ASHRAE Journal

Application Fundamentals Of Ice-Based Thermal Storage

By Brian Silvetti, P.E., Member ASHRAE

As might be expected, the recent turmoil in the electric power industry has focused considerable attention on thermal storage technologies. Commercial cooling is a major contributor to peak power demand, but it also represents one of the few areas where load management methods are practical, cost-effective and proven. A well designed thermal storage system will effectively and efficiently reduce electrical demand, exploit time-of-day rates and remain totally transparent to a building's occupants.

Although there is considerable variety in the types of available storage equipment, the majority of today's systems are chiller-based. In the case of ice storage systems, the chiller's secondary coolant is usually a 25% to 30% ethylene glycol/water solution. The coolant circulates through a heat exchanger that is submerged in a tank of water or through a tank packed with water filled containers.

In an "internal melt" system, the secondary coolant is used to both freeze (charge) and melt (discharge) the storage material (water). The water that is frozen never leaves the storage tank. In "external melt" equipment, the glycol coolant freezes the storage material, but unfrozen water surrounding the ice is used for discharge.

While most of this article is directed towards the design of internal melt systems, many of the principles are applicable to other types of storage equipment.

For a more complete and comprehensive discussion of different storage types and application techniques, the reader is referred to ASHRAE's *Design Guide for Cool Thermal Storage*.[1]

Component Sizing

While any portion of the cooling load can be served by thermal storage, the designer will typically be influenced by economic and practical factors that bound reasonable selection ranges.

What return on investment is acceptable to the customer? Are incentives or rebates available? Are there space or accessibility limitations? How is the utility rate structured? What are the occupancy and use characteristics of the application? What are the life-cycle costs of the equipment and the influences of seasonal changes and climate? Will operating or maintenance costs be a factor?

Figure 1 represents four different approaches to the same design day cooling load profile.

Our example building has a peak load of 1 ton (3.5 kW) with a total cooling requirement of 9.5 ton (33 kW) hours in a 12-hour cooling period. Consequently, chiller and storage requirements are presented on a "per ton of peak load" basis.

In a thermal storage system the building peak load (tons) no longer defines the required chiller capacity. Rather, the total integrated cooling load (ton-hours), must be met by the chiller over its entire operating period, with appropriate capacity adjustments for different conditions (*Equations 1, 2* and *3*).

For an ice storage system we commonly describe chiller capacity in two modes—a conventional daytime cooling capacity and a nighttime, ice-making capacity, which is typically 65% to 70% of the daytime value.

Note that "day" and "night" in this sense refers to the operating condition of the chiller and not necessarily the specific time of the day. Also, it is important to recognize that this is a capacity, and not an efficiency adjustment.

For each of the approaches we might consider, there is a minimum chiller capacity (*Equation 5*) that can supply all of the required cooling.

In simplified terms:

$$\text{total ton hours} = \text{chiller day capacity} + \text{chiller night capacity} \quad (1)$$

$$\text{chiller day capacity} = \text{chiller tons} \times \text{day hours} \quad (2)$$

About the Author

Brian Silvetti, P.E., is vice president, engineering, for Calmac Manufacturing, Englewood, N.J. He is secretary of ASHRAE Technical Committee 6.9, Thermal Storage.

$$\text{chiller night capacity} = \text{chiller tons} \times \text{derating} \times \text{night hours} \qquad (3)$$

$$\text{total ton hours} = \text{chiller tons (day hours} + \text{derating} \times \text{night hours)} \qquad (4)$$

$$\text{chiller tons} = \frac{\text{total ton hours}}{\text{day hours} + \text{derating} \times \text{night hours}} \qquad (5)$$

The minimum chiller is now defined in terms of its daytime capacity and the minimum storage capacity will equal the total ton hours less the daytime chiller contribution (*Equation 6*). Some approaches may use larger than minimum chillers that allow the use of either more or less storage, but the required storage capacity will still be accurate as long as the actual daytime chiller contribution is properly described.

$$\text{storage ton hours} = \text{total ton hours} - \text{chiller tons} \times \text{day hours} \qquad (6)$$

The simplest approach, in both selection and application, is "full storage." The chiller operates only during the 12-hour unoccupied period when there is no cooling load. All of the cooling is produced at the ice-making capacity, which we have estimated at 65% of the nominal value.

$$\text{chiller tons} = \frac{\text{9.5 total ton hours}}{\text{0 day hours} + \text{0.65 derating} \times \text{12 night hours}} = 1.2 \text{ tons} \qquad (7)$$

And, of course, the storage requirement is the entire 9.5 ton (33 kW) hours of design day cooling load. In this case, we see that the chiller is actually larger than the 1 ton (3.5 kW) that would have been needed in a non-storage application. If the cooling period was shorter, perhaps 10 hours, the chiller might calculate to approximately 0.9 tons (3 kW). However, we usually find that the chiller in a full storage application is approximately equal to the non-storage alternative. This is clearly the most expensive of our options and is most common where extended payback periods are acceptable or where incentives or rebates are offered. Recent developments in the cost of on-peak power, particularly during cooling intensive periods, have broadened the appeal of this approach.

In contrast to the full storage option, designers often elect a "partial storage" approach that reduces or minimizes installed chiller capacity. In this case, a fully loaded chiller operates continuously throughout a design cooling day. Application of the formula is identical, except that 12 hours of fully loaded daytime capacity would be included. Chiller tonnage is reduced to approximately 0.5 (1.7 kW) and the storage requirement drops to 3.75 ton (13.18 kW) hours. In fact, we often see chillers at 0.4 to 0.6 tons (1.4 to 2.1 kW) per peak load ton and storage capacities well under half the total ton hour cooling load. Due to the reduced chiller and storage capacities, there are many examples of partial storage systems that have been equal or less in cost than the conventional alternative.

These two approaches define the upper and lower bounds of chiller selection. As the chiller size is increased above the minimum, partial storage selection, we can apply larger storage capacities that eventually approach the demand avoidance of the full storage solution. Alternatively, larger than minimum chillers allow us to select reduced storage capacities to satisfy other design goals such as space restraints.

Figure 1 presents two other selection alternatives, although many others are possible. Systems are often designed with multiple chillers. The next approach incorporates this option in the selection procedure. Two chillers are operated at night to produce stored cooling, but only one runs during the daytime, on-peak, period. Two chillers, each 0.35 tons (1.2 kW) (0.7 tons per peak ton hour total) and 5.5 ton (19.3 kW) hours of storage are found to provide the entire cooling requirement with a 65% reduction in on-peak chiller demand.

In some areas of the country, parts of Florida, Texas and California for instance, utilities have established shorter on-peak periods, typically from noon to 6 p.m. These often are described as "window" rates. Because the ton-hours are considerably reduced for this compressed on-peak period, complete avoidance of on-peak chiller operation becomes economically viable.

When the simplified formula is applied, a minimum chiller size of 0.7 tons (2.5 kW) is calculated. However, the load profile reveals that this would require the installation of additional storage to meet some of the off-peak cooling load during hours 11 and 12. The chiller will normally be increased in capacity to handle the entire off-peak load. In this case, a 0.85 ton (3 kW) chiller is selected, but there is no increase in on-peak demand and storage is limited to 5.5 ton (19.3 kW) hours.

Each of these solutions is summarized in *Table 1*. The four design approaches satisfy different goals. The "full storage" option eliminates any chiller contribution to the on-peak demand and shifts most or all of the chiller energy to off-peak periods. "Partial storage" avoids half of the on-peak chiller demand but both chiller and storage capacities are well below half that required for full storage, minimizing initial investment.

Next, multiple chillers can be used to achieve an intermediate level of demand avoidance while enhancing redundancy. Sixty-five percent of the on-peak chiller demand is avoided, with equipment capacities only 40% to 45% greater than the minimum, partial storage, selection. And finally, where the on-peak period is of shorter duration, the entire on-peak chiller demand is eliminated with modest increases in equipment capacities. This approach is dependent on the available rate structure.

Since many designers will divide the chiller capacity into two machines, a final column has been added to illustrate the available cooling should a chiller fail for each case. All of the storage options provide more available cooling than the conventional system, except for the minimum partial storage selection. In this case, increasing total chiller capacity from 0.5 to 0.6 tons per peak ton will provide capacity equal to the non-storage approach, in the event of a chiller failure. Therefore, whatever redundancy the application calls for is easily accomplished with little or no change in design.

By properly adjusting chiller operating hours, numbers of chillers or applying derating factors, it is relatively simple to compare many different alternatives, in addition to those described earlier. Keep in mind that this approach is somewhat simplified and in some rare cases will provide incorrect results. The two most common instances are where a night load exceeds the ice-making capacity of the calculated chiller and secondly, where the calculated partial chiller size exceeds a daytime hourly cooling load, in other words, whenever our original assumptions of chiller contribution are incorrect. Manufacturer's selection programs should adjust for these cases.

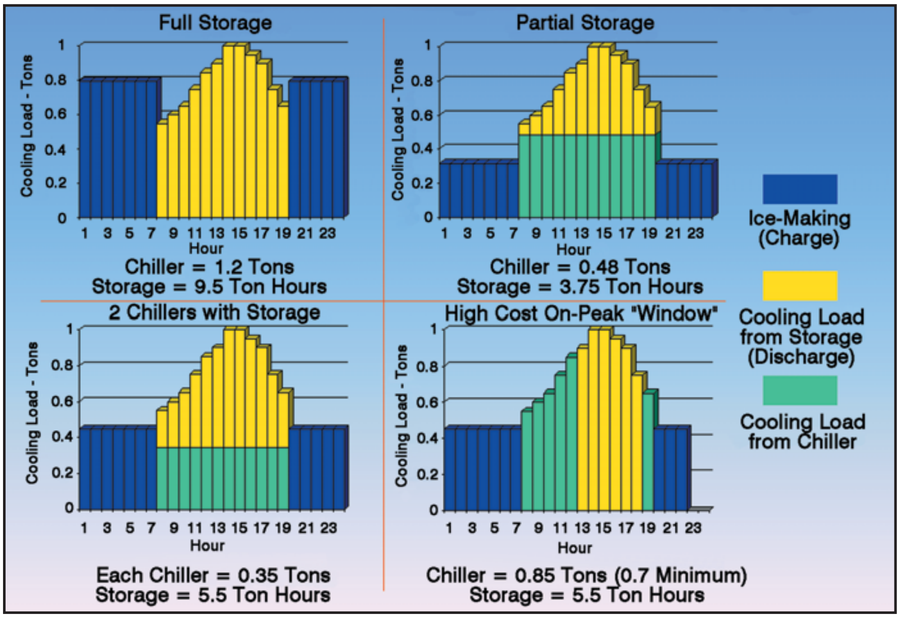

***Figure 1:** Chiller/storage selection (1 ton peak load, 9.5 total ton hours).*

Equipment Selection

Equipment must now be selected that will provide the necessary capacities. Thermal storage equipment is available in a range of designs, materials and configurations. Performance characteristics can vary significantly. Furthermore, ice storage systems are not steady state devices. In addition to the parameters that affect any heat exchanger, the critical physical dimensions for phase change thermal storage devices vary as storage material is frozen or melted.

High rates of discharge and/or lower temperatures are available early in the melting cycle when the ice surface is closest to the heat exchanger, with these capabilities diminishing as the ice surface recedes from the heat exchanger. This sometimes complex interaction of variable equipment performance with changing building load makes selection for discharge performance critically important.

Referring back to our example load profile, the worst case condition may be during a high load hour such as 15, or it may be later in the discharge where the loads are lower but the storage inventory has been reduced. Accordingly, the Air-Conditioning and Refrigeration Institute's Guideline T, *Specifying the Thermal Performance of Cool Storage Equipment,* requires that storage manufacturers provide hour-by-hour coolant temperatures for the specific equipment selection, load profile and chiller/storage arrangement, thereby guaranteeing adequate storage capacity throughout the design day.[2] Merely specifying ton hours of latent storage does not ensure that the offered equipment will adequately provide the desired performance.

Manufacturers have devised different methods of presenting performance information that is tailored to their particular product. *Figure 2* presents a segment of the discharge performance for one storage device with a constant coolant inlet temperature of 50°F (10°C).[3] The important relationship to recognize is the change in performance as storage inventory is expended, although the trends are predictable. This particular device is capable of providing 20 tons (70 kW) of discharge capacity with a leaving temperature of 44°F (6.7°C) and having expended 158 ton (556 kW) hours of stored cooling. If the discharge rate is increased to 30 tons (106 kW), the 44°F (6.7°C) leaving temperature will be exceeded if we attempt to discharge more than 126 ton (443 kW) hours. Likewise, if 40°F (4.4°C) coolant is required from storage at a 20 ton (70 kW) rate, 126 ton (443 kW) hours can be expected. As the rate of discharge is decreased, or the required leaving temperature is raised, the capacity of the storage equipment is increased. Most manufacturers have computerized the selection process so that each hour of the design day load profile is analyzed to ensure adequate storage capacity.

In determining a chiller's "charging" or ice-making performance, it is usually sufficient to establish the average chiller leaving temperature that the storage equipment requires. Again, charging performance, as reflected in chiller leaving temperatures, will gradually diminish as ice is formed. Because ice has almost four times the conductivity of liquid water, the variation in temperature over the cycle is usually fairly compact, although the behavior varies significantly with different equipment types.

Where centrifugals are contemplated, the minimum charging temperature should also be specified. This is the temperature of the coolant at the point the storage is fully charged. This is also useful for ensuring sufficient freeze protection for the secondary coolant. Most chiller manufacturers have become quite familiar with the use of their products in ice storage systems and some have incorporated additional logic in their controls to simplify the application.

System Configuration

Engineers have been very creative in combining chiller and

System Type	Chiller Tons	Storage Ton Hours	On-Peak Chiller	One Chiller Down Percent of Total Load
Conventional	1	0	1	63
Full Storage	1.2	9.5	0	125
Partial Storage	0.48	3.75	0.48	50
On-Peak Window	0.85	5.5	0	89
Two-Chiller Partial	2 × 0.35	5.5	0.35	72

Table 1: Design comparison, one-ton peak load.

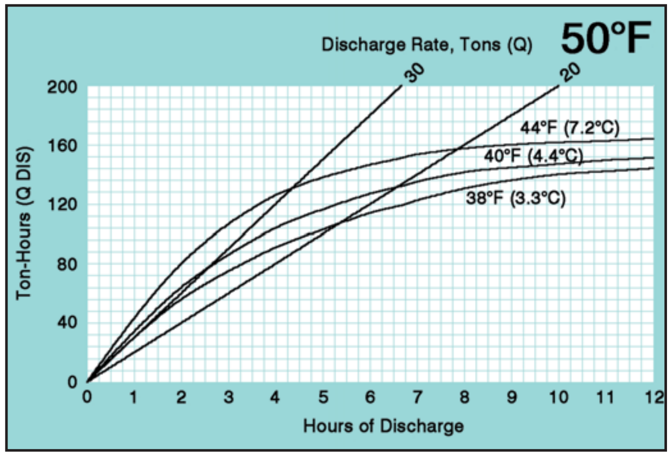

Figure 2: Sample discharge performance curve (50°F [10°C]) return temperature.

storage equipment in various arrangements to achieve a variety of goals. Rather than catalog all of the various configurations, inevitably omitting important ones, we will analyze a common arrangement that illustrates many of the essential application features. The system represented in *Figure 3* is commonly referred to as "series flow – chiller upstream." One advantage of placing the storage and chiller in series is that it does not require a change in flow path during the charging mode.

Furthermore, the manufacturer may recommend that flow through the storage equipment be in the same direction for charge and discharge. The series arrangement automatically accomplishes this while a parallel arrangement necessitates a change in flow path as the system cycles between charge and discharge. As "full storage" systems are fairly straightforward in selection and application, our focus will be on "partial storage" techniques, where controlling the contribution of chiller and storage are critical to system economy and comfort.

The first system characteristic of note is the wider differential in supply and return temperature, in this case 42°F (5.6°C) and 58°F (14.4°C). Partial storage systems use chiller capacities that are approximately half the peak load. It would be difficult to direct full system flow through the smaller chiller and storage at the more common 10°F (5.6°C) or 12°F (6.7°C) temperature range. Delta *T*s of 14°F (7.8°C) to 16°F (8.9°C) are fairly common with ranges of up to 20°F (11.1°C) often used. Flow rates are consequently lowered to levels compatible with the equipment, and pumping energy is reduced throughout the system. In the upstream position, the chiller often operates at higher daytime evaporator temperatures than it would have in the conventional system, although there may be a negative impact on storage capacity.

Reversing the arrangement retains all of the control flexibility as the storage modulating valve can be used to manage the relative contributions of storage and chiller. Storage capacity will be maximized at the expense of some chiller efficiency.

Before continuing, note two important features of the partial storage load profiles. First, even on the design day, there are hours that are less than peak load. And second, a conventional system chiller would unload during these hours, but the partial storage sizing calculations took advantage of the fact that we can keep our chiller fully loaded throughout the design day, minimizing the investment in equipment.

If we assume that our chiller capacity is about half of the peak load, the diagram represents system conditions at the peak load condition. The chiller reduces the return temperature by half the design Delta *T*, to 50°F (10°C), and storage reduces it further to the design supply temperature. However, if the chiller LCWT is simply set to 50°F (10°C), the chiller will unload any time the load is less than peak, as the return temperature decreases. This may shift cooling load to storage that should have been served by the chiller, resulting in premature depletion of storage capacity and the inability to meet the cooling load later in the day.

Alternatively, by setting our chiller LCWT at 42°F (5.6°C), the chiller will meet all cooling load up to its capacity, before any load is imposed on storage. In some cases, this can be the extent of the discharge control logic. In fact, through simple adjustment of chiller temperature setpoint, cooling load can be shifted between chiller and storage in any desired proportion in order to best exploit the electric rate in response to daily or seasonal load changes.

As the load and chiller contribution varies, the storage three-way modulating valve will automatically direct sufficient flow through the storage system to maintain 42°F (5.6°C) coolant delivered to the load.

On the design day, the operating logic is usually predetermined and obvious. The challenge in maximizing savings usually occurs on days with reduced load, which of course, comprise most of the operating hours. Control schemes can be as simple or complex as desired, consistent with the technical capabilities of operators, the utility rate and building load patterns. Very effective control schemes have been as straightforward as "hot day/mild day/cool day." On a hot day the chiller is fully loaded, half loaded on a mild day and off on a cool day.

An increased level of complexity might attempt to limit demand for each billing period. There is a minimum chiller demand that can be predicted for any billing period, either by analysis of the cooling loads, experience or established by a demand ratchet from a previous month. Since there is no avoid-

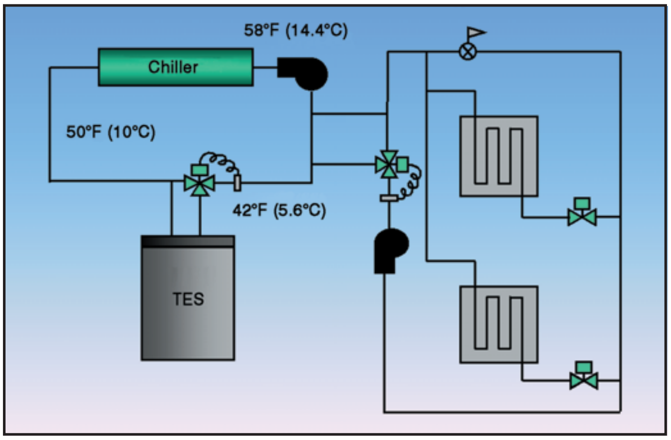

Figure 3: Series flow—chiller upstream.

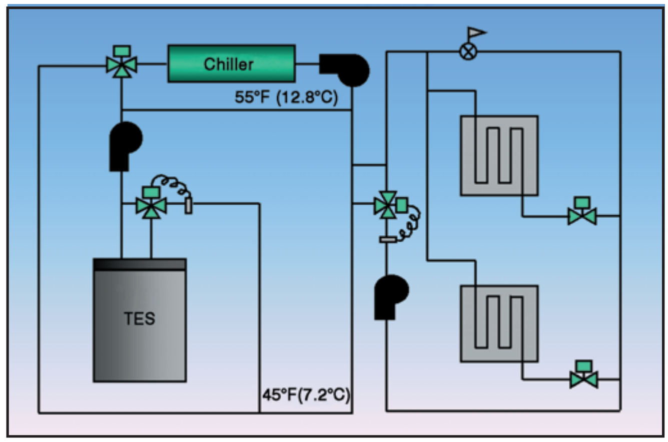

Figure 4: Parallel flow.

able demand penalty (kW) for chiller operation up to this level, the cost of energy (kWh) becomes the dominant influence when lower load days are addressed. If there is a significant differential in on and off-peak rates, further reduction in on-peak chiller operation is warranted, if practical. If the rates are approximately equal, chiller operation up to this pre-established limit carries no penalty. In either case, we have at least maximized demand savings. Even more complex methods are available that track storage inventory, cooling load, outdoor conditions etc., and will then modulate chiller loading to maximize savings under specific utility rates. In most cases, relatively simple control schemes are very effective. The challenge, of course, is to minimize operating cost while insuring adequate cooling capacity.

Certainly, there are times when a different configuration, such as a parallel flow system, is preferred. Perhaps the application is a retrofit with a fixed distribution Delta *T*. Consider how chillers in parallel load and unload in response to cooling needs. In storage applications it is essential that unloading of the chiller does not result in unanticipated depletion of storage. Referring to the simplified schematic of *Figure 4*, note that a two position, three-way valve at the chiller outlet is included to redirect flow for the charge and discharge modes.

During the discharge in parallel, the same return temperature fluid enters chiller and storage. With no other control, other than a fixed leaving temperature for both storage and chiller, the contribution of storage and chiller will be in a constant ratio as the return temperature varies. Keeping in mind that, even on a design day, the return temperature may be reduced during much of the day, it is apparent that the chiller will unload. If we have assumed full capacity from the chiller in our selection, the system will be undersized. Often, a designer will consider the control scheme at peak load, unaware that reduced return temperatures at part loads may inadvertently increase the load on storage resulting in premature depletion or limit the cooling capacity.

Solutions include but are not limited to, simple over-sizing of chiller and storage equipment (usually by about 15%), manipulation of chiller leaving temperature or the use of variable flow in the primary loops. In any case, it becomes more awkward to optimize the sharing of load between the chiller and storage. Rather than attempt to detail specific examples, the best advice, regardless of system configuration, is to apply the proposed control logic at a variety of cooling loads, calculate flows and temperatures, and insure that the results are consistent with the assumptions made during equipment selection.

The role of the three-way storage valve warrants some explanation. The valve responds to two separate system characteristics. The first is the variation in the required contribution from storage as the building load ramps up and down and the chiller capacity varies. Second is the storage system's variable performance. Referring back to *Figure 2*, the temperature of the coolant exiting the storage device is a function of flow, inlet temperature and ice inventory. The temperature modulating valve automatically adjusts to compensate for all of these effects, in addition to providing isolation of storage when necessary.

Focusing now on the charge mode, the series system allows us to simply position the storage three-way valve so that all flow is directed through the storage tank and reset the chiller temperatures. Where the chiller is operated at conventional daytime, as well as ice-making temperatures, the manufacturer may require that the chiller be fully loaded while in the ice-making mode. The storage charging range must therefore be consistent with the chiller capacity, or multiple chillers may be indicated. In most cases, storage equipment can be operated over a broad charging range and this only becomes a concern where the chiller capacity is very large in comparison to the storage sizing or perhaps where there are substantial night loads. The basic full and partial storage calculations almost always result in well-balanced selections.

It is often necessary to serve a cooling load during the charging mode. Actually, many designers consider the ability to efficiently meet small night, or even winter loads, one of the major advantages of storage systems. Obviously, the temperatures of the coolant circulating in the primary loop during the charge mode are below 32°F (0°C) and considerably lower than what is normally delivered to the secondary load loop. While a separate chiller operating at a higher temperature can be used, an additional three-way valve is often placed in the secondary loop so warm return fluid can be used to temper the coolant delivered to the load. This is critically important where pure water load loops are served through a heat exchanger. If no night loads (i.e., during ice-making) are anticipated, this valve becomes redundant as all daytime temperature control can be accomplished with the chiller or storage three-way valve.

If the system is designed to serve night loads, we must consider what happens if those loads may not be present. All of the chiller capacity would then be directed only to storage and the relative chiller to storage sizing must be evaluated as referred to previously. This is usually only an issue when the night load is a substantial percentage of the daytime peak, and a separate chiller may be preferred.

Once the storage is fully charged, the chiller can be reset to normal temperatures and the storage valve positioned so that all flow will bypass the storage equipment. Any loads can be met conventionally until it is time to discharge the storage. The completion of the charging mode will be indicated by a specific coolant temperature limit or, depending on the manufacturer, some type of inventory indicator.

If the storage has not been depleted, there may be an advantage in delaying ice-making until a cooler or lower cost period. In most cases ice-making is simply completed as quickly as possible with a fully loaded chiller.

Summary

The first step in thermal storage design is to establish an accurate design day cooling load profile. Rather than peak load, total ton hours of cooling load determine chiller and storage capacities. The procedure essentially equates full load chiller operating hours to the total cooling load. Chiller selection will be bounded by "full" or "partial" storage limits and simple economics and physical constraints will further define the basic approaches available to the designer. None of the design aspects are independent. Control logic must be developed to exploit the utility electrical rate and be consistent with the physical configuration of the equipment. The physical arrangement defines operating temperatures, which in turn, influence equipment capacities.

It was shown that a simple series system provides a straightforward means of implementing effective, efficient and exceptionally versatile control, although there are a wide variety of alternate arrangements.

Control can be designed to any level of complexity, while many utility rates are adequately served by uncomplicated logics. The designer must balance any benefits of added complexity with the technical sophistication of operating personnel.

Regardless of the design, control simulations should be applied over a range of cooling loads to insure that the assumptions made during equipment selection remain valid in practice. This is relatively simple for "full" storage systems, but "partial" storage designs demand greater scrutiny. A versatile storage design allows the system to shift load between chiller and storage to best exploit the utility rate. Accompanying this versatility is the responsibility to insure that loads are properly shared by chiller and storage under all conditions.

References

1. Dorgan, C.E., J.S. Elleson. 1994. *Design Guide for Cool Thermal Storage*. Atlanta: ASHRAE.

2. Air-Conditioning and Refrigeration Institute. 1994. Guideline T, *Specifying the Thermal Performance of Cool Storage Equipment.*

3. Levload® Ice Bank® Performance Manual-Form IB-102. 1995. Calmac Manufacturing.