

The Control of Ice Storage Systems

The tradeoffs between chiller and tank capacity for different load profiles and two storage strategies are evaluated

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Air-conditioning systems that employ ice storage incorporate equipment that produces ice during one period and melts it in another period to provide cooling for the building. In designing such systems, there are two basic strategies to consider: full-load and partial-load. In a full-load strategy, the entire daytime cooling energy is met using only cooling supplied by the ice storage tank. In a partial-load strategy, the chiller and the storage are used simultaneously to meet the load.

With a full-load strategy, the tank capacity must be sufficient to meet the entire energy requirement, and the chiller capacity must be sufficient to recharge the tank during the nighttime. In a partial-load strategy, a smaller chiller and tank than that for a full-load strategy are required, and there are many combinations of the two that will meet a given building load.

The design and sizing of the components of an ice storage system depend not only on the desired strategy and total daily cooling energy but also on other factors. The maximum load dictates the amount of cooling required at any time. The cooling rate provided by an ice filled tank is not constant, but decreases as the ice inventory drops, and the time that the maximum load occurs is important. Thus, there is an interaction between the building load profile and tank size.

In addition, the flow rate of the circulating fluid through the tank and the cooling coil may limit the supply air temperatures that may be reached. Thus, the circulating fluid flow rate must be sufficient to provide the desired rates of cooling to the building.

The challenge to the design engineer is to size the components to meet the building load at all times at the lowest system cost. Effective designs must acknowledge the dynamic performance of the ice storage system.

Design approach

A simulation program was developed for the general ice storage system, as shown in *Figure 1*. The system consists of a cooling coil, temperature-controlled valve, ice storage tank and a chiller. The secondary fluid is cooled in the ice storage tank and then circulates through the coil to meet the building load.

In a partial-load strategy, the chiller provides some cooling. The temperature-controlled valve proportions the flow from the tank and chiller return to meet the set temperature of the water into the cooling coil. The water set temperature is controlled by the desired air flow outlet temperature. The models for chiller, coil and controls were taken from the *TRNSYS* software library.¹

The most common system today employs ice-on-coil storage tanks.² These are

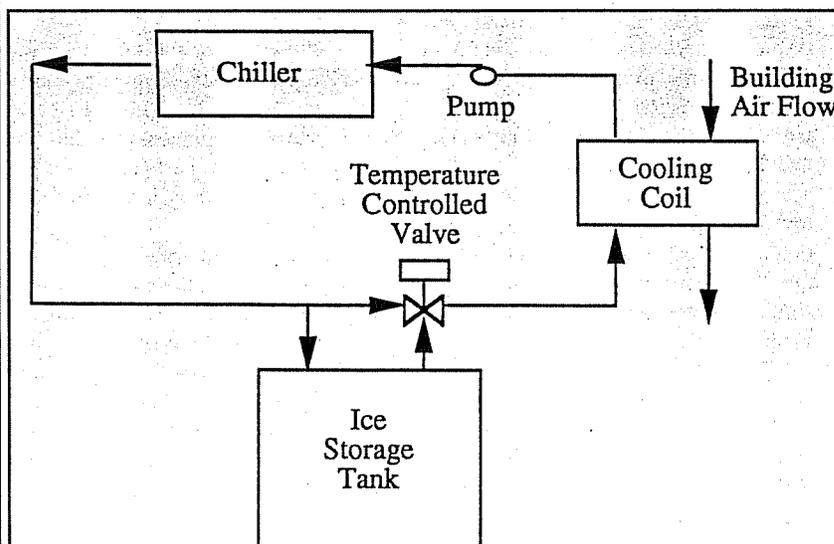


Figure 1. Typical ice storage system.

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water-filled tanks with coiled tubes that are stacked vertically. A header system provides a counterflow of secondary fluid between two adjacent coils.

During ice making, cold secondary fluid is circulated and ice builds around the coils, while during discharge, warm secondary fluid is pumped through the coils and the ice melts radially outward. The maximum rate of cooling decreases as ice melts and forms a layer of water around the tubes. The effectiveness concept³ is a convenient way to represent the performance of ice storage tanks, and was used in these simulations.

The effectiveness concept for ice storage systems is taken from heat exchanger theory, where the effectiveness is defined as the ratio of the actual heat flow to the secondary fluid to the maximum heat flow.

In an ice storage system, the actual heat flow is the secondary fluid capacity rate times the difference in temperature between the inlet and outlet. The maximum heat flow would occur if the secondary fluid outlet temperature was the lowest possible temperature, which is the freezing point temperature of water. The effectiveness is thus defined as,

$$\epsilon = \frac{T_{b_{in}} - T_{b_{out}}}{T_{b_{in}} - T_{b_{ice}}} \quad (1)$$

The effectiveness depends on the overall heat transfer conductance between the secondary fluid and the ice in the tank during discharging. The overall conductance includes the thermal resistance of the water between the tube wall and the ice. The conductance decreases as the ice melts and the layer of water builds up on the surface of the tubes.

The secondary fluid flow rate affects the effectiveness directly in that, with larger flow rates, the secondary fluid will not be cooled as much, and the tank outlet temperature will be higher. The effectiveness decreases as discharging progresses, and is lower for higher secondary fluid flow rates. During charging when ice builds up on the tube surface, there is a similar decrease in conductance with time.

The effectiveness as defined by Equation 1 was determined for the ice-on-coil tanks using a mechanistic model for ice forming and melting around the tubes. The development of the model is outlined briefly; the details are presented in Jekel.³

During discharge, the heat flow from the warm secondary fluid is given by the conductance-area product and the temperature difference between the secondary fluid and the ice. The temperature difference used is the log-mean temperature difference based on the secondary fluid inlet and outlet temperatures and the melting temperature of ice. The overall conductance-area product between the secondary fluid and the storage medium is the reciprocal of a sum of a series of thermal resistances.

The heat transfer mechanisms are convection from the secondary fluid flow to the tube wall, conduction through the tube wall, and conduction through the melted water to the ice. Average property values, temperatures and heat transfer coefficients over the tube length are employed in the model. The model uses the same mechanisms during charging, with the direction of heat flow reversed.

The heat transfer is determined as if there is an adiabatic surface halfway between adjacent tubes in the tank. This means that only one-half of the ice between tubes can be melted by one tube.

This is a good approximation early in the discharge of the tank, but becomes poorer as melting proceeds. Because the flow between adjacent tubes is counterflow, more than one-half of the ice between tubes can be melted by a tube near the entrance, and less near the exit.

The validity of this model was examined by axially subdividing the coil into a number of segments approximating the actual situation in which the heat transfer varies along the length. The effectiveness of the segmented model was slightly dependent on the number of segments.

It was found that using the entire length of a coil with average temperatures and heat transfer coefficients produces accurate results for most conditions. For low flow rates, outside the range of operation of the tanks that were modeled, more than one segment may be required. The single length model is adequate and is used in this study.^{4,5}

The effect of time in the discharging or charging of the ice storage tank was represented through the fraction of the tank capacity that was discharged or charged. The maximum tank capacity during discharge is the latent capacity plus the sensible capacity based on heating all of the tank water to the secondary fluid inlet temperature.

$$C_{max} = m_i [h_{if} + c_w (T_{b_{in}} - T_{ice})] \quad (2)$$

The amount of energy transferred to the ice (discharged capacity) is determined by integrating the rate of energy removal over time. The inlet and outlet secondary fluid temperatures and the secondary fluid flow rate are used to determine the rate of change in tank capacity with time. Heat losses to the surroundings are assumed negligible.

$$\frac{dC}{dt} = -\dot{m} c_{p,b} (T_{b_{in}} - T_{b_{out}}) \quad (3)$$

where C is the capacity remaining in the tank at any time. Equation 3 was solved numerically using a finite difference approximation for the derivative. With the inlet secondary fluid temperature, the instantaneous capacity remaining and the flow rate known, the effectiveness can be determined.

Once the effectiveness is known, the outlet secondary fluid temperature can be found using Equation 1. The discharged capacity is the total capacity minus the amount of capacity C remaining in the tank, and the fraction discharged is defined as,

$$\text{Fraction Discharged} = \frac{C_{max} - C}{C_{max}} \quad (4)$$

The effect of building load profile was studied by using profiles that were constant, increased and decreased linearly with time and that represented actual building loads. All of the loads had the same total integrated daily load and the same time period.

A variable air volume (VAV) system was modeled with the air flow rate determined by the load and set air temperatures out of and into the cooling coil of 75°F (24°C) and 60°F (16°C), respectively.

The controller determined the required temperature of the secondary fluid entering the cooling coil. The secondary fluid flow rate through the coil was constant and the required coil inlet temperature was attained by controlling the proportion of the flow through the tank. The hourly values for ambient air temperature

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and relative humidity were based on the design day for Madison, Wisconsin.⁶

Full-load strategy results

In the full-load strategy, the tank is charged during off-peak periods (night-time) and discharged during on-peak periods (daytime). The chiller runs only during the off-peak period. One-day simulations were performed to determine the minimum tank size that would meet the load under this strategy.

Because the tank effectiveness is continually changing during discharge, it is not possible to explicitly determine the required tank size *a priori*. An iterative process was used in which an initial guess for the tank size was made and then a simulation performed.

If the required effectiveness to meet the load exceeded that available from the tank at any time during discharge, the tank capacity was too low. The capacity was then increased until the tank was able to meet all hourly loads.

For example, the critical load for the linearly increasing load profile is the load at the end of the day when the rate of discharge is highest. For the minimum tank size that can meet this load, the flow through the tank is the maximum value and there is no bypass flow.

A commercially available ice storage tank with a capacity of 190 ton-hrs (670 kWh) was selected. It was able to just meet

the constant load over the 10.25-hour period at a secondary fluid flow rate of 40,000 lb/h (5 kg/s). This tank was chosen as the base for comparison and other tank sizes were normalized to its capacity.

For the linear profiles that increase with time, the required relative tank size is always greater than unity, and increases as the slope of the load profile increases. For increasing loads, significantly larger tanks are required because the tank effectiveness decreases throughout the discharge time, and the performance is most critical at the end of the day.

The effect of the secondary fluid flow rate on the required tank size is small for decreasing or constant loads, but is important for increasing loads over the range of secondary fluid flow rates evaluated. For a decreasing load profile, the relative required tank size is always less than unity and does not change significantly with load profile slope.

The load is largest when the tank effectiveness is very high, and because both the tank effectiveness and the load decrease with time, a smaller tank is required. For the linearly increasing loads, the effectiveness is low near the end of the day and either larger secondary fluid flow rates or larger tank sizes are required to meet the critical load.

A diversity factor (which is the ratio of the average load to the peak load for the design day) is sometimes used in design.² However, the diversity factor does not

account for the time at which the load occurs. A linearly increasing load and a linearly decreasing load can have the same diversity factor, but require drastically different tank sizes.

The representative load profiles are the same shape and have diversity factors of 0.83, 0.67 and 0.62. For the profile with a diversity factor of 0.83, a relative tank size of unity (190 ton-hrs; 670 kWh) is able to meet the load at all times during the day.

However, for the load with a diversity factor of 0.67, a relative size of 1.14 (217 ton-hrs; 760 kWh) is needed to meet the load. For the building load profile with a diversity factor of 0.62, a tank of relative size of 2.44 (464 ton-hrs; 1630 kWh) is required.

As with the linear load profiles, the critical element in sizing the tank is the time that the maximum load occurs. The effect of secondary fluid flow rate on the required tank size was similar to that for the linearly increasing load profiles; to some degree, increasing secondary fluid flow rate allowed smaller tank sizes.

The required tank sizes for all of the load profiles simulated are plotted against diversity in Figure 2. The diversity factor does not correlate the different profiles, especially at low values.

The time at which the peak load occurs is critical in sizing. For the linearly decreasing profiles, the peak loads are at the start of the day, and the required tank size actually decreases with decreasing diversity. In contrast, the required tank size increases with decreased diversity if the loads occur late in the day.

The linearly increasing loads have a maximum at the very end of the day, while the peak for the representative load profiles is two hours before the end. As a result, the required tank sizes are larger for the linear loads than for the representative loads.

In a full-load strategy, the ability of an ice tank in meeting the load is not ensured even though the tank has enough ice to meet the total integrated load. Different building load profiles that produce the same total integrated load over the same time span require different size tanks.

Because the cooling rate potential of an ice storage tank decreases as the ice melts, higher loads can be better met when they occur early in discharge when the tank effectiveness is high.

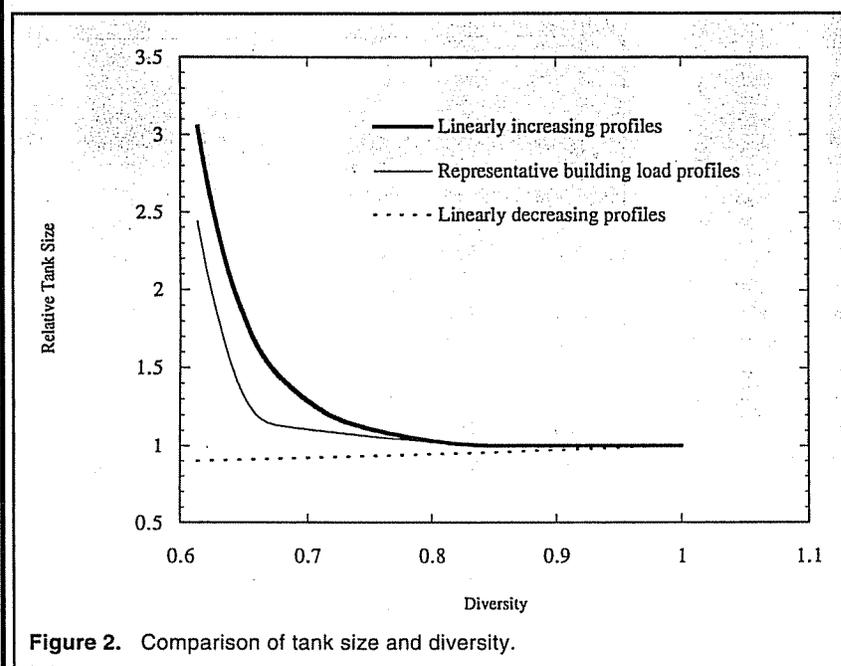


Figure 2. Comparison of tank size and diversity.

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Partial-load strategy results

In a partial-load strategy, the chiller and ice storage tank are used together in meeting the load, and the chiller charges the tank when there is no load. As a result, both charging and discharging periods must be analyzed.

Using the simulation methodology, the tank and chiller were sized for the same loads as in the full-load strategy. This was an iterative process that involved selecting a chiller capacity and then determining the minimum tank size.

The critical factor on tank capacity was the rate of cooling provided at the maximum load. The chiller capacity is that needed to recharge the tank. A three-day simulation was performed so that the initial ice storage tank charge had no effect.

Various combinations of chiller and ice storage tank size are possible for each load profile. The minimum required chiller capacity is one that allows charging the tank by operating at full load during the entire 12-hour nighttime period. If the chiller is smaller than this minimum size, it will not be able to fully recharge the ice storage tank for use on subsequent days.

The practical maximum chiller capacity is that required to meet the peak load without any storage. These chiller capacity limits also place limits on the required tank sizes.

The performance for different tank sizes and chiller capacities is shown schematically for the building load with a diversity of 0.83 in *Figure 3* for a supply air outlet temperature of 60°F (16°C). The chiller operation is also shown to indicate that, as the chiller size increases, the tank size decreases and the chiller operates during the off-peak period less of the time. There is a minimum chiller size required to charge the tank and a maximum size that meets the peak load.

Three different air outlet temperatures of 50°, 55° and 60°F (10°, 13° and 16°C) were simulated. For small tank capacity, essentially the same value was found to be required for all delivered air temperatures. It is only near the minimum chiller capacity that a slightly (5%) larger tank is required for the colder air temperatures. This is a result of the reduced actual capacity and higher effectiveness required for lower supply temperatures.

The effect of circulating secondary fluid flow rate was most pronounced at the

low chiller sizes. For flows less than 25,000 lb/h (3 kg/s), it is not possible to design a system to meet the building load in this example.

The system sizing for the other two representative building load profiles was also performed. The relations between tank size and chiller capacity are shown for all three representative load profiles for an air outlet temperature of 55°F (13°C) and a flow rate of 60,000 lb/h (8 kg/s) in *Figure 4*.

All profiles require the same minimum chiller size, but the tank size at the minimum chiller size depends on the profile. The maximum chiller size (at zero tank capacity) is different for each profile because the maximum load is different. A different tank size is required at the same chiller capacity for each profile shape.

The results presented in *Figure 4* illustrate the complex interrelation between chiller capacity, tank size, circulating fluid flow rate, and supply air temperature.

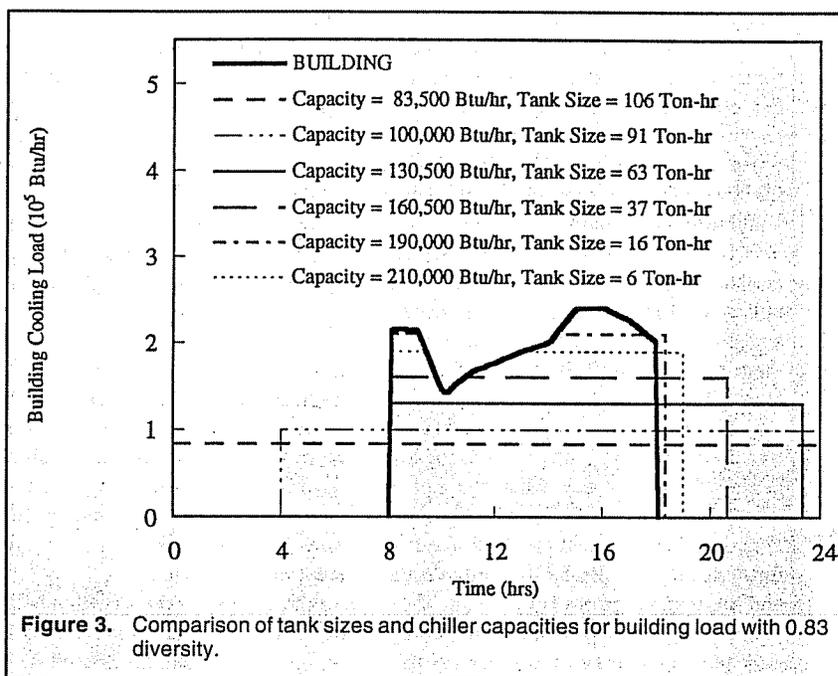


Figure 3. Comparison of tank sizes and chiller capacities for building load with 0.83 diversity.

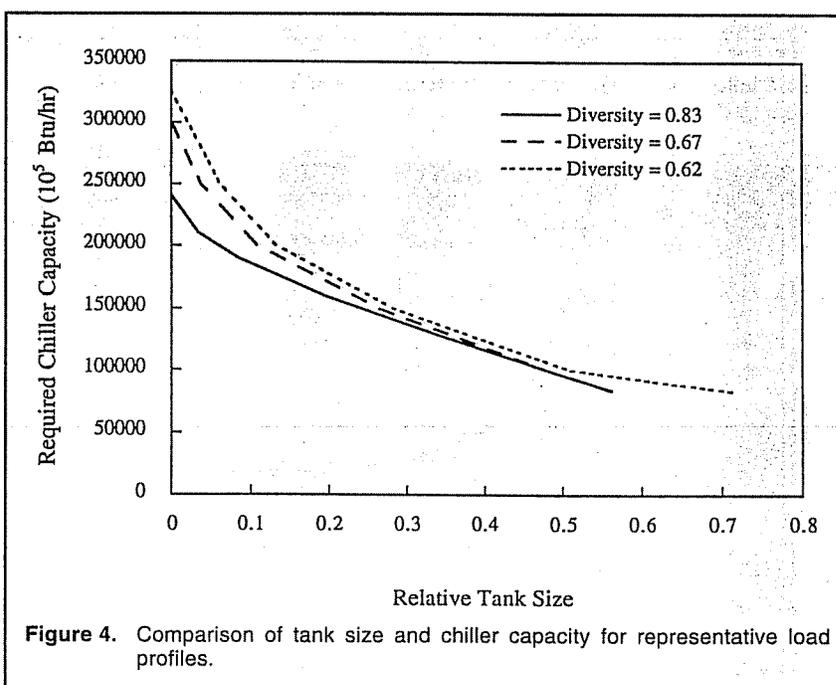


Figure 4. Comparison of tank size and chiller capacity for representative load profiles.

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There is a critical mass flow for each load profile. At flow rates greater than this value, the tank size does not depend on flow rate. Below this value, the coil performance is the limiting factor, and too low a flow rate will not allow the desired supply air temperature to be attained. These results were also found for the linear load profiles.

In the sizing of tanks and chiller under a partial-load strategy, the diversity factor is only a general indicator of the required tank capacity. At small capacities, the maximum load dictates the required chiller size. The maximum load is represented exactly by the diversity factor and, at small tank sizes, the diversity factor is a useful indicator.

The minimum chiller capacity is independent of the diversity factor and, at small chiller capacities, the profile shape is important. To satisfactorily design a partial-load system, it is important to know the desired profile shape accurately. A short-term (three-day) simulation can accurately size the chiller and tank.

System comparison

Comparisons were made between the component capacities for a conventional chilled water air-conditioning system, a full-load ice storage system, and a partial-load ice storage system for all three representative building profiles.

The required chiller and tank capacities are given in *Table 1*. For the partial-load system, the values are for the maximum tank capacity (minimum chiller capacity).

The chiller for the full-load system is 70% of that for the conventional system, for the building load profile with a diversity factor of 0.83. The chiller required by the partial-load strategy is 34% of the size of the chiller required in a traditional chiller

air-conditioning system, and 58% of that for the full-load system. The tank in a partial-load system is 56% of that for the full-load system.

For the building load with a diversity of 0.67, the chiller size for the conventional system is substantially larger than that required for the building load with a diversity factor of 0.83. The chiller size required in the ice storage systems is independent of diversity.

The chiller size for the conventional system is determined by the peak load, while the chiller size for the ice storage systems is determined by the recharging of the ice storage tank. The integrated load is the same for both profiles and thus the energy delivered from the tank is the same.

The ice storage tank capacity for the full-load system increases because of the decreasing tank effectiveness during discharge. The tank for the partial-load system is the same because the chiller and ice storage tank work together in the partial-load strategy. The effect of a change in load profile on the ice storage tank size is less than in the full-load system.

Similar results are found for the representative building load with a diversity of 0.62. For the full-load system, the required ice storage tank size is more than double that required with a diversity factor of 0.67. The ice storage tank size for the partial-load system is larger by 30%. The use of the chiller and ice storage collectively to meet the load in the partial-load strategy reduces the dependency of ice storage tank size on the load profile.

Conclusion

In a full-load strategy, the required size of the ice storage tank depends on both the total energy depleted from the tank during discharge and the load profile. There is a minimum chiller size, which is

that needed to recharge the tank at nighttime. Load profiles with high loads near the end of the day require larger ice storage tanks than profiles with high loads near the beginning of the day.

In a partial-load strategy, the building load profile also has a strong effect on the required chiller capacity and tank size. For any load profile, there is a maximum and a minimum required chiller capacity. The maximum capacity corresponds to a zero tank size, and is the capacity required to meet the maximum load. The minimum capacity is for a condition in which the chiller operates continuously and the tank size is the largest.

The chiller helps meet the building load during daytime and charges the storage tank during nighttime. The limit on minimum chiller capacity is that the chiller must be large enough to recharge the tank during nighttime.

In a partial-load strategy, there are many possible combinations of tank size and chiller capacity for any given load profile. For tank sizes smaller than the maximum, the chiller will operate only partially at night. For a given chiller capacity, the particular values of tank capacity depend on the profile shape, and the diversity factor is only a general indicator of the relative size.

Large tanks are required for profiles with large loads near the end of the day, while smaller tanks are sufficient when the loads are smaller near the end of the day even if the integrated load and diversity factors are equal. Further, there are complex interactions between the chiller capacity and tank size depending on circulating fluid flow rate and supply air temperature.

Chiller size is highly dependent on the peak load for a conventional air-conditioning system. In contrast, the chiller size for an ice storage system is dependent on the energy supplied by the ice storage tank. The chiller size is dictated by the need to recharge the storage tank and, for a partial-load strategy, to supplement the cooling delivered by the tank.

The peak load is met by the storage tank with the chiller providing a base load. Ice storage systems significantly reduce the required chiller sizes. The chiller and ice storage tank sizes for the partial-load system are almost one-half those required by a full-load system.

The ability of an ice storage tank to meet a load is highly rate dependent. Simu-

Table 1. Chiller Capacity and Ice Storage Tank Sizes

Load Profile Diversity	Conventional System	Full-Load System	Partial-Load System
0.83 Chiller Cap. (Btu/h)	240,000	143,000	83,300
Tank Size (ton-hr)	0	190	106
0.67 Chiller Cap. (Btu/h)	300,000	143,000	83,300
Tank Size (ton-hr)	0	217	106
0.62 Chiller Cap. (Btu/h)	325,000	143,000	83,300
Tank Size (ton-hr)	0	464	135

lation techniques may be used to advantage in design procedures. A three-day analysis is sufficient to eliminate initial effects and to properly size system components. ■

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References

1. Klein, S., et al. 1994. *TRNSYS: User's Manual*. Madison, Wisconsin: University of Wisconsin. Version 13.1.
2. Calmac. 1987. *Levload Ice Bank Performance Manual*. Englewood, New Jersey: Calmac Manufacturing Corporation. April.
3. Jekel, T., et al. 1992. "Operational strategies for reducing coil loads." *ASHRAE Transactions*. Atlanta, Georgia: ASHRAE. Vol. 98, Pt. 1, pp. 919-925.
4. Carey, C. 1993. *The Optimal Control of Ice-Storage Air-Conditioning Systems*. Masters thesis. Madison, Wisconsin: University of Wisconsin.
5. Kawashima, M. 1992. Personal communication.
6. ASHRAE. 1993. *ASHRAE Handbook-Fundamentals*. Atlanta, Georgia: ASHRAE.

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