OFF PEAK COOLING USING AN ICE STORAGE SYSTEM

by

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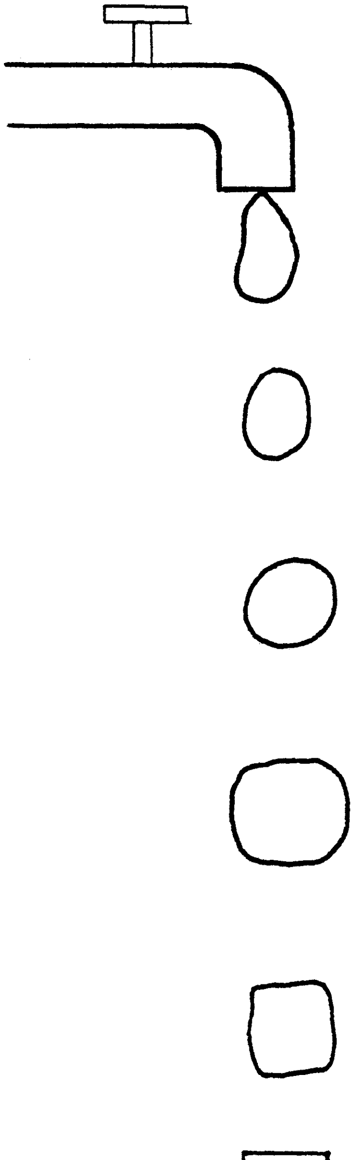

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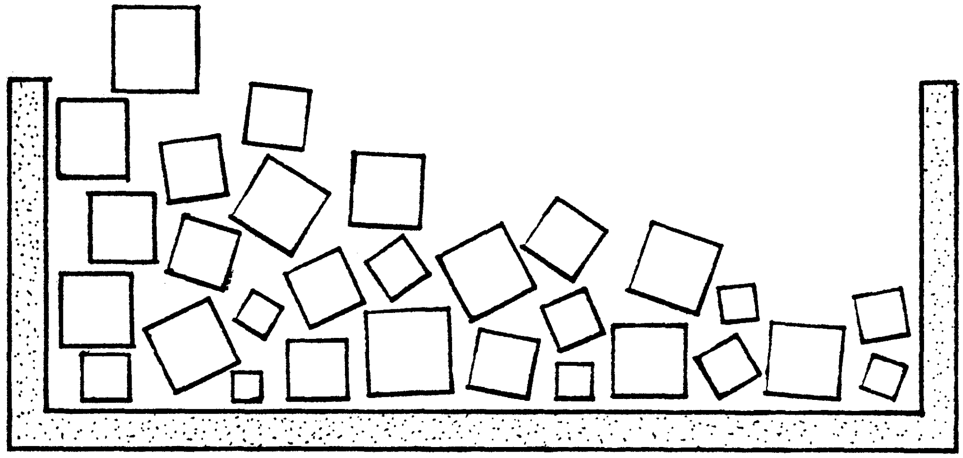

ABSTRACT

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Submitted to the Department of Architecture on June 16, 1980 in partial fulfillment of the requirements for the Degree of Master of Science.

The electric utilities in the United States have entered a period of slow growth due to a combination of increased capital costs and a staggering rise in the costs for fuel. In addition to this, the rise in peak power demand continues almost at historical levels resulting in lower plant utilization. Current rate schedules do little to improve the utilities' load factors and,in fact, encourage consumption. Time of day rate structures have been suggested as one load management device. This thesis investigates the impact of commercial cooling systems on the utilities' supply picture and describes an off-peak cooling system which would enable a building operator to shift chiller operation to off-peak hours.

The chillers draw heat from a water/glycol coolant, cooling it to 20°F. The coolant circulates through a series of coiled pipes inside a water filled storage tank. As heat is drawn from the water, ice forms around the pipe heat exchanger. With a coolant temperature of 20°F the ice cylinder will form out to a diameter of 3.4" in 10 hours. Optimum pipe spacing is 3.5" on center. Polyethylene pipe is preferred to copper pipe for cost and fabrication resaons. The plastic pipes are grouped in discrete modules which allow flexibility in design. Building cooling loads are managed by circulating the remaining 32°F tank water through a heat exchanger coupled to the air handling unit's cooling coils. The warm water is returned to the tank where the heat is absorbed by the ice.

Economic analysis using the present electric schedules indicate a favorable return on investment. Time of day rates would make the system look even more desirable.

Thesis Supervisor: Timothy E. Johnson
Title Research Associate

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I would like to express my gratitude to Timothy Johnson for his help throughout the course of my work here at MIT, not only on this thesis. He has been a constant source of intellectual stimulation (and money), without which I would have accomplished little. I deeply appreciate all he has done for me.

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Bonnie Blanchard helped me organize and edit this work and is responsible for its production.I would also like to thank her for all the assistance she provided me during my 2 years at MIT.

Alex Lohr and Chris Mathis helped me through the rough times of my thesis. Their supply of graphic assistance, interesting conversation, and bourbon provided a welcome help throughout the course of my work. I will definitely miss their company.

If ever there was a fundemental and perennial source of strength and wisdom, my parents are it. They are as much the authors of this thesis as I .

I would like to dedicate this thesis to Leslie. She managed to survive this ordeal with no complaints. Let's hope the years to come fare as well.

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INTRODUCTION

The electric utilities in the United States are no longer able to easily satisfy the rising demand for electricity. The utilities' load factors have been declining since the early sixties due, in large part, to the increasing summertime demand for cooling. The diminished capacity factors result in higher costs for the utilities and ultimately, the consumer. In addition, the peaking demand is typically satisfied with oil or gas fired turbines which exhibit low operating efficiencies. With declining domestic reserves of oil, the oil used for these peakers is coming increasingly from imported suppliers. The rate schedules used by the electric utilities only somewhat reflects the cost of this poor load factor and does nothing to slow down the growth in consumption. In fact, the rates were designed in an era when utility growth resulted in lower unit energy costs. While that situation no longer exists, the rates remain. A number of load management strategies are being investigated by the utilities. The purpose is to improve load and capacity factors, resulting in a reduced need for new power plants, especially peakers. Time of day rates are being introduced throughout the country as one way of reducing demand during a utilty's peak hours. Being similar to the phone company's rate structure, customers would be charged more for each KWH consumed during the utility's high demand period.

Unfortunately, cooling in buildings is needed during these peak hours. Most of the rate oriented load management strategies would drastically increase building operating costs due to the large cooling loads. Cool storage systems would allow the customer to run the cooling equipment at night during inexpensive off-peak hours and store the coolness for later use. A number of systems have been designed to date, with only the cool water storage type finding wide commercial application. Most of the systems suffer from bulkiness or cost or both. Smaller, cheaper systems are needed to insure widespread acceptance.

This thesis explores the evolution of the utilities and the underlying problems which gave rise to the need for cool storage systems. A variety of systems are described along with their inherent problems. A new type of modular ice storage system is introduced. Experiments run with a 30,000 BTU storage unit provided the data needed to confirm theoretical design methods. These methods were then used to determine optimum pipe length and space required for a given storage capacity. Commercial scale application is considered. This system should lower volume requirements by 80% with little or no additional cost whem compared to sensible water storage. In addition, an improvement to the ice making heat pump system is suggested in order to drastically lower capital costs and improve its compatibility with inexpensive packaged chillers. Finally, an economic accounting of the proposed pipe coil system is discussed using conventional rate structures.

OFF PEAK COOLING and the ELECTRIC UTILITIES

BIRTH OF AN INDUSTRY

The electric utilties were born in an era when coal and solar energy (wood) provided the power used to transform America from a rural, agrarian society into a robust industrial world power. The utilities growth mirrored this social evolution as much as it shaped it. The birth of the electric power company dates back to the 1880s when Thomas Edison's power plant in lower Manhatten provided electricity for the nighttime lighting of streets and shops in the neighborhood. At the time, lighting was the only available application of electric power resulting in an extremely inefficient use of the capital resouce embodied in the power plant. Edison embarked on a strategy which was to be characteristic of the utilities for the next 100 years, namely the development of home appliances and industrial equipment which required the reliable and continuous supply of electric power. Now the utility is not only insured of round the clock demand, but also of unparalleled growth in that demand. In the first quarter of this century, electricity was produced by small companies serving localized area. In the 1920s many of these companies were consolidated, by way of the holding company in order to realize advantages of economies of scale.[1] In the 1920s the Federal Trade Commission began a massive investigation of the holding companies. This investigation ultimately spawned the Federal Power Act which provided for federal regulation of the wholesale transactions of investor owned utilities. At the same time, the courts established the states prerogative to regulate retail power sales. The electric utility industry became an officially recognized monopoly under regulatory control. This control was exercised to limit profits to a "reasonable amount" of return on investment.[2] As profits were tied to investment, the utilities insured continual prosperity through growth. The utilities became a capital intensive industry with growth rates that required an investment to earnings ratio of 5 to 1.[3] In over 80 years the industry experienced an unbroken chain of technical advances and increasing returns to scale such that its average cost of production fell steadily. This caused the constant dollar prices for electricity generally to fall or remain steady while per capita disposable income, and many prices, actually rose. With utility system expanding under conditions of timely technological innovation and increasing economics of scale, virtually all customers were better off as electricity consumption expanded. New efficient means of production and labor saving devices were eagerly accepted by industry, commerce, and the home owner resulting in the growth patterns exhibited in Figs.1, 2. In the years before 1960, most utilities were winter peaking, that is, they experienced the greatest power demand during the winter months due to use of electric resistance heating. This changed dramatically as the public came to accept and demand air conditioning systems at the office place and in the home. By the early 1970s most utilities, except for those in the northern tiers of the country, had become summer peaking, resulting in an advertising push for new homes to be heated and cooled electrically, thereby attempting to balance demand through

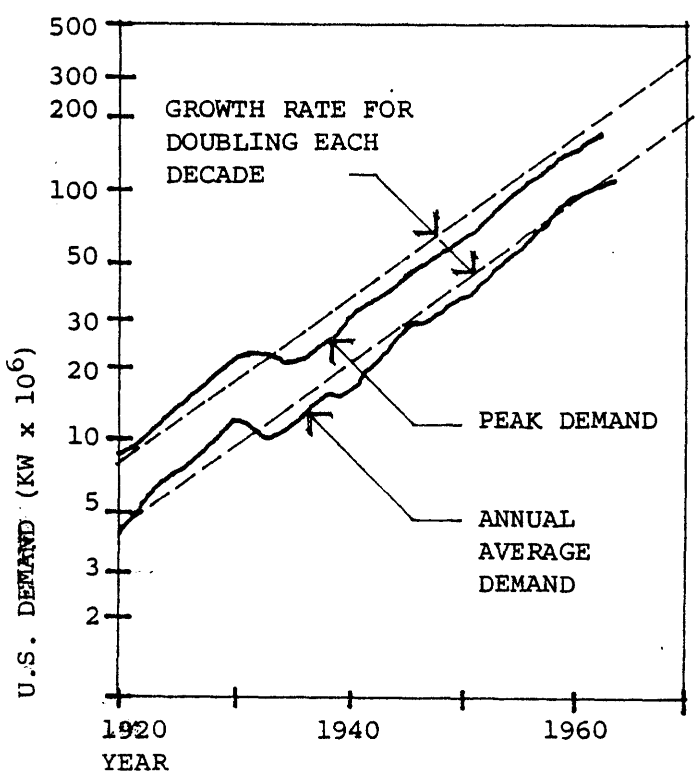

FIGURE 1 Electric Power Demand During the Past Half-Century
(courtesy C. Benton)

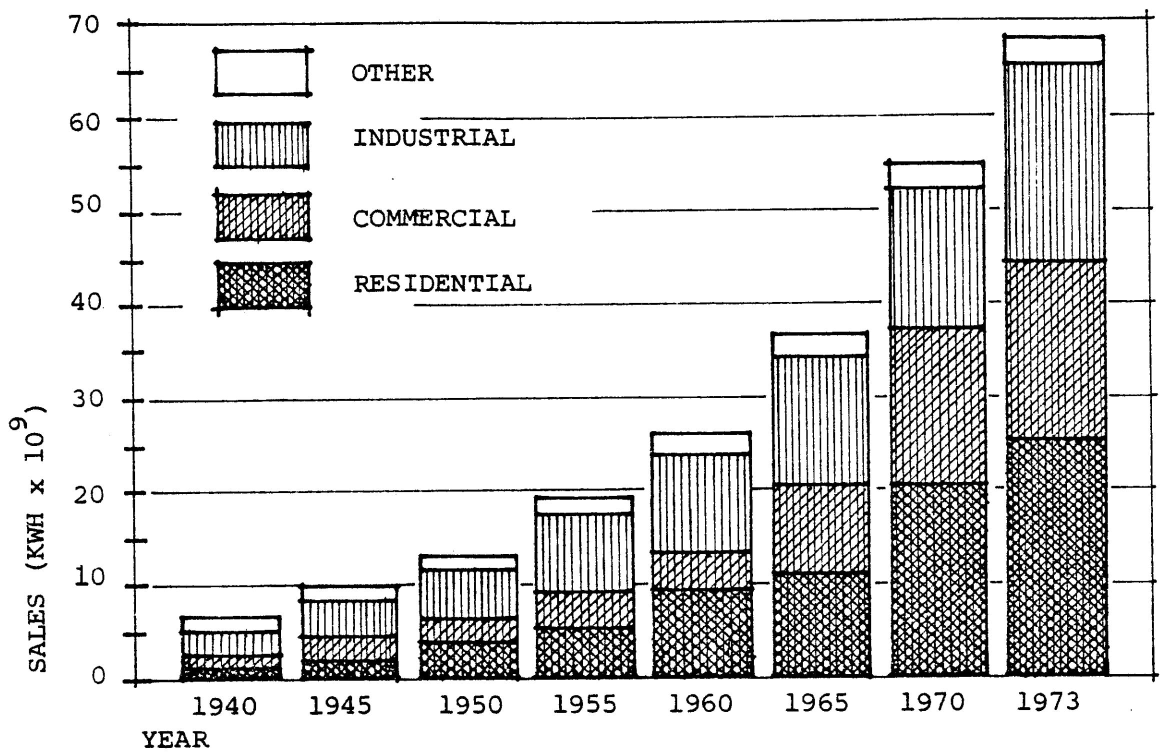

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FIGURE 2. Electrical Energy Sales in New England, 1940 to 1973

(courtesy C. Benton)

the year. Construction of office buildings underwent transformations during the 1950s and 1960s. Transparent operable windows were replaced with fixed tinted glass requiring both increased artificial lighting levels and mechanical ventilation for the perimeter zones. As buildings got larger, the percentage of interior zone increased along with the requirement of 100% artificial lighting. Recommended lighting levels continued to increase over the years as a result of studies, often times funded by the Electric Power Research Institute, thereby insuring future markets.

One of the most important contributing factors to the growth patterns experienced by the utilties was the way in which the price for power was determined. Throughout the country, higher consumption was reinforced through the use of declining rate block structure, which basically meant that the more a customer consumed, the less it cost per unit of energy. It is important to remember that as the growth of the electric utility sector continued, advances in power generation technology brought costs per unit of energy down. Since the unit cost of electricity declined with increasing consumption, there was an economic justification for establishing electricity rates in decling blocks which served to crudely mirror the cost outlook of utility systems. However, it must be recognized that such a rate design becomes onerous to society when the cost trend reverses as it has done in recent years for most utilities.

Plant expansion programs were designed to provide for this growth with new large efficient baseload plants. As the older inefficient plants were retired or moved to peaking duty, the cost of electricity decreased. Part of this plant expansion involved bringing on line large 1000 MW nuclear power plants which promised further economies of scale through lower fuel and production costs.[5]

TROUBLED TIMES FOR THE ELECTRIC UTILITIES

This bubble of prosperity burst on the utilties during the early 1970s. A combination of domestic inflation, rising labor costs and foreign cartels drove the cost of energy and its related technology up at unprecedented rates. Imported oil skyrocketed from under $3/barrel to $12/barrel overnight in 1973 and continues up at its present cost of $30/barrel. Gas and coal prices along with domestic supplies have followed this price rise with fuel costs increasing 250% between 1965 and 1975.[6] Electric power stations experienced similar cost escalation in the same period of time for a variety of reasons.[7] The increased cost of energy tended to fuel domestic inflation which resulted in higher labor and materials cost. More importantly, the utilities began to experience a credit squeeze. In years past, the utilities were able to generate much of the capital within house, ocassionally resorting to public money markets which were eager to buy the AAA rated utility bonds. This ease of financing became a thing of the past as the cost of electricity rose. Much of the frustration and anger of the public over the escalating energy costs was brought to bear on the utilities. Rightly or not, they were blamed in part for the suffering of the consumer, causing the Public Utilities Commissions throughout the country to assume a different posture. In the past the PUCs were generally a rubber stamp review board, understandable in light of the utilities outstanding successes in providing cheap reliable electric power. For the first time the PUCs experienced the outrage of a distressed populace, whose protection they were mandated to oversee. At this very critical time for the utilities, the PUC was forced to adopt a scrutinizing stance. Rate increases needed by the utilities to cover costs were slow in coming or refused . As production costs for utilties escalated at a rapid pace, capital funds for plant expansion became a major problem. For the first time, the lag between submission of utility requests for higher rates and regulatory approval became significant. Utilities were often caught selling at lower profit margins because of this lag. The resulting drop in revenues affected acquisition of capital funds by lowering the performances of utility stocks and bonds.[8]

The emergence of an environmental awareness on the part of consumers added another element to the utilities problems. Power plant siting, espceially nuclear plants, required increasing approval time. Questions of health and safety coupled to consumers concern about adverse property value impact is now causing lead times in excess of 12 years.[9] This long lead time makes it necessary for the utility to rely on exceptionally long range forecasting, precisely at a time when consumption patterns are increasingly hard to discern. In this situation, utilities are unwilling or unable (due to the capital markets) to embark on continued expansion of their large baseline plants (especially nuclear). Instead, they are filling the gap with fuel intensive gas turbines which are relatively inexpensive to build and have a very short lead time. Unfortunately, these plants rely on fuel oil and natural gas which merely aggravates an already severely stressed fuel supply.

It is becoming obvious that fundamental changes in utility operation and marketing must be undertaken in order to restore the industry to its former healthy state. However, this must not be done at the expense of the consumer, the environment, and future generations. It seems apparent that the conventional rate structures have served their purpose - perhaps too well. The existing rates are designed in a manner that encourages the growth of electrical use over time.These rates were established in a historical context reflecting decades of energy growth with declining costs and abundant fuel supplies. That era is over and along with it the usefullness of the rates which supported it. New rates which reflect the true cost of supplying energy are desperately needed. In conjunction with the new rate, new load management tools are needed to make more efficient use of our existing facilities. In order to discuss this in more detail, it is important to understand the basic principles and practices of a modern electric utility.

ENERGY USE IN AMERICA

As a large, affluent nation we have become ravenous consumers of energy. In order to satisfy this appetite we consume the energy equivalent of 1 1/2 billion gallons of oil per day, of which we import over 20% of our energy needs. This breaks down to 7 gallons of oil equivalent per day or 2500 gallons per year for every person in the country. Obviously, this energy supply consists not only of oil, but of a variety of fuels (Fig. 3).[10] However, we have put ourselves in the unenviable position where half of our energy needs are supplied by increasingly expensive oil. Coal supplies 18% of our total energy needs and 44% of our electrical requirements. Hydroelectric and nuclear power plants each contribute 13% to the electric power supply , and along with coal are mainly used as baseload electric capacity for this nation. Petroleum and natural gas add another 17% and 14% to the electric supply picture predominately as intermediate and peaking power plants. It is interesting to note that only 30% of the energy embodied in the various fuels used to fire the power plants ends up doing work at the consumers side of the power line. The rest of the energy is lost to the environment in the

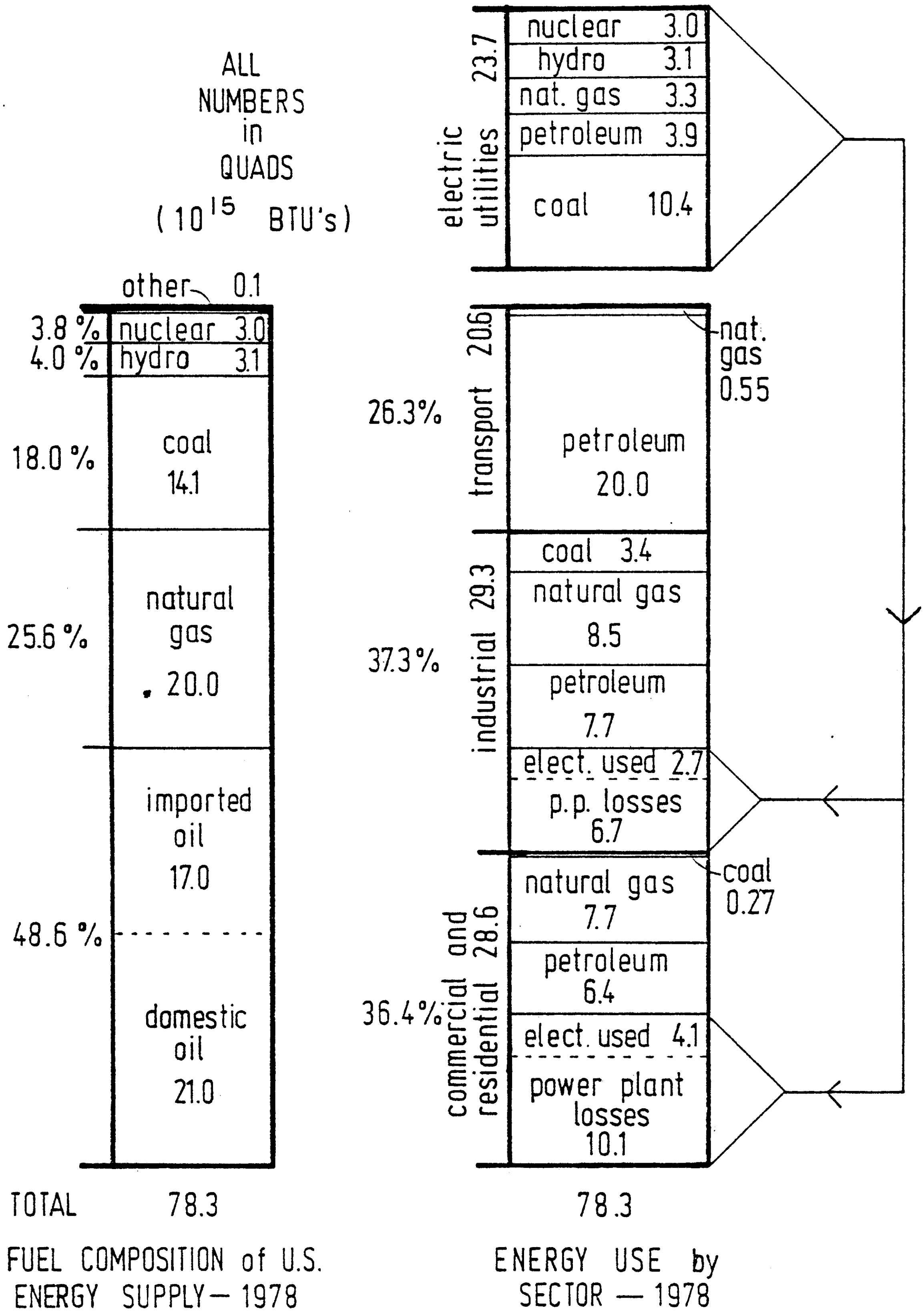
ALL NUMBERS in QUADS (10^{15} BTU's)
other 0.1
3.8 % nuclear 3.0
4.0 % hydro 3.1
18.0 % coal 14.1
25.6 % natural gas 20.0
imported oil 17.0
48.6 %
domestic oil 21.0
TOTAL 78.3
FUEL COMPOSITION of U.S. ENERGY SUPPLY — 1978
electric utilities 23.7
nuclear 3.0
hydro 3.1
nat. gas 3.3
petroleum 3.9
coal 10.4
transport 20.6
26.3%
nat. gas 0.55
petroleum 20.0
industrial 29.3
37.3%
coal 3.4
natural gas 8.5
petroleum 7.7
elect. used 2.7
p.p. losses 6.7
commercial and residential 28.6
36.4%
coal 0.27
natural gas 7.7
petroleum 6.4
elect. used 4.1
power plant losses 10.1
78.3
ENERGY USE by SECTOR — 1978

FIG 3 [10]

transmission lines and at the power plant as low grade heat, an inevitable consequence of the laws of physics. This does not mean to say that the low grade heat could not be used locally. Throughout Western Europe, many small power plants ship their waste heat to homes and industry located nearby. Due to the large scale of most of our power plants and the historically low cost of fuel, this was, for the most part, never done in a large scale way in this country. Thus, while the utilities consumed 30% of the nation's fuel supply in 1978, they contributed to only 9% of the end use energy demand.

POWER PLANT OPERATION

Generating capacity consists of a hierarchy of units from the most to least efficient. Within this hierarchy there may be a difference in operating efficiency of up to 60%.[11] Some large coal and nuclear fired power plants operate at a 35% efficiency while some small peak power gas turbines may be only 20% efficient. Unfortunately, these gas turbines are typically fueled with oil or natural gas. Fuel costs for peakers can run three times the cost of fossil fueled baseload plants and ten times the fuel costs for nuclear plants.[12]

The reason all the electric demand cannot be supplied by large efficient baseline power plants has to do with the nature of the utilities' load patterns throughout the day. Electric demand is not uniform; it is usually at a minimum between 11PM and 6AM and at a maximum between 10AM and 7PM. During the remaining hours, the demand is changing. The utility must be able to modulate the output from the power plants in accordance with the current demand. The minimum daily demand expected by the utility is known as the baseload. Demand from this baseload is satisfied by the large nuclear and coal power plants which are the cheapest to operate. Because of their size, the output from these plants cannot be easily modulated and they are usually run at their full rated output. As demand during the day rises, the plants which are more expensive to operate are brought on line, and referred to as the intermediate plants. To satisfy the peak daily loads which may only last 3 to 4 hours, smaller units, typically gas turbines or old inefficient oil boilers are brought on line as the "peakers". This hierarchy is established in such a way as to impose the lowest overall costs on the utility.

A utility's operating efficiency is characterized by a number of different indicators described below.[13]

LOAD FACTOR is the ratio of average demand to peak demand for a given time period, usually a day or year. This ratio can be calculated for either individual customers or for an entire utility system. Yearly load factors are an indicator of system utilization.

CAPACITY FACTOR is the ratio, for a given time period, of a system's total output in KWH to the system's maximum potential output (all generators running full time). This is a better indicator of system utilization because it reveals the magnitude of reserve equipment, forced outages, and scheduled maintenance.

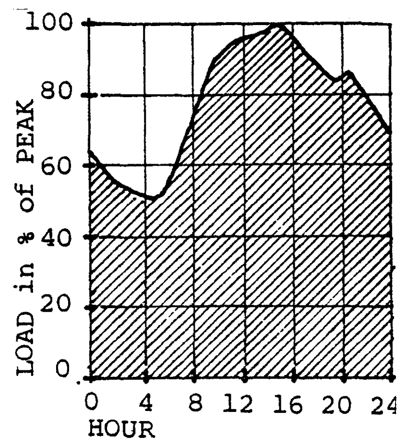

FIGURE 4

DAILY LOAD CURVE is a graphical representation of electrical demsnd vs. time over a 24 hour cycle. This demand may be the needs of a customer or the load placed on a utility system. The highest point on the graph represents the time and magnitude of the day's peak load. The illustration represents a summer peak day for Boston Edison.[14]

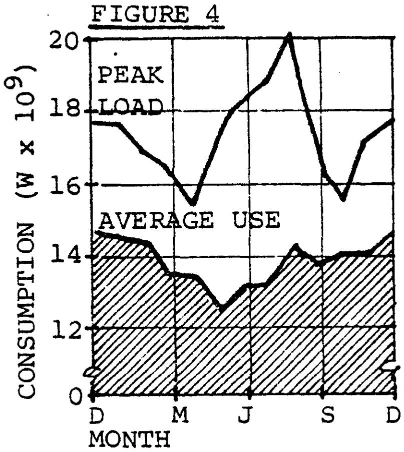

FIGURE 5

SEASONAL LOAD CURVE is a graph showing demand vs. time on a yearly basis. Daily peak loads and average loads are plotted. Graph peak represents the day and magnitude of the annual peak load. The illustration represents 1978 data for Boston Edison.[15]

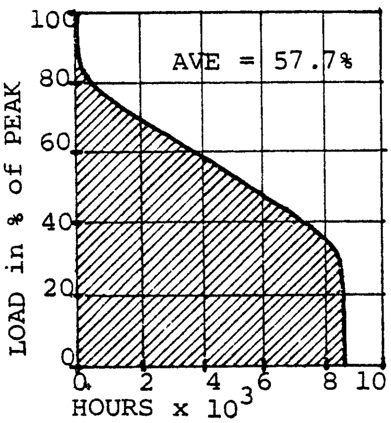

FIGURE 6

LOAD DURATION CURVE portrays the same information contained in the previous load curves except for the timing of demand. Demand values are plotted in descending order with 24 hour values for a yearly curve. This graph is normally constructed for utility systems and is useful for visualizing the character of peakload vs. baseload. This curve represents Boston Edison data in 1978.[16]

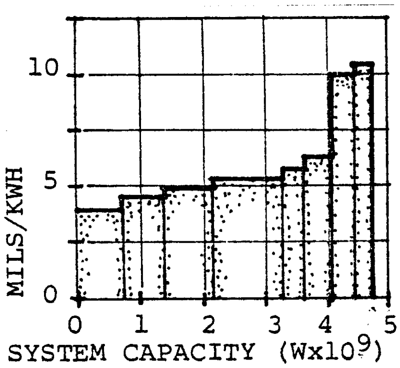

FIGURE 7

ORDER OF MERIT is a listing of power stations in an electrical system in the order of running costs. The cheapest stations are run as baseload stations, the most expensive only for infrequent peak loads, and the rest in between according to their merit order. This figure illustrates the order of merit for the Philadelphia area.[17]

(Fig. 4-7 courtesy C. Benton)

Utility load factors in the United States have been dropping since 1960 primarily due to the widespread acceptance of air conditioning. The national average load factor in 1973 was 62%, while in Europe it was 75%.[18] Although much of this difference can be accounted for in weather and national industrial differences, the Europeans long ago began to introduce load leveling incentives to customers. For winter peaking systems, as in Europe, the sophistication of residential heating systems significantly affects load factors. The Europeans have spent a considerable amount of money on thermal heat storage devices designed to utilize off-peak electricity.

In 1974, an average of less than 1/2 of installed U.S. capacity was utilized. To meet peak loads, more than twice the capacity needed to meet average demand was required.[19]

As former energy administrator Frank Zarb once said, "This nation cannot afford to continue building 100% excess capacity just to handle peak loads." Zarb went on to promote the following goals.

1. Improve load factor from 62% to 69%.

2. Improve capacity factor from 49% to 57%.

3. Expand baseload contribution to total capacity from 45% to 55%.

4. Increase end use efficiency by 10% (mainly by conservation).

It was claimed that these goals, if achieved, would reduce oil imports by 1.3 million barrels of oil per day.[20]

It seems that everyone is in agreement about the need to improve on load and capacity factors in order to make more efficient use of our energy resources and reduce our imported oil requirements for the "peakers". Disagreement arises as to what is the quickest, cheapest, and most equitable way of accomplishing the goal.

LOAD MANAGEMENT STRATEGIES

Historically, utilities have sought to balance their loads by bringing on more customers during the off-peak hours; load management through an increase in consumption. While this made sense in the past, it no longer is a sound policy due to the inability to bring on much more baseline capacity to handle these loads. The current strategy involves reducing the peak to approach the average demand. This load management may be of a direct or indirect nature as described below.

1. <u>Direct Load Management</u>

This management category leaves control of the loads with the utilities. Examples of this already exist. Many industrial uses of natural gas have interruptable service whereby the gas utility can cut off supply when demand from non-interruptable users exceeds a certain level. The same thing could be done with customers using electricity. Obviously, the customer would have to be compensated for this by enjoying reduced rates. In residences during the 50s, some utilities experimented with hot water tanks hooked to timers which would cut off the tank's electric supply during certain hours of the day. A more advanced form of direct load control involves the use of a signal transmitted over the power lines. This "ripple control" signal would decouple the user's load from the power grid for whatever period of time the utility deemed necessary. Experimental ripple control systems are currently being tested.

2. Indirect Load Management

The simplest form of this would be educational efforts by trained utility personnel directed towards large electrical customers. The utility people would try to explain how the customer could help manage their own loads in-house. The same would be done with home owners with educational packets included with the electric bill. This would probably not prove very effective unless it were backed up by some economic incentives for the customer. This incentive would most easily be embodied in the rate structure.

Presently, most rate structures for commercial and industrial users are 2 tier systems. There is a charge for the maximum load demanded during the billing period (demand charge) and another charge for the total energy consumed (energy charge). The more a customer consumes, the less he pays per energy unit, commonly referred to as a declining block rate structure. There is an implicit demand charge built into the energy charge such that if the customer's load factor is poor, his consumption is shifted into a more expensive block rate. This type of rate structure penalizes a customer for a poor load factor regardless of the relationship between the user's and the utility's daily load curve. A poor load factor for demand may be preferrable to a load factor of 100% as long as it improves the system's load factor. The existing rate structure has never borne more than a casual relationship to system costs since it is not keyed to the most important determinant of cost - the system's load factor. "A major fault with current rate structures is that they ignore peak load costs.... We recommend that cosideration be given to peak load pricing as a way to relieve some of the financial and operating stress on the system and to insure that the incidence of costs falls on the appropriate user."[21]

Peak load or "time of day" pricing schedules have been drawn up by a number of utilities throughout the nation. Energy consumption during a utility's peaking hours is discouraged by charging more for each KWH consumed. A simple non-utility example of this is the familiar rate structure of the phone system. Time of day rates would allow customers flexibility of choice that is not present in the direct load management designs. It is imperative that the off-peak rate schedules do not merely lower off-peak charges. This would simply lower the overall cost of electricity and encourage increased consumption, necessitating the construction of more power plants. The peak hour charges must also be raised so that the growth in power demand remains manageable. Off-peak electric energy can be sold at close to marginal cost,affording users the economic incentive to substitute electricity for consumption previously requiring oil and natural gas.[22]

The impact of time of day pricing would be felt throughout the economy. Where feasible, use would be shifted to the off-peak hours by changing habits and lifestyles. However, it is doubtful that lifestyle changes would go so far as to shift a significant portion of the work force to nighttime employment. This poses a serious problem for the business manager who is constantly trying to reduce costs. In office buildings, for example, the lighting, space conditioning and appliance power must be provided during the hours of employment, typically 8AM-6PM. This is precisely the same time that the utilities experience their peak loads. There is no economic alternative to operating the lighting and electrical equipment during the day as it is required. However, there exist a number of ways of storing "coolness" overnight so that the chillers may be shut off during the daytime hours. Since air conditioning represents 40%-50% of a building's power demand during a summer day, there is tremendous incentive to use such a storage system.

Traditionally, a cool storage system consists of water or rock which is cooled by chillers during the off-peak hours. During the peak hours,the chillers are shut down, and the building's heat gains are absorbed by the cool storage mass, thereby maintaining a comfortable environment. The consumer benefits derived from the use of cool storage are not totally dependant on time of day rates. Under existing rate structures, a customer still accrues substantial benefit by improving his individual load profile. By operating the chiller during nighttime hours, a customer can remove the chiller component from his daytime peak with a corresponding reduction in demand charges. In addition, a customer will improve chiller efficiency by operating the equipment at night, allowing the system to reject heat to more receptive, cooler atmospheric conditions.[23] Cool storage systems would be desirable even if the utilities adopted a direct load management strategy. In this case, the utility would decide when the storage system should be charged and discharged, rather than the customer.

There has been some discussion as to where off-peak storage systems should be located - on the customer's side or the utility's side of the power line. To date, the only economical utility side storage system is the pumped storage facilities around the country. Due to environmental pressure, and a lack of suitable sites, it appears that pumped storage facilities will not provide the quantity of storage capacity that is required for large scale full load off-peak power generation. Other centralized utility storage schemes such as battery and compressed air storage are still uneconomical.

When cooling or heating is the ultimate use of the electrical energy, thermal energy storage (TES) systems on the customer's side can provide a numner of advantages over utility side storage. 90%-95% of the energy stored in a customer's TES system can be recovered during cycles. Pumped storage facilities only recover 65% of the energy. The utilitiy transmission facilities are reduced with a TES system, and the geological and environmental constraints of customer side TES are non-existant. Depending on the length of the off-peak period, TES systems have the potential of reducing the cooling equipment capacity requirements. Since the cooling system is operated at a full constant load, the system does not have to be sized for the once or twice a year cooling peak, thereby realizing both capital and operating savings.

Cooling loads have been responsible for the majority of the increase in peak electric demand in this country. With or without off-peak time of day rates, tremendous savings in energy and money would be gained by the use of off-peak cooling systems. The technology for these thermal storage systems already exists; it is the reduction of size and cost that poses the real challenge. It is this challenge that will be addressed in the following chapters.

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21. "Should Utility Rates be Redesigned?", EPRI Journal, March 1976, p. 13.

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COOL STORAGE SYSTEMS

Large commercial and institutional office space usually consists of a combination of interior and perimeter floor area; the latter being defined as any floor area influenced by external weather conditions (usually floor area within 15 ft of the weather wall) with the interior comprising the remaining space. Because it depends on size and shape, the percentage of interior vs. perimeter varies widely among buildings. Interior spaces require cooling every day of the year while perimeter zones may experience both heating and cooling loads the same day. For this reason, the mechanical systems which heat and cool these areas may be separate and distinct. Any energy analysis of a 757,00 ft^2 commercial building in Boston[1] showed a design load condition of 25 $BTU/hrft^2$ for the interior and perimeter combined (Table 1). This building had undergone extensive retrofitting of mechanical and electrical systems in order to reduce energy consumption wherever it was shown to provide a return on the investment.

The ventilation load (8.5 $BTU/hrft^2$) could be reduced by an additional 50% merely by satisfying the Massachusetts Building Code minimum air requirements of 10 CFM/person assuming 1 worker/100 ft^2. If the National Bureau of Standards suggested codes were implemented this load would be reduced 50% further (5 CFM/person).[2] The electrical loads, 8.25 $BTU/hrft^2$ (2.4 w/ft^2) is generated primarily by the fluorescent lighting equipment in the building. This lighting load is quite low compared to loads encountered in older buildings. However, in discussion with lighting consultants[3], a lighting target of 1 $watt/ft^2$ is feasible with the proper lighting hardware and an average indoor lighting level of 30 foot candles. The solar load indicated in Table 1 is quite low due to the small percentage of overall glazing, especially

TABLE 1.

HEAT GAIN SUMMARY

BOSTON COMMERCIAL STRUCTURE (757,000 FT^2)

HEAT GAIN COMPONENT		COOLING LOAD ($BTU/HRFT^2$)*	
		Present	Theoretical
Conductance		3.12	3.12
** Occupants (sensible)		1.80	1.80
** Occupants (latent)		1.40	1.40
Ventilation		8.50	2.12
** Electrical		8.25	3.60
Solar		2.00	2.00
	TOTAL	25.17	14.04

NOTE:
* Design conditions, 88°F,DB, 71°F, WB
** Constant loads

on the southern exposure. To assume further reductions in this load would be unrealistic. Heat gain through the building envelope occurs primarily through the glass. Since the amount of glazing used in a building is primarily an architectural question, the size of the heat gain from this component will vary considerably from building to building. For this reason, no adjustment will be made to this figure. Nevertheless, it is imperative that designers understand the impact of fenestration on a building's cooling loads.

All of this heat that is generated within the structure must be removed by the building's air conditioning system. The system must be capable of sensibly cooling the air, dehumidifying it by removing latent heat, and cleaning the air with filters or air washers. The choice of mechanical systems used to condition air typically depends on the size of the building, its heating/cooling distribution system and the price and availability of local energy suppliers.

AIR CONDITIONING SYSTEMS

In residences, air to air cooling systems are usually the most economical approach for both centralized and decentralized (window) units (Fig. 8). The short refrigerant lines make it possible to reject heat directly to the environment in an air cooled condenser. Likewise, after passing through the expansion valve, the refrigerant cools the house air in a direct expansion (dx) evaporator in the air handling unit.

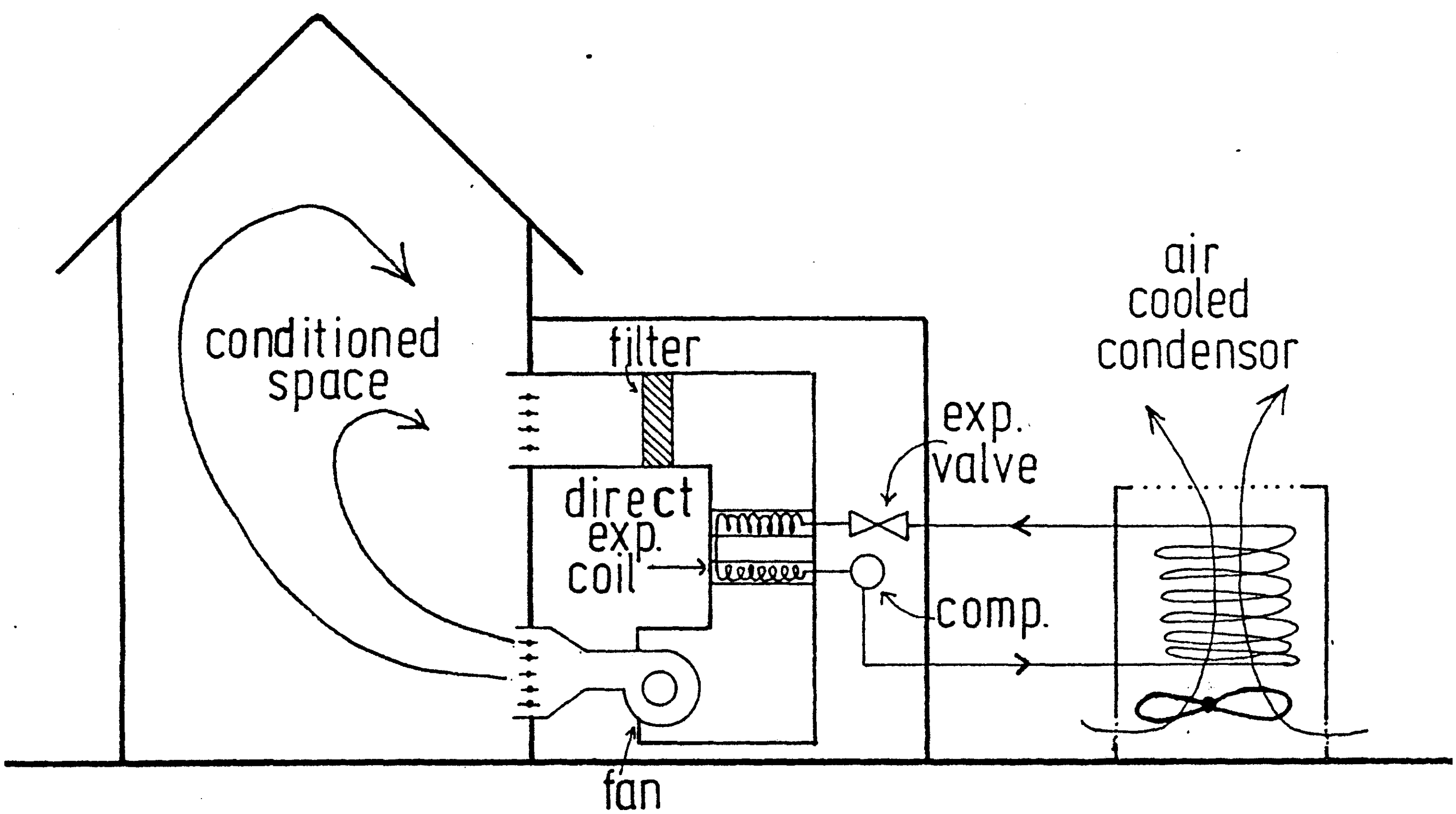


FIG 8

RESIDENTIAL COOLING SYSTEM

This investigation is primarily concerned with larger buildings where more complex refrigeration systems are required. It is not practical nor economical to ship liquid refrigerant throughout the building to decentralized dx evaporators. Instead, an intermediate heat transfer fluid, usually water, is used. Water is more economically distributed to the air handling units in the building where the dx evaporators are replaced with cooling coils supplied with 45-50°F chilled water. Water temperature in this range is needed to dehumidify warm, moist, incoming air. In order to lower the water content of the air its temperature must be dropped below its dew point. Under most situations 50°F chilled water will provide a sufficiently cold surface to insure adequate dehumidification of the air under summertime conditions.

On the other end of the chiller, the heat absorbed by the vapor refrigerant must be rejected. Once again it is more economical to transfer heat to a cooling tower with water rather than with a refrigerant. In this situation, a water cooled shell and tube condenser is used in conjunction with a cooling tower. This type of air conditioning system is shown diagramatically in Fig. 9.

The energy used to run the compressor and water pumps is typically 25-30% of the actual cooling provided. The relationship between work-out <u>vs</u>. work-in is commonly referred to as the system's Coefficient of Performance (COP) and is equal to (heat absorbed by evaporator)÷(work input to the compressor). COPs for the chillers alone are often times on the order of 3-4 during the summer operation. However, when the power used to run the pumps and air distribution system is included, the COP usually drops to 2-3. The amount of energy consumed by the compressor increases as the ΔT between the condenser and evaporator increases and/or as the absolute temperature of <u>both</u> decreases. This can be seen by solving the following equation[4] for a variety of operating conditions.

$$COP_{theoretical} = \frac{1}{\left(\frac{T_o}{T_r} - 1\right)} \qquad (1)$$

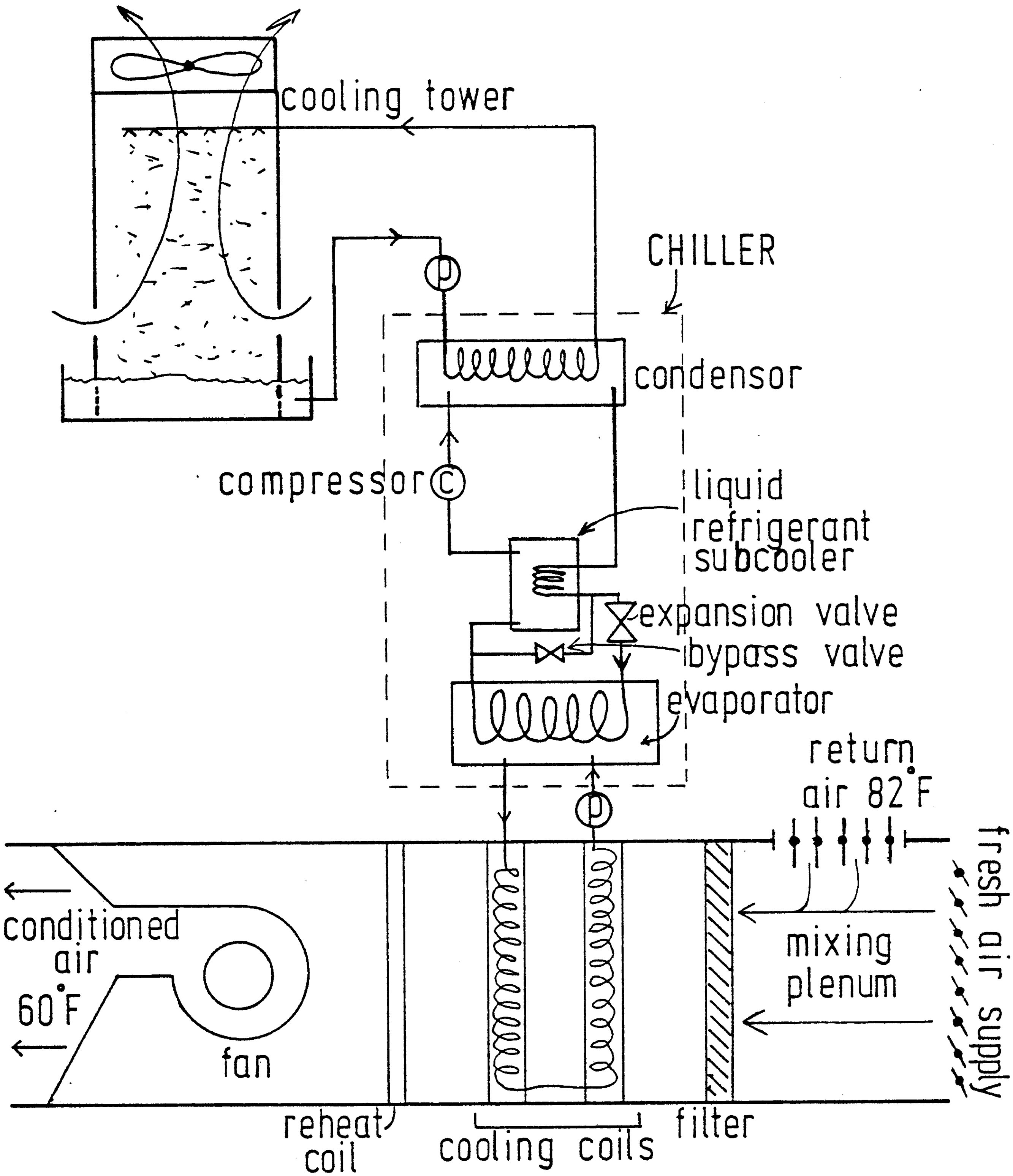


FIG 9

COMMERCIAL COOLING SYSTEM

where

T_o = average evaporator temperature (°K)

T_r = average condenser temperature (°K)

TABLE 2.

IMPACT OF WORKING TEMPERATURES ON COP

Case	Operating Conditions					COP	
	T_o		T_r		Lift	Theoretical	Manuf.Specs.[5]
	(°F)	(°K)	(°F)	(°K)	(°F)		
1	40	277.4	108	351.2	68	8.33	3.83
2	40	277.4	93	306.8	53	10.40	4.78
3	25	269.1	93	306.8	68	8.14	3.75

The first case represents typical summer, daytime operating conditions for a chiller producing 50°F chilled water. Case 2 is the same situation, except at night. In case 3 the chiller is producing ice at night during off-peak hours. These examples indicate that a chiller's COP is most sensitive to the temperature difference between the evaporator and condenser (commonly referred to as the "lift") and is relatively insensitive to the absolute temperatures for a given lift. However, the difference between the theoretical COP and a real chiller's COP is substantial. Departure from the ideal condition of negligible temperature difference during heat transfer and reversible processes of the reversed Carnot cycle require that the work input to the refrigeration cycle be greater than the ideal minimum. In addition, energy used by the pumps and fans associated with the rest of the air conditioning system reduces the system's overall COP even further than indicated in the above examples.

Nevertheless, the most revealing aspect of this exercise shows that the evaporator temperature can be lowered 15-20%F below normal and, as long as the condenser temperature is lowered by the same amount (resulting in the same lift), the chillers consumption of power per unit of output will remain roughly the same. Referring to the weather statistics in ASHRAE,[6] the mean daily summer range of temperature swing for Massachusetts is about 20°F. However, the average nighttime temperature is only about 14°F below the average daytime temperature. Therefore, assuming a normal chiller evaporator temperature of 40°F for daytime operation the evaporator temperature can be lowered to about 26°F without experiencing a drop in the system's COP.

From this analysis the temperature for storing heat from cooling systems ranges from 25°F to 50°F. The upper limit is set by the fact that chilled water must be supplied at a maximum temperature of about 50°F.

COOL STORAGE CONCEPT

Conceptually, a "cool storage" system should enable the building operator to run the cooling system at night and draw heat from a thermal mass thereby taking advantage of the off-peak electric rates while also reducing the building's electric demand charges. The storage mass must be capable of releasing its stored heat within the time constraints of the off-peak period now set by the utility as 12 hrs (8PM-8AM). It is anticipated from discussions with utility personnel that this off-peak period will be shortened to 10 hours. Conversely, the thermal mass must also be capable of absorbing peak cooling loads that are generated in the building throughout the day. This can potentially pose difficulties. The thermal mass is charged at a constant rate at night for perhaps 10 hours. During the day cooling loads are variable and, depending on the particular building, may peak out at rates 50-100% greater than the constant charging rate at night. The system must be capable of absorbing this heat if human comfort is to be maintained.

For cool storage systems considered in this thesis, energy can be stored as "latent" or "sensible" heat. When a material absorbs heat and rises in temperature without undergoing a change in state, the substance is said to have been heated "sensibly". For example, in raising the temperature of water from 40-50°F, every pound of water must absorb 10 BTU of sensible heat. Its specific heat is therefore 1 BTU/lb°F. When a change of state from solid to liquid or liquid to gas occurs, the material is said to absorb "latent" heat. For water to change from 32°F ice to 32°F liquid it must absorb 144 BTU/lb of material undergoing phase-change. Its heat of fusion is commonly expressed as 144 BTU/lbm. Due to the associated problems of pressure and volume changes with liquid to gas latent heat storage systems, discussion will be limited to liquid phase change materials (PCM).

SENSIBLE HEAT SYSTEMS

In sensible heat storage systems attractive materials are characterized by a high specific heat, high density and low cost. Referring to Table 3 it is obvious that water is by far the most logical choice

TABLE 3. THERMAL CHARACTERISTICS OF SENSIBLE HEAT STORAGE MATERIALS

Material	Specific Heat BTU/lb°F	Density lb/ft^3	Heat Capacity BTU/ft^3°F	¢/lb	Cost $/1000 BTU/°F
Brick*	0.2	123	24.6	1.0	50
Iron*	0.12	450	54.0	11.4	950
Lead*	0.031	707	21.9	42.0	13,548
Sand	0.191	95	18.0	0.26	13.6
Steel*	0.12	489	58.6	4.7	392
Stone	0.2	95	19.0	0.3	15
Water	1.0	62.4	62.4	~0.0	$\ll 1$

Note: *Scrap

for a centralized sensible heat thermal storage system. However, in a decentralized system where the thermal mass is distributed throughout the building, the situation changes. The mass may be already serving a structural function in which case its cost and use would already have been justified. The problem lies in designing the building and air distribution system in a way that will expose most of the surfaces to the chilled air at night and to the environment in which the heat is generated during the day.

A building for the[7] Department of Justice in Sacramento is being designed in this fashion. Chilled air is circulated throughout the building, cooling the structural slab 4-8°F. The daytime heat gains are absorbed by the slab and help to buffer the building's temperature swing. This system is limited by the heat transfer rate between the air stream and the concrete surface, and by the surface area of structure available for storage. Also, in order to maintain sufficient heat flow, the concrete surfaces must remain exposed, and not covered with floor rugs or acoustic ceiling tiles which will act as thermal insulators. In addition, if the surfaces are cooled too low, the room's inhabitants will be quite uncomfortable; therefore, the ΔT is limited. Concrete (and brick) are not exceptionally good thermal conductors, which may cause problems in fully charging or discharging the thermal mass. Finally, high rise towers would be heavily penalized in terms of structural costs for such a system. It should also be mentioned that this does not lend itself well to retrofits of existing buildings. Nevertheless, future buildings using concrete as structural mass would undoubtedly be able to realize significant savings in energy by designing from the start with this approach in mind.

Using water for sensible heat storage is experiencing an increasing amount of exposure and use in actual buildings. Early chilled water storage systems were relatively simple to design, install and operate. The primary cost for such systems was the tank itself. Conventional chiller equipment was used to cool the water and integration with the chilled water distribution system was fairly simple. However, chilled water storage systems have historically suffered from 2 problems. First, the sheer size of the tank is a major handicap and cost, and secondly, the water temperature supply to the air handling units is not constant due to the mixing between cold supply water and the warm returning water. This complicates the pumps and control systems which result in higher initial costs. Nevertheless, many systems have been installed throughout the country, primarily to reduce the building's power demand charge and reduce chiller size by operating the system constantly. In Minnesota, a 32,000 ft^2 multi-story office building is cooled with 45°F water stored in two 40,000 gallon, underground tanks.[8] This system lowers the building's peak load by 100 kw at an incremental construction cost of $115,000. Studies based on a similar system for a Toronto office building[9] indicated, in 1975 dollars, a cost of $270 per stored ton of cooling capacity for large systems and $392 per ton for small systems which include interconnecting hardware.

The ability to reduce the volumetric requirements of water storage systems is very limited. Given the specific heat of water (1 BTU/lb°F) the only other variable involved in determining tank size is the rise in temperature that the water can experience and still provide the required cooling. The lower water temperature limit is set at around 35°F, just above the freezing point (most systems designed to date do not go below 40°F). The upper limit is less defined and is dependent on the design of the tank and the heat exchangers in the air handling units. In well designed systems, the temperature rise can be stretched to 20°F (40°-60°F). This can be accomplished while still providing the required sensible and latent cooling of the air by employing a design such as the one shown conceptually in Fig. 10.[10,11] The engineers have designed a "reverse flow" heat exchanger system which allows tham to operate over this elevated temperature differential. In addition, the blending problem is eliminated by separating the tank into 2 zones with a "floating membrane" (patent held by R. Tamblyn, Engineering Interface, Ontario) that allows for the charging and discharging of the tank. In the past, mixing of the supply and return water was prevented by using multiple tanks and/or baffles. All these methods, to varying degrees, add to the complexity and cost of the system.

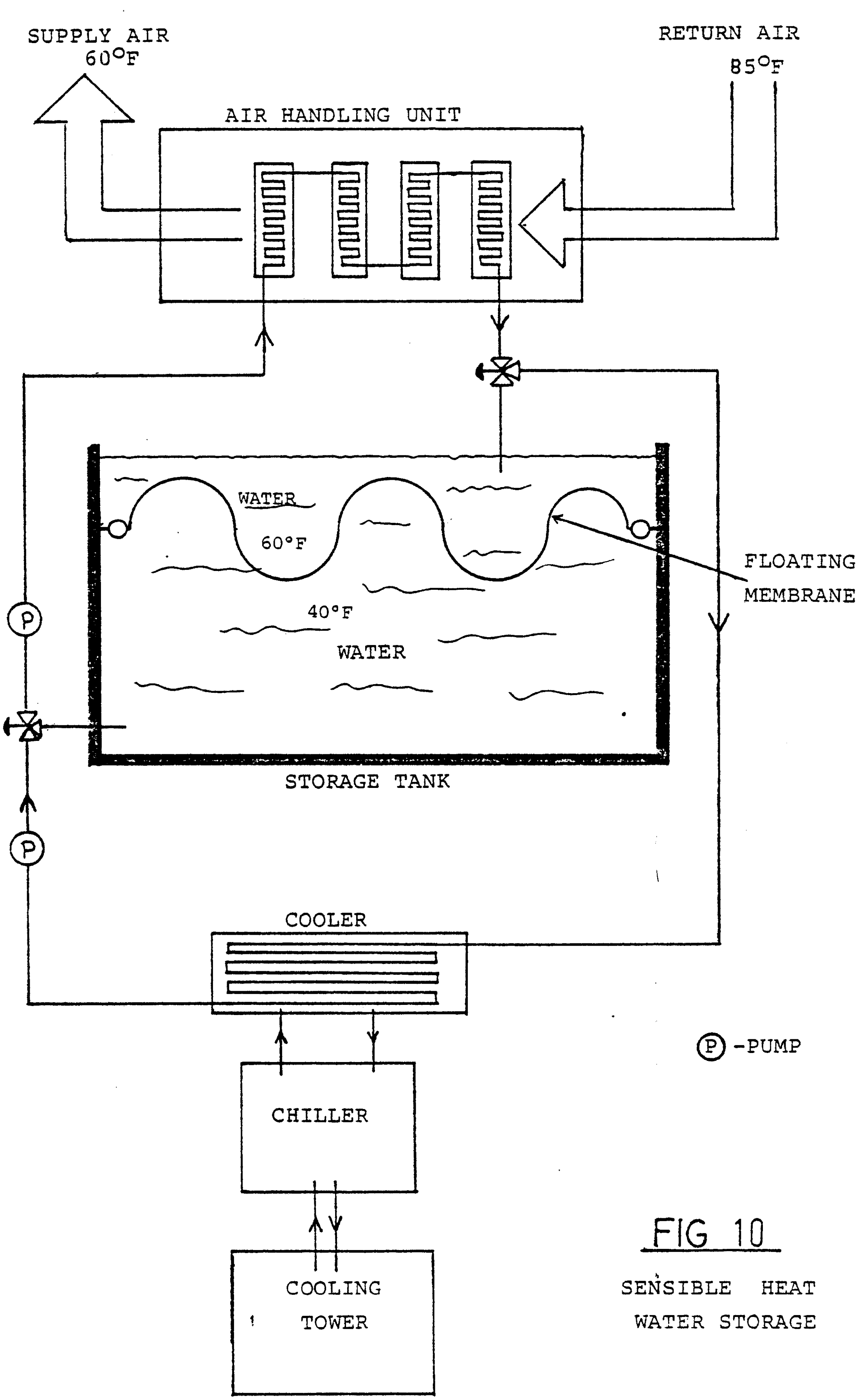


FIG 10

SENSIBLE HEAT
WATER STORAGE

In the New York building using the partitioned storage tank, the designers plan to operate the chilled water over a temperature range of 42°-60°F (Studies are underway to determine what penalties in performance will be experienced by chilling the water further.). The maximum cooling loads anticipated require 1,300,000 gallons of storage. No chillers are to be operated during the day, thereby taking full advantage of the off-peak rates. Preliminary cost estimates indicate a cost of $1.10/gallon of storage for tank construction costs which amounts to a total cost (storage only) of $1,430,000 ($0.74/BTU stored, $89/ton). In order to accommodate such a tank (174,000 ft^3 or equivalent to 18'x100'x100') a huge sub-basement area had to be added to the building plans. Excavation of it in New York bedrock undoubtedly added to the high $/gallon cost. Other figures for large storage tanks have ranged from $.40/gallon and up. The key to minimizing storage tank costs appear to be to include it in the sub-structural design from the start. In this way, tank walls and foundations can serve a dual purpose. An example of this was given by a competing New York design[12] in which the elevator shafts, which also will function as a structural core element, does triple duty as a water storage tank. Due to its function as a structural core, the elevator shafts will continue down to unoccupied depths. This "free" space will then be filled with water and chilled during off-peak hours. The floating membrane is used once again to prevent mixing of the warm and cold water.

All of these large water storage tanks are non-pressurized for economic and safety reasons, while the chilled water systems that they couple with are pressurized. For this reason, a heat exchanger is commonly used to join the two systems. However, heat exchangers require a small temperature differential in order to move heat (5°-10°F). Given the already limited temperature differential available to cool water storage systems (20°F max.) the ability to avoid this taxing heat exchanger is much desired. An interesting proposal by R. Tamblyn[13] calls for the placement of the storage tank on the top floor. In this fashion, the building's chilled water lines would be pressurized due to the gravitational head. In addition, the storage tank could also be used as a topside fire reservoir, reducing or eliminating the need for high pressure fire pumps and their emergency generators. Also, the tank could serve as an open expansion tank on top of the various piping systems. In many cases, topside storage may be prohibitively expensive. In order to avoid heat exchangers with basement storage, more elaborate and expensive systems using transfer or injection pumps with energy recovery turbines may be used.

In these more elaborate chilled water storage systems, the costs for plumbing hardware and controls start to mount. However, these systems are viable and apparently cost-effective for some buildings; especially when some of the tank expenses can be written off to foundation or structural costs. Nevertheless, a more compact system is highly desirable from a standpoint of both cost and the ability to retrofit existing buildings.

LATENT HEAT STORAGE

By taking advantage of the latent heat of fusion of a particular substance, a much higher volummetric heat storage efficiency can be realized. When a substance melts, it absorbs heat at a constant temperature until it has completely liquified, after which point the temperature rises. Depending on the material used, the amount of heat absorbed during phase-change is typically 50-200 times the amount absorbed as sensible heat per unit of temperature rise. Referring to the water storage systems where a temperature differential of approximately 25°F is feasible, it is possible to store almost 7 times as much heat in the same volume if both the sensible heat and latent heat of fusion are captured by cooling the water from 60°F to 31°F. However, the benefits of such systems do not come without a price. While the tank costs are greatly reduced, the heat exchanger required to charge and discharge the mass typically more than makes up for the price differential.

Potential phase-change materials (PCM) must satisfy a number of criteria before they can be considered suitable. No PCM satisfies all the criteria, but good ones will exhibit few of the problems discussed below.

1. Melting Point

The PCM should melt and fuse in a narrow range and at a suitable temperature. 45°F represents an upper temperature limit for providing chilled water at 50°F. The lower temperature limit is not as easily defined. Melt points much below 30°F are discouraged due to chiller performance penalties arising from the large lift between the evaporator and condensers. Freezing points are often depressed below the melting point due to impurities in the chemical. This freezing-melting band is especially a problem with the organics. This is distinct from problems with supercooling.

2. Heat of Fusion

The "heat of fusion" is the quantity of energy required to change 1 lb of material from a solid to a liquid. Most materials have a heat of fusion ranging from 20-150 BTU/lb.

3. Densities

Coupled to the heat of fusion, it is advantageous to have a high density. It is important to remember that the reason for using PCM is to economize on volume. Therefore, by multiplying the material's heat of fusion times its density, its volumetric heat capacity (BTU/ft^3) is indicated.

4. Congruent Melting[14]

In multi-component systems several phases are in equilibrium with one another at the melting point, and there is then a risk of spontaneous and irreversible phase separation caused by the different component densities. Species from a liquid phase and a solid phase are combined to form another solid phase in the boundary zone between the 2 other phases thus hindering the reacting species from reaching each other. As a result, the process becomes successively slower, and complete conversion seldom takes place. Only part of the latent heat stored can be used, and therefore the heat storage capacity is reduced. Congruent melting means that only one solid phase is involved, and that the composition of the solid and liquid phase is identical at the melting point. A congruently melting material can theoretically be cycled many times.

5. Thermal Conductivity

Ideally, as materials change phase, the temperature of the mass should remain uniform at the melt point until all the heat of fusion has been extracted. In reality, this is not the case. Due to the relatively low thermal conductivity of most PCM, the phase-change proceeds as a "front" across the material, originating at the heat exchanger. The velocity of this "front" is dependent on the conductivity and the temperature differential across the material. In a sense, most conventional PCM may be thought of as absorning large quantities of heat in a sensible manner over a low temperature differential. The impact of this is primarily manifested by the requirement of increasingly large heat exchanger and/or temperature gradients as the conductivity goes down. The first penalizes in the form of increased capital costs while the second in the form of higher operating costs (lower COP due to lower evaporator temperature).

6. Supercooling[15]

A substance that does not solidify at its melting point is exhibiting supercooling. Complicated structures tend to supercool more readily. This is due to the fact that the different species in the melt do not always diffuse together into the right crystal structure. Instead, they form so-called defect structures with a higher free energy and therefore a lower melting point. This is typical of eutectics and highly purified chemicals. Supercooling may amount to 20°-40°F in some cases. Once nucleation begins, the melt quickly rebounds to the melting point. In order to prevent supercooling, nucleating agents are employed. The simplest way to insure nucleation is to never completely melt the PCM. This could impose some expensive regulating costs on the system. A "cold finger" refrigeration device is sometimes used to freeze a fraction of the PCM and thereby initiate nucleation. Nucleating agents, with crystal structures similar to the PCM itself, are usually the cheapest and simplest solution to the problem. However, the nucleator must not be soluable in the PCM and must not chemically react with it.

7. Stability

The PCM must be able to withstand repeated cycling without decomposing. This is especially a problem with the organics which tend to oxidize and polymerize over time.

8. Inertness with Respect to Container

Many PCM are highly reactive with metals and/or plastics which are the two obvious container materials. Reactivity may lead to exotic containerization requirements which are usually quite costly.

9. Flammability and Toxicity

The use of combustible materials in commercial structures is severely restricted. The idea of placing 100,000 gallons of highly combustible fuel (i.e., paraffins) in the basement would no doubt be met with expensive sprinkler and fire detection equipment requirements. Some flammable materials can be made inflammable with compatible retardants. Toxic substances increase costs at the fabrication point. Workers handling and packing the materials must be protected against exposure and, after placed in the building, office workers must be shielded from ill effects in the event containment is ever breached.

10. Cost

As usual, it all comes down to the bottom line. Is the PCM and all the hardware required to insure efficient and safe operation worth the investment when compared to the alternatives (sensible heat water storage).

Reviews of the literature reveal a wide variety of PCM as potential candidates for a cool storage system. The prime consideration is that the chemicals' melt point lie within the 30°-45°F temperature range which is compatible with commercial chiller systems. Many of the candidates must immediately be dropped from consideration due to a variety of deficiencies which make them economically unfeasible. Table 4 lists 7 chemicals (or chemical groups) which are suitable, with limitations, for use as PCM. Thermophysical data, where available, is included. Prices quoted are from the respective distributors for car lot quantities as of December 1979. As detailed in the Appendix, the heat of fusion and melt points were measured here in the laboratory for water (as a reference), 2 paraffins (C_{14}-C_{16} and n-tetradecane) and deconal to insure that the published literature data was accurate. The eutectic mixtures were not tested due

TABLE 4. PHASE-CHANGE MATERIAL FOR COOL STORAGE SYSTEM**

tance	Melting Point	Form	BTU/lb	BTU/ft^3	Conductivity BTU/hrft°F Solid	Liq.	Flammability	Container Incompatibilit
ter	32(32)	H_2O	144(143)	8200(8150)	1.33	0.33	No	Unprotected steel
tradecane	38(<32)	$C_{14}H_{30}$	98	4600	0.083		Yes	Many plastic
ctic mix	39	31 Na_2SO_4 13 NaCl 16 KCl 40 H_2O	101*	8800*	-0.333		No	Most metals
hrate p	38-45	-	~110	-	-		Some in group	Most metals
ctic mix	41	80 $K_2HPO_4 \cdot 6H_2O$ 20 $Na_2HPO_4 \cdot 12H_2O$		5600	0.333			Most metals
ffin	42(46.5)	C_{14}-C_{16}	65.5(67)	3200(3275)	0.083		Yes	Many plastics
nal	42.8(45)	$C_{10}H_{22}O$	88.6(85)	4600(4400)	0.083		Yes	Many plastics

oretical; **For more information about PCM for cool storage systems, see Reference 30; ( ) Experimental data from tests conducted a to the time constraints which did not permit long term cycling experiments. In addition, the clathrate group was dropped from consideration after discussions with Dow Chemical Corporation revealed that studies undertaken by Dow show that the clathrates would cost in excess of $1.00 per pound to manufacture, even in carlot quantities. This is unfortunate, considering its high heat of fusion and congruent melting characteristics.

The samples obtained from the distributors were of industrial grade, as the technical grade, although more pure, was quite a bit more expensive and generally not available in the kinds of quantities required for commercialization. The tetradecane (from Humphrey Chemical Co., of Connecticut) was totally unsuitable. Attempts to freeze the material by cooling it to 32°F were futile. The sample of deconal (from Conoco in Saddlebrook, New Jersey) also exhibited a wide melt-freeze range. As explained in the Appendix, this was not supercooling, but rather a broadening of the melt-freeze band, apparently due to the presence of branched hydrocarbons. (Researchers from Penn. State who reported on deconal's performance[16] stated that they used scientific grade material rather than industrial grade.) Two samples of paraffins were obtained from Exxon (Baytown, Texas) and Conoco (Saddlebrook, New Jersey). The paraffins had quite narrow melt-freeze bands (1°-2°F) and exhibited no supercooling. The heat of fusion measured in the lab matched closely with the data in the literature. It should be noted that the organics tested all melt congruently, are quite stable and are quite compatible with metals and some plastics[17] (PVC, polypropylene). Their greatest drawback is their conductivity (1 BTU·in/hrft^2°F) and the high degree of flammability and typically low flash point. Results of the lab experiments limit the choice of PCM to the C_{14}-C_{16} paraffin, the eutectic salts and water. In order to fully appreciate why a further process of elimination leaves only water as a suitable PCM for use in conjunction with chiller systems, it is necessary to briefly review the research and development accomplished to date.

EXAMPLES OF LATENT HEAT COOL STORAGE SYSTEMS

Early attempts to overcome the difficulties associated with PCM centered on the eutectic salts, and in particular, hydrated forms of sodium sulphate (Na_2SO_4). This salt, in anhydrous form, costs only pennies per pound and when mixed with proper amounts of water and other salts (NaCl, NH_4Cl, KCl) melting points ranging from 40°-90°F (4°-32°C) can be obtained. The salt suffers form some serious drawbacks. Most importantly, it melts incongruently. In addition, it suffers from super-cooling and is corrosive to most metals. Theoretically, the salts should exhibit high heats of fusion (appeoximately 100 BTU/lb) but in reality, due to the chemical additives, the heats of fusion realized in practive are much lower (30-40 BTU/lb). Borax is commonly added to the mixture to prevent suprcooling, and various starches and gels have been used as thickening agents with variable success to prevent density induced separation of the liquid-solid phases.

The University of Delaware "Solar One" house[18,19] employed a cool storage system which consisted of a 216 ft^3 box filled with 6 ft by 1.25" 30 mil, plastic tubes spaced 3" on center. The PCM consisted of a mixture of Na_2SO_4, NaCl, NH_4Cl and water. Data from this experimental set-up shows the temperature of the air leaving the box as constantly rising, indicating a crystal build-up on the heat exchanger surface which tends to limit the heat flow into storage. The problems with incongruent melting have apparently not been eliminated as evidenced by declining heat storage ability of the unit over time. It should also be noted that only 14% of the storage volume was occupied by the PCM.

The General Electric Researcn Division has experimented with the same concept of the Delaware house with significant changes.[20] Horizontal cylinders containing PCM are equipped to rotate slowly to insure constant mixing which tends to break up the surface crystallization and incongruent melting problems. With laboratory models, researchers claim 100% crystallization, repeatable cycling, reliable nucleation, excellent heat transfer, and a high volumetric efficiency. The economics of a large scale set-up have yet to be demonstrated. Whether or not such a system can be designed with water (rather than air) as the heat transfer medium, is questionable. Both of these systems (University of Delaware and General Electric) are best suited for residential and small commercial applications where dx, air-air cooling systems are commonly used. With the sensible heat, cool water storage, 1250 BTU can be stored in a ft^3 of water (20°F ΔT, 1 BTU/lb°F, 62.4 lb/ft^3). Even when assuming a PCM to storage volume ratio of 1:2 (50% packing), only 1870 BTU/ft^3 can be stored using a proper mixture of Glaubers salts and additives and assuming 40 BTU/lb latent heat storage. These assumptions seem quite generous considering that the Delaware experiment had a 17% PCM to volume ratio and calorimeter tests on Glaubers salts with additives for cool storage application showed heats of fusion of 30-35 BTU/lb (tests conducted at MIT). If a cooling system using chilled water is specified, it makes no sense to apply the salt hydrate PCM when cool water storage is obviously simpler and cheaper.

Research at MIT by Timothy Johnson with the assistance of Cris Benton has produced a PCM which melts at 65°F and exhibits a heat of fusion equal to 35 BTU/lb. The melting point of Glauber salts ($Na_2SO_4 \cdot 10H_2O$) was lowered from 88°F by the addition of NH_4Cl as a eutectoid. Supercooling was minimized with Borax and stratification due to incongruent melting was solved in 2 ways. Fumed silica was added to the solution to retard component migration during the phase change. In addition, the salt mixture was packaged in 2 3/8" horizontal layers to minimize the gravitational pressure head while still allowing crystal growth by diffusion. This same combination of changes in the mixing and packaging of Glaubers salt is used in the MIT Solar 5 Building. The eutectoid in this case is NaCl and the melt point is 74°F. The mixture has undergone 4500 freeze-thaw cycles with minimal stratification and heat of fusion stabilized at 35 BTU/lb. The cooling mixture, with less testing, is behaving in a similar fashion.

The 10 lb salt bags are suspended on ceiling tiles in the room that is to be conditioned (See Fig. 11, reprinted with premission of author). The ceiling tiles must be made of a thermally conductive material ('U' approximately 15 $BTU/hrft^2°F$) so that the bags will remain in intimate thermal contact with the room. The bags are frozen overnight during off-peak hours with 55°F air which is circulated through the plenum. This air may be chilled with a refrigeration system or, when feasible, with outside air. During the day, heat gains from the space are absorbed by convective and radiative coupling between the space and the tiles. The room air must be dehumidified during working hours with a daytime chiller in order to prevent condensation on the ceiling system.

A 500 ft^2 test space was built, similar to Fig. 11. . A 1 1/2 ton chilller charged the salt bags for 12 hrs after which internal gains were simulated with electric resistance heaters. The behavior of the room air and PCM temperatures were monitored with results shown in Fig. 12. The temperature in the room went from 68°F in the morning to 79°F in late afternoon.[21]

This type of system would not be amenable to perimeter zone cooling due to the fact that within the course of a day perimeter offices may experience both a heating and cooling load. The "tug of war" that would result between the heating system and the cool ceiling would be unacceptable. Further tests were conducted to see if the perimeter zone air could be cooled by circulating it through the interior zone plenum containing the PCM. The results shown in Fig. 13 indicate that the amount of sensible cooling experienced by the perimeter air is insufficient to handle the anticipated loads. This is due to the "warm" temperature of the bags (65°F) and the low heat transfer from the air stream to the bags (experimentally found to be 1.7 BTU/hrft^2°F @ a velocity of 60 ft/min). In addition to the perimeter zone and dehumidification problems, the material composing the PCM tile will cause acoustic problems in the room. Due to the high conductivity requirement, the prototype material used is quite hard and dense. Conventional acoustic surface treatments would destroy the conductivity of the tile. Investigations on this problem are continuing. Nevertheless, this decentralized off-peak cooling system is one of the most novel to date. Due to the high melt point (65°F), the chiller COP's are the highest for any off-peak cooling scheme. In fact, for most of the year, the cool night air will be capable of charging the tile. Because the phase change material is located in the space to be conditioned, no fan or pumping power need be expended to transport the cooling medium

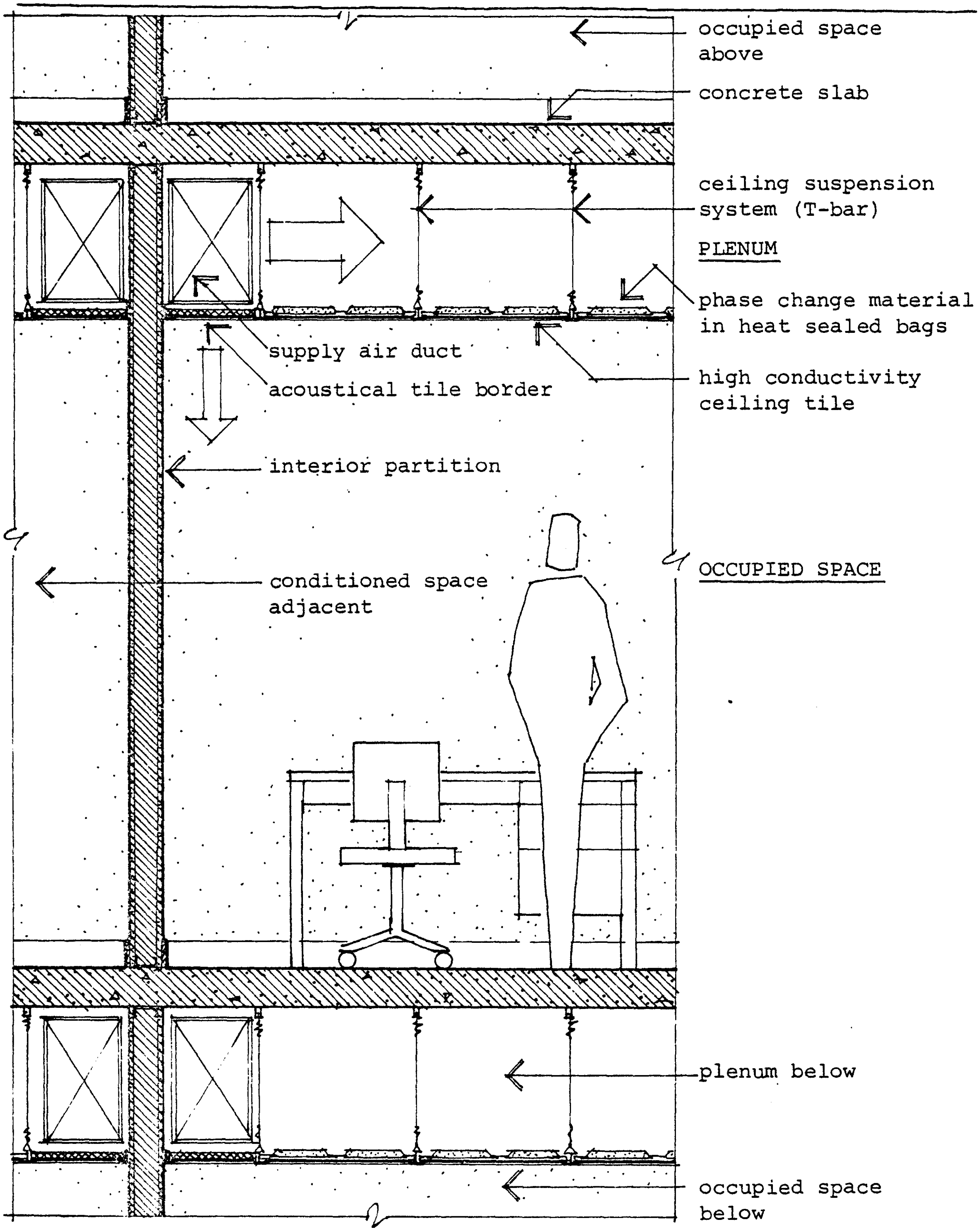


FIGURE 11 Vertical Section Through Interior Zone Space Showing System Components.

(courtesy C. Benton)

FIGURE NO. 12

Temperature profiles in test space for days 1 & 2

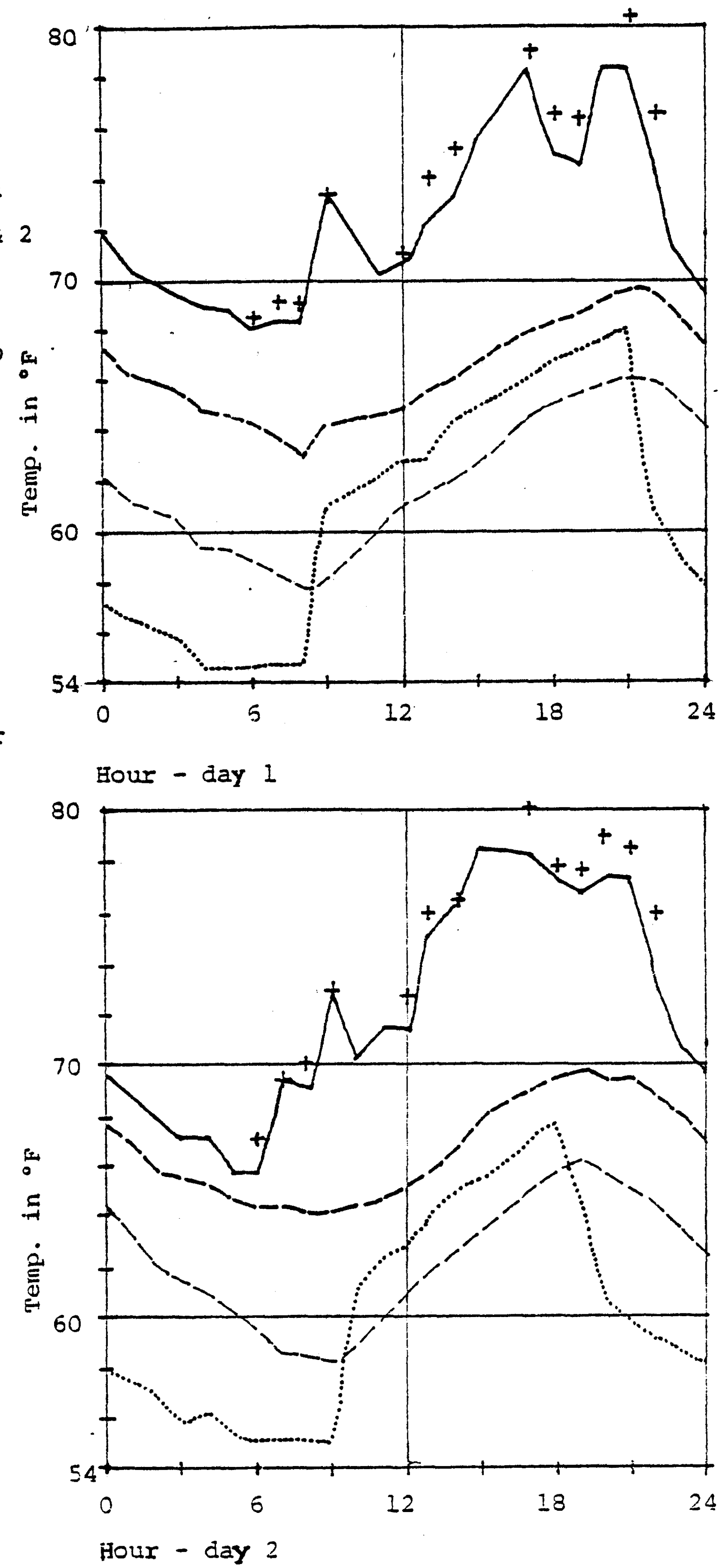
+ room air (drybulb thermometer)

——— room air (thermistor)

— — — PCM tile (center of room)

- - - - plaster ceiling

········ plenum air



(courtesy C. Benton)

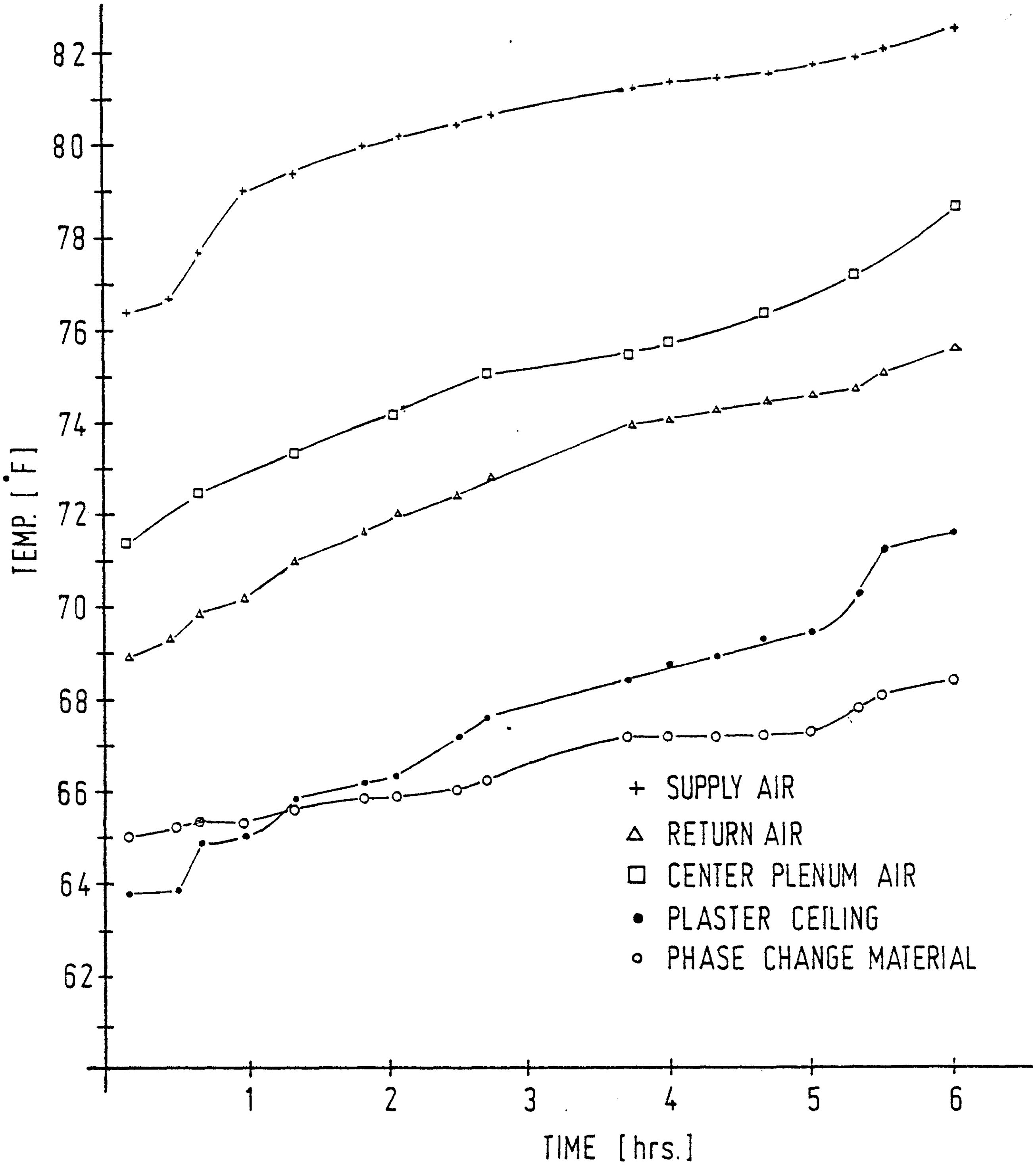


FIG 13

PERIMETER LOAD SIMULATION on the M.I.T. COOL STORAGE SYSTEM to the spaces during the day. Perhaps the most attractive feature of this scheme is the fact that it requires no additional building volume. Because the bags are placed throughout the building's plenum, no basement storage tank need be constructed. The remaining drawback to this system is cost. The tile-bag assembly costs about \$2.90 ft^2. Taking a \$.5/ft^2 credit for the acoustic tile it replaces, the net cost is \$2.40/ft^2. Assuming 5 lb of PCM per ft^2 and 35 BTU stored per lb, the assembly will store 175 BTU/ft^2. In addition, the surrounding concrete slab will provide an additional 25 BTU/ft^2 to yield a total 200 BTU/ft^2 of tile. The unit energy storage costs are 1.2¢/BTU stored. In order for this system to compete with the cool water storage systems, cost reductions will have to be realized.

Research on the use of organics for cool storage systems have received a fair amount of attention. A variety of independent investigations concluded that the paraffin hydrocarbons proved the most promising of the organic PCMs tested. Researchers at Penn. State University have extensively investigated a variety of PCM suitable for centralized off-peak cooling storage system. They concluded that the eutectic salt hydrates were not economical primarily due to the containerization requirements needed to prevent separation due to incongruent melting. They went on to recommend the Exxon paraffin (C_{14}-C_{16}). The major drawback to the paraffins are their poor thermal conductivity ($k = 1$ BTU·in/hrft^2°F) and the volumetric contraction on freezing (12-15%), which tends to pull the PCM away from the heat exchanger, causing a reduction in the system's heat transfer. Both problems make it necessary to increase the surface area of the heat exchanger.

E. Mehalick and A. Tweedie of General Electric's Space Division designed a cool storage system using paraffin as a PCM which overcame these difficulties.[22] The paraffin was micro-encapsulated with a flexible nylon wall material by Penwalt Corp. The encapsulation process is similar to the one used to encapsulate drugs and pesticides for time release action. The size of the capsules ranged from 50 to 2000 microns. The micro-capsules were suspended in water in the storage tank as a slurry. Maximum packing density was about 45 parts PCM to 55 parts water. Since the size of the individual particles was very small and suspended in water the heat transfer problem was effectively eliminated. In addition, the expansion and contraction of the PCM was accommodated for by the flexible microcapsule wall material. Leakage of the paraffin was undetectable after 300 cycles. However, leakage did occur when screens were placed in the tank to prevent the uptake of the micro-capsule. The capsules were ruptured when they accumulated against the screen. In addition, it was observed that the capsules were weakened and ruptured when exposed to most conventional pumping equipment. The impact of this on the system is significant. In order for the materials to be handled as a slurry and transported form manufacturing plant to site, pumping would be encountered a number of times. Stronger capsule materials are needed. In discussion with A. Tweedie, cost estimates obtained from Penwalt indicated a probable cost of $1.00/lb of encapsulated PCM ($1.42/BTU). This does not include storage tank and related hardware costs. Unless costs are reduced and encapsulation materials strengthened, this system will be unable to compete with the cool water storage systems.

A study conducted for NASA[23] in 1971 also identified the paraffins, as a group, as one of the more reliable and economic of the PCM they tested. Their approach to the conductivity problem involved packaging the PCM in a container with a metallic "sponge" filler which improves the heat transfer between the PCM and heat exchanger. Volume changes were accommodated for by either using container materials which could withstand the pressure changes or by using a prestressed "bellowed" container which would flex with the change of internal pressure. These measures never found widespread commercial application primarily due to cost constraints, something NASA was somewhat immune to 10 years ago.

The Annual Cycle Energy System (ACES) is an integrated space heating, cooling and domestic hot water system. As designed in 1976 its major elements are:

1. A high efficiency heat pump with refrigerant, to bring heat exchangers on both the evaporating and condensing side.
2. Thermal storage on the low temperature (evaporator) side.
3. An auxiliary heat source and sink.

4. A forced air circulating system with a fan coil for space heating and cooling.
5. A refrigerant to water heat exchanger for heating and a tank for storing domestic hot water.[24]

During the winter months of 1976-77 the heat pump extracted heat from the 18,000 gallon water tank via an intermediate brine circuit, and in the process formed ice around the 2000 ft of aluminum fin tubing spaced 13" on center which comprised the storage tank heat exchanger. Heat collected by rooftop solar collectors would melt ice when possible so as to maintain an adequate supply of PCM. As the winter progressed, the average evaporator temperatures fell due to the increasing ice cylinder, and the thermal resistance that is represented. Under average winter conditions and maximum ice thickness (6"), the brine to water ΔT was 9°F indicating an evaporator temperature below 20°F. The impact was that the COP of the heat pump in winter was often times lower than it would be if if had used the ambient air as a heat sink. (Knoxville winer design temperature = 17°F). Yet, this was necessary due to the requirement that full ice formation was needed in order to insure adequate cooling capacity in the summer. This system works best when both the winter and summer loads are about equal thereby taking maximum advantage of the system's capacity year round. In the summer, space cooling is accomplished by circulating the brine from the tank heat exchanger through the house's fan coil unit.

An interesting alternative to ACES for seasonal cooling only was proposed by T. Bligh in 1976 at the University of Minnesota.[25] Bligh designed a system which makes ice throughout the winter by circulating brine (methonal/water) through an externally mounted fin tube convector whenever ambient temperatures fell below freezing. The chilled brine was then circulated through 1/2" copper pipe spaced 18" on center in an 8000 gallon underground storage tank. Bligh's analysis of the ACES tank heat exchanger showed that the effect of the extruded aluminum fin tubing (1/2" pipe with 3" fins) was minimal. After the ice in the ACES system had grown to 5" or 6", the aluminum fin effect had practically disappeared. This fact, along with the extruded pipe cost, makes it logical that copper be substituted. (Polyethylene pipe would be even cheaper and the effect of its resistance would be minimal once the ice had formed out beyond 3".) At the end of the winter, the tankful of ice is ready to absorb the house heat by circulating the remaining tank water through the house's fan coil unit. In this manner, the only energy required to cool a house would be needed by the small circulating pumps. Unfortunately, the region of the country where this works best is also the same region which experiences only mild summertime cooling loads. It is questionable how effective such a system would be in regions such as Tennessee. With warmer ambient winter conditions, more tank heat exchanger piping would be necessary, driving the system costs even higher.

The ACES project underwent major redesign the following year (1977). The 2000 feet of pipe exchanger, along with the air to water heat pump, were eliminated. In order to increase heat transfer during the ice making stages and improve the system's COP, the heat pump was replaced with an ice-maker heat pump. In this approach, during the winter water is drawn from the storage tank and trickled over the ice maker's evaporator plate, The ice forms out to a distance of about 1/4". At this point warm liquid refrigerant from the receiver tank is circulated through the evaporator plate to loosen the ice from the surface after which it drops to the ice storage tank. Using the hot refrigerant in this manner reduces the system's COP by using energy to melt the ice. However, by removing the ice after a 1/4" build-up, the heat pump's evaporator is operated at a constant 32°F making this extra expenditure of enery worth it. As before, the domestic hot water is preheated by circulating the refrigerant from the compressors through a heat exchanger (de-superheater). Space heating is accomplished in similar fashion with a warm condenser coil located in the house's air handling unit, Fig. 14 .

A comparison of data[26] between the ice maker operation, water to air, and air to air heat pump indicate that the highest COP is realized with the ice maker system. This assumes an ice maker water temperature of 32°F, a ground water temperature of 60°F and an ambient air temperature of

47°F, respectively. One reason for the ice maker's remarkable performance is the fact that 144 BTU/lb of water is absorbed by the evaporator plate in the phase change process. This results in a low mass flow requirement for the ice maker which results in lower pumping costs. However, even when neglecting this, the ice maker still exhibits the highest COP. Considering the remarkably lower lift in the two conventional systems, this runs counter to convention. One possible explanation for this may be that efficiency improvements,which were not incorporated into the conventional systems, were built into the ice maker by the manufacturers. If this is the case, then the comparison is not a truly valid one.

One question that has not been asnwered to everyone's satisfaction is whether or nor the ACES system is truly cost effective. Although significant energy savings are possible, the capital costs for this type of system are very large. It appears doubtful that the expense can be justified for the single family residence. However, larger scale systems for apartment buildings and commercial structures may eventually prove to be cost effective.

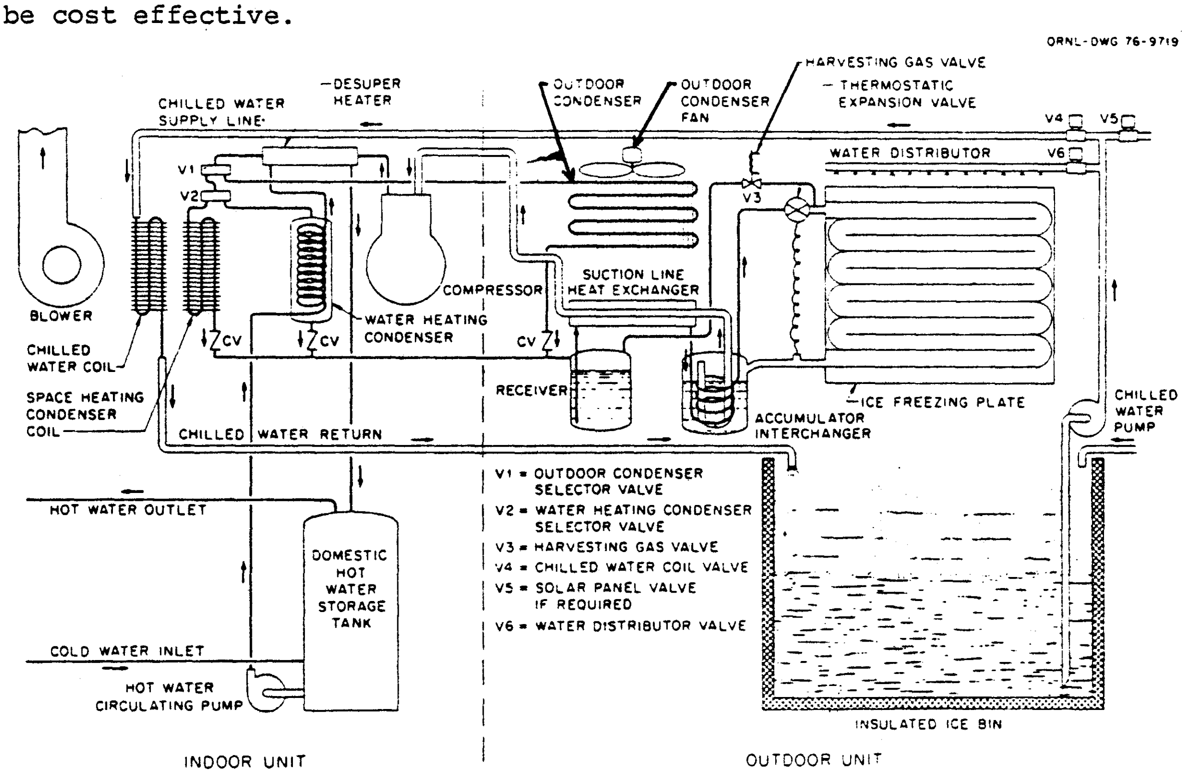


Fig.14. Schematic drawing of the ACES ice-maker heat pump

One of the first applications of the ice maker heat pump to commercial structures for off-peak cooling purposes is in the addition to Cray Research facilities in Mendota Heights, Minnesota. Architectural Alliance of Minneapolis is in charge of the project and Mason Somerville is the systems engineer from the University of North Dakota Engineering Extension Service. The ice maker is manufactured by TURBO Refrigerating Company of Denton, Texas and sells for approcimately $1000/ton (2-2.5 times the cost of a conventional chiller). A 30 ton heat pump is coupled to a 112,000 gallon storage tank ($.40/gallon).[27] The heat pump manufactures ice at night during the off-peak hours. Instead of rejecting heat to the environment, the heat at the condenser warms water to 110°F which is stored for any space heating needs. Additional boosting is possible to supply domestic hot water. When the condenser heat is not needed, it is rejected through a cooling tower. By utilizing both the heating and cooling outputs of the heat pump, the combined COP can be as high as 5.[28] This arrangement is most economical when the heating and cooling needs of a building are roughly equal.

There are two drawbacks to this approach. The first is cost. The ice makers are quite expensive and, due to their increased complexity, are subject to more mechanical failure problems than a conventional chiller. Secondly, when the tank is fully charged with ice "chips" only 1/2 of the volume is actually ice, the remainder is liquid water. Thus, the full potential of using ice as a PCM in order to reduce volumetric requirements is not being realized. This is due to the size and shape of the ice chunks in the storage tank. Also, in order to prevent uptake of ice by the water suction lines, ice making ceases once the level of the ice is within 2 feet of the bottom of the tank. The result is a volume which is less than 1/2 ice. Even though the ice level is 2 feet above the water uptake (located at the bottom of the tank) screens are placed over the pipe opening to prevent the uptake of slush. The screens initially used were susceptible to clogging and had to be replaced with more expensive ones.

It was difficult to determine the cost per energy unit stored for this system primarily due to the incomplete cost accounting at this point by Architectural Alliance. In addition, the decision was made to increase the size of the storage tank so that they would be able to have an ice capacity carry over of about four days. This,then,allowed them to reduce the capacity of their ice maker, the most expensive part of the system. In discussions with Architectural Alliance and TURBO manufacturing representatives it became clear that the cost of the ice makers installed was roughly $2000/ton ($1000/ton more than conventional equipment). The tank cost $.40/gallon. Assuming a 10 hr off-peak charge period, one day storage, and 50% volumetric change of phase, the incremental cost per million BTU of storage amounts to $8900 ($.89/BTU stored).

The problems encountered with the heat pump ice maker (complexity, cost, and percentage of volume involved in phase change) have been unknowingly addressed by the Wisconsin Electric Power Company.[29] This utility has experienced summer load peaking for a number of years, primarily due to the rapid investment in residential air conditioning systems. Due to the time of day rates that the utility was planning, and the hardship it would impose on consumers who have air conditioning Widconsin Electric, in 1975, undertook a development project in conjunction with A. O. Smith Corporation, a tank manufacturer, to develop an off-peak cooling storage system suited for residential application. The approach is quite similar to the ACES system except that it is cycled daily rather than seasonally. 200 ft of 3/8" copper tubing spaced 3 1/4" on center was built into a 180 gallon storage tank. The tubing functioned as the evaporator causing ice to form around the pipe out to a 3 1/4" diameter, at which point a controller shut off the compressor. Evaporator temperatures were maintained at 20°F. The system was operated at night taking advantage not only of the off-peak rates, but also of the lower temperatures available to the condensor. The drop in evaporator temperature below "normal" operation is matched by the drop in ambient air temperature at night vs. the day. Therefore, the lift is the same resulting in similar COP. The volume of ice made was approximately 80-85% of the tank volume. The remaining 32°F water was then circulated through a water to air heat exchanger in the air handling unit, slowly melting the ice cylinders while providing cooling and dehumidification to the home. The test modules constructed by the tank company for the utility cost $1800. The manufacturer reported that a limited quantity, commercially produced unit would cost \$400. Mass produced units are predicted to cost \$250. Assuming \$600/unit installed cost, 80% water to ice conversion, and a 120 gallon tank, the costs per unit of energy stored is \$.44/BTU.

Table 5 lists the cool storage systems and their characteristics as described in this section. The chilled water storage system has made the biggest inroads in the commercial market due to its relative simplicity and reasonable cost. The more advanced of these systems with floating membranes and elaborate plumbing is driving up the cost of the system in order to reduce the volumetric requirements. The G.E. salt and paraffin storage systems are bulky and expensive and cannot compete at all with the straightforward approach embodied in the water storage system. The MIT salt bag approach, although expensive, is the most space conserving (requires no central storage) and thermally efficient of the solutions. However, its limited applicability (interior zones) and severe design constraints are real handicaps.

The only logical and cost effective alternative to cool water storage is ice storage. The early concern of reduced COP due to low evaporator temperature was unfounded due to the lower night air temperature and the increasingly efficient low temperature refrigeration equipment.

The Wisconsin Electric ice storage design seemed to have the best change of providing an alternative to the water storage approach. However, their work was primarily centered on residential application. Commercial scale installations are possible but more detailed information is required about heat transfer in a pipe heat exchanger ice storage systems. Such information was obtained from experiments carried out at MIT, and are explained in the following chapter. These experiments reinforce the favorable conclusions of Wisconsin Electric and seem to indicate the viability of large scale commercial systems. In addition, a potential improvement in the ice maker heat pump concept is discussed.

TABLE 5. COOL STORAGE SYSTEMS

System	R&D Group	Operation Mode	BTU/ft^3	Testing	Cost $/million BTU)	Remarks
Cool water storage	in use for many years; membrane developed by R. Tamblyn, Ontario	coupled to chilled water system	1250, 20°F temperature differential	many installations	3000-6000	larger volume requirements limited ΔT
Rolling cylinders with Glauber salts	General Electric	air side storage	1870, 50% maximum packing density	laboratory	unknown	best suited for small commercial and residential
Glauber salts in bags	Massachusetts Institute of Technology	decentralized storage, air charge	3500	500 ft^2 test space	12,000	for use in interior zone of large commercial structures, may be charged with cool ambient air
Microencapsulated paraffins	General Electric	coupled to chilled water system	1750, 50% maximum packing density	laboratory	14,200	encapsulating material needs strengthening
Ice maker Heat pump	Oak Ridge National Laboratory	coupled to chilled water system	5280, 50% maximum packing density	40,000 ft^2 office, largest installation known	8900	equipment is twice as expensive as packaged chillers
Pipes in tank ice system	Wisconsin Electric with A. O. Smith Corporation	coupled to chilled water system	8100, 85% maximum packing density	residential tests conducted by Wisconsin Elec.	4400 (120 gallon tank)	major cost is pipe in tank

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EXPERIMENTAL MODELLING of an ICE STORAGE SYSTEM

EXPERIMENTAL OBJECTIVES

Because of the higher volumetric efficiency of the pipe heat exchanger ice system and the ability to couple it to conventional, cheap, packaged chillers, it was decided to carry out experiments which would quantify in an accurate way the system's heat transfer characteristics. The conceptual design of the system is illustrated in Figure 15. A water cooled chiller extracts heat from a brine solution (20% ethylene glycol,80% water) which is pumped through a series of pipes which comprise the ice forming heat exchanger in the storage tank. The brine extracts heat from the water, causing the water to change phase and form ice on the wall of the pipe (Fig. 17). This heat is carried by the brine back to the chiller's evaporator-cooler where it is transferred to the refrigerant,thus cooling the brine for the next pass through the tank.

Brine is used as an intermediate heat exchanger in these experiments for three reasons. From an experimental standpoint, it is easier to monitor the mass flow and temperatures of a liquid coolant than a refrigerant. Also, the technical feasibility of placing a direct expansion evaporation directly in the storage tank is questionable. It is critical that oil return from the evaporator is insured for proper lubrication of the compressor. Sufficient refrigerant velocities must be maintained along with suitable piping layout in order to facilitate this. The demands for an efficient ice making pipe configuration may not easily mesh with these demands. In addition, according to discussions with refrigeration engineers at Carrier Corporation, going to direct expansion, ice making evaporator piping will require custom built chillers. A brine system is compatible with the packaged chillers. Significant economies are realized by purchasing packaged chillers.

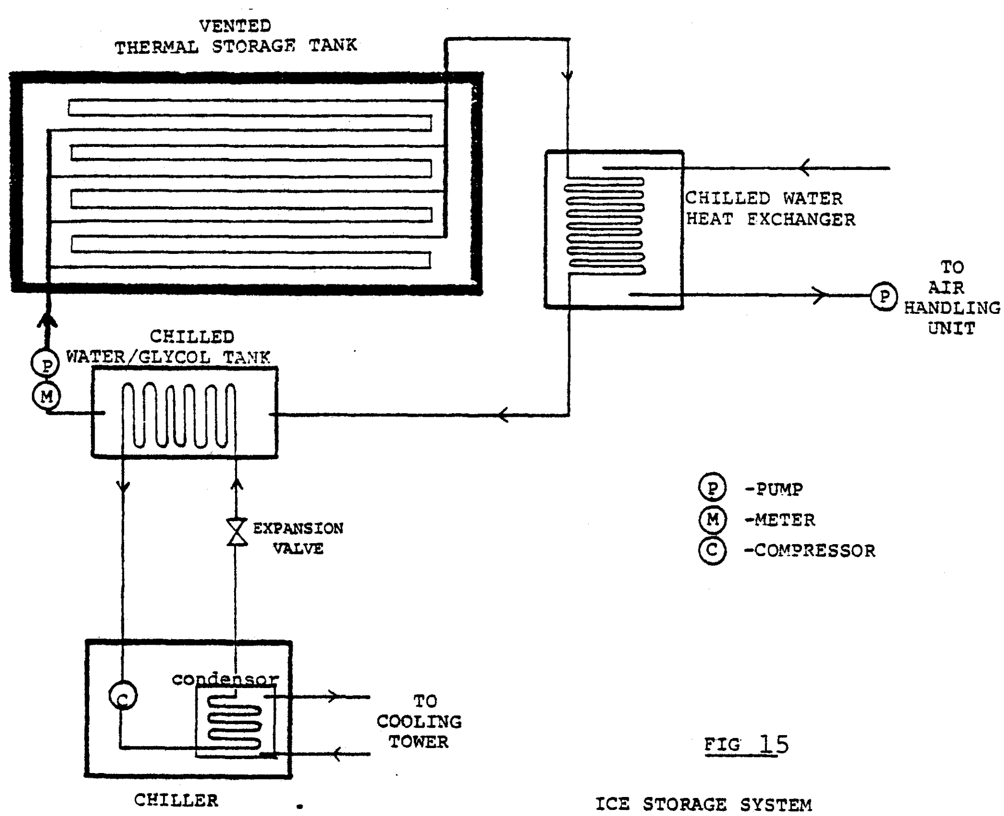


FIG 15

ICE STORAGE SYSTEM
INDIRECT DISCHARGE
CIRCUIT

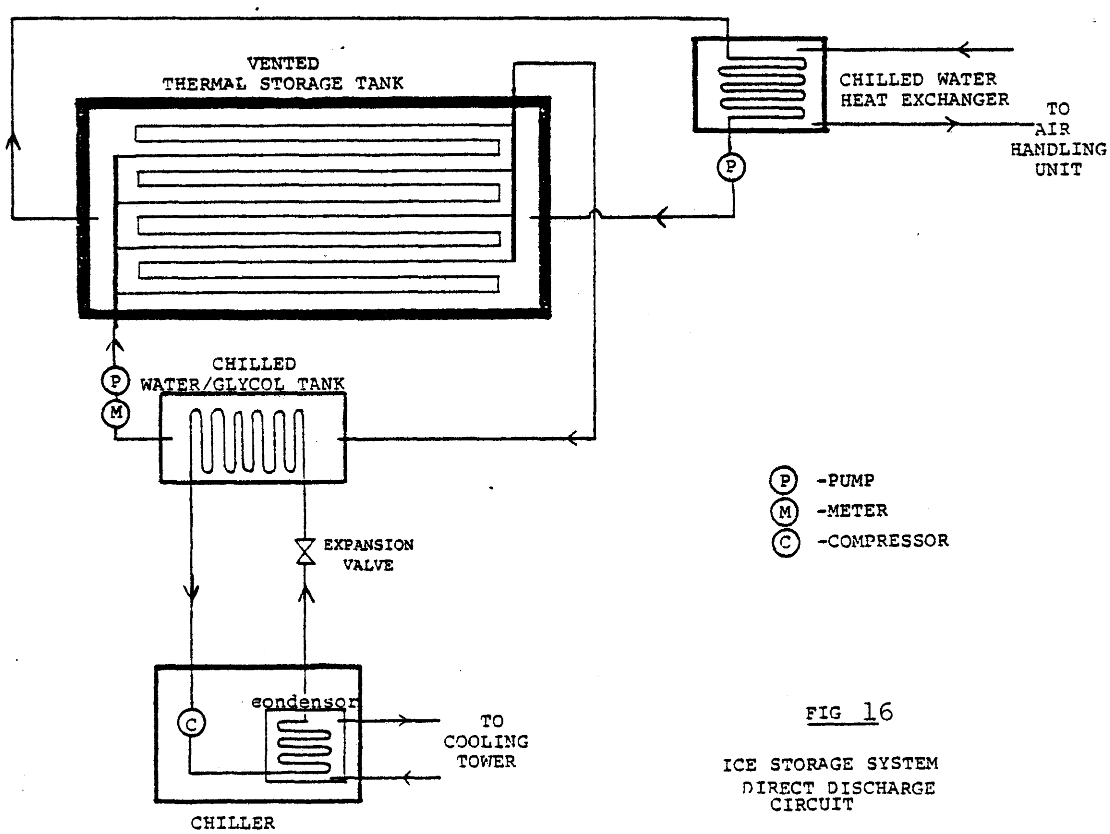
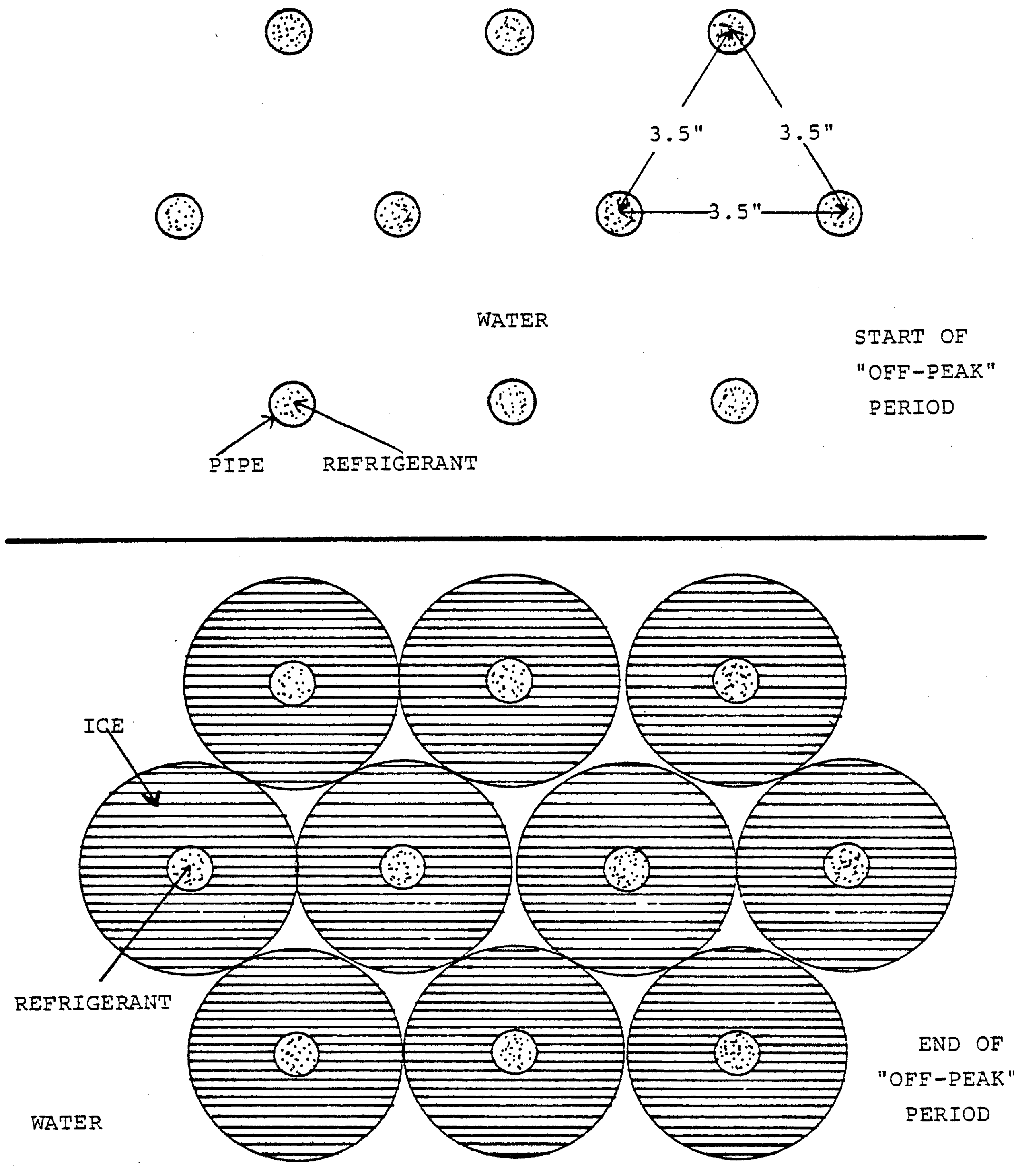


FIG 16

ICE STORAGE SYSTEM
DIRECT DISCHARGE
CIRCUIT



CROSS SECTIONAL VIEW OF PIPES IN

ICE STORAGE TANK

FIG 17

There are a series of heat transfers throughout this type of ice making process. The hot refrigerant from the compressor (105°F) loses heat to the cooling tower water across the condenser coils. After passing through the expansion valve, the cold refrigerant (about 20°F) picks up heat from the brine across the evaporator piping. The brine must draw heat through any ice build-up, through the pipe wall, and across the brine-pipe wall boundary layers. All of these heat transfers are limited by thermal resistances. The amount of heat flowing through a given thermal "resistor" is dependent on the temperature gradient across the resistor. By lowering the brine temperature, more ice will be formed per unit of time. That in turn means that the quantity of heat exchanger surface (pipe) will be reduced, thus lowering construction costs. However, by lowering the evaporator temperature, the temperature differential (lift) that the compressor has to operate over is increased, resulting in an increased power consumption for a given unit of cooling (a reduced COP). The system optimization process involves information and variables specific to each installation and region of the country. It is beyond the scope of this thesis to quantify these variables and perform an optimization. It will suffice at this time to assume that brine temperatures below 20°F will exact an unacceptable operating penalty on the chiller's COP during night-time operation (refer back to page 29 for additional information on this point).

At the beginning of the ice forming process, heat transfer from the water to the ice is limited by the conductivity of the pipe wall and the wall to brine boundary layer. Heat transfer across this boundary layer is influenced by the brine's conductivity, Reynolds number and Prandtl number. As the ice layer thickens around the pipe, the heat transfer is further reduced. Assuming a fixed brine supply temperature, the heat flow to the brine will fall off as time progresses. The amount of energy stored as ice over a given time period is equivalent to the area under the "heat flow vs. time" curve. The size of pipe used for the tank heat exchanger influences the Reynolds number of the brine due to the change in fluid velocity. It also influences the pressure drop across the pump which translates into the pumping energy requirements. A small pipe is desirable due to low cost and high fluid velocity (high Reynolds number), but undesirable due to greater pressure drops and smaller surface area. In order to allow flexibility in designing systems of various sizes and configuration, it is necessary to predict how pipe size will effect the rate of ice formation. Also, the pipe material may affect the ice formation. Copper, an excellent conductor, is also quite expensive. An economical alternative is plastic pipe, but its conductivity is much lower. The impact of this is evaluated. Most importantly, this information will indicate how thick the ice will be after a unit of time under varying operating conditions. This in turn will reflect the pipe spacing that is required to use the tank volume most efficiently which, in turn, indicates the quantity of pipe needed and its cost for a given cool storage requirement. Throughout the experiment, the charge period (off peak period) is assumed to be 10 hours. Longer charge periods will result in lower costs due to decreased chiller capacity and larger pipe spacing.

The ability of the storage system to discharge during cooling load periods is equally important. Discharging of the ice mass can be accomplished in two ways. With the indirect discharge system (Fig. 15), the building's heat is absorbed by the chilled water in the air handling units. This chilled water transfers its heat to the brine coolant via a heat exchanger. The brine is circulated through the storage tank heat exchanger releasing its heat through the pipe wall, to the melting ice. As the ice melts, the heat flow through the water/ice system changes. It is important that the heat transfer coefficient of such a system is sufficient to satisfy any peak cooling load the building may experience. An alternate way of discharging the ice mass is illustrated in Fig. 16. Here the charge and discharge circuits are separate. Making ice is accomplished as previously described. However, in discharging the system the remaining tank water is circulated directly around the ice and then through the heat exchanger that couples into the building's air handling units. The water is passed back to the storage tank where it transfers the heat directly to the ice mass. A further simplification would have the tank water circulated directly through the building's air handling units, avoiding the cost and inefficiencies of a heat exchanger.

However, because chilled water systems which supply air handling units are pressurized, this redesign would not be economically feasible. The storage tank would have to be built to withstand those pressures and would be prohibitively expensive. The intermediate heat exchanger does not degrade the performance of this system as it did to the cool water storage system. This is because the ice system gains its benefits not from the sensible heating of water which requires a substantial ΔT (See page 33) but from the latent heat of fusion which theoretically occurs at fixed ΔT. The difference in the discharging ability of the two ice storage designs are evaluated in the experiments.

MODELLING METHODS

Heat flow and ice formation in an ice storage system similar in concept to Figs. 15,16, theoretically can be accurately predicted with a two dimensional heat flow equation for a tube surrounded by an annular layer of ice.[1] The overall heat transfer coefficient can be determined by solving the equation

$$U_{theor.} = \frac{2\pi}{\dfrac{1}{h_{in} x R_{in}} + \dfrac{\ln(R_{out}/R_{in})}{K_{pipe}} + \dfrac{\ln(R_{ice}/R_{out})}{K_{ice}}} \qquad (2)$$

where

$U_{theor.}$ (BTU/hrft°F) = theoretical heat transfer coefficient at a given ice thickness

h_{in} (BTU/hrft^2°F) = convective heat trnasfer coefficient of the inside surface of the pipe

R_{in} (ft) = inside radius of pipe

R_{out} (ft) = outside radius of pipe

R_{ice} (ft) = outside radius of ice

K_{pipe} (BTU/hrft°F) = conductivity of pipe

K_{ice} (BTU/hrft°F) = conductivity of ice

$$h_{in} = \frac{K_b}{R_{in}} [0.0118 \, (Pr_b^{0.3})(Re_b^{0.9})] \tag{3}$$

where

K_b (BTU/hrft°F) = conductivity of brine

R_{in} (ft) = inside pipe radiue

Pr_b = Prandtl number of brine

Re_b = Reynolds number of brine

$$Pr_b = \frac{U_b \cdot C_b}{K_b} \tag{4}$$

where

U_b (lb/hrft) = absolute viscosity of brine

C_b (BTU/lb°F) = specific heat of brine

$$Re_b = \frac{\rho_b \cdot V_b \cdot R_{in}}{U_b} \tag{5}$$

where

ρ_b (lb/ft^3) = density of brine

V_b(ft/hr) = velocity of brine

therefore,

$$h_{in} = \frac{K_b}{R_{in}}[0.0118(\frac{U_b \cdot C_b}{K_b})^{0.3} \cdot (\frac{\rho_b \cdot V_b \cdot R_{in}}{U_b})^{0.9}] \qquad (6)$$

Experimental results are compared to the predictions in order to ascertain the validity of the mathematical model. Once confirmed, the model can then predict the impact of various pipe radii and pipe materials on ice formation without the expense of an experimental set-up.

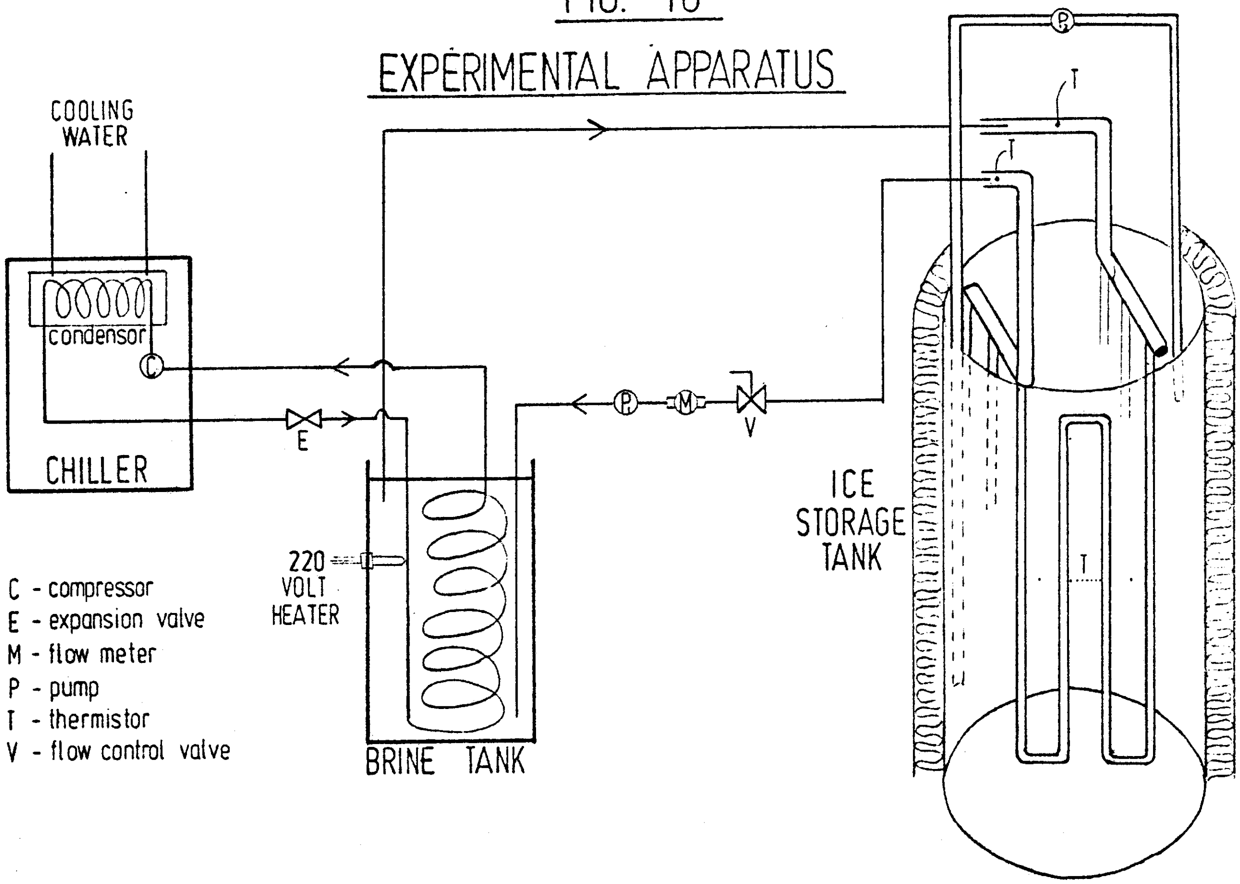
THE EXPERIMENT

A 65 gallon water tank is used as a storage vessel and is insulated with 3 1/2" of fiberglass (U = .09 BTU/hrft^2°F) on the sides and 4" of polystyrene on the top and bottom (U = .05 BTU/hrft^2°F). A heat exchanger built from copper tubing placed inside the tank consisting of three 17ft branches (1/2" I.D. copper M) which are fed from and returned to 1" manifolds in a reverse return fashion in order to insure a balanced flow. The brine is pumped through the heat exchanger by a fractional horsepower TACO circulator (008). The flow is monitored with a rotating disc type water meter and controlled manually with a butterfly valve. The brine is chilled in a 30 gallon tank by circulating it past the chiller's evaporator tubing. The chiller capacity is 1 1/2 ton (design shown in Fig. 18).

Heat flow is determined by monitoring the inlet and outlet bring temperature. The temperature difference times the mass flow rate times the specific heat of the brine is equal to the heat transferred from the water to the brine. Knowing the heat of fusion of water (144 BTU/lb), the quantity of ice being formed is determined by subtracting the heat gain to the tank from the cumulative amount of heat drawn from the tank. In addition, tempera-

FIG. 18

EXPERIMENTAL APPARATUS



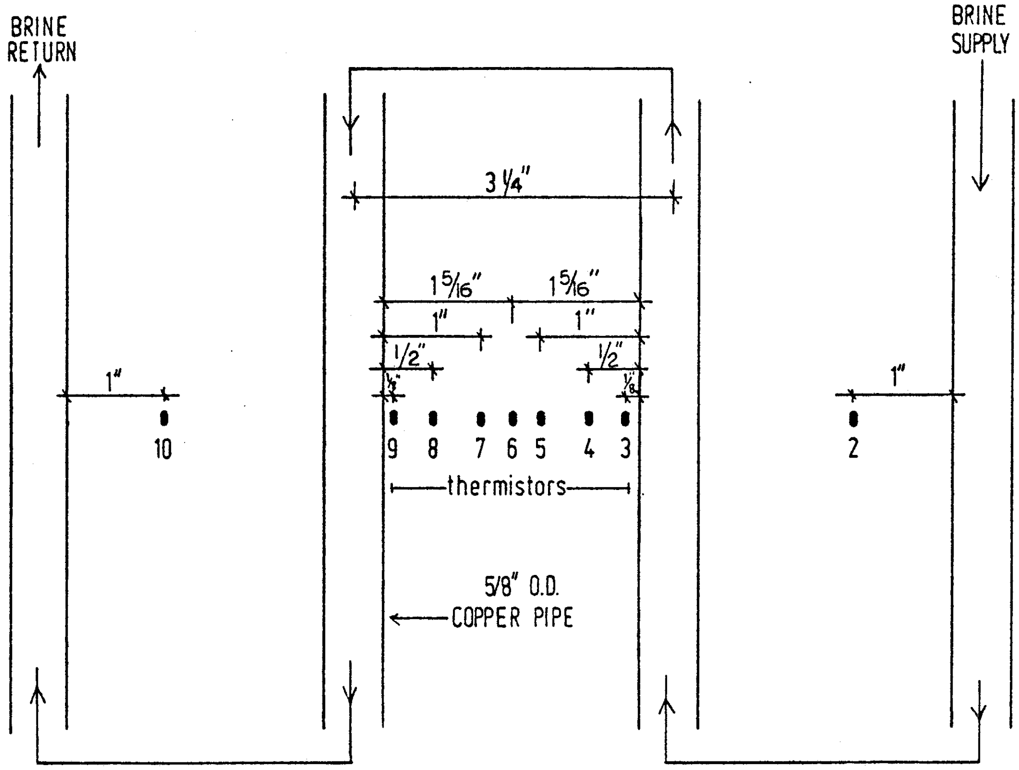


FIG. 19

THERMISTOR LOCATION in STORAGE TANK

ture sensors (thermistors) are suspended in the tank at various distances from the pipe along one of the brances (Fig. 18,19). When the ice had crystallized out to a thermistor, this is reflected by the drop in temperature below 32°F. Also, a set of calipers is used to manually check on the ice thickness in order to confirm the information obtained from the thermistors. The thermistor (a YSI 44203) is a composite device consisting of resistors and precise thermistors which produce an output voltage linear with temperature. A solid state low voltage power supply provides 3 volts to the thermistors. The output voltage is read manually off a digital voltmeter and converted to °F.

During the discharge mode, a 4500 watt heating element is immersed in the brine cooler to simulate a building's cooling load. The brine supply temperature ot the tank is held around 52°F. A smaller heating unit is used to heat the tank water during heat addition to the direct discharge design.

The experimental heat transfer coefficient is calculated in the following fashion.

$$\text{Heat flow (BTU/hr)} = (|T_{out} - T_{in}|) \times \frac{\text{lb coolant}}{\text{hr}} \times \begin{matrix}\text{specific heat}\\ \text{of coolant}\end{matrix} \qquad (7)$$

$$\text{Heat transfer coeffieicnt(BTU/hrft°F)}\,U_{exp.} = \frac{\text{Heat flow(BTU/hr)}}{ft_{pipe} \cdot \text{LMTD}} \qquad (8)$$

where

Ft_{pipe} = total length of 1/2" pipe (52 ft)

LMTD = log mean temperature difference between the inlet and outlet temperatures of the brine circulated through the heat exchanger and the temperature of the water in the ice bin.

$$\text{LMTD} = \frac{(T_{wt}-T_{in}) - (T_{wt}-T_{out})}{\ell n\left(\frac{T_{wt}-T_{in}}{T_{wt}-T_{out}}\right)} \tag{9}$$

where

T_{wt}(°F) = storage tank water temperature

T_{in}(°F) = brine temperature at heat exchanger inlet

T_{out}(°F) = brine temperature at heat exchanger outlet

The experimental heat transfer coefficient (BTU/hrft°F) is then compared to the theoretical heta transfer coefficient predicted by model at various ice thicknesses. No modelling was attempted for the discharge mode, although such modelling is possible. (See Ref. 1,2). As the data will later show, the direct discharge capability is more than sufficient to handle any foreseeable cooling loads. The limiting factor, therefore, is posed by the charging mode and it is here that the modelling is needed.

ANALYSIS OF THE HEAT EXTRACTING PROCESS

Data included in this report is taken from three experiments conducted at the Building Systems Laboratory at MIT. The difference between the three runs is the inlet-outlet brine temperatures and the corresponding rate at which ice is formed. Data from the first run is shown in Table 6. In this run, the tank water is initially cooled to a uniform 32°F and the brine flow rate is held at a constant 3.21 g.p.m. throughout the course of the experiment. As indicated by the "heat transfer coefficient" column, heat flow initially is quite high, with the inner pipe-brine convective heat transfer coefficient being the only real source of thermal resistance. Ice quickly solidifies out to a thickness of 1/4" before the thermal resistance

TABLE 6. HEAT EXTRACTION, 25.6°F AVERAGE BRINE TEMPERATURE

Time (hr)	Mass Flow (GPM)	Temp; °F: Tank Water	Temp; °F: Inlet Brine	Temp; °F: Outlet Brine	Temp; °F: ΔT Inlet-Outlet	Temp; °F: LMTD	Heat Flow (BTU/Hr)	Ice Thickness (in)	Experimental Heat Transfer
.10	3.21	32	24.54	27.06	2.52	6.11	4032		12.7
.75	"	"	24.90	26.52	1.62	6.26	2592		8.0
.90	"	"	"	26.70	1.80	6.16	2880	0.3	9.0
1.27	"	"	"	26.34	1.44	6.35	2304		7.0
1.50				26.70	1.80	6.16	2880	0.45	9.0
2.00	"	"	"	26.34	1.44	6.35	2304	0.50	7.0
2.25	"	"	"	26.52	1.62	6.26	2592	0.50	8.0
2.66	"	"	"	26.16	1.26	6.45	2016		6.0
3.00	"	"	"	26.34	1.44	6.35	2304	0.65	7.0
3.33	"	"	"	26.16	1.26	6.45	2016		6.0
3.75	"	"	"	26.34	1.44	6.35	2304	0.75	7.0
6.60	"	"	"	26.16	1.26	6.45	2016	1.0	6.0
7.10	"	"	"	25.98	1.08	6.55	1728	1.0	5.1
9.66	"	"	"	26.16	1.26	6.45	2016	1.2	6.0
10.00	"	"	"	25.98	1.08	6.55	1728	1.2	5.1

of the ice itself begins to dominate the rate of ice formation. Although the ice build-up acts to throttle down the flow of heat, the increasing ice circumference counteracts this somewhat by increasing the surface area through which the heat moves. As shown in Fig.20 the combination of these two effects produces a curve which approximates a log function whose slope approaches zero after the ice passes 1.2". This implies that the heat flow remains fairly constant (falls off very slowly) past this point. However, this does not mean that the ice build-up in terms of <u>ice thickness</u>, proceeds at an almost constant rate. Every additional incremental layer of ice embodies an increasing quantity of energy due to the Πr^2 relation between ice thickness and ice volume. Therefore, the thicker the ice, the longer it takes to add the next incremental layer - assuming a constant heat flow. This can be seen in the data in Table 6. The first 1/2" of ice is added after only 2 hours. During this same period, the heat flow has fallen from 12.7 to about 7.5 BTU/hrft$_{pipe}$°F. In order to add the next 1/2" layer of ice it takes 4.6 hours and yet the heat transfer has only fallen from 7.6-6.0 BTU/hrft°F. At the end of the 10 hour charge period, the ice had formed out to a thickness of 1.2" indicating that the optimum pipe center-to-center spacing would be 3". The ice thickness (1.2") plus the pipe radius (.3125") equals 1.5125". Since the adjoining pipes will have the same ice layer around them, a spacing of 3" will insure that all the neighboring ice cylinders just touch one another. This arrangement utilizes the tank volume most efficiently (See Fig. 17). The only remaining liquid water (about 10% tank volume) is needed to provide the channels through which the water will flow in order to efficiently and uniformly discharge the ice store. If, after the charge period is over, there is more water in the tank (i.e., the ice cylinders have not touched) then the pipe spacing was too large and consequently, the tank was excessively large.

Fig. 21 shows the thermistor data plotted as the experiment progressed (See Fig. 18,19 for location of thermistor). Thermistors 3 and 9 experienced freezing quite rapidly as they were located only 1/8" from the pipe. Ther-

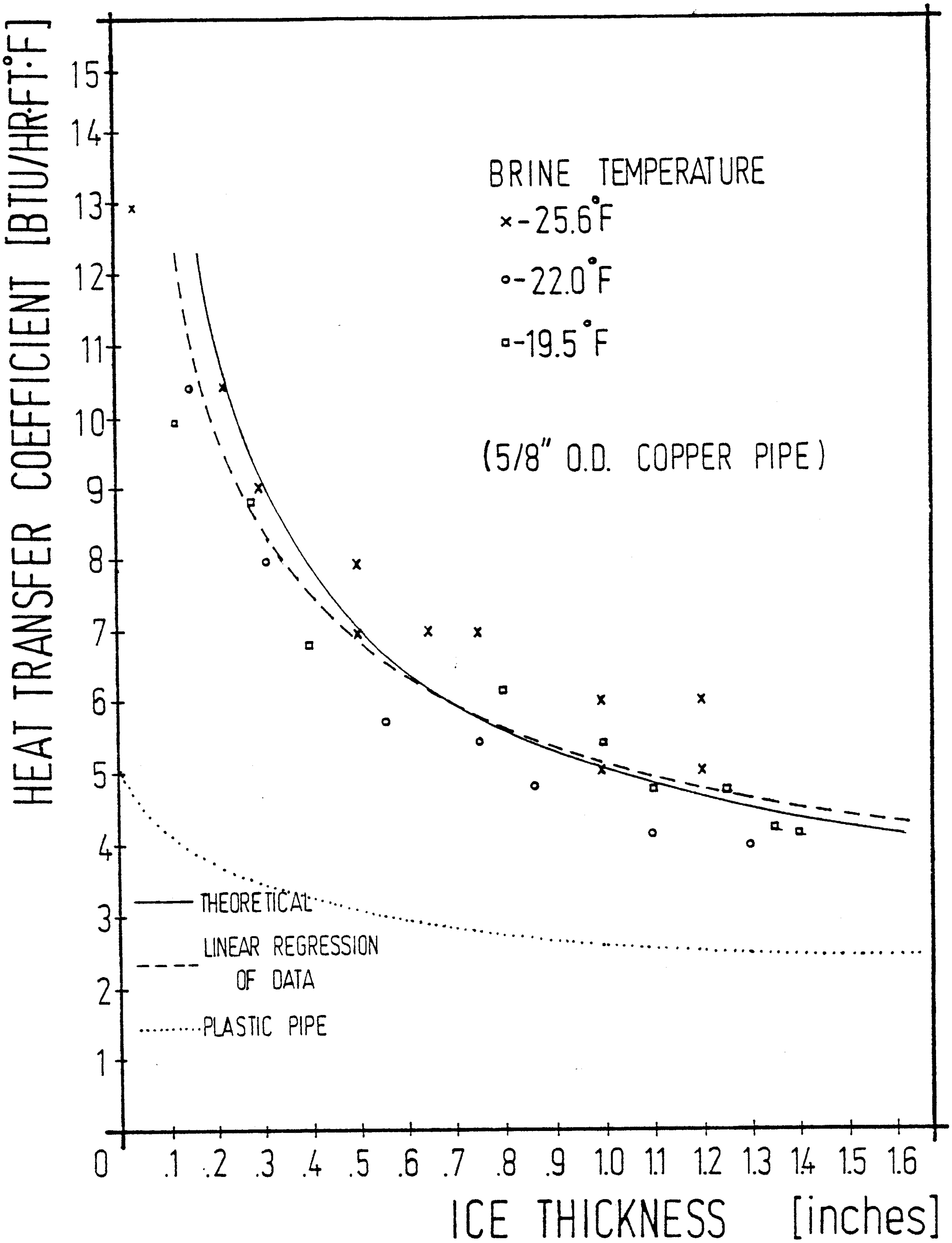


FIG 20

HEAT TRANSFER in an ICE STORAGE SYSTEM

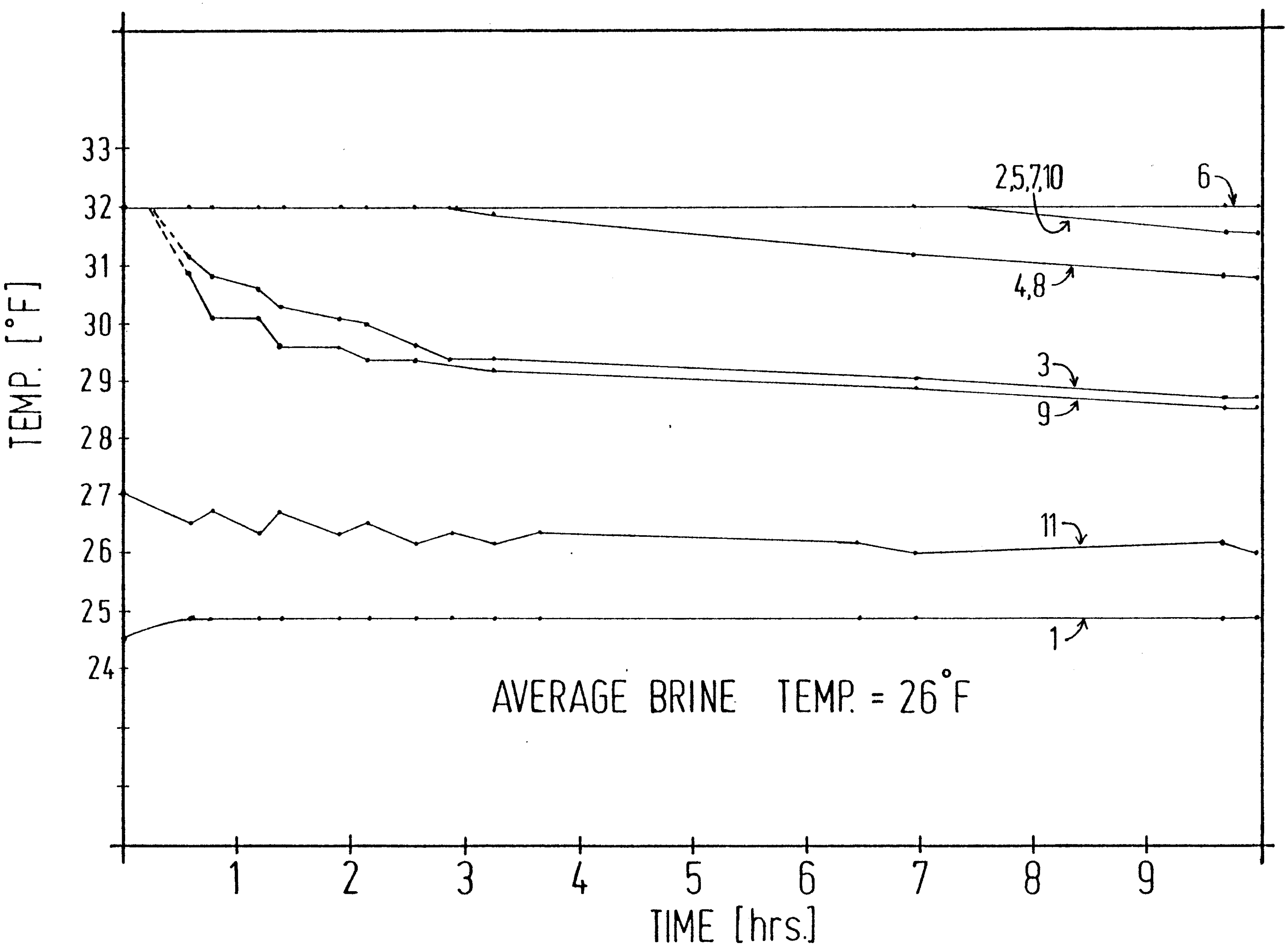


FIG. 21

THERMISTOR RESPONSE DURING HEAT EXTRACTION mistors 4, 8 (1/2" out) registered ice after 2.5 hours. The next thermistors 2,5,7,10 did not indicate the formation of ice until around the 7th hour. The location of the thermistors imply that the #2 thermistor should freeze before #10 since the brine is the coldest at #2 and has warmed up by the time it circulates to #10. However, looking at Table 6, the ΔT between inlet and outlet is typically 1.4°F, while between thermistor #2,10 the ΔT is probably only 1°F. This slight difference is not sufficient enough to be seen in the response of the thermistors. Had the mass flow rate been much less than 3.27 g.p.m., the thermistors would probably have responded differently.

Tables 7, 8, show the next two experimental runs conducted at lower brine temperatures (22°F, 19.5°F). In these two experiments, the tank water was initially at 34-35°F, resulting in some of the energy being expended to sensibly cool the water mass. At 22°F, the optimum pipe spacing should be 3 1/2". Experimentally, the ice only formed out to a thicnkess of 1.3", which is equal to a pipe spacing of 3 1/4". The difference in equivalent energy is equal to the amount of heat lost from the tank and the sensible cooling of the water mass at the start. At 19.5°F, the optimum pipe spacing is 3 3/4" while experimentally (for the same reasons) the spacing was effectively 3 1/2". One change that should be noted between the first experiment and the next two is the drop in the mass flow in runs #2 and 3. Evidently, the reduced temperature changed the velocity of the brine enough to have an effect on the capacity of the pump. This was monitored constantly and factored into the calculations.

The data from the three experiments is presented in Fig.20, showing the relationship between the ice thicnkess and the heat transfer coefficient. By taking the natural log of the data, an approximately straight line is generated, making it possible to run a linear regression. The dashed curve on the graph is the result of this analysis and the correlation coefficient is .905. In order to compare the data to the models prediction, equation 2, page 64, had to be computed for a series of ice radii (this is the only variable that changes). Before that equation can be solved, the pipes inside convective heat transfer coefficient (h_{in}) needs to be determined. The values for the variables (below) were obtained from the ASHRAE Handbook of Fundamentals, P. 17.7-17.8.

TABLE 7. HEAT EXTRACTION, 22°F AVERAGE BRING TEMPERATURE

Time (hr)	Mass Flow (GPM)	Temp, °F					Heat Flow (BTU/Hr)	Ice Thickness (in)	Experimental Heat Transfer
		Tank Water	Inlet Brine	Outlet Brine	ΔT Inlet-Outlet	LMTD			
0	3.28	34.5	21.74	25.52	3.78	10.66	6173		11.13
0.33	"	32.7	22.28	25.16	2.88	8.70	4703		10.39
1.00	"	32.4	21.74	24.08	2.34	9.24	3821	0.32	7.95
2.0	3.10	32.2	20.84	22.82	1.98	10.14	3055	0.56	5.79
3.0	"	32.2	20.30	22.28	1.98	10.68	3055	0.75	5.50
4.0	"	32.0	20.12	21.92	1.80	10.97	2778	0.86	4.87
6.5	3.00	32.0	19.76	21.38	1.62	11.40	2464	1.10	4.16
8.0	"	32.0	23.54	24.80	1.26	7.81	1882	1.22	4.63
10.0	"	32.0	19.40	21.02	1.62	11.77	2464	1.30	4.03

TABLE 8. HEAT EXTRACTION, 19.5°F AVERAGE BRINE TEMPERATURE

		Temp; °F							
Time (hr)	Mass Flow (GPM)	Tank Water	Inlet Brine	Outlet Brine	ΔT Inlet-Outlet	LMTD	Heat Flow (BTU/Hr)	Ice Thickness (in)	Experimental Heat Transfer
0.1	3.06	34.3	20.48	24.44	3.96	11.73	6039	0.1	10.0
0.50	"	33.6	21.56	24.26	2.70	10.63	4118		7.5
1.00	"	33.3	21.20	23.72	2.52	10.79	3843	0.4	6.9
1.50	3.03	32.2	19.76	22.28	2.52	11.13	3805		6.6
2.66	"	32.0	18.86	21.38	2.52	11.84	3805	0.8	6.2
3.50	"	"	18.32	20.66	2.34	12.47	3533		5.5
4.0	"	"	18.32	20.66	2.32	12.47	3533	1.0	5.5
5.0	2.98	"	18.14	20.30	2.16	12.75	3208	1.1	4.9
6.0	2.97	"	17.96	20.12	2.16	12.93	3197	1.25	4.8
7.0	"	"	17.78	19.76	1.98	13.21	2930	1.35	4.3
8.0	"	"	17.60	19.58	1.98	13.39	2930	1.40	4.2
9.5	2.96	"	17.24	19.22	1.98	13.75	2921	1.50	4.1

$K_b = 0.28$ BTU/hrft°F

$R_{in} = 0.021$ ft

$U_b = 5$ lb/hrft

$C_b = 0.9$ BTU/lb°F

$\rho_b = 64.9$ lb/ft^3

$V_b = 6084$ ft/hr

therefore

$h_{in} = 286$ BTU/hrft^2°F

In equation 2, page 64, all the products in the denominator are in the form of a thermal resistance. Table 9 compares the thermal resistance of each component. The copper pipe is practically invisible, thermally. The inside convective coefficient is a limiting factor only in the very beginning of the experiment and is soon overshadowed by the thermal resistance of the ice itself. By solving equation 2 (page 64) for a series of increasing ice radii, a curve begins to emerge, and is shown graphically in Figure 20. The dotted line represents the results of a linear regression of the experimental data. The results from equation 2 form the solid line and are quite close to what was observed. From this result, it seems fair to assume that the theoretical model provides an accurate means of predicting heat transfer characteristics of a piped ice storage system.

TABLE 9. R-VALUE OF THERMAL COMPONENTS IN A COPPER PIPE SYSTEM

Ice thickness (inches)	$\frac{1}{h_{in} \cdot R_{in}}$ (Hrft°F/BTU)	copper pipe 5/8"	Ice	Total
0	0.167	0.001	0	0.168
0.1	"	"	0.21	0.378
0.2	"	"	0.37	0.538
0.4	"	"	0.62	0.788
0.6	"	"	0.81	0.978
0.8	"	"	0.96	1.128
1.0	"	"	1.08	1.248

ANALYSIS OF MODELLING PREDICTIONS

When the copper pipe is replaced with similar sized plastic piping, heat transfer is substantially reduced as shown in Fig.20. When the individual "resistive" components are compared (Table 10), the pipe now becomes the limiting factor in the early stages of ice formation.

TABLE 10. R-VALUES OF THERMAL COMPONENTS IN A PLASTIC PIPE SYSTEM

Ice Thickness	$\frac{1}{h_{in} \cdot R_{in}}$ (Hrft°F/BTU)	plastic pipe 5/8"	Ice	Total
0	0.167	1.16	0	1.327
0.1	"	"	0.21	1.537
0.2	"	"	0.37	1.697
0.4	"	"	0.62	1.947
0.6	"	"	0.81	2.137
0.8	"	"	0.96	2.287
1.0	"	"	1.08	2.407

The plastic modelled here is polyethylene semi-rigid tubing 1/2" nominal I.D. with a .064" thick wall. The material is capable of handling 80 psi and its K value is 2.3 BTU/hrft^2°F/in (copper is about 2400). The only other readily available plastic that might be considered is PVC. However, its conductivity is 1/2 that of polyethylene and its cost is approximately three times as great. Both plastics are compatible with methanol or ethylene glycol - both suitable anti-freezes.

In order to accommodate the requirements of a variety of different sized ice storage systems, it is necessary to model the effect of pipe size and material, as well as brine temperature on the heat transfer coefficient. Smaller capacity systems may require only a 3/8" or 7/16" pipe while larger systems may be more efficiently served by 7/8" or 1 1/8" pipes. As discussed earlier, the inside diameter of the pipe affects the pressure drop through the pipe, the pumping requirements, and the inside convective heat transfer coefficient. More importantly, the outside diameter determines the actual surface area of heat exchanger per foot of pipe and consequently the overall heat transfer coefficient of the pipe. Fig.22 shows the effect of pipe size on heat transfer. To simplify matters, h_{in} is assumed to be equal in all the pipes which implies that the mass flow is higher in the larger pipes. As the pipe size increases the number of feet required to transfer a unit of heat goes down. As expected, plastic pipe impedes the overall system heat transfer.

The rate of ice formation is dependent on the heat exchanger size, material, temperature, and whatever ice has already formed on the pipe. By breaking the ice radii into 0.1" incremental cylinders which embody a certain amount of latent heat (Q_n) and dividing the quantity (BTU per ft of pipe) by the heat transfer coefficient "U" (BTU/hrft°F) for that distance (r_n), and by the ΔT between the evaporator and the water tank, the time required to freeze that cylinder of ice is determined (See Fig.24).

$$Q_n = [\pi(\Gamma_o)^2 - \Pi(\Gamma_i)^2] \times \frac{56 \text{ lb ice}}{\text{ft}^3 \text{ ice}} \times \frac{144 \text{ BTU}}{\text{lb ice}} \qquad (10)$$

where

Γ = ft.

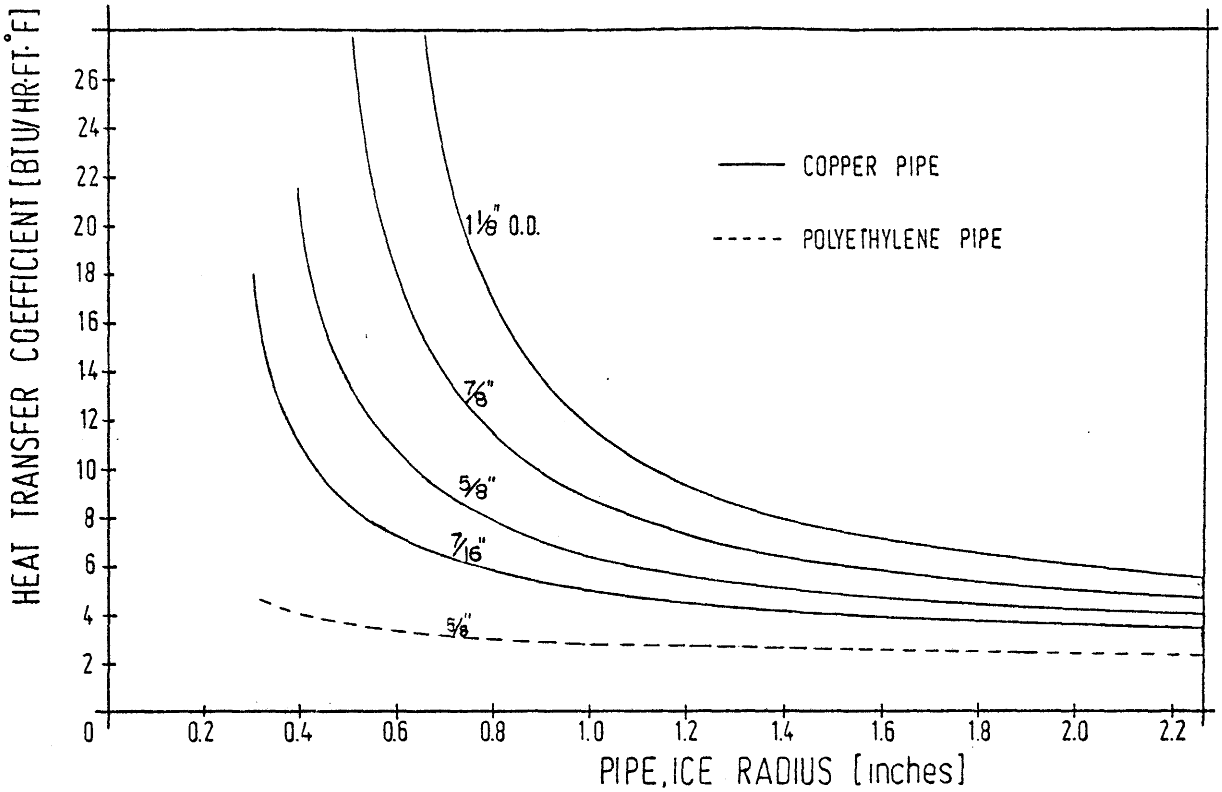


FIG 22

EFFECT of PIPE SIZE on HEAT TRANSFER

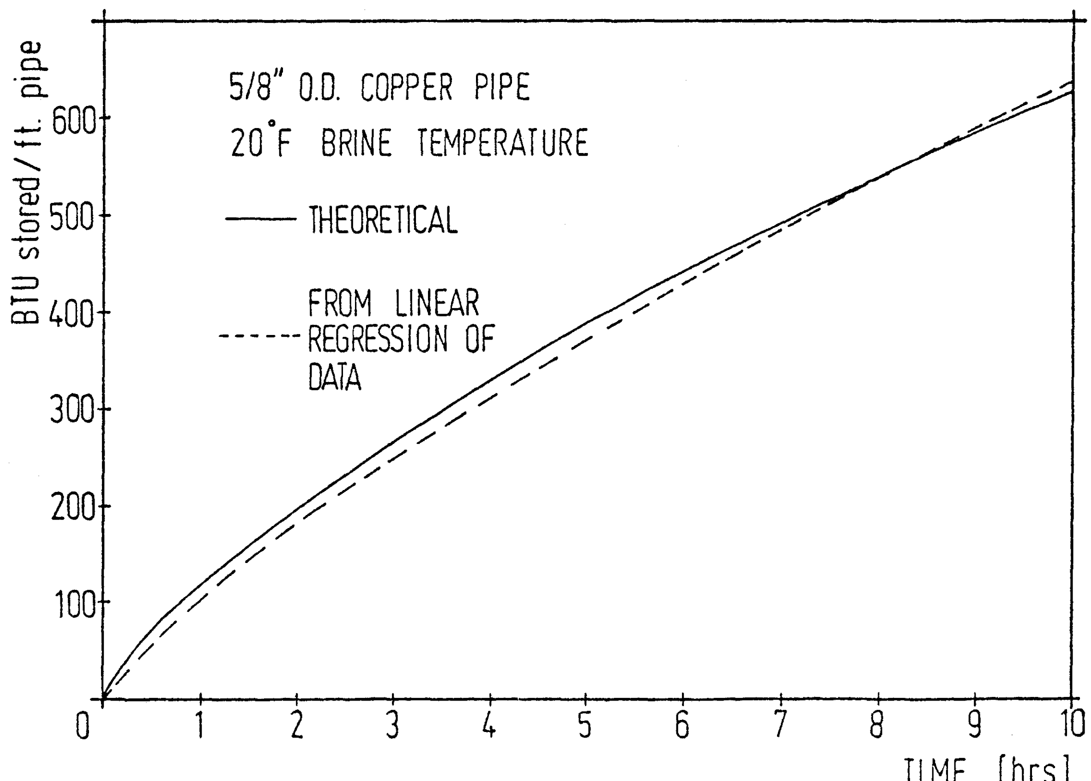


FIG 23

BTU's STORED (as ice) vs. TIME

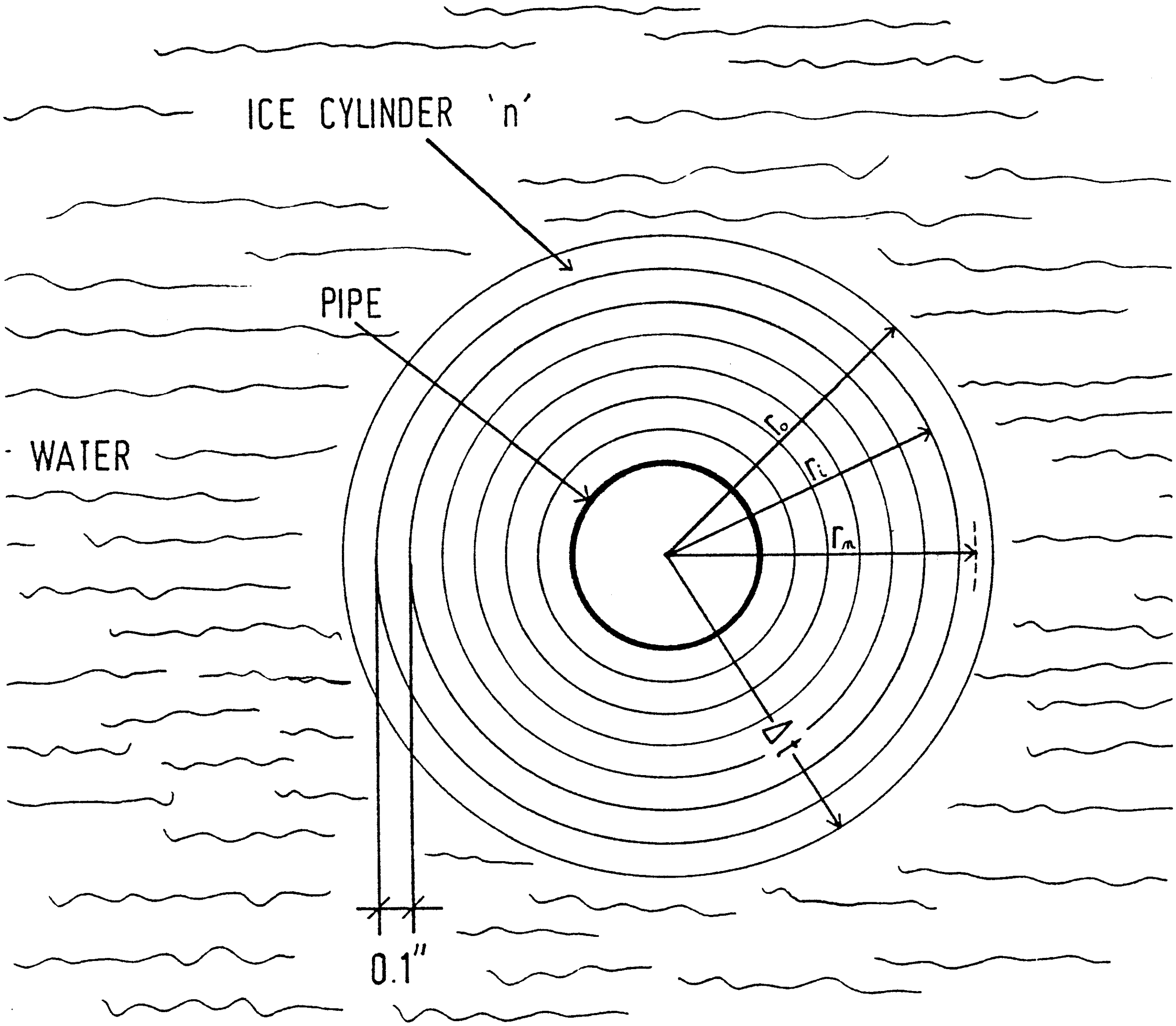


FIG. 24

TIME REQUIRED TO FORM ICE CYLINDER 'n'

Also,

$$U_n = f(r_n) \tag{11}$$

$$\frac{Q_n}{(U_n)(\Delta T)} = \text{time} \tag{12}$$

$$\frac{BTU/ft}{(BTU/hrft°F)(°F\ T)} = \text{hrs} \tag{13}$$

By reiterating this process for all the ice cylinders the time it takes to freeze the nth cylinder can be calculated. Conversely, by knowing the duration of the charge period (10 hrs), the radius of ice formed at the end of that period can be determined. A program was written to be used in conjunction with the TI-59 programmable calculator. The program was used to model ice formation under a variety of brine temperatures and for different pipe sizes. Fig.23 compares the predictions made with the two dimensional heat flow model to the real data for the conditions listed. After the end of the charge period, the difference between the two curves amounts to only 2%. In Fig. 25, the ice radius, rather than BTU stored, is plotted over time, the difference being less than 1%. It appears that the predictions made with the theoretical model are sufficiently accurate. As the brine temperature is reduced, a greater amount of ice is formed. For 5/8" copper pipe, the ice radius at the end of a 10 hour charge period is 1.6". At 20°F it is 1.9" while at 15°F, the ice radius would have grown to 2.15". There are two reasons why the increase in the ice radius between 20°F and 15°F is not as large as between 25°F and 20°F. First, going from 25°F to 20°F increases the temperature differential by 70%. Dropping from 20°F to 15°F only increases the temperature differential by 40%. In addition, the increase in the amount of energy embodied in the ice is not proportional to the increase in radius, but rather is a function of the square of the radius. Finally, the thicker ice radius at 15°F imposes a higher thermal resistance resulting in a reduced heat flow.

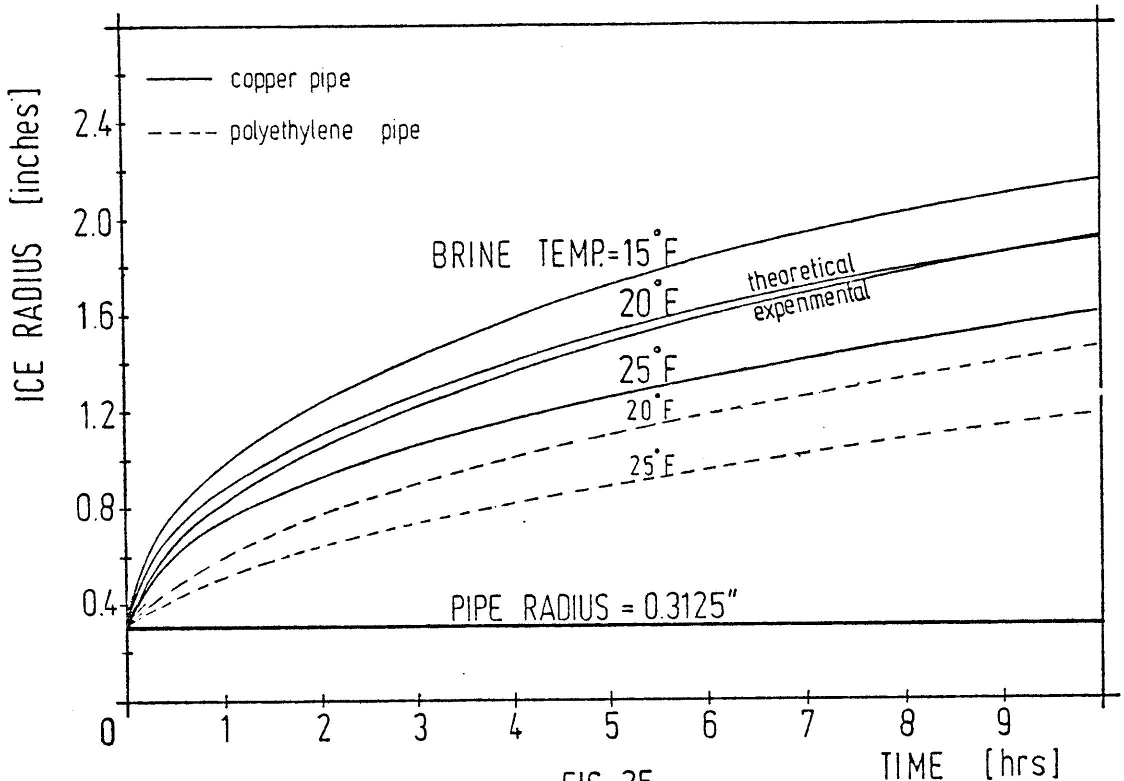


FIG 25
ICE FORMATION vs. TIME

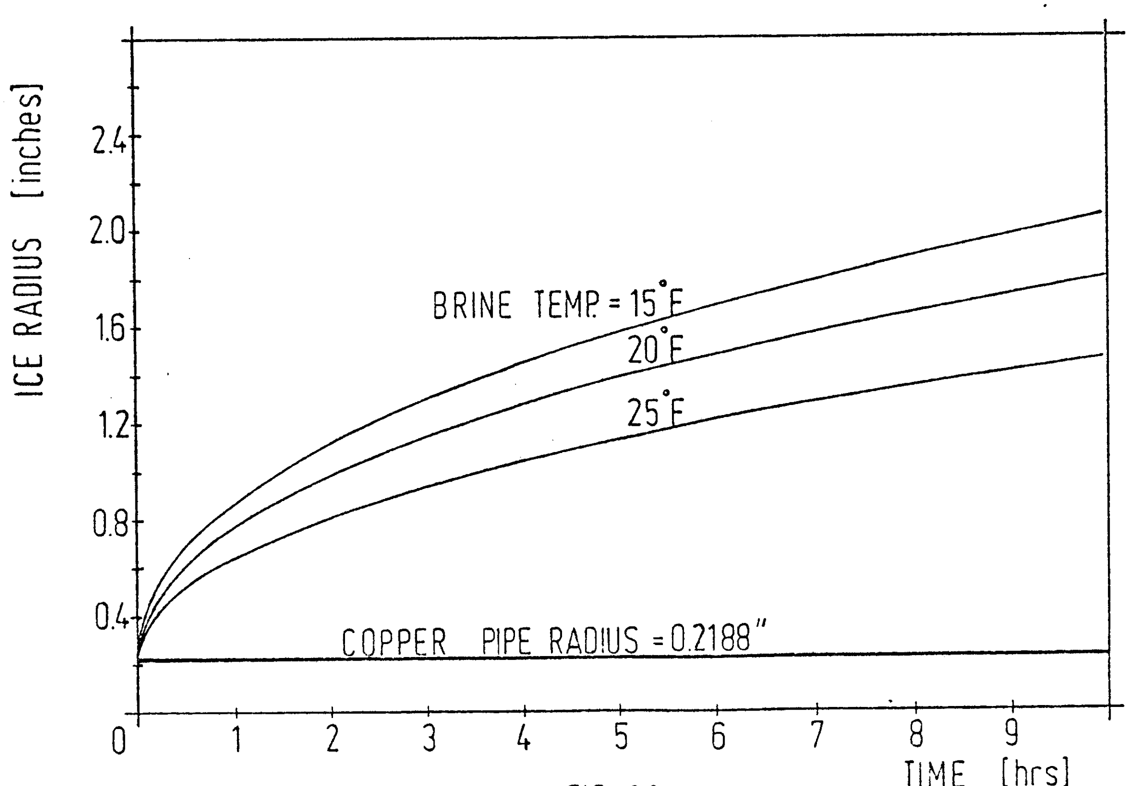


FIG 26
ICE FORMATION vs. TIME

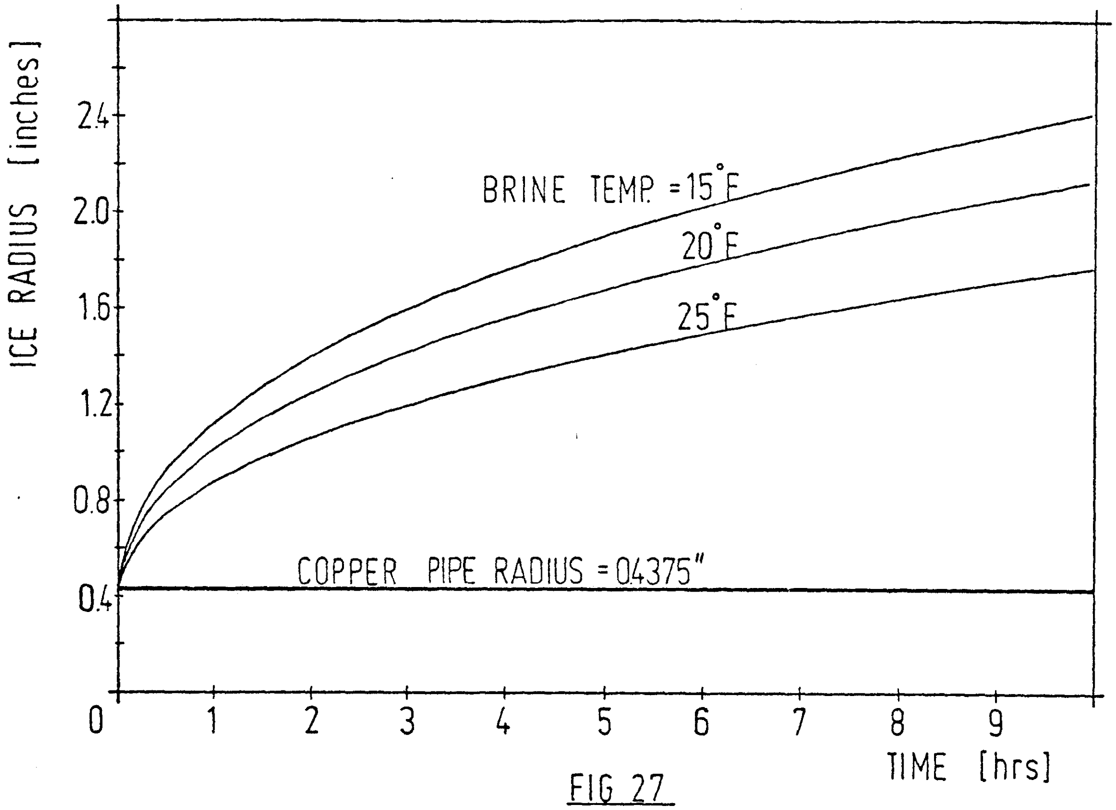


FIG 27
ICE FORMATION vs. TIME

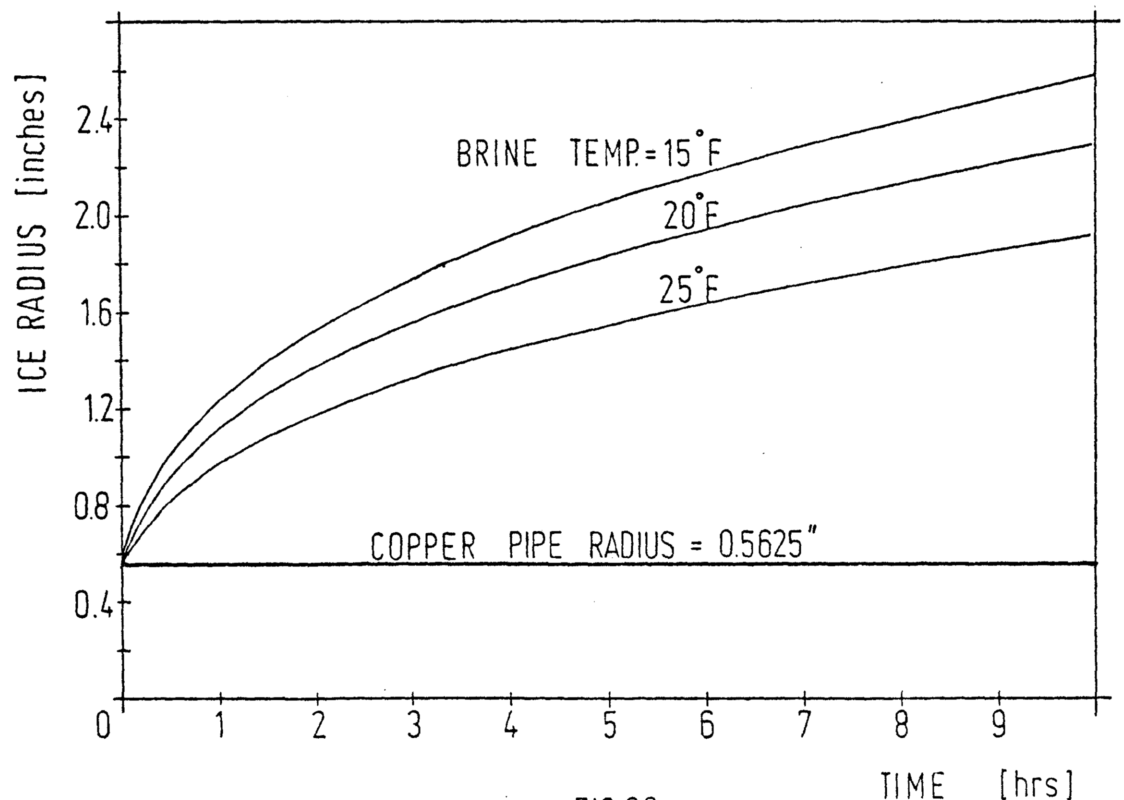


FIG 28
ICE FORMATION vs. TIME

The plastic pipe considerably retards the formation of ice due to its poor thermal conductivity for 5/8" pipe at 20°F. The copper pipe will form 77% more ice in 10 hours than the plastic pipe which means 77% more plastic pipe is needed to make the same amount of ice per unit of time. By knowing the ice radius, the proper pipe spacing is determined (double the radius). Figures 26, 27, 28 show ice radii for other pipe sizes and brine temperatures.

This data is compiled in a more usable form in Table 11. The optimum pipe spacing is listed along with the amount of heat stored per foot of pipe and the amount of pipe needed to store one million BTU. This information coupled with the cool storage capacity needs, provides the designer with the quantity of pipe needed and its proper spacing for a particular size of heat exchanger.

TABLE 11. DESIGN VALUES FOR AN ICE STORAGE SYSTEM USING A PIPE HEAT EXCHANGER (10 HOUR CYCLE)

Brine Temp. °F		O.D. Pipe Size (in)			
		7/16	5/8	7/8	1 1/8
	A	2.9	3.2(2.3)	3.5(2.8)	3.8
25	B	361.0	433(215)	505(300)	579
	C	2770	2309(4651)	1980(3333)	1727
	A	3.5	3.8(2.9)	4.2(3.4)	4.5
20	B	530	618(350)	742(470)	835
	C	1887	1618(2850)	1348(2128)	1198
	A	4.0	4.4	4.8	5.1
15	B	695	834	980	1088
	C	1439	1199	1020	919

A - Optimum pipe spacing, center to center (in.)

B - Heat stored per foot of pipe (BTU-heat of fusion only)

C - Feet of pipe per million BTU of storage

( ) - Polyethylene pipe

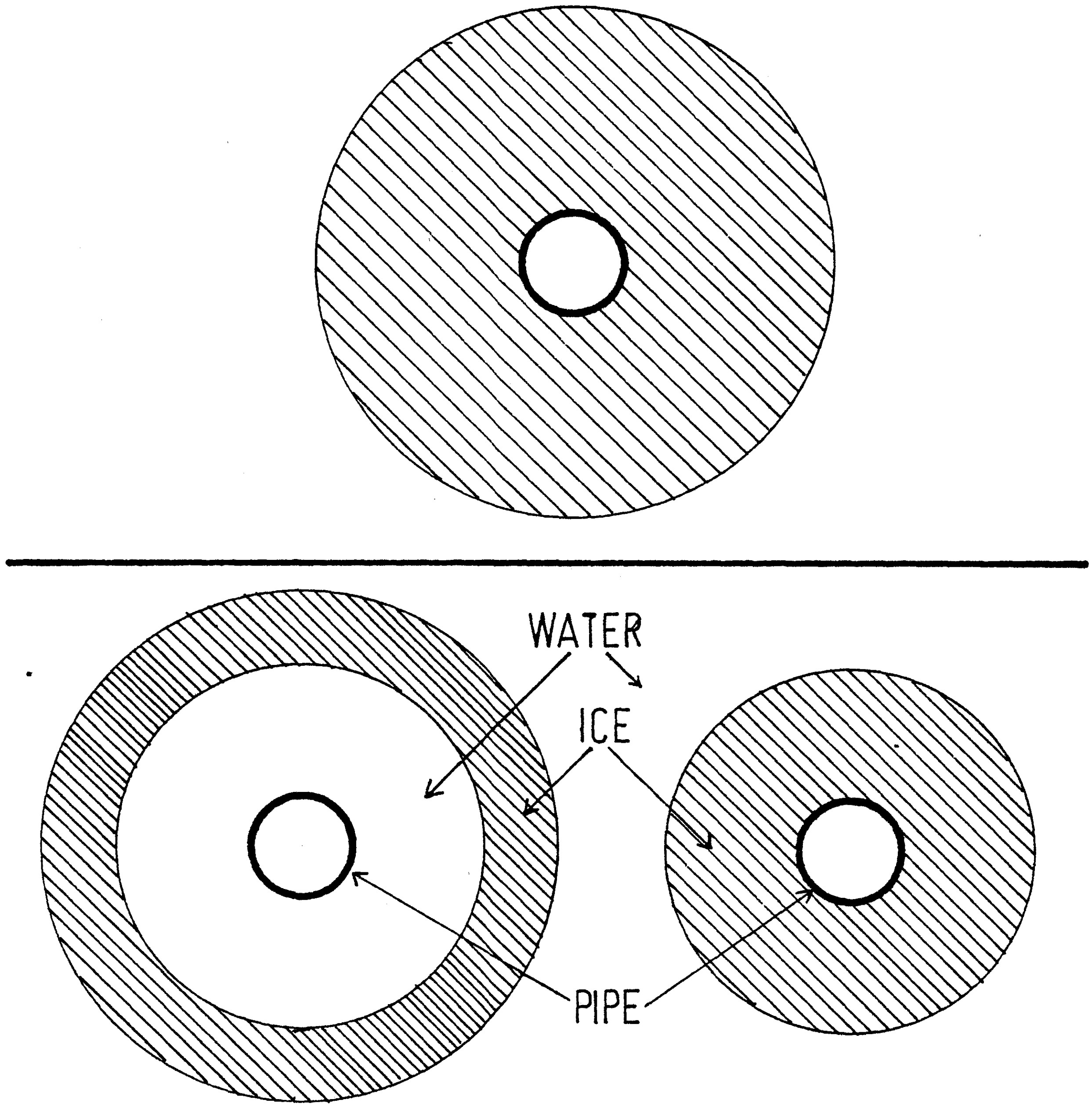
ANALYSIS OF HEAT ADDITION PROCESS

During the daytime hours, when the electrical rates are at a premium, the fully charged ice storage bank will be used to absorb the building's internal and weather related heat gains. There are two ways in which a pipe heat exchanger system can transfer the building's heat to the ice storage, indirectly, as in Fig.15, and directly, as in Fig.16.

With the indirect system, the brine is circulated through a shell and tube heat exchanger where it absorbs heat from the warm (60°F) water returning from the air handling units. The brine then passes through the now inactive chiller heat exchanger (or may bypass it with proper valving) and flows through the piping in the storage tank. The heat is tranferred through the pipe wall to the ice which melts from the pipe wall outward (Fig.29). The rate at which heat is transferred depends on the surface area of pipe, pipe material, brine flow rate, and the amount of ice already melted. Suitable methods for modelling the heat addition mode of the indirect system have been formulated[1], but are not addressed in this work.

A number of indirect discharge cycles were run with the experimental apparatus (Fig.18). Data from a representative run is listed in Table 12. A pump (P_2), separate from the brine loop, was switched on periodically to observe the effect that forced convection of the water between the pipe wall and ice layer would have on the heat transfer rate. The inlet and outlet brine temperatures remained around 50°F with the inlet to outlet differential varying from 1 to 3.8°F The heat flow (BTU/hr) was calculated in the same manner as in the heat removal mode (see equation 7 page 68). Likewise, the experimental heat transfer coefficient (BTU/hrft°F) was formulated as in equation 8. Fig. 30 shows how the heat transfer in the indirect discharge system changes as the ice cylinder melts from the inside out. Heat transfer is initially very high due to the low thermal resistance, consisting primarily of h_{in}. As the ice begins to melt, the water layer now separaring the pipe from the ice inhibits the heat flow. After approximately 2 hours, the heat flow begins to pick up. This occurs at a point where the ice has melted out to a radius of approximately 0.9". Evidently, convective heat transfer begins to supplement the heat flow enough to more

CROSS SECTION of a FULLY CHARGED ICE CYLINDER



50% DISCHAGED
via
INDIRECT MODE

50% DISCHARGED
via
DIRECT MODE

FIG 29

DISCHARGE MODES for the ICE SYSTEM

TABLE 12. HEAT ADDITION TO ICE STORAGE SYSTEM (INDIRECT MODE)

Time (hr)	Mass Flow (GPM)	Temp; °F Tank Water	Inlet Brine	Outlet Brine	ΔT Inlet-Outlet	LMTD	Heat Flow (BTU/Hr)	Ice Thickness (in)	Experimental Heat Transfer
.25	3.28	32	51.80	48.38	3.42	18.0	5585		6.0
.50	"	"	62.24	59.18	3.06	28.7	4997		3.4
1.00	"	"	54.50	52.34	2.16	21.4	3527		3.2
1.50	"	"	51.80	49.64	1.44	18.4	2351		2.5
2.00	3.35	"	50.72	49.46	1.26	18.1	2102		2.2
3.50	3.48	"	51.80	50.36	1.44	19.1	2495		2.5
4.00	"	"	52.34	50.72	1.62	19.5	2807		2.8
4.50	"	"	52.16	50.36	1.80	19.2	3119		3.1
4.75	"	"	51.98	50.18	1.80	19.1	3119		3.1
*5.00	"	"	53.96	50.18	3.78	20.0	6550		6.3
*5.33	"	"	51.08	47.48	3.60	17.2	6239		7.0
5.50	"	"	53.42	51.62	1.80	20.5	3119		2.9
7.75	"	"	51.98	50.72	1.26	19.3	2184		2.2
*8.0	"	"	51.62	49.28	2.34	18.4	4055		4.2
8.2	"	"	53.06	51.80	1.26	20.4	1872		1.8
8.6	"	"	52.70	51.62	1.08	20.2	1872		1.8

*Pump (P_2) on

TABLE 13. HEAT ADDITION TO ICE STORAGE SYSTEM (DIRECT MODE)

Time (hr)	Mass Flow (GPM)	Temp; °F: Tank Water	Temp; °F: Inlet Brine	Temp; °F: Outlet Brine	Temp; °F: ΔT Inlet-Outlet	Temp; °F: LMTD	Heat Flow (BTU/Hr)	Ice Thickness (in)	Experimental Heat Transfer
	3.48	32	36.75	34.15	2.6	3.28	4505	15/16	26.4
	3.48	"	37.07	34.47	2.6	3.61	4505	13/16	24.0
	3.48	"	37.35	34.85	2.5	3.97	4332	5/8	21.0
	2.13	"	38.10	34.0	4.1	3.67	4349	15/16	22.8
	2.13	"	39.0	34.61	4.4	4.44	4667	13/16	20.2
	2.13	"	39.25	35.05	4.2	4.85	4455	5/8	17.7
	0.90	"	43.3	33.6	9.7	4.96	4348	15/16	17.5
	0.90	"	43.58	33.98	9.6	5.43	4302	13/16	15.8
	0.90	"	44.0	34.51	9.5	6.06	4258	5/8	14.0

than make up for the increasing resistance associated with a purely conductive path. At the fifth hour, the small circulator (P_2) was turned on. Tank water was drawn out of the bottom of the tank and released at the top, thereby increasing the heat flow by 100% by raising the convective heat transfer component. When the pump was turned off, the heat flow dropped down to 3000 BTU/hr. Later on the pump was switched on again with similar results.

By directly discharging the ice storage tank much higher heat flows are possible (Table 13). Data was collected at three points during the discharge cycle, when the ice was 15/16", 13/16" and 5/8" thick. At each point the LMTD was checked at 0.9,2.13 and 3.48 gallons per minute. With

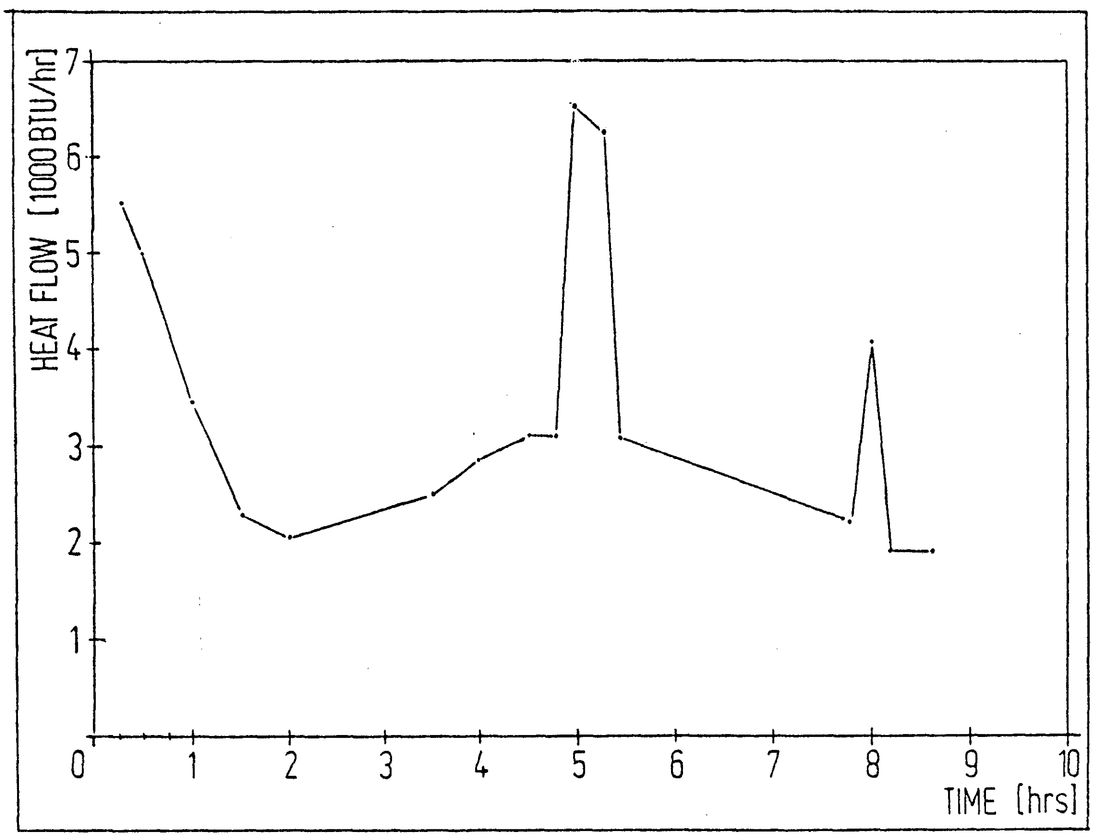


FIG 30

DISCHARGE of ICE STORAGE via INDIRECT MODE the LMTD and inlet/outlet temperature difference, the experimental heat transfer was calculated. As the mass flow declined, the heat transfer coefficient also declined, as result of the reduced fluid velocity (lower reynolds number). Also, as the diameter of the ice cylinder shrunk, the heat transfer coefficient dropped further, resulting from a combination of reduced ice surface area and a decreased fluid velocity. Nevertheless, the heat transfer coefficients were much higher than with the indirect discharge system, as shown in Fig. 31.

The ability of the ice storage system to absorb the building's heat as it is generated is critically important to the overall success of the idea. If the system has the thermal storage capacity, but not the necessary heat transfer capability to meet the peak loads, then the advantage of this concept is greatly diminished. This is because in such a situation, the chillers will have to be operated during the utilities' peak hours resulting in an increased energy charge and possibly a higher demand charge in addition. For this reason, it is highly desirable to design the storage system so that it can meet the maximum anticipated peak cooling load. Referring again to the energy analysis of the 757,000 ft^2 office building in Boston,[3] the cooling loads amounted to a total of 210 BTU/ft^2 for the hours of 6AM to 8PM (considered as the utilities' peak period). The cooling peak occurred on August 17 at 2PM and amounted to 21 $BTU/hrft^2$. A cool storage system for this building would require a capacity of 210 BTU/ft^2 of conditioned space and would need to have the capability of releasing 10% of its capacity per hour (210 $BTU/ft^2 \div 21$ BTU/ft^2 = .1 or 10% per hour).

With a capacity of approximately 25,000 BTU in the experimental apparatus, the minimum acceptable discharge capability is 2500 BTU/hr. In the indirect discharge mode without the pump on, the discharge capability drops to approximately 2 $BTU/hrft_{pipe}{}^\circ F$. With 52 feet of pipe and a maximum log mean temperature differential of 18°F, the discharge capability equals 1872 BTU/hr. The 18°F LMTD is determined by the fact that the ice storage is fixed at 32°F and in order to supply 55°F chilled water to the air handling units, the maximum average brine temperature is considered to be 50°F. Therefore, it is reasonable to conclude that the indirect discharge working under natural convection is not satisfactory. It appears that the indirect discharge with forced convection would be

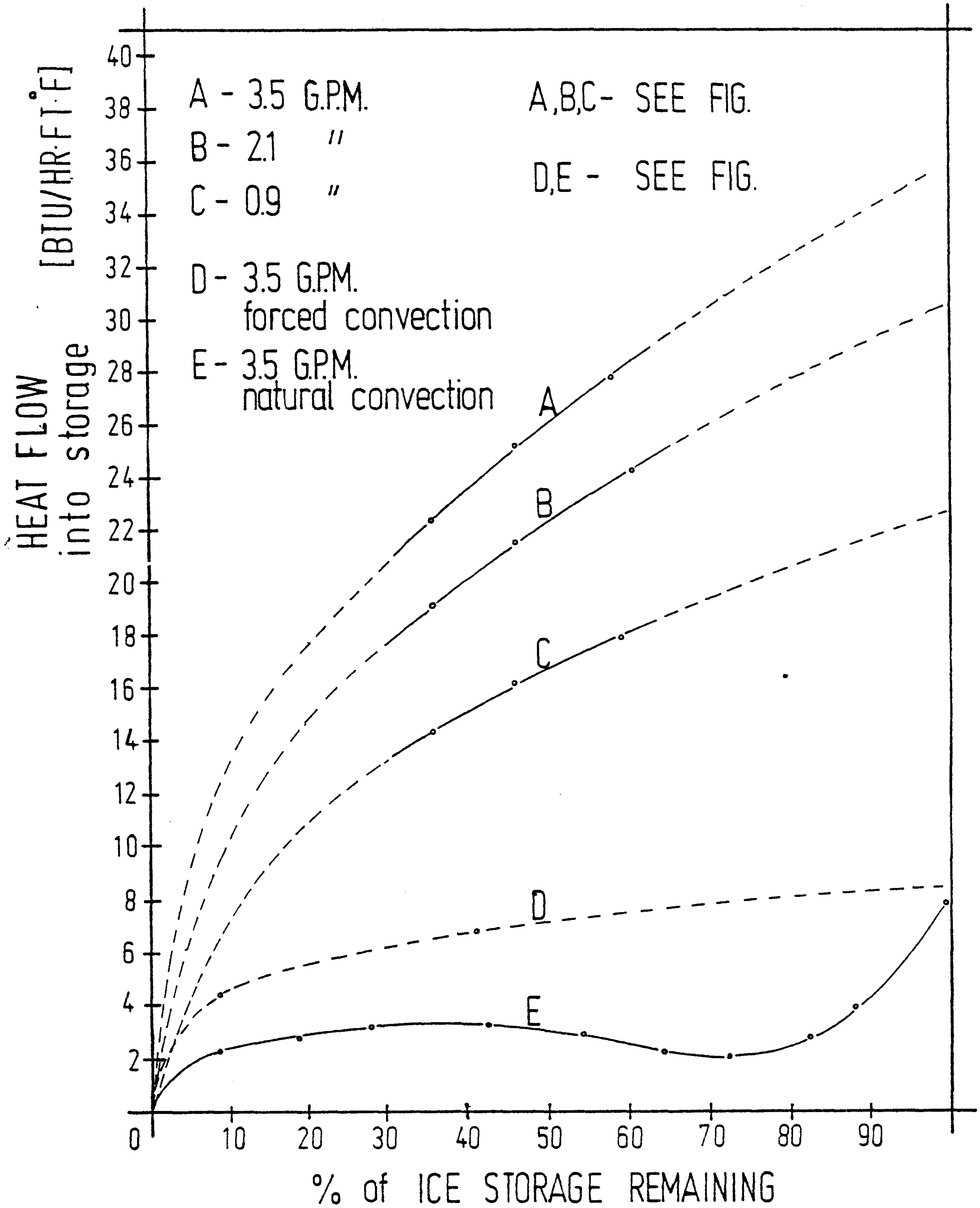


FIG 31
DIRECT vs. INDIRECT
DISCHARGE of ICE STORAGE

able to absorb any anticipated cooling load. This is only true if the tank water being pumped is able to enter the space between the pipe wall and the ice cylinder throughout the course of the discharge period. The piping would have to be designed to facilitate this.

The direct discharge design can very easily handle any conceivable cooling load. In fact, even at .9 g.p.m. flow rate, the heat transfer coefficient is 2.5 times that for the indirect system with a flow rate of 3.5 g.p.m. and operating under forced convection. The impact of this is that the pumping capacity is greatly reduced resulting in a smaller horsepower requirement, lower operating costs and smaller pipe size. It would be very simple to design for the direct discharge of the ice tank, leaving no reason to even consider using the indirect discharge approach.

REFERENCES

1. Bligh, T., "Heat Transfer in a Seasonally Charged Ice Storage Bin," University of Minnesota, Department of Mechanical Engineering, 1977 (unpublished).

2. Raithby, G.D., Hollands, K.G., Advances in Heat Transfer II,

3. C. Benton, Off-Peak Cooling Using Phase Change Materials, MIT Department of Architecture, M.Arch.A.S. Thesis, 1979.

DESIGN of a SYSTEM PROTOTYPE

In order to accommodate the utilities' off-peak schedule, a cool storage system must be able to charge and discharge within the time constraints imposed by the utilities' 10 hour off-peak period and the building's peak hour cooling load profile which may demand cooling at a rate 50% greater than the nighttime charging rate. In order to avoid turning on the chillers during the daytime, which would increase the demand charges and require expensive power, the storage system's discharge capabilities must be greater than its charging abilities.

Experimental data and modelling has shown that, with proper heat exchanger sizing, 10 hour system charging is feasible. In turn, by directly discharging the storage tank, heat transfer rates greater than any anticipated building load are possible. The temperature at which the tanks heat exchanger is operated directly determines the rate of ice formation. It also has an impact on the COP of the chiller. As illustrated on page 29, by operating the chillers at night during off peak hours, the evaporator temperature can be reduced by 15°-20°F without substantially reducing the chiller's COP, due to the lower ambient temperatures.

Strictly from an economic point of view, it would make sense to reduce the evaporator temperature even further. Since the off-peak electricity is so cheap, it would not increase the building operating costs significantly and in addition, it would enable the designers to install a smaller quantity of heat exchanger piping, lowering the overall capital costs. However, while there is some merit to this point, the ultimate goal of this storage system should be to reduce the consumption of power while also reducing costs. In order to accommodate both of these objectives, the minimum evaporator temperature to be considered for purposes of this thesis is 15°F. A 5°F minimum required temperature differential between the evaporator and brine loop results in a minumum brine temperature of 20°F.

Increasing the pipe diameter offers a greater initial surface area for the ice to form on, thereby allowing a greater heat flow during the charging period. Because of the higher heat flow per foot of pipe, the pipes can be spaced further apart resulting in a smaller quantity of pipe needed per unit of storage capacity. An additional benefit results from the fact that less pipes mean that less connections need to be made in the assembly process. Assembly costs can easily match, if not exceed, the materials cost. It should be obvious, however, that larger pipes cost more per foot. As long as the increased cost per foot for the pipe is cancelled out by the reduction in the overall number of feet required, it is probably worthwhile to use the larger pipe. The pipe size also affects the fluid velocity, which in turn, affects the convective heat transfer for the inside of the pipe (h_{in}). As a pipe gets larger, the fluid velocity drops, reducing the Reynolds number and introducing a larger thermal resistance. However, as the pipe gets larger, the mass flow must be higher in order to absorb the increased heat transfer over a fixed temperature differential, thus counteracting, somewhat, the reduction in velocity due to increased diameter. Thus point warrants a more detailed examination.

In the experimental apparatus, 52 feet of 1/2" I.D. copper pipe was used as the tank's ice coils. At a 1" ice radius, the heat flow was 6.5 BTU/hrft°F (Fig. 20). If the piping had been replaced with 3/4" I.D. pipe, the heat flow would have increased to 8.8 BTU/hrft°F. In order to accommodate this increased heat flow over an equivalent temperature differential as before, the mass flow would need to be increased by 35%. However, due to the increased cross-sectional area of the pipe, the fluid velocity would drop by 40%. Solving equation 6 (page 66), the h_{in} is found to drop by a corresponding 40%. In order to raise h_{in} back to its previous value, a larger, more energy consumptive pump is required. Eventually, the savings in heat exchanger costs are overshadowed by the increase in capital and operating costs for the pump.

The choice of pipe material should be limited to copper and plastic. Steel pipe is not competitive with copper below 1" and aluminum piping is very expensive and susceptible to galvanic corrosion. Polyethylene plastic pipe is commonly used in buildings for domestic water supplies and waste stacks. It is very cheap and quite durable, and is compatible with methanol or ethylene glycol which would be used as the the brine fluid. Referring again to Table 11, page 85, the lineal feet of pipe needed per million BTUs is given for each pipe size. In comparing 5/8", 7/8" O.D. copper and plastic pipe, and the respective costs, it quickly becomes clear that although the copper pipe system requires only 60% as much piping as the plastic system, the low costs for plastic pipe more than make up for the heat transfer deficiency.

TABLE 14. PIPING COSTS PER MILLION BTU OF STORAGE AT 20°F BRINE TEMPERATURE

Material		Required Feet	Cost per foot ($)*	Total Cost ($)
COPPER	5/8" O.D.	1618	.62	1003
	7/8" O.D.	1348	.95	1280
POLYETHYLENE	5/8" O.D.	2857	.109	311
	7/8" O.D.	2128	.133	283

*Prices obtained from local pipe wholesalers for standard lengths or rolls.

The polyethylene pipe is a much more cost effective material for use as an ice coil for this system. It can be easily extruded into a variety of different heat exchanger shapes and geometries while the copper piping must be fabricated in a metal shop, requiring much more time and labor. For small residential applications, a dx evaporator in place of the brine loop is more sensible. However, the evaporator would have to be made from copper tubing rather than polyethylene. Halogenated hydrocarbons attack polyethylene, and plastics suffer from water permeation which would damage the refrigeration components. For all but these small applications, plastic is the preferable choice of material for use in the ice storage system.

The optimum shape for the ice forming surface is determined by a combination of economics and heat transfer considerations. Simple round piping is not the most efficient surface geometry, but it is most likely the cheapest. In the first ACES design, the ice coil consisted of an extruded aluminum fin tube (Fig.32). The purpose of the fin was to increase the surface area of the heat exchanger,and thereby the heat transfer rate. Theoretically, the number of feet of pipe required should be reduced. Investigations conducted at the University of Minnesota[1] indicated that the effect of this fin in this geometry was relatively low, In fact, at an ice radius of 1", the rate of heat transfer in the fin tube was 65% greater than with conventional pipe. When the cost for fin tube piping versus a regular piping are examined, it becomes obvious that the fin tube does not make economic sense. For a plastic pipe heat exchanger, a plastic fin would be totally ineffective due to its extremely poor thermal conductivity. A simple round, plastic pipe is apparently the most cost effective approach for large commercial ice storage systems employing a piped ice coil heat exchanger. 20°F brine circulated through a 7/8" O.D. plastic pipe for a 10 hour charge period dictates that the pipe spacing should be 3.5" on center. Ice will form out to a diameter of 3.4" leaving 0.1" of space between the ice cylinders in order to permit movement of water around the ice. The height of the ice coil piping within the tank depends on what tank depth the building can accommodate. Normal floor to floor heights are 10'-12'. Sub-basement heights are not as restricted in this

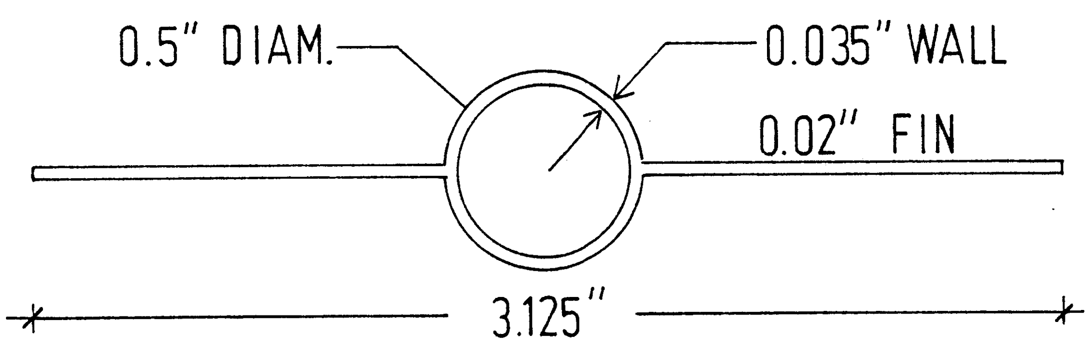


FIG. 32

ACES HEAT EXCHANGER TUBING — CROSS SECTION

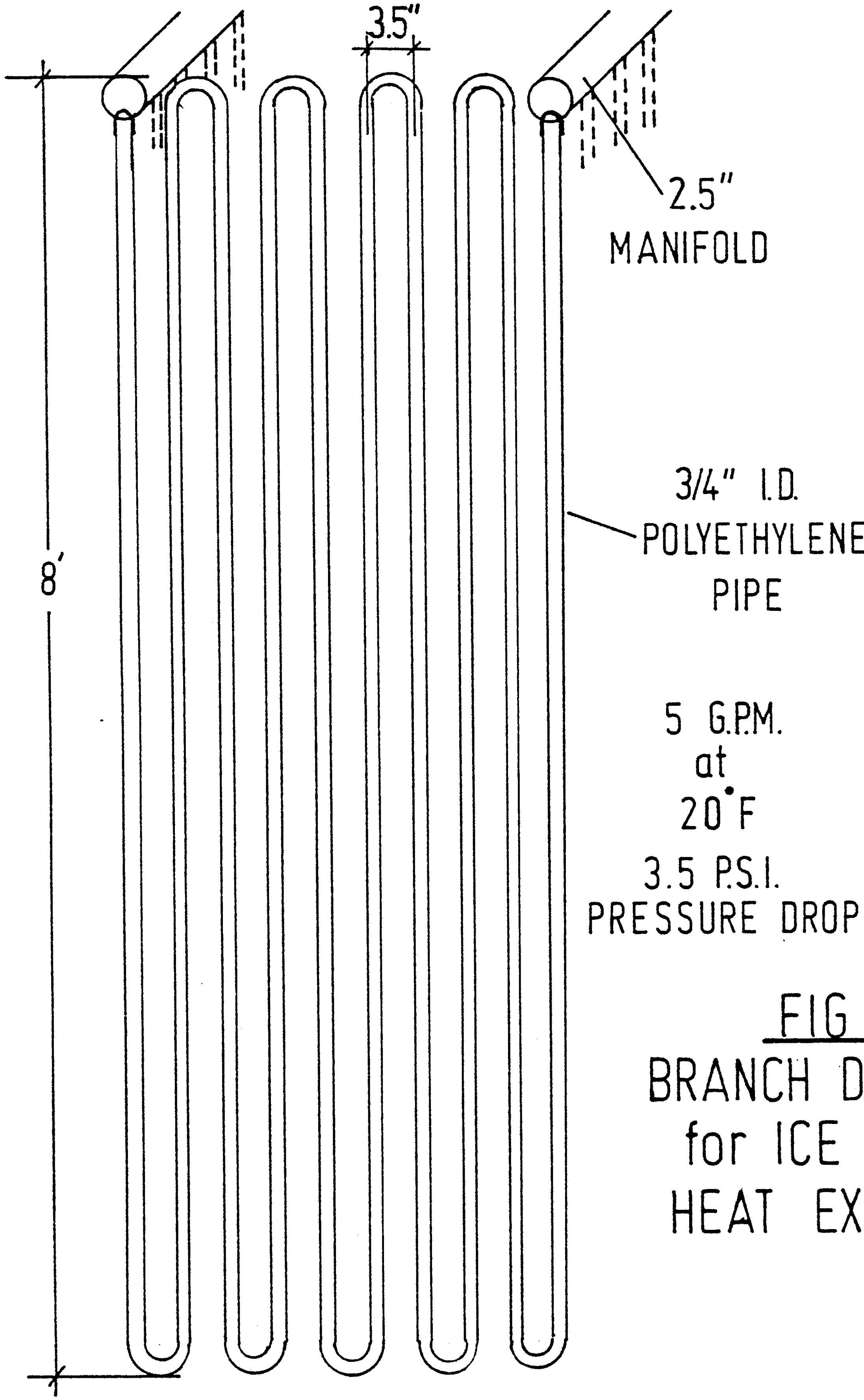


FIG 33
BRANCH DIMENSIONS
for ICE COIL
HEAT EXCHANGER

manner. A 10 foot tank depth is chosen for purposes of this design, though not necessarily restricted to this dimension. Some of the volume within this 10 foot height must be reserved for the supply and return manifolds and the 9 inch fluctuation in water level due to the volumetric expansion of the water. 2 feet of depth will easily accommodate these needs leaving the remaining 8 feet available for ice coil piping (Fig.33). An 80 foot long serpentine length of 7/8" pipe is the optimum branch unit. The total length of branch piping affects the temperature increase and pressure drops between the supply and return manifold. In order to minimize the temperature increase, a large fluid flow is required. However,

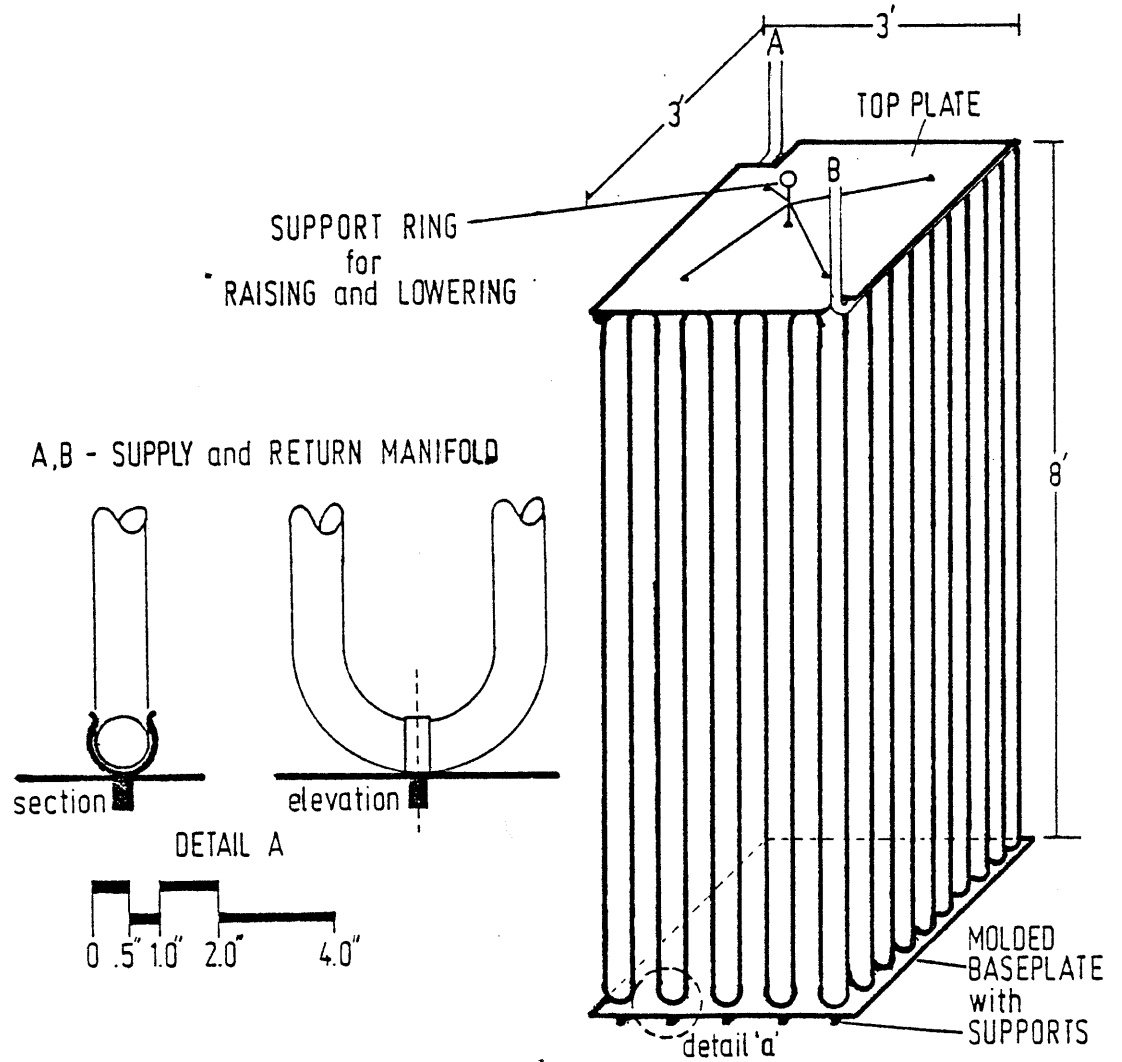


FIG 34

ICE COIL MODULE (376,000 BTU, h_f ONLY) a large mass flow results in a high fluid velocity and increased pressure drops. This necessitates a larger, more expensive circulating pump. A fluid velocity of 5 ft/sec should not be exceeded in plastic piping.[2] A mass flow of 5 gallons per minute at 20°F through an 80 foot long 7/8" O.D. pipe will experience a temperature rise of 1.2°F at the beginning and 2.0°F at the end of the freezing cycle. At that flow the velocity will be 3.6 ft/sec., and the pressure drop will be 3.5 psi (equivalent to a head of 8 feet of water). The energy required by a pump to move the fluid through the pipe is 1.5% of the amount of heat being transferred. With ten 80 foot branches per module (Fig. 34) a 2.5" I.D. manifold is required to insure uniform flow through each branch. It would be convenient if the manifold and branch pipes could be extruded as 1 piece. If this is not technically or economically feasible, the branches could be assembled into a module very quickly with an appropriate jig. In order to insure uniform spacing of the pipes, molded plates with clips for the top and bottom will be used (Fig. 34). The baseplate has 1" feet on the bottom in order to keep the ice forming surfaces away from the containment walls. The top plate includes an attachment which enables the unit to be raised and lowered from the storage tank. The weight of each module is 70 lb empty and 230 lb when filled with brine. The lifting straps must be capable of supporting this full load since the module will be full if removal is necessary.

Breaking the ice coils into discrete modules is an absolute necessity for commercial scale applications. While it is possible to build the storage coil 'in situ' as one large assembly, future repair of pipe leakage would necessitate the disassembly of a substantial portion of the piping in order to simply gain access to the problem spot. As shown in Figs. 35 and 36, the modules can be removed from the storage tank without affecting the rest of the unit by valving off the 4 module leg, decoupling the defective module from the 5" module manifold, and lifting out. Before opening the module couplings, the 5" manifold must be emptied of its brine by draining through a waste vent on the gate valve located at the end of each module manifold. (The flow regulator can double as a shut off valve on the supply side of the module manifold.) A new module is installed in reverse order.

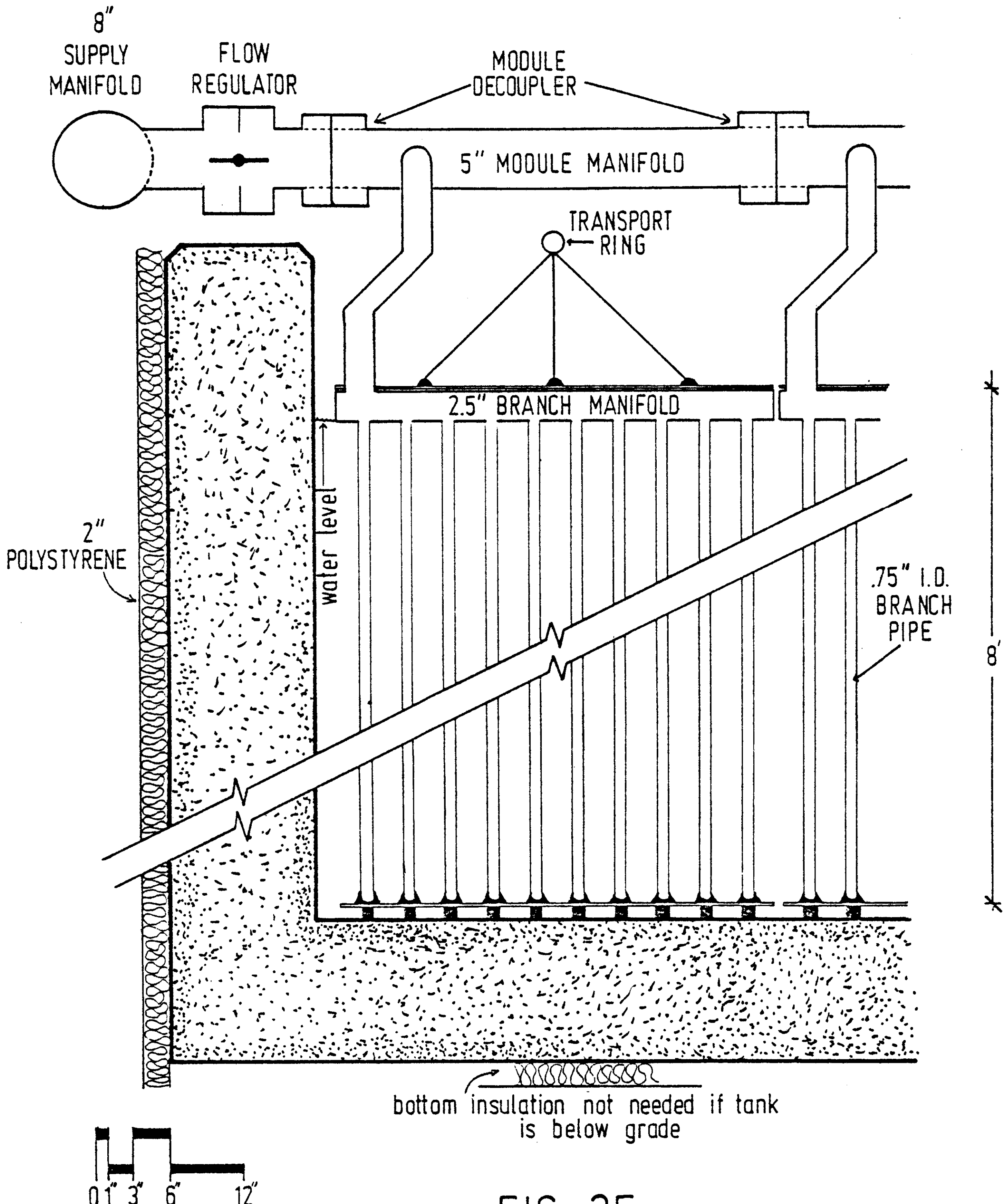


FIG 35
SECTION 'a' THROUGH
ICE STORAGE TANK

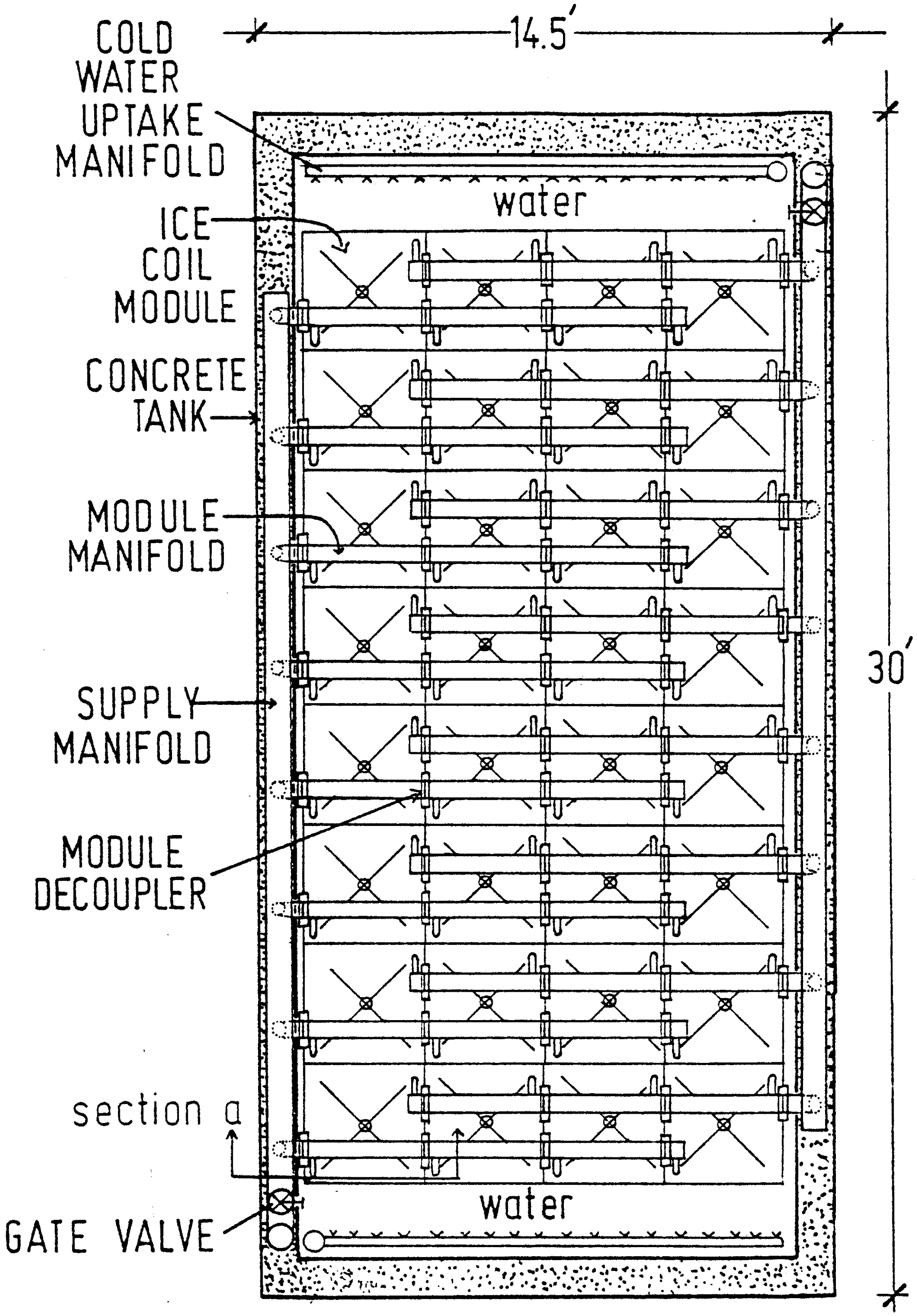


FIG 36
PLAN VIEW of STORAGE TANK

The module decouplers are necessary in order to remove the individual modules from the tank. Module decouplers would not be necessary if the module manifolds were not located directly above the module but instead, to either side. However,since there are other modules on both sides, a 1 foot spacing between modules would be necessary to insure free access to each unit. This would simplify the module manifold detail but add about 25% to the tank volume. One of the main advantages of this storage system is its volumetric efficiency;and it therefore seemed important to couple the modules in a manner that would maintain this efficiency.

The modules offer other cost savings over a site-built unit. Modules can be fabricated easily at a factory in a minimum of time and materials. The units can then be shipped to the site where they are assembled in the already constructed storage tank. Undoubtedly, this would be cheaper than field assembly of a single large unit. Modules allow for flexibility in enabling the designer to add however many modules are required to satisfy the storage capacity requirements.

The size of the module manifold will vary according to the number of units fed off the manifold. In order to insure uniform flow to each module, the module manifolds inside cross sectional areas must be equal to the sum of the branch manifold areas being fed. In this design, that necessitated a 5" module manifold. With coupling between the module and branck manifolds, new designs with different module manifold sizes can be used.

The main supply and return lines to the module manifolds should be sized such that pressure drops are kept to a moderate level. The pipe used in Fig. 35 is 8" in diameter (flow in the pipe is 1600 GPM). Flow requlators are employed at the beginning of the module manifolds to insure uniform fluid flow.

During the discharge mode when heat is transferred to the storage tank, water drawn into the cold water uptake manifold is pumped through a shell-in-tube heat exchanger where heat from the air handling unit is absorbed. The warm water then returns to the storage tank through another pipe manifold. By the time the water reaches the uptake manifold, it has been cooled to 32°F. Assuming a 10°F temperature differential at the heat exchanger feeding the air handling unit, and assuming that a maximum hourly demand will be 15% of total storage (1,800,000 BTU) then the coolant flow must be 360 GPM. A 2.5" pipe will be sufficient to carry this flow. The resident time of the water in the tank will be 18 minutes at the start and 72 minutes at the end of the discharge mode. The route around the ice cylinders will insure very high heat transfer. Because of this, the ice cylinders will not melt uniformly throughout the tank but rather from one end to the other. After days when the tank was not fully discharged, the brine will extract heat from all the ice cylinders, regardless of the ice thickness. It is necessary to stop the freezing process once the ice has reached a diameter of 3.4". Otherwise, the ice will meet the adjoining cylinder and obstruct the water flow path. To prevent this, thermocuples on the thermistor should be placed alongside a pipe in the middle of the one module in each row. The sensor will close the flow regulator to the module manifold when the temperature has reached a preset point. When the last valve is closed, the system is fully charged.

Details on tank construction are beyond the scope of this work. However, it is important to recognize that tremendous savings will be realized if the tank is incorporated into the structural system of the building. Overflow ports must be built into the tank to prevent flooding in case of pipe failure. In addition, the tank must be drainable in the event that tank reapairs need to be made. This is one of the tremendous advantages of using water as a PCM. If the tank needs to be drained the water can be replaced at a very low cost whereas other PCM would need to be stored for the interim due to high replacement costs. In addition, a 376,000 BTU module weighing 70 lb can be shipped from the factory without the PCM, whose delivery is facilitated by the building water service.

Heat loss from the tank is not a significant problem. The heat stored in a fully charged tank equals 16,000,000 BTU(sensible and latent heat). The surface area of the walls is 1760 ft^2 (tank top included). Assuming an ambient temperature of 72°F (40°F temperature differential), the heat loss through uninsulated walls would amount to 1,600,000 BTU/24 hr. By adding 2 inches of polystyrene to the surfaces, the heat loss drops to 160,000 BTU/24 hr. Since this is only 1% of the equivalent heat storage, no further insulation is recommended. If the tank is located below grade, the bottom of the tank need not be insulated. The equivalent resistance of the ground heat path to ambient will be greater than 2" of polystyrene.

SYSTEM DESIGN IMPROVEMENTS

The plastic coil ice module storage system is capable of storing 6470 BTU/ft^3, which includes the latent heat of fusion and the sensible heat of the water raised from 32°F to 52°F. Its heat storage density is 500% greater than the cool water storage system and 23% greater than the ice maker heat pump. The module is also compatible with packaged chillers. Despite these advantages, the storage system is quite complex. Over the course of a system's life, complexity usually generates unanticipated costs. 2 ways in which this system could be made more attractive would be to remove the heat exchanger from the tank making it more accessible, and to reduce the surface area of heat exchanger required. This must be done while not driving up equipment costs for the chillers. Theoretically, the ice maker heat pump concept could be adapted in a way that would make it compatible with packaged chillers while also improving the operating efficiency.

This new design would call for packaged chillers to supply ice forming plates with a 20-25°F brine. The plates would be coated with an ice resistance film which would cause the ice to flake off soon after forming, falling into the storage tank below. As a result of the minimal ice build-up extremely high heat transfer rates are possible, reducing the surface area of plate required. Alternately, if site-built chillers prove to be competitive with the packaged units (usually only when the capacity is greater than 300-400 tons) than refrigerant could be supplied to the plates directly, eliminating the brine loop and improving system COP.

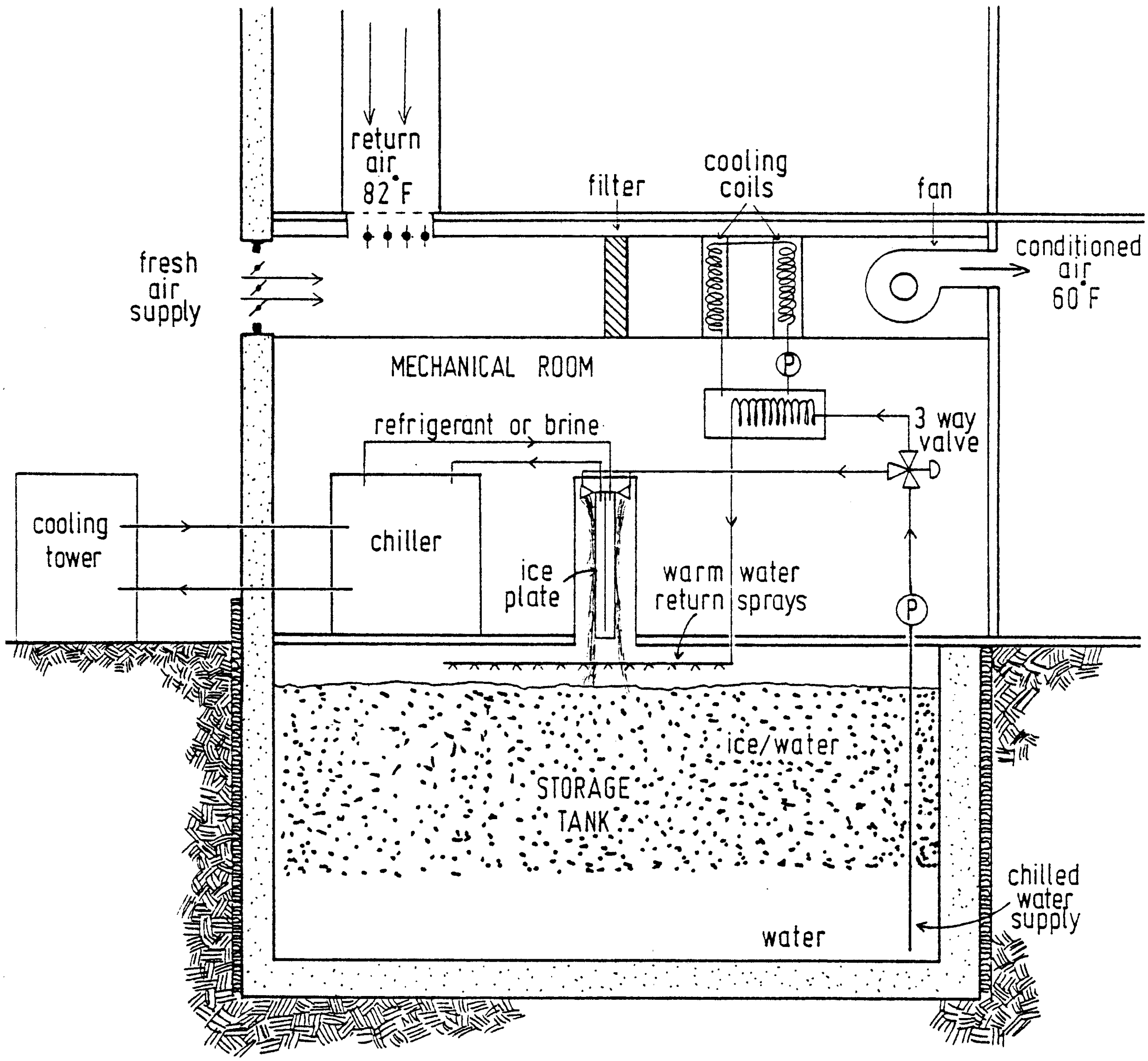


FIG 37

ICE STORAGE USING A FROSTLESS ICE PLATE

Fig. 37 details such a system. It is important that the ice resistant film be durable enough to withstand the mild abrasion of water and ice flakes, and thin enough to permit high heat transfer. It is conceivable that the plates could be removed periodically for recoating if that proved necessary. The plates would be located in the mechanical room for ease of access, and mounted vertically over openings in the tank, with water spraying both sides in order to take advantage of all the surface area. Ice would form readily on the 20°F-25°F surface and flake off into the water spray returning to the tank. Being lighter than the water, the ice flakes would float in the upper layer of the tank. As the formation of ice progressed, the water level in the tank would drop. If the water uptake pipe were designed properly, the velocity of the water entering the pipe would be low enough to permit the ice/water layer to approach within inches of the inlet.

Due to the high heat transfer across the ice plate, a large mass flow of refrigerant or brine would be required to keep the ΔT low. This large mass flow will result in a very high convective heat transfer between the fluid (refgigerant) and the plate, reinforcing the system's already high heat transfer. Because of the limited number of plates that would be required, the piping complexity will be substantially reduced.

Cooling needs for the building will be satisfied in a conventional manner. The warm, moist return air will be cooled and dehumidified by the cooling coils in the air handling units. The coils will be supplied with with 50°F water which has been chilled by the 32°F tank water in an intermediate heat exchanger. Because of this high ΔT, a smaller, cheaper heat exchanger may be used. In addition, the fluid flows will be lower, requiring smaller pipe. After the water leaves the heat exchanger, it returns to the tank through spray nozzles mounted inside the tank. This would result in a uniform and efficient melting of the ice. By the time the water had maneuvered its way through the ice/water layer it would have been chilled back to 32°F.

Assuming that a suitable ice resistant coating exists, it seems certain that this type of system would easily outperform the ice maker heat pump, the ice coil system, and the cool water storage designs. It takes advantage of the high heat storage capacity of PCM, overcomes the poor heat flow problems at the ice plate without expanding valuable energy, reduces piping and heat exchanger sizes, is relatively simple to construct and operate and is compatible with packaged chilled water equipment. It undoubtedly would make a strong impact on the market.

ECONOMIC ANALYSIS

Off peak cooling systems are beginning to be included in the designs for new commercial structures. This is not happening due to the development of some new technology, but rather because of increasing electric rates. Cool water storage systems have been used in Europe and Japan for many years with great success and are only now being seriously considered here in the U.S. Rising utility bills have provided the justification for designers to consider the extra capital expenditure embodied in off-peak cooling systems. More efficient, compact systems will undoubtedly be developed as the need for cool storage grows. If the recent trends in electric power cost escalations continue, this growth will be insured.

At present, the utilities in New England generate 35% of their power in nuclear power plants and the remainder with oil fired plants.

The nuclear plants run at full output 24 hours a day while the output from the oil fired plants are modulated according to demand. Peak power needs are met with inefficient gas turbines which operate only a few hours of the day. Consequently, the peak power costs more to produce since it is tied directly to the cost of fuel oil. In addition to this, peak power demands also require the utility to oversize feeder lines, transformer and switching yards, and other service equipment just to meet this infrequent demand. Electric utilities have attempted to reflect these costs in their rate structures with varying success. Time of day rates are being introduced by a number of utilities around the country in an attempt to encourage load management. To date, these types of rates have remained optional, with the vast majority of customers preferring the traditional declining block rate structure devoid of any time of day cost constraints. For this reason the discussion in this section will be limited to an analysis using the present rate structure for a test building in Boston.

This building offered detailed records of mechanical system operation and gross electrical consumption. In review, the building is a commercial office structure in the Boston Metropolitan area. This building has 757,000 SF of conditioned space on six floors. The occupancy represents a typical range of speculative office tenants from insurance agencies to computer firms. The building is arranged about a large central court. Exterior walls have a moderate ratio of glazed area to solid wall, with fenestration minimal on the southern exposure. Building management has actively pursued programs aimed at reducing energy waste, with moderately successful results. This building was selected for this study because it typifies the type of project well suited to off-peak cooling.

RATE STRUCTURE FOR TEST BUILDING

Most electric utility rate structures are organized on a step basis. The first units of energy purchased are more expensive than the next set of units, and so on in a series of predetermined blocks. When energy use is reduced, the energy is saved from the last block of usage. To determine the savings afforded by a reduction, the cost of energy purchased in the last block must be calculated and is referred to as the incremental energy cost.

Energy consumption for the model building is billed at the Massachusetts Electric Rate H. There is a charge, under this rate, for monthly consumption, measured in KWH. There is also a demand charge based on either the highest demand during a fifteen minute interval during the month, or on 80% of the highest demand measured in the preceeding 11 months,known as a 'peak ratchet'. There is a direct demand charge of $1.57 per KW and an implicit demand charge since the size of the billing blocks for electrical consumption is determined by the monthly demand. A fuel adjustment is added to the bill to allow the utiliy to pass along variations in the price it pays for the fuel used to generate the electricity.

TABLE 15 MASSACHUSETTS ELECTRIC COMPANY COMMERCIAL RATE H

Demand Charge

$830.00	for the 1st 500 KW or less
1.57	per KW for the excess

Energy Charge per KWH

0.02648	for the 1st 50,000 KWH
0.02350	for the next 50,000 KWH
0.02043	for excess of 100,000 KWH
0.01927	for excess of 200 hours use per KW demand
0.01464	for excess of 300 hours use per KW demand
0.01362	for excess of 400 hours use per KW demand
0.01300	for excess of 500 hours use per KW demand

Fuel Adjustment Charge

$0.025	per KWH

The energy charge under Rate H depends on the number of 'hours use' of billing demand for the month and is calculated by dividing the electrical consumption for the month by the billing demand for the same period. The 'hours use' is a load factor indicator. For example, if a customer consumes 1000 KW constantly for a month, energy use will amount to

$$1000 \text{ KW} \times 24 \text{ hrs/day} \times 30 \text{ days/month} = 720{,}000 \text{ KWH/month}$$

The customer's load factor is equal to 720 hours use. Because this customer is using the utility's output in a consistent, predictable manner, the utility will compensate the customer by selling the power at a reduced block rate. Another customer may use the same 1000 KW only 1 hour per day resulting in a consumption of 30,000 KWH and a load factor of 30 hours use. Existing rate structures penalize this customer due to the declining rate block structure and the poor load factor penalties. Most commercial office buildings exhibit load factors between 200-500 hours use.

INCREMENTAL ENERGY COST

An examination of recent consumption and demand data for the model building indicates that the load factor will tend to be close to 400 hours use (Fig. 38). Assuming that the value will be above this figure one third of the time, the cost would average

$$(\$0.01464/KWH \times 2/3) + (\$0.01362/KWH \times 1/3) = \$0.0143/KWH$$

Adding the fuel adjustment charge, the total cost would be

$$\$0.0143/KWH + \$0.025/KWH = \$0.0393/KWH$$

INCREMENTAL DEMAND COST

An increase in demand results in both a direct demand charge and an increase in the energy charge. This increase represents an implicit demand charge which must be accounted for in determining the cost of demand. If consumption falls between 300 and 400 hours use of demand then for each added KW of demand, 200 KWH are shifted into the \$0.02043/KWH block, and 300 KWH are shifted out of the \$0.01362/KW block. The result is a cost of

$$(200KWH/KW \times \$0.02043/KWH) + (100\ KWH/KW \times \$0.01927/KWH)$$

$$- (300\ KWH/KW \times \$0.01362/KWH)$$

$$= \$1.93/KW$$

If consumption falls between 400 and 500 hours use of demand, the cost by similar calculation is <u>\$2.05/KW</u>.

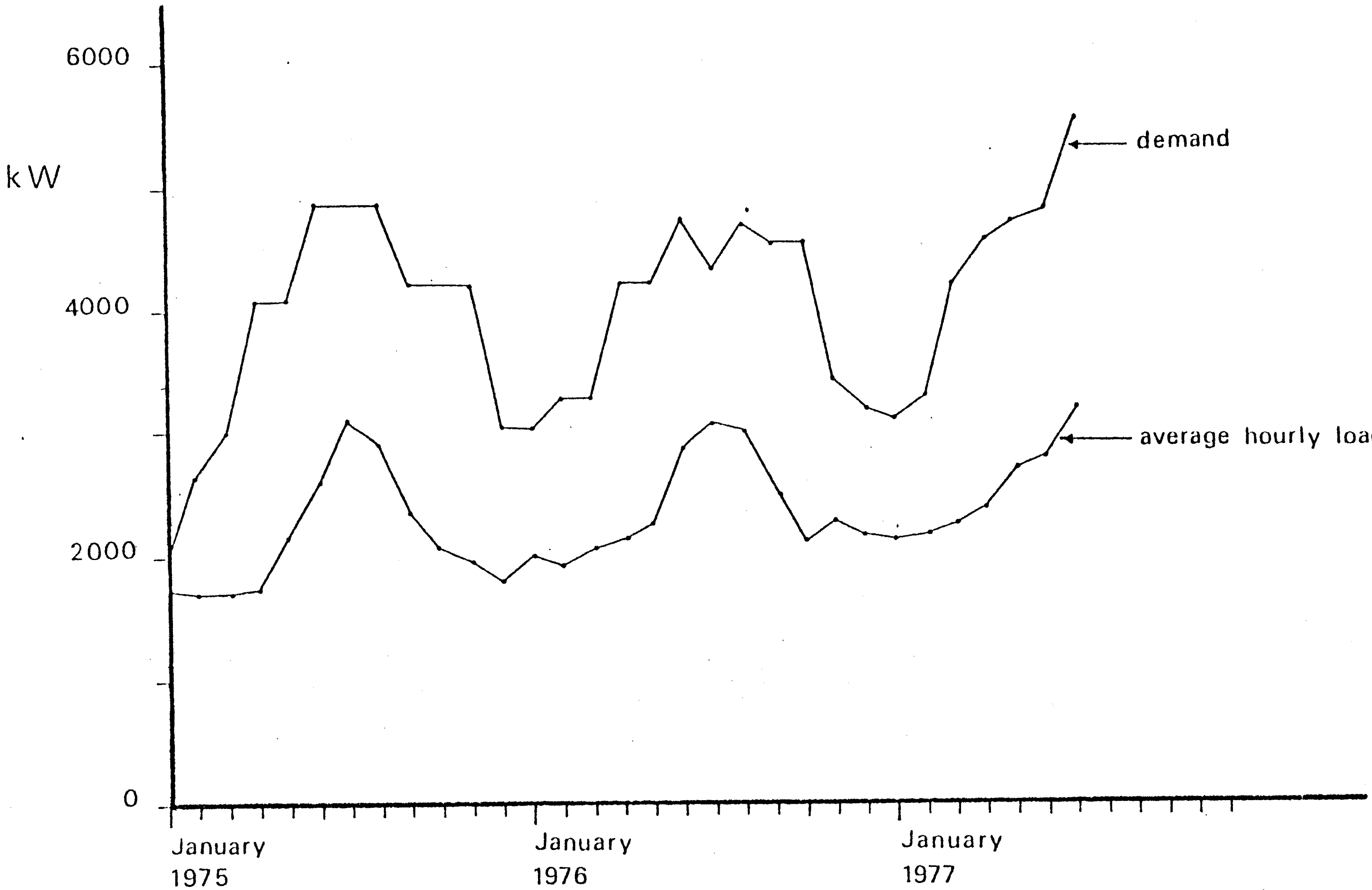


Figure 38: Average Hourly Electric Load

Assuming that consumption falls below 400 hours use of demand two thirds of the time, the average implicit demand charge if \$1.97/KW. Adding the direct demand charge results in a total of \$1.97/KW + \$1.57/KW = \$3.54/KW.

POTENTIAL SAVINGS WITH A COOL STORAGE SYSTEM

A. Reduced Demand Charges

Energy use in buildings generally changes according to the level of activity within the structure and the environmental conditions surrounding the envelope. These 2 factors typically generate peaks in power demand during the afternoon hours. This peak is troublesome to the utilities for a variety of reasons and results in an economic penalty to those who contribute to it. It is in everyone's interest to manage their loads in such a way that the peak is reduced or eliminated. The ice storage system will enable the building operator to eliminate this portion of the building's peak load and realize cost reductions in the electrical bill in the process. Savings will occur for practically all the months of the year. In summer the peak will be reduced by approximately 25%. Even though chillers are not operational during the winter, the peak generated by the chillers in the summer is reflected in the winter bill through the utility's 'ratchet' clause. Off peak cooling systems return the actual peak demand as the pricing index during the winter.

Data collected from the test building is listed in Table 16 . The normal peak demand is around 5500 KW. By running the chillers during off-peak hours, demand is reduced by 1218 KW. Similar reductions are realized for the months of April through September. Even though the chillers are not operational from November through February, the ratchet clause raises the power demand to 80% of the summer peak. The summer peak has now dropped from 5520 KW to 4302 KW. 80% of this is 3442 KW. Demand in November and December was 4080 KW and 3520 KW, meaning that the demand charges for those months reflect the actual load rather than the summer peak. Savings in March and October are zero since the chillers were generally not operational and the summer ratchet clause did not apply.

The savings in demand charges listed are equal to the incremental demand cost times the reduction in demand for each month.

TABLE 16 SAVINGS DUE TO PEAK LOAD REDUCTION

Month	Normal Demand Peak KW	Peak Reduction KW	Savings @ $3.54/KW
Jan.	(3776)†	(656)	2322
Feb.	(3776)	(496)	1756
March	4240	0	0
*April	4560	1048	3710
*May	4720	1226	4340
*June	4800	1270	4496
*July	5520	1270	4496
*August	4880	1270	4496
*September	5520	1218	4312
October	4560	0	0
November	(4416)	(336)	1189
December	(4416)	(896)	3172
			$34,288 saved/yr.

*Months chiller is operational

†Bracketed values indicate ratchet clause governs

B. Increased Chiller Efficiency Due to Operation at Design Load

The 1560 ton chiller presently used in our test building was designed to cool the building under worst case (2 1/2%) conditions. Because these conditions occur only during a few days of the summer, the chiller must operate at partial capacity the majority of the time. A chiller has its highest efficiency when operating at 100% with efficiency dropping as the load on the chiller is reduced. Fig. 39 shows the efficiency curve for the existing 1560 ton chiller; maximum efficiency at 100% load is 0.84 KW/ton.

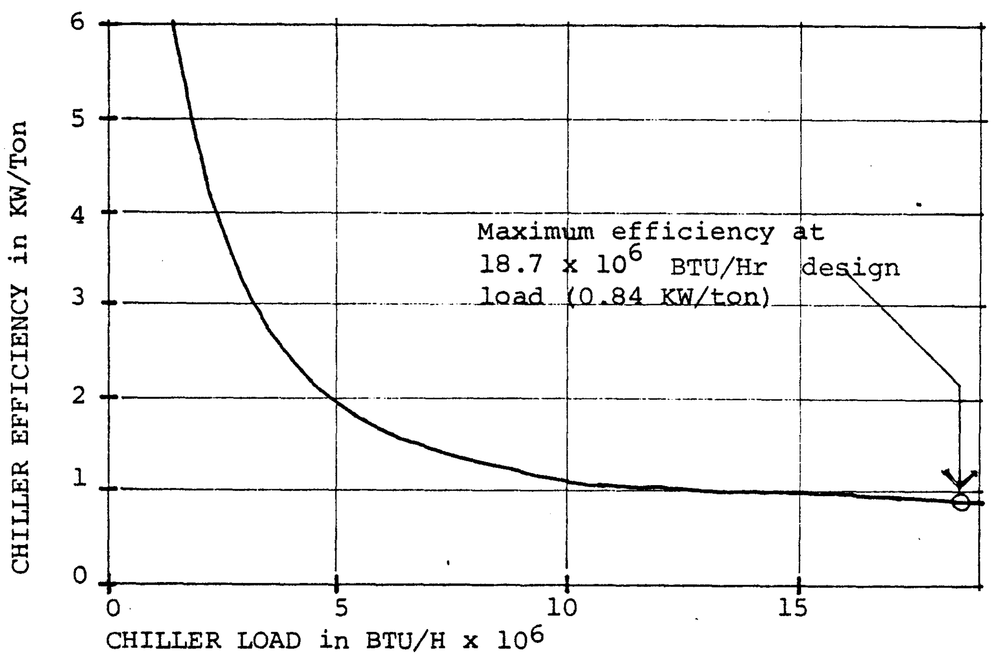


FIGURE 39 Chiller Efficiency vs. Load in Monitored Building (courtesy C. Benton)

The chiller involved in charging the proposed cool storage system can be run at 100% capacity until the cool storage is fully charged. Thus, losses in efficiency caused by partial load conditions do not exist.

To establish savings in this category, the chiller logs for the 1978 cooling season were examined. Pressure and temperature drops across the chiller were used to establish the cooling capacity provided by the chiller. The actual electrical consumption in providing this cooling capacity was compared, hour by hour, with the projected electrical consumption had the chiller been run at its maximum efficiency (0.84 KW/ton). The difference in these two electrical consumption rates was multiplied times the incremental energy costs to come up with savings. This information below is summarized by month.

TABLE 17 SAVINGS DUE TO DESIGN LOAD CHILLER EFFICIENCY

Month	KWH Saved	Cost @$.0393/KWH
April	5,112	201
May	99,433	3,908
June	150,024	5,896
July	228,970	8,999
August	269,587	10,595
September	73,783	2,900
Total	826,909	$ 32,499

Note: The 1978 cooling season was close to statistical normal for Boston

C. Lower Capital Costs for Chillers

An additional benefit of off-peak cooling systems is the ability to reduce the capacity of the chillers. This is possible by either running the chillers at a constant load for 24 hours, or at a constant load for some fraction of the day and at some reduced output during the peak load hours. In the test building, the cooling load profile for 3 hot August days is shown in Fig. 40 . During the 24 hour period of August 17, cooling demand amounted to 253 BTU/dayft^2. During the hours from 8AM to 8PM the load amounted to 168 BTU/12hrft^2 or an average of 14 BTU/hrft^2. The off-peak consumption was 7 BTU/hrft^2. A storage system designed for a 12 hr charge period for this building must have a chiller which can move 21 BTU/hrft^2°F (sum of the average peak and off-peak load). Without the storage system, the chiller must be capable of meeting the cooling loads as they are generated. For this building, that would require a chiller with a capacity of 21 BTU/hrft^2. While the off-peak system realizes substantial demand charges reduction, the capacity of the chiller (for a 12 hour charge cycle) is essentially the same. It should be noted that the 24 hour computer center in the test building adds a considerable amount of nighttime cooling load that would typically be absent from most commercial buildings. Taking this into account, the chiller capacity would be reduced somewhat by an off-peak storage system. Lowering the nighttime cooling loads from 7 to 4 BTU/hrft^2 reduces chiller capacity from 21 to 18 BTU/hrft^2. This 14% reduction would amount to an equivalent chiller reduction of 190 tons for this building (1325 to 1135 tons). Assuming an installed cost of $1000/ton the savings would amount to $190,000. Amortized over a 20 year period results in a savings of $9,500/yr.

It may be possible to increase savings even further if the off-peak period is extended further. This should only be done if the savings in capital costs for the chiller are greater than the decrease in demand cost savings. This area of optimization requires detailed and case by case study. For purposes of this thesis, using the office building chosen, no capital savings will be accrued with the cool storage system.

SUMMARY OF SAVINGS

A.	Reduced Demand Charges	$34,288/yr
B.	Increased Efficiency Savings	$32,499/yr
		$66,787

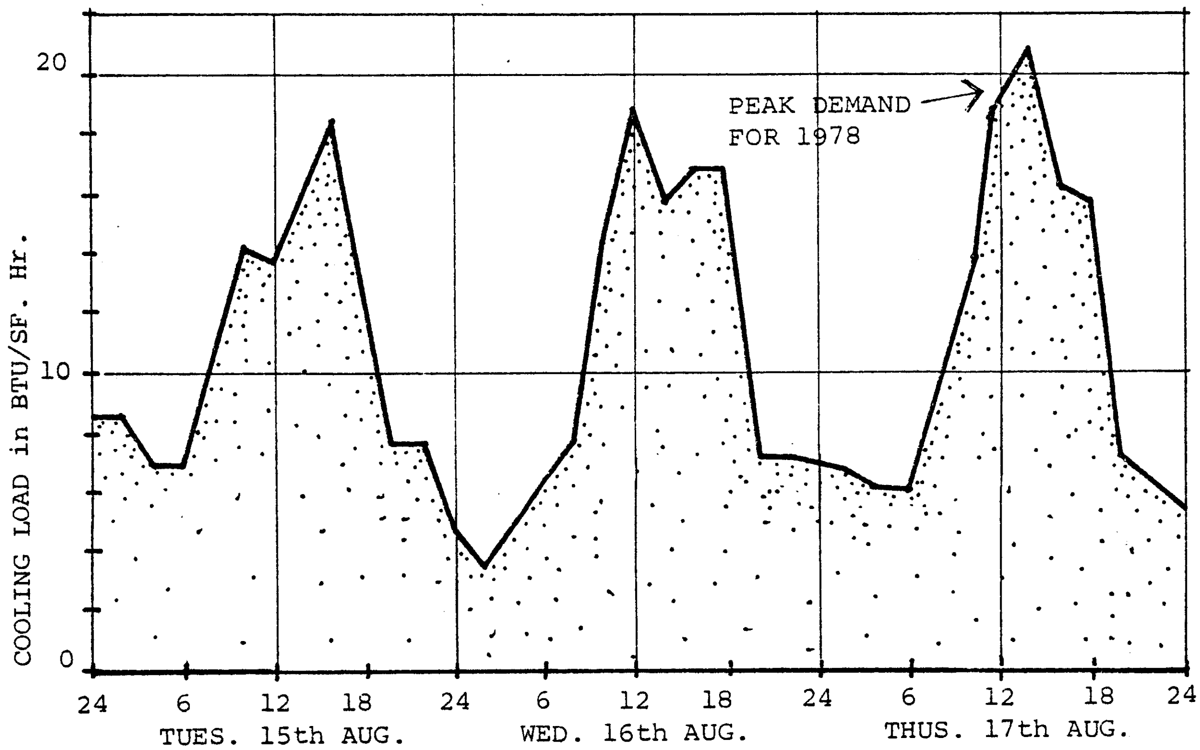


FIGURE 40 Actual Cooling Loads from Monitored Building, August 1978.

(courtesy C. Benton)

Costs for the plastic pipe coil ice storage system are unknown. The materials cost for the factory built, plastic modules would be lower than an equivalent copper pipe ice maker being tested by Wisconsin Electric. However, in a commercial scale application, the increased complexity could drive overall costs up to the level of the Wisconsin Electric prototype ($4400/million BTU). Therefore, a cost range of $3000-$4000/million BTU will be assumed for the plastic module storage system. The maximum cooling needs for the test building for the 12 hr peak period is 168 BTU/12hrft^2. With an area of 757,000 ft^2, the storage requirements would amount to 127 million BTU which would cost between $318,000-$508,000 ($3000-$4000/10^6 BTU). With a savings of $66,787/yr with storage, the straight line payback would be 5.7-7.6 years.

If the frostless ice plate storage system proved feasible, costs would be tremendously reduced. System components would include tank, ice plates, piping and control hardware. The tank costs $.35/gallon,and 1 gallon will store 764 BTU as latent and sensible heat (50% volume as ice) Tank costs per million BTU = $458. The cost for the frostless plates is proportional to the total area required. Heat transfer rates on the order of 70 BTU/hrft^2°F can be expected as indicated by the experimental data from Section IV . Assuming a 10°F ΔT, 83,000 BTU/hr ice making rate (for 1,000,000 BTU charged over 12 hrs), and both sides of the plate available for making ice, the amount of plate required equals 60 ft^2/10^6 BTU of storage. The plates would probably be similar to the Olin copper absorber plates made for solar collectors which sell for $4.50/ft^2. These would need a special coating (i.e., TEFLON^R) to insure frostless operation. Assuming the coating would double the cost to $9.00/ft^2, the plates would cost $540/million BTU. Adding an additional $500/million BTU for piping and controls brings the total cost to $1500/million BTU. The same 127 million BTU storage unit would cost $190,000. With operational savings equal to $66,787/yr, payback would take 2.8 years.

The same analysis using a cool water storage system yields costs of $381,000-$762,000 ($3000-$6000/million BTU of storage);payback would range from 5.4-10.8 years. All these cost estimates assume that extra space is available either in the sub-basement area of a building or elsewhere on the site for accommodation of the tank. For cool water storage systems this may be a very generous assumption, considering the large tank volume required.

The paybacks listed are for off peak cooling systems with costs in 1980 dollars and operated under the current electric rates. While this rate structure penalizes consumption with a poor load factor, it does not directly encourage consumers to shift consumption to off-peak hours, thereby alleviating the utility's peak load problems. Time of day rates would accomplish both of these goals by charging more for electricity consumed during the peak demand hours (usually 6AM-8PM). At present, time of day rates are optional. It would not be sensible for a building operator to choose these rates unless most of the power consumption can be shifted to the off peak hours. The air conditioning system represents only 35%-40% of a building's power consumption during a summer day. During the winter, only the fans are consuming power for air conditioning. The remainder of the electrical consumption is used to power lights and equipment which must be used while the building is occupied. Time of day rates would end up costing the operator more than the conventional rates, since a majority of electrical consumption would still occur during the day. Two separate metering systems would overcome this problem. The chilllers could be wired to a time of day meter while the rest of the building's electric load could be metered under the conventional rates. In this case, time of day rates would be extremely beneficial. Unfortunately, the experimental time of day rates do not allow this at present. All building loads have to be metered under 1 rate structure. Unless this changes in the future, commercial office buildings will be reluctant to purchase power under these rates. It is possible that the government may direct the Public Utilities Commission to make time of day rates mandatory. If this were ever to happen, energy use in buildings would undergo drastic changes. Undoubtedly, cool storage systems would then become an economic necessity.

CONCLUSION

As the United States enters the 1980s the greatest challenge it faces is the worsening energy picture. Our domestic supplies of easily recoverable oil and natural gas are approaching exhaustion. Coal and nuclear energy are beset by a variety of health and safety problems which require large investments of time and money in order to reconcile the dangers. Unfortunately, time and money are this issue's endangered species. As an industry, the electric utilities are feeling the effects of this situation in a dramatic way. New construction has slowed to a trickle as a result of reduced demand and higher construction costs. The impact on consumers is primarily manifested through unprecedented rate hikes which reinforces the drop in demand. While the growth in total consumption has dropped to 1-3% per year, the growth in peak demand continues at almost traditional rates of 4-5%. The resultant drop in capacity factor further aggravates the unhealthy state of the utilities.

In today's economy, load management rather than expansion of generating capacity is the rational way of optimizing a utilty's capacity factor. One of the most equitable load management strategies is time of day pricing. This type of rate schedule accurately reflects the cost of production by penalizing those who contribute to the utility's poor load factor. Cooling systems are prime contributors to a utility's summer peak. Cool storage systems would help to reduce this peak by shifting the chiller load to off-peak hours.

The cool storage system proposed in this thesis employs water as a phase change material. By forming ice, the quantity of mass is greatly reduced. When compared to a chilled water storage system, the containment volume will be reduced by 80%. The system is operated during the nighttime hours when ambient conditions enable the chiller to operate at a reduced evaporator temperature (20°F) without lowering the COP. A brine coolant is circulated through pipes located in a storage tank. The brine removes heat from the tank water, causing an ice cylinder to grow around the pipe. The ice formation stops after 80% of the water has changed phase. The amount of ice formed during the 10 hour off-peak charge period is dependent on the pipe material, pipe size, and brine temperature. Although a copper pipe system would allow a 60% higher heat transfer over plastic pipe, the cost for the copper and its fabrication (for commercial scale application) make plastic a more economical choice. A theoretical two dimensional heat flow analysis was verified through experimentation and used to predict pipe length and spacing for a variety of heat exchanger sizes.

The discharge of the ice store is facilitated by circulating the remaining water through a heat exchanger which is hydronically coupled to the building loads at the air handling unit. Discharge rates far in excess of any forseeable building load can be handled due to the direct heat exchange between the water and ice. This high heat transfer results in a lower mass flow which translates to lower pipe and pumping costs. This system satisfies the criteria of being compatible with conventional packaged chillers.

In order to facilitate assembly and repair, a modular system design is suggested. 3'x3'x8' plastic pipe modules should be able to be easily fabricated and moved into place. Coupling between modules allows for single unit removal without interrupting operation. The costs for such a system should be between $3000-$4000/million BTU of storage (including tank and hardware). Using a commercial building in Boston as an example, the system should pay for itself in 5-7 years assuming conventional rate schedule. The savings are due to a reduction in demand charges and a reduction in energy charges due to increased efficiency resulting from full load operation. This payback would be tremendously improved if time of day rates ever became mandatory. Ice maker heat pumps are considered uneconomical due to their high capital costs and low COP resulting from the superheat defrost cycle. If a frostfree surface treatment were devised, the resultant reduction in capital costs, operating costs, and system complexity would insure the widespread acceptance of this type of off-peak cooling system. Paybacks under current rate structures would be under three years.

Ice storage off-peak cooling systems will provide the means to avoid costly peak power demand penalties without resorting to storage systems that require the redesign of a building's structural and mechanical systems. These systems will help the utilities through a period of escalating fuel and credit costs by reducing the need for additional generating capacity.

Nevertheless, more basic questions will ultimately have to be addressed concerning the appropriate way of producing electricity and for what end use. In twenty years time will we think it appropriate to build a 2 billion dollar power plant that takes 10-12 years to construct, burns a finite fuel (coal, oil, uranium) at thousands of degrees, losing most of the energy in the process, all merely to provide 95° air to heat a home? Common sense says no. The era of cheap energy that gave rise to such scenarios is behind us, yet the institutions it created in its wake remain with us. Whether or not these institutions are flexible enough to change in the time period demanded by the circumstances is unknown. All that we can be sure of is that time is not on our side.

APPENXDIX

Determining PCM Melt Point and Heat of Fusion (H_F)

A large number of chemicals listed in the literature melt in the temperature range appropriate for a cool storage system. However, only a handful of these chemicals satisfy other criteria equally important for the economic operation of such a system (See page 36 for further discussion.) Table 4, page 40 lists seven potential PCM candidates. As explained in the text only water, n-tetradecane, the C_{14}-C_{16} paraffin, and deconal seemed to satisfy the economic and long term cycling constraints. These four chemicals were tested for melt point and heat of fusion. Water was used as a reference to check the accuracy of the methods. Although there is already a considerable amount of information in the literature about the thermophysical properties of these chemicals, most of this data is taken from tests using scientific grade chemicals. Because of the gross quantities of PCM required by a cool storage system, industrial grade quality material must be used. The impurities found in industrial grade chemicals can significantly alter the melting point of the compounds. The heat of fusion may also be affected. The industrial distributors are reluctant to supply exact freezing and H_F data, since the composition of their product may vary from lot to lot due to differences in the petroleum feedstocks. Industrial grade samples of the three organics were obtained from distributors and tested.

Melting Point Test

A sample of the PCM was drawn up into a capillary tube. The ends of the tube were then sealed. A 500 ml beaker was filled with water and ice so that the water bath starting point was always 32°F. The capillary tube was strapped alongside the bulb of a mercury thermometer having 0.5°F increments. The thermometer and tube were immersed in the water bath which was kept at a uniform temperature with a mechanical stirrer. As the temperature of the water bath rose, the point at which the PCM melted was noted. The experiment was then carried out in reverse order by starting with the PCM in the liquid state and watching for its freeze point by cooling the water bath slowly. The melt point and freeze point were usually different.

TABLE 18. MELT POINT TEST RESULTS

Substance	Supplier	Melt Point(°F) S.Grade*	Melt Point(°F) I.Grade**	Freeze Point(°F) S.Grade*	Freeze Point(°F) I.Grade**
n-tetradecane	Humphrey Chem. Co.	38.0	32	38.0	-
Paraffin	Conoco	44.0	46.5	42.0	44.5
C_{14}-C_{16}	Exxon	44.0	47.5	42.0	47.0
Deconal	Conoco	42.8	45.0	42.8	36.0

*Scientific grade; **Industrial Grade (MIT laboratory tests)

The decanol's wide melting/freezing band apparently resulted from the presence of branched hydrocarbon impurities in the industrial grade sample. This was not a supercooling effect. When nucleation started at 36°F, the sample was immediately raised to a temperature of 40°F and the nucleation process stopped. If supercooling was occuring, once nucleation began it should have continued until the sample was completely frozen, assuming the temperature was somewhere below the melt point (45°F). As the temperature of the deconal was raised from 36°F to 45°F an increasing percentage of the PCM solidified, indicating that the wide melt/freeze band was a result of the impurities. The industrial grade sample of n-tetradecane could not be frozen when taken down to 32°F. Further testing of the material was stopped for this reason.

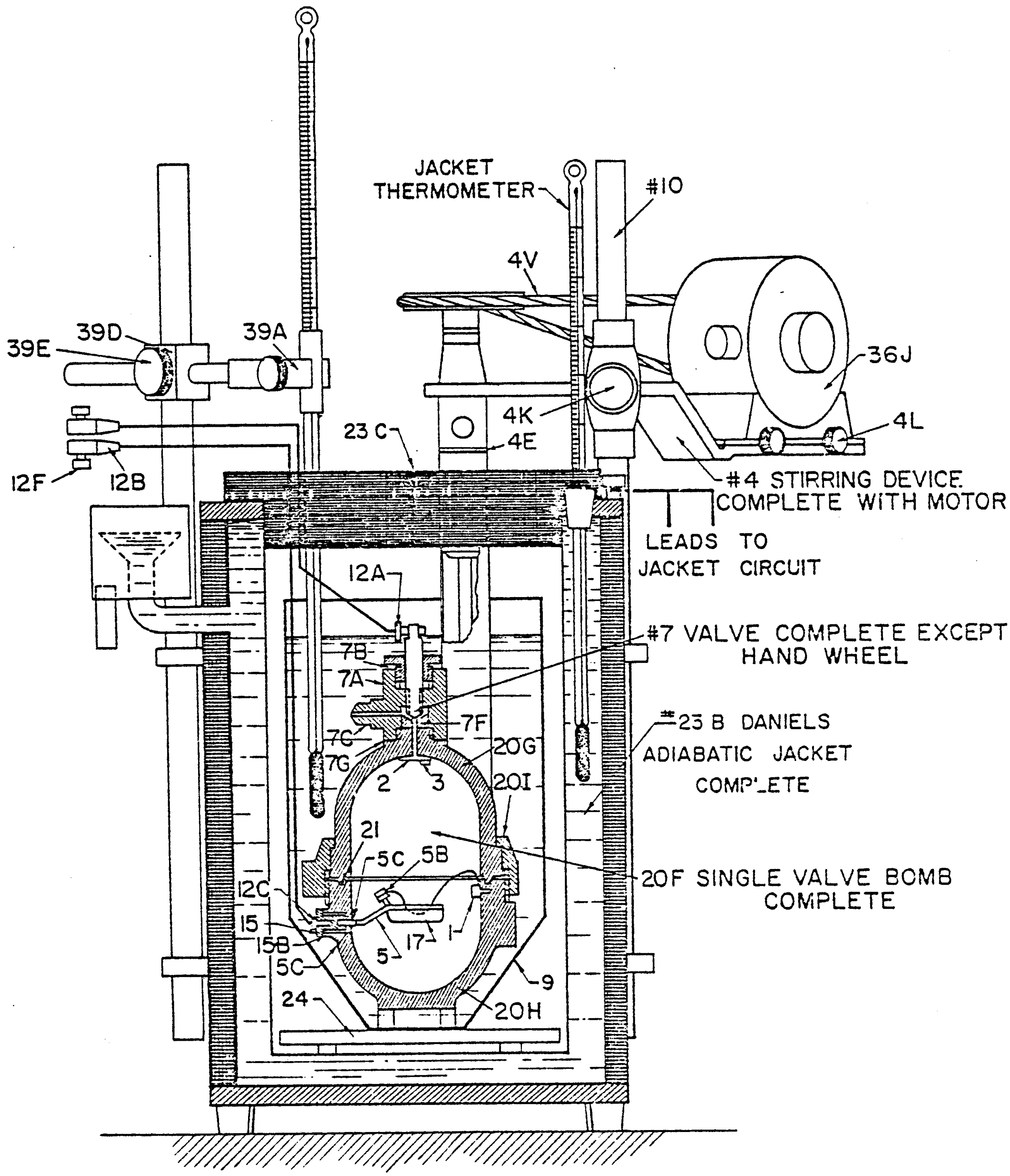
Heat of Fusion

An Emerson Fuel Calorimeter was adapted to be used for testing the heat of fusion for the three organics (the two paraffins, and deconal). Fig. 41 shows the device when it is used for bomb calorimeter tests. For these experiments, the bomb was removed and the conventional thermometers were replaced with two Beckman Thermometers having 0.01°C increments. The 500 ml stainless steel bucket was filled with a known quantity of water which had been chilled to 32°F. The adiabatic jacket was also filled with 32°F water in order to eliminate any heat flow between the bucket and the environment. A small stirring device kept the bath water at a uniform temperature. A 20 ml sample of PCM contained in a glass vial was then dropped into the bucket through an access port in the lid. The temperature change of the water bath indicated the sensible and latent heat content of the PCM. As the bucket temperature rose, the jacket temperature was increased by sending a small electric current through a resistance wire in the jacket. In this way, an adiabatic environment was maintained.

TABLE 19. HEAT OF FUSION TEST RESULTS

Substance	Heat of Fusion (BTU/lb)		
	Literature	Laboratory Exp.*	Deviation (%)
Water	144.0	143.0	-1
Deconal	88.6	85.0	-4
Exxon Paraffin	65.5	67.4	+3
Conoco Paraffin	65.5	60.0	-8

*Average of test runs



EMERSON FUEL CALORIMETER
WITH DANIELS ADIABATIC JACKET
Fig. 41

As a result of the experiments, it appears that the effect of impurities on the heat of fusion of the PCM is not very significant.