Baltimore Aircoil Company

July 23, 2008 **Sent via email**

Ms. Kristi Izzo, Secretary
New Jersey Board of Public Utilities
Two Gateway Center
Newark, New Jersey 07102

Re: Written comments regarding New Jersey's Draft Energy Master Plan

Dear Secretary Izzo:

Baltimore Aircoil Company (BAC) applauds the efforts of the BPU, the Governor's office, and the numerous coordinated agencies to create an Energy Master Plan for the State. We appreciate the opportunity to submit comments for consideration and respectfully request that the BPU consider and include these comments in preparing the final Energy Master Plan.

Baltimore Aircoil Company the worldwide leader in the manufacture and sales of heat transfer and ice thermal storage products that conserve resources and respect the environment. Serving air conditioning, refrigeration, industrial, process, and power generation customers, BAC's product offerings include ice thermal storage systems. As noted in the draft EMP:

"The high price periods generally coincide with the peak periods and accordingly the importance of peak load management. Reduction of the peak load has a direct correlation in reducing the marginal cost of electricity. In addition, peak load reduction can play a major role in reducing harmful environmental emissions. One commonly used measure of performance or efficiency of electricity use is the load factor. Load Factor is calculated as a ratio of the average load to the peak load during the period. The peakier the peak the worse the load factor…."

These issues are the basis on which most thermal storage systems are justified and installed. We respectfully recommend that the final report include incentives for demand shifting technology, and specifically include ice thermal storage as one alternative.

The other key benefit with a fully integrated thermal storage system is that it can reduce overall energy consumption while providing peak load reduction. Attached to the email are two documents, one piece of BAC literature that discuss the overall concept and benefit of ice thermal storage, the second being an ASHRAE (American Society of Heating, Refrigeration and Air Conditioning Engineers) publication that discuss the energy benefits of such systems.

If the regulatory parameters are established to encourage recognized technologies, such as the ice storage systems, to be rapidly implemented in the State, our experience has been that the proponents of such proven technology will seek to become active partners as the State moves toward the challenge of meeting its energy goals and legislative mandates. BAC looks forward to offering its experience to commercial customers within New Jersey to help bring a marked reduction to peak energy demands by how the economics and reliability of an ice thermal storage system is truly a "win-win" for the customer and the state.

Please contact me if I can be of service or if you would like to discuss in more detail.

Sincerely,

Bill Dietrich

Bill Dietrich
Director of Sales and Product Marketing, Commercial and Industrial Products - Americas
Baltimore Aircoil Company
410-799-6237 direct
bdietrich@baltimoreaircoil.com

ICE CHILLER® Thermal Storage Products

Product Detail

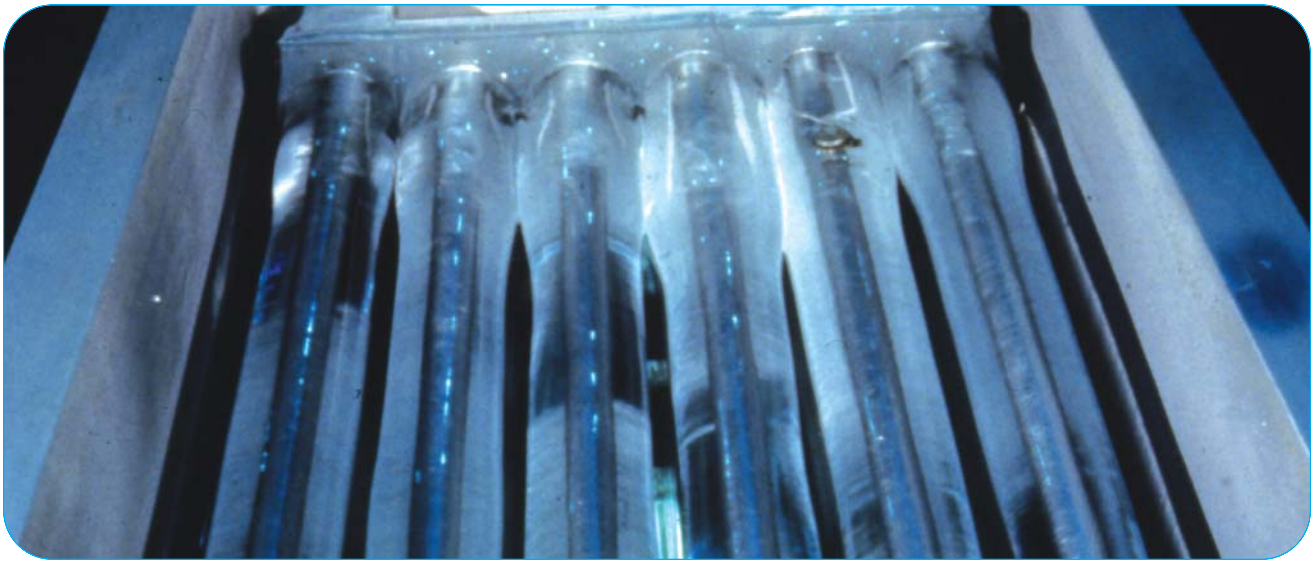
ICE CHILLER® Thermal Storage Products

Cooling with ice thermal storage can be the most cost-effective, reliable system approach to cooling offices, schools, hospitals, malls and other buildings, and provides a steady source of low temperature fluids for process cooling applications. These systems are environmentally friendly because they help lower energy consumption and reduce greenhouse gas emissions. With thousands of successful installations worldwide, BAC is the global leader in the application of ice thermal storage.

Ice Thermal Storage

- **Lowest first cost**
- **Reduced energy cost**
- **Variable capacity**
- **Improved system reliability**
- **Reduced maintenance**
- **Environmentally friendly**
- **Proven technology**


BAC



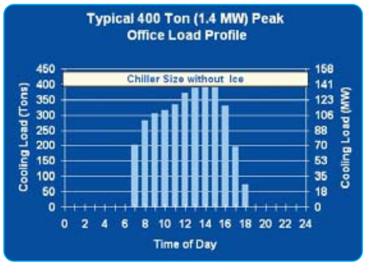

BAC

Benefits

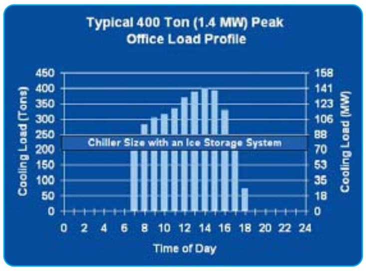
Lowest First Cost

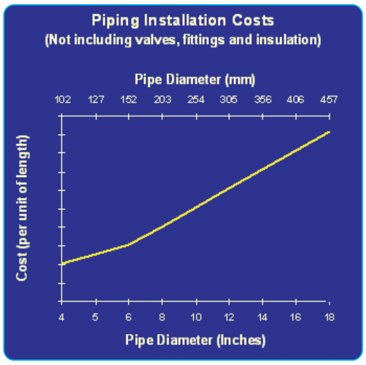
Systems with ice thermal storage can be installed at the same or lower first cost than traditional systems when designed with the colder supply water available from ice. The savings that result from the use of smaller chillers and cooling towers, reduced pump and pipe sizes, and less connected horsepower offset the cost of the ice thermal storage equipment. Additional savings can be found when using lower temperature air distribution, which allows for reduced ductwork and fan sizes.

- **Smaller Chillers and Heat Rejection Equipment:** By designing the system around 24-hour per day chiller operation, the size of the chillers and cooling towers or air-cooled condensers required for an ice storage system is significantly reduced, when compared to conventional chillers and heat rejection equipment sized for the instantaneous peak load. A typical thermal storage design includes chillers that provide 50 to 60% of the peak cooling load. The balance of the cooling requirement is provided from the ice storage system.



- **Reduced Pump and Pipe Sizes:** Pump and pipe sizes are also reduced in a properly designed ice storage system. Substantial savings in the chilled water distribution loop are realized when the system design incorporates reduced flow rates that result from using a larger temperature range in the water loop. Use of a larger temperature range, for example 18°F (10°C) instead of the more traditional 10°F (5.5°C) results in a reduction of pipe size. Condenser water pipe sizes are reduced due to lower flow requirements for the smaller chiller. Pump savings due to reduced chilled water and condenser water flow rates are also realized.





- **Reduced Cooling Coil and Supply Air Fan Sizes:** Cooling coils sized using lower supply water temperatures and traditional supply air temperatures are generally smaller due to fewer rows. The reduction in rows leads to lower supply fan HP (kW).

- **Reduced Air Handling Equipment:** When the air distribution is designed with lower supply air temperatures, the size of the ductwork, fans and fan motors are reduced.

- **Reduced Electrical Distribution:** Smaller chillers, heat rejection equipment and pumps require less horsepower than a traditional system, which results in smaller transformers, switchgear, wire sizes and starter panels.

- **Reduced Generator Size:** If a facility has a generator for daily or back-up power, the size of the generator will be significantly reduced when the peak electrical load of the facility is reduced using ice storage.

Reduced Energy Cost

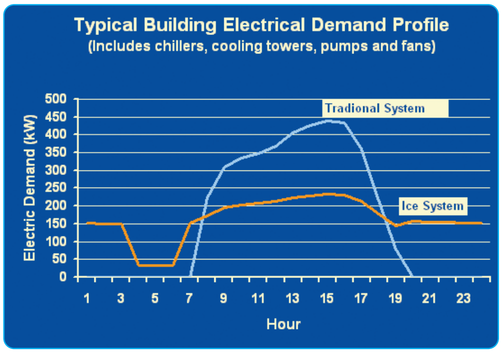
An ice thermal storage system reduces peak demand, shifts energy usage to non-peak hours, saves energy, and reduces energy costs.

- **Reduces Peak Demand and Shifts Energy Usage:** With less connected horsepower, ice storage can lower peak electrical demand for the HVAC or process cooling system by 50% or more. Since most electrical rates include demand charges during peak demand times and/or higher day versus night kWh charges, savings on electrical bills can be substantial. Peak electrical demand rates of $15 to $18 per kW are not uncommon. In areas with “real time pricing”, where the electric rate varies hour-by-hour based on the market price of electricity, day to night kWh costs can vary by 500 to 1000%. The use of electricity at night versus peak daytime hours can lead to large savings on energy bills.

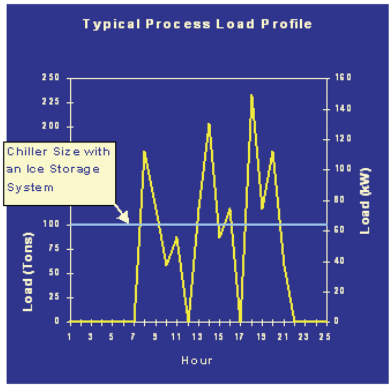
- **Saves Energy:** In addition, total annual kilowatt-hours used are less when the system is designed to take advantage of the low supply water temperature available from ice storage system. Lower kWh consumption is possible for several reasons:

    1. Although making ice requires more energy than producing chilled water, the efficiency penalty is not as large since the ice is made at night when condensing temperatures are lower, increasing the efficiency of the chiller.
    2. Ice systems typically operate the chiller at full load. Chillers are inefficient when run with low loads during the spring and fall. A conventional chiller will operate at less than 30% capacity for half the year.
    3. Reduced pumping horsepower.
    4. Reduced fan horsepower due to lower air pressure drop across the cooling coil. A higher chilled water temperature differential across the cooling coil usually results in fewer rows and therefore a lower pressure drop.
    5. The ability to recover waste heat from the chiller for heating water both night and day.

Benefits



Additional kWh savings are possible if the air distribution is designed to take advantage of the low temperatures available from the ice storage system. As the electric industry continues to deregulate, and time-of-use rates, real time pricing schedules and negotiated power prices become standard, ice storage can provide even greater future savings in operating costs.



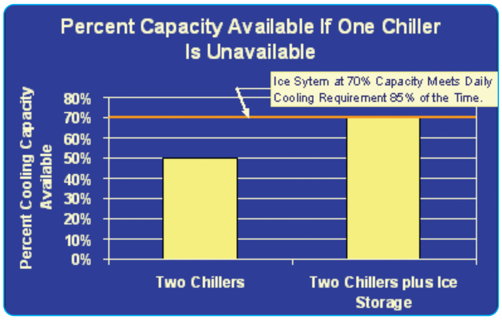
Variable Capacity

The ice storage system will maintain a constant supply temperature regardless of the variations in instantaneous cooling demand. The flow and entering water temperature set the instantaneous capacity.

Improved System Reliability

Ice storage systems provide the reliability necessary to ensure air-conditioning is available. With traditional systems, installing multiple chillers provides redundancy. In the event of a mechanical failure of one chiller, the second chiller provides limited cooling capacity. The maximum available cooling for the traditional system would only be 50% on a design day.

Most ice storage systems utilize two chillers in addition to the ice storage equipment. Two chillers are designed to provide approximately 60% of the required cooling on a design day while the ice storage provides the remaining 40% of the cooling capacity. In the event only one chiller is available to provide cooling during the day, up to 70% of the cooling capacity is available. The one operable chiller provides 30% of the cooling requirement while the ice provides up to 40%. Based on typical HVAC load profiles and ASHRAE weather data, 70% of the cooling capacity would meet the total daily cooling requirements 85% of the time.



Reduced Maintenance

The ice storage coils have no moving parts, so very little maintenance is required. Because the chillers, pumps and heat rejection equipment are smaller, ice storage systems will have less maintenance than a traditional system. The ice storage system also allows a chiller to undergo routine maintenance during the day when the ice storage can handle the system load.

Environmentally Friendly

Reducing energy consumption and using electricity at night will reduce global warming. Electricity generated at night generally has a lower heat rate (lower fuel use per power output), and therefore lower carbon dioxide and greenhouse gas emissions resulting in less global warming. The California Energy Commission concluded that the use of electricity at night created a 31% reduction in air emissions over the use of electricity during the day.

With smaller chillers, an ice storage system reduces the amount of refrigerant in a system. Most refrigerants in use today are slated to be banned in the future under the Montreal Protocol because they contribute to global warming. Using smaller amounts of refrigerant helps to save the ozone layer and reduce global warming.

Proven Technology

BAC has successfully applied ice storage technology to thousands of installations worldwide. BAC has the application and system experience to assist in the design, installation and operation of any ice storage system. BAC has supplied ICE CHILLER® Thermal Storage Products for projects that range in size from 90 to 125,000 ton-hours (0.3 to 441.3 MWh). Installations include office buildings, hospitals, manufacturing processes, schools, universities, sports arenas, produce storage facilities, hotels and district cooling applications.

The ICE CHILLER® Product includes a variety of factory-assembled units. For large applications, where space is limited or factory-assembled units are not cost effective, ICE CHILLER® Thermal Storage Coils are available for installation in field-erected tanks.

The BAC product offering provides system design flexibility. Ice can be built using various refrigerants or glycols on steel coils and is used to provide either chilled water or chilled glycol to the cooling system. This flexibility, combined with a broad range of application experiences, allows BAC to provide a cost-effective product to meet your specific requirements.

Merchandise Mart

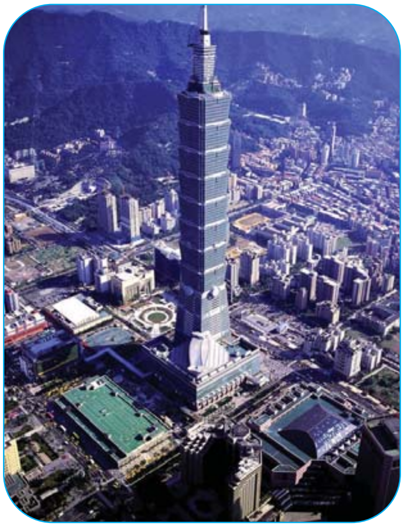
Merchandise Mart in Chicago, Illinois installed 26,400 ton-hours (93.2 MWh) of ICE CHILLER® Thermal Storage Coils in a retrofit of the building's air-conditioning system. The Merchandise Mart was built in 1930. The increased air- conditioning load on the building from computers, other electrical equipment and increased people density made the old system too small. Ice thermal storage, with low temperature water, allowed the retrofit of the air-conditioning system to go ahead without replacing piping and ductwork. Increasing the temperature ranges on the piping and air distribution system allowed the Merchandise Mart to install an ice storage system at a lower first cost than a conventional system.

Johns Hopkins Applied Physics Lab

The Johns Hopkins University Applied Physics Lab in Laurel, MD installed 5,600 ton-hours (19.8 MWh) of ICE CHILLER® Thermal Storage Coils to cool the new Steven Mueller Building which houses offices, labs and clean rooms. Another 2,800 ton-hours (9.9 MWh) of ICE CHILLER® Thermal Storage Coils were added to cool adjacent office and lab buildings. The ice thermal storage allows the Applied Physics Lab to save over $150,000 per year on its electric bill.

Friendship Annex 3 Office Building

The HVAC renovation of Friendship Annex (FANX) 2 and 3 in Baltimore, MD received the "Outstanding Engineering Achievement of the Year Award" from the Engineering Society of Baltimore. Ice with low temperature air distribution cools these renovated buildings. To meet federal guidelines, a comprehensive study of five alternate systems was made using life cycle costing. The analysis showed ice storage with low temperature chilled water and low temperature air to be the most economical system. A total of 15,230 ton-hours (53.8 MWh) of ICE CHILLER® Thermal Storage Units were installed for the two buildings.



Taipei 101

Taipei 101 is located in the central government and business district of Taipei, Taiwan. The building consists of a podium shopping and entertainment complex and office tower. Completed in August 2002, the 101-floor office tower is the world's tallest building at 508 meters.

BAC ice storage equipment (36,450 ton-hours) was selected because of its ability to provide low fluid temperatures, in this case 36°F (2°C). Low supply temperatures allowed economical selection of pressure isolation heat exchangers on the 42nd and 74th floors. Additionally, the low supply temperature allowed cold air distribution to be used throughout, thus reducing first costs and operating costs while providing improved occupant comfort.

IMUX Beijing District Cooling

IMUX Beijing District Cooling's first central cooling plant is located in Beijing's West Zone Zhongguancun Science and Technology Park, China's largest science park focused on developing high-tech enterprises. The plant, largely located underground, incorporates 29,800 ton-hours of BAC ice storage coils in a system which effectively uses less expensive nighttime power (75% less expensive than daytime power). Chilled water is supplied at 34°F (1°C) to a campus-style chilled water distribution loop. Many of the buildings served employ cold air distribution systems to achieve even lower construction and operating costs.

Proven Technology

Low Temperature Air

Omni Interlocken Resort Hotel

The Omni Interlocken Resort Hotel just outside of Denver, CO was designed with a low temperature air and water system using ICE CHILLER® Thermal Storage Units. The challenge was to design a high-quality HVAC system sensitive to building aesthetics, which would provide good guest comfort, low operating/maintenance costs and could be constructed within a tight construction budget. The first conceptual design was a four-pipe fan-coil system for the hotel rooms with air-cooled chillers and rooftop air-handler units for the public spaces. The final design was a low temperature air system with Modular ICE CHILLER® Thermal Storage Units. This low temperature air system was $500,000 less than the original conceptual design. In addition, the hotel's energy bills are $100,000 less than with a conventional system.

Villa Julie College

Modular ICE CHILLER® Thermal Storage Units were part of an expansion that doubled the size of this private college in Baltimore, MD. The new facilities added 135,700 ft^2 (12,620 m^2) of space to the campus and include a 400-seat auditorium and theater, gymnasium with showers and locker rooms, student center, video center, academic and computer classrooms, kitchen and administrative offices. The architect designed the new buildings with the intention that the structure be part of the visual space. This reduced the space allotted for the mechanical equipment. The engineer designed a low temperature air system that delivers 45°F (7°C) air temperature to VAV series fan powered boxes. The use of smaller piping and ductwork made it possible to avoid architectural changes that would affect the aesthetics of the design.

Food Processing

Zippy's Restaurant Central Facility

At Zippy's in Honolulu, HI, food is cooked in a central kitchen where it is cooled and packaged for use in local Zippy's restaurants. The FDA requires that the food in the cooking vessels be cooled to 45°F (7°C) in less than one hour to prevent contamination. The cooking vessels in the kitchen need varying amounts of cooling depending on the dish that is being prepared, and when it finishes its cooking cycle. Because of the varying cooling load from day to day and hour to hour and the need for a quick cool down period, standard chillers are not a good match for this application. Ice storage with its variable capacity and low supply temperature is an excellent match for this process cooling application.

Power Generation

Wolverine Power

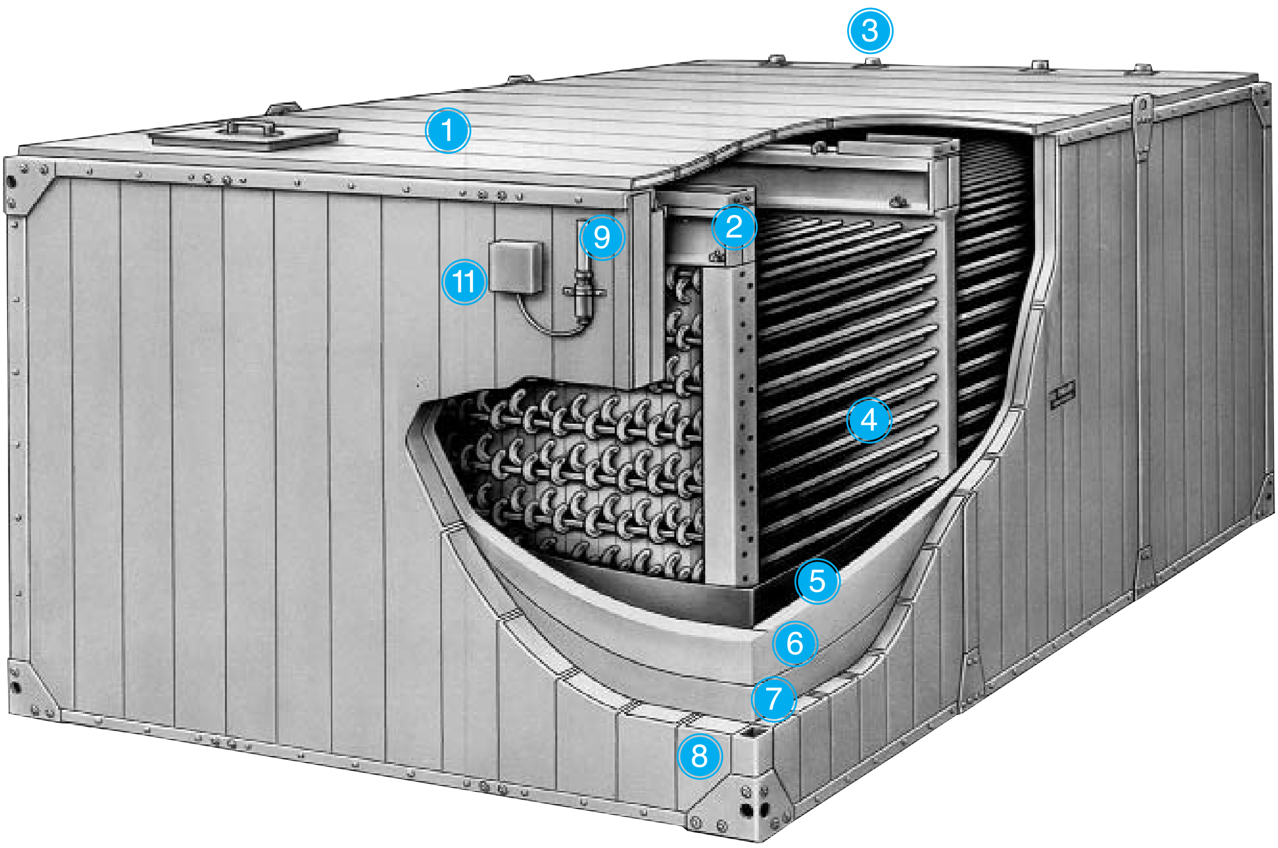
Wolverine Power, located in central Michigan, is a generation and transmission electric cooperative. For a new generating plant with (2) 22-megawatt Rolls Royce turbines, Wolverine Power elected to use ice storage for their turbine inlet air cooling. They installed 7,610 ton-hours (26.9 MWH) of ICE CHILLER® Thermal Storage Units to gererate 40°F (4.4°C) chilled water, which provides 55°F (13°C) inlet air. The generating plant's ice storage capacity can be used over a 16-hour period as partial storage or over a 4-hour period as full storage, depending on the value of power on the open market. During peak summer time, the increased power capacity is worth up to $3,500 per hour in electricity sales.

Emergency Cooling

Verizon

Verizon, the provider of telephone service to a large portion of the east coast, uses an ICE CHILLER® Thermal Storage Unit to provide back-up cooling to one of its computer centers in Silver Spring, MD. If the chiller that provides cooling goes down for any reason, power outage or alarm, the system immediately switches over to the ice storage system for cooling. The pump on the ice storage system is on the continuous power back up with the computers. There is enough ice to provide cooling for 30 minutes. This gives Verizon enough time to clear the alarm or get the back-up generator running and the chiller back on line.

Construction Details



1 Covers

- Watertight
- G-235 (Z700 Metric) Hot-dip galvanized steel panels
- Insulated with 2” expanded polystyrene insulation

2 Coil Support Beams

- Prevent contact between coil and primary liner

3 Glycol Connections

- Grooved for mechanical coupling

4 Galvanized Steel Coil

- Continuous serpentine, steel tubing
- Hot-dip galvanized after fabrication (HDGAF)
- Pneumatically tested at 190 psig
- Rated for 150 psig operating pressure

5 Primary Liner

- Single piece
- 48-hour integrity test before shipment

6 Extruded Polystyrene Insulation

- 1.5” thick, installed between primary and secondary liners

7 Secondary Liner/Vapor Barrier

- Prevents moisture from penetrating through the insulation

8 Wall Panel

- Heavy-gauge galvanized steel with double brake flanges
- 3” of expanded polystyrene insulation

9 Sight Tube

- Visual indicator of water level corresponding to the amount of ice in unit

10 Operating Control (Not Shown)

- High-level float switch and low water cutout mounted on the outside of the tank
- Can be provided for one tank (standard) or all tanks (optional)

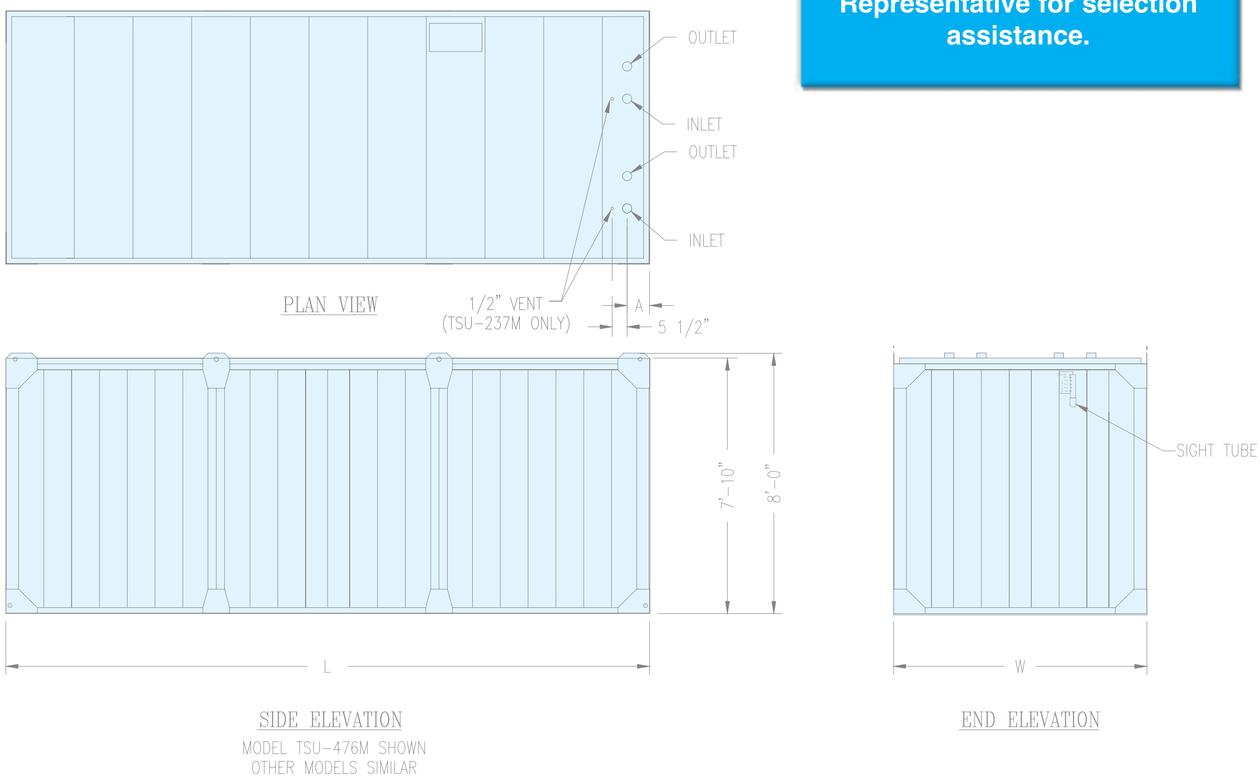
11 Ice Inventory Sensor (Optional)

- Differential pressure transmitter provides an electrical 4-20 mA output signal which is proportional to the amount of ice in inventory

Engineering Data

Do not use for construction. Refer to factory certified dimensions. This handbook includes data current at the time of publication, which should be reconfirmed at the time of purchase. Up-to-date engineering data and more can be found at **www.BaltimoreAircoil.com**.

Contact your local BAC Representative for selection assistance.



MODEL NUMBER	WEIGHTS (lbs)		VOLUMES (gal)		DIMENSIONS		CONNECTION SIZE
	OPERATING	SHIPPING	TANK WATER	COIL GLCOL	L	W	
TSU-237M	39,100	9,750	2,990	260	10' 7-5/8"	7' 10-3/8"	2"
TSU-476M	73,900	16,750	5,840	495	19' 10-1/4"	7' 10-3/8"	3"
TSU-594M	93,100	20,200	7,460	610	19' 10-1/4"	9' 9-1/4"	3"
TSU-761M	113,800	24,000	9,150	790	19' 10-1/4"	11' 9-3/4"	3"

Notes:

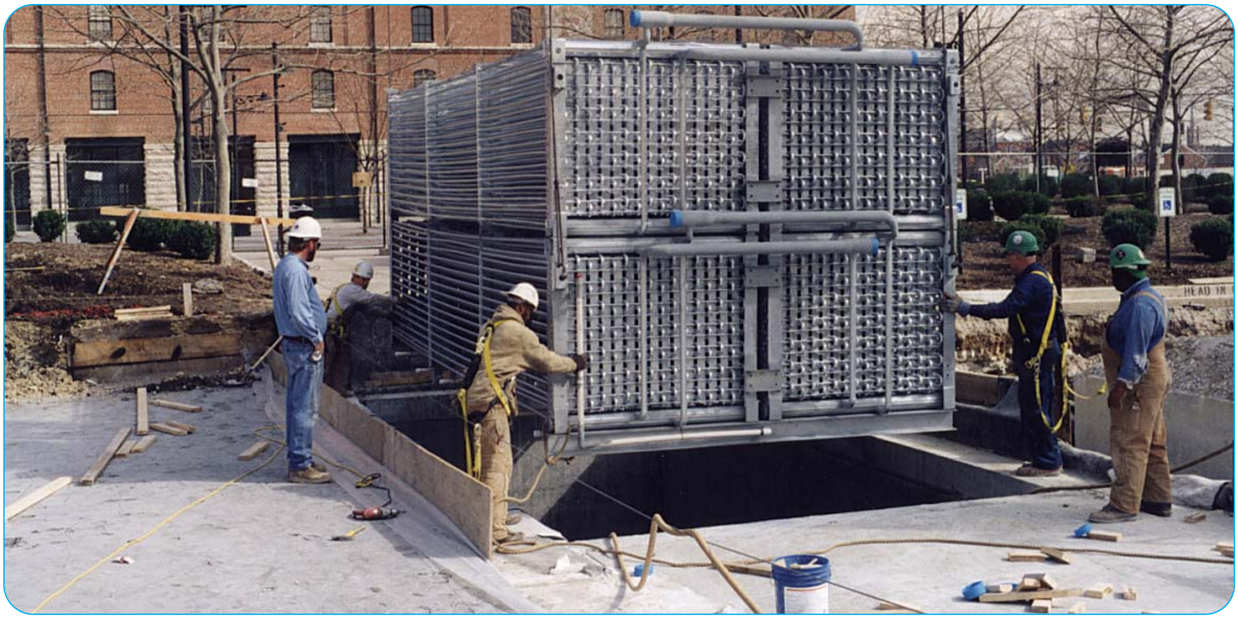
1. Unit should be continuously supported on a flat level surface.
2. All connections are grooved for mechanical coupling.

Custom Coils

BAC will manufacture custom ICE CHILLER® Thermal Storage Coils to meet project specific requirements. BAC has done extensive research and testing on the build and melt characteristics of ice storage. This research and testing has resulted in selection capabilities unmatched by any other company in the industry.

BAC can predict the temperatures required on an hour-by-hour basis for building ice on custom coils, over a variety of conditions and build times. The physical space available, load profile, discharge temperatures, chiller capacity and operating sequences can be evaluated to find the design that best meets the application.

The ICE CHILLER® Thermal Storage Coils are constructed of continuous 1.05" O.D. all prime surface serpentine steel tubing, with no intermediate butt welds. The coils are assembled in a structural steel frame designed to support the weight of the coil stack with a full ice build. After fabrication the coils are tested for leaks using 375 psig air pressure under water, then hot-dip galvanized for corrosion protection.



Installation of coil module at Oriole Park at Camden Yards, Baltimore, MD

For glycol applications the coils are configured to provide countercurrent glycol flow in adjacent circuits for maximum storage capacity.

Individual coils are factory-assembled into modules of two (2) coils. Glycol manifolds are coated with zinc-rich, cold galvanizing finish at the factory. Necessary support steel and lifting lugs are provided on the modules to allow for lifting into and final positioning within the storage tank.

Engineering Data:
Modes of Operation

The modular ICE CHILLER® Thermal Storage Unit can operate in any of five distinct operating modes. These modes of operation provide the flexibility required by building operators to meet their daily HVAC cooling requirements.

Ice Build

In this operating mode, ice is built by circulating a 25% solution (by weight) of inhibited ethylene/ propylene glycol through the coils contained in the ICE CHILLER® Thermal Storage Unit. Figure 1 illustrates typical chiller supply temperatures for 8, 10 and 12 hour build cycles. For a typical 10–hour build time, the supply glycol temperature is never lower than 22°F (-5.6°C). As the graph illustrates, for build times exceeding 10 hours, the minimum glycol temperature is greater than 22°F (-5.6°C). For build times less than 10 hours, the minimum glycol temperature will be lower than 22°F (-5.6°C) at the end of the build cycle. This performance is based on a chiller flow rate associated with a 5°F (2.8°C) range. When a larger temperature range is the basis of the chiller selection, the chiller supply temperatures will be lower than shown in Figure 1.

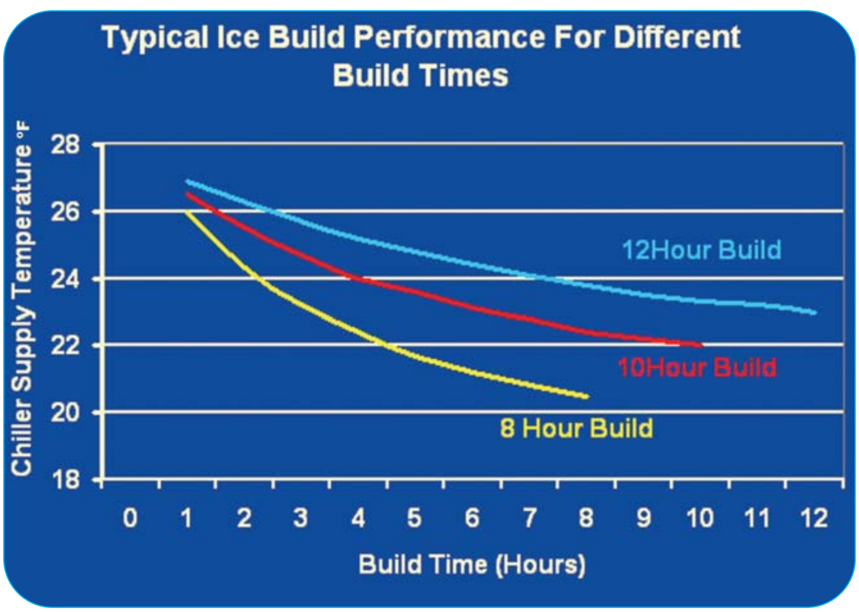


Figure 1

Ice Build with Cooling

When cooling loads exist during the ice build period, some of the cold glycol used to build ice is diverted to the cooling load to provide the required cooling. The amount of glycol diverted is determined by the building loop set point temperature. BAC recommends that this mode of operation be applied on systems using primary/secondary pumping. This reduces the possibility of damaging the cooling coil or heat exchanger by pumping cold glycol, lower than 32°F (0°C), to this equipment.

Cooling – Ice Only

In this operating mode the chiller is off. The warm return glycol solution is cooled to the desired set point temperature by melting ice stored in the modular ICE CHILLER® Thermal Storage Unit.

Cooling – Chiller Only

In this operating mode the chiller supplies all the building cooling requirements. Glycol flow is diverted around the thermal storage equipment to allow the cold supply glycol to flow directly to the cooling load. Temperature set points are maintained by the chiller.

Cooling – Ice with Chiller

In this operating mode, cooling is provided by the combined operation of the chiller and ice storage equipment. The glycol chiller precools the warm return glycol. The partially cooled glycol solution then passes through the ICE CHILLER® Thermal Storage Unit where it is cooled by the ice to the design temperature.

Modular ICE Thermal Storage Unit

Engineering Data:
System Schematics

ICE CHILLER®

Two basic flow schematics are applied to select ICE CHILLER® Thermal Storage Units. Figure 2 illustrates a single piping loop with the chiller installed upstream of the thermal storage equipment. This design allows the thermal storage system to operate in four of the five possible operating modes. They are Ice Build, Cooling-Ice Only, Cooling-Chiller Only and Cooling-Ice with Chiller.

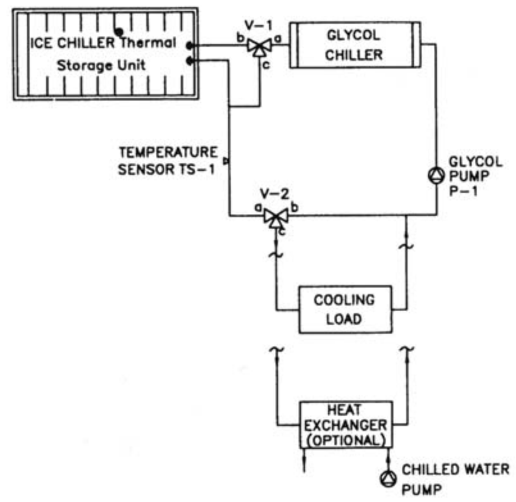


Figure 2

For Figure 2 the following control logic is applied:

MODE	CHILLER	P-1	V-1	V-2
Ice Build	On	On	A-B	A-B
Cooling – Ice Only	Off	On	Modulate	A-C
Cooling – Chiller Only	On	On	A-C	A-C
Cooling – Ice With Chiller	On	On	Modulate	A-C

Valve V-1 modulates in response to temperature sensor, TS-1. Valve V-2 could be positioned to either maintain a constant flow, less than P-1, or modulate in response to the return glycol temperature from the cooling load.

When the building loop contains chilled water, a heat exchanger must be installed to separate the glycol loop from the building's chilled water loop. On applications where an existing water chiller is available, it can be installed in the chilled water loop to reduce the load on the thermal storage system.

This design should not be used when there is a requirement to build ice and provide cooling. This would require the cold return glycol from the thermal storage equipment be pumped to the cooling load or heat exchanger. Since the glycol temperature is below 32°F (0°C), the cooling coil or heat exchanger is subject to freezing. The flow schematic illustrated in Figure 3 details a primary/secondary pumping loop with the chiller located upstream of the thermal storage equipment. This design allows the system to operate in all five operating modes.

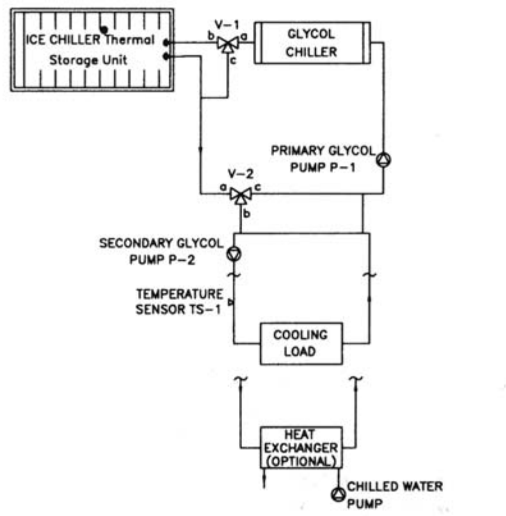


Figure 3

For Figure 3 the following control logic is applied:

MODE	CHILLER	P-1	P-2	V-1
Ice Build	On	On	Off	A-B
Ice Build With Cooling	On	On	On	A-B
Cooling – Ice Only	Off	On	On	Modulate
Cooling – Chiller Only	On	On	On	A-C
Cooling – Ice With Chiller	On	On	On	Modulate

Valve V-1 and Valve V-2 modulate, depending on the operating mode, in response to temperature sensor, TS-1. The benefit provided by the primary/secondary pumping loop is that the system can build ice and provide cooling without fear of freezing a cooling coil or heat exchanger. This system design also allows for different flow rates in each of the pumping loops. When the flow rates in the pumping loops are different, the glycol flow rate in the primary loop should be greater than or equal to the glycol flow rate in the secondary loop. As in the single loop schematic, a heat exchanger and a base water chiller can be added to the system schematic.

Variations to these schematics are possible but these are the most common for ice storage systems. One variation positions the chiller downstream of the ice storage equipment. By positioning the chiller downstream of the ice, the chiller is used to maintain the required supply temperature. In Figures 2 and 3, the chiller is installed upstream of the ice. This offers two significant advantages compared to system designs that locate the chiller downstream of the ice. First, the chiller operates at higher glycol temperatures to precool the return glycol. This enables the chiller to operate at a higher capacity which reduces the amount of ice required. Second, since the chiller is operating at higher evaporator temperatures, the efficiency (kW/TR) of the chiller is improved.

Engineering Specifications

See our website at www.BaltimoreAircoil.com for an electronic copy of product engineering specifications.

The ICE CHILLER® Thermal Storage Unit(s) shall be Baltimore Aircoil Company Model TSU-______. Each unit shall have a latent ton-hour storage capacity of _______ ton-hours to be generated in ____ hours when supplied with _______ USGPM (lps) of a 25% (by weight) solution of industrially inhibited ethylene/propylene glycol. The minimum glycol temperature required during the ice build operating mode shall be _______ °F(°C). Rated system performance shall be provided in the format recommended by the Air-Conditioning & Refrigeration Institute (ARI) Guideline T. The thermal storage units shall be modular in design. Unit design shall allow units of different sizes to be installed in order to optimize unit selection and minimize space requirements. Tanks sizes can be mixed due to internal piping arrangements that create a balanced flow due to uniform pressure drop through the coil circuits.

The tank shall be constructed of heavy-gauge galvanized steel panels and include double brake flanges for structural strength. The tank walls shall be supplied with a minimum of 4-1/2" of insulation that provides a total insulating value of R-18. The tank design shall utilize multiple liners. The primary liner, which forms the interior of the unit, shall be of single piece construction and be suitable for low temperature applications. The secondary liner/vapor barrier shall be separated from the primary liner by 1-1/2" of extruded polystyrene insulation. The tank bottom shall be insulated with 2" of expanded polystyrene insulation and 1" of extruded polystyrene insulation.

The ICE CHILLER® Thermal Storage Unit shall be provided with watertight, sectional covers constructed of hot-dip galvanized steel. The covers shall be insulated with a minimum of 2" of expanded polystyrene insulation.

Contained within the tank shall be a steel heat exchanger that is constructed of 1.05" O.D., all prime surface serpentine steel tubing encased in a steel framework. The coil, which is hot-dip galvanized after fabrication, shall be pneumatically tested at 190 psig and rated for 150 psig operating pressure. The coil circuits are configured to provide maximum storage capacity. The coil connections on the unit are galvanized steel and are grooved for mechanical coupling.

Each ICE CHILLER® Thermal Storage Unit shall be provided with a sight tube. The sight tube, which shall be fabricated from clear plastic pipe, shall display the tank water level and corresponding ice inventory.

Operating controls, consisting of two float switches, shall be mounted on the outside of the tank. The high level float switch terminates the build cycle when the tank water level reaches the 100% ice build level. The high level switch shall also prevent re-initiation of the build cycle until approximately 15% of the ice has been discharged. The second float switch is a low water cutout. The cutout requires that the water level in the ICE CHILLER® Thermal Storage Unit be at or above the 0% ice build level before the ice build cycle can begin. Operating control quantities vary based on project requirements. An optional differential pressure transmitter shall be available to supply an electrical output signal proportional to the amount of ice in inventory.

The heat transfer fluid shall be an industrially inhibited, 25% by weight, ethylene/propylene glycol solution specifically designed for HVAC applications. The 25% (by weight) solution is designed to provide freeze/burst and corrosion protection as well as efficient heat transfer in water based, closed loop systems. Corrosion inhibitors shall be provided to keep pipes free of corrosion without fouling. DOWTHERM® SR-1 and UCARTHERM® are acceptable fluids.

Overall unit dimensions shall not exceed approximately _____ ft (m) by ____ ft (m) with an overall height not exceeding ____ ft (m). The operating weight shall not exceed _______ lbs (kg).

Engineering Considerations - ICE CHILLER® Thermal Storage Products

Installation

ICE CHILLER® Thermal Storage Units are designed to be installed outdoors. The units must be installed on a continuous flat level surface. The pitch of the slab must not exceed 1/8" over a 10' span. Figure 1 details ICE CHILLER® Thermal Storage Unit layout guidelines. The units should be positioned so there is sufficient clearance between units and adjacent walls to allow easy access. When multiple units are installed, a minimum of 18" is recommended side-to-side and 3'-0" end-to-end for access to the operating controls.

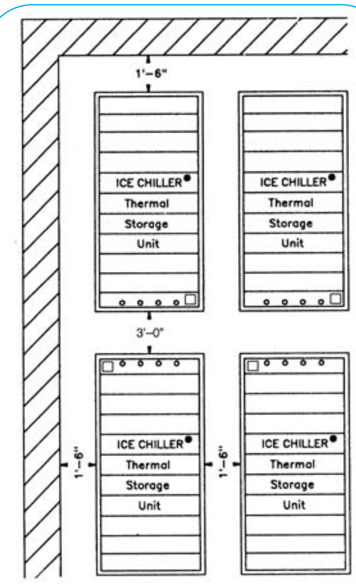


Figure 1

When installed indoors, the access and slab requirements described above also apply. The units should be placed close to a floor drain in the event they need to be drained. The minimum height requirement above the tank for proper pipe installation is 3 feet. Figure 2 illustrates the recommended overhead clearance for ICE CHILLER® Thermal Storage Units.

For large ton–hour applications, BAC will provide ICE CHILLER® Thermal Storage Coils for installation in field fabricated concrete tanks. When coils are required, BAC's manufacturing capabilities allow coils to be manufactured in the size and configuration necessary to meet specific site and performance requirements. The concrete tank design is to be completed by a qualified structural engineer. Figure 3 illustrates the ICE CHILLER® Thermal Storage Coil layout guidelines. For large projects that require ICE CHILLER® Coils, contact your local BAC Representative for selection and dimensional information.

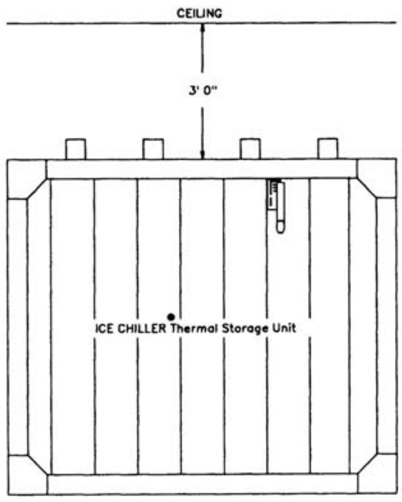


Figure 2

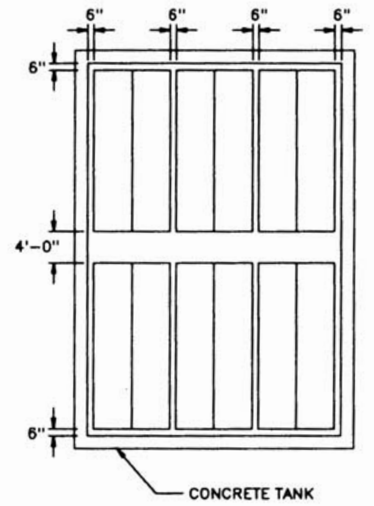


Figure 3

Unit Piping

Piping to the ICE CHILLER® Thermal Storage Unit should follow established piping guidelines. The coil connections on the unit are galvanized steel and are grooved for mechanical coupling.

For single tank applications, each pair of manifolded coil connections should include a shut-off valve, so the unit can be isolated from the system. Figure 4 illustrates the valve arrangement for a single unit. It is recommended that the piping include a bypass circuit to allow operation of the system without the ICE CHILLER® Thermal Storage Unit in the piping loop. This bypass can be incorporated into the piping design by installing a three way/modulating valve. This valve can also be used to control the leaving glycol temperature from the thermal storage unit. Temperature and pressure taps should be installed to allow for easier flow balancing and system troubleshooting. A relief valve, set at a maximum of 150 psi, must be installed between the shut-off valves and the coil connections to protect the coils from excessive pressures due to hydraulic expansion. The relief valve should be vented to a portion of the system which can accommodate expansion.

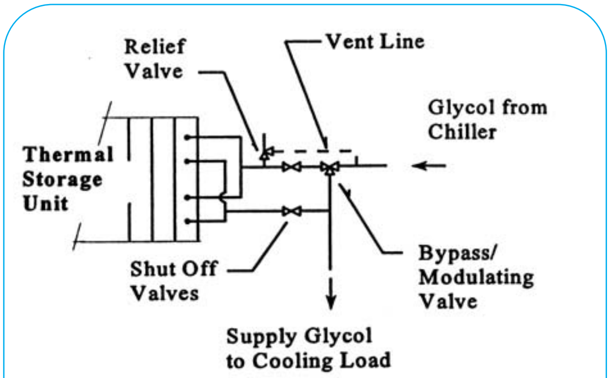


Figure 4

CAUTION:

The system must include an expansion tank to accommodate changes in fluid volume. Adequately sized air vents must be installed at the high points in the piping loop to remove trapped air from the system.

Figure 5 illustrates reverse return piping for multiple units installed in parallel. The use of reverse return piping is recommended to ensure balanced flow to each unit. Shut-off valves at each unit can be used as balancing valves.

When large quantities of ICE CHILLER® Thermal Storage Units are installed, the system should be divided into groups of units. Then, balancing of each unit can be eliminated and a common balancing valve for each group of units installed. Shut-off valves for isolating individual units should be installed but not used for balancing glycol flow to the unit.

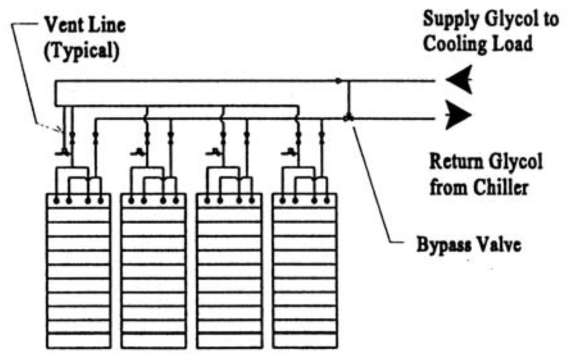


Figure 5

Controls

To ensure efficient operation of the ICE CHILLER® Thermal Storage Units, each system is provided with factory installed Operating Controls. A brief description of the controls follow.

Once the ice build cycle has been initiated, the glycol chiller should run at full capacity without cycling or unloading until the ICE CHILLER® Thermal Storage Units are fully charged. When the units are fully charged, the chiller should be turned off and not allowed to re-start until cooling is required. The ice build cycle is terminated by the Operating Control Assembly. This assembly includes a low water cutout and a shut-off switch. The low water cutout prevents the ice build mode from starting if there is insufficient water in the tank. The shut-off switch will terminate the build cycle when the units are fully charged and will prevent the next ice build mode from starting until 15% of the ice has melted.

Note: Multiple Operating Control Assemblies must be wired in series so that a full build signal from any one tank will terminate the ice build cyle.

An inventory control that provides a 4 - 20 mA signal is available. This control should be used for determining the amount of ice in inventory, but not to terminate the ice build cycle. Complete operating control details are provided in the Installation, Operation and Maintenance Manual.

Glycol

ICE CHILLER® Thermal Storage Units typically use a 25% (by weight) solution of industrially inhibited ethylene/propylene glycol for both corrosion protection and freeze protection. Industrial grade inhibited glycol is specifically designed to prevent corrosion in HVAC and heat transfer equipment. Inhibitors are used to prevent the ethylene glycol from becoming acidic and to protect the metal components in the thermal storage system. The system's lowest operating temperature should be 5°F to 7°F (2.8°C to 3.9°C) above the glycol freeze point. The freeze point for a system with 25% by weight ethylene glycol is 13°F (10.6°C); the freeze point for a system with 25% by weight propylene glycol is 15°F (9.4°C).

Acceptable industrial grade inhibited glycol solutions are DOWTHERM® SR–1, DOWFROST® HD and UCARTHERM®. Use of other brands of glycol in ICE CHILLER® Thermal Storage Products should be approved by BAC.

DOWTHERM® SR-1, DOWFROST® and UCARTHERM® are registered trademarks of The Dow Chemical Company or its subsidaries.

CAUTION:

Uninhibited glycol and automotive antifreeze solutions are NOT to be used on thermal storage applications.

Water Treatment

In the near freezing temperatures of the ICE CHILLER® Thermal Storage Unit, scale and corrosion are naturally minimized. Therefore, water treatment for these two conditions may not be required or may require minimal attention unless the water is corrosive in nature. To control biological growth, a biocide may be needed to prevent the spread of iron bacteria or other organisms. For specific recommendations, consult a reputable local water treatment company and follow the guidelines below:

PROPERTY OF WATER	RANGE
pH	7.0 to 9.0[1]
Hardness as CaCO3	30 to 500 ppm
Alkalinity as CaCO3	500 ppm maximum
Total Dissolved Solids	625 ppm maximum
Maximum Conductivity	1000 micromhos/cm @ 32°F
Chlorides	125 ppm maximum as Cl
Sulfates	125 ppm maximum

Note:

1. **A water pH of 8.3 or higher will require periodic passivation of the galvanized steel to prevent "white rust," the accumulation of white, waxy, nonprotective zinc corrosion products on galvanized steel surfaces.**

To assure full capacity of the ICE CHILLER® Thermal Storage Unit, the water treatment should not alter the freeze point of water.

Winterization

CAUTION:

Precautions must be taken to protect the unit and associated piping from freezing conditions. Heat tracing and insulation should be installed on all piping connected to the unit. The sight tube, operating controls and optional inventory sensor must be protected if the units are installed outdoors and exposed to sub-freezing ambient conditions.

For this purpose, BAC can provide an optional heated enclosure, complete with a 100 W heater. Otherwise, the sight tube, operating controls and optional inventory sensor must be heat traced and insulated. It is not necessary to drain the unit during cold weather.

Pressure Drop

The ICE CHILLER® Thermal Storage Unit is designed for low pressure drop. Figure 6 shows the pressure drop associated with each unit for a 25% solution of industrially inhibited ethylene glycol. Data for flow rates not shown should not be extrapolated from the performance curve. Pressure drops for flow rates not presented in this table, and for alternative fluids, are available by contacting the local BAC Representative.

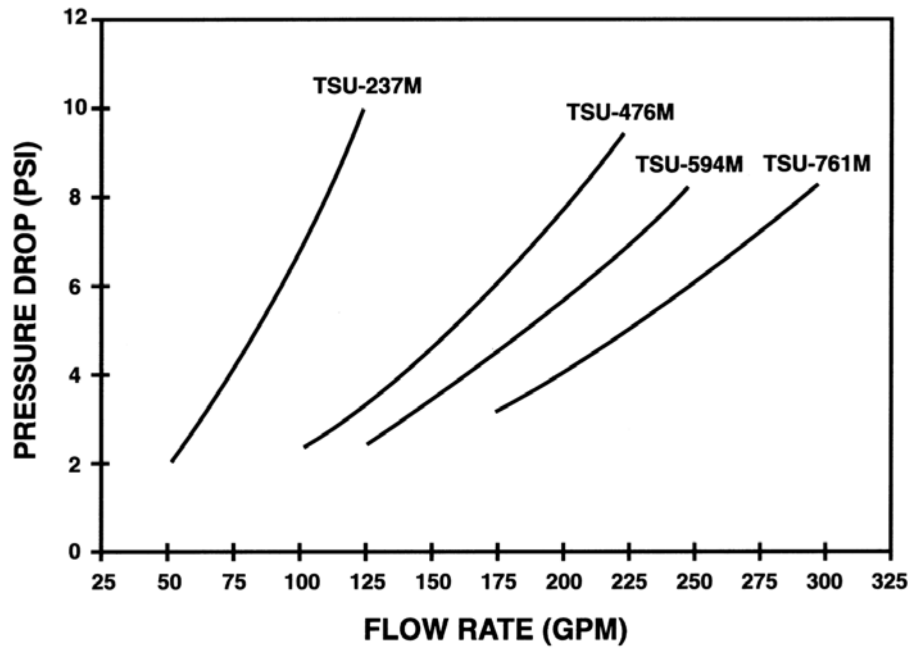


Figure 6

Warranties

Please refer to the Limitation of Warranties applicable to and in effect at the time of the sale/purchase of these products.

ACHIEVING ENERGY CONSERVATION WITH ICE-BASED THERMAL STORAGE

T.W. Brady
Member ASHRAE

ABSTRACT

In the last few years, studies have reported that, while reducing on-peak instantaneous demand for electrical energy, some existing ice-based thermal storage systems significantly increase overall annual energy consumption. It is thus feared that incentives offered by various electric utility companies to reduce on-peak demand for electricity, and thereby cost to the customer, will actually increase electrical generating requirements and lead to a waste of natural resources and increased environmental pollution. In this report, actual data collected from buildings that have at least two years of operating experience will be used to demonstrate that the negative impacts recognized by these studies can be avoided when designers take full advantage of the unique technological features offered by an ice-based thermal system. Specifically, it will be shown that a thermal storage system that is fully integrated with the total building design will reduce annual energy consumption, reduce peak demand, reduce annual operating costs, and reduce the environmental impact from the use of mechanical systems. The net impact of thermal storage systems upon the environment at the generation source will be discussed.

INTRODUCTION

Considerable attention is now being given to the relative annual energy consumption of building heating, ventilating, and air-conditioning (HVAC) systems that employ ice-based thermal energy storage (ITES) versus alternative system designs. Reports of the operating results achieved by several different ice storage systems (Gillespie and Turnbull 1991) have drawn the criticism of electric utility rate commissions and utility research institutes. This criticism emphasizes the importance of promoting HVAC system designs that reduce energy consumption, are environmentally benign, and, hopefully, reduce energy cost to the end-user.

For at least 15 years, ITES systems have been recognized for their ability to shift electric demand from on-peak electric rate periods to off-peak periods. As a result of utility rates that charge a high fee for on-peak demand kilowatts (kW), these systems have been able to reduce the overall cost of electric energy delivered. However, in most of the cases studied, there has been an increase in the annual energy consumption when compared to theoretically "conventional" HVAC systems. Any increase in energy requirements is seen as having a negative environmental impact due to the greater use of natural resources and the resultant potential for increased carbon dioxide (CO_2) emissions.

The criticism of ITES systems based upon these alleged negative attributes has led to reduced utility sponsorship of systems that utilize this technology and reduced funding for research on ITES-related subjects.

It is the author's experience that ITES systems that are properly designed to take advantage of all aspects of the technology reduce annual energy consumption when compared to alternative system designs. This paper compares system design features as well as annual operating results of the selected systems. Factors influencing the sizing and selection of components are discussed, and recommendations based upon field experience are presented. The discussion emphasizes that ITES technology must be fully integrated into a building and the HVAC system design in order for all of the benefits to be realized. Further, it shows that comparisons that rank the various types of thermal storage systems in order of energy efficiency are not useful for system selection purposes due to the impact of individual project specifics upon the features of each system type.

Following a discussion of the individual systems and associated energy consumption, the impact of thermal energy storage (TES) technology on energy use at the point of generation is investigated. It is the opinion of some researchers that TES technology increases the off-peak use of coal for electrical generation while reducing the on-peak use of nuclear energy. It has been assumed that this shift of primary fuel will increase CO_2 emissions.

The factors that influence the decision of the bulk power operators as to the most economical mix of generating capacity for any given time are many and vary considerably. It is nearly impossible, therefore, to arrive at a consensus as to whether the widespread use of TES systems will actually have the assumed negative impact on CO_2 emissions. From the fact that all utilities attempt to operate their most efficient mix of generating capacity at all times and due to the interconnecting grid that allows neighboring

Thomas W. Brady is president of Brady Consulting Services, Inc., Westchester, IL.

utilities to sell power to one another, it is reasoned that TES technology has a positive effect on CO_2 emissions. This paper presents that reasoning.

SYSTEM DESIGN COMPARISON

The size of the building under consideration determines, in part, the system configuration options that are available to the designer. This statement applies to both conventional HVAC systems and ITES systems. The considerations presented in this paper apply to buildings that are 150,000 gross ft^2 (13,935 m^2) and larger. The specific designs that will be analyzed are the HVAC systems that serve a multi-story office building of 500,000 gross ft^2 (46,450 m^2). A similar comparison can be drawn for smaller buildings; however, the type of equipment selected for both the conventional system and the ITES system would likely not be the same as that used in this comparison.

Building Design Considerations

As stated, the building to be analyzed contains 500,000 gross ft^2 (46,450 m^2), with 412,000 ft^2 (38,275 m^2) of general and private office space that is to be environmentally controlled by eight central station air-handling systems. There is also a fitness center and a kitchen that measure 18,000 ft^2 (1,672 m^2) and 9,750 ft^2 (906 m^2), respectively. The building is located in a geographical area that has an ambient design of 95°F (35°C) dry-bulb (DB) and 76°F (24.4°C) wet-bulb (WB).

The local utility has established an on-peak electrical use period of 9:00 a.m. to 6:00 p.m., Monday through Friday. All other hours are considered part of the off-peak electrical period. The charge for electrical consumption (per kWh) during the on-peak period is slightly more than twice the charge for consumption during off-peak hours. A charge is levied on the highest instantaneous rate of use (per kW) that is recorded during the on-peak hours in each monthly billing period. This charge exceeds $16.00 per kW and has the effect of making on-peak consumption cost nearly six times more than off-peak consumption.

Pertinent cooling load information on the building is given in Table 1. The size of the miscellaneous cooling loads and their location within the building make their inclusion in the building's central chilled-water system a logical choice. Thus, combining all cooling loads, the peak demand on the central system is 17,040 MBh (4,993.7 MW).

System Selection

The two central cooling systems selected for comparison in this analysis are shown in Figures 1 and 2. There is space for eight air-handling systems that will serve the office areas and individual air-handling systems for both the fitness center and the kitchen.

TABLE 1
Building Cooling Load Information

Office Area	
Coincident peak cooling load	14,891 MBh
Coincident peak room sensible heat gain	9,369 MBh
Minimum outdoor air required for ventilation (25 cfm/p)	42,800 cfm
Fitness Center	
Cooling load at peak	452 MBh
Minimum outside air	5,600 cfm
Kitchen	
Cooling load at peak	857 MBh
Minimum outside air	20,000 cfm
Miscellaneous cooling loads	
Heat rejection from kitchen refrigeration equipment	480 MBh
24-hour equipment cooling load	360 MBh
Total Building	
Coincident peak cooling load	17,040 MBh
Daily cooling load on peak day	187,200 MBh (15,600 ton-h)

As shown in Figure 1, the conventional system is composed of three centrifugal chillers, three cooling towers, individual chiller and condenser water-circulating pumps, and a variable-speed primary chilled-water distribution loop to serve the chilled-water coils in the air-handling units and any other uses. There is a heat exchanger that is used in conjunction with one of the cooling towers to provide free cooling to the miscellaneous cooling loads during the winter months.

The thermal storage system shown in Figure 2 has three rotary screw compressors, three evaporative condensers, a liquid overfeed pump/accumulator package, a chiller bundle, an ice-on-coil thermal storage tank, a heat exchanger between the storage system and the chilled-water system, ice/water pumps, secondary loop pumps, and a variable-speed primary chilled-water distribution loop to serve the chilled-water coils in the air-handling units and any other uses. In addition to providing compressor-driven water chilling to supplement the thermal storage in the summer months, the chiller bundle is used in conjunction with the evaporative condensers to provide thermosiphon cooling to the miscellaneous cooling loads during the winter months. The sumps of the evaporative condensers operate dry during this cooling mode.

The heating system serving the building is gas hot water with perimeter heating elements.

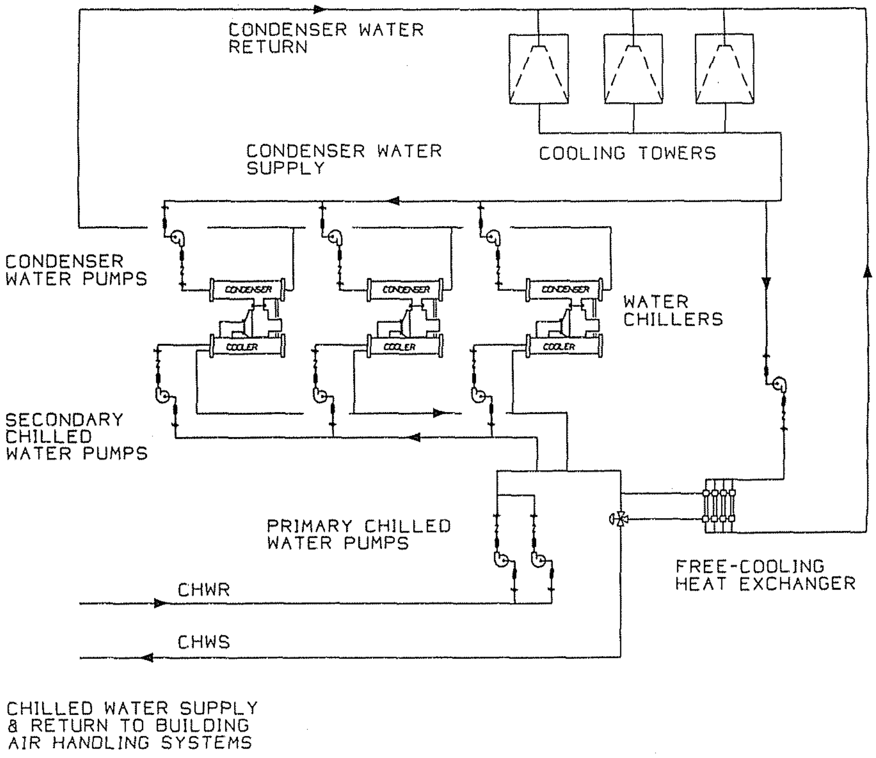


Figure 1 ***Conventional system flow diagram.***

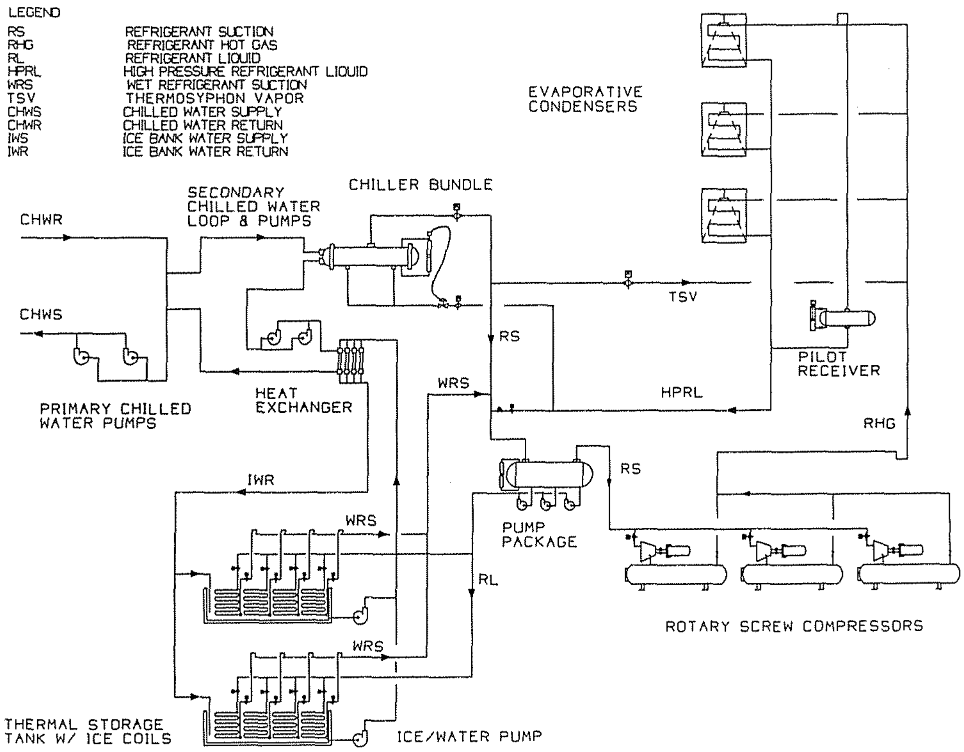


Figure 2 ***ITES system flow diagram.***

Equipment Selection

In order to compare the annual performance of the conventional system shown in Figure 1 with the thermal storage system shown in Figure 2, the equipment to meet the design requirements was selected. The performance published by the manufacturers in accordance with ARI standards was used in all calculations. Since the air-handling units serving the fitness center and kitchen as well as the equipment serving the miscellaneous loads apply equally to both system types, their selection information and energy use calculations have not been included as part of this paper. The equipment selected and the electrical consumption at the design operating point are shown in Tables 2 and 3.

The essential elements of the design shown in Table 2 include the following:

1. The central fan systems have been equally sized and deliver air to meet the space-cooling load at 56°F (13.3°C). The supply fan in each system is served by a variable-frequency drive that responds to the airflow requirements of variable-air-volume (VAV) terminal units.
2. The cooling towers have been equally sized and collectively have the capacity to accommodate the heat rejection of the selected chillers when delivering 85°F (29.4°C) water with a 78°F (25.6°C) WB ambient temperature and a 10°F (5.6°C) temperature range. Water delivered from the cooling towers is allowed to fall to 60°F (15.6°C) as the ambient temperature drops.

TABLE 2
Conventional System—Installed Cooling Equipment

Equipment	Capacity (each unit)	hp Installed	Bhp or kW
Fans			
8 supply	60,240 cfm	100	76.4 Bhp
8 exhaust/return	60,240 cfm	20	18.4 Bhp
Cooling Towers			
3 sections	7,125 MBh	30	27.0 Bhp
Condenser Water Pumps			
3 pumps	1,425 gpm @ 80 ft TDH	40	38.0 Bhp
Chiller Pumps			
3 pumps	812 gpm @ 80 ft TDH	30	22.5 Bhp
Primary Chilled-Water Pumps			
2 pumps	1,218 gpm @ 134 ft TDH	60	50.0 Bhp
Refrigeration Machines			
3 R-123 centrifugal chillers	475 T.R. @ 42 LWT & 85 ECWT	400	322.0 kW
Total Installed		2,580 hp	

3. The condenser water pumps have been sized to provide the flow required to offset the total heat rejection of the associated chiller at a 10°F (5.6°C) temperature rise.
4. The chiller pumps have been sized to provide the flow required through the evaporator of each chiller at a full-load condition and a 14°F (7.8°C) temperature rise.
5. The two primary chilled-water pumps have each been sized to provide 50% of the water flow required to offset the building's peak coincident cooling load at a 14°F (7.8°C) temperature rise. Each pump is served by a variable-frequency drive that responds to the water flow requirements of the cooling coils.
6. The three centrifugal chillers have each been sized to provide one-third of the capacity required to offset the building's peak coincident cooling load at a 14°F (7.8°C) temperature rise, from 42°F to 56°F (5.6°C to 13.4°C).

The essential elements of the design shown in Table 3 include the following:

1. The central fan systems have been equally sized and deliver air to meet the space-cooling load at 46°F (7.8°C). The supply fan in each system is served by a variable-frequency drive that responds to the airflow requirements of fan-powered VAV boxes.
2. The evaporative condensers have been equally sized and collectively have the capacity to reject 15,915 MBh (4,663.1 MW) at a condensing temperature of 95°F (35°C) DB and a 76°F (24.4°C) WB ambient temperature. The condensing temperature is allowed to fall as the ambient wet-bulb temperature drops.
3. The thermal storage pumps have been sized to provide the flow required at 34°F (1.1°C) to meet the building's coincident peak cooling load through a heat exchanger at a temperature range of 9.46°F (5.26°C).
4. The secondary pumps have been sized to provide the flow required through the chiller bundle and heat exchanger to meet the building's coincident peak cooling load at a 24°F (13.3°C) temperature rise.
5. The two primary chilled-water pumps have each been sized to provide 50% of the water flow required to offset the building's peak coincident cooling load at a 24°F (13.3°C) temperature rise. Each pump is served by a variable-frequency drive that responds to the water flow requirements of the cooling coils.
6. The three rotary screw compressors have each been sized to provide one-third of the capacity required to generate the 10,000 ton-hours (35.2 MWh) of ITES within a 12-hour period at a time-weighted average saturated refrigerant suction temperature of 24°F (−4.4°C). The same compressors are each able to produce 30% greater capacity when operating at a 38°F (3.3°C) saturated refrigerant suction temperature.

TABLE 3
ITES System—Installed Cooling Equipment

Equipment	Capacity (each unit)	HP Installed	BHP or kW
Fans			
8 supply	38,730 CFM	60	51.0 BHP
8 exhaust/return	38,730 CFM	15	13.2 BHP
Fan powered VAV	481,920 CFM total	90	102.4 kW
Evaporative Condensers			
3 sections	5,305 MBH	27.5	26.0 BHP
Thermal Storage Pumps			
2 pumps	1,800 GPM @ 50 Ft TDH	30	27.8 BHP
Secondary Pumps			
2 pumps	710 GPM @ 80 Ft TDH	20	17.0 BHP
Primary Chilled Water Pumps			
2 pumps	710 GPM @ 138 Ft TDH	40	30.0 BHP
R-22 Recirculation Package			
2 pumps	286 GPM @ 55 Ft TDH	10	7.2 BHP
Ice Tank Air Supply			
2 pumps	125 CFM @ 6.5 PSIG	7.5	6.9 BHP
Refrigeration Machines			
3 R-22 rotary screw compressors	291 T.R. @ 24 SST & 93 SDT	350	295.4 BHP
Ice-on-coil units			
8-10 ft.x 24 ft. x 12.5 ft modules	1,250 ton-hr at 24° F.	-	-
Chiller Bundle			
36" diameter x 24 Ft. long.	800 T.R @ 6° F approach	-	-
Total Installed		2,037.5 HP	

Projected System Energy Consumption

Frequently, the energy consumption of alternative conventional central cooling systems during the peak hour is used as the primary tool for evaluation of the relative energy efficiency. A percent of load output versus percent of electrical input curve is then used to evaluate part-load performance and derive annual energy consumption estimates.

While the peak-hour comparison of a conventional system versus a thermal storage system can be made, it is the author's opinion that the results will be erroneous. This is primarily due to the fact that an ITES system has four, and sometimes five, distinct modes of system operation that are selected by adaptive control algorithms to optimize performance. Each of these modes has a different coefficient of performance (COP), which can vary depending on the amount of cooling load being provided.

Selecting one peak point and mode of operation for comparison to a conventional chilled-water system disregards other ITES system modes that often increase the daily system efficiency. It is therefore necessary to compare the conventional system to the ITES system based on a full-day basis. A cooling load simulation program should be used to select the most efficient system mode at any given rate of cooling system output, while optimizing the use of storage to reduce peak electrical consumption.

The ITES system shown in Figure 2 is capable of providing more than 50% of the instantaneous capacity requirements of the building at maximum design ambient, with a compressor COP of 6.17. This is accomplished by operating two of the three compressors at full load to precool the return water prior to the ice/water heat exchanger. The chiller bundle that provides the heat transfer surface is sized to provide a close approach temperature, resulting in a refrigerant evaporating temperature of no less than 40°F (4.4°C) at peak load. Further, the saturated refrigerant discharge temperature at the compressor is held within 12°F (6.7°C) of the ambient wet-bulb temperature due to the fact that two compressors are operating on the full condensing surface that serves three compressors in the ice-building mode of operation. These factors combine to yield a low compression ratio of 2.28:1. The efficiency of the compressors during a water-chilling mode improves further at times when only one compressor operates on the full evaporating and condensing surfaces. Thus, part-load operation in an ITES system does not necessarily mean reduced efficiency. It is important to note that the location of the chiller relative to the thermal storage makes this compressor operation efficient. If the chiller was located on the leaving side of the thermal storage, then it would have to provide the 36°F (2.2°C) design chilled-water supply temperature. This location would reduce the compressor/chiller efficiency due to having to maintain a lower evaporator temperature and higher compression ratio.

Another beneficial feature of the system shown in Figure 2 is that the pumping horsepower is reduced compared to the requirements shown for Figure 1. This is primarily the result of designing the system for a 24°F (13.3°C) temperature range, with the supply water held at 36°F (2.2°C). Also, the condenser water pumps that are needed in the conventional system consume 35% more electrical power than all other pumps required in the ITES system.

Certainly a key ingredient to the overall efficiency of the ITES system lies in the comparison of the energy requirements for the building's air-moving apparatus to those same requirements in a conventional system. At maximum air delivery, the fans in the ITES design consume nearly 28% less electrical energy than the fans in the conventional system. This is due to the use of 36°F (2.2°C) water available from the ITES central system to generate supply air that has a 24% lower heat content than the air delivered from conventional systems. The supply fans in both systems are controlled by variable-frequency drives and, as a result, the percentage of energy savings remains relatively constant throughout all annual operating hours.

The combination of efficiency-based selection of the various modes of operation, low compression ratio, reduced pumping energy, and reduced fan energy can (and in this system comparison does) mean that the ITES system will consume less total annual energy than the conventional system. Tables 4 through 7 show the calculations of energy consumption during the peak hour, peak day type of operation, minimum load day type of operation, and intermediate load day type of operation.

Expansion of the data presented in Tables 4 through 7, given the operating characteristics of the systems described previously, yields an annual energy consumption for the conventional system of 2,452,107 kWh versus 2,365,911 kWh for the ITES system. Further, as shown in Table 4, the ITES system is expected to avoid 750 kW of demand charges during each month that the cooling system operates at peak load and some lesser amount during months in which the cooling system never reaches full load. Due to the electric rate structure, this demand avoidance and associated shift of electrical consumption to off-peak hours is projected to result in annual utility cost savings of $145,000 U.S. These savings are not, however, at the expense of energy efficiency.

Operating Results

The office building described in this paper was built in 1990 and included the ITES cooling system depicted in Figure 2. During the most recent 12-month period, the following information has been gathered:

1. The utility company's billing reflected a total building energy consumption for the period of 10,114,460 kWh.
2. The building's energy management system recorded a total air-handling system and central cooling plant energy consumption for the period of 1,893,130 kWh.
3. The utility company's billing indicated that the highest monthly building electrical demand was set in August at 2,368 kW. This is equal to 4.74 watts per gross square foot.
4. The utility company's billing indicated that the lowest monthly building electrical demand was set in April at 1,807 kW.
5. The electric utility charge for the period was $818,738.

These results do not necessarily validate the comparisons detailed in this paper; however, they do indicate that the ITES system consumption data used in the simulation are accurate.

Also, the company that owns and occupies this building occupies a similar building, of the same size and use, located within three miles. The cooling system in the second building is very similar to the conventional system shown in Figure 1, but it does not have a building energy management system to monitor the energy use. The only accurate information available at this time is the most recent 12 monthly utility bills, which indicate the following:

1. The utility company's billing reflected a total building energy consumption for the period of 11,695,468 kWh.
2. The utility company's billing indicated that the highest monthly building electrical demand was set in August at 3,307 kW. This is equal to 6.61 watts per gross square foot.

TABLE 4
Peak Hour Energy Comparison

Conventional System	Equipment	ITES System
	Fans	
508 kW	Supply	339 kW
122 kW	E/R	91 kW
-	VAV Boxes	102.4 kW
966 kW	Refrigeration Machines	447 kW
71 kW	Cooling Tower /Condenser	65 kW
83 kW	Primary Chilled Water Pumps	50 kW
56 kW	Secondary Chilled Water Pumps	28 kW
95 kW	Condenser Water Pumps	-
-	Ice/Water Pumps	23 kW
-	Ice Tank Air Supply Pumps	5.8 kW
-	R-22 Recirculation Package	-
-	Energy Required to Generate Storage	599 kW (.927 kW x 646)
1,901 kW	Total	1,750 kW

3. The utility company's billing indicated that the lowest monthly building electrical demand was set in March at 2,165 kW.
4. The electric utility charge for the period was $1,011,658.

Once again, these results do not necessarily validate the comparisons that were drawn in this paper; however, at a minimum, they do indicate that a building equipped with an ITES system can be more energy efficient than a similar building with a conventional design.

ITES SYSTEM IMPACT AT THE POINT OF POWER GENERATION

There is concern that widespread use of thermal energy storage (TES) technology will precipitate

- higher fuel consumption at the generating plants due to the possibility that TES systems may consume higher annual kWh and
- higher CO_2 emissions that result from
  (a) higher fossil fuel consumption due to systems consuming higher kWh and
  (b) higher fossil fuel consumption due to TES systems that shift generating requirements from daytime nuclear-based generation to nighttime coal-based generation.

Fuel Consumption

A definitive response to these concerns is elusive. The heat rate, or Btu input per kWh output, of a generator varies, with efficiency decreasing as the ambient temperature rises. Variation in the heat rate can be from 11,630

TABLE 5
Peak Day Type Energy Comparison
(15,000 T-H)

Conventional System	Equipment	ITES System
	Fans	
5,855 kWH	Supply	4,084 kWH
1,525 kWH	E/R	1,132 kWH
-	VAV Boxes	1,280 kWH
10,420 kWH	Refrigeration Machines	3,398 kWH
884 kWH	Cooling Tower /Condenser	487 kWH
981 kWH	Primary Chilled Water Pumps	625 kWH
662 kWH	Secondary Chilled Water Pumps	353 kWH
1,079 kWH	Condenser Water Pumps	-
-	Ice/Water Pumps	371 kWH
-	Ice Tank Air Supply Pumps	93 kWH
-	R-22 Recirculation Package	-
-	Energy Required to Generate Storage	9,033 kWH
21,406 kWH	Total	20,856 kWH

Btu/kWh to 7,700 Btu/kWh with changes in the ambient conditions. If it is assumed that TES systems increase annual kWh consumption, that increase would be seen at night when the ambient temperature is lower than during the day. This would appear to allow an increase in nighttime energy consumption at the point of use without increasing fuel consumption.

However, not all generating facilities experience the same variation in output, and the magnitude of the variation cannot be counted upon due to changes in weather and generating load requirements. It is, however, accurate to say that an opportunity exists to increase nighttime kWh consumption without increasing fuel consumption, but it is difficult to quantify.

Similar reasoning can be used with regard to generation and transmission efficiency losses and losses due to precipitators and other pollution control devices. All such losses are greater during the high-load periods experienced during the day. These losses would also appear to allow an increase in nighttime energy consumption at the point of use without increasing fuel consumption. Quantification of the amount of the increase that could be sustained is also difficult to obtain.

Given the fact that a TES system will consume nearly the same annual kWh, the higher generator efficiency during the nighttime hours should cause a decrease, albeit possibly slight, in generator fuel consumption.

CO_2 Emissions

From this discussion, it is assumed that there will be no increases in CO_2 emissions from a power-generating plant

TABLE 6
Intermediate Day Type Energy Comparison
(9,000 T-H)

Conventional System	Equipment	ITES System
	Fans	
4,572 kWH	Supply	3,053 kWH
1,464 kWH	E/R	1,087 kWH
-	VAV Boxes	1,280 kWH
5,769 kWH	Refrigeration Machines	-
564 kWH	Cooling Tower /Condenser	-
744 kWH	Primary Chilled Water Pumps	450 kWH
481 kWH	Secondary Chilled Water Pumps	228 kWH
768 kWH	Condenser Water Pumps	-
-	Ice/Water Pumps	276 kWH
-	Ice Tank Air Supply Pumps	70 kWH
-	R-22 Recirculation Package	-
-	Energy Required to Generate Storage	7,515 kWH
14,362 kWH	Total	13,959 kWH

resulting from increased fuel consumption attributed to TES technology.

Concerns of increased fossil fuel use relative to the use of nuclear fuel do not appear to be well founded. The mix of fuels used at any point on the generation curve varies from one utility to the next. Also, not all utilities have a nuclear base and therefore rely on the fossil fuels of coal, gas, and oil. Figure 3 depicts a 24-hour power generation curve for a utility that uses all four fuels.

It can be seen from the curve that any nighttime increase in generation will be met by loading up the nuclear capacity. The philosophy of the operators is to attempt to use the mix of fuels with the lowest cost for any given day. With the relative costs for fuel being $5.60/MW, nuclear; $9.50 to $11.00/MW, coal; $15.00/MW, gas; and $30.00, oil, any additional capacity requirement, whenever it occurs, will be met with nuclear generation if it is available.

The opposite side of the consideration is whether a decrease in the daytime generation requirement will be met with a reduction in nuclear fuel use or fossil fuel use. Once again, the economic considerations dictate that oil, gas, less efficient coal, and more efficient coal generation capacity, in that order, will be reduced prior to reducing nuclear generation.

By using the interconnecting grid that connects the generating capacity of neighboring utilities, operators regularly sell underutilized, lower cost generation capacity to other utilities to supplant their use of higher cost generation.

TABLE 7
Minimum Day Type Energy Comparison
(2,400 T-H)

Conventional System	Equipment	ITES System
	Fans	
3,906 kWH	Supply	2,640 kWH
1,464 kWH	E/R	1,087 kWH
-	VAV Boxes	1,280 kWH
1,517 kWH	Refrigeration Machines	-
426 kWH	Cooling Tower /Condenser	-
548 kWH	Primary Chilled Water Pumps	330 kWH
222 kWH	Secondary Chilled Water Pumps	170 kWH
380 kWH	Condenser Water Pumps	-
-	Ice/Water Pumps	155 kWH
-	Ice Tank Air Supply Pumps	70 kWH
-	R-22 Recirculation Package	-
-	Energy Required to Generate Storage	1,992 kWH
8,463 kWH	Total	7,724 kWH

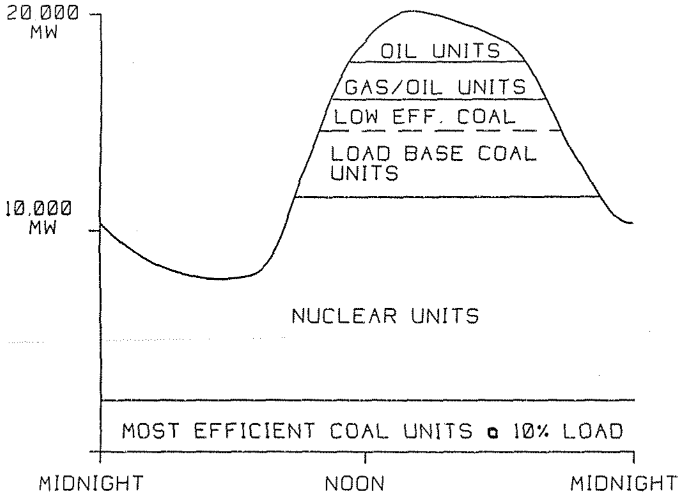


Figure 3 *Peak day power generation curve.*

If, due to TES technology, one utility experiences a reduction in daytime generation requirements, it will sell any resulting spare nuclear capacity to another utility that can the reduce its use of the higher cost fuels.

Therefore, it is not likely that greater nighttime use of electrical energy, with a corresponding reduction in daytime use, will result in a greater use of fossil fuel at the point of generation. In fact, it is evident that less fossil fuel will be used and CO_2 emissions will be reduced.

CONCLUSIONS

In order to minimize the annual energy consumption of an ITES system, these recommendations should be followed:

1. Design the system to achieve a compressor COP of 3.7

or higher when operating at the time-weighted average conditions in the storage generation mode.

2. Design the system to allow refrigerant condensing pressure to float down as the ambient temperature drops.
3. Select the storage capacity to be capable of providing 55% to 60% of the highest daily cooling load.
4. Design the chilled-water distribution system to deliver 36°F supply water to the cooling coils with a temperature range of 24°F to 28°F.
5. Design the primary air delivery system to provide 46°F to 48°F at the exit of the air-handling units at peak cooling load conditions.
6. Design the ITES system as an integral part of the building in order to maximize its capabilities within the specific application.
7. Use adaptive control logic to select ITES system modes of operation based upon maximum efficiency for the then current conditions.
8. Provide the flowmeters and temperature sensors required to record cooling system capacity output. These data are essential for the operation of the adaptive control routines and facilitate the commissioning of the system.
9. Design the storage system with inventory management capability. In an ice-based system, this means not allowing the inventory to accumulate in one area of the storage vessel. In a chilled-water storage system, this means providing effective separation between the warm and cold regions of the storage vessel.
10. Verify that system performance conforms to the design.

REFERENCES

Gillespie and Turnbull. 1991. How is your thermal energy storage system working? 14th World Energy Engineering Congress, Atlanta.

BIBLIOGRAPHY

Steam. Its generation and use. 1955. New York: Babcock & Wilcox Co.

Bauman, F.S., T. Borgers, P. LaBerge, and A.J. Gadgil. 1993. Cold air distribution in office buildings: Technology assessment for California. *ASHRAE Transactions* 99(2): 109-124.

Pearson, S.F. 1990. *Thermosyphon cooling*. London: The Institute of Refrigeration.