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HEAT PUMP CENTERED INTEGRATED COMMUNITY ENERGY SYSTEMS

System Development

Dubin-Bloome Associates Interim Report

ARGONNE NATIONAL LABORATORY

ENERGY AND ENVIRONMENTAL SYSTEMS
DIVISION

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HEAT PUMP CENTERED
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TABLE OF CONTENTS

PAGE

LIST OF FIGURES	<i>v</i>
LIST OF TABLES	<i>ix</i>
PREFACE	<i>xi</i>
ABSTRACT	<i>xii</i>
 I. SYSTEM DESCRIPTION	 1
1.1 SYSTEM CONCEPTUAL DESCRIPTION	1
1.2 OVERALL COEFFICIENT OF PERFORMANCE	12
1.3 OPTIMUM TEMPERATURE RANGE FOR COMPRESSOR OPERATION	17
1.4 REFRIGERANT INVESTIGATION	21
1.5 COMPRESSOR INVESTIGATION	28
1.5.1 PART LOAD COMPRESSOR PERFORMANCE	36
1.6 ANALYSIS OF STORAGE MEDIA PROPERTIES	39
1.7 ICE GENERATING METHODS	41
1.8 COMPONENT IDENTIFICATION	42
1.9 HP-ICES MODES OF OPERATION	45
 2. POTENTIAL APPLICATIONS	 55
2.1 INTRODUCTION	55
2.2 THE ENERGY BALANCE FOR HP-ICES	57
2.2.1 SENSITIVITY TO CLIMATE	57
2.3 COMMUNITY COMPOSITION	75
2.4 POTENTIAL ENERGY INPUTS FOR ENERGY REQUIREMENTS	77
2.5 MARKET AVAILABILITY FOR ICE MAKING EQUIPMENT	88
2.6 SYSTEM RELIABILITY	91
2.7 ECONOMICS	91
2.8 SYSTEM SIZE	92
2.9 PROJECTIONS OF ENERGY SAVINGS FOR THE YEAR 2000	94
 3. EXPECTED PERFORMANCE	 101
3.1 FIRST AND SECOND LAW ANALYSIS	101
3.1.1 THE FIRST LAW ENERGY BALANCE	101
3.1.2 SECOND LAW ANALYSIS	107
3.2 PERFORMANCE OF HP-ICES VS. A CONVENTIONAL SYSTEM (CASE STUDY)	114
3.2.1 SELECTION OF CONVENTIONAL SYSTEM (WASH.D.C.)	114
3.2.2 THE PERFORMANCE OF ICE GENERATING HP-ICES (WASH. D.C.)	115
3.2.3 RECAPITULATION OF RESULTS	121
3.2.4 THE ANNUAL OVERALL COP FOR THE HP-ICES	122
 4. EXPECTED ECONOMICS	 124
4.1 INTRODUCTION	124
4.2 ANALYSIS	124
4.2.1 ENERGY COST FOR THE CASE STUDY	125
4.2.2 FIRST COST	127
4.2.3 OPERATING COST	128
4.2.4 COMPARITIVE ANALYSIS	130
4.2.5 COST PER MILLION BTU.	131
4.3 CONCLUSION	132

5. EXPECTED ENVIRONMENTAL IMPACTS	133
5.1 AIR POLLUTION	133
5.2 WATER POLLUTION	135
5.3 NOISE POLLUTION	135
5.4 RESOURCE CONSUMPTION	136
6. PROJECTED GROWTH	138
7. SYSTEM MODIFICATION	141
7.1 SYSTEM MODIFICATION	141
7.2 MODIFICATION FOR SUPPLEMENTARY HEAT	141
7.3 MODIFICATION FOR SUPPLEMENTARY COOLING	142
8. COMPONENT TESTING REQUIREMENTS	148
8.1 PRE DEMONSTRATION TEST	148
8.2 TESTING PROCEDURES	150
APPENDIX	154
APPENDIX 1A	155
1A.1 REFRIGERANT INVESTIGATION	155
1A.2 REFRIGERATION COMPRESSOR PERFORMANCE	163
APPENDIX 2A	173
2A.1 HEATING AND COOLING ANNUAL ENERGY BALANCE	174
2A.2 MARKET SQUARE COMPLEX IN WASHINGTON D.C. (CASE STUDY)	178
2A.3 PROJECTION OF ENERGY SAVING. FOR THE YEAR 2000 BY USE OF HP-ICES	189
APPENDIX 4A	200
4A. ECONOMIC ANALYSIS PROGRAM	201
APPENDIX 7A	208
7A.1 EVALUATION OF THE ABILITY OF OUTSIDE AIR IN WINTER TO PRODUCE SUPPLEMENTARY ICE	209
7A.2 MODIFIED ACES SYSTEM WITH SOLAR RADIATION FOR DONALDSON CORPORATION, MINN.	213
7A.3 DIESEL ENGINE DRIVEN ICE GENERATING HP-ICES	218
7A.4 GAS ENGINE DRIVEN ICE GENERATING HP-ICES	221
BIBLIOGRAPHY	225

<u>LIST OF FIGURES</u>	<u>PAGE</u>
1.1 ICE GENERATING HEAT PUMP-ICES -ISOMETRIC	2
1.2 ICE GENERATING HEAT PUMP-ICES -SCHEMATIC	5
1.3 AUTOMATIC ICE MAKER	7
1.4 VARIATION OF COEFFICIENTS OF PERFORMANCE FOR HEATING WITH SATURATED DISCHARGE TEMPERATURE (SDT)	16
1.5 SCREW COMPRESSOR SLIDE VALVE - FULL LOAD	36
1.6 SCREW COMPRESSOR SLIDE VALVE - PART LOAD	36
1.7 TYPICAL PART LOAD POWER INPUT CURVE -ROTARY SCREW COMPRESSOR	37
1.8 CENTRIFUGAL COMPRESSOR - PART LOAD PERFORMANCE	38
1.9 BASIC WINTER MODE	48
1.10 SEEOND WINTER MODE	49
1.11 BASIC SUMMER MODE	50
1.12 SECOND WINTER MODE	51
1.13 THIRD SUMMER MODE (NIGHT-TIME)	52
1.14 FOURTH SUMMER MODE	54
2.1 ANNUAL HEATING DEGREE DAYS	66
2.2 ANNUAL DEGREE HOURS	67
2.3 ANNUAL ENERGY BALANCE	70
2.4 BOSTON ENERGY REQUIRED PER SECTORS - RESIDENTIAL, OFFICE, RETAIL	79
2.5 MINNEAPOLIS ENERGY REQUIRED PER SECTORS	80
2.6 SEATTLE ENERGY REQUIRED PER SECTORS	80
2.7 OKLAHOMA CITY ENERGY REQUIRED PER SECTORS	81
2.8 ELECTRICAL ENERGY INPUT REQUIRED TO OBTAIN:	82
1. Heating and Cooling	
2. Heating Only	
3. Cooling Only	

HP-ICES PROJECT

2.9	ENERGY COST FOR HEATING AND COOLING AS A FUNCTION OF ELECTRICITY COST	83
2.10	ENERGY COST FOR COOLING AS A FUNCTION OF ELECTRICITY COST	84
2.11	ENERGY COST FOR HEATING AS A FUNCTION OF ELECTRICITY COST	85
2.12	PERCENT OF POPULATION IN METROPOLITAN AREAS	99
2.13	CLIMATIC INFLUENCES IN GEOGRAPHIC DIVISIONS	100
3.1	FIRST LAW ANALYSIS BALANCE FOR HP-ICES SUMMER MODE	103
3.2	FIRST LAW ENERGY BALANCE ICE GENERATION HP-ICES WINTER MODE - BASED ON 12000 BTU/HR WATER	104
3.3	FIRST LAW ENERGY BALANCE FOR CONVENTIONAL BOILER SYSTEM WINTER MODE	105
3.4	FIRST LAW ENERGY BALANCE FOR CONVENTIONAL CHILLER SUMMER MODE	106
3.5	AVAILABILITY ANALYSIS ICE GENERATION HP-ICES BASED ON 12,000 BTU/HR WATER HEAT OF FUSION WINTER MODE	110
3.6	AVAILABILITY ANALYSIS ICE GENERATION HP-ICES SUMMER MODE	111
3.7	AVAILABILITY ANALYSIS OF CONVENTIONAL BOILER WINTER MODE	112
3.8	AVAILABILITY ANALYSIS FOR CONVENTIONAL CHILLER BASED ON 12,000 BTU/HR CHILLED WATER SYSTEM SUMMER MODE	113
7.1	HEATING AND COOLING OBTAINED BY VARIOUS SYSTEMS	145
7.2	HEATING AND COOLING OBTAINED BY VARIOUS SYSTEMS	146
1A.1	PRESSURE ENTHALPY DIAGRAM	156
1A.2	TWO STAGE COMPRESSION WITH WATER INTERCOOLING	156
1A.3	TWO STAGE COMPRESSION WITH LIQUID REFRIGERANT	158
1A.4 -1A.6	DUNHAM BUSH SCREW COMPRESSOR PERFORMANCE CHART	168-170
1A.7 -1A.8	QUICK SELECTION CHART CARRIER CENTRIFUGAL MPS COMPRESSOR	171-172

HP-ICES PROJECT

2A-1	HOUSING PLAN, MARKET SQUARE, WASHINGTON, DC	180
2A-2	HOUSING SECTION, MARKET SQUARE, WASHINGTON	179
4A-1	PRESENT VALUE ANALYSIS CONVENTIONAL SYSTEM BOILER AND CHILLERS	204
4A-2	PRESENT VALUE ANALYSIS HEAT PUMP SYSTEM ICE STORAGE HP-ICES	205
4A-3	RELATIVE PRESENT VALUE ANALYSIS CONVENTIONAL VS HEAT PUMP SYSTEM ICE STORAGE HP-ICES	207
7A-1	ICE STORAGE USING OUTSIDE AIR	211
7A-2	DUAL FUEL (NATURAL GAS OR OIL #2) ENGINE DRIVE ICE GENERATING HP-ICES	223
7A-3	ANNUAL OVERALL COP OF DIESEL ENGINE GAS ENGINE AND ELECTRIC DRIVE HEAT PUMP	224

<u>LIST OF TABLES</u>	<u>PAGES</u>
1.1 HOT WATER TEMPERATURES	19
1.2 SUMMARY OF ONE STAGE COMPRESSION PERFORMANCE FOR EVAPORATOR TEMPERATURE OF 20°F AND CONDENSER TEMPERATURE OF 130°F	25
1.3 SUMMARY OF TWO STAGE COMPRESSION PERFORMANCE FOR EVAPORATOR TEMPERATURE OF 20°F AND CONDENSER TEMPERATURE OF 130°F	26
1.4 COMPARISON OF PERFORMANCE OF SCREW COMPRESSORS (TYPE E(HP) X 2516 DUNHAM-BUSH USING R-12 AND R-22 REFRIGERANTS	29
1.5 PERFORMANCE OF MPS SIZE 40 CARRIER CENTRIFUGAL COMPRESSOR WITH R-12 AND R-22 AT 20°F AND 35°F SUCTION TEMPERATURES.	33
1.6 COMPARISON OF PERFORMANCE OF CARRIER MPS SIZE 40 CENTRIFUGAL COMPRESSOR VA 2516 DUNHAM-BUSH SCREW COMPRESSOR USING REFRIGERANT - 12	34
1.7 STORAGE MEDIA PROPERTIES	39
2.1 DESIGN CONDITIONS FOR SELECTED LOCATIONS	59
2.2 ANNUAL ENERGY REQUIREMENTS FOR COOLING	60
2.3 ANNUAL ENERGY REQUIREMENTS FOR HEATING	61
2.4 ANNUAL ENERGY REQUIREMENTS FOR DHW	62
2.5 HEATING AND COOLING ANNUAL ENERGY BALANCED	63
2.6 SUPPLEMENTARY COOLING AND HEATING REQUIREMENT	64
2.7 DHW AS PERCENTAGE OF THE TOTAL HEATING REQUIREMENT	73-74
2.8 ANNUAL ENERGY REQUIREMENT FOR RESIDENTIAL AND COMMERCIAL SECTORS	78
2.9 SCREW COMPRESSORS TONNAGE HOURS POWER AND COST	90
3.1 ICE BIN LOSSES	120
3.2 ENERGY INPUT (CASE STUDY WASHINGTON D.C.)	121
3.3 SUMMARY OF ANNUAL ENERGY INPUT	122
4.1 RECAPITULATION OF ECONOMICS	129

4.2	PAYBACK SENSITIVITY WITH ESCALATION RATE AND COST OF MONEY	130
4.3	OWNING AND OPERATING COST PER MILLION BTU	132
1A.1	RECAPITULATION OF PERFORMANCE OF R-12 PER ONE TON REFRIGERATION	159
1A.2	SUMMARY OF TWO STAGE COMPRESSION PERFORMANCE FOR EVAPORATOR TEMPERATURE OF 20°F AND CONDENSER TEMPERATURE OF 130°F	162
1A.3	COMPARISON OF PERFORMANCE OF SCREW COMPRESSORS TYPE E(HP X 2516) DUNHAM-BUSH USING R-12 AND R-22 REFRIGERANTS 20°F SST AND 130°F SDT	163
1A.4	COMPARISON OF PERFORMANCE OF SCREW COMPRESSORS TYPE E(HP X 2516) DUNHAM-BUSH USING R-12 AND R-22 REFRIGERANTS 35°F SST AND 105°F SDT	164
1A.5	REFRIGERANTS-12 SUCTION TEMPERATURE 20°F DISCHARGE TEMPERATURE 120°F COMPARISON OF PERFORMANCE OF CARRIER MPS SIZE 40 CENTRIFUGAL COMPRESSOR VS. 2516 DUNHAM-BUSH SCREW COMPRESSOR	165
1A.6	REFRIGERANT-12 SST 35°F SDT 105°F COMPARISON OF PERFORMANCE OF CARRIER MPS SIZE 40 CENTRIFUGAL COMPRESSOR VS. 2516 DUNHAM-BUSH SCREW COMPRESSOR	166
1A.7	PERFORMANCE OF MPS SOZE 40 CARRIER CENTRIFUGAL COMPRESSOR WITH R-12 AND R-22 AT 20° AND 35°F SUCTION TEMPERATURE	167
2A.1	HEATING AND COOLING ANNUAL ENERGY BALANCE	176
2A.2	WINTER AND SUMMER DOMESTIC HOT WATER ENERGY REQUIREMENTS AND COOLING CPAPCITY OF HEAT PUMP	177
2A.3	HEAT TRANSFER COEFFICIENTS FOR SELECTED LOCATIONS IN THE UNITED STATES	184
2A.4	PEAK HEATING LOAD	185
2A.5	PEAK COOLING LOAD	186
2A.6	DOMESTIC HOT WATER ENERGY REQUIREMENTS	187
2A.7	DETERMINATION OF SUPPLEMENTARY COMPRESSOR WORK, SUPPLEMENTARY HEATING & ICE STORAGE BIN SIZE	188
7A.1	ANNUAL ENERGY REQUIREMENTS FOR HEATING DHW AND COOLING	213

PREFACE

As the once popular perception of fuel being cheap and unlimited fades, increased emphasis is being given to more efficient use of energy resources and to the substitution of non-scarce energy forms for depleting and less abundant sources. Recognition of the need to conserve energy resources has been spurred in large part by the oil embargo of 1973 and subsequent escalation in fuel prices, by energy shortages in the harsh winter of 1976/77, by increasing electricity "brown-outs" in urban centers, and by the crippling effects of labor strikes and stoppages in energy supply industries. In recent years, development and acceptance of energy conserving systems which avoid dependence on scarce or interruptible fuels has increased markedly in the United States.

One comprehensive approach to energy conservation is the Integrated Community Energy Systems (ICES) concept, which is intended to increase the efficiency of the varied ways in which energy is provided to and utilized by a community. At the same time, the ICES approach seeks to reduce dependence on scarce fuels and other resources, to preserve environmental quality, to offer energy and energy consuming services in an economically attractive manner, and, perhaps most importantly, to meet the energy needs of communities without adversely affecting social and lifestyle objectives.

The ICES approach to meeting these goals is embodied in three levels of integration. First, by incorporating innovative technology to maximize the resource utilization efficiency, energy requirements can be reduced. Included in this method are cogeneration (or more correctly coproduction) and cascading uses of energy to minimize the thermodynamic mismatch of source energy qualities and actual energy needs. Also included is fuel substitution in centralized community-scale systems which would be impractical in independent, individual building and separate service energy systems. Second, by integrating the energy supply and energy consuming systems with the functional design and layout of the community, load management advantages can be achieved and distribution losses can be minimized. At the same time, resource needs can be reduced through appropriate land utilization and planned growth. And third, the community systems development is integrated with the financial and regulatory mechanisms common to communities to permit widespread implementation.

A specific ICES may consist of either a partial or complete integration of these approaches, as deemed necessary and appropriate to strike the desired balance among a community's economic, social, environmental, and energy conservation goals. An ICES can be applied to a total community as well as to portions of a community and the services provided need not be the same for all areas served. While ICES include a broad spectrum of technologies to meet energy service requirements, the ICES concept neither arbitrarily defines the energy services to be provided nor the type, size, or function of the area to be served. As a result, an ICES is tailored for each application and the determination of the kind and number of energy services provided and the size of the service area for each such service is based on an optimum combination of energy efficiencies, indigenous resource and labor supplies, economics, and environmental conditions.

The communities which ICES are intended to serve consist of multiple facilities in the residential, commercial, industrial, agricultural, recreational, or institutional sectors or in some combination of sectors.

Furthermore, these communities may be in various stages of development, e.g., planned, designed, constructed, or redevelopment.

Heat Pump Centered Integrated Community Energy Systems are energy systems for communities which provide heating, cooling, and/or other energy services through the use of heat pumps. Since heat pumps primarily transfer energy from existing and otherwise probably unused sources, rather than convert it from electrical or chemical to thermal form, HP-ICES are viewed as having significant potential for energy conservation. Furthermore, since conventional building heating and cooling systems would be replaced by this community energy system, nonscarce resources could be used instead of depleting fuels which are in short supply. This is accomplished by powering the heat pumps with nonscarce energy forms not practical for use in the smaller conventional systems. Secondary benefits expected to enhance the value of such systems include reduction of adverse environmental effects over conventional systems, reliable production of services in view of increasingly frequent utility curtailments and interruptions, and provision of services at costs more favorable to consumers than conventional system costs (including acquisition, operation, and maintenance).

The report which follows is a result of the System Development Phase of the HP-ICES Project. The objective of this multiphase project is development and demonstration of HP-ICES concepts leading to one or more operational systems by the end of 1983. The seven phases include System Development, Demonstration Design, Design Completion, HP-ICES Construction, Operation and Data Acquisition, HP-ICES Evaluation, and Upgraded Continuation.

This project is sponsored by the Community Systems Branch, Office of Buildings and Community Systems, Assistant Secretary for Conservation and Solar Systems, U. S. Department of Energy (DOE). It is a part of the Community Systems Program and is managed by the Energy and Environmental Systems Division of Argonne National Laboratory.

Nine parallel concepts are under development in the initial, System Development, phase. The report which follows presents the interim findings in the development and analysis of a concept being developed by Dubin-Bloome Associates, P.C.

ABSTRACT

This report deals with a Heat Pump Centered Integrated Community Energy System which should provide all the heating, cooling, and other energy requirements to an entire community.

The ice generating HP-ICES uses the heat of fusion of water as a heat source for the heat pump, thus converting the water into ice. The ice will be stored in a bin and used the following summer for cooling which, therefore, could be considered a "by-product" of heating.

The annual overall Coefficient of Performance is expected to reach a value of 4.85 and related to source energy a value of $4.85 \times 0.31 = 1.5$. This is greater than 0.713, the value for which the HP-ICES would perform equally well with conventional system having the same cooling and heating output.

Actually the HP-ICES performs better than a conventional system operating even with 100% boiler efficiency and a cooling COP equal to 5.

In a case study performed on Market Square, Washington, DC it was shown that for the HP-ICES the annual source energy input and the annual energy cost are 50% of the corresponding values for the conventional system.

The annual operating cost of the HP-ICES is 73% of the annual operating cost for the conventional while its first cost is 90% greater than the first cost of the conventional system. This leads to a pay-back of 11 years calculated with a 10% interest rate and a 12% escalation rate for the operating cost.

The rate of return calculated over a period of 20 years with the same escalation rate is 18% .

It is therefore, obvious that the HP-ICES is both energywise and costwise more feasible than a conventional system of the same cooling and heating output and holds great promises to be an energy saving and an attractive system.

1 SYSTEM DESCRIPTION

1.1 SYSTEM CONCEPTUAL DESCRIPTION

The Heat Pump Integrated Community Energy System described in this report is designed to provide energy for heating and domestic hot water to an entire community in the winter time. The system uses the latent heat of fusion of water as its low temperature heat source.

The ice generated by the heat pump during the winter is stored in a bin and used the following summer for cooling purposes. Ice water is circulated through the coils of the air conditioning equipment picking up heat and dumping it back into the ice bin, thus gradually melting the ice. Towards the end of the cooling season all the ice will be melted and the cycle will begin again.

The ice generated during the heat pump operation in the winter thus becomes "free" energy obtained as a by-product of the heating cycle. Conversely, the summer cooling load could be considered as an indirect heat source for the heat pump making the heating a "free" by-product of the cooling cycle.

Whichever way the system is looked upon, the ice generating HP-ICES is a highly efficient energy conserving arrangement. It not only utilizes the heat pumps concept, generally recognized as being highly efficient by elevating low grade energy to high potential form, but also utilizes both energy transfer levels of the heat pump to accomplish both heating and cooling.

This considerably increases the annual overall Coefficient of Performance (the sum of cooling and heating effects divided by the power input).

Another major benefit of HP-ICES is its utilization of seasonal storage

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FIG. 1.1 ICE GENERATING HEAT PUMP - ICS

HP-ICES PROJECT

which enables the transfer of cooling energy from the time of its generation to the time of its convenient use.

This moves the air conditioning demand from summer to winter, cutting the summer electrical peak demand and at the same time improving the electrical utility's load factor, enabling the existing electrical power capacity of the power plant to accommodate all the existing and even growing electrical energy demand.

Also, the concept does not exclude the application of diurnal storage on the cooling cycle when chilled water may have to be generated to supplement the cooling requirements. In such cases, the chilled water would be generated at night again reducing the peak electrical demand.

The community, as envisaged here, would be composed of private residences, apartments, shopping centers, offices, and educational facilities which, taken together, could constitute a suburb of a large city or an independent municipality.

The multisector community composition will cause a load diversity as the energy demand peaks are non-concurrent. This will allow the installation of smaller size equipment and thus reduce the first cost.

Likewise, the HP-ICES being a centralized system, with centralized storage bin will require a much lower capital outlay than several decentralized systems which would be required by the separate sectors of the community if HP-ICES were not applied.

The thermal characteristics of the community, together with the storage systems (both seasonal and diurnal), impart considerable "thermal inertia" to the whole system, which results in the leveling of load fluctuations and a reduction in the energy demand.

HP-ICES PROJECT

The HP-ICES is influenced to a large extent by the composition of the community. The predominance of either the residential sector or the commercial sector with their annual load profiles will decide whether the HP-ICES is applicable to the community or not.

Communities comprising many office buildings will have an internal cooling load in the winter. In such cases, the heating load may be considerably reduced. If the internal heat gain is very large some mechanical cooling in the affected zones will be necessary and the heat pump might be able to perform cooling and heating simultaneously, using the internal cooling load as a heat source.

Whether reducing the heating load or using the internal heat gain as a heat source, the heat pump will generate less ice. The reduced quantity may be insufficient for satisfying the summer cooling requirements, making it mandatory to provide supplementary cooling during the summer.

The central HP-ICES contains a modified refrigeration system with compressors, double bundle condensers, cooling towers, chillers, icemaking equipment, evaporators, circulating pumps and heating and cooling distribution systems.

The compressors should have the lowest horsepower per ton power consumption possible. They should be able to perform satisfactorily at low suction temperatures necessary for ice generation and at high discharge temperatures required for generation of hot water for heating and domestic hot water.

The compressors should be able to operate also under standard air conditioning conditions, that is, with a higher suction temperature and a lower discharge temperature and with a performance which is competitive with that of a conventional air conditioning chiller. The transition from one range of

FIG-1-2 ICE GENERATING HEAT PUMP ICES

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

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operation to the other should take place smoothly without any adverse influence on the system performance as a whole.

The type of compressor to be used will be the subject of a separate investigation but the most likely choice will be the positive displacement screw compressor.

Likewise, the type of the refrigerant used should have a high mass density which will reduce the required displacement of the compressor, and a high heat capacity which will increase the refrigerating effect. The refrigerant should be relatively inexpensive and available on the market. The type of the refrigerant to be used will be investigated separately.

Ice generators (evaporators) should have large heat transfer surfaces, should be prefabricated, easy to install, control and maintain, and be readily available on the market.

Ice generation could be accomplished in two ways (to be discussed later)- either by the use of "ice makers" (where water is converted to ice by being circulated by a pump and sprayed over evaporators) or by the use of chilled brine carrying coils immersed in the ice bin.

The ice storage bin should be able to accommodate all the ice which will be used for cooling in the summer. There are two criteria for sizing the ice bin. In locations where too much ice is generated - more than the summer cooling requirements - the ice bin is sized to accommodate only the amount of ice required for the summer cooling. The excess ice will be melted. In locations where the heat pump winter operation generates less ice than required for the summer cooling, the ice bin is sized to accommodate all of the ice that the heat pump is capable of generating.

fig 1-3

THE FREEZING PERIOD

-Water circulating through tubes of freezer.

-Refrigerant surrounds tubes and causes ice to form on inner wall of tubes.

-Ice cylinders forming in tubes.

THE THAWING PERIOD

-Hot gas warms refrigerant to release ice from the tubes.

-Ice descends from tubes to a revolving cutter and then to the ice chute.

-Water.

HP-ICES PROJECT

The insulation of the ice bin, likewise, will be dependent on the location.

In some locations the HP-ICES will contain standard chillers along with the ice making evaporators (connected in parallel) to enable chilled water generation if the ice obtained from the heat pump proves insufficient to satisfy the summer cooling requirements of the community.

Condensers will be of the double bundle type with as large of a surface area as possible in order to be able to supply the required heat to the community or, alternately, to reject heat to the cooling towers if necessary.

Pumps will circulate hot and chilled water between the central plant and the community through a 4-pipe distribution system arranged either in a single loop or in a primary-secondary pumping arrangement for each of the chilled and hot water systems. The choice of the pumping arrangement will depend on the spread and composition of the community.

The type of the heating system chosen, either an air or water system, will depend on the temperature of the hot water obtainable from the double bundle condenser.

Many interdependent factors will affect the temperature of the hot water - the COP of the heat pump, the size of the piping, the insulation of the piping and the pumping power of the hot water.

The final trade-off between all the factors mentioned is the utilization temperature. This will affect the size of the coils if air systems are used or the size of the radiators or convectors if water systems are used.

The successful operation of the HP-ICES will depend, to a large extent on the climate and the length of the seasons in addition to the aforementioned

dependance on the composition of the community and its internal load.

In moderate climates there may be a balance between the winter heating requirements and the summer cooling requirements. The ice generated in winter by the heat pump would sufficiently cover the summer cooling requirements. However, the existence of a large internal load will disturb the balance and cause a deficiency in the generated ice.

In northern latitudes where the winters may be severe and prolonged while the summers are mild and short, the heat pumps will produce more ice than is required for the summer. However, the existence of a large internal load will reduce the excess ice and in certain cases may cause a balance between the heating and cooling loads.

On the other hand, in southern areas where the summers may be hot and long while the winters are short and mild, the ice produced by the heating cycle is not sufficient to satisfy the summer cooling requirements of the community. The existence of a large internal load may cause this deficiency to become more severe.

Whichever way the inbalance points, certain modifications to the HP-ICES are necessary.

Wherever there is an excess of ice, it is dealt with first by deliberately leaving the ice storage bin uninsulated to encourage heat leakage into the bin and cause some melting of the ice. If this proves insufficient, supplementary heat, mostly solar heat, is used to melt the excess of ice.

Another method used to reduce the ice generated in cold climates is to generate ice sparingly by utilizing outside air as a heat source. This is accomplished with air temperature down to 45°F. Ice generation is resumed

only when the temperature drops below 45°F.

On the other hand, wherever there is a deficiency of ice produced by the heat pump, other means must be sought to overcome the deficiency in the available "cooling" energy for covering the summer cooling requirements. First, the bin is well insulated to reduce heat gain and thus prevent the loss of ice.

This, however, is usually insufficient and it may be necessary to generate chilled water by conventional means with heat rejection to the cooling towers. Alternately (and preferably), chilled water may be generated by the heat pump generating domestic hot water at the same time. Better yet, this could be done at night thereby cutting down the electrical demand if electric drive is used. This also implies the utilization of the diurnal chilled water storage concept with the use of the already available ice storage bin or part of it.

Another way to overcome the deficiency in ice is to supplement this deficiency by means other than the heat pump, namely to generate ice by passing cold, outside air (at a temperature below 25°F) over a brine carrying coil. The cold brine is circulated through another coil immersed in the ice bin producing ice. With the use of this method, the size of the storage bin might have to be increased slightly.

The heat pump of the HP-ICES does not necessarily have to be powered electrically. It could be powered by a thermal engine (oil or gas), or by a gas turbine.

This is especially beneficial in locations where the winter electrical demand is considerable. In such cases, additional heat recovery could be employed, thereby reducing the size of the heat pump. However, additional environmental problems might be caused, offsetting some of the benefits of heat recovery.

HP-ICES PROJECT

Therefore, the advantages and disadvantages of thermal engine drive should be compared with those of electric drive.

With regard to the management of the HP-ICES, it could be integrated with an existing thermal utility. The local power company would probably be very interested in the acceptance of the new concept which, by its versatility and many new possibilities, would enable the power company to maintain better power and load management in the utility.

The charge to end users could be based on a measurement of the energy supplied in chilled and hot water.

To sum up, the HP-ICES offers a great variety of options all of which tend to reduce the consumption of energy. It is the most sophisticated and efficient energy system to-date capable of supplying all the heating, domestic hot water, and cooling requirements to an entire community with the least expenditure of input energy.

1.2 OVERALL COEFFICIENT OF PERFORMANCE

For the comparison of the ice generating Heat Pump Integrated Community Energy System with an alternative conventional system where heating is accomplished with fossil fuel-fired boilers and cooling with centrifugal chillers, an annual overall coefficient of performance, should be computed for both systems.

This overall COP should reflect the source energy utilized and should be the ratio of both heating and cooling output to raw source energy input.

The COP for the ice generating HP-ICES based upon compressor shaft energy input is :

$$COP_c = \frac{Q_c}{W}$$

$$COP_h = \frac{Q_h}{W}$$

$$COP_{Overall} = \frac{Q_c + Q_h}{W}$$

Accounting for the thermal efficiency of the prime mover at the power station, for the transmission and for distribution losses and motor efficiency, a correction factor of 0.31 should be applied to adjust for source energy input. Therefore, the overall COP adjusted to reflect the source energy input is:

$$COP_{overall} = \frac{(Q_h + Q_c)}{W} 0.31$$

using $Q_h = Q_c + W$, $COP_{overall} = 0.31 (1 + 2 COP_c)$

The overall coefficient of performance for a conventional plant with the same heating and cooling output as the HP-ICES is determined as follows:

$$\text{COP overall} = \frac{Q_h + Q_c}{\frac{Q_h}{E} + \frac{W_1}{0.31}}$$

where Q_h and Q_c are the heating and cooling outputs assumed to be equal for both HP-ICES and conventional system: E is the heating plant efficiency of the conventional system and W_1 is the conventional cooling plant shaft work input.

To illustrate the point the overall COP's for both the conventional and HP-ICES will be calculated for $Q_h = 15,000$ Btu/hr and $Q_c = 10,000$ Btu/hr. Assume $E = 0.6$ and the coefficient of performance for cooling for the conventional system, $\text{COP}_1 = 4.4$.

Thus for the conventional system

$$\begin{aligned} \text{COP overall} &= \frac{Q_h + Q_c}{\frac{Q_h}{E} + \frac{Q_c}{\text{COP}_1 \times 0.31}} \\ &= \frac{15,000 + 10,000}{\frac{15,000}{0.6} + \frac{10,000}{4.4 \times 0.31}} = 0.773 \end{aligned}$$

For the HP-ICES

$$\begin{aligned} \text{COP}_c &= \frac{10,000}{15,000 - 10,000} = 2 \\ \text{COP overall} &= 0.31 (1 + 2 \text{ COP}_c) \\ &= 0.31 (1 + 2 \times 2) = 1.55 \end{aligned}$$

which is greater than the COP overall for the conventional system.

Moreover, if we assume a boiler efficiency of 100% and a $\text{COP}_1 = 5$ the COP overall of the HP-ICES will still be greater than the COP overall for

the conventional system.

Thus

$$\text{COP overall} = \frac{15000 + 10,000}{\frac{15000}{1} + \frac{10,000}{5 \times 0.31}} = 1.16$$

A general formula connecting the coefficients of performance for cooling for the two systems where the two overall COP's are equal will now be derived.

Heating plant efficiency

$$= \frac{Q_h}{E} = \frac{Q_c + W}{E}$$

where W is the shaft work input for the HP-ICES, the other terms having been already defined:

For the conventional system

$$\text{COP overall} = \frac{Q_c + W + Q_c}{\frac{Q_c + W}{E} + \frac{W}{0.31}}$$

Substituting $\text{COP}_h = Q_c/W$, and $\text{COP}_c = Q_c/W$

and simplifying

$$\text{COP overall} = \frac{(1 + 2 \text{ COP}_c) 0.31E}{0.31 (\text{COP}_c + 1) + \frac{(\text{COP}_c)_E}{(\text{COP}_1)}}$$

Setting the two expressions for overall COP equal to each other and solving for COP_c yields

$$\text{COP}_c = \frac{E - 0.31}{0.31 + E/\text{COP}_1}$$

The assumption of a heating plant seasonal efficiency of 60% and a con-

ventional cooling plant coefficient of performance, COP_1 of 4.4 (based upon 0.8KW per ton), will yield a COP_c of 0.65 for the HP-ICES. For these conditions the overall COP's for the two systems will be equal.

The overall COP would be

$$0.31 (1 + 2 \times 0.65) = 0.713$$

$$\text{and } COP_H (1 + 0.65) = 1.65$$

Inspection of manufacturer's data for refrigerant compressors reveals that a cooling COP_c of 0.65 is much lower than the COP_c obtainable within the range of saturated discharge and suction temperatures considered for the HP-ICES.

Therefore, the HP-ICES should be more efficient than a conventional plant in the use of source energy. From Fig. 1.4 the saturation discharge temperature has to be less than 160°F if the COP for heating is not to be lower than 1.65.

FIG. 1.4

Variation of Coefficients of Performance for Heating with Saturated Discharge Temperature (SDT).

1) Carnot Effect = $\frac{T_{SDT}}{T_{SDT} - T_{SST}}$ (All R Temperature)

2) Net refrigerating effect based on theoretical refrigerant curve (p-H) Diagram

$$COP_H = \frac{W + Q_c}{W} = \frac{H_{\text{condenser}}}{H_{\text{compressor}}}$$

3) Compressor Performance (Manufacturer's Data)
Dunham Bush

1.3 OPTIMUM TEMPERATURE RANGE FOR COMPRESSOR OPERATION

The saturated suction temperature (SST) and the saturated discharge temperature (SDT) are set as a consequence of many different and often conflicting factors.

The SST is set by the ice making conditions. The 20°F SST is the highest temperature feasible for ice making. This sets the lower limit for the temperature range of the compressor. In any other conditions, for example, chilled water generation in summer, the suction temperature will be higher, thus improving the COP.

The choice of the SDT is more involved. The ΔT employed for heating will determine the flow rate. The flow rate will determine the pipe size. The pipe size, in turn, will affect the pumping power. For a constant return temperature of, for example, 90°F, a higher ΔT means smaller flow and smaller pipe sizes. On the other hand, higher temperatures mean lower COP's.

The saturated discharge temperature of the compressor has a lower limit set by the utilization temperature of the hot water for heating and domestic hot water which may not go below 110°F, if it is to provide efficient heating.

Accounting for heat distribution losses of, say, 10%, the drop in temperature for a $\Delta T = 110^\circ\text{F} - 90^\circ\text{F} = 20^\circ\text{F}$ will be 2°F permitting the minimum temperature for the hot water to be 112°F. This would set the condensing temperature at 122°F.

The higher limit of the compressor saturated discharge temperature is set by cycle efficiency considerations resulting from the thermodynamic comparison of the overall COP of the heat pump system with the COP of the conventional system.

It was shown in Section 1.2, Page 15 that for the ice generating HP-ICES to be efficient, the overall COP of the heat pump should be greater than 0.713 accounting for the source energy considerations. From the analysis and graph in Section 1.2, Page 16, the resulting COP for heating should be greater than 1.65, and the highest corresponding saturated discharge temperature should be less than 160°F.

Table 1.1 shows that increasing the outlet temperature does not necessarily decrease the pipe size, nor does it reduce the pumping power. However, in every case the power input to the compressor of the heat pumps is increased.

It is evident from the table that going from as 110°F to a 120°F water temperature raises the power input to compressor by 176 hp while the saving in pumping power (without changing the pipe size) is 0.704 hp per 100 ft. of piping.

This means that if the piping is

$$\frac{173}{0.704} \times \frac{100}{1} = 24,574 \text{ ft.}$$

the saving in pumping horsepower will balance out the additional compressor power input. Up to 25,000 ft. the compressor power input exceeds the savings in pumping power. Above 25,000 ft. there is a saving in pumping power over the compressor power input.

The size of the piping being the same, it appears that the 110°F water temperature is more feasible than the 120°F temperature. Comparing the 120°F and 130°F water temperatures, it appears that at the 130°F water temperature, both the pumping power and the compressor power input are greater than at the 120°F water temperature. Comparing the 120°F with 140°F water temperature shows a compressor incremental power input of 490 hp that can be compensated by saving

in pumping horsepower with a distribution network larger than 17,500 ft.

The condensing temperature of the ice generating heat pump is within the limits, $122^{\circ}\text{F} < \text{SDT} < 160^{\circ}\text{F}$. Naturally, a SDT of 160°F would make the HP-ICES just competitive with the conventional system. This is from an energy usage point of view alone without consideration of the capital costs of the two systems.

Therefore, for the HP-ICES to be beneficial, the SDT should at least be 150°F which would result in a hot water temperature of 140°F . It thus appears that the two feasible limits for the hot water temperature are 110°F and 140°F .

The four candidate temperatures for the hot water, 110°F , 120°F , 130°F and 140°F were investigated in a brief analysis by comparing the pumping horse power of the hot water for each temperature with the compressor, horse power input required to produce and deliver 10,000,000 Btu/hr with a 90°F return water temperature.

The data are tabulated below.

TABLE 1.1

Pipe Dia- meter (in.)	Flow Velo- city ft/sec.	Hot Water Temp. $^{\circ}\text{F}$	Head Loss Ft per 100 ft	Gal. per Min.	hp per 100 ft of Pipe	COP	SDT $^{\circ}\text{F}$	hp to Com- pressor per 10^7 Btu/hr heat reject- ed
8	6.5	110	2.75	1000	0.992	3.50	120	1123
8	4.5	120	1.20	666	0.288	3.03	130	1296
6	5.5	130	2.5	500	0.451	2.75	140	1429
5	6.5	140	4.00	400	0.577	2.2	150	1786
6	4.4	140	1.8	400	0.26	2.2	150	1786

It is thus obvious that the hot water temperature has to be chosen for each particular community dependent on its size, the length of the distribution system, and the interrelation between the pumping power and the compressor input power.

Besides, we have to consider the degradation of the system with years of operation.

Considering the above, for the purpose of this report we are going to use the mean temperature from the preceding table, that is 120°F water which will set the saturated discharge temperature at 130°F.

1.4 REFRIGERANT INVESTIGATION

Although the primary function of a refrigerant is to remove heat, the choice of the refrigerant for a particular application is determined by consideration of all of its properties.

Its flammability, toxicity, density, viscosity and availability are taken into account in making the choice although its refrigerating capacity remains the deciding factor.

The Second Law of Thermodynamics, and following from it the Carnot Cycle, require that for a heat engine, heat be supplied from a high temperature source at as high a temperature as possible and be rejected to a low temperature sink at a temperature as low as possible. This would increase the area of temperature-entropy diagram and thus increase the work obtained from the heat engine.

For the refrigeration cycle (or the reversed cycle heat engine), we are interested in reducing the area on the temperature-entropy diagram, that is, the work input. Consequently, the heat absorbed from the low temperature source should be achieved at a temperature as high as possible and rejected to the high temperature sink at a temperature as low as possible.

Thus, according to Second Law, for the refrigeration cycle and consequently, for the heat pump cycle, the evaporation temperature at which heat is absorbed should be as high as possible, and the condensing temperature at which the heat is rejected should be as low as possible.

As shown in the temperature analysis (Section 1.3), the saturated suction temperature (SST) of the system was set at 20°F and the saturated discharge temperature (SDT) at 130°F.

For any medium, the pressures are proportional to the temperatures and not

being able to control the temperatures any more, we shall look for a refrigerant having the condensing and evaporative pressures as close to one another as possible, that is, a compression ratio as low as possible.

In other words, the most desirable properties of a refrigerant should be an evaporating pressure as high as possible and a condensing pressure as low as possible.

Coupled with this, a good refrigerant should have a high refrigerating capacity, that is, a large latent heat at evaporator pressure. A high refrigerating capacity, together with a low compression ratio, would necessarily lead to a low horsepower per ton refrigeration requirement which is the ultimate goal in any refrigeration or heat pump system.

In this study, R-11, R-12, R-22, R-113, R-500, R-502 and R-717 were investigated for suction temperatures at 20°F and condensing temperatures of 130°F. Single stage compression and two stage compression with intercooling between the stages were considered.

Generally, two stage compression was found to be more efficient, especially if the intercooling was done with water rather than with the refrigerant. In the latter case, refrigerant liquid from the condenser is injected into the first stage discharge to desuperheat the gas. This increases the mass flow through the second stage of compression, increasing the work or energy input to the second stage.

However, R-12 and R-502 show an opposite trend, namely that single stage compression is more efficient. The calculations are shown in Appendix 1A and were performed on certain assumptions like saturated suction conditions, isentropic compression and saturated liquid in the condenser. The results are

summarized on Pages 25-26, Tables 1.2 and 1.3.

R-11 and R-113 were discarded as possible candidates because of very large suction volumes requiring very large displacement compressors. Also, both have high compression ratios and very low evaporator pressures. Both are limited to centrifugal compressors and require multi-stage compression which is expensive. R-11 is more suited for water chilling than for ice making. In the case of R-113, it was found that the assumption of saturated suction conditions leads to discharge conditions in the wet region. To obtain saturated discharge conditions, a superheat of suction vapor would have to be assumed which was contrary to the assumption made in this study.

R-717 has some desirable properties like a large refrigeration capacity and consequently a small rate of flow per ton, a small volume rate of flow allowing a smaller compressor displacement, smaller piping, and generally a smaller system. Cost is about 15¢ per lb (as opposed to \$1.00 per lb for hydrocarbons). Finally, its odor makes it easily detectable in case of leakage which is not the case for other odorless refrigerants.

The ANSI B9.1-1971 Safety Code and ASHRAE 15-70 put the R-717 in Group 2 in the refrigerant Classification Section which classified the refrigerant by toxicity and flammability. Group 2 refrigerants are prohibited for use in direct systems, but are allowed in indirect systems in special machinery room (Class T), with some safety requirements. These requirements, however, will make the use of R-717 more complicated.

The central plant in a community could very easily overcome the limitations set by the Code, and thereby, benefit from the desirable properties of R-717, mentioned above.

R-502 was eliminated as its capacity is relatively low. Its COP is also low and its use is limited to cases of suction temperatures below 0°F. This limited the study to R-12, R-22 and R-500 which were singled out for further investigation.

All three are non-flammable and belong to Group I of ANSI B9.1-1971 Safety Code and Groups 5 and 6 of Underwriter's Laboratories Classification (ASHRAE Fundamentals Pg. 253). Both classify them as the least toxic refrigerants.

R-12, R-22 and R-500 were further studied for 35°F suction temperature and 105°F condensing temperature, that is, for conditions encountered in standard air conditioning work (see page 27a).

This was done in light of the fact that in some climatic zones the ice generated during the winter heat pump operation might not be sufficient to cover the summer cooling requirements and some supplementary cooling might prove necessary in the off electric peak periods during the summer months.

In such cases, the same ice making compressor might be called upon to operate under different conditions than ice making. Thus, the refrigerant chosen should be suitable for operation within a range of 20°F to 35°F saturated suction temperature and 105°F to 130°F saturated discharge temperature. The calculations are summarized in Table 1.3A on Page 27a.

It appears that for the air conditioning range of operation, R-22 has the smallest CFM/ton but its power requirement is somewhat larger than that of R-500. For the ice making range, R-12 appears to be the best choice. Ice making is the prevailing consideration since it is the primary purpose of the compressor in ice generating HP-ICES application.

TABLE 1.2

SUMMARY OF ONE STAGE COMPRESSION PERFORMANCE FOR EVAPORATOR TEMPERATURE OF 20°F AND CONDENSER TEMPERATURE OF 130°F

REFRI- GERANT	PRESSURE PSIA		COMPRESSOR RATIO	COP	HP/TON	REFRIGERANT FLOW PER TON		DIS- CHARGE TEMP. °F.
	P _s	P _d				LB/MIN*	CFM**	
R-11	4.35	38.67	8.9:1	3.80	1.23	3.37	28.6	148
R-12	35.74	195.71	5.5:1	3.11	1.52	4.90	5.38	145
R-22	57.73	311.50	5.4:1	3.08	1.53	3.49	3.27	180
R-113	1.53	18.45	12.1:1	3.17	1.48	4.23	76.14	130
R-500	41.96	231.90	5.5:1	3.10	1.50	4.09	4.61	150
R-502	67.16	335.54	5.0:1	2.55	1.85	6.09	3.71	150
R-717	48.21	330.00	6.8:1	3.32	1.42	0.48	2.82	290

P_s = Suction Pressure

* THROUGH THE EVAPORATOR

P_d = Discharge Pressure

** AT COMPRESSOR SUCTION

TABLE 1.3

SUMMARY OF TWO STAGE COMPRESSION PERFORMANCE FOR EVAPORATOR
TEMPERATURE OF 20°F AND CONDENSER TEMPERATURE OF 130°F

REFRI- GERANT	PRESSURE PSIA		COM- PRESSION RATIO	COP		HP/TON			REFRIGERANT FLOW PER TON					DIS - CHARGE TEMP. °F
									LB/MIN *		CFM **			
	P _S	P _D		INTER- COOLING		INTER- COOLING		FIRST STAGE	SECOND STAGE		FIRST STAGE	SECOND STAGE		
									INTER- COOLING			INTER- COOLING		
				WATER	REF.	WATER	REF.		WATER	REF.		WATER	REF.	
R-11	4.35	38.67	8.9:1	3.85	3.80	1.23	1.24	3.37	3.37	3.37	28.6	1.14	1.26	137
R-12	35.74	195.71	5.5:1	3.11	3.06	1.52	1.54	4.90	4.90	4.90	5.38	2.38	2.45	138
R-22	57.73	311.50	5.4:1	3.24	3.14	1.45	1.51	3.49	3.49	3.49	3.27	1.43	1.55	150
R-113	1.53	18.45	12.1:1	-	-	-	-	4.23	-	-	76.14	-	-	-
R-500	41.96	231.90	5.5:1	3.14	3.08	1.50	1.53	4.09	4.09	4.09	4.61	2.02	2.10	140
R-502	67.16	335.54	5.0:1	2.49	1.82	1.85	1.90	6.09	6.09	6.09	3.71	1.65	1.74	140
R-717	48.21	330.00	6.8:1	3.57	3.39	1.32	1.39	0.48	0.48	0.48	2.82	1.14	1.26	208

* THROUGH THE EVAPORATOR

** AT COMPRESSOR SUCTION

The theoretical study was useful in narrowing down the range of possible choices of refrigerants, in explaining the procedure probably followed by the manufacturers themselves, in their choice of refrigerants, and in enabling any reader of this report to better understand the underlying engineering principles on which this report is based (see Appendix 1A).

TABLE 1.3a

COMPARISON OF PERFORMANCE PER TON OF REFRIGERATION

REFRIGERANT	$T_S = 20^{\circ}\text{F}/T_D = 130^{\circ}\text{F}$				$T_S = 35^{\circ}\text{F}/T_D = 105^{\circ}\text{F}$			
	COP	HP/TON	FLOW LB/MIN.	CFM/TON	COP	HP/TON	FLOW LB/MIN.	CFM/TON
R-12	3.11	1.514	4.90	5.38	5.74	0.821	4.11	3.46
R-22	3.08	1.53	3.49	3.27	5.78	0.816	2.99	2.14
R-500	3.1	1.52	4.09	4.61	5.82	0.810	3.42	2.95

1.5 COMPRESSOR INVESTIGATION

The performance of a Durham-Bush E (HP) x 2516 screw compressor was investigated using R-12 and R-22, both at 20°F SST and 130°F SDT as well as at 35°F SST and 105°F SDT, that is, for conditions suitable both for ice making and standard air conditioning application respectively.

In both operating ranges, R-22 proved to be the superior of the two refrigerants producing a much larger tonnage (about 63% more) than the R-12 at a cost of 1% to 5% increase in HP/ton power input. (see Table 1.4)

This result is in agreement with the refrigerant investigation in which it was found that the CFM/ton for R-22 is smaller than the CFM/ton for R-12. Thus, a given compressor displacement will circulate per unit time more R-22 than R-12.

Likewise, the performance of the same compressor at 35°F - 105°F proved to be better for both refrigerants than the performance at 20°F - 120°F which, of course, was expected. The result was also obtained with the refrigerant investigation.

Table 1.5 confirms the above results for centrifugal compressors.- It appears that R-22 is the preferred refrigerant for the entire range of operation suitable for ice making and air conditioning. Table 1.6 shows a comparison of a centrifugal and a screw compressor. The comparison was limited to R-12, as the only full data available related to this refrigerant. Also, with the data available, the discharge temperature of the centrifugal compressor was limited to 120°F and, therefore, the performance of the screw compressor was also evaluated at this discharge temperature. For the latter, however, data were also available at 130°F and even higher temperatures.

•TABLE 1.4

COMPARISON OF PERFORMANCE OF SCREW COMPRESSORS (TYPE E (HP) X 2516)

DUNHAM-BUSH USING R-12 AND R-22 REFRIGERANTS. TABLES BASED ON

S.S.T. = 20°F

S.D.T. = 130°F AND

S.S.T. = 35°F

S.D.T. = 105°F

(TABLES ALSO BASED ON 10°F LIQUID SUBCOOLING & 10°F SUCTION SUPERHEAT)

REFRI- GERANT	$T_S = 20^\circ\text{F}/T_D = 130^\circ\text{F}$							$T_S = 35^\circ\text{F}/T_D = 105^\circ\text{F}$						
	REFRIGERANT CAPACITY		POWER REQUIREMENT			HEAT REJECTED	COOL- ING	REFRIGERANT CAPACITY		POWER REQUIREMENT			HEAT REJECTED	COOLING C.O.P.
	TONS	BTU/H	HP	HP/TON	BTU/H	BTU/H	C.O.P.	TONS	BTU/H	HP	HP/TON	BTU/H	BTU/H	
R-12	275	3,300,000	595	2.16	1,514,000	4,814,000	2.18	450	5,400,000	460	1.02	1,171,000	6,571,000	4.60
R-22	450	5,400,000	1025	2.28	2,609,000	8,009,000	2.07	735	8,820,000	760	1.03	1,934,000	10,754,000	4.56

At a lower suction temperature the centrifugal compressor performs slightly better, but at suction temperature of 35°F, the performance of the screw compressor is superior.

The centrifugal compressor, however, is not suitable for high discharge temperatures and the operation of the same centrifugal compressor within a range of 20°F to 35°F suction temperature is difficult to attain. The speed of the compressor would have to be re-adjusted each time a transition from one suction temperature to another is made.

This is an important consideration as a condition may arise where in the winter there is a cooling load, for example, in the interior zones, the heat pump might be called upon to perform simultaneous cooling and heating with the compressor switching over from 20°F suction temperature for ice making to 35°F standard air conditioning.

Reciprocating compressors are most efficient in small sizes, although they are suitable for operation over a wide pressure range. Inquires made with the manufacturers brought out the fact that reciprocating compressors are mostly manufactured up to sizes of 80 tons and not more than 350 tons.

Within the range of 200 ton - 1000 ton, the helical rotary compressor or screw compressor has earned its place in recent years, but above 1000 tons, the centrifugal compressor is used.

In our ice making application we shall use the screw compressor. They are a new generation of compressors introduced into the refrigeration industry in the late 50's; they earned almost instant acceptance.

Screw compressors belong to the broad class of positive displacement compressors. They are versatile and can be used for all common industrial gases,

for air, for refrigeration and air conditioning, and for heat pump applications. They operate stably over a wide range of evaporating and condensing temperatures and are capable of operating high pressure heads for all applications with the usual high pressure refrigerants like R-12, R-22, R-500 and R-717.

The oil flooded helical rotary compressors are mechanically simple with only two major moving parts - two rotating screws which turn at conservative speeds in an oil bath. The twin helical rotors compress refrigerant gas in a purely rotary motion thus assuring a non-pulsating, uniform gas flow and even torque with minimum noise, vibration and wear.

Compression is achieved by direct volume reduction with pure rotary motion. Variable capacity control from 100% down to 10% provides efficient part-load operation. The unique energy saving slide valve capacity control varies displacement for efficient part load performance and offers stepless capacity modulation in infinite stages down to 10% of full load. This permits the compressor to operate at the lowest KW input during part-load operation. The slide valve precisely controls the amount of refrigerant gas to be compressed, matching output with demand and thus significantly saving the power expenditure.

Capacity can also be varied automatically with the variation of suction temperature, that is, by switching from ice making to ordinary air conditioning. This is one of the most important advantages that the screw compressors has over centrifugal compressors. This alone would be sufficient cause to favor the use of screw compressors for our particular application.

Moreover, with screw compressors there are no speed increasing gears, no elaborate inlet guide vane assemblies, no crank shaft and pistons, no suction and discharge valves, and practically no clearance volume.

High adiabatic and volumetric efficiencies are achieved and the compressed gas is partly cooled by oil which may result in the possible use of a smaller condenser.

Size for size, the screw compressor is one of the smallest, lightest compressors on the market using far less space than any other equivalent capacity compressor. With no unbalanced forces, there is no need for costly foundations.

Two stage compression can be accomplished without the need of interstage desuperheating. The low stage compressor can discharge directly into the suction of the high stage compressor with a common oil separator serving both compressors.

As for maintenance, the solid steel screws practically never wear out. The smooth vibrationless operation with minimum working parts assures efficiency of operation, no loss of capacity, performance throughout the life of the compressors, and the ability to stand up under long hours of constant use.

Thus, there is no major mechanical maintenance to perform during the normal life of the machine. The estimated time between maintenance periods is 50,000 hours. The replacement parts are cheaper than those for a centrifugal or reciprocating compressor.

From the above, the conclusion can be drawn that, for our application, we would be inclined to use screw compressors with R-22.

The final decision, however, will also depend on the opinion of compressor manufacturers.

TABLE 1.5

PERFORMANCE OF MPS SIZE 40 CARRIER CENTRIFUGAL COMPRESSOR
WITH R-12 AND R-22 AT 20°F AND 35°F SUCTION TEMPERATURES

Refrigerant	Capacity at 20°F Suction Temperature	Capacity at 35°F Suction Temperature
R-12	1480 tons	1780 tons
R-22	2450 tons	3130 tons

This comparison shows that R-22 has a larger refrigeration capacity than R-12 when both are used in the same compressor.

HP-ICES PROJECT

TABLE 1.6

COMPARISON OF PERFORMANCE OF CARRIER MDS SIZE 40 CENTRIFUGAL COMPRESSOR
VS 2516 DUNHAM-BUSH SCREW COMPRESSOR USING REFRIGERANT - 12

SUCTION TEMPERATURE = 20°F DISCHARGE TEMPERATURE = 120°F AND
 SUCTION TEMPERATURE = 35°F DISCHARGE TEMPERATURE = 105°F

Type of Compressor	$T_S = 20^{\circ}\text{F}/T_D = 120^{\circ}\text{F}$		$T_S = 35^{\circ}\text{F}/T_D = 105^{\circ}\text{F}$	
	HP/Ton	COP Cooling	HP/Ton	COP Cooling
Centrifugal	1.65	2.86	1.1	4.28
Screw	1.80	2.62	1.03	4.57

1.5.1 PART LOAD COMPRESSOR PERFORMANCE

a. Screw Compressor

For the determination of part load performance, the data of leading manufacturers in the field were consulted. Among them were Durham-Bush, Vilters Co., Freezing Equipment Sales (FES), Inc., Sullair Corp. and others.

All use the same principle of capacity and load control. There is a slide valve in the rotor housing capable of moving back and forth in an axial direction. (See Figure 1.5 and 1.6)

FIGURE 1.5

FIGURE 1.6

The movement of the valve is programmed by a control arrangement which is pressure or temperature initiated and hydraulically activated.

When the compressor is fully loaded, the slide valve is in the closed position. Unloading starts when the slide valve is moved back away from the valve stop. Movement of the valve creates an opening in the bottom of the rotor housing through which suction gas can pass back from the rotor housing to the inlet port area before it has been compressed.

Capacity reduction down to 10% of full load is realized by progressive backward movement of the slide valve away from the valve stop. In principle, enlarging the opening in the rotor housing effectively reduces compressor displacement. Even though all the manufacturers use the same principle for capacity

control, the part load performance of screw compressor differs from manufacturer to manufacturer.

In actual selection of the screw compressor type and model, its most likely part load range of operation should be taken into account and matched with a compressor whose performance is best in the given range.

In Figure 1.7 the full and part load performance of screw compressors manufactured by Durham-Bush, Sullair Corporation and Freezing Equipment Sales are compared.

At part load and down to 50%, the Durham-Bush compressor requires less power input per unit capacity than at full load. From 50% load and down, the horse power per ton input is greater than at full load and tends to increase as the load becomes smaller.

The Sullair compressor has a similar curve shape to that of Durham-Bush, but the part load power input per unit capacity is greater than at full load, increasing with the decrease in load. At all part loads, it is greater than the corresponding power input for the Durham-Bush compressor.

The FES compressor has a greater power input at part load than at full load and a considerably greater power input than the corresponding power input for the compressors of the other two manufacturers.

Figure 1.8 shows the performance of a York centrifugal compressor. Down to 30%, the part load power input per unit capacity is less than at full load and less than the corresponding power input for screw compressors.

TYPICAL PART LOAD POWER INPUT CURVE

Rotary Screw Compressor

FIGURE 1.7

- ① "FES" Rotary Screw Compressor - Technical Manual
- ② "Sullair Corporation" - Selection Guide
- ③ "Durham-Bush" Screw Compressor - Catalog #6043C 20°F

YORK CENTRIFUGAL COMPRESSOR

At part load operation the power per ton is slightly less than at full load operation.

Thus, at 75% and 50% capacity the power input is 70% and 40% of full load input respectively.

However, as the capacity drops to 30% or 20% the power input per ton increases.

$$\text{Capacity ratio} = \frac{\text{designed or partial load (tons)}}{\text{rated capacity}}$$

Best part load efficiency is obtained with automatic inlet guide vane positioning with manual speed reset (A on chart).

Other part load efficiency conditions are: automatic inlet guide vane positioning together with automatic speed control (B), or automatic inlet guide vane positioning with constant compressor rpm (C).

Figure 1.8

At part load the centrifugal compressor is more efficient than the screw compressor. However, our operation is closer to full load operation, and for other reasons, specified in our compressor studies, we shall use screw compressors.

1.6 ANALYSIS OF STORAGE MEDIA PROPERTIES

Following is an evaluation of the properties of some storage media that are already being researched in the field. Table 1.7 below shows their main properties pertaining to thermal storage.

TABLE 1.7

Properties	Water (Latent Heat)	C-14, C-16 Paraffin	Water (Sensible Heat)
Specific Heat (Btu/lb-°F)	-	0.936	1
Heat of Fusion (Btu/lb)	144	71.1	-
Density (lb/ft ³)	55.4	48.0	62.4
Storage Temperature Range (°F)	32	35-40	100-200
Heat Storage Density (Btu/ft ³)	8000	3412	6240
Toxicity	No	No	No
Cost (\$)/lb	0	0.05	0
Corrosion	No	No	No
Availability	Plentiful	Scarc	Plentiful
Application	Cooling	Cooling	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/ft³) 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, it does not require special vessels, it is safe to handle, it is easy to handle in a distribution network, and the technology of its use is readily available and 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 the 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.

In conclusion, it should be stressed again that at the present time, water is the most suitable and efficient storage medium, for both cooling and heating applications.

1.7 ICE GENERATING METHODS

There are two methods of ice generation. In one, coils carrying chilled brine at 15°F 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.

The chilled brine system has a very low efficiency because ice has a low heat conductivity. In the process of ice making, the ice layers forming on the surface of the coils reduce heat transfer and thus impede further ice formation.

This method of ice formation also requires many coils to be installed in the bin. The coils cannot be prefabricated, only field installed, thus increasing the cost.

Moreover, this system is not suitable for large scale applications because of maintenance problems involved, and, in the case of coil fracture, the ice bin might have to be drained. This would cause pollution problems in the storm drainage system.

However, the chilled brine process has one advantage in that the ice that is formed is solid with no air spaces. This results in a higher energy density in the ice.

A provision is made to harvest the ice, that is, to separate it from the plates and collect it in the ice bin.

When the ice reaches a predetermined thickness on a plate, the plate is defrosted by passing hot refrigerant gas through it, thus melting the ice

around it. This detaches the ice from the plate, and it falls into the ice storage bin. If the bin is not located at the same place where ice is generated, a conveyor is used to carry the ice into the ice bin.

Usually, there are two ice makers to each compressor so that one plate is defrosted at a time while the other, in conjunction with the compressor, continues to produce ice. With the ice maker method, the generation of ice is more efficient than with the chilled brine carrying coils, the aforementioned reason of reduction in heat transfer between the brine and the water.

The ice maker system being part of the mechanical room is easier to maintain, can be prefabricated, and modularly expanded. It could also be located remotely from the ice bin. The system also causes no pollution problems.

The disadvantage of the ice maker operation is that the ice is not as solid as with the chilled brine method but contains air spaces in it. Instead of a solid block of ice, we get a mixture of ice, air and water, which, in fact, decreases the density of the ice. This requires a greater volume capacity of the bin.

The attached schematic (Figure 1.1, 1.3) conveys an idea of how ice is generated with the "ice maker" method.

1.8 COMPONENT IDENTIFICATION

In due consideration of the heating and cooling peak loads and monthly energy requirements, the selection of the equipment has to be made.

Compressor

The compressors should be able to operate in winter time with a SST of 20°F and SDT of 130°F and with as low horse power per ton power consumption as possible. In summer time, if there is need for supplementary cooling, the compressors should be able to operate under standard air conditioning conditions and with a performance competitive with that of conventional chillers.

The peak heating load should be the major factor in compressor selection and in determining their total capacity since the heating will be generated right at the time when it is required.

The system part load performance should be analyzed to determine whether one, two or multiple compressors should be used to cover the full and part load conditions. For example, if the system operates mostly near full load, one compressor is preferable but if part load conditions prevail, several smaller compressor might be preferable.

For summer operation the compressors need not match the peak cooling load, since a diurnal chilled water storage system is utilized.

Condenser

Since the refrigeration cycle rejects more heat in the summer time when the cycle operates under 35°F SST and 105°F SDT, the condensers should be sized for this condition.

Evaporators

The refrigerant pressure drop in the evaporators, suction lines and valves should be studied to determine whether a direct expansion or recirculation system is used.

Chillers

Selected according to compressor performance at 35°F SST and 105°F SDT.

Ice Makers

The ice maker will be selected to use the full capacity of the compressors for efficient ice generation, being, however, slightly oversized to account for intermittent operation. Both plate and tube ice makers will be considered.

Ice Storage Bin

The ice bin size will be based on two criteria: either the cooling requirements in the summer for colder locations or the ice generating capacity of the winter heat pump operation for warmer locations.

The same criteria will govern the ice bin insulation: in the colder areas the ice bin will be uninsulated, while in the warmer areas the insulation will vary with the climate.

Also, to reduce the size of the ice bin, the sensible heat of the chilled water will be used as a source for the heat pump. Thus, the chilled water will be first cooled, say, from 57°F down to 32°F and then converted to ice.

In summer, the chilled water will be used until its temperature goes up from 32°F to 57°F.

Distribution System

Considering the community composition and its multi-purpose uses, it may be necessary to supply cooling and heating at the same time. Therefore, a 4-pipe system is necessary, and it will enable the heat pump to transfer heat from the area which is being cooled to the area which is being heated.

Hot Water

Following the results of the analysis in Section 1.3, the hot water temperature used will be 120°F with a $\Delta T = 30^\circ\text{F}$.

Accordingly, an air system will be used as the hot water temperature is too low for radiators or convectors.

Chilled Water

The chilled water will come from the ice bin and its temperature will be mostly 32°F. After the melting of the ice, the temperature of chilled water will be allowed to go up to 57°F. The cooling coils will have to be sized for the highest temperature of the chilled water.

Domestic Hot Water

The domestic hot water system will be based on local heat exchangers with the heat supplied by the distribution system. 110°F is the maximum domestic water temperature attainable with a heating water supply temperature of 120°F. 110°F is considered adequate for most domestic usage. If higher temperatures are required for process use, supplementary heat will have to be provided.

1.9 HP-ICES MODES OF OPERATION

The nature of the HP-ICES renders it very feasible for operation in several modes. Since the system is designed to supply all the heating, domestic hot water and cooling requirements all year round, by the utilization of the heat pump during the winter for cooling in summer, the basic

mode of operation becomes the winter mode.

Besides the basic winter mode, the system has five additional modes of operation to accommodate the community requirements at any time. All this accomplished with maximum efficiency attainable.

1. Basic Winter Mode

When only heating is required and the ice storage temperature is above or equal to 32°F, the control center will set the (SST) Saturation Suction Temperature at 20°F and (SDT) Saturation Discharge Temperature at 130°F. Also, it will energize the compressors to generate hot water at 120°F and ice for storage and will let the hot water pump circulate the hot water through the community network.

2. Second Winter Mode

When heating and cooling is required and the water storage temperature in the bin is above or equal to 32°F, the control center will set one of the compressor at SST - 35°F and SDT - 130°F, and the other compressors at SST - 20°F and SDT - 130°F. The system in this case, will generate hot water to be distributed through the community network, chilled water with one compressor to accommodate the cooling required, while the remaining compressors will continue to generate ice for storage.

3. Summer Basic Mode

When only cooling is required and the water temperature in the storage bin is between 32°F and 50°F, the control center will

energize the ice storage circulation pump and will let the chilled water circulate through the community network.

4. Summer Second Mode

When cooling and heating is required and the ice storage temperature is between 32°F and 50°F, the control center will set one of the compressors at 35°F SST and 130°F SDT and will let it generate the hot water requirement for the community, while generating chilled water. If more chilled water is required it will be supplied from the ice storage bin.

5. Summer Third Mode

When the ice storage temperature increases above 50°F and there is low demand for cooling by the community (mainly during summer nights), the control center will set the compressors at SST - 35°F and SDT - 105°F to generate chilled water for storage to be used during the following day with heat rejection to the cooling towers.

6. Summer Fourth Mode

When only cooling is required and the ice storage temperature is above 50°F (insufficient to satisfy the community cooling requirement), the control center will set the compressor at SST - 35°F and SDT - 105°F, letting them generate enough chilled water to satisfy the demand with heat rejection to the cooling towers.

The ice generating HP-ICES could also operate with some other modes. These could be worked out according to the community's requirement by the control center as a combination of some of the above modes.

FIG.-1-9 BASIC WINTER MODE

FIG.-1-10 SECOND WINTER MODE

FIG. - 1-11 BASIC SUMMER MODE

FIG.-1-12 SECOND WINTER MODE

FIG.-1-13 THIRD SUMMER MODE (NITE TIME)

FIG. 1-14 FORTH SUMMER MODE

2. POTENTIAL APPLICATIONS

2.1 INTRODUCTION

Because the system uses the heat generated by the ice making process in winter and stores the ice for cooling in the summer, the annual efficiencies indicate excellent potential applications. Even if the loads on the system are not perfectly balanced, it can still save energy since any ice stored saves energy otherwise needed for a chiller operation during the summer. The following are the opportunities of the HP-ICES:

- a. High annual overall COP.
- b. Reduction in peak electrical demand
- c. Reduction in electrical energy consumption
- d. When both heating and cooling are required the economics are greatly enhanced
- e. The availability of equipment
- f. High energy density per unit volume of storage.
- g. Flexibility of the system (suitability of the system to operate in various conditions)
- h. The larger the ice storage bin volume the lower the relative heat gain and the lower the cost per unit volume.
- i. Multiple compressor application allows greater efficiency in part load situations.
- j. The control system can be made more sophisticated making the equipment more responsive to environmental changes and conditions.

HP-ICES PROJECT

- k. Large storage volume allows reduction in size of equipment
- l. Building structure could be utilized as storage.
- m. No environmental problems on site (some additional air or water pollution problems could be created at the generating plants, but these plants are usually equipped with sophisticated means to deal with such problems).
- n. Reduction in size of equipment due to load diversity and due to a centralized system.
- o. Reduction in maintenance and labor cost.
- p. Centralized HP-ICES will bring down the cost per unit energy for the customer.

It is also important to examine where the system becomes less effective.

The following are the constraints of the HP-ICES.

- a. Equipment not specifically designed for this concept.
- b. Lower COP at low saturated suction temperature for ice making.
- c. Heating and cooling loads profiles are not always compatible.
- d. Large storage volume required.
- e. High first cost.
- f. Many building trades and disciplines are involved.
- g. Relatively low hot water temperature requires a large distribution network which incurs a high first cost and results in high pumping power.
- h. Community distribution network will cause high losses.
- i. The chilled water system will be an open system causing additional hydraulic problems.
- j. Customer billing problems.
- k. Expert maintenance crew required.

2.2 THE ENERGY BALANCE FOR HP-ICES

Undoubtedly load balance is the most critical element for efficient operation. The ideal situation would be to have the heat pump provide all the heat required during the winter by generating exactly enough ice to cool in the summer. Extremes to either side negate the desirability of the annual storage cycle. If too much heat is required, ice generated and stored would require some sort of auxiliary heat source for melting. If the cooling load is too large, then the stored ice would often be insufficient and conventional cooling would reduce the savings in energy.

Essentially two conditions determine this: the climatological conditions and the composition of the community.

2.2.1 Sensitivity to Climate

The climate's role essentially is that the further north the greater the heating load and the smaller the cooling requirements. Going south the reverse is true. To examine this effect, the thermal load conditions for the Market Square Complex were developed and then shifted to various points in the country.

Market Square Complex in Washington, D.C.

This project, being developed by the Pennsylvania Avenue Re-development Corporation, is situated on Pennsylvania Avenue and consists of approximately 1,300,000 square feet above grade and 900,000 below.

It was selected because of the functional diversity in the program calling for 56% residential area, 13% retail department stores and offices, and about 30% of national archives and community storage above ground. The remaining 900,000 square feet below grade are devoted entirely to national archives.

Situated between 7th and 9th Street, Market Square Complex forms the focal point of the entire proposed Pennsylvania Avenue Development project. It is conceived as an integrated major city project containing multi-use facilities incorporating energy conservation techniques and load management concepts. (See Appendix 2A)

The following tables summarize the load as shifted to different locations in the United States.

TABLE 2.1

DESIGN CONDITIONS FOR SELECTED LOCATIONS

Locations	Latitude	Winter Design Temp °F	Summer Design Temp °F		Daily Temp Range Summer	Equivalent Full load Hours EFLH Cooling	Heating Degree Days
St. Petersburg FL	28°	42	91	80	17	1500-2700	683
New Orleans, LA	30°	35	91	80	16	1400-2800	1385
El Paso, TX	32°	25	98	69	27	1000-1400	2700
Oklahoma City, OK	35°	15	97	77	23	1100-2000	3725
Corpus Christi, TX	28°	36	93	80	19	2000-2500	914
Denver, CO	40°	3	90	64	28	400- 800	6283
Chicago, IL	42°	1	92	76	20	500-1000	6639
Washington, DC	39°	19	92	77	18	700-1200	4224
New York, NY	40°	15	91	76	17	500-1000	4871
Boston, MA	42°	10	88	74	16	400-1200	5634
Minneapolis, MN	45°	-10	89	75	22	400- 800	8382
Seattle, WA	48°	32	79	65	19	400-1200	4424
Rapid City, SD	44°	-6	94	71	28	800-1000	7345
Des Moines, IA	41°30'	-3	92	77	23	600-1000	6588

TABLE 2.2

ANNUAL ENERGY REQUIREMENTS FOR COOLING

Location	Tons	Equivalent Full load Hours (EFLH)	Ton-Hours	BTU x 10 ⁶
St. Petersburg, FL	3442	2100	7,228,200	86,738
New Orleans, LA	3450	2100	7,245,000	86,940
El Paso, TX	3462	1200	4,154,400	49,853
Oklahoma City, OK	3219	1550	4,989,450	59,873
Corpus Christi, TX	3452	2250	7,767,000	93,204
Denver, CO	2245	600	1,347,000	16,164
Chicago, IL	3120	750	2,340,000	28,080
Washington, DC	3205	950	3,044,750	36,537
New York, NY	3100	750	2,325,000	27,900
Boston, MA	2937	800	2,349,600	28,195
Minneapolis, MN	3036	600	1,821,600	21,859
Seattle, WA	2233	800	1,786,400	21,437
Rapid City, SD	2725	900	2,452,500	29,430
Des Moines, IA	3206	800	2,564,800	30,778

TABLE 2.3

ANNUAL ENERGY REQUIREMENTS FOR HEATING

Location	Design Temperature °F	Degree-Days	Peak Heating Load (BTUH)	Annual Energy Requirements BTU x 10 ⁶
St. Petersburg, FL	42	683	6,998,000	4,412
New Orleans, LA	35	1385	8,845,000	8,909
El Paso, TX	25	2700	11,481,000	17,302
Oklahoma City, OK	15	3725	14,213,000	23,974
Corpus Christi, TX	36	914	8,580,000	5,882
Denver, CO	3	6283	17,706,000	41,076
Chicago, IL	1	6639	18,232,000	43,358
Washington, DC	19	4224	13,063,000	27,026
New York, NY	15	4871	14,213,000	31,350
Boston, MA	10	5634	15,672,000	36,536
Minneapolis, MN	-10	8382	21,416,000	55,234
Seattle, WA	32	4424	9,635,000	28,417
Rapid City, SD	-6	7345	20,267,000	48,279
Des Moines, IA	-3	6588	19,423,000	43,265

TABLE 2.4 ANNUAL ENERGY REQUIREMENTS FOR DOMESTIC HOT WATER/MILLION BTU

(Data Developed From Appendix A-2)

Location	Winter	Summer	Total
St. Petersburg, FL	3534	4154	7,688
New Orleans, LA	3283	3682	6,965
El Paso, TX	4758	3182	7,940
Oklahoma City, OK	5115	3885	9,000
Corpus Christi, TX	4115	3823	7,938
Denver, CO	8285	2217	10,502
Chicago, IL	8002	2918	10,920
Washington, DC	6062	3149	9,211
New York, NY	6888	4095	10,983
Boston, MA	7541	2652	10,193
Minneapolis, MN	8278	2112	10,390
Seattle, WA	8294	2241	10,535
Rapid City, SD	8252	1857	10,109
Des Moines, IA	7044	2422	9,466

TABLE 2.5

HEATING AND COOLING ANNUAL ENERGY BALANCE

Location	Heating Energy Requirements Btu x 10 ⁶	Winter DHW Energy Requirements Btu x 10 ⁶	Total Heat- ing Energy Requirements Btu x 10 ⁶	Heat Pump Cooling Capacity With COP = 3.2 Btu x 10 ⁶	Energy Requirement for Cooling Btu x 10 ⁶	Insufficiency in Ice For Cooling Btu x 10 ⁶	Supple- mentary Heat For Ice Melt- ing Btu x 10 ⁶
St. Petersburg, FL	4412	3534	7946	5463	86,738	81275	
New Orleans, LA	3909	3283	12192	8382	86,940	78558	
El Paso, TX	17302	4758	22060	15166	49,853	34687	
Oklahoma City, OK	23974	5115	29089	19999	59,873	39874	
Corpus Christi, TX	5882	4115	9997	6873	93,204	86331	
Denver, CO	41076	8285	49361	33936	16,164		17772
Chicago, IL	43358	8002	51360	35310	28,080		7299
Washington, DC	27026	6062	33038	22748	36,537	13789	
New York, NY	31350	6888	38238	26289	27,900	1611	
Boston, MA	36536	7541	44077	30303	28,195		2108
Minneapolis, MN	55234	8278	63512	43665	21,859		21806
Seattle, WA	28417	8294	36711	25239	21,437		3802
Rapid City, SD	48279	8252	56895	39115	29,430		9685
Des Moines, IA	43265	7044	50309	34587	30,778		3309

Based on Table 2.5 the following listing was constructed. In it, the various cities are ranked in order of closeness to absolute load balance as indicated by our calculations.

TABLE 2.6 SUPPLEMENTARY COOLING & HEATING REQUIRED

City	Insufficiency in ice for Cooling BtuX10 ⁶	Supplementary Heat Required For Ice Melting BtuX10 ⁶	(1) DD Heating	(2) DH Cooling
New York, NY	1,611		4871	3,000
Boston, MA		2,108	5634	2,500
Seattle, WA		3,802	4424	213
Des Moines, IA		3,809	6588	4,211
Chicago, IL		7,299	6639	3,100
Rapid City, SD		9,685	7345	4,100
Washington, DC	13,789		4224	4,200
Denver, CO		17,772	6283	4,055
Minneapolis, MN		21,806	8382	2,500
El Paso, TX	34,687		2700	8,000
Oklahoma City, OK	39,874		3725	11,200
New Orleans, LA	78,558		1385	9,500
St. Petersburg, FL	81,275		683	10,500
Corpus Christe, TX	86,331		914	12,000

- 1) ASHRAE Handbook & Product Directory
(1976 System p 43.2 - 43.6)
- 2) Extrapolated from Magazine

Based on this table the following preliminary assessments were made:
Outside of the following ranges, the applicability of the system is drastically reduced:

Heating Degree Days - 2,000 and 8,000

Cooling Degree Hours - 2,000 and 9,000

The following maps present these zones.

fig. 2-2
annual degree hours
dry bulb greater
than 78°F

In making these assessments the following qualifications have to be stated and cannot be over emphasized:

- 1) The climatic conditions presented on the maps are general and in investigating a specific location the exact climatic data for the site should be used.
- 2) While the dividing lines presented on the maps are very distinct it should be realized that in fact the impact is actually graduated. The lines were selected based on a first analysis of where the climatic conditions begin to take on significance.
- 3) The climatic conditions are only one factor to be considered and that there may be various other factors which offset an otherwise bad climatic impact. For example, if a heavy heating requirement existed year round (e.g. DHW for restaurants), it may provide sufficient ice generation for cooling in a climate otherwise too hot.

To summarize, the further north the more supplemental heat required to melt excess ice while the further south the more supplementary operation of a chiller to overcome a cooling deficiency.

The deficiency in cooling energy in southern areas could be reduced, however, by generating domestic hot water in the summer with the use of the heat pump operating with a COP of 3.79*.

For the New York area this would do away entirely with the deficiency in cooling energy. For the northern areas, the excess ice would permit the use of uninsulated tanks, but, in addition, solar panels would have to be used to melt this excess.

In the case of Boston, the surplus of ice is small and the lack of ice bin insulation would suffice to melt the small excess ice.

* Durham-Bush Screw Compressors at 35°F SST and 130°F SDT.

Thus, New York and Boston are two locations where the heating and cooling energy requirements approximately balance out, with New York requiring a small amount of additional cooling energy produced with domestic hot water in the summer and Boston requiring the melting of a small amount of ice which can be accomplished with the omission of insulation.

The dividing line is thus 40° to 42° latitude, or the 4871 to 5634 degree day range, with latitude below 40° and below 4871 degree day requiring supplementary compressor work in the summer, while latitude above 42° and above 5634 degree days, including Denver and Seattle, requiring supplementary heat in the winter to melt the excess of ice.

As for the ice storage bin, its size is governed by two distinct factors depending on the latitude. Thus, for the southern latitudes the ice storage bin size is governed by the ice generating capacity of the heat pump winter operation, while for the northern latitudes the size is governed by the cooling energy requirements.

The above results are emphasized in the attached graph (Fig 2.3).

fig. 2-3

HP-ICES PROJECT

ANNUAL ENERGY BALANCE

The ice storage bin volume versus latitude (Fig. 2.3) shows two distinct straight lines, one sloping upwards from the lower latitudes up to about 40° , and the other sloping downwards from about 42° latitude down towards higher latitudes.

The first upwards sloping line reflects the fact that from the lower latitudes up to about 40° , the volume of heat pump generated ice depends on the heating requirements, and connected with it, the ice producing capacity of the heat pump. These increase with latitude. This is the region of cooling deficiency where the ice volume is governed by the heating requirements.

From 40° latitude and up the higher latitudes the ice storage volume is governed by the cooling energy requirements alone, as there is too much ice produced by the heat pump winter operation. This is the region of surplus ice generation, and the graph slopes downwards indicate that the cooling requirements for summer drop as the latitude becomes higher.

The graph of compressor supplementary work slopes down from the lower latitudes to about 40° , indicating that in this region there is a deficiency in cooling which is larger at lower latitudes but decreases towards higher latitudes.

The last graph concerns supplementary heat and it starts from about 40° latitude and slopes upwards towards the higher latitudes. This is the region of surplus ice generated by the winter heat pump generation and requires the surplus ice to be melted by supplementary heat. The upward slope indicates the fact that this supplementary heat requirement increases with latitude.

The graphs of supplementary compressor work and supplementary heat required were superimposed on the graph of ice storage bin volume. This was done on purpose as important conclusions can be drawn from this superimposition.

For the lower latitudes, the ice storage volume line represents the cooling energy generated by the heat pump winter operation where the energy is not sufficient to cover the summer cooling requirements and supplementary compressor work is therefore necessary. Thus, the sum of the ordinates of the two graphs modified by suitable proportionality factors represents the total cooling requirements in the summer for each location in this region, (latitude 28° - 40°)

On the other hand, for the upper latitudes (42° - 48°) the ice storage volume represents the summer cooling requirements of each location in the region and reflects a certain amount of heating (obtained by a multiplying of the cooling requirements by $\frac{3.2}{2.2}$), done by the heat pump. The supplementary heat required likewise reflects a certain amount of heating performed by the heat pump (similarly obtained as above). Thus, the sum of ordinates of the two graphs, to a certain scale, represents the total heating requirements of each location in the region.

To emphasize again - in the lower latitudes the sum of ordinates of ice volume and compressor work yields, to a certain side, the total summer cooling requirements for each location, while in the upper latitudes the sum of the ordinates of ice volume and supplementary heat yield the total heating requirements for each location.

As for the domestic hot water energy requirements, for the southern

locations it will be produced both in summer and winter with the heat pump operating with a COP of 3.2* in winter and COP of 3.79 in summer.

For the northern latitudes the winter domestic hot water requirements will be produced by the heat pump operating with a COP of 3.2, while in the summer the domestic hot water will be produced by other means.

The domestic hot water energy requirements represent a sizable amount of the heating energy requirements. As far as domestic hot water is concerned the 14 chosen locations can be divided into 3 distinct groups:

- 1) Latitudes 28° - 30° , degree days 683 - 1385, where the domestic hot water requirements either exceed or constitute a very large percentage of heating energy requirements,
- 2) latitude 32° - 40° (with the exception of Seattle) degree days 2700 - 4871 where the domestic hot water constitutes 34% to 46% of the heating energy requirements, and
- 3) latitude 42° - 48° (with the exception of Denver) where the domestic hot water requirements constitute 19% to 28% of the heating energy requirements.

These results are depicted in the following table:

TABLE 2.7

<u>Location</u>	<u>Latitude</u>	<u>Degree Days</u>	<u>% of Heating</u>
<u>Group 1</u>			
St. Petersburg, FL	28°	683	DHW exceeds heating
Corpus Christi, TX	28°	914	Exceeds
New Orleans, LA	30°	1385	80%

*Dunham-Bush Screw Compressors 20°F SST, 130°F SDT

Table 2.7 Con't

<u>Location</u>	<u>Latitude</u>	<u>Degree Days</u>	<u>% of Heating</u>
<u>Group 2</u>			
El Paso, TX	32°	2700	46%
Oklahoma City, OK	35°	3725	38%
Washington, DC	39°	4224	34%
New York, NY	40°	4871	35%
Seattle, WA	48°	4424	37%
<u>Group 3</u>			
Chicago, IL	42°	6639	25%
Boston, MA	42°	5634	28%
Rapid City, SD	44°	7343	21%
Des Moines, IA	42°	6588	22%
Minneapolis, MN	45°	8382	19%
Denver, CO	40°	6283	26%

2.3 COMMUNITY COMPOSITION

Equal in importance to the load balance are the community characteristics. Different categories of land use (i.e. residential, commercial, institutional, and industrial) each have different thermal characteristics which affect the load on the system. The load peaks vary in type (i.e. heating or cooling), time of day, and season of year.

To examine this factor, simple energy requirements for heating, DHW, and cooling were developed for three types of buildings in Boston. The loads were derived from Boston Redevelopment Project, Park Plaza - Energy Report June 1975 by Dubin-Mindell-Bloome Associates and a theoretical community of 10 million square feet was assumed. The requirements were then adjusted for the climatological differences of three other cities in the U.S.: Minneapolis, MN., Seattle, Wa., and Oklahoma City, OK. Using the energy required for space heating and domestic hot water as the determining factor, net requirements were derived for each type in each location. Positive numbers indicate an excess of ice being generated during the winter while the negative numbers indicated the insufficiency of ice for summer cooling requirement and therefore need supplemental compressor work. Table 2.8 summarize the findings. Figures 2-4 to 2-7 present this same data, graphically demonstrating the relative loads of each type of building from 0% of the community to 100%. These graphs can also give quick visualization of the combinations possible. In most locations the residences were on the side of the graph indicating excess ice, while the office and retail were on the side indicating insufficiency. The example given in Figure 2-4 shows how 6 million SF (60% of 10 million SF) of residential space

can be offset by 7 million SF of office space (70%) plus 8.5 million SF of retail space (85%). This is only one possibility of the many feasible.

If each category of land use type is examined, the following observations can be made:

1) Residential

Single Family, Low Rise Apartment Building, High Rise Apartment Building - In all cases the residential HVAC load would primarily be during off-peak hours. Use of domestic hot water is greater compared to most other building types. The cooling load is a small percentage of the annual energy requirement.

2) Commercial

Office Building, Retail Store, Recreation

Office building and retail store - load during normal working hours, heavy for both heating and cooling. Office buildings generally have some cooling year round. Recreational essentially a non-work hours load and mostly space conditioning. However, there may be other heavy loads such as domestic hot water year round for restaurants.

-
- REF: I Based on observations in An Assessment of the Potential For District Heating In Four Major Cities by Energy Systems Research Group Inc., BNL/ANL, August 1978, Page 19 - 21.
- II A Test Case For The Potential Application of District Energy Systems Using Thermal Energy Cogenerated At Existing Electric Power Plants by P.J. Santini and A.A. Davis, Argonne National Laboratory, January 1978, Page 15 - 16.
- III Performance Report For The Aces Demonstration House August 1976 through August 1977 by Eugene C. Hise, Oak Ridge National Laboratory, Page 17.

3) Institutional -

School, Museums, Church, Hospital -

School - peak hour operation for the most part and high ventilation rates that are required by codes. Annually the cooling load is less because of fewer pupils in school during the summer.

Church, Museum - lesser demands at off-peak times especially weekends. Hospital - intensive user similar to high use residential phase, the characteristic of a commercial load super-imposed because most hospital functions are carried on during peak day hours.

4) Industrial -

Light Manufacturing, Warehouse, Garage -

In all of these categories the load would probably be for space heating and cooling. It may follow the work schedule or be continuous such as in a refrigerated warehouse.

2.4 POTENTIAL ENERGY INPUTS FOR ENERGY REQUIREMENTS

The graphs in Figure 2.8 represent the required electrical energy input in kWh to obtain a certain amount of cooling and heating, of heating alone and cooling alone. To obtain 100×10^6 Btu of cooling and heating

$$\text{KWH} = \frac{100,000,000}{5.4 \times 3413} = 5426$$

The COP's for heating and cooling used were 3.2 and 2.2 respectively, hence the overall COP = 5.4.

The application of HP-ICES to obtain heating and cooling was already shown energywise to be more efficient than producing the same amount of heating and cooling by conventional means.

fig. 2-7
OKLAHOMA CITY

fig 2-8 Electrical Energy Input Required To Obtain:

1. heating and cooling
2. heating only
3. cooling only

fig 2-9 Energy Cost for Heating and Cooling as a Function of Electricity Cost

fig 2-10 Energy Cost for Cooling as a Function of Electricity Cost

fig 2-11 Energy Cost for Heating as a Function of Electricity Cost

However, applying the HP-ICES for heating alone (without utilizing the cooling effect) could also be energy saving. A comparison of the COP_H for heating related to energy source with a conventional boiler efficiency, shows that $COP_H = 3.2 \times 0.31 = 0.992$, while the boiler efficiency is only 0.6. Capital outlay would have to be considered also, but the possibility of HP-ICES being more efficient for heating alone does exist, especially in areas with long and cold winters where the summer cooling is of secondary importance.

Similarly, in areas where the summers are long and hot the cooling alone by HP-ICES could also be considered without utilizing the heating (that is using cooling towers).

Although with ice generation the $COP = 2.2$, much lower than the COP for producing chilled water, under certain circumstances ice generation in the summer might be feasible.

Consider a location like New York City where the summer demand is high and it might be efficient to generate chilled water at night and store it diurnally for day use.

If the space is also expensive as certainly it might be in New York, the generation of ice at night time and storing it for use the following day would require 8 times less storage space than storing chilled water, the energy density of ice being considerably larger than that of chilled water. Thus the space and billing demand factors could easily outweigh the COP factor making the HP-ICES more energy efficient than conventional production of cooling. Again, all the first costs would have to be considered.

The graphs in Figure 2.8 will enable a quick estimate of the energy re-

quirements if HP-ICES is applied.

The graphs (Figure No. 2-9, 2-10, 2-11) give the potential energy cost for the energy requirements as a function of unit energy cost.

2.5 MARKET AVAILABILITY FOR ICE MAKING EQUIPMENT

All of the components are readily available in the U.S. The piping and valves are off-the-shelf items easily procured. The mechanical components also are readily available. The heat exchangers and other ancillary equipment are not limited to a specific type or brand but instead can be chosen from any appropriate equipment sized and selected according to normal engineering practice.

The ice bin essentially must be custom made but no peculiar construction materials are called for.

There is at the present time a fully developed and quite sophisticated technology of ice making, but it is geared primarily to the food freezing industry.

Some leading ice maker manufacturers, The Turbo Company in Texas and the European Stal Company (affiliated in the USA with the Vilters Company) have indicated that given the demand and the incentive they could easily make the required adjustments in their manufacturing equipment and production methods and go over to producing ice making equipment required by the HP-ICES application.

For Ice Makers there are three leading manufacturers:

1. Turbo-Dento, Texas, Plate Ice Makers, -

up to 160 tons of ice per day.

Future availability - up to 250 tons of ice per day.

2. Henry Vogt Machinery, Louisville, Kentucky,

Tube Ice Makers, P-48AL, 66 tons of ice per day.

3. Stal - Refrigeration Company, Sweden, affiliated in the USA
with the Vilters Company, up to 350 tons of ice per day.

As for compressors there is a variety of manufacturers for all type
of compressors and the most known among them are listed below:

1. Reciprocating Compressors 30-100 hp

Vilter Company (multi-cylinder compressors)

York Company (multi-cylinder compressors)

Reco-Refrigeration Equipment Corporation

Schnacke-Grasso

2. Screw Compressors - Direct Drive - 3550 r/min

Horsepower

Displacement

Vilter Company	100 hp-2000 hp	-	55 ft ³ /min to 4040 ft ³ /min.
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Freezing Equipment Sales (FES)		-	55 ft ³ /min to 3337 ft ³ /min.
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Reco-Refrigeration Equipment Corporation

Sullair Corporation		-	445 ft ³ /min to 1710 ft ³ /min
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Ducham-Bush		-	278 ft ³ /min to 16673 ft ³ /min.
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3. Centrifugal Compressors

Carrier - Industrial Centrifugal Compressors

Series 17MPS, 2, 3, and 4 stage compression.

TABLE 2.9

SCREW COMPRESSORS, TONNAGE, HORSE-POWER AND COST

DUNHAM - EUSH							FES - FREEZING EQUIPMENT SALES, INC.						
Model	20°F SST-130°F SDT			35°F SST-105°F SDT			Model	20°F SST-130°F DST			35°F SST-105°F DST		
	Ton	Motor HP	\$/Ton	Ton	HP	\$/Ton		Ton	Motor HP	\$/Ton	Ton	HP	\$/Ton
PC x 350 W(350)x2510	250	650	406	435	465	233	305	250	650	298	400	450	186
PC x 750 W(1100) 2515	450	1100	291	735	750	178	575	450	1000	258	750	750	155
--	900	Currently in the Design Stage					1160	900	2500	222	1450	1550	138

\$/Ton Based On System Selected With Recirculation Refrigeration System Integrated With The Compressor.

2.6 SYSTEM RELIABILITY

Since most of the equipment has already been used in other configurations, each component taken separately is without doubt reliable with enough information about its performance available from manufacturers and past applications.

However, the HP-ICES configuration mainly the ice maker, the ice conveying system and the annual storage ice bin are new and the efficient performance of these components within the overall framework is still little known.

An engineering evaluation of the whole system will reduce the doubt to a minimum but there will still remain some stages and processes in the overall system operation that will prove difficult to evaluate theoretically and that will require an examination in a down scale model with the gathered information being logged for engineering evaluation.

A down scale model will enable the exact evaluation of the behavior of ice in the bin over a longer period of time, the influence of the ambient conditions on the ice, the distribution system and the distribution density of the ice, all of which have so far been very little investigated.

2.7 ECONOMICS

Because of the large scale of an HP-ICES system, financing may present some hurdles to potential applications. The problems facing new construction are somewhat different from those facing retrofit situations. Such a system can be incorporated more easily into planning for new construction because the heating systems for the community is yet to be built. However, there still are problems. The developing agency, may be unwilling to assume responsibility for coordinating or operating a community energy system. Most important however, is the hesitation of financial institu-

tions to take risks on untried and novel technologies. For this problem, there is help. The present trend towards life-cycle costing should help to overcome part of this hesitancy. Studies on district systems already indicate reasonable time periods (7-10 years) for simple payback analyses. If the fossil fuel supplies continue to dwindle then the rate of increase in fuel prices will easily exceed the rate of inflation making the HP-ICES system an even better investment. Another source of help would be if governmental agencies (eg. HUD, DOE) help to reduce the risk either by some sort of loan guarantees or by grants to reduce the principal of the loan.

Retrofit application face even more hurdles. Even assuming universal acceptance (a big and tenuous assumption), installing such a system would be quite costly, the highest being in urban environments where thermal density makes it most effective. The cost of installing the distribution system and the cost of land required would be significant additions to the price of the system. One last additional question concerns paying for converting individual buildings to the system. Unless replacement of the prime system was already being considered, the owner would be quite reluctant to make the capital investment for such a changeover. This would severely restrict the efficiency of the system in the early stages of growth.

2.8 SYSTEM SIZE

The HP-ICES size will be influenced by the size of the community and other factors which will characterize the community like:

1. An Assessment of the Potential for District Heating in Four Major Eastern Cities: Washington, D.C., Philadelphia, Baltimore, Boston, by Energy Systems Research Group, Inc., BNL/ANL, August 1978.

1. Thermal density-defined as the total thermal load of the built-up area in a community divided by the total acreage of the community. If the built-up area is more concentrated the thermal density will increase. A high thermal density requires a smaller distribution system and vice-versa. For example, a residential area where zoning regulations limit the size of the building over a certain acreage, the thermal density will be low and the distribution system will be long and costly, all of which would make the system unfeasible. However, in a community of high rise apartment buildings (like Fresh Meadows in New York City) the thermal density would be high, the distribution system would be short, which would make the HP-ICES feasible.

Thus, the thermal density has a significant influence on the distribution system and on the energy requirements and size of the system.

The distribution system in turn being part of the overall system by its very size and length will influence the cost of the system.

2. Storage bin size requirements.

The sizing of the bin may cause engineering problems which can be overcome by modular expansion of optimum modular size. More difficult however, are real estate problems which in new communities can still be overcome by pre-design planning (i.e., integrating the storage bin with the building of the community). In retrofit communities and metropolitan areas (like New York City) these problems may not be solved precluding, thereby, the possibility of installing large HP-ICES, although from the point of view of thermal density and energy management such systems would be very desirable.

3. Funding and Financial Feasibility-

Once the system is established as energy efficient and energy savings

it would encourage capital investment. As large investments involve large financial risks it is rather difficult to predict the optimum size of HP-ICES which would make capital investments forthcoming. However, the steadily increasing cost of energy coupled with its decreasing availability would tend to make the system attractive to investors by promising a high rate of return. Also, the rising inflation would certainly have an encouraging effect on prospective investors. All told, it is difficult to predict the ultimate size of system, which can only be done on the merits of the three above cited major factors.

2.9 PROJECTIONS OF ENERGY SAVINGS FOR THE YEAR 2000

Introduction

During the heating mode in winter times the ice generated is essentially a by-product of the heat pump operation. As a result, it could be considered "free" energy. With the expected growth of the built-up urbanized areas the potential savings in refrigeration energy is quite impressive. The purpose of this section is to establish some reasonable figures of savings in refrigeration energy by the year 2000, utilizing this system.

Methodology

This analysis is based on data contained in an article - "FORECAST" which appeared in the September issue of the Electrical World magazine, published by McGraw Hill, and on data obtained directly from the National Gas Survey, Federal Power Commission, Washington D.C. and the Edison Electrical Institute Washington, D.C. and New York.

In the cited article the number of dwelling units was projected to the year 1995. This growth rate was then assumed to hold also for

heating and air conditioning system in the residential sector.

For the commercial sector the growth data on built-up areas was not available and, therefore, was instead calculated in an indirect manner. By comparing the annual KWH sales for the residential and commercial sectors, a ratio was derived and applied to the residential figures.

The projection to the year 2000 was obtained by assumption that the growth rate from 1995 to 2000 is approximately the same as from the year 1990 to 1995.

The energy usage for heating, domestic hot water and air conditioning were obtained by fuels for the year 1971 from the National Gas Survey (FPC, Volume 1 Chapter 8), and were projected to the year 2000 based on figures developed above.

Using the cooling energy requirements for the year 2000 and using a heat pump COP of 3.2., the heat derived was unable to satisfy the requirements for the whole winter season.

The remaining in heating energy required was calculated in two ways:

- a) by directly supplying heat by means of a boiler.
- b) by supplying the heat to the heat pump system. (See Appendix 2A, P. 191)

This second method could be looked upon as an assisted heat pump which generates all the ice it must to satisfy the whole heating requirement with supplemental heat melting any excess ice.

The total energy inputs required to satisfy both heating and cooling with each of the above methods (a) and (b) were then compared with the energy quantities required to satisfy heating and cooling by the conventional method, i.e. heating with boiler and cooling with electrically driven centrifugal chillers.

One final assumption, important to the amount of energy saved

was that there would be no institutional and economic impediments to applying the HP-ICES system.

SUMMARY

a. Residential Sector

The energy requirements for heating domestic hot water by the year 2000 for the whole of the United States are estimated at $13,317 \times 10^{12}$ Btu. The corresponding energy for cooling is estimated at 327×10^{12} Btu. From this amount of cooling the ice generating HP-ICES will produce 476×10^{12} Btu of heating. The remaining heating requirements if produced by an oil fired boiler will result in a total energy expenditure for both heating and cooling of 3187×10^6 barrels of oil and $4,012 \times 10^6$ barrels for supplemental heat through the heat pump. With the conventional system that is an oil fired boiler for heating and electrically driven chillers for cooling, the total energy expenditure for both heating and cooling will be 3268×10^6 barrels of oil. Thus, the savings in applying the HP-ICES concept with direct boiler heat amount to 81 millions barrels of oil. Taking into account the savings in electrically driven refrigeration auxiliaries the total savings amount to 89 million barrels of oil.

b. Commercial Sector

The heating and domestic hot water and cooling requirements are estimated to reach by the year 2000 for the whole of the United States 7626×10^{12} Btu and 804×10^{12} Btu respectively.

The heating obtained from the cooling by the heat pump amounts to 1169×10^{12} Btu. The remainder in heating requirement if produced by an oil fired boiler, the total energy input for both heating and cooling is 1733×10^6 barrels of oil and 2148×10^6 barrels if by heat pump and supplementary heat source.

By using conventional means the total energy input is 1935×10^6 barrels of oil. The savings by using the direct heating approach are $(1935-1733) \times 10^6$ or 202 million barrels of oil and including the refrigeration auxiliaries the saving amount to 223 million barrels of oil.

Industrial Sector

Industrial portion of energy consumption will contribute to saving but it is difficult to compute because it is hard to estimate what percent of industrial consumption can be successfully integrated with the HP-ICES system. The suitable part of this sector would probably be light manufacture that does not need a high temperature source of energy. As such it would, therefore, not represent a significant portion of that sector's energy use.

Total Savings

The total savings for the entire country can then be stated as 89×10^6 barrels + 223×10^6 barrels = 312×10^6 barrels of oil in the year 2000.

Correction factors must be applied to take into account the probability of less than universal utilization of the HP-ICES system.

The two factors considered were the climatological conditions and percent of urban environment. (Figs. 2-1, 2-2, & 2-12) Based on the maps of heating degree days and cooling degree hours another map (Fig. 2-13) was constructed indicating these extremes. Areas covered by only one extreme were reduced by 50% whereas areas where both occurred were totally eliminated from consideration since these systems would be cost effective only in urban areas, energy consumption in each zone was reduced by the number of people not residing in metropolitan areas.

HP-ICES PROJECT

The calculations used can be found in Appendix 2A Section 2A3

As a result of these corrections, the total saving for the whole of the United States in the year 2000 will be 165 million barrels of oil.

GEOGRAPHIC DIVISIONS OF THE UNITED STATES

fig. 2-12
Percent of Population in Metropolitan Areas

U. S. DEPARTMENT OF COMMERCE
BUREAU OF THE CENSUS

fig. 2-13 CLIMATIC INFLUENCES IN GEOGRAPHIC DIVISIONS

3. EXPECTED PERFORMANCE

3.1 FIRST AND SECOND LAW ANALYSIS

The ice generating HP-ICES is a new concept and in order to ascertain its advantages and benefits, a detailed thermodynamic analysis of the First and Second Laws was performed.

The First Law determines the balance between the energy inputs and outputs for any system, stating the equivalence of heat and work but does not address itself to the amount of work obtainable from a certain amount of heat.

This is the domain of the Second Law which limits the work available from a certain amount of heat, in dependence on the available high and low temperatures at which heat is added and rejected respectively. Since, however, the refrigeration cycle is reversed - relating to a heat engine cycle - the main effort towards operating such a cycle is to minimize the difference between the temperatures of heat absorption and heat rejection, thereby reducing the work input.

The analysis of the First Law was made to point out the components of the HP-ICES having the greatest heat loss and to compare them with the corresponding losses of a conventional system with the same output.

The Second Law analysis was performed to point out the availability losses of the various components of both systems.

3.1.1 The First Law Energy Balance

This analysis is based on the energy equation as applied to each com-

ponent of the system. The analysis is based on the same output for both HP-ICES and conventional systems:

1. Winter - 16,290 Btu/hr (based on 12,000 Btu/hr of heat absorption from 32°F water which was converted to ice).
2. Summer - 12,000 Btu/hr supplied to the user.
3. Heat pump operation at COP = 3.2 (D-B screw compressors).
4. The conventional chiller operating at COP = 4.4 (Carrier).
5. Distribution piping - 10,000 Ft.
6. Friction losses - 5 Ft. per 100 Ft.
7. Pump efficiency 0.7
8. Hot water temperature differential of HP-ICES and boiler is 30°F.
9. Pressure drop for the air distribution system 2" WG.
10. Fan efficiency 0.6.
11. Condenser pump in conventional system working on a 80 Ft. head and 0.7 efficiency.
12. Condenser water temperatures 85°F - 95°F
13. For the conventional system, chilled water temperatures 42°F - 58°F and for the HP-ICES 32°F - 58°F.

Findings and Conclusions

The main heating losses occur at the major components of the system, like at the heat pump for the HP-ICES, at the boiler and chiller for the conventional system.

FIG. 3-1 FIRST LAW ANALYSIS BALANCE FOR HP-ICES
SUMMER MODE

FIG.-3-2 FIRST LAW ENERGY BALANCE ICE GENERATION HP-ICES
WINTER MODE - BASED ON 12000 BTU/HR WATER HEAT OF FUSION

**FIG.-3-3 FIRST LAW ENERGY BALANCE FOR CONVENTIONAL BOILER SYSTEM
WINTER MODE**

**FIG. 3-4 FIRST LAW ENERGY BALANCE FOR CONVENTIONAL CHILLER
SUMMER MODE**

	<u>Conventional</u>	<u>HP-ICES</u>
Winter Losses	8,727 Btu/hr	Neglected
Summer Losses	17,500 Btu/hr rejected to cooling tower	823 Btu/hr ice bin losses
Annual	26,227 Btu/hr	823 Btu/hr

The overall annual COP

$$\text{HP-ICES} = 1.063$$

$$\text{Conventional} = 0.572$$

accounting for all the inputs and outputs and related to source energy.

(See attached Diagrams). It follows that the HP-ICES is approximately twice as efficient as the conventional system (See Fig. 3.1 and 3.4).

3.1.2 Second Law Analysis

The Second Law analysis gives a quantitative account of what happens to available energy in a thermodynamic cycle and directs attention to significant losses.

The energy balance on the basis of the First Law and the general energy equation account for all energy quantities, but it does not reveal those transformations where the greatest losses of available energy occur.

Since the conservation of available energy is desired, a knowledge of the most serious losses is helpful and sometimes leads to ideas for measures of conservation.

Energy is available in the thermodynamic sense when it can be converted completely into mechanical work. The availability of any form of energy is the measure of the possibility of transforming that form

of energy into mechanical work.

Availability is defined by the equation:

$$\phi = -(\Delta H_{i-a} - T_a \Delta S_{i-a})$$

where the availability measures the work extractable from an energy stream in a steady flow process. It is the maximum amount of shaft work that can be extracted from a unit mass of matter as it flows into equilibrium with the atmosphere.

ϕ is always measured with respect to a rest state that in our usage will be 70°F and one atmosphere.

T_a is therefore 530°R

H_{i-a} is the enthalpy change of a unit mass of a material traversing the apparatus relative to a rest state.

S_{i-a} is the entropy change of unit mass of material traversing the apparatus relative to a rest state.

If a material moves through a flow process from an initial state (1) to a final state (2) that is not the atmospheric rest state, then the change in availability

$$\Delta \phi_{1-2} = \phi_2 - \phi_1$$

$$\Delta \phi_{1-2} = (H_2 - H_1) - T_a (S_2 - S_1) \text{ or}$$

$$\Delta \phi = H - T_a \Delta S$$

Analysis

This analysis will compute the losses in availability of the ice generating HP-ICES in winter and summer modes of operation and will find the availability loss in various stages of the cycle starting with the heat pump source and finishing with the end users. The analysis will be based on one

ton of refrigeration at the heat source that is for 12,000 Btu extracted from the water.

Basic Assumptions

1. Heat pumps operating conditions: SST - 20°F, SDT - 130°F
COP = 3.2 (Ref. Dunham-Bush Screw Compressor Model E(HP) x 2516)
2. $T_a = 70^\circ\text{F} = 530^\circ\text{R}$
3. Piping distribution system 10,000 Ft.
4. Friction pressure drop 5 Ft. per 100 Ft.
5. Piping heat loss 2°F over 10,000 Ft.
6. Pump efficiency 0.7
7. Water temperature drop $\Delta T_w = (120 - 90) = 30^\circ\text{F}$
8. Air temperature difference $\Delta T_a = (90 - 70) = 20^\circ\text{F}$
9. Pressure drop in air distribution system = 2" WG
10. Fan efficiency 0.6

Availability Calculations

$$\Delta \phi = -[\Delta H - T \Delta S]$$

where ΔH and ΔS refer to constant pressure conditions.

For water $\Delta H = C \Delta T$, $\Delta S = C \times \ln \frac{T_a}{T_i}$ for one lb.

$$\Delta \phi = - \left[1 \times (T_a - T_i) - T_a \ln \frac{T_a}{T_i} \right] \times \dot{m}$$

For air at constant pressure

$$\Delta S = C_p \ln \frac{T_a}{T_i}$$

FIG. 3-5 Φ = AVAILABILITY ANALYSIS ICE GENERATION HP-ICES

BASED ON 12,000 BTU/HR WATER HEAT OF FUSION
WINTER MODE

AVAILABILITY LOSSES:

$$1 \rightarrow 1A \quad \Delta\Phi = \Sigma\Phi_{1A} - \Sigma\Phi_1 = (-1057.24 + 1291.6) - (5454 + 224.7 + 119.2) = -5563.54 \text{ BTU/HR}$$

$$2 \rightarrow 2A \quad \Delta\Phi = \Sigma\Phi_{2A} - \Sigma\Phi_2 = (1192.7) - (534 + 1291.6) = -632.9 \text{ BTU/HR}$$

$$3 \rightarrow 3A \quad \Delta\Phi = \Sigma\Phi_{3A} - \Sigma\Phi_3 = (302 + 224.7) - (1009 + 1192.7 + 0) = -1675 \text{ BTU/HR}$$

$$4 \rightarrow 4A \quad \Delta\Phi = \Sigma\Phi_{4A} - \Sigma\Phi_4 = (0) - 302 = -302 \text{ BTU/HR}$$

FIG.-3-6 Φ = AVAILABILITY ANALYSIS ICE GENERATION HP-ICES
SUMMER MODE

AVAILABILITY LOSSES

$$1 \rightarrow 1A \quad \Delta\Phi = \Phi_{1A} - \Phi_1 = (660.4 + 119.2) - (-1057.24 + 63.6) = 1773.24 \text{ BTU/HR}$$

$$2 \rightarrow 2A \quad \Delta\Phi = \Phi_{2A} - \Phi_2 = (625.3) - (423.5 + 660.4) = -458.6 \text{ BTU/HR}$$

$$3 \rightarrow 3A \quad \Delta\Phi = \Phi_{3A} - \Phi_3 = (63.66 + 82.76) - (743 + 36) = -632.59 \text{ BTU/HR}$$

$$4 \rightarrow 4A \quad \Delta\Phi = \Phi_{4A} - \Phi_4 = (36 - 82.76) = -46.76 \text{ BTU/HR}$$

**FIG.-3-7 AVAILABILITY ANALYSIS OF CONVENTIONAL BOILER
WINTER MODE**

Ref. Temperature - 25°C - 77°F.

Flow rate equal to that for heat pump - 581.3 lb/hr

$$\phi_{90^\circ\text{F}} = - \left[1(537 - 550) - 537 \left[\ln \frac{537}{550} \right] \right] \times 581.3 = 90.02 \text{ Btu/hr}$$

$$\phi_{120^\circ\text{F}} = - \left[1(537 - 580) - 537 \left[\ln \frac{537}{580} \right] \right] \times 581.3 = 950.4 \text{ Btu/hr}$$

$$\phi_{\text{Fuel inlet}}^* = \frac{(581.3 \times 30)}{1000 \times 0.6} \times 953.77 = 27745.2$$

REF. FROM: M. V. SUSSMAN AVAILABILITY ANALYSIS

Page number 61

We take the availability of fuel air mixture at inlet to burner as

$$= 189,951 \text{ cal./gm.-mole CH}_4$$

which is equivalent to 21,388.33 Btu/lb or 953.77 Btu/ft³

FIG.-3-8 ϕ = AVAILABILITY ANALYSIS FOR CONVENTIONAL CHILLER

BASED ON 12,000 BTU/HR CHILLED WATER SYSTEM
SUMMER MODE

AVAILABILITY LOSSES:

$$1 \rightarrow 1A = \phi_{1A} - \phi_1 = (574.5 + 842.13) - (2730 + 103.3 + 306.9) = -1723.57 \text{ BTU/HR}$$

$$2 \rightarrow 2A = \phi_{2A} - \phi_2 = (534) - (688.6 + 574.5) = -729.1 \text{ BTU/HR}$$

$$3 \rightarrow 3A = \phi_{3A} - \phi_3 = (82.76) - (743 + 36 + 534) = -1230.24 \text{ BTU/HR}$$

$$4 \rightarrow 4A = \phi_{4A} - \phi_4 = (36) - (82.76) = -46.76 \text{ BTU/HR}$$

$$H = C_p \Delta T$$

$$= - \left[0.24(T_a - T_i) - T_a \times 0.24 \ln \frac{T_a}{T_i} \right] \times \dot{m}$$

For ice

$$= - \left[1(T_a - 492) + 144 - 530 \times \left[\frac{(-\ln T_a)}{492} + \frac{(-144)}{492} \right] \right] \times \dot{m}$$

Findings and Conclusions

In comparing the availability losses for the ice generating HP-ICES with those for the conventional system of boilers and chillers, main attention was concentrated on major components of both systems, namely 12,000 Btu/hr for cooling and 17,454 Btu/hr for heating.

The following results were obtained:

	<u>Conventional</u>	<u>HP-ICES</u>
Winter Losses	$\phi = -26,884.82 \text{ Btu/hr}$	$\phi = -5563.51 \text{ Btu/hr}$
Summer Losses	$\phi = -1,723.52 \text{ Btu/hr}$	$\phi = +1773.24 \text{ Btu/hr}$
Annual Losses	$\phi = -28,608.34 \text{ Btu/hr}$	$\phi = -3790.27 \text{ Btu/hr}$

The conclusion can be drawn that considering only the major components of the system, the losses in availability for the conventional system are 7.5 times as great as those for the HP-ICES (See Fig. 3.5 to 3.8).

3.2 PERFORMANCE OF HP-ICES VS. A CONVENTIONAL SYSTEM (CASE STUDY)

3.2.1 Selection of a Conventional System-(Washington, D.C.)

a. Equipment Selection

Heating: Two 10,000 MBH No. 2 oil fired boilers to cover heating peak load of $13.063 \times 10^6 \text{ Btu/hr}$. One

3,000 MBH No. 2 oil fired boiler to cover domestic hot water requirement of $2.95^* \times 10^6$ Btu/hr.

Cooling: Four 800 ton electrically driven centrifugal compressors and chillers to cover cooling load of 3200 ton. Four 800 ton cooling towers.

b. Energy Analysis

Energy Requirements:

$$\begin{aligned} \text{Heating and DHW } 27,026 \times 10^6 + 9,211 \times 10^6 \\ = 3.6237 \times 10^{10} \text{ Btu year} \end{aligned}$$

$$\text{Cooling} = 3,045,000 \text{ Ton-hr/year}$$

Annual Energy Consumption

$$\frac{36,237 \times 10^{60}}{140,000 \times 0.7} = 370,000 \text{ gallons of No. 2 oil/year}$$

Electrical energy input to the chillers is $3,045,000 \text{ Ton-hrs} \times 1\text{KW}^*/\text{Ton}$
 $= 3,045,000 \text{ kWh}$ (the cost of operation of chilled water pumps is not considered since their use applies to all the schemes).

Thus, annual energy input

$$\text{Heating} = 370,000 \text{ gallons of No. 2 oil}$$

$$\text{Cooling} = 3,045,000 \text{ kWh}$$

*Based on 0.81 KW/Ton for Carrier Hermatic Centrifugal Chillers, 19EA and 0.2 KW/Ton for condenser water pumps and cooling tower fans.

3.2.2 The Performance of Ice Generating HP-ICES
 (Washington, D. C.)

a. Energy Analysis

Winter Energy Requirements:

The annual energy requirements for heating were seen to be

* (See Page 187)

2.7026×10^{10} Btu per year and for DHW 0.9211×10^{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.

Thus: DHW Winter = $6,062 \times 10^6$ Btu/per year

DHW Summer = $3,149 \times 10^6$ Btu/per year

The heating and winter DHW requirements are therefore:

$$2.7026 \times 10^{10} + 0.6062 \times 10^{10} = 3.3088 \times 10^{10} \text{ Btu/year}$$

b. Summer Energy Requirements

Summer Cooling Load = 3,200 Ton

Summer energy requirements with 950 equivalent full load hours assumed:

Refrigeration = 3.045×10^6 Ton-hours

DHW = 0.3149×10^{10} Btu

c. "Cooling" Energy Available From Heat Pump

Using a heat pump with 20°F compressor suction temperature at 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:

$$\frac{3.3088 \times 10^{10}}{3.2} \times 2.2 \times 1/12,000 = 1.896 \times 10^6 \text{ Ton-hours}$$

which is

* Dunham-Bush Screw Compressors

$$\frac{1.896 \times 10^6}{3.05 \times 10^6} \times 100 = 62.2\% \text{ of summer cooling requirements.}$$

d. Refrigeration Energy Generated in the Summer from DHW

There is a deficiency in "cooling" energy of $(3.045 - 1.896) \times 10^6 = 1.149 \times 10^6$ 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 0.3149×10^{10} Btu. Therefore, the quantity of "cooling" energy that can be produced is:

$$\frac{3149 \times 10^6 \times 2.2}{3.2} = 2165 \times 10^6 \text{ Btu}$$

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 3.79.*

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

$$\frac{0.3149 \times 2.79 \times 10^{10}}{3.79} = 2318 \times 10^6 \text{ Btu}$$

$$\text{or } \frac{0.2318 \times 10^{10}}{12,000} = 1.9317 \times 10^5 \text{ Ton-hours}$$

Thus the total energy available for cooling is

$$\begin{aligned} & (1.896 + 0.1932) \times 10^6 \text{ Ton-hours per year} \\ & = 2.0892 \times 10^6 \text{ Ton-hours per year} \end{aligned}$$

* Dunham-Bush Screw Compressors for 35°F SST and 130°F SDT

e. Sizing and Storage Ice Bin

It is assumed that the entire refrigeration energy generated by the heat pump in the winter produces ice after the chilled water has been cooled sensibly from 57°F to 32°F. Thus the ice generated is

$$\frac{1.896 \times 10^6 \text{ Ton-hours} \times 12,000 \text{ Btu/Ton-hours}}{169 \text{ Btu/lb.}}$$

$$= 1.3463 \times 10^8 \text{ lbs. of ice.}$$

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

$$\frac{1.3463 \times 10^8}{50} = 2.692 \times 10^6 \text{ cubic feet}$$

$$\text{This yields } \frac{2.692 \times 10^6}{1.896 \times 10^6} = 1.42 \text{ cubic feet of ice storage}$$

per ton-hour. Comparing with chilled water storage with a 15°F temperature difference

$$\frac{62.4 \times 15}{12,000} = 0.078 \text{ ton-hours per cu. ft.}$$

$$\text{or } \frac{1}{0.078} = 12.8 \text{ cu. ft. per ton-hour}$$

Thus for the same refrigeration energy capacity the ice needs

$$\frac{12.8}{1.42} = 9.01 \text{ less storage volume than chilled}$$

water with a 15°F temperature difference.

f. 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:

$$\left[\frac{2.692 \times 10^6}{20} \right]^{\frac{1}{2}} = 367 \text{ feet square}$$

This yields a top and base area of 135,000 square feet each and an area of 30,000 square feet for the side walls. The bin will be insulated with 4 inch poly-urethane on the top (R-36.4) and with 2 inch poly-urethane (R-18.2 on the sides and bottom).

The losses were calculated by months and tabulated in Table 3.1.

TABLE 3.1 ICE BIN LOSSES
LOSSES IN BTU X 10⁸

	<u>Top</u>	<u>Sides</u>	<u>Bottom</u>	<u>Total</u>
June	0.99	0.263	0.90	2.153
July	1.02	0.348	1.01	2.378
August	0.911	0.294	1.08	2.285
September	0.697	0.393	1.05	2.140
October	0.511	0.393	1.00	1.904
November	0.33	0.311	0.66	1.301
December	0.23	0.250	0.87	1.35
January	0.23	0.178	0.008	0.416
February	0.307	0.121	0.67	1.098
March	0.33	0.120	0.73	1.18
April	0.70	0.142	0.74	1.582
May	0.912	0.210	0.832	1.594
				<u>19.741</u>

Thus, the total annual losses amount to:

$$\frac{19,741 \times 10^5}{12,000} = 165,000 \text{ ton hours}$$

which is: $\frac{165,000}{1.896 \times 10^6} \times 100 = 9\%$ of total "cooling" energy generated.

g. Net Available "Cooling" Energy

$$2,089,000 - 165,000 = 1,924,000 \text{ ton hours}$$

Deficiency in refrigeration energy

$$3,045,000 - 1,924,000 = 1,121,000 \text{ ton-hours}$$

h. Annual Energy Consumption

Winter electrical energy input is:

$$\frac{3.3092 \times 10^{10}}{3.2 \times 3413} = 3,030,000 \text{ kWh}$$

Summer DHW Generation

$$\frac{0.3149 \times 10^{10}}{3.79 \times 3413} = 243,500 \text{ kWh}$$

Summer electrical energy input to make up the deficiency

in "cooling" energy of 1,121,000 ton-hours using 1 kWh/Ton-hour

$$= 1,121,000 \text{ kWh}$$

$$\text{Subtotal for summer} = 1,364,500 \text{ kWh}$$

$$\text{Total for year} = 4,394,500 \text{ kWh}$$

3.2.3 Recapitulation of Results

Table 3.2 Energy Input

	W I N T E R		S U M M E R	
	Heating	DHW	Cooling	DHW
Conventional	370,000 gallons of No. 2 oil	Included in the heating load	3,045,000 kWh	Included in the heating load
HP-ICES	3,030,000 kWh	Included in heating load	1,121,000 kWh	243,500 kWh

The energy cost analysis will be made when economics are considered. However, source energy input will be calculated at this stage. Thus, assuming a heat rate of 11,000 Btu/kWh, the source energy inputs are as follows:

Conventional

$$\begin{aligned}
 \text{Winter} & \quad 370,000 \text{ gallons} \times 140,000 \text{ Btu/gal.} = 51,800 \times 10^6 \text{ Btu} \\
 \text{Summer} & \quad 3,045,000 \text{ kWh} \times 11,000 \text{ Btu/kWh} = 33,495 \times 10^6 \text{ Btu} \\
 \text{Total} & \quad = 85,295 \times 10^6 \text{ Btu}
 \end{aligned}$$

HP-ICES

$$\begin{aligned}
 \text{Winter} & \quad 3,030,000 \times 11,000 = 33,330 \times 10^6 \text{ Btu} \\
 \text{Summer} & \quad 1,364,500 \times 11,000 = 15,010 \times 10^6 \text{ Btu} \\
 \text{Total} & \quad = 48,340 \times 10^6 \text{ Btu}
 \end{aligned}$$

Thus the HP-ICES uses about half as much source energy as the Conventional System.

TABLE 3.3 SUMMARY OF ANNUAL ENERGY INPUT

	KWH	OIL GALLONS	SOURCE ENERGY BTU
Conventional System	3,045,000	370,000	$85,295 \times 10^6$
HP-ICES	4,394,500	-	$48,340 \times 10^6$

3.2.4 The Annual Overall COP For The HP-ICES

The COP of this system is 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:

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

(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

$$\begin{aligned} & 2.703 \times 10^{10} \text{ Btu} + 0.9211 \times 10^{10} \text{ Btu} \\ & = 3.6241 \times 10^{10} \text{ Btu per year} \end{aligned}$$

Cooling produced is 3.045×10^6 ton-hours per year

$$= 3.045 \times 10^6 \times 12,000 = 3.654 \times 10^{10} \text{ Btu per year}$$

$$\text{Total heating and cooling} = 7.2781 \times 10^{10} \text{ Btu per year}$$

Annual kWh input is 4,394,500 kWh per year

$$= 4,394,500 \times 3413 = 1.500 \times 10^{10} \text{ Btu per year}$$

$$\text{Annual overall COP} = \frac{7.2781 \times 10^{10}}{1.500 \times 10^{10}} = 4.85$$

$$\text{Related to source energy the COP} = 4.85 \times 0.31 = 1.50$$

4. EXPECTED ECONOMICS

4.1 INTRODUCTION

To analyse the expected economic feasibility of the HP-ICES, the concept was applied to the Market Square Project and compared with a conventional system.

The energy consumption figures used were those arrived at in the Section on Expected Performances.

The Conventional System is a central system selected for comparison capable of supplying the same heating, domestic hot water and cooling requirements as the HP-ICES.

Although the methods of generation of heating and cooling are different for the two systems, the distribution systems were assumed to be the same.

4.2 ANALYSIS

The first costs include the generation equipment and distribution systems, but do not include the end users equipment (air handling units).

Life of equipment was considered 20 years, life of the distribution system and ice bin 40 years, cost of money 10%, escalation of operation cost 12%.

The cost per million BTu was calculated by dividing the annual owning and operating cost by the energy produced (both heating and cooling) per year. This was done for both the conventional and HP-ICES. The procedure was repeated for 5, 10, 15 and 20 years. The results show that with time the cost per million BTu favors the HP-ICES. The escalation rate was varied between 12% and 18%, and the interest rate was varied between 10% and 18%. For each combination the pay-back period was obtained.

4.2.1 Energy Cost (See Section 3.2 for energy quantities) (Case Study)Conventional System

a) Winter

370,000 gallons of No. 2 oil at \$0.45/gallon yields cost of \$166,500.

Heating and DHW distribution energy 172,500 kWh

Energy cost $\$0.02158/\text{kWh} \times 172,500 = \$ 3,725$

Demand charge $20 \text{ KW} \times \$3.15/\text{KW} \times 7 \text{ mo.} = \underline{440}$

Subtotal = \$ 4,165

Total cost winter = \$166,500 + \$4,165

= \$170,665

b) Summer

Cooling

Energy charge $\$0.02261/\text{kWh} \times 3,045,000 \text{ kWh} = \$68,847$

Demand charge $3200 \text{ KW} \times \$5.25/\text{KW} \times 5 \text{ mo.} = \underline{\$84,000}$

Subtotal = \$152,847

Chilled water distribution system 533,000 kWh

Energy charge $\$0.02261/\text{kWh} \times 533,000 \text{ kWh} = \$ 12,050$

Demand charge $165 \text{ KW} \times \$5.25/\text{KW} \times 5 \text{ mo.} = \underline{\$ 4,330}$

Subtotal = \$ 16,380

Domestic hot water 66,500 kWh

Energy charge $\$0.02261/\text{kWh} \times 66,500 \text{ kWh} = \$ 1,505$

Demand charge $20 \text{ KW} \times \$5.25/\text{KW} \times 5 \text{ mo.} = \underline{\$ 525}$

Subtotal = \$ 2,030

Total cost for summer

= \$152,847 + \$16,380 + \$2,030 = \$171,257

$$\begin{aligned}
 \text{c) Total Winter and Summer} &= \$170,665 + 171,257 \\
 &= \$341,922 \text{ say } \$342,000
 \end{aligned}$$

HP-ICES

a) Winter

Heat Pump

$$\text{Energy } 3,030,000 \text{ kWh} \times \$0.02158/\text{kWh} = \$65,387$$

$$\text{Demand } 3 \times 875 \text{ HP} \times 0.746 \text{ KW/HP} \times \$3.15 \times 7 \text{ mo.} = \$43,179$$

$$\text{Subtotal} = \$108,566$$

$$\text{Ice Maker Pump (1 KW) } 5,000 \text{ kWh} \times \$0.02158 = \$108$$

$$\text{Heating and DHW Distribution Energy (as for conventional system)} = \$4,165$$

$$\text{Total for Winter } \$108,566 + \$108 + \$4,165 = \$112,839$$

b) Summer

$$\text{Energy } 1,364,500 \text{ kWh} \times \$0.02261/\text{kWh} = \$30,851$$

$$\text{Demand } 3 \times 670 \text{ HP} \times 0.746 \text{ KW/HP} \times \$5.25/\text{KW} \times 1.5 = \$11,808$$

$$\text{Subtotal} = \$42,659$$

$$\text{Chilled Water Distribution System} = \$16,380 \text{ (as for conventional)}$$

$$\text{Domestic Hot Water} = \$2,030 \text{ (as for conventional)}$$

$$\text{Total for Summer } \$42,659 + \$16,380 + \$2,030$$

$$= \$61,069$$

$$\text{c) Total Winter and Summer} = \$112,839 + \$61,069$$

$$= \$173,908$$

$$\text{say } \$174,000$$

4.2.2 First Cost (Case Study)Conventional System

a.	Four 800 ton centrifugal compressors	=	\$480,000
b.	Four cooling towers	=	\$120,000
c.	Two 10,000 MBH boilers	=	\$120,000
d.	One 3,000 MBH boiler	=	\$ 30,000
e.	Two chilled water pumps	=	\$ 16,000
f.	Two hot water pumps	=	\$ 6,000
g.	Chilled water distribution system (based on 20,000 linear feet of 6" diameter pipe)	=	\$800,000
h.	Hot water distribution system (based on 20,000 linear feet of 3" diameter pipe)	=	\$400,000

Total = \$1,972,000

Controls and other = \$ 28,000

\$2,000,000

HP-ICES

a.	Ice Bin Excavation (120,000 cu. yds. @ \$50/cu.yd.)	\$ 60,000
b.	Ice Bin Construction (2,700,000 cu. ft. @ \$0.50/cu.ft incremental cost; ice bin is part of the underground struc- ture)	\$1350,000

Ice Bin Subtotal \$1,410,000

c.	Three Dunham-Bush screw compressors including condensers, chillers, oil separators and fluid accumulators (450 ton at 20° F SST and 130° F (SDT))	= \$ 300,000
d.	Six ice makers with a total capacity of 1,200 tons	= 780,000
e.	Cooling towers with condenser pumps	= 100,000
f.	Two chilled water pumps	= 16,000
g.	Two hot water pumps	= 6,000
h.	Chilled water distribution system (as for conventional)	= 800,000
i.	Hot water distribution system (as for conventional)	= 400,000
	Subtotal (equipment and distribution system)	= \$2,402,000
	Total	= \$3,812,000

4.2.3 Operating Cost (Case Study)

Conventional System

a.	Energy cost	= \$342,000
b.	Maintenance (1% of first cost of equipment)	= 8,000
c.	Labor, 8 stationary engineers at \$20,000 p.a.	= 160,000
d.	Administration 6 people @ 15,000 p.a.	= 90,000
	Total	= \$600,000

HP-ICES

a.	Energy cost	= \$174,000
b.	Maintenance (1% of equipment)	= 12,000
c.	Labor, 8 stationary engineers	= 160,000
d.	Administration 6 people	= 90,000
	Total	= \$436,000

TABLE 4-1 - RECAPITULATION

	<u>Conventional System</u>	<u>HP-ICES</u>
First Cost	\$2,000,000	\$3,812,000
Energy Cost	\$ 342,000	\$ 174,000
Operation Cost	\$ 600,000	\$ 436,000
Present Worth (with 12% escalation over 20 years; 10% rate of interest)	\$14,927,000	\$13,076,000

HP-ICES PROJECT

4.2.4 Comparative Analysis

a. Incremental investment of HP-ICES

relative to conventional	\$3,812,000
	<u>- \$2,000,000</u>
	\$1,812,000

b. Savings in operating cost by using

HP-ICES relative to conventional	\$ 600,000
	<u>- 436,000</u>
	\$ 164,000

c. Simple Pay-back

=	<u>\$1,812,000</u>
	\$ 164,000
=	11 Years.

d. Discounted "pay-back" based on 12%

escalation of energy cost and 10% cost
of money is 11 years.

e. The rate of return over a life cycle of

20 years with 12% escalation of energy
cost is 18%.

TABLE 4-2

Pay-back sensitivity with escalation rate and cost of money.

Cost of money				
Escalation	10%	14%	16%	18%
12%	11½	14½	16½	20
14%	10½	12½	14½	16½
16%	9½	11½	12½	14½
18%	9	10½	11½	12½

4.2.5 Cost per Million BTU

Conventional System

Operating cost		\$ 600,000
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Amoritization of equipment

(10%, 20 years CRF = 0.11746)	= \$ 94,000
-------------------------------	-------------

Amoritization of distribution system

(10%, 40 years, CRF = 0.10226)	= \$ <u>123,000</u>
--------------------------------	---------------------

Annual owning & operating cost	\$ 817,000
--------------------------------	------------

Energy requirements per year

Heating	27,000x10 ⁶ BTU
---------	----------------------------

Cooling	37,000x10 ⁶ BTU
---------	----------------------------

DHW	<u>9,000x10⁶ BTU</u>
-----	---------------------------------

	73,000x10 ⁶ BTU
--	----------------------------

The cost per million BTU

	= \$ <u>817,000</u>
--	---------------------

73,000	= <u>\$11.2 / 10⁶ BTU</u>
--------	--------------------------------------

HP-ICES

Operating cost	= \$ 436,000
----------------	--------------

Amoritization of equipment	= \$ 141,000
----------------------------	--------------

Amoritization of distribution system

and ice bin	= \$ <u>267,000</u>
-------------	---------------------

Annual owning & operating cost	= \$ 844,000
--------------------------------	--------------

The cost per million BTU

	= \$ <u>844,000</u>
--	---------------------

73,000	= \$11.6 / 10 ⁶ BTU
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Table 4-3 Owning and Operating Cost per Million BTU

Conventional System			HP-ICES		
Year	AO & OC	Cost per 10 ⁶ BTU Dollars	AO & OC	Cost per 10 ⁶ BTU Dollars	Difference in cost per 10 ⁶ BTU
1-st	817,000	11.2	844,000	11.6	-0.4
5-th	1,274,400	17.45	1,146,380	15.70	1.75
10-th	2,080,500	28.50	1,762,150	24.10	4.40
15-th	3,501,140	48.00	2,794,474	38.20	9.80
20-th	6,004,800	82.25	4,613,800	63.20	19.05

4.3 Conclusions

The economic analysis shows that the HP-ICES has a smaller operating cost than the conventional system by \$164,000 and a smaller energy cost by \$168,000. Its first cost is by 1,812,000 greater than the first cost of the conventional system. The simple pay-back is 11 years and the pay-back as a function of both escalation rate and cost of money is tabulated in Table 4.2. The tabulated results show that the discounted pay-back with consideration of the escalation rate is not far off from the simple pay-back and centers around 11-12 years.

The owning and operating cost per million BTU favors the conventional system in the first year, but as the years go on, the escalation (12%) increases the cost for both systems, but the rate of increase is greater for the conventional system than it is for the HP-ICES, thus making the owning and operating cost per million BTU cheaper for the HP-ICES than for the conventional system.

The rate of return was calculated for 12% escalation over a period of 20 years and it amounts to 18%.

5. EXPECTED ENVIRONMENTAL IMPACTS

5.1 AIR POLLUTION

The heat pump centered community storage system does not cause any direct or localized air pollution. The nature of the system - two closed loops for heating and cooling - precludes any interaction with the environment.

5.1.1 Heating Cycle

Conventional systems (i.e. coal, oil or gas-fired), in contrast, require the use of a boiler which, in turn, can cause air pollution. Emissions from these conventional systems can range from carbon monoxide to sulfur dioxide to nitrous oxide to particulates.

A heat pump can cause indirectly some air pollution problems due to its electricity requirements. The pollution would stem from the power station generating this electricity. The pollution, however, would be confined to the immediate vicinity of the power station which would probably be equipped with sophisticated controls and means to combat the pollution. (See Section 5.1 for the overall effect).

5.1.2 Cooling Cycle

A conventional cooling system requires the use of a cooling tower which can create thermal pollution problems and chemical corrosion of the surroundings. The cooling cycle of the HP-ICES system is a by-product of the heating cycle, thus there exists no "cooling generation" in the system for summer needs. As previously mentioned, the cycle is completed in two closed loops. There is no thermal air

pollution associated with the HP-ICES system.

5.1.3 Peak Load and the Overall Effect

The overall effect of a HP-ICES system is a reduction in air pollution since the total electricity requirements from the power station for an HP-ICES system are considerably less than those required for a conventional system to produce an equal amount of hot and chilled water.

Most power stations are summer peaking. The equipment brought on line for this seasonal demand is generally less efficient and more polluting (i.e. combustion turbines) than the base load equipment. The seasonal storage of the HP-ICES reduces the peak load in that there is no "cooling generation" necessary in the summer. Thus, less pollution is generated from the power station. The HP-ICES system also indirectly reduces other forms of air pollution; the power station's fuel requirements are lowered and, in turn, less air pollution is generated by a reduction in transportation needs.

5.1.4 Refrigerant Leakage

The flat plate ice maker used in the HP-ICES system is easily maintained. Furthermore, it is designed so that leaks are easily detected. Due to the centralization and community-orientation of the system, it is probable that the entire system will be maintained better than a large number of small, individualized systems.

5.2 WATER POLLUTION

5.2.1 Normal Operation

Due to the closed loops, there also exists no thermal or chemical water pollution in a HP-ICES system. If a conventional system requires the use of a river or other waterway for operation, the temperature of the water will be increased (depending on the flow of the river and other factors). The resultant thermal pollution may be harmful to the habitat of the waterway. In the summer time less energy is produced at the power plant. Therefore, less heat is rejected into rivers, lakes or by the cooling towers and therefore the thermal pollution is not severe.

Leaks from the storage bin also would cause no pollution problems, as the bin contains only water. If drainage would become necessary, the water would flow into the local storm system.

The lack of any adverse effects from our HP-ICES system can be contrasted to the problems associated with the coil bin HP-ICES system.

The coil bin system utilizes a coil containing brine in the storage bin. If drainage would become necessary with this system, the brine fluid could cause pollution problems when it is drained into the storm system.

5.3 NOISE POLLUTION

Screw compressors used in the HP-ICES system operate with much less noise than centrifugal compressors used in conventional systems which often cause

damage to hearing unless drastically controlled. Thus, we do not foresee any noise pollution problems with the HP-ICES system.

5.4 RESOURCE CONSUMPTION

5.4.1 Land Utilization

The storage bin for the HP-ICES system is underground. It can be located underneath or beside the building. If located beside the building, a garden, park, or even parking lot, for examples, may be placed above it.

For the basic system, no cooling tower is required. If the system is altered to include a cooling tower, the tower would be considerably smaller than those used for conventional systems.

The total space requirements for the HP-ICES system (including the mechanical room, the storage bin, and the ice maker) are less than those required for a conventional system (with boilers, chillers, cooling towers, etc.).

5.4.2 Distribution of Piping Network

The piping network is necessarily underground. If the HP-ICES system is used for a new, undeveloped area, the network can be easily integrated with the community design.

Problems may arise if the system is to be used in a previously developed area. If there exists no central system for both hot and chilled water in these areas, it would be necessary to open the streets for installation. Thus, institutional problems (such as transportation,

inconvenience, etc.) would arise as with installation of any centralized system. These problems are especially aggravated in urban areas. When a centralized system for both hot and chilled water already exist in a given area, the HP-ICES system can be used with no alterations to the piping network.

5.4.3 Construction

No special materials are necessary to construct a HP-ICES system.

All materials are standard building items available today on the market.

5.4.4 Retirement

There are no environmental problems associated with retirement of the system. The storage bin, for example, can easily be transformed into an underground parking structure and the mechanical room can be used for another system.

6. PROJECTED GROWTH

Unlike other systems, the HP-ICES does not depend on any natural resources (well water, lakes, rivers, etc.) and therefore the growth of our HP-ICES system can be modularly expanded without being dependent on outside factors. Because of its many possible configurations, the HP-ICES system is quite flexible in its approach to growth. As a result, there are many factors which would affect this process. The analysis specifies some possible modes of expansion and then goes on to review the impact of these variables on the growth process. The expansion of the system can follow the increase in the population and the built up area by the modular addition of independent storage space for the required ice storage bin and the addition of space for new mechanical rooms.

The essence of this technique is that for every load that is added into the loop, units of mechanical systems and storage are also added on. This could be done by adding equipment at the central plant or adding subsidiary stations at the periphery of the loops. Central planning would help by insuring that the design of the system originally would take into consideration the projected growth that the system would have to accomodate. The limitations most clearly are that of the distribution system piping. Whenever possible the distribution systems should be viewed beforehand with a view of the projected growth of the community for a period not exceeding the

expected life of the distribution system.

The extension of the distribution system in relation to the extension of the community size and its building density may present a difficult problem when institutional limitations to the built-up density exist. Also, a distribution system extension may require the cooperation of private property owners which may involve a large capital expenditure. These two factors may contribute towards making the extension of the distribution system uneconomical.

Depending on the rise of growth of new communities, relation to the existing communities and also depending on the location of new communities with respect to existing, the expansion of the system could also be obtained by expanding the equipment of the existing system or building a new central plant with its own distribution network to accomodate the growth of the community.

Expanding the system by adding another complete modular unit it is easiest to maintain efficient use of all parts of the system within their design parameters. However, this method calls for strong central planning which would be capable of significant investment of funds in a system which would have to be underutilized until the design density is achieved or be capable of instantaneous development to that density. Lower density levels would probably be easier to arrive at using this technique, even though high density per unit of distribution is still the most efficient use. In its organizational arrangements the ICES system by nature lends itself to a central authority structure. As such,

central planning and the coordination with other community services are an advantage to growth becoming more of a necessity the more significant increases. Concomitant with this are problems of responsiveness and flexibility as the economies of scale approach limits to the rate of return. The key here is what sort of organization operates the system. It may be organized as a utility, either in the form of a private company or as a service agency for the community itself. It may also be operated by the developer as a part of the original development. All of these possibilities have different techniques of arriving at a decision on growth and expansion. Perhaps, more important is the spectrum of weight given to political as opposed to technical criteria in such decision making. Another consideration is whether the system is part of a new construction or retrofit in an existing community. New construction would be in a more speculative environment and probably more fiscally conservative. On the other hand, it should be more decisive and planned because the decision making process is more centralized. Retrofit, however, because it is in an existing community would almost always be with the involvement of the political or governmental entity. As such, the decisions would tend to lean towards political influences. The growth of the community does not present insurmountable problems for an ICES system. There is enough flexibility and variety in configuration to allow for most contingencies. The engineering possibilities are not that difficult, certainly not as the political influences will be.

7. SYSTEM MODIFICATION

7.1 SYSTEM MODIFICATION

The modification of the system should conform to local climatic conditions which primarily determine the heating and cooling loads.

7.2 MODIFICATION FOR SUPPLEMENTARY HEAT

In the northern areas where the ice generated exceeds the summer cooling requirements or if the ice generated in quantities just sufficient to cover the summer cooling requirements would result in a deficiency in heating, supplementary heat has to be supplied either by melting the ice or supplied directly to heat the building.

There are many means to remedy this condition, and one of the most efficient ways is to use inexpensive solar radiators-collectors which are $\frac{1}{2}$ " diameter aluminum extruded fin-tube pipe as shown in the figure below:

These finned tubes can be installed vertically on east, south, and west walls of a building and can collect solar energy which, when elevated by the heat pump to the required heating temperature, can supplement the deficiency in heating requirements. In Minneapolis, for example, such solar radiators can collect about 186,400 Btu/square feet per season. These

collectors operating at low temperatures (sometimes even lower than the ambient temperatures) are sufficient for the purpose of ice melting and do not require any insulation or encasing.

Moreover, these collectors can also collect thermal energy by convection and conduction.

This modification of the HP-ICES system to accommodate the radiators is relatively simple and inexpensive. Also, the radiators' system can be integrated in the overall construction of the building as a shading device thereby reducing the solar heat gain in the summer.

In less severe climates the heat pump can use outside air down to an outside temperature of 40°F as a heat source using an evaporative condenser acting as an evaporator and only below that temperature will ice generation begin. In such cases, the ice generated will be sufficient for the summer cooling requirements. Such a system could prove, however, slightly more complicated and require some modification in the system-like controls to prevent freezing problems on the surface of the coils. The use of the above modification has to be investigated in light of the local considerations.

7.3 MODIFICATION FOR SUPPLEMENTARY COOLING

In the southern areas 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 energy could be made up by freezing water or any other phase change material (with freezing tempera-

ture suitable for air conditioning operation) by circulating cold brine through it. The brine itself is 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, the other located in water. Whenever the air is 25°F or below it cools the brine which in turn cools the water, thus generating ice.

However, if the above method cannot be used because of relatively high winter temperatures, chilled water must be generated in the summer at off peak periods and the ice bin storage could serve as diurnal chilled water storage. This would considerably reduce the summer electric peak demand which is in keeping with our main purpose of cutting such a demand.

The diesel and gas engines could also be used in southern areas, especially if the billing demand charge is high. In such cases, the domestic hot water could be generated with the heat recovered from the engines.

Where space is scarce, ice could be generated at night and stored for use during the day.

Two examples presented in Appendix 7A were taken from our other design projects to show the possible modifications of HP-ICES.

The first one will show the use of outside air to produce the supplementary ice to do away with the deficiency in cooling.

The second project will show the utilization of HP-ICES system with solar radiators supplementing the deficiency in heating requirements.

In cold climates a diesel or gas engine could be used for the heat pump drive increasing considerably the heating and the overall COP's due to the

considerable amount of heat recovered from the exhaust and jacket water. For example, for the diesel engine the COP for heating related to source energy is 1.62 and related to shaft work, 4.61. The overall COP related to source energy is 2.39 and related to shaft work is 6.8.

The corresponding figures for the electrical drive are 1.55 and 5.4 respectively. The gas engine drive has a COP for heating related to source energy of 1.47 and related to shaft work of 4.87. Similarly, the corresponding overall COP's are 2.13 and 7.07 respectively.

The discrepancies in the COP's are due to different engine efficiencies and different proportions of the recovered heat.

Figures 7.1 and 7.2 compare the source energy requirements to obtain both cooling and heating, or heating only for the diesel engine, gas engine and electrically driven ice generating HP-ICES and conventional system. (See also Appendix 7A).

The diesel and gas engine drives were considered to operate with the same suction and discharge temperatures as the electrical drive. However, with the series arrangement of the heat pump condenser, the jacket water heat exchanger and exhaust silencer, we can reduce the condenser discharge temperature, thereby improving the performance of the heat pump cycle and, consequently, the performance of the whole system.

For example, reducing the discharge temperature by 10°F decreases the horse power input by about 10% and, at the same time, improves the COP's by the same percentage.

The condenser will account for about 60% of the hot water temperature difference while the recovered heat from the engine will account for 40% of

fig 7-1 Heating and Cooling Obtained by Various Systems

fig 7-2 Heating Obtained by Various Systems

the upper part of the hot water temperature difference. Actually, a trial and error approach will be necessary to determine the exact division of heat coming from the heat pump and the heat recovery.

8. COMPONENT TESTING REQUIREMENTS

8.1 PRE-DEMONSTRATION TEST

At the heart of this Integrated Community Energy System is the heat pump, the ice maker and the annual storage ice bin. Since the components of the system are known and have been tested separately in other systems and configurations and have been used in other commercial and industrial applications there is no need to perform any additional tests on these components. The performance data are readily available.

The relatively new aspect, however, is the annual cycle storage. Other seasonal storage systems already constructed may provide some information, but none have the equipment arrangement of this system. Its two main components are the ice maker and the ice bin. It is this aspect of the overall system that should be tested to gather more performance data.

The Ice Maker

The ice maker itself doesn't require any particular tests. It is used in a configuration that was designed to make ice for the Food Preservation Industry. Information on performance and optimization is easily available from the manufacturers.

Ice Bin

While present thermal storage systems are concerned with retaining heat, the ice bin requires that heat be kept out and deals with a phase change process in addition. As there are no large size ice bins designed for the purpose of annual storage, it is particularly important to examine

the integration of the ice maker with the ice storage bin and chilled water system loop from and into the ice storage bin.

The configuration of the components within this storage system is novel and testing of a scaled down mock-up should be done. This will enable evaluation of various configurations as well as the testing of the bin at the same time. The mock-up should examine the influence of the following variables:

Ice Delivery to the Bin

If it is desired that the bin be not directly under the ice maker, various configurations of conveyors can be tested for suitability. Temperature measurements along the travel path will provide criteria for determining optimum travel distances, shape of the ice and construction of the conduit to prevent heat gains.

Similarly, the conveying of the ice and its distribution in the ice bin should be investigated to see if the design density of the ice could be achieved. The conveying system of the ice from the ice maker to the bin has to be tested to check if it performs at minimum loss.

Likewise, the insulation required for the conveyor and the conveying methods (either a pneumatic or helical screw conveyor) will have to be determined. If a pneumatic conveyor is used, the air temperature and the losses caused by the warm air will have to be examined. To avoid losses, the air might have to be precooled or else a closed loop between the air exhaust from the storage bin and conveyor power fan established.

In large ice storage bins the ice can block part of the outlet to the chilled water pump suction and may cause non-uniform flow to the pump. To

avoid this, the layout and location of chilled water suction pipes within the bin have to be tested to secure that the number of suction points and suction surface required are adequate so as to secure a constant and unhindered flow to the chilled water loop.

It may be feasible to secure a layer of water beneath the ice at the bottom of the bin at all times so that the ice will be distributed homogeneously in the bin and not block the suction. This problem will also have to be examined to determine the amount of water required to achieve this condition and the influence of this condition on the volume of the bin.

The behaviour of the ice in the bin throughout the year under different ambient conditions (temperature, partial vapor pressure), will similarly have to be tested and logged in order to achieve all the information of the ice bin losses which will enable optimal bin insulation.

8.2 TESTING PROCEDURES

Once a system has been installed, a multilevel test of the system should be utilized. Because actual equipment and sub-systems have not at this time been selected, only general procedures can be outlined. This will cover testing of individual components, various sub-systems when installed and finally the system itself in the various modes of operation.

8.2.1 Component Testing

Testing of the various individual components should be performed at the factory or on-site according to procedures contained in industry standards (e.g. ASHRAE). If performed at the factory, complete records of the

procedure and the results should be forwarded with the other documentation.

The following standards shall be specifically utilized:

<u>Compressors</u>	ASHRAE Standard 23-67 compressors, positive displacement, refrigerant methods of testing for rating.
<u>Chillers</u>	ASHRAE Standard 30-78 liquid chilling packages, methods of testing for rating.
<u>Condensers</u>	ASHRAE Standard 22-71 water-cooled refrigerant, condensers methods of testing for rating.
<u>Ice Maker</u>	ASHRAE Standard 29-71 ice makers, methods of testing.
<u>Ice Bin</u>	ASHRAE Standard 94-77 methods of testing thermal storage devices based on thermal performance.
<u>Expansion Valves</u>	ASHRAE Standard 17-7 expansion valve, refrigerant, method of rating and testing.

All other equipment shall also be tested utilizing whatever industry standards are applicable.

8.2.2 Sub-system Testing

After proper installation the individual sub-systems shall be tested for performance. The testing will be in the following sequence:

a) Cooling Tower

The cooling tower will be tested for proper performance in accordance with procedures in Cooling Tower Institute ATP-105.

b) Refrigeration Operation

Each of the three heat pump units shall be tested using the cooling tower for heat rejection. Suitable instruments shall be attached at points determined by the designer in each of the units

to measure the actual load profiles for comparison with design parameters. Each unit shall be tested in the following modes at normal operating speeds and temperatures:

i) chilled water production:

- chilled water flow
- chilled water temperature
- automatic response to cooling load

ii) ice production:

- weight of ice produced per unit of ice making operation
- upper and lower limits of ice maker

c) Heat Production

Each of the heat pump units shall be tested using the ice making mode for the heat sink. Suitable instruments shall be attached at points determined by the designer to measure actual load profiles for comparison with design parameters. The unit shall be tested in the following mode at normal speeds and temperatures:

i) All three units tested for hot water production:

- hot water flow
- hot water temperature
- automatic response to space heating load

ii) One unit with the double bundle condenser heat for domestic hot water:

- hot water flow to heat exchanger

- hot water temperature to heat exchanger
- rise in temperature in domestic hot water tank per unit of operation

d) Ice Bin Cold Water Pumping

This system for supplying chilled water for cooling from the ice bin is to be tested with all heat pumps shut down, the valves connecting the system open and chilled water circulating pump operating. All elements should be operating at normal temperature and suitable instruments at points determined by the designer to measure the actual load profile for comparison with design parameters for:

- chilled water flow
- chilled water temperature
- change in bin temperature as a result of cooling load.

8.2.3 System Check

Once the various sub-systems have been checked and their performance verified, check of the system essentially entails balancing and check of the system controls. By manipulating the various sensor input, different climatic and elemental conditions can be simulated and the response of the system tested.

APPENDIX 1A

1A.1 REFRIGERANT INVESTIGATION

Single Stage Compression and Two Stage
Compression With IntercoolingRefrigerant - 12

Evaporator Temperature	=	20°F
Condenser Temperature	=	130°F
Evaporator Pressure	=	35.736 PSIA or 21.040 PSIG
Condenser Pressure	=	195.71 PSIA or 180.01 PSIG
Optimum Pressure For Intercooling		

$$P_1 = \sqrt{P_S P_D} = \sqrt{35.736 \times 195.71}$$

$$= 83.628 \text{ PSIA}$$

Enthalpies (See Fig. 1A.1)

$h_1 = h_g$ at 20°F	=	79.385 Btu/lb	
$h_2 = 85.6$ Btu/lb			Isentropic compression to intermediate pressure of 83.628 PSIA
$h_3 = 92.5$ Btu/lb			Isentropic compression to discharge pressure 195.71 PSIA
$h_4 = h_g$ at 83.628 PSIA	=	84.270 Btu/lb	
$h_5 = 91.2$ Btu/lb			Isentropic compression from intermediate to discharge pressure
$h_6 = h_7 = h_f$ at 195.71 PSIA	=	38.553 Btu/lb	

Single Stage Compression

$$\begin{aligned}\text{Work} &= h_3 - h_1 = 92.5 \text{ Btu/lb} - 79.385 \text{ Btu/lb} \\ &= 13.115 \text{ Btu/lb}\end{aligned}$$

$$\begin{aligned}\text{Refrigerating Effect} &= h_1 - h_7 = 79.385 - 38.553 \\ &= 40.832 \text{ Btu/lb}\end{aligned}$$

$$\begin{aligned}\text{Refrigerant Flow Per Ton} &= \frac{200 \text{ Btu/Min}}{40.832 \text{ Btu/lb}} \\ &= 4.898 \text{ lb/min per ton}\end{aligned}$$

Coefficient of Performance (COP)

$$\begin{aligned}&= \frac{h_1 - h_7}{h_3 - h_1} = \frac{79.385 - 38.553}{92.5 - 79.385} \\ &= \frac{40.832}{13.115} = 3.11\end{aligned}$$

Alternately

$$\begin{aligned}\text{COP} &= \frac{200 \text{ Btu/Min}}{13.115 \text{ Btu/lb} \times 4.898 \text{ Lb/Min}} \\ &= 3.11\end{aligned}$$

Energy Efficiency Ratio (EER)

$$= \text{COP} \times 3.413 = 10.626 \text{ Btu/h/Watt}$$

$$\begin{aligned}\text{Power} &= 13.115 \text{ Btu/lb} \times 4.898 \text{ Lb/Min} \\ &= 64.237 \text{ Btu/Min Per Ton} \\ &= \frac{64.237 \text{ Btu/Min} \times 778 \text{ Ft-Lb/Btu}}{33,000 \text{ Ft-Lb/Min/Hp}} = 1.514 \text{ HP/Ton}\end{aligned}$$

$$\begin{aligned}\text{Volume Flow} &= 4.898 \text{ Lb/Min} \times 1.0988 \text{ Cu. Ft./Lb (at } 20^\circ\text{F)} \\ &= 5.382 \text{ CFM/Ton}\end{aligned}$$

fig1A.1 Pressure Enthalpy Diagram

fig1A.2 Two Stage Compression With Water Intercooling

Two Stage Compression With Water Intercooling

First Stage Compression (Fig. 1A.2)

$$\text{Work} = h_2 - h_1 = 85.6 - 79.385 = 6.215 \text{ Btu/lb}$$

Second Stage Compression

$$\text{Work} = h_5 - h_4 = 91.2 - 84.270 = 6.930 \text{ Btu/lb}$$

$$\text{Total} = 13.145 \text{ Btu/lb}$$

$$\text{Refrigerating Effect} = 40.832 \text{ Btu/lb}$$

$$\text{Refrigerant Flow Per Ton} = 4.898 \text{ Lb/Min}$$

$$\text{COP} = \frac{40.832}{13.145} = 3.106$$

$$\text{Or COP} = \frac{200}{13.145 \times 4.898} = 3.106$$

$$\text{EER} = 3.106 \times 3.413 = 10.602 \text{ Btu/h/Watt}$$

$$\text{Power} = 13.145 \times 4.898 = 64.384 \text{ Btu/Min Per Ton}$$

$$= \frac{64.384 \times 778}{33,000} = 1.518 \text{ Hp/Ton}$$

$$\text{Volume Flow} = 5.382 \text{ CFM/Ton First Stage}$$

$$= 2.378 \text{ CFM/Ton Second Stage}$$

Two Stage Compression With Refrigerant Intercooling (See Fig. 1A.3)

Heat and mass balance at intercooler

$$m_6 + 1 = m_4$$

$$m_6 h_6 + m_2 h_2 = m_4 h_4$$

$$m_6 38.553 + 1 \times 85.6 = m_4 \times 84.270$$

$$\text{Yielding } m_4 = 1.029 \text{ Lb}$$

From Condenser, $m_6 = 0.029 \text{ Lb of Saturated Liquid}$

fig 1A.3 Two Stage Compression With Intercooling
With Liquid Refrigerant

First Stage Compression

$$\text{Work} = h_2 - h_1 = 85.6 - 79.385 = 6.215 \text{ Btu/Lb}$$

Second Stage Compression

$$\text{Work} = m_4(h_5 - h_4) = 1.029 \times 6.930$$

$$= 7.132 \text{ Btu/Lb}$$

$$\text{Total} = 13.347 \text{ Btu/Lb}$$

$$\text{Refrigerating Effect} = 40.832 \text{ Btu/Lb}$$

$$\text{Refrigerant Flow Per Ton} = 4.898 \text{ Lb/Min (through the evaporator)}$$

$$\text{COP} = \frac{40.832}{13.347} = 3.059$$

$$\text{Or} = \frac{200}{13.347 \times 4.898} = 3.059$$

$$\text{EER} = 10.442 \text{ Btuh/Watt}$$

$$\text{Power} = 13.347 \times 4.898 = 65.37 \text{ Btu/Min Per Ton}$$

$$= \frac{65.37 \times 778}{33,000} = 1.541 \text{ Hp/Ton}$$

Volume Flow

$$\text{First Stage} = 5.382 \text{ CFM/Ton}$$

$$\text{Second Stage} = 4.898 (1.029) \times 0.48555 = 2.447 \text{ CFM/Ton}$$

TABLE A.1.1 RECAPITULATION OF PERFORMANCE OF
R-12 PER ONE TONE REFRIGERATION

PERFORMANCE ITEMS	SINGLE STAGE COMPRESSION	TWO STAGE COMPRESSION	
		WATER INTERCOOLING	REFRIGERANT INTERCOOLING
Work Per Lb Of Refri- gerant Passing Evapor.	13.115 Btu	13.145 Btu	13.347 Btu
Refrigerating Effect	40.832 Btu/lb	40.832 Btu/lb	40.832 Btu/lb
Refrigerant Flow Per Ton	4.898 Lb/Min	4.898 Lb/Min	5.04 Lb/Min 2-ND Stage
COP	3.11	3.106	3.059
EER	10.626 Btuh/Watt	10.626 Btuh/Watt	10.442 Btuh/Watt
Power Expenditure	1.514 Hp/Ton	1.518 Hp/Ton	1.541 Hp/Ton
Volume Flow Per Ton	5.382 CFM	1. 5.382 CFM 2. 2.378 CFM	1. 5.382 CFM 2. 2.447 CFM
Discharge Temperature	145°F	138°F	138°F

TABLE A.1.2
SUMMARY OF TWO STAGE COMPRESSION PERFORMANCE FOR EVAPORATOR
TEMPERATURE OF 20°F AND CONDENSER TEMPERATURE OF 130°F

REFRI- GERANT	PRESSURE PSIA		COM- PRESSION RATIO	COP		HP/TON			REFRIGERANT FLOW PER TON.					DIS - CHARGE TEMP. °F
									LB/MIN *		CFM **			
	P _S	P _D		INTER- COOLING	REF.	INTER- COOLING	REF.	FIRST STAGE	SECOND STAGE	FIRST STAGE	SECOND STAGE	INTER- COOLING	REF.	
R-11	4.35	38.67	8.9:1	3.85	3.80	1.23	1.24	3.37	3.37	3.37	28.6	1.14	1.26	137
R-12	35.74	195.71	5.5:1	3.11	3.06	1.52	1.54	4.90	4.90	4.90	5.38	2.38	2.45	138
R-22	57.73	311.50	5.4:1	3.24	3.14	1.45	1.51	3.49	3.49	3.49	3.27	1.43	1.55	150
R-113	1.53	18.45	12.1:1	-	-	-	-	4.23	-	-	76.14	-	-	-
R-500	41.96	231.90	5.5:1	3.14	3.08	1.50	1.53	4.09	4.09	4.09	4.61	2.02	2.10	140
R-502	67.16	335.54	5.0:1	2.49	1.82	1.85	1.90	6.09	6.09	6.09	3.71	1.65	1.74	140
R-717	48.21	330.00	6.8:1	3.57	3.39	1.32	1.39	0.48	0.48	0.49	2.82	1.14	1.26	208

* THROUGH THE EVAPORATOR

** AT COMPRESSOR SUCTION

1A.2 REFRIGERATION COMPRESSOR PERFORMANCE

TABLE 1A.3

COMPARISON OF PERFORMANCE OF E(HP) x 2516 DUNHAM-BUSH
SCREW COMPRESSOR USING R-12 AND R-22 REFRIGERANTS.

SUCTION TEMP. = 20°F DISCHARGE TEMP. 130°F

(Based on 10°F Liquid Sub-cooling and 10°F Suction Superheat)

REFRIGERANT	$T_s = 20^\circ\text{F}/T_D = 130^\circ\text{F}$						
	REFRIGERANT CAPACITY		POWER REQUIREMENT			HEAT REJECTED	COOLING
	TONS	BTU/H	HP	HP/TON	BTU/H	BTU/H	C.O.P.
R-12	275	3,300,000	595	2.16	1,514,000	4,814,000	2.18
R-22	450	5,400,000	1025	2.28	2,609,000	8,009,000	2.07

This comparison shows that the performance of R-12 per ton is better than that of R-22, i.e. lower Hp/Ton and a higher COP which is a conclusion drawn from theoretical analysis (see Table above and also the written summary, Page 24).

However, the same compressor yields a much higher capacity when operating with R-22 than when operating with R-12, something which is not apparent from theoretical considerations alone.

One partial explanation could be the fact that the CFM/Ton for R-22 is smaller than the CFM/Ton for R-12, therefore, a given compressor having a certain displacement can handle more R-22 per unit time than R-12 thus

providing a larger tonnage.

TABLE 1A.4

COMPARISON OF PERFORMANCE OF E(HP) x 2516 DUNHAM-BUSH
SCREW COMPRESSOR USING R-12 AND R-22 REFRIGERANTS.

SUCTION TEMP. = 35°F, DISCHARGE TEMP. 105°F

(Based on 10°F Liquid Subcooling and 10°F Suction Superheat).

REFRIGERANT	$T_s = 35^\circ\text{F}/T_d = 105^\circ\text{F}$						
	REFRIGERANT CAPACITY		POWER REQUIREMENT			HEAT REJECTED	COOLING C.O.P.
	TONS	BTU/H	HP	HP/TON	BTU/H	BTU/H	
R-12	450	5,400,000	460	1.02	1,171,000	6,571,000	4.60
R-22	735	8,820,000	760	1.03	1,934,000	10,754,000	4.56

This comparison brings out the fact of R-22 having a much larger refrigerating capacity than R-12 when both are used in the same compressor, also with a suction temperature of 35°F and discharge temperature of 105°F as encountered in ordinary air conditioning work.

R-22 has a slightly higher Hp/Ton and lower COP than R-12 which differs slightly from theoretical results for these conditions.

Both R-22 and R-12 show a much better performance at 35°F suction and 105°F discharge temperatures than at 20°F and 130°F suction and discharge temperatures respectively which could be expected.

TABLE 1A.5

REFRIGERANT - 12. SUCTION TEMPERATURE 20°F

DISCHARGE TEMPERATURE 120°F

COMPARISON OF PERFORMANCE OF CARRIER MPS. SIZE 40

CENTRIFUGAL COMPRESSOR VS. 2516 DUNHAM-BUSH SCREW

COMPRESSOR.

TYPE OF COMPRESSOR	REFRIGERATION		POWER - INPUT			HEAT	COP	COP
	TONS	BTUH x 10 ⁶	BHP	HP/TON	BTUH x 10 ⁶	REJECTED BTUH x 10 ⁶	COOLING	HEATING
Centrifugal	1,480	17.76	2442	1.65	6.215	23.975	2.86	3.86
Screw	300	3.6	540	1.80	1.374	4.974	2.62	3.62

This comparison was made with a discharge temperature of 120°F as data on centrifugal compressors with a discharge temperature 130°F were unavailable. The comparison shows that centrifugal compressors have a superior performance over screw compressors.

However, for a 130°F discharge temperature screw compressor are available. Also, for a range 20°F to 35°F suction and 105 - 120°F discharge tem-

peratures, as required by our application, screw compressors are superior as centrifugal compressor would have to be of two stages, that is of large size. With gear boxes for speed change, the speed changeover from one suction temperature to another is difficult to attain.

TABLE 1A.6

REFRIGERANT - 12. SUCTION TEMPERATURE 35°F

DISCHARGE TEMPERATURE 105°F

COMPARISON OF PERFORMANCE OF CARRIER MPS. SIZE 40

CENTRIFUGAL COMPRESSOR VS. 2516 DUNHAM-BUSH SCREW

COMPRESSOR.

TYPE OF COMPRESSOR	REFRIGERATION		POWER - INPUT			HEAT REJECTED	COP COOLING	COP HEATING
	TONS	BTUH x10 ⁶	BHP	HP/TON	BTUH x10 ⁶	BTUH x10 ⁶		
Centrifugal	1780	21.36	1960	1.1	4.988	26.348	4.28	5.28
Screw	455	5.46	470	1.03	1.196	6.656	4.57	5.57

At 35°F suction temperature and 105°F discharge temperature the screw compressor performs better than the centrifugal compressor.

Again, both show a much better performance at 35°F ST and 105°F DT than at 20°F ST and 120°F DT which was to be expected.

TABLE 1A.7

PERFORMANCE OF MPS SIZE 40 CARRIER CENTRIFUGAL COMPRESSOR
WITH R-12 AND R-22 AT 20°F AND 35°F SUCTION TEMPERATURES

Refrigerant	Capacity at 20°F Suction Temperature	Capacity at 35°F Suction Temperature
R-12	1480 tons	1780 tons
R-22	2450 tons	3130 tons

This comparison shows that R-22 has a larger refrigeration capacity than R-12 when both are used in the same compressor.

FIG. 1A.4 DUNHAM-BUSH SCREW COMPRESSOR PERFORMANCE CHART

R22

HIGH TEMPERATURE

E(HP) X 2516 COMPRESSOR UNIT
W(HP) X 2516 CONDENSING UNIT

2516

Capacities Based on: 10°F Liquid Subcooling, 10°F Suction Superheat.

Ref: Table 1A.3
 Table 1A.4

Source:
 Dunham-Bush
 Screw Com-
 pressor

FIG. 1A.5 DUNHAM-BUSH, SCREW COMPRESSOR PERFORMANCE CHART

R12

HIGH TEMPERATURE

**E(HP) X 2516 COMPRESSOR UNIT
W(HP) X 2516 CONDENSING UNIT**

2516

Capacities Based on: 10°F Liquid Subcooling, 10°F Suction Superheat.

Ref: Table 1A.3
Table 1A.4

800
700
600
500
400
300
200
BRAKE HORSEPOWER - BHP

595 HP

SATURATED DISCHARGE TEMPERATURE - SDT (°F) 160°F Vi 2.4

145
135
125
115
105
95
85
75

4.4
4.8
5.2
5.6
6.0
6.4
6.8
7.2
7.6
8.0
8.4
8.8
9.2
9.6
10.0
10.4
SATURATED DISCHARGE TEMPERATURE - SDT (°F)

4.4
4.8
5.2
5.6
6.0
6.4
6.8
7.2
7.6
8.0
8.4
8.8
9.2
9.6
10.0
10.4
TOTAL HEAT OF REJECTION - THR (BTU/HR X 10⁶)

R12
55°F
10
20°F
2516

CAPACITY IN TONS OF REFRIGERATION - TR

450 TR

275 TR

SATURATED DISCHARGE TEMPERATURE - SDT (°F) 75°F
85
95
105
115
125
135
145
160

SATURATED SUCTION TEMPERATURE - SST (°F)

20 25 30 35 40 45 50 55

Source:
Dunham-Bush
Screw Com-
pressors

FIG. 1A.6 DUNHAM-BUSH SCREW COMPRESSOR PERFORMANCE CHART

R12

HIGH TEMPERATURE

E(HP) X 2516 COMPRESSOR UNIT
W(HP) X 2516 CONDENSING UNIT

2516

Capacities Based on: 10°F Liquid Subcooling, 10°F Suction Superheat.

Ref: Table 1A.3
 Table 1A.4

FIG. 1A.7

Quick Selection Chart

: For Carrier Centrifugal M.P.S. Compressor

Ref: Tables 1A.5
1A.6

This chart is for estimating purposes only. Actual compressor selection involves consideration of numerous factors beyond the scope of this publication.

For precise selection information, based on your individual job requirements, simply contact your Carrier Refrigeration Sales Engineer.

Source: Carrier Centrifugal
Compressor 17 mps

NOTES

1. Compressor capacity is based on 105 F condensing temperature, Refrigerant 12 or 22 as noted, and the use of an economizer in the refrigeration circuit.
2. Bhp: ton values are based on assumption of a 70% efficiency ratio, no friction losses, and the use of Refrigerant 12 and an economizer.

FIG. 1A.8

Quick Selection Chart

: For Carrier Centrifugal MPS Compressor

Ref. Table 1A.7

This chart is for estimating purposes only. Actual compressor selection involves consideration of numerous factors beyond the scope of this publication.

For precise selection information, based on your individual job requirements, simply contact your Carrier Refrigeration Sales Engineer.

Source: Carrier Centrifugal Compressor, 17 mps

NOTES

1. Compressor capacity is based on 105 F condensing temperature, Refrigerant 12 or 22 as noted, and the use of an economizer in the refrigeration circuit.
2. Bhp/ton values are based on assumption of a 70% efficiency ratio, no friction losses, and the use of Refrigerant 12 and an economizer.

APPENDIX 2A

2A.1 HEATING AND COOLING ANNUAL ENERGY BALANCE

Table 2A-1 summarizes the heat balance calculations. Column 3 is the sum of columns 1 and 2 and gives the total heating energy requirements for heating and domestic hot water. In the following columns are given the cooling capacities obtainable from a heat pump operating with a COP = 3.2. Thus for Washington,

$$\frac{33,088 \times 10^6 \times 2.2}{3.2} = 22,748 \times 10^6 \text{ BTU}$$

Thus the heat pump is capable of generating during an entire winter, $22,748 \times 10^6$ BTU cooling energy, which however, is not sufficient to cover the summer cooling requirements of $36,537 \times 10^6$ BTU (column 5). The deficiency in cooling is

$$(36,537 - 22,748) \times 10^6 = 13,789 \times 10^6 \text{ BTU (col 6)}$$

On the other hand in colder, northern locations, say in Minneapolis, MN the reverse is taking place, namely the "cooling" energy or ice generated during the winter is

$$63,512 \times 10^6 \times 2.2 = 43,665 \times 10^6$$

which exceeds the cooling requirements for the summer of $21,895 \times 10^6$ by $21,806 \times 10^6$ (column 7).

Table 2A-2 shows the domestic hot water energy requirements and the "cooling" energy obtainable from the heat pump by using a COP = 3.2 in winter and COP = 5 for summer.

The "cooling energy generated from the winter part of domestic hot water has been included in winter heat pump operation and has been accounted for.

However, the summer generation of domestic hot water can be done with the heat pump generating at the same time chilled water, possibly at night time and thereby reducing the deficiency of cooling.

Thus for Washington the deficiency in cooling was seen to be $13,789 \times 10^6$ BTU.

Generating domestic hot water in the summer can produce additional "cooling" energy

$$\frac{3149 \times 10^6 \times 2.79}{3.79} = 2318 \times 10^6 \text{ BTU}$$

Thus the deficiency is reduced to:

$$\begin{aligned} (13,789 - 2318) \times 10^6 &= 11471 \times 10^6 \text{ BTU} \\ &= 955,917 \text{ ton-hours} \end{aligned}$$

In the second table the domestic hot water in the summer is indicated only for locations having a deficiency in cooling as for the other locations which have surplus ice, hot water in the summer would be generated by other means.

Section 2A.2 gives all of the data used to develop the calculations and Tables 2A-3 through 2A-7 contain the calculations used to generate the finding.

TABLE 2A-1

HEATING AND COOLING ANNUAL ENERGY BALANCE

Location	Heating Energy Requirements BTU x 10 ⁶	Winter DHW Energy Requirements BTU x 10 ⁶	Total Energy Requirements BTU x 10 ⁶	Heat Pump Cooling Capacity With COP = 3.2 BTU x 10 ⁶	Energy Requirement for Cooling BTU x 10 ⁶	Defficiency in Cooling Energy BTU x 10 ⁶	Surplus in Cooling Energy Supplementary Heat for Ice Melting BTU x 10 ⁶
St. Petersburg, FL	4412	3534	7946	5463	86,738	81275	
New Orleans, LA	8909	3283	12192	8382	86,940	78558	
El Paso, TX	17302	4758	22060	15166	49,853	34687	
Oklahoma City, OK	23974	5115	29089	19999	59,873	39874	
Corpus Christi, TX	5882	4115	9997	6873	93,204	86331	
Denver, CO	41076	8285	49361	33936	16,164		17772
Chicago, IL	43358	8002	51360	35310	28,080		7299
Washington, DC	27026	6062	33038	22748	36,537	13789	
New York, NY	31350	6888	38238	26289	27,900	1611	
Boston, MA	36536	7541	44077	30303	28,195		2108
Minneapolis, MN	55234	8278	63512	43665	21,859		21806
Seattle, WA	28417	8294	36711	25239	21,437		3802
Rapid City, SD	48279	8252	56895	39115	29,430		9685
Des Moines, IA	43265	7044	50309	34587	30,778		3809

HP-ICES PROJECT

TABLE 2A-2

WINTER AND SUMMER DOMESTIC HOT WATER ENERGY REQUIREMENTS AND COOLING CAPACITY OF HEAT PUMP

Location	WINTER	Heat Pump	SUMMER	Heat Pump
	Winter Requirements BTU x 10 ⁶	Cooling Capacity with COP = 3.2 BTU x 10 ⁶	Summer Requirements BTU x 10 ⁶	Cooling Capacity with COP = 3.79 BTU x 10 ⁶
St. Petersburg, FL	3534	2430	4154	3058
New Orleans, LA	3283	2257	3682	2710
El Paso, TX	4758	3271	3182	2342
Oklahoma City, OK	5115	3517	3835	2823
Corpus Christi, TX	4115	2829	3823	2814
Denver, CO	8285	5696		
Chicago, IL	8002	5501		
Washington, DC	6062	4168	3149	2318
New York, NY	6888	4736	4095	3015
Boston, MA	7541	5184		
Minneapolis, MN	8278	5691		
Seattle, WA	8294	5702		
Rapid City, SD	8252	5673		
Des Moines, IA	7044	4843		

Note: Summer hot water generation with heat pump indicated only for locations where there is a deficiency in cooling.

2A.2 MARKET SQUARE COMPLEX IN WASHINGTON, D. C. (Case Study)

This project, being developed by the Pennsylvania Avenue Redevelopment Corporation, is situated on Pennsylvania Avenue and consists of approximately 1,300,000 square feet above grade and 900,000 below.

It was selected because of the functional diversity in the program calling for 56% residential area, 13% retail department stores and offices and about 30% of national archives and community storage above ground. The remaining 900,000 square feet below grade are devoted entirely to national archives.

Situated between 7th and 9th Street Market Square Complex forms the focal point of the entire proposed Pennsylvania Avenue Development project. It is conceived as an integrated major city project containing multi-use facilities incorporating energy conservation techniques and load management concepts.

Listed below is the data used to develop the thermal loads. The source for the information is Pennsylvania Avenue Energy Conservation and Alternate Energy Source Conceptual Plan by Dubin-Bloome Associates, July, 1977.

1. Gross Floor Areas

1.	776 Dwelling Units	734,970 sq. ft.
2.	Retail & Offices	170,973 sq. ft.
3.	Community Storage	39,772 sq. ft.
4.	Archive Above Grade	336,920 sq. ft.
5.	Archive Below Grade	<u>920,565 sq. ft.</u>
		2,203,200 sq. ft.

FIGURE 2A-2 MARKET SQUARE, WASHINGTON, DC

HOUSING SECTIONS

CROSS SECTION (A-A) LOOKING SOUTH TOWARD THE ARCHIVES

CROSS SECTION (B-B) ALONG 8TH STREET LOOKING EASTWARD

FIGURE 2A-1 MARKET SQUARE, WASHINGTON, DC

HOUSING PLAN

2. Envelope

Residential	404,000 sq. ft.
Commercial	<u>50,000 sq. ft.</u>
Total	454,000 sq. ft.

3. Roof

Residential	140,000 sq. ft.
Commercial	<u>124,000 sq. ft.</u>
Total	204,000 sq. ft.

4. Archives Below Grade

Walls	101,400 sq. ft.
Roof	321,600 sq. ft.
Floor	<u>321,600 sq. ft.</u>
Total	744,600 sq. ft.

5. "U" Factors

Wall	0.06 BTUH/sq. ft. - °F
Double Glass	0.6 BTUH/sq. ft. - °F
Roof	0.06 BTUH/sq. ft. - °F
Slab Below Grade	0.063 BTUH/sq. ft. - °F

Glass 0.2 of envelope area

Infiltration 1.2 LF/sq.ft. of glass

0.5 cfm/LF

6. Ventilation Load

Above Grade Areas:

Use 0.11 cfm/sq.ft. for summer only since in winter time infiltration has been accounted for.

Below Grade Areas:

Use 0.11 cfm/sq.ft. for summer and winter.

7. People

Residential 776 dwelling units, 3 person per unit
=2328 persons

Retail and Offices, 171,000 sq.ft., 100 sq. ft
per person =1710 persons

Community Storage, 40,000 sq.ft., 400 sq.ft.
per person = 100 persons

Archives, 1257,500 sq.ft., 1500 sq.ft. per
person = 838 persons

Total = 4976 persons

Use 450 BTUH per person

8. Light and Appliances

Residential 1.5 KW per D.U., 776 x 1.5 =	1,164 KW
Retail and offices 171,000 sq.ft. x 4W/sq.ft.=	684 KW
Community storage 40000 sq.ft. x 2W/sq.ft. =	80 KW
Archives 1,257,500 x 2W/sq.ft.	= <u>2,515 KW</u>
Total	= <u>4,443 KW</u>

TABLE 2A-3

HEAT TRANSFER COEFFICIENTS FOR SELECTED LOCATIONS IN THE UNITED STATES

Location	Heating Degree Days	Wall Overall Heat Transfer Coefficient		Roof Overall Heat transfer Coefficient		Below Grade Heat transfer Coefficient	Overall Thermal Transfer Value	
		Heating BTU/Hr-Ft ² -°F		BTU/Hr-Ft ² -°F		BTU/Hr-Ft ² -°F	Cooling (BTU/Hr-Ft ²)	
		ASHRAE	Bldg	ASHRAE	Bldg	Bldg	ASHRAE	Bldg
St. Petersburg, FL	683	0.465	0.305	0.100	0.06	0.063	30.100	19.394
New Orleans, LA	1385	0.450	0.305	0.100	0.06	0.063	30.700	19.576
El Paso, TX	2700	0.415	0.305	0.100	0.06	0.063	31.300	20.570
Oklahoma City, OK	3725	0.390	0.305	0.094	0.06	0.063	32.100	20.716
Corpus Christi, TX	914	0.460	0.305	0.100	0.06	0.063	30.100	19.634
Denver, CO	6283	0.325	0.305	0.073	0.06	0.063	33.500	20.310
Chicago, IL	6639	0.314	0.305	0.070	0.06	0.063	34.100	20.886
Washington, DC	4224	0.370	0.305	0.090	0.06	0.063	33.200	20.494
New York, NY	4871	0.360	0.305	0.085	0.06	0.063	33.50	20.370
Boston, MA	5634	0.340	0.305	0.079	0.06	0.063	34.100	20.406
Minneapolis, MN	8382	0.280	0.305	0.060	0.06	0.063	34.900	20.988
Seattle, WA	4424	0.370	0.305	0.088	0.06	0.063	35.700	20.278
Rapid City, SD	7345	0.294	0.305	0.065	0.06	0.063	34.650	21.420
Des Moines	6588	0.316	0.305	0.071	0.06	0.063	33.900	20.802

TABLE 2A-4

PEAK HEATING LOAD

<u>Location</u>	<u>Envelope and Roof Transmission BTUH</u>	<u>Below Grade Transmission BTUH</u>	<u>Ventilation Load BTUH</u>	<u>Total BTUH</u>
St. Petersburg, FL	4,012,000	141,000	2,845,000	6,998,000
New Orleans, LA	5,093,000	141,000	3,611,000	8,845,000
El Paso, TX	6,635,000	141,000	4,705,000	11,481,000
Oklahoma City, OK	8,179,000	235,000	5,799,000	14,213,000
Corpus Christi, TX	4,938,000	141,000	3,501,000	8,580,000
Denver, CO	10,031,000	563,000	7,112,000	17,706,000
Chicago, IL	10,338,000	563,000	7,331,000	18,232,000
Washington, DC	7,561,000	141,000	5,361,000	13,063,000
New York, NY	8,179,000	235,000	5,799,000	14,213,000
Boston, MA	8,950,000	376,000	6,346,000	15,672,000
Minneapolis, MN	12,037,000	845,000	8,534,000	21,416,000
Seattle, WA	5,555,000	141,000	3,939,000	9,635,000
Rapid City, SD	11,419,000	751,000	8,097,000	20,267,000
Des Moines, IA	10,956,000	704,000	7,768,000	19,428,000

HP-ICES PROJECT

TABLE 2A-5

PEAK COOLING LOAD

Location	Solar and Transmission BTUH	People BTUH	Lights and Appliances BTUH	Ventilation BTUH	Total	
					BTUH x 10 ³	Tons
St. Petersburg, FL	9,233,000	2,239,000	15,164,000	14,667,000	41,303	3442
New Orleans, LA	9,331,000	2,239,000	15,164,000	14,667,000	41,401	3450
El Paso, TX	9,798,000	2,239,000	15,164,000	14,344,000	41,545	3462
Oklahoma City, OK	9,896,000	2,239,000	15,164,000	11,324,000	38,623	3219
Corpus Christi, TX	9,357,000	2,239,000	15,164,000	14,667,000	41,427	3452
Denver, CO	9,538,000	2,239,000	15,164,000	0	26,941	2245
Chicago, IL	9,894,000	2,239,000	15,164,000	10,138,000	37,435	3120
Washington, DC	9,732,000	2,239,000	15,164,000	11,324,000	38,459	3205
New York, NY	9,660,000	2,239,000	15,164,000	10,138,000	37,201	3100
Boston, MA	9,645,000	2,239,000	15,164,000	8,196,000	35,244	2937
Minneapolis, MN	9,861,000	2,239,000	15,164,000	9,167,000	36,431	3036
Seattle, WA	9,396,000	2,239,000	15,164,000	0	26,799	2233
Rapid City, SD	10,010,000	2,239,000	15,164,000	5,285,000	32,698	2725
Des Moines, IA	9,745,000	2,239,000	15,164,000	11,324,000	38,472	3206

TABLE 2A-6

DOMESTIC HOT WATER ENERGY REQUIREMENTS
(Million BTU)

Location	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sept	Oct	Nov	Dec	Total
St. Petersburg, FL	777	648	647	627	647	628	520	520	565	647	691	771	7688
New Orleans, LA	712	648	647	627	556	440	455	455	502	647	628	648	6965
El Paso, TX	829	837	816	565	583	490	518	479	490	622	804	907	7940
Oklahoma City, OK	907	825	907	816	751	628	583	583	628	712	753	907	9000
Corpus Christi, TX	829	837	816	628	583	490	531	479	490	622	804	829	7938
Denver, CO	1049	943	997	892	842	754	738	725	716	829	942	1075	10502
Chicago, IL	1139	1037	1114	980	873	791	712	686	729	816	942	1101	10920
Washington, DC	1010	919	881	804	738	666	686	544	515	673	817	958	9211
New York, NY	1088	1002	1088	1018	945	829	803	777	741	816	905	971	10983
Boston, MA	1140	990	1049	854	803	616	596	686	754	829	905	971	10193
Minneapolis, MN	1140	1002	1036	1005	842	753	712	647	691	777	879	906	10390
Seattle, WA	1049	978	997	942	932	791	777	673	678	816	905	997	10535
Rapid City, SD	1140	1002	1136	879	842	691	583	583	691	777	879	906	10109
Des Moines, IA	1101	943	971	817	777	628	583	583	628	712	817	906	9466

HP-ICES PROJECT

*DHW demand based on 60 gallons per apartment per day, 12 hours usage;
2 gallons per person per day for other areas, 8 hours usage (see Page
115).

TABLE 2A-7

DETERMINATION OF SUPPLEMENTARY COMPRESSOR WORK, SUPPLEMENTARY HEATING AND ICE STORAGE BIN SIZE

Location	Defficiency In Cooling BTU x 10 ⁶	Supplementary Compressor Work KWH (=Ton-Hrs)	Cooling Required ⁶ BTU x 10 ⁶	Surplus Cool- ing Energy or Supplementary Heat Required BTU x 10 ⁶	Volume of Ice Bin Cu. Ft.
St. Petersburg, FL	81,275	6,773,000			646,000
New Orleans, LA	78,558	6,546,000			992,000
El Paso, TX	34,687	2,891,000			1,794,000
Oklahoma City, OK	39,874	3,323,000			2,367,000
Corpus Christi, TX	86,331	7,194,000			814,000
Denver, CO			16,164	17,772	1,913,000
Chicago, IL			28,080	7,299	3,323,000
Washington, DC	13,789	1,149,000			2,692,000
New York, NY	1,611	134,000			3,110,000
Boston, MA			28,195	2,108	3,337,000
Minneapolis, MN			21,859	21,806	2,587,000
Seattle, WA			21,437	3,802	2,537,000
Rapid City, SD			29,430	9,685	3,483,000
Des Moines, IA			30,778	3,809	3,643,000

Note: Cooling requirements given here only for locations where the cooling requirements govern the determination of ice storage bin size.

2A.3 PROJECTION OF ENERGY SAVINGS FOR THE YEAR 2000
BY THE USE OF HP-ICES

1. Fuel Usage by Sectors

(From National Gas Survey F.P.C., Chapter 6)

1.1 Residential Sector, 1971

Space Heating

Gas	$3,665 \times 10^{12}$	Btu
Oil	$2,968 \times 10^{12}$	Btu
Electric	215×10^{12}	Btu
Total =	$6,848 \times 10^{12}$	Btu

Domestic Hot Water

Gas	$1,112 \times 10^{12}$	Btu
Oil	143×10^{12}	Btu
Electric	293×10^{12}	Btu
Total =	$1,548 \times 10^{12}$	Btu

Space Heating and DHW =	$6,848 \times 10^{12}$	Btu
	$1,548 \times 10^{12}$	Btu
	$8,396 \times 10^{12}$	Btu

1.2 Commercial Sector, 1971

Space Heating

Coal	390×10^{12}	Btu
Gas	$1,397 \times 10^{12}$	Btu
Oil	$2,393 \times 10^{12}$	Btu
Total =	$4,180 \times 10^{12}$	Btu

Domestic Hot Water

Gas	458×10^{12}	Btu
Electric	87×10^{12}	Btu
Total =	545×10^{12}	Btu

Space Heating and DHW =	$4,180 \times 10^{12}$
	545×10^{12}
Total =	$4,725 \times 10^{12}$

1.3 Projection for the Year 2000 (From Electrical World Magazine - September 1978, issue)

Residential

<u>Year</u>	<u>Households Millions</u>	<u>Ratio of Annual Increase</u>
1971	65.2	
1976	72.9	1.118
1981	80.2	1.100
1986	87.2	1.087
1990	92.2	1.057
1995	97.7	1.060
2000	103.5	1.059

Heating and Domestic Hot Water

<u>Year</u>			<u>Ratio of Increase</u>
1971	$8,396 \times 10^{12}$	Btu	
1976	$9,387 \times 10^{12}$	Btu	1.118
1981	$10,325 \times 10^{12}$	Btu	1.100
1986	$11,224 \times 10^{12}$	Btu	1.087
1990	$11,863 \times 10^{12}$	Btu	1.057
1995	$12,575 \times 10^{12}$	Btu	1.060
2000	$13,317 \times 10^{12}$	Btu	1.059

Air Conditioning

Gas	5×10^{12}	Btu
Electrical	201×10^{12}	Btu
Total =	206×10^{12}	Btu

<u>Year</u>			<u>Ratio of Increase</u>
1971	206 x 10 ¹²	Btu	
1976	230 x 10 ¹²	Btu	1.118
1981	253 x 10 ¹²	Btu	1.100
1986	275 x 10 ¹²	Btu	1.087
1990	291 x 10 ¹²	Btu	1.057
1995	309 x 10 ¹²	Btu	1.060
2000	327 x 10 ¹²	Btu	1.059

2. Analysis of Residential Sector

2.1 Absolute Savings in Energy

By the year 2000 the required energy for air conditioning is estimated at 327×10^{12} Btu = $27,250 \times 10^6$ ton hours.

With 1 KW/Ton, the savings in energy input = $27,250 \times 10^6$ kWh

With an assumed heat rate of 9000 Btu/kWh and a fuel value of 140,000 Btu/gal. the savings in barrels of oil are estimated at

$$\frac{27,250 \times 10^6 \times 9000}{140,000 \times 42} = 42 \times 10^6 \text{ barrels}$$

2.2 Heat Pump System Plus Direct Heating

Heating obtained from heat pump on satisfying cooling requirements (COP 3.2)

$$= \frac{327 \times 10^{12} \times 3.2}{2.2} = 476 \times 10^{12} \text{ Btu}$$

Deficiency in available heating energy is

$$13,317 \times 10^{12} - 476 \times 10^{12} = 12,841 \times 10^{12} \text{ Btu}$$

Conventional (direct) heating with boiler thus requires

$$\frac{12,841 \times 10^{12}}{0.7 \times 140,000 \times 42} = 3120 \times 10^6 \text{ barrels}$$

assuming an utilization efficiency of 0.7.

Compressor input to heat pump

$$= \frac{327 \times 10^{12}}{2.2 \times 3413} \times 9000 = 392 \times 10^{12} \text{ Btu}$$

$$= \frac{392 \times 10^{12}}{140,000 \times 42} = 67 \times 10^6 \text{ barrels}$$

$$\begin{aligned} \text{Energy input for heating and cooling} &= (3120 + 67) \times 10^6 \\ &= 3,187 \times 10^6 \text{ barrels of oil} \end{aligned}$$

2.3 Using Supplementary Heat for Heat Pump Source, or Melting Excess Ice

Additional heating $12,841 \times 10^{12}$ Btu

Heat input required at heat source

$$= \frac{12,841 \times 10^{12} \times 2.2}{3.2} = 8,828 \times 10^{12} \text{ Btu}$$

$$= \frac{8,828 \times 10^{12}}{0.7 \times 140,000 \times 42} = 2,145 \times 10^6 \text{ barrels}$$

Compressor input to heat pump

$$= \frac{12,841 \times 10^{12} \times 9000}{3.2 \times 3413} = 10,582 \times 10^{12} \text{ Btu}$$

$$= \frac{10,582 \times 10^{12}}{140,000 \times 42} = 1,800 \times 10^6 \text{ barrels}$$

$$\text{Subtotal } (2,145 + 1,800) \times 10^6 = 3,945 \times 10^6 \text{ barrels}$$

Compressor input to ice generation (from above) = 67 barrels

$$\text{Total } (3945 + 67) \times 10^6 = 4,012 \times 10^6 \text{ barrels of oil}$$

Alternately

Generating ice all the way while satisfying the whole heating requirements and subsequently melting the excess of ice with supplementary heat.

$$\text{Heating and DHW} = 13,317 \times 10^{12} \text{ Btu}$$

$$\text{Cooling obtainable} = \frac{13,317 \times 2.2 \times 10^{12}}{3.2}$$

$$= 9,155.10^{12} \text{ Btu}$$

Compressor input

$$\frac{9,155 \times 10^{12}}{2.2 \times 3413} = 10,975 \times 10^{12} \text{ Btu}$$

$$= \frac{10,975 \times 10^{12}}{140,000 \times 42} = 1,866 \times 10^6 \text{ barrels}$$

$$\text{Excess Cooling } (9,155 - 327) \times 10^{12} \text{ Btu}$$

$$= 8,828 \times 10^{12} \text{ Btu}$$

Supplementary heat required to melt excess ice

$$= \frac{8,828 \times 10^{12}}{0.7 \times 140,000 \times 42} = 2145 \times 10^6 \text{ barrels}$$

$$\text{Compressor input} = 1,866 \times 10^6 \text{ barrels}$$

$$\text{Total} = 4,011 \times 10^6 \text{ barrels (as before)}$$

2.4 Conventional System

Cooling required 327×10^{12} Btu using 0.8 KW/Ton barrels of

$$\text{oil} = \frac{327 \times 10^{12} \times 0.8 \times 9000}{12,000 \times 140,000 \times 42} = 33 \times 10^6$$

Heating $13,317 \times 10^{12}$ Btu

$$= \frac{13,317 \times 10^{12}}{0.7 \times 140,000 \times 42} = 3,235 \times 10^6 \text{ barrels}$$

$$\text{Total} = (3,235 + 33) \times 10^6$$

$$= 3,268 \times 10^6 \text{ barrels of oil}$$

2.5 Recapitulation

Heat Savings With Direct Heating = $3,187 \times 10^6$ barrels

Heat Savings With Ice Melting = $4,012 \times 10^6$ barrels

Conventional = $3,268 \times 10^6$ barrels

2.6 Conclusions

Heat pump with direct heating is the best solution. Savings with relation to conventional system amount to

$$3,268 \times 10^6 - 3,187 \times 10^6 = 81 \times 10^6 \text{ barrels.}$$

In addition the best pump saves condenser water pumping energy and cooling tower fan energy assumed to amount to 0.2

KW/Ton. Thus additional savings in refrigeration auxiliaries

$$\frac{327 \times 10^{12} \times 0.2 \times 9000}{12,000 \times 140,000 \times 42} = 8 \times 10^6 \text{ barrels}$$

Total Savings $(81 + 8) \times 10^6 = 89 \times 10^6$ barrels of oil

3. Analysis of Commercial Sector

3.1 Project of Growth of Heating and Air Conditioning Energy Requirements.

Heat and DHW

<u>Year</u>			<u>Ratio of Increase</u>
1971	4725×10^{12}	Btu	
1976	5292×10^{12}	Btu	1.120
1981	5821×10^{12}	Btu	1.100
1986	6403×10^{12}	Btu	1.100
1990	6788×10^{12}	Btu	1.060
1995	7195×10^{12}	Btu	1.060
2000	7626×10^{12}	Btu	1.060

Air Conditioning

Gas	113×10^{12}	Btu
Electric	385×10^{12}	Btu
Total	$= 498 \times 10^{12}$	Btu

<u>Year</u>		<u>Ratio of Increase</u>
1971	498×10^{12} Btu	
1976	578×10^{12} Btu	1.120
1981	614×10^{12} Btu	1.100
1986	675×10^{12} Btu	1.100
1990	715×10^{12} Btu	1.060
1995	758×10^{12} Btu	1.060
2000	804×10^{12} Btu	1.060

3.2 Absolute Savings in Energy

$$\begin{aligned} \text{Cooling required by the year 2000} &= 804 \times 10^{12} \text{ Btu} \\ &= 67,000 \times 10^6 \text{ ton hours} \end{aligned}$$

$$\begin{aligned} \text{With 1 KW/Ton Savings} &= 67,000 \times 10^6 \text{ kWh} \\ &= \frac{67,000 \times 10^6 \times 9000}{140,000 \times 42} = 103 \times 10^6 \text{ barrels} \end{aligned}$$

3.3 Heat Pump System Plus Direct Heating

$$\text{Cooling Required} = 804 \times 10^{12} \text{ Btu}$$

$$\text{Heating Obtain by Heat Pump} = \frac{804 \times 10^{12} \times 3.2}{2.2} = 1169 \times 10^{12} \text{ Btu}$$

$$\begin{aligned} \text{Additional Heat Energy Required} &= (7626 - 1169) \times 10^{12} \\ &= 6457 \times 10^{12} \text{ Btu} \\ &= \frac{6457 \times 10^{12}}{0.7 \times 140,000 \times 42} = 1569 \times 10^6 \text{ barrels of oil} \end{aligned}$$

$$\begin{aligned}\text{Compressor input} &= \frac{804 \times 10^{12} \times 9000}{2.2 \times 3413} = 964 \times 10^{12} \text{ Btu} \\ &= \frac{964 \times 10^{12}}{140,000 \times 42} = 164 \times 10^6 \text{ barrels}\end{aligned}$$

$$\text{Total} = (1509 + 164) \times 10^6 = 1733 \times 10^6 \text{ barrels}$$

3.4 Using Supplementary Heat for Heat Pump Source, or Melting Excess Ice.

Additional heating required 6457×10^{12} Btu (twice above)

Heat required at heat source

$$\frac{6457 \times 10^{12} \times 2.2}{3.2 \times 0.7 \times 140,000 \times 42} = 1079 \times 10^6 \text{ barrels}$$

Compressor input

$$\frac{6457 \times 10^{12} \times 9000}{3.2 \times 3413 \times 140,000 \times 42} = 905 \times 10^6 \text{ barrels}$$

$$\text{Total} = (1079 + 905 + 164) \times 10^6 \text{ barrels} = 2,148 \times 10^6 \text{ barrels}$$

Alternately

$$\text{Heating \& DHW} = 7626 \times 10^{12} \text{ Btu}$$

Compressor input

$$\frac{7626 \times 10^{12} \times 9000}{3.2 \times 3413 \times 140,000 \times 42} = 1069 \times 10^6 \text{ barrels}$$

Cooling Obtainable

$$\frac{7626 \times 10^{12} \times 2.2}{3.2} = 5243 \times 10^{12} \text{ Btu}$$

$$\text{Cooling Required} = \frac{804 \times 10^{12}}{1} \text{ Btu}$$

$$\text{Excess} = 4439 \times 10^{12} \text{ Btu}$$

Supplementary Heat

$$= \frac{4439 \times 10^{12}}{0.7 \times 140,000 \times 42} = 1078 \times 10^6 \text{ barrels}$$

$$\text{Total} = (1069 + 1078) \times 10^6 = 2147 \times 10^6 \text{ barrels (as before)}$$

3.5 Conventional System

$$\begin{aligned} \text{Cooling Required} &= 804 \times 10^{12} \text{ Btu} \\ &= \frac{804 \times 10^{12}}{12,000} = 67,000 \text{ ton-hours} \end{aligned}$$

With 0.8 KW/Ton, Compressor input

$$\begin{aligned} &= 67,000 \times 0.8 = 53,600 \times 10^6 \text{ KW/T} \\ \text{Barrels of Oil} &= \frac{53,600 \times 10^6 \times 9000}{140,000 \times 42} = 82 \times 10^6 \text{ barrels} \\ \text{Heating} &= 7626 \times 10^{12} \text{ Btu} \\ \text{Barrels of Oil} &= \frac{7626 \times 10^{12}}{0.7 \times 140,000 \times 42} = \underline{1853} \text{ barrels} \\ \text{Total} &= 1935 \times 10^6 \text{ barrels} \end{aligned}$$

3.6 Recapitulation

$$\begin{aligned} \text{Heat Pump with Direct Heating} &= 1733 \times 10^6 \text{ barrels} \\ \text{Heat Pump with Ice Melting} &= 2148 \times 10^6 \text{ barrels} \\ \text{Conventional} &= 1935 \times 10^6 \text{ barrels} \end{aligned}$$

3.7 Conclusions

$$\begin{aligned} \text{Savings} &(1935 - 1733) \times 10^6 \text{ barrels} \\ &= 202 \times 10^6 \text{ barrels} \end{aligned}$$

Refrigeration auxiliaries 0.2 KW/Ton

$$= 67,000 \times 10^6 \times 0.2 = 13,400 \times 10^6 \text{ kWh}$$

$$= \frac{13,400 \times 9000 \times 10^6}{140,000 \times 42} = 21 \times 10^6 \text{ barrels}$$

$$\text{Total Savings} = (202 + 21) \times 10^6 = 223 \times 10^6 \text{ barrels}$$

4. Summary

Savings in barrels of oil by utilizing the ice generating HP-ICES for the whole of the United States, in the year 2000

Residential Sector 89×10^6 barrels

Commercial Sector 223×10^6 barrels

Total = 312×10^6 barrels

5. Calculations for Correction Factors Based on Climate and Metropolitan Areas

$$T_{us} \times \%_R \times \%_{mp} \times cc = T_R$$

Where T_{us} - Total U.S. Energy projected savings in year 2000.

$\%_R$ - Regional use percent of total U.S. energy use

Source: Fuels and Energy Data: United States by States and

Census Divisions 1973, by Lulie Crump, U.S. Department

$\% MP$ - Percent of regional population in Metropolitan Areas.

Source: Statistical Abstracts of the United States 1977,

U.S. Department of Commerce Bureau of the Census p. 17

CC - Climatic Condition Correction

Source: Extrapolated from Fig. 2-13

T_R - Total regional savings in year 2000

NEW ENGLAND -

$$312 \times 10^6 \text{ bbls} \times .05 \times .764 \times .837 = 9.98 \times 10^6 \text{ bbls}$$

MIDDLE ATLANTIC

$$312 \times 10^6 \text{ bbls} \times .15 \times .873 \times .908 = 37.10 \times 10^6$$

EAST NORTH CENTRAL

$$312 \times 10^6 \text{ bbls} \times .21 \times .773 \times .846 = 42.85 \times 10^6 \text{ bbls}$$

WEST NORTH CENTRAL

$$312 \times 10^6 \text{ bbls} \times .18 \times .507 \times .852 = 10.78 \times 10^6 \text{ bbls}$$

SOUTH ATLANTIC

$$312 \times 10^6 \text{ bbls} \times .12 \times .654 \times .701 = 17.16 \times 10^6 \text{ bbls}$$

EAST SOUTH CENTRAL

$$312 \times 10^6 \text{ bbls} \times .06 \times .521 \times .664 = 6.48 \times 10^6 \text{ bbls}$$

WEST SOUTH CENTRAL

$$312 \times 10^6 \text{ bbls} \times .17 \times .677 \times .330 = 11.85 \times 10^6 \text{ bbls}$$

MOUNTAIN

$$312 \times 10^6 \text{ bbls} \times .05 \times .597 \times .804 = 7.49 \times 10^6 \text{ bbls}$$

PACIFIC

$$312 \times 10^6 \text{ bbls} \times .11 \times .877 \times .706 = \underline{21.25 \times 10^6 \text{ bbls}}$$

Total Barrels of Oil Saving $164.94 \times 10^6 \text{ bbls}$
in the U.S. in the Year 2000

APPENDIX 4A

4A. ECONOMIC ANALYSIS PROGRAM

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

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

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:

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

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 t .

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 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.
- 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.
FUEL ESCALATION
12.00 %
DISCOUNT RATE
10.00 %
CAPITAL COST
2000000.00
OPERATION COST
600000.00
SALVAGE VALUE
600000.00

fig4A-1 Present Value Conventional System:**Boilers and Chillers**

*RUN					
1.00	YEAR	11.00	YEAR	SALVAGE VALUE	(PV)
-545454.55	COST	-653148.51	COST	89186.18	
-2545454.55	PVTL	-8576316.55	PVTL	PRESENT VALUE	
				-14926613.89	
2.00	YEAR	12.00	YEAR		
-555371.90	COST	-665023.94	COST		
-3100826.45	PVTL	-9241340.49	PVTL		
3.00	YEAR	13.00	YEAR		
-565469.57	COST	-677115.28	COST		
-3666296.02	PVTL	-9918455.77	PVTL		
4.00	YEAR	14.00	YEAR		
-575750.84	COST	-689426.47	COST		
-4242046.85	PVTL	-10607882.24	PVTL		
5.00	YEAR	15.00	YEAR		
-586219.03	COST	-701961.50	COST		
-4828265.89	PVTL	-11309843.73	PVTL		
6.00	YEAR	16.00	YEAR		
-596877.56	COST	-714724.43	COST		
-5425143.45	PVTL	-12024568.17	PVTL		
7.00	YEAR	17.00	YEAR		
-607729.88	COST	-727719.42	COST		
-6032873.33	PVTL	-12752287.59	PVTL		
8.00	YEAR	18.00	YEAR		
-618779.52	COST	-740950.68	COST		
-6651652.85	PVTL	-13493238.27	PVTL		
9.00	YEAR	19.00	YEAR		
-630030.05	COST	-754422.51	COST		
-7281682.90	PVTL	-14247660.78	PVTL		
10.00	YEAR	20.00	YEAR		
-641485.14	COST	-768139.29	COST		
-7923168.04	PVTL	-15015800.07	PVTL		

PROJECT LIFE
20.00
FUEL ESCALATION
12.00 %
DISCOUNT RATE
10.00 %
OPERATION COST
436000.00
CAPITAL COST
3812000.00
SALVAGE VALUE
1305000.00

fig 4A-2 Present Value Heat Pump System:**Ice Storage HP-ICES**

*RUN

1.00	YEAR	11.00	YEAR	SALVAGE VALUE (PV)
-396363.64	COST	-474621.25	COST	193979.93
-4208363.64	PVTL	-8590790.03	PVTL	PRESENT VALUE
				-13076168.12
2.00	YEAR	12.00	YEAR	
-403570.25	COST	-483250.73	COST	
-4611933.88	PVTL	-9074040.76	PVTL	
3.00	YEAR	13.00	YEAR	
-410907.89	COST	-492037.10	COST	
-5022841.77	PVTL	-9566077.86	PVTL	
4.00	YEAR	14.00	YEAR	
-418378.94	COST	-500983.23	COST	
-5441220.71	PVTL	-10067061.09	PVTL	
5.00	YEAR	15.00	YEAR	
-425985.83	COST	-510092.02	COST	
-5867206.55	PVTL	-10577153.11	PVTL	
6.00	YEAR	16.00	YEAR	
-433731.03	COST	-519366.42	COST	
-6300937.57	PVTL	-11096519.53	PVTL	
7.00	YEAR	17.00	YEAR	
-441617.05	COST	-528809.45	COST	
-6742554.62	PVTL	-11625328.98	PVTL	
8.00	YEAR	18.00	YEAR	
-449646.45	COST	-538424.16	COST	
-7192201.07	PVTL	-12163753.14	PVTL	
9.00	YEAR	19.00	YEAR	
-457821.84	COST	-548213.69	COST	
-7650022.91	PVTL	-12711966.84	PVTL	
10.00	YEAR	20.00	YEAR	
-466145.87	COST	-558181.22	COST	
-8116168.78	PVTL	-13270148.05	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.

fig 4A-3 Relative Present Value

Conventional vs. Heat Pump System:

Ice Storage HP-ICES

PROJECT LIFE
20.00
FUEL ESCALATION
12.00 %
DISCOUNT RATE
10.00 %
OPERATION SAVINGS
164000.00
CAPITAL COST
1812000.00
SALVAGE VALUE
705000.00

*RUN

1.00	YEAR	11.00	YEAR	SALVAGE VALUE (PW)
149090.91	COST	178527.26	COST	104793.76
-1662909.09	PVTL	-14473.48	PVTL	PRESENT VALUE
				1850445.78
2.00	YEAR	12.00	YEAR	
151801.65	COST	181773.21	COST	
-1511407.44	PVTL	167299.73	PVTL	
3.00	YEAR	13.00	YEAR	
154561.68	COST	185078.18	COST	
-1356545.76	PVTL	352377.91	PVTL	
4.00	YEAR	14.00	YEAR	
157371.90	COST	188443.23	COST	
-1199173.86	PVTL	540821.15	PVTL	
5.00	YEAR	15.00	YEAR	
160233.20	COST	191869.48	COST	
-1038940.66	PVTL	732690.62	PVTL	
6.00	YEAR	16.00	YEAR	
163146.53	COST	195358.01	COST	
-875794.12	PVTL	928048.63	PVTL	
7.00	YEAR	17.00	YEAR	
166112.83	COST	198909.98	COST	
-709681.29	PVTL	1126958.61	PVTL	
8.00	YEAR	18.00	YEAR	
169133.07	COST	202526.52	COST	
-540548.22	PVTL	1329485.13	PVTL	
9.00	YEAR	19.00	YEAR	
172208.21	COST	206208.82	COST	
-368340.01	PVTL	1535693.95	PVTL	
10.00	YEAR	20.00	YEAR	
175339.27	COST	209958.07	COST	
-193000.74	PVTL	1745652.02	PVTL	

7A.1 EVALUATION OF THE ABILITY OF OUTSIDE AIR IN WINTERTO PRODUCE SUPPLEMENTARY ICEFOR PENNSYLVANIA REDEVELOPMENT SCHEME, WASHINGTON, D.C.Energy Analysis

The system will utilize the ability of cold air in winter time to generate ice and make up the deficiency in "cooling" energy required which 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:

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

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:

HP-ICES PROJECT

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

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

$$\begin{aligned} \text{Total energy input} &= \text{fan energy} + \text{pump energy} = 53,370 \text{ KWH} + 65,000 \text{ KWH} \\ &= 118,370 \text{ KWH} \end{aligned}$$

$$\text{Winter heat pump energy input (see 14.1.8)} = 3,250,000 \text{ KWH}$$

$$\begin{aligned} &\text{Add} = \underline{118,370 \text{ KWH}} \\ \text{Subtotal for winter} &= \underline{3,368,370 \text{ KWH}} \\ \text{Summer heat pump energy input (DHW)} &= \underline{367,800 \text{ KWH}} \\ \text{Total annual energy consumption} &= \underline{3,736,170 \text{ KWH}} \end{aligned}$$

Cost Analysis

Energy Cost

Winter

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

Summer

Fans and Pumps

$$\begin{aligned} \text{energy } 118,370 \text{ KWH} \times \$0.02158 &= \$2,554 \\ \text{demand } (73+89) \text{ KW} \times \$3.17 \times 7 \text{ Mo} &= \underline{\$3,572} \\ \text{Subtotal} &= \underline{\$6,126} \end{aligned}$$

Heat Pump

$$\begin{aligned} \text{energy } 367,800 \text{ KWH} \times \$0.02261 &= \$8,315 \\ \text{demand } 650 \times 0.746 \text{ KW} \times \$2.4/\text{KWH} \times 1.5 \text{ Mo} &= \underline{1,746} \\ \text{Subtotal} &= \underline{\$10,061} \end{aligned}$$

$$\text{Total for year} = \$129,380$$

Equipment First Cost

$$\begin{aligned} \text{Heat Pump (see 14.2.12)} &= \$2,600,000 \\ \text{Additional ice bin storage} &= 750,000 \\ \text{42 Axial fans} &= 126,000 \\ \text{1 pump} &= 3,000 \\ \text{outside air coil } (\$0.18 \text{ per cfm}) &= \underline{270,000} \\ \text{Total} &= \underline{\$3,752,000} \end{aligned}$$

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

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

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 where there are sufficient degree hours below 25° to generate the deficiency in ice.

7A.2 MODIFIED ACES SYSTEM WITH SOLAR RADIATION
FOR DONALDSON CORPORATION, MINN.

(Table - 7A-1)

Annual Energy Requirements For Heating, Domestic Hot Water
And Cooling

MONTH	SPACE HEATING MILLION BTU	D.H.W. MILLION BTU	TOTAL HEATING & D.H.W. MILLION BTU	COOLING TON-HOURS
JANUARY	6834	35	6869	0
FEBRUARY	5271	31	5302	0
MARCH	4409	38	4447	0
APRIL	2020	33	2053	40,000
MAY	606	36	642	80,000
JUNE	94	36	130	104,000
JULY	15	33	48	176,000
AUGUST	23	38	61	176,000
SEPTEMBER	442	33	475	152,000
OCTOBER	1325	35	1360	72,000
NOVEMBER	4057	35	4092	0
DECEMBER	5691	33	5724	0
ANNUAL TOTAL	30,787	416	31,203	800,000

Ice Storage Bin Size

Annual heating requirement $31,203 \times 10^6$ Btu

With a heat pump COP = 3.2

The cooling potential is:

$$\frac{31,203 \times 10^6}{3.2} \times 2.2 = 21,452 \times 10^6 \text{ Btu}$$

$$\text{OR } \frac{21,452 \times 10^6}{12,000} = 1,787,672 \text{ Ton-Hours per Year}$$

This amounts to twice of the summer cooling requirement of 800,000 Ton-Hours.

The ice bin will thus be sized for the full cooling load plus the summer storage losses. (The ice bin will now be insulated). The losses are estimated at 12.5% of the cooling load.

Thus the storage capacity:

$$= 800,000 \times 1.125 = 900,000 \text{ Ton-Hours}$$

with 144 Btu/lb of latent heat and 50 lb per cubic foot for ice the storage volume is given by:

$$\frac{900,000 \times 12,000}{144 \times 50} = 1,500,000 \text{ Cu Ft.}$$

Assuming that the chilled water will be cooled down from 57°F to 32°F before being converted to ice, the heat extracted from 1 lb. of chilled water is:

$$(57 - 32) + 144 = 169 \text{ Btu/lb.}$$

Heating Capacity of the Heat Pump

Cooling capacity:

$$= 1,500,000 \times 50 \times 169$$

$$= 12,675 \times 10^6 \text{ Btu}$$

$$= \frac{12,675 \times 10^6}{12,000} = 1,056,250 \text{ Ton-Hours}$$

Energy Consumption Per Year

Heat pump compressor input:

$$\frac{31,203 \times 10^6}{3.2 \times 3413} = 2.857 \times 10^6 \text{ kWh}$$

$$\begin{aligned} \text{Energy cost} &= 2.857 \times 10^6 \times \$0.018 \\ &= \$51.426 \end{aligned}$$

$$\text{Heat supplied by heat pump} = 31,203 \times 10^6 \text{ Btu}$$

Heating load = 12,500,000 Btu (after deducting internal heat gain)

$$\text{Equivalent full load hours for heating} = \frac{31,203 \times 10^6}{12,500,000} = 2496$$

$$\text{KW} = \frac{\text{kWh}}{\text{Hours}} = \frac{2.857 \times 10^6}{2496} = 1145 \text{ kW}$$

(Chilled and hot water pumps not considered in any scheme)

Note: This figure is obtained also by considering part load.

$$\begin{aligned} \text{Billing demand charge} &= 1145 \times \$4.15 \times 5 \text{ months} \\ &= \$23,759 \end{aligned}$$

For pumps consider 96 kW =

$$\text{Charge} = 96 \times 4.15 \times 5 = \$1992$$

$$\text{Total Cost} = \$51,426$$

$$\$23,759$$

$$\underline{\$ 1,992}$$

$$\$77,177$$

The heating capacity is given by:

$$\frac{12,675 \times 10^6 \times 3.2}{2.2} = 18,436 \times 10^6 \text{ Btu}$$

$$\text{Heat required:} = 31,203 \times 10^6$$

Heat supplied by heat pump generating ice for cooling:

$$= 18,436 \times 10^6 \text{ Btu}$$

Heat supplied by heat pump generating ice melted:

$$= 12,767 \times 10^6 \text{ Btu (to assist heat pump)}$$

Supplementary Solar Heating

Use solar heat average incident solar energy on vertical wall facing south is 186,400 Btu per square foot per season (Ref. ACES H. FISHER)

$$\text{Solar energy required} = \frac{12,767 \times 2.2}{3.2} \times 10^6 = 8.777 \times 10^6 \text{ Btu}$$

Area of solar radiator (based on very low water temperatures)

$$= \frac{8,777 \times 10^6}{186,400} = 47,089 \text{ Sq. Ft.}$$

First Cost

Ice storage bin \$0.5/Cu Ft (part of building Structure)	\$750,000
Two ice makers	\$200,000
Two screw compressors (with the whole refrigerating system)	<u>\$200,000</u>
Subtotal	\$1,150,000
Solar collector system	<u>500,000</u>
Total	\$1,650,000

Annual Operating Cost

Energy	
Cooling and Heating	\$ 77,177
Maintenance	\$ 4,000
Insurance	\$ 3,000
Personnel	<u>\$ 40,000</u>
Total	\$ 124,177

Present Worth

26.78987 x 124,177	=	\$3,326,685
First Cost	=	<u>\$1,650,000</u>
PW		\$4,976,685
<u>Say PW</u>	=	<u>\$4,977,000</u>

Note: Solar radiators are used (instead of collectors) due to the fact that the water has a low temperature (close to 32°F), lower than ambient, and therefore, insulation and glazing not required. Moreover, the radiator will collect heat through conduction and convection.

Also the radiator system can be used to dissipate heat in summer time at night.

7A.3 DIESEL ENGINE DRIVEN ICE GENERATING HP-ICES

Energy Analysis

Dual fuel engine, Colt Industries, 960 HP, 720 RPM, 6 cylinder, opposed piston, Model Fairbanks Morse 38DD8.

$$\text{Heat input } 7256 \text{ Btu/bhp-hr (efficiency} = \frac{2545}{7256} = 0.351)$$

No. 2 oil, 19560 Btu/lb, 7.119 lb/gal, 139,250 Btu/gal, Sp. Gr. = 0.8546

Heat Balance (per bhp-hr)

Useful Work	=	2545 Btu (35%)	
Cooling Water	=	904 Btu (12%)	} 31%
Lube Oil	=	1357 Btu (19%)	
Exhaust	=	2352 Btu (32%)	
Radiation & Unacc	=	98 Btu (2%)	
Total	=	7256 Btu(100%)	

Heat Recovered

Exhaust 16,060 lb/hr at a temperature 630°F, 330°F temperature difference, sp. heat = 0.24 Btu/lb - °F

$$\text{Heat Recovered} = 16,060 \times 0.24 \times 330 \times \frac{1}{960}$$

$$= 1325 \text{ Btu/bhp-hr}$$

$$= \frac{1325}{2352} \times 100 = 56.3\% \text{ of exhaust heat}$$

Heat from lube oil recovered through jacket water i.e.

$$1357 + 904 = 2261 \text{ Btu/bhp-hr}$$

$$\text{Total recovered} = 2261 + 1325 = 3586 \text{ Btu/bhp-hr}$$

$$\text{Diesel Cycle Efficiency} = \frac{\text{Work} + \text{Heat Recovered}}{\text{Heat Input}}$$

$$= \frac{2545 + 3586}{7256} \times 100 = 84.5\%$$

For 960 HP.

Useful Work	960 x 2545	=	2,443,000 Btu/hr
Cooling Water	960 x 904	=	868,000 Btu/hr
Lube Oil	960 x 1357	=	1,303,000 Btu/hr
Exhaust	960 x 2352	=	2,258,000 Btu/hr
Radiation and Unacc.	960 x 98	=	<u>94,000 Btu/hr</u>
Total		=	6,966,000 Btu/hr

$$\text{Heat recovered from jacket water} = 2,171,000 \text{ Btu/hr}$$

$$\text{Heat recovered from exhaust} = \underline{1,272,000 \text{ Btu/hr}}$$

$$\text{Subtotal} = 3,443,000 \text{ Btu/hr}$$

$$\text{Work} = \underline{2,443,000 \text{ Btu/hr}}$$

$$\text{Total} = 5,886,000 \text{ Btu/hr}$$

$$\text{Efficiency (Check)} = \frac{5,886,000}{6,966,000} \times 100 = 84.5\%$$

Heat Pump

$$\text{Use COP}_c = 2.2$$

$$\begin{aligned} \text{Cooling obtained} &= 2.2 \times \text{work input} = 2.2 \times 2,443,000 \text{ Btu/hr} \\ &= 5,375,000 \text{ Btu/hr} \end{aligned}$$

Adjusted for Source Energy

$$\text{COP}_c = \frac{5,375,000 \text{ Btu/hr}}{6,966,000 \text{ Btu/hr}} = 0.772$$

Heating Obtained

$$\text{From heat pump} = 3.2 \times 2,443,000 = 7,818,000 \text{ Btu/hr}$$

$$\text{From recovered heat} = \underline{3,443,000 \text{ Btu/hr}}$$

$$\text{Total} = 11,261,000 \text{ Btu/hr}$$

 COP_H (related to shaft work)

$$= \frac{11,261,000 \text{ Btu/hr}}{2,443,000 \text{ Btu/hr}} = 4.61$$

 COP_H (related to source energy)

$$= \frac{11,261,000 \text{ Btu/hr}}{6,966,000 \text{ Btu/hr}} = 1.62$$

Overall COP

$$= \frac{\text{Cooling} + \text{Heating}}{\text{Work}}$$

$$\text{Cooling} = 5,375,000 \text{ Btu/hr}$$

$$\text{Heating} = \underline{11,261,000 \text{ Btu/hr}}$$

$$\text{Total} = 16,636,000 \text{ Btu/hr}$$

COP Overall

$$= \frac{16,636,000 \text{ Btu/hr}}{2,443,000 \text{ Btu/hr}} = 6.8$$

COP Overall

$$= \frac{16,636,000 \text{ Btu/hr}}{6,966,000 \text{ Btu/hr}} = 2.39$$

7A.4 GAS ENGINE DRIVEN ICE GENERATING HP-ICES

Caterpillar G398, 500 HP, 1,200 RPM, 12 Cylinder

Heat Balance

Network	=	1,272,600 Btu/hr	(30.2%)
Cooling Water	=	1,174,900 Btu/hr	(27.8%)
Lube Oil	=	211,100 Btu/hr	(5.0%)
Exhaust	=	1,310,300 Btu/hr	(31.0%)
Radiation and Unaccounted	=	253,300 Btu/hr	(6.0%)
Total	=	4,222,200 Btu/hr	(100%)
Heat Rate	=	$\frac{4,222,200}{500}$	= 8444 Btu/bhp (30.2% eff.)

Heat Recovered

Exhaust	=	733,700 Btu/hr	(56%)
Cooling Water (& Lube Oil)	=	<u>1,386,000 Btu/hr</u>	
Subtotal	=	2,119,700 Btu/hr	
Work	=	<u>1,272,600 Btu/hr</u>	
Total	=	3,392,300 Btu/hr	
Engine Cycle Efficiency	=	$\frac{3,392,200 \text{ Btu/hr}}{4,222,200} \times 100$	= 80.3%

Heat Pump

$$\text{Use COP}_c = 2.2$$

$$\text{Cooling obtained} = 1,272,600 \times 2.2 = 2,800,000 \text{ Btu/hr}$$

Adjusted for source energy

$$\text{COP}_c = \frac{2,800,000 \text{ Btu/hr}}{4,222,200 \text{ Btu/hr}} = 0.663$$

Heating Obtained

$$\text{From Heat Pump} = 3.2 \times 1,272,600 = 4,072,300 \text{ Btu/hr}$$

$$\text{From Heat Recovery} = \underline{2,119,700 \text{ Btu/hr}}$$

$$\text{Total} = 6,192,000 \text{ Btu/hr}$$

COP_H (related to shaft work)

$$= \frac{6,192,000 \text{ Btu/hr}}{1,272,600 \text{ Btu/hr}} = 4.87$$

COP_H (related to source energy)

$$= \frac{6,192,000 \text{ Btu/hr}}{4,222,200 \text{ Btu/hr}} = 1.47$$

Overall COP

$$= \frac{2,800,000 + 6,192,000}{1,272,600} = 7.07$$

related to shaft work

Overall COP

$$= \frac{8,992,000}{4,222,200} = 2.13$$

related to source energy

fig7A-3 Annual Overall COP of a Diesel Engine, Gas Engine,
and Electric Drive Heat Pump.

COP Related to Mechanical Work Input

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