CHAPTER 34

THERMAL STORAGE

THERMAL storage systems remove heat from or add heat to a storage medium for use at another time. Thermal storage for HVAC applications can involve storage at various temperatures associated with heating or cooling. High-temperature storage is typically associated with solar energy or high-temperature heating, and cool storage with air-conditioning, refrigeration, or cryogenic-temperature processes. Energy may be charged, stored, and discharged daily, weekly, annually, or in seasonal or rapid batch process cycles. The *Design Guide for Cool Thermal Storage* (Dorgan and Elleson 1993) covers cool storage issues and design parameters in more detail.

Thermal storage may be an economically attractive approach to meeting heating or cooling loads if one or more of the following conditions apply:

- Loads are of short duration
- Loads occur infrequently
- Loads are cyclical in nature
- Loads are not coincident with the availability of the energy source
- Energy costs are time-dependent (e.g., time-of-use energy rates)
- Charges for peak power demand are high
- Utility rebates, tax credits, or other economic incentives are provided for the use of load-shifting equipment
- Energy supply from the utility is limited, thus preventing the use of full-size nonstorage systems

Terminology

Heat storage. As used in this chapter, the storage of thermal energy at temperatures above the nominal temperature of the space or process.

Cool storage. As used in this chapter, the storage of thermal energy at temperatures below the nominal temperature of the space or process.

Mass storage. Storage of energy in building materials in the form of sensible heat.

Sensible energy storage (sensible heat storage). Heat or cool storage in which all of the energy stored is in the form of sensible heat associated with a temperature change in the storage medium.

Latent energy storage (latent heat storage). Heat or cool storage in which the energy stored is largely latent heat (usually of fusion) associated with a phase change (usually between solid and liquid states) in the storage medium.

Off-peak air conditioning. An air-conditioning system that uses cool storage during peak periods that was produced during off-peak periods.

Off-peak heating. A heating system that uses heat storage.

The preparation of this chapter is assigned to TC 6.9, Thermal Storage.

Storage Media

A wide range of materials can be used as the storage medium. Desirable characteristics include the following:

- Commonly available
- Low cost
- Environmentally benign
- Nonflammable
- Nonexplosive
- Nontoxic
- Compatible with common HVAC materials
- Noncorrosive
- Inert
- Well-documented physical properties
- High density
- High specific heat (for sensible heat storage)
- High heat of fusion (for latent heat storage)
- High heat transfer characteristics
- Storage at ambient pressure
- Characteristics unchanged over long use

Common storage media for sensible energy storage include water, soil, rock, brick, ceramics, concrete, and various portions of the building structure (or process fluid) being heated or cooled. In HVAC applications such as air conditioning, space heating, and water heating, water is often the chosen thermal storage medium; it provides virtually all of the desirable characteristics when kept between its freezing and boiling points. In lower-temperature applications, aqueous secondary coolants (typically glycol solutions) are often used as the heat transfer medium, enabling certain storage media to be used below their freezing or phase-change points. For high-temperature energy storage, the storage medium is often rock, brick, or ceramic materials for residential or small commercial applications; oil, oil-rock combinations, or molten salt are often used for large industrial or solar energy power plant applications. Use of the building structure itself as passive thermal storage offers advantages under some circumstances (Morris et al. 1994). A study by ASHRAE of operational experience with thermal storage systems was conducted in 1993 to 1994 (Potter et al. 1995).

Common storage media for latent energy storage include ice, aqueous brine-ice solutions, and other phase-change materials (PCMs) such as hydrated salts and polymers. Clathrates, carbon dioxide, and paraffin waxes are among the alternative storage media used for latent energy storage at various temperatures. For air-conditioning applications, ice is the most common storage medium; it provides virtually all of the previously listed desirable characteristics.

A challenge common to all latent energy storage methods is to find an efficient and economical means of achieving the heat transfer necessary to alternately freeze and melt the storage medium. Various methods have been developed to limit or deal with the heat transfer approach temperatures associated with freezing and melting; however, leaving fluid temperatures (from storage during melting) must be higher than the freezing point, and entering fluid temperatures (to storage during freezing) must be lower than the freezing point. Ice storage can provide leaving temperatures well below those normally used for comfort and nonstorage air-conditioning applications. However, entering temperatures are also much lower than normal. Certain PCM storage systems can be charged using temperatures near those for comfort cooling, but they produce warmer leaving temperatures.

Benefits of Thermal Storage

The primary reasons for using thermal storage are typically economic. The following are some of the key benefits of storage:

Reduced Equipment Size. If thermal storage is used to meet all or a portion of peak heating or cooling loads, equipment sized to meet the peak load can be downsized to meet an average load.

Capital Cost Savings. Capital savings can result both from equipment downsizing and from certain utility cash incentive programs. Even in the absence of utility cash incentives, the savings from downsizing cooling equipment can offset the cost of the storage system. Cool storage integrated with low-temperature air and water distribution systems can also provide an initial cost savings because of the use of smaller chillers, pumps, piping, ducts, and fans. Storage has the potential to provide capital savings for systems having heating or cooling peak loads of extremely short duration.

Energy Cost Savings. The significant reduction of time-dependent energy costs such as electric demand charges and on-peak time-of-use energy charges is a major economic incentive for the use of thermal storage.

Energy Savings. Although thermal storage is generally designed primarily to *shift* energy use rather than to *conserve* energy, storage can reduce energy consumption and cost. Cool-storage systems permit chillers to operate more at night, when lower condensing temperatures improve equipment efficiency; storage permits the operation of equipment at full load, avoiding inefficient part-load performance. Documented examples include chilled-water storage installations that reduce annual energy consumption on a kWh/ton-hour basis for air conditioning by up to 12% (Bahnfleth and Joyce 1994; Fiorino 1994). Heat recovery from chiller condensers can also reduce, or even potentially eliminate, the need for heating equipment and associated energy use (Goss et al. 1996).

Improved HVAC Operation. Storage adds an element of thermal capacitance to a heating or cooling system, allowing the decoupling of the thermal load profile from the operation of the equipment. This decoupling can be used to provide increased flexibility, reliability, or backup capacity for the control and operation of the system.

Other Benefits. Storage can bring about other beneficial synergies. As already noted, cool storage can be integrated with cold-air distribution. Thermal storage can be configured to serve a secondary function such as fire protection, as is often the case with chilled-water storage (Holness 1992; Hussain and Peters 1992; Meckler 1992). Some cool storage systems can be configured for recharge via free cooling. Thermal storage can also be used to recover waste energy from base-loaded steam plants by storing chilled water produced in an absorption chiller for later use upon demand.

ECONOMICS

Thermal storage is typically installed for two major reasons: (1) to lower initial cost and (2) to lower operating cost.

Lower Initial Cost. Applications such as churches and sports facilities, where the load is short in duration and there is a long time between load occurrences, generally have the lowest initial cost for thermal storage systems. Larger applications of nonmodular thermal storage technologies, such as chilled-water storage, can offer a significant economy of scale, and can be less expensive than equivalent nonstorage mechanical chilling systems (Benjamin 1994). Ice storage systems also offer added savings because of their compactness and further reduction in sizes of pumps, piping, air-handling units, ductwork, and space for these items.

Secondary capital costs may also be lower for thermal storage. For example, the electrical supply (kVa) can sometimes be reduced because peak energy demand is lower. For cold-air distribution systems, the volume of air is reduced; thus, smaller air-handling equipment and ducts may generally be used. Smaller air-handling unit sizes may reduce the mechanical room space required, providing more usable space. Smaller duct sizes may allow the floor-to-floor height of the building to be reduced.

An example of reduction in equipment size is shown in Figure 1. The load profiles shown are for a 100,000 ft^2 commercial building. If the 2040 ton·h load is met by a nonstorage air-conditioning system, as shown in Figure 1A, a 220 ton chiller is required to meet the peak cooling demand.

If a load-leveling partial storage system is used, as shown in Figure 1B, an 85 ton chiller meets the demand. The design-day cooling load, in excess of the chiller output (1020 ton·h), is supplied by the storage. The cost of storage approximates the amount saved by downsizing the chiller, cooling tower, electrical service, etc., so load-leveling partial storage is often competitive with nonstorage systems on an initial cost basis.

If a full-storage system is installed, as shown in Figure 1C, the entire peak load is shifted to the storage, and a 120 ton chiller is required. The size of the chiller equipment may be reduced, but the total equipment cost including the storage is usually higher for the full-storage system than for nonstorage systems. Although the initial cost is higher than for the load-leveling system (Figure 1B), full storage offers substantially reduced operating costs because the entire chiller demand is shifted to the off-peak period.

To prepare a complete cost analysis, the initial cost must be determined. Equipment cost should be obtained from each manufacturer under consideration, and an estimate of installation cost should be made.

Lower Operating Cost. The other reason for installing thermal storage is to reduce operating cost. Most electric utilities charge less during the night or weekend off-peak hours than during the time of highest electrical demand, which often occurs on hot summer afternoons because of air-conditioning use. Electric rates are normally divided into a demand charge and a consumption charge, although the use of real-time pricing (which substitutes time-varying consumption charges for the more common demand/consumption structure) is growing.

The monthly demand charge is based on the building's highest recorded demand for electricity during the month and is measured over a brief period (usually 1 to 15 min). Ratchet billing, another form of demand charge, is based on the highest annual monthly demand. This charge is assessed each month, even if the maximum demand in the particular month is less than the annual peak.

The consumption charge is based on the total measured use of electricity in kilowatt-hours (kWh) over a longer period and is generally representative of the utility's cost of fuel to operate its generation facilities. In some cases, the consumption charge is lower during off-peak hours because a higher proportion of the electricity is generated by baseload plants that are less expensive to operate. Rates that reflect this difference are known as time-of-use billing structures. Reductions in total energy consumption by some thermal storage systems can also directly reduce the consumption quantity and charge.

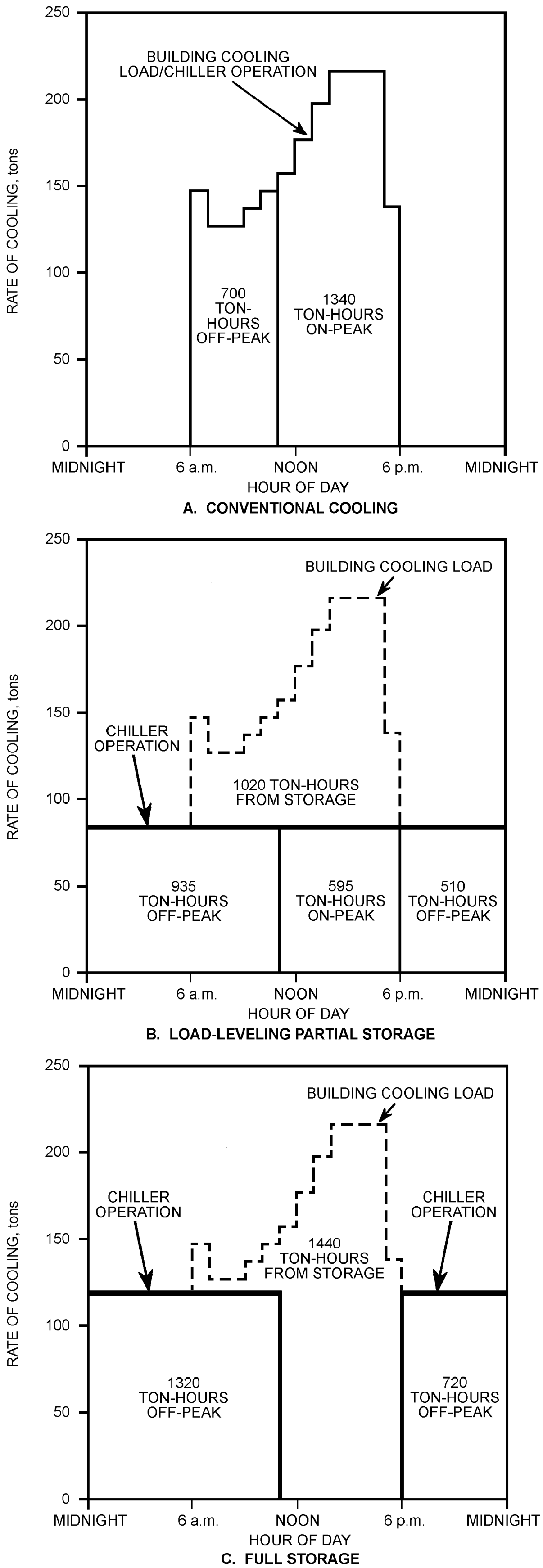

Fig. 1 Hourly Cooling-Load Profiles

To compare the costs of different systems, the annual operating cost of each system being considered must be estimated, including both electrical demand and consumption costs. To determine demand cost, the monthly peak demand for each system is multiplied by the demand charge and totaled for the year; any necessary adjustments for ratchet billing must be made. The electrical consumption cost is determined by totaling the annual energy use for each system in kilowatt-hours and multiplying it by the cost per kilowatt-hour. For time-of-use billing, energy use must be classified by time-of-use period and multiplied by the corresponding rate.

The first step is to estimate the annual utility cost for a nonstorage system. The average unit electric power consumption of the complete chilling system (including cooling tower fans, pumps, and other ancillary equipment) must be known or estimated, e.g., 1.0 kW/ton. For example, the peak electric power demand in Figure 1A, which occurs at 3 P.M. (220 ton × 1.0 kW/ton = 220 kW), is multiplied by the demand charge and, if necessary, adjusted for ratchet billing. The energy used during the off-peak period for the nonstorage system (corresponding to 700 ton·h of cooling) would be multiplied by the off-peak electrical rate. The energy used during the on-peak period (1340 ton·h) is multiplied by the on-peak electrical rate. This procedure is repeated (1) for each day of each month to determine the energy charge and (2) for the peak day in each month to determine the demand charge.

Annual cost for a load-leveling partial thermal storage is calculated the same way. The peak demand in Figure 1B (85 ton × 1.0 kW/ton = 85 kW) is multiplied by the demand charge and adjusted for any ratchet billing. This process is repeated for each month. The demand charge for the load-leveling system is lower than for a nonstorage system; the difference represents the demand savings. If time-of-use billing is used, the energy used during the off-peak period (935 + 510 ton·h) is multiplied by the off-peak electrical rate, and the energy consumed during the on-peak period (595 ton·h) is multiplied by the on-peak electrical rate. The total energy cost is lower because less energy is used during the on-peak hours.

The full-storage system in Figure 1C eliminates all energy use during the on-peak hours except energy used by the auxiliary equipment that transfers stored energy. The energy cost for the off-peak period (1320 + 720 ton·h) is multiplied by the off-peak rate. This procedure should be repeated for each month to determine the annual energy cost.

The operating cost and the net capital cost should be analyzed using the life-cycle cost method or other suitable method to determine which design is best for the project. Other items to be considered are space requirements, reliability, and ease of interface with the planned delivery equipment. An optimal application of thermal energy storage balances the savings in utility charges against the initial cost of the installation needed to achieve the savings. Many thermal storage systems are partial load shift systems because these systems often offer the lowest combination of capital and operating costs.

APPLICATIONS

Thermal storage can take many forms to suit a variety of applications. This section addresses several groups of thermal storage applications: off-peak air conditioning, retrofits, industrial/process cooling, combustion turbine inlet air cooling, off-peak heating, district heating and cooling, and other applications.

Thermal Storage Operation and Control

In general, thermal storage operation and control are more schedule-dependent than for instantaneous systems. Because thermal storage systems separate the generation of heating or cooling from its use, control of each of these functions must be considered separately. Also, many thermal storage systems offer the ability to provide heating or cooling either directly or from storage. With this flexibility comes the need to define how loads will be met at any time.

Table 1 Typical Thermal Storage Operating Modes

Operating Mode	For Heat Storage	For Cool Storage
Charging storage	Operating heating equipment to add heat to storage	Operating cooling equipment to remove heat from storage
Charging storage while meeting loads	Operating heating equipment to add heat to storage *and* meet loads	Operating cooling equipment to remove heat from storage *and* meet loads
Meeting loads, from discharging storage only	Discharging (removing heat from) storage to meet loads without operating heating equipment	Discharging (adding heat to) storage to meet loads without operating cooling equipment
Meeting loads, from discharging storage and direct equipment operation	Discharging (removing heat from) storage *and* operating heating equipment to meet loads	Discharging (adding heat to) storage *and* operating cooling equipment to meet loads
Meeting loads, from direct equipment operation only	Operating heating equipment to meet loads (no fluid flow to or from storage)	Operating cooling equipment to meet loads (no fluid flow to or from storage)

ASHRAE Research Project RP-1054, Cool Storage Operating and Control Strategies (Dorgan et al. 1999, 2001), developed a framework for describing and characterizing methods for providing required control functionality. These methods are defined in terms of (1) the operating modes available to the system, (2) the control strategies used to implement the operating modes, and (3) the operating strategies that determine when the various operating modes and control strategies are selected.

A complete thermal storage design includes a detailed description of the intended operating strategy and its associated control strategies and operating modes. Research Project RP-1054 provides specific recommendations for documenting each element of thermal storage operation and control. The researchers also recommend providing a graphical description to illustrate each operating mode and the logic used to select each mode. This documentation should address control for the complete storage cycle under full-load and part-load operation, including seasonal variations.

Design documentation should also include a description of design intent, including the sizing strategy (full or partial storage); applicable electricity or fuel cost structure; and reasons for selecting the operating strategy.

Operating Modes

A thermal storage **operating mode** describes which of several possible functions the system is performing at a given time. This should be defined by a specific control sequence.

The five most common thermal storage operating modes are described in Table 1. The available operating modes differ for individual thermal storage systems. Some systems may include fewer than the five basic modes. For example, the option to meet the load while charging may not be available. In some installations, operation to meet loads may be defined by a single operating mode that includes discharging-only at one end of a continuum and direct equipment-only at the other. In fact, many systems operate with just two modes: daytime and nighttime operation.

Many systems also include other operating modes. Some examples include

- Charging cool storage from free cooling
- Charging cool storage while recovering condenser heat
- Charging heat storage with recovered condenser heat
- Discharging at distinct supply temperatures
- Discharging in conjunction with various combinations of available equipment

In general, the control system selects the current operating mode based on the time of day and the day of the week. Other variables that may also be considered include outdoor temperature, current load, or total facility demand at the billing meter.

In some cases, particularly large cool-storage systems with multiple chillers and multiple loads, different operating modes or control strategies may be applied to different parts of a system at one time. For example, one chiller may operate in a charging-only mode, while another chiller operates to meet a load.

Control Sequences

An operating mode is defined for a given system by its control sequence. The control sequence defines what equipment is running, including the values of their set points, and what actions should occur in response to changes in load or other variables.

Charging Storage. Control sequences for the charging mode are generally easily defined. Typically the generation equipment operates at full capacity with a constant supply-temperature set point and a constant flow through the storage. This operation continues until the storage is fully charged or the period available for charging has ended. Under this basic charging control strategy, the entire capacity of the equipment is applied to charging storage.

Charging Storage While Meeting Load. A control sequence for charging storage while meeting load generally also operates the generation equipment at its maximum capacity. Capacity that is not needed to meet the load is applied to charging storage. Depending on system design, the load may be piped either in series or parallel with storage under this operating mode. Some systems may have specific requirements for this mode. For example, in an ice storage system with a heat exchanger between glycol and water loops, the control sequence may have to address freeze protection for the heat exchanger.

Meeting Load from Discharging Only. A control sequence for the discharging-only mode (full storage or load-shifting operation) is also straightforward. The generating equipment does not operate and the entire load is met from storage.

Meeting Load from Discharging and Direct Equipment Operation. Control sequences for this mode are more complex and must regulate what portion of the load, at any time, will be met from storage and what proportion will be met from direct generation. These partial-storage sequences have been primarily developed for and applied to cool storage. Although they could also be applied to heat storage, the following discussion is in terms of cool storage. Three common control sequences are chiller-priority, storage-priority, and constant-proportion or proportional.

Chiller-priority control operates the chiller, up to its available capacity, to meet loads. Cooling loads in excess of the chiller capacity are met from storage. If a chiller demand limit is in place, the available capacity of the chiller is less than the maximum capacity.

Chiller-priority control can be implemented with any storage configuration. However, it is most commonly applied with the chiller in series upstream of storage. A simple method of implementing chiller-priority control is to set both the chiller's onboard supply-temperature set point and the temperature set point downstream of storage to the desired chilled-water supply temperature. When the load exceeds the chiller capacity, the chiller discharge temperature exceeds its set point, and some flow is diverted through storage to provide the required additional cooling. Sensing errors in the storage downstream measurement may cause the storage to be unintentionally used before the chiller has reached full capacity.

Storage-priority control meets the load from storage up to its available discharge rate. If the load exceeds this discharge rate, the chiller operates to meet the remaining load.

A storage-priority control sequence must ensure that storage is not depleted too early in the discharge cycle. Failure to properly limit the discharge rate could cause loss of control of the cooling load, or excessive demand charges, or both. Load forecasting is required to maximize the benefits of storage-priority control. A method for forecasting diurnal energy requirements is described in Chapter 41, Supervisory Control Strategies and Optimization. Simpler storage-priority sequences using constant discharge rates, predetermined discharge rate schedules, or pseudopredictive methods have also been used.

A **constant-proportion or proportional control** sequence divides the load between chiller and storage. The load may be divided equally or in some other proportion. The proportion may change with time in response to changing conditions. A limit on chiller demand or storage discharge may be applied.

Demand-limiting control may be applied to any of the above control sequences. This type of control attempts to limit the facility demand either by setting a maximum capacity above which the chiller is not allowed to operate or by modulating the chiller set point. Because chiller demand may comprise 30% or more of the total demand, a large demand savings is possible. The demand limit may be a constant value or it may change with time in response to changing conditions. Demand limiting is most effective when the chiller capacity is controlled in response to the facility demand at the billing meter. In such cases, the chiller capacity is controlled to keep the total demand from exceeding a predetermined facility demand limit. A simpler approach, which generally achieves lower demand savings, is simply to limit chiller capacity or the chiller's electric demand without considering the total facility demand.

The storage discharge rate may also be limited, similar to the chiller demand limit. The discharge limit establishes a maximum discharge rate that the storage is allowed to provide. Such a limit is typically used with storage-priority control to ensure that sufficient capacity will be available for the entire discharge cycle. The discharge rate limit may change with time in response to changing conditions. The discharge limit may be defined in terms of maximum instantaneous cooling capacity supplied from storage. Alternatively, it may be defined either in terms of the maximum flow through storage, or the minimum mixed temperature leaving the storage and its bypass.

Operating sequences that seek to optimize system operation often recalculate the demand limit and discharge limit on a regular basis during the discharge period.

Nearly any partial-storage control sequence can be described by specifying (1) whether it is chiller-priority, storage-priority, or constant-proportion, and (2) by specifying the applicable chiller demand limit and storage discharge rate limit.

Applicable utility rates and system efficiency in various operating modes determine the selection of a control sequence. If on-peak energy cost is significantly higher than off-peak energy cost, the use of stored energy should be maximized and a storage-priority sequence is appropriate. If on-peak energy is not significantly more expensive than off-peak energy, a chiller-priority sequence is more appropriate. If demand charges are high, some type of demand-limiting control should be implemented.

Control Strategies

A thermal storage **control strategy** is the sequence of operating modes implemented under specific conditions of load, weather, season, etc. For example, different control strategies might be implemented on a design day in the summer, a cool day in the summer, and a winter day. The control strategy further defines the actions of individual control loops and the values of their set points in response to changes in load or other variables.

Operating Strategies

The **operating strategy** defines the overall method of controlling the thermal storage in order to achieve the design intent; it determines the logic that governs the selection of operating modes such as charging, meeting the load from chiller(s) only, or meeting the load from discharging storage. The operating strategy also determines which control strategy is implemented within each mode.

It is important to distinguish between the operating strategy, which defines the higher-level logic by which a system will be operated, and the various control strategies, which implement the operating strategy through selection of operating modes.

Dorgan and Elleson (1993) use the term *operating strategy* to refer to full-storage and partial-storage operation. That discussion focuses on design-day operation and does not discuss operation under all conditions. For example, a system that is designed for partial-storage operation on the design day may operate with a full-storage strategy during many times of the year.

Operating strategies that use sophisticated routines to optimize storage use have been investigated (Drees 1994). The cost-saving benefits of such optimal strategies are often small in comparison to well-designed logic that makes full and appropriate use of the principles described previously.

OFF-PEAK AIR CONDITIONING

Refrigeration Design

Packaged compression or absorption equipment is available for most of the cool storage technologies discussed in this chapter. The maximum cooling load that can be satisfied with a single package and the number of packages that must be installed in parallel to yield the required capacity depend on the specific technology. Individual components such as ice builders, chiller bundles, falling-film chillers, ice makers, chilled-water storage tanks, compressors, and condensers can be used and interconnected onsite.

Chillers used in chilled-water storage systems are typically conventional absorption, centrifugal, or screw chillers operating at conditions similar to those for nonstorage applications.

For ice storage systems, the lower suction temperature necessary for making ice imposes a higher compression ratio on the refrigeration equipment. Positive displacement compressors (e.g., reciprocating, screw, and scroll compressors) are usually better suited to these higher compression ratios than centrifugal compressors.

Ice storage systems must operate over a wider range of conditions than nonstorage systems. Care should be taken in design and installation to ensure proper operation at all operating points. The following general categorization of refrigeration equipment applies to ice storage systems:

- Direct-expansion systems feed refrigerant to the heat exchanger through expansion valves. The wide range of operating loads requires careful selection and control of expansion valves.
- Flooded or overfeed systems circulate liquid refrigerant from a large low-pressure receiver to the ice-making heat exchangers. Chapter 1 of the 2002 *ASHRAE Handbook—Refrigeration* covers liquid overfeed systems.
- Secondary coolant systems chill and circulate low-temperature secondary coolant to the ice-producing and storage equipment. The corrosion properties of the coolant should be considered; a solution containing corrosion inhibitors is normally added. The chiller must also be properly derated both for the lower heat transfer properties of the secondary coolant and for the lower-than-normal operating temperatures. Secondary coolant systems may use direct-expansion, flooded, or liquid overfeed refrigeration systems.

The following steps should be considered for all ice thermal storage refrigeration:

- ***Design for part-load operation***. Refrigerant flow rates, pressure drops, and velocities are reduced during part-load operation. Components and piping must be designed so that control of the system can be maintained and oil can be returned to the compressors at all load conditions.
- ***Design for pulldown load***. Because ice-making equipment is designed to operate at water temperatures approaching 32°F, a higher load is imposed on the refrigeration system during the initial start-up, when the inlet water is warmest. Components must be sized to handle this higher load.
- ***Plan for chilling versus ice making***. Most ice-making equipment has a much higher instantaneous water-/fluid-chilling capacity than ice-making capacity. This higher chilling capacity can be used advantageously if the refrigeration equipment and interconnecting piping are properly sized and selected to handle it.
- ***Protect compressors from liquid slugging***. Ice builders and ice harvesters tend to contain more refrigerant than chillers of similar capacity used for nonstorage systems. This provides more opportunity for compressor liquid slugging. Care should be taken to oversize suction accumulators and equip them with high-level compressor cutouts and suction heat exchangers to evaporate any remaining liquid.
- ***Oversize the receivers***. Every opportunity should be taken to make the system easy to maintain and service. Maintenance flexibility may be provided in a liquid overfeed system by oversizing the low-pressure receiver and in a direct-expansion system by oversizing the high-pressure receiver.
- ***Prevent oil trapping***. Refrigerant lines should be arranged to prevent the trapping of large amounts of oil and to ensure its return to compressors under all operating conditions, especially during periods of low compressor loads. All suction lines should slope toward the suction line accumulators, and all discharge lines should pitch toward the oil separators. Oil tends to collect in the evaporator because that is the location at the lowest temperature and pressure. Because refrigerant accumulators trap oil as well as liquid refrigerant, the larger accumulators needed for ice storage systems require special provisions to ensure adequate oil return to the compressors.

Cold-Air Distribution

Reducing the temperature of the distribution air is attractive because smaller air-handling units, ducts, pumps, and piping can be used, resulting in a lower initial cost. In addition, the reduced ceiling space required for ductwork can significantly reduce building height, particularly in high-rise construction. These cost reductions can make thermal storage competitive with nonstorage on an initial-cost basis (Landry and Noble 1991; Nelson 2000).

The optimum supply air temperature should be determined through an analysis of initial and operating cost for the various design options. Depending on the load, the additional latent energy removed at the lower discharge air temperature may be offset by the reduction in fan energy associated with the lower airflow.

The minimum achievable supply air temperature is determined by the chilled-water temperature and the temperature rise between the cooling plant and the terminal units. With some ice storage systems, the fluid temperature may rise during discharge; the supply temperatures normally achievable with various types of ice storage plants must therefore be carefully investigated with the equipment supplier.

A heat exchanger, which is sometimes required with storage tanks that operate at atmospheric pressure or between a secondary coolant and chilled-water system, adds at least 2°F to the final chilled-water supply temperature. The rise in chilled-water temperature between the cooling plant discharge and the chilled-water coil depends on the length of piping and the amount of insulation; the difference could be up to 0.5°F over long, exposed runs.

The difference between the temperature of the chilled-water entering the cooling coil and the temperature of the air leaving the coil is generally between 6 and 10°F. A closer approach (smaller temperature differential) can be achieved with more rows or a larger face area on the cooling coil, but extra heat transfer surface to provide a closer approach is often uneconomical. A 3°F temperature rise because of heat gain in the duct between the cooling coil and the terminal units can be assumed for preliminary design analysis. With careful design and adequate insulation, this rise can be reduced to as little as 1°F.

A blow-through configuration provides the lowest supply air temperature and the minimum supply air volume. The lowest temperature rise achievable with a draw-through configuration is 2 to 3°F because heat from the fan is added to the air. A draw-through configuration should be used if space for flow straightening between the fan and coil is limited. Blow-through units should generally be avoided in systems using a lined duct because air with a higher relative humidity, nearly 100% saturated air, enters the duct; draw-through units provide some reheat of the supply air before it enters the duct.

Face velocity determines the size of the coil for a given supply air volume. The coil size determines the size of the air-handling unit. A lower face velocity generates a lower supply air temperature, whereas a higher face velocity results in smaller equipment and lower first costs. The face velocity is limited by moisture carryover from the coil. The face velocity for cold air distribution should be 350 to 450 fpm, with an upper limit of 550 fpm.

Cold primary air can be tempered with room air or plenum return air by using series-style fan-powered mixing boxes or induction boxes. The primary air should be tempered before it is supplied to the space. The energy use of fan-powered mixing boxes is significant and negates much of the savings from downsizing central supply fans (Elleson 1993; Hittle and Smith 1994). Diffusers designed for cold-air distribution can provide supply air directly to the space without causing drafts, thereby eliminating the need for fan-powered boxes for the purpose of tempering the supply air.

If the supply airflow rate to occupied spaces is expected to be below 0.4 cfm per square foot, fan-powered or induction boxes should be used to boost the air circulation rate. At supply airflow rates of 0.4 to 0.6 cfm per square foot, a diffuser with a high ratio of induced room air to supply air should be used to ensure adequate dispersion of ventilation air throughout the space. A diffuser that relies on turbulent mixing rather than induction to temper the primary air may not be effective at this flow rate.

Cold-air distribution systems normally maintain space relative humidity between 30 and 45% rh, as opposed to the 50 to 60% rh generally maintained by other systems. At this lower humidity level, equivalent comfort conditions are provided at a higher dry-bulb temperature. The increased dry-bulb set point generally results in decreased energy consumption.

The surfaces of any equipment that may be cooled below the ambient dew point, including air-handling units, ducts, and terminal boxes, should be adequately insulated. All vapor barrier penetrations should be sealed to prevent migration of moisture into the insulation. Prefabricated, insulated round ducts should be insulated externally at joints where internal insulation is not continuous. If ducts are internally insulated, access doors should also be insulated.

Duct leakage is undesirable because it represents cooling capacity that is not delivered to the conditioned space. In cold-air distribution, leaking air can cool nearby surfaces to the point that condensation forms. Designers should specify acceptable methods of sealing ducts and air-handling units and establish allowable leakage rates and test procedures. During construction, these specifications must be followed up with on-site supervision and inspection by appropriate personnel.

A thorough commissioning process is important for the optimal operation of any large space-conditioning system, particularly thermal storage and cold-air distribution systems. Reductions in initial and operating costs are major selling points for cold-air distribution, but the commitment to provide a successful system must not be compromised by the desire to reduce first cost. A commissioning procedure may appear to involve additional expense, but it actually decreases cost by reducing future malfunctions and troubleshooting expense and provides increased value by ensuring optimal operation. The commissioning process is discussed in greater detail in the section on Implementation and Commissioning.

Storage of Heat in Cool Storage Units

Some cool storage installations may be used to provide storage for heating duty and/or heat reclaim. Many commercial buildings have areas that require cooling even during the winter, and the refrigeration plant may be in operation year-round. When cool storage is charged during the off-peak period, the rejected heat may be directed to the heating system or to storage rather than to a cooling tower. Depending on the building loads, the chilled water or ice could be considered a by-product of heating. One potential use of the heat may be to provide morning warm-up on the following day.

Heating with a cool storage system can be achieved using heat pumps or heat-reclaim equipment such as double-bundle condensers, two condensers in series on the refrigeration side, or a heat exchanger in the condenser water circuit. Cooling is withdrawn from storage as needed; when heat is required, the cool storage is recharged and used as the heat source for the heat pump. If needed, additional heat must be obtained from another source because this type of system can supply usable heating energy only when the heat pump or heat-recovery equipment is running. Possible secondary heat sources include a heat storage system, solar collectors, or waste heat recovery from exhaust air. When more cooling than heating is required, excess energy can be rejected through a cooling tower.

Individual storage units may be alternated between heat storage and cool storage service. The stored heat can be used to meet the building's heating requirements, which may include morning warm-up and/or evening heating, and the stored cooling can provide midday cooling. If both the chiller and storage are large enough, all compressor and boiler operation during the on-peak period can be avoided, and the heating or cooling requirements can be satisfied from storage. This method of operation replaces the air-side economizer cycle that involves the use of outside air to cool the building when the enthalpy of the outside air is lower than that of the return air.

The extra cost of equipping a chilled-water storage facility for heat storage is small. The necessary plant additions could include a partition in the storage tank to convert a portion to warm-water storage, some additional controls, and a chiller equipped for heat reclaim. The additional cost may be offset by savings in heating energy. In fact, adding heat storage may increase the economic advantage of cool storage.

Care should be taken to avoid thermal shock in cast-in-place concrete tanks. Tamblyn (1985) showed that the seasonal change from heat storage to cool storage caused sizable leaks to develop if the cooldown period was fewer than five days. Raising the temperature of a concrete tank causes no problems because it generates compressive stresses, which concrete can sustain. Cooldown, on the other hand, causes tensile stresses, under which concrete has low strength and is more prone to failure.

STORAGE FOR RETROFIT APPLICATIONS

Because latent energy storage (e.g., ice and other phase-change materials) typically has smaller volume than sensible energy (e.g., chilled-water) storage, it is often preferable to chilled-water storage for retrofit installations where space is limited. The relatively small size and modular nature of ice-on-coil units is often an advantage in cases where access to the equipment room is limited or where the storage units can be distributed throughout the building and located near the cooling coils.

Existing installations that already contain chilled-water piping and chillers are generally more easily retrofitted to accept chilled-water storage than latent energy storage. The requirement for secondary coolant in some ice storage systems may complicate pumping and heat exchange for existing equipment. A secondary coolant temperature colder than the chilled-water temperature in the original design typically offsets the reduction in heat transfer.

INDUSTRIAL/PROCESS COOLING

Refrigeration and process cooling typically require lower temperatures than air conditioning. Applications such as vegetable hydrocooling, milk cooling, carcass spray-cooling, and storage room dehumidification can use cold water from an ice storage system. Lower temperatures for freezing food, storing frozen food, and so forth can be obtained by using either a low-temperature sensible energy storage fluid or a low-temperature hydrated salt phase-change material (PCM) charged and discharged by a secondary coolant or a nonaqueous PCM such as CO_2.

Refrigeration applications have traditionally used heavy-duty industrial refrigeration equipment. The size of the loads has often required custom-engineered, field-erected systems, and the occupancy has often allowed the use of ammonia. Cool storage systems for these applications can be optimized by the choice of refrigerant (halocarbon or ammonia) and the choice of refrigerant feed control (direct-expansion, flooded, mechanically pumped, or gas-pumped).

Some of the advantages of cool storage may be particularly important for refrigeration and process cooling applications. Cool storage can provide a steady supply temperature regardless of large, rapid changes in cooling load or return temperature. Charging equipment and discharging pumps can be placed on a separate electrical service to economically provide cooling during a power failure or to take advantage of low, interruptible electrical rates. Storage tanks can be oversized to provide water for emergency cooling or fire fighting, as well. Cooling can be extracted very quickly from harvested ice storage or chilled-water storage to satisfy a very large load and then be recharged slowly using a small charging system. System design may be simplified when the load profile is determined by production scheduling rather than occupancy and outdoor temperatures.

COMBUSTION TURBINE INLET AIR COOLING

An industrial application for cool storage is the precooling of inlet air for combustion turbines. Oil- or gas-fired combustion turbines typically generate full rated output at an inlet air temperature of 59°F. At higher inlet air temperatures, the mass flow of air is reduced, as are the shaft power and fuel efficiency. The capacity and efficiency can be increased by precooling the inlet combustion air with chilled-water coils or direct contact cooling. The optimum inlet air temperature is typically in the range of 40 to 50°F, depending on the turbine model.

On a 100°F summer day, a combustion turbine driving a generator may lose up to 25% of its rated output. This drop in generator output occurs at the most inopportune time, when an electrical utility or industrial user is most in need of the additional electrical output available from combined cycle or peaking turbines. This loss in capacity can be regained if the inlet air is artificially cooled (Andrepont 1994; Ebeling et al. 1994; MacCracken 1994; Mackie 1994). Storage systems using chilled water or ice are well suited to this application because the cooling is generated and stored during off-peak periods, and the maximum net increase in generator capacity is obtained by turning off the refrigeration compressor during peak demand periods (Andrepont 2001; Cross et al. 1995). However, there is some parasitic power required to operate water pumps to use the stored cooling energy.

Although instantaneous cooling is also used for inlet air precooling, this refrigeration equipment often uses 25 to 50% of the increased turbine power output. Cool storage systems can be used to meet either the full- or partial-load requirements for inlet air precooling. Evaporative coolers are also a simple and well-established technology for inlet air precooling, but they cannot deliver the low air temperatures available from refrigeration. Additional information can be found in the *Design Guide for Combustion Turbine Inlet Air Cooling Systems* (Stewart 1999).

OFF-PEAK HEATING

Service Water Heating

The tank-equipped service water heater, which is the standard water heater in North America, is a thermal storage device; some electric utilities provide incentives for off-peak water heating. The heater is equipped with a control system that is activated by a clock or by the electric utility to curtail power use during peak demand. An off-peak water heater generally requires a larger tank than a conventional heater.

An alternative system of off-peak water heating consists of two tanks connected in series, with the hot-water outlet of the first tank supplying water to the second tank (ORNL 1985). This arrangement minimizes the mixing of hot and cold water in the second tank. Tests performed on this configuration show that it can supply 80 to 85% of rated capacity at suitable temperatures, compared to 70% for a single-tank configuration. The wiring for the heating elements in the two tanks must be modified to accommodate the dual-tank configuration.

Solar Space and Water Heating

Rock beds, water tanks, or PCMs can be used for solar space and water heating, depending on the type of solar system employed. Various applications are described in Chapter 33.

Space Heating

Thermal storage for space heating can be carried out by radiant floor heating, brick storage heaters (electric thermal storage or ETS heaters), water storage heaters, or PCMs. Radiant floor heating is generally applied to single-story buildings. The choice between the other storage methods depends on the type of heating system. Air heaters use brick, whereas water heaters may use water or a PCM. Water heaters can be charged either electrically or thermally.

Insulation is more important for heat storage than for cool storage because the difference between storage and ambient temperatures is greater. Methods of determining the cost-effective thickness of insulation are given in Chapter 23 of the 2001 *ASHRAE Handbook—Fundamentals*.

A rational design procedure requires an hourly simulation of the design heat load and discharge capacity of the storage device. The first criterion for satisfactory performance is that the discharge rate be no less than the design heating load at any hour. During the off-peak period, energy is added to storage at the rate determined by the connected load of the resistance elements, and the design heating load is subtracted. The second criterion for satisfactory performance, the daily energy balance, is checked at the end of a simulated 24 h period. Assuming that the design day is preceded and followed by similar days, the energy stored at the end of the simulated day should equal that at the start.

A simpler procedure is recommended by Hersh et al. (1982) for typical residential designs. For each zone of the building, the design heat loss is calculated in the usual manner and multiplied by the selected sizing factor. The resulting value (rounded to the next kilowatt) is the required storage heater capacity. Sizing factors in the United States range from 2.0 to 2.5 for an 8 h charge period and from 1.6 to 2.0 for a 10 h charge period. The lower end of the range is marginal for the northeastern United States. The designer should consult the manufacturer for specific sizing information. Figure 2 shows a typical sizing factor selection graph.

Brick Storage (ETS) Heaters. These heaters (commonly called ETS or electric thermal storage heaters) are electrically charged and store heat during off-peak times. In this storage device, air circulates through a hot brick cavity and then discharges into the area in which heat is desired. The brick in these heaters (commonly called ceramic brick) has a very dense magnetite or magnesite composition. The high density of the brick and its ability to store heat at a high temperature give the heater a large thermal storage capacity. Ceramic brick can be heated to approximately 1400°F during off-peak hours by resistance heating elements. Space requirements for brick storage heaters are usually much less than other storage mediums.

Figure 3 shows the operating characteristics of an electrically charged room storage heater. Curve 1 represents theoretical performance. In reality, radiation and convection from the exterior surface

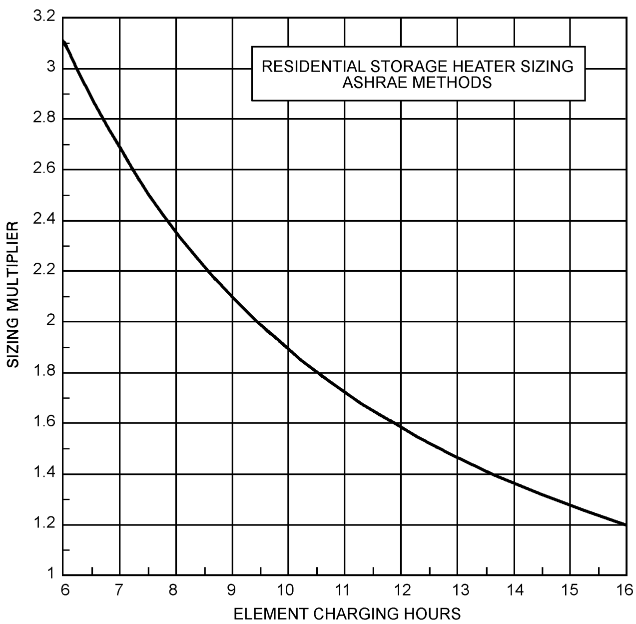

Fig. 2 Representative Sizing Factor Selection Graph for Residential Storage Heater

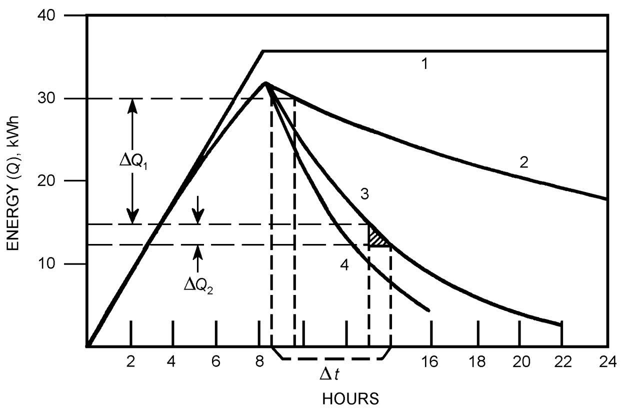

Fig. 3 Storage Heater Performance Characteristics
(Hersh et al. 1982)

of the device continually supplies heat to the room during charging. Curve 2 shows this static discharge. When the thermostat calls for heat, the internal fan starts operating. The resulting faster dynamic discharge corresponds to Curves 3 and 4. Because the heating elements of electrically charged room storage heaters are energized only during off-peak periods, they must store the total daily heating requirement during this period.

The three types of brick storage heaters currently available are room units, heat pump boosters, and central furnaces. The section on Electrically Charged Heat Storage Devices explains the various types of brick storage heaters.

Water Storage Heaters. Electrically charged pressurized water storage tanks, in most applications, must be able to recharge the full daily heating requirement during the off-peak period, while supplying heat during the off-peak period. The heat exchanger design allows a constant discharge and does not decrease near the end of the cycle.

Thermally charged hot water storage tanks are similar in design to the cool storage tanks described in the section on Off-Peak Air Conditioning. Many are also used for cooling. Unlike off-peak cooling, off-peak heating seldom permits a reduction in the size of the heating plant. The size of a storage tank used for both heating and cooling is based more on cooling than on heating for lowest life-cycle cost, except in some residential applications.

District Heating and Cooling. These systems distribute thermal energy (steam, hot- and chilled-water) from a central energy source (plant) to residential, commercial, and industrial consumers via a distribution system (piping) for use in comfort heating/cooling, service water heating, and process heating/cooling. Thermal energy storage has operated successfully in such district heating and cooling (DHC) applications as off-peak air conditioning, industrial/process cooling, etc. Because most storage tanks are open, attention should be given to system hydraulics when connecting several plants to a common distribution network where storage tank water levels are at different elevations. Chapter 11 of the 2000 *ASHRAE Handbook—HVAC Systems and Equipment* has additional information on DHC systems and related topics.

The distribution system in DHC applications is a large percentage of the total system cost. As a result, when higher temperature differentials available from thermal storage media are used, lower life-cycle costs are achieved through the use of smaller pipes, valves, pumps, motors, starters, etc. Furthermore, thermal storage may be used to increase the available cooling capacity of an existing chilled-water distribution system by providing cooler supply water. This modification would have little effect on the distribution system itself (i.e., pumping and piping) because the existing system flow rates and pressure drops are not changed.

OTHER APPLICATIONS

Storage in Aquifers

Aquifers can be used to store large quantities of thermal energy. Aquifers are underground, water-yielding geological formations, either unconsolidated (gravel and sand) or consolidated (rocks). In general, the natural aquifer water temperature is slightly warmer than the local mean annual air temperature. Aquifer thermal energy storage has been used for process cooling, space cooling, space heating, and ventilation air preheating (Jenne 1992). Aquifers can be used as heat pump sinks or sources and to store energy from ambient winter air, waste heat, and renewable sources. For more information, see Chapter 32, Geothermal Energy.

The thickness and porosity of the aquifer determine the storage volume. Two separate wells are normally used to charge and discharge aquifer storage. A well pair may be pumped either (1) constantly in one direction or (2) alternately from one well to the other, especially when both heating and cooling are provided. Backflushing is recommended to maintain well efficiency.

The length of the storage period depends on local climate and on the type of building or process being supplied with cooling or heating. Aquifer thermal energy storage may be used on a short-term or long-term basis, as the sole source of energy or as partial storage, and at a temperature useful for direct application or needing upgrade. It may also be used in combination with a dehumidification system such as desiccant cooling. The cost-effectiveness of aquifer thermal energy storage is based on the avoidance of equipment capital cost and on lower operating cost.

Aquifer thermal energy storage may be incorporated into a building system in a variety of ways, depending on the other components present and the intentions of the designer (CSA 1993; Hall 1993a). Control is simplified by separate hot and cold wells that operate on the basis that the last water in is the first water out. This principle ensures that the hottest or coldest water is always available when needed.

Seasonal storage has a large savings potential and began as an environmentally sensitive improvement on the large-scale mining of groundwater (Hall 1993a). The current method reinjects all pumped water to attempt annual thermal balancing (Morofsky 1994; Public Works Canada 1991, 1992; Snijders 1992). Rock caverns have also been used successfully in a manner similar to aquifers to store energy. For example, Oulu, Finland, stores hot water in a cavern for district heating.

STORAGE TECHNOLOGIES

SENSIBLE ENERGY CHANGE STORAGE

Water is well suited for both hot and cold sensible energy storage applications, in part because it has the highest specific heat (1 Btu/lb·°F) of all common materials. ASHRAE *Standard* 94.3 covers procedures for measuring the thermal performance of sensible heat storage. Tanks are available in many shapes; however, vertical cylinders are the most common. Tanks can be located above ground, partially buried, or completely buried. They can also be incorporated into the building structure. They usually operate at atmospheric pressure and may have clear-span dome roofs or column-supported shallow cone roofs.

Water thermal storage vessels must separate the cool and warm water. This section focuses on chilled-water thermal energy storage because it is the most common system type. However, similar techniques should apply to hot-water sensible energy storage for heating systems or to nonaqueous stratification fluids.

Temperature Range and Storage Size

The cooling capacity of a chilled-water storage vessel is proportional to the volume of water stored and the temperature differential (Δt) between the stored cool water and the returning warm water. For economical storage, the cooling coils should provide as large a Δt as practical. For example, chillers that cool water in storage to 40°F and coils that return water to storage at 60°F provide a Δt of 20°F. The cost of the extra coil surface required to provide this range can be offset by savings in pipe size, insulation, and pumping energy. Storage is likely to be uneconomical if the temperature differential is less than about 10°F because the tank must be so large (Caldwell and Bahnfleth 1997).

The initial cost of chilled-water storage benefits from a dramatic economy of scale; that is, large installations can be less expensive than equivalent nonstorage chilled-water plants (Andrepont 1992).

Techniques for Separating Cool and Warm Water Volumes

The following methods have all been applied in chilled-water storage. Thermal stratification has become the dominant method because of its simplicity, reliability, efficiency, and low cost. The other methods are described only for reference.

Thermal Stratification. In thermal stratified storage, the warmer, less dense returning water floats on top of the stored chilled water. The water from storage is supplied and withdrawn at low velocity, in essentially horizontal flow, so that buoyancy forces dominate inertial effects. Pure water is most dense at 39.2°F; therefore, it cannot be stratified below this temperature. However, low-temperature fluids (LTFs) can be used to achieve lower delivery temperatures.

When the stratified storage tank is charged, chilled supply water, typically between 39 and 44°F, enters through the diffuser at the bottom of the tank (Figure 4), and return water exits to the chiller through the diffuser at the top of the tank. Typically, the incoming water mixes with water in the tank to form a 1 to 3 ft thick thermocline, which is a region with vertical temperature and density gradients (Figure 5). The thermocline prevents further mixing of the water above it with that below it. The thermocline rises as recharging continues and subsequently falls during discharging. It thickens somewhat during charging and discharging due to heat conduction through the water and heat transfer to and from the walls of the tank. The storage tank may have any cross-section, but the walls are usually vertical. Horizontal cylindrical tanks are generally not good candidates for stratified storage.

Flexible Diaphragm. A flexible diaphragm (normally horizontal) is used in a tank to separate the cold water from the warm water.

Multiple Compartments. Multiple compartments in a single tank or a series of two or more tanks can also be used. Pumping is scheduled so that one compartment is always at least partially empty; water returning from the system during the occupied period and from the chiller during storage regeneration is then received into the empty tank. Water at the different temperatures is thus stored in separate compartments, minimizing blending.

Labyrinth Tank. This tank has both horizontal and vertical traverses. The design commonly takes the form of successive cubicles with high and low ports. Strings of cylindrical tanks and tanks with successive vertical weirs may also be used.

Performance of Chilled-Water Storage

A perfect storage tank would deliver water at the same temperature at which it was stored. This would also require that the water returning to storage neither mix nor exchange heat with the stored water or the tank. In practice, however, both types of heat exchange occur.

Typical temperature profiles of water entering and leaving a storage tank are shown in Figure 6. Tran et al. (1989) tested several large chilled-water storage systems and developed the figure of merit, which is used as a measure of the amount of cooling available from the tank.

$$\text{Figure of Merit (\%)} = \frac{\text{Area between A and C}}{\text{Area between A and D}} \times 100 \qquad (1)$$

Well-designed storage tanks have figures of merit of 90% or higher for daily complete charge/discharge cycles and between 80 and 90% for partial charge/discharge cycles.

Design of Stratification Diffusers

Stratification diffusers must be designed and constructed to produce and maintain stratification at the maximum flow through storage. The *Design Guide for Cool Thermal Storage* (Dorgan and Elleson 1993) discusses these design parameters. Wildin (1990) gives more information concerning the design of stratified storage hardware.

Inlet and outlet streams must be kept at sufficiently low velocities, so that buoyancy forces predominate over inertia forces in order to produce a gravity current (density current) across the bottom or top of the tank. Designers typically select a diffuser dimension to create an inlet Froude number of 1.0 or less. However, values up to 2.0 have been successfully applied (Yoo et al. 1986).

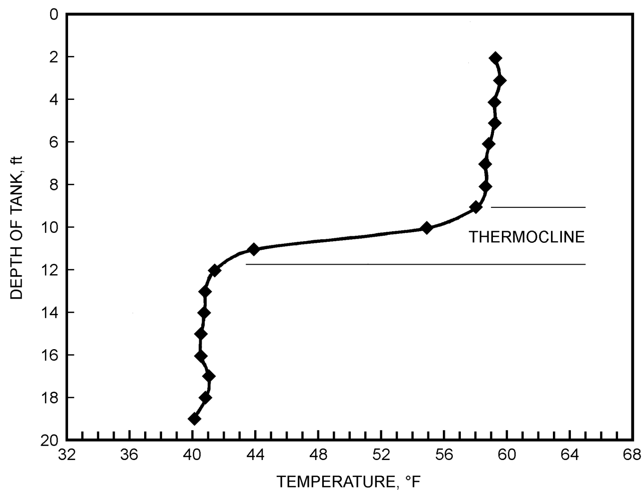

Fig. 5 Typical Temperature Stratification Profile in Storage Tank

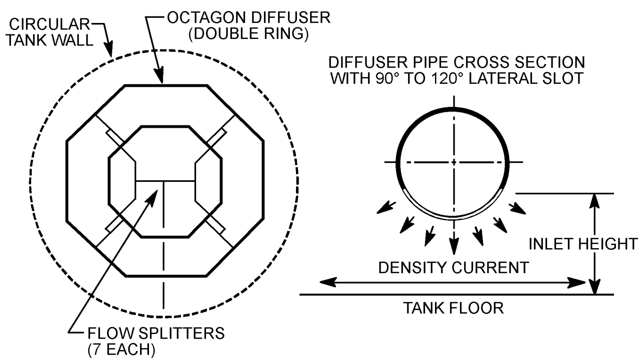

Fig. 4 Octagon Diffuser

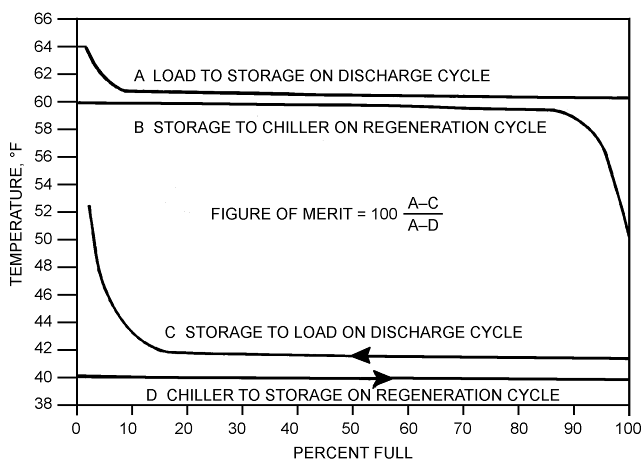

Fig. 6 Chilled Water Storage Profiles

Table 2 Chilled Water Density

°F	lb/ft³	°F	lb/ft³	°F	lb/ft³
32	62.419	44	62.424	58	62.378
34	62.424	46	62.421	60	62.368
36	62.426	48	62.417	62	62.357
38	62.427	50	62.411	64	62.344
39	62.428	52	62.404	66	62.331
40	62.427	54	62.396	68	62.316
42	62.426	56	62.387		

The inlet Froude number Fr is defined as

$$\mathrm{Fr} = \frac{Q}{\sqrt{gh^3(\Delta\rho/\rho)}} \tag{2}$$

where

Q = volume flow rate per unit length of diffuser, $\mathrm{ft^3/s \cdot ft}$
g = gravitational acceleration, $\mathrm{ft/s^2}$
h = inlet opening height, ft
ρ = inlet water density, $\mathrm{lb/ft^3}$
$\Delta\rho$ = difference in density between stored water and incoming or outflowing water, $\mathrm{lb/ft^3}$

The density difference $\Delta\rho$ can be obtained from Table 2. The inlet Reynolds number is defined as follows:

$$\mathrm{Re} = Q/\nu \tag{3}$$

where ν = kinematic viscosity, $\mathrm{ft^2/s}$.

Experimental evidence indicates that the intensity of mixing near the inlet diffuser is influenced by the inlet Reynolds number [Equation (3)]. In short tanks, in which the inlet diffuser affects a large portion of the tank volume, the inlet Reynolds number may significantly affect thermal performance. Also, tanks with sloping sidewalls, such as inverted truncated pyramids, may benefit from using lower inlet Reynolds numbers.

Wildin and Truman (1989), observing results from a 15 ft deep, 20 ft diameter vertical cylindrical tank, found that reduction of the inlet Reynolds number from 850 (using a radial disk diffuser) to 240 (using a diffuser comprised of pipes in an octagonal array) reduced mixing to negligible proportions. This is consistent with subsequent results obtained by Wildin (1991, 1996) in a 3 ft deep scale model tank, which indicated negligible mixing at Reynolds numbers below approximately 450, with the best performance achieved during testing at a Reynolds number of 250.

In larger tanks with water depths of 30 ft or more, or in tanks that are not fully discharged where cooled water remains at the bottom of the tank at the end of the discharging period, the inlet Reynolds number is less important. Field measurements indicate that inlet mixing does not significantly affect the thermocline when it is more than about 10 ft away from the inlet diffuser. Bahnfleth and Joyce (1994), Musser and Bahnfleth (1998, 1999), and Stewart (2001) documented the successful operation of tanks with water depths greater than 45 ft for design inlet Reynolds numbers as high as 10,000. However, improved thermal performance of these systems was observed at lower inlet Reynolds numbers.

A parametric study by Musser and Bahnfleth (2001) found that using Froude numbers less than 1.0 can significantly improve performance, and also confirmed that the Froude number was a parameter of first-order significance for radial diffuser inlet thermal performance, along with the ratio of diffuser diameter to diffuser height and the ratio of diffuser diameter to tank diameter.

Because a chilled-water system may experience severe pressure spikes (i.e., "water hammer") such as during rapid closing of control or isolation valves, the structural design of the diffusers should consider this potential event.

Storage Tank Insulation

Exposed tank surfaces should be insulated to help maintain the temperature differential in the tank. Insulation is especially important for smaller storage tanks because the ratio of surface area to stored volume is relatively high. Heat transfer between the stored water and the tank contact surfaces (including divider walls) is a primary source of capacity loss. Not only does the stored fluid lose heat to (or gain heat from) the ambient by conduction through the floor and wall, but heat flows vertically along the tank walls from the warmer to the cooler region. Exterior insulation of the tank walls does not inhibit this heat transfer.

The contents of chilled-water storage tanks are typically colder than the ambient dew-point temperature, so it is important that the insulation system use a high-integrity exterior vapor barrier to minimize the ingress of moisture and condensation into the insulation system.

Other Factors

The cost of chemicals for water treatment may be significant, especially if the tank is filled more than once during its life. A filter system helps keep the stored water clean. Exposure of the stored water to the atmosphere may require the occasional addition of biocides. Although tanks should be designed to prohibit leakage, the designer should understand the potential effect of leakage on the selection of chemical water treatment.

The storage circulating pumps should be installed below the minimum operating water level to ensure flooded suction. The required net positive suction pressure (NPSH) must be maintained to avoid subatmospheric conditions at the pumps.

Low-Temperature Fluid Sensible Energy Storage

Low-temperature fluids (LTFs) may also be used as a sensible thermal storage medium instead of water. Using an LTF can permit sensible energy thermal storage at temperatures lower than 39.2°F, the temperature at which the maximum density of plain water occurs. Thus, LTFs can permit lower-temperature applications of sensible energy storage, such as for low-temperature air conditioning and certain food processing applications. LTFs can be either aqueous solutions containing a chemical additive, or nonaqueous chemicals. A number of potential LTFs have been identified by Stewart (2000). One LTF has been in continuous commercial service in a stratified thermal storage tank in a very large district cooling system since 1994, and has exhibited good corrosion inhibition and microbiological control properties (Andrepont 2000)

LATENT ENERGY CHANGE STORAGE

Thermal energy can be stored in the latent heat of fusion of water (ice) or other phase change materials (PCMs). Water has the highest latent heat of fusion of all common materials: 144 Btu/lb at the melting or freezing point of 32°F. The PCM, other than water, most commonly used for thermal storage is a hydrated salt with a latent heat of fusion of 41 Btu/lb and a melting or freezing point of 47°F. The volume required to store energy in the form of latent heat is considerably less than that stored as sensible heat, which contributes to a lower capital cost of the storage system.

External Melt Ice-on-Coil Storage

The oldest type of ice storage is the refrigerant-fed ice builder, which consists of refrigerant coils inside a storage tank filled with water (Figure 7). A compressor and evaporative condenser freeze the tank water on the outside of the coils to a thickness of up to 2.5 in. Ice is melted from the outside of the formation (hence the term external melt) by circulating the return water through the tank, whereby it again becomes chilled. Air bubbled through the tank agitates the water to promote uniform ice buildup and melting.

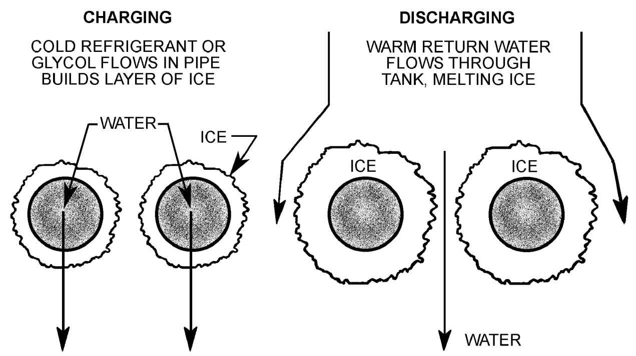

Fig. 7 Charge and Discharge of External Melt Ice Storage

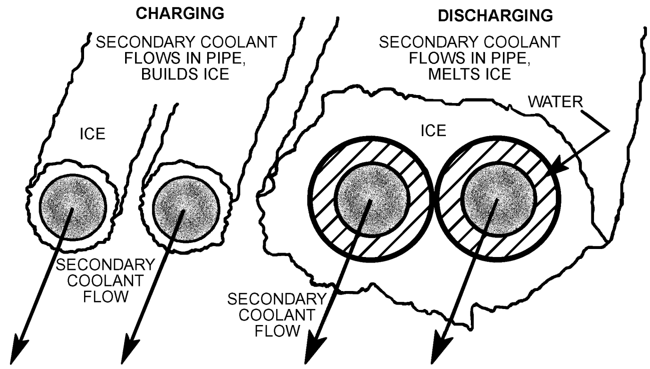

Fig. 8 Charge and Discharge of Internal Melt Ice Storage

Instead of refrigerant, a secondary coolant (e.g., 25% ethylene glycol and 75% water) can be pumped through the coils inside the storage tank. The secondary coolant has the advantage of greatly decreasing refrigerant inventory. However, a refrigerant-to-coolant heat exchanger is required.

Major concerns specific to the control of ice-on-coil storage are (1) limiting ice thickness (and thus excess compressor energy) during the build cycle and (2) minimizing the bridging of ice between individual tubes in the ice bank. Bridging must be avoided because it restricts the free circulation of water during the discharge cycle. Though not physically damaging to the tank, this blockage reduces performance, allowing a higher leaving water temperature because of the reduced heat transfer surface.

Regardless of the refrigeration method (direct-expansion, pumped liquid overfeed, or secondary coolant), the compressor is controlled by (1) a time clock or energy management system, which restricts operation to the periods dictated by the utility rate structure, and (2) an ice-thickness override control, which stops the compressor(s) at a predetermined ice thickness. At least one ice-thickness device should be installed per ice bank; the devices should be connected in series. If there are multiple refrigeration circuits per ice bank, one ice-thickness control per circuit should be installed. Placement of the ice-thickness device(s) should be determined by the ice-bank manufacturer, based on circuit geometry and flow pressure drop, to minimize bridging.

Ice-thickness controls are either mechanically or electrically operated. A mechanical control typically consists of a fluid-filled probe positioned at the desired distance from the coil. As ice builds, it encapsulates the probe, causing the fluid to freeze and apply pressure. The pressure signal controls the refrigeration system via a pneumatic-electric (PE) switch. Electric controls sense the three-times-greater electrical conductivity of ice as compared to water. Multiple probes are installed at the desired thickness, and the change in current flow between probes provides a control signal. Consistent water treatment is essential to maintaining constant conductivity and thus accurate control.

Because energy use is related to ice thickness on the coil, a partial-load ice inventory management system should be considered. This system maintains the ice inventory at the minimum level needed to supply immediate future cooling needs, rather than topping off the inventory after each discharge cycle. This method also helps prevent bridging by ensuring that the tank is completely discharged at regular intervals, thereby allowing ice to build evenly.

The most common method of measuring ice inventory is based on the fact that ice has a greater volume than water; thus, a sensed change in water level can be taken to indicate a change in the amount of stored ice. Water level can be measured by either an electrical probe or a pressure-sensing transducer.

Other methods, such as metering of energy flow rates (all temperatures and fluid flows on both sides of the coil, glycol/water and water) combined with strain gauges on the individual ice coils, can be used for calibrating inventory.

Internal Melt Ice-on-Coil Storage

In this type of storage device, coils or tubing are placed inside a water tank (Figure 8). The coils occupy approximately 10% of the tank volume; another 10% of the volume is left empty to allow for the expansion of the water upon freezing; and the rest is filled with water. A secondary coolant solution (e.g., 25% ethylene glycol and 75% water) is cooled by a liquid chiller and circulated through the coils to freeze the water in the tank. The thickness of ice on the coils and the percentage of the water in the tank that is frozen depend on the coil configuration and on the type of system. During discharge, the secondary coolant circulates to the system load and returns to the tank to be cooled again by the coils submerged in ice.

A standard chiller can provide the refrigeration for these systems. During the charging cycle for a typical system, the chilled secondary coolant exits from the chiller at a constant 25 to 26°F and returns at 31°F. When the tanks are 90% charged, the chiller inlet and outlet temperatures fall rapidly because there is little water left to freeze. When the chiller exit temperature reaches approximately 22°F, the chiller is shut down and locked off for the remainder of the charging period so that it does not short-cycle or recirculate because of convection flow through the pipes. As a result, the chiller remains fully loaded through the entire cycle and keeps the system running at its maximum efficiency because the exit-temperature thermostat, set at 22°F, prevents operation beyond that necessary to fully charge the storage tanks. The temperatures for a given system may vary from this example.

Because the same heat transfer surface freezes and melts the water, the coolant may freeze the water completely during each charge cycle, minimizing loss in efficiency. A temperature-modulating valve at the outlet of the tanks keeps a constant flow of liquid to the load (see Figure 11). Under a full-storage control strategy, the chiller is kept off during discharge, and the modulating valve allows some fluid to bypass the tanks to supply the load as needed. If a partial-storage control strategy is followed, the chiller thermostat is reset from the 22°F set point required for charging, up to the design temperature of the cooling coils (e.g., 44°F), during the discharge cycle. If the load on the building is low, the chiller operates to meet the 44°F setting without depleting the storage. If the load is greater than the chiller's capacity, the exiting temperature of the secondary coolant rises, and the temperature-modulating valve automatically opens to maintain the design temperature to the coils (Kirshenbaum 1991).

Because water increases 9% in volume when it turns to ice, the water level varies directly with the amount of ice in the tank as long as all of the ice remains submerged. This water displaced by the ice must not be frozen, or it will trap ice above the original water level.

Therefore, no heat exchange surface area can be above the original water level.

The change in water level in the tank from freezing and thawing is typically 6 in. This change can be measured with either a pressure gage or a standard electrical transducer. On projects with multiple tanks, reverse-return piping ensures uniform flow through all the tanks, so measuring the level on one tank is sufficient to determine the proportion of ice in all tanks.

Encapsulated Ice

This type of thermal storage relies on plastic containers filled with water and an ice-nucleating agent. Commercially available systems have either spherical containers of approximately 4 in. diameter (Figure 9) or rectangular containers approximately 1 3/8 by 12 by 30 in. These primary containers are placed in storage tanks, which may be either steel pressure vessels, open concrete or steel tanks, or suitable fiberglass or polyethylene tanks. In tanks with spherical containers, secondary coolant flows vertically through the tank; in tanks with rectangular containers, secondary coolant flows horizontally. The type, size, and shape of the storage tank are limited only by its ability to achieve even flow of heat transfer fluid between the containers. Refrigeration may be any type of liquid chiller rated for the lower temperatures required.

A secondary coolant (e.g., 25% ethylene glycol or propylene glycol and 75% water) is cooled to 24 to 26°F by a liquid chiller and circulates through the tank and over the outside surface of the plastic containers, causing ice to form inside the containers. As in internal melt ice-on-coil storage, the temperature at the end of the charge cycle is lower (e.g., 22°F), and the chiller must be capable of operating at this reduced temperature. The plastic containers must be flexible to allow for change of shape during ice formation; the spherical type has preformed dimples in the surface, and the rectangular type is designed for direct flexure of the walls. During discharge, coolant flows either directly to the load or to a heat exchanger, thereby removing heat from the load and melting the ice within the plastic containers. As the ice melts, the plastic containers return to their original shape.

For systems using pressure vessels, ice inventory is measured by an inventory/expansion tank and inventory control module connected directly to the main storage tank. As ice forms, the flexing plastic containers force the surrounding secondary coolant into the inventory tank. The liquid level in the inventory tank may be monitored to account for the ice available at any point during the charge or discharge cycle. For systems using open, atmospheric tanks, ice inventory is measured by changes in liquid levels within the storage tank.

Ice-Harvesting

Ice-harvesting systems separate the ice formation from storage. In one type of harvesting equipment, ice is typically formed on both sides of a hollow, flat plate or on the outside or inside (or both) of a cylindrical evaporator surface. The evaporators are arranged in vertical banks above the storage tank. Ice is formed to thicknesses between 0.25 and 0.40 in. This ice is then harvested, often by introducing hot refrigerant gas into the evaporator in a chiller defrost cycle. The gas warms the evaporator, which breaks the bond between the ice and the evaporator surface and allows the ice to drop into the storage tank below. Other types of ice harvesters use a mechanical means of separating the ice from the evaporator surface.

Ice is generated by circulating 32°F water from the storage tank over the evaporators for a 10 to 30 min. build cycle. The defrost time is a function of the amount of energy required to warm the system and break the bond between the ice and the evaporator surface. Depending on the control method, the evaporator configuration, and the discharge conditions of the compressor, defrost can be accomplished in 20 to 90 s. Typically, the evaporators are grouped in sections that are defrosted individually such that the heat of rejection from the active sections provides the energy for defrost. Knebel (1991) showed that the net available capacity from an ice harvester is the compressor gross capacity minus the fraction of time spent in defrost times the total heat of rejection from the compressor.

Tests done by Stovall (1991) indicate that, for a four-section plate ice harvester, all of the total heat of rejection is introduced into the evaporator during the defrost cycle. To maximize ice production, harvest time must be kept to a minimum.

In another type of harvesting equipment, ice is formed near the inside surface of a cylindrical evaporator but is prevented from forming on the evaporator surface by a mechanical device such as a rotating orbital rod. This arrangement reduces or eliminates the need for a defrost cycle, but adds mechanical components. The evaporators are also typically arranged in vertical banks above the storage tank.

Figure 10 shows an ice-harvesting schematic. Chilled water is pumped from the storage tank to the load and returned to the ice generator. A low-pressure recirculation pump is used to provide minimum flow for wetting the evaporator in the ice-making mode. This system may be applied to load-leveling or load-shifting applications.

In load-leveling applications, ice is generated and the storage tank charged when there is little or no building load. When a

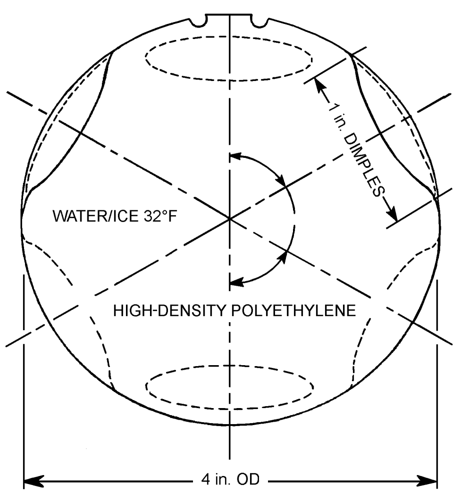

Fig. 9 Encapsulated Ice—Spherical Container
(Courtesy Cryogel)

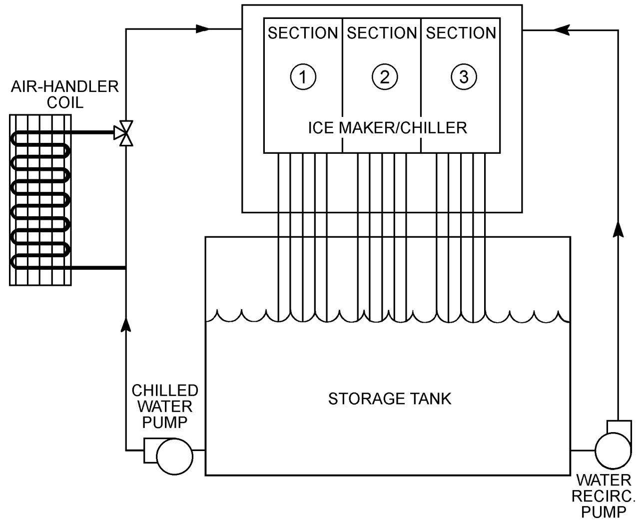

Fig. 10 Ice-Harvesting Schematic
(Courtesy Paul Mueller Company)

building load is present, the return chilled water flows directly over the evaporator surface, and the ice generator functions either as a chiller or as both an ice generator and a chiller. Cooling capacity as a chiller is a function of the water velocity on the evaporator surface and the entering water temperature. The defrost cycle must be energized any time the exit water from the evaporator is within a few degrees of freezing. When operated as a chiller only, maximum performance is obtained with minimum water flow and highest entering water temperature. In load-shifting applications, the compressors are turned off during the electric utility on-peak period.

Positive-displacement compressors are usually used with ice harvesters, and saturated suction temperatures are usually between 18 and 22°F. The condensing temperature should be kept as low as possible to reduce energy consumption. The minimum allowable condensing temperature depends on the type of refrigeration used and the defrost characteristics of the system. Several systems operating with evaporatively cooled condensers have operated with a compressor specific power consumption of 0.9 to 1.0 kW/ton (Knebel 1986, 1988a; Knebel and Houston 1989).

Ice-harvesting systems can melt the stored ice very quickly. Individual ice fragments are characteristically less than 6 by 6 by 0.25 in. and provide at least 152 ft^2 of surface area per ton-hour of ice stored. When properly wetted, a 24 h charge of ice can be melted in less than 30 min for emergency cooling demands.

During the ice generation mode, the system is energized if the ice is below the high ice level. A partial-storage system is energized only when the entering water is at or above a temperature that will permit chilling during the discharge mode; otherwise, the system is off, and the ice tank is discharged during the on-peak period to meet the load. The high ice level sensor can be mechanical, optical, or electronic. The entering water temperature thermostat is usually electronic.

When ice is floating in the tank, the water level will always be constant, so it is impossible to measure ice inventory by measuring water level. The following methods are generally used to determine the ice inventory:

Water Conductivity Method. As water freezes, dissolved solids are forced out of the ice into the liquid water, thus increasing their concentration in the water. Accurate ice inventory information can be maintained by measuring conductivity and recalibrating daily from the high ice level indicator.

Heat Balance Method. The cooling effect obtained from a system may be determined by measuring the power input to, and heat of rejection from, the compressor. The cooling load on the system is determined by measuring coolant flow and temperature. This may be accomplished by measuring the ice inventory and knowing the time to recharge the storage. The ice inventory is then determined by integrating cooling input minus load and calibrating daily from the high ice level sensor. A variant of the heat balance method is determined by running a heat balance on the compressor only, using performance data from the compressor manufacturer and measured load data (Knebel 1988b).

Optimal performance of ice harvesting may be achieved by recharging the ice storage tank over a maximum amount of time with minimum compressor capacity. Ice inventory measurement and the known recharge time are used to accomplish this goal. Knebel and Houston (1989) demonstrated that efficiency can be dramatically increased with proper selection of multiple compressors and unloading controls.

Design of the storage tank is important to system operation. The amount of ice stored in a storage tank depends on the shape of the storage, the location of the ice entrance to the tank, the angle of repose of the ice (between 15 and 30°, depending on the shape of the ice fragments), and the water level in the tank. If the water level is high, voids occur under the water because of the buoyancy of the ice. Ice-water slurries have been reported to have a porosity of 0.50 and typical storage densities of 2.92 $ft^3/ton \cdot h$). Gute et al. (1995), and Stewart et al. (1995a, 1995b) describe models for determining the amount of ice that can be stored in rectangular tanks and the discharge characteristics of various tank configurations. Dorgan and Elleson (1993) provide further information.

Ice Slurry Systems

Simply defined, an ice slurry is a suspension of ice crystals in liquid. In general, the working fluid's liquid state consists of a solvent (water) and a solute such as glycol, ethanol, or calcium carbonate. Depending on the specific slurry technology, the initial solute concentration varies from 2% to over 10% by mass. The solute depresses the freezing point of the solvent and buffers the production of ice crystals.

Slurry generation begins by lowering the working fluid to its initial freezing point. Extracting additional energy from the working fluid initiates the process of solidification. As solidification proceeds, solute is rejected to the solid-liquid interface (only the water is frozen). The solute-rich interface is eventually incorporated into the bulk fluid by convection and diffusion. As freezing progresses, the solute in the bulk field increases and the freezing point needed to sustain ice crystal production decreases.

Other Phase-Change Materials

Like ice and chilled-water storage, hydrated salts have been in use for many decades. PCMs with various phase-change points have been developed. To date, the hydrated salt most commonly used for cool storage applications changes phase at 47°F; it is often encapsulated in plastic containers, as described in the section on Encapsulated Ice. This material is a mixture of inorganic salts, water, and nucleating and stabilizing agents. It has a latent heat of fusion of 41 Btu/lb and a density of 93 lb/ft^3. The PCM's latent heat of fusion requires a capacity of about 5.5 $ft^3/ton \cdot h$ for the entire tank assembly, including piping headers and water in the tank.

Another option is an internal melt ice-on-coil system in which the water is replaced by salts or polymers that change phase at 12 or 28°F. The hydrated salts typically have a latent heat of fusion of 115 to 135 Btu/lb and a density of 65 to 70.5 lb/ft^3.

Typically, 40 to 42°F chilled water is used to charge the storage tank of systems using the 47°F PCM. The leaving temperature remains a relatively constant 47°F until the hydrated salts complete the phase-change process, at which point the temperature of the water leaving the tank begins to approach that of the entering water. It is usually preferable to charge the last 10 to 20% of the tank's capacity while there is a cooling load because of the low (3 to 4°F) temperature difference across the tank.

During discharge, the exit temperature begins at 42°F and rises to 48 to 50°F. The usable storage capacity is determined by the highest discharge temperature that can be used by the system.

Any new or existing centrifugal, screw, or reciprocating chiller can be used to charge the storage because conditions are comparable to those for standard air conditioning. As a result, this technology is particularly appropriate for retrofit applications.

Precoolers with partial storage provide some storage and a colder water temperature to the load. The 47°F PCM can be used with the chiller downstream of the storage to supply water at 44°F or lower to the load.

Circuits for Ice Storage

Piping for ice storage can be configured in a variety of ways. Numerous studies have been conducted to determine optimum configurations and control schemes for a variety of designs (Simmonds 1994). The optimum configuration depends on the type of system; the operation, performance, temperature requirements; and the configuration of the building loop. The chiller may be located upstream or downstream of the building load, and the cooling plant may also be decoupled from the building load.

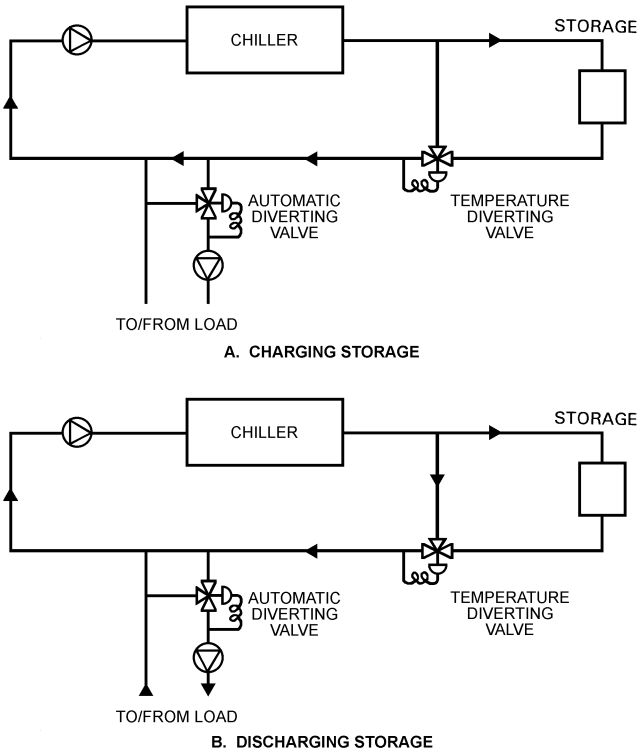

Fig. 11 Thermal Storage with Chiller Upstream

An example of a basic piping schematic for ice storage providing partial storage is shown in Figure 11. During off-peak periods, the loop supplying the building load is bypassed, and the chiller charges the ice storage unit (Figure 11A). When cooling is required during the on-peak period, the building load may be met by the chiller, the storage, or a combination of both (Figure 11B). A downstream modulating valve maintains the chilled-water supply to the building loop at the desired temperature. If demand limiting is desired, the chiller must be controlled in response to a demand limit set point. The remainder of the load is met by the storage.

Storage Tank Insulation

Because of the low temperature associated with ice storage, insulation is a high priority. In retrofit applications, the current insulation must be evaluated to ensure there is no condensation or excessive heat loss. All ice storage tanks located above ground should be insulated to limit standby losses. For external melt ice-on-coil systems and some internal melt ice-on-coil systems, the insulation and vapor barrier are part of the factory-supplied containers; most other storage tanks require that insulation and a vapor barrier be applied in the field. Below-ground tanks used with ice harvesters may not need insulation below the first few feet. Because the tank temperature does not drop below 32°F at any time, there is no danger of freezing and thawing groundwater.

All below-ground tanks using fluids below 32°F during the charge cycle should have a well-designed and properly installed insulation and vapor barrier, generally on the exterior. Interior insulation is susceptible to damage from the ice and should be avoided.

Because a hydrated salt solution operates at chilled-water temperatures of about 47°F, similar insulation practices apply.

ELECTRICALLY CHARGED HEAT STORAGE DEVICES

Thermal energy can also be stored in electrically charged, thermally discharged storage devices. For devices that use a solid mass

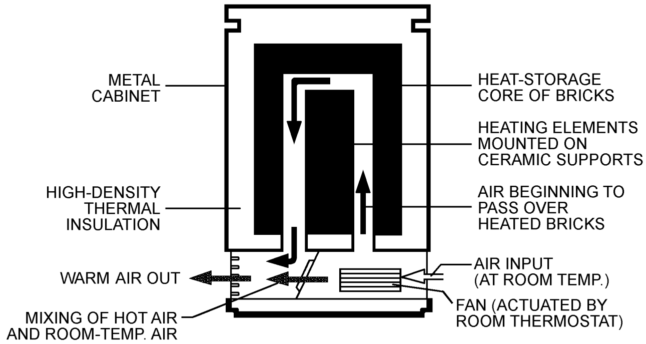

Fig. 12 Room Storage Heater
(Courtesy Steffes Corporation)

as the storage medium, equipment size is typically specified by the nominal power rating (to the nearest kilowatt) of the internal heating elements. The nominal storage capacity is taken as the amount of energy supplied during an 8 h charge period. For example, a 5 kW heater would have a nominal storage capacity of 40 kWh. ASHRAE *Standard* 94.2 describes methods for testing these devices. If multiple charge/off-peak periods are available during a 24 h period, an alternative method yields a more accurate estimate of equipment size. This method considers not only the nominal power rating, but also fan discharge rate and storage capacity. The equipment manufacturer should be consulted for more information on calculating the capacity of these devices.

Room Storage Heaters (Room Units)

Room storage heaters (commonly called room units) have magnetite or magnesite brick cores encased in shallow metal cabinets (Figure 12). The core can be heated to 1400°F during off-peak hours by resistance heating elements located throughout the cabinet. Room units are generally small heaters that are placed in a particular area or room. These heaters have well-insulated storage cavities, which help retain the heat in the brick cavity. Even though the brick inside the units get very hot, the outside of the heater is relatively cool, with surface temperatures generally below 180°F. Storage heaters are discharged by natural convection, radiation, and conduction (static heaters) or by a fan. The air flowing through the core is mixed with room air to limit the outlet air temperature to a comfortable range.

Storage capacities range from 13.5 to 60 kWh. Inputs range from 0.8 to 10.8 kW. In the United States, 120 V, 208 V, 240 V, and 277 V units are commonly available. The 120 V model is useful for heating smaller areas or in geographical areas with moderate heating days. Room storage heaters are for residential, motel, hotel, apartment, and office applications.

Operation is relatively simple. When a room thermostat calls for heat, fans (on dynamic units) located in the lower section of the room unit discharge air through the ceramic brick core and into the room. Depending on the charge level of the brick core, a small amount of radiant heat may also be delivered from the room unit. The amount of heat stored in the brick core of the unit can be regulated either manually or automatically in relation to the outside temperature.

These units fully charge in about 7 h (Figure 13), and they can be fully depleted in as little as 5 h. The equipment retains heat for up to 72 h if it has no fan discharge (Figure 14).

Choosing the appropriate size of room unit(s) depends on the rate structure of the power company (on-peak versus off-peak hours), outside design climate, and heat loss of the area or space. The manufacturer of the equipment may provide assistance in determining the heat loss for the area requiring heat. Based on the rate structure of the power company, the following two concepts can be used for sizing of the equipment:

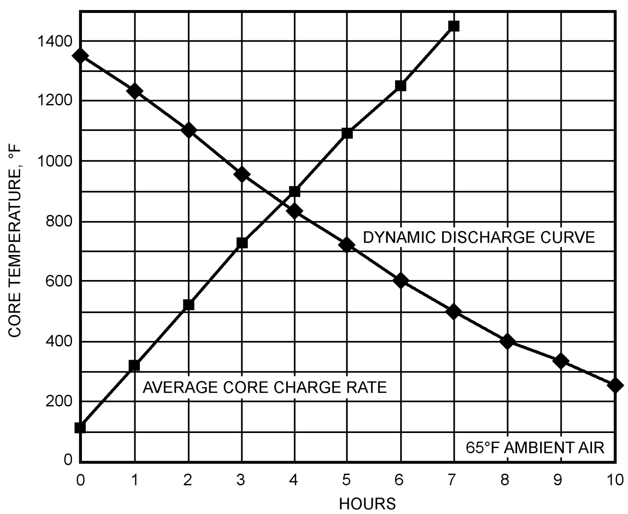

Fig. 13 Room Storage Heater Dynamic Discharge and Charge Curves

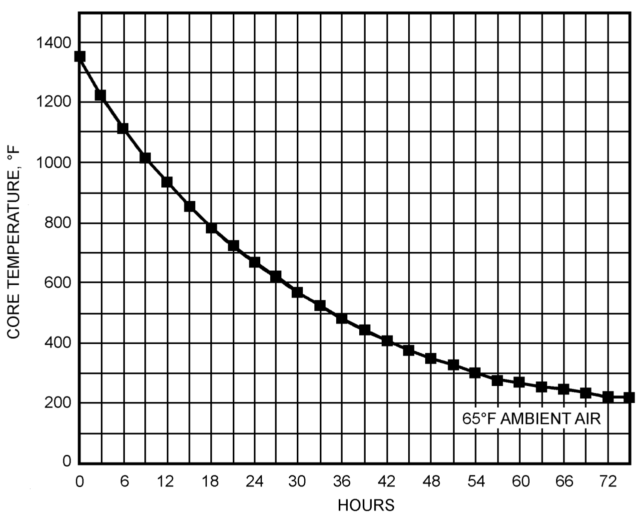

Fig. 14 Static Discharge from Room Storage Heater

Whole-House Concept. Under this strategy, room units are placed throughout the home. A room-by-room heat loss calculation must be performed. This method is used in areas where the power company has long on-peak hours, generally 10 h or more.

Warm-Room Concept. Under this strategy, one or two room units are generally used as the primary heating source during the on-peak periods. The units are placed in the area most often occupied (main area). Adjacent areas generally are kept cooler and have no operable heat during the on-peak time; however, some heat migrates from the main area.

When determining the heat loss of the main area, an additional sizing factor of approximately 25% should be added to allow for migration of heat to adjacent areas. Under the warm-room concept, sizing factors vary depending on the rate structure of the power company and the performance of the equipment under those conditions. The designer should consult the manufacturer of the equipment for specific sizing information.

The warm-room concept is the most common method used by power companies for their load management and off-peak marketing programs. It is successful in areas that have a small number of consecutive hours of control (generally less than 10 h of on-peak time), or have a midday block of off-peak time during which the equipment can recharge. The advantage of the warm-room concept is that it requires smaller sizes and quantity of equipment than the whole-house concept.

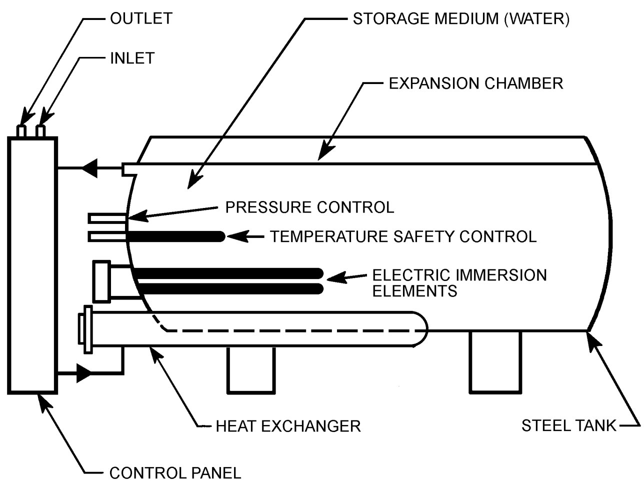

Fig. 15 Pressurized Water Heater

Heat Pump Booster

Air-to-air heat pumps generally perform well when outside temperatures are relatively warm. However, as outside temperature drops, the efficiency, output capacity, and temperature from a heat pump also decline. When the output of the heat pump drops below the heat loss of the structure, supplemental heat (typically from an electric resistance unit) must be added to maintain comfort. To eliminate the use of electric resistance heat during peak periods, storage heaters such as the heat pump booster (HPB) can be used to supplement the output of a heat pump. The HPB can also be used as a booster for stand-alone furnaces or to back up electric or fossil fuel equipment in dual-fuel programs.

Core charging of the HPB is regulated automatically based on outdoor temperature. The brick storage core is well insulated so that radiant or static heat discharge is small. Equipment input power ratings range from 14 to 38.4 kW. Storage capacities range from 86 to 240 kWh.

Central Furnace

The central storage furnace is a centrally ducted, heat storage product for residential and small commercial and industrial applications. These units are available with input ratings ranging from 14 to 38.4 kW. Storage capacities range from 180 to 240 kWh.

Pressurized Water Storage Heaters

This storage device consists of an insulated cylindrical steel tank containing immersion electrical resistance elements near the bottom of the tank and a water-to-water heat exchanger near the top (Figure 15). During off-peak periods, the resistance elements are sequentially energized until the storage water reaches a maximum temperature of 280°F, corresponding to 50 psig. The *ASME Boiler Code* considers such vessels unfired pressure vessels, so they are not required to meet the provisions for fired vessels. The heaters are controlled by a pressure sensor, which eliminates problems that could be caused by unequal temperature distributions. A thermal controller gives high-limit temperature protection. Heat is withdrawn from storage by running service water through the heat exchangers and a tempering device that controls the output temperature to a predetermined level. The storage capacity of the device is the sensible heat of water between 10°F above the desired output water temperature and 280°F. The output water can be used for

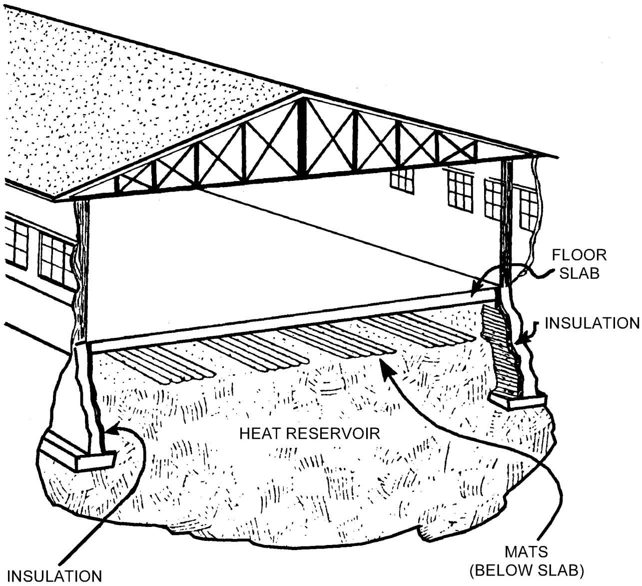

Fig. 16 Underfloor Heat Storage

space heating or service hot water. The water in the storage tank is permanently treated and sealed, requires no makeup, and does not interact with the service water.

Underfloor Heat Storage

This storage method typically uses electric resistance cables or hydronic tubing buried in a bed of sand 1 to 3 ft below the floor of a building. Underfloor heat storage is suitable for single-story buildings, such as residences, churches, offices, factories, and warehouses. An underfloor storage heater acts as a flywheel; although it is charged only during the nightly off-peak period, it maintains the top of the floor slab at a constant temperature slightly higher than the desired space temperature. Because the cables spread heat in all directions, they do not have to cover the entire slab area. For most buildings, a cable or tube location of 18 in. below the floor elevation is optimum. The sand bed should be insulated along its perimeter with 2 in. of rigid, closed-cell foam insulation to a depth of 4 ft (see Figure 16). Even with a well-designed and well-constructed underfloor storage, 10% or more of the input heat may be lost to the ground.

BUILDING MASS

Building Mass Effects

The thermal storage capabilities inherent in building mass can have a significant effect on the temperature within the space as well as on the performance and operation of the HVAC system. Effective use of structural mass for thermal storage reduces building energy consumption and reduces and delays peak heating and cooling loads (Braun 1990). In some cases, it improves comfort (Morris et al. 1994; Simmonds 1991). Perhaps the best-known use of thermal mass to reduce energy consumption is in buildings that include passive solar techniques (Balcomb 1983).

Cooling energy can be reduced by precooling the structure at night using ventilation air. Braun (1990), Ruud et al. (1990), and Andresen and Brandemuehl (1992) suggested that mechanical precooling of a building can reduce and delay peak cooling demand; Simmonds (1991) suggested that the correct building configuration may even eliminate the need for a cooling plant. Mechanical precooling may require more energy use; however, the reduction in electrical demand costs may give lower overall energy costs. Moreover, the installed capacity of air-conditioning equipment may also be reduced, providing lower installation costs.

The effective use of thermal mass can be considered incidental and be allowed for in the heating or cooling design, or it may be considered intentional and form an integral part of the design. The effective use of building structural mass for thermal energy storage depends on such factors as (1) the physical characteristics of the structure, (2) the dynamic nature of the building loads, (3) the coupling between the mass and zone air (Akbari et al. 1986), and (4) the strategies for charging and discharging the stored thermal energy. Some buildings, such as frame buildings with no interior mass, are inappropriate for thermal storage. Many other physical characteristics of a building or an individual zone, such as carpeting, ceiling plenums, interior partitions, and furnishings, affect thermal storage and the coupling of the building with zone air.

Incidental Thermal Mass Effects. A greater amount of thermal energy must be removed or added to bring a room in a heavyweight building to a suitable condition before occupancy than for a similar lightweight building. Therefore, the system must either start conditioning the spaces earlier or operate at a greater output. During the occupied period, a heavyweight building requires a smaller output, as a larger proportion of heat gains or losses is absorbed by the thermal mass.

Advantage can be taken of these effects if low-cost electrical energy is available during the night; the air-conditioning system can be operated during this period to precool the building. This can reduce both the peak and total energy required during the following day (Andresen and Brandemuehl 1992; Braun 1990) but may not always be energy-efficient.

Intentional Thermal Mass Effects. To make best use of thermal mass, the building should be designed with this objective in mind. Intentional use of the thermal mass can be either passive or active. Passive solar heating is a common application that uses the thermal mass of the building to provide warmth outside the sunlit period. This effect is discussed in further detail in Chapter 33. Passive cooling applies the same principles to limit the temperature rise during the day. The spaces can be naturally ventilated overnight to absorb surplus heat from the building mass. This technique works well in moderate climates with a wide diurnal temperature swing and low relative humidities, but it is limited by the lack of control over the cooling rate.

Active systems overcome some of the disadvantages of passive systems by using (1) mechanical power to help heat and cool the building and (2) appropriate controls to limit the output during the release or discharge period.

Systems

Both night ventilation and precooling have limitations. The amount of heat stored in a slab equals the product of mass, specific heat, and temperature rise. The amount of heat available to the space depends on the rate at which heat can be transferred between the slab and the surrounding spaces, which in simple terms is

$$q_s = \rho c_p V \frac{dt_s}{d\theta} = h_o A(t_s - t) \tag{4}$$

where

q_s = rate of heat flow between slab and indoor space, Btu/h
ρ = density, lb/ft^3
c_p = specific heat, Btu/lb·°F
V = slab volume, ft^3
θ = time, h
h_o = combined radiant panel heat transfer coefficient (radiant and convective), Btu/h·ft^2·°F
A = surface area of slab, ft^2
t_s = effective surface temperature of slab, °F
t = indoor temperature of space, °F

Equation (4) also applies to transferring heat to the storage medium; although the potential is equivalent to $c_p V(t_s - t)$, the heat released during the daytime period is related to the transfer coefficients. Building transfer coefficients are quite low; for example, a typical value for room surfaces is 1.4 Btu/h·ft^2·°F, which is the maximum amount of energy that can be released.

Effective Storage Capacity. The total heat capacity (THC) (Ruud et al. 1990) is the maximum amount of thermal energy stored or released because of a uniform change in temperature Δt of the material and is given by

$$\text{THC} = \rho c_p V \Delta t \tag{5}$$

The diurnal heat capacity (DHC) is a measure of the thermal capacity of a building component exposed to periodically varying temperature.

Many factors must be considered when an energy source is time dependent. The minimum temperature occurs around dawn, which may be at the end of the off-peak tariff; the optimum charge period may run into the working day. Beginning the charge earlier may be less expensive but also less energy-efficient. In addition, the energy stored in the building mass is neither isolated nor insulated, so some energy is lost during charging; the amount of available free energy (cooler outside air when suitable and available) varies and must be balanced against the energy cost of mechanical power. As a result, there is a tradeoff that varies with time between the amount of free energy that can be stored and the power necessary for charging.

Because the cooling capacity is, in effect, embedded in the building thermal mass, conventional techniques of assessing the peak load cannot be used. Detailed weather records that show peaks over 3 to 5 day periods, as well as data on either side of the peaks, should be examined to ensure that (1) the temperature at which the building fabric is assumed to be before the peak period is realistic and (2) the consequences of running with an exhausted storage after peak are considered. This level of analysis can only be carried out effectively using a dynamic simulation program. Experience has shown that these programs should be used with a degree of caution, and the results should be compared with both experience and intuition.

Storage Charging and Discharging

The building mass can be charged (cooled or warmed) either indirectly or directly. Indirect charging is usually accomplished by heating or cooling either the bounded space or an adjacent void. Almost all passive and some active cooling systems are charged by cooling the space overnight (Arnold 1978). Most indirect active systems charge by ventilating the void beneath a raised floor (Crane 1991; Herman 1980). Where this is an intermediate floor, cooling can be radiated into the space below and convected from the floor void the following day. By varying the rate of ventilation through the floor void, the rate of discharge can be controlled. Proprietary floor slabs are commonly of the hollow-core type (Anderson et al. 1979; Willis and Wilkins 1993). The cores are continuous, but when used for thermal storage, they are plugged at each end, and holes are drilled to provide the proper airflow. Charging is carried out by circulating cool or warm air through the hollow cores and exhausting it to the room. Discharge can be controlled by a ducted switching unit that directs air through the slab or straight into the space.

A directly charged slab, used commonly for heating and occasionally for cooling, can be constructed with an embedded hydronic coil. Refer to Chapter 6 in the 2000 *ASHRAE Handbook—HVAC Systems and Equipment* for further design details. The temperature of the slab is only cycled 3 to 5°F to either side of the daily mean temperature of the slab. Consequently the technique can use very-low-grade free cooling (approximately 66°F) (Meierhans 1993) or low-grade heat rejected from condensers (approximately 82°F). In cooling applications the slab is used as a cool radiant ceiling, and for warming it is usually a heated floor. Little control is necessary because of the small temperature differences and the high heat capacity of the slab.

INSTALLATION, OPERATION, AND MAINTENANCE

The design professional must consider that almost all thermal storage systems require more space than nonstorage systems. Having selected a system, the designer must decide on the physical location; the piping interface to the air-conditioning equipment; and the water treatment, control, and optimization strategies to transfer theoretical benefits into realized benefits. The design must also be documented, the operators trained, and the performance verified (i.e., the system must be properly commissioned). Finally, the system must be properly maintained over its service life. For further information on operation and maintenance management, see Chapter 38.

SPECIAL REQUIREMENTS

The location and space required by a thermal storage system are functions of the type of storage and the architecture of the building and site. Building or site constraints often shift the selection from one option to another.

Chilled-Water Systems

Chilled-water systems are associated with large volume. As a result, many stratified chilled-water storage systems are located outdoors (such as in industrial plants or suburban campus locations). A tall tank is desirable for stratification, but a buried tank may be required for architectural or zoning reasons. Tanks are traditionally constructed of steel or prestressed concrete. A supplier who assumes full responsibility for the complete performance often constructs the tank at the site and installs the entire distribution system.

Ice-on-Coil Systems (External and Internal Melt)

Ice-on-coil systems are available in many configurations with differing space and installation requirements. Because of the wide variety available, these offer solutions to meet the unique requirements of many types of buildings.

Bare coils are available for installation in concrete cells, which are a part of the building structure. The bare steel coil concept can be used with direct cooling, in which the refrigerant is circulated through the coils, and the water is circulated over the coils to be chilled or frozen. This external melt system has very stringent installation requirements. Coil manufacturers do not normally design or furnish the tank, but they do provide design assistance, which covers distribution and air agitation design as well as side and end clearance requirements. These recommendations must be followed exactly to ensure success.

The bare coil concept can also be used with a secondary coolant to provide the cooling necessary to build the ice. In an internal melt configuration, the ice and water, which remain in the tank and are not circulated to the cooling system, cool the secondary coolant during discharge. This indirect chilling can also be used with an external melt discharge if it is not desirable to circulate the secondary coolant to the cooling load. Indirect chilling can use either steel or plastic tubes in the ice builder.

Coils with factory-furnished containers come in a variety of sizes and shapes. A suitable style can usually be found to fit the available space. Round plastic containers with plastic coils are available in several sizes. These are offered only in an internal melt configuration and can be aboveground or partially or completely buried.

Rectangular steel tanks are available with both steel and plastic pipe in a wide variety of sizes and capacities. Steel coil modules have the option of either internal or external melt. These steel tank systems are not normally buried. Each system comes prepackaged; installation requires only placement of the tank and proper piping connections. Any special support or insulation requirements of the manufacturer must be strictly followed.

Encapsulated Ice

Cylindrical steel tanks with encapsulated ice modules are also available. These offer yet another shape to fit available space. With proper precautions, these tanks can be installed below grade. Standards and recommendations for corrosion protection published by the Steel Pipe Institute and the National Association of Corrosion Engineers should be followed, as should the manufacturer's instructions. These systems are shipped unassembled. For rectangular tanks, the ice modules must be placed in the shell at the job site so that the secondary coolant is channeled through passages where the desired heat transfer will be achieved. Spherical ice modules are naturally arranged in the tank by the coolant flow in a random manner such that channeled flow is not a concern.

Ice Harvesting

Field-built concrete tanks are generally used with ice harvesting. The ice harvester manufacturer may assist in tank design and piping distribution in the tank. The tank may be completely or partially buried or installed above ground. Where the ground is dry and free of moving water, tanks have been buried without insulation. In this situation the ground temperature eventually stabilizes, and the heat loss becomes minimal. However, a minimum of 2 in. of closed-cell insulation should be applied to the external surface. Because the shifting ice creates strong dynamic forces, internal insulation should not be used except on the underside of the tank cover. In fact, only very rugged components should be placed in the tank; exit water distribution headers should be constructed of stainless steel or rugged plastic suitable for the cold temperatures encountered. PVC is not an acceptable material because of its extreme brittleness at the ice-water temperature.

As with chilled-water and hydrated salt PCM tanks, close attention to design and construction is critical to prevent leakage. Unlike a system where the manufacturer builds the tank and assumes responsibility for its integrity, an ice-harvesting system needs an on-site engineer familiar with concrete construction requirements to monitor each pour and to check all water stops and pipe seals. Unlined tanks that do not leak can be built. If liners are used, the ice equipment suppliers will provide assistance in determining a suitable type; the liner should be installed only by a qualified installer trained in the proper methods of installation by the liner maker.

The sizing and location of the ice openings are critical; the tank design engineer should check all framed openings against the certified drawings before the concrete is poured.

An ice harvester is generally installed by setting in place a prepackaged unit that includes the ice-making surface, the refrigerant piping, the refrigeration equipment, and, in some cases, the heat rejection equipment and prewired controls. To ensure proper ice harvest, the unit must be properly positioned relative to the drop opening. As the internal piping is not normally insulated, the drop opening should extend under the piping so that condensate drops into the tank. A grating below the piping is desirable. To prevent air or water leakage, gasketing between the unit frame and caulking must be installed in accordance with the manufacturer's instructions. External piping and power and control wiring complete the installation.

Other PCM Systems

Coolant normally flows horizontally in salt and polymeric systems, so the tanks tend to be shallower than the ideal chilled-water storage tank. The chilled-water supply to and return from the tank must be designed to distribute water uniformly through the tank to prevent channeling. Tanks are traditionally made of concrete. The system supplier normally designs the tank and its distribution system, builds the tank, and installs the salt solution containers.

SYSTEM INTERFACE

Open Systems

Chilled-water, salt and polymeric PCMs, external melt ice-on-coil, and ice-harvesting systems are all open (vented to the atmosphere) systems. Draindown of water from the system back into the storage tank(s) must be prevented by isolation valves, pressure-sustaining valves, or heat exchangers. Because of the potential for draindown, the open nature of the system, and the fact that the water being pumped may be saturated with air, the construction contractor must follow the piping details carefully to prevent pumping or piping problems.

Closed Systems

Closed systems normally circulate an aqueous secondary coolant (25 to 30% glycol solution) either directly to the cooling coils or to a heat exchanger interfaced with the chilled-water system. A domestic water makeup system should not be the automatic makeup to the secondary coolant system. An automatic makeup unit that pumps a premixed solution into the system is recommended, along with an alarm signal to the building automation system to indicate makeup operation. The secondary coolant must be an industrial solution (not automotive antifreeze) with inhibitors to protect the steel and copper found in the piping. The water should be deionized; portable deionizers can be rented and the solution can be mixed on-site. A calculation, backed up by metering the water as it is charged into the piping system for flushing, is needed to determine the specified concentration. Premixed coolant made with deionized water is also available, and tank truck delivery with direct pumping into the system is recommended on large systems. An accurate estimate of volume is required.

INSULATION

Because the chilled-water, secondary coolant, or refrigerant temperatures are generally 10 to 20°F below those found in nonstorage systems, special care must be taken to prevent damage. Although fiberglass or other open-cell insulation is theoretically suitable when supplied with an adequate vapor barrier, experience has shown that its success is highly dependent upon workmanship. Therefore, a two-layer, closed-cell material with staggered, carefully sealed joints is recommended. A thickness of 1.5 to 2 in. is normally adequate to prevent condensation in a normal room. Provisions must be made to ensure that the relative humidity in the equipment room is less than 80%. This can be done with heating or with cooling and dehumidification.

Special attention must be paid to pump and heat exchanger insulation covers. Valve stem, gage, and thermometer penetrations and extensions should be carefully sealed and insulated to prevent condensation. PVC covers over all insulation in the mechanical room improve appearance, provide limited protection, and are easily replaced if damaged. Insulation located outdoors should be protected by an aluminum jacket.

REFRIGERATION EQUIPMENT

The refrigeration system may be packaged chillers, field-built refrigeration, or refrigeration equipment furnished as a part of a package. The refrigeration system must be installed in accordance with the manufacturer's recommendations. Because of the high cost of refrigerant, refrigerant vapor detectors are suggested even for Class A1 refrigerants. Equipment rooms must be designed and installed to meet ASHRAE *Standard* 15. Relief valve lines should be monitored to detect valve weeping; any condensate that collects in the relief lines must be diverted and trapped so that it does not flow to the relief valve and eventually damage the seat.

WATER TREATMENT

Open Systems

Water treatment must be given close scrutiny in open systems. Although the evaporation and concentration of solids associated with cooling towers does not occur, the water may be saturated with air, so the corrosion potential is greater than in a closed system. Treatment against algae, scale, and corrosion must be provided. No matter what type of treatment is chosen (i.e., traditional chemical or nontraditional treatment), effective filtration (at least down to 24 μm) should be provided. To prevent damage, water treatment must be operational immediately following the cleaning procedure. Corrosion coupon assemblies should be included to monitor the effectiveness of the treatment. Water testing and service should be performed at least once a month by the water treatment supplier.

Closed Systems

The secondary coolant should be pretreated by the supplier. A complete analysis should be done annually. Monthly checks on the solution concentration should be made using a refractive indicator. Automotive-type testers are not suitable. For normal use, the solution should be good for many years without needing new inhibitors. However, provision should be made for the injection of new inhibitors through a shot feeder if recommended by the manufacturer. The need for filtering, whether it be the inclusion of a filtering system or filtering the water or solution before it enters the system, should be carefully considered. Combination filter feeders and corrosion coupon assemblies may be needed for monitoring the effect of the solution on copper and steel.

For a more detailed description of water treatment technologies that can be used for thermal storage systems, refer to Chapter 48, Water Treatment.

CONTROLS

A direct digital control system to monitor and control all of the equipment associated with the central plant is preferred. Monitoring of electrical energy use by all primary plant components, individually if possible but at least as a group, is strongly recommended. Monitoring of the refrigeration capacity produced by the refrigeration equipment, by direct measurement where possible or by manufacturer's capacity ratings related to suction and condensing pressures, should be incorporated. This ensures that a performance rating can be calculated for use in the commissioning process and reevaluated on an ongoing basis as a management tool to gage performance.

Optimization Software

Optimization software should be installed to obtain the best performance from the system. This software must be able to predict, monitor, and adjust to meet the load, as well as adapt to daily or weekly storage, full or partial storage, chiller or ice priority, and a wide variety of electric rate schedules.

IMPLEMENTATION AND COMMISSIONING

Elleson (1997) identified the following key steps to designing a cool storage system:

- Calculate an accurate load profile.
- Use an hourly operating profile to size and select equipment.
- Develop a detailed description of the control strategy.
- Produce a schematic diagram.
- Produce a statement of design intent.
- Use safety factors with care.
- Plan for performance monitoring.
- Produce complete design documents.
- Retain an experienced cool storage engineer to review design.

Chapters 38 and 42 and ASHRAE *Guidelines* 1 and 4 provide more specific information regarding desired design documentation and operator training.

Performance Verification

ASHRAE *Standard* 150-2000, Method of Testing the Performance of Cool Storage Systems, provides a detailed test method for accurately determining the ability of a system to meet a given cooling load. The standard requires certain documentation before a performance test is carried out, including the design load profile, the criteria for determining the fully charged and fully discharged conditions, and a schematic diagram of the system illustrating intended measuring points. The *Standard* defines procedures for determining the charge and discharge capacities of a storage tank and the total capacity and efficiency of the entire system.

The commissioning authority should verify performance and document all operating parameters. This information should be used to establish a database for future reference. Some of the performance data for various ice storage systems are as follows:

External Melt Ice-on-Coil Storage

- Evaporator and suction temperatures at start of ice build
- Evaporator and suction temperatures at end of ice build
- Ice thickness at end of ice build
- Time to build ice
- Efficiency at start versus theoretical efficiency
- Efficiency at end versus theoretical efficiency
- Refrigeration capacity based on published ratings (deviation can indicate refrigerant loss or surface fouling)

Internal Melt Ice-on-Coil Storage

- Secondary coolant temperature and suction temperature at start
- Secondary coolant temperature and suction temperature at end
- Secondary coolant flow rate
- Tank water level at start
- Tank water level at end
- Time to build ice
- Efficiency at start versus theoretical efficiency
- Efficiency at end versus theoretical efficiency
- Capacity based on measured flow rate, heat balance, and published rating

Ice-Harvesting

- Suction temperature at start
- Suction temperature at harvest
- Harvest time/condensing temperature
- Time from start to "bin full" signal
- Efficiency
- Tank water level at start
- Tank water level at "bin full" signal
- Capacity based on published rating

Although tank water level cannot be used as an indicator of the amount of ice in storage in a dynamic system, the water level at the end of the discharge cycle is a good indicator of system conditions. In systems with no gain or loss of water, the shutdown level should be consistent, and it can be used as a backup to determine when the bin is full for shutdown requirements. Conversely, a change in level at shutdown can indicate a water gain or loss.

Maintenance Requirements

Following the manufacturer's maintenance recommendations is essential to satisfactory long-term operation. These recommendations vary, but their objective is to maintain the refrigeration equipment, the refrigeration charge, the coolant circulation equipment, the ice builder surface, the water distribution equipment, adequate water treatment, and controls so that they continue to perform at the same level as when the system was commissioned. Monitoring ongoing performance against commissioned kilowatt-hours per ton-hour of cooling capacity delivered gives a continuing report of system performance.

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