Effective Thermal Design Of Cooling Towers

A step-by-step approach to cooling-tower design, with an example calculation to make it clear

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Various misconceptions arise when it comes to the thermal design of cooling towers. Sometimes related parameters, such as range, approach, effectiveness, liquid-to-gas ratio (*L/G*), wet-bulb temperature, cooling water temperature, relative humidity, number of transfer units (NTU) and other terms create a confusion for the designer in effectively sizing, selecting and evaluating a particular cooling tower. This leads to inadequate design.

The objective of this article is to present a stepwise understanding of how to calculate the NTU for a cooling tower, and thus to understand the basis of thermal design of counter-flow cooling towers for optimizing cost and performance.

Definitions

First, let's look at some of the basic terms and briefly describe their significance and role in cooling tower design and performance.

Dry-bulb temperature. Dry-bulb temperature (t_{db}) — usually referred to as the air temperature — is the property of air that is most commonly used. When people refer to the temperature of the air, they are normally referring to its dry-bulb temperature. The dry-bulb temperature is an indicator of heat content and is shown along the bottom axis of a psychometric chart. The vertical lines extending upward from this axis are constant-temperature lines.

Wet-bulb temperature. Wet-bulb temperature (t_{wb}) is the reading when the bulb of a thermometer is covered with a wet cloth, and the instrument is whirled around in a sling. The wet-bulb temperature is the lowest temperature that can be reached by evaporation of water only.

Relative humidity (RH). RH is the ratio of the partial pressure of water vapor in air over the saturation vapor pressure at a given temperature. When the relative humidity is 100%, the air is saturated and therefore, water will not evaporate further. Therefore, when the RH is 100% the wet-bulb temperature is the same as the dry-bulb temperature, because the water cannot evaporate any more.

Range. The range is the difference in temperature of inlet hot water (t_2) and outlet cold water (t_1), $t_2 – t_1$. A high cooling-tower range means that the cooling tower has been able to reduce the water temperature effectively.

Approach. The approach is the difference in temperature of outlet cold water and ambient wet-bulb temperature, $t_1 – t_w$. The lower the approach, the better the cooling tower performance. Although both range and approach should be monitored, the approach is a better indicator of cooling tower performance.

Cooling tower capability. The capability of the cooling tower is a measure of how close the tower can bring the hot water temperature to the wet-bulb temperature of the entering air. A larger cooling tower (that is, more air or more fill) will produce a closer approach (colder outlet water) for a given heat load, flowrate and entering air condition. The lower the wet-bulb temperature, which indicates either cool air, low humidity or a combination of the two, the lower the cooling tower can cool the water. Capability tests are conducted per the ATC-105 Code of the Cooling Tower Institute (CTI; Houston; www.cti.org).

The thermal performance of the cooling tower is thus affected by the entering wet-bulb temperature. The entering air dry-bulb temperature has an insignificant effect on thermal performance.

Effectiveness. A cooling towers effectiveness is quantified by the ratio of the actual range to the ideal range, that is, the difference between cooling water inlet temperature and ambient wet-bulb temperature. It is defined in terms of percentage.

Nomenclature

t_2 Hot water temperature, °C

t_1 Cold water temperature, °C

t_{wb} Wet-bulb temperature, °C

t_{db} Dry-bulb temperature, °C

t_d Dew point temperature, °C

h_a Enthalpy of moist air, kJ/kg

h_1 Enthalpy of inlet air, kJ/kg

h_2 Enthalpy of exit air, kJ/kg

h' Enthalpy of fin, kJ/kg

F Flowrate, m^3/h

L Mass flowrate of liquid, lb/h

G Mass flowrate of gas, lb/h

Q Heat load, kcal/h

Z Altitude above sea level, m

p Barometric pressure, kPa

p_{ws} Saturation pressure of water vapor, kPa

p_w Partial pressure of water vapor, kPa

v Specific volume, m^3/kg

W Humidity ratio, kg water/kg air

W_s Humidity ratio at saturation air, kg water/kg moist air

ⓢ Relative humidity (RH), %

C Constant related to cooling tower design

m Slope of tower characteristic curve

$$\text{Effectiveness} = \frac{\text{Range}}{\text{Range} + \text{Approach}} \times 100\% \qquad (1)$$

Liquid-to-gas ratio (L/G). The L/G ratio of a cooling tower is the ratio of the liquid (water) mass flowrate (*L*) to gas (air) mass flowrate (*G*). Cooling towers have certain design values, but seasonal variations require adjustment and tuning of water and air flowrates to get the best cooling tower effectiveness.

Number of transfer units (NTU). Also called the tower coefficient, the NTU is a numerical value that results from theoretical calculations based on a set of performance characteristics. The value of NTU is also representative of the degree of difficulty for the cooling process. The NTU corresponding to a set of hypothetical conditions is called the *required coefficient* and is an evaluation of the problem. The same calculations applied to a set of test conditions is called the *available coefficient* of the tower involved. The available coefficient is not a constant but varies with operating conditions. The operating characteristic of a cooling tower is developed from an empirical correlation that shows how the available coefficient varies with operating conditions.

Cooling capacity. The cooling capacity of a tower is the heat rejected [kcal/h or TR (refrigeration tons; 1 TR = 12,000 Btu/h = 3,025.9 kcal/h)], and is determined by the product of mass flowrate of water, times the specific heat times the temperature difference.

Theory — the Merkel equation

In a cooling tower operating in counter current flow, there are two basic principles involved for removing heat by the cooling water:

1.sensible heat transfer due to a difference in temperature levels

2. latent heat equivalent of the mass transfer resulting from the evaporation of a portion of the circulating water

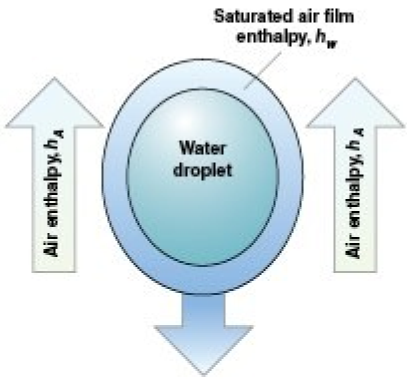

Figure 1. The Merkel equation is derived by considering a falling water droplet surrounded by saturated air

Merkel developed the basic equation based on the above principles. The Merkel model is universally accepted for designing and rating of counter-flow cooling towers. The model is based on a drop of water falling through an upstream flow of unsaturated air at a wet-bulb temperature of t_{wb} with enthalpy h_A (Figure 1), in a counter-flow cooling tower. The drop of water is assumed to be surrounded by a film of saturated air at the water temperature *WT* with saturation enthalpy h_W. As the drop travels downward, heat and mass transfer takes place from the interface air film to the upstream air, thereby cooling the water from hot temperature to a cold temperature.

The main assumptions of Merkel theory are the following:

1. The saturated air film is at the temperature of the bulk water.

2. The saturated air film offers no resistance to heat transfer.

3. The vapor content of the air is proportional to the partial pressure of the water vapor.

4. The heat transferred from the air to the film by convection is proportional to the heat transferred from the film to the ambient air by evaporation.

5. The specific heat of the air-water vapor mixture and the heat of vaporization are constant.

6. The loss of water by evaporation is neglected.

7. The force driving heat transfer is the differential enthalpy between the saturated and bulk air.

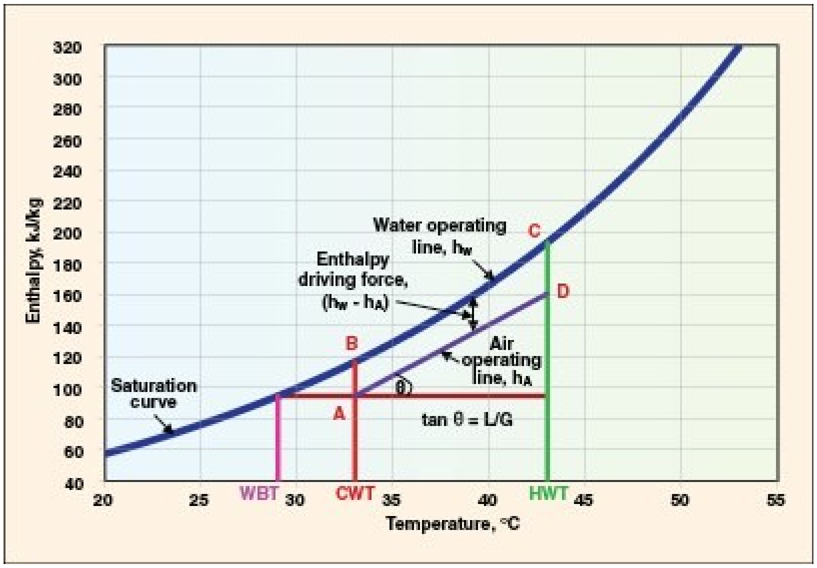

Figure 2. This plot, known as the driving force diagram, shows the enthalpy versus temperature for water and air

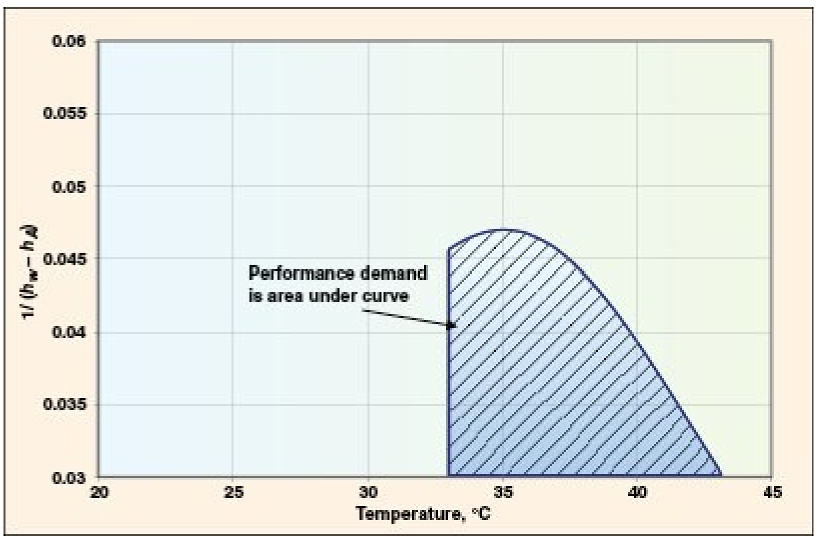

Figure 3. Solving the Merkel equation (Equation 2), is usually done graphically, where the integral is equal to the area under the curve

This cooling process can best be explained on a psychometric chart, which plots enthalpy versus temperature. The process is illustrated in the so-called driving-force diagram shown in Figure 2. The air film is represented by the water operating line on the saturation curve. The main air is represented by the air operating line, the slope of which is the ratio of liquid (water) to air (*L/G*). The cooling characteristic, a degree of difficulty to cooling is represented by the Merkel equation:

$$KaV / L = \int_{CWT}^{HWT} \frac{dT}{h_W - h_A} \qquad (\mathbf{2})$$

Where:

K = overall enthalpy transfer coefficient, lb/h-ft^2

a = Surface area per unit tower volume, ft^2/ft^3

V = Effective tower volume, ft^3

L = Water mass flowrate, lb/h

Equation 2 basically says that at any point in the tower, heat and water vapor are transferred into the air due (approximately) to the difference in the enthalpy of the air at the surface of the water and the main stream of the air. Thus, the driving force at any point is the vertical distance between the two operating lines. And therefore, the performance demanded from the cooling tower is the inverse of this difference. The solution of the Merkel equation can be represented by the performance demand diagram shown in Figure 3. The *KaV/L* value is equal to the area under the curve, and represents the sum of NTUs defined for a cooling tower range.

An increase in the entering t_{wb} moves the air operating line towards the right and upward to establish equilibrium. Both the cold water temperature (*CWT*) and hot water temperature (*HWT*) increases, while the approach decreases. The curvature of the saturation line is such that the approach decreases at a progressively slower rate as the t_{wb} increases. An increase in the heat load increases the cooling ranges and increases the length of the air operating line. To maintain equilibrium, the line shifts to the right increasing the*HWT*, *CWT*, and approach. The increase causes the hot water temperature to increase considerably faster than does the cold water temperature. In both these cases, the *KaV/L* should remain constant. However, a change in *L/G* will change the *KaV/L* value.

Cooling tower design

On the basis of the above discussion, it is clear that there are five parameters that, in combination, dictate and define the performance of a cooling tower, namely:

1. Hot water temperature, *HWT*

2. Cold water temperature, *CWT*

3. Wet bulb temperature, t_{wb}

4. Water mass flowrate, *L*

5. Air mass flowrate, *G*

The first four parameters are determined by the user of the cooling tower. It is the fifth quantity, *G*, that is selected by the designer of the cooling tower. Once these five quantities are available, the tower characteristic (*KaV/L*), can be calculated through the Merkel equation.

The first step in designing a cooling tower is the generation of a demand curve. In this curve, the *KaV/L*values are plotted against varying *L/G* ratios. The next step is to superimpose fill-characteristic curves and demand curves. The Cooling Technology Institute has tested a variety of fill configurations and generated fill characteristic curves for each type; CTI's Technical Paper TP88_05 can be referred to in this regard.

Cooling tower design is basically an iterative process. The factors that effect the selection of design *L/G* and consequently the fill height are: cell dimensions, water loading, air velocities across various cooling tower sections and pressure drops, and fan selection.

The classical method of thermal rating of cooling towers is to estimate the ratio of liquid to gas first and then find the proper tower volume by the means of trial and error using the tower performance curve. The *L/G* is the most important factor in designing the cooling tower and related to the construction and operating cost of cooling tower.

Finally we can summarize the importance of the *L/G* ratio with the following points.

A high *L/G* ratio means:

• More water to less air

• Air is more saturated — driving force is reduced

• More residence time of water needed

• Less cooling in given time

• Increase in required fan power

• Decrease in height of tower

• Low evaporation loss (under same water flowrate)

An example makes it clear

As an example, let us design a cooling tower with the following data:

Capacity (*F*): 3,000 m^3/h

Wet bulb temperature (t_{wb}): 29°C Relative humidity (s) 92%

Cooling water inlet (t_2): 43°C

Cooling water outlet (t_1): 33°C Altitude (*Z*): 10 m

Step I. This step involves heat load calculations as follows:

1. Range = ($t_2 - t_1$) = 43 – 33 = 10°C

2. Approach = $(t_1 - t_{wb})$ = 33 – 29 = 4°C

3. Heat load, $Q = mC_p(t_2 - t_1)$

= 998.13 x F x Range

= 998.13 x 3,000 x 10

= 29,943,900 kcal/h

Step II. This step involves total psychometric calculations as follows:

1. Barometric pressure (p) at the given altitude (Z) is calculated by using the following equation:

$$p = 101.325(1 - 2.25577 \times 10^{-5} Z)^{5.2559} \quad (3)$$

For an altitute of 10 m, this becomes

p = 101.2 kPa

2. Assume a dry bulb temperature of say, t_{db} = 32°C

3. Calculate water vapor saturation pressure (p_{ws}) at the assumed t_{db} for the temperature range of 0 to 200°C using the equation:

$$p_{ws} = \frac{\exp\left[C_1T^{-1} + C_2 + C_3T + C_4T^2 + C_5T^3 + C_6\ln(T)\right]}{1{,}000} \quad (4)$$

Where:

$C_1 = -5.8002206 \times 10^3$

$C_2 = 1.3914993 \times 10^0$

$C_3 = -4.8640239 \times 10^{-2}$

$C_4 = 4.1764768 \times 10^{-5}$

$C_5 = -1.4452093 \times 10^{-8}$

$C_6 = 6.5459673 \times 10^0$

and T represents the dry bulb temperature in Kelvin. This results in the value:

p_{ws} = 4.7585 kPa

4. The partial pressure of water (p_w) at given relative humidity is found using the following equation:

$$p_w = p_{ws} \times \varphi \quad (5)$$

p_w = 4.3779 kPa

5. The partial pressure (p_{ws}) is again calculated using Equation 4. This time T represents the wet bulb temperature in Kelvin, which calculates to:

p_{ws} = 4.0083 kPa

6. Using p_{ws} calculated in Step 5 we recalculate t_{wb} using the Carrier equation:

$$p_w = p_{ws} - \left\{ \frac{1.8 \times (p - p_{ws}) \times (t_{db} - t_{wb})}{\left[2{,}800 - 1.3 \times (1.8 t_{db} + 32)\right]} \right\} \qquad \textbf{(6)}$$

which gives the result:

t_{wb} = 37.7°C

7. This step is an iterative process, whereby the assumed value of t_{db} in Step 2 is varied in such a way that the calculated t_{wb} in Step 6 equals the actual (real) t_{wb}.

8. After a number of iterations, the calculated t_{db} value converges to 30.12°C.

Step III. This step involves the calculation of the inlet air enthalpy (h_1) as follows:

1. The humidity ratio (W) for dry air is calculated using the following equation:

$$W = \frac{0.62198 \times p_w}{p - p_w} \qquad \textbf{(7)}$$

W = 0.02515 kg water/kg dry air

2. The specific volume (v) for dry air is calculated using the following equation:

$$v = \frac{0.2871 \times (t_{db} + 273.15) \times (1 + 1.6078W)}{p} \qquad \textbf{(8)}$$

v = 0.89511 m^3/kg, dry air

3. Calculate the enthalpy of inlet air (h_1) using the following equation:

$$h_1 = 1.006 t_{db} + W(2{,}501 + 1.86 t_{db}) \qquad \textbf{(9)}$$

h_1 = 94.750 kJ/kg

4. Calculate the humidity ratio at saturation (W_s) for wet air using same Equation 7. Here we now use p_{ws}:

W_s = 0.02743 kg water/kg moist air

5. Calculate the specific volume (v) for wet air using Equation 8 with W_s.

v = 0.89827 m^3/kg moist air

Step IV.This step involves the calculation of the exit air properties, as follows:

1. Assume some value of the *L/G* ratio, say 1.575, and calculate h_2 for exit air using the following equation:

$$h_2 = h_1 + \left[(L/G) \times Range \times 4.186\right] \tag{10}$$

h_2 = 160.50 kJ/kg

2. Assume that the exit air has a relative humidity of 97–99% (design RH at the outlet), and also assume some value of exit air dry-bulb temperature.

3. Use the same partial pressure and humidity equations as discussed in Step II and Step III, to calculate the enthalpy of exit air at these assumed values in Point 2 above. At an assumed RH of 98.5% and exit air t_{db} of 37°C, we recalculate

h_2 = 141.18 kJ/kg

4. This is again an iterative process. Next, assume the value of exit air t_{db} in Point 2 (at same relative humidity) in such a way that the calculated h_2 in Point 3 equals the h_2 calculated in Point 1.

5. After a number of iterations, the calculated exit air t_{db} value converge to 39.55°C.

6. Now that the dry-bulb temperature and RH are known values, recalculate the wet-bulb temperature using Equation 6.

t_{wb} = 39.31°C

7. Calculate the dry and wet specific volume of exit air using Equation 8. Also calculate the density of dry air and wet air (for inlet and exit).

v = 0.9540 m^3/kg, dry air

v = 0.9551 m^3/kg, moist air

Average density dry = 1.0827 kg/m^3

Average density wet = 1.0801 kg/m^3

Step V. This step is to help draw the driving force diagram as follows:

1. Take different temperature ranges (covering cooling water inlet and outlet temperature) and calculate the enthalpy of air using Equation 9 and psychometric calculations discussed above. Plot the air saturation curve (enthalpy versus temperature) as shown in Figure 2.

2. Take cooling water outlet temperature and calculate the enthalpy of fin (h') using Equation 9. Here the partial pressure, saturation pressure and humidity ratio are calculated for the corresponding temperature taken (Table 1).

h' at 33°C = 116.569 kJ/kg

This enthalpy specifies Point B on the graph of Figure 2, and is the starting point of the water operating line.

3. Similarly calculate enthalpy of air h_a, at wet-bulb temperature (Table 1).

h_a = 94.667 kJ/kg

This enthalpy specifies Point A on the graph of Figure 2 and is the starting point of the air operating line.

4. Take incremental change in temperature (say 0.5 or 1.0) up to the hot water temperature and calculate the h' and h_a. The ending points are shown as C and D in Figure 2 on the water operating and air operating line respectively.

5. The difference between h' and h_a will give you the enthalpy driving force for incremental change in temperature.

6. Take the inverse of enthalpy difference in each incremental step (Table 1).

7. Calculate NTU = 4.18 x Δt x (Average of incremental increase in inverse of enthalpy difference).

Or for 0.5°C increment in temperature, calculated NTU = 0.096

8. Similarly, calculate the NTU for each step and add to get the total NTU for the particular assumed *L/G* ratio (Table 1). Or, for an assumed *L/G* of 1.575, Total NTU = 1.7533 — this is *KaV/L*.

9. Now to plot the tower characteristic curve, first we vary the *L/G* ratio and repeat all calculations discussed above to generate the data for various NTU to plot. The curve represents “Design NTU” on the graph, shown in Figure 4.

10. Take the design *L/G* ratio and plot the tower characteristic curve by assuming the slope of the line (m), which usually varies between –0.5 to –0.8. One can also consult with vendors for this value as it also depends on the type of fins used.

11. Calculate the value of the constant C, related to cooling tower design using equation:

NTU = C x (L/G) ^{m}

C = 2.522

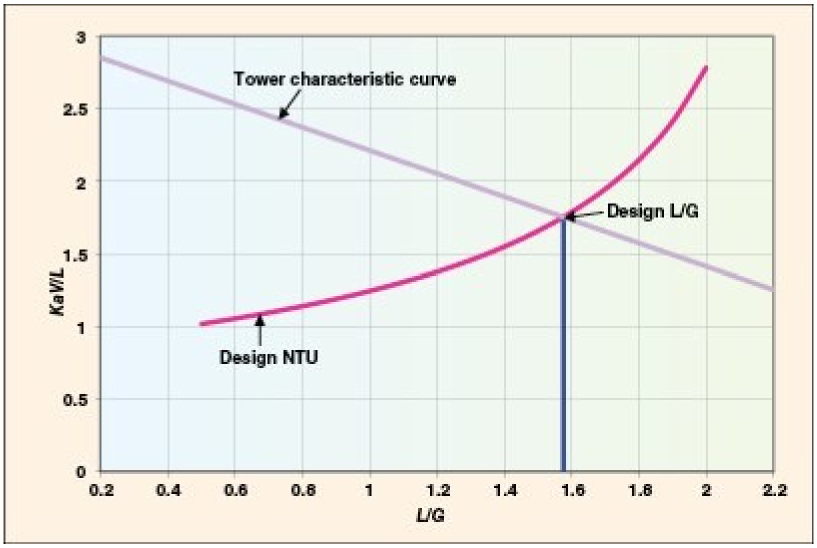

Figure 4. The intersection of the tower characteristic curve and the design NTU curve gives the design L/G ratio

Table 1.

Water Temperature, t		Water Vapor Saturation Pressure, pws
°C	K	kPa
33	306.15	5.0343
33.5	306.65	5.1774
34	307.15	5.3239
34.5	307.65	5.4740
35	308.15	5.6278
35.5	308.65	5.7853
36	309.15	5.9466
37	310.15	6.2810
38	311.15	6.6315
39	312.15	6.9987
40	313.15	7.3835
41	314.15	7.7863
42	315.15	8.2080
43	316.15	8.6492

Table 2. TYPICAL THERMAL CALCULATIONS OF COUNTER-FLOW COOLING TOWER

Flowrate

Wet-bulb temperature

Approach

Assumed dry-bulb temperature

Assumed L/G

No. of cells

Cell length

Cell width

Air inlet height

Design RH

Density of water

Altitude

Inlet Air Properties

Inlet t_{wb}

RH

Inlet t_{db} at above RH

Humidity ratio (W)

Specific volume (v)

Density

Humidity ratio at saturation (W_s)

Specific volume, (v) at saturation

Density

Enthalpy of moist air

Humidity Ratio at t_{wb}

Enthalpy at t_{wb}

Air Flow

Average density dry

Average density wet

Air flowrate at fin

Air flow at inlet, at rain zone

Air flowrate at fan

The significance of these calculations is that now we can directly calculate the cooling tower characteristic by using our equations and can compare with the vendor data if the provided height of the cell is adequate to meet the calculated NTU. Obviously the height also depends on the type of packing, but along with vendor input we can create a complete economical design of our cooling tower. Further, we can develop a calculation sheet in Microsoft Excel, which gives all the results of the psychometric chart as well as the cooling tower design. Typical thermal calculations for a counter-flow cooling tower can be seen in Table 2.

Edited by Gerald Ondrey

Reference:

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