OPTIMISATION OF MECHANICAL SYSTEMS IN AN INTEGRATED BUILDING ENERGY ANALYSIS PROGRAM: PART II: THERMAL STORAGE-BASED CENTRAL PLANT EQUIPMENT

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ABSTRACT

This is the second of two papers that describe the development of simulation methods for optimally controlled central plant equipment in IBLAST (Integrated Building Loads Analysis and System Thermodynamics). In Part II, the development and implementation of methods for simulating optimally controlled cold thermal storage in a building energy analysis program are described.

The goal of optimising a thermal storage system is to minimise the daily energy cost of operating the system. This means that the amount of ice built during off-peak operation, the quantity of ice used to provide cooling, and the time of use and replenishment must be determined. In this research, the optimal control problem was solved by discretising the allowable chiller part load fractions, only allowing a few to be considered at each time step. As a result, an approximate optimal solution could be found with a small number of search paths.

INTRODUCTION

Demand charges and time-of-day rates are the primary reason for the appeal of thermal storage systems since the chiller, which normally accounts for the largest fraction of the energy required to cool a building, is instead used to charge a storage tank during off-peak periods when utility rates are lowest. This also greatly reduces maximum electricity demand during on-peak periods, especially if the chiller is only used to fill a storage tank and not also to provide supplemental cooling during the day.

General criteria for sizing storage tanks and controlling thermal storage systems are still a matter of considerable debate within the thermal storage community. Evidence of this may be found in several of the papers in the bibliography which describe different control strategies. Thermal storage systems, and ice storage systems in particular, operate most efficiently when the amount of the stored medium produced is just enough to meet the next day's load. In BLAST, the system cooling load is already known by the time the plant is simulated so that optimal control, in this sense, is easy to simulate. In an integrated simulation, the simplest way to achieve the same effect is to run through the same 24 hour period several times updating the ice required each iteration. However, thermal storage systems frequently use a combination of storage and chiller capacity to meet on-peak loads. The relative fraction of on-peak ice storage to chiller use that also produces the lowest energy cost depends on the building load profile and also the utility rate structure. A method to determine the best combination of storage and mechanical cooling for a given building was the main focus of this research.

The principal requirement for optimising the use of thermal storage systems is to control the system so that it produces just enough ice during the off-peak hours to meet the following day's cooling loads. This is especially important for ice-on-coil systems since the production of ice reduces the performance of the refrigeration plant. However, since storage tanks are not perfectly insulated, all ice storage systems experience some heat leakage from the environment and additional ice must be produced to compensate for this. Therefore, the best control strategy is to start making ice at the latest possible time during the off-peak period while leaving enough time to make sufficient ice to meet the next day’s cooling load. Figure 1 shows the expected variation in tank capacity for a typical optimally controlled storage system.

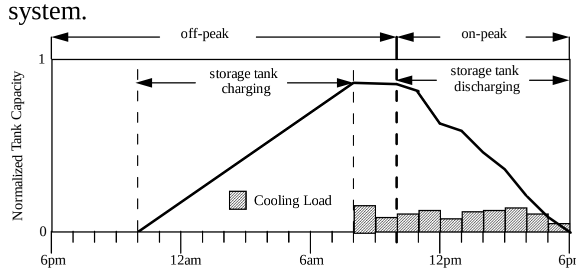

Figure 1: Charging and discharging of an optimally scheduled storage tank

The off-peak hours are from 6pm until 8am the following day. The hours from 8am to 10am are off-peak based on the utility rate and are not counted for the purpose of determining demand charges. In this example, the chiller operates to meet cooling loads during these hours instead of using ice from storage. The decrease in tank capacity between 8am and 10am is due to heat leakage into the storage tank. From 10am to 6pm, all cooling loads are met using stored ice; this is an example of a storage system using the full storage control strategy. The cooling load is also the fraction of tank capacity used during the indicated hour so that the total amount of cooling required by the system equals the initial tank capacity.

Figure 2 represents a storage system that is not scheduled optimally though it does not overproduce ice during the charging cycle. That is, at 6pm the storage tank has zero cooling capacity left. Clearly, this strategy consumes more energy than the one shown in Figure 1 since additional charging is required to "top-off" the storage tank due to the loss of ice that occurs between 4am and 8am because of heat leakage into the tank. The discharge process is, of course, identical to the previous case.

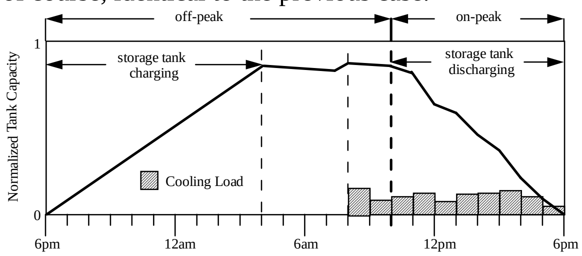

Figure 2: Charging and discharging of a non-optimally scheduled storage tank

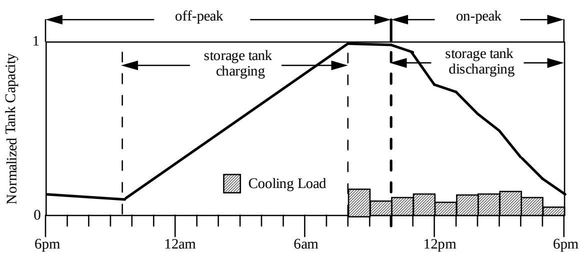

Figure 3: Operation of an optimally scheduled storage tank charged to full capacity each cycle

From Figure 3, it is clear that an optimally scheduled storage tank that is always charged to full capacity will consume more energy than a system only making enough ice to meet the next day's load. Again, additional chiller use is required to offset the loss of storage due to heat leakage through the tank walls

The efficiency of the charging process can vary as a function of the tank storage fraction, the ratio of the actual cooling capacity of the ice in the tank to the full storage capacity, especially in ice-on-coil ice storage systems. This effect is not shown on these figures and would cause even more energy consumption to occur than in the optimal case of Figure 1.

Finally, Figure 4 is the non-optimally scheduled system of Figure 2 that is charged to capacity each cycle. Unless the storage system performance is a function of the tank storage fraction, fully charging the tank each cycle does not incur any additional energy loss penalty due to leakage.

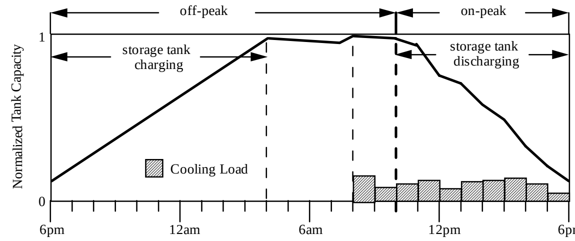

Figure 4: Operation of a non-optimally scheduled storage tank charged to full capacity each cycle

In summary, thermal storage systems have three potential benefits over systems without storage: 1) the bulk of the building cooling load is shifted to a period of the day when energy is at its cheapest, 2) the peak energy demand is reduced since the only energy expended to cool the building during on-peak hours is that required to circulate chilled water and air, and 3) the size of the chiller plant can be reduced. The chiller is also easier to size since this is based on the ability of the chiller plant to recharge the storage system with ice or chilled water during the off-peak hours.

METHOD

The ice storage models in the integrated building simulation IBLAST are limited to indirect ice storage systems and were originally developed for the BLAST program by Strand (1992). A previous paper by Taylor et al. (1994) describes the methods used to implement these models and operate them in an integrated building simulation.

In indirect systems the chiller is used to cool a secondary brine loop that circulates in the storage device where the ice is formed. The same brine flow may also be diverted through the coils when cooling loads exist and the chiller and stored ice may be used simultaneously. A common indirect ice storage system is the ice tank as illustrated in Figure 5. This system has a brine filled spiral wound coil inside a tank filled with water. The tank is charged by circulating refrigerated brine from the chiller through the 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 chilled brine is passed through the cooling coils. Relatively warm brine returns from the coils, entering the tank and causing the ice to melt.

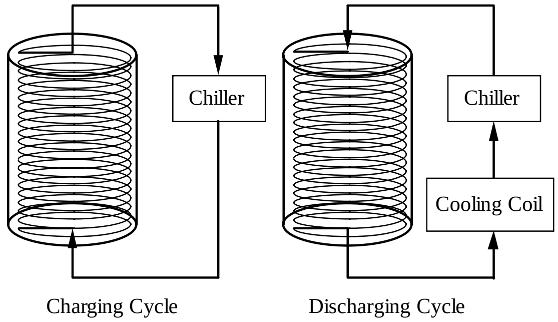

Figure 5: Simplified schematic of ice tank indirect storage system

The indirect ice storage control strategies are: full storage, ice priority and chiller priority. Full storage is defined the same way for indirect storage as for direct storage; all on-peak loads are met using ice. The ice priority control strategy uses ice until the load reaches a specified threshold with additional load being met by the chiller. Finally, the chiller priority strategy uses the chiller to meet cooling loads up to a pre-set level then additional load is met with ice.

The performance of the indirect ice storage models was simulated using curve fits that generate the normalised outlet temperature T_o^* according to a curve fit function of the form:

$$T_o^* = f_2\left(C_2, P_c, T_i^*, \dot{Q}^*\right) \tag{1}$$

where C_2 represents the coefficients of a polynomial curve fit to the performance data. The other parameters are the normalised inlet brine temperature, the normalised load on the storage tank $\dot{Q}^*$, and the ratio of the remaining storage unit capacity to its maximum capacity P_c. The normalised inlet and outlet temperatures required in this equation are calculated as shown in Equations 2(a and b):

$$T_o^* = \frac{T_o}{T_{freeze}} \tag{2a}$$

$$T_i^* = \frac{T_i}{T_{freeze}} \tag{2b}$$

The charging cycle of the ice tank requires the use of a second polynomial curve fit that returns the log mean temperature difference across the tank when the percentage of tank charged, normalised load, brine mass flow rate $\dot{m}$, and coefficients C_3 are specified as shown in Equation 3:

$$\Delta T_{lm}^* = f_3\left(C_3, P_c, \dot{Q}^*, \dot{m}^*\right) \tag{3}$$

Additional details on the development of the BLAST ice storage models and methods for determining the sets of polynomial coefficients from performance data can be found in Strand (1992) and the BLAST Users Manual (1993).

The thermal storage systems were to be optimised with the procedure described in Part I which was used for conventional central plant equipment. However, thermal storage systems have a characteristic that made direct implementation of this method impossible; the quantity of ice in the storage tank and teherefore tank performance is time dependent. Therefore, optimising the ice storage plant is not possible by simply piecing together a series of independent optimal combinations to get a complete optimal schedule. Each prospective optimal path must be simulated from beginning to end.

The control variable used to optimise the relative use of ice from storage and direct cooling from the compressor or chiller plant was the compressor part load fraction for each on-peak hour of the simulation. However, this value is continuously variable between 0 and 1 which means that infinitely many search paths exist from the beginning of the optimisation period to the end. The total number of paths was reduced by discretising the allowable variation in compressor part load fraction each time step to allow only a finite number of values to be considered. However, the number of paths that must be evaluated grows rapidly the finer the discretisation is i.e. as more compressor part load fractions are considered each hour. If there are six on-peak hours and two possible compressor part load fractions, the total number of paths is $2^6 = 64$; certainly a reasonable number. However, if the compressor part load fraction is allowed to take on 3 or 4 values then the total number of paths that must be calculated is more unreasonable: 729 or 4096 respectively. Figure 6 illustrates the growth in the number of possible chiller operating schedules when two chiller fractions are allowed each hour. Each path has a different associated cost and all paths must be searched to find the minimum cost. In the figure, the path resulting in COST 6 is shown as the optimal path.

On the other hand, the increase in the number of paths is accompanied by only a small a increase in accuracy of the optimal compressor schedule. Moreover, doubling the number of compressor operating states each hour increases the number of computations by a factor of 64, but only halves the uncertainty in the optimal path. In this context, accuracy implies the difference between the approximate optimal schedule calculated by this method and the true optimal schedule. Uncertainty, refers to the difference between the trial compressor part load fractions which is initially 0.34 when two part load ratios are allowed and 0.2 when four part load ratios are allowed. After one iteration, the uncertainties drop to 0.17 and 0.08, respectively.

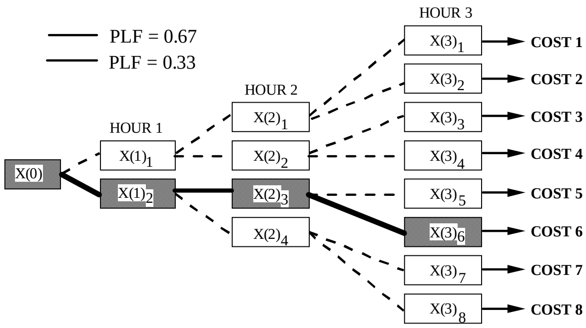

Figure 6: Growth of Storage Tank States With Time for Two Compressor Part Load Fractions

An alternative to using many compressor operating states each hour is to use as few states as possible and repeat the optimisation procedure several times to improve the accuracy of the solution. In each subsequent iteration, the interval between the allowable compressor part load fractions was narrowed while at the same time being centred on the previous iteration's optimal part load fraction for each hour.

This method yields an approximation to the global minimum energy cost path. An infinite number of compressor fractions, hence an infinite number of paths, would have to be considered for each time step of the simulation to obtain a true global minimum. However, the sensitivity of the solution to the number of compressor fractions allowed each hour should indicate whether the results obtained using 2 compressor part load fractions are reasonable. Therefore, the effects of using 2, 3, and 4 compressor operating states at each time step was evaluated assuming a four hour on-peak period to keep the total number of paths evaluated low. This resulted in 16, 81, and 256 paths respectively being generated. The compressor part load fractions used to initialise each test were: 0.33 and 0.67 for two allowable part load fractions per hour; 0.25, 0.5, and 0.75 for three; and finally, 0.2, 0.4, 0.6, and 0.8 for four. Figure 7 shows the results of these test cases after several days of iteration to ensure a converged solution (i.e. one with a small uncertainty). In this example, the approximate optimal paths, defined by hourly variations in chiller part load fraction, were within 5% of each other during each hour of the 4 hour on-peak period. Correspondingly, Figure 8 shows good agreement in the hourly change in storage tank capacity for each of the three cases. In conclusion, using only 2 paths for each hour and repeating the optimisation results in a reasonably good approximation to the global minimum cost path.

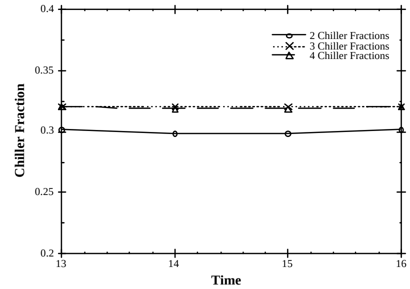

Figure 7: Comparison of Optimised Chiller Operating Fractions for Simulations Using 2, 3,, and 4 Chiller Fractions Per Time Step

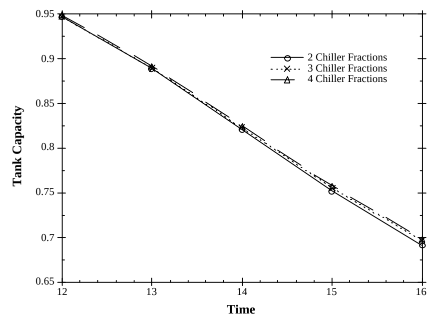

Figure 8: Comparison of Normalised Ice Storage Tank Capacity for Simulations Using 2, 3, and 4 Chiller Fractions Per Time Step

Finding the optimal compressor operating schedule through the on-peak hours required calculation of the total energy cost for that period of time. Selection of the optimal path was accomplished by direct comparison of the costs of each path. However, there are no direct costs associated with the consumption of ice from the storage tank because the energy required to build the ice was consumed at some prior time when, presumably, the cost of energy was cheaper. A way of including the cost of ice used to compute the total cost of each path was required since the ice is clearly not free. The average cost of ice per kilowatt-hour used was calculated by determining the amount of energy required to build ice off-peak and dividing the resultant cost of that energy by the total amount of ice built. This yields a direct cost for on-peak ice consumption that can be added to the cost of compressor electricity consumption, when ice is consumed, to generate a total energy cost for each path. Finally. the peak energy consumption for each path is determined in order to determine the correct demand charge to be added to the total path cost.

RESULTS

This section illustrates how the optimal use of cold thermal storage is affected by four variables: on-peak electricity cost, demand charge, chilling plant capacity, and storage tank capacity. The following examples show the best way to operate the chilling plant compressor by optimising its part load ratio hour-by-hour under a variety of energy cost and equipment size combinations. One immediate observation seen in all the examples, was an essentially constant compressor part load fraction over the optimisation period. This result reflects the fact that the optimal way for a compressor to provide a fixed amount of cooling is for it to operate at a constant part load fraction. This is true, even if that fraction is not the compressor's optimal part load fraction, because a deviation from a constant part load ratio might improve efficiency at one hour but would reduce it at another enough to offset the initial improvement. Thus, in the majority of the cases presented below, cold thermal storage is used to level the on-peak load on the compressor. In all the examples, the ice storage plant was connected to a VAV system serving a simple building with two thermal zones.

The first example shows the effects of varying the on-peak electricity cost without a demand charge. The thermal storage system consisted of a 440kWh storage tank and a 44kW compressor to charge the tank and meet other cooling loads. The system was simulated and optimised for four values of on-peak electricity rate multiplier: 1, 2, 4, and 10. The results are presented in Figures 9 and 10. Figure 9 shows the hour-by-hour fractional reduction in tank capacity as the on-peak rate multiplier is increased. As might be expected, as the cost of on-peak electricity increased, more of the cooling load was shifted to the storage tank from the compressor. This is seen directly from a corresponding reduction in compressor part load fraction. However, the transition was gradual, indicating that the cost of ice per kilowatt-hour of cooling was not constant, but increased as more ice was built. In addition, the optimisation algorithm seemed to find a local minimum, at small compressor part load fractions, where a decrease in the

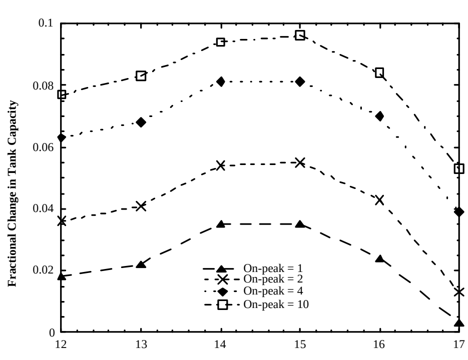

Figure 9: Hourly Variation in Consumption of Stored Ice for Four Values of On-Peak Rate Multiplier

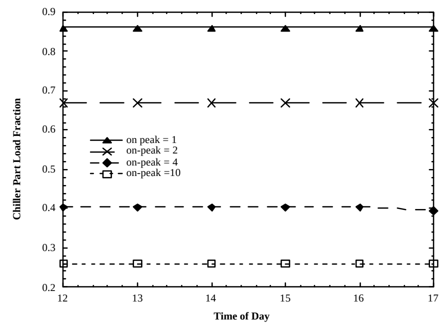

Figure 10: Hourly Variation in Chiller Part Load Ratio for Four Values of On-Peak Rate Multiplier

part load ratio resulted in an increase in compressor energy consumption. This, in combination with an increase in the stored energy required, drove the optimiser in the wrong direction. This problem could be overcome by simulating, as a separate case, the ice storage system using "full storage" control. Comparison of the two sets of results would determine the schedule giving the true global minimum energy cost.

Figures 11 and 12 show results obtained from the same central plant as the previous ice storage case, when the demand charge was varied and the on-peak rate multiplier was given a constant value of 2. Increasing the demand charge had similar effect to increasing the on-peak electricity rate; as demand charge increased a greater emphasis was placed on ice consumption rather than direct cooling.

A third example shows what effect a change in compressor capacity has on the optimal compressor part load fraction. Three compressor sizes, 66kW, 44kW, and 29kW, were selected, the on-peak rate multiplier was 2.0, and the demand charge was 0.0. The results are shown in Figure 13.

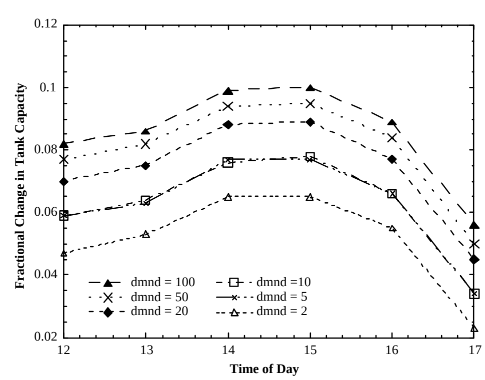

Figure 11: Hourly Variation in Consumption of Stored Ice for Six Values of Electricity Demand Charge

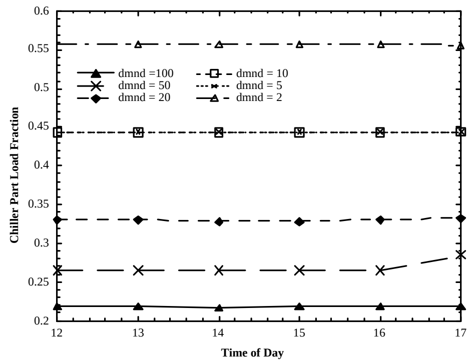

Figure 12: Hourly Variation in Chiller Part Load Ratio for Six Values of Electricity Demand Charge

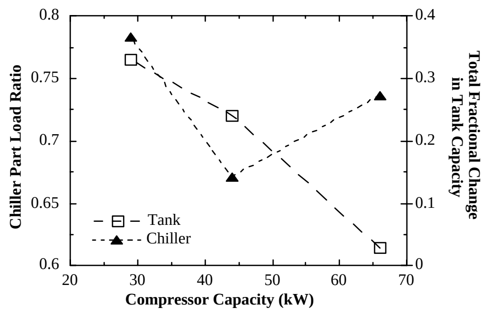

Figure 13: Effect of Compressor Capacity on Optimal Chiller Part Load Ratio and Fraction of Stored Ice Consumed (Tank Capacity = 440kWh, On-Peak Rate = 2.0/kWh)

Increasing the compressor capacity first results in a decrease in the optimal part load ratio, but for the 66kW compressor the optimal part load ratio is higher. Compressor energy consumption, shown in Figure 14 increased with each increase in compressor capacity. However, the plant with the 66kW compressor also consumed no ice. The changes in the amount of ice remaining in the tank were due to thermal gains from the outside environment into the storage tank. In this case, the compressor was large enough to meet all cooling loads so no ice was consumed. However, in the other cases, some ice had to be used off-peak as well as on-peak because the chiller was too small to meet the peak cooling loads. A final observation from this example, was that total on-peak energy cost was almost invariant for each of the three cases. Presumably, the lack of a demand charge was the principle reason for this as peak electrical consumption would be much higher for the 66kW compressor example than the other two examples, shown in Figure 14. This figure also shows opposite trends in compressor and storage tank energy consumption.

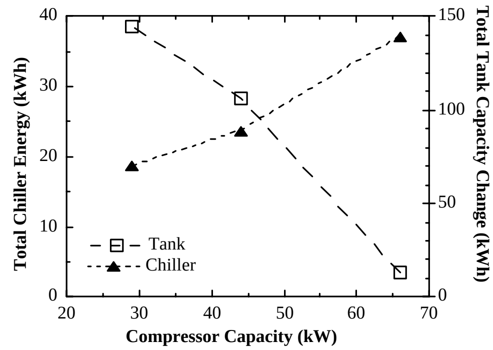

Figure 14 Effect of Compressor Capacity on Total Chiller Energy Consumption and Total On-Peak Consumption of Stored Ice (Tank Capacity = 440kWh, On-Peak Rate = 2.0/kWh)

In one final example, the compressor size was fixed at 44kW and the storage tank capacity varied from 307kWh to 613kWh. The on-peak rate multiplier was 1.5 and there was a demand charge of 2/kW. Figure 15 shows how the optimal chiller part load ratio and the fractional change in tank capacity were affected. As tank capacity was increased, the fraction of that capacity used to meet the on-peak loads decreased because the total amount of ice available in the tank was greater. The behaviour of the chiller part load ratio can be explained by noting that with a capacity of 300 kWh the storage tank was almost depleted by the end of hour 17. Therefore, the compressor had to be used to supplement the storage tank so that ice would be available through the end of the on-peak hours. Figure 16 provides the same information as Figure 15 in terms of compressor and storage tank energy consumption. Chiller energy consumption is relatively constant because of the insensitivity of this variable to part load ratio at small loads. The total change in tank capacity, which was limited by the storage tank capacity for the smallest storage tank, was highest for the 440kwh tank, and then decreased slightly for the largest storage tank. The total energy cost for the largest tank was 123.6 units, 122.0 units for the medium sized tank, and 122.5 for the smallest tank. The larger tank had higher thermal losses, while the smaller tank used more chiller energy than optimal, accounting for the small variation in total on-peak energy cost.

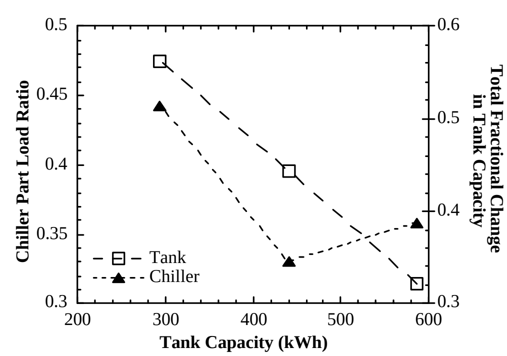

Figure 15: Effect of Storage Tank Capacity on Optimal Chiller Part Load Ratio and Fraction of Stored Ice Consumed (Compressor Capacity = 38kW, On-Peak Electric Rate Multiplier = 1.5, Demand Charge = 2.0/kW)

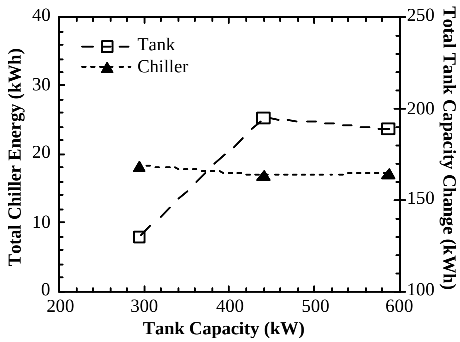

Figure 16: Effect of Storage Tank Capacity on Total Chiller Energy Consumption and Total On-Peak Consumption of Stored Ice (Compressor Capacity = 38kW, On-Peak Electric Rate Multiplier = 1.5, Demand Charge = 2.0/kW)

CONCLUSIONS

The objective of this work was to develop a method for optimally scheduling the use of thermal storage systems so as to minimise daily energy cost. The selected method assumes that a single constant cost per unit of ice removed from storage could be used to calculate the cost of using ice compared to the cost of direct cooling. In addition, in order to determine the lowest cost schedule, defined by the compressor part load ratio for each on-peak hour, all the possible schedules had to be considered to ensure that a global minimum was found. However, this would have resulted in an unwieldy computational problem so only two compressor part load fractions were allowed in each hour of the simulation. Subsequently, an iteration was performed to improve the accuracy of the solution. This method proved quite tractable from a computational point of view, as many one day iterations could be performed for minimal additional simulation time when the number of allowable compressor states is small for each hour. Use of additional compressor states each hour did not appreciably change the optimal schedule.

The results of the optimisation showed that increasing the on-peak cost of electricity, the demand charge, or a combination of the two caused the thermal storage system to favour the use of ice instead of direct cooling. Somewhat surprisingly, the optimised compressor part load fraction was constant for every hour of the interval being optimised. This indicates that the best efficiency can be obtained from mechanical cooling when it operates to meet a constant load. In that case, the melting of ice accommodates variations in the load. In other words, the compressor is base loaded at an efficient part load fraction, and ice is used to reduce the amount of on-peak energy consumption and reduce maximum on-peak demand. This makes sense since, for a fixed amount of cooling supplied by the compressor, a compressor part load schedule that was not constant would have a higher peak energy consumption and would spend periods of time operating at a less efficient setting. In future, it should only be necessary to optimise on a single compressor part load fraction which would then be applied to all the on-peak hours. A final observation from the use of this method is that obtaining the compressor part load fraction resulting in the true global minimum energy cost requires an additional simulation that assumes ice is used to meet all the on-peak loads. The results of this simulation must be compared to the optimised solution to find the best schedule. This step is necessary because, with the current compressor simulation in IBLAST, the optimiser got stuck in a local minimum as the on-peak cost of electricity or the demand charge got large. The predicted optimal compressor part load fraction for each hour was therefore small, but non-zero, which is not correct.

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