Number: H-002 Date: December 1982

TECHNICALREPORT

SPX
COOLING TECHNOLOGIES

SUBJECT:

The Application of Cooling Towers for Free Cooling

BACKGROUND

Considering the amount of recent coverage in trade journals concerning the use of cooling towers to achieve "free cooling" without the expense of operating the chiller, uninformed users may get the impression that such use represents ***new*** technology. Also, since many of the releases have come from a limited-scope cooling tower manufacturer, one might easily be led to believe that forced draft, centrifugal blower, counterflow type cooling towers are the best answer to a free cooling problem.

Neither of these impressions are true.

Without searching archives too deeply, the use of Marley towers in free cooling applications extends back at least 30 years, during which time Marley-manufactured crossflow, counterflow, induced draft, and forced draft towers were so applied. Therefore, if there is a storehouse of knowledge and experience on free cooling it belongs to Marley, and it was compiled from the use of various types of cooling towers—***not*** just one type.

Based upon the experience gained from having manufactured all types of cooling towers a previous article was written entitled "Cooling Tower Energy and its Management." Much of the operational-type information contained in that report applies equally to the application of towers on free cooling, and it is recommended that the reader obtain a copy to enhance full understanding of the present paper.

THE CLASSIC CHILLED WATER SYSTEM

Air conditioning and refrigeration systems, as well as numerous industrial processes, require cold water at a temperature well below that which a cooling tower is capable of producing during a normal summer. In those cases, various types of chilled water systems are utilized, the most common of which is depicted in Figure 1 for purposes of discussion.

In this system, a chilled water circuit picks up heat from the air conditioning or process load and transfers this heat to vaporize a refrigerant flowing through the evaporator. Having lost its added heat, the chilled water is thereby cooled for its return to the load source. Meanwhile, the refrigerant vapor is pressurized within a compressor (adding the heat equivalent of compression work) and flows to a condenser, where the total added heat is transferred to the condenser water circuit. Ultimately, of course, this total heat is rejected to the atmosphere by the cooling tower, and the water is cooled for its return to the condenser.

Important to note is the fact that the load rejected by the cooling tower exceeds the actual process load by the amount of heat (or work) necessary to affect the refrigeration function of the chiller. In the refrigerant compression system shown, this added "heat of compression" causes the tower to have to dissipate approximately 25% more load than that actually imposed by the process. Therefore, although a "ton" of refrigeration (by definition) is equivalent to a heat dissipation rate of 12,000 Btu/hr, cooling tower designers for this type system routinely think in terms of 15,000 Btu/hr/ton.

Similarly, in an absorption chiller system the cooling tower would also be required to dissipate the heat added to effect absorption and release of the refrigerant vapor. In that case, the load at the tower would be about 2.5 times the load imposed by the process, or approximately 30,000 Btulhr/ton.

The flow rates and temperatures indicated on Figure 1 are typical of those encountered in an air conditioning system operating at full load in summertime conditions. Note that the usual pumping rates are 3 gpm/ton in the condenser water circuit, and 2.4 gpm/ton in the chilled water circuit. These pumping rates are reflective of the aforementioned difference in heat content, and result in a 10°'F water temperature rise in each loop.

As a general rule, process loads do not require a temperature as low as that indicated on Figure 1. Typical low temperature processes might want temperatures between 55°F and 70°F and, for purposes of illustration in this paper,

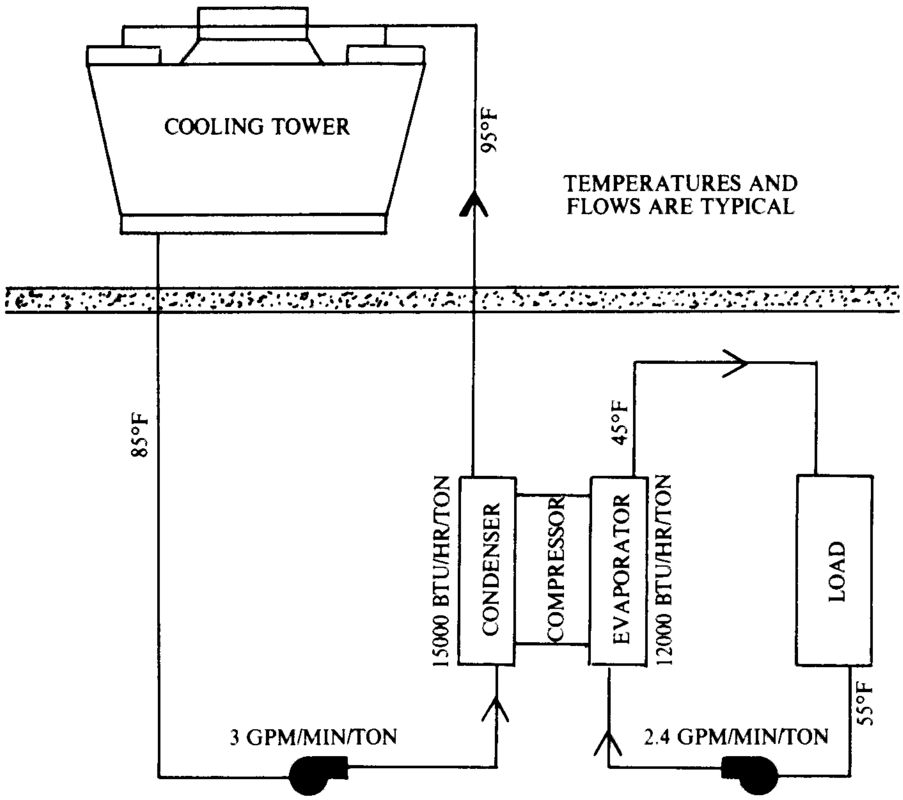

Figure 1 Refrigerant Compression Chiller System.

57°F process chilled water temperature has been arbitrarily selected.

The primary reason why 45°F chilled water is required for summertime air conditioning is to achieve dehumidification of the "conditioned" air. Oftentimes, it is then necessary to reheat the air to a temperature compatible with human comfort, or to a level that may be required for such considerations as laboratory temperature control. Since air's natural humidity reduces with its temperature, considerably less dehumidification (perhaps none at all) is required in the fall, winter, and spring than was needed in the summer. Accordingly, higher chilled water temperatures are usually permissible in the "off-seasons" which, as will be seen later, works to the benefit of free cooling.

The operating cost of providing an added heat source for an absorption system is both obvious and significant. In a refrigerant compression system, the operating expense is in the need for a continuous power input to the compressor of approximately 1 HP/ton. Utilization of the cooling tower for free cooling allows these operating costs to be avoided during a substantial portion of the year.

FREE COOLING DEFINED

Figure 2 shows how the cold water temperature produced by a cooling tower operating in conjunction with a refrigerant compression type chiller responds to wet-bulb temperature and load. This same relationship is depicted in Figure 3 for a tower operating with an absorption system. In either and all cases, as wet-bulb and/or heat load reduce, so does the temperature of the cold water coming from the tower.

With sufficient reduction in the wet-bulb temperature (as will occur seasonally and, in some locations, daily) there will obviously come a time when the cold water temperature produced by the tower is low enough to satisfy the requirements of the process or air conditioning system without assistance by the chiller. At those times, with a properly equipped and arranged piping system, the cooling tower water could serve the load directly — and the expense of added heat or compressor operation could be avoided.

That is the purpose of free cooling — to avoid the expense of unnecessary energy use — and the methods by which to accomplish this purpose will be discussed shortly. First, however, let's consider opportunity.

OPPORTUNITY FOR FREE COOLING

Depending on the variability of the load and the actual water temperatures required by the process, one can gather from Figures 2 and 3 that the wet-bulb temperature must depress to approximately 50°F, or below, before the use of free cooling becomes feasible. Consideration of free cooling, therefore, tends to be quite geographical in nature. In northern latitudes, where annual temperature variations are significant, the opportunity for free cooling may routinely exceed 75% of the total yearly operating hours. Conversely, in more southern climes a wet-bulb temperature af-

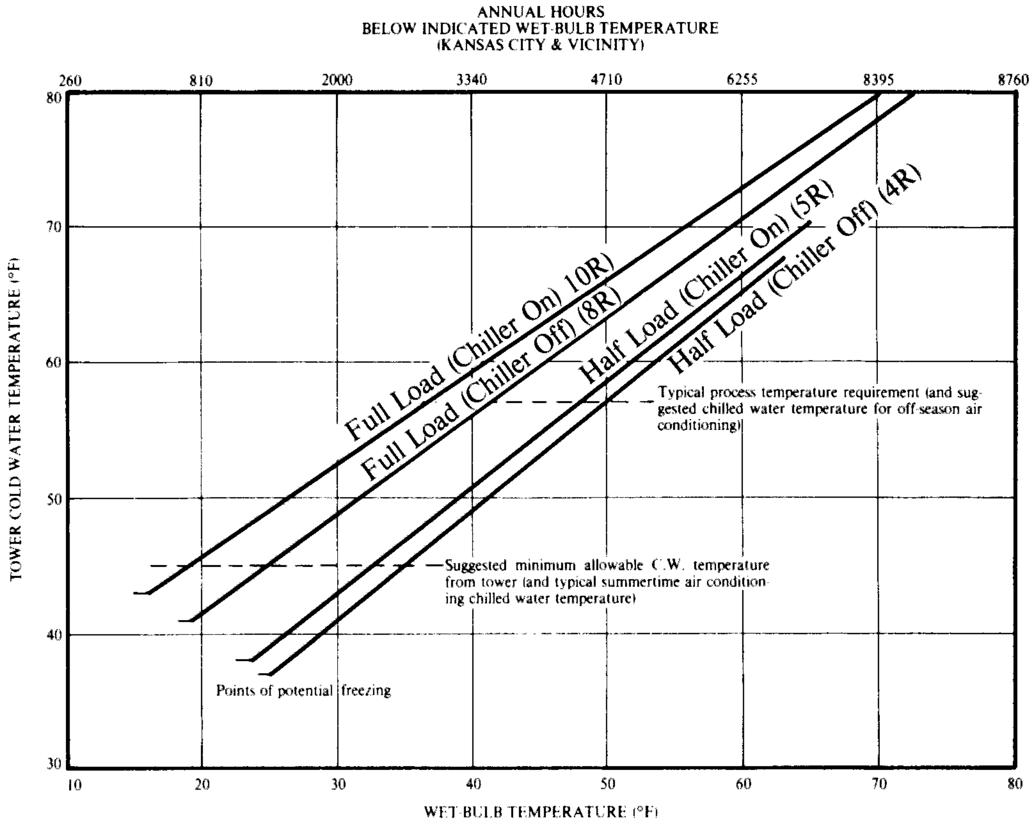

Figure 2 Typical Tower Capability with Refrigerant Compression Chiller.

fording the utilization of free cooling may be reached less than 20% of the time.

The numbers at the top of Figures 2 and 3 represent the total annual hours during which the wet-bulb temperature in the vicinity of Kansas City will be below that shown. Kansas City was chosen as an example because of its "heart of America" location, making for relatively simple comparisons in the minds of the readers. In more severe climates, the position of the numbers will shift to the left — and a greater number of hours will be available for free cooling. More temperate locations will see a shift to the right — allotting fewer hours to the use of free cooling.

Load reductions, as can be seen, affect the opportunity for free cooling dramatically. A 50% load decay (typical of off-season comfort air conditioning) allows a changeover to the free cooling mode at a wet-bulb temperature some 10°F higher than full load operation would permit. Considering geographical variations, this could add from 500 to 2000 hours to annual free cooling operation. Process loads, although sometimes variable, tend to be far less seasonal in their variations. In order to take full advantage of free cooling opportunities as they arise, process systems may require greater control sophistication for automated changeover from chiller — to free cooling — and back again. The "load sharing" system, later described, adapts well to this kind of variability.

Ultimately, of course, the cost of apparatus for free cooling must be weighed against the potential for energy savings. In many areas, the cost of energy is sufficiently high to make even limited use of free cooling worthwhile.

LOAD CHARACTERISTICS

Figures 2 and 3 are curves characteristic of the thermal performance of cooling towers operating at either full load or half load; with or without the added heat of chilling; for refrigerant compression and absorption systems respectively. For purposes of comparison, the cooling towers for both systems are assumed to be cooling the same water rate at a given process load (3 gpm/ton). Since the formula for heat load is:

$$L = \text{Lbs of water/hr} \times (t_1 - t_2) = \text{gpm} \times R \times 500 = \text{Btu/hr}$$

Where gpm = Flow rate over the tower. (gallons per minute)

R = "Range" = Difference between water temperature entering the tower and that leaving the tower.

500 = 8-1/3 Lbs/ gal of water times 60 min/hr.

you can see that the fully loaded tower operating with a refrigerant compression type chiller sees only a 10°F cooling range (15,000 Btu/hr/ton) whereas the base load for the tower operating with an absorption chiller (30,000 Btu/hr/ton) produces a required cooling range of 20°F. Therefore, absorption towers tend to be considerably larger than refrigerant compression towers, making them available for free cooling for a greater portion of the total annual operating hours.

The upper horizontal dashed line (arbitrarily placed at 57°F) represents a typical required cold water temperature level for a process load. On those systems, opportunity for the utilization of free cooling would begin when the cold water temperature from the tower depresses to a level from 3°F to 9°F above that operating point, depending upon the type of chiller.

With the absorption chiller operating (Fig. 3), it should be noted that it is very unlikely the tower would be permitted to cool the water below about 65°F for fear of causing crystallization of the absorbent brine. Fan manipulation and/or hot water by-pass (see "Cooling Tower Energy and its Management") would be utilized to maintain cold water temperatures at or above that level. Nevertheless, the operating line is continued downward for illustration and comparison purposes.

The lower horizontal dashed line (45°F cold water) represents the summertime chilled water temperature usually sought for purposes of comfort air conditioning, where considerable dehumidification of the air is required. In off-seasons, when significantly less dehumidification is required, this temperature is usually allowed to rise to 55 or 60°F. Not only does this assure greater usage of free cooling, but it is also more compatible with the heating that may be required in perimeter portions of the building. In other words, the heating and cooling systems will not be fighting each other for temperature control.

There are unique situations (laboratories, for example) where low temperature water year-round is mandatory. In those cases, the switch to free cooling must await a lower ambient wet-bulb temperature — and fewer operating hours are available. Most load systems, however, will operate successfully well above the 45°F level — and some will operate at water temperatures above 57°F. Users contemplating the use of free cooling should determine the highest allowable optimum operating temperature for their particular systems.

The lower horizontal dashed line is also arbitrarily selected as the operating line below which further reduction in the tower's cold water temperature should not be allowed. The aforementioned fan manipulation should be utilized to maintain the cold water temperature at or above that level. Otherwise, freezing on portions of the fill could begin to occur, depending upon the imposed load. As can be seen on the curves, higher imposed loads (larger cooling ranges) increase this potential for freezing in low temperature operation. Usually, for free cooling application, the imposed load will produce a cooling range no larger than

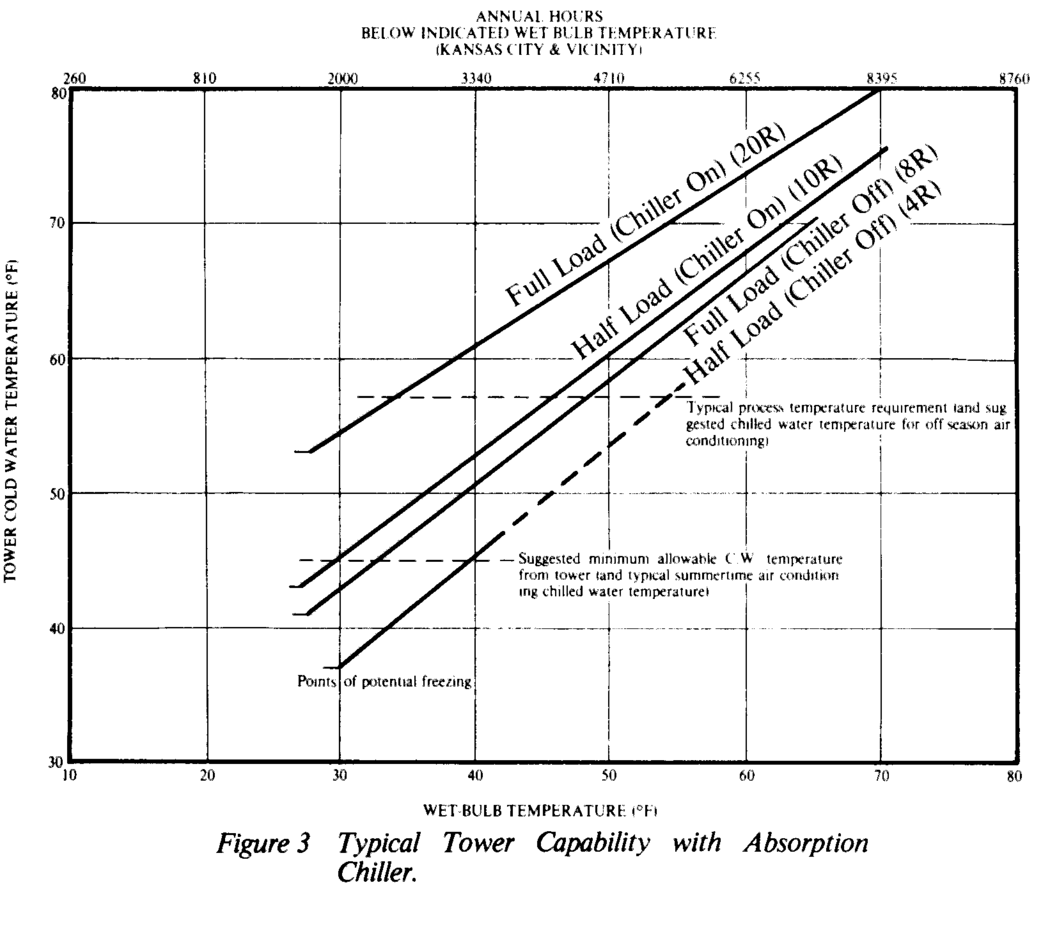

Figure 3 Typical Tower Capability with Absorption Chiller.

about 8 to 10°F, so 45°F cold water temperature is considered a safe minimum. Operating temperature requirements below 45°F should be thoroughly discussed with the cooling tower manufacturer.

METHODS OF FREE COOLING

Direct Free Cooling

The simplest and most thermally effective — yet least recommended — arrangement for free cooling is shown in Figure 4, where a simple by-pass system physically interconnects the condenser water and chilled water loops into one common water path between load and cooling tower. The dashed lines indicate the water flow path during the free cooling mode of operation.

The absence of a heat exchanger separating the two water loops precludes the need for a temperature differential, so the load benefits from the cooling tower's full capacity. The flow rate and temperatures indicated on the diagram are based upon the following assumptions:

1. One of the two circulating water pumps obviously must be by-passed. Under typical circumstances, the condenser water pump should be the one utilized. This assures that the tower will be operating at (or near) its design gpm and, therefore, will suffer no loss in efficiency. This aspect will be further discussed later in this paper, and should be read in conjunction with "Cooling Tower Energy and its Management". Suffice it here to say that the importance of maintaining design water flow over the tower cannot be overstated.
2. The imposed load is assumed to have remained constant (probable for a process load, but unlikely in a comfort air conditioning system), and a cold water temperature of 57°F to the load is acceptable.

From a transposition of the formula appearing on page 4, the temperature rise across the load will be:

$$\frac{12{,}000 \text{ Btu/hr/ton}}{3.\text{gpm/ton} \times 500} = 8°\text{F}$$

and the tower will be called upon to cool the water from 65°F to 57°F. Looking at the 8°F range line (8R) on Figure 2, you see that the tower would begin to be capable of producing 57°F cold water at a wet-bulb temperature of approximately 41°F and, in the Kansas City vicinity, would continue this capability for the remaining 3,475 hours of the year.

At half load (4R), free cooling would have begun at a wet-bulb of about 49°F and continued for some 4,570 hours.

The reason why this direct system is least recommended is because the intermixing of the two water streams con-

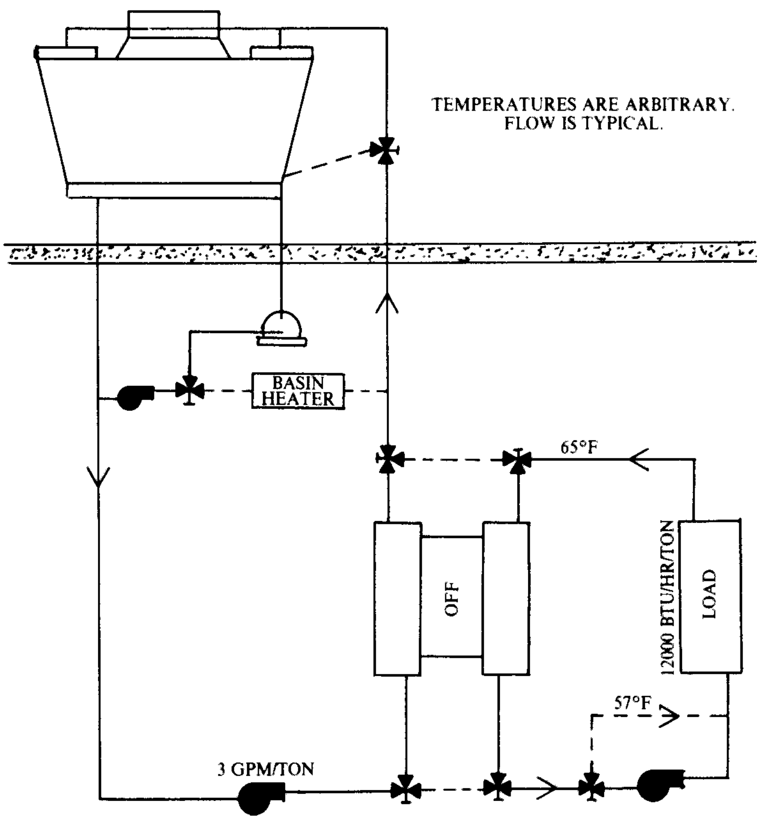

Figure 4 Direct Free Cooling System.

taminates the "clean" chilled water with "dirty" condenser water, which is a situation most practitioners are reluctant to allow. Those who have been most successful in the utilization of this direct-connected system have used a "side-stream" filtration arrangement to continuously filter a portion of the total water flow. Therefore, filtration is pictured as an integral component in this system.

Because cooling towers are naturally efficient air washers, airborne particulates are washed into the tower's cold water basin where those which do not remain in suspension gradually accumulate in low velocity areas. As build-up continues, water velocity increases in those areas and the resultant scouring action reintroduces particulates for potential fouling elsewhere in the system. An effective way to diminish this fouling potential is to direct return flow from the filter to the low velocity areas of the basin, as indicated in Figure 5. In this manner, particulates are maintained in suspension at a point closest to the filter intake.

The amount of side-stream flow necessary for effective system cleanliness, of course, varies with the qualities of the make-up water and the ambient air. Usually a quantity equivalent to from 5% to 10% of the system pumping rate is considered sufficient. Figure 4 also shows this side-stream flow being optionally directed through a heater to prevent freezing in the basin and exposed pipework during periods of winter shutdown. Since this subject is covered in a separate Marley publication, its finite function will not be discussed here.

Side-stream filtration (as opposed to full-flow filtration) is normally chosen because it adds nothing to the system's pumping head, and because it can be back-flushed at will without filter redundancy.

Indirect Free Cooling

By the simple expedient of installing a heat exchanger, piped in a parallel by-pass circuit with the chiller, free cooling is accomplished without concern for contamination of the chilled water loop (Fig. 6). In view of the moderate temperature and low pressures existing in each of these water circuits, plate-and-frame type heat exchangers have proven adequate. Also, because the plate exchanger can function properly with only a small temperature difference (as little as 2 degrees, depending on size), it permits separation of the water loops with minimal sacrifice of free cooling opportunity.

As indicated in Figure 6, full load free cooling operation would result in a 10°F temperature rise across the load and (without the added heat of compressor work) an 8°F cool-

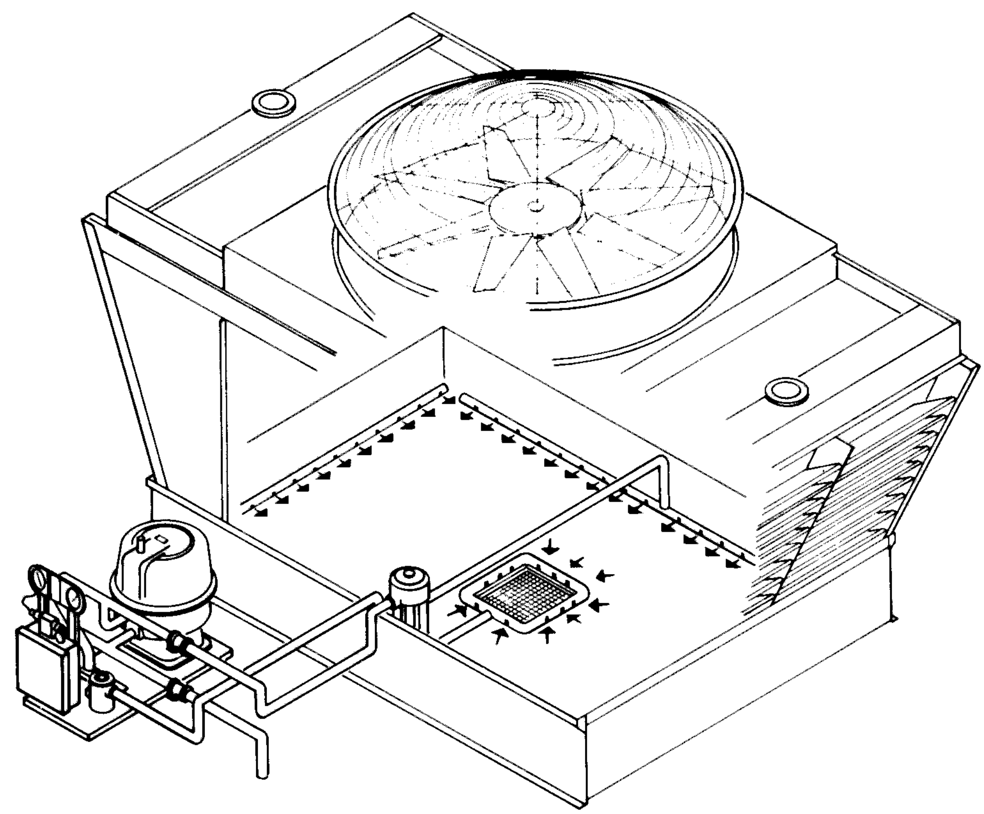

Figure 5 Efficient use of "side stream" filtration keeps cooling tower basin clean.

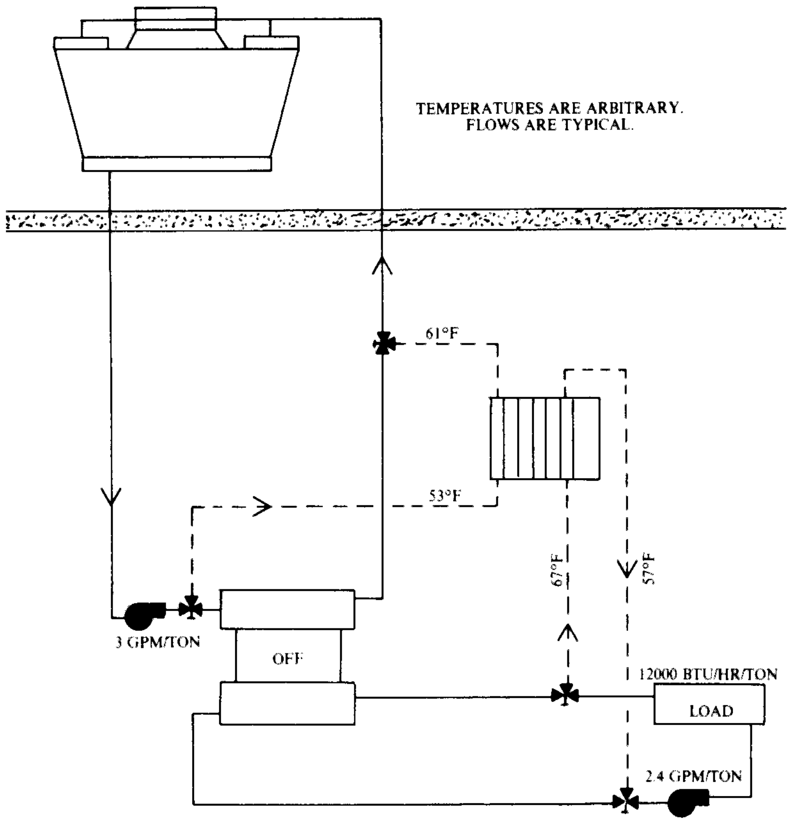

Figure 6 Indirect Free Cooling System.

ing range for the tower, because of the normal difference in flow rates in the separate water circuits. However, with an assumed temperature differential of 4°F required between sides of the heat exchanger, the tower would have to produce 53°F water in order to assure 57°F water at the load. This would result in a potential of about 2,670 hours of operation at full load, or 4,025 hours at half load.

Although a heat exchanger requiring only a 2°F temperature difference would obviously be more expensive, its use would increase the number of operating hours by 400 hours and 275 hours respectively, in the example geographical vicinity. The additional cost of the exchanger would, of course, have to be compared to the cost of continued operation of the chiller during those hours.

Pressure drop through the heat exchanger is also of prime importance. On the tower side of the exchanger, the pressure loss should not exceed that which existed in the condenser. Otherwise, the tower will see less than its design water flow and its efficiency will suffer. Reduced pressure loss, on the other hand, may require the inclusion of flow-regulator valves in the condenser water by-pass loop to prevent excess water flow over the tower and similar loss of efficiency. Occasionally, concern for pressure characteristics in the heat exchanger must take precedence over optimization of temperature difference. The Marley Cooling Tower Company's computer program permits optimization of the heat exchanger selection for both pressures and temperatures.

One obvious advantage of the indirect free cooling system is that it isolates the chiller for seasonal cleaning and maintenance.

Refrigerant Migration Free Cooling

For machines of about 200 tons and larger, many chiller manufacturers will sell an accessory package that will enable the free cooling method indicated in Figure 7 to be used. In this arrangement, valves are opened when the compressor is shut down to permit free migration of refrigerant vapor from the evaporator to the condenser, and the flow of liquid refrigerant from the condenser to the evaporator.

Because heat transfer is essentially limited to refrigerant phase-change, it is normally unlikely that the load capability of such a system will exceed about 25%, and that is the figure on which the temperatures indicated on Figure 7 are based. Furthermore, cold water temperatures at or below 45°F from the tower are usually required to effect heat transfer, which limits usage of this system to a relatively small portion of the year. A requirement for full load operation, of course, would preclude its use altogether.

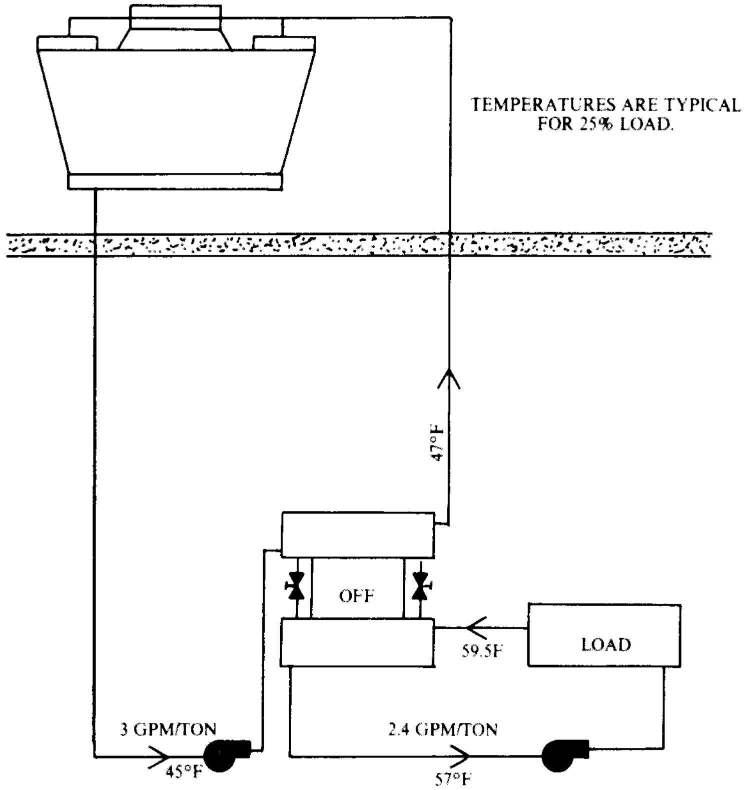

Figure 7 Refrigerant Migration Free Cooling.

Load Sharing

Those who have read to this point in the paper know that the cold water temperature from a cooling tower reduces with wet-bulb and load. Those who have also read "Cooling Tower Energy and its Management" know that compressor horsepower reduces with load and, usually, with the condenser water temperature.

Utilizing these facts, a designer can devise a "load sharing" system, as shown in simplified form in Figure 8, where a plate-and-frame heat exchanger placed in series ahead of the chiller progressively reduces the load imposed on the chiller, ultimately achieving total free cooling.

The operating sequence of such a system would be somewhat as follows:

1. As ambient and/or load reduces, the temperature sensed by thermostat T_1 will begin to approach that sensed by T_2. As soon as the temperature at T_1 is less than that at T_2, valve V_1 is repositioned to cause total flow from the cooling tower to go through the heat exchanger. This reduces the load imposed on the chiller, and continues to do so until the chiller load becomes zero.
   Up until the time valve V_1 is repositioned, note that the water flows through a throttling valve (TV) in its direct route to the condenser. This is to insure that the condenser water pump (P_1) sees a constant head and, therefore, that the tower will see only its design flow rate. The pressure drop through the throttling valve must, of course, equal the design pressure drop through the tower side of the heat exchanger.
2. Most chillers have a condenser water temperature below which no further reduction in compressor operating horsepower is realized, and head pressure problems may be encountered. Consequently, thermostat T_3 must sense that temperature and modulate valve V_2. As will be seen shortly, it is important that valve V_2 not be permitted to modulate to a full by-pass position. Note that there is also a throttling valve in this secondary circuit, whose pressure drop should equal that of the condenser at full flow.
   Thermostat T_3 must also actuate auxiliary pump P_2 in a tertiary by-pass circuit. This circuit provides a source of heat (leaving the condenser) to maintain condenser water temperature at an acceptable level, and assures proper flow through the condenser. Pump P_2 should be sized to provide at least the minimum required flow to the condenser, at the condenser's design pressure drop. Valve V_2 should always allow at least a small amount of water flow into the condenser circuit so that thermostat T_3 will have a meaningful temperature to sense in order to effect proper modulation of valve V_2.
3. Ultimately, thermostat T_4 will sense a temperature lower than that required by the load, and the compressor will

shut off. The compressor should be interlocked with pump P_2 so that pump will not run unless the compressor is energized. If desired, thermostat T_4 may also be used to actuate valve V_2 to effect full flow either through the condenser or through the throttling valve of the secondary by-pass circuit.

4. At this point in the sequence, a full free cooling mode will have been achieved, and thermostat T_5 will have begun manipulation of the fan or fans to maintain a predetermined cold water temperature from the tower, and to further conserve energy. A finite description of this function appears in "Cooling Tower Energy and its Management".

PROPER TOWER UTILIZATION

The primary consumer of energy in a chiller system is the compressor, and the goal of free cooling should be to diminish the total annual hours of compressor operation. Users who lose sight of this goal occasionally end up with a system which can work to their disadvantage. Having become very energy conscious, and having discovered free cooling as a means of reducing energy usage, they begin to look for ways to also reduce auxiliary power requirements and their attention usually focuses on the pumps.

In installations large enough to have multiple chillers, multiple pumps, and multiple towers or tower cells, users will sometimes attempt to operate only the number of condenser water pumps required to match the load. Unless each component is matched (and separately piped) to a tower or tower cell, this is not good practice because it will cause the tower to see something other than its design flow rate. Furthermore, maximizing load on individual components causes the net condenser water temperature rise to increase, which Figure 2 reveals is detrimental to the cold water capability of the cooling tower.

Similarly, when designing a free cooling system, users will consider the fewest possible number of heat exchangers in an effort to minimize flow, maximize temperature rise, and thereby hold down the first cost of the equipment. For the reasons stated before, this can work to the disadvantage of the user because it tends to minimize the amount of time free cooling can be utilized and, therefore, minimizes the potential reduction in gross annual operating cost.

Figure 9 shows a system which offers the user maximum utilization of free cooling along with minimum expenditure for plate heat exchanger. For purposes of illustration, the assumption is made that the summertime load is carried by three chillers of equal size, and that the anticipated off-

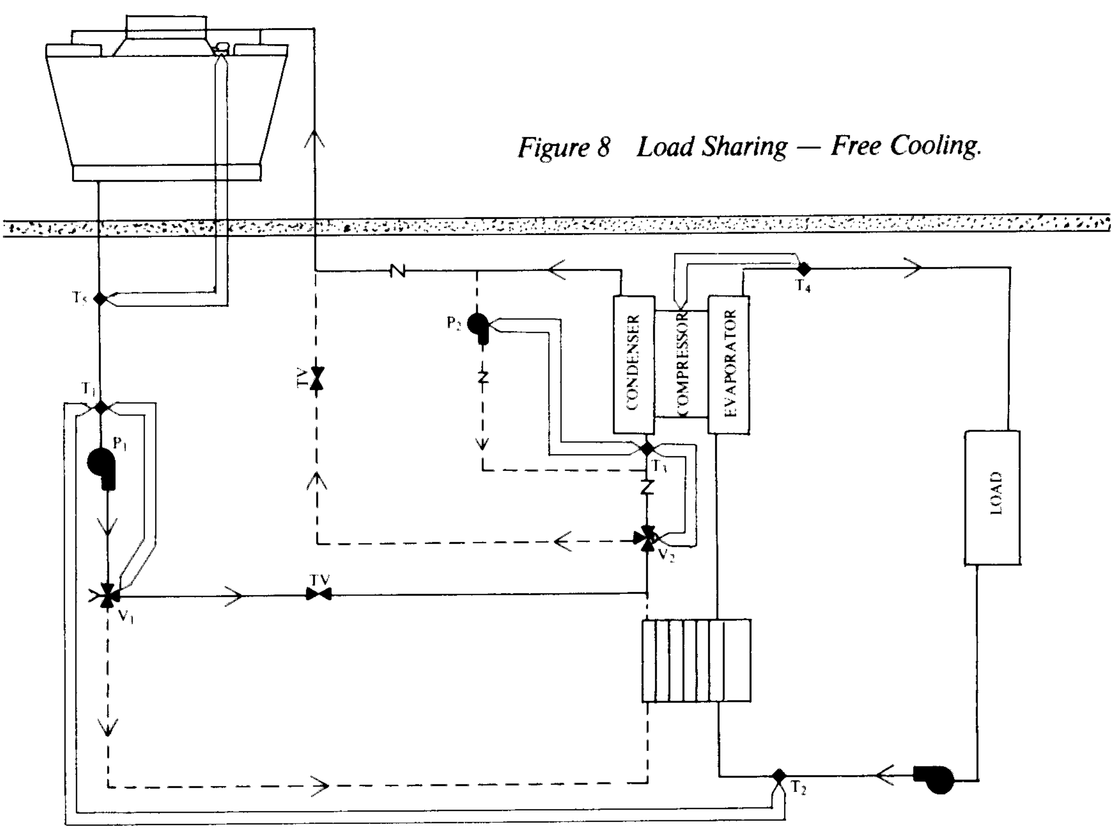

Figure 8 Load Sharing — Free Cooling.

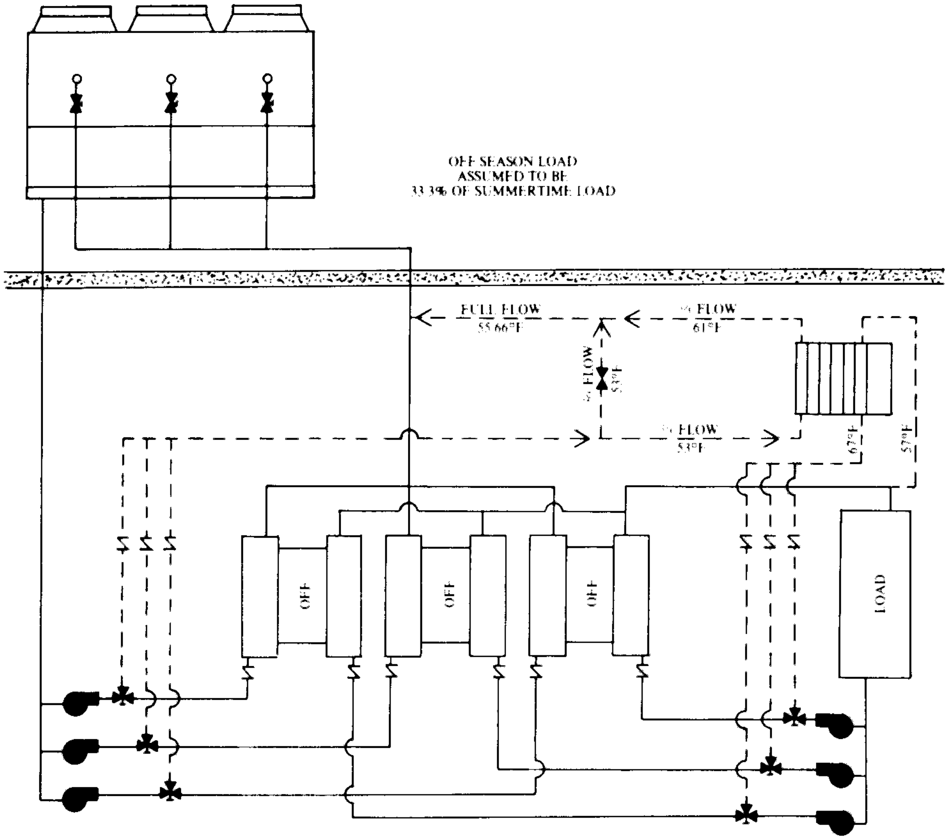

Figure 9 Typical Multiple Component Layout.

season load is ⅓ of the summertime load. Therefore, a small heat exchanger is selected which is capable of the flow and temperatures representative of ⅓ load. It would be sized for ⅓ of the total condenser water flow, at an operating pressure approximating the pressure drop in one of the condensers.

If we look at two of the possible operating combinations, the advantage of maintaining full condenser water flow to the tower becomes apparent:

The first combination is to operate one condenser water pump, with its total flow directed through the heat exchanger. Valves at the cooling tower would assure that this flow is directed through just one cell of the tower.

With this flow configuration, the operating cooling tower cell would see full load because the combination of ⅓ load and ⅓ total flow rate would result in an 8°F required cooling range. In this mode, Figure 2 reveals that the tower would be incapable of free cooling until the wet-bulb temperature depressed to 42°F, and the compressors would be inactive about 3600 annual hours.

To take the opposite extreme for the second possibility, let's assume that all of the condenser water pumps are operated, but only ⅓ of the total flow is allowed to go through the heat exchanger. The remaining ⅔ flow would be bypassed through a pressure-sensitive valve to mix with the water leaving the heat exchanger.

With all cooling tower cells operating, each cell would then see full design flow, at a required cooling range of only 2⅔°F. Figure 2 indicates that this would make the tower available for free cooling at a wet-bulb temperature of about 52°F, and the potential downtime of the compressors would increase by about 1400 annual hours.

The designer must remember that compressor horsepower is usually quite large compared to condenser water pump horsepower, and any ill-timed attempt at conservation of that pump horsepower can be false economy. Early in the fall and late in the spring, maximum usage of condenser water pumps and cooling tower cells will result in maximum utilization of free cooling.

However, as seen on the performance curves, depressed wet-bulb temperatures during colder portions of the year will permit further economies from manipulation of pumps and cooling tower cells. With some relatively simple control apparatus applied to the system shown in Figure 9, this manipulation would be handled automatically. Manipulation of the cooling tower fans, at the appropriate temperatures, would be as described in "Cooling Tower Energy and its Management".

Although, for purposes of clarity, Figure 9 shows chillers and pumps of equal size, this is not necessary for proper operation of the system. With the pressure-sensitive valve by-passing excess water, almost any combination of equipment sizes could be utilized. Where doubt exists, your local representative of The Marley Cooling Tower Company should be contacted.

COOLING TOWER SELECTION AND OPERATION

The primary motivation for spending the capital necessary to make a system capable of free cooling is the promise of reduced energy expenditure. Presumably, this thought pattern will carry through to the selection of equipment for projects in the design stage, and components will be chosen which contribute as little as possible to the total energy requirement.

Since the aforementioned publication compares the energy requirements of induced draft, propeller fan towers and forced draft, blower fan towers in some detail, the energy advantages of the induced draft, propeller fan tower will not be reiterated here. Suffice it to say that the forced draft, blower fan tower requires twice the operating horsepower and is, therefore, detrimental to a pure energy management system.

Wintertime operation and the potential for freezing is of prime concern in the use of towers for free cooling, and a separate technical report entitled "Operating Cooling Towers in Freezing Weather" deals with this aspect in considerable depth. In a nutshell, induced draft towers are the choice to make for wintertime operation. Not only are forced draft towers more conducive to the formation of ice, they are also most difficult to deice. Furthermore, ice on a forced draft tower tends to concentrate itself on the air intake (fan) area, where it tends to defeat attempted manipulation of air flow, and offers the greatest potential for calamity.

Evaporative condensers and closed-circuit fluid coolers have not been discussed in this paper because they offer no particular advantage in free cooling application. Although fluid coolers obviate the need for a separate heat exchanger, their cost usually exceeds that of a separate cooling tower and heat exchanger and, in many cases, their operating horsepower can be excessive.

Responsible users and engineers will, of course, compare the costs and energy management potential of various systems and components before arriving at a judicious decision.

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