PDHonline Course M425 (5 PDH)

HVAC & Cooling Towers– Practical Calculations

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HVAC – Practical Calculations:

HVAC (pronounced either "H-V-A-C" or, occasionally, "*H-vak*") is an acronym for “Heating, Ventilation and Air Conditioning”. HVAC is referred to climate control as a process of treating air to control its temperature, humidity, cleanliness and distribution to meet the requirements of the conditioned space.

The **HVAC** industry had been historically regulated by standards organizations such as **ASHRAE, SMACNA, ARI, ACCA**, Uniform Mechanical Code, International Building Code, and AMCA established to support the industry and encourage high standards and achievements.

The term ventilation is applied to processes that supply air to or remove air from a space by natural or mechanical means. Such air may or may not be conditioned. An air conditioning system has to handle a large variety of energy inputs and outputs in and out of the building where it is used.

The basic purpose of an **HVAC system** is to provide interior thermal conditions that a majority of occupants will find acceptable. Occasionally this may simply require that air be moved at an adequate velocity. However, occupant comfort will require that an **HVAC system** add or remove heat to or from building spaces.

1. HEAT AND TEMPERATURE:

• **Heat**: Heat may be defined as energy in transit from a high-temperature object to a lower-temperature object. This heat transfer may occur by the mechanisms of conduction, convection and radiation.

• **Sensible heat**: Kind of heat that increases the temperature of air. It is an expression of the molecular excitation of a given mass of solid, liquid, or gas.

• **Latent heat**: Heat that is present in increased moisture of air. It changes the matter from solid to liquid or from liquid to gas. Heat that is required to change solid to liquid is called latent heat of fusion, and that which is required to change liquid to gas is called latent heat of vaporization.

• **Enthalpy**: Sum of sensible and latent heat of a substance e.g. the air in our environment is actually a mixture air and water vapor. If the enthalpy of air is known, and the enthalpy of desired comfort condition is also known, the difference between them is the enthalpy that must be added (by heating or humidification) or removed (by cooling or dehumidification).

• **Temperature**: A measure of the degree of heat intensity. The temperature difference between two points indicates a potential for heat to move from the warmer point to the colder point. Unit in English system is Fahrenheit, and in International System is Celsius.

• **Dry-bulb temperature (DB)**: The dry-bulb temperature is the temperature of air measured by a thermometer freely exposed to the air but shielded from radiation and moisture.

More specifically, it is a measure of the intensity of kinetic energy of the molecules in the air. It is one of "the most important climate variables for human comfort and building energy efficiency”.
• **Wet-bulb temperature (WB)**: The temperature registered by thermometer whose bulb is covered by a wetted wick and exposed to a current of rapidly moving air. It is the temperature air would have if part of its energy were used to evaporate the amount of water it would absorb to become fully saturated.

• **Dew point temperature**: The temperature at which condensation begins when the air is cooled.

• **Relative humidity (RH)** = (actual vapor pressure of air-vapor mixture/pressure of water vapor when the air is completely saturated at the same DB temperature) x 100.

• **Vapor pressure** is the pressure exerted by the motion of molecules of water vapor. It is dependent on the amount of water vapor in the air and the temperature of the air.

2. THERMODYNAMICS BASIC CONCEPTS:

The **biggest problem** in **thermodynamics** is the student to learn and recognize **heat, work, force, energy, power** and other technical terms. So to facilitate the basic comprehension of the terms it is very important to remember some concepts below:

Cal - The **"Cal"** is the standard unit of measurement for heat. The gram calorie, small calorie or calorie (cal) is the amount of energy required to raise the temperature of one gram of water from **19.5 °C** to **20.5 °C** under standard atmospheric pressure of **1.033 Kg/cm² (14.7 psi)**.

Btu - **British Thermal Unit**. The **"Btu"** is the standard unit of measurement for heat. A Btu is defined as the amount of energy needed to raise the temperature of one pound of water from **58.5°F to 59.5°F** under standard pressure of **30 inches of mercury (14.7 psi)**.

- **Energy Unit Conversions:**

Unit	Multiply	To obtain
1 Btu	0.252	kcal
	107.7	Kgf.m
	778.7	ft.lbf
1Cal	0.00396	Btu
	0.00000116	kW.h
1 kcal	1000	cal
	3.9604	Btu

Watt – is the metric unit for power.

Unit	Multiply	To Obtain
1 W	0.001	kW
	0.00134	hp
	0.0002387	kcal/s
	44.2	ft.lbf/min
	0.000948	Btu/s
	0.000284	ton (refrig)

Unit	Multiply	To Obtain
1 Btu/s	0.3002	ton (refrig)
	1.055	kW
	1.435	hp
	106.6	kgf.m/s
	778.8	ft.lbf/s
1 joule/kilogram/K = J/(kg.K) = 1 joule/kilogram/°C = J/(kg.°C) =	0.001	kilojoule/kilogram/°C = kJ/(kg.°C)
	0.000239	kilocalorie /kilogram/°C = kcal/(kg.°C)
	0.000239	Btu/pound/°F = Btu/(lb.°F)
	0.000423	Btu/pound/°C = Btu/(lb.°C)
1 Btu/pound/°F = Btu/(lb°F)	1.8	Btu/pound/°C = Btu/(lb.°C)
	4186.8	joule/kilogram/°C = J/(kg.°C)
	4.1868	kilojoule/kilogram/°C = kJ/(kg.°C)
	778.2	pound-force.foot/pound/°R

Obs.:

1 Watt-Hour = 0.000948 (Btu/s) x 60 x 60 = **3.412 Btu/h**
1 Btu/h = 0.293 Watt = **0.000293 kW**

Celsius (also known as **centigrade**) is a temperature scale that is named after the Swedish astronomer Anders Celsius (1701–1744), who developed a similar temperature scale two years before his death. Then nominally, **0 °C** was defined as the freezing point of water and **100 °C** was defined as the boiling point of water, both at a pressure of one standard atmosphere **(1.033 Kg/cm²)**.

Fahrenheit is the temperature scale proposed in 1724 by, and named after, the physicist Daniel Gabriel Fahrenheit (1686–1736). On the Fahrenheit scale, the freezing point of water was **32 degrees** Fahrenheit (°F) and the boiling point **212 °F** at standard atmospheric pressure **(14.7 psi).**

Pressure (P) is the force per unit area applied in a direction perpendicular to the surface of an object. Gauge pressure is the pressure relative to the local atmospheric or ambient pressure.

Unit	**Pascal** (Pa)	**bar** (bar)	**atmosphere** (atm)	**Torr** (Torr)	**pound-force per square inch** (psi)
1 Pa	1 **N**/m²	0.00001	0.000009867	0.0075006	0.000145
1 bar	100000	106 dyn/cm²	0.9867	750	14.5
1 at	98066	0.980665	0.968	735.5	14.223
1 atm	101325	1.01325	**1 atm**	760	14.7
1 torr	133.322	0.013332	0.0013158	**1 mmHg**	0.0193
1 psi	0.006894	0.068948	0.068046	51.72	**1 lbf/in²**

3. TONS OF REFRIGERATION:

For commercial and industrial refrigeration systems most of the world uses the **kilowatt (kW)** as the **basic unit** refrigeration. Typically, commercial and industrial refrigeration systems are rated in **Tons of Refrigeration (TR)**.

One Ton of Refrigeration was defined as the energy removal rate that will freeze **1 ton of water at 0 °C (32 °F)** in one day, or, the amount of heat required to melt **1 ton of ice in 24 hours**. The unit's value is approximately **11,958 Btu/h (3.505 kW)**, redefined to be exactly:

1 Ton of Refrigeration = 12,000 Btu/h (3.517 kW) = 3,024 Kcal/h

4. METABOLIC RATE:

Metabolic rate is measured in **Met** units. A **Met** is the average amount of heat produced by a sedentary person (e.g. office work = **1 Met**).

1 Met unit corresponds approximately: **360 Btu/h = 90,718 cal/h = 90.718 kcal/h**.

Human beings are essentially constant-temperature creatures with a normal internal **body temperature of 98.6°F**. Heat is produced in the body as result of metabolic activity.

If the internal temperature rises or falls beyond its normal range, mental and physical operation is jeopardized or affected, and if the temperature deviation is extreme, then serious physiological disorders or even death can result.

5. COMFORT ZONE:

The comfort range of temperature during **summer varies between 70 to 76°F** dry bulb temperatures and **45 - 65% relative humidity**.

The comfort range during cold **winters** would be in the range of **65 to 68°F** dry bulb temperature and **relative humidity** of a minimum of **30%**.

6. HEAT TRANSFER:

Conduction: is the spontaneous transfer of thermal energy through matter, from higher temperature to lower temperature, and acts to equalize temperature differences. It is also described as heat energy transferred from one material to another by direct contact.

Convection: is usually the dominant form of heat transfer in liquids and gases. Convection is circulation of a fluid or gas/air caused by temperature difference. Commonly an increase in temperature produces a reduction in density.

Evaporation: It is exclusively a cooling mechanism. Evaporative losses become a predominant factor when ambient temperatures are very high. When surrounding temperature is about **70°F**, most people lose sensible heat.

Radiation: is the only form of heat transfer that can occur in the absence of any form of medium; it means heat transfers through a vacuum. Thermal radiation is a direct result of the movements of atoms and molecules in a material.

7. DRY BULB TEMPERATURE:

Thermal Comfort zone: It is an area plotted on the psychrometric chart. During **summer** the comfort range of temperature varies between **70 to 76°F** dry bulb temperatures, that is, **45 - 65% RH**.

During **cold winters** the comfort condition would be in the range of **65 to 68°F** dry bulb temperature and relative humidity of a minimum of **30%**.

Humidity: Density of water vapor per unit volume of air is expressed in units of **lbs. of water/ft^3 of dry air**. Specific humidity = weight of water vapor per unit weight of dry air, expressed in **grains/lb**.

Relative Humidity (RH): Human tolerance to humidity variations: in winter is from **20 to 50%**; in summer, the range extends up to **60% @ 75ºF**. High humidity causes condensation problems and reduces body heat loss by evaporative cooling.

8. 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"**.

Thermodynamic properties of moist air are affected by atmospheric pressure. The standard **temperature is 59°F** and standard **atmospheric pressure is 29.921 in-Hg (14.697 psi)** at sea level.

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 **water vapor is 1545.32/18.01528 = 85.778 ft-lb/lb°R**. The temperature and barometric pressure of atmospheric air vary with altitude as well as with local geographic and weather conditions.

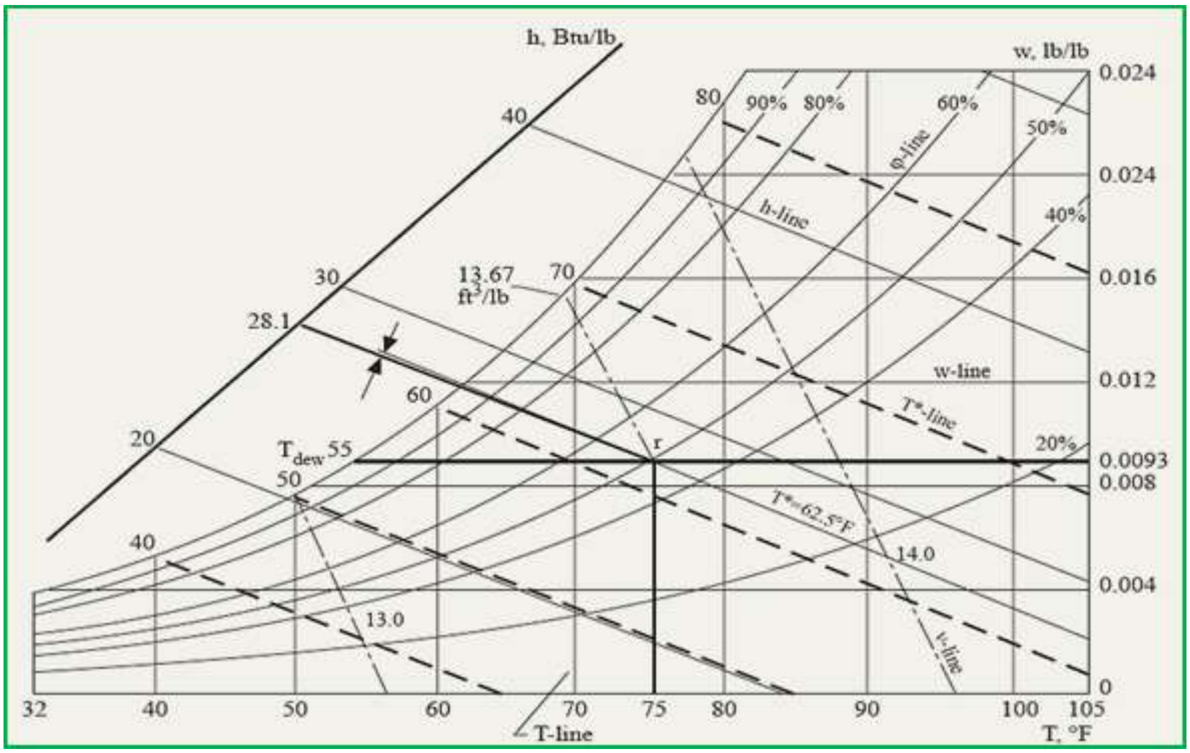
Gravity is also assumed constant at the standard value, **32.1740 ft/s^2**.

a) Psychrometric chart:

A **psychrometric chart** is a graphical presentation of the **thermodynamic properties** of moist air and various air-conditioning processes and air-conditioning cycles.

The most **common chart** is the **temperature - humidity ratio (w) chart**. The **Dry Bulb Temperature (DB)** appears horizontally as the abscissa and the **humidity ratios (w)** appear as the vertical axis.

Abridged sample of **psychrometric chart** is shown below:



b) How to Read a Psychrometric Chart:

• **Dry bulb** temperature **(DB)**: Verticals lines designate **DB**. A standard psychrometric chart for air conditioning applications has a temperature range of **32 to 120°F**.

• **Wet bulb** temperature **(WB):** Diagonals lines rising upward from left to right having negative slope slightly greater than that of the h-lines. A wet bulb line meets the DB line of the same magnitude on the saturation curve representing **100% RH**.

• **Dew point** temperature **(DP):** Follow the horizontal line from the point where the line from the horizontal axis arrives at **100% RH**, also known as the *saturation curve*. On a saturation curve the **dry bulb, wet bulb and dew point** temperature **(DP)** have the same value.

• **Relative humidity (RH%):** Curved lines that radiate from lower left to upper right are **RH** lines. Horizontal line at the bottom represents **0% RH**; the uppermost curved line is **100% RH** line.

• **Humidity ratio:** Humidity ratio or specific humidity w-lines are horizontal lines on the Y-axis, they range **from 0 to 0.28 lb/lb**. **Humidity** ratio **is dimensionless**, sometimes expressed **as grams of water per kilogram** of dry air or **grains of water/lb** of air.

• Specific enthalpy (h): Enthalpy lines incline downward to the right-hand side (negative slope) at an **angle of 23.5°** to the horizontal line and have a range of **12 to 54 Btu/lb** expressed in **Btu/lb of air**.

• **Specific volume (v):** Specific volume lines are represented by the diagonal lines close to 90º. The **moist volume** ranges from **12.5 - 15 ft³/lb**, also called **Inverse Density**, is the volume per unit mass of the air sample.

Example:

An air-conditioned room at sea level has an indoor design temperature of **75°F** and a relative humidity of **50%**. Determine the **humidity ratio, enthalpy, density, dew point, and thermodynamic wet bulb temperature** of the indoor air at design condition.

Solution:

Find the room temperature 75°F on the horizontal temperature scale. **Draw a line parallel to the 75°F** temperature line and establish the point where it meets the relative humidity curve of **50% at point (r).** This point denotes the **state point** of room air.

• **Draw a horizontal line** toward the humidity ratio scale from point (r). This line meets the ordinate and thus determines the room air humidity ratio **w = 0.0093 lb/lb**.

• **Draw a line** from point (r) parallel to the enthalpy line. This line determines the enthalpy of room air on the enthalpy scale, **h = 28.1 Btu/lb**.

• **Draw a line** through point (r) parallel to the moist volume line. The perpendicular scale of this line indicates **v = 13.67 ft³/lb**.

• **Draw a horizontal line** to the left from point (r). This line meets the saturation curve and determines the dew point temperature, **Tdew = 55°F**.

• **Draw a line** through point (r) parallel to the wet bulb line. The perpendicular scale to this line indicates that the thermodynamic wet bulb temperature **(WB) = 62.5°F**.

c) 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).

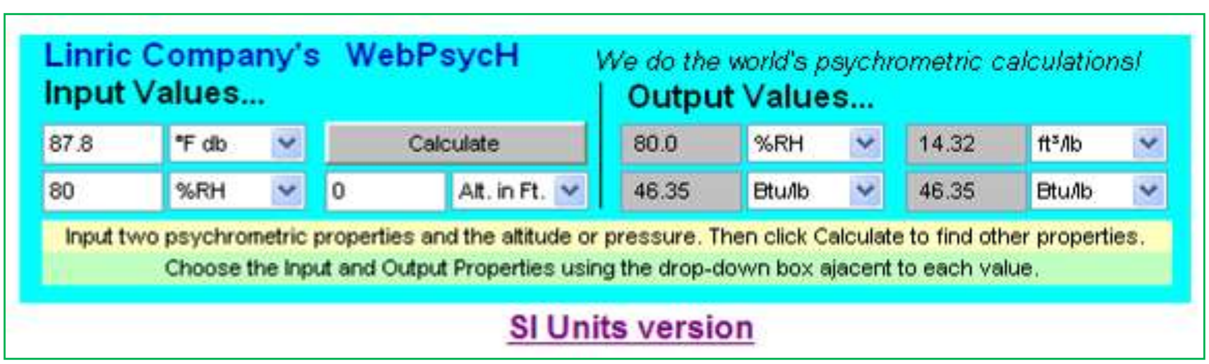
- Specific enthalpy symbolized by *h*, also called heat content per unit mass, is the sum of the internal (heat) energy of the moist air in question, including the heat of the air and water vapor within.

- 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.

d) 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**

9. BASIC AIR CONDITIONING CALCULATIONS:

The air conditioning loads are the **Sensible Heat Loads + Latent Heat Loads** as explained below:

a) **Sensible heat loads -** Sensible heat gain is directly added to the conditioned space by conduction, convection, and/or radiation. Sensible heat load is a total of:

   a. Heat transmitted thru floors, ceilings, walls
   b. Occupant's body heat
   c. Appliance & Light heat
   d. Solar Heat gain thru glass
   e. Infiltration of outside air
   f. Air introduced by Ventilation

b) **Latent Heat Loads:** Latent heat gain occurs when moisture is added to the space from internal sources or from outdoor air as a result of infiltration or ventilation to maintain proper indoor air quality. Latent heat load is a total of:

   a. Moisture-laden outside air form Infiltration & Ventilation
   b. Occupant Respiration & Activities
   c. Moisture from Equipment & Appliances

c) **Humidity ratio** or **specific humidity:** is dimensionless, sometimes expressed **as grams of water per kilogram** of dry air or **grains of water/lb** of air.

1) Sensible Heat Loads:

A sensible heating process adds heat to the moist air in order to increase its temperature. The rate of heat transfer from the hot water to the colder moist air is often called **sensible heating**, in **Btu/h (kW)**:

Qsensible **= m x Cp (To – Ti) [Btu/h]**

Where:

Q sensible = Sensible heat, **(kW) (Btu/h)**
m = Mass flow rate of air, **(Kg/h) (lb/h)**
Cp = Specific heat capacity of air (see tables), **0.24 Btu/lb°F (1.01 Kcal/Kg°C)**
ρair = Density of supply air, **(Kg/m³) (Lb/ft³)**
To, Ti = moist air temperature at final/initial states and the mass flow rate of supply air, (C°) (°F).

To maintain temperature requirements, the air inside a building keeps circulation through a cooling coil, empirically determined by:

Qsensible **= 1.08 x CFM (To – Ti) [Btu/h]**

Where:

CFM= Air circulation flow, **CFM**
Ti = Inside air temperature, **ºF**
To = Outside air temperature, **ºF**

The sensible heat loss from infiltration can also be calculated as:

Qsensible **= (60 x CFM) x (ρair x *C_p*) x (Ti – To) [Btu/h]**

Where:

Q = Heat loss in **(Btu/h)**
CFM = Volumetric air flow rate **(CFM)** - [CFH = Air flow in ft³/h **(CFM x 60)]**
ρair = Density of the air **(lb/ft³)** – [Air density = **0.075 lb/ft³]**
Cp = Specific heat capacity of air (see tables), **0.24 Btu/lb°F (1.01 Kcal/Kg°C)**
Ti = Inside air temperature **(°F)**
To = Outside air temperature **(°F)**

2) Latent Heat Loads:

The most commonly used method of removing **water vapor** from air (dehumidification) is to cool the air below its dew point. The dew point of air is when it is fully saturated, at **100%** saturation. The **latent heat** to be removed is:

Qlatent **= m x hfg x (Wo - Wi) [Btu/h]**

Where:

• Q = Cooling energy **(kW) (Btu/h**),
• m = Mass flow rate of air **(Kg/h) (Lb/h),**
• hfg = Latent heat of vaporization of water (see tables), **1,060 Btu/lb (589 Kcal/Kg)**,
• Wo, Wi = Moisture content of air **(Kg/Kg) (lb/lb-dry air).**

The amount of moisture added to the air must be removed through the cooling coil to maintain the humidity requirements, determined by:

Qlatent **= 4,840 x CFM (Wo – Wi)**

Where:

CFM= Air circulation flow, **CFM**
Wo = Outside moisture content of air (**lb/lb-dry air)**
Wi = Inside moisture content of air **(lb/lb-dry air).**

3) Humidity:

In a **humidifying** process, water vapor is added to moist air and increases the **humidity ratio** entering the humidifier if the moist air is not saturated. The humidifying capacity is given by:

$$H_m = CFM \times \rho \times (Ho - Hi)\ [lb/min]$$

Where:

CFM= Volume flow rate of supply air, **(CFM)**
ρ = Density of supply air, **(lb/ft³)**
Ho, Hi = Moisture content of air at final and initial states, **(Btu/lb of dry air)**.

4) Total heat loads:

$$Q_{total} = 4.5 \times CFM \times (Ho - Hi)\ [Btu/h]$$

Where:

CFM= Volume flow rate of supply air, **(CFM)**
Ho, Hi = Moisture content of air at final and initial states, **(Btu/lb of dry air**).

5) Design Volume Flow Rate:

The design **volume flow rate - V (Kg/m³) (CFM)** is calculated on the basis of the capacity to remove the space cooling load at summer design conditions to maintain a required space temperature T_r:

$$V = \frac{QTotal}{60 \times \rho air \times (Ho - Hi)} = \frac{QSensible}{60 \times \rho air \times (To - Ti)}$$

Where:

V = Design volume flow rate, **(m³/h) (CFM**)
Q Total, Q Sensible = Design cooling load, **(kW) (Btu/h)**
ρair = Air density - may vary with air systems (see tables), **(Kg/m³) (lb/ft³)**
To = Room temperature - normally 75°F (24°C) - for comfort applications)
Ti = Supply air temperature leaving the cooling unit, **(°F) (°C)**
Ho = Room enthalpy, **Kcal/Kg) (Btu/lb of dry air)**
Hi = Supply air enthalpy, **Kcal/Kg) (Btu/lb of dry air).**

6) Total Refrigeration Load:

Since the sensible heat **Qsensible** and the latent heat **Qlatent** are known, the total **Refrigeration Load** can be determined by:

$$RL = CFH \times \rho air \times (Ho - Hi)\ [Btu/h]$$

Where:

CFH = Air flow in ft³/h **(CFM x 60)**
ρair = Air density = **0.075 lb/ft³**
Hi = Enthalpy of the air - inside temperature
Ho = Enthalpy of the air – outside temperature

Obs.: CFM (Cubic feet per minute): amount of air that flows through a space in one minute. **1.0 CFM** equals approximately **2 liters/s (l/s)**. A typical system produces **400 CFM/ ton of air** conditioning.

. Constant of 4.5 = **1 CFM** airflow, there are 60 units in an hour, then, **60 CFM** = **4.5 lbs/air**.

. Constant of 1.08 = (Specific heat of air) 0.24 Btu/lb/°F x 4.5 = **1.08 Btu/lb/ºF**

. Constant of 0.68 = 1060/7000 x 4.5 = 0.68 Btuh as, **1,060 Btu/h** is latent heat water vaporization.

. Constant of 13.5 ft³/lb – dry air @ 70ºF, 50% RH = Specific volume of moist air

7) Specific Heat:

Is defined as the amount of heat energy needed to raise **1 gram of a substance 1°C** in temperature, or, the amount of energy needed to **raise one pound** of a substance **1°F** in temperature.

Q = m x Cp. (To – Ti)

Where:

Q = Heat energy needed (Joules) (Btu),
m = Mass of a substance (Kg) (lb),
Cp = Specific heat of air (see tables), **0.24 Btu/lb°F (1.01 Kcal/Kg°C)**
(*To – Ti*) = Dry Bulb temperature of air change (°C) (°F)

Substances with **higher specific heats require more heat energy** to lower temperature than do substances with a low specific heat.

Example:

Using metric units and imperial units, how much energy is required to heat **350 grams (0.77 pounds)** of gold from **10°C (50°F) to 50°C (122°F)**?

Mass = 350g = **0.35 Kg** = **0.77 lb**

Specific heat of gold (see tables) = 0.129 J/(g.°C) = 129 J/(Kg.°C) x 0.000239 = 0.0308 Btu/(lb.°F).

Q = m x Cp. (To – Ti)

Metric Units:

Q = (0.35 Kg) (129 J/(Kg.°C) (50°C - 10°C)

Q = **1,806 J** (1.71 Btu)

Q = m x Cp. (To – Ti)

Imperial Units:

Q = (0.77 lb) (0.0308 Btu/(lb.°F) (122°F - 50°F) =

Q = **1.71 Btu**

Notes:

1. A **greater ρ** means a smaller airflow rate **(Kg/m³) (CFM)** for a given supply mass flow rate.
2. Greater the sensible heat (Q Sensible), higher will be the airflow rate **(Kg/m³) (CFM)**.

10. HEAT LOSS BY CONDUCTION:

The different ways to calculate **heat losses** are: **"k"** values, **"C"** values, **"U"** values and **"R"** values.

a) ***k = Thermal Conductivity:*** Thermal conductivity means the rate of heat transfer through one inch of a homogeneous material expressed in **Btu-in/h.ft².°F** or **Btu-ft/h. ft².°F.** Materials with **lower "k"values** are better **insulators**.

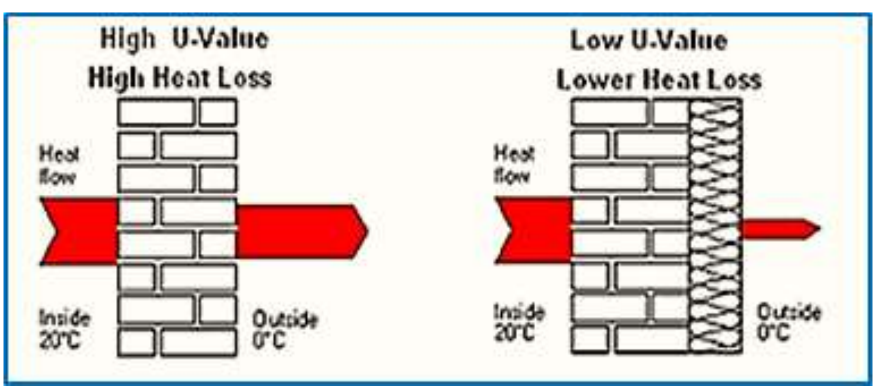
Q = k x A x ∆ T / t =

Where:

k = is thermal conductivity - see tables - (Btu-in/h ft² °F) or (Btu-ft/h ft² °F).
A = is the concerning area, ft²
∆ T = average temperature difference across the material, (F°)
t = thickness of a wall or some material (in)

b) ***C = Thermal Conductance:*** Thermal conductance is a specific factor is a heat transfer factor per inch of thickness. The **lower the "C" value**, the **better the insulator** and **the lower** is the heat loss. The overall **"C" value** must not be additive because **two insulating** materials with a **C-value** of **0.5** each, the result will be **1.0**.

c) ***U = Overall Coefficient of Heat Transmission:*** The **"U" value** is the rate of heat flow passing through a **square foot of a material per hour of each degree Fahrenheit difference in temperature** expressed in **Btu/h. ft².°F**. The **"U" value** is the inverse of the **"R" value**, ("U" = 1 / R) since ***the lower the "U" factor**, **the lower*** *is the heat loss.*



d) ***R = Thermal Resistance:*** The thermal resistance **"R" value** is a measure to retard heat flow. Thermal resistance is the reciprocal of a heat transfer coefficient. In other words, the **"R" value** is the inverse of the **"k" value (R = 1 / k)**, the **"C" value (R = 1 / C)** and the **"U" value (U = 1 / R)**.

Example.1:

Calculate the heat loss through a **3” thick** insulation board that has an area of **2 ft²** and has a **k-value** of **0.25**. Assume the average temperature difference across the material is **70°F**.

Q = k-value x A x ∆ T / t =

$$Q = \frac{0.25\ (k) \times 2\ (ft^2) \times 70°F\ (\Delta T)}{3\ (in)}$$

Q = 35 / 3 = **11.66 Btu/h**

Most good **insulating** materials have a **“k” value** of approximately **0.25 or less**, and rigid foam insulations have been developed with **“k” factors** as low as **0.12 to 0.15**.

Example.2:

Calculate the heat loss through a **100 ft²** wall with an inside temperature of **65°F** and an outside temperature of **35°F**. The wall is composed of **2" thickness** bricks having a **“k” factor of 0.80**. The insulation is **1" thickness** having a **“C” factor** of **0.16**.

Solution:

The **“U” value** is found as follows:

R total = 1 / k + 1 / C =
R total = 2”/0.80 + 1/0.16 =

R total = 8.75, Then:

U = 1 / 8.75 = **0.114 Btu/h ft² °F**

Once the **“U” factor** is known, the heat loss can be calculated by the **basic heat transfer equation**:

Q = A x U x (Ti – To) =
Q = 100 (ft²) x 0.114 (Btu/h ft² °F) x [65 (°F) - 35 (°F)] =
Q = 100 x 0.114 x 30 =

Q = 342 Btu/h

11. OVERALL COEFFICIENT OF HEAT TRANSMISSION – “U”:

The basic equation to calculate the **Overall Coefficient of Heat Transmission (“U”)** is:

U = 1/R Total, or:

$$U = \frac{1}{Ri + R1 + R2 + \ldots Ro}$$

Where:

Ri = resistance of a "boundary layer" of air on the inside surface;
R1, R2…= resistance of components of the walls, according to thickness of the component;

Ro = resistance of the "air boundary layer" on the outside surface of the wall.

Example:

Determine the **"U" value** and the **heat loss** through a **100 ft²** wall with a **70 °F** temperature difference for a layered wall. Construction composed of plywood **0.75-inch thick** (Cp = 1.25, expanded polystyrene **2-inches thick** (Cp = 4.00) and hardboard **1-inch thick** (Cp = 0.18).

Solution:

Inside air is assumed as **Ri = 0.68**

Plywood, 0.75-inch thick - **(R1 = 0.75 X 1.25 = 0.94)**

Expanded polystyrene, 2-inches thick - **(R2 = 2" X 4.00 = 8.00)**

Hardboard, 1.0-inch thick - **(R3 = 1 x 0.18 = 0.18)**

Outside air at 15 mph wind velocity is assumed as **Ro = 0.17.**

$$U = \frac{1}{Ri + R1 + R2 + R3 + Ro}$$

$$U = \frac{1}{0.68 + 0.94 + 8.0 + 0.18 + 0.17}$$

$$U = \frac{1}{9.97}$$

U = 0.10 Btu/h ft² °F

The heat loss **"Q"** for a **100 ft²** of wall with a **70° F** temperature difference will be:

Q = A x U x ∆ T =
Q = 100 x 0.10 x 70 =
Q = 700 Btu/h

12. HEAT LOSS CALCULATIONS:

The heat loss is divided into two groups:

1) The conductive heat losses through the building walls, floor, ceiling, glass, or other surfaces;
2) The convective infiltration losses through cracks and openings, or heat required to warm outdoor air used for ventilation.

The **heat loss** is determined by the **basic equation**:

Q = A x U x (To – Ti) = A x U x ∆ T =

Where:

Q = Total hourly rate of heat loss through walls, roof, glass, etc. (Btu/h);

U = Overall heat-transfer coefficient of walls, roof, ceiling, floor, or glass (Btu/h.ft².°F);
A = Net area of walls, roof, ceiling, floor, or glass (ft²);
Ti = Inside design temperature (°F);
To = Outside design temperature (°F).
Δ **T** = To – Ti (°F)

a) Net Area (A):

The **net area** of each building section is determined from **drawings or measurements** considering the areas of the four walls, floor, and ceiling, doors and windows and determine the volume of the building to estimate the rate of infiltration into the building measured in **air changes per hour (ACH)**.

b) Overall Coefficient of Heat Transfer (U):

The **U-value** measures how well a building component (wall, roof or a window), keeps heat inside a building. It is an indicator of how easy it is to keep the inside of the building cold.

A house built with low **U-value** building components will use less energy and thus using less energy is good for the environment.

The **inside design temperature** is traditionally taken as **65°F**. The temperature difference between the inside and outside of the building is the primary cause of heat loss in the winter months.

c) Outside Design Temperature (To):

The winter month **heating load conditions** are based on annual of **99.6 and 99%**, which suggests that the **outdoor temperature** is **equal to or lower** than design data **0.4% and 1%** of the time respectively. Then, commonly the **outside design temperature (To)** is **4°F**.

Example:

What is the value of the **heat loss** for a **10 ft²** building with a **single glass [U-value of 1.13]** with an inside temperature of **70°F** and an outside temperature of **0°**:

Q = A (10) x U (1.13) x Δ T (70 - 0) = **791 Btu/hr.**

Then, calculate the heat loss through each of the components separately and then **add their heat losses** together to get the total amount:

Q (wall) =Q (framed area) +Q (windows) +Q (door) =

13. HEAT CONDUCTION AND THERMAL RESISTANCE:

a) The **heat conducted** through a plane wall is:

$$Qw = \frac{k \times A\,(t1 - t2)}{L} \quad [Btu/h]$$

Where:

k = Thermal conductivity of the wall material, **Btu.in/h.ft². °F;**
A = Area of the wall, **ft²;**

t1, t2 = Temperature difference of the wall, **°F;**

L = Wall thickness, **inches.**

b) The same equation in terms of **thermal resistance** is:

$$Qw = \frac{A\ (To - Ti)}{R} = A \times U\ (To - Ti) \text{ - Btu/h}$$

14. HEAT LOSS DUE AIR CHANGES (ACH) AND VENTILATION:

To calculate this, you need to know how many times per hour the entire air in the building space is lost to outside referred to as **air changes per hour** or **ACH**. The infiltration can be considered to be **0.15 to 0.5 ACH** at winter design conditions.

a) Ventilation rate based on Air Change method:

V = ACH x A x H / 60

Where:

V = Ventilation air **(CFM) (m^3/h)**;
ACH = Air changes per hour **(ACH) – 0.15 to 0.5 CFM/ft^2**
A = Area of the space **(ft^2) (m^2);**
H = Height of the room **(ft) (m)**.

b) Ventilation rate based on Crack method:

V = I x A

Where:

V = Ventilation air **(CFM) (m^3/h)**;
I = Infiltration rate usually **0.15 CFM/ft^2**
A = Area of cracks/openings **(ft^2) (m^2)**;

c) Ventilation rate based on Occupancy method:

V = N x 20

Where:

V = Ventilation air, **(CFM) (m^3/h)**;
N = Number of people in space - usually **1 person per 100 ft^2 office**;
20 = Recommended ventilation rate is **20 CFM/person (34 m^3/h)** - (ASHRAE 62 standard.

15. TYPICAL CONCRETE FRAMES - THERMAL RESISTANCE:

For a **typical 2 x 4** concrete frame wall with **polystyrene insulation** and **drywall**, there are **five thermal resistance** layers due to convection and radiation.

Ri = 6.0 Btu/h ft^2 °F - then, **1/Ri = 1/6.0 Btu/h ft^2 °F**
Ro = 1.63 Btu/h ft^2 °F - then, **1/Ro = 1/1.63 Btu/h ft^2 °F**

So, the **thermal resistance** equation is:

R = Ri + R1 + R2 + R3 + R4 + R5 + Ro

Example:

A building, **35.0 ft wide, 73.0 ft long and 8.0 ft high**, is constructed with concrete **4 inches thick** and polystyrene insulation **2 inches thick on each side**. The building has a total of **4 windows** east, **16 windows** north measuring **2.5 ft by 4 ft.** Roof and ceiling are frame construction. The conditions are:

Inside: (Ti)
Dry bulb temperature = **80 °F**
Relative Humidity = **50%**

Outside: (To)
Dry bulb temperature = **95 °F**
Dew Point = **75 °F DP (dew point)**

Ventilation:
Supply air = **65 °F -** Dry bulb temperature.
Air Handling Unit (AHU) = Assume **4,000 CFM per AHU**

Others:

Average electrical usage = **1.0 watts/ft²**
Human activity per person = **180 Btu/h**
Infiltration rate, assume = **20 CFM**
Air density (**ρair**) = **0.075 lb/ft³**, at **80°F**, sea level.

The Thermal Conductivities **"k"**, the Thermal Resistances **"R"**, and the Overall Heat Transfer Coefficient **"U"** are shown in table below:

Material	k (Btu-in/ h ft² °F)	R (Btu/h ft² °F)	U (Btu/h ft² °F)
Siding		1.0	
Polystyrene - Wall Side 1 (2")	0.17		
Polystyrene - Wall Side 2 (2")	0.17		
Concrete (4")	10.0		
Drywall		0.45	
Pine 2 x 4	0.8		
Insulation	0.28		
Sheathing (0.5")	0.8		
Glass			1.13
Framed roof and ceiling			0.23

Considering that for a **typical 2 x 4 concrete frame** wall:

Ri = 1/Ri = 1/6.0 Btu/h ft² °F
Ro = 1/Ro = 1/1.63 Btu/h ft² °F

The **thermal resistance** of a **4-inches concrete** wall with **2-inches insulation** is:

Rtotal = Ri + R1 + R2 + R3 + R4 + R5 + Ro =

$$\textbf{Rtotal} = \frac{1.0}{6.0} + \frac{1}{0.17} + \frac{2.0}{0.17} + \frac{4.0}{10} + 0.45 + \frac{1}{1.63} =$$

R = 26.0 Btu/h ft² °F

1) Sensible Loads:

The **heat** conducted through the wall area **minus the windows area** is:

$$Q_{windows} = \frac{A(To - Ti)}{R} = \frac{[(A_{building} - A_{windows}) \times (To - Ti)]}{R} =$$

East wall **Qw** = [(8.0 x 35.0) - 4(2.5x4.0)] x (95 - 80) / 26 = 138 Btu/h
North wall **Qw** = [(8.0 x 73.0) - 16(2.5x4.0)] x (95 - 80) / 26 = 245 Btu/h

Qwindows **= 383 Btu/h**

There are **20 windows**, **10 ft² each**, the total glass area is **200 ft²,** considering that glass **U = 1.13,** the heat conducted through the glass is given by equation:

Qwindows **=** A x U x (To – Ti)
Qwindows = 200 x 1.13 (95 - 80) = **3,390 Btu/h**

The **infiltration** of outside air through cracks around windows and doors, a leakage rate of **20 CFM** assumed. The resulting sensible or infiltration heat gain inside is:

Qinfiltration **=** 1.08 x CFM (To – Ti)
Qinfiltration = 1.08 (20) (95 - 80) = **324 Btu/h**

For the medium size building considered here, the reference for average electrical usage is **1.0 watts/ft².** The heat gain from electrical appliances and lights is:

Remember:

1 Watt-Hour = 0.000948 (Btu/s) x 60 x 60 = **3.412 Btu/h**

Qlight **=** 3.412 x A
Qlight = 3.412 x (35.0 ft x 73.0 ft) = **8,717 Btu/h**

The **sensible heat** according to estimated light activity is **200 Btu/h per person.** For **twenty people** in the building, this gives a heat gain of:

Qoccupancy = 20 (200) = **4,000 Btu/h**

The heat conducted through the frame ceiling is:

Qceiling = A x U x (To – Ti) =
Qceiling **=** (35.0 x 73) x 0.23 x (95 – 80) = **8,815 Btu/h**

Adding all the heat gains the **total Qsensible load** is:

Qsensible **=** 383 + 3390 + 8717 + 330 + 4,000 + 8,815 =
Qsensible **= 25,635 Btu/h**

2) Latent Loads:

For **light activity**, people produce a latent gain of about 180 Btu/h per person, so for twenty persons:

Qlatent light = 20 (180) = **3,600 Btu/h**

Kitchen **appliances** add latent heat to the building as estimated below:

Dishwater = 420 Btu/h
Gas Oven = 1,200 Btu/h
Coffee maker = 2 x 1,540 = 3,080 Btu/h

Qkitchen **= 4,700 Btu/h**

The infiltration **humidity load** is determined by:

Qlatent **= 4,840 x CFM (Wo – Wi) =**

Using the **online WebPsych** can be found that **80^{o}F/50% RH** at sea level the air contains:

Inside: (Wi)
Dry bulb temperature = **80 °F**
Relative Humidity = **50%**

Wi = 76.8 gr/lb

Since there are **7,000 grains of moisture per pound** so we have:

Wi = $\mathbf{\frac{76.76}{7000}}$ **= ~ 0.011 lb of moisture/lb of air** or pounds of moisture per pound of air.

Outside: (Wo)
Dry bulb temperature = **95 °F**
Dew Point = **75 °F**

Wo = 131.81 gr/lb

Using the same considerations above:

Wo = $\mathbf{\frac{131.81}{7,000}}$ **= ~ 0.019 lb of moisture/lb of air** or pounds of moisture per pound of air.

Since that a leakage rate of **20 CFM** is assumed, the resulting latent heat gain inside is:

Qlatent = 4,840 x CFM (Wo – Wi) =
Qlatent = 4,840 x 20 (0.019 – 0.011) **= 774 Btu/h**

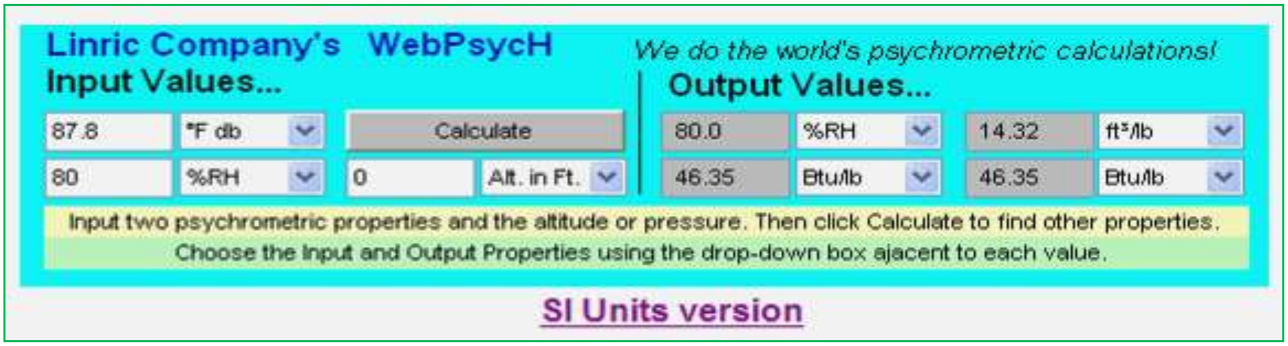
Adding all the heat gains the **total Qlatent load** is:

Qlatent **= 3,600 + 4,700 + 774** = **9,074 Btu/h**

Calculate the **Refrigeration Load**:

RL = CFH x ρair (Ho – Hi) =

a) First find **Hi** and **Ho** using the **online WebPsych**, as below:



Entering with the values:

Inside: (Ti)
Dry bulb temperature = **80 °F**
Relative Humidity = **50%**
Hi = 31.2 Btu/lb

Outside: (To)
Dry bulb temperature = **95 °F**
Dew Point = **75 °F**
Ho = 43.5 Btu/lb

b) Second find the **ventilation flow**:

Qsensible = 1.08 x CFM (To – Ti) =

Therefore,

$$\mathbf{CFM} = \frac{\mathbf{Qsensible}}{\mathbf{1.08\ (To - Ti)}}$$

$$\mathbf{CFM} = \frac{25{,}635}{1.08\ (15)} = \mathbf{1582}$$

CFH = 1,582 x 60 = **94,920 ft³/hr**

The **Refrigeration Load** is:

RL = CFH x ρair (Ho – Hi) =
RL = 94,920 x 0.075 (43.5 – 31.2) = **87,564 Btu/h**

The **Total Heat Loads** is:

Qtotal = **4.5 x CFM x (Ho – Hi) [Btu/h]**
Qtotal = 4.5 x 1,582 x (43.5 – 31.2) = **87,564 Btu/h (= RL)**

Considering the Total Heat Load (or Refrigeration Load) as Sensible Heat Load, the air conditioning loads are the **Sensible Heat Loads + Latent Heat Loads** as explained above, then:

Qload = **Q**sensible + **Q**Latent =
Qload = 87,564 + 9,074 = **96,638 Btu/h**

Calculating in **Tons of Refrigeration**:

$$\textbf{TR} = \frac{96{,}638 \text{ Btu/h}}{12{,}000 \text{ Btu/h}} = \textbf{8.0 tons}$$

The equipment size is based on the tons of cooling required, and a typical system produces **400 CFM/ ton of air** conditioning. Since, **1 ton of cooling = 12000 Btu/h = 400 CFM/ton** as explained above:

$$\textbf{CFM}(\text{equipment}) = \frac{96{,}638 \times 400}{12{,}000} = \textbf{3,221 CFM}$$

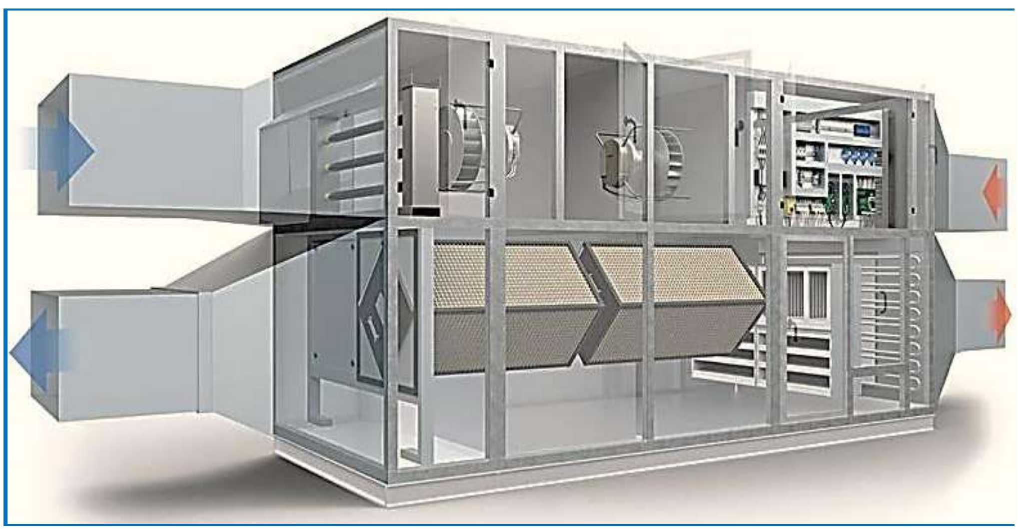
The system assumes **4,000 CFM** per Air Handling Unit (AHU), then:

$$\textbf{TR} = \frac{4000 \text{ CFM}}{400 \text{ CFM/ton}} = \textbf{10 tons = 1 Unit}$$

kW = 10 TR x 3.517 = **~35 kW**

Obs.:

An **air handler**, or **air handling unit** (abbreviated to **AHU**), is a device used to condition and circulate air as part of a heating, ventilating, and air-conditioning system. Air handlers usually connect to ductwork that distributes the conditioned air through the building and returns it to the **AHU**.



16. COOLING LOADS CONCEPTS:

Design **cooling loads** take into account all the loads experienced by a building under a specific set of assumed conditions. The assumptions for cooling loads are as follows:

a. Weather conditions are selected from a long-term statistical database, representative of the location of the building. ASHRAE has tabulated such data.
b. The solar loads on the building are assumed on a clear day in the month for calculations.
c. The building occupancy is assumed to be at full design capacity.
d. The ventilation rates are assumed on air changes or maximum occupancy expected.
e. All building equipment and appliances are considered to be operating.
f. Lights and appliances are considered for a typical day of design occupancy.
g. Latent as well as sensible loads are considered.

17. CLTD/SCL/CLF METHODS OF LOAD CALCULATION:

CLTD/SCL/CLF: Inside and outside theoretical air temperature differences for combined effects (**Cooling Load Temperature Difference/ Solar Cooling Factor/Cooling Load Factor**) for daily temperature range, solar radiation and heat storage.

CLTD factors: Are used for adjustment to conductive heat gains from walls, roof, floor and glass.

SCL factors: Are used for adjustment of heat gains from **glass** and are used for adjustment to heat gains from internal loads such as **lights, occupancy and power appliances**.

CLF: Is a **radiant energy** that enters the conditioned space at a particular time does not become a part of the cooling load instantly, calculated as functions of solar time and orientation and are available in the **form of tables in ASHRAE** Handbooks.

The basic conduction equation for heat gain is:

a) $\mathbf{Q = U \times A \times \Delta T}$ =

Where:

- **Q** = Heat gain in **Btu/h;**
- **U** = Thermal Transmittance for roof in **Btu/h.ft².°F;**
- **A** = Area of in **ft²;**
- **ΔT** = Temperature difference in **°F.**

The **heat gain** is converted to **cooling load** using the functions (sol-air temperature) for light, medium and heavy thermal characteristics. The equation is modified as:

b) **Q = U * A * (CLTD) =**

Where:

- Q = Cooling load, **Btu/h;**
- U = Coefficient of heat transfer roof or wall or glass, **Btu/hr.ft².°F;**
- A = Area of roof, **ft²;**
- CLTD = Cooling load temperature difference **°F** from tables in AHSRAE Handbook.

c) $\mathbf{Q = U \times A \times CLTD_{Corrected} =}$

$CLTD_{Corrected}$ ASHRAE Method: The **ASHRAE** tables provide hourly **CLTD values** for one typical set of conditions. Outdoor maximum temperature of **95°F** with mean temperature of **85°C** and daily range of **21°F**, the equation is adjusted to correction factors:

The typical equation for Roofs and Walls are:

$$\mathbf{CLTD_{Corrected} = CLTD + (78 - TR) + (TM - 85)}$$

Where:

- (78 – TR) = Indoor design temperature correction;
- (TM – 85) = Outdoor design temperature correction;

- TR = Indoor room temperature;
- TM = Mean outdoor temperature = Tmax – (Daily Range) / 2.

Practical Example: Estimate the cooling load using the Cooling Load Temperature Difference / Solar Cooling Load/ Cooling Load Factor **(CLTD/SCL/CLF)** method.

Given,

Type of building = Office;
Working = 8 hrs of working - 9.00 to 17.00 h;
Room Length x Width = 16 ft x 16 ft;
Room Height = 15 ft;
Window area = 20 % of the wall area;
Roof/Walls: U= 0.2 Btu/h.ft². ºF;
Windows: U= 0.55 Btu/h.ft².ºF;
Occupancy = 2 persons per room.

Design Conditions:

Indoor design Dry-bulb = 78 °F;
Outdoor design dry-bulb = 90 °F(max);
Outdoor design wet-bulb = 75 °F;
Daily Range = 20 °F.

Room Considerations – Roof and Walls:

Room area is (16 x 15) = **240 ft²;**
Window area for each wall is (16 x 15 x 0.2) = **48 ft²;**
Net area for each wall is 240 ft² - 48 ft² (window) =**192 ft².**

Calculation of Room CLTD Correction:

$\mathbf{CLTD_c = CLTD + (78 - TR) + (TM - 85) =}$

TR = Indoor design temperature = **78°F**
TM = Outdoor design dry bulb temperature - (Daily range /2) = 90 - (20 / 2) = 90 -10 = **80**

$\mathbf{CLTD_c}$ = CLTD + (78 - 78) + (80 - 85) = (**CLTD – 5)**

Table for Calculation of Heat Load due to Conduction from 9.00 to 17.00:

Time of Day	CLTD*	CLTDc = (CLTD – 5)	Q = U * A * CLTDc
9.00	9	4	154
10.00	9	4	154
11.00	9	4	154
12.00	10	5	192
13.00	10	5	192
14.00	11	6	230
15.00	12	7	269
16.00	14	9	346
17.00	15	10	384

***CLTD** = from tables in AHSRAE Fundamentals Handbook

U = 0.2 $Btu/h.ft^2.F$
A = 192 ft^2

a) Calculation of Radiation Load through the Windows from 9.00 to 17.00:

Q (Btu/hr) = U ($Btu/hft^2\,°F$) x A (ft^2) x CLTDc

U = 0.55 $Btu/h.ft^2\,°F$

Q (Btu/hr) = A x SC x SCL =

A = Glass area = 48 ft^2
SC = Shading Coefficient (ASHRAE Table 19-05F31.48)
SCL = Solar Cooling Load Factor (ASHRAE Table 36-89F26.41)

Time of Day	SCL	SC	Q = A x SC x SCL
9.00	27	0.72	933
10.00	30	0.72	1037
11.00	33	0.72	1140
12.00	34	0.72	1175
13.00	35	0.72	1210
14.00	34	0.72	1175
15.00	32	0.72	1106
16.00	29	0.72	1002
17.00	29	0.72	1002

b) Calculation of Internal Load for People:

Q sensible (Btu/hr) = N (number of people) x SHG (Btu/hr) x CLF =
Q latent (Btu/hr) = N (number of people) x LHG (Btu/hr) =

Practical Example Review: Calculate the **U-values** and the **Heat Loss** through the various components with the following data:

Given:

- Dry-bulb temperature = **70ºF**
- Outside Temperature = **22 º**F
- Dew point for the cooled air = **50ºF**
- Relative humidity = **40%**

1) Horizontal Roof - Roof area = 1200 ft^2:

- Built-up roofing, 0.375" thick - **Rb1 = 0.33 $Btu/h.ft^2.ºF$**;
- Extruded polystyrene, 3" - insulation smooth skin surface **– Rb2 =15 $Btu/h.ft^2.ºF$**;
- Plywood panels deck, 3/4" - **Rb3 = 0.93 $Btu/h.ft^2.ºF$**
- Air space (Tmean = 50ºF, DT = 10ºF) (air space 3.5") - **Rb4 = 0.93 $Btu/h.ft^2.ºF$**;

- Gypsum board, 0.5" – **Rb5 = 0.45 Btu/h.ft².ºF**;
- Acoustical ceiling tile, 0.5" – **Rb6 = 1.25 Btu/h.ft².ºF**.

2) Walls – Walls area = 1000 ft²:

- Common face brick, 4"density: 130 lb/ft 3 – **Rw1 = 0.56 Btu/h.ft².ºF;**
- Air space, 0.5" - 50ºF mean and 10ºF temperature difference – **Rw2 = 2.54 Btu/h.ft².ºF;**
- Nail-base sheathing with bright aluminum foil 0.5" to air space - **Rw3 = 1.06 Btu/h.ft².ºF;**
- Glass fiber 3.5", insulation(3.5") – **Rw4 = 12.98 Btu/h.ft².ºF;**
- Gypsum board, 0.5" – **Rw5 = 0.45 Btu/h.ft².ºF.**

And,

- Outside air film (15 mph) – **Ri = 0.17;**
- Inside air film (vertical surface) - **Ro = 0.68.**

Also,

- Windows glass, double pane with **0.5"** air space, **210 ft²**, U-value = **0.64 Btu/h.ft².ºF**;
- Door, 1 3/4" solid core with metal, area 24 ft², U-value = **0.26 Btu/h.ft².ºF**.
- Assume infiltration - Air change = **0.6** for building volume = **12,000 ft³**.
- Assume ten people in the building, **20 CFM per person**.

3) Using the **webpsy/linric** for **dry-bulb temperature 70ºF** and **relative humidity 40%** is found:

- **Wb** = 55.76 °F wb;
  Hi = 23.60 Btu/lb;
  Dp = 44.60 °F dp;

Wi = 43.66 gr/lb / 7,000 = **0.0062 lb/lb – dry air.**

4) Using the **webpsy/linric** for **dry-bulb temperature 70ºF** and **dew point 50ºF** is found:

- **Wb** = 58,18 °F wb;
- **RH** = 49,02 % RH;
- **Ho** = 25,15 Btu/lb;

Wo = 53.62 gr/lb / 7,000 = **0.0076 lb/lb – dry air.**

1) Determination of the Roof (U-Value) R:

Solution:

- Outside air film (15 mph) – **Rbi = 0.17**
- Built up Roofing (0.375") - 1/C = 1/3 - **Rb1= 0.33**
- 3" Extruded Polystyrene Insulation (smooth surface) – x/k = 3"/0.2 - **Rb2 = 15.00**
- 3/4" plywood deck - 1/1.07 - Rb3= **0.93**
- Non reflective air space (Tmean = 50ºF, DT = 10ºF) (air space 3.5") - **Rb4 = 0.93**
- 0.5" gypsum board - 1/2.22 - **Rb5 = 0.45**
- 0.5" acoustical tile - 1/0.8 – **Rb6 = 1.25**
- Inside air film (vertical surface) - **Rbo = 0.68**

Total (U-Value) - Rtot = 19.74

1/Rtot = 1/19.74 = **0.0506 Btu/h.ft².ºF**

2) **Determination of the Walls (U-Value) R:**

Solution:

- Outside air film – **Rwi = 0.17**
- 4" Common Face Brick – **Rw1 = 0.56**
- 0.5" air space (Tmean = 50ºF, DT = 10ºF) – **Rw2 = 2.54**
- 0.5" nail - base sheathing w/ aluminum foil – **Rw3 = 1.06**
- 3.5" glass fiber batt insulation – **Rw4 = 12.98**
- 0.5" gypsum board – **Rw5 = 0.45**
- Inside air film (For a vertical surface) – **Rwo = 0.68**

Total (U-Value) - Rtot = 18.44

1/Rtot = 1/18.44 **= 0.054 Btu/h.ft².ºF**

3) **Determination of Heat Loss through the roof - area of 1,200 ft²:**

Ti = 70ºF, To = 22ºF, A = **1,200 ft²**

Qroof = Uvalue x Aroof x (Ti - To) =

Qroof = 0.0508 x 1,200 x (48) = **2,926.08 Btu/h**

4) **Determination of Heat Loss through the walls - area of 1000 ft²:**

Ti = 70ºF, To = 22ºF, A = **1,000 ft²**

Qwall = Uvalue x Awall x (Ti - To) =

Qwall = 0.054 x 1,000 x (70 - 22) = **2,592 Btu/h**

5) **Determination of Heat Loss - windows clear glass, double pane, 0.5" air space:**

Solution:

Window area = 210 ft², U-value = **0.64 Btu/h.ft².ºF, Ti = 70ºF To = 22ºF**

Qwindow = U x A x (Ti - To) =

Qwindow = 0.64 x 210 x (48) = **6,451.2 Btu/h**

6) **Determination of Heat Loss - door, 1 3/4" solid core with metal:**

Solution:

Door Area = 24 ft², U-value = **0.26 Btu/h.ft².ºF, Ti = 70ºF To = 22ºF**

Qdoor = U x A x (Ti - To) =

Qdoor = 0.26 * 24 * (48) = **299.52 Btu/h**

7) Infiltration heat loss - Air change = 0.6, building volume: 12000 ft^3:

a) Sensible Heat Loss:

Qs = 1.08 x V_{CFM} x (Ti - To) =

V_{CFM} = (0.6*12,000) / 60 = **120 CFM**

Qsensible-inf. = 1.08 * 120 * 48 = **6,220.8 Btu/h**

b) Latent Heat Loss:

Qlatent-inf. = 4,840 x V_{CFM} x (Wo - Wi) =
Qlatent = 4,840 x 120 x (0.0076 – 0.0062) = **813.12 Btu/h**

8) Determination of Heat Loss by Ventilation - ten people, 20 CFM per person:

V_{CFM} = 10 persons x 20 CFM = **200 CFM**

Qsensible-vent. = 1.08 * V_{CFM} * (Ti - To) =
Qsensible-vent = 1.08 * 200 * 48 = **10,368 Btu/h.**

Qlatent-vent. = 4,840 * V_{CFM} * (Wi - Wo) =
Qlatent-vent = 4,840 x 200 x (0.0076 – 0.0062) = **1,355.2 Btu/h**

9) Total Heat Loss through the building:

Qroof = **2,926.08 Btu/h;**
Qwall = **2,592 Btu/h;**
Qwindow = **6,451.2 Btu/h;**
Qdoor = **299.52 Btu/h;**

Qsensible-inf. = **6,220.8 Btu/h;**
Qlatent-inf = **813.12 Btu/h;**
Qsensible-vent = **10,368 Btu/h ;**
Qlatent-vent = **1,355.2 Btu/h.**

Total Heat Loss = 31,026 Btu/h.

- ✓ Since, **1 Ton** of Refrigeration= **12,000 Btu/h:**
- ✓ Therefore, 31,026/12,000 = **2.56 tons** - select a **3.0 ton Air Handling Unit** (AHU) for heating.

18. SIZING DUCTS:

Sizing of ducts is given by the continuity equation like:

1) Area Method:

A = q / v =

a) *SI units:*

A = Duct cross sectional area (m^2);
q = Air flow rate (m^3/s);
v = Air speed (m/s).

b) Imperial units:

A = 144 q / v =

A = Duct cross sectional area (in^2);
q = Air flow rate (CFM);
v = Air speed (FPM).

2) Velocity Method:

a) SI units:

v = Q / A =

v = Air velocity (m/s);
Q = Air volume (m^3/s);
A = Cross section of duct (m^2).

Or,

$$v = q / A = 4\, q / (\pi\, d^2) = q / (a \times b) =$$

v = Air velocity (m/s);
q = Air flow (m^3/s);
A = Area of duct (m^2);
d = Diameter of duct (m);
a = Width of duct (m);
b = Width of duct (m).

b) Imperial units:

$$v = q / A = 576\, q / (\pi\, d^2) = 144\, q / (a \times b) =$$

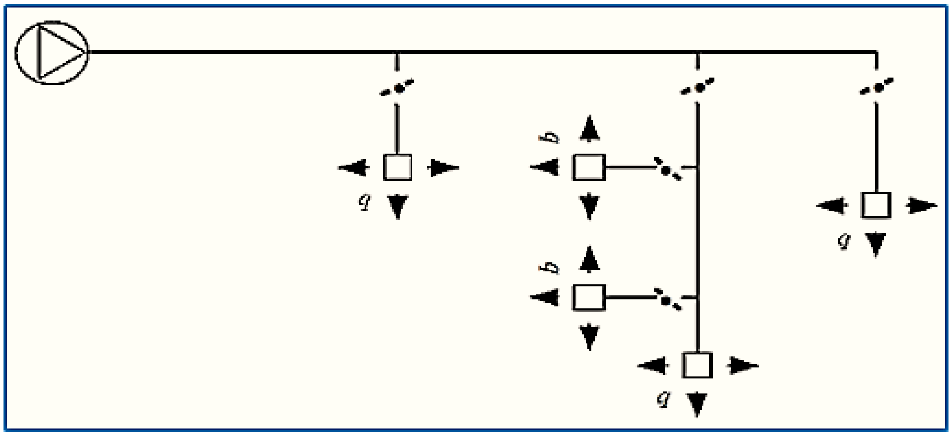
v = Air velocity (FPM);
q = Air flow (CFM);
A = Area of duct (ft^2);
D = Diameter of duct (inches);
a = Width of duct (inches);
b = Width of duct (inches).

3) Application of Proper Velocity:

A proper **velocity** for ventilation depends on the application and the environment. The **table below** indicate commonly used **velocity limits**:

Type of Duct	Comfort Systems		Industrial Systems		High Speed Systems	
	m/s	FPM	m/s	FPM	m/s	FPM
Main ducts	4 - 7	787 - 1378	8 - 12	1575 - 2362	10 - 18	1968 - 3543
Main branch ducts	3 - 5	590 - 984	5 - 8	984 - 1575	6 - 12	1181 - 2362
Branch ducts	1 - 3	197 - 590	3 - 5	590 - 984	5 - 8	984 - 1575

Obs.: Be aware that **high velocities** close to outlets and inlets may **generate unacceptable noise**. Maximum air velocity in the ducts should be kept below certain limits to avoid unacceptable generation of noise.



The values from the table below can be used to rough sizing of ducts in comfort, industrial and high speed ventilation systems. Commonly, the accepted duct velocities can be found in the **table below**:

Service	**Velocity – *v*** *(m/s and FPM - not equal conversions)*			
	Public buildings		**Industrial plants**	
	(m/s)	*FPM*	*(m/s)*	*FPM*
Air intake from outside	2.5 - 4.5	500 - 900	5 - 6	1000 - 1200
Heater connection to fan	3.5 - 4.5	700 - 900	5 - 7	1000 - 1400
Main supply ducts	5.0 - 8.0	1000 - 1500	6 - 12	1200 - 2400
Branch supply ducts	2.5 - 3.0	500 - 600	4.5 - 9	900 - 1800
Supply registers and grilles	1.2 - 2.3	250 - 450	1.5 - 2.5	350 - 500
Main extract ducts	4.5 - 8.0	900 - 1500	6 - 12	1200 - 2400
Branch extract ducts	2.5 - 3.0	500 - 600	4.5 - 9	900 - 1800

4) Pressure Loss in Ducts:

Dp = dp1 + dp2 + dp3 =

Where:

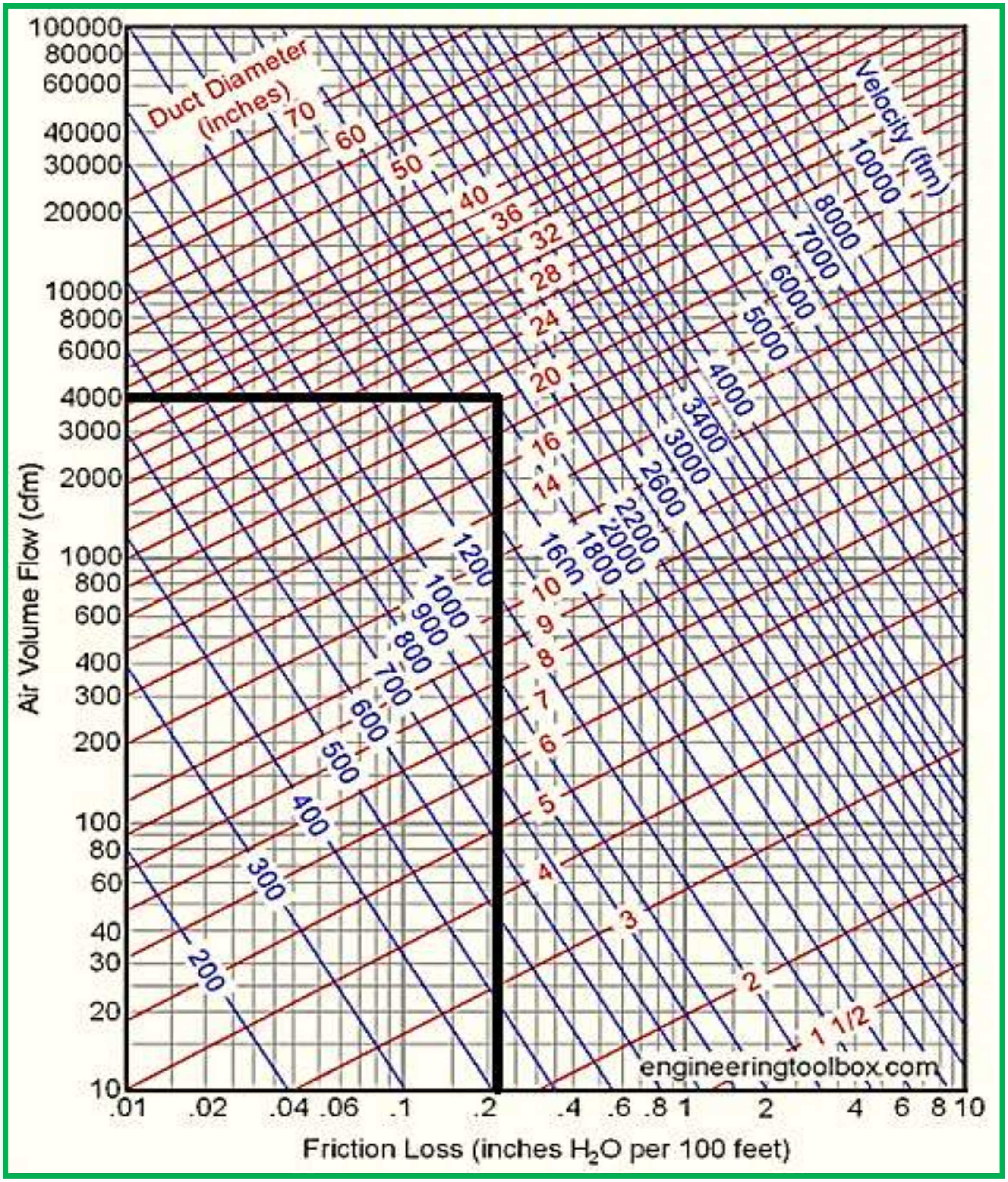
Dp = Total pressure loss in system – psi (Pa, N/m^2);
dp1 = Major pressure loss in ducts due to friction – psi (Pa, N/m^2);
dp2 = Minor pressure loss in fittings, bends etc. –psi (Pa, N/m^2);
dp3 = Minor pressure loss in components as filters, heaters etc. – psi (Pa, N/m^2).

5) Air Ducts Friction Loss Diagram:

Practical Example - Friction Loss in Air Ducts, using the diagram**:**

The friction loss in a **20 inches duct** with air volume flow at **4,000 CFM** can be estimated to be approximately **0.23 inches water, per 100 feet duct length**, as shown in the diagram below. The **air velocity** can be estimated to between **1,830** to **1,850 FPM**.

Friction loss (head loss) in standard air ducts are indicated in the diagram below:



Obs.: The diagram is based on **standard air 0.075 lb/ft³** in clean round galvanized metal ducts.

6) Pressure Loss in Ducts Due to Friction:

$Dp_f = R\ l$

Where:

R = Duct friction resistance per unit length - psi/ duct ft (Pa, N/m^2 per m duct);
l = Length of duct – inches (m).

7) Duct Friction Resistance per Unit Length:

$R = \lambda / D_h\ (\rho \times v^2 / 2) =$

Where:

R = Pressure loss – psi (Pa, N/m^2);
λ = Friction coefficient;
D_h *= Hydraulic diameter – inches (m).*

19. TEMPERATURE LOSS IN DUCTS:

The heat loss from a duct can be expressed as:

a) $H = A \times k\ (To + Ti) / (2 - Tr) =$

Where:

H = Heat loss – **Btu (W);**
A = Area of duct walls – **sq. inches (m^2);**
Ti = Initial temperature in duct - **ºF (^{o}C);**
To = Final temperature in duct - **ºF (^{o}C);**
k = Heat Transfer Coefficient = **(5.68 W/m^2 K for sheet metal, 2.3 W/m^2 K for insulated ducts) –** or,
k = Heat Transfer Coefficient = **(1.0 Btu/h.ft^2.ºF for sheet metal, 0.4 Btu/h.ft^2.ºF for insulated ducts);**
Tr = Surrounding room temperature - **ºF (^{o}C).**

The **heat loss** in the air flow can be expressed as:

b) $H = q \times cP\ (To - Ti) =$

Where:

*q = Mass of air flowing **– lb/s (kg/s);***
*cP = Specific heat capacity of air – **Btu/lb. ºF (kJ/kg K).***

Formulae (a) and (b) can be combined to:

c) $H = A \times k\ (To + Ti) / 2 - Tr) = q \times C_p\ (To - Ti) =$

Obs.: 1.0 inch water = 248.8 N/m^2 (Pa) = **0.0361 lb/in^2 (psi)** = **25.4 kg/m^2** = 0.0739 in mercury.

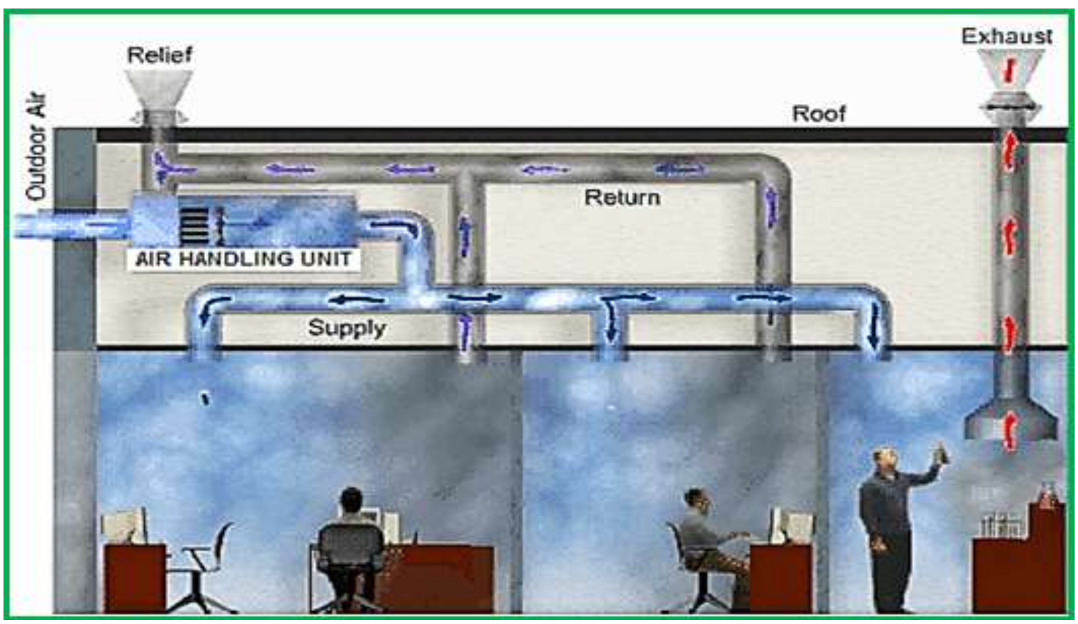
- **1.0 ft^3/min (CFM)** = 1.7 m^3/h = **0.47 l/s**;
- **1.0 ft/min (FPM)** = **0.00508 m/s**.

20. RULES OF THUMB FOR HVAC CALCULATION:

Property	*Units*	*Water*	*Air*
Heat Capacity	KJ/kg °C	4.2	1.0
	Btu/lb °F	1.0	0.239
Density	kg/m^3	1000	1.29@STP (1 bar, 0°C)
	lb/ft^3	62.29	0.075@STP (14.696 psia, 0°F)
Latent Heat	KJ/kg	1200 - 2100	
	Btu/lb	516 - 903	
Thermal Cond.	W/m °C	0.55 - 0.70	0.025 - 0.05
	Btu/h ft °F	0.32 - 0.40	0.014 - 0.029
Viscosity	cP	1.8 @ 0 °C	0.02 - 0.05
		0.57 @ 50 °C	
		0.28 @ 100 °C	
		0.14 @ 200 °C	
Prandtl Number		1 - 15	0.7

21. CHILLERS AND AIR HANDLING UNITS (AHU):

Chillers provide the cooling of the chilled water, which is then used in the air cooling coils for air conditioning. **Chillers** are used for relatively **large capacity systems**. **AH Units** provide the cooling by direct expansion of the refrigerant in the air cooling coils, commonly used for **relatively small capacity** systems. Typically an AHU system is not used for systems **larger than 100 ton capacity.**



AH Package Units: Are available in following major categories, based on the type of the compressor and packaged or split design. The air cooling coil in AH Units and the chilled water heat exchanger in chillers are also called an **"Evaporators"** since evaporations of the liquid refrigerant occurs in these components.

Chillers Package Units: Commonly mounted on one skid, with the water piping system being external to the unit. The **package unit** includes **the compressor, condenser, evaporator and associated components**.

- ✓ **Reciprocating or scroll compressor:** Package (condenser, compressor & evaporator) all installed on the same skid);
- ✓ **Reciprocating or scroll compressor:** Split systems (condenser, compressor, evaporator are located at separate locations - many systems have the condenser + compressor on the same skid as condensing unit, and the evaporator is located separately);
- ✓ **Centrifugal Compressor Chillers:** (used for medium to large capacity applications);
- ✓ **Screw Compressor Chillers:** (used for medium capacity applications).

Obs.: The condenser for both AH Units and chillers can be air cooled or water cooled for relatively smaller sizes. For the **larger units the condenser** is typically water cooled.

22. CHILLERS & AH SIZING RULES:

- ✓ **AH Unit Type – Reciprocating or Scroll**:

Capacity range for package units – **0.5 to 150 Tons**
Capacity range for split units – **1 to 70 Tons**
Power – **0.9 to 1.3 KW / Ton**
Min Capacity Turndown capability – depends on number of cylinders

- ✓ **Chiller Type – Reciprocating**:

Capacity range – **15 to 100 Tons**
Power – **0.9 to 1.3 KW / Ton**
Min. Capacity Turndown capability – depends on number of cylinders

- ✓ **Chiller Type – Scroll**:

Capacity range – **10 to 150 Tons**
Power – **0.9 to 1.3 KW / Ton**
Min. Capacity Turndown capability – depends on number of cylinders

- ✓ **Chiller Type – Rotary Screw**:

Capacity range – **70 to 500 Ton**
Power – **1 to 1.5 KW / Ton**
Min. Capacity Turndown capability – 25%

- ✓ **Chiller Type – Centrifugal**:

Capacity range – **200 Ton to 2,000 Ton**
Power – **0.5 to 0.85 KW / Ton**
Min. Capacity Turndown capability – 10%

23. HVAC REFRIGERANTS:

In general the most common refrigerants used in the industry belong to the following three categories:

- **CFC**: These are the Chloro Fluoro Carbon refrigerants, such as R11, R12, R113, R114, identified as the most **harmful to Ozone** layer by the Montreal Protocol, and were phased out in 2000.
- **R12:** is used commonly in the older cars for air conditioning.
- **HCFC**: The Hydro Chloro Fluoro Carbon refrigerants, such as R22, R123, etc., identified as **slightly harmful to the Ozone** layer by Montreal Protocol, and will be phased out by 2030.
- **R22 Refrigerant:** is commonly used in most reciprocating type of compressors, while **R123** is used in centrifugal chillers as a temporary replacement for **R11**.
- **HFC**: These are the Hydro Fluoro Carbon refrigerants, such as R134a, that **do not harm the Ozone** layer, and are being used in the newer machines to **replace the CFC and HCFC**.
- The **R134a** is now commonly used to replace either **R12 or R500**, and in all new cars air conditioning systems.

a) Refrigerant Analysis:

A periodic refrigerant analysis is important to detect and control contaminants in the refrigerant, which can result in degradation and failure of the various components, and cause inefficient operation of the unit. Refrigerants **should be tested** for the following contaminants:

- Moisture, Acid, Particulate/solids, Organic matter – sludge, wax, tars, Non-condensable gases.

b) Moisture:

The acceptable levels of moisture in new or reclaimed refrigerants are given in ARI 700. These levels are generally more demanding than what is typically feasible and acceptable in an operating system.

Refrigerant	Allowable Moisture Level per ARI 700 (ppm by wt)	Normal Operating Moisture Levels (ppm by wt) (Ref. ASHRAE)
R11	20	0 – 30 (Centrifugal Chillers)
R12	10	0 – 25 (Centrifugal Chillers)
R22	10	0 – 56 (Recip.& Screw Chillers)
R113	20	0 - 30* (similar to R11)
R114	10	0 - 25* (similar to R12)
R134a	10	0 – 25* (similar to R12)
R500	10	0 – 25* (similar to R12)

Notes:

1) R113, R114, R134a, R500 data are not available in ASHRAE.
2) There is extensive research going on to identify new refrigerants that can be used to replace the **CFC** and **HCFC** refrigerants. Currently **R134a is the most commonly** used new refrigerant.

24. REFERENCES:

ASHRAE: The American Society of Heating Air Conditioning and Refrigeration

- 2001 ASHRAE Handbook of Fundamentals
- 1997 ASHRAE Handbook of Fundamentals
- ASHRAE Cooling and Heating Load Calculation Manual

NIST Standard Reference Database 69: NIST Chemistry WebBook.
University of Arkansas – School of Architecture
Refrigerants Temperature/Pressure Table for common refrigerants

Links:

http://www.engineeringtoolbox.com/sizing-ducts-d_207.html
www.engineeringtoolbox.com/air-psychrometrics-properties-t_8.html
www.linric.com/webpsy.htm
http://www.flycarpet.net/en/download.asp
http://www.trane.com
http://www.refron.com/InfoCenter/Home.asp
http://www.chemicalogic.com/moistairtab/default.htm
http://www.epa.gov/ozone/title6/index.html
http://www.smacna.org/

CHAPTER 2:

MECHANICAL COOLING TOWERS DESIGN

1. INTRODUCTION:

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.

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

2. COOLING TOWERS 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/hr**. These types of towers are used only by utility power stations.

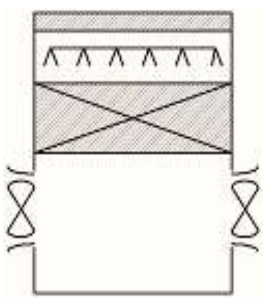
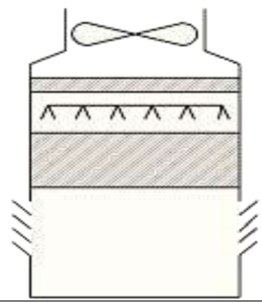
Mechanical Draft Towers: Use **large fans to force or suck the air** through the circulated water. The **water falls downward over fill surfaces**, which help increase the contact time between the water and the air - this helps maximize heat transfer between the two. An open circuit cooling tower is 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:

- Counter flow induced or forced draft cooling towers.
- Cross flow induced or forced draft cooling towers.

1. Counter Flow Draft: 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. Cross Flow Draft: 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**, e.g., 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** 

Make-up: To replenish the water lost due evaporation, is used the common called "**make-up water"** source. When the **hot water** is sent to the cooling tower, the **water is cooled through draft fans,** exits the cooling tower and is **sent back to the heat exchangers** or to other units for further cooling.

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.

Fill: 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.

Splash fill: The 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 fill: Consists 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 **film fill** is the more efficient and provides same heat transfer in a smaller volume than the splash fill. Due it offers greater heat transfer efficiency, the fill is a choice for applications where the circulating water is generally free of debris that could plug the fill passage ways.

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

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, located low on side or bottom of counter flow towers.

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: Widely used for cooling tower casings and basins, giving long life and protection from the harmful effects of many chemicals.

Plastics: 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. 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**.

Obs.: In some forced draft counter flow design, the water at the bottom of the fill, is channeled to a perimeter 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.

4. COOLING TOWERS PERFORMANCES:

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).

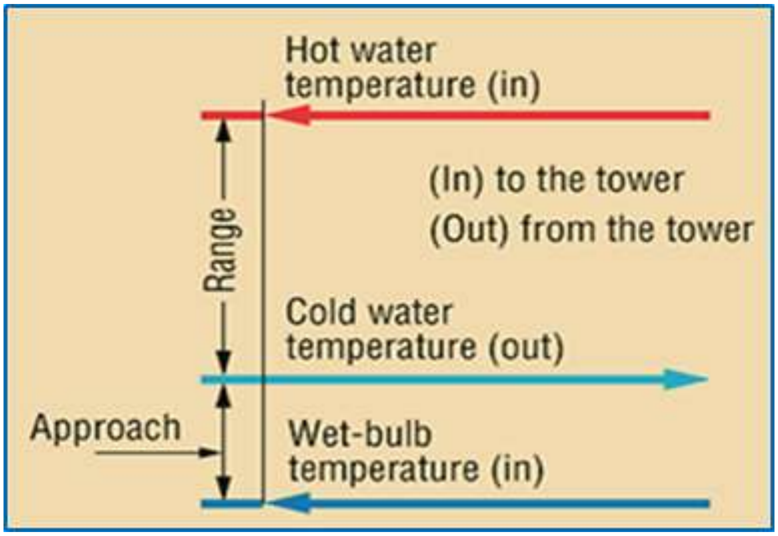
iv) **Cooling Capacity:** is the **heat rejected in kCal/hr (Btu/h)**, given as product of mass flow rate of water, specific heat and temperature difference.

v) **Evaporation Loss:** is the **water quantity evaporated** for cooling duty. Theoretically, for every **10,000,000 kCal/h (39,656,668 Btu/h)** of heat rejected, **the evaporation** quantity works out to **1.8 m³/h (63,566 ft³/h)**. So, an empirical relation is often used:

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

Where:

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



*Source: Perry's Chemical Engineers Handbook.

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

- ✓ **Blow Down** losses depend upon cycles of concentration and the evaporation losses and is given by relation:

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

- ✓ **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);**
T1 = hot water temperature **(°C) (°F);**
T2 = cold water temperature **(°C) (°F);**
h2 = enthalpy of air-water vapor mixture at exhaust wet-bulb temperature (units as above);
h1 = enthalpy of air-water vapor mixture at inlet wet-bulb temperature (units as above).

5. FACTORS AFFECTING COOLING TOWERS CAPACITY:

Cooling towers are specified to cool a certain flow rate **from one temperature to another temperature** at a **certain wet bulb temperature**.

5.1. Range: Range is a function of the **heat load and the flow** circulated through the system, 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.

Cold Water Temperature Range = 32.2°C (90°F);

Wet Bulb Temperature Range = (26.7°C) (80°F);

Approach Range = (5.5°C) (10°F).

Range °C (°F) = Heat Load (kcal/h) (Btu/h) / Water Circulation Rate - (l/h) (gal/h);

The 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. Commonly, **the closer the approach to the wet bulb**, **the more expensive** is 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.

Obs.: 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.**

5.2. Approach and Flow for Cooling Towers Sizing: 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 m3/h** (160,328 ft³/h) through a **16.67°C** (62°F) range at a **26.7°C** (80°F) design wet bulb.

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

Practical example: A cooling tower **sized** to cool **4540 m3/h** (160,328 ft³/h) through a **13.9°C** (57°F) range, **might be larger** than a cooling tower to cool **4540 m3/h** (160,328 ft³/h) through **19.5°C** (67°F) range. 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.

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

Practical example: 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.

Solution: The cooling tower cools **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).

- ✓ **Metric Units: 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.**
- ✓ **Imperial 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.**

The **table below 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).

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

5.4. Function of Fill Media in Cooling Towers: 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.
- ✓ Typical comparison of **Cross Flow Splash Fill**, **Counter Flow Tower with Film Fill and Splash fill** is shown in the table below. See below the comparison of a Cross-Flow Splash Fill X a Counter-Flow with film fill and splash fill:

Number of Towers : 2			
Water Flow : 16000 m^3/hr.			
Hot Water Temperature : 41.5°C			
Cold Water Temperature : 32.5°C			
Design Wet Bulb Temperature : 27.6°C			
	Counter Flow Film Fill	Counter Flow Splash Fill	Cross-Flow Splash Fill
Fill Height, Meter	1.5	5.2	11.0
Plant Area per Cell	14.4 × 14.4	14.4 × 14.4	12.64 × 5.49
Number of Cells per Tower	6	6	5
Power at Motor Terminal/Tower, kW	253	310	330
Static Pumping Head, Meter	7.2	10.9	12.05

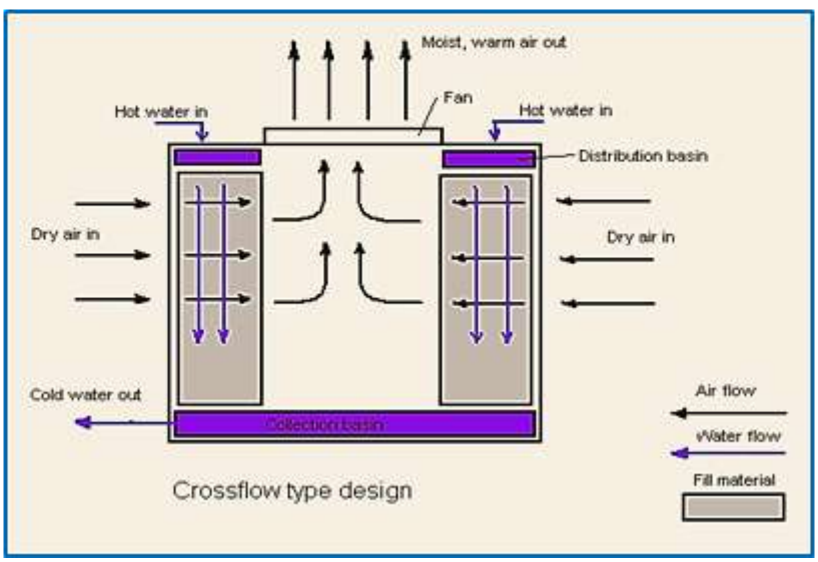
✓ ***Low-Clog Film*:** Have been developed to handle high turbid waters. For sea water, low clog film fills are considered as the best choice in terms of power saving and performance compared to conventional splash type fills. See below some comparisons between various fill media:

	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

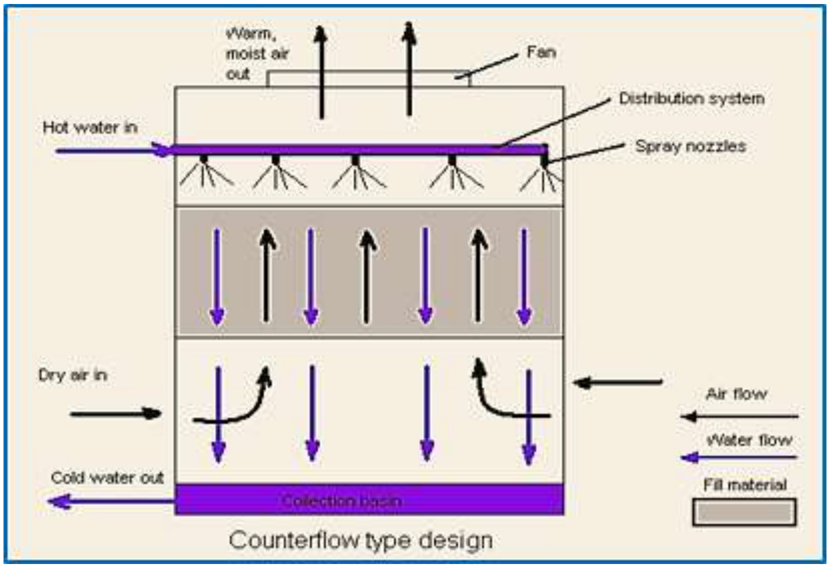
6. CHOOSING A COOLING TOWER TYPE:

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 Cooling Towers: Provided with **splash fill of concrete**, **wood or perforated PVC**.



c) Counter-Flow Cooling Towers: Are provided with **both film fill and splash fill**.



7. WATER FLOW AND HEAT TRANSFER:

First of all, the 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.

7.1. 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%.**

7.2. 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) =

Q2 = Heat transfer - Btu/h;
L = Mass flow of water entering the tower (water flow rate) - lb/min.;
t1= Hot water temperature entering the tower - °F;
t2 = Cold water temperature leaving the tower - °F.

7.3. 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.ft²;
S = heat transfer surface - ft²;
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" = Area of transfer surface per unit of tower volume (ft²/ft³);
V = Tower volume (ft³).

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 – lb/min.;

And the heat transfer rate from air side is:

Q = G x (ha2 - ha1) =

Where:

G = air mass flow rate - lb/min.;
ha1 = inlet enthalpy of air-water at the wet bulb temperature - Btu/lb - dry air;
ha2 = outlet enthalpy of air-water at the wet bulb temperature - Btu/lb - dry air.

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

7.4. Heat Balance: Is expressed in:

Cw x L1 x tw1 + G x ha2 =

Evaporation Loss is expressed in:

G x (w2 -w1) and is equal to L2 - L1 (water flow rate);

Therefore,

L1 = L2 - G x (w2 -w1) =

The enthalpy of outlet air is:

ha2 = ha1 + (Cw x L/G) x (tw2 - tw1) =

tw1 = temperature of water - inlet
tw2 = temperature of water - outlet

Or simply,

ha2 = ha1 + (L/G x Cooling Range) =

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. Calculate the ratio of **water flow and air rate** (L/G) for 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 (L) = GPM x (500 / 60) lb/min = 20,000 x (500 / 60) =**166,666.67 lb/min.;**
Air Flow Rate (G) = ACFM / Air Specific Volume = 1,600,000 / 14.3026 = **111,867.77 lb/min.**

Notes.:

a) 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.

b) Air Specific Volume @ **87.8°F (DBT)** and **80% (RH)** at sea level = **14.3026 ft³/lb**

7.6. Ratio of Water to Air (L/G):

Water Flow Rate / Air Flow Rate = 166,666.67 / 111,867.77 = **1.4898**

Example. Calculate the enthalpy **(ha2)** and the cooling range of air for the following Cooling Tower design conditions:

Ambient Wet Bulb Temperature = **82.4°F (WBT** (at sea level);
Relative Humidity = **80% (RH);**
L/G ratio = **1.4898**;
Entering (hot) Water Temperature = **107.6°F (tw1);**
Leaving (cold) Water Temperature = **89.6°F (tw2).**

Solution: According to above, the **enthalpy of exit air** is calculated from:

ha2 = ha1 + L/G x Cooling Range. It´s known that the enthalpy of the inlet air **(ha1)** at **82.4°F (WBT) and 80°F (RH)** at sea level is = **46.2528 Btu/lb of dry air.**

Cooling range = Entering Water Temp. - Leaving Water Temp. = **(tw2 - tw1) = 107.6 - 89.6 = 18°F**

Therefore, the enthalpy of exit air **(ha2)** is:

ha2 = ha1 + (L/G x Cooling Range) = 46.2528 + (1.4898 x 18) = **73.0692** = ~**73.07 Btu/lb.**

Note: The **value of air enthalpy** can be obtained from the **table** published by **Cooling Tower Institute** or other psychrometric curve, to find a temperature satisfying the same value of enthalpy varying a temperature by means of iteration.

8. NTU or KaV/L CALCULATION:

NTU means Number of Transfer Unit. The cooling towers calculation commonly uses a dimensionless factor designated as **KaV/L**. It is totally independent from the tower size and fill configuration and is often called, for lack of another name, **NTU**. This can be calculated using only the temperature and flows entering the cooling tower.

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

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/(hw - ha) at four points in the x axis (Temp.).

Example. Determine the NTU or KaV/L for the below given conditions:

Given,

Water Flow Rate = 16,000 GPM (L);
Air Flow Rate = 80,848 lb/min - of dry air (G);
Ambient Wet Bulb Temperature = 80.0°F (WBT);
Site Altitude = sea level;
Hot Water Temperature = 104.0°F (tw2);
Cold Water Temperature = 89.0°F (tw1).

Water Flow Rate (L) = 16,000 x (500 / 60) = **133,333 Lb/min.**

L/G Ratio = Water Flow Rate / Air Flow Rate = 133,333 / 80,848 = **1.6492**

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

- ✓ **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 Merkel curves are plotted with the thermal demand, **KaV/L** as a function of the liquid-to-gas ratio, **L/G**. The approach lines (tw1 - 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.

- ✓ **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.

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

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 flow 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).

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

8.2. 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 - 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.

8.3. 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). 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 (tw2 - tw1)] =

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.

Example. A Cooling Tower has a flow rate of **1000 GPM** from **95º F (tw2)** to **85º F (tw1)** at **72º F WBT** and operates at 3 cycles of concentration. Calculate the Cooling Range, Approach, Heat rejection, Drift loss, Evaporation rate, Bleed rate and Make up water requirements.

Solution:

1. Cooling Range:
(tw2 - tw1) = 95 - 85 = **10º F.**

2. Approach:
(CWT – WBT) = 85 - 72 = **13º F.**

3. Heat Rejection:
(Flow GPM x CWT x 500) = 1000 x 10 x 500 = **5,000,000 Btu/h.**

4. Drift Loss:
(0.002% x Flow Rate) = 0.00002 X 1000 = **0.02 GPM.**

5. Evaporation Rate:
(Flow rate x CWT / 1,000) = 1000 x 10 / 1,000 = **10 GPM.**

6. Bleed Rate:
[Evaporation Rate **/** (Cycles-1)] = 10 / (3-1) = **5 GPM.**

7. Make-up Requirement:
[Evaporation Rate x Cycles **/** (Cycles-1)] = 10 x 3/2 = **15 GPM.**

Example. Estimate the cold & hot water temperature when the water flow rate is increased from **16,000 to 20,000 GPM**, assuming no change in the entering air mass flow rate, wet bulb temperature, and heat load, using the following data below:

Given,

Water Flow Rate (L1) = **16,000 GPM;**
Air Flow Rate (G1) = **80,848 lb/min - of dry air;**
Ambient Wet Bulb Temperature = **80.0 °F (WBT);**
Site Altitude: **sea level;**
Hot Water Temperature (HWT, tw2) = **104.0 °F;**
Cold Water Temperature (CWT, tw1) = **89.0 °F;**
KaV/L = **1.4866;**
Characteristic Curve Slope (m) = **- 0.8;**
New Water Flow Rate (L2) = **20,000 GPM.**

Solution:

1. Cooling Range:
(tw2 - tw1) = 104 - 89 = **15°F.**

2. Water Flow Rate, L1:
Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = **133,333 lb/min.**

3. Heat Load, D1:
L1 x R1 = 133,333 x 15 = **2,000,000 Btu/min.**

4. Air Flow Rate, G1 = **80,848 lb/min.**

5. Liquid to Gas Ratio:
L/G1 = L1 / G1 = 133,333.3 / 80,848 = **1.6492.**

6. Water Flow Rate:
L2 = Water Flow Rate x (500 / 60) = 20,000 x (500 / 60) = **166,667 lb/min.**

7. Heat Load:
D2 = D1 = **2,000,000 Btu/min.**

8. Air Mass Flow Rate:
G2 = G1 = **80,848 lb/min.**

9. Liquid to Gas Ratio:
L/G2 = L2 / G2 = 166,666.7 / 80,848 = **2.0615**

10. Range:
R2 = D2 / L2 = 2,000,000 / 166,666.7 or = R1 x (L1 / L2) = **12°F**

8.4. Value of "C" of Tower Characteristic: For the design conditions as follows:

C = KaV/L / (L/G1)$^{-m}$ = KaV/L x (L/G1)$^{-m}$ = 1.4866 x (1.6492)$^{-0.8}$ = **2.21825**

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

New Tower Characteristic = C x (L/G2)$^{-m}$ = 2.21825 x (2.0615)$^{-0.8}$ = **1.2436**

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

New CWT (tw1) = WBT + New Approach = 80 + 10.45 = **90.45°F**
New HWT (tw2) = CWT + Range = 90.45 + 12 = **102.45°F**

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 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 (tw1) = WBT + New Approach = 80 + 10.35 = **90.35°F**
New HWT (tw2) = 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 graphical methods, is due to a very little difference in the enthalpy value between the formulas used.

Example. 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 has changed from **- 0.8** to **- 0.7.**

Others are the same as above example:

Solution:

First Step: Same as the example above.
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**

Third 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**

Note: 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 at the steps below, with 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 =

Fifth 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**

Through this example, it was proven that the cold water temperature at the slope of **- 0.7** is slightly lower than **- 0.8**.

Example. 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:

1. Range:
R1 = R2 = HWT - CWT = tw2 - tw1 = 104 - 89 = **15°F.**

2. Water Flow Rate:
L1 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = **133,333 lb/min.**

3. Heat Load:
D1 = L1 x R1 = 133,333 x 15 = **2,000,000 Btu/min.**

4. Air Mass Flow Rate:
G1 = G2 = **80,848 lb/min.**

5. Liquid to Gas Ratio:
L/G1 = L1 / G1 = 133,333 / 80,848 = **1.6492.**

6. Water Flow Rate:
L2 = Water Flow Rate x (500 / 60) = 20,000 x (500 / 60) = **166,667 lb/min.**

7. Heat Load:
D2 = L2 x R2 = 166,667 x 15 = **2,500,000 Btu/min.**

8. Liquid to Gas Ratio:
L/G2 = L2 / G2 = 166,667 / 80,848 = **2.0615.**

The new tower characteristic for the increased water flow is also same as the example 15-1. 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 =

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

Solution:

1. Range:
R1 = HWT - CWT = tw2 - tw1 = 104 - 89 = **15°F.**

2. Water Flow Rate:
L1 = L2 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = **133,333 lb/min.**

3. Air Mass Flow Rate:
G1 = G2 = **80,848 lb/min.**

4. Liquid to Gas Ratio:
L/G1 = L1 / G1 = L/G2 = 133,333 / 80,848 = **1.6492.**

5. Range:
R2 = **20°F.**

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

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. 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:

1. Range:
R1 = HWT - CWT = tw2 - tw1 = 104 - 89 = **15°F.**

2. Water Flow Rate:
L1 = L2 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = **133,333.3 lb/min.**

3. Air Mass Flow Rate:
G1 = **80,848 lb/min.**

4. Liquid to Gas Ratio:
L/G1 = L1 / G1 = 133,333.3 / 80,848 = **1.6492.**

5. Air Mass Flow Rate:
G2 = **53,900 lb/min.**

6. Liquid to Gas Ratio:
L/G2 = L2 / G2 = 133,333.3 / 53,900 = **2.4737.**

8.5. 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.**

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. 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 above.

Solution:

1. Range:
R1 = R2 = HWT - CWT = tw2 - tw1 = 104 - 89 = **15°F.**

2. Water Flow Rate:
L1 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = **133,333.3 lb/min.**

3. Air Mass Flow Rate:
G1 = **80,848 lb/min.**

4. Liquid to Gas Ratio:
L/G1 = L1 / G1 = L2 = 133,333.3 / 80,848 = **1.6492.**

5. Air Mass Flow Rate:
G2 = **53,900 lb/min.**

6. Liquid to Gas Ratio:
L/G2 = L2 / G2 = 133,333.3 / 53,900 = **2.4737.**

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.

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**

9. CONSIDERATION OF BY-PASS WALL WATER:

This consideration or factor accounts for the amount of water, which unavoidably by-passes 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 to account for it. It may be that in a very large or particularly in a small tower the wall effect can count to 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.

9.1. Estimation of the by-pass wall water: Let´s take an example to clarify this estimation. Considering that a **36 x 36 ft tower cell has 144 nozzles**. Assume that **40 nozzles** are near to the four walls each projecting **10 %** of their water onto those walls:

% nozzles near to 4 walls = 40 x 10% / 144 = **2.78 %.**

Then, 4 nozzles are in the corners and project **20%** of their water into the wall:

% water in walls = 4 x 20% / 144 = **0.56 %.**

There are 25 internal columns. Each column receives 5% of the water from 4 adjacent nozzles:

% water in columns = 25 x 4 x 5% / 144 = **3.47 %.**

Then, the total **by-pass water** is **2.78 % + 0.56 % + 3.47 % = 6.81%,** and the water amount for being **half-cooled** is:

% water to be half-cooled = 6.81 / 2 = **3.4%.**

This means that **3.4%** of the total water flow is **passing through the wall and 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 is **18 ft 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 passing through the cooling tower without being heated by the heat exchanger.

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

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.

1. Original Range:
R1 = HWT - CWT = tw2 - tw1 = 104 - 89 = **15°F.**

2. Water Flow Rate:
L1 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = **133,333.3 lb/min.**

3. Heat Load:
D1 = L1 x R1 = 133,333.3 x 15 = **2,000,000 Btu/min.**

4. Heat Load:
D2 = D1 = **2,000,000 Btu/min.**

5. Tower Water Flow Rate:
L2: 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.**

6. Range:
R2 = D2 / L2 = 2,000,000 / 128,000 = **15.625°F.**

7. Consideration:
L1 x R1 / {L1 x (1 - %By-Pass Wall Water / 100)} = R1 / (1 - % By-Pass Wall Water / 100) = (104 - 89) / (1 - 4 / 100) = **15.625°F.**

8. Tower Cold Water Temperature:
CWT2 = CWT1 + R1 - R2 = 89 + 15 - 15.625 = **88.375°F.**

This relation is obtained from the below derivations:

Heat Load, **D1** = L1 x R1 = L1 x (HWT1 - CWT1);
Heat Load, **D2** = L2 x R2 = L2 x (HWT1 - CWT2).

From the relation of **D1 = D2**,

L1 x (HWT1 - CWT1) = L2 x (HWT1 - CWT2);
L1 / L2 x (HWT1 - CWT1) = HWT1 - CWT2.

Therefore:

CWT2 = HWT1 - L1 / L2 x (HWT1 - CWT1) =
HWT1 - L1 / [L1 x (1 - % By-Pass Wall Water / 100) x (HWT1 - CWT1] =
HWT1 - 1 / (1 - % By-Pass Wall Water / 100) x (HWT1 - CWT1) = HWT1 - R2 =
[(HWT1 - CWT1) / (1 - % By-Pass Wall Water / 100) = R2] = CWT1 + R1 - R2 =

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

HWT = CWT1 + R1 = CWT2 + R2 =
CWT2 = CWT1 + R1 - R2 =

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, G2 = G1 = **80,848 lb/min**,
Liquid to Gas Ratio, L/G2 = L2 / G2 = 128,000.0 / 80,848 = **1.5832**

Example. 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:

1. Range:
R1 = tw2 - tw1 = 104 - 89 = **15.0°F.**

2. Tower Water Flow Rate:
L1 = Water Flow Rate x (500 / 60) = 16,000 x (500 / 60) = **133,333.3 lb/min.**

3. Liquid to Gas Ratio:
L/G1 = L1 / G1 = 133,333.3 / 80,848 = **1.6492.**

4. Air Mass Flow Rate:
G1 = G2 = 80,848 lb/min.

5. Tower Water Flow Rate:
L2 = 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.**

6. Liquid to Gas Ratio:
L/G2 = L2 / G2 = 160,000.0 / 80,848 = **1.9790.**

7. Range:
R2 = (tw2 - tw1) / (1 - % By-Pass Water / 100) = (104 - 89) / 0.96 = **15.625°F.**

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)$^{0.8}$ = **2.21825.**

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

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

9.2. New CWT through Tower:

WBT + New Approach + Design Range - Actual Range =
80 + 12.331 + 15 - 15.625 = **91.706**

9.3. Final CWT:

1. (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**

2. Water Flow Rate through Tower:
Alternative Water Flow x (1 - % By-Pass) = 20,000 x (1 - 0.04) = **19,200 GPM**

3. By-Pass Wall Water Flow:
Alternative Water Flow x % By-Pass = 20,000 x 0.04 = **800 GPM**

9.4. Final HWT:

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

Or,

2. New CWT through Tower + New Range through Tower = 91.706 + 15.625 = **107.331°F**

10. 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).

Most of air pressure drops at all the areas (excepting fill section) can be easily calculated as per the well-known formula shown below:

Pressure Drops = 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.

10.1. Density ratio: Is an actual air density divided by **0.075 lb/ft³ @ 70°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 indicated 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.**

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.** Anyway, 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, but it is in the range of **0.1 to 0.3**.

10.2. Pressure Drop: Is commonly determined by the following formula:

$$\textbf{Pressure Drop} = \mathbf{k.}\ \frac{\mathbf{p.v^2}}{\mathbf{2.g}} =$$

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 Water) =

or, with all conversions defined as:

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

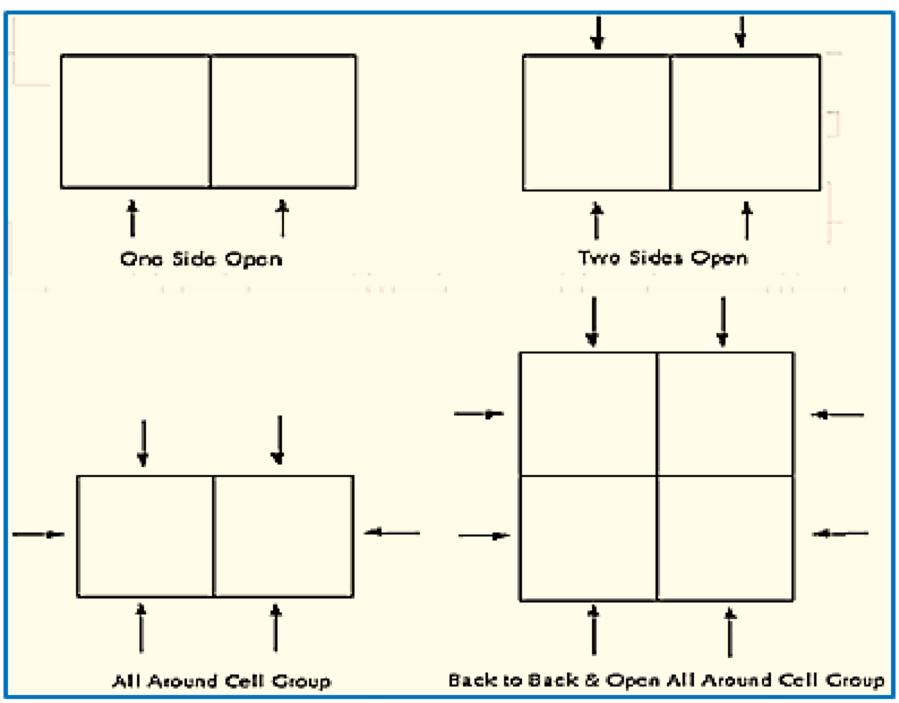
Where:

k = Pressure Drop Coefficient;
r = Air Density, lb/ft³;
V = Air Velocity, ft/min.;
G = Acceleration Gravity, ft/min² (1g = 32.172 ft/sec² = 115,820 ft/min²);
Density Ratio = Actual Air Density / 0.075 (1 lb/ft² = 0.1922 inch WG).

Note: 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**.

11. 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 Side Open & Ends Closed**: This arrangement is most general for the industrial cooling towers.
3) **All Around Cell Group Back To Back & Open All Round**: This is useful for a case where the area is limited.



Example. 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:

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) =

% By-Pass Water Calculation is as follows:

1) Water Flow Nozzle = 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 or 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 (%) =
% By-Pass Water = (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 following equation:

ha2 = ha1 + L/G x New Tower Range =

Then:

L/G = (ha2 - ha1) / 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**

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) =
L = 12,500 x (500 / 60) x (1 - 3.265 / 100) / 1.4760 = **68,271 lb/min.**

Remember: **(500 / 60)** - is a constant to convert water flow rate in **GPM to lb/min unit**.

Then, 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** 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.

1. **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 ft²**

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 ft²**

3. **Total Area of Obstructions:**

Total Obstruction Area = 70 + 56 = **126 ft²**

4. **Overall Area of Air Inlet**:

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

5. **Percent 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 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** = **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 ft³/lb**

1. **Air Mass Flow Volume @ Air Inlet**:

Air Mass Flow x Specific Volume @Air Inlet:
68,271 x 14.2230= **971,018 ft³/min.**

Compare this value with above airflow volume:

2. **Air Velocity @ Air Inlet**:

Airflow Volume @ Air Inlet / Net Area of Air Inlet:
971,018 / 1,134 = **856.27 ft/min (FPM)**

Air Density @ 85.24 DBT & 80% RH = **0.0718 lb/ft³**

Pressure Drop Coefficient for this arrangement = **2.5**

Then, pressure drop is obtained from below:

11.1. 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 Water.**

Obs.: For reference, the air density at the given wet bulb temperature is **0.0724 lb/ft³**. Compare this with the previous value of air density.

Example. Determine the pressure drop at the fill for the same example above.

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.

1. **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 = 14.5709 x 68,271 = **994,769 ft^3/min.**

2. **Average Air Velocity:**

a) Average Air Volume / Net Fill Area Net Fill Area:

(Cell Length x Cell Width) x (1 - % Fill Obstruction / 100) % Fill Obstruction:

Note: Safety margin for wood tower is about = **1.11%**

Therefore, the net fill area = (42 x 42) x (1 - 1.11 / 100) = **1,730.7 ft^2**

b) Average Air Velocity @Fill:

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

Thus, the water loading calculation is required as follows:

3. **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 = **0.0718 lb/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.**

Obs.: 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.

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

Example. Determine the pressure drop at the drift eliminator per given conditions in example above.

Solution:

In general, the obstruction area in the drift eliminator is considered same as the fill obstruction area:

Net drift eliminator area = (42 x 42) x (1 - 1.11 / 100) = **1730.7 ft².**

There is no change in the air mass flow throughout the cooling tower.

Therefore, the value of air mass flow is same as the above = **68,271 lb/min**.

The air density and specific volume at 97°F 100% RH are **0.0696 lb/ft³** = **14.9362 ft³/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 ft³/min.**

11.2. Air Velocity @ Drift Eliminator:

Airflow Volume @ Drift Eliminator / Net Area of Drift Eliminator

Air Velocity @ Drift Eliminator = **589 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:

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

Example. 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).**

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.

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

Note: 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 table could be **applied to the cooling tower fan stack** as a guide line in choosing the pressure drop coefficient.

(Solution):

Let's calculate the net fan area:

Fan Net Area = 3.1416 / 4 x (Fan Dia² -Air Seal Disk²) = **573.52 ft².**

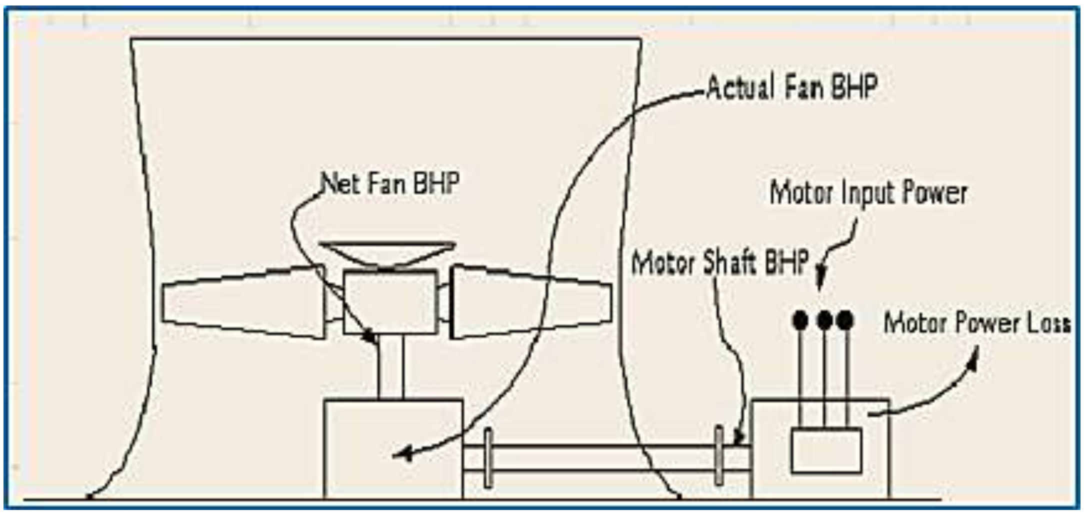
Air Velocity @ Fan:
Airflow Volume @ Fan / Net Fan Area = 101,971 / 573.52 = **1778 ft/min.**

Note: The air volume at fan is same as the air volume at the drift eliminator. Then, pressure drop is obtained from below:

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.**

12. MOTOR POWER SIZING:

The fan BHP 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.



The influence on the fan performance as follows:

1. Influence of Fan Inlet Shapes: 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.

3. The smaller of the ratio of distance and the larger of the ratio of area, the higher the resistance correction factor. In most cases, the additional pressure **drop coefficien**t due to the obstacles is within **0.1 to 0.15**.

4. 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.

Obs.: The additional **static pressure,** which increases due to obstacles could be obtained as adding the pressure drop factor. The influence of fan performance by the tip clearances could be achieved as adjusting the power transmission efficiency.

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. Considering data above and air volume = 101,971 ACFM, determine the fan brake horse-power and fan static efficiency for the design conditions with the following data:

Fan total efficiency = **80 %;**
Total Static Pressure = **0.4793 inch Water;**
Velocity Pressure = **0.1825 inch Water;**
Velocity Recovery = **0.0178 inch Water.**

Solution:

Fan BHP = Air Volume ACFM x Total Pressure in inch Water / (Fan Total Efficiency x 6356) =
Or, Air Volume ACFM x Static Pressure in inch Water / (Fan Static Efficiency x 6356) =

Total Static Pressure (considering the above data)**:**
PD @Air Inlet + PD @Fill + PD @Drift Eliminator + PD @Fan Inlet =
0.1092 + 0.3011 + 0.0361 + 0.0329 = **0.4793 inch Water**

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

Fan BHP = 101,971 x 0.6439 / (0.8 x 6356) = ~**13 BHP.** Consider **15 HP** as next size.

Fan Static Efficiency:
Air Volume @ Fan in ACFM x Static Pressure in inch Water / (Fan BHP x 6356) =
101,971 x (0.4793 - 0.0178) / (13 x 6356) = **57%**

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 program, since the efficiency of fan stack used by the fan makers is different each other.

Example. Determine the motor input power based on the example above.

Solution:

Actual Fan BHP:
Net Fan BHP / System Environmental Correction Factor =
13 / 0.85 = ~**15 BHP**

Motor Shaft BHP:
Actual Fan BHP / Efficiency of Power Transmission of Gear Reducer =
15 / 0.75= **20 BHP**

Obs.: The gear reducers waste **5 to 15%** of motor power, which depends on the number of reduction. These losses shall be turned to 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.

Example: Determine the rated motor power for motor shaft BHP = 140 and efficiency = 0.85.

Solution:

140 / 0.85 (Motor Efficiency: 85%) = ~**165 BHP**

Minimum Motor Power:
Motor Shaft BHP x Motor Minimum Margin x Operation Safety =
140 x 1.2 x 1.05 = **176 HP.** Consider = **~ 180 HP**

Obs.: The next available size of **motor power is 200 HP**. Note that the motor minimum margin depends on the type of cooling tower operation and ambient conditions.

13. WATER EVAPORATION:

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 wate**r. Thus, to evaporate a gallon of water, **8.3 x 1,000 or 8,300 Btu** is required. 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.

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, as the water is cooled by evaporative transfer in cooling towers. 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. Determine the evaporation loss in a percentage for the previous example above.

Solution:

13.1. 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%.**

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:

L2/G = {(ha2 - ha1) - (tw1 -32) x (w2 - w1)} / (tw2 - tw1) (tw2 - tw1 = Actual Range).
Air Enthalpy at Exit (97°F) = **66.5773 Btu/lb;**

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

Then, **L2/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/ (L2/G) x 100 = **1.28%**

Example. Determine the heat removal in the percentage by the evaporation for the example above.

Solution:

Evaporation Rate: (w2 - w1) x Latent Heat of Water / (Enthalpy @ Exit - Enthalpy @ Inlet)

Latent Heat of Water = ~ **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%**

Example. Determine the rate of heat removal by to the evaporation under the assumption that the **L/G ratio** was changed to **1.6** for the initial conditions of example above.

Solution:

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°F**

Absolute Humidity @ Tower Exit = **0.040639**
Absolute Humidity @ Tower Inlet = **0.021117**

Evaporation rate: (0.040639 - 0.021117) x 1040 / (68.5008 - 43.6907) x 100 (%) = **82%**

Example. Determine the rate of heat removal due to the evaporation under the assumption that **RH** was changed **from 80% to 60%** for the above example.

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**

Evaporation rate: (0.039167 - 0.019563) x 1040 / (66.5780 - 43.6907) x 100 (%) = **89%**

13.2. 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. Determine the cold water temperature for the following conditions.

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 net fan power is determined from the relation of - Motor HP x (1 - Motor Margin) x Power Transmission

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

13.3. 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.

It has 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)**. Then, the air mass flow rate and L/G Ratio can be obtained as below:

13.4. Air Mass Flow Rate:

Air Volume @ Fan / Specific Volume @ Fan =

The L/G Ratio:

Flow Rate in GPM:
Water through Tower x (500/60) / (Air Volume @ Fan / Specific Volume @ Fan) =
First Step: Design Water Flow Rate x (1 - % By-Pass Water) =

In order to obtain the specific volume the tower exit temperature should be calculated first. The exit temperature requires **L/G = (ha2 = ha1 + L/G x Range)** - ratio as the exit air enthalpy.

This is a summation of tower inlet air enthalpy and **L/G x cooling range**. The total static pressure is a summation of pressure drops obtained from each location.

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

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

Actual Cold Water Temperature:
Wet Bulb Temperature + Approach = 80.00 + 8.633 = **88.63°F**

1. 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 =

2. Also, the enthalpies for air side were based on the following:

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. Determine the L/G ratio and cold water temperature when the wet bulb temperature was downed to 70°F from design conditions described in the example above.

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) =

13.5. 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)

First, 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**

Second, 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**

13.6. 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. 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. 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 conditions given.

Solution:

First Step: Find a dry bulb temperature at the tower inlet for **80% of relative humidity** corresponding **80°F of wet 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 above.

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 value of Net Fan Power must equal the fan BHP varying air volume at the fan.

Net Fan Power = Fan BHP =

Motor HP x (1 - Motor Margin) x Power Transmission Efficiency = Air Volume @ Fan x Total Static Pressure / (6356 x Fan Efficiency). Then, the **L/G ratio** is obtained from below relations:

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 =

The L/G Ratio = Water Flow Rate in Lb/Min / Air Mass Flow Rate (lb/min):

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 = 110,836.7 / 69,283.3 = **1.59976**

14. DETERMINATION OF PUMPING HEAD:

For dimensioning of the pumping head the designer must consider the static lift (D) and vertical distance (h) of the pressure gauge above its base.

Example: 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 is **24** inch and inside 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;**
Welding Elbow 24", 90° (r/d = 1);
Equivalent Length = **37.7 ft** (from table of friction loss in term of length).

Total Equivalent Length = 30 + 5 + 37.7 = **72.7 ft** (based on 24 inch pipe).

2) 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 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 2.309 - (35 - 5) - 0.95 = **26.78 ft**

Note: 1 psi = 2.309 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] =
14,000 x 0.1336798 / [0.7854 x (22.624 / 12)^2] = 670.39 ft/min = **11.17 ft/sec**

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 =

Correcting 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**

Correcting the Pumping Head:

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

LINKS & REFERENCES:

ASHRAE: The American Society of Heating Air Conditioning and Refrigeration:

- 2001 ASHRAE Handbook of Fundamentals;
- 1997 ASHRAE Handbook of Fundamentals;
- ASHRAE Cooling and Heating Load Calculation Manual.

NIST Standard Reference Database 69: NIST Chemistry Web Book;

1. ASHRAE: 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.

http://www.smacna.org/
http://www.engineeringtoolbox.com/sizing-ducts-d_207.html
www.engineeringtoolbox.com/air-psychrometrics-properties-t_8.html
www.linric.com/webpsy.htm
http://www.flycarpet.net

