

# Cooling Tower Energy

## AND ITS MANAGEMENT

### Forward

Because of the fuel shortages which began to manifest themselves in the early 1970s, the cost of energy began an asymptotic rise. Since that time, we have seen rising and falling interest in energy savings. The most recent focus is on global warming and energy generation's impact. The level of attention to energy savings has risen as a part of a broader green approach to design, including carbon footprint. SPX Cooling Technologies has maintained its attention to energy efficient equipment and technologies throughout this period, continuing a much older tradition of energy conscious innovation.

From the time that their use began to become noteworthy, in the 1930s, much of the effort devoted to research and development of the mechanical draft cooling tower went toward achieving reductions in pump head and fan power. This effort resulted in the development, by Marley, of the induced draft, crossflow cooling tower in 1938, and a new standard of energy utilization was established — against which all other cooling tower designs are still compared.

The purpose of this report is to make those comparisons in a straightforward and explanatory manner, and to describe certain devices and techniques by which the ancient physical laws can be utilized for reduced energy consumption on existing installations, as well as on new projects.

The principles and results described in this paper apply universally to cooling towers — regardless of their size and irrespective of the sources of the heat loads dissipated. Cooling towers neither know nor care what produces the imposed load. They merely react to that load — and do so in a predictable fashion. Air conditioning was chosen as the load source for this paper only because it is most generally understood — and because the load variations which routinely occur within a given system offer the opportunity to analyze numerous operating situations.

### Cooling Tower Energy Defined

Energy is consumed in driving the fan, or fans, necessary to achieve proper air movement through a cooling tower. The pump-head of a cooling tower also contributes to the energy expended in the operation of the condenser water pump. Obviously, therefore, manipulation of one, or both, of these power consuming aspects as a means of adjusting for changing loads or ambient conditions should have some beneficial effect upon the cooling tower's energy requirement.

Which of these two aspects to control, or whether they should be manipulated in concert, is a question that has plagued operators for a long time, probably as long as cooling towers have been operated.

Deciding whether to attempt control of either air flow or water flow, as well as how to go about it, will depend primarily upon specific characteristics and limitations of the condenser/chilled water system. It also requires some understanding of the most-utilized types of cooling towers. **Figure 1** and **Figure 2** depict two basic types used for air conditioning service and for small to medium process loads.

### Tower Types

In a counterflow cooling tower, air moves vertically upward through the heat transfer media (fill) counter to the downward fall of water. In a crossflow cooling tower, movement of air through the fill is across (perpendicular to) the direction of water fall. Although both types of towers may be equipped with either propeller fans or blower fans, and the fans may be located either on the intake side to force air through the tower, or on the discharge side to induce air through, the definitive configurations indicated in **Figure 1** and **Figure 2** represent two of the most common arrangements utilized for the air conditioning and smaller process markets. Each type has distinctly different fan power and pump head energy consuming characteristics, which will be discussed in turn.



**FIGURE 1** Counterflow cooling tower with Centrifugal (Blower) Fan: Forced-Draft Configuration



**FIGURE 2** Crossflow cooling tower with Axial (Propeller) Fan: Induced-Draft Configuration

## Fan power requirements

Because of the nature of blower fans, when operating against the relatively low static pressures encountered in cooling towers, they characteristically require at least twice the applied horsepower of propeller fans to deliver equivalent air flow. Conversely, they are able to move predictable quantities of air against very high static pressures. They are also very compact and relatively inexpensive to adapt to cooling towers of commercial quality and limited size.

In an energy-conscious world, it is logical to assume that the use of blower fans will gradually become limited to those systems where pressures develop that would exceed the capabilities of a propeller type fan. In the realm of cooling towers, this would restrict their application to those towers which are designed to operate indoors, where supply and exhaust air must serve the tower through duct work. In recognition of this probability, *all* major cooling tower manufacturers currently market towers equipped with propeller type fans. However, because a large number of blower fan, counterflow towers are currently operating, a comparison of the two types of towers must be undertaken to establish the relative power levels, on which any future energy savings will be based.

**Table 1** illustrates this comparison of fan power requirements, as well as the subsequent pump head comparisons that will be made. Both types of cooling towers were selected for a 400 ton air conditioning application, requiring that each tower be capable of cooling 1200 gpm of condenser water from 95°F to 85°F at a 78°F design wet-bulb temperature.

Since the curves offered in this report denote load variations in terms of percent, *please keep in mind that the beginning power level of the blower-fan tower is twice that of the propeller-fan tower.*

## Pump Head Requirements

A study of **Table 1** reveals not only a 2:1 differential in fan power requirement between the two types of cooling towers, but also indicates a discrepancy of 2.3:1 in required pump head for the selections chosen. This is because of the distinctly different modes of water distribution that are necessary for the respective types.

The relatively enclosed nature of a counterflow cooling tower, coupled with the need for vertical air movement through the fill – **Figure 1**– dictates that distribution of the water be accomplished by means of a header and branch-arm piping system, terminating in multiple spray nozzles, critically spaced to achieve proper water coverage over the fill. Typically, these spray systems are designed for a water pressure at the inlet header of about 5 psig which, added to the static lift required from the cold water basin operating level to the inlet header elevation, resulted in the 23'-0" pump head indicated in **Table 1**. Furthermore, because the thermal efficiency of a counterflow tower can be increased either by elevating the spray system, or by increasing its operating pressure, with no change in fill volume or air flow, there is considerable motivation for the counterflow designer to achieve improved thermal ratings at the expense of pump head.

Conversely, the combination of "side-mounted" fill banks and horizontal air travel typical of the crossflow cooling tower – **Figure 2** – allows the use of open hot water basins located above the fill banks, out of which the water flows by gravity through metering orifices installed in the floors of those basins. Pump head consists of the static lift to the center of the inlet pipe, plus an approximate 1'-0" loss either in a downturned elbow, or in the flow-regulator valves that may be used to balance water flow to both hot water basins. Without increasing fill volume or air flow, increasing the elevation of the distribution system offers no benefit in thermal

Cooling Tower Type	Operating Fan Motor Horsepower	Fan Motor kW note 1	Tower Pump Head ft note 2	Additional Pump Motor kW note 3	Total Operating kW
Counterflow with Blower Fan	40	32.4	23	6.9	39.3
Crossflow with Propeller Fan	20	16.2	10	3.0	19.2

**TABLE 1** Comparison of a forced-draft counterflow cooling tower versus an induced-draft crossflow cooling tower for 400 tons.

- 1 Fan and pump motor efficiencies assumed to be 92%
- 2 That portion of total pump-head attributable to the cooling tower – sum of static lift plus losses in tower's internal water distribution system
- 3 Pump efficiency assumed to be 82%

performance capability, but it does increase the cost of the tower. Consequently, the crossflow designer's motivation is to decrease pump head as much as possible.

Regardless of which type tower exists on a present installation, or is chosen for a new project, once the water distribution system is installed for a particular flow rate, proper tower efficiency requires that flow rate to remain constant. Unless operating characteristics of the process dictate it, adjustment of the water flow rate should not be used as a means of varying a cooling tower's capacity. Such manipulation is not only self-limiting from a hydraulic and thermal performance standpoint, it also shows some lack of perspective in the approach to energy efficiency.

Let's look at the hydraulic and thermal limitations first.

Pressure required at a nozzle of given size varies approximately as the square of the flow quantity through the nozzle. A 10% increase in flow will require about 21% greater pressure. A 10% decrease in flow will decrease the required pressure by about 19%. In the direction of increase, hydraulic limitations manifest themselves quickly in the counterflow cooling tower by excessive pipe pressures, a measurable increase in drift loss, and in nozzles being blown off the ends of the branch arms. In the crossflow design, it is usually evidenced by excessive splash out and/or overflowing of the hot water basins.

It is in the direction of decreasing flow that thermal performance degradation is most noticeable. Water exits a spray nozzle in a conical pattern, and the nozzles are spaced for sufficient overlap of these "cones" to assure full water coverage on the fill. With a decrease in pressure, the cones of spray begin to "droop", the overlapping effect of the sprays is lost, and the intervening areas of fill are devoid of water.

The debilitating effect that this has upon the tower's thermal capacity goes well beyond the proportion of fill that is dry. Since air will naturally take the path of least resistance, it will concentrate its movement through those fill passages which are dry, providing very little cooling effect in the remaining wetted areas. In crossflow cooling towers, the head of water in the hot water basins may reduce to the point where peripheral orifices see little or no flow, and the net result is approximately the same as happens in the counterflow cooling tower.

As regards the energy efficiency perspective, a glance at **Table 1** will reveal that the energy demand created by the cooling tower

fan is 4 to 5 times that which is caused by contributory pump head. Therefore, manipulation of air rate (and fan power) is a far more energy-effective means of regulating thermal capacity than control of water rate could ever hope to be.

## Tower Operation

Having established that some means of air-side (fan) control is the proper method to use for varying the thermal performance level in a cooling tower, let's now look at the *necessity* for such control. Considering some of the extravagant claims that have been made concerning the capabilities of cooling towers operated at reduced fan loadings, and the exponential savings in power resulting, a normal reaction might be to stampede off in that direction without a thorough investigation of the net effect upon an overall system. This is not to say that cooling tower fan control is unwarranted. Without doubt, it is extremely beneficial when properly applied. However, misapplication can have a detrimental impact upon the ultimate energy costs of some air conditioning systems.

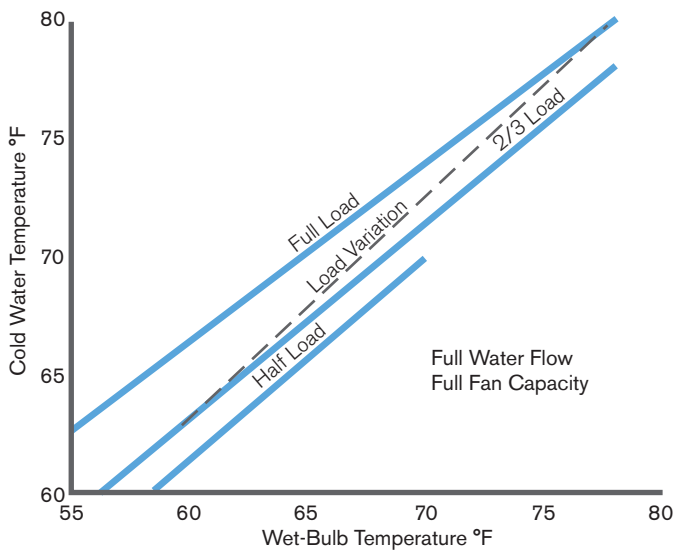
Ignoring, for the moment, the physical limitations that a cooling tower's water distribution system imposes on any attempt at water-side thermal control, the mere fact that the tower's fan constituted its greatest potential area of power consumption made air-side control the logical choice – that is, if energy use reduction is the criterion for judgement.

Extrapolating this logic to encompass a total air conditioning system, for example, one must conclude that the compressor is the system component on which to concentrate. Since its design power requirement is 10 to 15 times that of the cooling tower, any manipulation of the tower that does not come to the aid of the compressor must be considered to be wasted.

Barring any performance-degrading influences, **Figure 3** shows the cold water temperature capability of the example cooling tower at various wet bulb temperatures and load conditions, based upon full water flow and full fan operation. The diagonal dashed line represents what might be expected to happen to the overall air conditioning system load as outside ambient reduces, and indicates how the cooling tower's cold water temperature will track that load.

At nominal flow conditions, the design power requirement of a compressor is based upon the applied heat load, as well as a particular condenser water temperature. As load reduces, so does compressor horsepower and, for purposes of this report, that relationship will be considered to be linear. *Some* air conditioning





**FIGURE 3** Cold Water Temperature Variation with Wet Bulb and Load

units are designed such that compressor horsepower also reduces with a lowering of condenser water temperature, averaging approximately 1% reduction in horsepower for each degree that the cold water temperature is lowered. Operational characteristics usually limit this benefit to about 20% (with 85°F condenser water as a design) or what might be expected to accrue at a cold water temperature of 65°F.

Of the two factors which affect compressor horsepower (load and condensing temperature,) the cooling tower can only influence the latter. Load is established by the building being served by the air conditioning system, essentially irrespective of cooling tower operation. Looking again at **Figure 3**, by the time that the ambient has reduced to the point where the cooling tower is capable of producing 65°F cold water, it is assumed that the building load will probably have reduced to about 1/3 of design. Although the cooling tower has had nothing to do with the resultant savings of about 133 bhp on the compressor ( $1/3 \times 400 = 133.3$ ), it *does* benefit from the reduced load having allowed it to achieve 65°F cold water sooner than it would have at full load, and it takes full credit for the *additional* compressor power reduction of approximately 53 bhp ( $2/3 \times 400 \times 20\% = 53.3$ ) brought about by the colder water temperature.

In this case, continuous operation of the cooling tower fan resulted in a maximum power savings of 33 bhp (53-20) for the system served by the propeller fan tower, or 13 bhp in the case of the blower fan cooling tower.

The fact that actual energy cost savings are a function of time warranted an integration of this effect of cold water temperature on compressor power, along the line of expected heat load variation. The results were that compressor power reduction would have begun to exceed propeller fan power consumption at or below a wet-bulb temperature of approximately 72°F, whereas equality on blower fan operation would not have been reached until about 64°F wet bulb.

This takes on meaning only when you consider the operational time span that this temperature spread can represent. In Chicago, for example, considering only typical comfort air conditioning hours during working days in the months of May through September, this can mean a difference of some 600 hours. Indeed, in other more

southerly cities, the lower of these two wet bulbs may seldom be reached during summertime daylight hours.

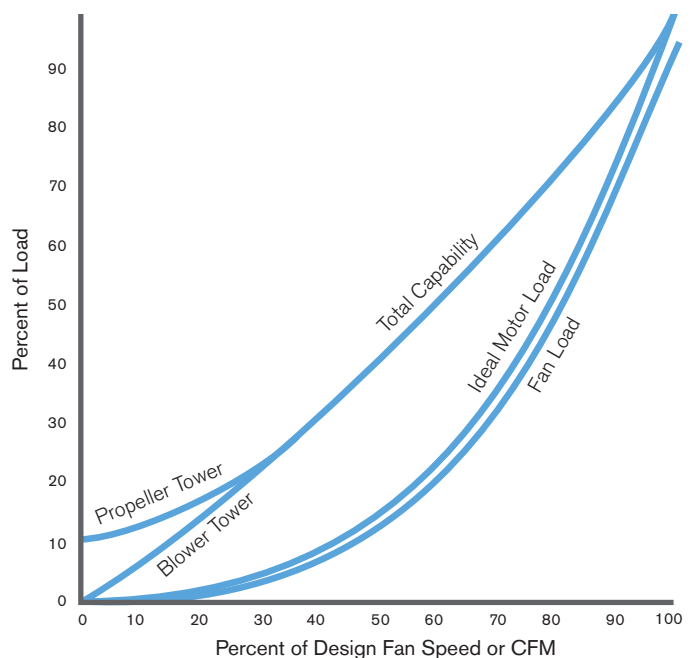
## Fan Laws

Analyzing what will happen if the capacity of the cooling tower fan is accurately varied to maintain a particular cold water temperature during a diminishing load and ambient requires understanding of some basic fan laws, which apply to both propeller and blower type fans. Because these laws depend upon constant fan efficiency, speed control reflects the method of capacity control that most nearly approximates that requirement.

1. The capacity (CFM) of a fan varies directly as the speed ratio. (A fan turning at 50% of design speed will move 50% of its volumetric air flow.)
2. The horsepower required to drive the fan varies as the cube of either the speed or the capacity ratio. (a fan turning at 50% of design speed will require 12.5% of design input hp)
3. At constant capacity and speed, horsepower varies directly with air density. (if air density increases 10%, so will fan hp)

The lower curve in **Figure 4** reflects the exponential change in *fan input* horsepower requirements with any variation of its air flow capacity. You will notice that the cooling tower motor load curve is slightly higher than the fan curve. This is because of the mechanical losses incurred in the transmission of power from the motor, through the Gear drive or V-belt drive system, to the fan.

It must be understood that this motor load curve is a measure of the *output* requirement of the motor, and does *not* necessarily reflect the comparative cost of running a given motor at varying speeds and loads. Since the efficiency and power factor of most motors change somewhat with load and rotational speed, a “cost of operation” curve could be expected to track slightly higher than that shown, with a greater speed at the low end than at full capacity.

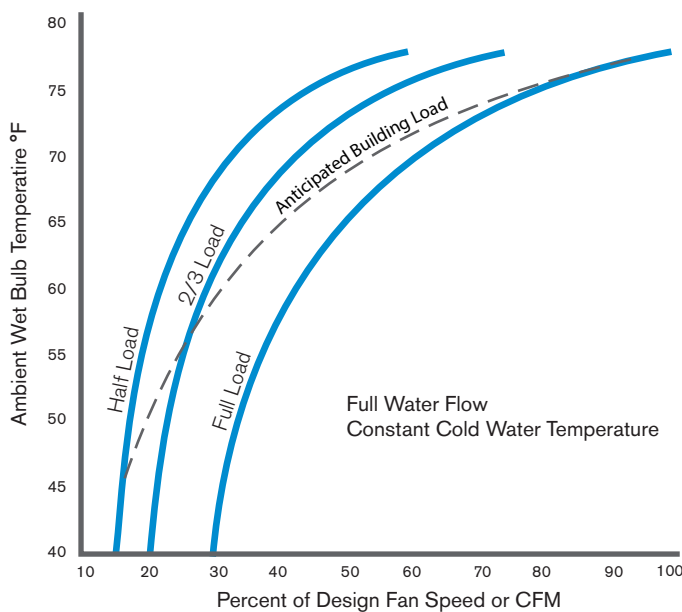


**FIGURE 4** Effect of Fan Speed on Fan Horsepower and Tower Capability

The top curve in **Figure 4** is meant to indicate the capability of the cooling tower at various conditions of fan capacity, but at constant water flow, ambient, and cold water temperature. The split at the lower end of this capability curve is brought about by the fact that an induced-draft crossflow tower achieves approximately 10 to 15% performance at zero fan speed by virtue of natural air movement through the tower. The enclosed nature of the forced-draft blower tower precludes this advantage.

With the help of **Figure 5** we can now do a quick analysis of what energy consumption level would have been reached in the previous example if we had chosen to maintain 85°F cold water temperature, instead of allowing the tower to cool the water as much as possible.

**Figure 5** shows that the building load was assumed to reach  $\frac{2}{3}$  of design at a wet bulb temperature of about 57°F, and indicates that the required fan capacity at that condition would have been about 27%. **Figure 4** shows that the motor output load at 27% fan capacity is approximately 4% of design. Considering the propeller fan tower – **Table 1** – we see that the ultimate motor output reduction would have been 19.2 hp (20 x .96). However, because the cold water temperature was not allowed to diminish, the previously indicated 53 hp reduction in compressor power would not have been realized, and a potential savings of 33.8 hp (53 - 19.2) would have been lost. A similar analysis for the blower tower shows that a potential savings of 14.6 hp would be lost through premature fan control.



**FIGURE 5** Fan Capacity Requirement at Varying Load and Ambient

## Effective Capacity Control

Obviously, on a cooling tower which is serving a process which benefits from colder water temperatures, any attempt at fan manipulation should be delayed until such time as the cold water temperature reaches a level that results in optimum system energy usage. Although that level was 65°F in the text example, it varies with system design and should be checked out with the equipment manufacturer. Some refrigerant compression systems, and the

majority of absorption systems, permit very little deviation from the design condenser water temperature. In those cases, effective control might begin at a temperature no lower than about 5° or 6° below design.

Once having determined the level at which control of the cooling tower's cold water temperature will be most conducive to total system power reduction, several energy-efficient methods by which to achieve this control are available to the operator. The most-used methods follow in general order of increasing effectiveness:

## Fan Cycling

Most operators have some familiarity with fan cycling, if only on an intuitive basis. Any time that the tower is capable of producing a water temperature that is appreciably colder than the control level, they see the opportunity to shut off the fan for the period of time that it takes the cold water temperature to build back up to that control point. In many cases, a thermostatic control device is used to accomplish this cycling automatically.

Automatic control devices can work well, provided that sufficient temperature spread is available between their make and break levels to insure that the fan motor insulation does not become overheated by an excessive number of starts per hour. Considering the fan sizes encountered on relatively small cooling towers (usually 120" diameter or less), five full starts of a single-speed motor per hour would not be considered excessive. This would also be equivalent to about seven speed changes per hour on a two-speed motor, from zero to half-speed, or from half-speed to full-speed. Larger fans may allow fewer starts.

Limitation of the number of motor starts, or speed changes, per hour is accomplished by setting the make level of the control device about 2° above the desired temperature from the tower, and setting the break level as far below that temperature level as the operating requirements of the system will tolerate. The amount of time that the fan will be off can be calculated by the following formula:

$$T = \frac{V \times 8.33 \times D}{C \times 1.25 \times K}$$

Where:

T = Period of time (min) fan is shut down.

V = Total amount of water (gal) in condenser water system, including cold water basin of cooling tower.

8.33 = Nominal weight of water (lb/gal).

D = Difference (°F) between minimum and maximum allowable cold water temperatures (make and break settings on controller).

C = Chiller operating load (Btu/min) at time of calculation (= 200 x actual tons).

1.25 = Factor to convert chiller load to condenser (cooling tower) load by the addition of heat of compression. For absorption systems, this factor is 2.5.

K = Factor reflecting tower's ability to achieve some cooling with fan off. K for blower tower = 1; K for crossflow, propeller fan tower = 0.9.



For example, if the controller were set to stop the fan at 80°F and restart at 87°F, **Figure 3** assumes that the load factor will be about 90% (360 tons) at the time that the cooling tower is first capable of producing 80°F cold water, and the wet bulb temperature at that time will be about 72°F. If the condenser water system volume were 6,000 gallons, those values inserted into the formula would indicate a fan off-period of 3.9 minutes for the blower fan cooling tower, and about 4.3 minutes for the crossflow cooling tower. Furthermore, although not calculable from the curves, it can be stated that with the fan restarted at 87°F cold water (at the combination of 360 tons and 72°F wet bulb) approximately 15 minutes will have elapsed before the water temperature from the cooling tower reaches 80°F, and the total of approximately 20 minutes between starts will have satisfied the limitation on number of starts per hour.

As a comparative example, let's assume that the wet-bulb temperature continues to drop, ultimately reaching 65°F, at which time the chiller load will have reached 70% (280 tons). At that time, the formula indicates that the fan off-period will have become 5.0 minutes for the blower fan cooling tower, and 5.6 minutes for the crossflow cooling tower. However, because the combination of reduced load and reduced ambient has drastically increased the cooling tower's capacity, the cold water temperature leaving the cooling tower when the fan is restarted will very quickly reduce to 80°F, probably in no more than a minute's time. Obviously, such a situation will cause the motor to "short cycle" about once every 6 minutes, running the risk of overheating the motor insulation.

As can be seen in these two examples, the number of motor starts tends to increase as ambient air temperature reduces, and soon will exceed the maximum allowable starts per hour. Because of this, the previously discouraged utilization of *water-side control must be brought into play* and, if intelligently accomplished, can still result in significant fan energy savings.

In addition to the previously mentioned temperature control device to start and stop the fan motor, two other pieces of equipment would be required: 1—a temperature-actuated three-way valve in the hot water return line to the tower that will divert a proportioned amount of hot water directly into the cold water basin, and 2—a time delay switch at the starter for the fan motor.

Given the problem of maintaining condenser cold water min/max temperature levels at 80°F and 87°F respectively, the operating sequence would be somewhat as follows:

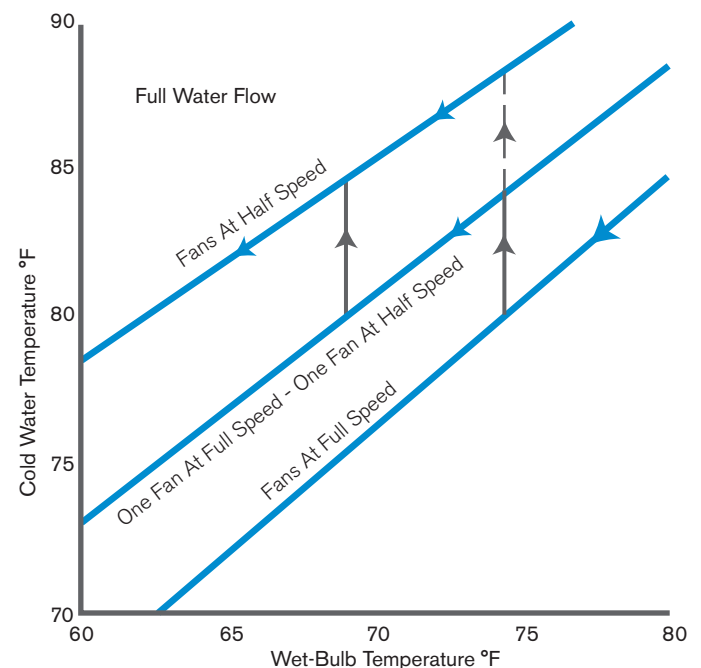
1. With the fan off, the water temperature would gradually rise to 87°F, at which time the fan control device would start the fan.
2. With the fan running, the water temperature would cool to 81°F, at which time the fan control device would signal the fan to stop. However, the time delay switch would not accept that signal until such time as 12 minutes have elapsed since the time the fan was previously stopped (instant of de-energization).
3. With the fan continuing to run, the tower will continue to cool the water, and the three-way valve will actuate at 80°F to maintain that temperature to the condenser.
4. As soon as the time delay has been satisfied, the fan will stop and the three-way valve will begin to restore full flow to the tower's water distribution system.

The time delay should be wired into the control circuit and, of course, should not obstruct normal manual shut-off or emergency controls, such as vibration limit switches, etc. Furthermore, *the use of proportional diversion valves on towers with single-speed fan motors is not recommended if the equipment is to be operated during freezing weather*. In such situations, the control flexibility afforded by two-speed motors should be considered mandatory.

The addition of two-speed fan motors on a cooling tower aids tremendously in both control flexibility and energy-use reduction. Although **Figure 6** is drawn to indicate the performance of a two-cell cooling tower with two-speed motors, the upper and lower curves are representative of what happens on a single-cell cooling tower with a two-speed motor operating at half-speed and full-speed respectively.

As can be seen, if a single-fan cooling tower operating at full-speed in a diminishing ambient condition is reduced to half-speed when the water temperature reaches 80°F, the cold water temperature will begin to rise toward the upper curve. However, because the tower is continuing to achieve effective cooling with half of its design air flow, the time that it takes for the water temperature to reach 87°F will be increased considerably beyond what would be indicated by the formula which, of course, represents a fan off condition. Indeed, in a situation of reducing ambient, an 87°F water temperature level might never be reached, and the fan would continue to operate indefinitely at half-speed. During that time, the motor would be operating at approximately 15% of its design horsepower, as seen in **Figure 4**.

**Figure 6** was drawn for a two-cell cooling tower in order to demonstrate the increased control flexibility afforded by multiple cells and fan speeds. The utilization of this flexibility merely requires a logical extrapolation of the control sequencing previously discussed, bearing in mind that the proportional diversion valve will probably not be required, and that the allowable number of starts or speed changes can be doubled on a two-cell cooling tower by alternating the motor to be controlled.



**FIGURE 6** Reduced Load Performance of a Two-Cell Cooling Tower Equipped with Two-Speed Motors

Operating Situation	Hours of Operation	Average kW Usage	Propeller Fan Energy kWh	Blower Fan Energy kWh
Constant Operation at Full Capacity	1202.2	P = 16.2 B = 32.4	19475.6	38951.2
Single-Speed Fan Cycling	P = 765.3* B = 852.7	P = 16.2 B = 32.4	12397.3	27627.5
Two-Speed Fan Cycling	P = 1132* B = 1146	P = 4.3 B = 8.55	4867.6	9798.3
Variable Controls at Constant Speed	1202.2	P = 2.72 B = 5.44	3270	6540
Variable Speed Control	1202.2	P = 1.99 B = 3.98	2392.4	4784.8

**TABLE 2** Comparative Energy Usage with Various Methods of Control.

\* The propeller fan will operate slightly fewer hours in these modes because of the crossflow tower's cooling effect with the fan off

## Fan Capacity Variation At Constant Speed

In order to achieve harmony between the operational requirements of the chiller and the desire to conserve cooling tower fan energy, control of fixed-speed fans, be they single-speed or two-speed, would seem to be limited to a series of "notches" — **Figure 6** — resulting in either an over achievement or an under achievement of the actual goal being sought. How much simpler the control effort would be if the fan capacity could be varied such that the tower would track the load at a constant cold water temperature.

The means by which to accomplish this result currently exist in the form of damper mechanisms for blower-type fans (not recommended for operation in freezing weather), and with Variable Frequency Drives (VFDs).

## Progressive Variation Of Fan Speed

Theoretically, the load of a cooling tower fan can be made to track the curve indicated in **Figure 4** exactly by the use of a Variable Frequency Drive. Since the speed of an AC motor is a direct function of input frequency, the ability of these controllers to infinitely vary frequency results in equal capability to infinitely vary the fan speed. There is one thing which restricts total variability on cooling tower application, however, and that is a characteristic of the fan itself.

The need to restrict overspeeding is obvious, and should require no further explanation. In addition, most fans have at least one critical speed that occurs between 0 and 100% of design RPM and some have several. Typically, fans are designed such that their critical speeds do not coincide with the RPMs that will be produced by normal motor speed changes. They are also, of course, designed to minimize the effect of any critical speed. Nevertheless, it would be foolhardy to run the risk of protracted operation at a critical speed and, therefore, it becomes necessary to predetermine the critical speeds, and to prevent corresponding frequencies from being utilized.

Fortunately, most VFDs permit avoidance of selected frequencies, so that critical speeds will be transited without risk of extended operation at those speeds. Avoiding critical speeds, however, reintroduces some miniature "notches" in an otherwise constant cold water temperature line and, to avoid "hunting", requires that some tolerance be designed into the control mechanism.

## Comparative Results

In order to determine the relative merits of the various methods of capacity control as regards energy usage, the example 400 ton air conditioning system was assumed to be operating five days per week; from 7 am to 6 pm from the first of May to the end of September in the city of Chicago. Load was assumed to vary as indicated in the figures, and attempted control was as described in the text.

The "constant operation at full capacity" section of **Table 2** represents no attempt at fan control whatsoever, and is indicative of usage on a system where the chiller rewards colder water with reduced power consumption. At the subject location, and for the time period under consideration, 67°F wet bulb temperature dominates the mid-range of ambient conditions and is considered to be a weighted average. At the coincident 75°F cold water and 83% load factor, the compressor kW savings, corrected for motor efficiency, is 27 kW, or 32,459 kWh for the time of use. This exceeds propeller fan demand by some 13,000 kWh.

The remaining sections of **Table 2** tabulate anticipated fan energy consumption utilizing the control methods that have been discussed. Since the values indicated will vary with load factor, nighttime use, year-round use, control settings, geographic location, etc, they are far more general than absolute. Relative to each other, however, their proportions are realistic.

## Innovative Thinking

Although some of the potential energy savings indicated in this paper require sophisticated auxiliary equipment to achieve the goal sought, others depend merely upon utilizing the cooling tower in a slightly different manner. One such innovation that the formula may have suggested to the imaginative reader is the possibility that a cooling tower operating at half fan speed during periods of low load, and low ambient (such as nighttime), could sufficiently cool the water in a large reservoir tank to carry the major portion of a succeeding high-ambient load.



## Comparative Costs

Because the costs of various control methods are so sensitive to size of unit, market conditions and escalation, as well as the condition and accessory equipment of an existing tower, no effort has been made to develop typical costs, even of the “ball park” variety. Given the expectation of a reduction in energy use, the user should obtain the absolute cost of appropriate equipment from his control supplier, or from SPX Cooling Technologies.

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