I am in charge of reducing our carbon footprint at W A C.
THE PROPERTIES WE WILL STUDY

1. DRY BULB TEMPERATURE
2. VAPOR PRESSURE
   (& THE SATURATION CURVE)
3. SPECIFIC VOLUME
4. HUMIDITY RATIO
5. RELATIVE HUMIDITY
6. DEW POINT TEMPERATURE
7. WET BULB TEMPERATURE
8. SPECIFIC ENTHALPY

WB & Other Useful Stuff That You Can Use This Afternoon
THE WET BULB – FORMAL (USELESS) DEFINITION

• THERMODYNAMIC WET-BULB TEMPERATURE $T^*$ IS THE TEMPERATURE AT WHICH WATER (LIQUID OR SOLID), BY EVAPORATING INTO MOIST AIR AT DRY-BULB TEMPERATURE $T$ AND HUMIDITY RATIO $W$, CAN BRING AIR TO SATURATION ADIABATICALLY AT THE SAME TEMPERATURE $T^*$ WHILE TOTAL PRESSURE $P$ IS CONSTANT.

• ASHRAE FUNDAMENTALS 2013
• THERMODYNAMIC WET-BULB TEMPERATURE T* IS THE TEMPERATURE AT WHICH WATER (LIQUID OR SOLID), BY EVAPORATING INTO MOIST AIR UNDER DRY-BULB TEMPERATURE T AND HUMIDITY RATIO W, CAN BRING AIR TO SATURATION ADIABATICALLY AT THE SAME TEMPERATURE T* WHILE TOTAL PRESSURE P IS CONSTANT.

• ASHRAE FUNDAMENTALS 2013
Therefore, if the process is strictly adiabatic, conservation of enthalpy at constant total pressure requires that:

\[ h + (W_s^* - W)h_w^* = h_s^* \]  

Equation (33)

\( W_s^* \), \( h_w^* \), and \( h_s^* \) are functions only of temperature \( t^* \) for a fixed value of pressure. The value of \( t^* \) that satisfies Equation (33) for given values of \( h \), \( W \), and \( p \) is the thermodynamic wet-bulb temperature.

\[ W = \frac{(1093 - 0.556t^*)W_s^* - 0.240(t - t^*)}{1093 + 0.444t - t^*} \]  

Equation (35)
Therefore, if the process is strictly adiabatic, conservation of enthalpy at constant total pressure requires that

\[ h + (W_s - W)h^*_w = h^* \]

\[ W_s^*, h_w^*, \text{ and } h_s^* \text{ are functions only of temperature at a fixed value of pressure. The value of } t^* \text{ that satisfies Equation (33) for given values of } h, W, \text{ and } p \text{ is the thermodynamic wet bulb temperature.} \]

\[ W = \frac{(1093 - 0.5t^*)W_s^* - 0.24t^*}{1093 + 0.444t - t^*} \]
• For our discussion, consider Enthalpy and Wet Bulb lines parallel – (they are not).
• Enthalpy Deviation. Will talk about this a little later.
• In the mean time, don’t mess with Deviation – use Psych Property Calculator.
• ASHRAE chart has separate Enthalpy and WB lines.
• 80°F DB/67°F is special because a lot of AHRI Standards for Unitary Equipment use this as the coil entering condition.
• At saturation DB, WB and DP are all the equal.
• Psych Charts are going away. If you are like me and still want stuff plotted on psych charts, then one good way is to plot the process on the psych chart but get all values from your smart phone or some other psych calc app. The one shown is free.
SO WHAT IS WET BULB?
SO WHAT IS WET BULB?

• AS A HVAC PROJECT MANAGER YOU JUST NEED TO KNOW 2 THINGS ABOUT THE WET BULB CONCEPT:

• FIRST:
  • IT IS A PROPERTY OF AIR WHICH TELLS US HOW MUCH IT CAN COOL WATER WHEN BOTH ARE BROUGHT IN INTIMATE CONTACT.

• AND SECOND:
  • THE WET BULB IN THE HVAC WORLD IS OFTEN USED AS A PROXY FOR ENTHALPY, (THE SO CALLED “TOTAL HEAT CONTENT”).

THAT IS IT. NO MORE - NO LESS!
Let us take a bowl of water
Put a fan near it
and blow air on the surface

Pathetic scientific set-up
BUT
It explains the concept of Wet Bulb well

Water will evaporate
Water will cool
It should get pretty close to the WB of the AIR

Now PRETEND there is no radiation, convection or conduction gain or loss with respect to the surroundings (adiabatic)

Water will drop to the WB of air and then stop dropping further

It is as if has hit an invisible low "floor"
Let us take another example:
Change the ambient air conditions.
Same thing happens … but now a new "floor" value pops up below which the water will not cool - this time 78°F

**NOTE:** IT IS NOT BECAUSE THE EVAPORATION STOPS – THE WATER IS STILL EVAPORATING AND WILL KEEP EVAPORATING TILL THE BOWL IS EMPTY

IT IS JUST THAT WITH THAT PARTICULAR STATE OF AMBIENT AIR (100°F & 41%RH) YOU CAN NEVER MAKE WATER COLDER THAN 78°F

What happens if I put a bigger fan with more CFM ???
… I will evaporate the water FASTER but it will not get any COOLER!

**THE WET BULB IS NATURE'S LOW LIMIT!**
"The Wet Bulb temperature is NOT, strictly speaking, a property of air, but it is a measurement of what the air can do to water when in intimate contact."

Now We Know What Happens To The Water

What Happens To The Air?

That Depends On How Efficient And Thorough The Air/Water Mixing Process
A LITTLE MORE FORMAL SET-UP

- Hypothetical 100% efficient Process.
- Perfectly insulated. Adiabatic – no heat in or out.
- WB and AST difference
- Thermodynamacists tell us that it is just a coincidence that they are the same for water. For e.g. if we were running 100 Proof Vodka through this apparatus the 2 temperatures may not be equal.
- We will ignore this useless information. Both are the same value for our HVAC world.
- Near 100 % efficiency is not totally out of reach. Good quality commercial pads can get within a few points of 100%.

**Diagram:**
- Insulated Duct
- Entering
- Leaving
- Wet Bulb Depression: 95°F – 73°F = 22°F
- Adiabatic Saturation Temperature

**Data Points:**
- **Entering**
  - Dry Bulb = 95°F
  - Wet Bulb = 73°F
  - RH % = 35.15%
- **Leaving**
  - Dry Bulb = 73°F
  - Wet Bulb = 73°F
  - RH % = 100%

**Make-up Water at 73°F**
MEASURING WET BULB?

- SLING PSYCHROMETER

- ASPIRATING PSYCHROMETER
Why and how does nature set such a definite low limit on how far we can cool the water?

At WTF Institute of Higher Learning our authoritative answer is: "Because it is the Wet Bulb. Now go sell some more jobs!"

It is not important to understand this, but there are always one or two perverts in the group that will lie awake at night, wondering why that is so! I would like to help them out. So in the next three slides we will take a look and see if we point them in the right direction.
SOME KEY FACTS TO REMEMBER BEFORE GETTING TECHNICAL ABOUT THIS

1. WATER WILL EVAPORATE AS LONG AS THE VAPOR PRESSURE OF WATER IS GREATER THAN THE VAPOR PRESSURE OF AMBIENT AIR

2. EVAPORATION TAKES ENERGY. FOR EXAMPLE 100°F WATER HAS A LATENT HEAT OF 1037 BTU/LB. THIS SIMPLY MEANS THAT IF EVAPORATION IS TAKING PLACE THEN "SOMETHING" IS PROVIDING THIS LATENT HEAT OF VAPORIZATION. (LIKE OUT OF A SWIMMING POOL YOUR SKIN IS PROVIDING THE HEAT OF VAPORIZATION, THEREFORE THE SKIN GETS COLD.)

3. AND FINALLY, JUST LIKE THE "SHIT FLOWS DOWN-HILL" PLUMBING AXIOM, WATER CAN ONLY TRANSFER HEAT TO MOIST AIR IF THE ENTHALPY OF WATER IS GREATER THAN THE ENTHALPY OF MOIST AIR. THERMODYNAMICISTS CALL THIS THE "ENTHALPY POTENTIAL". IT APPLIES TO ALL SITUATIONS WHERE UNSATURATED AIR COMES IN CONTACT WITH A WET SURFACE.
A DROP OF WATER FOLLOWING THROUGH BLOWING AIR

Let us follow a drop of water as it falls through air.
- The water temperature is 100°F.
- The air is at 95°F and 35% RH.
- We will plot this on a PSYCH chart in the next slide, but for the moment let us note a few thing:
  - The Vapor Pressure of water is higher than the Vapor Pressure of the air. The water will evaporate.
  - Evaporation takes energy. At 100°F roughly 1035 BTU/LB.

Where will this heat (energy) come from???
- At "A" mostly from water to cool itself, its temperature is higher than the air DB temperature.
- At "B" some from air by cooling itself. (Air is hotter than water now.) And some from water as it cools itself again. (Note: The source of energy cools and drops in temperature.)
- And so it goes ... water using part of its own heat to evaporate and cool itself and some energy coming from air as it cools itself.
- At "C" we finally see the mystery of the WET BULB revealed. The ENTHALPY of water at "C" is equal to the ENTHALPY of air.
- Water cannot provide any more heat. There is no "Enthalpy Potential" left. So it cannot cool itself.
Water is always shown on the saturation line on the psych chart.

All points A, B, C and D have higher Vapor Pressures than air and therefore evaporation will take place.

At A:
Some energy for evaporation comes from air, and some from water itself and therefore water temperature drops.

At B:
Some energy for evaporation comes from air, (but more than it did in A) and some from water itself (Less than it did at A) and therefore water temperature drops again.

At C:
Same as at B except that the air picks up an even higher share of the evaporation energy required.

At D:
The ENTHALPY of water is the same as the ENTHALPY of moist air (zero Enthalpy Potential) and therefore no energy for evaporation can come from the water itself, which in turn means that it cannot drop in temperature. The air is also at the same Enthalpy, but it can play a game where it gives up the exact amount of sensible heat that it gains in evaporation latent heat. The system then moves towards the saturation line as a constant Enthalpy process.

If the air is allowed to get to D (100% saturation) the Vapor pressure will be the same for water and air and the evaporation will stop.
EVAPORATIVE COOLERS
DIRECT EVAPORATIVE COOLING

110°F DB  12%RH
70°F Wet Bulb

WB Depression = 110 – 70 = 40°F

SUPPLY CONDITION:
LVG DB = 110 – (0.9 x 40) = 74°F Dry Bulb
70°F Wet Bulb
82%RH

90% Efficient Evaporative Cooler

Roughly ASHRAE
Palm Springs Airport
Example of a 90% Efficient Process:

Ambient 110°F Dry Bulb 70°F Wet Bulb
Supply 74°F DB  70°F WB

Formula for PAD Efficiency:

\[
\text{Eff.} = \frac{\text{Actual Dry Bulb Temp. Drop}}{\text{Wet Bulb Depression}}
\]

A 12" deep commercial PAD can easily deliver 90% efficiency when new. Maintenance is another story.
# Quickie - WET BULB BIN ANALYSIS

## SAN BERNARDINO CITY HALL

### EVAPORATIVE COOLING PADS - ENERGY SAVINGS

Prepared by: Western Allied Service Co.  [Sept, 2, 1994]

<table>
<thead>
<tr>
<th>BIN DRY BULB</th>
<th>BIN WET BULB</th>
<th>6am - 7pm YEARLY OCCUR.</th>
<th>AIR FLOW</th>
<th>AIR ENT. EVAP.</th>
<th>AIR LVG. EVAP</th>
<th>SENS. CLG. CAPACITY</th>
<th>DEMAND SAVINGS</th>
<th>ENERGY SAVINGS</th>
<th>COST SAVINGS</th>
</tr>
</thead>
<tbody>
<tr>
<td>TEMP. F</td>
<td>TEMP. F</td>
<td>HOURS</td>
<td>CFM</td>
<td>TEMP. F</td>
<td>TEMP. F</td>
<td>TONS</td>
<td>KW</td>
<td>KWh</td>
<td>$</td>
</tr>
<tr>
<td>115-119</td>
<td>110-114</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>105-109</td>
<td>72.0 F</td>
<td>90</td>
<td>40,000</td>
<td>84.0 F</td>
<td>73.2 F</td>
<td>39.6</td>
<td>47.5 kw</td>
<td>4,277 kwh</td>
<td>$342</td>
</tr>
<tr>
<td>100-104</td>
<td>70.0 F</td>
<td>540</td>
<td>40,000</td>
<td>84.0 F</td>
<td>71.4 F</td>
<td>46.2</td>
<td>55.4 kw</td>
<td>29,938 kwh</td>
<td>$2,395</td>
</tr>
<tr>
<td>95-99</td>
<td>67.0 F</td>
<td>420</td>
<td>40,000</td>
<td>84.0 F</td>
<td>68.7 F</td>
<td>56.1</td>
<td>67.3 kw</td>
<td>28,274 kwh</td>
<td>$2,262</td>
</tr>
<tr>
<td>90-94</td>
<td>65.0 F</td>
<td>360</td>
<td>40,000</td>
<td>84.0 F</td>
<td>68.9 F</td>
<td>62.7</td>
<td>75.2 kw</td>
<td>27,086 kwh</td>
<td>$2,167</td>
</tr>
<tr>
<td>85-80</td>
<td>64.0 F</td>
<td>330</td>
<td>40,000</td>
<td>84.0 F</td>
<td>66.0 F</td>
<td>66</td>
<td>79.2 kw</td>
<td>26,136 kwh</td>
<td>$2,091</td>
</tr>
<tr>
<td>80-84</td>
<td>58.0 F</td>
<td>660</td>
<td>40,000</td>
<td>84.0 F</td>
<td>60.6 F</td>
<td>85.8</td>
<td>103.0 kw</td>
<td>67,954 kwh</td>
<td>$5,436</td>
</tr>
<tr>
<td>75-70</td>
<td>57.0 F</td>
<td>420</td>
<td>40,000</td>
<td>84.0 F</td>
<td>59.7 F</td>
<td>89.1</td>
<td>106.9 kw</td>
<td>44,906 kwh</td>
<td>$3,593</td>
</tr>
<tr>
<td>70-74</td>
<td>56.0 F</td>
<td>300</td>
<td>40,000</td>
<td>84.0 F</td>
<td>58.8 F</td>
<td>92.4</td>
<td>110.9 kw</td>
<td>33,264 kwh</td>
<td>$2,661</td>
</tr>
<tr>
<td>65-69</td>
<td>56.0 F</td>
<td>540</td>
<td>40,000</td>
<td>84.0 F</td>
<td>58.9 F</td>
<td>92.4</td>
<td>110.9 kw</td>
<td>59,875 kwh</td>
<td>$4,790</td>
</tr>
<tr>
<td>60-64</td>
<td>54.0 F</td>
<td>420</td>
<td>40,000</td>
<td>84.0 F</td>
<td>57.0 F</td>
<td>99</td>
<td>118.8 kw</td>
<td>49,896 kwh</td>
<td>$3,992</td>
</tr>
<tr>
<td>55-59</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>50-54</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>$29,729</td>
</tr>
<tr>
<td>45-49</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>40-44</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>35-39</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>30-35</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Note: Mike Taylor has the CD and can get you the Bin Data for any Station.
INDIRECT EVAPORATIVE COOLING

Typical Efficiency for a good commercial unit = 70%
COOLING TOWERS
COOLING TOWER TEMPERATURES NOMENCLATURE

Tower Entering Water Temperature °F
Tower Leaving Water Temperature °F

RANGE
73°F
80°F
94°F

APPROACH

Note: there is a difference between Ambient Air Wet Bulb and Tower Entering Wet bulb.
For Design-Build Tower Selection use ASHRAE 0.4% WB. (Just my personal preference.)
COOLING TOWER SELECTION - APPROACH

SOME COMMON NUMBERS FOR CENTRIFUGAL CHILLER PLANTS

WET BULB + APPROACH = TOWER LEAVING WATER TEMPERATURE

<table>
<thead>
<tr>
<th>Wet Bulb</th>
<th>Approach</th>
<th>Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>5°F</td>
<td></td>
<td>5°F</td>
</tr>
<tr>
<td>6°F</td>
<td></td>
<td>6°F</td>
</tr>
<tr>
<td>7°F</td>
<td></td>
<td>7°F</td>
</tr>
<tr>
<td>8°F</td>
<td></td>
<td>8°F</td>
</tr>
<tr>
<td>9°F</td>
<td></td>
<td>9°F</td>
</tr>
<tr>
<td>10°F</td>
<td></td>
<td>10°F</td>
</tr>
<tr>
<td>11°F</td>
<td></td>
<td>11°F</td>
</tr>
<tr>
<td>12°F</td>
<td></td>
<td>12°F</td>
</tr>
</tbody>
</table>

Don't go here!

Design-Build Work:
- Don't be vague in your specification of the project "Design Conditions".
- Attach ASHRAE Data (Previous Slide) as part of your Proposal.
- Not only does it make it clear to the Customer, but also helps others on the team. Good examples are the Controls Programmer and the Start-Up/Troubleshooting technician.
- Taylor has the ASHRAE Weather CD and can print a hard copy for any station.
COOLING TOWER SELECTION - RANGE

SOME COMMON NUMBERS FOR CENTRIFUGAL CHILLER PLANTS

RANGE = TOWER ENT. WATER TEMP - TOWER LVG. WATER TEMP

- 15°F
- 14°F
- 13°F
- 12°F
- 11°F
- 10°F
- 9°F
- 8°F

3 Secrets of Chiller Plant Value Engineering:

1. Pump Less Water
2. Pump Less Water
3. Pump Less Water

ENT. WATER °F

RANGE or Tower Water ΔT
LOAD
RANGE
APPROACH
Materials of Construction
Discussion Points:

- General shape of curve is very different than that of swamp coolers – remember we are pumping heat into the water
- Air could be cooler or warmer
- Air mostly saturated at tower discharge
- Plume formation. Next Slide.
• Takes a lot of heat to move the curve to the right! Don’t make casual promises to the customer about plume abatement. The cost will shock you.

• Heat either leaving or entering air

• Potential heat recovery application?

• In normal commercial buildings, unless it is a real problem leave it alone

Any time the line joining the discharge air condition of the tower and the ambient air condition passes to the left of the saturation line … a plume will form on the outlet.
The most basic formula for heat flow into or out of water is:

\[ \text{BTU} = \text{MASS} \times \text{Sp. Heat} \times \Delta T \]

Where Sp.Heat of water = 1 BTU/Lb °F and Mass is in Lbm

So

\[ \text{BTU} = \text{Mass} \times \Delta T \]

Converting to a rate equation – divide both sides by Time (Hr)

\[ \text{BTU/Hr} = \text{LB/Hr} \times \Delta T \]

Converting Pounds to Gallons and Hours to Minutes

\[ \text{BTUH} = 500 \times \text{GPM} \times \Delta T \]

Dividing both sides by 12000 and rearranging:

\[ \text{GPM} = \frac{\text{Tons} \times 24}{\Delta T} \] (Must Know Equation!)

So a \( \Delta T \) of 10 gives 2.4 gpm per ton and a \( \Delta T \) of 16 gives 1.5 gpm per ton
Couple of notes:

• You will see some books and articles refer to a "Condenser TON". It is equal to 15,000 BTUH. DON"T use this term. It is really VERY inaccurate these days and can cause a lot of unnecessary confusion.

• For VERY accurate work you need to include condenser pump horsepower for CDW flow and tower sizing. Roughly every 5 hp of pump motor adds a Ton to the cooling tower load. Usually insignificant. Use only if very large HP CDW Pump.
We know the Evaporator load in TONs. That is the chiller design tonnage.

What about the Compressor load?

We have the KW/Ton (or COP).

If not from actual selection then use Title 24 maximums. For e.g. a 500 ton centrifugal in California must do better than 0.576 kW/ton (6.10 COP).

Since we know the chiller tons we can find the compressor kw by multiplication with above KW/Ton

Compr. kW max = 0.576 x 500 = 288 kW

1 Ton = 3.517 kW

Therefore this compressor adds:

288/3.517 = 82 Tons

Total tower load is 500 + 82 = 582 Tons

Correct Condenser GPM = 582 x 24/\Delta T
SIDE NOTE OF INTEREST:

Where did the 3 gpm per ton come from?

For a range of 95°F > 85°F (a 10°F ∆T) at 3 gpm equates to:

- $\text{BTUH} = \text{gpm} \times 500 \times \Delta T$
- $\text{BTUH} = 3 \times 500 \times 10 = 15000 \text{ BTUH}$ .... The so called "Condenser Ton"

Now we know that 12000 BTUH equals one Ton of cooling

So that leaves $15000 - 12000 = 3000 \text{ BTUH}$ for the COMPRESSOR HEAT per Ton of cooling

$3000\text{BTUH} \div 3412 \text{ BTUH/kW} = 0.88 \text{ kW per Ton of cooling}$

This was probably OK in the 60’s and 70’s but is way out of line today
Supposing you have a 500 Ton Tower shown on the Drawings and Equipment Schedule. Let us also assume that it is specified as a 0.50 kW/Ton machine. (Nothing special, and under the T-24 maximum.)

There is a good chance that it will show 95°F to 85°F @ 3 gpm per ton, which in this case will equal 500 x 3 = 1500 gpm. Assume 20 ft PD through the condenser barrel.

- Pipe size = 10” (6.1 ft/sec, 1.07’/100ft PD)
- Pump BHP @ 50’TDH and 75% eff. = 25.25 bhp (Probably a 30 hp motor)

Now we know this flow is not correct. (Previous slides.) \[ (500 + (500 \times 0.50)/3.517) \times (24/10) = 1,370 \text{ gpm} \]

- Pipe size = 8” (8.7 ft/sec, 2.83’/100ft PD)
- Pump BHP @ 40’TDH and 75% eff. = 18.4 bhp (Probably a 20 hp motor)

We have done nothing yet but just correctly computed the condenser flow. What if you design the range as 14°F?

Condenser Flow = 979 gpm. In this particular case you still can't get to 6” pipe but the new bhp comes out to 11.5 bhp

This is just one example. Depending on the tonnage of the machine and ΔT you might get significant cost savings.
Lower flow rate means:

- Smaller pipe size
- Smaller pipe accessories, like valves, strainers flow control elements etc.
- Reduced insulation costs
- Smaller pumps
- Smaller pump Wiring and VFDs
- Smaller Structural pipe supports
- LESS INSTALLATION LABOR
A LITTLE ABOUT DX/WATER HEAT EXCHANGERS

Some Useful Observations:
(Centrifugal Compressors)

kW/Ton goes up (less efficient) as lift INCREASES. The compressor is doing more work. Lift can increase if Cond pressure goes UP or Evap Pressure goes DOWN or both.

[BTW: If you keep increasing "lift", then at a certain point the centrifugal compressor cannot create enough head and the refrigerant is forced backwards through the wheel. This is called SURGING. (Sounds like a very large sea lion in heat.)]

The compressor does not "see" the condenser FLOW RATE or the RANGE. It only "sees" the LMTD (Log Mean Temp. Diff.) between the Refrigerant SDT and the Condenser Water. Higher LMTD equals greater heat transfer from refrigerant to water.

As you know by now, at WTF we hate formulas (especially those with natural logarithms in them) and so just do this:

When Value Engineering RANGE options keep an eye on what you are doing to the AVERAGE condenser water temperature.

If your proposed range has a higher average water temperature than spec. design, then you are hurting the LMTD and the performance will suffer (and vice versa).

Example: Engineer specified 85 > 95, the average is 90°F
You want 85 > 100, average 92.5°F … you have reduced condenser capacity. BAD!
You want 80 > 94, average 87°F … you have increased the condenser capacity. GOOD!

(Note, that this is not an exact method but will always get you making the right decision.)
• Condenser Pump BHP = (gpm x TDH x 0.0002525)/Pump Efficiency

• Use 0.7 Pump Efficiency if no data is available.

• Example 1000 gpm @ 50' TDH

\[
\text{BHP} = 1000 \times 50 \times 0.0002525 \div 0.70 = 18 \text{ BHP}
\]

(20 HP Motor?)

Note: 0.0002525 is the reciprocal of 3960. (Whichever is easier for you to remember.)
<table>
<thead>
<tr>
<th>Equipment Type</th>
<th>Total System Heat Rejection Capacity at Rated Conditions</th>
<th>Subcategory or Rating Condition</th>
<th>Fan Motor HP Estimation</th>
<th>Test Procedure(s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Propeller or Axial Fan Open Circuit Cooling Towers</td>
<td>All</td>
<td>95°F entering water 85°F leaving water 75°F entering wet-bulb</td>
<td>≥38.2 gpm/hp</td>
<td>CTI ATC-105 and CTI STD-201</td>
</tr>
<tr>
<td>Centrifugal Fan Open Circuit Cooling Towers</td>
<td>All</td>
<td>95°F entering water 85°F leaving water 75°F entering wet-bulb</td>
<td>≥20.0 gpm/hp</td>
<td>CTI ATC-105 and CTI STD-201</td>
</tr>
<tr>
<td>Propeller or Axial Fan Closed Circuit Cooling Towers</td>
<td>All</td>
<td>102°F entering water 90°F leaving water 75°F entering wet-bulb</td>
<td>≥14.0 gpm/hp</td>
<td>CTI ATC-105 and CTI STD-201</td>
</tr>
<tr>
<td>Centrifugal Closed Circuit Cooling Towers</td>
<td>All</td>
<td>102°F entering water 90°F leaving water 75°F entering wet-bulb</td>
<td>≥7.0 gpm/hp</td>
<td>CTI ATC-105 and CTI STD-201</td>
</tr>
<tr>
<td>Air-Cooled Condensers</td>
<td>All</td>
<td>125°F condensing temperature R-22 test fluid 190°F entering gas temperature 15°F sub-cooling 95°F entering dry-bulb</td>
<td>≥176,000 Btu/h-hp</td>
<td>AHRI 460</td>
</tr>
</tbody>
</table>
THOUGHTS ABOUT THE “SAFETY FACTOR” IN CHILLED WATER PLANTS

• This is a good place to add a few words of advice on the safety factor for DESIGN-BUILD chilled water plants:

  • You only apply the Safety Factor **ONCE**. Do it to the load estimate.

  • As an example, your load comes out 450 tons and you decide to install a 500 ton system. That is it. Size everything for 500 tons without slop.
    • Don’t cheat to a higher than 500 ton chiller or tower
    • Size piping aggressively
    • Size pump Flow and Head aggressively *(Oversizing pumps should be made a criminal offence.)*
    • Etc. etc.
    • By the word "aggressive" above I mean, right on the nose. And if your equipment selection straddles two close sizes – take the lower one.

  • The typical Consulting Engineer starts with a padded load, then pads the equipment selection, then pads pump Head, then pads pipe sizes …. and on and on …. till no one really knows how much unnecessary cost is built in the project.
SO WHAT IS WET BULB? PART-2

• AS A HVAC PROJECT MANAGER YOU JUST NEED TO KNOW 2 THINGS ABOUT THE WET BULB CONCEPT:

• FIRST:
  • IT IS A PROPERTY OF AIR WHICH TELLS US HOW MUCH IT CAN COOL WATER WHEN BOTH ARE BROUGHT IN INTIMATE CONTACT.

• AND SECOND:
  • THE WET BULB IN THE HVAC WORLD IS OFTEN USED AS A PROXY FOR ENTHALPY, (THE SO CALLED “TOTAL HEAT CONTENT”).
This camel jockey, like a lot of our HVAC engineers, does not understand Psychrometry.
• **WET BULB AND ENTHALPY ARE SEPARATE CONCEPTS.**

**BUT**

• WET BULB LINES BEING PARALLEL TO ENTHALPY LINES (FOR ALL PRACTICAL PURPOSES) TRULY REPRESENT THE RELATIVE TOTAL HEAT CONTENT OF AN AIR STREAM.

• IF STREAM "A" HAS A HIGHER WET BULB THAN STREAM "B" THAN STREAM "A" HAS A HIGHER HEAT CONTENT OF THE TWO.

• MANY PEOPLE THINK THAT WET BULB SOMEHOW REPRESENTS THE LATENT HEAT, OR HEAT DUE TO MOISTURE ONLY … THIS IS ABSOLUTELY WRONG! WET BULB INDICATES BOTH SENSIBLE AND LATENT HEAT.

• SEE THE PSYCH CHART.

---

**Wet Bulb & Enthalpy Line**

L to R:
Reduce humidity and grains/lb moisture.
But if you add enough sensible heat, Wet Bulb will increase
R to L:
Note what happens in the opposite direction
On an ASHRAE 1% Design Day

- Let us compare bringing OSA into the occupied space at 2 locations:
  - CA: Palm Springs (Intl Airport) 109°F Dry Bulb - 72.2°F WB (Mean Coincident)
  - FL: Tampa (Intl Airport) 91.4°F Dry Bulb - 77.2°F WB (Mean Coincident)

Don’t let the Dry Bulb throw you off. The Wet Bulb tells the correct story. IT TAKES MUCH MORE ENERGY TO COOL A GIVEN AMOUNT OF OSA (on a design day) IN TAMPA FLORIDA THAN IN PALM SPRINGS CALIFORNIA.

Caution: While there is no question that the Enthalpy in Tampa is higher than PS on a Design Day, things get messy when you start taking into account the dry coil versus wet coil efficiency of a cooling coil. But that’s another story and another Fachgespräch.
Palm Springs:
\[ h = 35.7 \text{ BTU/LB}_{DA} \]
\[ \text{Room} = 27.57 \text{ BTU/LB}_{DA} \]
Tons/1,000 CFM OSA = 3 tons

Atlanta Intl:
\[ h = 40.64 \text{ BTU/LB}_{DA} \]
\[ \text{Room} = 27.57 \text{ BTU/LB}_{DA} \]
Tons/1,000 CFM OSA = 5 tons

\[ \text{Tons} = \text{CFM} \times 4.5 \times \Delta h \div 12,000 \]
A coil selection TABLE from the 70's

Note only entering air WB info required to select the right capacity Table

Note only WB info provided for Leaving air
TAKE HOME MESSAGE

• AS A HVAC PROJECT MANAGER YOU JUST NEED TO KNOW 2 THINGS ABOUT THE WET BULB CONCEPT:

• FIRST:
  • IT IS A PROPERTY OF AIR WHICH TELLS US HOW MUCH IT CAN COOL WATER WHEN BOTH ARE BROUGHT IN INTIMATE CONTACT.

• AND SECOND:
  • THE WET BULB (IN THE HVAC WORLD) IS OFTEN USED AS A PROXY FOR ENTHALPY, (THE SO CALLED “TOTAL HEAT CONTENT”).
THE END OF FACHGESPRÄCH – 5 PART-WB

See you soon for the **Enthalpy** Fachgespräch

You know, my Friends, with what a brave Carouse
I made a Second Marriage in my house;
Divorced old barren PSYCH CHART from my Bed,
And took the Daughter of the Vine to Spouse.

Omar Khayyam