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# Prefeasibility Analysis and Study of the Seasonal Storage Systems for the Pennsylvania Avenue Redevelopment Project

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## 1. INTRODUCTION

The steady increase in fuel cost, coupled with its decreasing availability, has made energy conservation measures mandatory and has opened up new avenues for annual energy storages systems.

These systems enable the energy generated in one season to be stored for use in another, thus making possible the carry-over of energy from the point of its maximum generation to the point of its maximum use.

This concept is known as the Annual Cycle Energy System (ACES), and has already been initiated in the U.S.A.

The systems already in use in the U.S.A. are so far small systems, but the energy savings which they accomplish amount to 50 - 70%. There is good reason to believe that large scale systems can achieve even greater savings.

The main advantage of a seasonal storage system is the considerable cut in electrical peak demand which results in several beneficial savings, like smaller size for cooling equipment in summer time, smaller electrical equipment and installation and smaller billing demand charge.

The most favorable storage system of refrigeration energy is the ice making heat pump system, whereby a heat pump supplies the heating and domestic hot water requirements in winter time while using as a heat source water in a large bin, which could be part of the structure, and converting the water into ice to be used in the summer for cooling purposes. This is an example of cooling storage.

Solar energy storage can serve as an example of heating storage. Solar energy collectable when it is most available, that is in summer time, is stored and then used when it is least available, that is in winter time. Storage tanks are much cheaper than solar collectors which makes it very feasible to use large storage systems and smaller solar collectors.

Another energy storage system holding great promises for the future but still in the early stages of development is combined electric power and heat generation currently being developed both in Europe and in the U.S.

High temperature water (HTW) at a temperature of 350° F (175°C) is injected into confined underground aquifers, which are layers of sand, gravel and porous rock saturated with ground water, stored and later withdrawn with pumps, distributed and used for district heating in winter time. This high temperature water is generated in a series of heat exchangers which are fed by steam bled from a usually back-pressure turbine system at several pressure stages. The turbine system could be either a condensing or an entirely extraction system whereby none of the steam is condensing.

Such a system enable the differences in peak electrical and heating demands to be smoothed out and heat generated during summer months to be stored and used in winter months.

Other means of energy storage such as steam or hot water storage in man-made lakes insulated or thermally stratified have been considered.

This report will evaluate four different kinds of ACES to be supplied to the Market Square Complex of the Pennsylvania Avenue Development project in Washington, D.C.

## 2. PURPOSE AND SCOPE

The purpose of this prefeasibility study is to examine the application of the ACES concept to the proposed Pennsylvania Avenue Redevelopment Project which is conceived as a major inner city integrated project with innovative energy conservation and load management techniques.

Within the scope of this study the three already existing in different stages of design and completion, seasonal storage systems will be evaluated with a view of applying some of their concepts and methods to the PADC.

As a further step the systems, as applied to the PADC project, subsystems and their components will be identified and the candidate equipment selected and evaluated, with conceptual plans prepared for cooling and heating storage. The ACES will be compared with a Conventional system relative to which its costs and benefits will be evaluated.

Also, an estimation of electrical peak demand reduction will be made and other opportunities and constraints of the ACES versus a Conventional System discussed.

The utilization of below grade structures and their integration within the storage system will be investigated.

The inter-relationship of large scale systems and seasonal storage and their mutual bearing on one another will similarly be discussed.

Finally, an evaluation will be made whether the application of the ACES to the PADC project should be further pursued or not and if it should, a recommendation will be made to prepare a fully comprehensive feasibility study with an estimation of cost and time required to perform the task. In the feasibility study the parameters developed in the prefeasibility study will be fully utilized along with the identified technologies and methodology in order to continue to develop the new concept which holds great potential for energy conservation.

### 3. SUMMARY AND RECOMMENDATIONS

This report deals with Annual Cycle Energy Storage (ACES) concept as applied on a community scale to Market Square Complex on Pennsylvania Avenue, Washington, D.C.

Total area 2,200,000 square feet

Building Load estimation

	<u>Peak Loads</u>	<u>Annual Energy Consumption</u>
Heating & DHW	19.35 x 10 <sup>6</sup> BTUH	4.032 x 10 <sup>10</sup> BTU per yr
Cooling	3000 Ton	3,000,000 Ton-Hours

In this study we examined four alternatives of seasonal energy storage. Each alternative was examined on the energy saving aspect and its operational and economic feasibility.

Out of all the alternatives considered the most efficient system from the point of view of energy and economic feasibility was found to be system No. 3 dealing with heat pump generated ice for seasonal storage and it was thus selected and recommended for further study.

The system utilizes the heat pump for heating the buildings in winter and for meeting the DHW requirements. The heat pump obtains its heat by extracting the heat of fusion of water and thereby converting it to ice.

The method suggested is to use the ice maker evaporator with water sprayed over the evaporator coil and being converted to ice. The ice would be used to cool the buildings during the summer by circulating chilled water through the ice bin. The deficiency in cooling during the summer would be supplemented by generating chilled water during off peak periods and storing it for peak period use.

This system is expected to supply about 70% of the summer cooling requirements and provide a 100% cut in electric peak demand.

The heat pump system using the slab as storage of the heat rejected for reusing in winter time was found inefficient from the energy point of view. Only about 4% of the heat required during winter could be stored in the slab.

The solar energy annual storage was found efficient energywise but prohibitive from the economical point of view.

The winter cold air potential to make ice for storage was found efficient from the energy point of view but prohibitive from the economical point of view and because of unpredictability of system performance.

It is, therefore, recommended that the heat pump system with ice storage be taken up for a further feasibility study.

COMPARITIVE ANALYSIS OF FOUR CANDIDATE SYSTEMS

	Basic Conventional System Boilers & Chillers	1 Heat Pump System with Slab Storage	2 Solar Energy Annual Storage	3 Heat Pump for Generation of Ice. ACES	Outside Air Potential for Ice Making in Addition to System 3.	
Annual Heating Energy DIFW	4.032 x 10 <sup>10</sup> BTU	4.032 x 10 <sup>10</sup> BTU	4.032 x 10 <sup>10</sup> BTU	4.032 x 10 <sup>10</sup> BTU	4.032 x 10 <sup>10</sup> BTU	
Requirement Cooling	3.0 x 10 <sup>6</sup> TON H	3.0 x 10 <sup>6</sup> TON H	3.0 x 10 <sup>6</sup> TON H	3.0 x 10 <sup>6</sup> TON H	3.0 x 10 <sup>6</sup> TON H	
Energy Stored	0	7.5 x 10 <sup>8</sup> BTU	1.684 x 10 <sup>10</sup> BTU	2,114,000 TON H	3.0 x 10 <sup>6</sup> TON H	
Energy Saved	0	7.5 x 10 <sup>8</sup> BTU	4.032 x 10 <sup>10</sup> BTU	2.5379 x 10 <sup>10</sup> BTU	3.6 x 10 <sup>10</sup> BTU	
Energy Consumption	W	17 x 10 <sup>6</sup> KWH-T	3.447 x 10 <sup>6</sup> KWH	106,800 KWH	3,450,000 KWH	3,368,320 KWH
	S	3 x 10 <sup>6</sup> KWH	3.013 x 10 <sup>6</sup> KWH	3,105,200 KWH	1,252,870 KWH	367,770 KWH
	Total	20 x 10 <sup>6</sup> KWH	6.462 x 10 <sup>6</sup> KWH	3,212,000 KWH	4,502,870 KWH	3,736,140 KWH
Demand Reduction	0	0	+82 KWH-6 months	-3,000 KWH-4.5 mo.	-3,000 KWH-6 mo.	
Operation Annual Energy Cost	\$ 331,710	\$ 264,470	\$ 156,000	\$ 153,392	\$ 129,380	
Operation Annual Saving vs Conventional	0	\$ 67,241	\$ 175,710	\$ 178,318	\$ 202,330	
System Capital Cost	\$750,000	\$3,400,000	\$9,800,000	\$2,600,000	\$3,752,000	
Incremental Investment vs Conventional	0	\$2,650,000	\$9,050,000	\$1,850,000	\$3,002,000	
Absolute Present Value of The System	\$ - 10,684,269	\$ - 11,034,207	\$ - 14,321,807	\$ - 7,011,518	\$ - 7,361,254	
Present Value of The System Relative to The Conventional	\$ 0	\$ - 349,938	\$ - 3,637,538	\$ 3,672,751	\$ 3,323,015	

#### 4. SELECTION OF A BUILDING COMPLEX

The Market Square Complex situated at Pennsylvania Avenue and being part of the Pennsylvania Avenue Development project currently being developed by the Pennsylvania Avenue Redevelopment Corporation was selected as the object of application of the energy conservation and storage principles analyzed in this study.

This selection is being made in view of the functional diversity of the complex which has approximately 1,300,000 square feet area above grade and about 900,000 square feet below grade. The areas above grade comprise 56% of residential area, 13% of retail department stores and offices and about 30% of national archives and community storage. The 900,000 square feet below grade are devoted entirely to national archives.

The energy used in the residential areas is likely to be at its lowest during the day and at its highest during the evenings while in the stores and offices the reverse would occur--the latter being occupied mostly during the day.

The whole complex thus with its mixed uses and sizable cooling, heating and domestic hot water loads affords an opportunity to demonstrate the favorable interaction between areas having non-concurrent energy peaks.

The Market Square Complex situated between 7th and 9th Streets forms the focal point of the entire proposed Pennsylvania Avenue Development project which is conceived as a major city integrated project containing multi-use facilities whose major design elements incorporate modern and advance energy conservation techniques and load management concepts.

Pennsylvania Avenue is the link between the White House and the Capitol and as such it is the "Main Street of the USA," enjoying high public visibility. It contains existing buildings whose facades must remain intact, its new buildings must match the overall architecture of the area and its many other types of buildings include offices, hotels, commercial buildings and residences. It thus represents a cross section of most types of buildings typical of the US cities and therefore lends itself well as an ideal demonstration center to many different and new energy conservation and management techniques.

If successfully developed into an energy effective area, Pennsylvania Avenue would serve as a bright example to all branches of the government to follow suit and apply the example on a national scale. Also, the high visibility of the Pennsylvania Avenue project would serve as a constant reminder and spur to non-government agencies to sponsor and engage in similar projects. This would greatly further and enhance the commercialization and utilization of innovative systems and prototype equipment tested under field conditions in the Pennsylvania Avenue Redevelopment project, a "real-life" project as differentiated from simulated buildings or demonstration projects alone.

The climate in Washington, D.C. is moderate which makes it suitable for the application of a seasonal storage of energy system which with slight modification could be applied to other areas in the USA.

Finally, the last but not the least reason for selecting the Market Square Complex as the object of application of this study is the fact that there has already been prepared by Dubin-Bloome Assoc. an "Energy Conservation and Alternative Energy Source Conceptual Plan" for the Pennsylvania Avenue Development Corporation (PADC) which was devoted primarily to energy management programs, to the examination of architectural concepts and various control systems, distribution systems and end use of mechanical and electrical systems to reduce annual energy consumption and peak power demand.

The above study also singled out the Market Square Complex as an object for further attention.

It is therefore the intention of this pre-feasibility study to demonstrate how advanced concepts in energy conservation and energy management could be applied in the nation's capital to make it a showcase to the whole country and the world at large.

## 5. EVALUATION OF THREE EXISTING STORAGE SYSTEMS IN THE U.S.A.

Following is a review of completed or partly completed ACES for the purpose of extrapolating those portions which might be compatible with the seasonal storage systems for the Market Square Complex, PADC.

- A. ACES Demonstration House in Oak Ridge National Laboratory, Tennessee
  - B. The ACES Nursing Home for Veterans Administration, Wilmington, Delaware
  - C. Gettys ACES, Racine, Wisconsin
- A. The ACES Demonstration House in Oak Ridge is a 2,000 sq.ft. single family house.

The ACES is a water to air heat pump system with ice formation and thermal storage. The design calculations indicate that ACES can satisfy the space heating, space cooling and domestic hot water requirements for the whole year.

The system combines a heat pump mechanical package with hot gas heat exchanger, brine chiller, brine coil immersed in a water tank, heating/cooling fan coil units, domestic hot water storage tank and outdoor radiant convector coil and is thus capable of balancing the energy requirements for space heating cooling and domestic hot water over the whole year.

The analysis of the system brings out the following characteristics:

- a. The system shows about 42% savings in energy consumption compared with a conventional system.
- b. Equipping a new typical house with ACES incurs an extra cost of \$1,950 over the conventional air conditioning system with an electric hot water heater; retrofitting an old house, the extra cost would be between \$3,000 to \$4,000 depending on the size of the house, its construction and the previous heating system.
- c. Additional energy savings are limited in this system since the ice bin coil, carrying chilled brine, is inefficient as the ice forming on the coils. having a low heat conductance, impedes the heat transfer between the brine in the coils and the water and thus slows down any further ice formation reducing thereby substantially the COP of the system.

- d. The use of brine causes further reduction in the efficiency of the system because of the number of heat exchangers involved in such a system, lower suction temperatures employed and the usual corrosion problems that come up whenever brine is used.

The system being of small size and utilizing currently available on the market heat pumps, has a relatively low COP. However, with the new, specially designed for this application, high efficiency compressors that are beginning to appear on the market, a higher COP could be reached.

Since in large scale systems there are already compressors that can operate with high efficiency at low suction temperatures the ACES promises more savings when applied to large systems rather than to small ones.

- f. The system being equipped with an outdoor radiant convector coil is suitable for climates with long, severe winters where the ice generated during the winter months exceeds the summer cooling requirements and has to be controlled with solar energy.

- B. The ACES in the Veterans Administration Nursing Home operates as an air source heat pump down to 39.2°F and thereafter as a water source heat pump.

An outdoor unit serves as an evaporator for the heating cycle and as an evaporative condenser for the cooling cycle.

When the ambient temperature is above 39.2°F air is passed over the outdoor evaporator while a double bundle condenser delivers heat to the building. Below 39.2°F the brine chiller is energized and brine circulated through the water in the ice bin removes the latent heat of fusion given up by the water (144 BTU/lb) causing ice formation.

The whole system besides the above mentioned outdoor unit, a double bundle condenser and brine chiller contains also a water chiller in parallel with the brine chiller and a radiant convector panel which in winter can collect solar energy to melt the excess ice if any, and on cool summer nights, when some cooling is required can dissipate heat along with the evaporative condenser.

The whole system is located in a 40 ft. x 50 ft. building called the "Energy Bank." A micro processor within the Energy Bank will periodically accumulate data from a multitude of sensing locations and execute COP calculations.

This is to determine if some other mode of operation could be employed which would result in the use of less energy.

There are seven basic modes of operation in the Energy Bank. Four are used to produce heating and the remaining three are used to produce cooling.

Although the system has numerous "modes" of operation there are three overriding objectives that ACES seeks to attain: energy conservation, load management and the use of renewable resources.

The expected annual overall COP is 4.5 to 5.0, the energy savings are about 60% compared to a conventional system and the pay back period is estimated as 11 years, taking into account the cost of money and allowing for fuel escalation costs.

Evaluating the system we can specify the following points:

- a. The system would promise a higher COP than the ACES Demonstration House since it is being applied on a larger scale and for large scale applications, the equipment is more readily available on the market.
- b. The system is applicable to the Nursing Home because of the compatibility of the load which has a large cooling, heating and domestic hot water demand.
- c. The many subsystems, like the evaporator-condenser, the radiant convector panel, the parallel water chiller (for conventional cooling in the summer) together with micro-processor-controller enable the system to attain a larger flexibility and consequently a larger COP.
- d. The system is particularly well suited in climates having severe winters when the ice accumulated during the winter months exceeds the summer cooling requirements and has to be partially melted by using solar energy.
- e. The use of the ice bin coils to freeze water should be limited and replaced by "ice-maker" equipment that promises higher efficiencies and better performance of the systems.

- C. The Gettys ACES in Racine, Wisconsin has a cooling capacity of 160 tons and operates in the following mode:

During the winter nights ice is being generated by the ice-making evaporators while the heat rejected in the condenser is utilized to heat the building. The condenser can be either water cooled or air cooled for winter and summer operation respectively. Heating on winter days is done by circulating hot water from the hot water storage tank through the building distribution system, the storage tank being charged during the winter nights by an electric boiler.

Heat generated in the condenser of the heat pump could also be stored in the hot water storage tank.

In order to control the quantity of ice generated by the heat pump winter operation and melt the excess of it solar energy is utilized.

This is accomplished by energizing a solar pump which will circulate water from the ice bin through the solar collectors.

On cloudy days when solar energy is not available, controlling of ice generation is done by utilizing the electric boiler and boiler pump to pump heated water through a water-to-water heat exchanger, through the other side of which water from the ice bin is being circulated by temporarily utilizing the evaporator pumps.

There is no chiller in the system and cooling on summer and winter days is accomplished by circulating ice water from the ice bin through the air conditioning equipment, utilizing the chilled water pump, which is shut off during the nights in order to conserve energy.

Ice is also being generated during summer nights and summer weekends utilizing air to cool the condensers.

The system is controlled automatically with a micro-processor.

The following are the main characteristics of the system:

- a. The system operates mainly with nightly and weekly off-peak production and storage of ice both in winter and summer time and also nightly and weekly off-peak production and storage of hot water and therefore could be classified as a diurnal rather than seasonal storage system.
- b. Cooling and heating peak electrical loads are reduced through storage and this results in reducing demand for the electric utility.
- c. Ice generation in summer time is necessarily accomplished with low saturation suction temperatures at the expense of a lower COP. Chilled water production in summer time would result in a higher COP.
- d. Ice production is accomplished efficiently by the use of ice-maker evaporators.

## 6. METHODOLOGY

1. Select a building complex on the basis of:
  - a. Functional diversity
  - b. Large size
  - c. Has heating, cooling and DHW loads
  - d. Real building
  - e. Funding by PADC
  - f. High visibility
  - g. Sensitivity to climate
  - h. Previous work done on loads and energy conservation
  
2. Select Conventional System
  - a. Oil fired boilers for heating and DHW and
  - b. Electric chillers for cooling
  
3. Evaluate existing ACES type systems in the U.S.A.
  - a. ACES Demonstration House, Oak Ridge National Laboratory, University of Tennessee
  - b. ACES Nursery Home for Veterans Administration, Wilmington, Delaware
  - c. Gettys ACES, Racine, Wisconsin
  
4. Evaluate the properties of various storage media that are already being researched in the field.
  - a. Specific Heat
  - b. Heat of fusion
  - c. Density

- d. Temperature range.
- e. Toxicity
- f. Corrosion
- g. Availability
- h. Cost

5. Select Candidate Seasonal Storage System

- a. Heating storage
  - 1. Slab storage; heat pump performing cooling in summer and storing rejected heat in the building slab for winter use.
  - 2. Solar energy hot water storage
- b. Cooling storage
  - 1. Ice storage
  - 2. Ice storage using cold air

All the candidate systems were already researched and designed on a small scale.

- 6. Recalculate the load - heating, cooling and DHW
  - a. Peak (Demand)
  - b. Seasonal (Consumption)
- 7. Calculate peak electrical demand for Conventional System.
- 8. Analyze the electric and fuel cost rates.
- 9. Assume escalation rate for energy (refer Appendix).
- 10. Evaluate heat pump with slab.
- 11. Evaluate solar collectors and collector performance for Washington, D.C. (using weather data).
- 12. Evaluate the heat-pump with the ice maker and ice storage.

13. Evaluate the ability of outside air in winter time to make ice.
14. Estimate the initial cost of components for each system.
15. Evaluate maintenance and operation expenses for each system (preventive, progressive and breakdown maintenance).
16. Comparative analysis of all candidate systems on a present worth basis.
17. Comparative evaluation of all candidate systems (opportunities and constraints.)
18. Method of calculation (do now show the calculation).
19. Sources of information.
20. Summarize results in report.

7. KEY ASSUMPTIONS IN THIS STUDY

Following are all the key assumptions used in this study:

A. Climatic Condition

1. Location Washington (WBCO) Lat 38° - 51'N.
2. Winter Design temperature 15°F.
3. Summer Design temperature 93°F DB, 78°F WB.
4. Winter Degree days 4,224 below 65°F.
5. Summer equivalent full load air conditioning hours - 1,000.
6. Winter months November through May.
7. Winter mean total hours of sunshine 1,297 hours.
8. Summer months June through October.
9. Summer mean total hours of sunshine 1,283 hours.
10. Direct sunshine annual possible 58%.
11. The average city water temperature 55°F.

B. Inside Design Conditions

1. Winter - 68°F DB no humidity control, set back temperature 65°F DB
2. Summer - 78°F DB, 65°F WB, no night control.
3. Domestic hot water temperature 110°F.

C. Building Description

- |  |                  |
|--|------------------|
| 1. Total area                          | 2,203,192 sq.ft. |
| 1.1 776 Dwelling Units 950 sq.ft. each | 734,970 sq.ft.   |
| 1.2 Retail and Offices                 | 120,973 sq.ft.   |

- 1.3 Community Storage 39,772 sq.ft.
- 1.4 Archive above grade 336,915 sq.ft.
- 1.5 Archive below grade 920,562 sq.ft.
2. Building walls heat transfer coefficient 0.06 BTU/hr-sq.ft.  
OF (R=16.6).
3. Building roof heat transfer coefficient 0.05 BTU/hr-sq.ft.  
OF (R=20).
4. Windows area 15% of walls area.
5. Windows heat transfer coefficient 0.56 BTU/hr-sq.ft.  
OF (Double Glass).
6. Windows shading coefficient 0.56
7. Infiltration through windows 0.5 CFM/LF of window.
8. Infiltration through doors 0.6 CFM/LF of door.
9. Domestic hot water demand.
  - a. Residential based on 60 gal/day/dwelling unit.
  - b. Commercial areas based on 100 sq. ft./person  
2/gal/person/day.

D. Cost Assumption

1. The chilled water pumps and the hot water distribution pumps not considered since their use applies to all the schemes.
2. The operation annual cost based on annual energy cost only, because maintenance and labor applies to all the schemes. (See Section 18)
3. Energy Cost

Electric energy cost based on: Response of Potomac Electric Power Company to orders nos. 5739 and 5869 for Time of Day Peak Load Pricing Rates case no. 630 dated May 20, 1977.

Energy Charge	Summer	Winter
On peak period 12:00 noon to 8 PM.	2.57¢ per KWH	2.36¢ per KWH
Intermediate period 8 AM to 12:00 noon and 8 PM to 12:00 Midnight	2.36¢ per KWH	2.26¢ per KWH
Off peak period 12 Midnight to 8 AM	1.855¢ per KWH	1.855¢ per KWH
Average cost of Energy Charge	2.261¢ per KWH	2.158¢ per KWH
Demand Charge	Summer	Winter
On peak period billing demand	\$5.25 per KW	\$3.15 per KW
Intermediate period billing demand	3.15 per KW	2.40 per KW
Off Peak Period billing demand	2.4 per KW	2.4 per KW

Saturdays, Sunday and Holidays off peak period all hours.

The billing demands shall be the maximum 30 minute demands recorded during each rating period of the billing month.

Fuel oil No. 2 cost - \$0.5 per gallon

The present worth of energy cost escalated over a period of 20 years at a rate of 12% and cost of money at 8%.

E. Solar Collector Annual Storage

1. Annual average collector efficiency 32%.
2. Collector tilt 54°.
3. Azimuth South
4. Collector rate flow 0.025 gpm/sq. ft.
5. Solar system pressure head loss - 50 ft.

6. Collector average inlet temperature 140°F.
7. Storage loss assume 10%.
8. Annual operation hours - 2,580 (mean sunshine annual hours).

F. Slab Storage Assumption

1. Specific weight of concrete 150 lbs./cu.ft.
2. Specific heat of concrete 0.2 BTU/lb.-°F.
3. The ground floor and the walls below grade with total area of 500,000 sq. ft. assumed as a slab storage.
4. Walls slab and floor assumed 1 ft. thickness.
5. Slab temperature difference assumed as 60°F.
6. The heat pump system COP assumed as 3.

G. Ice Storage Assumption

1. Ice heat of fusion 144 BTU/lb.
2. Ice storage density 50 lb./cu. ft.
3. Ice storage insulation walls and bottom (R=18.2) equivalent to 2" polyurethane; top (R=36.4) equivalent to 4" polyurethane.

H. Ice Storage System Using the Ability of Winter Air to Generate Ice

1. Number of degree hours below and including 25°F - 6,727
2. Number of hours of occurrence is 728
3. Brine used 20% menthol, 80% water
4. Brine density 60.4 lb/cu.ft. specific gravity 0.968
5. Specific heat 0.97 BTU/lb -°F
6. Freezing point 4.5°F.

8. EVALUATION OF STORAGE MEDIA PROPERTIES

Following is an evaluation of the properties of various storage media that are already being researched in the field. Table 8.1 below shows their main properties pertaining to thermal storage.

Table 8.1

<u>Properties</u>	<u>Water</u>	<u>Paraffin C14-C16</u>	<u>Water</u>	<u>Slab Concrete</u>	<u>Salt Sodium Sulfate</u>
Specific Heat (BTU/lb/°F)	1	0.936	1	0.2	0.78
Heat of fusion (BTU/lb)	144	71.1	-	-	108
Density (lb/cu.ft.)	55.4	48	62.4		83
Temperature Range (°F)	32	35.40°	100°-200°	100°-400°	89°
Heat Storage Density (BTU/cu.ft.)	8000	3412	6240	3000	9612
Toxicity	No	No	No	No	No
Cost (\$)/lb	0	0.05	0	-	0.37
Corrosion	No	No	No	No	Yes
Availability	Plentiful	Scarce	Plentiful	Avail- able	Scarce
Application	Cooling	Cooling	Heating	Heating	Air Heating

From the data given in the table it follows that water is the most suitable medium for cooling storage because of its high heat storage density (8000 BTU/cu.ft.) and a temperature range suitable for cooling applications. It is also the most suitable storage medium for heating applications.

Its availability is limitless, its cost is low, it has an appreciable specific heat, does not require special vessels, it is safe to handle, non toxic, easy to handle in a distribution network and the technology of its use is readily available and well familiar to engineers.

The next storage medium, paraffin, has a very high cost and is not yet available in commercial quantities. Being derived from crude oil, its cost of production is likely to increase as the price of oil goes up.

Its fusion temperature falls within the range of air conditioning cooling applications but its energy density is about 2.4 times less than that of water. It contains about 10% by volume of air entrained in it which effectively reduces heat transfer.

Paraffin requires containers made of special material and the use of plastic containers would have to be carefully investigated for the possibility of environmental stress cracking.

Because of rarity of utilization there is not as yet any proven technology and established engineering knowledge of its handling and utilization.

Although suitable for storage, paraffin is not suitable as an energy transfer medium, thus requiring special heat exchangers for energy transfer.

The slab concrete is suitable as a sensible heat storage medium with a heat storage density of about 3000 BTU/cu ft. which is about half of the heat storage density of water in the same temperature range. It requires, however special technology for heat exchange and energy transfer.

Although the building slab does not require any additional cost (beyond that of the building structure) the installation of water coils in the concrete requires a cost which may be prohibitive.

Sodium sulfate (Glaubert salt) has a high heat of fusion at the temperature of 89°F which is suitable for direct air heating applications.

This salt is useful where minimum space or weight is a consideration and where a fairly constant storage temperature is desired. However, the expense is prohibitive, the storage containers are more expensive than conventional ones, and the availability of long lasting eutectics is limited. It is still not available in commercial quantities and the technology of handling is not yet fully developed. Special containers are necessary as the salt is capable to penetrate materials which results in water evaporating from the solution raising the concentration and reducing thereby the efficiency of the solution. The recommended materials for containers are high density polyethylene or polypropylene.

Concluding, it should be stressed again that at the present time water is the most suitable and efficient storage medium.

## 9. SELECTION OF CANDIDATE SEASONAL STORAGE SYSTEMS

### a. Heating storage

#### 1. Slab storage

The heat rejected by the refrigeration cycle during the summer time could be stored in the building slab mass, by means of condenser water coils embedded in the concrete slab. During the winter this stored heat could be used as a heat source for a heat pump to heat the building.

The water coil circuit embedded in the slab will be connected to both the condenser and chilled water circuits.

In summer time the rejected by the refrigeration cycle heat will charge the slab and will also be used to preheat the domestic hot water.

After the slab is charged and reaches a certain predetermined temperature, further heat dissipation will take place through an evaporative condenser. In winter time this evaporative condenser will serve as an evaporator and will absorb heat from outside air serving as a heat source for the heat pump which will heat the building. This will occur when the outside temperature is above 40°F. Below this temperature the chilled water circuit will connect to the coils embedded in the slab and the heat stored in the slab in the summer will serve as a heat source for the heat pump heating the building.

#### 2. Solar Energy Hot Water Storage

Solar energy collected in the summer will be used to preheat domestic hot water and the excess of it will be stored in a daily storage tank. When the temperature in the tank reaches a certain predetermined value hot water will be transferred to a large annual storage tank usually located below grade. In winter time the solar energy will be first extracted from the daily storage tank and when the temperature in the tank drops below a certain value, the annual storage tank will be used. The fundamental advantage of such a system is that solar energy is collected when it is most available, namely in the summer and then stored in a storage tank for later use when direct solar energy is least available, that is in the winter time.

b. Cooling storage

1. Ice storage

This concept uses an electrically driven heat pump for heating the buildings in winter and for meeting the DHW requirements. The heat pump uses as its heat source water contained in a large underground bin which could be part of the building structure.

At the beginning of the cycle the water is assumed to be at 56°F. In the course of the heat pump operation the water would cool down to 32°F and then its heat of fusion of 144BTU/lb would be extracted turning it gradually into ice.

There are two methods of ice generation. In one, coils carrying chilled brine are immersed in the ice bin and ice is formed in the immediate vicinity of the brine-carrying coils. The other method, suggested here, uses an "ice maker" which is a direct expansion evaporator over which water is sprayed and is being converted into ice.

Over the duration of the winter enough ice is accumulated and stored in the bin to provide "free" cooling in summer time, the "cooling" energy being a "by-product" of the heat pump operation in winter time. During the summer ice water is circulated through the coils of the air conditioning equipment extracting heat from the air passing over the coils. As heat is being dumped into the ice bin the ice begins to melt progressively until all of it is converted to water which then begins to heat up until it reaches 56°F when no more cooling is available. Some ice may also be produced during the summer when domestic hot water is generated, although it is more efficient to produce chilled water instead of ice. If there is a deficiency in "cooling" energy in the summer, it would be supplemented by generating chilled water during off peak periods and storing it for peak period use. The annual cycle is then completed.

The system comprises therefore cooling towers and chillers, in parallel with the ice making evaporators, for off peak summer operation when the chilled water generated could be stored in the ice storage bin for on peak operation. Chilled water could also be supplied directly from the chillers to the buildings distribution system.

The condensers could operate in summer time in conjunction with the cooling towers but one condenser is provided with a double bundle for simultaneous operation with the cooling tower and domestic hot water heater. Provision is made to separate the ice from the ice maker and drop it into the storage bin.

When the ice reaches a predetermined thickness hot refrigerant gas is passed through the coils and melts the ice around it which detaches from the coils and falls into the bin.

With the ice maker the generation of ice is more efficient than with chilled brine carrying coils where the ice layers formed around the coils reduce the heat transfer between the chilled brine and the water and impede further ice formation.

But the disadvantage of the ice maker operation is that the ice is not solid as with the chilled brine method but contains air spaces in it so that instead of a solid block of ice we get a mixture of ice, air and water which in fact decreases the density of ice thus requiring a greater volume of the bin.

## 2. Ice Storage using Cold Air

In some cases the ice generated during the winter as a "by product" of heating and DHW is not sufficient to cover all the summer cooling requirements. The deficiency in cooling requirements could be made up by freezing water or any other phase change material (with freezing temperature suitable for air conditioning operation), by circulating cold brine through it. The brine itself being cooled by outside winter cold air.

A brine circuit includes a circulating pump and two coils: one located outside across which cold air is blown while the other is located in water. Whenever the air is 25°F or below it cools the brine which in turn cools the water thus generating ice.

10. COOLING AND HEATING PEAK LOADS AND ANNUAL ENERGY REQUIREMENTS

The weather data used were those of the Andrews Air Force Base, Washington, D.C.

The number of heating degree days is 4224 per year and for cooling 700-1200 full load hours were assumed.

The summer outside design temperatures are 93°F DB and 78°F WB while the inside conditions are 78°F DB and 65°F WB.

For winter the outside conditions are 15°F and inside 68°F.

The average city water temperature is 55°F.

A. Heating and Domestic Hot Water Peak Loads

	<u>Heating Million BTUH</u>	<u>DHW* Million BTUH</u>	<u>Total Million BTUH</u>
1. Residential (776 dwelling units)	7.65	2.10	9.75
2. Commercial Areas above grade	4.30	0.50	4.80
3. Archives below grade	4.80	Included in the above	4.80
Total	16.75	2.60	19.35

The heating equipment will thus be sized according to the total heating demand.

---

\*Based on 53,000 gallons per day:

Residential	60 gal/day per dwelling unit	=	47,000
Offices		=	3,400
Archives		=	2,600
Total		=	53,000

B. The Annual Energy Requirements for Heating

- 1. Residential..... $1.460 \times 10^{10}$  BTU per year
- 2. Commercial areas above grade..... $0.825 \times 10^{10}$  BTU per year
- 3. Archives below grade..... $0.920 \times 10^{10}$  BTU per year
- TOTAL..... $3.205 \times 10^{10}$  BTU per year

C. Annual Energy Requirements for DHW

- 1. Residential..... $7.60 \times 10^9$  BTU per year
- 2. Commercial..... $0.67 \times 10^9$  BTU per year
- TOTAL..... $8.27 \times 10^9$  BTU per year

D. Cooling Peak Loads

- 1. Residential (776 Dwelling Units).....800 tons
- 2. Commercial Areas above grade.....1,000 tons
- 3. Archives below grade.....1,200 tons
- TOTAL.....3,000 tons

E. Annual Energy Requirements for Cooling

Based on 1,000 equivalent full load hours, the annual energy requirement is:

$$3,000 \times 1,000 = 3 \times 10^6 \text{ Ton - Hours}$$

## 11. SELECTION OF A CONVENTIONAL SYSTEM

### 11.1. Equipment Selection

Heating: Two 12,000 MBH No.2 oil fired boilers to cover heating peak load of  $16.75 \times 10^6$  BTUH. One 3,000 MBH No.2 oil fired boiler to cover domestic hot water requirement of  $2.6 \times 10^6$  BTUH.

Cooling: Four 750 ton electrically driven screw compressors and chillers to cover cooling load of 3000 ton. Four 750 ton cooling towers.

### 11.2. Energy Analysis

Annual Energy Requirements:

Heating and DHW	$4.032 \times 10^{10}$ BTU/year
Cooling	3,000,000 Ton - HR/year

Annual Energy Consumption

$$\frac{4,032 \times 10^{10}}{140,000 \times 0.7} = 411,400 \text{ gallons of No.2 oil/year}$$

Electrical energy input to the chillers is  
 $3,000,000 \text{ Ton} - \text{Hrs} \times 1\text{KW/Ton} - \text{Hr} = 3,000,000 \text{ KWH}$  (the cost of operation of chilled water pumps is not considered since their use applies to all the schemes).

### 11.3. Cost Analysis

At \$0.45/gallons, 411,400 gallons of No.2 oil yields cost \$185,130.

Using Potomec Electric Company (Pepco) electrical rates.

energy charge

$$3,000,000 \text{ KWH} \times \$ 0.02261/\text{KWH} = \$ 67.830$$

demand charge

$$3000 \text{ KW} \times \$ 5.25/\text{KW} \times 5 \text{ Mo} = \underline{\$ 78.750}$$
$$\text{Total} = \underline{146.580}$$

The total annual energy cost is thus  
 $\$185,130 + \$146,580 = \$331,710$

The System First Cost:

Four 750 ton screw compressors	= \$ 480,000
Four cooling towers	= \$ 120,000
Two 12,000 MBH boilers	= \$ 120,000
One 3,000 MBH boiler	= \$ 30,000
Total	= <u>\$ 750,000</u>

The distribution systems are the same for all schemes and therefore will not be considered in the discussion of the systems.

## 12. EVALUATION OF HEAT PUMP WITH SLAB

### 12.1 Energy Analysis

The ground floor and the below grade walls of the Archives structure, with a total volume of 500,000 cubic feet, will serve as the slab storage.

Using the specific heat of concrete of 0.2 BTU/lb -°F, the specific weight of 150 lb/cu. ft. and assuming further a temperature difference of the slab storage as 60°F, the heat capacity of the slab storage will be:

$$500,000 \times 150 \times 0.2 \times 60 = 900 \times 10^6 \text{ BTU}$$

Using a heat pump with a COP = 3 with the slab as the heat source the heat obtainable is:

$$\frac{900 \times 10^6}{2} \times 3 = 1,350 \times 10^6 \text{ BTU}$$

This is only a small percentage of the winter heating and domestic hot water requirements. Thus:

$$\frac{1,350 \times 10^6 \text{ BTU}}{3.205 \times 10^{10} \text{ BTU}} \times 100 = 4.2\%$$

Whenever outside air cannot be used as a heat source with a high COP the slab will assist the heat pump as a heat source. The annual energy input with COP=3 for winter heating and DHW is:

$$\frac{3.205 \times 10^{10} \text{ BTU} + 0.345 \times 10^{10} \text{ BTU}}{3413 \text{ BTU/KWH} \times 3} = 3,447,600 \text{ KWH}$$

For summer cooling the energy input is:

$$\frac{3 \times 10^6 \text{ Ton} - \text{HR} \times 12,000 \text{ BUT/Ton-HR}}{3414 \text{ BTU/KWH} \times 3.5} = 3,013,700 \text{ KWH}$$

The COP with the slab operation will be lower than with the evaporative condensers operation, therefore an average COP of 3.5 was assumed.

Annual electrical energy consumption is thus

$$3,447,600 + 3,013,700 = 6,461,300 \text{ KWH}$$

12.2. Cost Analysis

12.2.1. Energy Cost

Annual energy cost using Potomac Electric Company (Pepco) electric rates:

Summer

Energy charge 3,013,700 KWH x \$0.02261/KWH = \$ 68,140

Demand charge 3000 KW x \$5.25/KW x 5 Mo = \$ 78,750  
Total = \$146,890

Winter

Energy charge 3,447,600 KWH x \$0.02158/KWH= \$ 74,400

Demand charge 3x875x0.746x\$3.15/KWx7 Mo = \$ 43,180  
(3 compressors 875HP each) \$117,580

Total annual energy cost \$ 146,890  
\$ 117,580  
\$ 264,470

Annual energy cost savings relative to a conventional system  
= \$331,710 - \$264,470 = \$67,240

12.2.2. System First Cost:

4 Screw Compressors = \$ 600,000

Piping assembled in the  
concrete = \$ 2,669,000

Evaporative condensers = \$ 100,000

Control system = \$ 31,000  
\$ 3,400,000

12.2.3. Comparison With Conventional System

Incremental investment cost relative to conventional system  
= \$3,400,000 - \$750,000 = \$2,650,000

simple "payback" =  $\frac{2,650,000}{67,240} = 39.4$  years

The Present Worth of savings using 8% cost of money and an escalation rate of 12% over a period of 25 years = \$2,791,000 (see economic analysis) which slightly exceeds the incremental cost of \$2,650,000.

Thus the discounted "payback" is less than 25 years.

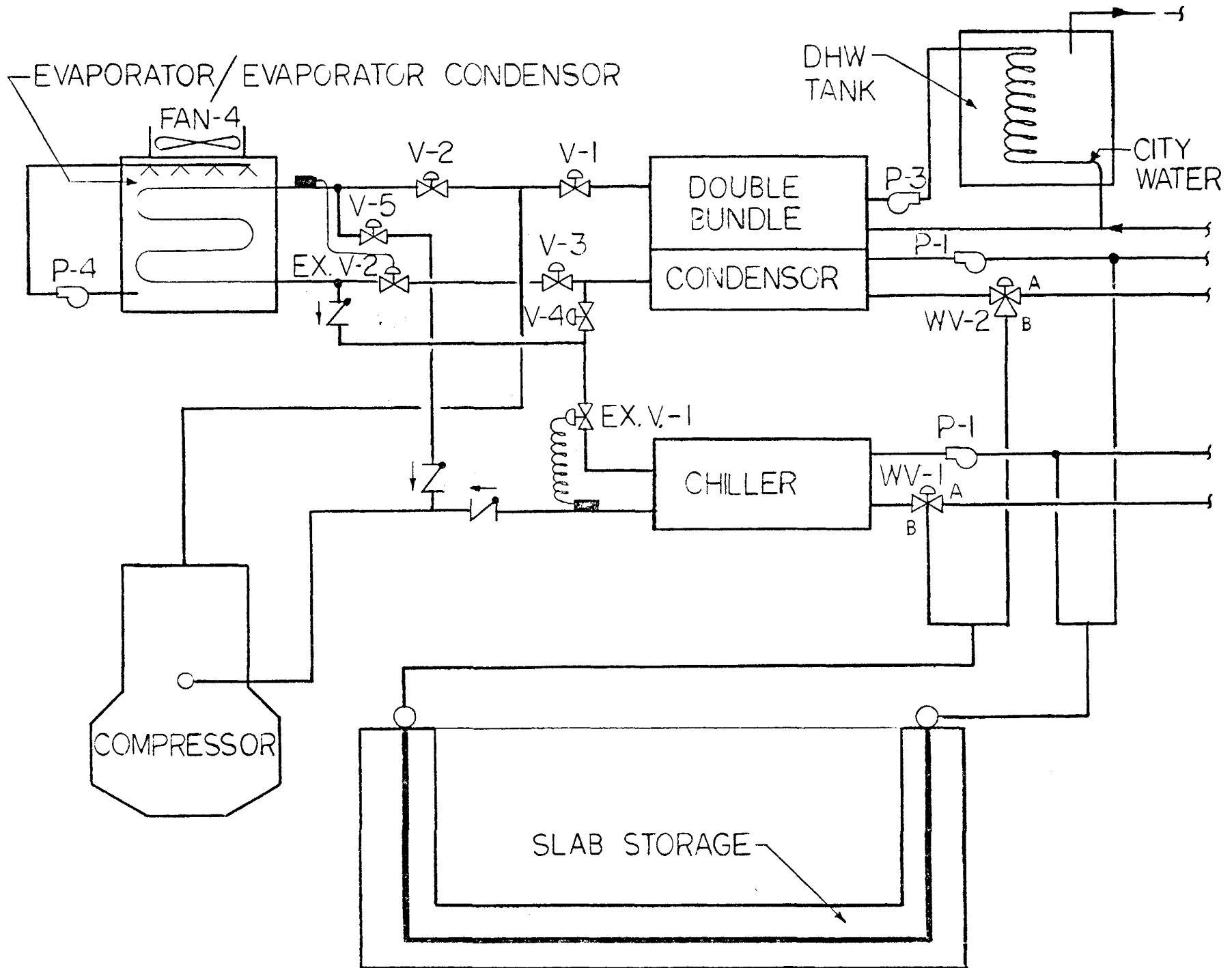
### 12.3 General Evaluation

Obviously, the slab storage is inadequate for heating purposes. Furthermore the initial cost of installing a pipe network within the slab cannot be justified by the savings in energy.

From the temperature point of view the temperature of the condenser water cannot be higher than 130°F (which implies a low COP) and even then the available temperature for heating will be low because of the heat exchangers involved in the functioning of the system each with its own efficiency and approach especially with heat transfer between pipes and slab.

In our case, as said above, the heat stored in the slab is not sufficient for heating requirements and this scheme must therefore be discarded. It should be borne in mind, however, that slab storage could, in certain cases, prove efficient; namely, where waste heat at high temperatures is available concurrently with large slab volumes.

In the design of such slab storage system, care must be taken to provide for temperature stresses brought about by high temperature fluctuations. These stresses could have an adverse influence on the building structure stability.



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HP-SLAB STORAGE SYSTEM

HP SLAB STORAGE SYSTEM MODE OF OPERATION

MODE DESCRIPTION	v-1	v-2	v-3	v-4	v-5	EXV-1	EXV-2	WV-1	WV-2	P-1	P-2	P-3	P-4	f-1
1) Cooling required Slab storage below 120°F DHW below 120°F	Open	cl.	cl.	open	cl.	open	cl.	Open A	open B	on	on	on	off	off
2) Cooling required Slab storage above 120°F DHW below 120°F	Open	cl.	cl.	open	cl.	open	cl.	Open A	Cl.	on	off	on	off	off
3) Cooling required Slab storage above 120°F DHW above 120°F	cl.	open	cl.	cl.	cl.	open	cl.	Open A	cl.	on	off	off	on	on
4) Heating and DHW required No cooling required Outside air above 40°F	open	cl.	open	cl.	open	cl.	open	cl.	open A	off	on	on	off	on
5) Heating DHW and cooling required simultaneously	open	cl.	cl.	open	cl.	open	cl.	Open A	open A	on	on	on	off	off
6) Heating DHW required No cooling required Outside air below 40°F	open	cl.	cl.	open	cl.	open	cl.	Open B	open A	on	on	on	off	off
7) DHW required No heating or cooling required Outside air above 40°F	open	cl.	open	cl.	open	cl.	open	cl.	cl.	off	off	on	off	on
8) DHW required No heating or cooling required Outside air below 40°F	open	cl.	cl.	open	cl.	open	cl.	open B	cl.	on	off	on	off	off

13. EVALUATION OF SOLAR COLLECTORS AND COLLECTOR PERFORMANCE

WASHINGTON, D.C. (Using Weather Data)

13.1 Solar Collector System

Total space heating and DHW required per year  $4.1 \times 10^{10}$  BTU/year

Solar Collector System

Collector tilt  $54^\circ$

Collector area 260,000 sq ft

Average collector efficiency 39%

Average collector temperature  $140^\circ\text{F}$

Storage volume  $20.2 \times 10^6$  Gal =  $2.74 \times 10^6$  cubic feet

13.2 Energy Analysis

13.2.1. Solar Energy

Solar energy collectable per square foot per year 162,419 BTU

Total solar collectable  $4.2 \times 10^{10}$  BTU/year

Solar energy used directly in winter and summer  $2.54 \times 10^{10}$  BTU/year

Solar energy stored from summer for use in winter  $1.6836 \times 10^{10}$  BTU/year

Storage loss (10% of energy stored)  $1.683 \times 10^9$  BTU/year

Deficiency in energy in winter time  $1.66 \times 10^{10}$  BTU/year

13.2.2. Electrical Energy Input

Solar collector pump power required 110 HP

The annual energy consumption for solar collector pump working  
2580 hours (sunshine hours)

$110\text{HP} \times 0.746 \text{ KW/HP} \times 2580 \text{ hours} = 211715 \text{ KWH/year}$

Summer energy consumption

Cooling -	3,000,000 KWH
Solar Collector pump -	105,206 KWH

Winter energy consumption

Solar collector pump	106,354 KWH
Annual energy consumption heating & cooling	\$3,212,000 KWH

13.3 Cost Analysis

13.3.1. Energy Cost

Annual Energy Cost = \$156,000

13.3.2. Equipment First Cost

Solar Collector System (at \$30/sq.ft.) = \$7,800,000

Storage tank (at \$0.50/cu.ft.) = \$1,400,000

Chillers and pumps \$ 480,000

Cooling towers \$ 120,000  
\$9,800,000

13.3.3. Comparison With Conventional System

Incremental Cost = \$9,800,000 - \$750,000  
\$9,050,000

Annual savings in energy = \$331,710 - \$156,000  
= \$175,710

Simple "payback"  $\frac{\$9,050,000}{\$175,710} = 59 \text{ years}$

Discounted "payback" with 12% fuel escalation rate and 8% cost of money = 29 years.

# SOLAR ENERGY ANNUAL STORAGE

DHW COLLECTOR  
85,000  
SQ. FT.

DOUBLE BUNDLE  
CONDENSOR CONNECTION

DHW  
STOR.

CITY WATER

DHW  
BACK-UP  
BOILER

HEATING COLLECTOR  
174,000  
SQ. FT.

TO LOAD

ANNUAL STORAGE TANK  
2800,000 GALLONS

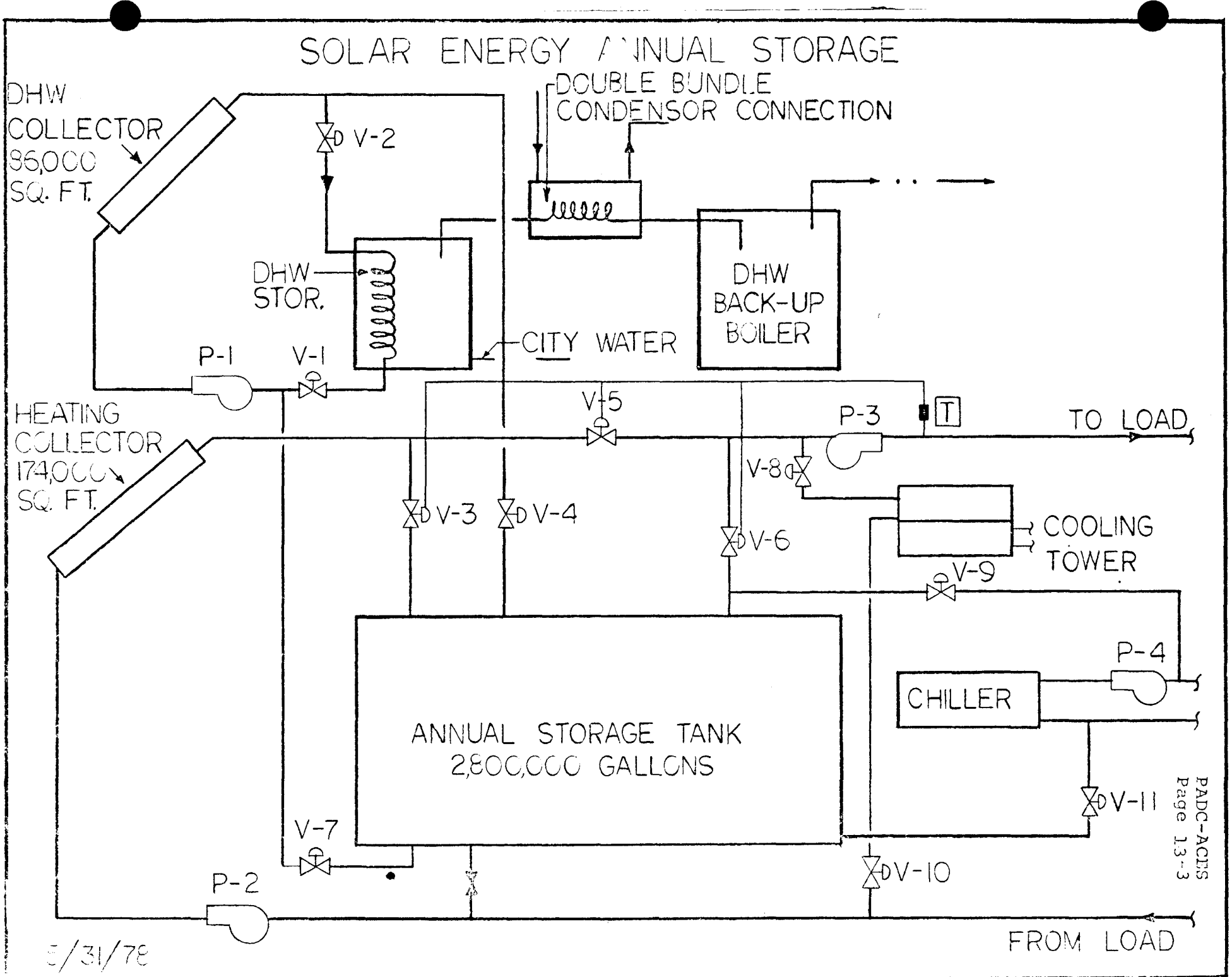
COOLING  
TOWER

CHILLER

PADC-ACLS  
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FROM LOAD



SOLAR ENERGY ANNUAL STORAGE MODE OF OPERATION

MODE DESCRIPTION	v-1	v-2	v-3	v-4	v-5	v-6	v-7	v-8	v-9	v-10	v-11	p-1	p-2	p-3	p-4
1) Summer DHW required No heating required	open	open	open	cl.	cl.	cl.	cl.	cl.	cl.	cl.	cl.	on	on	off	on
2) Summer No heating or DHW required	cl.	cl.	open	open	cl.	cl.	open	cl.	cl.	cl.	cl.	on	on	off	on
3) Winter DHW and heating required Solar energy available	open	open	mod. open	cl.	mod. open	cl.	cl.	cl.	cl.	cl.	cl.	on	on	on	off
4) Winter DHW and heating required No solar available	cl.	cl.	cl.	cl.	cl.	open	cl.	cl.	cl.	cl.	cl.	off	off	on	off
5) Winter DHW and heating required No solar available Storage tank temperature below -90°	cl.	cl.	cl.	cl.	cl.	cl.	cl.	open	open	open	open	off	off	on	on

cl. = closed  
mod.= modulate

General Evaluation

The absolute savings in energy per year are:

$$\frac{4.1 \times 10^{10}}{140,000 \times 0.71} = 410,000 \text{ gallons of No.2 oil}$$

which at \$0.45/gallon amounts to \$205,000.

The system affords big energy savings although the initial cost lay-out is high.

The collector area is based on annual collected energy and is substantially smaller than if it were based on winter months collectable energy.

This is due to the use of a storage system which is much cheaper to install than solar collectors. The storage tank cost is only \$0.5/cu.ft. as compared to the cost of solar collectors which is about \$30.0/sq ft.

Also, the larger the storage system the less the cost per cu/ft and the smaller the losses.

The table below emphasizes the fact of reduced collector area when it is based on annual collected solar energy instead of being based on collected solar energy in winter months.

Ratio of collector area based on winter demand with 100% participation to collector area based on annual collected solar energy.

Nov.	Dec.	Jan.	Feb.	March
1.57	2.70	2.70	2.25	1.5

14. EVALUATION OF HEAT PUMP WITH ICE MAKE AND ICE STORAGE

14.1 Energy Analysis

14.1.1 Winter Energy Requirements

The annual energy requirements for heating were seen to be  $3.205 \times 10^{10}$  BTU per year and for DHW  $0.827 \times 10^{10}$  BTU per year.

Assuming five heating months and seven cooling months, the DHW could be split up into winter and summer requirements by using the same ratio of heating and cooling months.

$$\text{Thus: DHW winter} \approx \frac{5}{12} \times 0.827 \times 10^{10} = 0.345 \times 10^{10} \text{ BTU Per year}$$

$$\text{DHW summer} = \frac{5}{17} \times 0.827 \times 10^{10} = 0.482 \times 10^{10} \text{ BTU Per year}$$

The heating and winter DHW requirements are therefore:

$$3.205 \times 10^{10} + 0.345 \times 10^{10} = 3.55 \times 10^{10} \text{ BTU Per year}$$

14.1.2 Summer Energy Requirements

Summer Cooling Load = 3,000 Ton

Summer Energy Requirements with 1000 Equivalent Full Load hours assumed

$$\text{Refrigeration} = 3.0 \times 10^6 \text{ Ton-Hours}$$

$$\text{DHW} = 0.482 \times 10^{10} \text{ BTU}$$

14.1.3 "Cooling" Energy Available From Heat Pump

Using a heat pump with 20° F. compressor suction temperature and 130° F. compressor discharge temperature a COP of 3.2 is obtained. This means that for every BTU energy input we take from the heat source 2.2 BTU and deliver on heating 3.2 BTU.

Thus the "cooling" energy available from heat pump operation satisfying the winter heating and DHW requirement is

$$\frac{3.55 \times 10^{10}}{3.2} \times 2.2 \times 10^6 / 12000 = 2.03 \times 10^6 \text{ Ton-Hours}$$

which is

$$\frac{2.03 \times 10^6}{3.0 \times 10^6} \times 100 = 67.7\% \text{ of summer cooling requirements.}$$

14.1.4 Refrigeration Energy Generated in the Summer from  
DHW

There is a deficiency in "cooling" energy of  $(3.0 - 2.03) \times 10^6 = 0.97 \times 10^6$  Ton Hours.

During the summer we can generate DHW using the heat pump and thereby generate ice. The DHW requirement for summer was seen to be  $0.482 \times 10^{10}$  BTU. Therefore, the quantity of "cooling" energy that can be produced is:

$$\frac{0.482 \times 10^{10}}{3.2} \times 2.2 = 0.331 \times 10^{10} \text{ BTU}$$

$$\text{or } \frac{0.331 \times 10^{10}}{12,000} = 2.76 \times 10^5 \text{ Ton-Hours}$$

However, if in generating DHW in the summer we produce chilled water instead of ice we can operate the compressor at 35°F saturation suction temperature (instead of 20°F as for ice) and 130°F saturation discharge temperature and thereby obtain a better COP, say 3.84.

In such case the annual chilled water energy obtained from DHW generation would equal to

$$\frac{0.482}{3.84} \times 2.84 \times 10^{10} = 0.3565 \times 10^{10} \text{ BTU}$$

$$\text{or } \frac{0.3565 \times 10^{10}}{12000} = 2.97 \times 10^5 \text{ Ton-hours}$$

Thus the total energy available for cooling is

$$\begin{aligned} &(2.03 + 0.297) \times 10^6 \text{ Ton-Hours per year} \\ &= 2.327 \times 10^6 \text{ Ton-Hours per year} \end{aligned}$$

14.1.5 Sizing the Storage Ice Bin

It is assumed that the entire refrigeration energy generated by the heat pump in the winter produces ice. Sensible heat of the ice water is neglected. Thus the ice generated is

$$\begin{aligned} &\frac{2.03 \times 10^6 \text{ Ton-Hours} \times 12000 \text{ BTU/Ton-Hour}}{144 \text{ BTU/lb.}} \\ &= 1.68 \times 10^8 \text{ lbs. of ice} \end{aligned}$$

with an assumed density of ice of 50 lb./cu. ft. the volume of the ice bin is

$$\frac{1.68 \times 10^8}{50} = 3.4 \times 10^6 \text{ cubic feet}$$

This yields  $\frac{3.4 \times 10^6}{2.03 \times 10^6} = 1.68$  cubic feet of ice storage per Ton-Hour. Comparing with chilled water storage with a 15° temperature difference

$$\frac{62.4 \times 15}{12,000} = 0.078 \text{ Ton-Hour per cub. ft.}$$

or  $\frac{1}{0.078} = 12.8$  Cub. ft. per Ton-Hour

Thus for the same refrigeration energy capacity the ice needs  $\frac{12.8}{1.68} = 7.6$  less storage volume than chilled water with a 15°F temperature difference.

#### 14.1.6 Ice Bin Storage Loss

Since the surface to volume ratio is more favorable for large tanks than for small ones, both the cost and losses per unit volume are smaller for large scale storage systems.

In any study of heat leakage, one has to consider the top, sides and bottom of the bin as separate problems since each obeys a different set of rules.

Assuming a height of the ice bin as 20 feet, the dimensions of the bin are:

$$\left( \frac{3.4 \times 10^6}{20} \right)^{\frac{1}{2}} = 412 \text{ feet square.}$$

This yields a top and base area of 170,000 square ft. each and an area of 33,000 square feet for the side walls. The bin will be insulated with 4 in. polyurethane on the top (R-36.4) and with 2 in. polyurethane (R-18.2) on the sides and bottom.

It was found that the annual heat leakage into the bin is equivalent to 212,100 Ton-Hours which represents 9% of the cooling energy stored.

#### 14.1.7 Net Available "Cooling" Energy

$$2,327,000 - 212,100 = 2,114,900 \text{ Ton-Hours}$$

Deficiency in refrigeration energy

$$3,000,000 - 2,114,900 = 885,100 \text{ Ton-Hours}$$

#### 14.1.8 Annual Energy Consumption

Winter electrical energy input is  $\frac{3.55 \times 10^{10}}{3.2 \times 3413} = 3,250,000 \text{ KWH}$

Summer DHW generation

$$\frac{0.482 \times 10^{10}}{3.84 \times 3413} = 367,800 \text{ KWH}$$

Summer electrical energy input to make up the defficiency in "cooling" energy of 885,100 Ton-Hours using 1 KWH/Ton-Hour = 885,100 KWH.

Subtotal for summer = 1,252,900 KWH

Total for year = 4,502,900 KWH

## 14.2 Cost Analysis

### 14.2.1. Energy Cost

Winter

Using the Potomac Electric Company (Pepco) rates for winter of \$0.02158/kwh energy charge and \$3.15/kw demand charge the annual cost comes to:

energy	$3.25 \times 10^6 \times \$0.02158$	= \$ 70,135
demand	$3 \times 875 \times 0.746 \times \$3.15 \times 7 \text{ (months)}$	= <u>\$ 43,118</u>
	<b>Total</b>	= \$113,253

Summer

energy	$1,252,900 \times \$0.02261$	\$ 28,327
demand	$3 \times 650 \times 0.746 \times \$5.25 \times 1.5 \text{ Mo} =$	<u>\$ 11,812</u>
	<b>Total =</b>	<u>\$ 40,139</u>

Thus, the total annual energy cost to satisfy the heating, DHW and cooling requirements amounts to:

$$\$ 113,253 + \$ 40,139 = \$ 153,392$$

#### 14.2.2 Equipment First Cost

Three screw compressors including condensers, chillers, oil seperators and fluid accumulators:	\$ 300,000
6 ice makers with a total capacity of 1125 ton	\$ 500,000
Cooling towers and condenser pumps	\$ 100,000
Ice storage bin structure	<u>\$1,700,000</u>
Total	\$2,600,000

#### 14.2.3 Comparison with Conventional System.

The incremental investment cost relative to a conventional system

$\$2,600,000 - \$750,000 = \$ 1,850,000$

Annual energy cost savings  
compared with a conventional system  $\$331,710 - \$153,392 = \$178,318$

The simple "Payback" 11 years

Present value "payback" (Discounted  
"payback" with an 8% interest rate  
and 12% energy escalation rate) 9 years

#### 14.3 General Evaluation

The system has a very favorable "payback" both simple and  
discounted.

The heat pump produces 2,114,900 Ton-Hours of "cooling" energy.

This energy could be considered "free" energy since it is a "by-product" of the generated heating and DHW requirements.

The savings in acquiring this energy could be calculated by assuming it to be produced by a conventional refrigeration system with 1.0 KW/ton electrical input (compressor, condenser water pumps, and cooling tower fans, chilled water pumps excluded).

Using the Potomac Electric Company (PEPCO) service rates of \$0.02261/KWH energy charge and \$5.25/KW demand charge the annual savings amount to:

Energy charge	2,114,900 kwh × \$0.02261/kwh	= \$ 47,818
Demand charge	3,000 tons × kw/ton × \$5.25/kw × 4½ months	= \$ <u>70,875</u>
	Total	\$118,693

In addition there is a cut in Demand of 3,000 KW. These are absolute savings to be distinguished from savings relative to a conventional system.

The COP of this system is also very favorable. Both the heating capacity and the cooling capacity are being utilized in the annual cycle system and the annual COP is calculated as follows:

$$\text{Annual COP} = \frac{\text{heat of rejection + heat of absorption}}{\text{electric input in BTU}}$$

(heating)                      (cooling saved)

which is approximately equal to 5. However, pumping power and heat leakage into the ice bin (approximately 3% of bin capacity per month) reduce the COP so that with the present technology the COP is 4.25 and with the new high efficiency compressors the COP is again approximately 5.

Thus the heating and the domestic hot water requirements  
are  $3.205 \times 10^5$  BTU +  $0.827 \times 10^{10}$  BTU

$$= 4.032 \times 10^{10} \text{ BTU per year}$$

Cooling produced is  $3.0 \times 10^6$  Ton-Hours per year

$$= 3.0 \times 10^6 \times 12000 = 3.6 \times 10^{10} \text{ BTU per year}$$

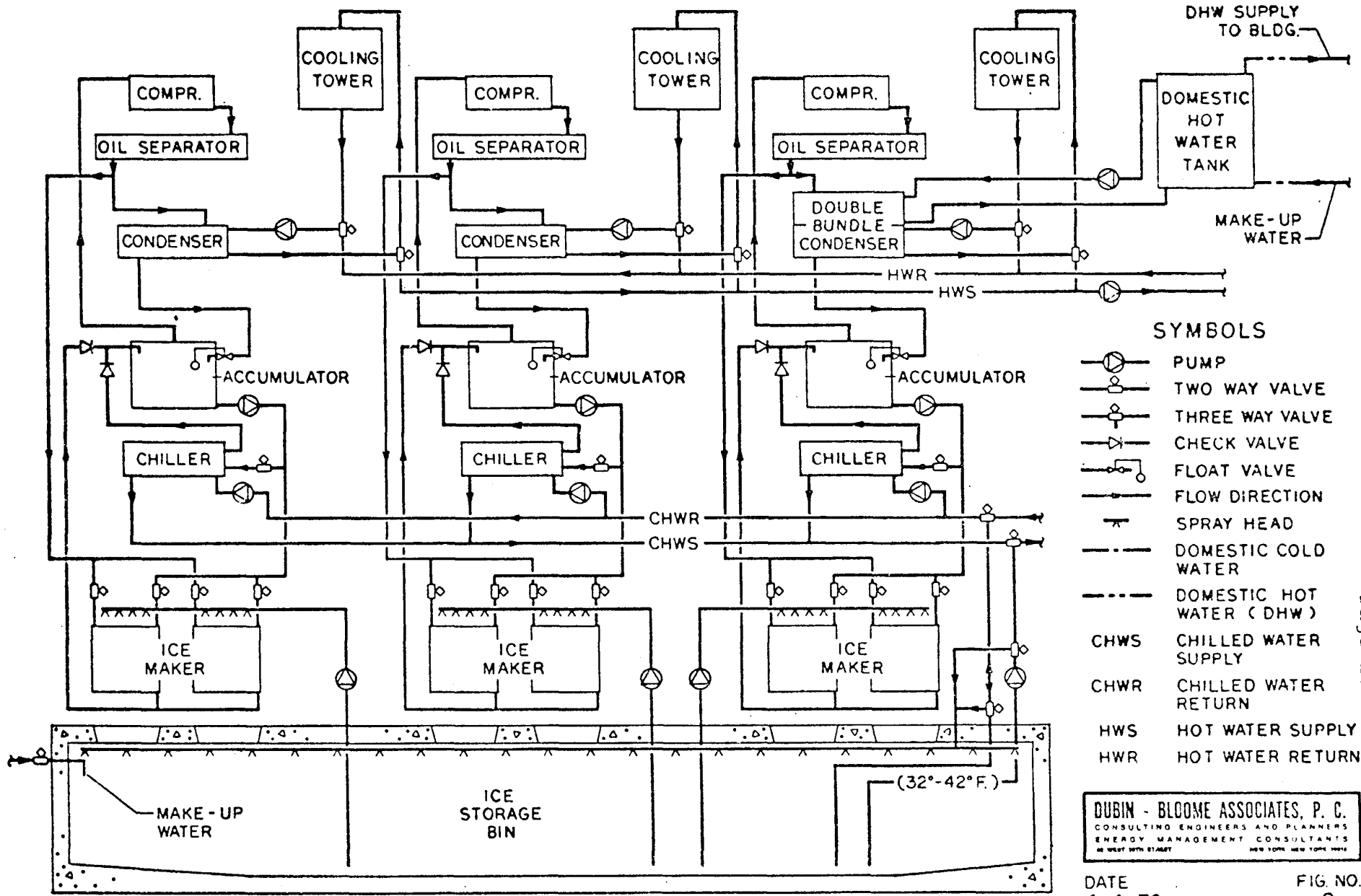
Total heating and cooling =  $(4.032 + 3.6) \times 10^{10} = 7.632 \times 10^{10}$  BTU/year

Annual KWH input is 4,502,900 KWH per year

$$= 4,502,900 \times 3413 = 1.537 \times 10^{10} \text{ BTU per year}$$

$$\text{Annual overall COP} = \frac{7,632 \times 10^{10}}{1,537 \times 10^{10}} = 4.97$$

# ANNUAL CYCLE ENERGY SYSTEM WITH ICE STORAGE



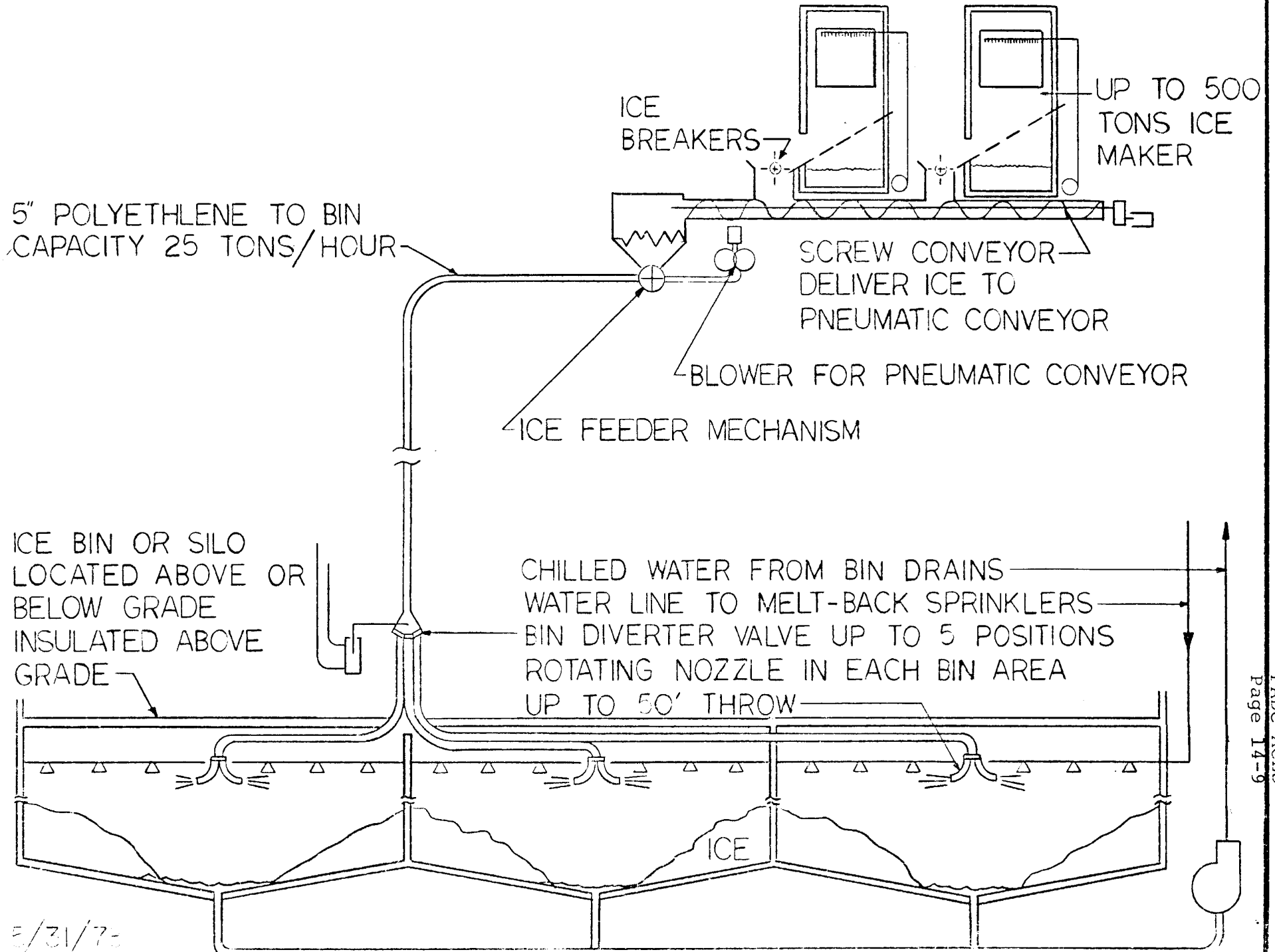
- SYMBOLS**
- PUMP
  - TWO WAY VALVE
  - THREE WAY VALVE
  - CHECK VALVE
  - FLOAT VALVE
  - FLOW DIRECTION
  - SPRAY HEAD
  - DOMESTIC COLD WATER
  - DOMESTIC HOT WATER (DHW)
  - CHWS CHILLED WATER SUPPLY
  - CHWR CHILLED WATER RETURN
  - HWS HOT WATER SUPPLY
  - HWR HOT WATER RETURN

**DUBIN - BLOOME ASSOCIATES, P. C.**  
 CONSULTING ENGINEERS AND PLANNERS  
 ENERGY MANAGEMENT CONSULTANTS  
40 WEST 60TH STREET NEW YORK, NEW YORK 10019

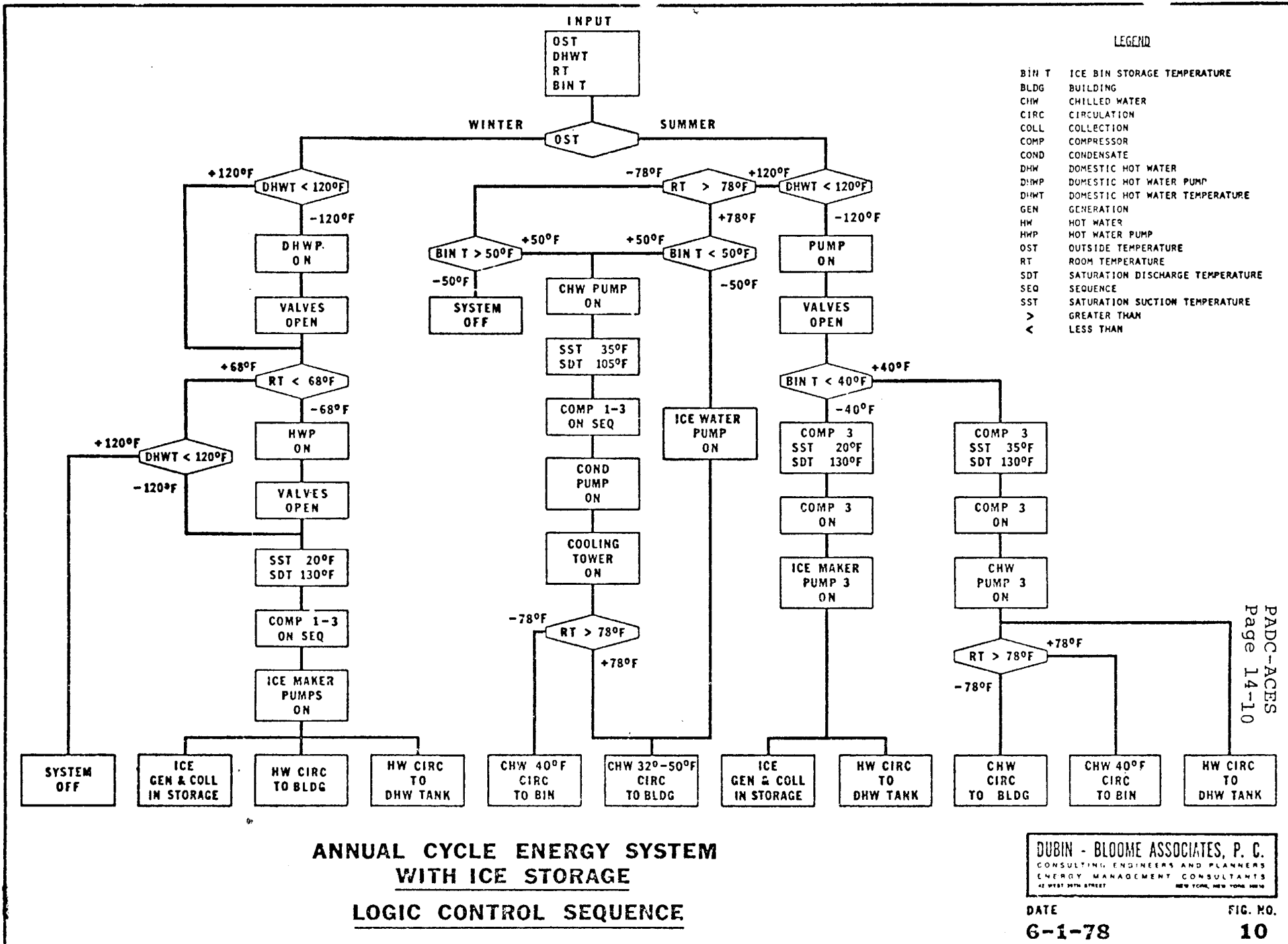
DATE 4-4-78 FIG. NO. 9

PADC-ACES  
 Page 14-8

# SCHEMATIC OF LARGE ANNUAL ICE STORAGE SYSTEM



E/31/75



14.4 EXPLANATION OF ACES LOGIC CONTROL

- 14.4 . 1. The system will operate as a heat pump and ice maker when the room thermostat calls for heating and/or the hot water heater thermostat calls for heating.
- 14.4 , 2. The system will operate as a heat pump and chilled water generator when the hot water heater thermostat calls for heat, the room thermostat indicates no heat required and the ice bin thermostat indicates a temperature of above 50°F.
- 14.4 . 3. The ice water pump is on when the room thermostat calls for cooling and the ice water thermostat indicates that the ice water thermostat indicates that the ice water temperature is below 50°F.
- 14.4 , 4. The system will produce chilled water when the room thermostat calls for cooling and the ice bin thermostat shows a temperature of above 50°F or when the room thermostat does not call for cooling but the ice bin thermostat indicates a temperature above 50°F.

## 14.5 EQUIPMENT SELECTION

### Refrigeration Units

Three screw compressors selected to meet the winter heat load.

Saturated suction temperature (SST)	20°F
Saturated discharge temperature (SDT)	130°F
Cooling capacity per unit	375 Ton
Heat rejection per unit	6.75 x 10 <sup>6</sup> BTUH
Power input per unit	875 BHP

### Cooling Towers

Three cooling towers selected for eventual summer operation

Entering condenser water	95°F
Leaving condenser water	85°F
Cooling capacity per unit	610 Ton
Condenser flow rater per unit	1830 GPM
Summer design temperature	93°F DB, 78°F WB

### Ice Makers

- a) 6 ice makers selected, 2 for each compressor. Each ice maker has the capability to generate about 200 tons of ice during 24 hours.
- b) The amount by weight of the make up water required per day will be equal to the amount by weight of ice generated, that is for 1125 tons of ice per day the daily make up water requirement is about 270,000 Gal.
- c) The three ice maker water pumps supply the make up water for ice generation. The flow rate required is 65 GPM per pump. However to assure this rate of ice generation we multiply this rate by two since only part of the water will be converted to ice.

### Condensers

The condensers were selected for summer time operation since during the summer more heat is being rejected, about  $9 \times 10^6$  BTUH per condenser compared with  $6.7 \times 10^6$  BTUH rejected during winter operation.

### The Ice Bin

The ice bin shall have a volume of  $3.4 \times 10^6$  cub. ft. in order to accommodate and store the ice that was being generated during the winter.

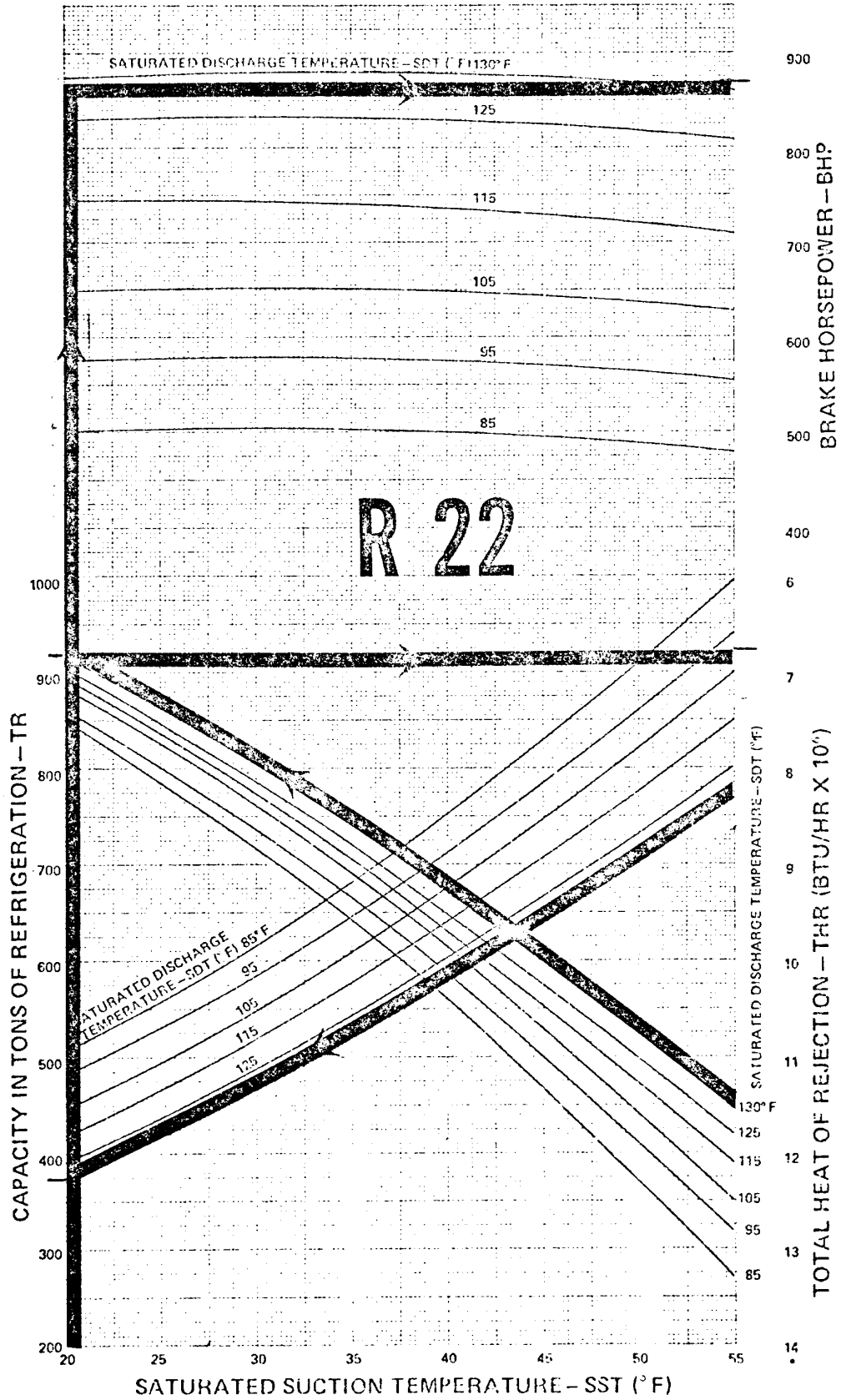
### Chillers

The chillers were selected for the eventual summer operation with 610 ton cooling capacity each and  $10^{\circ}\text{F}$  temperature difference.

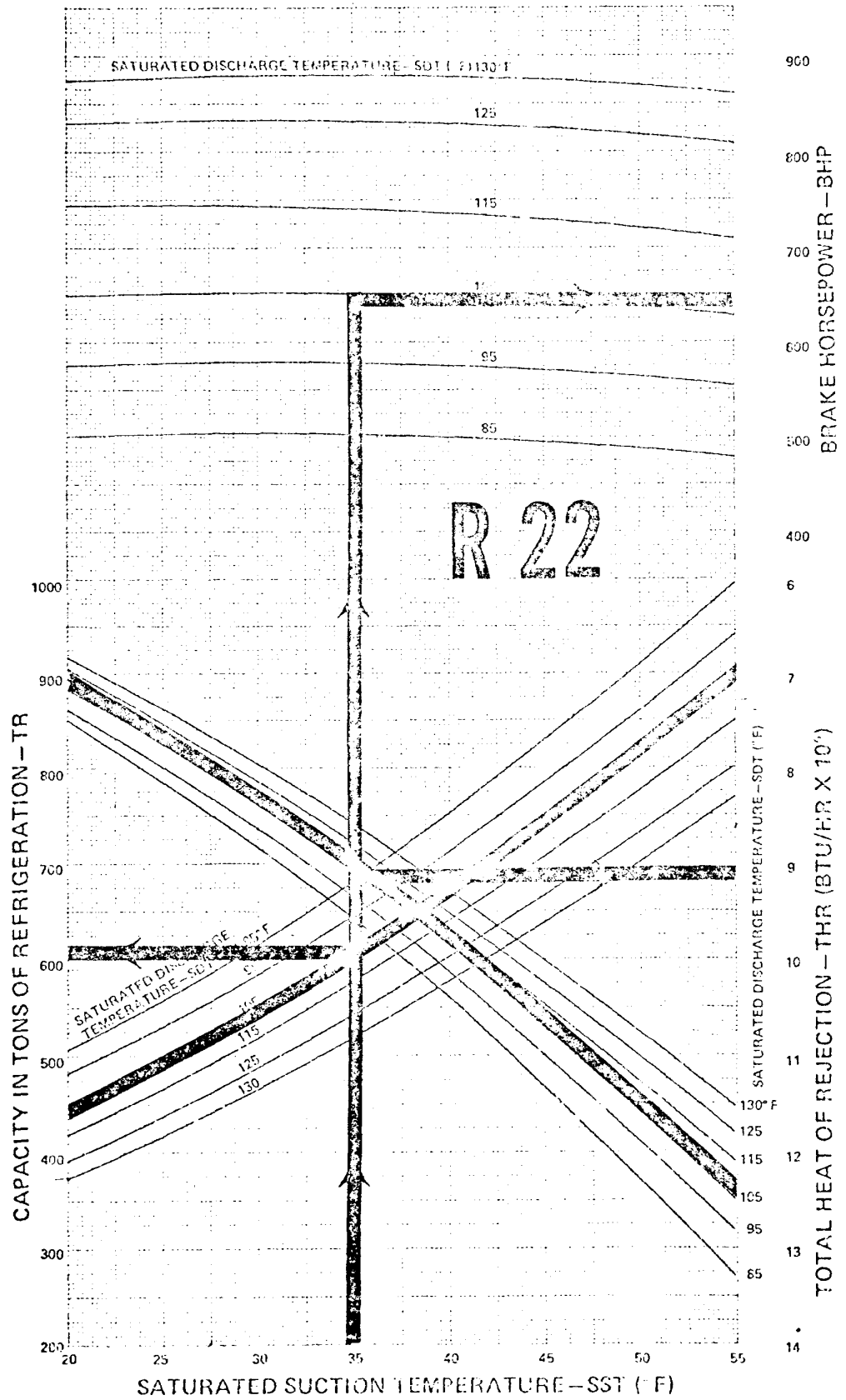
### Domestic Hot Water Storage

The domestic hot water storage tank was selected to have a capacity of 150,000 gallons in order to meet the daily requirements of the Complex with a  $20^{\circ}\text{F}$  temperature difference.

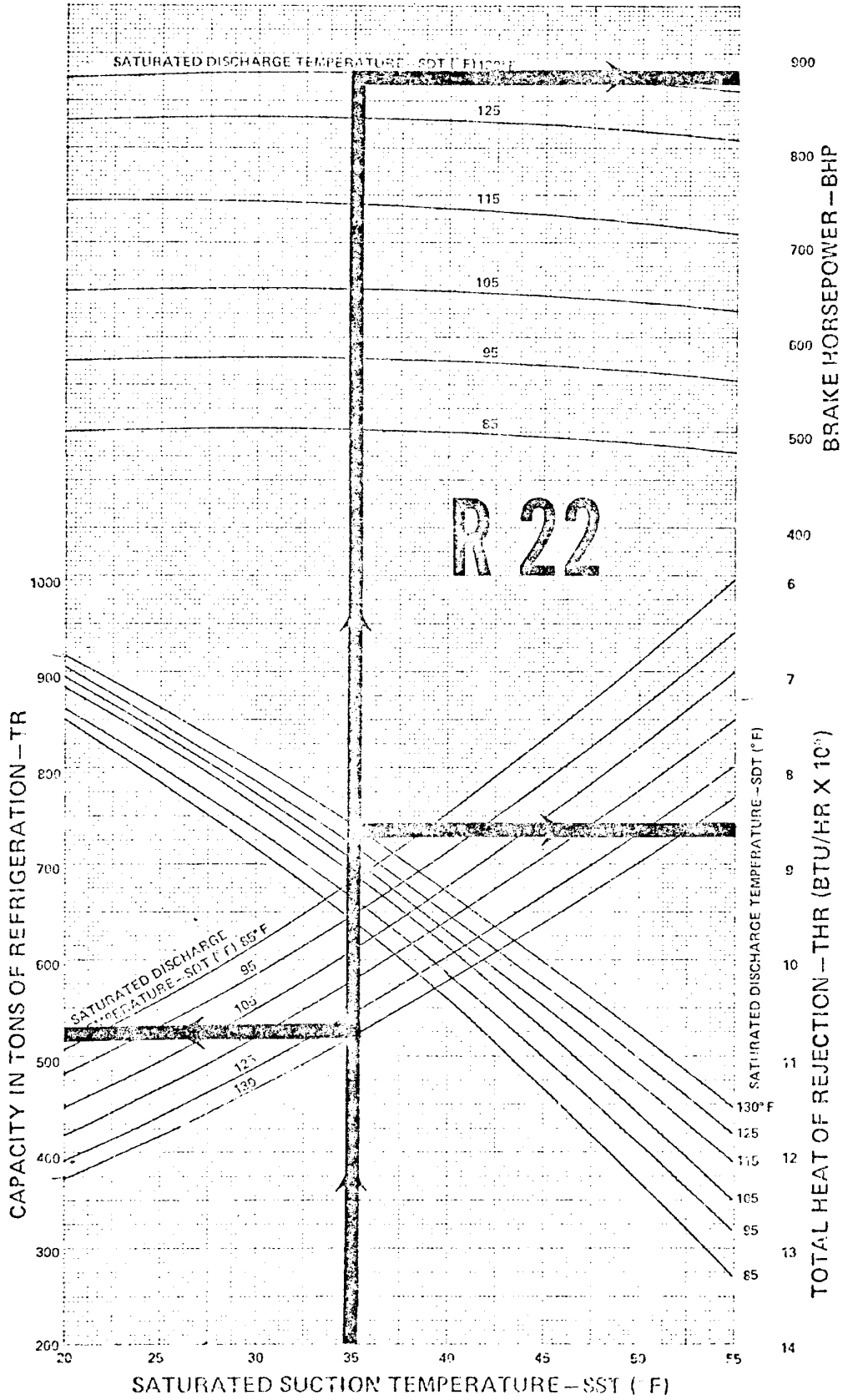
# COMPRESSOR PERFORMANCE - WINTER MODE



# COMPRESSOR PERFORMANCE - SUMMER MODE

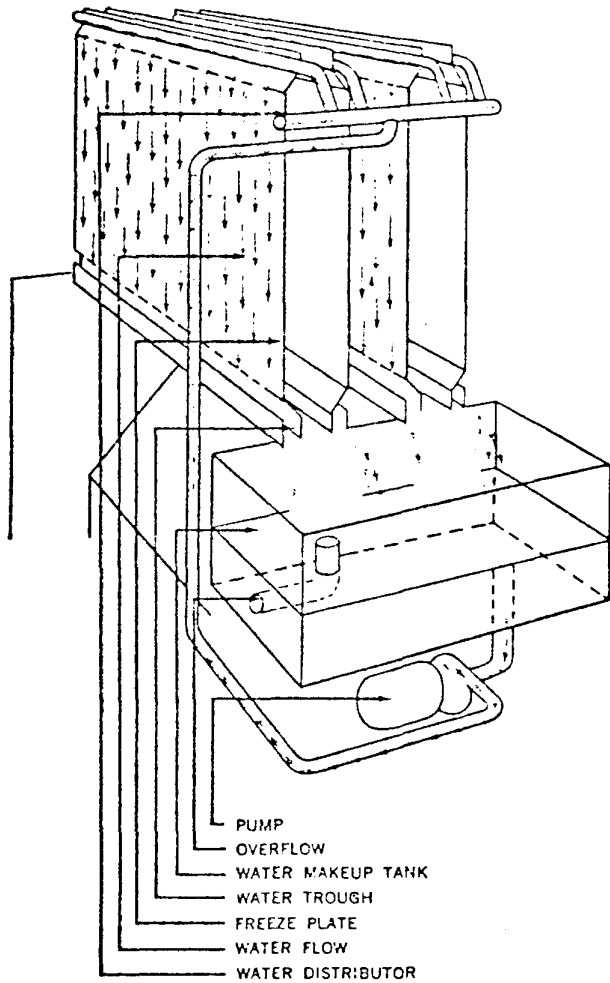


COMPRESSOR PERFORMANE FOR DOMESTIC HOT WATER  
GENERATION AT NIGHT IN SUMMER TIME



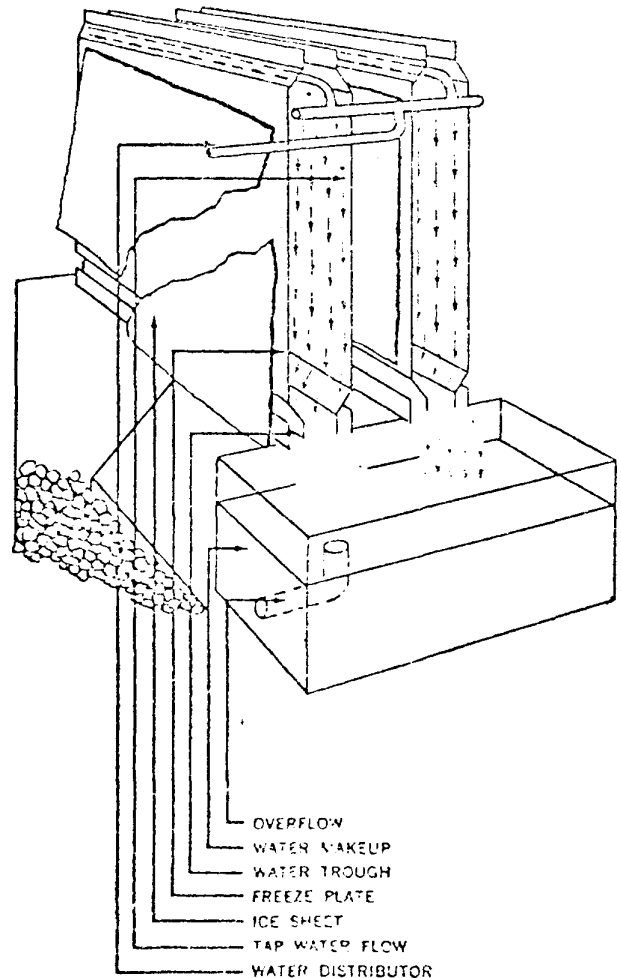
# TURBO PLATE ICE MAKERS DELIVER SELECTION OF MACHINES IN THE WORLD.

## ICE FREEZING CYCLE



Water is sprayed on freezing plates and flows by gravity over plates in freezing cycle.

## ICE HARVESTING CYCLE



For harvesting, fresh water is sprayed on back of plates, releasing ice which falls by gravity into a breaker bin where the ice is broken into pieces of the selected size. Exclusive ice drying cycle is built in.

#### 14.6 SENSITIVITY OF THE ICE GENERATION SYSTEM TO CLIMATE

The quantity of ice that a heat pump can generate on heating a building varies with the climate. As one moves north where the winters are long and severe more ice could be generated during the winter than is necessary to meet the cooling requirements of the relatively short summers. In such cases only enough ice is generated to satisfy the cooling capacity required by the summer while for the remainder of heating requirements other heat sources for the heat pumps are used.

The foremost of these is solar energy which could be used as an indirect or direct heat source for the heat pump. In the first case more ice is generated than required and subsequently melted with solar heat while in the second case solar heated water in a storage tank is circulated directly through the chiller thus serving as a heat source for the heat pump.

There are also other heat sources that could supplement the ice generated during long winter months like internal cooling load of the building, outside air above 40°F utilizing an outdoor evaporator - evaporative condenser, or even well water.

On the other hand, when one moves south where the winters are mild and short with longer, hot summers prevailing, the opposite occurs, namely the ice generated during the heating season by space heating and domestic hot water generation is insufficient to cover the summer cooling requirements.

Also the generation of domestic hot water in summer time in off peak periods is insufficient to make up for the deficiency in cooling requirements. In such cases chilled water is generated in off peak periods with heat rejection to cooling towers. In summertime, whenever possible, chilled water should be generated rather than ice, this preference stemming from the fact that chilled water generation permits the compressor to operate at higher saturation suction temperatures which results in a higher COP.

However, here too, there are several ways to supplement the deficiency in cooling requirements. Besides generating chilled water in off peak periods for diurnal storage or even for direct use, cold winter air could be used to generate additional ice for summer use, or evaporative cooling could be used to supplement cooling in summer time.

In certain geographical latitudes, however the ice generated in winter time covers exactly or nearly exactly the cooling requirements in summertime. We refer to such a location as a "balance point."

North of the balance point the size of the ice bin is governed by the summer cooling requirements and solar panels usually control the ice generation. South of the balance point the size of the ice bin is determined by the ability of heat pump winter operation to produce ice. Compressors usually supplement the deficiency in cooling requirements in summer time, while in winter time solar collectors regulate the ice generation. These points are brought out by the attached chart (page 14-20). (From Annual Cycle Energy System--by H. C. Fisher, Consultant Energy Division ORNL, Oak Ridge, Tenn.).

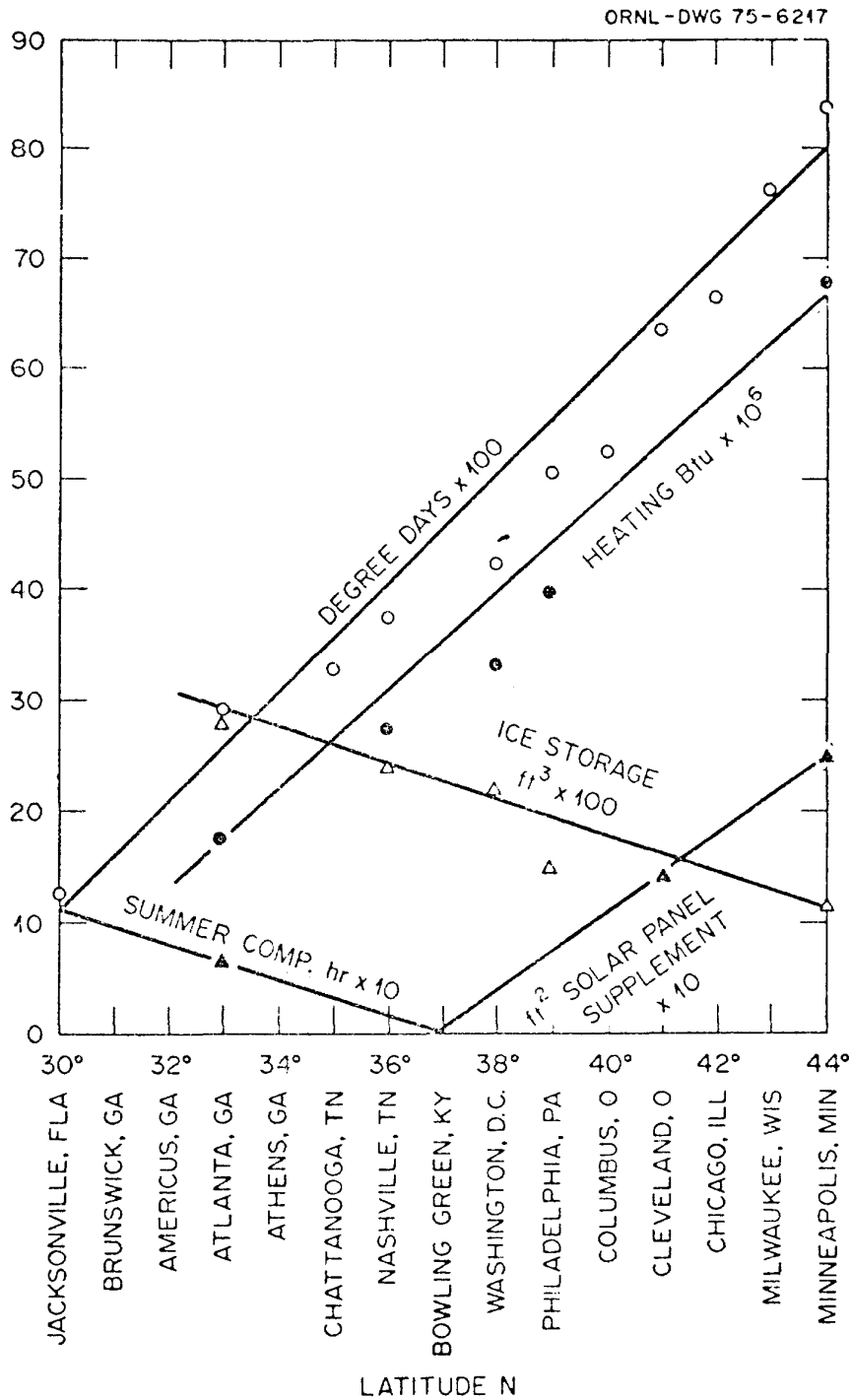


Fig.14.6 Effect of latitude on heating load ice storage, solar panel, and supplemental summer compressor operation for ACES equipped house.

15. EVALUATION OF THE ABILITY OF OUTSIDE AIR IN WINTER  
TO PRODUCE SUPPLEMENTARY ICE

15-1 Energy Analysis

The system will utilize the ability of cold air at winter time to generate ice and make up the deficiency in "cooling" energy required, which energy could not be generated by the heat pump system.

The deficiency in cooling was found to be 885,100 Ton-Hours. The number of degree-hours below and including 25°F and a base of 32°F. is 6727 and the number of hours of occurrence is 728.

Writing the energy relation

$$885,100 \times 12,000 = \text{C.F.M.} \times 1.08 \times 6727$$

$$\text{CFM} = \frac{885,100 \times 12,000}{1.08 \times 6727} = 1,462,000 \text{ say } 1,500,000 \text{ CFM}$$

This happens during 728 hours--hence energy input is:

$$\text{KWH} = \frac{1500,000 \times 0.25'' \text{ WG} \times 0.746 \times 728}{6360 \times 0.6} = 73 \text{ KW} \times 728 = 53,370 \text{ KWH}$$

The brine used would be 20% methanol and 80% water having a density of 60.4 lb/cubic foot (specific gravity  $\frac{60.4}{62.4} = 0.968$ );

specific heat 0.97 BTU/lb-°F. and freezing point of 4.5°F. With a  $\Delta T = 10^\circ\text{F}$  the energy relationship would be:

$$885,100 \times 12,000 = \text{gpm} \times 10 \times 8.33 \times 0.968 \times 60 \times 0.97 \times 728$$

$$\text{yielding gpm} = 3110$$

with an assumed head of 100 feet, the pumping energy would be:

$$\text{KWH} = \frac{3110 \times 8.33 \times 0.968 \times 100 \times 0.746 \times 728}{33,000 \times 0.75 \times 0.85} =$$

$$= 89 \text{ KW} \times 728 = 65,000 \text{ KWH}$$

Total energy input = fan energy + pump energy = 53,370 KWH + 65,000 KWH  
= 118,370 KWH

Winter heat pump energy input (see 14.1.8) = 3,250,000 KWH

	Add =	118,370 KWH
Subtotal for winter		<u>3,368,370 KWH</u>
Summer heat pump energy input (DHW) =		367,800 KWH
Total annual energy consumption =		<u>3,736,170 KWH</u>

## 15.2 Cost Analysis

### 15.2.1 Energy Cost

Winter

Heat Pump (see 14.2.1) = \$ 113,253

Summer

Fans and Pumps

energy 118,370 KWH x \$0.02158 =	\$2,554
demand (73+89) KW x \$3.17x7 Mo =	<u>\$3,572</u>
Subtotal=	\$6,126

Heat Pump

energy 367,800 KWH x \$0.02261=	\$ 8,315
demand 650x0.746 KW x \$2.4/KWH 1.5mo	<u>1,746</u>
Subtotal=	\$10,061

Total for year = \$129,380

### 15.2.2 Equipment First Cost

Heat Pump (see 14.2.2)=	=	\$ 2,600,000
Additional ice bin storage	=	750,000
42 Axial fans	=	126,000
1 pump	=	3,000
outside air coil (\$0.18 per cfm)	=	<u>270,000</u>
Total	=	\$ 3,752,000

### 15.2.3 Comparison With Conventional System

Energy savings related to conventional system  
= \$331,710 - \$129,380 = \$202,330

Incremental investment cost = \$ 3,752,000 - \$750,000  
= \$ 3,002,000

Simple "payback" = 15 years

Discounted "payback" = 12 years

#### 15-3 General Evaluation

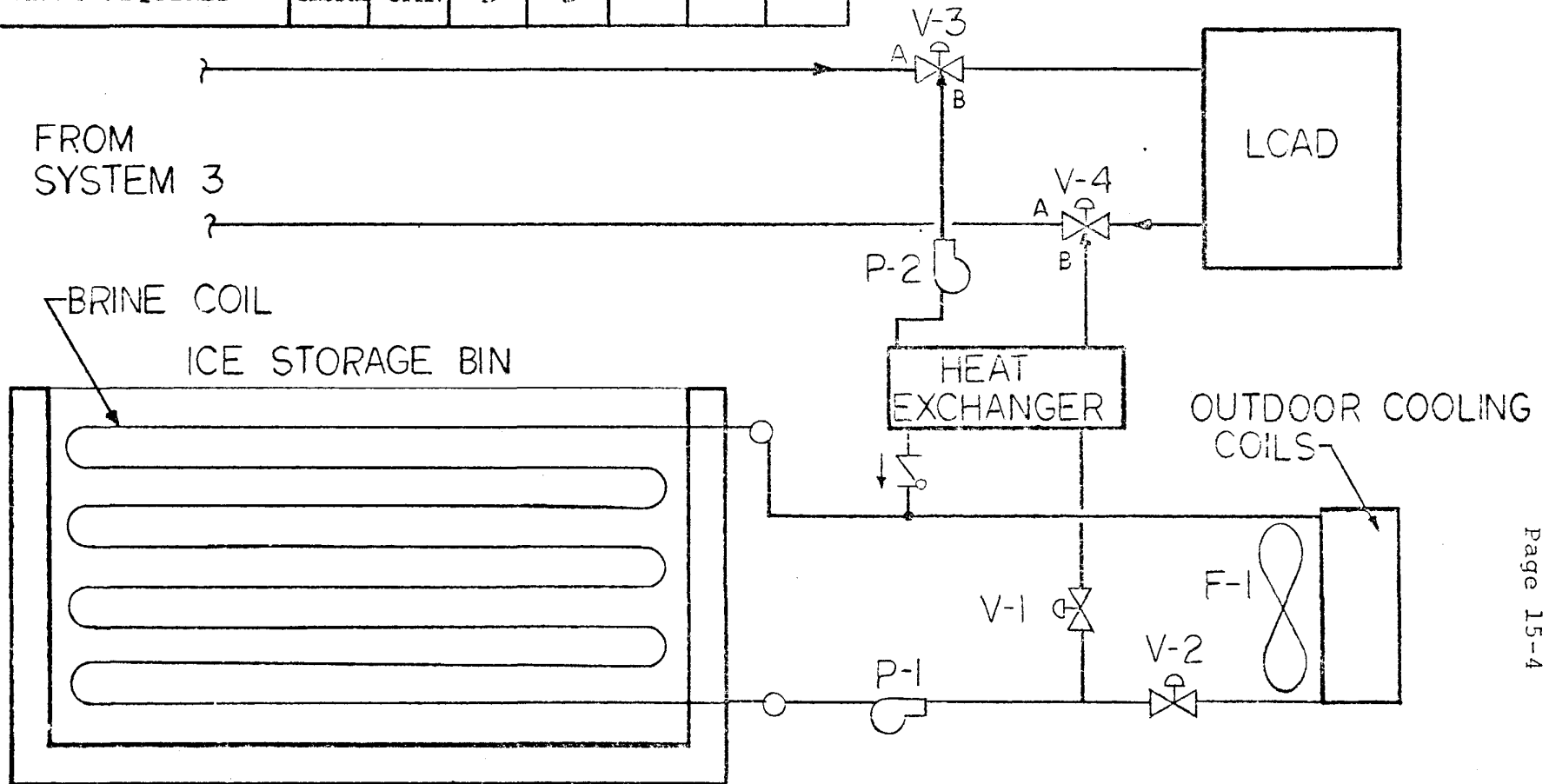
The system cuts the electrical demand by 3000 KW.

The limitless availability of low temperature air makes the system suitable for climates where the ice generated in winter time is not sufficient to cover the summer cooling requirements but there are sufficient degree hours below 25°F to generate the deficiency in ice.

# ICE STORAGE USING OUTSIDE AIR

## MODES OF OPERATION

MODE DESCRIPTION	V-1	V-2	V-3	V-4	P-1	P-2	F-1
WINTER MODE OUTSIDE TEMP BELOW 25°F	OPEN	CLOSED	OFF	OFF	OFF	OFF	ON
SUMMER MODE COOLING REQUIRED	CLOSED	OPEN	OPEN B	OPEN B	ON	ON	



5/31/78

16. COMPARATIVE ANALYSIS OF FOUR  
CANDIDATE SYSTEMS

16. COMPARITIVE ANALYSIS OF FOUR CANDIDATE SYSTEMS

	BASIC Conventional System Boilers & Chillers	1 Heat Pump System with Slab Storage	2 Solar Energy Annual Storage	3 Heat Pump for Generation of Ice. ACES	Outside Air Potential for Ice Making in Addition to System 3.	
Annual Heating Energy DHW	$4.032 \times 10^{10}$ BTU	$4.032 \times 10^{10}$ BTU	$4.032 \times 10^{10}$ BTU	$4.032 \times 10^{10}$ BTU	$4.032 \times 10^{10}$ BTU	
Requirement Cooling	$3.0 \times 10^6$ TON H	$3.0 \times 10^6$ TON H	$3.0 \times 10^6$ TON H	$3.0 \times 10^6$ TON H	$3.0 \times 10^6$ TON H	
Energy Stored	0	$7.5 \times 10^8$ BTU	$1.684 \times 10^{10}$ BTU	2,114,000 TON H	$3.0 \times 10^6$ TON H	
Energy Saved	0	$7.5 \times 10^8$ BTU	$4.032 \times 10^{10}$ BTU	$2.5379 \times 10^{10}$ BTU	$3.6 \times 10^{10}$ BTU	
Energy Consumption	W	$17 \times 10^6$ KWH-T	$3.447 \times 10^6$ KWH	106,800 KWH	3,450,000 KWH	3,368,320 KWH
	S	$3 \times 10^6$ KWH	$3.013 \times 10^6$ KWH	3,105,200 KWH	1,252,870 KWH	367,770 KWH
	Total	$20 \times 10^6$ KWH	$6.462 \times 10^6$ KWH	3,212,000 KWH	4,502,870 KWH	3,736,140 KWH
Demand Reduction	0	0	+82 KWH-6 months	-3,000 KWH-4.5 mo.	-3,000 KWH-6 mo.	
Operation Annual Energy Cost	\$ 331,710	\$ 264,470	\$ 156,000	\$ 153,392	\$ 129,380	
Operation Annual Saving vs Conventional	0	\$ 67,241	\$ 175,710	\$ 178,318	\$ 202,330	
System Capital Cost	\$750,000	\$3,400,000	\$9,800,000	\$2,600,000	\$3,752,000	
Incremental Investment vs Conventional	0	\$2,650,000	\$9,050,000	\$1,850,000	\$3,002,000	
Absolute Present Value of The System	\$ - 10,684,269	\$ - 11,034,207	\$ - 14,321,807	\$ - 7,011,518	\$ - 7,361,254	
Present Value of The System Relative to The Conventional System.	\$ 0	\$ - 349,938	\$ - 3,637,538	\$ 3,672,751	\$ 3,323,015	

PAYBACK PERIOD

SYSTEM NO.	SIMPLE PAYBACK	DISCOUNTED PAYBACK
1	40	25
2	52	29
3	11	9
4	15	12

17. EVALUATION OF OPPORTUNITIES  
AND CONSTRAINTS OF CANDIDATE  
SYSTEMS

Evaluation of the Opportunities  
for 4 Candidate Seasonal Energy Storage Systems.

OPPORTUNITIES	Basic Conventional System Boilers & Chillers	Heat Pump System with Slab Storage	Solar Energy Annual Storage	Heat Pump Generates Ice Storage ACES	Outside Air Potential for Ice Making in Addition to System 3
1. Reduction in peak electrical demand	0	0	0	100%	100%
2. Reduction in electrical energy consumption	0	68%	84%	77.5%	82%
3. When both heating and cooling are required the economics are greatly enhanced	NOT AP	AP	NOT AP	AP	AP
4. The availability of equipment	AP	AP	AP	AP	AP
5. High energy density per unit volume of storage	NOT AP	little AP	well AP	well AP	well AP
6. Flexibility of the system	AP	limited	limited	well AP	AP
7. Suitability of the system to operate in various conditions.	AP	limited	limited	well AP	AP
8. Large storage volume causes lower relative heat losses	NOT AP	NOT AP	well AP	well AP	well AP
9. The larger the storage volume the lower the cost per unit volume	NOT AP	NOT AP	well AP	well AP	well AP

Evaluation of the opportunities  
for 4 Candidate Seasonal Energy Storage Systems.

(Continued)

-2-

OPPORTUNITIES	Basic Conventional System Boilers & Chillers	Heat Pump System with Slab Storage	Solar Energy Annual Storage	Heat Pump Generates Ice Storage ACES	Outside Air Potential for Ice Making in Addition to System 3
10. Multiple compressor application at part load	AP	AP	AP	AP	AP
11. Large storage volume allows reduction in size of equipment	NOT AP	NOT AP	NOT AP	well AP	well AP
12. Air Pollution	Yes	No	No	No	No
13. Additional space required	No	No	Yes	Yes	Yes
14. The limitless availability of low temperature air	NOT AP	NOT AP	NOT AP	NOT AP	well AP
15. Smallest collector areas could be installed than would be necessary to meet energy demands during periods of low isolation and high demand	NOT AP	NOT AP	AP	NOT AP	NOT AP

EVACUATION OF THE CONSTRAINTS

for 4 Candidate Seasonal Energy Storage Systems

CONSTRAINTS	Basic Conventional System Boilers & Chillers	1 Heat Pump System with Slab Storage	2 Solar Energy Annual Storage	3 Heat Pump for Ice Generation with ACES Storage	4 Outside Air Potential for Ice Making in Addition to System 3
1. No larger system as yet in operation	Not AP	AP	AP	AP	AP
2. Equipment not specifically resigned for this- concept.	Not AP	AP	Not AP	AP	AP
3. Lower COP at low Saturated temperature for ice making	Not AP	Not AP	Not AP	AP	AP
4. Heating and Cooling loads profiles are not always compatible	Not AP	Not AP	Not AP	AP	AP
5. Large Storage Area required	Not AP	Not AP	AP	AP	AP
6. High first cost	Not AP	AP	AP	AP	AP
7. Future conventional electricity and fuel cost not predictable hence the economic viability is diffic- ult to qualify	Not AP	AP	AP	AP	AP

## for 4 Candidate Seasonal Energy Storage Systems

CONSTRAINTS	Basic Conventional System Boilers & Chillers	1 Heat Pump System with Slab Storage	2 Solar Energy Annual Storage	3 Heat Pump for Ice Generation with ACES Storage	4 Outside Air Potential for Ice Making in Addition to System 3
8. Architects and Engineers are not familiar with principles and technical personnel for analysis and design are in short supply.	NOT AP	well AP	AP	well AP	well AP
9. Existing buildings and systems are difficult and costly to retrofit for seasonal storage; space may be difficult to find.	NOT AP	AP	AP	AP	AP
10. Many building trades and disciplines are involved.	NOT AP	AP	AP	AP	AP
11. Seasonal Storage not mutually compatible with some other energy system (T.E.)	NOT AP	AP	AP	AP	AP
12. Difficult to estimate cost before complete design is done; difficult to obtain a real economic analysis.	NOT AP	AP	AP	AP	AP
13. Long term pay-back.	0	25 years	29 years	9 years	12 years
14. Performance unpredictability of the system.	NOT AP	AP	AP	NOT AP	AP

## for 4 Candidate Seasonal Energy Storage Systems

CONSTRAINTS	Basic Conventional System Boilers & Chillers	1 Heat Pump System with Slab Storage	2 Solar Energy Annual Storage	3 Heat Pump for Ice Generation with ACES Storage	4 Outside Air Potential for Ice Making in Addition to System 3
15. Large space required for 2 outside air coils (or collectors)	NOT AP	NOT AP	AP	NOT AP	AP
16. Back up system required.	NOT AP	NOT AP	AP	NOT AP	NOT AP
17. Low capacity Heat Storage in air conditioning temperature operation range.	NOT AP	AP	NOT AP	NOT AP	NOT AP
18. High heat loss brought about by the high storage surface.	NOT AP	AP	NOT AP	NOT AP	NOT AP
19. Lower COP during summer influenced by high condensing temperatures.	NOT AP	AP	NOT AP	NOT AP	NOT AP
20. May influence the building structure with high temperature fluctuations.	NOT AP	AP	NOT AP	NOT AP	NOT AP
21. The adjustment of the system to a given location is sensitive to prevailing climatic	NOT AP	NOT AP	AP	AP	AP
22. Difficulty in analysis the heat transfer problems in the slab.	NOT AP	AP	NOT AP	NOT AP	NOT AP
23. Can satisfy only the heating requirements.	NOT AP	NOT AP	AP	NOT AP	NOT AP

18. MAINTENANCE AND LABOR EVALUATION FOR EACH SYSTEM

Due attention was given to the maintenance and labor required by each of the four candidate systems and for each system a corresponding estimate was made.

A. Conventional System

1. The system contains boilers and chillers and therefore is required by code to employ two stationary engineers for operating the equipment. For a 24-hour operation the number of men should be tripled.
2. The refrigeration compressors are operating during summer months only, which is about one half of the operating hours required for the heat pump system and therefore relatively little maintenance is required.
3. However this will be offset by the increased maintenance and labor cost required by the boilers of the conventional system.

B. Heat Pump with Slab Building Storage

1. Due to the fact that the heat pump system comprises both heating and cooling, only one stationary engineer is required to maintain the system. However three shifts will be required both summer and winter.
2. It can be expected that the compressors of the heat pump system would operate more hours, winter and summer and therefore more hours would be required for preventive, progressive and breakdown maintenance.
3. Due to the fact that in the slab storage system there are some subsystems performing the same duty (slab storage condenser and evaporative condenser, chiller and outdoor evaporator, simultaneous cooling and heating whereby the double bundle condenser supplies the heating required), each of the subsystems would work less and thus require less maintenance.
4. However it is difficult to determine the actual maintenance cost of the system due to the fact that the compressor would work at a variable discharge temperature for long periods of time and it could be expected that more preventive maintenance would be required.

C. Solar Energy System

1. The solar system is expected to require minimum preventive and progressive maintenance due to the fact that few moving parts are involved in such a system.
2. The collector array will require a minimum maintenance depending on location, mainly cleaning and from time to time replacing the broken down collectors.
3. The solar collector does not require any permanent attendance. The same stationary engineers operating the refrigeration plant could look after the solar collectors.
4. The cooling system would operate only during summer months and therefore require less maintenance.
5. The storage tank would require some preventive maintenance mainly cleaning about once in five years.

D. Heat Pump with Ice Maker and Annual Storage

1. Due to the fact that only one type of system is involved, one stationary engineer would be required. However the system requires three shift operation.
2. In winter time the compressors would operate between extreme conditions and it is expected therefore that preventive and breakdown maintenance might be necessary.
3. In summer time, on the other hand, there is little need to operate the compressors due to the fact that the winter operation of the heat pump produced enough ice to cover most of the summer cooling requirements. To supplement the deficiency in cooling the compressors would operate at off-peak periods to charge the chilled water storage tanks and mainly at night, when the temperatures are cool and thus lower condensing temperatures would result in good compressor performance and little maintenance required.
4. The control system of the ice generation heat pump is rather sophisticated and could prevent maintenance and breakdown problems.
5. The system is new and operating engineers may not

be familiar with its operation which fact might cause some additional maintenance problems.

6. It is difficult to determine the maintenance expenses of such a system since there is no such system in operation as yet.

E. Supplementary Ice Generated by Outside Cold Air.

1. The heat pump system would operate mainly in winter time and the compressors would therefore have less operating hours although they would work between extreme temperatures.
2. There is only one type of system and therefore it would require only one stationary engineer. Three shift operation would be required.
3. The supplementary ice generating system with an outdoor brine coil, ice bin coil, pumps and heat exchangers would require additional maintenance, especially due to corrosion problems caused by the brine system.
4. It is difficult to determine the maintenance cost required by such a system since it has never been tried.

Summarizing, it can be said that each system has its advantages and disadvantages with regard to maintenance, so that the maintenance and labor cost are roughly the same and would not affect the balance between the systems as obtained by considering the energy and first cost alone. For example, the solar system has a relatively low maintenance cost but this cannot offset its large first cost.

Similarly the ice generating heat pump remains the favored system with a predictable high maintenance in winter time, which is offset by the predictable low maintenance in summer time.

The supplementary ice generation system using cold outside air and the slab storage system appear to have a high maintenance cost and this confirms only the previously deduced conclusion of the nonfeasibility of these systems. It is felt therefore that the conclusions drawn before are not affected by the maintenance considerations and therefore no maintenance and labor cost were considered at this stage. However, in the proposed feasibility study, if performed, a detailed maintenance and labor costs analysis will have to be made.

## 19. METHOD OF CALCULATION

The peak cooling load was calculated by hand for the residential areas taking into account the transmission, solar and internal loads. For other areas the cooling loads were calculated on the basis of square foot per ton considering the use of each area.

Similarly, the heating loads were calculated by hand for the residential areas, while for other areas the method of BTU per hour per square foot was applied.

Annual energy requirements were calculated by using the Degree-Day method for heating and the assumed Equivalent Full Load Hours for cooling. The energy consumptions of the equipment were calculated by utilizing the performance data supplied by the manufacturers of the equipment.

The solar system design was based on weather data for Washington, D.C. supplied by HUD and on the solar collector performance supplied by the manufacturers.

20. CALCULATIONS

PL29-73

PLANN. AVE DEVELOPMENT PROJECT  
SLAB STORAGE

1  
JAN. 65-18

HEAT PUMP WITH SLAB STORAGE

1. OPERATION ENERGY COST

SUMMER

ENERGY CHARGE  $3.013 \times 10^6 \text{ KWH} \times \$0.0226 = \$ 68,140$

DEMAND CHARGE  $3000 \text{ KW} \times \$5.25/\text{KW} \times 5 \text{ Mo} = \$ 78,750$

TOTAL = \$ 146,890

WINTER

ENERGY CHARGE  $3.447 \times 10^6 \text{ KWH} \times \$0.02158 = \$ 74,400$

DEMAND CHARGE  $3 \times 653 \text{ KW} \times \$3.15/\text{KW} \times 7 = \$ 43,179$

TOTAL = 117,579

ANNUAL ENERGY CONSUMPTION = \$ 264,470

THE SYSTEM WILL OPERATE WITH DISCHARGE TEMPERATURES VARYING BETWEEN 105°F AND 125°F; THE POWER INPUT WILL FLUCTUATE BETWEEN 750 HP AND 900 HP.

2. SYSTEM DESCRIPTION

FOUR SCREW COMPRESSORS WITH 900 HP MOTORS EACH

TOTAL CONDENSER WATER FLOW IS

$$4 \times 750 \text{ TON} \times 2 \text{ GPM/TON} = 9000 \text{ GPM}$$

3. SLAB PIPING NETWORK

TOTAL BELOW GRADE AREA = 920,000 SQ FT

DIVIDED BETWEEN 3 FLOORS, YIELDS 306,000 SQ FT/FLOOR

DEPTH OF UNDERGROUND STRUCTURE 30 FT

ASSUME A SQUARE FLOOR, DIMENSION =  $\sqrt{306,000} = 552 \text{ FT}$

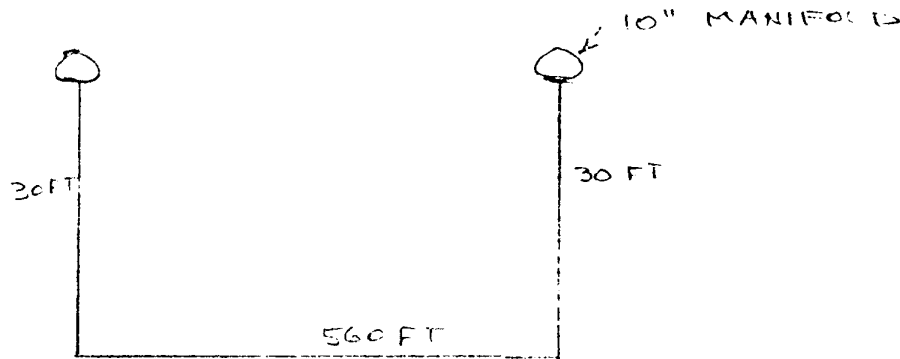
ASSUME PIPES SPACED 1 FT APART

THUS THERE ARE 552 PIPES FOR FOUR COMPRESSORS  
OR 138 PIPES PER COMPRESSOR

R24-77  
PTM. AYE DEVELOPMENT PROJECT  
SLAB STORAGE

AH

Z.  
6.6.78



10" MANIFOLD FOR EACH COMPRESSOR SYSTEM  
AND 138 2"  $\phi$  PIPES

$$\begin{aligned} \text{TOTAL LENGTH OF 2" PIPES} &= (560 + 30 + 30) \times 138 \\ &= 342,240 \text{ LINEAR FEET} \end{aligned}$$

$$\text{TOTAL LENGTH OF 10" PIPES} = 138 \times 8 = 1104, \text{ SAY 1500 L.F.}$$

3. SYSTEM FIRST COST

a.	2" PIPES	$342,240 \times \$7.5/\text{FT}$	$= \$2,568,750$
b.	10" PIPES	$1500 \times \$52/\text{FT}$	$= \$78,000$
c.	FITTINGS AND VALVES		$= \$22,500$
		SUBTOTAL	$= \$2,669,000$

d.	chiller Package	$= \$600,000$
e.	COOLING TOWERS	$= \$100,000$
f.	CONTROLS	$= \$31,000$
		SUBTOTAL $= \$731,000$

$$\text{TOTAL} = \$3,400,000$$

SOLAR SYSTEM CALCULATIONS

	1	2	3	4	5		6
Month	Horizontal Insolation $\bar{I}_H$ Btu/(Day·ft <sup>2</sup> )	Extra-terrestrial Insolation $\bar{I}_0$ Btu/(Day·ft <sup>2</sup> )	Ratio Horizontal to Extra-terrestrial $\bar{K}_t$	Ratio Horizontal to Tilt $\bar{R}$	Monthly Avg. Daily Rad. on Tilt Surf. $\bar{I}_T$ Btu/(Day·ft <sup>2</sup> )	No. of Days in Month N	Tot. Monthly Radiation on Tilt Surf. S Btu/(Mo·ft <sup>2</sup> )
Jan.	639.4	1377	0.445	1.807	1139.6	31	35327.1
Feb.	901.5	1836	0.470	1.427	1340.5	28	37535
March	1255	2478	0.496	1.19	1478.4	31	45821
April	1600.4	3040	0.504	0.94	1504.4	30	45131
May	1846.8	3450	0.516	0.79	1459	31	45228
June	2050.2	3620	0.53	0.725	1529.4	30	45881
July	1999.9	3645	0.524	0.759	1471.5	31	44930
Aug.	1717.1	3102	0.516	0.877	1430.2	31	44172
Sept.	1446.1	2727	0.512	1.09	1322.6	30	42280
Oct.	1033.4	2002	0.506	1.24	1172	31	47019
Nov.	672.5	1420	0.464	1.73	1274	30	38907
Dec.	594.1	1313	0.445	1.93	1153	31	35913

A-119

- From Table A-4 or Fig. A-29 section 5, or known data.
- From Table A-5 section 5, used only for eq. 11.
- From Table A-4 section 5, or eq. 11.
- From Table A-6 section 5, latitude ( $\phi$ ) = 39°, with collector tilt ( $\theta$ ) = 54°, and latitude - tilt = -15°.
- From eq. 12,  $\bar{I}_T = (\bar{I}_H)(\bar{R})/50000$
- From eq. 13,  $S = (\bar{I}_T)(N)$ .

811-V

Month	1 Monthly Degree Days DD °F-days	2 Monthly Space Htg Load Q <sub>s</sub> Btu/Mo.	No. of Days/ Mo. N	3 Vol. of DHW Used/Mo. Gal./Mo.	Temp. Water Main Sup. t <sub>m</sub> °F	4 DHW Temp. Rise t <sub>s</sub> - t <sub>m</sub> °F	5 Monthly DHW Load Q <sub>w</sub> Btu/Mo.	6 Total Heating Load L Btu/Mo.
Jan.	271	1500	31	52,000	55	65	2589	7366
Feb.	769	1500	28	52,000	55	65	7775	6470
March	696	1500	31	52,000	55	65	8534	6754
April	388	2150	30	52,000	55	65	3316	2920
May	74	500	31	52,000	55	65	2589	1900
June	0	0	30	52,000	55	65	3316	2316
July	0	0	31	52,000	55	65	3524	2316
Aug.	0	0	31	52,000	55	65	3524	2316
Sept.	32	947	30	52,000	55	65	2316	1700
Oct.	617	1021	31	52,000	55	65	3524	2470
Nov.	510	3278	30	52,000	55	65	3524	4709
Dec.	825	5431	31	52,000	55	65	3524	7030

$q_d = \frac{311,320}{\text{Btu/h/ft}^2}$   
 (Given data or calculate from Manual J or equivalent.)

$\Delta t_d = 70 - t_w$   
 $= 70 - \text{_____} = \text{_____}$   
 Where:  $t_w = 97 \frac{1}{2}\%$  winter design temperature  
 (From ASHRAE Fundamentals, Table A-3, section 5 or known weather data.)  
 $70^\circ =$  indoor design temperature

$UA = \frac{q_d}{\Delta t_d} = \text{_____}$

$t_s = \text{_____}$

- From ASHRAE Systems, Climatic Atlas or Table A-3, section 5.
- $Q_s = (PF)(24)(UA)(\text{Degree Day}) = \text{_____} \times (\text{Degree Day}) = \text{_____}$   
 Where: PF = 0.75 or more appropriate value.
- $(\text{Vol/day})(\text{no. days/mo.}) = \text{_____} (\text{gal./day})(\text{no. days/mo.})$
- May be constant or may vary.
- $Q_w = (\text{vol. of water}) \times 8.33 \times 1 \times (t_s - t_m)$ .
- $L = Q_s + Q_w$

P Heat load 16,500,000 Btu/yr

SUNSYM COLLECTOR PERFORMANCE 1977 DEC 19

TI=MONTHLY AVE. SOLAR RADIATION FRACTION REACHING EART  
 TA=MONTHLY AVE. DAYTIME DRY BULB TEMPERATURE (F)  
 TI=CONSTANT INLET TEMPERATURE (F) WAZ=HALL AZIMUT

COVER N<sub>h</sub>K = 1.520 .125 .035

30 1 5 .04 .02 .310 .0 .095 0 0 1 0 0 1 1  
 36. 27.0 1736. .315 .375 4.28 .01 224. 19.50 1.

ABS---.940 .940 .940 .935 .927 .914 .886 .825 .691 .00  
 TRANS-.914 .914 .914 .912 .905 .897 .839 .721 .453 .00

WASHINGTON, D.C. (WECO)														LAT=38.9		A=.940		TILT=43.9		WAZ= 0.		ANNUAL TOTAL		AVE MONTHLY							
BTU/SF/DAY														IT= 5		1.00GPM		FL=36		1-COVER		E=.120		WIND= 5.0		R= 10.0		BTU/SF/YR		BTU/SF/MO	
TI	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC																			
KT	.447	.475	.508	.525	.546	.591	.560	.547	.543	.520	.470	.462																			
TA	38.	40.	48.	58.	68.	76.	80.	78.	72.	61.	50.	40.																			
Q! 100!	440	564	695	730	767	871	854	881	888	740	507	418!	254317	21193																	
Q! 110!	400	516	641	670	705	805	789	817	824	683	463	379!	234142	19511																	
Q! 120!	360	470	588	613	644	740	725	753	762	630	421	341!	214511	17075																	
Q! 130!	322	425	536	558	585	676	662	690	702	578	379	303!	195306	16275																	
Q! 140!	288	381	486	505	529	615	602	632	644	527	339	270!	177112	14759																	
Q! 150!	256	341	437	453	475	556	544	574	589	478	302	239!	159441	13303																	
Q! 160!	225	303	391	404	422	499	488	519	535	429	268	209!	142837	11903																	

WASHINGTON, D.C. (WECO)														LAT=38.9		A=.940		TILT=53.2		WAZ= 0.		ANNUAL TOTAL		AVE MONTHLY							
BTU/SF/DAY														IT= 5		1.00GPM		FL=36		1-COVER		E=.120		WIND= 5.0		R= 10.0		BTU/SF/YR		BTU/SF/MO	
TI	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC																			
KT	.447	.475	.508	.525	.546	.591	.560	.547	.543	.520	.470	.462																			
TA	38.	40.	48.	58.	68.	76.	80.	78.	72.	61.	50.	40.																			
Q! 100!	489	579	656	634	630	703	706	774	841	755	556	477!	237329	19777																	
Q! 110!	448	531	603	577	572	642	645	712	778	698	511	437!	217673	18139																	
Q! 120!	408	485	551	523	516	583	586	651	718	644	467	397!	198653	16554																	
Q! 130!	368	439	500	471	462	525	529	593	659	593	425	358!	180185	15015																	
Q! 140!	331	394	451	421	411	470	473	537	603	542	383	322!	162419	13534																	
Q! 150!	297	354	403	373	361	417	421	483	549	492	344	289!	145528	12127																	
Q! 160!	265	316	359	327	314	366	370	431	496	443	308	257!	129367	10780																	

COVER N,H,K = 1.520 .125 .035  
 30 1 5 .04 .02 .310 .0 .095 0 0 1 0 0 1 1 ABS---.940 .940 .940 .935 .927 .914 .886 .826 .691 .000  
 36. 27.0 1736. .315 .375 4.28 .01 224. 18.50 1. TRANS-.914 .914 .914 .912 .905 .887 .839 .721 .453 .000

!WASHINGTON, D.C. (WECO)														!ANNUAL TOTAL!		!AVE MONTHLY!	
!BTU/SF/DAY IT= 5 1.00GPM FL=36 1-COVER														!BTU/SF/YR		!BTU/SF/MO	
! TI !	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC					
KT!	.447!	.475!	.508!	.525!	.546!	.591!	.560!	.547!	.543!	.520!	.470!	.462!					
TA!	38.!	40.!	48.!	58.!	68.!	76.!	80.!	78.!	72.!	61.!	50.!	40.!					
Q! 100!	440	564	695	730	767	971	854	981	988	740	507	418!	254317	21193			
Q! 110!	400	516	641	670	705	905	789	917	924	683	463	379!	234142	19511			
Q! 120!	360	470	588	613	644	740	725	753	762	630	421	341!	214511	17875			
Q! 130!	322	425	536	558	585	676	662	690	702	578	379	303!	195306	16275			
Q! 140!	288	381	486	505	529	615	602	632	644	527	339	270!	177112	14759			
Q! 150!	256	341	437	453	475	556	544	574	589	478	302	239!	159641	13303			
Q! 160!	225	303	391	404	422	499	488	519	535	429	268	209!	142837	11903			

!WASHINGTON, D.C. (WECO)														!ANNUAL TOTAL!		!AVE MONTHLY!	
!BTU/SF/DAY IT= 5 1.00GPM FL=36 1-COVER														!BTU/SF/YR		!BTU/SF/MO	
! TI !	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC					
KT!	.447!	.475!	.508!	.525!	.546!	.591!	.560!	.547!	.543!	.520!	.470!	.462!					
TA!	30.!	40.!	48.!	58.!	68.!	76.!	80.!	78.!	72.!	61.!	50.!	40.!					
Q! 100!	468	576	680	686	701	789	783	831	870	753	536	451!	247235	20602			
Q! 110!	428	528	627	627	641	726	720	768	806	696	491	411!	227305	18942			
Q! 120!	388	482	574	572	582	664	657	706	745	642	448	372!	207917	17326			
Q! 130!	348	437	523	518	526	603	598	645	686	590	406	334!	189110	15759			
Q! 140!	313	392	473	467	472	544	540	588	628	540	364	299!	171041	14253			
Q! 150!	280	351	425	416	420	488	484	532	573	490	326	267!	153756	12815			
Q! 160!	248	314	379	368	370	434	431	478	520	441	291	235!	137224	11435			

PAD-CACES  
 Page 20-  
 8

!WASHINGTON, D.C. (WECO)														!ANNUAL TOTAL!		!AVE MONTHLY!	
!BTU/SF/DAY IT= 5 1.00GPM FL=36 1-COVER														!BTU/SF/YR		!BTU/SF/MO	
! TI !	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC					
KT!	.447!	.475!	.508!	.525!	.546!	.591!	.560!	.547!	.543!	.520!	.470!	.462!					
TA!	30.!	40.!	48.!	58.!	68.!	76.!	80.!	78.!	72.!	61.!	50.!	40.!					

PROJECT NO. PC 29-77  
 PROJECT NAME: PENNSYLVANIA AVENUE DEV. PROJECT OWNED BY A.H. DATE: 6.178  
 SUBJECT: DETERMINATION OF SOLAR COLLECTOR AREA

SELECTION OF OPTIMUM SOLAR COLLECTOR AREA

MONTH	TOTAL LOAD BTU x 10 <sup>6</sup>	SOLAR ENERGY COLLECTED BTU/SQ.FT.	COLLECTOR AREA REQUIRED FOR 100% PARTICIPATION	TOTAL ENERGY COLLECTED ON 200,000 SQ. FT. COLL. AREA BTU x 10 <sup>6</sup>	ENERGY COLLECTED ON 260,000 SQ. FT. COLLECTOR AREA BTU x 10 <sup>6</sup>	ENERGY TO AND FROM STORAGE BTU x 10 <sup>6</sup>
JAN.	7366	10261	717860	2052.2	2667.9	-4698.1
FEB.	6470	11032	586480	2206.4	2868.3	-3601.7
MAR.	5536	13961	395970	2796.2	3635.1	-1900.9
APR.	2983	12630	23618	2526	3283.8	+ 300.8
MAY	1411	12741	11024	2548.2	3312.7	+1901.7
JUNE	831	14100	58936	2820	3666	+2835
JULY	858	14663	58515	2932.6	3812.4	+2954.4
AUG.	858	16647	51541	3329.4	4328.2	+3470.2
SEP.	1078	18090	59591	3618	4703.4	+3625.4
OCT.	2479	16260	15246	3252	4227.6	+1748.6
NOV.	4709	11470	409630	2298	2987.4	-1721.6
DEC.	7089	9962	711080	1996.4	2595.3	-4493.7
TOTAL	41680	162419	256620	32485.	42229.	-

12-24-77  
PENNSYLVANIA AV. DEVELOPMENT PROJECT  
DETERMINATION OF STORAGE TANK VOLUME

A.H. 6-27-78

ENERGY FROM STORAGE DURING MONTHS  
NOVEMBER, DECEMBER, JANUARY, FEBRUARY  
AND MARCH =  $16,416 \times 10^6$  BTU

ENERGY INTO STORAGE DURING MONTHS  
APRIL THROUGH OCTOBER =  $16,836 \times 10^6$  BTU  
LOSSES 10% =  $-1,684 \times 10^6$  BTU  
ENERGY INTO STORAGE =  $15,152 \times 10^6$  BTU

SOLAR ENERGY USED DIRECTLY

$$= (42,229 - 16,836) \times 10^6$$

$$= 25,393 \times 10^6 \text{ BTU}$$

THE SIZE OF THE STORAGE TANK IS DETERMINED  
BY THE AMOUNT OF ENERGY STORED DURING THE  
MONTHS APRIL THROUGH OCTOBER, I.E.  $16,836 \times 10^6$  BTU  
WITH A 100°F TEMPERATURE DIFFERENCE.

THE SIZE OF THE TANK IS

$$\frac{16,836 \times 10^6}{8.33 \times 10^3} = 2.02 \times 10^7 \text{ GAL}$$

$$= 2.74 \times 10^6 \text{ CU. FT.}$$

HOURS OF SUNSHINE ARE GIVEN BY THE TABLE  
BELOW

MONTH	J	F	M	A	M	J	J	A	S	O	N	D	ANNUAL
NO. OF HOURS	138	160	205	226	267	288	291	264	233	207	162	135	2580

PROJECT # PC29-77

LOCATION: PENNSYLVANIA AVE. DEV. PROJECT  
ANNUAL ENERGY CONSUMPTION

AREA: 6,578

OPERATION ENERGY CONSUMPTION

1. FLOW RATE IN COLLECTOR LOOP

$$260,000 \text{ SQ. FT} \times 0.025 \text{ GPM/SQ FT} = 6,500 \text{ GPM}$$

2. PUMPING POWER REQUIRED

$$\frac{6500 \text{ GPM} \times 8.33 \text{ LB/GAL} \times 55 \text{ FT}}{33,000 \times 0.75} = 110 \text{ HP}$$

ENERGY CONSUMPTION

$$110 \text{ HP} \times 0.746 \text{ KW/HP} \times 2580 \text{ HR} = 211,715 \text{ KWH/YEAR}$$

3. ENERGY COST

a. SUMMER

COLLECTOR PUMP 1283 HRS. OF SUMMER OPERATION

ENERGY CHARGE  $110 \times 0.746 \times 1283 \text{ KWH} \times \$0.0261/\text{KWH} = \$2379$

DEMAND CHARGE  $110 \times 0.746 \times \$5.25/\text{KW} \times 5 \text{ Mo} = \$2152$

SUBTOTAL = \$4531

COOLING

ENERGY CHARGE  $3,000,000 \text{ KWH} \times \$0.0276 = \$82,800$

DEMAND CHARGE  $3000 \text{ KW} \times \$5.25/\text{KW} \times 5 = \$78,750$

SUBTOTAL = \$146,550

TOTAL SUMMER = \$146,550 + \$4,531 = \$151,111

b. WINTER

COLLECTOR PUMP OPERATES 1297 HOURS

ENERGY CHARGE

$82 \text{ KW} \times 12.97 \text{ HRS} \times \$0.02158 = \$2295$

DEMAND CHARGE  $82 \text{ KW} \times \$3.157 = \$2589$

TOTAL WINTER = \$4103

c. GRAND TOTAL SUMMER AND WINTER ENERGY

COST = \$4103 + \$151,111 = \$155,214

SAY \$156,000

A-108

Latitude	$\bar{I}_o$ , Monthly Average Daily Extraterrestrial Radiation Btu/Day·ft <sup>2</sup>											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
20	2349	2671	3019	3301	3421	3445	3423	3332	3106	2763	2421	2246
25	2103	2747	2891	3266	3463	3524	3485	3329	3013	2588	2192	1995
30	1851	2260	2740	3206	3482	3581	3526	3303	2877	2395	1950	1735
35	1590	2030	2570	3124	3479	3619	3546	3254	2759	2184	1698	1468
40	1324	1788	2380	3019	3454	3637	3545	3183	2600	1958	1438	1149
45	1056	1535	2172	2892	3409	3636	3525	3090	2421	1720	1174	921
50	791	1275	1948	2746	3346	3621	3489	2979	2225	1470	910	669
55	535	1011	1769	2582	3269	3596	3441	2856	2012	1212	651	422
60	299	747	1459	2403	3185	3571	3389	2709	1784	950	405	200

Table A-5 Monthly Average Daily Extraterrestrial Radiation Btu/Day·ft<sup>2</sup>,  $\bar{I}_o$

S

$\bar{R}$  for  $K_t = .30$

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
(Latitude-Tilt) = 15.0												
25	1.09	1.06	1.03	1.00	.98	.98	.98	.99	1.02	1.05	1.08	1.09
30	1.15	1.10	1.05	1.01	.98	.97	.97	.99	1.03	1.08	1.13	1.16
35	1.23	1.15	1.07	1.01	.97	.96	.96	1.00	1.05	1.12	1.20	1.25
40	1.34	1.22	1.11	1.02	.97	.95	.96	1.00	1.07	1.18	1.30	1.38
45	1.51	1.31	1.15	1.03	.97	.94	.95	1.00	1.10	1.25	1.45	1.58
50	1.77	1.44	1.21	1.05	.97	.93	.95	1.01	1.13	1.35	1.67	1.91
55	2.24	1.65	1.29	1.07	.96	.93	.94	1.02	1.18	1.50	2.04	2.53
(Latitude-Tilt) = .0												
25	1.17	1.11	1.04	.97	.93	.91	.92	.95	1.01	1.08	1.16	1.19
30	1.24	1.15	1.05	.97	.92	.90	.91	.95	1.02	1.11	1.21	1.27
35	1.33	1.20	1.08	.97	.91	.89	.90	.95	1.03	1.16	1.29	1.38
40	1.46	1.27	1.11	.98	.90	.87	.89	.94	1.05	1.21	1.41	1.53
45	1.65	1.37	1.15	.99	.90	.86	.88	.94	1.08	1.29	1.57	1.76
50	1.96	1.52	1.21	1.00	.89	.85	.87	.95	1.11	1.40	1.82	2.14
55	2.51	1.75	1.29	1.01	.89	.84	.86	.95	1.16	1.56	2.25	2.88
(Latitude-Tilt) = -15.0												
25	1.21	1.11	1.00	.91	.84	.82	.83	.88	.96	1.07	1.18	1.24
30	1.28	1.15	1.01	.90	.83	.80	.81	.87	.97	1.10	1.24	1.32
35	1.37	1.20	1.03	.90	.82	.79	.80	.86	.97	1.14	1.32	1.43
40	1.51	1.27	1.06	.90	.81	.77	.79	.86	.99	1.19	1.44	1.60
45	1.71	1.37	1.10	.90	.80	.76	.77	.85	1.01	1.27	1.61	1.84
50	2.04	1.52	1.15	.91	.79	.74	.76	.85	1.04	1.38	1.88	2.26
55	2.63	1.76	1.23	.92	.78	.73	.75	.85	1.08	1.54	2.33	3.05
Vertical Surface												
25	.94	.78	.62	.48	.42	.40	.41	.45	.56	.73	.90	.99
30	1.04	.85	.67	.52	.44	.42	.43	.48	.60	.79	.99	1.10
35	1.17	.94	.72	.55	.47	.44	.45	.51	.65	.86	1.10	1.24
40	1.33	1.04	.78	.59	.50	.47	.48	.55	.70	.95	1.25	1.44
45	1.57	1.18	.86	.64	.53	.49	.51	.59	.76	1.06	1.45	1.72
50	1.93	1.36	.95	.68	.56	.52	.54	.63	.82	1.20	1.75	2.17
55	2.55	1.62	1.06	.74	.60	.55	.57	.67	.91	1.40	2.24	3.00

A-103

Table A-6 Ratio of Monthly Average - Daily Radiation on a Tilted Surface to that on a Horizontal Surface

4930.2

S

 $\bar{R}$  for  $K_t = .50$ 

A-III

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
(Latitude-Tilt) = 15.0												
25	1.14	1.09	1.05	1.01	.98	.97	.97	1.00	1.03	1.08	1.12	1.15
30	1.23	1.16	1.08	1.02	.97	.96	.96	1.00	1.06	1.13	1.21	1.26
35	1.37	1.24	1.13	1.03	.97	.95	.96	1.01	1.09	1.20	1.33	1.41
40	1.55	1.36	1.19	1.05	.97	.94	.96	1.02	1.13	1.30	1.49	1.62
45	1.82	1.51	1.26	1.08	.98	.94	.96	1.03	1.18	1.42	1.72	1.93
50	2.24	1.73	1.36	1.12	.99	.94	.96	1.06	1.25	1.59	2.08	2.45
55	2.99	2.06	1.50	1.16	1.00	.94	.96	1.08	1.34	1.83	2.67	3.44
(Latitude-Tilt) = .0												
25	1.29	1.19	1.08	.98	.91	.88	.90	.95	1.04	1.15	1.26	1.32
30	1.40	1.26	1.11	.99	.91	.87	.89	.95	1.06	1.21	1.36	1.45
35	1.56	1.35	1.16	1.00	.90	.86	.88	.96	1.09	1.28	1.50	1.63
40	1.77	1.48	1.22	1.02	.90	.86	.88	.97	1.13	1.38	1.68	1.87
45	2.08	1.65	1.30	1.04	.90	.85	.87	.98	1.18	1.52	1.95	2.25
50	2.57	1.89	1.40	1.08	.91	.85	.87	1.00	1.25	1.70	2.36	2.86
55	3.44	2.26	1.54	1.12	.92	.85	.88	1.02	1.34	1.97	3.04	4.02
(Latitude-Tilt) = -15.0												
25	1.38	1.22	1.05	.91	.81	.77	.79	.86	.99	1.16	1.33	1.43
30	1.50	1.29	1.09	.91	.80	.76	.78	.86	1.01	1.22	1.44	1.57
35	1.66	1.39	1.13	.92	.80	.75	.77	.86	1.04	1.30	1.58	1.75
40	1.89	1.52	1.19	.94	.79	.74	.76	.87	1.08	1.40	1.78	2.02
45	2.22	1.69	1.26	.96	.79	.73	.76	.88	1.12	1.53	2.06	2.43
50	2.75	1.94	1.36	.98	.79	.73	.76	.89	1.19	1.72	2.49	3.09
55	3.68	2.32	1.50	1.02	.80	.72	.75	.91	1.27	1.99	3.22	4.34
Vertical Surface												
25	1.13	.89	.63	.42	.32	.29	.30	.37	.53	.80	1.06	1.21
30	1.29	1.00	.71	.47	.35	.32	.33	.41	.60	.89	1.20	1.38
35	1.48	1.13	.79	.53	.40	.35	.37	.47	.67	1.01	1.38	1.60
40	1.74	1.29	.89	.59	.44	.39	.41	.52	.75	1.14	1.61	1.91
45	2.11	1.50	1.00	.66	.49	.44	.46	.58	.84	1.31	1.92	2.34
50	2.67	1.78	1.14	.73	.54	.48	.51	.64	.95	1.54	2.39	3.04
55	3.64	2.19	1.32	.81	.60	.53	.56	.71	1.08	1.84	3.15	4.34

61.30

Table A-6 (continued) Ratio of Monthly Average - Daily Radiation on a Tilted Surface to that on a Horizontal Surface

$\bar{R}$  for  $K_t = .40$

A-110

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
(Latitude-Tilt) = 15.0												
25	1.11	1.08	1.04	1.01	.98	.97	.98	1.00	1.03	1.07	1.10	1.13
30	1.20	1.13	1.07	1.01	.98	.96	.97	1.00	1.05	1.11	1.18	1.22
35	1.31	1.20	1.11	1.03	.97	.95	.96	1.00	1.07	1.17	1.28	1.34
40	1.46	1.30	1.15	1.04	.97	.94	.96	1.01	1.10	1.25	1.41	1.52
45	1.69	1.43	1.21	1.06	.97	.94	.95	1.02	1.15	1.35	1.61	1.79
50	2.04	1.61	1.30	1.09	.98	.94	.95	1.04	1.20	1.49	1.90	2.22
55	2.68	1.89	1.41	1.12	.98	.93	.95	1.06	1.27	1.70	2.41	3.06
(Latitude-Tilt) = .0												
25	1.24	1.15	1.06	.98	.92	.90	.91	.95	1.03	1.12	1.22	1.27
30	1.34	1.21	1.09	.98	.91	.88	.90	.95	1.04	1.17	1.30	1.38
35	1.46	1.29	1.13	.99	.91	.87	.89	.95	1.07	1.23	1.41	1.52
40	1.64	1.39	1.17	1.00	.90	.86	.88	.96	1.10	1.31	1.57	1.73
45	1.90	1.53	1.23	1.02	.90	.86	.88	.96	1.14	1.42	1.79	2.04
50	2.32	1.74	1.32	1.04	.90	.85	.87	.93	1.19	1.58	2.13	2.56
55	3.05	2.04	1.43	1.07	.90	.84	.87	.99	1.27	1.80	2.71	3.54
(Latitude-Tilt) = -15.0												
25	1.31	1.17	1.03	.91	.82	.79	.80	.87	.98	1.12	1.27	1.35
30	1.41	1.23	1.06	.91	.81	.77	.79	.86	.99	1.17	1.36	1.46
35	1.54	1.31	1.09	.91	.80	.76	.78	.86	1.01	1.23	1.47	1.62
40	1.73	1.41	1.13	.92	.80	.75	.77	.86	1.04	1.31	1.64	1.84
45	2.01	1.56	1.19	.93	.79	.74	.76	.87	1.08	1.42	1.87	2.18
50	2.45	1.77	1.27	.95	.79	.73	.76	.88	1.12	1.58	2.23	2.74
55	3.24	2.08	1.39	.98	.79	.72	.75	.89	1.19	1.81	2.85	3.80
Vertical Surface												
25	1.05	.84	.63	.44	.36	.34	.35	.40	.54	.77	.99	1.12
30	1.18	.94	.69	.49	.39	.36	.37	.44	.60	.85	1.11	1.26
35	1.35	1.05	.76	.54	.43	.39	.41	.49	.66	.95	1.26	1.45
40	1.57	1.18	.84	.59	.47	.42	.44	.53	.73	1.06	1.46	1.71
45	1.88	1.36	.94	.65	.51	.46	.48	.58	.81	1.21	1.73	2.08
50	2.36	1.60	1.06	.71	.55	.50	.52	.63	.90	1.39	2.12	2.68
55	3.18	1.95	1.21	.78	.60	.54	.56	.69	1.00	1.66	2.76	3.78

Table A-6 (continued) Ratio of Monthly Average - Daily Radiation on a Tilted Surface to that on a Horizontal Surface

$\bar{R}$  for  $K_t = .60$ 

LATITUDE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
(Latitude-Tilt) = 15.0												
25	1.15	1.11	1.06	1.01	.98	.96	.97	1.00	1.04	1.09	1.14	1.17
30	1.27	1.18	1.10	1.02	.97	.95	.96	1.00	1.07	1.15	1.24	1.29
35	1.41	1.28	1.15	1.04	.97	.94	.96	1.01	1.10	1.23	1.37	1.46
40	1.62	1.40	1.21	1.07	.98	.94	.95	1.02	1.15	1.34	1.56	1.70
45	1.92	1.58	1.30	1.10	.98	.94	.96	1.04	1.21	1.48	1.82	2.05
50	2.40	1.83	1.41	1.14	.99	.94	.96	1.07	1.29	1.67	2.22	2.64
55	3.24	2.20	1.57	1.19	1.01	.94	.97	1.10	1.39	1.95	2.89	3.75
(Latitude-Tilt) = .0												
25	1.33	1.21	1.09	.98	.91	.87	.89	.95	1.05	1.17	1.30	1.37
30	1.46	1.30	1.13	.99	.90	.86	.88	.95	1.08	1.24	1.41	1.51
35	1.63	1.40	1.19	1.01	.90	.85	.87	.96	1.11	1.33	1.57	1.71
40	1.88	1.55	1.26	1.03	.90	.85	.87	.97	1.16	1.44	1.78	1.99
45	2.23	1.74	1.35	1.06	.91	.85	.87	.99	1.22	1.59	2.08	2.41
50	2.78	2.02	1.47	1.10	.92	.85	.88	1.01	1.30	1.81	2.54	3.10
55	3.76	2.43	1.63	1.15	.93	.85	.88	1.05	1.40	2.11	3.31	4.41
(Latitude-Tilt) = -15.0												
25	1.43	1.26	1.07	.91	.80	.75	.77	.86	1.00	1.19	1.39	1.49
30	1.57	1.34	1.11	.92	.79	.74	.76	.86	1.03	1.26	1.51	1.65
35	1.76	1.45	1.16	.93	.79	.73	.76	.86	1.06	1.35	1.67	1.86
40	2.02	1.60	1.23	.95	.79	.73	.75	.87	1.11	1.47	1.90	2.17
45	2.40	1.80	1.32	.98	.79	.72	.75	.89	1.16	1.62	2.22	2.62
50	2.99	2.09	1.44	1.01	.80	.72	.75	.91	1.24	1.84	2.70	3.37
55	4.04	2.52	1.59	1.05	.81	.72	.76	.93	1.34	2.15	3.52	4.78
Vertical Surface												
25	1.20	.92	.63	.39	.28	.25	.26	.34	.53	.82	1.12	1.28
30	1.37	1.04	.72	.46	.32	.28	.30	.39	.60	.93	1.26	1.48
35	1.59	1.19	.81	.52	.37	.32	.34	.45	.68	1.06	1.48	1.73
40	1.88	1.37	.92	.59	.42	.37	.39	.51	.77	1.21	1.73	2.07
45	2.30	1.61	1.05	.66	.48	.42	.44	.58	.87	1.40	2.09	2.56
50	2.93	1.93	1.21	.75	.54	.47	.50	.65	.99	1.65	2.61	3.34
55	4.01	2.39	1.41	.84	.60	.52	.55	.72	1.13	2.00	3.46	4.80

Table A-6 (continued) Ratio of Monthly Average - Daily Radiation on a Tilted Surface to that on a Horizontal Surface

State	Station	97 1/2% Winter Design Temp. <sup>a</sup>	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Yearly Total	
Ala.	Birmingham.....	A	22	0	0	6	93	363	555	592	462	363	108	9	0	2551
	Huntsville.....	A	17	0	0	12	127	426	663	694	557	434	138	19	0	3070
	Mobile.....	A	29	0	0	0	22	213	357	415	300	211	42	0	0	1560
	Montgomery.....	A	26	0	0	0	68	330	527	543	417	316	90	0	0	2291
Alaska.	Anchorage.....	A	-22	245	291	516	930	1284	1572	1631	1316	1293	879	592	315	10854
	Fairbanks.....	A	-50	171	332	642	1203	1833	2254	2359	1901	1739	1068	555	222	14279
	Juneau.....	A	-4	301	338	485	725	921	1135	1237	1070	1073	810	601	381	5075
	Nome.....	A	-28	481	496	693	1094	1455	1820	1879	1666	1770	1314	930	573	14171
Ariz.	Flagstaff.....	A	5	46	68	201	558	867	1073	1169	991	911	651	437	180	7152
	Phoenix.....	A	34	0	0	0	22	234	415	474	328	217	75	0	0	1765
	Tucson.....	A	32	0	0	0	25	231	406	471	344	242	75	6	0	1800
	Winslow.....	A	13	0	0	6	245	711	1008	1054	770	601	291	96	0	4782
	Yuma.....	A	40	0	0	0	0	108	264	307	190	90	15	0	0	974
Ark.	Fort Smith.....	A	19	0	0	12	127	450	704	781	596	456	144	22	0	3292
	Little Rock.....	A	23	0	0	9	127	465	716	756	577	434	126	9	0	3219
	Texarkana.....	A	26	0	0	0	78	345	561	626	468	350	105	0	0	2533
Calif.	Bakersfield.....	A	33	0	0	0	37	282	502	546	364	267	105	19	0	2122
	Bishop.....	A	0	0	0	48	260	576	797	874	680	555	366	143	36	4275
	Blue Canyon.....	A	28	37	108	347	594	781	896	795	806	597	412	195	0	5596
	Burbank.....	A	38	0	0	6	43	177	301	366	277	239	138	81	18	1646
	Eureka.....	C	35	270	257	258	329	414	499	546	470	505	438	372	285	4643
	Fresno.....	A	31	0	0	0	84	354	577	605	426	335	162	62	6	2611
	Long Beach.....	A	38	0	0	9	47	171	316	397	311	264	171	93	24	1803
	Los Angeles.....	A	43	28	28	42	78	180	291	372	302	255	219	158	81	2061
	Los Angeles.....	C	44	0	0	6	31	132	229	310	230	202	123	68	18	1349
	Mt. Shasta.....	C	25	34	123	406	696	902	983	784	738	525	347	159	0	5722
	Oakland.....	A	37	53	50	45	127	309	481	527	400	353	255	180	90	2870
	Red Bluff.....	A	0	0	0	0	53	318	555	605	428	341	168	47	0	2515
	Sacramento.....	A	32	0	0	0	56	321	546	583	414	332	178	72	0	2502
	Sacramento.....	C	0	0	0	0	62	312	533	561	392	310	173	76	0	2419
	Sandberg.....	C	0	0	0	30	202	480	691	778	661	620	426	264	57	4209
	San Diego.....	A	44	9	0	21	43	135	236	298	235	214	135	90	42	1458
	San Francisco.....	A	37	81	78	60	143	306	462	508	395	363	279	214	126	3015
San Francisco.....	C	42	192	174	102	118	231	388	443	336	319	279	239	180	3301	
Santa Maria.....	A	34	99	93	96	146	270	391	459	370	363	282	235	165	2967	
Colo.	Alamosa.....	A	-13	65	99	279	639	1065	1420	1476	1162	1020	696	440	168	8529
	Colorado Springs.....	A	4	9	25	132	456	825	1032	1128	938	893	582	319	84	6423
	Denver.....	A	3	6	9	117	428	819	1035	1132	938	887	558	285	66	6287
	Denver.....	C	0	0	0	90	306	714	905	1004	851	800	492	254	48	5524
	Grand Junction.....	A	11	0	0	30	313	786	1113	1209	907	729	387	146	21	5641
	Pueblo.....	A	-1	0	0	54	326	750	986	1085	871	772	429	174	15	5462
Conn.	Bridgeport.....	A	8	0	0	66	307	615	906	1079	966	853	510	268	27	5617
	Hartford.....	A	5	0	12	117	394	714	1101	1190	1042	808	519	205	33	6235
	New Haven.....	A	9	0	12	87	347	648	1011	1097	991	871	543	245	48	5897
Del.	Wilmington.....	A	15	0	0	51	270	588	927	980	874	735	387	112	6	4930
D.C.	Washington.....	A	19	0	0	33	217	519	834	871	762	626	288	74	0	4224
Fla.	Apalachicola.....	C	0	0	0	16	153	319	347	260	180	33	0	0	0	1308
	Daytona Beach.....	A	36	0	0	0	0	75	211	248	190	140	15	0	0	879
	Fort Myers.....	A	42	0	0	0	0	24	109	146	101	62	0	0	0	442
	Jacksonville.....	A	32	0	0	0	12	144	310	332	246	174	21	0	0	1239
	Key West.....	A	58	0	0	0	0	0	28	40	31	9	0	0	0	106
	Lakeand.....	C	39	0	0	0	0	57	164	195	146	99	0	0	0	661
Miami.....	A	47	0	0	0	0	0	65	74	56	19	0	0	0	214	

<sup>a</sup>Data for United States cities from a publication of the United States Weather Bureau, *Monthly Normals of Temperature, Precipitation and Heating Degree Days, 1962*, are for the period 1931 to 1960 inclusive. These data also include information from the 1963 revisions to this publication, where available.

<sup>b</sup>Data for airport stations, A, and city stations, C, are both given where available.

<sup>c</sup>Data for Canadian cities were computed by the Climatology Division, Department of Transport from normal monthly mean temperatures, and the monthly values of heating degree days data were obtained using the National Research Council computer and a method devised by H. C. S. Thom of the United States Weather Bureau. The heating degree days are based on the period from 1931 to 1960.

<sup>d</sup>Date from ASHRAE Handbook of Fundamentals, 1972.

Table A-3 Average Monthly and Yearly Degree Days (Base 65°F) and 97 1/2% Winter Design Temperatures (a, b, c, d)

From: ASHRAE Systems Handbook 1976

4930.2

Table A-4 (Continued)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
<b>DISTRICT OF COLUMBIA</b>													
Washington (WBCO).....	$\bar{I}_H$	632.4	901.5	1255	1600.4	1846.8	2080.8	1923.9	1712.2	1446.1	1083.4	763.5	594.1
Lat. 38°51'N.....	$K_t$	0.445	0.470	0.496	0.504	0.516	0.553	0.524	0.516	0.520	0.506	0.464	0.460
El. 64 ft.....	$t_0$	38.4	39.6	48.1	57.5	67.7	76.2	79.9	77.9	72.2	60.9	50.2	40.2
<b>FLORIDA</b>													
Apalachicola.....	$\bar{I}_H$	1107	1378.2	1654.2	2040.9	2288.6	2195.9	1978.6	1912.9	1703.3	1544.6	1243.2	982.3
Lat. 29°45'N.....	$K_t$	0.577	0.584	0.576	0.612	0.620	0.594	0.542	0.538	0.559	0.608	0.574	0.543
El. 135 ft.....	$t_0$	57.3	59.0	62.9	69.5	76.4	81.5	83.1	83.1	80.6	73.2	63.7	56.55
Gainesville.....	$\bar{I}_H$	1036.9	1324.7	1635	1956.4	1934.7	1960.9	1895.4	1873.8	1615.1	1312.2	1169.7	919.5
Lat. 29°39'N.....	$K_t$	0.535	0.56	0.568	0.587	0.538	0.531	0.519	0.547	0.529	0.515	0.537	0.508
El. 165 ft.....	$t_0$	62.1	63.1	67.5	72.8	79.4	81.4	83.8	84.1	82	75.7	67.2	62.4
Miami.....	$\bar{I}_H$	1292.2	1554.6	1828.8	2020.6	2068.6	1991.5	1992.6	1890.8	1646.8	1436.5	1321	1183.4
Lat. 25°47'N.....	$K_t$	0.604	0.616	0.612	0.600	0.578	0.545	0.552	0.549	0.525	0.534	0.559	0.588
El. 9 ft.....	$t_0$	71.6	72.0	73.8	77.0	79.9	82.9	84.1	84.5	83.3	80.2	75.6	72.6
Tampa.....	$\bar{I}_H$	1223.6	1461.2	1771.9	2016.2	2228	2146.5	1991.9	1845.4	1687.8	1493.3	1328.4	1119.5
Lat. 27°55'N.....	$K_t$	0.605	0.600	0.606	0.602	0.620	0.583	0.548	0.537	0.546	0.572	0.590	0.569
El. 11 ft.....	$t_0$	64.2	65.7	68.8	74.3	79.4	83.0	84.0	84.4	82.9	77.2	69.6	65.5
<b>GEORGIA</b>													
Atlanta.....	$\bar{I}_H$	848	1080.1	1426.9	1807	2618.1	2002.6	2002.9	1898.1	1519.2	1290.8	987.8	751.1
Lat. 33°39'N.....	$K_t$	0.493	0.496	0.522	0.551	0.561	0.564	0.545	0.559	0.515	0.543	0.510	0.474
El. 976 ft.....	$t_0$	47.2	49.6	55.9	65.0	73.2	80.9	82.4	81.6	77.4	66.5	54.8	47.7
Crittin.....	$\bar{I}_H$	889.6	1135.8	1450.9	1923.6	2163.1	2174	2064.9	1961.2	1605.9	1352.4	1073.8	781.5
Lat. 33°15'N.....	$K_t$	0.513	0.517	0.528	0.586	0.601	0.593	0.562	0.578	0.543	0.565	0.545	0.487
El. 980 ft.....	$t_0$	48.9	51.0	59.1	66.7	74.6	81.2	83.0	82.2	78.4	68	57.3	49.4
<b>IDAHO</b>													
Boise.....	$\bar{I}_H$	518.8	884.9	1280.4	1814.4	2169.3	2376.7	2590.3	2149.4	1717.7	1178.4	678.6	456.6
Lat. 43°34'N.....	$K_t$	0.446	0.533	0.548	0.594	0.619	0.631	0.684	0.660	0.656	0.588	0.494	0.442
El. 2844 ft.....	$t_0$	29.5	36.5	45.0	53.5	62.1	69.3	79.6	77.2	66.7	56.3	42.3	33.1
<b>ILLINOIS</b>													
Lemont.....	$\bar{I}_H$	(590)	879	1255.7	1481.5	1866	2041.7	1900.8	1836.9	1469.4	1015.5	(639)	(531)
Lat. 41°40'N.....	$K_t$	(0.464)	0.496	0.520	0.477	0.525	0.542	0.542	0.559	0.547	0.506	(0.433)	(0.467)
El. 595 ft.....	$t_0$	28.9	30.3	39.5	49.7	59.2	70.8	75.6	74.3	67.2	57.6	43.0	30.6
<b>INDIANA</b>													
Indianapolis.....	$\bar{I}_H$	526.2	797.4	1184.1	1481.2	1828	2042	2039.5	1832.1	1513.3	1094.4	662.4	491.1
Lat. 39°44'N.....	$K_t$	0.330	0.424	0.472	0.47	0.511	0.543	0.554	0.552	0.549	0.520	0.413	0.391
El. 793 ft.....	$t_0$	31.3	33.9	43.0	54.1	64.9	74.8	79.6	77.4	70.6	59.3	44.2	33.4
<b>KANSAS</b>													
Dodge City.....	$\bar{I}_H$	953.1	1186.3	1565.7	1975.6	2126.5	2459.8	2403.7	2210.7	1841.7	1471	1065.3	873.8
Lat. 37°46'N.....	$K_t$	0.639	0.598	0.606	0.618	0.594	0.655	0.652	0.663	0.654	0.650	0.625	0.622
El. 2592 ft.....	$t_0$	33.8	38.7	46.5	57.7	66.7	77.2	83.8	82.4	73.7	61.7	46.5	36.8
<b>KENTUCKY</b>													
Lexington.....	$\bar{I}_H$	-	-	-	1834.7	2171.2	-	2246.5	2064.9	1775.6	1316.8	-	681.5
Lat. 38°02'N.....	$K_t$	-	-	-	0.575	0.606	-	0.610	0.619	0.631	0.634	-	0.513
El. 979 ft.....	$t_0$	36.5	38.8	47.4	57.8	67.5	76.2	79.8	78.2	72.8	61.2	47.6	35.5
<b>LOUISIANA</b>													
Lake Charles.....	$\bar{I}_H$	899.2	1145.7	1487.4	1801.8	2080.4	2213.3	1968.6	1910.3	1678.2	1505.5	1122.1	875.6
Lat. 30°13'N.....	$K_t$	0.473	0.492	0.521	0.542	0.578	0.597	0.538	0.558	0.553	0.597	0.524	0.494
El. 12 ft.....	$t_0$	55.3	58.7	63.5	70.9	77.4	83.4	84.8	85.0	81.5	73.8	62.6	56.9
<b>MAINE</b>													
Caribou.....	$\bar{I}_H$	497	861.6	1360.1	1495.9	1779.7	1779.7	1898.1	1675.6	1254.6	793	415.3	398.9
Lat. 46°52'N.....	$K_t$	0.504	0.579	0.619	0.507	0.509	0.473	0.522	0.527	0.526	0.455	0.352	0.470
El. 628 ft.....	$t_0$	11.5	12.8	24.4	37.3	51.8	61.8	67.2	65.0	56.2	44.7	31.3	18.8
Portland.....	$\bar{I}_H$	565.7	874.5	1329.5	1526.4	1923.2	2017.3	2095.6	1799.2	1428.8	1035	591.5	527.7
Lat. 43°39'N.....	$K_t$	0.482	0.524	0.569	0.570	0.544	0.536	0.572	0.554	0.546	0.539	0.431	0.491
El. 63 ft.....	$t_0$	23.7	24.5	34.4	44.8	55.4	65.1	71.1	69.7	61.9	51.8	40.3	28.0
<b>MANITOBA</b>													
Winnipeg.....	$\bar{I}_H$	488.2	835.4	1354.2	1641.3	1904.4	1962	2123.8	1761.2	1193.4	787.5	444.6	345.7
Lat. 49°54'N.....	$K_t$	0.601	0.636	0.661	0.574	0.550	0.524	0.587	0.567	0.504	0.487	0.436	0.503
El. 786 ft.....	$t_0$	3.2	7.1	21.3	40.9	55.9	65.3	71.5	69.4	58.8	45.8	25.2	10.1
<b>MASSACHUSETTS</b>													
Blue Hill.....	$\bar{I}_H$	555.3	797	1145.9	1438	1776.4	1943.9	1881.5	1622.1	1314	941	592.2	482.3
Lat. 42°13'N.....	$K_t$	0.445	0.458	0.477	0.464	0.501	0.516	0.513	0.495	0.442	0.427	0.406	0.436
El. 629 ft.....	$t_0$	26.4	28.3	36.9	45.9	56.5	67.2	74.3	70.6	64.2	54.1	43.3	31.5
Boston.....	$\bar{I}_H$	505.5	734	1070.1	1355	1709	1864	1889.5	1577.1	1287.5	894	535.9	422.8
Lat. 42°22'N.....	$K_t$	0.471	0.426	0.445	0.438	0.489	0.497	0.507	0.483	0.477	0.453	0.372	0.409
El. 29 ft.....	$t_0$	31.4	31.4	39.9	49.5	60.4	69.8	74.5	71.8	60.8	52.4	46.4	34.9

ICE STORAGE SYSTEM

SYSTEM CONCEPTUAL DESIGN AND DESIGN PARAMETERS

1. Equipment Selection

Positive displacement direct drive screw compressor selected because of efficient operation over a wide range of suction and discharge temperatures, small size and simple capacity control.

The compressor has to be selected to be able to satisfy the winter heat requirements for space heating and DHW at peak time which amount to  $19.5 \times 10^6$  BTUH at a temperature of about 120°F. Three units are selected, model W900 x 2514Q with refrigerant k2-22.

Performance

	Winter	Summer
Saturation		
Suction Temperature (SST)	20°F	35°F
Saturation		
Discharge Temperature (SDT)	130°F	105°F
Heat rejection $6.75 \times 10^6$ BTUH		Heat rejected $9.2 \times 10^6$ BTUH
Refrigeration capacity = 375 ton		Refrigeration capacity = 610 ton
BHP	875	BHP 650

The peak summer cooling demand is 3000 ton and the three compressors can generate simultaneously only 1830 ton. However, this difference between the peak and available capacity will be covered by a chilled water storage system whereby the chillers will operate at night. This chilled water system will enter into operation after the ice generated during the winter will be exhausted.

Part of the ice bin will serve for chilled water storage.

The compressor units can be switched from summer to winter operation and vice versa by automatic suction and discharge temperature control. The condenser has to be selected according to the summer requirements of 9,000,000 BTUH heat rejection at a discharge temperature of 105°F. Thus with condenser water  $\Delta T$  of 10°F the flow will be 1800 gpm.

In winter time, assuming a  $\Delta T$  of 20°F, the flow will only be 675 gpm.

There will be two ice makers for each compressor unit operating alternately in accordance with full winter compressor capacities (375 ton each at 20°F SST). The chillers will be selected according to summer operation conditions (35°F SST, 105°F SDT, 610 ton). The accumulator on the low pressure side of the refrigeration cycle will have the capacity of 1783 lbs. of R-22 at 90°F.

In this configuration the compressor is not influenced by brief changes in load.

## 2 Comparative analysis of DHW Generation in Summertime with Ice Making vs. Chilled Water Generation.

### 2.1 Ice generation (ice - heat source for heat pump).

SST -20°F, SDT - 130°F,  $6.75 \times 10^6$  BTUH heat rejection per unit, "by-product" 375 ton refrigeration available for ice generation, power input 875 BHP (cop =  $\frac{6,750,000}{875 \times 2545} = 3.03$ )

$$\begin{aligned} \text{Overall cop} &= \frac{\text{Refrigeration, effect} + \text{heat rejected}}{\text{power input}} \\ &= \frac{375 \times 12,000 + 6,750,000}{875 \times 2545} = 5.052 \end{aligned}$$

2.2 Chilled Water Generation (chilled water-heat source).

SST - 35°F SDT - 130°F, 860 x 10<sup>6</sup> heat rejection per unit,

"by-product" 520 ton, power input 880BHP (cop =  $\frac{8,600,000}{880 \times 2545} = 3.84$ )

$$\text{Overall cop} = \frac{520 \times 12,000 + 8,600,000}{880 \times 2545} = 6.625$$

Thus the overall cop with chilled water is

$$\frac{6.625 - 5.052}{5.052} \times 100 = 31.1\% \text{ higher}$$

### 3. Night summer operation

3.1 DHW daily load 53,000 gallons. Heating the water from 55°F to 120°F the daily load is:

$$53,000 \times 8.33 \times (120 - 55) = 28.70 \times 10^6 \text{ BTU per day}$$

Using one compressor SST - 35°F, SDT - 130°F we get 8.60 x 10<sup>6</sup>BTUH heat rejection and 520 ton refrigeration. Number of hours of

$$\text{compressor operation} = \frac{28.70 \times 10^6}{8.60 \times 10^6} = 3.3 \text{ hours}$$

Storage required for DHW with an assumed  $\Delta T = 20^\circ\text{F}$

$$= \frac{28.7 \times 10^6}{20 \times 8.33} = 172,000 \text{ gallons}$$

Energy input 880 x 0.746 x 3.3 = 656 kw x 3.3 hours = 2166 kwh per day chilled water ("by-product") produced

$$= 520 \cdot x \ 3.3 = 1716 \text{ ton - hours}$$

### 3.2 Diurnal Chilled Water Storage.

It was seen above that the ice produced during winter operation is not sufficient to cover the cooling requirement in the summer causing a deficiency of "cooling" energy which amounts to 885,100 ton-hours. This inefficiency will have to be covered by the three machines, selected for winter heating, operating, however, between SST - 35°F and SDT - 105°F and using cooling towers for heat rejection. We could use the machines to produce ice with SST - 20°F and SDT - 105°F which results in 450 ton refrigeration, power input 650 BHP, KW/ton = 1.078 and cop = 3.26. However, producing chilled water we operate at SST of 35°F and SDT - 105°F which results in 610 tons of refrigeration, BHP = 650, KW/ton = 0.795 and cop = 4.42. Chilled water generation is thus by far more efficient which was also demonstrated in the case of DHW generation.

Three machines are capable of producing 3 x 610 = 1830 tons. The required peak is 3000 tons. Thus storage of chilled water can solve the problem cutting, in addition, the billing demand charge.

Assuming 5 hours of full load operation on a summer peak day the energy to be stored is 3000 ton + 5 hours = 15000 ton-hours using a  $\Delta T$  - 10°F the volume of the chilled water storage tank comes to

$$\frac{15000 \times 12000}{8.33 \times 10} = 2,160,000 \text{ gallons.}$$

As the volume of the ice bin is 3,400,000 = 25,400,000 gallons we have ample space for the chilled water storage accommodation.

It was found above that the chilled water produced in generating DHW was equivalent to 1716 ton-hours. Thus the additional chilled water energy required is  $15,000 - 1716 = 13,284$  ton-hours. This amount of cooling energy can be produced with one 610 ton machine operating during  $\frac{13,284}{610} = 21.8$  hours say 22 hours. This means that three machines can produce the required daily amount of chilled water and DHW, while operating at night only. Thus

2 machines operating 8 hours = 16 hours

1 machine operating 6 hours = 6 hours

This 22 hour operation using cooling towers would produce 13.284 ton-hours of refrigeration. One machine generating DHW and chilled water would produce 1716 ton-hours of refrigeration. Thus satisfying the chilled water and DHW requirements.

7.4. Cop of Chilled Water and DHW Generation. The energy input in DHW generation in summer time was seen to be 2166 kwh per day (3.1). As a "by-product" 1716 ton-hours of chilled water energy were produced.

The energy input to produce the 13,284 ton-hours of chilled water energy was  $13,784 \text{ ton-hours} \times 0.795 \text{ kw/ton} = 10,561 \text{ kwh}$  per day. Thus the summer night

$$\text{cop} = \frac{15000 \times 12000 \text{ (ch.water)} + 28,700,000 \text{ (DHW)}}{(2166 + 10,561) \times 3413} = 4.81$$

#### 7.5. Annual Overall Cop

On an annual basis the cop can be calculated by considering the sum of all the useful heat transfers and dividing it by the total energy invested in all the systems.

Thus Annual Overall cop =

$$\frac{(\text{HRWT} + \text{ICWT}) - \text{ICSL} + (\text{DHWS}) + \text{CHWS-1} + \text{CHWS-2}}{\sum \text{Annual Energy Input}}$$

Where the terms in the formula are defined as follows:

HRWT - heat rejected in winter time which was used for space heating and DHW generation

ICWT - heat extracted from water in producing ice during winter time

ICSL - ice storage loss

DHWS - domestic hot water generated in the summer

CHWS-1 - chilled water generated in summer time while producing DHW

CHWS-2 - chilled water produced in summer using cooling towers

Substituting the corresponding values

$$\begin{aligned} \text{cop} &= \frac{(3.55 \times 10^{10} + 2.44 \times 10^{10}) - 0.2545 \times 10^{10} + 0.482 \times 10^{10} +}{(3.55 - 2.44) \times 10^{18}} \\ &\quad + \frac{0.482 \times 10^{10} \cdot \frac{2.84}{3.84} + 0.804 \times 10^{10}}{3.84} + \frac{0.804 \times 10^{10}}{12,000} \times 0.795 \times 3413 \\ &= \frac{7.382}{1.4173} = 5.21 \end{aligned}$$

Ice Storage Bin Losses

The storage bin volume is  $3.4 \times 10^6$  cub. ft. The area of the base and top is 170,000 square feet each and the total area of the sides is 33,000 square feet. The bin will be insulated with 4-inch polyurethane on the top ( $R=36.4$ ) and with 2-inch polyurethane ( $R=18.2$ ) on the sides and bottom.

The heat losses were calculated by month thus:

<u>Losses in BTU x 10<sup>8</sup></u>				
June	1,248	0.290	1.138	2.680
July	1,290	0.383	1.280	2.950
August	1,150	0.324	1.366	2.840
September	0.880	0.432	1.320	2.635
October	0.645	0.432	1.265	2.342
November	0.416	0.342	0.832	1.590
December	0.291	0.275	1.100	1.667
January	0.291	0.196	0.010	1.500
February	0.388	0.133	0.845	1.366
March	0,417	0.132	0.923	1.472
April	0.890	0.157	0.930	1.980
May	1.151	0.230	1.050	<u>2.43</u>
			Total =	25.450 x 10 <sup>8</sup>

Thus, the total annual losses amount to:

$$\frac{25,450 \times 10^5}{12,000} = 212,100 \text{ ton hours}$$

which is:  $\frac{212,100}{2.33 \times 10^6} \times 100 = 9.1\%$  of total "cooling" energy generated.

ECONOMIC ANALYSIS PROGRAMS

## ECONOMIC ANALYSIS PROGRAMS

### I ABSOLUTE PRESENT VALUE ANALYSIS

This program generates a discounted cash flow of a given system, based on: system life, capital cost of the system, operation cost in year 0, fuel cost escalation rate, discount rate, and system salvage value at the end of its life. A year by year discounted cash flow is produced for the entire life of the system. At the end of the system life, the discounted salvage value is added to the present value of the system and the life cycle cost is printed. The following formula is used for the analysis

$$PV_s = -C - \left( \sum_{t=1}^N \frac{O(1+i_F)^t}{(1+i_D)^t} \right) + \frac{SV}{(1+i_D)^N}$$

Where:

- $PV_s$  = Present value of the system.  
 $C$  = Capital cost of the system.  
 $N$  = System life.  
 $O$  = Operation energy cost in year 0.  
 $i_F$  = Fuel cost escalation rate.  
 $i_D$  = Discount rate/cost of capital  
 $SV$  = Salvage value of the system after N years.

THE FOLLOWING ASSUMPTIONS WERE MADE:

1. Operation energy costs in year I are escalated and discounted.
2. Fuel cost escalation rate remains constant through the payback period.

### II RELATIVE PRESENT VALUE ANALYSIS

This program generates a discounted cash flow analysis for an energy conserving system relative to a conventional system, based on: system life, incremental capital cost of the system, operation savings in year 0, fuel cost escalation

rate, discount rate, salvage value of the system. A year by year discounted cash flow is produced for the entire life of the system. At the end of the system life, the discounted salvage value is added to the present value of the system and the life cycle savings of the system is printed.

The following formula is used:

$$PV_s = -C_I + \left( \sum_{t=1}^N S \frac{(1+i_F)^t}{(1+i_D)^t} \right) + \frac{SV}{(1+i_D)^N}$$

Where:

- $PV_s$  = Present value of the incremental cost and savings of the system.
- $C_I$  = Incremental capital cost of the system.
- $N$  = System life.
- $S$  = Operation Energy savings in year 0.
- $i_F$  = Fuel cost escalation rate.
- $i_D$  = Discount rate/cost of capital
- $SV$  = Salvage value of the system after N years.

This program allows energy conserving systems to be directly compared to alternate or conventional systems. By examining the payment stream, the discounted payback period may be determined, provided that the payback is within the system life.

ABSOLUTE PRESENT VALUE ANALYSIS

SYMBOL DEFINITIONS:

- PROJECT LIFE - Life of the system used for economic analysis.
- Capital Cost - Absolute capital cost of the system.
- Operation Cost - Operation energy cost in year 0.
- Fuel Escalation - Fuel escalation rate per year.
- Discount Rate - Discount rate/cost of capital per year.
- Salvage Value - Salvage value of the system at the end of the project life.
- Year - The previous year.
- Cost - Discounted and escalated energy costs for that year.
- PVTL - Accumulated total of the present value of energy costs and capital costs.
- Salvage Value (PV) - Present value of the salvage value of the system.
- Present Value - Present value of the total system including - capital cost, energy costs, and salvage value.

PROJECT LIFE  
20.00  
CAPITAL COST  
750000.00  
OPERATION COST  
331710.00  
FUEL ESCALATION  
12.00 %  
DISCOUNT RATE  
8.00 %  
SALVAGE VALUE  
0.00

1. CONVENTIONAL SYSTEM: BOILERS & CHILLERS.

\*RUN

1.00	YEAR	10.00	YEAR	19.00	YEAR
-343995.56	COST	-477200.88	COST	-661987.27	COST
-1093995.56	PVTL	-4823744.65	PVTL	-9997763.68	PVTL
2.00	YEAR	11.00	YEAR	20.00	YEAR
-356736.13	COST	-494874.99	COST	-686505.32	COST
-1450731.69	PVTL	-5318619.64	PVTL	-10684269.00	PVTL
3.00	YEAR	12.00	YEAR	SALVAGE VALUE	(PV)
-369948.58	COST	-513203.69	COST	0.00	
-1820680.27	PVTL	-5831823.33	PVTL	PRESENT VALUE	
				-10684269.00	
4.00	YEAR	13.00	YEAR		
-383650.38	COST	-532211.23	COST		
-2204330.65	PVTL	-6364034.56	PVTL		
5.00	YEAR	14.00	YEAR		
-397859.65	COST	-551922.76	COST		
-2602190.30	PVTL	-6915957.32	PVTL		
6.00	YEAR	15.00	YEAR		
-412595.20	COST	-572364.35	COST		
-3014785.50	PVTL	-7488321.67	PVTL		
7.00	YEAR	16.00	YEAR		
-427876.50	COST	-593563.02	COST		
-3442662.00	PVTL	-8081884.69	PVTL		
8.00	YEAR	17.00	YEAR		
-443723.78	COST	-615546.84	COST		
-3886385.78	PVTL	-8897431.53	PVTL		
9.00	YEAR	18.00	YEAR		
-460157.99	COST	-638344.87	COST		
-4346543.77	PVTL	-9335776.40	PVTL		

PROJECT LIFE  
20.00  
CAPITAL COST  
3400000.00  
OPERATION COST  
264470.00  
FUEL ESCALATION  
12.00 %  
DISCOUNT RATE  
8.00 %  
SALVAGE VALUE  
1334500.00

2. HEAT PUMP SYSTEM WITH SLAB STORAGE.

\*RUN

1.00	YEAR	10.00	YEAR	19.00	YEAR
-274265.19	COST	-380468.88	COST	-527797.70	COST
-3674265.19	PVTL	-6647967.34	PVTL	-10773175.54	PVTL
2.00	YEAR	11.00	YEAR	20.00	YEAR
-284423.16	COST	-394560.27	COST	-547345.76	COST
-3958688.34	PVTL	-7042527.61	PVTL	-11320521.30	PVTL
3.00	YEAR	12.00	YEAR	SALVAGE VALUE	(PV)
-294957.35	COST	-409173.62	COST	286314.58	
-4253645.69	PVTL	-7451701.23	PVTL	PRESENT VALUE	
				-11034206.72	
4.00	YEAR	13.00	YEAR		
-305881.69	COST	-424328.19	COST		
-4559527.38	PVTL	-7876029.42	PVTL		
5.00	YEAR	14.00	YEAR		
-317210.64	COST	-440044.05	COST		
-4876738.02	PVTL	-8316073.48	PVTL		
6.00	YEAR	15.00	YEAR		
-328959.19	COST	-456341.98	COST		
-5205697.21	PVTL	-8772415.46	PVTL		
7.00	YEAR	16.00	YEAR		
-341142.86	COST	-473243.54	COST		
-5546840.07	PVTL	-9245658.99	PVTL		
8.00	YEAR	17.00	YEAR		
-353777.78	COST	-490771.07	COST		
-5900617.85	PVTL	-9736430.07	PVTL		
9.00	YEAR	18.00	YEAR		
-366880.66	COST	-508947.78	COST		
-6267498.51	PVTL	-10245377.85	PVTL		

PROJECT LIFE  
20.00  
CAPITAL COST  
9800000.00  
OPERATION COST  
156000.00  
FUEL ESCALATION  
12.00 %  
DISCOUNT RATE  
8.00 %  
SALVAGE VALUE  
700000.00

3. SOLAR ENERGY ANNUAL STORAGE SYSTEM.

\*RUN

1.00	YEAR	10.00	YEAR	19.00	YEAR
-161777.78	COST	-224422.95	COST	-311326.20	COST
-9961777.78	PVTL	-11715842.65	PVTL	-14149133.68	PVTL
2.00	YEAR	11.00	YEAR	20.00	YEAR
-167769.55	COST	-232734.91	COST	-322856.80	COST
-10129547.33	PVTL	-11948577.56	PVTL	-14471990.48	PVTL
3.00	YEAR	12.00	YEAR	SALVAGE VALUE	(PV)
-173983.23	COST	-241354.72	COST	150183.75	
-10303530.56	PVTL	-12189932.29	PVTL	PRESENT VALUE	
				-14321806.74	
4.00	YEAR	13.00	YEAR		
-180427.06	COST	-250293.79	COST		
-10483957.62	PVTL	-12440226.07	PVTL		
5.00	YEAR	14.00	YEAR		
-187109.54	COST	-259563.93	COST		
-10671067.16	PVTL	-12699790.00	PVTL		
6.00	YEAR	15.00	YEAR		
-194039.52	COST	-269177.41	COST		
-10865106.68	PVTL	-12968967.41	PVTL		
7.00	YEAR	16.00	YEAR		
-201226.17	COST	-279146.94	COST		
-11066332.86	PVTL	-13248114.35	PVTL		
8.00	YEAR	17.00	YEAR		
-208678.99	COST	-289485.72	COST		
-11275011.85	PVTL	-13537600.07	PVTL		
9.00	YEAR	18.00	YEAR		
-216407.85	COST	-300207.41	COST		
-11491419.70	PVTL	-13837887.48	PVTL		

PROJECT LIFE  
20.00  
CAPITAL COST  
2600000.00  
OPERATION COST  
153392.00  
FUEL ESCALATION  
12.00 %  
DISCOUNT RATE  
8.00 %  
SALVAGE VALUE  
850000.00

4. HEAT PUMP SYSTEM: ICE STORAGE (ACES).

\*RUN

1.00	YEAR	10.00	YEAR	19.00	YEAR
-159073.19	COST	-220671.06	COST	-306121.47	COST
-2759073.19	PVTL	-4483813.69	PVTL	-6876425.09	PVTL
2.00	YEAR	11.00	YEAR	20.00	YEAR
-164964.78	COST	-228844.06	COST	-317459.30	COST
-2924037.97	PVTL	-4712657.75	PVTL	-7193884.39	PVTL
3.00	YEAR	12.00	YEAR	SALVAGE VALUE (PV)	
-171074.59	COST	-237319.77	COST	182365.98	
-3095112.56	PVTL	-4949977.52	PVTL	PRESENT VALUE	
				-7011518.41	
4.00	YEAR	13.00	YEAR		
-177410.69	COST	-246109.39	COST		
-3272523.25	PVTL	-5196086.91	PVTL		
5.00	YEAR	14.00	YEAR		
-183981.45	COST	-255224.55	COST		
-3456504.70	PVTL	-5451311.46	PVTL		
6.00	YEAR	15.00	YEAR		
-190795.58	COST	-264677.31	COST		
-3647300.28	PVTL	-5715988.78	PVTL		
7.00	YEAR	16.00	YEAR		
-197862.08	COST	-274480.18	COST		
-3845162.37	PVTL	-5990468.95	PVTL		
8.00	YEAR	17.00	YEAR		
-205190.31	COST	-284646.11	COST		
-4050352.68	PVTL	-6275115.06	PVTL		
9.00	YEAR	18.00	YEAR		
-212789.95	COST	-295188.56	COST		
-4263142.63	PVTL	-6570303.62	PVTL		

PROJECT LIFE  
20.00  
CAPITAL COST  
3752000.00  
OPERATION COST  
129380.00  
FUEL ESCALATION  
12.00 %  
DISCOUNT RATE  
8.00 %  
SALVAGE VALUE  
1237500.00

5. HEAT PUMP SYSTEM: ICE STORAGE (ACES) WITH  
OUTSIDE AIR SOURCE.

\*RUN

1.00	YEAR	10.00	YEAR	19.00	YEAR
-134171.85	CDST	-186127.19	CDST	-258201.18	CDST
-3886171.85	PVTL	-5340921.29	PVTL	-7358993.05	PVTL
2.00	YEAR	11.00	YEAR	20.00	YEAR
-139141.18	CDST	-193020.79	CDST	-267764.19	CDST
-4025313.03	PVTL	-5533942.08	PVTL	-7626757.24	PVTL
3.00	YEAR	12.00	YEAR	SALVAGE VALUE (PV)	
-144294.56	CDST	-200169.71	CDST	265503.41	
-4169607.59	PVTL	-5734111.79	PVTL	PRESENT VALUE	
				-7361253.83	
4.00	YEAR	13.00	YEAR		
-149638.80	CDST	-207583.40	CDST		
-4319246.39	PVTL	-5941695.19	PVTL		
5.00	YEAR	14.00	YEAR		
-155180.98	CDST	-215271.67	CDST		
-4474427.37	PVTL	-6156866.86	PVTL		
6.00	YEAR	15.00	YEAR		
-160928.42	CDST	-223244.70	CDST		
-4635355.79	PVTL	-6380211.56	PVTL		
7.00	YEAR	16.00	YEAR		
-166888.73	CDST	-231513.02	CDST		
-4802244.52	PVTL	-6611724.58	PVTL		
8.00	YEAR	17.00	YEAR		
-173069.80	CDST	-240087.58	CDST		
-4975314.32	PVTL	-6851812.16	PVTL		
9.00	YEAR	18.00	YEAR		
-179479.79	CDST	-248979.71	CDST		
-5154794.11	PVTL	-7100791.87	PVTL		

## Relative Present Value Analysis

### Symbol Definitions:

- Project Life - Life of the system used for economic analysis.
- Capital Cost - Incremental capital cost of the system relative to a conventional system.
- Operation Savings - Operation Savings in year o.
- Fuel Escalation - Fuel escalation rate per year.
- Discount rate - Discount rate/cost of capital per year.
- Salvage value - Salvage value of the system at the end of the project life.
- Year - The previous year.
- SVNG - Discounted and escalated energy cost savings for that year.
- PVTL - Accumulated total of the present value of energy cost savings.
- Salvage value - Present value of the salvage value of the system.
- Present value - Present value of the system, including: incremental capital cost, energy cost savings, and salvage value.

PROJECT LIFE  
20.00  
CAPITAL COST  
2650000.00  
OPERATION SAVINGS  
67240.00  
FUEL ESCALATION  
12.00 %  
DISCOUNT RATE  
8.00 %  
SALVAGE VALUE  
1334500.00

1 VS. 2 CONVENTIONAL VS HEAT PUMP SYSTEM WITH  
SLAB STORAGE.

\*RUN

1.00	YEAR	10.00	YEAR	19.00	YEAR
69730.37	SVNG	96732.05	SVNG	134189.58	SVNG
-2580269.63	PVTL	-1824222.69	PVTL	-775411.87	PVTL
2.00	YEAR	11.00	YEAR	20.00	YEAR
72312.98	SVNG	100314.72	SVNG	139159.56	SVNG
-2507956.65	PVTL	-1723907.98	PVTL	-636252.31	PVTL
3.00	YEAR	12.00	YEAR	SALVAGE VALUE	
74991.24	SVNG	104030.07	SVNG	286314.58	
-2432965.42	PVTL	-1619877.90	PVTL	PRESENT. VALUE	
				-349937.72	
4.00	YEAR	13.00	YEAR		
77768.69	SVNG	107883.04	SVNG		
-2355196.73	PVTL	-1511994.86	PVTL		
5.00	YEAR	14.00	YEAR		
80649.01	SVNG	111878.71	SVNG		
-2274547.72	PVTL	-1400116.15	PVTL		
6.00	YEAR	15.00	YEAR		
83636.01	SVNG	116022.36	SVNG		
-2190911.71	PVTL	-1284093.79	PVTL		
7.00	YEAR	16.00	YEAR		
86733.64	SVNG	120319.49	SVNG		
-2104178.07	PVTL	-1163774.30	PVTL		
8.00	YEAR	17.00	YEAR		
89946.00	SVNG	124775.77	SVNG		
-2014232.07	PVTL	-1038998.53	PVTL		
9.00	YEAR	18.00	YEAR		
93277.33	SVNG	129397.09	SVNG		
-1920954.74	PVTL	-909601.44	PVTL		

PROJECT LIFE  
20.00  
CAPITAL COST  
9050000.00  
OPERATION SAVINGS  
175710.00  
FUEL ESCALATION  
12.00 %  
DISCOUNT RATE  
8.00 %  
SALVAGE VALUE  
700000.00

1 VS 3. CONVENTIONAL VS SOLAR ENERGY ANNUAL  
STORAGE SYSTEM.

\*RUN

1.00	YEAR	10.00	YEAR	19.00	YEAR
182217.78	SVNG	252777.93	SVNG	350661.07	SVNG
-8867782.22	PVTL	-6892098.00	PVTL	-4151370.00	PVTL
2.00	YEAR	11.00	YEAR	20.00	YEAR
188966.58	SVNG	262140.07	SVNG	363648.52	SVNG
-8678815.64	PVTL	-6629957.93	PVTL	-3787721.49	PVTL
3.00	YEAR	12.00	YEAR	SALVAGE VALUE	
195965.35	SVNG	271848.97	SVNG	150183.75	
-8482850.29	PVTL	-6358108.96	PVTL	PRESENT VALUE	
				-3637537.74	
4.00	YEAR	13.00	YEAR		
203223.32	SVNG	281917.45	SVNG		
-8279626.97	PVTL	-6076191.52	PVTL		
5.00	YEAR	14.00	YEAR		
210750.11	SVNG	292358.83	SVNG		
-8068876.86	PVTL	-5783832.68	PVTL		
6.00	YEAR	15.00	YEAR		
218555.67	SVNG	303186.94	SVNG		
-7850321.18	PVTL	-5480645.74	PVTL		
7.00	YEAR	16.00	YEAR		
226650.33	SVNG	314416.08	SVNG		
-7623670.86	PVTL	-5166229.66	PVTL		
8.00	YEAR	17.00	YEAR		
235044.78	SVNG	326061.12	SVNG		
-7388626.07	PVTL	-4840168.54	PVTL		
9.00	YEAR	18.00	YEAR		
243750.15	SVNG	338137.48	SVNG		
-7144875.93	PVTL	-4502031.08	PVTL		

PROJECT LIFE  
20.00  
CAPITAL COST  
1850000.00  
OPERATION SAVINGS  
178318.00  
FUEL ESCALATION  
12.00 %  
DISCOUNT RATE  
8.00 %  
SALVAGE VALUE  
850000.00

1 VS 4. CONVENTIONAL VS HEAT PUMP SYSTEM:  
ICE STORAGE (ACES).

\*RUN

1.00	YEAR	10.00	YEAR	19.00	YEAR
184922.37	SVNG	256529.82	SVNG	355865.81	SVNG
-1665077.63	PVTL	339930.96	PVTL	3121338.59	PVTL
2.00	YEAR	11.00	YEAR	20.00	YEAR
191771.35	SVNG	266030.92	SVNG	369046.02	SVNG
-1473306.28	PVTL	605961.88	PVTL	3490384.61	PVTL
3.00	YEAR	12.00	YEAR	SALVAGE VALUE	
198873.99	SVNG	275883.92	SVNG	182365.98	
-1274432.29	PVTL	881845.80	PVTL	PRESENT VALUE	
				3672750.59	
4.00	YEAR	13.00	YEAR		
206239.69	SVNG	286101.84	SVNG		
-1068192.60	PVTL	1167947.65	PVTL		
5.00	YEAR	14.00	YEAR		
213878.20	SVNG	296698.21	SVNG		
-854314.40	PVTL	1464645.86	PVTL		
6.00	YEAR	15.00	YEAR		
221799.61	SVNG	307687.03	SVNG		
-632514.79	PVTL	1772382.89	PVTL		
7.00	YEAR	16.00	YEAR		
230014.42	SVNG	319082.85	SVNG		
-402500.37	PVTL	2091415.74	PVTL		
8.00	YEAR	17.00	YEAR		
238533.47	SVNG	330500.73	SVNG		
-163966.90	PVTL	2423316.47	PVTL		
9.00	YEAR	18.00	YEAR		
247366.04	SVNG	343156.31	SVNG		
83401.14	PVTL	2765472.78	PVTL		

PROJECT LIFE  
20.00  
CAPITAL COST  
3002000.00  
OPERATION SAVINGS  
202330.00  
FUEL ESCALATION  
12.00 %  
DISCOUNT RATE  
8.00 %  
SALVAGE VALUE  
1237500.00

1 VS 5. CONVENTIONAL VS HEAT PUMP SYSTEM:  
ICE STORAGE (ACES) WITH OUTSIDE  
AIR SOURCE.

\*RUN

1.00	YEAR	10.00	YEAR	19.00	YEAR
209823.70	SVNG	291073.69	SVNG	403786.09	SVNG
-2792176.30	PVTL	-517176.65	PVTL	2638770.63	PVTL
2.00	YEAR	11.00	YEAR	20.00	YEAR
217594.95	SVNG	301854.20	SVNG	418741.13	SVNG
-2574581.34	PVTL	-215322.45	PVTL	3057511.76	PVTL
3.00	YEAR	12.00	YEAR	SALVAGE VALUE	
225654.02	SVNG	313033.98	SVNG	265503.41	
-2348927.32	PVTL	97711.54	PVTL	PRESENT VALUE	
				3323015.17	
4.00	YEAR	13.00	YEAR		
234011.58	SVNG	324627.83	SVNG		
-2114915.74	PVTL	422339.37	PVTL		
5.00	YEAR	14.00	YEAR		
242678.68	SVNG	336651.09	SVNG		
-1872237.06	PVTL	758990.46	PVTL		
6.00	YEAR	15.00	YEAR		
251666.78	SVNG	349119.65	SVNG		
-1620570.29	PVTL	1108110.10	PVTL		
7.00	YEAR	16.00	YEAR		
260987.77	SVNG	362050.00	SVNG		
-1359582.52	PVTL	1470160.11	PVTL		
8.00	YEAR	17.00	YEAR		
270653.98	SVNG	375459.26	SVNG		
-1088928.54	PVTL	1845619.37	PVTL		
9.00	YEAR	18.00	YEAR		
280678.20	SVNG	389365.16	SVNG		
-808250.34	PVTL	2234984.53	PVTL		

DISCUSSION OF ECONOMIC ANALYSIS:

CONVENTIONAL SYSTEM: BOILERS AND CHILLERS VS.  
HEAT PUMP SYSTEM: ICE STORAGE (ACES).

The absolute present value analysis of each of the two systems reveals that the life cycle cost of the conventional system is \$10,684,269 and the heat pump system is \$7,011,518. Therefore the life cycle cost of the heat pump system is \$3,672,751 less than a conventional system over the twenty year life of the system. This analysis includes a salvage value of \$850,000 for the heat pump system to account for the 40 year life of the ice storage tank, while the salvage value of the conventional system is 0, because the equipment has a twenty year life. The salvage value is added in at the end of 20th year and discounted back to year 0.

The relative present value analysis of the conventional vs. the heat pump system describes the discounted cash flow of savings vs. incremental capital cost. This indicates that the accumulated discounted savings equals the incremental capital cost of the system in just under 9 years. Over 20 years, the present value of the savings vs. the incremental costs equals \$3,672,751, including the discounted salvage value of \$182,366. This clearly indicates that the heat pump system has a life cycle cost that is \$3,672,751 less than a conventional system and it pays for the incremental first cost in just under 9 years.

ECONOMIC ANALYSIS PROGRAM

Discounted Payback

This program generates a discounted cash flow analysis of an energy conservation measure, based on following data: Capital cost of the measure, operation savings in year 0, fuel cost escalation rate, and the discount rate. What is produced is a year by year cash flow for the entire payback period. The following formula is used for the analysis:

$$C = \sum_{X=1}^N S \frac{(1 + i_F)^X}{(1 + i_D)^X}$$

Where:

S = Operation Energy Savings In Year 0.

= Fuel Cost Escalation Rate

= Discount Rate

N = Years To Payback

C = Incremental Capital Cost Vs.

Conventional

The analysis assumes the following:

- (1) Savings in year 1 are escalated and discounted.
- (2) All savings go to amortizing the capital cost.
- (3) Fuel cost escalation rate remains constant through payback period.
- (4) Operation energy savings are relative conventional system.

The program allows energy conservation measures of different magnitudes to be compared.

Symbol Definitions:

- 1) YEAR - Previous year
- 2) SVNG - Escalated and discounted energy operation savings.
- 3) PVTL - Accumulated present value of savings.
- 4) PV PACKBACK - Present value payback.

FUEL ESCALATION  
 12.00 %  
 DISCOUNT RATE  
 8.00 %  
 OPERATION SAVINGS  
 67241.00  
 CAPITAL COST  
 2650000.00

\*RUN

1.00	YEAR	11.00	YEAR	21.00	YEAR
69731.41	SVNG	100316.21	SVNG	144315.76	SVNG
69731.41	PVTL	926105.79	PVTL	2158093.41	PVTL
2.00	YEAR	12.00	YEAR	22.00	YEAR
72314.05	SVNG	104031.62	SVNG	149660.79	SVNG
142045.46	PVTL	1030137.42	PVTL	2307754.20	PVTL
3.00	YEAR	13.00	YEAR	23.00	YEAR
74992.35	SVNG	107884.65	SVNG	155203.79	SVNG
217037.81	PVTL	1138022.06	PVTL	2462957.98	PVTL
4.00	YEAR	14.00	YEAR	24.00	YEAR
77769.84	SVNG	111880.37	SVNG	160952.07	SVNG
294807.65	PVTL	1249902.43	PVTL	2623910.06	PVTL
5.00	YEAR	15.00	YEAR	25.00	YEAR
80650.21	SVNG	116024.09	SVNG	166913.26	SVNG
375457.86	PVTL	1365926.52	PVTL	2790823.32	PVTL
6.00	YEAR	16.00	YEAR	PV PAYBACK IN-	
83637.25	SVNG	120321.28	SVNG	25.00	YRS.
459095.12	PVTL	1486247.80	PVTL		
7.00	YEAR	17.00	YEAR		
86734.93	SVNG	124777.62	SVNG		
545830.05	PVTL	1611025.42	PVTL		
8.00	YEAR	18.00	YEAR		
89947.34	SVNG	129399.02	SVNG		
635777.38	PVTL	1740424.44	PVTL		
9.00	YEAR	19.00	YEAR		
93278.72	SVNG	134191.57	SVNG		
729056.10	PVTL	1874616.01	PVTL		
10.00	YEAR	20.00	YEAR		
96733.49	SVNG	139161.63	SVNG		
825789.59	PVTL	2013777.64	PVTL		

SOLAR ENERGY ANNUAL STORAGE

FUEL ESCALATION  
12.00 %  
DISCOUNT RATE  
8.00 %  
OPERATION SAVINGS  
175710.00  
CAPITAL COST  
9050000.00

#RUN

1.00	YEAR	11.00	YEAR	21.00	YEAR
182217.78	SVNG	262140.07	SVNG	377116.98	SVNG
182217.78	PVTL	2420042.07	PVTL	5639395.50	PVTL
2.00	YEAR	12.00	YEAR	22.00	YEAR
188966.58	SVNG	271848.97	SVNG	391084.28	SVNG
371184.36	PVTL	2691891.04	PVTL	6030479.77	PVTL
3.00	YEAR	13.00	YEAR	23.00	YEAR
195965.35	SVNG	281917.45	SVNG	405568.88	SVNG
567149.71	PVTL	2973808.48	PVTL	6436048.65	PVTL
4.00	YEAR	14.00	YEAR	24.00	YEAR
203223.32	SVNG	292358.83	SVNG	420589.95	SVNG
770373.03	PVTL	3266167.32	PVTL	6856638.60	PVTL
5.00	YEAR	15.00	YEAR	25.00	YEAR
210750.11	SVNG	303186.94	SVNG	436167.36	SVNG
981123.14	PVTL	3569354.26	PVTL	7292805.96	PVTL
6.00	YEAR	16.00	YEAR	26.00	YEAR
218555.67	SVNG	314416.08	SVNG	452321.70	SVNG
1199678.82	PVTL	3883770.34	PVTL	7745127.66	PVTL
7.00	YEAR	17.00	YEAR	27.00	YEAR
226650.33	SVNG	326061.12	SVNG	469074.36	SVNG
1426329.14	PVTL	4209831.46	PVTL	8214202.02	PVTL
8.00	YEAR	18.00	YEAR	28.00	YEAR
235044.78	SVNG	336137.46	SVNG	486447.48	SVNG
1661373.93	PVTL	4547963.92	PVTL	8700649.50	PVTL
9.00	YEAR	19.00	YEAR	29.00	YEAR
243750.15	SVNG	350661.07	SVNG	504464.06	SVNG
1905124.07	PVTL	4898630.00	PVTL	9205113.56	PVTL
10.00	YEAR	20.00	YEAR	29.00	YEAR
252777.93	SVNG	363648.52	SVNG	522911.00	SVNG
2222111.93	PVTL	5322222.00	PVTL	9205113.56	PVTL

PM PAYBACK IN-  
29.00 YRS.

FUEL ESCALATION  
12.00 %  
DISCOUNT RATE  
8.00 %  
OPERATION SAVINGS  
178318.00  
CAPITAL COST  
1850000.00

*RUN		
	1.00	YEAR
	184922.37	SVNG
	184922.37	PVTL
	2.00	YEAR
	191771.35	SVNG
	376693.72	PVTL
	3.00	YEAR
	198873.99	SVNG
	575567.71	PVTL
	4.00	YEAR
	206239.69	SVNG
	781807.40	PVTL
	5.00	YEAR
	213878.20	SVNG
	995685.60	PVTL
	6.00	YEAR
	221799.61	SVNG
	1217485.21	PVTL
	7.00	YEAR
	230014.42	SVNG
	1447499.63	PVTL
	8.00	YEAR
	238533.47	SVNG
	1886033.10	PVTL
	9.00	YEAR
	247368.04	SVNG
	1933401.14	PVTL
PV PAYBACK IN-	9.00	YRS.

OUTSIDE AIR POTENTIAL FOR ICE GENERATION  
IN ADDITION TO SYSTEM (3)

FUEL ESCALATION  
12.00 %  
DISCOUNT RATE  
8.00 %  
OPERATION SAVINGS  
202330.00  
CAPITAL COST  
3002000.00

\*RUN

1.00	YEAR	11.00	YEAR
209823.70	SVNG	301854.20	SVNG
209823.70	PVTL	2786677.55	PVTL
2.00	YEAR	12.00	YEAR
217594.95	SVNG	313033.98	SVNG
427418.66	PVTL	3099711.54	PVTL
3.00	YEAR	PV PAYBACK IN-	
225654.02	SVNG	12.00	YRS.
653072.68	PVTL		
4.00	YEAR		
234011.58	SVNG		
887084.26	PVTL		
5.00	YEAR		
242678.68	SVNG		
1129762.94	PVTL		
6.00	YEAR		
251666.78	SVNG		
1381429.71	PVTL		
7.00	YEAR		
260987.77	SVNG		
1642417.48	PVTL		
8.00	YEAR		
270653.98	SVNG		
1913071.46	PVTL		
9.00	YEAR		
280678.20	SVNG		
2193749.66	PVTL		
10.00	YEAR		
291073.69	SVNG		
2484823.35	PVTL		

COMPATATIVE ANALYSIS 7 FOUR CANDIDATE SYSTEMS

	Basic Conventional System Boilers & Chillers	Heat Pump 1. System with Slab Storage	Solar 2. Energy Annual Storage	Heat Pump 3. Generate Ice Storage ACFS	4. Outside Air Potential for Ice Making in Addition to System 3.	
Annual Heating Energy DIW	$4.032 \times 10^{10}$ BTU	$4.032 \times 10^{10}$ BTU	$4.032 \times 10^{10}$ BTU	$4.032 \times 10^{10}$ BTU	$4.032 \times 10^{10}$ BTU	
Requirement Cooling	$3.0 \times 10^6$ TON H	$3.0 \times 10^6$ TON H	$3.0 \times 10^6$ TON H	$3.0 \times 10^6$ TON H	$3.0 \times 10^6$ TON H	
Energy Stored	0	$7.5 \times 10^8$ BTU	$1.684 \times 10^{10}$ BTU	2,114,000 TON H	$3.0 \times 10^6$ TON H	
Energy Saved	0	$7.5 \times 10^8$ BTU	$4.032 \times 10^{10}$ BTU	$2.5379 \times 10^{10}$ BTU	$3.6 \times 10^{10}$ BTU	
Energy Consumption	W	$17 \times 10^6$ KWH-T	$3.447 \times 10^6$ KWH	106,800 KWH	3,450,000 KWH	3,368,370 KWH
	S	$3 \times 10^6$ KWH	$3.013 \times 10^6$ KWH	3,105,200 KWH	1,252,870 KWH	367,770 KWH
	Total	$20 \times 10^6$ KWH	$6.462 \times 10^6$ KWH	3,212,000 KWH	4,502,870 KWH	3,736,140 KWH
Demand Reduction	0	0	+82 KWH-6 months	-3,000 KWH-4.5 mo.	-3,000 KWH-6 mo.	
Operation Annual Energy Cost	\$ 331,710	\$ 264,470	\$ 156,000	\$ 153,392	\$ 129,380	
Operation Annual Saving vs Conventional	0	\$ 67,241	\$ 175,710	\$ 178,318	\$ 202,330	
System Capital Cost	\$750,000	\$3,400,000	\$9,800,000	\$2,600,000	\$5,752,000	
Incremental Investment vs Conventional	0	\$2,650,000	\$9,050,000	\$1,850,000	\$3,002,000	
Discounted Present Worth, PB	0	25 years	29 years	9 years	12 years	
Simple Payback	0	40 years	52 years	11 years	15 years	

D A T A

DC - GT

SCHEDULE OF MONTHLY CHARGES -

	<u>SUMMER MONTHS</u>	<u>WINTER MONTHS</u>
A. Customer Charge	\$210.00 per month	\$210.00 per month
B. Energy Charge		
On Peak Period	1.120¢ per kwhr	0.910¢ per kwhr
Intermediate Period	0.910¢ per kwhr	0.810¢ per kwhr
Off Peak Period	0.405¢ per kwhr	0.405¢ per kwhr
C. Demand Charge		
On Peak Period Billing Demand	\$5.25 per kw	\$3.15 per kw
Intermediate Period Billing Demand	\$3.15 per kw	\$2.40 per kw
Off Peak Period Billing Demand	\$2.40 per kw	\$2.40 per kw
D. Minimum Charge - The Customer Charge		

SEASON DESIGNATION -

SUMMER MONTHS, for purposes of application of this rate schedule, are the billing months of June through October, and WINTER MONTHS are the billing months of January through May, plus November and December.

RATING PERIODS -

Weekdays - (Excluding Holidays)

On Peak Period	12:00 noon	to	8:00 p.m.
Intermediate Period	8:00 a.m.	to	12:00 noon
	and	8:00 p.m.	to 12:00 midnight
Off Peak Period	12:00 midnight	to	8:00 a.m.

Saturdays, Sundays and Holidays

Off Peak Period All Hours

Holidays

New Year's Day, Washington's Birthday, Memorial Day, Independence Day, Labor Day, Columbus Day, Veterans Day, Thanksgiving Day, Christmas Day.

BILLING DEMANDS -

The billing demands shall be the maximum 30 minute demands recorded during each rating period of the billing month.

Date of Issue:

Date Effective:

Issued by Frank S. Walters, Vice President  
1900 Pennsylvania Avenue, N. W.  
Washington, D. C. 20068

POTOMAC ELECTRIC POWER COMPANY

SCHEDULE OF MONTHLY CHARGES - (Including 1.45¢ of Fuel Adjustment)

	<u>SUMMER MONTHS</u>	<u>WINTER MONTHS</u>
A. Customer Charge	\$210.00 per month	\$210.00 per month
B. Energy Charge		
On Peak Period	2.570¢ per kwhr	2.360¢ per kwhr
Intermediate Period	2.360¢ per kwhr	2.260¢ per kwhr
Off Peak Period	1.855¢ per kwhr	1.855¢ per kwhr
C. Demand Charge		
On Peak Period Billing Demand	\$5.25 per kw	\$3.15 per kw
Intermediate Period Billing Demand	\$3.15 per kw	\$2.40 per kw
Off Peak Period Billing Demand	\$2.40 per kw	\$2.40 per kw
D. Minimum Charge - The Customer Charge		





Energy Estimating Methods

$$E = (KW) (F)/S \quad (2)$$

where

- E = energy required for auxiliaries, kilowatt hours.
- KW = connected load of fans or pumps.
- F = fuel estimate for annual heating season.
- S = size of heating unit in fuel units input.

The above procedure is only valid for intermittent fan and/or pump operation of fossil fueled systems. For continuous fan operation, the energy use is simply the connected load times the hours of operation. For electric resistance systems, the auxiliary energy is included in the modified degree day calculation.

Estimating Electrical Energy

These procedures are only applicable to resistance type heating devices such as baseboard, electric furnaces, and electric boilers. Eq 2 has been used for heat pumps with the addition of a "seasonal performance factor", but the estimate of this factor, lacking comparable building data, is a hazardous procedure. The bin method calculation presented later is recommended for heat pumps. For other electric heat systems, Eq 1 applies.

*Example 3:* Estimate the energy requirements to heat a residence in Washington, D.C. The design heat loss is 65,000 Btuh based on 10 F outdoor and 70 F indoor.

Solution: From Eq 1:

$$E = \left( \frac{65,000 \times 4224 \times 24}{60 \times 1.0 \times 3413} \right) (0.79) \quad (1.0)$$

$$E = 25,420 \text{ kwh}$$

Energy Conservation Effects

No simple, valid procedures have been developed for evaluating the effects of suggested residential energy conservation practices. Those suggestions that reduce heat loss per degree difference, such as storm windows or increased insulation, may be approximated using the modified degree day procedure. Several studies indicate that reduction of thermostat set point may reduce fuel consumption by approximately 3% per degree. Reducing appliance and lighting usage will lower energy requirements but increase heating needs in winter. Evaluation of other suggestions is not yet on a base sufficiently valid to draw quantitative conclusions.

EQUIVALENT FULL-LOAD HOURS

This procedure simply consists of using an estimate, based on local experience, of the ratio of annual cooling energy requirements to rated energy input of the cooling equipment.

Table 4 . . . Approximate Power Inputs

System	Compressor Kw/Design Ton	Auxiliaries Kw/Design Ton
Window Units	1.46	0.32
Through-Wall Units	1.64	0.30
Dwelling Unit, Central Air-Cooled	1.49	0.14
Central, Group, or Bldg. Cooling Plants		
(3 to 25 tons) Air-Cooled	1.20	0.20
(25 to 100 tons) Air-Cooled	1.18	0.21
(25 to 100 tons) Water-Cooled	0.94	0.17
(Over 100 tons) Water-Cooled	0.79	0.20

The operating cost of cooling equipment during a particular summer depends of variables such as the amounts of sunshine and rain, the number of abnormally hot or cool days, the efficiency of the equipment, and the local power rate. It is also influenced by human factors, such as operation of equipment only during the hottest weather, opening windows at night, and difference in preferred indoor temperatures. Nevertheless, it is important that lending agencies and prospective buyers of equipment be given a reasonably accurate estimate of the operating cost during normal summer weather and under usual operating conditions. Adjustments can then be made for any special conditions anticipated. The approximate electrical power inputs for the various motorized components in mechanical cycle air conditioners are shown in Table 4.

Energy cost per hour can be estimated by multiplying the estimated power per ton, the cooling capacity in tons and the cost per kilowatt-hour. Thus, the estimated cost per hour for a central 3-ton air conditioner with an air-cooled condenser will be 1.63 kw × 3 tons × the electric rate. It is essential to use the correct step of the utility residential rate structure to get a good estimate. The basis of the method used requires the use of a table of the estimated annual hours of operation for properly sized equipment in typical cities. See Table 5.

The values in Table 5 are estimated ranges based on a survey of electric utility companies and are based on an indoor temperature of 75 F. In general, residential units will be toward the lower end of the range, and light commercial toward the higher.

It must be impressed on the buyer that the energy consumption arrived at by this method is purely an estimate, as it takes into consideration no abnormally hot weather or the buyer's preference for lower indoor temperatures.

Water usage is another important factor with water-cooled equipment. Various manufacturers have published water usage data for their equipment at varying summer water temperatures. Representative water consumption values are given in Table 6.

Table 5 . . . Estimated Equivalent Rated Full-Load Hours of Operation for Properly Sized Equipment During Normal Cooling Season

Albuquerque, NM	800-2200	Indianapolis, IN	600-1600
Atlantic City, NJ	500-800	Little Rock, AR	1400-2400
Birmingham, AL	1200-2200	Minneapolis, MN	400-800
Boston, MA	400-1200	New Orleans, LA	1400-2800
Burlington, VT	200-600	New York, NY	500-1000
Charlotte, NC	700-1100	Newark, NJ	400-900
Chicago, IL	500-1000	Oklahoma City, OK	1100-2000
Cleveland, OH	400-800	Pittsburgh, PA	900-1200
Cincinnati, OH	1000-1500	Rapid City, SD	800-1000
Columbia, SC	1200-1400	St. Joseph, MO	1000-1600
Corpus Christi, TX	2000-2500	St. Petersburg, FL	1500-2700
Dallas, TX	1200-1600	San Diego, CA	800-1700
Denver, CO	400-800	Savannah, GA	1200-1400
Des Moines, IA	600-1000	Seattle, WA	400-1200
Detroit, MI	700-1000	Syracuse, NY	200-1000
Duluth, MN	500-500	Trenton, NJ	800-1000
El Paso, TX	1000-1400	Tulsa, OK	1500-2200
Honolulu, HI	1500-3500	Washington, DC	700-1200

Materials	Melting Point		Heat of Fusion		Heat Capacity	
	°F	°C	Btu/lb	kJ/kg	Btu/ft <sup>3</sup>	kJ/m <sup>3</sup>
a) <u>Salt Hydrates</u>						
Na <sub>2</sub> SO <sub>4</sub> · 1/2NH <sub>4</sub> Cl · 1/2NaCl · 10H <sub>2</sub> O	55	12.8	78	181	7200	268,265
K <sub>2</sub> HPO <sub>4</sub> · 6H <sub>2</sub> O	52-56	11.1-13.3	47	109	4900	182,570
Ca(NO <sub>3</sub> ) <sub>2</sub> · 4H <sub>2</sub> O	117	47.2	66	154	7650	285,032
Na <sub>2</sub> S <sub>2</sub> O <sub>3</sub> · 5H <sub>2</sub> O	113-120	45.0-48.9	90	209	9200	342,784
Na <sub>2</sub> SO <sub>4</sub> · 10H <sub>2</sub> O (Glauber Salt)	90	32.2	108	251	9900	368,365
MgCl <sub>2</sub> · 6H <sub>2</sub> O	239	115	71	165	6940	258,578
b) <u>Waxes</u>						
C14-C16 Paraffin	35-45	1.7-7.2	65.4	152	3185	118,670
C15-C16 Paraffin	40-50	4.4-10.0	65.7	153	3200	119,229
1 - Decanol	40-45	4.4-7.2	88.6	206	4590	171,019
C14 Paraffin	35-40	1.7-4.4	71.1	165	3420	127,426
C16 Paraffin	58-65	14.4-18.3	86.2	200	4190	156,116
P116 Paraffin	116	46.7	90	209	4380	163,195
c) <u>Plastics</u>						
High Density Polyethylene	230-255	110-123.9	108	251	7200	268,266

TABLE 3-2, PHASE CHANGE THERMAL ENERGY STORAGE MATERIALS

21. SOURCES OF INFORMATION

- 20 - 1. PENNSYLVANIA AVENUE ENERGY CONSERVATION AND ALTERNATIVE ENERGY SOURCE CONCEPTUAL PLAN - Dubin Bloome Associates, July 1977
- 20 - 2. SEASONAL STORAGE IN DISTRICT HEATING - by General Electric Company Tempo Center for Advanced Studies, Santa Barbara, California - District Heating July - 1977
- 20 - 3. Summary of Annual Cycle Energy Systems Workshop I held October 29-30, 1975 at Oak Ridge, Tennessee, by H. C. Fisher ORNL/TM-t243, July 1976.
- 20 - 4. Design Report for the ACES DEMONSTRATION HOUSE, Oak Ridge National Laboratory, by E. C. Hise, ORNL/CON-1, October 1976
- 20 - 5. THE ANNUAL CYCLE ENERGY SYSTEM, Initial Investigations, Oak Ridge National Laboratory, ORNL/TM-5525, October 1976
- 20 - 6. THERMAL ENERGY STORAGE UNIT FOR AIR CONDITIONING SYSTEMS USING PHASE CHANGE MATERIAL, by James C. Dudley, University of Pennsylvania, August 1972, Report No. NSF/RANN/SE/GI27976/TO7218
- 20 - 7. LATENT HEAT AND SENSIBLE HEAT STORAGE FOR SOLAR HEATING SYSTEMS, by Harold G. Lorsch, University of Pennsylvania, December 1972, Report No. NSF/RANN/SE/GI22976/TR 72/20
- 20 - 8. CONGRUENTLY MELTING MATERIALS FOR THERMAL ENERGY STORAGE IN AIR CONDITIONERS, by Kenneth Kauffman, Yen Chi Pan University of Pennsylvania, June 1973 Report No. NSF/Rann/SE/GI27976/TR 73/5

- 20 - 9. THE DEVELOPMENT AND TESTING OF A SINGLE PLATE AND TWO PLATE ICE MAKER HEAT PUMP, by H. C. Fischer
- 20 - 10. Performance Report for the ACES DEMONSTRATION HOUSE, August 1976 through August 1977, ORNL/CON-19.
- 20 - 11. THE ANNUAL CYCLE ENERGY SYSTEM, A HYBRID HEAT PUMP CYCLE, Richard A. Buhl, Ashrae Journal, July 1977.
- 20 - 12. Report No. UCB-ME 26-1, January 1976, A SOLAR HEATING SYSTEM, University of California, Berkeley.
- 20 - 13. HUD Intermediate Minimum Property Standards Supplement, 1977 Edition
- 20 - 14. U.S.A. Climatic Atlas
- 20 - 15. ASHRAE Guides
- 20 - 16. Manufacturers Data  
Dunham - Bush Screw Compressors  
Carrier Multistage Centifugal Compressors  
Turbo Refrigeration Company, Ice Makers  
Sunwork Collector Co.
- 20 - 17. Adviser Henry C. Fisher  
Energy Management Consulting, Florida