# A COMPARISON OF WET TYPE AND DRY TYPE COOLING TOWERS BY ENERGY AVAILABILITY METHODS

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A COMPARISON OF WET TYPE AND DRY TYPE COOLING TOWERS
BY ENERGY AVAILABILITY METHODS

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NOMENCLATURE

| | |
|---|---|
| a | mean area of water-air interface per cubic foot of packed volume |
| A | cooling tower base area $ft^2$ or available energy |
| B | energy availability BTU or a parameter defined by Equation IV-53 |
| b | energy availability in flow systems BTU/lbm |
| C | Chilton coefficient or specific heat of water |
| Ce,Ch,Cr | constants defined by Equation III-39 |
| D | duty coefficient or tube diameter |
| $D_e$ | equivalent diameter |
| $D_v$ | volumetric diameter |
| e | effectiveness defined by Equation III-1 or a parameter defined by Equation IV-54 |
| E | total energy |
| f | mean driving force factor defined by Equation IV-7 or fin-side frictional factor defined by Equation IV-41 |
| G | air load in cooling towers $lbm/ft^2$-hr |
| $G_m$ | air flow rate through minimum cross-sectional area |
| $G'_m$ | average air mass flow rate over total area including finned tube projected area and free spacing area in a dry type cooling tower |
| $G_w$ | water flow rate through a tube $lbm/ft^2$-hr |
| H | height of cooling tower or total enthalpy of air-vapor mixture |
| h | enthalpy of air-vapor mixture per lbm of air or enthalpy of steam or air side heat transfer coefficient for finned tubes |

| | |
|---|---|
| $h_i$ | tube side heat transfer coefficient |
| $h_m$ | mean enthalpy corresponding to the temperature $\frac{T_1+T_2}{2}$ |
| $h_{md}$ | mean driving enthalpy force |
| ID | inner diameter of a tube |
| j | air side heat transfer factor defined by Equation IV-38 |
| $j_h$ | tube side heat transfer factor |
| K | heat conductivity or Merkel's heat transfer coefficient defined by Equation IV-3, or a constant defined by Equation III-39 |
| Ka | a combined coefficient of k and a |
| $K_p$ | pumping factor defined by Equation III-25 |
| L | water flow rate lbm/ft$^2$-hr in a cooling tower or length of a tube |
| $L_l$ | liner thickness of a finned tube |
| LMTD | log mean temperature difference |
| $\dot{m}$ | steam mass flow rate |
| m | index defining the variation of mass-transfer coefficient with water loading in Equation IV-15 (a constant for a given packing) |
| n | index defining variation of mass-transfer coefficient with air loading in Equation IV-15 (a constant for a given packing) |
| N | a resistance to air flow in velocity heads referred to the reference plane as defined in Equation IV-21 or number of rows of finned tube per cooling unit or mass of component defined by Equation III-3 |
| $N_{ct}$ | number of cooling towers |
| $N_t$ | number of finned tubes per ft of pitch length |
| OD | outer diameter of a tube |
| P | pressure |

| | |
|---|---|
| Pr | power plant profit |
| Q | amount of heat transfer or heat input in a thermal cycle |
| $R_1$ | tube side heat transfer resistance |
| $R_2$ | fouling resistance |
| $R_3$ | liner resistance |
| $R_4$ | bond resistance |
| $R_5$ | root tube resistance |
| $R_t$ | total resistance |
| $R_b$ | bond resistance factor |
| $R_{di}$ | fouling factor |
| Re | Reynolds number |
| $Re_v$ | volumetric Reynolds number defined by Equation IV-40 |
| s | entropy per unit of mass |
| S | entropy or contact area of liquid with air defined by Equation IV-3 or contact area of finned tube with air defined by Equation IV-51 |
| $T_d$ | dry bulb temperature |
| $T_{dp}$ | dew point temperature |
| $T_w$ | wet bulb temperature |
| $T_h$ | heat input temperature for a thermal cycle |
| $T_l$ | heat rejection temperature for a thermal cycle |
| $T_{hav}$ | average heat input temperature |
| $T_o$ | cooling water temperature leaving cooling system-recooled water temperature |
| Tr | a reference temperature defined by Equation III-39 |
| $T_{oav}$ | average cooling water temperature in condenser |

| | |
|---|---|
| $T_{lav}$ | average heat rejection temperature |
| $\Delta T_1$ | the difference between $T_{oav}$ and $T_{lav}$ |
| $\Delta T_o$ | cooling water cooling range |
| $\Delta T_t$ | true temperature difference |
| TTD | terminal temperature difference |
| $T_1$ | hot fluid inlet temperature |
| $T_2$ | hot fluid outlet temperature |
| $t_1$ | cold fluid inlet temperature |
| $t_2$ | cold fluid outlet temperature |
| U | overall heat transfer coefficient |
| u | air velocity in a cooling tower |
| v | specific volume |
| V | height of packing |
| W | work output of a thermal cycle or cooling water total flow rate |
| w | work per unit mass |
| X | steam quality or air absolute humidity |
| Y | unknown temperature of cooling water |
| z | potential height |

<u>Greek Notation</u>

| | |
|---|---|
| $\rho$ | air density |
| $\alpha$ | Merkel's cooling factor or a factor defined by Equation III-44 |
| $\lambda$ | latent heat of steam or a constant for a given packing defined by Equation IV-16 |
| $\mu$ | viscosity or Gibb's free energy |
| $\mu_w$ | viscosity at wall temperature |

| | |
|---|---|
| $\eta$ | Carnot thermal efficiency or thermal efficiency |

Prefix Notation

| | |
|---|---|
| $\Delta$ | the change of value between conditions 1 and 2 |
| d | differentiation |
| $\partial$ | partial differentiation |

Superscript Notation

| | |
|---|---|
| c | creation |
| ' | isentropic process or saturated state |

Subscript Notation

| | |
|---|---|
| a,b | state a or b |
| av | average |
| c | condenser |
| e | exit |
| f | fin-side |
| g | gas |
| h | high temperature side |
| i | inlet |
| l | low temperature side |
| m | mean |
| max | maximum |
| min | minimum |
| o | cooling water or equilibrium with surroundings |
| p | pump |
| pi | pipe |
| r | root tube |
| L | liner |

rej rejection

t turbine, tube side or total

w water or wall temperature

1 and 2 above or below packing or inlet and exit

Constant Notation

g gravitational acceleration, taken as 32.2 $ft/sec^2$

$g_c$ universal constant

## SUMMARY

The objective of this thesis is to present an investigation comparing the wet type and dry type cooling tower by energy availability methods. The current various cooling systems for power plants are discussed. A suitable equation, via energy availability methods, is developed to clearly reflect the effect of cooling water temperature on the power production rate. In order to facilitate the comparison, a 1000 MW modern power plant is considered using both the wet type and dry type cooling tower for rejecting its waste heat. Successful comparisons are made by using the derived equation to evaluate and compare the performances of each type of cooling tower under different weather conditions.

CHAPTER I

# INTRODUCTION

During the past several decades, electric power loads have approximately doubled every ten years in the United States and they are expected to continue increasing at this rate through 1990. At present, more than 80 percent of the electric energy produced in the United States is generated in steam-electric plants. Favorable sites for new hydroelectric developments are comparatively limited, and other noncondensing types of generating plants now in use are not likely to account for a substantial portion of future energy requirements. Thus, even considering the results of research under way to develop new means of energy conversion, it appears likely that for the foreseeable future the bulk of electric generation will be produced by steam-electric plants, either nuclear or fossil fuel. Figure 1 shows the projected corresponding generation to 1990 [1].

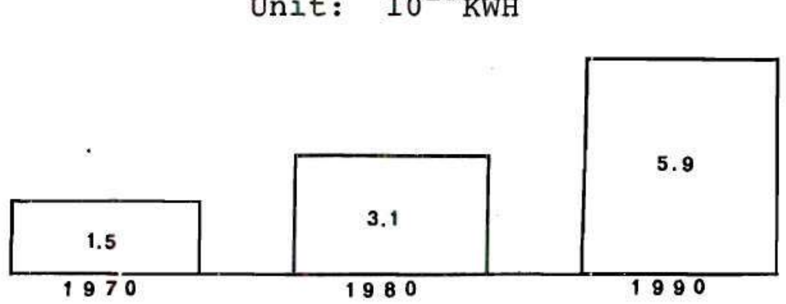



Figure 1. Electricity Demand Through 1990

In the operation of a steam electric plant, steam is produced at high temperature and pressure in the boiler or reactor, then flows through the turbine giving up energy which drives the generator to produce electricity. At the exhaust of the turbine, steam is condensed so as to maximize the energy conversion. A large amount of heat is given up to the cooling water in the condensing process. The amount of heat discharged to the condenser is related to the plant efficiency. Normally stating, current thermal efficiency of 33 to 40 percent for modern steam power plants results in a mistaken notion that the efficiency is extremely low. This occurs because any thermal cycle is subject to the unbreakable second law of thermodynamics, in other words, heat rejection is an inherent byproduct of all heat machines [2]. The amount of heat is inversely proportional to the cycle efficiency. Thus, an increasing demand for the electric energy means an increasing amount of heat to be rejected to the surroundings. Figure 2 illustrates the total estimated waste heat to be

Unit: $10^{15}$BTU

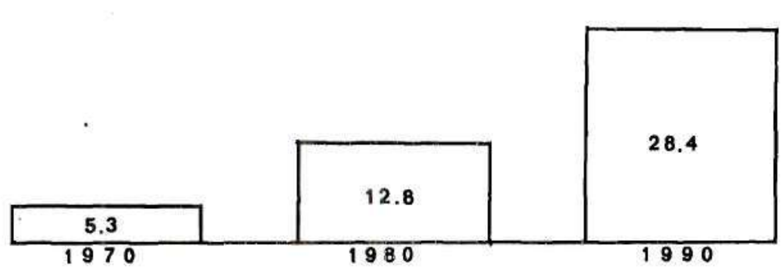



Figure 2. Amount of Heat to be Rejected Through 1990

discharged by the fossil and nuclear steam plants projected for operation to 1990. For comparative purposes, it may be noted that the total estimated waste heat for 1990 is 40 percent greater than the $20 \times 10^{15}$ BTU equivalent of the electricity generation in that year by all types of generation plants. Some even have forecast the electrical generation waste heat from steam power plants will amount to $55 \times 10^{15}$ BTU in the year of 2000 [3].

Attendant upon such a large amount of heat to be rejected, there arise two problems which interest power plant engineers most: (1) How to dispose of this vast amount of waste heat without adversely affecting the environment, and (2) How to choose heat rejection systems taking in consideration the capital investment and the thermal efficiency of the power plant.

In summary, the power industry faces the problem of meeting the growing demand for electrical energy and at the same time controlling thermal pollution with the view of thermal efficiency and capital investments. The contents of this thesis will focus on the discussion of the thermal efficiency based on each different heat rejection system.

The Carnot thermal efficiency $\eta = \frac{T_h - T_l}{T_h}$ tells us that for any thermal cycle, the heat input temperature $T_h$ should be as high as possible while the heat rejection temperature $T_l$ should be as low as possible. These two requirements must be fulfilled when trying to increase thermal efficiencies of power cycles. The heat input temperature $T_h$ depends on the type of heat source. Generally speaking, a fossil-fired boiler can produce higher temperature and pressure steam than a nuclear reactor. Therefore, in real practice an efficiency of almost 40 percent is attainable for a fossil-fired power plant, while an efficiency of only 33 percent is obtained by nuclear power plants. The heat rejection temperature $T_l$ of a power plant is referred to the steam exhaust temperature from turbines. $T_l$ is determined by the temperature of cooling water in the condensers, or further by the cooling systems. The steam exhaust pressure $P_l$ is fixed also since the steam is condensed along the steam saturation line as shown on Figure 1-3. It is apparent that a lower cooling water temperature will reduce the steam exhaust temperature and pressure and vice versa. Therefore, the thermal efficiency of a power plant is intimately related to the performance of its heat rejection system [4].

There are several typical heat rejection systems available now. As they are superimposed on Figure 3, the cooling systems can roughly be divided into three types according to that heat transfer mechanism: (1) Once-through cooling range, (2) Wet-tower cooling range, and (3) Dry-tower cooling range. There is considerable current interest in comparing the relative merits of these methods. Much of this interest has recently centered upon a comparison of evaporative versus dry cooling towers. Some previous work comparing the recooled

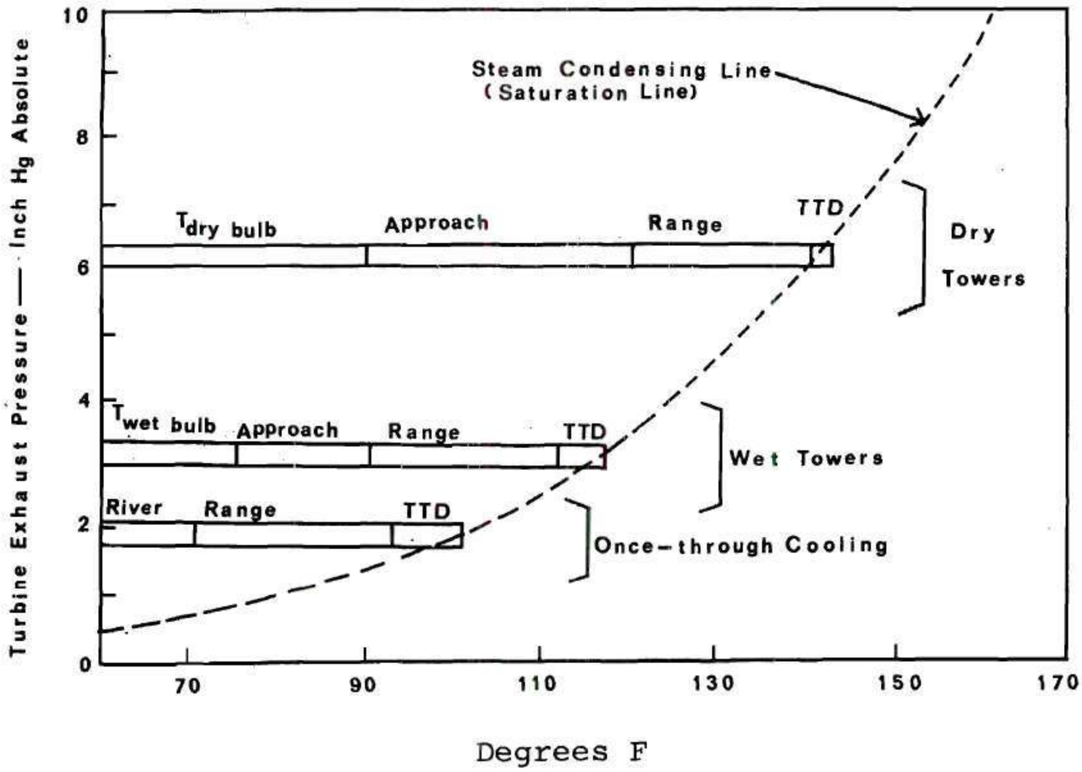



Figure 3. Cooling System Temperatures Determine Turbine Exhaust Temperature and Pressure Along the Steam Condensing Line. (Ranges of exhaust pressures are indicated for basic types of cooling systems according to heat transfer mechanism employed.)

water temperatures of these two types of cooling towers has been done by H. Heeren and L. Holly [27]. However, no one has as yet compared the effects of the recooled water temperatures of these two cooling systems on the thermal efficiencies of power plants. Such a special comparison can easily be done if we have certain analytical procedures to evaluate how the performance of a power plant varies with the cooling water temperature, i.e., the performances of the cooling systems being evaluated by the performances of its power plant.

The purpose of this thesis is to use energy availability methods to demonstrate a direct means of comparison which will reflect clearly the comparative effects of recooled water temperature on power production rates from modern power plants--an essential step in making comprehensive decisions regarding the design and use of cooling systems.

In order to facilitate the comparison, a power plant of 1000 MW employing a supercritical thermal cycle is considered for the design of both the dry and wet type of cooling towers to dispose of its waste heat. A more comprehensive comparison of their recooled water temperatures than that of Heeren and Holly is obtained. Finally, the efficiencies of the power plant, calculated by an equation derived from energy availability methods, are also compared by plotting against different dry bulb temperatures.

# CHAPTER II

# TECHNICAL DISCUSSION

Before entering the main subject of this thesis, a brief discussion on the recent techniques of discharging waste heat from a power plant is in order. As was mentioned in the previous chapter, many electric utilities have been confronted with the problem of how to reject waste heat from future electric generating steam power plants. Several heat rejection systems have been devised and are now being used for the purpose of fitting the special situations of different plant sites. As a consequence of their employing different heat transfer mechanisms, each type has some advantages and disadvantages, respectively. A literature survey of conventional and modern heat rejection systems is presented in this chapter, together with a comparative analysis of their merits and demerits.

## Classification of Heat Rejection Systems

The cooling systems can be divided into four systems according to their devices [4]:

(1) Once-through cooling system
(2) Cooling lake system
(3) Spray pond cooling system
(4) Cooling tower cooling system

The cooling tower system can further be separated into several types:

- Evaporative Cooling Towers
  - Natural Draft
    - Cross Flow
    - Counter Flow
  - Mechanical Draft
    - Forced
    - Induced
      - Counter Flow
      - Cross Flow
- Dry Cooling Towers
  - Direct
  - Indirect
    - Mechanical Draft
    - Natural Draft
- Evaporative/Dry Cooling Towers
  - Series Types
  - Parallel Types

Each cooling system will be discussed separately in the following sections [5].

(1) Once-through cooling system

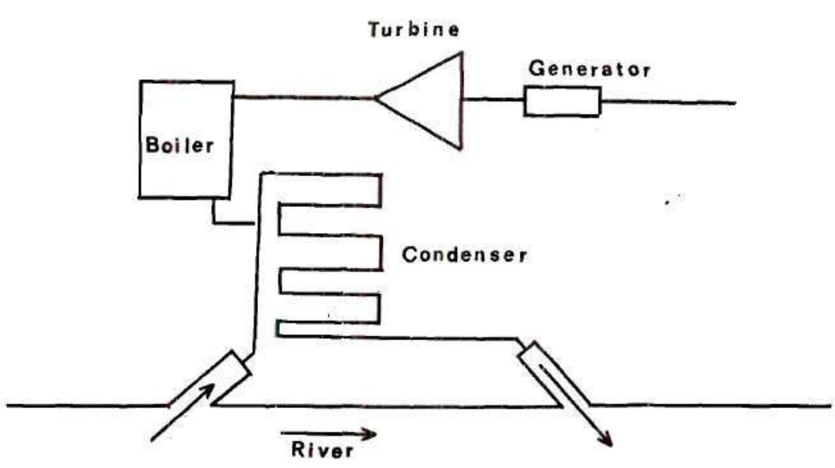



Figure 4. Once-through Cooling System

Once-through cooling takes water from a lake or river at temperatures about 70°F and heats it about 20 to 25° F in the condenser, and then discharges it at a point downstream from the plant. With such a cooling system, the cooling water doesn't form a closed cycle. Therefore, the once-through cooling systems provide the lowest naturally occurring condensing temperature available to the steam turbine, i.e., provide the most efficient utilization of turbine heat input, since the steam can expand to a low exhaust pressure and produce more useful work. Recently, because of the widespread adoption of federal and state thermal pollution control regulations, there are fewer and fewer new applications of this conventional cooling system.

Advantages:

1. The simplest and most economical method.
2. Minimum water consumption.

Disadvantages:

1. Limited availability of large supplies of cooling water.
2. May violate water quality standards.

(2) Cooling lake system

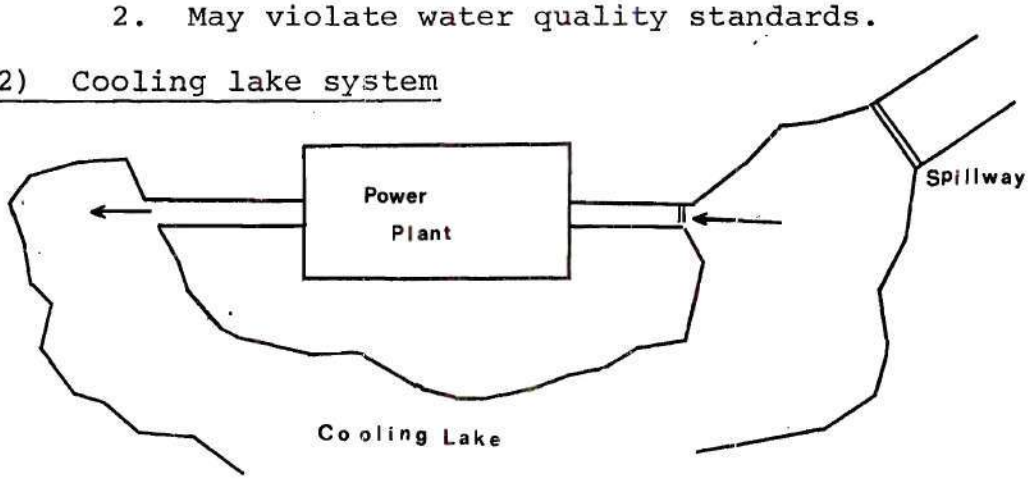



Figure 5. Cooling Lake System

This system resembles the once-through system, with the exception that the cooling water is recirculated. The water surface temperature will be closely identical with a once-through cooling system, if the lake size is large enough. However, many cooling lakes are much smaller, and therefore operate at much higher surface water temperature and, likewise, higher turbine exhaust pressure. An approximate estimate indicates that the minimum area required for a cooling lake is 2 acre/MW for a fossil plant, or 3 acre/MW for a nuclear power plant. At such rates, the cooling lake will perform as well as a once-through system.

Advantages:

1. Reasonable construction costs where soil conditions permit.
2. Can possibly operate over a long period of time without make-up water.
3. May be beneficial for other purposes--sailing.

Disadvantages:

1. Requires large land area.
2. The basin soil of low permeability is seriously required.
3. Possibility of fogging and icing.
4. Concentrate dissolved solid.

(3) Spray-pond cooling system

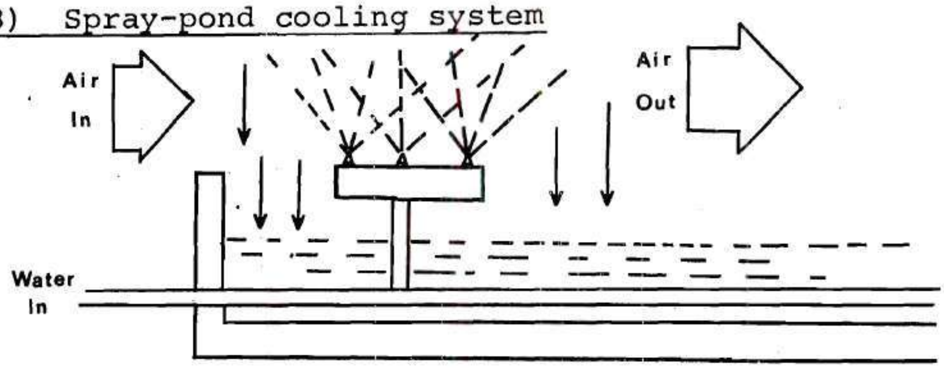



Figure 6. Spray-pond Cooling System

A spray-pond cooling system is shown in Figure 6. The spray nozzles atomize the droplets into fine sprays, thereby increasing the heat transfer per unit area of land. Heat is rejected by direct contact of ambient air with the water sprays direct from condenser. A nominal water loading rate of 500 lbm/hr-ft$^2$ of pond area, and wind speeds of 6 miles/hr would be typical design parameters for such a cooling system.

Advantages:

1. Reduces required area compared to cooling lake.
2. Relatively simple and economic compared to cooling lakes.

Disadvantages:

1. Increased water losses due to drift.
2. Performance strongly depends on wind speed and direction.
3. May cause localized icing and fogging.

(4) Wet or evaporative cooling towers

There are two basic classes of evaporative cooling towers:

(a) Mechanical draft type -- a fan is used to produce air draft through the tower.

(b) Natural draft type -- the air draft is produced by the "chimney effect" of the tower height.

For both types, heat transfer takes place within the cooling tower by direct contact of cooling water with air. Most of the heat is dissipated by evaporating a portion of the circulating water, while the remaining heat is lost by sensible heat transferring to the air.

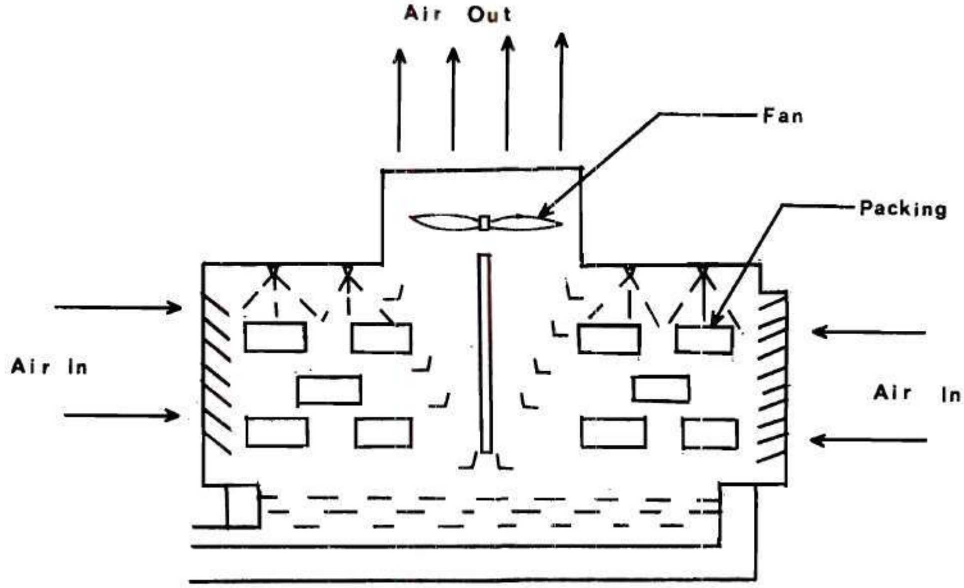



Figure 7. Mechanical Draft Wet Type Cooling Tower

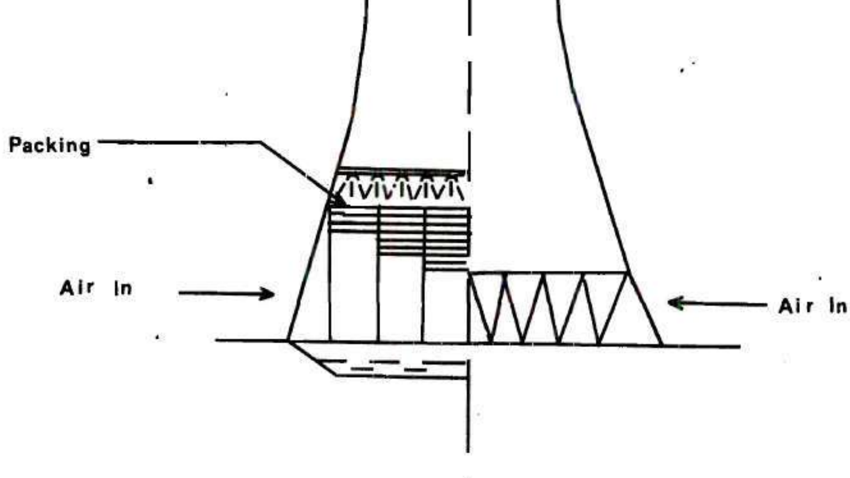



Figure 8. Natural Draft Wet Type Cooling Tower

The towers are also divided into two types according to the flow direction of air in the tower relative to the flow direction of water.

1. Counter-flow -- The air flow direction is just the opposite of that of cooling water. Such an arrangement provides the most efficient means of heat transfer.

2. Cross-flow -- The air flow is perpendicular to the water flow.

Advantages of mechanical wet type cooling tower:

1. Positive control over air supply.
2. Pumping head is low.
3. Close control of cold water temperature.
4. A minimum effect on performance by relative humidity.
5. Lower capital cost than natural draft tower.

Disadvantages of mechanical wet type cooling tower:

1. Subject to mechanical failure.
2. Subject to recirculation of the humid exhaust air.
3. Operation and maintenance costs are higher than natural-draft tower.
4. Exhaust air may cause icing and fogging.

Advantages of natural wet type cooling tower:

1. No mechanical or electrical components.
2. Low maintenance costs.
3. Large water loading capacity.
4. Use comparatively small ground area.
5. Local icing and fogging may be eliminated by high level plume discharge.

Disadvantages of natural wet type cooling tower:

1. Internal resistance to air flow must be kept to a minimum.
2. Great tower height is necessary to produce draft, thus capital investment is higher than for mechanical type.
3. Exact control of outlet temperature is difficult.
4. Blow down disposal problem.

(5) Dry type cooling towers

The shape of dry type cooling towers are very similar to that of wet type towers except the internal construction. A dry type cooling system operates on the same principles as an automobile radiator. Thus, there are no evaporative losses. Heat is rejected through a fin-tube exchanger. Another type of dry cooling system, known as an air-cooled condenser, will condense steam directly inside the finned tubes. The flow of cooling air, in either dry cooling design could be promoted by fans or a natural draft stack.

Advantages:

1. Eliminate fogging, mist, icing.
2. Eliminate water problems, such as availability of water, evaporative losses, blow down and thermal pollution.

Disadvantages:

1. High construction costs.
2. High maintenance costs.
3. Large volume of air flow is needed.
4. Turbine output is limited by high cooling temperature.
5. Larger land area is required than for wet tower.

(6) Wet/dry type cooling tower

The vapor-plume emissions or large water consumption rate of wet type cooling towers and the high condensing temperature of dry cooling towers are undesirable to power plant cooling systems. A newly proposed method, known as a wet/dry cooling tower, provides great flexibility in plant design and siting because of its ability to reduce or even to eliminate visible plumes. It can also reduce annual water consumption to perhaps 20% of the conventional wet type tower value, without increasing the economic penalties associated with dry cooling systems. Figure 9 illustrates the construction of one of this type of cooling towers. The upper part of the tower is of dry type and the lower part is of wet type.

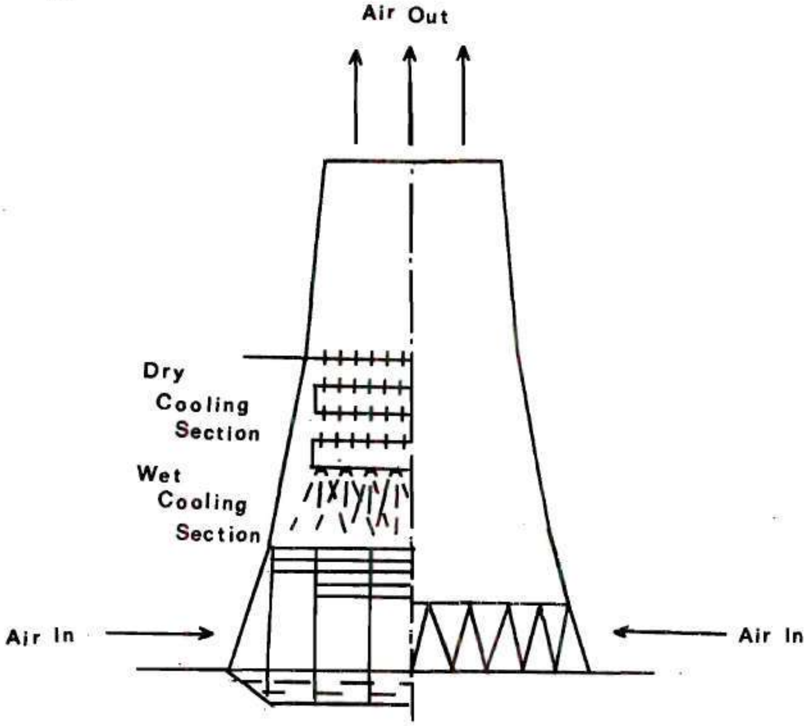



Figure 9. Wet/Dry Type Cooling Towers

There are several basic configurations for wet/dry tower design. Parallel flow designs have separate air passages through convective and evaporative sections and rely on the fan to mix the warm dry air with the warm saturated air. In series design, the dry cooling section can be located either behind or ahead of the wet cooling sections [6].

The theoretical analysis of a wet/dry cooling tower as compared with a wet type tower can easily be done on a psychrometric chart. The conventional wet type cooling processes result in warm-saturated exhaust air, which becomes supersaturated as it mixes with the cooler ambient air. Supersaturated air is a mixture of the moist air and water droplets that have condensed to form visible plumes. These processes can be shown on a psychrometric chart. As for the wet/dry type towers, the air never becomes saturated. The air exhausts from the tower in a warm but unsaturated condition. As it mixes with the cooler ambient air, the mixture follows line 3-1 (Figure 10), never becomes supersaturated and never forms a visible vapor plume.

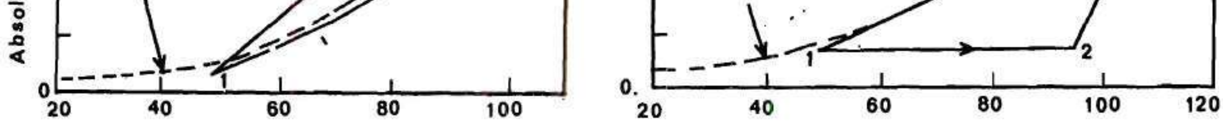



Figure 10. Psychrometric Chart for Wet Cooling Towers and Wet/Dry Type Cooling Towers

Summary:

Four principal types of cooling systems have been devised. Owing to the recent widespread adoption of federal and state thermal pollution regulations, fewer and fewer conventional once-through condenser cooling systems are being installed. This has resulted in an increasing trend toward using supplementary methods to reject waste heat at future plant sites. Man-made cooling lakes, spray ponds, wet type cooling towers, dry type cooling towers and wet/dry cooling towers appear to be popular and satisfactory devices for some specific purposes. As a consequence of their employing different modes of heat transfer, each cooling system has its own advantages and disadvantages, respectively.

# CHAPTER III

## THEORY OF ENERGY AVAILABILITY METHODS

As we pointed out in Chapter I, it is well known that the cooling water temperature is one of the factors which affect the output of turbines. Before building a new power plant, a method which will reflect directly the effect of cooling water temperature on the power production rate is quite necessary for making a comparative analysis on the cooling systems available. In this chapter, we are going to develop such a method using availability methods which employ the concepts of available energy and then derive an equation containing all the pertinent parameters.

### Discussion on energy availability

There are two forms of energy to be considered--heat and work [7]. The first law of thermodynamics shows a balance between them. However, the second law of thermodynamics marks the distinction between them. The concepts of energy availability are derived from a combination of these two laws of thermodynamics. Since the time of Carnot (1824), the concept of potential work--in the sense of the maximum work which can be produced by a system or process--has been of concern to engineers dealing with power systems. The concept was inherent in the free energy and available energy functions of von Helmholtz and Gibbs (1873). In 1941, Keenan formulated the following measure of the maximum work of closed systems--a measure which he called "Availability":

$$A = E+P_oV-T_oS-(E_o+P_oV_o-T_oS_o) \qquad \text{(III-1')}$$

The subscript "o" denotes the closed system when it is in equilibrium with the surrounding medium so that the quantities $P_o$, $T_o$ and $(E_o+P_oV_o-T_oS_o)$ are constants. Since A is thus a function of the system properties E, V, and S, it may be regarded as being a property of the system for any given surrounding medium. Keenan refers to the property A as being "the maximum work which can be delivered to things other than the system and medium by the two unaided by any change in external things." The availability A is a measure of the potential work of systems. In regard to the potential work of processes, Keenan pointed out that the steady flow availability derived earlier by Darrieus and Keenan is given simply by:

$$A_{process} = (E+PV-T_oS)-(E_o+P_oV_o-T_oS_o) \qquad \text{(III-2')}$$

In 1958, Tribus suggested that the potential work of processes should be given by a balance of availability rather than by a balance equation for the term $E+P_oV-T_oS$, since $E+P_oV-T_oS$ is not a general measure of the potential work of open systems. Dr. R. B. Evans, the advisor of this thesis, carried out the generalization by replacing the term $(E_o+P_oV_o-T_oS_o)$ in availability by the term $\sum_c \mu_{co}N_c$; where $\mu$ is Gibb's free energy and N is the quantity of mass [36].

$$A_{generalization} = E+P_oV-T_oS-\sum_c \mu_{co}N_c \qquad \text{(III-3')}$$

Equation III-1′ and Equation III-2′ are special cases of Equation III-3′. This is a brief history of available energy. Evans also proved that Equation III-3 is the only consistent measure of potential work for a very large class of systems [36]. Tribus and McIrvine [35] have recently displayed the relationship between this function and statistical thermodynamics and information theory closely related to the energy availability is effectiveness. A further discussion on energy availability and effectiveness is given in Appendix I.

Availability analysis of a Rankine cycle

The available energy in a steady flow system may be defined as $b = h-T_os$ by neglecting the term $\sum_c \mu_{co}N_c = E_o+P_oV_o-T_oS_o$ which cancels out of steady flow processes. There is an inherent decrease in energy availability when heat transfers across a definite temperature difference or when a fluid flows through a pipe with friction. Both processes cause increases in entropy [9].

Any Rankine power cycle will consist of the following four main processes:

1. Heat input in the boiler or superheater.
2. Work output from the turbines.
3. Heat rejection in the condenser.
4. Work input in the pumps.

Processes 1 and 3 concern the heat transfer across a temperature difference. Processes 2 and 4 include some friction work. All of these four processes will cause a change in the energy availability of the working fluid. Thus, when analyzing thermal power cycles, two theoretical analyses can be made, i.e., the first law analysis and the second law analysis. An example is given here:

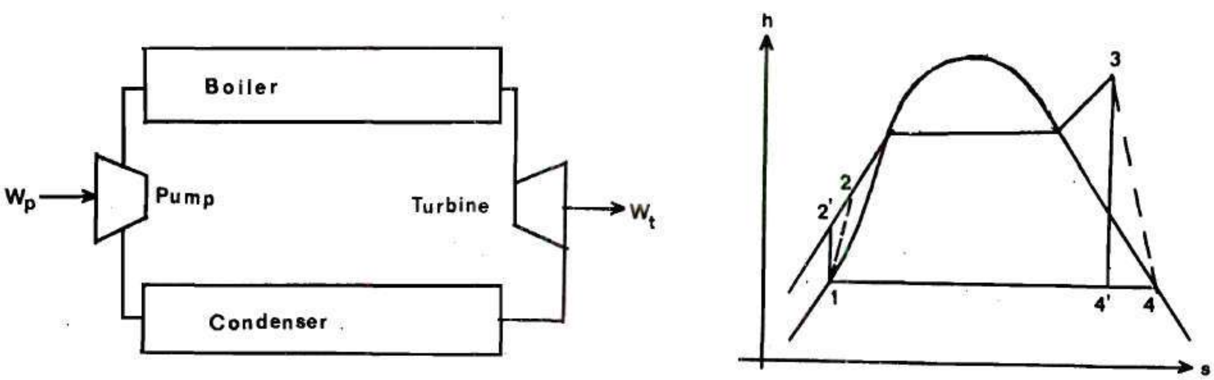



Figure 11. Simple Power Cycle Analysis

The following states are given

$P_1$ = 1 psia $\qquad$ $n_t = n_p = 0.8$

$P_3$ = 600 psia $\qquad$ $T_o$ = 70°F

$T_3$ = 800°F

From steam tables

$$h_1 = 69.7 \text{ BTU/lbm} \qquad s_1 = 0.1326 \text{ BTU/lbm-F}$$

$$b_1 = h_1 - T_o s_1 = 69.7 - 530 \times 0.1326 = 0$$

The pumping work

$$w_{sp} = v\,dp = o.o1614\ (600-1) \times 144/778 = 1.8 \text{ BTU/lbm}$$

$$w_p = w_{sp}/n_p = 1.8/0.8 = 2.25 \text{ BTU/lbm}$$

$$h_2 = h_1 + w_p = 69.7 + 2.25 = 71.95 \text{ BTU/lbm}$$

From steam table h = 71.95, P = 600 psia, we find that

$$s_2 = o.134 \text{ BTU/lbm-F}$$

$$b_2 = h_2' - T_o s_2' = 71.95 - 530 \times 0.134 = 1.03 \text{ BTU/lbm}$$

From steam table P = 600 psia, T = 800°F

$$h_3 = 1407.7 \text{ BTU/lbm} \qquad s_3 = 1.6343 \text{ BTU/lbm-F}$$

$$b_3 = h_3 - T_o s_3 = 1407.7 - 530 \times 1.6343 = 541.52 \text{ BTU/lbm}$$

The entropy of state 4' can be calculated by

$$s_4' = s_3 = s_{4f}' + X_4' \, s_{fg4}'$$

$$1.6343 = 0.1326 + X_4' \times 1.8456 \qquad X_4' = 0.8135$$

$$h_4' = h_{4f}' + X_4' h_{fg4}' = 69.7 + 0.8135 \times 1036.3 = 913.3 \text{ BTU/lbm}$$

The isentropic work is thus;

$$w_{st} = h_3 - h_4' = 1407.3 - 913.3 = 494.4 \text{ BTU/lbm}$$

The actual work is

$$w_t = w_{st} \times n_t = 494.4 \times 0.8 = 395.5 \text{ BTU/lbm}$$

The enthalpy of state 4 can be calculated

$$h_4 = h_3 - w_t = 1407.7 - 395.5 = 1012.2 \text{ BTU/lbm}$$

The quality of state 4

$$h_4 = h_{4f} + X_4 \times h_{fg4} \qquad 1012.2 = 69.7 + x_4 \times 1036.3$$

$$X_4 = 0.9095$$

The entropy of state 4 is

$$s_4 = s_{f4} + X_4 \times s_{fg4} = 0.1326 + 0.9095 \times 1.8456 = 1.6785 \text{ BTU/lbm-F}$$

$$b_4 = h_4 - T_o s_4 = 1012.2 - 530 \times 1.6785 = 121.6 \text{ BTU/lbm}$$

Summaries are given as follows:

| state | h BTU/lbm | b BTU/lbm |
|---|---|---|
| 1 | 69.7 | 0 |
| 2 | 71.95 | 1.03 |
| 3 | 1407.7 | 541.52 |
| 4 | 1012.2 | 121.6 |

(1) The First Law Analysis

| Energy in | BTU/lbm | percentage |
|---|---|---|
| Feedpump | 2.25 | 0.168% |
| Boiler | 1335.75 | 99.832% |
| Total | 1338 | 100 % |

| Energy out | BTU/lbm | percentage |
|---|---|---|
| turbines | 395.5 | 29.5 % |
| condenser | 942.5 | 70.5 % |
| Total | 1338 | 100 % |

Overall thermal efficiency = 395.5/1335.75 = 0.296

(2) The Second Law Analysis

| Availability gain | BTU/lbm | percentage |
|---|---|---|
| feed pump | 1.03 | 0.19 % |
| boiler | 540.49 | 99.81 % |
| Total | 541.52 | 100 % |

| Availability loss | BTU/lbm | percentage |
|---|---|---|
| turbine | 419.92 | 77.3% |
| condenser | 121.6 | 22.7% |
| Total | 541.52 | 100 % |

Overall thermal effectiveness = 395.5/540.49 = 0.731

A more complicated example is presented in Thermodynamics--Keenan [8].

It is obvious that the second law analysis provides much more detailed information than the simple first law analysis because several ideas are suggested as follows:

(1) The ambient temperature $T_o$, which can be taken to be the cooling water temperature leaving the cooling system before entering the condenser, enters the whole calculation as a variable parameter. This is a special merit of 2nd law analysis that the 1st law analysis doesn't reveal.

(2) The effect of cooling water temperature on the power output rate can be investigated by applying the effectiveness of the power plant. The definition of the effectiveness of a power plant is

$$e = \frac{\text{work output}}{\text{energy availability input}} \tag{III-1}$$

We know that both the work output and the availability energy input increase as $T_o$ decreases and vice versa. Thus, if we can find an equation of effectiveness as a function of $T_o$, namely,

$$e = f(T_o)$$

Then the work output rate can be calculated by

$$W_{net} = e \times \text{availability input}$$

$$W_{net} = f(T_o) \times \text{availability input} \qquad \text{(III-2)}$$

Letting the energy availability input = heat input x $g(T_o)$ and substituting into Equation III-2 yields,

$$W_{net} = \text{heat input} \times g(T_o) \times f(T_o)$$

$$= \text{heat input} \times k(T_o) \qquad \text{(III-3)}$$

Such an equation will reflect directly the effect of cooling water on the power output rate.

(3) The second law analysis also suggests some detailed improvements since the various irreversibilities may readily be isolated and their importance compared. We find that the most serious availability loss occurs in the boiler due to the very large temperature difference between the heat source and steam. In practice, the availability loss is reduced by incorporating superheaters which abstract some heat from the furnace gases before the boiler, and economizers which preheat

the feed water and abstract heat from the furnace gases after the boiler [10].

(4) Considering the effect of cooling water temperature on the power cycle and repeating the same calculation, we find that, with respect to $T_o$, the effectiveness of a power cycle is more stable than its efficiency. The statement can be proved by considering a simple power cycle in which heat is supplied to the steam in a boiler. The energy availability gained by the steam is

$$\begin{aligned} \dot{B}_{in} &= \dot{m}(h_e - T_o s_e) - (h_i - T_o s_i) \\ &= \dot{m}(h_e - h_i) - T_o(s_e - s_i) \end{aligned} \qquad \text{(III-4)}$$

where the subscripts e and i refer to the inlet and exit of the boiler. Since

$$\dot{m}(h_e - h_i) = \dot{Q}_h;\ \dot{m}(s_e - s_i) = \int \frac{d\dot{Q}_h}{T} = \frac{\dot{Q}_h}{T_{hav}} \qquad \text{(III-5)}$$

Substituting Equation III-5 into Equation III-4 yields:

$$\dot{B}_{in} = \dot{Q}_h\left(1 - \frac{T_o}{T_{hav}}\right) \qquad \text{(III-6)}$$

where the $\dot{Q}_h$ is the heat absorbed by steam and the $T_{hav}$ is the average temperature of the steam in the boiler. Substituting Equation III-6 into Equation III-1 we get

$$e = \frac{\dot{W}_{net}}{\dot{Q}_h(1 - \frac{T_o}{T_{hav}})} \qquad \text{(III-7)}$$

Since $W_{net}/Q_h = n_{power\ cycle}$, thus

$$e = n_{power\ cycle}\ (\frac{T_{hav}}{T_{hav} - T_o}) \qquad \text{(III-8)}$$

It is clear from Equation III-8 that when $T_o$ increases, the efficiency of the power plant tends to decrease, meanwhile, the denominator $T_{hav} - T_o$ also tends to decrease. Thus, we can conclude that the effectiveness of a power cycle is more stable, with respect to $T_o$, than its efficiency. The efficiency may be equal to the effectiveness only when $T_o$ is equal to absolute zero. It is impossible to reach such a low temperature. Therefore, the effectiveness is always larger than the efficiency of a power cycle. Also, as pointed out by Kreith [34], the effectiveness tends to be a constant, as will be demonstrated in this thesis.

The previous four statements are all derived from a comparison of the first and second law analysis and will be quite useful later on.

Derivation of Equation $e = f(T_o)$

Figure 12 illustrates the typical arrangement of a simplified power plant. The subscripts h and l refer to boiler and condenser. The output of the turbine is equal to the isentropic work minus the lost work and the pumping work

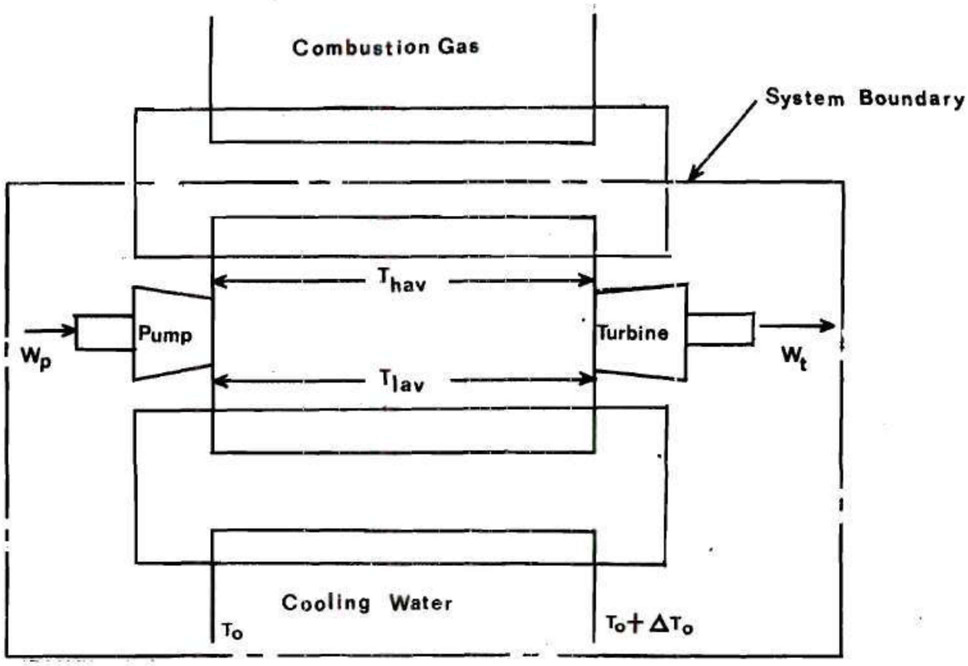



Figure 12. The Typical Simplified Power Plant

plus the pumping loss:

$$\dot{W}_t = \dot{m} \int_{p_l}^{p_h} v_t (dp)_s - \dot{m}\, h_{loss\ t} = \eta_t \dot{m} \int_{p_l}^{p_h} v_t (dp)_s \quad \text{(III-9)}$$

$$\dot{W}_p = \dot{m} \int_{p_l}^{p_h} v_p (dp)_s + \dot{m}\, h_{loss\ p} = \frac{1}{\eta_{pump}} \dot{m} \int_{p_l}^{p_h} v_p (dp)_s \quad \text{(III-10)}$$

The net work of the output is thus,

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_p \quad \text{(III-11)}$$

$$\dot{W}_{net} = \eta_t \dot{m} \int_{p_l}^{p_h} v_t (dp)_s - 1/\eta_p\, \dot{m} \int_{p_l}^{p_h} v_p (dp)_s \quad \text{(III-12)}$$

The energy availability supplied to the boiler is,

$$\dot{B}_{in} = \dot{W}_{net} + T_o\dot{m}(\dot{S}^c_{turbine}+\dot{S}^c_{pump}+\dot{S}^c_{condenser}+\dot{S}^c_{pipe}) + \dot{m}b_{out} \tag{III-13}$$

where the superscript c denotes entropy creation [35]. Repeating the effectiveness of the power cycle,

$$e = \dot{W}_{net}/\dot{B}_{in} \tag{III-14}$$

Substituting Equation III-12 and III-13 into Equation III-14, yields

$$e = \frac{\eta_{turbine}\ \dot{m}\int_{p_1}^{p_h} v_t(dp)_s - \frac{1}{\eta_{pump}}\ m\int_{p_1}^{p_h} v_p(dp)_s}{\dot{m}\int_{p_1}^{p_h} v_t(dp)_s - \dot{m}\Delta h_{loss\ t} + T_o\dot{m}(s_t^c+s_p^c+s_c^c+s_{pi}^c) + \dot{m}b_{out} - \frac{\dot{m}}{\eta_{pump}}\int_{p_1}^{p_h} v_p(dp)_s} \tag{III-15}$$

The enthalpy loss can be related to the entropy increase in a turbine. Since most of the turbines try to expand the steam to a low pressure, i.e., the state of exhaust steam is normally in the two phase region or near the saturation line on a Mollier diagram. In these regions, there are two special advantages that we have: (1) In two phase region, the constant pressure lines are straight lines and in the region near saturation line, the constant pressure lines bend upward a little bit; and (2) In the two phase region, the constant pressure and constant temperature lines overlap, and nearly overlap in the region near the saturation line. Figure 13 shows such a diagram duplicated from a Mollier diagram [11,12]. The enthalpy loss is

$$h_{loss\ t} = h_1 - h_1' \qquad \text{(III-16)}$$

From the thermodynamic basic relationships [13]

$$dh = T\ ds + v\ dp = \left(\frac{\partial h}{\partial s}\right)_p ds + \left(\frac{\partial h}{\partial p}\right)_s dp \ldots \qquad \text{(III-17)}$$

thus

$$T = \left(\frac{\partial h}{\partial s}\right)_p \qquad \text{(III-18)}$$

Since the constant pressure lines are constant temperature lines in the two phase regions, thus

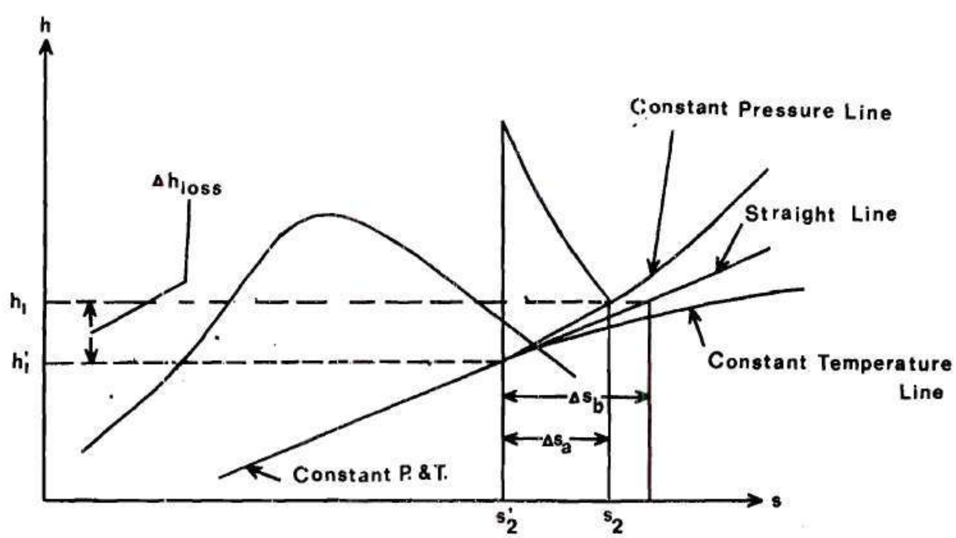



Figure 13. h-s Diagram

$$s^c_{turbine} = \frac{\Delta h_{loss\ t}}{T_1} \qquad \text{(III-19)}$$

or

$$\Delta h_{loss\ t} = T_1 \times s^c_{turbine} \qquad \text{(III-20)}$$

These two equations are exactly valid for the exhaust steam in the two phase region. In the superheated region near the saturation line, it still holds approximately well. Here a real example is checked.

| | |
|---|---|
| $P_h = 3000$ psia | $s_1' = s_h = 1.7163$ BTU/lbm-F |
| $T_h = 1600°F$ | $h_1' = 959.47$ BTU/lbm |
| $s_h = 1.7163$ BTU/lbm-F | $s_1 = 2.144$ BTU/lbm-F |
| $n_t = 0.7$ | $h_1 = 1217.6$ BTU/lbm |
| $P_1' = 1$ psia | $p_1 = 1$ psia |
| $T_1' = 101.74°F$ | $T_1 = 350°F$ |

The theoretical entropy increase is $\Delta s_a = s_2 - s_1 = 0.4277$. The entropy increase calculated by our method is

$$\Delta s_b = \frac{h_{loss\ t}}{T_1} = \frac{h_1' - h_1}{T_1'} = 0.46$$

The percentage of error $= \frac{0.46-0.4277}{0.4277} = 7.5\%$.

If we check the point of h = 1217.6 BTU/lbm and s = 2.144 BTU/lbm-F on a Mollier diagram, we find that the point is not near the saturation line, however, the error is limited to 7.5%. Thus, we can surely conclude that our method holds pretty well for a turbine which exhausts steam in the superheat region near the saturation line.

I also have to point out that the entropy increase $\Delta s_b$ calculated by Equation III-19 is always larger than the real entropy increase $\Delta b_a$ because the constant pressure line in the superheated region is an upward bended curve relative to the straight line lengthened from the same pressure line in the two-phase region. This fact has enabled our method of calculating the entropy increase to closely approximate to the real entropy increase.

Now we return to the derivation of our equation. Substituting Equation III-19 into Equation III-15 and rearranging $s^c_{tur}$, yields

$$e = \frac{\eta_t \dot{m}\int_{p_1}^{p_h} v_t (dp)_s - 1/\eta_{pump}\dot{m}\int_{p_1}^{p_h} v_p (dp)_s}{\dot{m}\int_{p_1}^{p_h} v_t (dp)_s + (T_o - T_1)\dot{m}\, s^c_{tur} + T_o \dot{m}(s^c_p + s^c_c + s^c_{pi}) + \dot{m}b_{out} - \frac{1}{\eta_{pump}}\int_{p_1}^{p_h} v_p (dp)_s} \tag{III-21}$$

Again,

$$\dot{m}s^c_{tur} = \frac{\dot{m}\Delta h_{loss}}{T_1} = \frac{(1-\eta_t)\ \dot{m}\int_{p_1}^{p_h} v_t (dp)_s}{T_1} \tag{III-22}$$

Substituting Equation III-22 into Equation III-21, yields

$$e = \frac{\eta_t \dot{m} \int_{p_1}^{p_h} v_t (dp)_s - 1/\eta_p \dot{m} \int_{p_1}^{p_h} v_p (dp)_s}{\dot{m} \int_{p_1}^{p_h} v_t (dp)_s + (T_o - T_1) \frac{(1-\eta_t)}{T_1} \dot{m} \int_{p_1}^{p_h} v_t (dp)_s + T_o \dot{m} (s_p^c + s_c^c + s_{pi}^c) + \dot{m} b_{out} - \frac{1}{\eta_{pump}} \dot{m} \int_{p_1}^{p_n} v_p (dp)_s \cdots} \tag{III-23}$$

Dividing Equation III-23 by $\dot{m} \int_{p_1}^{p_h} v_t (dp)_s$ yields

$$e = \frac{\eta_{tur} - \dfrac{\int_{p_1}^{p_h} v_p (dp)_s}{\eta_p \int_{p_1}^{p_h} v_t (dp)_s}}{1 + \left(\dfrac{T_o - T_1}{T_1}\right)(1-\eta_t) + \dfrac{T_o (s_p^c + s_c^c + s_{pi}^c) + b_{out}}{\int_{p_1}^{p_h} v_t (dp)_s} - \dfrac{\frac{1}{\eta_p} \int_{p_1}^{p_h} v_p (dp)_s}{\int_{p_1}^{p_h} v_t (dp)_s} \cdots} \tag{III-24}$$

For simplification, we neglect some insignificant terms in the above equation.

(1) The entropy increase in the pipe and pump are almost negligible in comparison with the entropy increase in the condenser.

(2) The ratio of isentropic pumping work to isetropic turbine work tends to be a very small number and for a given cycle, the variation in this ratio is almost negligible that for a given power cycle, it can be treated as a constant

without causing too much error.

Let

$$\frac{\int_{p_1}^{p_h} v_p (dp)_s}{\eta_p \int_{p_1}^{p_h} v_t (dp)_s} = K_p \tag{III-25}$$

Neglecting the insignificant terms and substituting Equation III-25 into III-24 results in

$$e = \frac{\eta_{turbine} - K_p}{1+(\frac{T_o - T_1}{T_1})(1-\eta_t) + \frac{T_o s^c_{condenser} + b_{out}}{\int_{p_1}^{p_h} v_t (dp)_s} - K_p} \tag{III-26}$$

<u>The entropy creation for a heat exchanger</u>

$T_a$ and $T_b$ are the average temperature of cooling water and heated water [14].

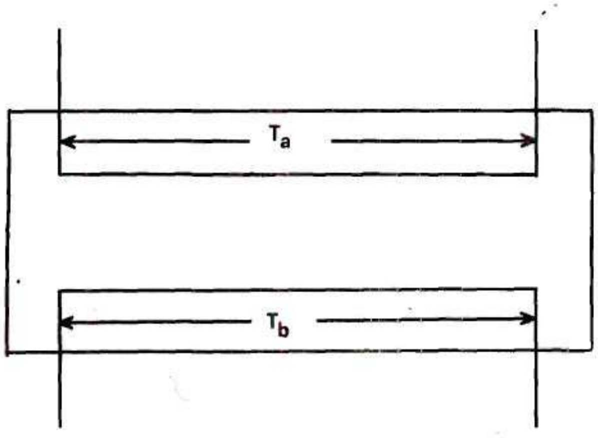



Figure 14. Heat Exchanger

$$S_c = -\frac{Q}{T_a} + \frac{Q}{T_b} = Q\left(\frac{T_a - T_b}{T_a T_b}\right) = Q\left(\frac{\Delta T_{ab}}{T_a T_b}\right) \qquad \text{(III-27)}$$

Therefore, the entropy increase in the condenser is

$$s_c^c = q_1\left(\frac{\Delta T_1}{T_{1av}\ T_{oav}}\right) \qquad \text{(III-28)}$$

The availability energy leaving the condenser is

$$b_{out} = q_1 - T_o \int_{T_o}^{T_o+\Delta T_o} dq_1/T = q_1\left(1 - \frac{T_o}{T_{oav}}\right) \qquad \text{(III-29)}$$

Substituting Equation III-29 and III-28 into III-26 yields

$$e = \frac{\eta_{turbine} - k_p}{1 + \left(\frac{T_o - T_1}{T_1}\right)(1 - \eta_{tur}) + \frac{q_1\left(\frac{\Delta T_1 T_o}{T_{1av} T_{oav}} + \frac{T_{oav} - T_o}{T_{oav}}\right)}{\int_{p_1}^{p_h} v_t (dp)_s} - K_p} \ldots \text{(III-30)}$$

From the first law of thermodynamics

$$\dot{Q}_1 = \dot{Q}_h - \eta_t \dot{m} \int_{p_1}^{p_h} v_t (dp)_s + \dot{W}_{pump} \qquad \text{(III-31)}$$

In this case, we can neglect the pumping work because it is so small in comparison with the heat input $\dot{Q}_h$. Thus, dividing by the mass flow rate $\dot{m}$

$$q_1 = q_h - \eta_t \int_{p_1}^{p_h} v_t (dp)_s \tag{III-32}$$

Dividing Equation III-32 by the isentropic work yields

$$\frac{q_1}{\int_{p_1}^{p_h} v_t (dp)_s} = \frac{q_h}{\int_{p_1}^{p_h} v_t (dp)_s} - \eta_t \tag{III-33}$$

From Equation III-6 with $b_{in} \cong \int_{p_1}^{p_n} v_t (dp)_s$ and $T_o$ replaced by $T_{lav}$, we have,

$$\frac{q_h}{\int_{p_1}^{p_h} v_t (dp)_s} = \frac{T_{hav}}{T_{hav} - T_{lav}} \tag{III-34}$$

Substituting Equation III-33 and III-34 into Equation III-30 yields

$$e = \frac{\eta_t - k_p}{1 + \left(\frac{T_o - T_1}{T_1}\right)(1 - \eta_t) + \left(\left(\frac{T_{hav}}{T_{hav} - T_{lav}}\right) - \eta_t\right)\left(\frac{\Delta T_1 T_o}{T_{lav} T_{oav}}\right) + \left(\frac{T_{oav} - T_o}{T_{oav}}\right)) - k_p} \tag{III-35}$$

...

To get a good approximation $T_{oav} = T_o + \frac{1}{2} \Delta T_o$;

$$\frac{T_{oav} - T_o}{T_{oav}} = \frac{\Delta T_o}{2T_o + \Delta T_o} \tag{III-36}$$

Substituting Equation III-36 into Equation III-35 yields

$$e = \frac{\eta_t - K_p}{1+\left(\frac{T_o - T_l}{T_l}\right)(1-\eta_t)+\left(\left(\frac{T_{hav}}{T_{hav}-T_{lav}}\right)-\eta_t\right)\left(\left(\frac{\Delta T_L T_o}{T_{lav}T_{oav}}\right)+\left(\frac{\Delta T_o}{2T_o+\Delta T_o}\right)\right)-k_p} \quad \ldots \quad \text{(III-37)}$$

$$= \frac{\eta_t - K_p}{1 + \text{correction term}} \quad \text{(III-38)}$$

The correction term represents the sum of the corresponding terms in Equation III-37.

It is obvious from Equation III-37 that the efficiency of a turbine is an important factor in the effectiveness of a power plant. For a given turbine and inlet steam, the efficiency is a function of the back pressure. Figure 15 illustrates how the steam turbine efficiency varies as the back pressure increases in a 400 MW power plant [15].

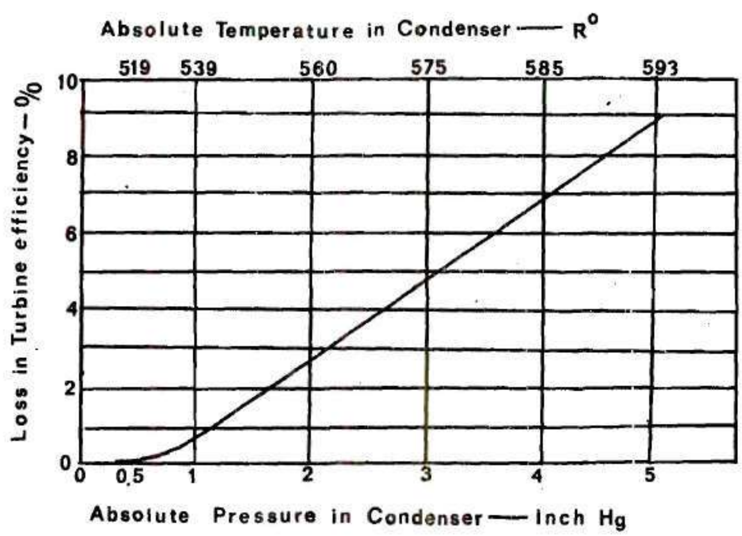



Figure 15. Loss in Turbine Efficiency with Drop in Back Pressure Vacuum

An example will be given here to check Equation III-37. The given conditions are listed as follows.

1. The efficiency at the design point is 0.9 and decreases according to Figures 3-5 with increase in back pressure.

2. In most cases, the condensers now being used have a typical range of 20°F, and temperature across the condenser is 15°F.

$$\Delta T_1 = 15°F \quad \Delta T_o = 20°F$$

When the efficiency of the turbine changes, then $\Delta T_L$ and $\Delta T_c$ also change, however, the amount of variation is so small in comparison with $T_{lav}$ or $T_o$. Thus, $\Delta T_c$ and $\Delta T_l$ are taken to be constant.

3. $T_{hav}$ is selected to be 1000°R.

4. $K_p$ is selected to be a typical value of 0.02.

The results are tabulated in Table 1. The effectivenesses are plotted against the cooling temperature $T_o$ on Figure 16.

Figure 16 clearly shows that the effectiveness of a power plant is almost a constant if the working range is limited to some range around the design point of its turbine and this is the significant range of interest for evaluation. Thus, we can conclude that the significant effectiveness of a power plant is almost a constant, a principle which may

Table 1. Effectivenesses of a Power Plant

| $\eta_{tur}$ | $T_1$ | $\Delta T_1$ | $\Delta T_o$ | $T_o$ | $T_{oav}$ | $T_{hav}$ | Corr | $K_p$ | e |
|---|---|---|---|---|---|---|---|---|---|
| 0.897 | 500 | 15 | 20 | 475 | 485 | 1000 | 0.05000 | 0.02 | 0.835238 (impossible)* |
| 0.900 | 519 | 15 | 20 | 494 | 504 | 1000 | 0.05205 | 0.02 | 0.836462 |
| 0.895 | 539 | 15 | 20 | 514 | 524 | 1000 | 0.05423 | 0.02 | 0.830000 |
| 0.875 | 560 | 15 | 20 | 535 | 545 | 1000 | 0.05686 | 0.02 | 0.809000 |
| 0.855 | 575 | 15 | 20 | 550 | 560 | 1000 | 0.05882 | 0.02 | 0.788613 |
| 0.833 | 585 | 15 | 20 | 560 | 570 | 1000 | 0.06024 | 0.02 | 0.766808 |
| 0.812 | 593 | 15 | 20 | 568 | 578 | 1000 | 0.06143 | 0.02 | 0.746163 |

*$T_o$ must be greater than the freezing temperature (492 R°).

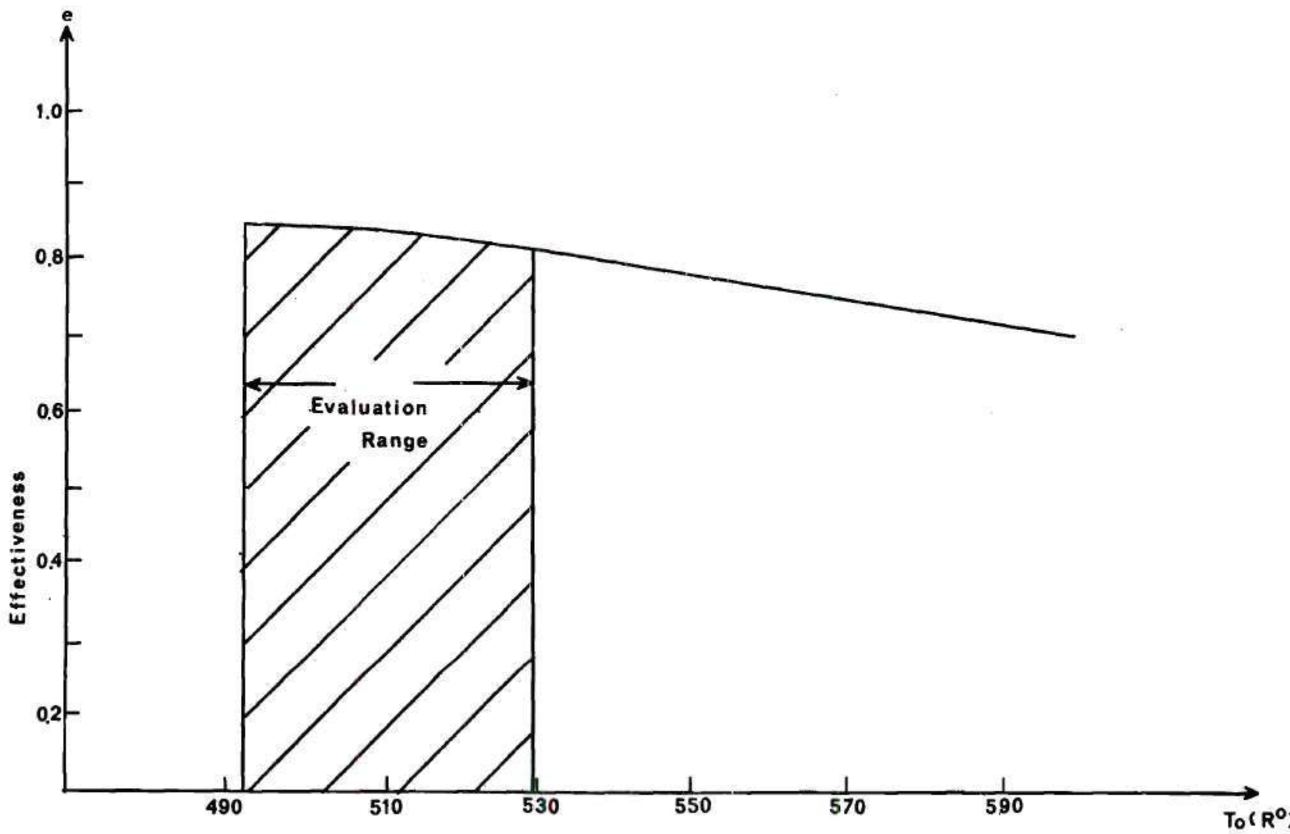



Figure 16. Effectiveness of a Power Plant as a Function of Cooling Water Temperature $T_o$

be called the principle of constant effectiveness.

Figure 16 applies to a given plant with a fixed design point. For design purposes, it would be better to have a plot of effectiveness versus $T_o$ as an envelope of optimum design points as shown in Figure 16a. Since optimization curves generally tend to be rather flat, the principle of constant effectiveness will still apply, as Figure 16a will indicate.

When designing a power plant, a comparison of capital costs of each type of cooling system is definitely needed.

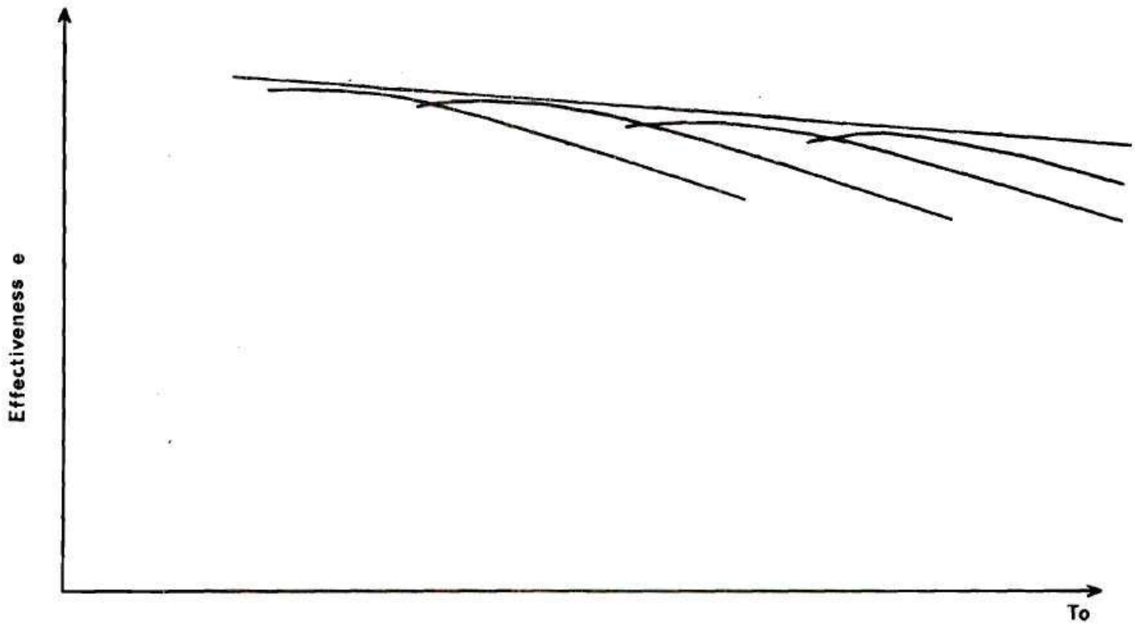



Figure 16a. Effectiveness as a Function of Cooling Water Temperature for Optimum Design Points

Such a comparison includes [33]:

1. Investment cost comparison--cost for the cooling tower and its equipment, land cost, etc.

2. Operating cost comparison--cost for pumping water, makeup water, steam generation rate, maintenance, etc.

The principle of constant effectiveness is quite useful in selecting the optimum cooling water temperature $T_o$ in relation to the above cost. The optimization is carried out by letting the profits of a power plant $\dot{P}_r$ be

$$\dot{P}_r = C_e \dot{W} - \frac{\dot{K}}{T_{o'} - T_r} - C_h \dot{Q}_h - \dot{C}_r \qquad \text{(III-39)}$$

where

$Ce\,\dot{W}$ is the profit of the power output.

$\frac{\dot{K}}{T_o - T_r}$ is the cost of the cooling towers including construction and operation. $T_r$ is a fixed wet-bulb temperature and $\dot{K}$ is a constant for this simplified analysis, while $T_o \leqq$ surrounding temperature.

$C_h \dot{Q}_h$ is the cost of fuel consumption, $C_h$ is a constant, $\dot{C}_r$ is the remaining plant cost.

From Equation III-7

$$\dot{Q}_h = \frac{\dot{W}}{e(1 - T_o/T_{hav})}$$

Substituting Equation III-7 into III-39 yields

$$\dot{P}_r = Ce\,\dot{W} - \frac{\dot{K}}{T_o - T_r} - \frac{C_h \dot{W}}{e(1 - T_o/T_{hav})} - \dot{C}_r \qquad \text{(III-40)}$$

By applying the principle of constant effectiveness e can be treated as a constant as discussed previously. The optimum cooling water temperature $T_o$ under the condition of constant power output is thus (noting that Ce, $\dot{W}$, and $\dot{C}r$ will be constant with respect to $T_o$)

$$\frac{\partial \dot{P}_r}{\partial T_o} = \frac{\dot{K}}{(T_o - T_r)^2} - \frac{T_{hav}\, C_h\, \dot{W}}{e} \times \frac{1}{(T_{hav} - T_o)^2} = 0$$

Rearranging the above equation yields

$$(T_o)_{optimum} = \frac{T_{hav} + (\frac{T_{hav}\ C_h\ \dot{W}}{e\ K})^{1/2}\ T_r}{1 + (\frac{T_{hav}\ C_h\ W}{e\ K})^{1/2}} \qquad \text{(III-41)}$$

Equation III-41 is valid only under the given assumptions, so that a cooling tower will be used only if K is small enough so that $(T_o)_{optimum} \leqq$ Surrounding Temperature.

Equation III-37 is only valid for simple power cycles. In real cases, the regenerative, topping, and reheat processes may be added to simple cycles. Therefore, it is necessary that Equation III-37 be extended to apply in general, so that the principle of constant effectiveness can be used in general too. The complete analytical work is suggested for further research. Only a brief investigation is presented here.

1. Regenerative cycles. Assuming that the steam mass for regenerative processes is $m_1$ and for the original cycle is $m_2$, the overall effectiveness of the whole cycle will be

$$e_{overall} = \frac{m_1}{m_1+m_2}\ e_1 + \frac{m_2}{m_1+m_2}\ e_2 \qquad \text{(III-43)}$$

Where $e_1$ and $\dot{e}_2$ can be approached by the same analytical processes as discussed in previous sections, so that $e_1$ and $e_2$ will tend to be constant as before. Hence, $e_{overall}$ will tend to be constant. The principle of constant effectiveness remains valid for regeneration.

2. <u>Topping cycles.</u> The topping cycles may be composed of two, three or more cycles in general. Thus,

$$e_1 = \frac{w_1}{b_1}$$

$$e_2 = \frac{w_2}{b_2}$$

$$e_3 = \frac{w_3}{b_3}$$

Where $e_1$, $e_2$, and $e_3$ will be similar to Equation III-37. There will be certain factors which relate $b_1$, $b_2$ and $b_3$ as follows:

$$b_2 = b_1 \times \alpha$$

$$b_3 = b_2 \times \beta = b_1 \times \alpha\beta$$

Therefore, the overall effectiveness will be

$$e_{overall} = \frac{w_1+w_2+w_3}{b_1} \qquad \text{(III-44)}$$
$$= e_1 + e_2\alpha = e_3\alpha\beta$$

The effectiveness values $e_1$, $e_2$ and $e_3$ will be virtually constant as before. Thus, for given values of $\alpha$ and $\beta$, the effectiveness e will be constant, so the principle of constant effectiveness remains valid for topping processes.

3. Reheat cycles. The reheat cycles consist of several stages of reheat and turbine. Therefore, the effectiveness equation will not be identical to Equation III-37. Basic procedures are suggested here.

$$e = \frac{\sum_i w_i}{\sum_i b_i} \tag{III-45}$$

$$= \sum_i \frac{w_i}{\sum_i b_i}$$

Let $\alpha_i = \frac{b_i}{\sum_i b_i}$ and noting that $\frac{w_i}{b_i}$ is the effectiveness $e_i$ of the i'th reheat stage, then we have

$$e_{overall} = \sum_i \alpha_i e_i$$

The effectiveness values $e_i$ each remain virtually constant as before. Thus for given values of $\alpha_i$, $e_{overall}$ will remain virtually constant. Similar results will, of course, occur if we consider systems which combine regeneration, reheat and topping. We may thus conclude that the principle of constant effectiveness applies to steam power plants in general.

# CHAPTER IV

# THE PERFORMANCE OF NATURAL-DRAUGHT WET AND DRY COOLING TOWERS

Among the heat rejection systems that we presented in Chapter II, natural draught wet cooling towers recently have interested power engineers the most, owing to the following several reasons:

1. Reasonable costs
2. Easy maintenance
3. Large capacity for heat rejection
4. Eliminating thermal pollution

In this chapter, such two types of cooling towers are designed to dispose a 1000 MW power plant waste heat. The performances of each type of cooling tower are then investigated and compared by our methods of energy availability.

## Part A: Natural Draught Wet Type Cooling Towers

The design of wet type cooling towers requires a knowledge of psychrometry, which is a study of moisture content of air. Air is composed mainly of oxygen, nitrogen, rare gases, and water vapor in varying quantities depending on its temperature and humidity.

### Total Heat of Air

The total heat of air is the arithmetical sum of sensible heats of air and water vapor, plus the latent heat of vaporization of the water [16]. For saturated air:

$$h' = X'_{air}T_d + X'_{air}\lambda + 0.24T_d \qquad \text{(IV-1)}$$

For unsaturated air:

$$h = X_{air}T_{dp} + X_{air}\lambda_{dp} + 0.45X_{air}(T_d - T_{dp}) + 0.24T_d \qquad \text{(IV-2)}$$

Where the subscripts d and dp refer to dry bulb temperature and dew point temperature. 0.45 and 0.24 are the specific heats of vapor and air, respectively, from 0°F to 120°F. The total heat of air calculated by Equation IV-1 and IV-2 can be measured from a fixed-datum such as 0°F or 32°F. Table A-1 in the appendix shows total heat of saturated air taking 32°F as datum.

The properties of air and water vapor mixtures can be represented graphically on a single chart referred to as the Psychrometric chart. On this chart, absolute humidities are plotted against dry bulb temperatures and lines of constant relative humidity and wet bulb temperatures are added [17] (Figure A-1).

Heat Transfer in the Cooling Tower

Consider a droplet of water falling through the cooling tower. Heat is transferred in four ways, namely, conduction, convection, radiation, and evaporation. Quantitative treatment of cooling tower performances by dealing with mass and heat transfer separately is very laborious. Therefore, Merkel's total heat theory has been almost universally adopted for the calculation of cooling tower heat transfer. Briefly, Merkel's theory states that all the heat transfer taking place at any position in the cooling tower is proportional to the difference between the total heat of air and the total heat of saturated air at the temperature of water at that point in the tower.

$$Q = K \cdot S(H_w - H_g) \qquad \text{(IV-3)}$$

Adapting this equation to the cooling tower yields [18]

$$dq = K \cdot a(h' - h)dV = Ka(h' - h)dV \qquad \text{(IV-4)}$$

where

- K  Merkel's heat transfer coefficient
- a  Mean area of water-air interface per cubic foot of packed volume
- V  Height of the packing
- h'  Saturated air enthalpy at water temperature
- h  Ambient air enthalpy

K and a are generally combined together to form Ka.

Another basic equation for heat transfer in a wet cooling tower can be derived simply by considering a total heat diagram, Figure 17. Cooling water is cooled along the

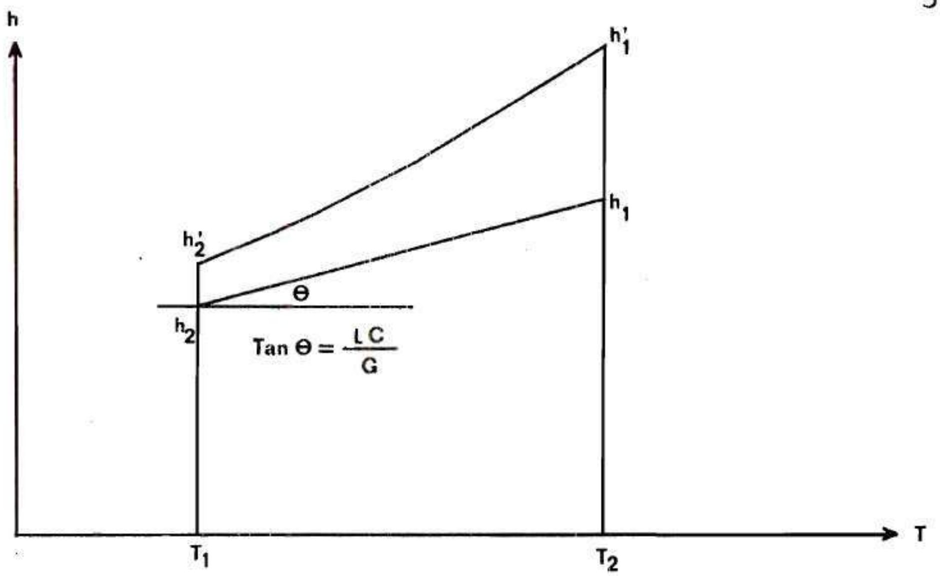



Figure 17. Total Heat Diagram

saturation line, while the air is heated through the tower [19]. The total heat decrease of water must be equal to the total heat increase of air. Thus,

$$dQ = dLCT = G\ dh \tag{IV-5}$$

(C = 1 BTU/lbm-F for water). The gas loading G remains constant throughout the tower because it is based on bone-dry gas only. The liquid loading L is not quite constant, owing to the evaporation of water into bone-dry air. The saturation line loss from water to air amounts to 2% of water circulated over the tower and may be considered constant without causing serious error [16].

$$dq = LC\ dT = G\ dh \tag{IV-6}$$

Mean Driving Force

Merkel's theory of heat transfer refers only to a single point in the tower whereas water and air conditions vary throughout the tower. It is, therefore, necessary to use the mean value of the enthalpy difference. The "mean position" is that position at which the water temperature is the arithmetic mean of inlet and outlet water temperature. A chart has been prepared which indicates the value of a factor f by two parameters $\Delta h_m/\Delta h_1$ and $\Delta h_m/\Delta h_2$. Such a chart is devised by W. L. Stevens and is presented in [17] and [20] (Figure A-2). The mean driving force is thus

$$h_{md} = \Delta h_m \times f \qquad \text{(IV-7)}$$

Required Height of Packing. Combining Equations IV-4 and IV-6 yields

$$dq = LC\ dt = G\ dh = K_a(h'-h)dV \qquad \text{(IV-8)}$$

Integrating Equation IV-8 gives

$$q = LC\Delta T = Ka\ h_{md}V \qquad \text{(IV-9)}$$

Thus, the required height is

$$V = \frac{L \times \Delta T}{Ka \times h_{md}} \qquad \text{(IV-10)}$$

Merkel's Cooling Factor. From Equation IV-8 we have

$$\frac{Ka}{G} \int_{o}^{V} dV = \int_{h_1}^{h_2} \frac{dh}{h'-h} \qquad \text{(IV-11)}$$

By integrating Equation IV-11 we get

$$\frac{Ka\ V}{G} = \frac{\Delta h}{h'_m - h_m} \qquad \text{(IV-12)}$$

Substituting $h_m = h_1 + 1/2\Delta h$, dividing by G and letting $G\ h = L\Delta T$ yields

$$\frac{h'_m - h_1}{\Delta T} = \frac{L}{Ka\ V} + \frac{1}{2\ G} \qquad \text{(IV-13)}$$

Merkel's cooling factor is defined as

$$\alpha = \frac{h'_m - h_1}{\Delta T} = \frac{L}{KaV} + \frac{1}{2\ G} \qquad \text{(IV-14)}$$

From the definition of Merkel's cooling factor, it is obvious that $\alpha$ depends on cooling range $\Delta T$, $h'_m$ and $h_1$. Wood and Betts have prepared charts expressing Merkel's cooling factor in terms of wet-bulb temperature, cooling range, and recooled water temperature. Such a figure is shown on [21] (Figure A-3).

Volume-Transfer Coefficient Ka. The experiments of a given cooling tower packing show that Ka depends only on the design of the packing itself and air and water loading. It is practically independent of water temperature. Thus,

$$Ka \alpha L^{m} G^{n} \tag{IV-15}$$

Combining with its height V results in

$$KaV = \lambda L^{m} G^{n} \tag{IV-16}$$

Substituting Equation IV-16 into IV-14 yields

$$\alpha = \frac{L}{\lambda L^{m} G^{n}} + \frac{L}{2G} \tag{IV-17}$$

Since L/2G is dimensionless, $L/L^{m}G^{n}$ must be also dimensionless. Thus,

$$1 - m - n = 0 \tag{IV-18}$$

or

$$1 - m = n \tag{IV-19}$$

Substituting Equation IV-19 into Equation IV-16 yields

$$\frac{Ka\ V}{L} = \lambda \left(\frac{L}{G}\right)^{-n} \tag{IV-20}$$

The transfer characteristics of a packing may therefore be defined by the two factors m and n. Table A-2 shows some basic forms of packing tested and their experimental data [22].

Air Flow Equation. The draught of natural draught cooling tower is due to the difference of air density between inside space and outside space of the tower and may be equated to the pressure difference necessary to maintain a flow of air through the tower. If the resistance of the tower to air flow can be regarded as being predominantly due to inertia losses occasioned by solid system, as distinct from friction loss and the drag of the falling water [21],

$$H\Delta\rho = \frac{N\rho u^2}{2g} \tag{IV-21}$$

$$= \frac{N\ 1/\rho\ \rho^2 u^2}{2g} \tag{IV-22}$$

$$= \frac{N\ v\ G^2}{2g} \tag{IV-23}$$

Several further steps have been carried out and result in

$$G^3N = 8210\ H\ L\Delta T(\frac{\Delta T}{\Delta h} + 0.3124) \tag{IV-24}$$

The derivation of this equation is shown in Appendix II. Let

$$F(T_d,T_w) = 8210\ (\frac{\Delta T}{\Delta h} + 0.3124) \tag{IV-25}$$

$F(T_d,T_w)$ depends on $\Delta T$ and $\Delta h$ only, i.e., it is a function of dry bulb and wet bulb temperature only. Figure A-4 indicates the value of $F(T_d,T_w)$ against wet bulb temperature and the difference between dry and wet bulb temperature. Substituting Equation IV-25 into Equation IV-24 and taking its cubic root result in

$$G\ N^{1/3} = \sqrt[3]{H\ L\Delta T\ F(T_d,T_w)} \qquad \text{(IV-26)}$$

Dividing water loading L by Equation IV-26 yields

$$\frac{L}{G\ N^{1/3}} = \frac{L}{\sqrt[3]{H\ L\Delta T\ F(T_D,T_W)}} \qquad \text{(IV-27)}$$

From Equation IV-27, $\frac{L}{G\ N1/3}$ can be evaluated simply by calculating the right side of the equation.

Chilton Performance Coefficient. The Chilton coefficient is defined as

$$C = \frac{\alpha}{(\frac{L}{GN^{1/3}})} \qquad \text{(IV-28)}$$

By substituting Merkel's cooling factor into Equation IV-28, we have

$$C = \frac{N^{n/3}}{(\frac{L}{GN^{1/3}})}(\frac{L}{G\ N^{1/3}})^{n-1} + \frac{N^{1/3}}{2} \qquad \text{(IV-29)}$$

The Chilton coefficient of a cooling tower has been investigated and the results show that for a given cooling tower, the Chilton coefficient tends to remain constant under all different working conditions such as dry and wet bulb temperature, water load, cooling range, wind speed, or wind direction. This fact can reasonably be regarded as a strong indication, though not as a conclusive proof, that there is a linear correlation between $\alpha$ and $L/GN^{1/3}$, i.e., a constant Chilton coefficient for a given tower. This concept will provide a valid basis for the comparison of different tests on the same tower [21].

Some of the experimental data on Chilton coefficients are tabulated on Table A-3 which will be quite useful when designing a new tower.

Duty Coefficient [23]. Substituting Equation IV-27 into Equation IV-28 and then replacing L by W/A yields,

$$\frac{C}{A^{2/3}\ H^{1/3}} = \frac{\sqrt[3]{\alpha \Delta T\ F(T_d, T_w)}}{W^{2/3}} \qquad \text{(IV-30)}$$

Duty coefficient is defined as

$$D = \frac{A}{C}\left(\frac{H}{C}\right)^{1/2} = \frac{W}{\alpha\sqrt{\alpha\ \Delta T\ F(T_d, T_w)}} \qquad \text{(IV-31)}$$

This equation can be used to evaluate the dimension of a tower for a given water load W. The Chilton coefficient can be estimated from the previous experimental data that is tabulated in Table A-3. The required dimension of a cooling tower is thus,

$$A(H)^{1/2} = \frac{C(C)^{1/2}}{\alpha\sqrt{\alpha\Delta T\ F(T_d,T_w)}} \qquad \text{(IV-32)}$$

<u>The Number of Towers.</u> Currently, the ratio of height of a cooling tower to its base diameter is usually taken to be 3:2 or 5:4 and the height of the natural draught cooling tower is limited to 120m or 390 ft by commerical availability. If the ratio of 3:2 is employed

$$A(H)^{1/2} = \frac{\Pi D^2 H^{1/2}}{4} = \frac{\Pi}{4}\left(\frac{2H}{3}\right)^2 (H)^{1/2} = 0.3491\ H^{2.5} \qquad \text{(IV-33)}$$

The maximum of $A(H)^{1/2}$ is thus determined if $H_{max} = 390$ ft. Thus,

$$A(H)^{1/2}_{max} = 1.05 \times 10^6 \qquad \text{(IV-34)}$$

The minimum number of cooling towers required is

$$N_{ct\ min} = \frac{C(C)^{1/2}W}{1.05 \times 10^6\alpha(\alpha\Delta T\ F(T_d,T_w))^{1/2}} \qquad \text{(IV-35)}$$

Other factors that affect the performance of a wet type cooling tower: Several other factors may affect the performance of a natural draught cooling tower, namely, wind velocity, barometric pressure, concentration or composition of dissolved or suspended solids in the circulating water, dynamic stability of the atmosphere and type of water spray.

Among these factors, the wind speed is the most significant [24]. Experiments show that strong wind has an adverse effect on wet cooling towers. This effect is attributed mainly to the disturbance of the air velocity distribution caused by the wind. This nonuniformity reduces the heat transfer and at the same time increases the effective resistance of the cooling stack and eliminator. Wind is also known to cause a small increase of draught but experimental evidence suggests that the beneficial effect is more than offset by the disturbance of the air velocity distribution [25].

The above analysis of designing a wet type cooling tower will be applied to design wet cooling towers for a 1000 MW power plant.

A supercritical steam power plant has a maximum steam pressure of 4000 psia and a temperature of 1000°F. Steam is expanded 115°F or 1.483 psia. The overall efficiency of turbines (several turbines are installed in series) is 0.9. The pump efficiency is taken to be 0.6. The output rate is 1000 MW.

Power cycle analysis:

| | | | |
|---|---|---|---|
| $h_1$ = 82.93 BTU/lbm | | $w_s$ | = 581.15 |
| $w_{ps}$ = 11.97 BTU/lbm | | $w$ | = 523.05 |
| $w_p$ = 19.95 BTU/lbm | | $h_4$ | = 883.8 |
| $h_2$ = 102.88 BTU/lbm | | $X_4$ | = 0.778 |
| $h_3$ = 1406.8 BTU/lbm | | $w_{net}$ | = 503.71 |
| $s_3$ = 1.4482 BTU/lbm-F | | $q_{rej}$ | = 800.87 |
| $x_4'$ = 0.722 | | | |
| $h_4'$ = 825.65 | | | |

The steam flow rate is

$$\dot{m}_{steam} = \frac{\text{power output rate}}{w_{net}}$$

$$= 6.776 \times 10^6 \text{ lbm/hr}$$

The waste heat rejection rate is thus

$$\dot{Q}_{rej} = \dot{m}_{steam} \times q_{rej} = 5.4267 \times 10^9 \text{ BTU/hr}$$

The heat rejection rate of this cycle at different steam exhaust temperatures is plotted in Figure 18.

Design Conditions: The design conditions of the heat rejection system are listed below:

| | |
|---|---|
| Terminal temperature difference | 5°F |
| Range | 20°F |
| Approach | 20°F |
| Cooling water inlet temperature | 110°F |
| Cooling water outlet temperature | 90°F |
| Dry bulb temperature | 80°F |

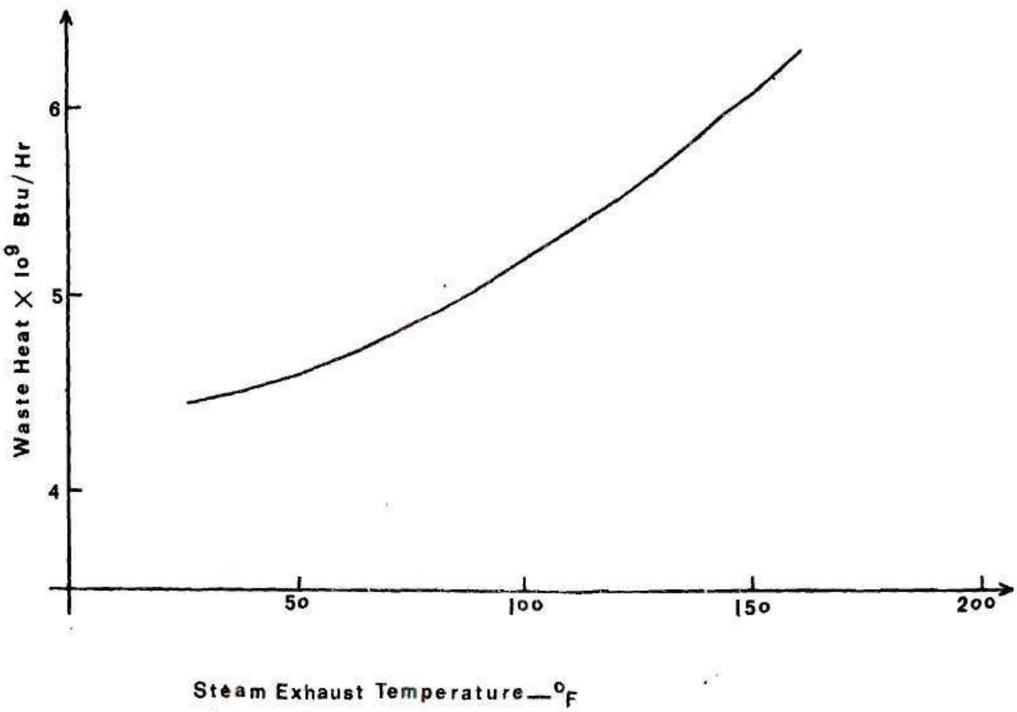



Figure 18. The Heat Rejection Rate of a Supercritical Cycle 4000 psia, 1000°F at Different Steam Exhaust Temperatures

Wet bulb temperature 70°F
Packing (Table A-2 No. 21)

Cooling water flow rate $W_a = \frac{\dot{Q}_{rej}}{C\Delta T_o} = 2.713 \times 10^8$ lbm/hr.
Merkel's cooling factor is decided by $T_w$ - 70°F, $\Delta T_o$ = 20°F, recooled temperature $T_o$ = 90°F and Figure A-2.

$$\alpha = 1.9$$

$F(T_d, T_w)$ = 5700 (From Figure A-4)

$$D = \frac{W_a}{\alpha(\alpha \Delta T_o F(T_d, T_w))^{1/2}} = 3.068 \times 10^5 = \frac{A}{C}(\frac{H}{C})^{1/2}$$

The Chilton coefficient C can be expected to be around 5.5 by Table A-3, No. F if the packing is about 3.5 ft in height. Thus,

$$A(H)_a^{1/2} = 3.957 \times 10^6$$

Thus, from Equation IV-35 the minimum number of towers is 4 by commercial availability. Let the number of towers be 8. The water flow rate for each tower is $\frac{W_a}{N_{ct}} = 3.391 \times 10^7$ lbm/hr and $A(H)^{1/2}$ for each tower is $\frac{A(H)_a^{1/2}}{N_{ct}} = 4.946 \times 10^5$.
Letting the height and base diameter ratio be 3:2 yields,

$$H = 288.755 \text{ ft} \quad A = 2.910 \times 10^4 \text{ ft}^2$$

From Table A-2 No. 21 $\lambda = 0.21$ and $n = 0.69$ and we assume that the height of packing is 4 ft. Substituting $\lambda = 0.21$, $n = 0.69$ and $V = 4$ into Equation IV-17 and IV-20 results in

$$1.9 = \frac{1}{4 \times 0.21}\left(\frac{L}{G}\right)^{0.69} + \frac{L}{2G}$$

L/G is found to be 1.16. We have to check the height of packing that we assumed. From a psychrometric chart we find that $h_{g1} = 26.4$ BTU/lbm and $h_{g2} = h_{g1} + L\Delta T/G = 49.6$ BTU/lbm. Thus $h_{gm} = 38.15\ \frac{BTU}{lbm}$. From Table A-1 $h_{w1} = 48.6$, $h_{w2} = 85.5$ and $h_{wm} = 67.05$ BTU/lbm

$$\Delta h_1 = 22.2 \quad \Delta h_m = 28.9 \quad \Delta h_2 = 35.9$$

and

$$\frac{\Delta h_m}{\Delta h_1} = 1.30 \quad \frac{\Delta h_m}{\Delta h_2} = 0.805$$

From Figure A-2 we have $f = 0.982$. Thus, the mean driving force $h_{md} = 0.982 \times 28.9 = 28.4$ BTU/lbm. The required height is calculated by Equation IV-10 $V = 3.72$ ft. The calculated height is very close to the assumed height. The Chilton coefficient that we assumed to be 5.5 has to be checked also. We know that $L = \frac{W}{A} = 1165\ lbm/ft^2\text{-}hr$, $\Delta T_o = 20°F$, $F(T_d, T_w) = 5700$ and $H = 288.75$.

$$G^3N = H \times L \times \Delta T_o \times F(T_d, T_w) = 38.35 \times 10^9$$

Thus $GN^{1/3} = 3.3722 \times 10^3$. Substituting L, and $GN^{1/3}$ into Equation IV-28 yields

$$C = 5.499$$

This also proves that our previous assumption of the Chilton coefficient is correct.

Part B: Natural Draught Dry Type Cooling Tower

There are two alternative systems, as we discussed in Chapter II, for dry type cooling towers [26,27]:

1. Direct system--the steam exhausted from the turbine is made to condense in air-cooled extended-surface condensers. The condensate is then pumped back into the boiler feed circuit (GEA concept).

2. Indirect system--the steam leaving the turbine is condensed by mixing with water in a direct-contact or spray condenser. A proportion of the condensate water equivalent to the exhaust steam condensed is returned into the boiler feed circuit, while the balance, which is the greater amount representing the spray water quantity, is circulated through an air cooled heat exchanger. The cooled water is then sprayed into the condenser again. There is no need for any make-up water (Heller concept).

A brief comparison is made here:

a. The capital investment is lower for a direct system than an indirect system.

b. Large-bore exhaust piping may prove difficult to accommodate to a direct system.

c. The extensive vacuum system is more susceptible to air leakage for direct type.

d. When starting up, the direct system has a large space to be evacuated.

e. The direct system is generally of mechanical draught type and the cooling elements have to be located close to the turbine, usually installed on the roof of the turbine house, in order to limit the pressure drop in the exhaust piping. Such a requirement also limits the application of direct systems to a maximum rejecting load of 200 MW power plants currently.

Owing to the reasons that we listed above, our discussion of designing a dry cooling tower for a 1000 MW power plant will be based on indirect systems.

<u>Heat transfer in dry cooling towers</u>. The heat transfer in a dry cooling tower is by conduction and convection of heat to air only. However, the low specific heat of air makes it inferior to water as a cooling medium so much that an air mass flow four times that of water is needed for the same cooling load. Hence, the performances of indirect system cooling elements, which reject heat from water through tube

wall to the air, are governed by the air-side heat transfer coefficient which is much lower than the water-side heat transfer coefficient. For the purposes of increasing air-side heat transfer amount, several extended surface finned tubes of various types have been developed.

Types of Finned Tubes

There are four types of finned tubes now prevailing.

1. The Heller-Forgo tube. Both the tube and fin are made of aluminum. Special characteristics are a large secondary surface, high performance with low air velocity and low pressure drop [28].

2. Integral finned tubes. Integral finned tubes are made by rolling an endless aluminum band onto a steel or copper tube for the finning.

3. Extended finned tubes. Extended finned tubes have aluminum fins expanded from a tube which are bonded or embedded onto the central tubes.

4. Elliptical finned tubes. Elliptical finned tubes are made of an elliptical central tube and extended square fins provided with turbulators for intensifying the heat transfer. This type of finned tube is an improvement of circular finned tubes with less pressure drop and more heat transfer duty [29].

The arrangement of finned tubes. The arrangements of finned tubes also influence the heat transfer of finned tubes. There are two types of arrangements commonly used [29].

1. In-line arrangement
2. Staggered arrangement

The two types of arrangement, when applied to the same finned tubes, will not affect the heat transfer but the pressure drop increases significantly for in-line arrangements.

Layout of Dry Cooling Tower

There are also two types of layout of dry cooling towers.

1. Vertical element arrangement. The vertical element design has individual cooling elements arranged around the periphery at the bottom of the tower.

2. Horizontal element arrangement. The horizontal element design has the individual cooling elements arranged in a horizontal plane at the bottom of the tower.

The horizontal arrangement has proved to be superior to the vertical arrangement owing to the following two facts [26]:

a. The vertical arrangement causes an adverse negative pressure by the wind at the lee side of the tower which interfers with the up-current through the tower.

b. The air all over the tower base of a horizontal arrangement is heated approximately uniformly and thus avoids the occurrence of cool air core. Further, the substantial horizontal tubes are also self-compensating in avoiding air flow maldistribution [30].

Finned Tubes Dimensional Analysis [31]

The heat transfer and pressure drop of an air-finned tube is dependent on the tubing dimension such as the space of air fins and its thickness, height, and materials. Table A-4 presents the dimensional data of a tube of type 2.

Air-Side Heat Transfer and Pressure Drop

The experimental data of air-side heat transfer and pressure drop of tubes (Table A-4) are shown on Figure A-5 and Figure A-6. These figures employ several parameters.

Air-Side Heat Transfer

1. Equivalent diameter $D_e$.

$$D_e = \frac{2(\text{total outside surface})}{\pi(\text{projected perimeter})} \qquad (\text{IV-36})$$

2. Reynolds number Re.

$$Re = \frac{D_e\ G_m}{\mu} \qquad (\text{IV-37})$$

3. Heat transfer factor j.

$$j = \frac{h}{CG_m}\left(\frac{C\mu}{K}\right)^{2/3} \qquad (\text{IV-38})$$

where $G_m$ is the mass velocity through the minimum cross-sectional flow area in the section. The fin efficiency is shown on Figure A-7.

Air-Side Pressure Drop

1. Volumetric diameter.

$$D_v = \frac{4(\text{net free volume})}{\text{total surface}} \qquad \text{(IV-39)}$$

2. $\underline{Re_v}$.

$$Re_v = \frac{D_v\ G_m}{\mu} \qquad \text{(IV-40)}$$

3. Fin-side friction factor f.

$$f = \frac{2\ \Delta P\ g_c \rho}{4\ G_m^2\ N} \qquad \text{(IV-41)}$$

where N is the number of rows.

Tube-Side Heat Transfer

The heat transfer of a tube when fluid flowing through it can be divided into three regions [32]:

1. Laminar flow region
2. Transition flow region
3. Turbulent flow region

The corresponding heat transfer equations are shown below:

1. Laminar flow, Reynolds number < 2100.

$$\left(\frac{h_i\ D}{K}\right) = 1.86\ \left(\left(\frac{DG_w}{u}\right)\left(\frac{C\mu}{K}\right)\left(\frac{D}{L}\right)\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad \text{(IV-46)}$$

2. Transition flow, Reynolds number 2100 < Re < 1000.

$$\left(\frac{h_i\ D}{K}\right) = 0.116\left(\left(\left(\frac{DGw}{u}\right)-125\right)\left(\frac{C\mu}{K}\right)^{1/3}(\mu/\mu_w)^{0.14}\left(1+\left(\frac{D}{L}\right)^{2/3}\right)\right) \qquad \text{(IV-47)}$$

3. Turbulent flow Reynolds number > 10000

$$\left(\frac{h_i D}{K}\right) = 0.023\left(\frac{DG_w}{\mu}\right)^{0.8}\left(\frac{C\mu}{K}\right)^{1/3}\left(\frac{\mu}{\mu w}\right)^{0.14} \qquad \text{(IV-48)}$$

These three equations are plotted on Figure A-8 by $\frac{DG_w}{\mu}$ (vs) $j_h = \left(\frac{h_i D}{K}\right)\left(\frac{C\mu}{K}\right)^{1/3}\left(\frac{\mu}{\mu w}\right)^{-0.14}$

Total thermal resistance of finned tubes. It is important that all the thermal resistances of the air finned tube be properly corrected and added.

1. Tube-side heat transfer coefficient

$$R_1 = \frac{1}{h_i}\left(\frac{OD_r}{ID_1}\right)$$

2. Tube-side fouling factor

$$R_2 = R_{di}\left(\frac{OD_r}{ID_L}\right)$$

3. Liner resistance

$$R_3 = \frac{L_1}{K_L}\left(\frac{OD_r}{ID_{L\ mean}}\right)$$

4. Bond resistance

$$R_4 = R_b\left(\frac{OD_r}{ID_r}\right)$$

5. Root tube resistance

$$R_5 = \frac{L_r}{K_r}\left(\frac{OD_r}{ID_r}\right)$$

$h_i$ tube-side heat transfer coefficient

$ID_L$ inner diameter of liner

$ID_L$ mean of ID and OD of liner

$ID_r$ inner diameter of root tube

$ID_r$ mean mean of ID and OD of root tube

$K_L$ thermal conductivity of liner

$K_r$ thermal conductivity of root tube

$L_L$ liner thickness

$L_r$ root tube thickness

$R_{di}$ tube-side fouling factor

$R_b$ bond resistance

The total thermal resistance $R_t$

$$R_t = R_1 + R_2 + R_3 + R_4 + R_5 \qquad \text{(IV-49)}$$

The air side fouling factor is usually neglected because it is small compared with the air heat transfer resistance.

Overall heat transfer coefficient. The overall heat transfer coefficient U can be calculated since the air side heat transfer coefficient and the thermal resistances of the tube are known

$$U = \frac{1}{(\frac{1}{h}) + R_t} \qquad \text{(IV-50)}$$

True temperature difference in cross flow arrangement. When air passes the finned tubes, the flow pattern is one of cross flow. Cross flow dictates a temperature difference distribution different from that of counter flow or parallel flow of which the heat transfer is governed by

$$q = U\ S\ (\mathrm{LMTD}) \qquad \text{(IV-51)}$$

where LMTD is the log mean temperature difference. This is due to that, for a cross flow pattern, the temperature difference between the tube and air varies from row to row and section to section.

A method has been suggested to relate the true temperature difference of cross flow with the LMTD of counter flow.

$$T_t = F(t)\ \mathrm{LMTD} \qquad \text{(IV-52)}$$

F(t) is plotted against two parameters B and e where

$$B = \frac{T_1-T_2}{t_2-t_1} \qquad \text{(IV-53)}$$

$$e = \frac{t_2-t_1}{T_1-t_1} \qquad \text{(IV-54)}$$

The results are plotted on Figure A-9. The total heat transferred can then be written as

$$\begin{aligned}\dot{Q} &= U\ S\Delta T_t = U\ S\ F(t)\ \mathrm{LMTD} \\ &= \dot{m}_{air}\ c_{p\ air}\Delta T_{air} \qquad \text{(IV-55)} \\ &= \dot{m}_{water}\ c_{p\ water}\Delta T_{water}\end{aligned}$$

Number of rows of air-finned tubes. When air flows across the finned tubes, it is found that the pressure drop and heat transfer coefficient vary considerably at the first few rows. Since air is compressible and water essentially is not, only a small pressure drop can be expanded for air circulating across the finned tubes lest the cost of the air compression work become prohibitive. In most applications, only 3 or 4 rows are used for each element since more rows of finned tubes will cause a poor performance in heat transfer and an increase in pressure drop.

Height of tower. The height of dry cooling tower can be calculated by

$$H \times \Delta\rho = \frac{v_{av} G'^2_m}{2\ g} + \Delta P = \frac{G'^2_m}{2\ g\rho_{av}} + \Delta P \qquad \text{(IV-56)}$$

where $G'_m$ is the average air mass flow rate of total area including finned tube projected area and spacing area.

Air density difference $\Delta\rho$. The density of air with 60% relative humidity at different temperatures is tabulated on Table A-5. The air temperature difference must be carefully evaluated.

The above analysis of designing a dry type cooling tower will be applied to design a dry tower for a 1000 MW power plant.

The power plant employs the same power cycle except the following design conditions:

| | |
|---|---|
| Dry-bulb temperature | 95°F |
| Relative humidity | 60% |
| Approach | 25°F |
| Range | 20°F |
| Terminal temperature difference | 5°F |

The steam exhaust temperature is thus 145°F.

Power cycle analysis:

| | | | |
|---|---|---|---|
| $h_1$ | = 87.92 BTU/lbm | $h_4$ | = 918.1 BTU/lbm |
| $w_p$ | = 20 BTU/lbm | $w_t$ | = 488.66 BTU/lbm |
| $h_2$ | = 107.92 BTU/lbm | $w_{net}$ | = 468.66 BTU/lbm |
| $h_3$ | = 1406.8 BTU/lbm | $q_{rej}$ | = 830.18 BTU/lbm |

The steam flow rate $\dot{m}_{steam} = \frac{\dot{P}}{W_{net}} = 7.282 \times 10^6$ lbm/hr.

The waste heat rejection rate $\dot{Q} = \dot{m}_{steam}\, q_{rej} = 6.044 \times 10^9$ BTU/hr. The required cooling water flow rate $\dot{m}_{water} = \frac{Q_{rej}}{C\Delta T_o} = 3.022 \times 10^8$ lbm/hr $3.022 \times 10^8$ lbm/hr of water is to be cooled from 140°F to 120°F by air on a summer day $T_d = 95°F$. Properties of water at $T_{mean} = 130°F$ are

$$C = 1 \text{ BTU/lbm-hr}$$

$$u = 1.26 \text{ lbm/ft-hr}$$

$$K = 0.373 \text{ BTU/hr-ft-F}$$

The dimensional data to be used are tabulated in Table A-4 with Bond resistance 0.00067 ft-hr-F/BTU and fouling factor 0.002. The arrangement of finned tubes will be in a staggered arrangement with 2.125 inch equal triangular pitch. The number of finned tube rows is 4.

Tube-side heat transfer coefficient. The flow area per tube is 0.594 in$^2$ or 0.00412 ft$^2$ and the water mass flow rate is taken to be $3.48 \times 10^5$ lbm/ft$^2$-hr. The tube-side Reynolds number is thus $\frac{DG_w}{\mu} = 20000$. From Figure A-8, it is found that $j_h = 70$. The tube side heat transfer coefficient

$$h_i = \frac{K\ j_h}{D}\left(\frac{C\mu}{K}\right)^{1/3}(\mu/\mu_w)^{0.14} = 444\ \text{BTU/ft}^2\text{-hr-F}$$

where $\mu/\mu_w$ is taken to be 1.

Total resistance of finned tubes. Substituting $h_i$, $R_{di}$, $R_b$ and the data in Table A-4 into Equation IV-49 gives

$$R_t = 0.00634$$

Air side heat transfer coefficient. The properties of air at 100°F and 1 atm

$$\rho = 0.070\ \text{lbm/ft}^3$$

$$\mu = 0.045\ \text{lbm/ft-hr}$$

$$C_{air} = 0.24\ \text{BTU/lbm-F}$$

Thus,

$$\left(\frac{C_{air} \times \mu}{K}\right)^{2/3} = 0.795$$

Equivalent diameter $D_e = \dfrac{2\text{(total outside surface)}}{\pi\text{(projected perimeter)}}$

From Table A-4, the total surface per ft of tube is 3.59 $ft^2$. The projected perimeter per ft of tube 2 + 9 x 12 x (2-1.08) x 2/12 = 16.56 ft/ft.

Thus, $D_e = 0.14$. Net free volume per ft of tube = $12(\frac{1}{2}(2.125)^2\cos 30^\circ - \frac{1}{2}(\frac{\pi}{4})(1.08)^2 - \frac{1}{2}(9)\frac{\pi}{4}(((2.00)^2-(1.08)^2)$ $(0.019)))) = 15.62$ in$^3$/ft.

The volumetric diameter $D_v = \dfrac{4(\text{net free volume})}{(\text{total surface})}$

The total surface of free volume contacting with the tube = $\frac{1}{2}(3.59) = 1.795$ ft$^2$/ft. Thus,

$$D_v = \frac{4 \times 15.62 \times 1/1728}{1.795} = 0.0202 \text{ ft}$$

The flow area per ft of fin tube:

$$\begin{aligned} S_{fin} &= \text{projected fin area per 1 ft of tube length} \\ &= (\text{fin thickness})(OD_{fin} - OD_{root\ tube})(\text{no. of fin per ft}) \\ &= 0.019 \times (2-1.08) \times 9 \times 12 \times 1/144 \\ &= 0.0131 \text{ ft}^2/\text{ft} \\ S_{tube} &= \text{projected area of root tube per ft} \\ &= OD_{root\ tube} \times \text{tube length} \\ &= 1.08 \times 1/12 \times 1 = 0.09 \text{ ft}^2/\text{ft of tube} \end{aligned}$$

The total projected area per ft of finned tube:

$$S_{ft} = S_f + S_t = 0.0131 + 0.09 = 0.1031 \text{ ft}^2/\text{ft of tube length}$$

The number finned tube per ft of pitch length:

$$N_t = \frac{1}{\text{pitch}} = \frac{1 \times 12}{2.125} = 5.65 \text{ tubes}$$

Total projected area of finned tube per $ft^2$ = 0.1031 x 5.65
= 0.583 $ft^2$

Total air flow area per $ft^2$ = 1 - 0.583 = 0.417 $ft^2$

Assume that the air flow rate $G_m$ = 4500 lbm/$ft^2$-hr. From Figure A-5, j = 0.0071,

$$h = \frac{j\ C\ G_m}{(C\mu/K)^{2/3}} = 9.78$$

The fin efficiency is estimated to be 0.9.

$$h' = h \times 0.9 = 8.8$$

The overall heat transfer coefficient:

$$U = \frac{1}{\frac{1}{h'} + R_t} = 8.31$$

Contact area between finned tube and air per $ft^2$ of projected area = 3.59 x 5.65 x 4 = 81.12 $ft^2/ft^2$ of projected area. The total amount of heat transferred will be

$$q = U\ A_t (81.12) \Delta T_t$$
$$= G_m\ C_{air} \Delta T_{air}\ A_t (0.417)$$

We get

$$\frac{\Delta T_t}{\Delta T_{air}} = 1.496$$

By trial and error,

| | |
|---|---|
| 140 | Y |
| 120 | 95 |

Y is found to be 124°F. This is checked as follows:

| | | |
|---|---|---|
| 140 | 124 | 16 |
| 120 | 95 | 25 |

$$\text{LMTD} = \frac{25-16}{\ln\frac{25}{16}} = 20.2$$

$$B = \frac{20}{29} = 0.69 \qquad e = \frac{29}{45} = 0.645$$

From Figure A-9d, F(t) is found to be 0.96, thus, $\Delta T_t$ = LMTD x F(t) = 20.2 x 0.96 = 19.38°F. The ratio $\Delta T_{air}/\Delta T_t$ = 29/19.38 = 1.496. Substituting $\Delta T_t$ = 19.38°F back into heat transfer equation gives

$$6.044 \times 10^9 = 8.31 \times 81.12 \times 19.38 \times A_t$$

At the beginning, we let the water mass flow rate to be $3.48 \times 10^5$ lbm/ft$^2$-hr and we know that the required water flow rate is $3.022 \times 10^8$ lbm/hr. The required cross-sectional area of tube $\frac{3.022 \times 10^8}{3.48 \times 10^5} = 8.7 \times 10^2 ft^2$. The cross-sectional area per tube is 0.00412 ft$^2$ and the number of tubes per ft is 5.65. Thus, the cross-sectional area per ft is $0.00412 \times 5.65 = 0.02325$ ft$^2$/ft. Two rows of the four rows per section are for cooling water entry. Thus, total cross-sectional area of root tube per foot = 0.02325 x 2 = 0.0465 ft$^2$/ft. Required cross-sectional area length $L = \frac{8.7 \times 10^2}{0.0465} = 1.87 \times 10^4$ ft. Required tube length $l = \frac{A_t}{L_{cross\text{-}section}} = \frac{4.63 \times 10^5}{1.87 \times 10^4} = 24.8$ ft. Let the dimension of each cooling unit be 25 ft in length and 17.7 ft in cross-section length, i.e., 100 tubes per row. The total units required are $\frac{1.87 \times 10^4}{17.7} = 1100$ units.

<u>The height of cooling tower</u>. $\frac{D_v G_m}{\mu} = 2020$ and from Figure A-6 we find f = 0.33. The pressure drop across the cooling element (from Equation IV-41)

$$\Delta P = \frac{2 \times 0.33 \times (4.5)^2 \times 10^6 \times 4}{4.18 \times 10^8 \times 0.07} = 1.83 \text{ psia}$$

From Table A-5

$$T_d = 95°F \qquad \rho = 0.07060 \text{ lbm/ft}^3$$

$$T_d = 124°F \qquad \rho = 0.06594 \text{ lbm/ft}^3$$

The density difference is = 0.0046 $lbm/ft^3$ and the average density is 0.06827 $lbm/ft^3$. The average air mass flow rate $G'_m = G_m \times 0.417 = 2623$ $lbm/ft^2$-hr. Substituting $_{av}$, $G'_m$ and P into Equation IV-56 yields

$$H = 27.23 + 398 = 425.23 \text{ ft}$$

Let the required number of tower be 8. Base area for each tower $\frac{A_t}{8} = 5.7875 \times 10^4$ $ft^2$.

| Summary | Wet Type | Dry Type |
|---|---|---|
| Power output | 1000 MW | 1000 MW |
| Steam exhaust temperature | 115°F | 145°F |
| Heat rejection | $5.4267 \times 10^9 \frac{BTU}{hr}$ | $6.044 \times 10^9 \frac{BTU}{hr}$ |
| TTD | 5°F | 5°F |
| Range | 20°F | 25°F |
| Approach | 20°F | 20°F |
| Cooling Water inlet temperature | 110°F | 140°F |
| Cooling water outlet temperature | 90°F | 120°F |
| Dry bulb temperature | 80°F | 95°F |
| Wet bulb temperature | 70°F | 80°F |
| Number of towers | 8 | 8 |
| Tower base area | $2.910 \times 10^4 ft^2$ | $5.7875 \times 10^4 ft^2$ |
| Tower height | 289 ft | 425 ft |

# CHAPTER V

## A COMPARISON OF WET AND DRY TYPE COOLING TOWERS

In this chapter, the performances of wet and dry type cooling towers, designed in Chapter IV for a 1000 MW power plant, will be investigated by applying the principle of constant effectiveness theory derived in Chapter III with changing dry bulb temperature while the following conditions derived in the previous chapter are held constant:

| | Wet Type | Dry Type |
|---|---|---|
| Power output | 1000 MW | 1000 MW |
| Turbine efficiency | 0.9 | 0.9 |
| Terminal Temperature difference | 5°F | 5°F |
| Humidity | 60% | 60% |
| Cooling water flow rate | $2.713 \times 10^8$ lbm/hr | $3.022 \times 10^8$ lbm/hr |
| Number of towers | 8 | 8 |
| Tower base area | $2.910 \times 10^4$ | $5.7875 \times 10^4 ft^2$ |
| Height of tower | 289 ft | 425 ft |

### Part A: Wet Type Cooling Tower

The performance of a wet type cooling tower can be investigated by assuming that the Chilton coefficient is constant. This assumption is based on experience, not theoretical proof. The procedures of calculating the recooled temperature are illustrated in Table 2. The same procedures are repeated for different dry bulb temperatures and the results are tabulated in Table 3.

Part B: Dry Type Cooling Tower

The procedures of calculating the recooled temperatures of a dry type cooling tower are much more complicated than that of wet type cooling tower. Trial and error is applied to solve some unknown factors. Again, the procedures are shown here and repeated for different dry bulb temperatures. The results are tabulated in Table 4.

Let the dry bulb temperature be 85°F and relative humidity 60%. From Table A-5, we have the air-vapor mixture density $\rho = 0.07215$ lbm/ft$^3$. Assuming that the outlet temperature of air is 113.5°F, $\rho = 0.06780$ lbm/ft$^3$, the density difference is thus $\Delta\rho = 0.00435$ lbm/ft$^3$. Substituting $\Delta\rho$, H, and Equation IV-41 into IV-56 gives

$$425.23 \times 0.00435 = \frac{2 x f x G_m^2 x 4}{4.18 x 10^8 x 0.070} + \frac{0.417^2 x G_m^2}{2x32.2x3600^2 x0.070}$$

By using Figure A-6 and trial and error we get

$$f = 0.334 \qquad G_m = 4.39 \times 10^3 \text{ lbm/ft}^2\text{-hr}$$

The total heat rejected will be $\dot{Q} = G_m x C_p x T_{air} x A_t x 0.417 = 5.8 \times 10^9$, since $G_m = 4.39 \times 10^3$ lbm/ft$^2$-hr, $R_{e\ air} = 12100$.

Table 2. Procedures of Calculating the Recooled Temperature of Wet Type Cooling Towers

| | Design Conditions | Calculating Conditions | Step |
|---|---|---|---|
| Assumed steam exhaust temp. | 115°F | 110°F | 1 |
| Heat rejection rate per tower | $6.783 \times 10^8$ BTU/hr | $6.70 \times 10^8$ BTU/hr | 2 |
| Water flow rate per tower | $3.391 \times 10^7$ lbm/hr | $3.391 \times 10^7$ lbm/hr | |
| Base area | $2.9106 \times 10^4$ $ft^2$ | $2.9106 \times 10^4$ $ft^2$ | |
| Water load | 1165 lbm/$ft^2$-hr | 1165 lbm/$ft^2$-hr | |
| $AT_o$ | 20°F | 19.85°F | 3 |
| $T_d$ | 80°F | 70°F | |
| $T_w$ | 70°F | 60°F | |
| $T_d-T_w$ | 10°F | 10°F | |
| $F(T_d,T_w)$ | 5700 | 7100 | 4 |
| Height of tower | 288.75 ft | 288.75 ft | |
| $H \times L \times \Delta T_o \times F(T_d,T_w)$ | $38.34955 \times 10^9$ | $47.41 \times 10^9$ | 5 |
| $L/GN^{1/3}$ | 0.3454 | 0.32188 | 6 |
| $\alpha GN^{1/3}/L=C$ | 5.499 | 5.499 | 7 |
| $\alpha$ | 1.9 | 1.770 | 8 |
| $T_o$ | 90°F | 84°F | 9 |
| $T_o+\Delta T_o$ | 110°F | 103.5°F | 10 |
| Estimated steam exhaust temp. | 115°F | 108.5°F | 11 |

From Figure A-5, $j = 0.0073$ and then substituting j into Equation IV-38 yields $h = 9.75$. Assume fin efficiency = 0.9, $h' = h \times 0.9 = 8.78$. Substituting h' and $R_t$ into Equation IV-50 gives U = 8.31 BTU/ft-F-hr. Substituting $\dot{Q} = 5.8 \times 10^9$ BTU/hr, $U = 8.31$, $S = 4.63 \times 10^5 \times 81.12\ ft^2$ into Equation IV-55 yields

$$\Delta T_t = 18.6°F$$

Assuming $F(t) = 0.96$, hence LMTD = 19.4°F. Substituting $\dot{Q}$, $\dot{m}_{water}$ into Equation IV-55 results in $\Delta T_{water} = 19.2°F$. We have the following conditions to estimate the recooled temperature of cooling water:

| Cooling Water | Cooling Air | |
|---|---|---|
| $Y_1$ | 113.5 | $Y_1 - Y_2 = 19.2°F$ |
| $Y_2$ | 85 | LMTD = 19.4°F |

By trial and error, we get

$$Y_1 = 129.2°F \qquad Y_2 = 109°F$$

Our assumption of $F(t) = 0.96$ has to be checked.

$$\text{LMTD} = 19.4°F \qquad B = \frac{19.2}{28.5} = 0.674 \qquad e = \frac{28.5}{44.2} = 0.645$$

From Figure A-9d, $F(t) = 0.96$. The result is consistent with our previous assumption.

At the very beginning, we also assumed that the outlet temperature of air was 113.5°F and thus has to be checked. The outlet water temperature is 129.2°F. The terminal temperature difference is 5°F. The estimated steam exhaust temperature is 129.2 + 5 = 134.2°F. From Figure IV-2, the rejected waste heat is $5.8 \times 10^9$ BTU/hr. Substituting $\dot{Q} = 5.8 \times 10^9$ BTU/hr, $G = 4.39 \times 10^3$ lbm/ft$^2$-hr, $S = 4.63 \times 10^5 \times 0.417$ ft$^2$ into Equation IV-55 gives $\Delta T_{air} = 28.5$°F. The outlet temperature of air is thus 85 + 28.5 = 113.5°F. Thus, our previous assumption is correct. The same procedures have been repeated and the results are tabulated in Table 4.

The data tabulated on Tables 3 and 4 are plotted on Figure 19. It is clear that the dry type cooling curve has a larger slope than that of wet type cooling curves. Thus, on a cold day the dry type towers seem to be superior to wet type towers.

Very fruitful results can be obtained by using the effectiveness of power plants to evaluate the performances of such two types of cooling towers.

$$\frac{W}{Q_h} = e\left(1 - \frac{T_o}{T_{hav}}\right)$$

Table 3. The Calculated Recooled Temperatures of Wet Type Cooling Towers at Different Dry Bulb Temperatures

| Dry Bulb Temperature | Recooled Water Temperature |
|---|---|
| 90°F | 95°F |
| 80°F | 90°F |
| 70°F | 84°F |
| 60°F | 78°F |
| 50°F | 72°F |
| 40°F | 66°F |

Table 4. The Calculated Recooled Temperatures of Dry Type Cooling Towers at Different Dry Bulb Temperatures

| Dry Bulb Temperature | Recooled Water Temperature |
|---|---|
| 95°F | 120°F |
| 85°F | 109°F |
| 75°F | 99°F |
| 65°F | 89°F |
| 55°F | 78°F |
| 45°F | 66°F |
| 35°F | 56°F |

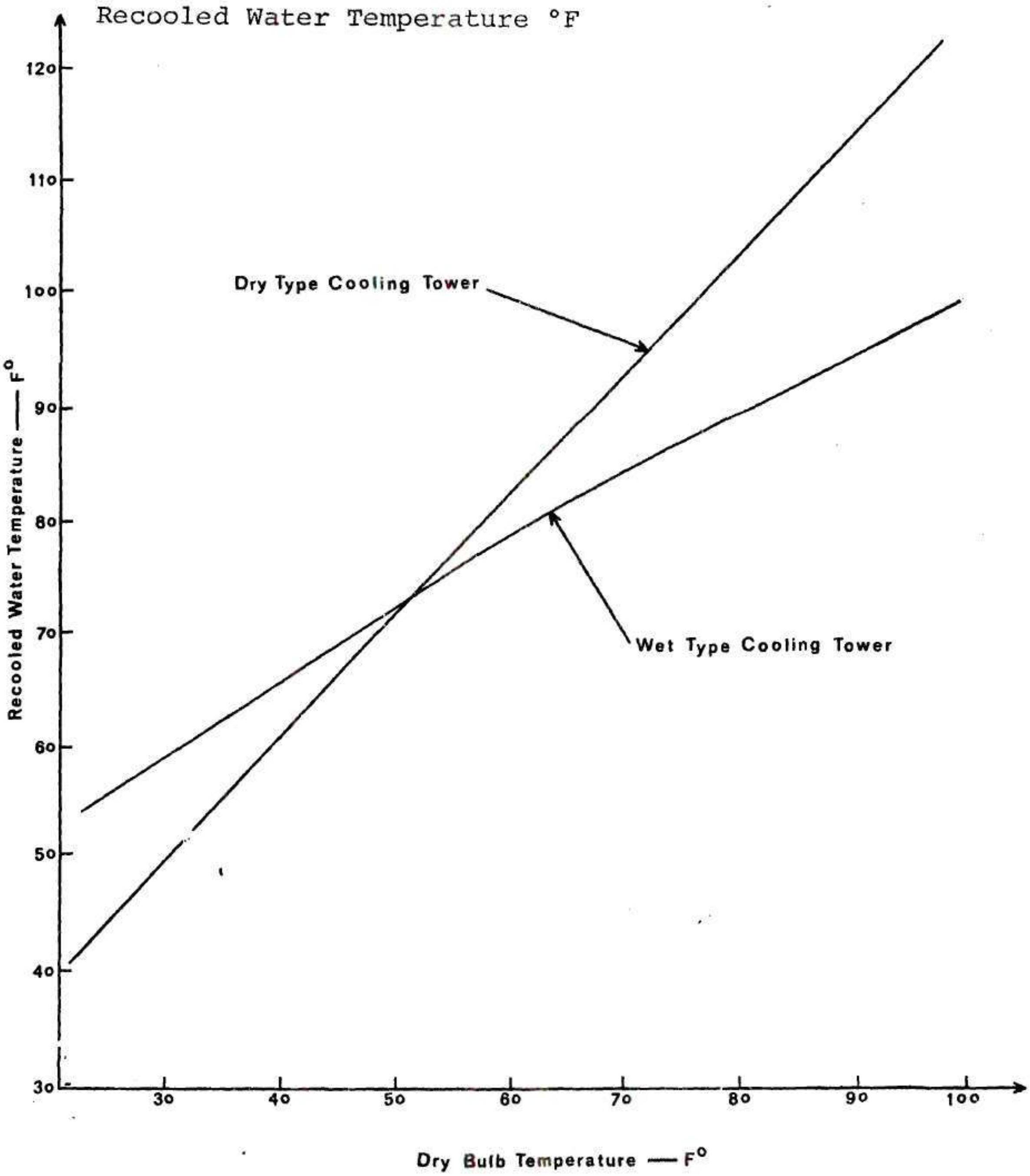



Figure 19. The Recooled Temperatures of Wet Type and Dry Type Cooling Tower at Different Dry Bulb Temperatures (Relative Humidity 60%)

where the effectiveness e can be calculated by using Equation III-37. Tables 5 and 6 show the calculated effectiveness of the 1000 MW power plant using the dry and wet type cooling towers respectively. Also, the efficiency of the power plant is calculated and the results are plotted on Figure 20 against dry bulb temperatures.

It is apparent from Figure 20 that the dry type cooling tower will make the efficiency of a power plant vary more than the wet type tower does. This is an important fact that has to be taken into consideration when designing a power plant.

Table 5. The Effectiveness of a 1000 MW Power Plant Using Wet Type Cooling Towers at Various Dry Bulb Temperatures

| $T_{dry\ bulb}$(°F) | $T_o$(°F) | effectiveness e | $e(1 - \frac{T_o}{T_{hav}}$ |
|---|---|---|---|
| 90 | 95 | 0.82075 | 0.38120 |
| 80 | 90 | 0.82152 | 0.38576 |
| 70 | 84 | 0.82218 | 0.39059 |
| 60 | 78 | 0.82301 | 0.39575 |
| 50 | 72 | 0.82379 | 0.40090 |
| 40 | 66 | 0.82464 | 0.40609 |

Note: $T_{hav}$ = 1037°R; $K_p$ = 0.034

Table 6. The Effectiveness of a 1000 MW Power Plant Using Dry Type Cooling Towers at Various Dry Bulb Temperatures

| $T_{dry\ bulb}$(°F) | $T_o$(°F) | effectiveness e | $e(1 - \frac{T_o}{T_{hav}})$ |
|---|---|---|---|
| 95 | 120 | 0.82075 | 0.36192 |
| 85 | 109 | 0.82079 | 0.37086 |
| 75 | 99 | 0.82264 | 0.37961 |
| 65 | 89 | 0.82345 | 0.38793 |
| 55 | 78 | 0.82667 | 0.39820 |
| 45 | 66 | 0.82792 | 0.40838 |
| 35 | 56 | 0.82945 | 0.41712 |

$$\text{Power Plant Efficiency} = e\left(1 - \frac{T_o}{T_{hav}}\right)$$

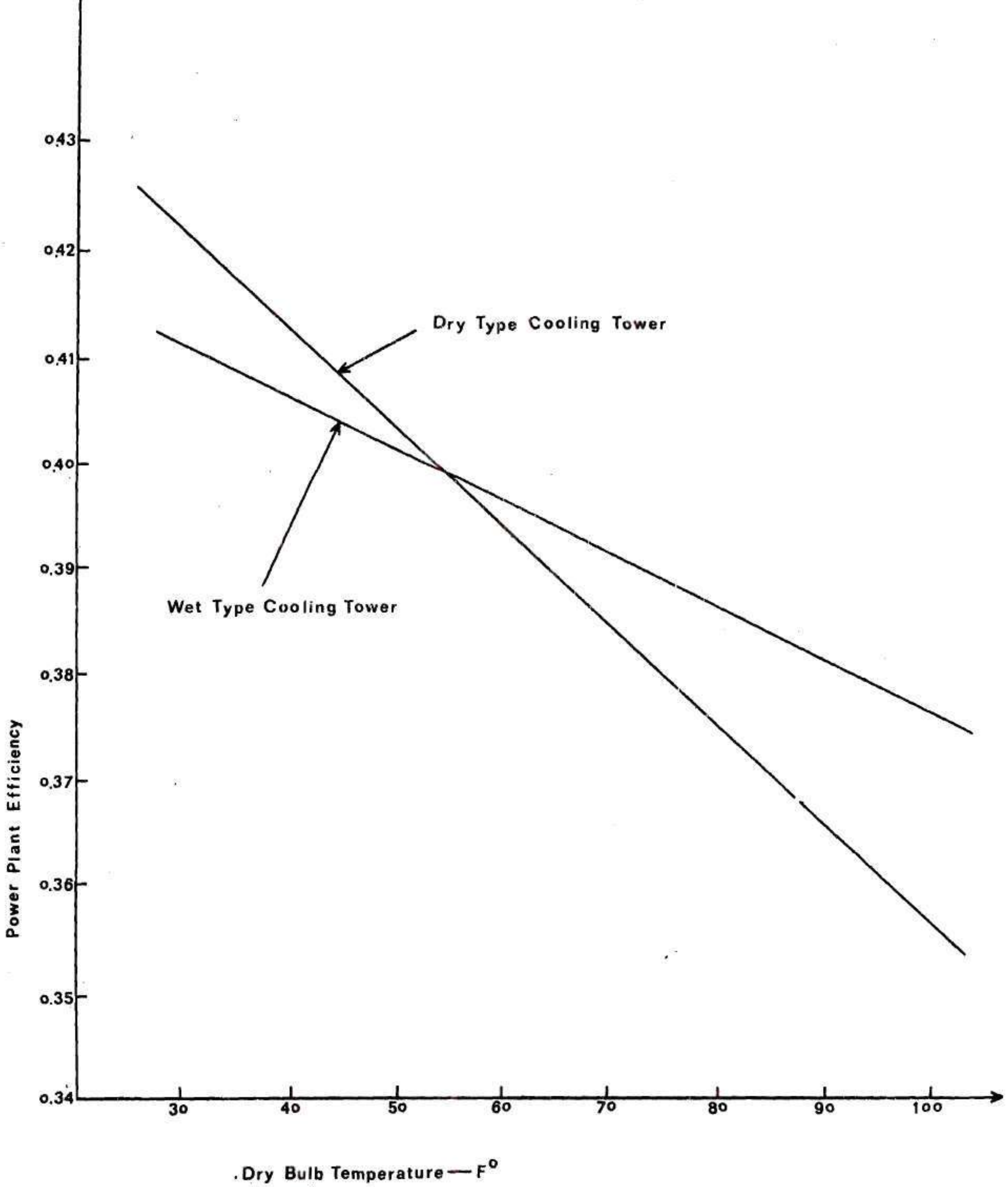



Figure 20. The Power Plant Efficiency Corresponding to Figure 19

# CHAPTER VI

## SUMMARY, CONCLUSIONS, AND SUGGESTIONS

The heat rejection of a power plant is an inheritant byproduct of its thermal cycle. Many heat rejection systems have been devised.

1. Once-through cooling system
2. Cooling lake system
3. Cooling pond system
4. Wet type cooling tower
5. Dry type cooling tower
6. Wet-dry type cooling tower

Owing to the recent wide-spread adoption of federal and state thermal pollution regulations, there is an increasing interest in the application of dry and wet type cooling towers. The different modes of heat transfer being employed by these two types of cooling towers have resulted in different performances for each type. The recooled water temperature is one of the most significant factors to the power plant itself, since the steam exhaust temperature is dependent on that temperature. A lower cooling water temperature means that the steam can be expanded to a lower pressure, hence, more work is converted from enthalpy. In this thesis, a procedure is developed which reflects the comparative effects of cooling water temperature on the power production rate; in other words, which evaluates the cooling tower by the performance of the power plant itself.

Energy availability methods are used to derive the following equations which relate the heat input, work output, cooling water temperature and other power plant factors:

$$w = e\,Q\left(1 - \frac{T_o}{T_{hav}}\right)$$

where

$$e = \frac{\eta_{turbine} - K_p}{1+\left(\frac{T_o - T_1}{T_1}\right)(1-\eta_{turbine}) + \left(\left(\frac{T_{hav}}{T_{hav}-T_1}\right) - \eta_{turbine}\right)\left(\left(\frac{T_1 T_o}{T_{hav} T_1}\right) + \left(\frac{\Delta T_o}{2T_o + T_o}\right)\right) - K_p \ldots}$$

See Equation III-37 on page 38.

It is found that the effectiveness of a power plant remains almost constant when working conditions are limited to some range of the design point of the steam turbine. We have thereby established a principle of constant effectiveness. Such a statement is very helpful when trying to optimize the cooling water temperature. As shown on page 44, the effectiveness equation may be extended to apply to regeneration, reheat and topping and the effectiveness will still remain virtually constant over a wide range of design conditions. Thus, the principle of constant effectiveness may be applied in general. The complete generalization is left for further study.

A power plant of 1000 MW employing a supercritical thermal cycle is considered using both wet and dry type cooling towers. The recooled temperatures of these two types of cooling towers are investigated respectively and tabulated in Tables 3 and 4. Finally, the above effectiveness equation is used to calculate the effectivenesses and efficiencies of the power plant. The results are plotted on Figure 20 against dry bulb temperatures. A complete comparison for the purpose of design requires a consideration of capital cost as discussed at the end of Chapter III. As shown there, the principle of constant effectiveness which resulted from energy availability methods greatly facilitate the determination of optimum parameters. This principle may also obviously be used for comparing cooling towers against the other heat rejection systems listed on the preceding pages. The results could then be weighed against the relative advantages and disadvantages which are listed on pages 11-14. These comparisons are left for further study.

## Suggestion for Further Study

1. We have shown via energy availability methods that the principle of constant effectiveness may be applied to the simple Rankine cycle shown on page 41. (The deviation from this principle being shown in Figure 16 on page 41.) We have also indicated that the principle may be applied to

more general Rankine cycles including regenerative, reheat, and topping cycles, but the deviation from the principle has not been worked out in detail. For accurate design work Equation III-37 should be extended to include the effect of regeneration, reheat, and topping, resulting in a more general correction curve of the type shown on Figure 16.

2. The use of the principle of constant effectiveness in the selection of such cooling tower parameters as geometry (height, shape, etc.), packing, flow-rate, etc. For this purpose, our studies indicate that it will be better to take the equilibrium temperature $T_o$ to be exit temperature of the surface condenser, rather than the entrance temperature as was done in this thesis. This will result in a still more constant value of the effectiveness.

3. Application of the principle of constant effectiveness in comparing other heat rejection systems (once-through cooling system, cooling lake cooling system, cooling pond cooling system, wet-dry cooling system).

4. Application of these methods of energy availability to derive the principle of constant effectiveness for other thermal cycles (Brayton, Stirling, and Ericson) for comparisons and design of the corresponding systems (gas turbine, Stirling engine, etc.).

5. Extension of these comparisons to more general, dimensionless comparison, making use of the principles of modeling and scaling. For example, division of certain of

the dry tower equations by corresponding wet tower equations will yield relevant dimensionless parameters.

# APPENDIX I

## DISCUSSION OF ENERGY AVAILABILITY

The 1st law and 2nd law of thermodynamics determine the relationship between heat and work. The 1st law tells us how much change in stored energy results when the system goes from one state to another. The 2nd law, however, marks the distinction between work and heat by stating that heat is limited to convert into work by any continuous operating device while work can always be converted into heat completely and continuously. These facts go to prove that there is a maximum amount of work which can be obtained when a system changes from one state to another state while exchanging heat only with its surroundings.

The maximum work delivered by a steady flow system is the sum of that delivered by the system and that produced by a reversible heat engine as heat exchanging between the system and the surroundings. Figure I-1 illustrates the arrangement.

$$\delta W_{max} = W_{shaft} + W_{engine}$$

$$\delta W_{shaft} = dm(h_1 + \frac{V_1}{2g_c} + \frac{g}{g_c} z_1) - dm(h_2 + \frac{V_2^2}{2g_c} + \frac{g}{g_c} z_2) - Q$$

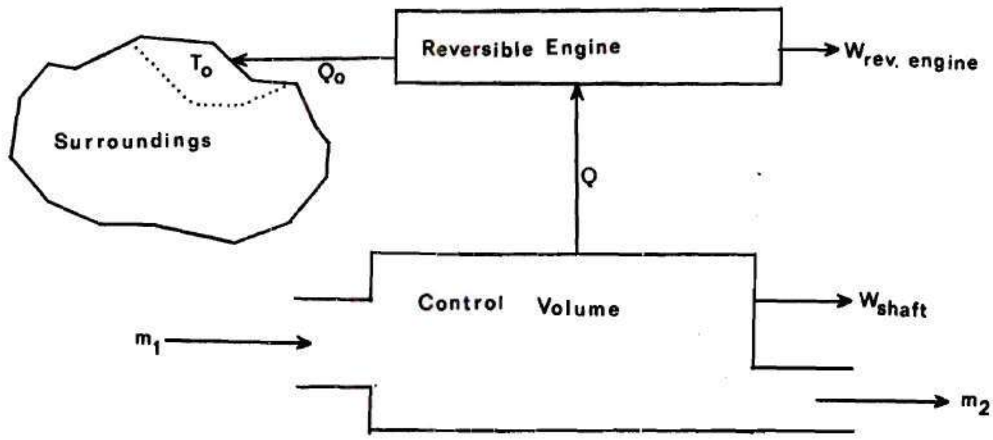



Figure I-1. Composite Control-Volume-Surroundings Producing Maximum Work

$$\delta W_{engine} = Q-Q_o = Q-Q(\frac{T_o}{T}) = Q-T_o(\frac{Q}{T}) = Q-T_oS$$

Substituting $W_{shaft}$ and $W_{engine}$ into $W_{max}$ gives

$$\delta W_{max} = dm(h_1 + \frac{V_1^2}{2g_c} + \frac{gz_1}{g_c} - T_os_1)-dm(h_2 + \frac{V_2^2}{2g_c} + \frac{g}{g_c} z_2-T_os_2)$$

For simplicity, we neglect the kinetic and potential energy.

$$\delta W_{max} = dm((h_1-T_os_1)-(h_2-T_os_2))$$

Let $b = h-T_os$, thus,

$$\delta W_{max} = dm(b_1-b_2)$$

Energy Availability Loss due to Heat Transfer

Figure I-2 illustrates the heat exchange of a boiler. From the 1st law of thermodynamics, we have the following heat balance:

$$m_{product} \times \Delta h_{product} = m_{steam} \times \Delta h_{steam}$$

hence

$$\frac{m_{product}}{m_{steam}} = \frac{\Delta h_{steam}}{\Delta h_{product}}$$

The availability loss is thus

$$\Delta B = m_{product}((h_a-h_b)-T_o(s_a-s_b))-m_{steam}((h_2-h_1)-T_o(s_2-s_1)) =$$

$$-(m_{product} \times T_o\Delta s_{product}-m_{steam} \times T_o\Delta s_{steam})$$

$$= -T_o(m_{product} \times \Delta s_{product}-m_{steam} \times \Delta s_{steam})$$

$$= -T_o m_{steam}((\frac{m_{product}}{m_{steam}}) \times \Delta s_{product}-\Delta s_{steam}))$$

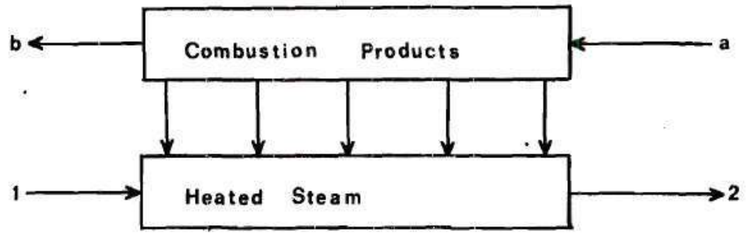



Figure I-2. Heat Exchange of a Boiler

## The Energy Availability Loss Due to Friction

For an open system, the entropy change can be expressed as

$$dS_o = \frac{Q}{T} + dm(s_1 - s_2)$$

Introducing the concept of lost work by friction, $Q = Q_o + LW$

$$dS_o = \frac{dQ_o + dLW}{T} + dm(s_1 - s_2)$$

For a steady flow system, $dS_o = 0$.

$$dLW = T\ dm(s_2 - s_1) - dQ_o$$

Since the lost work is in a form of work not heat before the process, the decrease in lost work is equal to the lost energy availability.

$$dB = dLW = T\ dm(s_2 - s_1) - dQ_o$$

The lost work can be applied to any frictional flow such as steam flowing through a turbine or a pipe.

# APPENDIX II

## DERIVATION OF APPROXIMATE AIR FLOW EQUATION

From Equation IV-23

$$-H\ \Delta\rho = \frac{N\ v\ G^2}{2g} \tag{1}$$

Since G is expressed in lbm/hr but g is in ft/sec$^2$

$$-H\ \Delta\rho = \frac{N\ v\ G^2}{2 \times 32.2 \times 3600^2}$$

Taking the value of v as that of saturated air at 60°F, $v = 13.53$ ft$^3$/lbm. The last equation becomes

$$-H\ \Delta\rho = 1.639 \times 10^{-8}\ N\ G^2 \tag{2}$$

The air density difference must be carefully evaluated, since it changes very little. A method for evaluating $\Delta\rho$ is presented here. Consider 1 lbm of air at 60°F and 1000 millibars total pressure (29.53 inch $H_g$), the volume is 13.29 ft$^3$, containing z lbm of vapor. The total volume is therefore

$$13.29 \times \left(1 + \frac{\lambda}{0.622}\right)\ \text{ft}^3$$

where 0.622 is the ratio of specific volume of vapor to air at the same temperature $^{M}H_2O/^{M}air = 0.622$. Assuming that air-vapor is an ideal gas, the specific volume of the mixture at any temperature T is

$$v = 13.29 \times (1 + \frac{\lambda}{0.622})(\frac{460+T}{520}) \quad (3)$$

Let $\theta = (T-60/520)$.

$$v = 13.29 \times (1 + \frac{\lambda}{0.622}) \times (1+\theta) \quad (4)$$

The density of the mixture is

$$\rho = \frac{1}{13.29} (1 + \frac{\lambda}{0.622})^{-1} (1+\theta)^{-1} \quad (5)$$

Expanding by the binomial theorem, multiplying and collecting terms, and neglecting higher order than the first in z and $\theta$.

$$\rho = \frac{1}{13.29} (1-\theta-0.608\ \lambda) \quad (6)$$

The total heat of mixture per lbm of air is

$$h = 0.241(T-32) + \lambda h_s \quad (7)$$

where $h_s$ is the total heat of 1 lbm of water vapor at temperature T. From the steam tables it is found that the following formula gives an excellent approximation to $h_s$ between 32°F and 100°F.

$$h_s = 1061.2 + 0.44\ T \tag{8}$$

Substituting Equation 8 into 7 gives

$$h = 0.241(T-32) + \lambda(1061.2 + 0.44\ T) \tag{9}$$

By substituting for T for θ and neglecting higher order than the first in q and θ, it is found that

$$h = 6.75 + 125.3\theta + 1087.6\lambda \tag{10}$$

or

$$\Delta h = 125.3\Delta\theta + 1087.6\Delta\lambda \tag{11}$$

Hence

$$\Delta\lambda = \frac{\Delta h - 125.3\Delta\theta}{1087.6} \tag{12}$$

From Equation 6

$$\Delta\theta = \frac{\Delta T}{520} \qquad \Delta\rho = \frac{1}{13.29}\ (\Delta\theta + 0.608\Delta\lambda)$$

Substituting the above two equations into Equation 12

$$-\Delta\rho = 13.465 \times 10^{-5}\Delta h(\frac{\Delta T}{\Delta h} + 0.3124) \qquad (13)$$

Substituting Equation 13 into Equation 2 gives

$$1.639 \times 10^{-8}\ \frac{N}{H}\ G^2 = 1.3465 \times 10^{-4}\Delta h(\frac{\Delta T}{\Delta h} + 0.3124) \qquad (14)$$

Since $G\Delta h = L\Delta T_o$

$$1.639 \times 10^{-8}\ \frac{N}{H}\ G^2 = 1.3465 \times 10^{-4}\ \frac{L}{G}\ \Delta T_o(\frac{\Delta T}{\Delta h} + 0.3124)$$

or

$$G^3N = 8210\ H\ L\Delta T_o(\frac{\Delta T}{\Delta h}) + 0.3124) \qquad (15)$$

This is the equation that we need in Chapter IV.

Table A-1. Enthalpy of Moist Saturated Air British Units, 32F Datum

Total Pressure: 1 atm.abs (Btu/lb of dry air)

| Temp (°F) | 0.0 | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
|---|---|---|---|---|---|---|---|---|---|---|
| 32 | 4.1 | 4.2 | 4.2 | 4.2 | 4.3 | 4.3 | 4.3 | 4.4 | 4.4 | 4.5 |
| 33 | 4.5 | 4.7 | 4.6 | 4.6 | 4.7 | 4.7 | 4.8 | 4.8 | 4.9 | 4.9 |
| 34 | 4.9 | 5.0 | 5.0 | 5.8 | 5.1 | 5.2 | 5.2 | 5.2 | 5.3 | 5.3 |
| 35 | 5.4 | 5.4 | 5.5 | 5.5 | 5.5 | 5.6 | 5.6 | 5.7 | 5.7 | 5.7 |
| 36 | 5.8 | 5.8 | 5.9 | 5.9 | 6.0 | 6.0 | 6.1 | 6.1 | 6.2 | 6.2 |
| 37 | 6.3 | 6.3 | 6.3 | 6.4 | 6.4 | 6.5 | 6.5 | 6.6 | 6.6 | 6.7 |
| 38 | 6.7 | 6.7 | 6.7 | 6.8 | 6.9 | 6.9 | 7.0 | 7.0 | 7.1 | 7.1 |
| 39 | 7.2 | 7.2 | 7.2 | 7.3 | 7.3 | 7.4 | 7.4 | 7.5 | 7.5 | 7.6 |
| 40 | 7.6 | 7.7 | 7.7 | 7.8 | 7.8 | 7.8 | 7.9 | 7.9 | 8.0 | 8.0 |
| 41 | 8.1 | 8.1 | 8.2 | 8.2 | 8.3 | 8.3 | 8.4 | 8.4 | 8.5 | 8.5 |
| 42 | 8.6 | 8.6 | 8.7 | 8.7 | 8.8 | 8.8 | 8.9 | 8.9 | 9.0 | 9.0 |
| 43 | 9.1 | 9.1 | 9.2 | 9.2 | 9.3 | 9.3 | 9.3 | 9.4 | 9.4 | 9.5 |
| 44 | 9.5 | 9.6 | 9.6 | 9.7 | 9.7 | 9.8 | 9.9 | 9.9 | 10.0 | 10.0 |
| 45 | 10.1 | 10.1 | 10.2 | 10.2 | 10.3 | 10.3 | 10.4 | 10.4 | 10.5 | 10.5 |
| 46 | 10.6 | 10.6 | 10.7 | 10.7 | 10.8 | 10.8 | 10.9 | 10.9 | 11.0 | 11.0 |
| 47 | 11.1 | 11.1 | 11.2 | 11.3 | 11.3 | 11.4 | 11.4 | 11.5 | 11.5 | 11.6 |
| 48 | 11.6 | 11.7 | 11.7 | 11.8 | 11.8 | 11.9 | 11.9 | 12.0 | 12.1 | 12.1 |
| 49 | 12.2 | 12.2 | 12.3 | 12.3 | 12.4 | 12.4 | 12.5 | 12.6 | 12.6 | 12.7 |
| 50 | 12.7 | 12.8 | 12.8 | 12.9 | 12.9 | 13.0 | 13.1 | 13.1 | 13.2 | 13.2 |
| 51 | 13.3 | 13.3 | 13.4 | 13.5 | 13.5 | 13.6 | 13.6 | 13.7 | 13.7 | 13.8 |
| 52 | 13.9 | 13.9 | 14.0 | 14.0 | 14.1 | 14.2 | 14.2 | 14.3 | 14.3 | 14.4 |
| 53 | 14.4 | 14.5 | 14.6 | 14.6 | 14.7 | 14.7 | 14.8 | 14.9 | 14.9 | 15.0 |
| 54 | 15.1 | 15.1 | 15.2 | 15.2 | 15.3 | 15.4 | 15.4 | 15.5 | 15.5 | 15.6 |
| 55 | 15.7 | 15.7 | 15.8 | 15.9 | 15.9 | 16.0 | 16.0 | 16.1 | 16.2 | 16.2 |
| 56 | 16.3 | 16.3 | 16.4 | 16.5 | 16.5 | 16.6 | 16.7 | 16.7 | 16.8 | 16.9 |
| 57 | 16.9 | 17.0 | 17.1 | 17.1 | 17.2 | 17.3 | 17.4 | 17.4 | 17.5 | 17.5 |
| 58 | 17.6 | 17.7 | 17.7 | 17.8 | 17.8 | 17.9 | 18.0 | 18.0 | 18.1 | 18.2 |
| 59 | 18.2 | 18.3 | 18.4 | 18.4 | 18.5 | 18.6 | 18.7 | 18.7 | 18.8 | 18.9 |
| 60 | 18.9 | 19.0 | 19.1 | 19.1 | 19.2 | 19.3 | 19.3 | 19.4 | 19.5 | 19.5 |
| 61 | 19.6 | 19.7 | 19.8 | 19.8 | 19.9 | 20.0 | 20.0 | 20.1 | 20.2 | 20.3 |
| 62 | 20.3 | 20.4 | 20.5 | 20.6 | 20.6 | 20.7 | 20.8 | 20.8 | 20.9 | 21.0 |
| 63 | 21.1 | 21.1 | 21.2 | 21.3 | 21.4 | 21.4 | 21.5 | 21.6 | 21.6 | 21.7 |
| 64 | 21.8 | 21.9 | 21.9 | 22.0 | 22.1 | 22.2 | 22.3 | 22.3 | 22.4 | 22.5 |
| 65 | 22.6 | 22.6 | 22.7 | 22.8 | 22.9 | 22.9 | 23.0 | 23.1 | 23.2 | 23.2 |
| 66 | 23.3 | 23.4 | 23.5 | 23.6 | 23.6 | 23.7 | 23.8 | 23.9 | 24.0 | 24.0 |
| 67 | 24.1 | 24.2 | 24.3 | 24.4 | 24.4 | 24.5 | 24.6 | 24.7 | 24.8 | 24.9 |
| 68 | 24.9 | 25.0 | 25.1 | 25.2 | 25.3 | 25.4 | 25.4 | 25.5 | 25.6 | 25.7 |
| 69 | 25.8 | 25.9 | 25.9 | 26.0 | 26.1 | 26.2 | 26.3 | 26.4 | 26.4 | 26.5 |
| 70 | 26.6 | 26.7 | 26.8 | 26.9 | 27.0 | 27.0 | 27.1 | 27.2 | 27.3 | 27.4 |

Table A—1 Continued

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 71 | 27.5 | 27.6 | 27.7 | 27.7 | 27.8 | 27.9 | 28.0 | 28.1 | 28.2 | 28.3 |
| 72 | 28.4 | 28.5 | 28.6 | 28.7 | 28.7 | 28.8 | 28.9 | 29.0 | 29.1 | 29.2 |
| 73 | 29.3 | 29.4 | 29.5 | 29.6 | 29.7 | 29.7 | 29.8 | 29.9 | 30.0 | 30.1 |
| 74 | 30.2 | 30.2 | 30.4 | 30.5 | 30.6 | 30.7 | 30.8 | 30.9 | 31.0 | 31.1 |
| 75 | 31.2 | 31.3 | 31.4 | 31.5 | 31.6 | 31.7 | 31.7 | 31.8 | 31.9 | 32.0 |
| 76 | 32.1 | 32.2 | 32.3 | 32.4 | 32.5 | 32.6 | 32.7 | 32.8 | 32.9 | 33.0 |
| 77 | 33.1 | 33.2 | 33.3 | 33.3 | 33.5 | 33.6 | 33.8 | 33.9 | 34.0 | 34.1 |
| 78 | 34.2 | 34.3 | 34.4 | 34.5 | 34.6 | 34.7 | 34.8 | 34.9 | 35.0 | 35.1 |
| 79 | 35.2 | 35.3 | 35.4 | 35.5 | 35.6 | 35.8 | 35.9 | 36.0 | 36.1 | 36.2 |
| 80 | 36.3 | 36.4 | 36.5 | 36.6 | 36.7 | 36.9 | 37.0 | 37.1 | 37.2 | 37.3 |
| 81 | 37.4 | 37.5 | 37.6 | 37.7 | 37.8 | 38.0 | 38.1 | 38.2 | 38.3 | 38.4 |
| 82 | 38.5 | 38.6 | 38.7 | 38.9 | 39.0 | 39.1 | 39.2 | 39.3 | 39.5 | 39.6 |
| 83 | 39.7 | 39.8 | 39.9 | 40.0 | 40.1 | 40.3 | 40.4 | 40.5 | 40.6 | 40.7 |
| 84 | 40.8 | 40.9 | 41.1 | 41.2 | 41.3 | 41.5 | 41.6 | 41.7 | 41.8 | 42.0 |
| 85 | 42.1 | 42.2 | 42.3 | 42.5 | 42.6 | 42.7 | 42.8 | 42.9 | 43.1 | 43.2 |
| 86 | 43.3 | 43.4 | 43.6 | 43.7 | 43.8 | 44.0 | 44.1 | 44.2 | 44.3 | 44.5 |
| 87 | 44.6 | 44.7 | 44.9 | 45.0 | 45.1 | 45.3 | 45.4 | 45.5 | 45.6 | 45.8 |
| 88 | 45.9 | 46.0 | 46.2 | 46.3 | 46.4 | 46.6 | 46.7 | 46.8 | 47.0 | 47.1 |
| 89 | 47.3 | 47.4 | 47.5 | 47.7 | 47.8 | 47.9 | 48.1 | 48.2 | 48.3 | 48.5 |
| 90 | 48.6 | 48.7 | 48.9 | 49.0 | 49.2 | 49.3 | 49.5 | 49.6 | 49.8 | 49.9 |
| 91 | 50.1 | 50.2 | 50.3 | 50.5 | 50.6 | 50.8 | 50.9 | 51.1 | 51.2 | 51.4 |
| 92 | 51.5 | 51.7 | 51.8 | 52.0 | 52.1 | 52.3 | 52.4 | 52.6 | 52.7 | 52.9 |
| 93 | 53.0 | 53.2 | 53.3 | 53.5 | 53.6 | 53.8 | 53.9 | 54.1 | 54.2 | 54.4 |
| 94 | 54.5 | 54.7 | 54.8 | 55.0 | 55.1 | 55.3 | 55.5 | 55.6 | 55.8 | 55.9 |
| 95 | 56.1 | 56.3 | 56.4 | 56.6 | 56.7 | 56.9 | 57.1 | 57.2 | 57.4 | 57.5 |
| 96 | 57.7 | 57.9 | 58.0 | 58.2 | 58.4 | 58.5 | 58.7 | 58.9 | 59.0 | 59.2 |
| 97 | 59.4 | 59.5 | 59.7 | 59.8 | 60.0 | 60.2 | 60.3 | 60.5 | 60.7 | 60.8 |
| 98 | 61.0 | 61.2 | 61.4 | 61.5 | 61.7 | 61.9 | 62.1 | 62.2 | 62.4 | 62.6 |
| 99 | 62.8 | 62.9 | 63.1 | 63.3 | 63.5 | 63.6 | 63.8 | 64.0 | 64.2 | 64.3 |
| 100 | 64.5 | 64.7 | 64.9 | 65.1 | 65.3 | 65.5 | 65.6 | 65.8 | 66.0 | 66.2 |
| 101 | 66.4 | 66.6 | 66.8 | 67.0 | 67.2 | 67.4 | 67.5 | 67.7 | 67.9 | 68.1 |
| 102 | 68.3 | 68.5 | 68.7 | 68.9 | 69.1 | 69.3 | 69.5 | 69.7 | 69.9 | 70.1 |
| 103 | 70.3 | 70.4 | 70.6 | 70.8 | 71.0 | 71.2 | 71.4 | 71.6 | 71.8 | 72.0 |
| 104 | 72.2 | 72.4 | 72.6 | 72.8 | 73.1 | 73.3 | 73.5 | 73.7 | 73.9 | 74.1 |
| 105 | 74.3 | 74.5 | 74.8 | 75.0 | 75.2 | 75.4 | 75.6 | 75.8 | 76.0 | 76.2 |
| 106 | 76.5 | 76.7 | 76.9 | 77.1 | 77.3 | 77.5 | 77.7 | 77.9 | 78.2 | 78.4 |
| 107 | 78.6 | 78.8 | 79.0 | 79.2 | 79.4 | 79.6 | 79.9 | 80.1 | 80.3 | 80.5 |
| 108 | 80.7 | 80.9 | 81.2 | 81.4 | 81.7 | 81.9 | 82.1 | 82.4 | 82.6 | 82.9 |
| 109 | 83.1 | 83.3 | 83.6 | 83.8 | 84.1 | 84.3 | 84.5 | 84.8 | 85.0 | 85.3 |
| 110 | 85.5 | 85.7 | 86.0 | 86.2 | 86.5 | 86.7 | 86.9 | 87.2 | 87.4 | 87.7 |
| 111 | 87.9 | 88.1 | 88.4 | 88.6 | 88.9 | 89.1 | 89.3 | 89.6 | 89.8 | 90.1 |
| 112 | 90.3 | 90.6 | 90.8 | 91.9 | 91.4 | 91.6 | 91.9 | 92.2 | 92.4 | 92.7 |
| 113 | 93.0 | 93.2 | 93.5 | 93.7 | 94.0 | 94.3 | 94.5 | 94.8 | 95.1 | 95.3 |
| 114 | 95.6 | 95.9 | 96.1 | 96.4 | 96.7 | 96.9 | 97.2 | 97.5 | 97.7 | 98.0 |
| 115 | 98.2 | 98.5 | 98.8 | 99.0 | 99.3 | 99.6 | 99.8 | 100.1 | 100.4 | 100.6 |
| 116 | 100.9 | 101.2 | 101.5 | 101.8 | 102.1 | 102.4 | 102.7 | 103.0 | 103.3 | 103.6 |
| 117 | 103.9 | 104.2 | 104.5 | 104.8 | 105.1 | 105.4 | 105.7 | 106.0 | 106.3 | 106.6 |
| 118 | 106.9 | 107.1 | 107.4 | 107.7 | 108.0 | 108.3 | 108.6 | 108.9 | 109.2 | 109.5 |
| 119 | 109.8 | 110.0 | 110.4 | 110.7 | 111.0 | 111.3 | 111.6 | 111.9 | 112.2 | 112.5 |
| 120 | 112.8 | 113.1 | 113.5 | 113.8 | 114.1 | 114.5 | 114.8 | 115.1 | 115.5 | 115.8 |
| 121 | 116.2 | 116.5 | 116.8 | 117.2 | 117.5 | 117.8 | 118.2 | 118.5 | 118.8 | 119.7 |
| 122 | 119.5 | 119.8 | 120.2 | 120.5 | 120.8 | 121.2 | 121.5 | 121.8 | 122.2 | 122.5 |
| 123 | 122.9 | 123.2 | 123.5 | 123.9 | 124.2 | 124.5 | 124.9 | 125.2 | 125.5 | 125.9 |
| 124 | 126.2 | 126.6 | 127.0 | 127.3 | 127.7 | 128.1 | 128.5 | 128.8 | 129.2 | 129.6 |
| 125 | 130.0 | 130.3 | 130.7 | 131.1 | 131.5 | 131.8 | 132.2 | 132.6 | 133.0 | 133.3 |
| 126 | 133.7 | 134.1 | 134.5 | 134.8 | 135.2 | 135.6 | 136.0 | 136.3 | 136.7 | 137.1 |
| 127 | 137.5 | 137.8 | 138.2 | 138.6 | 139.0 | 139.3 | 139.7 | 140.1 | 140.5 | 140.8 |
| 128 | 141.2 | 141.6 | 142.1 | 142.5 | 142.9 | 143.3 | 143.8 | 144.2 | 144.6 | 145.0 |
| 129 | 145.5 | 145.9 | 146.3 | 146.8 | 147.2 | 147.6 | 148.0 | 148.5 | 148.9 | 149.3 |
| 130 | 149.8 | 150.2 | 150.6 | 151.0 | 151.5 | 151.9 | 152.3 | 152.7 | 153.2 | 153.6 |
| 131 | 154.0 | 154.5 | 154.9 | 155.3 | 155.7 | 156.2 | 156.6 | 157.0 | 157.4 | 157.9 |
| 132 | 158.3 | 158.8 | 159.3 | 159.7 | 160.2 | 160.7 | 161.2 | 161.7 | 162.2 | 162.6 |
| 133 | 163.1 | 163.6 | 164.1 | 164.6 | 165.1 | 165.5 | 166.0 | 166.5 | 167.0 | 167.5 |
| 134 | 168.0 | 168.4 | 168.9 | 169.4 | 169.9 | 170.4 | 170.8 | 171.3 | 171.8 | 172.3 |
| 135 | 172.8 | 173.3 | 173.7 | 174.2 | 174.7 | 175.2 | 175.7 | 176.2 | 176.6 | 177.1 |

Table A-2. Basic Forms of Packing [22]

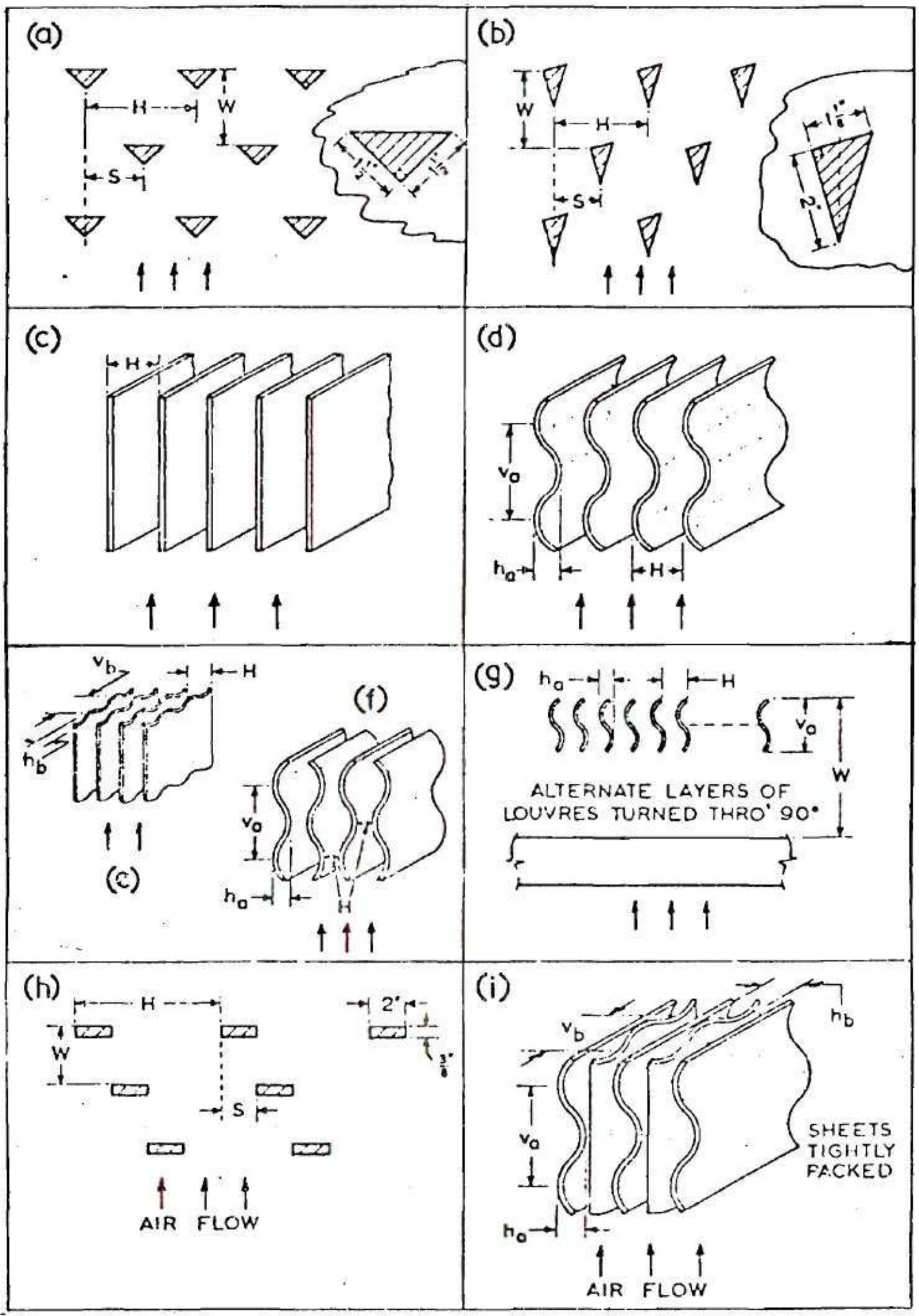


Table A-2 (continued) Cooling Tower Packing Transfer Number

| Packing No. | Description of Packing | Figure No. | Dimensions | | | | | Transfer | |
|---|---|---|---|---|---|---|---|---|---|
| | | | $h_o$ (inches) | $v_o$ (inches) | $H$ (inches) | $W$ (inches) | $S$ (inches) | $\lambda$ | $n$ |
| 7 | Triangular Splash Bar | 2 (a) | | | 6 | 9 | 3 | 0.09 | 0.50 |
| 8 | " | " | | | 6 | 6 | 3 | 0.094 | 0.50 |
| 9 | " | " | | | 6 | 5 & 13 Alternately | 3 | 0.096 | 0.45 |
| 10 | " | " | | | 6 | 12 | 3 | 0.075 | 0.42 |
| 11 | " | " | | | $4\frac{1}{2}$ | 18 | $2\frac{1}{4}$ | 0.072 | 0.47 |
| 14 | Flat Asbestos Sheets | 2 (c) | | | $1\frac{3}{4}$ | | | 0.088 | 0.70 |
| 15 | " | " | | | $1\frac{1}{2}$ | | | 0.11 | 0.72 |
| 16 | " | " | | | $1\frac{1}{4}$ | | | 0.12 | 0.76 |
| 17 | " | " | | | 1 | | | 0.14 | 0.73 |
| 19 | Triangular Splash Bar | As 2 (a) | With Bars Upside Down | | 6 | 9 | 3 | 0.084 | 0.49 |
| 21 | Corrugated Asbestos Sheets | 2 (d) | $2\frac{1}{8}$ | $5\frac{3}{4}$ | $1\frac{3}{4}$ | | | 0.21 | 0.69 |
| 22 | " | " | $2\frac{1}{8}$ | $5\frac{3}{4}$ | $1\frac{1}{4}$ | | | 0.22 | 0.61 |
| 23 | " | " | $2\frac{1}{8}$ | $5\frac{3}{4}$ | $2\frac{1}{4}$ | | | 0.18 | 0.68 |
| 24 | " | 2 (e) | $h_b = 2\frac{1}{8}$ | $v_b = 5\frac{3}{4}$ | $1\frac{3}{4}$ | | | 0.11 | 0.65 |
| 25 | " | 2 (f) | $2\frac{1}{8}$ | $5\frac{3}{4}$ | 1 | | | 0.17 | 0.58 |
| 26 | Triangular Splash Bar | 2 (b) | | | 4 | 8 | 0 | 0.074 | 0.52 |
| 27 | " | " | | | 4 | 8 | 2 | 0.087 | 0.55 |
| 28 | " | " | | | 4 | 10 | 2 | 0.079 | 0.58 |
| 29 | " | " | | | 4 | 10 | 0 | 0.072 | 0.54 |
| 30 | " | " | | | 4 | $7\frac{1}{2}$ | 2 | 0.095 | 0.53 |
| 31 | " | " | | | 4 | 6 | 2 | 0.098 | 0.54 |
| 32 | " | " | | | 5 | 8 | $2\frac{1}{2}$ | 0.093 | 0.46 |
| 37 | " | " | | | 2 | 6 | 1 | 0.137 | 0.65 |
| 38 | Asbestos Louvres | 2 (g) | 1 | $5\frac{3}{4}$ | 1 | $10\frac{3}{4}$ | | 0.203 | 0.70 |
| 39 | " | " | 1 | $5\frac{3}{4}$ | 1 | $6\frac{3}{4}$ | | 0.237 | 0.68 |
| 40 | " | " | 1 | $5\frac{3}{4}$ | 1 | $20\frac{3}{4}$ | | 0.118 | 0.69 |
| 41 | " | " | 1 | $5\frac{3}{4}$ | 1 | $15\frac{3}{4}$ | | 0.154 | 0.67 |
| 42 | Triangular Splash Bar | 2 (b) | | | 5 | $7\frac{1}{2}$ | $2\frac{1}{2}$ | 0.095 | 0.49 |
| 43 | " | " | | | 6 | $7\frac{1}{2}$ | 3 | 0.089 | 0.47 |
| 45 | Asbestos Louvres | 2 (g) | $1\frac{1}{2}$ | $5\frac{1}{4}$ | 1 | $6\frac{1}{4}$ | | 0.351 | 0.66 |
| 47 | " | " | $1\frac{1}{2}$ | $5\frac{1}{4}$ | $1\frac{1}{2}$ | $3\frac{1}{4}$ | | 0.247 | 0.66 |
| 48 | " | " | $1\frac{1}{2}$ | $5\frac{1}{4}$ | $1\frac{1}{2}$ | $15\frac{1}{4}$ | | 0.169 | 0.65 |
| 49 | " | " | $1\frac{1}{2}$ | $5\frac{1}{4}$ | $1\frac{1}{2}$ | $20\frac{1}{4}$ | | 0.101 | 0.63 |
| 50 | Rectangular Splash Bar | 2 (h) | | | 8 | 9 | 2 | 0.086 | 0.52 |
| 51 | " | " | | | 8 | 12 | 2 | 0.08 | 0.53 |
| | | | Corrugations Horiz. | | Corrugations Vert. | | | | |
| | | | $h_o$ | $v_c$ | $h_b$ | $v_b$ | | | |
| 55 | Corrugated Asbestos Sheets | 2 (i) | $2\frac{1}{4}$ | $5\frac{1}{4}$ | $2\frac{1}{4}$ | $5\frac{1}{4}$ | | 0.186 | 0.73 |
| 57 | " | " | $1\frac{7}{8}$ | $2\frac{7}{8}$ | $1\frac{7}{16}$ | $2\frac{7}{8}$ | | 0.308 | 0.80 |
| 58 | " | " | $1\frac{7}{16}$ | $2\frac{7}{8}$ | $2\frac{1}{2}$ | $5\frac{3}{4}$ | | 0.207 | 0.79 |
| 59 | " | " | $2\frac{1}{4}$ | $5\frac{3}{4}$ | $1\frac{7}{16}$ | $2\frac{7}{8}$ | | 0.248 | 0.79 |
| 61 | " | " | $2\frac{1}{2}$ | 7 | $2\frac{1}{2}$ | 7 | | 0.163 | 0.71 |
| 62 | " | " | $1\frac{7}{16}$ | $2\frac{7}{8}$ | $8\frac{3}{4}$ | $2\frac{15}{16}$ | | 0.133 | 0.72 |

Table A-3. Experimental Data of Chilton Coefficient [21]

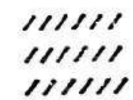
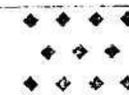
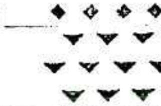
| Tower | Design data | | | Range of variables | | | | Number of tests | Mean per-formance coefficient | Type of packing | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Height | Internal base diameter | Depth of packing | Wet-bulb tempera-ture | Dry-bulb tempera-ture | Water loading | Cooling range | | | | |
| | ft | ft | ft | °F | °F | lb/h-ft² | °F | | | | |
| A | 290 | 194·75 | 22·5 | 40·72 to 52·96 | 42·5 to 59·4 | 782 to 840 | 12·75 to 16·28 | 9 | 5·05 |  | Upper packing |
| B | 140 | 92 | 23 | 32·53 to 58·5 | 37·1 to 74·1 | 1 160 to 1 780 | 8·0 to 22·5 | 16 | 4·79 |  | Lower packing |
| C | 125 | 95 | 20 | 36·0 to 70·5 | 38·7 to 77·0 | 56·4 to 1 156 | 8·3 to 20·0 | 117 | 5·49 | | |
| D | 175 | 119 | 18 | 32·53 to 66·34 | 37·1 to 84·6 | 740 to 1 088 | 8·8 to 20·2 | 26 | 5·41 |  | Square packing |
| E | 175 | 119 | 18 | 32·53 to 56·43 | 37·1 to 64·2 | 742 to 1 090 | 7·5 to 20·0 | 19 | 5·46 |  | Triangular packing |
| F | 125 | 95 | 3·5 | 42·2 to 65·9 | 46·2 to 73·7 | 960 to 1 030 | 9·3 to 14·2 | 62 | 5·69 |  | Corrugated asbestos sheets |

Table A-4. Integral Aluminum High Finned Tubing [31]

| Type of finned tube, in. | Tube material | Thickness of liner, in. | Thickness of liner, BWG | Liner OD, in. | Minimum fin tube OD (root-tube OD), in. | Minimum fin tube wall, in. | Fin height, in. | OD of fins, in. | Mean fin thickness, in. | Fins per in. | Outside surface, $ft^2/ft$ | Ratio outside to inside surface | Maximum recommended metal temperature, °F |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Aluminum | Alloy copper | 0.028 | 22 | 1 | 1.08 | 0.04 | $\frac{1}{2}$ | 2 | 0.019 | 9 | 3.59 | 14.58 | 400 600 |
| | | 0.035 | 20 | | | | | | | | | 14.78 | |
| | | 0.042 | 19 | | | | | | | | | 14.98 | |
| | | 0.049 | 18 | | | | | | | | | 15.28 | |
| | | 0.065 | 16 | | | | | | | | | 15.78 | |
| | | 0.083 | 14 | | | | | | | | | 16.58 | |
| | | 0.095 | 13 | | | | | | | | | 16.98 | |

Table A-5. Air Density at 60% Relative Humidity and Different Dry Bulb Temperatures

| $T_{dry\ bulb}$ (°F) | $\rho$ ($lbm/ft^3$) |
|---|---|
| 35 | 0.080048 |
| 40 | 0.07922 |
| 45 | 0.07840 |
| 50 | 0.07760 |
| 55 | 0.07680 |
| 60 | 0.07602 |
| 65 | 0.07523 |
| 70 | 0.07445 |
| 75 | 0.07369 |
| 80 | 0.07291 |
| 85 | 0.07215 |
| 90 | 0.07138 |
| 95 | 0.07060 |
| 100 | 0.06983 |
| 105 | 0.06905 |
| 110 | 0.06825 |
| 115 | 0.06745 |
| 120 | 0.06663 |
| 125 | 0.06581 |

P: 1000 millibars

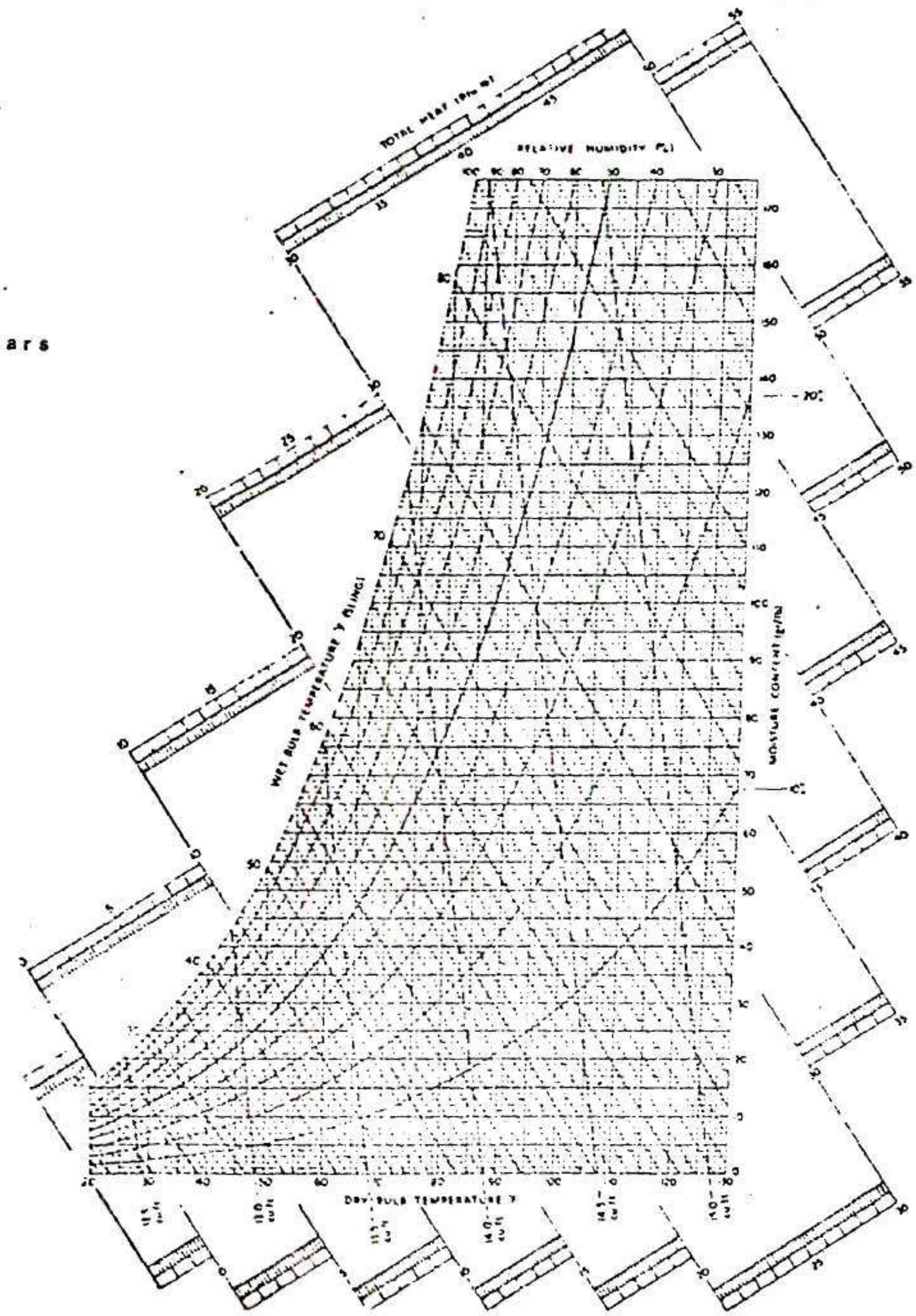



Figure A-1. Psychrometric Chart [17]

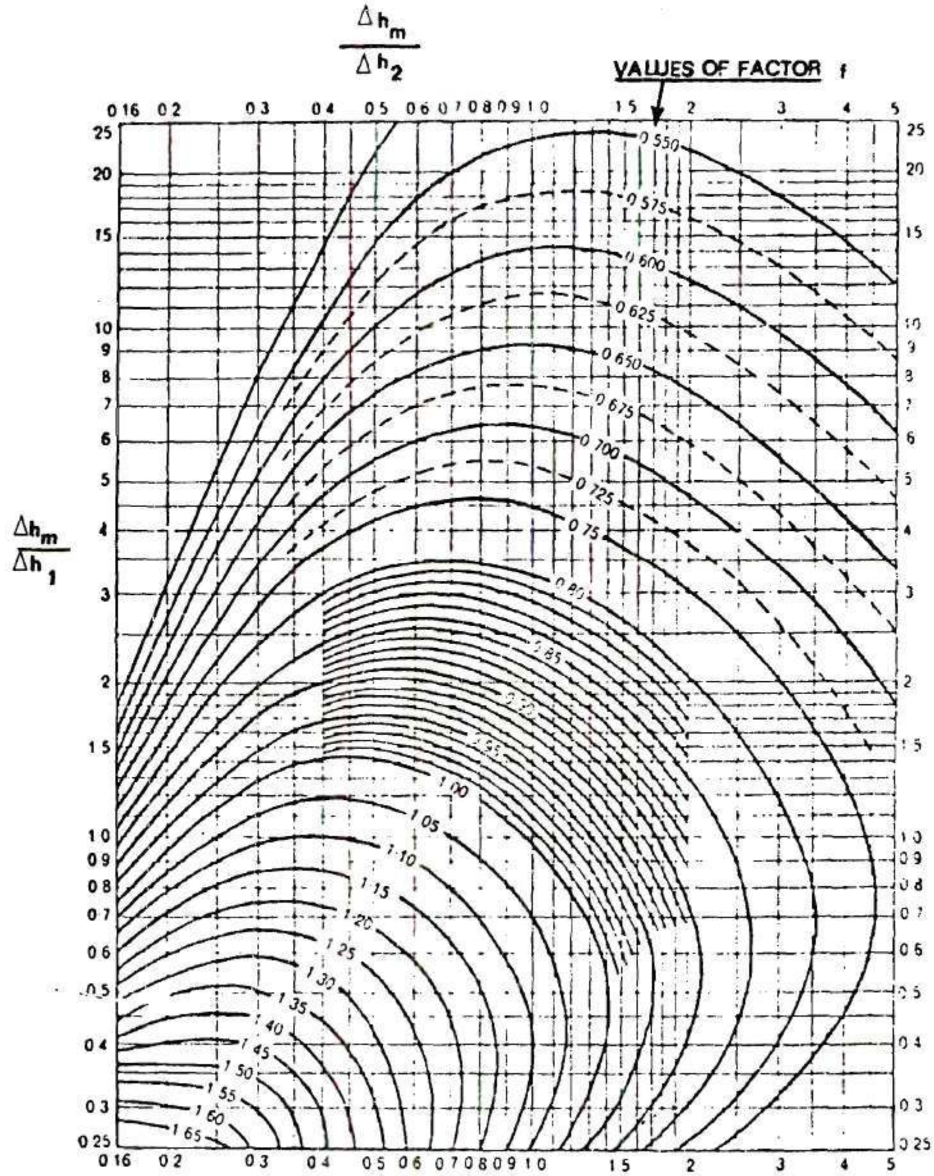



Figure A-2. Chart for Determination of Mean Driving Force [17]

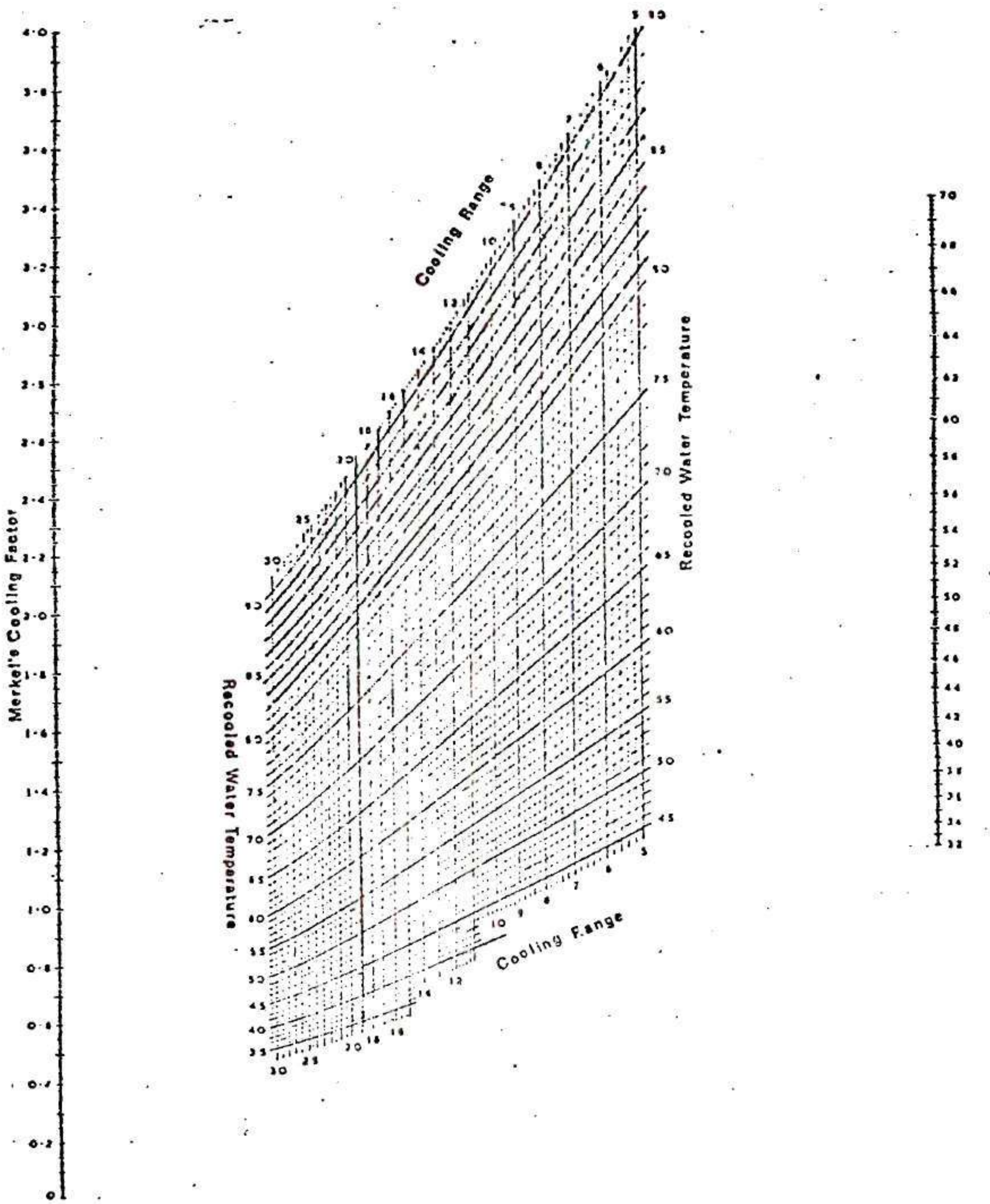



Figure A-3. Cooling Tower Performances Nomogram Based on Merkel's Approximate Integration [21]

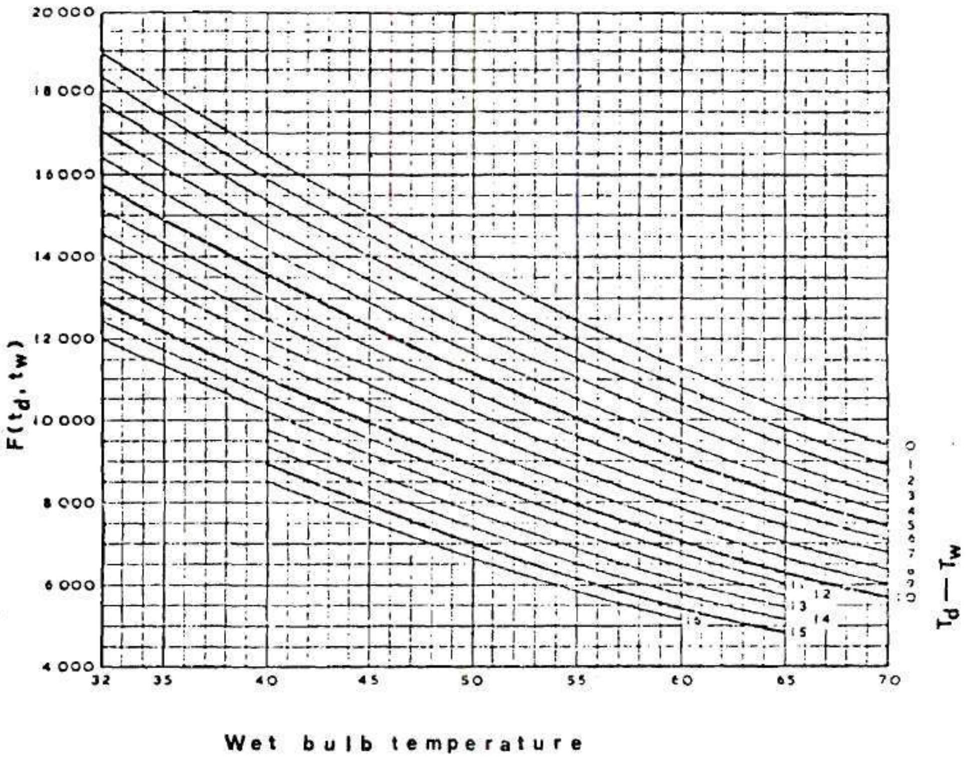



Figure A-4. Values of $F(t_d, t_w)$ [21]

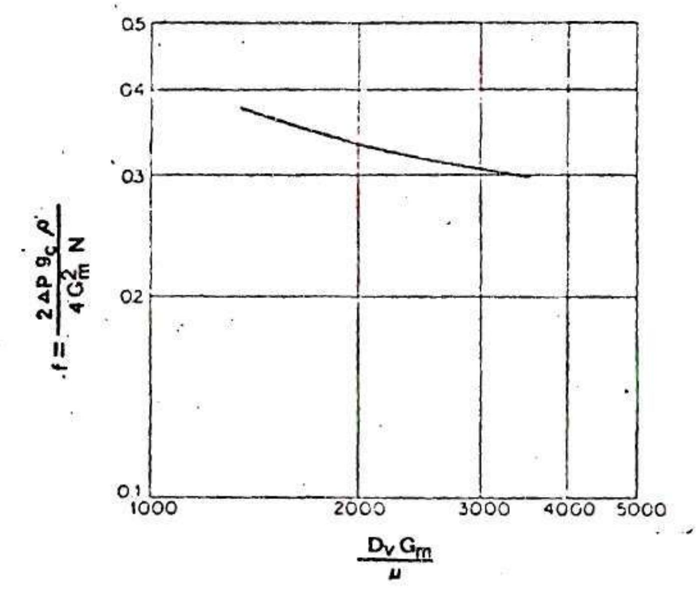



Figure A-6. Fin-Side Friction Factor [31]

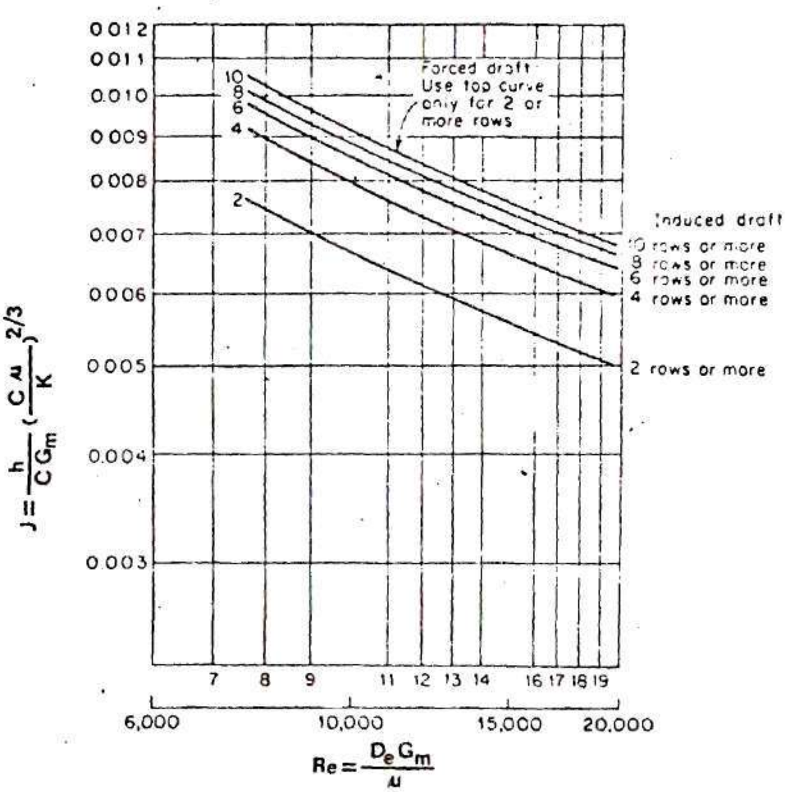



Figure A-5. Air-Side Heat Transfer Factor j [31]

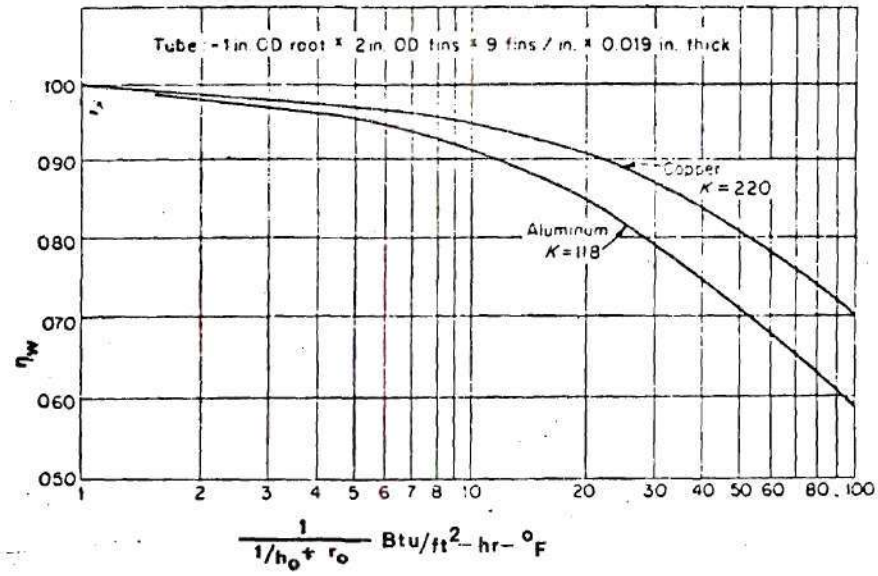



Figure A-7. Fin Efficiency [31]

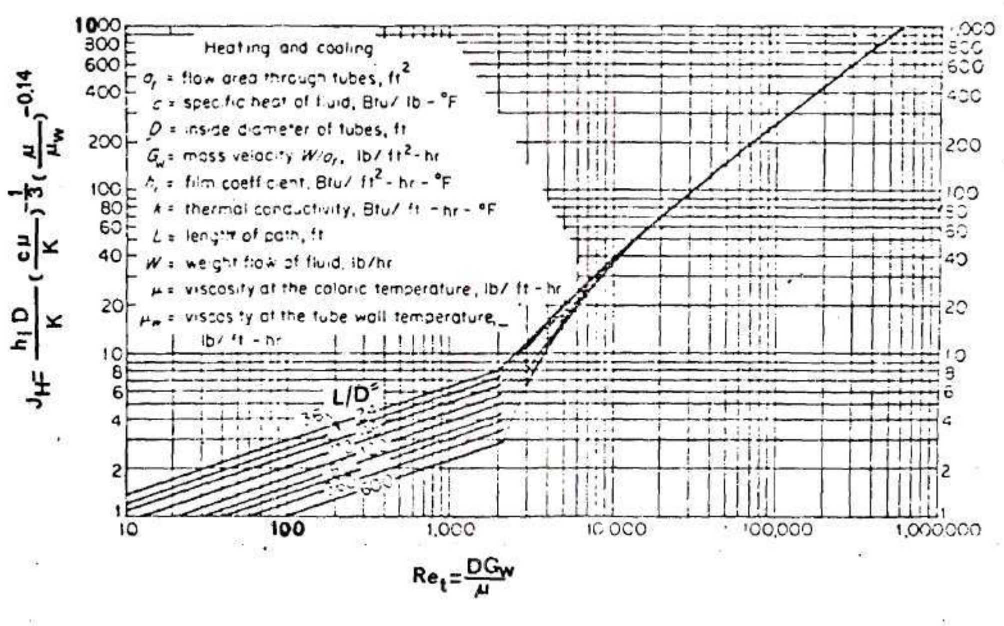



Figure A-8. Tube-Side Heat Transfer Factor $j_h$ [31]

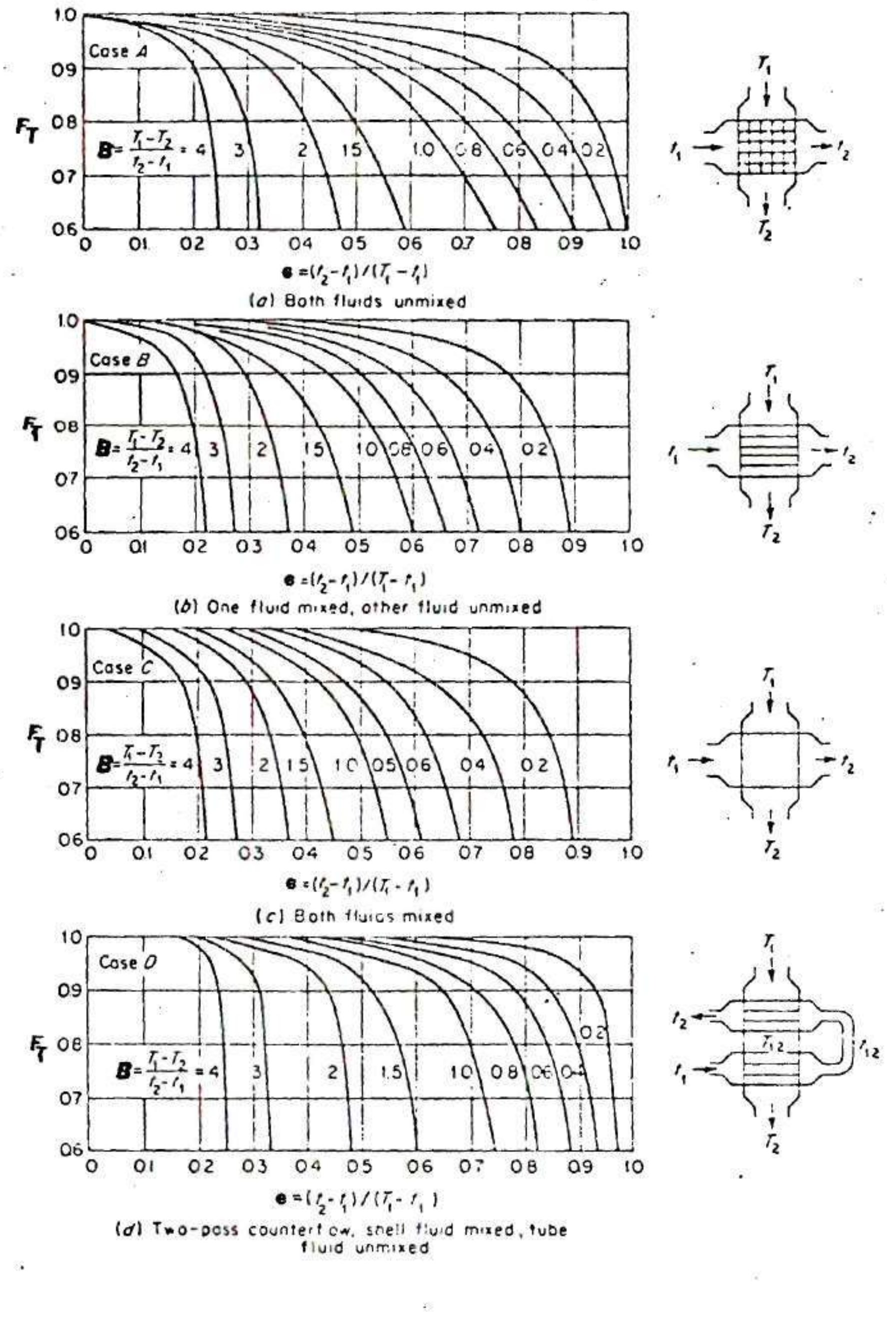



Figure A-9. Values of F(t) [31]

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