PDHonline Course M374 (2 PDH)

Cooling Towers - Basic Calculations

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Course Content:

1. Introduction:

1. To design **Cooling Towers** someone can find a forest of requirements, guidance construction activities and technical formulae. Every day a student or a professional is looking for a short and timely handbook with practical information and comprehensive calculations the way he can conclude a work without wasting too much his precious time. This is the main subject of this sketch.

2. Cooling towers are a very important part of many chemical plants. The primary task of a cooling tower is to reject heat into the atmosphere. They represent a relatively inexpensive and dependable means of removing low-grade heat from cooling water.

3. Following the rules here described someone can easily calculate a process for a basic Mechanical Draft Cooling Tower.

2. Cooling Tower Types:

Cooling towers fall into two main categories: **Natural draft and Mechanical draft**.

Natural Draft Towers - use very large concrete chimneys to introduce air through the media. Due to the large size of these towers, they are generally used for water flow rates **above 45,000 m^3/h**. These types of towers are used only by utility power stations.

Mechanical Draft Towers - utilize large fans to force or suck the air through circulated water. The water falls downward over fill surfaces, which help increase the contact time between the water and the air - this helps maximize heat transfer between the two.

An open circuit cooling tower is a specialized heat exchanger in which two fluids (air and water) are brought into direct contact with each other to affect the transfer of heat.

Mechanical Draft Towers are available in the following airflow arrangements:

1. **Counter flow** - induced or forced draft;
2. **Cross flow** - induced or forced draft.

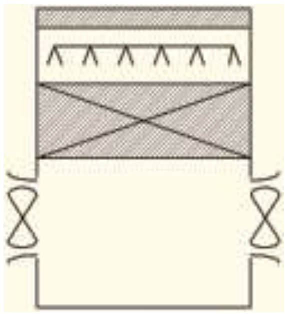
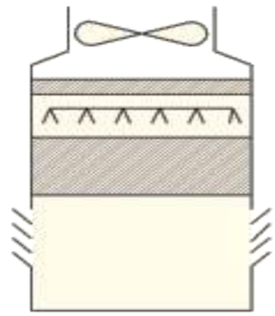
1. **In the counter flow draft** – in this typical design, hot water enters at the top, while the air is introduced **at the bottom** and exits through the top as warm water falls downward. Both forced and induced draft fans are used.

Because of the need for extended intake and discharge plenums; the use of high pressure spray systems and the typically higher air pressure losses, **some of the smaller counter flow towers** are physically higher; require more pump head and utilize more fan power than their cross flow counterparts.

2. **In the cross flow draft** – in this special, but common design, the air flows horizontally, across the downward fall of water. The air, however, is introduced at one side **(single-flow tower)** or opposite sides **(double-flow tower)**.

An induced or forced draft fan draws the air **across the wetted fill** and expels it through the top of the structure as the water cascades down through the tower.

Many cooling towers are assemblies of two or more individual cooling towers or "cells." Multiple-cell towers as an eight-cell tower, can be lineal, square, or round depending upon the shape of the individual cells and whether the air inlets are located on the sides or bottoms of the cells.

Forced Draft Axial Fans 	**Induced Draft Axial Fans** 

The common called "**make-up water"** source is used to replenish water lost to evaporation. Hot water from heat exchangers is sent to the cooling tower. The water exits the cooling tower and is sent back to other units for further process conditions.

3. Components of Cooling Towers:

The **basic components** of an evaporative tower are: Frame and casing, fill, cold water basin, drift eliminators, air inlet, louvers, nozzles and fans.

Frame and casing: Most towers have structural frames that support the exterior enclosures (casings), motors, fans, and other components. With some smaller designs, such as some glass fiber units, the casing may essentially be the frame.

Fills: Most towers employ fills (made of plastic or wood) to facilitate heat transfer by maximizing water and air contact. Fill can either be splash or film type.

With **splash fill**, water falls over successive layers of horizontal splash bars, continuously breaking into smaller droplets, while also wetting the fill surface. Plastic splash fill promotes better heat transfer than the wood splash fill.

Film fills – these components consist of thin, closely spaced plastic surfaces over which the water spreads, forming a thin film in contact with the air. These surfaces may be flat, corrugated, honeycombed, or other patterns. The film type of fill is the more efficient and provides same heat transfer in a smaller volume than the splash fill.

Cold water basin: is located at or near the bottom of the tower, receives the cooled water that flows down through the tower and fills. The basin usually has a sump or low point for the cold water discharge connection. In many tower designs, the cold water basin is beneath the entire fill.

In some forced draft counter flow design, however, the water at the bottom of the fill is channeled to a perimeter trough that functions as the cold water basin. Propeller fans are mounted beneath the fill to blow the air up through the tower. With this design, the tower is mounted on legs, providing easy access to the fans and their motors.

Drift eliminators: These capture water droplets entrapped in the air stream that otherwise would be lost to the atmosphere.

Air inlet: This is the point of entry for the air entering a tower. The inlet may take up an entire side of a tower–cross flow design– or be located low on the side or the bottom of counter flow designs.

Louvers: Generally, cross-flow towers have inlet louvers. The purpose of louvers is to equalize air flow into the fill and retain the water within the tower. Many counter flow tower designs do not require louvers.

Nozzles: These provide the water sprays to wet the fill. Uniform water distribution at the top of the fill is essential to achieve proper wetting of the entire fill surface. Nozzles can either be fixed in place and have either round or square spray patterns or can be part of a rotating assembly as found in some circular cross-section towers.

Fans: Both axial (propeller type) and centrifugal fans are used in towers. Generally, propeller fans are used in induced draft towers and both propeller and centrifugal fans are found in forced draft towers. Propeller fans are fabricated from galvanized, aluminum, or molded glass fiber reinforced plastic.

Glass fiber - is also widely used for cooling tower **casings and basins**, giving long life and protection from the harmful effects of many chemicals.

Plastics - are widely used for fill, including **PVC, polypropylene, and other polymers**. Treated wood splash fill is still specified for wood towers, but plastic splash fill is also widely used when water conditions mandate the use of splash fill.

Film fill - because it offers greater heat transfer efficiency, is the fill of choice for applications where the circulating water is generally free of debris that could plug the fill passageways.

Plastics also find wide use as **nozzle materials**. Many nozzles are being made of PVC, ABS, polypropylene, and glass-filled nylon. Aluminum, glass fiber, and hot-dipped galvanized steel are commonly used fan materials.

4. Psychrometrics Concepts:

Psychrometrics or **psychrometry** are terms used to describe the field of engineering concerned with the determination of physical and thermodynamic properties of gas-vapor mixtures. The term derives from the Greek “*psuchron*” meaning "cold" and “*metron*” meaning "means of measurement".

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).

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 °R**.

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 °R**.

The temperature and barometric pressure of atmospheric air vary considerably with altitude as well as with local geographic and weather conditions.

- **At sea level**, standard temperature is **59°F**; standard barometric pressure is **29.921 inch Hg**.
- 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**.

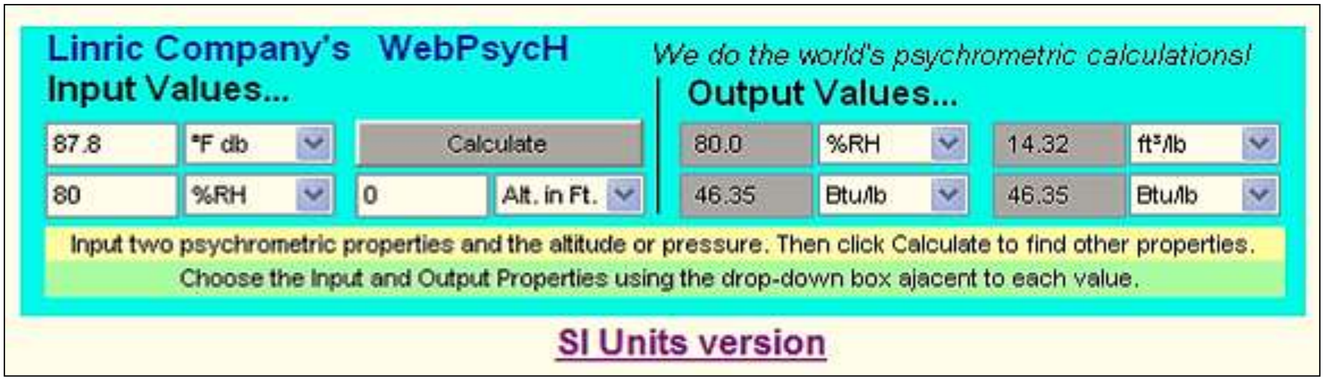
5. Main Properties of Psychrometrics:

- Dry-bulb temperature (*DBT*) is that of an air sample, as determined by an ordinary thermometer, the thermometer's bulb being dry. The SI units for temperature are kelvins or degrees Celsius; other units are degrees Fahrenheit and degrees Rankine.
- Wet-bulb temperature (*WBT*) is that of an air sample after it has passed through a constant-pressure, ideal, adiabatic saturation process, that is, after the air has passed over a large surface of liquid water in an insulated channel. When the air sample is saturated with water, the WBT will read the same as the DBT. The slope of the line of constant WBT reflects the heat of vaporization of the water required to saturate the air of a given relative humidity.
- Dew point temperature (*DPT*) is that temperature at which a moist air sample at the same pressure would reach water vapor "saturation." At this point further removal of heat would result in water vapor condensing into liquid water fog or (if below freezing) solid hoarfrost.
- Relative humidity (*RH*) is the ratio of the mole fraction of water vapor to the mole fraction of saturated moist air at the same temperature and pressure. RH is dimensionless, and is usually expressed as a percentage. Lines of constant RH reflect the physics of air and water: they are determined via experimental measurement.
- Humidity ratio (also known as moisture content or mixing ratio) is the proportion of mass of water vapor per unit mass of dry air at the given conditions (DBT, WBT, DPT, RH, etc.). It is typically the ordinate (vertical axis) of the graph. For a given DBT there will be a particular humidity ratio for which the air sample is at 100% relative humidity: the relationship reflects the physics of water and air and must be measured. Humidity ratio is dimensionless, but is sometimes expressed as grams of water per kilogram of dry air or grains of water per pound of air (7000 grains equal 1 pound). Specific humidity is related to humidity ratio but always lower in value as it expresses the proportion of the mass of water vapor per unit mass of the air sample (dry air plus the water vapor).
- Specific enthalpy symbolized by *h*, also called heat content per unit mass, is the sum of the internal (heat) energy of the moist air in question, including the heat of the air and water vapor within. In the approximation of ideal gases, lines of constant enthalpy are parallel to lines of constant WBT. Enthalpy is given in (SI) joules per kilogram of air or BTU per pound of dry air.

- Specific volume, also called *inverse density*, is the volume per unit mass of the air sample. The SI units are cubic meters per kilogram of dry air; other units are cubic feet per pound of dry air.

5.1 Psychrometrics Calculation:

There are hundreds of psychometrics charts and calculation spreadsheets to be downloaded. For the examples below try http://www.linric.com/webpsy.htm



Example 5-1. Calculate the air density, specific volume, and enthalpy in US units at the ambient conditions of DBT 87.8°F, RH 80% and sea level.

Answers:
Air Density = 0.0714 lb/ft³
Air Specific Volume = 14.32 ft³/lb - dry air
Air Enthalpy = 46.35 Btu/lb - dry air

Example 5-2. Calculate the air density, specific volume, and enthalpy in US at the ambient conditions of DBT 87.8°F, RH 0% (Dry Air), and sea level.

Answers:
Air Density: 0.0723 lb/ft³
Air Specific Volume: 13.8224 ft³/lb - dry air
Air Enthalpy: 21.1196 Btu/lb - dry air

Example 5-3. Calculate the air density, specific volume, and enthalpy in US at the ambient conditions of DBT 87.8°F, RH 80%, and 1,000 feet in altitude.

Answers:
Air Density: 0.0688 lb/ft³
Air Specific Volume: 14.8824 ft³/lb - dry air
Air Enthalpy: 47.3494 Btu/lb - dry air

6. Cooling Tower Performance:

The important parameters, from the point of determining the performance of cooling towers, are:

i) **"Range"** - is the difference between the cooling tower water inlet and outlet temperature.

ii) **"Approach"** - is the difference between the cooling tower outlet cold water temperature and ambient wet bulb temperature. Although, both range and approach should be monitored, the 'Approach' is a better indicator of cooling tower performance.

iii) **Cooling Tower Effectiveness (%)** - is the ratio of range, to the ideal range, i.e., difference between cooling water inlet temperature and ambient wet bulb temperature, or in other words it is = Range / (Range + Approach).

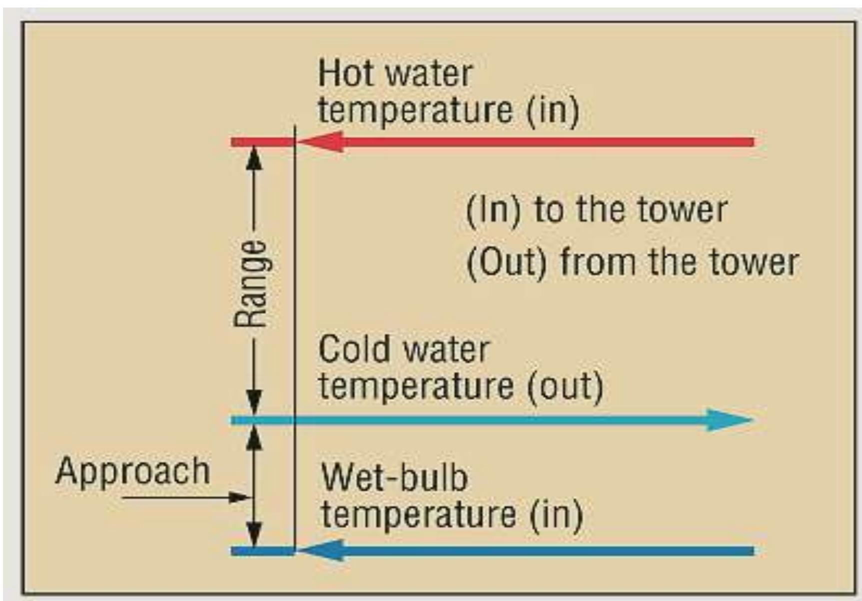
iv) **Cooling Capacity** - is the heat rejected in kCal/hr (Btu/h), given as product of mass flow rate of water, specific heat and temperature difference.

v) **Evaporation Loss** - is the water quantity evaporated for cooling duty and, theoretically, for every **10,000,000 kCal (39,656,668 Btu)** of heat rejected the evaporation quantity works out to **1.8 m³ (63,566 ft³)**.

So, an empirical relation is often used:

Evaporation Loss (m³/h) = 0.00085 x 1.8 x Circulation Rate (m³/h) x (T1-T2).

Where:
T1; T2 = Temperature (°C) difference between inlet and outlet water.



*Source: Perry's Chemical Engineers Handbook

Cycles of Concentration (C.O.C) - is the ratio of dissolved solids in circulating water to the dissolved solids in makeup water.

vii) **Blow Down** – are losses that depend upon cycles of concentration and evaporation. This calculation is given by the following relation:

Blow Down = Evaporation Loss / (C.O.C. – 1)

viii) **Liquid/Gas (L/G) ratio**, of a cooling tower is the ratio between the water and the air mass flow rates. Thermodynamics also dictate that the heat removed from the water must be equal to the heat absorbed by the surrounding air, giving the following equation:

$$L(T_1 - T_2) = G(h_2 - h_1)$$

$$\frac{L}{G} = \frac{h_2 - h_1}{T_1 - T_2}$$

Where:

L/G = liquid to gas mass flow ratio (kg/kg) (lb/lb);
$\mathbf{T_1}$ = hot water temperature (°C) (°F);
$\mathbf{T_2}$ = cold water temperature (°C) (°F);
$\mathbf{h_2}$ = enthalpy of air-water vapor mixture at exhaust wet-bulb temperature (Btu/lb);
$\mathbf{h_1}$ = enthalpy of air-water vapor mixture at inlet wet-bulb temperature (Btu/lb).

7. Factors Affecting Cooling Tower Performance Capacity:

Cooling towers are specified to cool a certain flow rate from **one temperature to another temperature** at a certain **wet bulb temperature**. For **example**, a cooling tower sized to cool **4540 m^3/h (160,328 ft^3/h) through a 13.9°C (57°F) range**, might be larger than a cooling tower to cool **4540 m^3/h (160,328 ft^3/h) through 19.5°C (67°F) range**.

Range - is determined not by the cooling tower, but by the process it is serving. The range at the exchanger is determined entirely by the heat load and the water circulation rate through the exchanger and on to the cooling water.

Range °C (°F) = Heat Load (kcal/h) (Btu/h) / Water Circulation Rate (l/h) (gal/h). Thus, Range is a function of the heat load and the flow circulated through the system.

Cold Water Temp. **32.2°C (90°F) –** Wet Bulb Temp**. (26.7°C) (80°F) = Approach (5.5°C) (10°F)**.

Commonly, the closer the approach to the wet bulb, the more expensive the cooling tower due to increased size. Usually a **2.8°C approach to the wet bulb** is the coldest water temperature that manufacturers will guarantee.

If flow rate, range, approach and wet bulb had to be ranked in the order of their importance in sizing a tower, **approach would be first with flow rate** closely following the range and **wet bulb would be of lesser importance.**

Wet bulb temperature - is an important factor in performance of evaporative water cooling equipment. It is a controlling factor from the aspect of minimum cold water temperature to which water can be cooled by the evaporative method.

7.1. Approach and Cooling Tower Size: The table below illustrates the effect of the **approach on the size and power** difference of a cooling tower. The towers included were sized to cool **4540 m^3/h (160,328 ft^3/h)** through a **16.67°C (62°F)** range at a **26.7°C (80°F)** design wet bulb. The overall width of all towers is **21.65 m (71.0 ft)**; the height, **15.25 m (50.0 ft)**, and the pump head, **10.7 m (35 ft)** approximately.

APPROACH X COOLING TOWER SIZE

Metric: (4540 m3/hr; 16.67°C Range; 26.7°C Wet Bulb; 10.7 m Pump Head)
US Units: (160328 ft³/h; 62°F Range; 80°F Wet Bulb; 35 ft Pump Head)

Approach °C	2.77	3.33	3.88	4.44	5.0	5.55
Hot Water °C	46.11	46.66	47.22	47.77	48.3	48.88
Cold Water °C	29.44	30	30.55	31.11	31.66	32.22
No. of Cells	4	4	3	3	3	3
Length of Cells Mts.	10.98	8.54	10.98	9.76	8.54	8.54
Overall Length Mts.	43.9	34.15	32.93	29.27	25.61	25.61
No. of Fans	4	4	3	3	3	3
Fan Diameter Mts.	7.32	7.32	7.32	7.32	7.32	6.71
Total Fan kW	270	255	240	202.5	183.8	183.8

8. Approach and Flow:

Suppose a cooling tower is installed that is **21.65 m (71 ft)** wide x **36.9 m (121 ft)** long x **15.24 m (50 ft)** high, has **three 7.32 m (24 ft)** diameter fans and each powered by **25 kW (33 hp) motors**. The cooling tower cools from **3632 m3/h (128,263 ft³/h)** water from **46.1°C (115°F) to 29.4°C (85°F)** at **26.7°C (80°F).** Wet Bulb dissipating **60.69 million kCal/hr (240.7 million Btu/h).**

Flow m³/hr	Approach °C	Cold Water °C	Hot Water °C	Million kCal/hr
3632	2.78	29.40	46.11	60.691
4086	3.33	29.95	46.67	68.318
4563	3.89	30.51	47.22	76.25
5039	4.45	31.07	47.78	84.05
5516	5.00	31.62	48.33	92.17
6060.9	5.56	32.18	48.89	101.28
7150.5	6.67	33.29	50.00	119.48
8736	8.33	35.00	51.67	145.63
11590	11.1	37.80	54.45	191.64
13620	13.9	40.56	57.22	226.91
16276	16.7	43.33	60.00	271.32

FLOW X APPROACH FOR A GIVEN TOWER

Metric: (Tower is 21.65 m × 36.9 m x 15.24 m; three Ø7.32 m fans; three 25 kW motors; 16.7°C Range with 26.7°C Wet Bulb)
US Units: (Tower 71 ft wide x 121 ft long x 50 ft high; three Ø24 ft fans; three 33 hp motors; 62°F Range with 80°F Wet Bulb)

Note: The table above shows what would happen with additional flow but with the range remaining constant at **16.67°C**. The heat dissipated varies from **60.69 million kCal/h (240.7 million Btu/h)** to **271.3 million kCal/h (1075.8 million Btu/h).**

9. Function of Fill media in a Cooling Tower:

Heat exchange between air and water is influenced by surface area of tower, time of heat exchange and turbulence in water effecting thoroughness of intermixing. Fill media is to achieve all of above.

Film Fill: In a film fill, water forms a thin film on either side of fill sheets. Thus area of heat exchange is the surface area of the fill sheets, which is in contact with air.

Splash Fill Media: As the name indicates, splash fill media generates the required heat exchange area by splashing action of water over fill media and hence breaking into smaller water droplets. Thus, surface of heat exchange is the surface area of the water droplets, which is in contact with air.

Low-Clog Film**:** have been developed to handle high turbid waters. For sea water, low clog film fills are considered as the best choice in terms of power saving and performance compared to conventional splash type fills.

Typical comparison of **Cross Flow Splash Fill**, **Counter Flow Tower with Film Fill and Splash fill** is shown in the table below:

TYPICAL COMPARISONS BETWEEN VARIOUS FILL MEDIA

	Splash Fill	Film Fill	Low Clog Film Fill
Possible L/G Ratio	1.1 – 1.5	1.5 – 2.0	1.4 – 1.8
Effective Heat Exchange Area	30 – 45 m^2/m^3	150 m^2/m^3	85 – 100 m^2/m^3
Fill Height Required	5 – 10 m	1.2 – 1.5 m	1.5 – 1.8 m
Pumping Head Requirement	9 – 12 m	5 – 8 m	6 – 9 m
Quantity of Air Required	High	Much low	Low

9. Cooling Water Treatment:

Cooling water treatment is mandatory for any cooling tower whether with **splash fill** or with **film type fill** for controlling suspended solids, algae growth, etc. With increasing costs of water, efforts to increase **Cycles of Concentration (COC)**, by Cooling Water Treatment would help to reduce make up water requirements significantly.

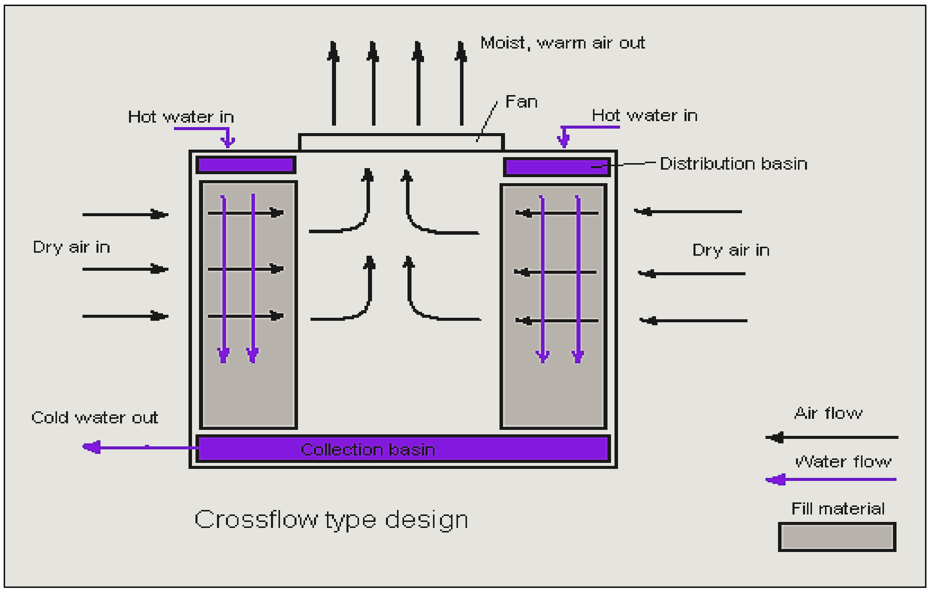
10. Drift Loss in the Cooling Towers:

Most of the end **user specification** calls for **0.02% drift loss**. With technological development and processing of PVC, manufacturers have brought large change in the drift eliminator shapes and the possibility of making efficient designs of drift eliminators that enable end user to **specify the drift loss requirement** to as low as **0.003 – 0.001%.**

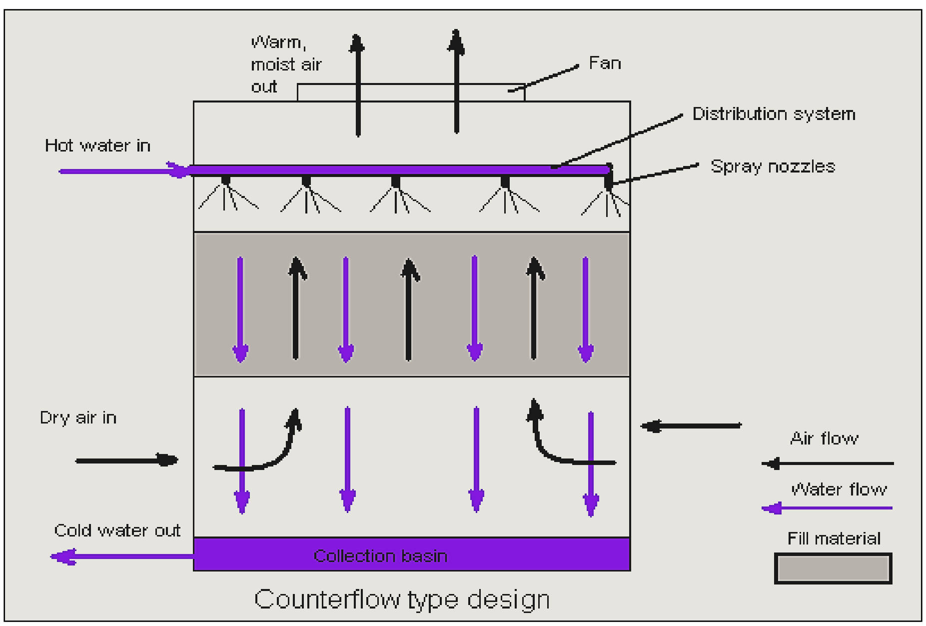
11. Choosing a Cooling Tower:

Counter-Flow and Cross Flow: are two basic designs of cooling towers based on the fundamentals of heat exchange. It is well known that a **counter flow** cooling tower **is more effective** as compared to **cross flow** or parallel flow heat exchange.

a. **Cross-Flow** – these models of cooling towers are provided with **splash fill of concrete**, **wood or perforated PVC**:



b. **Counter-Flow** – these models of cooling towers are provided with **both film fill and splash fill**:



12. Basic Transfer Rate:

Ignoring any negligible amount of sensible heat exchange, the heat gained by the air must equal the heat lost by the water. Within the air stream, the rate of heat gain is identified by the expression:

Q1 = G (h2 – h1) =

Where:

Q1 = Heat transfer - Btu/h;
G = Mass flow of dry air through the tower—lb/min.;
h1 = Enthalpy (total heat content) of entering air—Btu/lb of dry air;
h2 = Enthalpy of leaving air—Btu/lb of dry air.

Within the water stream, the **rate of heat loss** would appear to be:

Q2 = L (t1 – t2) =

Q1 = Heat transfer - Btu/h
L = Mass flow of water entering the tower—lb/min.
t1= Hot water temperature entering the tower—°F.
t2 = Cold water temperature leaving the tower—°F.

13. Heat & Mass Transfer Fundamentals:

The **Merkel theory** 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;
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.

[**S = a x V**, **"a"** means area of transfer surface per unit of tower volume (ft^2/ft^3), and "**V"** means the tower volume (ft^3)]. The heat transfer rate from water side is:

Q = Cw x L x Cooling Range

where:

Cw = specific heat of water = **1.0**
L = water flow rate.

Air Side Transfer Rate - also, the heat transfer rate from air side is:

Q = G x (ha2 - ha1) =

Where:

G = air mass flow rate **- lb/min.**

For the determination of **KaV/L**, **rounding off these values** to the nearest tenth is entirely adequate.

13.1. Heat Balance:

HEAT$_{in}$ = HEAT$_{out}$ + WATER HEAT$_{in}$ + AIR HEAT$_{in}$ = WATER HEAT$_{out}$ + AIR HEAT$_{out}$ + (C_w L_2 t_{w2}) + G h_{a1} =

(C_w L_1 t_{w1}) + G h_{a2} =

Evaporation Loss: 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**). Then, evaporation loss is expressed in:

G x (w_2 -w1) and is equal to L_2 - L_1.

Therefore,

L_1 = L_2 – [G x (w_2 -w_1)] =

Therefore, the enthalpy of exit air is:

h_{a2} = h_{a1} + Cw x [L/G x (t_{w2} - t_{w1})]**,** or,
h_{a2} = h_{a1} + (L/G x Range) **=**

Example 13-1. 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) =**1,66666.67 lb/min.**

OBS.:

1) The weight of **1 gallon of water at 60°F = 8.345238 pounds** and **500 was obtained from 8.345238 x 60** for simplifying the figure.
2) Specific Volume of Air @ 87.8°F, 80% & sea level = **14.32 ft^3/lb**.

Air Flow Rate = **ACFM / Air Specific Volume** = 1,600,000 / 14.32 = **111731.84 lb/min;**

L/G Ratio of Water to Air = Water Flow Rate / Air Flow Rate = 166666.67 / 111731.84 = **1.4916.**

Example 13-2. Calculate the enthalpy and temperature of exit air for the following cooling tower design conditions.

Given,

Ambient Wet Bulb Temperature = **82.4°F;**
Relative Humidity = **80%;**
Site Altitude = sea level L/G Ratio - **1.4928;**
Entering Water Temperature = **107.6°F;**
Leaving Water Temperature = **89.6°F.**

(Solution):

The **enthalpy of exiting** air is calculated from:

$\mathbf{h_{a2}} =$ h_{a1} + (L/G x Range) **=**

$\mathbf{h_{a1}}$ = at 82.4°F WBT & sea level = **46.3624 Btu/lb (enthalpy of inlet air).**

Range = Entering Water Temp. - Leaving Water Temp. = **(t_{w2} - t_{w1}) = 107.6 - 89.6 = 18°F**

Therefore, the **enthalpy of exit air (h_{a2})** is:

$\mathbf{h_{a2}} =$ h_{a1} + (L/G x Range) = 46.3624 + [1.4928 x (107.6 - 89.6)] = **73.2328 Btu/lb**

Note: A temperature corresponding to this value of air enthalpy can be obtained from the table published by Cooling Tower Institute or other psychrometric curve. 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.

14. NTU (Number of Transfer Unit) Calculation:

The equation gives a dimensionless factor as **KaV/L**. 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**.

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

Example 14-1. Determine the **L/G** ratio for the below given conditions, as a function of **NTU** or **KaV/L.**

Given,

Water Circulation Rate = **16,000 GPM;**
Entering Air Flow Rate = **80,848 Lb of dry air / min;**
Ambient Wet Bulb Temperature = **80.0°F;**
Site Altitude = **sea level.**

(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;**

Obs.: 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.).

15. Tower Demand & Tower Characteristic – KaV/L:

15.1. Tower Demand: 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 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. The curves are plotted with the thermal demand, **KaV/L** as a function of the **liquid-to-gas ratio, L/G**. The approach lines (t_{W1} - 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.

15.2. Tower Characteristic - KaV/L: The **KaV/L** is a measure of the rate of evaporative and convective cooling reported as a non-dimensional number and pressure drop combining to create the relative thermal performance of the fill known as the Merkel Equation. The **KaV/L vs. L/G** relationship - is a linear function on log-log demand curve - **KaV/L = C $(L/G)^{-m}$**.

$$KaV/L = \int_{CWT}^{HWT} \frac{dT}{h_W - h_A}$$

Where:

- **KaV/L** = tower characteristic (dimensionless);
- **K** = mass transfer coefficient (lb water/h ft²);
- **a** = contact area/tower volume;
- **V** = active cooling volume/plan area;
- **L** = water rate (lb/h ft²);
- **dT** = bulk water temperature (°F or °C);
- **hw** = enthalpy of air-water vapor mixture at bulk water temperature (J/kg dry air or Btu/lb dry air);
- **ha** = enthalpy of air-water vapor mixture at wet bulb temperature (J/kg dry air or Btu/lb dry air).

Where:

KaV/L = Tower characteristic;
C = Constant related to the cooling tower design, **L/G=1.0;**
m = Exponent related to the cooling tower design (called slope), determined from the test data.

The **tower 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 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 **L/G ratio** is then calculated as follows:

1. Knowing **wet bulb temperature** at the inlet of tower, the **enthalpy increase** of the air stream can be obtained from a psychrometric chart. 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. Considering data from a heat and mass balance the dry air rate and the prevailing **L/G ratio** in the tower can be calculated:

L/G = ha / [Cw x (tw_2 - tw_1)] **= Cw** = specific heat of water = **1.0.**

Obs.: 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.

Example 15-1. A Tower cools **1000 GPM** from **95º F to 85º F at 72º F wet bulb temperature** and operates at **3 cycles of concentration**. Calculate Range, Approach, Heat Rejection, Drift Loss, Evaporation Rate, Bleed Rate and Make-up water requirements.

(Solution):

1. **Range**: (HWT – CWT) = 95 - 85 = **10º F**
2. **Approach**: (CWT – WBT) = 85 - 73 = **13º F**
3. **Heat Rejection**: (Flow GPM X Range ºF X 500) = 1000 x 10 x 500 = **5,000,000 Btu/hr**
4. **Typical Drift Loss**: (0.002% X Flow Rate) = 0.00002 X 1000 = **0.02 GPM**
5. **Evaporation Rate**: (Flow GPM X Range ºF / 1,000) = 1000 X 10 / 1,000 = **10 GPM**
6. **Bleed Rate**: (Evaporation Rate GPM / (Cycles-1) = 10 / (3-1) = **5 GPM**
7. **Make-up Requirements**: (Evaporation Rate GPM X [Cycles/(Cycles-1)]) = 10 x 3/2 = **15 GPM**

Example 15-2. Estimate the **cold & hot water temperature** for a Mechanical Draft Cooling tower when the **water flow rate is increased to 20,000 GPM**. Assume 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)** = 16,000 GPM;
Alternative Water Flow Rate **(L_2)** = 20,000 GPM;

Entering Air Flow Rate **(G_1)** = 80,848 Lb of dry air / min;
Ambient Wet Bulb Temperature = 80.0 °F;
Site Altitude: sea level;
Hot Water Temperature **(HWT, t_{w2})** = 104.0 °F;
Cold Water Temperature **(CWT, t_{w1})** = 89.0 °F;
Characteristic Curve Slope **(m)** = -0.800.

(Solution):

Water Flow Rate, **L_1** = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = **133,333 lb/min.;**
Heat Load, **D_1** = L_1 x 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.**
Range, **R_1** = HWT - CWT = t_{w2} - t_{w1} = 104 - 89 = **15°F;**

Water Flow Rate, **L_2** = Water Flow Rate x (500 / 60) = 20,000 x (500 / 60) = **166,666 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 x (L_1 / L_2) = **12°F.**

Calculate a value of **"C"** of tower characteristic for the design conditions as follows;

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

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

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

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**

From the curve of **80 WBT** and **15°F 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 intersection of **New KaV/L** = **1.2436** and **L/G** = **2.0615** on of **L/G2** = **2.0615** on the approach line, thus determine 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**

Note: Notice that there is a little difference in the values between the computer aid calculation and the CTI graphical methods. This is due to a very little difference in the enthalpy value between the formula used by this and CTI.

Example 15-3. 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 3-1.

(Solution):

First Step: Same as example 3-1.

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

C = KaV/L / (L/G)$^{-m}$ = KaV/L x (L/G)^{m} = 1.4866 x (1.6492)$^{0.7}$ = **2.11001.**

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

New Tower Characteristic = C x (L/G)$^{-m}$ = 2.11001 x (2.0615)$^{-0.7}$ = **1.2716.**

Obs: The **"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.

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

New Cold Water Temperature (New CWT) = Wet Bulb Temperature + New Approach =

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

New CWT = WBT + New Approach = 80 + 10.32 = **90.32°F**
New HWT = CWT + Range = 90.32 + 12 = **102.32°F**

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

Example 15-4. 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):

Water Flow Rate, **L_1** = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = **133,333.3 lb/min.;**
Heat Load, **D_1** = L_1 x 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, **L_2** = Water Flow Rate x (500 / 60) = 20,000 x (500 / 60) = **166,666.7 lb/min.;**
Heat Load, **D_2** = L_2 x 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.**
Range, **R_1** = R_2 = HWT - CWT = tw_2 - tw_1 = 104 - 89 = **15°F;**

Obs.: The value of **NTU** at the design conditions is same as a value calculated in the example 3-1. The value of **"C"** of tower characteristic for the design conditions same as the example 3-1. The new tower characteristic for the increased water flow is also same as the example 3-1. 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 =

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

New CWT = WBT + New Approach = 80 + 12.01 = **92.01°F**
New HWT = CWT + Range = 92.01 + 15 = **107.01°F**

Example 15-5. Assume again the conditions of **example 3-1** and determine the cold and hot water temperature when the heat load is added to increase the **cooling range from 15 to 20°F**, assuming no change in the water circulation rate or in entering air mass flow rate or wet bulb temperature.

(Solution):

Range, $\mathbf{R_1}$ = HWT - CWT = $tw_2 - tw_1$ = 104 - 89 = **15°F;**
Water Flow Rate, $\mathbf{L_1}$ = L_2 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = **133,333.3 lb/min.;**
Air Mass Flow Rate, $\mathbf{G_1}$ = G_2 = **80,848 lb/min.;**
Liquid to Gas Ratio, $\mathbf{L/G_1}$ = L_1 / G_1 = L/G_2 = 133,333.3 / 80,848 = **1.6492;**
Range, $\mathbf{R_2}$ = 20°F;

New Cold Water Temperature = Wet Bulb Temperature + New Approach =

Second 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 15-6. 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):

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

First Step: Calculate a new tower characteristic for the decreased air mass flow. Iterate until the value of new characteristic is equal to the new NTU varying the value of approach:

New Tower Characteristic = $\mathbf{C \times (L/G)^{-m}}$ = $2.21825 \times (2.4737)^{-0.8}$ = **1.0748.**

New Cold Water Temperature = Wet Bulb Temperature + New Approach =

Second 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 15-7. 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 15-1.

(Solution):

Water Flow Rate, 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**
Range, $R_1 = R_2$ = HWT - CWT = $t_{w2} - t_{w1}$ = 104 - 89 = **15°F**

First Step: 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.**

Second 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 =

Third 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**

16. 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. 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? See example below, where the estimation can be discussed:

Example: A **36 x 36 ft** tower cell has **144 nozzles**. Consider that **40 nozzles** are near to the four walls each projecting **10 %** of their water onto those walls: There are 25 internal columns. Each column receives **5%** of the water from **4 adjacent nozzles**:

(Solution):

40 x 10% / 144 = **2.78 %.**

Then, **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. 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 %**

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

Example 16-1: Let's assume that the some by-pass wall water was **4%** and compare the tower demand using the example 15-1.

(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 natural that the cooling range is increased and the tower demand must be based on these new cooling range and cold water temperature.

Water Flow Rate, L_1 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = **133,333.3 lb/min.;**
Heat Load, D_1 = L_1 x R_1 = 133,333.3 x 15 = **2,000,000 Btu/min.;**
Heat Load, D_2 = D_1 = **2,000,000 Btu/min.;**
Range, R_1 = HWT - CWT = t_{w2} - t_{w1} = 104 - 89 = **15°F;**

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.;**
Tower Cold Water Temp., CWT_2 = CWT_1 + R_1 - R_2 = 89 + 15 - 15.625 = **88.375°F;**
Range, R_2 = D_2 / L_2 = 2,000,000 / 128,000 = **15.625°F;**

Obs.: Range can be also calculated, $R_2 = L_1 \times R_1 / \{L_1 \times (1 - \%$ By-Pass Wall Water $/ 100)\} = R1 / (1 - \%$ By-Pass Wall Water $/ 100) = (104 - 89) / (1 - 4 / 100) =$ **15.625°F.**

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 **$D_1 = D_2$**,

$L_1 \times (HWT_1 - CWT_1) = L_2 \times (HWT_1 - CWT_2)$, then, $L_1 / L_2 \times (HWT_1 - CWT_1) = HWT_1 - CWT_2 =$

Therefore:

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

Thus, from the condition that the design hot water temperature must be equal, regardless the By-Pass Wall Water. When the cold water temperature through the cooling tower is being considered will be lower than not to consider the By-Pass Wall Water.

HWT $= CWT_1 + R_1 = CWT_2 + R_2$, then, **$CWT_2$** $= CWT_1 + R_1 - R_2 =$

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 (°F)	hw (But/Lb)	Description	ha (Btu/Lb)	1/(hw-ha)
$tw_1 + 0.1 \times R$	89.94	55.8639	$ha_1 + 0.1 \times L/G \times R$	46.1645	0.1031
$tw_1 + 0.4 \times R$	94.63	62.7545	$ha_1 + 0.4 \times L/G \times R$	53.5858	0.1091
$tw_1 + 0.6 \times R$	97.75	67.8345	$ha_1 + 0.6 \times L/G \times R$	58.5334	0.1075
$tw_1 + 0.9 \times R$	102.44	76.2814	$ha_1 + 0.9 \times L/G \times R$	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

Note: 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 16-2. 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):

Tower Water Flow Rate, **L_1** = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = **133,3333 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, **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_1** = tw_2 - tw_1 = 104 - 89 = **15.0°F;**
Range, **R_2** = (tw_2 - tw_1) / (1 - % By-Pass Water / 100) = (104 - 89) / 0.96 = **15.625°F.**

Calculate a value of **"C"** of tower characteristic and iterate until the value of new characteristic is equal to the new NTU varying the value of approach, for the design conditions as follows:

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

New Tower Characteristic = C x (L/G)$^{-m}$ = $2.21825 \times (1.9790)^{-0.8}$ = **1.2848.**

New Cold Water Temperature = Wet Bulb Temperature + New Approach =

New **CWT** through Tower. Compute the cold water temperature with the result of iteration as follows;

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**: **CWT + Heat Build Up from Heat Exchanger (Range)** = 92.331 + 15.0 = **107.331°F.**
New **HWT**: **CWT through Tower + New Range through Tower** = 91.706 + 15.625 = **107.331°F.**

Obs.: Therefore, **the hot water temperature when to consider the by-pass wall water** is higher than the example 15-3 by **0.321°F**.

17. Pressure Drops in Cooling Towers:

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);

Obs.: Most of air pressure drops at all the areas (excepting fill section) can be calculated by the formula:

Pressure Drops = K x (Air Velocity / 4008.7)^2 x Density Ratio. Notice that **K value** 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°F dry air conditions.

In cooling tower, these pressure losses are called "**Static Pressure Loss**", "**Static Pressure**", or "**System Resistance**". 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°, 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.**

Pressure Drop Calculation: 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** are in the range of **1.6 to 3.0,** commonly 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, but it is in the range of **0.1 to 0.3**.

$$\mathbf{k\ \frac{(p \times v^2)}{2.g} =}$$

Pressure Drop = k x (1 / 2) x Air Density x V^2 / 115,820 (lb/ft^2) =
Pressure Drop = k x 0.1922 x (1/2) x Air Density x V^2 / 115,820 (inch WG = inch Water) =

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/min^2);
Density Ratio = Actual Air Density / 0.075 (1 lb/ft^2 = 0.1922 inch WG).

This calculation can also be:

Pressure Drop = k x 0.1922 x (1/2) x (Density Ratio x 0.075) x V^2 / 115,820 (inch WG = inch Water) =
Pressure Drop = k x 0.1922 x (1/2) x 0.075 / 115,820 x V^2 x Density Ratio (inch WG = inch Water) =
Pressure Drop = k x V^2 x 1 / 16,069,371 x Density Ratio =

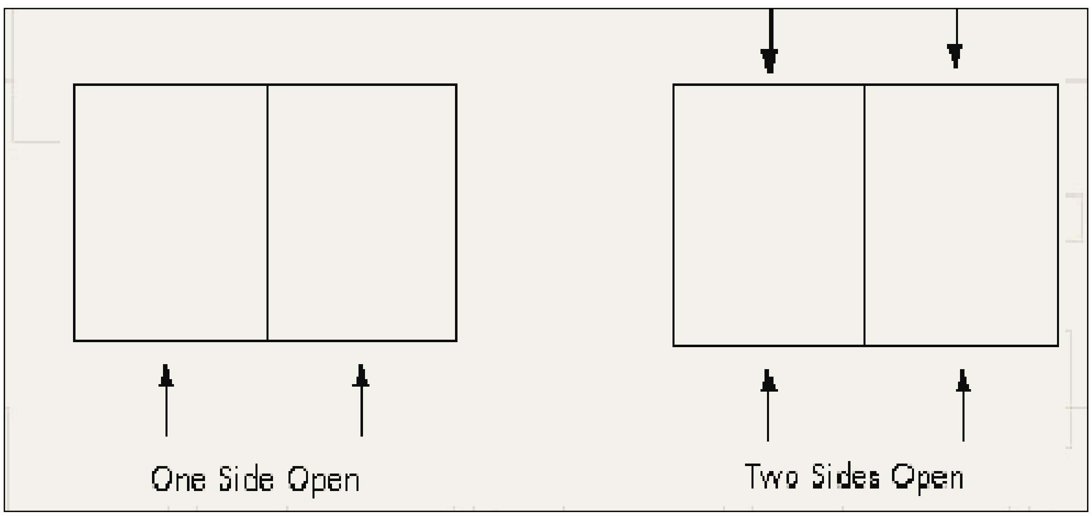
Pressure Drop = k x V² x 1 / 4008.72 x Density Ratio =
Pressure Drop = k x (V / 4008.7)² x Density Ratio =

Obs.: 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/ft³ unit of air density**.

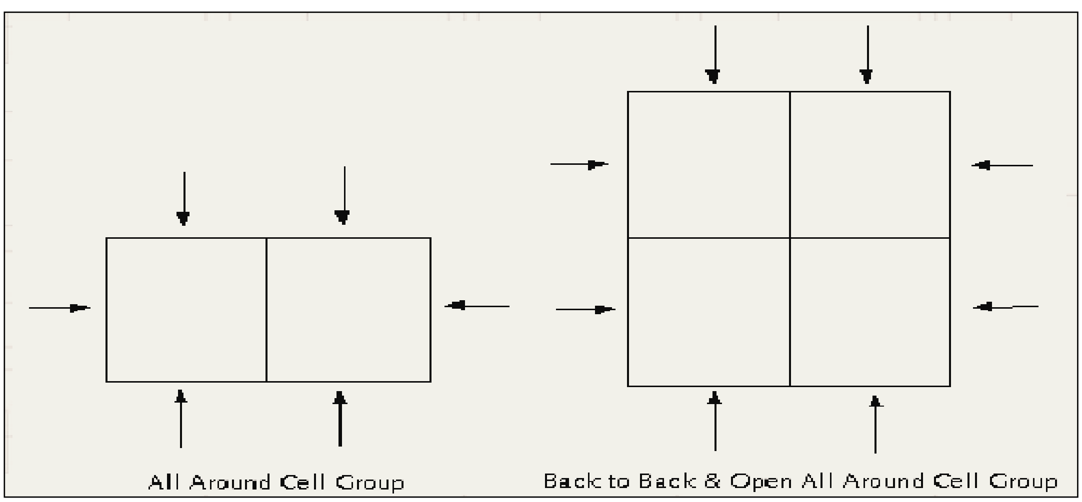
18. Air Flow Arrangements:

1) **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.

2) **Two Sides Open & Ends Closed**: This arrangement is most general for the industrial cooling towers.



1) **All Around Cell Group Back To Back & Open All Round**: This is useful for a case where the area is limited.



Example 18-1. 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 = 12,500 GPM;
Exit (Entering) Water Temperature = 89°F;
Inlet (Leaving) Water Temperature = 104°F;
Fill Depth = 4 feet;
Fill Flute Size = 19 mm;
Entering Wet Bulb Temperature = 80°F;
Relative Humidity = 80.0%;
Site Elevation = 0 feet;
Exit Air Temperature: 97°F.

Arrangement of Air Inlet: Two Sides Open & Ends Closed;
Material of Tower Framework: Wood;
Type of Air Inlet Louver: Large, Widely Spaced.

(Solution):

New Tower Range = Design Range / (1 - % by pass wall water / 100) = 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.

% 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 or 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:
{(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%.**

New Tower Range = 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 **h_{a2} = h_{a1} + L/G x New Tower Range**.

L/G = (h_{a2} - h_{a1}) / New Tower Range:

Air Enthalpy at Exit (97°F) = **66.5773 Btu/lb;**
Air Enthalpy at Inlet (80°F) = **43.6907 Btu/lb.**

Therefore:

L/G = (66.5773 - 43.6907) / 15.5063 = **1.4760.**

First Step: The air mass is calculated from the relation: **G = L / (L/G)**. Here the value of L is a net water flow rate through the cooling tower.

L = Design Water Flow Rate x (500 / 60) x (1 - % By-Pass Water / 100) =
L = 12,500 x (500 / 60) x (1 - 3.265 / 100).

Obs.: (500 / 60) is a constant to convert water flow rate in GPM to lb/min unit.). Then, the value of air mass flow:

Flow Rate = 12,500 x (500 / 60) x (1 - 3.265 / 100) / 1.4760 = **68,271.5 lb/min.**

Second Step: Calculate the area of obstruction in the air inlet. In case of wood structure, **one bay (between centers of columns)** is based on **6 feet** and the traversal member is based on **6 feet** height.

Therefore, the number of bay for the **42 feet of cell length is 7** and the **width of column is 4 inch**. For transversal member, two beams are required for this air inlet height.

Area of Obstruction due to Columns: No. of Bays x Width of Column x Air Inlet Height x No. of Air Inlet = 7 x (4 / 12) x 15 x 2= **70 ft²**.

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 ft²**.

Total Area of Obstructions = 70 + 56 = **126 ft²**.

Overall Area of Air Inlet: Cell Length x Air Inlet Height x No. of Air Inlet = 42 x 15 x 2 = **1,260 ft²**.

% Obstruction = 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 ft²**.

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.

The **dry bulb temperature** corresponding **80% RH at 80°F WBT**, is = **85.24°F**.

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 = 68,2715 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:

Air Velocity = Airflow Volume @ Air Inlet / Net Area of Air Inlet =
Air Velocity = 971,028 / 1,134 = **856.29 ft/min (FPM)**

Air Density = @ **85.24 DBT** & **80% RH** = **0.0718 lb/ft^3**.

The Pressure Drop Coefficient for this "narrow, small louvers", is = **2.5.** Then, the pressure drop is:

Pressure Drop = **K (V / 4008.7)² x Density Ratio** =
Pressure Drop = 2.5 x (856.29 / 4008.7)² x (0.0718 / 0.0750) = **0.1092 (inch Water).**

Obs.: 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.

Example 18-2. Determine the pressure drop at the fill for the same example 18-1.

(Solution):

First Step: 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.

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

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

Average Specific Volume = 2 / (1 / Specific Volume @ Tower Inlet Temp. + 1 / Specific Volume @ Tower Exit Air Temp.) = **14.5709 ft³/lb.**

Average Air Volume = Average Specific Volume x Air Mass Flow = **994,776.8 ft³/min.**

Average Air Velocity = Average Air Volume / Net Fill Area Net Fill Area =
Average Air Velocity = (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 = ~3.6 = **1.11%.**

Net Fill Area = (42 x 42) x (1 - 1.11 / 100) = **1,730.7 ft².**

Average Air Velocity @ Fill = Average Air Volume @Fill / Net Fill Area= **574.78 ft/min.**

Third Step: The water loading calculation is required as follows:

Water Flow = 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/ft².**

Air Density @ 85.24 DBT & 80% RH = **0.0718 lb/ft³;**
Air Density @ 97.0 DBT & 100% RH = **0.0696 lb/ft³.**

Average Air Density (at fill) = **0.0707 lb/ft³.**

Pressure Drop Parameters: 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 is:

Pressure Drop @ Fill = **0.3011 (inch WG)**.

Example 18-3. Determine the pressure drop at the drift eliminator per given conditions in example 18-1.

(Solution):

Net Fill Area = **1730.7 ft².**
Air Mass Flow = **68,2715 lb/min**.
Specific Volume @ 97.0 DBT & 100% RH = **14.9362 ft³/lb.**

Drift Eliminator (obtained from Specific Volume x Air Mass Flow) = **1,019,7163 ft³/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**

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

Example 18-4. Determine the pressure drop at the fan inlet of fan stack per the given conditions in example 18-1. 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. 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.

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

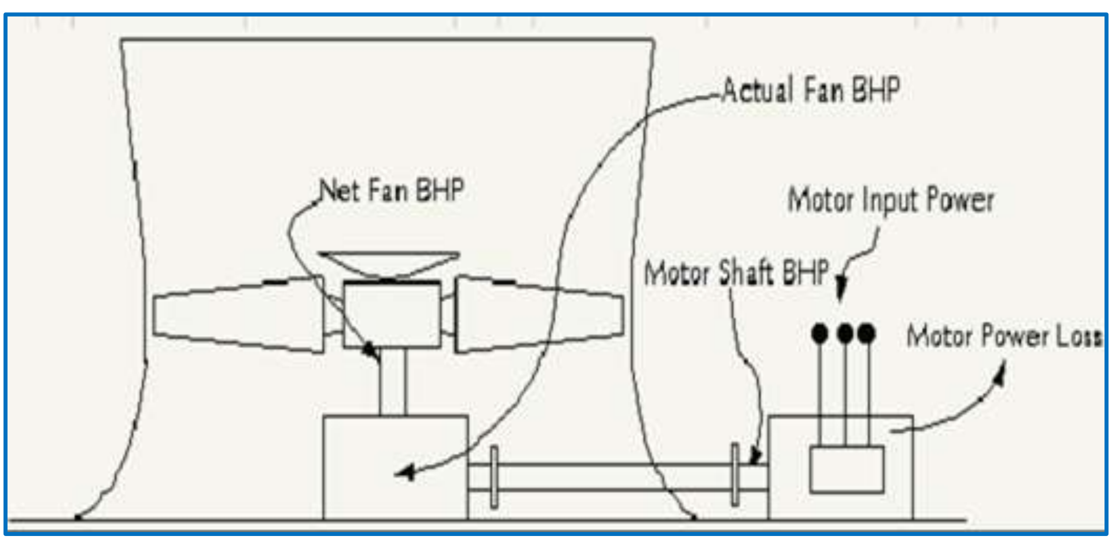
Calculate the net fan area:

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 = 1,019716.3 / 573.52 = **1778.00 ft/min.**

Pressure drop = **K (V / 4008.7)^2 x Density Ratio =**
Pressure drop = 0.18 x (1778.0 / 4008.7)^2 x (0.0696 / 0.0750) = **0.0329 (inch Water).**

19. 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 for the fan performance as follows:

1) Influence of Fan Inlet Shapes = **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. When smaller is the ratio of distance and the larger is the ratio of area, the higher is the resistance correction factor. The additional pressure drop coefficient due to the obstacles is within **0.1 to 0.15**.

3) Influence of Tip Clearance: The tip clearance less than **1% to the fan diameter** does not effect to the fan performance.

Note: 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. Thus, the suggestion is to use the following guidelines:

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

Example 19-1. Determine the fan brake horsepower and fan static efficiency for the design conditions, with 1,019716.28 ft³/min of air volume, 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 (inch Water)**.

Note: 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.

Total Pressure:
Total Static Pressure + Velocity Pressure - Velocity Recovery =
0.4793 + 0.1825 - 0.0178 = **0.6439 (inch Water)**.

Fan BHP = 1,019716.28 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) =
1,019716.28 x (0.4793 - 0.0178) / (128.98 x 6356) = **57.4%**

Example 19-2. Determine the motor input power based on the example 19-1.

(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**

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

Example 19-3. 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**

Obs.: 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.

Note: The gear reducer wastes **3 to 5%** of motor power, which depends on the number of reduction. These losses shall be turned to the heat due lubricant oil and a proper cooling of lubricant oil is required. 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.

20. Evaporation:

1. Cooling towers **evaporate about 2 gallons** of water every hour for each ton of refrigeration. A gallon of water weighs **8.3 pounds, and about 1, 000 Btu** is needed to **evaporate 1 pound of water**. Thus, to evaporate a gallon of water, **8.3 x 1,000 or 8,300 Btu** is required.

2. 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.

3. 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.

4. 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 20-1. 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%**

Obs.: 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:

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

Air Enthalpy at Exit (97°F) = **66.5773 Btu/lb**
Air Enthalpy at Inlet (80°F) = **43.6907 Btu/lb**

Then, L_2/G = {(66.5773 - 43.6907) - (89 - 32) x (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%**

Example 20-2. Determine the heat removal in the percentage by the evaporation for the example 7-1. Assume Latent Heat of Water, about **1,040 Btu/lb** of Water**.**

(Solution):

For **each pound of water** that a cooling tower evaporates, it removes somewhere near **1,040 Btu from water.** This heat removal is the result of the evaporation of water into air stream during the direct contact cooling process.

Evaporation Loss Rate:
($w_2 - w_1$) x Latent Heat of Water / (Enthalpy @ Exit - Enthalpy @ Inlet) =

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

21. Estimation of Actual Cold Water Temperature:

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

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 in GPM through Tower x (500/60) / (Air Volume @ Fan / Specific Volume @ Fan) =

Example 20.3. Determine L/G ratio and cold water temperature when the wet bulb temperature was downed to 70◦F from design conditions described in the example 21-1.

(Solution):

First, find a dry bulb temperature for **80% of relative humidity corresponding 70◦F 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 (lb/min) = Air Volume @ Fan / Specific Volume @ Fan =

L/G ratio is obtained from the relation:

Water Through Tower (lb/min) / Air Mass (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 =
Net Fan Power = 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 =
KaV/L 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) =
Total Kav/L 1.3890 / (1 - 9.9% / 100) = **1.5416.**

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

Actual Cold Water Temperature = Wet Bulb Temperature + Approach = 70 + 11.891 = **81.89 °F.**

21. Determination of L/G:

As seen in the equations of **NTU or Tower Demand**, the right side of formula is obviously a **dimensionless factor.** 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 21.1. Determine the **L/G ratio** under the assumption that the water flow rate was increased to **13,750 GPM.** The wet bulb temperature remains unchanged from conditions given in the example 21-1.

(Solution):

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

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

Second Step: Find an exit air temperature and air volume of fan. The procedure is exactly same as the contents described in the example 21-1, obtained from this relationships:

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) =

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

Water Flow Rate (GPM) through Tower = Design Water Flow Rate x (1 - % By-Pass Water) =
Water Flow Rate (lb/min) = Water Flow Rate in gpm through Tower x (500/60) =
Air Mass Flow Rate = Air Volume @ Fan / Specific Volume @ Fan =

Water Flow Rate (lb/min) = 13,300.4 x (500/60) = **110,836.7;**
Air Mass Flow Rate (lb/min) = 1,039,249.8 / 15.000 = **69,283.3.**

L/G Ratio =Water Flow Rate in Lb/Min / Air Mass Flow Rate (lb/min);
L/G = 110,836.7 / 69,283.3 = **1.59976.**

22. Determination of the Pumping Head:

The below example will show how to compute the pumping head.

Example 22.1: Determine the pumping head considering the **static lift (D) is 35 feet** and vertical **distance (h)** of the pressure gauge above the basin curb **is 5 feet**. The pressure gauge indicates **25 psig**. The **pipe 24 inch inner diameter is 22.624 inch**. Assume that the test water flow rate is **14,000 GPM**.

(Solution):

1) 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", 90◦ Welding Elbow (r/d = 1);

Equivalent Length = **37.7 ft** (table of friction loss in term of length).
Total Equivalent Length = 30 + 5 + 37.7 = **72.7 ft** (based on 24 inch pipe).

2) 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 for 14,000 GPM** of 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.**

3) Determine the **static pressure** of test water flow at the center of inlet pipe.

SPt = Test Pressure - (D - h) - Friction Loss =
SPt = 25 psig x 2.31 - (35 - 5) - 0.95 = **26.78 ft -** (1 psi = 2.31 feet).

4) Determine the **velocity pressure** of test water flow at the entering of inlet pipe.

Water Velocity @ 24 inch pipe = GPM x 0.1336798 / [0.7854 x (Inner Diameter / 12)^2] =
Water Velocity = 14,000 x 0.1336798 / [0.7854 x (22.624 / 12)^2] = 670.39 ft/min = **11.17 ft/s.**

Velocity Pressure = Velocity^2 / 2g = 11.17^2 / (2 x 32.174) (1g = 32.174 ft/sec^2) = **1.94 ft.**

5) Compute the **test pumping head**:

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

6) 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.**

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

References and Related Links:

1. ASHRAE: at: www.ashrae.org.
2. The Tubular Exchanger Manufacturers Association at: www.tema.org.
3. OSHA Technical Manual at: www.osha.gov.
4. Cooling Tower Thermal Design Manual at: www.daeilaqua.com.
5. Tower Design Free Online eBook Collection at: www.pdftop.com/ebook/tower+design.

