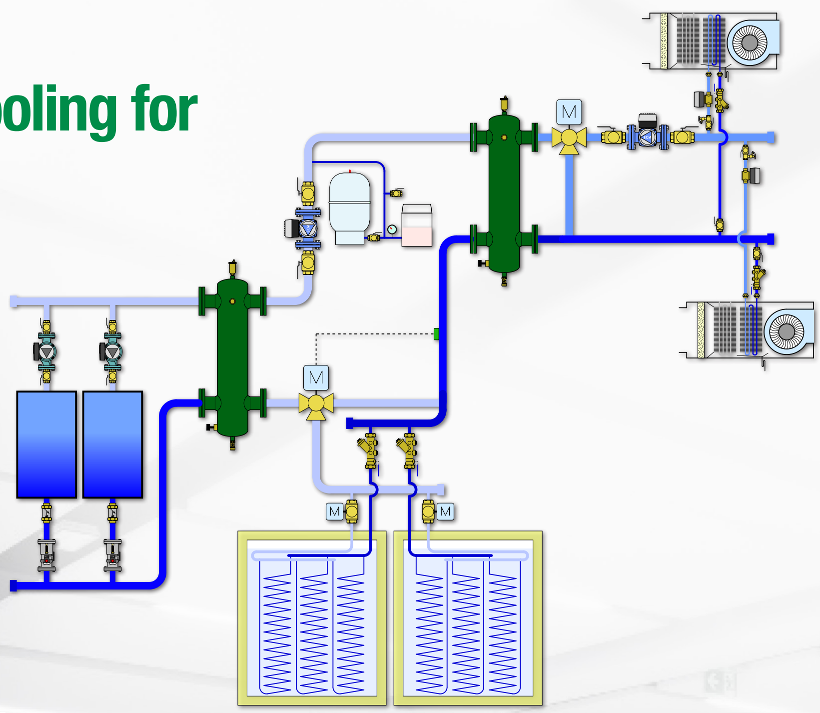
CALEFFI
Hydronic Solutions

idronics™

JOURNAL OF DESIGN INNOVATION FOR HYDRONIC AND PLUMBING PROFESSIONALS

28
January 2021

Contemporary Hydronic Cooling for Commercial Buildings

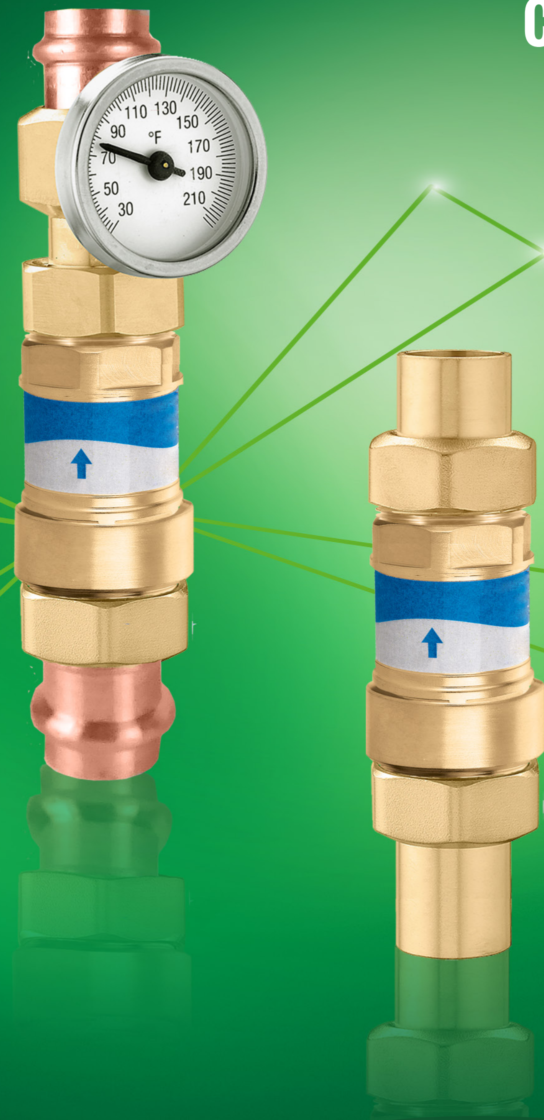


CALEFFI
Hydronic Solutions

www.caleffi.com

AUTOMATIC

CONSTANT GPM
CIRCUIT BALANCING



The FlowCal™ patented series of recirculation balancing valves automatically maintain the desired hydronic or plumbing flow rate, despite differential pressure changes across the circuits, using a cartridge with spring-loaded piston and characterized orifices. The compact design features union end connections and a choice of NPT, Sweat, Press, and PEX fittings for ease of installation and maintenance. **CALEFFI GUARANTEED.**

FROM THE GENERAL MANAGER & CEO

Dear Hydronic and Plumbing Professional,

When it comes to cooling large buildings many HVAC engineers simply default to standard practices.

I recall testing this notion early in my Caleffi career. I was staying on the 20th floor of a hotel in Dallas. Being curious, and still on the steep end of the hydronics learning curve, I managed to remove the panel in my room from which cool air was flowing. I peered in and spotted a standard 4-pipe fan coil. Two pipes for heating and two for chilled water cooling. Standard practice for sure. The fact that flow to each coil was controlled by a Caleffi zone valve was a bonus!

Although standard 4-pipe fan-coil systems work, they're not the only option for cooling commercial and institutional buildings. Other options, such as rooftop units, PTACs, and VRF systems all compete in this market space. However, none of these deliver the efficiency, versatility, comfort, and proven safety that's available using contemporary hydronics technology.

This issue of idronics addresses these contemporary methods and details several modern hydronic concepts such as water source heat pumps, ice storage, heat recovery chillers, radiant cooling, chilled beams, variable speed pumping, hydraulic separation and heat metering. These concepts provide engineers with a wide palette of design opportunities, and the ability to go far beyond "standard practice."

We hope you enjoy this issue of idronics and encourage you to send us any feedback by e-mailing us at idronics@caleffi.com.

For prior issues please visit us at www.caleffi.us, and click on the idronics icon. There you can download the PDF files. You can also register to receive hard copies of future issues.

Mark Olson

Mark Olson

General Manager and CEO

p.s. If a maintenance manager for a major Dallas hotel ever mentions the curious way a customer was able to remove a tamper-proof HVAC panel, mums the word!

CALEFFI Hydronic Solutions

idronics

28

January 2020

TABLE OF CONTENTS

A Technical Journal

from

CALEFFI NORTH AMERICA, INC

3883 W. Milwaukee Rd

Milwaukee, Wisconsin 53208
USA

Tel: 414-238-2360

FAX: 414-238-2366

E-mail: idronics@caleffi.com

Website: www.caleffi.us

To receive future idronics issues

FREE, register online

www.caleffi.us

Printed: Milwaukee, Wisconsin
USA

Disclaimer: Caleffi makes no warranty that the information presented in idronics meets the mechanical, electrical or other code requirements applicable within a given jurisdiction. The diagrams presented in idronics are conceptual, and do not represent complete schematics for any specific installation. Local codes may require differences in design, or safety devices relative to those shown in idronics. It is the responsibility

INTRODUCTION

Most people living in developed countries take building cooling for granted. Most expect it in office buildings and stores. They also expect it in cars, buses, airplanes and trains. With the exception of some far northern homes, or those at high elevations, most homeowners now expect cooling in newly constructed homes.

Attempts to limit the temperature of occupied spaces date back to early civilizations living in caves. Even in tropical climates, the temperature of soil and rock in caverns, and the ability to escape from direct sunlight, provided some relief from what most of us would now consider unbearably high temperatures and humidity levels.

Figure 1-1

Ancient Egyptians learned to improve the comfort of their interior spaces during warm and dry weather by hanging reeds in windows and trickling water down them. The water evaporating from the surface of the reeds absorbed heat from the dry desert air as it passed through. Today this is recognized as an early form of evaporative cooling.

The Romans routed water, supplied from aqueducts, through pipes in the walls of buildings to cool them. Today, cool water flowing through PEX tubing embedded in building surfaces provides radiant cooling.

The Persians fashioned "wind towers" into their structures. The buoyancy of a column of warm air on the downwind side of a nominal 50-foot-tall structure that resembled a large chimney created an upward draft. The exiting air created a downward draft on the windward side of the tower that routed air into lower levels of the building. These wind towers are still used in parts of the Middle East.

Figure 1-2

In 1758, Benjamin Franklin demonstrated that evaporation of volatile compounds, such as ether, could cause the object from which the evaporation was occurring to drop below room temperature. During one of his experiments, he managed to lower the temperature of a thermometer from 64ºF down to 7ºF. Today, the evaporation of refrigerant compounds is an essential process in millions of cooling systems.

Figure 1-3

Prior to electricity, there was little relief from oppressive summer heat and humidity. Cross ventilation through open windows provided some solace if a breeze was present but remained unable to lower oppressive humidity.

When electricity became available, motorized fans provided some relief from the discomfort associated with high air temperatures and high humidity. Increased air velocity across exposed skin encourages evaporation of perspiration, and thus increases heat dissipation from the body. However, fans alone cannot reduce interior humidity, and thus, were not a complete comfort solution.

Advances in chemistry during the 1800s laid the foundation for the first mechanical cooling devices. The

Figure 1-4

first use of closed-cycle refrigeration-based cooling for industrial applications dates back to 1902, when Willis Carrier developed an ammonia-based refrigeration system for a Brooklyn printing plant.

Refrigeration-based cooling for human comfort dates back to 1924, when a centrifugal chiller also designed by Willis Carrier was installed in a Detroit department store. The luxury it provided soon lead to a market for cooling in movie theaters and other public buildings. The continued development of refrigeration-based air conditioning played a major role in the development of the southern United States during the 1900s. *Some historians consider it one of the major technical accomplishments of the 20th century.*

Before widespread use of refrigeration-based cooling, most buildings equipped with central heating used steam or hot water to deliver heat from a boiler to radiators in various parts of a building. As devices for cooling were developed and electricity became increasingly available, ducted forced-air systems became the dominant method for delivering cooling comfort. The primary reason was that a forced-air delivery system could deliver heating, cooling and ventilation, whereas hydronic systems of that era could only provide heating.

However, when the objective is to deliver cooling exactly when and where it's needed, especially in larger buildings, *water remains far superior to air.*

Figure 1-5

The same physical properties that make water ideally suited for transporting heat through buildings also give it distinct performance advantages in removing heat from those buildings. Modern hydronics technology allows the development of water-based cooling systems that can be highly customized and highly efficient in a wide range of commercial and institutional buildings.

Figure 1-6

Courtesy of Aermec

This issue of *idronics* focusses on several *contemporary* approaches for hydronic-based cooling in commercial and institutional buildings. These approaches leverage many modern hydronic design concepts, such as variable-speed pumping, hydraulic separation, thermal metering, heat pump technology, thermal storage, radiant panels, and waste heat recovery. This issue provides an overview of several approaches, summarizes the benefits they offer, and explains the underlying technical concepts.

ADVANTAGES OF HYDRONIC-BASED COOLING

Several methods for cooling commercial and institutional buildings have been developed based on the use of chilled water as the primary energy transport media. These methods offer several distinct advantages that cannot be matched by "all-air" systems. They include:

- Minimally invasive installation
- Reduced electrical energy usage
- Adaptability to many sources of chilled water
- Easy zoning
- No coil frosting
- Adaptability to radiant panel cooling
- Adaptability to chilled-beam cooling
- Possibility of use of rejected heat
- Adaptability to thermal storage
- Low refrigerant volume relative to VRF systems

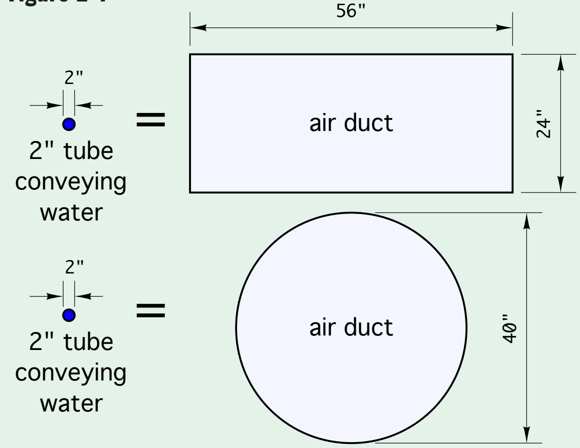
Each of these advantages is briefly described in this section. They will be more fully explained and applied in later sections.

MINIMALLY INVASIVE INSTALLATION

The heat capacity of water is 62.4 Btu/ft^3/ºF. The heat capacity of air is 0.018 Btu/ft^3/ºF. The difference in this thermal property allows liquid water to absorb almost 3,500 times more heat than the same volume of air. This has profound implications regarding the size of the piping required to convey chilled water through a building versus the size of ducting required to move a thermally equivalent amount of air through that building.

For example, a 2-inch copper tube carrying chilled water at a flow rate of 40 gallons per minute to a large air handler, and undergoing a temperature increase of 15ºF across the coil of that air handler, is absorbing heat at a rate of 300,000 Btu/hr. To do this with a duct operating at a face velocity of 1,000 feet per minute and an air temperature change of 30ºF would require a duct 24 inches deep and 56 inches wide, or a round duct about 40 inches in diameter, as shown in Figure 2-1. Accommodating such large ducts in commercial and institutional buildings usually requires a mechanical chase located above a suspended ceiling. This adds to building height as well as cost.

Figure 2-1



Surface area of 2"copper tube = 0.556 sq ft/ linear ft
Surface area of 24"x56" duct = 13.33 sq ft/ linear ft
Surface area ratio = 24:1

Piping carrying chilled water and ducting carrying chilled air require insulation to prevent surface condensation. The 24-inch x 56-inch rectangular duct shown in Figure 2-1 has 24 times more surface area compared to the 2-inch copper tube. Insulating the duct against surface condensation would require 24 times as much insulation material as using the same insulation on the copper tube. Undesirable heat gain into the duct will be substantially greater than that into the tube, assuming approximately equal internal temperatures and insulation R-values.

REDUCED ELECTRICAL ENERGY USAGE

A cooling distribution system operating with chilled water can use significantly less electrical power compared to a forced-air distribution system of equivalent thermal capacity. The difference can be expressed through a calculation of *distribution efficiency*, which is defined by Formula 2-1:

Formula 2-1

$$\text{Distribution efficiency} = \frac{\text{rate of heating or cooling delivery(Btu/hr)}}{\text{electrical power input to operate delivery system(watt)}}$$

Here's one comparison: Published data for the blower in a geothermal water-to-***air*** heat pump using forced-air delivery indicates that a 3/4-horsepower electronically commuted motor is required to deliver approximately 1500 CFM airflow through a duct system. The estimated electrical input power to this motor when operating at full capacity is 690 watts. The rated total cooling capacity of this heat pump is about 53,000 Btu/hr (based on 60ºF entering fluid temperature from the geothermal source). The heat pump's distribution efficiency under these conditions is:

$$\text{Distribution efficiency} = \frac{53{,}000\text{Btu/hr}}{690\text{ watt}} = 76.8\frac{Btu/hr}{watt}$$

An equivalent hydronic system is assumed to have 200 feet equivalent length of 1-inch copper tubing. It will operate with a 10ºF chilled-water temperature rise across a radiant ceiling panel. The hydronic system uses a circulator with an electronically commutated motor with a wire-to-water efficiency of 40%.

The water flow rate required for delivering 53,000 Btu/hr at a 10ºF temperature change is:

$$f = \frac{53{,}000}{500 \times \Delta T} = \frac{53{,}000}{500 \times 10} = 10.6\text{gpm}$$

Assuming a 200-foot total equivalent length for the circuit, the pressure drop in the circuit is 5.6 psi. The power supplied to the circulator can be estimated using Formula 2-2:

Formula 2-2:

$$\text{w} = \frac{0.4344 \times f \times \Delta P}{e} = \frac{0.4344 \times 10.6 \times 5.6}{0.40} = 64.5 watt$$

Where:
w = electrical input power to circulator (watts)
f = flow rate through circulator (gpm)
ΔP = pressure increase across circulator (psi)
e = wire-to-water efficiency of circulator (decimal %)

Assume that 30% of the 53,000 Btu/hr total cooling load is latent cooling. This is handled by a small air handler with a blower operating at 150 watts.

The distribution efficiency of this hydronic delivery system would be the rate of heat transfer divided by the wattage of the circulator plus the wattage of the air handler blower. It can also be calculated using Formula 2-1:

$$\text{Distribution efficiency} = \frac{53{,}000 Btu/hr}{\left(64.5w + 150w\right)} = 247 \frac{Btu/hr}{watt}$$

This comparison shows that the chilled-water delivery system is using only about 36% of the electrical power required by the equivalent forced-air system. The circulator wattage (64.5 watts) was only about 9.4% of the power required by the blower in the heat pump. The 150-watt blower in the small air handler adds substantially to the total power required for the hydronic cooling method.

Hydronic cooling systems using radiant panels or chilled beams will have minimal ratios of total cooling capacity divided by fan or blower input power. Commercial and institutional buildings using these systems can substantially reduce the electrical energy required to convey cooling throughout a building. Radiant panel cooling and chilled beams are discussed in more detail in section 7.

It's important to remember that all electrical input to a cooling distribution system ultimately becomes a heat gain to the building. When high energy efficiency is a primary design goal, it is imperative to minimize the electrical input power required to create cooling effect AND move that effect through the building to where it's needed. Properly designed hydronic distribution systems require significantly less electrical energy compared to thermally equivalent "all-air" systems.

CHILLED WATER OPTIONS

A wide variety of devices are now available to produce chilled water. They include the large dedicated (non-reversible) chillers often used in large high-rise buildings, as well as reversible heat pumps, and heat recovery chillers, such as shown in Figure 2-2. In some limited circumstances, it is also possible to use water from a lake to directly cool a building. These options are described in section 5.

Figure 2-2

Courtesy of MultiAqua

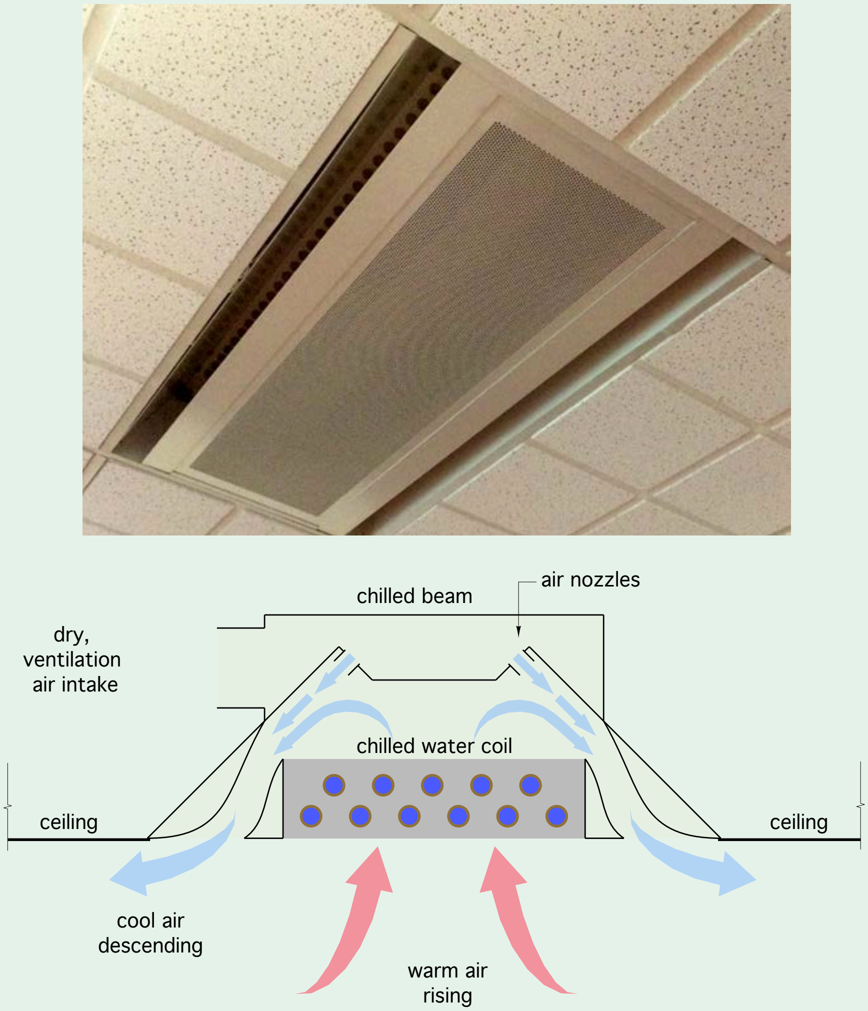
EASE OF ZONING

Chilled-water cooling systems are very easy to zone. They can be designed around electrically operated zone valves in combination with variable-speed pressure-regulated circulators. This approach substantially reduces electrical energy consumption relative to systems using multiple zone circulators. Zoning also allows unoccupied areas to be minimally cooled, and thus reduces the energy used by the chiller.

NO COIL FROSTING

Many air handlers and fan coils used for cooling have direct expansion ("DX") coils. Liquid refrigerant flows

Figure 2-3



into these coils and evaporates as it absorbs heat from the air stream passing across the coil. In some cases, the temperature of the refrigerant within the coil can be lower than 32ºF. This creates a potential for frost to form on the exterior surfaces of the coil — depending on the temperature and humidity of the air entering the coil. This undesirable effect is exasperated if the actual airflow rate across the coil is lower than the specified airflow, which could result from improperly designed ducting systems. Coil frosting will not occur with chilled-water coils, which operate well above 32ºF.

RADIANT PANEL COOLING

Chilled water can be used for radiant panel cooling. Ceiling surfaces are ideal for absorbing heat from the occupied space below. An example of a radiant ceiling that can provide both heating and radiant cooling is described in section 7.

CHILLED-BEAM COOLING

Chilled beams are specially designed heat absorption units that use chilled water to generate a gentle cooling airflow within a room using natural convection. They also serve as a terminal to gently supply ventilation air to occupied spaces, as shown in Figure 2-3.

Although relatively new to North America, chilled beams have been used in European buildings since the 1970s. Like radiant ceilings, they are only intended to handle a portion of the sensible cooling load. The distribution energy saved through use of chilled beams has proven to be as much as 50% lower than that required by VAV (Variable Air Volume) systems. Chilled-beam cooling is also known to be very quiet and comfortable. It is discussed in more detail in section 7.

POSSIBLE USE OF REJECTED HEAT

The heat rejected from a water-cooled chiller may be of use for simultaneous space heating or process heating loads. One example is preheating (or fully heating) domestic water using the heat rejected from a water-to-water heat pump supplying chilled water for space cooling. Another is using the heat rejected from a water-cooled chiller to warm a swimming pool. In commercial buildings, it is common to have situations in which the core areas of the building require cooling, while the exposed perimeter areas of the building require heating. This is an ideal application for a water loop heat pump system, which is described in section 8.

THERMAL STORAGE

Chilled-water cooling is adaptable to thermal storage where preferential time-of-use electrical rates or lower nighttime ambient temperatures

Figure 2-4

make this approach feasible. Ice-based thermal storage cooling systems use large water-filled tanks, such as shown in Figure 2-4.

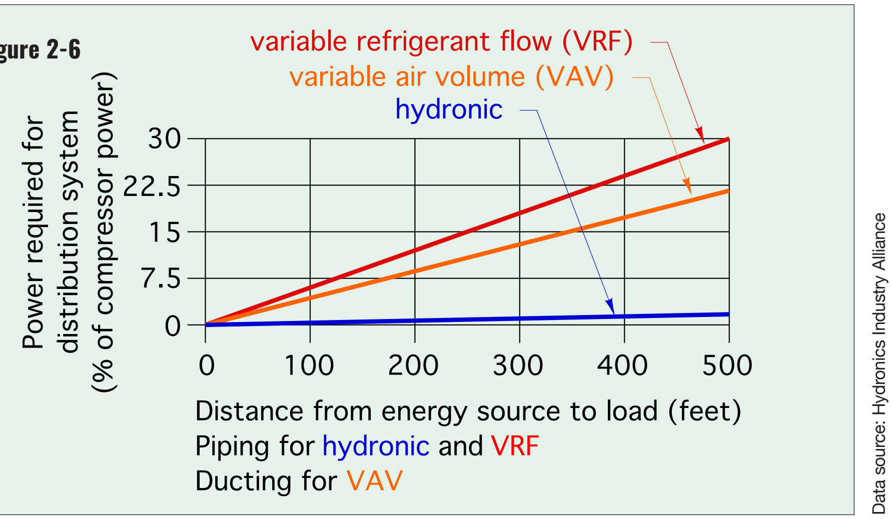
Integrating ice-based thermal storage into hydronic-based cooling can also significantly reduce electrical power demand charges, as well as the size of the chiller(s) used. These systems are described in section 9.

LOWER REFRIGERANT VOLUME

Chilled-water cooling systems contain far less refrigerant than equivalent variable refrigerant flow (VRF) systems. This is important for several reasons.

First, a leak in a commercial (VRF) system could lead to the loss of hundreds of pounds of refrigerant, in some cases inside the building. This is a dangerous possibility that could require emergency evacuation of the building. It's also an expensive failure, and undesirable from the standpoint of releasing gases that contribute to climate change. ASHRAE standards 15 and 34 define specific refrigeration concentration limits. Those designing VRF systems should verify that the amount of refrigerant that could be released due to an interior leak, and the smallest interior space where such a leak could occur, are in conformance with these standards.

Figure 2-6



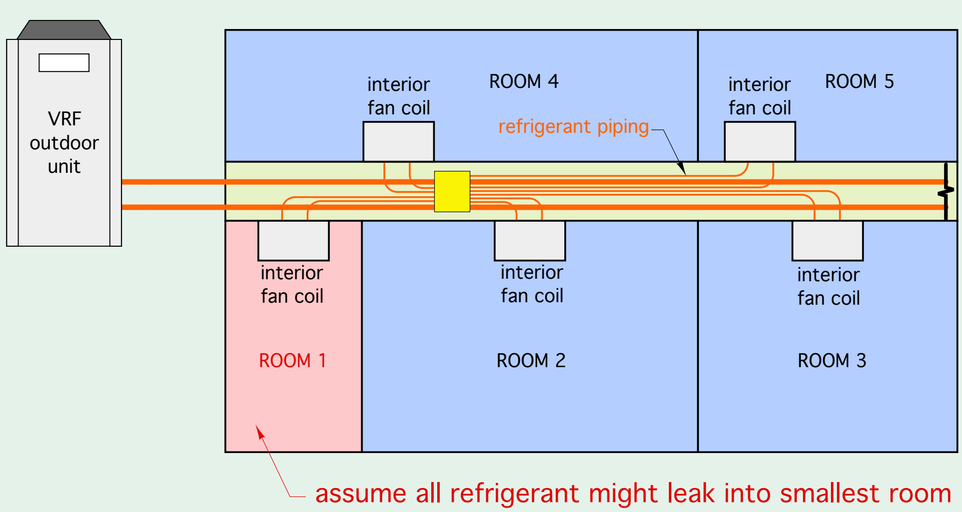
Second, the type of refrigerants used in current generation VRF systems may not be the same as those used in the future. There is currently a wide variety of research underway to develop new refrigerants with lower global warming potential. It's also likely that when such refrigerants are available, environmental regulations aimed at replacing legacy refrigerants will be enacted. There is no guarantee that a currently installed VRF system will be compatible with future refrigerants or oils. Incompatibility could require a major equipment changeout. In contrast, any future hydronic chiller will remain compatible with a water-based distribution system.

Third, most VRF systems use proprietary controls that must be commissioned and serviced by specially trained technicians. This limits the building owner's choices on maintenance as well as parts availability and cost.

Fourth, hydronic-based systems are adaptable to low power terminal units such as radiant panels and chilled beams, whereas almost all VRF systems use fan-equipped terminal units.

Finally, the electrical energy used to transport heating and cooling energy through a building, expressed as a percent of the chiller compressor power, is substantially lower in a properly designed hydronic system compared to either a VRF system, or a variable air volume (VAV) system of equivalent capacity, as shown in Figure 2-6.

Figure 2-5



A SURVEY OF COMMERCIAL COOLING OPTIONS

There are many options when it comes to cooling commercial and institutional buildings. They range from inexpensive (but intrusive) approaches, to sophisticated systems that can independently cool and dehumidify hundreds of rooms with nearly imperceptible noise and drafts.

WINDOW AIR CONDITIONERS

The low-end approach to cooling large buildings, especially older buildings used for apartments or condominiums, is to use window air conditioners that are temporarily installed in late spring and removed in fall. Window air conditioners are only useful in rooms with exterior walls and suitably sized windows. Although such rooms are common in residential buildings, this requirement significantly impedes their use in many non-residential commercial buildings, (i.e., those having rooms without exterior walls). Window air conditioners are also aesthetically unappealing. The sound that they generate is present in the space being conditioned. Their cooling efficiency is also lower than that of many other central cooling options. They are very limited in terms of how the forced-air stream is directed into the occupied space, and thus may create objectionable drafts. They require much more effort to install and remove each year compared to other, permanently installed options. In most cases, their use in commercial buildings is based on low installation cost, or the inability to accommodate any other cooling hardware.

Figure 3-1

PACKAGED TERMINAL AIR CONDITIONERS (PTAC UNITS)

Other commercial cooling options included "through-the-wall" packaged terminal air conditioner (PTAC) units, as seen in Figures 3-2a and 3-2b.

Figure 3-2a

Figure 3-2b

PTAC units are self-contained, and most are all-electric. They use a reversible heat pump refrigeration system to provide heating and cooling. They are installed through openings in exterior walls, and thus are not applicable in spaces without such walls. They supply heating or cooling to a room using forced-air circulation. They allow for individual temperature control in each room, and thus are popular in hotels.

Still, most people who have spent time in hotel rooms conditioned by PTAC units can attest that the comfort they provide is not ideal. A single point of high velocity air delivery into a room tends to create objectionable drafts. Undesirable fan noise is also present.

DUCTLESS MINI-SPLIT HEAT PUMPS

Another commercial cooling option is a multiple-head ductless mini-split air conditioner (or heat pump). In these systems, a single outdoor condenser can supply chilled refrigerant to several indoor wall or ceiling "cassettes," which in turn supply cooled and dehumidified fan-forced air to occupied spaces. A typical outdoor condenser for small system is shown in Figure 3-3a. One of its associated wall cassettes is shown in Figure 3-3b.

Figure 3-3a

Figure 3-3b

Ductless systems require two refrigerant tubes between each indoor cassette and the outdoor condenser. They also require individual condensate drain tubes from each cassette to a suitable drain and wiring between each cassette and the condenser. Like PTAC units, ductless mini split systems allow individual temperature control within each space where a cassette is mounted.

Figure 3-4 shows the outdoor portion of a ductless mini-split cooling system that serves four interior wall cassettes.

Figure 3-4

Ductless mini-split systems tend to be more expensive on a dollar per unit of cooling capacity basis compared to larger centralized systems. They are popular in retrofit applications because they eliminate the need for ducting. However, routing the refrigeration tubing, wiring and condensate drain to each interior cassette, especially in retrofit situations, can be challenging. When interior routing is difficult or impossible, the tubes and wires are routed along the exterior of buildings, which compromises aesthetics, as seen in Figure 3-5.

Figure 3-5

VARIABLE REFRIGERANT FLOW (VRF) SYSTEMS

In larger commercial buildings the concept of ductless mini-split systems can be expanded into a variable refrigerant flow (VRF) system. One or more outdoor condenser units can supply refrigerant to many (in some cases over 100) indoor cassettes. Figure 3-6 shows the basic topology of a VRF system.

Unlike ductless mini-split systems, most VRF systems allow some indoor cassettes to operate in cooling while others operate in heating. A common situation in commercial systems is when a room with an exterior exposure requires heating while an interior room requires cooling. VRF systems can handle these situations using refrigerant management controls. This makes them popular for hotels and other buildings where many independently controlled zones are required.

Large VRF systems can contain miles of copper refrigerant piping routed throughout the building, as seen in Figure 3-7. There can be hundreds of brazed or compression connections connecting this tubing. There can also be hundreds of pounds of refrigerant contained in the system, much of it within interior space. A leak or rupture in an interior refrigerant line could create a very dangerous situation requiring evacuation of the building.

ROOFTOP UNITS

Air conditioning, ventilation, and in some cases heating, can also be supplied from rooftop forced-air units. These systems are commonly used on low-rise commercial or institutional buildings with flat roofs.

Figure 3-6

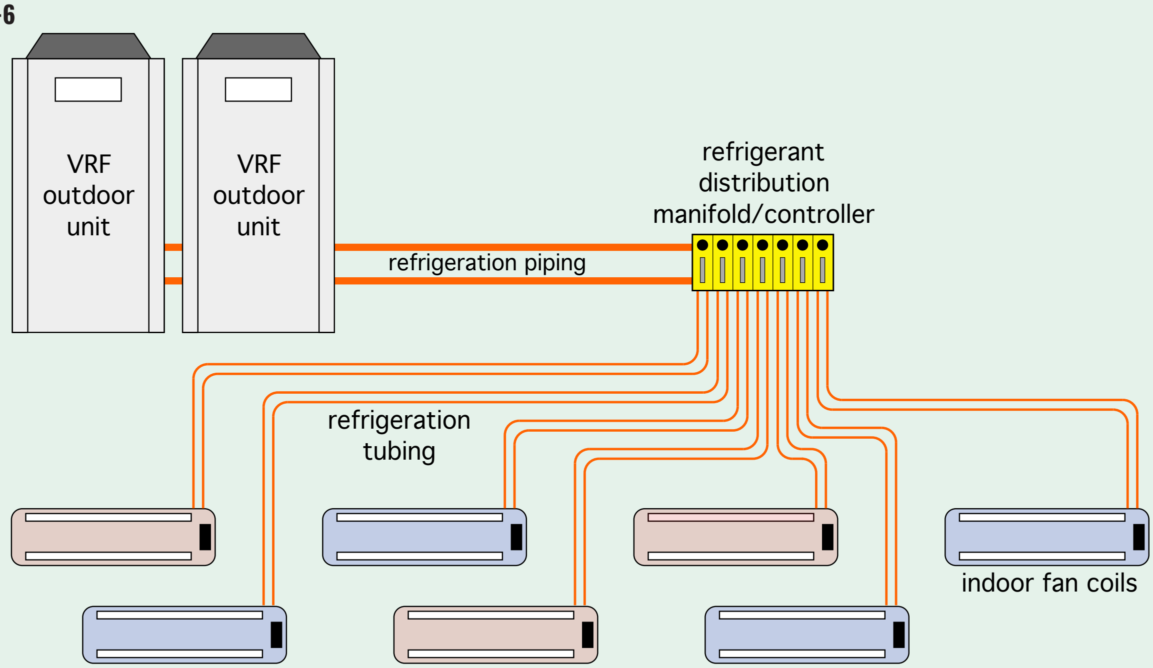


Figure 3-7

A typical rooftop unit mounts to a curb that has been secured and sealed to the flat roof of a building, as shown in Figure 3-8. Supply and return air ducts connect to the rooftop unit through the opening in the curb. Cooling is provided by a direct expansion refrigeration system. Outside air for ventilation enters the unit through a modulating damper. Heating can be supplied by the refrigeration system (operating as a heat pump), or by burners supplied by natural gas. The ducting system can be routed to several air diffusers within the building.

Because they are "all-air" systems, rooftop units are limited to applications where the supply and return ducting can be accommodated within the building. In most cases, this requires a mechanical plenum space above the suspended ceiling at each floor of the building. The height of the plenum space depends on the size of the ducting that must be routed through it. The ducting for all-air systems must be sized to accommodate the full (sensible + latent) cooling load, as well as the ventilation airflow requirement. Rooftop units are also

Figure 3-8

Figure 3-9a

Figure 3-9b

directly exposed to the elements, which tends to shorten their service life relative to cooling systems with mostly indoor components. Rooftop units that provide heating experience higher heat losses compared to heating appliances that are housed inside buildings. Some rooftop units can also be configured for ground mounting. However, some ground-mounted "rooftop" units infringe on building appearance, as shown in Figures 3-9a and 3-9b. The long-term maintenance of insulated exterior ducting can also prove challenging.

Although they are a convenient means of providing cooling in low rise commercial or institutional buildings, they do not offer the versatility of hydronic-based systems.

"CLASSIC" CHILLER/TOWER COOLING FOR LARGER BUILDINGS

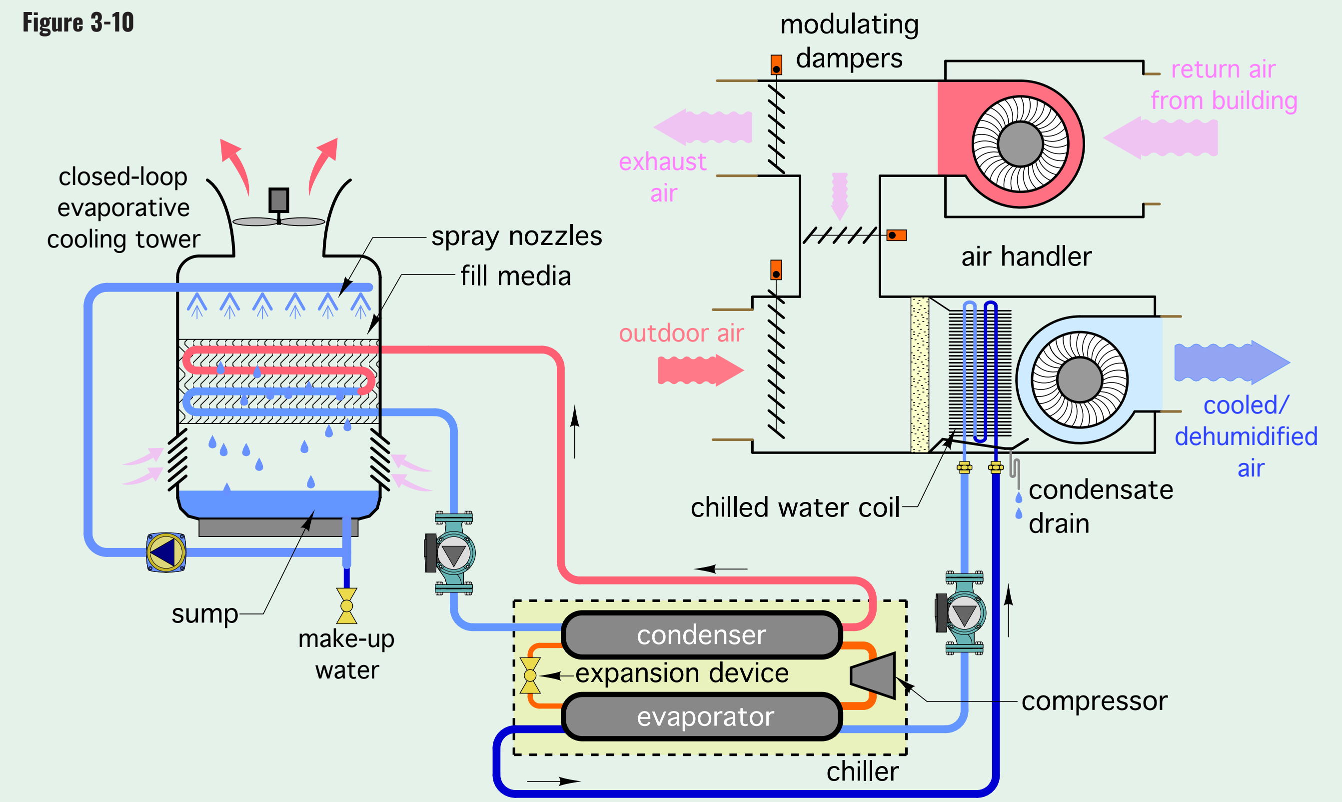
Chilled water cooling has been used for several decades in commercial and institutional buildings such as high-rise hotels and offices buildings. One of its chief attractions, as previously discussed, is the ability to move "cooling effect" over large distances using pipes that are much smaller than ducting of equivalent thermal capacity.

Figure 3-10 illustrates the make-up of a "classic" — and very basic — commercial chilled water-cooling system. It will be referred to as a "chiller/tower" system.

The chiller in this system is shown in the dashed lines. It contains the major components of any electrically powered refrigeration system: a compressor, evaporator, condenser and expansion device. The evaporator and condenser are both heat exchangers that separate the refrigerant from water but allow rapid heat transfer between these fluids.

Small chillers typically use hermetically sealed compressors (e.g., where the motor and mechanical portions of the compressor as enclosed in a sealed steel shell). Older chillers may have reciprocating compressors that use one or more pistons mounted to a crankshaft to compress the vaporous refrigerant. Modern hermetic compressors typically use orbiting scrolls to compress the refrigerant. Larger chillers often use screw compressors or centrifugal compressors driven by large 3-phase electric motors. In all cases, the compressed refrigerant exits the compressor at much higher

Figure 3-10



pressures and temperatures in comparison to those entering the compressor.

The condenser releases heat to a stream of water that flows through a cooling tower. As this warm water flows through the cooling tower, it is cooled by the combined action of upward airflow and a downward spray of water.

The two most common types of electrically powered chillers are:

- air-cooled chillers
- water-cooled chillers

Air-cooled chillers are usually located outside buildings, often on rooftops. They contain a complete refrigeration system, which absorbs heat from a stream of water supplied to the chiller from the building's hydronic cooling system. The temperature of this heat is increased by the refrigeration system and released to outside air through large refrigerant-to-air heat exchanger coils that serve as condensers for the refrigeration cycle. Large fans create a strong airflow past these coils. Cold water produced by the chiller returns to the building's hydronic cooling system.

Figure 3-11 shows a large, multi-stage, air-cooled chiller mounted outside a cool storage warehouse. This unit includes several staged hermetic compressors, along with multiple condenser coils and multiple fans.

Water-cooled chillers are usually located inside buildings and are more commonly used in larger systems. They also consist of fundamental refrigeration components including an evaporator, condenser, compressor and expansion device. However, water-cooled chillers transfer heat gathered from the building into a stream of water rather than directly to outside air. That water stream is circulated through a heat dissipation device, such as a cooling tower or geothermal earth loop, where the heat is rejected. Figure 3-12 shows an example of a large water-cooled chiller with a centrifugal compressor. This type of chiller is commonly used in large commercial and institutional chilled water-cooling systems.

In most applications, the typical leaving water temperature from modern chillers ranges from about 40 to 60ºF. However, some chillers can create significantly lower temperatures when operating with an antifreeze solution rather than water.

Cold water from chillers is routed through piping within the building to one or more terminal units. In Figure 3-10,

Figure 3-11

Courtesy of cfm Distributors, Inc

the terminal unit is an air handler equipped with a chilled-water coil. That coil consists of an array of smaller diameter copper tubes mechanically bonded to closely spaced aluminum fins. Figure 3-13 shows an example of a "thick" chilled-water coil with several rows of tubes.

Return air from the building, as well as some outdoor air to meet ventilation requirements, is pulled across the coil by a blower in the air handler. As it passes through the coil, the air is cooled (e.g., its temperature is lowered), and in most cases, the moisture content of the air is reduced. The latter is called dehumidification, or "latent" cooling, which will be explained in more detail in the next section.

The "conditioned" air leaving the air handler is pushed through a ducting system to several air diffusers in the building. Larger buildings generally have multiple air handlers to reduce the space requirement and routing constraints associated with large ducts.

Figure 3-12

Figure 3-13

The air handler shown in Figure 3-10 also has an exhaust blower and modulating dampers. These components are used to regulate the amount of outside air admitted to the building and to provide a controlled means of exhausting stale air from the building.

The cooling tower shown in Figure 3-10 is more specifically classified as an "evaporative" cooling tower. It is further classified as a "closed" cooling tower.

The water stream from the condenser remains within piping inside the tower and doesn't mix with the "sump" water being sprayed over the piping to create evaporative cooling. A large fan at the top of the tower pulls outside air through louvers at the base. Some cooling towers are designed as "open" units. The warm water stream from the chiller's condenser is directly exposed to outside air within the tower. The water stream leaving the cooling tower flows back to the condenser to absorb additional heat as it is being created.

Figure 3-14

Water from the sump at the base of the tower is pumped through an array of nozzles near the top of the tower. Some of this water evaporates when it contacts the piping carrying warm water from the chiller's condenser. This significantly increases the heat dissipation ability of the tower, especially on low-humidity days. Additional water is automatically added to the sump at the base of the tower to replace the water that has evaporated. The cooled (but not chilled) water then flows back to the chiller's condenser to absorb additional heat. Figure 3-14 shows an example of a cooling tower that is part of a commercial chilled-water cooling system.

There are many variations of the "classic" chilled-water cooling system just described. Larger systems may have two or more chillers. Those chillers may have the ability to modulate refrigerant flow, and thus vary cooling capacity. Multiple variable-speed circulators and modulating valves may be used to manage chilled water flow through the chilled-water distribution system. Two or more cooling towers, each with variable water and airflow capability might be used to adjust heat dissipation capacity to the current cooling load.

WATER-SIDE ECONOMIZERS

Some large chilled-water cooling systems are designed around "water-side economizers." These systems have a large plate & frame heat exchanger, commonly referred to as an "economizer," between the cooling tower and chilled-water distribution system. On days when the outdoor air temperature is relatively low, it's possible to supply the chilled-water distribution system directly from this heat exchanger without operating the chiller, or with the chiller operating at minimal cooling capacity. These modes of operation use much less electrical energy compared to a standard mode where all cooling is provided by the chiller. ASHRAE Standard 90.1 now requires evaluation of water-side economizers in otherwise "classic" chiller/tower cooling systems for installations in climate zone 4 or higher.

Figure 3-15 shows one piping configuration for a chiller/tower system that includes a water-side economizer.

The system in Figure 3-15 uses two 3-way diverter valves and one 3-way modulating valve. It also uses a SEP4 hydraulic separator to interface the evaporator side of the chiller to the system. This separator allows the chiller to maintain a minimum flow rate when the demand from a zoned distribution system is low. It also provides air, dirt and magnetic particle separation for the system.

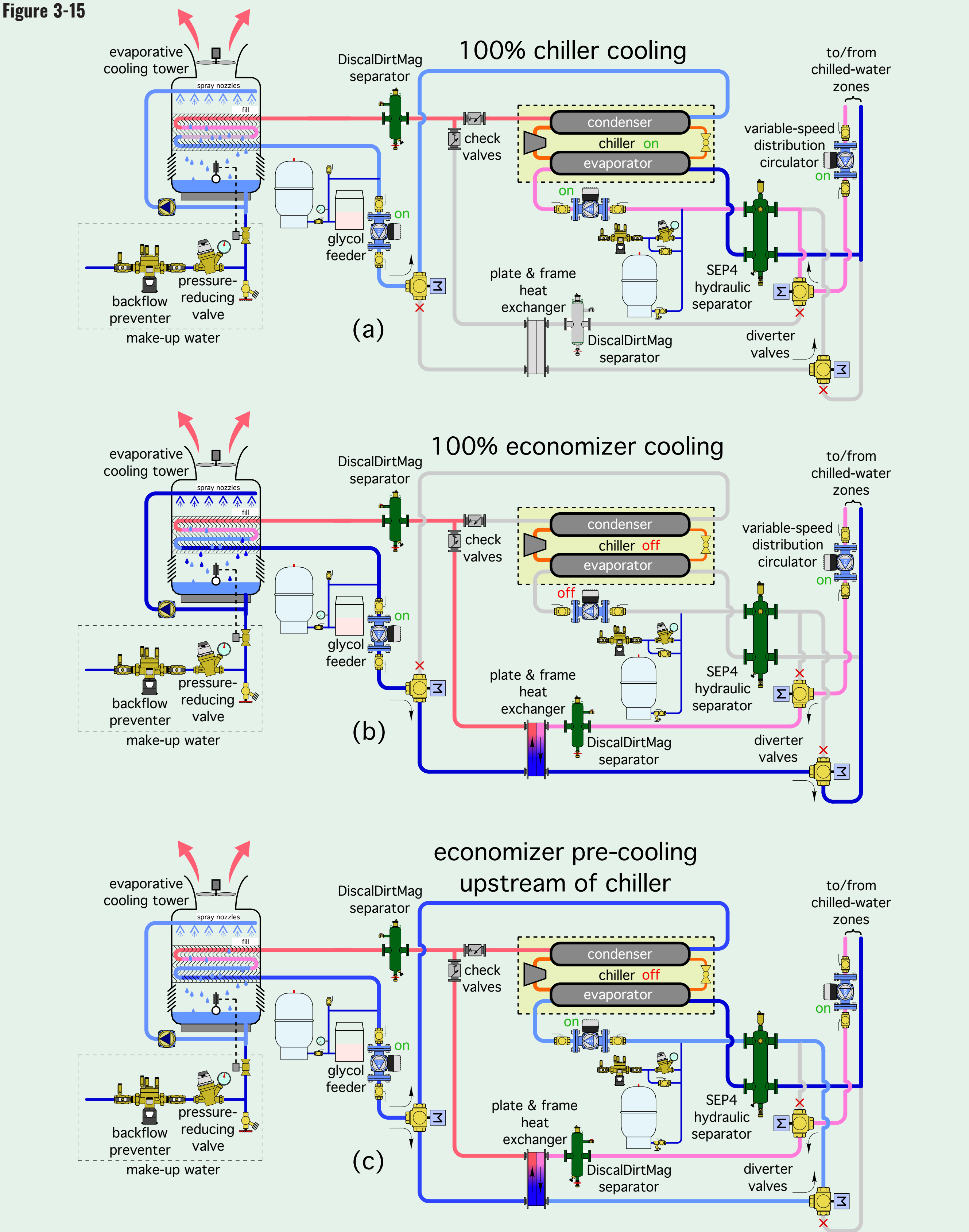
In Figure 3-15a, the position of the valves and circulator operation are in the mode where the chiller provides 100 percent of the cooling load. This mode applies when outdoor conditions are warmer than the return water from the cooling distribution system. The water-side economizer is not functioning in this mode.

In Figure 3-15b, the position of the valves and circulator operation allows the plate & frame heat exchanger (e.g., the economizer) to assume 100 percent of the cooling load. This mode applies when the outdoor temperature is below the return temperature from the cooling distribution system — but is still several degrees above freezing. This is also the mode that yields the greatest reduction in operating cost since the chiller is completely off. The only significant electrical power being used is for the circulators.

In Figure 3-15c, the system is using the plate & frame heat exchanger to pre-cool the water returning from the cooling load, and then passing that water to the chiller to further lower its temperature to the desired setpoint. The pre-cooling removes some of the chiller load and lowers its power demand. A modulating 3-way valve controls the portion of flow from the cooling tower to the heat exchanger and the chiller. This mode of operation is appropriate when the outdoor temperature is low enough to allow some pre-cooling, but not enough to drop the chilled water temperature to the desired setpoint. As the outdoor temperature decreases, more of the tower flow is directed through the heat exchanger and less through the chiller, and vice versa.

Cold water from the building is supplied to the sump in the cooling tower through an automatic make-up assembly. A float switch inside the cooling tower monitors the water level in the sump. When the water level drops, the float switch operates a solenoid valve to allow water into the sump. This make-up water passes through a

Figure 3-15



evaporative cooling tower
spray nozzles
fill
DiscalDirtMag separator
100% chiller cooling
to/from chilled-water zones
condenser
chiller on
evaporator
check valves
variable-speed distribution circulator
on
on
on
glycol feeder
backflow preventer
pressure-reducing valve
make-up water
plate & frame heat exchanger
SEP4 hydraulic separator
DiscalDirtMag separator
diverter valves
(a)
evaporative cooling tower
spray nozzles
fill
DiscalDirtMag separator
100% economizer cooling
to/from chilled-water zones
condenser
chiller off
evaporator
check valves
variable-speed distribution circulator
on
off
on
glycol feeder
backflow preventer
pressure-reducing valve
make-up water
plate & frame heat exchanger
SEP4 hydraulic separator
DiscalDirtMag separator
diverter valves
(b)
evaporative cooling tower
spray nozzles
fill
DiscalDirtMag separator
economizer pre-cooling upstream of chiller
to/from chilled-water zones
condenser
chiller off
evaporator
check valves
on
on
on
glycol feeder
backflow preventer
pressure-reducing valve
make-up water
plate & frame heat exchanger
SEP4 hydraulic separator
DiscalDirtMag separator
diverter valves
(c)

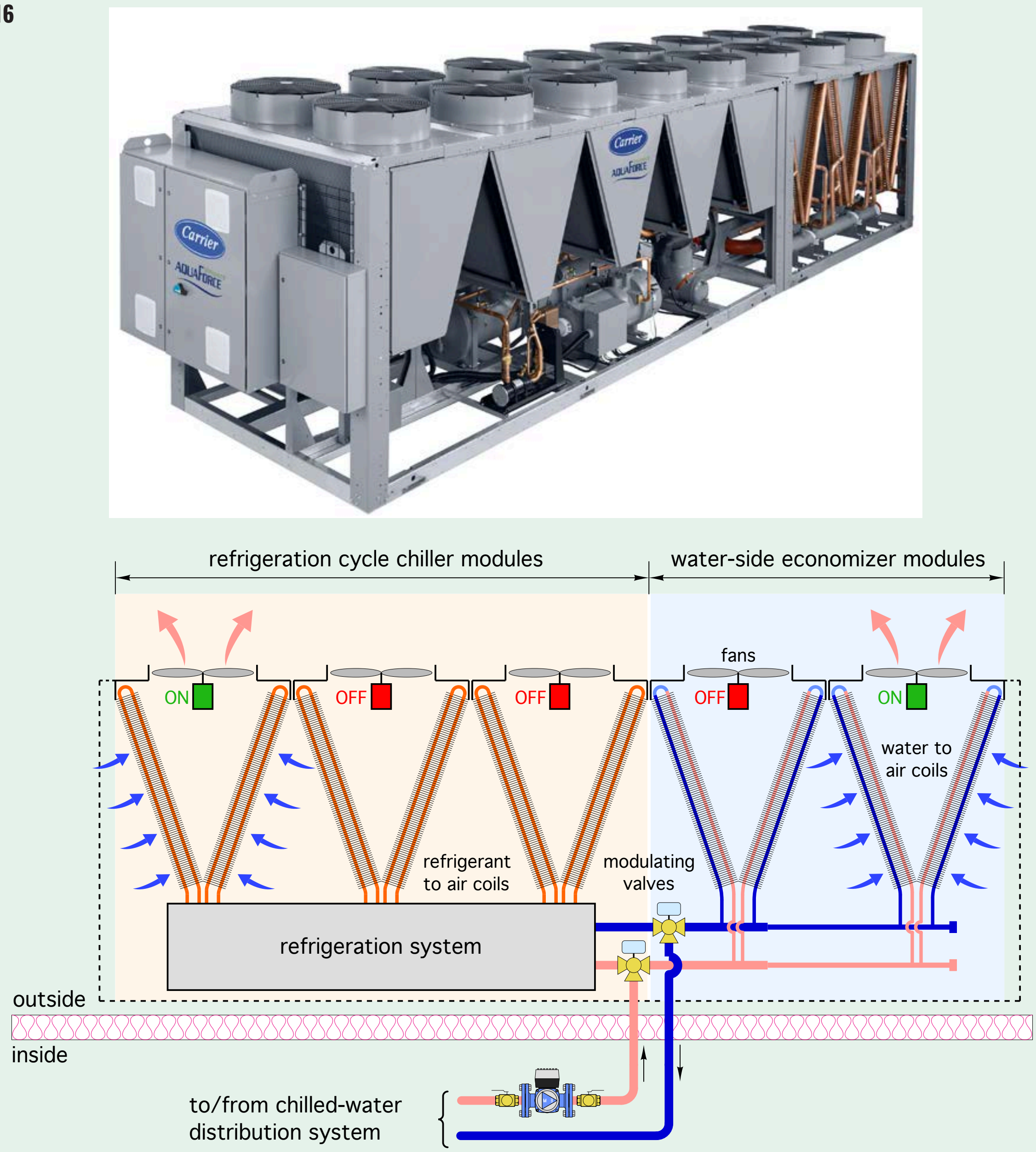
testable backflow preventer and a pressure-reducing valve upstream of the solenoid valve.

The closed cooling tower circuit operates with an antifreeze solution. If the tower will be inactive in winter, its sump water would be drained.

The heat exchanger in the cooling tower and the economizer heat exchanger are protected against dirt that could foul the heat transfer surfaces by DiscalDirtMag separators. These separators also collect air entrained with the water and eject it from the system.

Water-side economizers are especially well-suited to chiller/tower systems serving with a high percentage of sensible cooling load. Examples include data centers or industrial cooling in arid climates. These loads allow for higher chilled water temperatures, and thus increase the hours where the economizer can provide most, if not all, of the cooling requirement. They are also well-suited for applications where a high percentage of the sensible cooling load is handled by radiant panel cooling or chilled beams, and the latent load is handled by a separate chiller operating at a lower water temperature.

Figure 3-16



Some air-cooled chillers are now available with optional water side economizers. One modular approach places a section of water-to-air coils that form the economizer next to the refrigeration-based hardware, as illustrated in Figure 3-16.

The number of economizer modules, as well as the number of refrigeration-based modules, can be varied depending on the cooling capacity required, the ratio of sensible versus latent cooling, and the climate where the unit is installed. Applications with high sensible cooling ratios, or those using radiant cooling and/or chilled beams, allow for warmer chilled water temperatures, and thus extended the range of outdoor conditions over which the economizer can be used. Applications in hot/humid climates would likely require more refrigeration-based modules but still provide for limited use of the water-side economizer during winter. The modular approach also allows cooling capacity to be controlled in stages, and sometimes with variable-speed modulating within stages.

CHILLER EFFICIENCY

There are several indices that are used to express the cooling efficiency of chillers. The one that is used usually depends on the cooling capacity of the chiller and if it operates as an on/off device or a variable-capacity device.

For large water-to-water chillers, the efficiency index used to measure efficiency has been kilowatts of electrical input power per ton (12,000 Btu/hr) of cooling capacity. Modern centrifugal chillers operating at full load, and using HFC-134a refrigerant, can attain efficiencies in the range of 0.55 Kw/ton.

The efficiency of smaller chillers is often expressed as an Energy Efficiency Ratio. EER is the cooling output in Btu/hr divided by the electrical input power in watts, both of which are measured at industry standardized conditions for entering air and water.

Another efficiency index used for variable-capacity chillers is called Integrated Part Load Value (IPLV). This rating, which is based on AHRI standard 550/290, is meant to approximate the *seasonal* efficiency that a variable-capacity chiller could achieve when subject to a wide range of operating conditions over a representative cooling season. IPLV is based on Formula 3-1.

Formula 3-1:

$$IPLV = 0.01A + 0.42B + 0.45C + 0.12D$$

Where:
IPLV = Integrate Part Load Value
A = EER of chiller operating at 100% capacity
B = EER of chiller operating at 75% capacity
C = EER of chiller operating at 50% capacity
D = EER of chiller operating at 25% capacity

Formula 3-1 "weights" the EER of the chiller based on the assumption that it operates at full capacity for 1% of the cooling season, 75% capacity for 42% of the season, 50% of capacity for 45% of the season, and 25% capacity for 12% of the season. When chillers operate at reduced capacity, their instantaneous EER increases based on lower rates of heat transfer through the same amount of heat exchange area. The IPLV is meant to capture this improved performance under part load conditions in a standardized manner. It allows designers to compare the performance of different variable-capacity chillers.

CONTINUING IMPROVEMENT

Commercial chilled-water cooling systems have evolved significantly since first being used in the late 1920s. The two primary goals of this evolution have been to:

- Reduce the electricity used, or in the case of absorption chillers, the fossil fuel used, to create suitable chilled water.
- Improvements to hydronic delivery systems that reduce circulator power in chilled-water distribution systems.
- Other refinements including:
- Improved load matching between the devices producing chilled water and concurrent cooling loads.
- Use of naturally cool outside air or water when available to reduce (or eliminate) the need for refrigeration-based cooling.
- Use of alternative heat dissipation sub-systems, such as geothermal earth loops, that reduce reliance on cooling towers.
- Designing systems to reduce peak loading on electrical utilities.

Later sections provide overviews of several contemporary hydronic-based cooling systems for commercial and institutional buildings that leverage these refinements.

Optimal operation of any chilled-water cooling system requires good hydronic design and detailing. For example, it's imperative to keep all heat transfer surfaces in the chiller clear of dirt and scale. It's also important to properly balance water flow through each of several chilled-water terminal units in the system. Chilled-water systems also need to be free of air and have properly sized expansion tanks. These details can be accomplished with modern hydronic system components. Several will be discussed as you read on.

TEMPERATURE & HUMIDITY INDICES

Air is a combination of gases including oxygen (21%), nitrogen (78%), and small amounts of carbon dioxide, hydrogen and neon. Most air — with the exception of what is possible in a laboratory — contains water vapor.

All cooling systems create changes in the temperature and water vapor content (e.g., humidity) of interior air. Most cooling systems attempt to maintain the temperature and humidity of interior air at some set combination of conditions deemed to provide human comfort during warm weather. A typical combination would be an interior air temperature of 75ºF and an interior humidity level of 50% during cooling season operation.

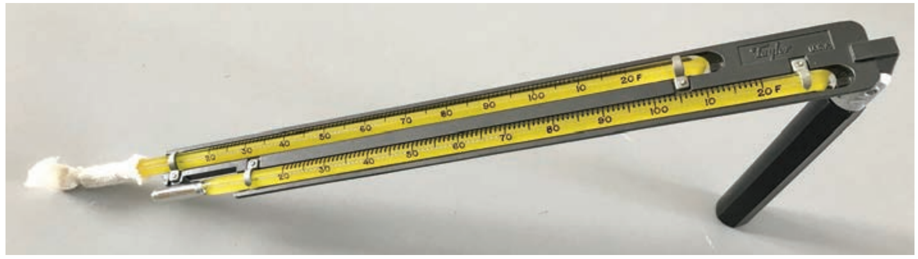
To appreciate how any cooling system affects the temperature and humidity of interior air, it's necessary to understand several specific physical quantities that are used to quantitatively describe the condition of air. Those physical quantities include:

- dry-bulb temperature
- wet-bulb temperature
- dewpoint temperature
- relative humidity
- absolute humidity (aka humidity ratio)

DRY-BULB TEMPERATURE

One of the simplest indexes that describe the condition of air is called dry-bulb temperature. It's simply the temperature of air read from a thermometer after the thermometer has stabilized within an interior or exterior space. The word "bulb" makes reference to mercury bulb thermometers that were extensively used for most air temperature measurements prior to development of electrically based temperature sensors such a thermocouples, thermistors and resistance temperature detectors (RTDs).

Figure 4-1



WET-BULB TEMPERATURE

Anyone who has ever stood in a wind while wearing wet clothes can attest to the cooling effect created as the moving air causes water to evaporate from those clothes. This same effect (e.g., cooling caused by evaporating water) can be used to determine the amount of moisture contained in air.

The classic method of doing so is to place a small cotton "sock" over the bulb of a mercury thermometer, saturate the sock with water, and then pass the thermometer through the air at a speed that expedites evaporation from the sock. After approximately 30 seconds, the moisture content of the sock will stabilize at a saturation condition, which implies that no further evaporation can occur. At that point, the thermometer with the sock over its bulb indicates the air's wet-bulb temperature.

The classic device for determining wet-bulb temperature, as well as dry-bulb temperature, is a sling psychrometer, an example of which is shown in Figure 4-1.

This instrument has two identical thermometers mounted side by side on a plate. The plate is mounted to a small shaft that extends to a handle. The person using the instrument first wets the small cotton sock stretched over the bulb of one thermometer. They then use the handle to spin both thermometers around the shaft for about 30 seconds. This causes the water on the sock to evaporate until its moisture content reaches that of the surrounding air. The evaporation of water from the sock cools the thermometer's bulb to a stable value, which is the air's current wet-bulb temperature.

The recorded dry-bulb and wet-bulb temperatures can be used along with a psychrometric chart to determine several indices related to the moisture content of the air. Psychrometric charts are described later in this section.

DEWPOINT TEMPERATURE

The ability of air to contain water vapor is very dependent on the air's temperature. The warmer the air, the greater its ability to contain water vapor. If a given sample of air is progressively cooled to lower and lower temperatures, it eventually reaches a saturation condition at which it contains all the water vapor it's capable of holding. The temperature at which this condition occurs is called the dewpoint of the air.

The dewpoint temperature is very important in the design and operation of cooling systems. For example, if air contacts a surface that cools it to (or below) its dewpoint temperature, water vapor in the surrounding air will condense into liquid water on that surface. A glass containing a cold drink and surrounded by warm/moist air is a common example of such condensation, as shown in Figure 4-2.

Figure 4-2

The formation of condensation is very desirable when it occurs *intentionally*, such as on the surfaces of a chilled-water coil within an air handler. This dehumidification is the intended action. However, *unintentional* condensation on surfaces of piping, circulator volutes, ducting or valves is a condition that must be avoided. If allowed to occur, the condensation can quickly create stains or oxidize steel and cast iron surfaces. Figure 4-3 shows a portion of a commercial chilled-water system that includes uninsulated CPVC pipe and a steel air/dirt separator. Both the piping and separator are covered with condensate. Repeated formation of condensate will eventually cause surface oxidation and staining of the floor under steel or cast iron components. Repeated formation of condensation

Figure 4-3

Figure 4-4a

Courtesy of Foley Mechanical, Inc.

can also encourage mold growth, and eventually destroy materials such as wood and drywall.

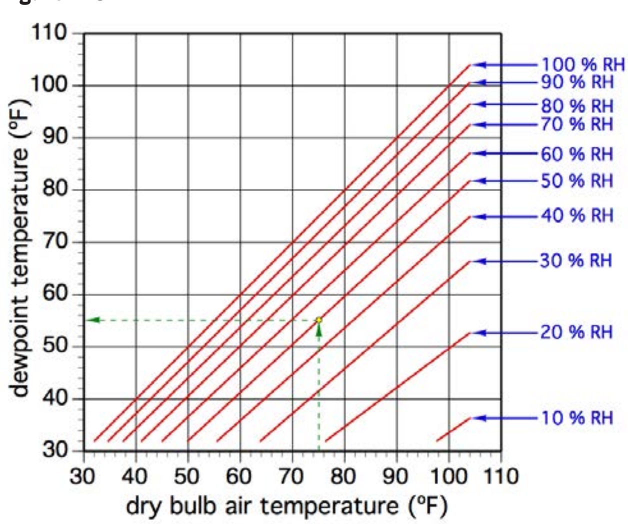
Most chilled-water cooling systems operate at water temperatures that are well below the dewpoint of surrounding air. To prevent surface condensation, all piping components in contact with chilled water must be insulated and that insulation must be vapor sealed. Figure 4-4 shows a pair of base-mounted circulators that are part of a commercial chilled-water cooling system. After all piping components were assembled and pressure tested, they were carefully insulated with an elastomeric foam. Note that the circulator motors and coupling assemblies are not — and never should be — insulated, to prevent heat buildup.

Figure 4-4b

Courtesy of Foley Mechanical, Inc.

See *idronics* 27 for information on the minimum thicknesses of elastomeric insulation needed to prevent surface condensation on chilled-water piping.

Figure 4-5



The dewpoint temperature of air is generally considered the same as the air's current wet-bulb temperature and can be read from a psychrometric chart. It can also be read from Figure 4-5 based on the air's dry-bulb temperature and relative humidity.

ABSOLUTE HUMIDITY

There are several indices used to represent the amount of water vapor contained in air. One that is particularly useful for cooling system design is called absolute humidity. It is commonly expressed as the pounds of water contained in one pound of dry air. Perfectly dry air would have an absolute humidity of 0. However, perfectly dry air is not naturally occurring. Air at 70ºF and 50% relative humidity has an absolute humidity of 0.0078 lb/lb indicating that one pound of air at this condition contains 0.0078 pounds of water, and 0.9922 pounds of the gases that make up air.

Absolute humidity is also sometimes stated in *grains* of water per pound of dry air. There are 7000 grains in one pound. So, air at 70ºF and 50% relative humidity also has an absolute humidity of about 55 grains per pound of dry air. The absolute humidity of air can be read from the vertical right side axis of a psychrometric chart. Another term sometimes used for absolute humidity is "specific humidity."

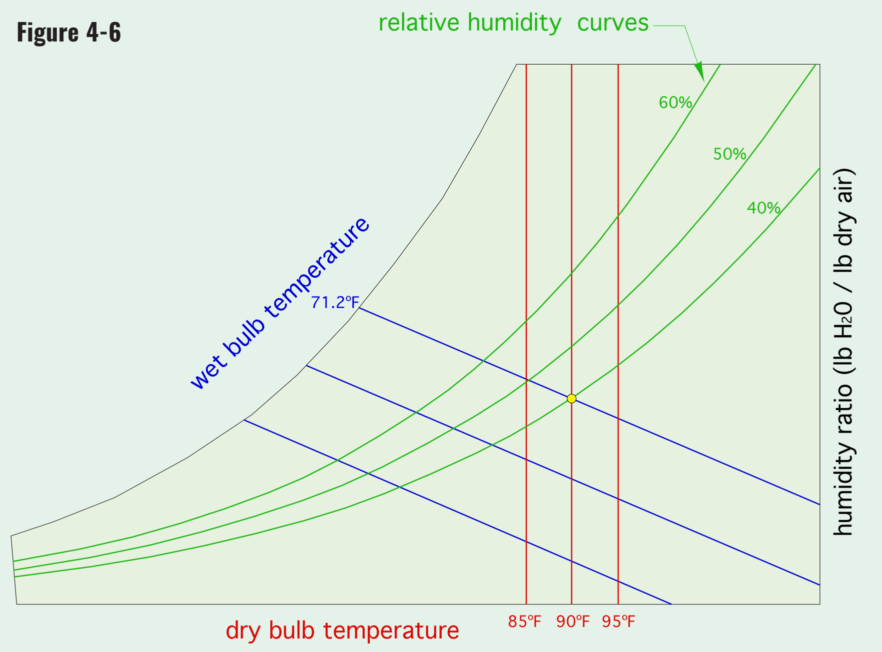
RELATIVE HUMIDITY

When people discussing weather talk about "humidity" they are usually referring to *relative* humidity. This is simply the ratio of the actual water mass in a given mass of air divided by the maximum mass of water the air could hold at its current temperature. Thus, air at 60% relative humidity only contains 60% of the water vapor that it is capable of holding at that temperature. A relative humidity of 100% means that the air is saturated and holding the maximum amount of water possible given the air's temperature. Air reaches 100% relative humidity at the dewpoint temperature. Relative humidity is plotted as a series of curves on a psychrometric chart.

PSYCHROMETRIC CHART

Willis Carrier is considered by most historians as "the father of air conditioning." In 1905, Carrier developed the first psychrometric chart that quantified the properties of air. That chart has been refined since its inception, but modern versions still bear a close resemblance to the original chart developed over a century ago. A full psychrometric chart is provided in appendix B. Several websites also have this chart available. Figure 4-6 shows a limited approximation of the psychrometric chart.

Figure 4-6



All the previously discussed thermodynamic properties of air are represented on a modern psychrometric chart. Dry-bulb air temperatures are listed along the chart's bottom horizontal axis (shown in red in Figure 4-6). Wet-bulb temperatures are listed along the curved upper left side of the chart (shown in blue in Figure 4-6). The relative humidity of the air is indicated by the curve that span from lower left to upper right (shown in green in Figure 4-6). The absolute humidity (lb H_2O/lb dry air) is listed along the right side vertical axis of the chart. Lines are drawn perpendicular to the bottom and right side axes, allowing dry-bulb temperature and absolute humidity values to be projected into the field of the chart. Sloping lines are also drawn from the wet-bulb temperatures indicated on the chart's upper left curved edge.

When any two of the thermodynamic properties plotted on this chart are known, the remaining properties can be determined. For example, if the dry-bulb and wet-bulb temperature of the air are known, the psychrometric chart can be used to find relative humidity and absolute humidity.

Some psychrometric charts also plot the density of air, the enthalpy of the air when fully saturated with moisture, and an index called sensible heat ratio.

SENSIBLE HEAT RATIO

As previously mentioned, all building cooling loads consist of sensible cooling (e.g., a measurable drop in dry-bulb temperature), and latent cooling (e.g., moisture removed from air). Different types of cooling systems, operating in different locations, need to operate at different proportions of these two load components. For example, a cooling system removing heat from a computer server room in an arid climate such as Santa Fe, NM, operates at a very high percentage of sensible cooling — likely 90+ percent. A commercial kitchen in New Orleans, LA, will face a much different cooling load, perhaps 60%sensible cooling and 40% latent cooling.

An index called sensible heat ratio has been developed to assess these different proportions of sensible and latent cooling load. The sensible heat ratio is defined as the sensible portion of the cooling load, divided by the total cooling load, which can be mathematically expressed as Formula 4-1:

Formula 4-1:

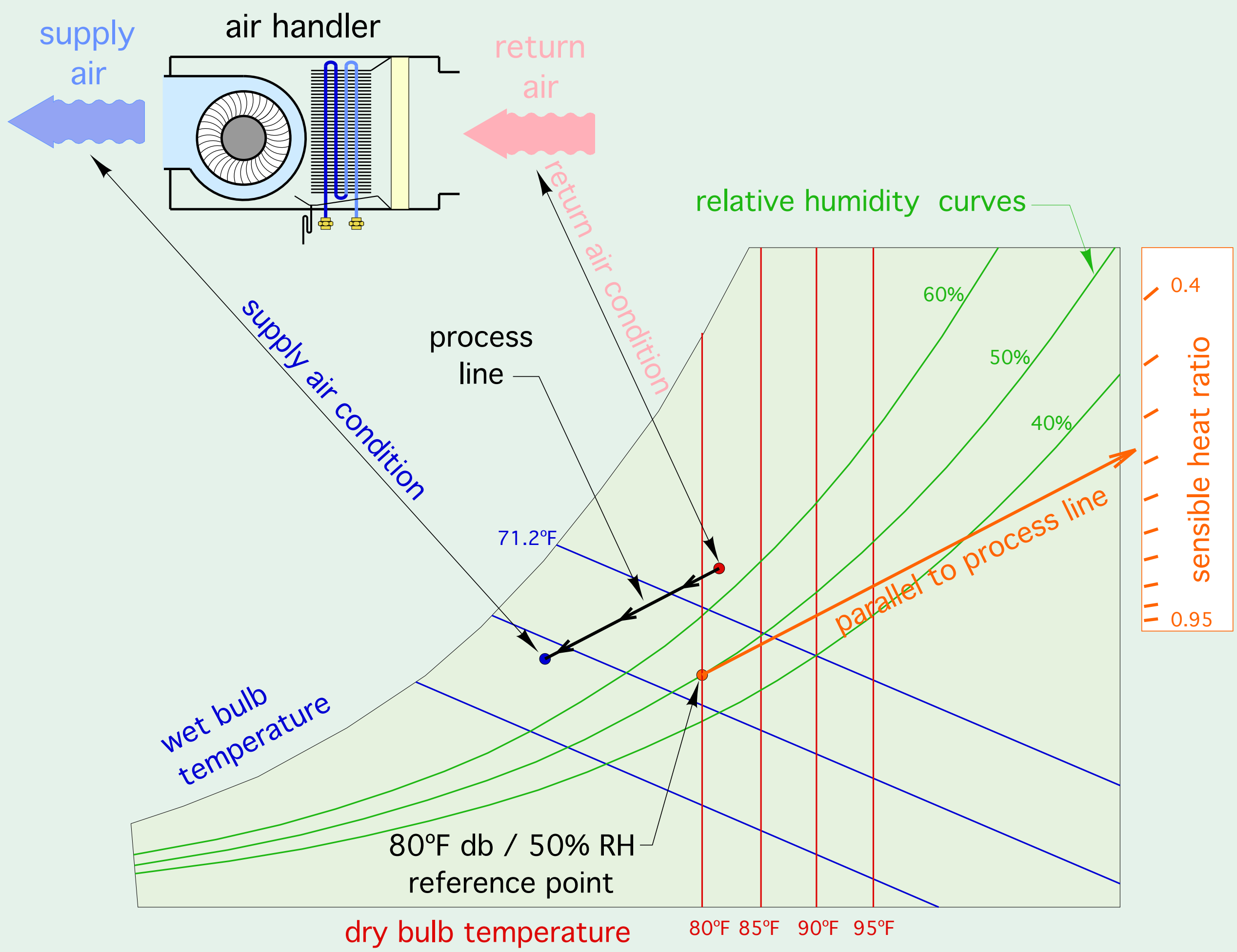
$$SHR = \frac{q_s}{q_T} = \frac{q_s}{q_s + q_L}$$

Where:
SHR = sensible heat ratio (unitless decimal number between 0 and 1)
q_S = sensible cooling load (Btu/hr)
q_T = total cooling load (Btu/hr)
q_L = latent cooling load (Btu/hr)

It is possible to determine the sensible heat ratio of a specifically defined cooling process using a psychrometric chart. The cooling process is usually defined starting with the condition of air *returning* from a space being cooled. That condition can be a combination of dry-bulb and wet-bulb temperatures, or a combination of either the wet-bulb temperature or the dry-bulb temperature, along with a known relative humidity. This combination establishes a point on a psychrometric chart. Another point on the chart is established based on the desired condition of the air *supplied* to the space being cooled. A line is drawn between these two points. That line is called the *process line.* An example of a psychrometric process line is shown in Figure 4-7.

The sensible heat ratio is found by constructing another line, parallel to the process line, and passing through a reference point defined as (80ºF db/50% RH) on the psychrometric chart. The parallel line is extended to the upper right portion of the chart and passes

Figure 4-7



through a scale labeled sensible heat ratio (or sometimes labelled sensible heat factor).

Knowing the sensible heat ratio helps determine the type of cooling equipment used. Cooling scenarios with high SHR are well-suited to cooling using thinner cooling coils, radiant panels or chilled beams. Cooling scenarios with low SHR will require significantly more dehumidification and favor thicker, multi-row cooling coils.

COOLING LOADS

The starting point for designing any chilled-water cooling system is to establish the building's cooling load. Those loads involve several influencing factors, each of which can change with time. These include:

- Building occupancy
- Solar heat gain through windows and opaque exterior surfaces of buildings
- Operation of interior equipment (computers, office machines, production equipment, cooking, etc.)
- Lighting schedules
- Ventilation requirements during occupied and unoccupied periods
- Building thermal mass
- Combined temperature and humidity (e.g., enthalpy) of outside air

Some of these influencing factors do not add moisture to spaces, and thus only effect the sensible cooling load.

The sensible portion of the cooling load consists of:

- Heat transfer through the exposed building surfaces
- Heat gain from sunlight through windows
- Heat gain from interior lighting
- Heat gain from equipment operating within the conditioned space
- A *portion* of the metabolic heat output from occupants

Other sources of internal heating such as occupant respiration/perspiration, cooking, showering, washing and infiltration of humid outside air increase the humidity of inside air. These sources contribute to the *latent* cooling load as well as the sensible cooling load. In this context, the word latent implies "hidden." Latent heat gains do not change the temperature of interior air but have a profound impact on human thermal comfort and the total cooling load of the building.

The latent portion of the cooling load consists of:

- Moisture that diffuses through exterior building surfaces due to vapor pressure difference
- Moisture carried in by air infiltration
- Moisture carried in by ventilation air
- Moisture given off by occupant respiration and perspiration
- Moisture given off by cooking, bathing, showering, laundry, interior plants and washing

The total cooling load, at any time, is the sum of the sensible and latent heating loads. This is almost never equal to the sum of the *maximum* expected rate of heat gain (sensible or latent) attributable to each of the previously stated influencing factors.

For example, the thermal mass of buildings having large exposed interior masonry walls can absorb some of the sensible internal heat gains, especially if that mass has been cooled prior to the occurrence of those sensible heat gains. This causes a time delay between the occurrence of the internal gain and when that energy becomes part of the instantaneous cooling load.

The diversity of the factors that influence total cooling loads in commercial and industrial buildings has been extensively studied over several decades. This study has led to development of detailed methods for determining hourly cooling loads.

The often-cited reference for methods of calculating cooling loads in commercial and institutional building is chapter 18 in the 2017 *ASHRAE Book of Fundamentals*. The methods recommended by ASHRAE include *Radiant Time Series* method and *Transfer Function* Method. These methods include consideration of how all the factors listed above interact over time to determine the building's cooling load profile.

There are also several commercially available software programs that can be used to estimate sensible and latent cooling loads in commercial and institutional buildings. Some examples include:

Chvac: from Elite software
Block Load (BLK): from Carrier Corporation
Right-N: from Wrightsoft (based on ACCA Manual N)

VENTILATION COOLING LOAD

Commercial and institutional buildings intended for human occupancy require ventilation. The recommended rates of outdoor air supplied to the building vary with the intended occupancy and activity.

For example, ASHRAE recommends the following ventilation rates in a school building:

Occupied area	ventilation rate
Classroom	15 CFM per person
Science laboratory	17 CFM per person
Art room	19 CFM per person
Cafeteria	9 CFM per person

Other ventilation rates apply to office and industrial buildings. In general, the greater the potential for undesirable substances such as carbon dioxide, carbon monoxide, aldehydes, smoke, chemical vapors or biological contaminants (e.g., viruses and bacteria) to be created or present in the space, the higher the recommended ventilation rate.

The cooling load associated with providing ventilation depends on the dry-bulb temperature and absolute humidity of the inside and outside air. The sensible portion of the ventilation cooling load is based on the change in dry-bulb temperature between that of the incoming outside air, and the desired delivery temperature and humidity ratio to the occupied spaces. It can be estimated using Formula 4–2:

Formula 4-2

$$q_{vent(sensible)} = 1.08(CFM)\left(T_{out} - T_s\right)$$

Where:
$q_{vent(sensible)}$ = sensible cooling load due to ventilation (Btu/hr)
CFM = ventilation airflow rate (ft^3/minute)
T_{out} = outdoor air dry-bulb temperature (ºF)
T_s = dry-bulb temperature of air delivered to ventilated spaces (ºF)

The latent cooling associated with the ventilation air can be estimated based on the ventilation airflow rate and change in moisture content between outside air and the air supplied to the occupied spaces. It can be estimated using Formula 4-3:

Formula 4-3:

$$q_{vent(latent)} = 4840(CFM)\left(AH_{out} - AH_s\right)$$

Where:
$q_{vent(latent)}$ = latent cooling rate of ventilation air stream (Btu/hr)
CFM = ventilation air flow rate (ft^3/minute)
AH_{out}= absolute humidity of incoming outdoor air (lb_{water}/lb_{dryair})
AH_S= absolute humidity of air supplied to the conditioned space (lb_{water}/lb_{dryair})

The absolute humidity of air can be read on the vertical axis of a psychrometric chart (shown in appendix B).

It's worth noting that condensing water vapor into a liquid to remove it from an air stream is an energy-intensive process. The average heat removal necessary to condense one pound of water from typical summertime air is about 1076 Btu.

Consider a classroom of 29 students and 1 instructor (30 people in total). Based on ASHRAE recommendations, the ventilation air supply should be 15 CFM per person, and thus the ventilation airflow rate would be 450 CFM. If the outdoor air was at 90ºF dry-bulb temperature and 80% relative humidity, its absolute humidity (found on a psychrometric chart) would be 0.0245 pounds of water per pound of dry air. If the ventilation air stream was required to have a delivery temperature of 60ºF and relative humidity of 50%, its absolute humidity would be 0.0056 pounds of water per pound of dry air. Based on these conditions, the total ventilation air cooling load (e.g., the sum of the sensible and latent loads) would be:

Sensible cooling load for ventilation air:

$$q_{vent(sensible)} = 1.08(450)\left(90-60\right) = 14{,}580\frac{Btu}{hr}$$

Latent cooling load for ventilation air:

$$q_{vent(latent)} = 4840(450)\left(0.0245-0.0056\right) = 41{,}164\frac{Btu}{hr}$$

Total cooling load for ventilation air = 14,580 + 41,164= 55,744Btu/hr

Under the stated conditions, the cooling capacity needed to lower the temperature of the ventilation air is only about 26% of the total cooling load for the ventilation air. This demonstrates that a large amount of cooling capacity is needed to reduce the moisture level of what — in this case — is "tropical" outside air. Also keep in mind that these calculations only pertain to the cooling load associated with ventilation. The sensible and latent cooling load of the classroom due to other heat gains (lights, solar, etc.) and moisture gains from people and diffusion through building envelope surfaces must be added to the ventilation cooling load to arrive at the total cooling load.

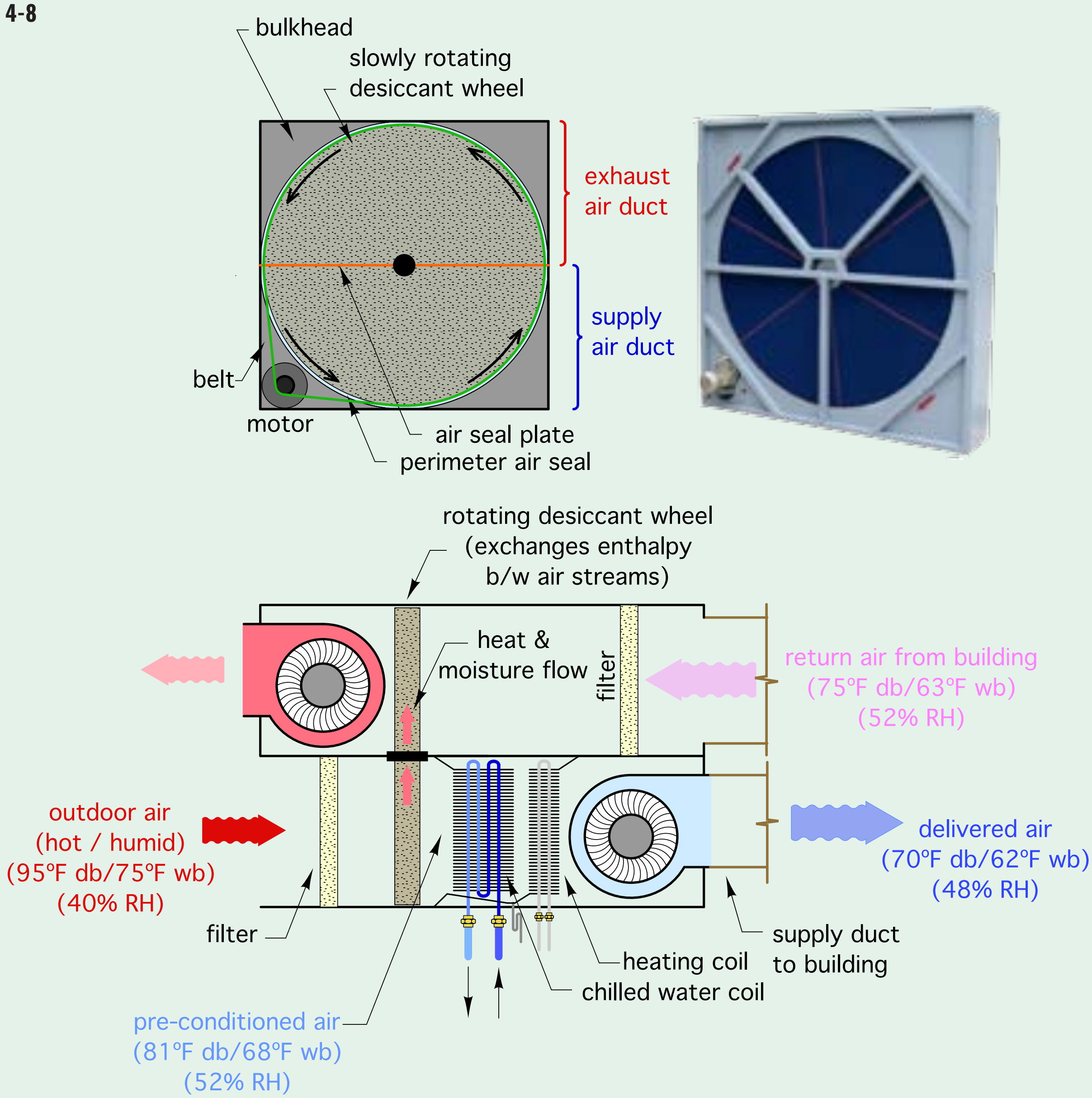
ENERGY RECOVERY VENTILATION SYSTEMS

Some modern air handlers that also control ventilation airflow are equipped with heat recovery or energy recovery devices that capture a portion of the cooling effect available from the exhaust air stream and use it to partially condition the incoming air stream. Heat recovery systems work with sensible heat exchange (e.g., heat transfer based solely on the temperature difference between incoming and leaving air streams). Energy recovery systems exchange both sensible heat and latent heat. The latter are also sometimes called "enthalpy" recovery systems.

Figure 4-8 illustrates the concept of an air handler containing an "enthalpy wheel."

The enthalpy wheel is composed of a cellular media that allows air to flow through it. This media contains a desiccant that absorbs moisture when cooled and releases moisture when heated. The upper half of the wheel is in the exhaust duct, while the lower portion of the wheel is in the supply air duct. The wheel is slowly rotated by an electric motor. The media in the lower half of the wheel absorbs heat and moisture from the incoming outside air causing the air temperature and moisture level to drop. This "pre-conditioned" air decreases the cooling and dehumidification load on the downstream cooling coil. The cells that absorbed the heat and moisture eventually rotate into the upper duct, where they release heat and moisture into the exhaust air stream. Enthalpy wheels can significantly reduce the cooling load associated with building ventilation. They can also recover up to 70% of the thermal energy that would otherwise be exhausted with stale air during the heating season.

Figure 4-8


bulkhead
slowly rotating
desiccant wheel
exhaust
air duct
supply
air duct
belt
motor
air seal plate
perimeter air seal
rotating desiccant wheel
(exchanges enthalpy
b/w air streams)
heat &
moisture flow
filter
return air from building
(75ºF db/63ºF wb)
(52% RH)
outdoor air
(hot / humid)
(95ºF db/75ºF wb)
(40% RH)
delivered air
(70ºF db/62ºF wb)
(48% RH)
filter
heating coil
chilled water coil
supply duct
to building
pre-conditioned air
(81ºF db/68ºF wb)
(52% RH)

Section 2 discussed the types of chillers often found in larger commercial and institutional cooling systems, specifically:

- air-cooled chillers
- water-cooled chillers

There are many electrically powered air-cooled and water-cooled chillers available in the 10- to 500-ton range (1 ton = 12,000 Btu/hr). These chillers are designed to operate on 3-phase electrical power, which is usually available in urban areas, as well as areas designated for commercial building development. However, 3-phase power may not be available in rural areas. This doesn't mean that buildings requiring commercial-capacity cooling cannot be sited in such areas, which may only have single phase utility power available.

One solution could be an array of 2 or more single phase chillers operated in stages. In this context, the word "chiller" could represent a device that only provides chilled water, or a heat pump that can provide chilled water for cooling and warm water for heating. Single phase chillers are available with capacities up to about 5 tons. Thus, a 20-ton cooling load could be handled by four 5-ton chillers operated in stages.

The use of multiple smaller-capacity chillers has advantages and limitations. One advantage is redundancy — if one of the chillers is down for maintenance, the other chillers can still provide partial load capacity.

Another advantage is load matching. An array of four fixed-speed 5-ton chillers could provide four incremental steps of cooling capacity. This allows instantaneous cooling capacity to be better matched to variable cooling loads. In some systems, it may also eliminate the need for a buffer tank.

When multiple chillers are used, they are typically controlled by a system that uses a closed-loop PID (proportional-integral-derivative) control algorithm that's configured for a specific chilled water target temperature to the load. The controller varies the number of operating chillers to "steer" the delivered water temperature as close as possible to that target temperature.

One potential drawback of multiple chillers is that they typically require more piping and valving relative to a single large chiller. Another is that the "footprint" required for multiple chillers is usually more than a single chiller of equivalent capacity.

MULTIPLE AIR-TO-WATER HEAT PUMPS AS CHILLERS

An array of two or more single phase electric chillers (or heat pumps) can be piped very similar to how multiple boilers are piped. Figure 5-1 shows an example based on using three monobloc-style air-to-water heat pumps as chillers.

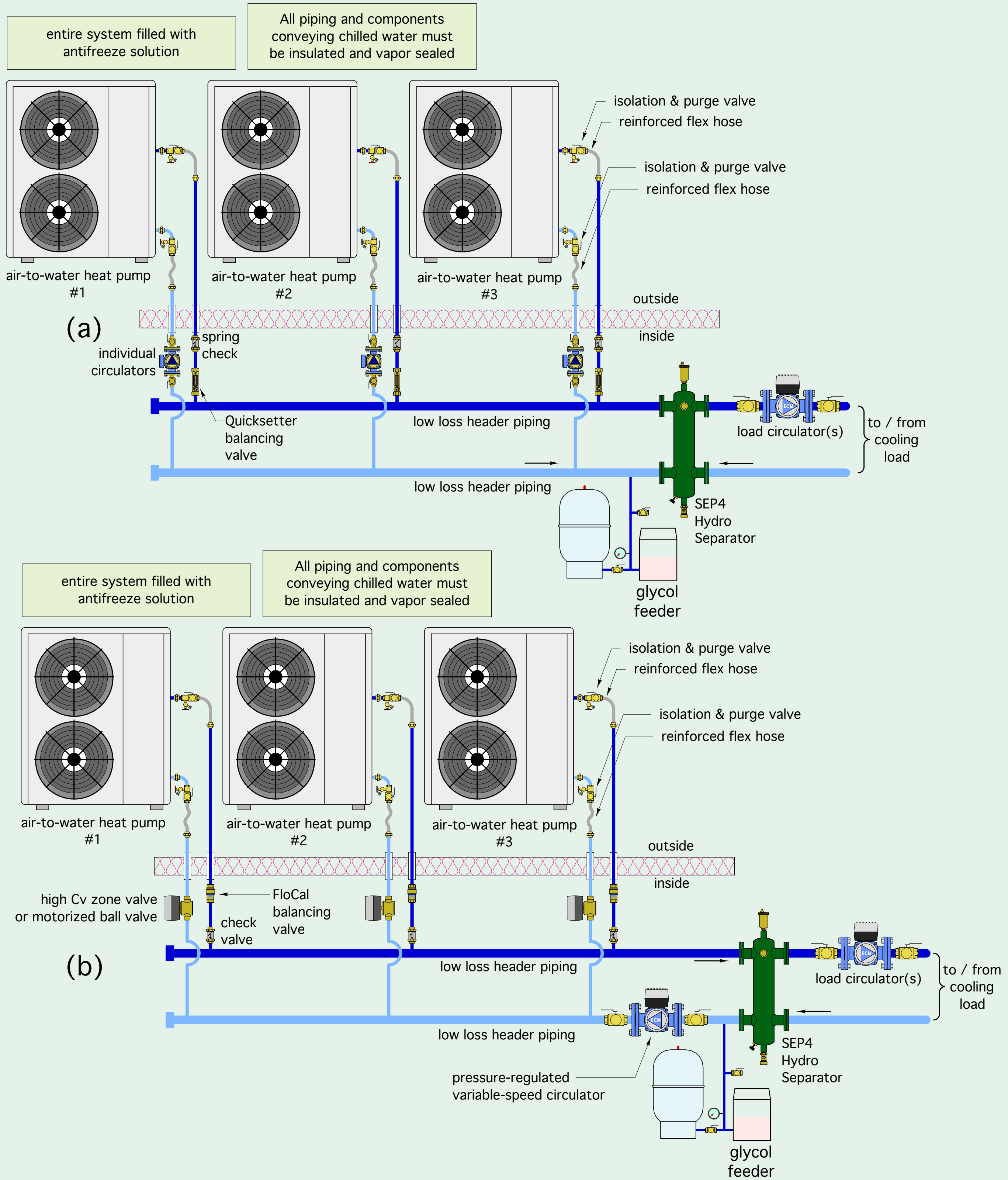
The system in Figure 5-1a uses individual circulators to create flow through each heat pump. Each circulator operates only when its associated heat pump operates. Each heat pump has combination isolation/purging valves and unions, allowing it to be isolated from the remainder of the system and serviced, or even temporarily removed if necessary. Each heat pump also has reinforced flexible hose connectors that reduce vibration transfer to rigid piping. A spring-loaded check valve is installed in the piping at each heat pump to prevent reverse flow when some heat pumps are on and others are off. A QuickSetter automatic balancing valve in the branch piping to each heat pump allows flow rates to be balanced.

All the heat pumps connect to low head loss (generously sized) headers that lead to a hydraulic separator equipped with a magnet for capturing dirt and iron oxide particles. The entire system operates with a propylene glycol antifreeze solution sufficiently concentrated to prevent freezing in any exterior piping. All piping is also insulated to prevent condensation and heat gain during cooling, as well as to reduce heat losses during heating mode.

The system in Figure 5-1b uses a single variable-speed, pressure-regulated circulator to provide flow to all heat pumps. Each heat pump branch is equipped with a high Cv zone valve (or motorized ball valve) that opens when its associated heat pump is operating. The variable-speed circulator operates in constant differential pressure mode and automatically adjusts speed based on the number of operating heat pumps. FloCal balancing valves automatically maintain a pre-set flow rate through each heat pump when it operates. *This piping configuration would have a lower electrical power demand in comparison to use of individual circulators for each heat pump.*

The piping shown in Figure 5-1 assumes that heat pumps — rather than dedicated chillers — were selected so that the system can provide heating and cooling. The piping at the hydraulic separators is optimized for heating mode operation (e.g., heated fluid passed through the upper portion of the separator while cooler fluid passes through the lower portion). If *dedicated chillers* were used for a system that only provides cooling, the coldest fluid leaving the chillers should pass through the lower portion

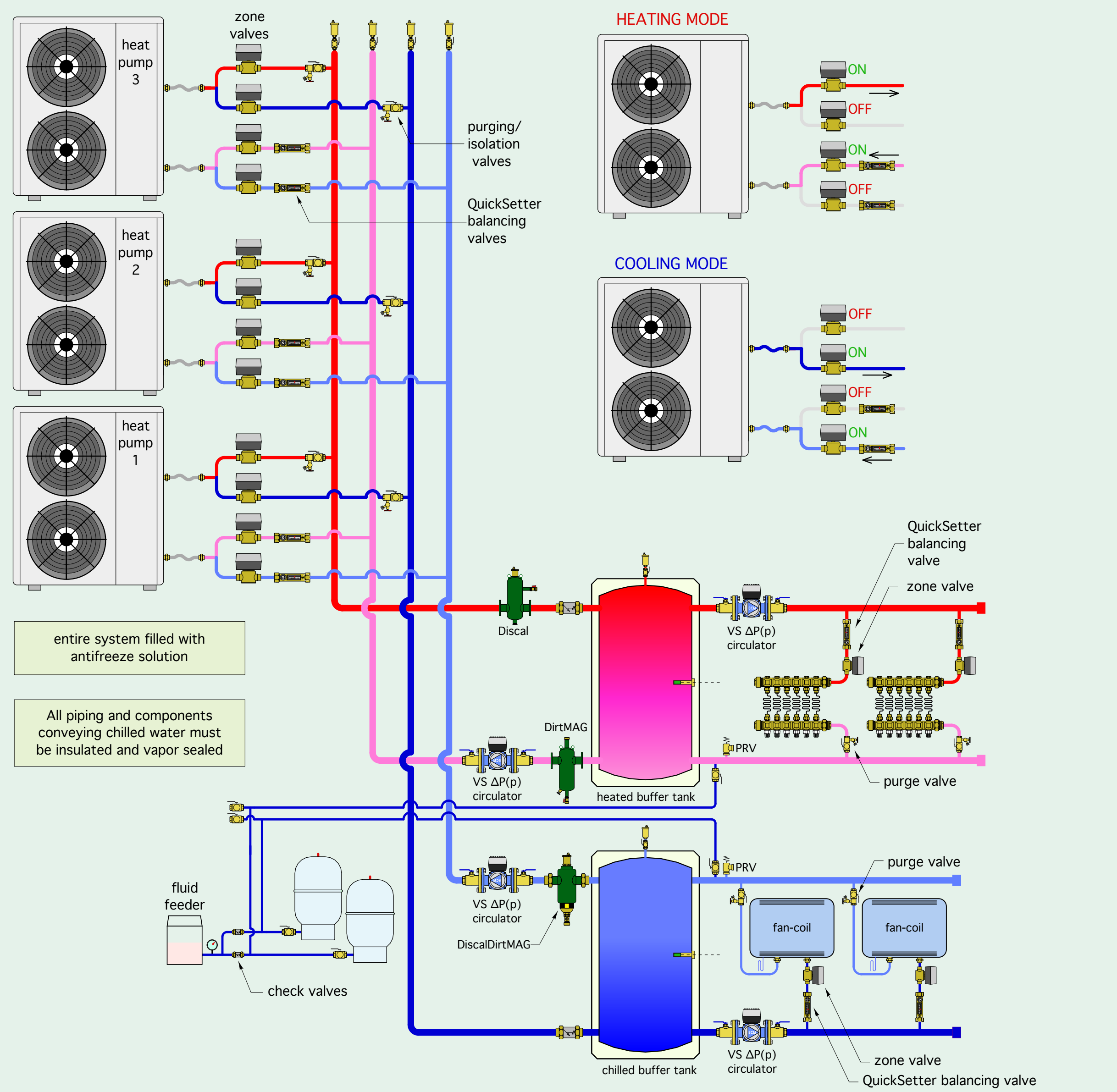
Figure 5-1



entire system filled with antifreeze solution
All piping and components conveying chilled water must be insulated and vapor sealed
isolation & purge valve
reinforced flex hose
isolation & purge valve
reinforced flex hose
air-to-water heat pump #1
air-to-water heat pump #2
air-to-water heat pump #3
outside
inside
(a)
individual circulators
spring check
Quicksetter balancing valve
low loss header piping
load circulator(s)
to / from cooling load
low loss header piping
SEP4 Hydro Separator
glycol feeder
entire system filled with antifreeze solution
All piping and components conveying chilled water must be insulated and vapor sealed
isolation & purge valve
reinforced flex hose
isolation & purge valve
reinforced flex hose
air-to-water heat pump #1
air-to-water heat pump #2
air-to-water heat pump #3
outside
inside
high Cv zone valve or motorized ball valve
FloCal balancing valve
check valve
(b)
low loss header piping
load circulator(s)
to / from cooling load
low loss header piping
SEP4 Hydro Separator
pressure-regulated variable-speed circulator
glycol feeder

Figure 5-2

Figure 5-3



of the separator. The slightly warmer fluid returning from the cooling load would pass through the upper portion of the separator. The warmer fluid would have a slightly lower solubility for dissolved gases, and thus enhance the air-separating function within the upper portion of the SEP4 separator.

Figure 5-2 shows an array of six nominal 5-ton air-to-water heat pumps that supply chilled water for cooling a small industrial facility equipped with chilled beam terminal units (to be discussed in section 7). These heat pumps also supply warm water for heating.

It's important to understand that the piping shown in Figures 5-1a and 5-1b allows any of the heat pumps to operate, in the same mode, as required by the load, but does *not* allow some of the heat pumps to operate in cooling mode while others simultaneously operate in heating mode. Mixed mode operation is useful in some buildings, especially during swing season conditions. On a cool day in fall or spring, perimeter areas of a building may require heating, while core areas of the building require cooling.

Simultaneous heating and cooling from a multiple heat pump array is possible, but requires more piping, valves and controls. Figure 5-3 shows one example of piping that could be used with an array of three air-to-water heat pumps to allow simultaneous heating and cooling.

This system has a "hot" buffer tank and a "cold" buffer tank. The system controls monitor the temperature of each tank and call for a heat pump to turn on when the temperature of either tank deviates slightly from its target temperature or outside of a set temperature range. The water temperature in the hot buffer tank would be regulated based on outdoor reset control. The water for the cold buffer tank would be maintained between upper and lower temperature setpoints, such 45ºF and 60ºF whenever a cooling load is present.

When a heat pump is called to operate, the zone valve pairs associated with its mode of operation open. The status of the heat pump's reversing valve is also set. A variable-speed pressure-regulated circulator operates to create flow through the appropriate buffer tank. The speed of the circulator is based on *proportional* differential pressure control. The speed automatically increases or decreases depending on how many heat pumps are operating.

The valving at each heat pump is also arranged so that the zone valves or heat pump can be isolated from the balance of the system if necessary for service.

Each buffer tank provides hydraulic separation between the heat pump circulators and the load circulators.

The heating zones are supplied by low temperature radiant panels. Flow to each manifold station is controlled by a zone valve. Each manifold station piping assembly is also equipped with a balancing valve and purging valve. These valves are arranged so that each manifold station and its associated zone valve can be completely isolated from the balance of the system if necessary for service.

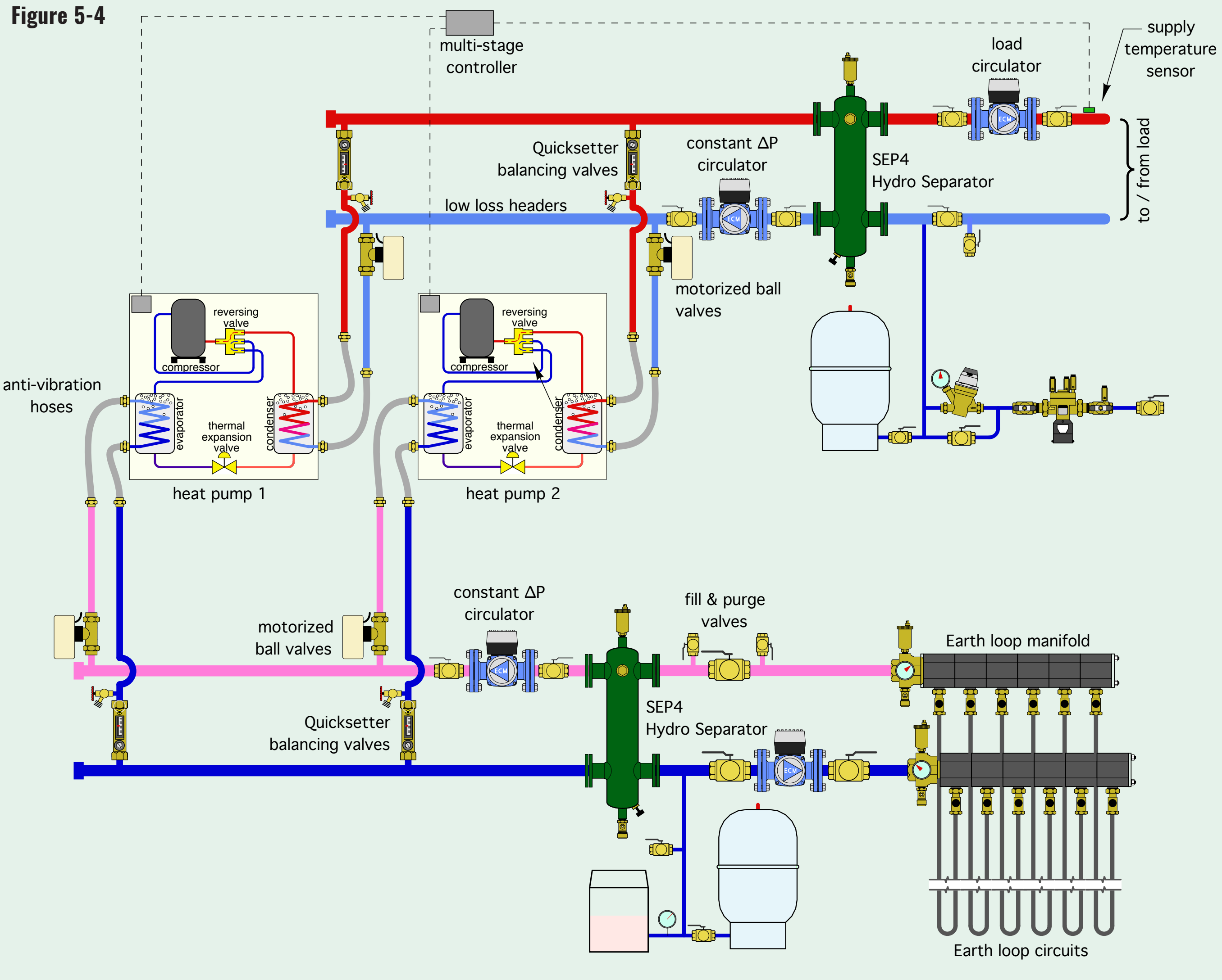
The cooling zones are supplied by fan-coils. Each fan-coil piping assembly is equipped with a balancing valve and purging valve. These valves are arranged so that each fan-coil and its associated zone valve could be completely isolated from the balance of the system if necessary for service.

Because this system can simultaneously supply heating and cooling, one mode of operation must take priority when staging the heat pumps. Several possibilities exist. For example, during heating season the ability to maintain adequate water temperature in the "hot" buffer would likely be the priority. Once that temperature is established, at least one of the heat pumps would be allowed to operate in cooling mode if a cooling load is present. During the cooling season, it's likely that all heat pumps would be prioritized to satisfy the cooling load.

MULTIPLE WATER-TO-WATER HEAT PUMPS AS CHILLERS

The same piping concepts discussed for multiple air-to-water heat pumps can also be applied to multiple water-to-water heat pumps. In some applications, the heat pumps would be dissipating heat to earth loops or "open" groundwater sources. In other applications, the heat of rejection from the heat pumps may be used to heat domestic water, or to preheat process water that would eventually be used for washing or sterilization in food processing facilities. Figure 5-4 shows an array of two water-to-water heat pumps that could provide either staged heating capacity or staged cooling capacity, but not both at the same time.

Flow to both sides of each heat pump is provided by a variable-speed pressure-regulated circulator. A pair of motorized ball valves, one on the condenser side of the heat pump and the other on the evaporator side, open when their associated heat pump operates. These valves have relatively high Cv ratings, and thus minimize head loss in their associated branch piping paths. An end switch within the actuator of each motorized ball valve closes when the valve reaches 80% of its fully open status. This switch can be used as part of a "safety switching" circuit to verify that both flow pathways (e.g., evaporator



and condenser) through the heat pump are open before allowing the heat pump's compressor to operate.

The two SEP4 separators allow the possibility of independent variable-speed control of all four circulators. They also provide high-efficiency air, dirt and magnetic particle separation in all portions of the system. The flow rate through the earth loop could be varied depending on how many heat pumps are operating.

This approach to flow control (e.g., use of variable-speed high-efficiency circulators and valve-based branch flow control) significantly reduces the electrical power required to operate the hydraulic portions of the system. This is especially relevant in cooling mode operation, since all electrical power supplied to the circulators adds to the overall cooling load or heat dissipation requirements of the system.

Figure 5-5 shows an array of two 5-ton rated water-to-water heat pumps that are used in a maintenance facility. These heat pumps provide staged heat input to a radiant floor slab. They also provide cooling through multiple chilled-water air handlers.

Figure 5-5

Each water-to-water heat pump is equipped with a desuperheater for domestic water heating (uninsulated piping connected to upper two ports on each heat pump). The two pipes that lead to the earth loop and the two pipes the lead to the heating and cooling distribution system are connected to the heat pump using reinforced flexible hose connectors to minimize vibration transfer. The earth loop piping and distribution piping are insulated with an elastomeric foam to prevent surface condensation and conserve energy. Each heat pump is mounted so that front and side service panels can be accessed.

HEAT RECOVERY CHILLERS

One variation on standard air-cooled chillers is to add an internal refrigerant-to-water heat exchanger that can transfer some or all of the heat of rejection produced during cooling mode operation to a stream of water instead of dissipating all that heat to outside air. The heated water can then be used for loads such as space heating, domestic water heating, pool heating or other process heating. Any heat not transferred to the warm water stream is dissipated to outside air by a condenser coil and fan.

Figure 5-6 shows an example of a 5-ton rated heat recovery chiller. One set of pipes connects to the chilled-water distribution system, the other set connects to the heated water distribution system.

Figure 5-6

Courtesy of Multiaqua

Heat recovery chillers are ideal in situations where simultaneous heating and cooling are required or possible. Potential applications include:

- Cooling a building while simultaneously heating a swimming pool
- Cooling the core area of a building while simultaneously heating perimeter areas
- Cooling a building while providing domestic water heating
- Cooling a building while providing hot water for an industrial process such as laundry, food preparation or greenhouse soil warming

Because both the chilled water stream and hot water stream are being used, the *effective COP* of a heat recovery chiller can be significantly higher than the COP of a stand-alone chiller or a stand-alone heat pump. In theory, the effective COP of a heat recovery chiller would be:

$$COP_e = \frac{Q_t}{e_{input} \times 3413} = \left(2 \times COP_h - 1\right)$$

Where:
COP_e = Effective COP based on total useful energy output, which is the sum of heating and cooling outputs
Q_t = Total useful energy output of hot and cold water streams (Btu/hr)
e_i = electrical input to heat recovery chiller (kilowatts)
COP_h = Measured heating COP of the unit based on *heat output only* and electrical input
3413 = Conversion factor from kilowatt to Btu/hr

Thus, a heat recovery chiller operating at a COP of 4.0 based on measured *heating output* would achieve an *effective* COP of (2x4-1) = 7 based on the total useful output of heating and cooling flow streams.

Some heat recovery chillers can also switch between outside air or a geothermal earth loop as the “source” for low temperature heat, or the media to which excess heat (e.g., heat not used by the hot stream load) is dissipated.

Figure 5-7 shows a possible application using a heat recovery chiller to supply either 2-pipe air handlers or 4-pipe air handlers. Note that each air handler can independently operate in heating or cooling mode. It would also be possible to add different types of chilled-water terminal units or low temperature heat emitters such as radiant panels to the distribution systems shown.

Figure 5-7

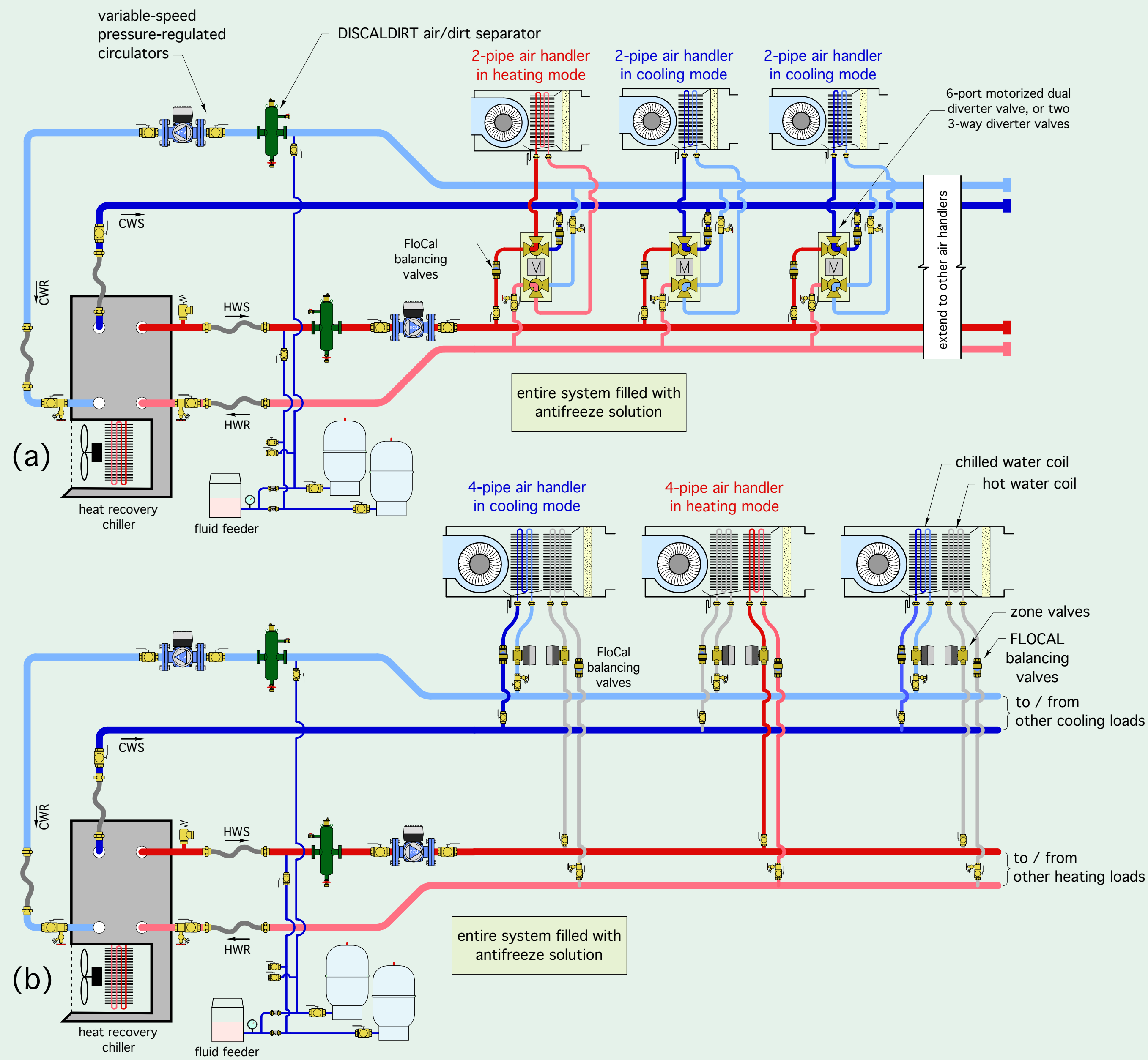
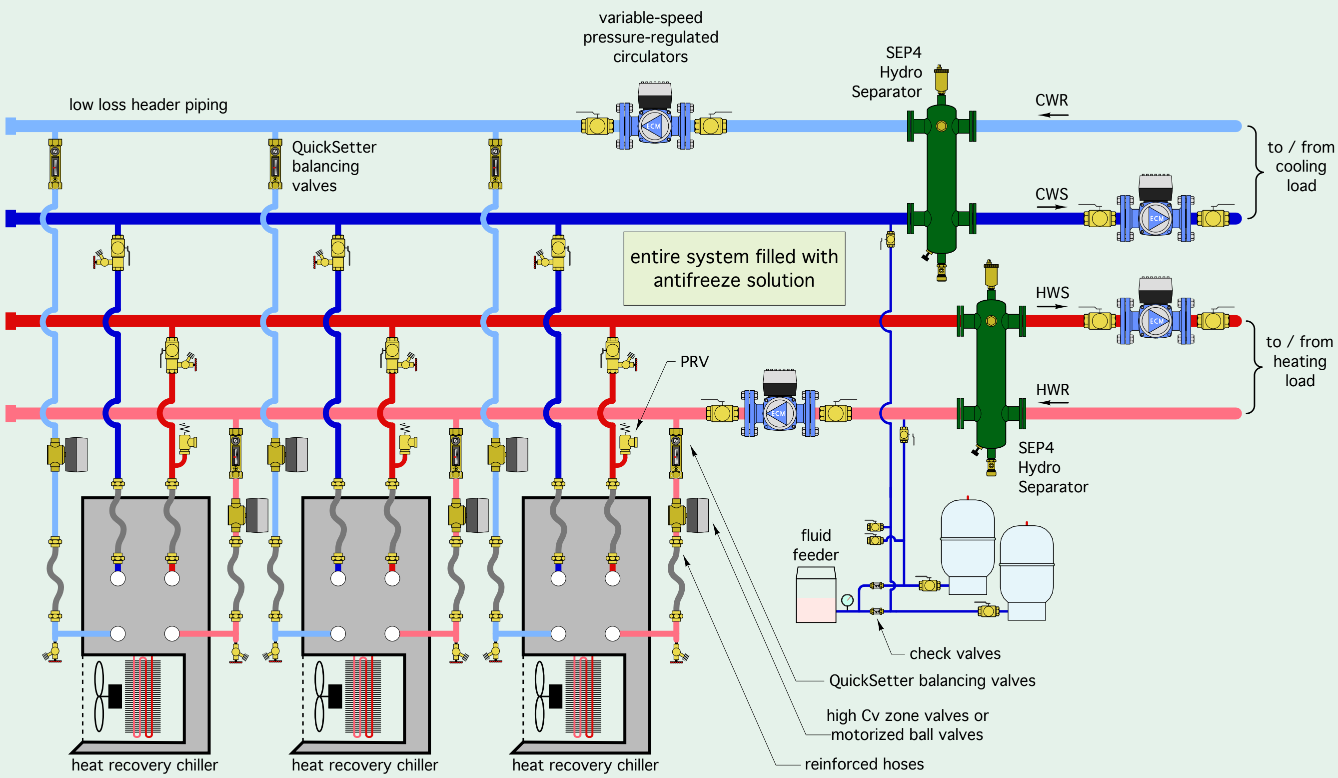


Figure 5-8 shows how multiple heat recovery chillers can be piped to provide staged input to either the heating or cooling distribution system.

The "hot side" and "cold side" of each chiller are connected to low head loss header piping that leads to two hydraulic separators. Depending on the chiller's flow requirements, a pair of high Cv zone valves or a pair of motorized ball valves open and close to allow or prevent flow through the chiller. Two variable-speed pressure-regulated circulators automatically adjust speed to maintain a constant differential pressure across the headers. The distribution system is shown with variable-speed pressure-regulated circulators. The hydraulic separators prevent interaction

Figure 5-8



between the chiller circulators and the load circulators. They also provide high-efficiency air, dirt and magnetic particle separation. Each chiller can be fully isolated for service if necessary.

Because the chillers are located outside, the entire system operates with an antifreeze solution. A fluid feeder is used to maintain system pressure as air is vented from either side of the system.

AIR-COOLED CONDENSERS WITH DIRECT EXPANSION HEAT EXCHANGERS

It's also possible to produce chilled water by combining one or more outdoor condensers with an external refrigerant-to-water heat exchanger. The heat exchanger serves as the evaporator for the refrigeration system. This heat exchanger must be properly sized based on the refrigerant used, the desired evaporator temperature, the superheat setting of the thermal expansion valve in the condenser unit, and the required chilled-water temperature and flow rate.

In systems that require 10 or more tons of cooling, it's possible to use two more 5-ton single phase condensers and operate them in stages based on the demand for chilled water. Figure 5-9 shows how each condenser unit is piped to a separate heat exchanger. A flow switch should be installed on each branch, as shown in Figure 5-9a.

Each flow switch verifies that a suitable flow rate exists through the water side of the refrigerant-to-water evaporator before allowing the condenser unit to operate. Insufficient water flow through the heat exchanger could result in freezing. Notice that all piping, as well as the lower body of the flow switch, are insulated to prevent condensation formation on components carrying chilled water.

Figure 5-9a

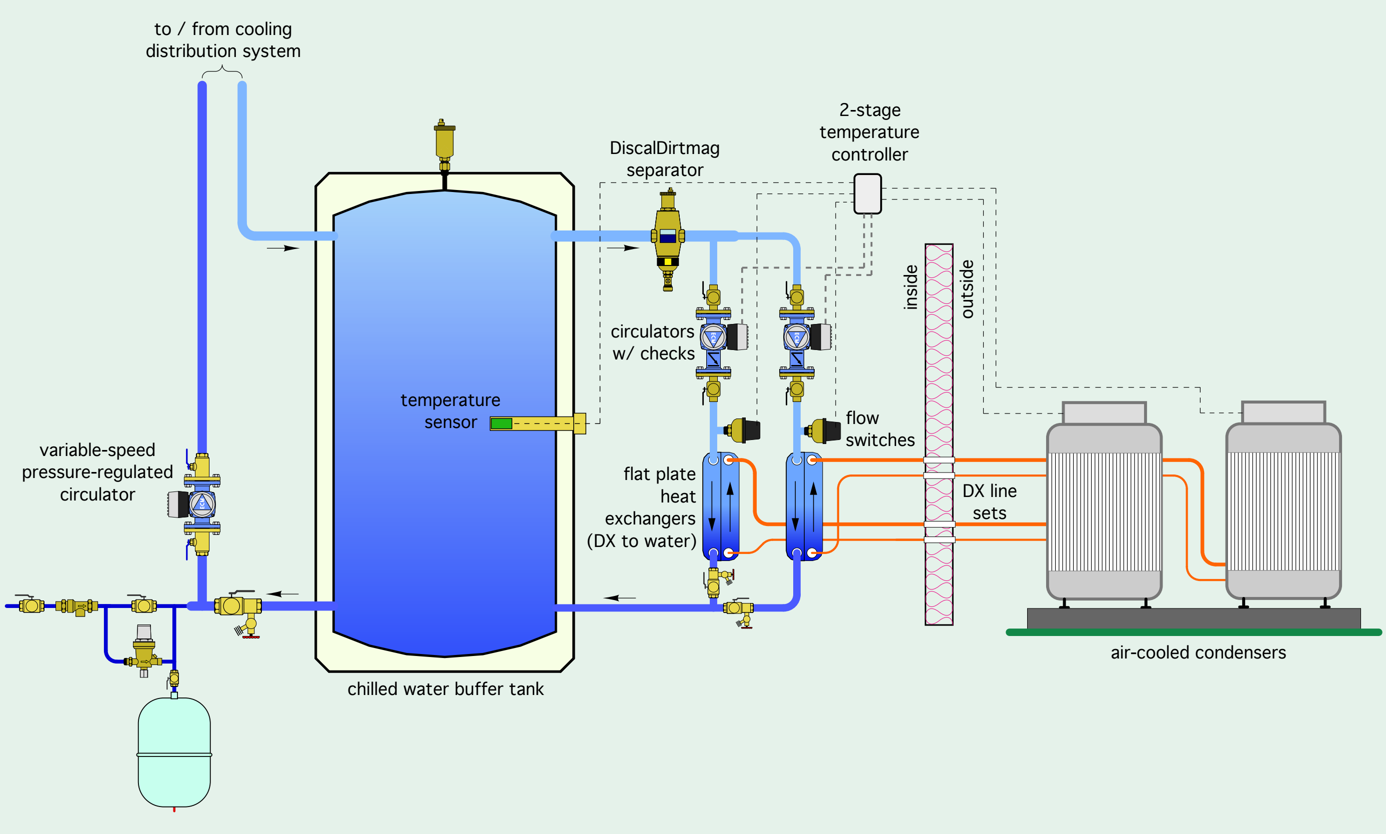


Figure 5-9b

Figure 5-9c

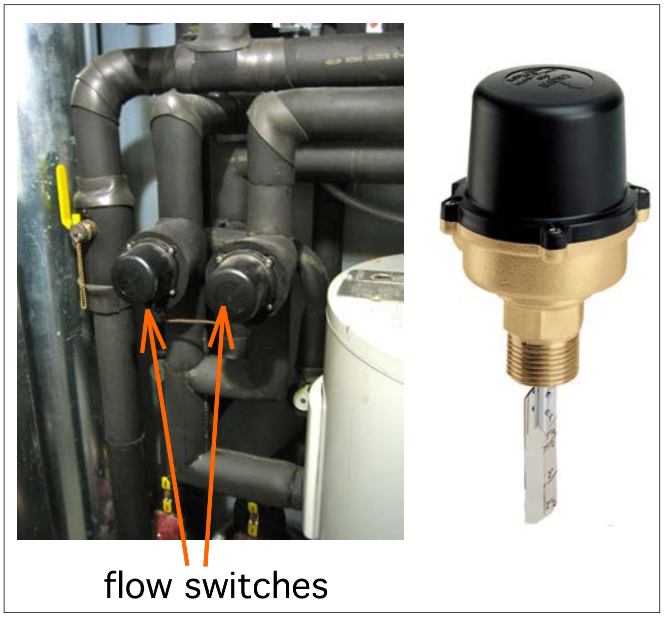


Figure 5-10

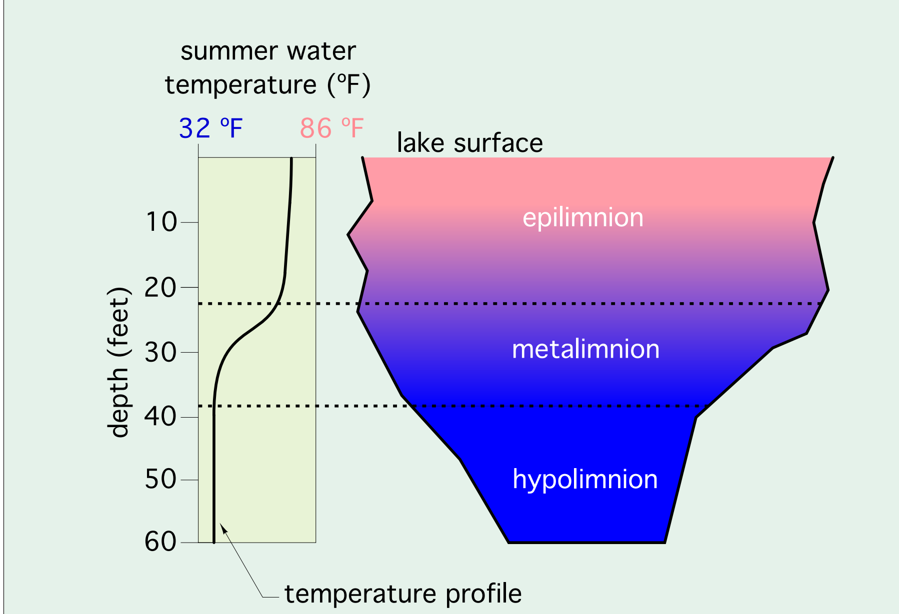


Figure 5-11

Courtesy of HPAC Magazine

LAKE SOURCE COOLING

Anyone who has swam in a lake in the northern half of the United States can verify that the water temperature gets noticeably cooler just a few feet below the surface, even on a hot summer day. Measurements have shown that the water temperatures at depths of approximately 40 feet or more below the surface of lakes in climates that experience several weeks of below-freezing air temperature during winter, experience very little temperature variation on an annual basis. This is illustrated in Figure 5-10.

Water attains its maximum density at a temperature of 3.98ºC (39.2ºF). In cold climates, water at this temperature will permanently accumulate in the lowest regions of lakes having depths of at least 40 feet. Water at such a temperature is very adequate for chilled-water cooling systems, if it can be accessed from shore.

Lake source cooling has been done on several large-scale projects. The cool waters of Cayuga Lake in upstate New York provide approximately 20,000 tons of cooling capacity for Cornell University in Ithaca, NY. The city of Toronto, Ontario, also uses the cool water from about 600 feet below the surface of Lake Ontario to provide 59,000 tons of cooling for downtown high-rise office spaces. The lake water absorbs heat from a large district cooling system that is connected to several high-rise office buildings in downtown Toronto. Several large plate & frame heat exchangers, seen in Figure 5-11, provide the interface between the lake water and the water in the district cooling system.

The cost and complexity of such systems is beyond what would be practical for a single commercial building. Its feasibility depends on a cluster of larger buildings, located reasonably close to a large lake, that could connect to a district cooling system.

The feasibility study of a potential lake water cooling system should begin with assessment of any codes or regulations regarding use of lake water. Different requirements may apply to "navigable waters" versus lakes in which boating is not allowed. If these criteria allow further pursuit of the project, water temperature measurements should be taken at several depths below the lake surface to determine available water temperatures during months when cooling is needed. This helps determine the required depth of the water intake pipe.

See *idronics* #13 for more details on lake water cooling systems.

Figure 5-12

Courtesy of AWEB Supply.

LAKE WATER HEAT EXCHANGERS

Another method for harvesting the cooling potential of deep lakes or large ponds uses a closed plate-type heat exchanger, an example of which is shown in Figures 5-12 and 5-13.

This lake heat exchanger consists of multiple stainless steel plate assemblies. Each assembly has two stainless steel plates that are specially patterned to create flow channels and are welded together along their perimeter. Each plate assembly is connected to a supply and return header. The overall assembly is welded to a stainless steel base that supports the plates several inches above the surface they rest on. HDPE tubing is routed from the headers to the shore.

This type of heat exchanger is designed to be used with water source heat pumps. However, when properly sized and used in a lake where the water temperatures at the lakebed are stable and relatively low, it could provide direct heat exchange to a chilled-water cooling system. Figure 5-13 shows one concept for such a system.

This system uses a SEP4 hydraulic separator to provide air, dirt and magnetic particle separation of the system water. A variable-speed pressure-regulated circulator is used on the load side of this separator. The speed of this circulator automatically changes based on proportional differential pressure control as the zone valves on each air handler open and close. The SEP4 also allows for different flow rates between the lake heat exchanger and load side of the system. The lake heat exchanger allows the lake's cooling potential to be harvested using a completely closed loop system. It eliminates the need for filtering lake water.

Manufacturers of lake water heat exchangers provide design assistance software that can be used to select a specific heat exchanger based on total cooling load, lake water temperature, flow rate through the heat exchanger and temperature change across the heat exchanger.

BUFFER TANKS FOR MULTIPLE CHILLER SYSTEMS

Some chiller manufacturers specify minimum "loop volumes" in dedicated chilled-water distribution systems. The suggested value is typically 3 gallons of water in the chilled-water circuit *per ton of cooling capacity* the loop delivers. Specialized applications that require more precise control of chilled water temperature may require

Figure 5-13

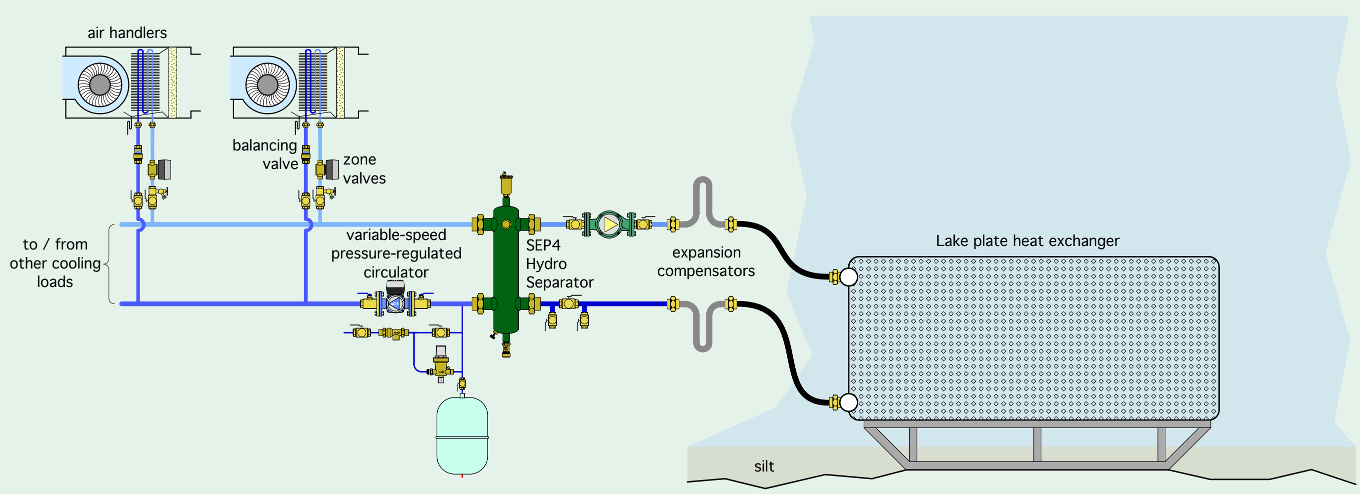


Figure 5-14

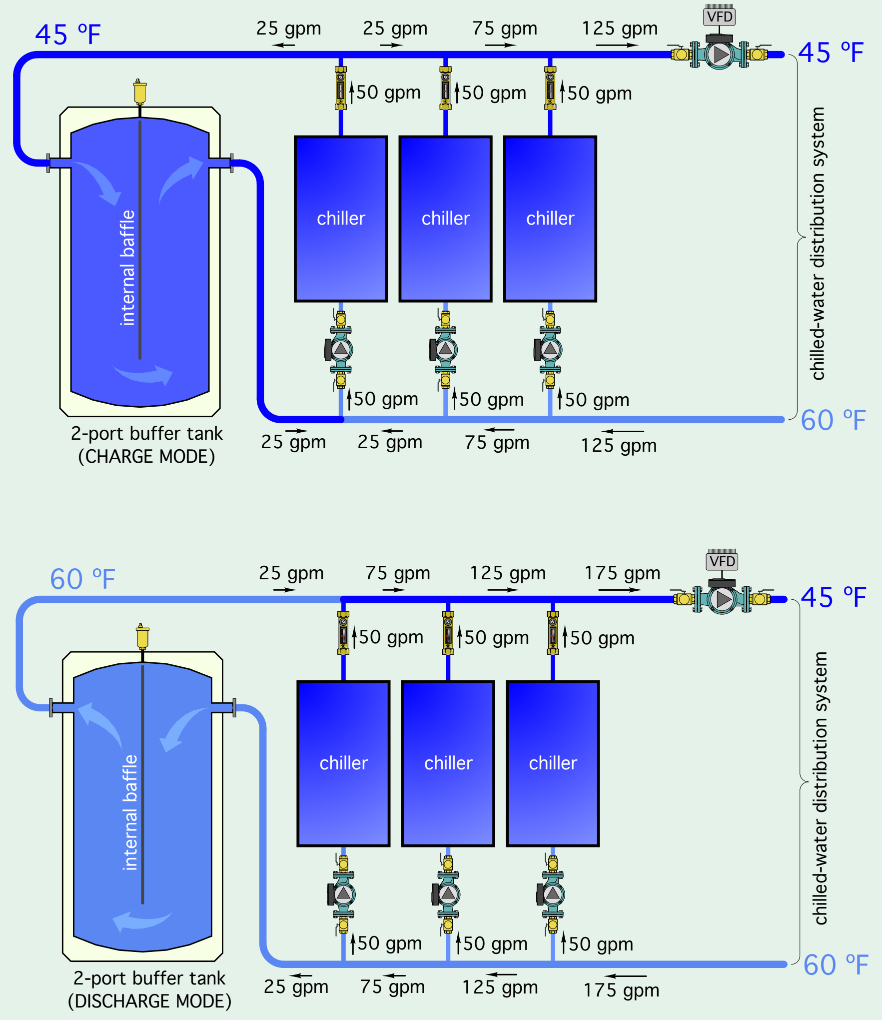


Figure 5-15

Courtesy of cfm Distributors, Inc

from 6 to 10 gallons of water volume per ton of cooling capacity.

If the distribution system does not provide the required volume, a “2-port” baffled buffer tank is often suggested by the chiller manufacturer to bring the circuit volume up to the required value. A typical installation for such a tank is shown in Figure 5-14.

Notice that flow through the 2-port buffer tank changes direction depending on the flow rate of the distribution system versus the flow rate through the chillers. In either case, the buffer tank provides the necessary water volume to stabilize chiller operation. Verify the recommended sizing and placement of 2-port buffer tanks with chiller manufacturers.

Figure 5-15 shows an example of an air-cooled chiller connected to a “2-pipe” buffer tank. The piping to the left of the buffer tank has a DiscalDirt separator for eliminating air and capturing dirt in the hydronic system.

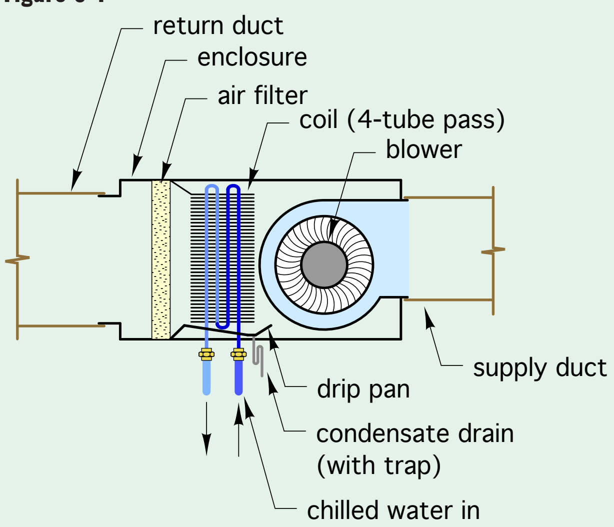
TRADITIONAL CHILLED-WATER TERMINAL UNITS

There are several devices that can absorb heat from a room and transfer that heat to a stream of chilled water. Some have been used for several decades, while others are relatively new. Some are delivered to a project ready to be placed on a floor or suspended from a ceiling and then piped to the chilled-water distribution system. Others are integrated into the construction of the building. This section provides a survey of traditional chilled-water terminal units that are intended to handle both sensible and latent cooling loads. Section 7 will cover radiant cooling panels and chilled beams, which are used in systems that separate the sensible and latent cooling loads.

AIR HANDLERS

One of the most common chilled-water terminal units is an air handler. A basic air handler consists of a chilled-water coil, condensate drip pan, blower, air filter and enclosure, as illustrated in Figure 6-1.

Figure 6-1



Air handlers are available in a wide range of rated cooling capacities, ranging from about 1 ton (e.g., 12,000 Btu/hr) to several hundred tons. Smaller air handlers are typically used in highly zoned systems, or systems where it is impractical to route large ducting. Larger air handlers are more common in systems where large cooling loads are present in a single space or a group of adjacent spaces that can be served by a ducted distribution system.

Smaller air handlers used in commercial chilled-water cooling systems are usually configured with a horizontally oriented cabinet, an example of which is shown in Figure 6-2.

Figure 6-2

Courtesy of SpacePak

Horizontal air handlers are usually suspended within a mechanical chase above a suspended ceiling. The weight of the air handler and its associated ducting and piping is supported by an overhead structure such as a concrete slab, wood framing or steel trusses. Figure 6-3 shows an example where a small chilled-water air handler is suspended on steel channel strut that is supported by threaded rods attached to a structure ceiling. Each steel rod includes a vibration dampener that minimizes vibration transfer to the structure.

The small air handler in Figure 6-3 is contained within the thermal envelope of the building. This protects the water within the coil from freezing. The air handler is also

Figure 6-3

placed above a secondary drip pan. This is a shallow, sealed metal or plastic pan that is equipped with a condensate drainage pipe. Its purpose is to capture any condensate that might collect if the primary drip pan within the air handler were to leak or the drainage pipe from the primary drip pan were to plug. The goal is to provide maximum protection against condensate leakage in areas above finished spaces. Also notice that all piping to the air handler is insulated to prevent condensation.

Figure 6-4

Figure 6-5

Courtesy of cfm Distributors, Inc

Figure 6-4 shows a large air handler designed for a sports arena. The piping leading to the cooling and heating coils within the air handler is insulated with a fibrous material enclosed in PVC jacketing to prevent moisture migration into the insulation.

Large air handlers in the range of 50 to several hundred tons of cooling capacity are usually highly customized products that are configured to the specific requirements of each project. Their modular construction allows for many more options compared to those available in smaller air handlers. Those options include: chilled-water coils, DX refrigerant coils, heating coils, mixing dampers that can blend outside air with return air, variable-speed blowers that can operate independently to control building pressurization, heat recovery wheels, multiple levels of filtration, UV light sterilization hardware, acoustical suppression and humidifiers. The air handler shown in Figure 6-4 has doors allowing service technicians to work inside the unit. The overall functionality of such complex devices is usually coordinated by a building automation system.

Figure 6-5 shows another commercial air handler mounted from the ceiling structure of a cold storage warehouse. This air handler is non-ducted. The photo shows several air filters on the intake side of the unit. Note the insulated supply and return piping that supply twin chilled-water coils within the air handler. The small piping near the bottom of the unit is a condensate drain equipped with a trap.

"2-PIPE" & "4-PIPE" AIR HANDLERS

Many air handlers can be ordered in either "2-pipe" or "4-pipe" configurations. A 2-pipe air handler contains a single coil that could be used for either heating or cooling. In heating mode, heated water flows through the coil. In cooling mode, chilled water flows through the coil. Several 2-pipe air handlers are typically combined on a single distribution system, an example of which is shown in Figure 6-6.

Most 2-pipe distribution systems can operate with either heated water or chilled water. However, all air handlers on the distribution system can only operate in the same mode (e.g., heating or cooling) at a given time.

A "4-pipe" air handler is supplied with two coils: one for heating and the other for cooling. Each coil has a supply and return connection to the appropriate distribution

Figure 6-6

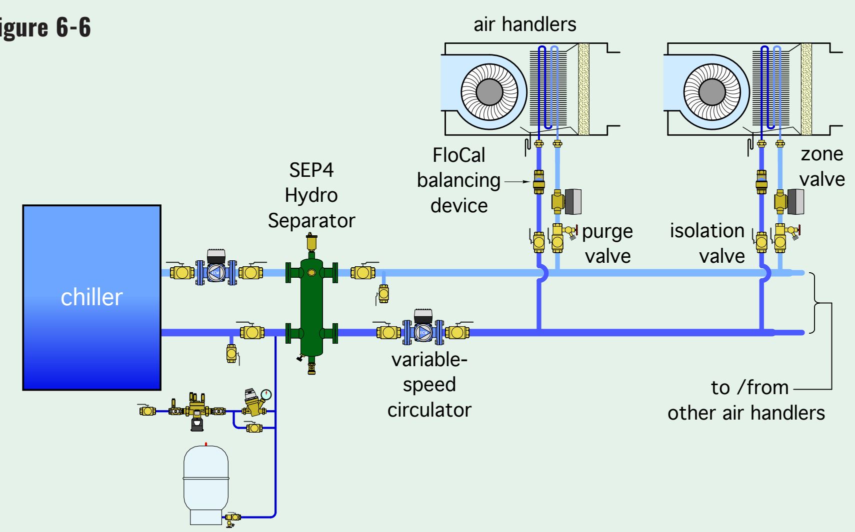
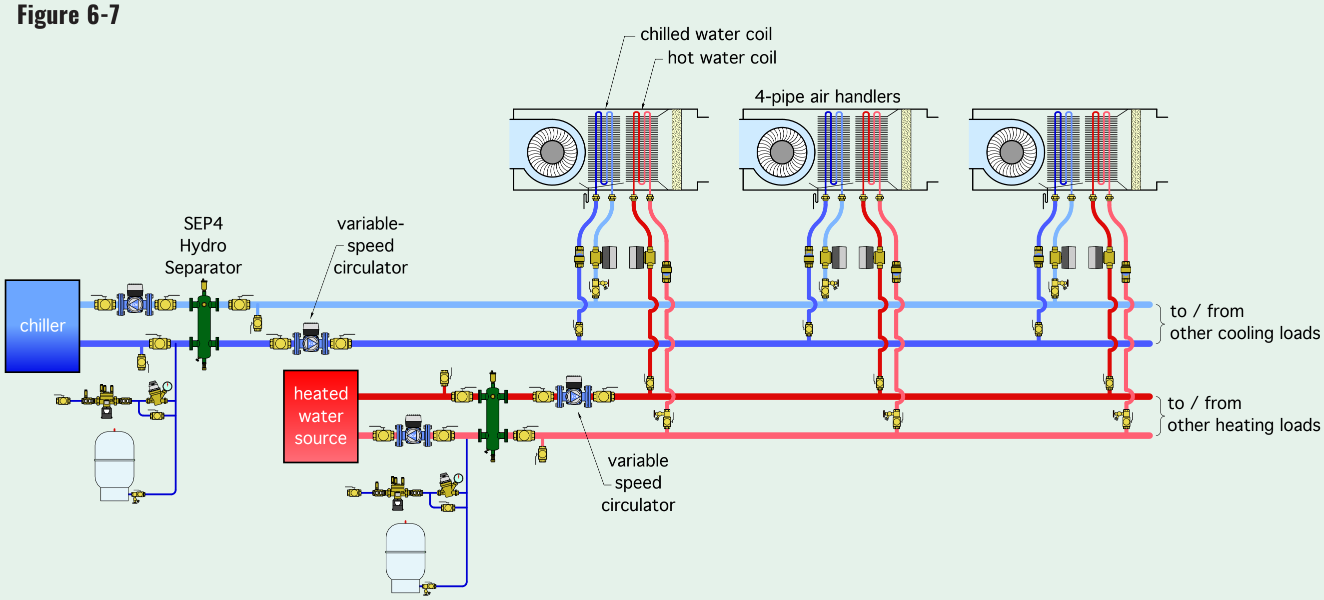


Figure 6-7



mains. A 4-pipe system allows each air handler to operate in either heating or cooling, independent of the other air handlers in the system. Figure 6-7 shows an example of a distribution system using 4-pipe air handlers.

Notice that the heated water and chilled water circuits are completely separated. As such, each should be equipped with make-up water systems, expansion tanks, air/dirt/hydraulic separators and safety devices.

Most air handlers are rated at specific operating conditions for both the incoming chilled water temperature and the incoming air. Those conditions are typically as follows:

- Entering chilled water temperature: 45ºF
- Incoming air conditions
  Dry-bulb temperature = 80ºF, wet-bulb temperature = 67ºF
  Dry-bulb temperature = 75ºF, wet-bulb temperature = 63ºF

The thermal output ratings for chilled-water air handlers are usually expressed as:

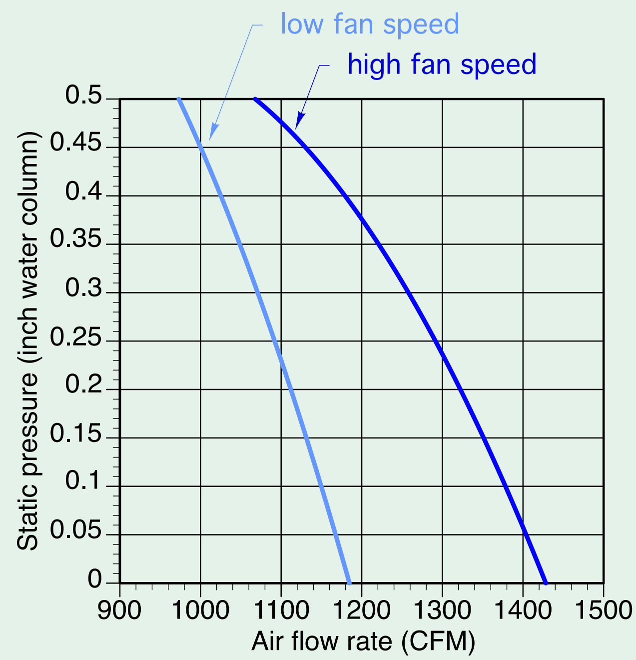
- Total cooling capacity (Btu/hr)
- Sensible cooling capacity (Btu/hr)

Sensible cooling capacity refers to the air handler's ability to lower the temperature of the air as it passes through the unit. Latent cooling capacity is a measure of the air handler's ability to remove moisture from the air stream. Total cooling capacity is the sum of sensible cooling capacity and latent cooling capacity. Latent cooling capacity can be obtained by subtracting sensible cooling capacity from total cooling capacity.

Other performance measures for air handlers include:

- Fluid head loss or pressure drop across the coil as a function of the flow rate through the coil.

Figure 6-8



• Airflow rate produced by the blower as a function of the external static pressure of the ducting system. This is typically stated in cubic feet per minute (CFM), versus the static pressure of the duct system, which is stated in inches of water column (e.g., inch w.g.). If the air handler has a multiple-speed motor, this information is often given for each speed setting.

Figure 6-8 shows an example of airflow rate produced by a nominal 3-ton air handler with a 3-speed motor versus the static pressure developed by its blower. The static pressure at which the air handler operates is determined by the design of the ducting system it supplies.

A typical airflow rate for a chilled-water air handler is 400 CFM per ton of delivered cooling capacity.

FAN COILS:

Another commonly used chilled-water terminal unit is called a fan coil. These units can be mounted into a recessed wall cavity or fastened to a wall surface. An example of two surface-mounted “console” fan coils that can be used for chilled-water cooling, as well as hydronic heating, are shown in Figure 6-9.

Figure 6-9a

Courtesy of Myson

Figure 6-9b

Courtesy of Mestek

Like air handlers, fan coils have an internal blower or fan, a chilled-water coil and a condensate drip pan. Some fan coils also have intake air filters. They are designed to deliver cooling (and in some cases heating) to a single room or space, and are not connected to ducting. Fan coils come in a range of capacities, as well as configured for either a 2-pipe or 4-pipe distribution system, as discussed for air handlers. They often have electrical controls that allow for multiple-speed operation of the blower or fan during heating or cooling, or in some cases just the fan for air circulation. Some also have programmable controls.

Most fan coils have very low internal water volume. This allows them to respond to demands for cooling (or heating) very quickly.

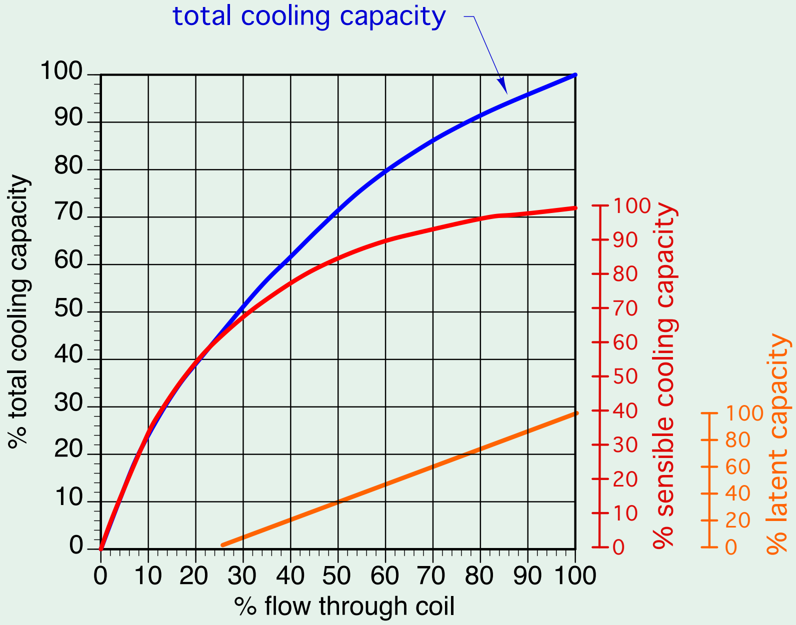
Fan coils equipped with condensate drain pans, and thus suitable for chilled-water cooling, are also available in other mounting configurations. One example is the “high wall” cassette shown in Figure 6-10.

Figure 6-10

High wall cassettes typically have a small tangential blower that draws air in through the front grill and discharges the cooled/dehumidified air from the bottom slot. The angle at which air is discharged into the room can be varied depending on the unit’s mounting height. This adjustment can often be made using a handheld remote control. All wall cassettes used for chilled-water cooling must be equipped with a condensate drip pan and an associated drainage tube. The drainage line can be routed outside or into the building’s sanitary drainage piping, where allowed by code. In the latter case, the drainage pipe must be equipped with a trap to prevent sewer gas from migrating into the unit.

Designers planning to use fan coils or air handlers need to reference specific thermal, hydraulic and airflow data available from manufacturers. Since these terminal units deliver a cooled and dehumidified air stream, cooling

Figure 6-11



Cooling capacity increases as chilled water temperature decreases, especially when the supplied chilled water temperature is well below the dewpoint of air entering the terminal unit.

Cooling capacity is also influenced by flow rate. Figure 6-11 shows a typical relationship.

The non-linear relationship between total cooling capacity and flow rate implies that small changes in flow rate at lower flow rate operating conditions have significantly more effect on cooling capacity than will the same change in flow rate at high flow rates. This characteristic is similar to that of hydronic heat emitters operating at a constant supply water temperature.

If a modulating valve will be used to control the total cooling capacity of an air handler or fan coil by controlling flow rate through it, that valve should use an internal flow control element with an equal percentage characteristic.

capacity is usually listed in a way that allows sensible and latent cooling capacity to be determined based on the chilled water temperature and flow rate supplied, along with typical indoor air and humidity conditions assumed for the rated outputs.

Figure 6-11 also shows that the sensible cooling capacity of the coil begins to rise at lower flow rates, while the coil's latent cooling capacity begins at approximately 25% of total flow rate, and rises in a quasi-linear manner with increasing flow.

The classic approach to absorbing heat and humidity from occupied spaces is to pass air from those spaces through a chilled-water coil in an air handler. That coil provides both sensible and latent cooling. The extent of total cooling capacity and the proportions of sensible versus latent cooling are constrained by the coil selection, as well as the temperature and flow rates of the chilled water stream and airflow through the coil. This approach "couples" the sensible and latent cooling loads and handles both through a common device (e.g., the coil in the air handler). This approach also delivers both sensible and latent cooling using a forced-air stream, typically delivered through ducting.

Another option that has enjoyed increasing interest over the last two decades, is based on "decoupling" the sensible and latent cooling loads. The majority of the sensible cooling load is handled by absorbing heat into radiant panels built into floors or ceilings, or into devices commonly known as "chilled beams." This heat absorption occurs above the dewpoint of the interior air, and thus does not create any surface condensation. The remainder of the cooling load (e.g., small portion of the sensible load and all of the latent load) is handled by a chilled-water coil operating at temperatures well below the dewpoint of the interior air. That coil is typically integrated into an air handler that also provides ventilation to interior spaces.

RADIANT PANELS FOR COOLING

The term radiant panel is generally used in the context of heating. It is defined as any temperature-controlled interior surface that releases 50% or more of its total heat output in the form of infrared radiation.

It's also possible for the temperature of interior surfaces, such as floors and ceilings, to be lowered by circulating cool water through tubing embedded in or fastened to those surfaces. This allows those surfaces to absorb long-wave thermal radiation given off by occupants or other warmer objects in the room. It also allows the surfaces to absorb some of the incident solar radiation entering through windows or skylights.

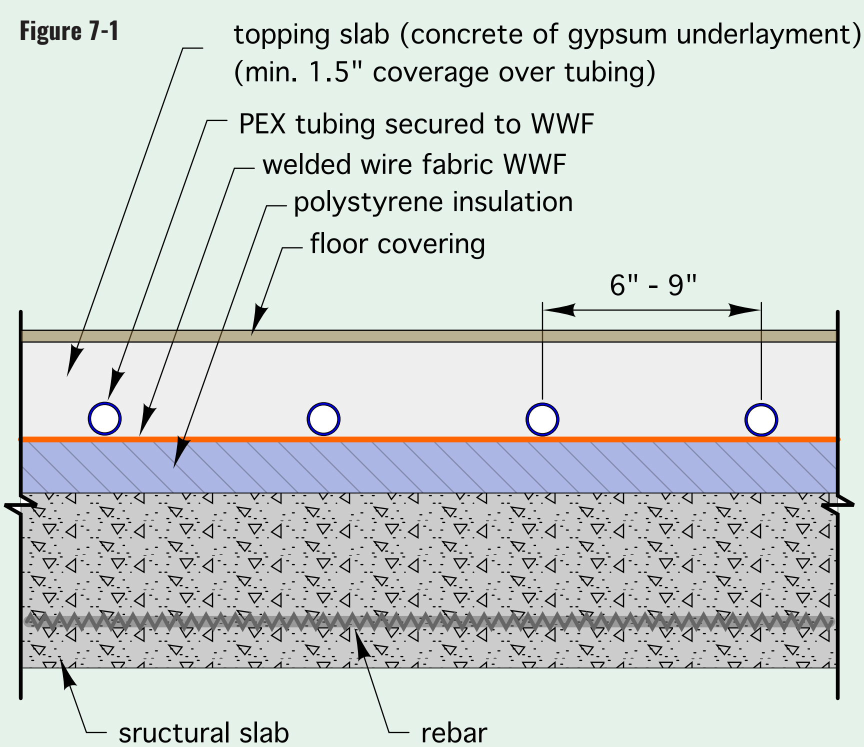
To avoid condensation, it is critically important that cooled ceilings or floors operate at temperatures above the dewpoint temperature of the spaces they serve. . This also applies to components that serve those surfaces, such as piping in a mechanical chase above a cooled ceiling. The chilled water supplied to radiant cooling panels and their associated piping should be a minimum of 3ºF above the current dewpoint temperature of the room. Designers should also consider how the dewpoint temperature varies in different spaces within a building. For example, on a humid day the dewpoint temperature within a frequently used entry vestibule is likely to be higher than that within an interior building space. This is also true for spaces used for food preparation, commercial laundry, washing or exercising. Such areas should be zoned and use independent controls for regulating chilled water supply temperature.

RADIANT FLOOR COOLING

One common radiant panel configuration uses PEX, PERT or PEX-AL-PEX tubing embedded in a concrete floor slab. This approach has been applied in a wide range of commercial and institutional buildings. Due to the thermal mass of the concrete, this approach is often referred to as a "high-mass" radiant panel. This type of panel can be used for heating and sensible cooling.

Many multi-story commercial and institutional buildings have structural

Figure 7-1



concrete slab floors. Figure 7-1 shows an example of how a “topping slab” with embedded tubing can be placed on a layer of rigid polystyrene insulation over a structural concrete floor.

A simplified method for estimating the heat absorption from a cooled floor is based on the difference between the average floor surface temperature and the operative temperature of the room. This is expressed as Formula 7-1.

Formula 7-1:

$$q_c = 1.23\left(T_o - T_f\right)$$

Where:
q_c = cooling absorption flux (Btu/hr/ft^2)
1.23 = combined convective+radiative surface coefficient for cooling (Btu/hr/ft^2/ºF)
T_o = *operative* temperature of the room (ºF)
T_f = *average* floor surface temperature (ºF)

The operative temperature of a room is the *average* between the room’s dry-bulb air temperature and the room’s mean radiant temperature.

Formula 7-2:

$$T_o = \frac{T_{air} + T_{MR}}{2}$$

Where:
T_o = operative temperature of the room (ºF)
T_{air} = dry-bulb temperature of air in the room (ºF)
T_{MR} = mean radiant temperature of surfaces in the room (ºF)

A simplified approach for determining the mean radiant temperature (T_{MR}) of a space is given in Formula 7-3.

Formula 7-3

$$MRT = \frac{\sum_{\text{all surfaces}} T_i A_i}{\sum_{\text{all surfaces}} A_i}$$

Where:
MRT = estimated mean radiant temperature of a space (ºF)
T_i = surface temperature of surface (i) in the room (ºF)
A_i = area of surface (i) in the room (ft^2)

For a typical rectangular room with 6 surfaces (4 walls, ceiling, floor), Formula 7-3 is just shorthand for the following:

$$MRT = \frac{\sum_{\text{all surfaces}} T_i A_i}{\sum_{\text{all surfaces}} A_i} = \frac{T_1A_1 + T_2A_2 + T_3A_3 + T_4A_4 + T_5A_5 + T_6A_6}{A_1 + A_2 + A_3 + A_4 + A_5 + A_6}$$

When the mean radiant temperature of a room with radiant heating or cooling is calculated, that calculation would be _for the unheated or uncooled surfaces_. Thus, for a room with floor cooling, the surface temperature of the floor and the area of the floor would *not* be used in Formula 7-3.

Here’s an example: The mean radiant temperature of a room with floor cooling is determined to be 78ºF. The room’s dry-bulb air temperature is 75ºF. The floor area of the room is 500 ft^2. Estimate the rate of heat absorption into a floor, assuming the average floor surface temperature is 69ºF.

Solution: The room’s operative temperature is found using Formula 7-2:

$$T_o = \frac{T_{air} + T_{MR}}{2} = \frac{75+78}{2} = 76.5^\circ F$$

The cooling flux into the floor is calculated using Formula 7-1:

$$q_c = 1.23\left(T_o - T_f\right) = 1.23\left(76.5 - 69\right) = 9.2\frac{Btu}{hr \cdot ft^2}$$

The rate of heat absorption would be the cooling flux multiplied by the floor area:

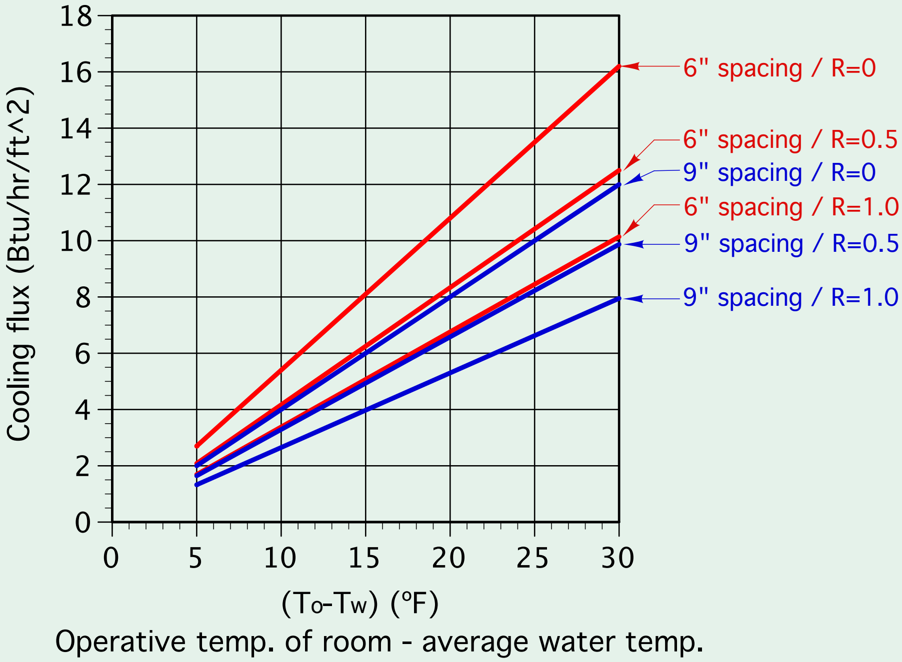
$$Q = \left(9.2\frac{Btu}{hr \cdot ft^2}\right)500\,ft^2 = 4600\frac{Btu}{hr}$$

There is also a relationship between the water temperature in the embedded tubing, the tube spacing and the thermal resistance of any floor covering over the slab. In qualitative terms, the closer the tube spacing, the lower the water temperature, and the lower the thermal resistance of any floor covering, the greater the heat absorption capacity of the cooled slab.

Figure 7-2 shows the relationship between heat absorption flux (in Btu/hr/ft^2) and the temperature difference between the operative temperature of the space and the average water temperature in the embedded tubing circuits. Tube spacings of 6 inches and 9 inches are shown, along with floor covering resistances of 0 (bare or painted slab), 0.5 and 1.0 ºF•hr•ft^2/Btu.

Here’s an example: A room has a floor area of 500 square feet. The entire bare floor slab has embedded

Figure 7-2



The total sensible cooling provided is this cooling flux multiplied by the room area.

$$Q = \left(6.0 \frac{Btu}{hr \cdot ft^2}\right) 500\, ft^2 = 3000 \frac{Btu}{hr}$$

RADIANT CEILING PANELS

Ceilings are ideal surfaces for radiant cooling. One type of "modular" radiant ceiling panel intended for commercial and institutional buildings is shown in Figure 7-3.

This panel is factory-assembled and can also be installed as a free-hanging "cloud" element below a finished ceiling, or within a typical "T-bar" suspended ceiling grid. These panels are available in nominal widths of 24 inches, and nominal lengths ranging from 24 to 120 inches.

tubing spaced at 9 inches apart. Assume that a room has a dry-bulb air temperature of 75ºF, and that the mean radiant temperature of the space, excluding the cooled floor, is 79ºF. The average water temperature in the tubing circuits (e.g., [supply temperature + return temperature]/2) is 62ºF. Determine the total sensible cooling provided by the floor slab.

The operative temperature of the room is the average of the room's dry-bulb air temperature and the room's mean radiant temperature based on all surfaces other than the cooled floor.

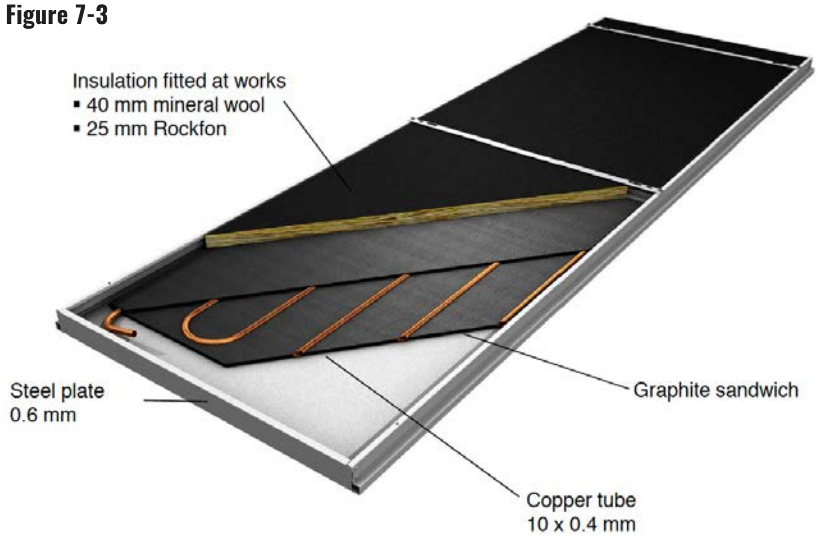
$$T_o = \frac{T_{air} + T_{MR}}{2} = \frac{75 + 79}{2} = 77^o F$$

The difference between the room's operative temperature and the average water temperature in the embedded tubing is:

$$\Delta T = 77^o F - 62^o F = 15^o F$$

Using the graph in Figure 7-2, the cooling flux is found to be 6.0 Btu/hr/ft^2.

Figure 7-3



Courtesy of Zehnder Rittling.

Each panel consists of a painted steel frame pan into which is fitted a copper tube circuit, graphite heat diffusion layers and top-side insulation. The graphite layers provide lateral heat conduction between the tubing and panel areas between the tubing. Although these panels are relatively light, they still require support from light gauge chain or steel cable connected to structural elements such as bar joints or concrete slabs.

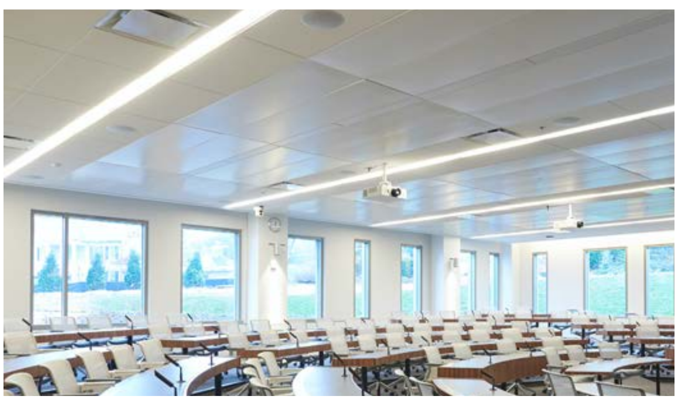
Figure 7-4

Courtesy of Zehnder Rittling.

Figure 7-4 shows modular radiant panels mounted as “clouds” suspended below a finished ceiling. This type of mounting provides slightly higher heat absorption rates due to more surface area exposed to the conditioned space and better convective airflow.

Figure 7-5 shows “flush-mounted” radiant panels in a T-bar suspended ceiling.

Figure 7-5



Courtesy of Zehnder Rittling.

Figure 7-6

Courtesy of Zehnder Rittling.

Figure 7-7

Courtesy of Zehnder Rittling.

After the panels are suspended, they are connected into circuits using small-diameter reinforced hoses with push-on connectors, as seen in Figure 7-6.

Figure 7-7 shows an example of piping above a ceiling using flush-mounted modular radiant panels. The flexible interconnecting piping is corrugated stainless steel tubing (CSST).

The heat absorption rate of modular ceiling panels is also a function of the room’s operative temperature, the average chilled water temperature in the panels, and how the panel is mounted (e.g., flush or “cloud”). The cooling flux for a flush-mounted panel operating with 58ºF entering water temperature, in a room with a 75 ºF dry-bulb temperature, is 30-35 Btu/hr/ft^2. Under the same conditions, a “cloud” mounted panel can absorb 40-50 Btu/hr/ft^2.

CHILLED BEAMS

Although relatively new in North America, chilled beams have been used for cooling in European buildings for more than four decades. *They operate at chilled water temperatures above the room’s dewpoint, and thus can only absorb sensible heat.* They must therefore be

Figure 7-8a

Courtesy of Dadanco div. of Mestek

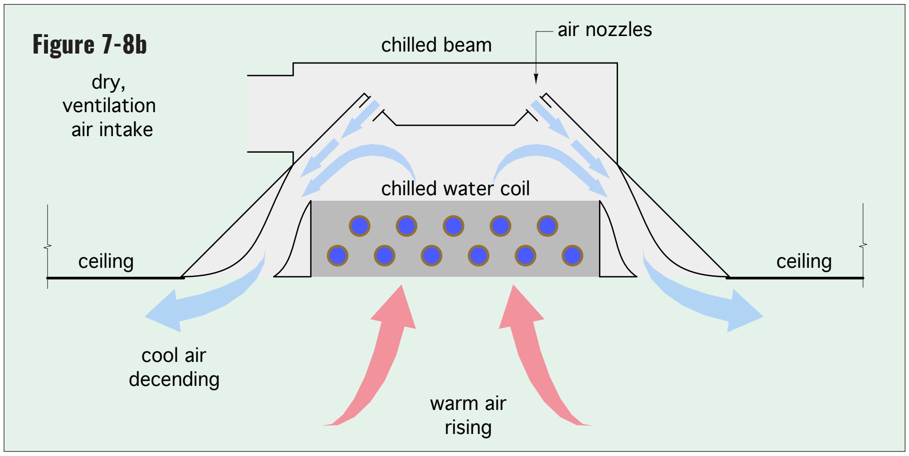


Figure 7-8b

supplemented by an air-handling system that handles latent cooling (moisture removal).

Chilled beams are categorized as “active” or “passive.” Active chilled beams have ventilation air ducted to them. This air has been preconditioned in both temperature and moisture content before it is sent to the chilled beam. This preconditioning allows the air to absorb moisture from the space, and thus manage the latent portion of the cooling load. The preconditioned “dry” air enters the chilled beam and passes through nozzles that increase airflow velocity and decrease local air pressure. The reduced pressure induces airflow through the chilled-water coil, where the temperature of the air is reduced. This reduced temperature air then mixes with the dry ventilation air, and it is reintroduced to the room through the slot diffusers near the outer edges of the chilled beam.

Figure 7-9a

Courtesy of Dadanco div. of Mestek

Passive chilled beams do not have ventilation airflow. They are used to supplement heating or cooling capacity in spaces where sufficient ventilation air is introduced through active chilled beams or other means.

Figure 7-8a shows a typical active chilled beam. Figure 7-8b illustrates the flow of ventilation air into the unit and the induced flow of room air through the coil.

A typical 6-foot-long active chilled beam, when supplied with 58ºF chilled water and 40 CFM of ventilation air, will provide approximately 3,600 Btu/hr of sensible cooling at very low sound levels of about 25 decibels.

Figure 7-9a shows an active chilled beam in an exposed ceiling application. A small duct supplies ventilation air to each chilled beam. The chilled beam and its

Figure 7-9a

Courtesy of Dadanco div. of Mestek

connected piping and ducting are suspended from the steel bar joists in the ceiling. Figure 7-9b shows chilled beams installed flush within a suspended ceiling.

Chilled-beam systems operate in combination with an air-handling system that provides both ventilation air and controls the humidity of the air supplied to the active chilled beams. These air-handling systems vary in how they condition air supplied to the chilled beams. The goal is to reduce the humidity level to a point where the air volume supplied to the chilled beams is no more than 15% higher than the airflow rate required for ventilation.

Active chilled beams significantly reduce the size of ducting required in the building. In dry climates, the ducting is primarily sized for the peak ventilation airflow. In climates with higher humidity, the airflow rate is based on handling the latent load and is usually higher that the airflow required for ventilation. Reduced airflow means that smaller ducting can be used, as can lower power blowers. This can result in operating costs that are up to 50% lower than those of variable air volume (VAV) systems.

DEDICATED OUTDOOR AIR SYSTEMS (DOAS)

Chilled beams, as well as radiant cooling panels, are frequently combined with an air handling/dehumidification system called a DOAS, which stands for Dedicated Outdoor Air System.

DOAS systems use 100% outside air, rather than recirculating a portion of the air in the occupied space. During cooling operation, the objective of a DOAS air handler is to provide sufficient airflow to meet ventilation requirements AND to reduce the absolute humidity of that air to a condition that allows it to absorb the entire latent cooling load of the space. Because the very low-humidity air handles the full latent cooling load, devices such as chilled beams and radiant cooling panels can be operated at temperatures above the dewpoint of the room air and not experience surface condensation.

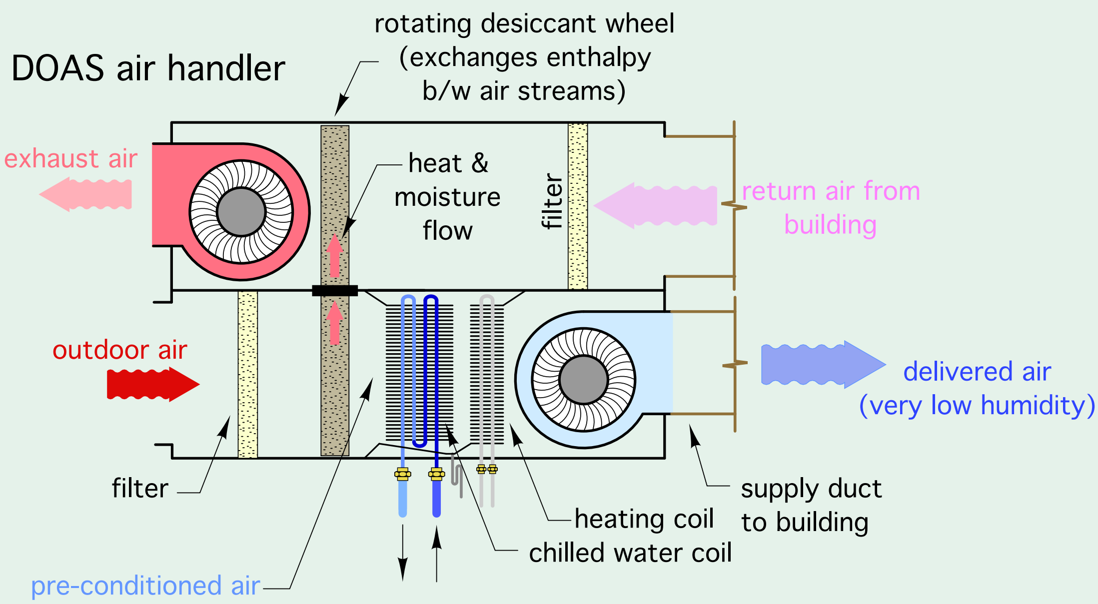
Figure 7-10 shows a typical DOAS air handler.

DOAS air handlers use a multi-row chilled-water coil to extract a high amount of moisture from the ventilation air. The highly dried air is then introduced to the occupied space in a manner that mixes it with the more humid air in the space, such as through an active chilled beam. This mixing allows the moisture being released into the space by occupants, activities or infiltration of humid outside air to be absorbed at a rate that doesn't allow the relative humidity of that space to rise above an established threshold, such as a typical 50% RH.

Return air from the space is routed through an enthalpy wheel within the DOAS unit to recover some sensible and latent cooling effect before being exhausted outside.

During winter, a heating coil in the DOAS air handler can warm incoming ventilation air prior to introducing it into occupied spaced.

Figure 7-10



The combination of chilled beams or radiant panels operating just above room dewpoint temperature, along with a DOAS system for ventilation and latent cooling, is highly efficient from the standpoint of distribution energy. This is largely based on shifting the majority of the sensible cooling load from "air-side" delivery using ducting to "water-side" delivery using a hydronic distribution system. With good design, the electrical power required for the hydronic portion of the system can be approximately 10% of the power required for an equivalent forced-air distribution system.

DOAS systems also allow the distribution ducting to be significantly smaller, and thus less expensive, compared to ducting that must handle the full sensible, latent and ventilation load. Smaller ducting can also reduce the height requirements for mechanical chases above suspended ceilings.

LATENT COOLING LOADS FOR CHILLED BEAMS & RADIANT PANELS

Since chilled beams and radiant panels are limited to sensible cooling, it is necessary to provide latent cooling (e.g., moisture removal from the air) by other means.

The following procedure can be used to determine the operating conditions of the radiant panel or chilled beam, and the DOAS air handler. It assumes a situation in which outside air at design cooling load conditions is drawn into the building to provide the required ventilation airflow.

Step 1: Establish the design values for dry-bulb and wet-bulb air temperature both inside and outside. Typical indoor design conditions are 75ºF and 50% relative humidity (which yields a wet-bulb temperature (e.g., dewpoint temperature) of 55ºF). Typical outdoor design conditions

Figure 7-11

Activity	Male TOTAL heat output (Btu/hr)	Adjusted total heat output (Btu/hr)	Sensible heat output (Btu/hr)	Latent heat output (Btu/hr)
Seated at rest	393	341	205	136
Seat light work	478	410	222	188
Light bench work	870	785	341	444
Heavy work	1604	1604	563	1041
Athletics	1997	1792	631	1161

can be found in Chapter 14 of the 2017 ASHRAE Handbook of Fundamentals, or similar references.

Step 2: Using these design conditions, perform calculations to determine the sensible and latent cooling load of the space.

Step 3: Use a psychrometric chart (provided in Appendix B) to determine the absolute humidity for the indoor air (w_i), as well as the outdoor air at design conditions. Absolute humidity is expressed as either pounds of water per pound of dry air or grains of water per pound of dry air. 7000 grains of water = 1 pound of water.

Step 4: Determine the outside airflow rate required for ventilation based on occupancy. Recommended ventilation airflow rates for different building types and activities can be found in the 2019 ASHRAE HVAC Applications Handbook. These rates can vary from 5 to 20 CFM per person. In general, the higher the metabolic activity level, and the higher the potential for undesirable pollutants to be generated within a space, the higher the recommended ventilation airflow rate per occupant.

Step 5: Determine the increase in absolute humidity (Δw) created by occupants or other processes within the room giving off moisture using Formula 7-2:

Formula 7-2:

$$\Delta w = \frac{q_L}{4842(CFM)}$$

where:
Δw = increase in absolute humidity caused by occupants (lb water/lb dry air)
q_L= latent load due to occupancy (Btu/hr)*
CFM = required airflow rate of outside air (ft^3/minute)

*A typical adult engaged in very light activity creates a latent load of about 188 Btu/hr. The latent load for other activity levels can be found in Figure 7-11.

Step 6: Determine the increased absolute humidity when the room is occupied by adding the absolute humidity at the design indoor air conditions (i.e., 75ºF and 50% RH) to the increase in humidity caused by occupants:

Formula 7-3:

$$w_{max} = w_{di} + \left(\Delta w\right)_{occupants}$$

where:
w_{max} = absolute humidity of space based on design load conditions plus moisture added by occupants and activities (lb water/lb dry air)
$(\Delta w)_{occupants}$ = increase in absolute humidity caused by occupants or other moisture-generating processes (lb water/lb dry air)

w_{di} = absolute humidity of indoor air at design conditions (lb water/lb dry air)

Step 7: Use a psychrometric chart (provided in Appendix B) to determine the elevated dewpoint temperature corresponding to this maximum absolute humidity. The minimum chilled water temperature supplied to the chilled beams or radiant panels should be 3ºF above this elevated dewpoint temperature.

Step 8: Select a dry-bulb temperature at which the air handler will deliver cooled/dehumidified air to the space (T_{dbc}). Typical values range from 46ºF to 60ºF.

Step 9: Use Formula 7-4 to determine the portion of the sensible cooling load that is carried by the air handler.

Formula 7-4:

$$q_{sah} = 1.08(CFM)(T_{dbo} - T_{dbi})$$

where:
q_{sah} = sensible load carried by the air handler (Btu/hr)
CFM = airflow rate through air handler
T_{dbi} = dry-bulb temperature of air delivered to space (ºF)
T_{dbo} = dry-bulb temperature of outdoor air at design conditions (ºF)

Step 10: The sensible cooling load that remains for the radiant panels is the total sensible cooling load minus the sensible load carried by the air handler.

Formula 7-5:

$$q_{srad} = q_{stotal} - q_{sah}$$

Step 11: When radiant panel cooling is used, the sensible heat flux required of the panel is the total sensible load carried by the radiant panel divided by the area of the panel.

Step 12: Determine the required chilled water supply temperature to the chilled beam or radiant panel to provide the heat absorption rate calculated in step 11. Refer to the manufacturer's ratings of panel heat absorption per unit of area versus chilled water supply temperature.

Step 13: Verify that the required chilled water supply temperature is at least 3ºF above the dewpoint temperature of the room when it is fully occupied. If it is not, consider shifting more of the sensible cooling load to the air handler to reduce the required load on the chilled beams or radiant panel to a condition where it can safely operate above the maximum (occupied) dewpoint temperature of the room.

Step 14: The latent load that must be carried by the coil in the air handler is based on the change in air conditions across the coil and can be calculated using Formula 7-6.

Formula 7-6:

$$q_L = 4842(CFM)(\Delta w)$$

where:
q_L= latent load due to occupancy (Btu/hr)
Δw = increase in absolute humidity caused by occupants or other moisture-generating processes (lb water/lb dry air)
CFM = required airflow rate of outside air (ft^3/minute)

The design goal in selecting the air handler is to produce relatively dry air (e.g., with a low dewpoint temperature) at a flow rate that satisfies the ventilation requirement of the space. When introduced into the space for ventilation, this dry air will absorb moisture at a rate that satisfies the latent cooling load and maintains a reasonable relative humidity within the space.

PIPING FOR DECOUPLED SENSIBLE & LATENT LOADS

Systems using chilled beams or radiant panels require piping and controls that can maintain the chilled water temperature to the chilled beams or radiant panels a minimum of 3ºF above the dewpoint temperature of the room. The 3ºF margin allows the chilled beams or radiant panels to absorb as much of the sensible cooling load as possible, while still providing a slight "safety factor" against formation of surface condensation due to sensor and controller accuracy tolerances.

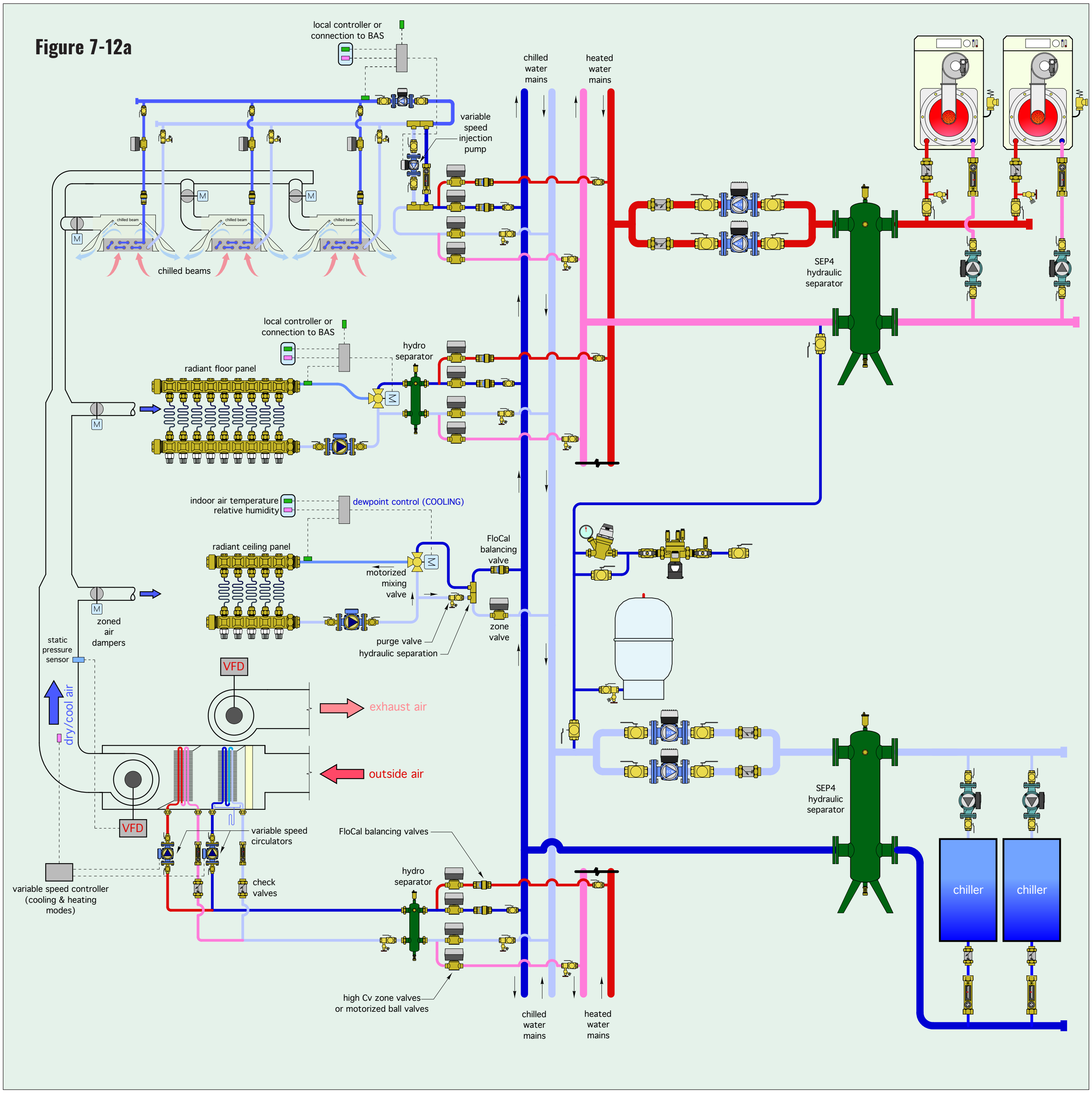
These systems also require control of chilled water through the coil of a DOAS air handler so that ventilation air can be conditioned to a low dewpoint temperature, allowing it to absorb the full latent load of the space, as well as a portion of the sensible cooling load.

Figure 7-12a shows a concept system that uses a combination of chilled beams and radiant panels to supply the majority of sensible cooling, along with a DOAS air handler for the remainder of the sensible load and all of the latent load.

Figure 7-12b shows a closer view of the three chilled beams in Figure 7-12a.

The system in Figure 7-12a illustrates several possibilities. Fundamentally, it's a "4-pipe" distribution system capable

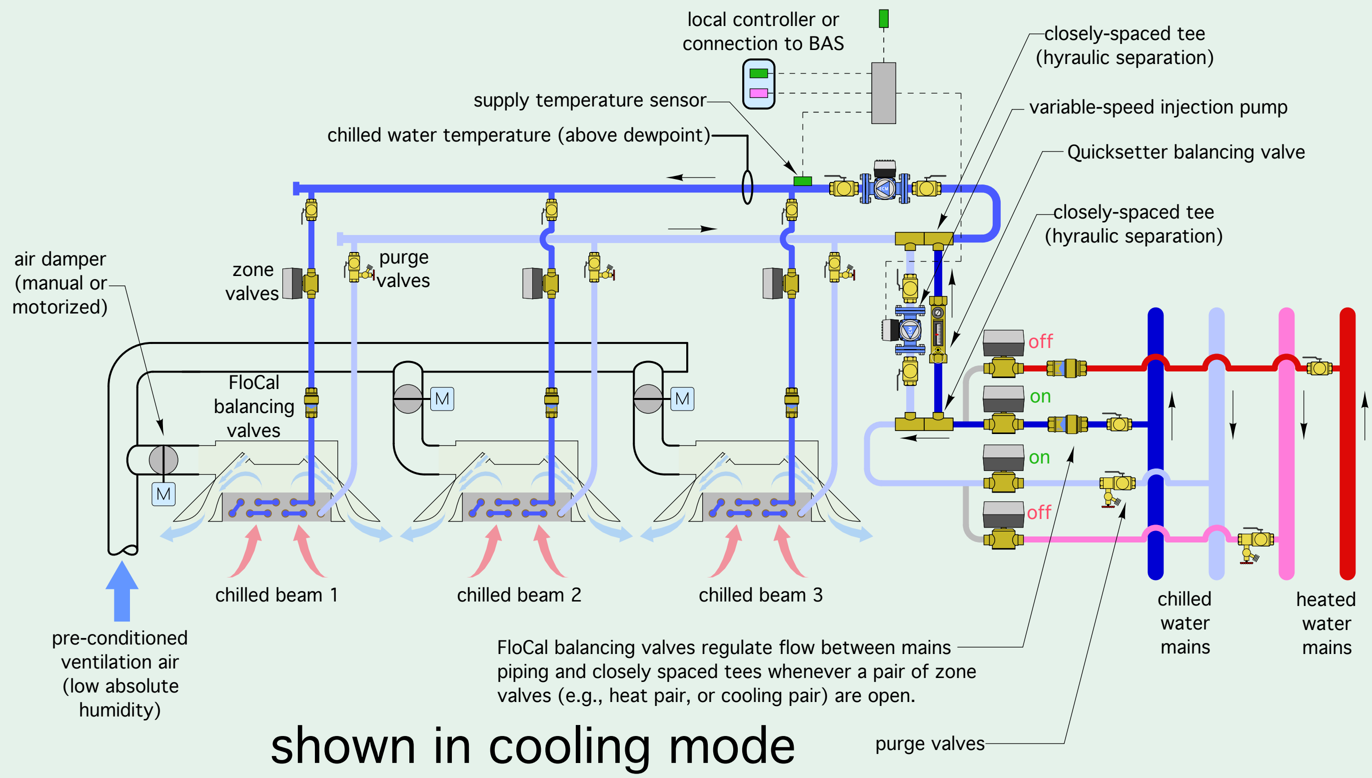
Figure 7-12a



of simultaneous heating and cooling. It also shows a mix of radiant panels and chilled beams, either of which can operate in heating or cooling mode.

An array of three chilled beams operating in cooling mode is shown at the top of the schematic and in more detail in Figure 7-12b. These are active chilled beams that provide sensible cooling and also introduce ventilation air into their associated spaces. Water flow to each chilled beam is controlled by a zone valve. FloCal balancing valves are used to automatically regulate the flow rate through each chilled beam when it is operating. Flow to the chilled beams is provided by a variable-speed pressure-regulated circulator operating in proportional differential pressure mode. The speed of this circulator automatically changes as the zone valves at each chilled beam open and close. The water *temperature* supplied to the chilled beams is regulated by another variable-speed "injection" pump.

Figure 7-12b



The speed of this injection pump is regulated to maintain a set chilled water supply temperature to the chilled beams that is slightly above the current dewpoint temperature of the space. Flow from the chilled-water mains is allowed to pass through a pair of closely spaced tees when any of the chilled beams are active. This flow rate is regulated by FloCal balancing valves. The closely-spaced tees provide hydraulic separation between the pressure across the chilled-water mains and the differential pressure of the injection circulator. Another set of closely-spaced tees separate the injection pump from the differential pressure of the variable-speed circulator supplying the chilled beams. The injection pump can also be controlled to regulate the supply water temperature to the chilled beams during heating mode. That temperature could be a setpoint or based on outdoor reset control.

Ventilation air that has been conditioned to a low dewpoint temperature is supplied from a DOAS air handler to each chilled beam and regulated by a motorized damper. The speed of the intake blower is regulated based on maintaining a set static pressure in the ducting system. The speed of the exhaust blower is regulated based on maintaining a set pressure differential between inside and outside air. Although not shown in Figure 7-12a, an enthalpy wheel could be added to the DOAS air handler to enable energy recovery from the building exhaust air stream.

The other cooling loads on the system are supplied by radiant panels. The upper radiant panel manifold can supply heating or cooling. The smaller radiant panel manifold only supplies cooling. The upper radiant panel manifold is separated from the heating and cooling mains by a hydraulic separator. A 3-way motorized mixing valve is used to keep the chilled water temperature to the radiant panel circuits slightly above dewpoint. This same mixing valve can be operated in heating mode based on outdoor reset control.

The DOAS air handler has a “deep” chilled-water coil capable of lowering the dewpoint of the ventilation air to a very dry condition. Flow through this coil is regulated by a variable-speed circulator operated by a controller that monitors the dewpoint of the air leaving the air handler. Flow through the heating coils is also regulated by a variable-speed circulator. Hydraulic separation of these injection circulators from the mains is provided by a hydraulic separator.

The chilled-water plant consists of two chillers, each with its own circulator. The chiller output is linked to the mains using a hydraulic separator. A duplex pair of variable-speed circulators creates flow in the chilled-water mains. The speed of the active circulator automatically varies as the zone valves that lead to the loads open and close.

Heated water is provided by a pair of modulating/condensing boilers. These boilers are also linked to the heated water mains through a hydraulic separator. . A duplex pair of variable-speed circulators creates flow in the heated water mains. The speed of the active circulator automatically varies as the zone valves that lead to the heating loads open and close.

There are many possible variations of this concept drawing. The key underlying concepts are:

• Separation of the sensible and latent cooling loads.

• Use of water-based heating and cooling conveyance to reduce air-side delivery to the ventilation requirements and a small portion of the sensible cooling load.

• Using chilled beams and/or radiant panels to absorb the majority of the sensible cooling load.

• Using a DOAS air handler with variable-speed blowers to supply chilled ventilation air at a low dewpoint temperature to absorb the latent cooling load. The same air handler can also heat incoming ventilation air during winter.

• Use of variable-speed pressure-regulated circulators with on/off flow control through branch circuits to reduce electrical energy consumption.

• Use of automatic flow-balancing valves to control flow rates in key areas of the system.

• Using a staged approach to chilled water production and heated water production.

• Using central hydraulic separators to isolate the pressure dynamics of the chiller and boiler plants from those of the distribution system.

• Using sets of closely-spaced tees or hydraulic separators to isolate the pressure dynamics established by the main distribution circulators from those established by the branch circulators.

• The option to use either 3-way motorized valves or variable-speed injection pumps to regulate heat transfer rates at coils.

WATER LOOP HEAT PUMP SYSTEMS

Many commercial and industrial buildings have heating and cooling load characteristics that are very different from smaller buildings. One of those characteristics is a frequent need for simultaneous heating and cooling in different parts of the building. On many days during fall and spring, the perimeter areas of commercial buildings may require heating, while the building's core areas (e.g., those without exposed surfaces) require cooling, as depicted in Figure 8-1.

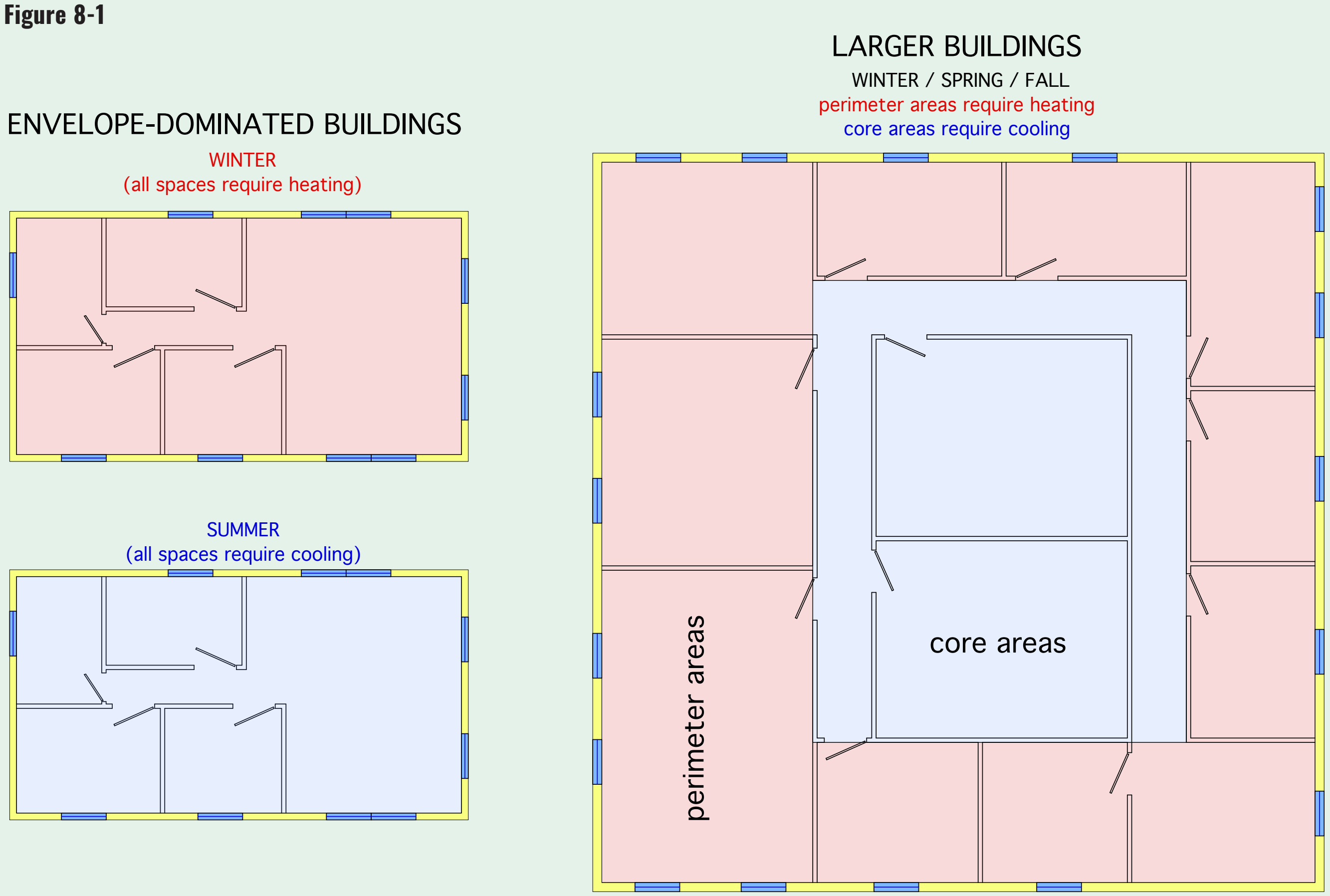
Although it's possible to operate two separate systems — one for heating and the other for cooling — and use devices such as zoning dampers or zone valves to direct the heating or cooling where needed, this is not the most efficient approach. The heating system would be burning fuel or consuming electricity to *create* heat for the perimeter areas. The cooling system would be consuming electricity to absorb heat (e.g., *create* cooling effect). Neither system is attempting to *move* heat from where it's present in access to where it's needed in the building.

Buildings with simultaneous heating and cooling loads are well-suited to a water loop heat pump system, which can *move* heat from core areas to perimeter areas.

Water loop heat pump systems consist of several water-to-air heat pumps. Each heat pump has a refrigerant-reversing valve, allowing it to operate in either heating or cooling mode. All the heat pumps are connected to a common hydronic distribution system. Heat pumps operating in heating mode extract heat from the water circulating in the hydronic distribution system. Heat pumps operating in cooling mode dissipate heat into this hydronic distribution system. This allows excess heat in the building's core areas to be moved to where it's needed in the perimeter areas. Heat that is *moved* is heat that doesn't have to be *created*.

Figure 8-2 shows an example of a typical water-to-air heat pump that would be used in this type of system. Heat pumps with horizontal orientation are typically installed above suspended ceilings in commercial buildings. Those with vertical orientation are usually installed in mechanical

Figure 8-1

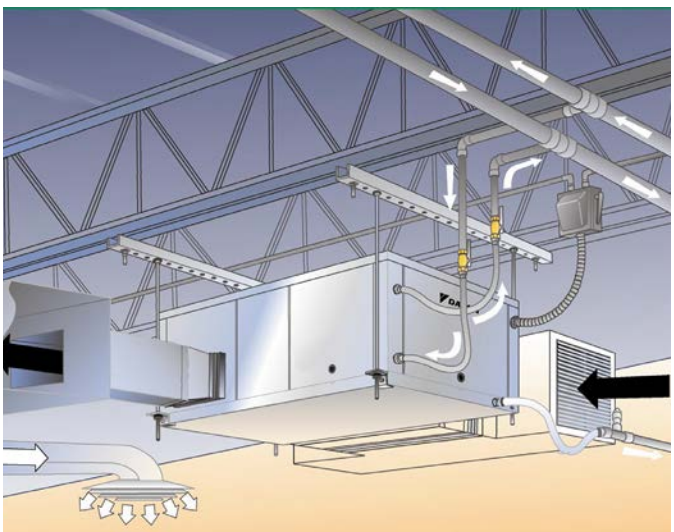


rooms or mechanical closets. These heat pumps are available in nominal cooling capacities ranging from 0.5 to 5.0 tons (6,000 to 60,000 Btu/hr).

Figure 8-2a

Courtesy of McQuay div. of DaikinApplied

Figure 8-2b



Courtesy of McQuay div. of DaikinApplied

An ideal situation for a water loop heat pump would allow all the heat collected from areas of the building requiring cooling to be moved to areas requiring heating. Under such conditions, and depending on the water temperature in the common water "loop" piping, the building would be heated and cooled at a fraction of what it would otherwise cost using an electric resistance heating system and a separate electrically powered cooling system.

This ideal situation would be when the building's total heating requirement is equal to the building's total cooling requirement plus the total electrical energy to run all the heat pumps, as expressed in Formula 8-1. Under these conditions, the water temperature in the piping loop would remain constant.

Formula 8-1:

$$Q_{heating} = Q_{cooling} + P_{electrical}$$

Where:
$Q_{heating}$ = total heating requirement of the building (Btu/hr)
$Q_{cooling}$ = total concurrent cooling load of building (Btu/hr)
$P_{electrical}$ = total electrical power supplied to all operating heat pumps (Btu/hr) (kw x 3413)

Although these balanced load conditions are possible, they will not occur during winter or summer. During winter, it's possible, even likely, that the building's heating load will be significantly higher than its cooling load. Likewise, during summer, the entire building is likely to require cooling.

These situations create the need for supplemental heat input, and the ability to dissipate the heat of rejection from all the heat pumps running simultaneously in cooling mode.

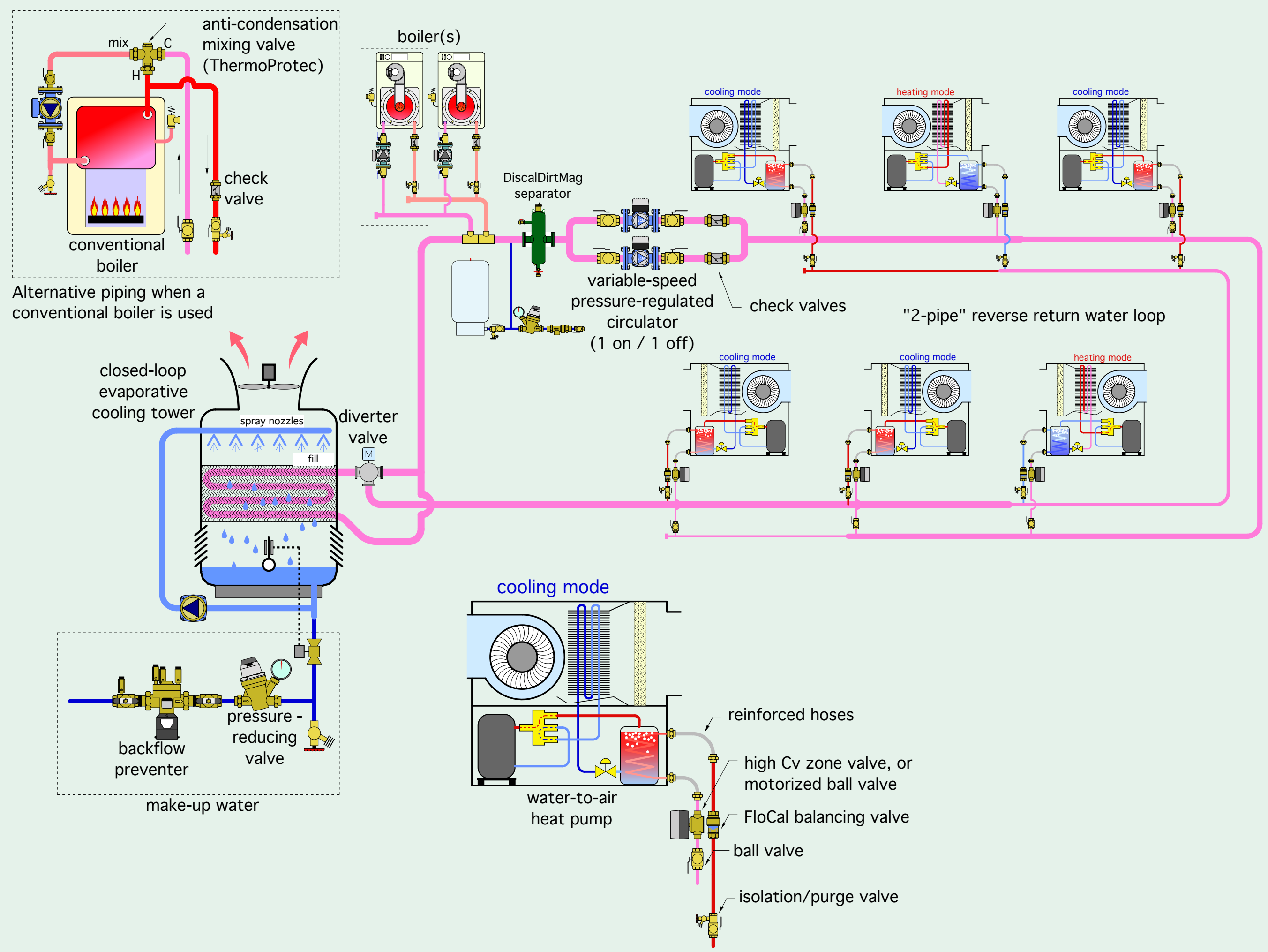
The most common way to add supplemental heat to the water loop is using one or more boilers. A typical way to dissipate heat that's in excess of the building's needs is by including a cooling tower in the system. These supplemental heating and cooling devices, along with several heat pumps, are shown connected to a common "2-pipe" reverse return piping loop in Figure 8-3.

A typical control criterion for a water loop heat pump system is to allow the loop temperature to "float" between 70ºF and 90ºF whenever possible, based on the rates of heat input and extraction from all operating heat pumps. The thermal mass of the water in the loop provides some buffering effect. If the heat input to the loop forces the water temperature above 90ºF, flow is directed through the cooling tower. If the loop temperature drops below 70ºF on a cold winter day, one or more boilers are operated. The working temperature of the loop makes it an ideal application for modulating/condensing gas-fired boilers. *If conventional boilers are used in this type of system, they should be installed with anti-condensation mixing valves, as shown in Figure 8-3.*

The operating temperature range of 70ºF to 90ºF is close enough to typical conditioned space temperatures that the water loop is usually not insulated and not subject to condensation.

Each heat pump branch contains a zone valve with a high Cv rating, or a motorized ball valve, that opens when the heat pump is operating. Each branch also contains a FloCal balancing valve that automatically maintains a

Figure 8-3



set flow rate through the heat pump when it operates. Most heat pump manufacturers specify flow rates of 2 to 3 gallons per minute *per ton* of heat pump heating capacity. Reinforced hoses are used to reduce vibration transfer between the heat pump and rigid piping. Each heat pump branch is also equipped with isolation and purging valves. These allow any heat pump to be isolated from the distribution loop and removed for servicing if necessary, without affecting operation of the other heat pumps.

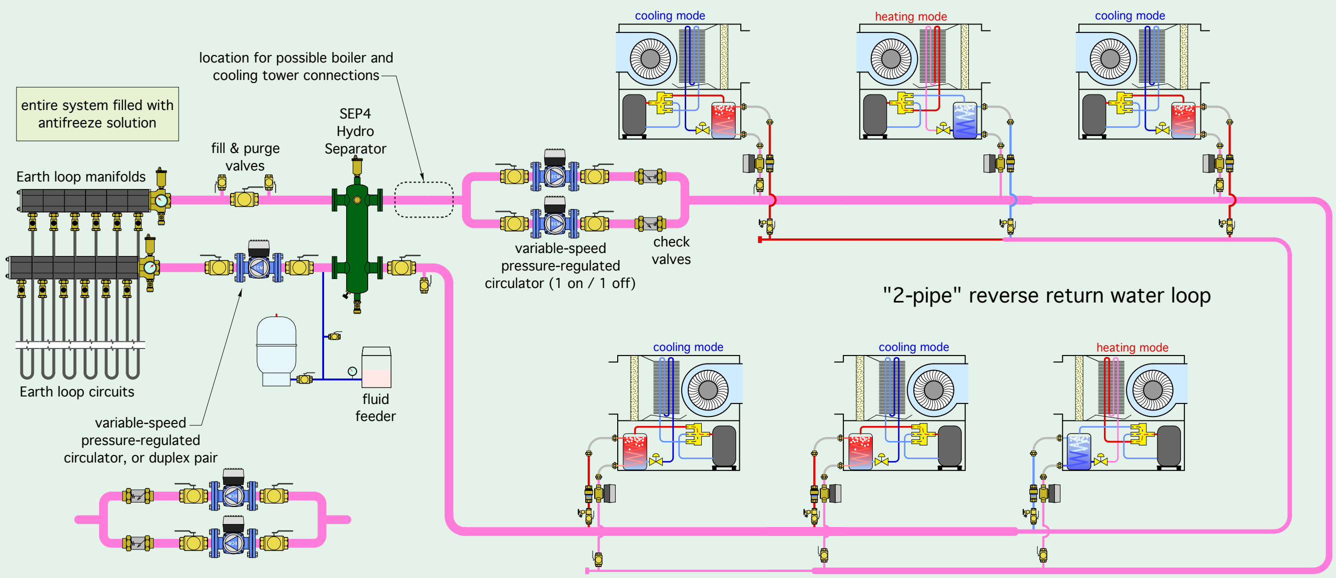
Flow through the building loop is typically maintained 24/7. It's provided by a duplex set of variable-speed pressure-regulated circulators that automatically adjust flow rate based on proportional differential pressure control. As more heat pumps turn on, flow in the loop is automatically increased, and vice versa. Only one of the two circulators operates at a time. The other serves as a backup. A controller automatically determines which circulator is operating and attempts to creates approximately equal elapsed run time for each circulator.

A DiscalDirtMag separator provides high-efficiency air, dirt and magnetic particle separation for the system.

Some water loop heat pump systems use closed-loop cooling towers. The same fluid that passes through the water loop also passes through closed piping paths within the cooling tower. There is no contact between this water and the sump water used to enhance the tower's evaporative heat dissipation. This is the type of cooling tower shown in Figure 8-3.

If the building loop system operates with all water, the cooling tower must be protected from freezing in winter. In some systems, the tower is isolated from the

Figure 8-4



balance of the system and drained during winter. In other systems, a small flow of water from the loop is maintained through the tower to keep it from freezing. The modulating 3-way motorized valve shown in Figure 8-3 can provide this flow. The latter approach allows the tower to come online quickly during an abnormally warm day in winter.

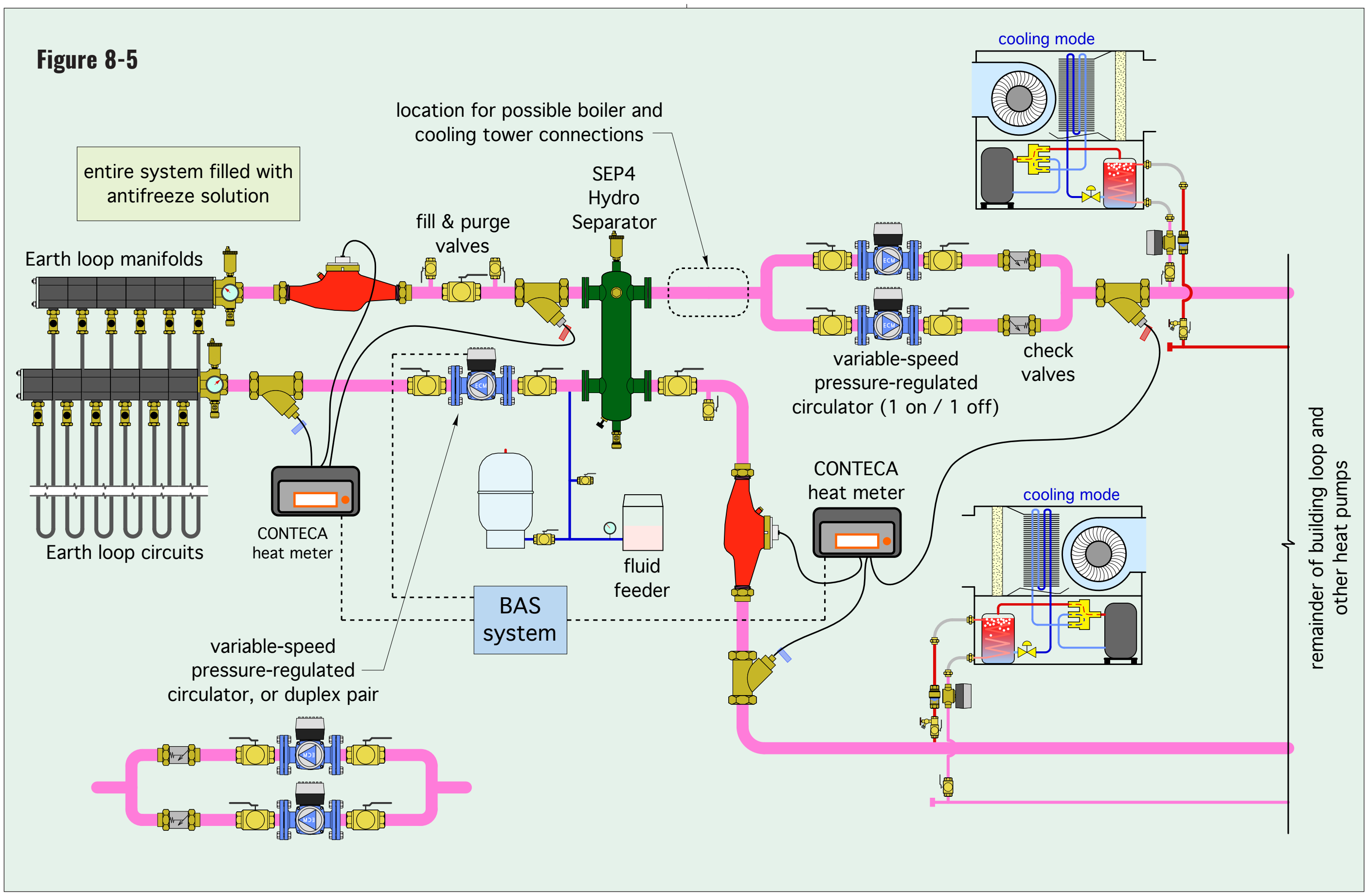
Other systems use open cooling towers. In these systems, the tower is typically separated from the closed/pressurized building water loop by a plate & frame heat exchanger. That heat exchanger should be sized for a small approach temperature difference in the range of 2 to 4ºF. This helps keep the loop temperature down during cooling mode operation on hot summer days, and thus minimizes the drop in heat pump cooling performance.

In most systems, the cooling tower is located outside the building. However, it is possible to use a cooling tower mounted inside the building. Outside air enters the tower through a large louvered panel in an exterior wall. After passing through the tower, this air is discharged back outside through another larger louvered panel. When this type of tower is used, the discharge air should be directed away from nearby windows, doors, parked vehicles or areas of outdoor gatherings. This prevents water droplets in the discharge air, or residual water treatment chemicals in the sump water, from being discharged where they could cause adverse effects.

GEOTHERMAL WATER LOOP HEAT PUMP SYSTEMS

Water loop heat pump systems can also be adapted to geothermal earth loops. In some systems, the earth loop is sized to eliminate the need for a cooling tower and auxiliary boiler. In other systems, the earth loop may be sized to minimize the size of the cooling tower and boiler. Figure 8-4 shows an example of a system where the boiler and cooling tower have been eliminated.

The earth loop portion of this system is interfaced to the building loop using a SEP4 hydraulic separator. This component provides high-efficiency air, dirt and magnetic particle separation. It also provides hydraulic separation between the earth loop circulator and the building loop circulator. The hydraulic separation also allows the potential for the speed of the earth loop circulator to be regulated independently of the building loop circulator. For example, the speed of the earth loop circulator could be ramped up and down depending on the thermal load of the building. If there is a near balance between the heating and cooling loads in the building, the earth loop circulator could be operated at a very low speed since there is very little need for additional heat input to, or heat dissipation from the building loop.



The earth loop circulator speed could be regulated based on the rate of heat transfer supplied from the earth loop compared to the measured load in the building loop. These two rates of heat transfer can be monitored by CONTECA heat meters and transferred to a building automation system that in turn regulates the speed of the earth loop circulator. The piping for this concept is shown in Figure 8-5.

Aside from the source of supplemental heat supply and dissipation, there are important differences between a water loop heat pump system supplied by an earth loop versus one configured around a cooling tower and supplemental boiler.

One is the water temperature in the building loop. Whereas a “boiler/tower” heat pump system typically maintains the loop temperature between 70 and 90ºF, a building loop supplied from a geothermal system will operate over a much wider temperature range. Depending on the earth loop design, the site location and the cooling load, the summer loop temperatures can be comparable to those achieved using a properly sized cooling tower. However, during winter, the earth loop temperatures could drop much lower than 70ºF, perhaps as low as 30ºF.

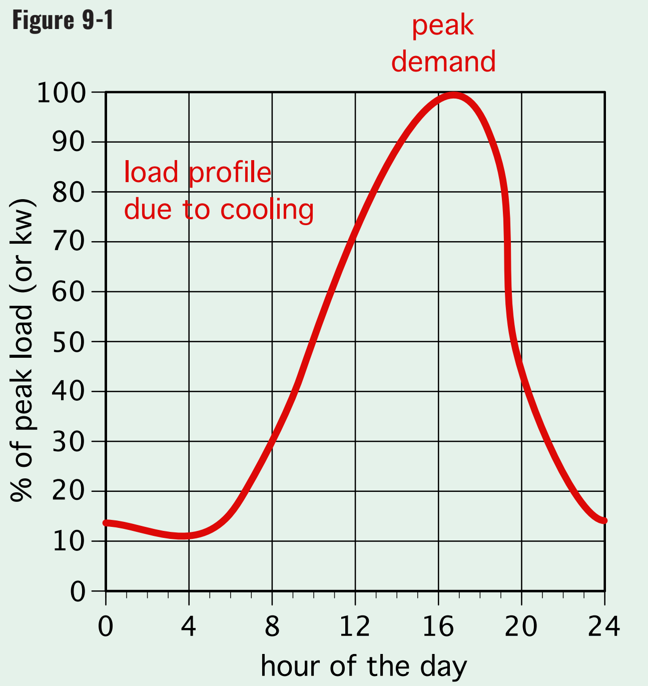
Modern water-to-air heat pumps intended for geothermal applications can operate across these wide temperature ranges. Still, if the minimum loop temperature is expected to approach freezing, the system should be operated with an antifreeze solution. *All piping and piping components must also be insulated to prevent surface condensation when operating at fluid temperatures below the dewpoint of interior air*.

Another possibility is to connect one or more *water-to-water* heat pumps to the building loop. These heat pumps could be used to heat domestic hot water, supply radiant panels in some areas of a building, or supply heated or chilled water for a process load.

When water-source heat pumps are used for heating and cooling, the ventilation air is typically provided by a dedicated outdoor air system (DOAS). . During warm months, the DOAS cools and dehumidifies entering outside air. This air can be introduced directly into spaces by a separate duct system, or it can be mixed into the supply air stream leaving each heat pump. During winter, the DOAS would pre-heat the incoming ventilation air.

As the use of electrical-powered building cooling continues to expand, electric utilities face a challenge: how to supply the necessary electrical energy in ways that do not create increasingly sharp peaks in their power demand profile. For many utilities, especially those in warmer locations, peak power demand occurs in late afternoon on hot and humid days, as represented in Figure 9-1.

Figure 9-1



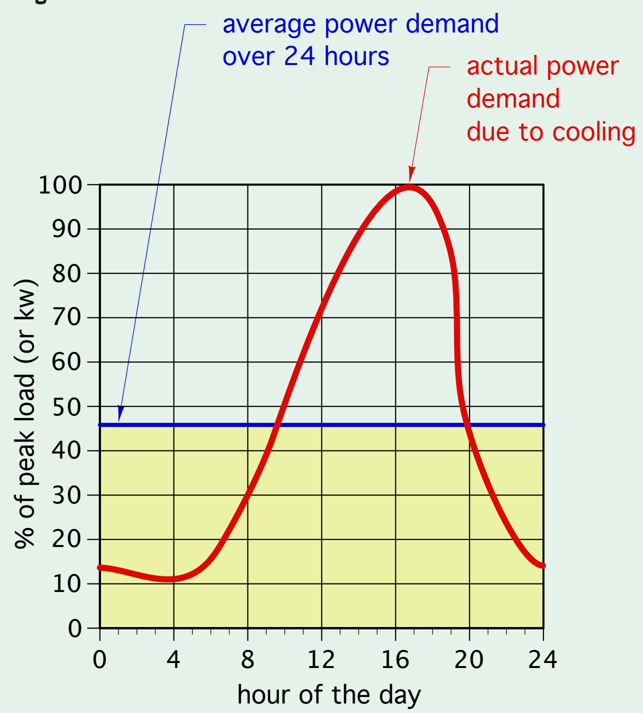
These peaks in demand necessitate preemptive operation of peak power-generating facilities, which often cost significantly more to operate per kilowatt•hour of delivered energy compared to base load generating facilities. Peaks that are largely created by motors also increase the reactive power loading on distribution grids, increasing transmission line losses.

The area under the load profile curve in Figure 9-1 represents the total electrical energy (not power) supplied to the load over the 24 hour period. Figure 9-2 shows the average power that — if maintained for 24 hours — would result in the same total electrical energy supply.

For this specific load profile, the average power is only about 46% of the peak power.

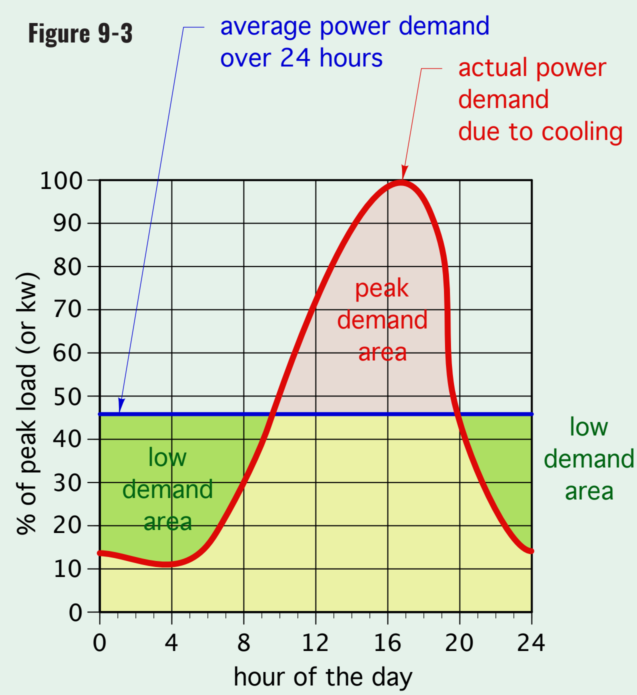
An "ideal" scenario for an electric utility would be to provide the same total amount of electrical energy over the 24-hour period, but for the load profile shown in Figure 9-2, supply it at a constant power of 46% of the peak. This scenario is represented by the yellow area in Figure 9-2.

Figure 9-2



One way to conceptually achieve this "ideal" scenario would be to "clip off" the high-demand area shown in red in Figure 9-3 and use this area to "fill in" the low-demand areas shown in green in Figure 9-3.

Figure 9-3



One method that has been used to reduce peaks and shift more electrical power demand to periods of low demand is called "demand-side management." In essence, electric utilities encourage customers to install equipment that can operate during low-demand periods and store energy until it's needed. Financial incentives, such as time-of-use electrical rates and demand charges, is one of the key methods used for demand-side management. Time-of-use electrical rates offer electrical energy at significantly lower cost per kilowatt•hour when it is used during pre-defined periods of low demand. Those periods include nighttime hours, and in some cases weekends and holidays. Each utility offering these rates will have specific rate structures and times at which they apply.

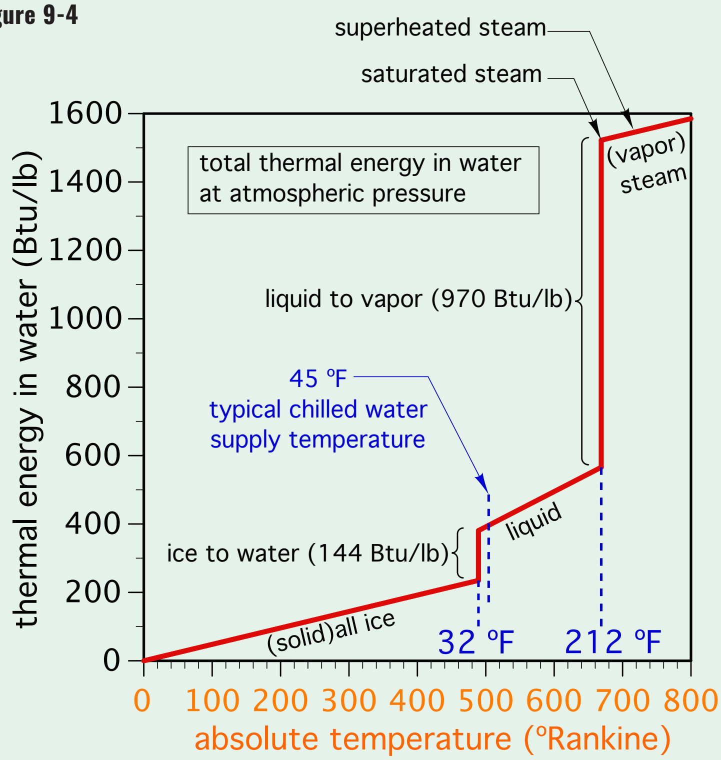
One way to apply demand-side management to commercial cooling loads is through ice storage systems. *Each pound of liquid water at 32ºF must give up 144 Btus to form one pound of ice at 32ºF. This allows ice to store much more cooling effect per pound of water compared to simply lowering the water's temperature.*

Figure 9-5

Courtesy of CALMAC®

Figure 9-4 shows the total thermal energy in water versus its absolute temperature.

Figure 9-4



Notice the significant increase in energy as a pound of water changes from ice to water. This transition can also be viewed in reverse, as a large increase in "cold storage" as a pound of liquid water changes to a pound of ice. A pound of *liquid* water would have to change its temperature by 144ºF to store an equivalent amount of thermal energy.

Also notice that the change between ice and liquid water occurs at a constant temperature of 32ºF. This temperature is very close to the typical supply water temperature of 45ºF in chilled-water cooling systems.

The fundamental concept of an ice storage cooling system is to operate a chiller during periods of low utility rates (typically at night) to transform a volume of liquid water, held in one or more large, unpressurized, insulated containers, into ice. This ice is then melted to supply cooling during the subsequent peak loading period.

One possibility is a configuration called a "full storage" system. In this approach, the amount of ice created during a given "off-peak" period will be sufficient to meet the building's entire cooling load during the subsequent "on-peak" period. In theory, this eliminates the need to operate the chiller during the "on-peak" period, and significantly lowers both the cost of operation and the peak power demand of the system. Most commercial buildings are billed for *both* the amount of electrical *energy* used over a billing period (e.g., kilowatt•hours), as well as the peak *power* demand (e.g., kilowatts) over any 15-minute period during that billing period.

Another scenario, referred to as a "partial storage" system is to displace the majority (but not all) of the on-peak cooling load using ice storage, and supply the balance of the load by operating the chiller. This approach typically has a higher operating cost relative to a full storage system, but also has a significantly lower installation cost since the chiller and ice bank storage can be smaller.

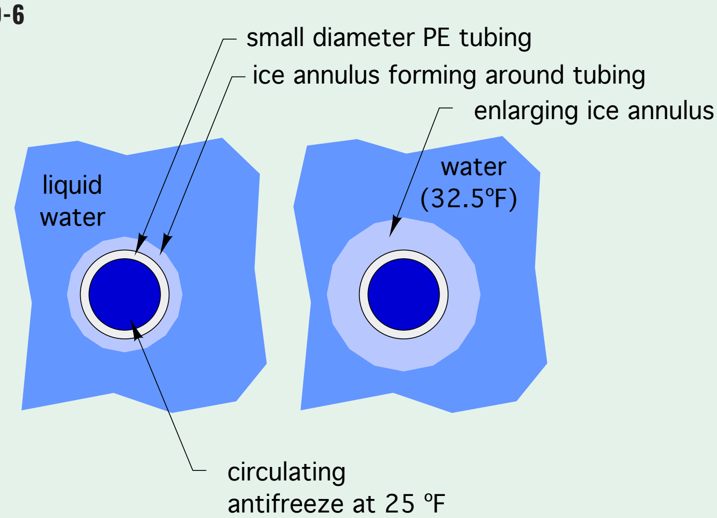
Figure 9-5 shows an array of large unpressurized insulated thermal storage tanks that serve as the "ice bank®."

These tanks can range in volume from approximately 500 to 5000 gallons. They are available in a range of diameters and heights to coordinate with different installation locations and constraints. Multiple tanks can be combined in parallel to increase ice storage capacity.

Each tank contains several thousand feet of small diameter polyethylene tubing arranged in coils. The coils are immersed in water, which nearly fills the shell of the tank. The coils are also manifolded together, resulting in two 4-inch piping connections between the ice bank tank and the remainder of the system. Up to 9 tanks can also be connected in parallel using an internal header system.

During the ice-making phase of operation, which typically lasts 8 to 10 hours, a propylene glycol or ethylene glycol antifreeze solution at approximately 25ºF is circulated

Figure 9-6



through the tubing coils. An annulus of ice forms around the chilled tubing, as depicted in Figure 9-6.

The phase change from liquid water to ice creates a slight volumetric expansion within the tank. The water that is not yet frozen absorbs this expansion, resulting in a slight rise in water level within the tank. This approach prevents stresses within the tank. At the end of the ice-making phase, approximately 95%of the water in the tank has been converted to ice.

One or more of the ice storage tanks are combined with one or more chillers and a hydronic cooling distribution system. Figure 9-7 illustrate one possible arrangement, shown during the ice-making phase.

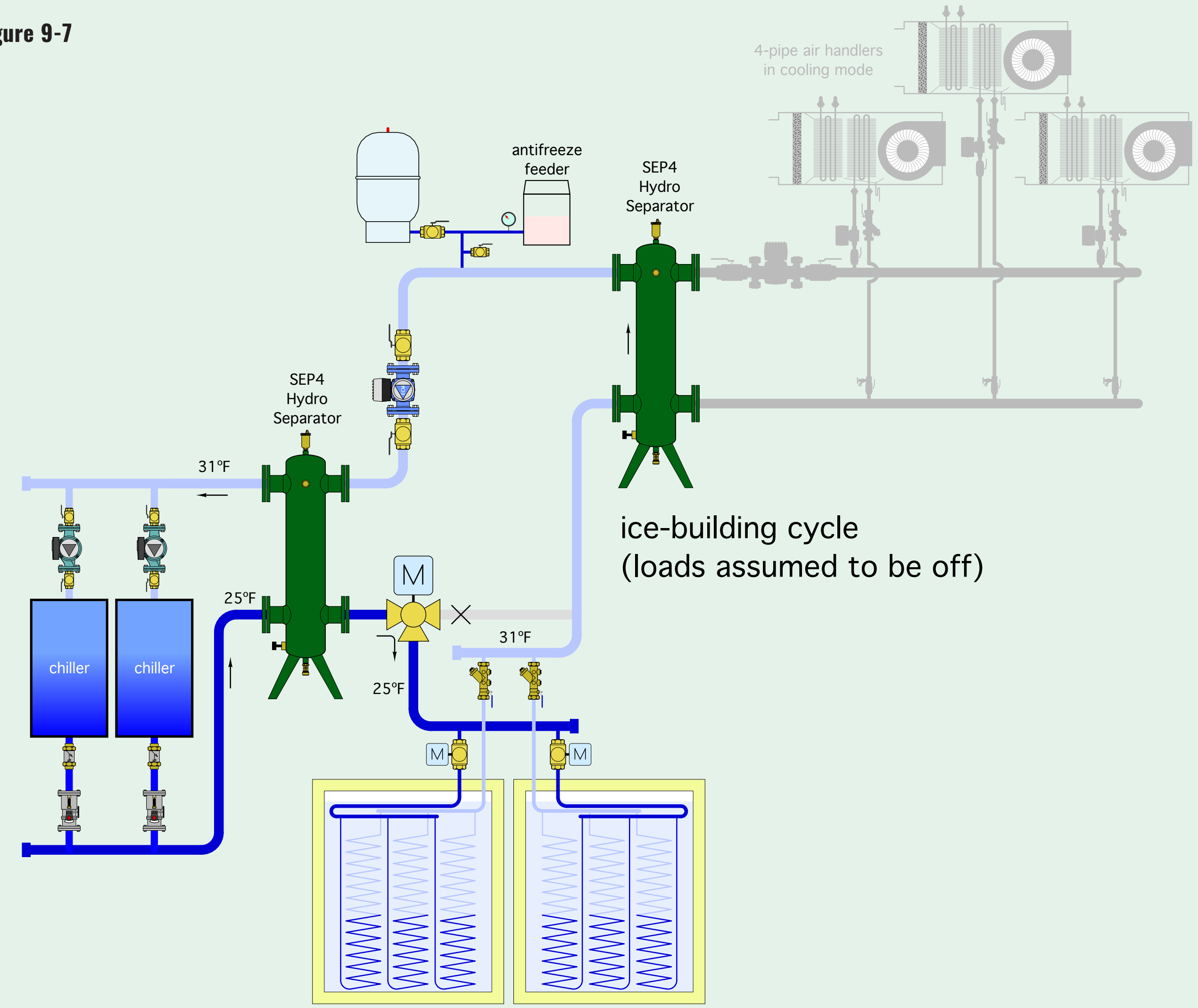
The chilled antifreeze solution exits the chillers at approximately 25ºF. A hydraulic separator prevents the pressure dynamics of the chiller circulators from interfering with the pressure dynamics of the main loop circulator and allows the chillers to maintain their minimum operating flow rate under other operating conditions. This separator also eliminates air, dirt and magnetic particles from the fluid.

The chilled antifreeze is routed into the coils of the ice storage tanks by a modulating valve. The slightly warmer antifreeze exiting the tanks flows through another hydraulic separator, through the main loop circulator, and eventually back to the chillers.

Although the loads are shown to be off during the ice-making cycle, which typically occurs at night, it is possible for one or more of the loads to operate during this time if needed. A 3-way motorized valve mixes some of the 31ºF antifreeze solution leaving the ice storage tanks and into the hydraulic separator, with some of the water returning from any active load to achieve a supply fluid temperature of approximately 44ºF to the active cooling load. The ice production cycle continues until approximately 95% of the water in the ice bank tanks has been frozen, or possibly until the off-peak period ends.

Figure 9-8 shows the system during the ice-*melting* mode. The chillers are assumed to be off, and the cooling load is completely supplied from the ice bank tanks.

Figure 9-7



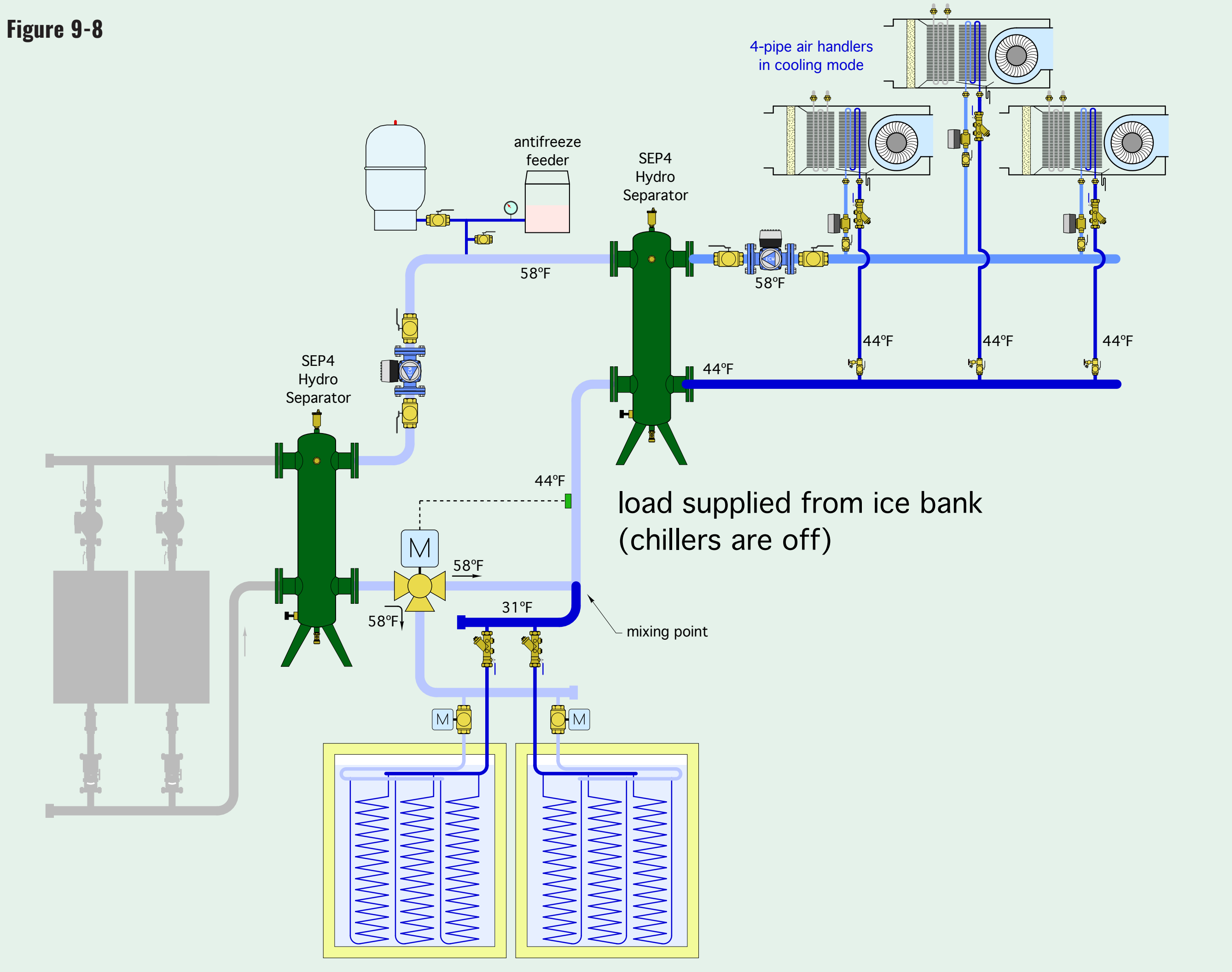
If a condition is reached where the ice storage tanks can no longer supply the cooling load, one of the chillers would be operated, perhaps at partial capacity, to supplement the output of the ice storage tanks. The modulating 3-way valve would determine the proportions of flow through the ice bank versus straight through the valve.

Figure 9-9 shows a scenario where the cooling effect available from the ice storage tank has been depleted (e.g., all ice melted and the water in the tanks elevated to a temperature that can no longer provide adequate cooling at the loads).

At this point, the chillers supply the entire cooling load. The two hydraulic separators allow the chiller flow rates to be independent of the main loop flow. They also allow the load flow rate to be independent of the main loop flow rate.

The load consists of multiple 4-pipe air handlers. Flow to each chilled-water coil is controlled by a zone valve. A pressure-independent balancing valve maintains a stable predetermined flow rate through each coil when the associated zone valve is open, regardless of the on/off status of the other air handlers. A variable-speed pressure-regulated circulator automatically adjusts its speed based on proportional differential pressure control. The 3-way mixing valve downstream of the load circulator would have its bypass fully closed. System fluid would be supplied to each air handler at 44ºF. The fluid leaving each active air handler at approximately 58ºF would pass through the hydraulic separator and back to the chiller. If modern ECM circulators are used, a building automation system can be used to match flow rate of the main loop circulator to the flow rate of the load circulator. This prevents mixing within the hydraulic separator and assures that the warmest fluid is returned to the chiller.

Figure 9-8



Ice-based thermal storage can be retrofitted to existing chilled-water distribution systems. In these situations, it is possible to install a generously sized (e.g., low approach temperature difference) plate & frame heat exchanger to separate the antifreeze solution in the thermal storage portion of the system from the water in the existing portion of the system.

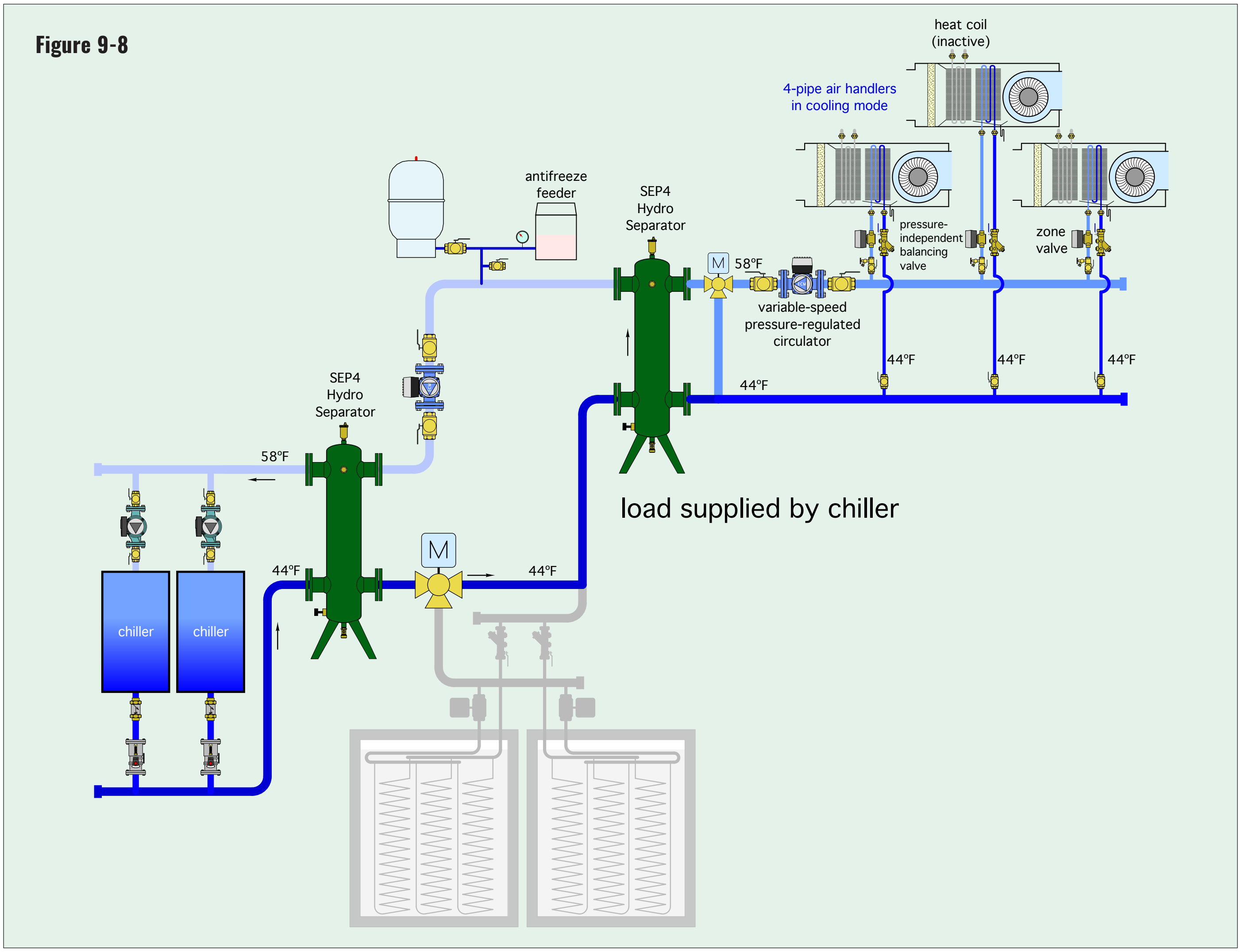
Ice-based thermal storage cooling systems provide several benefits, including:

- Lower operating cost based on off-peak electrical rates.
- Reduced capacity chiller sizing relative to peak load (66% of peak load is a starting point estimate).
- Reduced utility demand charges due to reduced chiller capacity.
- Ability to provide several hours of cooling if the chiller is temporarily offline for maintenance following a full ice-making cycle.
- Ability to provide some cooling capacity during a power outage, assuming a sufficient backup generator is available to power the circulators.
- Some utilities offer incentives to reduce the installation cost of this type of system.

SUMMARY

Today, there are very few new or remodeled commercial buildings or institutional buildings that do not have a means of cooling. Its appeal, and availability in many different forms, make it an expectation rather than a luxury. As population patterns within the U.S. have progressively shifted to warmer climates, the market for cooling in nearly all commercial and institutional buildings has created continuously expanding market opportunities. This issue of *idronics* has discussed the unique and unmatched role that modern hydronics technology can play in supplying that market.

Figure 9-8



The following points summarize the design considerations for these systems:

- The starting point for all cooling system design is an accurate estimate of sensible and latent cooling loads.

- Piping for conveyance of chilled water is much smaller than ducting of equivalent cooling capacity.

- Hydronic-based cooling systems have much higher distribution efficiency compared to air-based cooling distribution systems.

- All terminal units that provide latent cooling must have drip pans.

- All chilled-water piping must be insulated and vapor-sealed to prevent condensation.

- Multiple staged chillers or heat pumps can provide commercial cooling capacity in locations that only have single phase power.

- Heat recovery chillers can serve simultaneous heating and cooling loads. So can water loop heat pump systems.

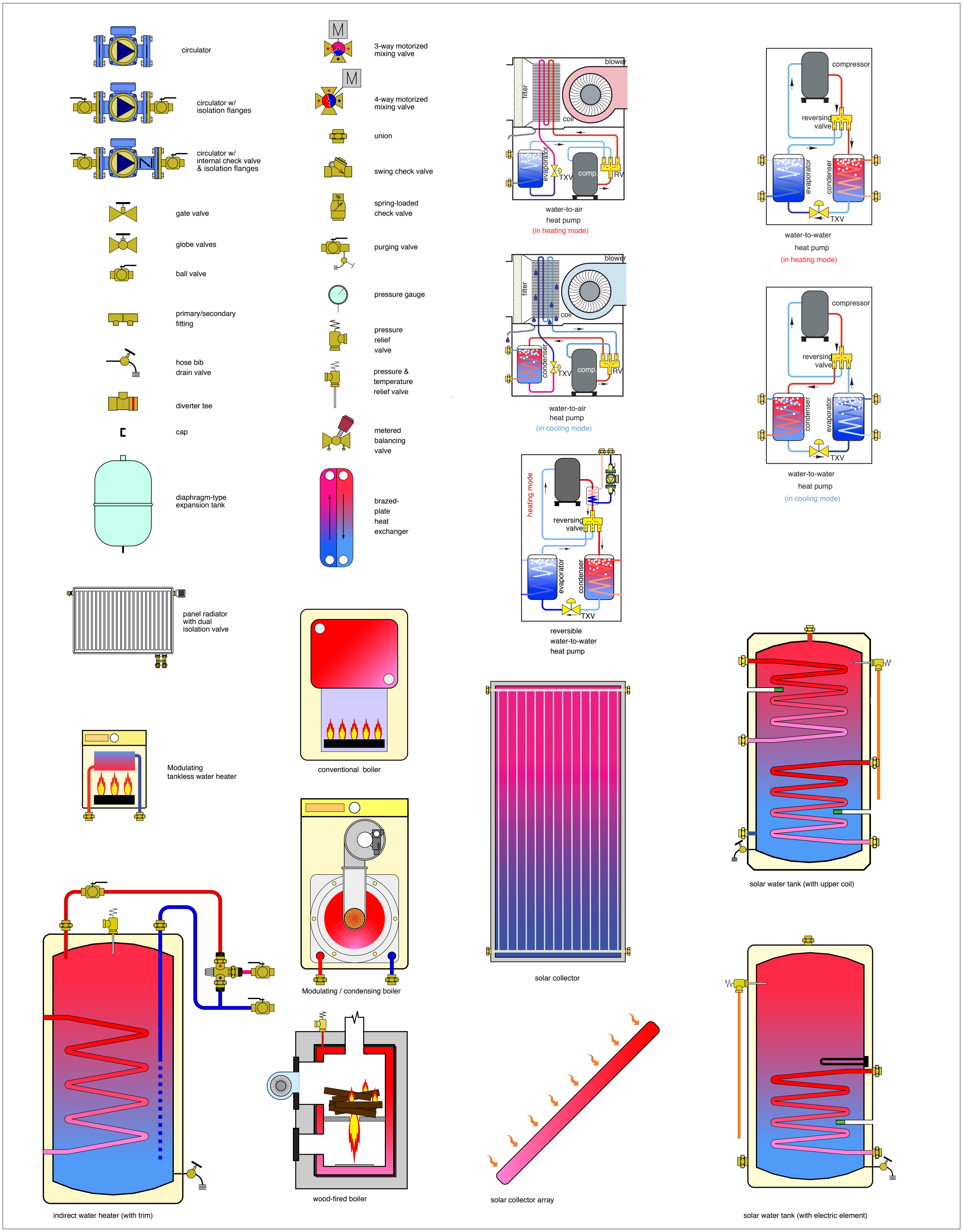
- Circulator power should be minimized, since it all adds to the cooling load. Use of variable-speed circulators in combination with valve-based zoning will significantly lower operating costs in multi-zone systems.

- Radiant cooling panels and chilled beams must remain above the current dewpoint temperature of the room to avoid condensation.

- Ice storage cooling systems can significantly lower operating cost and reduce the size of required chiller plants.

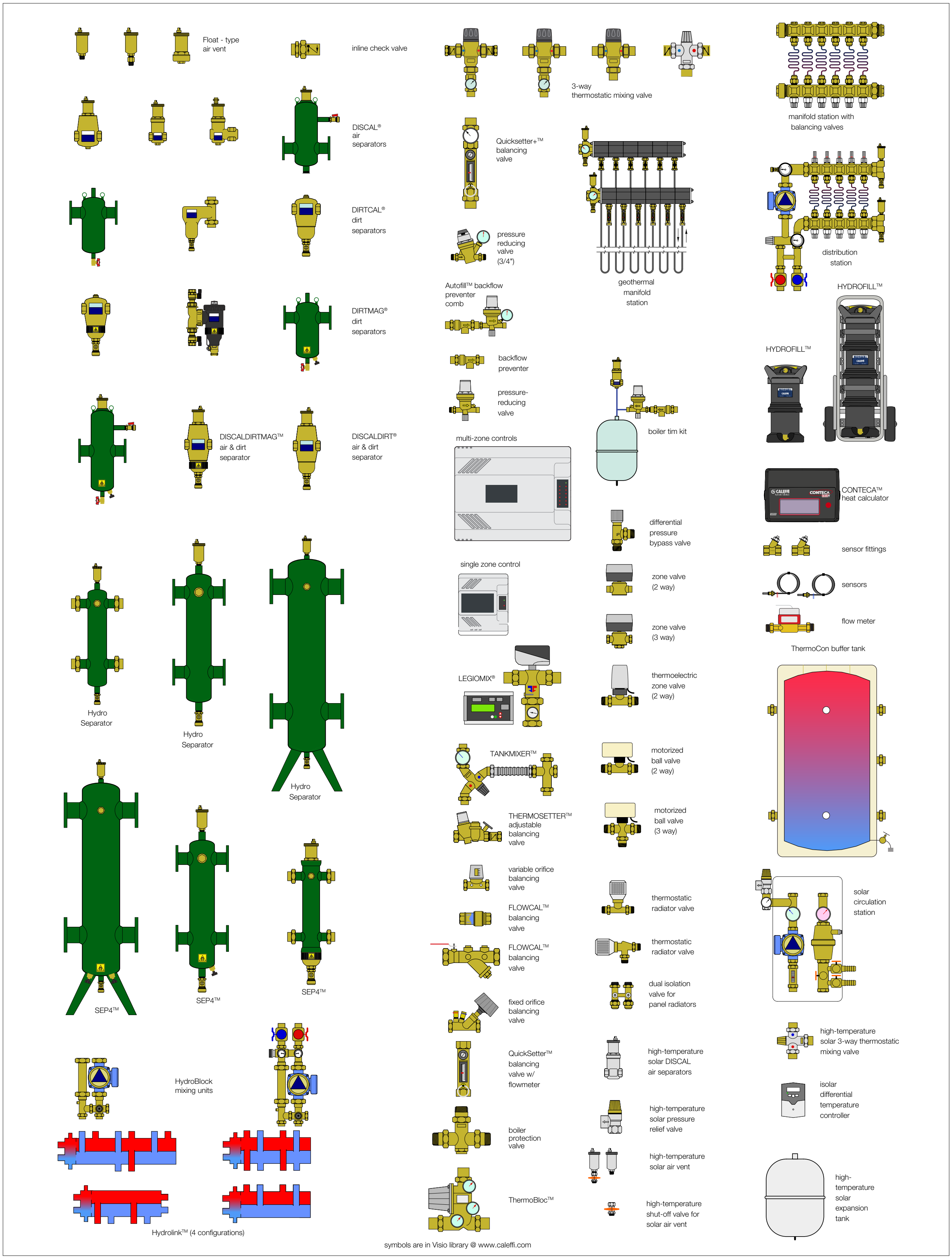
APPENDIX A: GENERIC PIPING SYMBOL LEGEND

GENERIC COMPONENTS



APPENDIX A: CALEFFI COMPONENT SYMBOL LEGEND

CALEFFI COMPONENTS



APPENDIX B: PSYCHROMETRIC CHART