Fans Key to Optimum Cooling-Tower Design

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The questions most frequently asked of a fan engineer about axial-flow fans for today's wet cooling towers generally cover:

- Performance
- Efficiency
- Corrosion resistance
- Noise

This article reviews such fundamentals and hopefully gives new insight for optimum tower design.

Optimum. Optimum fan performance is several things other than just delivery of design-air quantity:

- Lowest first cost. This means optimizing the fan diameter and number of blades. Why buy extra blades if they aren't needed?
- Lowest horsepower requirements. The fan should work as close to an efficient operating point as possible.
- Lowest noise without extra cost. Since noise emitted by the fan is a function of tip speed to almost the 6th power, even a small reduction of speed has significant effect on the environment. If this can be accomplished and the required air duty attained at no extra cost, why not?
- Lowest maintenance costs. The optimum choice of fan blades and hub materials ensures against corrosion and subsequent replacement.

Before going into some suggestions on how to achieve the proceeding goals, a review of basic fan principles may be helpful. Starting from ground zero, they are:

Basic Fan Equations Fig. 1

- TP = VP + SP (in. H_2O)
  TP = total pressure, VP = velocity pressure,
  Static pressure = SP
  (All pressures must be relative to the same air density)
- ACFM = actual cfm
- Air hp $= \dfrac{\text{TP actual} \times \text{ACFM}}{6{,}356 \times \text{Efficiency}}$
- of Efficiency $_{total} = \dfrac{\text{TP actual} \times \text{ACFM}}{6{,}356 \times \text{HP actual}}$
- Efficiency $_{static} = \dfrac{\text{SP actual} \times \text{ACFM}}{6{,}356 \times \text{hp actual}}$
- Air velocity $= \dfrac{\text{ACFM}}{\text{Net free area of fan}}$ fpm
- Velocity pressure $_{std.} = \left(\dfrac{V}{4{,}005}\right)^2$ in.-H_20
- Air density $_{std} = 0.075 \dfrac{\text{lb}}{\text{cu ft}}$
- Basic fan law = CFM = f (rpm)^1 , TP = f (rpm)^2 , HP = f (rpm)^3
- Solidity ratio $= \dfrac{\text{sum of tip chords}}{\pi \text{ diameter}}$

When V.R. stacks are used

- Velocity recovery = (VP $_{fan}$ ⁻VP $_{exit}$) Efficiency of recovery
  (Both velocity pressures must be calculated at same density
  Efficiency of recovery will usually be in the range of 0.6 to 0.8)
- TP $_{effective}$ = TP $_{std}$ – V.R.

- A fan is supposed to move air and do work.

- This air is supposed to be evenly distributed over the entire exit area of the fan.

To move air, the fan must overcome two resistances, which are measured as pressure drops across the fan.

The first is a parasitic loss called the velocity-pressure loss. I see this as the energy required to move the required air quantity without doing any work to overcome the system resistances. Work is done however to move the hot air away from the equipment.

The second resistance is the static pressure loss. It is the accumulated losses due to duct, fill, and mist-eliminator pressure drops. This would be the "work" to be accomplished and reflects the design of the total system, including inlet conditions.

Whether the air is distributed evenly across the fan is primarily a function of the blade and hub design. A properly designed blade will have adequate chord width and "twist" to ensure an even distribution of velocity pressure over its entire length.

A properly designed hub will include a center air-seal disk which prevents negative air flow at the center of the fan.

Some basic fan equations are shown in Fig. 1.

Cases. Following are two cases that most everyone has faced:

- Designing a new tower from scratch, attempting to get the best fan design possible.

- Replacing a fan on an old tower where practically nothing is known.

Selection for new tower design Fig. 2

Total pressure

Density ratio: $\frac{0.0734}{0.0750}$ saturated air

$SP_{Std.} = \frac{0.477}{0.978} = 0.488$

$VP_{Std.} = 0.220$ (read from 28ft fan curve or calculated using NFA of fan.)

$TP_{Std.} = 0.488 + 0.220$
$= 0.708$ in. H_2O

Speed Factor

Speed factor (flow) = $\frac{12,000}{10,000} = 12$

Speed factor (pressure) = $(1.2)^2 = 1.44$

Speed factor (hp) = $(1.2)^3 = 1.73$

Corrections

CFM corrected = 1,100,000 x 1.2 = 1,320,000 ACFM

TP corrected = 0.708 x 1.44 = 1.02 in H_20

The check list for a brand new design includes:
1. Fan-diameter or cell-size limitations.
2. Actual cfm of air.
3. Actual static pressure.
4. Air temperature and elevation (density).

Comparison of Selections Table 1

Fan Diameter ft	No. of blade s	Pitch Angle	Fan total Efficiency	Curve horse-power	Actual horsepower	Fan cost ($)
28	9	20º	0.730	290	164	3,621
28	12	16º	0.784	270	152	4,737
26	11	19º	0.823	290	164	4,074
30	10	17º	0.757	260	147	4,345

Effect of velocity recovery stack Table 2

Fan Dia.	Total Pressure standard	Total pressure effective	Horse-power	Horse-power w/V.R.	% Savings of horsepower
26 ft	0.798	0.672	164	138	16
30 ft	0.678	0.596	147	124	15

5. Limitations on tip speed other than standard gearbox ratios.
6. Should a velocity-recovery stack (VR) be used?
7. Are there noise-level specifications involved?
8. Are any unusual corrosion problems expected?

Factors that must be known when replacing a fan on an existing installation are:

1. Fan diameter.
2. Installed horsepower.
3. Gearbox ratio.
4. Shaft size or gearbox model.
5. Some estimate of elevation above sea level of installation.

Of course what we are seeking in Case 1 is an optimum fan diameter, number and type of blades, required pitch angle, fan rpm, and some estimate of horsepower. In some cases, we are looking for an estimated sound pressure level or possibly a sound power level to satisfy OSHA requirements for working-area noise levels or more importantly noise levels at a plant boundary.

In Case 2 we are looking for a fan that at least will be an adequate replacement for the original fan.

Fan diameter. The fan diameter bears on performance primarily because it affects: (a) magnitude of the velocity pressure, which is a parasitic loss; and (b) the pressure capability of the fan.

In our estimation, velocity pressure should fall in the range of 0.15 to 0.25 in. for optimum performance.

Of course, other factors influence the choice of fan diameter, such as cell-size limitations due to plot plan, economics, or selection of a fan for an existing installation.

Excessively high velocity pressures are the result of high flow requirements for the fan size required, resulting in a waste of horsepower.

It may require adding blades just to cope with a high velocity-pressure requirement. However, this presents a good case for converting velocity pressure into useful static-pressure capability by adding a velocity-recovery stack.

The solidity ratio is a way to compare a fan's pressure capability. The higher the ratio, usually the more work the fan can do. Still another aspect of optimum fan diameter is cost. Nonstandard sizes mean special handling by the fan manufacturer at additional cost.

Other factors. Besides fan diameter we must also consider:

- Actual cfm, or the design-air quantity at the fan required to do a specified job.

- Actual static pressure, usually stated at the actual exit-air temperature to the fan, and at the elevation of the cooling tower. This must be converted to static pressure at standard conditions before being used with the fan-performance curves.

- Air temperature, elevation. These factors are important in fan selection because of their effect on fan-pressure capability and horsepower requirements through changes in air density. These factors are combined into a density ratio. Hudson's fan catalog has a nomograph for the density of dry air which is accurate enough in most cases to use for saturated air as well. This nomograph gives the reciprocal of the density ratio.

New tower. Let's look at a typical case of fan selection for a new tower design. Considered will be a 26-ft, 28ft, or 30-ft-diameter fan. Duty requirements are 1.1 million actual cfm, 0.477-in. actual static pressure, 95º F. outlet-air temperature at sea level. It is to be restricted to a maximum tip speed of 10,000 fpm.

To be determined are diameter, number of blades, pitch angle, and estimated horsepower.

Since all fan curves are based on standard conditions (70º F., sea level or 0.075 lb/cu ft density) let's took at the standard total pressure, considering 28-ft-diameter fans (Fig. 2).

Then we must consider the effect of the reduced tip speed by calculating a speed factor (Fig. 2).

Since the fan loses capability as it slows down, we now increase our requirements to include these losses before we enter the fan curves. These corrections are shown in Fig. 2.

As you can see, the speed limitation has imposed a 20% penalty on flow and a 44% penalty on pressure.

To select a 28-ft-diameter fan, we can go to the performance curves and enter them for 1.32 by 101, actual cfm and 1.02 in (H,O). We may then read pitch angle and curve horsepower (standard conditions).

This problem-example work could be done by a 28-ft, nine-bladed fan at 20° pitch, or a 12-bladed fan at 16º pitch. We could also consider 26-ft diameter fans or 30-ft fans. Table 1 shows a comparison of selections.

Note that fan cost/cell could vary from $3,621 to $4,737 for a difference of $1,116/fan.

Also note the difference in power requirements from 147 to 164 hp. These differences are due to difference in fan efficiency at different blade angles and due to differences in velocity-pressure requirements for the different-size fans. Any one of these fans would handle the air duty. The final choice could be affected by the economics of the tower design and possibly gearbox costs.

The effect of adding a velocity-recovery stack to the 26-ft and 30-ft selections should also be considered. How does it affect fan-horsepower requirements?

To calculate the velocity-recovery stack, we must know the exit area of the particular stack being considered and the net free area at the fan. For our cases we will assume exit diameters based on 14-ft height with an 8º slope/side. An efficiency of regain of 0.80 was used. A summary of results is shown in Table 2. The regain efficiency is a function of the L/D ratio of stack height to fan diameter.[1]

Fan replacement. A problem that arises frequently is fan replacement for an old tower that practically has no design information available.
In this case, the only approach is to calculate the curve horsepower that will allow for the actual gear ratio and approximate density. An example follows:

Assume the tower has an old obsolete 9-ft-diameter fan, with eight blades, 20-hp motor, and 4.5 ratio (in Houston). What is required for a replacement fan?

- Density. Most wet towers have roughly 95º to about 125º F. outlet-air temperature. If we assume 125º F. and sea level, density is 0.065 giving a density ratio of: (0.065/0.075) = 0.867.

- Speed. At 12,000-fpm tip speed (usually standard in the U.S.), a 9-ft fan requires 424 rpm. In this case we have (1,750/4.5) or 389 rpm or 11,000 fpm tip speed.

The speed factor is: (424/389) = 1.09.

Since horsepower is affected by the speed cubed, we have $1.09^3 = 1.29$.

Since we will decrease horsepower by 29% due to speed, we add this loss plus the density loss to determine a "curve" horsepower:

(20)(1.29)/(0.867) = 29.8

This means that if the fan is loaded to 29.8 hp on the curve for 12,000 tip speed and 70º F. air, when run at 11,000 tip speed in 125º F. air, the actual motor load would be 20 hp.

Considering a four-blade fan first for economy, we see that about a 201 pitch angle would be required. If we choose a six-blade fan, operating pitch would be between 14º and 16º, a more efficient operating range. The actual pitch would be selected by field adjustment by comparing motor amperage with nameplate amperage. See the fan curve (Fig. 3).

Fan curves. Some comments are in order on Hudson fan curves since they use different parameters than most. Basically, Hudson uses total pressure vs. flow rather than just static pressure vs. flow.

Reason is that velocity pressure can become a major resistance for the fan to overcome and needs to be considered.

Hudson curves have a convenient line to read velocity pressure directly.

Another major point of difference is that the "stall" area of the curve is shown, as well as normal working range. It is risky to extrapolate fan performance in the stall region.

Tip clearance. Data for all Hudson fan curves were acquired by testing 5-ft diameter scale fans in a test tunnel at Texas A&M University with essentially "zero" tip clearance.

Losses up to 20% of fan efficiency are possible with excessive clearance. Since most of the work is done by the outer third of the fan blade, excessive tip clearance allows "spillover" of, the air flow from the high-pressure region to the low-pressure region in the inlet side.

By "excessive" tip clearance, we mean greater than about 0.3% of fan diameter for cooling-tower fans. This would be no more than 1-in. clearance for a 29-ft-diameter fan or about 0.32in. for a 9-ft-diameter fan. Sometimes this may not be possible to attain, but the consequences should be recognized.

Corrosion Control. The effect of corrosion must be considered on the two major elements of any cooling tower fan the blades and hub. The severity of such problem depends largely on the efficiency of the mist eliminators. An airstream with a lot of entrained water droplets can also actually erode the blades' leading edges due to the impact at rather high velocities.

We believe the optimum blade material is plastic. Fiber-glass-reinforced epoxy was chosen over polyester because of higher strength, higher heat resistance, and greater chemical resistance. Epoxy is impervious to any element normally found in the water vapor. No additional corrosion protection is required. Further, any scratch or minor surface damage to the plastic does not weaken its resistance to corrosion. We do use fiber-glass-reinforced polyester for our air-seal disks, which are only lightly loaded in service.

For corrosion protection of the hub, our standard method consists of galvanizing over steel. On some cast components a zinc-rich paint is applied. When the user feels additional protection is necessary, coal-tar epoxy coatings or polyvinyl chloride coatings can be applied with minimum difficulty. Chlorinated rubber coatings are difficult to apply because of the large thickness buildup necessary. Coatings requiring 20 to 30 mils thickness make accurate hub assembly difficult.

Costs can be high when certain grades of stainless-steel fasteners are specified. For instance, we frequently have requests for 18-8 grade stainless fasteners and these are readily available. However, if 316 grade stainless is specified, costs doubled and reach $20/blade, just for hardware.

The 316 grade stainless is basically the same as the 18-8 but has molybdenum added for greater strength at elevated temperatures. Since outlet temperatures rarely exceed 125ºF., 316 grade stainless would seem uneconomical.

Monel fasteners are sometimes requested. When we have to quote an additional $60/blade, another suitable material is usually found.

On a few rare occasions we have had requests for complete hubs made from 316 stainless steel. Such construction could double or even triple the cost of the whole fan. We feel that an all-stainless hub would really not be satisfactory. In some applications where the makeup water has high chlorine contents, stress corrosion could cause catastrophic failures in short periods.

As an alternate protection for fasteners in light of today's material shortages and costs, we suggest some of the dry-film coatings.

Many new coatings have been developed for corrosion protection of offshore platforms, chemical plants, and other severe atmospheres.

These coatings on standard fasteners seem an economical alternative and in many cases provide superior protection than some expensive alloys.

Noise. It is difficult to calculate and guarantee the maximum noise level from a fan in a new installation without having tested a previous installation. Thousands of man-hours have been invested in the study of fan noise in air-cooled heat exchangers. These noise levels can be guaranteed based on full-scale tests. Field tests of standard cooling-tower modules must now be made to allow guaranteeing of cooling-tower fan noise. The effect of water noise further complicates the problem.

There is no real agreement between engineering societies on even the basic parameters for calculating expected fan noise for a particular set of conditions. Some say noise is a function of tip speed, static pressure, horsepower, flow, diameter, or number of blades.

Each manufacturer has his own method. The parameters we at Hudson feel most important are tip speed and pressure differential across the fan. Hudson's position is given in a paper by K. V. Shipes[2].

The one general noise specification designed to protect "inplant" workers is the Occupational Safety and Health Act of 1970, paragraph 1910.95. This criterion is based on sound-pressure levels in dba and lists nine discreet pairs of sound levels and associated permissible hours of duration. The sound levels range from 90 dba for 8-hr exposure to only 15-min allowable exposure to 115 dba.

Much more difficult criteria have been established in Europe and some states, notably California, which limits the total noise at the plant boundary. In these cases the sound energy, or gound-power level of the total fan installation must be studied.

One of the most important factors in evaluating noise is obtaining a precise definition of the point or locus of points at which the noise specification must be met. It is not sufficient to state: "sound-pressure levels must not exceed 90 dba" without stating where measurements will be taken.

Generally, for cooling-tower work, the main point of concern for the effect of noise on workmen is on the fan deck. A more-precise definition of a noise specification would be for example:

"The sound-pressure level at X feet radially from any fan stack and at Y feet above the fan deck shall not exceed 90 dba."

When it is necessary for tower/suppliers to furnish noise guarantees to customers, it can be done if the fan manufacturer is given sufficient data concerning the fan environment.

If the measurement is to be made at a point on grade level, a sketch is helpful if it shows the orientation and dimensions of the tower with respect to adjacent buildings or unusual terrain.

The height of fans above grade, height of velocity-recovery stacks, and exact location of the measurement point or points is necessary.
Noise criteria should be relayed to the manufacturer exactly as stated by the specifications.

Generally, OSHA requirements are not difficult to meet if the concern is primarily fan-deck noise. If a guaranteed noise level in a community several miles away is required, the noise analysis becomes very complicated because prevailing background noise and attenuation of noise by the natural surroundings must be considered.

Fan manufacturers must exempt motor noise, gearbox noise, and water noise from any guarantee. They can be included for special cases if sufficient data are given.
We are now studying the attenuation of sound-absorbing covers for the high-frequency motor and gearbox to help simplify this problem.

If any type of silencers is being considered for the fans, check the economics. Most of our particular fan noise is in the 125-500-hz bands. It may be cheaper to slow the fan down, add more blades, and avoid the silencer treatment. Each case must, of course, be considered individually.

It is possible to decrease fan noise about 10 db by reducing tip speed from 12,000 fpm to 8,000 fpm. This reduction, however, would be possible only if the fan being considered had the capability of handling 125% more pressure and 50% more flow without stalling.

A rough estimate of fan cost vs. decibel reduction is shown in Fig. 4. A 14-ft fan was used in this analysis but the costs would be proportional for any fan.

References

1. Buffalo Forge Co., "Fan. Engineering," Sixth Edition.

2. Shipes, K. V,, "Noise Control of Air Cooled Heat Exchangers," API No. 34-72, 1972.

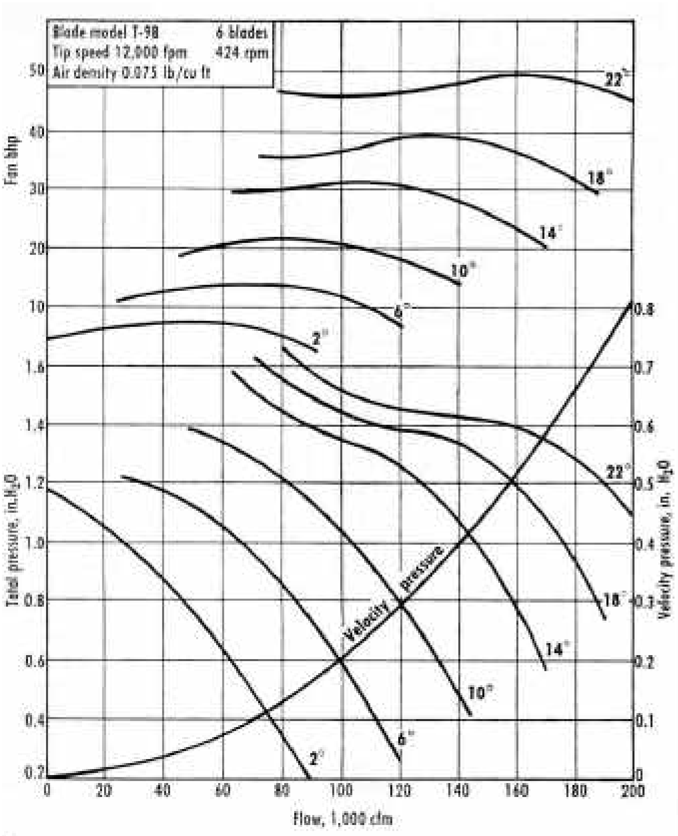

Fig. 3 Fan Performance Curves*

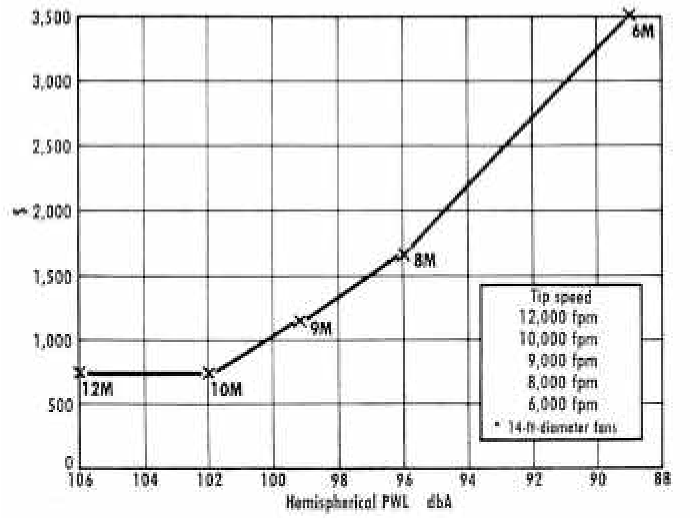

Fig. 4 Fan Cost Vs. Decibel Reduction*

*Hudson Engineering Corp. (Ratings are result of tests run in accordance with Fig. 3 and 6 of Bulletin 210, April 1962 edition, standard test code for air-moving devices adopted by the air-moving and conditioning