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Design and Construction of an Ice-in-Tank Diurnal Ice Storage Cooling System for the PX Building at Fort Stewart, GA

by
Chang W. Sohn
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Diurnal cold storage cooling systems may be effective tools for reducing peak electrical demand at Army installations. Ice, water, or eutectic salts can be used to store refrigeration produced during offpeak hours. Then, during peak load periods, the stored cold can be used to cool a facility. Such a reduction in the peak electrical demand can result in substantial savings in utility bills throughout the year. However, there is a lack of standard design guidance on these systems for Army engineers. As a first step toward remedying that situation, a demonstration ice-in-tank diurnal ice storage (DIS) cooling system was designed and installed—using standard engineering practices—at Fort Stewart, GA. Although the concept of cold storage is new, it can be implemented using familiar engineering procedures. This report documents the design and installation of the ice-in-tank system. The results of performance and operation testing will appear in a future technical report.

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FOREWORD

This study was carried out for the Facilities Engineering Division (now part of the U.S. Army Engineering and Housing Support Center), Office of the Assistant Chief of Engineers (OACE), under the Facility Technology Application Test (FTAT) work unit "Diurnal Energy Storage System Demonstration." The Technical Monitor was Mr. B. Wasserman, CEHSC-FU.

This work was performed by the Energy Systems Division, U.S. Army Construction Engineering Research Laboratory (USA-CERL-ES) and the Oak Ridge Nationa. Laboratory (ORNL), in cooperation with the Engineering Services Division, Fort Stewart, GA (AFZP-DEE). Dr. Gilbert Williamson is Chief of USA-CERL-ES. Dr. Sohn is a Principle Investigator at USA-CERL; Mr. Tomlinson is associated with ORNL. The technical editor was Ms. Jane Andrew, Information Management Office, USA-CERL.

COL Norman C. Hintz is Commander and Director of USA-CERL, and Dr. R. L. Shaffer is Technical Director.

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1 INTRODUCTION

Background

The U.S. Army Construction Engineering Research Laboratory (USA-CERL) recently summarized available energy storage technologies appropriate for Army applications.[1] Among them, the storage of cold water or ice is considered to be the most cost-effective technology. Army facilities possess ideal characteristics for using storage cooling systems. They have centralized district cooling systems with generous space for storage installation. Potentially, a standard system could be developed for a very large number of facilities which have similar design parameters. To demonstrate the technical and economic feasibility of these systems for Army facilities, a diurnal ice storage (DIS) cooling system was selected for a demonstration as a part of the Facility Technology Application Test (FTAT) program.

In the private sector, cold storage for space cooling applications has been rapidly developing, with hundreds of successfully operating systems.[2] Within the Army, engineers from installations and districts have expressed interest in this developing technology. However, the consensus is that there is an absence of design guidance from the Army. USA-CERL is developing design guidelines, and the FTAT program is a part of an effort to provide this information to Army engineers.

Before any design guidance for cold storage cooling systems can be written, extensive data are needed for evaluation and comparison of their technical and economic feasibility. Ice storage has the most potential for standardization throughout the Army because of the modular design of commercial DIS cooling systems. Four categories of these commercially available ice storage systems will be field tested. During fiscal year (FY) 86, an ice-in-tank DIS cooling system was installed for a Post Exchange (PX) building at Fort Stewart, GA. An ice-on-coil DIS system is being installed at an office/barracks complex at Yuma Proving Ground, AZ in FY 88. An ice-shucking and a eutectic salt DIS cooling system are scheduled to be installed in the coming years. The general principles of these systems are discussed in Chapter 2.

Concurrent with USA-CERL's field demonstration program, Oak Ridge National Laboratory (ORNL) is conducting a series of laboratory experiments under the sponsorship of Electric Power Research Institute (EPRI) to rate the performance of each type of DIS cooling system.[3] The USA-CERL and ORNL programs will complement each other, thereby presenting accurate design and operation data for DIS cooling systems.

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[1] R. J. Kedl and C. W. Sohn, *Assessment of Energy Storage Technologies for Army Facilities*, Technical Report E-86/04/ADA171513 (USA-CERL, May 1986).

[2] *Current Trends in Commercial Cool Storage*, EPRI EM-4125, Project 2036-13, Final Report (Electric Power Research Institute, [EPRI] , July 1985).

[3] J. J. Tomlinson, "Results of EPRI Laboratory Testing of Ice-Storage Systems," presented at Seminar on Commercial Cool Storage: State of the Art (sponsored by EPRI, Denver, CO, 19-20 February 1987).

Objective

The objective of this work was twofold: (1) to design and install a demonstration ice-in-tank DIS cooling system at Fort Stewart, and (2) to provide a design reference on ice-in-tank DIS cooling systems for Army engineers.

Approach

Using the criteria given in Chapter 3, Fort Stewart was selected for the first demonstration. The ice-in-tank concept was chosen for the first demonstration because it best matched the operating modes and space needs at Fort Stewart. The system was designed using standard engineering practices. A preliminary economic analysis was done to estimate the savings. The system was installed (successfully) by a contractor with no prior experience with storage cooling systems.

Scope

This report describes only the design and installation of the ice-in-tank system at Fort Stewart, GA. Results from the operation and performance monitoring of the Fort Stewart DIS cooling system will be topics of a future report.

Mode of Technology Transfer

A part of this report has appeared in the OCE Daily Staff Journal, and an entry will appear in the FTAT Notebook. It is recommended that information about DIS cooling systems be summarized in an Engineering Technical Note (ETN). The intermediate outcome of the overall testing program will be a series of technical reports discussing design/installation and operation/performance of each type of DIS cooling system, which will serve as interim guidelines. At the conclusion of the demonstration program, USA-CERL will develop a general design and operation guide on DIS cooling systems.

2 DIURNAL COLD STORAGE COOLING SYSTEMS

General System Description

A diurnal cold storage cooling system is an air-conditioning system which shifts electrical utility demand for air-conditioning from onpeak to offpeak periods. Rather than operating a chiller to meet the cooling load as it arises, the chiller is operated either partially or solely during the offpeak period, and the refrigeration produced is stored to meet the next day's onpeak cooling requirements. It can be stored in several media: for example, chilled water, ice, or freezing eutectic salts. A diurnal ice storage (DIS) cooling system uses ice as a storage medium.

Electrical Demand Characteristics of Army Facilities

Electrical demand charges provide the economic incentive for developing storage cooling technology. A close examination of the electrical demand cost is essential to understand the concept, rationale, and methodology of the technology. The characteristics of electrical demand by the Army facilities are discussed by analyzing electrical utility consumption records and monthly utility bills of a typical Army installation.

Army installations use large amounts of electricity for a variety of applications in many different buildings. Typically, an Army installation has one or more master meters which measure its total power consumption of the installation. The installation is billed for its electrical utility charges based on those meter readings. Typical monthly electrical utility bills for an Army installation consist of two parts: cost of energy consumption--based on kilowatt hours used, and the demand charge--based on the *billing peak* demand in kilowatts. Table 1 shows monthly electrical utility bills for an Army installation (Post A) in the Southeast for 1985.

Table 1

Monthly Electrical Utility Charges for Post A

Mon	Days	Actual Demand kW	Billing Demand kW	Energy Total kWH	Demand Charge ($)	Energy Charge ($)	Total Charge ($)
Jan	29	12,269	17,485	6,144,000	171,379	153,600	324,979
Feb	30	12,355	17,485	6,480,000	171,379	162,000	333,379
Mar	28	10,627	17,485	5,376,000	171,379	134,400	305,779
Apr	32	12,528	17,485	6,312,000	171,379	157,800	329,179
May	30	16,589	17,485	7,368,000	171,379	191,568	362,947
Jun	29	19,354	19,354	9,168,000	189,620	238,368	427,988
Jul	32	19,354	19,354	10,704,000	189,620	278,304	467,924
Aug	29	19,267	19,267	9,648,000	188,771	250,848	439,619
Sep	28	19,008	19,008	8,280,000	186,243	215,280	401,523
Oct	34	15,379	17,485	8,232,000	171,379	214,032	385,411
Nov	31	14,342	17,485	6,384,000	171,379	159,600	330,979
Dec	32	11,232	17,485	5,976,000	171,379	149,400	320,779
Total					2,125,283	2,305,200	4,430,483

The basic rate structure for Post A consists of an energy charge of \$0.025/kWh and a demand charge of \$10.485/kW for the first 1000 kW and \$9.76/kW for the excess over that amount. The billing demand is based on whichever of the following is greatest: (1) the maximum integrated 15-minute demand (which may be on a rolling time interval) measured during the current month, (2) 80 percent of the highest demand occurring during the 11 preceding months, (3) the contract demand, or (4) 1000 kW. For Post A, the contract demand is set at 17,485 kW, which is reflected during the nonsummer months. Notice that in this example, the demand portion of the annual electrical utility cost is over \$2.1 million, which constitutes 48 percent of the total bill.

Power Demand Characteristics

Figure 1 shows the power demand curve for Post A on the day (1 August 1985) when the peak demand of the month was recorded. The power demand for that day fluctuated from 11,923 kW at 2:30 a.m. to 19,267 kW at 3:30 p.m. If a perfect energy storage system were available, the power demand for that day would have been 15,238 kW:

$$P = (1/T) \int_0^T D(t)dt = 15{,}238 \text{ (kW)} \qquad \text{[Eq 1]}$$

where D(t) is the power demand, T is the time period (one day), and P is the average power demand. The horizontal line in Figure 1 shows the daily average power demand with an ideal energy storage.

Many of the utilities in the United States experience a summer peak in their electrical demand. As shown in Figure 1, the peak demand occurs typically during the early afternoon (12 to 4 p.m. in this example). The cause of this peak is attributed to summer air-conditioning during these hours. Notice that the demand is relatively flat during the same period on a weekend (also shown in Figure 1). If the compressors could be unloaded during that 4-hour period, the demand could be reduced by the number of kilowatts the compressors would have required.

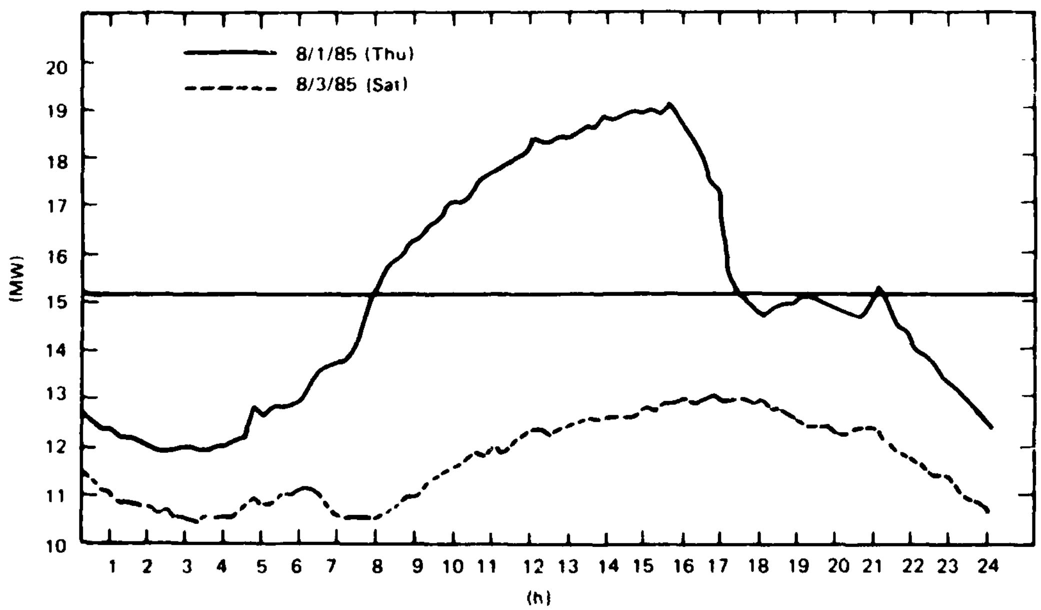

Figure 1. Peak day power demand profile for Post A.

However, to provide air-conditioning during that period a source of refrigeration is required. Examination of Figure 1 shows that the power demand is low during the night. The compressors can be activated to produce refrigeration during that period without increasing the peak power demand. However, the refrigeration produced must be stored to meet the cooling load the following day. Therefore, a means of diurnal cycle cold storage is required to use the offpeak power for air-conditioning.

Site Characteristics

A typical Army installation is served by a few centralized cooling plants. An extensive chilled water distribution loop serves a number of buildings, originating from the central cooling plants, whose capacities are typically on the order of 1000 tons of cooling. A schematic system layout is shown in Figure 2. A retrofit application of a diurnal storage cooling system is also shown in the same figure.

The advantages of an Army central cooling plant for storage cooling application are:

- A significant reduction in electrical demand is possible with a relatively minor alteration of existing piping. Existing distribution loops can be used effectively.

- The space for a storage location may be easily available near the cooling plant. The centralized plant and storage would reduce the costs of initial construction and also improve the efficiency of operation and maintenance of the system. The latter is one of the most important criteria for the Army.

- Hundreds of Army installations have similar designs and operating conditions. Therefore, a successful storage cooling system for an Army installation can be easily copied at other installations with little modification. An operation and maintenance lesson from one installation can also be easily applied to other installations.

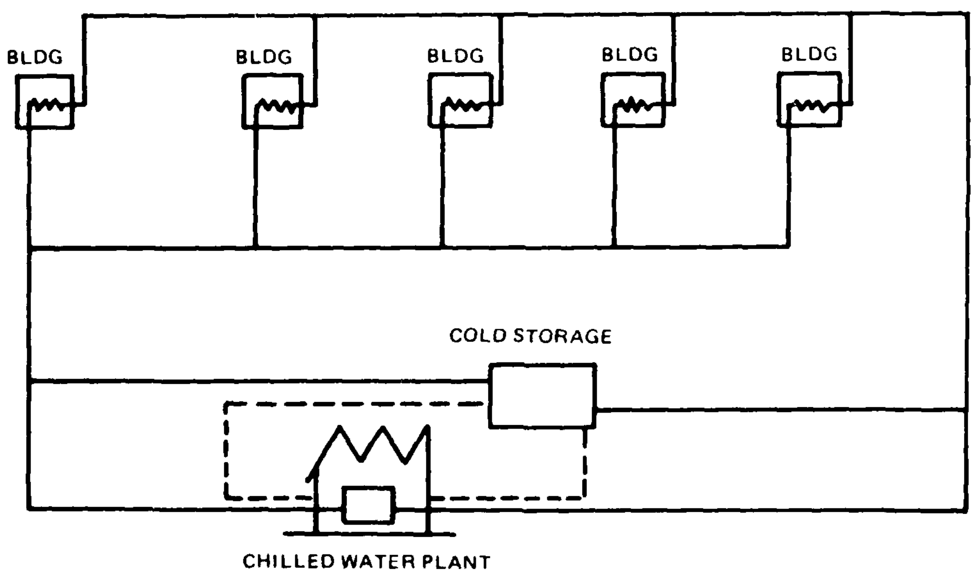

Figure 2. Schematic drawing of central district cooling system.

Types of Diurnal Cold Storage Cooling Systems

The refrigeration produced during the offpeak period can be stored in several forms: chilled water, ice, or freezing eutectic salt. Ice and salt systems can be grouped together as phase-change systems. In a recent survey of over a hundred newly installed cold storage systems,[4] ice and chilled water were found to be the predominant storage media. To date, there is no clear answer on which is the best medium.[5] The characteristics of chilled water and ice for cold storage cooling systems are summarized in Table 2. Technologies developed for each of the three media are discussed below.

Chilled Water Storage Cooling Systems

When water is the storage medium, blending of warm water and cold water must be minimized. The blending problem has been addressed by the following techniques.[6]

Membrane Storage of Chilled Water. This system uses a coated fabric membrane or sock that is fitted to the tank. Cold and warm water are stored in the same tank--cold water below the membrane and warm water above it, as shown in Figure 3. The membrane moves up as the tank is charged and cold water displaces warm water; it moves down as cold water is used and displaced with warm water.

Multiple Tank Storage. In this approach, the cold water and warm water are stored in separate tanks as shown in Figure 4. The tank farm contains one more tank than is required for storage. To charge the system, warm water is drawn from one tank, cooled by the chiller, and added to the empty tank. In the discharge mode, cold water is drawn from a cold tank, used for space cooling, and placed in a warm water tank. Thus, the tank farm always contains one empty tank or two partially empty tanks.

Baffled Tank Storage. Conceptually, natural thermal stratification can best be achieved in a tall, narrow tank. The baffle arrangement shown in Figure 5 can be envisioned as a scheme that conceptually makes a tall, narrow tank out of a short, wide tank. The arrangement shown was developed in Japan, where it has been estimated[7] that hundreds of these tanks were installed. After extensive testing,[8] it was concluded that better stratification was achieved by removing the baffles and installing a carefully designed diffuser system to provide natural stratification.

Natural Thermal Stratification. This system (shown in Figure 6) uses the slightly lower density of warm water to cause it to "float" on the more dense cold water. Because the density difference is not great, natural stratification is difficult to achieve. For successful stratification, the inlet and outlet manifolds must be carefully designed.

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[4] *Survey of Thermal Energy Storage Installations in the United States and Canada* (American Society of Heating, Refrigerating, and Air Conditioning Engineering [ASHRAE], 1984).

[5] R. H. Stamm, "Thermal Storage Systems," *Heating/Piping/Air Conditioning*, January 1985.

[6] *Commercial Cool Storage Design Guide*, EPRI EM-2981, Project 2036-3, Final Report (EPRI, May 1985).

[7] Ayers Associated, *A Guide for Off-Peak Cooling of Buildings* (Southern California Edison Co., September 1980).

[8] M. W. Wildin, *Results from Use of Thermally Stratified Water Tanks to Heat and Cool the Mechanical Engineering Building at the University of New Mexico*, ORNL/SUB/80-7967/1 (ORNL, June 1983).

Table 2

Chilled Water Vs. Ice as a Cold Storage Medium

Characteristics	Ice	Chilled Water
Volume/space	Compact	Large
System	Modularized	Customized
Initial cost	Low	Relatively high
Compressor derating	High	Low to none
Blending control	Simple	Relatively complicated
Application type	Small or retrofit	New or large

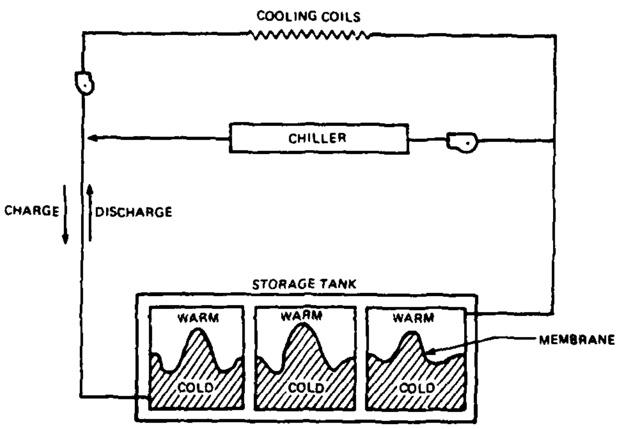

Figure 3. Membrane storage of chilled water.

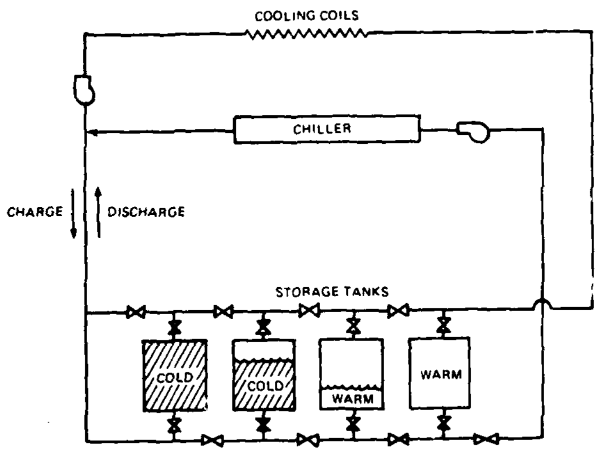

Figure 4. Multiple tank storage of chilled water.

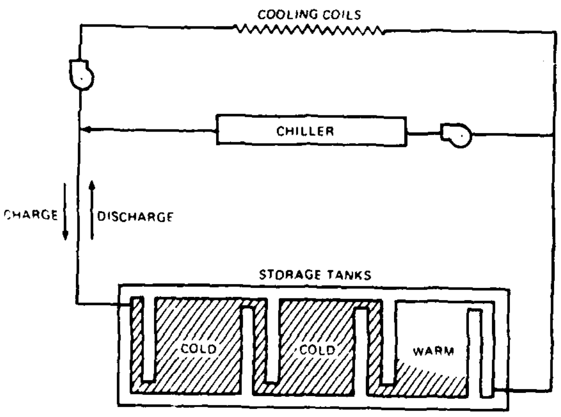

Figure 5. Baffled tank storage of chilled water.

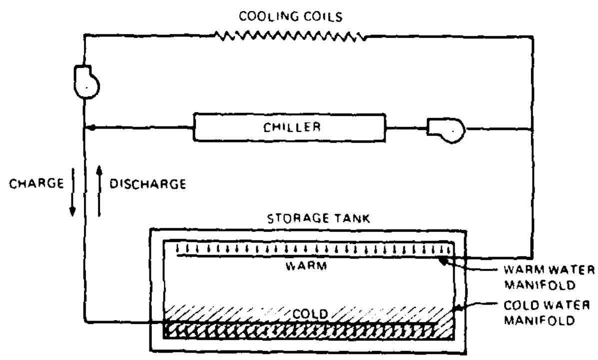

Figure 6. Natural stratification storage of chilled water.

Recent advances in understanding the dynamics of natural stratification[9] have increased confidence in the ability to design a successful installation.

Phase Change Storage Cooling Systems

The heat of fusion for water is 144 Btu/lb. This means that 1 lb of water will store 144 Btu of cold as it freezes (but it will only store 20 Btu of cold between 35 °F and 55 °F). Thus, ice storage units are much more compact than chilled water storage units. Ice storage cooling systems can be categorized as follows based on the methods of ice freezing and harvesting.

Ice-on-Coil System. This is the most common type (Figure 7). Tubes are supported in a serpentine fashion throughout a tank filled with water. The tubes act as an evaporator for the chiller cycle, so that ice forms logs around the tubes. Controls are used to prevent the ice logs from bridging, thus maintaining flow passages through the storage unit. Agitation or baffling for flow control is required to prevent short circuiting and to keep water in contact with ice.

Ice-in-Tank System. In this approach, an antifreeze solution is cooled by a standard chiller and is pumped through a heat exchanger, where it extracts heat from the water in the tank until the water freezes solid (Figure 8). The unit is discharged through the same heat exchanger. The tubes are plastic and much closer together than in the ice-on-coil type. Since the water does not have to be pumped through the ice side of the storage unit, it can be frozen completely, thus increasing the volumet ic storage density. Neither bridging control nor agitation is needed in this approach. However, the system requires an additional heat exchanger for the intermediate loop.

Ice-Shucking System. An alternate form of icemaker is often referred to as a dynamic ice system or an ice shucker. In one arrangement of this system (Figure 9), the evaporator is a vertical flat plate. Water is sprayed on the plate and freezes to form a layer of ice. Periodically, condenser heat is routed through the evaporator to release the ice that falls into a bin below the evaporator. Cold is extracted from the ice by pumping water through the ice storage bin.

Eutectic Salt System. A typical evaporator temperature for the icemaker is about 20 °F.* The lowered evaporator temperature for the icemaker, compared to the chilled water system, derates the chiller and adversely affects the kW/ton ratio. Eutectic salts are being developed to freeze and thaw at around 47 °F.[10] Their latent heat of fusion is about 41 Btu/lb, approximately 3.5 times less than that of ice. Theoretically, the eutectic salt storage combines the desirable features of both the chilled water and the ice storage systems. A typical system layout is shown in Figure 10. Long term performance, however, is yet to be established due to its limited exposure on the market so far.

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[9] R. T. Tamblyn, "Thermal Storage: Resisting Temperature Blending," *ASHRAE Journal*, January 1980; R. L. Cole and F. O. Bellinzer, *Natural Thermal Stratification in Tanks*, ANL-82-5 (Argonne National Laboratory, February 1982); C. Hiller, "Stratified Chilled Water Storage Techniques," presented at the Seminar on Commercial Cool Storage: State of the Art (sponsored by EPRI, Denver, CO, February 1987).

*Metric conversions are listed on p 46.

[10] Commercial literature from Transphase Systems, Inc., 16552 Burke Lane, Huntington Beach, CA 92647.

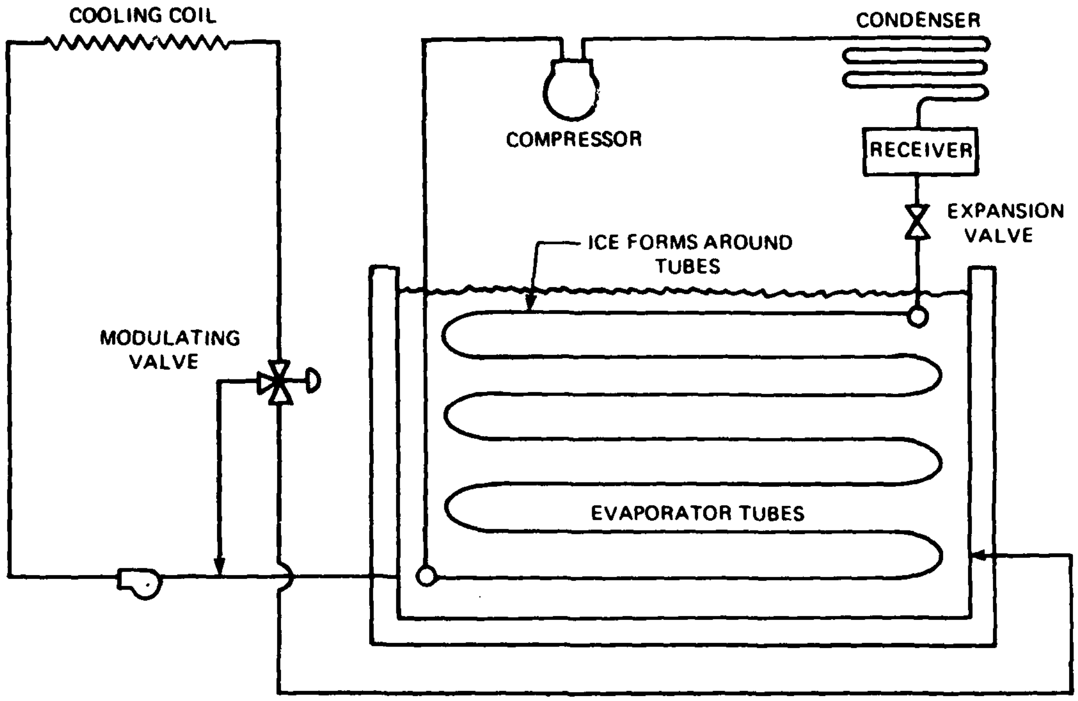


Figure 7. Ice-on-coil DIS cooling system.

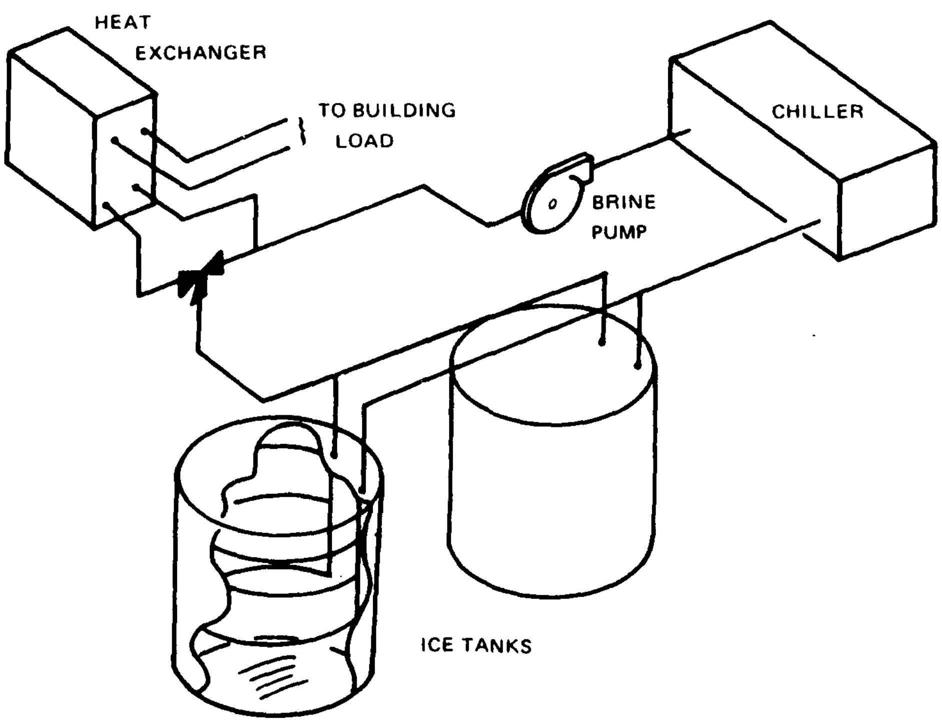


Figure 8. Ice-in-tank system.

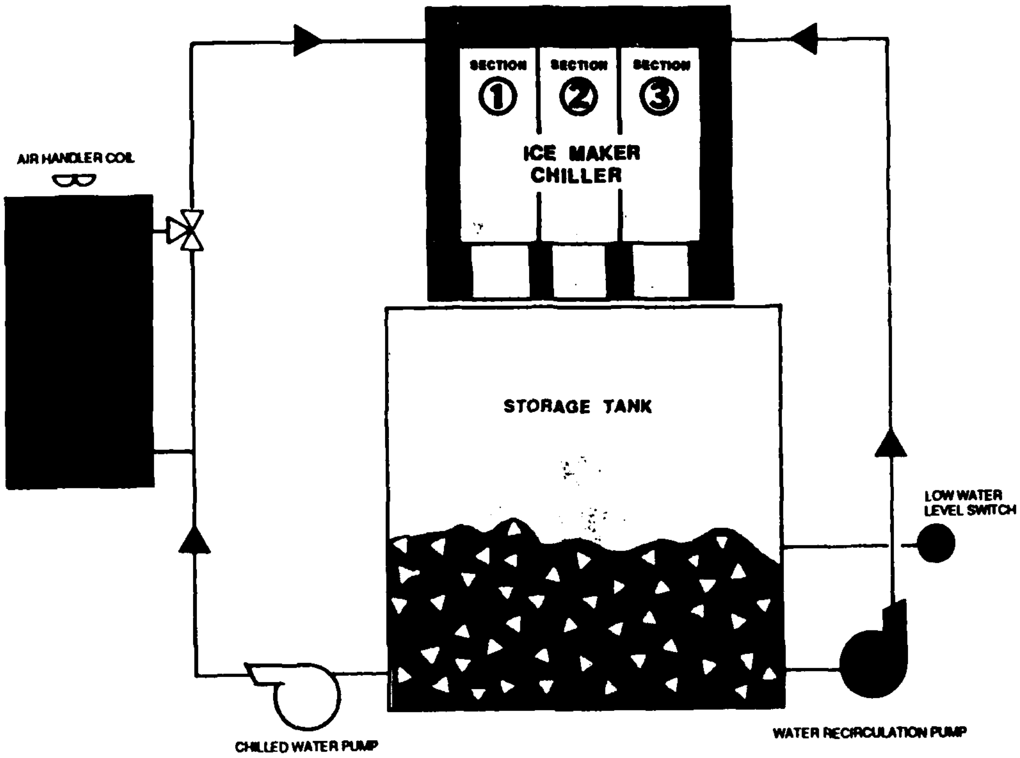


Figure 9. Ice-shucking system.

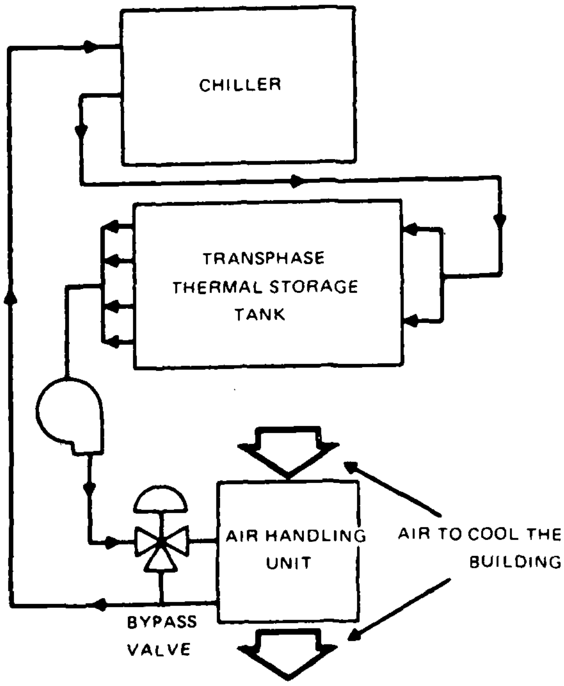


Figure 10. Typical layout of a eutectic salt system.

3 SITE CHARACTERISTICS OF FORT STEWART

Site Selection

The keys to a successful installation of storage cooling systems are as follows:[11]

1. Choose a building whose owner is sympathetic and well informed.
2. Perform sound financial analysis in advance.
3. Provide adequate safety factors in system design.
4. Keep the system simple.
5. Design the system for overall energy and cost effectiveness.
6. Perform a third-party system design review.
7. Specify and purchase the system with single-source contractor responsibility, backed up by a performance bond.
8. Implement a microprocessor based control system.
9. Secure support from the serving utility company.
10. Design for flexibility for future expansion.
11. Minimize parasitic heat gains and energy consumption.
12. Be prepared for unexpected future changes in utility rate schedules and building usages.

In addition to these considerations, the following guidelines were developed for selecting a candidate Army facility that would allow the most informative and effective demonstration of a DIS cooling system.

1. The facility has a sharp peak load coinciding with the installation's peak electrical demand.
2. The installation's electrical demand charge is high.
3. The facility has a well defined occupancy schedule.
4. The installation will strongly support the project.
5. The facility is separately metered, or at least has its own chiller to cool it (for monitoring performance).

[11] D. P. Gatley, "Cooling Thermal Storage," *Heating/Piping/Air Conditioning*, April 1987.

6. The facility has space available for installing the DIS cooling system.

7. Experienced contractors are available locally.

During the course of selecting an ideal candidate facility, USA-CERL and ORNL visited Fort Jackson, SC, Fort Stewart, GA, Fort Benning, GA, and Fort Polk, LA. Among the facilities studied, the PX building at Fort Stewart was selected to be the ice-in-tank demonstration facility.

PX Building at Fort Stewart

The building has 51,000 sq ft of floor area containing a department store and auxiliary facilities such as a candy shop, gift shop, and barber and beauty shops. It is cooled by a 175-ton centrifugal chiller. (A later addition has its own cooling system.) The mechanical room could not accommodate the ice storage system; however, room was available in an adjacent fenced area enclosing a cooling tower and a transformer. The building has a distinctive 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 energy characteristics of the building for a typical hot day are displayed in Figure 11.* The electrical demand of the chiller is a well defined step function: a 90 to 100 kW power demand during the hours the building is occupied, including early morning cool down and maintenance, and a 40 kW demand occurs during the night. The nighttime cooling is required for the area where food is stored and prepared. It should be remembered that the day was not a design day (see Chapter 4), therefore the chiller was running on part load condition. Figure 11 simply reveals that the building power demand is a well defined step function with respect to time of day.

The building does not have its own meter by which a demand charge is billed. The power to the building comes through a master meter which measures the total electrical demand of the installation. The effect of demand reduction by the DIS cooling system should be examined through a power demand curve for Fort Stewart in total. Figure 12 shows the annual peak day power demand for 1984, established on 20 June 1984. It shows a significant swing in power demand from the valley (15 MW) to the peak (25 MW). Any amount of power shifted from the 1400 to 1600 period to an offpeak period will effectively reduce the base peak power demand by the same amount, which will then reduce the monthly electrical utility bill for the next 11 months.

Preliminary Economic Studies

The most important benefit of a DIS cooling system is reduced electrical utility costs for air-conditioning. Expected savings in electrical cost by the DIS cooling system for the PX building can be calculated by finding how much power will be shifted away from the peak period. Then the dollar savings from this shift can be calculated using the rate schedule.

To calculate the power shift, ORNL first measured the performance of the chiller for the PX building with the existing cooling system from 7 August 1985 through

*Figures and tables for this chapter begin on p 21.

19 September 1985. Figure 13 shows the results of chiller performance for the day of 21 August 1985. According to Figure 13, the chiller efficiency for the day in terms of kW/ton is

$$\text{Performance} = 0.75 \text{ kW/ton} \qquad \text{[Eq 2]}$$

The specified rating of the chiller at full load condition is

$$\text{Capacity} = 175 \text{ ton} \qquad \text{[Eq 3]}$$

Assuming that the chiller is running at full load condition for a design day, at the peak recording hour the chiller will draw

$$\text{Power} = \text{Performance} \times \text{Capacity} = 130 \text{ kW} \qquad \text{[Eq 4]}$$

where Power is the power that would be shifted from onpeak to offpeak periods for a peak setting day by unloading the chiller during onpeak periods. The estimate in Equation 4 corroborates the measurements shown in Figure 11, which show a demand of over 100 kW by the chiller for the day even though it is not the peak-setting day.

A review of Georgia Power's utility rate schedule is needed to calculate the amount of dollar savings from this shift in demand using a DIS cooling system. A shortcut rate schedule applicable to Fort Stewart is shown in Table 3. It is important to note that the following calculation is unique to Fort Stewart and Georgia Power Company. The "hours use of demand," needed to identify which demand rate schedule should be used, is determined by

$$\text{Hours Use of Demand} = \text{kWh}/\textit{Billing Demand} \qquad \text{[Eq 5]}$$

where kWh is the total kilowatt hours consumed during the billing period (Table 4), and Billing Demand is the peak billing demand recorded during the corresponding billing period (Table 4). The hours of demand for 1986 are given under "Hours" in Table 4.

A billing demand is based both on the billing period, i.e., whether it is a summer month or a winter month, and on the actual demand during the current month and the preceding 11 months. The months June through September are considered summer (peak demand) months, and October through May are considered winter months. For the summer billing months, the billing demand is the greatest of the following: the current actual demand, 95 percent of the highest demand occurring in any previous applicable summer month, or 60 percent of the highest demand occurring in any previous applicable winter month. For the winter months, the billing demand is the greater of 95 percent of the highest demand occurring in any previous applicable summer month, or 60 percent of the highest demand occurring in any applicable winter month, including the current month.

Figure 14 shows the monthly actual peak demands by Fort Stewart during 1986. The summer peak is significantly higher. Therefore the monthly billing demand is the higher of either the actual monthly demand or 95 percent of the highest peak recorded during the preceding summer months. Unloading a chiller during the summer months will decrease monthly utility bills for a whole year. Based on Equation 4 and the definition of billing demand (95 percent ratchet schedule), the monthly effective demand reduction can be calculated:

$$PON = 130 \text{ kW} \quad [Eq\ 6]$$

$$POFF = 0.95\ PON = 123.5 \text{ kW} \quad [Eq\ 7]$$

where PON is the amount of power shifted away from the peak for the peak-setting month, and POFF is the power shifted for the immediately following eleven months. The actual monthly electrical utility costs for Fort Stewart are shown in Table 4. For each month, the Hours Use of Demand from Equation 5 is used to find the cost/kW in Table 3. With this and the power shifts from Equations 6 and 7, the annual total savings in demand cost by the DIS cooling system is calculated as follows (for 5 summer months and 7 winter months):

$$\begin{aligned} SVNG &= [(5\text{ mo})(130\text{ kW})(6.69\ \$/\text{kW})] + [(7\text{ mo})(123.5\text{ kW})(6.69\ \$/\text{kW})] \\ &= \$10{,}132/\text{yr} \end{aligned}$$

Due to its experimental nature the construction cost for the Fort Stewart system has been inflated to include facilities for instrumentation and data collection. Also, to prevent any compromise in comfort of the host facility (PX operation) during the construction and any catastrophe due to unexpected failure of the system, the DIS cooling system has been designed for redundancy. The PX building can be cooled either by the original cooling system or by the newly installed DIS cooling system. Such a redundancy was necessary not only for the comfort of the hosts but also to analyze the performance of the DIS cooling system with respect to that of a conventional cooling system.

The final total construction cost of the DIS cooling system is $153,295, excluding the cost of the redundant chiller. A detailed cost breakdown and analysis is given in Chapter 5. Based on the estimated annual savings of $10,132/yr in demand charges and the construction cost of $153,295, the simple payback period of the system is about 15 years. At this point, it should be emphasized again that the rather prolonged payback period of this system should not discourage potential customers. The redundancy and research capability which had to be built into the present system cause it to present a biased picture in economic performance. A number of utility companies are willing to subsidize construction of storage cooling systems to the point that they will pay back within 3 years.

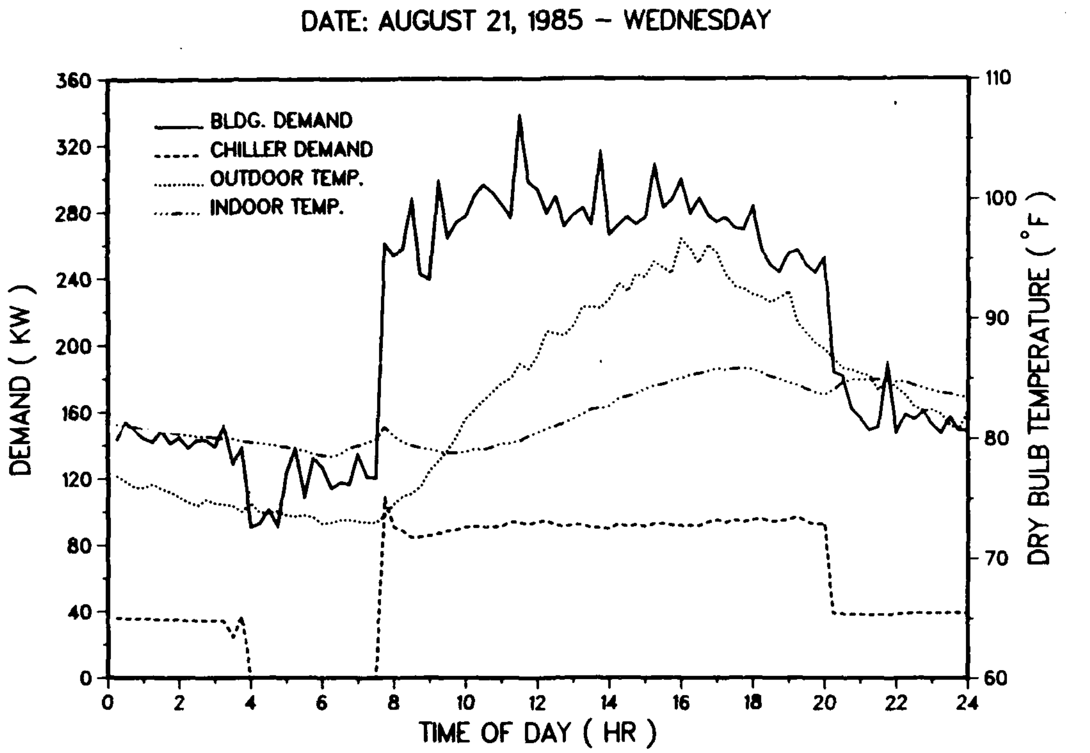


Figure 11. Energy characteristics of Fort Stewart PX on a hot day.

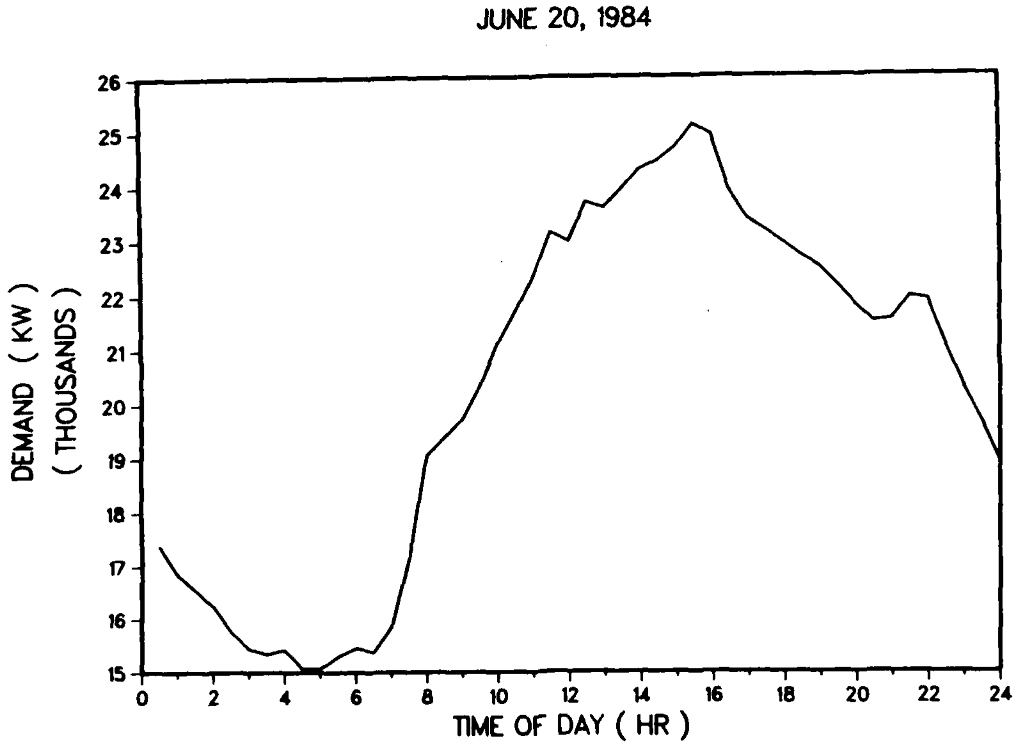


Figure 12. Peak day load profile for Fort Stewart.

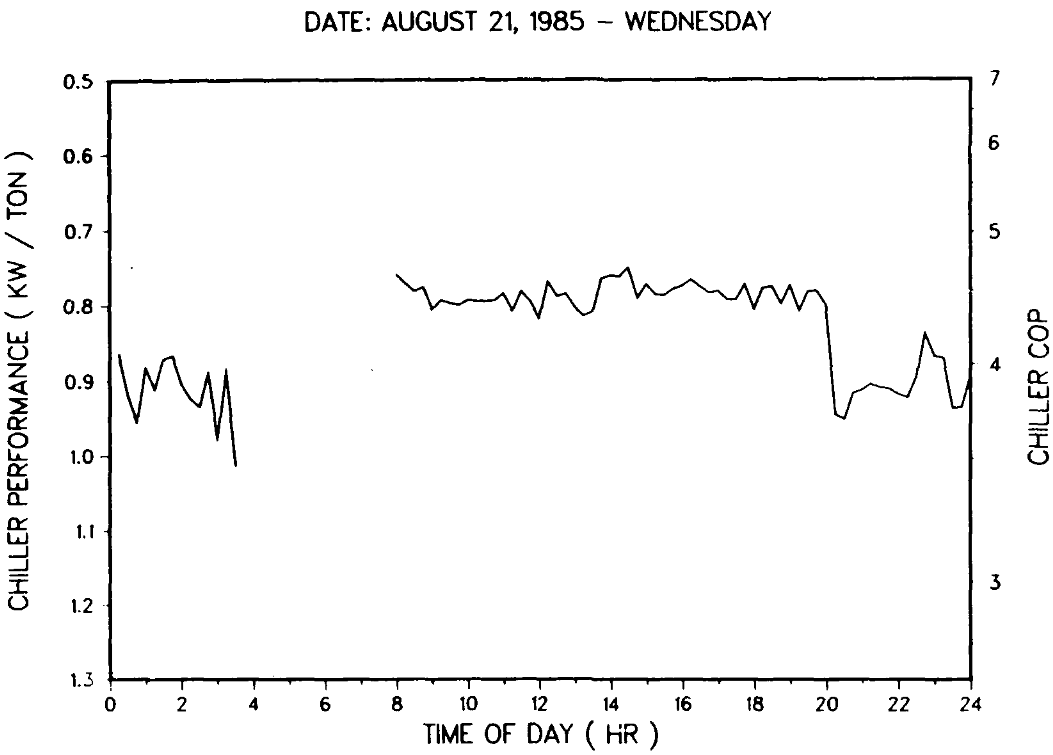


Figure 13. Chiller performance in existing cooling system at Fort Stewart PX.

Table 3

Shortcut Rate Schedule for Fort Stewart

Under 300 Hr Use of Demand

KWH Range	Correction Factor ($)	Cents/KWH
1 - 50,000	10.00	4.45 + FCR
50,000 - 200,000	125.00	4.22 + FCR
200,000 - 1,000,000	1945.00	3.31 + FCR
Over - 1,000,000	4245.00	3.08 + FCR

Over 300 Hr Use of Demand

KW Range	Correction Factor ($)	$/kW	Cents/kWH
0 - 166	10.00	10.80	0.85 + FCR
167 - 666	125.00	10.11	0.85 + FCR
667 - 3333	1945.00	7.38	0.85 + FCR
Over - 3333	4245.00	6.69	0.85 + FCR

Add excess reactive demand at $0.27 per KVAR
Add sales tax when applicable
Minimum bill: $6.00/kW of billing demand in excess of 30 kW, plus $10.00 per meter, plus excess reactive demand charge, plus FCR.

Table 4

Summary of 1986 Monthly Utility Bills for Fort Stewart

Date Read	Actual Demand	Billing Demand	Kilowatt Hours	Fuel Charge	Bill Amount	Demand* Charge	Hour* (Billing)
01 24	17,510	24,697	9,676,800	183,936.61	435,673.54	169,455.42	391
02 24	19,680	24,697	9,542,400	181,381.94	431,972.96	169,455.42	386
03 24	17,856	24,697	8,505,600	161,674.44	403,904.10	169,455.42	344
04 23	17,500	24,697	8,697,600	165,323.98	408,766.60	169,455.42	352
[illegible] 23	23,155	24,697	10,809,600	205,468.88	467,604.92	169,455.42	437
[illegible] 24	26,112	26,112	14,342,400	272,620.34	574,377.35	178,921.77	543
07 24	26,918	26,918	14,630,400	267,048.69	576,719.82	184,313.91	543
08 25	27,379	27,379	15,436,800	281,767.91	601,324.34	187,398.00	563
09 24	27,360	27,360	12,614,400	230,520.64	525,615.04	187,270.89	461
10 23	26,419	26,010	11,750,400	214,480.05	493,381.47	178,239.39	452
11 20	19,085	26,010	8,659,200	158,056.38	410,441.49	178,239.39	333
12 22	17,587	26,010	9,696,800	176,630.63	437,489.03	178,239.39	392
12 MO ENDING	266,561	309,284	134,342,400	2,498,640.49	5,767,270.66	2,119,899.84	----

*Reconstructed

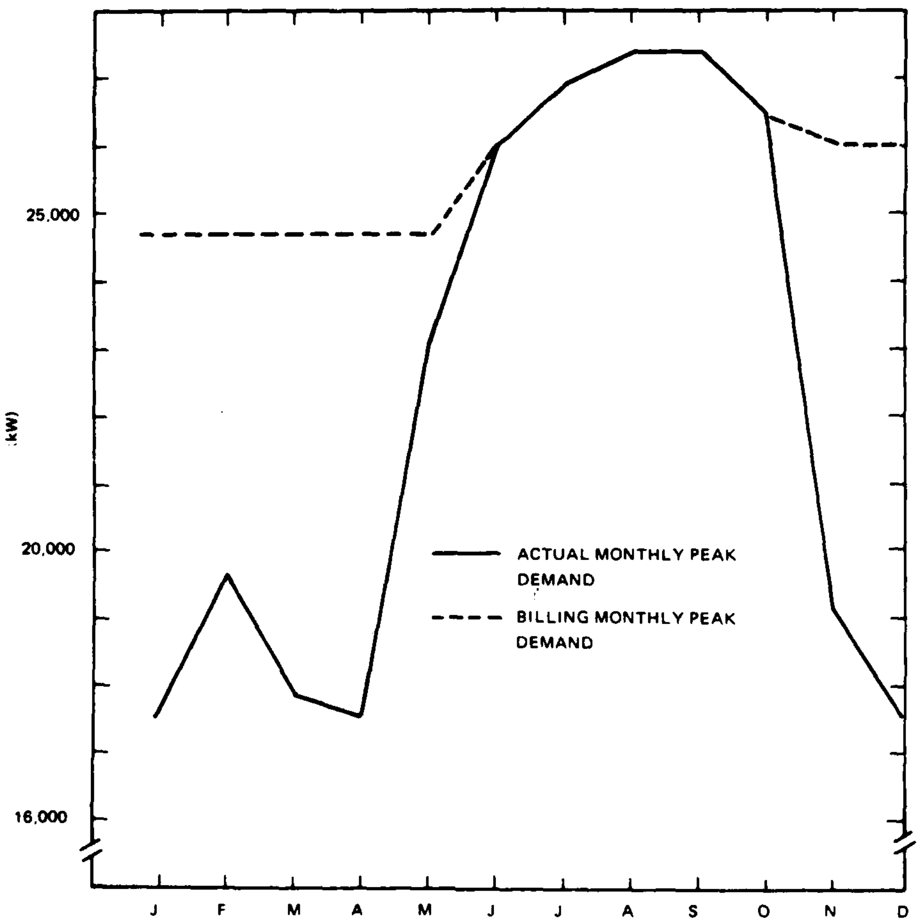


Figure 14. Actual and billing peak demands for Fort Stewart.

4 SYSTEM DESIGN

Design Rationale

At Fort Stewart, as at many other Army installations, all electrical power is supplied to the installation through a single master meter which measures electrical energy (kWh) as well as electrical power (kW) supplied to the installation. The electrical demand at any instant is simply the sum of the instantaneous demands for each electrical component in the installation. To reduce the total electrical demand at a particular time of the day, some electrical component (a chiller in the case of a DIS cooling system) must be turned off during the typical 15-minute peak demand period.

The first step in designing a DIS cooling system was examining the 24-hr peak day electrical demand profile for the installation and defining a window during which the chiller should be turned off. The 1984 summer peak day demand profile, provided by the Georgia Power Company, is shown in Figure 12. It indicates a definite peak of about 25 MW that occurs around 3:30 p.m. A 6-hour window from noon to 6:00 p.m. was selected to provide a sufficient margin to cover variations in the peaking hour.

The next step was determining the 24-hr cooling load of the PX building on a design day so that the sizes of the system components could be determined. A demand-limited storage design was selected in which the entire cooling load of the building during the 6-hr window would be met by the storage cooling system. In late summer of 1985, the building's cooling load was instrumented by measuring the temperatures of the chilled water supply and return mains, and the flow rates through the chiller. These data determined the total (sensible and latent) building cooling load for a design day. Two criteria were used for selecting the design day: (1) the day had to be a usual business weekday so that the effect of internal heat loads from people, interior lighting, computers, etc. would be included, and (2) the exterior dry bulb temperature must approach the ASHRAE summer design day temperature profile.[12] For Savannah, the location nearest to Fort Stewart for which data were available, the 1 percent design day maximum temperature is 96 °F, reached at 3:00 p.m. The design day temperature range is 20 °F. The design day information was fitted to a sine curve (Figure 15)* with an amplitude of 10 °F and period of 24 hour, so that dry bulb temperatures could be determined for any hour of the theoretical design day. An examination of all data collected during the summer of 1985 revealed that the measured outdoor temperature on Monday, 19 August 1985, approximated the theoretical design day temperature profile. Thus, it was reasonable to assume that the building cooling load measured on 19 August 1985 can be considered the design cooling load on which a DIS cooling system design could be based.

The capacity required of the ice storage system is determined by the building cooling load during the window selected:

$$C = \int_{t_1}^{t_2} \dot{Q}\, dt \qquad \text{[Eq 9]}$$

where C is the storage capacity, t_1 and t_2 the beginning and closing hours of the window, respectively, and $\dot{Q}$ is the instantaneous building cooling load. Figure 15 shows the

[12] *1985 Fundamentals Handbook* (ASHRAE, 1985).
*Figures and tables for this chapter begin on p 31.

measured building cooling load profile for the design day as determined from the chilled water temperatures and flow rate. The capacity, C, in Equation 9 corresponds to the area bounded by the load profile and the x-axis during the window. Based on 10-minute data collections on 19 August 1985, a nomograph, shown in Figure 16, was developed to allow the storage capacity to be sized as a function of beginning and ending hours of the window of demand shift. Selecting the onpeak beginning time (beginning of the window) and onpeak ending time (ending of the window) on the nomograph determines a point whose y-coordinate provides the required storage capacity. For a window beginning at noon and ending at 6:00 p.m., the nomograph shows about 700 ton-hr for the required storage capacity.

Ice Storage System Selection and Sizing

The decision to test an ice-in-tank DIS cooling system at Fort Stewart rather than another type of DIS cooling system was based on the following considerations: (1) operating modes required by the building, and (2) available space in the existing mechanical yard adjacent to the building. It can be seen from Figure 17 that the building has a nighttime cooling load as well as the typical daytime load. To accommodate the nighttime load, the storage system must be designed so the chiller builds ice while still satisfying the nighttime cooling load. A simple approach would be to bleed some of the stored refrigeration from the ice tank as it is being charged. A second (and more efficient) method would be providing a brine (heat transfer fluid) loop between the chiller evaporator, the ice tank, and a heat exchanger to provide cooling to the building. In this way, cold brine from the evaporator will make the ice first, and then pass through the heat exchanger to provide cooling to the building before returning to the evaporator.

Further, the system must provide for normal chiller operation in the morning when the storage system is fully charged. With the brine loop in place, the storage tank could be valved out of the system so that cold brine from the evaporator could be sent directly to the building heat exchanger. A DIS cooling system in which the storage unit is remote from the chiller evaporator had an advantage in this approach.

A secondary consideration was efficient utilization of the available space for the system. Although ample space was available around the building, a modularized storage system consisting of a battery of closely packed storage tanks had the advantage of maximum compactness and reduced installation costs for ground preparation and security fencing.

Several Calmac models (the most typical ice-in-tank DIS cooling systems) were investigated. These systems consist of one or more insulated polyethylene tanks containing a multicircuited, spiral-wound, plastic-tube heat exchanger surrounded with water as shown in Figure 18. At night, during offpeak hours, a 25 percent brine (ethylene glycol) solution from a standard packaged chiller circulates through the heat exchanger (plastic tube) in the tank, and extracts heat from the surrounding body of water until all the water in the tank is frozen solid. During the daytime (onpeak hours), the circulating brine, carrying heat from the building, is cooled by the melting ice. Calmac manufactures several sizes of ice tank (all vertical cylinders) with various amounts of heat exchange surface. The specifications for the Calmac models are given in Table 5.

The next step in the design process was selecting the size and number of ice storage tanks. From an examination of the available space, the total storage capacity required (700 ton-hr) and the nominal discharge time (6 hr), Calmac Models 1100 and 2090 were selected for further evaluation. Several possible configurations were

analyzed, and the results are given in Table 6. Case 7 will be discussed in detail here to illustrate the steps in the analysis.

Step 1: Assume the following: the ice tanks (Model 2090) will be discharged in a 6-hr period; the brine returning to the tanks will be at 50 °F; the brine leaving the tanks will be blended to 42 °F; and the offpeak, ice building time is 10 hr. From the discharge data provided by Calmac in Figure 19, with a 6-hr discharge under the stated conditions, the available capacity is 81 ton-hr rather than the nominal 90 ton-hr. As shown in Figure 19, this derating of the ice tanks worsens with lowered blended outlet temperatures and can be an important parameter, especially if low temperatures from storage are required.

In addition, the discharge performance curves in Figure 19 indicate that the discharge power, Q, under the assumed conditions is 13.6 tons/tank. If there were 10 Model 2090 tanks in the storage system, a total of 810 ton-hr would be available with a discharge power of 136 tons.

Step 2: The brine flow rate (in gpm) during discharge can be calculated from the brine specific heat, the rate at which ice is melted, the discharge power (Q), and the entering/leaving brine temperature difference (ΔT). A simple formula for this calculation is the following:

$$\text{flowrate(gpm)} = (25.5/\Delta T)Q \qquad \text{[Eq 10]}$$

Under the conditions specified in Step 1, this formula yields

$$\text{flow rate} = (25.5/8)(13.6) = 43.3 \text{ gpm/tank} \qquad \text{[Eq 11]}$$

Step 3: If the same flow rate calculated in Step 2 is maintained during system charging and if the charging time is known, the brine temperature when it enters and leaves the ice tank array during this period can be determined. The charging rate is given by

$$810 \text{ ton-hr}/[(10 \text{ tanks})(10 \text{ h})] = 8.1 \text{ tons/tank} \qquad \text{[Eq 12]}$$

With this rate and the manufacturer's data in Figure 20, the average entering brine temperature is found to be 26 °F, the brine temperature rise is about 5 °F, and from the table at the bottom of this figure, the minimum temperature for brine leaving the chiller evaporator and entering the ice tank is 23.6 °F.

Step 4: The final information required for the system design is the brine pressure drop across an ice tank. From the manufacturer's data for tank Model 2090 shown in Figure 21, the pressure drop is 16.8 psi (or 39.3 ft of water).

Chiller Selection/Sizing

The existing chiller in the PX building is a 175-ton Trane Centravac centrifugal chiller which is not capable of efficient icemaking. It was decided at the outset of the project to keep the existing HVAC system (including chiller) intact, and to design the ice storage system as a separate package with minimal impact on the existing HVAC components and controls. The rationale of this approach was discussed in the economic analysis section of Chapter 3. By keeping the existing chiller operational, tests to compare the efficiency of the ice system and the conventional system could be performed easily.

Therefore the design called for a new, reciprocating chiller to be installed as part of the ice system. The following is an outline of the process used in selecting this chiller.

A rough idea of the chiller capacity required was determined by examining the nighttime building cooling load and the capacity required for charging storage since both of these loads occur simultaneously. From Figure 16, the design day cooling load is reasonably uniform at 40 tons throughout the night. To charge 810 ton-hr in 10 hr period requires an average cooling capacity of 81 tons at 26 °F. Thus the total chiller capacity required is

$$\begin{array}{rl} & \text{81 tons (icemaking)} \\ + & \underline{\text{40 tons (nighttime cooling)}} \\ & \text{total 121 tons at 26 °F.} \end{array} \qquad \text{[Eq 13]}$$

Nominal chiller capacity would be higher at Air-Conditioning and Refrigeration Institute (ARI) rated conditions: 85 °F entering condensing water temperature (ECWT) and 44 °F leaving chilled water temperature. The low evaporator temperatures required for icemaking give rise to a "chiller derating" typically taken to be about 30 percent.[13] With this derating value used as a starting point, the nominal chiller size would be

$$121 \text{ tons}/(1 - 0.3) = 175 \text{ tons.} \qquad \text{[Eq 14]}$$

Based on this nominal size, a representative chiller was chosen and a detailed analysis of the performance of the chiller under icemaking conditions was compared with the ice system performance found in Steps 1 through 4. The liquid chiller chosen was a dual-compressor, semi-hermetic, water-cooled package (Trane Model CGWB-D18E). Available chiller performance data, shown in Table 7, cover the range of leaving chilled water temperatures suitable for nonstorage applications. The trend in capacity reduction with decreasing leaving chilled water temperature is about 3 tons/°F. The rate was verified by the manufacturer as acceptable for extrapolation to 26 °F for icemaking, provided a capacity adjustment factor of 0.986 is also included due to the use of a 25 percent glycol solution rather than water. Thus, the chiller capacity available for icemaking is

$$C = 0.986[155.6 \text{ tons} - (3 \text{ tons/°F})(40 \text{ °F} - 26 \text{ °F})] = 112.0 \text{ tons} \qquad \text{[Eq 15]}$$

This is a conservative value: the capacity is rated at 90 °F ECWT, but with the lower nighttime ECWT, the capacity will be greater. Since nighttime cooling requires about 40 tons (Figure 17), the difference (72 tons) is available for icemaking. At this capacity (72 tons), 10 units of Model 2090 tanks can be charged to a capacity of 720 ton-hr during the 10-hr charging period.

During the morning, the chiller operates in the conventional manner with about 160 tons of capacity available at 90 °F ECWT and 42 °F, leaving brine temperature more than adequate for the 125-ton morning load experienced on the design day (see Figure 17).

From the analysis of several configurations (Table 6) it appears that the discharge power (column 10) is the limiting parameter, rather than storage capacity. In case 2, with nine Model 2090 tanks, the usable storage capacity of 729 ton-hr is adequate; however, the discharge power of 122 tons is marginal. Adding the tenth tank (case 7) brings

[13] *1987 HVAC Systems and Applications* (ASHRAE, 1987).

the discharge power up to an ample 136 tons; yet at 810 ton-hr the system now has more storage capacity than needed. Apparently, for this particular application and type of ice storage system, heat transfer to the melting ice during discharging imposes a limitation. For short discharge periods, ice storage systems that present a larger ice surface area to the heat exchange fluid would be favored.

The system selected for installation at the PX building is case 7. This provides a good balance between chiller capacity, storage capacity, and discharging capacity; each is adequate to meet the peak and offpeak cooling demands of the building with a safe margin. During the second year of this project, after the system design was completed, a new addition to the building was constructed. The new addition has its own cooling plant to meet the increased load it created. However, the new addition may alter the cooling load of the original PX building and thus may affect the performance data of the DIS system.

System Layout

The ice storage system was designed to minimize the impact on the existing HVAC components of the PX building and to permit the facility to use either the new system or the conventional centrifugal chiller. By keeping the existing chiller operational, in situ performance testing/comparison of the storage cooling system and the conventional cooling system is possible. The design selected and shown schematically in Figure 22 accomplishes this objective.

The existing mechanical room (shown at the right of Figure 22) consisted of two system pumps in parallel (P-4 and P-5), a 175-ton centrifugal chiller (CH-1) coupled to a matched cooling tower, and a condensing water pump (P-3). The existing control system was based on maintaining a fixed return water temperature from the building at all times. This was accomplished by modulating the vanes on the chiller to regulate cooling capacity. The ice storage system design (shown on the left side of Figure 22) was located in the mechanical yard outside the building, requiring two wall penetrations to make connections to the supply and return water lines. Since a brine-based storage system was selected, a heat exchanger (HX-1) was required to transfer heat from the chilled water distribution system in the building to the storage system. Two pumps (P-1 and P-2) were used to circulate the brine solution to the evaporator of the new chiller (CH-2), to the battery of ten ice storage tanks, to the heat exchanger (HX-1), and back to the pump suction. The flow control valve (FCV-2) is a modulating valve designed to maintain a fixed return water temperature from the building. A pressurized expansion tank (ET-1) is used to maintain brine operating pressure and to accommodate volumetric changes with temperatures. An isometric drawing (Figure 23) shows the physical arrangement of the various system components.

Auxiliary Equipment Selection/Sizing

The remaining components of the ice storage system design were selected and sized according to conventional engineering practice. From sizing the ice tanks described earlier, it was determined that the brine flow rate through the system would be 434 gpm (43.4 gpm through each ice tank). To keep the piping pressure losses at reasonable levels, preliminary piping selections were made such that the flow velocities were 10 ft/s or less. To accommodate this guideline, pipe sizes of 4 in. and 5 in. were chosen initially, and equipment connections were specified to accommodate this range of pipe sizes.

The heat exchanger (HX-1) was specified as a plate frame type for reasons of compactness, generally low pressure drop, and availability of low approach temperatures. Heat exchanger specifications were determined based on chilled water flow rate and chilled water return temperature (those existing in the system before the ice storage system was installed), and on the capacity of the new chiller while providing cooling directly to the building through the heat exchanger. A 2 °F approach temperature was arbitrarily selected as an initial value for the following iteration procedure:

1. Select a chiller leaving brine temperature;

2. From chiller data (Table 7) determine the chiller capacity;

3. Based on the chiller capacity in the last step, the chilled water return temperature, and flow rate to the heat exchanger from the building calculate the chilled water temperature entering the load side of the heat exchanger;

4. Compare the calculated approach temperature with the approach temperature initially selected. If these two temperatures do not agree, select another approach temperature and repeat the procedure once again.

It was found that the 2 °F selection was reasonable, yielding the heat exchanger specifications given in Figure 24.

Due to the limited availability of published data on plate frame heat exchanger performance, manufacturers were contacted to determine the size of a unit with the specifications given in Figure 24 and the pressure drops that would be found on the glycol and water sides. A single-pass model with 239 plates, water and glycol side pressure drops of 1.6 psi and 2.2 psi respectively, and a "footprint" of 3 ft by 7 ft was selected.

The expansion tank (ET-1) was sized based on an analysis of the quantity of brine in the system, the specific gravity of the brine, and the temperature range to which the brine would be subjected. These data are as follows:

Brine volume:	72 ft of 5-in. pipe @ 1.04 gal/ft	=	75 gal
	225 ft of 4-in. pipe @ 0.66 gal/ft	=	149
	Heat exchanger (volume = 10 cu ft)	=	75
	10 Ice tanks (88 gal/tank)	=	880
	Chiller evaporator	=	67
	TOTAL		1246 gal

Brine specific gravity = 1.043 at 15 °F

= 1.022 at 110 °F.

Based on these data and ASHRAE sizing practices[14] the required volume of the expansion tank was calculated to be 36 gal. A diaphragm-type expansion tank that could accept this volumetric change was installed. The diaphragm allowed the tank to be pressurized so that the available positive suction head at the brine circulating pumps could be made to exceed the required head without overpressurizing the plastic heat exchanger tubing in the ice tanks. As in standard engineering practice, the expansion tank was

[14] *1982 Systems Handbook* (ASHRAE, 1982).

located at the pump suction side to give a point of no pressure change with system operation.

The piping system designed for the ice storage system consisted of 6-in. steel pipe connected at two locations to the existing chilled water distribution system; these pipes run through the back wall of the building and connect to the heat exchanger. For freeze protection, the chilled water piping outside the building was provided with thermostatically controlled heat tape. Steel piping (5-in.) was specified for the brine loop between the heat exchanger, brine pumps, ice tank header, and chiller evaporator. The piping to the ice tank array was polyvinyl-chloride (PVC) in a full reverse-return configuration so that the pressure drops across each tank are equal, resulting in even brine distribution to/from each tank.

The tank header piping is 4-in. PVC; 2-in. stubs lead to each tank. The ice tank connections were made according to the manufacturer's suggestion using reinforced flexible hoses between the 2-in. stubs and the tank connections.

The design called for every component (except the pumps) to be provided with isolation valves (ball or gate) for ease in servicing the system. The diverting valve (FCV-1) and mixing valve (FCV-2) (Figure 22) were specified as 6-in. for minimum pressure drop over all modes of system operation. The flow coefficients for FCV-1 and FCV-2 were 250 and 390 respectively so that pressure drop under actual flow conditions could be determined. The pressure drop for the chiller evaporator was determined from information provided by Trane to be 6.9 psi using the 25 percent glycol solution, and the 434 gpm flow rate needed for the ice tank array was well within acceptable flow rates and will not cause abnormal erosion of the evaporator tubes.

Based on pressure drop data, a preliminary brine pump selection was made. This selection consisted of a pair of end-suction, close-coupled, centrifugal pumps, each developing 220 gpm at 107 ft head, piped in parallel. Final pump selection was the contractor's responsibility and was based on actual piping/equipment installed.

Modes of Operation/Controls

To achieve the design objective of preventing onpeak chiller operation, three operating modes were established. Using an electronic, programmable timer, the system operates in each of the three modes once each day. Figure 22 shows the valves that are used to establish these modes of operation.

Mode 1: 6:00 a.m. to 12:00 noon

Brine is circulated by pumps P-1 and P-2 through the 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, while 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, the 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 flowmeter 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 set so that brine is circulated by pumps P-1 and P-2 through the battery of ice tanks. Valve FCV-2 operates as a control valve in the same manner as before.

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

The system can easily be returned to conventional (nonstorage) operation by closing valves V-2, V-3, V-6, and V-7 and opening valves V-1 and V-8.

Electrical service to the new chiller (CH-2) was provided by wiring taps into the circuit for the existing chiller (CH-1). The control circuit for CH-1 was modified through electrical interlocks such that if CH-2 were operating, CH-1 could not be turned on. A simple switch located in the mechanical room selectively activates CH-1 or CH-2.

The system design also included capabilities for monitoring brine flow rate, brine temperatures, and electrical power consumption of the chiller CH-2 and brine pumps P-1 and P-2. These instrumentation locations are shown in Figure 22, with JE standing for electrical power, TE for temperature, and FE for flow rate. Data from these instruments will be taken during the 1987 and 1988 cooling seasons to evaluate the system performance.

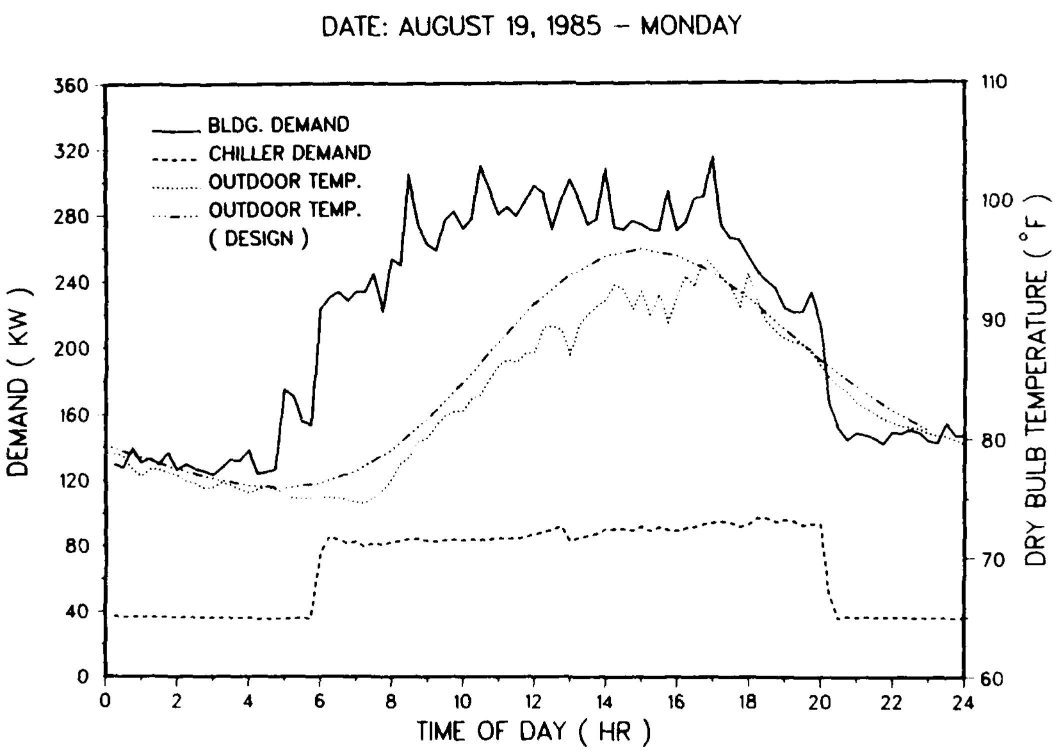


Figure 15. Main exchange power requirements on a design day.

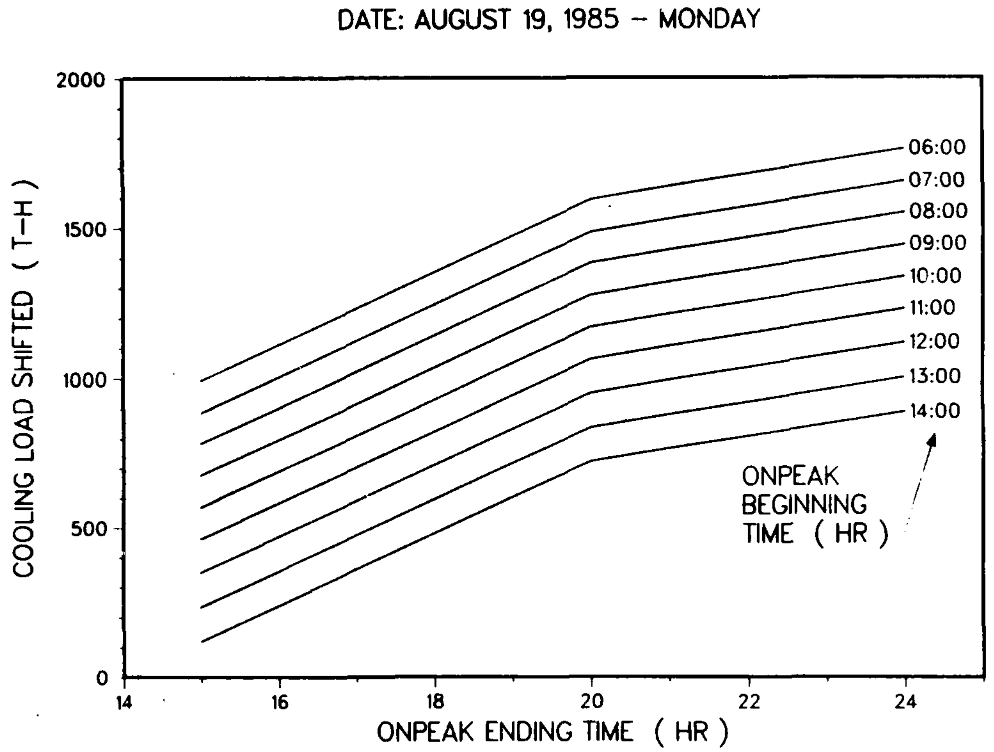


Figure 16. Dependency of cooling load shift on operating time (nomograph).

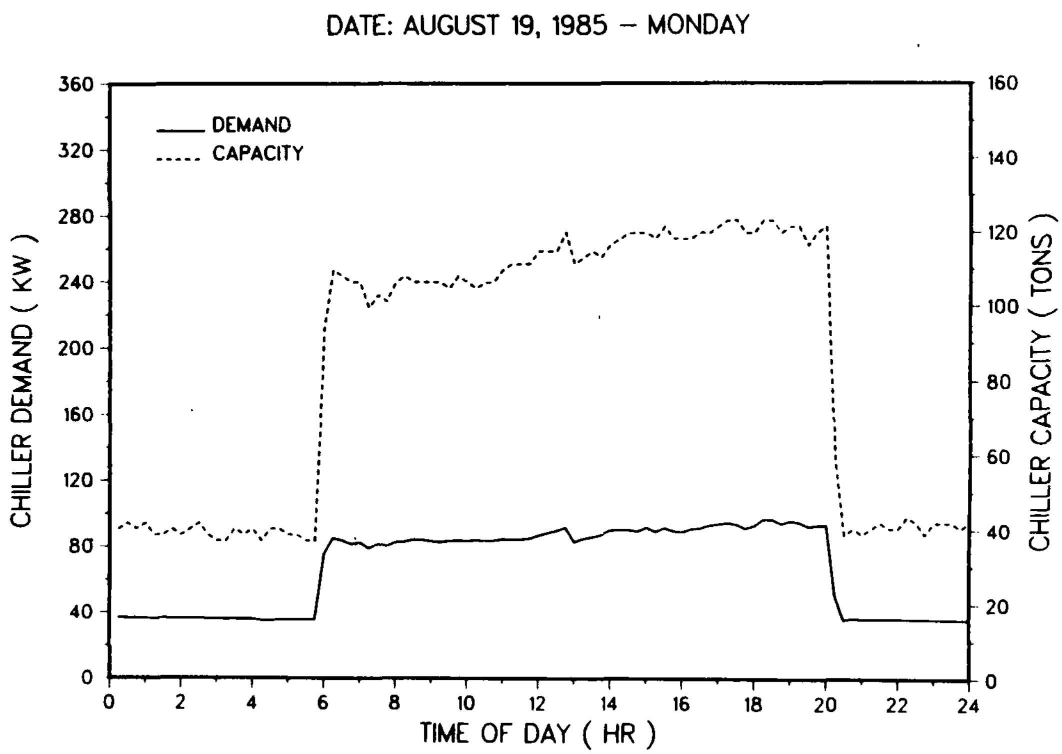


Figure 17. Demand and capacity of existing cooling system.

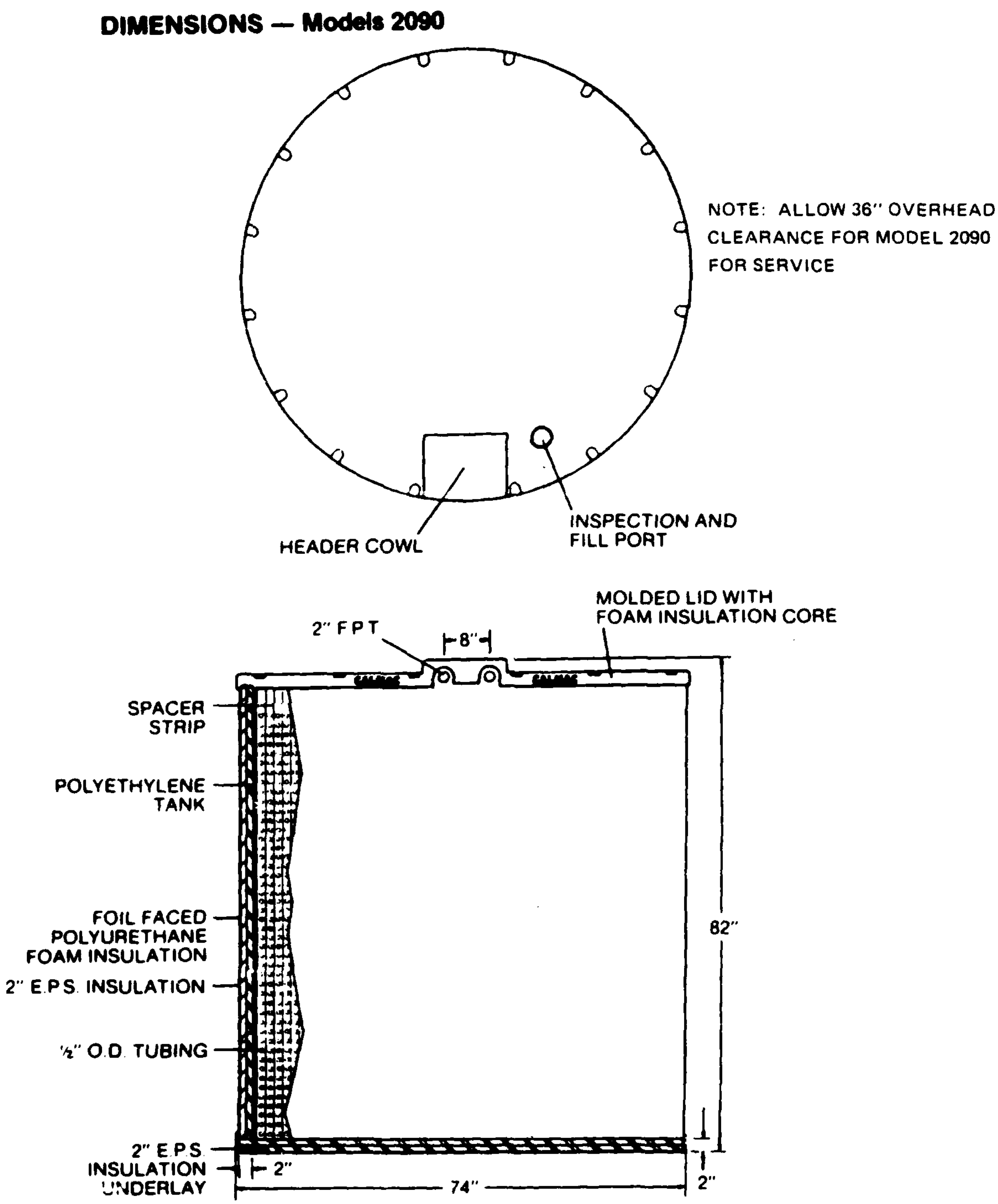


Figure 18. Detail of Calmac Model 2090 ice bank.

Table 5

Calmac Ice Tank Specifications

Specifications	Model			
	3060	2090	1100	1190
Total ton-hr capacity	60	90	100	190
Tube surface/ton-hr sq ft	34.8	22.1	10.9	12.0
Nominal discharge time, hr	1-4	4-8	6-12	6-12
Latent storage capacity, ton-hr	51	76	85	162
Sensible storage capacity, ton-hr	9	14	15	28
Max. operating temp, °F	100	100	100	100
Max. operating pressure, psi	90	90	90	90
Outside diameter, in.	56	74	74	89
Height, in.	98	82	82	100
Weight, unfilled, lb	700	1,080	850	1,550
Weight, filled, lb	5,520	8,300	8,300	16,750

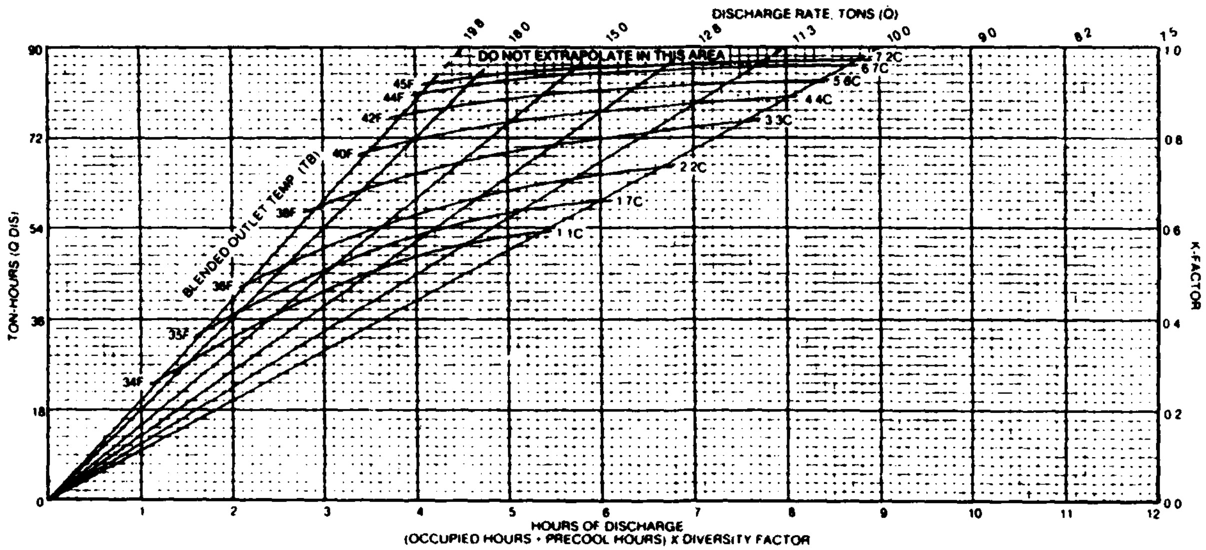
Table 6

Summary of Design Studies

Case No.	Discharge time (h)	T(out) (°F)	T(in) (°F)	Tank Model	K-factor	Discharge power (tons/tank)	Number of tanks required (nominal)	Energy stored (Ton-h)	Discharge power (Tons)	Discharge flowrate (gpm)
1	6.0	40.0	50.0	1100	0.68	11.3	10.3(11)	748	124	316
2	6.0	42.0	50.0	2090	0.90	13.6	8.6(9)	729	122	390
3	6.0	44.0	50.0	2090	0.94	14.2	8.2(9)	761	128	543
4	6.0	45.0	50.0	2090	0.96	14.6	8.1(9)	778	131	670
5	6.0	42.0	50.0	1100	0.71	11.9	9.9(10)	710	119	379
6	6.0	44.0	50.0	1100	0.75	12.5	9.3(10)	750	125	531
7	6.0	42.0	50.0	2090	0.90	13.6	10.0	810	136	434
8	6.0	44.0	50.0	2090	0.94	14.2	10.0	846	142	604
9	6.0	42.0	50.0	1100	0.71	11.9	9.9(10)	710	119	379
10	6.0	44.0	50.0	1100	0.75	12.5	9.3(10)	750	125	531
11	6.0	42.0	50.0	2090	0.90	13.6	10.0	810	136	433
12	6.0	44.0	50.0	2090	0.94	14.2	10.0	846	142	603

Case No.	Charge rate (tons/tank)	Charge flowrate (gpm/tank)	Min. ent. brine temp (°F)	Chiller* capacity (tons)	Chiller charging capacity (T)	Ice build time (h)	Tank Pressure drop during charging (ft H_2O)	Tank Pressure drop during discharging (ft H_2O)
1	6.8	28.7	22.9	108.0	68.0	10.0	31.7	31.7
2	8.1	43.4	23.6	112.0	72.0	10.0	39.3	39.3
3	8.5	60.4	24.0	112.0	72.0	10.0	----	----
4	8.7	74.5	----	----	----	10.0	----	----
5	7.1	37.9	23.7	112.0	72.0	10.0	48.6	48.6
6	7.5	42.0	25.8	120.0	80.0	10.0	----	----
7	8.1	43.4	23.6	112.0	72.0	10.0	39.3	39.3
8	8.5	60.4	23.9	112.0	72.0	10.0	----	----
9	5.9	37.9	26.8	123.0	83.0	12.0	47.6	47.6
10	6.3	32.0	25.8	120.0	80.0	12.0	----	----
11	6.8	43.4	26.8	123.0	83.0	12.0	39.3	39.3
12	7.1	30.0	25.4	120.0	80.0	12.0	----	----

Notes: ---- = Exceeds maximum allowable pressure drop.
* = Assumed chiller: Trane CGWB-D18E at 85 °F ECWT and minimum entering brine temperature.

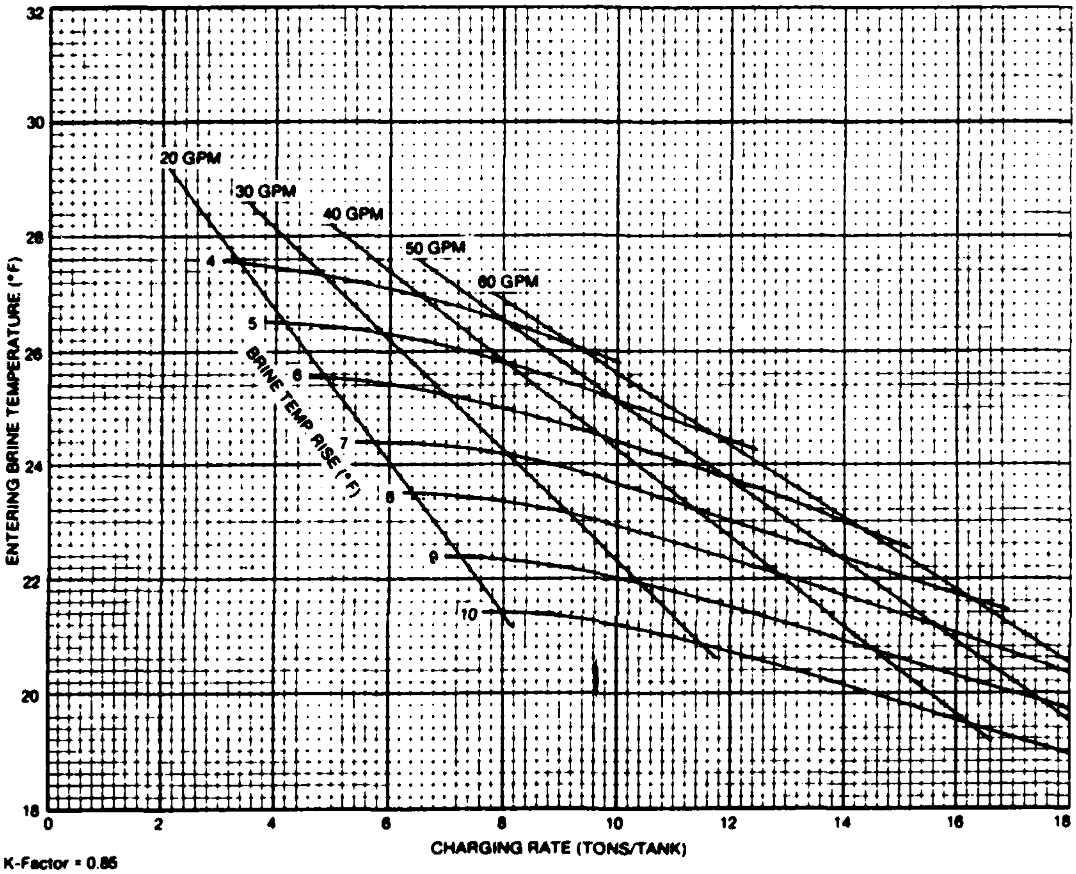


Model 2090
Nominal 90 Ton-Hours
Constant Inlet Temperature - 50F (T in)
Constant Discharge Rate ($\dot{Q}$)
Constant System Flow (GPM-$_{sys}$)
For 25% by volume EG/H_2O

$$GPM\text{-}_{sys} = \frac{25.5}{(T\ in - TB)} \times (\dot{Q})$$

Figure 19. Discharge performance—Calmac Model 2090 with 50 °F return to ice tank.

Model 2090

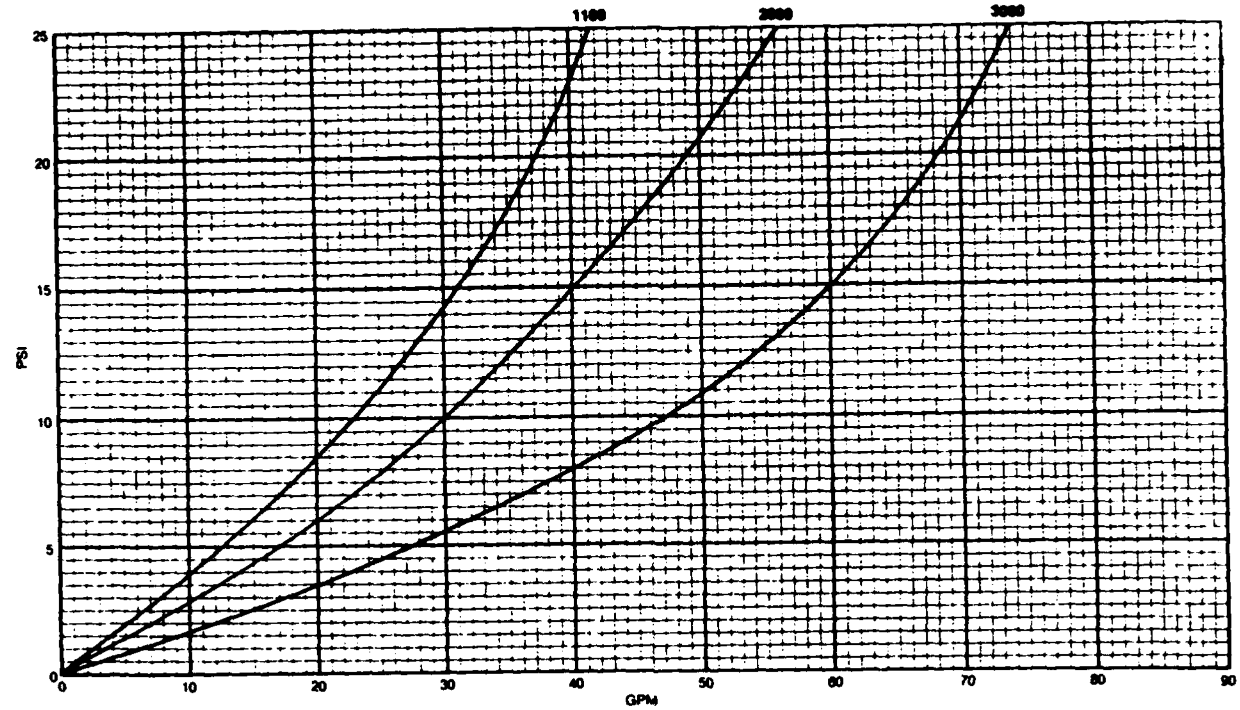


K-Factor = 0.85
Brine Solution = 25% EG/H_2O

Minimum Entering Brine Temperature To Ice Bank For Full Charge

Average Charging Temp.	Tons/Tank								
	2	4	6	8	10	12	14	16	18
28		26.0							
27		25.1	24.6	24.0					
26		24.4	24.1	23.6	23.0				
25		23.6	23.2	22.8	22.4				
24			22.4	22.2	21.7	21.3			
23			22.0	21.6	21.1	20.4	21.2		
22				21.0	20.6	20.0	19.4	19.0	
21				20.2	20.0	19.6	18.9	18.4	18.0
20							18.7	18.1	17.5
19								17.6	17.2
18								17.1	16.8

Figure 20. Charging performance—Calmac Model 2090.



Model 1100: 1½" Inlet, 1½" Headers
Model 2090: 2" Inlet, 1½" Headers
Model 3080: 2" Inlet, 1½" Headers
All Models: 25% by Weight EG/H_2O

28F, 4.0 CP, 1/18/85

Figure 21. Pressure drop data—Calmac Model 2090.

Table 7

Performance of Trane Liquid Chiller (Model CGWB-D18E)

Leaving chilled water temperature (°F)	Entering condenser water temperature (°F)									
	75		80		85		90		95	
	tons	kW	tons	kW	tons	kW	tons	kW	tons	kW
50	200.0	153.3	195.0	148.9	190.0	154.4	184.8	160.0	179.5	165.6
48	193.7	152.0	188.9	147.5	183.9	152.9	178.8	158.3	173.6	163.7
46	187.5	140.7	182.8	146.1	177.9	151.0	173.0	156.6	167.9	161.7
45	184.5	140.0	179.8	145.4	175.0	150.6	170.1	155.7	165.1	160.7
44	181.4	139.4	176.8	144.6	172.1	149.7	167.3	154.8	162.3	159.8
42	175.4	138.0	170.9	143.0	166.3	148.0	161.5	152.9	156.7	157.7
40	169.4	136.5	165.1	141.5	160.6	146.3	155.6	151.0	151.2	155.7

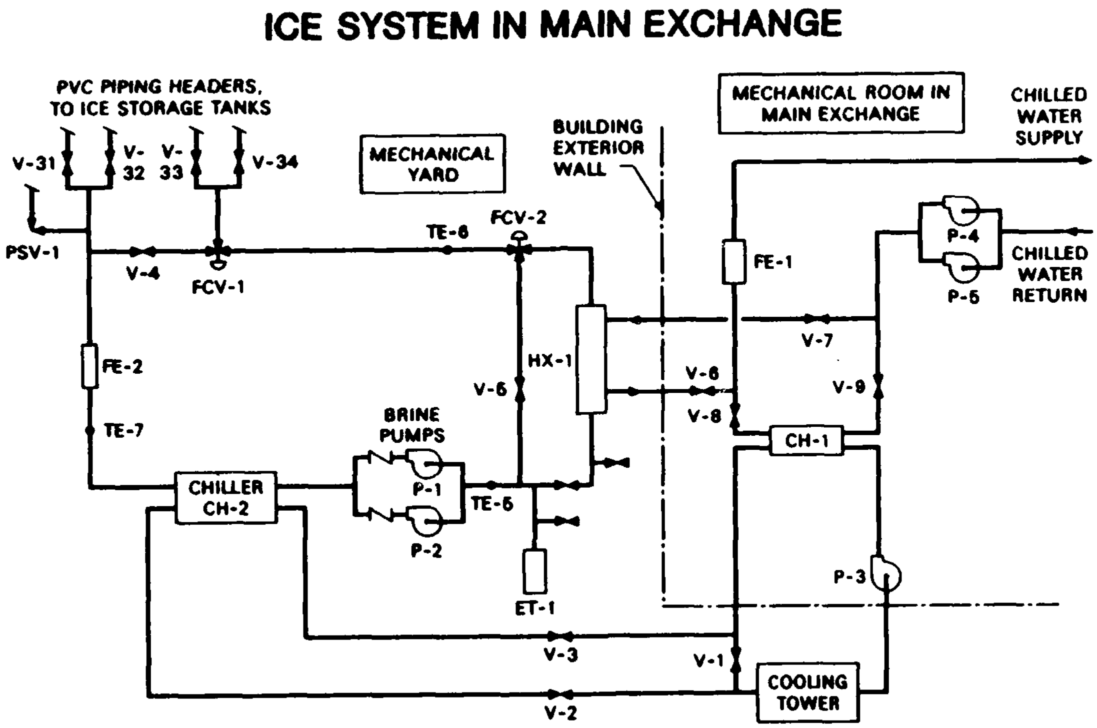


Figure 22. Ice system schematic.

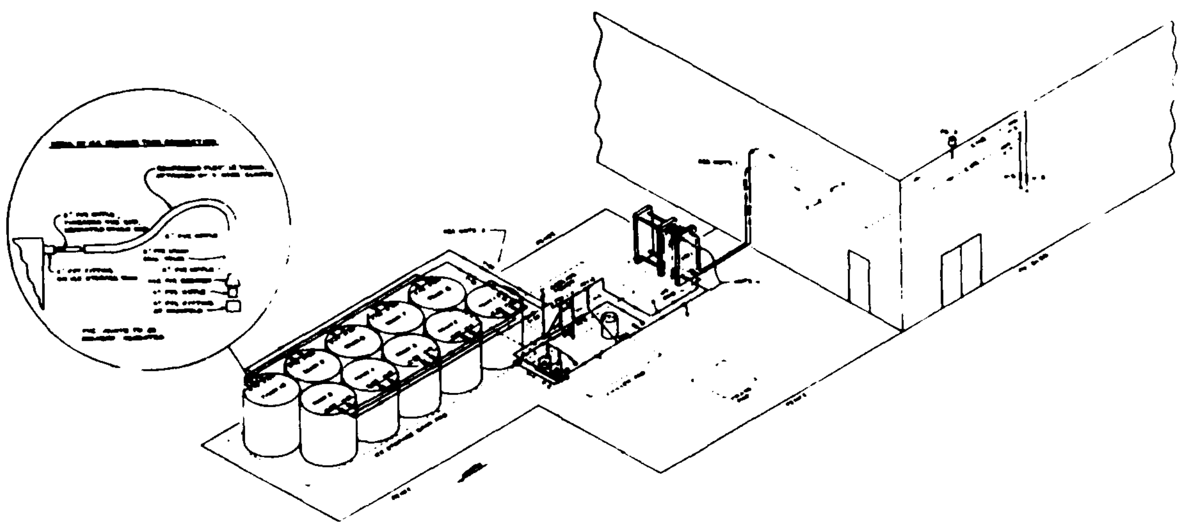


Figure 23. Ice system isometric.

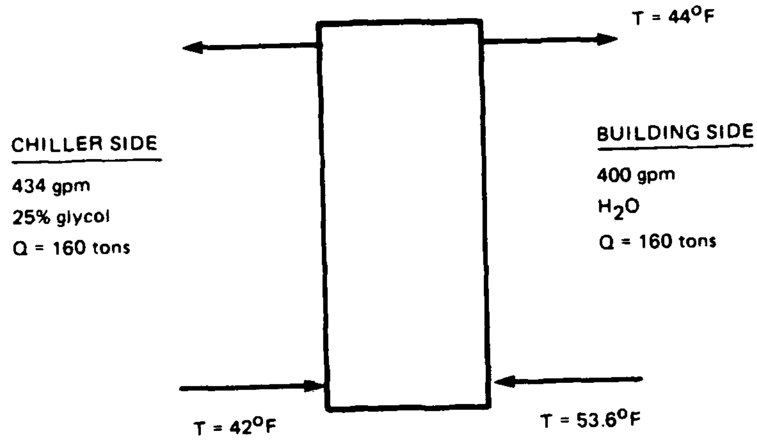


Figure 24. Heat exchanger specifications.

5 SYSTEM CONSTRUCTION

Construction Logistics

ORNL and USA-CERL designed the demonstration system and produced construction specifications in cooperation with the Engineering Services Division, Fort Stewart. Major equipment (ice storage tanks, a chiller, and a heat exchanger) were procured by ORNL and provided to the contractor as Government furnished equipment. This step was necessary to accelerate the project schedule, although it may not be an ideal approach. (The selection criteria in Chapter 3 recommend single-source responsibility.) A construction contract was awarded by the Fort Stewart Contracting Office to a local firm through a competitive bidding process. Erickson's, Inc., the awardee, installed the DIS cooling system for the PX building. USA-CERL managed the project execution from inception of the project to acceptance of the installed DIS cooling system by Fort Stewart. Table 8 summarizes the chronology of the project. Figures 25 through 27 show the progress at the site: (1) before the installation, (2) in the middle of installation, and (3) after completion of installation of the DIS cooling system.

Table 8

Chronology of the Project

01 October 1985	Funding authorized.
01 November 1985	Ft. Stewart, GA was slected to be the demo site; initial project conference at Fort Stewart.
27 November 1985	ORNL's system design and draft constuction specifications com pleted and shipped to Fort Stewart.
06 February 1986	Project advertised through the *Commercial Business Daily* by Fort Stewart.
12 February 1986	ORNL initiated major equipment procurement process.
07 May 1986	Construction specifications completed, and RFP issued by Fort Stewart.
09 June 1986	Bid opening (three bids received).
10 July 1986	Major equipment delivered to Fort Stewart.
20 July 1986	Contract awarded to Erickson's Inc.
01 August 1986	Preconstruction conference; notice to proceed issued to contractor.
07 November 1986	Pre-final-acceptance test.
01 April 1987	Final acceptance of system by Fort Stewart.

Figure 25. Mechanical yard of Fort Stewart PX before DIS installation.

Figure 26. Mechanical yard of PX during DIS system installation.

Figure 27. Completed DIS system.

Problems Encountered and Lessons Learned

It took 18 months for the project to be completed from the initial funding stage to the final acceptance of the completed system by the user, as summarized in Table 8. It can be divided into 2 months of system design, 8 months of construction contract awarding, 4 months of system installation, and another 4 months of fine tuning and system acceptance. The 8-month period of the contracting process was longer than the typical period (90 to 120 days) required for awarding a minor construction contract. It was due to uniqueness of this project in coordinating activities among multiple organizations, i.e., ORNL, USA-CERL, and Fort Stewart. The actual construction of the DIS cooling system for the PX building proceeded rather smoothly. The following are experiences from the construction phase of the project which could be of interest for future projects.

Construction Cost

ORNL estimated the net construction cost to be $59,280 to install the DIS cooling system with the Government furnished equipment, 10 ice storage tanks, a plate heat exchanger, and a reciprocating chiller. At the opening of the bids, Fort Stewart received three bids which quoted $83,900, $113,900, and $114,900. The $83,900 bidder, Erickson's, Inc., was awarded the contract. A breakdown of Erickson's quote was not available due to the proprietary nature of the information. However, through an informal discussion with the contractor it was learned that the quote included roughly $15,000 and $10,000 worth of subcontracting for insulation and control system work, respectively. The Calmac ice tanks are factory insulated and shipped to the site; therefore, the Erickson's quote should have been highly conservative. In retrospect, it is conjectured that the

unexpectedly high quote of the cost for the control work could have been due to the uniqueness of the project, i.e., its research application, and the contractor's unfamiliarity with storage cooling systems. Excluding these components, ORNL's cost estimate ($59,280) and the projected contractor's bid ($58,900) are within 1 percent of each other. A breakdown of the ORNL's construction cost estimate is given in the Appendix.

Although DIS cooling is a newly emerging technology, installation of the system is a routine process. The close agreement between the ORNL's cost estimate and the contractor's bid shows that an able, practicing contractor can competently plan his/her work and budget schedules for system installation. Therefore, installation of a DIS cooling system is not a major venture either for the Army or for a contractor.

Contractors, however, are still somewhat uneasy about installing storage cooling systems, as shown by the conservative overbids for the project. Unfamiliarity with the storage cooling technology, rather than the complexity of the system, probably would be the main reason for a conservative approach. An informal interview with the contractor after the system was installed illustrates that experience creates a more positive attitude: when asked whether he would bid on another DIS cooling system installation, the contractor was more than willing to undertake another project. With further dissemination of the storage cooling technology in the private sector as well as in the Army, more contractors would be exposed to the new technology and their bids would become more competitive.

The major purpose of this demonstration project is a rapid dissemination of the storage cooling technology within the Army. In order to accelerate the construction schedule, major equipment was purchased by ORNL while Fort Stewart conducted the contracting process. This shortened construction time by at least 5 months because the equipment was delivered to the site ready for installation when the construction contract was awarded to a contractor. It took 5 months for the equipment delivery from the time it was ordered. If a contractor is required to furnish the equipment, which might be preferable for a routine installation an extended construction period must be provided to take into account the equipment delivery schedule.

Construction Period

The contractor was required to commence within 10 calendar days and complete the work within 60 days from the date of notice to proceed. A preconstruction conference was held on 1 August 1986, and the notice to proceed was issued at the conclusion of the conference. Therefore, a pre-final-acceptance test was scheduled on 1 October 1986; however, the actual test was conducted on 7 November 1986 due to a delay caused by a routine contract modification as discussed in the following section. The system was put into continuous operation on the day of the test. Installation of the ice storage tanks and the chiller offered no difficulties, and controls for operating the chiller for icemaking performed as designed. However, after a few weeks the chiller developed a malfunction. Ordering replacement parts, repairing, and testing the system took roughly 3 months. The system was again put into operation on 15 March 1987, and was formally accepted by Fort Stewart on 1 April 1987. During the test period for system operation before the final acceptance by Fort Stewart, the system successfully provided cooling to the PX building as designed. The events which delayed construction of the DIS cooling system were routine ones. They did not originate from unexpected complexities with the storage cooling technology. Each of these events is discussed below.

Modification of contract. A modification was required to reinforce the concrete pads for the ice storage tanks. Half of the ground surface planned for the tank pad was a part of the existing concrete paved parking lot. Fort Stewart expressed concern about the strength of the proposed concrete pad along the boundary between the existing concrete surface and the fresh soil. The pad was redesigned to reinforce the potential fault line, and the cost was reflected in the contract modification. Another modification of the contract was related to the uprating of the new chiller. The original design specified a 175-ton reciprocating chiller, and Trane Co. was selected to supply one through an open bidding. The chiller supplier was behind in its delivery schedule for the ordered unit and proposed to deliver a similar reciprocating chiller with a 200-ton-capacity unit at no extra cost. Therefore, power lines to the chiller were upgraded, and this was reflected in the modified contract. The last modification of the contract was procurement of additional ethylene glycol by the contractor for the loop between the chiller and the ice storage tanks. The modifications required an additional $4,120 for the project.

Malfunction of the reciprocating chiller. A few days after the system pre-final-acceptance test was completed, a seal in the chiller barrel broke. A new part was supplied by the Trane Co., and it was replaced by the contractor. Since the chiller was covered under the manufacturer's warranty, all costs associated with the repair were paid by the chiller manufacturer.

6 CONCLUSION

Design

The ice-in-tank cold storage system that was installed in the PX building at Fort Stewart was designed for complete redundancy with the existing system (for performance comparison and back up). ORNL completed the design of the DIS colling system and auxiliary equipment using standard practices. It is expected that the reduction in the peak demand will save Fort Stewart over $10,000/yr. The salient features of the design are listed below.

PX floor area	51,000 sq ft
Hours of operation	9:30 a.m. - 7:30 p.m., Monday - Saturday; 11:00 a.m. - 5:00 p.m., Sunday
Chiller shut-off window	12:00 noon - 6:00 p.m.
Minimum storage capacity (peak load, design day)	700 ton-hr
Tanks	10 in two rows; Calmac Model 2090
Actual storage capacity	810 ton-hr
Discharge power	136 tons
Charging time	10 hours
Charging rate	8.1 tons/tank
Coolant	25 percent brine (ethylene glycol)
Brine flow rate	43.3 gpm/tank
Entering brine temperature	25 °F
Temperature rise	5 °F
Pressure drop across a tank	16.8 psi
Reciprocating chiller	Trane Co. CGWB-D18E, designed for 175 ton unit, 200 ton delivered and installed

Installation

Contract awarding and supervision	Fort Stewart (AFZP-DEE)
Contractor	Erickson's Inc.
Project management	USA-CERL-ES
Schedule	Design--2 months Contract award--8 months Installation--4 months Fine tuning, acceptance--4 months Total--18 months

The contracting process was lengthened by the need for coordination among three organizations. On the other hand, the construction phase was shortened by procuring major equipment through Government channels while the contract was being awarded, saving at least 5 months. Installation of the system proceeded smoothly, with only a few complications, all of which were routine problems typical of any construction project and not unique to cold storage technology. After the system was complete, the contractor felt confident about bidding on other cold storage projects. As the private sector

becomes more familiar with this technology, contractors will be more confident and bids may be lower.

Testing

The performance of the Fort Stewart DIS cooling system is currently being measured and will be reported in the near future. These results will be used to evaluate the efficiency of the present design and to compare it with the other types of DIS systems which will be demonstrated in future phases of this work.

Guidance

Although storage cooling is a so-called new technology, especially for Army engineers, a system was designed and installed using standard engineering practices, as discussed in this report. However, the design and construction process described here should not be taken as a cookbook to be followed strictly. Army engineers are encouraged to improve on the design given here, using this report as a starting point.

METRIC CONVERSION FACTORS

1 in.	=	2.54 cm
1 sq ft	=	0.0929 m^2
1 lb	=	0.453 kg
1 psi	=	6.894 kPa
1 gal	=	3.78 l
°C	=	(°F-32) x (5/9)
1 ton of cooling	=	3.517 kW
1 ton-hr	=	3.517 kWh

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APPENDIX:

COST ESTIMATE FOR SYSTEM CONSTRUCTION

Table A1

Itemized Hardware Cost Estimates

Description	Amount Required	Crew	Daily Output	Unit	Bare Costs Mat.	Bare Costs Inst.	Bare Costs Total	Unit Including O & P	Total Including O & P
A. Piping (chilled water) to HX									
1. 6" gale valve	2	Q-2	3	EA	250	114	364	440	880
2. 6" Tee	2	Q-16	3	EA	93	120	213	276	552
3. 6" Pipe	137 ft	Q-16	36	LF	23	10	33	51	6,960
4. 6" 90° Ell	11	Q-16	5	EA	70	72	142	181	1,991
5. 6" Weld Flange	2	Q-16	6	EA	35	60	95	126	252
6. 10" Core Drill	2	A-1	3.3	EA	11	26	37	50	100
									10,735
B. Piping - HX to Pump Inlet									
1. 5" Pipe	30	Q-15	32	LF	22	8	30	36	1,080
2. 5" 90° Ell	3	Q-16	5	EA	50	72	122	159	477
3. 6 X 5 Reducer	1	Q-16	5	EA	50	72	122	159	159
4. 5" Gate Valve	1	Q-2	3.4	EA	250	101	351	421	421
5. 5" Strainer	1	Q-6	3.4	EA	685	101	786	900	900
6. 5" Tee	2	Q-16	3	EA	93	120	213	276	552
7. 5 x 5 x 2 Red. Tee	2	Q-16	3	EA	70	120	190	251	502
8. 5" Weld Flange	3	Q-15	5	EA	30	48	78	103	309
									4,400
C. Piping - Pump Outlet to Evaporator									
1. 4" Pipe	10 ft	Q-15	37	LF	8	7	15	19	190
2. 4" Weld Flange	3	Q-15	6	EA	22	40	62	82	246
3. 4" Check Valve	2	Q-1	3	EA	645	74	719	817	1,634
4. 4" 90° Ell	1	Q-15	5	EA	26	48	74	98	98
5. 4" Tee	1	Q-15	3	EA	58	80	138	180	180
6. 4" 45° Ell	1	Q-15	5	EA	18	48	66	89	89
									2,437
D. Brine Pump/Motor	2	Q-2	0.4	EA	3,800	857	4,657	5,423	10,846
E. Piping - Evaporator Outlet to Ice Bank									
1. 4" Pipe	23 ft	Q-15	37	LF	8	7	15	19	437
2. 4" Weld Flange	4	Q-15	6	EA	22	40	62	82	328
3. 4" 45° Ell	1	Q-15	5	EA	18	48	66	89	89
4. 4" 90° Ell	3	Q-15	5	EA	26	48	74	98	294
5. 5" Tee	1	Q-16	3	EA	93	120	213	276	276
6. 5 x 5 x 2 Red. Tee	1	Q-16	3	EA	70	120	190	251	251
									1,675
F. Piping-Ice Bank to Heat Exchanger									
1. 5" Pipe	60 ft	Q-15	32	LF	22	8	30	36	2,160
2. 6" 3-Way Valve	2	Q-2	3	EA	1,450	114	1.564	1,760	3,520
3. 5" Butterfly Valve	2	Q-2	3.4	EA	115	101	216	273	546
4. 5" Weld Flange	1	Q-15	5	EA	30	48	78	103	103
5. 6 x 5 Reducer	7	Q-16	5	EA	50	72	122	159	1,113
6. 5" 90° Ell	7	Q-16	5	EA	50	72	122	159	1,113
7. 6" Weld Flange	1	Q-16	6	EA	35	60	95	126	126
									8,681
G. Fill System & Expansion Tank									
1. Ethylene Glycol	250 gal			Gal	3.50	--	3.50	4	1,000
2. 30 Gal. Tank	1	Q-5	12	EA	215	18	233	263	263
3. 2 x 2 x 1 Tee	1	Q-1	15	EA	12	15	27	35	35
4. 2" Pipe	6 ft	Q-1	64	LF	3	3	6	8	48
5. 1" Pipe	3 ft	1 Plumb.	53	LF	2	7	9	12	36
6. 2 x 1 Red	1	Q-1	17	EA	23	13	36	44	44
7. 2 Valves	2	1 Plumb.	19	EA	59	6	65	74	148
									1,574

Table A1 (Cont'd)

Itemized Hardware Cost Estimates

Description	Amount Required	Crew	Daily Output	Unit	Bare Costs Mat.	Bare Costs Inst.	Bare Costs Total	Unit Including O & P	Total Including O & P
H. PVC Manifold Piping									
1. 4" Pipe	200 ft	Q-1	48	LF	4	5	9	12	2,400
2. 4" 90° Ell	9	Q-1	14	EA	5	16	21	29	406
3. 4 x 4 x 2 Tee	20	Q-1	9	EA	6	25	31	43	387
4. 2" Ball Valve	20	1 Plumb.	17	EA	56	7	63	72	1,440
5. 4" Ball Valve	4	Q-1	20	EA	225	11	236	263	105
6. 4" Flange	2	Q-1	16	EA	31	14	45	54	108
7. 2" Flex Tubing	40 ft	1 Plumb.	20	LF	3	6	9	12	48
									6,273
I. Electrical									
1. Motor Starter	2	Elec.	1.2	EA	535	85	620	712	1,424
2. 3-Way Switch/Wiring/ Conduit		2 Elec. 4 Days			50				1,238
									2,662
J. Miscellaneous									
1. Cooling Tower Pump	1	Q-2	0.4	EA	3,800	857	4,657	5,423	5,423
2. Install 175-T Chiller	1	Q-1	1	EA	--	417	--	605	605
3. Install Plate Heat Exchanger	1	J-1	2	EA	--	208	--	302	302
4. Fill System		B-10J	1	EA	0	161	--	233	233
5. Install 6' Chain Fence	85 ft	J-2	75	LF	10	2	12	13	1,105
6. Install Curb	60 ft	2 Mason	30	LF	5.50	4.30	9.80	12.30	738
7. Remove Curb/Fence/Vent		21 Labors 3 Days						418	418
8. Replace Gravel		2 Labors 1/2 Day			75				145
9. Form & Pour Concrete Pads		2 Masons 2 Days			650				1,030
									10,000

Table A2

Labor Crew Cost Estimates

	BASE		O'HEAD & PROFIT (X1.45)	
	HOUR	DAILY	HOUR	DAILY
A-1 CREW				
1 Bldg. Laborer	6.20	49.60	9.00	72.00
1 Gas Engine Tool		34.90		38.40
		84.50		110.40
B-38 CREW				
2 Bldg. Laborers	6.20	99.20	9.00	144.00
1 Equip. Oper. (Light)				
Q-i CREW				
1 Plumber	15.30	122.40	22.20	177.60
1 Plumber Apprentice	12.25	98.00	17.75	142.00
		220.40		319.60
Q-15 CREW				
1 Plumber	15.30	122.40	22.20	177.60
1 Plumber Apprentice	12.25	98.00	17.75	142.00
1 Elec. Weld. Mach.		18.15		20.00
		238.55		339.60
J-1 CREW				
2 Plumbers	15.30	244.80		
1 Hydraulic Crane		87.30		
1 Crane Operator	10.57	84.56		
		416.66		
Q-16 CREW				
2 Plumbers	15.30	244.80	22.20	355.20
1 Plumber Apprentice	12.25	98.00	17.75	142.00
1 Elec. Weld. Mach.		18.15		20.00
		360.95		517.20

Table A2 (Cont'd)

Labor Crew Cost Estimates

	BASE		O'HEAD & PROFIT (X1.45)	
	HOUR	DAILY	HOUR	DAILY
Q-8 CREW				
1 Steamfitter Foreman	15.80	126.40	22.90	183.20
1 Steamfitter	15.30	122.40	22.20	177.60
1 Welder (Steamfitter)	15.30	122.40	22.20	177.60
1 Steamfitter Apprentice	12.25	98.00	17.75	142.00
1 Elec. Weld. Mach.		18.15		20.00
		487.35		700.40
Q-6 CREW				
2 Steamfitters	15.30	244.80	22.20	355.20
1 Steamfitter Apprentice	12.25	98.00	17.75	142.00
		342.80		497.20
Q-2 CREW				
2 Plumbers	15.30	244.80	22.20	355.20
1 Plumber Apprentice	12.25	98.00	17.75	142.00
		342.80		497.20
Q-5 CREW				
1 Steamfitter	15.30	122.40	22.20	177.60
1 Steamfitter Apprentice	12.25	98.00	17.75	142.00
		220.40		319.60
B-10 J CREW				
1 Equipment Operator	10.00	80.00		
1 Laborer	5.50	44.00		
1 Water Pump		22.90		
1 3" Suct. Hose		4.60		
2 50' Disch. Hose		9.20		
		160.70		
J-2 CREW				
2 Skilled Laborers	6.00	96.00		

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