CHAPTER
# ONE

## INTRODUCTION

An ice-storage air-conditioning system involves making ice at night and melting the ice for cooling during the day. The driving force in using an ice-storage system is that some electric companies charge a fraction of the daytime rate for usage during the night to try to reduce peak demand. By reducing peak demand, the utility may not need to fire up a less efficient gas fired generator and possibly not have to build more power plants. An additional incentive is often implemented in the form of utility rebate on the first cost of the ice system.

Although the operating cost can be substantially lower for an ice-storage system, the efficiency of an ice-storage system is almost always lower than the efficiency of a traditional chiller at full load. This is because the chiller COP associated with making ice is lower than the COP associated with cooling water. In addition, there are tank losses to the ambient conditions.

By improving tank design, system design, and control strategies, the efficiency of the system can be improved and the costs can be further lowered.

There are two strategies in ice-storage systems. In a full load strategy, the entire daytime load is met using only the cooling capacity of the ice-storage tank. In a partial load strategy, the load is met using both a chiller and an ice-storage tank. The partial load strategy is often used because the ice-storage system has a chiller to freeze the water in the ice tank. In addition, a partial load strategy requires smaller chiller and tank sizes relative to the sizes required in a full storage to meet the same given load, and thus, a lower first cost. One strategy in using a chiller is to completely discharge all the ice during the day to minimize daytime chiller energy.

Within the partial load strategy there are two different methods of operation. Chiller priority designates usage of a chiller to meet a building load and usage of the ice-storage tank to meet any additional load requirements above this base load. Ice priority designates usage of the ice-storage tank to meet the building load first. If the load for the day cannot be met using the ice, the chiller is run. When the ice priority system is able to meet the entire load without the chiller, the system operates the same as a full load strategy.

## 1.1 SCOPE OF STUDY

Jekel [1] developed a one dimensional model for an ice-storage tank, and average property values, film temperatures, and surface temperatures were assumed over the entire tube length. In this thesis the discretization of the tank coils into finite lengths was performed. The model was then run at different flow rates, different brine inlet temperatures, and with varying numbers of finite lengths. The convergence of the outlet brine temperature with increasing disretization was studied. The resulting data was plotted against the Calmac performance data to determine how many discretizations of the coils are needed to obtain agreement between model and experiment.

A model of the ice-storage tank in a full storage strategy was developed. The model for the tank used effectiveness performance curves developed by Jekel. A design procedure using TRNSYS was developed that allows sizing of the tank to meet a given building load profile. The effect of different load profiles on tank size, the impact of cooling due to ice tank depletion, and the sizing of both the cooling coil and ice-storage tank to meet various load profiles were determined.

A TRNSYS program was developed to model a partial storage configuration during charging and discharging of an ice-storage system. The model for the tank used performance curves developed by Jekel. The effect of various methods of chiller operation on the performance of the ice-storage tank during discharge and the optimum sizing of both the chiller and the ice-storage tank during both charging and discharging were studied. A comparison of the partial load strategy to the full load strategy was made with respect to chiller and ice-storage tank sizes.

## 1.2 Calmac Ice-on-Coil Storage Tank

The model for an ice-on-coil tank was based on the Calmac 1190 ice -storage tank [2]. This is a cylindrical tank with tubes that coil axially. The coils are stacked vertically. A header system that allows counterflow between two adjacent coils is built into the tank. The brine that flows through the coils is a 25%-75% mixture of ethylene glycol and water by volume.

Ice builds around the coils by pumping brine at a temperature less than the freezing point of water. During discharge the ice melts to cool the warm brine being pumped through the coils. The ice melts radially outward leading to a water formation diameter.

## 1.3 The Effectiveness Concept

Jekel developed the idea of an ice-storage tank effectiveness [1]. The effectiveness is defined as the ratio of the actual heat flow to the maximum heat flow. The actual heat flow is the brine flow rate times the difference in temperature of the inlet and outlet brine temperatures. The maximum heat flow would occur if the brine outlet temperature were the freezing point temperature of water. The effectiveness equation (eq 1.1) works for both charging and discharging. The true effectiveness would be defined in the same terms except the freezing point temperature of water would be replaced with the ice tank temperature (eq 1.2). Although the effectiveness developed by Jekel is not the true effectiveness, it is correct to use if it is used consistently. The effectiveness, defined by Jekel, indicates the basic physical phenomena that occur in the discharge of an ice-storage tank.

$$\varepsilon = \frac{Tb_{in} - Tb_{out}}{Tb_{in} - T_{ice}} \qquad (1.1)$$

$$\varepsilon = \frac{Tb_{in} - Tb_{out}}{Tb_{in} - T_{tank}} \qquad (1.2)$$

# *REFERENCES 1*

1. Jekel, Todd, "Modeling of Ice-Storage Systems," M.S. Thesis, University of Wisconsin - Madison, December 1991.

2. *Levload Ice Bank Performance Manual*, Product Literature, Calmac Manufacturing Corporation, Englewood, New Jersey, April 1987.

# CHAPTER TWO

## TANK MODEL

In Jekel's master's thesis [1] a mechanistic model of the Calmac 1190 series tank was developed. The thesis gives, in detail, the model relations used to predict perfomance. These are available in Appendix A.

During discharge, the conductance area product between the brine and the storage medium is based on a series of thermal resistances of convection from the brine flow, conduction through the tube, conduction through the ice, and convection to the water (eq 2.1, Jekel).

$$UA_{tot} = \left[ \frac{1}{A_i\, h_b} + \frac{\ln (D_o/D_i)}{2\, \pi\, k_{tube}\, L} + \frac{\ln (D_w/D_o)}{2\, \pi\, k_w\, L} + \frac{1}{A_w h_w} \right]^{-1} \qquad (2.1)$$

The convection coefficient for the internal brine flow is obtained from the Nusselt number (eq 2.2).

$$h_b = \frac{\overline{Nu_D}\; k_b}{D_i} \qquad (2.2)$$

If the brine flow is turbulent, the Nusselt number is found using equation 2.3, the Dittus-Boelter correlation for a circular tube [2].

$$\overline{Nu_D} = .023\, Re_D^{4/5}\, Pr^{4/10} \qquad (2.3)$$

Equation 2.4, a relation for constant wall temperature that takes into account developing thermal and hydrodynamic boundary layers, is used if the flow is laminar [3].

$$\overline{Nu_D} = 3.66 + \frac{.0534\ (Re_D\ Pr\ D / L)^{1.15}}{1 + 0.0316\ (Re_D\ Pr\ D / L)^{.84}} \qquad (2.4)$$

In this model, the tube within the tank was analyzed by assuming that film temperatures, surface temperatures, and convection coefficients were constant over the tube length. The tube analyzed is 250 feet long. Physically, this is very long, but a more relevant question is whether it is thermally long. A measure of the thermal length is the number of transfer units, which is related to the overall heat transfer coefficient, the area, and the minimum capacitance (eq 2.5).

$$NTU = \frac{UA}{C_{min}} \qquad (2.5)$$

To examine this question, the Jekel model was reconfigured as a subroutine and a main program was developed to analyze the tubes within the tank by discretizing the tubes into finite lengths. This program calculates new film temperatures, surface temperatures, tank temperatures, water formation diameters, and convection coefficients for each finite length. Average property values are calculated along each finite length.

In solving for the inlet and outlet brine temperatures of the finite lengths, there are two useful pieces of information available. The inlet brine temperature to the first finite length is simply the inlet brine temperature given. The inlet brine temperature for all subsequent finite lengths is simply the outlet brine temperature of the previous finite length.

The heat flow from the warm brine solutions is the product of the conductance-area product and the temperature difference (eq. 2.6). The conductance varies due to the change in diameter of the ice-water interface. The temperature difference used is the log-mean temperature difference.

$$Q = UA(LMTD) \quad (2.6)$$

## 2.1 Developing Equations of Finite Tank Model

In Jekel's model, an initial outlet temperature guess is given and the model solves iteratively for the correct outlet temperature. For the finite length analysis, a single coiled tube is divided lengthwise into a given number of finite lengths and each outlet temperature is found using the same iteration scheme. The log-mean temperature difference for each finite length is:

$$LMTD(i) = \frac{\left[[T_{ice}\text{-}T(i+1)]\text{-}[T_{ice}\text{-}T(i)]\right]}{\log\left[\frac{[T_{ice}\text{-}T(i+1)]}{[T_{ice}\text{-}T(i)]}\right]} \quad (2.7)$$

where LMTD(i) = Log-mean temperature difference for a finite length

T(i)= Inlet brine temperature to the finite length

T(i+1)=Outlet brine temperature from the finite length

Tice=Freezing point temperature of ice

The temperature of the ice-water mixture for each finite length are calculated as if there is an adiabatic wall between adjacent tubes in the tank. By assuming there is an adiabatic wall halfway between adjacent tubes, 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 worsens as most of the ice in the tank has melted. In the case of counterflow, which really occurs in the tank, more than one half of the ice between tubes can be melted by a single tube since the tubes are coiled with the inlet of one next to the outlet of the other.

## 2.2 Convergence of Outlet Brine Temperature

In figure 2.1 the brine outlet temperature is plotted against time for a flow rate of 20 GPM at three levels of discretization. Convergence is attained when the change in outlet brine temperature with an increase in dicretization is negligible. The difference in brine outlet temperature between using four finite lengths and using six finite lengths is negligible, so for this case, four finite lengths are required for convergence.

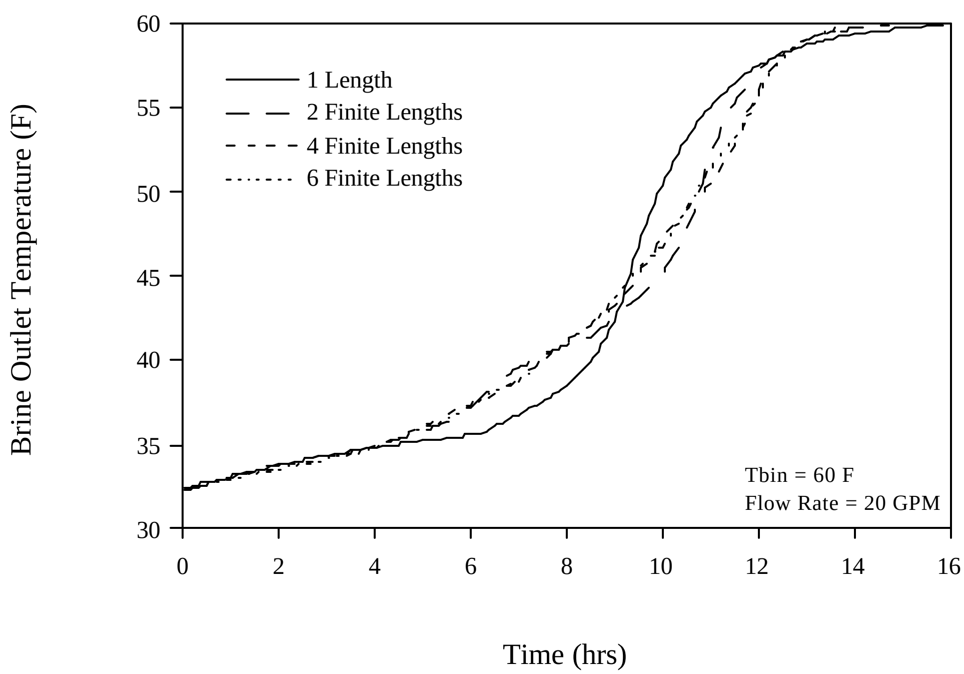


**Figure 2.1** Brine outlet temperature as a function of time of discharge for a flow rate of 20 GPM at various levels of discretization.

In figure 2.2 the brine outlet temperature is plotted against time for a flow rate of 100 GPM at three levels of discretization. Figure 2.2 shows that, for the higher flow rate, the outlet brine temperature is not affected appreciably by discretization. It is interesting to note that several finite lengths are needed when the flow rate is small (20 GPM) and one length is sufficient when the flow rate is large (100 GPM).

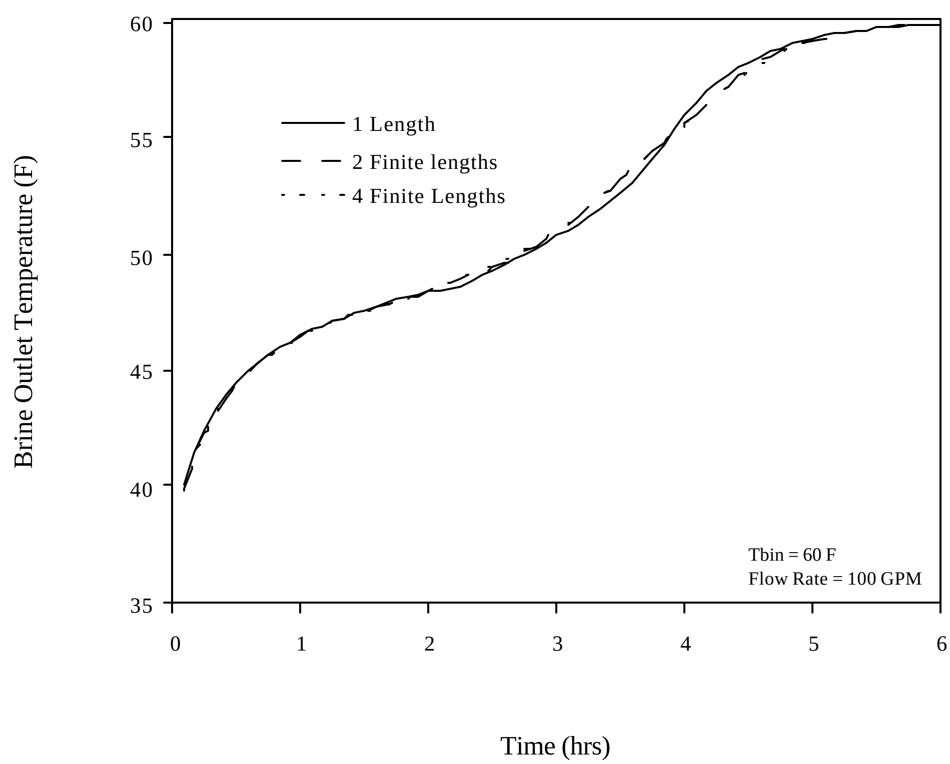


**Figure 2.2** Brine outlet temperature as a function of time of discharge for a flow rate of 100 GPM at various levels of discretization.

The reason for the difference in results for figure 2.1 and figure 2.2 is due to the capacitance of the brine. The number of transfer units (Ntu) is inversely proportional to the brine capacitance. In figure 2.3 the maximum number of transfer units for a finite length is plotted against the level of discretization for various flow rates. The number of transfer units increases with decreasing brine flow rate, and the maximum Ntu of a finite length to achieve good results is about 1.5. When the maximum Ntu of a finite length is less than 1.5, further discretization does not significantly change the maximum Ntu of a finite length for any flow rate. For the 20 GPM flow, four finite lengths were required for convergence (fig 2.1) which corresponds to a maximum Ntu of a finite length of about 1.5. For the 100 GPM, one length was enough convergence (fig 2.2) which corresponds to a maximum Ntu of a finite length of about 1.5. Thus, the maximum value of 1.5 for the maximum Ntu of a finite length is verified.

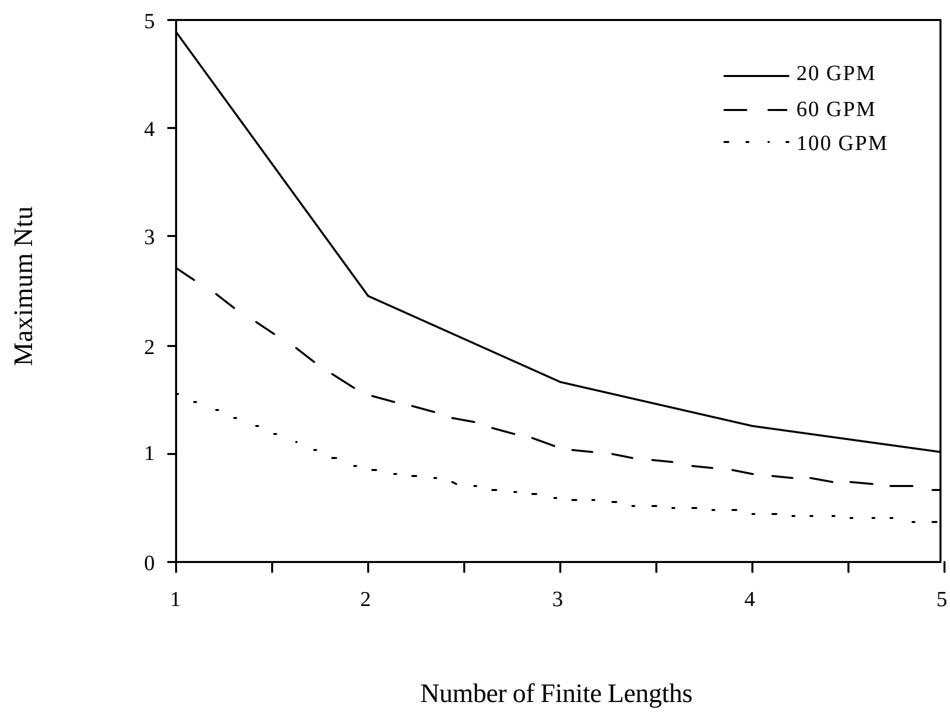


**Figure 2.3** The maximum finite Ntu vs number of finite lengths for various flow rates.

## 2.3 Convergence and Comparison to Calmac Data

In using the finite length method, the discharge of the tank was performed at different flow rates and brine inlet temperatures. The number of finite lengths was increased until a satisfactory convergence of the outlet brine temperature was reached. Satisfactory convergence is attained when an increase in dicretization changes the outlet brine temperature negligibly. Another way to study the convergence of the outlet brine temperature is to study the convergence of the tank effectiveness. The tank effectiveness for a fixed flow rate and a fixed inlet brine temperature is only a function of the outlet brine temperature (eq 2.8).

$$\varepsilon = \frac{Tb_{in} - Tb_{out}}{Tb_{in} - T_{ice}} \qquad (2.8)$$

In figures 2.4 through 2.13 the tank effectiveness has been plotted for each combination of flow rate and inlet brine temperature. As can be seen in figures 2.4 through 2.13, the number of finite lengths required for convergence depends on the flow rate and the inlet brine temperature; but mainly the flow rate. At very low flow rates the number of finite lengths required is larger than for higher flow rates. Physically, at lower flow rates, the flow is laminar, thus leading to a more thermally stratified flow along the length of the tube. At very high flow rates the heat loss from the brine over the finite length is very high. This means that the water formation diameter does not vary as much along the tube when compared to low flow rates, and discretization is not needed at higher flow rates.

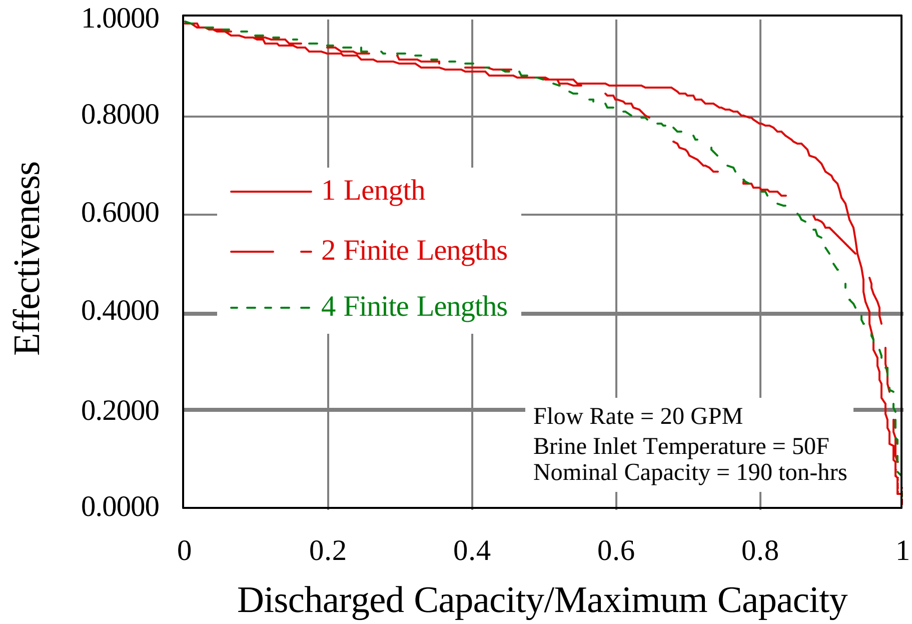


**Figure 2.4** Effectiveness as a function of discretization for 20GPM and $50^0$F

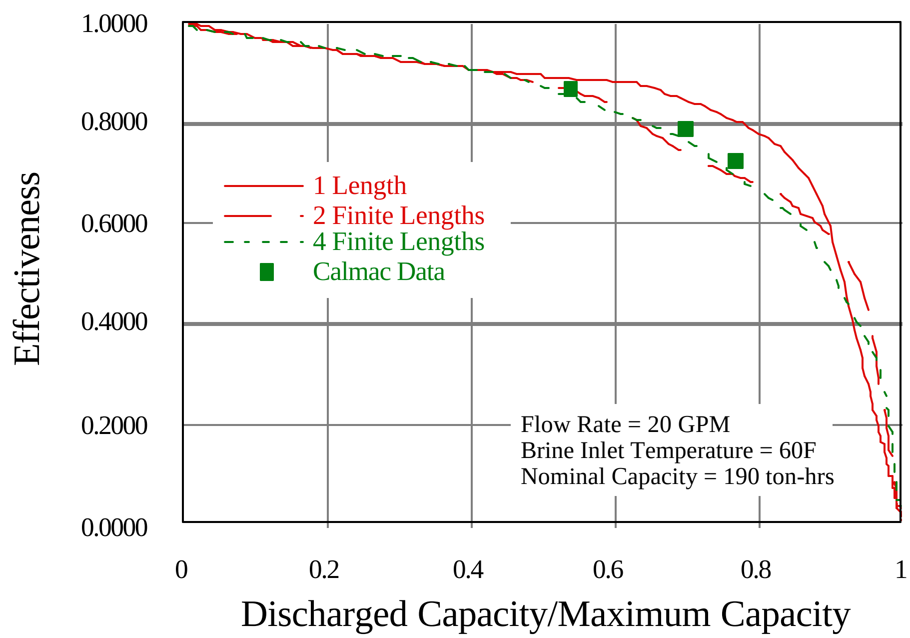


**Figure 2.5** Effectiveness as a function of discretization for 20GPM and $60^0$F

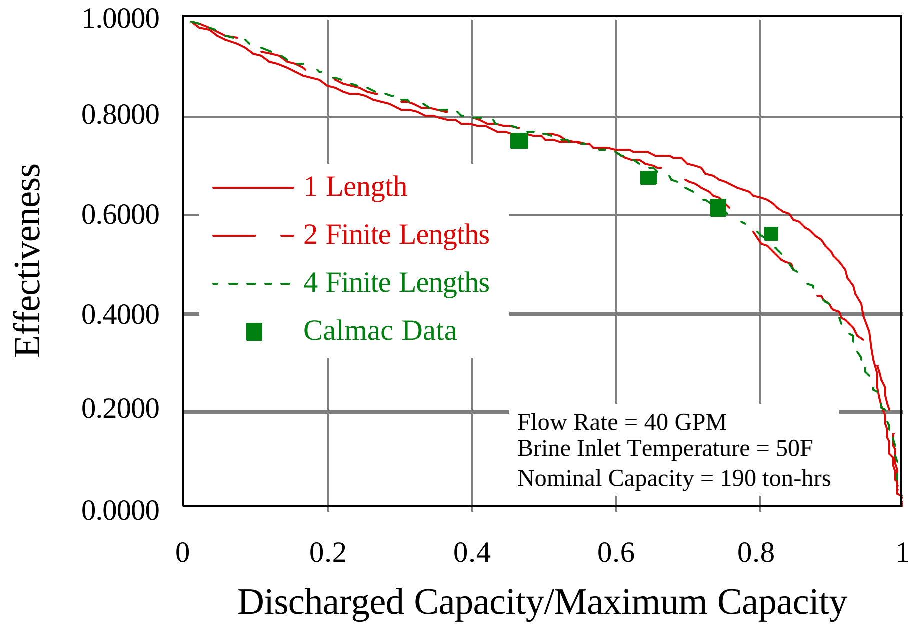


**Figure 2.6** Effectiveness as a function of discretization for 40GPM and $50^0$F

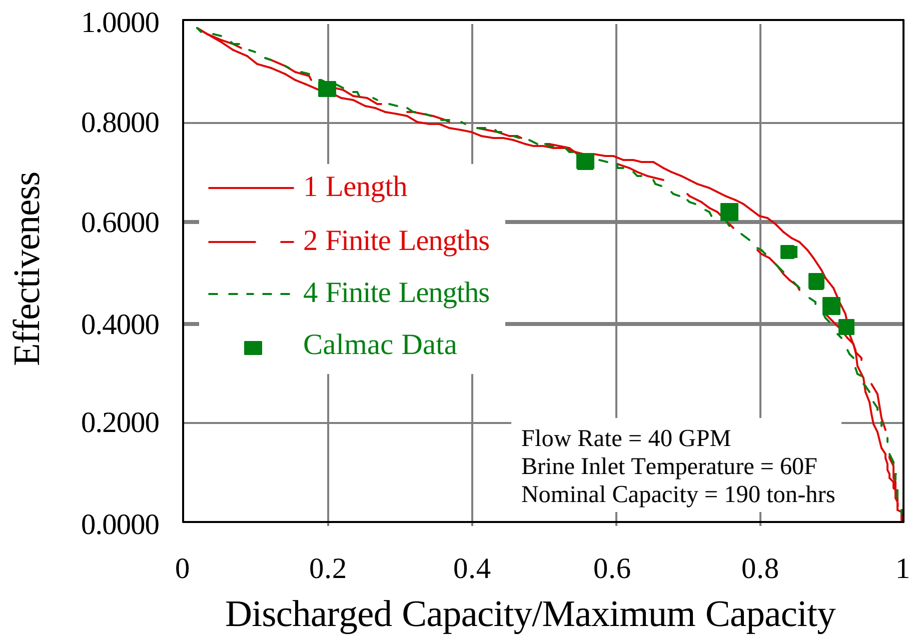


**Figure 2.7** Effectiveness as a function of discretization for 40GPM and $60^0$F

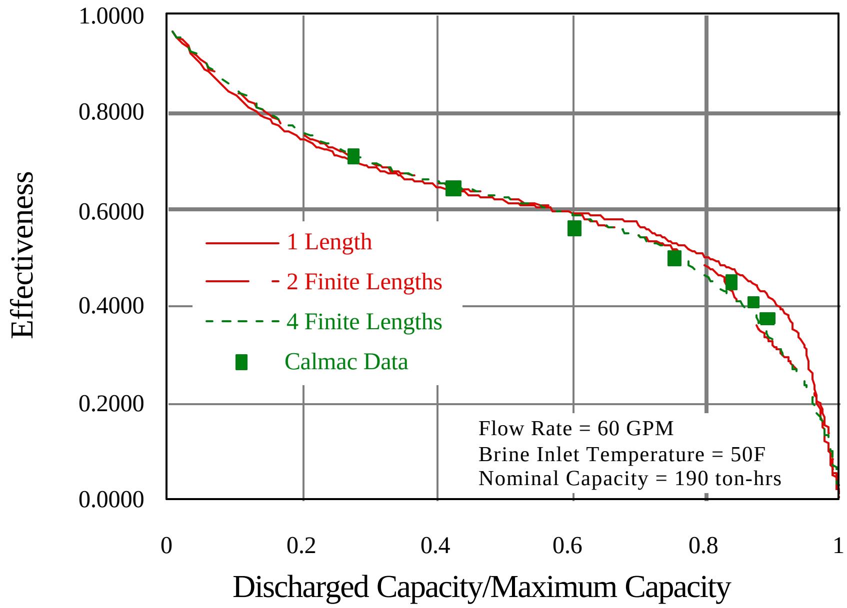


**Figure 2.8** Effectiveness as a function of discretization for 60GPM and $50^0$F

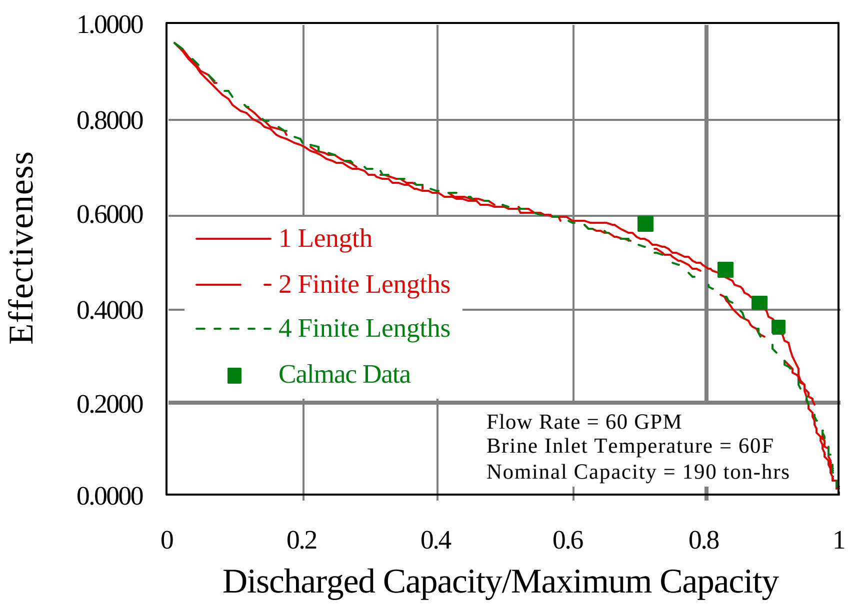


**Figure 2.9** Effectiveness as a function of discretization for 60GPM and $60^0$F

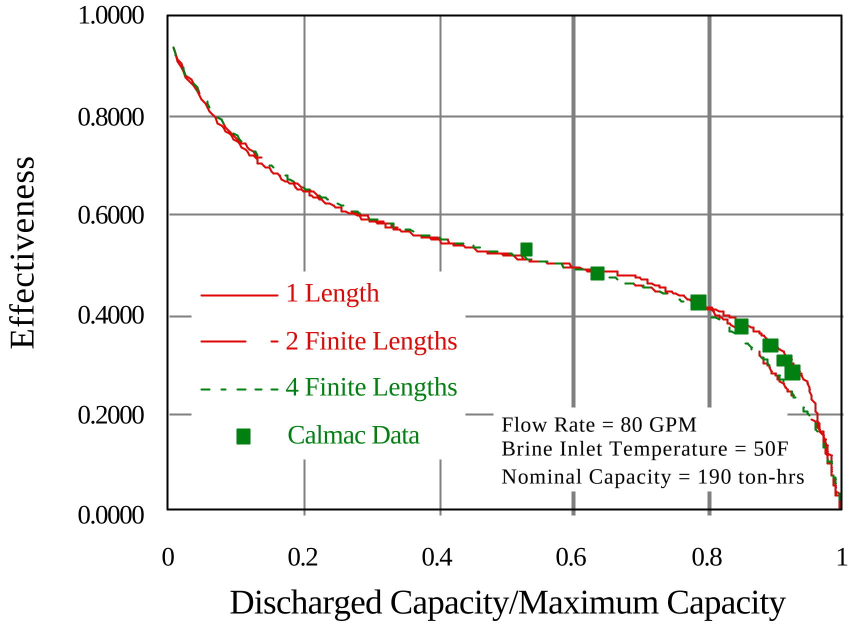


**Figure 2.10** Effectiveness as a function of discretization for 80GPM and $50^0$F

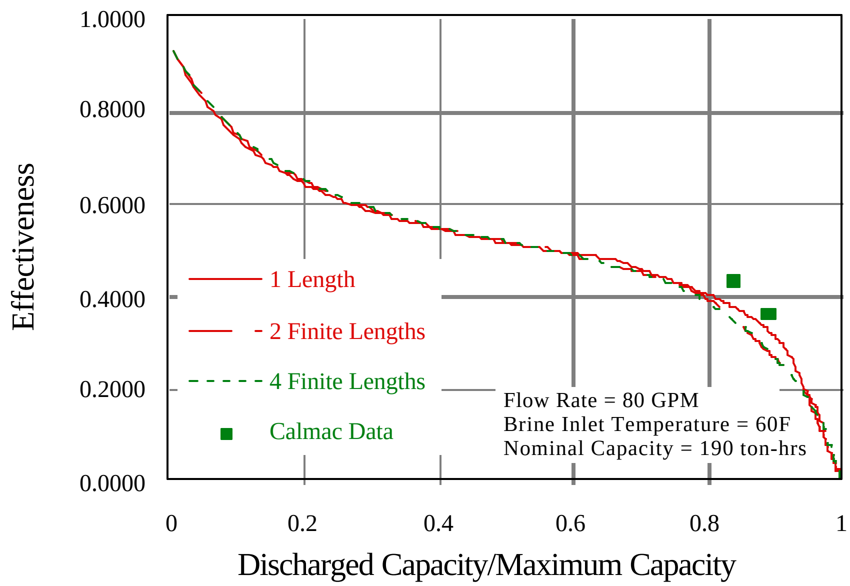


**Figure 2.11** Effectiveness as a function of discretization for 80GPM and $60^0$F

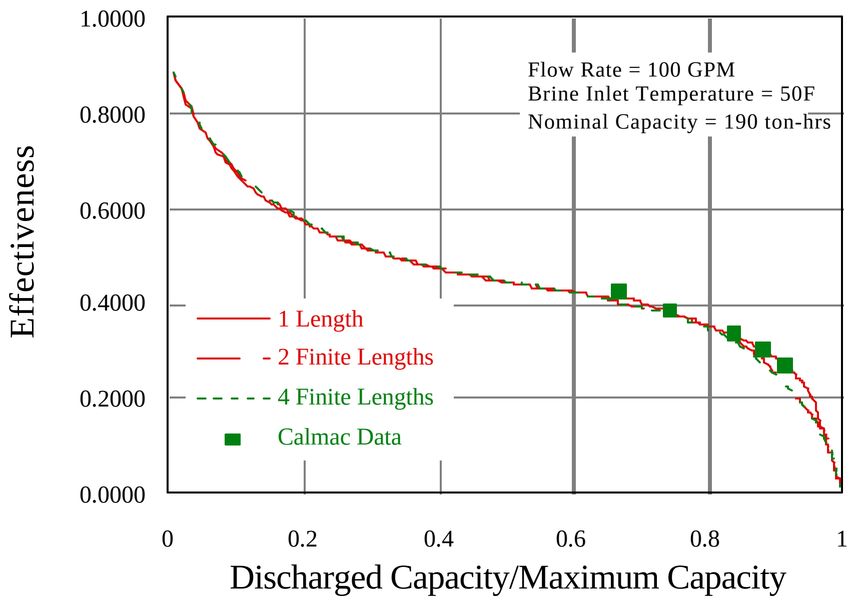


**Figure 2.12** Effectiveness as a function of discretization for 100GPM and 50$^0$F

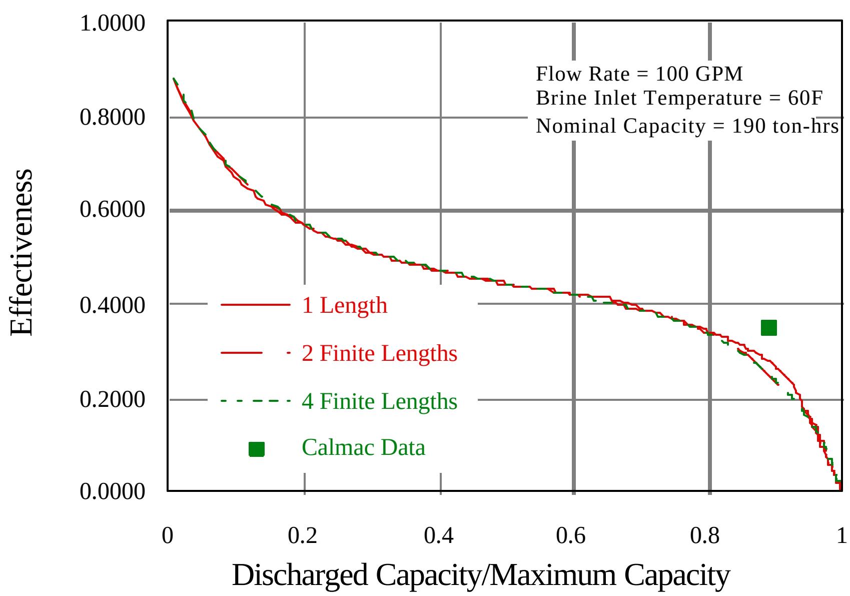


**Figure 2.13** Effectiveness as a function of discretization for 100GPM and 60$^0$F

For flow rates less than 60 GPM discretization into at least two finite lengths is required. At higher flow rates discretization is not required.

In figures 2.4 through 2.13, it can be seen that the model under predicts the performance of the tank at high inlet brine temperatures and high brine flow rates [4]. This underprediction is due, in part to the assumption in the model that the flow between two adjacent coils is parallel. In fact, the flow is counterflow, thus for a given axial position of a coil, more ice can be melted than just one-half the space between coils.

In figures 2.4 through 2.13, the lines for the analysis of the entire length of the tube and the analysis of four finite lengths are both continuous, but the lines for the four finite length analysis are close to being continuously differentiable. A line is continuously differentiable if its derivative is also continuous [5]. The one length analysis shows a sharp change in slope particularly in the cases of low flow rate. This can be explained by studying the water formation diameter. The water formation diameter is the diameter of the water-ice interface, and increases during the discharge of the tank. In figure 2.14 the water formation diameter is plotted against the position along the length of the tube for various levels of discretization. Figure 2.14 shows the water formation diameter distribution afte five minutes of discharge for a flow of 100 GPM and a brine inlet temperature of 60°F. When the entire length is analyzed, the water formation diameter is constant with length. As the tube is discretized, it is found that the water formation depends on the axial position along the tube. For several discretizations the water formation diameter is higher at the inlet when compared to the average value and is lower at the outlet. Thermally, this make sense. The highest temperature gradient occurs at the inlet of the tube with a corresponding high heat transfer rate. The water formation diameter is a geometric measure of the amount of heat transfer, and the higher the heat transfer, the larger the amount of ice that is melted, and thus the larger the water formation diameter. It can be seen in figure 2.14 that as the level of discretization of the tube length is increased, the diameter of the water formation varies more significantly along the length.

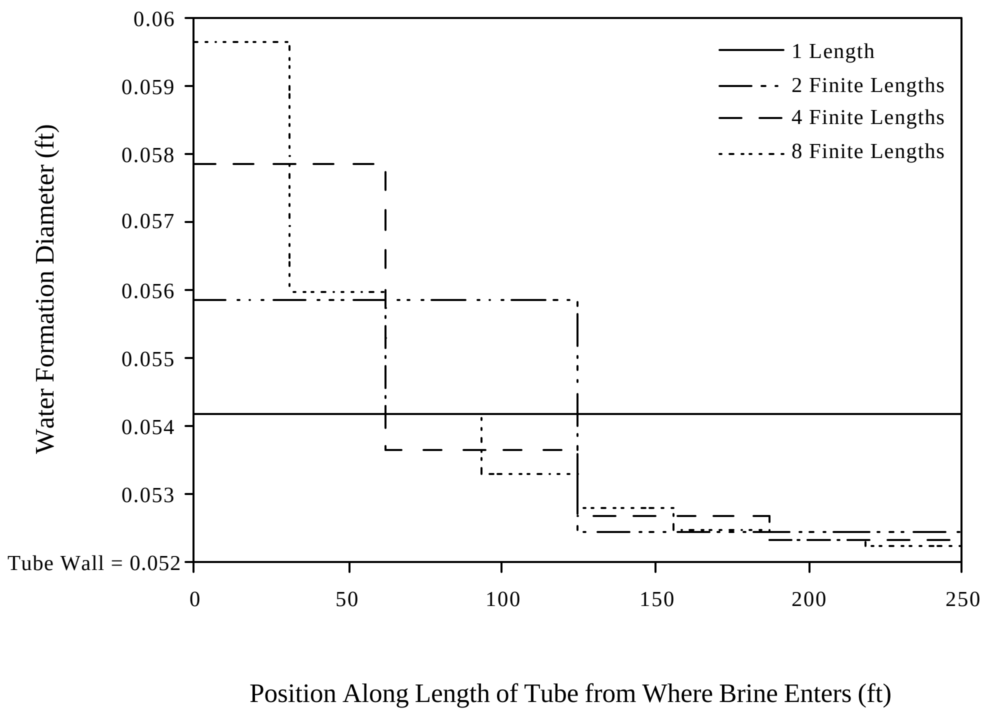


**Figure 2.14** Water formation diameter profile along the length of a coil for various levels of discretization.

As the water formation diameter grows, it will approach a critical diameter where ice can no longer be melted radially. There is still some ice left in the corners where the critical diameters of adjacent tubes do not intersect. The mode of heat transfer to melt this ice is much different. It is much harder to melt this ice. In the effectiveness curves for the average length, there is a sharp point where the effectiveness begins to drop dramatically. This point is where the critical diameter is reached. This sharp point does not appear in the discretized curves because the critical diameter for each finite length is reached at a different time in discharge. Thus, the curves are more continuous.

## 2.4 Chapter Summary

In this chapter it was found that using one length with average temperatures and heat transfer coefficients works well for most conditions. For low flow rates, four finite lengths were required for convergence of the outlet brine temperature, while for higher flows, one length was adequate for convergence of the outlet brine temperature.

# *REFERENCES 2*


1. Jekel, Todd, "Modeling of Ice-Storage Systems," M.S. Thesis, University of Wisconsin - Madison, December 1991.

2. Incropera, F.P., and D.P. DeWitt, *Introduction to Heat Transfer,* John Wiley & Sons, New York, 1990.

3. Duffie, J. A. and W. A. Beckman, *Solar Engineering of Thermal Processes,* John Wiley & Sons, New York, 1985.

4. *Levload Ice Bank Performance Manual*, Product Literature, Calmac Manufacturing Corporation, Englewood, New Jersey, April 1987.

5. Greenberg, Michael D., *Advanced Engineering Mathematics*, Prentice Hall, New Jersey, 1988.

CHAPTER

# THREE

## FULL STORAGE STRATEGY

In this chapter, the effectiveness model for the ice-storage tank in discharge is used to study a full storage strategy. A schematic of a full storage system consisting of a cooling coil, temperature-controlled valve, ice-storage tank, and a chiller is shown in figure 3.1 (Jekel, 1991). The temperature controlled valve is controlled by a set water temperature into the cooling coil. The valve proportions the flow from the tank to meet the set temperature. In the simulations in this thesis, the performance of the temperature controlled valve is modeled using a temperature controlled flow diverter at the inlet of the ice-storage tank. In the full storage strategy, the chiller is used only for charging the tank.

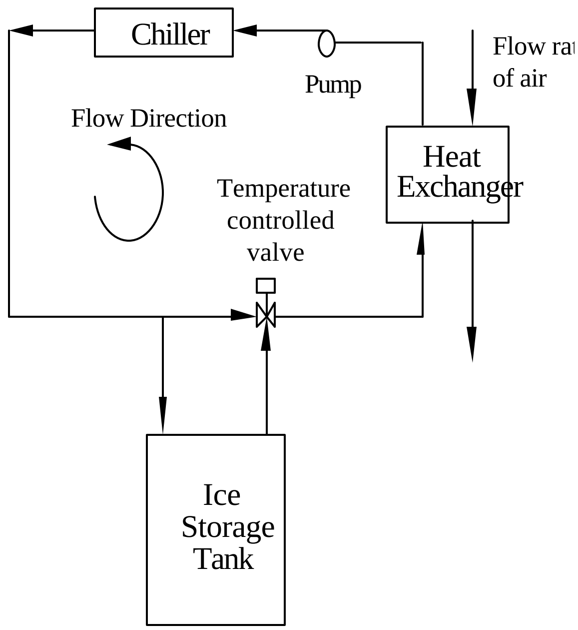


**Figure 3.1** Chiller and ice tank in series with the chiller upstream of the ice tank.

Using the effectiveness curves in figure 3.2 (Jekel, 1991), a simplified performance model was constructed for use with TRNSYS [1]. The effectiveness was regressed (Kawashima, 1992) with respect to the volumetric flow rate and the fraction of the capacity discharged (Eq. 3.1). The coefficients are given in Appendix B.

$$\varepsilon = f\left(Q, \frac{\text{Discharged Capacity}}{\text{Maximum Capacity}}\right) \tag{3.1}$$

The maximum capacity is a function of the mass of ice and the inlet brine temperature of the flow (Eq. 3.2), and consists of the latent heat of fusion and sensible heat. By knowing the inlet brine temperature, the fraction of energy discharged, and the flow rate, the effectiveness can be found.

$$\text{Maximum Capacity} = m_i \left(u_{if} + C_{v,w}\left(Tb_{in} - T_{ice}\right)\right) \tag{3.2}$$

Once the effectiveness is known, the outlet brine temperature can be found using the definition of the tank effectiveness (Eq. 3.3).

$$Tb_{out}=Tb_{in}-\varepsilon\,(Tb_{in}-T_{ice}) \qquad (3.3)$$

The capacity discharged is added internally within the ice-storage tank model at each time step to the total capacity discharged. The new total capacity discharged is the sum of the old total capacity discharged and the heat transfer from the brine for the given time step. At each time step the load, the inlet brine temperature, and the flow rate are required. At any time the change in tank capacity with time is the ice cooling load (eq 3.4).

$$\frac{dC}{dt} = -\, m\, C_{p,b}\, (Tb_{in}-Tb_{out}) \qquad (3.4)$$

In the simulations, a finite difference approximation was used (eq. 3.5).

$$\Delta C = -\, m\, C_{p,b}\, (Tb_{in}-Tb_{out})\Delta t \qquad (3.5)$$

Knowing the change in capacity and the capacity at the previous time, the new capacity can be found (eq 3.6).

$$C_{t+\Delta t} = C_t+\Delta C \qquad (3.6)$$

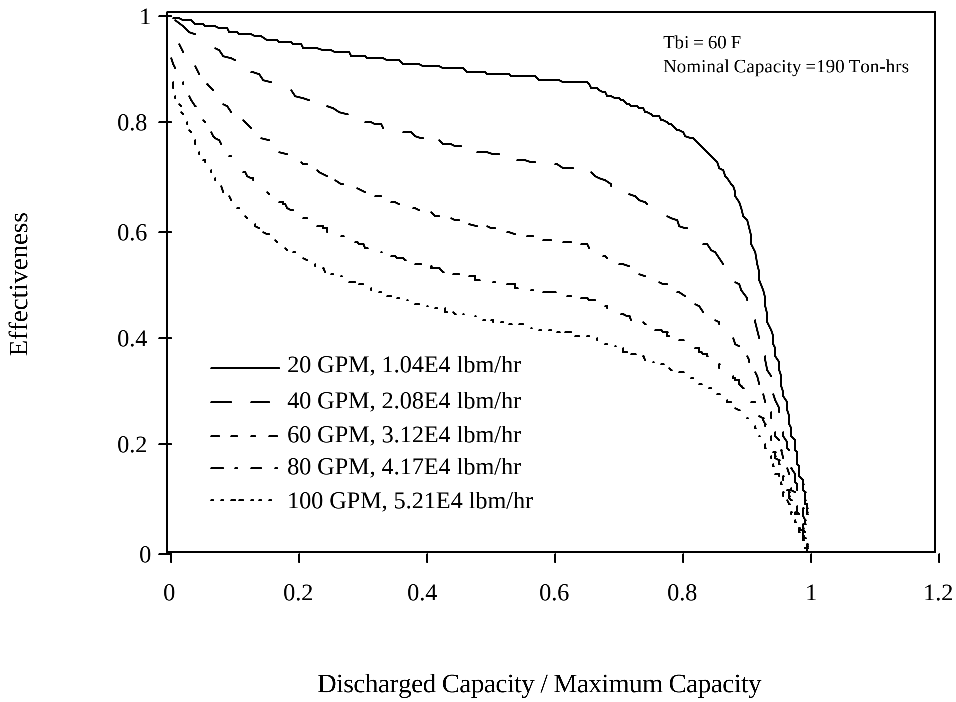


**Figure 3.2** Effectiveness of discharge vs. fraction of capacity discharged for various flow rates at an inlet brine temperature of $60^0$F.

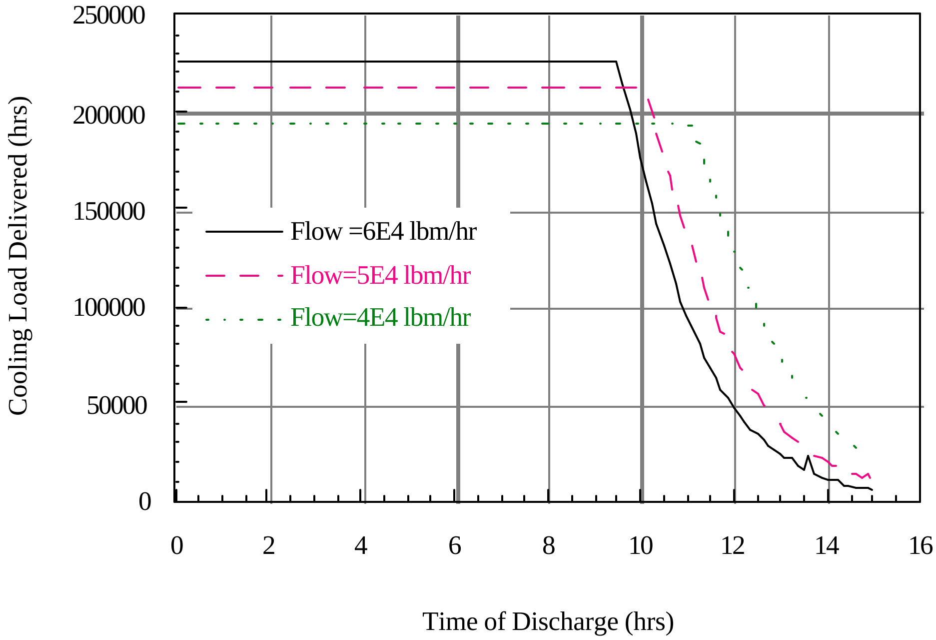


**Figure 3.3** Cooling load delivered as a function of time for different flow rates at a constant Tset of 50F.

Simulations were performed with a constant air flow rate and temperature into the cooling coil. The initial simulations involve fixing the set temperature of the water to the cooling coil at 50$^o$F and changing the water flow rate through the coil.

Figure 3.3 shows that for a constant set water inlet temperature to the cooling coil, increasing the water flow rate through the coil increases the cooling load. When the flow rate is increased, however, the duration at which the constant load can be met is decreased.

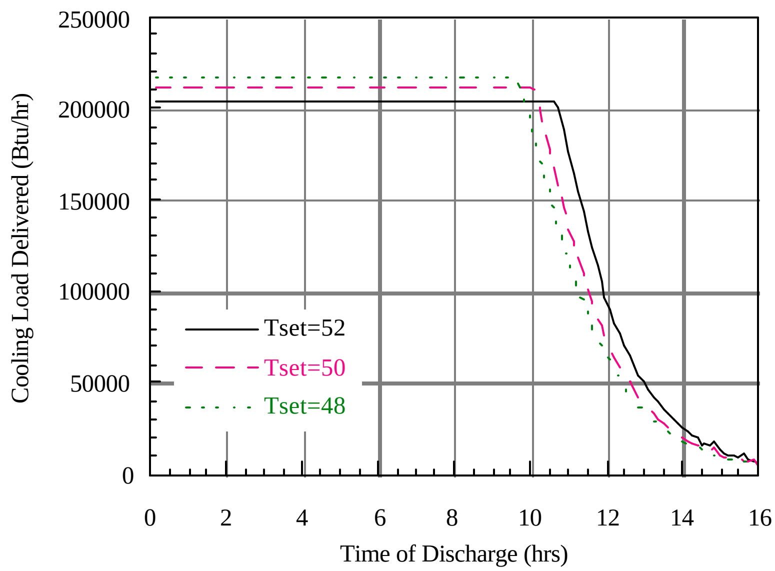


**Figure 3.4** Cooling load delivered as a function of time for different set temperatures for a brine flow rate of 50,000 lbm/hr.

Simulations were performed with dry air, at a constant temperature, entering the cooling coil at a constant flow rate. The simulations involve fixing the water flow rate at 50,000 lbm/hr (100 GPM) through the coil and changing the set temperature of the water to the cooling coil

In figure 3.4 it can be seen that when the set water inlet temperature was increased, for a constant flow rate through the cooling coil, the cooling load delivered decreased. The duration at which a constant load can be met increases, however, with decreasing cooling load.

Combining the behavior observed in figure 3.3 and 3.4, multiple combinations of flow rate and set temperature can be found that will meet the same cooling load. They do not, however, meet the same cooling load rate for the same discharge time. At higher flow rates, and consequently, higher set temperatures, the tank is able to deliver the load for a longer period of time as shown in figure 3.5. Although the combinations cannot deliver the same load rate for the same amount of time, they all deliver the same total energy when they are all fully discharged. The tank can be fully discharged by allowing the set temperature to increase as needed which means the cooling load delivered will be less than the amount required. It is the rate of cooling that is of importance in air-conditioning, however.

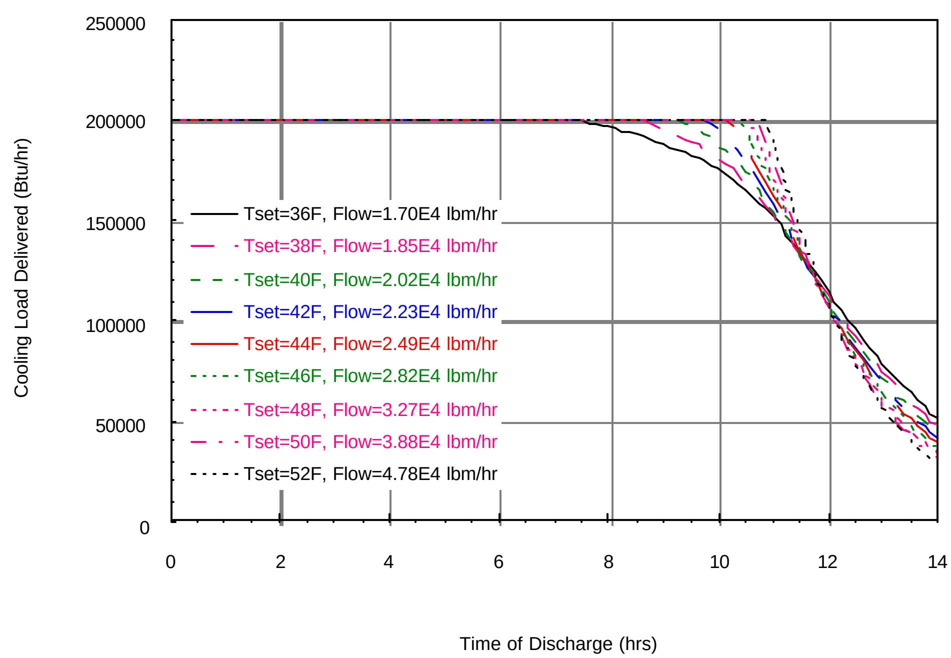


**Figure 3.5** Cooling load delivered as a function of time for various combinations of set water inlet temperatures and flow rates to the cooling coil.

## 3.2 Relation Between Load Profile and Required Tank Size

For section 3.1, the effect of set temperature and flow rate on discharge time for a constant load profile was studied. In this section the effects of load profile on the required size of an ice-storage tank will be studied.

Four types of load profiles were studied: Constant load profile, load profile increasing with time, load profile decreasing with time, and a representative building load profile. To study the effects of the profiles on system performance, the constant load profile of 200,000 Btu/hr was chosen as a reference. The Calmac 1190 Ice-storage tank was able to meet this load for 10.25 hours. The increasing and decreasing loads were designed to yield the same total integrated load over a period of 10.25 hours. The rate of change of the load profile was varied. In the simulations, the air temperature into the cooling coil is constant at 75°F, the air into the cooling coil is dry, and the air temperature out of the cooling coil is constant at 61.1°F. The air flow rate is then determined by the load through a controller, which then determines the required set temperature of the water entering the cooling coil. Therefore, in this section, the simulations are run using a variable air volume system (VAV).

In figure 3.6 some of the linearly increasing load profiles with the same total integrated load are seen. Not all are shown to maintain clarity in the figure. Figure 3.7 shows some of the linearly decreasing load profiles.

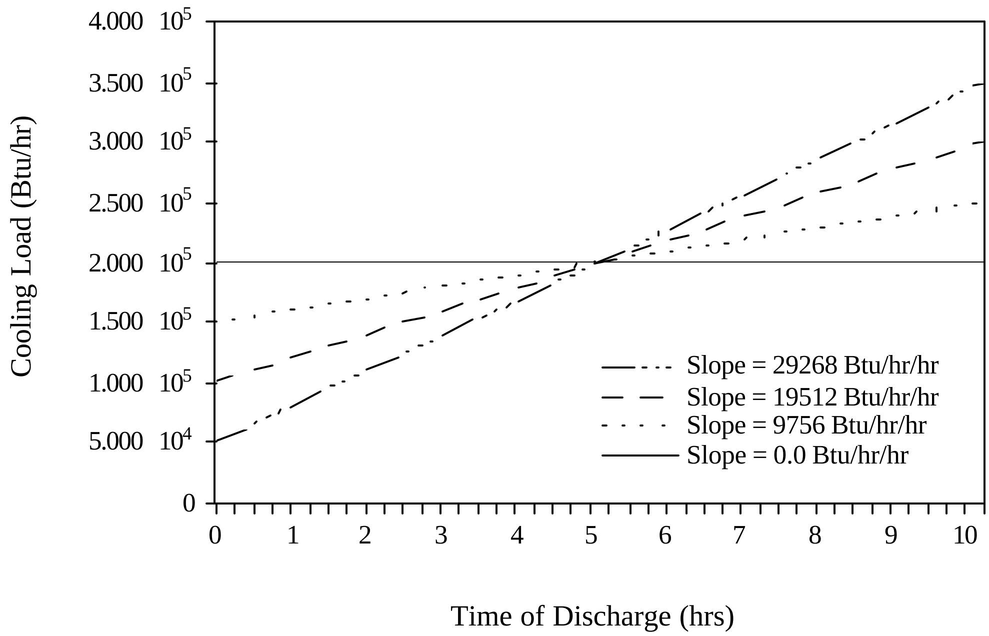


**Figure 3.6** Various linearly increasing load profiles

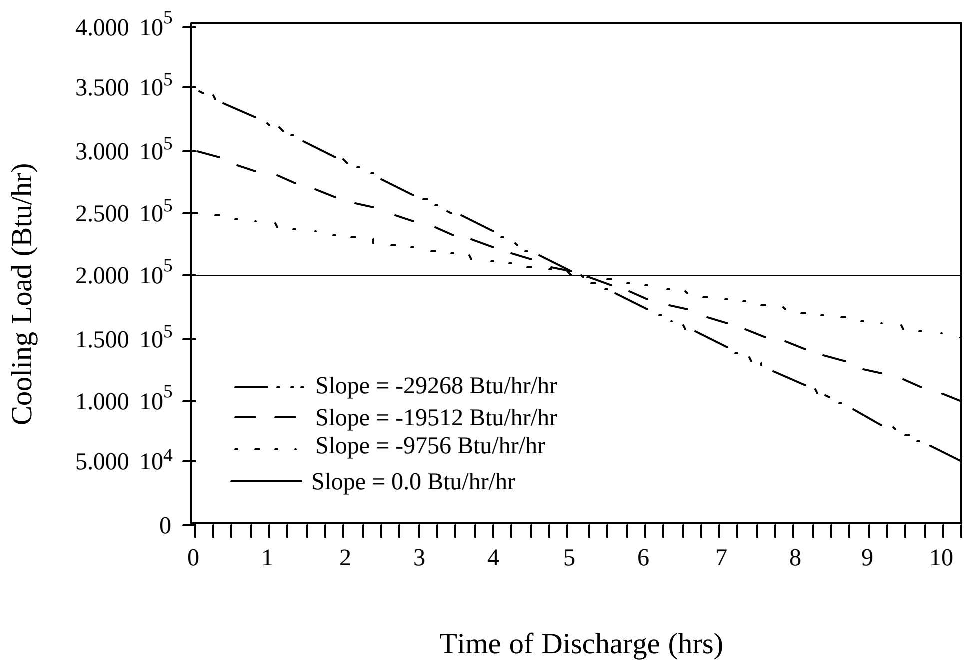


**Figure 3.7** Various linearly decreasing load profiles

The effect of load profile on tank size was evaluated. The Calmac 1190 tank was given a base tank size of 1.0 (1.0 = 190 Ton-hrs). In other words, the 190 Ton-hr Calmac tank can meet the constant load for 10.25 hours.

For an increasing load profile, the required tank size is always greater than unity. As the slope of the load profile increases, the equivalent tank size increases at an increasing rate as shown in figure 3.8. For an increasing load with a slope of 25,000 Btu/hr/hr the tank size is four times the size required for a constant load. A larger tank is required to handle an increasing load, because the tank effectiveness decreases throughout the discharge time. The performance of the ice tank is most critical at the end of the day.

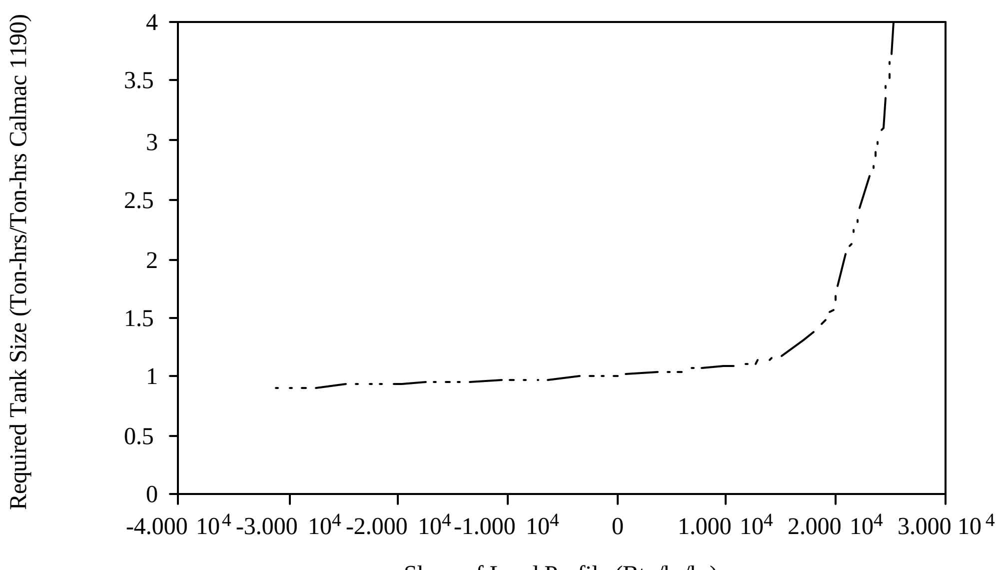


**Figure 3.8** Required tank size as a function of rate of change of cooling load for a water flow rate of 40,000 lbm/hr through the cooling coil.

For a decreasing load profile, the required tank size is always less than unity as shown in figure 3.8. The required tank size is less than unity, because the load is largest when the tank effectiveness is very high. Since both the tank effectiveness and the load profile decrease with time, a smaller tank is required. Figure 3.8 also shows that the tank size does not change significantly for a load that decreases with time.

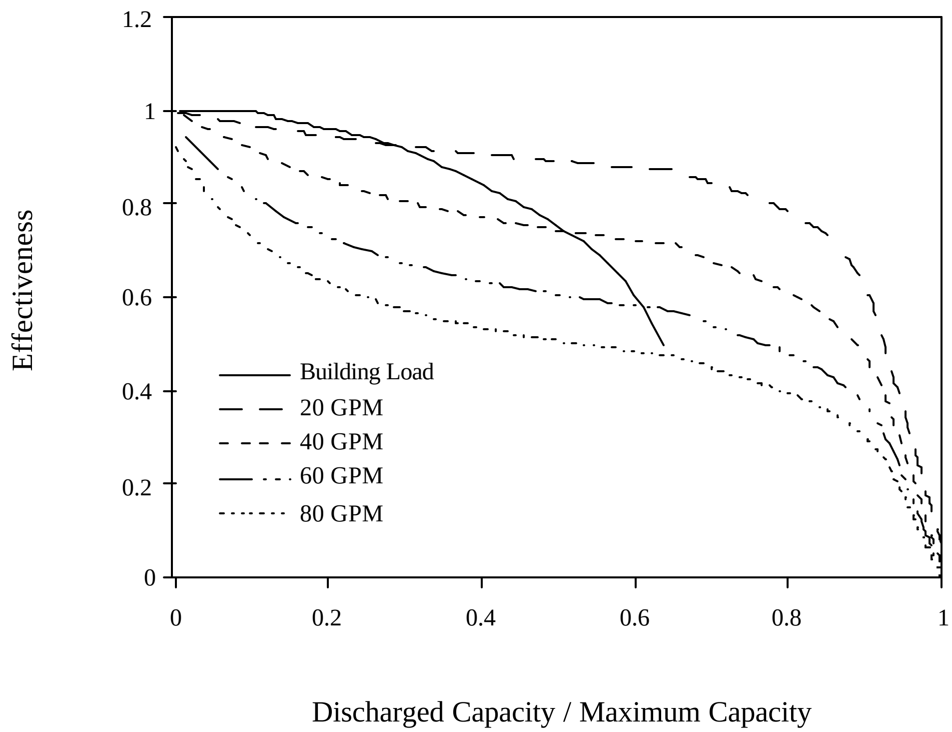


**Figure 3.9** Effectiveness mapping of linearly increasing load with a slope of 19512 Btu/hr/hr.

In figure 3.9 the effectiveness history of the ice-storage tank meeting an increasing load with slope 19,512 Btu/hr/hr is superimposed on the constant flow curves. The drop in effectiveness can be attributed to both the increasing load and the depletion of ice.

For a typical building load profile, the required tank size could be different from unity. In the Calmac Performance Manual [2], a diversity factor, which is the ratio of the average load to the peak load for the design day is used. This diversity factor is used to find the required number of tanks for a given design day load. The diversity factor does not weight the time at which the load occurs. A linearly increasing load and a linearly decreasing load can have the same diversity factor, but have drastically different tank sizes as seen in figure 3.8.

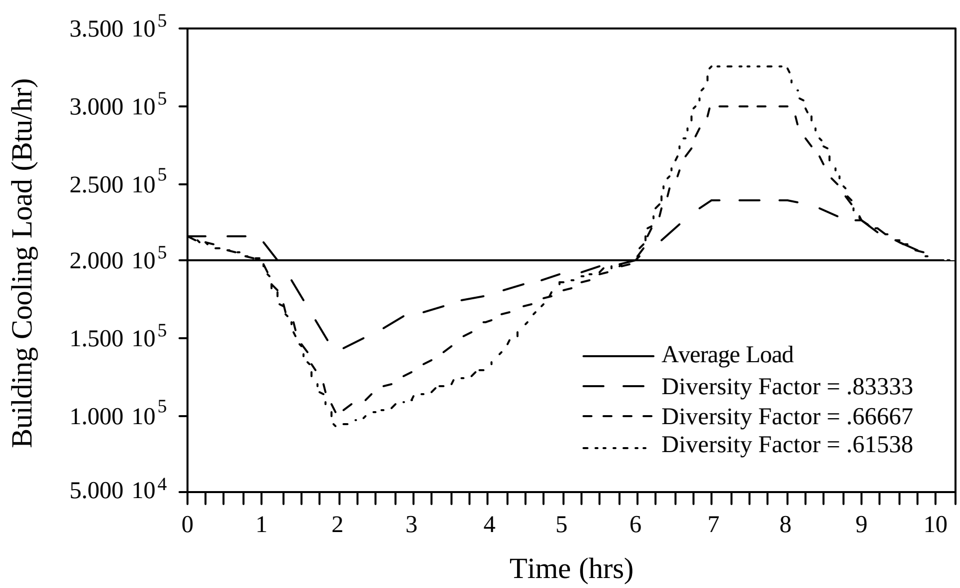


**Figure 3.10** The three representative building loads plotted against time.

In figure 3.10 three representative building loads with diversity factors of .83333, .66667, and .61538 are plotted against time. The building loads were, to some degree, based on building loads from Cummings [3]. The building loads were used in simulations with a constant air inlet temperature of 75°F, dry air into the cooling coil, and a constant air outlet temperature of 61.1°F. The effect of lower coil air outlet temperatures on the results will be discussed later.

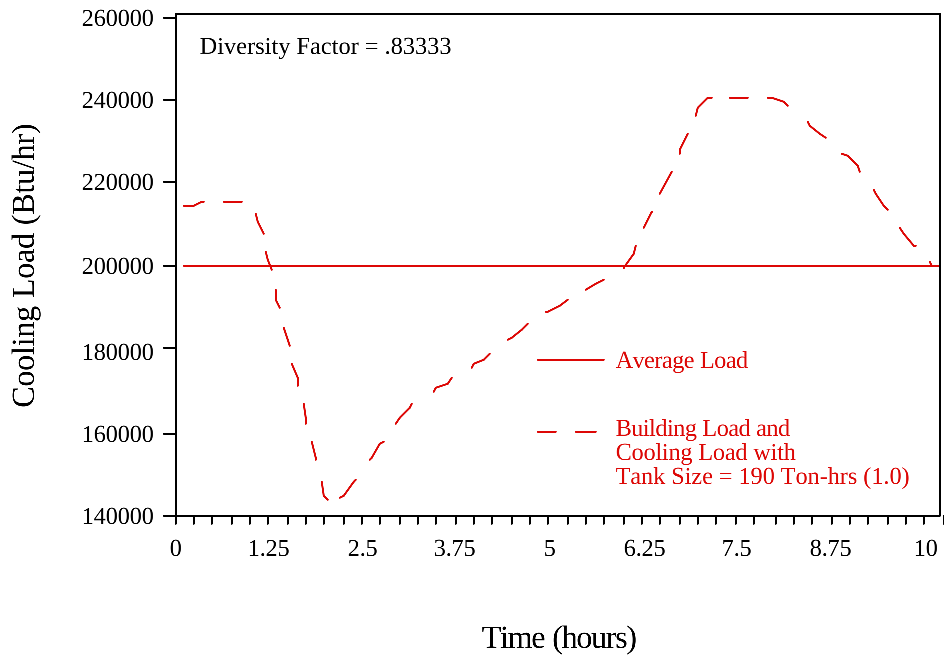


**Figure 3.11** Cooling load delivered vs. time. The building load has a diversity of .83333.

In figure 3.11 the nominal tank size of unity used to deliver the average load can deliver the required load for the given building load profile with a diversity factor of .83333.

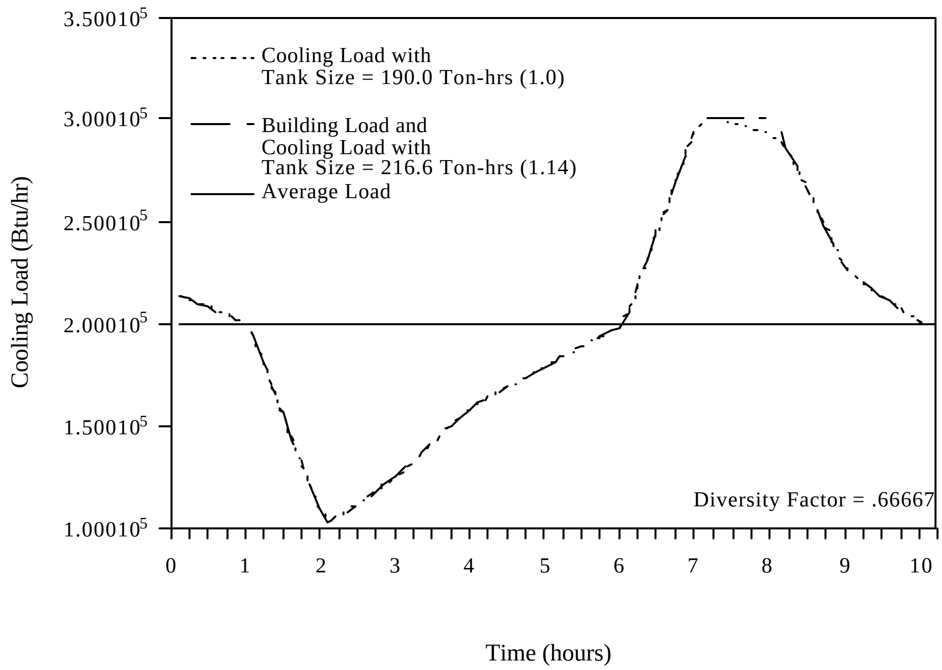


**Figure 3.12** Cooling load as a function of time for various tank sizes. The building cooling load has a diversity of .66667.

For the load with a diversity of .66667 the nominal tank size of unity cannot fully meet the load requirements as shown in figure 3.12. After 7.25 hours of discharge time, the nominal tank size cannot meet the required rate. It was found that a tank that is 14% larger (216.6 Ton-hrs) can just meet the load.

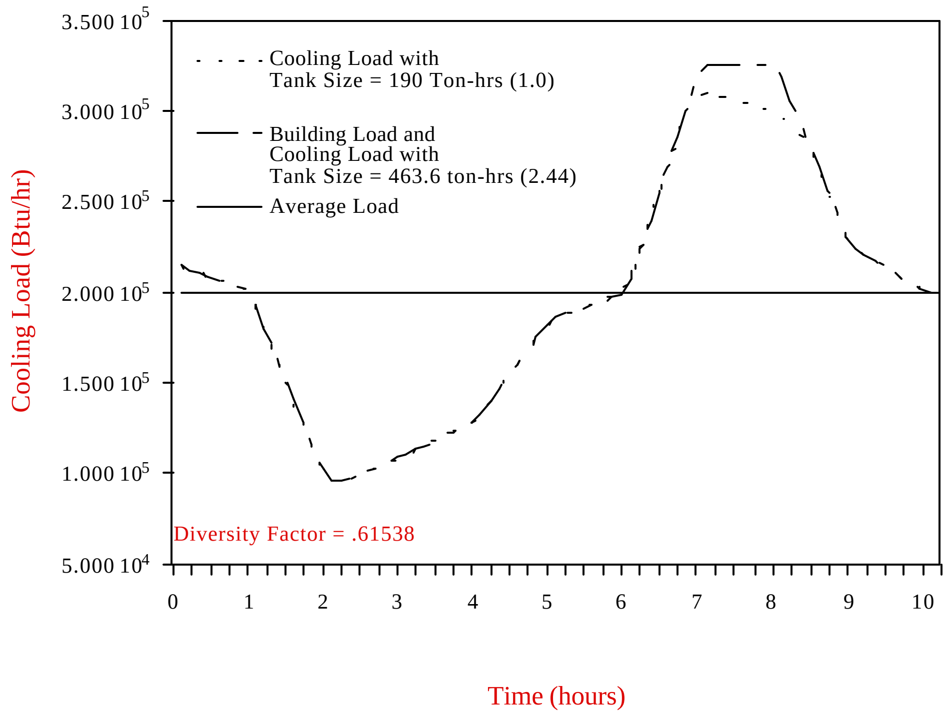


**Figure 3.13** Cooling load delivered vs. time for various tank sizes. The building load has a diversity of .61538.

In figure 3.13, the nominal tank size used to deliver the average load cannot deliver the required load for the building load profile with a diversity factor of .61538.

It is of interest to examine how the ice tank behaves during discharge to meet a real load profile. The model of the ice-storage tank was based on a regression of a series of constant flow curves, and the discharge flow was superimposed on these curves as shown in figure 3.14. The building load used was that of figure 3.12. In figure 3.14, the effectiveness is high initially because the tank is fully charged and the cooling load demand is not high. The effectiveness drops dramatically when the tank has about half the capacity left and the peak load occurs. During the peak load, the set temperature for the water into the cooling coil must be very low which is attainable only if most of the constant flow through the cooling coil is pumped through the ice-storage tank. After the peak load, the effectiveness increases but cannot return to the levels prior to the peak load because a large amount of the capacity was used to meet the peak load.

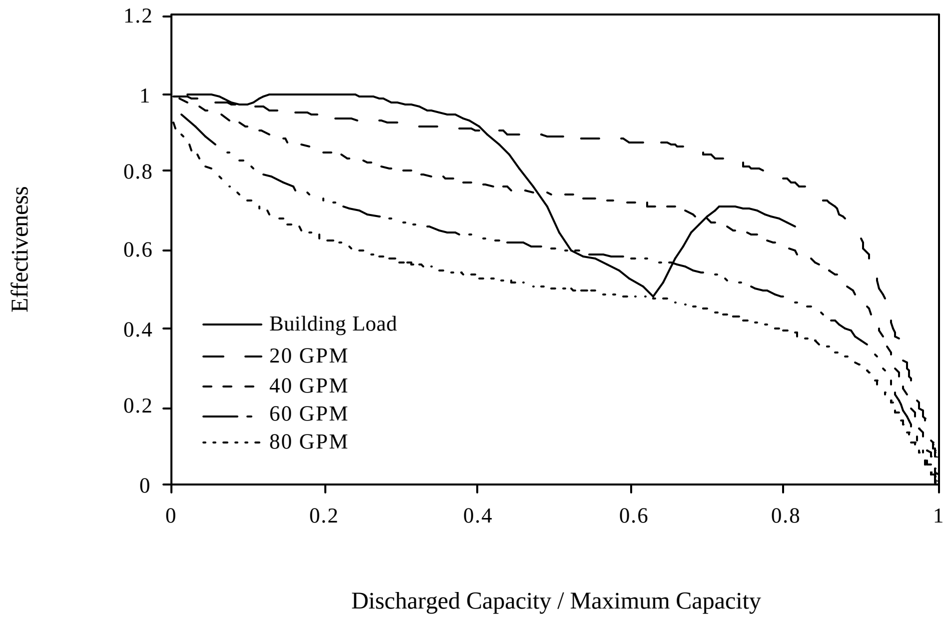


**Figure 3.14** Tank discharge effectiveness vs. fraction of capacity discharged. The building load varies the flow rate through the ice-storage tank.

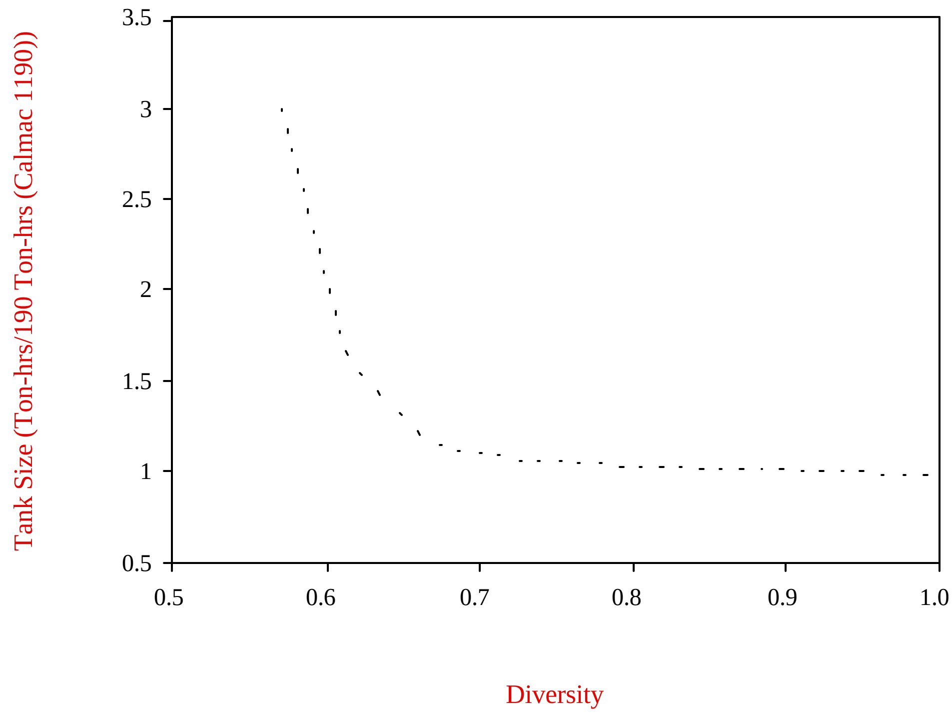


**Figure 3.15** Tank Size vs. Diversity Factor for a Flow Rate of 50000 lbm/hr for the linearly increasing load profiles.

To further study the effect of the diversity factor on the tank size, tank sizes for the linearly increasing load profiles were plotted against diversity. In figure 3.15 the tank size required for a constant flow rate is shown as a function of diversity. The size increases with decreasing diversity factor. The large increase in tank size when the diversity is less than 0.65 can also be seen in figure 3.8 for a slope of 20,000 Btu/hr/hr. The increasing load at a time when the tank is partially discharged and has a low effectiveness means that a much larger tank is required.

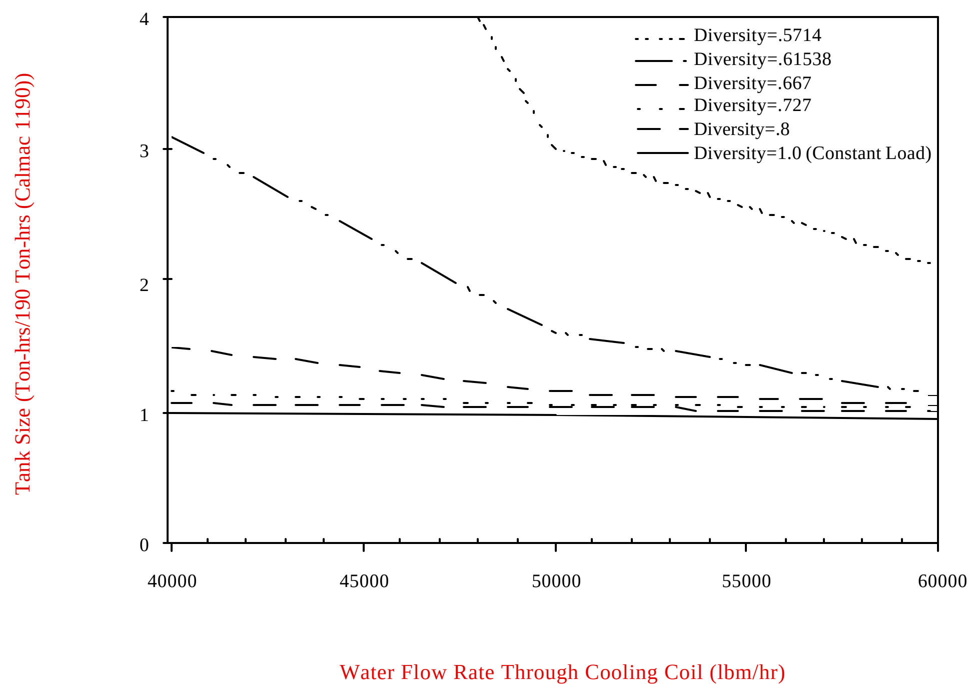


**Figure 3.16** Tank Size vs. Water Flow Rate Through Cooling Coil for Various Linearly Increasing Building Load Profiles.

Figure 3.16 shows that for a given load profile, there are an infinite number of combinations of water flow rate through the coil and tank size that will meet the load. The graph also shows that as the diversity factor becomes much less than one, the tank size required grows very large. This is further proof that the proper tank size is highly dependent on the diversity factor.

In sizing the tank, the time at which the load occurs relative to the total time of discharge from the ice-storage tank is important. From the simulations it is obvious that for a given total load the diversity factor alone does not size the tank. This is especially true when larger loads occur during the latter part of the day. This was seen most clearly in examining the linearly increasing and linearly decreasing loads. For a fixed tank size and a fixed flow rate through the cooling coil, the linearly decreasing load could be met, while the linearly increasing load could not be met.

## 3.3 Analysis of the Diversity Factor

In the previous section it was found that the diversity factor was a good indicator in determining the proper tank size for a given building load profile. It is important to also find out how accurate the diversity factor is as a function of the building load profile. To test the diversity factor two different load profiles were used. One profile was a linearly increasing load profile and the other profile was the building load profile used in figure 3.13. Both load profiles have a diversity factor of .61538. Simulations were run at various water flow rates to find corresponding tank sizes. The results can be seen in figure 3.17.

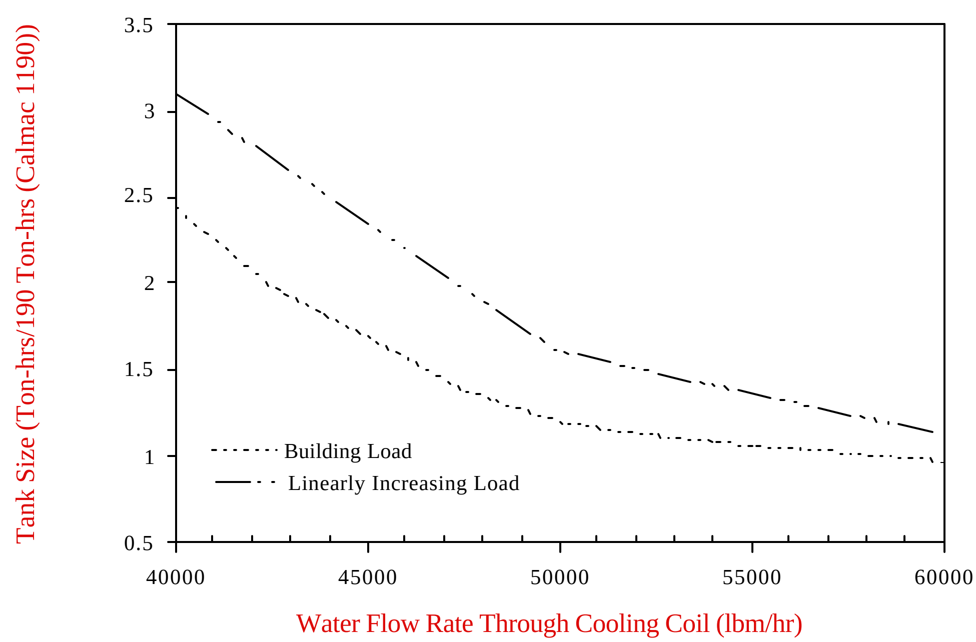


**Figure 3.17** Tank Size vs. Water Flow Rate Through Cooling Coil for Two Different Load Profiles with the Same Diversity Factor of .61538.

If the diversity factor was an accurate way of reducing a building load profile down to a single number, the two load profiles would have produced the same curve in figure 3.17; however the two curves are different. Thus, the diversity factor is not very accurate for use in tank sizing and flow selection, and is only an indicator of a building load profile.

## 3.4 Effect of Water Flow Rate on Required Tank Size

In the previous section it was seen that the equivalent tank size is strongly dependent on the shape of the load profile. The equivalent tank size is also strongly dependent on the water flow rate. In a series of simulations the water flow rate through the cooling coil was fixed at 30,000 lbm/hr, 40,000 lbm/hr, 50,000 lbm/hr, and 60,000 lbm/hr. The air inlet temperature to the cooling coil was constant at 75°F, the air into the cooling coil was dry, and the air outlet temperature from the cooling coil was fixed at 61.1°F. By increasing the water flow rate, the equivalent tank size can be reduced as seen in figure 3.18.

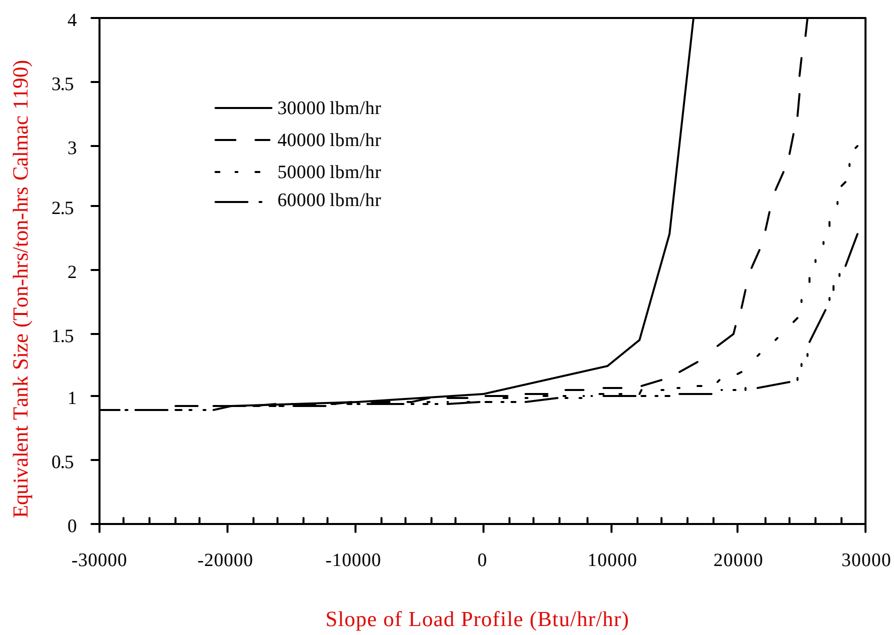


**Figure 3.18** Equivalent tank size as a function of rate of change of cooling load for various water flow rates.

Figure 3.18 shows that for slopes less than zero, the profile does not affect the required tank size appreciably. When the slope is greater than 10000 Btu/hr/hr, increasing the flow dramatically reduces the tank size required to meet the load profile.

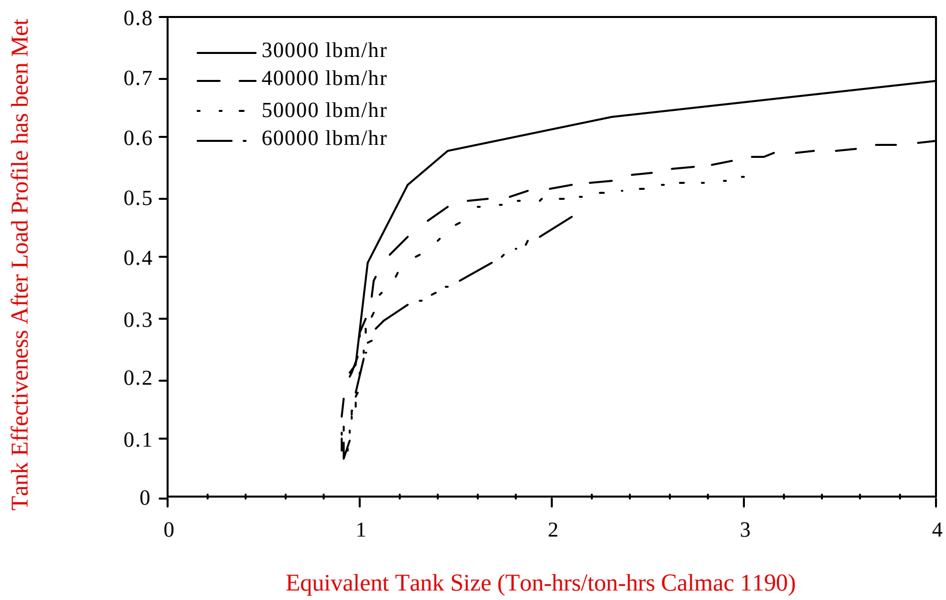


**Figure 3.19** Tank effectiveness after load profile has been met as a function of tank size for various water flow rates through the cooling coil.

Figure 3.19 shows the tank effectiveness at the end of the discharge period as a function of tank size for various mass flow rates. In general the higher the tank effectiveness at the end of the discharge period, the larger the tank size that is required. Increasing the tank size increases the maximum capacity, which will decrease the ratio of the discharged capacity to the maximum capacity for a fixed load. This ratio is inversely proportional to the effectiveness as shown in figure 3.2. Thus, a high tank effectiveness indicates that the building load was too much for the nominal size tank (190 ton-hours) to handle, not because the tank does not have enough capacity, but rather because the tank does not have a high enough effectiveness at the end of the discharge period. Figure 3.19 also shows that the effectiveness at the end of the discharge period can be increased by increasing the flow rate which is also shown in figure

3.2. Thus the tank size required can be decreased by increasing the flow rate which is also shown in figure 3.18.

Once the correct size of the ice-storage tank(s) is found using the design day analysis, the flow rate can be increased to handle the rare case when the design day load is exceeded. Similarly, the flow rate can be decreased for daytime loads, that are less than the design day load, to the point where the fixed tank size will just be able to meet the load, which will minimize pumping power.

## 3.5 Load Failure and Temperature Flotation

So far, all simulations have been performed using a variable air volume system. The air handling unit adjusts the flow rate to give a constant outlet air temperature. If a constant air volume system would be used, the air supply temperature from the cooling coil could be higher at low loads due to high flow rates. This will allow the ice-storage system to better meet the load. While the ice-storage system can better handle the load, the comfort level associated with the air temperature will be degraded, since the air inlet temperature to the cooling coil is a measurement of the comfort in the cooling space. To study this, the increasing and decreasing load profiles used previously were used in simulations involving the variable air volume system and constant air volume system. For the variable air volume system the air inlet temperature was held constant at 75°F, the outlet air temperature was held constant at 61.1°F, and the air flow rate was varied to meet the load. For the constant air volume system, the air flow rate was held constant at 40,000 lbm/hr, the air outlet temperature was held constant at 61.1°F, and the air inlet temperature floated to meet the load.

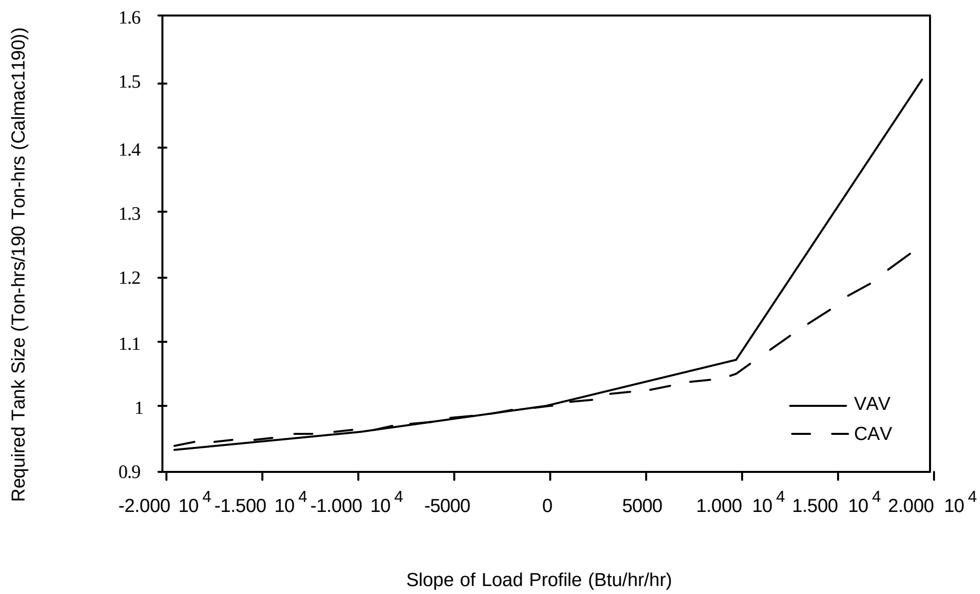


**Figure 3.20** Required tank size as a function of slope of building load profile for both a variable air volume system (VAV) and a constant air volume system (CAV).

In figure 3.20 the required tank size is plotted against the slope of the building load profile. It can be seen that the constant air volume system requires less tank capacity than the variable air volume system for higher slopes. In other words a larger linearly increasing load can be handled with a smaller tank size by the constant air volume system than with a variable air volume system. The fact that the air inlet temperature increases indicates a degradation of the air temperature quality. Thus while the load may be met, the room temperature may not be comfortable.

## 3.6 Cooling Coil Sizing

In addition to the building load profile affecting the size of the ice-storage tank, the size of the cooling coil also affects the size of the ice-storage tank. In the simulations, a detailed geometry is specified to determine the size of the heat exchanger. For this section, a FORTRAN program was written that has exactly the same components as in the previous TRNSYS simulations except that the cooling coil size is simply input as a constant UA value. The cooling coil is modeled using the effectiveness-Ntu method for an unmixed counterflow heat exchanger using equation 3.7 [4].

$$\varepsilon_{cc} = \frac{[1\text{-}\exp(\text{-}Ntu\ (1\text{-}\ C_r))]}{[1\text{-}C_r\ \exp(\text{-}Ntu\ (1\text{-}\ C_r))]} \qquad (3.7)$$

where:

$$Ntu = \frac{UA}{C_{min}} \qquad (3.8)$$

$$C_r = \frac{C_{min}}{C_{max}} \qquad (3.9)$$

where:

$\varepsilon_{cc}$ = Cooling coil effectiveness

Ntu = Number of transfer units

$C_r$ = Critical capacitance

The effectiveness can then be related to the air temperatures and brine temperatures by equation 3.10.

$$\varepsilon_{cc} = \frac{[C_{air}\ (T_{air,in} - T_{air,out})]}{[C_{min}\ (T_{air,in} - T_{brine,in})]} \qquad (3.10)$$

The same constant load of $2x10^5$ Btu/hr for 10.25 hours of discharge was used to find the multiple combinations of cooling coil sizes and ice-storage tank sizes that will just meet this load for the specified time; the tank sizes are given in figure 3.21. Since the load was constant, the flow air flow rate through the coil was also constant, even though it is a variable air volume (VAV) system. Correspondingly, the capacitance rate and, thus, the number of transfer units were constant. This enabled the cooling coil size, in terms of Ntu, to be plotted against tank size. The relation seen in figure 3.21 shows that there are two limits in the sizing of the cooling coil and the ice-storage tank. The limit along the cooling coil size axis shows that if the tank is too small, an infinitely large heat exchanger will not be able to meet the required building load. In another words, perfect heat exchange cannot make up for the fact that the tank does not have enough capacity to meet the building load. The limit along the tank size axis shows that an infinitely large capacity cannot make up for the fact that there is not enough heat exchange area to provide the required temperature change to meet the load.

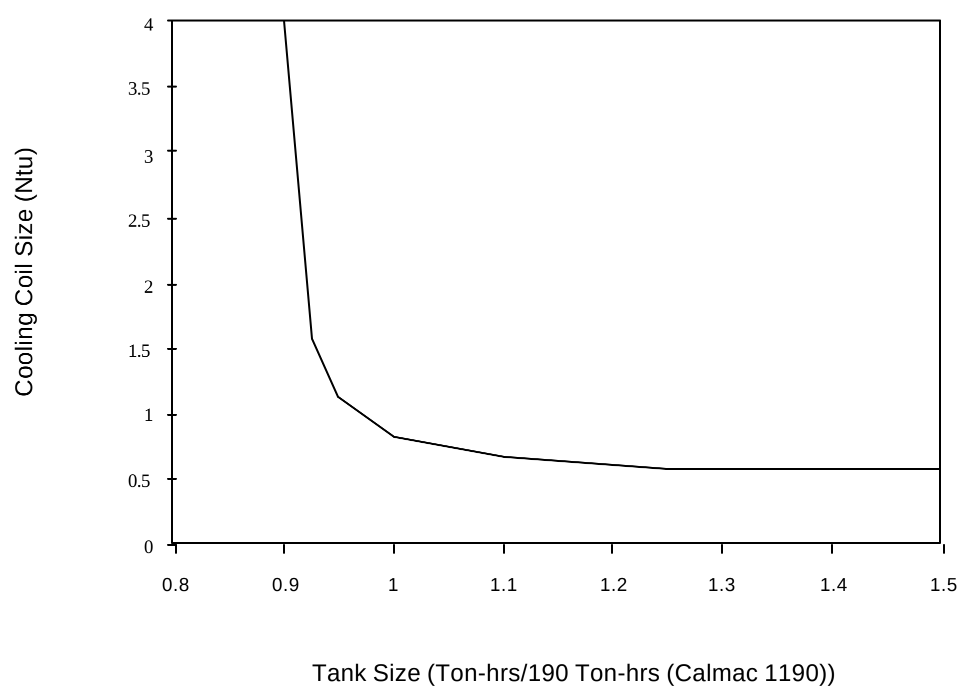


**Figure 3.21** Cooling coil size as a function of tank size for a constant load of 2E5 Btu/hr for 10.25 hours.

To examine the effects of the building load profile on cooling coil sizing and ice-storage tank sizing, simulations were run with some of the linearly increasing building profiles from figure 3.6. Since the building load changes with time, the air flow rate and the capacitance rate of the air change with time. Since the number of transfer units is the UA value divided by the capacitance, the number of transfer units cannot be used for size comparison. Instead the UA value will be used since it stays constant regardless of the building load profile. In figure 3.22 it can be seen that as the slope of the building load increases, both the cooling coil size and ice-storage tank size must increase. The cooling coil size must increase because the conductance area is not large enough to handle the higher peak load.

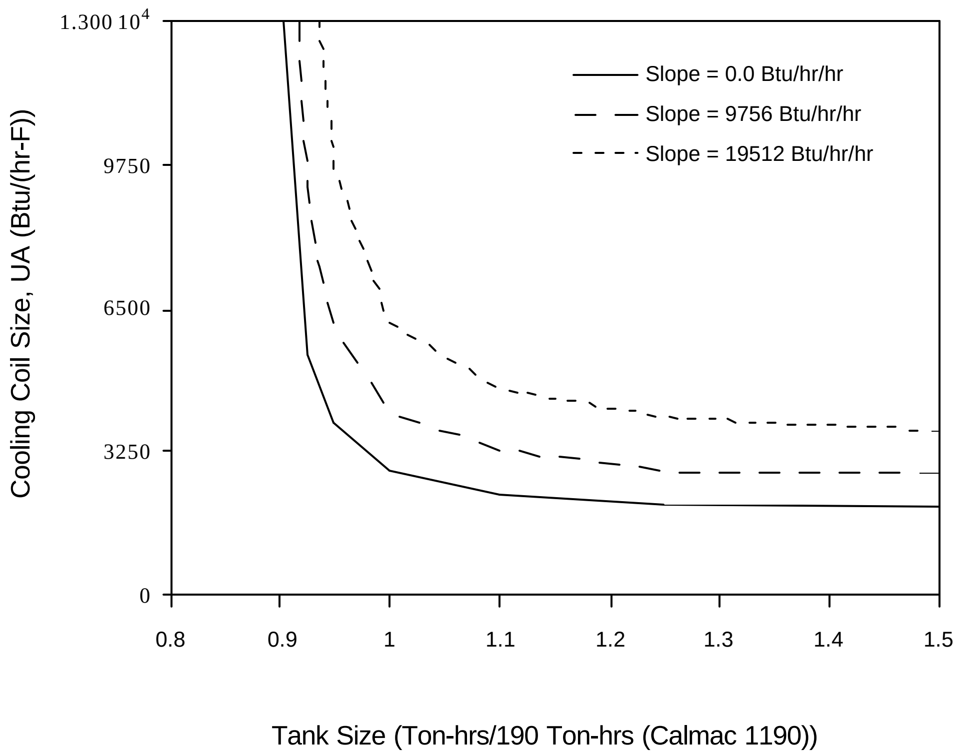


**Figure 3.22** Cooling coil size as a function of tank size for various load profiles that produce the same total integrated load.

## 3.7 Flow Reduction

In all the previous sections a full storage system with a flow diverter to the ice-storage tank was in place that allowed the total flow to be proportioned to the ice-storage tank to be remixed to a set temperature through a mixing valve. One question to be addressed is the effect on performance by eliminating the flow diverter and varying the flow rate through the cooling coil. Since the extra pumping associated with having a bypass line would be eliminated, the pumping power requirements would certainly be reduced, but other performance parameters may be affected.

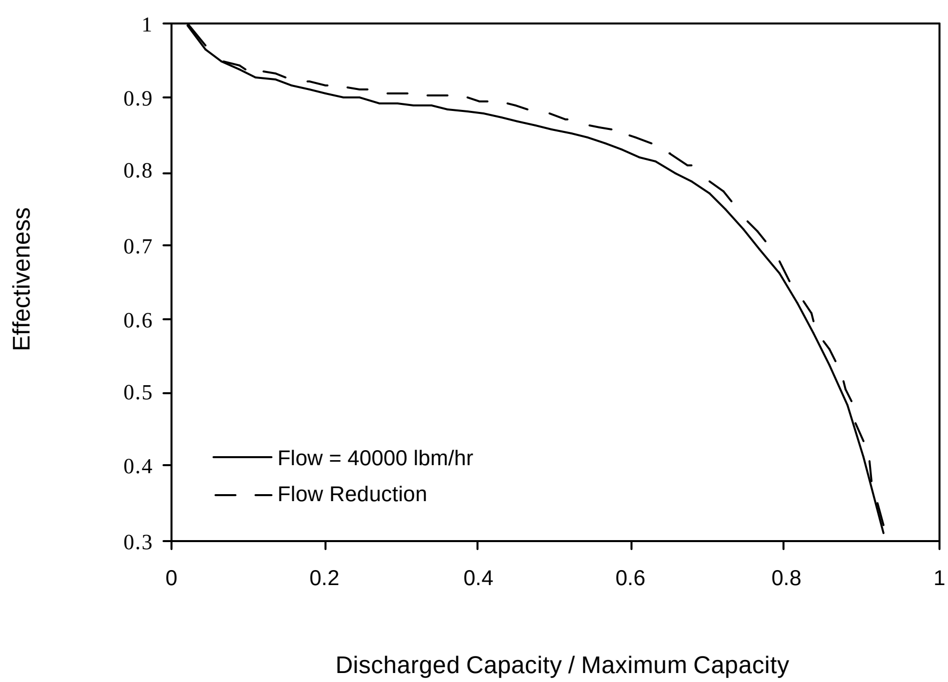


**Figure 3.23** Effectiveness of the tank vs. capacity discharged for both a constant flow and a varied flow.

To study the effects of flow reduction on performance, simulations were run with the same constant load of $2x10^5$ Btu/hr for a 10.25 hour time span. Figure 3.23 shows the effectiveness of the ice-storage tank as a function of the capacity discharged for two cases: A case of constant flow of 40,000 lbm/hr through the cooling coil with a diverter and a case of variable flow through both the cooling coil and the ice-storage tank. It can be seen that through the most of the discharge, the variable flow has the higher effectiveness. The higher effectiveness can be attributed to two factors. First, the tank effectiveness is inversely proportional to flow rate; thus the reduced flow has a higher effectiveness. The second factor is that the tank effectiveness is proportional to the inlet brine temperature. Since the flow through the cooling coil is reduced, the temperature rise of the brine through the coil must be higher, thus raising the effectiveness.

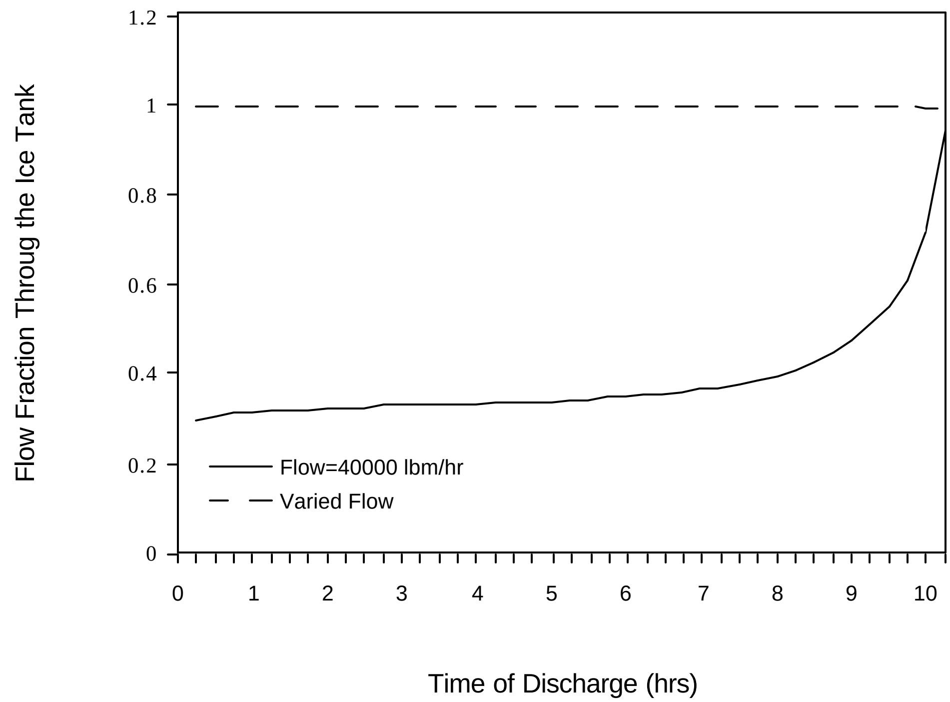


**Figure 3.24** Flow fraction through the ice as a function of time for both the constant 40,000 lbm/hr flow with a flow diverter and the varied flow.

The flow fraction (eq 3.11), which is the ratio of the flow rate through the ice-storage tank to the flow rate through the cooling coil, was used to describe the behavior in a full-storage system.

$$m^* = \frac{m_{tank}}{m_{cooling\ coil}} \qquad (3.11)$$

In figure 3.24 the flow fraction through the ice is plotted against time of discharge. Since the varied flow has no diverter, the flow fraction to the ice is one. The constant flow of 40,000 lbm/hr requires a diverter, thus the flow fraction of the flow is less than one. At the end of discharge both flows approach the same value. This occurs because the tank effectiveness is so low, that a high flow rate must be pumped to the ice-storage tank to meet the load. If the rate is high enough the diverter will not be able to bypass any of the flow. Hence, the flow fractions converge near the end of discharge.

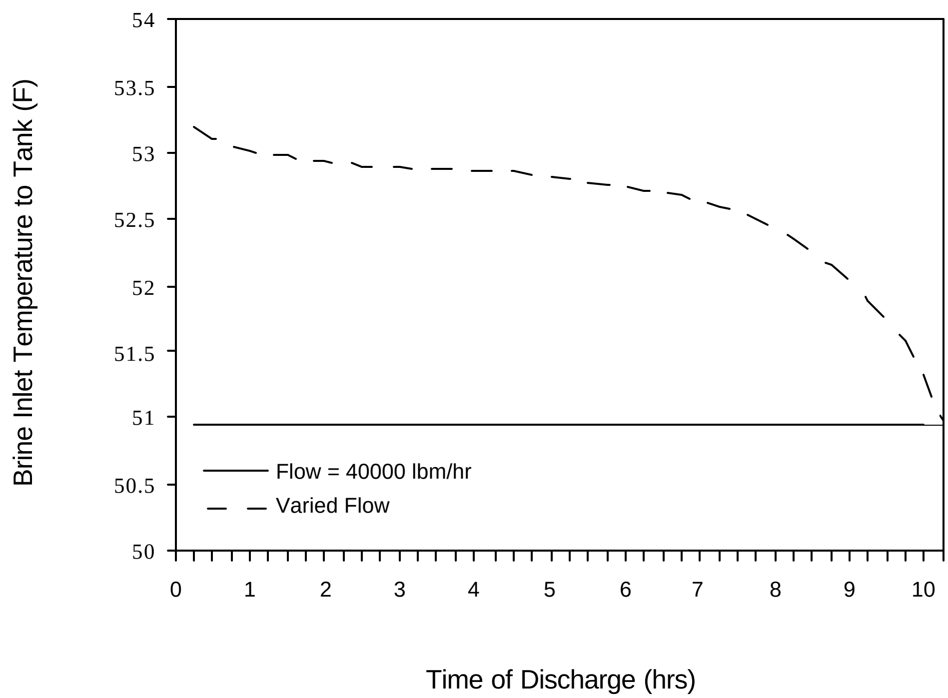


**Figure 3.25** Inlet brine temperature to the tank as a function of time for both the constant 40,000 lbm/hr flow with a flow diverter and the varied flow.

In figure 3.25, the inlet brine temperature to the tank is plotted against discharge time for both cases of variable flow and constant flow with diverter. By varying the flow, it can be seen that the inlet brine temperature to the tank, which is the temperature from the cooling coil, is higher than in the constant flow case. This higher inlet temperature translates to a slightly higher tank effectiveness. The largest part of the increase in tank effectiveness as seen in figure 3.23 is due to the decreased flow, however. The increase in inlet brine temperature becomes significant when a partial storage strategy is considered. If a chiller were placed upstream of the ice-storage tank the chiller would have a higher inlet temperature when the flow is varied. The higher inlet temperature would translate to a much higher chiller COP, which would mean the same amount of cooling could be provided with a lower electric power requirement.

In the preceding discussion the advantages of using the variable flow rate instead of using a bypass with the tank have been shown. This may cause one to wonder why the diverter is used in the first place. It is important to remember that the simulations were with dry air. When water is present in the air, there may be need for dehumidification. The amount of dehumidification is dependent on the temperature of the inlet brine flow to the cooling coil. As the brine temperature decreases further below the dew point of the air, more water will be removed. Thus, the diverter is required to control dehumidification.

In figure 3.26 the inlet brine temperature to the cooling coil is plotted against discharge time for both the variable flow and the constant flow with diverter. It can be seen that the varied or reduced brine flow temperature to the coil is significantly lower than the brine temperature to the coil produced by the constant flow through the coil with diverter. If the air did have some water content the amount of dehumidification would be much higher with the varied flow. Thus, the main reason for the diverter is to control the amount of dehumidification. Even though it is not practical to pump all the flow through the ice-storage tank, some flow reduction may be possible and would optimize performance.

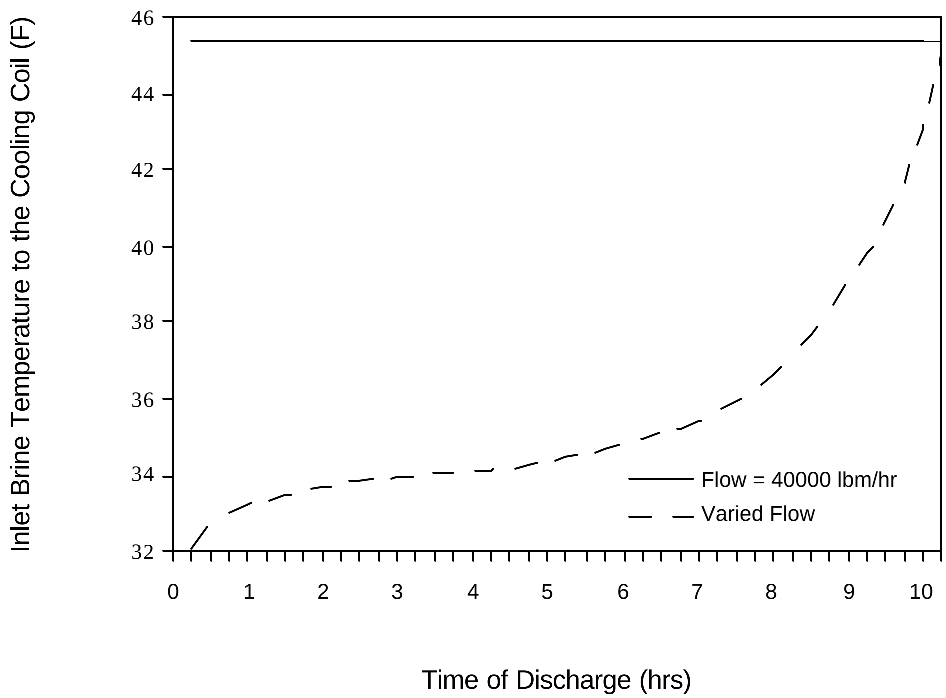


**Figure 3.26** Inlet brine temperature to the cooling coil as a function of time for both the constant 40,000 lbm/hr flow with a flow diverter and the varied flow.

## 3.8 Conclusions

In this chapter it was seen that for a constant building load, increasing both the water flow rate to the cooling coil and the inlet water temperature to the coil can significantly increase the time span at which the load can be met.

For different building load profiles that produce the same total integrated load over the same time span, 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. The ice-storage tank's capacity to meet a load is highly rate dependent. It was seen that higher loads can be better handled by the tank when they occur early in discharge.  A TRNSYS model was developed to properly size the ice-storage tank given a building load profile.

The use of the diversity factor as an indicator of tank size requirements was demonstrated. While the diversity factor gives an approximate figure of the required tank size, it cannot accurately predict the correct tank size.  The diversity factor was tested by taking two different building load profiles with the same diversity factor and checking if they required the same tank size; different tank sizes were required.

It was shown that by changing the variable air volume system to a constant air volume system with a higher air supply temperature to the coil, the same size tank can deliver a larger load. This can be useful when the amount of ice made is slightly less than what is required.

The effects of cooling coil size on ice-storage tank size were shown.  A model was developed that can properly size both was created.  The shape of the building load profile affects both the cooling coil size and ice-storage tank size.

The advantages of reducing the flow through the cooling coil in terms of lower pumping power and increased chiller COP (partial storage strategy) were discussed.  Complete flow reduction involving the elimination of the flow diverter is not possible, because dehumidification may require an upper limit on coil outlet temperature.

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2. *Levload Ice Bank Performance Manual*, Product Literature, Calmac Manufacturing Corporation, Englewood, New Jersey, April 1987.

3. Cummings, M. S., "Modeling, Design, and Control of Partial Ice-Storage Air-Conditioning Systems," M.S. Thesis, University of Wisconsin-Madison, 1989.

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CHAPTER

# FOUR

## PARTIAL LOAD STRATEGY

A partial load strategy uses a chiller and ice storage tank together to meet the building load. The partial load strategy is the most commonly used configuration because a chiller must be in place to make the ice. Figure 4.1 shows a system with the chiller and ice-storage tank in series with the chiller upstream. In the simulations cooling towers are used to condense the chiller brine. The cooling tower model and chiller used in the simulations in this chapter were developed by Braun [1].

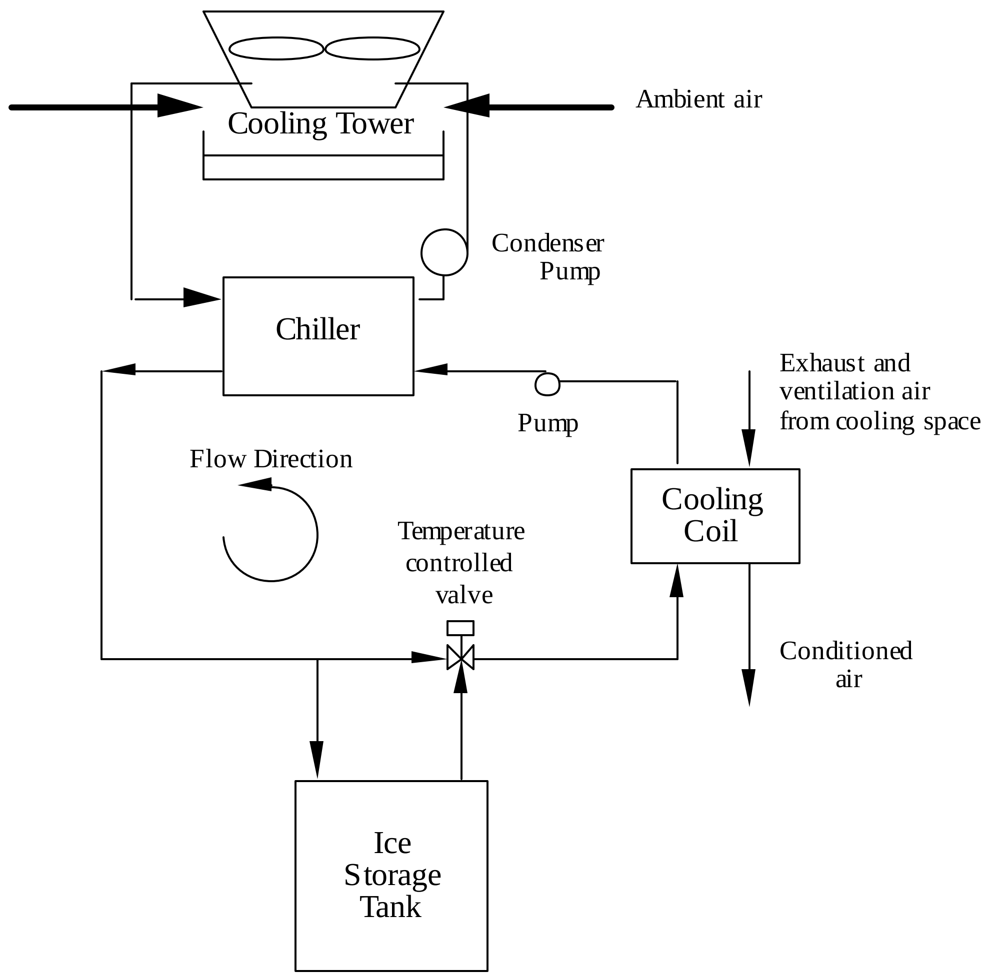


**Figure 4.1** Chiller and ice tank in series with the chiller upstream of the ice tank.

## 4.1 Chiller Operation Strategy

In this section the operation of a chiller, with a maximum capacity of $2.85 \times 10^5$ Btu/hr, with an ice-storage tank size of 190 Ton-hrs during discharge will be studied. The air inlet temperature to the coil was held constant at 75°F, the air into the cooling coil had no humidity, the air outlet temperature was held constant at 61.1°F, the ambient air was dry with temperature of 85$^o$F, and the water flow rate through the cooling coil was held constant at 40,000 lbm/hr. The constant load of $3.0 \times 10^5$ Btu/hr spanning over 12 hours was studied.

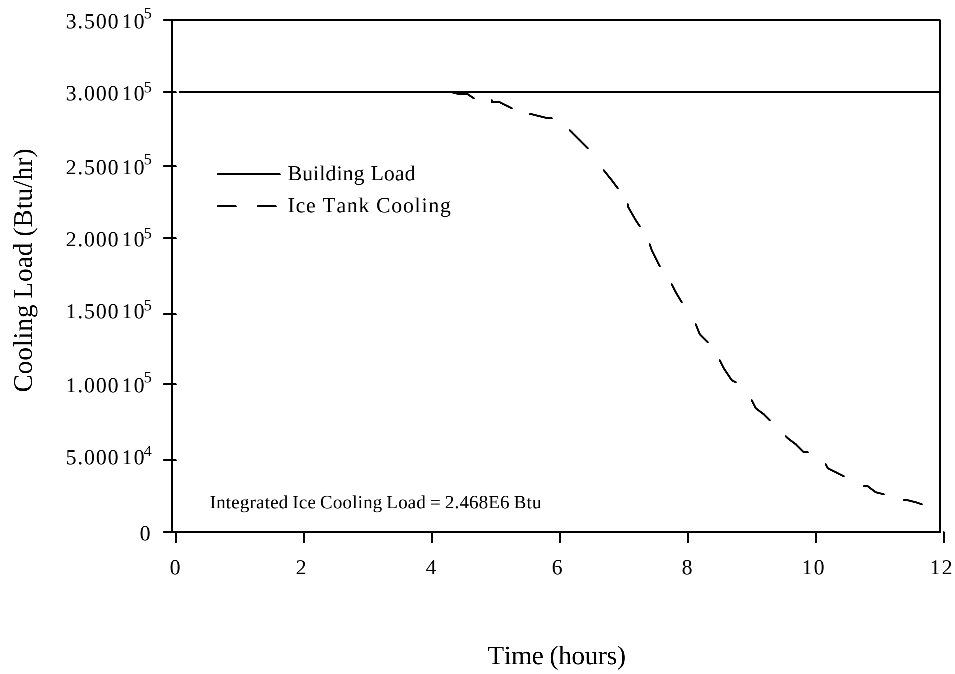


**Figure 4.2** Load met by ice-storage system when chiller is off.

In figure 4.2 the building load and load met by the ice-storage tank when the chiller is off are plotted with time. It is clearly seen that the ice-storage tank cannot meet the load without the help of a chiller, and after four hours the chiller must be turned on. The total integrated building over the twelve hours of discharge is $3.6 \times 10^6$ Btu, and the total integrated load met by the ice-storage tank over is $2.468 \times 10^6$ Btu. Thus, the ice-storage tank is able to meet 68.6% of the total building load when the chiller is off.

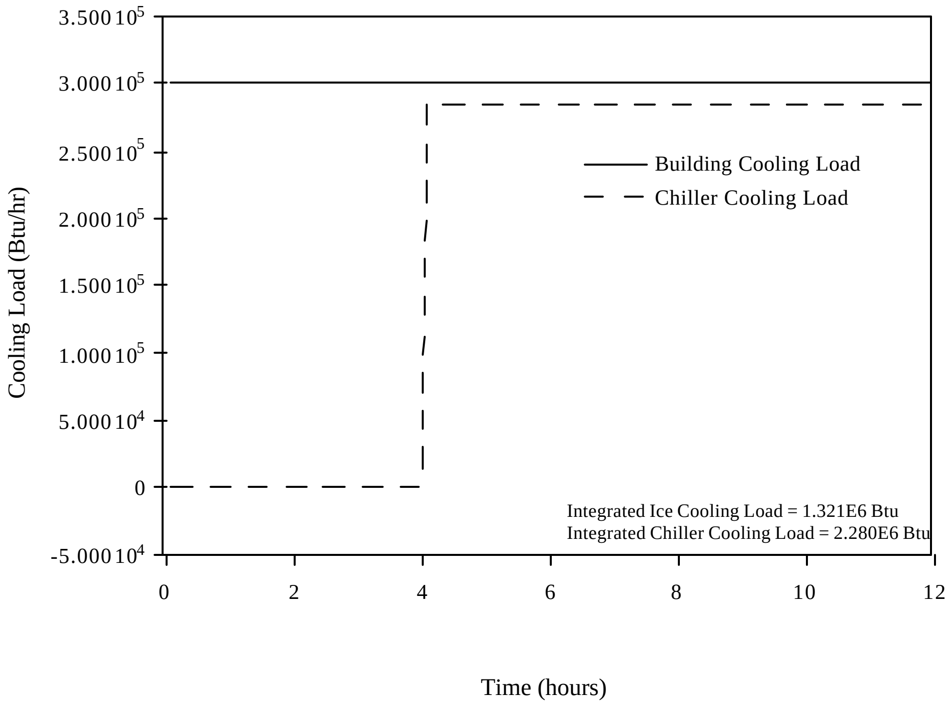


**Figure 4.3** Total load met and load met by chiller plotted against time.

In figure 4.3, the results for a chiller turned on when the ice-storage tank cannot meet the load rate required is shown. The chiller then is turned on to try to meet the entire load. In the figure it can be seen that the chiller does not have the capacity to meet the entire load by itself. The chiller capacity is $2.85 \times 10^5$ Btu/hr (23.75 Tons). Since the building load is $3.0 \times 10^5$ Btu/hr, the ice-storage tank must deliver the remaining $0.15 \times 10^5$ Btu/hr. In this strategy, the entire load is met, but the percentage of the load met by the ice-storage tank over the twelve hours of discharge is only 36.7%. Only 59.7% of the ice-storage tank's nominal capacity was discharged.

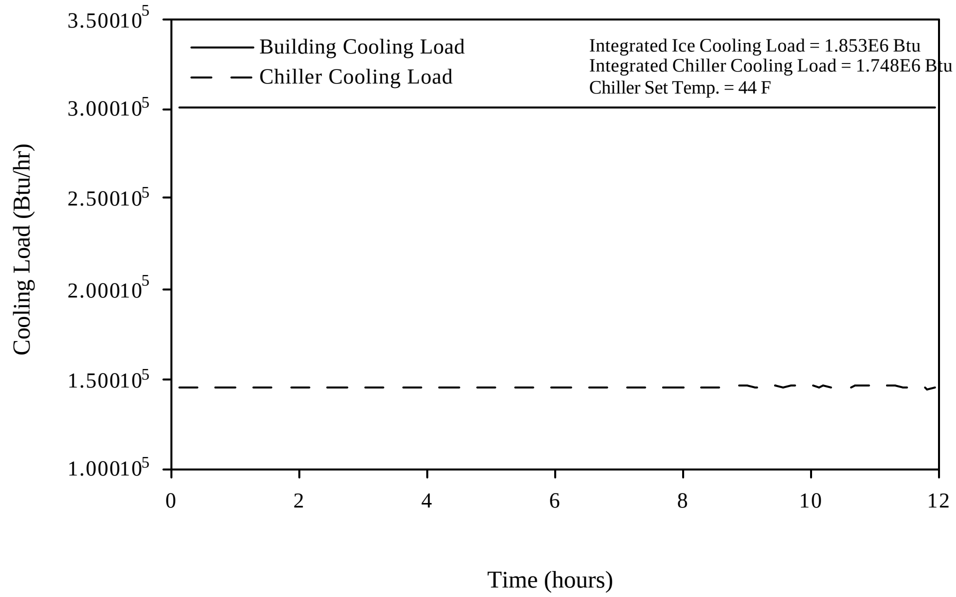


**Figure 4.4** Total load met and load met by chiller plotted against time.

In figure 4.4, the results for meeting the total building load by running the chiller at a constant set chilled water temperature are shown. In this strategy, 51.5% of the total building load is met by the ice-storage tank. 81.3% of the tank's nominal capacity was used to help meet the building load. In comparing these numbers to the numbers from figure 4.3, the numbers from figure 4.4 are much higher. Thus, in terms of building energy supplied by the ice, running a chiller at the maximum allowable chilled water set temperature is much better than just letting the chiller take over when the load cannot be met by the ice.

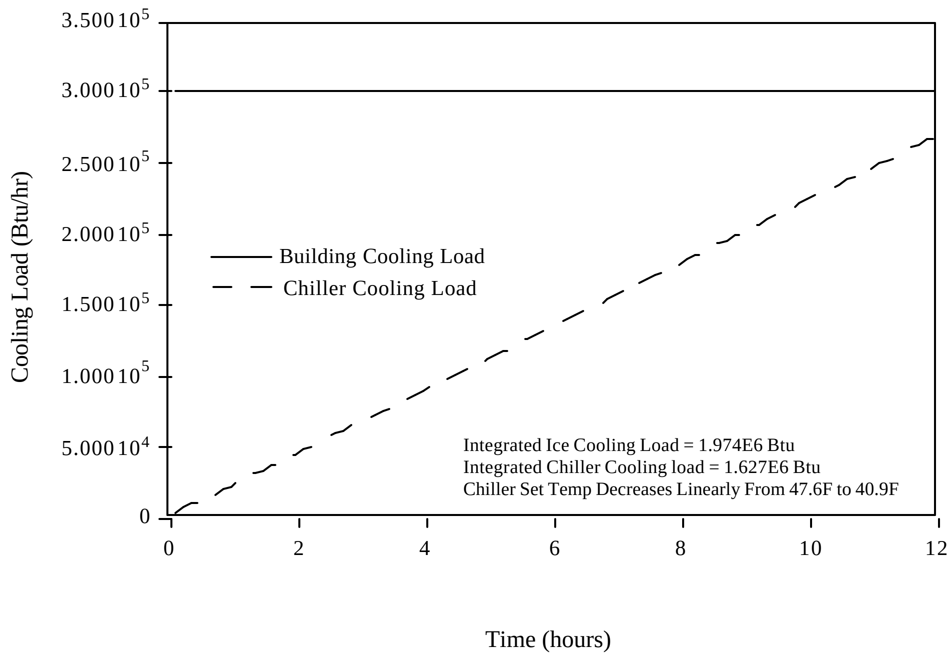


**Figure 4.5** Total load met and load met by chiller plotted against time.

In figure 4.5, the results for running the chiller with a linearly decreasing chilled water set temperature are shown. The percentage of the building load met by the chiller is linearly increasing over the twelve hour discharge period. In this strategy, 54.8% of the building load is met by the ice-storage tank. 86.6% of the tank's nominal capacity was used to help meet the building load.

In looking at figures 4.2 thru 4.5, there are some interesting information that can be culminated from them. In figures 4.3 thru 4.5 the chiller was run with different chiller load profiles, but in none of the cases, could the ice-storage tank discharge as much energy as the case when the chiller was off and the building load was not being met (figure 4.2). It is important to remember that the maximum capacity of the ice-storage tank is directly proportional to the inlet water temperature to the tank. When the chiller is running the inlet water temperature to the ice-storage tank is the same as the chilled water set temperature. This means that when the chiller is running to help meet the load the maximum capacity is smaller because the water temperature is not able to exceed the chilled water set temperature.

| | Chiller Off | Chiller Turned on to Full Capacity After 4 Hours | Chiller on at a Constant Chilled Water Set Temperature | Chiller on at a Linearly Decreasing Chilled Water Set Temperature |
|---|---|---|---|---|
| Integrated Load Met by the Ice-Storage Tank (Btu) | $2.468 \times 10^6$ | $1.321 \times 10^6$ | $1.853 \times 10^6$ | $1.974 \times 10^6$ |
| Percentage of Building Load Met by the Ice-Storage Tank (%) | | 36.7 | 51.5 | 54.5 |
| Percentage of Ice-Storage Tank's Nominal Capacity Discharged (%) | | 59.7 | 81.3 | 86.6 |

**Table 4.1** Ice-storage performance for various chiller operation strategies.

Another important finding was that among the three different chiller load profiles used to meet the constant building load, the linearly increasing chiller load maximized the fraction of the building load met by the ice as shown in table 4.1. It is important to note that the linearly increasing chiller load worked best for a constant building load, but other chiller load profiles may work better for other building load profiles.

## 4.2 Effect of Water Flow Rate on Performance

In most ice-storage systems used today the chiller is run at a constant load. For a constant building load, the chilled water set temperature will remain constant for a fixed water flow rate through the cooling coil. For each flow rate, there is a maximum chilled water set temperature that will just allow the total building load to be met for the specified time. Each flow rate has an optimum chilled water set temperature. Simulations were performed with a constant air inlet temperature of 75$^o$F, no humidity, a constant air outlet temperature from the cooling coil, an ambient air temperature of 85$^o$F, an ambient relative humidity of zero, and with water flow rates of 30,000 lbm/hr, 40,000 lbm/hr, 50,000 lbm/hr, and 60,000 lbm/hr. In figure 4.6 it can be seen that by increasing the water flow rate through the cooling coil, the percentage of the building load delivered by the chiller decreases.

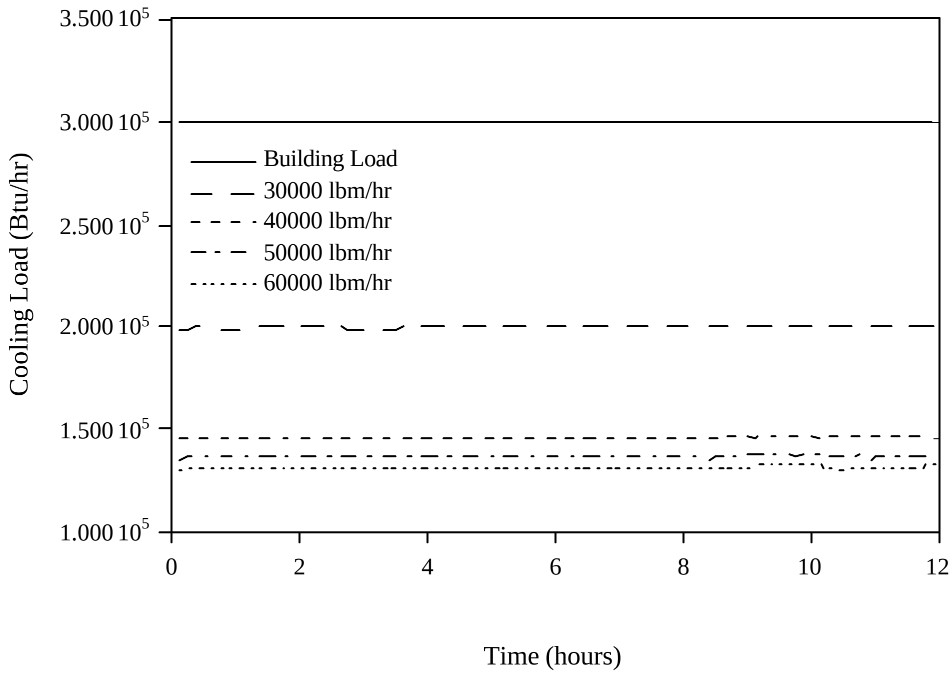


**Figure 4.6** Chiller load deliverd with time for various water flow rates through the cooling coil

In figure 4.7, the fraction of the building load met by the ice-storage tank is plotted against the flow rate through the cooling coil. It can be seen that the total load delivered by the ice increases with flow rate through the cooling coil at a decreasing rate.

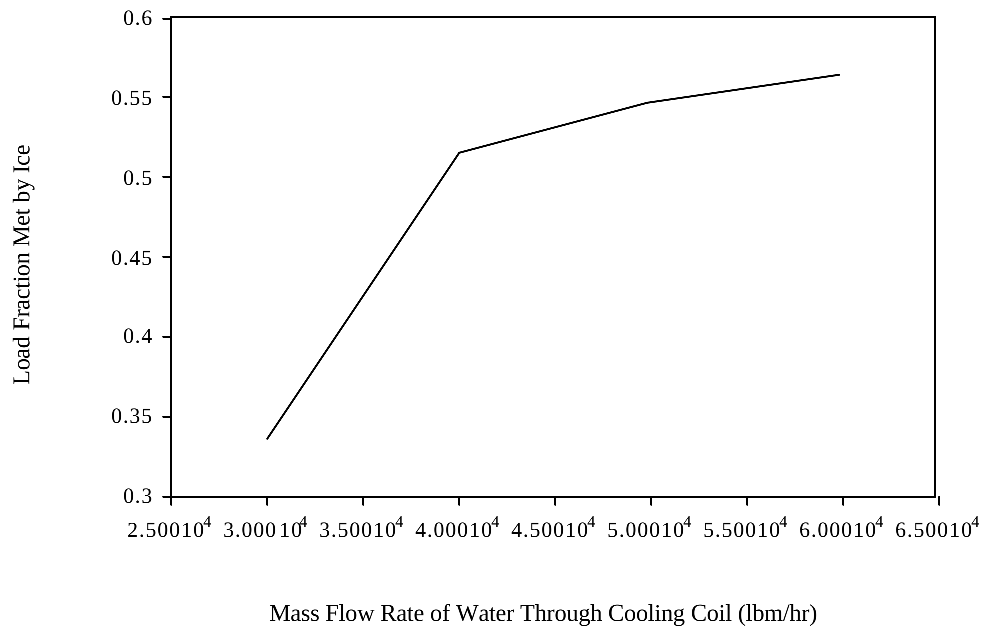


**Figure 4.7** Fraction of building load met by ice as a funtion of flow rate through cooling coil.

By increasing the flow rate through the cooling coil, the fraction of the building load met by the ice-storage tank increased. This means that the chiller load delivered must decrease when the flow rate increases. In addition, increasing the flow rate increases the set temperature of the water into the cooling coil. This, in turn, increases the chilled water set temperature. When the chilled water set temperature increases, the COP of the chiller increases, because the temperature difference between the evaporator and condenser is smaller.

In figure 4.8 the chiller COP is plotted against the fraction of the building load met by the ice-storage tank. The chiller COP increases because the chilled water set temperature increases, which will increase the fraction of the load met by the ice-storage tank.

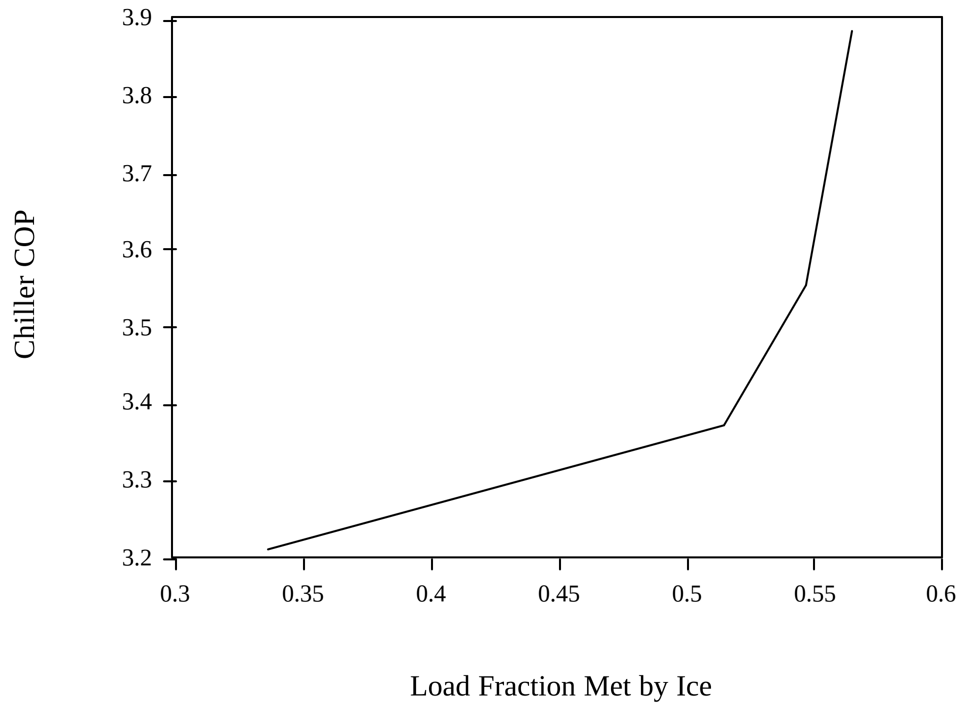


**Figure 4.8** Chiller COP as a function of the load fraction met by ice

Since the load required from the chiller decreases and the chiller COP increases with increasing water flow rate through the cooling coil, the culmination of the two effects result in a decreasing chiller power with flow rate through the cooling coil as seen in figure 4.9.

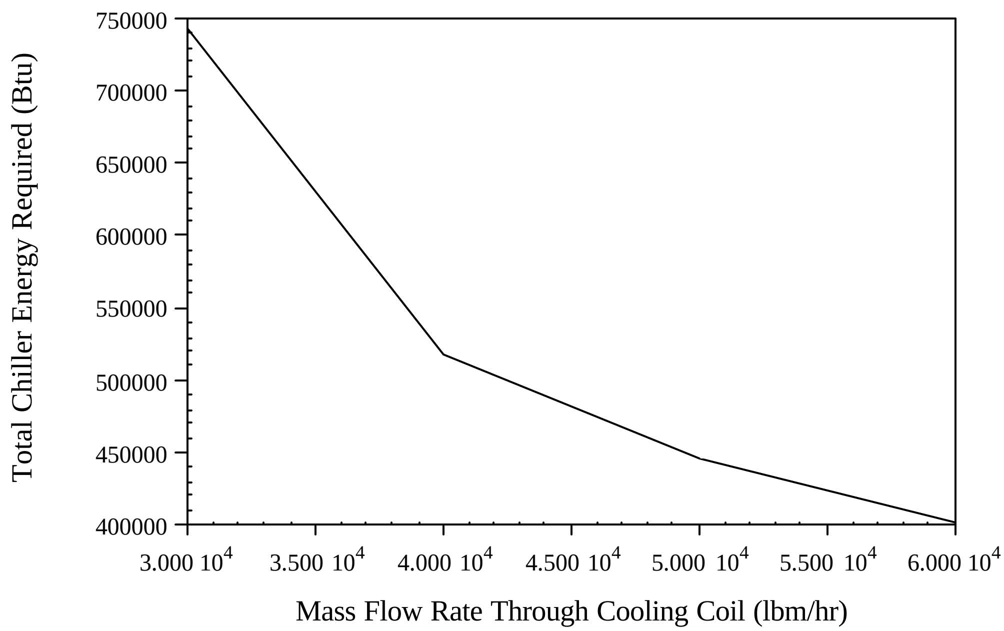


**Figure 4.9** Total chiller electrical energy required as a function of mass flow rate through the cooling coil

## 4.3 Effect of Load Profile on Tank Size and Chiller Size

To this point only the discharge period of the tank has been considered. In analyzing the full storage strategy, the required tank size for a given load profile is, to some degree, independent of chiller size, because the chiller is not run during the discharge period. In the partial load strategy the chiller and ice-storage tank are used together to meet the load. Since they must work together, they are coupled. This coupling requires the analysis to span both the charging and discharging periods. To begin a study of the sizing of both the chiller and the ice-storage tank, three different load profiles were studied. Simulations were run with the same conditions as in the previous section except the water flow rate to the cooling coil was constant at 40,000 lbm/hr.

The three load profiles used are shown in figure 4.10. The three loads consist of: a linearly decreasing load, a constant load, and a linearly increasing load. All three load span a total discharge period of 10 hours and have the same total energy requirement of $2.0 \times 10^6$ Btu. Figure 4.10 shows the same load profiles for a period of three days. A three day analysis allows the effect of the initial ice-storage tank charge to disappear.

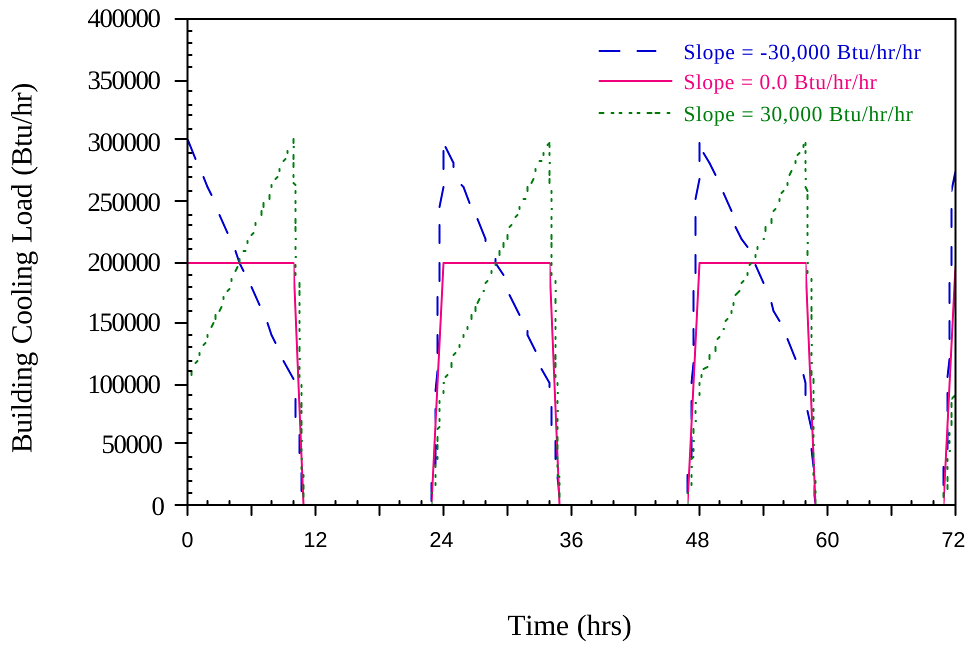


**Figure 4.10** Three building cooling load profiles as a function of time.

In the simulation, the chiller was placed upstream of the ice-storage tank. The chiller was cooled using three cooling towers. The chiller was run at maximum capacity. In a real chiller the maximum capacity is a rating at certain room and ambient conditions. In operation the cooling space and ambient conditions will vary. This means that the maximum cooling load delivered by a chiller will vary. The chiller model used in the simulation assumes a constant maximum capacity regardless of cooling space and ambient conditions. For the simulations the chiller size was changed by changing the maximum capacity of the chiller. In the simulations combinations of chiller size and ice-storage tank size were found that will just meet the load.

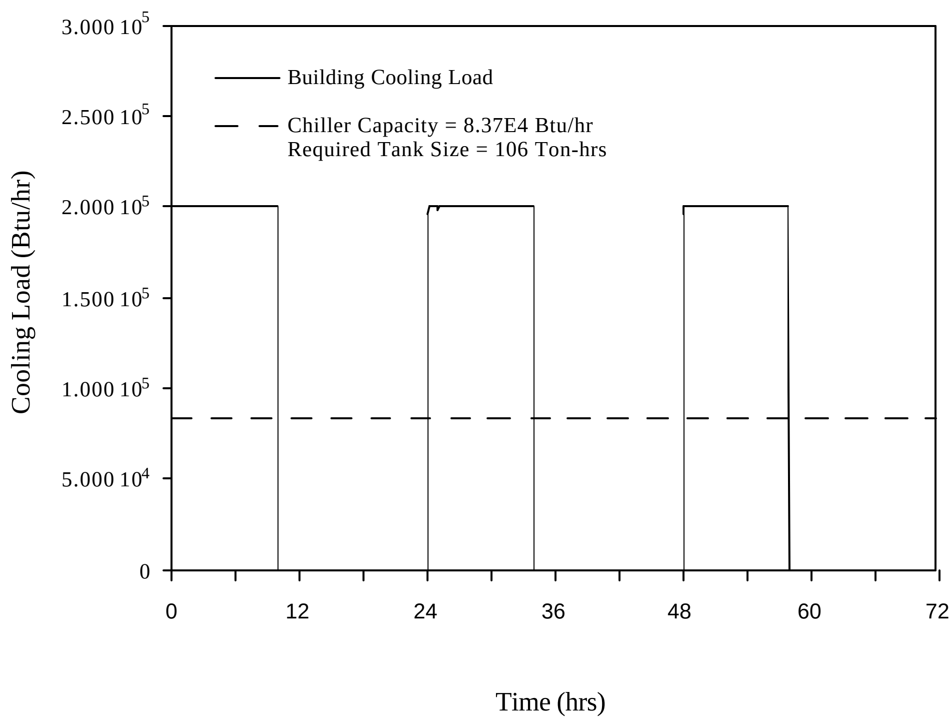


**Figure 4.11** The building cooling load and the minimum chiller load possible plotted against time for the constant building load of 2.0E5 Btu/hr.

In the simulations it was found that there is a minimum allowable chiller capacity. If the chiller is any smaller, it will not be able to fully recharge the ice-storage tank. In figure 4.11 the minimum allowable chiller cooling load is plotted for the constant building load of $2.0 \times 10^5$

Btu/hr. If, in the simulations, the chiller size was smaller than this minimum value and the ice-storage tank size appropriately increased to just meet the load for the first day, the same load could not be met for any subsequent day. The chiller capacity would be to small to replenish the ice depletion in the ice-storage tank. The minimum chiller size will replenish the ice to the same energy storage found at the same time twenty-four hours earlier.

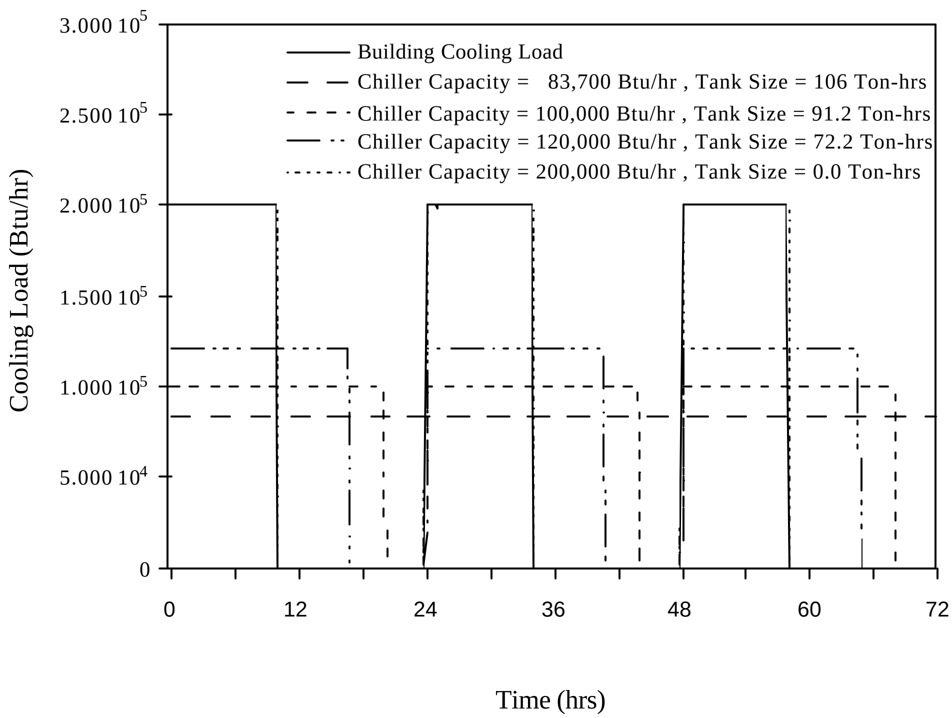


**Figure 4.12** Chiller cooling loads vs. time for a constant building cooling load of 2e5 Btu/hr.

In meeting the constant load there are a combination of chiller sizes and ice-storage tank sizes that will meet the load requirements for the three day period. In figure 4.12 various chiller cooling loads are plotted with respect to the total cooling building load. As the chiller capacity is increased, the percentage of the cooling load delivered by the ice-storage tank is decreased. This means that the ice-storage tank size can be decreased. Since the chiller size increases and the ice-storage size decreases, the time required to fully charge the ice-storage tank size decreases. In the simulation the chiller is turned off when the ice-storage tank is fully charged. In figure 4.12 it can be seen that the time of charging does decrease with increasing chiller size.

In figure 4.13 the linearly increasing load profile is plotted with a series of chiller capacities that will meet the load. It can be seen in this figure that as the chiller size is increased there will be times when the chiller capacity exceeds the building cooling load requirements. In this circumstance, the chilled water set temperature is increased until the chiller just meets the building cooling load.

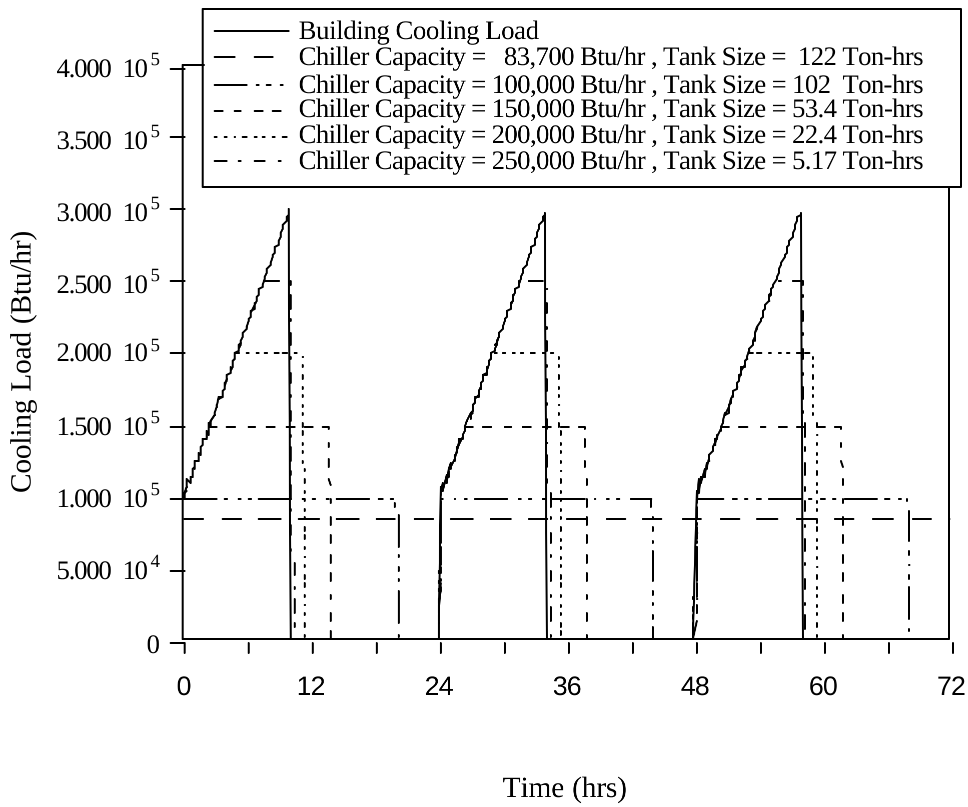


**Figure 4.13** Chiller cooling loads vs. time for a linearly increasing building cooling load with slope of 30,000 Btu/hr/hr

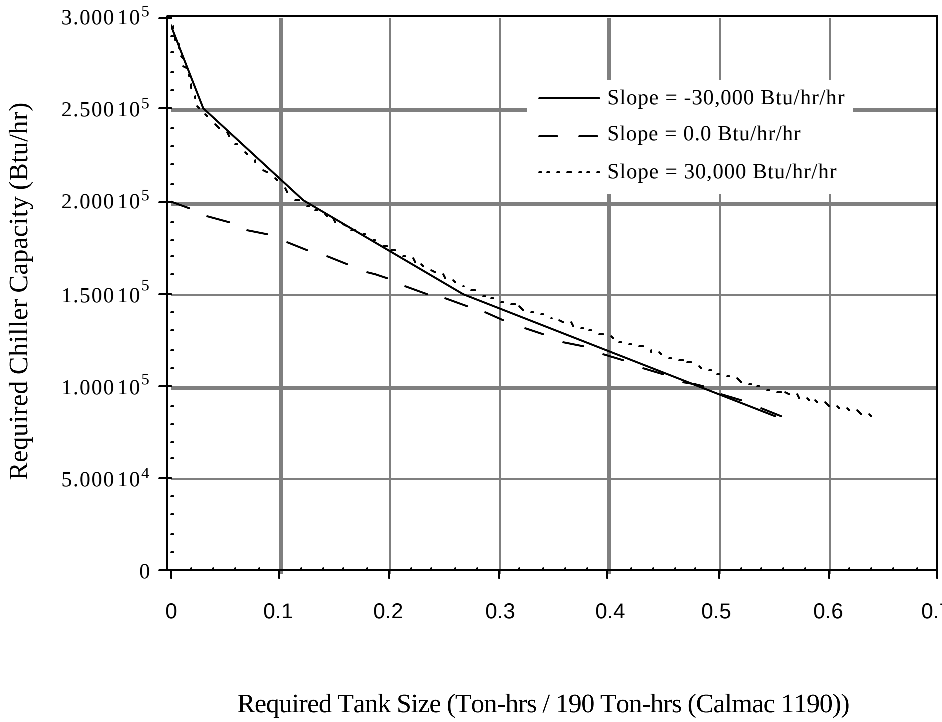


**Figure 4.14** Required chiller capacity as a function of tank size for three linear load profiles with different slopes.

In figure 4.14, the various combinations of chiller size are plotted against the required ice-storage tank size for the three linear load profiles. It can be seen that there is a minimum chiller size that is independent of the load profile. The reason that the minimum chiller size is independent of the load profile is that the amount of energy depletion in the ice storage tank is constant. This means that even though a larger or smaller tank is required, the total energy discharged is the same. Since the energy depletion is constant for the load profiles, the same minimum chiller size will replenish this energy. The minimum chiller size can be related to the average building load, the total charging time, and the total discharging time by equation 4.1.

$$\text{Min. Chiller Capacity} = \frac{\text{Ave. Load}}{\left(1+\dfrac{t_{charge}}{t_{discharge}}\right)} \quad (4.1)$$

For the average building load of $2.0 \times 10^5$ Btu/hr, a total charging time of fourteen hours, and a total discharging time of ten hours, the minimum chiller capacity found through simulations was $8.37 \times 10^4$ Btu/hr. Equation 4.1 predicts a chiller capacity of $8.33 \times 10^4$ Btu/hr. Thus, there is a .44% difference between equation 4.1 and the simulation results. This error is within the tolerances set in the simulation.

Why is the chiller size a linear function of the required tank size for the constant load profile in figure 4.14 ? The case of the constant load is a unique. It is the only case where the chiller capacity never exceeds the cooling load during daytime hours. Since the amount of energy depletion from the ice-storage tank is dependent on both the load profile and the chiller capacity, the tank size will be a linear function of chiller size since the load is constant. The linearly increasing and decreasing load profiles also exhibit this linear relation until the chiller capacity exceeds 100,000 Btu/hr. Any larger capacity chiller will exceed the minimum cooling load of 100,000 Btu/hr and will result in a nonlinear relation between the chiller size and the ice-storage tank size.

Why do the linearly increasing and linearly decreasing loads have very different ice-storage tank sizes at low chiller sizes and the same ice-storage tank sizes at high chiller capacities? At the minimum chiller capacity of 83,700 Btu/hr, the ice-storage tank is discharging for 9.875 of the total 10 hours of daytime load. At a chiller capacity of 250,000 Btu/hr, the ice-storage tank is discharging for only 2.5 hours of the total 10 hours of daytime load. As the hours of ice-storage tank discharge decrease, the change in load profile as seen by the tank is decreased. Since the load profile seen by the ice-storage tank is decreased, the tank is less dependent on the load profile.

## 4.4 Analysis of Representative Building Loads

In the previous section a linearly increasing load, a constant load, and a linearly decreasing building load were studied. In this section three simulated building loads will be studied. The representative building loads are the same as the simulated building loads used in the full storage strategy. The easiest way to identify the three loads is by there diversity factor. The three representative building loads have diversities of .83333, .66667, and .65138. In the simulations, the same conditions were used as in the previous section except that the ambient air temperature and relative humidity were based on the design day for Madison [2].

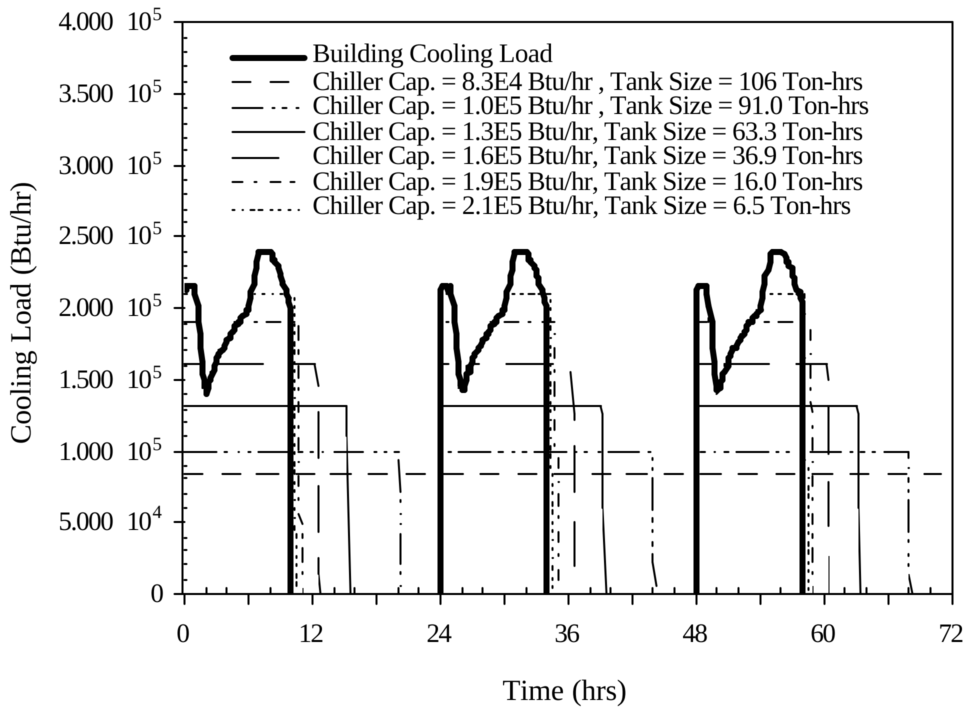


**Figure 4.15** Chiller cooling loads vs. time for a representative building load with a diversity of .83333.

In figure 4.15 the representative building load with a diversity of .83333 is plotted with various chiller sizes. The minimum building load is $1.4x10^5$ Btu/hr and occurs at the second hour of discharge. In figure 4.16 the required chiller capacity is plotted against the required tank size for the simulated building load with a diversity of .83333 for air outlet temperatures of 61.1$^o$F and 50$^o$F. The relation between chiller size and ice-storage tank size is linear until the chiller size exceeds $1.4x10^5$ Btu/hr. For the 61.1$^o$F air outlet temperature, the nonlinear relation above a chiller size of $1.4x10^5$ Btu/hr is where a linear increase in chiller size does not correspond to a linear increase in cooling load delivered by the chiller. This is the same relation seen for the linearly increasing and decreasing load profiles. The relation between chiller size and ice-storage tank size remains linear for a large part of figure 4.16. Thus, a diversity factor close to one translates to a largely linear relation between chiller size and ice-storage tank size.

The effect of air temperature was evaluated by simulating the system with a coil outlet temperature of 50°F. As shown in figure 4.16, as the desired air outlet temperature is decreased, the tank size required to meet the building load increases for a given chiller capacity. A decrease in air outlet temperature will require a decrease in the water temperature to the cooling coil which will increase the flow rates through the ice-storage tank. Since the maximum flow rate through the ice-storage tank is fixed, a larger tank size will be required to meet the building load. The tank sizes, for the two different air outlet temperatures, converge as the chiller capacity increases.

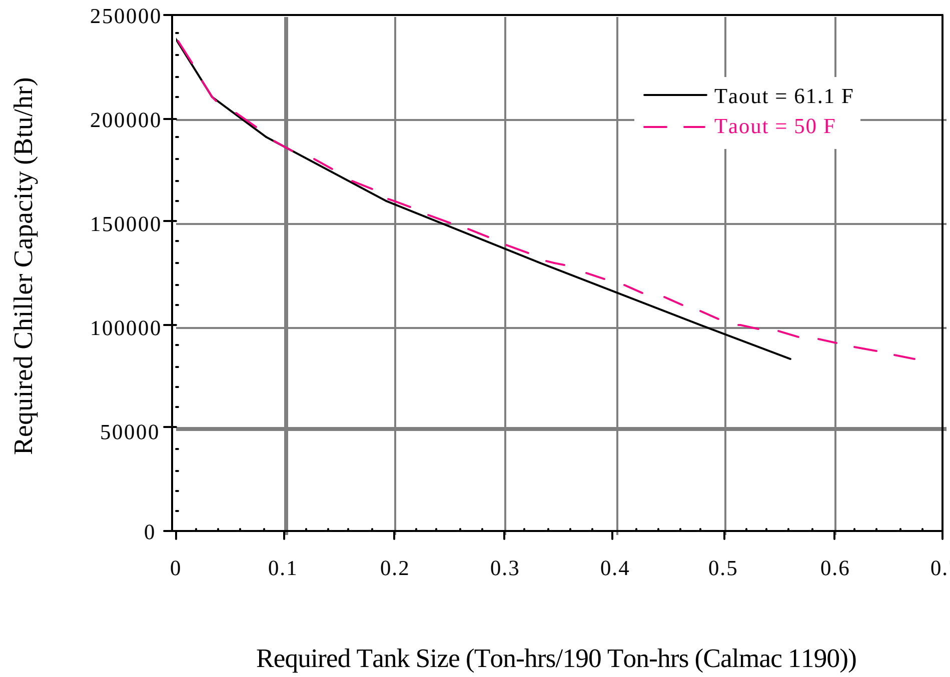


**Figure 4.16** Required chiller capacity as a function of tank size for the representative building load with a diversity of .83333

In figure 4.17 the representative building load with a diversity of .66667 is plotted along with various chiller capacities. The minimum building load of $1.0x10^5$ Btu/hr occurs at the second hour of discharge.

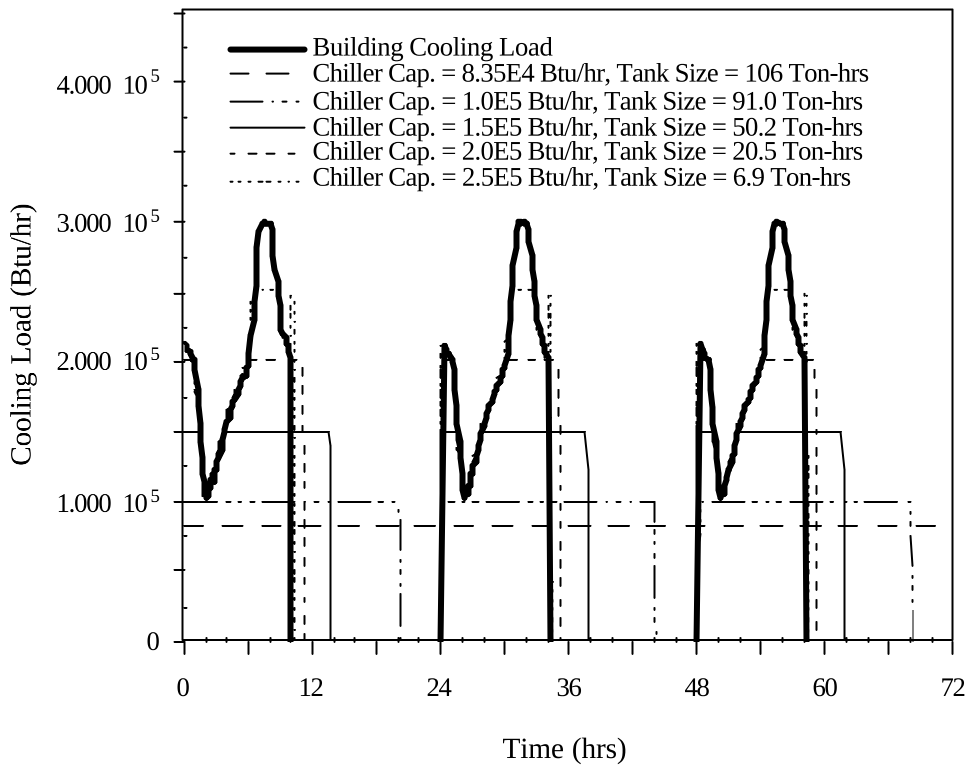


**Figure 4.17** Chiller cooling load vs. time for a representative building load with a diversity of .66667.

The required chiller capacity is plotted against the require ice-storage tank size for the representative building load with a diversity factor of .66667 in figure 4.18 for air outlet temperatures of 61.1°F and 55°F. In comparison to the line for a diversity factor of .83333, the line for a diversity factor of .66667 is less linear. Because the increase in chiller capacity does not correspond to a linear increase in percentage of the building load met by the chiller, the relation between chiller capacity and ice-storage tank size is nonlinear.

Figure 4.18 shows that as the desired air outlet temperature is decreased, the tank size required to meet the building load increases for a given chiller capacity. A decrease in air outlet temperature will require a decrease in the water temperature to the cooling coil which will increase the flow rates through the ice-storage tank. Since the maximum flow rate through the ice-storage tank is fixed, a larger tank size will be required to meet the building load. The tank sizes, for the two different air outlet temperatures, converge as the chiller capacity increases.

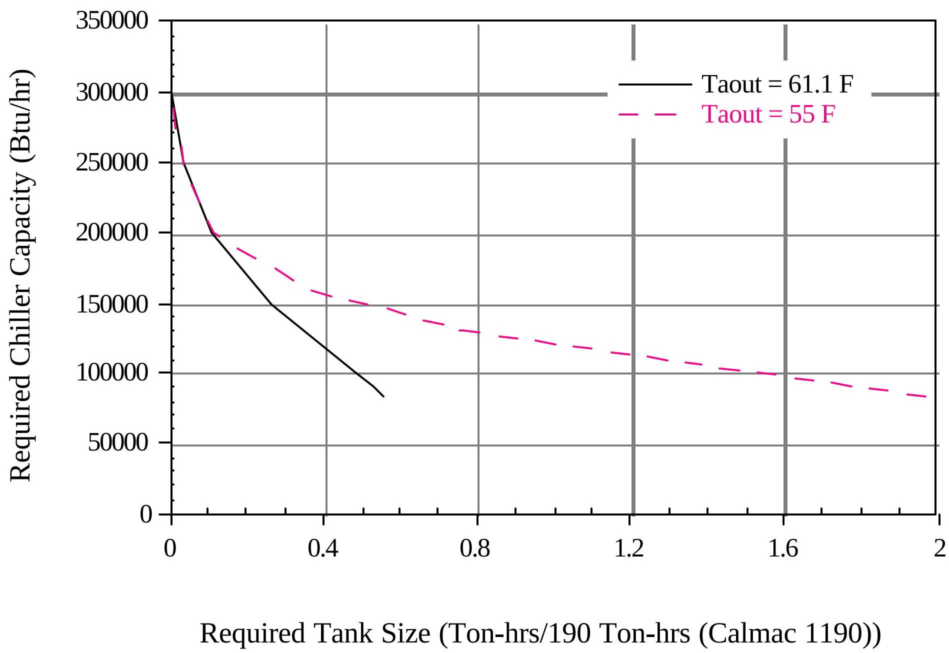


**Figure 4.18** Required chiller capacity as a function of tank size for a representative building load with a diversity of .66667.

Various chiller load capacities are plotted with the representative building load with diversity factor of .61538 as shown in figure 4.19. The peak load is $3.25 \times 10^5$ Btu/hr. With the increasing peak load, the chiller, for each size except for the minimum size, is not run at full capacity. Thus, the chiller sizes will not be linearly related to the percentage of the building load met by the chiller.

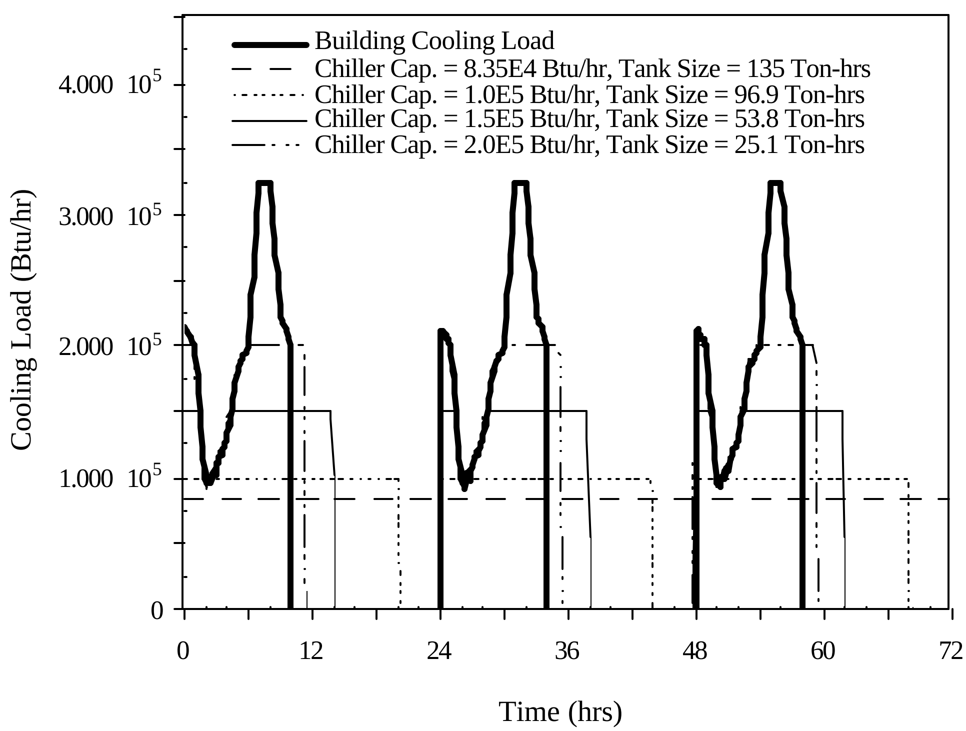


**Figure 4.19** Chiller cooling load vs. time for a representative building load with a diversity of 61538.

The nonlinear relation between chiller capacity and cooling load met by chiller translates into a nonlinear relation between the chiller capacity and ice-storage tank size for the representative building load with diversity factor of .61538 as shown in figure 4.20. With the higher peak load, there is an associated higher chiller capacity for the chiller strategy (no ice-storage).

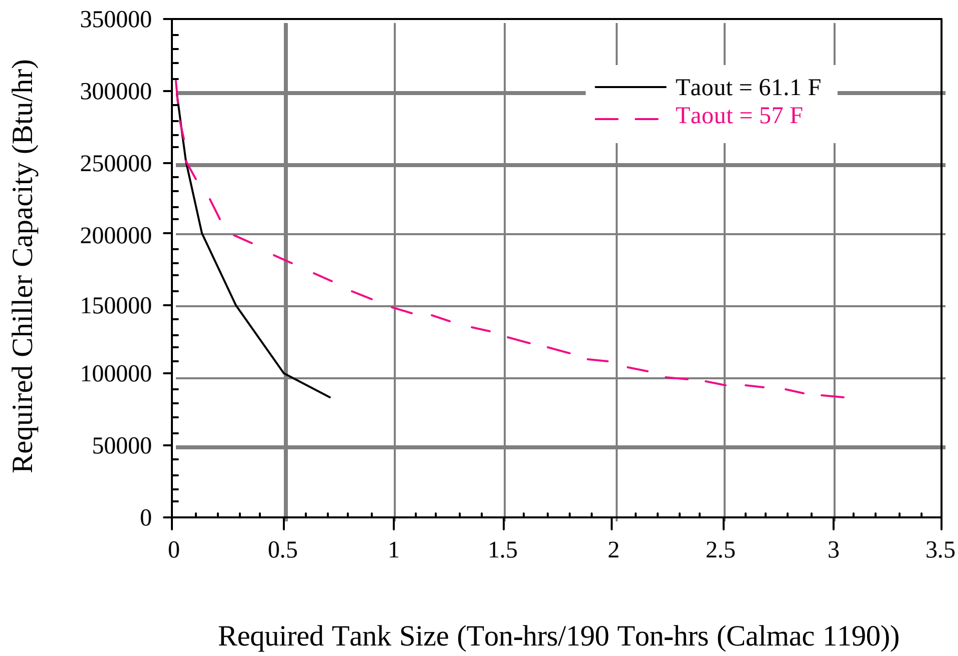


**Figure 4.20** Required chiller capacity as a function of tank size for a representative building load with a diversity of .61538.

Figure 4.20 shows that as the desired air outlet temperature is decreased, the tank size required to meet the building load increases for a given chiller capacity. A decrease in air outlet temperature will require a decrease in the water temperature to the cooling coil which will increase the flow rates through the ice-storage tank. Since the maximum flow rate through the ice-storage tank is fixed, a larger tank size will be required to meet the building load. The tank sizes, for the two different air outlet temperatures, converge as the chiller capacity increases.

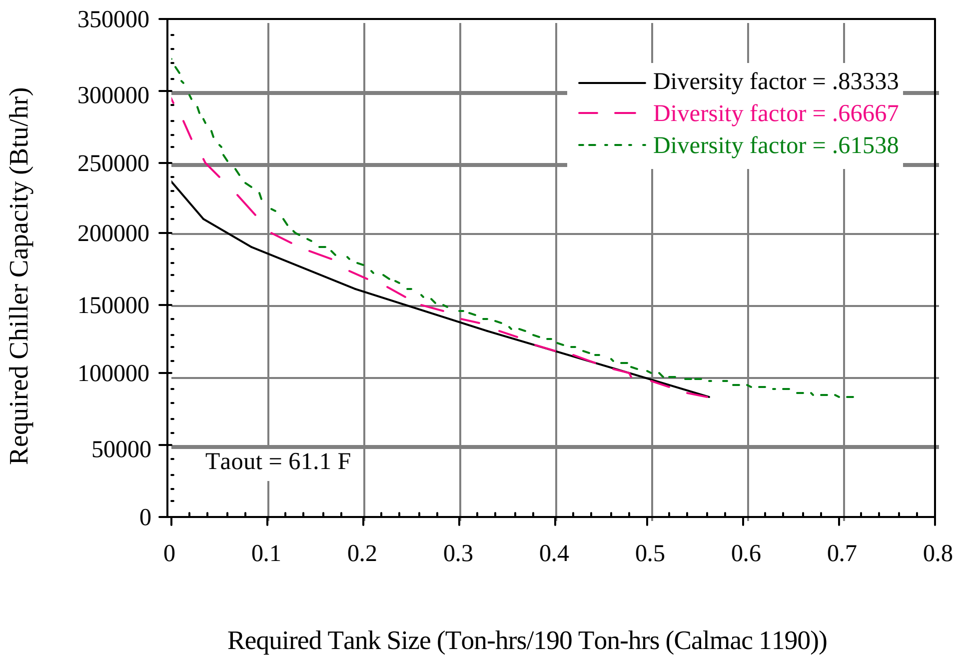


**Figure 4.21** Required chiller capacity as a function of tank size for the three building load profiles for an air outlet temperature of 61.1°F.

In figure 4.21 the required chiller size is plotted against the required ice-storage tank size for the three building load profiles with an air outlet temperature of 61.1°F. As the peak load increases, the required tank size increases at the minimum chiller capacity. The increased tank capacity is required to increase the cooling rate sufficiently to meet the load.

## 4.5 Operation Guideline After Sizing

In sizing the chiller and ice-storage tank, the chiller was sized to replenish the energy depleted from the ice-storage tank within the available charging time. The energy taken from the ice-storage tank may not be all the available energy within the tank since the tank is never fully discharged. Thus the chiller may not be large enough to completely recharge an ice-storage tank that is initially at zero capacity within the required time.

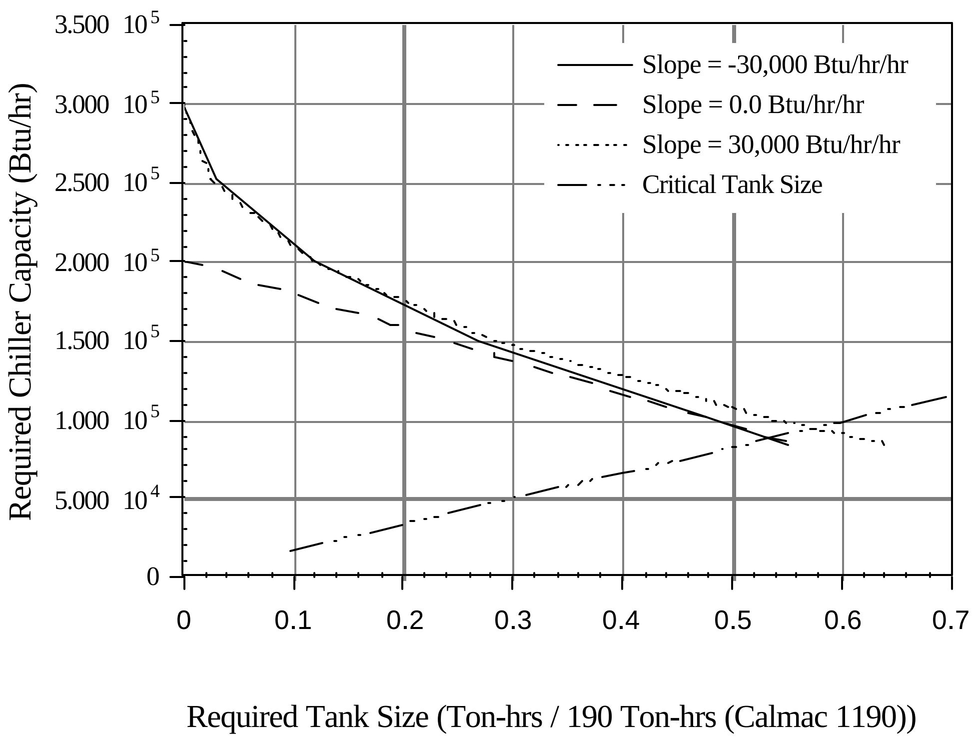


**Figure 4.22** Required chiller capacity as a function of tank size for three linear load profiles. The required chiller capacity vs. critical tank size.

In figure 4.22, the required chiller capacity is plotted against required tank size for three linear load profiles with different slopes and the critical tank size. The critical tank size is the maximum allowable size that will allow the required chiller to completely charge the ice-storage tank in the specified charging time. The critical tank size line shown in figure 4.22 is for a fourteen hour charging period. A decrease in the charging time would decrease the critical tank size for a fixed chiller capacity.

The critical tank size represents a caution in the operation of the ice-storage tank. If the tank size is smaller than the critical tank size, then the system can be charged in one charging period without regard to the the second forthcoming day's load. If the tank size is larger than the critical tank size, then an analysis of the week should be performed to make sure that the tank is not completely drained prior to a design day load.

## 4.6 Chapter Summary

In this chapter it was seen that the method of operation of the chiller during discharge can significantly affect the percentage of the load met by the ice-storage tank and the electric power requirement of the chiller. For a constant load profile, a linearly increasing chiller cooling load, during discharge, delivered maximized the usage of energy from the ice-storage tank.

For a chiller operating with constant cooling load, increasing the water flow rate through the cooling coil will increase the chilled water set temperature, increase the COP, increase the fraction of cooling load delivered by the ice-storage tank, and decrease the chiller's electrical power requirement.

For three linear load profiles, a minimum chiller capacity was found that was the same in all three cases, even though the required ice-storage tank sizes were different. For the constant load the relation between chiller capacity and ice-storage tank size was linear. While the increasing and decreasing load profiles had different tank sizes at the minimum chiller capacity, the ice-storage tank sizes of the load profiles converge with increasing chiller capacity.

The three representative building loads, with diversity factors of .83333, .66667, and .61538, showed that the relation between chiller size and required ice-storage tank size goes from nonlinear to linear as the diversity factor goes from .5 to 1.0.

In the operation of the partial load system, the critical tank size for the tank should be checked for a given chiller capacity. If the tank size is larger than the critical tank size, then an analysis should be done for the week to make sure the ice-storage tank can be fully charged during every charging period.

# *REFERENCES 4*

1. Braun, James Edward, "Methodologies for the Design and Control of Central Cooling Plants," Ph.D Thesis, University of Wisconsin - Madison, 1988.

2. *ASHRAE Handbook*, *Fundamentals Volume*, 1989, American Society of Heating, Regfrigeration, and Air-Conditioning Engineers Inc., Atlanta, Georgia.

CHAPTER

# FIVE

# SYSTEMS COMPARISON

In chapters three and four, three representative building loads were used to determine the minimum chiller capacity and the minimum ice-storage tank size required. In this chapter, the sizes of both the chiller and ice-storage tank will be compared for a conventional chilled water air-conditioning system, a full load ice-storage system, and a partial load ice-storage system for alll three building profiles. For the partial load strategy, the sizing of the ice-storage tank corresponding to the minimum chiller capacity was chosen for comparison.

## 5.1 Systems Comparison

For the representative building load with a diversity of .83333, the required chiller capacities and tank sizes for the three systems are shown in table 5.1. The chiller required by the partial load strategy is almost three times smaller than the chiller required in a traditional chiller air-conditioning system. The partial storage strategy requires both chiller and ice-storage tank sizes that are almost two times smaller than the sizes required by the full storage system.

| | Conventional System | Full Storage System | Partial Load System |
|---|---|---|---|
| Chiller Capacity (Btu/hr) | $2.40 \times 10^5$ | $1.43 \times 10^5$ | $.835 \times 10^5$ |
| Ice-Storage Tank Size (Ton-hrs) | 0.0 | 190 | 106 |

**Table 5.1** Required chiller capacity and ice-storage tank sizes for three systems for the building load with a diversity factor of .83333.

For the representative building load with a diversity of .66667, the required chiller capacities and tank sizes for the three systems are shown in table 5.2. The chiller size for the conventional system is substantially larger than that required for the building load with a diversity factor of .8333, while the chiller size required in both ice-storage systems does not change. The chiller size for the conventional system is determined by the peak load, while the chiller size for the ice-storage systems is mainly determined by the energy depletion from the ice-storage tank. The ice-storage tank size for the full storage system increases from the decrease in diversity factor while the ice-storage tank size for the partial load system doe not change. Since 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 storage system.

| | Conventional System | Full Storage System | Partial Load System |
|---|---|---|---|
| Chiller Capacity (Btu/hr) | $3.00x10^5$ | $1.43x10^5$ | $.835x10^5$ |
| Ice-Storage Tank Size (Ton-hrs) | 0.0 | 217 | 106 |

**Table 5.2** Required chiller capacity and ice-storage tank sizes for three systems for the building load with a diversity factor of .66667.

For the representative building load with a diversity of .661538, the required chiller capacities and tank sizes for the three systems are shown in table 5.3. With this further decrease in diversity, the ice-storage tank size more than doubles from the size required for a representative building load with a diversity factor of .66667 for the full load system, while the ice-storage tank size for the partial load system increases by 30%. The use of the chiller and ice-storage collectively to meet the load reduces the ice-storage tank's size dependence on the load profile.

| | Conventional System | Full Storage System | Partial Load System |
|---|---|---|---|
| Chiller Capacity (Btu/hr) | $3.25x10^5$ | $1.43x10^5$ | $.835x10^5$ |
| Ice-Storage Tank Size (Ton-hrs) | 0.0 | 464 | 135 |

**Table 5.3** Required chiller capacity and ice-storage tank sizes for three systems for the building load with a diversity factor of .61538.

## 5.2 Chapter Summary

In this chapter it was shown that while the chiller size is highly dependent on the peak load for a conventional air-conditioning system, the chiller size for an ice-storage system is dependent on the energy depletion of the ice-storage tank. Thus, an ice-storage system can significantly reduce the required chiller size for a given design day load.

The chiller size and ice-storage tank size for the partial storage system were almost half the sizes required by a full storage system. In addition, the partial storage system's ice tank size was less affected by a change in load profile than in the full storage system.

CHAPTER

# SIX

# CONCLUSIONS & RECOMMENDATIONS

## 6.1 CONCLUSIONS

The following conclusions can be drawn from the results presented in this thesis.

**TANK MODEL**

It was found that discretization of the tank model developed by Jekel [1] into two finite lengths adequately converged the outlet brine temperature. Discretization is not required for flow rates greater than 40 GPM. Discretization at flow rates less than or equal to 40 GPM converged the model's tank performance to the Calmac experimental performance data [2].

**FULL LOAD STRATEGY**

By increasing both the flow rate and the set temperature of the water into the cooling coil, the discharge time for a fixed load can be extended.

It was found that the size of the ice-storage tank is not only dependent on the total energy depleted from the tank during discharge, but is also highly dependent on the load profile. For load profiles with loads that increase near the end of the day, the required ice-storage tank size increases substantially. For profiles with loads that decrease near the end of the day, the tank size decreased a small amount.

The Calmac diversity factor [3] is a means to quantify the shape of a load into a single number. The diversity factor is not an accurate measure of load profile.

**PARTIAL LOAD STRATEGY**

Chiller operation strategy was examined. For maximum ice-storage usage, decreasing the chilled water set temperature during discharge was the optimum chiller operation strategy.

For a constant chiller load, increasing the water flow rate through the cooling coil raised the chilled water set temperature, thus increasing the percentage of the load met by ice-storage, increasing the chiller COP, and decreasing the chiller electric power requirement.

The effect of load profile on chiller capacity and ice-storage tank size was examined. For the same total load, there exists a constant minimum chiller capacity independent of load profile. The ice-storage tank size, however, changes. As the chiller capacity increases, the ice-storage tank size decreases, the tank size becomes increasingly independent of the load profile, and the required charging time decreases.

Three representative building loads were examined. As the peak load increased, the relation between chiller size and ice-storage tank became increasingly nonlinear and the ice-storage tank size increased for a fixed chiller capacity.

**SYSTEM COMPARISON**

In comparing the required chiller capacities and ice-storage tank sizes for a conventional air-conditioning system, a full storage system, and a partial storage system, the partial storage system minimized chiller size, had a smaller ice-storage tank size than the full storage system, and the load profile had less impact on the ice-storage tank size. The partial storage system ice storage tank size was mainly affected by the amount of energy depletion from the tank. In design of an ice-storage air-conditioning system, the system should be designed for a partial load strategy for the design day load.

## 6.2 RECOMMENDATIONS FOR FUTURE WORK

The finite length ice-storage tank model should be modified to evaluate the counterflow situation, which is the real flow in the ice-storage tank.

In analyzing the load profiles, the load profile was assumed to consist entirely of sensible cooling. Dehumidification and its effect, if any, on chiller capacity and ice-storage tank size should be studied. The influence of coil outlet temperature on sizing needs to be evaluated.

In the simulations of the ice-storage air-conditioning systems, water had to be used as the working fluid to properly use the chiller and cooling coil models. Models of these components for use with ethylene glycol mixtures would better predict the performance of the systems.

Simulations of the system should be done over an entire cooling season to determine the chiller capacity and ice-storage tank size combinations that will minimize cost for different rate structures.

# REFERENCES 6

1. Jekel, Todd, "Modeling of Ice-Storage Systems," M.S. Thesis, University of Wisconsin - Madison, December 1991.

2. *Levload Ice Bank Performance Manual*, Product Literature, Calmac Manufacturing Corporation, Englewood, New Jersey, April 1987.

3. *Levload Ice Bank Performance Manual*, Calmac Manufacturing Corporation.

# APPENDIX A

## FINITE LENGTH TANK MODEL

```
c***************************************************************
c*                                                             *
c*            Finite Length Ice-Storage Tank Model             *
c*                                                             *
c*                         Front End                           *
c*                                                             *
c*                      by  Colin Carey                        *
c*                                                             *
c***************************************************************
      program ft
      implicit none
      real*8 T(10),T_ti(10),Capacity,Cap_nom,mw(10),NTUM
      real*8 mwtot,Dw(10),hw(10),hwi(10),Q(10),Ntu(10),qtot
      integer N,i,j,k
      open (unit=20,file='discout.dat',status='new')
      Capacity=0.0
      mwtot=0.0
      qtot=0.0
c     ***T(1)=inlet brine temperature***
      T(1)=60.0
      NTUM=0.0
      Cap_nom=13527.*(144.0+1.01*(T(1)-32.0))/12000.0
c     ***N=number of finite lengths***
      N=6
      do 5 j=1,N
      T_ti(j)=32.0
      Dw(j)=.0521
      hw(j)=0.0
      hwi(j)=0.0
      mw(j)=0.0
```

```
    5   continue
        do while (Capacity .le..9999*Cap_nom)
        do 10 i=1,N
        call disc(T(i),T(i+1),N,Capacity,T_ti(i),mw(i),Dw(i),hw(i),
     &            hwi(i),Cap_nom,Q(i),Ntu(i))
        mwtot=mwtot+mw(i)
        qtot=qtot+Q(i)/(12000.*60)
c       write(20,100)Capacity/Cap_nom,Dw(i)
        if (Ntu(i) .GT. NTUM) then
        NTUM=Ntu(i)
        ENDIF
   10   continue
        mwtot=0.0
        write(20,100)Capacity/Cap_nom,T(N+1),NTUM
  100   Format (5x,F8.6,5x,F6.3,5x,F8.5)
        end do
        stop
        end

      subroutine disc(T_bi,T_bo,N,Capacity,T_ti,m_w,D_o_temp,
     &  h_w,h_wi,Cap_nom,Q_net,Ntu)
c*****************************************************************
c*                                                               *
c*                    Ice Storage Tank Model                 *
c*                    ======================                     *
c*                    CALMAC Ice-on-Coil Tank                    *
c*                                                               *
c*                        by  Todd Jekel                         *
c*****************************************************************

c----Variable Declaration--------------------------------------

      implicit none

      real*8 A_crit,A_o,A_ice,Acs_tube,alpha_b,AR
      real*8 B_w,Bw,Bw_min
      real*8 Cap_nom,Capacity,CF,Cp_b,Cp_w,Cv_b,Cv_i,Cv_w,check
      real*8 D_crit,D_i_tube,D_max,D_o,D_o_temp,D_o_tube,ddm,
     &    dD_o_tempdO,dD_odO,dm_w,dO,dT,dT_lm,dTbdO_w,dTdO,
     &    dU_w,dTT,dTlm
      real*8 f,ff,Flow,fract
```

```
      real*8 g,Gr_w
      real*8 h_b,h_if,h_w,h_wi,h_w1
      real*8 k_b,k_i,k_tube,k_w,kv_b,kv_w,kvb,kvw
      real*8 L_tube,Lat_cap
      real*8 m_w,m_w_crit,m_w_tot,mf_tot,mf_tube,mf_tank
      real*8 Ntu,Nu_b,Nu_w,Nu_b1,Nu_b2
      real*8 Old_dm,Old_T,one
      real*8 Per_store,pi,Pr_b,Pr_w,Prw
      real*8 Q_des,Q_gain,Q_ice,Q_net,Q_tot,Q_tube
      real*8 Re_b,rho_b,rho_i,rho_w,Ra_w,RR,RePrDL
      real*8 stp_hr
      real*8 T_amb,T_bbar,T_bi,T_bo,t_crit,T_des,T_film,T_frz,
     &      T_wi,T_wo,T_min,T_t,T_ti,T_tank,T_temp,Tb,Tb_w,time,
     &      time_end,Thick,tol,Ts_avg,TT,T_b_diff
      real*8 U_b,U_t,U_tot,U_tube,U_w,UA,UA_tank,U_tott
      real*8 v,x
      real*8 zero,zz
      integer N_tube,mode,step,N

      common /H2O_PROP/Bw,k_w,kvw,Prw
      common /TANK_GEOM/D_o_tube,L_tube

      parameter (pi = 3.14159)
      parameter (g = 32.2)          !c [ft/s^2]
      parameter (one = 1.)
      parameter (zero = 0.)
      parameter (Bw_min = 5.e-6)

c----Statement functions for properties--------------------------

      kv_b(T_bbar) = 0.1416 - 0.005158*(T_bbar-32.)/1.8 + 0.0001097*
     &            ((T_bbar-32.)/1.8)**2.
      B_w(T_film) = abs((-0.279 + 0.00854*(T_film+1.) - 0.0000411*
     &           (T_film+1.)**2.)/1000.)
      kv_w(T_film) = (12.24 - 0.2047*T_film + 0.001223*T_film**
     &            2.)/100.  !c [ft^2/hr]
      Pr_w(T_film) = 25.17 - 0.4577*T_film + 0.002843*T_film**2.

c----CALMAC tank------------------------------------------------

c--------Geometry data-----------------------------------------
```

```
          D_i_tube=.0425
          D_o_tube=.0521
          t_crit=.04042
          L_tube=250.0/N
          N_tube=56
          UA_tank=0.00
          dO=5.0/60.0
c         ***Flow=flow rate of brine through tank (GPM)***
          Flow=20.0
      D_i_tube = D_i_tube/1.      !c [ft]
      D_o_tube = D_o_tube/1.      !c [ft]
      t_crit = t_crit/1.      !c [ft]
      D_crit = D_o_tube + 2.*t_crit
      A_crit = pi*D_crit*L_tube
      D_max = D_crit*sqrt(2.)

c--------Thermal data------------------------------------------

      T_amb = 70.                 !c [F]
      k_tube = 0.30               !c [Btu/hr-ft-F]
      m_w_tot = 13527./N                  !c [lbm]

c----Brine Properties------------------------------------------

      alpha_b = 0.00489
      Cp_b = 0.9                  !c [Btu/lbm-F]
      k_b = 0.303                 !c [Btu/hr-ft-F]
      rho_b = 64.93               !c [lbm/ft^3]

c----Water/Ice Properties--------------------------------------

      Cp_w = 1.01                 !c [Btu/lbm-F]
      Cv_w = 1.01                 !c [Btu/lbm-F]
      rho_w = 62.4                !c [lbm/ft^3]
      T_frz = 32.         !c [F]

      Cv_i = 0.487                !c [Btu/lbm-F]
      k_i = 1.09          !c [Btu/hr-ft-F]
      rho_i = 62.4                !c [lbm/ft^3]
      h_if = 144.0                !c [Btu/lbm]
```

```
c----Determine relavent constants--------------------------------

      Acs_tube = pi*D_i_tube**2./4.
      U_tube = log(D_o_tube/D_i_tube)/k_tube
      Lat_cap = m_w_tot*h_if
      m_w_crit=rho_w*pi*(D_crit**2-D_o_tube**2)*N_tube*L_tube/4.
      mf_tot = 36455.5*Flow/70.             !c [lbm/hr] about 70 GPM

c-------------------------------------------------------------
c----Discharging Analysis---------------------------------------
c-------------------------------------------------------------

      dTT = 0.
      mf_tank = mf_tot
      mf_tube = mf_tank/N_tube
      v = mf_tube/(rho_b*Acs_tube)

c----Estimate brine outlet temperature and tube/ice surface temp-

      T_bo = T_bi + 0.75*(T_ti-T_bi)
      Ts_avg = T_bi - (T_bi - (T_bo+T_bi)/2.)/2.

c----Set initial conditions--------------------------------------

      T_tank = T_ti
      if (T_tank .GE. T_bi) return
      D_o = D_o_temp
      Old_T = T_ti
      time = 0.0
      time_end = time_end+dO
      tol = 1.E-5
      dm_w = 0.

c----Determine heat transfer coeff. inside tube-----------------

  100   T_bbar = (T_bi+T_bo)/2.
        kvb = kv_b(T_bbar)
        Re_b = v*D_i_tube/kvb
        Pr_b = kvb/alpha_b
        RePrDL = (D_i_tube/L_tube)*Re_b*Pr_b
```

```
      Nu_b1 = 3.66+(0.0534*(RePrDL)**(1.15))/(
     &        1.+0.0335*(RePrDL)**(0.82))
      Nu_b2 = 0.023*Re_b**0.8*Pr_b**0.4
      if (Re_b .lt. 700.) then
        Nu_b = Nu_b1
      else if (Re_b .gt. 1300.) then
        Nu_b = Nu_b2
      else
        Nu_b = (Nu_b1*(1300.-Re_b)+Nu_b2*(Re_b-700.))/(1300.-700.)
      end if
      h_b = Nu_b*k_b/D_i_tube

c----Determine the properties of the water----------------------

      T_film = (Ts_avg + T_tank)/2.
      kvw = kv_w(T_film)
      Prw = Pr_w(T_film)
      k_w = kvw*rho_w*Cp_w/(Prw)
      Bw = dmax1(Bw_min,B_w(T_film))

c----Determine the effective thermal conductivity---------------

      if (m_w .lt. m_w_crit) then

c----Determine the total conductance----------------------------

        A_o = pi*D_o_temp*L_tube
        U_b = D_o_temp/(h_b*D_i_tube)
        U_t = U_tube*D_o_temp/2.
        U_w = D_o_temp*log(D_o_temp/D_o_tube)/(2.*k_w)
        U_tot = 1./(U_b+U_t+U_w)
        UA = U_tot*A_o
        Ntu = UA/(mf_tube*Cp_b)

        T_bo = T_frz + (T_bi-T_frz)*exp(-Ntu)
        dT_lm = ((T_frz-T_bo)-(T_frz-T_bi))/log((T_frz-T_bo)/
     &        (T_frz-T_bi))
        Q_tube = abs(UA*dT_lm)
        Q_tot = N_tube*Q_tube
        Q_gain = UA_tank*(T_amb - T_frz)
        Q_net = Q_tot+Q_gain
```

```
        if (m_w .ne. 0.) then
       T_wi = T_frz + Q_tube/(1./(log(D_o_temp/D_o_tube)
     &          /(2.*pi*k_w*L_tube)))
         T_wo = T_frz
         T_tank = T_wo - (T_wo-T_wi)*(1./(2.*log(D_o_temp/
     &          D_o_tube)) - D_o_tube**2/(D_o_temp**2-
     &               D_o_tube**2))
         dTbdO_w = abs(T_ti - T_tank)/dO
         Ts_avg = T_wi
        else
         Ts_avg = T_frz
         dTbdO_w = 0.
        end if
        dU_w = m_w*Cp_w*dTbdO_w
        Q_ice = Q_net-dU_w
        dm_w = (Q_net-dU_w)*dO/(h_if+Cp_w*(T_tank-T_frz))
        D_o_temp = dsqrt(D_o**2+4.*dm_w/(rho_w*pi*L_tube*
     &                N_tube))
        if (m_w .gt. (m_w_crit-500.)) then
         h_wi = (Q_net-dU_w)/(N_tube*A_o*(T_tank-T_frz))
         T_t = T_tank
         dTlm = ((T_t-T_bo)-(T_t-T_bi))/log(abs((T_t-T_bo)/
     &           (T_t-T_bi)))
         U_tott = Q_tot/abs(pi*D_o_tube*L_tube*N_tube*dTlm)
         h_w = 1./(1./U_tott-D_o_tube/(h_b*D_i_tube)-D_o_tube*
     &              U_tube/2.)
        end if

       else if (m_w .gt. m_w_crit) then

        if (m_w .gt. m_w_tot) m_w = m_w_tot

c----Determine the total conductance----------------------------

        A_o = pi*D_o_tube*L_tube
        U_tot = 1./(D_o_tube/(h_b*D_i_tube)+D_o_tube*U_tube/2.+
     &         1./h_w)

        UA = U_tot*A_o
        Ntu = UA/(mf_tube*Cp_b)
```

```
      T_bo = T_tank + (T_bi-T_tank)*exp(-Ntu)
      dT_lm = ((T_tank-T_bo)-(T_tank-T_bi))/log((T_tank-T_bo)/
&        (T_tank-T_bi))
      Q_tube = abs(UA*dT_lm)
      Q_tot = N_tube*Q_tube
      Q_gain = UA_tank*(T_amb - T_tank)
      Q_net = Q_tot+Q_gain
      Ts_avg = T_tank+(Q_net/N_tube)/(h_w*A_o)

      if (m_w .ge. m_w_tot) then
       m_w = m_w_tot
       dm_w = 0.
       dTdO = Q_net/(m_w*Cv_w)
       T_tank = T_ti + dO*dTdO
      else
       check = D_o_temp*(pi-4.*acos(D_crit/D_o_temp))*L_tube
       A_ice = dmax1(zero,D_o_temp*(pi-4.*acos(D_crit/D_o_temp)
&             )*L_tube)
       Q_ice = h_wi*N_tube*A_ice*(T_tank-T_frz)
       dm_w = Q_ice*dO/(h_if+Cp_w*(T_tank-T_frz))
       dTdO = (Q_net-Q_ice)/(m_w*Cv_w)
       T_tank = T_ti + dO*dTdO
     call dia(D_crit,D_o_temp,D_o_tube,dm_w,L_tube,m_w,
&           N_tube,rho_w)
      end if
     end if

     ddm = abs(dm_w-Old_dm)
     if (m_w .lt. m_w_crit) then
      TT = 0.
     else
      TT = abs(T_tank-Old_T)
     end if
     if ((ddm .gt. tol) .or. (TT .gt. .01)) then
      Old_dm = dm_w
      Old_T = T_tank
      goto 100
     else
110   m_w = m_w+dm_w
      D_o = D_o_temp
      Thick = (D_o - D_o_tube)/2
```

```
          T_temp = T_tank
        end if
        Capacity = Capacity+Q_net*dO/12000.
        dTT = dTT+(T_bo-T_bi)*dO
        T_ti=T_tank
        write(9,111)time,T_bo,Q_net/12000.,dT_lm,
     &             Capacity/Cap_nom,UA
      ZZ = ((T_frz-T_bo)-(T_frz-T_bi))/log((T_frz-T_bo)/
     &           (T_frz-T_bi))
      write(12,99)Capacity/Cap_nom,Ntu*dT_lm/ZZ,(T_bi-T_bo)/(T_bi-32.)
      if (Q_net/12000. .lt. .05) goto 200
        time = time + dO
       dT = Ts_avg-T_tank
 2000 continue
   99 format(1x,3f10.5)
  111 format(1x,f7.4,5f15.5)
  112 format(1x,f7.4,3e15.5,f15.5)
  200 print*,h_w
      write(*,1000)T_ti,T_bo,m_w,Capacity/Cap_nom
 1000 format(1x,5F15.4)
      continue
c     STOP
      end

      subroutine dia(D_crit,D_o_temp,D_o_tube,dm_i,L_tube,m_i,
     &              N_tube,rho_i)

      implicit none

      real*8 D_crit,D_o_temp,D_o_tube,dm_i,L_tube,m_i,rho_i
      real*8 dMdD,f,M,mass,pi,tol
      integer i,N,N_tube

      parameter (pi = 3.14159)

      mass = (m_i+dm_i)/N_tube
      tol = 1.E-8
      N = 50
      do 10, i = 1,N
        M = rho_i*(D_crit*L_tube*(D_o_temp**2-
     &      D_crit**2)**0.5 + (pi/4.-acos(D_crit/D_o_temp))*
```

```
     &     L_tube*D_o_temp**2 - pi*D_o_tube**2*L_tube/4.)
        dMdD = rho_i*(pi*D_o_temp*L_tube/2. -
     &     2*D_o_temp*L_tube*acos(D_crit/D_o_temp))
        f = D_o_temp - (M-mass)/dMdD
        if ((f-D_o_temp) .lt. tol) then
C          write(20,*)m_i,dm_i
           return
          else
           D_o_temp = f
          end if
   10 continue
      write(7,*)'Method failed after 50 iterations!'
      stop
      end
```

## Discin.dat

```
.0425,.0521,.04042
250.0, 56
0.000
.083333,40.0
```

# APPENDIX B

## REGRESSION OF EFFECTIVENESS CURVES

**Regression performed by Minoru Kawashima**

$\varepsilon$ = C0+C1+C2+C3+C4+C5+C6+C7+C8+C9+C10+C11+C12+C13+C14+C1

$$B = \frac{\text{Capacity Discharged}}{\text{Maximum Capacity}}$$

q = Flow Rate through Ice-Storage Tank (GPM)

where:

### ***Dicharging:***

If $\frac{\text{Capacity Discharged}}{\text{Maximum Capacity}} < .66$ then:

C0 = 0.84119769
C1 = 0.200276759 *B
C2 = 1.636547199 $*B^2$
C3 = - 5.204433828 $*B^3$
C4 = 4.196217689 $*B^4$
C5 = 0.015118414 *q
C6 = - 0.000390064 $*q^2$
C7 = $3.64763 \times 10^{-6}$ $*q^3$
C8 = $- 1.24338 \times 10^{-8}$ $*q^4$

C9 = - 0.053871746 $*B*q$
C10 = 0.064822502 $*B^2*q$
C11 = 0.000354565 $*B*q^2$
C12 = - 0.034354947 $*B^3*q$
C13 = - 0.000142311 $*B^2*q^2$
C14 = $-9.15865\text{x}10^{-7}$ $*B*q^3$
C15 = 0.0

If $\dfrac{\text{Capacity Discharged}}{\text{Maximum Capacity}} \geq .66$ then:

C0 = 25.62156701
C1 = - 110.463303 $*B$
C2 = 176.6331532 $*B^2$
C3 = - 114.555632 $*B^3$
C4 = 22.86186786 $*B^4$
C5 = - 0.01026212 $*q$
C6 = - 0.0004725 $*q^2$
C7 = $5.03616\text{x}10^{-7}$ $*q^3$
C8 = $-2.1181\text{x}10^{-9}$ $*q^4$
C9 = 0.105010295 $*B*q$
C10 = - 0.27724386 $*B^2*q$
C11 = 0.001260003 $*B*q^2$
C12 = 0.179974285 $*B^3*q$
C13 = - 0.00078403 $*B^2*q^2$
C14 = $-1.8073\text{x}10^{-7}$ $*B*q^3$
C15 = 0.0

***Charging:***

If $\frac{\text{Capacity Discharged}}{\text{Maximum Capacity}} \leq .755$ then:

C0 = 1.077255269

C1 = - 0.079156996 *B

C2 = - 0.0046742 *q

C3 = 0.0

C4 = 0.0

C5 = 0.0

C6 = 0.0

C7 = 0.0

C8 = 0.0

C9 = 0.0

C10 = 0.0

C11 = 0.0

C12 = 0.0

C13 = 0.0

C14 = 0.0

C15 = 0.0

If $\frac{\text{Capacity Discharged}}{\text{Maximum Capacity}} > .755$ then:

C0 = 1.511144226

C1 = 0.22757868 $*\log_{10}(-B+1.0)$

C2 = - 0.009864783 *q

C3 = $3.83656 \times 10^{-5}$ $*q^2$

C4 = - 0.281804303 *B

C5 = 0.0

C6 = 0.0

C7 = 0.0

C8 = 0.0

C9 = 0.0

C10 = 0.0

C11 = 0.0

C12 = 0.0

C13 = 0.0

C14 = 0.0

C15 = 0.0

# APPENDIX C

## FULL STORAGE STRATEGY

### TRNSYS DECK

| Deck | Corresponding Diversity | Corresponding Building Data |
|---|---|---|
| SYS4.DCK | .61538 | fbld3.dat |
| SYS5.DCK | .66667 | fbld2.dat |
| SYS6.DCK | .83333 | fbld1.dat |

```
ASSIGN sys4.LST 6
* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
*                   FULL LOAD STRATEGY                   *
*             ICE-STORAGE SYSTEM WITH COOLING COIL       *
*                    NOVEMBER 1992                       *
* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
ASSIGN C1.OUT 7
ASSIGN sys4.PLT 8
ASSIGN sys4.OUT 9
assign fbld3.dat 10

EQUATIONS 35
*|*        University of Wisconsin - Solar Energy Laboratory *|*
*******Ice-Storage Air-Conditioning System******* *|*
***************Full Load Strategy*************** *|*
*|*                Date:|February 15,1993
*|*         Name of System:|C2-ICE-F1
*|*       Input Prepared by:|COLINATOR
*|*
*|*            *****Ice-Storage Tank Parameters*****
*|*
```

```
TANKSZ=  2.436842E+00
*|Tank Size                |Ton-hrs/190 Ton-hrs|       Ton-hrs|0|190.0|.1|1000.0|1 *|*
*|*                   *****Cooling Coil*****
*|*
MODE=  1.000000E+00
*|<Cooling Coil Selection    |Coil.dat|1|2|1
UNITS=  2.000000E+00
*|<Units|Coil.Dat|0|3|1
Nrows=  5.000000E+00
*|<Nrows|Coil.Dat|0|4|1
Ntubes=  1.200000E+01
*|<Ntubes|Coil.Dat|0|5|1
Lduct=  2.000000E+00
*|<Lduct|Coil.Dat|0|6|1
Wduct=  2.000000E+00
*|<Wduct|Coil.Dat|0|7|1
do=  8.330000E-02
*|<do|Coil.Dat|0|8|1
di=  7.917000E-02
*|<di|Coil.Dat|0|9|1
ktube=  2.270000E+02
*|<ktube|Coil.Dat|0|10|1
ft=  8.330000E-04
*|<ft|Coil.Dat|0|11|1
fs=  8.330000E-03
*|<fs|Coil.Dat|0|12|1
Nfin=  2.400000E+02
*|<Nfin|Coil.Dat|0|13|1
kfin=  2.270000E+02
*|<kfin|Coil.Dat|0|14|1
finm=  1.000000E+00
*|<finm|Coil.Dat|0|15|1
dcr=  1.670000E-01
*|<dcr|Coil.Dat|0|16|1
dclr=  1.670000E-01
*|<dclr|Coil.Dat|0|17|1
*|*
*|*                 *****System Parameters*****
*|*
END=  1.000000E+01
*|Simulation Duration       |hrs|      hrs|0|1.0|.125|1000.0|1
```

```
STEP=  1.250000E-01
*|Time Increment            |hrs|     hrs|0|1.0|.005|500.000|1 FLOW=
4.000000E+04
*|Water Flow Rate           |lbm/hr|       lbm/hr|0|1|30000|60000|1 CPA=.2399
TAOUT=  6.110000E+01
*|Air Outlet Temperature    |F|       F|0|1|40|70.00|1
ma=[1,3]/(CPA*max(1,([1,1]-TAOUT)))
TSET=62.0
Ca=CPA*ma
CPW=1.00
Cw=CPW*FLOW
Cmin=MIN(Ca,Cw)
E1=Ca*([1,1]-[3,1])
E2=MAX(0.1,(Cmin*([1,1]-[6,1])))
E3=E1/E2
EHX=MIN(1,E3)
AIRLOAD=ma*CPA*([1,1]-[3,1])
WATERLOAD=[3,5]*CPW*([3,4]-[6,1])
ICELOAD=[4,2]*CPW*([4,1]-[5,3])

SIMULATION 0.0  END  STEP
*11 HOURS OF DISCHARGE AT 5 MINUTE INCREMENTS
WIDTH 120
TOLERANCES .0002 .0002
LIMITS 200 10

UNIT 1 TYPE 9 DATA READER (FREE)
PARAMETERS 13
3 1.0 1 1 0 2 1 0 3 1 0 10 -1

UNIT 13 TYPE 13 CONTROLLER
PARAMETERS 2
0.2399 1.0
INPUTS 5
*LOAD EFF TAIN MAIN MWIN
 1,3  EHX 1,1  ma  FLOW
 100000 0.7 75.0 30000 FLOW
*TRACE 0 1

UNIT 2 TYPE 33 PSYCHROMETRIC CHART
PARAMETERS 2
2 2
```

INPUTS 2
*Tai,db RH
1,1 1,2
75.0 0.0

UNIT 3 TYPE 52 COOLING COIL
PARAMETERS 16
Mode Units Nrows Ntubes Lduct Wduct do di ktube ft fs Nfin kfin finm dcr dclr INPUTS 5
*Tai,db wai ma Twi mw
1,1 2,1 ma 6,1 FLOW
75.0 .05 30000.0 TSET FLOW
*trace 0 1

UNIT 11 TYPE 24 QUANTITY INTEGRATOR
INPUTS 1
3,6
0.0
*trace 0 1

UNIT 4 TYPE 11 FLOW DIVERTER
PARAMETERS 2
4 150
INPUTS 4
*Tbi Mb Tbout_tank Tset
3,4 3,5 5,3 13,1
90.0 FLOW 33.0 50.0
*trace 10.0 10.25
*TRACE 0 1

UNIT 5 TYPE 75 ICE STORAGE
TANK PARAMETERS 5
32.0 64.93 STEP TANKSZ 153
INPUTS 3
*Tbin LOAD Mb
4,1 3,6 4,2
61.11 .5E5 8500.0
*trace 0.0 1.0

UNIT 6 TYPE 11 TEE-PIECE
PARAMETERS 1
1
INPUTS 4

```
*Tbint mbt Tbdiv mbdiv
5,3 5,4 4,3 4,4
32.0 8500.0 60.0 51500.0
*trace 14.75 15.0
*TRACE 0 1

UNIT 8 TYPE 25 PRINTER
PARAMETERS 4
STEP 0 30.0 7
INPUTS 2
1,3 3,6
*11,1 1,3 3,6 5,1 5,6
*3,1 ICELOAD 3,6 AIRLOAD WATERLOAD *TLOAD Tao Qcoil EFFECT ACCUM
BLDD CLLD

unit 10 type 25 printer
parameters 4
STEP 0 30 9
inputs 5
3,5 4,2 4,4 5,4 6,2
m1 m1 m2 m3 m4 m5

unit 9 type 25 printer
parameters 4
STEP 0 30 8
inputs 5
5,6 4,1 4,3 5,3 6,1
t1 t2 t3 t4 t5

EQUATIONS 2
IN1=[1,3]/1000
IN2=[3,6]/1000

UNIT 65 TYPE 65 ONLINE
PARAMETERS 9
* # UPPER  # LOWER
2 3
*MIN & MAX UPPER,LOWER
0 400 20 68
* TIMESTEP FOR PRINT REFRESH
UNITS (3=HR) 1 1 3
INPUTS 20
```

```
IN1 IN2 3,4 5,3 6,1 0,0 0,0 0,0 0,0
0,0 0,0 0,0 0,0 0,0 0,0 0,0 0,0 0,0 0,0
0,0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0
0 0

END
```

## Modification of Type 11: Temperature-controlled Flow Diverter for use with Ice-Storage

```
C     MODES 4 AND 5:  TEMPERATURE CONTROLLED FLOW DIVERTER
400     NSTK=IFIX(PAR(2)+0.5)
      IF(INFO(7).GT.NSTK)THEN
        TSOURC=XIN(1)
        FLOW=XIN(2)
        TCOLD=XIN(3)
        TSET=XIN(4)
        OUT(1)=XIN(1)
        OUT(2)=GAM*FLOW
        OUT(3)=XIN(1)
        OUT(4)=FLOW-OUT(2)
        OUT(5)=GAM
        RETURN
      ENDIF
      TSOURC=XIN(1)
      FLOW=XIN(2)
      TCOLD=XIN(3)
      TSET=XIN(4)
      GAM = 1.0
         if (TSOURC .GE. TSET) THEN
                 if (Tset .GT. TCOLD) GAM=(TSET-TSOURC)/(TCOLD-TSOURC)
                 if (TSET .LE. TCOLD) GAM=1.0
         else
                 gam=0.0
```

```
          endif
c        IF (TCOLD .LE. TSET) GAM = (TSET-TSOURC)/(TCOLD-TSOURC)
C  *** NOTE CHANGE
c        IF (TCOLD .GT. TSET .AND. MODE.EQ.4) GAM=1.0
c        IF (THOT.LT.TSOURC .AND. MODE.EQ.5) GAM = 0.0
C        MODE 4 AND 5 OUTPUTS
       OUT(1)=XIN(1)
       OUT(2)=GAM*FLOW
       OUT(3)=XIN(1)
       OUT(4)=FLOW-OUT(2)
       OUT(5)=GAM
      RETURN
```

## Type 13 Controller

```
          SUBROUTINE TYPE13(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
          DIMENSION XIN(5),OUT(1),T(1),DTDT(1),PAR(1),INFO(10)
          Real Tset,Load,EHX,Tain,ma,mw,Cpa,Cpw,Ca,Cw,Cmin
C

       NI=5
       NP=2
       ND=0
       INFO(6)=1
       CALL TYPECK(1,INFO,NI,NP,ND)
          LOAD=XIN(1)
          EHX=XIN(2)
          Tain=XIN(3)
          ma=XIN(4)
          mw=XIN(5)
C
          Cpa=PAR(1)
          Cpw=PAR(2)
C
          Ca=ma*Cpa
          Cw=mw*Cpw
          Cmin=min(Ca,Cw)
C
```

```
      Tset=Tain-LOAD/(max(2000.0,(Cmin*EHX)))
C
      OUT(1)=Tset
C
      RETURN
C
      END
```

## Type 75 Ice-Storage Tank
## (Discharge Only)

```
      SUBROUTINE TYPE75(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
C     ICE STORAGE TANK MODEL USING A REGRESSION OF
C                    EFFECTIVENESS CURVES
C
C     CALMAC 190 Ton Hour Tank
C
      DIMENSION XIN(3),OUT(6),T(1),DTDT(1),PAR(4),INFO(10)
      REAL C0,C1,C2,C3,C4,C5,C6,C7,C8,C9,C10,C11,C12,C13,C14,C15
      REAL densityb,Vb,MaxCap,Hif,Cvw,mwtot,B,PLOAD,LOAD,Mb
C
    IF(INFO(7).LT.0) THEN
    A=0.
    PLOAD=0.
    LOAD=0.
    NI=3
    NP=5
    ND=0
    INFO(6)=6
    CALL TYPECK(1,INFO,NI,NP,ND)
    RETURN
    ENDIF

      Tfp=PAR(1)
      Cpb=PAR(2)
      deltat=PAR(3)
      tnksize=PAR(4)
    NSTCK=PAR(5)
```

```
C
      IF(INFO(7).eq.0) THEN
        PLOAD=PLOAD+LOAD
      ENDIF

         Tbin=XIN(1)
         LOAD=deltat*XIN(2)
         Mb=XIN(3)
         A=PLOAD+LOAD
C        densityb=64.93
         densityb=62.4
C
         Vb=Mb*7.481/(densityb*60.0)
         mwtot=13527.
         Hif=144.0
         Cvw=1.01

      IF(INFO(7).LT.NSTCK) THEN
           MaxCap=tnksize*mwtot*(Hif+Cvw*(Tbin-Tfp))
           if(A .ge. MaxCap) then
             A=MaxCap
           ENDIF
           B=A/MaxCap
      ENDIF
C
C***Regression performed by Minoru Kawashima***
C
         C0=1.07725166843407537
         C1=1.75144166459073890*B
         C2=-44.3276572271372681*B**2
         C3=428.326569085308112*B**3
         C4=-2098.54851528811492*B**4
         C5=5877.82456866507374*B**5
         C6=-9806.03356404150013*B**6
         C7=9618.07618747049688*B**7
         C8=-5113.11667741195139*B**8
         C9=1135.2085285916837*B**9
         C10=-.00461366093990033812*Vb
         C11=.0000287634886505943191*Vb**2
         C12=-6.23979241136891786E-8*Vb**3
         C13=-.0194263060424152013*B*Vb
```

```
      C14=.0179765346269569237*(B**2)*Vb
      C15=.000013575187035953475l*B*(Vb**2)
C
      E=C0+C1+C2+C3+C4+C5+C6+C7+C8+C9+C10+C11+C12+C13+C14+C15
      if (E .ge. 1.0) E=1.0
      if (E .le. 0.0) E=0.0
      if (B .ge. 1.0) E=0.0
C
      Tbout=Tbin-E*(Tbin-Tfp)
C
      OUT(1)=E
      OUT(2)=Tbin
      OUT(3)=Tbout
      OUT(4)=Mb
      OUT(5)=LOAD/deltat
      OUT(6)=B
C
      RETURN
C
      END
```

## Cooling Coil Data

**(Coil.dat)** (ABC used in simulations)

**Number of Different Coils**

**Name Mode Units Nrows Ntubes Lduct Wduct do di ktube finthickness**

**finspacing Nfins kfin finmode dcr dclr**

2

Cooling Coil ABC,1,2,6,12,2,2,.0833,.07917,227,8.33E-4,

8.33E-3,240,227,1,.167,.167

Cooling Coil XYZ,1,2,5,12,2,2,.0833,.07917,227,8.33E-4,

8.33E-3,240,227,1,.167,.167

## Building Load Data with Diversity = .83333

Tain (ºF) , RH (%) , Building Load (Btu/hr)
75.0,0.0,215000.0
75.0,0.0,215000.0
75.0,0.0,140000.0
75.0,0.0,165000.0
75.0,0.0,177000.0
75.0,0.0,190000.0
75.0,0.0,200000.0
75.0,0.0,240000.0
75.0,0.0,240000.0
75.0,0.0,225000.0
75.0,0.0,200000.0
75.0,0.0,200000.0

## Building Load Data with Diversity = .66667

75.0,0.0,215000.0
75.0,0.0,200000.0
75.0,0.0,100000.0
75.0,0.0,127500.0
75.0,0.0,160000.0
75.0,0.0,180000.0
75.0,0.0,200000.0
75.0,0.0,300000.0
75.0,0.0,300000.0
75.0,0.0,225000.0

75.0,0.0,200000.0
75.0,0.0,200000.0

## Building Load Data with Diversity = .61538

75.0,0.0,215000.0
75.0,0.0,200000.0
75.0,0.0,92000.0
75.0,0.0,110000.0
75.0,0.0,130000.0
75.0,0.0,185000.0
75.0,0.0,200000.0
75.0,0.0,325000.0
75.0,0.0,325000.0
75.0,0.0,225000.0
75.0,0.0,200000.0
75.0,0.0,200000.0

# APPENDIX D

## COOLING COIL SIZING AND FLOW REDUCTION

**Fortran program for use in cooling coil sizing and flow reduction in a full storage strategy.**

```
C       ICE STORAGE PROGRAM
C
C               by COLIN CAREY
C
C       This program can be used to properly size the cooling coil in a full storage
C               strategy.
C
C       This program also allows the study of flow reduction in a full storage strategy
C

        REAL ICELD, MB, MBX, TBIN, TBOUT, TBM, MI,Cr,UA,NTU
        REAL MAXCAP,REALLD,X,B0,B1,B2,B3,B4,CPB,UIF,CVW
        REAL DLOAD,VB,RHOB,G,R,S,J,CPA,PAIR
        REAL TAOUT,TAIN,TBOUTF,CFM,Cair
        REAL Cbrine,Cmin,Cmax,BRINELD,EXC,P,PP,delT,absdlt,sig,rig,time
        REAL MWTOT,TNKSZ,FR,DT,TEND
        DOUBLE PRECISION C0,C1,C2,C3,C4,C5,C6,C7
        DOUBLE PRECISION C8,C9,C10,C11,C12,C13,C14,C15
        DOUBLE PRECISION B,VBX,E1,E2,E3,E,CAP,A
        INTEGER F,FF,FFF
        CPA=.2399
```

```
      CPB=.9
      CVW=1.01
      MWTOT=13527.0
      PAIR=.075165
      UIF=144.0
c     DT=15.0
      OPEN (UNIT=40, FILE='SIZE.DAT',STATUS='OLD')
      OPEN (UNIT=50, FILE='ICEDATA2.DAT',STATUS='OLD')
      OPEN (UNIT=60, FILE='ICEOUT',STATUS='NEW')
      WRITE(60,*)
      DLOAD=0
      READ (40,*) UA,TNKSZ,FR,DT
      READ (50,*,END=99) REALLD, TAIN
      REALLD=REALLD*DT/60.0
      TIME=0.0
C  Tend = time at which simulation should be terminated
      TEND=10.25
10    TIME=TIME+(DT/60.0)
      REALLD=REALLD+(162.601*(DT**2)/60)
      VB=76.81
      FFF=0
5     CONTINUE
      Cbrine=VB*CPB*64.93/7.481
      TAOUT=61.11
      CFM=REALLD/(PAIR*DT*(TAIN-TAOUT))
c     CFM=4000
c     TAOUT=TAIN-(REALLD/(PAIR*CFM*DT))
      Cair=CFM*PAIR*CPA
      IF (Cair .LE. Cbrine) THEN
      Cmin=Cair
      Cmax=Cbrine
      ELSE
      Cmin=Cbrine
      Cmax=Cair
      ENDIF
      Cr=Cmin/Cmax
C     units of UA=Btu/min-F
      NTU=UA/Cmin
      EXC=(1-exp(-NTU*(1-Cr)))/(1-Cr*exp(-NTU*(1-Cr)))
C     EXC=(Cair*(TAIN-TAOUT))/(Cmin*(TAIN-TBOUTF))
      TBM=TAIN-(Cair*(TAIN-TAOUT)/(Cmin*EXC))
```

```
      TBIN=TBM+(EXC*Cmin*(TAIN-TBM)/Cbrine)
      MAXCAP=TNKSZ*MWTOT*(UIF+(CVW*(TBIN-32.0)))
C     MAXCAP=1.944e6+(13500.7*(TBIN-32))
      VBX=5.0
      R=-1
      F=0
      DLOAD=DLOAD+REALLD
      IF (DLOAD .LE. MAXCAP) THEN
      A=DLOAD/MAXCAP
      ELSE
c     WRITE (60,*) 'OVERLOAD'
      GOTO 88
      ENDIF
20    CONTINUE
      F=F+1
      IF (F .GT. 5000) THEN
      FF=666
      GOTO 50
      ENDIF
C     IF (VBX .GE. VB) THEN
C     WRITE (60,*) 'PARTIAL LOAD MET'
C     VBX=VB
C     ENDIF
C
C     Calculation of Effectiveness
C
100   CONTINUE
      C0=1.0474847639D0
      C1=-2.5507795485D0
      C2=38.497719302D0
      C3=-293.18368689D0
      C4=1323.0858682D0
      C5=-3657.0553227D0
      C6=6218.4776547D0
      C7=-6330.9777019D0
      C8=3532.9306972D0
      C9=-830.06879468D0
      C10=0.0008128231D0
      C11=-.0000388508D0
      C12=.0000002083D0
      C13=-.0243846202D0
```

```
      C14=.0186705717D0
      C15=.0000425368D0
C
      B=VBX
      E1=C0+C1*A+C2*A**2+C3*A**3+C4*A**4+C5*A**5+C6*A**6+C7*A**
7
      E2=C8*A**8+C9*A**9+C10*B+C11*B**2+C12*B**3+C13*A*B
      E3=C14*A**2*B+C15*A*B**2
      E=E1+E2+E3
C
      IF (E .GT. 1.0) THEN
      E=1.0
      ENDIF
C
      RHOB=64.93
      MBX=VBX*RHOB*DT/7.481
      ICELD=E*MBX*CPB*(TBIN-32.0)
C     IF (VBX .EQ. VB) THEN
C     GOTO 50
C     ENDIF
      J=ICELD-REALLD
      G=ABS(J)
      IF (G .LE. 25.0) THEN
      GOTO 50
      ENDIF
      S=SIGN(1.0,J)
      IF (S .EQ. R) THEN
      VBX=VBX+.02
      R=S
      P=VBX
      GOTO 20
      ELSE
      VBX=.5*(P+VBX)
      GOTO 20
      ENDIF
C
50    CONTINUE
      TBOUT=TBIN-(ICELD/(MBX*CPB))
C
70    CONTINUE
      MB=VB*RHOB*DT/7.481
```

```
      TBM=TBIN-(REALLD/(MB*CPB))
      X=(TBM-TBIN)/(TBOUT-TBIN)
      IF (FFF .EQ. 10) THEN
      GOTO 77
      ENDIF
C
C     FR=desired flow fraction through the ice
C     if FR=.999 then the flow will be reduced so that all flow will go through the tank
C     if FR=.001 then the flow through the cooling coil will be held constant
C
      IF (X .LE. FR) THEN
      VB=X*VB
      DLOAD=DLOAD-REALLD
      FFF=FFF+1
      GOTO 5
      ENDIF
      IF (X .GT. 1.0) THEN
      VB=VB/X
      DLOAD=DLOAD-REALLD
      GOTO 5
      ENDIF
77    CONTINUE
C     WRITE (60,*) E,VBX,REALLD,ICELD,TBIN,TBOUT,TBM,X,A
C     WRITE (60,*) EXC,TAIN,TAOUT,UA,NTU,VB
88    WRITE (60,500)A,E,X,VB,TBIN,EXC,NTU,TBM,REALLD/.125,TIME
C     WRITE (60,*) Cair,Cbrine,Cmin,Cmax
C     WRITE (60,*)
C     WRITE (60,*)
      IF (TIME .GE. TEND) GOTO 99
      GOTO 10
99    CONTINUE
500   FORMAT(2x,F4.3,2x,F5.3,2x,F5.3,2x,F6.2,2x,F5.2,2x,F4.3,2x,F5.3,
    & 2x,F5.2,2x,F7.0,2x,F5.2)
      CLOSE (50)
      CLOSE (60)
      STOP
      END
```

## Size.dat (sample)

UA value, Tank Size (Ton-hrs/190 Ton-hrs (Calmac 1190)), Flow Fraction Desired (.0001(no flow reduction), Time increment (minutes)

219.0, .915, .0001, 7.5

## Icedata.dat (sample)

Load (Btu), Tain (°F)
60950,75
60950,75
60950,75
60950,75
40825,75
40825,75
40825,75
40825,75
47150,75
47150,75
47150,75
47150,75
54050,75
54050,75
54050,75
54050,75
60950,75
60950,75
60950,75
60950,75
60950,75
60950,75
60950,75
60950,75
67850,75
67850,75
67850,75

67850,75
67850,75
67850,75
67850,75
67850,75
60950,75
60950,75
60950,75
60950,75
54050,75
54050,75
54050,75
54050,75
54050,75

# APPENDIX E

## PARTIAL STORAGE STRATEGY

### TRNSYS DECK

| Deck | Corresponding Diversity | Corresponding Building Data |
|---|---|---|
| SYS1.DCK | .61538 | pbld3.dat |
| SYS2.DCK | .66667 | pbld2.dat |
| SYS3.DCK | .83333 | pbld1.dat |

```
ASSIGN sys1.lst 6
* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *

*                   PARTIAL LOAD STRATEGY                   *

*                   ICE-STORAGE SYSTEM                      *

*                   FEBRUARY 1993                           *

* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *

*ASSIGN run.dat  5
ASSIGN sys1.plt 7
ASSIGN sys1.out 8
ASSIGN icec.out 9
*CHIILER PERFORMANCE DATA
X,Y,Z (18data) ASSIGN chil.dat 15
```

```
*SYSTEM DRIVING DATA
ASSIGN pbld3.dat 20

EQUATIONS 33
*|*     ********University of Wisconsin - Solar Energy Laboratory********
*|*             *****Ice-Storage Air-Conditioning System*****
*|*                  *****Partial Load Strategy*****
*|*
*|*                  Date:|March 17, 1993
*|*          Name of system:|C2-ICE-P1
*|*        Input prepared by:|COLINATION, INC.
*|*
*|*                *****Ice-Storage Tank Parameters*****
*|*
TNKSZ=  5.684211E-01
*|Tank Size          |Ton-hrs/ton-hrs|       Ton-hrs|0|190.0|0|1000.0|1 *|*
*|*                    *****Chiller Parameters*****
*|*
CHLCAP=  8.330000E+04
*|Capacity           |Btu/hr|     Btu/hr|0|1.0|1|1000000.0|1
STORE=(TNKSZ*2166485.0)+[11,2]
TSET=48.68
*|*
*|*                      *****Cooling Coil*****
*|*
MODE=  1.000000E+00
*|<Cooling Coil Selection |Coil.dat|1|2|1
UNITS=  2.000000E+00
*|<Units|Coil.Dat|0|3|1
Nrows=  6.000000E+00
*|<Nrows|Coil.Dat|0|4|1
Ntubes=  1.200000E+01
*|<Ntubes|Coil.Dat|0|5|1
Lduct=  2.000000E+00
*|<Lduct|Coil.Dat|0|6|1
Wduct=  2.000000E+00
*|<Wduct|Coil.Dat|0|7|1
do=  8.330000E-02
*|<do|Coil.Dat|0|8|1
di=  7.917000E-02
*|<di|Coil.Dat|0|9|1
```

```
ktube=  2.270000E+02
*|<ktube|Coil.Dat|0|10|1
ft=  8.330000E-04
*|<ft|Coil.Dat|0|11|1
fs=  8.330000E-03
*|<fs|Coil.Dat|0|12|1
Nfin=  2.400000E+02
*|<Nfin|Coil.Dat|0|13|1
kfin=  2.270000E+02
*|<kfin|Coil.Dat|0|14|1
finm=  1.000000E+00
*|<finm|Coil.Dat|0|15|1
dcr=  1.670000E-01
*|<dcr|Coil.Dat|0|16|1
dclr=  1.670000E-01
*|<dclr|Coil.Dat|0|17|1
*|*
*|*                    *****System Parameters*****
*|*
END=  4.800000E+01
*|Simulation Duration     |hrs|      hrs|0|1.0|.125|10000.0|1 STEP=  1.250000E-01
*|Time Increment          |hrs|      hrs|0|1.0|0.005|500.000|1 FLOW=  4.000000E+04
*|Water Mass Flow Rate    |lbm/hr|       lbm/hr|0|1.0|30000|60000|1 TAOUT=
5.500000E+01
*|Air Outlet Temperature  |F|        F|0|1.0|40|80.0|1
CPA=.2399
CA=CPA*[17,2]
CPW=1.0
CW=CPW*FLOW
CMIN=MIN(CA,CW)
E1=CA*([1,3]-[3,1])
E2=MAX(.1,(CMIN*([1,3]-[6,1])))
E3=E1/E2
EHX=MIN(1,E3)
*CHPOW=3412*12,6


SIMULATION 0.0  END  STEP
*11 HOURS OF DISCHARGE AT 7.5 MINUTE INCREMENTS
WIDTH 132
TOLERANCES .0005 .0005
```

```
LIMITS 200 10

*------------------------------------------------------UNIT 1
TYPE 9 DATA READER (FREE)   #4.1.1
PARAMETERS 25
*Ndata   deltaT
*(-)     (hr)
7 1.0 1 1 0 2 1 0 3 1 0 4 1 0 5 1 0 6 1 0 -7 1 0 20 0

*------------------------------------------------------UNIT 17
TYPE 73 CONTROLLER
PARAMETERS 2
.2399 1.0
INPUTS 11
*LOAD TAIN TAOUT MW TWIN MODE
1,5 1,3 TAOUT FLOW 6,1 1,6 EHX 12,6 0,0 CHLCAP 1,7
0.0 75.0 61.0 0.0 30.0 1 .7 5.0 0.0 100000 23.0

UNIT 2 TYPE 33 PSYCHROMETRIC CHART
PARAMETERS 2
2 2
INPUTS 2
1,3 1,4
75.0 70.0

UNIT 3 TYPE 52 COOLING COIL  #4.6.8
PARAMETERS 16
Mode Units Nrows Ntubes Lduct Wduct do di ktube ft fs Nfin kfin finm dcr dclr
INPUTS 5
1,3 2,1 17,2 6,1 FLOW
85.0 .05 0. 29.0 0.0

UNIT 15 TYPE 33 PSYCHROMETRIC CHART
PARAMETERS 2
2 2
INPUTS 2
1,1 1,2
88.0 20.0

UNIT 12 TYPE 53  PARALLEL CHILLERS(1 UNIT)
#4.6.9 PARAMETERS 12
```

```
2   0.80 CHLCAP 0.0 15 18 CHLCAP 70.0 17.0 0.998
1.0 1
INPUTS 6
17,6 3,4 3,5 13,1 13,2 5,7
23.0 29.0 47750. 95.0 40000. 1

UNIT 13 TYPE 51 COOLING TOWER   #4.6.7
PARAMETERS 11
*Note:Enter 12 parameters for mode 2, 11 parameters for mode 1
*Units          Mode      Geom      Ncell   Va,cell,max   Pcell,max
*(2=eng)        (2=data)  (2=X flow)  (#)     (ft3/h)       (KW)
   2            1            1        4       4.42E5        2.0

*Va,off         Vs        Ti,sump   LU/C      Ndata/n     Print
*(ft3/hr)       (ft3)      (F)       (#)        (#)       (1=print)

6.63E4           30.       85.0      2.0       -0.63        2

INPUTS 9
*Twi            Mwi      Tai      Twb      Tsump      rel-fun speed
*(F)   (lbm/h)   (F)      (F)      (F)      (0-1)
12,3 12,4 1,1    15,2     0,0      0,0 0,0 0,0 0,0 95.0 40000. 95.
78.  80.0      1.0 1.0 1.0 1.0
*TRACE 12.0 12.5
*---------------------------------------------------------------
UNIT 4 TYPE 70 FLOW DIVERTER   #4.5.2
PARAMETERS 2
*MODE                            NSTK
*(4=tempering valve)             (iteration)
 4                               151

INPUTS 5
*Tbi     Mb        Tbout_tank    Tset   MODE2
*('F)    (lbm/hr)   ('F)                 ('F)   (1=CHARGE, 2=DISCHARGE)
12,1     12,2       5,3                  17,1     1,6
80.0     47750.0    33.0                 TSET     1
*TRACE 0.0 1.0

*---------------------------------------------------------------
UNIT 5 TYPE 74 ICE STORAGE TANK   #?.?.?
PARAMETERS 5
*Tfrz    Cp-brine                  deltaT          TankSize    NSTK
```

```
*('F)     (Btu/lb'F)                    (hr)               (-)
(#)
32.0     1.00     STEP      TNKSZ     151

INPUTS 3
*Tbin                Mb                   Mode
*('F)                (lbm/hr)             (1=charge,
2=discharge)
4,1                  4,2                  1,6
29.0                 47750.               1
TRACE 14.0 14.25

*-------------------------------------------------------------
UNIT 6 TYPE 11 TEE-PIECE   #4.5.2
PARAMETERS 1
*MODE
*(#)
1

INPUTS 4
*Tbint                  m-tank    Tbdiv   mbdiv
*('F)                   (lbm/h)            ('F)
(lbm/h)
5,3          5,4          4,3         4,4
32.0     20000.0      60.0    40000.0
*TRACE 0.0       1.0

*-------------------------------------------------------------
UNIT 11 TYPE 24 QUANTITY INTEGRATOR  #4.1.5
INPUTS 4
*No
*(#)
3,6 5,5 17,4 17,5
0.0 0.0 0.0 0.0

*-----------------------------------------------------------
UNIT 8 TYPE 25 PRINTER   #4.10.1
PARAMETERS 4
*deltaT               P-start   P-end    LU
*(h)                  (h)       (h)      (#)
STEP                  0         72.0     7
```

```
INPUTS 5
*No.
*(#)
3,6 5,1 5,6 STORE 4,2
COIL EFFECT DISCH ICES MFICE

*-----------------------------------------------------------
UNIT 9 TYPE 25 printer   #4.10.1
PARAMETERS 4
*deltaT          P-start          P-end    LU
*(h)             (h)              (h)      (#)
STEP             0 72      8

INPUTS 9
*No.
*(#)
4,2 .....3,4.....12,1.....4,1     4,3     5,3     6,1     13,1     2,2
MFICE Tcoilo    Tchil     Ttk,i   Tdv,o   Ttk,o    Ttee,o   Tct,0     Twb

*-----------------------------------------------------------
UNIT 10 TYPE 25 printer   #4.10.1
PARAMETERS 4
*deltaT            P-start         P-end     LU
*(h)               (h)             (h)       (#)
STEP               0               72        9
INPUTS 5
*No.
*(#)
*12,2    4,2      4,4      5,4       6,2       1,7   1,8    1,6 *Mchi,o  Mtk,i    Mdv,o
Mtk,o      Ttee,o    Mcon   mode  conoff 11,1 3,1 12,8 11,3 11,4
load taout cop oppow ppow

*-----------------------------------------------------------

equations 3
in1=[17,3]/1000
in2=[3,6]/1000
in3=[12,5]/1000
UNIT 65 TYPE 65 ONLINE
PARAMETERS 9
* # UPPER  # LOWER
3 4
```

*MIN & MAX UPPER,LOWER
0 400 20 68
* TIMESTEP FOR PRINT REFRESH
UNITS (3=HR) 1 1 3
INPUTS 20
IN1 IN2 in3 3,4 12,1 5,3 6,1 0,0 0,0
0,0 0,0 0,0 0,0 0,0 0,0 0,0 0,0 0,0 0,0
0,0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0
0 0

END

## Type 70 Flow Diverter

```
      SUBROUTINE TYPE70(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
C  THIS SUBROUTINE SIMULATES A FLOW DIVERTER, MIXER, OR T-PIECE.
C  MODES 1-5 MIX/DIVERT A FLUID WITH 1 PROPERTY (USUALLY
TEMPERATURE)
C  MODES 6-10 MIX/DIVERT A FLUID WITH 2 PROPERTIES (USUALLY TEMP.
AND HUMIDITY)
C    IN MODES 1 AND 6, A MIXING T-PIECE IS SIMULATED
C    IN MODES 2 AND 7, THE FLOW DIVERTER HAS 1 INLET AND 2 POSSIBLE
OUTLETS
C    IN MODES 3 AND 8, THE FLOW MIXER HAS 2 POSSIBLE INLETS AND 1
OUTLET
C    IN MODES 4 AND 9, THE COMPONENT FUNCTIONS AS A FLOW
DIVERTOR WITH INTERNAL
C       CONTROL DERIVED FROM THE INPUTS.
C    IN MODES 5 AND 10, THE COMPONENT FUNCTIONS AS IN MODE 4
EXCEPT THAT ALL
C       FLOW IS DIVERTED WHEN THE SOURCE TEMPERATURE IS GREATER
THAN THE
C       HEATED FLOWSTREAM TEMPERATURE (TYPICALLY THE TANK
TEMPERATURE).
C
      DIMENSION XIN(10),OUT(20),PAR(15),INFO(10)
```

```
C    DETERMINE THE MODE OF OPERATION
      MODE=IFIX(PAR(1)+0.5)
      IF (MODE .LT. 1 .OR. MODE .GT. 10) THEN
        CALL TYPECK(4,INFO,0,0,0)
      ENDIF
C    DETERMINE IF THIS IS THE FIRST CALL OF SIMULATION
      IF (INFO(7) .LE. -1) THEN
C      FIRST CALL OF SIMULATION
        INFO(9)=0
C      CASE ON (MODE)
        GO TO (21,22,23,24,24,26,27,28,29,29),MODE
C      MODE 1
21        INFO(6)=2
          CALL TYPECK(1,INFO,4,1,0)
        GO TO 90
C      MODE 2
22        INFO(6)=4
          CALL TYPECK(1,INFO,3,1,0)
        GO TO 90
C      MODE 3
23        INFO(6)=2
          CALL TYPECK(1,INFO,5,1,0)
          GO TO 90
C      MODES 4 AND 5
24        INFO(6)=5
          CALL TYPECK(1,INFO,5,2,0)
          GO TO 90
C      MODE 6
26        INFO(6)=3
          CALL TYPECK(1,INFO,6,1,0)
          GO TO 90
C      MODE 7
27        INFO(6)=6
          CALL TYPECK(1,INFO,4,1,0)
          GO TO 90
C      MODE 8
28        INFO(6)=3
          CALL TYPECK(1,INFO,7,1,0)
          GO TO 90
C      MODE 9 AND 10
29        INFO(6)=7
```

```
          CALL TYPECK(1,INFO,5,2,0)
          GO TO 90
C         END CASE
90        CONTINUE
      END IF
C
C     BRANCH ACCORDING TO MODE  (CASE)
      GO TO (100,200,300,400,400,600,700,800,900,900), MODE
C     MODE 1:  T-PIECE
100     T1=XIN(1)
        FLOW1=XIN(2)
        T2=XIN(3)
        FLOW2=XIN(4)
        FLOWO=FLOW1+FLOW2
        IF(FLOWO.GT.0.0) THEN
          TO=(FLOW1*T1+FLOW2*T2)/FLOWO
          OUT(1)=TO
        END IF
        OUT(2)=FLOWO
      RETURN
C     MODE 2:  FLOW DIVERTER
200     TIN=XIN(1)
        FLOW=XIN(2)
        GAM=XIN(3)
        FLOW2 = FLOW*GAM
        FLOW1 = FLOW - FLOW2
        IF (FLOW1 .GT. 0.) OUT(1) = TIN
        OUT(2) = FLOW1
        IF (FLOW2 .GT. 0.) OUT(3) = TIN
        OUT(4) = FLOW2
      RETURN
C     MODE 3:  FLOW MIXER
300     T1=XIN(1)
        FLOW1=XIN(2)
        T2=XIN(3)
        FLOW2=XIN(4)
        GAM=XIN(5)
        FLOWO=FLOW1*(1.-GAM)+FLOW2*GAM
        IF (FLOWO .GT. 0.0) THEN
          OUT(1)=(T1*FLOW1*(1.-GAM)+T2*FLOW2*GAM)/FLOWO
        END IF
```

```
      OUT(2)=FLOWO
     RETURN
C    MODES 4 AND 5:  TEMPERATURE CONTROLLED FLOW DIVERTER
400     NSTK=IFIX(PAR(2)+0.5)
      IF(INFO(7).GT.NSTK)THEN
        TSOURC=XIN(1)
        FLOW=XIN(2)
        TCOLD=XIN(3)
        TSET=XIN(4)
        OUT(1)=XIN(1)
        OUT(2)=GAM*FLOW
        OUT(3)=XIN(1)
        OUT(4)=FLOW-OUT(2)
        OUT(5)=GAM
        RETURN
      ENDIF
      TSOURC=XIN(1)
      FLOW=XIN(2)
      TCOLD=XIN(3)
      TSET=XIN(4)
      MODE2=XIN(5)
      GAM = 1.0
C^^^^^^^^^^^^^^^^^^^ Jan 21 '93  by kawashima
      IF(MODE2 .EQ. 1) GO TO 2000
      IF(TSOURC .GE. TSET) THEN
        IF(TSET .GT. TCOLD) GAM = (TSOURC-TSET)/(TSOURC-TCOLD)
        IF(TSET .LE. TCOLD) GAM =1.0
      ELSE
        GAM = 0.0
      ENDIF
2000    CONTINUE
C       WRITE(*,60001) GAM,TSOURC,TSET,TCOLD
60001   FORMAT(1X,'+++++++GAM,TSOURCE,TSET,tcold',4F10.4)

C  *** NOTE CHANGE
C       IF (TCOLD .GT. TSET .AND. MODE.EQ.4) GAM=1.0
C       IF (THOT.LT.TSOURC .AND. MODE.EQ.5) GAM = 0.0
C      MODE 4 AND 5 OUTPUTS
      OUT(1)=XIN(1)
      OUT(2)=GAM*FLOW
      OUT(3)=XIN(1)
```

```
      OUT(4)=FLOW-OUT(2)
      OUT(5)=GAM
     RETURN
C    MODE 6:  T-PIECE
600    T1=XIN(1)
      OMEGA1=XIN(2)
      FLOW1=XIN(3)
      T2=XIN(4)
      OMEGA2=XIN(5)
      FLOW2=XIN(6)
      FLOWO=FLOW1+FLOW2
      IF(FLOWO.GT.0.0) THEN
        TO=(FLOW1*T1+FLOW2*T2)/FLOWO
        OUT(1)=TO
        OMEGAO=(FLOW1*OMEGA1+FLOW2*OMEGA2)/FLOWO
        OUT(2)=OMEGAO
      END IF
      OUT(3)=FLOWO
     RETURN
C    MODE 7:  FLOW DIVERTER
700    TIN=XIN(1)
      OMEGAI=XIN(2)
      FLOW=XIN(3)
      GAM=XIN(4)
      FLOW2 = FLOW*GAM
      FLOW1 = FLOW - FLOW2
      IF (FLOW1 .GT. 0.) THEN
        OUT(1) = TIN
        OUT(2) = OMEGAI
      END IF
      OUT(3) = FLOW1
      IF (FLOW2 .GT. 0.) THEN
        OUT(4) = TIN
        OUT(5) = OMEGAI
      END IF
      OUT(6) = FLOW2
     RETURN
C    MODE 8:  FLOW MIXER
800    T1=XIN(1)
      OMEGA1=XIN(2)
      FLOW1=XIN(3)
```

```
      T2=XIN(4)
      OMEGA2=XIN(5)
      FLOW2=XIN(6)
      GAM=XIN(7)
      FLOWO=FLOW1*(1.-GAM)+FLOW2*GAM
      IF (FLOWO .GT. 0.0) THEN
        OUT(1)=(T1*FLOW1*(1.-GAM)+T2*FLOW2*GAM)/FLOWO
        OUT(2)=(OMEGA1*FLOW1*(1.-GAM)+OMEGA2*FLOW2*GAM)/FLOWO
      END IF
      OUT(3)=FLOWO
     RETURN
C    MODES 9 AND 10:  TEMPERATURE CONTROLLED FLOW DIVERTER
900     NSTK=IFIX(PAR(2)+0.5)
      IF(INFO(7).GT.NSTK)RETURN
      TSOURC=XIN(1)
      OMEGAS=XIN(2)
      FLOW=XIN(3)
      THOT=XIN(4)
      TSET=XIN(5)
      GAM = 1.0
      IF (THOT .GT. TSET) GAM = (TSET-TSOURC)/(THOT-TSOURC)
      IF (THOT.LT.TSOURC .AND. MODE.EQ.9) GAM=1.0
      IF (THOT.LT.TSOURC .AND. MODE.EQ.10) GAM = 0.0
C      MODE 9 AND 10 OUTPUTS
      OUT(1)=TSOURC
      OUT(2)=OMEGAS
      OUT(3)=GAM*FLOW
      OUT(4)=TSOURC
      OUT(5)=OMEGAS
      OUT(6)=FLOW-OUT(3)
      OUT(7)=GAM
     RETURN
     END
```

## **Type 73 Controller**

```
      SUBROUTINE TYPE73(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
      DIMENSION XIN(11),OUT(6),T(1),DTDT(1),PAR(1),INFO(10)
      Real Tset,Load,EHX,Tain,ma,mw,Cpa,Cpw,Ca,Cw,Cmin,CHILCAP
      Real TAOUT,TWIN,MODE,POWER,OPOW,PPOW,TOWPOW,TSETCH
C     Integer Mode
C

    NI=11
    NP=2
    ND=0
    INFO(6)=1
    CALL TYPECK(1,INFO,NI,NP,ND)
      LOAD=XIN(1)
      TAIN=XIN(2)
      TAOUT=XIN(3)
      MW=XIN(4)
      TWIN=XIN(5)
      MODE=XIN(6)
      EHX=XIN(7)
      POWER=XIN(8)
      TOWPOW=XIN(9)
      CHILCAP=XIN(10)
      TSETCH=XIN(11)

C
      Cpa=PAR(1)
      Cpw=PAR(2)
C
      if (MODE .LT. 2) then
      ma=0.0
      Tset=35.0
      LOAD=0.0
      OPOW=POWER+TOWPOW
      PPOW=0.0
      else
      ma=LOAD/(Cpa*(TAIN-TAOUT))
C
      Ca=ma*Cpa
      Cw=mw*Cpw
      Cmin=min(Ca,Cw)
```

```
C
C        EHX=(Ca*(TAIN-TAOUT))/(Cmin*(Tain-TWIN))
C
         Tset=Tain-LOAD/(max(20.0,(Cmin*EHX)))
C
         IF (LOAD .LE. CHILCAP) THEN
         TSETCH=TSET
         ENDIF
C
         OPOW=0.0
         PPOW=POWER+TOWPOW
         endif
C
         OUT(1)=Tset
         OUT(2)=ma
         OUT(3)=LOAD
         OUT(4)=OPOW
         OUT(5)=PPOW
         OUT(6)=TSETCH
C
         RETURN
C
         END
```

## Type 74 Ice-Storage Tank

```
         SUBROUTINE TYPE74(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
C      ICE STORAGE TANK MODEL USING A REGRESSION OF EFFECTIVENESS
CURVES
C
C      CALMAC 190 Ton Hour Tank
C
C      TEMP.....'F              1 BTU     =251.9958  cal
C      HEAT.....BTU             1 ft3     =7.480520  gal
C      FLOW.....lb/h            1 lb      =0.4535924 Kg
C      DENSITY..lb/ft3          1 g/cm3   =62.42795  lb/ft3
C                          1 BTU/lb'f=1.000000  Kcal/Kg'C
C
```

```
      DIMENSION      XIN(3),OUT(7),T(1),DTDT(1),PAR(4),INFO(1)
      DOUBLE PRECISION C0,C1,C2,C3,C4,C5,C6,C7,C8,C9,C10,C11,
     #          C12,C13,C14,C15
      REAL          DENSIT,MAXCAP,HIF,CVW,MWTOT,STORE,QIO,MB
      INTEGER              NCH
      DOUBLE PRECISION VB,B
C
C++++++++++++++++++++++++++++++
C     WRITE(*,60005) INFO(7)
60005 FORMAT(1X,'     INFO(7)=',I3)
C  if initial call INFO(7)=-1
C
      IF(INFO(7).LT.0) THEN
C        STORE=0.
         QIO=0.
         NI=3
         NP=5
         ND=0
         INFO(6)=6
         CALL TYPECK(1,INFO,NI,NP,ND)
C
C        PAR(1); FREEZING POINT=32.0
C        PAR(2); SPECIFIC HEAT OF BRINE =1.0 (Btu/lb'F)
C        PAR(3); TIME STEP=SETP=0.125
C        PAR(4); TANK SIZE=1.0
C        PAR(5); NSTK ITERATION LIMIT=151
C
         TFP=PAR(1)
         CPB=PAR(2)
         DELTAT=PAR(3)
         TNKSZ=PAR(4)
         NSTCK=PAR(5)
      STORE=TNKSZ*2166485.0
         ENDIF

C
C if normal time step INFO(7)=0 is the first call
C if iteration step   INFO(7)=1,2,3,4......
C
      IF(INFO(7) .EQ. 0) THEN
       STORE=STORE+QIO
```

```
      ENDIF
C
C          XIN(1); BRINE INLET TEMP.
C          XIN(2); BRINE MASS FLOW Kg/h
C          XIN(3); MODE =1;CHARGING
C                      =2;DISCHARGING
C
C          Mb; brine mass flow lb/h
C          DENSIT; BRINE DENSITY=64.93 lb/ft3=(1.04E+3 kg/m3)
C          Vb; BRINE MASS FLOW, gal/min (1 ft3=7.480520 gal)
C          mwtot; VOLUME OF WATER IN THE TANK, lb(1 lb=0.4535924 Kg)
C          Hif; LATENT HEAT=144 BTU/lb =(80 Kcal/Kg) (1 BTU=251.9958cal)
C          Cvw; SPECIFIC HEAT OF WATER=1.01 BTU/lb'F (10.1 Kcal/kg'C)

      TBIN=XIN(1)
      MB=XIN(2)
      MODE=XIN(3)
C
      DENSIT=64.93
      VB=MB*7.481/(DENSIT*60.0)
      MWTOT=13527.
      HIF=144.0
      CVW=1.01
      MAXCAP=TNKSZ*MWTOT*(HIF+CVW*(48.0 -TFP))
C
      IF(MODE .EQ. 1) GO TO 1000
      IF(MODE .NE. 2) GO TO 999
C
C      ------ MODE=2  DISCHARGING PERIOD  ------
           NCH=1
        IF(INFO(7).LT.NSTCK) THEN
          IF(STORE .GE. MAXCAP) STORE=MAXCAP
          IF(STORE .LE. 0.0   ) STORE=0.0
          B=1-STORE/MaxCap
        ELSE
          GO TO 999
        ENDIF
C
        IF(B .LT. 0.66) THEN
         C0=0.84119769D0
         C1=  0.200276759D0      *B
```

```
      C2=  1.636547199D0      *B**2
      C3= -5.204433828D0      *B**3
      C4=  4.196217689D0      *B**4
      C5=  0.015118414D0      *VB
      C6= -0.000390064D0      *VB**2
      C7=  3.64763D-6         *VB**3
      C8= -1.24338D-8         *VB**4
      C9= -0.053871746D0      *B*VB
      C10= 0.064822502D0      *B**2*VB
      C11= 0.000354565D0      *B*VB**2
      C12=-0.034354947D0      *B**3*VB
      C13=-0.000142311D0      *B**2*VB**2
      C14=-9.15865D-7         *B*VB**3
      C15=0.0
      GO TO 2000
     ELSE
      C0=  25.62156701D0
      C1=-110.463303D0        *B
      C2= 176.6331532D0       *B**2
      C3=-114.555632D0        *B**3
      C4=  22.86186786D0      *B**4
      C5=  -0.01026212D0      *VB
      C6=  -0.0004725D0       *VB**2
      C7=   5.03616D-7        *VB**3
      C8=  -2.1181D-9         *VB**4
      C9=   0.105010295D0     *B*VB
      C10= -0.27724386D0      *B**2*VB
      C11=  0.001260003D0     *B*VB**2
      C12=  0.179974285D0     *B**3*VB
      C13= -0.00078403D0      *B**2*VB**2
      C14= -1.8073D-7         *B*VB**3
      C15=0.0
      GO TO 2000
     END IF


C
C      -------  MODE=1  CHARGING PERIOD  ---------
1000   IF(INFO(7) .LT. NSTCK) THEN
        IF(STORE .GE. MAXCAP)THEN
             STORE=MAXCAP
```

```
            NCH=0
        ELSE
            NCH=1
        ENDIF
        IF(STORE .LE. 0.0   ) STORE=0.0
        B=STORE/MAXCAP
        IF (B .GE. 1.0) GOTO 2500
    ELSE
        GO TO 999
    ENDIF
C
      IF(B .LE. 0.755) THEN
        C0=1.077255269D0
        C1=-0.079156996D0  *B
        C2=-0.0046742D0   *VB
        C3=0.0
        C4=0.0
        C5=0.0
        C6=0.0
        C7=0.0
        C8=0.0
        C9=0.0
        C10=0.0
        C11=0.0
        C12=0.0
        C13=0.0
        C14=0.0
        C15=0.0
        GO TO 2000
        ELSE
        C0= 1.511144226D0
        C1= 0.22757868D0    *DLOG10(-B+1.0D0)
        C2=-0.009864783D0   *VB
        C3= 3.83656D-5      *VB**2
        C4=-0.281804303D0   *B
        C5=0.0
        C6=0.0
        C7=0.0
        C8=0.0
        C9=0.0
        C10=0.0
```

```
          C11=0.0
          C12=0.0
          C13=0.0
          C14=0.0
          C15=0.0
          GO TO 2000
          END IF
C       -------------------------------
C
2000    E=C0+C1+C2+C3+C4+C5+C6+C7+C8+C9+C10+C11+C12+C13+C14+C15

        IF (E .GE. 1.0) E=1.0
        IF (E .LE. 0.0) E=0.0
2500    IF (B .GE. 1.0) E=0.0
C
        TBOUT=TBIN-E*(TBIN-TFP)
        QIO=MB*(TBOUT-TBIN)*DELTAT
C
        OUT(1)=E
        OUT(2)=TBIN
        OUT(3)=TBOUT
        OUT(4)=MB
C       OUT(5);BTU/h
        OUT(5)=QIO/DELTAT
        OUT(6)=B
        OUT(7)=NCH
C
C++++++++++++++++++++++++++++++++++++++++++++++++++
C    ICE TANK OUTPUT CAPACITY CHECK
C
C     WRITE(*,60006) TBIN,TBOUT,MB, QIO
60006 FORMAT(1X,'TBIN=',F6.3,'   TBOUT=',F6.3,'   MB=',F10.1,/,
    +       1X,' QIO=',F12.3)

        RETURN
C
C       999     WRITE(*,600)
999     CONTINUE
600    FORMAT(1X,'----Something wrong in TYPE75-----')
        RETURN
c
```

END

## Chiller Data (Chil.dat)

| | | |
|---|---|---|
| 0.566648 | 0.714286 | 0.728784 |
| 0.653609 | 0.642857 | 0.758392 |
| 0.737764 | 0.571429 | 0.763216 |
| 0.82192 | 0.5 | 0.769843 |
| 0.90327 | 0.428571 | 0.74502 |
| 0.993036 | 0.357143 | 0.721882 |
| 0.532986 | 0.857143 | 0.83451 |
| 0.614336 | 0.785714 | 0.867412 |
| 0.695687 | 0.714286 | 0.875294 |
| 0.774232 | 0.642857 | 0.876706 |
| 0.858387 | 0.571429 | 0.864 |
| 0.934127 | 0.5 | 0.848824 |
| 0.502129 | 1 | 0.933608 |
| 0.577869 | 0.928571 | 0.961333 |
| 0.656414 | 0.857143 | 0.981882 |
| 0.732154 | 0.785714 | 0.982588 |
| 0.807894 | 0.714286 | 0.971294 |
| 0.883634 | 0.642857 | 0.951176 |

## Cooling Coil Data

## (Coil.dat) (ABC used in simulations)

**Number of Different Coils**

**Name Mode Units Nrows Ntubes Lduct Wduct do di ktube ft**

**fs Nfin kfin finmode dcr dclr**

2

Cooling Coil ABC,1,2,6,12,2,2,.0833,.07917,227,8.33E-4,

8.33E-3,240,227,1,.167,.167

Cooling Coil XYZ,1,2,5,12,2,2,.0833,.07917,227,8.33E-4,

8.33E-3,240,227,1,.167,.167

## Building Load Data
## for Diversity of .83333
## (Pbld1.dat)

**Tamb, RHamb, Tain, RHain, Load, Mode, Tchilled water set**

76,56,75,0,215000,2,23.0
78,52,75,0,215000,2,30.0
81,46,75,0,140000,2,30.0
82,44,75,0,165000,2,30.0
84,40,75,0,177000,2,30.0
87,34,75,0,190000,2,30.0
88,32,75,0,200000,2,30.0
91,26,75,0,240000,2,30.0
91,26,75,0,240000,2,30.0
89,30,75,0,225000,2,30.0
88,32,75,0,200000,2,30.0
86,36,75,0,0,1,23.0
82,44,75,0,0,1,23.0
80,48,75,0,0,1,23.0
78,52,75,0,0,1,23.0
76,56,75,0,0,1,23.0

74,60,75,0,0,1,23.0
72,64,75,0,0,1,23.0
70,68,75,0,0,1,23.0
69,70,75,0,0,1,23.0
71,66,75,0,0,1,23.0
72,64,75,0,0,1,23.0
73,62,75,0,0,1,23.0
74,60,75,0,0,1,23.0
76,56,75,0,215000,2,23.0
78,52,75,0,215000,2,30.0
81,75,0,140000,2,30.0
82,75,0,165000,2,30.0
84,75,0,177000,2,30.0
87,75,0,190000,2,30.0
88,75,0,200000,2,30.0
91,75,0,240000,2,30.0
91,75,0,240000,2,30.0
89,75,0,225000,2,30.0
88,75,0,200000,2,30.0
86,75,0,0,1,23.0
82,75,0,0,1,23.0
80,75,0,0,1,23.0
78,75,0,0,1,23.0
76,75,0,0,1,23.0
74,75,0,0,1,23.0
72,75,0,0,1,23.0
70,75,0,0,1,23.0
69,75,0,0,1,23.0
71,75,0,0,1,23.0
72,75,0,0,1,23.0
73,75,0,0,1,23.0
74,75,0,0,1,23.0
76,75,0,215000,2,23.0
78,75,0,215000,2,30.0
81,75,0,140000,2,30.0
82,75,0,165000,2,30.0
84,75,0,177000,2,30.0
87,75,0,190000,2,30.0
88,75,0,200000,2,30.0
91,75,0,240000,2,30.0
91,75,0,240000,2,30.0

89,75,0,225000,2,30.0
88,75,0,200000,2,30.0
86,75,0,0,1,23.0
82,75,0,0,1,23.0
80,75,0,0,1,23.0
78,75,0,0,1,23.0
76,75,0,0,1,23.0
74,75,0,0,1,23.0
72,75,0,0,1,23.0
70,75,0,0,1,23.0
69,75,0,0,1,23.0
71,75,0,0,1,23.0
72,75,0,0,1,23.0
73,75,0,0,1,23.0
74,75,0,0,1,23.0
76,75,0,215000,2,23.0

## Building Load Data for Diversity of .66667 (Pbld2.dat)

76,56,75,0,215000,2,23.0
78,52,75,0,200000,2,30.0
81,46,75,0,100000,2,30.0
82,44,75,0,127500,2,30.0
84,40,75,0,160000,2,30.0
87,34,75,0,180000,2,30.0
88,32,75,0,200000,2,30.0
91,26,75,0,300000,2,30.0
91,26,75,0,300000,2,30.0
89,30,75,0,225000,2,30.0
88,32,75,0,200000,2,30.0
86,36,75,0,0,1,23.0
82,44,75,0,0,1,23.0
80,48,75,0,0,1,23.0
78,52,75,0,0,1,23.0

76,56,75,0,0,1,23.0
74,60,75,0,0,1,23.0
72,64,75,0,0,1,23.0
70,68,75,0,0,1,23.0
69,70,75,0,0,1,23.0
71,66,75,0,0,1,23.0
72,64,75,0,0,1,23.0
73,62,75,0,0,1,23.0
74,60,75,0,0,1,23.0
76,56,75,0,215000,2,23.0
78,52,75,0,200000,2,30.0
81,46,75,0,100000,2,30.0
82,44,75,0,127500,2,30.0
84,40,75,0,160000,2,30.0
87,34,75,0,180000,2,30.0
88,32,75,0,200000,2,30.0
91,26,75,0,300000,2,30.0
91,26,75,0,300000,2,30.0
89,30,75,0,225000,2,30.0
88,32,75,0,200000,2,30.0
86,36,75,0,0,1,23.0
82,44,75,0,0,1,23.0
80,48,75,0,0,1,23.0
78,52,75,0,0,1,23.0
76,56,75,0,0,1,23.0
74,60,75,0,0,1,23.0
72,64,75,0,0,1,23.0
70,68,75,0,0,1,23.0
69,70,75,0,0,1,23.0
71,66,75,0,0,1,23.0
72,64,75,0,0,1,23.0
73,62,75,0,0,1,23.0
74,60,75,0,0,1,23.0
76,56,75,0,215000,2,23.0
78,52,75,0,200000,2,30.0
81,46,75,0,100000,2,30.0
82,44,75,0,127500,2,30.0
84,40,75,0,160000,2,30.0
87,34,75,0,180000,2,30.0
88,32,75,0,200000,2,30.0
91,26,75,0,300000,2,30.0

91,26,75,0,300000,2,30.0
89,30,75,0,225000,2,30.0
88,32,75,0,200000,2,30.0
86,36,75,0,0,1,23.0
82,44,75,0,0,1,23.0
80,48,75,0,0,1,23.0
78,52,75,0,0,1,23.0
76,56,75,0,0,1,23.0
74,60,75,0,0,1,23.0
72,64,75,0,0,1,23.0
70,68,75,0,0,1,23.0
69,70,75,0,0,1,23.0
71,66,75,0,0,1,23.0
72,64,75,0,0,1,23.0
73,62,75,0,0,1,23.0
74,60,75,0,0,1,23.0
76,56,75,0,215000,2,23.0

## Building Load Data
## for Diversity of .61538
## (Pbld3.dat)

76,56,75,0,215000,2,23.0
78,52,75,0,200000,2,30.0
81,46,75,0,92000,2,30.0
82,44,75,0,110000,2,30.0
84,40,75,0,130000,2,30.0
87,34,75,0,185000,2,30.0
88,32,75,0,200000,2,30.0
91,26,75,0,325000,2,30.0
91,26,75,0,325000,2,30.0
89,30,75,0,225000,2,30.0
88,32,75,0,200000,2,30.0
86,36,75,0,0,1,23.0
82,44,75,0,0,1,23.0

80,48,75,0,0,1,23.0
78,52,75,0,0,1,23.0
76,56,75,0,0,1,23.0
74,60,75,0,0,1,23.0
72,64,75,0,0,1,23.0
70,68,75,0,0,1,23.0
69,70,75,0,0,1,23.0
71,66,75,0,0,1,23.0
72,64,75,0,0,1,23.0
73,62,75,0,0,1,23.0
74,60,75,0,0,1,23.0
76,56,75,0,215000,2,23.0
78,52,75,0,200000,2,30.0
81,46,75,0,92000,2,30.0
82,44,75,0,110000,2,30.0
84,40,75,0,130000,2,30.0
87,34,75,0,185000,2,30.0
88,32,75,0,200000,2,30.0
91,26,75,0,325000,2,30.0
91,26,75,0,325000,2,30.0
89,30,75,0,225000,2,30.0
88,32,75,0,200000,2,30.0
86,36,75,0,0,1,23.0
82,44,75,0,0,1,23.0
80,48,75,0,0,1,23.0
78,52,75,0,0,1,23.0
76,56,75,0,0,1,23.0
74,60,75,0,0,1,23.0
72,64,75,0,0,1,23.0
70,68,75,0,0,1,23.0
69,70,75,0,0,1,23.0
71,66,75,0,0,1,23.0
72,64,75,0,0,1,23.0
73,62,75,0,0,1,23.0
74,60,75,0,0,1,23.0
76,56,75,0,215000,2,23.0
78,52,75,0,200000,2,30.0
81,46,75,0,92000,2,30.0
82,44,75,0,110000,2,30.0
84,40,75,0,130000,2,30.0
87,34,75,0,185000,2,30.0

88,32,75,0,200000,2,30.0
91,26,75,0,325000,2,30.0
91,26,75,0,325000,2,30.0
89,30,75,0,225000,2,30.0
88,32,75,0,200000,2,30.0
86,36,75,0,0,1,23.0
82,44,75,0,0,1,23.0
80,48,75,0,0,1,23.0
78,52,75,0,0,1,23.0
76,56,75,0,0,1,23.0
74,60,75,0,0,1,23.0
72,64,75,0,0,1,23.0
70,68,75,0,0,1,23.0
69,70,75,0,0,1,23.0
71,66,75,0,0,1,23.0
72,64,75,0,0,1,23.0
73,62,75,0,0,1,23.0
74,60,75,0,0,1,23.0
76,56,75,0,215000,2,23.0

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