SIMULATION OF THERMAL STORAGE SYSTEMS IN AN INTEGRATED BUILDING SIMULATION PROGRAM

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ABSTRACT

This paper presents simulation results for an ice tank thermal storage system using non optimal control strategies. The simulation has been implemented in the IBLAST integrated energy analysis program which is capable of a building, fan systems, and central plant simultaneously. The system-plant water loop solution technique, which was developed to enable the ice storage system to be simulated is also described as is the implementation of detailed cooling coil models. In addition, results are presented which demonstrate the effects of an improperly sized storage device on zone conditions and energy consumption.

INTRODUCTION

Thermal storage systems in the U.S. have been developed in response to the widespread use of time of day rates by electric utility companies to encourage large consumers of electricity to limit their "on-peak" electric power consumption. By producing ice or chilled water during the overnight "off-peak" hours, all or a fraction of the next day's cooling load can be met without using chillers. The only on-peak energy consumption being that required to circulate air in the zones and chilled water between the storage tank and the cooling coils. Generally, ice storage systems can be divided into two categories: direct and indirect storage. In direct storage systems, for example the ice harvester, ice is formed on the evaporator plates. In indirect storage a brine solution (usually a mix of water and ethylene glycol) is circulated between the evaporator and the storage tank.

Ice storage systems can be operated in several modes: full storage, chiller priority, storage priority, and perfect or optimal control. In full storage storage is used to meet the entire on-peak cooling load. Chiller priority and storage priority both utilise storage to meet a fraction of the on-peak cooling load. Optimal control requires that the next day's cooling load be predicted to determine how much ice must be built overnight to meet the cooling demand. However, this is very difficult to accomplish in real systems. Building excess ice does adversely affect the performance of direct storage systems, however with indirect storage the penalties for ice remaining at the end of the discharge period are usually small. Current practice in the control of thermal storage systems is usually to fully charge the storage tank during the off-peak hours, guaranteeing that sufficient storage will be available to meet the next day's load.

In this paper, the simulation of indirect ice storage systems in a building simulation program is considered. Previously, methods have been detailed which simulate the interactions of zones and air handling systems in a simulation program based on a zone energy balance (Taylor et. al. 1990, 1991). In the program, called IBLAST (Integrated Building Loads Analysis and System Thermodynamics), the system-plant steady-state energy balance has been formulated and solved simultaneously. This has made possible implementation in the program of detailed models of thermal storage systems and the water loop required to connect them to the cooling coils. Indirect ice storage systems were chosen for simulation because they can be operated efficiently without using perfect control.

PLANTS INTEGRATION IN IBLAST

Integration of the central plants simulation in IBLAST required several new models to be implemented to allow for realistic interactions between the system and plant components. In order to simulate ice storage equipment it was critical to have detailed models of the heat exchange that occurs in the cooling coils, and the control of the water and air temperatures and flow rates through the coils. These models simulate the interdependencies between the systems and plant a detailed coil model which predicted changes in air and water flow variables across the coil based on the coil geometry was required. The input required to model the coil includes a complete geometric description which in most cases should be derivable from manufacturer's data. A similar model is has been previously presented (Elmahdy and Mitalas, 1977).

In IBLAST, the coil model has been developed in two forms. One form calculates all the coil outlet conditions when the coil inlet condtions are provided. The second form solves for the coil water flow rate and the coil water outlet temperature when the coil inlet and outlet air temperatures, coil air flow rate, and coil entering water temperature are specified. This is useful when it is desired to control the coil outlet air temperature.

The development of cooling and heating coil models allows interactions between water and air side variables to occur. But, the coil models alone are not sufficient because they provide no way of determining coil air and water inlet conditions. However, coil air inlet conditions are relatively easy to specify since they are dependent on the ambient outside air and zone return air conditions. The water side inlet conditions are a function of the capacity of the chillers or boilers and their ability to provide a certain flow rate of water at a specified temperature. Conservation of mass and energy between the coils and the plant allow the following relationships to be defined:

$$\sum_{\substack{\text{cooling}\\\text{coils}}} \dot{m}_{w,in} = \sum_{\substack{\text{cooling}\\\text{coils}}} \dot{m}_{w,out} = \dot{m}_{chiller,in} = \dot{m}_{chiller,out} \qquad (1)$$

$$T_{cooling\ coil,in} = T_{chiller,out} \qquad (2)$$

$$\sum_{i=1}^{\#\text{cooling coils}} \dot{m}_{i,w,out} T_{i,w,out} = \dot{m}_{chiller,in} T_{chiller,in} \qquad (3)$$

Additionally controls must be specified which are used by the system and plant to regulate the quantity of air flowing out of the coil and the outlet temperature. This can be expressed mathematically in terms of the steady state zone energy balance as follows:

$$\dot{Q}_{load} + \dot{m}_{air,in} C_{p,air} \left(T_{\substack{supply\\air}} - T_{zone} \right) = 0 \qquad (4)$$

where $\dot{Q}_{load}$ is the rate of energy transfer to the zone due to external and internal loads: radiation, conduction through the walls, infiltration, people, electrical equipment, etc. Since it is impossible to control $\dot{Q}_{load}$ in any practical way the goal of the building air conditioning system is to adjust $T_{supply\ air}$ and $\dot{m}_{air,in}$ so that Equation 4 is satisfied and T_{zone} is maintained within a range consistent with the comfort requirements of the buildings occupants. Implementation of controls in IBLAST was accomplished using the two forms of the cooling coil model.

SIMULATION PROCEDURE

In order to control the temperature in each zone, an air handling system and a sequence of zone temperature setpoints which define when the system heats and cools the zone, and also the desired zone temperature must be specified. These setpoints are analogous to the settings of a thermostat within each zone. The system response is then determined from the lagged zone temperature and the steady state zone energy balance. In effect, a zone temperature based on the steady state energy balance is first computed. The system response based on this temperature determines the system response which is then input to the transient zone energy balance to update the zone temperature. This methodology has been previously documented (Taylor et. al. 1990, 1991).

In IBLAST a time step for the zone temperature update of 0.25 hours resulted in a stable simulation. However, after implementation of coil controls, a much shorter time step was needed. An adaptive time step which selectively is shortened to maintain stability of the zone air energy balance when zone conditions were changing rapidly was therefore adopted. This required a two time step approach in which the zone air temperature is updated using the adaptive time step to ensure stability; and the contributions to the zone loads from the surfaces, infiltration, mixing, and user specified internal loads are updated at the default or user specified time step.

Simultaneous solution of the system and plant operating parameters required that the temperature of the water entering the coils must be the same as the temperature leaving the chillers or boilers. In addition, the temperature of the return water from the coils must be equal to the chiller or boiler entering water temperature. When the plant is not out of capacity the leaving water temperature is a constant equal to the design value. If either the chiller or boiler plant is overloaded then the secant method is used to iterate to the correct plant leaving water conditions with the plant operating at its maximum capacity.

ICE STORAGE MODELS

The ice tank model used in this work was based on the one developed for the BLAST program (Strand, 1992). In the ice tank system, a brine solution is circulated inside a spiral wound coil to freeze the water in the tank. When there is a cooling load on the plant the circulation in the brine loop is reversed and the cooling coils are brought into the loop. The performance of the indirect ice storage tank is modeled by relating the normalized outlet brine temperature, the normalized inlet brine temperature, and the normalized load on the storage tank q^*. Separate curve fits are required to model the charging and discharging cycles for each type of storage device. The ice tank discharge cycle uses a function for normalized outlet temperature T^*_o of the form:

$$T^*_o = f_2\left(C_2, Pc, T^*_i, q^*\right) \qquad (5)$$

where C_2 represents the coefficients of the polynomial fit of the performance data and P_c is the storage unit charged fraction. The normalized inlet and outlet temperatures are calculated as shown in Equation 6:

$$T_o^* = T_o/T_{freeze} \qquad T_i^* = T_i/T_{freeze} \tag{6}$$

where T_{freeze} is the absolute freezing temperature of the storage medium. The charging cycle of the ice tank requires the use of a second polynomial curve fit in which the percentage of tank charged, normalized load, brine mass flow rate $\dot{m}$, and coefficients C_3 are specified:

$$\Delta T_{lm}^* = f_3\left(C_3, P_c, q^*, \dot{m}^*\right) \tag{7}$$

The value returned ΔT^*_{lm} is the log mean temperature difference which is calculated from:

$$\Delta T_{lm}^* = \frac{T_{brine,in} - T_{brine,out}}{\ln\left(\dfrac{T_{brine,in} - T_{freeze}}{T_{brine,out} - T_{freeze}}\right)} \Bigg/ \Delta T_{nominal} \tag{8}$$

Additional details on methods for determining the sets of polynomial coefficients from performance data are available (BLAST Support Office, 1993).

During the tank discharge cycle, the correct temperatures and brine flow rate to pass to the ice storage system were determined from the coil simulation and also the location of the chiller with respect to the storage tank; three possibilities exist: upstream, downstream, and parallel. In each case the ice storage entering and leaving brine temperatures and the brine flow rate must satisfy mass and energy conservation around the brine loop. When the tank is charging the location of the chiller with respect to the ice storage unit is unimportant since the coldest possible brine temperature is desired for making ice. Therefore, the storage unit always receives brine at the chiller outlet temperature. The chiller inlet temperature thus is given by:

$$T_{chiller,in} = T_{chiller,out} + \frac{\dot{Q}_{absorbed,ice} + \dot{Q}_{absorbed,system}}{\dot{m}_{chiller} C_{brine}} \tag{9}$$

where, $\dot{Q}_{absorbed,ice}$ is the rate of energy absorption by the brine stream from the ice storage unit, and $\dot{Q}_{absorbed,system}$ is the rate of energy absorption from the cooling coils.

RESULTS

In order to evaluate the implementation of the ice storage models in IBLAST an input deck with a simple two zone building was designed. The plan view of the building, was rectangular with a total floor area of 6235 square feet, and a wall height of 10 feet. The walls and roof had an R value of 13 and the floor R value was 30. The external walls of the building have substantial glazing and, in addition, significant internal latent and electric loads were specified, as shown in Figure 1. The two zones were served by a variable air volume (VAV) system which supplied a maximum of 3800 cubic feet of air per minute to zone 1 and 2500 cubic feet per minute to zone 2. The nominal desired zone temperature for both zones was 78°F with the VAV system throttling the supply air flow rate from minimum at 77°F to maximum flow rate at 79°F. This control schedule was in effect from 8am to 6pm. Outside of these hours the zones were put in setback mode allowing the air temperatures to float. The on-peak period for the ice storage unit was from 10am to 6pm meaning that the chiller would not operate between these hours when full storage control was specified.

The ice tank model was tested in all three control modes and using all three chiller locations with each control strategy. However, the results indicated that chiller location had little effect on the performance of the ice storage system therefore all data subsequently presented is for an upstream chiller. Two additional cases were added to evaluate the effect of selecting an undersized storage tank on the zone temperatures when full storgae control was selected. Table 1 shows the values of compressor capacity, ice tank capacity, and PSHAVE used for the six groupings of three cases and the three cases with insufficient ice capacity. PSHAVE specifies the amount of cooling to be provided by the chiller during the on peak hours for and is applicable to ice priority control only.

Grouping	Chiller Capacity (kBtu/hr)	Ice Storage Capacity (kBtu)	PSHAVE (kBtu/hr)
Full Storage	200	1800	N/A
Ice Priority	200	1800	100
Chiller Priority	120	1800	N/A
Insuff. Cap.	200	1200	N/A
Minimal Cap.	100	1000	N/A

Table 1: Chiller capacity, ice storage capacity, and PSHAVE values used for simulation test

Figures 2 through 6 show the performance of the ice tank system when there is sufficient storage capacity to meet the on peak cooling loads. Each graph shows the variation temperature through one day of simulation for each zone of the building, the amount of ice remaining in the storage tank and the electric power consumed by the compressor and condenser.. The outside dry bulb temperature is also plotted for reference. In each graph it was observed that the temperature in the zones holds relatively constant at around 78°F between the hours of 8am and 6pm. After 6pm, the system goes into setback mode so that only heating is available. Because the outside dry bulb temperature is above the zone temperature and there is still a solar load on the building, the temperature in each zone drifts upward until around 8pm (hour 20 of the simulation). At this point, the outside dry bulb dips below the zone temperatures which also begin to fall since the sun has by now set. The zone temperatures then track with the outside temperature with an apparent lag which is due to the capacitance of the zone air.

Figure 2 shows operation in full storage mode so that no chiller power is consumed between the hours of 10am and 6pm. There is, however, a cooling load which starts between 7am and 8am and is met by the chiller. At 10am the chiller turns off and consumption of stored ice begins, the tank capacity decreases fairly linearly until 6pm when the system goes into setback. Figure 3 shows the same information for an ice tank operating in ice priority mode. In this figure, the chiller operates at half its maximum capacity throughout the on peak period and the rest of the cooling load is met with stored ice. Clearly, for this case the storage tank is very oversized since only one third of its capacity is used during the day. After 6pm the chiller begins operating at its maximum capacity to recharge the storage tank, completing the charging operation at approximately 11pm. Finally, Figure 4 is for the same system type in chiller priority mode. Since the chiller operates at maximum capacity throughout the day even less ice is used in this case than for ice priority control.

Figure 5 shows the effect of an inadequately sized ice tank storage system on the zone temperatures for an ice container system operating in full storage mode. Only stored ice can be used for cooling between 10am and 6pm and in this case the storage unit runs out of capacity at 4pm. The zone temperatures take a corresponding dramatic rise in response to the high outside air temperatures, the solar load, and the internal loads. Finally, Figure 6 represents a very undersized ice tank storage unit. In this case, the chiller is so small that it cannot meet the off peak load at 8am so ice must be used instead. Because of the earlier start time for ice consumption and the smaller tank capacity the ice is exhausted at around 1pm causing the zone temperatures to rise rapidly. Note that the chiller runs continuously during the off peak hours.

CONCLUSIONS

This paper has demonstrated the feasibility of simulating thermal storage systems within a large integrated building simulation program. Such a simulation gives users a valuable tool for evaluating system sizing requirements which is particularly critical for thermal storage systems. Systems which run out of storage too early result in uncomfortable zone conditions or use chillers during the most expensive period of the day with regard to energy consumption. The methods used in this paper can easily be adapted to other ice storage models, for example direct storage systems where ice is formed directly on the evaporator plates, or stratified chilled water storage tanks. Implementation of models for both these systems in IBLAST is currently in progress. Finally, additional work needs to be performed to develop optimal control strategies for thermal storage units within the context of an integrated simulation. This will require the development of predictive methods to determine the next day's storage requirements and allow the correct amount of ice or chilled water to be made during the off-peak hours and use the optimum combination of stored ice and chiller capacity for on-peak cooling.

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FIGURES

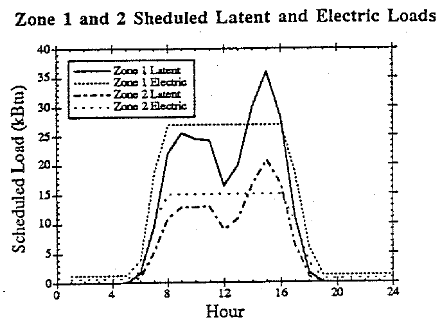

Figure 1: Test building zone 1 and 2 scheduled loads

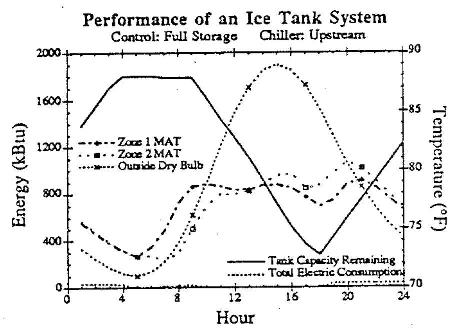

Figure 2: Ice tank system operating in full storage mode

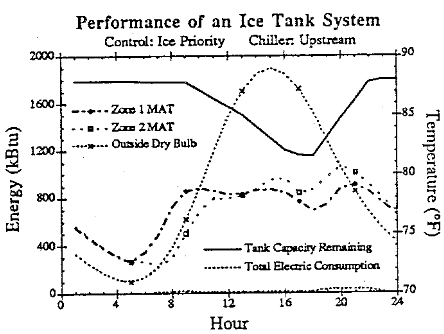

Figure 3: Ice tank system operating in ice priority mode

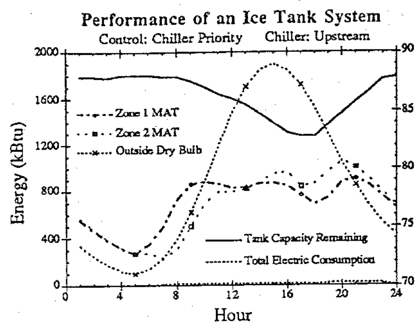

Figure 4: Ice tank operating in chiller priority mode

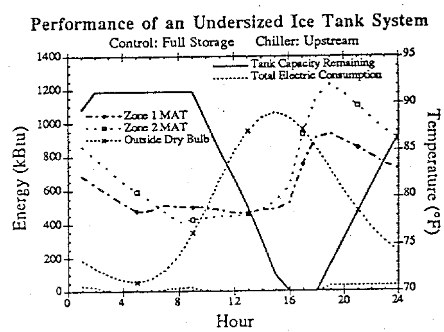

Figure 5: Undersized ice tank system operation

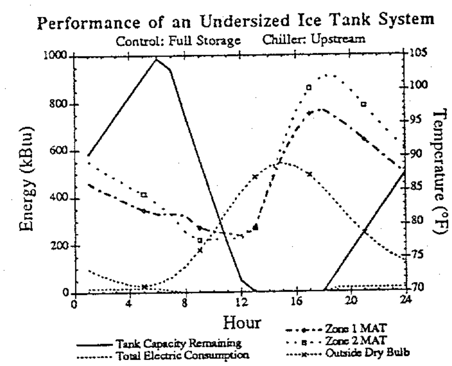

Figure 6: Very undersized ice tank system operation