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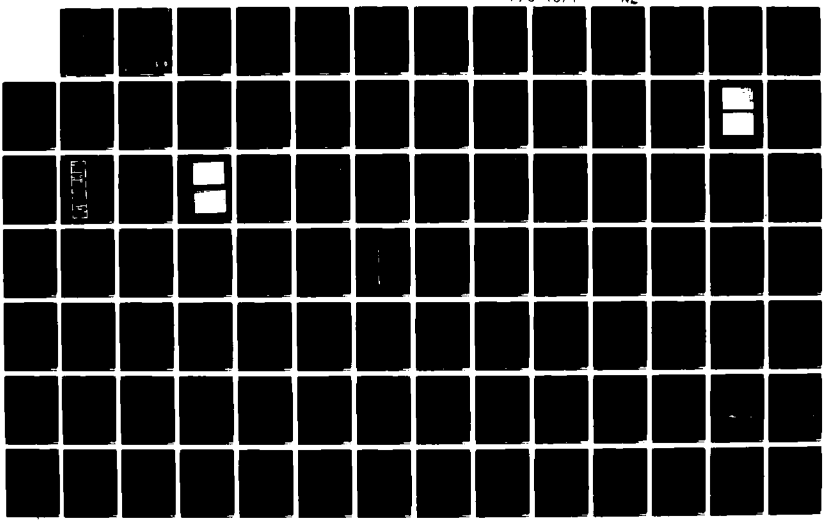

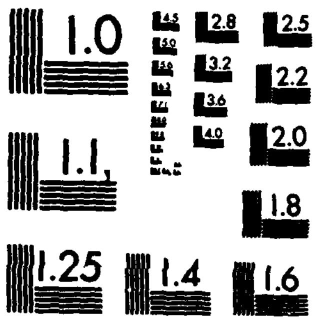

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REPORT DOCUMENTATION PAGE

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1. REPORT NUMBER AFIT/NR 82-83T	2. GOVT ACCESSION NO. AD-A128582	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) Design, Construction, Testing and Evaluation of a Residential Ice Storage Air Conditioning System		5. TYPE OF REPORT & PERIOD COVERED THESIS/~~DISSERTATION~~
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) J. Jay Santos		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS AFIT STUDENT AT: University of Florida		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS AFIT/NR WPAFB OH 45433		12. REPORT DATE Nov 1982
		13. NUMBER OF PAGES 96
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)		15. SECURITY CLASS. (of this report) UNCLASS
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

APPROVED FOR PUBLIC RELEASE; DISTRIBUTION UNLIMITED

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

APPROVED FOR PUBLIC RELEASE: IAW AFR 190-17

LYNN E. WOLAVER 17 Aug 83
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19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

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DESIGN, CONSTRUCTION, TESTING AND EVALUATION OF A RESIDENTIAL ICE STORAGE AIR CONDITIONING SYSTEM

By

J. Jay Santos
Thomas A. Ritz

A MAJOR PROJECT PRESENTED TO THE COLLEGE OF ENGINEERING OF THE UNIVERSITY OF FLORIDA IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF MASTER OF ENGINEERING

University of Florida

November 1982

ACKNOWLEDGEMENTS

We wish to express our sincere gratitude to our supervisory committee, Dr. H.A. Ingley and Dr. E.A. Farber, for their constant support and advice throughout the project.

We would like to thank Rick Dixon, John Murdoch, and Howard Purdy of the Energy Research and Education Park for their invaluable assistance during the construction and testing of the project.

We are indebted to the sponsors of this research. We want to thank Mr. Jack Martin of Crystal Tips Ice Products, McQuay Group, McQuay-Perfex, Inc., who arranged for the use of an ice machine to carry out our research. We'd also like to thank Gainesville Regional Utilities for their major participation in our project.

We would like to extend our appreciation to Denise Jobb for her diligent efforts in typing this report.

Last, but not least, we'd like to thank our wives, Lisa and Karen, for their patience and support over the past six months.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

ABSTRACT

DESIGN, CONSTRUCTION, TESTING AND EVALUATION OF A RESIDENTIAL ICE STORAGE AIR CONDITIONING SYSTEM

By

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Thomas A. Ritz

November 1982

The purpose of this project was to construct and evaluate a residential size air conditioning system which utilizes bulk ice as a thermal storage medium. An experimental system was constructed at the Energy Research and Education Park at the University of Florida.

The experimental system was used to supply cooling to a single wide trailer and performance data were compared to a conventional air conditioning system of the same capacity. Utility rate information was collected from over one hundred major utility companies and used to evaluate economic comparison of the two systems. The ice storage system utilized reduced rate time periods to accomodate ice while providing continuous cooling to the trailer.

The economic evaluation resulted in finding that the ice storage system required over 50% more energy than the conventional system. Although a few of the utility companies offered rate structures which would result in savings of up to $200 per year, this would not be enough to offset higher initial costs over the life of the storage system.

82-83

Recommendations include items that would have to be met in order for an ice storage system to be an economically viable alternative to the conventional system.

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INTRODUCTION

Concept

The use of ice for air conditioning is not a new idea. In fact, the first recorded use was in 1833, when a doctor utilized air blown over buckets of ice to cool a hospital for malaria and yellow fever patients. During the late 1800's many prominent hotels, restaurants, and theaters were cooled by ice ventilating systems. But, by the early 1900's most cooling was done by mechanical refrigeration and the ice systems became obsolete. Cool storage is also an old principle, applied in ancient times to dwellings in hot climates by constructing massive walls or roofs. These walls would lose heat (store cold) during the cool nights and absorb heat the following day (Bullock, Reedy, and Groff, 1979).

The combination of these two concepts lead to the use of ice making equipment and storage vessels to cool facilities that had little or no air conditioning load most of the time, but extremely large loads once or twice per week. Typical applications included churches, gymnasiums, and auditoriums. In fact, the University of Florida's Florida Gym was cooled in this manner as late as 15 years ago. Three or four ice machines would be making ice two to three days before a basketball game or commencement ceremony. This ice would be stored in insulated vessels and used during the period of the event. These ice storage systems were built with first cost savings in mind. It was less expensive to purchase a few small ice machines and a storage system rather than expensive refrigeration equipment to meet that very large peak load. Over the past decade, the rise in electricity rates has created new incentives for ice storage systems. Specifically, the large demand charge penalties and the advent of time-of-day rate structures have made the concept of ice storage extremely attractive in some areas. Figures 1A and 1B show the power demand profile of a conventional unit versus that of an ice storage unit. Also depicted are conventional electrical rates and time of day rates.

Initial interest in this project began with a design project for Dr. H.A. Ingley's Advanced Refrigeration class. Our problem statement was:

> Design a thermal energy storage system for cooling utilizing ice as the storage medium for a residence with a cooling load of 38,000 Btuh. Assume full load equivalent operating hours (FLEOH) of approximately 1600 hours.

The design consisted of a commercial ice maker mounted on top of a well insulated storage box. This was sized to satisfy the total air conditioning load. The required quantity of ice would be produced during off peak hours, taking advantage of substantial cost savings. Water circulates from the bottom of the box to a coil inside the air handler and then back to the top of the box where it is sprayed over the existing water/ice mixture (see Appendix A). Although the project looked promising on paper, many assumptions were made in the sizing of an ice maker and storage box and questions

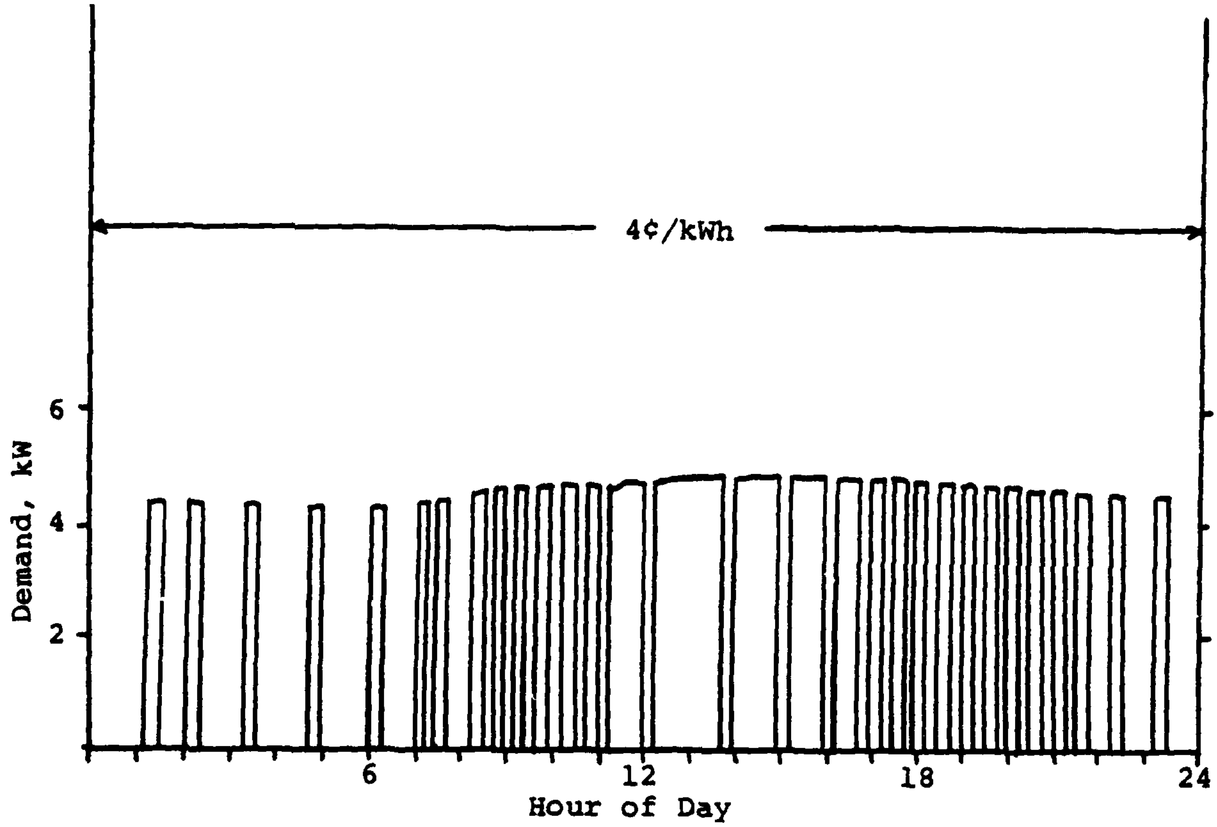


Figure 1A. Conventional Unit Demand Profile

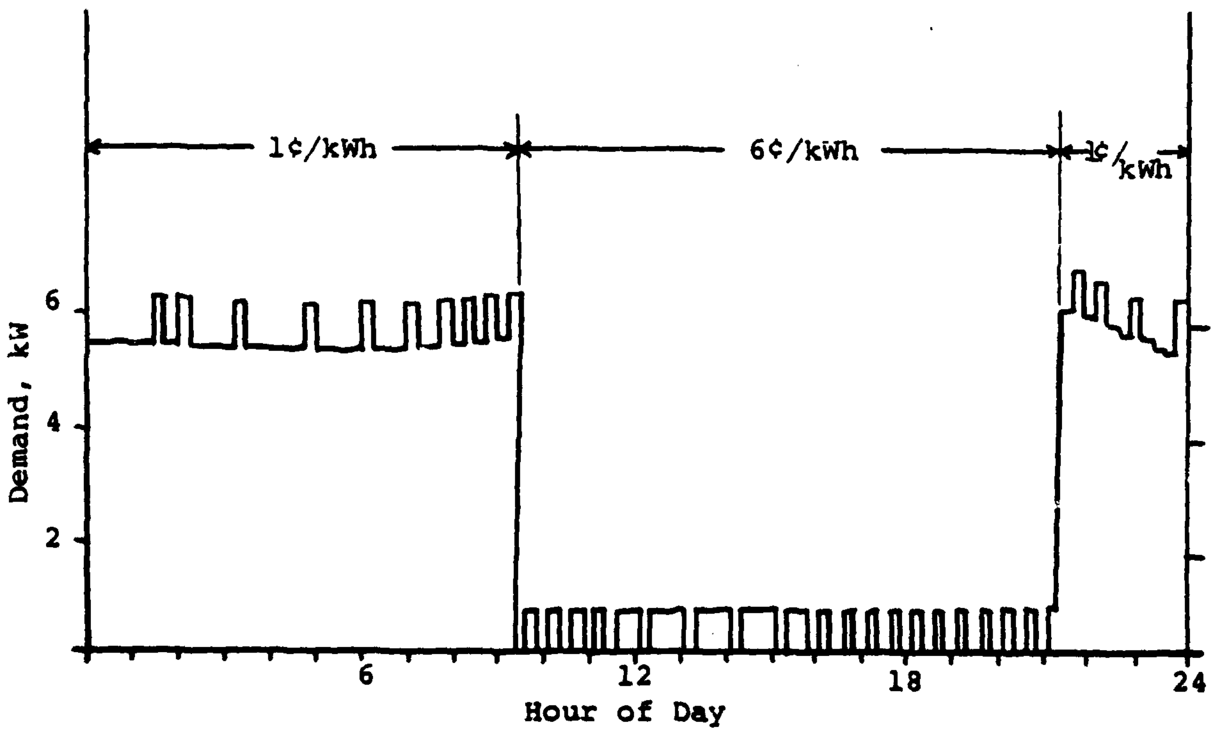


Figure 1B. Ice Storage System Demand Profile developed on how best to control the quantity of ice made. The only way to answer those questions and verify the accuracy or inaccuracy of the assumptions was to build, test, and analyze the design.

Scope

The scope of the project included satisfying all of the following objectives:

1. Construct an operational unit similar in design to the unit designed for Advanced Refrigeration, EML 5605.

2. Test and evaluate this unit under a variety of time-of-day rate structures (i.e. 12 hr off peak/12 hr on peak, 16 hr off peak/8 hr on peak, etc.).

3. Determine rate structures required to make this type of unit economically feasible.

4. Compare results to those predicted or expected in the design phase.

5. Compare results with those of a conventional air conditioning system.

6. Based on results, estimate additional benefits of a larger unit (small commercial application) which would be realized due to demand charge reduction.

7. Gather information on time-of-day rate structures that are in use, being tested, or under consideration by utility companies in different areas of the country.

8. Gather information on other residential and commercial projects using ice for cold storage.

9. Gather information on various sizes and types of commercial ice makers to use in evaluation of larger systems.

10. Make recommendations on changes to ice making control strategy and to sizing of ice maker and storage box.

Literature Review

The use of ice for comfort cooling is an old principle, but the recent energy shortage has sparked renewed interest in ice as a cold storage medium. The researchers' interest in ice storage was restricted to its use to reduce system demand and take advantage of time of day electric rates.

A good deal of emphasis was placed on chilled water storage in the seventies for these same reasons. Recently, the emphasis has shifted to ice storage systems. The main advantage of ice storage over chilled water storage is that less storage volume is required due to the latent heat of fusion of water. This will allow a 75% - 90% reduction in storage box size (Kohlenberger, 1981).

There are two basic types of ice storage systems. A static system is defined as one where ice is frozen on evaporator coils and left there (see Figure 2A). A dynamic system is one where the ice is periodically removed as in an automatic ice maker (see Figure 2B). The latter system employs a separate ice storage unit, while the static system uses the same box for freezing and storage (Bullock, Reedy, Groff, 1979).

Over the past few years, several applications of commercial ice storage systems have been designed or constructed. The subsequent paragraphs briefly describe some of these.

Wisconsin Electric Power Company, of Milwaukee, Wisconsin, was building an electric utility service center, with 10,750 square feet of conditioned space, which utilized a static ice storage system. The ice storage tank capacity was selected based on a maximum condition of 50% ice and 50% water. The tank provided storage for 12,730 pounds of ice. The system used a 25 ton compressor (rated @ 40°F suction) which would freeze the 12,730 pounds of ice in 13.5 hours,

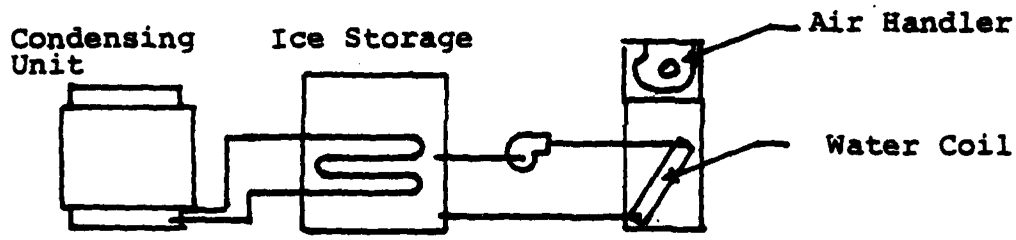


Figure 2A. Static Ice Storage System

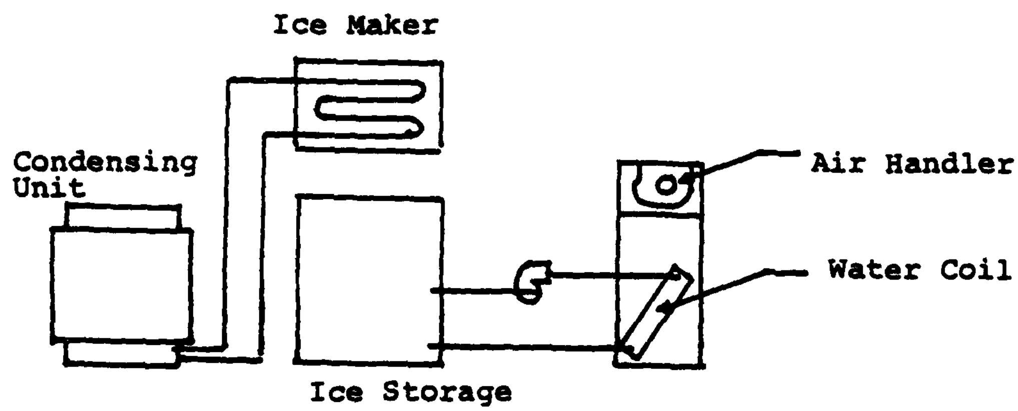


Figure 2B. Dynamic Ice Storage System

starting with 52°F water. The estimated additional cost of this system, over a conventional direct expansion system was $12,000 (Jorgensen, 1979).

Kohlenberger Associates of Fullerton, California performed a feasibility study for the use of an "ice builder" system to supplement an existing chilled water system in a complex of eleven buildings. Maximum daily load analysis indicated an ice storage capacity of 1,200,000 pounds would be required. The total estimated cost of this system was a little over $1 million, $190,000 more than a conventional system. The maximum kilowatt demand for the ice system was 1,082 kW while that of the conventional system was 1,875 kW. This represents a 42% reduction in maximum kW demand. It was estimated that the ice storage system would consume 5% more energy annually. This is due to lower chiller suction temperatures and storage system losses. Even though the ice system uses more energy than the conventional system it will save $87,000 per year in energy costs due to time of day rates and demand charge reduction. The additional cost of the ice system would pay back in about 2.2 years (Kohlenberger, 1981).

In Texas, a 495,000 square foot high school is being planned using a static ice storage system. The peak electrical demand will be reduced 43-48%. The ice building apparatus in this application will consist of 4,000 PVC pipe sections 20 feet long and 6 inches in diameter filled with 31 gallons of water each. This water will be frozen during the weekends and evenings and a water-glycol mixture will be circulated over the pipes and piped throughout the building (Engineering News Record, September, 1980).

The St. Mary's Health Center in St. Louis, Missouri is cooled with an "ice bank" air conditioning system. A study revealed an extreme variable load of 45 tons peak loading and 15 tons average loading. Refrigeration was supplied by two 7 ton units which builds up ice on a 1 inch evaporator coil inside a 300 cubic foot storage tank with 2,250 gallons of water. At night ice would accumulate (optimal thickness of 1.5 inches) in order to satisfy the following days' peak loading. The savings of this system were three-fold: (1) initial equipment cost halved that for a conventional 45-ton unit, (2) a 45-ton unit required triple the available electric service capability, and (3) peak demand service charges have been significantly reduced (Rublee, 1979).

An expansion of a Union Oil Company of California research center in Buea, California will incorporate a static ice storage system. Ice is built up during off peak times in an ice bank storage unit. During peak cooling periods, the conventional chillers are turned off and return water is circulated through the ice bank and mixed with some 32°F water from the ice storage unit, supplying water to the air handlers at about 42°F. Rough estimates put the cost of the new system at $250,000 to $300,000 with annual savings of about $60,000 (Engineering News Record, June 1979).

The new Iowa Public Service Company Corporate Headquarters in Sioux City, Iowa makes use of a dynamic ice storage system. This 167,635 square foot building has, instead of conventional equipment, six large commercial ice makers and a 75,000 gallon ice storage pit. The units used were North Star Model 60 ice machines, each capable of producing 50,000 pounds of ice per 24 hours (Progressive Architecture, April, 1982).

The increased interest in thermal storage as an economical means to air condition or heat a facility has brought about the development of some packaged thermal storage units. Some of these systems are summarized in the following paragraphs.

Caloskills Thermal Energy Storage System is a heavily insulated water vessel with internal refrigeration coils. The unit may be equipped with electric resistance heaters to provide hot water storage in the winter in addition to ice storage for air conditioning purposes. During off peak hours, up to 50% of the water may be turned into ice. The system would shut down the compressors during peak periods, utilizing the ice build up for cooling. An alternative operating strategy would allow the compressor to operate continuously producing a load-leveling effect and decreasing the equipment size required. This unit is available in a variety of sizes, ranging from 48 ton hours to 540 ton hours storage capacity. The current price for these units range from $6,000 - $28,000 for cooling only systems. Additional cost for heating storage capability range from $3,000 - $6,000 for the above sizes.

Calmac Manufacturing makes an ice storage module which freezes water solid. This system is available in two sizes, 36 ton hours and 54 ton hours, and may be connected in series to provide for larger applications. The ice banks are made of all plastic construction using a spiral wound mat type heat exchanger. These units contain 300 and 450 gallons of water respectively. The company estimates that installed costs of their ice storage system is less than conventional systems when partial storage is desired (i.e. 24 hour operation of a smaller unit to achieve load leveling). It also estimates typical paybacks of 2.3 years for systems desiring 100% energy storage (i.e. total ice production during off peak periods).

Baltimore Aircoil also manufactures an ice bank type ice storage system (static). This unit is available in sizes ranging from 145 to 1200 ton hours. For larger application, they sell their ice chiller evaporator coils for custom "field erected" storage units.

Crepaco also manufactures similar ice builders that range in size from 12 to 600 ton hours. They also build custom field units for larger applications.

All of the previous listed manufacturers make units for fairly large commercial applications. A. O. Smith makes an Ice Bank Tank, equipped with evaporator coils designed to be used with R-22 air conditioning condensing units and chilled water air handling units. This unit comes in 9 ton hour or 13 ton hour sizes which held 108 and 155 gallons of water, respectively.

A study of residential cool storage, conducted by five utility companies under contract to Oak Ridge National Laboratory concluded the following:

> ".....residential cool storage as a load management option is not ready for commercialization in its present state of development. Problems were experienced with inadequate compression capacity, inadequate storage capacity, high energy usage, and poor equipment reliability..."

The report went on to say that results suggested when equipment was properly designed, cold storage could provide a viable option to shift a large portion of peak power to off peak hours (Kuliasha, 1981).

A study conducted by Southern California Edison Company of a commercial application of ice storage shows better results. The ice storage system is used to cool a 168,000 square foot electronics assembly plant in Garden Grove, California. The system, in operation since 1979, consists of two 100,000 pound ice storage banks in series with a conventional chiller. The ice banks are supplied by two 150 HP screw compressors and evaporative condensers. Extensive data were collected and results showed the system met design objectives most of the time. Tabulation of one years' data showed the system consumed 22% more energy at a 5% reduction in cost over a conventional system. During the test period, control equipment malfunctions had allowed compressors to run during peak demand hours. A correction of this problem would yield an additional 5.3% in dollar savings. The higher energy usage was predominantly attributed to low compressor operating suction temperatures of about 0°F. Design and/or control changes would allow raising this to about 13°F and reducing the additional energy requirement of the ice storage system from 22% to about 5% (Southern California Edison Company, 1982).

EXPERIMENTAL VERIFICATION

Approach

A single wide trailer at the Energy Research and Education Park was chosen as the site for the construction and testing of the ice storage system (see Figure 3, and photo, Figure 4). This trailer was well suited for use due to the following reasons:

1. It was equipped with an air handler and chilled water coil from previous research. This was a necessity because of the financial constraints of the project.

2. The trailer was cooled by a 3 ton package system and data were available on its power consumption.

3. Weather data were available at the site.

4. The trailer had been used for various HUD research projects and excellent air conditioning load data were available.

5. Ample space existed for pouring a slab and constructing a roof for the ice maker and storage box.

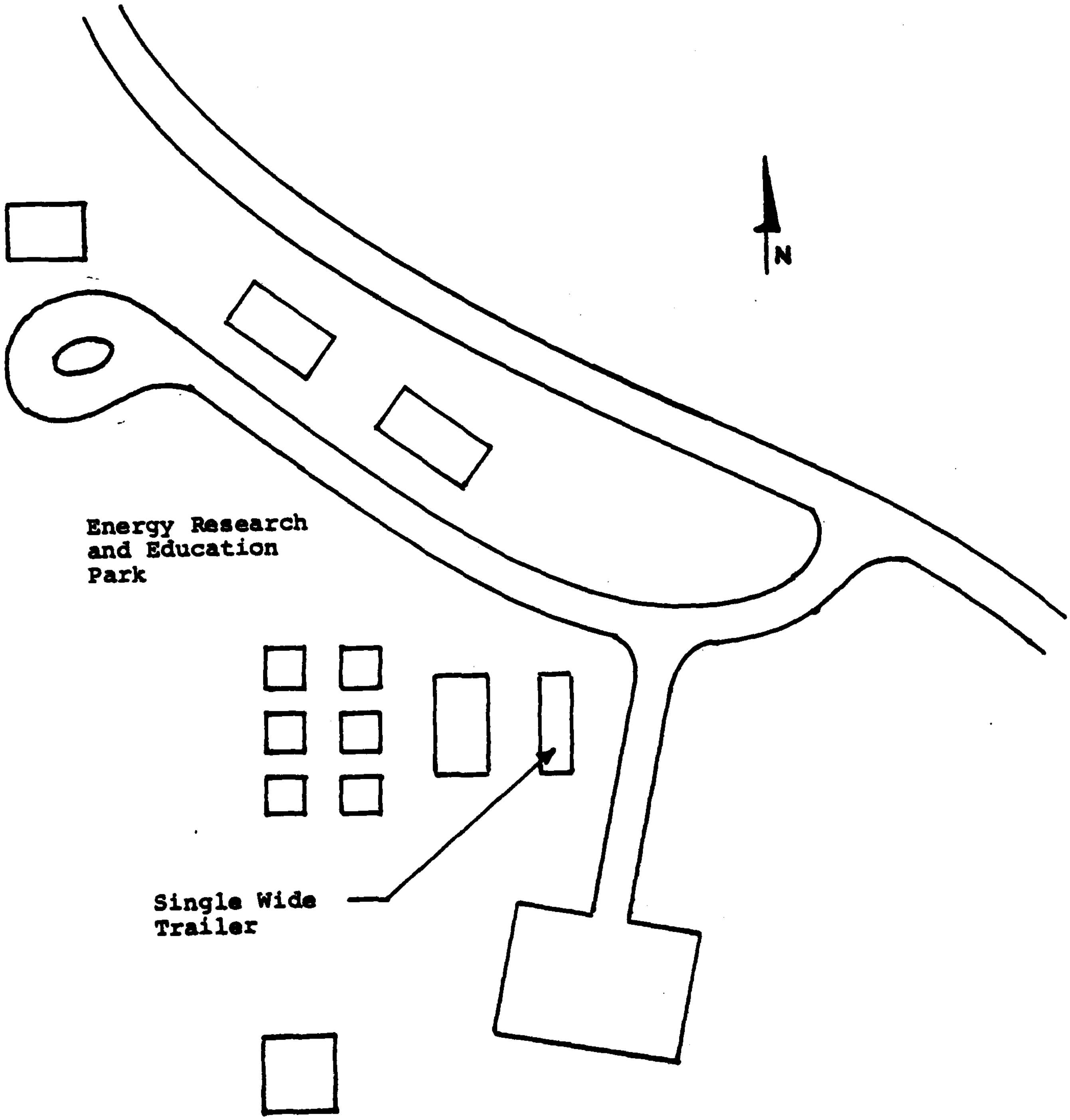


Figure 3. Trailer Location

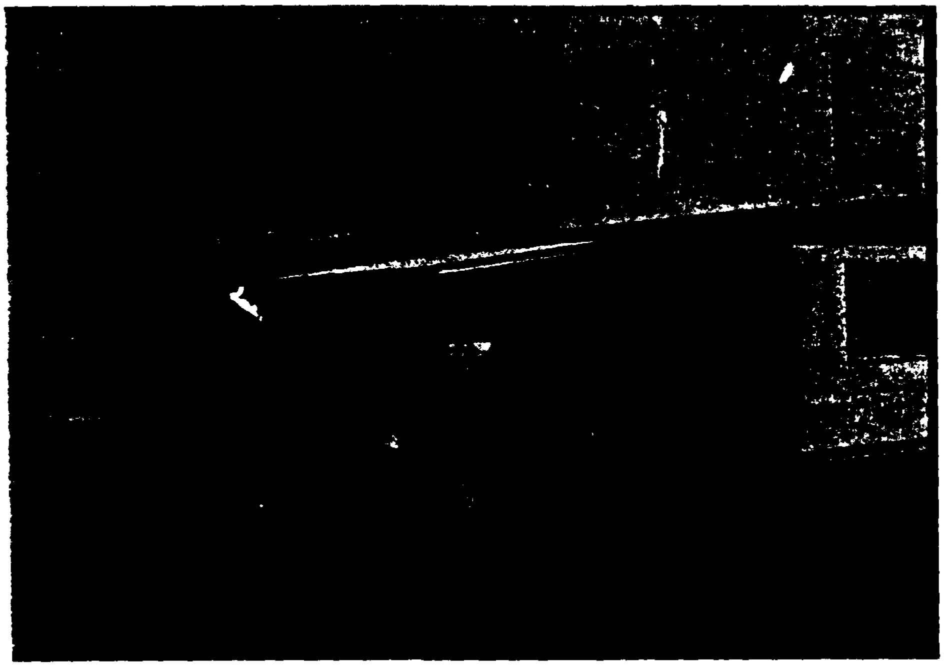


Figure 4. Single Wide Trailer

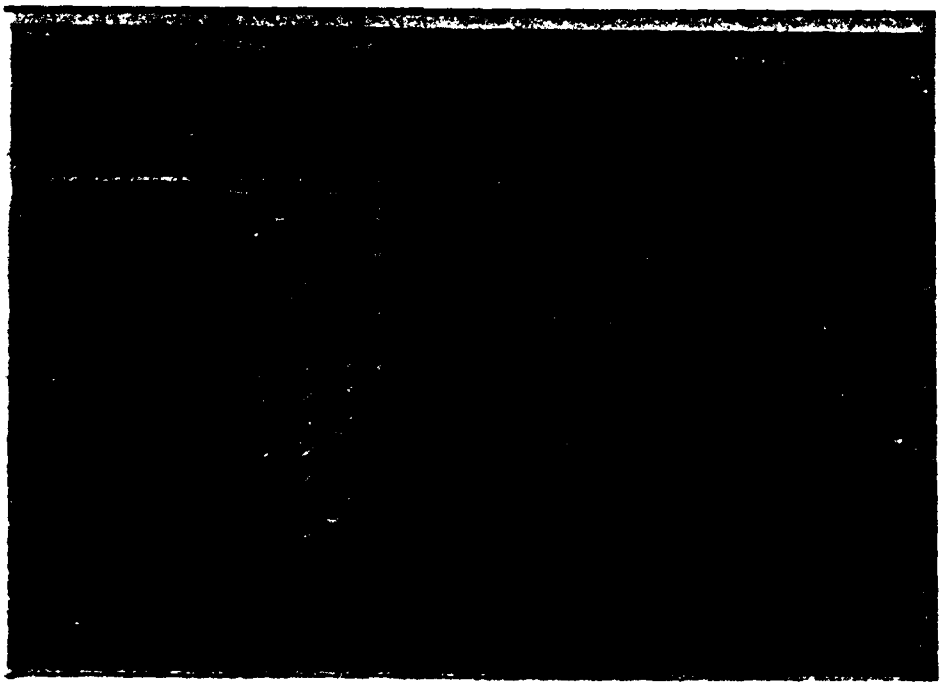


Figure 5. Air Handler with Water Coil

The trailer was 784 square feet in area (11 x 56 ft). The U-values were given as follows:

Wall	0.099 Btuh/ft^2•F
Ceiling	0.067 Btuh/ft^2•F
Floor	0.075 Btuh/ft^2•F

The air handling ducts were located below the floor and were insulated (U-value = 0.147 Btuh/ft^2•F). The existing air conditioner was a 34,000 Btuh Whirpool packaged unit.

A Singer air handler Model #B-DHU 341000, with a 3/4 horsepower blower, had been retrofitted with a 3 1/2 ton chilled water coil (see photo, Figure 5). The coil has 6 rows, 12 circuits, and is 18 x 15 inches. Design specifications were given as follows:

$T_{inlet\ air} = 80°F$

$T_{outlet\ air} = 50\text{-}55°F$

$T_{inlet\ H_2O} = 40°F$

Design and Construction

The first objectives were ice maker selection and ice storage box sizing. Due to budgeting constraints, an ice maker suitable for the design needed to be located that could be donated for the period of our research. While this search was on-going, site preparations continued. The ice maker and storage box would require a concrete slab for support and a roof for shelter from the sun and weather. The location was selected due to its proximity to the air handler, existing plumbing, and electrical power (see Figure 6). A concrete slab, 8 ft x 12 ft x 5 in, was poured and an aluminum roof was constructed.

The original design was for a house with a 38,000 Btuh load and 8 FLEOH per day. This application would require an ice maker capable of producing 4200 lb/day which equates to 2100 lb/12 hr if a 12 hr off-peak rate structure is considered. The air conditioning load on the trailer was difficult to determine. The following data were available:

Method of Calculation	Load Btuh	lbs ice/12 hr req'd @ 8 FLEOH
1. NFPA 501 BM	25,043	1391
2. ACCA Manual J using manufacturer's data	27,571	1537
3. ACCA Manual J using test data	44,441	2479

These data provided varied results. An estimate was made based on KWH consumption of the air conditioner to determine compressor run time which would yield approximate Btu's/day. This estimate, conducted for one week in June, yielded a maximum of 2416 lb ice required.

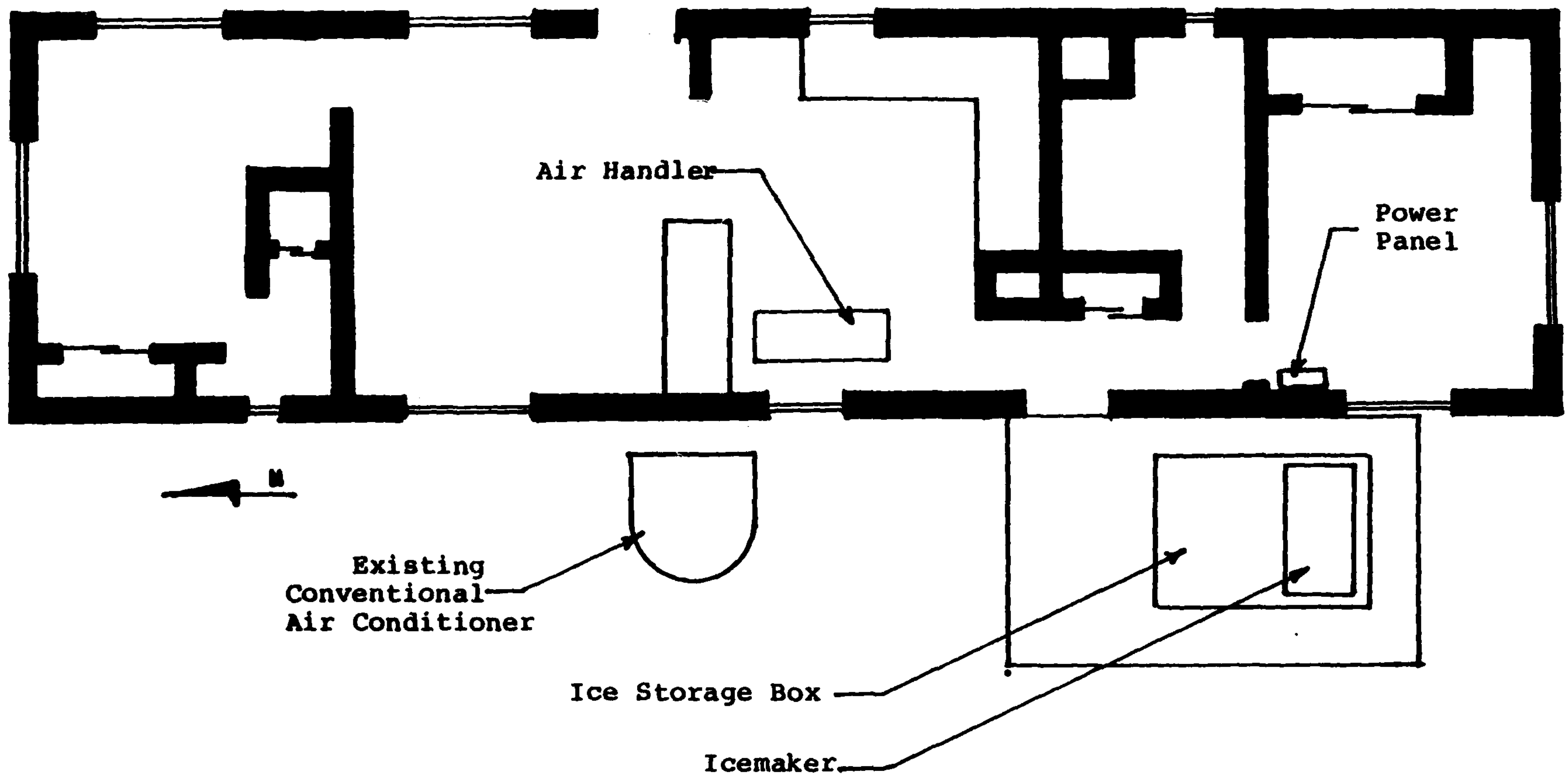


Figure 6. Trailer Floor Plan

These data indicated that a 5000 lb/24 hr ice machine would be required. An intensive search was conducted for a manufacturer that would participate in the research. McQuay Perfex, Inc., Crystal Tips Ice Equipment, agreed to loan a unit that can produce 3300 lb ice/24 hr for a period of 6 months. Although this machine was smaller than the design indicated, the experimental procedure could be modified to provide realistic results. In addition, the unit was an excellent physical size for the requirement of mounting on the storage box and the power requirements were compatible with existing capacity.

The Crystal Tips Model FA-229 Flake Ice Maker (see photo, Figure 7) is an air cooled unit with a 3 Hp compressor and two evaporator/ice making chambers. For specifications and operation principles see Appendix E. The only change from standard operation of this unit was to recirculate water from an ice storage box instead of using city water. This enables one to increase ice making production by using colder water. In addition, this provides a closed system and maintains a constant water level.

The ice storage box was sized to accomodate the ice maker on top and provide storage for 1 day's cooling, 2400 pounds of ice (see photo, Figure 8). The volume of 2400 lb ice (ice density = 56 lb/ft^3) is 43 ft^3. This figure was doubled to compensate for voids and the piling of ice. Taking into consideration the dimensions of the ice maker and

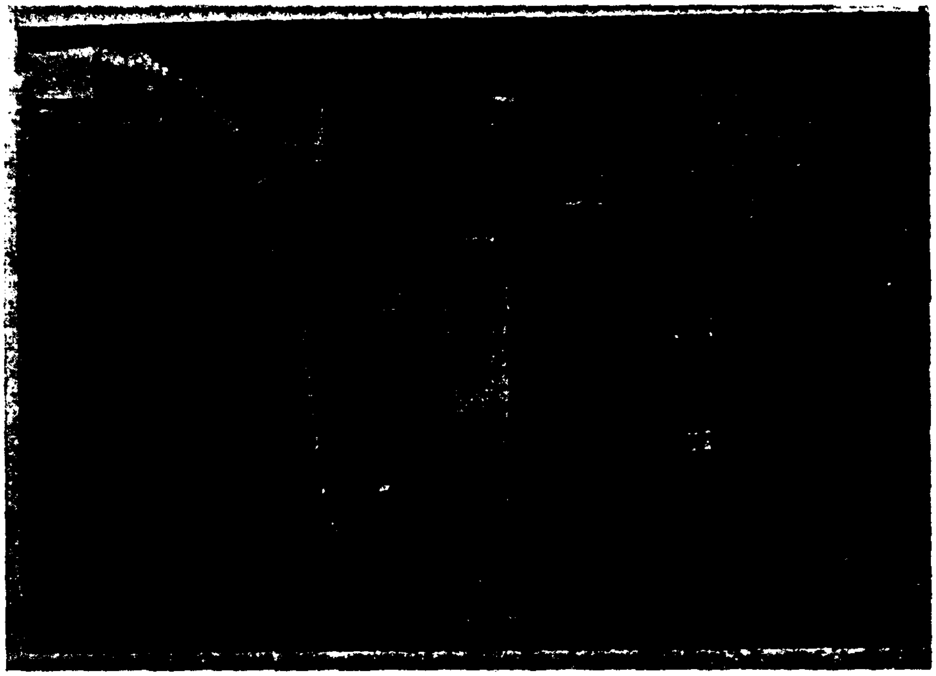


Figure 7. Ice Maker

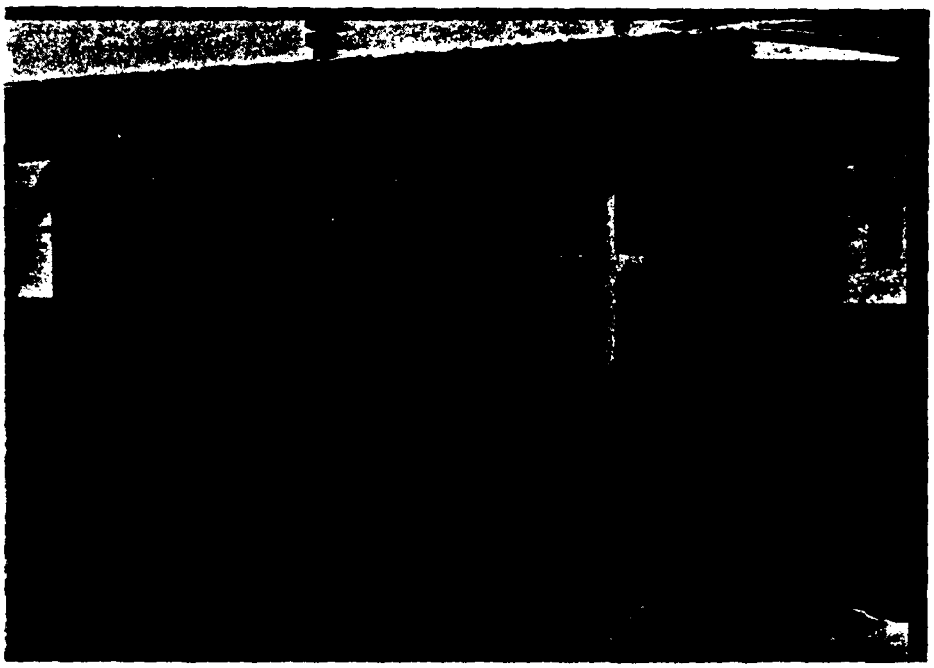


Figure 8. Ice Storage Box

standard building material sizes, we made the storage box 4 ft x 5.5 ft 4 ft high (interior dimensions). The box was made with 2 in x 6 in framing, 16 inches on center, to provide structural support for stored ice and ice maker and to allow spacing required for insulation. Plywood was used for interior and exterior walls. Several layers of fiberglass lay-up resin were applied to interior wall for waterproof lining. Seams were caulked inside and outside. Exterior was painted with weather resistant paint. Overall R-value of 30.1 was obtained using one 5 1/2" (R-19) batt and two sheets of 3/4" (R-12) Thermax, polyisocyanurate (see Figures 9 and 10). Penetrations were provided for the following:

1. 1 in inlet for return water.
2. 1 in outlet for supply water.
3. Two 1/2 in holes for sight glass (to determine water level).
4. 1/2 in outlet for ice maker supply water.
5. Two 2 in x 4 in chutes in top for ice delivery.

A sprayer was attached to the return water piping to distribute water over the ice. One half of top was constructed to be removable. A tight seal was provided using 1/8 in foam weatherstripping and turnbuckles.

Chilled water was circulated through 42 feet of 1 in nominal PVC piping. The piping was insulated with 1 3/4 in "Solar- 7" (R-12). This chilled water circulating pump selected was a Little Giant, Model 3E12NR rated at 11.2 GPM at 1 foot.

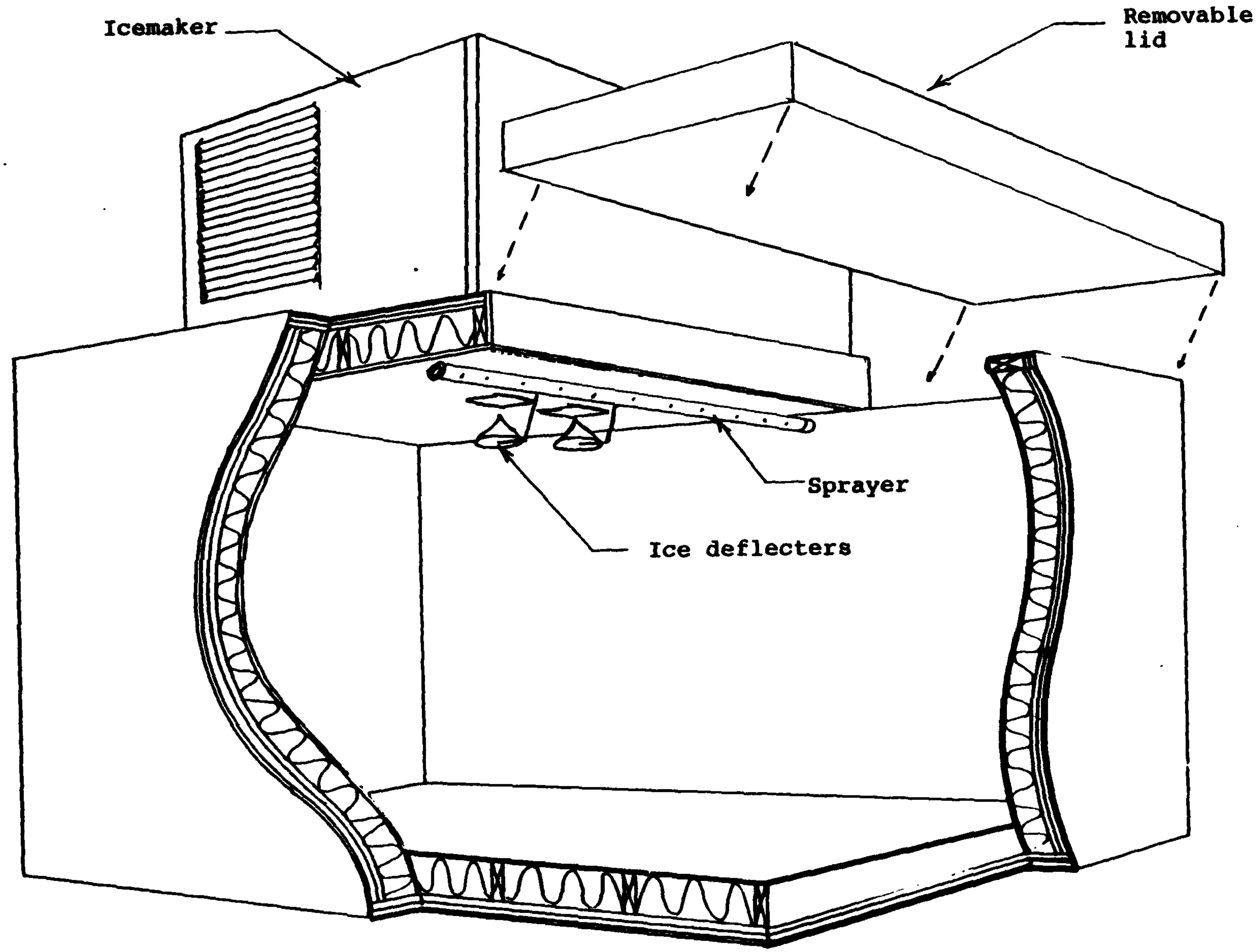


Figure 9. Ice Storage Box Cutaway

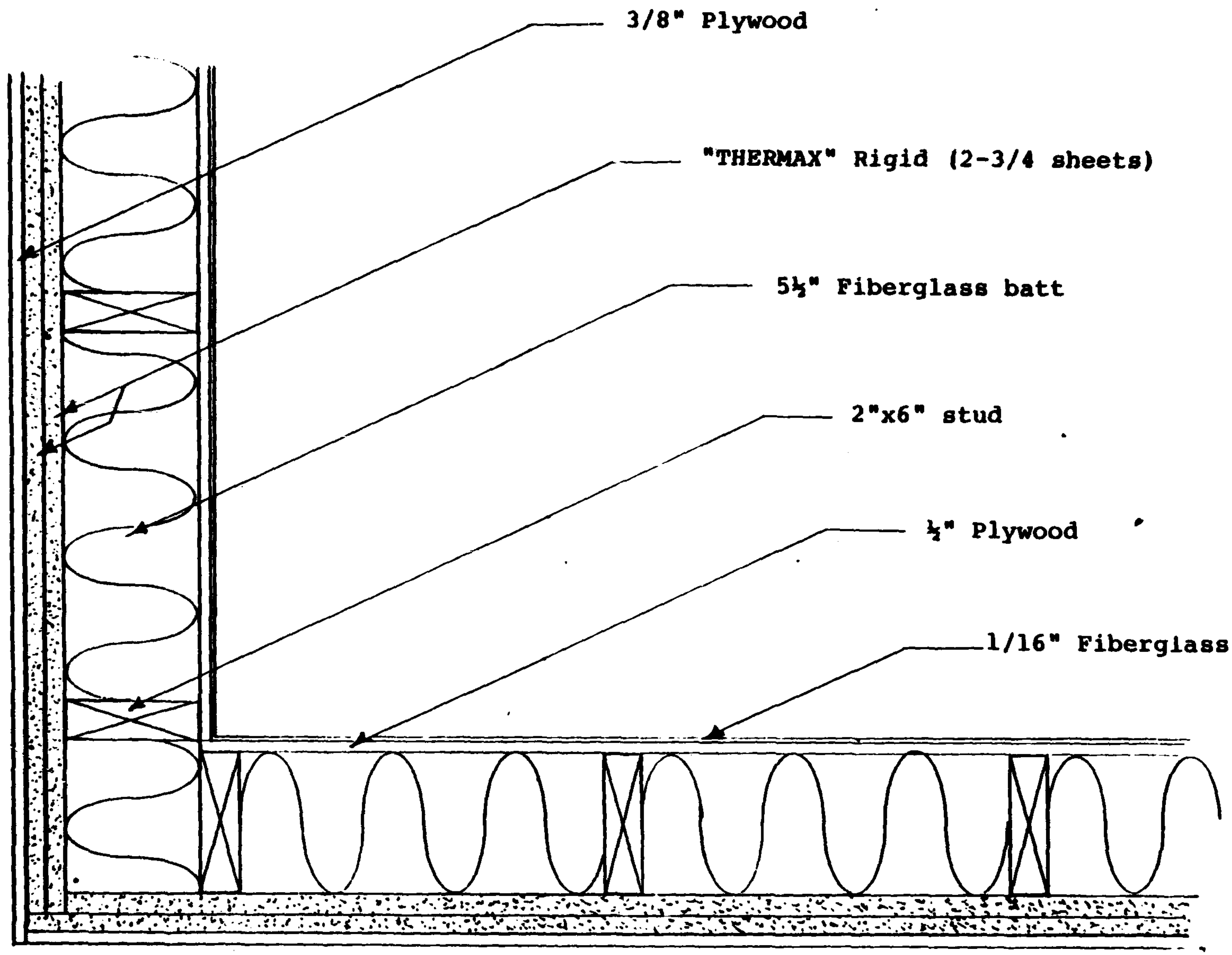


Figure 10. Ice Storage Box Section

of head. The ice machine supply pump selected was a Beckett, Model 6-150 rated at 2.5 GPM at 1 foot of head. This pump was submerged inside the box.

The basic advantage of using ice storage is to produce ice during inexpensive off-peak hours. This requires that the ice maker run independently of the air handler. A time clock would normally control this function and allow flexibility to adjust to different off-peak hours. Ice maker control was accomplished manually since data were taken at the same time the unit was to be turned off or on. The ice maker came equipped with shut-off switches on each ice chute that shut down the unit when chutes fill with ice.

The ice maker supply water pump was wired directly into the ice maker control relay so that it runs whenever the ice maker is on. The air handler was controlled by a room thermostat. Additional contacts were installed so that the circulating pump came on when the air handler was on. A complete system sketch is shown in Figure 11.

Instrumentation

In order to evaluate the operation of the system, the following needed to be determined:

1. Ice maker power consumption.
2. Ice production.
3. Air handler power consumption.
4. Cooling provided to trailer.
5. Losses.

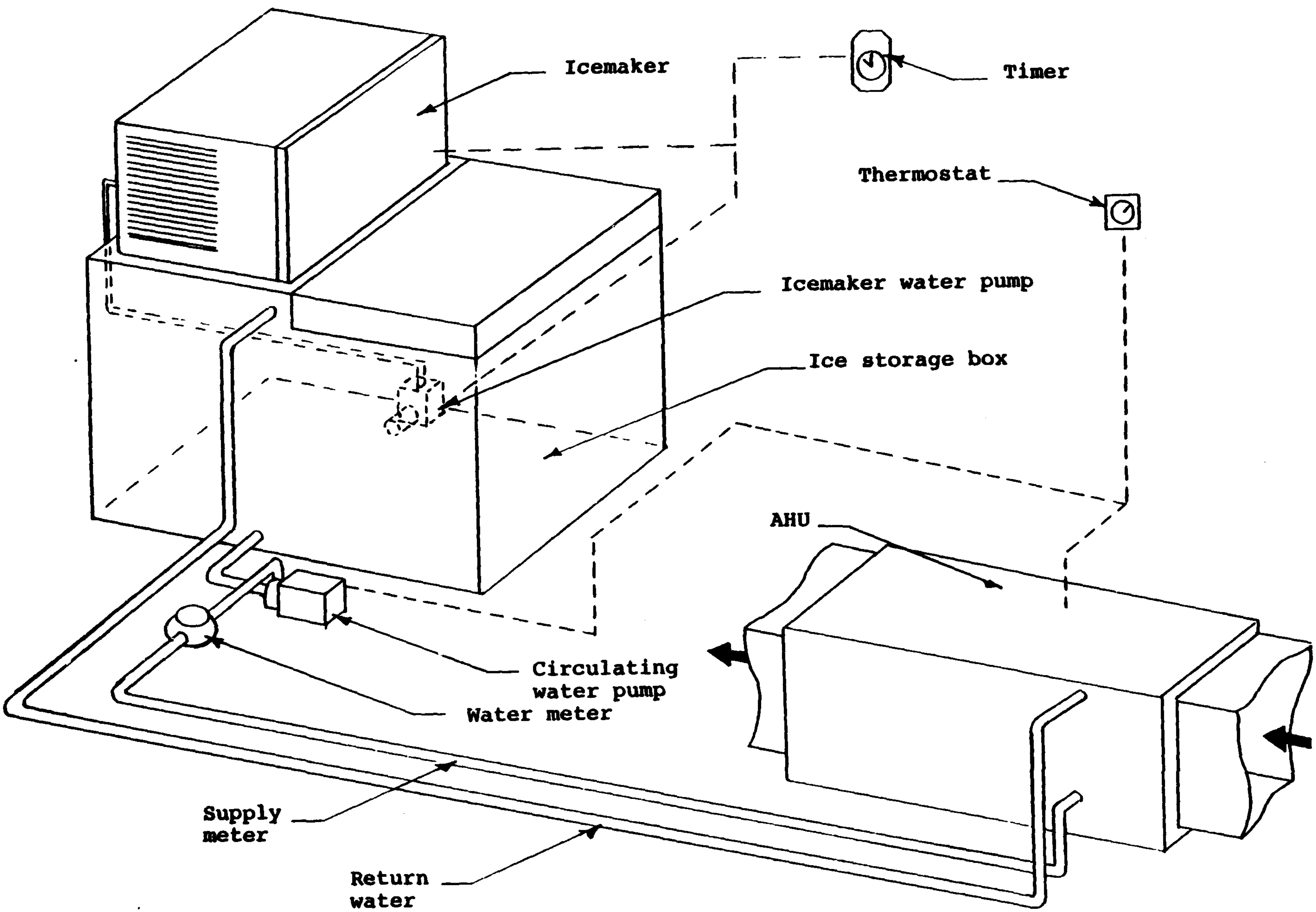


Figure 11. System Schematic

Thermocouples, a flow meter, watt hour meters, and running time clocks were installed to provide data to determine the above. A list of these instruments appears in Table 1.

Manufacturer provided data for ice production at supply water temperatures varying between 55°F and 90°F while ambient varies between 60°F and 100°F (see Appendix E). In order to establish ice production rates in the range of the water temperatures (32°F - 55°F), ice was collected at various conditions. Figure 12 shows these results.

Experimental Prodedure

A survey of 108 utility companies (see Appendix B) indicated that 78 (65%) offered some sort of time-of-day rate structure and 55 (51%) offered residential time of day rates. Of these, 20 (36%) offered a 12 hour off peak/12 hour on peak rate schedule. All of these included weekends as "off peak hours" and most included holidays. The experimental procedure was structured to coincide with this most popular rate schedule.

The original design included a timer that could be adjusted to various time-of-day rate schedules. This was not used for the experiment since manual data collection was required at the same time the timer would have turned the ice maker on/off.

Table 1

Instrumentation List

Instrument	Purpose
Thermocouple 1	Temperature of ice storage box outlet
Thermocouple 2	Temperature of ice storage box inlet
Thermocouple 3	Temperature of ice storage box at 1 foot
Thermocouple 4	Temperature of ice storage box at 2 feet
Thermocouple 5	Temperature of ice storage box at 3 feet
Thermocouple 6	Temperature of ice maker supply reservoir
Thermocouple 7	Temperature of room air
Thermocouple 8	Temperature of water entering coil
Thermocouple 9	Temperature of water leaving coil
Thermocouple 10	Temperature of outside air
Thermocouple 11	Temperature of air entering coil
Thermocouple 12	Temperature of air leaving coil
ΔT Thermocouple 1	Temperature difference across the coil
ΔT Thermocouple 2	Temperature difference across the storage box
Clock 1	Running time of ice maker
Clock 2	Running time of air handler
KWH Meter 1	Power consumption of ice maker/pump
KWH Meter 2	Power consumption of air handler/pump
Flowmeter	Water volume into coil
Recorder 1	Record thermocouples 1 through 12
Recorder 2	Record ΔT thermocouples

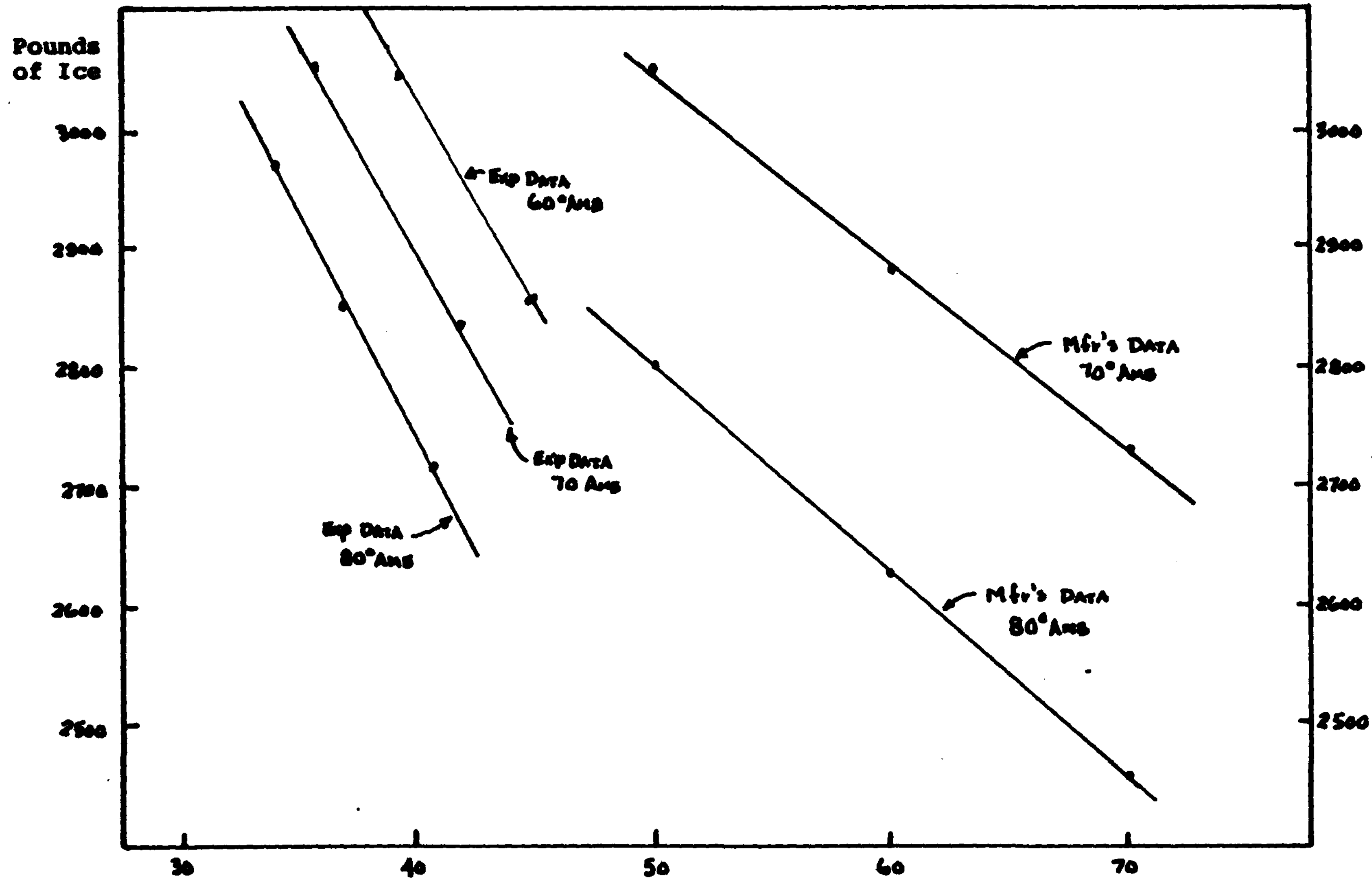


Figure 12. Ice Production Chart

Most of the time-of-day weekday off peak rate schedules were from 9:00 p.m. to 9:00 a.m. A 12 hour off peak time period was selected from 8:00 p.m. to 8:00 a.m. as a convenience to the researchers. It was felt the one hour shift would not affect the results.

One major uncertainty of the system operation was how fast the ice would stack under the diffusers, filling the ice chutes and shutting off the ice maker (See Figure 13). The objective was to find the optimum water level to allow maximum ice production. If the water level was too low, all water would be made into ice. The ice maker reservoirs would run dry shutting down the unit on low pressure. In addition, this would not utilize the full capacity of the storage box. If the water level was too high, the ice would begin to stick on top of the water, filling the chutes, and shutting off the unit prematurely. After observing the system operation at various water levels, it was concluded that a level of 2 feet (2750 lb H_2O) was best. Although this level occasionally caused the ice to back up and shut off the unit, it was compensated by periodically smoothing down of ice peaks in the storage box. If the unit did not run the full 12 hour period because of ice buildup, it was simulated by running during "on peak hours" without penalty.

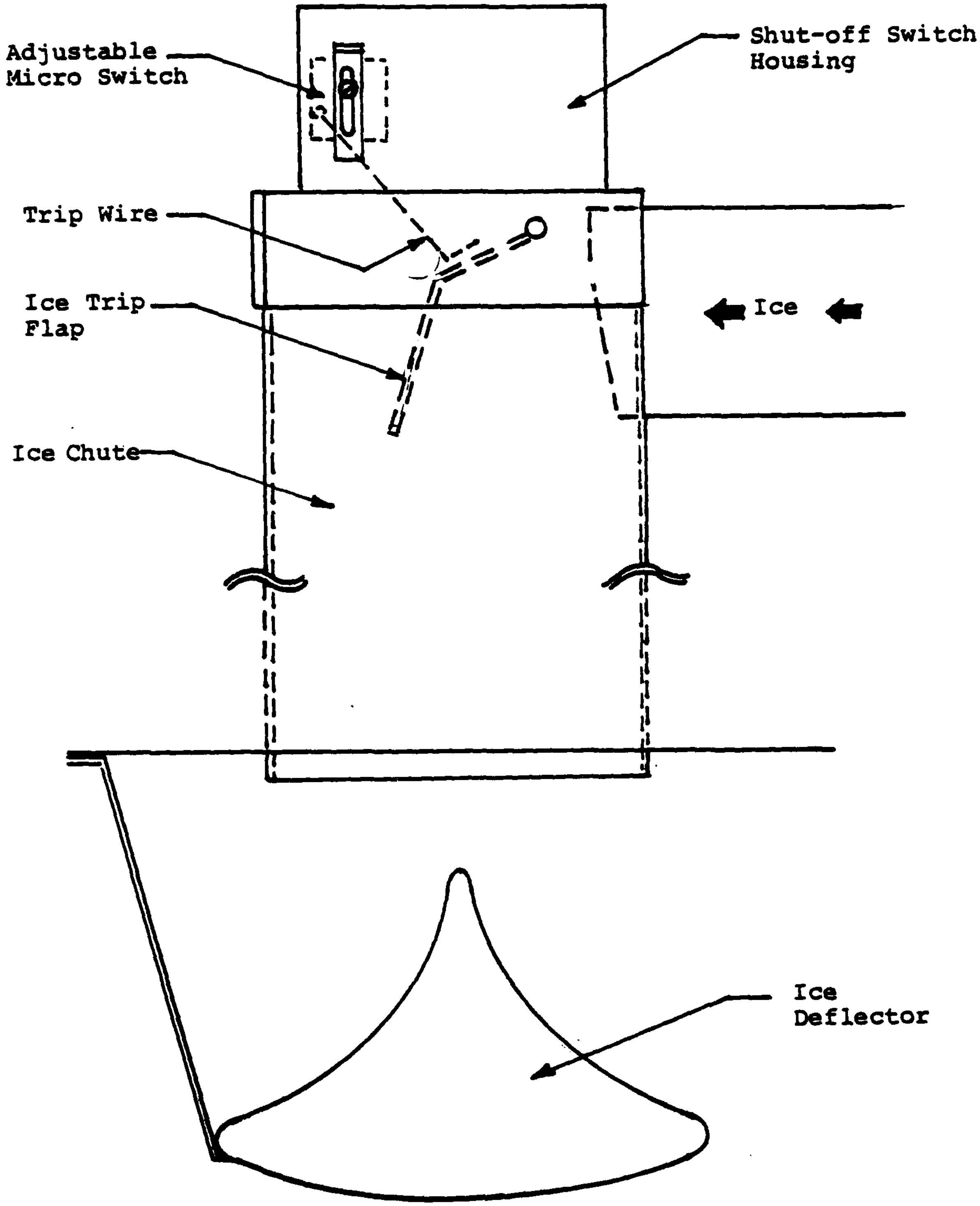


Figure 13. Ice Chute Detail

The twelve temperature points were recorded every 15 minutes continuously. The two delta T points were recorded continuously. The flow meter, running clocks and kilowatt-hour meters were read manually at twelve hour increments.

DATA AND RESULTS

Table 2 shows energy consumption and running time for the ice maker and air handler. In addition, water flow and temperature drops across the storage and cooling coil are shown. These temperature differences are weighted averages based on continuously recorded data. The data were taken during a 12½ day test from 7 October to 19 October, 1982. The initial strategy was to run the ice maker during the 12 hour off peak period on weekdays and all weekend if necessary. Because of ice build-up and control switch problems, many times the ice maker would shut off prematurely and not provide enough ice for the following days cooling load. The unit was run during the day to compensate for this lost ice production time. As a result, it was easier to show the data daily rather than at half day periods.

Figure 14 shows outside air, inside air, and ice storage box (at one foot) temperatures for a 48 hour period beginning on 7 October at 0900. By observing the storage box temperature, it can be seen that there was an adequate supply of ice on the first day (7 Oct.), whereas the next day the ice supply ran out in late afternoon causing the box temperature to rise dramatically before the icemaker was turned back on.

Figure 15 shows air handler cooling coil entering and leaving temperatures for both water and air. Also shown is air handler running time. During the first day, when an adequate ice supply was available, the air temperature difference was about 10°F, while the water was about 6°F. The following day, when the ice ran out, the air temperature difference dropped to less than 6°F as a result of a water temperature difference of only 3°F.

Figures 16 to 18 show daily electric consumption, estimated run time and equivalent ice requirement for the conventional air conditioner supplying the trailer. The data are shown for a 3 month period beginning on 21 June 1982. Kilowatt hour consumption for the air conditioner was recorded daily at the same time. Condensing unit run time was estimated using the energy efficiency ratio and the rated unit capacity.

$$\text{Run Time } \left(\frac{\text{hrs}}{\text{day}}\right) = \frac{6.7 \dfrac{\text{Btu}}{\text{watt hr}} \times N \dfrac{\text{kilowatt hr}}{\text{day}} \times \dfrac{1000 \text{ watt}}{\text{kilowatt}}}{34{,}000 \text{ Btu/hr}}$$

$$= 0.197 \left(N \frac{\text{kilowatt hr}}{\text{day}}\right)$$

The equivalent ice requirement was estimated using the above data and 144 Btu/pound of ice. This does not account for any losses.

$$\text{Ice Required (lbs)} = \frac{\text{Run Time (hrs)} \times 34{,}000 \text{ Btu/hr}}{144 \text{ Btu/lb}}$$

Table 2

System Performance Data

Time Period	Hrs	Ice kWh	Ice Hrs	Ice kW	AHU kWh	AHU Hrs	AHU kW	H2O Gals.	H_2O GPH	$\overline{\Delta T}^*$, °F Storage	$\overline{\Delta T}^*$, °F Coil
0900 7 Oct - 2000 7 Oct	11	0	0	0	7.6	9.2	.83	2465	268	6.5	5.6
2000 7 Oct - 2000 8 Oct	24	42.5	12.1	3.51	7.5	9.9	.76	2592	262	6.2	5.4
2000 8 Oct - 1900 8 Oct	23	71.0	20.0	3.55	10.5	12.3	.85	3205	261	6.5	5.6
1900 9 Oct - 1900 10 Oct	24	7.30	19.3	3.78	10.0	12.5	.80	3260	261	6.8	5.9
1900 10 Oct- 2000 11 Oct	25	42.2	11.8	3.57	13.0	17.5	.74	4400	251	4.7	4.0
2000 11 Oct- 1600 12 Oct	20	70.8	20.0	3.54	9.0	11.7	.77	2985	255	6.5	5.6
1600 12 Oct- 2200 13 Oct	30	34.0	9.6	3.54	13.1	16.8	.78	4378	261	5.4	4.8
2200 13 Oct- 2200 14 Oct	24	65.5	18.8	3.48	11.2	14.3	.78	3712	260	4.8	4.2
2200 14 Oct- 1200 16 Oct	38	67.7	20.0	3.39	3.7	4.4	.84	1145	260	6.6	5.9

Table 2, Continued

Time Period	Ice Hrs	Ice kWh	Ice Hrs	AHU kW	AHU kWh	AHU Hrs	AHU kW	H_2O Gals.	H_2O GPH	$\overline{\Delta T}^*$, °F Storage	$\overline{\Delta T}^*$, °F Coil
1200 12 Oct– 1900 17 Oct	31	39.4	11.5	3.43	4.9	6.0	.82	1533	256	6.2	5.4
1900 17 Oct– 2200 18 Oct	27	10.4	4.0	2.60	.8	1.0	.80	252	252	5.7	5.0
2200 18 Oct– 2100 19 Oct	23	24.1	7.7	3.13	6.3	8.0	.79	2040	255	6.2	5.4
TOTAL	300	541.1	154.8	3.50	98.6	123.6	.80	32395	262	5.9	5.1

*Weighted average.

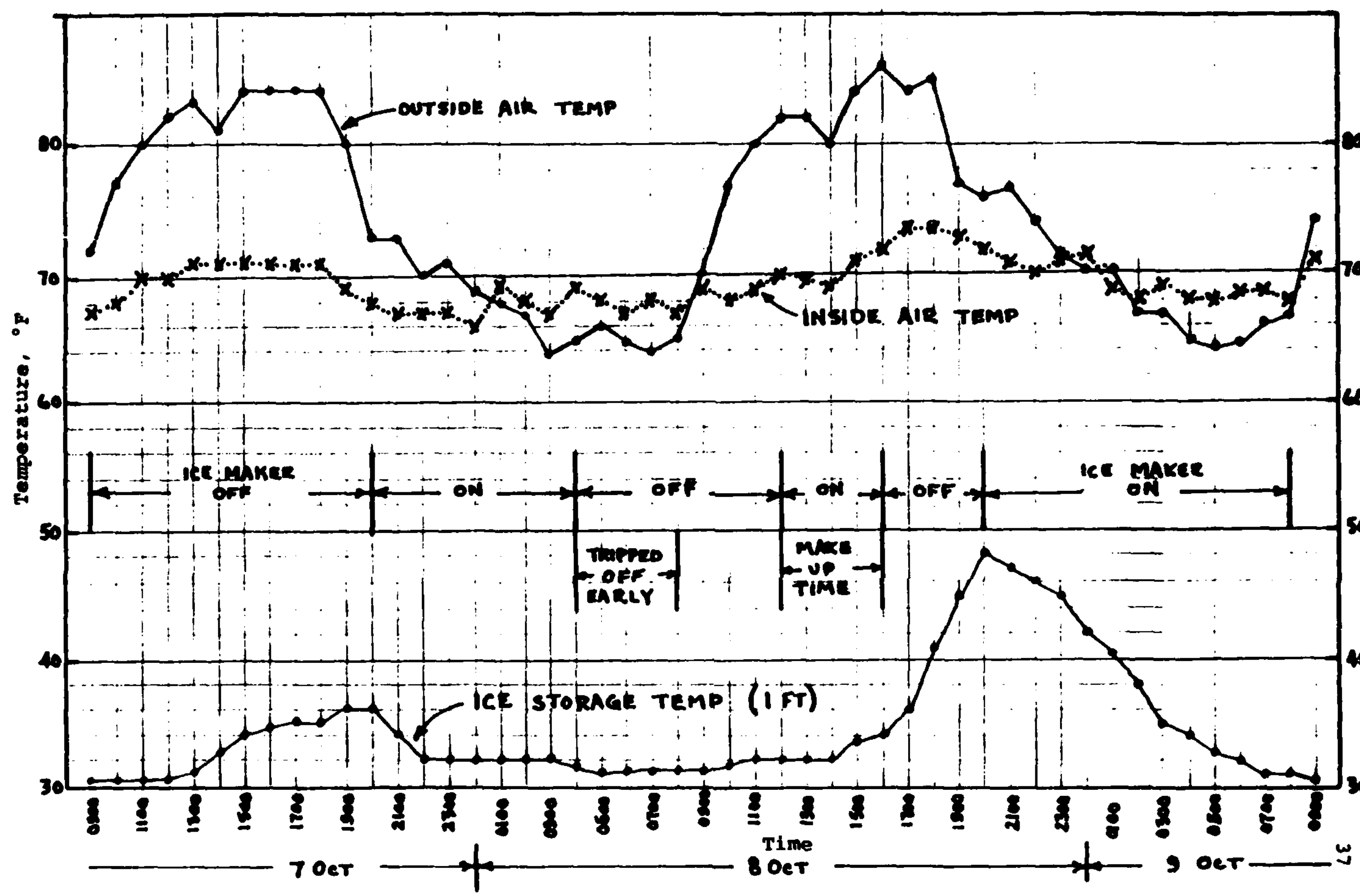


Figure 14. Temperature Profiles

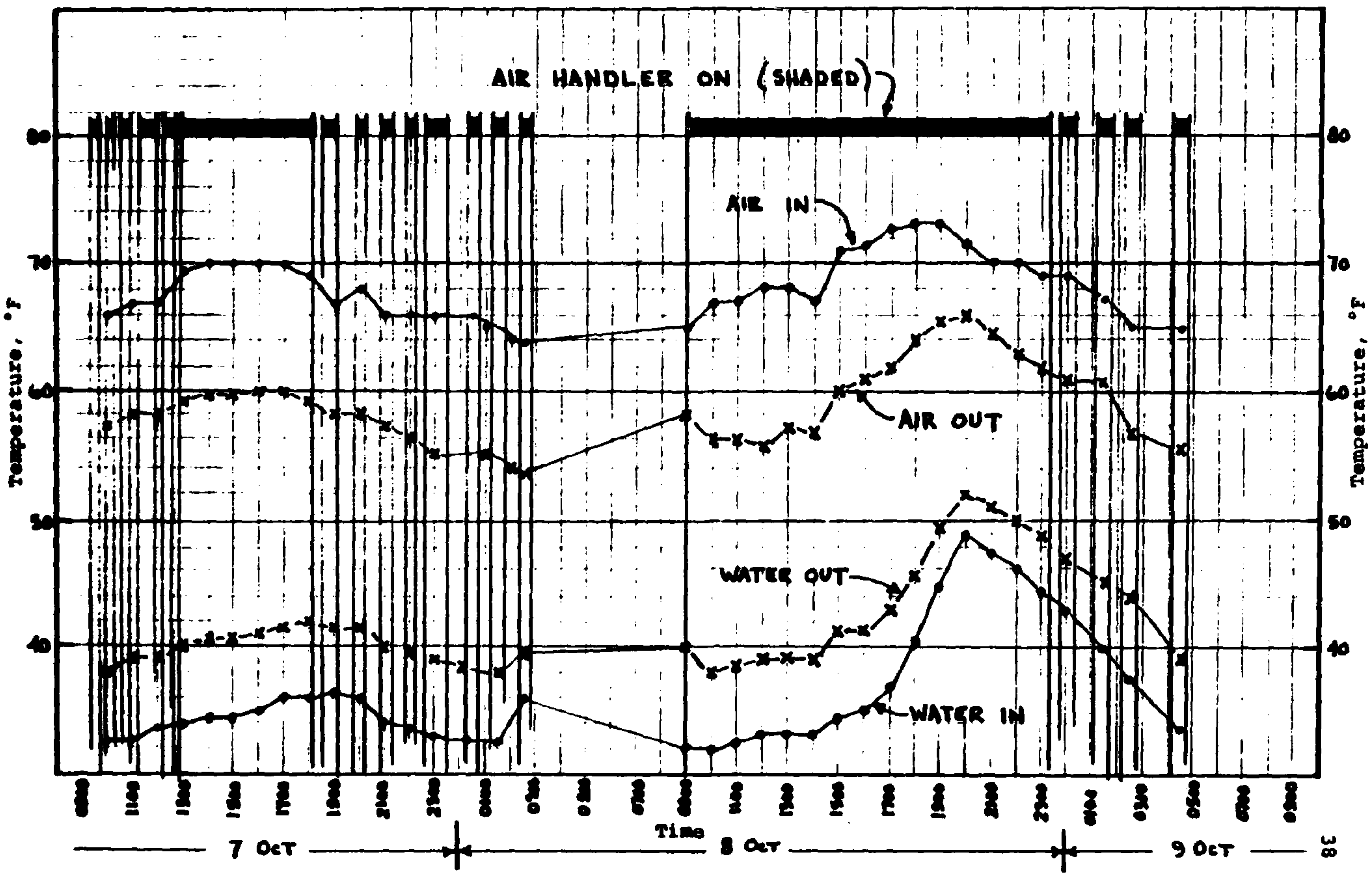


Figure 15. Coil Temperature Profiles

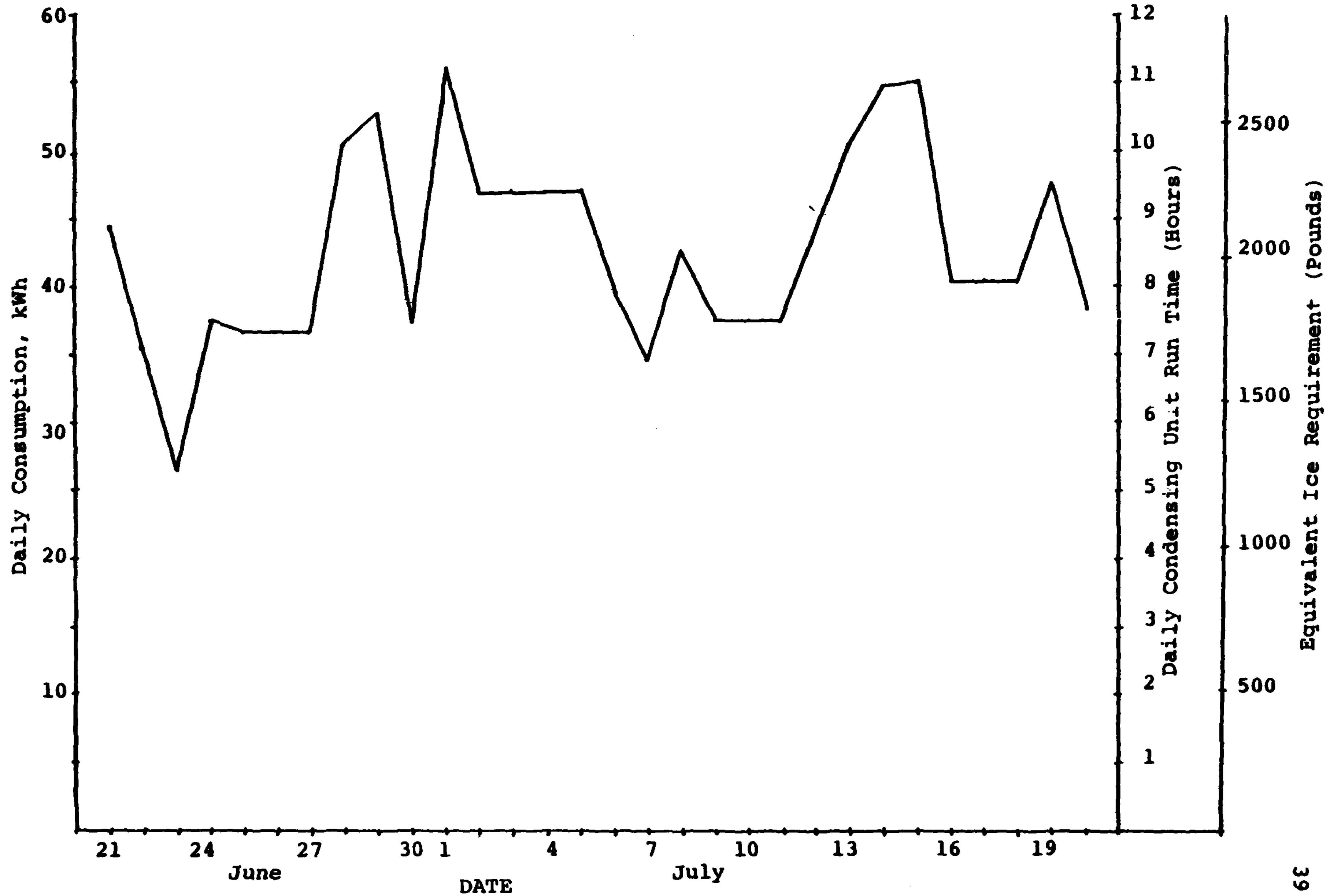


Figure 16. Conventional System Operation

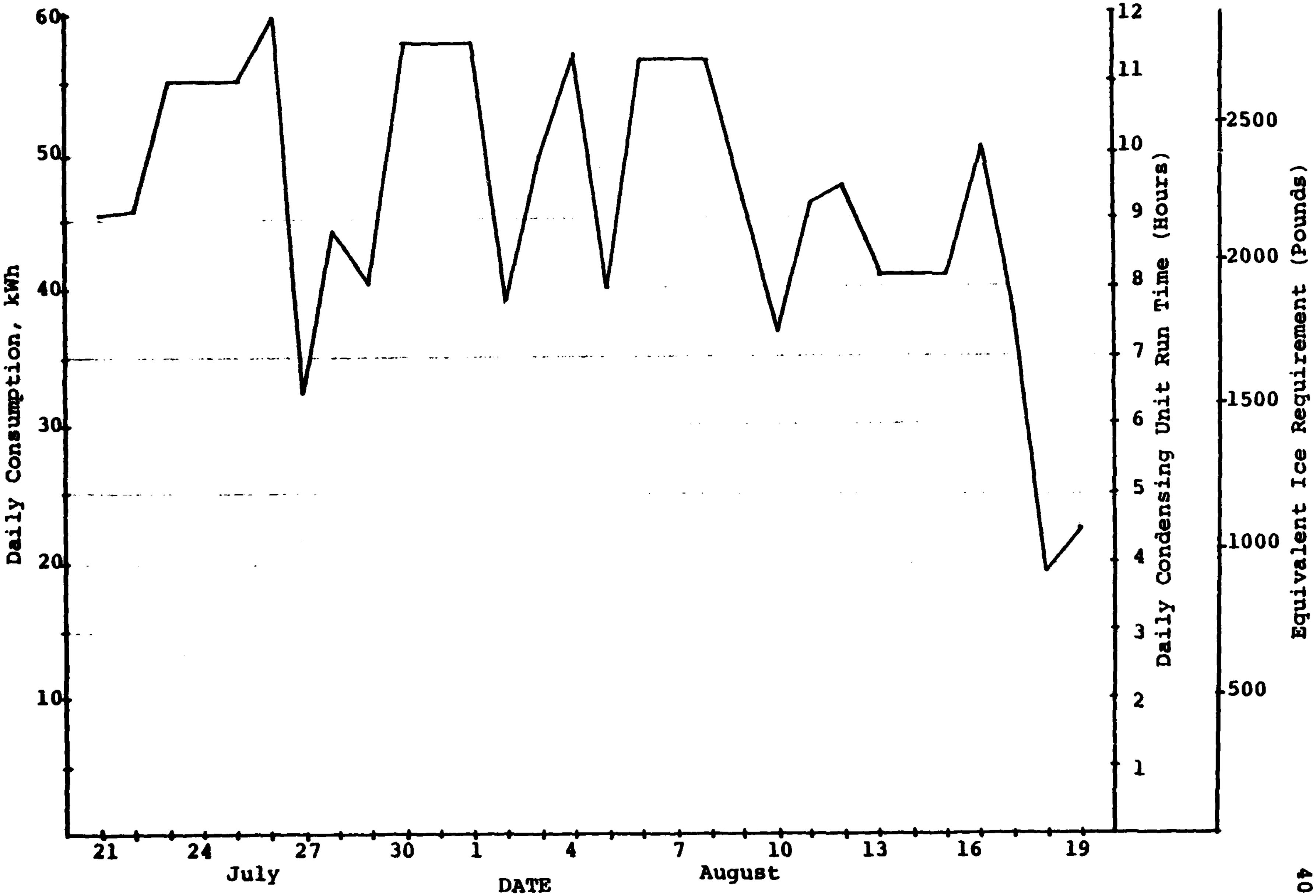


Figure 17. Conventional System Operation (Cont.)

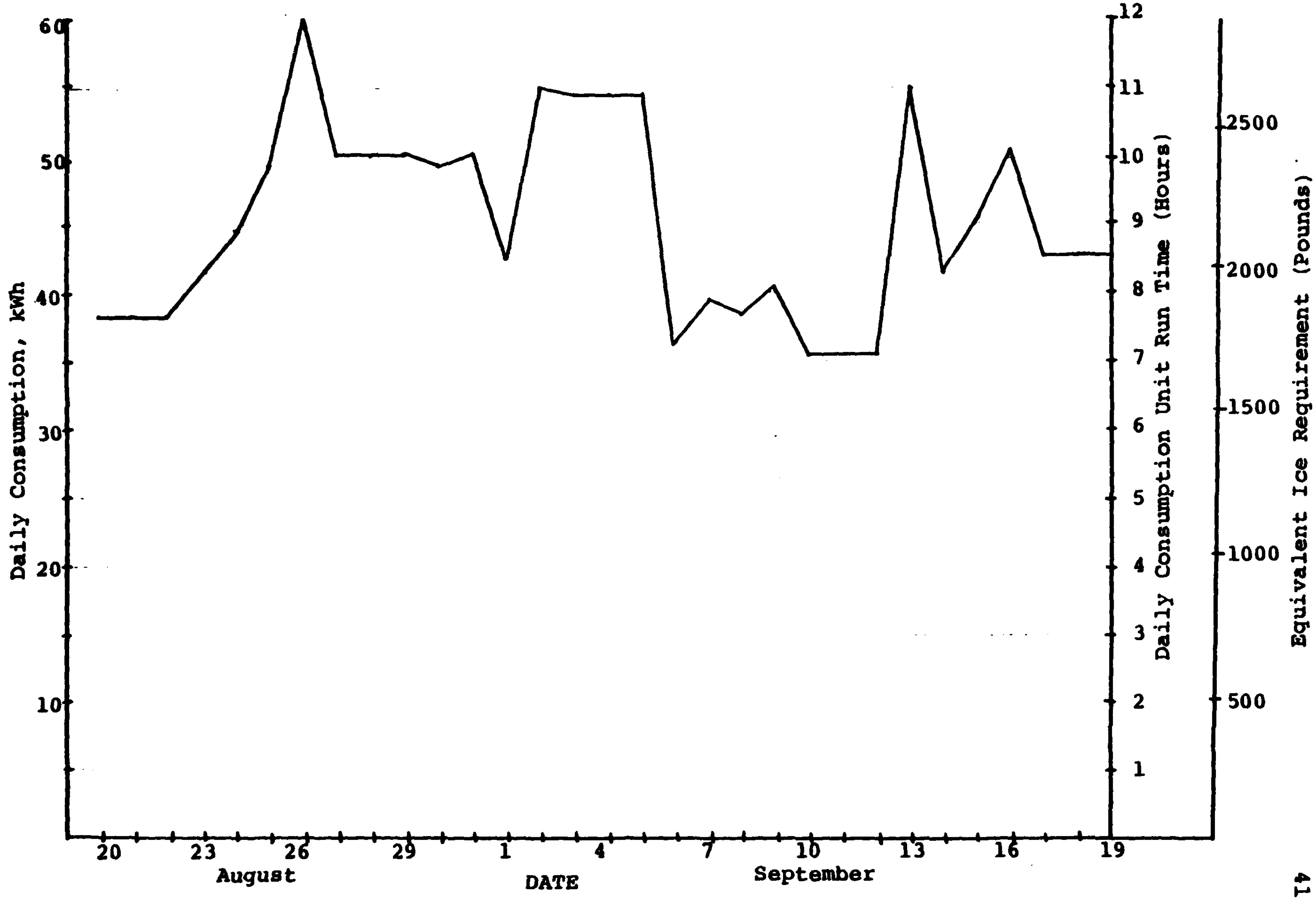


Figure 18. Conventional System Operation (Cont.)

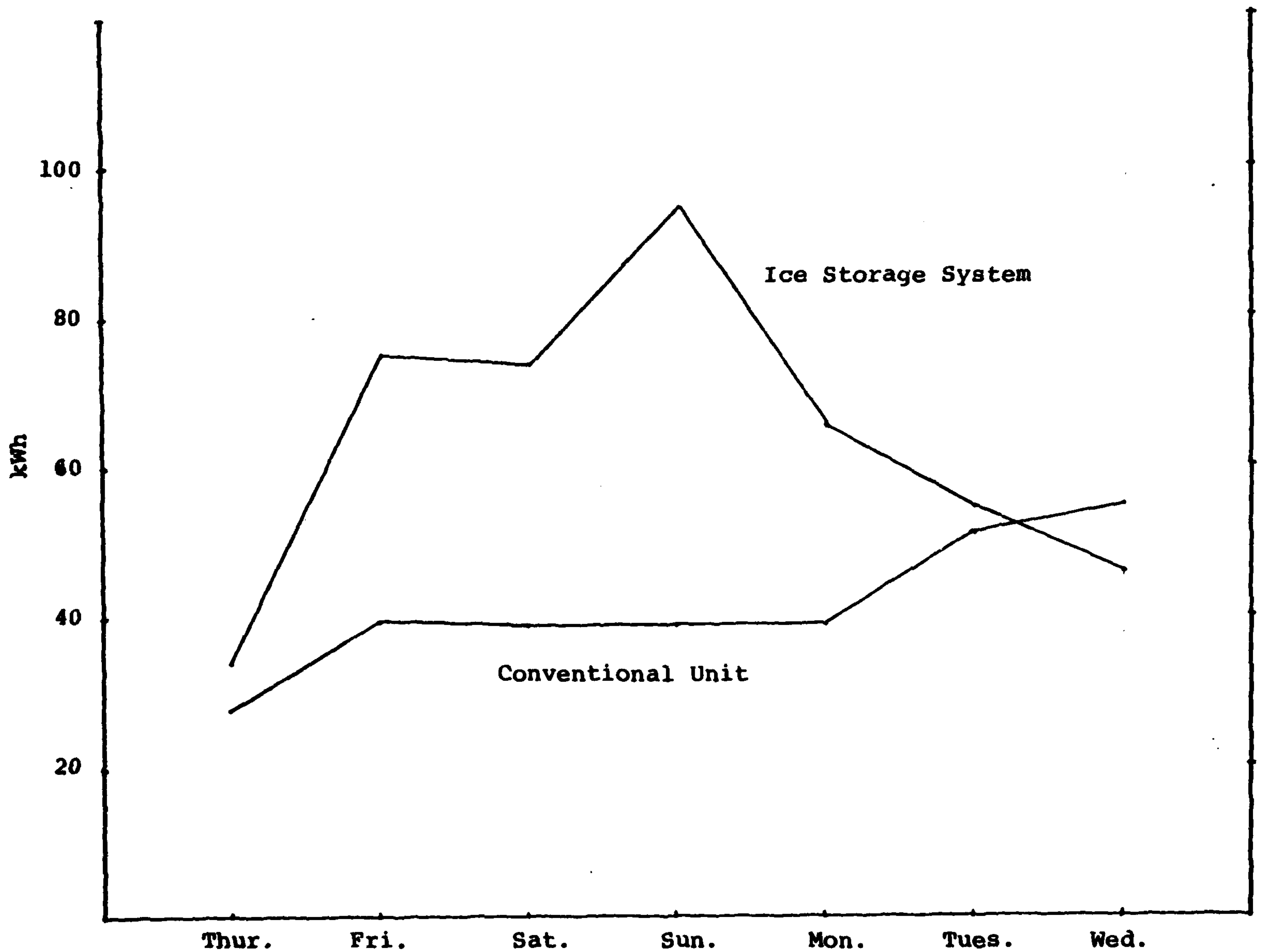


Figure 19. Kilowatt Hour Consumption

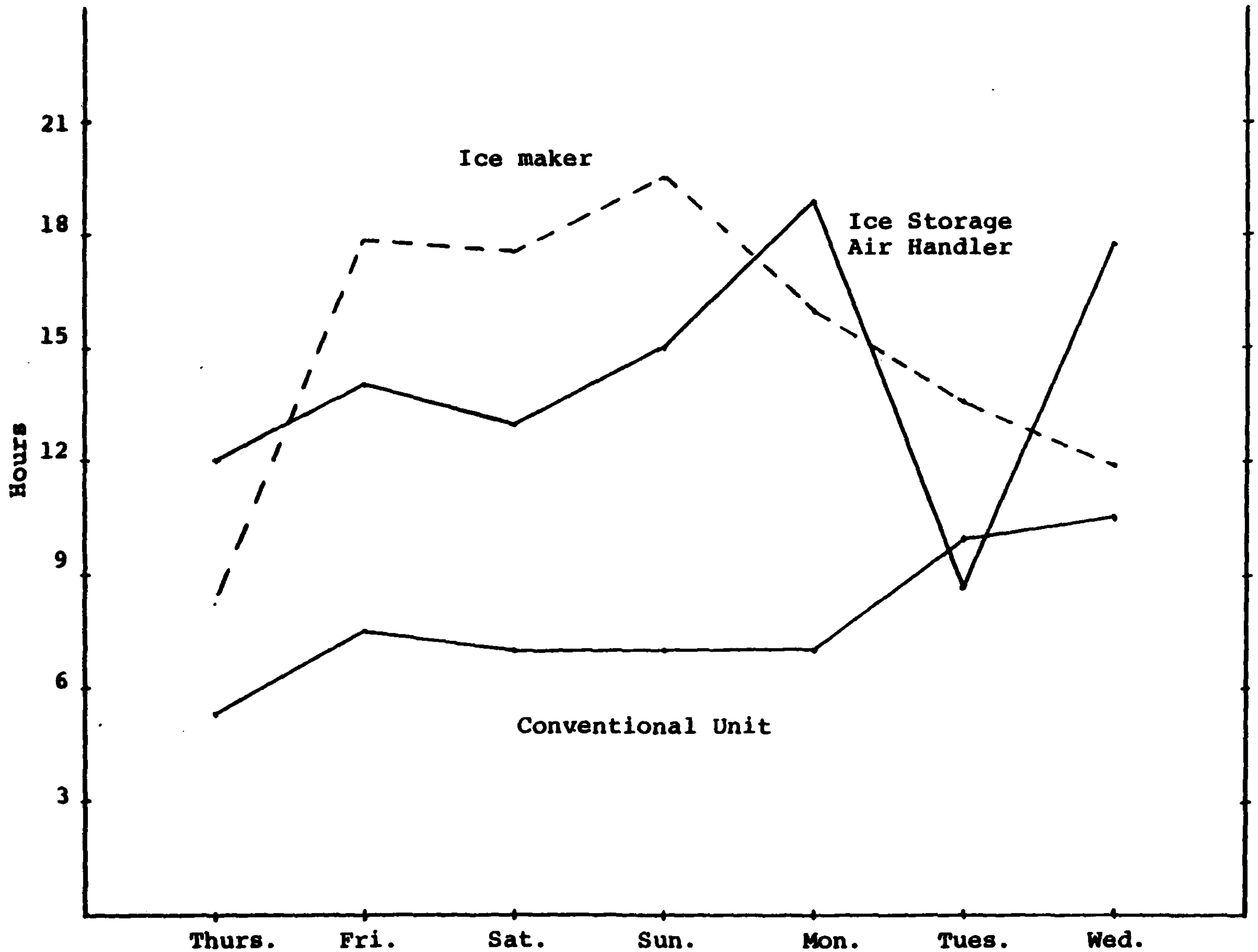


Figure 20. Equipment Run Time (hours)

CALCULATIONS AND DISCUSSION

Storage Box Losses

The ice storage box R-value was computed as follows:

Material	Thickness (in)	Thru Insulation ($ft^2 °F/Btuh$)	Thru Stud ($ft^2 °F/Btuh$)
Inside Surface	-	.3	.3
Fiberglass	1/16	.25	.25
Plywood	1/2	.62	.62
Batt Insulation	5 1/2	19.0	-
Wood Stud	5 1/2	-	7.14
"Tuff R" Insulation	1 1/2	10.8	10.8
Plywood	3/8	.47	.47
Paint	-	-	-
Outside Surface	-	.17	.17
	7 15/16	31.31	19.45

Table 3. Box R-Value

The 2 x 6 inch studs were 16 inches on center, which corresponds to 10% of the cross sectional area. Therefore,

$$R = 0.9(3.13.) + 0.1(19.45) = 30.12 \text{ ft}^2 °F/Btuh$$

$$U = 1/R = 1/30.12 = 0.0332 \text{ Btu/hrft}^2 °F$$

The inside area of the box was 120 square feet. Based on average conditions of outside temperature 74°F and storage box temperature of 32°F, the following heat loss is estimated.

$$\dot{Q} = UA\Delta T$$

$$= 0.0332\ \text{Btu/hrft}^2\,{}^\circ\text{F}\ (120\text{ft}^2)(72-32)\,{}^\circ\text{F}$$

$$= 159\ \text{Btu/hr}$$

This is the equivalent of about 1.1 pounds of ice per hour.

The estimate was checked by producing 2100 pounds of ice and leaving it in the storage box without any water circulating. This ice lasted 28 days, which equated to an average of 450 Btu/hr. This was 2.8 times the calculated value. It was assumed the additional losses were caused by box penetrations (mainly the ice chutes), air leakage through lid, radiation gains, and construction imperfections. The total box losses per day were estimated by the following equation:

$$Q_{\substack{\text{Box}\\\text{Loss}}} = 2.8 \times 0.0332\ \text{Btu/hrft}^2\,{}^\circ\text{F} \times 120\ \text{ft}^2 \times (\bar{T}_0 - \bar{T}_B)\,{}^\circ\text{F} \times \tau_T$$

where,

$\bar{T}_0$ = average outside temperature (°F)

$\bar{T}_B$ = average box temperature (°F)

τ_T = total time (hours)

<u>Pipe Losses</u>

The pipe losses were estimated using data collected during test period:

$$Q_{\substack{\text{Pipe}\\\text{Losses}}} = m_c c_p (\overline{\Delta T})_p$$

where,

m_c = mass of water circulated (lbs)

c_p = constant pressure specific heat of water (1 Btu/lb°F)

$(\Delta T)_p$ = average temperature drop through piping (°F)

Supply Water Losses

Ice maker supply water losses were estimated using the following equation:

$$Q_{\substack{\text{Supply}\\ \text{Water}}} = \dot{m}_s c_p (\overline{\Delta T})_s \tau_{im}$$

where,

$\dot{m}_s$ = average flow rate (lb/hr)

c_p = constant pressure specific heat of water (1 Btu/lb°F)

$(\overline{\Delta T})_s$ = average temperature drop of water

τ_{im} = ice maker run time

Cooling Coil

Cooling coil heat gain was estimated using data collected during test period as follows:

$$Q_{\substack{\text{Cooling}\\ \text{Coil}}} = m_c c_p (\overline{\Delta T})_c$$

where,

m_c = mass of water circulated (lbs)

c_p = constant pressure specific heat of water (1 Btu/lb°F)

$(\overline{\Delta T})_c$ = average temperature drop through cooling coil (°F)

Ice Production

The ice use was estimated by adding the total losses to the cooling provided.

$$Q_{\substack{\text{ice}\\\text{used}}} = Q_{\substack{\text{box}\\\text{loss}}} + Q_{\substack{\text{pipe}\\\text{loss}}} + Q_{\substack{\text{supply}\\\text{water}\\\text{loss}}} + Q_{\substack{\text{cooling}\\\text{coil}}}$$

This equation was valid since we started and finished with approximately equal amounts of ice in storage. The following data were calculated from our results fro use in the above equations:

$\overline{T}_0 = 75°F$

$\overline{T}_B = 35.5°F$

$m_c = 270{,}000$ lbs

$(\overline{\Delta T})_p = 0.8°F$

$\dot{m}_s = 315$ lb/hr

$(\Delta T)_s = 1.5°F$

$\tau_T = 300$ hrs

$\tau_{lm} = 154.8$ hrs

$(\Delta T)_c = 5.1°F$

Table 4 shows the results of the substitution of these values into the equations.

Table 4

Heat Balance (in Btu's)

Box Losses	Pipe Losses	Supply Water Losses	Coil Cooling	Ice Used
132,000	216,000	73,000	1,380,000	1,800,000

The calculated ice use of 1,800,000 Btu over the 154.8 ice maker running hours corresponds to an average ice production of about 81 pounds per hour. This value is considerably less than the measured rates and manufacturer's predicted rates (see Figure 12). This discrepancy indicates an error in the heat balance calculations or an inaccuracy in the measurement of ice production. All measurement devices involved in heat balance calculation were double checked for calibration and found to be accurate.

Based on observations, when the storage box becomes full of ice (dropping water level), the water supply water to the ice maker decreased, slowing down ice production. This accounts for most of the discrepancy and the results displayed in Table 4 are accurate.

Economic Comparison with Conventional Unit

One of the project's major objectives was to evaluate the economics of the ice storage unit versus that of a conventional unit. This was accomplished using data collected from 7 October to 14 October, 1982, for the ice storage system and comparing it with energy consumption data for the conventional unit operating between 23 June and 30 June, 1982. The evaluation was qualitatively accurate based on a comparison of temperature and humidity data for the two periods. Also, inside temperature was controlled to the same temperature.

Figure 19 shows kilowatt hour consumption for both systems. As can be seen, the ice storage system consumes considerably more energy than the conventional system. In fact, the ice storage system consumed 451 kilowatt hours, while the conventional system only used 288 kilowatt hours during this time period. This represents a 56% increase in power consumption.

Figure 20 shows the running time of the ice maker, ice storage air handler, and the conventional condensing unit. The ice maker ran for 105 hours, while its air handler ran 98 hours for the week. The condensing unit ran only 55.5 hours for the similar period.

Although the ice storage system used 56% more power, it could still save money if used in conjunction with a time of day electrical rate structure. The objective was to simulate at least one weeks use of the ice storage system. The initial plan was to operate the ice maker during all the off peak hours. These hours were between 9 p.m. and 9 a.m. Monday through Thursday (48 hours) and from 9 p.m. Friday to 9 a.m. Monday (60 hours), for a total of 108 hours. As stated earlier, some problems were experienced with premature ice maker shut-off due to a variety of causes. Early shut down was compensated for by running the unit an equal time during the day. Despite these problems, ice maker run time was achieved during 105 out of the 108 off peak hours.

Some of the more cost advantageous time of day rate structures were selected to evaluate the results of the one week test (see Table 7, Appendix B). It was estimated that a 35 week cooling season (times one week) would simulate actual annual energy consumption and cost for the cooling system. Table 5 shows pertinent data from the rate structures used in the evaluation.

Table 5

Rate Data

Utility	Off Peak Rate[1]	On Peak Rate[1]	Reg. Rate[1]	Extra TOD Meter Chrg.
	(¢/kWh)			($/mo)
Union Electric Company	2.283	8.223	5.80	0
Central Illinois Public Service	2.08	10.59	5.96	4.20
Salt River Project	2.31	7.28	6.163	10.00
Gulf States Utilities	1.71[2]	12.31	6.03	0
Public Service Electric and Gas Company	3.935	15.957	9.797	11.10
Northeast Utilities	3.634	8.734	7.368	3.00
Omaha Public Power District	1.691	4.291	4.641	2.70

1. Includes fuel adjustment, if applicable.

2. Rate structure has 8 hour intermediate rate of 5.41¢/kWh for 8 hours, and peak period is only 4 hours long.

The following energy consumption data were used in the annual energy cost estimate.

Ice storage system:

Off Peak kWh/week: 397

On Peak kWh/week: 54

Conventional System kWh/week: 288

Table 6 displays the results of the estimated annual cooling energy cost for the various rate structures.

The annual savings of the ice storage system using these time of day rates ranged from \$50 to \$216 with an average of \$101. The major reasons for these moderate savings were as follows:

1. Large fuel adjustment charges significantly reduced the cost advantage of off peak rates.
2. High on peak rates added substantial cost to the total bill.
3. Additional monthly meter charges were large in some instances.

In addition, this estimate does not include additional cost benefits and penalties of other household power use with these varied rates.

Table 6

Annual Energy Cost

Utility	Ice Storage System with TOD Rates: Off Peak Power Cost	Ice Storage System with TOD Rates: On Peak Power Cost	Ice Storage System with TOD Rates: Extra Meter Cost	Total Cost	Conventional System & Rates Total Cost	Cost Difference
Union Electric Company	$317	$155	$ 0	$472	$585	$ -113
Central Illinois Public Service	289	200	34	523	601	- 78
Salt River Project	321	138	80	539	621	- 82
Gulf States Utilities	238	154[1]	0	392	608	-216
Public Service Electric & Gas Company	547	302	89	938	988	- 50
Northeast Util.	505	165	24	694	762	- 68
Omaha Public Power District	235	81	22	338	468	-130

1. Includes 60% of kWh (32) during intermediate rate.

CONCLUSIONS AND RECOMMENDATIONS

Based on the results and experience of this project, there are some design changes that could be made to reduce or eliminate problems encountered. The most frequent nuisance experienced was the early shutdown of the ice maker due to ice buildup. Possible ways to alleviate this problem would be:

1. Construct the ice storage box to be much taller than wide and deep. This will allow more height so storage volume is not wasted.
2. Include in the storage box, a better way of ice dispersal. This could be accomplished with a mechanically operated rake or even an electric fan. Another way might be to incorporate a system of multiple ice chutes, creating many piles rather than two.
3. Size the storage box, to accomodate a 50-50 mixture of ice and water at its maximum capacity. This has a disadvantage of requiring a larger volume.
4. Select an ice maker that produces ice in a shape that does not stack as easily as flake ice. Although this type of unit may not be as efficient. It was found that the flake ice product would stack in very steep cones.

During the conceptual design phase of the project, the allowance to be taken for the existence of voids in an ice pile was unknown. During the last phase of the project, when 2100 pounds of ice was made, and the percentage of voids was calculated to be about 50% (density equals 31 lb/ft^3).

Much larger than expected pipe losses were experienced. It is believed that this is due to the difficulties of insulating the water meter, water pump, and some piping irregularities. Pipe losses accounted for a 13.5% reduction in cooling potential leaving the storage box. The easiest way to reduce these losses would be to shorten the distance between the ice storage and air handler. (And eliminate the meter in an actual system.)

The project's greatest uncertainty was the ice production rate. Although ice production rates were measured under various conditions, the results indicated a large fluctuation in rates as supply water flow changed. The water flow rate to the ice makers decreased as the storage box water level decreased (pump head increased). The problem could be avoided by increasing the water level to insure adquate water flow to the ice maker.

Ice makers, in general, require more maintenance than a conventional residential air conditioner. This is primarily due to the mechanism required to make and transfer the ice. Additionally, the ice maker/ice storage system would cost about four times that of a similar capacity residential system, at present costs. Obviously, as shown in Table 6, savings of $100-$200 per year would not justify this large first cost investment.

Based on the results of this project and the experience with the ice storage system, it is believed the following points must be considered before ice storage systems are a viable economic alternative:

1. An economically competitive, low maintenance, ice storage system must be developed. This system must be a self-contained packaged unit, capable of providing 150 to 250 pounds per hour

while drawing only 5 kw to 8 kw, respectively. A possible solution might be a static system, where ice is produced on evaporator coils in the storage box, thereby eliminating the need for ice transfer. The cost of this unit cannot exceed that of a conventional unit by more than $2000.

2. The survey of time of day rate structures indicated most of the rates offered too little economic incentive to shift to off peak power usage with too large a penalty for continued on peak power usage. The average customer will still use about 20% of his power during the peak period, even after shifting air conditioning and hot water savings. It is believed the most advantageous rate structures are those that offer an intermediate period in addition to on peak and off peak periods (see Appendix B).

3. These rate structures must be guaranteed over the life of the thermal storage system. This is not to say, they may not fluctuate; but they must consistently offer comparative savings over the years.

An addendum to this report will be submitted in April 1983. This report will contain an analysis of the data from this project applied to a continuous operation strategy and to larger system applications.

APPENDIX A

DESIGN OF AN ICE AIR CONDITIONING AND STORAGE SYSTEM

Introduction

Problem Statement: "Design a thermal energy storage system for cooling utilizing ice as the storage media for a residence with a cooling load of 38,000 Btuh. Assume full load equivilent operating hours (FLEOH) of approximately 1600 hours."

The above general problem statement was further defined by using the following assumptions:

1. The project location will be Gainesville, Florida. Design conditions and electrical rates of Gainesville are applicable for analysis.
2. The ice air conditioning system will be the sole source of cooling for the residence. Therefore, design will be made for peak load conditions.
3. We are interpreting "thermal energy storage system" to infer the production of ice during possible off peak electrical rate hours (i.e. nightime) for use during higher demand daytime hours.
4. 1600 FLEOH equates to an approximate maximum of 8 FLEOH per day. This assumption dictates our design condition. It was verified with weather data found in Air Force Manual 88-8.
5. Off peak electrical rates will begin at 7:00 p.m. and end at 7:00 a.m.

6. Inside design conditions will be 80°F dry bulb and 67°F wet bu'b.

The above assumptions were made to define the problem to a point where our conceptual design process could begin. Additional assumptions will be made in the concept development and calculation part of this paper. These assumptions are more specific in nature.

Literature Review

The following is a reference list used in the design of our unit. Each entry is followed by a summary of the information used in this report.

1. ASHRAE 1969 Guide and Data Book, Chapter 43, Automatic Icemakers.

   This chapter described the various methods of producing ice, as well as the different forms of ice that can be made. It suggested that flake ice machines were the most efficient at making large quantities of ice. Typical operating performance data of an ice maker was given for various air and water temperatures. These data were useful in determining the effect of using cold water for ice production versus make-up water. The increase in efficiency was substantial. Other useful data included typical compressor sizes, unit dimensions, refrigerants, for various capacities.

2. ASHRAE, 1981 Fundamentals Book.

This text was primarily used to obtain design conditions, size piping, select pumps, and calculate R values.

3. Baumeister, Avallone and Baumeister, Marks' Standard Handbook for Mechanical Engieers, McGraw-Hill, 1978.

This text provided a summary of the various methods used to produce ice, but was used mostly to provide physical constants.

4. Carrier specification sheet for a residential air conditioner.

The specification sheet contained data used to estimate annual energy consumption which was compared to the ice air conditioner. It also provided typical specifications for the residential air handler, which will be the same for the ice air conditioner with the exception of the chilled water coil.

5. Dossat, Roy J., Principles of Refrigeration, John Wiley & Sons, 1971.

This text was used predominantly to size pipes and select pumps.

6. Jorgensen, Jay F., Cold Energy Storage, Heating/Piping/ Air Conditioning, April 1979.

This article describes a commercial application in Wisconsin using ice storage for air conditioning. It mentions that many electric companies are investigating "time-of-day" electrical rates. Two design limitations were set for this design. The first was that components of the design are readily available. The second stipulated the use of less stringent design conditions, "since this point is reached only several times a year," in determining required storage quantity. We agreed with the former stipulation, but felt that the latter would not apply in Gainesville.

7. Malloy and Turner, Thermal Insulation Handbook, McGraw-Hill, 1981.

   This reference was used to size insulation thicknesses.

8. Sakshaug, Thomas M., Performance of an Ice Air Conditioner, University of Florida Master's Thesis, August, 1951.

   This thesis described the performance of an ice air conditioner which used chilled water produced by ice and sprayed into the air stream. The results of this analysis showed the humidity problems associated with this type of system.

9. Scotsman, specification sheet for a flake ice maker.

   The specification sheet contained data such as capacity, dimensions, compressor size, electrical requirements, and power consumption.

10. Stubbefield, Richard R., Energy Efficiency Through Ice Storage, Heating/Piping/Air Conditioning, December 1979.

    This article contains the design of a commercial ice air conditioner that produces ice between the hours of 5:00

p.m. and 12 noon and uses ice for cooling between 7:00 a.m. and 7:00 p.m. It justifies the design based on "the advantage of future off peak rate schedules."

Concept Development and Calculations

Three methods of cooling air with ice were considered:

1. Return air would be blown directly across the ice.
2. Water would be chilled by the ice and sprayed into the air stream.
3. Water would be chilled by the ice and circulated through a coil which would cool the air.

The first two methods were discarded due to their inherent humidity problems. We judged the method of circulating ice chilled water through a coil as most promising and concentrated our design on this method.

Load Calculation: The maximum ice load requirement was calculated to be 2112 lbs/day.

38,000 Btu/hr x 8 hrs/day = 304,000 Btu/day

(Assuming 144 Btu/lb)

$$\frac{304{,}000\ \text{Btu/day}}{144\ \text{Btu/lb}} = 2112\ \text{lb/day}$$

Ice Production: We assumed off peak hours of 7:00 p.m. to 7:00 a.m. or a total of 12 hours for ice production. Therefore, we require the production of 2112 lbs ice per 12-hour period, or since capacities of ice makers are normally given per 24-hour

4" foamed polyurethane-------------R=23.53

Outside surface-------------------R= 0.68

Inside surface--------------------R= neg.

Metal-----------------------------R= neg.

R_t=24.21

$U \approx 0.04$ Btu/hr-ft^2-°F.

Estimate of storage losses:

$$Q_{loss} = UA(t_{out}-t_{in})$$

$$A = 52"\text{x}48"\text{x}2 + 52"\text{x}34"\text{x}2 + 48"\text{x}34"\text{x}2 = 11{,}792 \text{ in}^3$$

$$A = 11{,}792 \text{ in}^3 \times \frac{\text{ft}^2}{144 \text{ in}^2} \approx 82 \text{ ft}^2$$

Assume t_{out} = 82°F. Assume t_{in} = 32°F.

$$Q_{loss} = 0.04 \text{ Btu/hr-ft}^2\text{-°F} \times 82 \text{ ft}^2 \times (82-32)\text{°F.}$$

$$= 164 \text{ Btu/hr} \times 24 \text{ hr/day} = 3939 \text{ Btu/day}$$

Chilled water flow rate: The chilled water coil will be specified to yield a temperature difference (water) of 10°F. The flow rate of chillei water required for design conditions was calculated to be 7.6 GPM.

$$\dot{m}_{water} = \frac{Q_{coil}}{c_p(\text{temp. diff.})} = \frac{38{,}000 \text{ Btu/hr}}{1 \text{ Btu/lb- F}(10 \text{ F})}$$

$$= 3{,}800 \text{ lb/hr} \times \frac{\text{gal}}{8.34 \text{ lb}} \times \frac{\text{hr}}{60 \text{ min}} = 7.6 \text{ GPM}$$

period, 4224 lbs/24 hours. The Scotsman Volume Flaker, Model SF8, which produces 4330 lbs/24 hour, was selected for our application.

Ice Storage Sizing: The volume of ice required to satisfy our maximum load was calculated to be 37.7 cubic feet.

$$\text{Volume} = \frac{2112 \text{ lbs/day}}{56 \text{ lbs/ft}^3} = 37.7 \text{ ft}^3$$

The storage unit was sized based on the following criteria:

1. The size of the ice maker which will be mounted on top of the storage unit.
2. Allowance for a spray device at the top of the storage unit.
3. Allowance for minimum water requirement at the base of the storage unit.
4. Allowance of ice flake spacing and piling.

The following dimensions were determined to satisfy the above criteria without being too large:

Dimension	Height	Width	Depth
Outside	60"	56"	42"
Inside	52"	48"	23"

The difference in the above dimensions is attribued to 4" of insulation all around the storage box. See page 73, Figure 25.

Insulation sizing (storage box): Four inches of polyurethane insulation was determined to be adequate for use on the storage box.

<u>Chilled water supply and return pipe sizing</u>: 1¼" Type L Copper tubing will be used. This will yield a friction loss of 1.6 ft/100 ft at 7.6 GPM (from 1981 ASHRAE Fundamentals, Figure 2, page 34.3).

<u>Ice maker water supply requirement</u>: Water to be used by the ice maker will be drawn from the bottom of the storage tank. This is the water left over from air conditioning, which will be colder than make-up water. The water required by this ice maker will be 0.36 GPM.

$$4330 \text{ lb}/24 \text{ hr} = 180.4 \text{ lb/hr} \times \frac{\text{gal}}{8.34 \text{ lb}} \times \frac{\text{hr}}{60 \text{ min}}$$

$$= 0.36 \text{ GPM}$$

<u>Ice maker water supply pipe sizing</u>: 3/8" Type L Copper tubing will be used. This will give a friction loss of 1.4 ft/100 ft at 0.36 GPM (from 1981 ASHRAE Fundamentals, Figure 2, page 34.3).

<u>Chilled water pump sizing</u>: It was assumed that the maximum equivalent length of pipe would be 200 feet (including 100 feet of pipe, elbows, valves, coil). In addition, there is 5 feet of head to pump from the bottom of storage back to top. Therefore the total head equals 8.2 feet.

$$(1.6 \text{ feet}/100 \text{ feet})(200 \text{ feet}) + 5 \text{ feet} = 8.2 \text{ feet}$$

The pump efficiency was assumed to be 80%. The required pump brake horsepower was calculated to be 1/50 horsepower.

$$BHp = \frac{\text{Head x GPM}}{n \times 3960}$$

$$= \frac{8.2 \times 7.6}{0.80 \times 3960} = 0.02 \text{ Hp} = 1/50 \text{ Hp}$$

We chose 1/20 Hp as the size of the chill water pump, since pumping costs will be minimal.

Ice Maker supply pump size: The total head for this pump would be 8 feet. The required Brake horsepower for this pump was calculated to be 0.0011 horsepower, that is, a very small pump.

$$BHp = \frac{8 \times 0.36}{0.80 \times 3960} = 0.0011 \text{ Hp} = 1/1100 \text{ Hp}$$

We called Robbie's Reef of Gainesville, to see if they carried any aquarium pumps this small. They had several pumps that could be used for our application for about $15. These pumps would circulate about 20 gallons per hour.

Chilled water coil: The coil would be sized on the basis of the following data:

	$\dot{m}$	t_{in}	t_{out}
Water	7.6 GPM	33°F	43°F
Air	1400 GPM	80°F	56°F

Air handling unit: The air handler would be no different than the commercially available units with the exception of a chilled water coil in place of a refrigerant coil. We assumed typical air inlet and supply air temperatures given for commercially available units (see Figure 21, page 71).

Sprayer design: When cold water returns to the storage unit at about 43°F, it must be sprayed over the existing ice to be cooled back down to around 33°F. To accomplish this with minimal pressure drop but still maintain a good distribution of water over the ice, we designed the total area of holes in the sprayer equal to the area of our supply pipe. The design consists of a 3 foot copper tube with 1/8" holes on both sides, spaced every inch, for a total of 72 holes. This tube would oscillate, powered by the waper flow, similar to lawn sprinklers (see Figure 22, page 72).

Pipe insulation: We want a maximum temperature difference between storage and coil of 1°F. It was assumed that 3/4" insulation, with an R value of 4, was used. It was calculated that a maximum loss of 3800 Btu/hr could be tolerated for a 1°F temperature rise.

$$Q_{loss\ max} = \dot{m} \times c_p \times \text{(temp. diff.)}$$
$$= 3800 \text{ lb/hr} \times 1 \text{ Btu/lb-°F} \times 1\text{°F} = 3800 \text{ Btu/hr}$$

It will be shown in the following section that this thickness of insulation will allow a heat loss of about 432.0 Btu/hr for a 50 foot length of chilled water piping. Therefore, 3/4" insulation providing an R value of 4 is plenty for this application. This same insulation will be used on the ice maker water supply line.

Pipe losses: Chilled water pipe losses were estimated to be 432.0 Btu/hr.

$$Q_{loss} = U \times A \times (\text{temp. diff.})$$
$$= 0.25 \times (3.14 \times D \times L) \times (\text{temp. diff.})$$

The diameter of the pipe is 1¼" plus the 3/4" insulation times 2, for a total diameter of 2.75". The temperature difference was assumed to be 80°F minus 32°F, which equals 48°F.

$$Q_{loss} = 0.25 \times (3.14 \times \frac{2.75}{12} \times 50) \times 48°F$$
$$= 0.25 \text{ Btu/hr-ft}^2\text{-°F} \times 36 \text{ ft}^2 \times 48°F$$
$$= 432 \text{ Btu/hr}$$

The ice maker water supply line losses were calculated to be 56.6 Btu/hr.

$$Q_{loss} = U \times A \times (\text{temp. diff.})$$

We assumed the use of ½ the insulation of this application, which gives an R value of 2. The diameter would then be 3/8" plus the 3/8" insulation times 2, for a total diameter of 1.125".

$$Q_{loss} = 0.5 \times (3.14 \times \frac{1.125}{12} \times 8) \times 48°F$$
$$= 0.5 \text{ Btu/hr-ft}^2\text{-°F} \times 2.36 \text{ ft}^2 \times 48°F$$
$$= 56.6 \text{ Btu/hr.}$$

<u>Controls</u>: It was decided that our operating strategy would be to keep our ice storage unit at full load capacity each night. We also considered a control strategy that would produce part load capacities based on anticipated weather conditions. This idea was discarded because of the more complex controls requirement, the daily required occupant attention, and the low thermal losses of the storage box. Therefore, a control strategy was selected that would replace, each night, the previous days' cooling use plus the system losses. This will be accomplished by a shut off switch at the base of the storage unit that will shut down the ice maker when a full load of ice is produced. The switch will be activated when the buoyant screen separater (see Figure 24, page 73), is pushed down by the weight of ice. In addition, a timer will be installed giving the operator the flexibility to vary the operating hours of the ice maker.

The operation of the air handling unit and chilled water pump will be controlled by an indoor thermostat.

<u>Electrical</u>: The ice maker requires 208V/60 Hz/3 power with a 30 amp fuse. The running and maximum operating amperage is 20.4 amps. This power will be supplied through the disconnect, timer, and cut off switch, to the ice maker (see Figure 21, page 71).

Mounting and Unit Location: The ice maker will be mounted on top of the storage unit, so that the ice delivery chutes will supply ice to the center of the box. The total unit should be located in a garage or carport if available. Otherwise, it should be placed on a shaded slab, preferably on the northside close to the air handler.

Annual Energy Cost

The estimated annual energy cost for the ice air conditioner was $596. This was calculated considering the current electrical rate of Gainesville, Florida, which is without any off peak rate.

$$\frac{38{,}000 \text{ Btu/hr}}{144 \text{ Btu/lb}} = 264 \text{ lb/hr} \times 1600 \text{ hrs/yr} = 422{,}400 \text{ lb/yr}$$

Box losses = 3939 Btu/day x 270 days/yr = 1,063,530 Btu/yr

Chilled water pipe losses = 432 Btu/hr x 1600 hrs/yr
= 691,200 Btu/yr

Ice maker supply losses = 56.6 Btu/hr x 2194 hrs/yr
= 124,180 Btu/yr

Total losses = 1.878,910 Btu/yr x 1/144 Btu/lb = 13,048lb/yr

Ice maker load = cooling load + losses
= 422,400 lb/yr + 13,048 lb/yr
= 435,448 lb/yr

We assumed an increase in ice production capability of about 10% due to the colder water used for the making of ice, approximately 33°F versus 50°F.

$$\frac{435{,}448 \text{ lb/yr}}{1.1 \times 4330 \text{ lb/day}} = 91.42 \text{ days/yr} \times 24 \text{ hrs/day}$$

$$= 2194 \text{ hrs/yr running time}$$

2194 hrs/yr x 4.365 kwatt = 9577 kwh/yr

9577 kwh/yr x \$0.0619/kwh = \$593/yr

The pumping cost was calculated to be about \$3/year.

0.02 Hp x 1 kw/0.75 Hp x 1600 hrs/yr = 42.7 kwh/yr

0.0011 Hp x 1 kw/0.75 Hp x 2194 hrs/yr = 3.2 kwh/yr

45.9 kwh/yr x \$0.0619/kwh = \$2.84/yr ≈ \$3/yr

In addition, we calculated a typical energy cost for a conventional residential air conditioner. It was estimated that the annual energy cost of a Carrier Weathermaker Model 38GS036 was \$545.

EER = 6.9 Btuh/watt

$$\frac{38{,}000 \text{ Btu/hr}}{6.9 \text{ Btu/hr-watt}} = 5.507 \text{ kw} \times 1600 \text{ hrs/yr}$$

$$= 8812 \text{ kwh/yr}$$

8812 kwh/yr = \$0.0619/kwh = \$545/yr

Conclusions

Based on our assumptions and calculations, we conclude that it would be interesting to build a similar design and compare the energy consumption to that of a conventional unit. Our estimates show that the cost to run this unit would be about 10% more than the conventional unit, assuming no variable rate structure.

The first cost would be greater for this unit. It would be interesting to compute, based on an experimental study, the required rate structure adjustment necessary to make this unit a viable alternative.

FIGURE 21

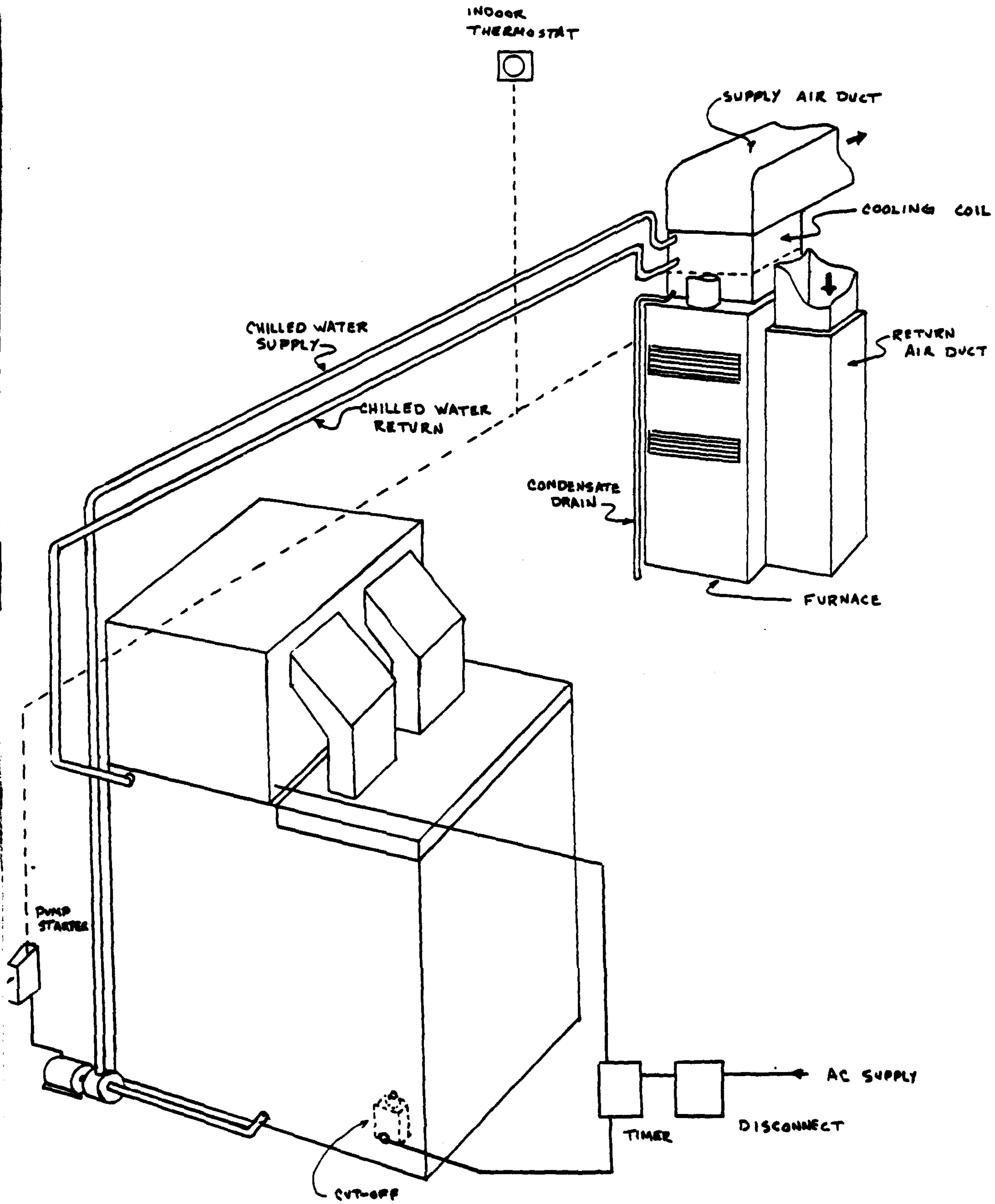
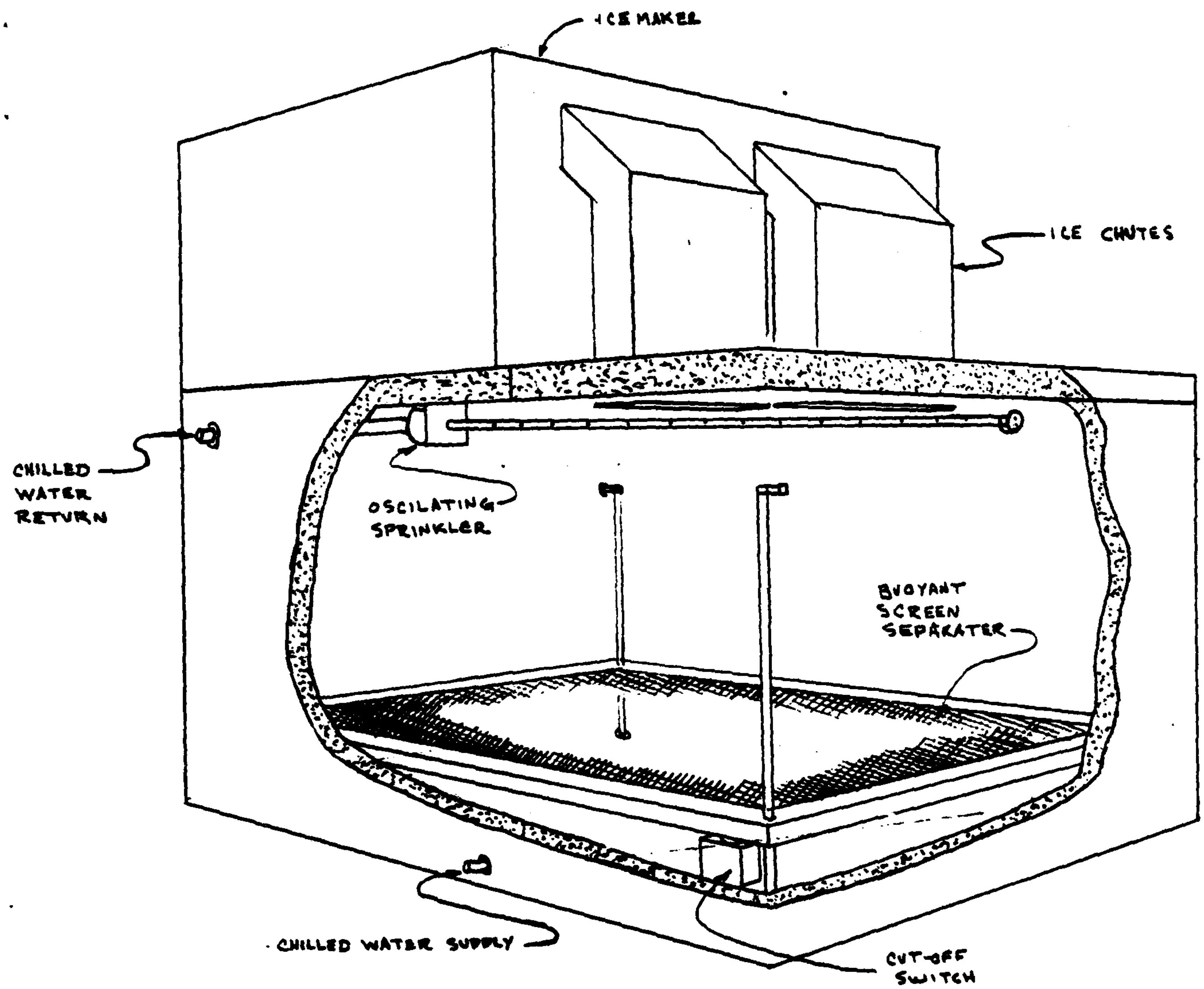

INDOOR
THERMOSTAT
SUPPLY AIR DUCT
COOLING COIL
RETURN
AIR DUCT
CHILLED WATER
SUPPLY
CHILLED WATER
RETURN
CONDENSATE
DRAIN
FURNACE
PUMP
STARTER
AC SUPPLY
TIMER
DISCONNECT
CUT-OFF

FIGURE 22



ICEMAKER AND STORAGE BOX

FIGURE 23

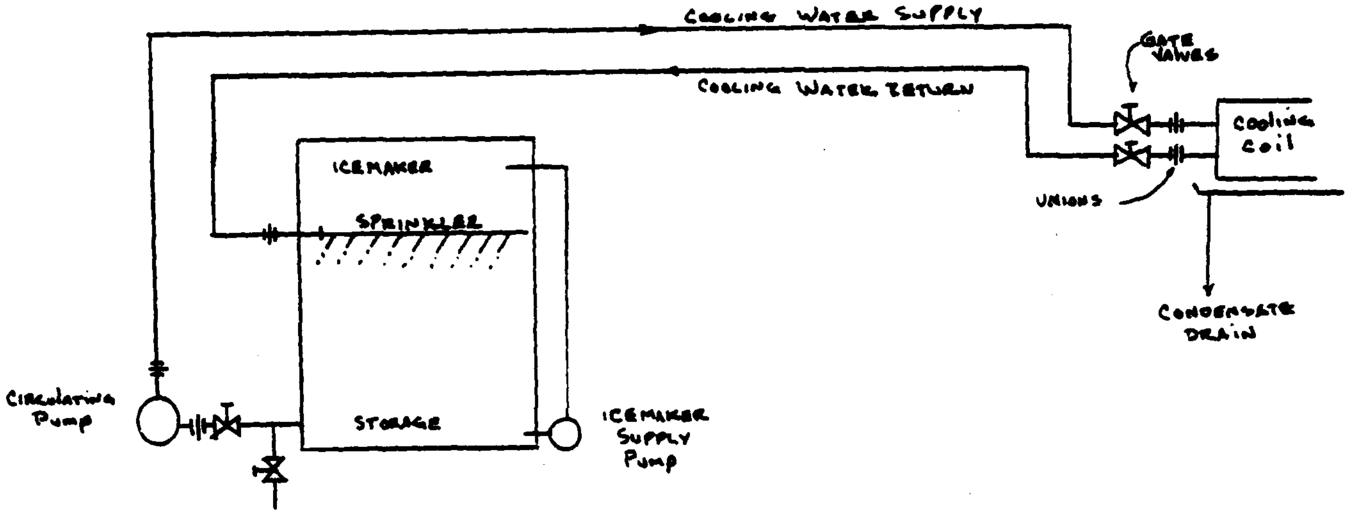


FIGURE 24

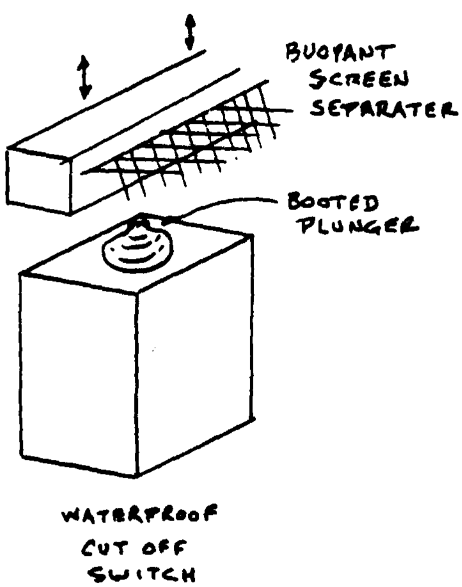
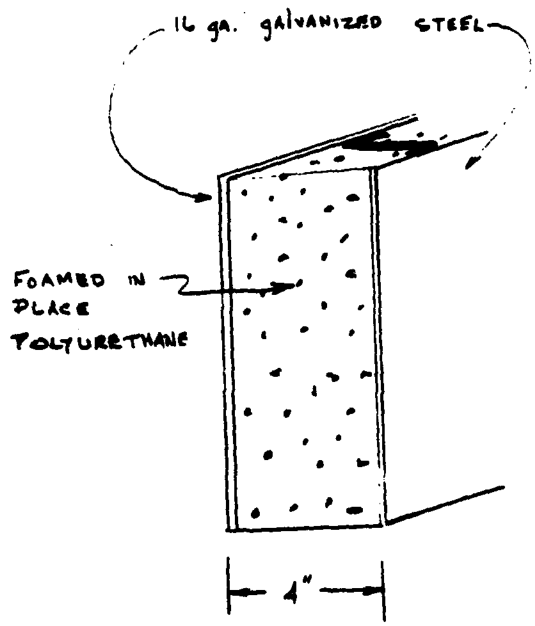


FIGURE 25



APPENDIX B

TIME-OF-DAY RATES

It is more expensive for an electric company to produce a kwh of energy during peak hours than during off-peak hours. Less efficient, stand-by generators generally are used to accomodate peak loads while all off peak power normally can be provided by the most efficient base-load equipment. In addition, the utility company must plan for future generating capacity based on anticipated peak loads. Large savings can be realized if unnecessary new power plants are not built (Reason, 1980). For these reasons it is advantageous to shift some peak energy, if possible, to less expensive off peak hours. Time-of-day pricing is one way of encouraging this shift. Electricity costs more during peak periods, reflecting higher utility costs, and costs less during off peak periods when production costs are less.

Time-of-day rate structures divide the day into two periods. The peak period may be between 9 and 16 hours in length and occurs normally during the day. A typical peak period (summer) will run from 9:00 a.m. to 9:00 p.m. The off peak period also varies from 8 to 15 hours and occurs during nighttime hours, typically from 9:00 p.m. to 9:00 a.m. Some time-of-day rates divide the day into three periods, incorporating an intermediate period in addition to on and off-peak periods. Most of the utilities had different on peak/ off peak hours for winter

Time of day metering requires the use of a more complex, and more expensive, meter. The meter keeps a cumulative total of electricity used during both peak and off-peak time periods. This additional cost is normally absorbed by the customer in the form of a higher basic customer charge.

A survey was conducted of 182 of the larger utility companies in the country requesting information on their experience with time-of-day rates. Of these, 108 (59%) responded, of which, 55 (51%) included information on residential time-of-day rate structures that were offered (see Table 7). The table contains summer rate information only, since the study involved ice storage for cooling purposes. The table gives a variety of information on 32 utility companies' regular residential and off-peak residential rate structures. Many of the off-peak residential rates are of an experimental or trial nature. Most of the time-of-day rates include a higher customer charge to cover additional metering charges. The average additional cost was about $5 and ranged between $1 and $23. Two companies charged a one-time fee for the meter changeover (see footnotes 4 & 7).

The most popular off-peak period ran for 12 hours, usually from 10:00 p.m. to 10:00 a.m. Most included weekends and holidays as all off peak hours. One company, Dallas Power and Light, included all winter (November through May) as off peak hours. The longest weekday off peak time period was 15 hours and ran from 9:00 p.m. to 12:00 noon. The shortest off peak time period was 8 hours running from 11:00 p.m. to 7:00 a.m. Gulf States Utility Company offered a time of day rate which included an intermediate period. This rate had off peak hours of 10:00 p.m. - 10:00 a.m. at 0.1¢/kwh, intermediate hours of 10:00 a.m. to 2:00 p.m. and 6:00 p.m. - 10:00 p.m. at 3.7¢/kwh, and on peak hours of 2:00 p.m. - 6:00 p.m. at 10.7¢/kwh. A fairly long off peak period would be required to make ice storage economically feasible. An off peak period of 12-16 hours would allow smaller ice making equipment, thus keeping down first cost. The rate with an intermediate period looks very promising since you could size the system to handle 80% of the cooling days during the 12 hour off peak period and handle design days by using the intermediate period whose rate is still below the regular residential rate of 4.42¢/kwh.

Columns 9 and 10 of the table calculate ratios of rate data. Column 9 is the ratio on peak rate to regular rate and column 10 is the ratio regular rate to off peak rate. To make ice storage an economic alternative will require rates where the on peak penalty, column 9, is not too large (not greater than 1.5). The average home will still consume about 20% of its power during on peak periods (10:00 a.m. - 10:00 p.m.) even if air conditioning and hot water is shifted to off peak times. Refrigerators/freezers, air handlers, televisions, and lights will still operate during this period. Demand penalties of 2 and 3 to 1 will offset any savings gained by off peak rates. On the other hand, off peak rates

Table 7

Utility Company	State	TOD Cust. Chg.	Off Peak Hours[1]	Off Peak Rate	On Peak Rate	Standard Cust. Chg.	Standard Residential Rate	Ratio On Peak / Reg. Rate	Ratio Reg. Peak / Off Rate	Fuel Adjustment[2]
Wheeling Electric Company	W.Va.	$ 8.36	11pm-7am +wkends.	3.365¢/kWh	5.962¢/kWh	$ 5.57	5.363¢/kWh	1.11	1.59	0[10] ¢/kWh
Texas Power & Light Company	Tex.	10.30	10pm-10am +wkends, hol.	1.25	9.70	7.00	4.09	2.37	3.23	2.25[11]
Northern States Power Company	Wisc.	5.50	9pm-9am +wkends., hol.	2.10	9.00	3.00	4.85	1.86	2.33	NA
Mississippi Power Company	Miss.	7.25	7pm-11am + wkends.	4.195	11.30	4.10	5.74	1.97	1.37	.662¢[11]
Municipal Light and Power	Alaska	10.39	7pm-7am	2.25	5.75	4.31	5.19	1.11	2.33	.155¢[12]
Salt River Project	Ariz.	15.00	10pm-10am	2.38	7.35	5.00	6.233[3]	1.18	2.63	-.07[10]
Central Maine Power Company	Maine	9.69	9pm-8am + wkends.	4.0093	8.7134	3.38	6.771	1.29	1.69	.207[11]
Florida Power Corporation	Fla.	8.50	9pm-12noon + wkends., hol.	4.116	8.026	5.00	5.371	1.49	1.30	1.003[10]

Table 7, Continued

Utility Company	State	TOD Cust. Chg.	Off Peak Hours[1]	Off Peak Rate	On Peak Rate	Standard Cust. Chg.	Standard Residential Rate	Ratio On Peak / Reg. Rate	Ratio Reg. Peak / Off Rate	Fuel Adjustment[2]
Central Hudson Gas & Electric	N.Y.	$11.00	10pm-8am + wkends.	6.3¢	11.0¢	$ 3.64	8.453¢	1.30	1.33	.155[10]
Union Electric Company	Mo.	7.29[4]	10pm-10am + wkends.	2.283	8.223	8.06	5.80	1.42	2.45	0
United Illuminating	Ct.	9.05	9pm-7am	6.76	10.76	5.00	9.9265	1.08	1.47	-.36[12]
Iowa Power and Light Company	Ia.	6.00	10pm-9am + wkends., hol.	3.19	11.05	4.00	6.23	1.77	1.96	.432[11]
Commonwealth Electric Co.	Mass.	10.76	9pm-9am + wkends.	0.55	6.33	2.18	3.175[3]	1.99	5.88	1.873[11]
Arkansas Power & Light Co.	Ark.	9.25	9pm-11am + wkends.	2.205	8.416	6.25	3.943[3]	2.13	1.79	.265[11]
Central Illinois Public Service	Ill.	6.80	10pm-10am + wkends., hol.	2.08	10.59	2.60	5.96	1.78	2.86	NA
Pennsylvania Power & Light Co.	Penn.	11.00	9pm-7am + wkends.	2.5028	6.8028	6.00	2.74[3]	2.48	1.10	NA
Indiana & Mich. Electric Co.	In.	5.95	11pm-7am + wkends.	3.349¢	5.311¢	3.45	4.121[3]	1.29	1.23	NA

Table 7, Continued

Utility Company	State	TOD Cust. Chg.	Off Peak Hours[1]	Off Peak Rate	On Peak Rate	Standard Cust. Chg.	Standard Residential Rate	Ratio On Peak / Reg. Rate	Ratio Reg. Peak / Off Rate	Fuel Adjustment[2]
Gulf States Utility Co.	Tex. La.	$ 5.50	10pm-10am + wkends.	0.10	10.7(2-6) 3.7(10-2, 6-10)[5]	$ 7.00	4.42	2.42 0.84[9]	44.2	1.61[11]
Public Service Electric & Gas Company	N.J.	15.50	9pm-7am + Sunday	4.70	16.722[6]	4.40	10.562	1.58	2.27	-.7654¢[10]
Gulf Power Corp.	Fla.	8.00[7]	9pm-noon + wkends., hol.	4.065	9.612	5.00	6.226	1.54	1.54	.337
Dayton Power & Light	Oh.	5.25	9pm-11am + wkends., hol.	2.024	8.161	5.00	4.576[3]	1.78	2.27	2.048[10]
Northern States Power Co.	Minn.	5.85	9pm-9am + wkends., hol.	2.76	10.4	2.85	5.78	1.80	2.08	NA
Omaha Public Power District	Neb.	6.65	10pm-9am	1.85	4.45	3.95	4.80	0.93	2.63	-0.159[11]
Metropolitan Edison Company	Pa.	9.00	8pm-8am + wkends.	3.50	8.60	6.00	5.672	1.52	1.61	1.144[10]
Northeast Utilities	Ct.	10.35	8pm-8am + wkends.	3.50	8.60	7.35	7.434[3]	1.16	2.31	.134[10]

Table 7, Continued

Utility Company	State	TOD Cust. Chg.	Off Peak Hours[1]	Off Peak Rate	On Peak Rate	Standard Cust. Chg.	Standard Residential Rate	Ratio On Peak / Reg. Rate	Ratio Reg. Peak / Off Rate	Fuel Adjustment[2]
Central Illinois Light Company	Ill.	$ 7.00	10pm-8am + wkends.,	2.80¢	12.00¢	4.07	7.50	1.6	2.70	1.23[11]
Iowa Southern Utility Co.	Ia.	26.70	11pm-10am + wkends.	4.154	7.244	3.60	5.284[3]	1.37	1.27	NA
Commonwealth Edison Company	Ill.	9.00	10pm-9am + wkends., hol.	4.022	19.376	1.00	8.9075	2.18	2.22	0.69[11]
Dallas Power & Light Company	Tx.	8.42	10pm-10am[8] + wkends., +Nov.-May	1.20	12.41	5.53	4.08	3.04	3.45	2.66[10]
New England Power Service	Mass.	11.06	9pm-8am + wkends.	2.098	7.745	2.62	5.025	1.54	2.40	NA
Wisconsin Power & Light	Wisc.	4.00	10pm-8am + wkends., hol.	2.50	9.10	3.00	5.96	1.95	2.38	.311[10]
Public Service Co. of New Hampshire	N.H.	7.00	10pm-7am + wkends., hol.	6.22	10.67	4.50	8.586[3]	1.24	1.39	NA

1. On peak hours are all hours not listed as off peak.
2. Fuel adjustment in ¢/kWh.
3. Incremental rate. Rate listed in average rate for 2000 kWh.
4. Plus $35 meter changeover charge.
5. Intermediate time period from 10 am to 2 pm and from 6 pm to 10 pm, cost is 3.7¢/kWh.
6. Average rate, different rate offered for Saturday and Monday through Friday.
7. Plus $154.40 meter changeover charge.
8. Wintertime hours (Nov.-May) are all off peak.
9. For intermediate period.
10. Fuel adjustment for last available month.
11. Fuel adjustment averaged over last 12 months.
12. Fuel adjustment averaged for 1982.

must show savings of 3 to 1 or better (over regular rates, column 10) to offset first cost of an energy storage system. In addition, the fuel adjustment charge is another important factor in time-of-day rates. For some utilities, the fuel adjustment is becoming such a large part of the total bill, that potential time-of-day benefits are not realized. Most utilities add fuel adjustments to all rates, which tends to level out the advantage of time-of-day rates.

In 1975, the Federal Energy Administration (FEA), now the Department of Energy, started a series of studies which included emphasis on time-of-day rates. These projects were accomplished in cooperation with individual state and local utility organizations (Gorzelnik, November, 1978). The program objectives were:

1. To demonstrate the technical and administrative feasibility of implementing innovative electric rates.
2. To demonstrate and gauge the precise extent of custom acceptance of such rates.
3. To gather and provide data for the analysis of impacts of such rates on customer and class-load patterns.

The results of these time-of-day experiments were mixed. Most of the studies showed little effect in overall consumption, but did show a shift of power to off peak hours, as expected (Gorzelnik, May, 1980). One study showed no total kilowatt hour consumption savings in winter, but a 5%

reduction in summer. This study showed that reduction of system peak increased with the customer's financial incentive as seen below.

Peak/Off Peak Pricing Ratio	Summer System Reduction %
2:1	18%
4:1	24%
8:1	30%

Financial incentive had no significant bearing on winter peak system reduction which averaged 5-10% (Gorzelnik, February, 1980).

Since these studies were temporary, they show only the shift of energy caused by consumer's change in electrical consumption habits. They do not show potential savings that could be realized from more active storage devices that require a capital investment. Customers will need a guarantee of 10 years or more on relative savings that can be gained from time-of-day rates before they can invest in hot or cold storage systems. The only affordable option they may exercise is a hot water timer.

APPENDIX C

BUDGET

Item No.	Materials Purchased	Cost
1	Lumber for roof	$ 73.50
2	Aluminum roofing	73.56
3	Lumber, materials for storage box	190.73
4	Insulation for storage box	119.88
5	Fiberglass for storage box	124.94
6	Circulating pumps	124.85
7	Pipe Insulation	132.56
8	Paint, caulk, miscellaneous supplies	45.21
9	2x4's, nails, aluminum tape, relay, alarm clocks, turnbuckles, micro switch...	72.97
		$958.20

The project was partially funded by a Gainesville Regional Utiities grant for $1,200.00, less 15% University of Florida overhead ($180.00) which left $1,020.00 available for the project.

In addition to the materials purchased above, some items were obtained at no cost. Foremost, was the ice maker which was generously donated by Crystal Tips Ice Equipment, McQuay Perfex, for the purpose of the project. The air handler, with chilled water coil was in place from a previous project. Also, the PVC piping, cement for the slab, and recording instruments were used at no cost.

APPENDIX D

ICE MAKERS

There are four basic types of commercial ice makers available. They are named for the type of ice they produce. Block ice makers use cannisters containing 300-400 pounds of water which are immersed in a brine for more than a day. This is often an uneconomical way of producing ice. Other ice makers may produce the ice in cylinders or sheets and mechanically chop them into ice cubes. Cracked or crushed ice may be produced in the same way, but the ice is mechanically broken before depositing into storage. Flake ice is made on the inside surface of a refrigerated cylinder partially submerged in water. As ice builds up, a revolving helix, sweeps the cylinder walls, pushing the ice out an opening at the top.

Flake ice machines are more efficient since they produce only a thin layer of ice and do not require any defrost cycle to discharge ice. Information was collected on various sizes of this type machine the results of our research could be extrapolated to larger applications. This will be accomplished in a supplement to this report to satisfy the conditions of the contract with Gainesville Regional Utilities.

The survey indicated a large range of ice maker sizes ranging from 200 to 170,000 pounds of ice produced per 24 hours. The typical company offered sizes in the 2000 lb - 20,000 lb per day range. The list price ranged from $6,000 to $11,000 per ton per day for the smaller units and $4,500 - $5,000 per ton per day for units in the 4-10 ton per day range. No prices were obtained for units in the 10 ton - 85 ton per day range. The actual price of these units would be between 55-60% of list price.

AD-A128 582   DESIGN CONSTRUCTION TESTING AND EVALAUTION OF A RESIDENTIAL ICE STORAGE AIRCONDITIONING SYSTEM(U) AIR FORCE INST OF TECH WRIGHT-PATTERSON AFB OH   J J SANTOS

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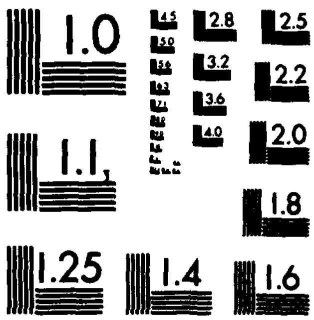

MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

APPENDIX E

PRINCIPLES OF ICEMAKER OPERATION

Mechanical

The evaporator surface, where ice is formed, is a vertical cylindrical tube with formed corrugations running horizontally for formation of ice.

A second cylindrical tube (housing) encases the evaporator and is sealed water-tight at its base by means of an "O" ring seal at the base of the evaporator.

As ice is formed on the evaporator surface, it is removed by a cage assembly, which revolves in a counterclockwise direction around the cylindrical evaporator tube.

Ice is allowed to build up in thickness until it is intercepted or contracted by a raised segment of the cage bar. Pressure from the segment fractures the ice to the bottom of the corrugation and removes it from the evaporator surface.

As ice is removed, the rotation at the cage bars moves the ice to the outer area of the annular space where it comes in contact with the spiral groove formed in the housing tube. By rotation against the spiral groove in the annular space, the ice is elevated to the spout in the front of the housing tube above the water level where it is ejected and drops through the ice chute into the storage bin.

A gearmotor drives the cage by a means of a square drive shaft which fits into the cage assembly.

The gearmotor assembly is mounted above the housing tube and the entire assembly is held together by two (2) tie bolts from the gearmotor assembly to the bottom of the evaporator base and two (2) nuts at the upper rear gearmotor support bracket.

The shut-off of the unit, when the storage bin is full of ice, is achievewd by using a mechanical bin control. The ice leaving the evaporator builds up in the ice discharge chute until in contacts the bin control blade, pushing it forward to actuate a micro-switch which shuts the unit off.

When ice is removed from the storage bin, the ice build-up in the chute drops into the storage bin, allowing the bin control blade to drop down, closing the micro-switch contacts and re-starting the unit.

Refrigeration Circuit

The Crystal Tips unit employs a hermetic compressor, air cooled condenser, drier, 2 evaporators, and thermostatic expansion valve refrigerant feed with a low side accumulator with heat exchanger.

A normally open hot gas solenoid is utilized to equalize high and low side pressures and temperatures upon unit shutdown to permit easy restart of the compressor and gearmotors. (See Electric Circuit for details.) Access to low side and high side are provided by compressor service valves.

Water Circuit

Make-up water (for ice production) is fed through a fitting at the rear of the unit into a float reservoir which maintains a predetermined water level in the reservoir.

The water in the reservoir is gravity fed into the evaporator chamber, replenishing the supply as the water is frozen into ice and discharged.

Electric Circuit

A momentary push button switch and lockout relay are provided for protection of the unit in event of power failure. If power to the unit and crankcase heater is removed for any reason, this switch must be pushed to reset the relay. Reset only after the crankcase heater has been energized for a period of time to warm the compressor.

A relay is wired in series with the crankcase heater. Its contacts are in series with the lockout relay coil described above. If the heater burns out, the relay contact will open, de-energizing the lockout relay described, shutting the unit down until the heater is replaced.

The on-off switch opens all control circuitry when in the off position. This switch must be in the "on" position and the lockout relay set before the unit will run. The control relay coil is energized through a series of circuits with the on-off switch, ice level controls, the N.C. contacts of a thermal delay relay, and the right hand gearmotor overload.

Once the control circuit is "made," the thermal delay relay heater is energized. When fully heated (5 to 15 seconds) the relay contacts open, transferring the control circuit to a set of contacts on the control relay. An interruption of the control circuit at this time, by on-off switch, or full bin, will shut the machine down until the thermal delay relay cools and resets. This takes approximately 2 to 5 minutes, depending on ambient air temperature and length of time unit has been running.

The delay feature permits the refrigeration system time to balance and evaporator surface to defrost sufficiently to permit the compressor and gearmotor to restart easily.

The second set of contacts on the control relay energizes the main relay coil and the left gearmotor.

If the gearmotor overload (manual reset) opens, the main relay contacts will open and shut the unit down. The three sets of contacts energize 1) the hot gas solenoid, fan motor and compressor, 2) right hand gearmotor and 3) the third power line on three phase units.

A dual pressure control is wired into the circuit ahead of the compressor to cycle the compressor in the event of fan failure, refrigerant circuit malfunction, or water loss to the evaporator or condensor.

The compressor capacitors and relays are located in the unit electrical control box for accessibility and ease in checking circuits.

A frozen evaporator will overload the gearmotor manual reset overload and cause it to open, shutting the system down. To reset, the red reset button must be pushed.

Table 8

GENERAL SPECIFICATIONS

Model No.:	FA-229BP-01
Dimensions (In.) W x H x D:	48 x 26 x 31 3/4
Production Capacity/24 hr:	3300 lbs.
Condenser:	Air Cooled
Compressor:	3 HP
Electrical:	230/60/1
Min. Circuit Amps:	25
Max. Fuse Size:	40 Amps
Shipping Weight:	550 lbs.

Table 9

Model	FA*229C
Compressor	3 HP Semi Hermetic
Condensor	Air Cooled
Condensor Fan Motor	1/6 HP, 1050 RPM
Gear Motor	1/4 HP, 12 RPM
Evaporator (2 each)	Flooded Cylinder
Head Pressure	See Table 11
High Pressure Control	Out 380 PSIG, diff. 65 PSI
Low Pressure Control	Out 3 PSIG, diff. 45 PSI
Refrigerant Charge R-502	52 oz
Refrigerant Control	T.E.V. 10°F superheat
Suction Pressure	Non-adjustable, see Table 11
Electrical	See Table 8
Water Inlet (Ice Maker)	1/4 inch F.P.T.
Shipping Weight	550 lbs

Table 10
Ice Production Chart

Air Temp. F	Ice Making Capacities Pounds per 24 hours — Incoming Water Temperature of 50	60	70	80	90
60	3300	3130	2950	2780	2610
70	3050	2880	2710	2540	2360
80	2800	2630	2460	2280	2040
90	2570	2400	2220	2130	1880
100	2340	2170	2100	1820	1640

Table 11
Systems Pressure

Air Temp.	Pressure PSIG Suction	Head
60	24	196
70	24	206
80	27	230
90	30	268
100	32	298

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ADDENDUM

to

DESIGN, CONSTRUCTION, TESTING AND EVALUATION OF A RESIDENTIAL ICE STORAGE AIR CONDITIONING SYSTEM

By

Thomas A. Ritz

ADDENDUM TO A MAJOR PROJECT PRESENTED
TO THE COLLEGE OF ENGINEERING OF THE UNIVERSITY
OF FLORIDA *IN* NOVEMBER 1982
BY J. JAY SANTOS AND THOMAS A. RITZ

UNIVERSITY OF FLORIDA

April 1983

Introduction

This addendum is a supplement to the project report "Design, Construction, Testing and Evaluation of a Residential Ice Storage Air Conditioning System" dated November 1982.

This supplement will address the use of an ice storage system in a commercial size application. The design and costs of a commercial size system will be discussed. The conclusions and recommendations resulting from the evaluation of the residential size system will be applied to the commercial size design and changes or alterations to the system will be made. Finally, the improved commercial size design will be compared with a conventional system based on both continuous operation and limited operation strategies.

For the purpose of designing and analyzing a commercial size system as indicated above, the following criteria have been selected:

1. A cooling load of 600,000 Btuh (50 tons), with 8 full load equivalent operating hours (FLEOH) per day the maximum. This system could be utilized in a moderate sized office complex or commercial establishment with normal daytime hours.
2. A seasonal load of 1800 FLEOH per year or 225 days at full daily load.
3. The ice storage system will be analyzed at 12 hour charge capacity and at 24 hour charge capacity (continuous operation during full load).

System Design

The most important consideration in the design of this system is the selection of the ice storage unit. In the residential unit prototype tested, the ice was produced in the icemaker and transferred to the ice storage box, a dynamic storage system. This process was the most troublesome part of the project. As noted in the report, problems centered around the "stacking" or buildup of ice which often shut down the ice maker before sufficient ice was produced. The control mechanism, a trip switch on the ice chute, was difficult to adjust and maintain. Leaks were experienced at the base of the ice making compartments. In addition to the mechanical problems of the transfer of ice, this dynamic system required excess storage space for voids and stacking. It also contributed to system efficiency losses by bogging down the ice maker when ice accumulated in the chutes. The chutes themselves were an additional area for losses from heat transfer. These problems can be eliminated in a static storage system, when refrigerant is passed through coils inside the ice storage container and ice forms on the outside of the coils. This type of storage system allows solid ice buildup on the coils and does not require a control system for ice buildup.

As in the prototype residential unit, the ice storage container is sized to hold about 1 1/2 times the full load daily ice requirement. For a 50 ton commercial unit with 8 FLEOH per day, storage requirement is 25 tons of ice. The static system does not acquire the void and stacking allowance, but does require space for coils, the volume of the storage container must be approximately 1500 ft^3.

Although a wooden storage box was suitable for the prototype unit for the project, it is not practical or desirable for a larger system. A fiberglass or polyethylene tank would provide adequate strength and watertightness and still be economical. A tank of 16 ft. diameter and 8 ft. high can be used for this storage system. Other sized or even multiple smaller units could be used as conditions dictate. The tank could be buried or partially buried as needed. Poured or foamed insulation of 5 inches thickness will result in an R value of 30 or greater, similar to prototype storage. Expanded polystyrene or polyurethane foam or a combination of both can be used. An outer cover of sheet metal should also be used for insulation protection. Pumping and piping from the storage vessel to the AHU should be similar to the prototype unit except sized to larger capacity. Molded polyurethane pipe insulation with aluminum faced Kraft paper or plastic covering would provide an R value of 20 or more for 2 1/2" thickness.

Since the larger system will utilize a static storage system, an "ice maker" is not required. Refrigeration unit will be a low temprature direct expansion unit. The compressor will run steadily with no cycling and no unloading is required. In order to provide the required 50 tons of cooling for 8 hours with an ice storage system operating during 12 hours off peak time, a 40 ton refrigeration unit is required. (Actual requirement is 33 tons, but units are usually only available in 10 ton increments for this range.) For continuous operation a 20 ton unit is required.

The air handling unit for the commercial size system is a standard centrifugal fan unit with chilled water coil. Because the temperature of the chilled water in the ice storage system is colder (33-35°F) compared with normal chilled water systems (45-55°) smaller fan units and ducting can be used. The air handling unit for this design is assumed to be 80% of similar conventional system size.

System Comparison

The prototype residential size unit tested for this project consumed over 50% more electricity than a conventional air conditioning system. The ice maker performance was felt to be the largest factor in this analysis. Ice production was generally at 70 to 80% of expectations based on manufacturers' data. Problems with the water feed system and ice delivery contributed to some of the performance loss. Another area which reduced the overall efficiency of the prototype system was the high heat losses in the piping system. As noted in the report, these were felt to be due to difficulty in insulating components such as the water meter and pump.

Ice storage systems require an additional heat transfer circuit and therefore should generally not be expected to obtain efficiencies of conventional systems. Our research of performance of other experimental and commercial ice storage systems does indicate that they can commonly be 90% as efficient or more. This is generally true of the larger static systems.

Due to the lower chilled water temperatures and greater temperature drops in an ice storage system, the size of the air handling unit can be reduced as indicated in the previous section. Similar reductions in piping, pumping and auxiliary electrical demands can be made when comparing an ice storage system with a conventional chilled water system. In some buildings such as offices or schools, the cost of ducts and piping can account for more than half of the installed cost of an HVAC system. Therefore, reductions of 25 to 50% in the size of these components can have considerable effect on total initial cost.

The lower coil temperatures of the ice storage system allow a faster response time to deliver cooled air to room outlets. This reduces the time needed to precool offices before business hours.

The lower coil temperatures also provide more condensate removal resulting in lower relative humidity. For the purpose of making cost comparisons of the ice storage system and a conventional system, the assumption will be made that reductions of fan and pump sizes for the ice storage system will offset the inherent loss of efficiency due to the additional circuit. The following cost comparisons are based on general estimated equipment costs, current local electrical rates and demand charges, and hypothetical off peak rates based on some of the more typical used in the project report.

Conventional System

Normal Electric Rates

50 ton refrigeration unit	\$ 30,000
Distribution System	25,000
Total installed system	\$ 55,000
*1800 FLEOH @ 4.80c/kwh (assume 1.5 kw/ton)	\$ 6,480
**75 kw demand @ \$4.30/kw ea.mo.	3,870
Yearly electrical cost	\$ 10,350

*50 tons x 1800 hr x 1.5 kw/ton x \$.048/kwh

**50 tons x 1.5 kw/ton x \$4.30/kw x 12 months

Full Ice Storage System

Normal Electric Rates

40 ton refrigeration unit	$ 24,000
Distribution System	20,000
Ice Storage System	12,000
Total installed system	$ 56,000
1800 FLEOH @ 4.80 ¢/kwh	$ 6,480
60 kw demand @ $4.30/kw	3,096
Yearly electrical cost	$ 9,576

Yearly savings in electrical cost: $774

Payback period for additional cost:

$$\frac{1000}{774} = 1.3 \text{ years}$$

Continuous Operation System

Normal Electric Rates

20 ton refrigeration unit	$ 15,000
Distribution System	20,000
Ice Storage System	12,000
Total installed system	$ 47,000
1800 FLEOH @ 4.80¢/kwh	$ 6,480
30 kw demand @ $4.30/kw	1,548
Yearly electrical cost	$ 8,028

Conventional System

Time of Day Rate Structure*

50 ton refrigeration unit	$ 30,000
Distribution System	25,000
Total installed system	$ 55,000
1800 FLEOH @ 8.60 ¢/kwh	11,610
or	
1800 FLEOH @ 6.80 ¢/kwh	9,180

*Hypothetical time of day rate structure:

Standard Rate	6.80¢/kwh
On peak	8.60¢/kwh
Off peak	3.80¢/kwh

Full Ice Storage System

Time of Day Rate Structure

40 ton refrigeration unit	$ 24,000
Distribution System	20,000
Ice Storage System	12,000
Total installed system	$ 56,000
1800 FLEOH @ 3.80 ¢/kwh	5,130
Yearly savings over conventional at on peak rate	6,480
Yearly savings over conventional at standard rate	4,050

Continuous Operation System

Time of Day Rate Structure

20 ton refrigeration unit	$ 15,000
Distribution System	20,000
Ice Storage System	12,000
Total installed system	47,000
900 FLEOH @ 8.60 ¢/kwh	5,805
+	
900 FLEOH @ 3.80 ¢/kwh	2,565
Yearly electrical cost	$ 8,370

Conclusions

A survey of the above cost summaries shows that ice storage systems can be economically competitive with conventional air conditioning systems. The economical advantage of the ice storage system depends entirely on the electrical rate structure available. A straight rate will not yield any savings in energy cost since the same amount of electricity is used. Initial costs can be reduced, however, with the ice storage systems. Under a straight rate structure, a continuous ice storage system can save about 15% in initial cost.

Under a normal commercial electrical rate structure where a demand charge is included, the full ice storage system will result in enough annual energy savings to pay back the additional system cost in one or two years. A continuous ice storage system will offer even better annual energy savings and a smaller installed cost under the normal rate structure with a demand charge.

Where off peak rate reductions are available the full ice storage system has the greatest economic potential. In the above example the annual energy cost could be reduced by about 45% if compared to standard rates. It must be taken into account though that almost all rate structures that offer off peak reductions also include on peak penalties. The energy costs in the above examples reflected only electricity to operate the cooling system. Under the time of day structure the user must pay for normal usage such as lighting and other equipment at the penalized on peak rate. Some time of day rate structures include so much of an on peak penalty that this portion of electrical load which cannot be shifted to off peak periods will more than off set the savings of shifting the cooling load.

It must be concluded that the variety and severity of utility rates available makes it impossible to draw any hard conclusions of whether an ice storage system is the better choice.

Each application must be individually evaluated with its particular operating criteria, such as loads, operating hours and portion of total energy used against the exact rate structure available. There will be many times though when an ice storage system will prove the best for a commercial size application.