COOL STORAGE PERFORMANCE

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ABSTRACT

Utilities have promoted the use of electric heat and thermal storage to increase off peak usage of power. High daytime demand charges and enticing discounts for off peak power have been used as economic incentives to promote thermal storage systems.

This article covers three thermal storage topics. The first section catalogs various thermal storage systems and applications. Included are: load shifting and load leveling, chilled water storage systems, and ice storage systems using Refrigerant 22 or ethylene glycol.

The second section discusses the result of system simulation computer studies (TRACE) that show the energy efficiencies of various cool storage methods. Results are shown as energy cost in dollars per square foot.

The last section addresses the need for advanced energy management control systems for use with cool storage systems and studies the success of such a system at a corporate engineering building for The Trane Company in La Crosse, Wisconsin.

SYSTEMS AND EQUIPMENT

Today the HVAC system designer is beset by a number of advocates of various systems all singing a litany of praises for the concept they promote. The fact that most of the praise is well founded only compounds the decision-making process. However, as we will see in the case of thermal storage, seemingly insignificant differences in design can have a profound impact on system performance and, ultimately, operating cost. Fortunately, today's information society provides the tools design professionals need to make the correct evaluation and even improve system performance.

We do know that several utilities are offering substantial discounts to a number of their customers and even waiving demand charges on the use of power. However, there is a catch; to qualify for these discounts, the power must be used at night. A number of utilities, for financial or capacity reasons, must increase their production of nuclear power. Since construction programs have been curtailed, several utilities are left with a surplus of power only during off-peak periods. By offering time-of-day rates, utilities pass the low cost of nuclear power on to their customers.

Time-of-day rates charge a premium during periods of high demand (on-peak) and offer reduced rates during periods of low demand (off-peak). The charge may be based on demand, usage, or both. Time-of-day rates have revived the concept of using thermal storage for cooling applications.

Thermal storage uses large volumes of water or ice in central or modular packages to store refrigeration produced during off-peak periods for later use during periods of peak demand. Thermal storage systems can be placed in two general categories: partial storage and total storage, Figure 1. Partial storage trims the peak refrigeration load only and requires some chiller operation during on-peak periods. The storage volumes are manageable and the demand savings are attractive. Total storage systems must produce the entire daily refrigeration load the evening before. No chiller operation is allowed during on-peak periods. Total storage systems require enormous thermal storage volumes and are usually attractive only where time-of-day rates offer substantial discounts for off-peak power.

Chilled water has been the preferred choice of storage medium. Chilled water storage systems have lower energy requirements when compared to conventional systems. Producing and storing chilled water requires no unusual hardware. However, chilled water storage demands a fair amount of space. Approximately 10 cubic feet of water are required to store one ton-hour of cooling. This storage volume can be reduced by a factor of two to three by using ice storage. Can this space savings justify the additional installed cost and higher energy cost of ice storage? Some recent innovations in packaging give the system designer some options when designing ice storage systems. A review of ice storage systems, past and present, would be helpful.

Ice is stored on the heat transfer surface that produces the ice. The size of the ice tank is then a function of the heat transfer efficiency and the arrangement of the heat transfer surface as well as the amount of cooling the designer wishes to store. The ice tank is a large insulated tank containing several feet of steel pipe. The ice is built up on the outside of the pipe while the refrigerant (R22) is circulated inside the pipe. When fully charged, two and a half inches of ice surround the pipe or 13 pounds of ice for each linear foot of pipe. This equates to five feet of steel pipe for every ton-hour of cooling stored.

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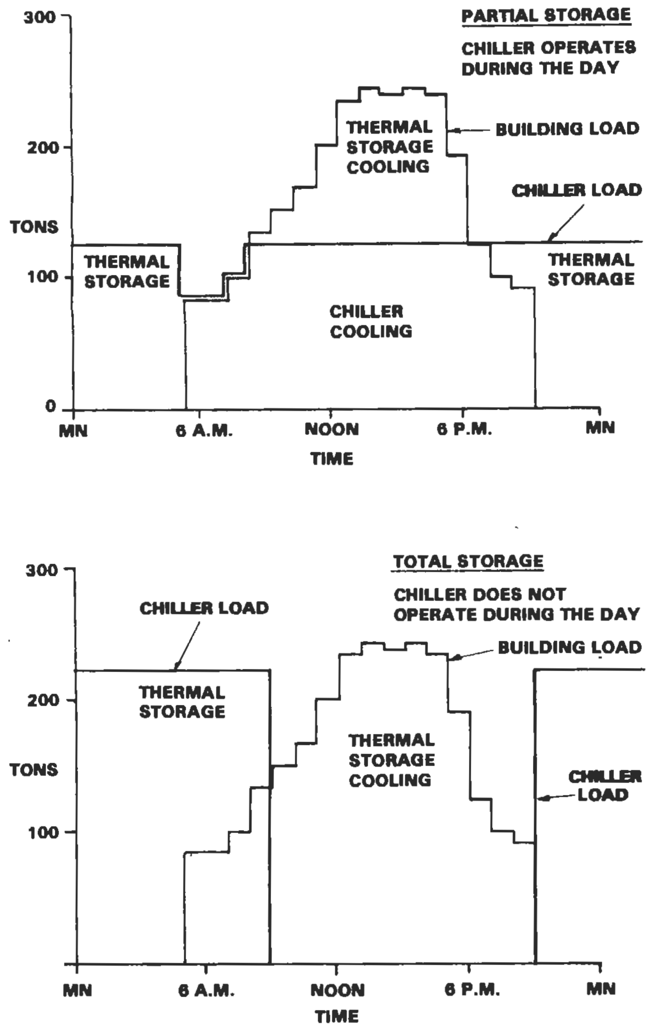
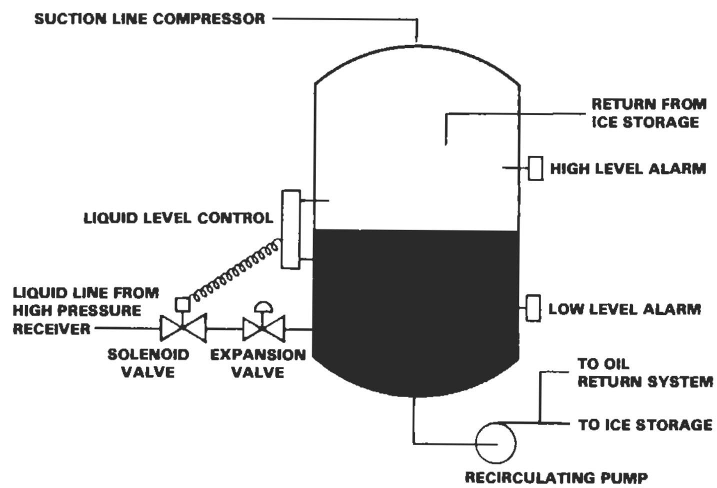


FIGURE 1

Liquid recirculation systems use either the compressor or a separate liquid refrigerant feed pump to circulate the refrigerant, Figure 2. The recirculation rate is two to three times the evaporation rate. This causes liquid overfeed systems, as they are sometimes called, to return a two-phase mixture from the ice tank to a low pressure receiver. From the low pressure receiver, refrigerant vapor is returned to the compressor while the refrigerant liquid is available for recirculation to the ice tank. The liquid level control meters additional refrigerant from the high pressure receiver to the low pressure receiver, to make up for refrigerant returned to the compressor. The design of the feed pump is critical in liquid recirculation systems. Proper design of the pumping system must be followed to prevent flashing of the refrigerant at the pump suction or in the pump itself. Semihermetic pumps or open pumps with specially designed seals are used to keep refrigerant losses to a minimum.



LOW PRESSURE RECEIVER

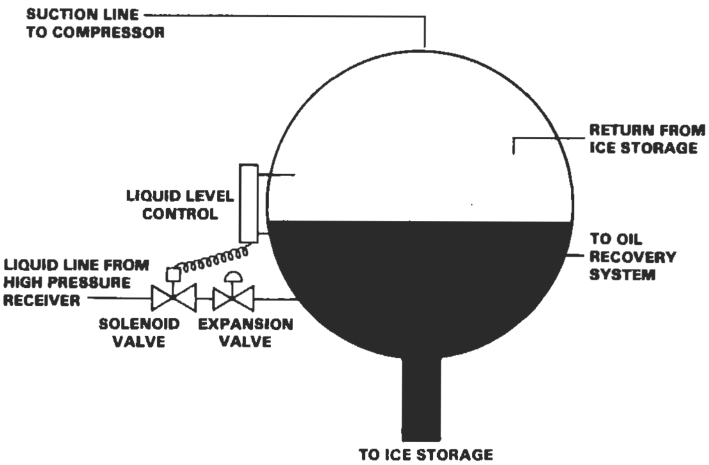
FIGURE 2

There are several methods to circulate the refrigerant within the tubes. Pumping large volumes of Refrigerant 22 is possible, but the pumping system must not allow the premature formation of flash gas in the system. Refrigerant 22 is circulated through the ice tank tubes by direct expansion, liquid recirculation or gravity (flooded systems).

Direct expansion employs the pressure difference between the high side receiver and the suction line accumulator to transport refrigerant through the ice tank. With direct expansion, proper sizing of the suction line will minimize oil return problems. From a design standpoint, direct expansion offers a simple and reliable system. However direct expansion suffers some inefficiencies compared to other systems. Fifteen to 20 percent of the tube surface must be used to provide superheat and is not available for making ice. This results in an installed cost penalty for additional surface area and/or lower efficiencies due to lower suction temperatures.

Pumping drums use hot gas to recirculate refrigerant, thereby eliminating the need for the refrigerant feed pump. The "pumper" is a smaller receiver that is located below the low pressure receiver. When vented to the low pressure receiver, liquid refrigerant drains by gravity into the pumper drum. The pumper is then isolated from the low pressure receiver by means of solenoid valves. Hot gas is used to pressurize the pumper forcing refrigerant from the pumper through the ice tank. Two pumping drums are used so that one is filling while the other is draining. The "double pumper" replaces the expensive and high maintenance liquid feed pump.

Gravity feed systems eliminate the need for pumping refrigerant altogether. With gravity systems, a low pressure receiver, or surge drum, is paired with each ice tank as a coil header. The vertical header carries the refrigerant to the ice tank inlet while a suction line returns refrigerant from the ice tank back to the surge drum. At the surge drum, vapor is returned to the compressor while liquid is recirculated to the ice tank. Due to the large refrigerant charge and the cost of multiple surge drums and controls, gravity flooded systems may be impractical in larger ice storage systems.



HORIZONTAL SURGE DRUM

FIGURE 3

Oil return is a special concern with all liquid recirculation systems. At the low pressure receiver, refrigerant vapor is drawn off by the compressor, but the oil remains in the receiver. When a liquid feed pump or pumper drum is used, sufficient pressure drop is available to employ an oil return system. The liquid line to the ice tank is tapped and a small amount of refrigerant is bled off to an expansion valve. The line downstream of the expansion valve is sized to maintain sufficient velocity to carry the entrained oil back to the compressor inlet. With a gravity flooded system, sufficient head is not available and a separate oil recovery system must be used. In addition to an oil return or oil recovery system, an oil separator in the compressor discharge is a must.

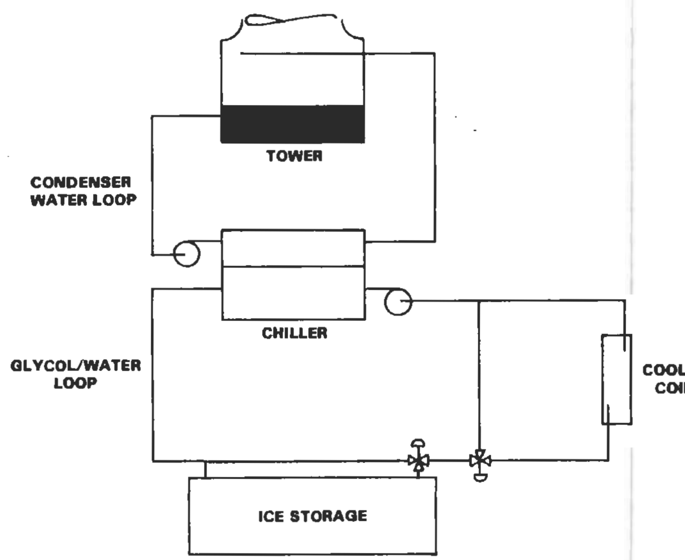
Finally, there are ice tank accessories. To maintain uniform ice thickness, the water in the ice tank must be thoroughly mixed. An air pump is used to agitate the ice tank water during both the freezing and melting cycles. During the ice melting cycle, chilled water must circulate around the ice covered pipes. An ice thickness probe is required to prevent the ice tank from overfreezing or "bridging." Bridges of ice forming between the ice covered pipes will impede the flow of chilled water through the ice bundle during the melting cycle, resulting in uneven ice melting.

Ice storage systems of this nature require very large refrigerant charges and incur substantial construction costs due to the number of field erected components. Due to these construction costs, many ice systems are difficult to justify except in larger tonnage applications. Recent innovations in the packaging of glycol type ice tanks may make the option of ice storage a greater possibility.

Glycol systems use an ethylene glycol/water mixture as a low temperature heat transfer medium to transfer heat from the ice storage tanks to a packaged chiller and from the cooling coils to either the ice storage tanks or the chiller. The use of freeze protected chilled water eliminates the design time, field construction, large refrigerant charges and leaks formerly associated with ice systems.

Glycol ice storage tanks create ice by circulating the low temperature fluid through half-inch polyethylene tubing. The polyethylene tubing is coiled inside insulated polystyrene tanks. The ethylene glycol/water mixture is also used for cooling by circulating the warm fluid through the tubing, melting the ice. The water that experiences a phase change remains in the tank. Since no water circulates around the tubing, the ice tank may be frozen solid. The problems of ice bridging and an air pump for agitation are eliminated. In this configuration the glycol ice tank is a sealed system similar to a packaged chiller or a car battery.

The heat transfer surface of the glycol ice tank can be increased by four to five times the area used in refrigerant ice tanks due to the low cost of polyethylene. The extended heat transfer area decreases the approach temperature required to make ice. Centrifugal or reciprocating chillers producing 23 to 26 degree ($^{\circ}$F) glycol are well suited for this application. Centrifugal chillers have an excellent track record in low temperature applications including food processing, cosmetics, pharmaceuticals, clean rooms, other industrial applications and, of course, ice rinks.



GLYCOL SYSTEM

ETHYLENE GLYCOL/WATER

FIGURE 4

PERFORMANCE

The performance of these cool storage systems can be compared by using currently available hour-by-hour computer simulation programs. This examination is based on a 150,000 square foot office building located in the Midwest. The initial HVAC design included a fan-powered variable air volume system and large tonnage, air-cooled DX equipment. A review of the local utility rates confirmed a high energy cost premium is charged for power during daytime hours.

DEMAND	$9.44/kw	
ELECTRICITY	$0.06/kwh	On Peak
	$0.03/kwh	Off Peak

On peak is defined as 9 am to 9 pm, Monday through Friday.

When the building owner expressed concern about energy cost, four alternative air conditioning systems were investigated as potential methods to reduce HVAC energy cost:

1. High efficiency water-cooled chillers
2. Chilled water storage
3. Ice storage employing a packaged chiller
4. Ice storage employing DX equipment

A computer analysis of the annual air conditioning load also hinted at the potential for thermal storage:

Design cooling load	350 tons
Refrigeration operating hours	1,862 hours
Annual refrigeration load	278,293 ton-hours
Average refrigeration load	149 tons (43%)

The average refrigeration load is not an indication of the required chiller size for thermal storage. It only indicates the operational diversity of the building. It does suggest the potential for energy cost savings with thermal storage. A lower average refrigeration load suggests that fewer tonhours of thermal storage are required to minimize the peak refrigeration load. An hourly load profile based on maximum daily cooling requirements is required to size the refrigeration equipment and thermal storage volume.

Ton-hours design day	3,371 ton-hours
Peak tons	316 tons
Hours of operation	16 hours

The system designer elected to design a partial storage system. By assigning only a portion of the daily cooling requirements to the thermal storage system, the required storage volume could be significantly reduced. The final design allowed only 1200 ton-hours of thermal storage. The 150 ton chiller selected could provide as much as 3600 ton-hours daily; more than enough to carry the maximum daily cooling requirement. The packaged chiller and the DX equipment for the ice storage systems were selected to provide 45 F chilled water during daytime operation as well as produce ice at night. The high efficiency chillers were selected at 45 F. The packaged chiller for the chilled water storage system was selected at 40 F. In addition to providing 1200 ton-hours of cooling, the thermal storage system must be able to deliver cooling at a 200 ton per hour rate and fully recharge in the allowable time period. It is the second requirement, recharge time, that can cause problems. This is particularly true with ice storage systems employing DX equipment.

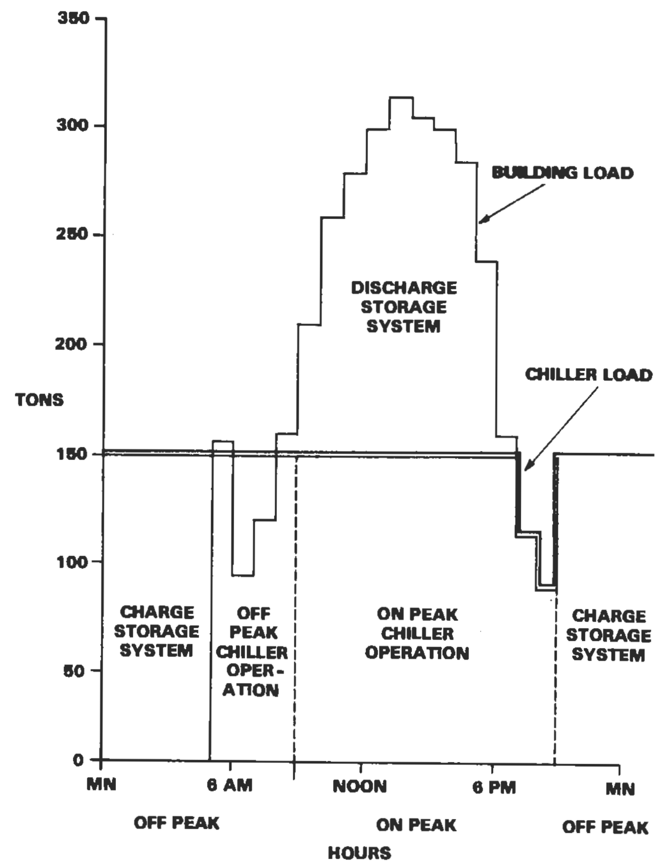


FIGURE 5

Building operation allows only eight hours to recharge the 1200 ton-hour thermal storage system, Figure 5. This should not be a problem as long as the chiller or compressor continues to deliver its nominal 150 tons of refrigeration. This is not possible with ice storage systems. Compressor capacity declines as the storage system recharges. Ice storage systems store ice on the same heat transfer surface that produces the ice. As the ice thickness increases, the heat transfer rate decreases due to the insulating qualities of ice. This reduction in heat transfer lowers suction temperature and, as a result, compressor capacity also declines, Table 1. Reciprocating compressor capacity was increased to 225 nominal tons to meet the recharge time requirement. This decline in chiller capacity also afflicts glycol ice storage systems. The centrifugal chiller produces 24 F leaving chilled water to recharge the ice storage system. The rated capacity of this same chiller at ARI conditions with the appropriate motor is 307 tons. However, with glycol ice storage systems, chiller capacity remains constant throughout the recharge cycle.

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Hour	Suction Temp	Tank Temperature /Ice Thickness	KW/ Ton	Compressor Capacity	Tonhours Stored
1	29.8	39.81	0.67	158.6	159
2	18.8	0.00	0.93	113.1	272
3	16.4	0.19	0.95	110.0	382
4	14.3	0.60	1.00	103.5	485
5	12.4	0.95	1.05	98.0	583
6	10.9	1.24	1.10	93.3	676
7	9.5	1.50	1.14	89.3	766
8	8.4	1.72	1.18	85.9	852
9	7.4	1.91	1.22	83.1	935
10	6.5	2.08	1.25	80.7	1015
11	5.8	2.22	1.28	78.7	1094
12	5.3	2.33	1.31	77.2	1171
13	4.8	2.43	1.33	76.0	1247

Table 1

The computer analysis provided some interesting but predictable results. Advocates of thermal storage tout reduction of peak demand and a reduction of annual energy costs as the primary benefits of thermal storage systems. Installed cost savings due to smaller refrigeration equipment is often cited as a secondary benefit. Designers less enthused about thermal storage point to the very poor system coefficient of performance (COP) due to the lower suction temperatures required and the high cost of thermal storage equipment and refrigerant accessories. With the notable exception of chilled water storage, none of the thermal storage systems received high marks for energy conservation, Table 2, although all four alternatives, including the high efficiency chiller without thermal storage, significantly reduced the peak demand and annual energy costs as promised. The final test: Does the annual energy cost savings justify the added capital cost of thermal storage? Or perhaps the more important question: Will careful analysis of each system lead to discovery of design techniques that may benefit any system design?

Location - Midwest System	Cooling Energy(MWH)	Building Energy Cost (Per Sq. Ft.)
Air-Cooled DX/VAV	417	$1.039
High-Efficiency Chillers	376	0.987
Chilled Water Storage	370	0.937
Ice Storage With A Packaged Chiller	456	0.983
Ice Storage With DX Equipment	477	0.983

Table 2

Consider the performance of chilled water storage. Combatting extended hours of pump operation, higher evaporator pressure drop due the large delta T and a slightly lower compressor COP, it performed within the ranks of the high efficiency chiller. Certainly rejecting heat at night during lower ambient temperatures improved performance of the chilled water storage system. But the major benefit is not chiller performance, it is system performance.

High efficiency chiller	
chiller	155,627 kwh
Pumps (excluding distribution)	52,139 kwh
Chilled water storage	
chiller	180,247 kwh
Pumps (excluding distribution)	26,771 kwh

Not always can accessory equipment enjoy the same part load performance as the central chiller. A conventional approach to producing chilled water required 840 gpm (350 tons and 2.4 gpm/ton). The chilled water storage system required only 180 gpm (150 tons and 1.2 gpm/ton). Condenser gpm was 3 gpm per ton in both cases. Distribution pumping requirements are a function of the load in a variable flow system and will be approximately the same for both systems. Chilled water system designs that emphasize chiller performance without adequately addressing accessory and system performance are incomplete. A second factor is lower average annual leaving chilled water temperature. Chilled water storage systems cannot benefit from chilled water reset while replenishing the storage volume.

The performance of the two ice storage systems differs dramatically, although the economic performance is essentially equal. A centrifugal chiller was suggested in the glycol ice storage system and it performed very well (0.84 kw/ton). This performance endures during the entire ice building cycle. The performance of the reciprocating compressors, although improved by the 95 F condensing temperature of evening operation, declines steadily as ice is produced. The only pumping power assigned directly to the production of ice in the DX system is condenser water and this can be reduced if the building operator is willing to maintain an evaporative condenser. However, pumping power is a major consumer in the glycol ice system. The combination of glycol at low temperatures and high water velocity to enhance heat transfer causes high pressure drops and additional pumping requirements.

Ice storage with packaged chiller	
Chiller	216,645 kwh
Pumps (excluding distribution)	76,086 kwh
Ice storage with DX equipment	
Compressors	289,020 kwh
Pumps (condenser only)	28,180 kwh

It is the compressor that is responsible for most of the mass transfer in the DX ice storage system. As suction temperature falls, the energy requirement to "pump" refrigerant steadily increases. In both ice storage systems, producing more ice than required can be counterproductive. With thermal storage systems, energy management is no longer a luxury, but an absolute necessity. Operating a building air conditioning system with substantially undersized equipment mandates careful planning. The penalty for errors can be several hours of loss of comfort or, worse, failure to realize potential energy cost savings. It may be misleading to draw conclusions from a single analysis of thermal storage. As seen, each thermal storage concept brings a different set of advantages and disadvantages. Many times, simply improving the performance of conventional systems can lead to significant operating cost savings with minimal increases in capital cost. Since accessory performance plays a major role in determining system performance, subtle changes in building operation or climate can produce dramatically different results. Obviously, utility charges play a major role in selecting the best cost performing system.

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Utility charges in the Southwest are higher and the mild climate of the region suggest an even greater potential for thermal storage. However that analysis draws completely different conclusions:

Demand	$7.30/kw
Electricity	$0.12/kw On Peak
	$0.09/kw Off Peak

Design cooling load	325 tons
Refrigeration operating hours	3687 hours
Annual refrigeration load	60,663 ton-hours
Average refrigeration load	98 tons (30%)

The results in Table 3 show different conclusions than drawn earlier. With the exception of chilled water storage, none of the lower energy systems could develop any substantial savings when compared to air-cooled DX equipment. Certainly extended hours of operation at low load and low ambient temperatures greatly benefitted DX equipment. However, the most significant difference is the utility rate. The demand charge is lower and a smaller discount for off peak power is offered (only 25percent compared to 50percent in the Midwest study).

Location - Southwest System	Cooling Energy (MWH)	Building Energy Cost (Per Sq. Ft.)
Air-Cooled DX/VAV	442	$1.551
High-Efficiency Chillers	473	1.559
Chilled Water Storage	415	1.493
Ice Storage With A Packaged Chiller	515	1.567
Ice Storage With DX Equipment	529	1.549

Table 3

AUTOMATION

The control of thermal storage systems and equipment mandates the uses of automated control systems. The complexity of controls should not exceed the complexity of the system. However, more than a time clock and a few extra thermostats are required to insure the maximum benefit of the thermal storage system. Control concepts can range from simple to comprehensive.

The simplest control is chiller priority. This control scheme allows the chiller to accept all of the cooling load until the load exceeds chiller capacity. Excess load is then met by melting ice. Control defaults to limiting chiller capacity by chiller selection or some form of demand limiting. Control is simple, but use of thermal storage is limited to cooling in excess of chiller capacity. This control may not provide the maximum financial return on the thermal storage investment.

There are several variations of ice priority control schemes. The intent of ice priority control is to gain maximum benefit of time-of-day rates by maximizing ice usage each day. During intermediate seasons when the daily cooling load is less than thermal storage capacity, chiller operation can be limited to off-peak hours. When daily cooling load exceeds thermal storage capacity, chiller and ice must share the on-peak cooling load. The task of assigning cooling load to either ice or chiller is well-suited for building automation systems with equation processing capabilities. If ice is used too early or too rapidly, the ice will be depleted before the end of the daytime cooling load, leaving several hours of building load in excess of chiller capacity. On the other hand, if the ice is not used to its maximum potential, the operating cost benefits of thermal storage are not totally realized.

How does the automation system predict the daily cooling load the day before? Most of the required tools are already in place. The automation system has the ability to monitor several building or load producing trends. What the automation system does not possess is the ability to equate these trends to building cooling load. Fortunately, several HVAC system simulation programs are available for this task. Simulation programs can isolate single load components and plot their dependence to an easily monitored variable, Figure 6. Internal loads are a function of time and calendar. Ventilation and solar loads can be equated to monitored weather variables. A daily cooling load profile can now be constructed by measurement of the monitored variables, profiling those variables for the period of prediction, and summing the results. Automation systems that can produce an expected daily load profile provide the building operator with a very powerful management tool. Only informed and prudent operation of thermal storage systems can guarantee their success.

SUMMARY

In the past, ice systems were used to reduce the installed tonnage of expensive refrigeration systems that were generally constructed in the field. The advent of the packaged chiller saw the end of ice storage systems in the commercial market. Today we see the seeds of a rejuvenation of ice systems. If the ice systems of today are to succeed, they must embrace the concept of packaging and advanced controls that the commercial HVAC market now enjoys.

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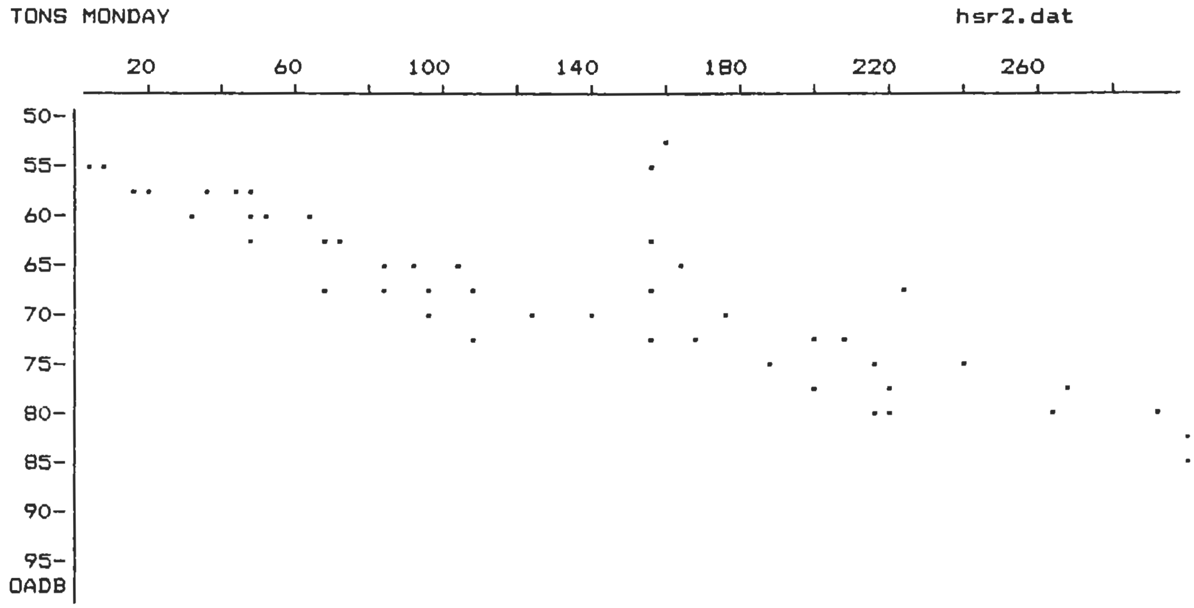
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Figure 6