Rehabilitating A Thermal Storage System Through Commissioning

Mingsheng Liu, Ph.D., P.E.
Energy Systems Laboratory
Texas A&M University
College Station, Texas

Bryan Veteto
Energy Systems Laboratory
Texas A&M University
College Station, Texas

David E. Claridge, Ph.D., P.E.
Energy Systems Laboratory
Texas A&M University
College Station, Texas

Abstract

A thermal storage system was installed in a 37,000 ft^2 hospital in April 1992. The chiller had to be operated every summer during peak demand periods until the authors rehabilitated system through building commissioning in late 1995. This paper presents information about the building systems, commissioning activities, and the measured energy savings which resulted from the process.

Introduction

A thermal storage system stores heat in a thermal storage medium during periods of low cooling demand. The stored cooling is later used to meet an air-conditioning or process cooling load. Interest in thermal storage for commercial applications grew during the 1970s and 1980s, when electric utility companies recognized the need to reduce the peak demand on their

successful examples have been published [Bartlett et al, 1995; Siverling et al, 1995; Bahnfleth et al., 1995; Crane et al., 1994, etc]. However, unsuccessful projects are seldom documented and published. This limits the information available for engineers and operational staff to trouble shoot thermal storage systems and maintain suitable performance. This paper presents a case study of rehabilitating a thermal storage system. It describes building and its systems, operational problems, the analysis procedures applied and results of the systems analysis, the commissioning activities, and measured results of the system modifications on performance.

Building and System Information

The hospital has a total floor area of 37,000 square feet which includes 32 patient rooms with an area of approximately 13,000 square feet. It was built in 1978 and a chilled-water thermal storage system was installed in 1992 to provide cooling during the electric utility's peak hours of 12:00 noon to 8:00 PM from May to October. During the summer of 1992, it was frequently necessary to operate the chiller during peak hours in order to maintain building comfort. Evaluations performed by a consulting firm in 1993 and 1994 concluded that a supplementary chiller (50 tons) was needed due to an under-sized storage tank and an under-sized chiller.

In 1995, the authors were asked to investigate the problems and provide possible solutions. The thermal storage system was subsequently rehabilitated in 1996 as part of a building commissioning process.

The peak cooling load was determined, using the ASHRAE TETD method, to be 93 tons. The design-day total cooling consumption was determined to be 1375 ton-hours, with 639

day are presented in Figure 1.

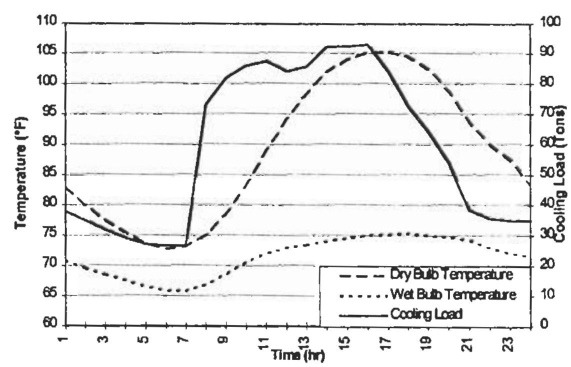

Figure 1: Profiles of Design Day Cooling Load and Outside Air Dry Bulb and Wet Bulb Temperatures

A roof top single duct constant volume system provides conditioned outside air (2,000 cfm, or 0.15 cfm/ft^2) to each patient room where a fan coil unit (FCU) maintains suitable room comfort conditions. Other areas are conditioned by 12 multi-zone constant volume systems. The design air flow rate is 40,300 cfm (1.67 cfm/ft^2) with 8,800 cfm (0.37 cfm/ft^2) outside air. The installed coil capacity is 130 tons for the 13 AHUs excluding the FCUs. The summary of design air-handler information is listed in Table 1.

Chilled water is supplied to the building through two loops: a loop for the 32 FCUs in the patient-rooms and a loop for the 13 roof top air handlers. The loop for the fan coil units is piped with reverse return while the loop for the roof top air handlers has direct return. The FCUs in the patient rooms have three-way valves while the roof top units have two-way valves.

Table 1: Summary of Design Characteristics of the AHUs

AHU	Total Air Flow (cfm)	Outside Air Flow (cfm)	Outside Air Fraction (%)	Fan Power (hp)	Coil Capacity (kBtu/hr)
1	4,800	300	6%	5	110,000
2	8,000	2,000	25%	7.5	245,000
3	2,600	200	8%	2	85,000
4	3,500	500	14%	3	115,000
5	1,800	400	22%	1.5	78,000
6	1,800	300	17%	1.5	70,000
7	4,000	750	19%	5	130,000
8	2,700	600	22%	2	120,000
9*	2,000	2,000	100%	1.5	76,000
10	4,500	1,000	22%	7.5	200,000
11	2,400	900	38%	2	105,000
12	1,000	350	35%	1.5	60,000
13	3,200	1,500	47%	5	170,000
Total	42,300	10,800	26%	45	1,564,000
	1.68	0.31	26%		130

* Outside air unit for the patient rooms.

Figure 2a presents a schematic diagram of the water loop outside the building which supplies chilled water to the AHU and FCU water loops in the building. The system setup shown represents the thermal storage system in the harvesting mode. The building chilled water pump draws water from the bottom of the tank and sends the return water to the top of the tank. Valve V4 isolates the chiller from the building and the tank. In the charging mode (Figure 2b), valves V3 and V4 are open while valve V1 is closed. The tank pump sends chilled water to the bottom of the tank. A building pump (the second pump shown in Figure 2 is used only as a backup) supplies chilled water to the building directly from the chiller.

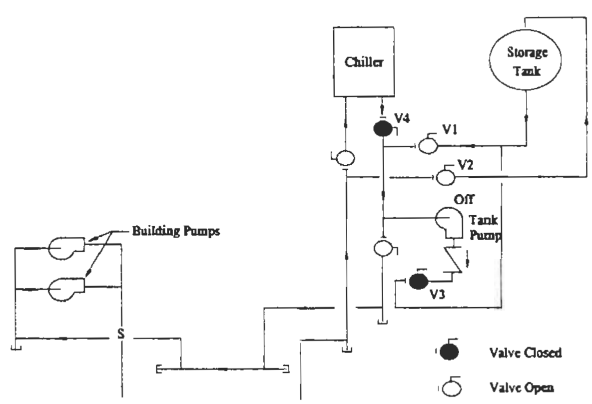

Figure 2a: Schematic Diagram of Thermal Storage System in the Harvesting Mode

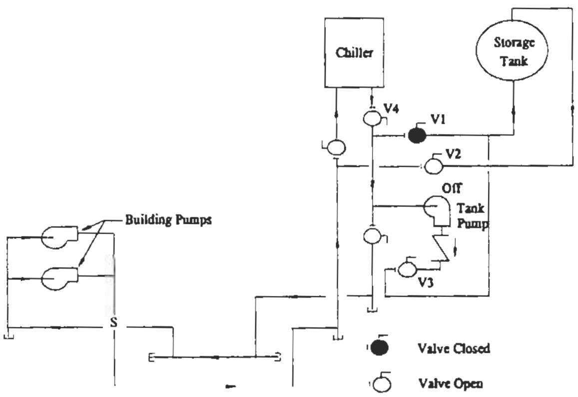

Figure 2b: Schematic Diagram of Thermal Storage System in the Charging Mode

The chiller is a 120 ton air cooled chiller. The actual capacity varies from 88 tons to 110 tons (based on chiller specifications) as the ambient air temperature decreases from 115°F to 85°F. The water tank is designed to store 639 ton-hours of cooling in a tank volume of 80,675 gallons at an assumed differential temperature for the tank of 12°F and a volumetric efficiency of 95%.

The working point of the tank pump is designed to be 120 GPM at 40 feet of water column. At this pumping rate, the pump can fully charge the tank in 11 hours. The working point of the building pump was designed to be 180 GPM with 75 feet of water column. The chiller has a rated pressure drop of 13.4 feet when the flow rate is approximately 300 GPM. When the pressure drop from the pump to the chiller is included, the total pressure drop is approximately 30 feet. However, the pump for the building chilled water loops is significantly over-sized, so most of the 300 GPM total flow goes through the building if the control valves do not function perfectly. Figure 3 presents the pump curves and the relationship of water flow rate and the pressure drop across the chiller.

Systems Analysis

In the first summer of operation (1992), the thermal storage system exhibited two problems: (1) the cold water ran out as early as 4:00 PM which led to comfort complaints; and (2) the tank could not be fully charged during off-peak hours. Due to these problems, the chiller had to be turned on during peak hours to maintain suitable building comfort. Figure 4 presents the measured whole building electricity consumption versus the time of day for June and July of 1992. The chiller was turned on every weekend to charge the tank as much as possible. Even so, the chiller had to be turned on during the peak hours of three weekdays. The causes of these problems are analyzed below.

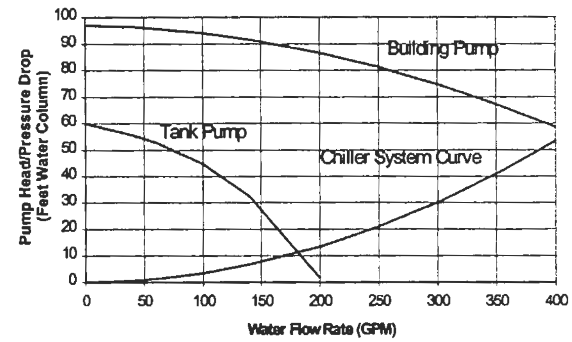

Figure 3: Chilled Water Performance Curves and the Relationship Between Chilled Water Flow Rate and the Pressure Drop Across the Chiller Loop Section

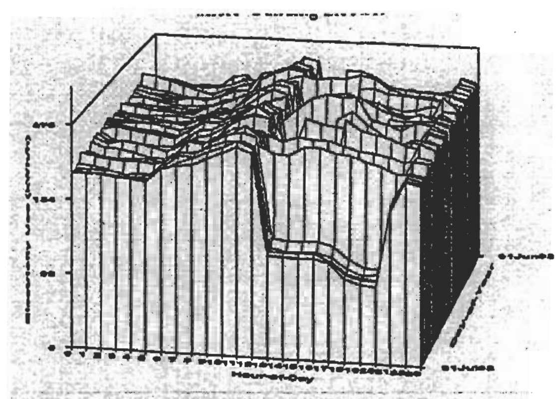

Figure 4: Measured Whole Building Electricity Consumption Versus the Time of Day for the Months of June and July 1992

Tank Size

The tank has a volume of 80,675 gallons. When the peak differential temperature is controlled at 12°F, the tank can provide the rated cooling capacity (638 ton-hours) to the building during peak hours with a 5% safety factor. However, the actual peak summer time chilled water differential temperature across the tank as measured by the consultant during their evaluation in 1993-94 varied from 4°F to 10°F. The tank was undercharged due to the operation of the: (1) three-way valves in 32 patient rooms; (2) wide open valves in the roof top units; and (3) over-sized building pump. If the peak chilled water differential temperature is 8°F, the tank capacity is reduced from 638 ton-hours to 425 ton-hours for a complete cycle of water flow. Under the design day conditions, with an 8 °F temperature differential, all of the chilled water in the tank will be used up by 4:30 PM. Hence, there is not adequate tank capacity unless the peak differential temperature is controlled at 12°F as designed.

Tank Pump

Although both the water tank pump and the building pump are over-sized, the chilled water flow rate to the tank can be significantly smaller than the design value due to the oversized building pump. A simplified system diagram is shown in Figure 5 for the whole water system which includes the chiller section (from joint 2 through chiller to joint 1); tank section (from joint 1 through tank pump and tank to joint 2); and the building section (from joint 1 through building pump and building to joint 2).

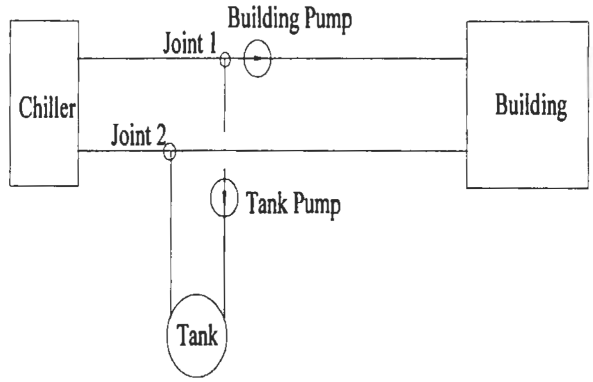

Figure 5: Simplified Water System Diagram

Based on the system layout, the hydraulic performance of the system was estimated. The loop and component results are summarized in Table 2.

A constant loop resistance factor was assumed for the building loop since most control valves were fully open due to extremely low cold deck set-points.

Table 2: Summary of Hydraulic Performance for Water Loop and Each Component

Component	Rated ΔP (psi)	Rated Flow (GPM)	Relationship Between ΔP and Flow
Building loop	30	180	$\Delta P = 30\left(\frac{GPM}{180}\right)^2$
Tank loop	30	100	$\Delta P = 30\left(\frac{GPM}{100}\right)^2$
Chiller loop	30	300	$\Delta P = 30\left(\frac{GPM}{300}\right)^2$
Tank pump	$\Delta P = 60 - 0.09961 \times GPM - 0.000061 \times GPM^2 - 0.0000045 \times GPM^3$		
Building pump	$\Delta P = 97 - 0.00783 \times GPM - 0.00022 \times GPM^2 - 0.00000000077 \times GPM^3$		
VFD	$\frac{\Delta P_1}{\Delta P_2} = \left(\frac{n_1}{n_2}\right)^2\left(\frac{GPM_1}{GPM_2}\right)^2$		

The results of loop simulation show that the tank pump can only provide 64 GPM of chilled water to the tank when the building pump provides 250 GPM to the building. Therefore, the tank pump can only provide a total of 61,440 gallons of water (76% of tank capacity) to the tank during the 16 off-peak hours.

The water flow to the tank can be increased by either using a larger tank pump or reducing the building loop flow. Building loop flow can be reduced : (1) by increasing the building loop resistance by closing a valve or valves; or (2) by slowing down the pump by using a variable frequency drive (VFD).

Increasing the tank pump size would increase the chilled water flow rate through the chiller. Since the chilled water flow rate is already higher than the rated flow rate of 228

GPM, the supply water temperature control would be even worse.

Partially closing the manual valve on the return section of the building loop would reduce the chilled water flow to the building and increase the chilled water flow to the water tank. This is an easy and quick solution to the problem. However, the best working point varies as the building load changes. Since the system capacities (both chiller and tank) are marginal for the building, a VFD was suggested for the system.

The chilled water flow was simulated at different pump speeds by assuming that a VFD was installed to control the building pump. The simulation results are presented in Figure 6. When the speed of the building pump is controlled from 60% to 70% of capacity, the chilled water flow rate varies from 88 GPM to 95 GPM for the tank, and from 139 GPM to 168 GPM for the building.

During off-peak hours, the cooling load varies from 35 tons to 88 tons with an average of 50 tons for the design day. After the building is commissioned, the differential temperature may be controlled with DT as small as 8°F due to water balance problems. Under the average design-day cooling load condition, we can run the chilled water pump at 65% speed. The tank will then receive 91.5 GPM, 150 GPM will flow to the building, and 241.5 GPM will flow through the chiller. The tank can then be fully charged in less than 15 hours, while the chiller has enough capacity to control the chilled water supply temperature at 44 °F or lower.

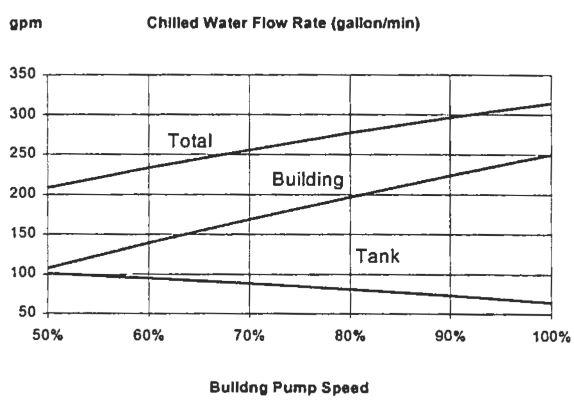

Figure 6: Simulated Chilled Water Flow Rate through Chiller, Tank, and Building When the Building Pump is Controlled at Different Speeds

System Commissioning

As part of the system commissioning, the room comfort conditions, air flow rates of AHUs, and chilled water flow conditions were measured in November 1995.

The room CO_2 concentration was lower than 500 ppm while the outside air CO_2 was over 350 ppm. This indicated that the outside air intake was much higher than required. Reducing outside air intake can greatly reduce the cooling load during the summer.

Table 3 compares the measured air flow rates with the design air flow rates. The outside air intake could not be measured accurately due to field restrictions. The designed average air flow rate is 1.68 cfm/ft^2. The measured average air flow rate was 1.96 cfm/ft^2, or 17% higher than the designed flow

The pressure of the mixing chamber varied from -0.6"$_{H_2O}$ to -0.8"$_{H_2O}$ in the AHUs. Due to this negative pressure combined with poor outside air dampers (without rubber seals), the actual outside air intake is much higher than the design flow of 8,800 cfm (0.37 cfm/ft^2). It appeared that both total and outside air flow requirements should be re-evaluated and adjusted based on the actual use of each room.

Table 3: Summary of AHU Flows

AHU	CFM_{design}	$CFM_{measured}$
1	4,800	5,264
2	8,000	10,105
3	2,600	2,998
4	3,500	4,000
5	2,050	2,030
6	1,800	3,353
7	4,000	3,826
8	2,700	3,363
10	4,500	4,500
11	2,400	2,727
12	1,000	1,425
13	3,200	3,475
Total	40,300	47,064

The field inspection also found that a number of economizer controllers were set incorrectly, such as a high temperature limit of 80°F, and a cold deck set point of 35°F. These improper set-points were corrected during the site inspection.

The thermal storage system operation was evaluated by measuring: chilled water supply and return temperatures at the chiller (sensors were put into the sensor wells), and at the thermal storage tank (the sensors were put underneath the insulation). The measured results are summarized in Table 4.

Table 4: Summary of Measured Chilled Water Differential Temperatures (°F) during November 1995 Site Inspection

	Charging	Harvesting
Chiller	2 - 7	0
Tank	0 - 8	5 - 7
Building	3 - 4	5 - 7

During harvesting, the differential temperature was generally in the range of 5°F to 7°F across the tank (significantly smaller than the design value of 12°F and a narrower range than the summertime values measured earlier by the consultant).

During the charging process, the differential temperature across the tank varied from 0°F to 8°F, 2°F to 7°F across the chiller, and 3°F to 4°F across building, **which was 3 to 4 times smaller than the design value of 12°F.** It appears that the chilled water flow to the building was 3 to 4 times higher than required. This appears to limit the water flow used to charge the tank which in turn prevents the system from charging the storage tank during the summer months.

In March 1996, a stand alone controller and VFD system were designed and installed by the authors. Two thermo-couples are used to measure the building chilled water supply and the return temperatures. The controller receives these signals and compares with the set-point differential (12°F). When the differential temperature is lower than the set-point, the controller slows down the pump through a PI loop. When the differential temperature is higher than the set-point, the controller speeds up the pump.

Before the controller sends the command to the VFD, it compares the output of the PI loop with the low limit of the pump speed (60%). If the command is lower than 60%, the pump speed is set at 60%.

This low limit was determined based on the need to: (1) fully charge the tank during off-peak hours; (2) prevent negative pressure at the highest point of the water loop; and (3) avoid frequent maintenance of water loop balance.

In April 1996, the air flow rate to each zone was adjusted to 1.1 cfm/ft^2 by using zone dampers and return air dampers. All the outside air dampers were totally closed to avoid excessive outside air intake. Due to the poor quality of the dampers and the very negative pressure in the mixing chamber, the average building CO_2 concentration is still lower than 600 ppm. After adjustments, most AHUs still supply more outside air than necessary due to over-sized fans. Consequently, we recommended changing the pulleys to reduce the total air flow and save electricity. We also recommended replacing the three-way valves with two-way valves in the FCUs in the patient rooms.

Measured Results

After the VFD installation and the AHU commissioning, the thermal storage system has performed well. Figure 7 presents the measured whole building electricity consumption during June and July 1996. The rehabilitation of the thermal storage system decreased the peak demand by 140 kW.

Figure 8 compares the measured daily average whole building electricity consumption before and after the commissioning. The pre-commissioning period is from January 1, 1995 to December 31, 1995. The post period is from April 1, 1996 to December 31, 1996. The AHU commissioning reduced the electricity consumption by approximately 20 kW

Figure 9 compares the measured daily average electricity consumption of the building pump pre- and post-commissioning. The electricity consumption of the building chilled water pump varied from 3.5 kW to 6.5 kW pre-commissioning. Post-commissioning, it varies from 1 kW to 4 kW. The pump power consumption has been reduced by 10% to 70% depending on the ambient temperature.

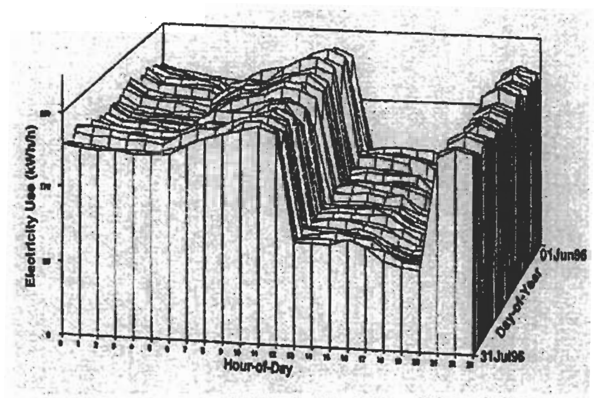

Figure 7: Measured Whole Building Electricity Consumption During June and July of 1996

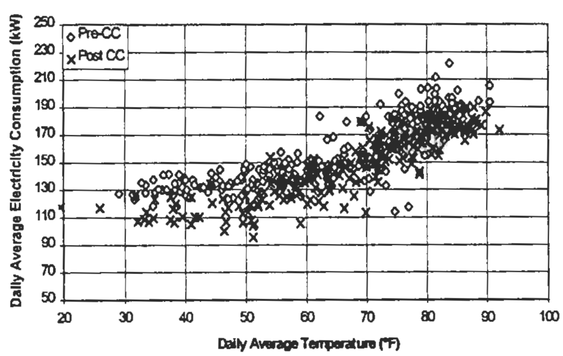

Figure 8: Comparison of Measured Whole Building Electricity Before (Pre) and After (Post) Building Commissioning

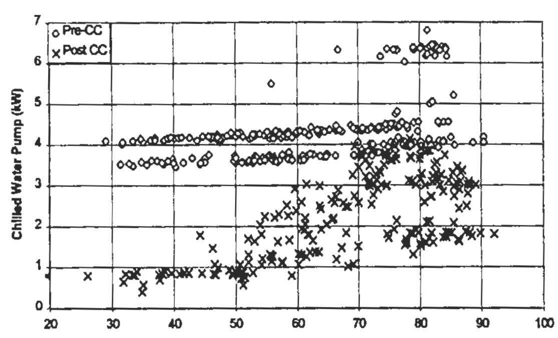

Figure 9: Comparison of Measured Electricity Consumption of the Building Chilled Water Pump Before (Pre) and After (Post) Building Commissioning

The rehabilitation of the thermal storage system allowed the hospital to realize annual peak demand savings of $14,190/yr. Building commissioning also reduced the annual electricity consumption by 134,000 kWh/yr which provides additional cost savings of $5,540/yr for total savings of $19,730/yr.

Conclusions

A thermal storage system was successfully rehabilitated through a building commissioning process. The building commissioning process also reduced the whole building electricity consumption by 134,000 kWh/yr by reducing the cooling load, chilled water pump power, and fan power. The total annual cost savings are $19,730.

It is important to point out that (1) the actual differential temperature needs to be considered when sizing a thermal storage tank; and (2) a single over-sized part in a complicated system may make effective operation of the system impossible.

Acknowledgements

This paper was prepared with financial support from the Texas State Energy Conservation Office.

References

Bahnfleth W. P. and W. S. Joyce, 1995. Stratified Storage Economically Increases Capacity and Efficiency of Campus Chilled Water System. ASHRAE Journal, March 1995.

Bartlett T. A and R. Froebe, 1995. The Design and Operation of J C Penny's Ice Storage System Takes Conversion and/of Addition of Future Controls and Equipment Into Consideration. Heating Piping Air Conditioning, April, 1995.

Crane J. M and C Dunlop, 1994. Ice Storage System for a Department Store. ASHRAE Journal, January 1994.

Dorgan C. E. and J. S. Elleson, 1994. Design Guide For Cool Thermal Storage. Atlanta, Georgia: ASHRAE.

Siverling A. M. and K. J. Kressler, 1995. Ice Storage System Assures Data Center Cooling. Heating Piping Air Conditioning, April, 1995.