



The Society shall not be responsible for statements or opinions advanced in papers or in discussion at meetings of the Society or of its Divisions or Sections, or printed in its publications. Discussion is printed only if the paper is published in an ASME Journal. Papers are available from ASME for fifteen months after the meeting.  
Printed in USA.

## The Theory and Operation of Evaporative Coolers For Industrial Gas Turbine Installations

R. S. JOHNSON, Sr., P.E.

Principal Engineer

Solar Turbines Incorporated

San Diego, California

Member ASME

### ABSTRACT

This paper discusses the theory of evaporative cooling and describes the application of wetted rigid media evaporative coolers to gas turbines. Calculations of parameters used to predict evaporative cooler performance are included. Also included are discussions of evaporative cooler design, installation, operation, feedwater quality, and the causes and prevention of water carry-over.

### NOMENCLATURE

A	= Feedwater rate, gpm
B	= Blowdown rate, gpm
D	= Weight of water, lb/gal.
E	= Water evaporation rate, gpm
H	= Water hardness in reservoir, as $\text{CaCO}_3$
N	= Evaporative cooler effectiveness, %
$\dot{Q}$	= Thermal energy flow rate from water to air, Btu/hr
T	= Contaminant concentration as Total Fuel Equivalent, ppm by weight
$\dot{V}$	= Gas turbine air volume flow rate, $\text{ft}^3/\text{min}$ .
X	= Feedwater hardness as $\text{CaCO}_3$ , ppm
AFR	= Air/fuel mass flow ratio
CFR	= Water carry-over/fuel mass flow ratio
LHV	= Lower heating value of fuel, Btu/lb
SFR	= Steam/fuel mass flow ratio
WFR	= Water/fuel mass flow ratio
$A_i$	= Concentration of the "i"th contaminant in air, ppm by weight
$C_i$	= Concentration of the "i"th contaminant in carry-over water, ppm by weight

$F_i$	= Concentration of the "i"th contaminant in fuel, ppm by weight
$\dot{m}_a$	= Air mass flow rate, lb/hr
$\dot{m}_w$	= Water mass flow rate, lb/hr
$m_1$	= Moisture in incoming air, lb moisture/lb dry air
$m_2$	= Moisture in leaving air, lb moisture/lb dry air
$P_a$	= Density of air at ambient temperature, $\text{lb}/\text{ft}^3$
$S_i$	= Concentration of the "i"th contaminant in steam, ppm by weight
$T_{DB1}$	= Dry-bulb temperature of air entering the evaporative cooler, °F
$T_{DB2}$	= Dry-bulb temperature of air leaving the evaporative cooler, adiabatic condition, °F
$T_{DN1}$	= Dry-bulb temperature of air leaving the evaporative cooler, nonadiabatic condition, °F
$T_{WB}$	= Ambient wet-bulb temperature, °F
$W_i$	= Concentration of "i"th contaminant in water, ppm by weight

### CONVERSION FACTORS

°C	= $(^{\circ}\text{F} - 32)/1.8$
kg/L	= 8.33 lb/U.S. gal.
kg/m <sup>3</sup>	= 6.25 E-02 lb/ft <sup>3</sup>
kg/s	= 7.937 E+03 lb/hr
kJ/kg	= 4.292 E-01 Btu/lb
kJ/kg·°C	= 2.387 E-01 Btu/lb·°F
L/s	= 1.585 E+01 U.S. gal./min.
m <sup>3</sup> /sec	= 2.120 E+03 cfm
m	= 3.28 ft
W	= 3.413 Btu/hr

## INTRODUCTION

Evaporative coolers are used with gas turbines to increase the density of the combustion air, thereby increasing power output. The air density increase is accomplished by evaporating water into the inlet air, which decreases its temperature and correspondingly increases its density. The water vapor passes through the turbine, causing a negligible increase in fuel consumption.

Water used with evaporative coolers often contains dissolved solids such as sodium and potassium, which, in combination with sulfur in the fuel, are principal ingredients in hot gas path corrosion. For this reason, water quality and the prevention of water carry-over are important considerations in the use of evaporative coolers. The prevention of water carry-over is accomplished by correct design of the evaporative cooler, and proper installation and operation. Water quality requirements depend on the amount of water carry-over expected (or allowed) and can vary from the use of deionized water to water with significant concentrations (as much as several hundred ppm by weight, in water) of sodium and potassium.

## THEORY

Evaporative cooling involves heat and mass transfer, which occurs when water and the unsaturated air-water mixture of the incoming air are in contact. This transfer is a function of the differences in temperatures and vapor pressures between the air and water. Heat and mass transfer are both operative in the evaporative cooler because heat transfer from the air to the water evaporates water, and the water evaporating into the air constitutes mass transfer.

Heat inflow can be described as either latent or sensible heat. Whichever term is used depends on the effect. If the effect is only to raise or lower temperature, it is sensible heat. Latent heat, on the other hand, produces a change of state, e.g., freezing, melting, condensing, or vaporizing.

In evaporative cooling, sensible heat from the air is transferred to the water, becoming latent heat as the water evaporates. The water vapor becomes part of the air and carries the latent heat with it. The air dry-bulb temperature is decreased because it gives up sensible heat. The air wet-bulb temperature is not affected by the absorption of latent heat in the water vapor because the water vapor enters the air at the air wet-bulb temperature.

Theoretically, the incoming air and the water in the evaporative cooler may be considered an isolated system. Because no heat is added to or removed from the system, the process of exchanging the sensible heat of the air for latent heat of evaporation from the water is adiabatic. Evaporative cooler performance, therefore, is based on the concept of an adiabatic process.

## RIGID MEDIA EVAPORATIVE COOLERS

Evaporative coolers used in gas turbine applications today are the wetted rigid media type (Fig. 1). In this type of evaporative cooler, the evaporating medium is a saturated porous pad. Water is introduced through a header at the top of the media, sprays into the top of an inverted half-pipe, and is deflected downward onto a distribution pad on top of the media. Water drains through the distribution pad into the media, by gravity action downward through it, and wets enormous areas of media surface contacted by air passing through the cooler.

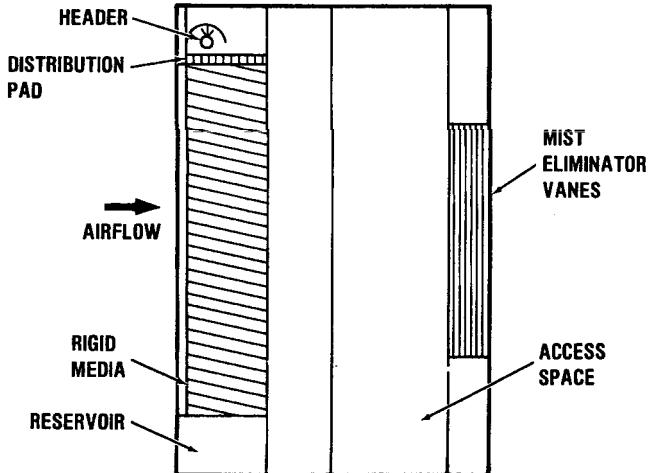


FIG. 1 RIGID MEDIA EVAPORATIVE COOLER

An evaporative cooler can be either recirculating or nonrecirculating. A recirculating evaporative cooler holds water in a reservoir and uses a pump to supply water to the header pipe. Water drains through the media into the reservoir below. Make-up water replacing evaporated water and water lost through blowdown (which is explained later) flows into the reservoir. For a nonrecirculating cooler, new water is continuously introduced directly into the header from the water supply outside the cooler, passes once through the media, and is discarded. The water flow rate to the header should be the same for both types of coolers.

## OPERATING PARAMETERS

In determining the effect of the evaporative cooler on turbine performance, the dry-bulb temperature ( $T_{DB1}$ ) of the air leaving the evaporative cooler must be known. This is calculated using Eq. (1):

$$T_{DB2} = T_{DB1} - \frac{N}{100\%} (T_{DB1} - T_{WB}) \quad (1)$$

In most applications, the relative humidity (RH) of the incoming air is known rather than the wet-bulb temperature. The wet-bulb temperature can be found from the psychrometric chart in Fig. 2. For example, assume that it is desired to consider the effect of an evaporative cooler on turbine performance with the following conditions:

- RH = 20%
- Ambient Temperature ( $T_{DB1}$ ) = 105°F
- Evaporative Cooler Effectiveness ( $N/100\%$ ) = 0.90

Enter the bottom of the chart at 105°F and proceed upward to intersect the 20% RH line. Then follow the diagonal line to the wet-bulb and dew-point temperature lines, and read 72.3°F for  $T_{WB}$ . Using Eq. (1), calculate  $T_{DB2}$ :

$$T_{DB2} = 105 - 0.90 (105 - 72.3) = 75.6°F$$

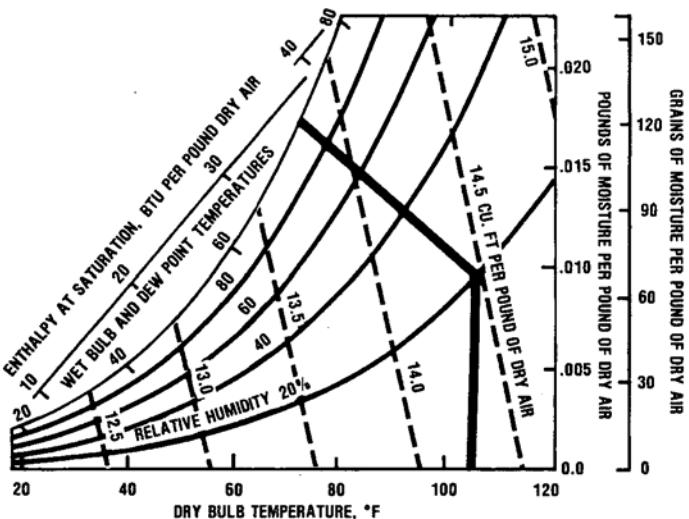


FIG. 2 PSYCHROMETRIC CHART A  
(BAROMETRIC PRESSURE 29.92 INCHES OF MERCURY)

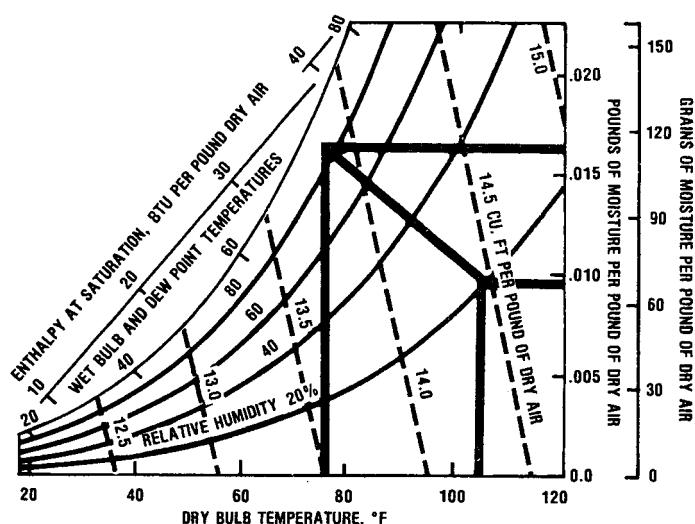


FIG. 3 PSYCHROMETRIC CHART B  
(BAROMETRIC PRESSURE 29.92 INCHES OF MERCURY)

This is the temperature of the turbine combustion inlet air under the conditions described.

If the evaporative cooler being used is nonrecirculating, this is the only calculation needed. If the cooler is a recirculating type, it is necessary to determine the blowdown and make-up water flow rates. Blowdown is a continuous wastage or bleed-off of water from the reservoir to carry away accumulating minerals. This controls the concentration level of dissolved solids in, and/or hardness of, the water in the reservoir. The rate of blowdown is determined by the initial condition of the water, the rate of evaporation, and the degree of concentration to be allowed. Make-up water is the water added to the reservoir to make up for water lost from blowdown and evaporation.

In order to calculate the blowdown rate (B), it is necessary to first calculate the rate of water evaporation (E). Using the conditions given above and the calculated exiting air dry-bulb temperature ( $T_{DB2}$ ), Eq. (2) is employed:

$$E = \frac{\dot{V} [m_2 - m_1] P_a}{D} \text{ gpm} \quad (2)$$

To determine the values for  $m_2$  and  $m_1$ , enter Fig. 3 at the 105°F ( $T_{DB1}$ ) and proceed upward to the 20% RH line. Move perpendicularly to the right and read 0.0094 lb moisture/lb of dry air ( $m_1$ ).

Return to the 20% RH line and move diagonally left to intersect the 75.6°F ( $T_{DB2}$ ) line. Now move perpendicularly to the right and read 0.0162 lb moisture/lb of dry air ( $m_2$ ).

From the psychrometric chart, Fig. 3, the specific volume of air at 105°F and 20% relative humidity is 14.442 ft<sup>3</sup>/lb. Therefore,  $P_a = 0.0692 \text{ lb/ft}^3$ .

Assume that the gas turbine combustion airflow rate ( $\dot{V}$ ) is 30,000 cfm. Take  $D = 8.33 \text{ lb/gal}$ . and return to Eq. (2):

$$E = [30,000 \text{ cfm} (0.0162 \text{ lb/lb dry air} - 0.0094 \text{ lb/lb dry air}) 0.0692 \text{ lb/ft}^3] \div 8.33 \text{ lb/gal.}$$

Therefore,

$E = 1.7 \text{ gal./min.}$  (The rate at which water evaporates into the inlet air)

The feedwater and blowdown rates can now be determined if the hardness of the feedwater is known. If the feedwater has been tested and found to have, for example, a hardness of 150 ppm as  $\text{CaCO}_3$ , the blowdown rate can be determined. From the curve in Fig. 4, which is widely used in the evaporative cooler industry to estimate blowdown ratio, the blowdown rate (B) should equal four times the rate of evaporation (E):

$$B = 4 (E \text{ gpm}) = 4 (1.7) = 6.8 \text{ gpm} \quad (3)$$

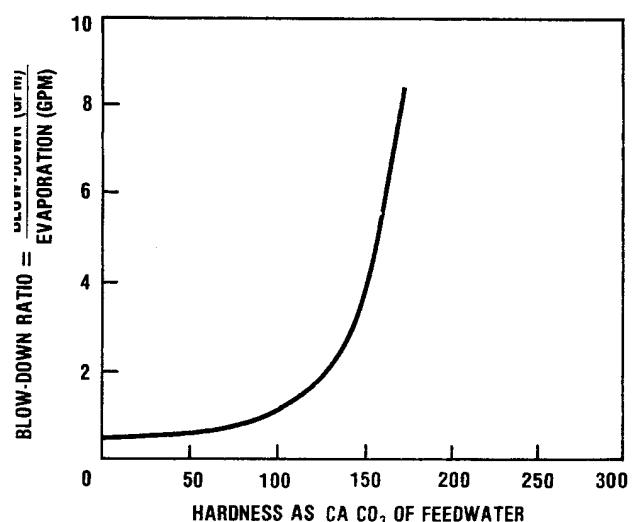


FIG. 4 HARDNESS AS  $\text{CaCO}_3$  OF FEEDWATER

The feedwater rate is found from:

$$A = B + E \text{ gpm} \quad (4)$$

$$= 6.8 + 1.7 = 8.5 \text{ gpm}$$

The water hardness in the reservoir, measured as  $\text{CaCO}_3$ , can be calculated from Eq. (5):

$$H = \frac{AX}{B} \text{ ppm} \quad (5)$$

$$\frac{(8.5 \text{ gpm})(150 \text{ ppm})}{6.8 \text{ gpm}} = 187.5 \text{ ppm}$$

The same procedure can be used to calculate the concentrations of dissolved solids in the reservoir from known concentrations in the feedwater.

A question frequently asked about evaporative coolers involves the use of make up water from discarded warm water from a manufacturing or industrial process. If the water has satisfactorily low levels of hardness, or dissolved solids, the question then becomes one of whether the water can be used despite its elevated temperature. The following discussion of adiabatic and nonadiabatic cooling attempts to answer this question.

The temperature of the water used with the evaporative cooler is also limited by the type of media. If a cellulose-based rigid media is used, which is the type typically installed in evaporative coolers used with gas turbines, water temperature above 130°F may damage it.

## ADIABATIC COOLING

Adiabatic cooling occurs when the temperature of water delivered to the header of the evaporative cooler is equal to the incoming air wet-bulb temperature ( $T_{WB}$ ), and no external heat enters the process.

Because the ambient wet-bulb temperature is constantly changing, this condition only rarely occurs when a nonrecirculating evaporative cooler is being used. In most cases in hot weather, however, it is sufficiently accurate to assume that the water supply temperature approximates the air wet-bulb temperature. This is because the temperature of many water sources is naturally close to, or slightly lower than, the wet-bulb temperature. However, the temperature of water in long runs of sun-struck pipes can be significantly increased.

In a recirculating evaporative cooler, the water temperature will approach the wet-bulb temperature so closely that for all practical purposes, the two temperatures can be taken as equal. This is true even if the water used is significantly above the wet-bulb temperature when it is delivered to the reservoir.

In all cases, some external heat enters the process from the sun shining on the cooler. However, because the air's mass flow rate is so large, its effect is minimal.

The adiabatic process is explained in Fig. 5. The incoming air dry-bulb temperature enters at position 1 and moves at constant enthalpy toward the entering air wet-bulb temperature at position 2. The air does not reach position 2, however, because the evaporative cooler is not 100% efficient; the process stops at position 3. Position 3 is the dry-bulb temperature of the air leaving the evaporative cooler.

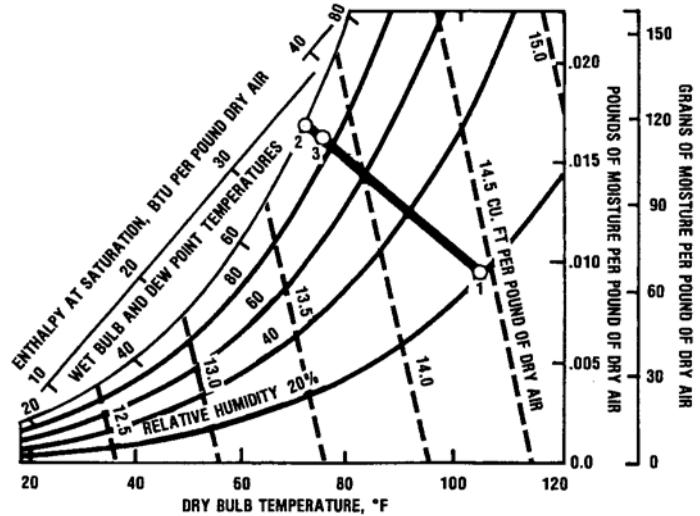


FIG. 5 PSYCHROMETRIC CHART C  
(BAROMETRIC PRESSURE 29.92 INCHES OF MERCURY)

## NONADIABATIC COOLING

Two conditions describe nonadiabatic cooling. The first condition occurs when the evaporative cooler water is delivered to the header at a temperature above the wet-bulb temperature of the air, but below the entering air dry-bulb temperature. In this case, the incoming air dry-bulb temperature enters at position 1 (Fig. 6) and moves toward position 3 at increasing enthalpy and wet-bulb temperature, as the air absorbs the extra heat from the water. Again, position 2 is the air wet-bulb temperature and line 1-2 represents the direction of adiabatic cooling, which does not occur. Position 3 cannot be determined theoretically because the final condition of the air after it has been cooled is indeterminate. Position 4 is the final temperature of the air, which, as in the adiabatic process, is above the saturation temperature because the evaporative cooler is less than 100% efficient. Position 4 is the dry-bulb temperature of the air leaving the evaporative cooler.

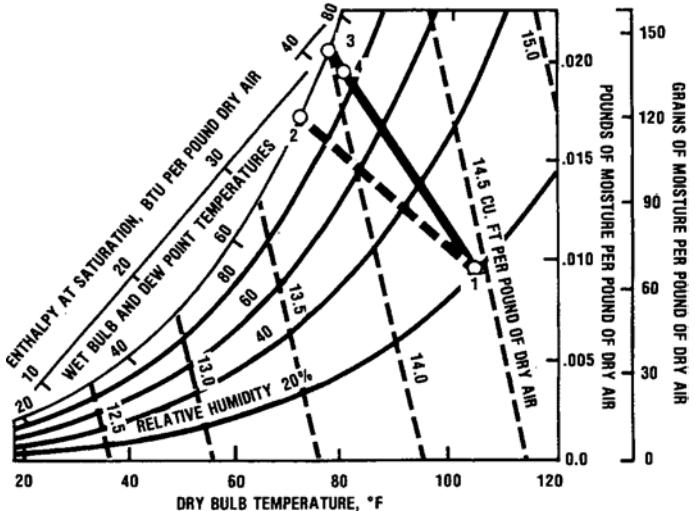


FIG. 6 PSYCHROMETRIC CHART D  
(BAROMETRIC PRESSURE 29.92 INCHES OF MERCURY)

The second nonadiabatic cooling condition occurs when the water delivered to the header is above the entering air dry-bulb temperature, as well as above the wet-bulb temperature. The incoming air dry-bulb temperature (Fig. 7) enters at position 1 and moves to position 2 as the sensible heat transfer from the water to the air increases the air dry-bulb temperature. The air then moves toward position 4 with increasing enthalpy and wet-bulb temperature, as noted earlier. Position 3 is the air wet-bulb temperature, and line 2-3 is the path of adiabatic cooling, which does not occur. Position 5 is the final, dry-bulb temperature of the air leaving the evaporative cooler.

Calculating the evaporative cooler's exiting dry bulb temperature cannot be done solely by means of Eq. (1) in a nonadiabatic condition. This is because the process of cooling the air has not been done at constant enthalpy. Some of the enthalpy of the incoming water has produced sensible heat gain in the air. Although the effect of this sensible heat gain cannot be evaluated theoretically, it is possible to estimate its effect. This procedure assumes that the temperature difference between the incoming water and the air wet-bulb temperature represents the available thermal energy given up by the water to the air, as sensible heat, as the water is cooled to the air wet-bulb temperature. However, this is not strictly true because some water will be evaporated and will pass latent heat into the air, which slightly raises the air wet-bulb temperature. But, as with the final dry-bulb temperature, the final wet-bulb temperature is also indeterminate. Therefore, the procedure predicts a dry-bulb temperature ( $T_{DB2}$ ) that is slightly higher than the actual temperature, if other heat does not enter the cooler. This procedure is slightly conservative and prevents overestimating the effect of the evaporative cooler. The procedure is detailed in the following example:

#### Example

##### Assumptions:

1.  $T_W = 100^\circ\text{F}$ , incoming water temperature
2.  $T_{DB1} = 105^\circ\text{F}$ , incoming air dry-bulb temperature

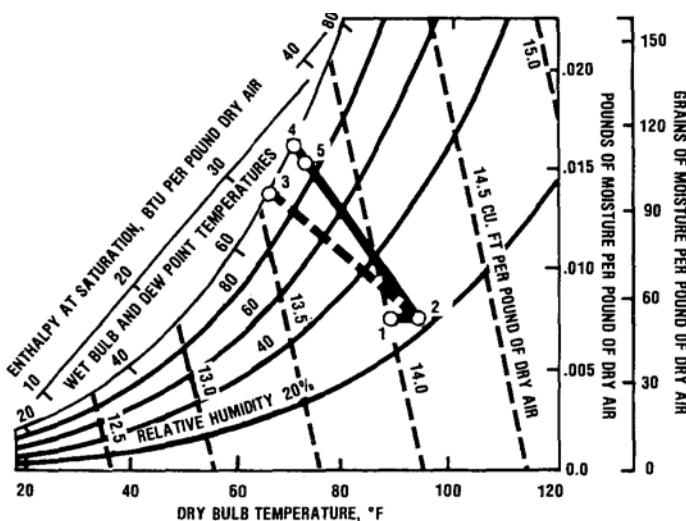


FIG. 7 PSYCHROMETRIC CHART E  
(BAROMETRIC PRESSURE 29.92 INCHES OF MERCURY)

$$3. T_{WB} = 72^\circ\text{F}, \text{air wet-bulb temperature}$$

$$4. \dot{m}_a = 126,558 \text{ lb/hr}, \text{air mass flow rate}$$

$$5. \dot{m}_w = 5010 \text{ lb/hr}, \text{water mass flow rate}$$

First, using Eq. (1), calculate the exiting air dry-bulb temperature from the evaporative cooler, assuming adiabatic conditions:

$$T_{DB2} = 105^\circ\text{F} - 0.9(105^\circ\text{F} - 72^\circ\text{F}) = 75^\circ\text{F} \quad (6)$$

Now, using Eq. (6), calculate the available heat from the water:

$$\begin{aligned} \dot{Q} &= \dot{m}_w c_p \Delta T \\ &= (5010 \text{ lb/hr})(1 \text{ Btu/lb} \cdot ^\circ\text{F})(100^\circ\text{F} - 72^\circ\text{F}) \\ &= 140,280 \text{ Btu/hr} \end{aligned}$$

Finally, calculate the final dry-bulb temperature of the air using Eq. (7):

$$\begin{aligned} T_{DB2} &= T_{DB2} + \frac{\dot{Q}}{\dot{m}_a c_p} \\ &= 75^\circ\text{F} + \frac{140,280 \text{ Btu/hr}}{(126,558 \text{ lb/hr})(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})} \\ &= 80^\circ\text{F} \end{aligned} \quad (7)$$

This is the air temperature used to predict the gas turbine's theoretical performance. Remember, the correction shown in this example is used only for nonrecirculating evaporative coolers. For equivalent recirculating evaporative coolers adiabatic conditions can be assumed, even when using make-up water at temperatures above the incoming air wet-bulb temperature.

This 5°F temperature difference can have a significant impact on turbine performance. With a recirculating cooler and the conditions used for the example, the available shaft horsepower (neglecting inlet and exhaust losses) at sea level from a 3481-hp compressor set (80°F matched engine) without evaporative cooling is 4052 hp with a recirculating cooler and 4007 hp with a nonrecirculating cooler, which represents gains of 16.4 and 15.1%, respectively. Obviously, the recirculating evaporative cooler is recommended for installations using water at elevated temperatures. However, for best results, the water temperature should be close to, or below, the ambient air wet bulb temperature.

#### DESIGN

Basically, evaporative cooler design involves the packaging of the media in an appropriate housing and the incorporation of several common features which prevent (or minimize) water carry-over in the air leaving the cooler and reduce maintenance. A typical evaporative cooler is shown in Fig. 8.

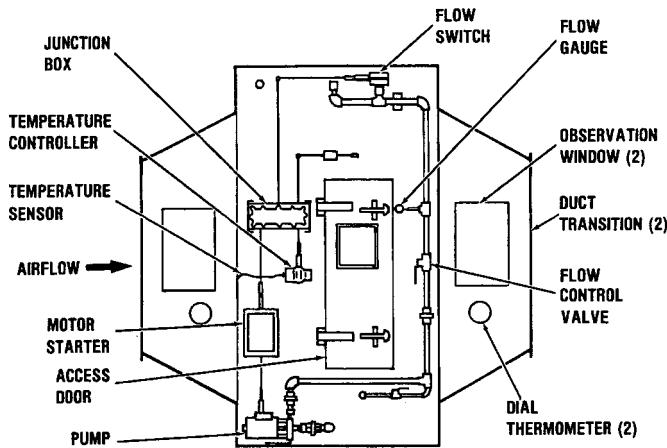


FIG. 8 TYPICAL EVAPORATIVE COOLER

### Common Features

1. **Header** - Delivers the water to the distribution pad at the top of the cooler. Should be located toward the front (air-entering side) of the cooler (Fig. 1), which delivers more water to the front of the cooler than a header located at the center of the pad.
2. **Media Retainers** - Must be gasketed. These are surfaces on the downstream side of the media which secure the media pack in the housing, shown as surfaces "A" in Fig. 9. A closed cell, neoprene material, 1/16-inch thick, is recommended for the gaskets. The media must be in contact with the gaskets. This prevents water carry-over in the leaving air caused by water leaking between the media and the retaining surfaces.
3. **Vane-Type Mist Eliminator** (Fig. 10) - Used on the cooler's downstream side. This is intended to remove air-entrained water droplets, if necessary. The mist eliminators should have a minimum droplet cut-off size of 100 micron, which means the mist eliminator has a removal efficiency of 99.9% on droplets of 100 micron and larger.

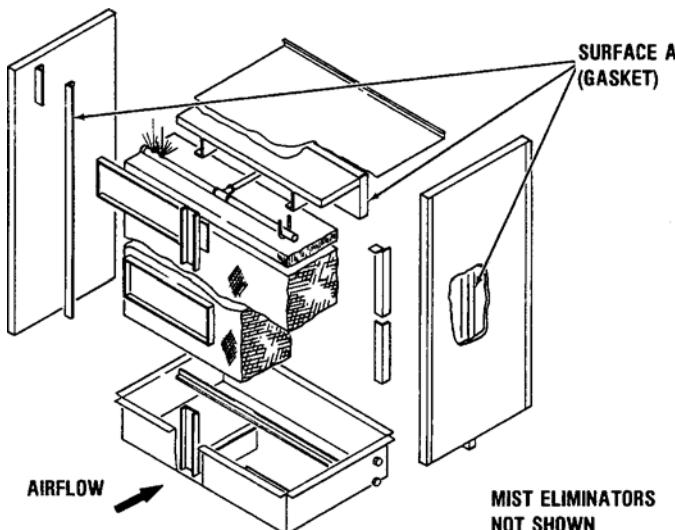


FIG. 9 EVAPORATIVE COOLER EXPLODED VIEW

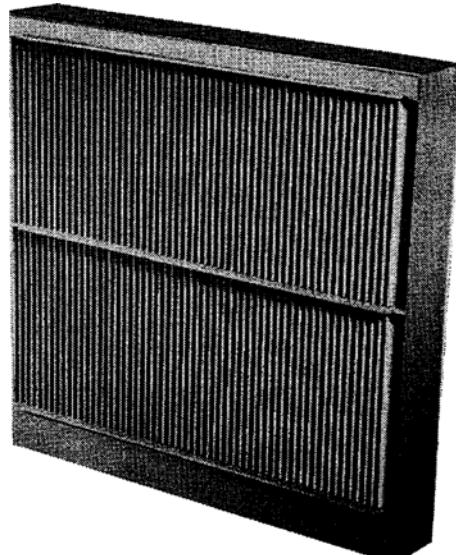


FIG. 10 VANE-TYPE MIST ELIMINATOR

4. **Observation Window** - Installed in the cooler housing sidewall between the media and the mist eliminators, to allow observation of the air-leaving side of the media. Large amounts of water leaving the media indicate an upset condition which should be investigated. Duct sections immediately upstream and downstream of the cooler should have observation windows. This allows examination of the upstream side for dry spots and heavy scale formation on the media.
5. **Thermometers** - Placed in the ductwork upstream and downstream of the cooler to monitor the effectiveness of the cooler.
6. **Temperature Sensor** - Measures incoming dry-bulb temperature. Should be in the airstream on the air inlet side of the cooler. Recommended setting is 55°F, below which temperature the cooler is shut off.
7. **Water Flow Gauge and Valve** - Used to monitor and adjust water flow to the header.
8. **Movable Wall in the Cooler Housing** - Considerably reduces the difficulty of installing or removing the media. This feature is highly beneficial. Without it, installing media in the field is difficult and often leads to damaged media. The wall is backed off the media for media removal and pushes against the media after the media has been reinstalled, compressing the media strips tightly together. Such compression is necessary to prevent water leakage between the strips, which would lead to water entrainment.
9. **Water Flow Monitor** - Should produce an alarm signal to notify the operator if water flow to the cooler is interrupted.
10. **Recirculating Pump** - Installed outside the cooler. This makes it easier to service the pump and motor while the turbine is operating.

11. Media Face Velocity - 500 feet/minute. Higher face velocities of up to about 625 feet/minute can be used, but lower face velocities greatly reduce the risk of water entrainment from the media.

These features, except for the thermometers, are standard with the evaporative coolers used by the author's Company. Thermometers are a convenience preferred by many operators and are supplied when specified by the user. Media retainer gaskets are not included sometimes, if it is determined the cooler will function satisfactorily without them. However, the vertical retainers on the side wall of the cooler housing are mandatory.

## WATER FLOW RATES

Evaporative cooler manufacturers recommend water flow rates to the evaporative cooler header ranging from 1 to 2 gal./min. for each square foot of surface area of the distribution pad (Fig. 1) for coolers up to 6-feet high, and 1-1/4 to 3 gal./min. for coolers up to 12-feet high. A flow rate of 1 gal./min./ft<sup>2</sup> of distribution pad surface area for evaporative coolers up to 6-feet high has been shown to work satisfactorily. Accordingly, for evaporative coolers 7 to 12-feet high, 1-1/4 gal./min./ft<sup>2</sup> of distribution pad surface area is recommended. Higher flow rates have the advantage of minimizing the rate of scale formation on the media surface, but lower flow rates minimize the potential for water carry-over. (Water carry-over is discussed in the next section.) These recommended flow rates are considered a practical result of the consideration of these two effects.

According to a major manufacturer of the media used in rigid media evaporative coolers, these flow rates could be reduced as much as 20% and still retain the full effectiveness of the evaporative cooler. However, lower flow rates are not recommended because the 20% margin is a useful safety factor.

A water flow rate that is too low will be indicated by dry spots on the media's upstream surface. If dry spots occur, the water flow rate to the header should be gradually increased. If the dry spots do not disappear after 15 to 20 minutes, the water flow rate should be increased further. This procedure should be followed until the dry spots are gone.

Although scale will form on any evaporative cooler media over time, heavy scale formation on the front of the media after several weeks or a few months of operation may also indicate a water flow rate which is too low. As with dry spots, the water flow rate should be increased. In both cases, observe the downstream side of the media to look for water carry-over. The mist eliminators will remove entrained water, but water blowing from the media should be minimized, nevertheless. At the first opportunity, the media should be removed from the cooler and washed down or cleaned to remove the scale.

The water flow rate to the header should be set at the time the evaporative cooler becomes operational. If the flow rate is initially set correctly, it should not be necessary to adjust it later. The water flow rate should not be adjusted because of changes in the ambient temperature. Unnecessary water flow rate adjustments, especially those made because of ambient temperature changes, are a primary cause of evaporative cooler problems associated with too much - or too little - water.

## WATER CARRY-OVER

Water carry-over is the presence of water droplets in the air leaving the evaporative cooler. Water carry-over will be negligible if the cooler is properly designed, sized, installed, operated, and

maintained. However, water carry-over seems to be affected by a large number of parameters and, because these conditions can (and seem to) appear in most evaporative cooler installations, it seems prudent to accept the inevitability of water carry-over in evaporative coolers which do not use vane-type mist eliminators.

Accordingly, in order to eliminate water carry-over, the author's Company requires the use of vane-type mist eliminators for all evaporative coolers supplied with its gas turbines. The mist eliminator is installed on the downstream side of the media. Even though a mist eliminator is installed after the media, spray leaving the media should still be investigated and reduced as much as possible because the mist eliminator efficiency is less than 100%. This means that some water will pass through the mist eliminator. The greater the water volume impinging on the mist eliminator, the greater the water volume entrained in the turbine's inlet air. The vane-type mist eliminator is extremely effective in removing water droplets, but reducing airborne water droplets to the maximum extent possible upstream of the mist eliminator is advisable and will ensure that water carry-over does not occur.

The ten causes of water carry-over discussed below apply to water carried over from the air-leaving side of the media. These causes are based on field investigation of water carry-over from the evaporative cooler media:

1. Incorrect Media Polarity (Fig. 11) - Media may be installed incorrectly by the manufacturer or removed at some point before the evaporative cooler's installation and then replaced incorrectly.
2. Damaged Media - Damage may occur when the media is being reinserted into the evaporative cooler after it has been removed in the field. Damage usually occurs at the edges of the media when the media strips are being forced into position. This can prevent proper alignment of the media strips and can be the cause of open cracks between the media (See Item 3). Damage also occurs when the media strips are placed on the ground and then stepped upon. This crushes the media, and water carry-over often occurs at the point of damage.

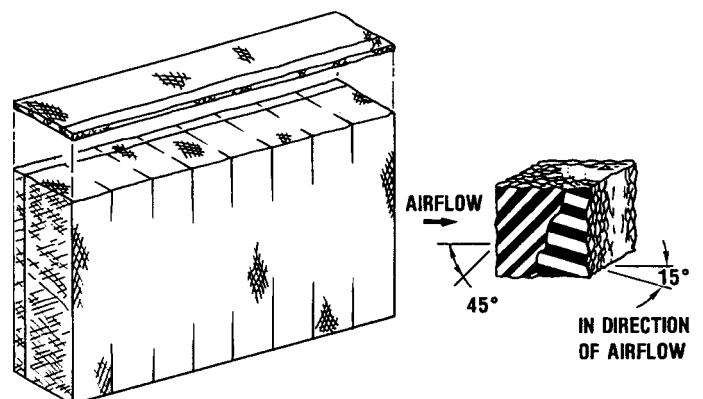


FIG. 11 MEDIA POLARITY

3. **Media Strips Improperly Aligned** - The vertically installed media strips may not be flush with each other on the air-leaving side. Excessive misalignment, 3/4 inch or more, can allow water carry-over. In addition, there should be no open cracks between the media strips.
4. **Media Not Sealed against Retainers** - Media must be firmly sealed against the media retainers which hold the top and sides of the media on the downstream side of the cooler.
5. **Excessive Water Flow** - Too much water delivered to the header will flood the media and cause water carry-over. The water flow rate should be no more than needed to wet all media internal surfaces without flooding them.
6. **Uneven Water Distribution from the Header** - Holes in the water header can become clogged, causing uneven water distribution to the media pad, which means one side of the cooler receives too much water.
7. **Incorrectly Installed Baffle Sheet** - If the evaporative cooler uses a rubber baffle sheet, this sheet should be installed on top of the media pad.
8. **Uneven Airflow through the Media** - Most likely the result of improper ducting design. The inlet ducting must be designed so that the airflow distribution through the media is as even as possible.
9. **Scale Deposits on the Media** - If the evaporative cooler has been in service for a long time, it may have scale deposits on the media surface, causing water entrainment and necessitating media replacement.
10. **Unsaturated Media** - Some water entrainment may be caused by starting the turbine before the evaporative cooler media is saturated. Normally, the water carry-over will be insignificant because the period required to saturate the media lasts for only a few minutes.

## WATER TREATMENT

Field experience from the author's Company has demonstrated the necessity of setting a quality specification for water used with evaporative coolers. This specification has established limits for turbidity, pH, hardness as  $\text{CaCO}_3$ , and the concentration of sodium (Na) plus potassium (K).

The specification imposes the following limits:

1. Turbidity	- 5000 turbidity units
2. pH	- 6-9
3. Hardness as $\text{CaCO}_3$	- 160 ppm
4. Na + K	- (See below)

The Na + K concentration limit is based on the assumption that water carry-over from the evaporative cooler without the mist eliminator is not more than 10% of the water flow rate delivered to the header. Concentrations of several hundred ppm (up to about 1,000 ppm) Na + K can be tolerated if mist eliminators having a removal efficiency of at least 99.9% on water droplets larger than 100 microns are used.

The 160 ppm for hardness as  $\text{CaCO}_3$  is a recommended maximum, generally used as a rule of thumb. Water having higher concentrations can be used. However, when the water is saturated with  $\text{CaCO}_3$ , scaling occurs on the evaporative cooler media. In addition to calcium hardness, scaling is also a function of the total dissolved solids, pH, and water temperature. A more accurate determination of the  $\text{CaCO}_3$  saturation point can be found using the Langelier Saturation Index, Ryznar Stability Index, or the Puckorius Scaling Index.

Typically, water treatment recommendations should be obtained from a water treatment company or a water treatment consultant doing business in the area of the gas turbine installation.

At the author's Company, engineering personnel review each evaporative cooler installation, calculating the maximum anticipated carry-over of elemental contaminants in terms of a Total Fuel Equivalent using Eq. (8). (For the derivation and application of this equation, see Ref. 3.)

$$T = (18,380/LHV)$$

$$[F_i + (AFR)A_i + (WFR)W_i + (SFR)S_i + (CFR)C_i] \quad (8)$$

where:

i = Na, K, V, Pb, etc.

The calculated total fuel equivalent cannot exceed the allowable limit for the gas turbine.

Proper control and treatment of feedwater is necessary to maintain good evaporative cooler performance. If a recirculating evaporative cooler is used, this introduces the added requirement for setting an appropriate blowdown rate and selecting suitable feedwater treatment to prevent scale formation on the media and pH changes of the water.

Scale forms from the precipitation and deposition of dissolved solids, thereby decreasing media performance and shortening its life. Scale formation in the holes of the header reduces water flow and also decreases performance. If the hardness of untreated feedwater exceeds 160 ppm of  $\text{CaCO}_3$ , treatment is recommended because blowdown alone will not prevent scale formation. The ratio of blowdown to evaporation is 5.7 to 1 at 160 ppm, but this ratio becomes excessively high above 160 ppm. Below 135 ppm, feedwater can be used without treatment. Between 135 and 160 ppm, feedwater is usually not treated, but water availability, water costs, and cost of treatment are factors in deciding whether to treat the water.

Even though feedwater treatment due to hardness may be unnecessary, treatment may still be required because of pH changes caused by evaporation and aeration of the water in the reservoir of a recirculating evaporative cooler. Aeration, for example, removes dissolved  $\text{CO}_2$  and, thereby, increases the alkalinity, which aids the precipitation of  $\text{CaCO}_3$ .

Deionized water can be used with the evaporative cooler, but it reacts with the stiffening agents in the media, softening the media to the point of collapse. It is also very corrosive to unprotected metal parts of the cooler. If deionized water is used, galvanized materials should not be used. The cooler housing and water piping should be constructed of 304 stainless steel, and the media should be specially constructed with increased stiffening agents.

## INSTALLATION

Generally, in a gasturbine inlet air system, it is recommended the evaporative cooler be placed after the inlet air cleaner, not before it. This arrangement will protect the media from the dust and other airborne contaminants that would otherwise impinge upon it. Rigid media evaporative coolers have not been tested for filtration efficiency, but it is known that they can remove substantial amounts of airborne particulate. The effect this particulate removal has on the evaporative cooler's operation will depend on a number of factors, including the average airborne dust concentration, the maximum anticipated airborne dust concentration, and whether the evaporative cooler is recirculating or nonrecirculating.

### Recirculating Evaporative Coolers

Light dust concentrations (less than about 1 ppm in air) will not create operational problems for the evaporative cooler. As a rule of thumb, though, the evaporative cooler should follow an air cleaner, if the average dust concentration exceeds 1 ppm. Higher concentrations, especially in dust and sandstorms where concentrations can easily exceed 500-600 ppm, can cause heavy silt in the reservoir and result in damage to pumps and in reduced recirculating water flow.

Water flow rates higher than those recommended for normal operation are needed to wash the dirt from the evaporative cooler media. These higher flow rates greatly increase the risk of water entrainment from the media and of greater water loading on the mist eliminator, which, in turn, increases the risk of water carryover from the evaporative cooler.

Organic growths also become a problem for recirculating evaporative coolers exposed to sunlight. Growths form on the media and in the reservoir, necessitating chemical treatments to control them.

### Nonrecirculating Evaporative Coolers

The nonrecirculating evaporative cooler does not have the silt problems of the recirculating cooler, but it is subject to the same dust loading problems. The higher water flow rates and subsequent risk of water carry-over that apply to the recirculating cooler also apply to the nonrecirculating cooler. Organic growth can also occur on the media.

## ACKNOWLEDGMENTS

The author wishes to acknowledge John W. Adler, The Munters Corporation; Harvey von E. Doering, Consultant; Francis R. Story, P.E., Pneumafil Corporation; and Dr. John R. Watt, P.E., Consultant, for their valuable contributions to the quality and accuracy of this paper.

## REFERENCES

1. ASHRAE Handbook, Chapter 36, 1980 System Volume, American Society of Heating, Refrigerating and Air-Conditioning Engineers, New York, NY.
2. Gonzales, D.J. and Short, D.T., "Some Environmental Control Techniques Affecting Gas-Turbine Performance," ASME Paper No. 61-WA/GTP-10, September 1964.
3. Hsu, L.L., "Total Corrosion Control for Industrial Gas Turbines: Airborne Contaminants and Their Impact on Air/Fuel/Water Management," ASME, 1988.
4. Puckorius, Paul, "Get a better reading on scaling tendency of cooling water," Power, September 1983, pp 79-81.
5. Watt, John R., P. E. , Evaporative Air Conditioning Handbook, 2nd ed., Chapman & Hall, 1986.