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COOLING TOWERS WORK AS A SYSTEM

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Cooling Towers Work as a System

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There are two principal concepts that often confuse people who are not familiar with cooling towers and how they operate:

- 1) The cooling tower is part of a bigger system: it does not set the heat load.
- 2) The cooling tower and its components work as a system.

The first concept recognizes that the tower will get rid of all the heat put on it. The cooling system of the building or plant sets the heat load and it will try to reach equilibrium at any given point in time. The heat exchangers or contact surfaces or fluids transfer heat to the water. When the water circulates over the tower it contacts the air and some of the water is evaporated. A rough rule of thumb is that you will evaporate 556 Kilocalories for each kg of water evaporated (1000 BTU/lb). But this says nothing about the **level** at which the heat is removed.

If a tower is too small the equilibrium will be reached when the temperatures (both the cold water temperature and the hot water temperature) rise to the point where the system shuts down or water boils as steam or the plant cannot get the production needed or any number of other bad scenarios. If a tower is too big it is possible to lower the air rates, shut off pumps, use a fewer number of cells, or buy a smaller tower and save money.

Also, the wet bulb is constantly changing, and the hot and cold water temperatures will fluctuate accordingly. The tower size, its components, and the prevailing weather set the **level** at which the system operates, but they do not set the heat load.

One comment heard often is. *“We are not getting the range we used to get.”* or, *“The tower was designed to cool 10 degrees, and I’m only getting 8 degrees. It’s not working.”*

Again, the tower does not set the heat load. The heat load is defined as:

$$\text{Heat load} = \text{flow rate} \times \text{range} \times K$$

K is a constant that makes the units come out right: example: K= 500 (for 60 minutes an hour and 8.33 for pounds of water per gallon) for an equation of BTU/hr, flow in gpm, and range in degrees F.

If the flow rate is constant and the heat load goes down, then the range will go down. If a pump is turned off, or a valve is opened, or some other change is made, then all of these components of the

basic heat equation will change. If the range is not what was expected then something else must have changed.

Another related concept is that the range or heat load does not define the tower capacity. Capacity is a matter of cooling a specific heat load at a specific set of temperatures. If the equipment being cooled is not operating at the those same temperatures, or the design flow rate, or the design heat load, then it becomes difficult to say the tower is meeting its design capacity unless you have the aid of a set of performance curves, generally supplied by the manufacturer, that show what should happen at other conditions. Fortunately, there is a theory of how a tower operates within the system of the plant, and we can determine what will happen at off design conditions. Several CTI papers have been presented about these theories, so they will not be discussed in detail here. You can purchase software to calculate performance curves.

NOTE: Theoretical performance curves may not be quite correct. Nozzle pressures variations may cause more or less wall water (over-spray water runs down the walls rather than through the fill) or poor distribution or better performance because of high nozzle pressure, or fan efficiencies may not be constant because the water loading or pressure drops are not quite the same as at the design point. As an example, a rough estimate is that a 10% increase in water flow rate will reduce the air flow (and performance) by 1%. Again, the tower is a system in itself, and it is working in the system of the plant. More will be discussed below about the effects of changing one component and how it may change something else. The point is that theoretical calculations assume a constant system, and the system may not be constant.

The Art

Selecting the economic size for the tower is part art and part science. The science is knowing how all the components work together, and the art is picking the right “box size”. The science will be discussed momentarily, but first lets discuss the “box size”.

One can have a great big tower with a little bit of power or a small tower that uses a lot of power. The tower can be either a cross flow or a counter flow tower, forced draft or induced draft. They all can be designed to have the same “capacity.” That means they all will cool the same amount of water through the same temperatures at the same wet bulb. However, one may have a high fan power, or a high pump head while the another takes more real estate or requires more starters and controls or has better maintenance preferences.

Deciding which tower design is best for a given plant is usually either an economic decision or a physical need decision, and sometimes compromises are needed.

The economics usually involve a life cycle analysis of the cost of parasitic power for fans and pumps, the size of available real estate, piping and electrical equipment costs, cost of the basin and its supports, the cost of capital, whether or not the taxes are involved for a new capital expense or a write off for repairs.

Another factor is often dictated by the banks: how long will they give you to pay back the loan? It used to be that banks would loan money for seven years for a refinery or chemical plant and thirty years for a power plant. It took three years to build a refinery and five years to build a power plant. Therefore, the economic analysis was based on power costs of three years for a refinery or chemical

plant and twenty-five years for a power plant, regardless of how long the plant may operate. Since the power crisis of a few years ago there have been reassessments of these types of projects by the lenders, and they may consider long term payouts poor investments which raises the cost of capital.

The compromise comes when the economic analysis is for only a few years, yet it is known that the plant may run for fifty years or more. Perhaps there needs to be a consideration for the changes that could be made after the initial plant payout period has elapsed. Is it possible to add another cell or change the fan power to provide more cooling that would increase plant output or efficiency? What effect would these changes have on the tower or other equipment in the plant? Remember, everything works as a system. Even though capital resources may be limited, buying on a low first cost may not always be the best answer.

Physical needs may also limit the design options. Perhaps the expected water quality will be contaminated by oil or excessive dirt, fouling particles, biological growth and other contaminants. These potential problems may dictate splash fill rather than a more efficient film type fill, or less thermally efficient open flute film type fills. Maybe larger nozzle openings or different drift eliminator designs will be required. Easy, open access to the distribution system available in a cross flow tower may be necessary if nozzle plugging is expected and safety regulations do not allow entering a tower when the plant is in operation.

The interaction of tower components:

Shown below is a typical computer program printout that lists just about everything you need to know to design a tower.

English Units

Hot Water	95.00	Stack Ht.	6.00	Pressure Drops		
Cold Water	85.00	Stack Dia	18.00	InletLuvs	0.223	Plen Ht W
Wet Bulb	78.00	Fan Dia	18.0	Fill	0.457	6.000
Dry Bulb	95.00	Hub Dia	4.42	Eliminator	0.048	Plen Ht L
Altitude	634	Fan Type	HUDSON	Plenum	0.026	3.000
Fill Type	BA1900	No Blades	7	SubTtl<hs>	0.753	FanDk Ht W
Fill Ht.	4.00	Motor RPM	1765	Extnl <hs>	0.000	22.000
Inlet Ht.	9.00	Gear Ratio	8.50	Total <hs>	0.753	FanDk Ht L
Mode	CTI	Fan RPM	208	VeloRecury	0.000	19.000
Cp	1.000	Tip Speed	11742	Net <hs>	0.753	1.855
=====		Gear Eff	95	Velo <hv>	0.188	1.910
GPM	27000	GPM/Cell	5400	Total <ht>	0.942	0.720
No Cells	5	#da/m/sqft	42.22	U In/Out	14.71	14.96
Cell W	30.00	GPM/sqft	7.87	Dens In/Out	0.069180	0.069059
Cell L	24.00	L/G	1.554	Temp Out	91.1	Computer
BlkgFactor	1.05	NTU	1.391	GPM Evap	273.7	Fan Curve
Elim Type	CDX080	Velo In	986	CFM/Fan	433124	Output
LuvsSide	N	Velo Out	1702	Static Eff	57.4	0.0
Luvs on W	N	Fill Velo	626	Total Eff	-	71.8
LuvsSpacng	100.00	Elim Velo	632	PHP	89.4	0.0
RecircFlow	1.000	PresFactor	13.5	BHP	94.1	0.0

F19 Type / for Commands

Practical limits apply to each of the tower components. The discussion that follows will look at these limits and how changes in one will affect all the others (and, sometimes the system in which the tower is operating).

The design temperatures

High hot water temperatures affect:

- Choice of materials which are often limited to 52° C (125° F)
- Long term creep of standard PVC: slumping of fill sheets and eliminators can occur if hot water temperatures are above 53° C (127° F.)
- High temperature PVC or other plastic components may allow higher hot water temperatures
- Increased cost of water treating
- Scaling in heat exchangers
- Plant performance: It is suggested that running an overall economic analysis of flow rate and temperatures would be advisable.
- Evaporation rates may increase because the discharge air will hold more water per volume unit.
- Most tower fill tests have been run by manufacturers at 38° C (100° F) hot water temperature. Several authors have documented that operation at hot water temperatures above 38° C (100° F) will likely result in design deficiencies if “hot water correction factors” or revised rating theories are not used. Since many power plants operate at hot water design temperatures between 46° C (115° F) and 52° C (125° F), it should be noted that necessary corrections from simplified Merkel performance theory can be as high as 7% to 10% at these conditions.
- FRP (fiber reinforced plastic) structures may require special design considerations.

Cold Water Temperature

- One good question to ask is, “What is a degree of cold water worth?” If the cold water temperature is reduced one degree will production increase? If the cold water temperature is raised a degree will the plant shut down?
- How much will reducing the cold water temperature affect the cost of heat exchangers?
- Compared to the cost of shell and tube or other types of heat exchangers, the cooling tower is usually the cheapest surface in the plant, and it may pay to buy a bigger cooling tower to give colder water rather than by a bigger heat exchanger.
- As a rough guide, the cold water temperature changes about 0.7° to 1° for each 1° change in wet bulb temperature.

Long range and low flow rates

- Water distribution at very low flow rates (less than 2.0 l/s/m² or 3.0 gpm/ft²) are often a major cause of performance being less than expected when laboratory testing was done at higher flow rates. This also often occurs when a low flow rate is used for design and the system is run with fewer than the design number of pumps at low ambient temperatures without taking cells out of service.
- Using nozzles selected for very low flow rates requires careful consideration of possible variations in plant operations.
- Although low flow rates will usually result in less pumping costs, the overall affect on heat exchanger design should be carefully considered. Low flow rates with high LMTD may result in higher backpressure on the system and low water velocities may lower the heat transfer coefficients to the extent that larger heat exchangers are required.

Choosing the optimum design wet bulb is usually an economic decision:

- ASHRAE and USAF and others have studied the statistical weather data for most regions of the world.
- Selections based on the statistical maximum are rarely economical.
- Selecting a design wet bulb that is too low can result in loss of production on a hot day, or even plant shut down.

“Tough duty”: (Long range, close approach to a low wet bulb temperature)

- Requires a bigger tower
- More first cost and power
- Light water loadings
- Taller tower
- Closer spaced fill
- More fill height
- Smaller nozzle orifice or higher nozzle pressure.

“Easy duty” (Short range, long approach to a higher wet bulb temperature)

- Wider spaced fill
- Heavy water loading can cause plugging
- Lower height tower
- Larger nozzle orifices
- Lower fill heights and less fill volume.

An “easy duty” design might actually work better with half as much fill. For example, a cross flow tower with splash fill might be selected with closely spaced splash bars, but the water loading for an easy duty is high and so is the air pressure drop. Removing half of the splash bars might result in a lower fan power for the same box size. However, the air velocity will increase, and it is necessary to be sure the velocity limits for the drift eliminators have not been exceeded. Again, the tower components work as a system.

Approach

The “approach” (cold water temperature minus the wet bulb temperature) has probably the most affect on tower performance. While most manufacturers do not want to make selections based on designs with less than a 2.7°C (5° F) approach because a small deviation in temperature can sometimes mean a large variation in percentage capacity, a tower *will* operate at lower approaches. However, this usually means very low flow rates per unit area of tower and it may be difficult to guarantee the actual performance will agree with predictions based on tests at higher flow rates.

A rough general rule is that the required tower size (and cost) varies directly with the “approach”. As an example, a tower with an 4°C (8° F) approach will cost 25% more than a tower with a 4°C (10° F) approach. This larger tower might be in the form of a bigger box, or the same box with a different fill, fan, fan stack height, and/or more power, or a combination of all of these options.

Range

Changes in the “Range (hot water temperature minus the cold water temperature) generally do not have as great of an effect on the tower size as changes in the approach given a fixed heat load.

Altitude

The altitude or barometric pressure will definitely affect tower performance. The driving force for cooling water is related to the difference in the vapor pressure of water at the film of the water droplet and vapor pressure of water in the air stream. (Actually, vapor pressure can be converted to enthalpy, and standard cooling tower theories usually use enthalpy differences between the air and the water film as the driving force). Changes in the barometric pressure affect the vapor pressure and air density. There is a direct relationship between the density and the pressure drop through each of the tower components. There is a direct effect on the fan power and fan speed or pitch. Fan stall considerations will be changed with changes in altitude. Computer aided selections can easily and accurately account for changes in altitude.

Fill dimensions

Counter flow tower fill considerations:

- Close spaced film fill sheets would normally have a maximum effective height of 1.2m (4 ft) and a minimum height of 0.5m (1.5 ft.)
- Wider spaced film fill sheets can have a maximum effective height of 0.6m to 2.3m (2 ft to 7 ft.).
- Very wide spaced film fill sheets used for high fouling applications can have effective heights of as high as 3.75 m (12 ft).
- The diminishing return of taller fill heights is a logarithmic function, and it is influenced by the increased pressure drop as well as the reduction in driving force between the air and water enthalpies.
- Cross corrugated sheets may encourage plugging
- Splash fill requires much taller fill heights 4.5 m to 12 m (15 ft to 40 ft)
- Splash fill placed in the air inlet plenum below the top of the air inlet opening has been found to be only about 50% as thermally effective as the fill within the tower casing above the top of the air inlet opening.
- There is cooling in the rain zone below the fill, however, the laboratory tests were conducted with some rain zone effect, so only the marginal difference between the test rain zone height and the actual rain zone height should be added.
- A lower performance, non-plugging fill is infinitely better in the long run, even if it requires a larger tower. *A fill that will plug in your plant's specific application is never a viable option.*
- Tall single lift fill packs are difficult to handle if they plug. Cleaning or replacement costs will be increased.
- Cross-oriented, lower fill height packs may encourage plugging and add to the pressure drop, but they can also improve water distribution, and even air and water distribution throughout the tower is critical to obtain full performance.
- The air inlet height is critical to good performance. One of the most frequent causes of poor performance is because of poor air distribution in counter flow towers caused by high air inlet velocities. More comments on this are listed below.

- Fill packs should be properly supported to prevent long term structural failure.

Cross flow tower fill considerations:

- In general a tough duty requires a higher fill height to air travel ratio.
- Selecting the optimum height to air travel ratio is a major factor is selecting the optimum “box size”.
- Film fill crossflow fills often have only a few feet of air travel, while splash type fills require much wider towers.
- Slope of the fill or fill supports may affect the performance when high or low air flow rates change the draw of water through the tower as it falls. The object is to keep the flowing water within the fill, not on the louvers or the eliminators.
- Freezing weather may dictate the need for a heavy water loading at the air inlets and ice breaker bars within the louvers to prevent damage from thawing ice blocks.

Water distribution:

- Good water distribution is at least as important as good air distribution.
- Optimum performance is obtained when air and water distribution is uniform throughout the tower.
- *It is not possible to separate the performance of the distribution system from the fill.* Many authors have attempted to calculate the difference between the two so they can apply one fill or nozzle with an alternate fill or nozzle using the same set of test data. This cannot be done with any degree of accuracy. *To be reliable, a given fill must be tested with a given nozzle.* Failure to recognize this is often a cause of poor tower performance.
- Low pressure nozzles usually give much less performance than high pressure nozzles. As much as 25% of the cooling capacity can be produced by using high pressure nozzles which usually produce finer droplet sizes with resulting greater surface exposed to the air stream.
- Low pressure nozzles usually have larger orifices and can be used to reduce potential of clogging.
- Good low pressure nozzles produce uniform distribution with minimum pump head over the entire top surface of the fill.
- Daisy petal type nozzles seldom produce uniform distribution, and as a result performance can be significantly reduced.
- Nozzle spacing is critical for good distribution. Overlap of sprays is not always helpful.
- Up-spray nozzles do not appear to provide better performance than down-spray nozzles.
- Cross flow towers may need a “distribution decking” below the nozzles to produce even water distribution. This is especially true for film type cross flow fills.

How many cells?

Part of the art of selecting the “box size” is selecting the optimum number of cells. Often at least two cells are required to assure system availability. Beyond that, the optimum number of cells is associated with fill height, type of fill, air and water rate limits, fan size and fan and fan stack properties. The cost of piping and valves, pump head cost evaluation, electrical and electrical wiring and control costs and cold water basin design and costs should also be considered when selecting the number of cells.

Cell size

Cell sizes for package and factory assembled towers are usually based on practical shipping limitations for truck (and sometimes rail) shipment.

Field erected towers are usually sized on 4 ft or 6 ft (or 2 meter) structural spacing.

- Column spacing larger than 2m X 2m (6 ft X 6 ft.) would generally result in larger transverse and longitudinal structural members.
- Practical fill pack sizes often are based on a maximum support span of 2m (6 ft) for reasons of supports, shipping and handling.
- Vertical structural spacing greater than 2m (6 ft) can cause increased erection costs because workmen have more trouble climbing through the tower. For this reason some manufacturers may limit the maximum fill height to avoid the cost of an extra girt level and to avoid water and air re-distribution problems as well as possible extra pressure drop through the transition area.

It is usually better to make the cell length the longest direction in order to minimize the inlet air velocity for a set inlet height. (Usually the length is the direction of a series of cells in a row, and the width is the dimension across one cell). Very wide cells require extra consideration of air distribution to the middle of the tower. A few guidelines will be discussed later.

The relationship of the fan to the cell size sets the fan plenum size.

- For counter flow towers the upper plenum height is usually set by extending a 45 degree line from the top of the drift eliminators to the fan blade tip using the longest dimension of cell width or cell length.
- Low height upper plenums can cause significant performance problems due to mal-distribution of air in the fill. This is the most important reason to get the plenum height correct.
- For cross flow towers the bottom plenum width should be adequate to keep the air velocity leaving the bottom 2m of fill height to no more than the fill air velocity.
- Long cell length, small fan diameter, cross flow towers should have fan decks raised above the top of the fill to allow air to properly approach the fan stack opening.

Tower orientation and configuration

In general the best layout is to have the longitudinal axis of the tower in the same orientation as the prevailing summer (maximum wet bulb) wind direction.

- While space restrictions may require them, back-to-back counterflow cell configurations are discouraged because they usually result in excessive recirculation which reduces performance. If this type of design is used it should include low inlet velocities, high discharge velocities, and widely-spaced tall fan stacks.
- If the prevailing wind is cross-wise or quartering to the longitudinal axis, it is recommended the cell and fan sizes be selected to maximize the space between fan stacks.
- The distance between towers should generally be one tower length to minimize discharge air interference from one tower to the other.

- Air inlet openings on all sides of a counterflow tower will reduce the required inlet height and lower the pump head accordingly. However, if louvers are not provided it is necessary to have wind baffles to prevent water from blowing from the tower air inlets.
- *It is critical that multiple-cell counterflow towers be checked to assure that the inlet air velocities to the interior cells are within practical limits: this is especially true if there are transverse cell partitions, and failure to check this design consideration has often been a major source of poor tower performance.*
- Air inlets on both sides of a cross flow tower produces desirable fan requirements because there will be twice the amount of air flow for the same amount of pressure drop. This is a major advantage of a cross flow tower.
- Cross flow designs are often the best option for small package towers with propeller fans because counterflow designs do not have room enough for large diameter fans to match the required air flow. This is the main reason small packaged counter flow towers are provided with blower fans.
- For smaller applications in congested areas it is often desirable to orient the tower to provide the minimum noise pollution. Noise levels off the cased side of a tower will often be 6 to 12 dB less than the levels off the louver side.

Fans

Fan selection is an extremely critical part of designing a cooling tower and changes made to a tower will usually affect fan performance.

- Fan operating power costs are often evaluated on a life cycle basis.
- Induced draft fan discharge velocities should be at least 6 m/s (1200 fpm) to minimize recirculation
- Maximum fan discharge velocities may be as high as 11 m/s (2,200 fpm), however, such high velocities are seldom economical.
- Maximum fan speeds are usually limited to blade tip speeds of 61 m/s (12,000 fpm) to minimize stress on blade attachments, keep noise to reasonable levels and reduce blade air loading.
- *What really counts is total fan efficiency.* One unique reality of a cooling tower is the linear relationship between thermal performance, first cost, and fan efficiency. If the fan efficiency can be increased the tower size and cost can be reduced proportionately.
- A major objective is to have the fan operate in what is normally called an optimum fan efficiency zone. This can be done by changing the fan design, number of blades, fan speed or stack design.

There have been many cases where changes have been made to existing cooling towers that result in performance decreases when performance increases were expected. This is often due to changing fill types or adding fill without careful consideration of the affect on fan performance. This can be seen in the following example:

- Fan curves are often as shown in Figure 1 with plots of *total* pressure vs. air flow rate and power vs. air flow rate for lines of constant fan pitch

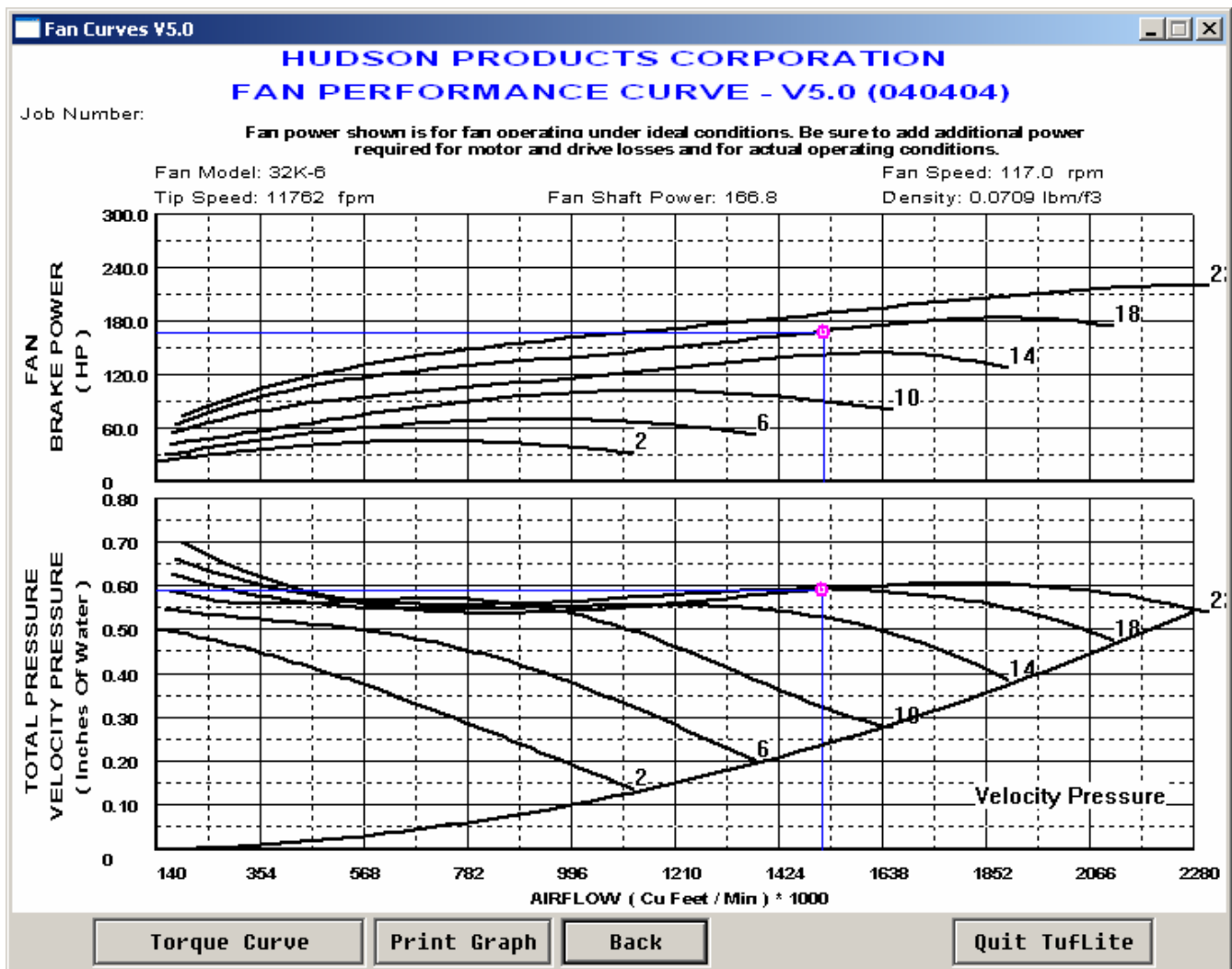


FIGURE 1 (to be revised for final paper)

- A more useful plot ignores the pitch (which is often a guess anyway) and plots power vs. air flow with lines of constant *static* pressure as shown in Figure 2.
- Using the original tower design it is possible to plot what is normally called an “operating line” which varies the *static* pressure as the square of the velocity (fan laws apply).
- Changing the fan speed has the affect of moving the operating point up or down the “operating line”
- If the fill is changed, say to a denser fill or taller fill height in a counterflow tower, the static pressure at the same air rate will be increased. The static pressure can also be changed by other factors such as adding louvers, changing eliminators, adding other restrictions to air flow, or plugging of the fill with algae or other contaminants. If something is changed it is not reasonable to expect that there will always be a positive affect on performance. It just might go the other way.
- The effect of increasing the pressure drop is to move the operating line to the left. This reduces the air rate at the same power level, and in addition it pushes the operating line close to possible fan stall.

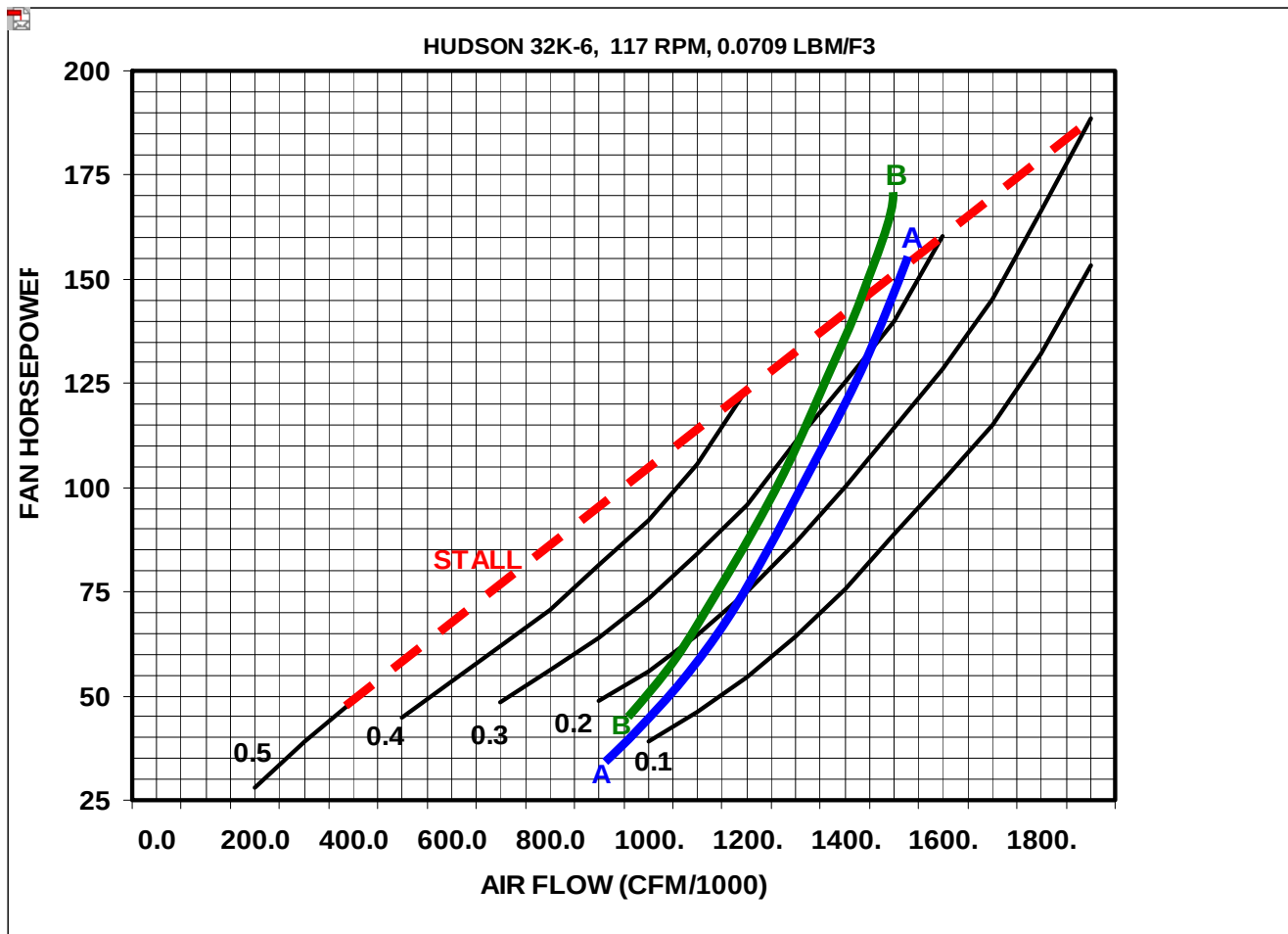


FIGURE 2

- If the fan is operating in the fan stall region it will be inefficient and performance will be erratic.
- If fan changes are necessary they can be made with changes to the pitch, fan speed, or number of blades or a more efficient fan blade design. Often it is possible to change to a larger diameter fan with the knowledge that the driveshaft, gear box, fan stack and mechanical equipment supports will also have to change.
- Making changes without considering the total consequence is often undesirable.
- Changing the hub disk seal diameter will change the velocity pressure, and it will affect the total efficiency. A hub disk seal will reduce or prevent air recirculation at the center of the fan.
- Two speed fans can save considerable power expense. According to the fan laws the fan power decreases proportional to the speed cubed, and the noise level can be as much as 13 dB less at half speed.

Fan blade tip clearance

Most fan manufacturers publish fan performance data based on tests with very close tip clearances. Tip clearances need to be increased due to fan stack deflections from wind, vibration and erection tolerances. Increased tip clearances will reduce total fan efficiencies from most standard published data.

Practical tip clearances	Practical reduction in total efficiency
o 5 ft diameter – ½ inch	7.5%
o 14 ft diameter – 13/16 inch	7.0%
o 20 ft diameter – 1 1/8 inch	6.5%
o 28 ft diameter – 1 ½ inch	6.0%
o 40 ft diameter – 2 inch	5.0%

Fan power should be increased to account for the lower expected total fan efficiency.

Fan stacks

Fan stacks designs are usually selected for economic reasons.

- Package towers may use low straight sided stacks to keep the first costs to a minimum while ignoring the higher power costs. Increasing power a few kW will usually not cause a problem for a company.
- Larger towers will benefit greatly from eased inlet stack designs, but tall stacks are not always justified.
- Tall fan stacks will usually be justified when high power evaluations are required or when it allows use of a smaller motor. Changing the fan stack height can be one way of fine tuning a design.
- An extended height fan stack will improve fan efficiency. In general, there will be recovery of the system velocity pressure loss to the extent of approximately 70% of the difference between the velocity head at the fan net disk area (area based on fan diameter less area based on seal disk diameter) less the velocity head at the fan discharge (area based on stack discharge diameter). This has the same effect on the fan as lowering the static pressure loss through the rest of the tower system.
- Extending the fan stack height not only costs more for the taller stack, but it also may affect the structural design due to higher wind load considerations.
- Low height fan stacks (2 m / 6 ft high) on large fans (over 7 m (22 ft) diameter) are not recommended. The wind effects on fan performance can be significant due to eddies formed by the upwind edge of the stack. The minimum stack height should be 3 m (10 ft.) for applications with large diameter fans.
- Straight-sided or truncated-cone type fan stacks may be economical to build, however the reduction in fan efficiency is usually quite significant – maybe more than 15%.

Water and air rate limits

Water rates

- To obtain good water distribution the water rate per unit area should normally be greater than 3.0 gpm/ sq. ft. for either counter flow or cross flow towers. It is difficult to assure good distribution below this value, but this does not necessarily mean a tower will not perform at lower water rates. A good distribution system and even air rates are critical.
- Cross flow maximum water rates can be as high as 13 to 17 l/s/m² (20 to 25 gpm/ft²) for splash type fills with practical limits of 6.8 to 10 l/s/m² (10 to 15 gpm/ft²).
- Counter flow maximum water rates are usually 5.8 to 6.8 l/s/m² (8.5 to 10 gpm/ft²).

- Some close-spaced counter flow film type fills can have a plugging or percolating action with high air velocities and heavy water loadings which cause unpredictable and erratic performance. This can sometimes be overcome by spacing the fill sheets so that every other sheet has the bottom edge recessed into the pack (alternate tips recessed). With this design the thickness of the fill sheets should be increased to provide adequate structural support strength.

Air rates

- It is usually best to analyze air flow rates in terms of mass flow rather than volume flow because the mass flow rate of dry air is the only thing that stays constant in a cooling tower. Users of published performance data need to make adjustments if the analysis is based on velocity as it is necessary to correct for density and specific volume changes throughout the tower.
- Normal design air rates are 1.6 to 4 kg/s/m² (20 to 50 lb dry air/min/ft²) .
- Normal design air velocities for counter flow towers are as follows:
 - *Absolute maximum air inlet velocity – 5.333 m/s (1050 fpm).* Air velocities greater than this value will likely cause unexpected performance problems because the inlet air by-passes the outer few feet of fill. Sometimes air inlet guide vanes will be used to “pre-turn” the air, but it is not a certainty that full performance will be available. One tool used to determine the affinity of the air to evenly distribute below the fill is to calculate a pressure ratio (the fill and eliminator pressure drops divided by the total inlet pressure drop consisting of louver losses, velocity head losses at the inlet, and rain zone losses). The Pressure ratio should be a minimum of 5 and preferably 10 or more.
 - Minimum fan outlet velocity – 6 m/s (1200 fpm.) - Lower outlet velocities will increase the likelihood of recirculation or interference.
 - Maximum average fill velocity – 3.2 m/s (625 fpm). Some manufacturers may want to use higher velocities, and for small towers the increased fan power required may be justified. A 3 m/s (600 fpm) maximum fill velocity is usually more practical. High power evaluations often dictate use of fill velocities as low as 2.3 to 2.5 m/s (450 to 500 fpm).
 - Maximum eliminator velocity – 3.3 m/s (650 fpm). Some new eliminator designs may be available that allow higher air rates, however, water begins to float at 3.5 to 4 m/s (700 to 800 fpm) depending on the droplet size. Practical designs will usually limit eliminator velocities to less than 3.2 m/s (625 fpm). Modern cellular eliminators can significantly improve drift rates, and pressure drops can be less than the original blade or wave form types. Cross flow cellular eliminators should have drain passages to prevent build-up of water that might result in extra drift.
 - For counter flow towers the maximum rates to avoid plugging or percolating are as follows;

Fill sheet spacing 12 mm				Fill sheet spacing 19 mm			
m/s	l/s/m ²	fpm	Gpm/ft ²	m/s	l/s/m ²	fpm	Gpm/ft ²
3	13.6	600	20	3	15	600	22
3.55	6.8	700	10	3.55	11	700	16
3.8	4.0	750	6				

Pressure drops

Sometimes the ability to select a proper fan is limited by high static pressure losses. Propeller fans usually do not perform well at total static pressures above 250 Pa (1.0 inches wg) without an excessive number of blades or high speed. High static pressures also can result in noisy fans.

Inlet and discharge pressure losses by enclosures, louvers, screens or obstructions must be accounted for.

Fan power

The output from fan curves is called fan horsepower or fan kW. It is necessary to add drive losses created by gear boxes or belts and bearings to obtain the brake horsepower or brake kW. Spiral bevel gears, most commonly used in cooling towers, have an inherent efficiency of 98%, and in addition a gear box will have a no load loss due to friction and oil viscosity which brings the total drive loss to a total of about 4% to 5%. Actual no load losses will vary from about 0.5 kW for small gear boxes to 3.75 kW for very large boxes.

Motor efficiency

High efficiency motors are generally economical for cooling tower fans. Since certified tower testing is based on readings of kW input to the motors, it behooves the manufacturer to use the best motor efficiency possible. On the other hand, owners and operators that need to repair a motor should seriously consider new motors rather than re-winding the old one to get the best efficiency.

Conclusion

Designing a cooling tower is partially art and mostly science.

The art is picking the best box size and the right combination of fill, fan, and other components to give the most economical selection. The most economical selection may be low first cost or low evaluated cost.

The science is adequately measuring the test data, developing a theory that properly correlates the data and allows for application of the data in a manner which does not produce distorted results. Practical review of the results should show the laws of physics still apply, i.e., similar fills should produce similar performance characteristics, and similar fans should have similar efficiencies. There is no "Black Magic".

Air and water distribution are all important. The best performance is going to occur when the distribution is even over the full area of the tower. This is difficult to achieve. Some water nozzles and distributors are better than others, and sometimes a manufacturer may claim performance capabilities based on testing with one type of nozzle apply to their nozzle when this is really not the case.

Fan testing may have been done in ideal eased inlet stacks with very close tip clearances. Actual field installations seldom have extremely close tip clearances to allow for wind load deflections, thermal expansion of the fan blades or the stack, possible slight slumping of a stack over time, and construction assembly tolerances. The shape of the stack being used may not be the same as the fan manufacturer used for its tests. Improper fan hub seal disks may be used.

Air distribution can be affected by high inlet velocities, louver spacing and angles, wind, partition location, obstructions and structural blockage, etc.

Static pressure losses for various components are the most difficult data to measure and predict. Different manufacturers may show significant variations in the pressure drop across individual components, but the total pressure drop will normally be quite consistent.

These are just a few of the many considerations that create uncertainty. The object is not to hedge: it is to try the best possible to predict something close to reality.

The important point to remember is that a cooling tower works as a system, and changes to one part will usually affect others, and the design of the tower can significantly affect the overall plant operation.

ABOUT THE AUTHOR

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Prior to formation of his own company in 1984 he worked for twenty-five years with the Marley Co and its subsidiaries. He has represented several different cooling tower, heat exchanger and industrial equipment manufacturers, been Principal Engineer for an air pollution control company, and designed and fabricated several different types of evaporative cooling equipment. He is the author of cooling tower and psychrometric computer programs and CTI paper TP92-01.