CASE STUDY OF STRATIFIED CHILLED WATER STORAGE UTILIZATION FOR COMFORT AND PROCESS COOLING IN A HOT, HUMID CLIMATE

William P. Bahnfleth, Ph.D., P.E.
Assistant Professor
Department of Architectural Engineering
The Pennsylvania State University
University Park, PA

Amy Musser
Graduate Student
Department of Architectural Engineering
The Pennsylvania State University
University Park, PA

ABSTRACT

The advantages of thermal storage are enhanced in hot and humid climates. Year-round cooling loads increase thermal storage operating cost savings. The absence of a long winter during which major maintenance tasks can be accomplished without compromising system reliability increases the importance of thermal storage as back-up capacity. In an industrial setting, operating cost savings due to thermal storage go directly to the bottom line of a manufacturing process and the avoidance of lost production due to process cooling outages can save millions of dollars per year. This paper presents a case study of chilled water storage use at the campus of a major US electronics manufacturer located in Dallas, TX. An overview of the system and its operation is followed by presentation of operating data taken during 1997.

INTRODUCTION

Chilled water thermal energy storage (TES) in naturally stratified tanks has been shown to be a valuable central cooling plant load management technology in climates as diverse as those of upstate New York (Bahnfleth and Joyce 1994) and central Texas (Fiorino 1994a). Both of these systems have produced substantial operating cost savings for their owners, increased the energy efficiency of chilled water production, and have made plant operation simpler and more flexible.

A typical stratified chilled-water storage installation has a volume in excess of 1 million gallons (3.8 million L) and a peak output of more than 3,000 tons (10,550 kW). The unit cost of such systems is typically less than $90/ton-hr ($26/kWh), which is more than twice the unit cost of typical ice storage systems (Potter 1994). The cost of chilled water thermal storage capacity frequently is lower than the cost of equivalent conventional refrigeration plant capacity. Consequently, chilled water storage is not threatened by deregulation of the US electric industry and the associated disappearance of incentive programs to the same extent as other thermal storage technologies. Under certain circumstances, chilled water storage may be economically feasible even in the absence of electric demand charges and time-of-use energy charges (Caldwell and Bahnfleth 1997).

Applied in a hot and humid climate, thermal storage stands to yield even greater benefits than it does in temperate regions for a variety of reasons. Not the least of these is the simple fact that mechanical cooling loads in such climates typically exist year-round. Consequently, storage is likely to be utilized for more hours per year. A second important factor is the influence of humidity on cooling loads. Thermal storage is a cost-effective technology for helping to meet high latent loads and is especially beneficial when the thermal storage supply temperature is lower than common practice levels.

This paper describes the performance of a stratified chilled water storage system serving an office/manufacturing complex in a hot, humid region of the United States and presents quantitative information regarding its performance on the basis of several months of typical operating data. Performance characteristics summarized include charge and discharge cycle flow rate distributions, thermocline thickness and lost capacity as a function of flow rate, and thermal efficiency. This case study illustrates the utilization pattern of chilled water storage in a hot and humid environment and presents new data on the performance of a large stratified chilled water storage tank.

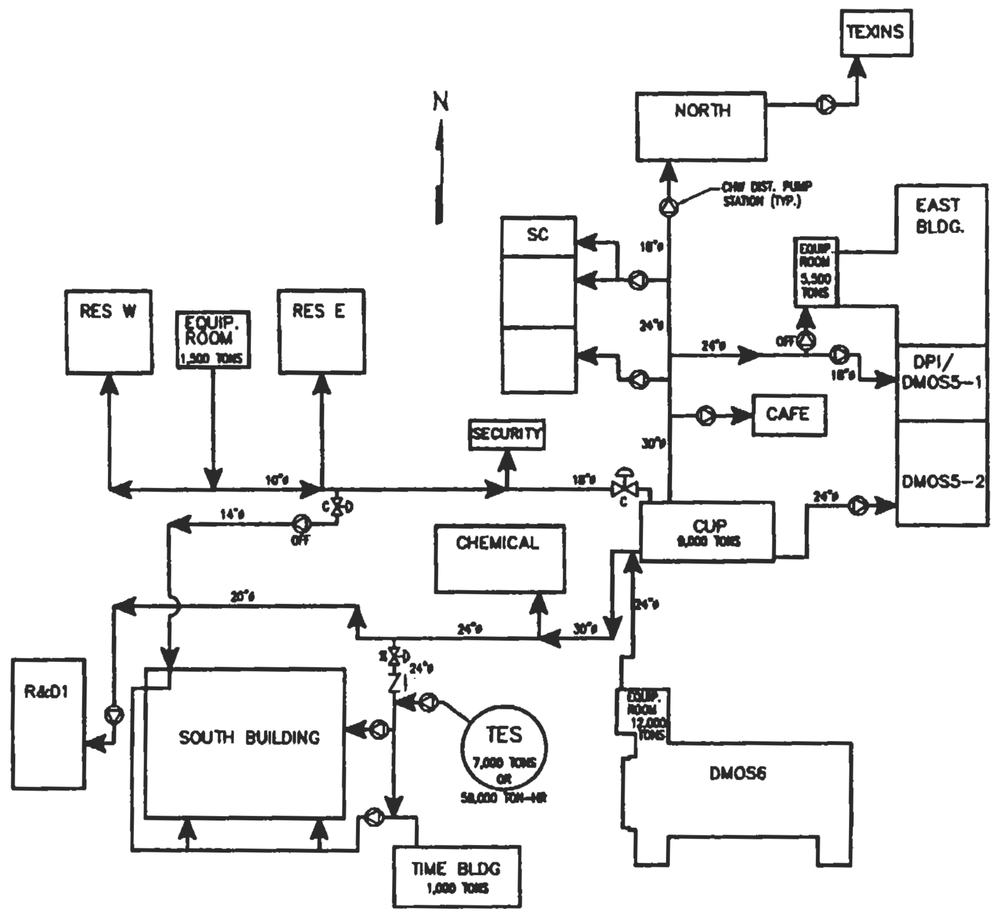

Figure 1. Plant Schematic.

SITE

The case study site is the Dallas, TX world headquarters of a major semiconductor manufacturer. The 6 million-plus square foot (557,400 m^2) complex includes office spaces as well as semiconductor development and manufacturing facilities. Buildings range from new to more than 30 years old. Air-conditioning loads exist year-round and include large ventilation air quantities, which must be cooled and dehumidified, and large sensible process loads. The peak load currently stands in excess of 33,000 tons (116,160 kW). These loads are served by a distributed district chilled water system with twenty-four chillers totaling more than 37,500 tons (132,000 kW) in capacity and the chilled water storage system. The system is being improved continuously. Recent changes include measures to increase chilled water temperature differential and reduce pumping energy. A detailed description of the site and the ongoing development of its cooling systems may be found in a recent publication (Fiorino 1994b). A schematic of the chilled water system is shown in Figure 1.

THERMAL STORAGE SYSTEM

The chilled water storage system was built during 1993 and 1994 at a cost (before utility rebate) of $7 million. The addition of chilled water storage avoided $4.25 million in conventional chiller plant addition and has reduced net operating expenses for the site by approximately $1.5 million per year.

The thermal storage tank is a fully buried cylindrical, pre-cast, pre-stressed tank with four-ring single pipe octagonal diffusers. It holds 5.2 million gallons (19.7 million L) of water, and is 140 ft (42.7 m) in diameter and has a water depth of 45 feet (13.7 m). The design charge and discharge rates of the system are 5,800 and 9,000 gpm (366 and 568 L/s) respectively. Typical charging temperature is approximately 42°F (5.6°C), and the system return temperature is roughly 57°F (13.9). At these temperatures, the nominal tank storage capacity is 60,500 ton-hr (213,000 kWh) and the instantaneous output at the maximum discharge flow rate is 5,600 tons (19,700 kW). The tank can discharge at this rate for nearly 11 hours. At design flow rates, the inlet Froude number of the diffusers is 0.35 or less, and the inlet Reynolds number varies from 1,500 for the inner ring to 400 for the outer ring.

The system operates in a storage priority, partial storage operating mode from June 1 through September 30 during on-peak hours that run from noon to 8 p.m. Controls are configured to obtain maximum efficiency from refrigerant equipment on-line by maintaining each on-line chiller at its peak capacity. During the on-peak period the tank is discharged to limit peak site electrical demand to 86 MW. Implementation of this strategy during August 1997 reduced on-peak demand by 6.7 MW, saving the owner $60,970 in demand charges.

Existing instrumentation was used to monitor the flow and temperatures within the tank. Temperature in the tank and its supply lines is measured by thermocouple transmitters, specified by the manufacturer as accurate to ±0.25°F (±0.14°C). The installed flow meters are uni-directional insertion-type paddle wheel meters. Their accuracy is specified as 1% of scale over an 8,000 gpm (505 L/s) scale, therefore an accuracy of 1.6% of design flow can be expected when the flow meter is installed and calibrated in accordance with manufacturer's specifications.

Perfect installation and precise calibration of flow meters is not the norm in real installations. Consequently, flow meters in the field generally do not approach the best-case performance described in the manufacturer's literature. The authors have observed flow measurements obtained from a high quality instrument to differ from the actual flow rate by as much as 30% (Musser and Bahnfleth 1998). In order to analyze the performance of these systems, Musser and Bahnfleth developed a method for correcting the readings of installed flow meters using the observed transit time of a thermocline moving between two sensors located inside the tank during constant flow rate tests.

Flow measurement based on thermocline transit time was found to be very accurate in this case. Uncertainty in this technique is due to the combination of uncertainties in sensor location, time increment, and temperature reading at the two sensors involved. Using conservative estimates for component uncertainties, the uncertainty was calculated to be ±2.5% or less. Comparison of thermocline transit time flow rate measurements with uncorrected flow meter readings indicated that the meter in the present case was in error by roughly ±10%. Therefore, the transit time procedure was used to correct the readings of the existing meter in reduction of the data presented in this paper.

ANALYSIS OF THERMAL STORAGE UTILIZATION

The performance of the tank was monitored for two periods during the summer of 1997: July 21 through August 12 and September 8 through October 1. During this period, the tank was discharged 45 times and charged 44 times. Inlet flow rate and temperatures at the tank inlet, outlet and at two foot (0.61 m) vertical intervals inside the tank were recorded at one minute time intervals throughout the monitoring period.

Typical Operating Strategies

The relatively long period of observation permits description of typical operating practice and calculation of long term efficiency of thermal storage in this application. As noted above, the basic operating strategy employed at this site is storage priority, which endeavors to make maximum use of the tank during each operating cycle, but the implementation of this strategy is conservative. It is typical practice to charge the tank fully each day in order to have the maximum possible capacity available. However, the tank is rarely discharged completely. Of the 45 discharge cycles monitored, the supply chilled water level dropped below 6 feet (1.8 m, 13% of tank volume) only twice. By comparison, the supply chilled water level rose above 39 feet (11.9 m, 87% of tank volume) at the completion of 41 of the 44 charge cycles monitored. This leaves a substantial emergency reserve to provide additional load shifting in case of an unexpected increase in cooling demand late in the on-peak period or to cover an unscheduled chiller outage. This practice has the additional benefit of preserving the thermocline through many cycles, which tends to increase thermal efficiency.

Figure 2 shows typical sequential charge and discharge data taken over a three day period in late September 1997. In each of the three charge cycles, the tank is initially charged at a high flow rate, then the flow rate is reduced for the remainder of the process. The step change in charging rate after several hours suggests the use of two chillers for charging, and a subsequent reduction to only one. The three discharge cycles vary in average flow rate and in pattern as storage follows the load.

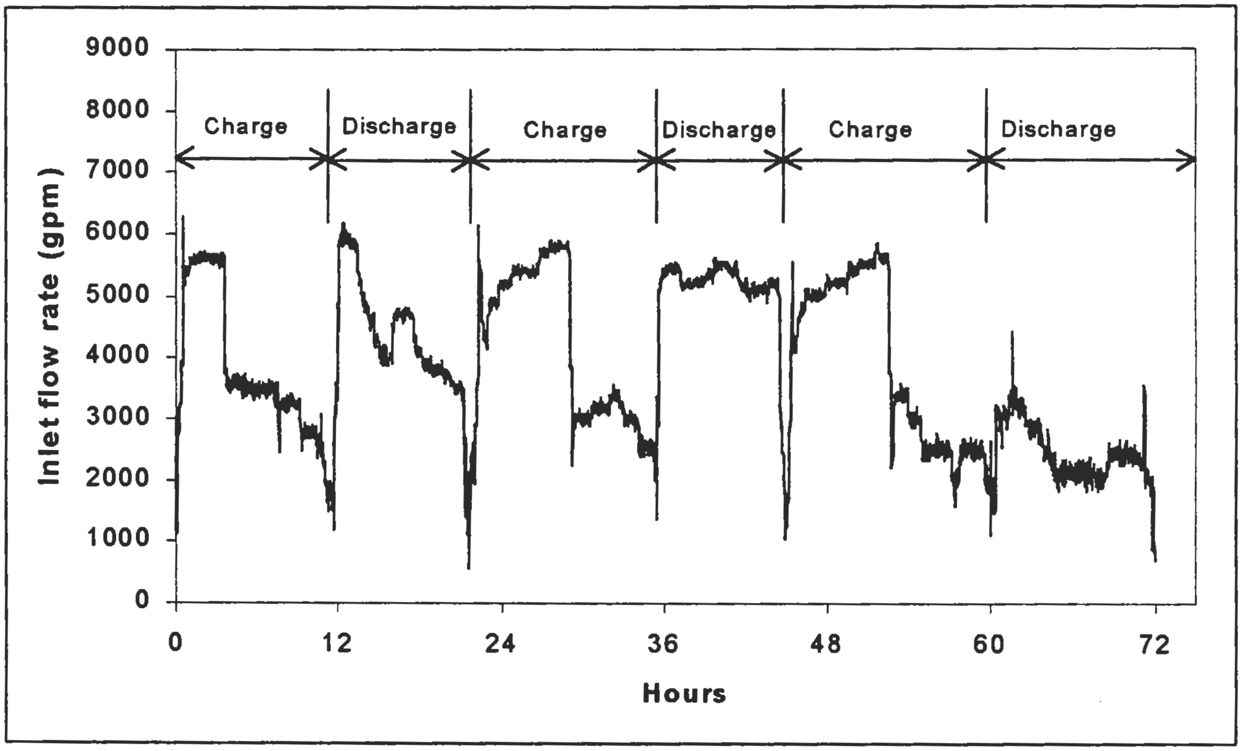

Figure 2. Typical inlet flow profile. Three days are shown, beginning at midnight on the first day

Figures 3 and 4 are flow rate duration curves for, respectively, the charge and discharge modes over the entire interval monitored. This interval contained roughly 420 hours in discharge mode and 640 hours in charge mode. It is evident that a significant percentage of operation occurred at flow rates much lower than the design charge and discharge flow rates of 5,800 gpm (366 L/s) and 9,000 gpm (568 L/s), respectively. This point is made even more clearly by Figures 5 and 6, which are flow rate frequency distributions for the tank during charge and discharge expressed in terms of percentage of design flow rates. Figure 4 shows that the tank was discharged at 80% of design flow or more during only 15% of its operating hours. The tank was discharged at between 20 and 40% of design flow for 35% of its operating hours. Operation at less than 60% of design discharge flow rate occurred during more than 80% of the hours during the observation period. As shown in Figure 6, the tank charged at flow rates closer to design, yet nearly half of its charge mode operation was at flow rates lower than 60% of design charging flow rate. These observations are significant because stratified chilled water storage tank diffusers are designed to provide acceptable stratification at design conditions, typically the highest inlet flow rate expected. Operation at lower flow rates should result in some degree of performance improvement. Similar utilization patterns, having many low flow rate hours, have been observed at other sites (Bahnfleth and Musser 1997).

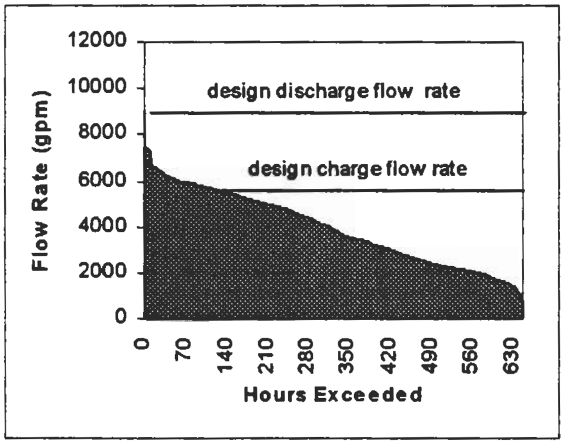

Figure 3. Charge process flow duration curve.

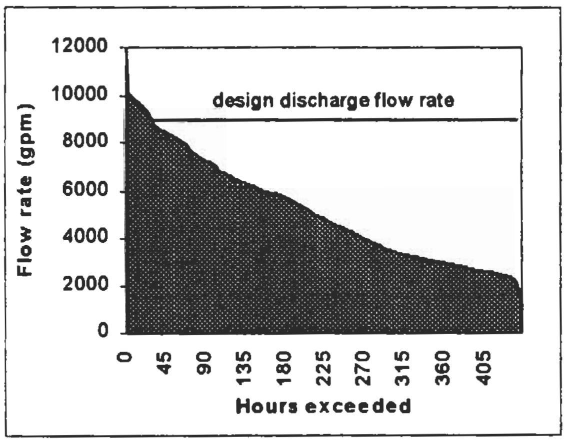

Figure 4. Discharge process flow duration curve.

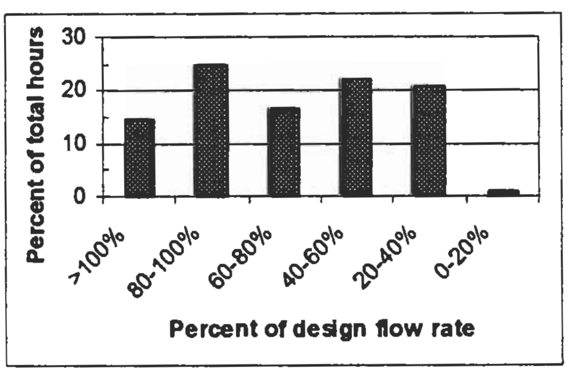

Figure 5. Charge flow rate frequency distribution.

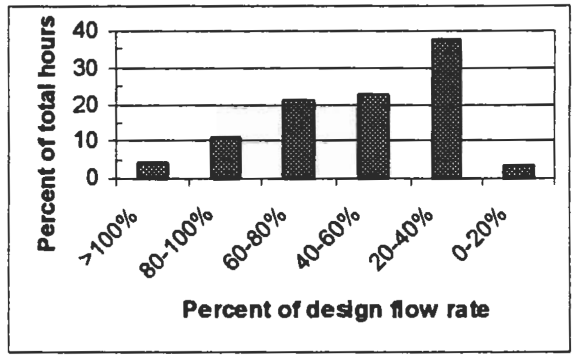

Figure 6. Discharge flow rate frequency distribution.

Thermal Efficiency

Data collected under normal operation allows a long term, or "seasonal" thermal storage efficiency to be calculated. This is done by dividing the capacity discharged by the capacity charged over a period of several consecutive cycles. Averaging of a large number of cycles increases the accuracy of the estimate because it reduces the error resulting from differences between the initial and final states of the storage tank. For the case study tank, long-term efficiency was calculated for the thirteen day interval between July 21 and August 2 and for the 22 day interval from September 9 to 30. In both cases, the thermal efficiency was calculated to be a very high 97.7 %.

Thermocline Thickness

A thermocline is the thin thermal transition region that separates the warm water in the upper portion of a stratified tank from the cool water in the lower portion. Diffuser designs that cause more mixing of warm and cool water produce thicker thermoclines. Thermocline thickness is a widely accepted quantitative measure of stratification effectiveness and inlet diffuser performance.

A definition of thermocline thickness proposed by the authors (Musser and Bahnfleth 1998) is adopted in this paper. It is based on variation of a dimensionless temperature difference Θ:

$$\Theta = \frac{T - T_c}{T_h - T_c} \qquad (1)$$

where T is the temperature at an arbitrary vertical position in the tank and T_h and T_c are reference warm and cool ambient temperature values. In a discharge cycle, T_h might be interpreted as the warm tank inlet temperature (system return) while T_c

would be the average temperature of cool water in the tank at the beginning of the cycle. Clearly, Θ varies from a value of zero at the cool end of the temperature profile to 1 at the warm end.

For practical purposes, thermocline thickness can be defined as the thickness of a region symmetric about the average temperature (Θ = 0.5) over which Θ changes by some arbitrary fraction of the difference between T_h and T_c. The distance over which Θ varies from 0.1 to 0.9 (i.e., 80% of the maximum temperature difference) includes most of this transition region without becoming overly sensitive to small fluctuations in measured cutoff Θ values near 0 or 1 (Musser and Bahnfleth 1998). The application of this definition to data from a case study tank is shown in Figure 7.

To observe the effects of inlet conditions, thermocline thickness should be measured near the inlet diffuser in a tank that is initially isothermal. In this study, it was not possible to perform complete controlled flow rate tests because of constraints on plant operation. Because it is normal practice at the site not to fully discharge the tank, it was not possible to obtain temperature profiles generated by the lower diffuser at the start of a charge cycle. However, since the tank typically is charged completely and a new thermocline is generated at the start of the discharge cycle, it was possible to obtain estimates of thermocline thickness formed by the upper diffuser at the beginning of discharge.

Temperature profiles generated during discharge cycles were observed at a sensor 36 ft (11.0 m) above the tank floor using a time series method for reduction of data from tanks with sparse vertical temperature sensors described previously by Musser and Bahnfleth (1998). This location chosen was near the inlet diffuser, but sufficiently far removed from it that the thermocline was observed not to change shape significantly after passing it. In thirteen of the recorded discharge cycles, a nearly constant flow rate was maintained until after the thermocline had moved below the 36 ft (11.0 m) sensor. Temperature in the tank was nearly uniform at the start of these cycles and inlet temperature was essentially constant over the period during which the thermocline was generated.

Figure 8 shows calculated thermocline thickness plotted as a function of percent of design discharge flow rate. For these cycles, the inlet flow rate varied from roughly 25% to 110% of design. Thermocline thickness was relatively insensitive to inlet flow rate for flows less than 80% of design. Most thermoclines in this range of flow rate were slightly less than 2 ft (0.61 m) thick (4.4% of the overall tank height). Based on the discharge flow rate frequency distribution shown in Figure 5, thermoclines of this thickness would be expected to occur 85% of the time. At higher flow rates, slightly thicker thermoclines were formed. In the worst case, the thickness of the thermocline was measured to be 3.2 ft (0.98 m), about 7% of the overall tank height.

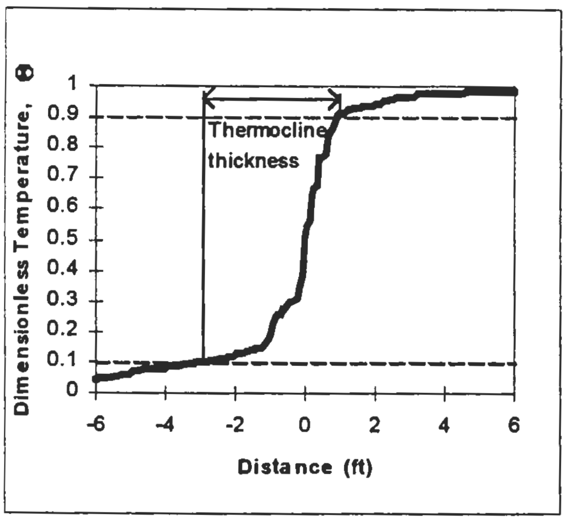

Figure 7. Definition of thermocline thickness.

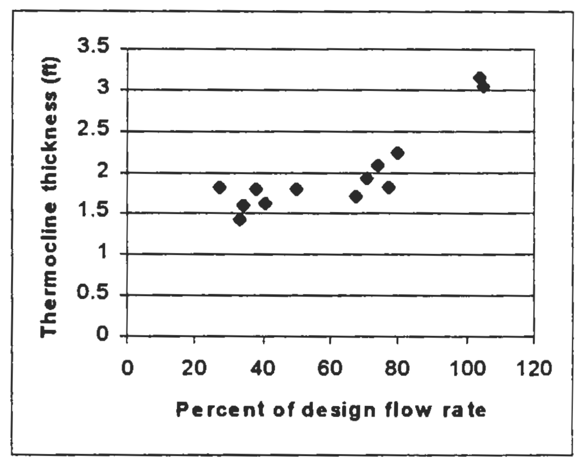

Figure 8. Thermocline thicknesses for thirteen discharge cycles.

The thermocline thicknesses generated by the octagonal diffusers in this tank compare very favorably with those generated by the upper diffusers of two other tanks studied by the authors (Bahnfleth and Musser 1997). In the referenced study, a tank with a single ring octagonal diffuser with design Froude and inlet Reynolds numbers of 1.0 and 4,400, respectively, generated a 12 ft (3.7 m) thick thermocline at roughly 70% of its design discharge flow rate. At approximately 80% design flow rate, a tank with radial diffusers having design Froude and inlet Reynolds numbers of 0.35 and 6,200, respectively, produced thermoclines roughly 3.5 ft (1.1 m) thick during charge cycles, and 6 ft thick during discharge cycles.

Lost Capacity

The thermal performance of entire cycles has been quantified using the Figure of Merit (Tran, et al. 1989). Figure of Merit (FoM) can be calculated for cycles in which a complete charge process is followed by a complete discharge process. The integrated discharge capacity is then calculated as a fraction of the ideal tank capacity, or the capacity represented by one entire tank volume of cool water.

Two major difficulties arise when calculating Figure of Merit. First, the tank must be fully charged and subsequently fully discharged. Measurement uncertainty is also an issue when calculating Figure of Merit. The first of these problems can be overcome by defining a Figure of Merit applicable to half cycles (a single, complete charge or discharge process) has been proposed (Bahnfleth and Musser 1998), however, this still requires the tank to be either charged or discharged completely, which is not done in the normal operation of this tank. In previous studies, Tran et al. (1989) estimated FoM uncertainties of ±7% and Wildin and Truman (1985) estimated uncertainties of ±5%. The magnitude of these uncertainties is most directly affected by uncertainties in flow measurement. Since differences in the thermal performance of many tanks are within this range, it can be difficult to assess factors relating to thermal performance.

Figure of Merit is defined in terms of integrated capacity, which is the capacity delivered or stored. An alternative approach is to evaluate the capacity *lost* to mixing and conduction during the course of a cycle. The lost capacity associated with a cycle is that capacity which cannot be removed from the tank due to an outlet temperature limitation. It is "lost" to the cycle because it is unavailable at a temperature that can be utilized by the process served by the system. The shaded regions in Figure 9 illustrate lost capacity for a hypothetical storage tank temperature profile and limiting outlet temperature, T_{limit}.

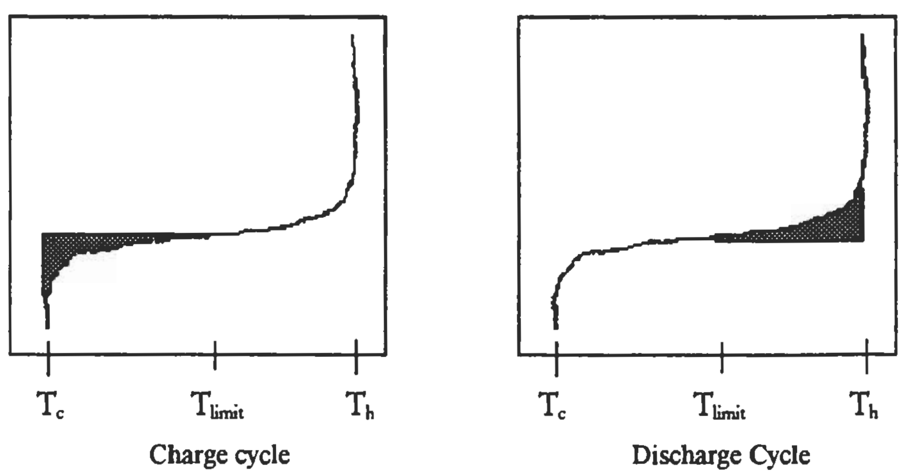

Figure 9. Illustration of Lost Capacity.

Lost capacity is described mathematically by an integral taken from the edge of the thermocline to the point within the thermocline with temperature T_{limit} (Bahnfleth and Musser 1998). This calculation can be done at any point in the cycle, using a temperature profile measured at any point inside the tank. The lost capacity calculated using a profile measured near the tank inlet will account only for capacity lost due to inlet mixing. Lost capacity calculated at a temperature sensor located near the tank outlet will account for the effects of both inlet mixing and conduction over the remainder of the cycle. Typically, the two values will be very close to one another, since the effects of conduction throughout the length of a single cycle are small for typical full-scale tanks (Homan et al. 1996).

Because calculation of lost capacity involves only the thermocline region, rather than the entire tank volume, it is based on fewer flow measurements, and the error (in absolute terms) is reduced relative to a Figure of Merit calculation (Bahnfleth and Musser 1998). Another advantage particular to the current study is that thermal performance can be quantified without discharging the tank completely. The predicted lost capacity for the cycle will be in error only to the extent that conduction alters the temperature profile during the course of the cycle.

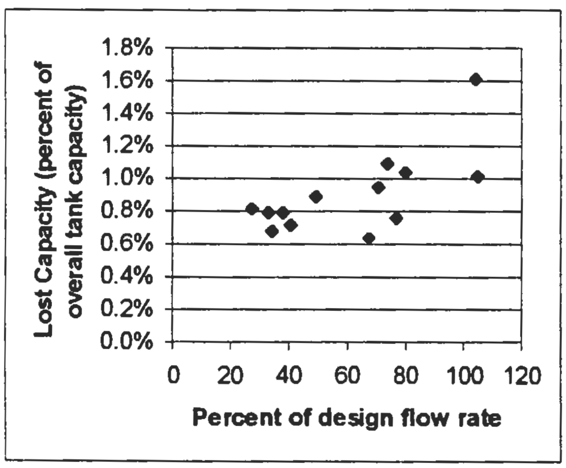

Figure 10. Lost Capacity associated with thirteen discharge cycles.

Figure 10 shows the lost capacity calculated for the thirteen discharge processes analyzed. For this analysis, the cut-off temperature was assumed to be the midpoint of the warm and cool temperatures in the tank (Θ=0.5). Trends in Lost Capacity tend to track trends in thermocline thickness. Again, there is little difference in lost capacity for profiles generated at less than 80% of the design flow rate. Lost capacity is increased for one of the higher flow rate cases, but not the other. This is unexpected since the thermocline thicknesses for both profiles were similar. However, inspection of the temperature profiles for the two cases showed that their shapes differed, with one having a steeper central temperature gradient. No conclusive explanation for this result could be identified, and more testing is needed to establish clear trends. However, the significant conclusion to be reached is that, in all cases, lost capacity represented less than 1.6% of the overall tank capacity.

An uncertainty analysis of calculated lost capacities was performed using techniques described by Doeblin (1975). Uncertainty in lost capacity depends on uncertainties in the measurement of flow, temperature, and the time interval at which readings are taken. For the cases plotted in Figure 10, the maximum uncertainty in lost capacity was estimated to be about ±3%. In the worst case, this uncertainty is about 26 ton-hr, (91 kWh, 0.04% of the overall tank capacity).

Because the tank is rarely discharged completely, the lower diffuser does not affect tank thermal performance. Therefore, the only opportunity for mixing to occur exists at the upper diffuser. For representative flow rates, the capacity lost due to mixing at the upper diffuser of this tank represents 1% of the overall tank capacity.

DISCUSSION

The two monitoring periods reported took place in August and September, two of the hottest months of the year in Dallas, TX. Utilization of the storage tank was high throughout this period and environmental heat gains to the storage tank could be expected to be near their maximum values. Therefore, the performance observed is indicative of the worst case behavior of the system.

The low Reynolds number four ring octagonal upper diffuser of this tank produces excellent stratification at the start of discharge cycles. Thermocline thicknesses produced by this diffuser are much smaller than those produced by a single ring high Reynolds number diffuser and significantly smaller than typical thermoclines produced by radial diffusers. More investigation is needed to determine which parameters control radial and slotted pipe diffuser performance; however the performance of this tank suggests that lower inlet Reynolds numbers or a related parameter results in improved stratification..

SUMMARY AND CONCLUSIONS

The utilization of a stratified chilled water storage system serving a manufacturing facility in a hot, humid climate has been described.

The efficiency of the thermal storage system, over the period of observation was nearly 98 percent, indicating that environmental heat gains to the storage tank are small.

Lost capacity, which indicates the effects of inlet mixing, was less than 1.6% of nominal capacity over the range of flow rates observed, which included operation at design flow rates.

The discharge control strategy for the system is storage priority; however, complete discharge cycles are uncommon. Typically, a reserve of 10% or more of tank volume remains at the end of a discharge cycle. As a result, thermoclines are not generated during typical charge cycles. Thermoclines generated during discharge are very sharp by comparison with those found in other full-scale tanks.

Typical flow rates during both charge and discharge cycles were substantially lower than design values. Because inlet diffuser mixing increases with flow rate, the performance of the system on average may be better than at design conditions.

The system has been a success for the owner on a number of levels. It has saved energy, simplified system operation, reduced energy cost. The effectiveness of this installation indicates that chilled water storage is an excellent technology for application in hot and humid climates.

ACKNOWLEDGMENT

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