Minimizing Fan Energy Costs

Parts 1 and 2

Robert C. Monroe
Hudson Products Corporation, Houston, Texas

Of five ways for reducing the energy cost of operating axial fans, two are discussed in Part1: Improving the fan and the drive system. The three other methods will be taken up in Part 2.

Reprinted from Chemical Engineering Cost File, 1993

Part 1 — Minimizing Fan Energy Costs

The causes of fan system inefficiency are sometimes obvious, but usually not. For instance, blade design itself is critical. Axial-fan blades are usually made of extruded aluminum or molded fiberglass. The former are always of uniform chord width, whereas the latter can have an irregular shape (Fig. 1).

A basic design criterion is that a blade should produce uniform flow over the entire plane of the fan. An aerodynamic principle involved says that the work done at any radius along the blade is a function of blade width, angle of attack, and tangential velocity squared. Angle of attack in airfoil design dictates the amount of blade twist at any radius along the blade.

At points along a blade from the tip to the hub, tangential velocity decreases sharply. To compensate for this, so as to produce uniform flow, the blade must be wider and its twist greater, closer to the hub. If it cannot be made wider, its twist must be correspondingly greater.

In the case of the extruded blade, twist is created by mechanical bending. Of course, limits of elasticity prescribe the maximum twist. In the case of the molded blade, however, there is no such limitation to chord width or twist, so it can be made closer to the ideal shape.

The point being made is that blade design itself can cause poor flow distribution and inefficiency.

Also an inherent problem with the axial fan is swirl-the tangential deflection of exit-flow caused by the effect of torque. Air vectors at the extreme inboard section of a blade actually reverse direction, diminishing the net flow. Swirl can be prevented with an inexpensive hub component, which usually covers the inner 25-35% of the fan diameter.

Checking Fan Efficiency

Basic efficiency can be checked by means of the fan's performance curve. For the fan's operating point, the following must be known: fan diameter (ft), air flow (ft^3/min), total pressure (in. water), air density, design horsepower, and fan speed (rpm). Total pressure is the sum of static pressure and velocity pressure.

Actual static pressure (referred to actual density) is the total of all the pressure drops through the tower fill, or tube bundle, plus that caused by the tower or air cooler. Velocity pressure is the force needed to simply move the desired volume of air. Most fan curves show velocity pressure as a function of airflow for a fan.

With the foregoing known, total efficiency can be calculated: Total efficiency = [(actual total pressure)(actual ft^3/min)] / [(6,356)(actual horsepower)].

Total pressure and horsepower must be based on the same density, either actual or for the standard condition of 0.075 lb/ft^3. Check that operating speed is as specified for the fan curve. If in doubt, calculate the fan's efficiency from published performance data, keeping in mind that these data were probably collected under conditions as perfect as possible, and so will not reflect actual air-cooler or tower conditions. Although actual efficiencies will be less than curve efficiencies, fan curves can, and should, be the common denominator for comparing *fan* performance. *System* efficiency will be discussed later.

Total efficiencies of aerodynamically well-designed fans can range between 0.75 and 0.85. However, actual efficiencies rarely exceed 0.75 — not because of the fans, but because *system* efficiencies tend to be lower. This is why critical judgment must be exercised in evaluating horsepower requirements from ideal "curve" conditions. In some cases, it may be prudent to actually perform scale-model tests on a new tower configuration in a wind tunnel so as to define the "system resistance" line.

As a rule of thumb, fewer blades translate into higher axial-fan efficiency. A crowded inboard area creates hub turbulence, which lessens efficiency.

Fixed vs. Variable Pitch

Whether the pitch of fan blades is fixed or variable also affects energy consumption. Consider what happens when airflow is changed across a resistance (e.g., tower fill or tube bundle). Assume flow can be throttled, as by mechanically actuated louvers.

Take, for example, a 14-ft-dia. fan with 14-deg pitched blades operating at its design Point 1 (Fig. 2). The system resistance line (dashed line) is the locus of points obtained by summing static and velocity pressure losses vs. flows through the bundle. The pitch angle of 14 deg represents the delivery characteristic of the fan. Assume density remains constant and lower air flow results from decreased duty.

The louver throttles the airflow to control its outlet temperature. In Fig. 2, the points shown as 1, 2', 3' and 4' represent the total pressure output, and the horsepower consumed by the fixed-pitch fan. If the airflow were being controlled by means of a variable-pitch fan, instead of via throttling, the total pressure output would exactly match the system requirements, as shown by points 2, 3 and 4. This would significantly reduce horsepower requirements. In this case, the variable-pitch fan saves 26%, 51% and 73% of the horsepower required by the fixed fan at points 2', 3' and 4'.

A variable-pitch fan typically is controlled by a 3-15 psi instrument signal. In the event of signal failure, it automatically moves to full design pitch, functioning very much like a diaphragm-operated, spring-return valve operator. Its airflow usually ranges from design to zero, although it can provide up to 60% of its upflow in a negative, or down, direction. Fig. 3 shows the decrease in flow and power for a set of 45-deg travel. Note how horsepower falls to a minimum "no load" motor condition, then increases as work is being done in the negative airflow mode. (No-load losses are a function of iron and magnetic losses in the motor.)

Operating Cooling-Tower Fans to Save Energy

A study by the predecessor to the U.S. Dept. of Energy of a typical cooling tower in Houston, Tex., operating with 100 million Btu/h heat load and a 110°F-to-95°F cooling water range, revealed the following relative power consumptions:

No control, full speed	100% (base)
Single-speed, automatic on/off	65% of base
Two-speed, automatic on/off	59% of base
Automatic variable-pitch fan	50% of base

The tower was equipped with four fans, each run by a 25-hp motor. The results typify the power savings possible in Houston, where annual dry-bulb temperatures vary from 30°F to over 100°F.

If airflow were controlled with a fixed-pitch fan and a variable-speed drive, rather than by a variable-pitch fan, the results would be similar. Speed changing saves power because horsepower consumption is a cubic function of airflow — i.e., $hp_2 = hp_1 (cfm_2/cfm_1)^3$.

There is a slight advantage in controlling flow by varying speed rather than adjusting blade pitch angle, because the operating-point efficiency is maintained at all lower speeds. The efficiency of a variable-pitch fan declines with decreasing pitch angle. Because, in both cases, power consumption drops off dramatically, the actual savings would have to be carefully evaluated. However, power savings of 50%/yr, or greater, can be realized by replacing a fixed-pitch, continuously operating fan with one whose blade pitch or speed is automatically varied.

Improving the Drive System

Cooling-tower fans are generally gear-driven. The efficiencies of such drives fall between 96% and 98%. Not much can be done to improve these efficiencies, which depend on the type and class of gear.

Efficiencies of smaller fans driven by V-belts range between 92% and 95%, because of slippage. Newer synchronous drives can eliminate slippage and improve efficiency by 3-4%. Besides saving energy, these drives reduce the costs of periodically tightening drive belts and replacing worn sheaves. Lower tension also lessens motor-bearing wear.

J. Matley, Editor

Author

Robert C. Monroe is Manager of Fan Engineering for Hudson Products Corporation, Houston. He has 23 years of experience as fan engineer and two years experience as Manager R&D. He was engaged in the design of oilfield tools for 17 years. He holds a B.S.M.E. from the University of Houston. Formerly President of the Cooling Tower Institute and a member of its Board of Directors, he is a member of ASME and the Texas Society for Professional Engineers.

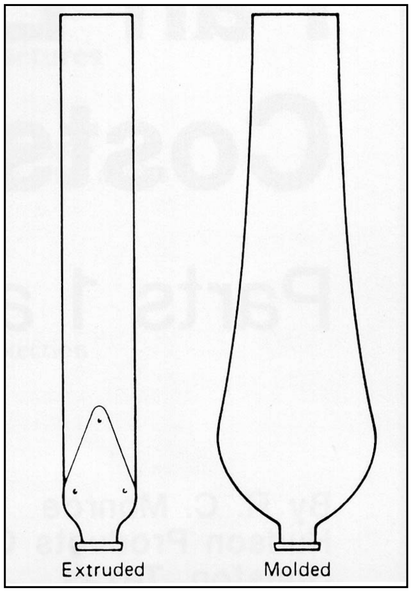

Figure 1 Extruded blades are uniform in shape; molded ones are irregular

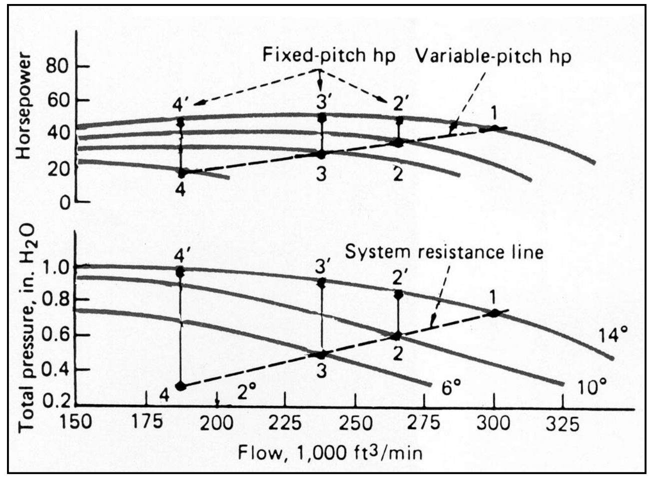

Figure 2 Flow control by varying blade pitch consumes less energy

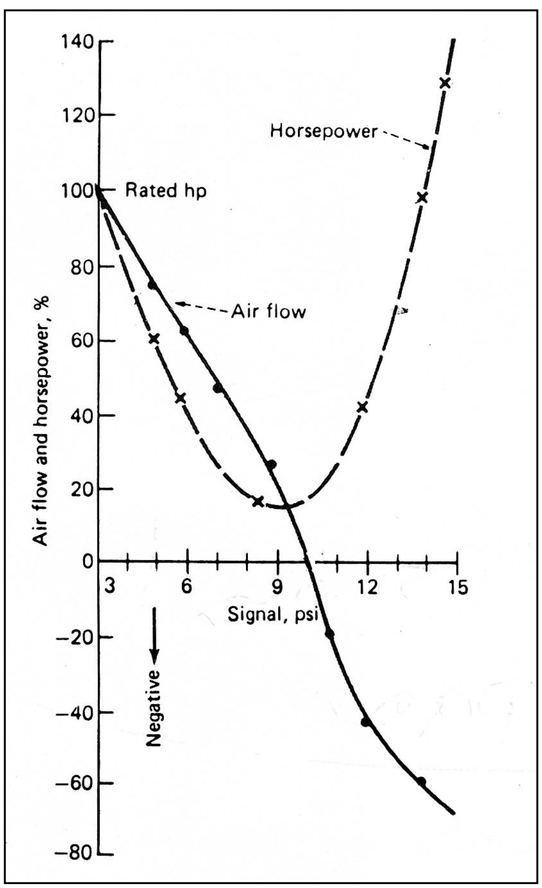

Figure 3 Signal controls blade pitch, changing air flow and hp

Part 2 — Minimizing Fan Energy Costs

The configuration of a cooling tower or air cooler can contribute to fan inefficiency. Fan efficiency can be undermined by too much tip clearance, poor inlet conditions, and excessive approach and discharge velocities.

Too much tip clearance allows discharge air to circulate back around blade tips and into the fan inlet. In Fig. 1, losses in efficiency and total discharge pressure are related to tip clearance for a small fan at one flowrate. The data are from tests with a 5-ft-dia. fan having an initial clearance of 0.074 in., which is increased to 0.464 in. (A clearance of 0.27 in. with a 5-ft-dia. fan is equivalent to 0.75 in. with a 14-ft-dia. fan; a clearance of 0.18 in. with a 5-ft-dia. fan is equivalent to 1 in. with a 28-ft-dia. fan.)

Inlet conditions are especially critical with forced-draft coolers. A total pressure loss of 26% and an efficiency drop of 18%, at a velocity pressure of 0.2 in. water, were revealed by tests on fans not equipped with an inlet bell on the fan ring. These losses, which are a function of velocity pressure, increase with higher air flows.

Excessive approach velocity can occur if the air-cooler fan inlet is too close to grade or the cooler is crowded by adjacent structures. Estimate approach velocity from the cylinder projected from the fan ring — i.e., the projected area. Therefore, approach velocity = $(ft^3/min)/\pi Dh$. With an air cooler, the approach velocity should not exceed 50% of the discharge velocity. Discharge velocity = $(ft^3/min)/(0.785D^2)$.

Cooling-tower approach velocity is measured at the inlet louvers. For counterflow towers, the tendency is to allow 800-1,200 ft/min through the louvers. For crossflow towers, the range is typically 400-600 ft/min.

Excessive discharge velocity wastes horsepower. This loss is gauged by fan velocity pressure. For fans of 6-40 ft dia., typical velocity pressures range from 0.1-0.2 in. water. If the velocity pressure approaches 0.3-0.35 in., a velocity-regain stack should be considered. The excessive velocity pressure recovered by a stack is used to lower pressure, and thus horsepower consumption. Typical horsepower savings are shown in Fig. 2. Cooling-tower fans of 16-ft dia. and larger are usually equipped with such stacks.

Fan Operating Point

The point where the system resistance line meets the fan performance line at the desired air flow defines a fan's operating point. At this point, the fan's output exactly meets the air-flow and pressure-drop requirements. Such a point will be represented by a specific blade pitch angle, actual ft^3/min and total pressure air output, and fan rpm. Fan rpm is usually set by noise restrictions, available gear or belt ratios, or the manufacturer's standard practice. Most large fans operate at a tip speed of about 11,000 ft/min.

The most obvious thing to check pertaining to operating point is whether the fan is "stalled." A stalled fan is like a cavitating pump: it consumes a lot of energy but produces no work. If a poorly operating fan is suspected of stall, try lowering blade pitch and see if the static pressure (measured with a water manometer) in the plenum changes. If the pressure does *not* change, the fan may be stalled. A stalled fan draws more horsepower with increasing pitch, but air flow and static pressure may actually decrease.

Because efficiency varies along pitch lines and with air flow, power can be saved and noise and vibration reduced by simply fine-tuning a fan's operating point.

At various speeds, calculate operating points using speed factors, and check efficiencies at these points on the fan curve. Speed factor = curve speed/actual speed. If, for example, the curve tip-speed was 12,000 ft/min, and the new speed is 10,000 ft/min, the speed factor = 12,000/10,000 = 1.2. (This can also be calculated in rpm. Tip speed = (rpm)(π)(fan dia.), or rpm = tip speed/πD.)

After calculating the speed factor, find the fan's new operating point: $(ft^3/min)_2 = (ft^3/min)_1$ (speed factor), or $(\text{total pressure})_2 = (\text{total pressure})_1 (\text{speed factor})^2$.

Using the speed factor, the fan speed can be changed at will. Each new speed and pitch angle will improve or worsen the efficiency of the original starting point. Plot total pressure vs. ft^3/min air flow for various pitch angles on the appropriate fan curve to obtain the horsepower requirement. Note that the pressure and flow work are the same at all the operating points, at which pitch angles differ. Many manufacturers refine cooling-tower design by such an optimization procedure before selecting an optimum gear ratio.

Variable Pitch vs. Variable Speed

The choice between automatically-varied fan pitch and variable fan speed depends on such factors as: (1) initial costs and payback period, (2) the availability of suitable fans, and (3) the number of fans performing the same duty.

Automatically-variable-pitch systems are simple and low-cost compared with variable-speed drive systems. A typical variable-pitch fan hub costs an additional $2,000 to $4,000 per fan. This includes the cost of a valve positioner for controlling pitch precisely. A typical variable-frequency type of variable-speed drive for a 40-hp motor can cost upwards of $10,000 per fan. The advantage of the latter drive is that any fan system can be retrofitted with it, normally without fan or gearbox modifications.

Payback period is calculated from power savings, which depend on the "degree days" at the site, power cost, required air flow vs. ambient temperature, and installation cost. The $/hp cost of a variable-speed drive drops significantly the larger the unit. For example, a 50-hp variable-speed drive may cost $12,500 (or $250/hp), but a 400-hp one may cost $46,000 (or $115/hp).

Since, for heat-transfer purposes, an even air flow across a bundle is desirable, the maximum number of fans should be operated by a single large controller, rather than by several smaller ones. A variable-speed drive can handle multiple motors, up to its rated capacity.

Variable-pitch fans range up to 20 ft dia. The limiting factor is the pneumatic power available for controlling blade pitch. Variable-speed drives can exceed 1,000 hp.

First cost for an automatically-variable-pitch fan — including optional hub and valve positioner, and control-signal and air lines — is typically $4,000. First cost for a variable-speed unit — including reversing magnetic starter, signal conditioner and wiring — is typically $4,500/fan, plus the drive cost.

A reversing magnetic starter is required because a variable-speed fan must be operated in reverse to overcome the natural draft of an air cooler and attain "zero" air flow.

The results of a study comparing costs and energy savings from automatically-variable-pitch and variable-speed fans are shown in Fig. 3. (Fig. 3 assumptions are: all fan motors are 30-hp, three-phase; power costs

$0.05/kWh; cost of variable-pitch fan includes controls and piping; and cost of variable-speed fan includes starter and signal controller.)

Note that the "break-over point" at which variable-speed fans offer more savings than variable-pitch fans is between 4 and 5 two-fan bays. This indicates that, with fewer than 8 fans, variable-pitch fans would be more cost-effective. Fig. 3 also shows that, with larger numbers of units, the variable-speed fan becomes more economical. In terms of horsepower, at less than about 300 total hp, the variable-pitch fan is the likely choice; at greater than 300 hp, the variable-speed fan should be selected. With either variable-pitch or variable-speed fans, power savings of 50%, or more, can be realized.

J. Matley, Editor

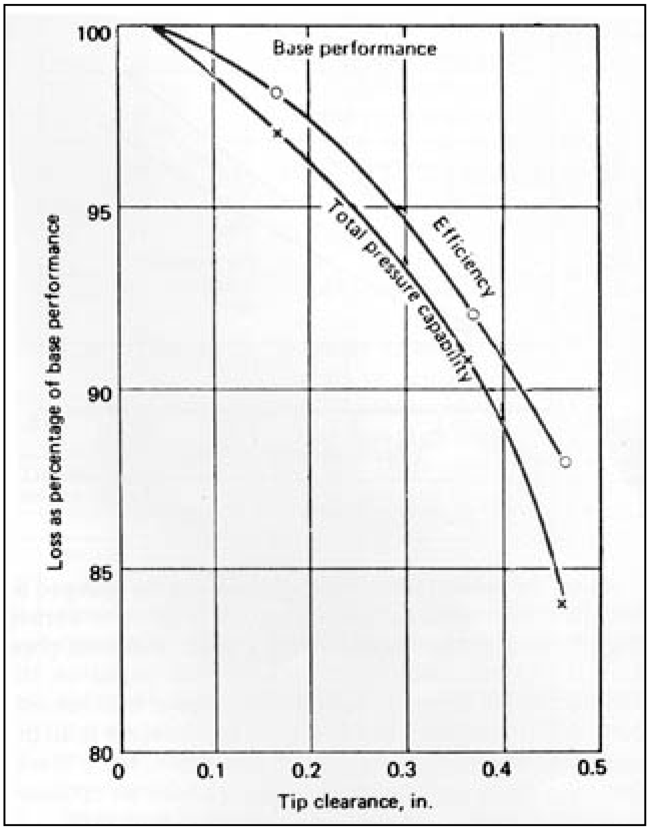

Figure 1 Efficiency declines with larger tip clearance

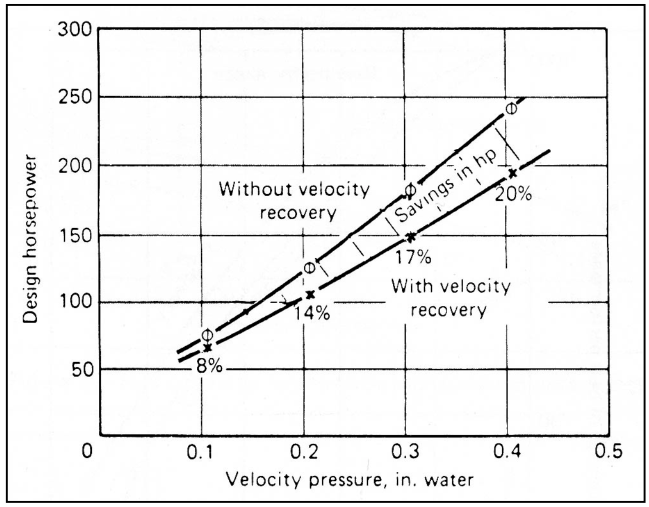

Figure 2 Velocity-recovery stacks save horsepower

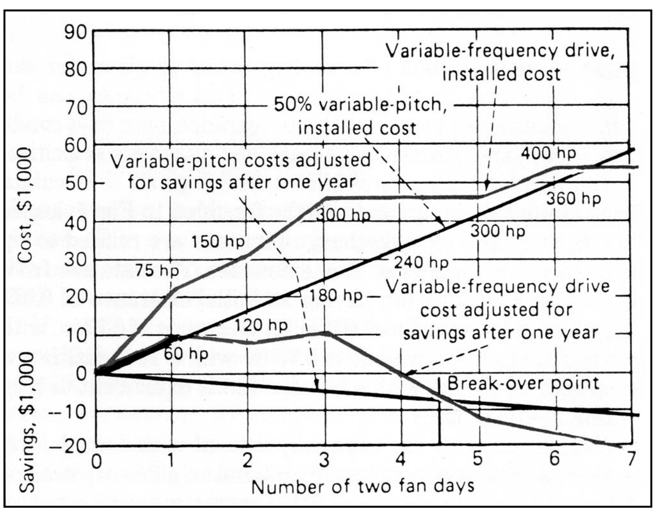

Figure 3 Costs and savings are compared for 50%-automatically-variable-pitch fans and 100%-variable-speed fans