Cooling Tower Technical Site of Daeil Aqua Co., Ltd. for
Cooling Tower Engineers, Operators and Purchasers

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Preface to Fifth Edition

This engineering book was prepared for educating the cooling tower engineers of a company in Taiwan. So, this deals with a very specific subjects, which were never written as of now. I prepared the documents so that the engineers can easily understand the cooling tower theory and can make the design program by themselves through the actual examples. All were run on MS Excel and you will see this new approach in computerizing the cooling tower thermal design. You can download all the examples through this homepage.

The major concern during I study the cooling tower theory was how to computerize the cooling tower theory from the calculation of NTU to the cooling tower performance analysis. If you read this book carefully, you can make any cooling tower design programs by yourself.

Again, this will be a first issue releasing the actual engineering approach of cooling tower with the examples in the world. If any questions on this issue, please send your mail to me.

1. Psychrometrics

Psychrometrics deals with thermodynamic properties of moist air and uses these properties to analyze conditions and process involving moist air. Atmospheric air contains many gases components as well as water vapor and miscellaneous contaminants (e.g., smoke, pollen and gaseous pollutants). Dry air exists when all water vapor and contaminants have been removed from atmospheric air. The composition of dry air is relatively constant, but small variations in the amounts of individual components occur with time, geographic location, and altitude. The apparent molecular mass or weighted average molecular weight of all components, for dry air is 28.9645, based on the carbon-12 scale. The gas constant for dry air, based on the carbon-12 scale is 1545.32/28.9645 =53.352 ft lbf / lbm ^oR.

Moist air is a binary mixture of dry air and water vapor. The amount of water vapor in moist air varies from zero (dry air) to a maximum that depends on temperature and pressure. The latter condition refers to saturation, a state of neutral equilibrium between moist air and the condensed water phase. Unless otherwise stated, saturation refers to a flat interface surface between the moist air and the condensed phase. The molecular weight of water is 18.01528 on the carbon-12 scale. The gas constant for water vapor is 1545.32/18.01528 = 85.778 ft lbf / lbm ^oR

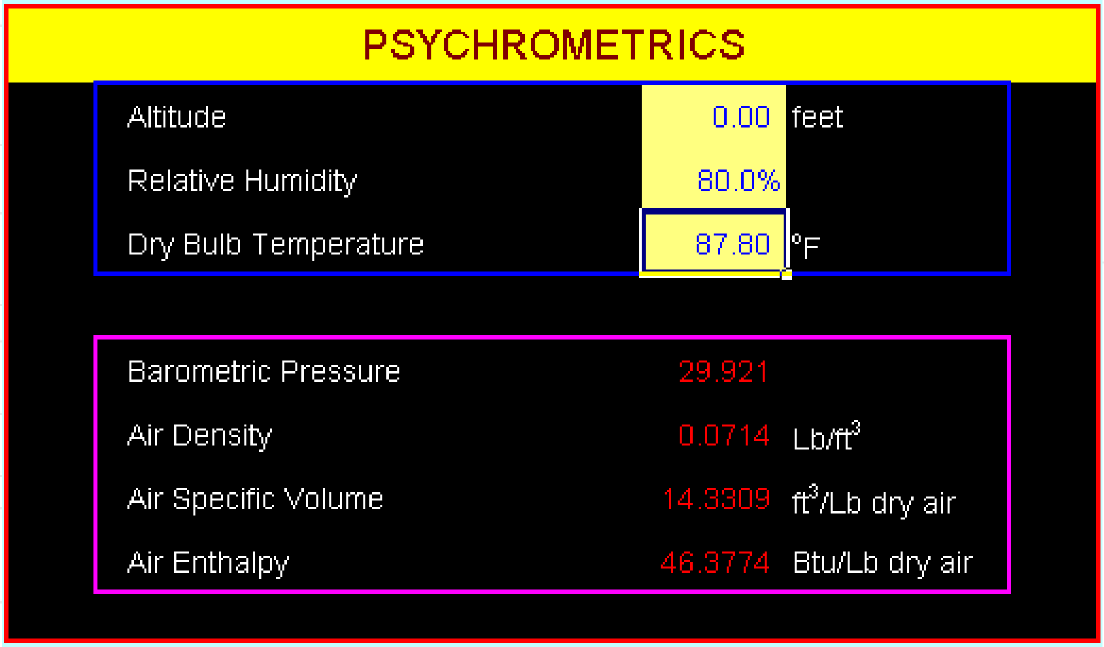
The temperature and barometric pressure of atmospheric air vary considerably with altitude as well as with local geographic and weather conditions. The standard atmospheric gives a standard of reference for estimating properties at various altitudes. At sea level, standard temperature is 59^{o}F; standard barometric pressure is 29.921 inch Hg. The temperature is assumed to decrease linearly with increasing altitude throughout the troposphere (lower atmosphere), and to be constant in the lower reaches of the stratosphere. The lower atmosphere is assumed to constant of dry air that behaves as a perfect gas. Gravity is also assumed constant at the standard value, 32.1740 ft/s^2.

Humidity ratio (alternatively, the moisture content or mixing ratio) is defined as the ratio of the mass of water vapor to the mass of dry air. Specific humidity is the ratio of the mass of water vapor to the total mass of the moist air. Absolute humidity (alternatively, water vapor density) is the ratio of the mass of water vapor to the total volume of the moist air. Saturation humidity ratio is the humidity ratio of moist air saturated with respect to water at the same temperature and pressure. Degree of saturation is the ratio of the air humidity ratio to humidity ratio of saturated air at the same temperature and pressure. Relative humidity is the ratio of the mole fraction of water vapor in a given moist air to the mole fraction in an air saturated at the same temperature and pressure.

The enthalpy of a mixture air is the sum of the individual partial enthalpies for dry air and for saturated water vapor at the temperature of the mixture.

Example 1-1. Calculate the air density, specific volume, and enthalpy in US units at the ambient conditions of DBT 87.8^{o}F, RH 80% and sea level.

- Air Density: 0.0714 Lb/ft^3
- Air Specific Volume: 14.3309 ft^3/Lb dry air
- Air Enthalpy: 46.3774 Btu/Lb dry air



Download the example file (exe1_1.zip)
This file covers the examples of 1-1 through 1-4.

Example 1-2. Calculate the air density, specific volume, and enthalpy in US at the ambient conditions of DBT 87.8^{o}F, RH 0% (Dry Air), and sea level.

- Air Density: 0.0723 Lb/ft^3
- Air Specific Volume: 13.8224 ft^3/Lb dry air
- Air Enthalpy: 21.1196 Btu/Lb dry air

Example 1-3. Calculate the air density, specific volume, and enthalpy in US at the ambient conditions of DBT 87.8^{o}F, RH 100%, and sea level.

- Air Density: 0.0711 Lb/ft^3
- Air Specific Volume: 14.4639 ft^3/Lb dry air
- Air Enthalpy: 52.9849 Btu/Lb dry air

Example 1-4. Calculate the air density, specific volume, and enthalpy in US at the ambient conditions of DBT 87.8^{o}F, RH 80%, and 1,000 feet in altitude.

- Air Density: 0.0688 Lb/ft^3
- Air Specific Volume: 14.8824 ft^3/Lb dry air
- Air Enthalpy: 47.3494 Btu/Lb dry air

Example 1-5. Find a relative humidity which the relationship of 1/ air density = specific volume is established at an ambient condition of DBT 87.8^{o}F and sea level.

- Air Density: 0.0723 Lb/ft^3
- 1 / Air Density: 1 / 0.0723 = 13.8224 ft^3/Lb dry air
- Air Specific Volume: 13.8224 ft^3/Lb dry air

The relationship of 1/air density = specific volume is only valid at the point that the relative humidity is zero. That is, only valid for the dry air condition.

PSYCHROMETRICS

Altitude	0.00	feet
Relative Humidity	0.0%	
Dry Bulb Temperature	87.80	°F

Barometric Pressure	29.921	
Air Density	0.0723	Lb/ft^3
1 / Air Density	13.8224	ft^3/Lb dry air
Air Specific Volume	13.8224	ft^3/Lb dry air

Download the example file (exe1_5.zip)

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2. Heat & Mass Transfer Fundamental

Many theories have been developed since the early 1900s describing the heat and mass transfer phenomenon which takes place in several types of atmospheric water cooling devices. Most of these theories are based on sound engineering principles. The cooling tower may be considered as a heat exchanger in which water and air are in direct contact with one another. There is no acceptable method for accurately calculating the total contact surface between water and air. Therefore, a "K" factor, or heat transfer coefficient, cannot be determined directly from test data or by known heat transfer theories. The process is further complicated by mass transfer. Experimental tests conducted on the specified equipment designs can be evaluated using accepted and proven theories which have been developed using dimensional analysis techniques. These same basic methods and theories can be used for thermal design and to predict performance at the operating conditions other than the design point.

Many types of heat and mass transfer devices defined the solution by theoretical methods or dimensional analysis. Design data must be obtained by the full-scale tests under the actual operating conditions. Items such as evaporative condensers in which an internal heat load is being applied, along with water and air being circulated over the condenser tubes in indefinable flow patterns, presents a problem which cannot be solved directly by mathematical methods. The boundary conditions have not been adequately defined and the fundamental equations describing the variables have not been written. Other devices such as spray ponds, atmospheric spray towers, and the newer spray canal systems have not been accurately evaluated solely by mathematical means. This type of equipment utilizes mixed flow patterns of water and air. Many attempts have been made to correlate performance using "drop theories", "cooling efficiency", number of transfer units, all without proven results. Accurate design data are best obtained by the actual tests over a wide range of operating conditions with the specified arrangement.

The development of cooling tower theory seems to begin with Fitzgerald. The American Society of Civil Engineers had asked Fitzerald to write a paper on evaporation, and what had appeared to be a simple task resulted in a 2 year investigation. The result, probably in keeping with the time, is more of an essay than a modern technical paper. Since the study of Fitzgerald, many peoples like Mosscrop, Coffey & Horne, Robinson, and Walker, etc. tried to develop the theory.

1) Merkel Theory

The early investigators of cooling tower theory grappled with the problem presented by the dual transfer of heat and mass. The Merkel theory overcomes this by combining the two into a single process based on enthalpy potential. Dr. Frederick Merkel was on the faculty of the Technical College of Dresden in Germany. He died untimely after publishing his cooling tower paper. The theory had attracted little attention outside of Germany until it was discovered in German literature by H.B. Nottage in 1938.

Cooling tower research had been conducted for a number of years at University of California at Berkley under the direction of Professor L.K.M. Boelter. Nottage, a graduate student, was assigned a cooling tower project which he began by making a search of the literature. He found a number of references to Merkel, looked up the paper and was immediately struck by its importance. It was brought to the attention of Mason and London who were also working under Boelter and explains how they were able to use the Merkel theory in their paper.

Dr. Merkel developed a cooling tower theory for the mass (evaporation of a small portion of water) and sensible heat transfer between the air and water in a counter flow cooling tower. The theory considers the flow of mass and energy from the bulk water to an interface, and then from the interface to the surrounding air mass. The flow crosses these two boundaries, each offering resistance resulting in gradients in temperature, enthalpy, and humidity ratio. For the details for the derivation of Merkel theory, refer to Cooling Tower Performance edited by Donald Baker and the brief derivation is introduced here. Merkel demonstrated that the total heat transfer is directly proportional to the difference between the enthalpy of saturated air at the water temperature and the enthalpy of air at the point of contact with water.

Q = K x S x (hw - ha)
where,

- Q = total heat transfer Btu/h
- K = overall enthalpy transfer coefficient lb/hr.ft^2
- S = heat transfer surface ft^2. S equals to a x V, which "a" means area of transfer surface per unit of tower volume. (ft^2/ft^3), and V means an effective tower volume (ft^3).
- hw = enthalpy of air-water vapor mixture at the bulk water temperature, Btu/Lb dry air
- ha = enthalpy of air-water vapor mixture at the wet bulb temperature, Btu/Lb dry air

The water temperature and air enthalpy are being changed along the fill and Merkel relation can only be applied to a small element of heat transfer surface dS.

dQ = d[K x S x (hw - ha)] = K x (hw - ha) x dS
The heat transfer rate from water side is Q = Cw x L x Cooling Range, where Cw = specific heat of water = 1, L = water flow rate. Therefore, dQ = d[Cw x L x (tw2 - tw1)] = Cw x L x dtw. Also, the heat transfer rate from air side is Q = G x (ha2 - ha1), where G = air mass flow rate Therefore, dQ = d[G x (ha2 - ha1)] = G x dha.
Then, the relation of K x (hw - ha) x dS = G x dha or K x (hw - ha) x dS = Cw x L x dtw are established, and these can be rewritten in K x dS = G / (hw - ha) x dha or K x dS / L = Cw / (hw - ha) x dtw. By integration,

$$\frac{KS}{L} = \frac{KaV}{L} = \frac{G}{L}\int_{ha1}^{ha2} \frac{dh}{hw - ha} \qquad \frac{KS}{L} = \frac{KaV}{L} = Cw\int_{tw1}^{tw2} \frac{dtw}{hw - ha}$$

This basic heat transfer equation is integrated by the four point Tchebycheff, which uses values of y at predetermined values of x within the interval a to b in numerically evaluating the integral $\int_a^b y\,dx$. The sum of these values of y multiplied by a constant times the interval (b - a) gives the desired value of the integral. In its four-point form the values of y so selected are taken at values of x of 0.102673.., 0.406204.., 0.593796.., and 0.897327..of the interval (b - a). For the determination of KaV/L, rounding off these values to the nearest tenth is entirely adequate. The approximate formula becomes:

$$\int_a^b y\,dx = (b - a) \times (y_1 + y_2 + y_3 + y_4) / 4$$

where, y_1 = value of y at $x = a + 0.1 \times (b - a)$ = CWT + 0.1 x Range

y_2 = value of y at $x = a + 0.4 \times (b - a)$ = CWT + 0.4 x Range

y_3 = value of y at $x = b - 0.4 \times (b - a)$ or $x = a + 0.6 \times (b - a)$ = CWT + 0.6 x Range

y_4 = value of y at $x = b - 0.1 \times (b - a)$ or $x = a + 0.9 \times (b - a)$ = CWT + 0.9 x Range

For the evaluation of KaV/L,

$$\frac{KaV}{L} = Cw\int_{tw1}^{tw2} \frac{dtw}{hw - ha} = (tw_2 - tw_1) \times [(1 / Dh_1) + (1 / Dh_2) + (1 / Dh_3) + (1 / Dh_4)] / 4$$

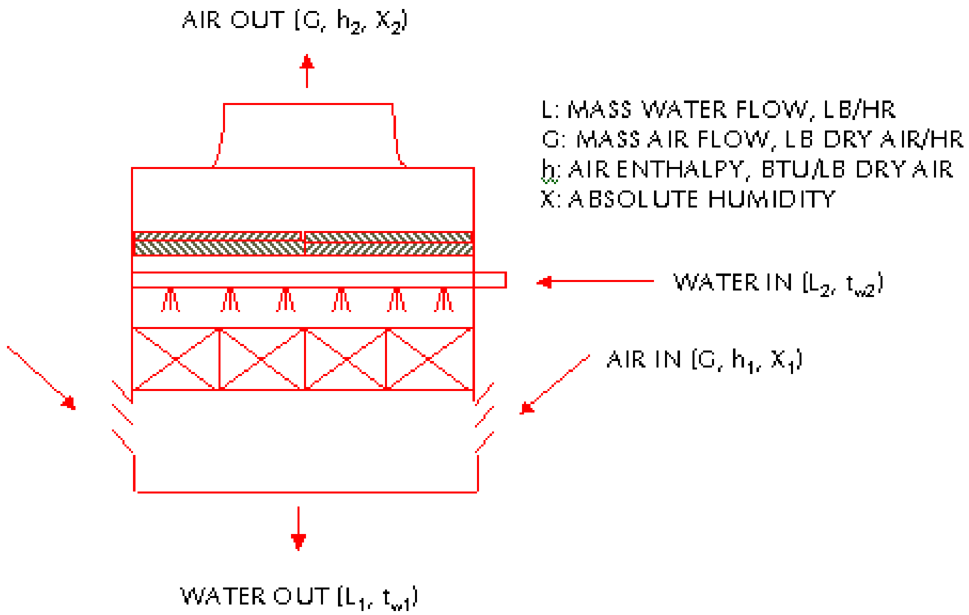
where, Dh_1 = value of (hw - ha) at a temperature of CWT + 0.1 x Range

Dh_2 = value of (hw - ha) at a temperature of CWT + 0.4 x Range

Dh_3 = value of (hw - ha) at a temperature of CWT + 0.6 x Range

Dh_4 = value of (hw - ha) at a temperature of CWT + 0.9 x Range

2) Heat Balance



$HEAT_{in} = HEAT_{out}$

WATER HEAT$_{in}$ + AIR HEAT$_{in}$ = WATER HEAT$_{out}$ + AIR HEAT$_{out}$
$Cw\ L_2\ tw_2 + G\ ha_1 = Cw\ L_1\ tw_1 + G\ ha_2$ Eq. 2-1
(The difference between L_2 (entering water flow rate) and L_1 (leaving water flow rate) is a loss of water due to the evaporation in the direct contact of water and air. This evaporation loss is a result of difference in the water vapor content between the inlet air and exit air of cooling tower. Evaporation Loss is expressed in G x (w_2 -w1) and is equal to $L_2 - L_1$. Therefore, $L_1 = L_2 - G \times (w_2 - w_1)$ is established.)

Let's replace the term of L_1 in the right side of Eq. 2-1 with the equation of $L_1 = L_2 - G \times (w_2 - w_1)$ and rewrite. Then, $Cw\ L_2\ tw_2 + G\ ha_1 = Cw\ [L_2 - G \times (w_2 - w_1)] \times tw_1 + G\ ha_2$ is obtained. This equation could be rewritten in $Cw \times L_2 \times (tw_2 - tw_1) = G \times (ha_2 - ha_1) - Cw \times tw_1 \times G \times (w_2 - w_1)$. In general, the 2nd term of right side is ignored to simplify the calculation under the assumption of $G \times (w_2 - w_1) = 0$.

Finally, the relationship of $Cw \times L_2 \times (tw_2 - tw_1) = G \times (ha_2 - ha_1)$ is established and this can be expressed to $Cw \times L \times (tw_2 - tw_1) = G \times (ha_2 - ha_1)$ again. Therefore, the enthalpy of exit air, $ha_2 = ha_1 + Cw \times L / G \times (tw_2 - tw_1)$ is obtained. The value of specific heat of water is Eq. 2-1 and the term of tw_2 (entering water temperature) - tw_1 (leaving water temperature) is called the cooling range.

Simply, $ha_2 = ha_1 + L/G \times Range$ Eq. 2-2

Consequently, the enthalpy of exit air is a summation of the enthalpy of entering air and the addition of enthalpy from water to air (this is a value of L/G x Range).

Example 2-1. Calculate the ratio of water and air rate for the 20,000 gpm of water flow and 1,600,000 acfm of air flow at DBT 87.8°F, 80% RH, and sea level.

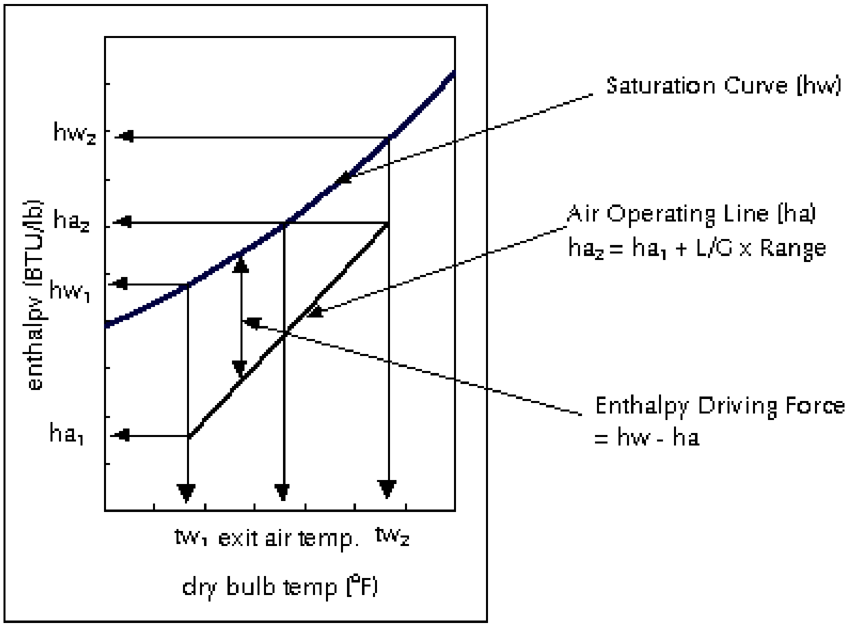
(Solution)
Water Flow Rate = GPM x (500 / 60) lb/min = 20,000 x (500 / 60) =166,666.67 lb/min
(The weight of 1 gallon of water at 60°F equals to 8.345238 pounds and 500 was obtained from 8.345238 x 60 for simplifying the figure.)

Air Flow Rate = ACFM / Specific Volume =1,600,000 / 14.3309 =111,646.76 lb/min
(Specific Volume @ 87.8°F, 80% & sea level = 14.3309 ft^3/lb)

Ratio of Water to Air = Water Flow Rate / Air Flow Rate =166,666.67 / 111,646.76 =1.4928

Example 2-2. Why is L/G in the equation of $ha_2 = ha_1 + L/G \times Range$ called a slope?

(Solution)
This curve is exactly same as a linear function of y = a + b x. The ha_1 corresponds to "a" , L/G corresponds to "b", and the cooling range corresponds to "x". So, L/G is a slope of linear curve.



Example 2-3. Calculate the enthalpy and temperature of exit air for the following cooling tower design conditions.

Given,

- Ambient Wet Bulb Temperature: 82.4ºF
- Relative Humidity: 80%
- Site Altitude: sea level
- L/G Ratio: 1.4928
- Entering Water Temperature: 107.6ºF
- Leaving Water Temperature: 89.6ºF

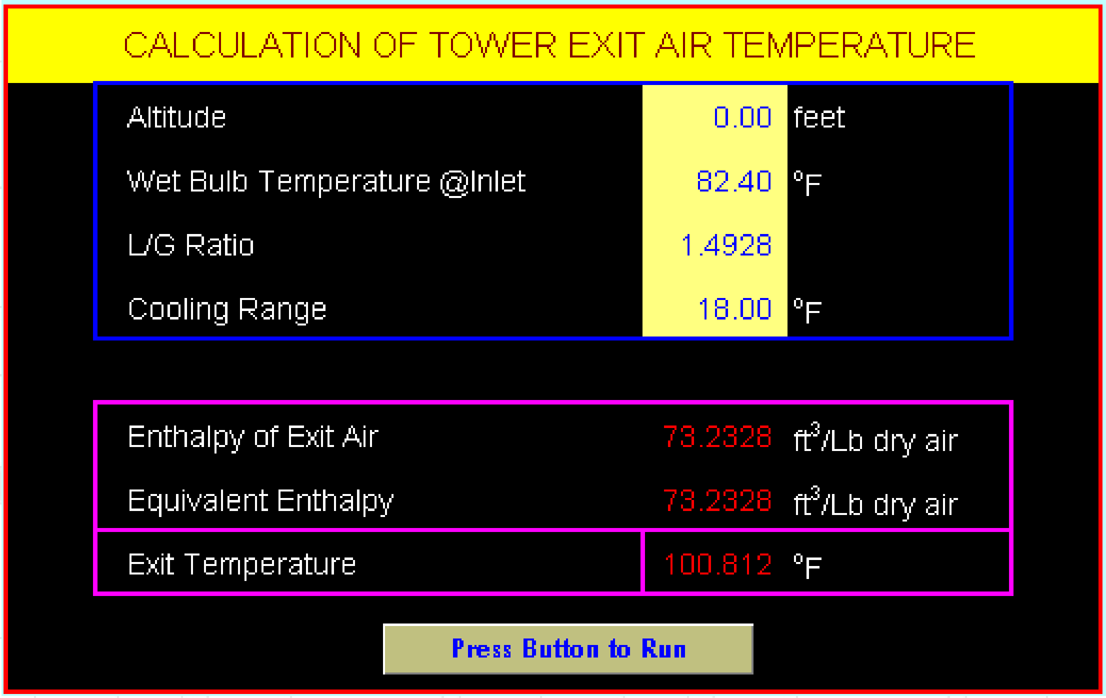
(Solution)
The enthalpy of exit air is calculated from Eq. 2-2 which was derived above. That is, $ha_2 = ha_1 + L/G \times Range$. The enthalpy of inlet air (ha_1) at 82.4ºF WBT & sea level is 46.3624 Btu/Lb dry air.

The cooling range = Entering Water Temp. - Leaving Water Temp. = $(tw_2 - tw_1)$ = 107.6 - 89.6 = 18ºF

Therefore, the enthalpy of exit air (ha_2) is obtained as below.

$ha_2 = ha_1 + L/G \times Range = 46.3624 + 1.4928 \times (107.6 - 89.6) = 73.2328$ BTU/lb

A temperature corresponding to this value of air enthalpy can be obtained from the table published by Cooling Tower Institute or other psychrometric curve. However, this can be computed from the computer program. The procedure of computing a temperature at a given enthalpy is to find a temperature satisfying the same value of enthalpy varying a temperature by means of iteration.

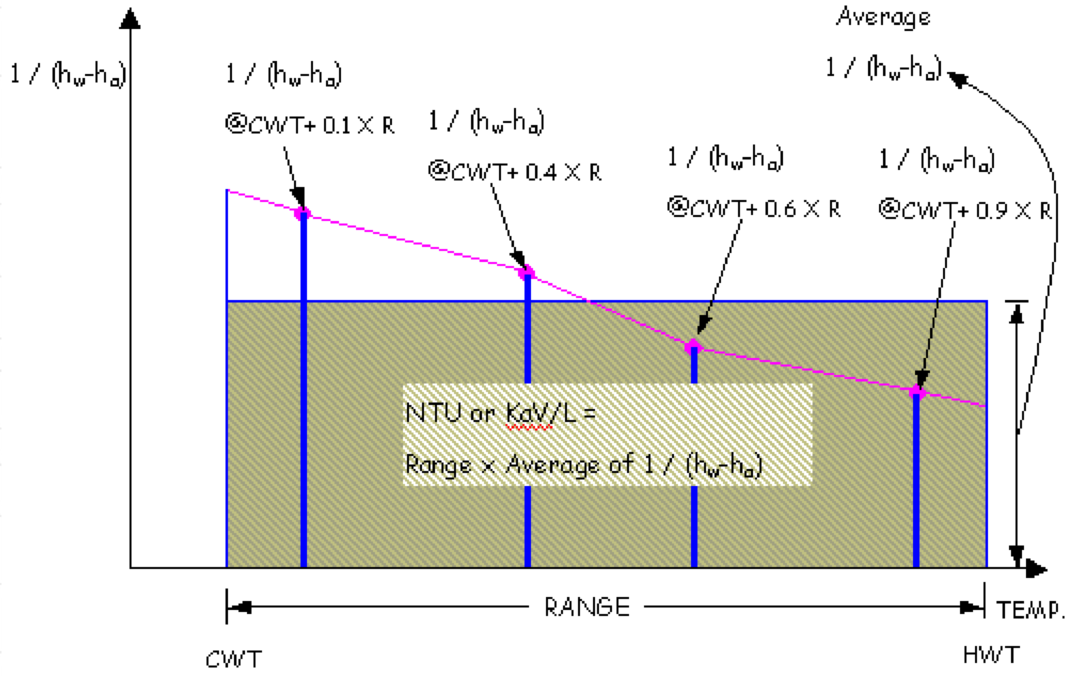


Download the example file(exe2_3.zip)

3) NTU (Number of Transfer Unit) Calculation

$$NTU = \frac{KaV}{L} = Cw\int_{T2}^{T1} \frac{dT}{hw - ha} \quad \text{or} \quad = \frac{G}{L}\int_{ha1}^{ha2} \frac{dha}{hw - ha}$$

The right side of the above equations is obviously dimensionless factor. This can be calculated using only the temperature and flows entering the cooling tower. It is totally independent from the tower size and fill configuration and is often called, for lack of another name, NTU. Plotting several values of NTU as a function of L/G gives what is known as the "Demand" curve. So, NTU is called Tower Demand too.



As shown on above, NTU is an area of multiplying the cooling range by the average of 1/ $(h_w - h_a)$ at four points in the x axis (Temp.).

NTU or KaV/L = Cooling Range x [Sum of 1 / (hw - ha)] / 4

Example 2-4. Determine the tower demand (called Number of Transfer Unit) for the below given conditions.

Given,

- Water Circulation Rate: 16000 GPM
- Entering Air Flow Rate: 80848 Lb of dry air / min
- Ambient Wet Bulb Temperature: 80.0ºF
- Site Altitude: sea level
- Hot Water Temperature: 104.0ºF
- Cold Water Temperature: 89.0ºF

(Solution)
Water Flow Rate = 16,000 x (500 / 60) = 133,333 Lb/min
L/G Ratio = Water Flow Rate / Air Flow Rate = 133,333 / 80,848 = 1.6492

It is very convenient to use the below tool in calculating NTU.

TOWER DEMAND (NTU) CALCULATION						
Altitude (feet)		0.00	Hot Water Temperature			104.00
Wet Bulb Temperature @Inlet (°F)		80.00	Cold Water Temperature			89.00
Water Flow Rate (gpm)		16,000	L/G Ratio			1.6492
Air Mass Flow Rate (Lb'/min)		80,848	Cooling Range			15.00
WATER SIDE			AIR SIDE		ENTHALPY DIFF.	
DESCRIPTIONS	tw (°F)	hw (BTU/Lb')	DESCRIPTIONS	ha (BTU/Lb')	hw - ha	1/(hw-ha)
tw_1 + 0.1 x Range	90.50	56.6478	ha_1 + 0.1 x L/G x Range	46.1645	10.4833	0.0954
tw_1 + 0.4 x Range	95.00	63.3426	ha_1 + 0.4 x L/G x Range	53.5858	9.7567	0.1025
tw_1 + 0.6 x Range	98.00	68.2591	ha_1 + 0.6 x L/G x Range	58.5334	9.7257	0.1028
tw_1 + 0.9 x Range	102.50	76.4013	ha_1 + 0.9 x L/G x Range	65.9547	10.4466	0.0957
Sum of 1 / (hw - ha)............						0.3964
Tower Demand (NTU) = Sum of 1 / (hw - ha) / 4 * RANGE						1.4866

Download the example file(exe2_4.zip)

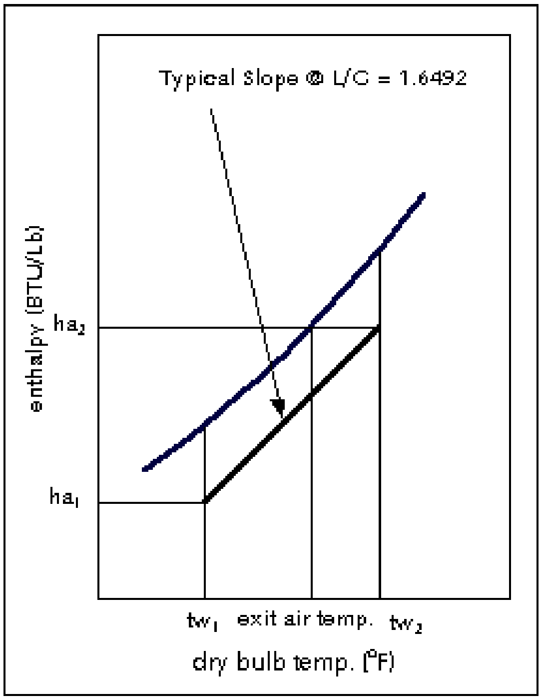
Example 2-5. Compare NTU at the same given conditions as above example 2-4 excepting L/G = 1.2540.

(Solution)

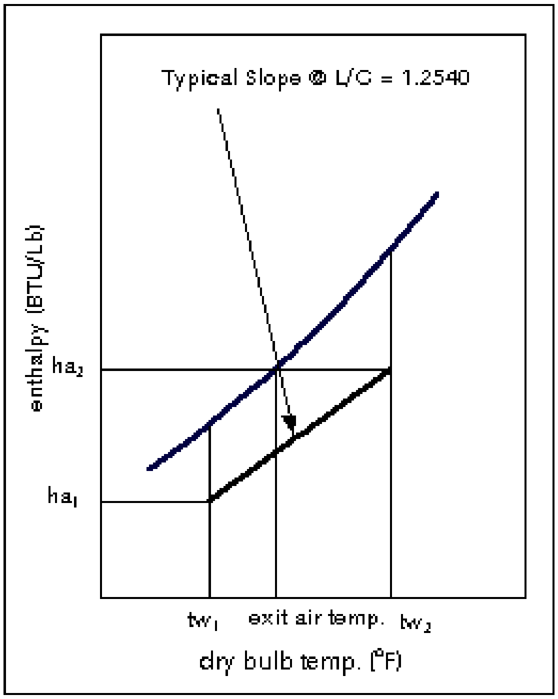
TOWER DEMAND (NTU) CALCULATION						
Altitude (feet)		0.00	Hot Water Temperature			104.00
Wet Bulb Temperature @Inlet (°F)		80.00	Cold Water Temperature			89.00
L/G Ratio		1.2540	Cooling Range			15.0000
WATER SIDE			AIR SIDE		ENTHALPY DIFF.	
DESCRIPTIONS	tw (°F)	hw (BTU/Lb')	DESCRIPTIONS	ha (BTU/Lb')	hw - ha	1/(hw-ha)
tw_1 + 0.1 x Range	90.50	56.6478	ha_1 + 0.1 x L/G x Range	45.5717	11.0761	0.0903
tw_1 + 0.4 x Range	95.00	63.3426	ha_1 + 0.4 x L/G x Range	51.2147	12.1278	0.0825
tw_1 + 0.6 x Range	98.00	68.2591	ha_1 + 0.6 x L/G x Range	54.9767	13.2824	0.0753
tw_1 + 0.9 x Range	102.50	76.4013	ha_1 + 0.9 x L/G x Range	60.6197	15.7816	0.0634
Sum of 1 / (hw - ha)............						0.3114
Tower Demand (NTU) = Sum of 1 / (hw - ha) / 4 * RANGE						1.1677

Download the example file(exe2_5.zip)
(This file covers the examples of 2-6 & 2-7.)

The NTU at L/G = 1.2540 is smaller than NTU at L/G = 1.6492 under the same design conditions. What the L/G is reduced to 1.2540 from 1.6492 under the same water flow rate means that the air mass is increased. In other word, the decrease of L/G for the same water flow rate means the decrease of enthalpy in the air side and a value of 1 / (hw - ha) is consequently decreased. Also, the exit enthalpy per pound dry air is decreased and the temperature of exit air is reduced.



<Example 2-4>



<Example 2-5>

In the actual cooling tower, what the water is evenly distributed on the entire top of fill is very rare. If the temperature is measured onto the top of drift eliminator, the temperature at the area where the water is smaller than other locations is always lower than the water is larger. This is because the air at the area where the water is small can go easily up due to less pressure drop with the water loading.

Example 2-6. Compare NTU at the same given conditions as above example 2-4 excepting that the ambient wet bulb temperature has been changed to 81.0ºF from 80.0ºF.

(Solution)

WATER SIDE			AIR SIDE		ENTH DIFF.
Descriptions	tw (ºF)	hw (But/Lb)	Description	ha (Btu/Lb)	1/(hw-ha)
tw_1 + 0.1 x R	90.50	56.6478	ha_1 + 0.1 x L/G x R	47.2587	0.1065
tw_1 + 0.4 x R	95.00	63.3426	ha_1 + 0.4 x L/G x R	54.6800	0.1154
tw_1 + 0.6 x R	98.00	68.2591	ha_1 + 0.6 x L/G x R	59.6276	0.1159
tw_1 + 0.9 x R	102.50	76.4013	ha_1 + 0.9 x L/G x R	67.0489	0.1069
Sum of 1 / (hw - ha)					0.4447
Total Tower Demand (NTU) = Cooling Range x Sum of 1 / (hw - ha)					1.6677

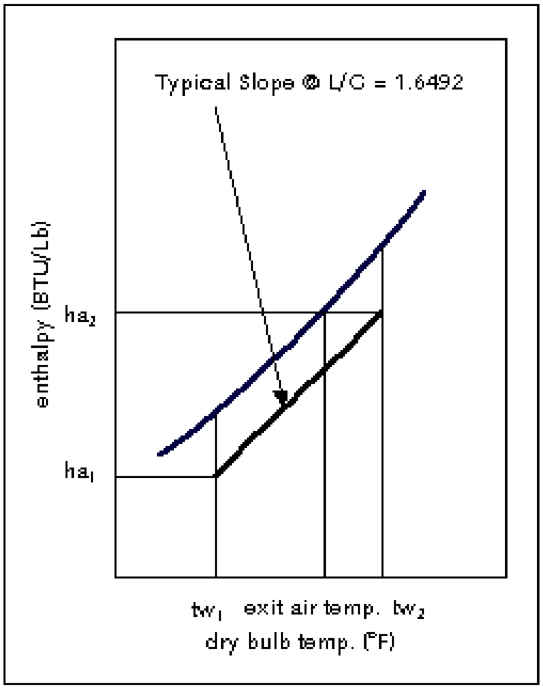
Through this example, the higher ambient wet bulb temperature (approach is smaller), the larger NTU. That is, the enthalpy driving force (hw - ha) is reduced as the ambient wet bulb temperature is increased. This means that less driving force requires more heat transfer area or more air. (Sometimes, NTU calls "Degree of Difficulty".)

Example 2-7. Compare NTU at the same given conditions as above example 2-4 excepting that the entering water temperature has been changed to 101.0ºF from 104ºF.

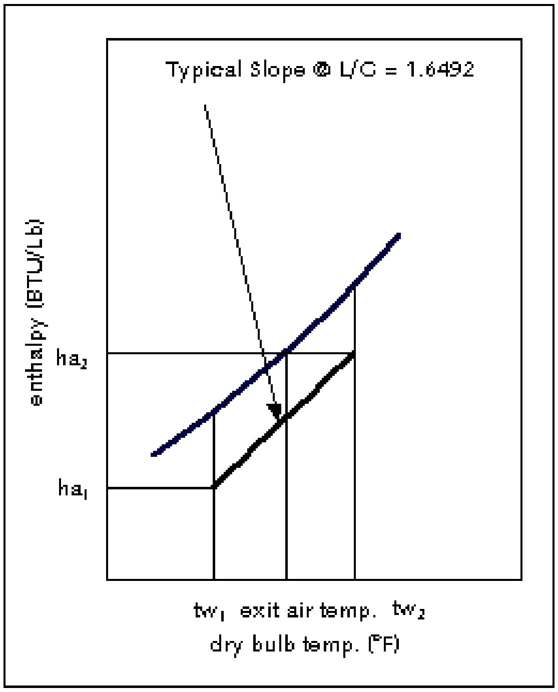
(Solution)

WATER SIDE			AIR SIDE		ENTH DIFF.
Descriptions	tw (°F)	hw (But/Lb)	Description	ha (Btu/Lb)	1/(hw-ha)
tw_1 + 0.1 x R	90.20	56.2283	ha_1 + 0.1 x L/G x R	45.6697	0.0947
tw_1 + 0.4 x R	93.80	61.4808	ha_1 + 0.4 x L/G x R	51.6068	0.1013
tw_1 + 0.6 x R	96.20	65.2631	ha_1 + 0.6 x L/G x R	55.5648	0.1031
tw_1 + 0.9 x R	99.80	71.4001	ha_1 + 0.9 x L/G x R	61.5019	0.1010
Sum of 1 / (hw - ha)					0.4001
Total Tower Demand (NTU) = Cooling Range x Sum of 1 / (hw - ha)					1.2004

This example presents that the smaller range under the same approach, the smaller NTU.



<Example 2-6>



<Example 2-7>

COMPARISON TABLE				
Descriptions	Exe. 2-4	Exe. 2-5	Exe. 2-6	Exe. 2-7
Range (°F)	15.0	15.0	15.0	12.0
Approach(°F)	9.0	9.0	8.0	9.0
Wet Bulb Temp.(°F)	80.0	80.0	81.0	80.0
L/G Ratio	1.6492	1.2540	1.6492	1.6492
KaV/L	1.4866	1.1677	1.6677	1.2004
Driving Force (BTU/Lb)	10.1031	13.0670	9.0089	10.0073
Order of Cooling Difficulty	2nd	4th	1st	3rd

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3. Tower Demand & Characteristic Curves

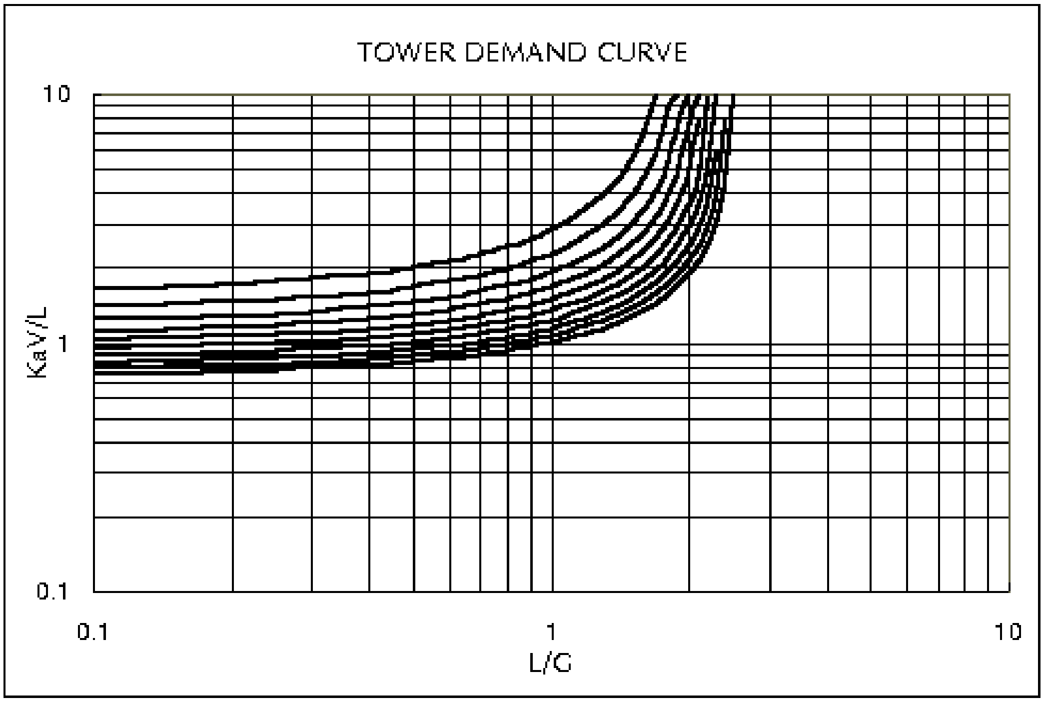
1) Tower Demand

Liechtenstein introduced the "Cooling Tower" equation in 1943 and he used Merkel theory in conjunction with differential and fundamental equations to define cooling tower boundary conditions. The resulting dimensionless groups related the variables for heat and mass transfer on the counter flow type tower. Liechtenstein determined by experimental testing that his equation did not fully account for the air mass rate or velocity. He also implies in the original paper that tests conducted at the University of California suggested a variation in the tower characteristic due to the inlet water temperature. A method is given for adjusting the tower characteristic for the effect. Several investigators have substantiated the effect of hot water temperature and air velocity on the counter flow tower.

The Merkel equation is used to calculate the thermal demand based on the design temperature and selected liquid-to-gas ratios (L/G). The value of KaV/L becomes a measure of the order of difficulty for the liquid cooling requirements. The design temperature and L/G relate the thermal demand to the MTD (Mean Temperature Difference) used in any heat transfer problem. As stated by Liechtenstein the use of his method required a laborious trial-and-error graphical integration solution for tower design. During his employment with the Foster-Wheeler Corporation, he published a limited edition of "Cooling Tower Black Book" in 1943. The tower demand calculations were incorporated into a volume of curves eliminating the need for tedious busy work. For many years the publication was the industry standard for evaluating and predicting the performance of tower.

A similar publication entitled "Counter Flow Cooling Tower Performance" was released during 1957 by J. F. Pritchard and Co. of California. The so-called "Brown Book" presented a change in format to a multi-cycle log plot. This format allows the cooling tower characteristic curves to be plotted as straight lines. The publication include cooling tower design data for various types of counter flow fill. Design procedures and factors affecting cooling tower selection and performance are discussed.

With the advent of the computer age the Cooling Tower Institute published the "Blue Book" entitled "Cooling Tower Performance Curves" in 1967. The availability and use of the computer allowed the Performance and Technology Committee to investigate several methods of numerical integration to solve the basic equation. The Tchebycheff method was selected as being of adequate consistency and accuracy for the proposed volume. The CTI curves were calculated and plotted by computer over a large span of temperature and operating conditions. The curves are plotted with the thermal demand, KaV/L as a function of the liquid-to-gas ratio, L/G. The approach lines (tw_1 - WBT) are shown as parameters. The curves contain a set of 821 curves, giving the values of KaV/L for 40 wet bulb temperature, 21 cooling ranges and 35 approaches.



2) Tower Characteristic

An equation form used to analyze the thermal performance capability of a specified cooling tower was required. Currently, the following equation is widely accepted and is a very useful to be able to superimpose on each demand curve, since KaV/L vs. L/G relationship is a linear function on log-log demand curve.

$KaV/L = C\,(L/G)^{-m}$

where,

KaV/L=Tower Characteristic, as determined by Merkel equation
C=Constant related to the cooling tower design, or the intercept of the characteristic curve at L/G=1.0
m=Exponent related to the cooling tower design (called slope), determined from the test data

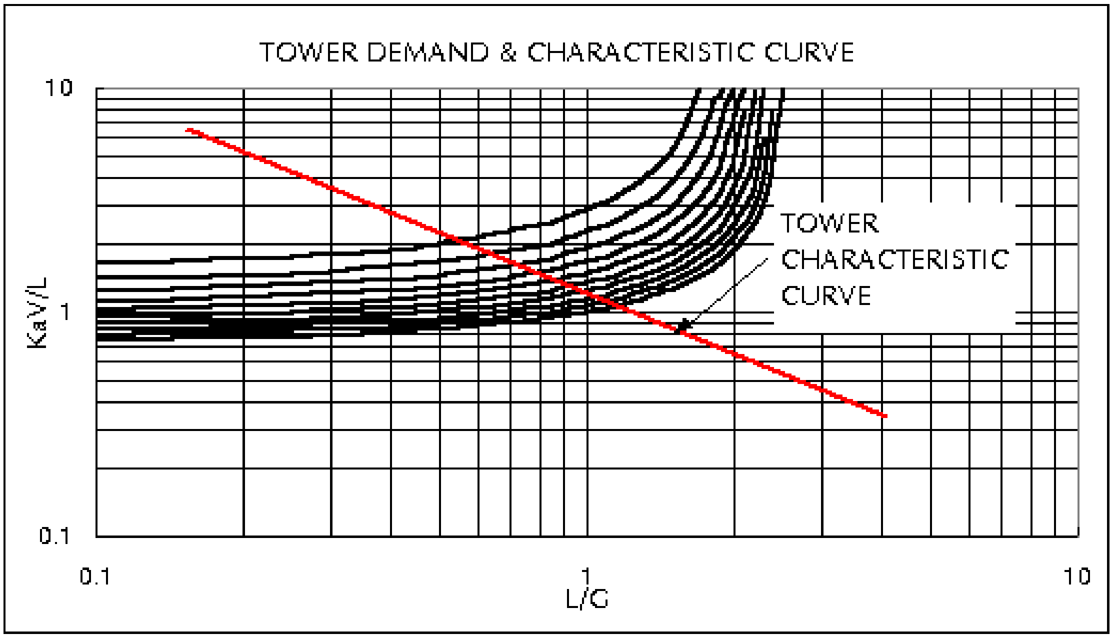
The characteristic curve may be determined in one of the following three ways;

(1) If still applicable and available, the vendor supplied characteristic curve may be used. In all cases the slope of this curve can be taken as the slope of the operating curve.
(2) Determine by field testing one characteristic point and draw the characteristic curve through this point parallel to the original characteristic curve, or a line through this point with the proper slope (- 0.5 to - 0.8).
(3) Determine by field testing at least two characteristic points at different L/G ratios. The line through these two points is the characteristic curve. The slope of this line should fall within the expected range, and serves as a check on the accuracy of the measurement.

A characteristic point is experimentally determined by first measuring the wet bulb temperature, air discharge temperature, and cooling water inlet and outlet temperature. The L/G ratio is then calculated as follows;

(1) It may be safely assumed that the air discharge is saturated. Therefore, the air discharge is at its wet bulb. Knowing wet bulb temperature at the inlet of tower, the enthalpy increase of the air stream can be obtained from a psychrometric chart. Air and water flow rates have to be in the proper range for uniform flow distribution. In case of recirculation of the air discharge, the inlet wet bulb may be 1 or 2°F above the atmospheric wet bulb temperature.
(2) From a heat and mass balance the dry air rate and the prevailing L/G ratio in the tower can be calculated [L/G = D ha / (Cw x (tw_2 - tw_1))]

Next, the corresponding KaV/L value has to be established. This is simply done by plotting the calculated L/G and approach on the demand curve for the proper wet bulb and range.



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4. Cooling Tower Performance Variables

There are a lot of parameters which effects to the cooling tower design and operation. Some will be discussed here through the examples below.

Example 4-1. The water circulating rate is one of most important primary variables. Obviously it is a key number in the original design. A problem frequently encountered is the prediction of the effects of changes in water circulation rate on the temperatures of the water entering and leaving an existing cooling tower. Assume an existing mechanical draft cooling tower is operating at the following conditions, and estimate the cold & hot water temperature when the water flow rate is increased to 20,000 GPM, assuming no change in the entering air mass flow rate, wet bulb temperature, and heat load. (Actually, the air mass is decreased due to the increase of pressure drop at the fill with the increase of water.)

Given,

- Water Flow Rate (L_1): 16000
- Entering Air Flow Rate (G_1): 80848
- Ambient Wet Bulb Temperature: 80.0
- Site Altitude: sea level
- Hot Water Temperature (HWT, tw_2): 104.0
- Cold Water Temperature (CWT, tw_1): 89.0
- Characteristic Curve Slope (m): -0.800
- Alternative Water Flow Rate (L_2): 20000

(Solution)
Range, R_1 = HWT - CWT = $tw_2 - tw_1$ = 104 - 89 = 15ºF
Water Flow Rate in Pound, L_1 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = 133,333.3 lb/min
Heat Load, $D_1 = L_1 \times R_1$ = 133,333.3 x 15 = 2,000,000 BTU/min
Air Mass Flow Rate, G_1 = 80,848 lb/min
Liquid to Gas Ratio, $L/G_1 = L_1 / G_1$ = 133,333.3 / 80,848 = 1.6492
Water Flow Rate in Pound, L_2 = Water Flow Rate x (500 / 60) = 20,000 x (500 / 60) = 166,666.7 lb/min
Heat Load, $D_2 = D_1$ = 2,000,000 BTU/min
Air Mass Flow Rate, $G_2 = G_1$ = 80,848 lb/min
Liquid to Gas Ratio, $L/G_2 = L_2 / G_2$ = 166,666.7 / 80,848 = 2.0615
Range, $R_2 = D_2 / L_2$ = 2,000,000 / 166,666.7 or = $R_1 \times (L_1 / L_2)$ = 12ºF
(The range must be calculated since the heat load is same as the design condition but water flow rate was changed.)

In estimating the cold water temperature with the new water flow rate, there are two methods.

One is to find a new approach by means of the computer. Another is to find it using the CTI performance curves. Two methods shall be discussed. With the use of the computer, the iteration is required until the value of the new tower characteristic is exactly equal to the new KaV/L (NTU) varying the approach by means of computer.

First Step: Calculate NTU at the design conditions as follows;

WATER SIDE			AIR SIDE		ENTH DIFF.
Descriptions	tw (ºF)	hw (Btu/Lb)	Description	ha (Btu/Lb)	1/(hw-ha)
tw_1 + 0.1 x R	90.50	56.6478	ha_1 + 0.1 x L/G x R	46.1645	0.0954
tw_1 + 0.4 x R	95.00	63.3426	ha_1 + 0.4 x L/G x R	53.5858	0.1025
tw_1 + 0.6 x R	98.00	68.2591	ha_1 + 0.6 x L/G x R	58.5334	0.1028
tw_1 + 0.9 x R	102.50	76.4013	ha_1 + 0.9 x L/G x R	65.9547	0.0957
Sum of 1 / (hw - ha)					0.3964
Total Tower Demand (NTU) = Cooling Range x Sum of 1 / (hw - ha)					1.4866

Second Step: Calculate a value of "C" of tower characteristic for the design conditions as follows;

$C = KaV/L / (L/G_1)^{-m} = KaV/L \times (L/G_1)^{m} = 1.4866 \times (1.6492)^{0.8} = 2.21825$

Third Step: Calculate a new tower characteristic for the increased water flow as follows;

New Tower Characteristic = $C \times (L/G_2)^{-m} = 2.21825 \times (2.0615)^{-0.8} = 1.2436$

(Note that "C" value is a constant value regardless the change of water flow rate in finding the approach at the alternative temperature conditions.) The new tower characteristic for the increased water flow rate can be calculated as above.

Forth Step: Iterate until the value of new characteristic is equal to the new NTU varying the value of approach.

New Cold Water Temperature = Wet Bulb Temperature + New Approach

DATA INPUT	
INPUT PARAMETERS	DATA
1. Design L/G	1.6492
2. Design HWT (°F)	104.00
3. Design CWT (°F)	89.00
4. Wet Bulb Temp. (°F)	80.00
5. Characteristic Slope	-0.800
6. Altitude (Feet)	0
7. Design Cooling Range	15.00
8. Design Approach	9.00
9. Design KaV/L	1.4866
10. C VALUE IN CHA.	2.2183
11. New L/G	2.0615
12. New Cooling Range (°F)	12.00
13. New KaV/L	1.2436

CALCULATION OF KaV/L @ DESIGN			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	56.6478	46.1645	0.09539
0.4	63.3426	53.5859	0.10249
0.6	68.2591	58.5335	0.10282
0.9	76.4013	65.9549	0.09573
			0.39643

CALCULATION OF KaV/L @ A POINT			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	58.2789	46.1645	0.08255
0.4	63.7301	53.5859	0.09858
0.6	67.6573	58.5335	0.10960
0.9	74.0323	65.9549	0.12380
		1.2436	0.41453
		10.445	
New Cold Water Temperature			90.45
New Hot Water Temperature			102.45

Fifth Step: Compute the cold water temperature with the result of iteration as follows;

New CWT = WBT + New Approach = 80 + 10.45 = 90.45°F
New HWT = CWT + Range = 90.45 + 12 = 102.45°F

The following is a way to estimate the new cold water temperature with the use of CTI performance curve. From the curve of 80 WBT and 15 range, the initial operating point is located at the intersection of L/G = 1.6492 line and approach 9°F curve. The corresponding value of KaV/L is 1.502.

The point of intersecting with the values of KaV/L and L/G for new conditions is then plotted on the curve of 80 WBT and 12 range, and the point of intersecting with the approach line and this point is a new approach which intended to obtain.

The intersection of New KaV/L = 1.2436 and L/G = 2.0615 on of L/G2 = 2.0615 on the approach line determines the new approach = 10.35°F, and then the water temperatures can be predicted:

New CWT = WBT + New Approach = 80 + 10.35 = 90.35°F
New HWT = CWT + Range = 90.35 + 12 = 102.35°F

The reason that there is a little difference in the values between the computer aid calculation and the CTI graphical methods is due to a very little difference in the enthalpy value between the formula used by this and CTI

Download the example file(exe4_1.zip)
This file covers the examples of Example 4-2 through 4-5.

Example 4-2. Estimate the cold & hot water temperature when the water flow rate is increased to 20,000 GPM from 16,000 GPM and the slope of tower characteristic was changed to - 0.7 from - 0.8. Others are same as above example 4-1.

(Solution)
First Step: Same as example 4-1.

Second Step: Calculate a value of "C" of tower characteristic for the design conditions as follows;

$C = KaV/L / (L/G)^{-m} = KaV/L \times (L/G)^{m} = 1.4866 \times (1.6492)^{0.7} = 2.11001$

Third Step: Calculate a new tower characteristic for the increased water flow as follows;

New Tower Characteristic $= C \times (L/G)^{-m} = 2.11001 \times (2.0615)^{-0.7} = 1.2716$
(Note that "C" value is a constant value regardless the change of water flow rate in finding the approach at the alternative temperature conditions.) The new tower characteristic fourth increased water flow rate can be calculated as above.

Forth Step: Iterate until the value of new characteristic is equal to the new NTU varying the value of approach.

New Cold Water Temperature = Wet Bulb Temperature + New Approach

DATA INPUT	
INPUT PARAMETERS	DATA
1. Design L/G	1.6492
2. Design HWT (°F)	104.00
3. Design CWT (°F)	89.00
4. Wet Bulb Temp. (°F)	80.00
5. Characteristic Slope	-0.700
6. Altitude (Feet)	0
7. Design Cooling Range	15.00
8. Design Approach	9.00
9. Design KaV/L	1.4866
10. C VALUE IN CHA.	2.1100
11. New L/G	2.0615
12. New Cooling Range (°F)	12.00
13. New KaV/L	1.2716

CALCULATION OF KaV/L @ DESIGN			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	56.6478	46.1645	0.09539
0.4	63.3426	53.5859	0.10249
0.6	68.2591	58.5335	0.10282
0.9	76.4013	65.9549	0.09573
			0.39643
CALCULATION OF KaV/L @ A POINT			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	58.1007	46.1645	0.08378
0.4	63.5346	53.5859	0.10052
0.6	67.4492	58.5335	0.11216
0.9	73.8034	65.9549	0.12741
		1.2716	0.42387
		10.322	
New Cold Water Temperature			90.32
New Hot Water Temperature			102.32

Fifth Step: Compute the cold water temperature with the result of iteration as follows;

New CWT = WBT + New Approach = 80 + 10.32 = 90.32ºF
New HWT = CWT + Range = 90.32 + 12 = 102.32ºF

Through this example, it was proven that the cold water temperature at the slope of - 0.7 is slightly lower than - 0.13.

<u>**Example 4-3**</u>. The example number 18 was based on the assumption that the heat load is constant for the increase of water flow rate. Estimate the cold & hot water temperature under the assumption that the cooling range is constant for the increase of water flow rate to 20,000 from 16,000 GPM.

(Solution)
Range, $R_1 = R_2$ = HWT - CWT = $tw_2 - tw_1$ = 104 - 89 = 15ºF
Water Flow Rate in Pound, L_1 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = 133,333.3 lb/min
Heat Load, $D_1 = L_1 \times R_1$ = 133,333.3 x 15 = 2,000,000 BTU/min
Air Mass Flow Rate, $G_1 = G_2$ = 80,848 lb/min
Liquid to Gas Ratio, $L/G_1 = L_1 / G_1$ = 133,333.3 / 80,848 = 1.6492
Water Flow Rate in Pound, L_2 = Water Flow Rate x (500 / 60) = 20,000 x (500 / 60) = 166,666.7 lb/min
Heat Load, $D_2 = L_2 \times R_2$ = 166,666.7 x 15 = 2,500,000 BTU/min
Liquid to Gas Ratio, $L/G_2 = L_2 / G_2$ = 166,666.7 / 80,848 = 2.0615

The value of NTU at the design conditions is same as a value calculated in the example 4-1. The value of "C" of tower characteristic for the design conditions same as the example 4-1. The new tower characteristic for the increased water flow is also same as the example 4-1. Iterate until the value of new characteristic is equal to the new NTU varying the value of approach.

New Cold Water Temperature = Wet Bulb Temperature + New Approach

DATA INPUT	
INPUT PARAMETERS	DATA
1. Design L/G	1.6492
2. Design HWT (°F)	104.00
3. Design CWT (°F)	89.00
4. Wet Bulb Temp. (°F)	80.00
5. Characteristic Slope	-0.800
6. Altitude (Feet)	0
7. Design Cooling Range	15.00
8. Design Approach	9.00
9. Design KaV/L	1.4866
10. C VALUE IN CHA.	2.2183
11. New L/G	2.0615
12. New Cooling Range (°F)	15.00
13. New KaV/L	1.2436

CALCULATION OF KaV/L @ DESIGN			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	56.6478	46.1645	0.09539
0.4	63.3426	53.5859	0.10249
0.6	68.2591	58.5335	0.10282
0.9	76.4013	65.9549	0.09573
			0.39643
CALCULATION OF KaV/L @ A POINT			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	61.0459	46.7830	0.07011
0.4	68.2833	56.0597	0.08181
0.6	73.6048	62.2442	0.08802
0.9	82.4285	71.5210	0.09168
		1.2436	0.33162
		12.014	
New Cold Water Temperature			92.01
New Hot Water Temperature			107.01

Fifth Step: Compute the cold water temperature with the result of iteration as follows;

New CWT = WBT + New Approach = 80 + 12.01 = 92.01ºF
New HWT = CWT + Range = 92.01 + 15 = 107.01ºF

Example 4-4. Assume again the conditions of example 4-1 and determine the cold and hot water temperature when the heat load is added to increase the cooling range from 15 to 20ºF, assuming no change in the water circulation rate or in entering air mass flow rate or wet bulb temperature.

(Solution)

Range, R_1 = HWT - CWT = tw_2 - tw_1 = 104 - 89 = 15°F

Water Flow Rate in Pound, L_1 = L_2 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = 133,333.3 lb/min

Air Mass Flow Rate, G_1 = G_2 = 80,848 lb/min

Liquid to Gas Ratio, L/G_1 = L_1 / G_1 = L/G_2 = 133,333.3 / 80,848 = 1.6492

Range, R_2 = 20°F

The value of NTU, and "C" at the design conditions is same as a value calculated in the example 4-1. Also, the new tower characteristic for even a increased cooling range is same as the example 4-1. Iterate until the value of new characteristic is equal to the new NTU varying the value of approach. (New Cold Water Temperature = Wet Bulb Temperature + New Approach)

DATA INPUT	
INPUT PARAMETERS	DATA
1. Design L/G	1.6492
2. Design HWT (°F)	104.00
3. Design CWT (°F)	89.00
4. Wet Bulb Temp. (°F)	80.00
5. Characteristic Slope	-0.800
6. Altitude (Feet)	0
7. Design Cooling Range	15.00
8. Design Approach	9.00
9. Design KaV/L	1.4866
10. C VALUE IN CHA.	2.2183
11. New L/G	1.6492
12. New Cooling Range (°F)	20.00
13. New KaV/L	1.4866

CALCULATION OF KaV/L @ DESIGN			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	56.6478	46.1645	0.09539
0.4	63.3426	53.5859	0.10249
0.6	68.2591	58.5335	0.10282
0.9	76.4013	65.9549	0.09573
			0.39643
CALCULATION OF KaV/L @ A POINT			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	59.7433	46.9891	0.07841
0.4	69.3684	56.8843	0.08010
0.6	76.6812	63.4811	0.07576
0.9	89.2334	73.3763	0.06306
		1.4866	0.29733
		10.646	
New Cold Water Temperature			90.65
New Hot Water Temperature			110.65

Fifth Step: Compute the cold water temperature with the result of iteration as follows;

New CWT = WBT + New Approach = 80 + 10.65 = 90.65°F

New HWT = CWT + Range = 90.65 + 20 = 110.65°F

Example 4-5. Assume the existing mechanical-draft cooling tower is operating at the initial conditions of example 4-1. Determine the cold & hot water temperature if the air mass flow rate is reduced to 53,900 lb/min by the adjustment of the fan pitch angle and/or fan speed.

(Solution)

Range, R_1 = HWT - CWT = tw_2 - tw_1 = 104 - 89 = 15°F

Water Flow Rate in Pound, L_1 = L_2 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = 133,333.3 lb/min

Air Mass Flow Rate, G_1 = 80,848 lb/min

Liquid to Gas Ratio, L/G_1 = L_1 / G_1 = 133,333.3 / 80,848 = 1.6492

Air Mass Flow Rate, G_2 = 53,900 lb/min

Liquid to Gas Ratio, L/G_2 = L_2 / G_2 = 133,333.3 / 53,900 = 2.4737

The value of NTU, and "C" at the design conditions is same as a value calculated in the example 4-1.Calculate a new tower characteristic for the decreased air mass flow.

New Tower Characteristic = C x $(L/G)^{-m}$ = 2.21825 x $(2.4737)^{-0.8}$ = 1.0748
Iterate until the value of new characteristic is equal to the new NTU varying the value of approach. New Cold Water Temperature = Wet Bulb Temperature + New Approach

DATA INPUT	
INPUT PARAMETERS	DATA
1. Design L/G	1.6492
2. Design HWT (°F)	104.00
3. Design CWT (°F)	89.00
4. Wet Bulb Temp. (°F)	80.00
5. Characteristic Slope	-0.800
6. Altitude (Feet)	0
7. Design Cooling Range	15.00
8. Design Approach	9.00
9. Design KaV/L	1.4866
10. C VALUE IN CHA.	2.2183
11. New L/G	2.4737
12. New Cooling Range (°F)	15.00
13. New KaV/L	1.0748

CALCULATION OF KaV/L @ DESIGN			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	56.6478	46.1645	0.09539
0.4	63.3426	53.5859	0.10249
0.6	68.2591	58.5335	0.10282
0.9	76.4013	65.9549	0.09573
			0.39643
CALCULATION OF KaV/L @ A POINT			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	65.5009	47.4013	0.05525
0.4	73.2951	58.5329	0.06774
0.6	79.0325	65.9540	0.07646
0.9	88.5571	77.0857	0.08717
		1.0748	0.28663
		14.846	
New Cold Water Temperature			94.85
New Hot Water Temperature			109.85

Fifth Step: Compute the cold water temperature with the result of iteration as follows;

New CWT = WBT + New Approach = 80 + 14.85 = 94.85°F
New HWT = CWT + Range = 94.85 + 15 = 109.85°F

Example 4-6. Assume that the cold & hot water temperature at the conditions where the wet bulb temperature is decreased to 77°F from 80°F and the air mass flow is changed to 53,900 lb/min. Others remain unchanged from example 4-1.

(Solution)
Range, $R_1 = R_2$ = HWT - CWT = $tw_2 - tw_1$ = 104 - 89 = 15°F
Water Flow Rate in Pound, L_1 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = 133,333.3 lb/min
Air Mass Flow Rate, G_1 = 80,848 lb/min
Liquid to Gas Ratio, $L/G_1 = L_1 / G_1 = L_2$ = 133,333.3 / 80,848 = 1.6492
Air Mass Flow Rate, G_2 = 53,900 lb/min
Liquid to Gas Ratio, $L/G_2 = L_2 / G_2$ = 133,333.3 / 53,900 = 2.4737

The value of NTU, and "C" at the design conditions is same as a value calculated in the example 4-1. Calculate a new tower characteristic for the decreased air mass flow.

New Tower Characteristic = C x $(L/G)^{-m}$ = 2.21825 x $(2.4737)^{-0.8}$ = 1.0748

Forth Step: Iterate until the value of new characteristic is equal to the new NTU varying the value of approach. New Cold Water Temperature = Wet Bulb Temperature + New Approach

DATA INPUT	
INPUT PARAMETERS	DATA
1. Design L/G	1.6492
2. Design HWT (°F)	104.00
3. Design CWT (°F)	89.00
4. Wet Bulb Temp. (°F)	80.00
5. Characteristic Slope	-0.800
6. Altitude	0
7. Design Cooling Range (°F	15.00
8. Design Approach (°F)	9.00
9. Design KaV/L	1.4866
10. C VALUE IN CHA.	2.2183
11. New L/G	2.4737
12. New Cooling Range (°F)	15.00
13. New Wet Bulb Temp. (°F	77.00
14. New KaV/L	1.0748

CALCULATION OF KaV/L @ DESIGN			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	56.6478	46.1645	0.09539
0.4	63.3426	53.5859	0.10249
0.6	68.2591	58.5335	0.10282
0.9	76.4013	65.9549	0.09573
			0.39643
CALCULATION OF KaV/L @ A POINT			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	62.9515	44.2731	0.05354
0.4	70.4262	55.4047	0.06657
0.6	75.9249	62.8258	0.07634
0.9	85.0471	73.9575	0.09017
		1.0748	0.28662
		16.251	
New Cold Water Temperature			93.25
New Hot Water Temperature			108.25

Fifth Step: Compute the cold water temperature with the result of iteration as follows;

New CWT = WBT + New Approach = 77.0 + 16.25 = 93.25°F
New HWT = CWT + Range = 93.25 + 15 = 108.25°F

Download the example file(exe4_6.zip)

5. Consideration of By-Pass Wall Water

This factor accounts for the amount of water which unavoidably bypasses the fill along the outside and partition walls, internal columns, internal risers etc. This water is not cooled as much as the water passing through the fill. This effect is well known and recognized as the WALL EFFECT but there is no precise theory on how to predict and account for it. This is not a reason for neglecting it in the calculations. It may be very large particularly in a small tower where it can be as big as 20%. Even large towers can have 2% to 5% on the walls. The approach to this problem is very simple. The by-pass wall water is assumed to be only half cooled.

How to estimate the by-pass wall water? Through an example, the estimation can be discussed. A 36 x 36 ft tower cell has 144 nozzles. 40 nozzles are near to the four walls each projecting 10 % of their water onto those walls. 40 x 10% / 144 = 2.78 %. 4 nozzles are in the corners and project 20% of their water into the wall. 4 x 20% / 144 = 0.56 %. There are 25 internal columns. Each column receives 5% of the water from 4 adjacent nozzles 25 x 4 x 5% / 144 = 3.47 %. Then total by-pass is 6.81% and the water amount for being half cooled is 6.81 / 2 = 3.4%. This means that 3.4% of total water flow is passing through the wall under not being cooled. This is not an exaggerated number. Obviously this evaluation largely depends on water distribution design and the type of nozzles used. A lot of precautions can be taken to minimize this value but it must be kept in mind that that it is better to have a little water on the walls than leaving dry spots with no water at all. Many cooling tower fills do not redistribute the water very well and air will rush through a dry spot where there is less resistance.

If the tower was 18 x 18 ft the same type of evaluation would give:

16 x 10% / 36 = 4.4 %
4 x 20% / 36 = 2.2 %
4 x 4 x 5% / 36 = 2.2 %
Total = 8.8 %

This means that the total 4.4% of water flow is being passed through the cooling tower without the heat exchange.

Example 5-1: Let's assume that the % by-pass wall water was 4% and compare the tower demand using the example 4-1.

(Solution)
Since the 4% of water flow rate is considered not to be completely cooled, the cooling tower has to remove the heat for the original heat load duty and reduced water flow rate. Therefore, it is nature that the cooling range is increased and the tower demand must be based on these new cooling range and cold water temperature.

Original Range, R_1 = HWT - CWT = $tw_2 - tw_1$ = 104 - 89 = 15ºF
Water Flow Rate in Pound, L_1 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) =

133,333.3 lb/min
Heat Load, $D_1 = L_1 \times R_1$ = 133,333.3 x 15 = 2,000,000 BTU/min
Heat Load, $D_2 = D_1$ = 2,000,000 BTU/min
Tower Water Flow Rate, L_2 = Water Circulation Rate x (1 - % By-Pass Wall Water / 100) x (500 / 60)
= 16,000 x (1 - 4 / 100) x (500 / 60) = 128,000.0 lb/min
Range, $R_2 = D_2 / L_2$ = 2,000,000 / 128,000 = 15.625ºF
= $L_1 \times R_1$ / {L_1 x (1 - %By-Pass Wall Water / 100)}
= R1 / (1 - % By-Pass Wall Water / 100) = (104 - 89) / (1 - 4 / 100) = 15.625ºF

Tower Cold Water Temp., $CWT_2 = CWT_1 + R_1 - R_2$ = 89 + 15 - 15.625 = 88.375ºF

(This relation is obtained from the below derivations;

Heat Load, $D_1 = L_1 \times R_1 = L_1 \times (HWT_1 - CWT_1)$
Heat Load, $D_2 = L_2 \times R_2 = L_2 \times (HWT_1 - CWT_2)$
From the relation of $D_1 = D_2$,
$L_1 \times (HWT_1 - CWT_1) = L_2 \times (HWT_1 - CWT_2)$
$L_1 / L_2 \times (HWT_1 - CWT_1) = HWT_1 - CWT_2$

Therefore, CWT_2 = $HWT_1 - L_1 / L_2 \times (HWT_1 - CWT_1)$
= $HWT_1 - L_1$ / [L_1 x (1 - % By-Pass Wall Water / 100) x ($HWT_1 - CWT_1$]
= HWT_1 - 1 / (1 - % By-Pass Wall Water / 100) x ($HWT_1 - CWT_1$)
= $HWT_1 - R_2$
[($HWT_1 - CWT_1$) / (1 - % By-Pass Wall Water / 100) = R_2]
= $CWT_1 + R_1 - R_2$

Or from the condition that the design hot water temperature must be equal regardless By-Pass Wall Water,

$HWT = CWT_1 + R_1 = CWT_2 + R_2$
$CWT_2 = CWT_1 + R_1 - R_2$

Also, it is obvious that the cold water temperature through the cooling tower when by-pass wall water is being considered will be lower than when not to consider the by-pass wall water.)

Air Mass Flow Rate, $G_2 = G_1$ = 80,848 lb/min, Liquid to Gas Ratio, $L/G_2 = L_2 / G_2$ = 128,000.0 / 80,848 = 1.5832

WATER SIDE			AIR SIDE		ENTH DIFF.
Descriptions	tw (ºF)	hw (But/Lb)	Description	ha (Btu/Lb)	1/(hw-ha)
tw_1 + 0.1 x R	89.94	55.8639	ha_1 + 0.1 x L/G x R	46.1645	0.1031

tw_1 + 0.4 x R	94.63	62.7545	ha_1 + 0.4 x L/G x R	53.5858	0.1091
tw_1 + 0.6 x R	97.75	67.8345	ha_1 + 0.6 x L/G x R	58.5334	0.1075
tw_1 + 0.9 x R	102.44	76.2814	ha_1 + 0.9 x L/G x R	65.9547	0.0968
Sum of 1 / (hw - ha)					0.4165
Total Tower Demand (NTU) = Cooling Range x Sum of 1 / (hw - ha)					1.6270

This example shows that the tower demand is increased by about 9.44% when the by-pass wall water is considered. That is, the degree of cooling difficulty with the consideration of by-pass wall water is higher than the degree with the ignorance of by-pass wall water.

Example 5-2. The example number 18 was based on the assumption that the heat load is constant for the increase of water flow rate. Estimate the cold & hot water temperature under the assumption that the cooling range is constant for the increase of water flow rate to 20,000 from 16,000 GPM, and the assumption of 4% of total water is being by-passed without the heat removal through the tower.

(Solution)
Range, $R_1 = tw_2 - tw_1$ = 104 - 89 = 15.0ºF
Tower Water Flow Rate in Pound, L_1 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = 133,333.3 lb/min
Liquid to Gas Ratio, $L/G_1 = L_1 / G_1$ = 133,333.3 / 80,848 = 1.6492
Air Mass Flow Rate, $G_1 = G_2$ = 80,848 lb/min
Tower Water Flow Rate in Pound, L_2 = Water Flow Rate x (1 - % By-Pass Wall Water / 100) x (500 / 60)
= 20,000 x (1 - 4 / 100) x (500 / 60) = 160,000.0 lb/min
Liquid to Gas Ratio, $L/G_2 = L_2 / G_2$ = 160,000.0 / 80,848 = 1.9790
Range, $R_2 = (tw_2 - tw_1)$ / (1 - % By-Pass Water / 100) = (104 - 89) / 0.96 = 15.625ºF

The value of NTU is same as a value calculated in the example 4-1.
Calculate a value of "C" of tower characteristic for the design conditions as follows;

$C = KaV/L / (L/G)^{-m} = KaV/L \times (L/G)^{m} = 1.4866 \times (1.6492)^{0.8} = 2.21825$

Calculate a new tower characteristic for the increased water flow.
New Tower Characteristic = C x $(L/G)^{-m}$ = 2.21825 x $(1.9790)^{-0.8}$ = 1.2848
Iterate until the value of new characteristic is equal to the new NTU varying the value of approach. New Cold Water Temperature = Wet Bulb Temperature + New Approach

DATA INPUT	
INPUT PARAMETERS	DATA
1. Design L/G	1.6492
2. Design HWT (°F)	104.00
3. Design CWT (°F)	89.00
4. Wet Bulb Temp. (°F)	80.00
5. Characteristic Slope	-0.800
6. Altitude (Feet)	0
7. Design Cooling Range (°F	15.00
8. Design Approach (°F)	9.00
9. Design KaV/L	1.4866
10. C VALUE IN CHA.	2.2183
11. By-Pass Wall Water	4.00%
12. New L/G	1.9790
13. New Cooling Range (°F)	15.625
14. New KaV/L	1.2849

CALCULATION OF KaV/L @ DESIGN			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	56.6478	46.1645	0.09539
0.4	63.3426	53.5859	0.10249
0.6	68.2591	58.5335	0.10282
0.9	76.4013	65.9549	0.09573
			0.39643
CALCULATION OF KaV/L @ A POINT			
	WATER SIDE	AIR SIDE	1/(Hw-Ha)
0.1	60.6750	46.7829	0.07198
0.4	68.1847	56.0595	0.08247
0.6	73.7287	62.2438	0.08707
0.9	82.9604	71.5204	0.08741
		1.2849	0.32894
		12.331	
New Cold Water Temperature			92.331
New Hot Water Temperature			107.331

Finally, compute the cold water temperature with the result of iteration as follows;

New CWT through Tower = WBT + New Approach + Design Range - Actual Range = 80 + 12.331 + 15 - 15.625 = 91.706
Final CWT = (New CWT through Tower x Water Flow through Tower + New HWT x By-Pass Wall Water Flow) / Total Water Flow Rate = (19,200 x 91.706 + 800 x 107.331) / 20,000 = 92.331°F
Water Flow Rate through Tower = Alternative Water Flow x (1 - % By-Pass) = 20,000 x (1 - 0.04) = 19,200 GPM
By-Pass Wall Water Flow = Alternative Water Flow x % By-Pass = 20,000 x 0.04 = 800 GPM

Final HWT = Final CWT + Heat Build Up from Heat Exchanger (Range) = 92.331 + 15.0 = 107.331°F

Or, Final HWT = New CWT through Tower + New Range through Tower = 91.706 + 15.625 = 107.331°F

Therefore, the hot water temperature when to consider the by-pass wall water is higher than example no. 4-3 by 0.321°F.

Download the example file (exe5_2.zip)

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6. Pressure Drops in Cooling Tower

The air pressures are always dropped in the area where the direction of air flow is changed or the velocity of air flow is decreased suddenly. Representative areas where the pressure losses of air are occurring in the induced draft counter flow cooling tower are as follows;

- Air Inlet (Entrance Loss)
- Fill
- Water Distribution Piping
- Drift Eliminator
- Fan Inlet (Sometimes called plenum losses)

Most of air pressure drops at all the areas excepting fill section can be easily calculated as per the well known formula of K x (Air Velocity / 4008.7)^2 x Density Ratio. K is a pressure drop coefficient and depends on the shape of obstruction laid in the air stream. Density ratio is an actual air density divided by 0.075 lb/ft^3 @ 70^{o}F dry air conditions. In cooling tower, these pressure losses are called "Static Pressure Loss", just "Static Pressure", or "System Resistance. The performance of cooling tower fans depends on the calculation degree of static pressures at the cooling tower.

The minimum value of pressure drop coefficient at the air inlet is including the two turns of air stream directions and is 1.0 for a hypothetical perfect bell inlet. As a guide line, K values at the air inlet are as below;

A) Without Louvers

Square edge beams and square columns: 1.5
Rounded beams (R = 0.04 x H) and columns (R = 0.04 x W): 1.3
Tapered beams and columns, 30^o, H = 0.1 x W: 1.2

B) With Louvers

Large, widely spaced louvers: 2.0 to 3.0
Narrow, small louvers: 2.5 to 3.5

In most cases, the pressure drops at the water distribution piping zone are included into the pressure drops at drift eliminators because the drift eliminators are installed onto the water distribution pipes or within 2 feet from pipes. In this case, K values is in the range of 1.6 to 3.0. Of course, it must be based on the data provided by manufacturer. The pressure drop coefficient at the fan inlet will be discussed in the examples related to the fans later again, but it is in the range of 0.1 to 0.3.

Pressure Drop

$= k\dfrac{\rho v^2}{2g}$ = k x (1 / 2) x Air Density x V^2 / 115,820 (lb/ft^2)

= k x 0.1922 x (1 / 2) x Air Density x V^2 / 115,820 (inch WG = inch Aq. = inch Water)

or,

= k x 0.1922 x (1 / 2) x (Density Ratio x 0.075) x V^2 / 115,820 (inch WG = inch Aq. = inch Water)

= k x 0.1922 x (1 / 2) x 0.075 / 115,820 x V^2 x Density Ratio (inch WG = inch Aq. = inch Water)

= k x V^2 x 1 / 16,069,371 x Density Ratio

= k x V^2 x 1 / 4008.72 x Density Ratio

= k x (V / 4008.7)^2 x Density Ratio

where,

k: Pressure Drop Coefficient

r: Air Density lb/ft^3

V: Air Velocity, ft/min

g: Acceleration Gravity, ft/min^2 (1g = 32.172 ft/sec^2 = 115,820 ft/min2)

Density Ratio: Actual Air Density / 0.075

(1 lb/ft^2 = 0.1922 inch WG)

Therefore, a constant of 4008.7 is obtained from above in order to convert the unit of pressure drop to inch Aq. using the ft/min unit of air velocity and lb/ft^3 unit of air density.

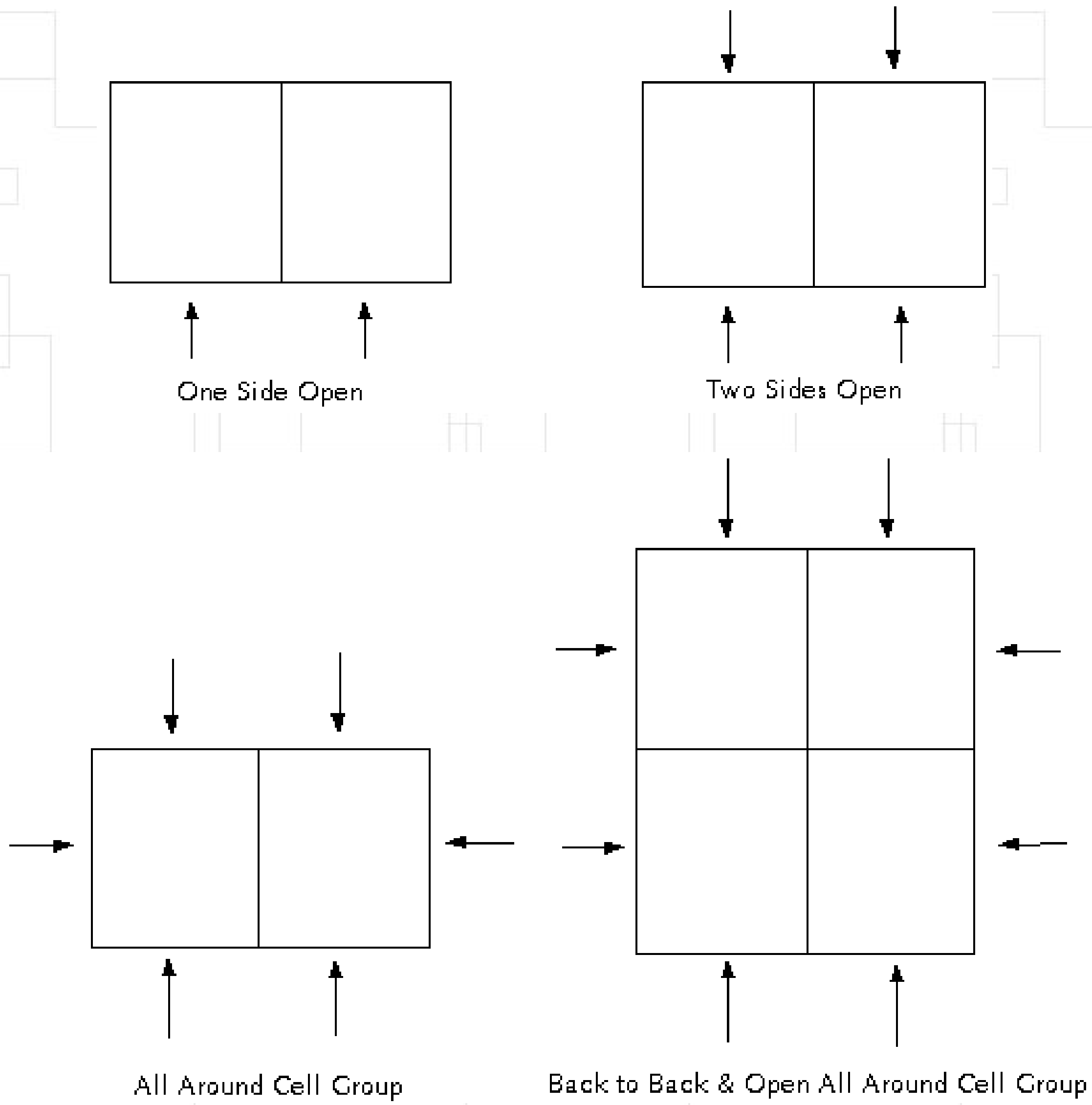
It is important to predict the obstructions in the air stream. The obstructions which must be considered in designing the cooling tower are as follows;

- Obstruction at the air inlet area
- Obstruction at the fill

The obstructions at the air inlet area are the area of preventing the air flow and are a summation of area projected to the air inlet with the columns, beams, or bracing, etc. There is no need to consider the air flow obstruction due to the inlet louvers. The obstruction at the fill is a plain area which is not filled due to the columns or bracing, etc.

The types of air inlet for the counter flow induced draft cooling tower as below are being used.

- One Side Open: This arrangement is useful for the area where the obstruction to be able to disturb the air flow or to increase the inlet wet bulb temperature due to the adjacent building or the heat sources to be able to affect the entering wet bulb temperature are located to the one side of cooling tower. When to design the cooling tower with this arrangement of air inlet, a special attention is required for the even air distribution into the fill section.
- Two Side Open & Ends Closed: This arrangement is most general for the industrial cooling towers.
- All Around Cell Group
- Back To Back & Open All Round: This is useful for a case where the area is limited.



Example 6-1. Determine the pressure drop at the air inlet for the below given conditions.

Given,
Cell Length: 42.0 feet
Cell Width: 42.0 feet
Air Inlet Height: 15.0 feet
Number of Spray Nozzle: 196 each (Center to Center Distance of Nozzles: 3 feet)
Water Flow Rate: 12500 GPM
Exit (Entering) Water Temperature: 89^{o}F
Inlet (Leaving) Water Temperature: 104^{o}F
Fill Depth: 4 feet
Fill Flute Size: 19 mm
Entering Wet Bulb Temperature: 80^{o}F
Relative Humidity: 80.0%
Site Elevation: 0 feet
Exit Air Temperature: 97^{o}F
Arrangement of Air Inlet: Two Sides Open & Ends Closed
Material of Tower Framework: Wood
Type of Air Inlet Louver: Large, Widely Spaced

(Solution)
In order to obtain the air mass flow, the following calculation must be first accomplished. The actual cooling range through the tower must be calculated because there is a by-pass wall water in the tower.

New Tower Range = Design Range / (1 - % by pass wall water / 100)
(Note: This was already discussed in example 5-2.)
% By-Pass Water Calculation is as follows:

1) Water Flow Rate per Nozzle = Design Water Flow Rate / Total Number of Nozzles= 12,500 GPM / 196 = 63.78 GPM/Nozzle

2) By-Pass Wall Water from Spray Nozzles;

By Pass Wall Water	= [{(Cell Length / Center to Center Distance of Nozzle) - 2} x 2 + {(Cell Width / Center to Center Distance of Nozzle) - 2} x 2] x 10% x GPM / Nozzle + 4 Nozzles x 20% x GPM / Nozzle = [{(42 / 3) - 2} x 2 + {(42 / 3) - 2} x 2] x 10% x 63.776 + 4 x 20% x 63.776 = 357.14 GPM

3) By-Pass Column Water due to Spray Nozzles near to Tower Internal Columns

By-Pass Column Water	= {(Cell Length / Bay Distance) - 1} x {(Cell Width / Bay Distance) -1} x 4 Nozzles x 5% x GPM / Nozzle = {(42 / 6) - 1} x {(42 / 6) - 1} x 4 x 5% x 63.776 = 459.18 GPM
% By-Pass Water	= (By-Pass Wall Water + By-Pass Column Water) / GPM / 2 x 100 (%) = (357.14 + 459.18) / 12,500 / 2 x 100 = 3.265%

Therefore, the actual range through tower is obtained from relation of Design Range / (1 - % By-Pass Water / 100)

Actual Range = (104 - 89) / (1 - 3.265 / 100) = 15.5063
A value of L/G is obtained from the equation of $ha_2 = ha_1$ + L/G x New Tower Range.
L/G = (ha_2 - ha_1) / New Tower Range
Air Enthalpy at Exit (97ºF) = 66.5773 Btu/lb
Air Enthalpy at Inlet (80ºF) = 43.6907 Btu/lb
Therefore, L/G = (66.5773 - 43.6907) / 15.5063 = 1.4760

The air mass is calculated from the relation of G = L / (L/G). Here the value of L is a net water flow rate through the cooling tower. That is, L = Design Water Flow Rate x (500 / 60) x (1 - % By-Pass Water / 100) = 12,500 x (500 / 60) x (1 - 3.265 / 100). (Note: (500 / 60) is a constant to covert water flow rate in GPM to lb/min unit.) Then, the value of air mass flow = 12,500 x (500 / 60) x (1 - 3.265 / 100) / 1.4760 = 68,271.5 lb/min

Second, let's calculate the area of obstruction in the air inlet. In case of wood structure, one bay (between center of columns) is based on 6 feet and the traversal member is based on 6 feet in the height. Therefore, the number of bay for the 42 feet of cell length is 7 and the width of column is 4 inch. In the traversal member, two beams are required for this air inlet height.

Area of Obstruction due to Columns = No. of Bay x Width of Column x Air Inlet Height x No. of Air Inlet = 7 x (4 / 12) x 15 x 2= 70 ft^2

Area of Obstruction due to Traversal Members = No. of Members x Height of Members x Cell Length x No. of Air Inlet = 2 x (4 / 12) x 42 x 2= 56 ft^2

Total Area of Obstructions = 70 + 56 = 126 ft^2

Overall Area of Air Inlet = Cell Length x Air Inlet Height x No. of Air Inlet = 42 x 15 x 2= 1,260 ft^2

% Obstruction @ Air Inlet = Total Area of Obstructions / Overall Area of Air Inlet x 100 (%)= 126 / 1,260 x 100(%)= 10.0%
Net Area of Air Inlet = 1,260 - 126 = 1,134 ft^2

Air Density and Specific Volume @ Air Inlet must be based on the dry bulb temperature at a relative humidity, not on wet bulb temperature. Let's find a dry bulb temperature from Psychrometric chart or from the following computer calculation method.

CALCULATION OF INLET DRY BULB TEMPERATURE		
Altitude	0.00	feet
Relative Humidity	80.0%	
Wet Bulb Temperature @Inlet	80.00	°F
Enthalpy @ WBT	43.6907	Btu/Lb dry air
Equivalent Enthaply	43.6907	Btu/Lb dry air
Air Density @Tower Inlet	0.0718	Lb/ft^3
Air Specific Volume @Tower Inlet	14.2230	ft^3/Lb dry air
Equivalent Dry Bulb Temperature	85.242	°F

The dry bulb temperature corresponding 80% RH at 80ºF WBT is 85.24ºF. (Note: Some engineers are using the air density and specific volume at the air inlet using the web bulb temperature. This is totally wrong and is quite different from the value at the dry bulb temperature & relative humidity.)

Specific Volume @ 85.24 DBT & 80% RH = 14.2230 ft^3/lb
Airflow Volume @ Air Inlet = Air Mass Flow x Specific Volume @Air Inlet= 68,271.5 x 14.2230= 971,028 ft^3/min

(For reference, the specific volume at the given wet bulb temperature is 14.1126 ft^3/lb and airflow volume becomes 963,485 ft^3/min. Compare this value with above airflow volume.)

Air Velocity @ Air Inlet = Airflow Volume @ Air Inlet / Net Area of Air Inlet= 971,028 / 1,134 = 856.29 ft/min (FPM)
Air Density @ 85.24 DBT & 80% RH = 0.0718 lb/ft^3
Pressure Drop Coefficient for this arrangement = 2.5

Then, pressure drop is obtained from below:

Pressure Drop = K $(V / 4008.7)^2$ x Density Ratio= 2.5 x $(856.29 / 4008.7)^2$ x (0.0718 / 0.0750)= 0.1092 inch Aq.
(For reference, the air density at the given wet bulb temperature is 0.0724 lb/ft^3. Compare this with the previous value of air density.)

Download the example file (exe6_1.zip)

Example 6-2. Determine the pressure drop at the fill for the same example 6-1.

(Solution)
First, it is to calculate the average air velocity through the fill. The reasons why the average air velocity must be calculated are based the assumptions below;

1) The heat exchange in the rain zone is negligible and there is no change in the air between the entering air into the tower inlet and into the bottom of fill.
2) The heat is completely exchanged at the fill section & water distribution zone.
3) The exit air from the fill is 100% saturated and the heat of exit air transferred from the water is considered as an adiabatic process.

To calculate the average air velocity, the average air volume and specific volume through the fill must be calculated.

Average Specific Volume = 2 / (1 / Specific Volume @ Tower Inlet Temp. + 1 / Specific Volume @ Tower Exit Air Temp.)

Specific Volume @ 85.24 DBT & 80% RH = 14.2230 ft^3/lb
Specific Volume @ 97.0 DBT & 100% RH = 14.9362 ft^3/lb (The exit temp. was guessed.)

Therefore, the average specific volume at the fill = 14.5709 ft^3/lb
Then, the average air volume at the fill is obtained from Average Specific Volume x Air Mass Flow. That is, the average air volume at the fill = 994,776.8 ft^3/min
Average Air Velocity = Average Air Volume / Net Fill Area
Net Fill Area = (Cell Length x Cell Width) x (1 - % Fill Obstruction / 100)
% Fill Obstruction = (Sectional Area of Column x Number of Columns) / (Cell Length x Cell Width) x Margin x 100(%)= (4 x 4 / 144 x 7 x 7) / (42 x 42) x 3.6 x 100 (Note: Safety margin for wood tower is about 3.6)= 1.11%
Therefore, the net fill area = (42 x 42) x (1 - 1.11 / 100) = 1,730.7 ft^2
Average Air Velocity @Fill = Average Air Volume @Fill / Net Fill Area= 574.78 ft/min

Second, the water loading calculation is required as follows;

Water Loading = Tower Water Flow Rate / Net Fill Area
= Design Water Flow Rate x (1 - % By-Pass Water / 100) / Net Fill Area = 12,500 x (1 - 3.27 / 100) / 1,730.7 = 6.99 GPM/ft^2

Air Density @ 85.24 DBT & 80% RH = 0.0718lb/ft^3
Air Density @ 97.0 DBT & 100% RH = 0.0696 lb/ft^3
Then, average air density at fill = 0.0707 lb/ft^3

Now, all the parameters are ready to compute the pressure drop at the fill. The calculation of pressure drop at the fill is very complicated and it is impossible to predict the pressure drop if the formula for the pressure drop is not available. The formula of calculating the pressure drop at the fill is a proprietary data of fill maker.

Pressure Drop @Fill = 0.3011 inch WG.

Example 6-3. Determine the pressure drop at the drift eliminator per the given conditions in example 6-1.

(Solution)
In general, the obstruction area in the drift eliminator is considered same as the fill obstruction area. Therefore, the net drift eliminator area = (42 x 42) x (1 - 1.11 / 100) = 1,730.7 ft^2. There is no change in the air mass flow through out the cooling tower. Therefore, the value of air mass flow is same as the above obtained value of 68,271.47 lb/min. The air density and specific volume at 97^{o}F 100% RH are 0.0696 lb/ft^3, 14.9362 ft^3/lb respectively.

Then, the air volume at the drift eliminator is obtained from Specific Volume x Air Mass Flow. That is, the air volume at the drift eliminator = 1,019,716.3 ft^3/min
Air Velocity @ Drift Eliminator = Airflow Volume @ Drift Eliminator / Net Area of Drift Eliminator
Air Velocity @ Drift Eliminator = 589.19 ft/min
Pressure Drop Coefficient for a general module type of drift eliminator = 1.6 to 2.0

Then, pressure drop is obtained from below:

Pressure Drop = K $(V / 4008.7)^2$ x Density Ratio= 1.8 x $(589.19 / 4008.7)^2$ x (0.0696 / 0.0750)= 0.0361 inch Aq.

Example 6-4. Determine the pressure drop at the fan inlet of fan stack per the given conditions in example 6-1. Let's assume that the 28 feet of fan in the diameter with the 88 inch of air seal disk was used and the fan inlet shape is rounded. (R/D = 0.10)

(Solution)
The pressure drop is occurring at the fan inlet of fan stack unless the shape of fan inlet is elliptical bell and no obstruction under the fan in case of induced draft fan arrangement. The following table could be applied to the cooling tower fan stack as a guide line in choosing the pressure drop coefficient.

Inlet Shape	K	Extra Factor	Total Factor
Elliptical (L/D = 1:1.5)	0.00	0.10	0.10
R/D = 0.15	0.00	0.10	0.10
R/D = 0.10	0.04	0.14	0.18
R/D = 0.05	0.13	0.15	0.28
R = 0	0.40	0.20	0.60

In practice, it is quite essential to add some extra to the above K value since there are a lot of obstructions under the fan. It is considered that there is no change in the heat from the drift eliminator to the fan. Accordingly, the specific volume at the fan is same as the value at the drift eliminator. Let's calculate the net fan area.

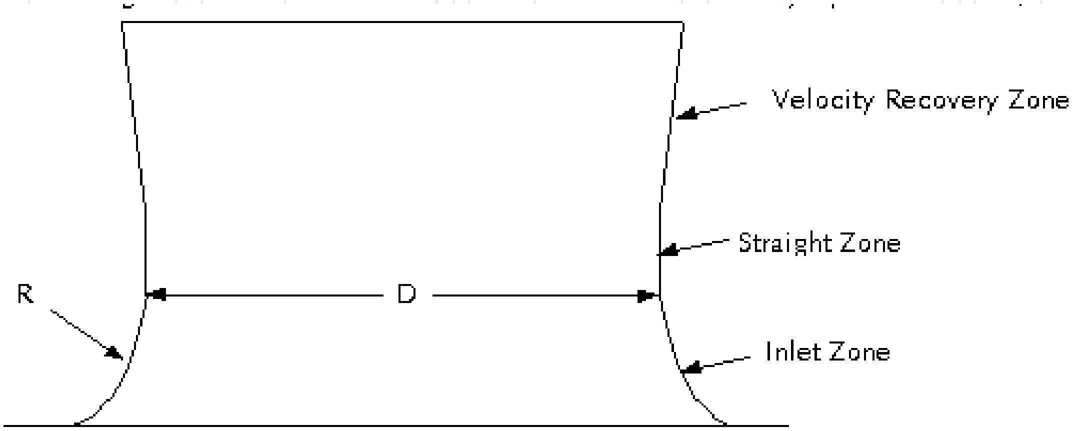
Fan Net Area = 3.1416 / 4 x (Fan Dia^2 -Air Seal Disk^2)= 573.52 ft^2
Air Velocity @ Fan = Airflow Volume @ Fan / Net Fan Area= 1019716.3 / 573.52 = 1778.00
Air Velocity @ Fan = 1,778.00 ft/min

(Note: The air volume at fan is same as the air volume at the drift eliminator.)

Then, pressure drop is obtained from below:
Pressure Drop = K (V / 4008.7)^2 x Density Ratio= 0.18 x (1778.0 / 4008.7)^2 x (0.0696 / 0.0750) = 0.0329 inch Aq.

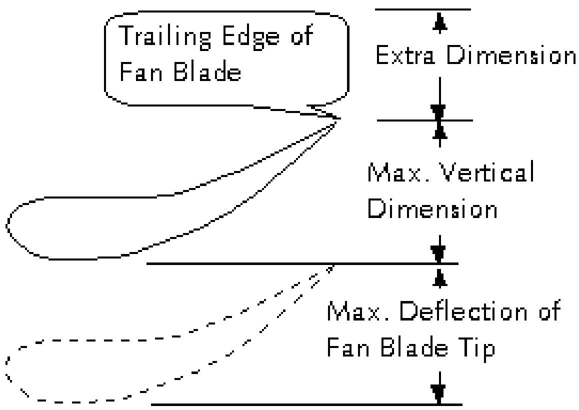
7. Velocity Recovery at Fan Stack

The fan stacks are used for maximizing the fan efficiency, for preventing the reverse running of fan, and for minimizing the discharge air recirculation. The fan stack consists of three major parts as follows;



As the air is induced out of the plenum chamber beneath the fan, it tends to flow very predictable streamline into the fan stack. The inlet section of fan stack must be designed to induce the air smoothly and to minimize the air turbulence. In most cases, R/D = 0.15 or R/D = 0.10 is recommended.

The straight zone of fan stack is also very important to the fan performance. The fan blade are deflected downward during the operation due to the axial load onto the fan blade surface. Therefore, the movement of fan blade tips must be limited within the straight zone of fan stack. The minimum height of straight zone in the fan stack is a summation of the vertical dimension at the maximum blade pitch angle, the maximum deflection of fan blades tip, and some extra allowance.



Once the air properly directed into the fan stack, the close tip clearance must be kept. The greater the tip clearance the less efficient the fan. The space between the fan tip and fan stack allows the creation of air vortex at the blade tips which shorten the effective length of the blade, reducing the fan performance. (A vortex from upper section of the fan blades back to the low pressure area beneath the fan allows; this produces a lowered air flow rate and reduced fan efficiency.)

Close tip clearance minimizes the magnitude of the disturbances, maximizing the fan performance. However, the tip clearance must be designed to accommodate the wind-affected deformation of the fan stack, thermal expansion of the fan blades, and the possible build-up of ice inside the fan stack under the reverse fan operation. Fans are often installed in cooling tower with the tip clearance of up to 2 inches because of the manufacturing tolerances inherent in large fiberglass stack segments.

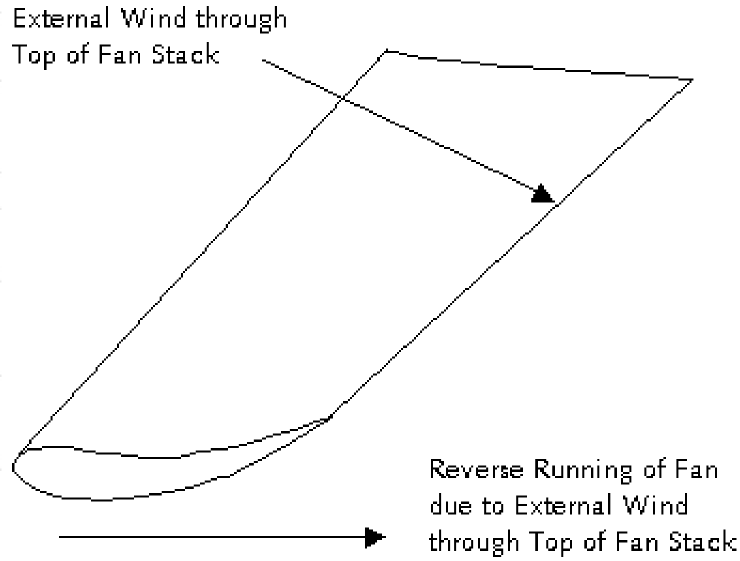
If the tip clearance is larger than the below maximum values, a pressure loss due to the increase of fan stack sectional area will occur. A rapid decline in the fan efficiency due to the decrease of total pressure and airflow will be resulted in and the brake horsepower under this situation will be slightly decreased.

Fan Diameter	Minimum	Maximum
up to 9 feet	1/4"	1/2"
10 - 14 feet	3/8"	3/4"
16 - 20 feet	1/2"	1"
22 - 30 feet	3/4"	1-1/4"

The power consumption is generally decreased as much as the tip clearance is increased, since the volumetric air flow rate is significantly decreased. The efficiency at the larger tip clearance is decreased. The efficiency of fan at the larger tip clearance is decreased, because the input power is not reduced as much as the airflow is decreased.

At a slightly tapered exit cone the velocity pressure compared to the plane of fan is significantly reduced. The recovery of velocity pressure is converted into static regain which lowers the total pressure requirements of the fan.

A poorly designed and fabricated fan stack is a potential cause of poor air distribution, low fan stack efficiency, and significant vibration of fan stack due to the resonant frequency of fan. For high efficient fan stack design, the normal height of total fan stack is ranged in the 0.6 to 1.0 to the fan diameter. The taller height of fan stack than 1 x fan diameter does not useful for the velocity recovery and only makes the problems like the heavy fan deck load and higher wind load. The short height of fan stack is making a problem of the reverse running of fan due to the external wind under the situation of the fan is off.



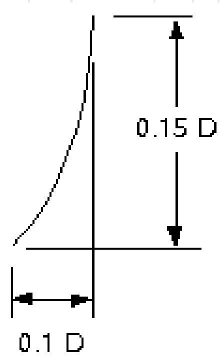
If the power is applied to the motor under the fan is reversibly running, all the mechanical equipment as well as fan, gear reducer, and coupling shaft shall be broken.

So, the height of fan stack must be taller than the fan diameter by at least0.6 times and sometimes the back stop device, which protects the reverse running of fan, shall be installed to the motor or gear reducer.

Example 7-1. Estimate the height of inlet, straight, and velocity recovery zones of fan stack for the 28 feet of fan in the diameter and 10 feet of fan stack in the height.

(Solution)
1) Fan Inlet Zone

Let's use the R/D = 0.15.



Inlet Zone Height
= 0.15 x Fan Dia.
= 0.15 x 28 feet
= 0.15 x 28 x 12
= 50.4 inch

This inlet shape was proven to be the ideal inlet shape, because the air flows along the wall with a uniform velocity and to the fan with the slightest possible turbulence.

2) Straight Zone

- Vertical Dimension of Blade Tip @ Max. Pitch Angle: 5.73 inch
- Maximum Deflection of Blade Tip: 14 inch
- Extra Dimension from the trailing edge of blade: 6 inch
- Then, the height of straight zone is 25.73 inch (= 5.73 + 13 + 6)

3) Velocity Recovery Zone

Velocity Recovery Zone Height = Total Fan Stack Height - Fan Inlet Zone Height - Straight Zone Height =10 x 12 - 50.4 - 25.73 =43.87 inch

Example 7-2. Calculate the velocity recovery at the above given design conditions.

(Solution)
There is no regulation in estimating the velocity recovery at the fan stack, which is generally accepted by every one, and the designers have to decide it with the experience. For the angle of taper, 7 degree is most efficient through the a lot of tests. The following formulas could be used for estimating the velocity recovery.

1) Formulated by Hudson Products Corp.

Basically, Hudson's velocity recovery formula is based on the 7 degree of taper angle and 70% of fan stack efficiency.

Velocity Recovery = 70% of Fan Stack Efficiency x (Velocity Pressure @Fan - Velocity Pressure @Top of Fan Stack)

2) Formulated by MRL Corp.

The degree of taper at the venturi zone is same as Hudson, but the fan stack efficiency is differently obtained as 0.8 - 0.2 x (Venturi Height / Fan Diameter)
Velocity Recovery = 0.8 - 0.2 x (Venturi Height / Fan Diameter)
x (Velocity Pressure @Fan - Velocity Pressure @Top of Fan Stack)

In order to obtain a velocity pressure at the top of fan stack for a given fan stack, the area at the top of fan stack must be calculated first as follows;

Diameter of Fan Stack Top = Fan Diameter + 2 x Tan 7^o x Venturi Height
Area of Fan Stack Top = 0.7854 x (Diameter of Fan Stack Top^2 - Air Seal Disk^2)
= 0.7854 x [28 + 2 x Tan 7^o x 43.87 / 12)^2 - (88 / 12)^2] = 613.6 ft^2

Air Velocity @Fan Stack Top = Air Volume @ Fan / Area of Fan Stack Top = 1019716.289 / 613.6 = 1,661.86 ft/min

Velocity Pressure @Fan Stack Top = (Air Velocity @ Fan Stack Top / 4008.7)^2 x (Air Density / 0.075) = (1661.86 / 4008.7)^2 x (0.0696 / 0.0750) = 0.1594 inch Aq.

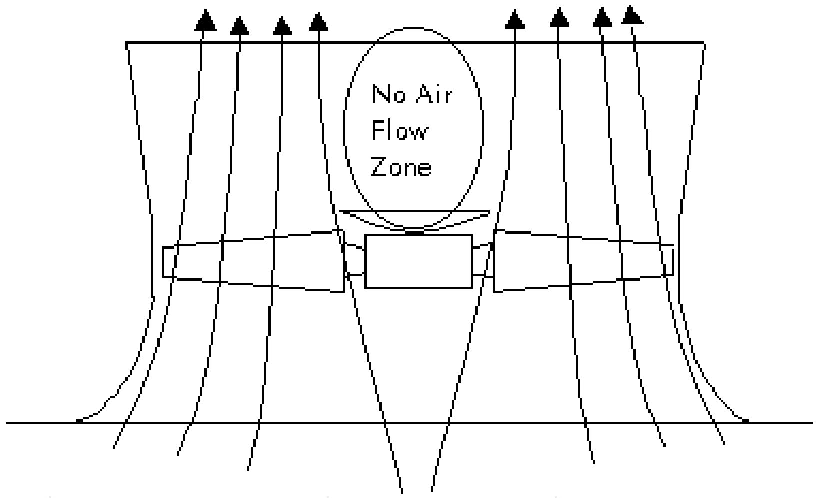
Let's fan stack efficiency using the formula of MRL Corp.

Fan Stack Efficiency = [0.8 - 0.2 x (Venturi Height / Fan Diameter)] x 100(%) = {0.8 - 0.2 x [(43.87 / 12) / 28]} x 100 = 77.4%

Velocity Pressure @ Fan = (Air Velocity @Fan / 4008.7)^2 x (Air Density @Fan / 0.075) = (1778.0 / 4008.7)^2 x (0.0696 / 0.0750) = 0.1825 inch Aq.

Velocity Recovery = Fan Stack Efficiency x (Velocity Pressure @Fan - Velocity Pressure @Fan Stack Top) = 0.774 x (0.1825 - 0.1594) = 0.0178 inch Aq.

(Note: The reason why the area of air seal disk must be subtracted from the above equation in calculating the area of fan stack top is because the air streamline does not exist above the air seal disk.)



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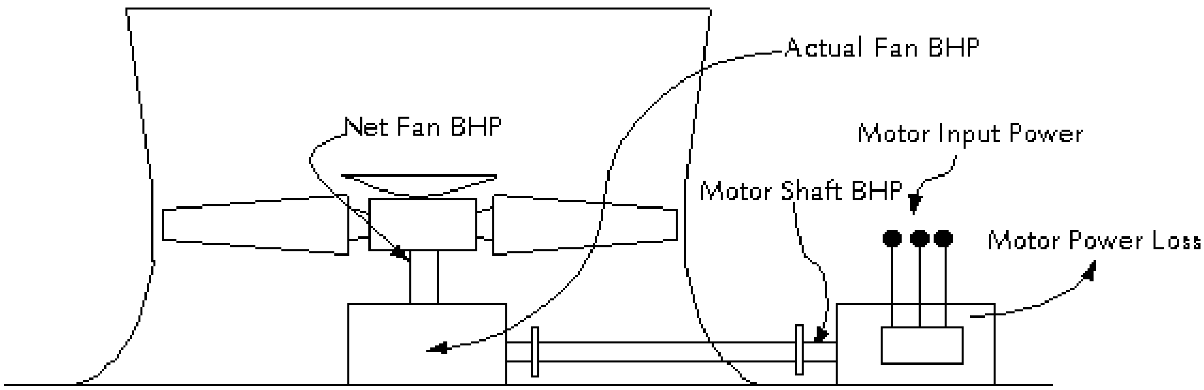
8. Motor Power Sizing



The fan BHP shown on the fan rating sheet is the net fan brake horsepower based on the ideal conditions of fan test. The actual operating conditions of cooling tower is quite different from the test conditions of fan maker and the actual fan efficiency will be different from the environmental factor like the inlet and exit air flow conditions, tip clearance, obstructions to air flow, plenum geometry, etc. Therefore, a proper environmental correction factor should be considered to both total pressure and horsepower.

Ventilatoren Sirroco Howden who is supplying the fans had published a paper about the influence on the fan performance as follows;

1) Influence of Fan Inlet Shapes

Refer to example 6-4 how much the resistance is increased for the inlet shape other than R/D = 0.15.

2) Influence of Obstacles present in the air flow of the fan

The influence of fan performance due to the obstacles under the fan depends on the ratio of distance of leading edge of fan blade from the obstacles and the fan stack throat diameter, and on the ratio of area of obstacles and area of fan stack throat. The smaller of the ratio of distance and the larger of the ratio of area, the higher of resistance correction factor. In most cases, the additional pressure drop coefficient due to the obstacles is within 0.1 to 0.15.

3) Influence of Tip Clearance

VSH is describing that the tip clearance less than 1% to the fan diameter does not effect to the fan performance. The author has a different opinion against the publication of VSH and suggests to use the following guideline.

Tip Clearance	Multiplying Factor	Tip Clearance	Multiplying Factor
<= 0.1% to Fan Dia	1.000	<= 0.5% to Fan Dia.	0.950
<= 0.2% to Fan Dia.	0.990	<= 0.6% to Fan Dia.	0.925
<= 0.3% to Fan Dia.	0.975	<= 0.7% to Fan Dia.	0.900
<= 0.4% to Fan Dia.	0.965	<= 0.8% to Fan Dia.	0.875

The additional static pressure increase due to the obstacles could be obtained as adding the pressure drop factor due to the obstacles. The influence of fan performance due to the tip clearance could be achieved as adjusting the power transmission efficiency, which shall be discussed.

Example 8-1. Determine the fan brake horsepower and fan static efficiency for the design conditions dealt above under the assumption that the fan total efficiency is 80.1%.

(Solution)
Fan BHP = Air Volume @Fan in ACFM x Total Pressure in inch Aq. / (Fan Total Efficiency x 6356), or = Air Volume @Fan in ACFM x Static Pressure in inch Aq. / (Fan Static Efficiency x 6356)

Total Static Pressure = PD @Air Inlet + PD @Fill + PD @Drift Eliminator + PD @Fan Inlet = 0.1092 + 0.3011 + 0.0361 + 0.0329 = 0.4793 in Aq.

(Note that the static pressure for rating the fan must be a value of Total Static Pressure - Velocity Recovery unless the venturi height is input to the fan rating program. The suggestion is to use this method instead of inputting the venturi height into the fan rating program, since the efficiency of fan stack used by the fan makers is different each other.)

Total Pressure = Total Static Pressure + Velocity Pressure - Velocity Recovery = 0.4793 + 0.1825 - 0.0178 = 0.6439 inch Aq.
Fan BHP = 1019716.28 x 0.6439 / (0.801 x 6356) = 128.98 BHP
Fan Static Efficiency = Air Volume @ Fan in ACFM x Static Pressure in inch Aq. / (Fan BHP x 6356) = 1019716.28 x (0.4793 - 0.0178) / (128.98 x 6356) = 57.4%

Example 8-2. Determine the motor input power based on the example 8-1.

(Solution)
Actual Fan BHP = Net Fan BHP / System Environmental Correction Factor= 128.98 / 0.95= 135.77 BHP

Motor Shaft BHP = Actual Fan BHP / Efficiency of Power Transmission of Gear Reducer= 135.77 / 0.96= 141.43 BHP

The gear reducer wastes 3 to 5% of motor power, which depends on the number of reduction. The factors influencing the efficiency of gear reducer are:

- Frictional loss in bearings
- Losses due to pumping or splashing the lubricant oil
- Frictional loss in gear tooth action.

All these losses shall be turned to the heat build up of lubricant oil and a proper cooling of lubricant oil is required.

Motor Input Power = Motor Shaft BHP / Motor Efficiency= 141.43 / 0.89 (Motor Efficiency: 89%) = 158.91 BHP

Example 8-3. Determine the rated motor power for above examples.

(Solution)
Minimum Motor Power = Motor Shaft BHP x Motor Minimum Margin x Operation Safety = 141.43 x 1.1 x 1.03 = 160.24 HP

The next available size of motor power is 175 HP. Note that the motor minimum margin depends on the type of cooling tower operation and ambient conditions.

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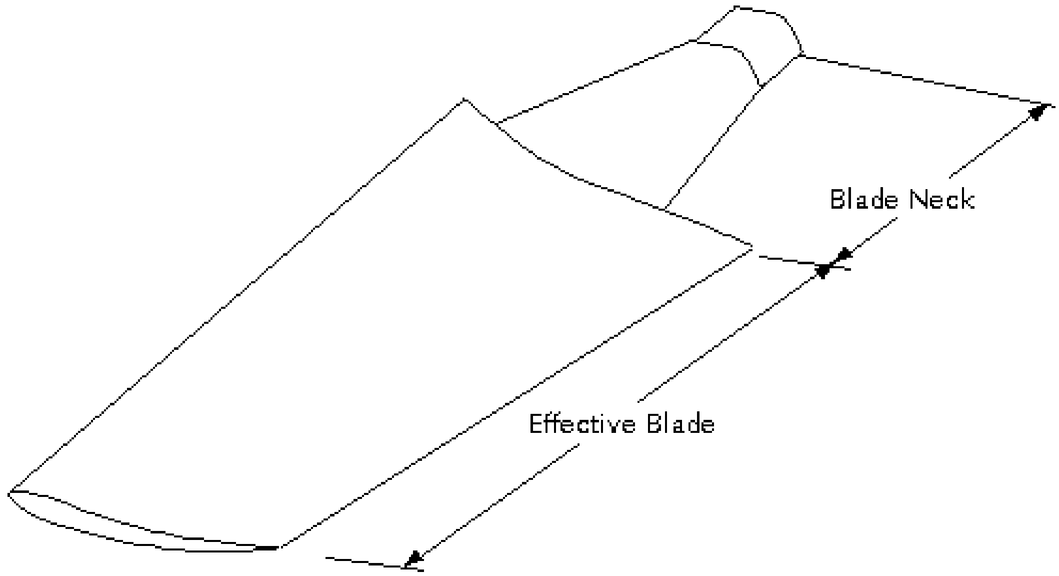
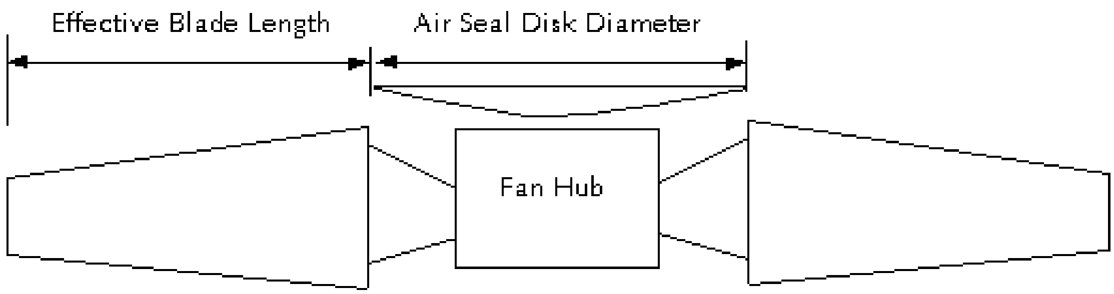
9. Fan Components & Sizing

1) Fan Components: The fan hub is a component to connect the fan blades and to be mounted to the low speed shaft of gear reducer. The fan hubs must be enough strong to overcome the air load imparted onto the fan blades and the centrifugal force due to the rotation of the fan blades. The fan blades have a certain neck to be connected to the fan hub and the effective length of fan blade is reduced as much as the radius of fan hub and the length of fan blade neck. Therefore, the air is not delivered through such fan hub and blade neck and the air pressure at this area is less than the fan blade surface. The air delivered through the fan blades flows back through the area of fan blade neck, where the air pressure is relatively lower than the fan blade area. To prevent this back flow (called recirculation) the aerodynamic seal disk is mounted onto the fan hub.





In general, the diameter of seal disk is about 20 to 25% to the fan diameter. If the hub is too large for the required performance, the result will be an increase in the velocity pressure due to the smaller net opening, and subsequent waste of power. If the seal disk is too small, the result will be deterioration of the flow near the fan hub, possibly even a reversal of flow in this area.

2) Fan Coverage: For an even air suction from the drift eliminator section and to have a smooth

entrance of air into the fan, the fan coverage must not be smaller than 30% of the cross sectional area of cell. Less fan coverage than 30% will returned to a poor intake from the entire drift eliminator section. Therefore, the overall performance of cooling tower will be dramatically reduced.

The fan diameter affects the performance of fan primarily because the magnitude of the velocity pressure depends on the fan diameter. The pressure capability of the fan could be changed by changing the number of fan blades, but the fan must be rated to overcome more static pressure, which is a cooling tower system resistance, as having less velocity pressure with keeping a low air velocity through the fan.

General speaking, the velocity pressure through the fan should be within 0.14 to 0.25 inch Aq. or the air velocity should be raged within 1600 to 2100 FPM for the optimum rating of fan.

3) Fan Sizing: The major factors in deciding the number of fan blades are as below:

(1) Blade Strength

There is a limit of blade strength in bearing the torque or horsepower. In case of Hudson Products Corp., the maximum and Trouble Free BHP/Blade by the fan diameter are as follows;

Fan Dia.	Max. BHP/Blade	Trouble Free BHP/Blade	Fan Dia.	Max. BHP/Blade	Trouble Free BHP/Blade
12 ft	8	4	22 ft	18	14
13 ft	9	5	24 ft	20	16
14 ft	10	6	26 ft	22	18
16 ft	12	8	28 ft	24	20
18 ft	14	10	30 ft	26	22
20 ft	16	12	32 ft	28	24

As a general rule, do not select the fans near to the limit of BHP/Blade specified like above. The high BHP/Blade will cause a fatigue in a short period due to the high blade air loading, and will make a trouble for the vibration noise. Author's experience is the less number of fan blades causes the severe vibration (called Throat Flutter) in the fan stack, unless a special attention in making the fan stack is paid.

Any fan that is effectively moving air at the tips of the blades will develop a reduced pressure area (or suction) on the fan throat at the tip of the blade. This suction tends to draw the throat toward the tip of each blade, which means that a four blade fan would tend to draw the throat into something approaching a square while a six blade fan would draw it into something resembling a hexagon, etc. Since the fan is rotating, the effect on the throat is that of continually drawing it into a rotating polygon. The resulting throat flutter is frequently mistaken for fan unbalance.

A substantial throat will be sufficiently rigid that flutter will not exist. A weak or flexible throat, particularly when used with a fan of a low number of blades, will be greatly affected by this type of vibration. Throat flutter is easily detected due to the fact that it is invariably of a frequency of the fan RPM times the number of blades on the fan. If in doubt that throat flutter is the cause of vibration, reduce the angle of the blades until the fan is doing little or no work. If the vibration ceases under this condition, it is certain that throat flutter is present when the blades are loaded. Throat flutter will cause no damage to the fan so long as the throat does not disintegrate and fall into the fan blades. It may be eliminated by stiffening or bracing the throat.

(2) Material Constructions of Tower Structure and Fan Stack

Common practice in deciding the number of fan blades is to maintain the level of vibration below 80 micron at the gear reducer. A general guideline with Hudson's fans is as below;

Structure Material	Fan Stack Material	Fan Diameter	Minimum Blades No
Concrete	Concrete or FRP	7 - 14 ft	4 each
		16 - 20 ft	5 each
		22 - 24 ft	6 each
		26 - 32 ft	7 each
Wood or Steel	FRP	7 - 14 ft	5 each
		16 - 20 ft	6 each
		22 - 24 ft	7 each
		26 - 32 ft	8 each

Example 9-1. Determine the axial thrust load produced from the fan using the above examples.

(Solution)
This is an axial force opposite the airflow direction and is necessary for engineering the supporting beam of gear reducer and for checking if the bearing thrust capacity for the selected size of gear reducer is larger than this axial load. Ignorance for checking the thrust capacity will result in an early failure of bearings of gear reducer.

Axial Thrust Load = 5.202 x Total Pressure in inch Aq. x Net Fan Area in ft^2 + Fan Weight
= 5.202 x 0.6439 x 573.52 + 1639 = 3,561.0 LB

10. Air-Water Distribution System Design

A cooling tower is an air and water management device, which consists of fan stacks, fans, drift eliminators, fill and water/air distribution systems. For the fans and fan stacks were previously discussed. So, the explanation shall be focused to the air/water distribution systems.

1) Water Distribution: The distribution of water to the top of counter flow fill is a key aspect of assured performance. It is a function of nozzle design, nozzle installation pattern, nozzle distance, and the structural cleanliness of the spray chamber. The impact of water distribution on performance is a combination of uniformity of water distribution, air-side pressure drop through the spray chamber, and heat transfer occurring in the spray zone.

The challenge for a spray system designer is to accomplish an optimum balance of design parameters with practical considerations such as resistance to silt build-up, and the ability to pass objects from trash to Amertap balls. To provide the primary function of precise water distribution, the nozzle must be designed with other considerations in mind:

- The location of counter flow nozzles and the potential for poor quality circulating water demands that the nozzle system be designed to minimize fouling.
- The nozzle must be capable of providing uniform distribution over a wide range of flows, without significant loss in nozzle performance.
- The nozzle must be capable of efficient operation while consuming a minimum of expensive pump energy.

The nozzle arrangement, and the design of the tower structure in the spray chamber, are critical to provide uniform distribution to the top of the fill. The placement of the nozzles must accommodate the tower geometry and still provide even coverage for all parts of the plan area. In general, a criterion such as 90% of the plan area within 5% of the average gpm/square foot, and no areas varying more than 10% from the average will still require several percent conservatism in the thermal performance.

Structure in the spray chamber should be avoided, to prevent spray pattern interference and because any water hitting it tends to fall in concentrated zones on the fill. The impact depends entirely on the extent of structural blockage but can be very substantial for large elements like distribution pipes placed within the spray zone.

Spray water which hits walls or partitions may bypass the fill altogether, with direct impact on performance. Some hollow cone nozzle designs are more prone to structure and wall interference due to the requirement for large overlapping spray patterns. Providing uniform coverage to the edges of the fill requires nozzle placement near the walls to maintain the overlap pattern. As a result, a significant part of the water from the edge nozzles becomes wall water.

The influence of the spray system design on performance is dramatic. Even small changes in the layout of a good spray system, or variations on a nozzle design can have an effect on tower performance of 10% or more. For this reason, it is absolutely imperative that the performance of the fill and spray system be tested as they will be installed. Fill performance data in only valid with the exact spray system configuration used in the test.

2) Air Distribution: Three variables control the distribution of air to the fill in a counter flow configuration. The first is the air inlet geometry. The second is called the pressure ratio. The third is the fan coverage over the eliminators. Extensive aerodynamic modeling studies have been conducted to evaluate the impact of the air inlet design on distribution, and therefore on performance. It is especially important with film fill that air flow reach the entire plan area, including the region adjacent the air entrance. Any region having significantly reduced air flow will effectively allow a bypass of hot water to the cold water basin.

Studies showed that the portion of fill plan area adjacent to the air inlet plan is substantially starved from air flow. Since the air approaching the tower is coming from above the air inlet as well as horizontally, the air has a large downward component adjacent to the tower casing. When this air stream passes the air inlet plane, it is still moving downwards, and does not turn into the fill nearest the inlet. In round tower, this can become a very significant percentage of the total area. In a rectangular tower the effect is still significant, but less.

Critical to the effectiveness of any design, even with an inlet air guide, is that structural interference near the fill and air inlet be minimized. Since inlet velocity is highest in this zone, the wakes behind structural elements can shadow significant areas of fill. Structural interference in this area is meticulously avoided to maximize the effectiveness of a design. The wakes around structural elements at the air entrance also lead to growth of ice in freezing conditions, so avoidance of structure in the air entrance reduces tendencies for icing problems as well. Baffles used for the purpose of changing the direction of air flow in a uniform parallel manner, also utilized to prevent water droplets from splashing out of the tower on their descent through the structure.

The second variable, the pressure ratio, is the ratio of system pressure drop (from the air inlet to the eliminator exit plane) to velocity pressure at the average entrance velocity. The pressure ratio reflects the ratio of resistance to available entering air energy. The higher the ratio, the better entering air will be spread out before entering the fill. The lower the pressure ratio, the less uniform, and less stable the distribution of air flow becomes. The degradation of air flow uniformity is readily apparent, particularly at the inlet.

(Pressure Ratio = Static Pressure / Velocity Pressure at Air Inlet)

It should be noted that ambient winds can decrease the effective pressure ratio in relation to the square of wind speed. Added entering air velocity due to winds increases the velocity pressure as the square of wind velocity. A safety margin is necessary to prevent moderate (10 mph = 4.47 m/s) winds from degrading air distribution. The chosen practice is not to apply towers below a pressure ratio of 5, which is of importance particularly for highly evaluated cases.

The tendency for optimized selections is toward selections with low pressure drop (low fan power, or draft requirement) and high entrance velocity (low pump head). The pressure ratio limitation is a frequent limiting factor in optimization, so that a manufacturer who is unaware of the limitation could have a better evaluated bid - which is not likely to perform as the manufacturer might expect. A manufacturer who recognizes the limitation may be unable to respond in this case, while an unaware manufacturer and the user may discover a serious performance problem after the tower is in service.

Modeling and full scale tower studies have shown that fan plenum pressure drop is related to fan coverage, the third variable. Inadequate fan coverage has been shown to lead also to poor air flow distribution over the fill plane area. Fan coverage is a function of the size of the fan deck opening, the cell size, and the plenum height.

An approximate rule of thumb which has been shown to provide good air distribution and a low plenum pressure drop is as follows; If a circle is projected on the eliminator plan area at a 45 degree angle from the fan stack opening, the percentage of the eliminator are covered by the projected circle is the percent fan coverage. A fan coverage percentage of 30% or greater generally limits the plenum pressure drop to about 10% of the total system pressure drop, and provides good air distribution.

Ignoring this sort of guideline will allow a shorter plenum height, and lower cost tower, but higher plenum pressure drops and uncertain air distribution lead to lower and less predictable performance.

3) Exit Air Velocity: Low fan exit velocity have a two-fold effect on susceptibility to influence by ambient winds. First, at low exit velocity relative to ambient wind speed, the effect of wind is greatest on the velocity profile leaving the fan stack. With tall velocity recovery stacks, the effect is limited primarily to a reduction of the velocity recovery stack. Depending on the magnitude of recovery expected in relation to the total system head, this can be a significant loss. The shorter the recovery stack, or the closer ambient wind can penetrate the cylinder toward the fan itself, the greater will be the direct influence on the fan efficiency. For fans and recovery stacks as commonly applied in industrial applications, a minimum stack exit velocity is approximately 1.4 times the maximum wind speed for guaranteed tower performance (10 mph = 880 fpm). Use of any lower exit velocity requires substantial performance conservatism to compensate for wind effects.

It should be noted also, that tower performance capacity at lower exit velocities relative to the ambient wind speed becomes increasingly sensitive to the wind and inherently as unsteady as the wind speed is variable. It is entirely in the tower owner? best interest to avoid a tower configuration which will have highly variable performance in winds from this effect alone.

The second consequence of excessively low stack exit velocity is the tendency for effluent air to be caught in the ambient wind stream and entrained in the aerodynamic wake downstream of the tower. Since the tower generally has an air entrance face on the downstream side, a portion of the effluent air is recirculated back through the tower. The effluent air is, of course, at a higher wet bulb temperature, so the tower as if subject to hotter ambient temperature.

11. Recirculation of Exit Air

Many works had been studied by several institutes, but the study by CTI is most acceptable for the tower designs. The recirculation in the cooling towers is defined as an adulteration of the atmosphere entering the tower by a portion of the atmosphere leaving the tower. This adulteration by the exhaust air raises the wet bulb temperature of the entering air above that of the ambient air, reducing the tower overall performance.

The recirculation phenomenon arises primarily because of the negative pressure produced on the leeward side of the cooling tower by the wind blowing across the structure. Thus, any factors which enhances this lee-side negative pressure will increase recirculation. At the same time, those elements of tower design or meteorological phenomena which increase the amount of ambient air which mixes with the exhaust air before entering the lee-side air inlets will reduce the magnitude of recirculation. Recirculation is therefore a complex result of factors which affect the lee-side negative pressure and the amount of exhaust air dilution.

From the mathematical standpoint recirculation can be expressed as the percent of the exhaust air which reenters the tower at the air inlets. Thus, considering a heat balance on the air around the tower:

Heat (Q) = $G\ h_1 = G\ (1 - R_c/100)\ h_a + G\ (R_c\ /100)\ h_2$

This equation could be written as below:

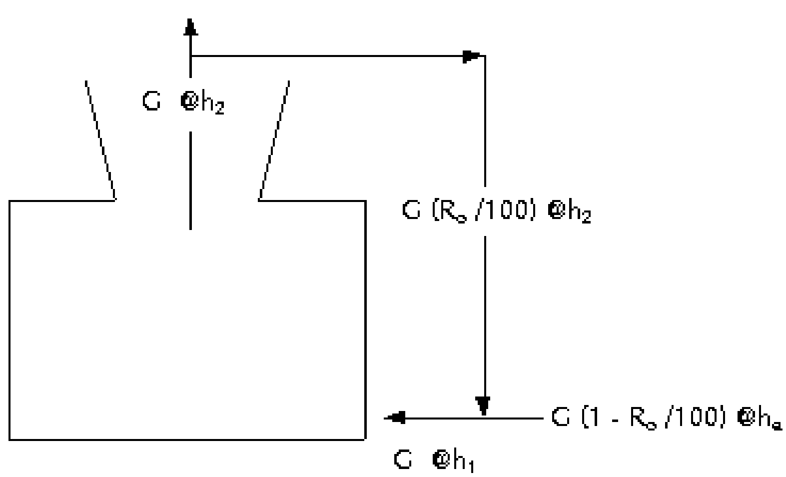
$G\ h_1 = G\ h_a - G\ R_c\ h_a\ /\ 100 + G\ R_c\ h_2\ /\ 100$
$G\ (h_1 - h_a) = G\ R_c\ /100\ (h_2 - h_a)$

Solving for R_c,
$R_c = (h_1 - h_a)\ /\ (h_2 - h_a) \times 100$

From the equation of h_2 = h1 + L/G Range, the enthalpy difference between h_1 and h_a can be obtained as follows;

$(h_1 - h_a)\ 100 = R_c\ (h_1 + L/G\ \text{Range} - h_a) = R_c\ h_1 + R_c\ L/G\ \text{Range} - R_c\ h_a = R_c\ (h_1 - h_a) + R_c\ L/G\ \text{Range}$

$(h_1 - h_a)\ 100\ (1 - R_c) = R_c\ L/G\ \text{Range}$



Solving for $(h_1 - ha)$, $(h_1 - h_a) =$
R_c L/G Range / $(1 - R_c)$ 100

12. Evaporation

When water is cooled in a direct contact cooling tower of air and water, some of the heat is removed by the sensible heat of air in contact with the water. Most of the heat is removed by evaporation of a portion of the circulating water. This mass transfer of water occurs normally from the water stream to the air stream. However, this transfer will be in the opposite direction if the entering water temperature is lower than the entering air wet bulb temperature. In the usual cooling tower operation the water evaporation rate is essentially fixed by the rate of removal of sensible heat from the water, and the evaporation loss can be roughly estimated as 0.1% of the circulating water flow for each degree F of cooling range.

Sensible heat transfer involves an increase in the dry bulb temperature of mixture but evaporation heat transfer involves a change in the humidity ratio of the mixture. Thus, a sensible heat transfer from water to air inside a cooling tower involves a horizontal change on the psychrometric chart while evaporative transfer involves a vertical movement as is illustrated in psychrometric curve. Sensible heat transfer refers to heat transferred by virtue of a temperature difference between the water and air. Evaporative heat removal refers to the energy removal from the water as latent heat of evaporation; this heat removal is the result of the evaporation of water into air during the direct-contact cooling process. In a wet cooling tower, where the temperature of water is greater than the ambient wet bulb temperature, the air humidity always increases as the air passes through the tower. Sensible heat transfer may be either positive or negative. When the temperature is less than the ambient dry bulb temperature, the sensible heat transfer may be negative and the air dry bulb temperature will be lowered as the air passes through the tower; under these circumstance, the air as well as the water is cooled by evaporative transfer in the cooling tower.

In normal cooling tower operation the amount of heat removal by the evaporation is about 60 to 95% to the total heat, and it varies upon the cooling range, air flow rate, relative humidity, and dry bulb temperature, etc.

Example 12-1. Determine the evaporation loss in a percentage for the previous example 6-1.

(Solution)
Evaporation Loss Rate = (Absolute Humidity @ Tower Exit - Absolute Humidity @ Tower Inlet) x 1/(L/G) x 100

Absolute Humidity @ Tower Exit (97°F WBT) = 0.039166
Absolute Humidity @ Tower Inlet (85.24°F DBT & 80% RH)= 0.021117
Evaporation Loss Rate = (0.039166 - 0.021117) x 1 / 1.4760 x 100 = 1.22%

EVAPORATION RATE

Altitude	0.00	feet
Relative Humidity	80.0%	
Wet Bulb Temperature @Inlet	80.00	°F
L/G Ratio	1.4760	
Cooling Range	15.506	°F

Enthalpy of Air @Inlet	43.6907	ft^3/Lb dry air
Equivalent Enthalpy @Inlet	43.6908	ft^3/Lb dry air
Equivalent Dry Bulb Temperature	85.242	°F
Absolute Humidity @Inlet	0.021117	
Enthalpy Air @ Exit	66.5776	ft^3/Lb dry air
Equivalent Enthalpy @Exit	66.5775	ft^3/Lb dry air
Exit Wet Bulb Temperature	97.000	°F
Absolute Humidity	0.039166	
Evaporation Rate	1.223%	

Download the example file (exe12_1.zip)

The above calculation is based on a value of L/G , which was obtained from a result of ignoring the term of evaporation loss in the heat balance. In case of considering the loss of water due to the evaporation, L/G must be computed again as follows;

$L_2/G = \{(ha_2 - ha_1) - (tw_1 - 32) \times (w_2 - w_1)\} / (tw_2 - tw_1)$ $(tw_2 - tw_1 = \text{Actual Range})$

Air Enthalpy at Exit (97º F) = 66.5773 Btu/lb

Air Enthalpy at Inlet (80ºF) = 43.6907 Btu/lb

Then, $L_2/G = \{(66.5773 - 43.6907) - (89 - 32) \times (0.039166 - 0.021117)\} / 15.507 = 1.4096$

Evaporation Loss Rate = (Absolute Humidity @ Tower Exit - Absolute Humidity @ Tower Inlet) x 1/ (L_2/G) x 100 = 1.28%

EXACT EVAPORATION RATE		
Altitude	0.00	feet
Relative Humidity	80.0%	
Wet Bulb Temperature @Inlet	80.00	°F
L/G Ratio	1.4760	
Cooling Range	15.506	°F
Cold Water Temperature	89.00	°F
Enthalpy of Air @Inlet	43.6907	ft^3/Lb dry air
Equivalent Enthalpy @Inlet	43.6907	ft^3/Lb dry air
Equivalent Dry Bulb Temperature	85.242	°F
Absolute Humidity @Inlet	0.021117	
Enthalpy Air @ Exit	66.5776	ft^3/Lb dry air
Equivalent Enthalpy @Exit	66.5776	ft^3/Lb dry air
Exit Wet Bulb Temperature	97.000	°F
Absolute Humidity	0.039166	
Exact L/G Ratio	1.4096	
Evaporation Rate	1.280%	

Download the example file (exe12_1A.zip)

Example 12-2. Determine the heat removal in the percentage by the evaporation for the example 6-1.

(Solution)
Evaporation Rate = (w_2 - w_1) x Latent Heat of Water / (Enthalpy @ Exit - Enthalpy @ Inlet)
Latent Heat of Water: About 1,040 BTU/Lb of Water
(Note: For each pound of water that a cooling tower evaporates, it removes somewhere near 1,040 BTU from water. Evaporative heat removal refers to the energy removal from water as latent heat of evaporation. This heat removal is the result of the evaporation of water into air stream during the direct contact cooling process.)

Evaporation Rate = (0.039166 - 0.021117) x 1040 / (66.5773 - 43.6907) x 100 (%) = 82.02%

Example 12-3. Determine the rate of heat removal by to the evaporation under the assumption that the L/G ratio was changed to 1.600 for the initial conditions of example 6-1.

(Solution)
First, let's calculate the enthalpy of exit air.

Enthalpy of Exit Air = Enthalpy of Inlet Air + L/G x Actual Range = 43.6907 + 1.6 x 15.506 = 68.5019 BTU/lb

Exit Air Temperature = 98.14°F

TOWER EXIT AIR TEMPERATURE

Altitude	0.00	feet
Wet Bulb Temperature @Inlet	80.00	°F
L/G Ratio	1.6000	
Cooling Range	15.50628	°F

Enthalpy of Exit Air	68.5008	ft^3/Lb dry air
Equivalent Enthalpy	68.5008	ft^3/Lb dry air
Exit Wet Bulb Temperature	98.142	°F

Download the example file (exe12_3.zip)

Absolute Humidity @ Tower Exit = 0.040639
Absolute Humidity @ Tower Inlet = 0.021117
Therefore, evaporation rate = (0.040639 - 0.021117) x 1040 / (68.5008 - 43.6907) x 100 (%) = 81.83%

HEAT REMOVAL RATE BY EVAPORATION

Altitude	0.00	feet
Relative Humidity	80.0 %	
Wet Bulb Temperature @Inlet	80.00	°F
L/G Ratio	1.6000	
Cooling Range	15.506	°F

Enthalpy of Air @Inlet	43.6907	ft^3/Lb dry air
Equivalent Enthalpy @Inlet	43.6908	ft^3/Lb dry air
Equivalent Dry Bulb Temperature	85.242	°F
Absolute Humidity @Inlet	0.021117	
Enthalpy Air @ Exit	68.5008	ft^3/Lb dry air
Equivalent Enthalpy @Exit	68.5007	ft^3/Lb dry air
Exit Wet Bulb Temperature	98.142	°F
Absolute Humidity	0.040639	
Heat Removal Rate By Evaporation	81.834 %	

Download the example file (exe12_3A.zip)

Through above two examples the heat removal rate by the evaporation varies with the ratio of water and air mass flow rate. Under the same water flow rate, the higher L/G the smaller evaporation rate.

Example 12-4. Determine the rate of heat removal due to the evaporation under the assumption that RH was changed to 60% from 80% for the example 6-1.

(Solution)
First, calculate the dry bulb temperature of inlet air and find the humidity ratio with the dry bulb temperature & relative humidity.

Absolute Humidity @ Tower Exit = 0.039167
Absolute Humidity @ Tower Inlet = 0.019563
Therefore, evaporation rate = (0.039167 - 0.019563) x 1040 / (66.5780 - 43.6907) x 100 (%) = 89.08%

HEAT REMOVAL RATE BY EVAPORATION

Altitude	0.00	feet
Relative Humidity	60.0%	
Wet Bulb Temperature @Inlet	80.00	
L/G Ratio	1.4760	
Cooling Range	15.506	
Enthalpy of Air @Inlet	43.6907	ft
Equivalent Enthalpy @Inlet	43.6907	ft
Equivalent Dry Bulb Temperature	92.018	
Absolute Humidity @Inlet	0.019563	
Enthalpy Air @ Exit	66.5780	ft
Equivalent Enthalpy @Exit	66.5779	ft
Exit Wet Bulb Temperature	97.000	
Absolute Humidity	0.039167	
Heat Removal Rate By Evaporation	89.078%	

Download the example file (exe12_4.zip)

Note that the evaporation rate of heat removal is being highly effected by the change of relative humidity. Sensible heat transfer involves an increase in the dry bulb temperature of the mixture but evaporative heat transfer involves a change in the humidity ratio of the mixture. Therefore, a sensible heat transfer from water to air inside a cooling tower involves a horizontal change on the psychometric chart while evaporative transfer involves a vertical movement on the psychometric chart. In a wet cooling tower, which the inlet water temperature is greater than the ambient wet bulb temperature, the air humidity always increase as the air passes through the tower. However, Sensible heat transfer may be either positive or negative. When the inlet water temperature is less than the ambient air dry bulb temperature, the sensible heat transfer may be negative and air dry bulb temperature will be lowered as the air passes through the tower. Under these circumstances, the air as well as the water is cooled by evaporative transfer in the cooling tower.

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13. Estimation of Actual Cold Water Temperature

The following steps are being practically applied to design the cooling tower and the current computer thermal programs are based on this concept.

Example 13-1. Determine the cold water temperature for the following conditions.

First Step: Find a dry bulb temperature at the tower inlet.

Tower Design Conditions					
Site Altitude	0	ft	% By-Pass Wall Water	3.27%	
Wet Bulb Temperature	80.00	°F	PD Coefficient @Drift Eliminator	1.80	
Relative Humidity	80.0%		Fan Total Efficiency	79.2%	
Number of Cells	1		Power Transmission Efficiency	91.2%	
Design Water Flow Rate	12,500	gpm	Motor Power Margin	13.3%	
Cell Length	42.0	ft	Motor Power	175	hp
Cell Width	42.0	ft	Fan Diameter	28	ft
Type of Air Inlet	Two Sides Open		Number of Fan per Cell	1	
Air Inlet height	15.0	ft	Seal Disk Diameter	88.0	inch
% Obstruction @Air Inlet	10.0%		PD Coefficient @Fan Inlet	0.18	
PD Coefficient @Air Inlet	2.50		Venturi Height of Stack	3.66	ft
Fill Model	CF1900		Design Hot Water Temperature	104.00	°F
Fill Depth	4.0	ft	Design Cold Water Temperature	89.00	°F
PD Fill Multiplying Factor	1.00		Design Cooling Range	15.0	°F
Fill KaV/L Multiplying Factor	1.00		Actual Range through Tower	15.507	°F
KaV/L Correction Factor	0.09900				
% Obstruction @Fill	1.11%				

Inlet Dry Bulb Temperature Estimation		
Altitude	Feet	0
Inlet Wet Bulb Temperature	°F	80.00
Inlet Air Enthalpy @ WBT	BTU/LB	43.6907
Relative Humidity	%	80.0%
Inlet Air Enthalpy @ DBT & RH	BTU/LB	43.6907
Inlet Dry Bulb Temperature	°F	85.242
Inlet Air Density	Lb/FT^3	0.0718
Inlet Air Specific Volume	FT^3/LB	14.2230

Second Step: Find an exit air temperature and air volume of fan. The net fan power is determined from the relation of Motor HP x (1 - Motor Margin) x Power Transmission Efficiency. It is to iterate the calculation until the net fan power obtained from this equals to the fan bhp which is formulated with (ACFM x Total Pressure) / (6356 x Fan Efficiency). Two variables in the fan bhp equation are unknown, but can be computed from below relationships.

The main idea is to iterate the calculation until the net fan power equals to the calculated fan bhp varying the air volume, static pressure and tower exit temperature at the fan.

The air mass flow rate through the tower is always constant because the air mass is being considered as dry gas. Then, the air mass flow rate and L/G Ratio can be obtained as below:

Air Mass Flow Rate = Air Volume @ Fan / Specific Volume @ Fan

The L/G Ratio =Water Flow Rate in gpm through Tower x (500/60) / (Air Volume @ Fan / Specific Volume @ Fan)

Water flow rate in gpm through tower in this formula is a net water flow rate considering the % by-pass wall water. That is, Design Water Flow Rate x (1 - % By-Pass Water).

In order to obtain the specific volume the tower exit temperature should be computed first. The exit temperature requires L/G ratio as the exit air enthalpy ($ha_2 = ha_1$ + L/G x Range) is a summation of tower inlet air enthalpy and L/G x cooling range. L/G ratio can be given or calculated if the tower dimensions are given and the value of fill characteristic is known.

The air volume at each location of cooling tower is obtained from the relationship of Air Mass Flow Rate x Specific at each location, and then the air velocity at each location is calculated by dividing the air volume by the net area at each location. The total static pressure is a summation of pressure drops obtained from each location.

Exit Air Wet Bulb Temperature Estimation		
L/G Ratio		1.4413
Cooling Range Through Tower	°F	15.5071
Exit Air Enthalpy Based On Estimation	BTU/LB	66.0411
Equivalent Exit Air Enthalpy	BTU/LB	66.0411
Equivalent Exit Wet Bulb Temperature	°F	96.676
Exit Air Density	LB/FT^3	0.0696
Exit Specific Volume	FT^3/LB	14.9183

Fan Air Volume Estimation		
Fan Net Power @ Design	HP	138.37
Total Pressure Drop	Inch Aq.	0.4768
Velocity Pressure @ Fan	Inch Aq.	0.1910
Fan Efficiency		79.20%
Fan Net Power	HP	138.37
Predicted Air Volume	ACFM	1,042,925.7

Third Step: Calculate the tower characteristic in accordance with above results. To calculate this the performance data of fill manufacturer is required.

Fourth Step: Determine NTU(=KaV/L) satisfying the value of tower characteristic by the method of iteration with changing the approach.

Approach Estimation		
Tower Characteristic		
Water Side	Air Side	1/(hw-ha)
88.13	43.6907	
55.5036	45.9257	0.10441
62.2935	52.6309	0.10349
67.2947	57.1010	0.09810
75.6032	63.8061	0.08477
NTU		1.5149
NEW APPROACH		8.633

Download the example file, Version ID-THERMAL/TOWER (idthermal.zip)
This file covers the example no. 13-4, too.

Therefore, Actual Cold Water Temperature = Wet Bulb Temperature + Approach = 80.00 + 8.633 = 88.63°F

Above the enthalpies for water side were computed at the below temperatures.

- Water Temperature @ 0.1 x Range = Wet Bulb Temperature + Approach + Range - New Range + 0.1 x New Range
- Water Temperature @ 0.4 x Range = Wet Bulb Temperature + Approach + Range - New Range + 0.4 x New Range
- Water Temperature @ 0.6 x Range = Wet Bulb Temperature + Approach + Range - New Range + 0.6 x New Range
- Water Temperature @ 0.9 x Range = Wet Bulb Temperature + Approach + Range - New Range + 0.9 x New Range

Also, the enthalpies for air side were based on the followings.

- Enthalpy @ 0.1 x Range = Inlet Air Enthalpy + L/G x 0.1 x New Range
- Enthalpy @ 0.4 x Range = Inlet Air Enthalpy + L/G x 0.4 x New Range
- Enthalpy @ 0.6 x Range = Inlet Air Enthalpy + L/G x 0.6 x New Range
- Enthalpy @ 0.9 x Range = Inlet Air Enthalpy + L/G x 0.9 x New Range

Example 13-2. Check the result if the cold water temperature obtained from example 13-1 is correct.

(Solution)
The actual cold water temperature obtained from example 13-1 is exactly same as the combine temperature of cold water temperature through the tower and hot water temperature of by-pass wall water.

Cold Water Temp. through Tower (Ctemp) = Wet Bulb Temp. + Approach + Design Range - New Range

Hot Water Temp. of By-Pass Wall Water (Btemp) = Ctemp + New Range

Final Cold Water = (Ctemp x Water Flow through Tower + Btemp x By-Pass Wall Water Flow) / Water Flow
= (Ctemp x Water Flow x (1 - %BP) + Btemp x Water Flow x %BP)/ Water Flow
= Ctemp x (1 - %BP) + Btemp x %BP
= Ctemp x (1 - %BP) + (Ctemp + New Range) x %BP
= Ctemp - Ctemp x %BP + Ctemp x %BP + New Range x %BP
= Ctemp + New Range x %BP
= WBT + Approach + Design Range - New Range + New Range x %BP
= WBT + Approach + Design Range - New Range x (1 - %BP)
= WBT + Approach + Design Range - Design Range / (1 - %BP) x (1 - %BP)
= WBT + Approach

- Cold Water Temperature Through Tower (Ctemp) = Actual Cold Water Temp. + Range - Actual Range = 88.633 + 15.0 - 15.507 = 88.126ºF
- Hot Water Temperature into Tower (Btemp) = Ctemp + Actual Range or = Actual Cold Water Temp + Range = Ctemp + 15.507 = 103.633ºF
- Water Flow Through Tower (W_1) = Design Water Flow x (1 - % By-Pass / 100) = 12,500 x (1 - 3.27 /100) = 12,091.25 GPM
- By-Pass Water Flow without Cooling (W_2) =Design Water Flow x % By-Pass / 100 = 12,500 x 3.27 / 100 = 408.75 GPM
- Cold Water Temperature at Water Basin = (W_1 x Ctemp + W_2 x Btemp) / (W_1 + W_2) = (12,091.25 x 88.126 + 408.75 x 103.633) / (12,091.25 + 408.75) = 88.633ºF

Example 13-3. Check if the relation of HEATin = HEATout is established from the above example

(Solution)
HEATin = Total Heat Removal from Water
= Water Flow Rate in GPM x (500 / 60) x Cooling Range
= 12,500 x (500 / 60) x (104 - 89)
= 1,562,500 BTU/Min

or = Water Flow Through Tower in GPM x (500 / 60) x Cooling Range Through Tower
= 12,500 x (1 - % By Pass / 100) x (500 / 60) x Cooling Range / (1 - % By Pass / 100)
= 12,500 x (1 - 3.27 / 100) x (500 / 60) x 15 / (1 -3.27 / 100)
= 1,562,500 BTU/Min

HEATout = Total Heat Gain from Air
= Air Mass Flow in LB/Min x (Exit Air Enthalpy - Inlet Air Enthalpy)
= 69,909.2 x (66.0411 - 43.6907)
= 1,562,500 BTU/Min

Example 13-4. Determine L/G ratio and cold water temperature when the wet bulb temperature was downed to 70ºF from design conditions described in the example 13-1.

(Solution)
First, find a dry bulb temperature for 80% of relative humidity corresponding 70ºF of wet bulb temperature.

Second, find an exit air temperature and air volume of fan until these are ultimately equal.

Water Through Tower in LB/Min = Water Through Tower in GPM x (500 / 60)

Air Mass in LB/Min = Air Volume @ Fan / Specific Volume @ Fan

L/G ratio is obtained from the relation of Water Through Tower in LB/Min / Air Mass in LB/Min

Exit Air Enthalpy = Inlet Air Enthalpy + L/G x Range Through Tower = Inlet Air Enthalpy + {Water Through Tower x (500 / 60) / (Air Volume @ Fan / Specific Volume @ Fan)} x Range Through Tower

Net Fan Power = Motor HP x (1 - Motor Minimum Margin) x Power Transmission Efficiency = Air Volume @ Fan x Total Pressure / (Fan Efficiency x 6356)

Third, calculate the tower characteristic in accordance with above computed results.
KaV/L = 1.864 x {1 / (L/G)}0.8621 x Fill Air Velocity-0.1902 x Fill Height = 1.864 x (1 / 1.4105)$^{0.8621}$ x 578.9$^{-0.1902}$ x 40.8764 = 1.3890
Total Kav/L = KaV/L @ Fill / (1 - % of Heat Transfer at Rain & Water Spray Zone / 100) = 1.3890 / (1 - 9.9% / 100) = 1.5416

Inlet Dry Bulb Temperature Estimation		WBT	DBT
Altitude	Feet	0	0
Inlet Wet Bulb Temperature	°F	70.00	70.00
Inlet Air Enthalpy @ WBT	BTU/LB	34.0743	30.5493
Relative Humidity	%	80.0%	80.0%
Inlet Air Enthalpy @ DBT & RH	BTU/LB	34.0743	49.7362
Inlet Dry Bulb Temperature	0	74.618	85.242
Inlet Air Density	Lb/FT^3	0.0735	0.0742
Inlet Air Specific Volume	FT^3/LB	13.8087	12.4336

Exit Air Wet Bulb Temperature Estimation		
L/G Ratio		1.4150
Cooling Range Through Tower	°F	15.5071
Exit Air Enthalpy Based On Estimation	BTU/LB	56.0164
Equivalent Exit Air Enthalpy	BTU/LB	56.0164
Equivalent Exit Wet Bulb Temperature	°F	90.048
Exit Air Density	0	0.0708
Exit Specific Volume	FT^3/LB	14.5729

Fan Air Volume Estimation		
Fan Net Power @ Design	HP	138.37
Total Pressure Drop	Inch Aq.	0.4790
Velocity Pressure @ Fan	Inch Aq.	0.1922
Fan Efficiency		79.20%
Fan Net Power	HP	138.37
Predicted Air Volume	ACFM	1,037,739.7

Fourth, determine the NTU satisfying the value of tower characteristic by the method of iteration with the change of approach figure.

Approach Estimation		
Tower Characteristic		
Water Side	Air Side	1/(hw-ha)
81.38	34.0743	
46.9790	36.2685	0.09337
52.7058	42.8511	0.10147
56.9123	47.2396	0.10338
63.8808	53.8222	0.09942
NTU		1.5416
NEW APPROACH		11.891

Therefore, Actual Cold Water Temperature = Wet Bulb Temperature + Approach = 70 + 11.891 = 81.89 deg. F

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14. Determination of L/G

The classical method of thermal rating of cooling tower is to estimate the ratio of liquid to gas first and is to find the proper tower volume by the means of trial & error using the tower performance curve. This was the most convenient solution when the computer was not readily accessible. The major problems with this solution are not to consider the actual geography. As seen in the equations of NTU or Tower Demand, the right side of formula is obviously a dimensionless factor. It can be calculated using only the temperatures and flows entering the tower. It is totally independent from the tower size and fill configuration.

Now, the best way to design the cooling tower is based on the actual sizes of tower and is to find a proper L/G satisfying such sizes of cooling tower. The L/G is the most important factor in designing the cooling tower and related to the construction & operating cost of cooling tower. The fooling example will explain about the procedure of determining the L/G ratio.

Example 14-1. Determine the L/G ratio under the assumption that the water flow rate was increased to 13,750 GPM and the wet bulb temperature remains unchanged from design conditions given in the example 13-1.

(Solution)

Tower Design Conditions					
Site Altitude	0	ft	% By-Pass Wall Water	3.27%	
Wet Bulb Temperature	80.00	°F	PD Coefficient @Drift Eliminator	1.80	
Relative Humidity	80.0%		Fan Total Efficiency	79.2%	
Number of Cells	1		Power Transmission Efficiency	91.2%	
Design Water Flow Rate	13,750	gpm	Motor Power Margin	13.3%	
Cell Length	42.0	ft	Motor Power	175	hp
Cell Width	42.0	ft	Fan Diameter	28	ft
Type of Air Inlet	Two Sides Open		Number of Fan per Cell	1	
Air Inlet height	15.0	ft	Seal Disk Diameter	88.0	inch
% Obstruction @Air Inlet	10.0%		PD Coefficient @Fan Inlet	0.18	
PD Coefficient @Air Inlet	2.50		Venturi Height of Stack	3.66	ft
Fill Model	CF 1900		Design Hot Water Temperature	104.00	°F
Fill Depth	4.0	ft	Design Cold Water Temperature	89.00	°F
PD Fill Multiplying Factor	1.00		Design Cooling Range	15.0	°F
Fill KaV/L Multiplying Factor	1.00		Actual Range through Tower	15.507	°F
KaV/L Correction Factor	0.09900				
% Obstruction @Fill	1.11%				

First, find a dry bulb temperature for 80% of relative humidity corresponding 80°F of wet bulb temperature at the tower inlet.

First Step: Find a dry bulb temperature at the tower inlet.

Inlet Dry Bulb Temperature Estimation		
Altitude	Feet	0
Inlet Wet Bulb Temperature	°F	80.00
Inlet Air Enthalpy @ WBT	BTU/LB	43.6907
Relative Humidity	%	80.0%
Inlet Air Enthalpy @ DBT & RH	BTU/LB	43.6907
Inlet Dry Bulb Temperature	°F	85.242
Inlet Air Density	Lb/FT^3	0.0718
Inlet Air Specific Volume	FT^3/LB	14.2230

Second Step: Find an exit air temperature and air volume of fan. The procedure is exactly same as the contents described in the example 13-1.

Net Fan Power = Motor HP x (1 - Motor Margin) x Power Transmission Efficiency
Fan bhp = Air Volume @ Fan x Total Static Pressure / (6356 x Fan Efficiency)
Exit Air Enthalpy = Inlet Air Enthalpy + L/G x Actual Cooling Range
Actual Cooling Range = Design Range / (1 - % By-Pass Water)

The iteration is continued until the value of Net Fan Power equals to Fan bhp varying air volume at the fan and the pressure drops corresponding to the air volume at each location of cooling tower.

Net Fan Power = Fan bhp
That is, Motor HP x (1 - Motor Margin) x Power Transmission Efficiency = Air Volume @ Fan x Total Static Pressure / (6356 x Fan Efficiency)

The air volume is finally obtained from this relationship. Then, the L/G ratio is obtained from below relations.

Water Flow Rate in gpm through Tower = Design Water Flow Rate x (1 - % By-Pass Water).
Water Flow Rate in Lb/Min = Water Flow Rate in gpm through Tower x (500/60)
Air Mass Flow Rate = Air Volume @ Fan / Specific Volume @ Fan
The L/G Ratio =Water Flow Rate in Lb/Min / Air Mass Flow Rate in Lb/Min

Water Flow Rate in Lb/Min = 13,300.4 x (500/60) = 110,836.7
Air Mass Flow Rate = 1,039,249.8 / 15.000 =69,283.3

L/G = 110,836.7 / 69,283.3 = 1.59976

Exit Air Wet Bulb Temperature Estimation		
L/G Ratio		1.5998
Cooling Range Through Tower	°F	15.5071
Exit Air Enthalpy Based On Estimation	BTU/LB	68.4982
Equivalent Exit Air Enthalpy	BTU/LB	68.4982
Equivalent Exit Wet Bulb Temperature	°F	98.140
Exit Air Density	LB/FT^3	0.0694
Exit Specific Volume	FT^3/LB	15.0000

Fan Air Volume Estimation		
Fan Net Power @ Design	HP	138.37
Total Pressure Drop	Inch Aq.	0.4812
Velocity Pressure @ Fan	Inch Aq.	0.1890
Fan Efficiency		79.20%
Fan Net Power	HP	138.37
Predicted Air Volume	ACFM	1,039,249.8

Basic Thermal Rating Solving					
Water Through Tower	13,300.4	gpm	Net Fan Area per Fan	573.52	ft^2
Computed L/G Ratio	1.5998		Air Velocity @Fan	1,812.1	fpm
Air Mass Flow Rate	69,283.5	Lb/min	Velocity Pressure @Fan	0.1890	inch Aq.
Tower Exit Air Temperature	98.14	°F	Net Fan Power	138.37	bhp
Air Volume per Fan	1,039,250	acfm	Total Fan Static Pressure	0.4812	inch Aq.
Pressure Drops Calculation					
1) Air Inlet			3) Drift Eliminator		
- Total Net Air Inlet	1,134.0	ft^2	- Net Area	1,744.4	ft^2
- Air Density	0.0718	Lb/ft^3	- Air Density	0.0694	Lb/ft^3
- Specific Volume	14.2230	ft^3/Lb	- Specific Volume	15.0000	ft^3/Lb
- Total Air Volume	985,422	acfm	- Air Volume	1,039,250	acfm
- Air Velocity	869.0	fpm	- Air Velocity	595.8	fpm
- Pressure Drop	0.1125	INCH Aq	- Pressure Drop	0.0368	inch Aq.
2) Fill			4) Fan Inlet		
- Total Net Fill Area	1,744.4	ft^2	- Air Density	0.0694	Lb/ft^3
- Water Loading	7.62	gpm/ft^2	- Air Velocity @Fan	1,812.1	fpm
- Average Air Density	0.0706	Lb/ft^3	- Pressure Drop	0.0340	inch Aq.
- Average Air Specific Volume	14.6012	ft^3/Lb	5. Velocity Recovery		
- Average Air Volume per Cell	1,011,621	acfm	- Efficiency of Fan Stack	77.4%	
- Average Fill Air Velocity	579.9	fpm	- Air Density	0.0694	Lb/ft^3
- Pressure Drop	0.3154	inch Aq.	- Velocity Recovery	0.0174	inch Aq.

Download the example file, Version ID-THERMAL/TOWER (idthermal.zip)
This file is same as the example file discussed in example 13-1.

Example 14-2. The value of slope in the tower characteristic was just estimated like the above examples. Determine the actual slope using the design conditions of the example no.13-1 and 14-1.

(Solution)

NTU = C x $(L/G)^{-m}$ ------------------------------ Eq.14-1
Log(NTU) = Log C - m x Log(L/G) -------------- Eq. 14-2
Log C = Log(NTU) + m x Log(L/G) -------------- Eq. 14-3

The fact, which that the value of C for a designed cooling condition is same regardless the change of water flow rate, was already mentioned previously. From this rule, the value of slope can be derived as follows;

Log C @ 100% Water = Log(NTU @ 100% Water) + m x Log(L/G @ 100% Water) -- Eq. 14-4
Log C @ 110% Water = Log(NTU @ 110% Water) + m x Log(L/G @ 110% Water) -- Eq. 14-5

Eq. 14-4 and Eq. 14-5 can be written as below using the relation of Log C @ 100% Water = Log C @ 110% Water.
Log(NTU @100% Water) + m x Log(L/G @ 100% Water) = Log(NTU @ 110% Water) + m x Log(L/G @ 110% Water)
Log(NTU @100% Water) - Log(NTU @ 110% Water) = m x Log(L/G @ 110% Water) - m x Log(L/G @ 100% Water)

This form can be changed to:

Log(NTU @ 100% Water / NTU @ 110% Water) = m x Log (L/G @ 110% Water / L/G @ 100% Water)

Finally, this equation can be solved for m as follows;

m = Log (NTU @ 100% Water / NTU @ 110% Water) / Log (L/G @ 110% Water / L/G @ 100% Water)

	100% OF WATER FLOW	110% OF WATER FLOW
L/G	1.4413	1.5998
NTU	1.5149	1.3863

Slope = Log (1.5149 / 1.3863) / Log (1.5998 / 1.4413) = 0.8506

15. Comparison of Tower Performance at Sea Level and Altitude

In regard to the volume of air required for a particular cooling tower requirement and a particular tower at high altitude when compared to sea level, the effect of altitude is two folds. Since it is the mass (or weight) of air and not the volume that is vital to the tower performance, the first effect is that, because of altitude and corresponding reduction in air density, it takes a larger volume to obtain the required pounds. On the other hand, because of the higher water partial pressure which increases the evaporative tendency, the actual mass of air required for the duty is reduced.

Although it has been recognized by most cooling tower manufacturers that, for the majority of operating conditions, the more significant of these two effect is the evaporative effect, it is generally assumed they are sufficiently canceling to ignore elevation in the calculation of the volume of air, and consider only the reduced static pressure and corresponding reduction in horsepower resulting from the lower air density.

At the reduced atmospheric pressure associated with high elevation, the higher partial pressure of water results in a higher moisture content in the air at any temperature. This higher moisture content increases the heat content of the air, or the temperature-enthalpy plot of saturated air at high elevation is above the sea level curve.

Example 15-1. Discuss about the effect of altitude on cooling tower rating and performance at 2000 feet in the altitude using the example 13-1.

(Solution)

Tower Design Conditions					
Site Altitude	2000	ft	% By-Pass Wall Water	3.27%	
Wet Bulb Temperature	80.00	°F	PD Coefficient @Drift Eliminator	1.80	
Relative Humidity	80.0%		Fan Total Efficiency	79.2%	
Number of Cells	1		Power Transmission Efficiency	91.2%	
Design Water Flow Rate	12,500	gpm	Motor Power Margin	13.3%	
Cell Length	42.0	ft	Motor Power	175	hp
Cell Width	42.0	ft	Fan Diameter	28	ft
Type of Air Inlet	Two Sides Open		Number of Fan per Cell	1	
Air Inlet height	15.0	ft	Seal Disk Diameter	88.0	inch
% Obstruction @Air Inlet	10.0%		PD Coefficient @Fan Inlet	0.18	
PD Coefficient @Air Inlet	2.50		Venturi Height of Stack	3.66	ft
Fill Model	CF 1900		Design Hot Water Temperature	104.00	°F
Fill Depth	4.0	ft	Design Cold Water Temperature	89.00	°F
PD Fill Multiplying Factor	1.00		Design Cooling Range	15.0	°F
Fill KaV/L Multiplying Factor	1.00		Actual Range through Tower	15.507	°F
KaV/L Correction Factor	0.09900				
% Obstruction @Fill	1.11%				

First, find a dry bulb temperature at 2000 feet in the altitude for 80% of relative humidity corresponding 80ºF of wet bulb temperature.

First Step: find a dry bulb temperature for 80% of relative humidity corresponding 80ºF of wet bulb temperature at the tower inlet.

Inlet Dry Bulb Temperature Estimation		
Altitude	Feet	2000
Inlet Wet Bulb Temperature	ºF	80.00
Inlet Air Enthalpy @ WBT	BTU/LB	45.6157
Relative Humidity	%	80.0%
Inlet Air Enthalpy @ DBT & RH	BTU/LB	45.6157
Inlet Dry Bulb Temperature	ºF	85.337
Inlet Air Density	Lb/FT^3	0.0667
Inlet Air Specific Volume	FT^3/LB	15.3442

Second Step: Find an exit air temperature and air volume of fan. The procedure is exactly same as the contents described in the example 14-1. Refer to it for the details.

Exit Air Wet Bulb Temperature Estimation		
L/G Ratio		1.5167
Cooling Range Through Tower	ºF	15.5071
Exit Air Enthalpy Based On Estimation	BTU/LB	69.1349
Equivalent Exit Air Enthalpy	BTU/LB	69.1349
Equivalent Exit Wet Bulb Temperature	ºF	96.470
Exit Air Density	LB/FT^3	0.0647
Exit Specific Volume	FT^3/LB	16.1115

Fan Air Volume Estimation		
Fan Net Power @ Design	HP	138.37
Total Pressure Drop	Inch Aq.	0.4639
Velocity Pressure @ Fan	Inch Aq.	0.1868
Fan Efficiency		79.20%
Fan Net Power	HP	138.37
Predicted Air Volume	ACFM	1,070,364.8

Third Step: Calculate the tower characteristic for the results obtained above per the performance data provided by the fill manufacturer.

Fourth Step: Iterate until NTU satisfies the value of tower characteristic by changing of approach figure.

Approach Estimation		
Tower Characteristic		
Water Side	Air Side	1/(hw-ha)
87.92	45.6157	
57.9063	47.9676	0.10062
65.1346	55.0234	0.09890
70.4698	59.7272	0.09309
79.3522	66.7830	0.07956
NTU		1.4428
NEW APPROACH		8.425

Therefore, Actual Cold Water Temperature = Wet Bulb Temperature + Approach = 80 + 8.425 = 88.425ºF

Download the example file, Version ID-THERMAL/TOWER (idthermal.zip)
This file is same as the example file discussed in example 13-1.

16. Evaluation of Tower Performance at Design & Off Design

The prediction of cold water temperature at the off-design points (wet bulb temperature and cooling range other than design conditions) is to find an approach satisfying the cooling tower characteristic value at the design condition. In finding an approach at the off-design points, the L/G at the off-design points must be first assumed. There are three methods in assuming L/G ratio at the off-design point.

- Constant Fan BHP (BHP off = BHP dsn)
- Constant Fan Pitch (VOL off = VOL dsn)
- Constant Air Mass Flow Rate (GAS off = GAS dsn)

1) Relationship between Design & Off-Design L/G Ratio and Design & Off-Design BHP @ Constant Fan BHP

First, let's see the fan bhp formula.

Fan BHP =VOL x TP / (6356 x Fan Effi.)
= VOL x (VP + SP) / (6356 x Fan Effi.)
= VOL x (1/2g x Density x VEL^2 + K x 1/2g x DEN x Vel^2) / (6356 x Fan Effi.)
= VOL x DEN x VEL^2 x (1 + K) /1/2g / (6356 x Fan Effi.)
= VOL x DEN x VEL^2 x (Area^2 /Area^2) x (1 + K) /1/2g / (6356 x Fan Effi.)
= VOL x DEN x VOL^2 x 1 / Area^2 x (1 + K) /1/2g / (6356 x Fan Effi.)
= VOL^3 x DEN x 1 / Area^2 x (1 + K) /1/2g / (6357 x Fan Effi.)
(The term of 1 / Area2 x (1 + K) /1/2g / (6357 x Fan Effi.) could be considered as a constant under the assumption that the fan efficiency at the design conditions is equal to the fan efficiency at the off-design conditions.)
= Constant x VOL3 x Density

Where,
VOL = Air Volume @ Fan (ACFM)
TP = Total Pressure @ Fan (Inch Aq.)
VP = Velocity Pressure @ Fan (Inch Aq.)
SP = Static Pressure (Inch Aq.)
g = Acceleration Gravity (ft/min^2)
DEN = Air Density @ Fan (Lb/ft^3)
VEL = Air Velocity @ Fan (FPM)
K = Overall Pressure Drop Coefficient
Area = Net Fan Area (ft^2)

The relation of BHP off = BHP dsn is established under the assumption of constant fan BHP, which means that the fan bhp at the off-design is always equal to the fan bhp at the design regardless the off-design conditions.

BHP off = Const. x VOL off^3 x DEN off

$$BHP\ dsn = Const. \times VOL\ dsn^3 \times DEN\ dsn$$
$$BHP\ off = BHP\ dsn \text{ -- Eq. 16-1}$$

Since BHP off = BHP dsn, the Eq. 16-1 presenting the relation with VOL off and VOL dsn can be written in the following form.

$$Const. \times VOL\ off^3 \times DEN\ off = Const. \times VOL\ dsn^3 \times DEN\ dsn$$
$$VOL\ off = VOL\ dsn \times (DEN\ dsn / DEN\ off)^{1/3} \text{ --------------------------- Eq. 16-2}$$
$$VOL\ dsn / SV\ dsn = G\ dsn \text{ (SV = Specific Volume of Air at Fan)}$$
$$VOL\ dsn = G\ dsn \times SV\ dsn$$
$$= L\ dsn / L/G\ dsn \times SV\ dsn \text{ (L = Water Flow in Pound)--------- Eq. 16-3}$$

Substitute VOL dsn in the right side of Eq. 16-2 by Eq. 16-3. Then, the below form is obtained.

$$VOL\ off = L\ dsn \times (1 / L/G\ dsn) \times SV\ dsn \times (DEN\ dsn / DEN\ off)^{1/3} \text{------ Eq. 16-4}$$
$$L/G\ off = L\ off / G\ off = L\ off / (VOL\ off / SV\ off) \text{ ------------------------ Eq. 16-5}$$

Substitute VOL off in the denominator of right side of Eq. 16-5 by Eq. 16-4.

$$L/G\ off = L\ off / [(L\ dsn \times (1 / L/G\ dsn) \times SV\ dsn \times (DEN\ dsn / DEN\ off)^{1/3}) / SV\ off]$$
$$= L/G\ dsn \times (L\ off / L\ dsn) \times (DEN\ off / DEN\ dsn)^{1/3} \times (SV\ off / SV\ dsn) \text{ --- Eq. 16-6}$$

Therefore, L/G at off-design point can be obtained from Eq. 16-6.

2) Relationship between Design & Off-Design L/G Ratio Design & Off-Design BHP @ Constant Fan Pitch

The relation of VOL off = VOL dsn is established under the assumption of constant fan pitch, which means that the air volume at the off-design is always same as the air volume at the design regardless the off-design conditions.

$$BHP\ off = Const. \times VOL\ off^3 \times DEN\ off$$
$$VOL\ off^3 = BHP\ off / (Const. \times DEN\ off)$$
$$VOL\ off = BHP\ off^{1/3} / (Const. \times DEN\ off)^{1/3} \text{ ------------------------------ Eq. 16-7}$$

$$BHP\ dsn = Const. \times VOL\ dsn^3 \times DEN\ dsn$$
$$VOL\ dsn^3 = BHP\ dsn / (Const. \times DEN\ dsn)$$
$$VOL\ dsn = BHP\ dsn^{1/3} / (Const. \times DEN\ dsn)^{1/3} \text{ ---------------------------- Eq. 16-8}$$

From the assumption of constant fan pitch, the relation of VOL off = VOL dsn is established and the following forms are obtained.

$$BHP\ off^{1/3} / (Const. \times DEN\ off)^{1/3} = BHP\ dsn^{1/3} / (Const. \times DEN\ dsn)^{1/3}$$
$$BHP\ off = BHP\ dsn \times (DEN\ off / DEN\ dsn) \text{ --------------------------------- Eq. 16-9}$$

$$L/G\ dsn = L\ dsn / G\ dsn = L\ dsn / (VOL\ dsn / SV\ dsn) \text{ --------------------- Eq. 16-10}$$

Solve Eq. 16-10 for VOL dsn and rewrite.

VOL dsn = (1 / L/G dsn) x L dsn x SV dsn ---------------------------------- Eq. 16-11

VOL off = VOL dsn = (1 / L/G dsn) x L dsn x SV dsn ---------------------- Eq. 16-12

L/G off = L off / G off = L off / (VOL off / SV off) --------------------------- Eq. 16-13

Substitute VOL off in the denominator of right side of Eq. 16-13 by Eq. 16-12 and rewrite.

L/G off = L off / [((1 / L/G dsn) x L dsn x SV dsn) / SV off]
= L/G dsn x (L off / L dsn) x (SV off / SV dsn) ------------------------ Eq. 16-14

3) Relationship between Design & Off-Design L/G Ratio Design & Off-Design BHP @ Constant Gas

The relation of GAS off = GAS dsn is established under the assumption of constant gas mass flow rate, which means that the air mass flow rate at the off-design is always equal to the air mass flow rate at the design regardless the off-design conditions.

BHP off = Const. x VOL off^3 x DEN off
VOL off = GAS off x SV off
BHP off = Const. x (GAS off x SV off)^3 x DEN off
GAS off^3 = BHP off / (Const. x DEN off x SV off^3)
GAS off = BHP off$^{1/3}$ / (Const. x DEN off$^{1/3}$ x SV off) ----------------------- Eq. 16-15

BHP dsn = Const. x VOL dsn^3 x DEN dsn
VOL dsn = GAS dsn x SV dsn
BHP dsn = Const. x (GAS dsn x SV dsn)^3 x DEN dsn
GAS dsn^3 = BHP dsn / (Const. x DEN dsn x SV dsn^3)
GAS dsn = BHP dsn$^{1/3}$ / (Const. x DEN dsn$^{1/3}$ x SV dsn) -------------------- Eq. 16-16

From the assumption of constant fan pitch, the relation of GAS off = GAS dsn is established and the following forms are obtained.

BHP off$^{1/3}$ / (Const. x DEN off$^{1/3}$ x SV off) = BHP dsn$^{1/3}$ / (Const. x DEN dsn$^{1/3}$ x SV dsn)

Therefore, BHP off = BHP dsn x (DEN off / DEN dsn) x (SV off / SV dsn)^3 --- Eq. 16-17

L/G dsn = L dsn / G dsn --- Eq. 16-18
L/G off = L off / G off --- Eq. 16-19

Eq. 16-19 can be written as Eq. 16-20 using the relation of G dsn = G off
L/G dsn = L dsn / G off --- Eq. 16-20
Also, Eq. 16-20 can be solved for G off as Eq. 16-21.
G off = (1 / L/G dsn) x L dsn -- Eq. 16-21

Substitute the G off of right side of Eq. 16-19 by Eq. 16-21 and rewrite it as below.

L/G off = L off / [(1 / L/G dsn) x L dsn] = L/G dsn x (L off / L dsn) ------------- Eq. 16-22

Example 16-1: Determine the KaV/L, L/G, and Fan BHP at the off-design points (60, 72.50, and 85°F of wet bulb temperature) by the simple performance prediction and under the conditions of constant fan bhp, constant fan pitch constant gas using example 13-1.

PERFORMANCE CURVE PLOTTING BY SIMPLE METHOD

DESCRIPTIONS	DESIGN	UNIT
1. Method of Performance Prediction	Constant Fan Pitch	
2. Circulated Water Flow	12500	GPM
3. Range	15	°F
4. Cold Water Temperature	89	°F
5. Inlet Wet Bulb Temperature	80	°F
6. Minimum WBT for Curve	60	°F
7. Maximum WBT for Curve	85	°F
8. Relative Humidity	80%	
9. Altitude	0	FEET
10. Design L/G Ratio	1.4413	
11. Design KaV/L (Uncorrected)	1.3469	
12. Slope of Tower Characteristic	-0.8000	
13. Net Fan Horsepower	138.37	HP
14. Name of Customer	Chungrok ENC Company	
15. Model of C/Tower	Sample	
16. Person in Charge	Oick Kwon	

(Solution)
1) Tower Performance by Method of Constant Fan BHP

PERFORMANCE DETAILS WITH 100 % OF WATER FLOW						
WATER FLOW (GPM)	WET BULB TEMP (°F)	DRY BULB TEMP (°F)	RANGE (°F)	L/G	KaV/L	FAN POWER (HP)
12500	60.00	63.97	12.00	1.3813	1.3935	138.37
12500	66.25	70.63	12.00	1.3949	1.3827	138.37
12500	72.50	77.28	12.00	1.4098	1.3709	138.37
12500	78.75	83.92	12.00	1.4264	1.3581	138.37
12500	80.00	85.24	12.00	1.4299	1.3555	138.37
12500	85.00	90.54	12.00	1.4450	1.3441	138.37
12500	60.00	63.97	15.00	1.3937	1.3836	138.37
12500	66.25	70.63	15.00	1.4069	1.3732	138.37
12500	72.50	77.28	15.00	1.4215	1.3619	138.37
12500	78.75	83.92	15.00	1.4378	1.3495	138.37
12500	80.00	85.24	15.00	1.4413	1.3469	138.37
12500	85.00	90.54	15.00	1.4561	1.3359	138.37
12500	60.00	63.97	18.00	1.4059	1.3740	138.37
12500	66.25	70.63	18.00	1.4187	1.3640	138.37
12500	72.50	77.28	18.00	1.4331	1.3531	138.37
12500	78.75	83.92	18.00	1.4491	1.3411	138.37
12500	80.00	85.24	18.00	1.4526	1.3385	138.37
12500	85.00	90.54	18.00	1.4673	1.3278	138.37

PERFORMANCE DETAILS WITH 90% OF WATER FLOW						
WATER FLOW (GPM)	WET BULB TEMP (°F)	DRY BULB TEMP (°F)	RANGE (°F)	L/G	KaV/L	FAN POWER (HP)
11250	60.00	63.97	12.00	1.2387	1.5204	138.37
11250	66.25	70.63	12.00	1.2510	1.5085	138.37
11250	72.50	77.28	12.00	1.2646	1.4955	138.37
11250	78.75	83.92	12.00	1.2796	1.4814	138.37
11250	80.00	85.24	12.00	1.2828	1.4784	138.37
11250	85.00	90.54	12.00	1.2965	1.4660	138.37
11250	60.00	63.97	15.00	1.2488	1.5106	138.37
11250	66.25	70.63	15.00	1.2608	1.4991	138.37
11250	72.50	77.28	15.00	1.2741	1.4866	138.37
11250	78.75	83.92	15.00	1.2889	1.4729	138.37
11250	80.00	85.24	15.00	1.2921	1.4700	138.37
11250	85.00	90.54	15.00	1.3055	1.4578	138.37
11250	60.00	63.97	18.00	1.2587	1.5010	138.37
11250	66.25	70.63	18.00	1.2705	1.4899	138.37
11250	72.50	77.28	18.00	1.2835	1.4778	138.37
11250	78.75	83.92	18.00	1.2981	1.4645	138.37
11250	80.00	85.24	18.00	1.3012	1.4617	138.37
11250	85.00	90.54	18.00	1.3145	1.4498	138.37

PERFORMANCE DETAILS WITH 110% OF WATER FLOW						
WATER FLOW (GPM)	WET BULB TEMP (°F)	DRY BULB TEMP (°F)	RANGE (°F)	L/G	KaV/L	FAN POWER (HP)
13750	60.00	63.97	12.00	1.5250	1.2875	138.37
13750	66.25	70.63	12.00	1.5397	1.2776	138.37
13750	72.50	77.28	12.00	1.5559	1.2669	138.37
13750	78.75	83.92	12.00	1.5741	1.2552	138.37
13750	80.00	85.24	12.00	1.5779	1.2528	138.37
13750	85.00	90.54	12.00	1.5944	1.2424	138.37
13750	60.00	63.97	15.00	1.5398	1.2775	138.37
13750	66.25	70.63	15.00	1.5541	1.2681	138.37
13750	72.50	77.28	15.00	1.5700	1.2578	138.37
13750	78.75	83.92	15.00	1.5878	1.2465	138.37
13750	80.00	85.24	15.00	1.5916	1.2441	138.37
13750	85.00	90.54	15.00	1.6079	1.2341	138.37
13750	60.00	63.97	18.00	1.5544	1.2679	138.37
13750	66.25	70.63	18.00	1.5684	1.2589	138.37
13750	72.50	77.28	18.00	1.5840	1.2489	138.37
13750	78.75	83.92	18.00	1.6015	1.2380	138.37
13750	80.00	85.24	18.00	1.6053	1.2357	138.37
13750	85.00	90.54	18.00	1.6213	1.2259	138.37

Download the example file: Version IDPC/TOWER 2.01 (idpcsim.zip: 157Kb)

2)Tower Performance by Method of Constant Fan Pitch

PERFORMANCE DETAILS WITH 100% OF WATER FLOW						
WATER FLOW (GPM)	WET BULB TEMP (°F)	DRY BULB TEMP (°F)	RANGE (°F)	L/G	KaV/L	FAN POWER (HP)
12500	60.00	63.97	12.00	1.3622	1.4091	144.08
12500	66.25	70.63	12.00	1.3802	1.3944	142.66
12500	72.50	77.28	12.00	1.4000	1.3786	141.20
12500	78.75	83.92	12.00	1.4218	1.3616	139.66
12500	80.00	85.24	12.00	1.4265	1.3581	139.35
12500	85.00	90.54	12.00	1.4461	1.3433	138.06
12500	60.00	63.97	15.00	1.3785	1.3957	142.79
12500	66.25	70.63	15.00	1.3960	1.3817	141.48
12500	72.50	77.28	15.00	1.4153	1.3666	140.11
12500	78.75	83.92	15.00	1.4367	1.3503	138.67
12500	80.00	85.24	15.00	1.4413	1.3469	138.37
12500	85.00	90.54	15.00	1.4606	1.3326	137.15
12500	60.00	63.97	18.00	1.3946	1.3829	141.59
12500	66.25	70.63	18.00	1.4116	1.3695	140.37
12500	72.50	77.28	18.00	1.4305	1.3550	139.08
12500	78.75	83.92	18.00	1.4515	1.3393	137.72
12500	80.00	85.24	18.00	1.4560	1.3360	137.43
12500	85.00	90.54	18.00	1.4751	1.3221	136.27

PERFORMANCE DETAILS WITH 90% OF WATER FLOW						
WATER FLOW (GPM)	WET BULB TEMP (°F)	DRY BULB TEMP (°F)	RANGE (°F)	L/G	KaV/L	FAN POWER (HP)
11250	60.00	63.97	12.00	1.2200	1.5390	144.61
11250	66.25	70.63	12.00	1.2364	1.5226	143.16
11250	72.50	77.28	12.00	1.2544	1.5051	141.65
11250	78.75	83.92	12.00	1.2743	1.4864	140.07
11250	80.00	85.24	12.00	1.2785	1.4825	139.75
11250	85.00	90.54	12.00	1.2962	1.4662	138.44
11250	60.00	63.97	15.00	1.2334	1.5257	143.42
11250	66.25	70.63	15.00	1.2493	1.5101	142.07
11250	72.50	77.28	15.00	1.2669	1.4933	140.65
11250	78.75	83.92	15.00	1.2864	1.4752	139.16
11250	80.00	85.24	15.00	1.2905	1.4714	138.85
11250	85.00	90.54	15.00	1.3080	1.4556	137.60
11250	60.00	63.97	18.00	1.2465	1.5128	142.30
11250	66.25	70.63	18.00	1.2621	1.4979	141.03
11250	72.50	77.28	18.00	1.2793	1.4817	139.69
11250	78.75	83.92	18.00	1.2984	1.4642	138.28
11250	80.00	85.24	18.00	1.3025	1.4606	137.99
11250	85.00	90.54	18.00	1.3198	1.4452	136.79

PERFORMANCE DETAILS WITH 110% OF WATER FLOW						
WATER FLOW (GPM)	WET BULB TEMP (°F)	DRY BULB TEMP (°F)	RANGE (°F)	L/G	KaV/L	FAN POWER (HP)
13750	60.00	63.97	12.00	1.5057	1.3006	143.55
13750	66.25	70.63	12.00	1.5252	1.2873	142.18
13750	72.50	77.28	12.00	1.5468	1.2729	140.75
13750	78.75	83.92	12.00	1.5706	1.2574	139.26
13750	80.00	85.24	12.00	1.5757	1.2542	138.95
13750	85.00	90.54	12.00	1.5971	1.2407	137.69
13750	60.00	63.97	15.00	1.5252	1.2873	142.18
13750	66.25	70.63	15.00	1.5442	1.2746	140.92
13750	72.50	77.28	15.00	1.5652	1.2609	139.59
13750	78.75	83.92	15.00	1.5886	1.2461	138.19
13750	80.00	85.24	15.00	1.5935	1.2429	137.90
13750	85.00	90.54	15.00	1.6147	1.2299	136.71
13750	60.00	63.97	18.00	1.5445	1.2744	140.90
13750	66.25	70.63	18.00	1.5630	1.2623	139.73
13750	72.50	77.28	18.00	1.5835	1.2492	138.48
13750	78.75	83.92	18.00	1.6064	1.2349	137.16
13750	80.00	85.24	18.00	1.6113	1.2319	136.89
13750	85.00	90.54	18.00	1.6322	1.2193	135.76

3) Tower Performance by Method of Constant Gas

PERFORMANCE DETAILS WITH 100% OF WATER FLOW						
WATER FLOW (GPM)	WET BULB TEMP (^oF)	DRY BULB TEMP (^oF)	RANGE (^oF)	L/G	KaV/L	FAN POWER (HP)
12500	60.00	63.97	12.00	1.4413	1.3469	122.36
12500	66.25	70.63	12.00	1.4413	1.3469	125.84
12500	72.50	77.28	12.00	1.4413	1.3469	129.77
12500	78.75	83.92	12.00	1.4413	1.3469	134.25
12500	80.00	85.24	12.00	1.4413	1.3469	135.22
12500	85.00	90.54	12.00	1.4413	1.3469	139.40
12500	60.00	63.97	15.00	1.4413	1.3469	125.65
12500	66.25	70.63	15.00	1.4413	1.3469	129.07
12500	72.50	77.28	15.00	1.4413	1.3469	132.96
12500	78.75	83.92	15.00	1.4413	1.3469	137.40
12500	80.00	85.24	15.00	1.4413	1.3469	138.37
12500	85.00	90.54	15.00	1.4413	1.3469	142.53
12500	60.00	63.97	18.00	1.4413	1.3469	128.89
12500	66.25	70.63	18.00	1.4413	1.3469	132.27
12500	72.50	77.28	18.00	1.4413	1.3469	136.12
12500	78.75	83.92	18.00	1.4413	1.3469	140.54
12500	80.00	85.24	18.00	1.4413	1.3469	141.50
12500	85.00	90.54	18.00	1.4413	1.3469	145.65

PERFORMANCE DETAILS WITH 90% OF WATER FLOW						
WATER FLOW (GPM)	WET BULB TEMP (^oF)	DRY BULB TEMP (^oF)	RANGE (^oF)	L/G	KaV/L	FAN POWER (HP)
11250	60.00	63.97	12.00	1.2972	1.4653	121.03
11250	66.25	70.63	12.00	1.2972	1.4653	124.53
11250	72.50	77.28	12.00	1.2972	1.4653	128.49
11250	78.75	83.92	12.00	1.2972	1.4653	132.98
11250	80.00	85.24	12.00	1.2972	1.4653	133.96
11250	85.00	90.54	12.00	1.2972	1.4653	138.15
11250	60.00	63.97	15.00	1.2972	1.4653	124.01
11250	66.25	70.63	15.00	1.2972	1.4653	127.46
11250	72.50	77.28	15.00	1.2972	1.4653	131.37
11250	78.75	83.92	15.00	1.2972	1.4653	135.83
11250	80.00	85.24	15.00	1.2972	1.4653	136.80
11250	85.00	90.54	15.00	1.2972	1.4653	140.97
11250	60.00	63.97	18.00	1.2972	1.4653	126.95
11250	66.25	70.63	18.00	1.2972	1.4653	130.36
11250	72.50	77.28	18.00	1.2972	1.4653	134.23
11250	78.75	83.92	18.00	1.2972	1.4653	138.66
11250	80.00	85.24	18.00	1.2972	1.4653	139.62
11250	85.00	90.54	18.00	1.2972	1.4653	143.78

PERFORMANCE DETAILS WITH 110% OF WATER FLOW						
WATER FLOW (GPM)	WET BULB TEMP (°F)	DRY BULB TEMP (°F)	RANGE (°F)	L/G	KaV/L	FAN POWER (HP)
13750	60.00	63.97	12.00	1.5854	1.2480	123.68
13750	66.25	70.63	12.00	1.5854	1.2480	127.14
13750	72.50	77.28	12.00	1.5854	1.2480	131.05
13750	78.75	83.92	12.00	1.5854	1.2480	135.52
13750	80.00	85.24	12.00	1.5854	1.2480	136.48
13750	85.00	90.54	12.00	1.5854	1.2480	140.66
13750	60.00	63.97	15.00	1.5854	1.2480	127.27
13750	66.25	70.63	15.00	1.5854	1.2480	130.68
13750	72.50	77.28	15.00	1.5854	1.2480	134.54
13750	78.75	83.92	15.00	1.5854	1.2480	138.97
13750	80.00	85.24	15.00	1.5854	1.2480	139.94
13750	85.00	90.54	15.00	1.5854	1.2480	144.09
13750	60.00	63.97	18.00	1.5854	1.2480	130.81
13750	66.25	70.63	18.00	1.5854	1.2480	134.17
13750	72.50	77.28	18.00	1.5854	1.2480	138.01
13750	78.75	83.92	18.00	1.5854	1.2480	142.41
13750	80.00	85.24	18.00	1.5854	1.2480	143.37
13750	85.00	90.54	18.00	1.5854	1.2480	147.52

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17. Plotting of Tower Performance Curves

The performance curves consist of a minimum 3 sets of three curves each, which are presented as a plot of wet bulb temperature as abscissa versus cold water temperature as ordinate with the cooling range as parameter. According to CTI ATC 105 (Acceptance Test Code for Water Cooling Tower), graphical scaling shall be incremented so as to provide a minimum of 0.5ºF increments and no more than 5ºF per inch for both wet bulb and cold water temp.

The curves shall fully cover the range of variables specified in ATC 105 as follows;

- Wet Bulb Temperature: +/- 15ºF from Design WBT
- Dry Bulb Temperature: +/- 25ºF from Design DBT
- Cooling Range: +/- 20% from Design Range
- Water Flow Rate: +/- 10% from Design Flow Rate

The performance curves could be prepared by the simple method and detail method, which shall be discussed later. The performance (= cold water temperature) prediction of cooling tower by means of the simple method is made by a few design parameters as well as water flow rate, L/G, KaV/L, range, cold water temp., wet bulb tem., and fan bhp, while the performance prediction by the detail method is requiring all the actual cooling tower dimensions, thermal rating conditions, and all the mechanical rating conditions.

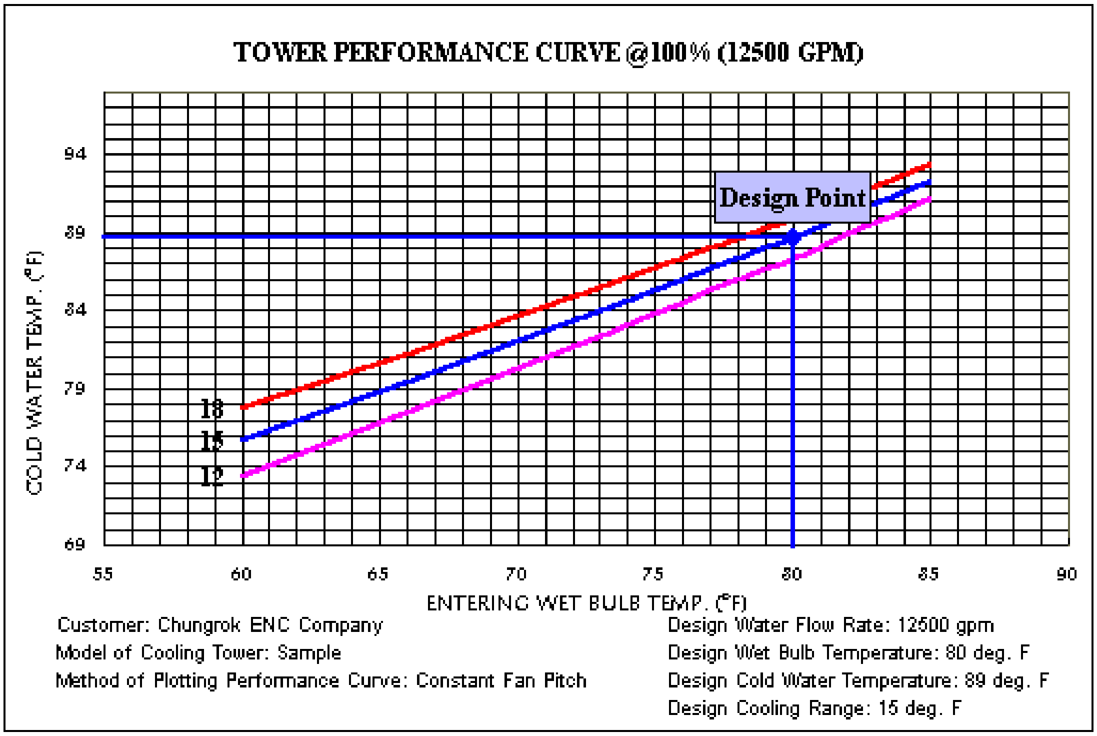
Example 17-1: Plot the performance curve by the method of constant fan pitch and simple method using the example 16-1.

(Solution)
The cold water summaries for the previous result of calculating the tower performance are as follow and the performance curves are being plotted.

1) Performance Curve @100% of Design Water Flow Rate

PERFORMANCE CURVE @ 100% OF WATER FLOW

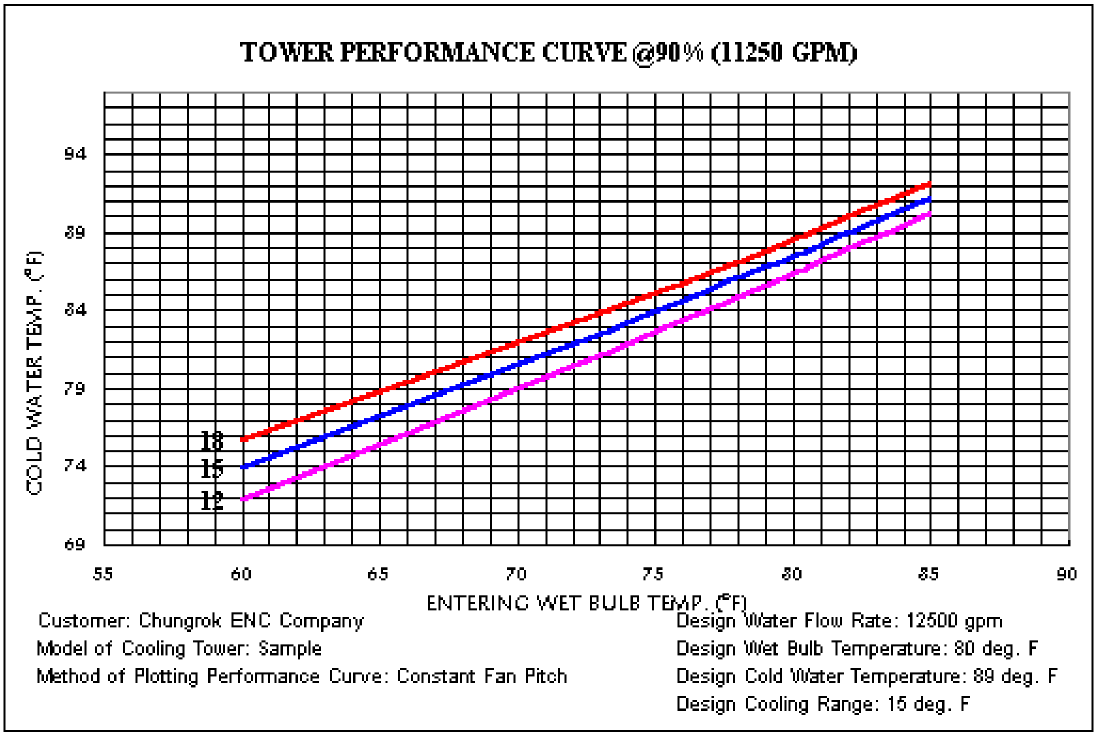
RANGE (°F)	WET BULB TEMPERATURE (°F)					
	60	66.25	72.50	78.75	80.00	85.00
12.00	73.43	77.61	81.95	86.47	87.39	91.17
15.00	75.72	79.58	83.62	87.85	88.72	92.29
18.00	77.72	81.31	85.09	89.08	89.91	93.30



2) Performance Curve @90% of Design Water Flow Rate

PERFORMANCE CURVE @ 90% OF WATER FLOW

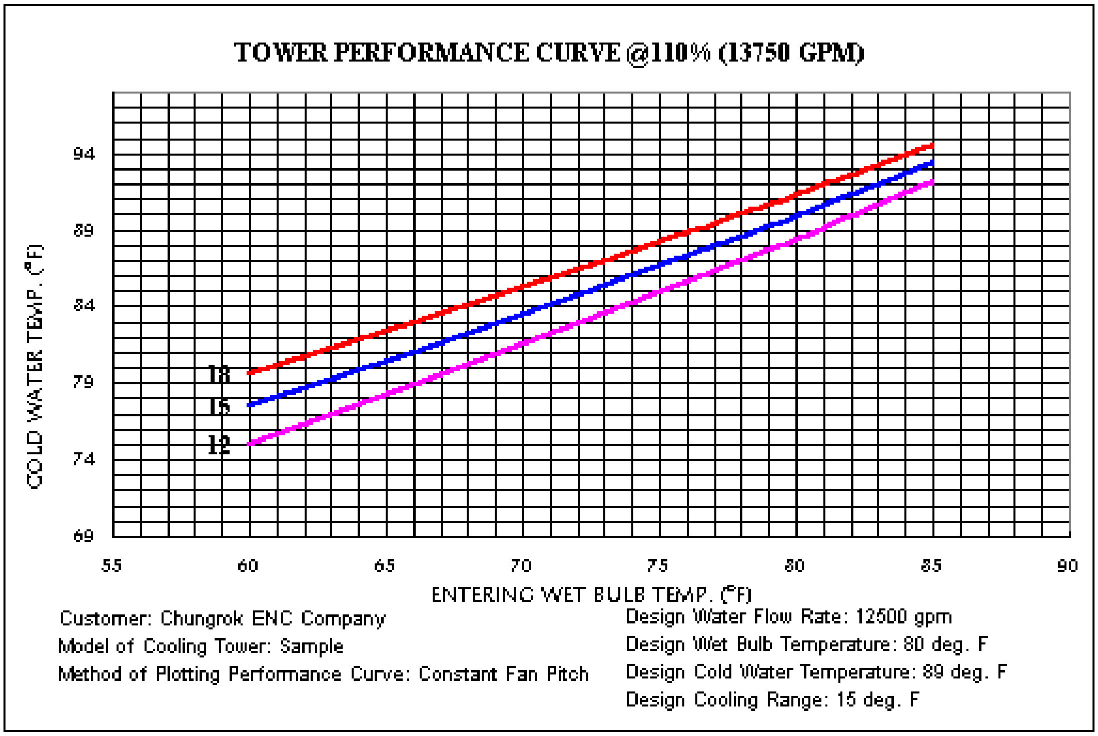
RANGE (°F)	WET BULB TEMPERATURE (°F)					
	60	66.25	72.50	78.75	80.00	85.00
12.00	71.88	76.21	80.71	85.38	86.33	90.23
15.00	73.94	77.97	82.18	86.59	87.49	91.19
18.00	75.76	79.53	83.49	87.66	88.52	92.06



3) Performance Curve @110% of Design Water Flow Rate

PERFORMANCE CURVE @ 110% OF WATER FLOW

RANGE (°F)	WET BULB TEMPERATURE (°F)					
	60	66.25	72.50	78.75	80.00	85.00
12.00	74.95	78.98	83.17	87.55	88.44	92.11
15.00	77.45	81.14	85.02	89.10	89.94	93.39
18.00	79.62	83.04	86.65	90.47	91.27	94.52



The file used here is same as one used for example 16-1.

<u>**Example 17-2**</u>. Plot a set of performance curves presenting the cold water temperature vs wet bulb temperature by the constant fan pitch and the detail method using the same example 16-1.

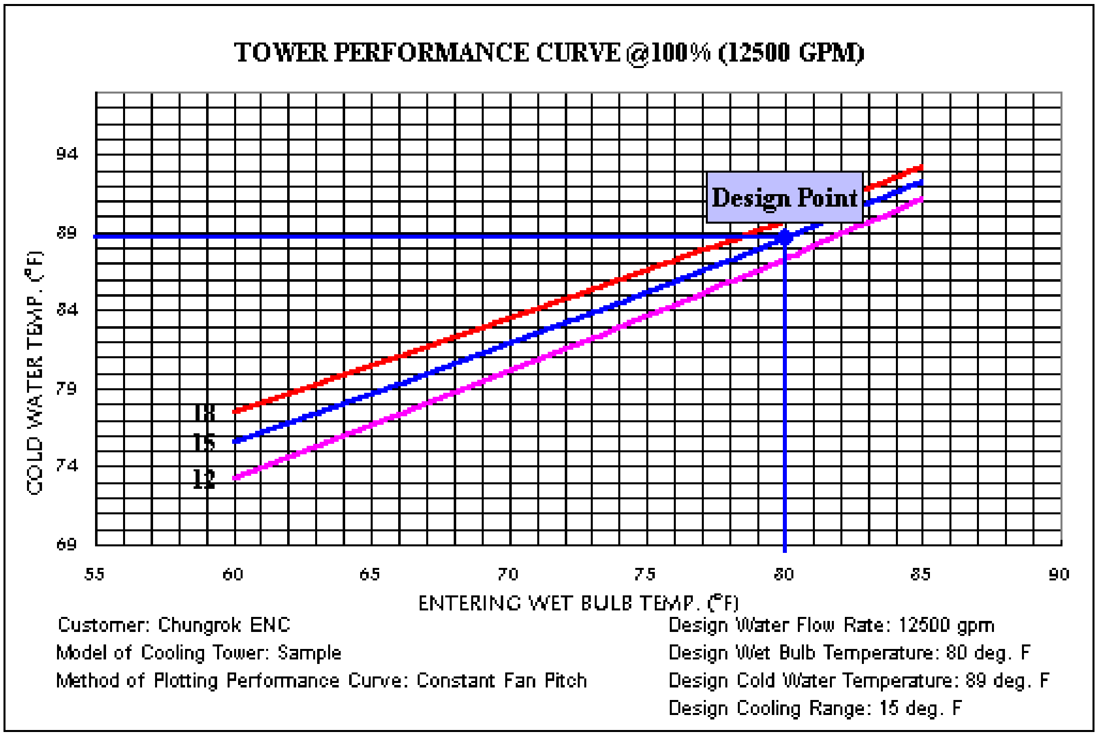
(Solution)

Tower Design Conditions			
Method of Performance Prediction	Constant Fan Pitch	% By-Pass Wall Water	3.27%
Site Altitude	0 ft	PD Coefficient @Drift Eliminator	1.80
Wet Bulb Temperature	80.00 °F	Fan Total Efficiency	79.2%
Relative Humidity	80.0%	Power Transmission Efficiency	91.2%
Number of Cells	1	Motor Power Margin	13.3%
Design Water Flow Rate	12,500 gpm	Motor Power	175 hp
Cell Length	42.0 ft	Fan Diamter	28 ft
Cell Width	42.0 ft	Number of Fan per Cell	1
Type of Air Inlet	Two Sides Open	Seal Disk Diameter	88.0 inch
Air Inlet height	15.0 ft	PD Coeficient @Fan Inlet	0.18
% Obstruction @Air Inlet	10.0%	Venturi Height of Stack	3.66 ft
PD Coefficient @Air Inlet	2.50	Design Hot Water Temperature	104.00 °F
Fill Model	CF1900	Design Cold Water Temperature	89.00 °F
Fill Depth	4.0 ft	Design Cooling Range	15.0 °F
PD Fill Multiplying Factor	1.00	Actual Range through Tower	15.507 °F
Fill KaV/L Multiplying Factor	1.00	Indicating of Wet Bulb Temperature	
KaV/L Correction Factor	0.09900	Minimum WBT for Plotting Curve	60.00 °F
% Obstruction @Fill	1.11%	Maximum WBT for Plotting Curve	85.00 °F
		Customer	Chungrok ENC
		Model of Cooling Tower	Sample

The curves are plotted for 80, 100 & 120% of cooling range on 90, 100 & 110% of design water flow rate. Note that these performance curves are based on the constant fan pitch and are plotted by the detail method. For further details of calculation, download the file.

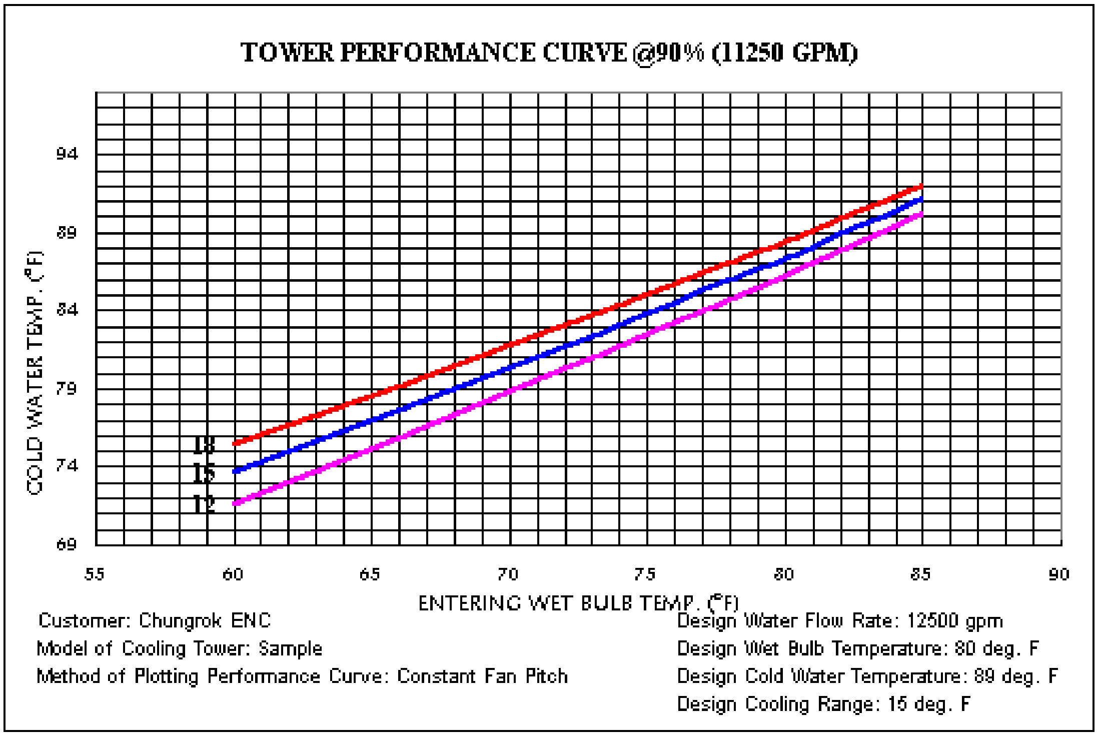
PERFORMANCE CURVE @ 100% OF WATER FLOW

RANGE (°F)	WET BULB TEMPERATURE (°F)					
	60.00	66.25	72.50	78.75	80.00	85.00
12.00	73.26	77.46	81.83	86.38	87.32	91.12
15.00	75.53	79.42	83.49	87.77	88.65	92.25
18.00	77.52	81.14	84.96	89.00	89.84	93.27



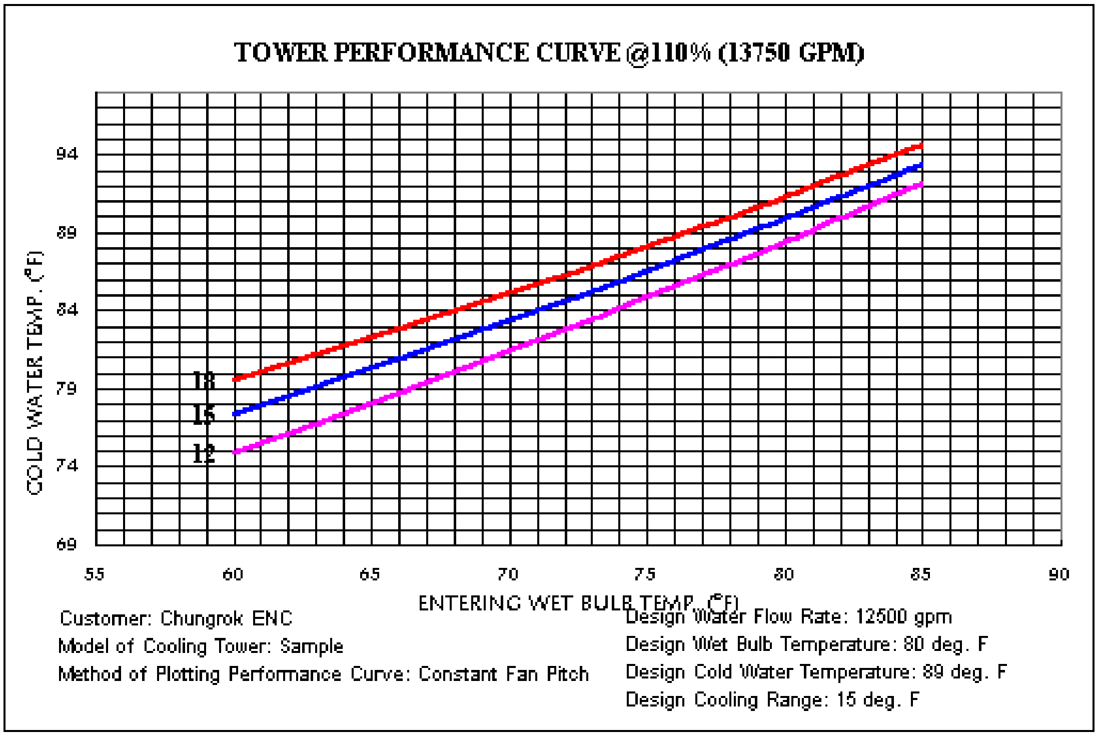
PERFORMANCE CURVE @ 90% OF WATER FLOW

RANGE (°F)	WET BULB TEMPERATURE (°F)					
	60.00	66.25	72.50	78.75	80.00	85.00
12.00	71.66	76.03	80.57	85.27	86.24	90.16
15.00	73.70	77.77	82.03	86.48	87.39	91.14
18.00	75.50	79.31	83.33	87.56	88.43	92.02



PERFORMANCE CURVE @ 110% OF WATER FLOW

RANGE (°F)	WET BULB TEMPERATURE (°F)					
	60.00	66.25	72.50	78.75	80.00	85.00
12.00	74.82	78.87	83.09	87.49	88.39	92.08
15.00	77.30	81.03	84.93	89.05	89.89	93.37
18.00	79.47	82.92	86.57	90.43	91.23	94.52



Please compare these results of cold water temperature obtained by detail method and simple method. You will see that the difference between these two results is very minor. So, the simple method is strongly recommended to use in practice.

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18. Estimation of Air Flow at No-Load Condition

After finishing the construction of cooling tower and before running the cooling tower commercially, the cooling tower at no-load condition is tested. So, it is a meaningful study to estimate the fan brake horsepower and air volume at fan under the condition of no water flow. Let's see these through the actual sample.

Example 18-1: Determine the fan bhp and air volume at the no-load condition under the assumption that the fan efficiency remains unchanged from the design conditions. The ambient dry bulb temperature is 63°F and relative humidity is 68%.

Tower Design Conditions					
Method of Performance Prediction	Constant Fan Pitch		% By-Pass Wall Water	3.27%	
Site Altitude	0	ft	PD Coefficient @Drift Eliminator	1.80	
Wet Bulb Temperature	80.00	°F	Fan Total Efficiency	79.2%	
Relative Humidity	80.0%		Power Transmisison Efficiency	91.2%	
Number of Cells	1		Motor Power Margin	13.3%	
Design Water Flow Rate	12,500	gpm	Motor Power	175	hp
Cell Length	42.0	ft	Fan Diamter	28	ft
Cell Width	42.0	ft	Number of Fan per Cell	1	
Type of Air Inlet	Two Sides Open		Seal Disk Diameter	88.0	inch
Air Inlet height	15.0	ft	PD Coeficient @Fan Inlet	0.18	
% Obstruction @Air Inlet	10.0%		Venturi Height of Stack	3.66	ft
PD Coefficient @Air Inlet	2.50		Design Hot Water Temperature	104.00	°F
Fill Model	CF 1900		Design Cold Water Temperature	89.00	°F
Fill Depth	4.0	ft	Design Cooling Range	15.0	°F
PD Fill Multiplying Factor	1.00		Actual Range through Tower	15.507	°F
Fill KaV/L Multiplying Factor	1.00		No Load Conditions		
KaV/L Correction Factor	0.09900		Ambient Dry Bulb Temperature	63.00	°F
% Obstruction @Fill	1.11%		Relative Humidity	68.0%	

(Solution)
The fan bhp at no-load condition could be estimated by three ways as described before. That is, constant fan pitch, constant fan bhp or constant gas. The constant fan pitch would be most proper choice of estimating the fan bhp and air volume at the no load condition. Below results were obtained by the method of constant fan pitch. The fan bhp @no load condition is considerably increased while the air volume is slightly increased. So, this is why the fan test must be carefully done at the no load and cold weather conditions.

Basic Thermal Rating Solving					
Air Mass Flow Rate	82,158.9	Lb/min	Air Velocity @Fan	1,915.7	fpm
Air Volume per Fan	1,098,683	acfm	Velocity Pressure @Fan	0.2296	inch Aq.
Net Fan Area per Fan	573.52	ft^2	Net Fan Power	149.85	bhp
			Total Fan Static Pressure	0.4570	inch Aq.
Pressure Drops Calculation					
1) Air Inlet			3) Drift Eliminator		
- Total Net Air Inlet	1,134.0	ft^2	- Net Area	1,744.4	ft^2
- Air Density	0.0754	Lb/ft^3	- Air Density	0.0754	Lb/ft^3
- Specific Volume	13.3727	ft^3/Lb	- Specific Volume	13.3727	ft^3/Lb
- Total Air Volume	1,098,683	acfm	- Air Volume	1,098,683	acfm
- Air Velocity	968.9	fpm	- Air Velocity	629.8	fpm
- Pressure Drop	0.1468	INCH Aq	- Pressure Drop	0.0447	inch Aq.
2) Fill			4) Fan Inlet		
- Total Net Fill Area	1,744.4	ft^2	- Air Density	0.0754	Lb/ft^3
- Water Loading	0.00	gpm/ft^2	- Air Velocity @Fan	1,915.7	fpm
- Average Air Density	0.0754	Lb/ft^3	- Pressure Drop	0.0413	inch Aq.
- Average Air Specific Volume	13.3727	ft^3/Lb	5. Velocity Recovery		
- Average Air Volume per Cell	1,098,683	acfm	- Efficiency of Fan Stack	77.4%	
- Average Fill Air Velocity	629.8	fpm	- Air Density	0.0754	Lb/ft^3
- Pressure Drop	0.2453	inch Aq.	- Velocity Recovery	0.0211	inch Aq.

Download Example File: ID-NOLOAD/TOWER, Version 1.03, File Size (no_load.zip)

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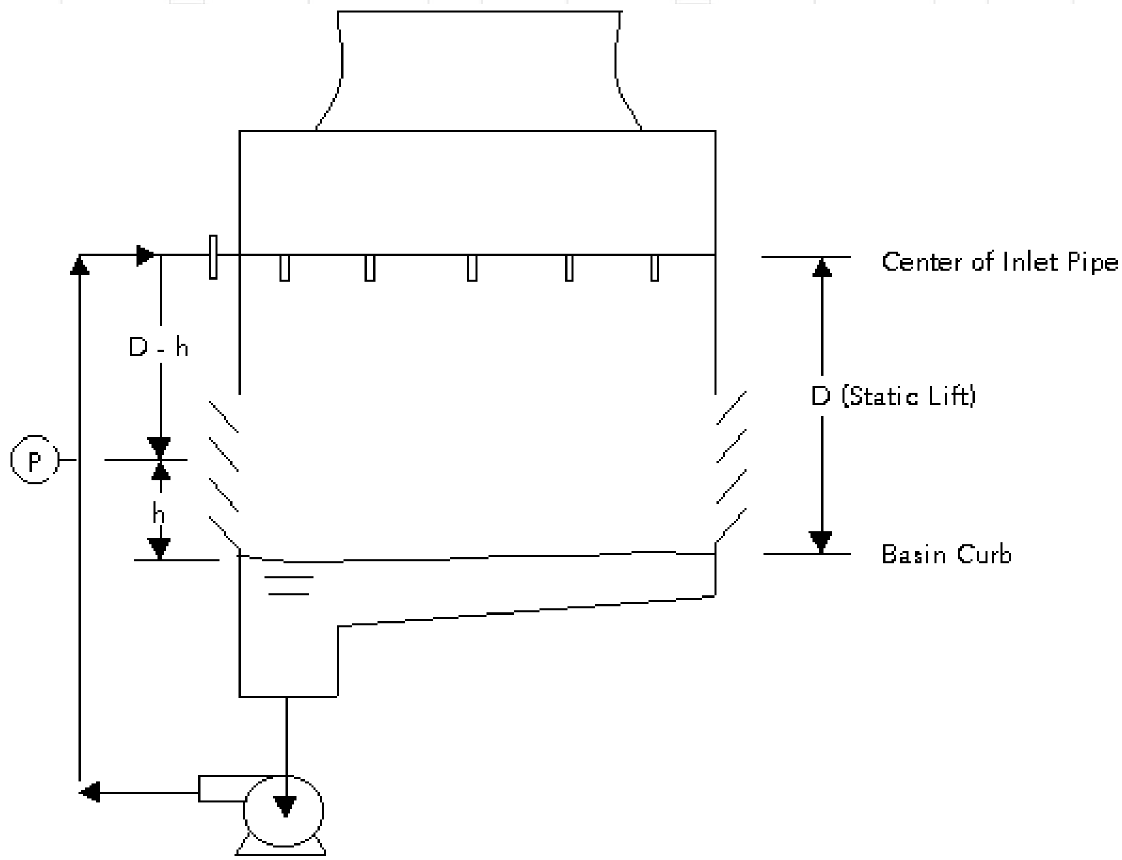
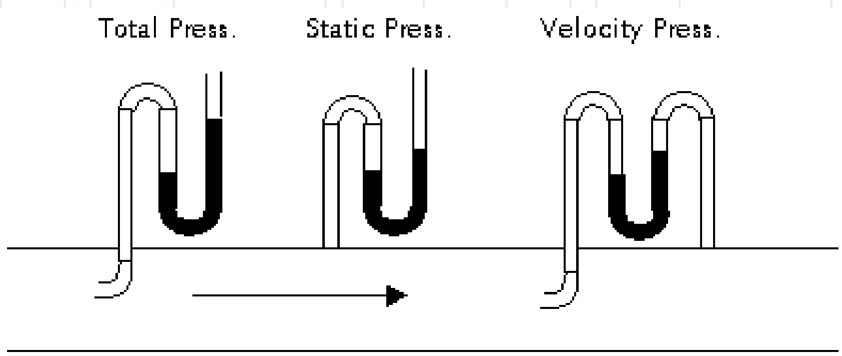
19. Determination of Pumping Head

What to predict the pumping head at the design water flow rate from measurement made at the test water flow rate is very important. The below example will show how to compute the pumping head.

Example 19-1: Determine the pumping heat for the assumption which the static lift (D) was 35 feet and vertical distance (h) of the pressure gauge above the basin curb was 5 feet. At this point, the pressure gauge indicated 25 psig. The 24 inch of pipe was used and the inner diameter was 22.624 inch. Let's assume that the test water flow rate was 14,000 GPM.





(Solution)
First, let's determine the equivalent length of piping and fittings between the point of pressure gauge and the center of inlet pipe.

- Vertical Leg, Length of D - h = 35 - 5 = 30.0 ft
- Horizontal Leg from the center of riser pipe and inlet pipe = 5 ft
- 24", 90º Welding Elbow (r/d = 1), Equivalent Length = 37.7 ft (from table of friction loss in term of length) Then, total equivalent length = 30 + 5 + 37.7 = 72.7 ft (based on 24 inch pipe)

Second, determine the friction loss in piping and fitting between the point of pressure gauge and center of inlet pipe. The head loss for 24 inch pipe per 100 feet and for 14,000 of test water flow rate is 1.30 ft from the friction table of steel pipe. Then, the friction loss in the feet could be obtained from below;

- Friction Loss = Head Loss per 100 ft x Equivalent Pipe Length = 1.30 / 100 x 72.7 = 0.95 ft

Third, determine the static pressure of test water flow at the center of inlet pipe.

- SPt = Test Pressure - (D - h) - Friction Loss = 25 psig x 2.309 - (35 - 5) - 0.95

(Note: 1 psi = 2.309 feet) = 26.78 ft

Fourth, determine the velocity pressure of test water flow at the enter of inlet pipe.

- Water Velocity @ 24 inch pipe = GPM x 0.1336798 / (0.7854 x (Inner Diameter / 12)^2) = 14,000 x 0.1336798 / (0.7854 x (22.624 / 12)^2 = 670.39 ft/min = 11.17 ft/sec
- Velocity Pressure = Velocity2 / 2g = 11.172 / (2 x 32.174) (1g = 32.174 ft/sec^2) = 1.94 ft

Fifth, let's compute the test pumping head

Test Pumping Head = SPt + Velocity Pressure + Static Lift = 26.78 + 1.94 + 35 = 63.72 ft

Sixth, determine the corrected total pressure to the design water flow rate.

- Test Pumping Head = Test Static Pressure + Test Velocity Pressure + Static Lift
- Test Total Pressure = Test Static Pressure + Test Velocity Pressure
- Test Pumping Head = Test Total Pressure + Static Lift
- Test Total Pressure = Test Pumping Head - Static Lift
- Corrected Total Pressure = Test Total Pressure x (Design Water Flow Rate / Test Water Flow Rate)^2 = (Test Pumping Head - Static Lift) x (Design Water Flow Rate / Test Water Flow Rate)^2 = (63.72 - 35) x (12,500 / 14,000)^2 = 22.90 ft

Finally, determine the predicted pumping head at the design water flow rate.

- Corrected Pumping Head = Corrected Total Pressure + Static Lift = 22.90 + 35 = 57.90 ft

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20. Determination of Line Voltage Drop

It is essential to measure the motor Kw input when to analysis the tower performance. This measurement could be done at the motor terminal box, but it is sometime more reasonable to measure the motor Kw input at the motor control center because the voltage and power factor indicators are usually not available at the tower fan deck. Now, the discussion will be focused how to determine the line voltage drop from the motor control center to the motor terminal box.

Example 20-1: Determine the line voltage loss for 1200 feet away from the motor control center to the motor using the example 6-1. Let's assume that the 175HP of motor is applied for the initial conditions of example 6-1 and the motor efficiency is 92.8% at the full load. The power supply is 460VAC, 60 Hz, 3 phases, and 4 poles. The cable between motor control center and the motor is "Bare Copper, 250 AWG. The measured ampere at the motor control center was 203A.

(Solution)
First, find DC resistance in Ohm per 1000 ft of cable length from the table of properties of conductors

- DC Resistance per 1000 feet = 0.0431 Ohms / 1000 ft

Multiplying Factor for Converting DC Resistance to AC Resistance from the table = 1.06
Then, AC Resistance = DC Resistance x (Cable Length / 1000) x Multiplying Factor = 0.0431 x (1,200 / 1,000) x 1.06 = 0.05482 Ohms

Second, determine the KW loss per the given power supply specifications.

- KW Loss for 3 Phases = 3 x I2R / 1000 = 3 x 2032 x 0.05482 / 1000 = 6.78 KW

Third, compute the net KW input to motor.

- Net KW Input to Motor = Measured KW @ Motor Control Center
- KW Line Loss - Measured KW = 1.7321 x Ampere x Voltage x Power Factor / 1000 = 1.7321 x 189 x 460 x 0.86977 / 1000 (Power Factor = 0.86977) = 140.68 KW

(The KW could be measured with the wattmeter or calculated after measuring ampere, voltage, and power factor.)

Finally, determine the motor shaft BHP. - Motor Shaft BHP = Measured KW @ Motor Control Center x Motor Efficiency @Full Load / 0.746 = 140.68 x 0.928 / 0.746 = 174.77 HP

21. Calculation of Tower Capability by Tower Characteristic Curve

The characteristic curve serves as a measure of the capability of the cooling tower to which it applies. It relates the familiar design term of KaV/L and L/G, and is of the form;

$KaV/L = C\ L/G^{-m}$

C and m are constant for a given cooling tower and are determined by the characteristics of the fill, while m is determined by end effects. The characteristic curve is used in conjunction with a KaV/L vs L/G relationship to determine performance. This curve may be termed "Design requirement" curve, since it is a measure of the degree of difficult of the design requirements, and has nothing to to with the physical characteristics of the tower. It is constructed by assuming values of L/G and computing the corresponding values of KaV/L using the following equations.

$$\frac{KaV}{L} = Cw\int_{tw1}^{tw2} \frac{dtw}{hw - ha} \quad \text{---------------- Eq. 21.1}$$

where, hw is the enthalpy of air-water vapor mixture at the bulk water temperature and ha is the enthalpy of air-water vapor mixture at the equilibrium wet bulb temperature.

The intersection of the characteristic and design requirements curves locates the design point. The manufacturer predicts that, when operating at the L/G value so located and at design water circulation rate, inlet water temperature, and wet bulb temperature, design outlet water temperature will be attained.

The test value of L/G is determined from Eq. 22-1, 22-10 & 22-12 which were derived in the chapter 22. From these equations, L/G dsn is the L/G value at the intersection of the characteristic and design requirement curves. The corresponding value of KaV/L is computed from above Eq. 63-1 using the test wet bulb and water temperatures. This point is then plotted, and a line a parallel to the characteristic curve is drawn through it. The intersection of this line and design requirements curve locates the L/G capability at the design conditions.

Example 21-1: Determine the tower capability using the characteristic curve for the initial design conditions and below test records.

COOLING TOWER TEST DATA INPUT

DESCRIPTIONS	DESIGN	TEST	UNIT
1. Circulated Water Flow	10,000.00	9,150.00	GPM
2. Hot Water Temperature	115.00	104.70	°F
3. Cold Water Temperature	85.00	79.30	°F
4. Inlet Wet Bulb Temperature	80.00		°F
5. Inlet Dry Bulb Temperature		78.46	°F
6. Relative Humidity	80.00%	78.00%	
7. Barometric Pressure	29.921	29.870	INCH HG
8. L/G Ratio	0.8600		
9. Fill Characteristic Slope	-0.6000		
10. Fan Horsepower	240	216	HP

(Solution)
First, let's compute the dry bulb temperature for the design and wet bulb temperature for the test conditions.

Inlet Dry Bulb Temperature Estimation for Design Condition		
Barometric Pressure	Inch hg	29.921
Inlet Wet Bulb Temperature	°F	80.00
Inlet Air Enthalpy @ WBT	BTU/LB	43.6907
Relative Humidity	%	80.0%
Inlet Air Enthalpy @ DBT & RH	BTU/LB	43.6907
Inlet Dry Bulb Temperature	°F	85.242
Inlet Air Density	Lb/FT^3	0.0718
Inlet Air Specific Volume	FT^3/LB	14.2230

Inlet Wet Bulb Temperature Estimation for Test Condition		
Barometric Pressure	Inch hg	29.870
Inlet Dry Bulb Temperature	°F	78.46
Relative Humidity	%	78.0%
Inlet Air Enthalpy @ DBT & RH	BTU/LB	36.8485
Inlet Air Enthalpy @ WBT	BTU/LB	36.8485
Inlet Wet Bulb Temperature	°F	73.098
Inlet Air Density	Lb/FT^3	0.0736
Inlet Air Specific Volume	FT^3/LB	13.7695

Exit Enthalpy @ Design = Inlet Enthalpy @ Design + L/G design x Range @ Design = 43.6907 + 0.8600 x (115 - 85) = 67.4907 BTU/LB

Exit Air Wet Bulb Temperature Estimation for Design		
Barometric Pressure	Inch hg	29.921
Design L/G Ratio		0.8600
Design Cooling Range	°F	30.0000
Exit Air Enthalpy	BTU/LB	69.4907
Equivalent Exit Air Enthalpy	BTU/LB	69.4907
Equivalent Exit Wet Bulb Temperature	°F	98.716
Exit Air Density	LB/FT^3	0.0693
Exit Specific Volume	FT^3/LB	15.0327

Exit Enthalpy @ Test = Inlet Enthalpy @ Test + L/G test x Range @ Test

The L/G test is calculated from the below formula.

L/G test = L/G design x (Water Flow test / Water Flow design) x (Fan BHP design / Fan BHP test)$^{1/3}$ x (Exit Air Density test / Exit Air Density design)$^{1/3}$ x (Exit Air Specific Volume test / Exit Air Specific Volume design)

Derivation details of L/G Test are as below:

Fan BHP = VOL x TP / (6356 x Fan Effi.)
= VOL x (VP + SP) / (6356 x Fan Effi.)
= VOL x (1/2g x Density x Vel^2 + K x 1/2g x Density x Vel^2) / (6356 x Fan Effi.)
= VOL x Density x Vel^2 x (1 + K) / 1/2g / (6356 x Fan Effi.)
= VOL x Density x Vel^2 x (Area^2 / Area^2) x (1 + K) /1/2g / (6356 x Fan Effi.)
= VOL x Density x VOL^2 x 1 / Area^2 x (1 + K) /1/2g / (6356 x Fan Effi.)
(The term of 1 / Area^2 x (1 + K) / 1/2g / (6357 x Fan Effi.) could be considered as a constant under the assumption that the fan efficiency at the design conditions is equal to the fan efficiency at the test conditions.)

Then, above equation could be expressed to Constant = Fan BHP / (VOL^3 x Density). Therefore, the following relationship is established.

Constant = BHP dsn / (VOL dsn^3 x Density dsn) = BHP test / (VOL test^3 x Density test)
Let's rewrite this relationship for the term of Vol test.
VOL test^3 = (BHP test / BHP dsn) x (Density dsn / Density test) x VOL dsn^3

VOL test = (BHP test / BHP dsn)$^{1/3}$ x (Density dsn / Density test)$^{1/3}$ x VOL dsn
= (BHP test / BHP dsn)$^{1/3}$ x (Density dsn / Density test)$^{1/3}$ x L dsn / (L/G dsn) x SV dsn (VOL dsn = L dsn / (L/G dsn) x SV dsn)

L/G test = L test / G test
= L test / (VOL test x SV test)
= L test x SV test / VOL test
= L test x SV test / ((BHP test / BHP dsn)$^{1/3}$ x (Density dsn / Density test)$^{1/3}$ x L dsn / (L/G dsn) x SV dsn)
= L/G dsn x (L test / L dsn) x (BHP dsn / BHP test)$^{1/3}$ x (Density test / Density dsn)$^{1/3}$ x (SV test /SV dsn)

Then,

L/G test = 0.860 x (9,150 / 10,000) x (240.0 / 216.0)$^{1/3}$ x (Exit Air Density test / 0.0693)$^{1/3}$ x (Exit Air Specific Volume test / 15.0327)

The two factors in the right side of above equation are unknown and these must be computed by the method of try and error. The approximation of air temperature at the first step is an average temperature of test hot and cold water temperatures. Density and S/Volume are computed per the approximated exit air temperature, and L/G test is calculated as per the above equation which is including two unknown factors. The enthalpy under the approximation is obtained and then the new exit air temperature is computed when to iterate until the approximated enthalpy equals to the equivalent enthalpy with varying the temperature.

Finding of L/G Test by Try & Error mehtod

	Air Temp	Density	S/Volume	L/G test	Enthalpy	Air Temp	Equi. Enth.
1st Trial	92.00000	0.07031	14.69718	0.80078	57.18830	90.83850	57.18829
2nd Trial	90.83850	0.07051	14.63844	0.79832	57.12591	90.79448	57.12591
3rd Trial	90.79448	0.07051	14.63624	0.79823	57.12357	90.79283	57.12357
4th Trial	90.79283	0.07051	14.63615	0.79823	57.12348	90.79276	57.12348
5th Trial	90.79276	0.07051	14.63615	0.79823	57.12348	90.79276	57.12348
6th Trial	90.79276	0.07051	14.63615	0.79823	57.12348	90.79276	57.12348

The L/G test calculated by the try & error method is 0.79823.
Then,

Exit Enthalpy @ Test = 36.8485 + (104.7 - 79.3) x L/G test
= 36.8485 + 25.4 x 0.79823
= 57.1235

Second, calculate of NTU at Design and Test respectively.

Tower Demand (NTU) Calculation @ Design						
WATER SIDE			AIR SIDE		ENTHALPY DIFF.	
DESCRIPTIONS	tw (°F)	hw (BTU/lb)	DESCRIPTIONS	ha (BTU/lb)	hw - ha	1/(hw-ha)
tw_1 + 0.1 x Range	88.00	53.2477	ha_1 + 0.1 x L/G x Range	46.2707	6.9770	0.1433
tw_1 + 0.4 x Range	97.00	66.5773	ha_1 + 0.4 x L/G x Range	54.0107	12.5666	0.0796
tw_1 + 0.6 x Range	103.00	77.3676	ha_1 + 0.6 x L/G x Range	59.1707	18.1969	0.0550
tw_1 + 0.9 x Range	112.00	97.2029	ha_1 + 0.9 x L/G x Range	66.9107	30.2922	0.0330
Sum of 1 / (hw - ha).........						0.3109
Tower Demand (NTU) = Sum of 1 / (hw - ha) / 4 * RANGE						2.3315

Tower Demand (NTU) Calculation @ Test						
WATER SIDE			AIR SIDE		ENTHALPY DIFF.	
DESCRIPTIONS	tw (°F)	hw (BTU/lb)	DESCRIPTIONS	ha (BTU/lb)	hw - ha	1/(hw-ha)
tw_1 + 0.1 x Range	81.84	45.7711	ha_1 + 0.1 x L/G x Range	38.8760	6.8951	0.1450
tw_1 + 0.4 x Range	89.46	55.2675	ha_1 + 0.4 x L/G x Range	44.9585	10.3090	0.0970
tw_1 + 0.6 x Range	94.54	62.6941	ha_1 + 0.6 x L/G x Range	49.0135	13.6806	0.0731
tw_1 + 0.9 x Range	102.16	75.8453	ha_1 + 0.9 x L/G x Range	55.0960	20.7494	0.0482
Sum of 1 / (hw - ha).........						0.3633
Tower Demand (NTU) = Sum of 1 / (hw - ha) / 4 * RANGE						2.3071

Third, find a value of L/G which the test tower characteristic curve intersects with the design NTU curve. The test C value of tower characteristic is obtained from the relationship of NTUtest = C x L/G^{-m}.

Test C Value = NTUtest / $L/Gtest^{-m}$
= NTUtest x $L/Gtest^{m}$
= 2.3071 x $0.7982^{0.600}$
= 2.0153

It is to find the value of L/G intersection with the iteration until the test NTU equals to Design NTU varying L/C Intersection.

Test NTU = Test C x L/C $Intersection^{-m}$
Design NTU at L/G Intersection must be calculated using the below NTU calculation table. The L/G in Air Side of NTU calculation table is to use L/C Intersection.

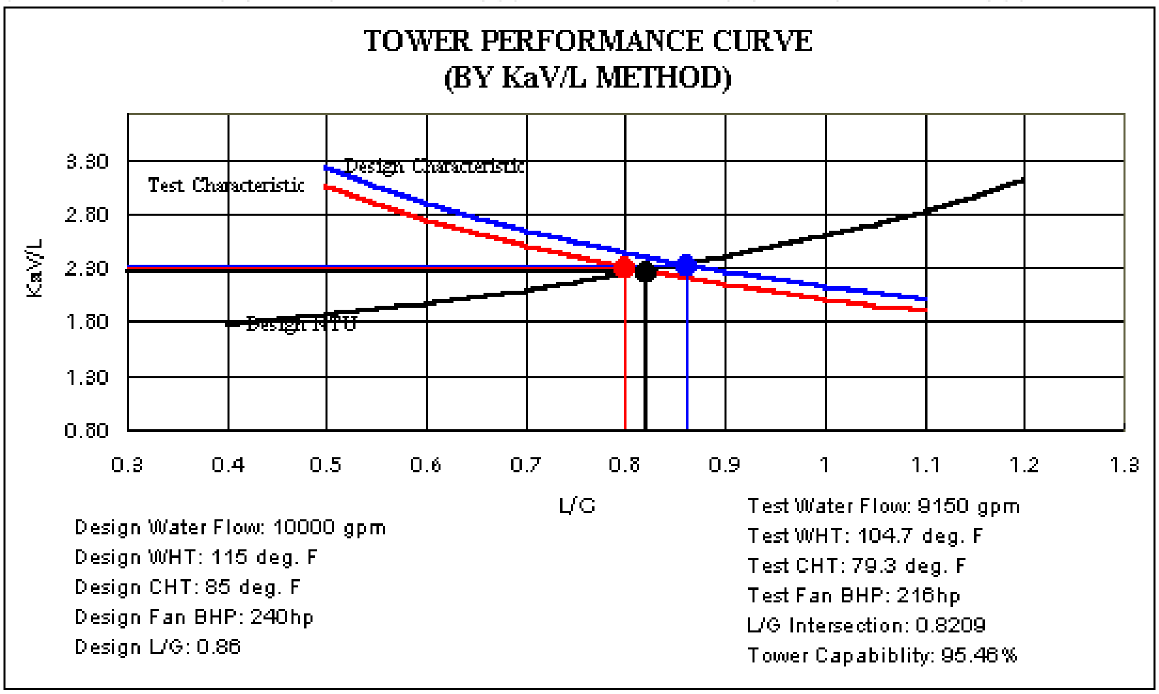
Intersection Calculation of Test Characteristic Curve to Design NTU						
WATER SIDE			AIR SIDE		ENTHALPY DIFF.	
DESCRIPTIONS	tw (°F)	hw (BTU/lb)	DESCRIPTIONS	ha (BTU/lb)	hw - ha	1/(hw-ha)
tw_1 + 0.1 x Range	88.00	53.2477	ha_1 + 0.1 x L/G x Range	46.1535	7.0942	0.1410
tw_1 + 0.4 x Range	97.00	66.5773	ha_1 + 0.4 x L/G x Range	53.5418	13.0355	0.0767
tw_1 + 0.6 x Range	103.00	77.3676	ha_1 + 0.6 x L/G x Range	58.4673	18.9003	0.0529
tw_1 + 0.9 x Range	112.00	97.2029	ha_1 + 0.9 x L/G x Range	65.8557	31.3473	0.0319
Sum of 1 / (hw - ha).........						0.3025
Tower Demand (NTU) = Sum of 1 / (hw - ha) / 4 * RANGE						2.2686

L/G Intersection of Test Characteristic Curve to Design NTU	
Slope of Tower Characteristic Curve	-0.6000
Test NTU	2.3071
Test C Value of Tower Characteristic	2.0153
Test NTU @ Intersected L/G	2.2686
Design NTU @ Intersected L/G	2.2686
L/G Intersection	0.8209

Finally, determine the tower capability as per the equation below.

Tower Capability = (L/G intersection / L/G design) x 100 (%)
= (0.8209 / 0.8600) x 100 = 95.46%

The below curve represents the tower capability using Design NTU, Test NTU and Design Characteristic curves.



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22. Calculation of Tower Capability by Tower Performance Curve

When to calculate the tower capability by the method of tower performance curves, it is required to convert the test water flow rate to the water flow rate at the design conditions.

The equation is necessary to predict the amount of water that the tower can cool, at test temperatures, if the fan drives were loaded to design power, and is based on the following assumptions:

- The GPM capacity of a cooling tower is directly proportional to the air flow.
- The air flow is proportional to the cube root of the power delivered to the fans.

Actually the GPM capacity of a cooling tower deviates from the straight line relationship with air flow, due to changes in drop size, interfacial area, and distribution, but the error is small for small changes in air flows. Also, air flow deviates from the cube root relationship with power, due to the fact that a change in water loading is involved, and to the fact that fan efficiency does not remain exactly constant as air and water flows, and hence static pressure, are changed. For these reasons it is desirable that water circulation rate and fan power beheld reasonably close to design during test.

In summary, the closer water circulation rate and fan power are to design, the less will be the error due to the adjustment of test water circulation rate by means of equations below.

The determination of predicted GPM from the performance curves is accomplished in the following manner:

- Outlet water temperatures at the test wet bulb temperature are read from the performance curve. These values are shown on a table titled "Cold Water Temp. @ Test WBT".
- The data from table are then plotted to obtain the curves shown in "Cold Water Temp. vs Ranges".
- The cold water temperatures at the test range are then read from the curves. These are shown in table titled "Cold Water Temp. @ Test WBT & Range".
- The data in table are plotted to produce the curve shown in "Water Flow Rate vs Cold Water Temp.". The predicted GPM is found from this curve.
- Compute the adjusted test GPM.
- Compute the performance from the ratio of adjusted test GPM to predicted GPM.

There are three methods in converting the test water flow rate to the water flow of the design conditions. They vary on the assumption as follows;

- Constant Fan BHP (BHP off = BHP dsn)
- Constant Fan Pitch (VOL off = VOL dsn)
- Constant Air Mass Flow Rate (GAS off = GAS dsn)

1) Adjusted Test Water Flow Rate @ Constant Fan BHP

The relation of L/G off at the constant fan bhp to L/G dsn was already discussed and is given to Eq. 16-6. This equation could be expressed to the relation of L/G test and L/G dsn.

L/G test = L/G dsn x (L test / L dsn) x (DEN test / DEN test)$^{1/3}$ x (SV test / SV dsn) Eq. 22-1

L/G test = L test / G test
= L test / (Vol test / SV test)
= L test x SV test / Vol test

Eq. 22-2

BHP test = VOL test3 x DEN test VOL test

VOL test = BHP test$^{1/3}$ / DEN test$^{1/3}$ Eq. 22-3

Substitute VOL test of right side in Eq. 22-2 by Eq. 22-3.

L/G test = L test x SV test / (BHP test$^{1/3}$ / DEN test$^{1/3}$)

Eq. 22-4

L/G dsn = L dsn / G dsn
= L dsn / (Vol dsn / SV dsn)
= L dsn x SV dsn / Vol dsn

Eq. 22-5

BHP dsn = VOL dsn^3 x DEN dsn VOL dsn = BHP dsn$^{1/3}$ / DEN dsn$^{1/3}$

Eq. 22-6

Substitute VOL dsn of right side in Eq. 22-5 by Eq. 22-6.

L/G dsn = L dsn x SV dsn / (BHP dsn$^{1/3}$ / DEN dsn$^{1/3}$)

Eq. 22-7

Substitute L/G test and L/G dsn in Eq. 22-1 by Eq. 22-4 and Eq. 22-7.

L test x SV test / (BHP test$^{1/3}$ / DEN test$^{1/3}$)
= L dsn x SV dsn / (BHP dsn$^{1/3}$ / DEN dsn$^{1/3}$) x (L test / L dsn) x (DEN test / DEN test)$^{1/3}$ x (SV test / SV dsn)

1 = (BHP test / BHP dsn)$^{1/3}$

Eq. 22-8

Therefore, Eq. 22-8 can be rewritten to Eq. 22-9.

L adj = L test x (BHP dsn / BHP test)$^{1/3}$

Eq. 22-9

2) Adjusted Test Water Flow Rate @ Constant Fan Pitch

The relation of L/G off at the constant fan pitch to L/G dsn was discussed and is given to Eq. 16-14. Also, this equation could be expressed to the relation of L/G test and L/G dsn.

L/G test   L/G dsn x (L test / L dsn) x (SV test / SV dsn)   Eq.22-10

Substitute L/G test and L/G dsn in Eq. 22-10 by Eq. 22-4 and Eq. 22-7 which were derived previously.

L test x SV test / (BHP test$^{1/3}$ / DEN test$^{1/3}$)
= L dsn x SV dsn / (BHP dsn$^{1/3}$ / DEN dsn$^{1/3}$) x (L test / L dsn) x (SV test / SV dsn)

1 = (BHP test / BHP dsn)$^{1/3}$ x (DEN dsn / DEN test)$^{1/3}$

Eq. 22-11

Therefore, Eq. 22-11 can be rewritten to Eq. 22-12.

L adj   = L test x (BHP dsn / BHP test)$^{1/3}$ x (DEN test / DEN dsn)$^{1/3}$

3) Adjusted Test Water Flow Rate @ Constant Gas

The relation of L/G off at the constant gas to L/G dsn was discussed and is given to Eq. 16-22. Also, this equation could be expressed to the relation of L/G test and L/G dsn.

L/G test   = L/G dsn x (L test / L dsn)   Eq. 22-12

Substitute L/G test and L/G dsn in Eq. 64-12 by Eq. 64-4 and Eq. 64-7 which were derived previously.

L test x SV test / (BHP test$^{1/3}$ / DEN test$^{1/3}$) = L dsn x SV dsn / (BHP dsn$^{1/3}$ / DEN dsn$^{1/3}$) x (L test / L dsn)

1 = (BHP test / BHP dsn)$^{1/3}$ x (DEN dsn / DEN test)$^{1/3}$ x (SV dsn / SV test)

Eq. 22-13

Therefore, Eq. 22-13 can be rewritten to Eq. 22-14.

L adj   L test x (BHP dsn / BHP test)$^{1/3}$ x (DEN test / DEN dsn)$^{1/3}$ x (SV test / SV dsn)

Eq. 22-14

Example 22-1: Determine the capability of cooling tower on the basis of Constant Fan Pitch by the analysis method of performance curve using the same design and test conditions as example 21-1.

(Solution)
Sorry. Will add the solution later.

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