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USACERL TECHNICAL REPORT E-90/10
June 1990
FEAP: Diurnal Energy Storage System

AD-A224 739

Performance of an Ice-in-Tank Diurnal Ice Storage Cooling System at Fort Stewart, GA

by
Chang W. Sohn
Gerald Cler
Robert J. Kedl

Diurnal cold storage cooling systems provide an effective means for reducing peak electrical demand at Army installations. The U.S. Army Construction Engineering Research Laboratory (USACERL) demonstrated an ice-in-tank diurnal ice storage (DIS) cooling system at the Post Exchange (PX) building, Fort Stewart, GA in April 1987 as part of the Facility Engineering Applications Program (FEAP). Design and construction of the system have been documented in USACERL Technical Report E-88/07.

The system was instrumented during the 1987 cooling season to test its performance. This report documents the system energy performance, peak shaving capability, operation and maintenance experience, and lessons learned from the Fort Stewart demonstration. The efficacy of a DIS cooling system as a means of reducing peak electrical demand has been verified.

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY (Leave Blank)	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	June 1990	Technical Report

4. TITLE AND SUBTITLE

Performance of an Ice-in-Tank Diurnal Ice Storage Cooling System at Fort Stewart, GA

5. FUNDING NUMBERS

WU FEAP-EB-FF9

6. AUTHOR(S)

Chang W. Sohn, Gerald Cler, and Robert J. Kedl

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

U.S. Army Construction Engineering Research Laboratory (USACERL)
PO Box 4005
Champaign, IL 61824-4005

8. PERFORMING ORGANIZATION REPORT NUMBER

TR E-90/10

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)

US Army Engineering and Housing Support Center
ATTN: CEHSC-FU
Ft. Belvoir, VA 22060-5000

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

11. SUPPLEMENTARY NOTES

Copies are available from the National Technical Information Service, 5285 Port Royal Road, Springfield, VA 22161

12a. DISTRIBUTION/AVAILABILITY STATEMENT

Approved for public release; distribution is unlimited.

12b. DISTRIBUTION CODE

13. ABSTRACT (Maximum 200 words)

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14. SUBJECT TERMS

Fort Stewart, GA
Dirunal ice storage cooling system

15. NUMBER OF PAGES

40

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	SAR

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std 239-18
298-102

FOREWORD

This work was performed for the U.S. Army Engineering and Housing Support Center (EHSC) under the FY88 Facility Engineering Applications Program (FEAP). The Work Unit was FEAP-EB-FF9, "Diurnal Energy Storage System." The EHSC Technical Monitor was B. Wasserman, CEHSC-FU.

The project was conducted by the U.S. Army Construction Engineering Research Laboratory Energy Systems Division (USACERL-ES). Dr. G. R. Williamson is Chief of USACERL-ES.

LTC E.J. Grabert, Jr. is Commander of USACERL and Dr. L.R. Shaffer is Director.

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PERFORMANCE OF AN ICE-IN-TANK DIURNAL ICE STORAGE SYSTEM AT FORT STEWART, GA

1 INTRODUCTION

Background

As part of the Facility Engineering Applications Program (FEAP), the U.S. Army Construction Engineering Research Laboratory (USACERL) installed an ice-in-tank diurnal ice storage (DIS) cooling system on the Post Exchange (PX) building at Fort Stewart, GA in early 1987. Design and construction of the Fort Stewart DIS cooling system have been documented elsewhere.[1]

The FEAP demonstration was conducted to determine the system's operating and performance characteristics. This information will allow installation Directorates of Engineering and Housing (DEHs) to select suitable DIS technologies.

Objective

The objective of this work was to document first-year system operation and performance data from the Fort Stewart ice-in-tank DIS cooling system.

Approach

The conventional cooling system was instrumented before the DIS cooling system was installed to generate a reference for comparison studies. Type-T (copper-constantan) thermocouple temperature sensors and vortex-shedding flow meters were added after the DIS cooling system was installed. Hourly data were taken 24 hr/day during the 1987 cooling season. The collected data were stored on a cassette tape and transferred to Oak Ridge National Laboratory (ORNL), which is cooperating with USACERL in the DIS research. ORNL conditioned the data and provided them to USACERL in legible format through a personal computer (PC) version of BASIC programming language. USACERL reduced and analyzed the data.

Scope

The data discussed in this report apply to the system installed at Fort Stewart, GA only. The performance may be similar for systems with the same configurations as the one installed at Fort Stewart. However, generalization of performance for generic ice-in-tank systems is not recommended due to the limited amount of data available from only one system.

USACERL Technical Report E-88/07 covers principles in storage cooling technologies, market availability of the systems, characteristics of storage cooling systems for Army application, the USACERL

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[1] C. W. Sohn and J. Tomlinson, *Design and Construction of an Ice-In-Tank Diurnal Ice Storage Cooling System for the PX Building at Fort Stewart, GA*, Technical Report E-88/07/ADA197925 (U.S. Army Construction Engineering Research Laboratory [USACERL], June 1988); C. W. Sohn and J. J. Tomlimson, *Diurnal Ice Storage Cooling Systems in Army Facilities*, (ASHRAE Transactions, V. 95, Pt. 1, 1989).

demonstration program in DIS cooling systems for the Army, and execution of the demonstration program in cooperation with ORNL.

Mode of Technology Transfer

This demonstration has been described in the FEAP Notebook. It is recommended that information on DIS cooling systems be summarized in a Technical Note (TN) upon completion of the DIS cooling systems demonstration. Technical reports discussing design/installation and operation/performance of each type of DIS cooling system will serve as interim design guidance. At the end of the demonstration program, USACERL will develop design and operating instructions for inclusion in the appropriate Army criteria documents.

2 SYSTEM DESCRIPTION

Building Description

The main PX building at Fort Stewart was selected for demonstration of the ice-in-tank DIS cooling system. The building has 51,000 sq ft* of floor area containing a department store and auxiliary facilities such as a candy shop, gift shop, barber and beauty shops, and a small food service. It has a controlled occupancy schedule: 9:30 a.m. to 7:30 p.m. on Monday through Saturday, and 11:00 a.m. to 5:00 p.m. on Sunday. The building is cooled by a 178-ton Trane Centravac centrifugal chiller. During business hours, the electric demand of this chiller is typically 100 to 120 kW for a hot day. It is this electric demand that was to be shifted by the DIS cooling system.

After the DIS cooling system design was completed, the building was expanded by 14,400 sq ft. The expanded section contains its own independent cooling system, including a 35-ton reciprocating chiller and two air handlers. The expansion was designed so as to not affect the thermal load of the original system. However, it was observed that the expansion has significantly increased the cooling load of the original system.

System Schematics

Figure 1 is a schematic of the DIS cooling system. This system was designed as an "add-on" to the existing chilled water system. In this way, the building can be cooled by either the existing centrifugal chiller (CH-1) or the DIS system, which includes an ice-making reciprocating chiller (CH-2).

The ice-making loop requires a fluid temperature less than 32 °F; therefore, it must circulate brine (25 percent ethylene glycol). The brine loop must be isolated from the building chilled water loop, with a heat exchanger (HX-1) used between the two loops. The ice tank battery consists of 10 Calmac Manufacturing Corp. Model 2090 ice storage tanks, arranged as shown in Figure 2. Their total nominal capacity is 900 ton-hr, with an actual storage capacity of 810 ton-hr.

The ice maker ordered was a nominal 175-ton dual reciprocating compressor that derates to 112 tons when supplying brine at 26 °F to the ice storage tanks. The ice maker that was supplied by the manufacturer and installed for the DIS cooling system had a nominal rating of 200 tons. Both chillers share the same cooling tower; control interlocks prevent their simultaneous operation. Control valve FCV-2 is a three-way modulating valve that maintains a constant return water temperature from the building distribution system. Control valve FCV-1 is a three-way diverting valve that allows for the various modes of operation. Manifold and feeder pipe and fittings to the ice storage tanks are made of polyvinyl chloride (PVC). All other brine loop piping is steel. As shown in Figure 3, the system is located outside and enclosed by a chainlink fence.

System Operation and Control

Control over daily system operation consists of three modes: (1) direct cooling from 8:00 a.m. to 12:00 noon, (2) cooling by stored ice from 12:00 noon to 6:00 p.m. and (3) ice building and night-time cooling from 6:00 p.m. to 8:00 a.m. These operating modes are selected by an electronic programmable timer.

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*A metric conversion chart is on p 30.

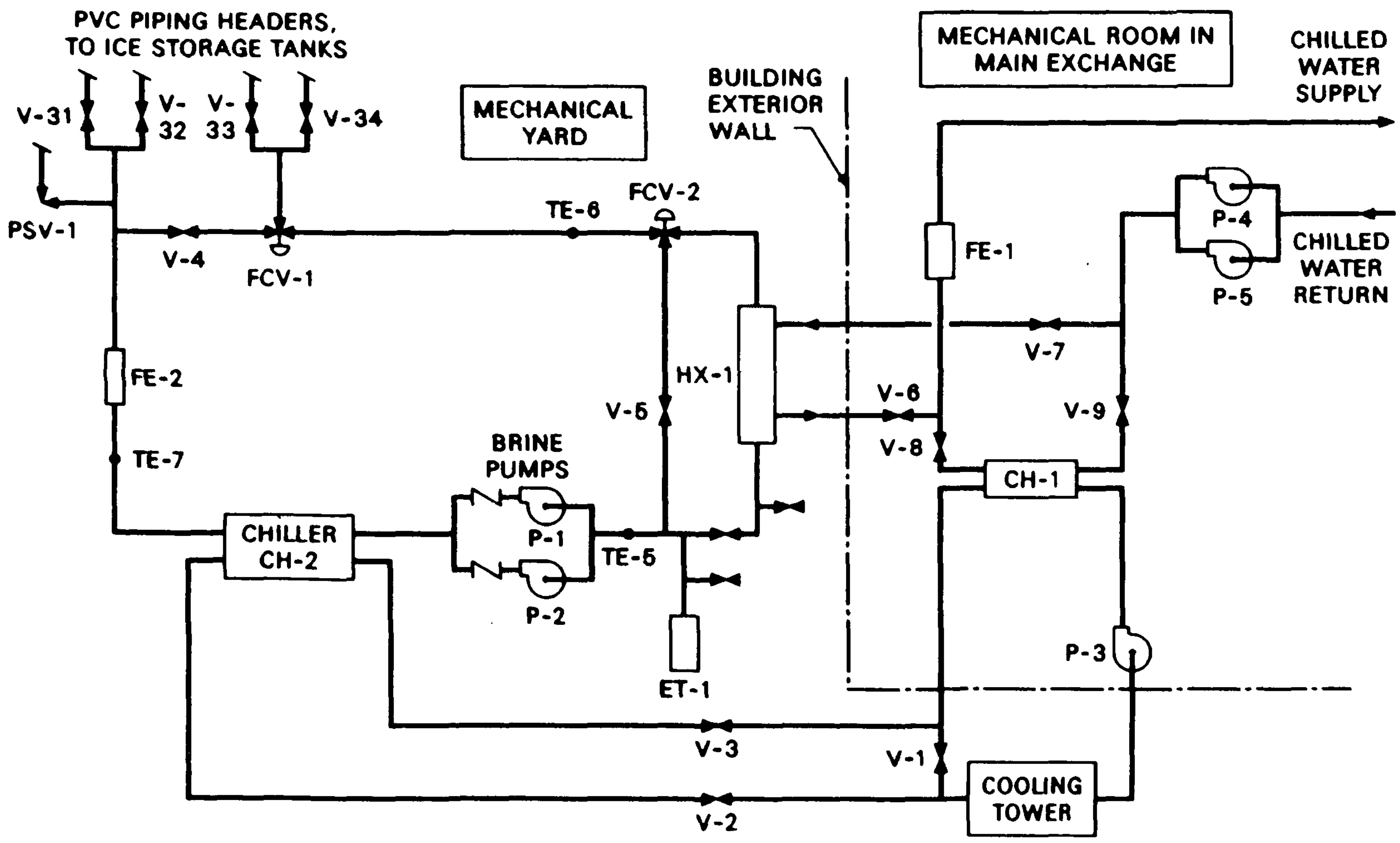

Figure 1. Ice system in main Post Exchange.

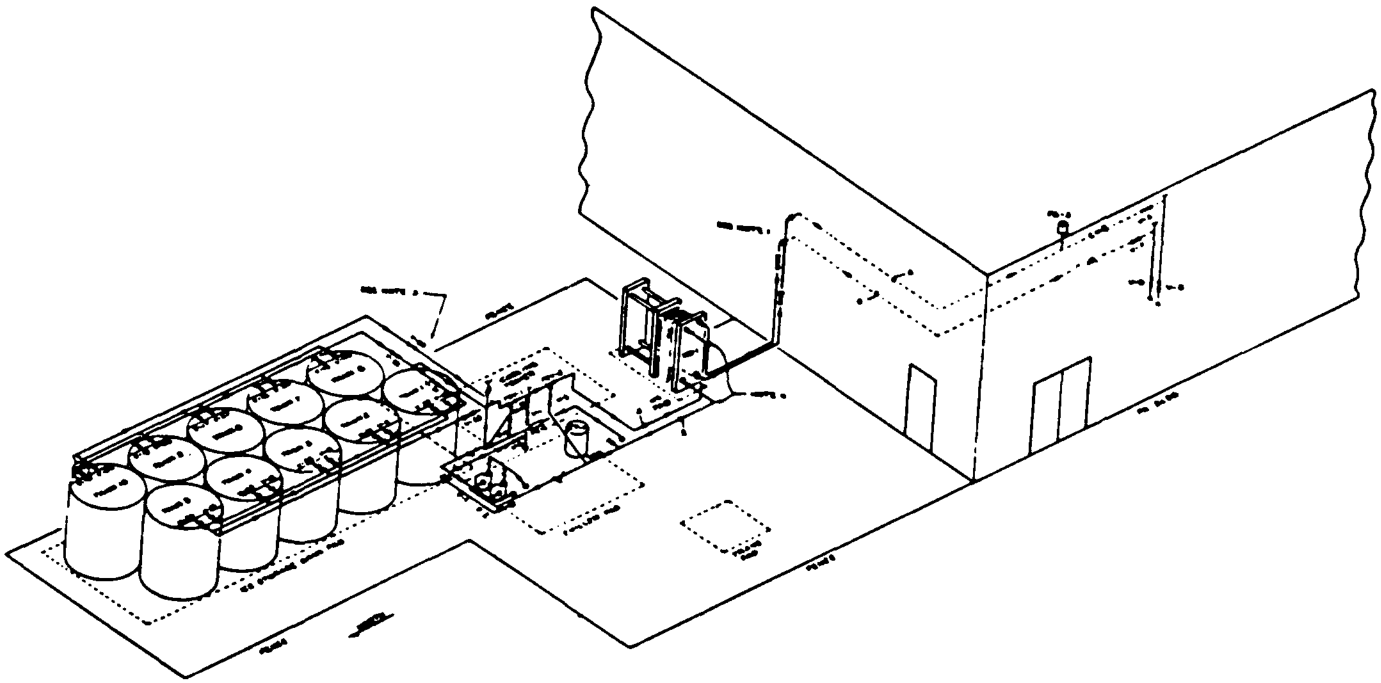

Figure 2. Arrangement of the ice storage tanks.

Figure 3. The installed DIS system.

Mode 1: 8:00 a.m. to 12:00 Noon

Brine is circulated by pumps P-1 and P-2 through chiller CH-2, through hand valve V-4, flow control valve FCV-1, flow control valve FCV-2, and the plate heat exchanger, HX-1. Valve FCV-1 is used as a diverting valve, whereas valve FCV-2 is a modulating valve that controls the flow through HX-1 in response to the building return water temperature. If this water temperature rises above 53 °F, valve FCV-2 reduces the flow through hand valve V-5, sending more brine through HX-1. Valves V-5 and V-4 are balancing valves that maintain a relatively constant head pressure on the system regardless of the mode of operation. FE-2 is a flow meter used to monitor brine flow rates continuously as part of an experimental evaluation of the system.

Mode 2: 12:00 Noon to 6:00 p.m.

During this period, the chiller CH-2 is turned off and valve FCV-1 is diverted so that brine is circulated by pumps P-1 and P-2 through the ice tank battery. Valve FCV-2 operates as a control valve in the same manner as before.

Mode 3: 6:00 p.m. to 8:00 a.m.

During this period, the chiller CH-2 is used to make ice and satisfy a small nighttime building cooling load at the same time. Valve FCV-1 remains in the same position as in Mode 2 and the chiller is activated. Control valve FCV-2 continues to modulate in response to the building cooling load. A temperature sensor in the brine return line from the tanks will turn the chiller off when the ice tank is fully charged.

During operation of the DIS cooling system, it was noted that the addition to the building had significantly increased the cooling requirements. The increased nighttime cooling requirement resulted in the ice tanks not being fully charged on some occasions. This situation, in turn, caused a shortage of stored cooling capacity for the next day's cooling. Figure 4 shows the brine temperatures at the outlet of the storage tank and chiller power for a typical 2-day period, 23 and 24 July 1987. At about 5:00 a.m. on 23 July, the storage tank was fully charged, with the phase change complete. This condition is indicated on the figure by the rapid decrease in brine temperature. At 7:00 a.m., the temperature of the brine returning to the chiller reached a lower limit (20 °F) and the chiller shut down. The chiller was directly cooling the building from 8:00 a.m. to 12:00 noon. A 12:00 noon, the chiller was turned off and cooling was supplied by the stored ice. The gradual increase in brine outlet temperature (Figure 4) indicates ice melting.

At 6:00 p.m., the next ice production cycle began. Notice that there was no decrease in brine outlet temperature near the end of this cycle. This condition indicates the possibility of incomplete freezing. Again at 8:00 a.m., the chiller began direct cooling of the building. At 12:00 noon, the chiller stopped and the cooling was supplied by the stored ice. At about 4:00 p.m., the brine outlet temperature began to rise very rapidly. This increase occurred because the ice had completely melted and the sensible energy of the water in the tanks has very little cooling capacity compared with the latent energy stored in ice. Also to be noted is the long time required to bring the water temperature down to the freezing point once the next ice production period began at 6:00 p.m. on 24 July 1987. This situation further reduced the amount of ice that could be produced during that period.

The problem was resolved by two actions. First, the main building air handler controls were routed through a 7-day timer, and the air handler was turned off from 10:00 p.m. to 6:00 a.m. This arrangement effectively placed the air-conditioning system on night setback, with chiller CH-2 dedicated to making ice. Second, the time interval of Mode 2 was reduced by 2 hr from 12:00 noon to 6:00 p.m. to 12:30 p.m. to 4:30 p.m. With these two corrections, chiller CH-2 delivered ample refrigeration to freeze ice in the tanks as shown in Figure 5. However, this was not the end of the problem. As Figure 5 shows, adequate ice was available to meet the cooling load between 2:00 p.m. and 6:00 p.m. At night, with the air handler off and a shorter ice melting period, the tanks were frozen solid each night at around 2:00 a.m. This freezing resulted in intermittent chiller operation from 2:00 a.m. to 8:00 a.m. and was a very undesirable operating condition for the cooling equipment. The problem was solved by extending the ice cooling period by 1 hr.

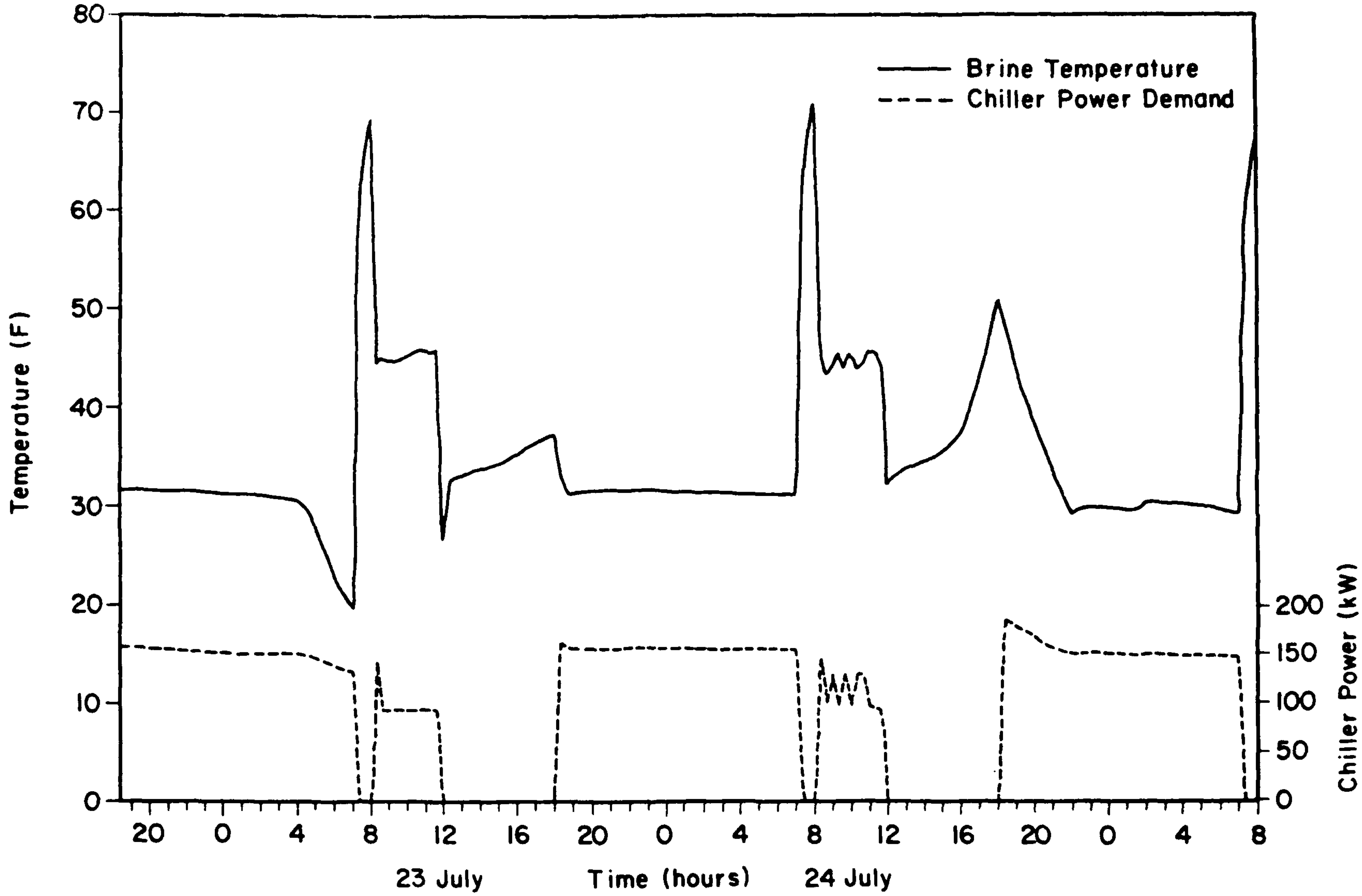

Figure 4. Brine temperature at the tank outlet and chiller power demand (23-24 July 1987).

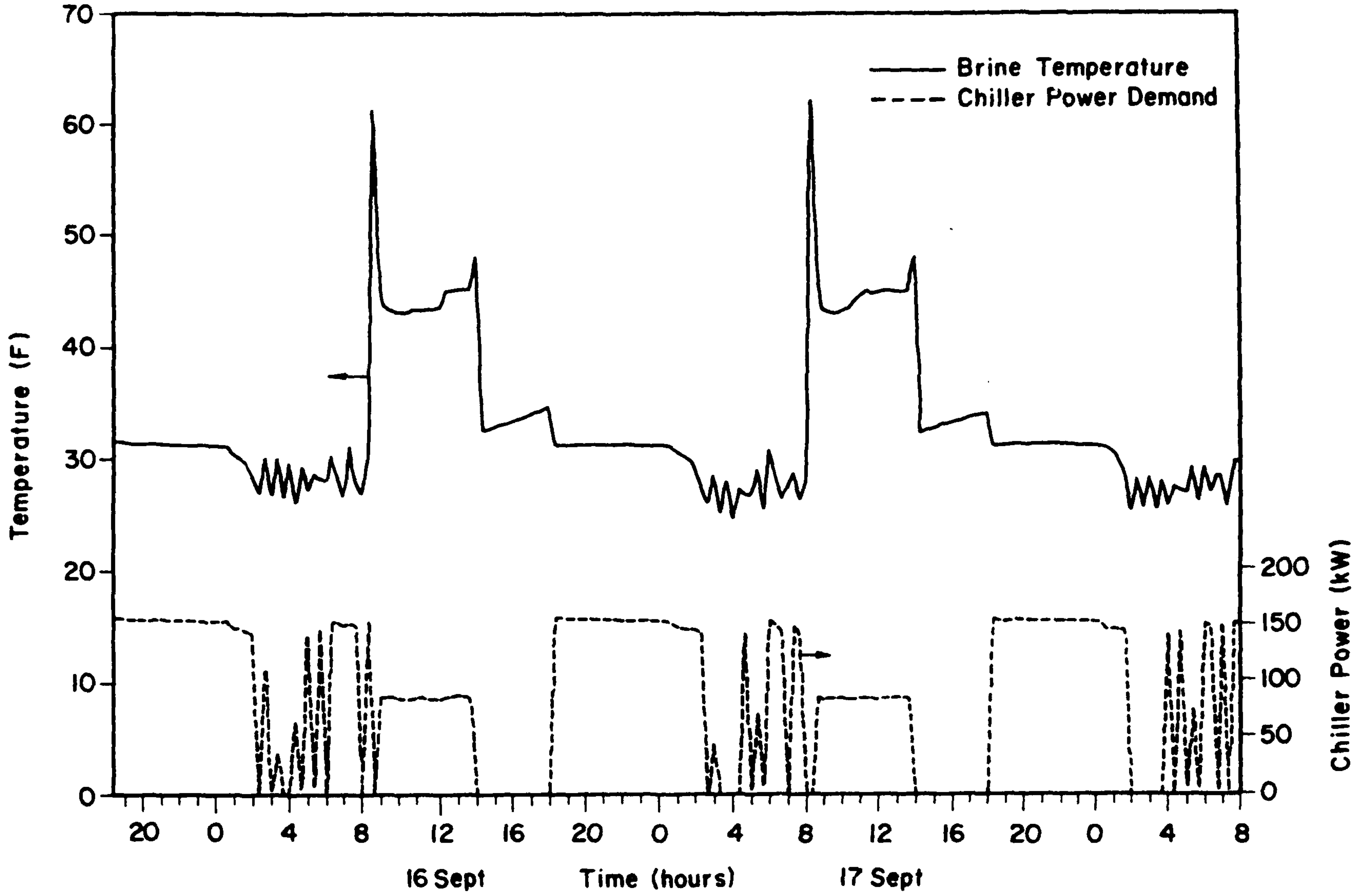

Figure 5. Brine temperature at the tank outlet and chiller power demand (16-17 September 1987).

3 SYSTEM PERFORMANCE

Instrumentation

Performance of the DIS cooling system was measured using two factors: (1) the ability to shift electrical demand for cooling from on-peak to off-peak periods and, (2) the energy consumption rate of the chiller compressor (kW/ton ratio) in cooling the building. The energy consumption rate of the DIS cooling system was compared with that of the conventional cooling system. The demand shift capability was measured by a power transducer (kW/kWh) installed on chiller CH-2. The compressor energy consumption rate (kW/ton) was measured by the power transducer, a flow meter, and two associated temperature sensors. It is the ratio of power consumed by the chiller (CH-2) versus cooling provided to the building which is measured by the flow meter (FE-1) and two temperature sensors (TE-8 and TE-9). The other quantities measured and sensors used are listed in Table 1.

Temperature was measured by type-T (copper-constantan) thermocouples which were sheathed in 1/8-in. stainless steel tubing and inserted into the flowing stream. The inside air temperature was measured in the return air duct of the main air handler. The outside air temperature was measured in a short section of PVC pipe equipped with a fan to supply fresh air. Flow rates were measured with vortex-shedding type flow meters that feature no moving parts. Electrical power and energy were measured with kW/kWh transducers equipped with appropriate current and potential transformers.

The signals from all sensors were transmitted to an Acurex Autodata 10 data logger (installed at the site) and recorded onto cassettes. Data from the DIS cooling system were taken every 15 min at the beginning, then at 20-min intervals. Recorded values of the temperature and electric power are the instantaneous values at the end of each time period. Recorded values of the flow rate and electrical energy are the averages for each time period. The recorded cassette data tapes were shipped to ORNL weekly for processing.

Data Reduction

The collected raw data were conditioned by ORNL to be legible through a PC version of BASIC program language. The conditioned ORNL data were mailed to USACERL on floppy diskettes and the primary data (temperatures, flow rates, and electrical power) were saved to a hard disk. These data were used to calculate the cooling delivered to the PX building by the DIS cooling system and the power consumed by the chillers to produce the refrigeration effect.

The cooling delivered to the building is given by:

$$Q = \int \dot{m}\ c_p\ (T_r - T_s)\ dt \qquad \text{[Eq 1]}$$

where the integration is taken over the time period of interest, Q is the cooling delivered during that period, T_r is the return temperature from the building (TE-9), T_s is the supply temperature (TE-8), and c_p is the specific heat of water.

The energy supplied to the compressor of the chiller during the same period is given by:

$$E = \int P\ dt \qquad \text{[Eq 2]}$$

where E is the total electrical energy in kWh and P is the power to the compressor (KW-2).

Table 1

Sensors Used for Instrumentation

Sensor	Location	Measured Parameter
Temperature		
TE-9	Upstream of HX-1	Building return water temp.
TE-8	Downstream of HX-1	Building supply water temp.
TE-7	Between chiller and ice bank	Chiller outlet brine temp.
TE-6	Between ice bank and 3-way valve	Ice bank outlet brine temp.
TE-5	Between HX and Chiller	Chiller inlet brine temp.
TE-3	In shaded open air	Ambient air temperature.
TE-2	Inlet to CH-1 (old chiller)	Building return water temp.
TE-1	Outlet of CH-1 (old chiller)	Building supply water temp.
Flow Rate		
FE-2	Between chiller and ice bank	Chiller loop flow rate
FE-1	In the load loop	Flow rate to air handler
Electrical Energy		
KW-3	At building elec. panel	Building elec. power (kW)
KH-3	At building elec. panel	Building elec. energy (kWh)
KW-2	At CH-2	CH-2 elec. power (kW)
KH-2	At CH-2	CH-2 elec. energy (kWh)
KW-1	At CH-1	CH-1 elec power (kW)
KH-1	At CH-1	CH-1 elec energy (kWh)

The collected data is discrete, i.e., measurements were taken every 15-min interval. Therefore, the integration operations in Equations 1 and 2 are replaced by the summations of the integrand for the time interval of interest.

Performance of the Conventional System

During the period 7 August to 17 October 1985, the performance of the 178-ton centrifugal chiller was measured. During that 72-day period, data were collected on the daily electrical energy input to the chiller (E) and the cooling provided to the building (Q) by the chiller. The daily data during that period are provided in Appendix A. The total electrical energy input to the chiller and the cooling delivered by the chiller were found to be 91,054 kWh and 110,008 ton-hr, respectively. Therefore, the seasonal average power consumption factor (PCF) of the 178-ton centrifugal chiller was:

$$PCF = E/Q \qquad [Eq\ 3]$$

$$= 0.83\ kW/ton$$

The daily average PCF of the chiller during that period ranged from 0.80 on 19 August 1985 to 1.02 on 7 October 1985. The variations in PCF are due to the loading factor for the chiller in meeting the building cooling load. Figure 6 shows the daily data for 19 August 1985. This was a hot day when the temperature reached the mid 90s. During the early morning and late evening hours, the building required about 40 tons to meet the cooling load and the system operated with a PCF of 0.92. During the day the chiller was more fully loaded. The building cooling requirements were about 120 tons and the PCF was about 0.76. The chiller was running in steady state during the building operating hours, thereby providing cooling at the lowest PCF.

In contrast, 7 October 1985 was rather cool day with outdoor temperatures only reaching the lower 70s. Data for this day are shown in Figure 7. In the early morning and late evening, the chiller was loaded very lightly, operating at an average of about 20 tons during this period. The corresponding PCF was quite high at 1.34. During the daytime hours the cooling load increased to about 50 tons on average and the PCF for this period was 0.84. This poor performance was due to the low loading factor which exacerbated the energy efficiency of the chiller.

Performance of the DIS Cooling System

DIS cooling system performance was monitored through the 1987 cooling season. Performance was studied through two parameters: the capability of electrical demand shifting from the on-peak windows to the off-peak hours and the energy consumption factor for the chiller.

Demand Shifting

During the 1987 cooling season, the DIS cooling system successfully shifted electrical demand from the on-peak windows to the off-peak hours according to the operation control logic as described under **System Operation and Control** in Chapter 2. A typical result of shifting is shown in Figure 8. The chiller provided cooling directly, drawing electrical power at the rate of 94 kW and bypassing the storage loop, during the morning hours 08:00 a.m. to 12:00 noon. During the on-peak windows 12:00 noon to 6:00 p.m., the chiller remained off, thereby shifting the electrical demand by at least 94 kW. During the night, starting at 6:00 p.m., the chiller resumed operation to provide night-time cooling as well as to freeze ice in the tanks. In the later part of the 1987 cooling season, the on-peak window had been reduced from 6 hr (12:00 noon to 6:00 p.m.) to 4 hr (12:30 p.m. to 4:30 p.m.) to accommodate the increased building nighttime cooling load as discussed in Chapter 2. The 4-hr window is still wide enough to catch the peak demand at Fort Stewart. Thus, the DIS cooling system successfully shifted at the minimum of 94 kW of the Fort Stewart peak demand.

Energy Performance of DIS Cooling System

For the DIS cooling system, the daily PCF of the chiller is difficult to determine. The problem is that the ice produced during the previous night may not be fully harvested during the next day, and if ice remains in the tank after the daytime cooling, the chiller does not have to provide all the refrigeration needed for freezing water as is the case when the tank is full of water only. Therefore, the PCF for the ice system must be measured over an extended test period so as to minimize the carryover effect of remaining ice. During the period 29 May to 17 June 1987, a set of continuous operating data was collected. Table 2 shows the daily data during this period.

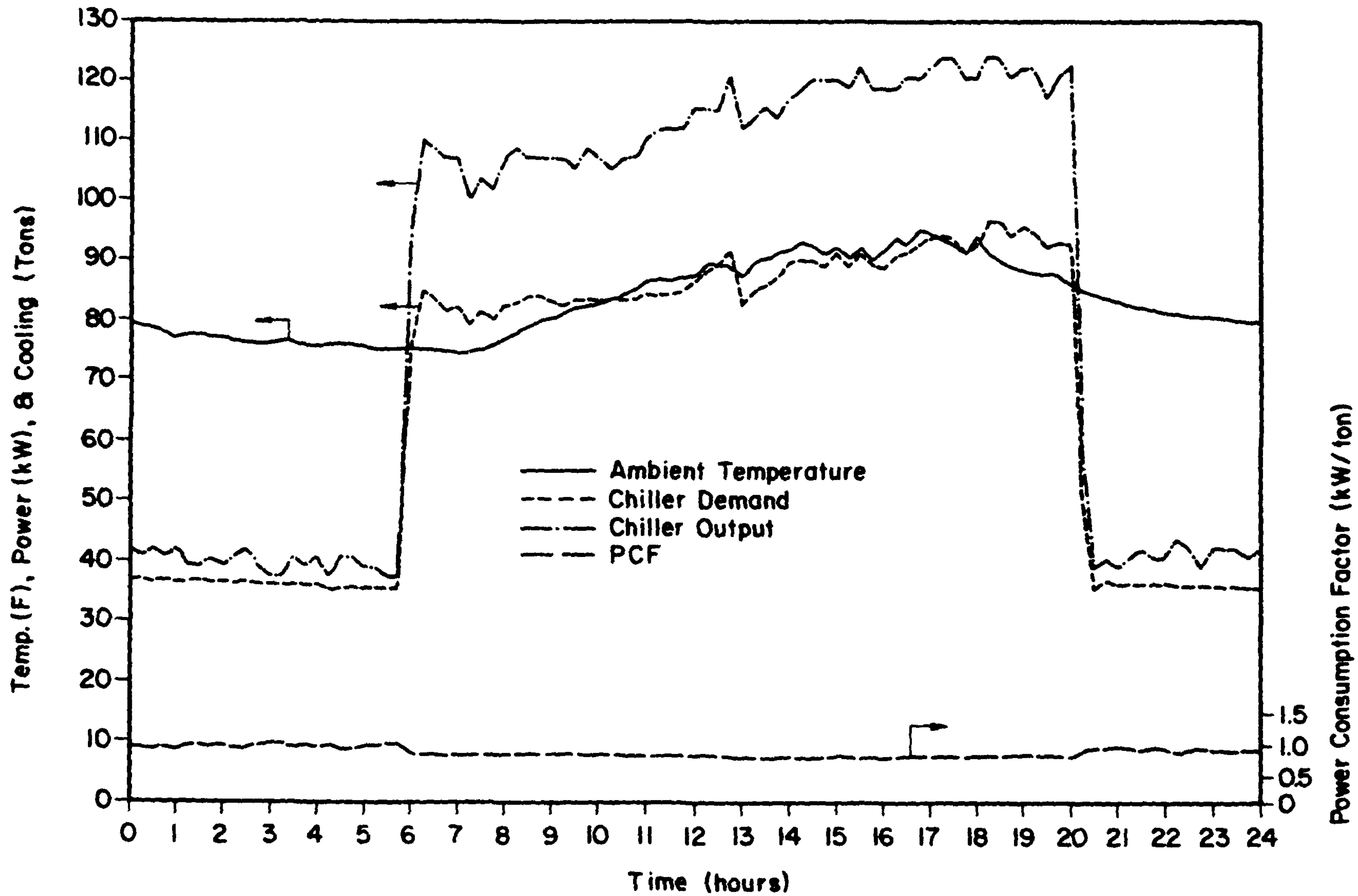

Figure 6. Fort Stewart's conventional chiller performance on 19 August 1985.

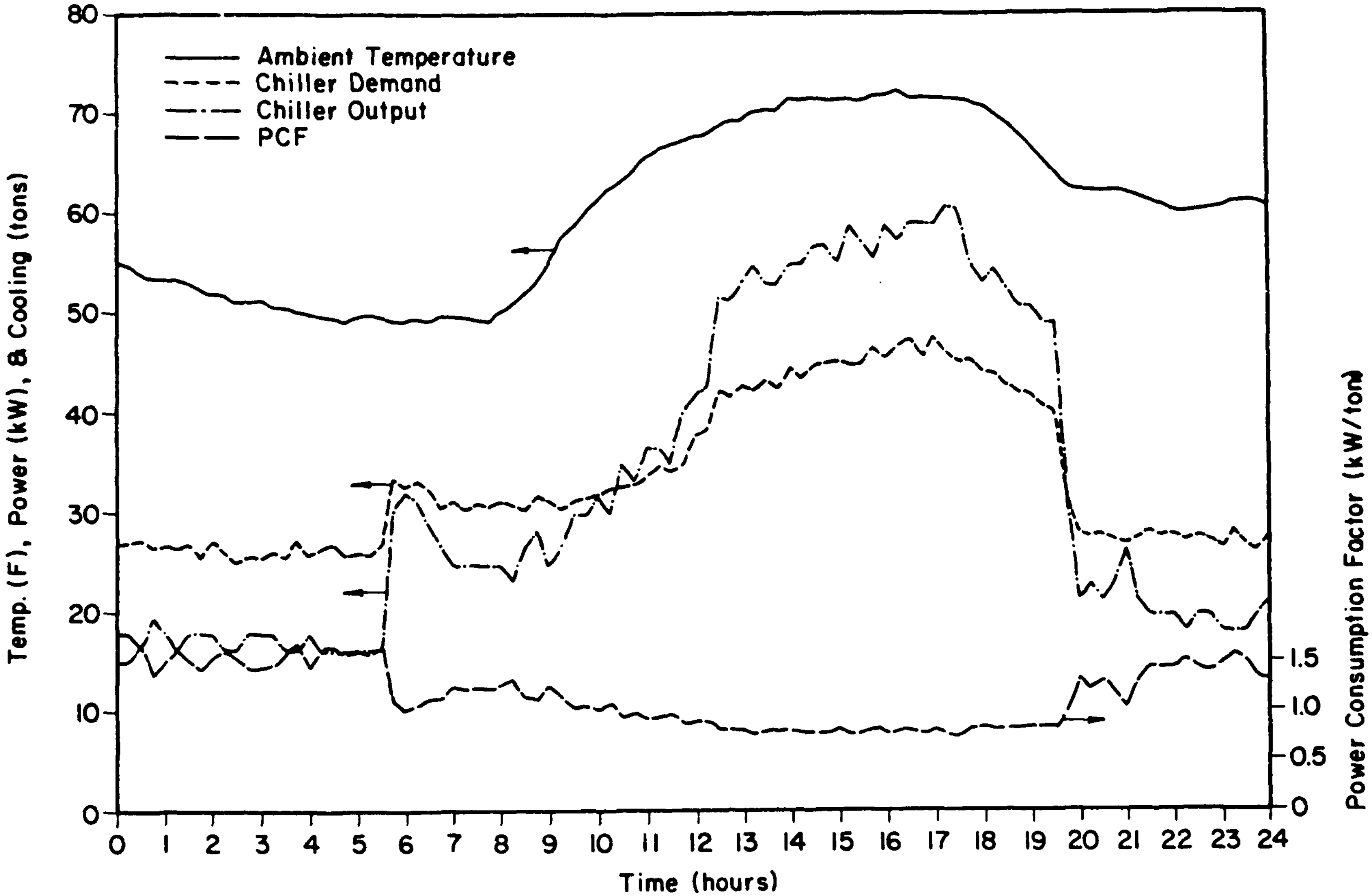

Figure 7. Fort Stewart's conventional chiller performance on 7 October 1985.

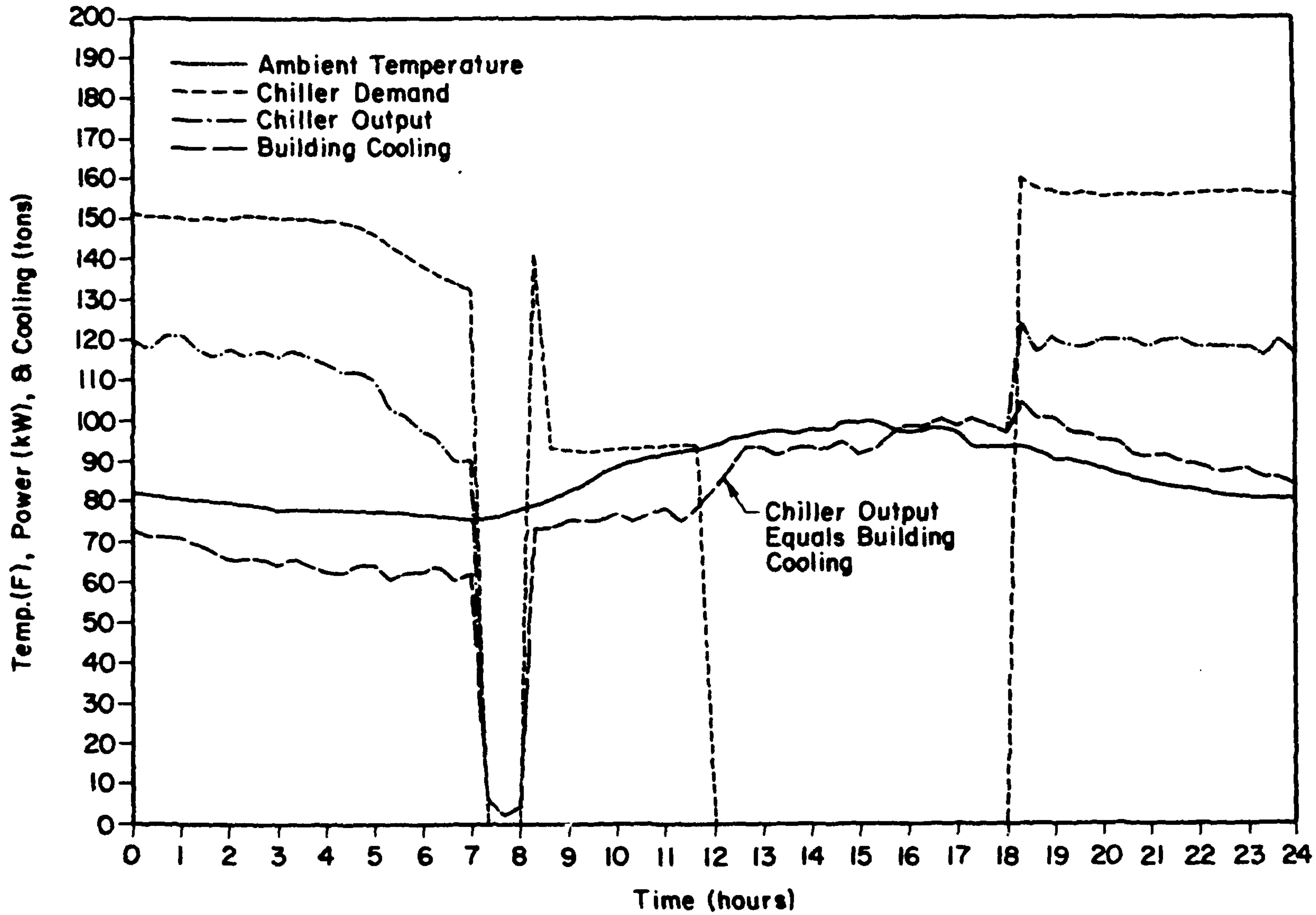

Figure 8. Typical Fort Stewart DIS cooling system performance.

Table 2

DIS Cooling System Performance During May and June 1987*

			Qstorage		Bldg. Load			Chiller Energy			Chil COP		Chil PCF	
DAY	Tmax amb.	Tmax indoor	1	3	1	2	3	1	2	3	1	2	1	2
29	89	80	473	640	972	368	616	1824	363	0	2.79	3.56	1.26	0.99
30	91	80	649	647	926	367	596	1862	361	0	2.97	3.58	1.18	0.98
31	94	81	333	572	1192	339	665	1829	301	0	2.93	3.97	1.20	0.89
1	96	82	121	620	1469	370	552	1828	374	0	3.06	3.48	1.15	1.01
2	96	82	545	618	1059	411	602	1904	397	0	2.96	3.64	1.19	0.97
3	99	84	542	603	1105	420	559	1893	400	0	3.06	3.69	1.15	0.95
4	94	85	547	455	1070	409	445	1883	405	0	3.02	3.55	1.16	0.99
5	87	81	806	490	765	374	524	1854	369	0	2.98	3.56	1.18	0.99
6	88	80	618	598	942	297	526	1882	280	0	2.92	3.73	1.21	0.94
7	90	79	746	605	745	304	587	1810	282	0	2.90	3.79	1.21	0.93
8	91	80	461	582	1006	367	569	1792	312	0	2.88	4.14	1.22	0.85
9	94	81	474	256	1012	342	305	1788	290	0	2.92	4.15	1.20	0.85
10	96	81	519	593	1114	374	530	1829	369	0	3.14	3.56	1.12	0.99
11	93	81	853	511	708	386	490	1829	375	0	3.00	3.62	1.17	0.97
12	98	82	652	606	905	427	528	1878	408	0	2.92	3.68	1.21	0.96
13	91	82	516	681	1068	409	619	1910	404	0	2.92	3.56	1.21	0.99
14	82	78	640	760	941	359	682	1906	365	0	2.92	3.46	1.21	1.02
15	95	80	432	750	1037	378	681	1861	386	0	2.78	3.44	1.27	1.02
16	95	82	519	494	1060	403	507	1923	407	0	2.89	3.48	1.22	1.01
17	94	83	550	529	1063	425	506	1937	409	0	2.93	3.66	1.20	0.96
total			10998	11612	20157	7529	11092	37220	7257	0				
ave.	93	81	550	581	1008	376	555	1861	363	0	2.94	3.65	1.19	0.96

*1: ice production and night-time cooling; 2: direct cooling in the morning; 3: cooling by ice during on-peak window.

During the 20-day test period, the chiller consumed 7257 kWh and 37,220 kWh of energy during the morning (mode 1) and nighttime periods (mode 2), respectively. Meanwhile, the cooling delivered to the building was 7529 ton-hr during the morning periods (mode 1) and 31,249 ton-hr during the window and the nighttime. Therefore, the average PCF for the direct cooling mode (mode 1) and the storage cooling mode are:

$$PCF_d = 7257/7529$$

$$= 0.96 \text{ (kW/ton)}$$

$$PCF_s = 37{,}220/31{,}249$$

$$= 1.19 \text{ (kW/ton)}$$

The complete set of daily DIS cooling system performance data is provided in Appendix B.

4 DISCUSSION

System Energy Performance

The power consumption factors can be compared between the DIS cooling system and the conventional chiller as follows:

Conventional System	DIS Cooling System
178-ton Centrifugal Chiller	200-ton Reciprocating Chiller
0.83	0.96 (Direct Cooling)
	1.19 (Storage Cooling)

These data show that the DIS cooling system consumed more energy than the conventional system in providing the same amount of cooling. This result is generally true because of a lower suction temperature for the compressor in the ice-making operation. (The effect of lower suction temperature is discussed in detail below.) However, the dramatic difference in PCFs between the conventional and DIS cooling systems warrants further clarification.

The important difference is that the conventional system uses a centrifugal chiller whereas the DIS cooling system is equipped with a reciprocating chiller. Typically, a centrifugal chiller is more energy-efficient than a reciprocating chiller in the 150-ton capacity range. Therefore, a direct comparison of the power consumption factors listed above cannot provide unbiased results on the DIS cooling system's energy efficiency.

A more meaningful comparison can be made using PCFs between the direct cooling mode (0.96 kW/ton) and the storage cooling mode (1.19 kW/ton). In the direct cooling mode, the chiller was running at normal evaporator temperature (42 °F--see Figure 9) to provide chilled water directly to the building. The penalty for the ice storage system is due to the lowered evaporator temperature to supply low-temperature brine (25 °F--see Figure 9) to the ice storage tanks. According to the power consumption factors listed above, the penalty for ice storage cooling was 19 percent extra power.

A theoretical estimate of the penalty for running the chiller at a lower evaporator temperature can be made in two ways: (1) a change in chiller coefficient of performance (COP) based on typical R-22 compressor ratings and engineering data from a manufacturer for several operating conditions, and (2) a change in chiller COP based on conventional and ice-making operating conditions based on R-22 p-h diagram. An isentropic compression and an isenthalphic expansion were assumed in the p-h diagram analysis. In both the cases, the following operation conditions were assumed:

Conventional: 110 °F condensing temperature with 40 °F saturated suction temperature.

Ice-making: 90 °F condensing temperature with 20 °F saturated suction temperature.

Table 3 lists compressor ratings and engineering data for a typical reciprocating compressor running on R-22 refrigerant. For this machine, the power consumption of the compressor for the ice-making mode (20 °F suction and 90 °F condensing) is 0.99 ton/kW (71.5 ton/72 kW), and 1.05 ton/kW (97.3 ton/93 kW) for the conventional operating mode (40 °F suction and 110 °F condensing); these data are underlined in Table 3. The COP of the compressor is a dimensionless number that can be obtained by multiplying the power consumption factor (ton/kW) by a normalizing factor of 3.516 (1 ton = 3.516 kW). The COP of this compressor was 3.48 and 3.69 for ice-making and conventional operations, respectively. The energy penalty due to the lowered suction temperature is therefore 6 percent.

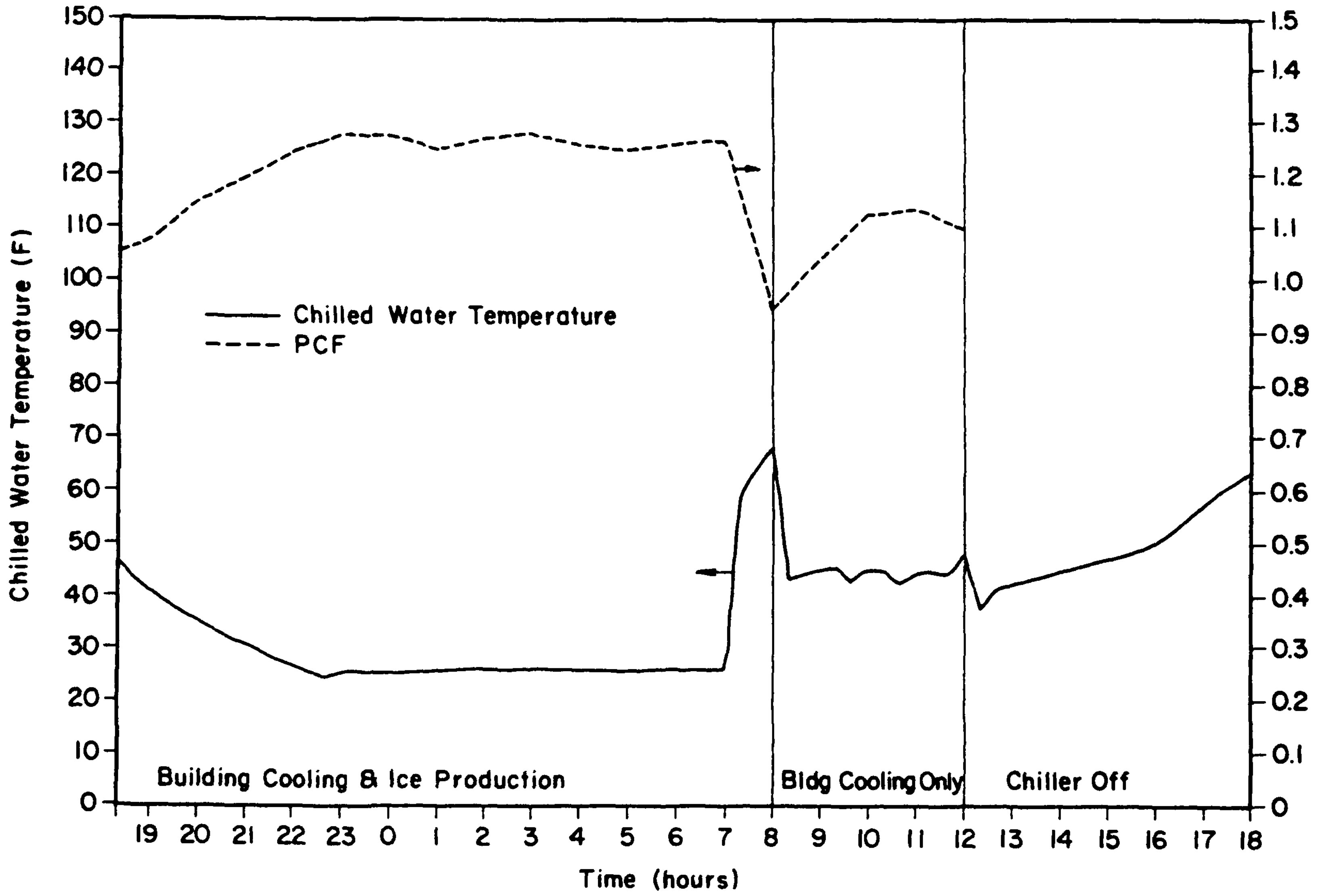

Figure 9. DIS system performance during direct cooling mode.

Table 3

Compressor Ratings and Engineering Data for a Typical R-22 Unit (tons/kW)

Condensing Temperature	Saturated Suction Temperature (°F) 20	30	40	50
80	76.4/66.1	95.1/67.9	116.9/69.2	141.9/70.4
90	71.5/72.0	89.5/75.2	110.5/77.0	134.6/78.5
100	66.4/77.1	83.8/82.0	104.0/85.1	127.2/87.0
105	64.0/79.3	80.9/85.0	100.7/89.0	123.4/91.6
110	61.5/81.3	77.9/88.1	97.3/93.0	119.6/96.2
120	56.4/84.5	72.1/93.3	90.5/100.2	111.9/105.2

A p-h diagram of R-22 with prescribed operating conditions is shown in Figure 10. In conventional operating mode, it takes 12 Btu of energy to compress 1 lb of saturated R-22 vapor, and the same pound of liquid refrigerant will remove 66 Btu of heat from the evaporator. The COP of this cycle is 5.5 (66 Btu/12 Btu). In the ice-making mode, the compression work is 14 Btu, and the refrigeration effect at the evaporator is 70 Btu. The COP in ice making mode is 5.0 (70 Btu/14 Btu). Therefore, the reduction of COP due to the lower suction temperature is calculated to be roughly 9 percent.

The calculated COPs based on the p-h diagram analysis are higher than those in the manufacturer's table. This is expected since the analysis based on the p-h diagram assumes isentropic compression (which is not true for a real compressor) and no superheating (i.e., at the evaporator outlet the refrigerant is assumed to be saturated vapor). The more important consideration is the degree of penalty (a ratio of reduction in COP) due to a lowered suction temperature for ice making. In both theoretical analyses, a penalty of roughly 10 percent reduction in COP was expected.

For the Fort Stewart DIS cooling system, the measured reduction in COP during ice-making mode was 19 percent. In the experiments, the refrigeration effect was measured at the delivery point to the building, not at the chiller outlet. The measured refrigeration effect may include heat gains through the piping, ice tank, and heat exchanger. Therefore, the measured reduction of COP in the ice-making mode should be higher than those from the theoretical analysis. Another source of discrepancy in COP reductions between the predicted and measured values would be the uncertainties in system operating conditions. As can be seen in Table 3, a slight variation in operating conditions (suction and condensing temperature) could result in appreciable changes in compressor COP. Taking into account these considerations, the measured data are in good agreement with the predicted performance.

Benefits

The objective of a DIS cooling system is to shift electrical power demand required for space cooling from on-peak to off-peak periods (typically from day to night), thereby achieving savings in electrical

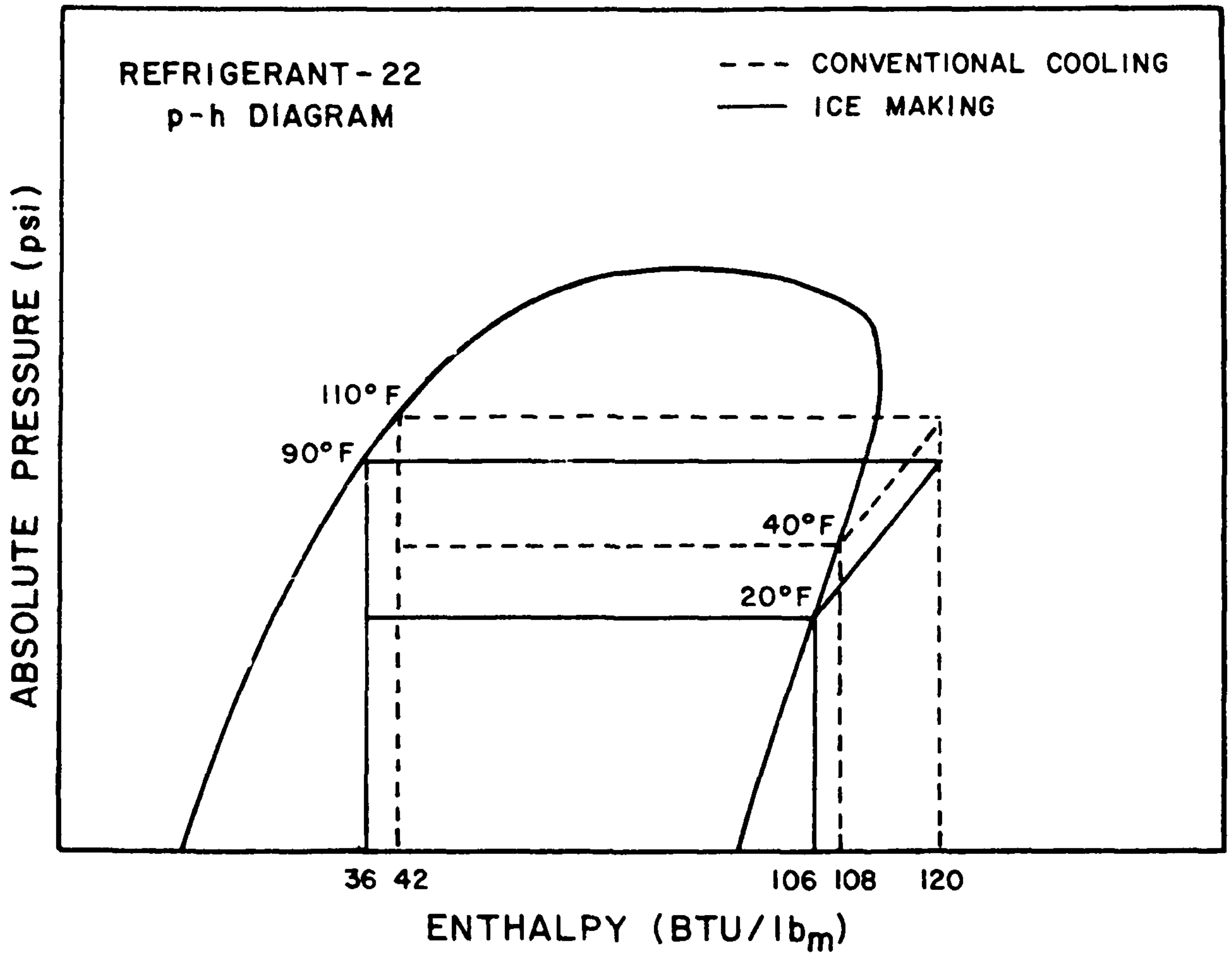

Figure 10. P-h diagram for R-22 under prescribed operating conditions.

utility costs for air-conditioning. The demand shifted by the DIS cooling system is evident by comparing Figures 11 and 12. Figure 11 shows the electrical demand of the conventional chiller for a typical summer day. The chiller is fully loaded from 6:00 a.m. to 8:30 p.m. while the building needs cooling. Fort Stewart's peak demand occurs around 2:30 p.m. when the chiller is on and contributing to the demand. Figure 12 shows the same information for the DIS cooling system. Note that the chiller is off between 12:00 noon and 6:00 p.m., thereby reducing Fort Stewart's peak demand by 120 kW. With the current electrical rate schedule at Fort Stewart, reduction of the peak demand by 1 kW translates into $70 of savings in electrical utility cost every year.

Operation and Maintenance

The ice-in-tank DIS cooling system installed in the PX building at Fort Stewart was the first ice system ever built for an Army facility. Like most first-in-kind systems, it experienced several interruptions in operation. The causes of the problems, however, were trivial. The unexpected expansion of the PX building added another problem by increasing the nighttime cooling load from 40 tons (design value) to 80 tons. Adjustment of the on-peak window to resolve this problem was discussed earlier.

A more annoying problem was leakage in the brine loop. One of the few routine inspections of system operation is to check the brine loop pressure with a pressure gauge installed at the suction side of the brine circulating pump (P-1 and P-2 in Figure 1). The operator was instructed to shut down the operation of the system if the gauge pressure reads below 30 psig. An expansion tank (ET-1 in Figure 1) is installed to maintain the brine loop pressure at or above 30 psig. This pressure is maintained to prevent cavitation in the circulation pump.

The leakage occurred at the connection between the tanks and the main supply and return headers (see Figures 2 and 3). The L-shape hose and fittings connecting the header and the leaking tank have been replaced by Fort Stewart personnel. During replacement of the hose, a few gallons of brine were drained. The system was replenished with brine to bring the pressure up to 30 psig. Later, the system pressure dropped below 30 psig and operation was interrupted. While the DIS cooling system was undergoing repair, the conventional cooling system (CH-1 loop in Figure 1) was reactivated to cool the building. The data in Appendix B show the DIS system's down period (a day with chiller CH-1 input data other than zero corresponds to the repair period). These data show that the system was down 31 days during the 1987 cooling season from 1 May 1987 to 30 October 1987. Repairs often were delayed by the lack of personnel as well as difficulty in procuring makeup ethylene glycol. The relatively long period of system downtime was mainly caused by logistical problems rather than technical difficulties. Since the system can be easily converted back to the previous cooling system (by closing valves V-6 and V-7 and opening valves V-8 and V-9--see Figure 1), the building cooling service was not interrupted. This easy remedy may have eliminated a pressing need to service the system.

Another problem associated with the brine leakage was recharging the brine back to the system. It was learned that, on several occasions, the operator charged the system without bleeding the expansion tank (ET-1). When the system was accepted from the contractor, the brine loop was pressurized to 30 psig. This pressurization was done in two steps: first, the loop was filled with brine while bleeding out air in the loop through bleeders installed at the top of supply and return main headers (Figure 3). Second, the loop was sealed and the expansion tank (ET-1) pressurized to 30 psig. Unless the expansion tank bladder pressure is relieved before filling the loop, recharging the system by simply refilling it will not achieve the desired system pressure. Instructions on the proper sequence of recharging the system were developed and left with the system operator. This sequence is also included as Appendix C.

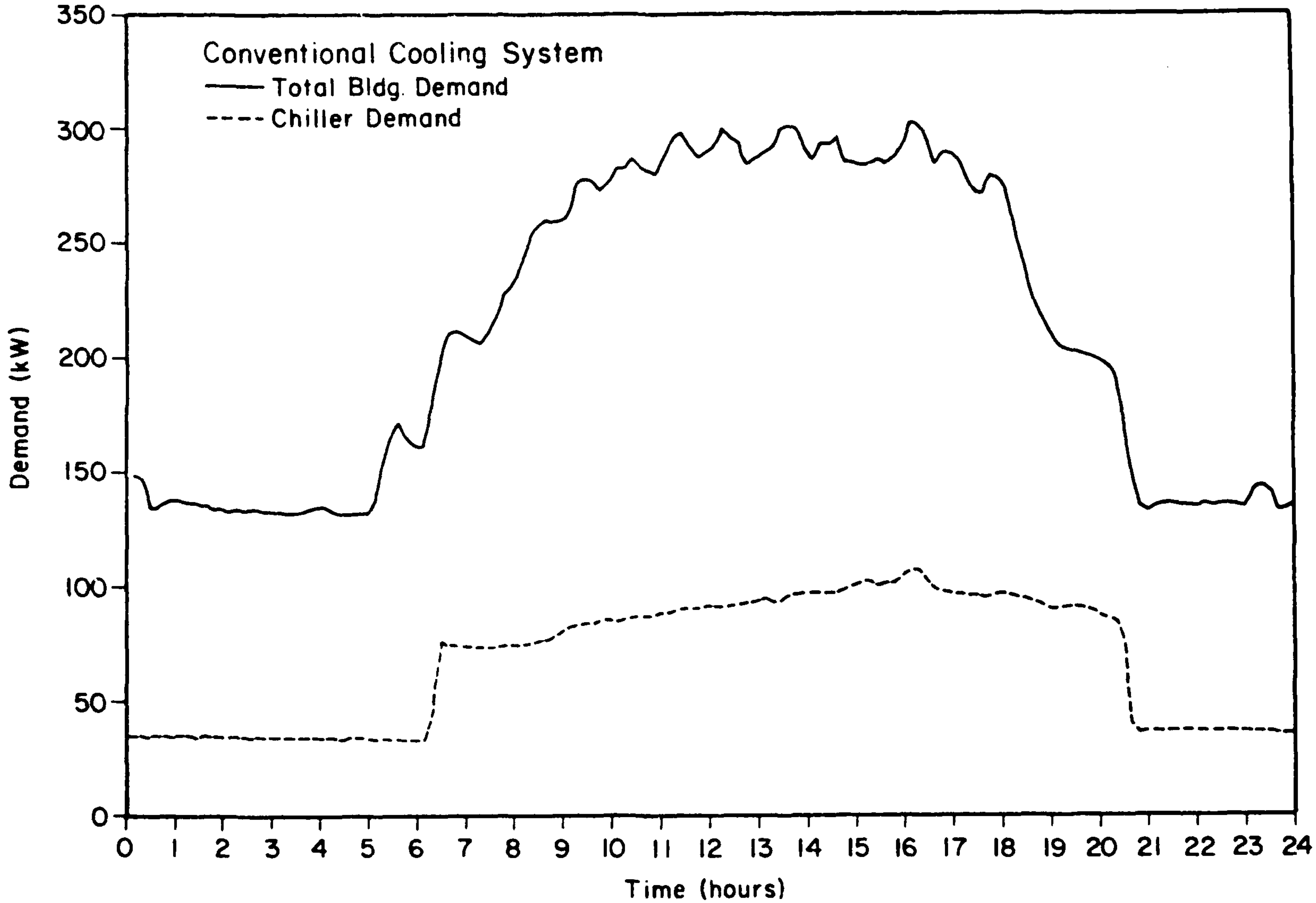

Figure 11. Electrical demand for conventional chiller on a typical summer day.

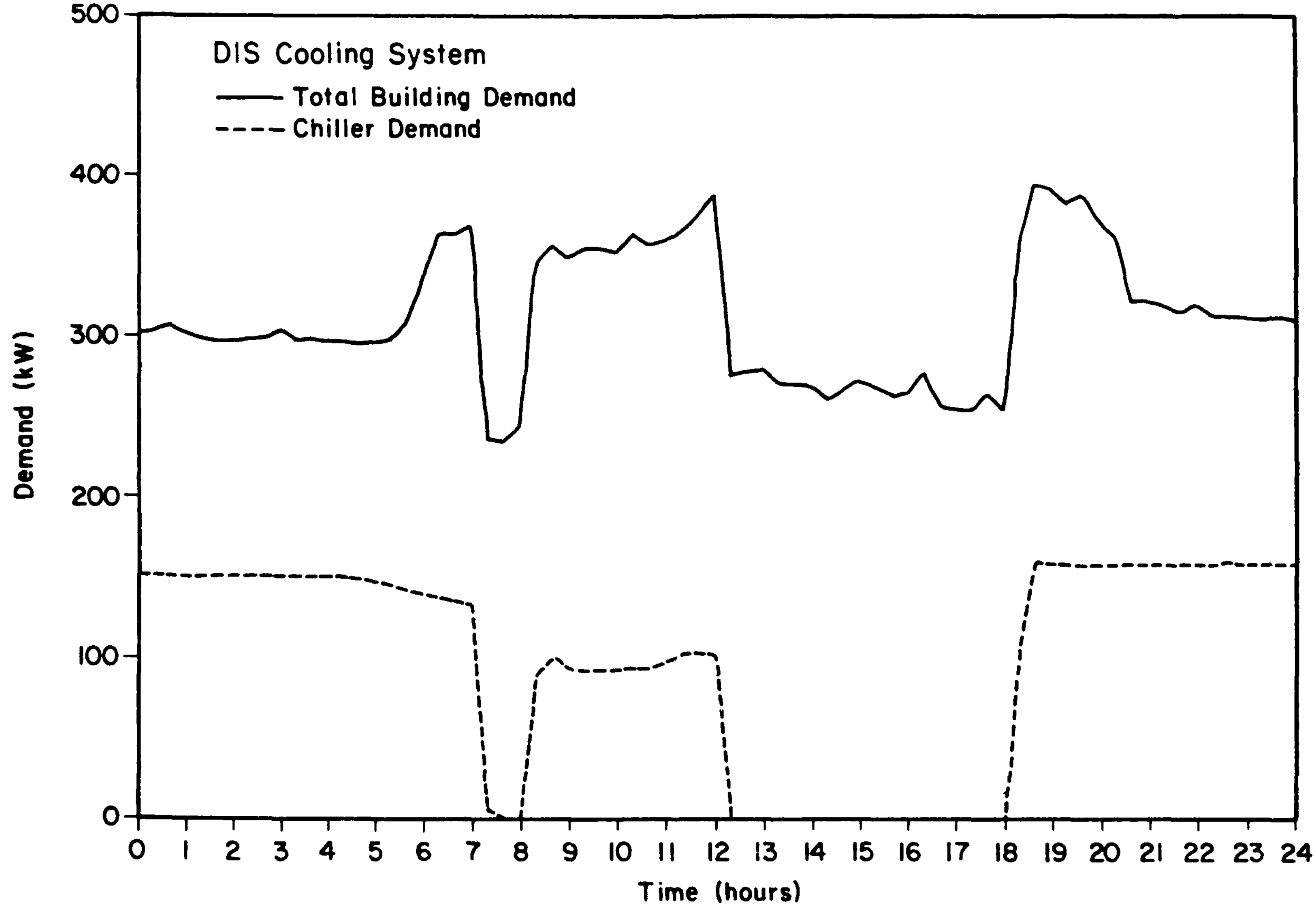

Figure 12. Electrical demand for DIS system on a typical summer day.

Lessons Learned

The DIS cooling successfully shifted electrical demand for cooling the building from the on-peak window to off-peak hours. It proved to be an effective method of electrical demand management.

The energy penalty observed in this DIS cooling system due to lowered suction temperature is 19 percent, which is relatively minor. However, compared with the centrifugal chiller system, the penalty is 43 percent, which is rather substantial. The literature reports that a cold air system (less than 40 °F supply air) can be as competitive as a centrifugal chiller-based conventional system in terms of overall energy consumption. Moreover, a chilled *water* storage cooling system, which does not have the low suction temperature effect, can be an alternative if the energy penalty is of concern.

Maintenance of the system requires no more attention than a conventional cooling system. The leak in the brine loop is a rather routine occurrence in piping systems. In October 1988, the tank manufacturer's representative inspected the system and informed USACERL that the header assemblies are available as an off-the-shelf factory prepackaged unit. Furthermore, since late 1988, prepackaged complete DIS cooling systems have become available from several manufacturers. This purchase option will help eliminate problems in field plumbing and the associated maintenance. The factory prepackaged systems are discussed further in the next section.

The brine leakage was a minor design rather than a maintenance problem. In the design, the supply and main headers over the tanks were supported by a steel angle structure that was intended to withstand the vertical load only. During startup, slight lateral movement of the main headers was observed due to the momentum of initial fluid flow. The angle structure can be reinforced with X-members between the vertical support structure. Again, a factory prepackaged system will eliminate this kind of minor problem.

Factory Prepackaged Systems

Quite recently (1988), prepackaged ice storage cooling systems became available on the market.[2] This is a major development in DIS cooling technology. For the system installed at Fort Stewart, the ice maker, storage tank, pump, and other associated hardware as well as the system controller were purchased from different vendors. Then the components were assembled by a general contractor. This approach required not only extra work in system design but also a very significant extra cost for system integration by the installation contractor. The enforcement of a system warranty is also complicated due to the multiplicity of equipment suppliers. Prepackaged units will avoid these problems.

For the Fort Stewart system, the installation cost ($83,900) for the general contractor represented about 40 percent of the total system cost. That portion can be greatly reduced when prepackaged systems (PPS) are used. All system assembly works will be completed at the factory. The only work required in the field will be to connect the unit to the existing chilled water supply and return mains and to the cooling tower loop, which can be done easily by in-house labor.

Another advantage of PPS will be the elimination of maintenance problems. As discussed earlier, the major trouble with the Fort Stewart system has been the leakage of brine at the connections between the ice tank and main headers. This problem originated from the assembly procedures and the system support design. Such problems would be eliminated with factory prepackaged systems.

The most important benefit of the PPS will be the single source responsibility. With this approach, the factory should provide a warranty covering not only the quality of the equipment, but also the workmanship of the system assembly. Elimination of multiple equipment suppliers will also expedite maintenance activities (e.g., one telephone call to the system manufacturer rather than trying to track a number of equipment manufacturers who may tend to "pass the buck").

[2] *Energy User News* (February 1989).

5 CONCLUSIONS AND RECOMMENDATIONS

The daily performance data for an ice-in-tank DIS cooling system at the PX building, Fort Stewart, GA were collected during the 1987 cooling season (Appendix B). The system energy consumption factor in direct cooling mode (for the conventional cooling operation) was measured as 0.96 kW/ton for the cooling delivered to the building. In the ice storage cooling mode, it was measured as 1.19 kW/ton. The energy penalty for the ice storage cooling operation (due to lowered evaporator temperature from 42 °F conventional to 25 °F ice making) is 19 percent. The theoretical estimate of the penalty (excluding the heat gain from the ice tank, heat exchanger, and piping) is 9 percent, which supports the reliability of the measured data.

While the DIS cooling system was operating, it successfully shifted electrical demand for building cooling from the on-peak windows (12:00 noon to 6:00 p.m. until 5 August 1987, and 1:00 p.m. to 5:00 p.m. thereafter) to off-peak hours. Therefore, the system's ability to shift demand has been verified as successful.

The only maintenance problem encountered during the 1987 cooling season was leakage of brine from the connections between the ice storage tank and the supply/return main header. This failure was caused by inadequate design of the support structure for the header assembly. The design did not take into account the lateral movement of the header resulting from the momentum of brine flow inside the header during the start period. Reinforcement of the supporting structure with cross members would eliminate the cause of the brine leakage.

Except for the problem of brine leakage, the system did not require any additional attention for operation and maintenance. The system operating mode was controlled by the timer clock with no operator assistance. The clock was adjusted once to change the on-peak window from 6 hr to 4 hr. The brine leakage was the result of inadequate design, and was not due to the system's operation and maintenance characteristics. The system at Fort Stewart was installed during the early stages of development for the DIS. Since then, the header assembly as well as supports became available as a factory prepackaged unit from the tank manufacturer. Therefore, similar shortcomings in accessory design would be eliminated for future systems. Furthermore, factory prepackaged DIS cooling systems became available in 1988. Such units will drastically eliminate most of the field plumbing requirements, thereby ensuring the integrity of the DIS cooling system.

Factory prepackaged DIS cooling systems represent a major development in the storage cooling industry. It has a strong potential to reduce the system construction cost by as much as 30 percent (based on the system installation cost at Fort Stewart). It will also greatly reduce the potential for maintenance problems associated with field installation of the systems. An analogy is the difference between buying a factory built automobile and assembling the vehicle with separately purchased engine, frame, and chassis.

Several different types of DIS cooling systems are currently available; the ice-in-tank type described here is just one example. Two other popular DIS cooling systems are being demonstrated by USACERL (an ice-on-coil system at Yuma Proving Ground, AZ and an ice harvester system at Fort Bliss, TX). Technical reports will be published in FY90 and FY91 to document these tests.

The DIS cooling system was demonstrated to be an effective tool to reduce electrical demand charges. It is recommended that installations incurring high charges during peak usage consider obtaining a DIS system. Guidance for estimating the potential benefit of storage cooling systems is available in two

USACERL Technical Reports.[3] If the benefit in demand charge reduction is estimated to be more than \$100/year for each kilowatt shifted from on-peak to off-peak periods, the DIS cooling system will have a 3-year payback.

If the installation decides to purchase a DIS system, it is recommended that the market be surveyed to determine if a factory prepackaged unit is available.

METRIC CONVERSION FACTORS

1 psig	=	6.895 kPa
1 lb	=	0.453 kg
1 Btu	=	1.055 kJ
1 kWh	=	3.6 MJ
1 sq ft	=	0.092 m^2
1 ton cooling ≈ 12000 Btu/hr	=	3.517 kW
1 ton-hour cooling = 12000 Btu	=	3.517 kWh
1 °F	=	1.8 (°C) + 32
1 in.	=	25.4 mm
1 gal	=	$3.8 \times 10^{-3} m^3$

[3] C.W. Sohn and J. Tomlinson; C. W. Sohn, *Market Potential of Storage Cooling Systems in the Army* (USACERL Technical Report E-89/13/ADA213977, September 1989).

APPENDIX A:

PERFORMANCE DATA FOR THE ORIGINAL CHILLER

August 1985

DAY	Tamb. ave(F)	chiller (kWh)	cooling (t-h)	PCF
7	75.9	659	817	0.81
8	74.2	1299	1564	0.83
9	76.8	1467	1801	0.81
10	79.5	1470	1807	0.81
11	81.1	1375	1636	0.84
12	82.1	1410	1726	0.82
13	80.6	1614	1967	0.82
14	81.3	1511	1881	0.80
15	82.0	1577	1866	0.85
16	83.3	1603	1958	0.82
17	83.5	1528	1913	0.80
18	83.4	828	1134	0.73
19	83.2	1604	2008	0.80
20	82.1	1640	2035	0.81
21	82.9	1440	1787	0.81
22	82.6	1591	1976	0.80
23	78.4	1645	2018	0.82
24	77.1	1511	1851	0.82
25	79.5	1452	1720	0.84
26	77.7	1018	1254	0.81
27	78.1	756	1048	0.72
28	77.9	1264	1608	0.79
29	77.1	1552	1865	0.83
30	76.5	1479	1802	0.82
31	74.4	1290	1577	0.82
total		34580	42617	
average	79.6	1383	1705	0.81

September 1985

DAY	Tamb. ave(F)	chiller (kWh)	cooling (t-h)	PCF
1	77.3	1266	1510	0.84
2	81.3	1457	1764	0.83
3	81.1	1540	1866	0.83
4	80.9	1519	1836	0.83
5	79.2	1464	1774	0.83
6	80.5	1218	1558	0.78
7	81.8	1590	1956	0.81
8	82.3	1481	1782	0.83
9	83.1	1613	1983	0.81
10	84.0	1637	2028	0.81
11	84.4	1607	1977	0.81
12	78.6	1478	1794	0.82
13	72.5	1197	1473	0.81
14	64.5	875	898	0.97
15	69.1	883	905	0.98
16	70.5	992	1121	0.88
17	72.3	1067	1255	0.85
18	74.1	1124	1354	0.83
19	73.1	1165	1372	0.85
20	74.6	1245	1494	0.83
21	74.9	1200	1431	0.84
22	75.4	1139	1327	0.86
23	75.2	1194	1424	0.84
24	78.4	1323	1596	0.83
25	76.3	1369	1676	0.82
26	76.3	1248	1535	0.81
27	72.0	1050	1251	0.84
28	64.7	884	943	0.94
29	70.2	974	1114	0.87
30	73.9	1179	1444	0.82
total		37978	45439	
average	76.1	1266	1515	0.84

October 1985

DAY	Tamb. ave(F)	chiller (kWh)	cooling (t-h)	PCF
1	76.1	1345	1639	0.82
2	72.7	1247	1519	0.82
3	77.9	1402	1663	0.84
4	80.0	1501	1848	0.81
5	75.7	1277	1559	0.82
6	60.6	807	790	1.02
7	60.3	810	783	1.03
8	66.1	930	1011	0.92
9	72.8	1082	1291	0.84
10	72.3	987	1166	0.85
11	75.4	1006	1248	0.81
12	73.8	1225	1493	0.82
13	74.6	636	835	0.76
14	76.0	1200	1458	0.82
15	77.6	1322	1595	0.83
16	78.0	1375	1680	0.82
17	73.0	346	375	0.92
total		18496	21952	
average	73.1	1088	1291	0.84

APPENDIX B:

PERFORMANCE DATA FOR THE DIS SYSTEM

May 1987

DAY	Tmax amb.	Tmax indoor	Qstorage 1	Qstorage 3	Bldg. Load 1	Bldg. Load 2	Bldg. Load 3	Chiller Energy 1	Chiller Energy 2	Chiller Energy 3	Chil COP 1	Chil COP 2	Chil PCF 1	Chil PCF 2
7	85	80	0	58	0	0	68	0	0	0	----	----	----	----
8	88	82	636	523	492	203	430	1440	43	0	2.76	16.75	1.28	0.21
9	85	82	867	244	500	158	244	1526	22	0	3.15	24.73	1.12	0.14
10	82	79	131	260	558	115	245	916	4	0	2.65	107.96	1.33	0.03
11	85	81	263	247	469	222	242	907	34	0	2.84	22.65	1.24	0.16
12	83	83	140	189	653	225	187	872	141	0	3.20	5.60	1.10	0.63
13	80	83	232	164	607	230	157	843	196	0	3.50	4.12	1.00	0.85
14	81	81	243	185	593	200	178	841	218	0	3.49	3.22	1.01	1.09
15	86	81	723	378	832	281	416	1789	57	0	3.06	17.30	1.15	0.20
16	90	81	925	377	701	189	399	1819	84	0	3.14	7.93	1.12	0.44
17	85	81	-128	0	147	0	0	8	0	0	7.51	----	0.47	----
18	91	82	0	0	0	20	54	0	0	0	----	----	----	0.00
19	89	82	0	0	-49	-23	-106	0	0	0	----	----	0.00	0.00
20	91	80	0	0	0	0	0	0	0	0	----	----	----	----
21	87	80	0	0	-463	-117	-223	0	0	0	----	----	0.00	0.00
22	86	80	0	-369	-546	-128	434	0	0	801	----	----	0.00	0.00
23	89	80	576	330	916	351	349	1846	137	0	2.84	8.97	1.24	0.39
24	93	81	665	293	826	331	317	1879	99	0	2.79	11.74	1.26	0.30
25	90	81	403	346	1124	269	358	1866	79	0	2.88	11.96	1.22	0.29
26	90	81	526	420	912	279	407	1868	97	0	2.71	10.13	1.30	0.35
27	87	81	802	451	802	344	377	1847	99	0	3.05	12.18	1.15	0.29
28	88	80	596	157	940	200	143	1867	73	0	2.89	9.67	1.22	0.36
29	89	80	473	640	972	368	616	1824	363	0	2.79	3.56	1.26	0.99
30	91	80	649	647	926	367	596	1862	361	0	2.97	3.58	1.18	0.98
31	94	81	333	572	1192	339	665	1829	301	0	2.93	3.97	1.20	0.89
total			9054	6110	13104	4421	6554	27650	2409	801				
ave.	87	81	362	244	524	177	262	1106	96	32	2.82	6.46	1.25	0.54

June 1987

DAY	Tmax amb.	Tmax indoor	Qstorage 1	Qstorage 3	Bldg. Load 1	Bldg. Load 2	Bldg. Load 3	Chiller Energy 1	Chiller Energy 2	Chiller Energy 3	Chil COP 1	Chil COP 2	Chil PCF 1	Chil PCF 2
1	96	82	121	620	1469	370	552	1828	374	0	3.06	3.48	1.15	1.01
2	96	82	545	618	1059	411	602	1904	397	0	2.96	3.64	1.19	0.97
3	99	84	542	603	1105	420	559	1893	400	0	3.06	3.69	1.15	0.95
4	94	85	547	455	1070	409	445	1883	405	0	3.02	3.55	1.16	0.99
5	87	81	806	490	76[illegible]	374	524	1854	369	0	2.98	3.56	1.18	0.99
6	88	80	618	598	942	297	526	1882	280	0	2.92	3.73	1.21	0.94
7	90	79	746	605	745	304	587	1810	282	0	2.90	3.79	1.21	0.93
8	91	80	461	582	1006	367	569	1792	312	0	2.88	4.14	1.22	0.85
9	94	81	474	256	1012	342	305	1788	290	0	2.92	4.15	1.20	0.85
10	96	81	519	593	1114	374	530	1829	369	0	3.14	3.56	1.12	0.99
11	93	81	853	511	708	386	490	1829	375	0	3.00	3.62	1.17	0.97
12	98	82	652	606	905	427	528	1878	408	0	2.92	3.68	1.21	0.96
13	91	82	516	681	1068	409	619	1910	404	0	2.92	3.56	1.21	0.99
14	82	78	640	760	941	359	682	1906	365	0	2.92	3.46	1.21	1.02
15	95	80	432	750	1037	378	681	1861	386	0	2.78	3.44	1.27	1.02
16	95	82	519	494	1060	403	507	1923	407	0	2.89	3.48	1.22	1.01
17	94	83	550	529	1063	425	506	1937	409	0	2.93	3.66	1.20	0.96
18	96	85	40	88	653	262	202	682	295	83	3.58	3.12	0.98	1.13
19	97	85	214	456	1571	411	474	1962	407	0	3.20	3.55	1.10	0.99
20	90	85	416	438	1153	426	466	1891	410	0	2.92	3.66	1.20	0.96
21	92	80	0	0	371	52	18	0	0	0	----	----	0.00	0.00
22	88	79	0	0	-144	-52	-108	0	0	0	----	----	0.00	0.00
23	88	78	0	0	-314	-87	-154	0	0	0	----	----	0.00	0.00
24	95	81	0	-236	-424	257	606	0	416	897	----	2.17	0.00	1.62
25	94	83	275	462	1228	432	455	1953	445	0	2.71	3.42	1.30	1.03
26	87	85	0	-46	56	437	240	0	498	291	----	3.08	0.00	1.14
27	91	81	458	632	1060	344	599	1953	379	23	2.74	3.19	1.29	1.10
28	91	85	156	259	873	258	258	1034	248	15	3.50	3.66	1.01	0.96
29	94	85	204	112	633	210	337	816	195	217	3.61	3.78	0.97	0.93
30	89	85	0	0	683	282	643	681	367	834	3.52	2.70	1.00	1.30
total			11308	11918	24466	9686	13251	42676	9892	2359 -				
ave.	92	82	377	397	816	323	442	1423	330	79	2.95	3.44	1.19	1.02

July 1987

DAY	Tmax amb.	Tmax indoor	Qstorage 1	Qstorage 3	Bldg. Load 1	Bldg. Load 2	Bldg. Load 3	Chiller Energy 1	Chiller Energy 2	Chiller Energy 3	Chil COP 1	Chil COP 2	Chil PCF 1	Chil PCF 2
1	94	83	342	552	1190	387	533	1951	398	0	2.76	3.42	1.27	1.03
2	98	86	430	461	1171	454	453	1937	438	0	2.91	3.64	1.21	0.97
3	96	86	429	419	1200	400	408	1928	410	0	2.97	3.43	1.18	1.02
4	93	83	594	525	1022	390	508	1889	387	0	3.01	3.55	1.17	0.99
5	96	81	578	578	1025	373	543	1915	365	0	2.95	3.59	1.19	0.98
6	96	84	507	485	1127	412	466	1925	406	0	2.99	3.57	1.18	0.99
7	95	85	-1	533	1	402	512	0	385	0	----	3.67	0.00	0.96
8	97	86	451	443	1184	394	442	1936	400	0	2.97	3.47	1.18	1.01
9	94	86	352	365	1295	439	355	1953	436	0	2.97	3.54	1.19	0.99
10	99	88	430	407	1230	420	395	1956	428	0	2.98	3.45	1.18	1.02
11	97	88	447	-25	1221	416	734	1935	394	916	3.03	3.71	1.16	0.95
12	95	83	269	467	1295	400	462	1987	414	0	2.77	3.40	1.27	1.03
13	99	85	507	548	1043	401	533	1915	410	0	2.85	3.44	1.24	1.02
14	97	86	353	397	1307	408	387	2014	415	0	2.90	3.45	1.21	1.02
15	96	87	468	432	1159	384	429	1916	388	0	2.99	3.48	1.18	1.01
16	93	86	530	488	1104	357	465	1921	315	0	2.99	3.99	1.18	0.88
17	90	85	492	-70	1133	288	669	1944	529	918	2.94	1.92	1.20	1.84
18	88	79	421	646	1080	364	608	1885	329	0	2.80	3.89	1.26	0.90
19	92	79	884	615	659	337	594	1876	285	0	2.89	4.15	1.22	0.85
20	93	80	549	629	996	368	611	1888	368	0	2.88	3.51	1.22	1.00
22	95	83	523		1030	390		2005	395		2.72	3.47	1.29	1.01
23	100	83	512	626	978	372	570	1860	383	0	2.82	3.42	1.25	1.03
24	94	82	441	646	1092	402	605	2005	435	0	2.69	3.25	1.31	1.08
25	90	80	614	566	938	334	523	1992	333	0	2.74	3.53	1.28	1.00
26	97	87	67	239	929	252	226	1096	212	0	3.20	4.18	1.10	0.84
27	101	84	143	-203	588	302	607	740	452	979	3.47	2.35	1.01	1.50
28	98	83	331	275	1022	394	635	1856	430	452	2.56	3.22	1.37	1.09
29	96	84	404	580	1040	375	544	1990	401	0	2.55	3.29	1.38	1.07
30	96	87	112	73	937	327	392	1134	394	335	3.25	2.92	1.08	1.20
31	94	85	73	33	989	338	452	1148	419	417	3.25	2.84	1.08	1.24
total			12251	11732	30984	11279	14662	52495	11753	4017				
ave.	95	84	408	405	1033	376	506	1750	392	139	2.90	3.38	1.21	1.04

August 1987

DAY	Tmax amb.	Tmax indoor	Qstorage 1	Qstorage 3	Bldg. Load 1	Bldg. Load 2	Bldg. Load 3	Chiller Energy 1	Chiller Energy 2	Chiller Energy 3	Chil COP 1	Chil COP 2	Chil PCF 1	Chil PCF 2
1	99	89	64	242	994	351	232	1133	434	0	3.28	2.84	1.07	1.24
2	94	85	214	204	963	346	193	1267	438	0	3.27	2.78	1.08	1.27
3	98	88	-65	0	53	31	71	0	17	0	----	6.41	0.00	0.55
4	97	85	0	0	0	-24	-76	0	0	0	----	----	----	0.00
5	99	88	0	-48	-402	-191	-146	0	0	26	----	----	0.00	0.00
6	101	88	227	383	1035	567	172	1346	726	0	3.30	2.75	1.07	1.28
7	98	87	190	145	1057	582	141	1343	737	0	3.27	2.78	1.08	1.27
8	98	89	151	181	1057	561	172	1308	727	0	3.25	2.71	1.08	1.30
9	99	89	182	175	1079	461	167	1376	525	0	3.22	3.09	1.09	1.14
10	94		227		1034	202		1364	230		3.25	3.09	1.08	1.14
11											----	----	----	----
12		88	-20		67			45			3.67	----	0.96	----
21	93	82		7			314			332	----	----	----	----
22	96	83	-7	12	1089	460	404	1406	593	495	2.71	2.73	1.30	1.29
23	99	82	-18	11	1044	434	357	1335	523	458	2.70	2.92	1.30	1.21
24	95	84	0	-73	1010	542	299	1294	642	399	2.75	2.97	1.28	1.18
25	95	81	94	8	1049	491	389	1250	626	469	3.22	2.76	1.09	1.27
26	97	82	-7	4	1030	412	416	1776	493	491	2.03	2.94	1.74	1.20
27	98	82	-33	6	1113	515	387	1478	674	492	2.57	2.69	1.37	1.31
28	96	83	0	9	1123	434	408	1476	547	521	2.68	2.79	1.31	1.26
29	99	82	-14	7	1181	506	401	1529	649	506	2.68	2.74	1.31	1.28
30	93	84	-10	-4	1123	380	40	1461	498	19	2.68	2.68	1.31	1.31
31	93	80	83	-103	1024	517	416	1245	654	645	3.13	2.78	1.12	1.26
total			1258	1166	17723	7577	4757	23432	9733	4853				
ave.	97	85	60	58	844	379	238	1116	487	243	2.85	2.74	1.23	1.28

September 1987

DAY	Tmax amb.	Tmax indoor	Qstorage 1	Qstorage 3	Bldg. Load 1	Bldg. Load 2	Bldg. Load 3	Chiller Energy 1	Chiller Energy 2	Chiller Energy 3	Chil COP 1	Chil COP 2	Chil PCF 1	Chil PCF 2
1	87	82	20	230	990	442	220	1398	491	6	2.54	3.17	1.38	1.11
3	81	75		354		436	321		520	0	----	2.95	----	1.19
4	77	74	510	360	923	398	328	1996	479	0	2.52	2.92	1.39	1.20
5	82	75	415	369	932	375	336	1848	448	0	2.56	2.94	1.37	1.19
6	85	75	401	371	984	350	331	1913	372	15	2.55	3.31	1.38	1.06
7	88	75	508	163	960	402	147	1778	435	0	2.90	3.25	1.21	1.08
8	97	80		381		444	347		498	0	----	3.14	----	1.12
9	90	74	493		1023	400		1961	382		2.72	3.68	1.29	0.96
11	94	81		388		441	357		489	0	----	3.17	----	1.11
12	87	78	7	5	81	426	326	35	508	430	8.84	2.95	0.40	1.19
13	87	77	-16	7	969	387	321	1178	459	401	2.85	2.97	1.24	1.19
14	90	80	-17	18	922	402	332	1140	482	437	2.79	2.93	1.26	1.20
15	89	80	8	-71	783	460	358	1055	498	551	2.61	3.25	1.35	1.08
16	89	79	443	372	784	443	339	1660	490	0	2.60	3.18	1.35	1.11
17	90	80	390	367	801	455	330	1693	498	0	2.47	3.21	1.42	1.09
18	93	82	392	0	752	307	55	1722	365	0	2.34	2.96	1.51	1.19
22	84	76	225	285	575	313	252	1052	368	0	2.67	2.99	1.32	1.18
23	83	78	290	267	566	269	237	1237	348	0	2.43	2.72	1.45	1.29
24	81	78	345	273	554	292	239	1186	350	0	2.67	2.93	1.32	1.20
25	83	78	292	270	534	272	244	1121	340	0	2.59	2.81	1.36	1.25
26	83	78	297	307	532	329	268	1181	358	0	2.47	3.23	1.42	1.09
27	84	77	343	328	610	352	293	1297	384	0	2.58	3.22	1.36	1.09
28	83	77	318	328	607	369	293	1342	444	0	2.42	2.92	1.45	1.20
29	87	78	406	339	652	388	302	1465	453	0	2.54	3.01	1.38	1.17
30	81	76	373	325	722	412	294	1466	465	0	2.63	3.12	1.34	1.13
total			6435	6036	16256	9564	6870	30724	10924	1840				
ave.	86	78	293	252	739	383	286	1397	437	77	2.60	3.08	1.35	1.14

October 1987

DAY	Tmax amb.	Tmax indoor	Qstorage 1	Qstorage 3	Bldg. Load 1	Bldg. Load 2	Bldg. Load 3	Chiller Energy 1	Chiller Energy 2	Chiller Energy 3	Chil COP 1	Chil COP 2	Chil PCF 1	Chil PCF 2
1	76	76	371	259	597	273	226	1305	321	0	2.61	2.99	1.35	1.18
2	80	77	273	261	465	262	229	1034	326	0	2.51	2.83	1.40	1.24
3	77	74	308	220	507	249	209	1139	304	0	2.52	2.88	1.40	1.22
4	68	76	86	220	404	18	204	654	34	0	2.64	1.86	1.33	1.89
5	75	79	274	254	421	18	226	858	7	0	2.85	9.04	1.23	0.39
total			1312	1214	2394	820	1094	4990	992	0				
ave.	75	76	262	243	479	164	219	998	198	0	2.61	2.91	1.35	1.21

APPENDIX C:

PROCEDURE FOR FILLING THE BRINE LOOP

Filling an Empty Brine Loop

Much of this procedure is dictated by the fact that the expansion tank must be filled with brine when the loop is filled. The recommended procedure, starting with an empty loop, is as follows:

1. The 25 percent ethylene glycol must be prepared outside the loop. This can be done in a 55-gal. drum.

2. Pressurize the airside of the expansion tank about 10 psig. This will collapse the expansion tank diaphragm.

3. Pump brine into loop with a portable transfer pump. Vent all high sections of the loop and repeatedly work the two three-way valves. Continue adding brine and working valves until a steady stream of liquid is vented. Close the vent valves.

4. Release the air pressure on the expansion tank and remove the air valve. Continue pumping brine into the loop until the brine pressure increases to about 12 psig. This will expand and fill the expansion tank diaphragm with brine. Close loop fill valve.

5. Turn the brine pumps on for a period of no longer than about 5 seconds. Reverse the positions of the two three-way valves and start the pump again for about 5 seconds. The purpose is to move air pockets around the loop and allow them to collect in high regions of the loop for venting. The time limit of 5 seconds is set so that a minimum amount of air goes through the pump and becomes froth. Vent all high sections of the loop. If venting allows considerable air to escape and the loop pressure is reduced significantly, then pump more brine into the system until the brine pressure is again up to 12 psig. Repeat this step until no more air is released when venting the loop and the pressure remains at 12 psi.

6. Reinstall the air valve in the expansion tank. Pressurize the expansion tank with air to about 20 psig. It would be advantageous to pressurize the expansion tank with instrument air. Thus, the pressure will always be maintained at a constant 20 psi. When the brine pumps are off, the brine system pressure gage near the expansion tank will also be a constant 20 psig. When the brine pumps are on, the same brine system pressure gage will read a higher value but constant. These pressures should be marked on the pressure gage.

   As brine leakage proceeds, the point will eventually be reached when the expansion tank is empty. When this happens, the diaphragm in the expansion tank will collapse and seal the outlet pipe. Air pressure will be maintained in the expansion tank at 20 psig. but the pressure will not be transmitted to the brine loop. The result is that the pressure in the brine loop will begin to drop. Thus, when the pressure in the brine loop. The result is that the pressure in the brine loop will begin to drop. Thus, when the pressure in the brine loop is below the marked value on the gage, brine must be added to the system.

7. The pressure gages should be inspected periodically to determine if the brine system is running out of brine. Gages should be checked daily for a while, and then weekly. The inspection schedule can be adjusted as more experience is gained with the diurnal ice storage system.

Filling the Brine Loop When It Is Low on Brine

Step 6 of the previous section (filling an empty brine loop) describes the symptoms when the brine loop is running low on brine and requires refilling.

The historic procedure of using the small vacuum at the brine circulating pump suction to introduce brine to the system should be avoided. The reason is that the vacuum condition will not allow the expansion tank to fill, and its internal bladder will remain collapsed on the outlet pipe.

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