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**COOLING METHODS STUDY
FOR THE PROPOSED RAFT RIVER
GEOTHERMAL POWER PLANT**

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ABSTRACT

Several different condensate cooling methods are available for use on the proposed Raft River Geothermal Power Plant. A study was made to determine the sensitivity of the power output of steam and organic vapor turbines to changes in the ambient conditions of the Raft River Valley for various cooling methods. The exhaust vapor from a turbine is condensed by a cooling fluid, such as water or air, in a condenser cooling system. It is known that deviations from the design temperature or flow rate of the cooling fluid can result in a decrease in the power output of the turbine. Since changing ambient conditions will affect the condition of the cooling fluid, they can also affect the power output.

Another consideration was to determine if lower ambient design conditions could be used to significantly reduce the size requirements of the various cooling systems without adversely affecting output power capabilities.

The major parameters and results are discussed and comments are presented regarding the effect of variations in ambient conditions on power output and cooling system size requirements. Conclusions were drawn from the results and recommendations made.

SUMMARY

Several different condensate cooling methods are available for use on the proposed Raft River Geothermal Power Plant. A study was made to determine the sensitivity of the output power of steam and organic vapor turbines to changes in the ambient conditions of the Raft River Valley for various cooling methods. The exhaust vapor from a turbine is condensed by a cooling fluid, such as water or air, in a condenser cooling system. It is known that deviations from the design temperature or flow rate of the cooling fluid can result in a decrease in the power output of the turbine. For example, if the temperature of the cooling fluid drops below the design value, the capacity of the condenser to condense the turbine exhaust vapor is reduced. This results in less vapor being condensed, which raises the condenser operating pressure, which in turn causes a reduction in turbine output power. Since the cooling system is dependent on ambient conditions to provide proper cooling of the condenser cooling fluid, any variations in the ambient conditions can result in a reduced power output.

Another consideration was to determine if lower ambient design conditions could be used to reduce significantly the size requirements of the various cooling systems without adversely affecting output power capabilities.

The study showed that deviations from ambient design conditions did not significantly affect the output power, and reduced design conditions showed little effect on the size requirements of the various cooling systems. The results indicate that the most efficient cooling system for use with the proposed geothermal power plant in the Raft River area is the wet cooling tower designed to a wet-bulb temperature of 68°F.

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I. INTRODUCTION

In the proposed geothermal power plant in the Raft River Valley, a vapor driven turbine will be used to generate electrical power. The exhaust vapor from the turbine will enter a condenser where the vapor will be condensed. For a given system, the output power capability of the turbine will be determined by the ability of the condenser to condense the vapor. A reduction in cooling ability requires a reduction in the vapor flow through the turbine or a rise in condenser pressure, thus lowering the power from design conditions. The condenser pressure and temperature must be maintained relatively constant in order to provide the required cooling. This is done by maintaining a constant flow of cooling water at a constant inlet temperature.

Most cooling systems that are in use today are sensitive to changes in weather conditions. For example, in a cooling tower, if the ambient temperature rises above the design wet-bulb temperature, the cooling ability of the tower decreases. This causes an increase in the temperature of the condenser cooling water which reduces the amount of vapor that can be condensed at the design condenser pressure. The resulting reduction of vapor flow causes a reduction in the power production of the turbine generator. The purpose of this study was to take a quick look at various cooling methods for both binary and dual flash steam geothermal power plants and determine the sensitivity of power output to changes in atmospheric conditions, especially temperature. The study was also to determine if lower design temperatures could decrease the size of the cooling system without adversely affecting the power output capabilities of the turbine.

II. COOLING SYSTEMS

The following systems were evaluated as a source of condenser cooling:

- (1) Cooling ponds
- (2) Spray ponds
- (3) Wet and dry cooling towers
- (4) Fan cooled condensers
- (5) Cold water aquifer.

A brief description of each system is provided in the following sections.

1. COOLING PONDS

Where large ground areas are available, cooling ponds offer a satisfactory method of removing heat from condenser cooling water. The cooling action is independent of the pond depth. Factors considered to affect pond performance are air temperature, relative humidity, wind speed, and solar radiation. The required pond area depends on the number of degrees of cooling required and the net heat loss from each square foot of pond surface. A pond may be constructed at a relatively small investment by pushing up an earth dike 6 to 10 feet high.

2. SPRAY PONDS

Spray ponds provide an arrangement for lowering the temperature of water by increasing the evaporative surface and, in so doing, greatly reducing the cooling area required in comparison with a cooling pond. A spray pond uses a number of nozzles which spray water into contact with the surrounding air. The water issuing from the nozzles creates a draft which, aided by the natural breeze, effects the necessary evaporation for cooling. Final temperatures depend on the cooling range, atmospheric conditions, arrangement of nozzles, and storage capacity of the pond.

3. WET COOLING TOWER

A cooling tower consists of a wooden or sheet-iron housing, open at the top and bottom, and so arranged that the condenser cooling water may be elevated to the top and

distributed in such a manner that it falls in thin sheets or sprays into a reservoir at the bottom. At the same time, air is being drawn in at the bottom by natural draft or induced in by a fan. The water gives up its heat to the ascending current of air by evaporation and convection. If the air supply is dependent entirely upon the chimney action of the device, the system is known as a natural-draft cooling tower; if the air is forced into the device by fans, the system is called an induced-draft cooling tower.

The rate and degree of cooling is a function of the amount of water surface exposed to the air per unit of time. The required tower size is a function of the following:

- (1) Cooling range (hot-water temperature minus cold-water temperature)
- (2) Approach (cold-water temperature minus wet-bulb temperature)
- (3) Quantity of water to be cooled
- (4) Wet-bulb temperature.

4. DRY COOLING TOWER

Dry cooling towers are used to reject steam condenser heat without the use of evaporative cooling; rather, dry cooling towers transmit heat directly to the atmosphere by conduction only. The heat is rejected through fin-tube heat exchangers with air flow promoted by fans or a natural draft stack.

In most situations, the dry tower must be quite large compared to evaporative-type equipment simply because size is necessary in order to get the heat transfer area. Two types of dry tower systems are in common usage: the direct cycle in which steam is condensed directly in the cooling tower finned piping; and the indirect cycle in which steam is condensed in a barometric condenser by spraying water on it, following which the combination of condensate and circulating water is passed through the cooling tower for direct conductive cooling. The advantage of dry cooling towers is that no water is evaporated, thus eliminating the vapor plume commonly seen with wet cooling towers. Also, the consumptive use of water is eliminated.

5. FAN COOLED CONDENSER

The fan cooled condenser is another name for the direct dry cooling tower. The turbine exhaust steam is conveyed directly to the air-cooled coils where cooling air passing over the finned coil surfaces condenses the steam.

The performance of a dry-type cooling system is measured by the temperature difference between the condensing steam of the turbine exhaust and the ambient air entering the cooling coils required to reject the design heat load.

6. COLD WATER AQUIFER

The cold water aquifer is a once-through cooling system in which water from a well is circulated through the condenser and reinjected into the ground. This system was considered in the cooling methods study for comparison only since the cold water aquifer is not affected by changing weather conditions.

III. DESIGN TEMPERATURES

Before the cooling systems can be evaluated, design temperatures must be established. Care must be taken not to choose design temperatures that are too high or too low. Low temperatures can cause excessive power losses during periods of high ambient temperatures. If the design temperature is too high, the cooling system would operate at full capacity only a few hours per day during peak temperature periods; thus, the additional cost and size required for such a system would not be justified. The normal procedure is to choose temperatures that are high enough to include 95-99% of the maximum temperatures incurred during the four hottest months^[1]. This allows full capacity operation of the power plant at all times except during the few hours when summer temperatures exceed the design temperature. The percent of power decrease during peak temperature periods depends on the type of cooling system used.

Recommended design dry-bulb and wet-bulb temperatures for the Raft River area are presented in Table I. All the data are based on Fire Weather information from the Malta, Idaho, Weather Station as recorded by the U. S. Forest Service. Data were obtained for a sufficient period of years (1965-1974) to permit a reasonably accurate definition of climatic conditions.

TABLE I
DESIGN TEMPERATURES

Temperatures	<u>Dry-Bulb</u> <u>Temperatures (°F)</u>			<u>Wet-Bulb</u> <u>Temperatures (°F)</u>		
	1%	2.5%	5%	1%	2.5%	5%
Design Value	96	94	92	68	66	65
Average Above Design Value	98	96.5	95	71	68.5	67
Maximum Expected Value	105			75		

The design data on dry-bulb and wet-bulb temperatures presented in Table I represent the highest 1, 2.5, and 5% of all the daily maximum temperatures incurred during the summer months of June through September. In a normal summer, there would be approximately 30 hours at temperatures above the 1% design value and approximately 150 hours at temperatures above the 5% design value^[2].

All data presented are based on the number of days the temperature goes above a given design value over a period of several seasons. It follows that about half the summers

would have more days of temperatures above a particular design level, and half would have fewer such days.

The design temperatures presented in Table I were used to determine the operating characteristics for each of the cooling systems being considered.

IV. POWER OUTPUT VERSUS AMBIENT TEMPERATURE

Each cooling system would provide 100% of the required cooling for full power operation at temperatures less than or equal to the design temperature. As the temperature increases above the design temperature, during the summer months, the performance of the cooling system begins to decrease which results in a decrease in the output of the power plant. Some cooling systems show only a small decrease in cooling ability at high temperatures while other systems would have to be shut down completely.

To determine the sensitivity of a cooling system, it is necessary to obtain a relationship between the power output of the turbine generator and the ambient air temperature. Since the power is directly related to steam flow and the steam flow is directly related to the condenser cooling water flow rate and temperature change, the following relationship was established:

$$\frac{P_{\text{actual}}}{P_{\text{design}}} = \frac{W_s \text{ actual}}{W_s \text{ design}} = \frac{W_w \text{ actual}}{W_w \text{ design}} \times \frac{\Delta T_w \text{ actual}}{\Delta T_w \text{ design}} \quad (1)$$

where P = power output; W_s = steam flow rate; W_w = condenser cooling water flow rate; ΔT_w = exit temperature minus inlet temperature of condenser cooling water.

Each of the cooling systems studied has a different equation relating the conditions of the cooling water to the ambient air temperature. These equations are determined by the performance characteristics of the cooling system. By combining these equations with Equation (1) above, the relationship between power and the ambient temperature can be obtained for each cooling system.

To maintain a constant power output, the water flow rate (W_w) and the temperature difference (ΔT_w) must remain constant at the design values. However, the values of W_w and ΔT_w will tend to increase as the air temperature decreases and, similarly, these values will tend to decrease as the air temperature increases. In the case of decreasing air temperatures, the air flow rate or the water to air exposure time can be reduced to help maintain constant values of W_w and ΔT_w . However, when the air temperature increases, the system design limits prevent an increase in the air flow rate or the water to air exposure time which will result in a decrease in W_w or ΔT_w . When either W_w or ΔT_w decrease, the power output will also decrease.

Five different cooling systems were evaluated in this study to determine, for each system, how sensitive the power output is to increasing air temperatures. The results are shown in Figure 1.

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Fig. 1 Cooling systems performance curves.

V. COOLING SYSTEMS PERFORMANCE RESULTS

Figure 1 shows the ratio of actual power to design power as a function of the ambient air temperature for the five systems that were studied. The systems on the left side of Figure 1 are sensitive to changes in the wet-bulb temperature, and the systems on the right are sensitive to changes in the dry-bulb temperature. The slopes of the curves indicate the sensitivity of power output to increasing temperatures. For a given system, as the design temperature is decreased, the slope of the performance curve decreases. This means that the power output is less sensitive to temperature changes when a low design temperature is used. However, as indicated in Figure 1, the power loss is greater for the lower design temperatures than for the high design temperature.

1. DUAL FLASH STEAM SYSTEM

To evaluate the five cooling systems, the following basic design parameters were used^[3]:

- (1) Steam flow rate (W_s) = 2.78×10^5 lb/hr
- (2) Heat released from steam (Δh_s) = 950 Btu/lb
- (3) Cooling water flow rate (W_w) = 16,000 gpm
- (4) Cooling water temperature drop (ΔT_w) = 33°F

The individual performance characteristics of each cooling system will be discussed in the following sections.

1.1 Wet Cooling Tower

The performance of the wet cooling tower is determined by the ambient wet-bulb temperature. Table II lists the three design temperatures along with the cooling tower air flow requirements, the high temperature power losses, and the power sensitivity associated with these temperatures. The maximum air flow requirements are an indication of the size of the cooling tower. Note that the lower design temperatures require smaller towers (less air flow required). The average power loss for periods of high temperatures can best be defined by an example. Suppose there are 10 days during the year when the daily maximum wet-bulb temperature goes above the design wet-bulb temperature of 68°F. If the maximum temperature is 69°F for 2 days, 71°F for 4 days, 72°F for 3 days, and 74°F for 1 day, the weighted average of the maximum temperatures above 68°F would be

$$T_{avg} = \frac{[2(69) + 4(71) + 3(72) + 1(74)]}{10} = 71.2^\circ\text{F} \quad (2)$$

TABLE II

WET COOLING TOWER PERFORMANCE CHARACTERISTICS

<u>Design Temperature (°F)</u>	<u>Maximum Air Flow Required (lb/hr)</u>	<u>Average Power Loss For Periods of High Temperatures (%)</u>	<u>Maximum Expected Power Loss (%)</u>	<u>Power Sensitivity (%/°F)</u>
68	6.16×10^6	4.5	11.0	1.6
66	5.99×10^6	3.5	13.5	1.5
65	5.91×10^6	2.5	14.5	1.4

From Figure 1, at a wet-bulb temperature of 71.2°F, the power loss with a design temperature of 68°F is 1-0.95 or 5%. This is the average power loss for periods of high temperature. As shown in Table II, the weighted average method used here will sometimes indicate a lower average power loss for low design temperatures than would be expected. This is true because the maximum temperatures within 2 or 3°F of the design value occur more frequently than the higher temperatures. The maximum expected power loss is the power loss that would occur at the maximum expected wet-bulb temperature. The power sensitivity is taken from Figure 1 as the percent power loss per degree increase in ambient temperature.

With a wet cooling tower, when the ambient temperature drops below the design temperature, the air flow can be decreased to maintain constant values of W_w and ΔT_w . When the ambient temperature goes above the design temperature, the air flow cannot be increased due to the design restriction. Therefore, either W_w or ΔT_w must decrease. In this study, ΔT_w was held constant while W_w was decreased for high temperatures, which resulted in a decrease in power output, as shown in Figure 1. The same results would have been obtained if W_w was held constant and ΔT_w allowed to decrease with increasing temperatures. Sample calculations for wet tower performance are included in the appendix.

As indicated in Figure 1, the power output is least sensitive to increasing temperature when the cooling system is a wet cooling tower.

1.2 Spray Pond

As with the wet cooling tower, the performance of the spray pond is determined by the ambient wet-bulb temperature. The wind speed also affects spray pond performance; however, for simplicity in calculations, a constant value of five mph was assumed as an average wind speed. Table III lists the three design temperatures along with the spray pond size, the high temperature power losses, and the power sensitivity associated with these temperatures.

With a spray pond, when the ambient temperature drops below the design temperature, the values of W_w and ΔT_w can be maintained constant by diverting some of the hot water directly to the pond, bypassing the spray nozzles. When the ambient temperature goes above the design temperature, the spray nozzles will be operating at full capacity. The high temperatures decrease the cooling ability of the spray pond, thus resulting in a decrease of either W_w or ΔT_w . As with the wet cooling tower, if ΔT_w is maintained constant, W_w will have to decrease; or if W_w is maintained constant, ΔT_w will be decreased. The same amount of power loss results with either method. As shown in Table III, the sensitivity of power loss to increasing temperatures for a spray pond is about three times greater than the power sensitivity for a wet cooling tower.

1.3 Cooling Pond

Cooling pond performance is basically dependent on the wet-bulb temperature and air movement; however, in this case, the dry-bulb temperature was used. To determine the

TABLE III

SPRAY POND PERFORMANCE CHARACTERISTICS

<u>Design Temperature (°F)</u>	<u>Pond Size Required (ft²)</u>	<u>Average Power Loss For Periods of High Temperatures (%)</u>	<u>Maximum Expected Power Loss (%)</u>	<u>Power Sensitivity (%/°F)</u>
68	54,700	14.5	34.0	4.9
66	50,100	10.0	39.2	4.4
65	48,200	7.5	41.5	4.2

performance characteristics of the cooling pond, a nomograph from *Perry's Chemical Engineer's Handbook* was used. The variables employed by the nomograph are relative humidity, dry-bulb temperature, and wind speed. The wind speed was assumed constant at an average value of five mph. The relative humidity was assumed constant at the value obtained from a psychrometric chart using the design dry-bulb temperature with the corresponding wet-bulb temperature obtained from the weather data. Thus, the only variable remaining was the dry-bulb temperature; therefore, the cooling pond performance was determined as a function of the dry-bulb temperature. Table IV lists the three design temperatures and corresponding values of relative humidity along with the cooling pond size, the high temperature power losses, and the power sensitivity associated with these temperatures.

With the cooling pond, when the ambient temperature drops below the design temperature, the values of W_w and ΔT_w can be maintained constant by allowing some of the hot water to bypass the cooling pond. When the ambient temperature goes above the design temperature, the value of ΔT_w will decrease because the average pond temperature will increase. With a cooling pond, the value of ΔT_w cannot be maintained constant by decreasing W_w as with other systems. Therefore, the cooling water flow rate will be held constant at the design value; and the decrease in ΔT_w will account for high temperature power losses.

In comparison, the power sensitivity for the cooling pond is slightly less than for a spray pond. This might indicate that the cooling pond is more desirable; however, the cooling pond performance is determined by dry-bulb temperatures which have a greater range of variability for a given period of time than the wet-bulb temperatures. Thus, for a given time period, the power loss will be greater with the cooling pond than with a spray pond.

1.4 Fan Cooled Condenser

Since cooling water is not used in the fan cooled condenser, and no water comes into contact with the air, the dry-bulb temperature is the important factor in determining the condenser cooling performance. Table V lists the three design temperatures along with the air flow requirements, the power losses, and the power sensitivity associated with these temperatures. As with the wet cooling tower, the required air flow is an indication of the size of the condenser.

The power output of the turbine generator is directly related to the quantity of steam condensed by the condenser. A fan cooled condenser will condense all of the steam for full power output with ambient temperatures equal to or less than the design temperature. When the air temperature goes above the design temperature, the quantity of steam condensed will decrease, resulting in a decrease in power. As indicated in Table V, the power is more sensitive to increasing temperatures for the fan cooled condenser than with any of the previously discussed systems. The fan cooled condenser also causes greater power losses at high temperatures.

TABLE IV

COOLING POND PERFORMANCE CHARACTERISTICS

<u>Design Temperature (°F)</u>	<u>Relative Humidity (%)</u>	<u>Pond Size Required (ft²)</u>	<u>Average Power Loss For Periods of High Temperatures (%)</u>	<u>Maximum Expected Power Loss (%)</u>	<u>Power Sensitivity (%/°F)</u>
96	11	3.15 x 10 ⁶ (73 acres)	5.8	40.0	4.4
94	12	3.00 x 10 ⁶ (69 acres)	7.0	46.5	4.2
92	13	2.96 x 10 ⁶ (69 acres)	9.0	51.5	4.0

TABLE V

FAN COOLED CONDENSER PERFORMANCE CHARACTERISTICS

<u>Design Temperature (°F)</u>	<u>Air Flow Required (lb/hr)</u>	<u>Average Power Loss For Periods of High Temperatures (%)</u>	<u>Maximum Expected Power Loss (%)</u>	<u>Power Sensitivity (%/°F)</u>
96	10.00×10^7	18.5	82.0	9.1
94	8.46×10^7	19.0	85.0	7.7
92	7.34×10^7	20.0	87.0	6.7

1.5 Dry Cooling Tower

The dry cooling tower is similar to the fan cooled condenser in that no water comes into contact with the air and the performance is a function of the dry-bulb temperature. The main difference between the two systems is the cooling water cycle used with a dry cooling tower. The additional heat load on the cooling system of removing heat from the steam then removing the heat from the cooling water causes the dry cooling tower to be less efficient than the fan cooled condenser.

Design temperatures are also a problem with the dry cooling tower. In the preliminary power plant design, the cooling water was specified to enter the condenser at 75°F. In a dry cooling tower, the cold water temperature will always be at least 5 to 10°F greater than the ambient air temperature; therefore, the design temperature is restricted to 65°F or less. Table VI shows the performance characteristics of the dry tower with a design temperature of 65°F. When designed to this temperature, the dry tower will be completely ineffective at temperatures above 98°F; see Figure 1. Because of the poor performance at a design temperature of 65°F, lower design temperatures were not considered.

If a dry cooling system is necessary, the fan cooled condenser, which does not use a cooling water cycle, would be the best system.

2. BINARY SYSTEM

The performance of a cooling system on a binary geothermal power plant is identical to the performance of the same cooling system on the dual flash steam power plant. The only difference in the two cooling systems is the size requirements. The size of a cooling system used with a binary power plant is dependent on the type of organic fluid used. Different organic fluids will require different quantities of cooling water to condense the fluid from the vapor state to the liquid state because of the different thermodynamic properties associated with each fluid. In this study, the binary fluid was Freon-12, which requires less cooling water for condensing than is required with steam. The performance characteristics for both steam and Freon-12 cycles were the same for each type of cooling system studied; however, with Freon-12 as the operating fluid, the cooling systems were roughly 20% smaller in size.

TABLE VI

DRY COOLING TOWER PERFORMANCE CHARACTERISTICS

<u>Design Temperatures (°F)</u>	<u>Air Flow Required (lb/hr)</u>	<u>Average Power Loss For Periods of High Temperatures (%)</u>	<u>Maximum Expected Power Loss (°F)</u>	<u>Power Sensitivity (%/°F)</u>
65	31.43×10^6	44.0	Complete power loss above 98	3.3

VI. CONCLUSIONS

The main purpose of this study was to determine the sensitivity of power output to increasing temperatures for various cooling systems. Another consideration was to determine if lower design temperatures have a significant effect on the size of the cooling systems and the maximum power loss associated with each system.

The study indicated that lower design temperatures caused an increase in the maximum expected power loss; however, Figure 1 shows that the increased power loss is less than 10% for most of the systems studied. Since the maximum power loss is effective for such a short time, the increased loss associated with lower design temperatures is relatively insignificant.

Tables II through V indicate that lower design temperatures allow the use of smaller cooling systems. Of the systems considered, the fan cooled condenser had the greatest decrease in size with decreases in design temperatures. By decreasing the design temperature from the 1% value to the 5% value, the condenser size was reduced by more than 25%. Since the increased power loss is insignificant, a fan cooled condenser should be designed to the 5% design temperature. The other cooling systems studied had size reductions of less than 10% when designed to the lower temperature. The savings associated with a 10% reduction in size would be insignificant relative to the overall cost of the system. Therefore, these systems should be designed to the 1% design temperature.

The performance curves in Figure 1 indicate that the power output is least affected by increasing temperatures when a wet cooling tower system is used to condense the turbine exhaust vapor. The size of the wet cooling tower is also an advantage over other systems. As shown in the tables of performance characteristics, the wet tower requires less air flow and, thus, less ground area for construction than the other four systems considered in this study. In conclusion, the most efficient cooling system for use with the proposed geothermal power plant in the Raft River area is the wet cooling tower designed to the 1% design temperature of 68°F.

VII. RECOMMENDATIONS

Since there are several different types of wet cooling towers in existence, a more detailed study should be made to determine which of these systems would give the best performance for operation in the Raft River area. The following systems should be considered in the study:

- (1) Mechanical draft towers
 - (a) induced-draft -- counterflow or crossflow arrangement
 - (b) forced-draft
- (2) Natural draft towers
- (3) Atmospheric cooling towers.

After a system has been chosen, the system should be optimized for least cost, size, and power loss.

VIII. REFERENCES

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APPENDIX

1. SAMPLE CALCULATIONS

2. REFERENCES

APPENDIX

1. SAMPLE CALCULATIONS FOR WET COOLING TOWER PERFORMANCE

A schematic for induced-draft, counterflow, cooling tower installation is shown in Figure A-1. Applicable equations are:

$$W_{a_1} h_{a_1} + W_{v_1} h_{v_1} + W_{w_A} h_{f_A} = W_{a_2} h_{a_2} + W_{v_2} h_{v_2} + W_{w_B} h_{f_B} \quad (A-1)$$

$$W_{w_B} = W_{w_A} - (W_{v_2} - W_{v_1}); \quad W_{a_1} = W_{a_2} \quad (A-2)$$

$$W_{w_A} (h_{f_A} - h_{f_B}) = (W_{a_2} h_{a_2} + W_{v_2} h_{v_2}) - (W_{a_1} h_{a_1} + W_{v_1} h_{v_1}) - \quad (A-3)$$

$$(W_{v_2} - W_{v_1}) h_{f_B}$$

$$h_{f_A} - h_{f_B} = t_{w_A} - t_{w_B} \quad (A-4)$$

and

$$W_a h_a + W_v h_v = \text{total heat, from psychrometric chart} \quad (A-5)$$

$$W_{w_A} (t_{w_A} - t_{w_B}) = \text{total heat at 2} - \text{total heat at 1} - \quad (A-6)$$

$$(W_{v_2} - W_{v_1}) h_{f_B}$$

where

W = flow (lb/hr)

a = air

w = water

v = vapor

h = enthalpy (Btu/lb)

f = fluid

t = temperature ($^{\circ}\text{F}$)

1, 2, A, B = locations.

Fig. A-1 Schematic for induced-draft, counterflow, cooling tower installation.

$$\text{Total heat} = W_a C_p (t_a) + W_v h_g \text{ at } t_a \quad (\text{A-7})$$

$$W_v = W_a w_v \quad (\text{A-8})$$

where w_v = specific humidity, grains/lb of dry air.

Therefore,

$$\begin{aligned} W_{wA} (t_{wA} - t_{wB}) &= (W_{a2} C_p t_{a2} + W_{a2} w_{v2} h_{g2}) - (W_{a1} C_p t_{a1} \\ &+ W_{a1} w_{v1} h_{g1}) - (W_{a2} w_{v2} - W_{a1} w_{v1}) h_{fB} \end{aligned} \quad (\text{A-9})$$

$$W_{wA} (t_{wA} - t_{wB}) = W_a \left[C_p (t_{a2} - t_{a1}) + w_{v2} (h_{g2} - h_{fB}) - w_{v1} (h_{g1} - h_{fB}) \right] \quad (\text{A-10})$$

Example:

$$\begin{aligned} t_{a2} &= 100^\circ\text{F saturated} \\ t_{wA} &= 108^\circ\text{F} \\ t_{wB} &= 75^\circ\text{F} \\ C_{p_a} &= 0.24 \text{ Btu/lb}^\circ\text{F} \\ w_{v2} &= 300 \text{ grains/lb}_a \text{ at } 100^\circ\text{F saturated air} \\ h_{g2} &= 1105.2 \text{ Btu/lb at } 100^\circ\text{F} \\ h_{fB} &= 43.03 \text{ Btu/lb at } 75^\circ\text{F}. \end{aligned}$$

Apply these values to Equation (A-10):

$$\begin{aligned} W_{wA} (33) &= W_a \left[0.24 (100 - t_{a1}) + \frac{300}{7000} (1105.2 - 43.03) - w_{v1} (h_{g1} - 43.03) \right] \\ &= W_a \left[24.0 - 0.24 t_{a1} + 45.52 - w_{v1} (h_{g1} - 43.03) \right] \quad (A-11) \end{aligned}$$

This equation shows W_w as a function of t_{a1} if W_a is constant.

Given the following design parameters:

Water flow (W_w)	= 16,000 gpm
Design wet-bulb temperature (t_{a1})	= 68°F
Specific humidity (w_{v1})	= 69 grains/lb _a
Enthalpy of air vapor (h_{g1})	= 1091.5 Btu/lb

the maximum design air flow may be calculated,

$$\begin{aligned} W_{ad} &= \frac{(16,000)(500)(33)}{69.52 - 0.24(68) - \frac{69}{7000}(1091.5 - 43.03)} \\ W_{ad} &= 6.16 \times 10^6 \text{ lb/hr.} \end{aligned}$$

Thus,

$$W_w \Delta t_w = 6.16 \times 10^6 \left[69.52 - 0.24 t_{a1} - w_{v1} (h_{g1} - 43.03) \right] \quad (A-12)$$

To maintain a constant power output, the water flow rate (W_w) and the temperature difference (Δt_w) must remain constant at

$$W_w = 16,000 \text{ gpm}$$

$$\Delta t_w = 33^\circ\text{F.}$$

These values are directly related to t_{a1} . If t_{a1} is less than the design temperature, then the air flow (W_a) can be decreased to maintain constant values of W_w and Δt_w . However, as t_{a1} goes above the design temperature, the air flow cannot be increased; therefore, either W_w or Δt_w will decrease. In this case, Δt_w will be held constant and W_w will decrease.

At

$$\Delta t_w = 33^\circ\text{F}$$

$$W_w = 16,000 \text{ gpm if } T_{a1} \leq 68^\circ\text{F} \quad (\text{A-13})$$

$$W_w = 373.25 \left[69.52 - 0.24 t_{a1} - w_{v1} (h_{g1} - 43.03) \right] \text{ if } T_{a1} \geq 68^\circ\text{F}. \quad (\text{A-14})$$

The relationship of W_w to t_{a1} is shown in Figure A-2. Figure A-3 shows how much the power is reduced when t_{a1} goes above the design temperature. This relationship comes from the equation:

$$\frac{P}{P_d} = \frac{W_w}{W_{wd}}.$$

Fig. A-2 Wet cooling tower performance curve.

Fig. A-3 Wet cooling tower performance curve.

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