MARLEY

thermal science

Free Cooling

MINIMIZING ENERGY COSTS

Background

Energy costs and operating efficiency have gained considerable importance in the minds of many building owners and plant operators in recent years. Current prospects for future prices of energy resources suggest that these issues will become even more urgent as environmental concerns and the high cost of money exert an ever greater impact on building design and operation.

Equipment expenditures for energy-saving systems made little sense in the days of cheap energy. Now, these systems are gaining popularity as a means to control total life-cycle costs.

Free cooling systems can generate significant savings for the owners of such systems. However, the amount of potential energy savings available depends almost totally on the overall system design and on the selection of equipment for use in the system. In general, the designer must balance higher equipment cost with greater opportunity for energy savings. Fortunately, these savings — and their associated costs —are reasonably quantifiable so that designers can make intelligent choices guided by reliable information.

This article will describe several general free cooling design schemes. In doing so, we will review the basic air conditioning scheme as it applies to free cooling and we will discuss the trade-offs involved in equipment selection.

We will also see that the designer's choices determine the potential energy savings available from free cooling for a given project.

The Classic Chilled Water System

Air conditioning and refrigeration systems, as well as most industrial processes, demand cold water — much colder than a cooling tower can provide during a normal summer. Designers select various types of chilled water systems for those applications. The basic chilled water system appears in **Figure 1** as a model for this discussion.

In this system, a chilled water circuit transfers heat from the air conditioning or process load to vaporize a refrigerant flowing through the evaporator. The chilled water then returns to the load source. Meanwhile, the refrigerant vapor is pressurized within a compressor (adding the heat of compression work). The refrigerant then flows to a condenser, where its total added heat content is transferred to the condenser water circuit. Ultimately, of course, this total heat is rejected to the atmosphere by the cooling tower, which cools the condenser water for its return to the condenser.

Notice that the load rejected by the cooling tower exceeds the actual process load by the amount of heat (or work) imposed by the refrigeration function of the chiller. In the refrigerant compression system shown, the additional "heat of compression" increases the load on the cooling tower by approximately 25% over the load imposed by the process. Therefore, although a "ton" of refrigeration is equivalent by definition to a heat dissipation rate of 12,000 Btu/hr, the actual load on the cooling tower for this type of system is actually 15,000 Btu/hr/ton.

Similar logic applies to an absorption chiller system. The cooling tower must dissipate the heat added to effect absorption and release of the refrigerant vapor. The load at the cooling tower in an absorption system is about 2.5 times the load imposed by the process, approximately 30,000 Btu/hr/ton.

For purposes of illustration throughout this article, we will look at the 300-ton system described by **Figure 1**. The flow rates and temperatures indicated on **Figure 1** are typical of an air conditioning system operating at full load in summer 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 reflect the difference in heat content between the condenser water loop and the chilled water loop, and result in a 10°F water temperature rise in each loop. Motor power for the chiller is based on 0.85 hp/ton, a fairly typical figure for many existing machines.

The difference in heat content between the two water circuits will become very important in free cooling system design later in this paper.

As a general rule, process loads do not require a temperature as low as the 45°F shown in **Figure 1**. Typical low temperature processes might demand temperatures between 55°F and 70°F and, for purposes of illustration in this paper, 55°F process chilled water temperature has been arbitrarily selected.

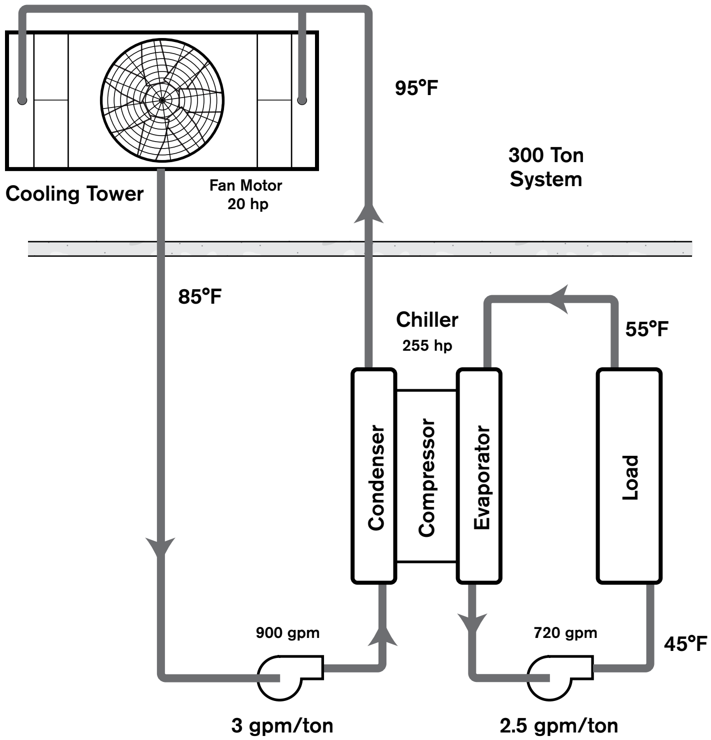

Figure 1 *Refrigerant Compression Chiller System*

In most cases, the primary reason for selecting 45°F chilled water is the need to dehumidify the "conditioned" air during the summer months. Some systems then reheat the air to a proper temperature for human comfort or for laboratory temperature control. Since the natural humidity of the air declines as air temperature falls, little or no dehumidification is necessary during "off-season" months. So, higher chilled water temperatures are usually acceptable during the off-season. As we will see later, these higher allowable temperatures offer greater opportunity to benefit from 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 roughly 0.60 - 0.85 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

A quick glance at **Figure 1** shows that the chiller uses the most energy in the system, by far. Simple logic leads immediately to the conclusion that the greatest possible energy savings would accrue from turning off the chiller.

This, then, is the goal of free cooling – to avoid the energy costs associated with operating the chiller. Obviously, some other means of producing the necessary chilled water must be available. Under suitable conditions of weather and heat load, the cooling tower can act as the source of chilled water.

As most specifiers are aware, the cold water temperature coming from a cooling tower declines as the wet-bulb temperature and/or heat load declines.

At some wet-bulb temperature the cold water temperature produced by the cooling tower will be low enough to satisfy the requirements of the process or air conditioning system without assistance from the chiller. At those times, with a properly equipped and arranged piping system, the cooling tower water could serve the load directly, avoiding the expense of added heat or compressor operation.

Heat Load Characteristics

Figure 2 and **Figure 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:

$$\text{Load} = \text{lb of water/hr x } (t_1 - t_2) = \text{Btu/hr}$$
$$= \text{gpm x 500 x range}$$

Where: gpm = flow rate over the tower (gallons per minute)

range = difference between water temperature entering the tower and temperature leaving the tower

500 = 8.33 lb/gal of water × 60 min/hr

As a general rule, both operating efficiency and control of cooling tower freezing dictate that the flow rate over the cooling tower should be constant at all times. So, it is obvious that reductions in heat load translate directly to reduced cooling range (ΔT across the cooling tower).

The typical system shown in **Figure 1** with 10°F range during chiller operation therefore sees only an 8° range when the chiller is off under conditions of full building load. Similarly, a 75% building load with the chiller off produces only a 6°F range; and a 50% building load produces only a 4°F range.

Knowing these relationships, the designer can easily expand these examples to cover virtually any combination of load and cooling range.

Effective free cooling system design, then, depends on intelligent application of these principles to select design conditions and to develop equipment requirements that will maximize free cooling opportunity at a reasonable installed cost. Before examining system design, let's look at the basic types of free cooling systems in common use today.

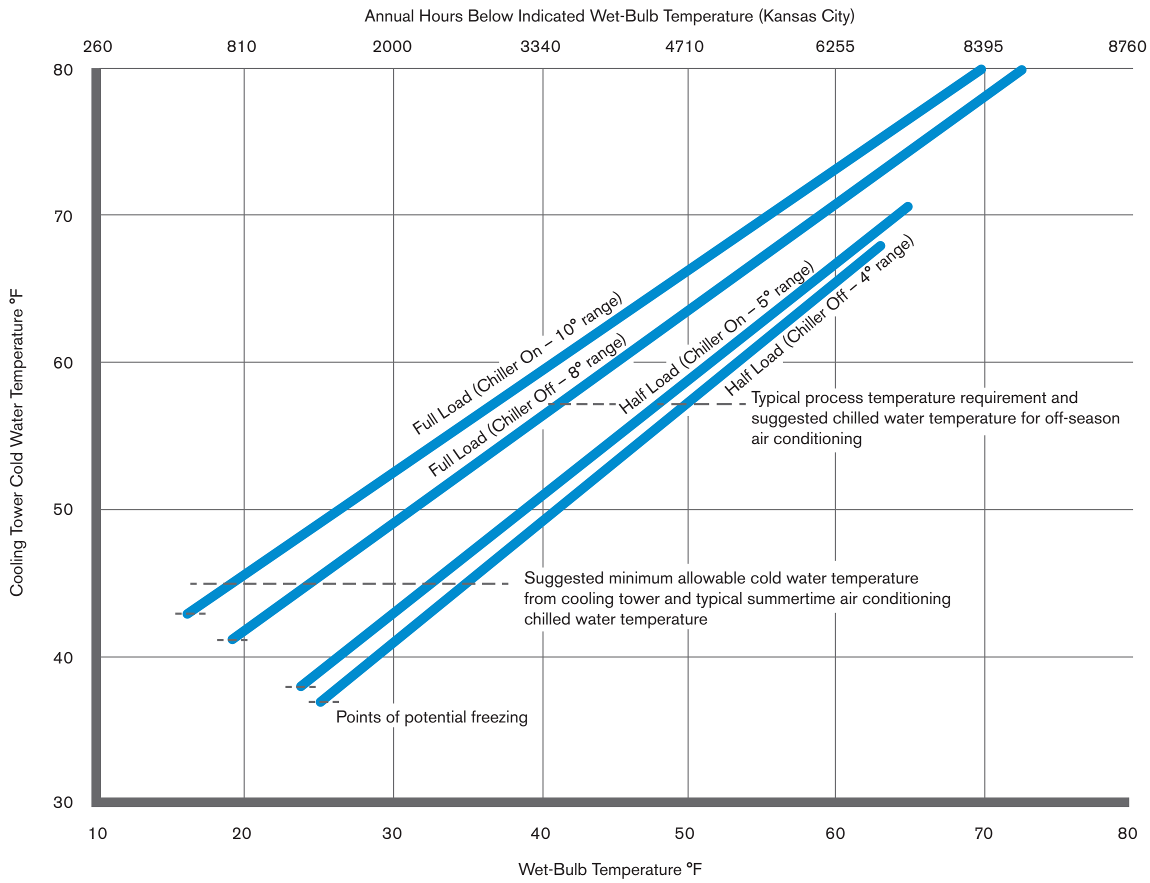

Figure 2 *Typical Cooling Tower Capability with Refrigerant Compression Chiller*

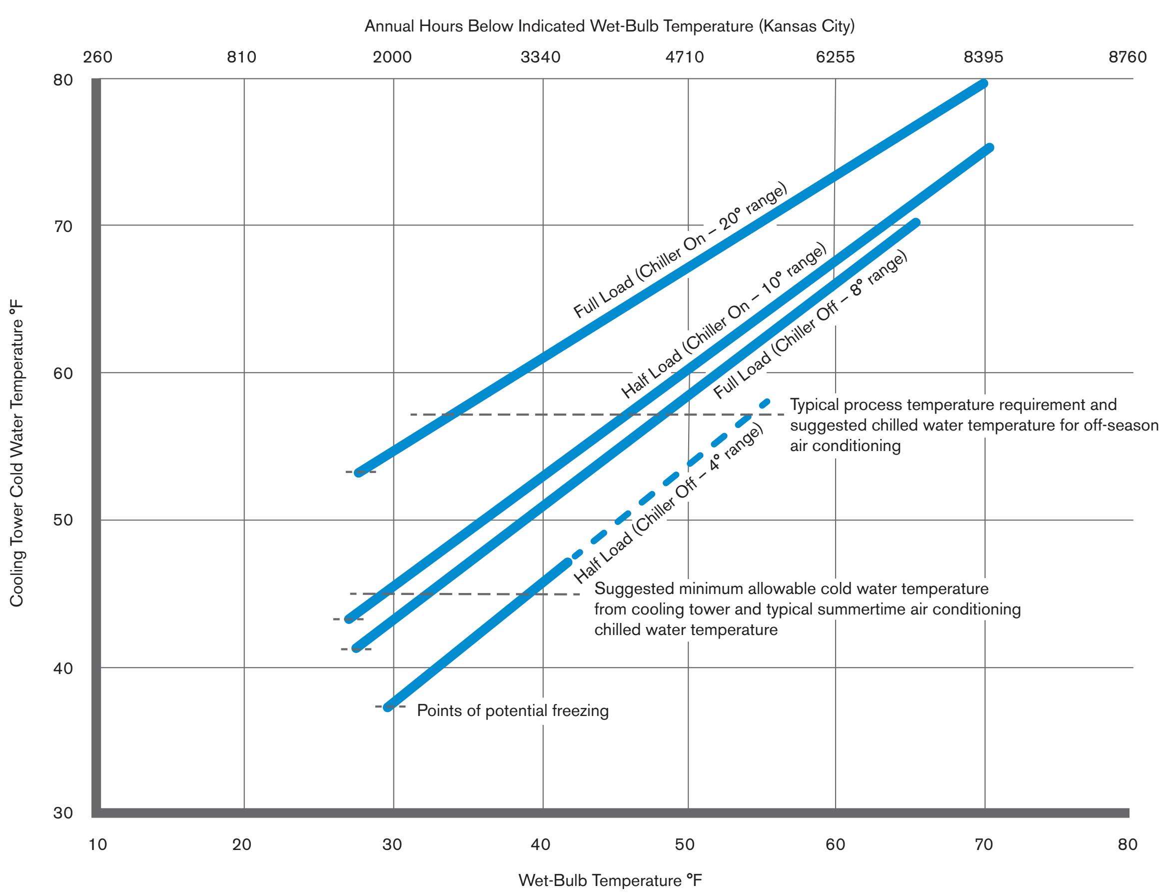

Figure 3 *Typical Cooling Tower Capability with Absorption Chiller*

Direct Free Cooling

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

The direct interconnection of the two water loops permits the load to benefit 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 — usually the chilled water pump. Use of the condenser water pump maintains the cooling tower's efficiency by ensuring that the cooling tower will operate at (or near) its design gpm at all times. This aspect will be further discussed later in this article and should be read in conjunction with *Cooling Tower Energy and its Management*.
2. The imposed load is assumed to remain constant (probable for a process load, but usually unlikely in a comfort air conditioning system), and a cold water temperature of 57°F to the load is acceptable.

The formula for heat load shows that the temperature rise across the load becomes:

So the cooling tower must cool the water from 65°F to 57°F.

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

Looking at the 8°F range line on **Figure 2**, you see that the cooling tower would begin to produce 57°F cold water at a wet-bulb temperature of approximately 44°F and would continue to produce water this cold or colder at all lower wet-bulbs.

At half load (4° range), free cooling would have begun at a wet-bulb of about 51°F and continued for all lower wet-bulbs.

This direct system (often called a "strainer cycle" for historical reasons) is usually least recommended because the intermixing of the two water streams contaminates the "clean" chilled water with "dirty" condenser water — a situation most users are reluctant to allow. In order to minimize the likelihood of fouling heat exchanger surfaces in the chilled water loop, most designers of direct-connected systems include a "side-stream" filtration arrangement to continuously filter a portion of the total water flow. The filter shown in **Figure 4** should be considered an integral component in this type of system.

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.

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

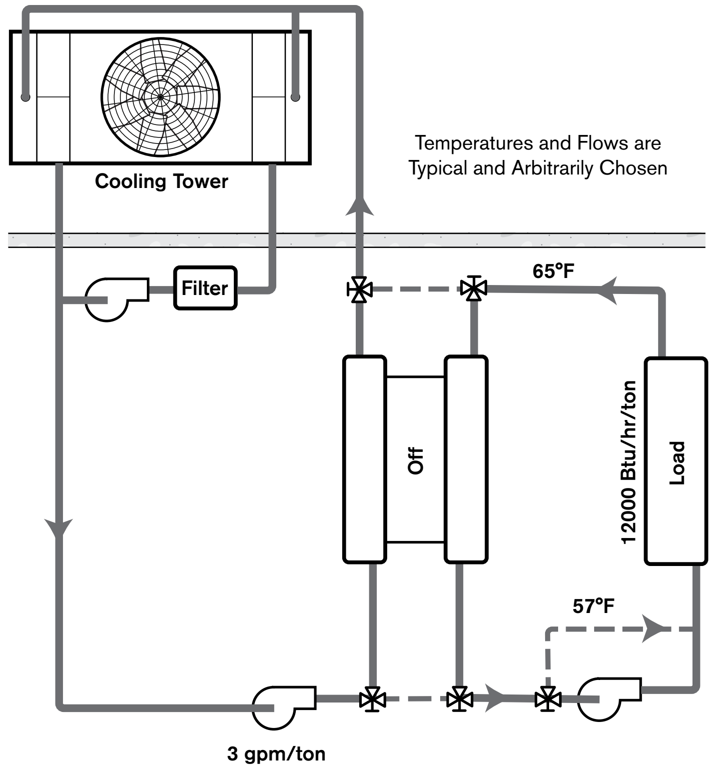

Figure 4 *Direct Free Cooling System*

Indirect Free Cooling

Addition of a heat exchanger, piped in a parallel by-pass circuit with the chiller, maintains complete isolation of the chilled water and condenser water loops during the free cooling cycle — **Figure 5**. Plate-and-frame heat exchangers are usually acceptable for the moderate temperatures and low pressures that occur in these water circuits. Also, because the plate exchanger can function properly with only a small temperature difference (as low as 2°F, depending on size), it permits separation of the water loops with minimal sacrifice of free cooling opportunity.

As shown in **Figure 5**, full load free cooling operation results in a 10°F temperature rise across the load and an 8°F cooling range for the cooling tower. Cooling tower range is only 8°F because the heat of compressor work has been eliminated and because the normal difference in flow rates in the separate water circuits requires only an 8°F ΔT across the cooling tower. However, the cooling tower would have to produce 55°F water in order to assure 57°F water at the load — assuming that the heat exchanger had been selected for a 2°F temperature differential (57°F - 55°F).

While a heat exchanger requiring only a 4°F temperature difference would obviously save on equipment cost, its use would significantly decrease the number of free cooling operating hours as we shall soon see; demanding 53°F cold water from the cooling tower. The potential savings on the lower exchanger cost would, of course, have to be compared to the additional cost of continued chiller operation during the hours lost by the demand for colder water.

Pressure drop through the heat exchanger is also of primary importance. On the cooling tower side of the exchanger, the pressure loss should not exceed that which existed in the condenser. Otherwise, the cooling tower will see less than its design water flow and its efficiency will suffer.

Reduced pressure loss, on the other hand, may require flow-regulator valves in the condenser water by-pass loop to prevent excess water flow over the cooling tower and similar loss of efficiency. Occasionally, concern for pressure characteristics in the heat exchanger must take precedence over optimization of temperature difference.

In addition to the obvious advantage of separate water circuits, the indirect free cooling system also permits the system's operator to isolate the chiller for seasonal cleaning and maintenance.

A properly designed indirect system permits nearly maximum use of free cooling; and also offers obvious advantages in terms of system operation and maintenance.

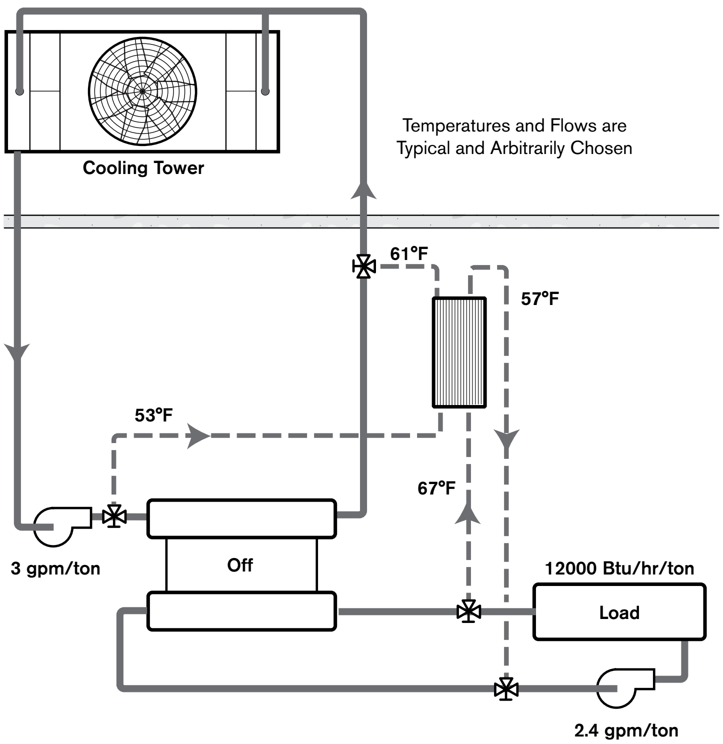

Figure 5 *Indirect Free Cooling System*

Refrigerant Migration Free Cooling

Many chiller manufacturers offer an accessory package that can enable the use of the free cooling method depicted in **Figure 6**. In this arrangement, as the compressor shuts down, valves open to permit the 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, the load capability of these systems rarely exceeds about 25%, although some chiller manufacturers can provide systems capable of up to 35% full load. The temperatures shown in **Figure 6** are based on 25% load.

In addition to the limited load capability of this type of system, the cold water from the cooling tower must usually be 45°F or colder from the cooling tower in order to accomplish sufficient heat transfer. This requirement limits use of this system to a relatively small portion of the year. And, of course, a requirement for full load operation would completely preclude the use of this type of system.

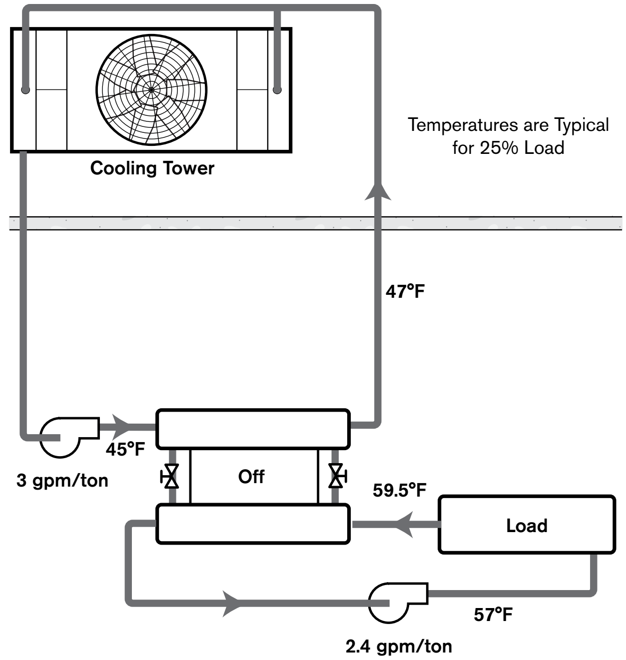

Figure 6 *Refrigerant Migration Free Cooling System*

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 7**, 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 thermostat 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 cooling 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 cooling 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

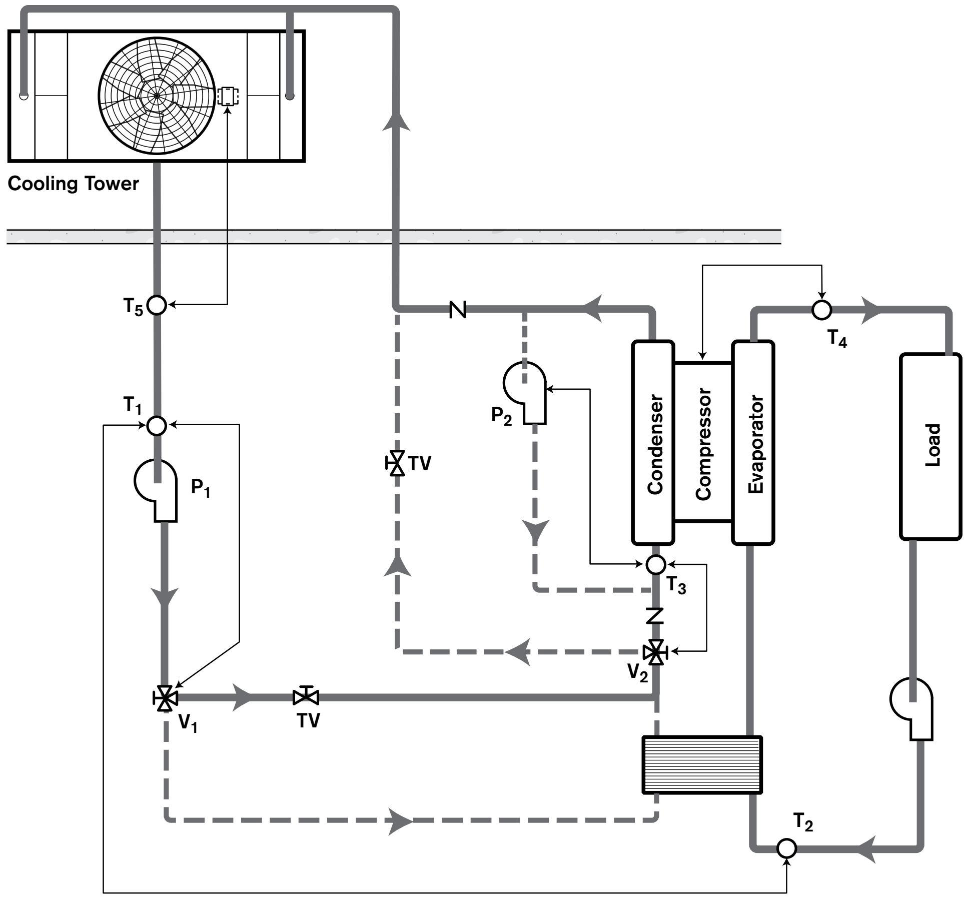

Figure 7 *Load Sharing Free Cooling System*

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 cooling tower, and to further conserve energy. A finite description of this function appears in *Cooling Tower Energy and its Management.*

Proper Cooling 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 cooling towers or cooling 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 cooing tower or cooling tower cell, this is not good practice because it will cause the cooling 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 8 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-season load is one-third of the summertime load. Therefore, a small heat exchanger is selected which is capable of the flow and temperatures representative of one-third load. It would be sized for one-third 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 cooling 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 cooling tower.

With this flow configuration, the operating cooling tower cell would see full load because the combination of one-third load and one-third 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 one-third of the total flow is allowed to go through the heat exchanger. The remaining two-thirds flow would be by-passed 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 cooling 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 8**, 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 8** 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 Marley representative should be contacted.

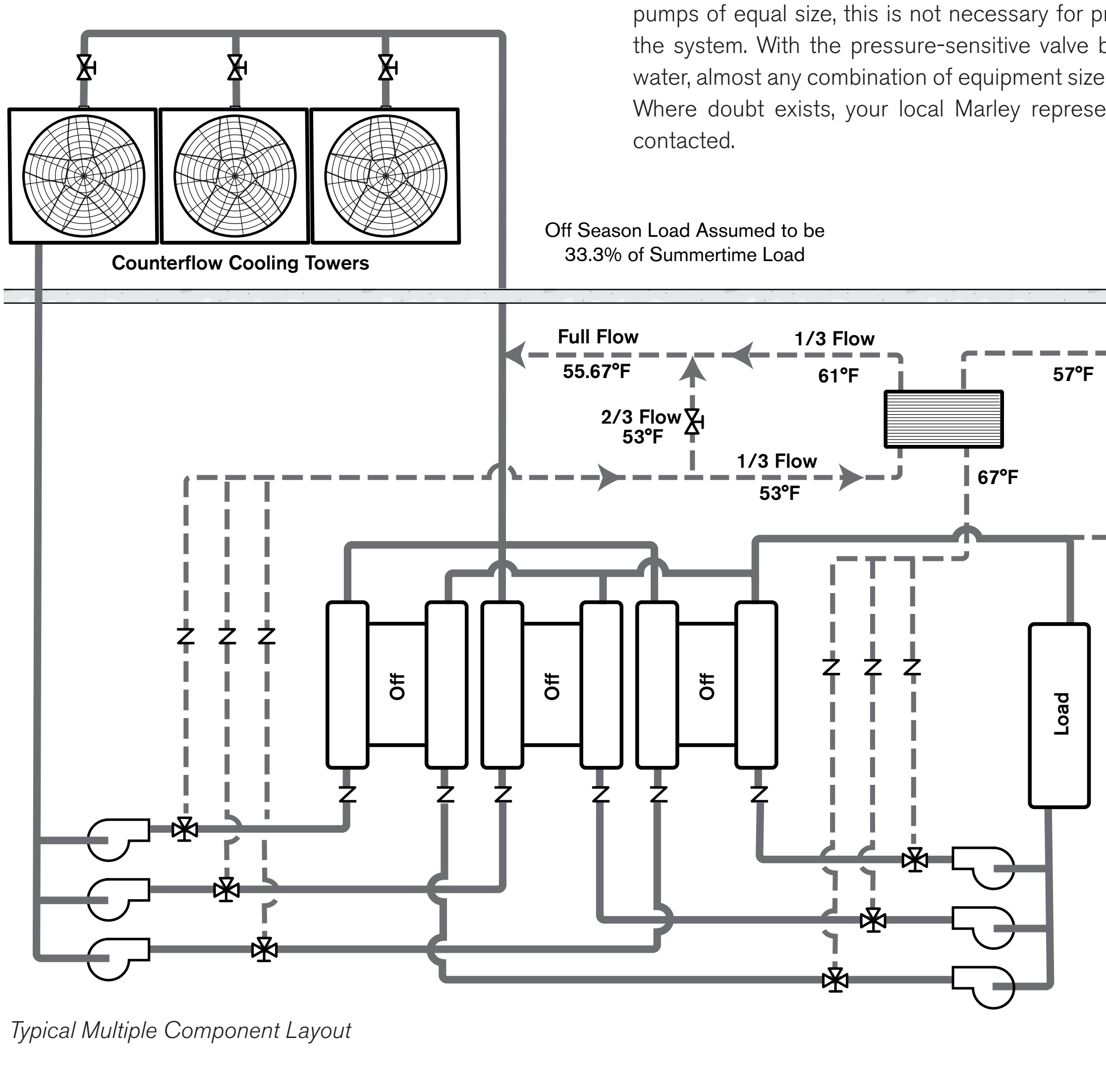

Figure 8 *Typical Multiple Component Layout*

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 white paper compares the energy requirements of induced draft, propeller fan cooling towers and forced draft, blower fan cooling towers in some detail, the energy advantages of the induced draft, propeller fan cooling tower will not be reiterated here. Suffice it to say that the forced draft, blower fan cooling 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 cooling towers for free cooling, and a separate white paper entitled *Cooling Towers and Freezing Weather* deals with this aspect in considerable depth. In a nutshell, induced draft cooling towers are the choice to make for wintertime operation. Not only are forced draft cooling towers more conducive to the formation of ice, they are also most difficult to deice. Furthermore, ice on a forced draft cooling 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.

Free Cooling Opportunity

Simply stated, free cooling opportunity is the number of hours per year a given system can operate in the free cooling mode. The designer controls the amount of free cooling opportunity, to a very large extent, by the choices made in the design process.

Three primary factors under the designer's control determine the amount of free cooling: design chilled water temperature, heat exchanger capacity (for an indirect system) and selected cooling tower capacity. Two other significant variables, load profile and local weather patterns are — quite obviously — beyond the designer's control; but consideration of their impact can help to maximize the availability of the system for free cooling and assist the designer in balancing initial cost with potential operating savings.

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 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.

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H-002A | ISSUED 10/2016

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