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Heat-Pump-Centered Integrated Community Energy Systems

System Development

Dubin-Bloome Associates Final Report

MASTER

ICES
INTEGRATED COMMUNITY ENERGY SYSTEMS

ENERGY AND ENVIRONMENTAL SYSTEMS DIVISION

ARGONNE NATIONAL LABORATORY

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ANL/CNSV-TM-8

ARGONNE NATIONAL LABORATORY
9700 South Cass Avenue
Argonne, Illinois 60439

HEAT-PUMP-CENTERED INTEGRATED COMMUNITY ENERGY SYSTEMS

System Development
Dubin-Bloome Associates Final Report*

by

Fred S. Dubin, Amos Halfon, and Paul Herzog

DUBIN-BLOOME ASSOCIATES, P.C.
42 West 39th Street
New York, N.Y. 10018

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PREFACE

More than one-quarter of the energy consumed in the United States is used to heat and cool buildings and to heat service water. Approximately 83% of the energy for these services is supplied from natural gas and oil. Recognition of the need to conserve these and other natural resources has been spurred, in large part, by the oil embargo of 1973 and subsequent escalation in fuel prices, by energy shortages, by increasing electricity "brown-outs" in urban centers, and by the crippling effects of labor strikes and stoppages in the energy supply industries. In recent years, development and acceptance of energy conserving systems which avoid dependence on scarce or interruptible fuels has increased markedly both in the United States and abroad.

Integrated community energy systems (ICES) are a comprehensive approach to increasing the efficiency of the varied ways in which energy is provided to and utilized by a community. ICES also offer opportunities to reduce dependence on scarce resources, protect environmental quality, reduce costs of energy and energy-consuming services, and, perhaps most importantly, meet energy needs without adversely affecting lifestyles.

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 precisely 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 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 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 does not arbitrarily define either the energy services to be provided or the type, size, or function of the area to be served. As a result, an ICES is tailored for each application. 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.

Heat-pump-centered integrated community energy systems (HP-ICES) are energy systems for communities which provide heating, cooling and/or other thermal 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 offer significant potential for energy savings. By powering these heat pumps with nonscarce fuels, the use of which would be impractical in most conventional systems, less-abundant fuels including natural gas and oil can be conserved. Secondary benefits of HP-ICES include reduction of adverse environmental effects as compared to conventional systems, reliable production of services in contrast to the increasingly frequent utility curtailments and interruptions, and delivery of services to consumers at costs lower than those for conventional systems (including acquisition, operation, and maintenance costs).

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 1984. 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 Urban Waste and Municipal Systems Branch, Community Systems Division, Office of Buildings and Community Systems, Assistant Secretary for Conservation and Solar Applications, 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.

The report which follows presents the findings in the development and analysis of one concept under investigation. This report was prepared by Dubin-Bloome Associates, P.C.

The HP-ICES concept under examination uses the latent heat of fusion of water as a heat source for the heat pump, thus, converting the water to ice. The ice is stored in a bin and used the following summer for cooling. The development of this concept is presented in Part I. The concept is then applied to two communities, in Part II, with modifcations to meet the unique requirements of the specific community and climatic conditions. Both communities contain commercial, retail, office, hotel, and residential facilities.

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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 a 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 detailed case study on the Market Square project in Washington, D.C., it was found that for the HP-ICES the annual source energy input is about 60% and the life cycle annual average cost is 40% of the corresponding quantities for a conventional central system with equal heating and cooling capacity.

The annual average operating and administration cost for the HP-ICES is less than 70% of the corresponding costs for the conventional system, while the first cost of the HP-ICES is about 70% larger than the first cost of the conventional system. With the values assumed for the discount rate, interest rate, etc., the return on investment was found to be about 15%, which gives a discounted payback period of about 6.7 years.

For the Park Plaza in Boston, the annual source energy input for the HP-ICES is 35% and the energy cost is about 30% of the corresponding quantities for the conventional system.

The annual average operating and administration cost for the HP-ICES is 41% of the conventional system cost while the first cost of the HP-ICES is 4.5 times as great as the first cost for the conventional system. The return on investment is 13% and the payback is 8 years.

These results show that the HP-ICES can be better both in energy usage and in life cycle cost than a conventional system of the same heating and cooling capacity, and holds great promise as an energy saving system.

PART I

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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 throught 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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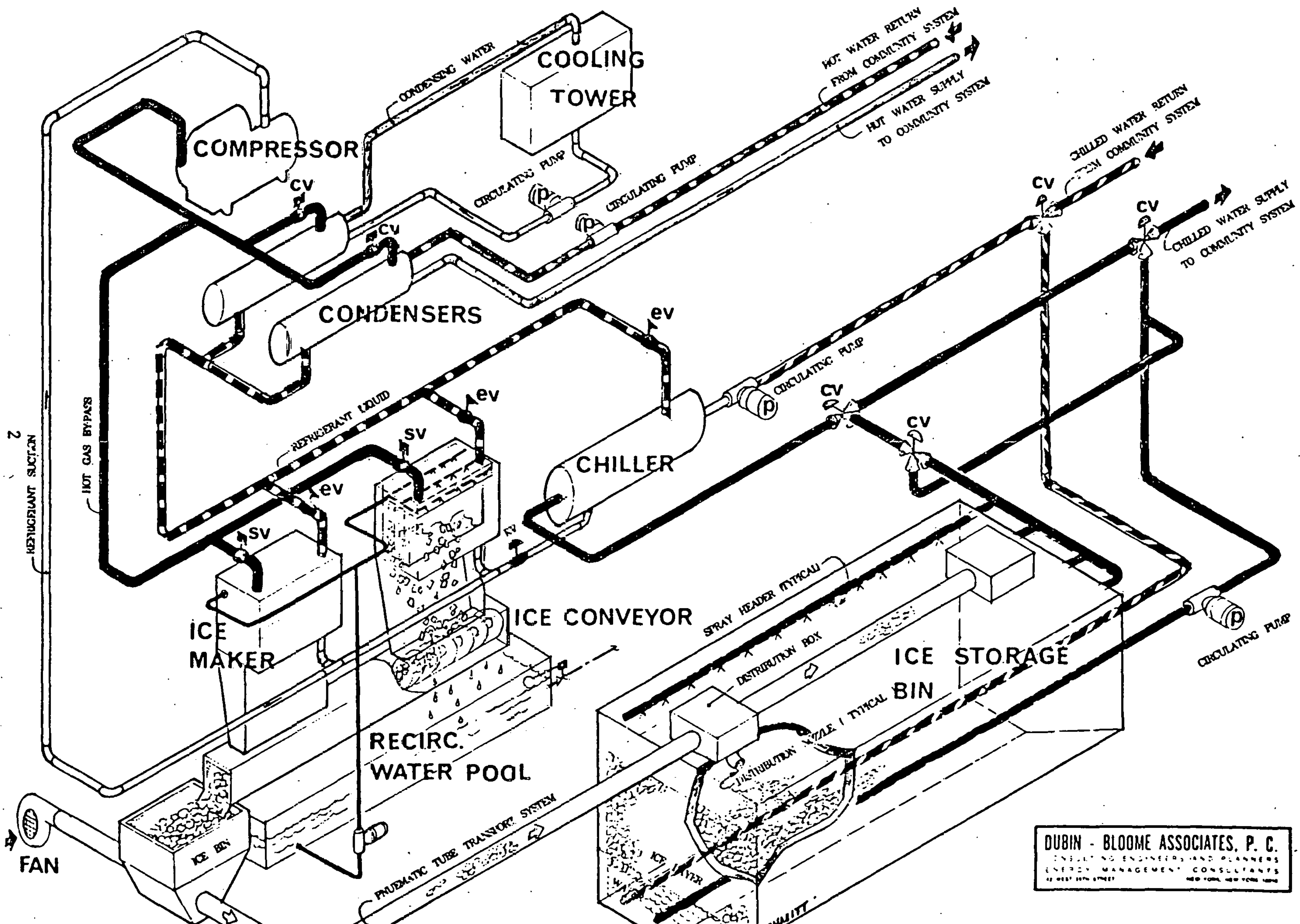


FIG. 1.1 ICE GENERATING HEAT PUMP - ICES

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.

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 sorce.

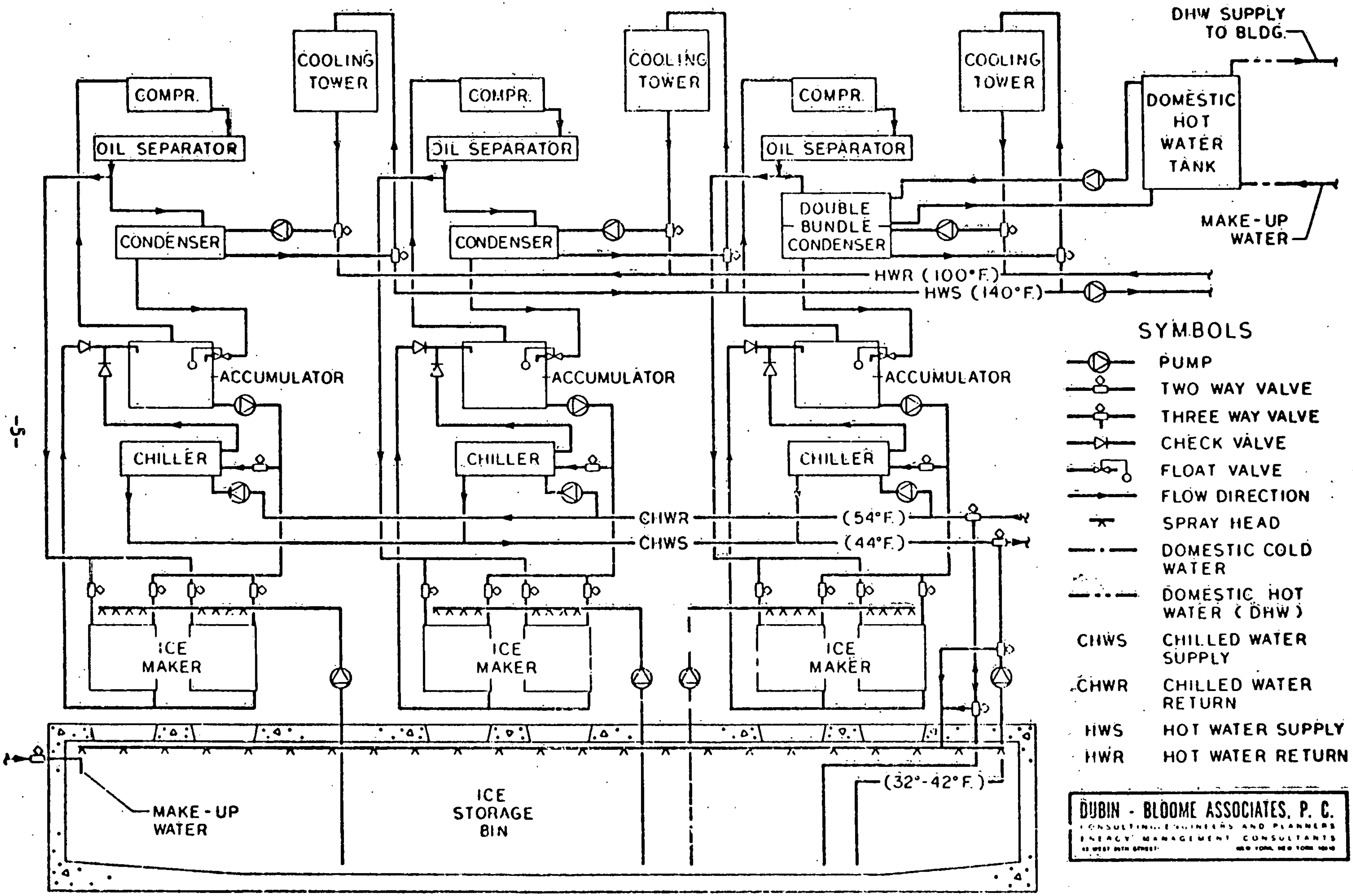
When the heating load is reduced by using the internal heat gain as a heat sorce, the heat pump will generate less ice. The reduced quantity may be insufficient for satisfying the summer cooling requirements, making it mandatory to provide supplementary cooling during the summer.

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



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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.

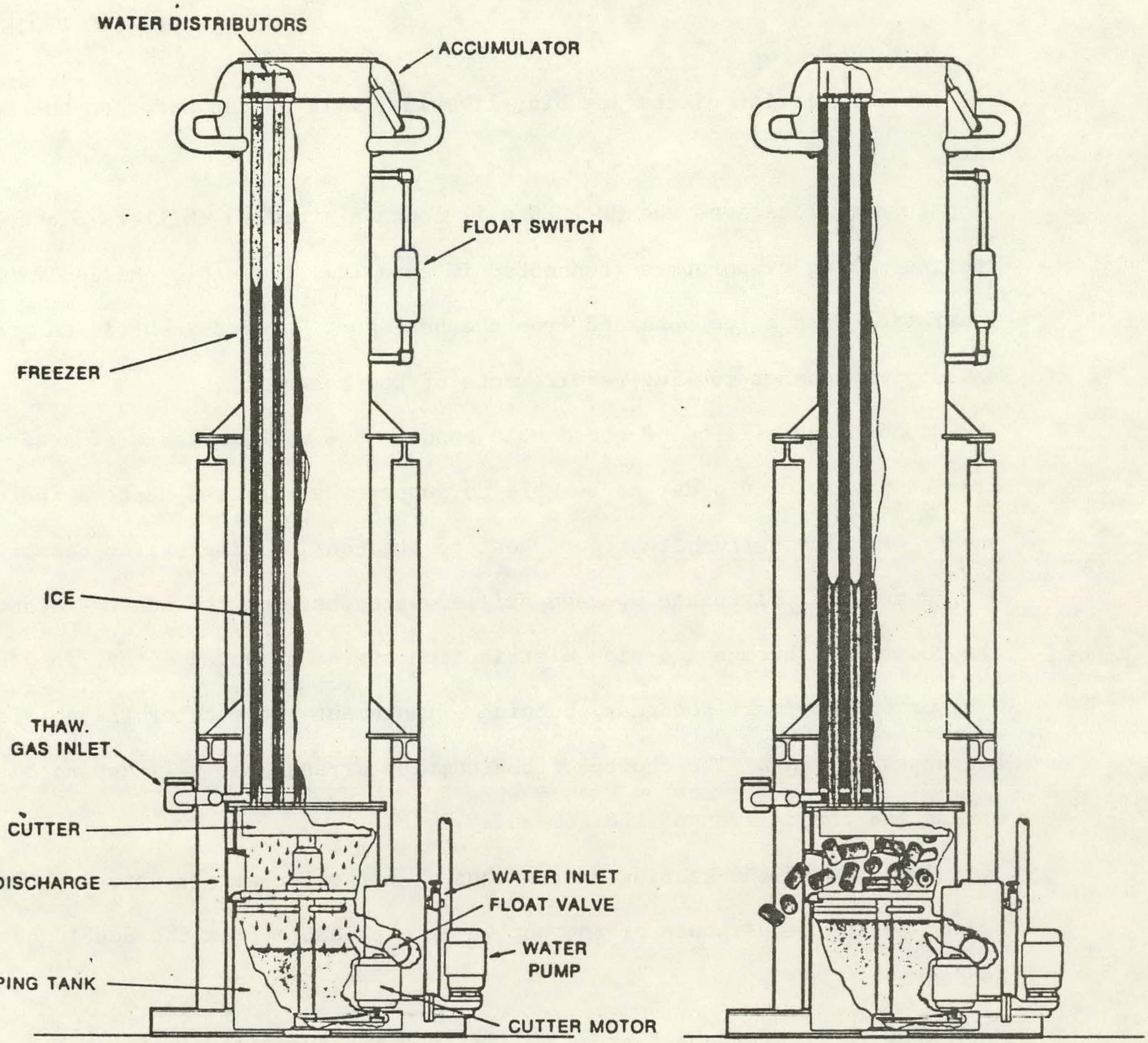
Likewise, the type of the refrigerant used should have a high mass density which will reduce the required displacement of the compressor, and a high heat capacity which will increase the refrigerating effect. The refrigerant should be relatively inexpensive and available on the market. The type of the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

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, alternatively, 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 imbalance points,certain modifications to the HP-ICES are necessary.

Whenever 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 done for air temperatures 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 supplying 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. Alternatively (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.

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:

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

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, COP_1 = 4.4.

Thus for the conventional system

$$\text{COP overall} = \frac{Q_h + Q_c}{\frac{Q_h}{E} + \frac{Q_c}{COP_1 \times 0.31}}$$

$$= \frac{15000 + 10,000}{\frac{15000}{0.6} + \frac{10,000}{4.4 \text{x} 0.31}} = 0.773$$

For the HP-ICES

$$COP_c = \frac{10,000}{15,000 - 10,000} = 2$$

$$\text{Cop overall} = 0.31\ (1 + 2\ COP_c)$$

$$= 0.31\ (1 + 2 \times 2) = 1.55$$

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

Moreover, if we assume a boiler efficiency of 100% and a 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.

$$\text{Heating plant efficiency} = \frac{Q_h}{E} = \frac{Q_c + W}{E}$$

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

For the conventional system

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

Substituting $COP_1 = Q_c/W_1$ and $COP_c = Q_c/W$ and simplifying

$$\text{COP overall} = \frac{(1 + 2\ COP_c)\ 0.31E}{0.31\ (COP_c+1)+\frac{(COP_c)}{(COP_1)}E}$$

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

$$COP_c = \frac{E - 0.31}{0.31 + E/COP_1}$$

The assumption of a heating plant seasonal efficiency of 60% and a conventional 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.

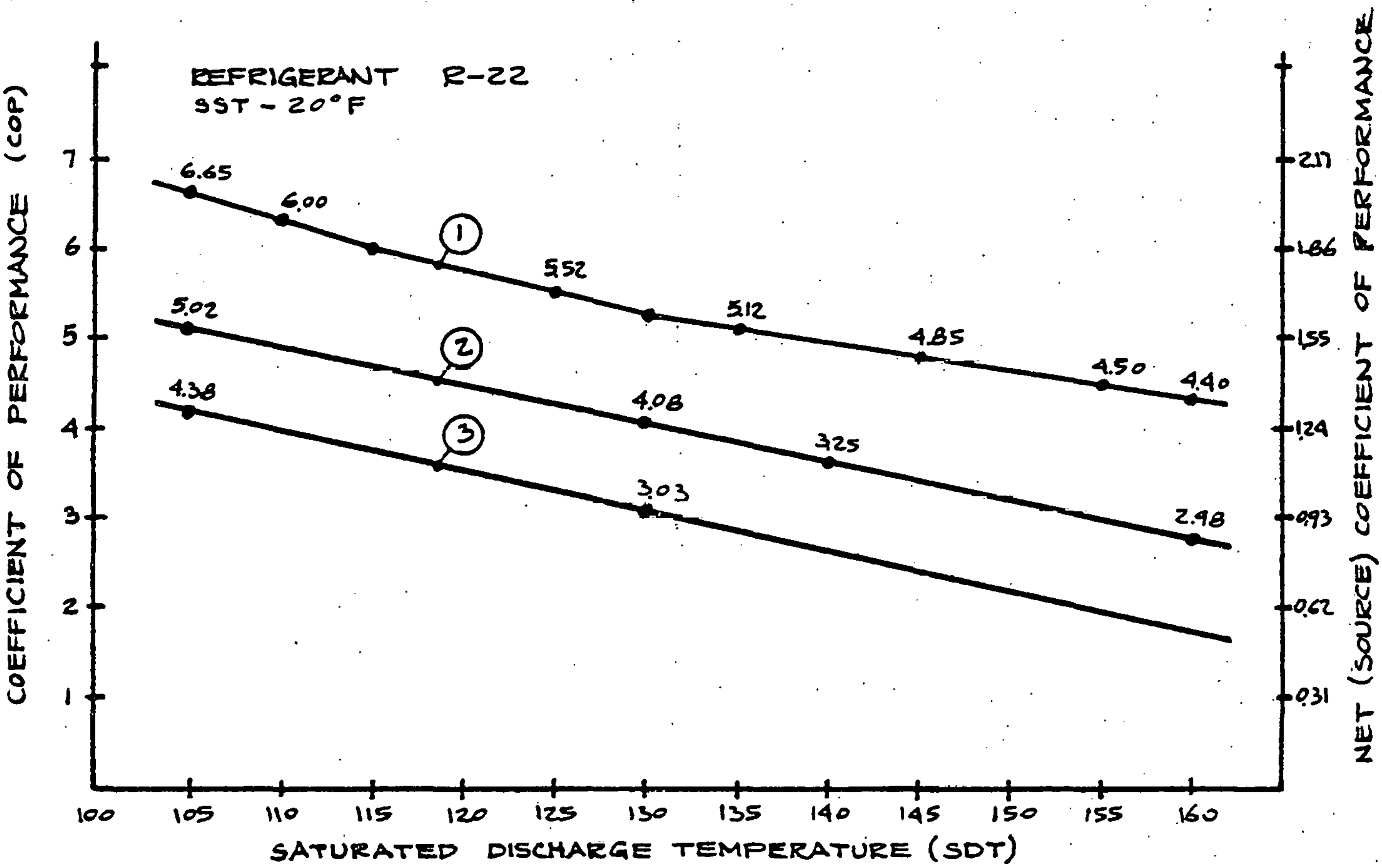


FIG. 1.4

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

DESCRIPTION OF CURVES

1) Carnot Efficiency $= \frac{TSDT}{TSDT-TSST}$ (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 determines 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 ΔT = 110°F - 90°F = 20°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 a 110°F to a 120°F water temperature raises the power input to the 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°F < SDT < 160°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 Diameter (in.)	Flow Velocity ft/sec.	Hot Water Temp. °F	Head Loss Ft per 100 ft	Gal. per Min.	hp per 100 ft of Pipe	COP	SDT F°	hp to Compressor per 10^7 Btu/hr heat rejected
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 degredation 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^{o}F water which will set the saturated discharge temperature at 130^{o}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 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 the 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

REFRIGERANT	PRESSURE PSIA P_s	PRESSURE PSIA P_d	COMPRESSOR RATIO	COP	HP/TON	REFRIGERANT FLOW PER TON LB/MIN*	REFRIGERANT FLOW PER TON CFM**	DISCHARGE TEMP. °F.
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

P_d = Discharge Pressure

* THROUGH THE EVAPORATOR

** AT COMPRESSOR SUCTION

TABLE 1.3

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

REFRIGERANT	PRESSURE PSIA		COMPRESSION RATIO	COP		HP/TON		REFRIGERANT FLOW PER TON						DISCHARGE TEMP. °F
								LB/MIN *			CFM **			
								FIRST STAGE	SECOND STAGE		FIRST STAGE	SECOND STAGE		
				INTER-COOLING		INTER-COOLING			INTER-COOLING			INTER-COOLING		
	P_S	P_D		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	$T_S = 20^oF/T_D = 130^oF$				$T_S = 35^oF/T_D = 105^oF$			
	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 Dunham-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 that 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)

REFRIGERANT	$T_S = 20^oF/T_D = 130^oF$							$T_S = 35^oF/T_D = 105^oF$						
	REFRIGERANT CAPACITY		POWER REQUIREMENT			HEAT REJECTED	COOLING	REFRIGERANT CAPACITY		POWER REQUIREMENT			HEAT REJECTED	COOLING
	TONS	BTU/H	HP	HP/TON	BTU/H	BTU/H	C.O.P.	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	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.

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	$T_S = 20^\circ F / T_D = 120^\circ F$		$T_S = 35^\circ F / T_D = 105^\circ F$	
	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 Dunham-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)

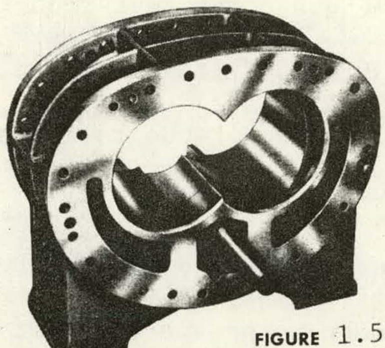


FIGURE 1.5

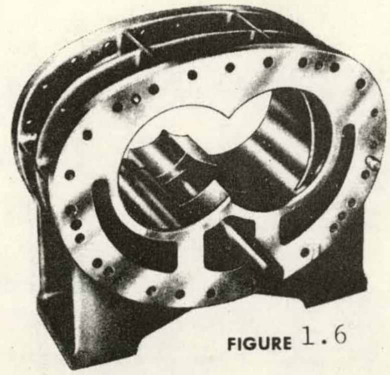


FIGURE 1.6

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 Dunham-Bush, Sullair Corporation and Freezing Equipment Sales are compared.

At part load and down to 50%, the Dunham-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 Dunham-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 Dunham-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

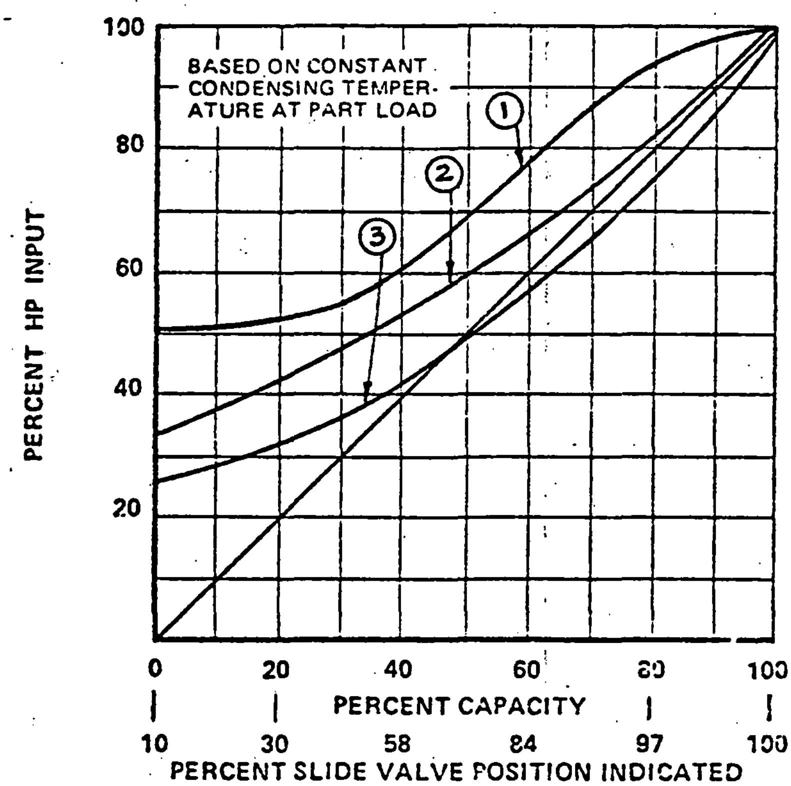


FIGURE 1.7

DESCRIPTION OF CURVES (and Reference for Source of Information)

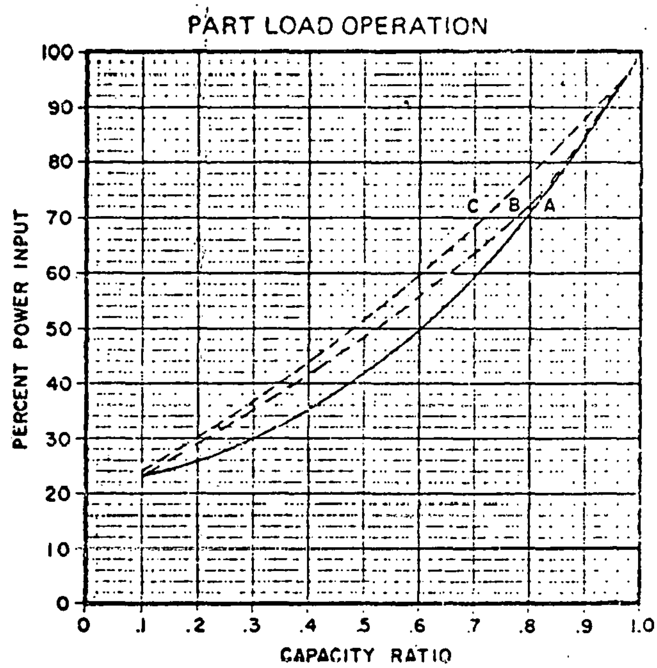
1. "FES" Rotary Screw Compressor - Technical Manual
2. "Sullair Corporation" - Selection Guide
3. "Dunham-Bush" Screw Compressor - Catalog #6043C 20°F

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-^{o}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 (^{o}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	Scare	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/ft^3) 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

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

Compressor

The compressors should be able to operate in winter time with a SST of $20^{\circ}F$ and SDT of $130^{\circ}F$ 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 compressors 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 $35^{\circ}F$ SST and $105^{\circ}F$ 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 ΔT = 30°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 both heating and cooling are required and the water storage temperature in the bin is above or equal to 32°F, the control center will set one of the compressors 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 are 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.

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FIG.-1-9 BASIC WINTER MODE

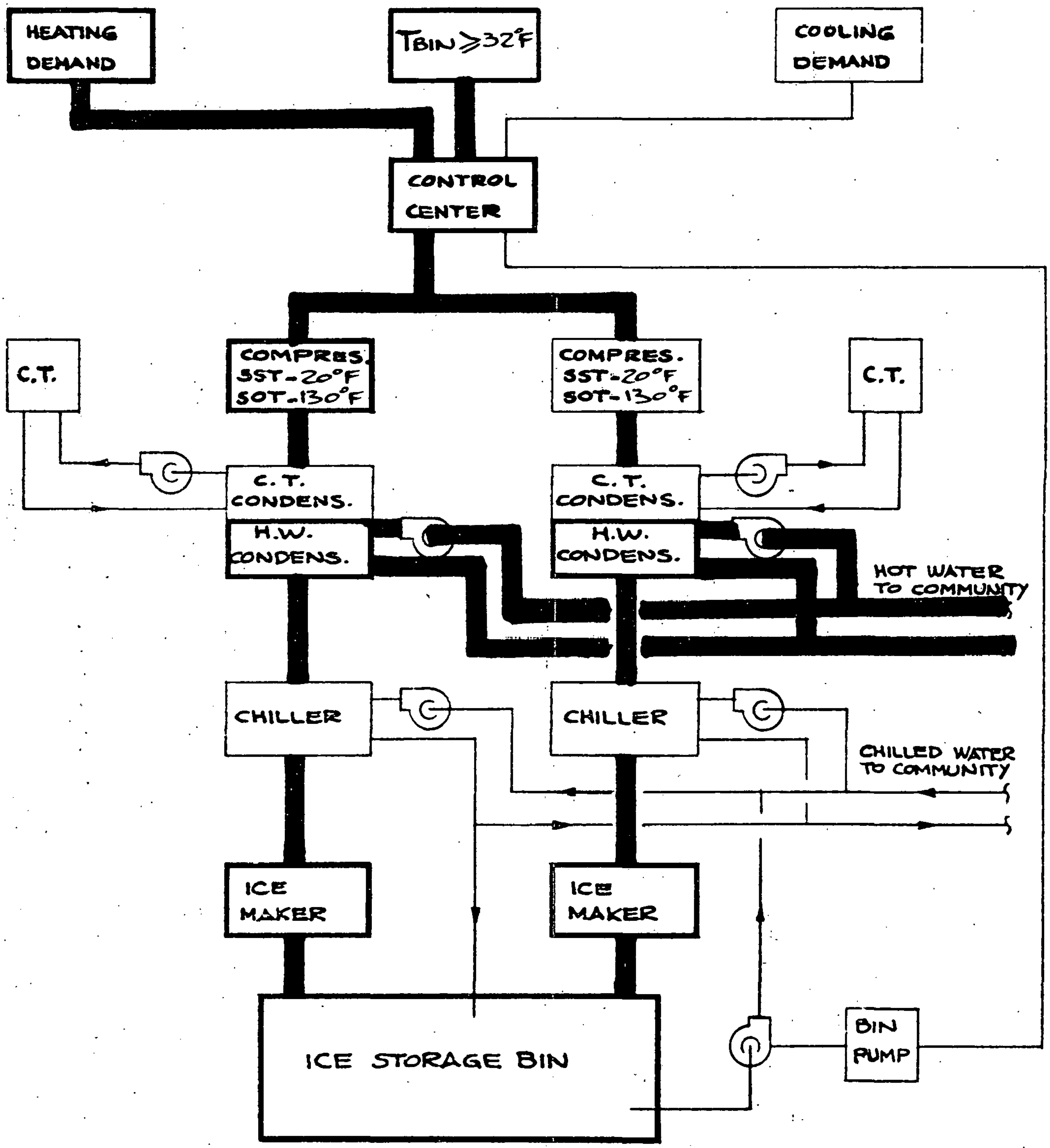


FIG.-1-10 SECOND WINTER MODE

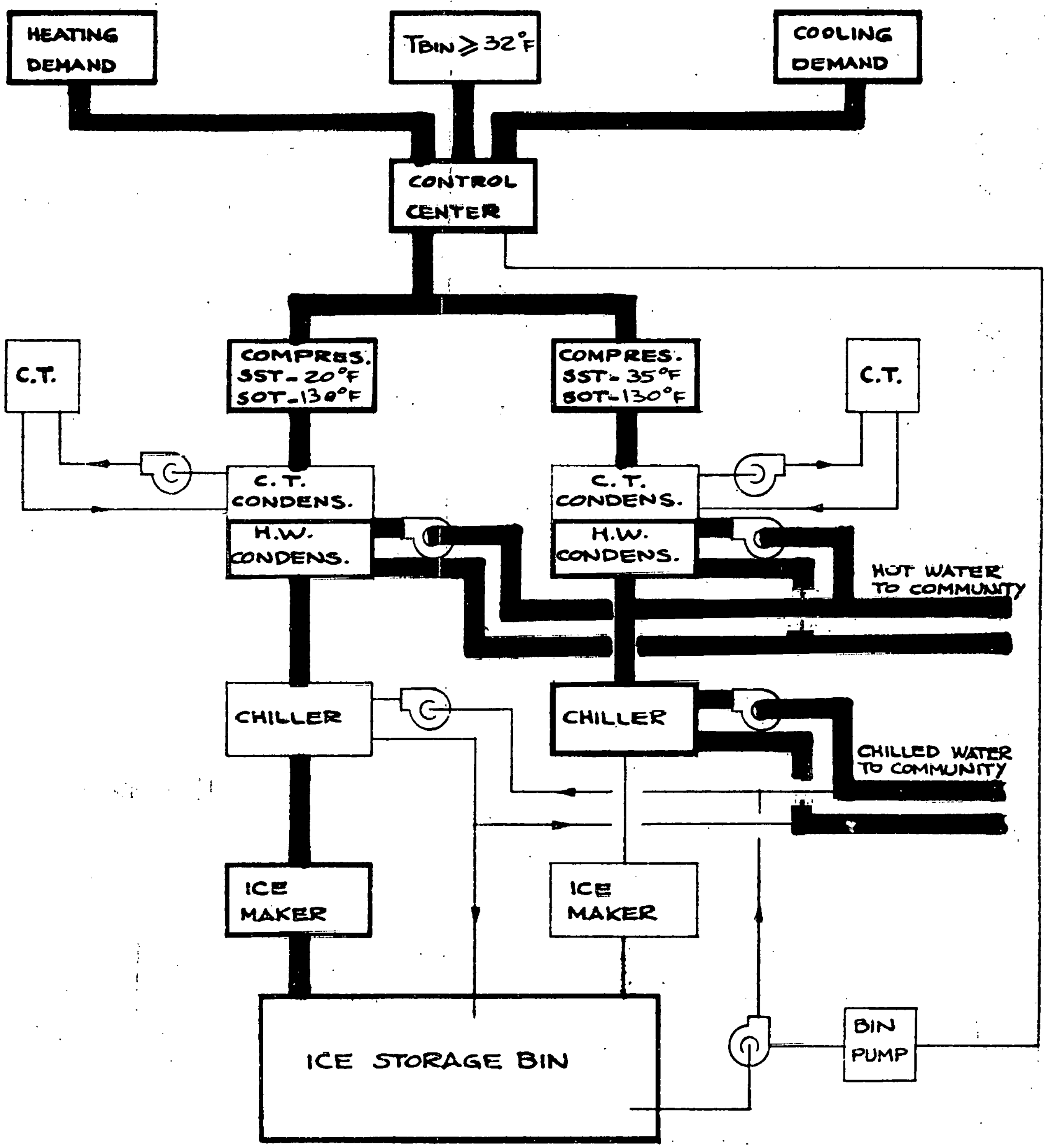


FIG. — 1-11 BASIC SUMMER MODE

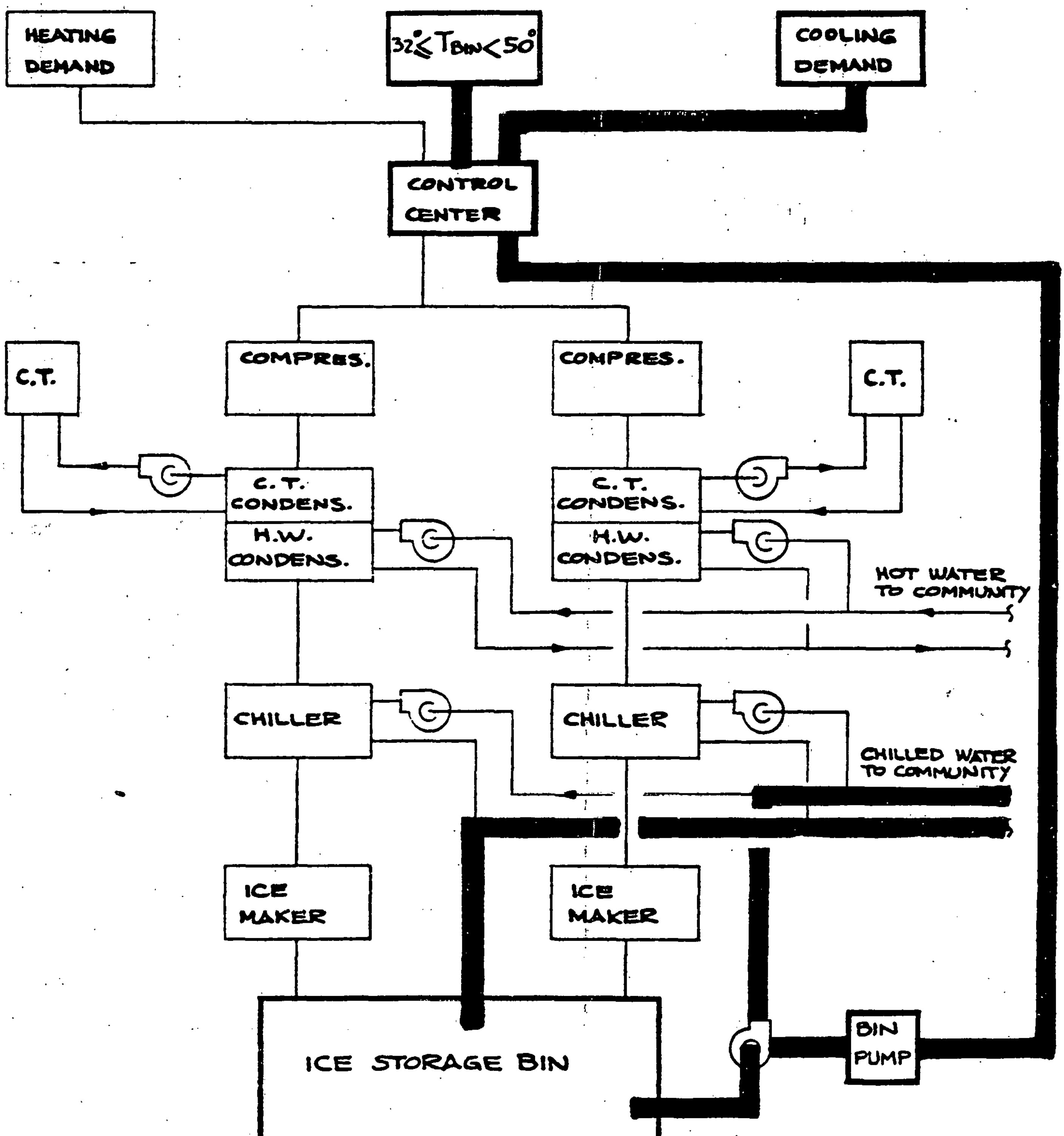


FIG.-1-12 SECOND SUMMER MODE

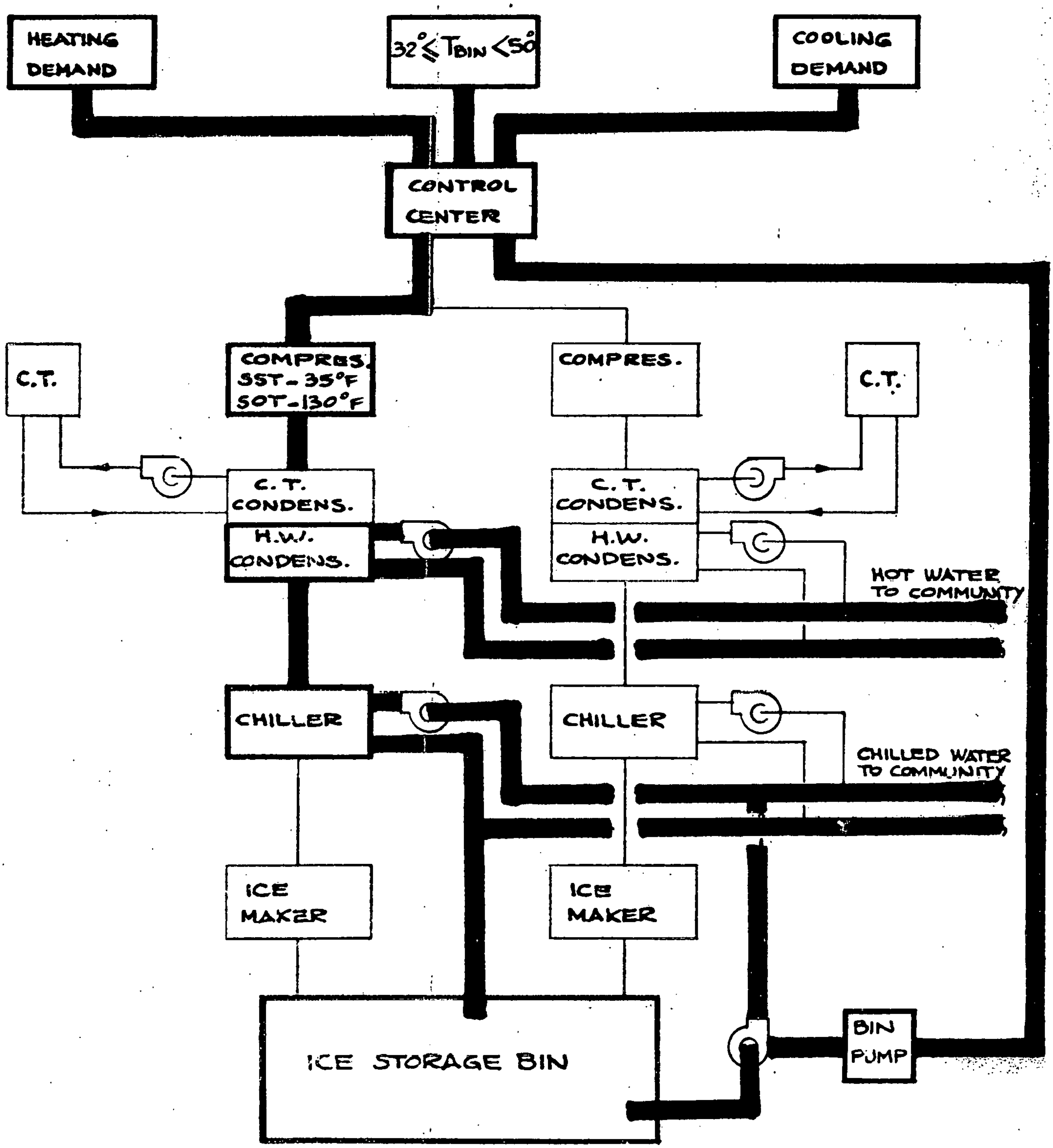


FIG.-1-13 THIRD SUMMER MODE (NITE TIME)

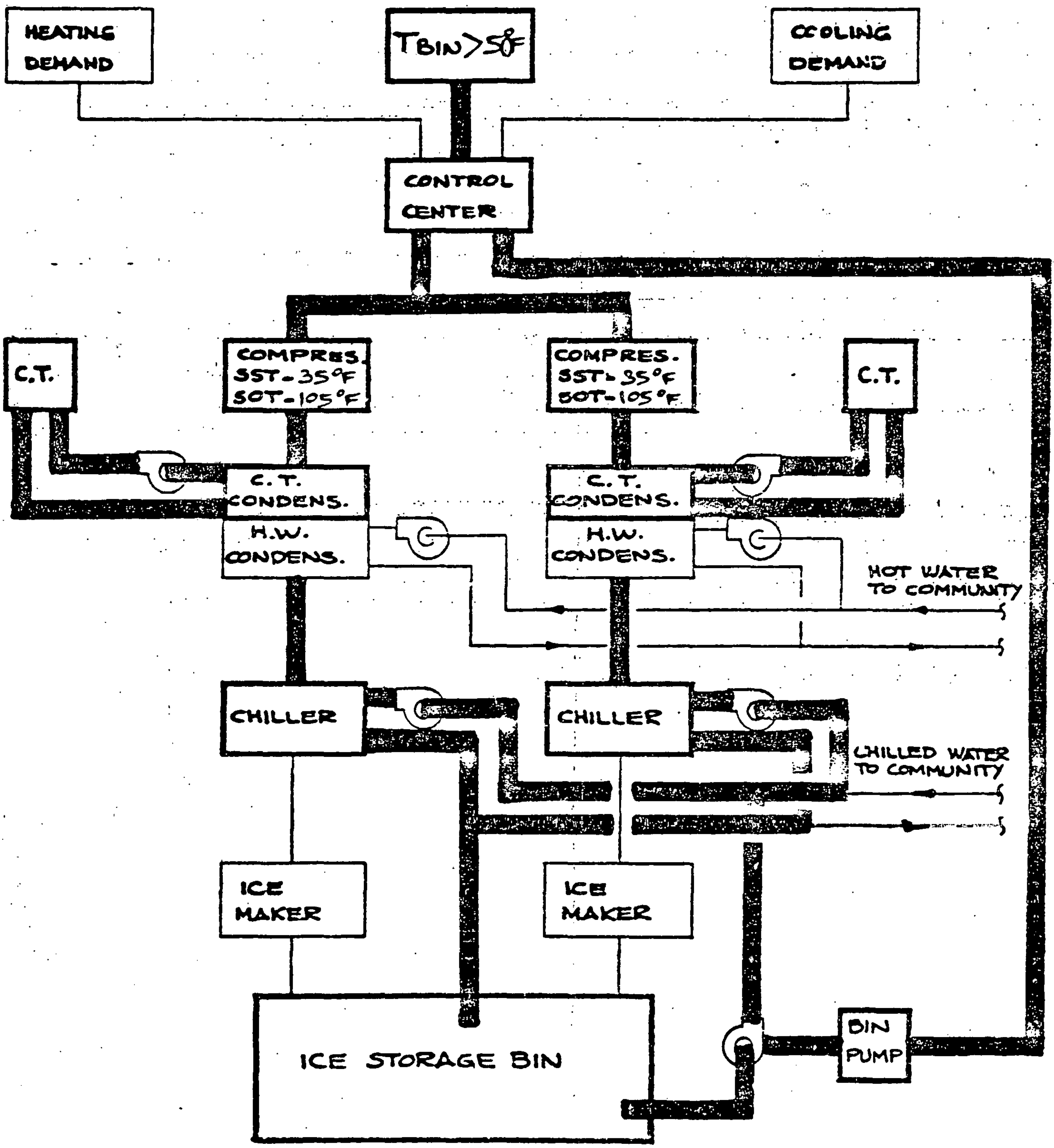
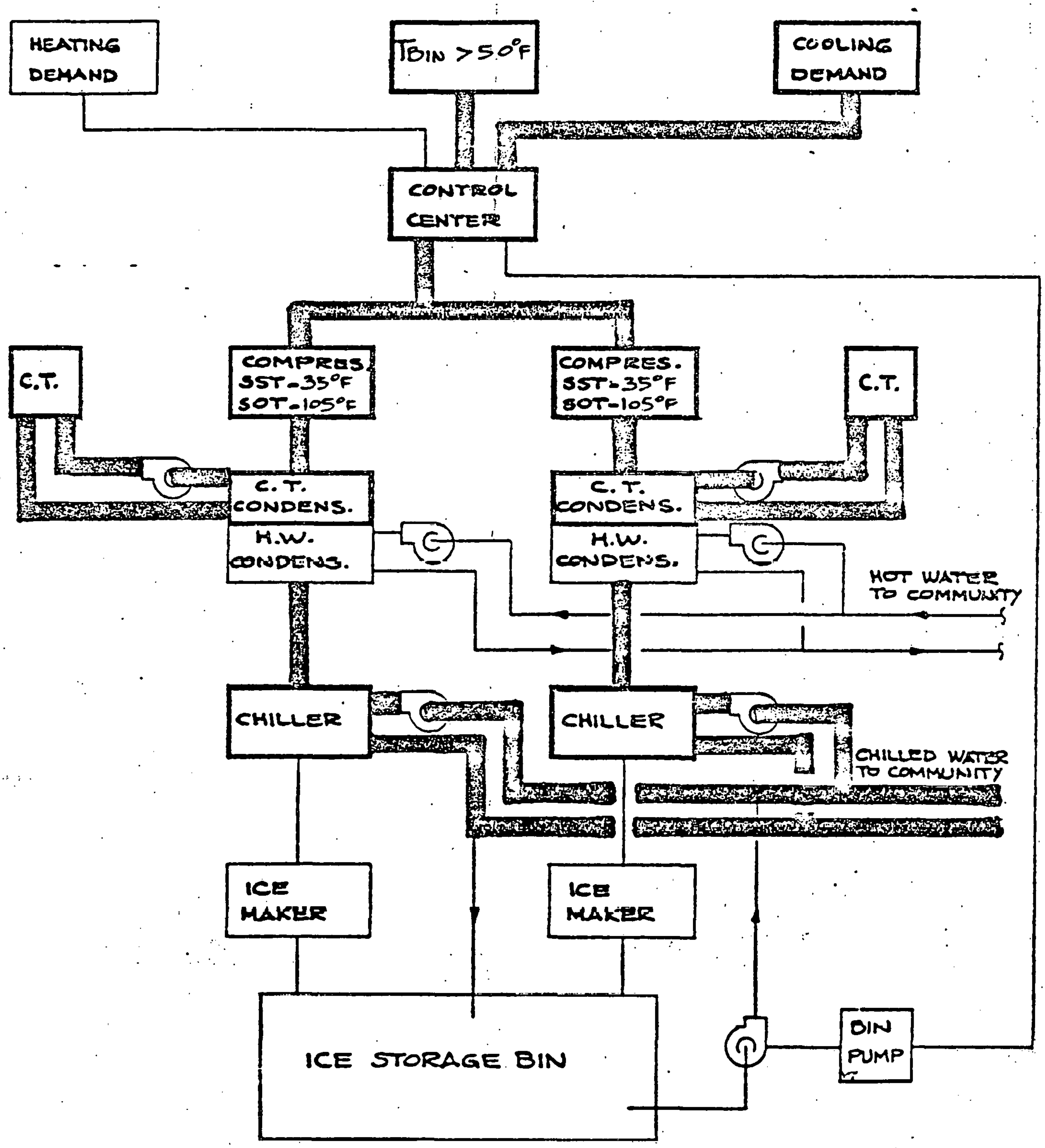


FIG.-1-14 FOURTH SUMMER MODE



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.

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 shortcomings and difficulties in the use of the HP-ICES:

a. Equipment not specifically designed for this concept may have to be used.

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 efficent 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 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, the Market Square Complex forms the focal point of the entire proposed Pennsylvaina 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 DB	Summer Design Temp °F WB	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 Heating 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	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	33088	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 Christi, 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.

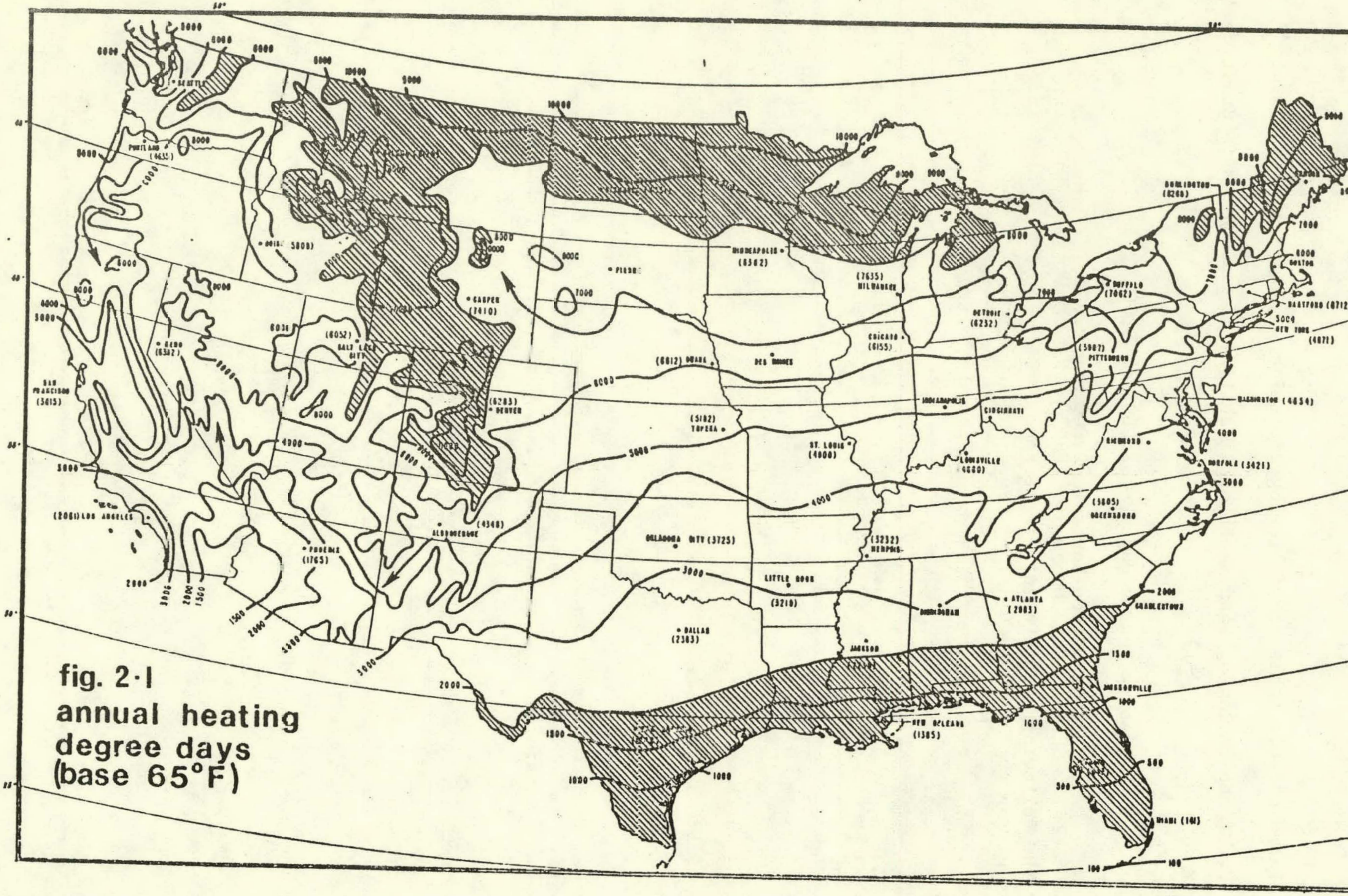


fig. 2·1
annual heating
degree days
(base 65°F)

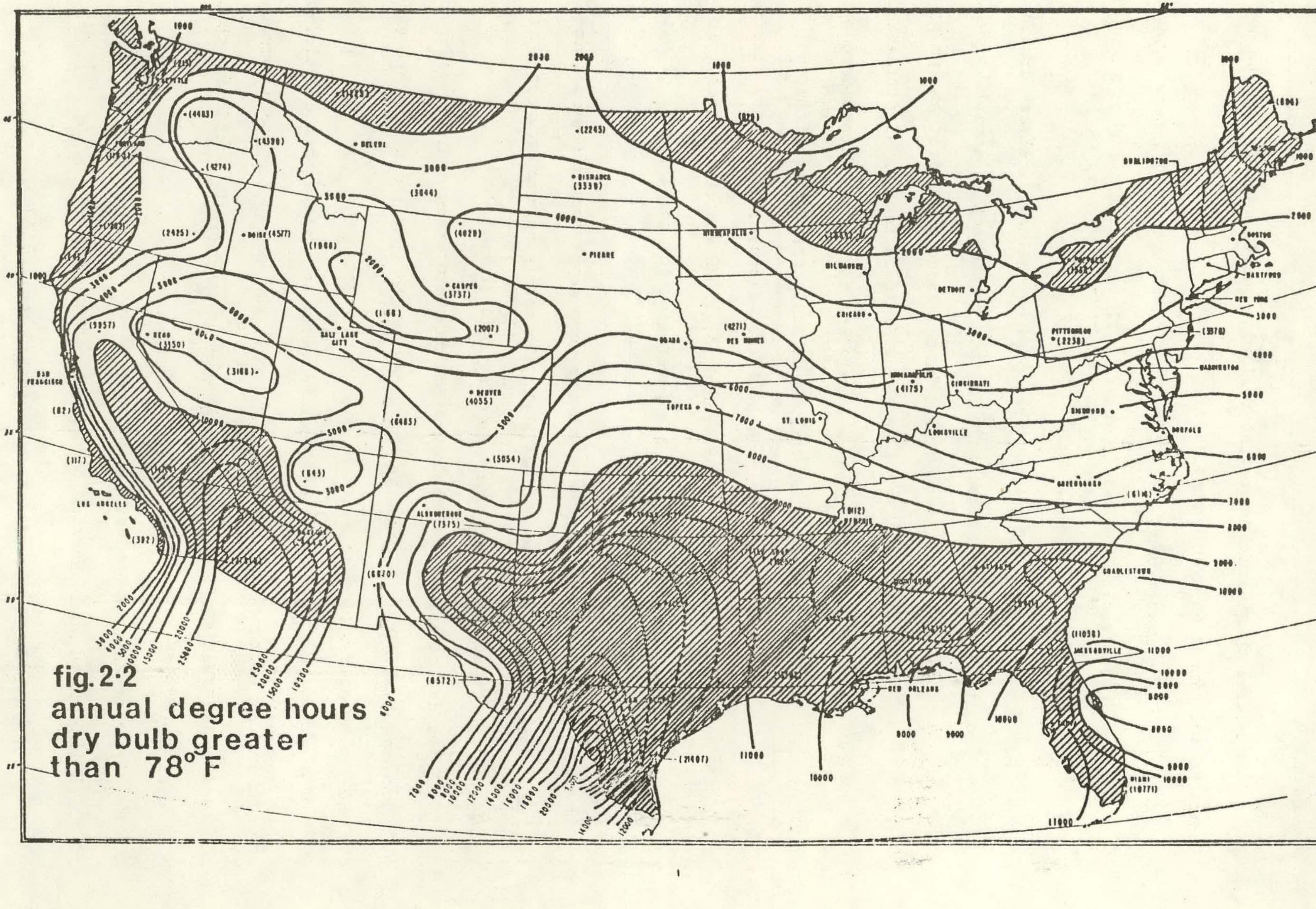


fig. 2·2
annual degree hours
dry bulb greater
than 78°F

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 required 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.

*Dunham-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^{o} to 42^{o} latitude, or the 4871 to 5634 degree day range, with latitude below 40^{o} and below 4871 degree day requiring supplementary compressor work in the summer, while latitude above 42^{o} and above 5634 degree days, including Denver and Seattle, requiring supplementary heat in the winter to melt the excess of ice.

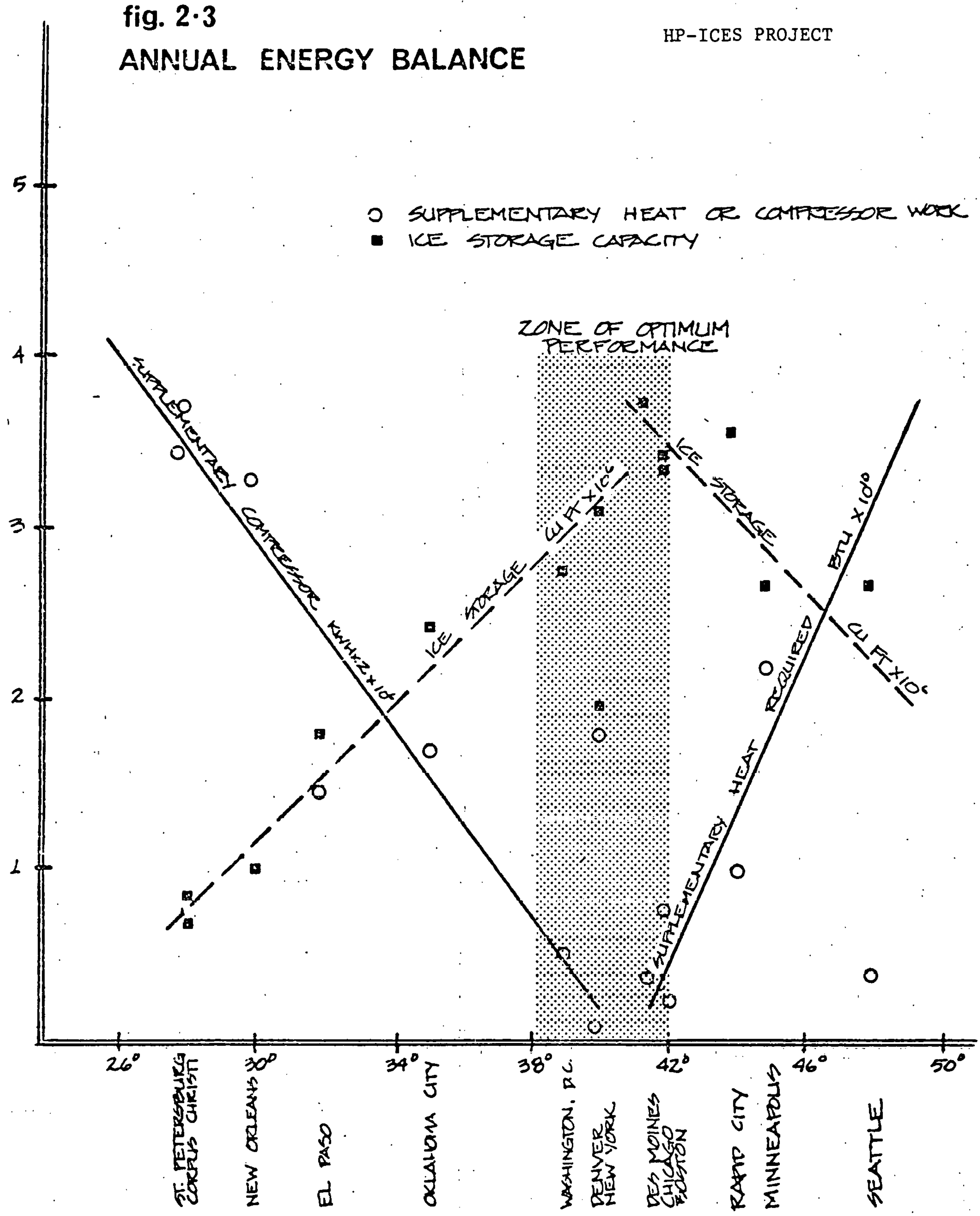
As for the ice storage bin, its size is governed by two distinct factors depending on the latitude. Thus, for the southern latitudes the ice storage bin size is governed by the ice generating capacity of the heat pump winter operation, while for the northern latitudes the size is governed by the cooling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

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

fig. 2·3

ANNUAL ENERGY BALANCE

HP-ICES PROJECT



Cities according to Latitude

The ice storage bin volume versus latitude curve (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^{o} - 40^{o})

On the other hand, for the upper latitudes (42^{o} - 48^{o}) 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

Location	Latitude	Degree Days	% of Heating
Group 1			
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

Location	Latitude	Degree Days	% of Heating
Group 2			
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%
Group 3			
Chicago, IL	42°	6639	26%
Boston, MA	42°	5634	28%
Rapid City, SD	44°	7343	21%
Des Moines, IA	42°	6588	22%
Minneapolis, MN	45°	8382	18%
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) <u>Residential</u>

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) <u>Commercial</u>

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 Eastern Cities, Argonne National Laboratory Report ANL/ICES-TM-11, prepared by Energy Systems Research Group Inc., Aug. 1978.

II A Test Case For The Potential Application of District Energy Systems Using Thermal Energy Cogenerated At Existing Electric Power Plants D.J. Santini and A.A. Davis, Argonne National Laboratory Report ANL/ICES-TM-13, January 1978.

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 week-ends. Hospital - intensive user similar to high use residential phase, the characteristic of a commercial load super-imposed be-cause 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 con-tinuous 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

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

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 heat-ing and cooling by conventional means.

TABLE 2-8 ANNUAL ENERGY REQUIREMENT FOR FOR RESIDENTIAL AND COMMERCIAL SECTORS ($Btu \times 10^6$)

			COMMERCIAL	
		RESIDENTIAL	RETAIL	OFFICE
Boston	Space HTG	79.12	53.77	53.84
	DHW	9.00	2.00	3.84
	Total HTG	88.12	55.77	57.68
	Total Cool	10.20	70.56	48.00
	Net Energy Req.	+ 50.38 Excess Ice	- 32.22 Ice Deficiency	- 8.34 Ice Deficiency
Minneapolis	Space HTG	117.71	80.00	80.1
	DHW	9.00	2.00	3.84
	Total HTG	126.71	82.00	83.94
	Total Cool	7.68	52.92	57.71
	Net Energy Req.	+ 79.43 Excess Ice	+ 3.46 Excess Ice	+ 21.71 Excess Ice
Seattle	Space HTG	62.13	42.22	42.28
	DHW	9.00	2.00	3.84
	Total HTG	71.13	44.22	46.12
	Total Cool	48.90	70.56	48.00
	Net Energy Req.	+ 38.70 Excess Ice	- 40.16 Ice Deficiency	- 16.29 Ice Deficiency
Oklahoma City	Space HTG	52.31	35.55	35.60
	DHW	9.00	2.00	3.84
	Total HTG	61.31	37.55	39.44
	Total Cool	20.4	136.80	93.00
	Net Energy Req.	+ 21.75 Excess Ice	- 110.98 Ice Deficiency	- 65.88 Ice Deficiency

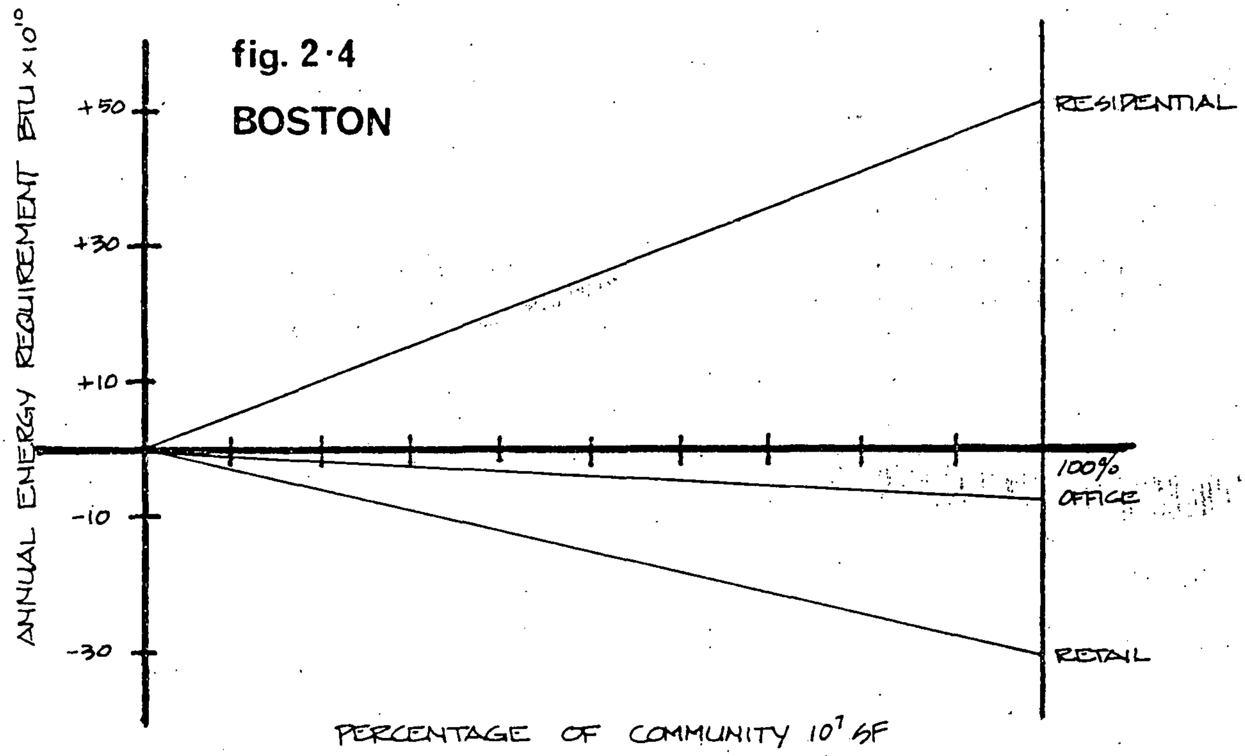
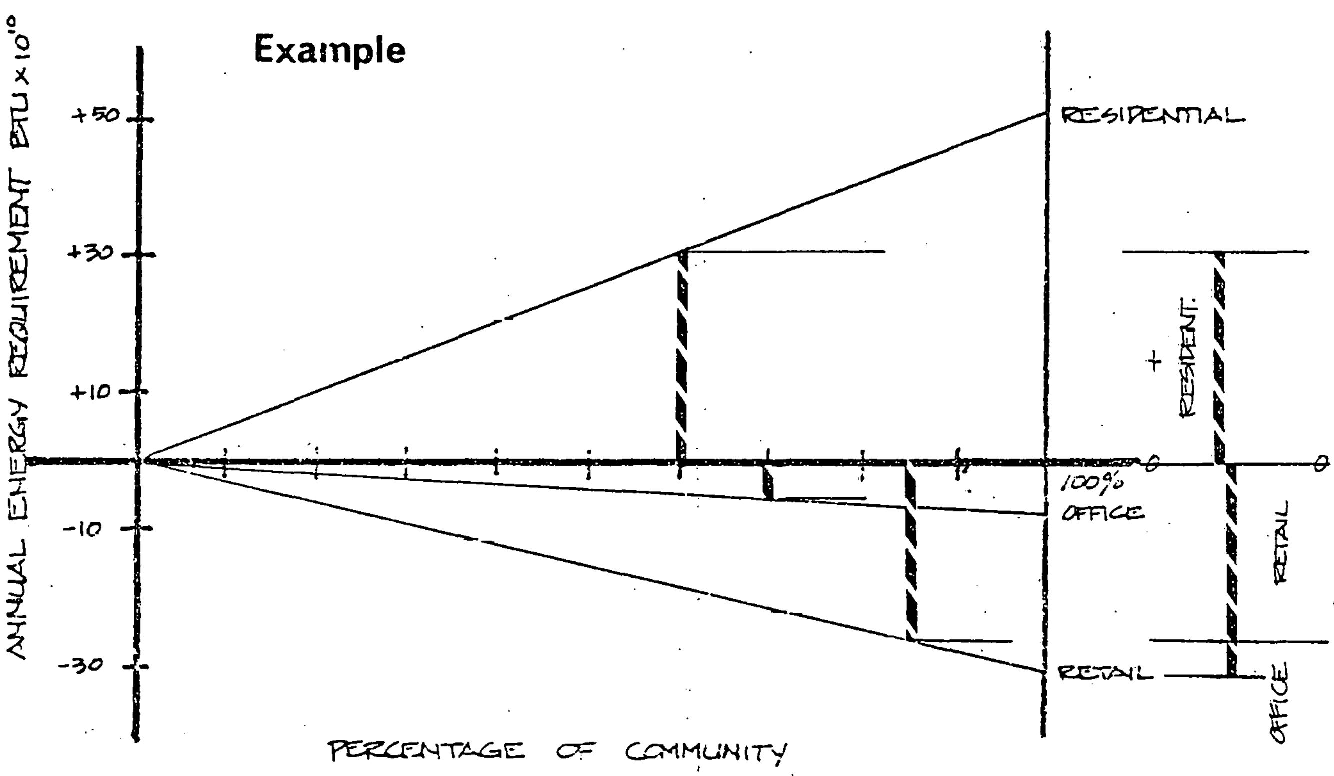


fig. 2·4
BOSTON
ANNUAL ENERGY REQUIREMENT BTU x 10⁶
+50
+30
+10
-10
-30
RESIDENTIAL
100%
OFFICE
RETAIL
PERCENTAGE OF COMMUNITY 10⁷ SF


Example
ANNUAL ENERGY REQUIREMENT BTU x 10⁶
+50
+30
+10
-10
-30
RESIDENTIAL
100%
OFFICE
RETAIL
0
0
+
RESIDENT.
RETAIL
OFFICE
PERCENTAGE OF COMMUNITY

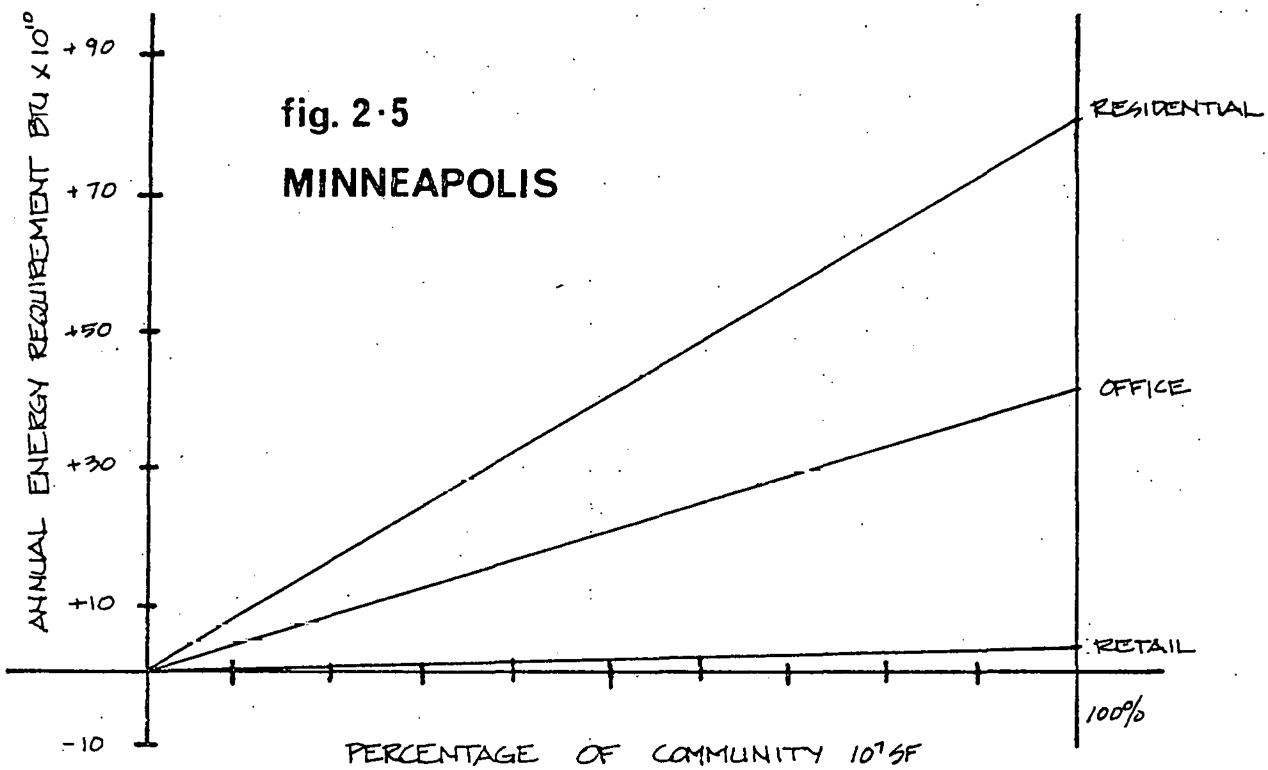


fig. 2·5
MINNEAPOLIS
ANNUAL ENERGY REQUIREMENT BTU x 10^10
+90
+70
+50
+30
+10
-10
RESIDENTIAL
OFFICE
RETAIL
100%
PERCENTAGE OF COMMUNITY 10^7 SF

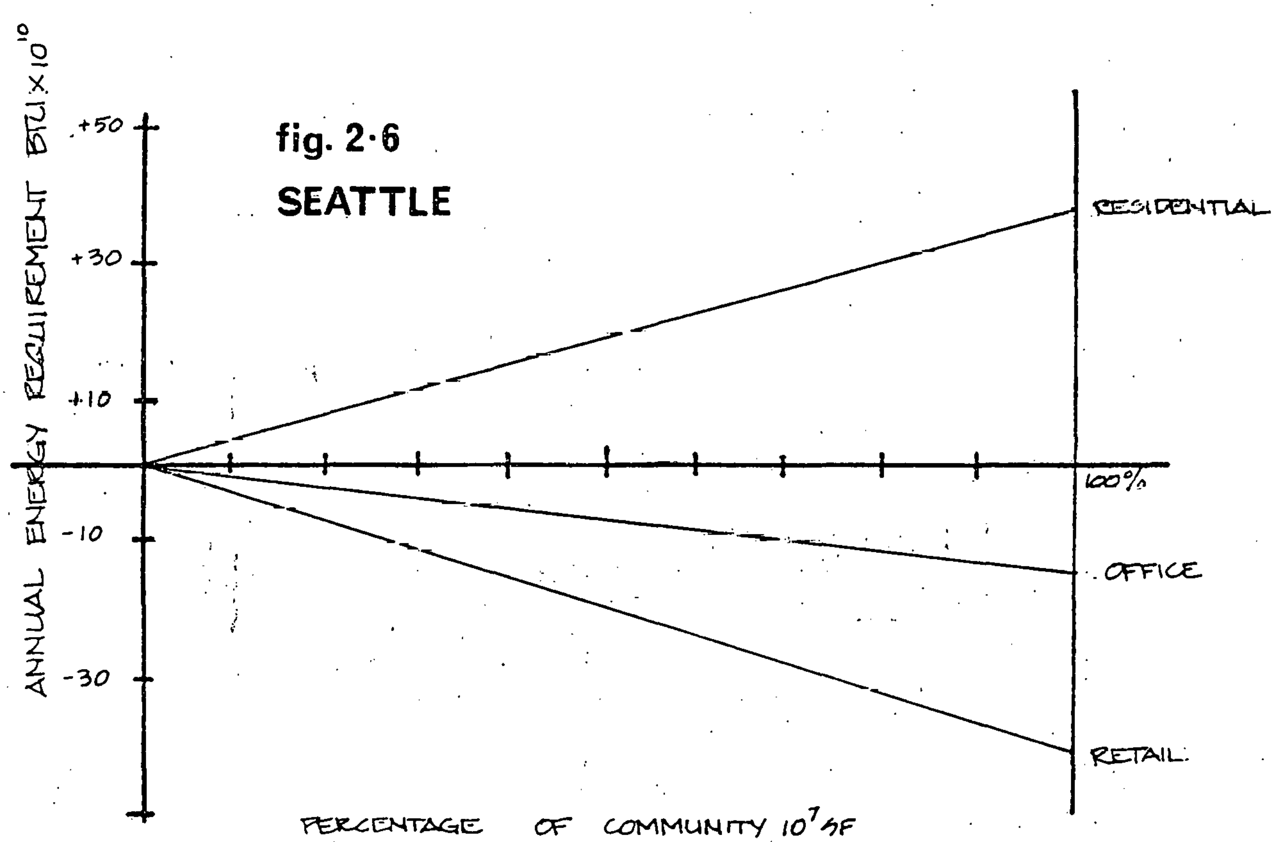
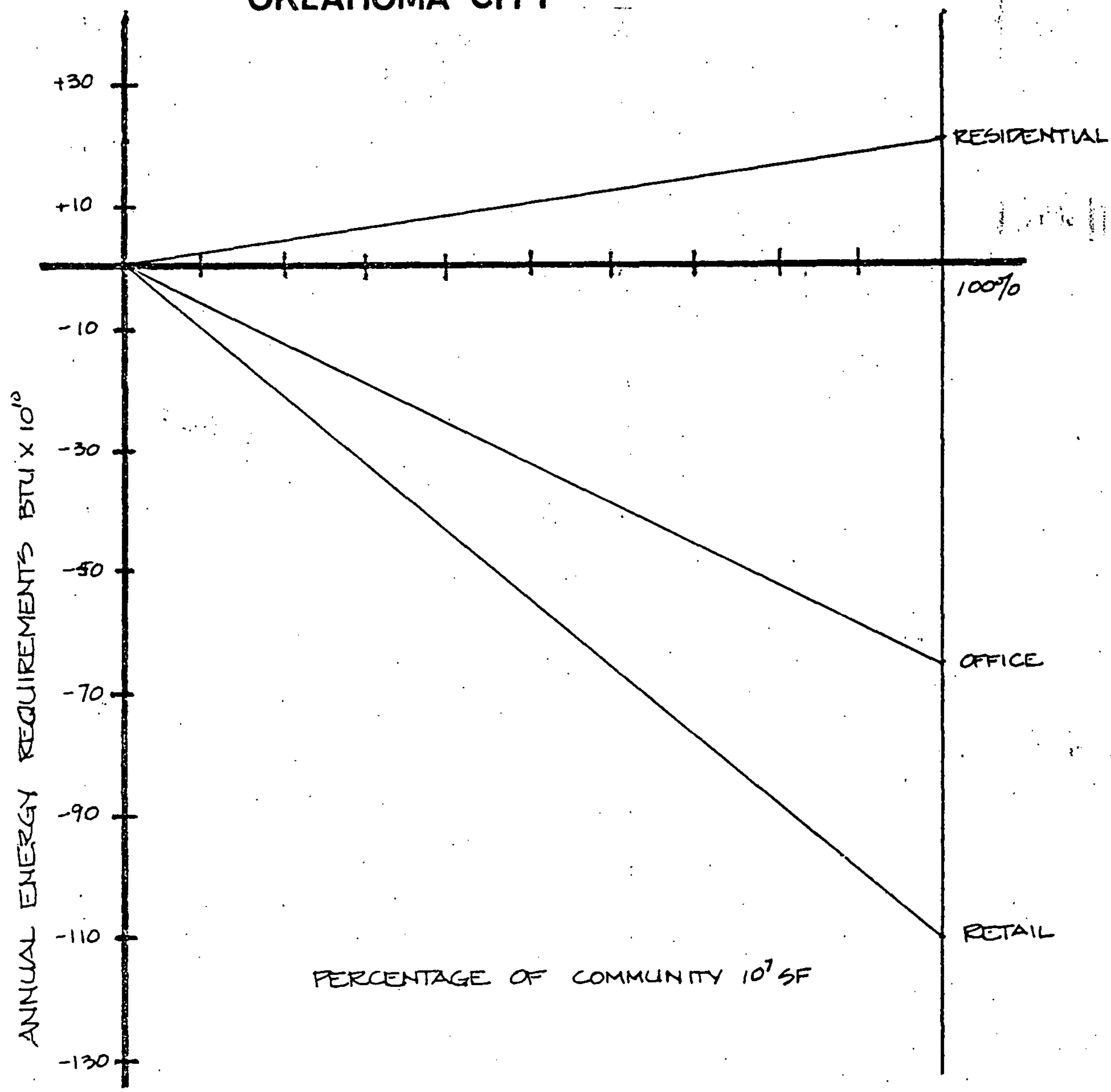


fig. 2·6
SEATTLE
ANNUAL ENERGY REQUIREMENT BTU x 10^10
+50
+30
+10
-10
-30
RESIDENTIAL
100%
OFFICE
RETAIL
PERCENTAGE OF COMMUNITY 10^7 SF

fig. 2·7

OKLAHOMA CITY



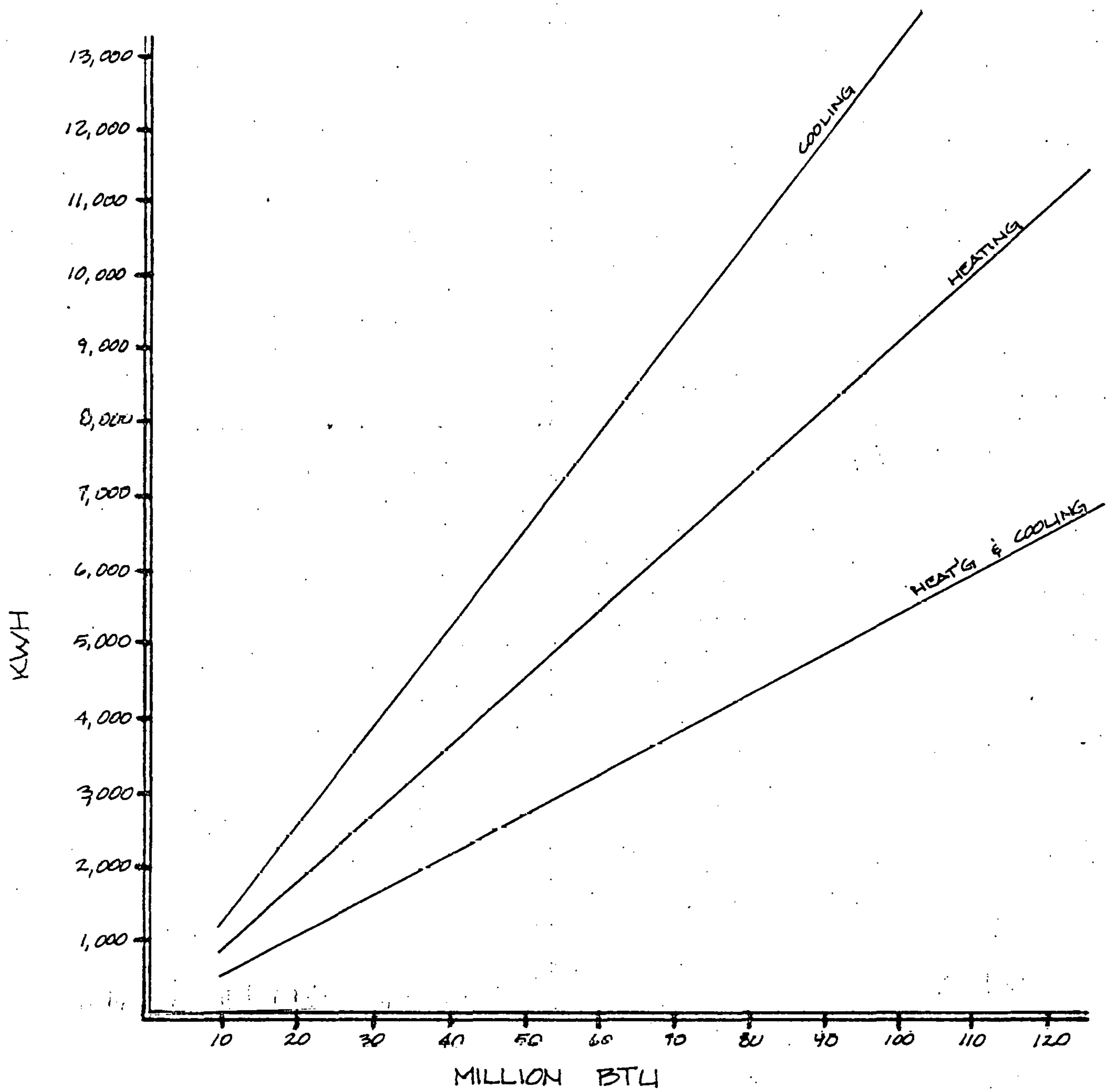


fig 2·8 Electrical Energy Input Required To Obtain:

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

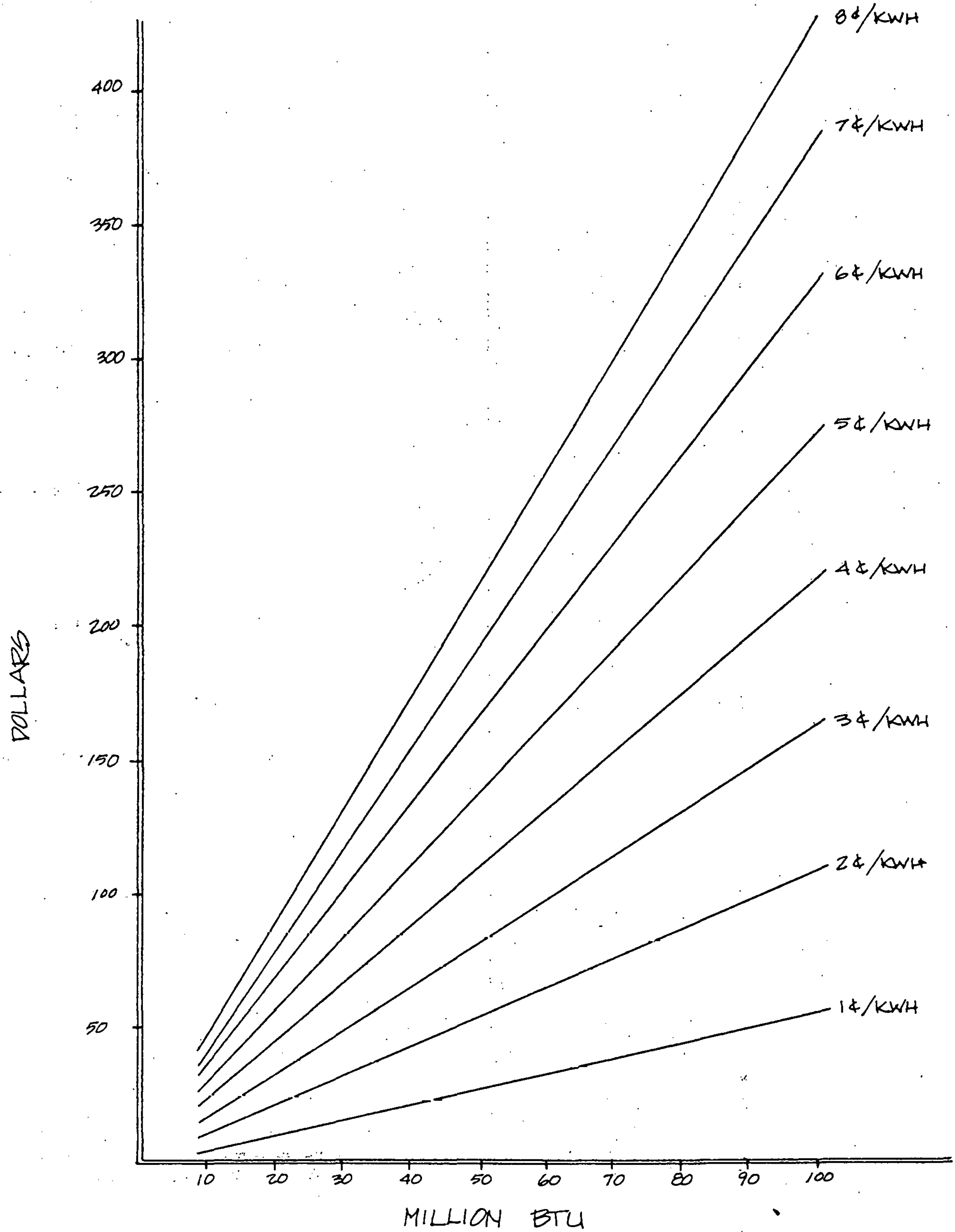


fig 2·9 Energy Cost for Heating and Cooling as a Function of Electricity Cost

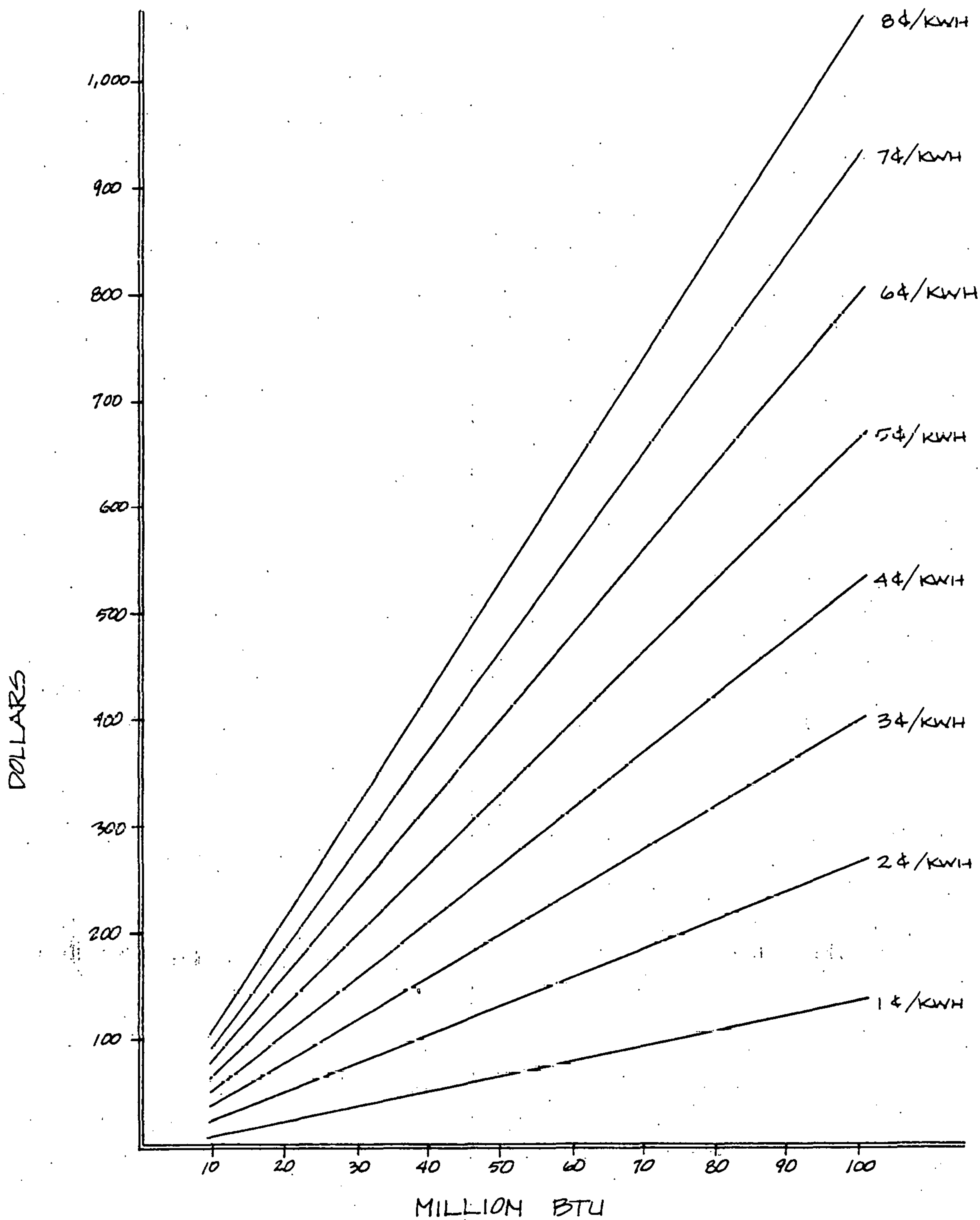


fig 2·10 **Energy Cost for Cooling as a Function of Electricity Cost**

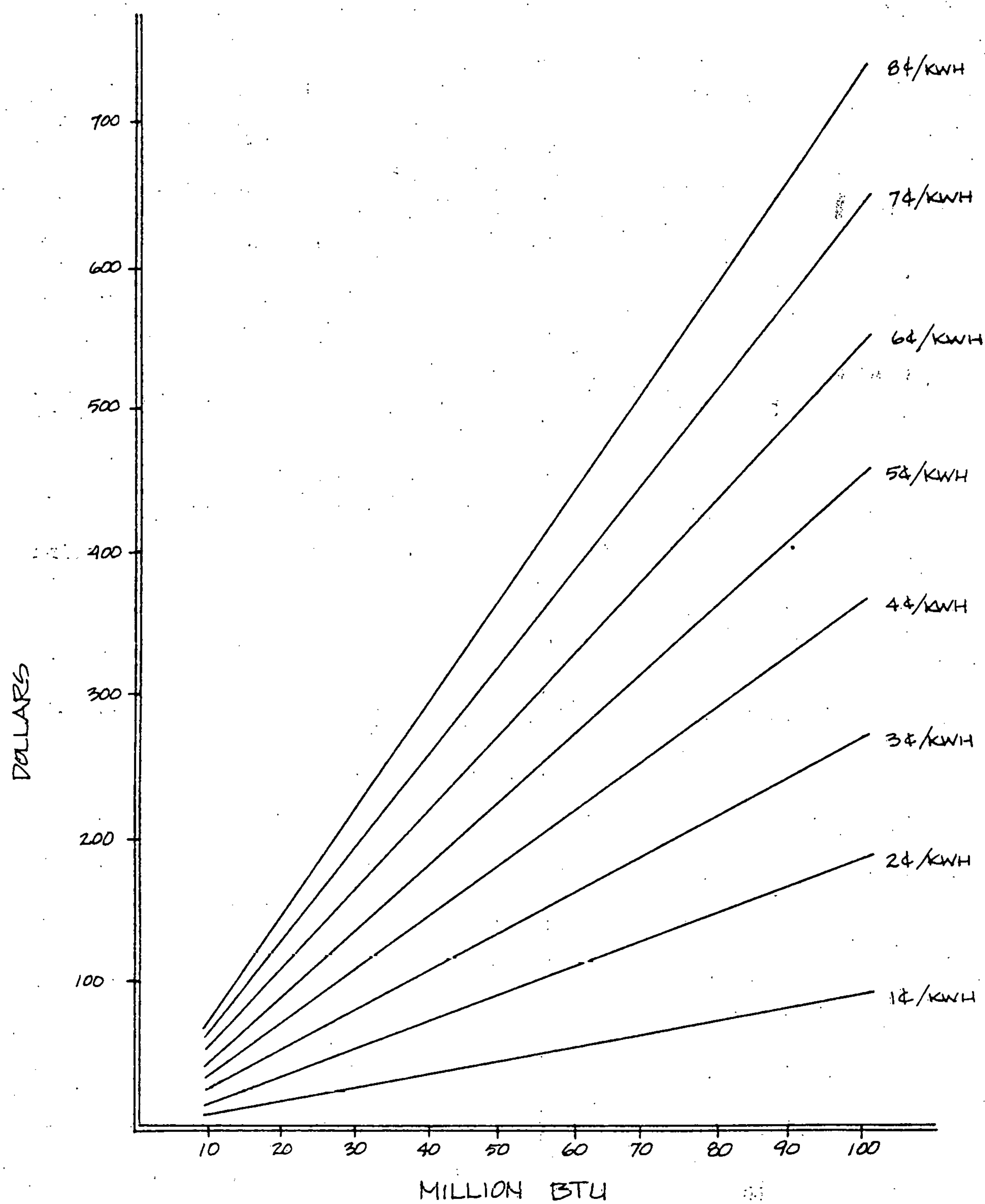


fig 2·11 **Energy Cost for Heating as a Function of Electricity Cost**

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 the 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 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 requirements 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.
Freezing Equipment Sales (FES)		-	55 ft^3/min to 3337 ft^3/min.
Reco-Refrigeration Equipment Corporation			
Sullair Corporation		-	445 ft^3/min to 1710 ft^3/min
Dunham-Bush		-	278 ft^3/min to 16673 ft^3/min.

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 - BUSH							FES - FREEZING EQUIPMENT SALES, INC.						
Model	20^oF SST-130^oF SDT			35^oF SST-105^oF SDT			Model	20^oF SST-130^oF DST			35^oF SST-105^oF 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. This will require an examination in a scaled-down model with the gathered information being logged for engineering evaluation.

A scaled-down 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 institutions 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 applicationsface 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, Argonne National Laboratory Report ANL/ICES-TM-11, prepared by Energy Systems Research Group, Inc., 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 over come 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 soluble 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 rest of the 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 needs 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 x 10^{12} Btu. The corresponding energy for cooling is estimated at 327 x 10^{12} Btu. From this amount of cooling the ice generating HP-ICES will produce 476 x 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 x 10^6 barrels of oil and 4,012 x 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 x 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 x 10^{12} Btu and 804 x 10^{12} Btu respectively.

The heating obtained from the cooling by the heat pump amounts to 1169 x 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 x 10^6 barrels of oil and 2148 x 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.

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

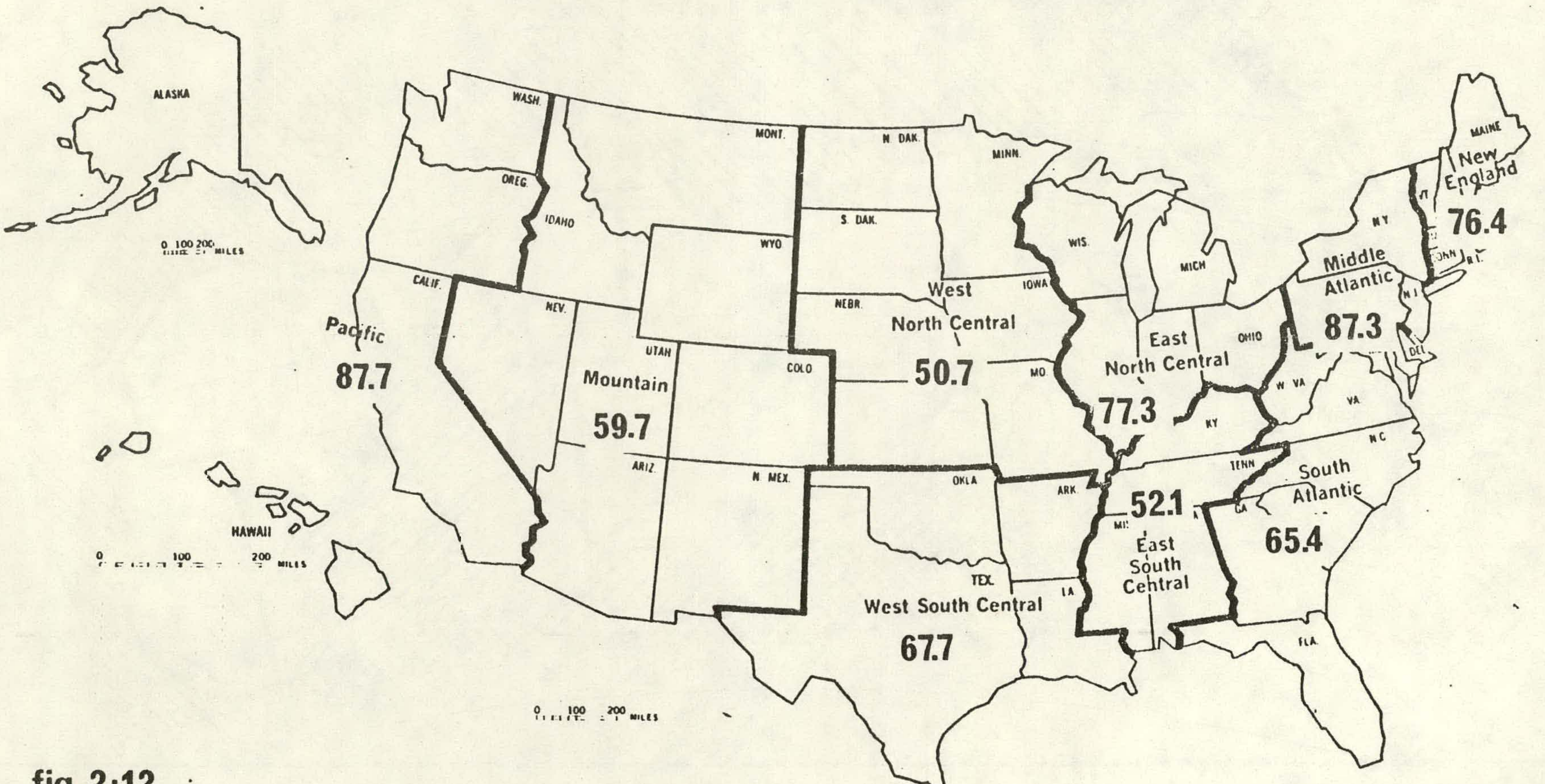


fig. 2·12
Percent of Population in Metropolitan Areas

U. S. DEPARTMENT OF COMMERCE
BUREAU OF THE CENSUS

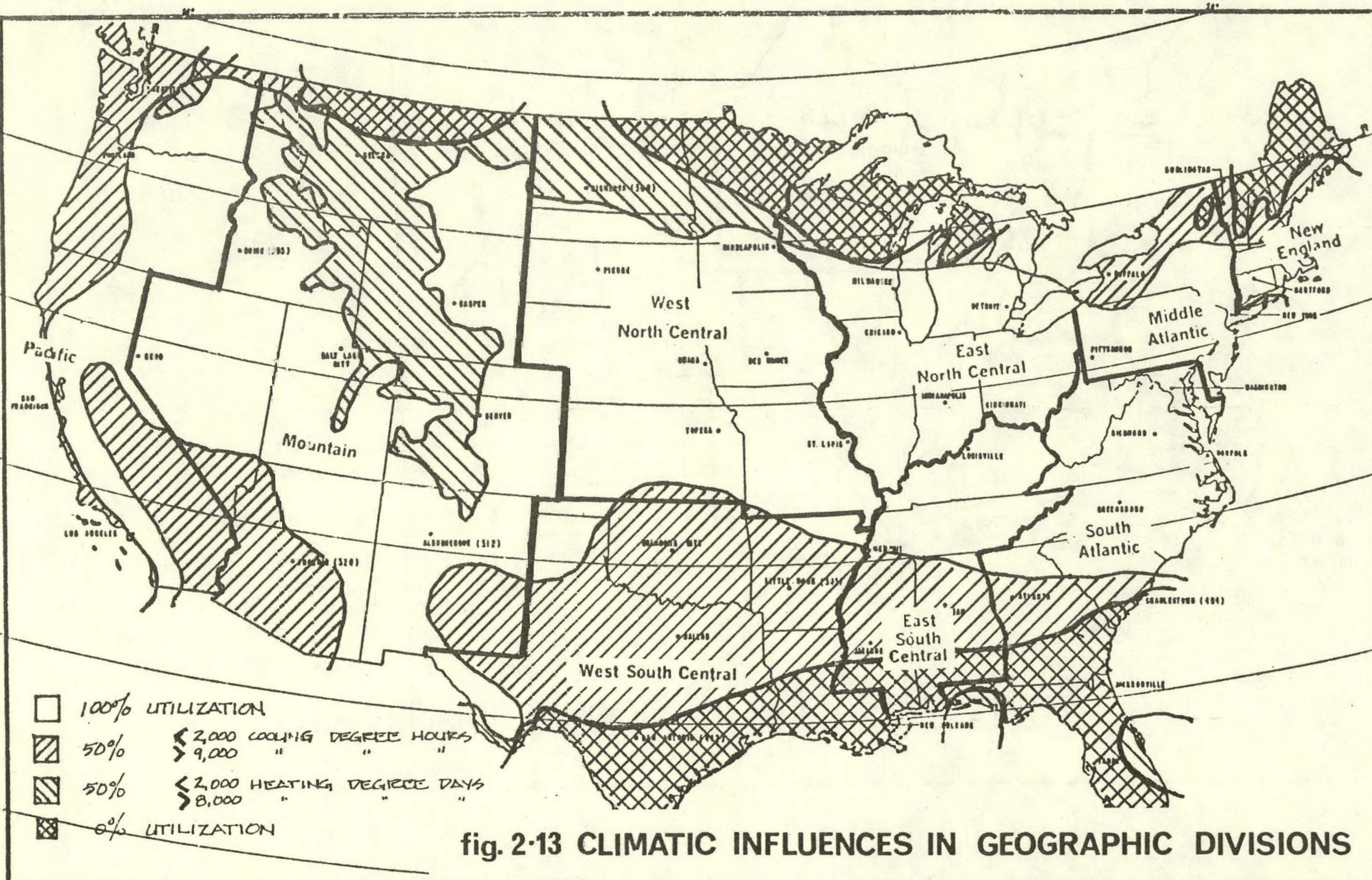


fig. 2·13 CLIMATIC INFLUENCES IN GEOGRAPHIC DIVISIONS

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 component 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

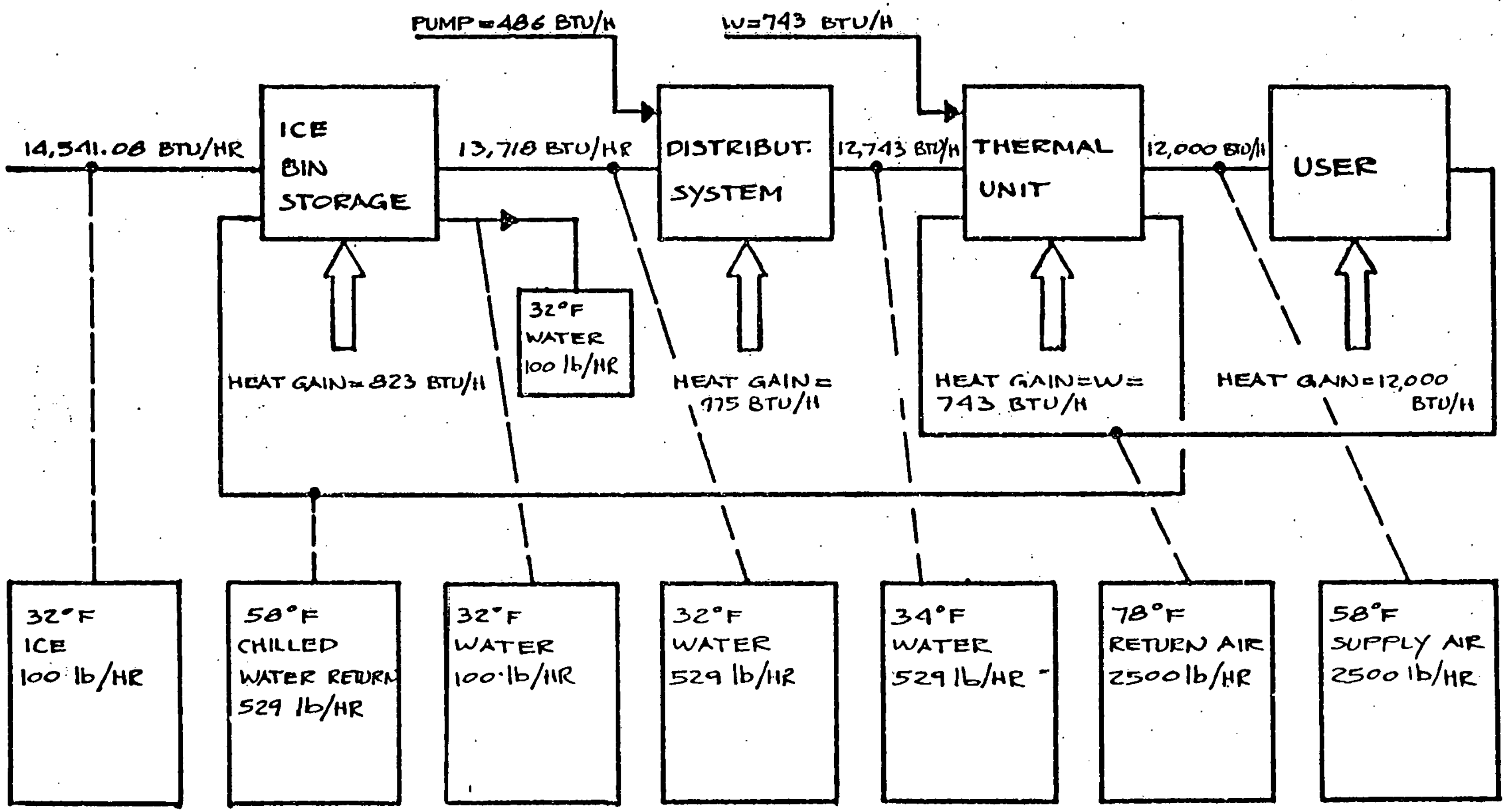


FIG.- 3-2 FIRST LAW ENERGY BALANCE ICE GENERATION HP-ICES
WINTER MODE - BASED ON 12000 BTU/HR WATER HEAT OF FUSION

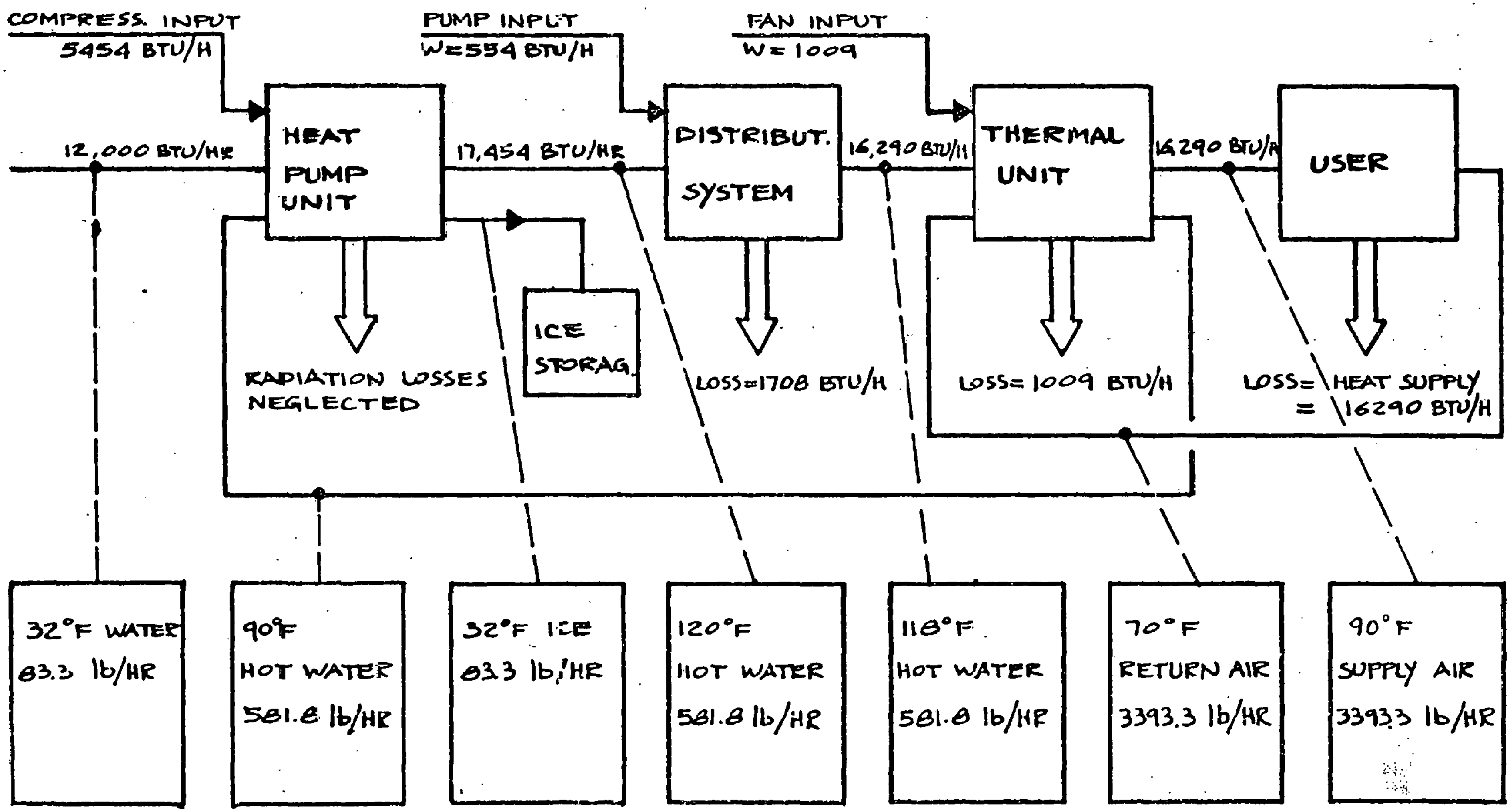


FIG.-3-3 FIRST LAW ENERGY BALANCE FOR CONVENTIONAL BOILER SYSTEM WINTER MODE

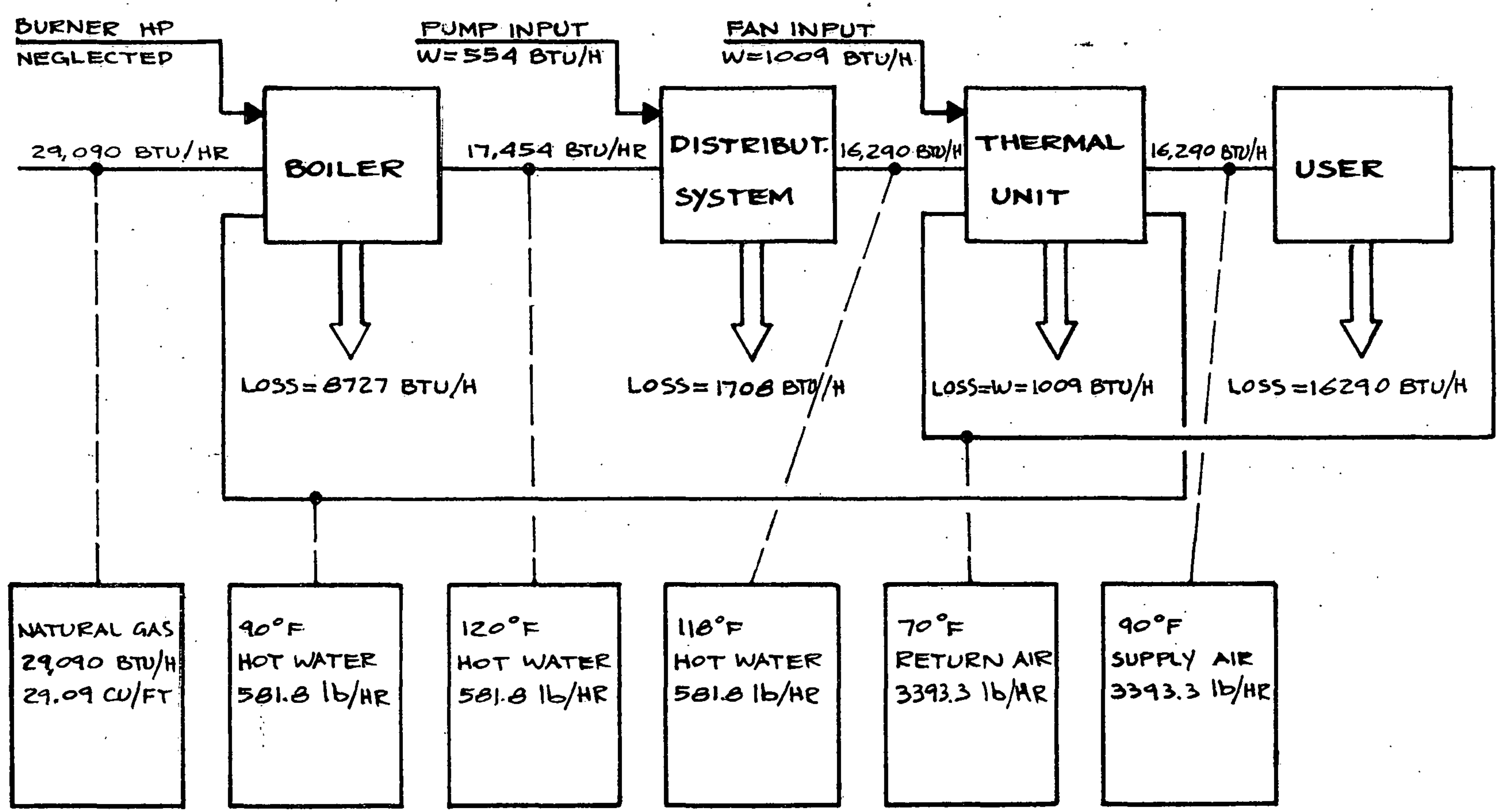
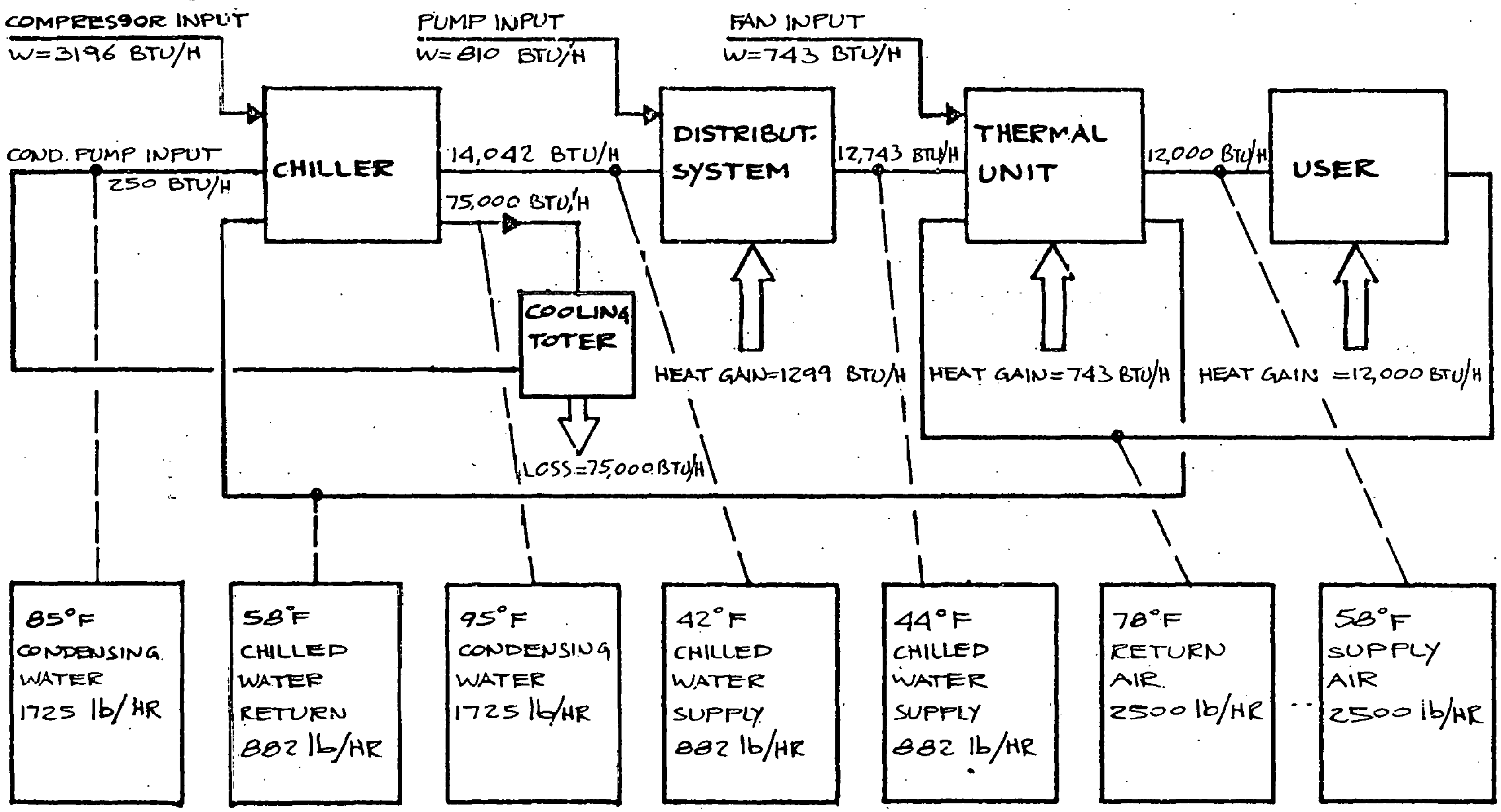


FIG.-3-4 FIRST LAW ENERGY BALANCE FOR CONVENTIONAL CHILLER SUMMER MODE



	Conventional	HP-ICES
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

HP-ICES	=	1.063
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:

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

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.

$\emptyset$ is always measured with respect to a rest state that in our usage will be $70^{o}F$ and one atmosphere.

T_a is therefore $530^{o}R$

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

$$\Delta \emptyset_{1-2} = \emptyset_2 - \emptyset_1$$

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

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

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 - $20^{o}F$, SDT - $130^{o}F$

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

Availability Calculations

$$\Delta \emptyset = -\left[\Delta H - T\Delta S\right]$$

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.

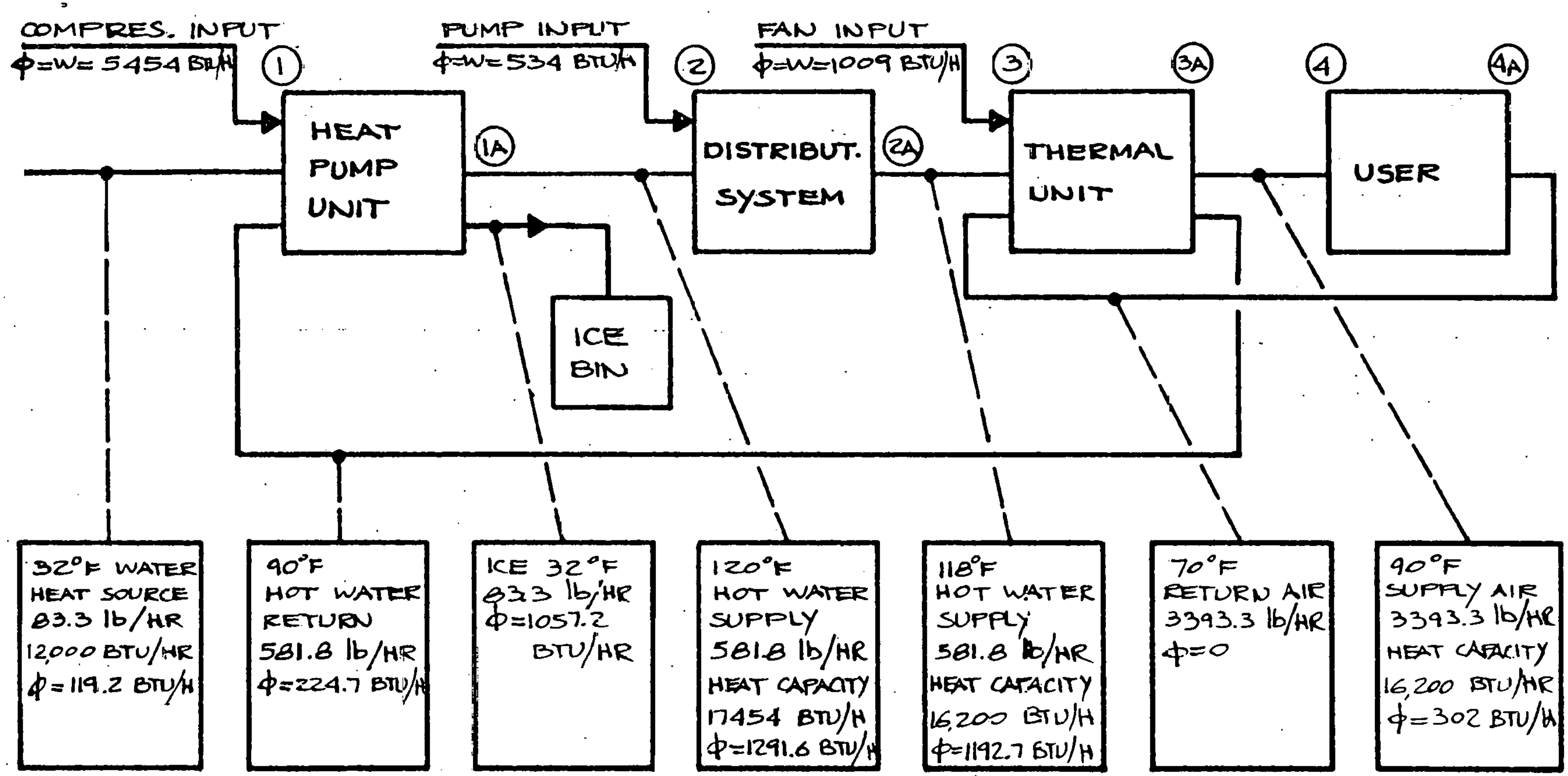
$$\Delta \emptyset = -\left[1 \times (T_a - T_i) - T_a \ln \frac{T_a}{T_i}\right] \times \dot{m}$$

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



AVAILABILTY LOSSES:

1 → 1A $\Delta\phi = \Sigma\phi_{1A} - \Sigma\phi_1 = (-1057.24+1291.6)-(5454+224.7+119.2) = -5563.54$ BTU/HR

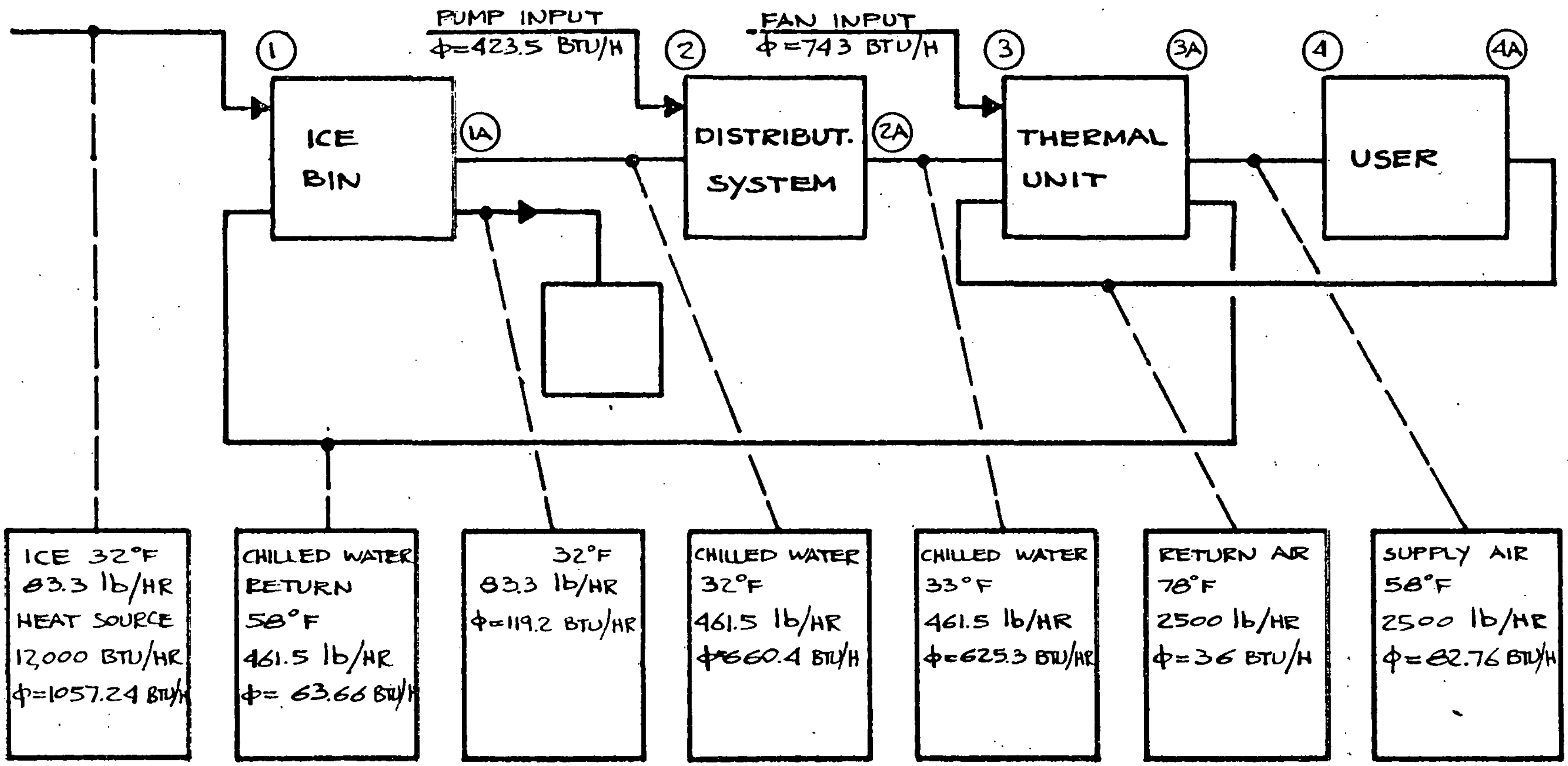
2 → 2A $\Delta\phi = \Sigma\phi_{2A} - \Sigma\phi_2 = (1192.7)-(534+1291.6) = -632.9$ BTU/HR

3 → 3A $\Delta\phi = \Sigma\phi_{3A} - \Sigma\phi_3 = (302+224.7)-(1009+1192.7+0) = -1675$ BTU/HR

4 → 4A $\Delta\phi = \Sigma\phi_{4A} - \Sigma\phi_4 = (0)-302 = -302$ BTU/HR

FIG.-3-6 ϕ = AVAILABILITY ANALYSIS ICE GENERATION HP-ICES

SUMMER MODE



AVAILABILITY LOSSES

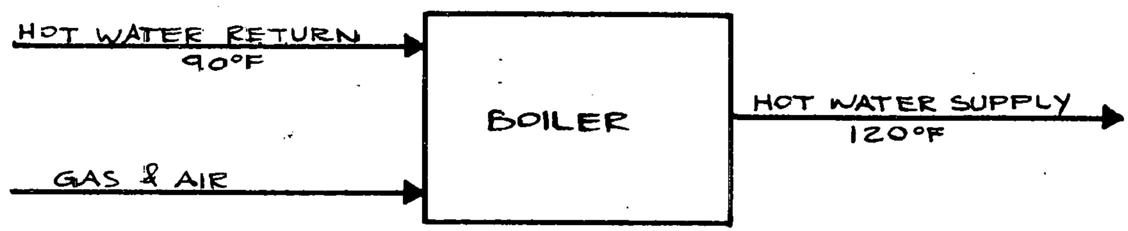
1 → 1A $\Delta\phi = \phi_{1A} - \phi_1 = (660.4 + 119.2) - (-1057.24 + 63.6) = 1773.24$ BTU/HR

2 → 2A $\Delta\phi = \phi_{2A} - \phi_2 = (625.3) - (423.5 + 660.4) = -458.6$ BTU/HR

3 → 3A $\Delta\phi = \phi_{3A} - \phi_3 = (63.66 + 82.76) - (743 + 36) = -632.59$ BTU/HR

4 → 4A $\Delta\phi = \phi_{4A} - \phi_4 = (36 - 82.76) = -46.76$ 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

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

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

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

REF. FROM: M. V. SUSSMAN AVAILABILITY ANALYSIS

Page number 61

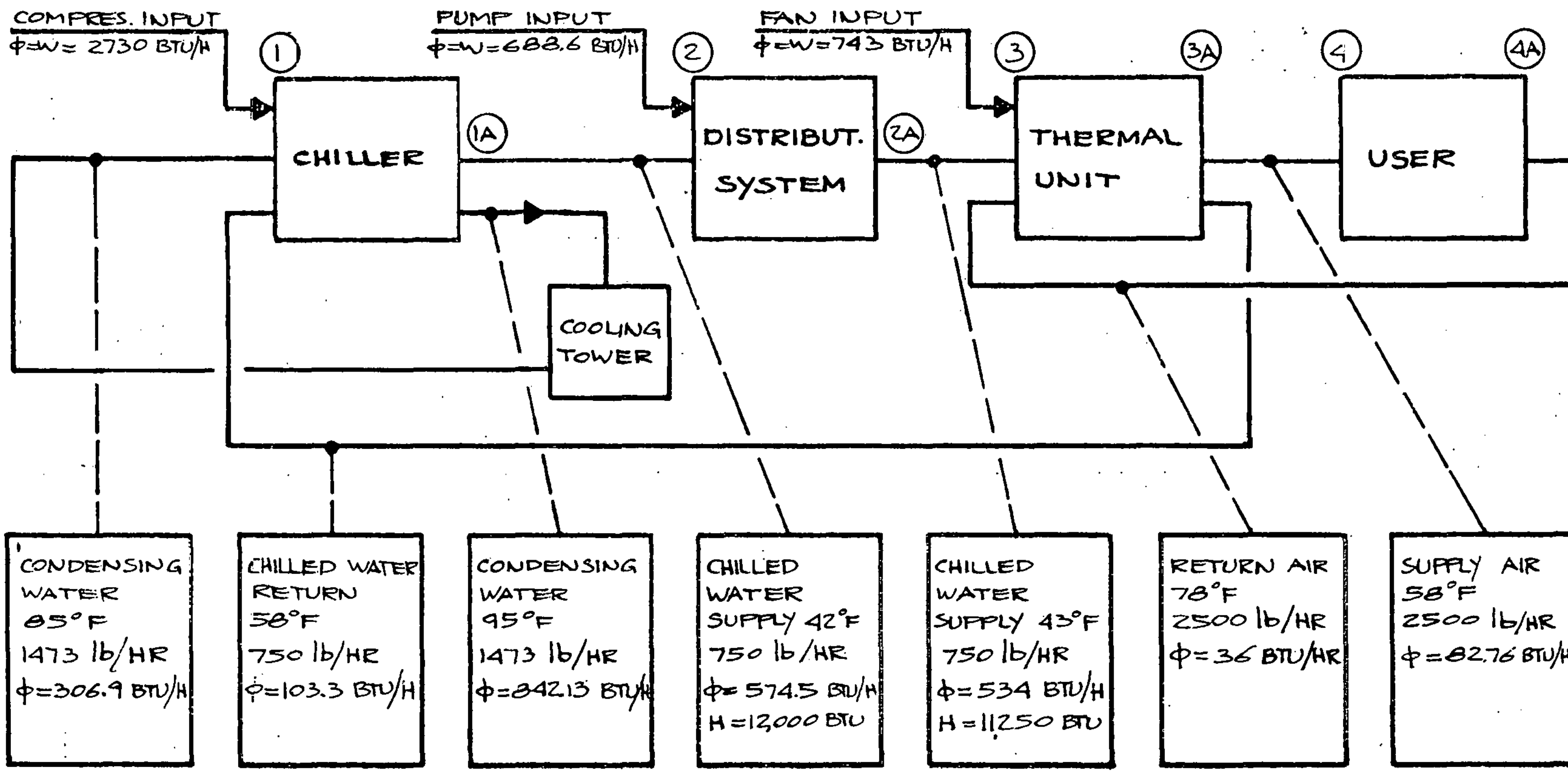
We take the availability of fuel air mixture at inlet to burner as

$= 189{,}951$ cal./gm.-mole CH_4

which is equivalent to 21,388.33 Btu/lb or 953.77 Btu/ft^3

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

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



AVAILABILITY LOSSES:

$1 \rightarrow 1A = \phi_{1A} - \phi_1 = (574.5+842.13)-(2730+1033+305.9) = -1723.57$ BTU/HR

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

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

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

$$H = C_p \Delta T$$

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

For ice

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

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:

	Conventional	HP-ICES
Winter Losses	Ø = -26,884.82 Btu/hr	Ø = -5563.51 Btu/hr
Summer Losses	Ø = - 1,723.52 Btu/hr	Ø = +1773.24 Btu/hr
Annual Losses	Ø = -28,608.34 Btu/hr	Ø = -3790.27 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 13.063×10^6 Btu/hr. One

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

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

b. <u>Energy Analysis</u>

Energy Requirements:

Heating and DHW $27{,}026 \times 10^6 + 9{,}211 \times 10^6$

$= 3.6237 \times 10^{10}$ Btu year

Cooling = 3,045,000 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 Ton-hrs x 1KW/*Ton = 3,045,000 kWh (the cost of operation of chilled water pumps is not considered since their use applies to all the schemes).

Thus, annual energy input

Heating	=	370,000 gallons of No. 2 oil
Cooling	=	3,045,000 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. <u>Energy Analysis</u>

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:

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

b. <u>Summer Energy Requirements</u>

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. <u>"Cooling" Energy Available From Heat Pump</u>

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:

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

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. <u>Refrigeration Energy Generated in the Summer from DHW</u>

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

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

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

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

*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 $57^{o}F$ to $32^{o}F$. 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

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

This yields $\frac{2.692 \times 10^6}{1.896 \times 10^6} = 1.42$ cubic feet of ice storage per ton-hour. Comparing with chilled water storage with a 15^{o} temperature difference

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

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 $15^{o}F$ 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:

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

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 10^8

	Top	Sides	Bottom	Total
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
				19.741

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 ton hours

Deficiency in refrigeration energy

3,045,000 - 1,924,000 = 1,121,000 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 kWh

Subtotal for summer = 1,364,500 kWh

Total for year = 4,394,500 kWh

3.2.3 Recapitulation of Results

Table 3.2 Energy Input

	WINTER		SUMMER	
	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

Winter	370,000 gallons x 140,000 Btu/gal.	=	$51,800 \times 10^6$ Btu
Summer	3,045,000 kWh x 11,000 Btu/kWh	=	$33,495 \times 10^6$ Btu
	Total	=	$85,295 \times 10^6$ Btu

HP-ICES

Winter	3,030,000 x 11,000	=	$33,330 \times 10^6$ Btu
Summer	1,364,500 x 11,000	=	$15,010 \times 10^6$ Btu
	Total	=	$48,340 \times 10^6$ Btu

Thus the HP-ICES uses uses up 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:

$$\text{Annual COP} = \frac{\underset{\text{(heating)}}{\text{heat of rejection}} + \underset{\text{(cooling saved)}}{\text{heat of absorption}}}{\text{electric input in Btu}}$$

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

$$2.703 \times 10^{10} \text{ Btu} + 0.9211 \times 10^{10} \text{ Btu}$$

$$= 3.6241 \times 10^{10} \text{ Btu per year}$$

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

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

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

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

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

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

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/kWh x 172,500	=	$ 3,725
Demand charge 20 KW x $3.15/KW x 7 mo.	=	440
Subtotal	=	$ 4,165

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

= $170,665

b) Summer

Cooling

Energy charge $0.02261/kWh x 3,045,000 kWh	=	$68,847
Demand charge 3200 KW x $5.25/KW x 5 mo.	=	$84,000
Subtotal	=	$152,847

Chilled water distribution system 533,000 kWh

Energy charge $0.02261/kWh x 533,000 kWh	=	$ 12,050
Demand charge 165 KW + $5.25/KW x 5 mo.	=	$ 4,330
Subtotal	=	$ 16,380

Domestic hot water 66,500 kWh

Energy charge $0.02261/kWh x 66,500 kWh	=	$ 1,505
Demand charge 20 KW x $5.25/KW x 5 mo.	=	$ 525
Subtotal	=	$ 2,030

Total cost for summer

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

c) Total Winter and Summer = \$170,665 + 171,257

= \$341,922 say \$342,000

- HP-ICES

a) Winter

Heat Pump

Energy 3,030,000 kWh x \$0.02158/kWh = \$65,387

Demand 3 x 875 HP x 0.746KW/HP x \$3.15 x7 mo. = \$43,179

Subtotal = \$108,566

Ice Maker Pump (1 KW) 5,000 kWh x \$0.02158 = \$ 108

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

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

b) Summer

Energy 1,364,500 kWh x \$0.02261/kWh = \$30,851

Demand 3 x 670 HP x 0.746 KW/HP x \$5.25/KW x 1.5 = \$11,808

Subtotal = \$42,659

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

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

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

= \$61,069

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

= \$173,908

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 structure)	$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 <u>Operating Cost</u> (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

<u>HP-ICES</u>

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

	Conventional System	HP-ICES
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

4.2.4 Comparative Analysis

a. Incremental investment of HP-ICES relative to conventional

$$\begin{array}{r} \$3,812,000 \\ -\ \underline{\$2,000,000} \\ \$1,812,000 \end{array}$$

b. Savings in operating cost by using HP-ICES relative to conventional

$$\begin{array}{r} \$\ 600,000 \\ -\ \underline{436,000} \\ \$\ 164,000 \end{array}$$

c. Simple Pay-back

$$= \frac{\$1,812,000}{\$\ 164,000}$$

$= 11$ 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 esculation 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

a. Conventinnal System

Operating cost		\$ 600,000
Amoritization of equipment		
(10%, 20 years CRF = 0.11746)	=	\$ 94,000
Amoritization of distribution system		
(10%, 40 years, CRF = 0.10226)	=	\$ 123,000
Annual owning & operating cost		\$ 817,000

Energy requirements per year

Heating	$27,000\text{x}10^6$ BTU
Cooling	$37,000\text{x}10^6$ BTU
DHW	$9,000\text{x}10^6$ BTU
	$73,000\text{x}10^6$ BTU

The cost per million BTU

$$= \frac{\$\ 817,000}{73,000} = \underline{\underline{\$11.2\ /\ 10^6\ \text{BTU}}}$$

b. HP-ICES

Operating cost	=	$ 436,000
Amoritization of equipment	=	$ 141,000
Amoritization of distribution system and ice bin	=	$ 267.000
Annual owning & operating cost	=	$ 844,000

The cost per million BTU

$$= \frac{\$\ 844{,}000}{73{,}000} = \$11.6\ /\ 10^6\ \text{BTU}$$

Table 4-3 Owning and Operating Cost per Million BTU

Conventional System			HP-ICES		
Year	AO & CC	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%.

4.4 Refined Economic Analysis

Section A-4, Expected Economics for the Market Square project and Section B-4, the expected economics for the Park Plaza project illustrate a different approach to economic analysis which considers also taxes, insurance and different escalation rates for maintenance, labor and energy costs.

This results in different paybacks (6.7 years for the Market Square project and 8 years for the Park Plaza project).

The cost for the users as based on the new analysis averages $4.56/kw demand and 2.9¢/kwH energy for the Market Square project and $6.78/kw demand and 2.6¢/kwH thermal energy for the Park Plaza Project.

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 under-utilized 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

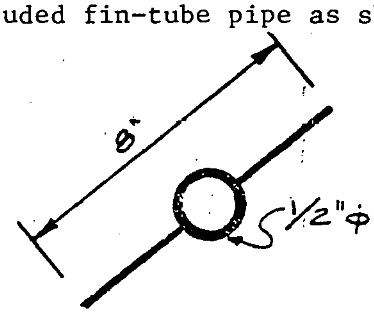
7.1 SYSTEM MODIFICATION

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 ½" 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 requre 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 temperature 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 that related to shaft work is 4.61. The overall COP related to source energy is 2.39 and that 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

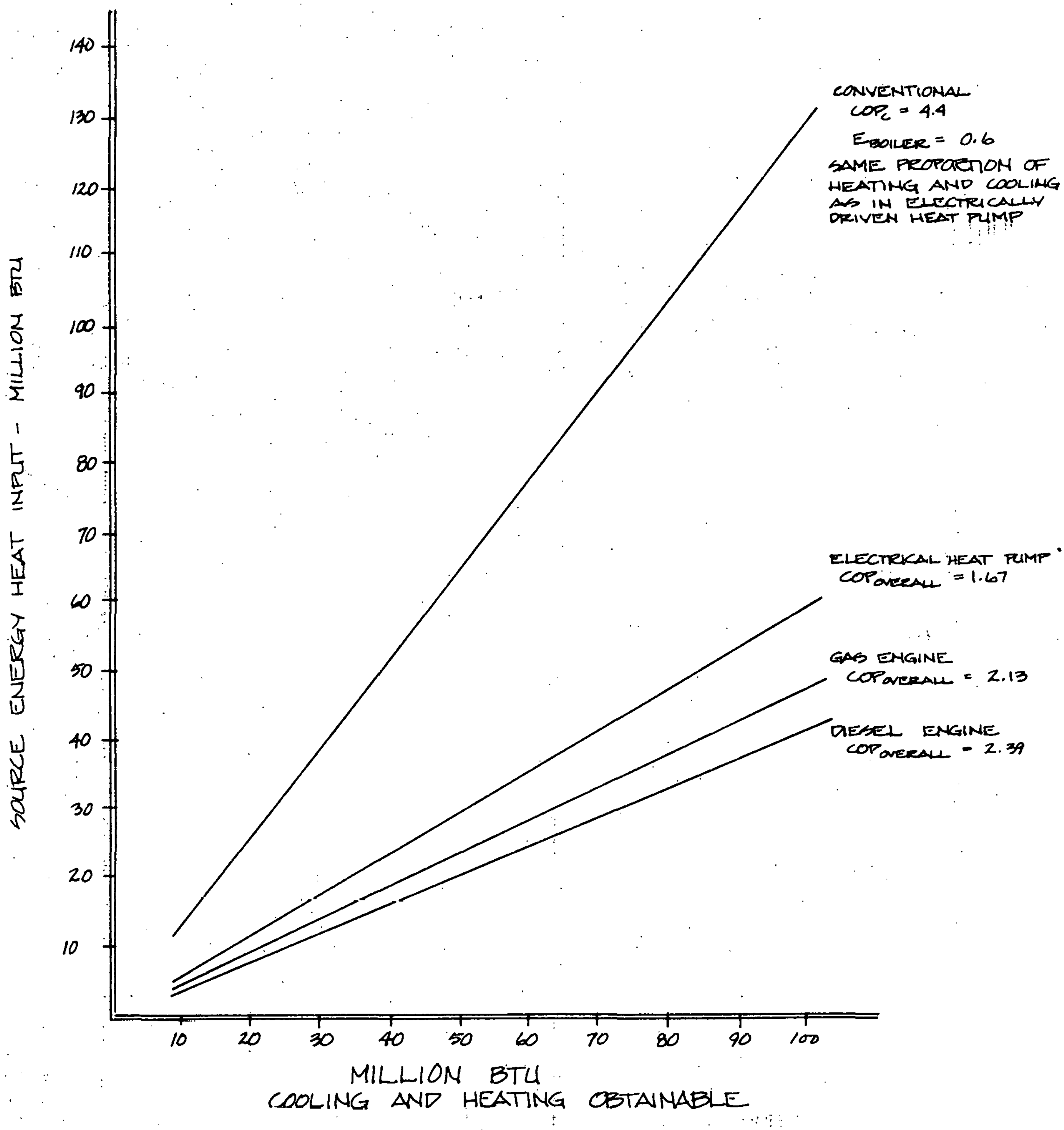


fig 7·1 Heating and Cooling Obtained by Various Systems

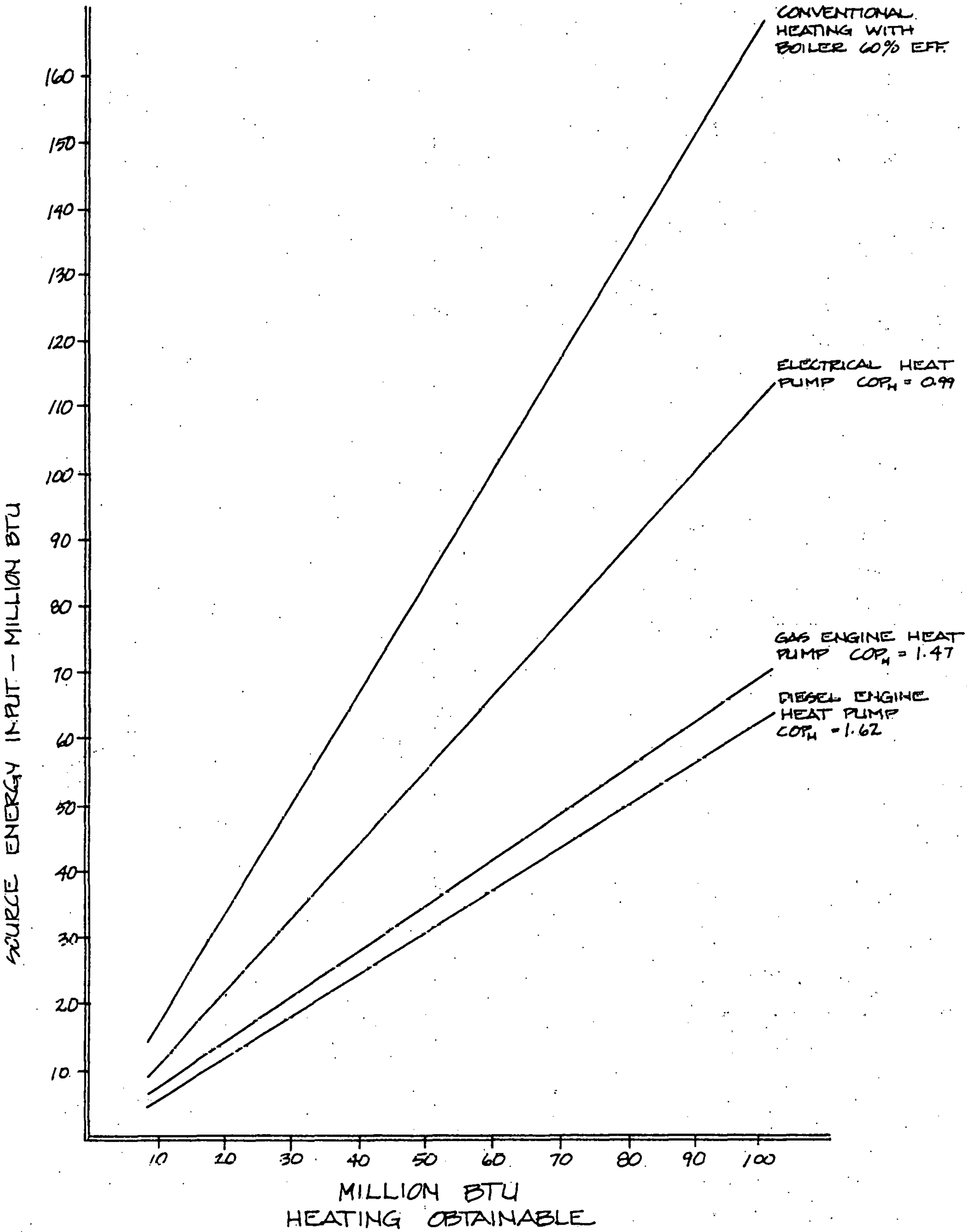


fig 7·2 Heating Obtained by Various Systems the upper part of the hot water temperature difference. Actually, a trial and error approach will be necessary to determine the exact division of heat coming from the heat pump and the heat recovery.

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 homogenically 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 documention. The following standards shall be specifically utilized:

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

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APPENDIX 1A

1A.1 REFRIGERANT INVESTIGATION

Single Stage Compression and Two Stage Compression With Intercooling

Refrigerant - 12

Evaporator Temperature	=	$20^{o}F$
Condenser Temperature	=	$130^{o}F$
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 $20^{o}F$ = 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

Work $= h_3 - h_1 = 92.5$ Btu/lb $- 79.385$ Btu/lb

$= 13.115$ Btu/lb

Refrigerating Effect $= h_1 - h_7 = 79.385 - 38.553$

$= 40.832$ Btu/lb

Refrigerant Flow Per Ton

$$= \frac{200 \text{ Btu/Min}}{40.832 \text{ Btu/lb}}$$

$= 4.898$ lb/min per ton

Coefficient of Performance (COP)

$$= \frac{h_1 - h_7}{h_3 - h_1} = \frac{79.385 - 38.553}{92.5 - 79.385}$$

$$= \frac{40.832}{13.115} = 3.11$$

Alternately

$$\text{COP} = \frac{200 \text{ Btu/Min}}{13.115 \text{ Btu/lb} \times 4.898 \text{ Lb/Min}}$$

$= 3.11$

Energy Efficiency Ratio (EER)

$=$ COP x 3.413 = 10.626 Btuh/Watt

Power $= 13.115$ Btu/lb x 4.898 Lb/Min

$= 64.237$ Btu/Min Per Ton

$$= \frac{64.237 \text{ Btu/Min} \times 778 \text{ Ft-Lb/Btu}}{33{,}000 \text{ Ft-Lb/Min/Hp}} = 1.514 \text{ HP/Ton}$$

Volume Flow $= 4.898$ Lb/Min x 1.0988 Cu. Ft./Lb (at 20°F)

$= 5.382$ CFM/Ton

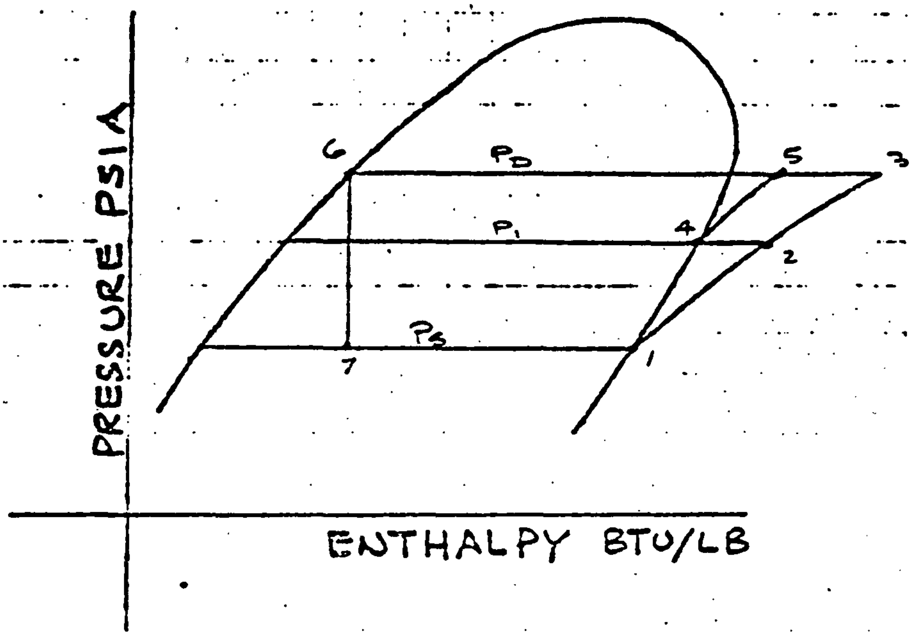


fig 1A.1 Pressure Enthalpy Diagram

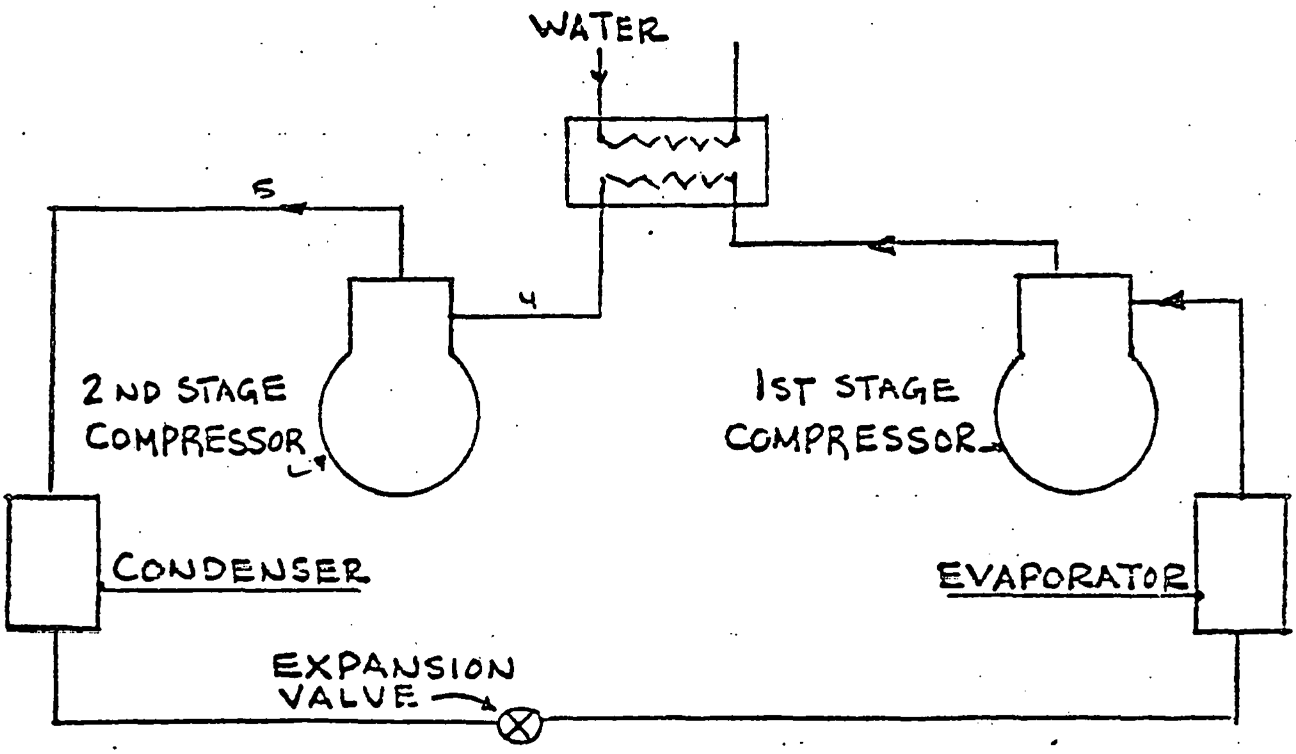


fig 1A.2 Two Stage Compression With Water Intercooling

Two Stage Compression With Water Intercooling

First Stage Compression (Fig. 1A.2)

Work = $h_2 - h_1$ = 85.6 - 79.385 = 6.215 Btu/lb

Second Stage Compression

Work = $h_5 - h_4$ = 91.2 - 84.270 = 6.930 Btu/lb

Total = 13.145 Btu/lb

Refrigerating Effect = 40.832 Btu/lb

Refrigerant Flow Per Ton = 4.898 Lb/Min

COP $= \dfrac{40.832}{13.145} = 3.106$

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

EER = 3.106 x 3.413 = 10.602 Btuh/Watt

Power = 13.145 x 4.898 = 64.384 Btu/Min Per Ton

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

Volume Flow = 5.382 CFM/Ton First Stage

= 2.378 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 x 85.6 = m_4 x 84.270

Yielding m_4 = 1.029 Lb

From Condenser, m_6 = 0.029 Lb of Saturated Liquid

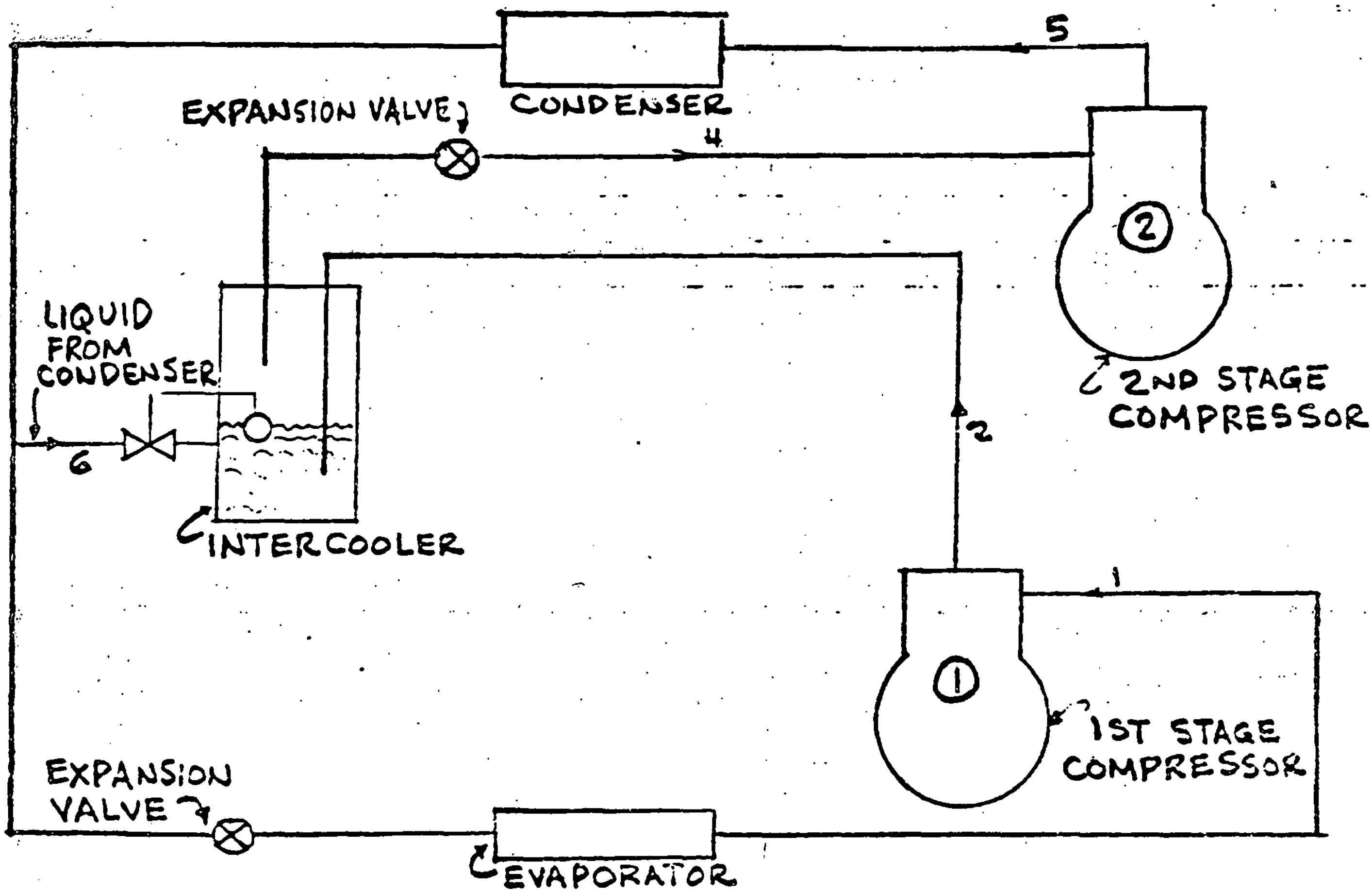


fig 1A.3 Two Stage Compression With Intercooling

With Liquid Refrigerant

First Stage Compression

Work $= h_2 - h_1 = 85.6 - 79.385 = 6.215$ Btu/Lb

Second Stage Compression

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

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

Total $=$ 13.347 Btu/Lb

Refrigerating Effect = 40.832 Btu/Lb

Refrigerant Flow Per Ton = 4.898 Lb/Min (through the evaporator)

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

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

EER = 10.442 Btuh/Watt

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

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

Volume Flow

First Stage = 5.382 CFM/Ton

Second Stage $= 4.898\ (1.029) \times 0.48555 = 2.447$ 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 Refrigerant 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

REFRIGERANT	PRESSURE PSIA		COMPRESSION RATIO	COP		HP/TON		REFRIGERANT FLOW PER TON						DISCHARGE TEMP. °F
								LB/MIN *			CFM **			
								FIRST STAGE	SECOND STAGE		FIRST STAGE	SECOND STAGE		
				INTER-COOLING		INTER-COOLING			INTER-COOLING			INTER-COOLING		
	P_S	P_D		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	195.72	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	231.95	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	1.32	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

HP-ICES PROJECT

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	$T_S = 20°F/T_D = 130°F$						
	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 yeilds 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°F/T_D = 105°F						
	REFRIGERANT CAPACITY		POWER REQUIREMENT			HEAT REJECTED	COOLING
	TONS	BTU/H	HP	HP/TON	BTU/H	BTU/H	C.O.P.
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 REJECTED	COP COOLING	COP HEATING
	TONS	BTUH x 10^6	BHP	HP/TON	BTUH x 10^6	BTUH x 10^6		
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 temperatures, 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	COP	COP
	TONS	BTUH $\times 10^6$	BHP	HP/TON	BTUH $\times 10^6$	REJECTED BTUH $\times 10^6$	COOLING	HEATING
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.

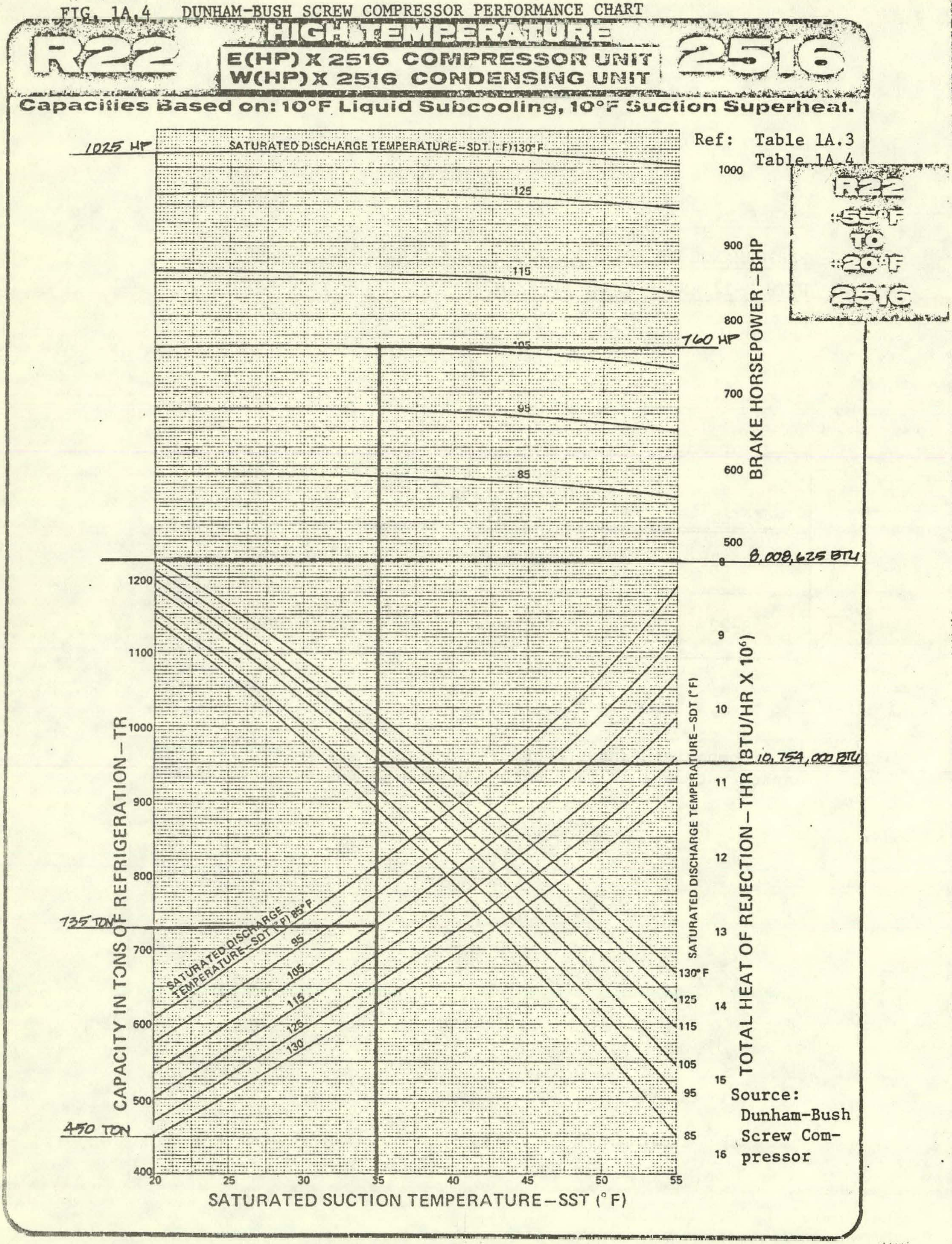
TABLE 1A.7

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



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

Source:
Dunham-Bush
Screw Compressor

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

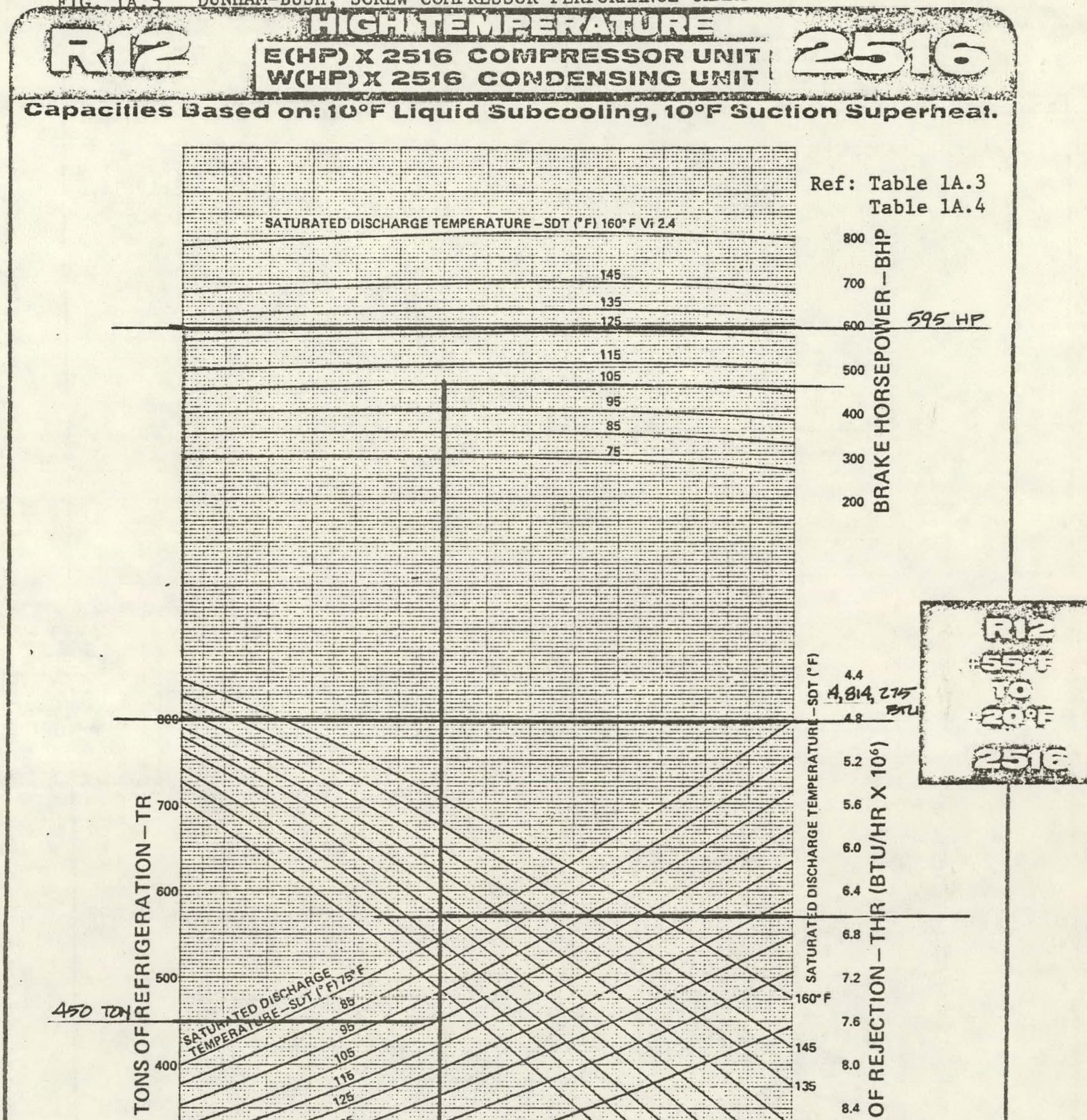


FIG. 1A.6 DUNHAM-BUSH SCREW COMPRESSOR PERFORMANCE CHART

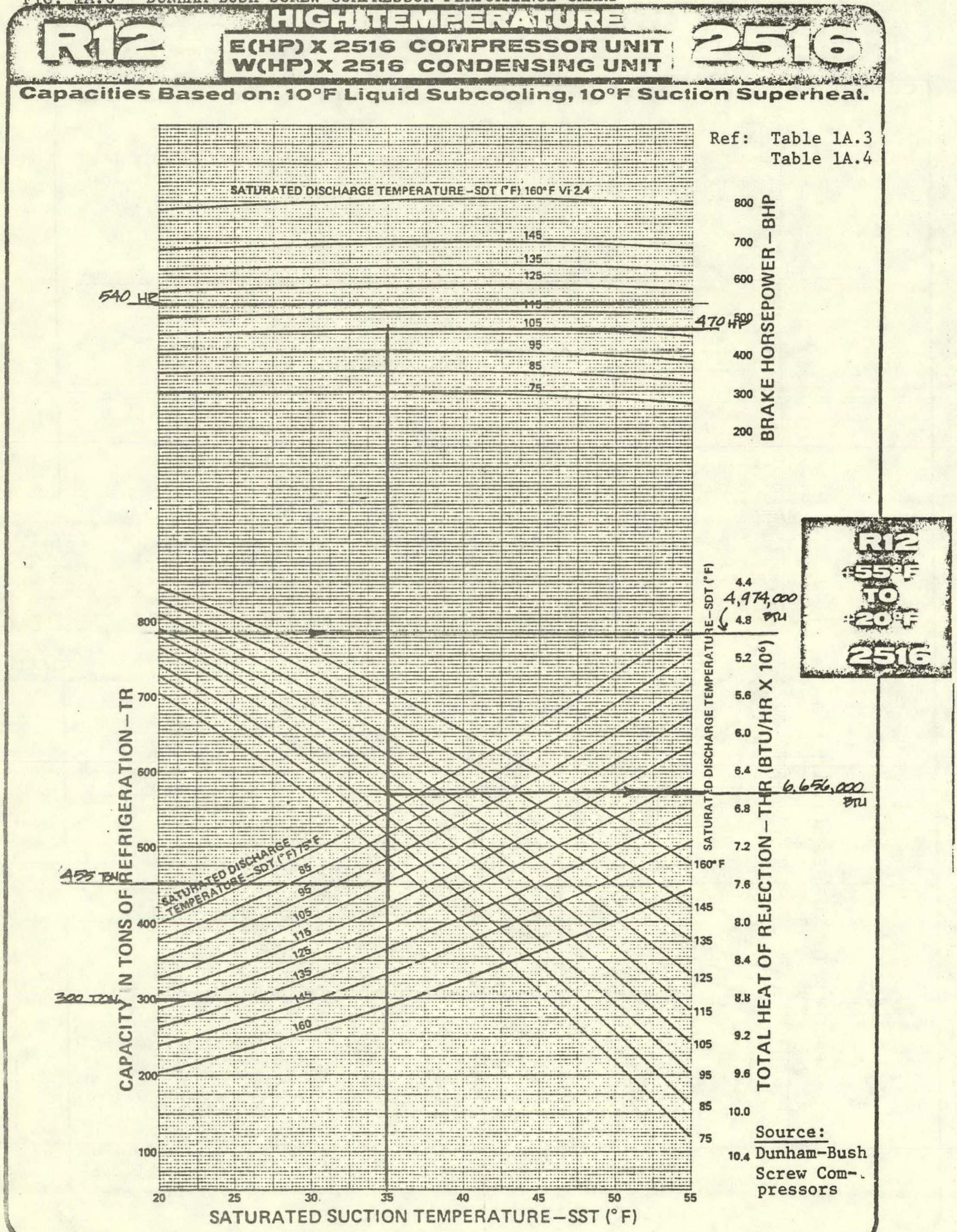
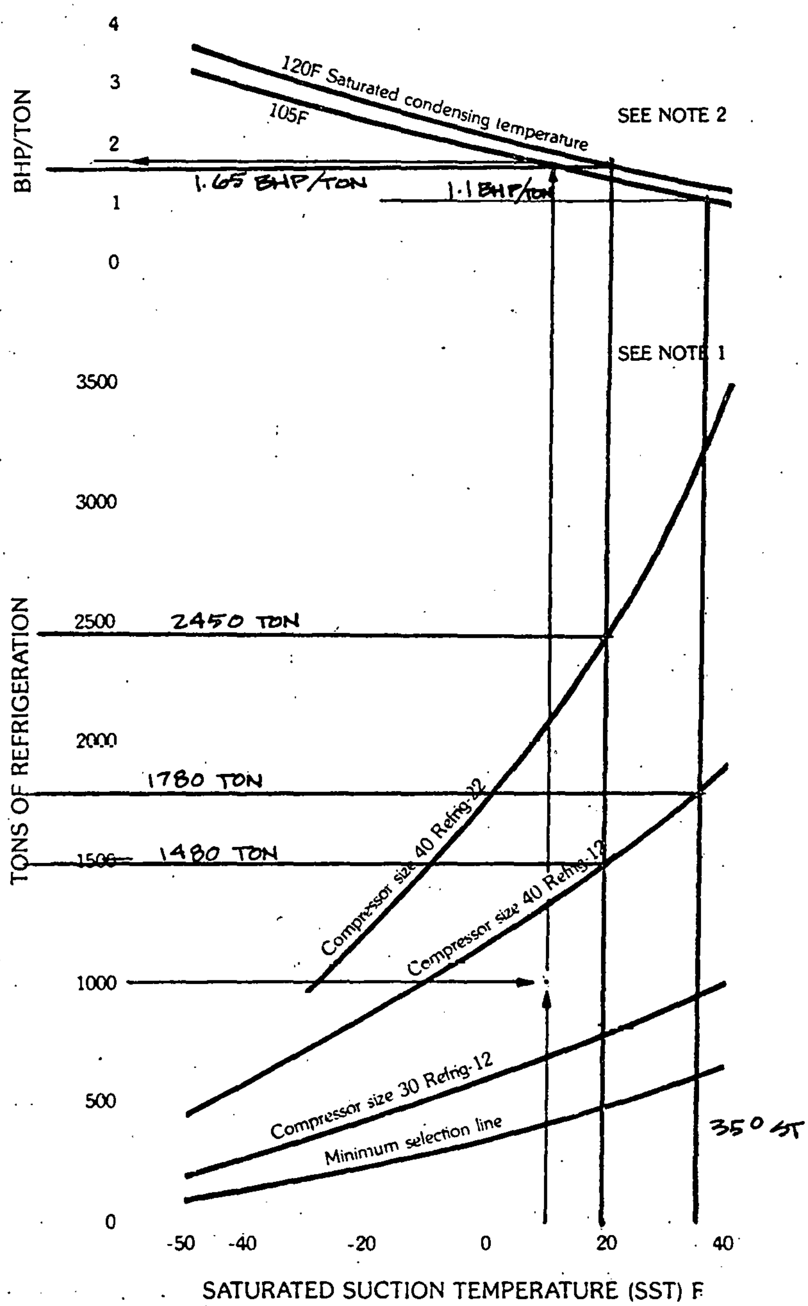


FIG. 1A.7

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.

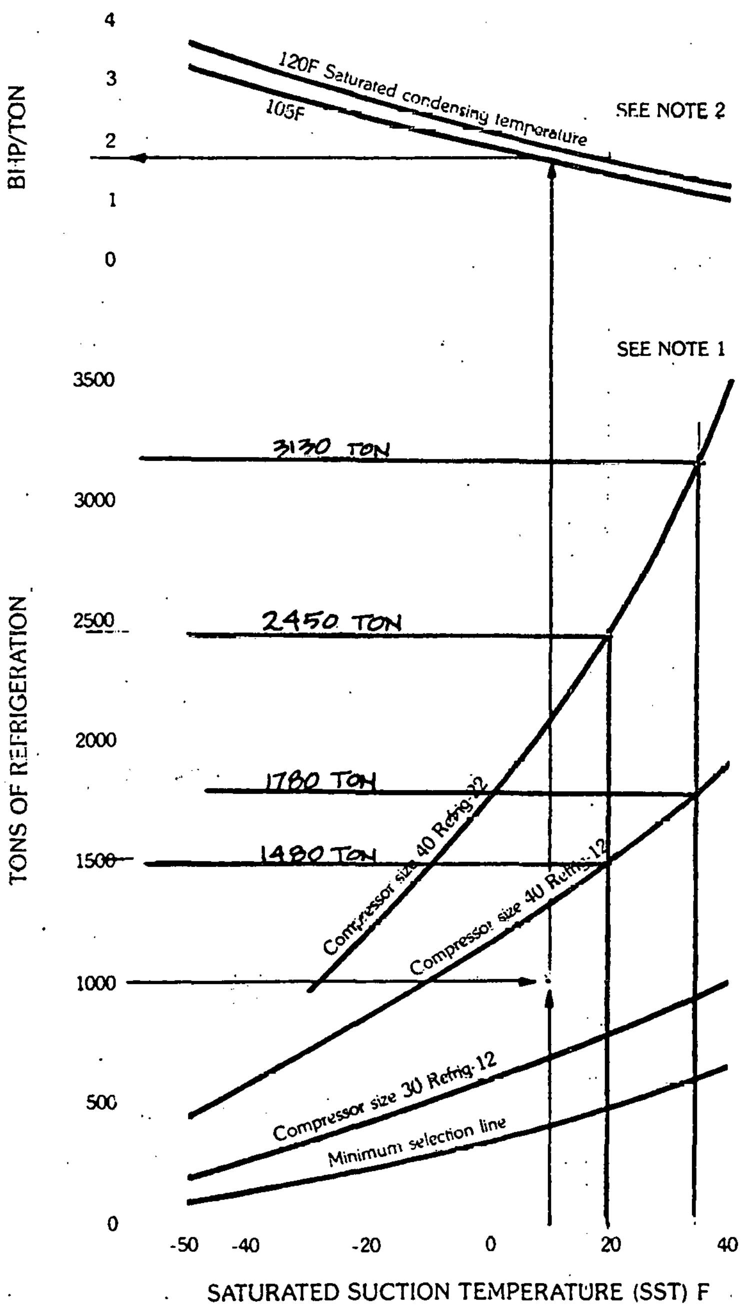
FIG. 1A.8

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, nothern 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:

$$(13{,}789 - 2318) \times 10^6 = 11471 \times 10^6 \text{ BTU}$$

$$= 955{,}917 \text{ ton-hours}$$

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 10^5	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 Supple-mentary 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	33088	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

TABLE 2A-2

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

Location	WINTER: Winter Requirements BTU x 10^6	WINTER: Heat Pump Cooling Capacity with COP = 3.2 BTU x 10^6	SUMMER: Summer Requirements BTU x 10^6	SUMMER: Heat Pump 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	920,565 sq. ft.
		2,203,200 sq. ft.

FIGURE 2A-1 MARKET SQUARE, WASHINGTON, DC

HOUSING PLAN

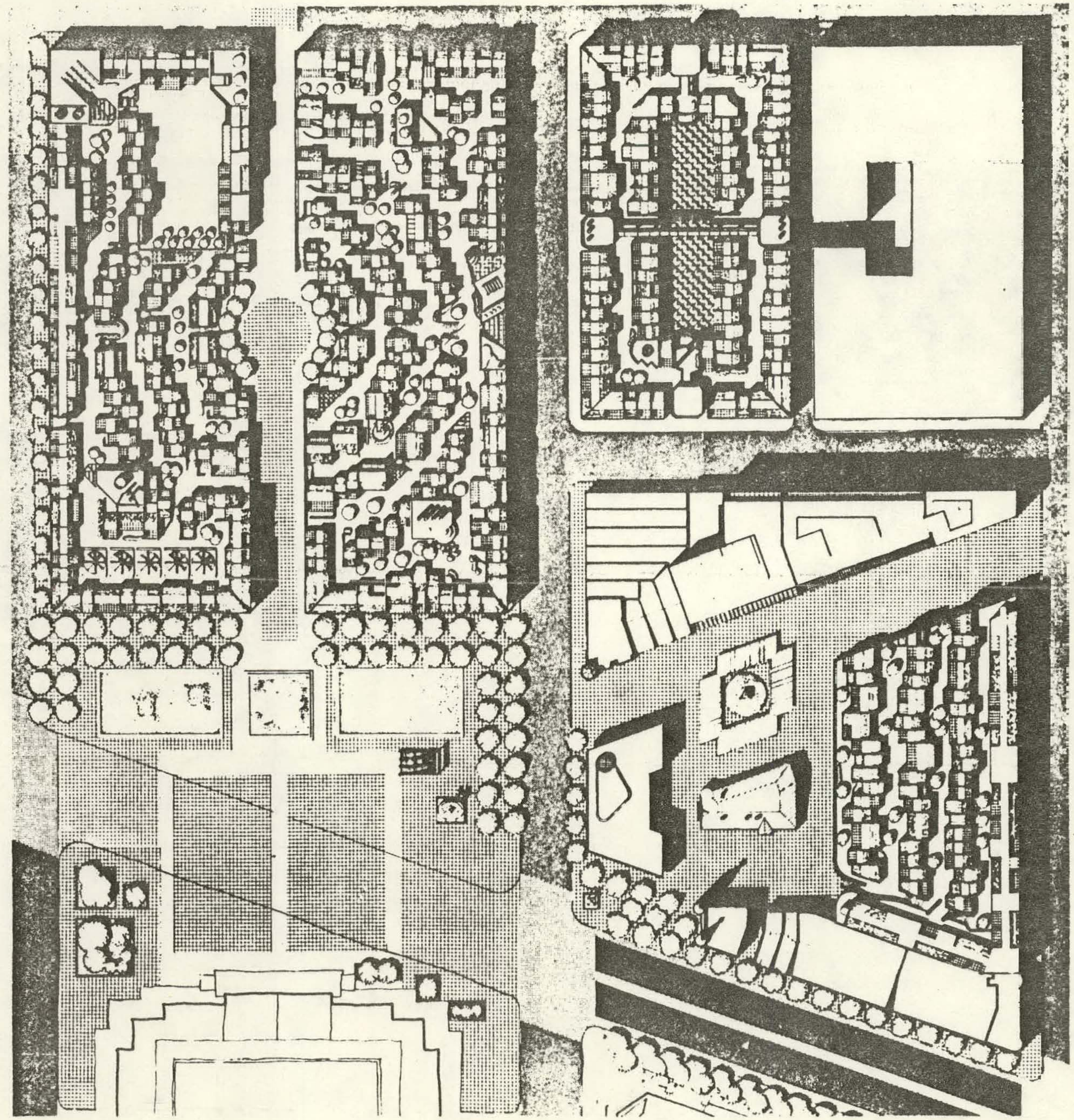
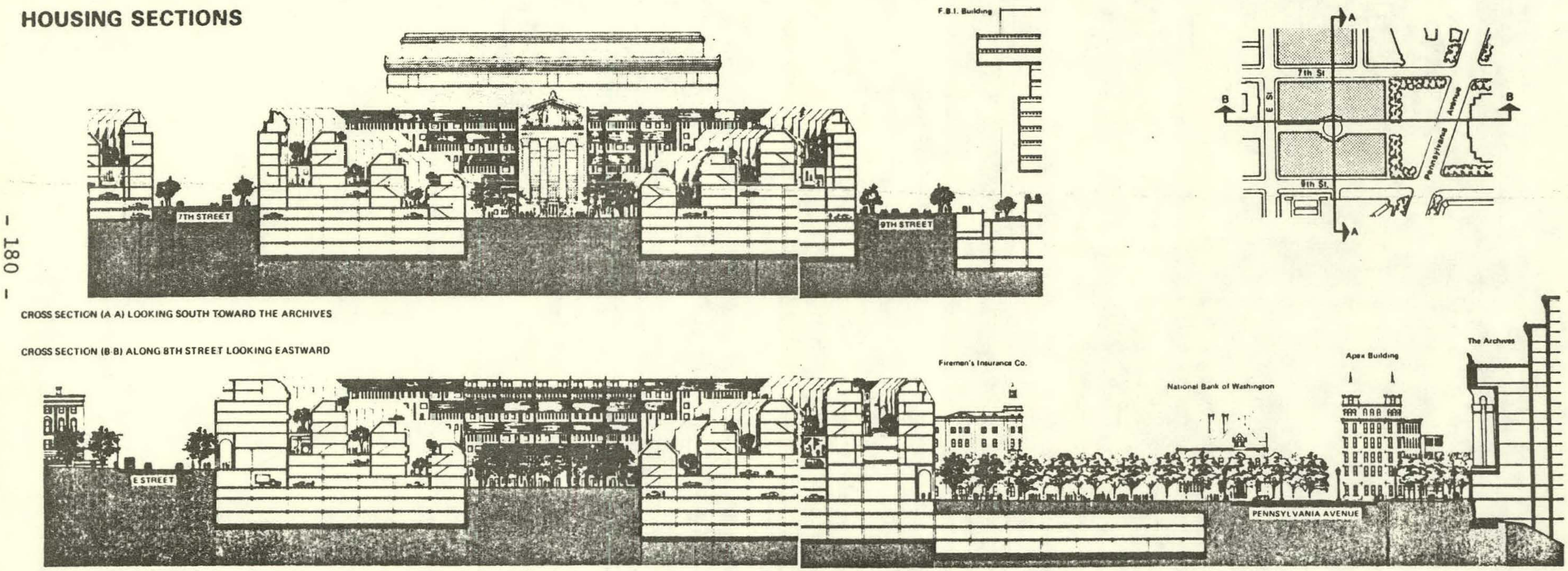


FIGURE 2A-2 MARKET SQUARE, WASHINGTON, DC



2. Envelope

Residential	404,000 sq. ft.
Commercial	50,000 sq. ft.
Total	454,000 sq. ft.

3. Roof

Residential	140,000 sq. ft.
Commercial	124,000 sq. ft.
Total	204,000 sq. ft.

4. Archives Below Grade

Walls	101,400 sq. ft.
Roof	321,600 sq. ft.
Floor	321,600 sq. ft.
Total	744,600 sq. ft.

5. "U" Factors

Wall	0.06 BTUH/sq. ft - $^{\circ}$F
Double Glass	0.6 BTUH/sq. ft.- $^{\circ}$F
Roof	0.06 BTUH/sq. ft. -$^{\circ}$F
Slab Below Grade	0.063BTUH/sq. ft.-$^{\circ}$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 inflitration 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. =	2,515 KW
Total =	4,443 KW

TABLE 2A-3

HEAT TRANSFER COEFFICIENTS FOR SELECTED LOCATIONS IN THE UNITED STATES

Location	Heating Degree Days	Wall Overall Heat Transfer Coefficient Heating BTU/Hr-Ft2-°F		Roof Overall Heat transfer Coefficient BTU/Hr-Ft2-°F		Below Grade Heat transfer Coefficient BTU/Hr-Ft2-°F	Overall Thermal Transfer Value Cooling (BTU/Hr-Ft2)	
		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

Location	Envelope and Roof Transmission BTUH	Below Grade Transmission BTUH	Ventilation Load BTUH	Total BTUH
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

TABLE 2A-5

PEAK COOLING LOAD

Location	Solar and Transmission BTUH	People BTUH	Lights and Appliances BTUH	Ventilation BTUH	Total BTUH x 10^3	Total 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	593	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

*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 BTU x 10^6	Surplus Cooling 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

Year	Households Millions	Ratio of Annual Increase
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

Year		Ratio of Increase
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

Year		Ratio of Increase
1971	206×10^{12} Btu	
1976	230×10^{12} Btu	1.118
1981	253×10^{12} Btu	1.100
1986	275×10^{12} Btu	1.087
1990	291×10^{12} Btu	1.057
1995	309×10^{12} Btu	1.060
2000	327×10^{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

$13,317 \times 10^{12} - 476 \times 10^{12} = 12,841 \times 10^{12}$ 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

$$= \frac{327 \times 10^{12}}{2.2 \times 3413} \times 9000 = 392 \times 10^{12} \text{ Btu}$$

$$= \frac{392 \times 10^{12}}{140{,}000 \times 42} = 67 \times 10^{6} \text{ barrels}$$

Energy input for heating and cooling = $(3120 + 67) \times 10^6$

= $3{,}187 \times 10^6$ barrels of oil

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

$$= \frac{12{,}841 \times 10^{12} \times 9000}{3.2 \times 3413} = 10{,}582 \times 10^{12} \text{ Btu}$$

$$= \frac{10{,}582 \times 10^{12}}{140{,}000 \times 42} = 1{,}800 \times 10^{6} \text{ barrels}$$

Subtotal $(2{,}145 + 1{,}800) \times 10^6 = 3{,}945 \times 10^6$ barrels

Compressor input to ice generation (from above) = 67 barrels

Total $(3945 + 67) \times 10^6 = 4{,}012 \times 10^6$ 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.

Heating and DHW = $13,317 \times 10^{12}$ Btu

Cooling obtainable = $\dfrac{13,317 \times 2.2 \times 10^{12}}{3.2}$

$= 9,155.10^{12}$ Btu

Compressor input

$$\frac{9,155 \times 10^{12}}{2.2 \times 3413} = 10,975 \times 10^{12} \text{ Btu}$$

$$= \frac{10,975 \times 10^{12}}{140,000 \times 42} = 1,866 \times 10^{6} \text{ barrels}$$

Excess Cooling $(9,155 - 327) \times 10^{12}$ Btu

$= 8,828 \times 10^{12}$ Btu

Supplementary heat required to melt excess ice

$$= \frac{8,828 \times 10^{12}}{0.7 \times 140,000 \times 42} = 2145 \times 10^{6} \text{ barrels}$$

Compressor input = $\underline{1,866 \times 10^{6}}$ barrels

Total = $4,011 \times 10^{6}$ barrels (as before)

2.4 Conventional System

Cooling required 327×10^{12} Btu using 0.8 KW/Ton barrels of

oil = $\dfrac{327 \times 10^{12} \times 0.8 \times 9000}{12,000 \times 140,000 \times 42} = 33 \times 10^{6}$

Heating $13,317 \times 10^{12}$ Btu

$$= \frac{13,317 \times 10^{12}}{0.7 \times 140,000 \times 42} = 3,235 \times 10^{6} \text{ barrels}$$

Total $= (3,235 + 33) \times 10^{6}$

$= 3,268 \times 10^{6}$ 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

Year		Ratio of Increase
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

Year		Ratio of Increase
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

Cooling required by the year 2000 $= 804 \times 10^{12}$ Btu

$= 67{,}000 \times 10^{6}$ ton hours

With 1 KW/Ton Savings $= 67{,}000 \times 10^{6}$ kWh

$$= \frac{67{,}000 \times 10^{6} \times 9000}{140{,}000 \times 42} = 103 \times 10^{6} \text{ barrels}$$

3.3 Heat Pump System Plus Direct Heating

Cooling Required $= 804 \times 10^{12}$ Btu

Heating Obtain by Heat Pump $= \dfrac{804 \times 10^{12} \times 3.2}{2.2} = 1169 \times 10^{12}$ Btu

Additional Heat Energy Required $= (7626 - 1169) \times 10^{12}$

$= 6457 \times 10^{12}$ Btu

$$= \frac{6457 \times 10^{12}}{0.7 \times 140{,}000 \times 42} = 1569 \times 10^{6} \text{ barrels of oil}$$

$$\text{Compressor input} = \frac{804 \times 10^{12} \times 9000}{2.2 \times 3413} = 964 \times 10^{12} \text{ Btu}$$

$$= \frac{964 \times 10^{12}}{140,000 \times 42} = 164 \times 10^{6} \text{ barrels}$$

$$\text{Total} = (1509 + 164) \times 10^{6} = 1733 \times 10^{6} \text{ barrels}$$

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

$$\frac{6457 \times 10^{12} \times 2.2}{3.2 \text{x} 0.7 \text{x} 140,000 \text{x} 42} = 1079 \times 10^{6} \text{ barrels}$$

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

Heating & DHW $= 7626 \times 10^{12}$ 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

$$\frac{7626 \times 10^{12} \times 2.2}{3.2} = 5243 \times 10^{12} \text{ Btu}$$

Cooling Required $= \underline{804 \times 10^{12} \text{ Btu}}$

Excess $= 4439 \times 10^{12}$ Btu

Supplementary Heat

$$= \frac{4439 \times 10^{12}}{0.7 \times 140{,}000 \times 42} = 1078 \times 10^{6} \text{ barrels}$$

Total = $(1069 + 1078) \times 10^{6} = 2147 \times 10^{6}$ barrels (as before)

3.5 Conventional System

Cooling Required $= 804 \times 10^{12}$ Btu

$$= \frac{804 \times 10^{12}}{12{,}000} = 67{,}000 \text{ ton-hours}$$

With 0.8 KW/Ton, Compressor input

$$= 67{,}000 \times 0.8 = 53{,}600 \times 10^{6} \text{ KW/T}$$

Barrels of Oil $= \dfrac{53{,}600 \times 10^{6} \times 9000}{140{,}000 \times 42} = 82 \times 10^{6}$ barrels

Heating $= 7626 \times 10^{12}$ Btu

Barrels of Oil $= \dfrac{7626 \times 10^{12}}{0.7 \times 140{,}000 \times 42} = \underline{1853}$ barrels

Total $= 1935 \times 10^{6}$ barrels

3.6 Recapitulation

Heat Pump with Direct Heating $= 1733 \times 10^{6}$ barrels

Heat Pump with Ice Melting $= 2148 \times 10^{6}$ barrels

Conventional $= 1935 \times 10^{6}$ barrels

3.7 Conclusions

Savings $(1935 - 1733) \times 10^{6}$ barrels

$= 202 \times 10^{6}$ barrels

Refrigeration auxiliaries 0.2 KW/Ton

$$= 67{,}000 \times 10^6 \times 0.2 = 13{,}400 \times 10^6 \text{ kWh}$$

$$= \frac{13{,}400 \times 9000 \times 10^6}{140{,}000 \times 42} = 21 \times 10^6 \text{ barrels}$$

Total Savings $= (202 + 21) \times 10^6 = 223 \times 10^6$ 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: Extropolated from Fig. 2-13

T_R – Total regional savings in year 2000

NEW ENGLAND -

312×10^6 bbls x .05 x .764 x .837 = 9.98×10^6 bbls

MIDDLE ATLANTIC

$312\ 10^6$ bbls x .15 x .873 x .908 = 37.10×10^6

EAST NORTH CENTRAL

312×10^6 bbls x .21 x .773 x .846 = 42.85×10^6 bbls

WEST NORTH CENTRAL

312×10^6 bbls x .18 x .507 x .852 = 10.78×10^6 bbls

SOUTH ATLANTIC

312×10^6 bbls x .12 x .654 x .701 = 17.16×10^6 bbls

EAST SOUTH CENTRAL

312×10^6 bbls x .06 x 521 x .664 = 6.48×10^6 bbls

WEST SOUTH CENTRAL

312×10^6 bbls x .17 x .677 x .330 = 11.85×10^6 bbls

MOUNTAIN

312×10^6 bbls x .05 x .597 x .804 = 7.49×10^6 bbls

PACIFIC

312×10^6 bbls x .11 x .877 x .706 = 21.25×10^6 bbls

Total Barrels of Oil Saving in the U.S. in the Year 2000 — 164.94×10^6 bbls

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APPENDIX 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 of the system, operation cost in year o, 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 o.
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 o, 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 salavage value is added to the present value of the system and the life cycle savings of the system is printed.

The following formula is used:

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

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 o.
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

fig 4A·1 Present Value Conventional System: Boilers and Chillers

#RUN

1.00	YEAR	11.00	YEAR
-545454.55	COST	-653148.51	COST
-2545454.55	PVTL	-8576316.55	PVTL
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
-607729.00	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

SALVAGE VALUE (PV)
89186.18
PRESENT VALUE
-14926613.89

fig 4A-2 Present Value Heat Pump System:

Ice Storage HP-ICES

PROJECT LIFE
20.00
FUEL ESCALATION
12.00 %
DISCOUNT RATE
10.00 %
OPERATION COST
436000.00
CAPITAL COST
3812000.00
SALVAGE VALUE
1305000.00

*RUN

1.00	YEAR	11.00	YEAR
-396363.64	COST	-474621.25	COST
-4208363.64	PVTL	-8590790.03	PVTL
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

SALVAGE VALUE (PV)
193979.93
PRESENT VALUE
-13076168.12

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
149090.91	COST	178527.26	COST
-1662909.09	PVTL	-14473.48	PVTL
2.00	YEAR	12.00	YEAR
151801.65	COST	181773.21	COST
-1511107.44	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

SALVAGE VALUE (PV)
104793.76
PRESENT VALUE
1850445.78

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APPENDIX 7A.1 EVALUATION OF THE ABILITY OF OUTSIDE AIR IN WINTER TO PRODUCE SUPPLEMENTARY ICE FOR 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 occurence 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-$^{\circ}$F and freezing point of 4.5°F. With a Δ T = 10°F the energy relationship would be:

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

yielding gpm = 3110

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

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

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

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

= 118,370 KWH

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

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

Cost Analysis

Energy Cost

Winter

Heat Pump (see 14.2.1) = $113,253

Summer

Fans and Pumps

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

Heat Pump

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

Total for year = $129,380

Equipment First Cost

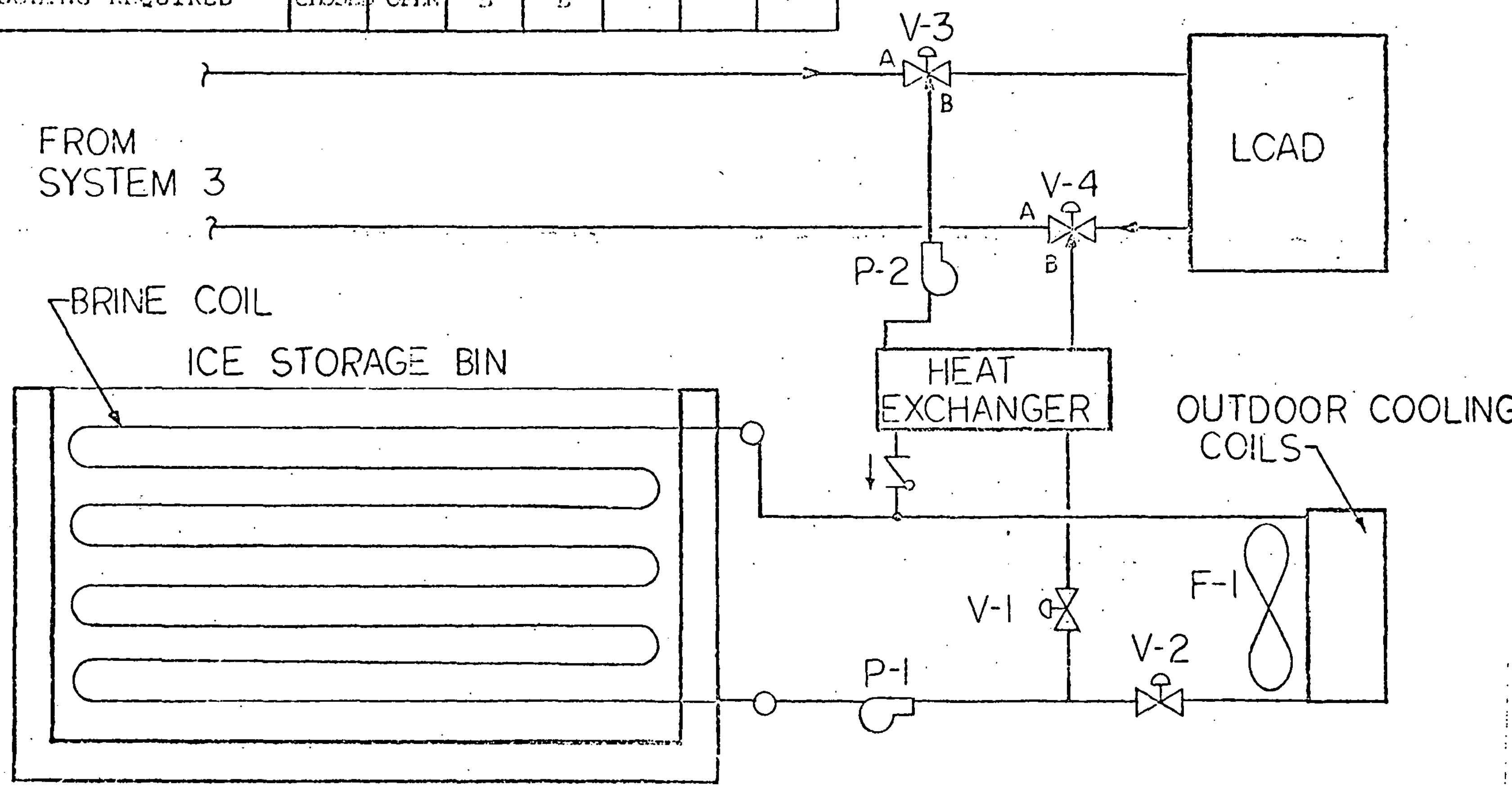
Heat Pump (see 14.212)	=	$ 2,600,000
Additional ice bin storage	=	750,000
42 Axial fans	=	126,000
1 pump	=	3,000
outside air coil ($0.18 per cfm)	=	270,000
Total	=	$ 3,752,000

FIGURE 7A-1

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	ON	OFF	ON
SUMMER MODE COOLING REQUIRED	CLOSED	OPEN	OPEN B	OPEN B	ON	ON	--



5/31/78

HP-ICES PROJECT

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:

$$\frac{31,203 \times 10^6}{3.2} \times 2.2 = 21,452 \times 10^6 \text{ Btu}$$

$$\text{OR} \quad \frac{21,452 \times 10^6}{12,000} = 1,787,672 \text{ Ton-Hours per Year}$$

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:

$$\frac{31,203 \times 10^6}{3.2 \times 3413} = 2.857 \times 10^6 \text{ kWh}$$

Energy cost $= 2.857 \times 10^6 \times \$0.018$

$= \$51.426$

Heat supplied by heat pump $= 31,203 \times 10^6$ Btu

Heating load = 12,500,000 Btu (after deducting internal heat gain)

Equivalent full load hours for heating $= \frac{31,203 \times 10^6}{12,500,000} = 2496$

$$KW = \frac{kWh}{Hours} = \frac{2.857 \times 10^6}{2496} = 1145 \text{ kW}$$

(Chilled and hot water pumps not considered in any scheme)

Note: This figure is obtained also by considering part load.

Billing demand charge = 1145 x \$4.15 x 5 months

= \$23,759

For pumps consider 96 kW =

Charge = 96 x 4.15 x 5 = \$1992

Total Cost = \$51,426

\$23,759

\$ 1,992

\$77,177

The heating capacity is given by:

$$\frac{12{,}675 \times 10^6 \times 3.2}{2.2} = 18{,}436 \times 10^6 \text{ Btu}$$

Heat required: $= 31{,}203 \times 10^6$

Heat supplied by heat pump generating ice for cooling:

$= 18{,}436 \times 10^6$ Btu

Heat supplied by heat pump generating ice melted:

$= 12{,}767 \times 10^6$ 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)

Solar energy required $= \frac{12{,}767 \times 2.2}{3.2} \times 10^6 = 8.777 \times 10^6$ 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)		\$200,000
	Subtotal	\$1,150,000
Solar collector system		500,000
	Total	\$1,650,000

Annual Operating Cost

Energy		
Cooling and Heating		$ 77,177
Maintenance		$ 4,000
Insurance		$ 3,000
Personnel		$ 40,000
	Total	$ 124,177

Present Worth

26.78987 x 124,177		=	$3,326,685
First Cost		=	$1,650,000
	PW		$4,976,685
	Say PW	=	$4,977,000

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.

Heat input 7256 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^{o}F, 330^{o}F temperature difference, sp. heat = 0.24 Btu/lb − ^{o}F

Heat Recovered = $16{,}060 \times 0.24 \times 330 \times \frac{1}{960}$

= 1325 Btu/bhp-hr

= $\frac{1325}{2352} \times 100$ = 56.3% of exhaust heat

Heat from lube oil recovered through jacket water i.e.

$$1357 + 904 = 2261 \text{ Btu/bhp-hr}$$

Total recovered = $2261 + 1325 = 3586$ Btu/bhp-hr

$$\text{Diesel Cycle Efficiency} = \frac{\text{Work + Heat Recovered}}{\text{Heat Input}}$$

$$= \frac{2545 + 3586}{7256} \times 100 = 84.5\%$$

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	=	94,000 Btu/hr
	Total	=	6,966,000 Btu/hr

Heat recovered from jacket water	=	2,171,000 Btu/hr
Heat recovered from exhaust	=	1,272,000 Btu/hr
Subtotal	=	3,443,000 Btu/hr
Work	=	2,443,000 Btu/hr
Total	=	5,886,000 Btu/hr

$$\text{Efficiency (Check)} = \frac{5,886,000}{6,966,000} \times 100 = 84.5\%$$

Heat Pump

Use $COP_c = 2.2$

Cooling obtained = 2.2 x work input = 2.2 x 2,443,000 Btu/hr
= 5,375,000 Btu/hr

Adjusted for Source Energy

$$COP_C = \frac{5,375,000 \text{ Btu/hr}}{6,966,000 \text{ Btu/hr}} = 0.772$$

Heating Obtained

From heat pump	= 3.2 x 2,443,000	=	7,818,000 Btu/hr
From recovered heat		=	3,443,000 Btu/hr
	Total	=	11,261,000 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

$$= \frac{\text{Cooling + Heating}}{\text{Work}}$$

Cooling	=	5,375,000 Btu/hr
Heating	=	11,261,000 Btu/hr
Total	=	16,636,000 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

Caterpiller G398, 500 HP, 1,200 RPM, 12 Cylinder

Heat Balance

Net Work	=	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%)

$$\text{Heat Rate} = \frac{4,222,200}{500} = 8444 \text{ Btu/bhp} \quad (30.2\% \text{ eff.})$$

Heat Recovered

Exhaust	=	733,700 Btu/hr (56%)
Cooling Water (& Lube Oil)	=	1,386,000 Btu/hr
Subtotal	=	2,119,700 Btu/hr
Work	=	1,272,600 Btu/hr
Total	=	3,392,300 Btu/hr

$$\text{Engine Cycle Efficiency} = \frac{3,392,200 \text{ Btu/hr}}{4,222,200} \times 100 = 80.3\%$$

Heat Pump

Use $COP_c = 2.2$

Cooling obtained = 1,272,600 x 2.2 = 2,800,000 Btu/hr

Adjusted for source energy

$$COP_c = \frac{2,800,000 \text{ Btu/hr}}{4,222,200 \text{ Btu/hr}} = 0.663$$

Heating Obtained

From Heat Pump	=	3.2 x 1,272,600	= 4,072,300 Btu/hr
From Heat Recovery	=		= 2,119,700 Btu/hr
		Total	= 6,192,000 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

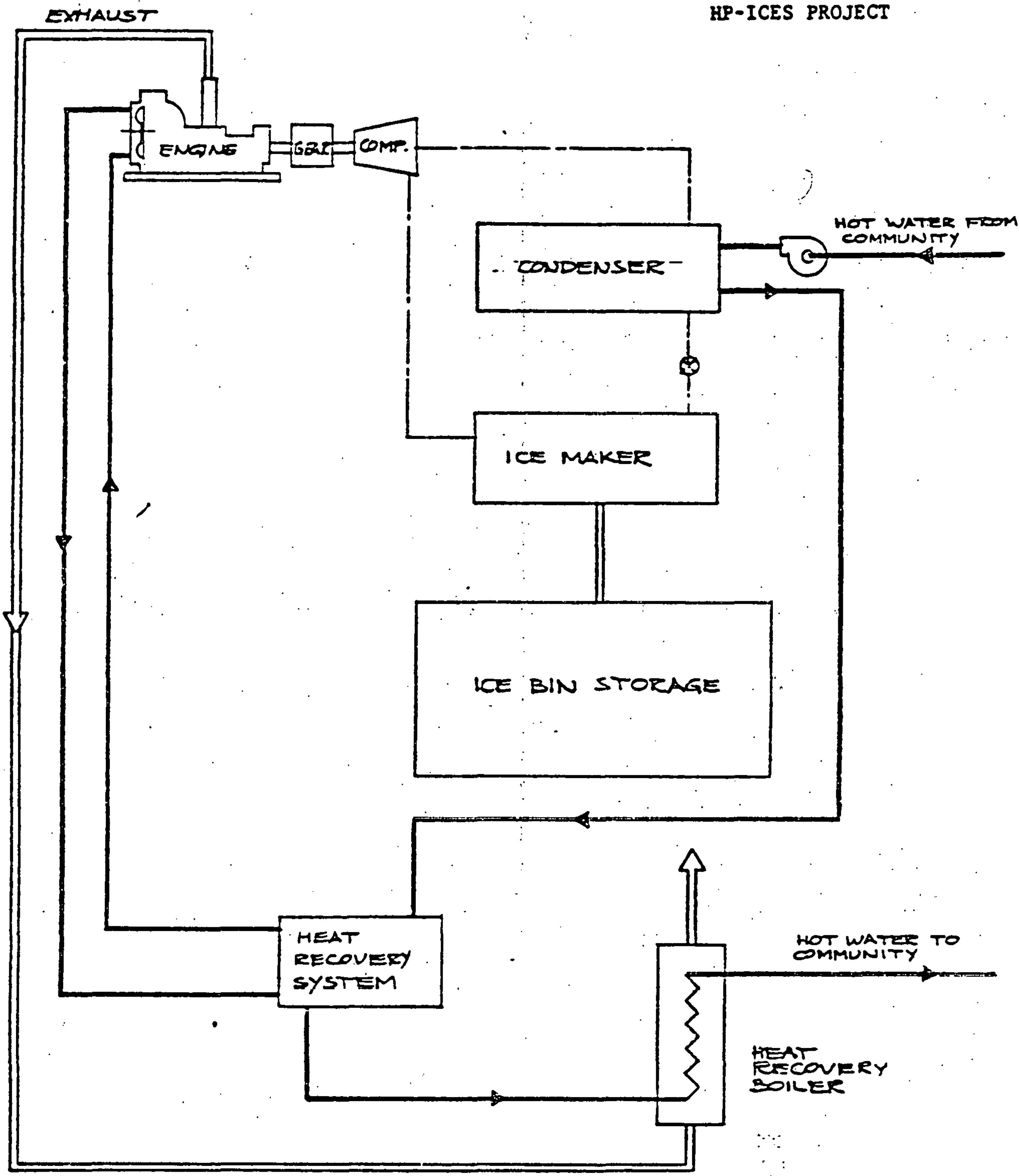


FIG 7A-2. DUAL FUEL (NATURAL GAS OR OIL #2) ENGINE DRIVE ICE GENERATING HP-ICES

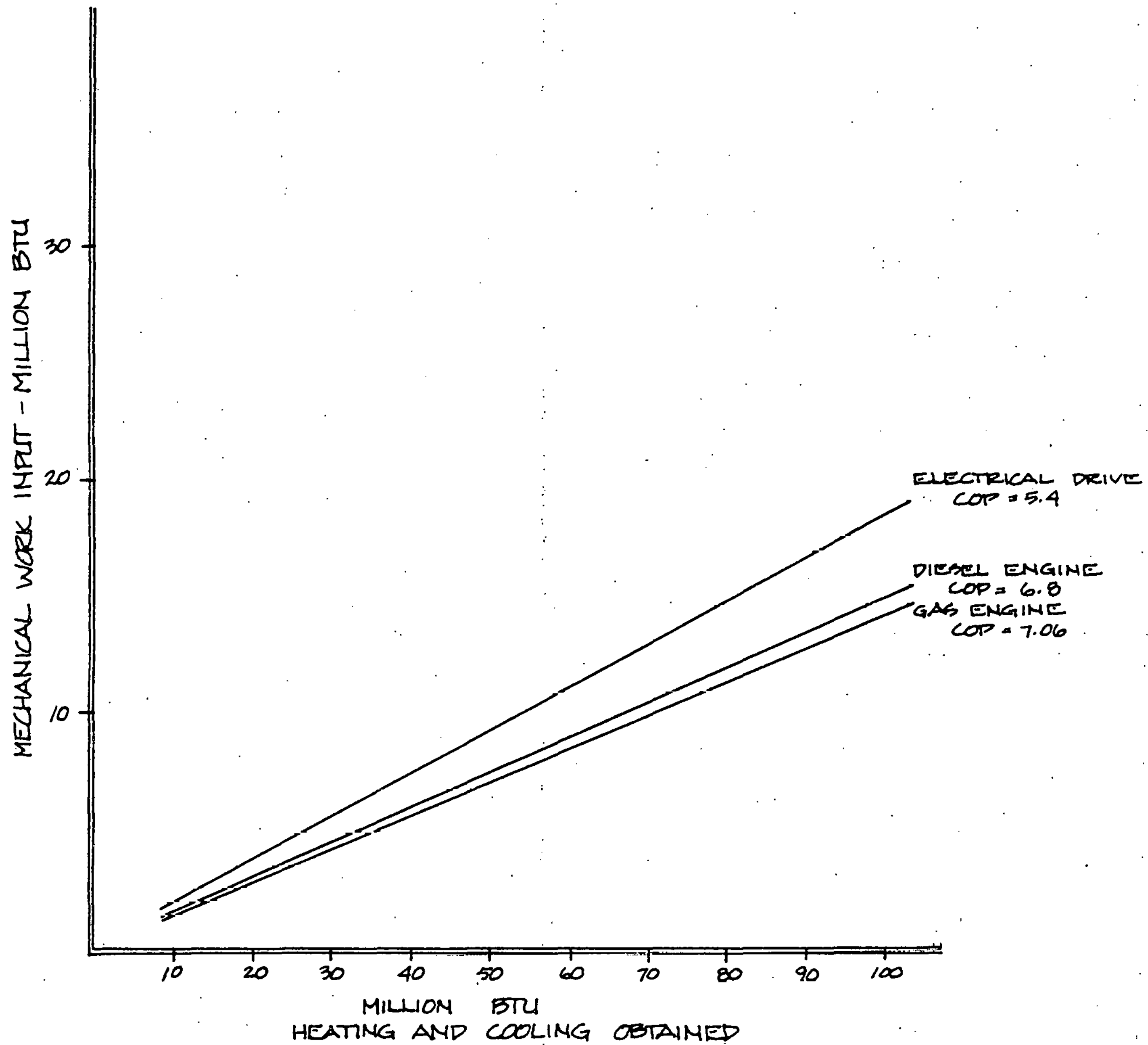


fig 7A·3 Annual Overall COP of a Diesel Engine, Gas Engine, and Electric Drive Heat Pump.

COP Related to Mechanical Work Input

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A. MARKET SQUARE COMPLEX

WASHINGTON, D.C.

SUMMARY

1. AREAS

Sector	Area ft^2	Per cent of Total
Residential (776 dwelling units)	735,000	33.4
Commercial		
a. Department Stores & Offices	171,000	7.8
b. Community Storage	40,000	1.8
c. National Archives	1,258,000 (921,000 below grade)	57.0
Total	2,204,000	100.0

2. LOADS

Heating		Domestic Hot Water		Cooling	
Demand	Annual Energy Requirement	Demand	Annual Energy Requirement	Demand	Annual Energy Requirement
13,064,000 Btu/HR	$27,028 \times 10^6$ BTU	3,086,000 BTU/hr	9211×10^6 BTU	3205 Tons	3,009,000 ton-hr $= 36,108 \times 10^6$ BTU

3. COOLING SAVED BY THE USE OF HP-ICES (BY-PRODUCT OF HEATING).

$23,258 \times 10^6$ BTU or 64% of $36,108 \times 10^6$ BTU

4. PERFORMANCE AND COSTS

	HP-ICES System	Conventional System
1. Source Energy Consumption		
Winter	$33,573\times10^6$ BTU	$51,770\times10^6$ BTU
Summer	$17,439\times10^6$ BTU	$33,439\times10^6$ BTU
Total	$51,013\times10^6$ BTU	$85,209\times10^6$ BTU
Energy Savings BTU	$34,197\times10^6$	---
2. Annual Energy Cost	$100,568	$263,048
Energy Cost Saving	$162,480	---
3. Annual Operating Cost	$380,568	$540,048
Oper. Cost Savings	$159,480	---
4. First Cost	$4,547,130	$2,697,255
Incremental First Cost	$1,849,875	
5. Pay-back	7 years	
6. Rate of Return	15%	

A.1 COMMUNITY DESCRIPTION

1.1 INTRODUCTION

The Market Square complex is situated at Pennsylvania Avenue, Washington, D. C., between 7th and 9th streets and is part of Pennsylvania Avenue Development project.

In fact, it is its central point and its design incorporates modern energy conservation techniques and advanced load management concept.

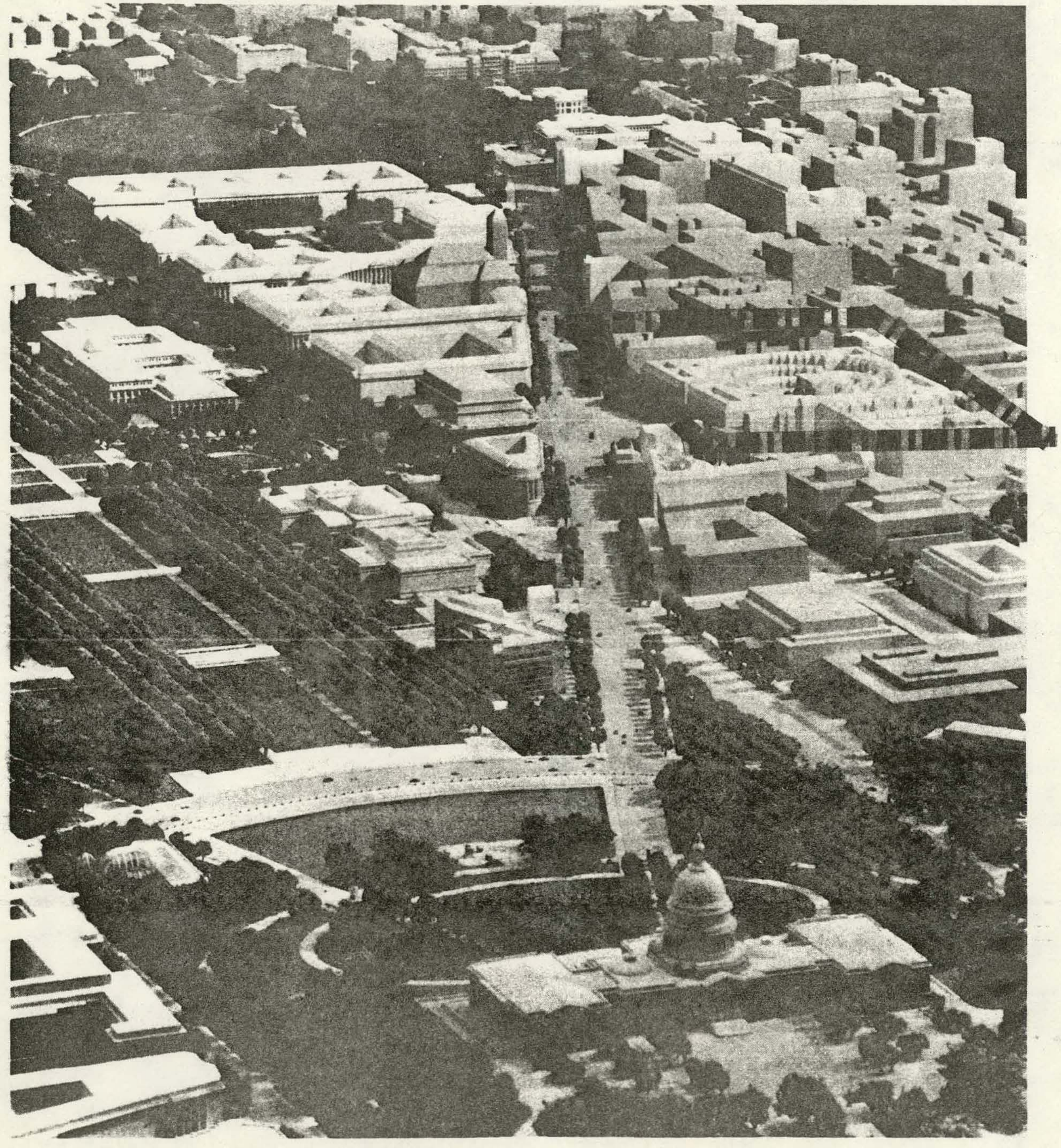
The project is being sponsored and developed by the Pennsylvania Avenue Redevelopment Corporation and it should serve as a showcase, a shining model and example for government and private agencies to follow suit and apply the principles developed for Market Square to other projects, both in the city and country at large.

The Market Square complex conains residential areas, retail and offices, national archives and community storage areas. It is conceived as a major integrated city project with multi-use facilities and a wide functional diversity.

It contains a total area of 2,200,000 ft.^{2} out of which 735,000 ft.^{2} or 33.4% are occupied by 776 dwelling units; 171,000 ft.^{2} or 7.8% are occupied by department stores and offices; 40,000 ft.^{2} or 1.8% are community storage areas and the remaining 1,258,000 ft.^{2} or 57.0% are occupied by national archives, out of which 921,000 ft.^{2} are below grade.

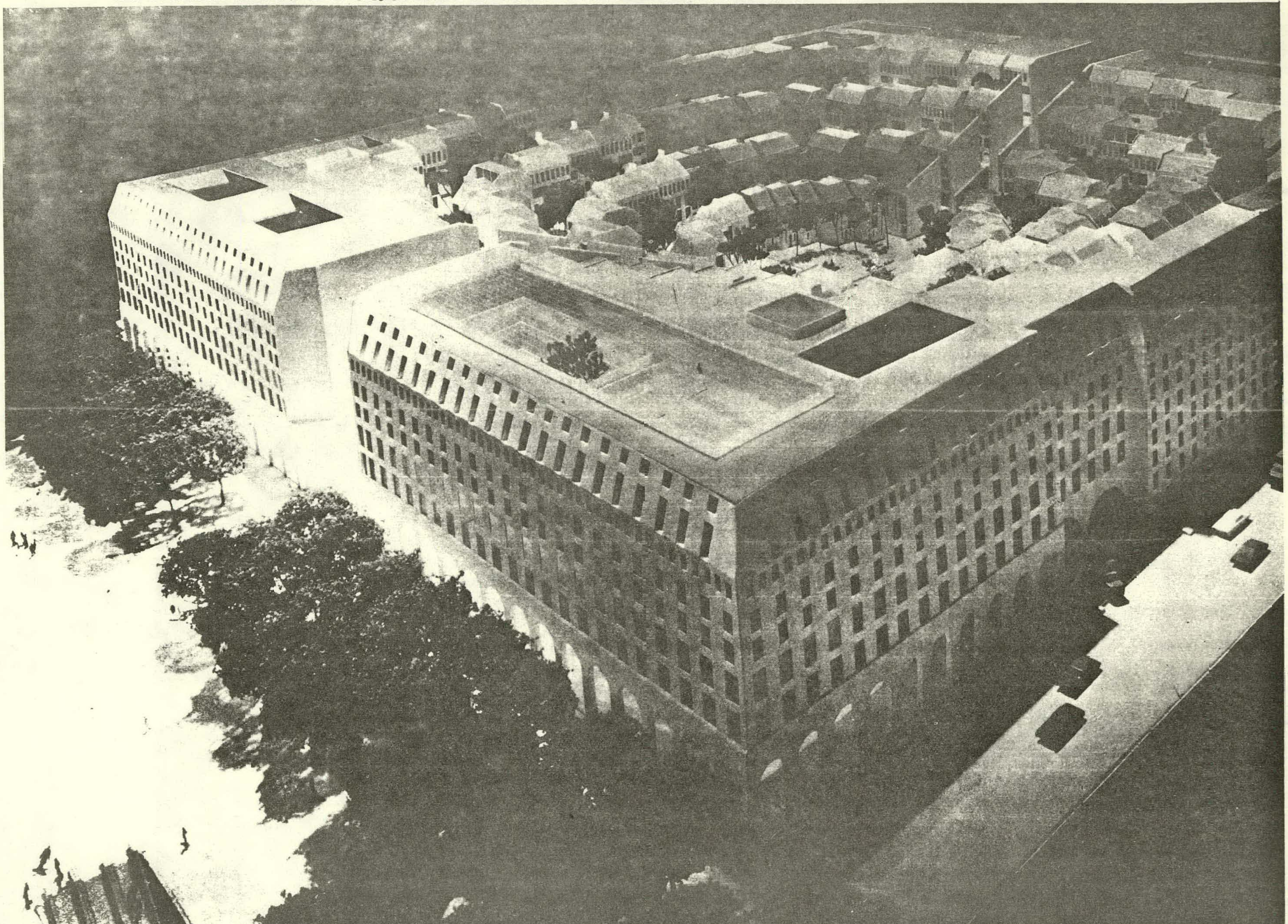
The general building design and construction emphasizes opportunities for the reduction of energy consumption by the building and their associated environmental system (heating, cooling, lighting, etc.) by applying the energy conservation concept on Architectural Design and treatment of the building and building material.

FIG.-A1.1- PENNSYLVANIA AVENUE AERIAL VIEW



MARKET SQUARE OUTLINED

FIG.-A1.2-MARKET SQUARE WASH. D.C. AERIAL VIEW.



Vie 'roposed Housing between 7th and 9th Streets

All exposed walls and roof will be designed with high mass construction of 100 lbs/ft^2 and insulated to achieve an overall "U" value of 0.06 $Btuh/ft^2/^oF$, combined window/wall area in any given space will have a combined "U" value of no greater than 0.12 on north exposure and 0.22 on south.

Where possible, insulation will be located on the external surface of walls and roof so that advantage is taken of the structure's thermal mass, which will act as a heat reservoir and smooth out peak loads.

Interior partition and floor slabs will be designed for 60 lbs/ft^2. High mass construction gives the building a high thermal inertia characteristic that reduces the effect of rapidly changing outside conditions. The high mass makes the building react as a large thermal storage system which obviates wasteful short cycling.

The exterior wall surfaces will have light color and absorption coefficient not greater than 0.3. All windows will be double glazed to reduce heat conduction and sound transmission. Windows facing south will be provided with external shading devices designed to exclude high angle summer sun, but allow entry of low angle winter sun. All windows will be carefully caulked and weatherstripped to reduce infiltration of outside air.

Concerning energy conservation, space like corridors, wash rooms, utility rooms, vertical transport and other similar spaces that do not have such stringent temperature condition requirements should be located in those areas where heat loss is potentially the greatest (usually at the north wall). These spaces can then act as a buffer between outside and inside occupied areas.

FIG.- A1.3

HOUSING PLAN

MARKET SQUARE OUTLINED

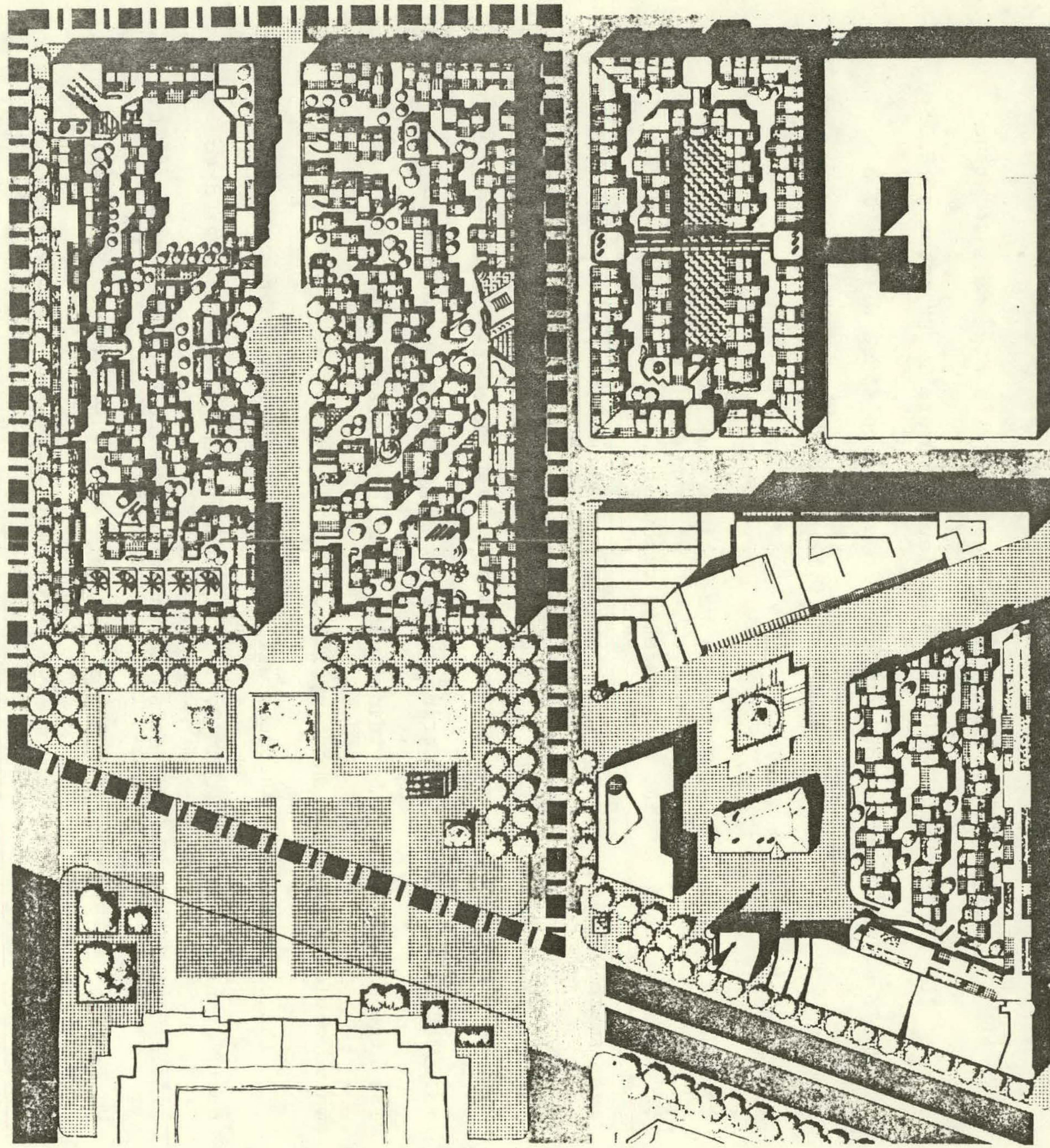
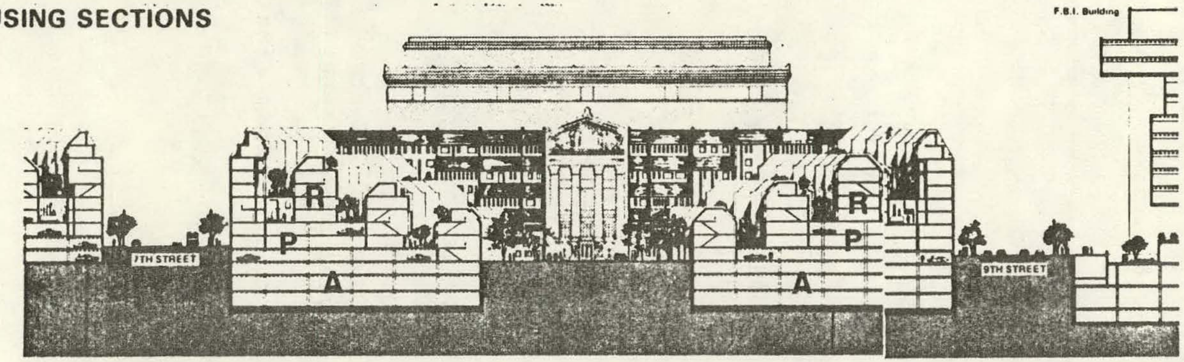
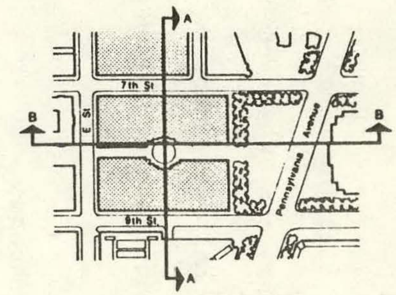


FIG.-A-1.4 MARKET SQUARE

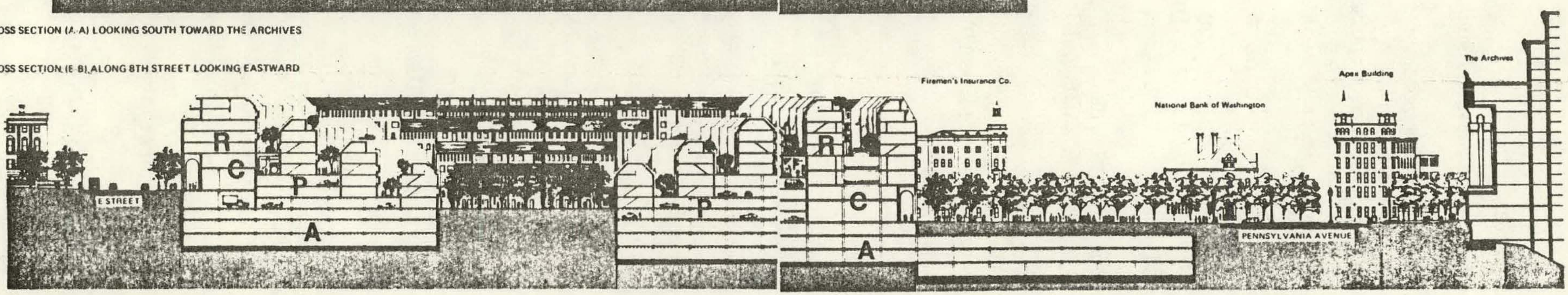
HOUSING SECTIONS





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

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



R – RESIDENTIAL

C – COMMERCIAL

P – PARKING

A – ARCHIVES

Areas of common requirements such as lighting level - same occupancy hours etc. should be grouped together so they can be served by one system designed for those conditions and operated only when required.

The loads and annual energy requirements are as follows:

(a) Cooling:
Demand = 3205 tons
Annual Requirement = 3,009,000 ton-hours
= $36,108 \times 10^6$ BTU

(b) Heating:
Demand = 13,064,000 BTU/HR
Annual Requirement = $27,028 \times 10^6$ BTU

(c) Domestic Hot Water:
Demand = 3,086,000 BTU/HR
Annual Requirement = 9211×10^6 BTU

1.2 DESIGN CONDITIONS*

Latitude 39°

Winter design temperature 19°F

Summer DB-92°F, WB-77°F

Inside temperatures, winter 68°F, night set back 58°F

Summer 78°F DB, 50% RH

Daily temperature range summer 18°F

Equivalent full load hours cooling 700-1200

Heating degree-days 4224

Overall thermal transfer value for cooling

= 20.494 BTU/HR ft^2
(Table 2A-3 System Development Report)

Overall wall heat transfer coefficient for heating (including infiltration)

=0.305 BTU/HR-ft^2-°F

Below grade heat transfer coefficient

= 0.063 BTU/HR-ft^2-°F

*ASHRAE GUIDE 1) Fundamentals ch. 23
2) Systems ch. 43

Roof u factor = = 0.06 BTU/HR-ft^2-^{o}F

Ventilation 0.11 cfm/ft^2 for above grade areas for summer only; below grade for summer and winter.

1.3 BASIC ASSUMPTIONS

a. People

Residential - 3 persons for dwelling unit

Retail and offices - 100 ft^2 per person

Community Storage and Archives - 2 W/ft^2

1.4 AREA TABULATION

RESIDENTIAL		
776 dwelling units		735,000 ft^2
COMMERCIAL		
Retail & Office		171,000 ft^2
Community Storage		40,000 ft^2
Archive Above Grade		337,000 ft^2
Archive Below Grade		920,000 ft^2
	Sub-Total	1,468,000 ft^2
TOTAL (Residential and Commercial)		2,203,000 ft^2

1.5 COOLING PEAK DEMAND ESTIMATE

RESIDENTIAL

a. Solar and transmission

Envelope 404,000 ft^2 x 20.494 BTU/HR-ft^2	= 8,280,000 BTU/HR
Roof 140,000 ft^2 x 0.06 BTU/HR-ft^2-^{o}F x 27^{o}F	= 227,000 BTU/HR
Sub-Total	8,507,000 BTU/HR

b. Ventilation

735,000 x 0.11 x 4.45 x (40.5 -30.0) = 3,778,000 BTU/HR

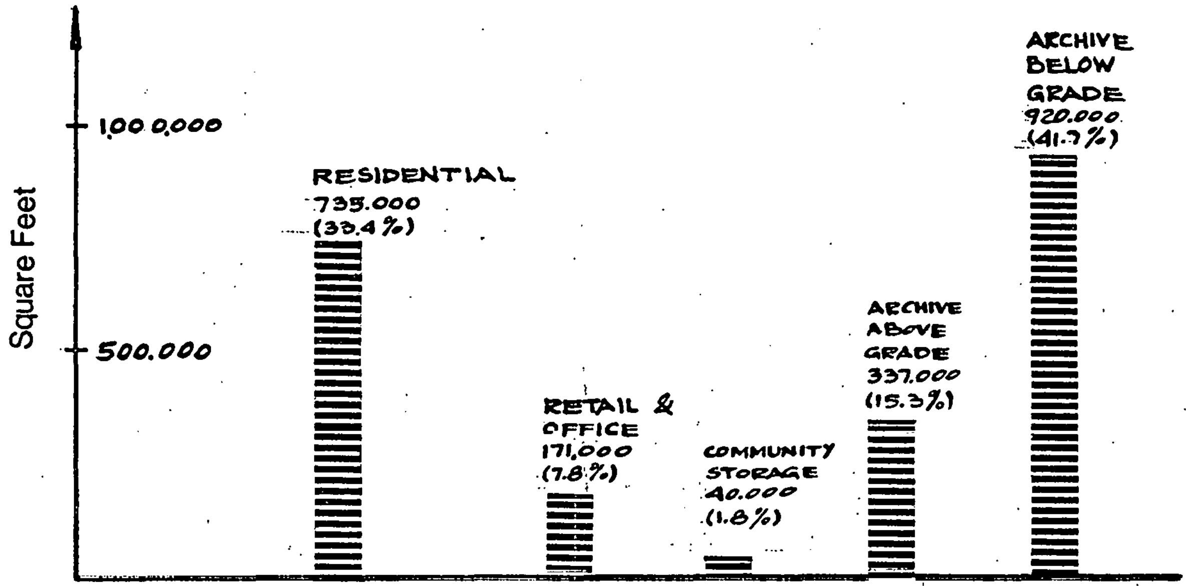
c. Light and Appliances

1.5 KW per D.U., 776 x 1500 x 3.413 = 3,973,000 BTU/HR

d. People

Three persons per D.U., 776 x 3 x 450 = 1,048,000 BTU/HR

TOTAL RESIDENTIAL 17,306,000 BTU/HR



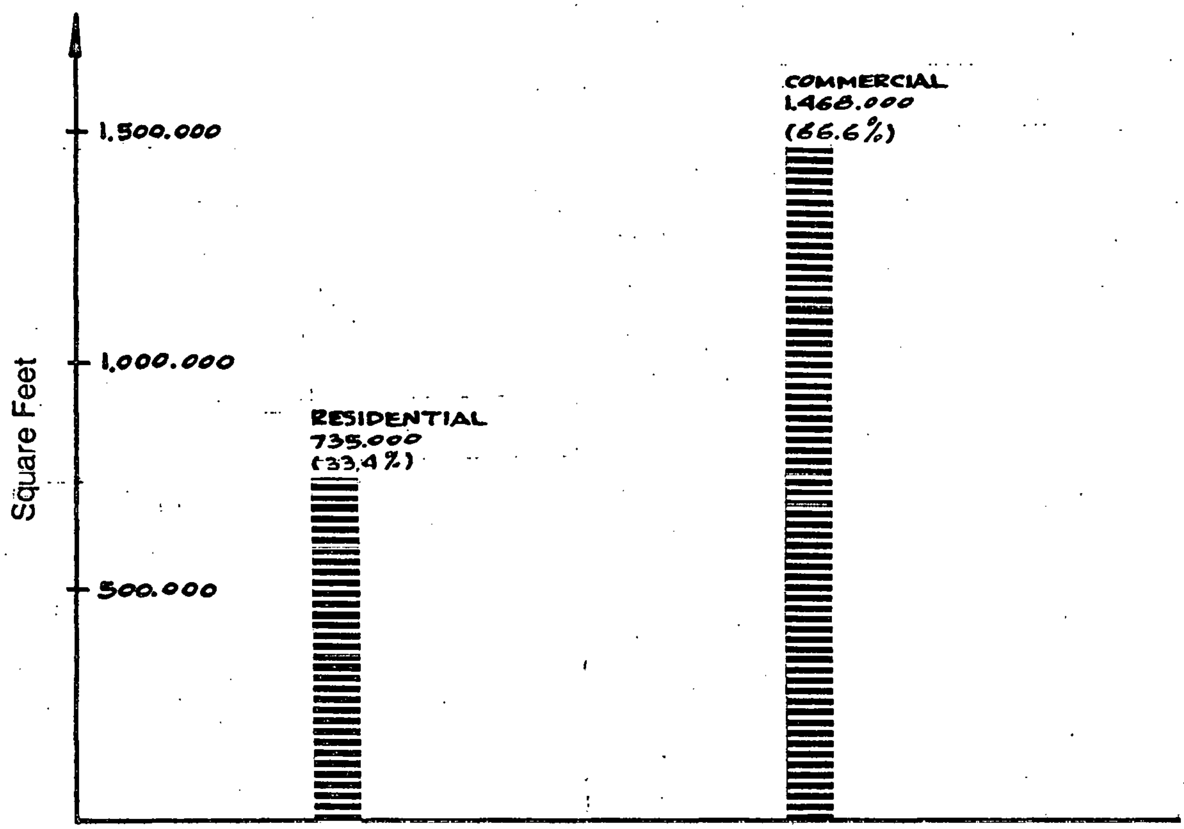


FIG.- A1.5

Area Distribution By Sectors

COMMERCIAL

a. Solar and Transmission
   Envelope 50,000 x 20,494 = 1,025,000 BTU/HR
   Roof 124,000 x 0.06 x 27 = 201,000 BTU/HR

Sub-Total = 1,226,000 BTU/HR

b. Ventilation
   Retail, Community Storage, Archives
   1 1,468,000 x 0.11 x 4.45 x 10.5 = 7,545,000 BTU/HR

c. Light and Appliances
   Retail and Offices 171,000 ft^2 x 4 W/ft^2 = 684,000 W
   Community Storage 40,000 x 2 w/ft^2 = 80,000 W
   Archives 1,256,000 x 2 W/ft^2 = 2,514,000 W

Sub-Total = 3,278,000 W

Load 3,278,000 x 3.413 = 11,188,000 BTU/HR

d. People
   Retail and Offices, 171,000 ft^2:
   100 ft^2/person = 1,710 persons
   Community Storage, 40,000 ft^2: = 100 persons
   Archives 1,257,000: 1500 ft^2/person = 838 persons

Sub-Total = 2,648 persons

Load = 2,648 x 450 = 1,192,000 BTU/HR

e. Total Commercial = 21,151,000 BTU/HR

TABLE A-1.1 MARKET SQUARE HEAT GAIN RECAPITULATION (BTU/hr)

	Solar & Transmission	Ventilation	Light & Appliances	People	Total	
Residential	8,507,000	3,778,000	3,973,000	1,048,000	17,306,000	45.0
Commercial	1,226,000	7,545,000	11,188,000	1,192,000	21,151,000	55.0
Total	9,733,000	11,323,000	15,161,000	2,240,000	38,457,000	100.0
% of Total	25.3	29.4	39.5	5.8	100.0	

Residential $\frac{17,306,000 \text{ BTU/HR}}{12,000 \text{ BTU/HR-ton}}$ = 1442 Ton

Commercial $\frac{21,151,000 \text{ BTU/HR}}{12,000 \text{ BTU/HR-ton}}$ = 1763 Ton

TOTAL = 3205 Ton

1.6 WINTER PEAK COOLING LOAD IN COMMERCIAL SECTOR

Internal heat gain	=	$12,380 \times 10^6$ BTU/HR
Heat Losses in Commercial Area	=	$6,614 \times 10^6$ BTU/HR
Net Heat Gain	=	$5,766 \times 10^6$ BTU/HR
	=	480.5 tons

1.7 ENERGY REQUIREMENT FOR COOLING

Equivalent full load hours

Residential	1200
Commercial	1100

Diversity Factor:

We assume a diversity factor of 0.82 for the cooling energy requirements, and a diversity factor of 1 for the heating energy requirements.

The diversity factor is obtained by assuming load profiles for the residential cooling requirements and the commercial cooling requirements. The ratio of the maximum total load to the sum of the residential peak load and the commercial peak load gives the diversity factor. The value of the diversity factor estimated here is 0.82.

The reason for a diversity factor for the cooling load is that the peak cooling energy usages for the residential and commercial sector do not coincide. In the residential sector, there is only a small occupancy during most of the daytime hours, so that the air-conditioning in the summer would only be used for a small number of hours in the evening (to take care of the time lagged cooling load).

The diversity factor for the heating demand is taken to be 1. The heating demand is assumed to be relatively uniform, and it appears reasonable to require that the maximum heating load to be allowed for is the sum of the peak heating loads.

Cooling Energy Requirement:

(a) Summer:

Residential Sector: 1442 ton x 1200 EFLH x 0.82 = 1,419,000 Ton-HR

Commercial Sector: 1763 ton x 1100 EFLH x 0.82 = 1,590,000 Ton-HR

(b) Winter:

Residential Sector: 0

Commercial Sector: 1763 ton x 367 EFLH = 648,675 Ton-HR

Total Annual Cooling Energy Requirement = 3,657,675 Ton-HR

TABLE A 1.2 ANNUAL ENERGY REQUIREMENT FOR COOLING *

Month	% Cooling Energy Requirement	Cooling Energy Requirements BTU x 10^6
January	3.55	1560
February	3.55	1560
March	3.55	1560
April	4.8	1850
May	8.22	3611
June	10.7	4694
July	18.1	7944
August	18.1	7944
September	15.62	6860
October	6.71	3250
November	3.55	1560
December	3.55	1560
Total Annual	100	43,908

* In the winter, cooling is treated by enthalpy control.

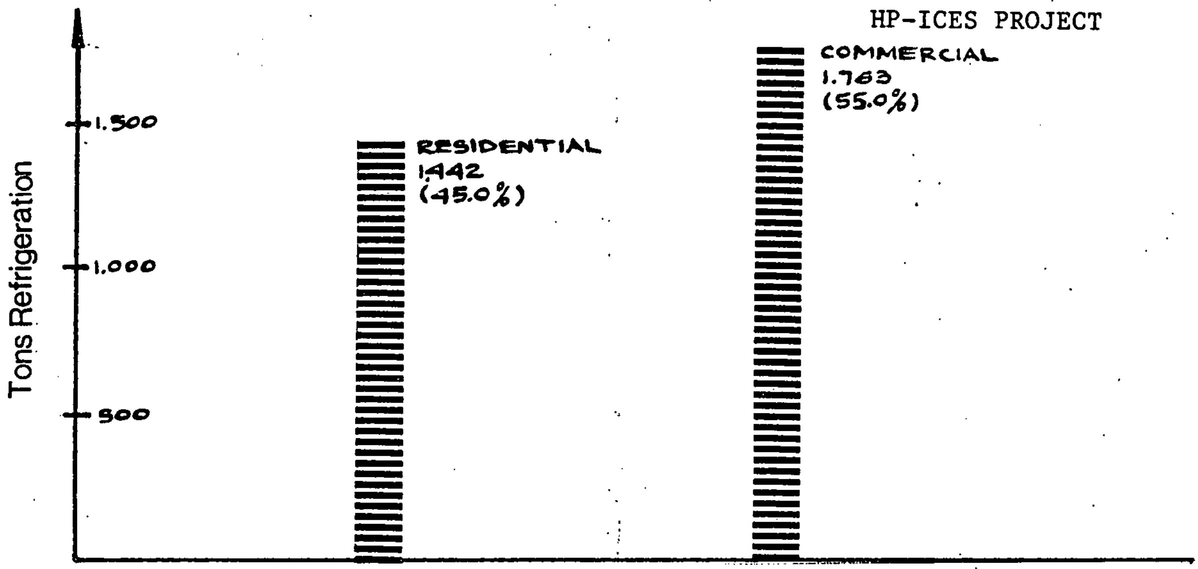


FIG.- A1.6

Tonnage Distribution By Sectors

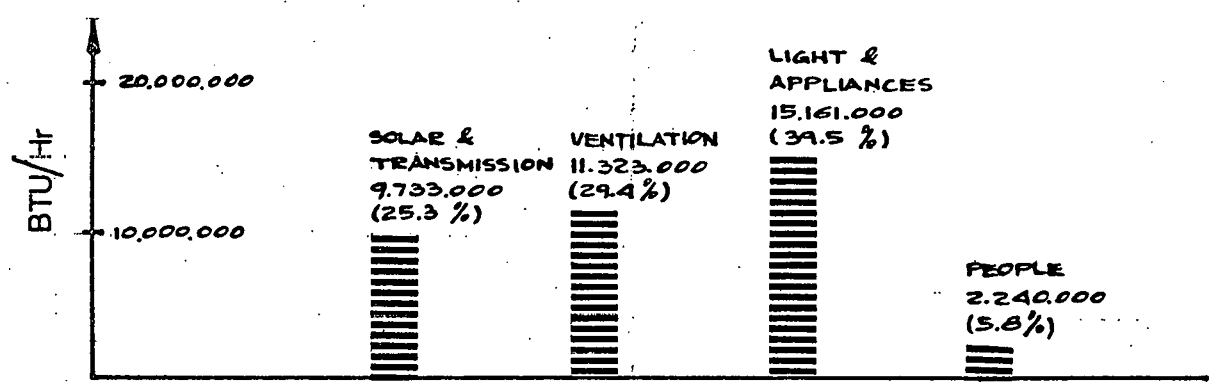


FIG.- A1.7

Cooling Load Distribution

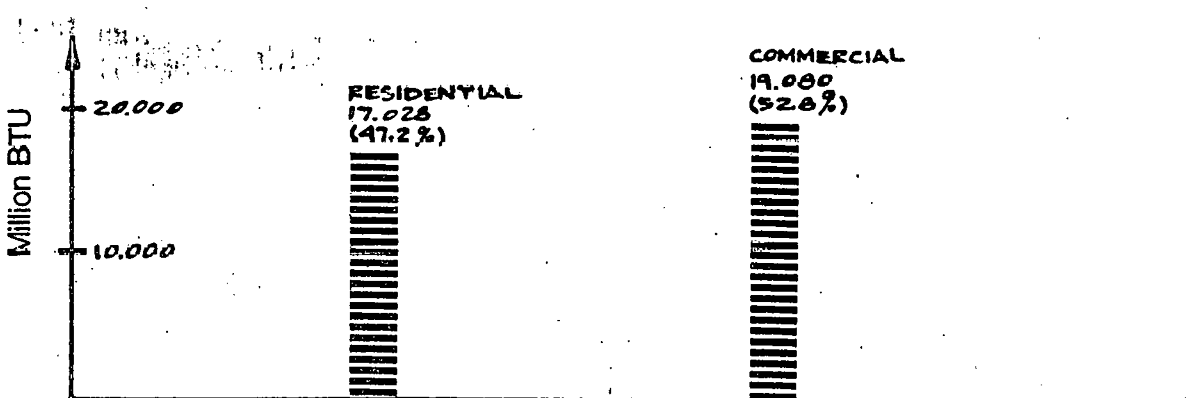


FIG.- A1.8

Annual Energy Requirement For Cooling

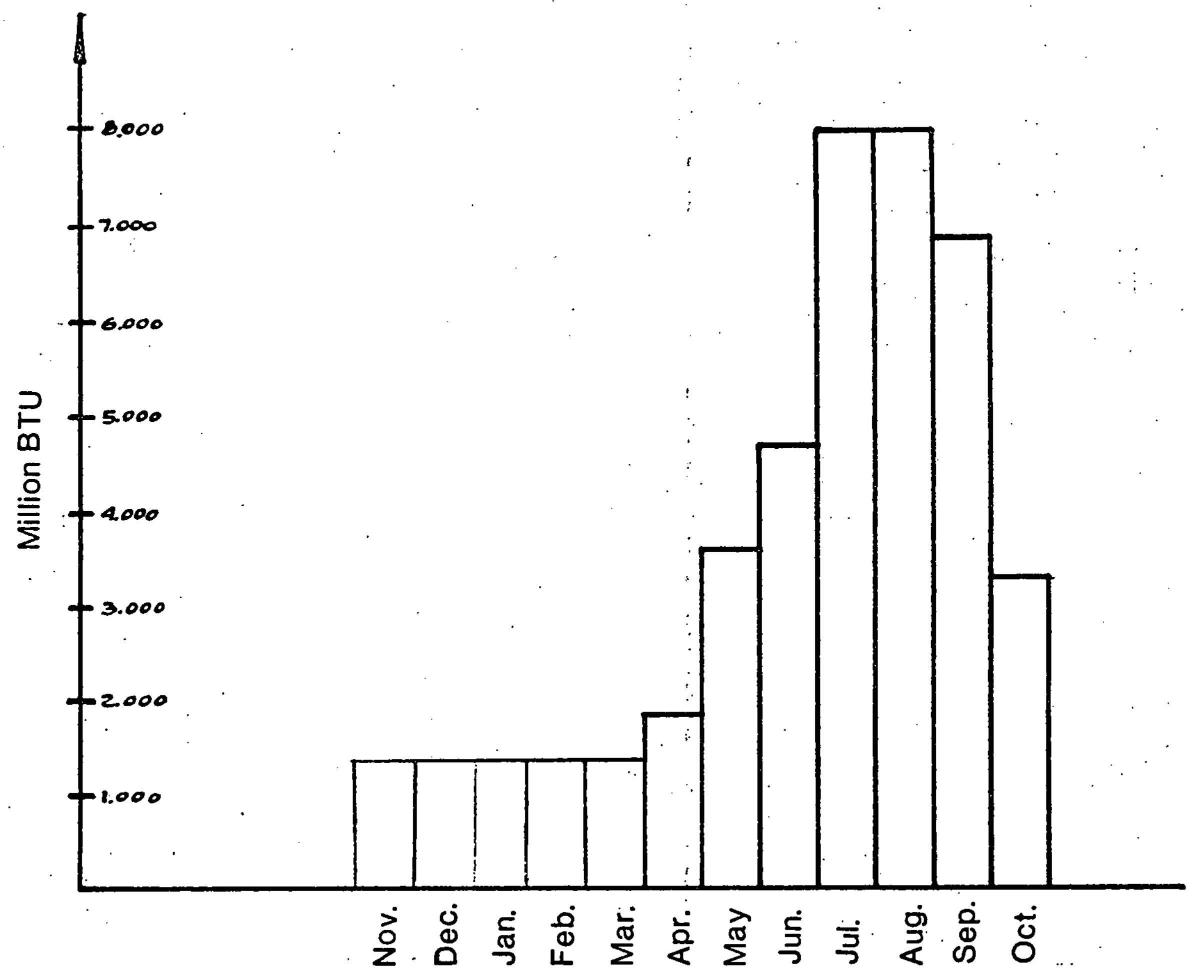


FIG.- A1.9

Annual Cooling Requirement By Month

1.8 HEATING PEAK DEMAND ESTIMATE

Residential

a. Transmission and infiltration

Envelope	404,000 x 0.305 x (68-19)	= 6,038,000 BTU/HR
Roof	140,000 x 0.06 x 49	= 412,000 BTU/HR
	TOTAL	= 6,450,000 BTU/HR

Commercial

a. Transmission and Infiltration

Envelope	50,000 x 0.305 x 49	= 747,000 BTU/HR
Roof	124,000 x 0.06 x 49	= 365,000 BTU/HR
	Sub-Total	=1,112,000 BTU/HR

Below Grade 422,600 x 0.063 x 5.3 = 141,000 BTU/HR

b. Ventilation (Below Grade)

921,000 x 0.11 x 1.08 x 49 = 5,361,000 BTU/HR

Total = 6,614,000 BTU/HR

TABLE A1.3 MARKET SQUARE HEAT LOSSES RECAPITULATION (Btu/hr)

	Transmission & Infiltration	Ventilation	Total
Residential	6,450,000	-	6,450,000
Commercial	1, 253,000	5,361,000	6,614,000
Total	7,703,000	5,361,000	13,064,000

1.9 ENERGY REQUIREMENT FOR HEATING

Residential

$$\frac{6,450,000 \times 4224\ (DD) \times 24}{49} = 13,344 \times 10^6$$

Commercial

$$\frac{6,614,000 \times 4224\ (DD) \times 24}{49} = 13,684 \times 10^6 \text{BTU}$$

TOTAL = $27,028 \times 10^5$ BTU

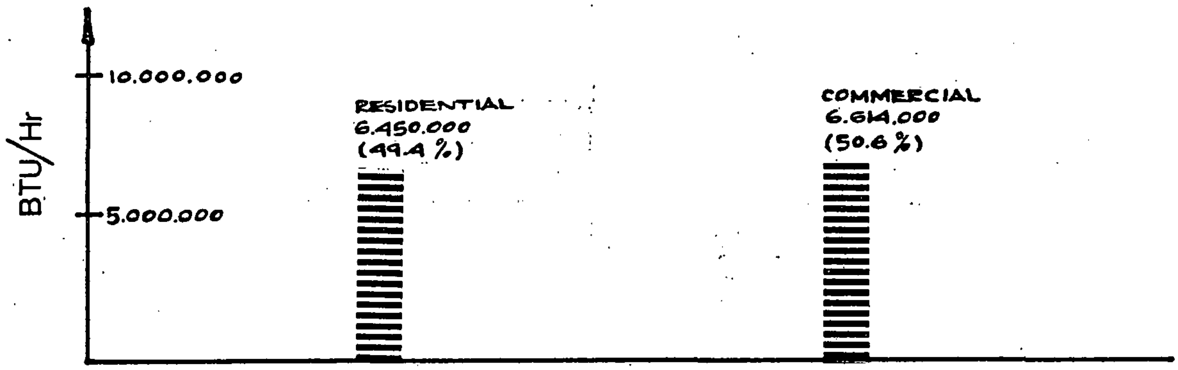


FIG.- A1.10

Heating Peak Demand Distribution By Sectors

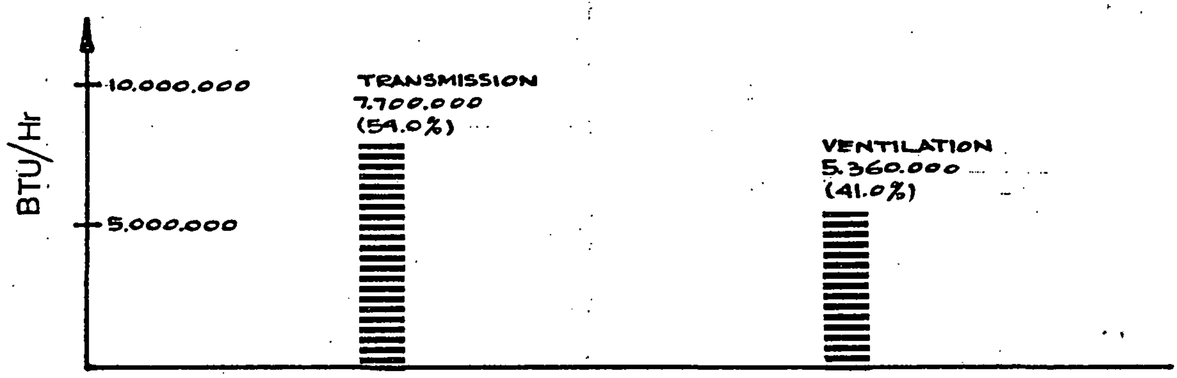


FIG.- A.11

Heating Load Distribution

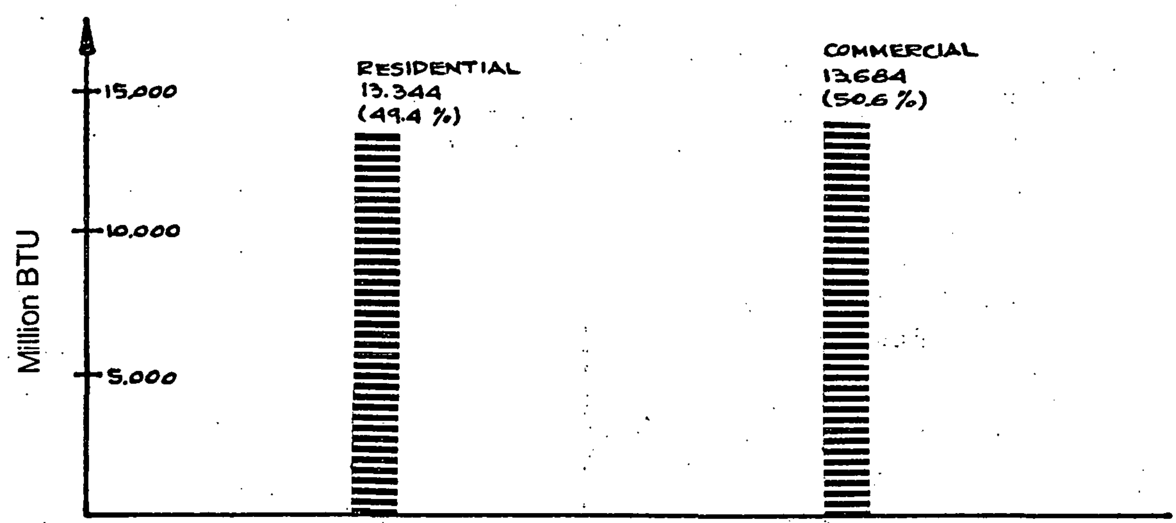


FIG.- A1.12

Annual Energy Requirement For Heating

TABLE A-1.4 ANNUAL HEATING REQUIREMENT

MONTH	DEGREE DAYS	ENERGY REQUIREMENT BTU 10^6
January	871	5573
February	762	4876
March	626	4006
April	288	1843
May	74	475
June	0	0
July	0	0
August	0	0
September	33	212
October	217	1386
November	519	3321
December	834	5336
Total Annual	4224	27,028

1.10 DOMESTIC HOT WATER DEMAND ESTIMATE

The assumed DHW requirement is 60 gallons per day per apartment delivered at a temperature of 120°F.

The recommended ASHRAE requirements for apartments are lower, i.e. between 35 and 37 gallons per day at 140°F.

The present system does not have a central boiler. The DHW will be provided by individual units in the apartments, each with a tank and heat exchanger coil.

Because of the difference in the delivery temperatures and the characteristics of the system, we have assumed a larger DHW requirement (of 60 gallons per day).

The results are the following:

<u>Calculation of Average Hot Water Loads</u>

Residential

60 gallons per day per apartment

$= 60 \times 776$ (No. of D.U.) $= 46{,}560$ gal/day

12 hours usage

$= \dfrac{46{,}560 \times 8.33}{12}$ $= 32{,}320$ lb/hr

Commercial

2 gallons per day per person

$= 2 \times 2648$ $= 5{,}296$ gal/day

5 hours usage $\dfrac{5296 \times 8.33}{5}$ $= 8{,}823$ lb/hr

The DHW energy requirement for the coldest month is the following:

32,320 + 8823
with $\Delta T = 75^{o}F$

Demand = 41,143 x 75 - 3,086,000 BTU/HR

Table A-1.5 gives the results for the monthly DHW energy requirements. The supply water temperatures have been taken from the Handbook of Air Conditioning, Heating and Ventilating, by Strock and Koral, pages 1-157. (1965)

TABLE A-1.5 DOMESTIC HOT WATER ENERGY REQUIREMENTS
(Based on monthly city water temperatures)

Month	BTU x 10^6
January	1010
February	919
March	881
April	804
May	738
June	666
July	686
August	544
September	515
October	673
November	817
December	958
Total Annual	9211

Winter months - October through April
DHW requirements = 6062 x 10^6 BTU

Summer months - May through September
DHW requirements = 3149 x 10^6 BTU

Total = 9211 x 10^6BTU

Ref.: Handbook of Air Conditioning, Heating and Ventilating, by Strock and Koral, pages 1-157. (1965).

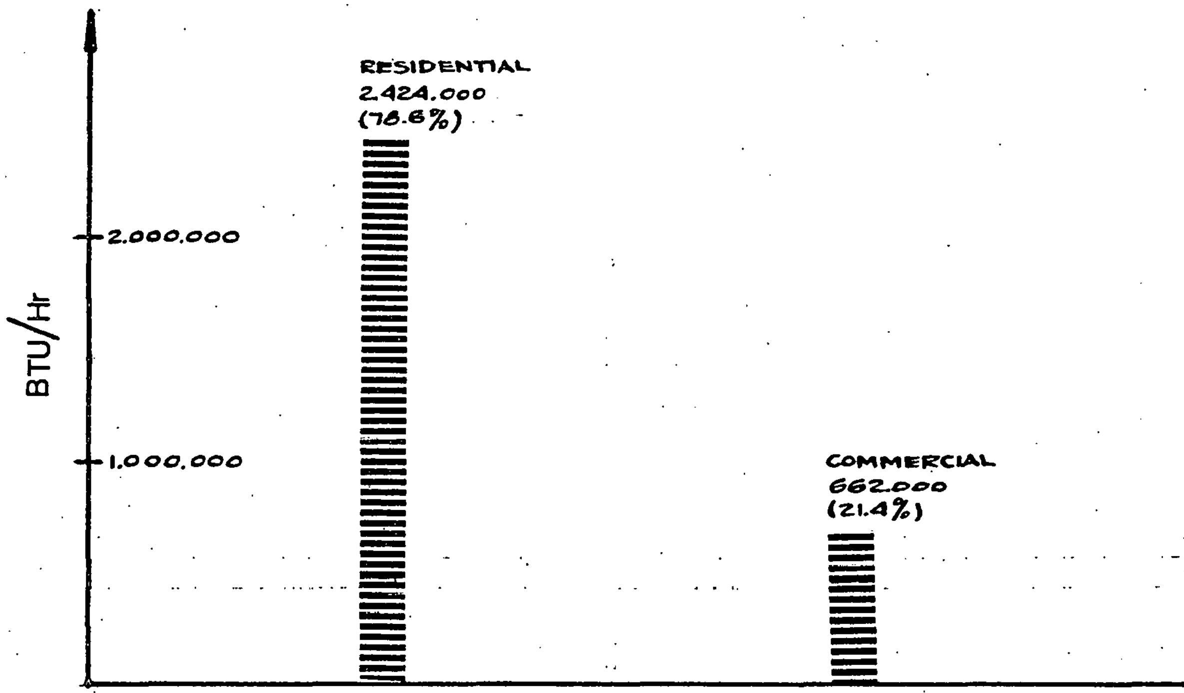


FIG.- A1.13

Domestic Hot Water Demand Distribution

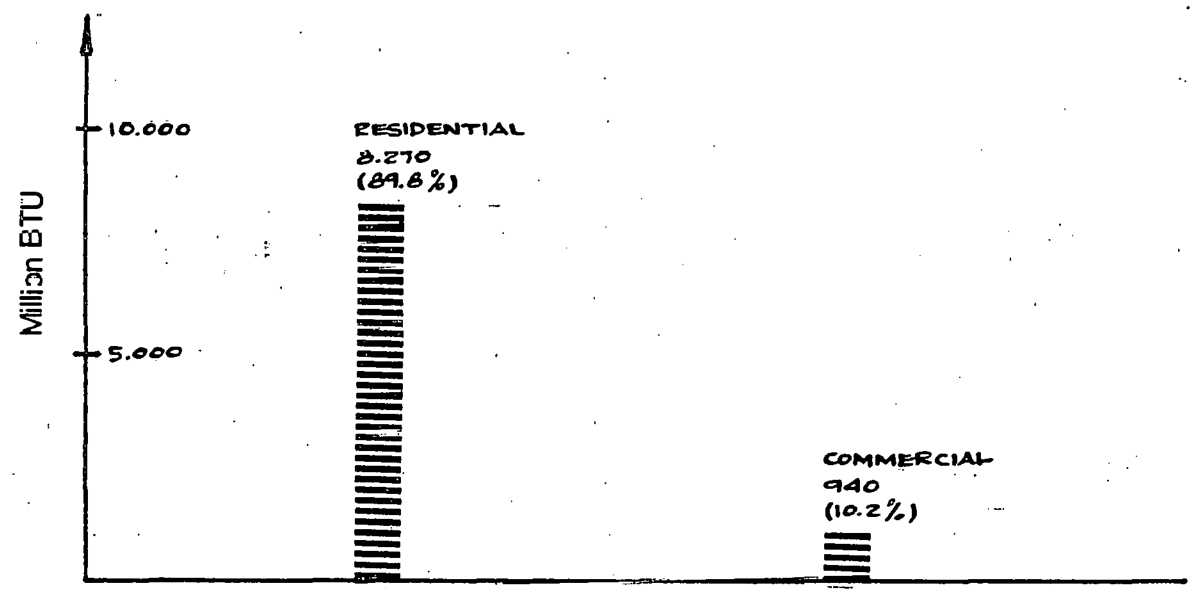


FIG.- A1.14

Annual Energy Requirement For Domestic Hot Water

TABLE A-1.6 ENERGY REQUIREMENTS FOR HEATING & DHW

MONTH	HEATING	DHW	SUBTOTAL
January	5573	1010	6583
February	4876	919	5795
March	4006	881	4887
April	1843	804	2647
May	475	738	1213
June	0	666	666
July	0	686	686
August	0	544	544
September	212	515	727
October	1386	673	2059
November	3321	817	4138
December	5336	958	6294
Total Annual	27,028	9,211	36,239

TABLE A1-7 RECAPITULATION OF ANNUAL ENERGY REQUIREMENTS FOR COOLING,HEATING & DHW

	Residential	Commercial	Total
Cooling	17,043 x 10^6BTU	26880 x 10^6BTU	43908 x 10^6 BTU
Heating	13,352 x 10^6BTU	13,675 x 10^6BTU	27028 x 10^6 BTU
DHW	8271 x 10^6BTU	940 x 10^6BTU	9211 x 10^6 BTU

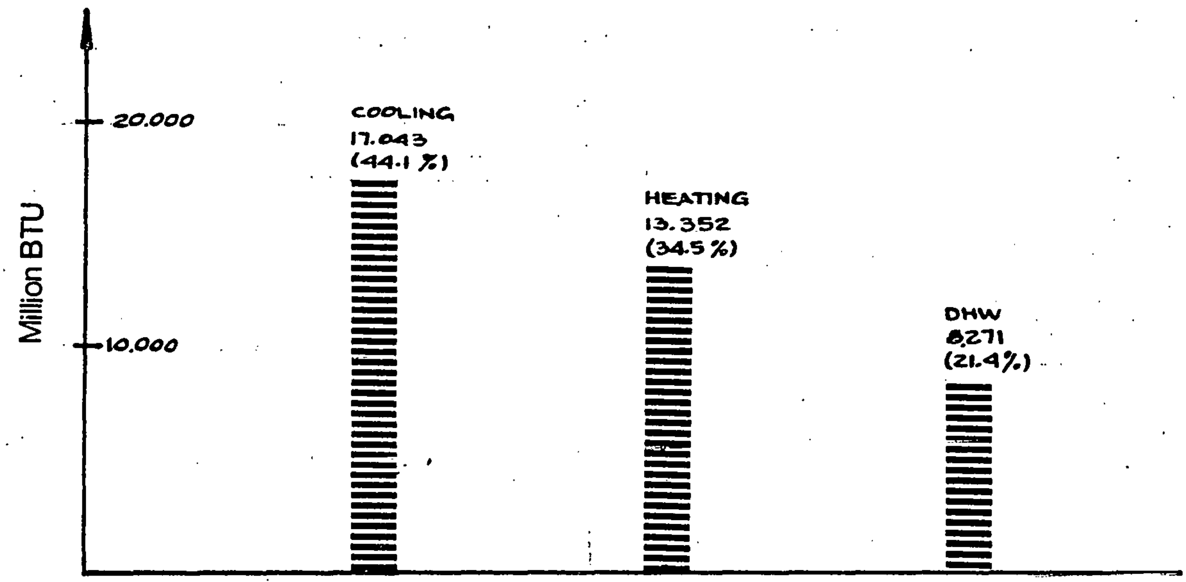


FIG.- A1.15

Residential Sector Comparison Of Annual Energy

Requirement For Cooling , Heating & DHW

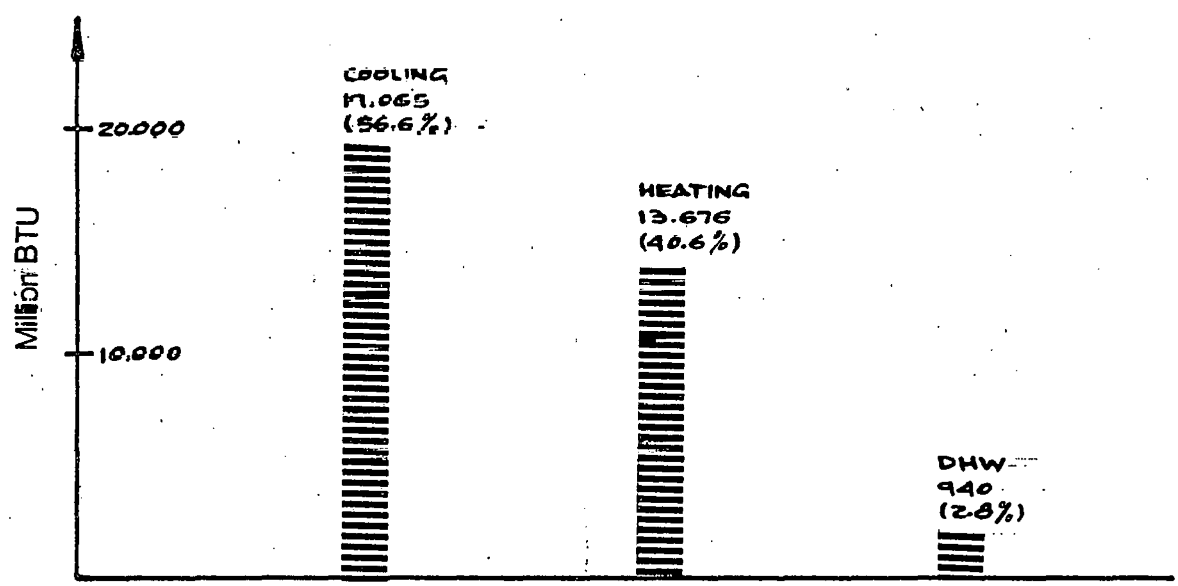


FIG.- A1.16

Commercial Sector Comparison Of Annual Energy

Requirement For Cooling , Heating & DHW

A.2 SYSTEM DESCRIPTION

2.1 System Components

The system will be a heat pump generating ice on heating and will consist of three main components:

- the energy center located in three mechanical rooms and generating the heating, domestic hot water and cooling requirements for the entire complex
- a distribution system
- terminal units

2.1.1 Energy Center

The energy center will include:

a. Three electrically driven screw compressors capable of operating at 20°F SST and 130°F SDT, at 35°F SST and 130°F SDT for summertime generation of D H W with the simultaneous generation of chilled water and at 35°F SST and 105°F SDT for the conventional air conditioning operation in summertime with heat rejection to the cooling towers.

b. Three double bundle condensers

c. Six flat plate ice makers

d. Three chillers

e. Circulating pumps:

- three primary hot water pumps
- two primary chilled water pumps, operating between the ice bin and the primary loop
- three primary chilled water pumps, operating between the chillers and the primary loop and also between the chillers and the ice bin.
- condenser water pumps
- ice plate circulating pumps

2.1.2 Distribution System

The distribution system will consist of:

a. Primary distribution piping

b. Secondary hot water pumps

c. Secondary chilled water pumps

d. Heat exchangers for domestic hot water

2.1.3 Terminal Units

For the residential areas and small offices, there will be a four-pipe fan coil unit system, while for the commercial areas (department stores, landscape offices, places of public assembly, archives) air handling units will be used with proper air distribution systems. For the commercial sector, for which a cooling load may be expected in the winter, enthalpy controllers will be incorporated into the air handlers to minimize the use of mechanical cooling when the outside air conditions will permit.

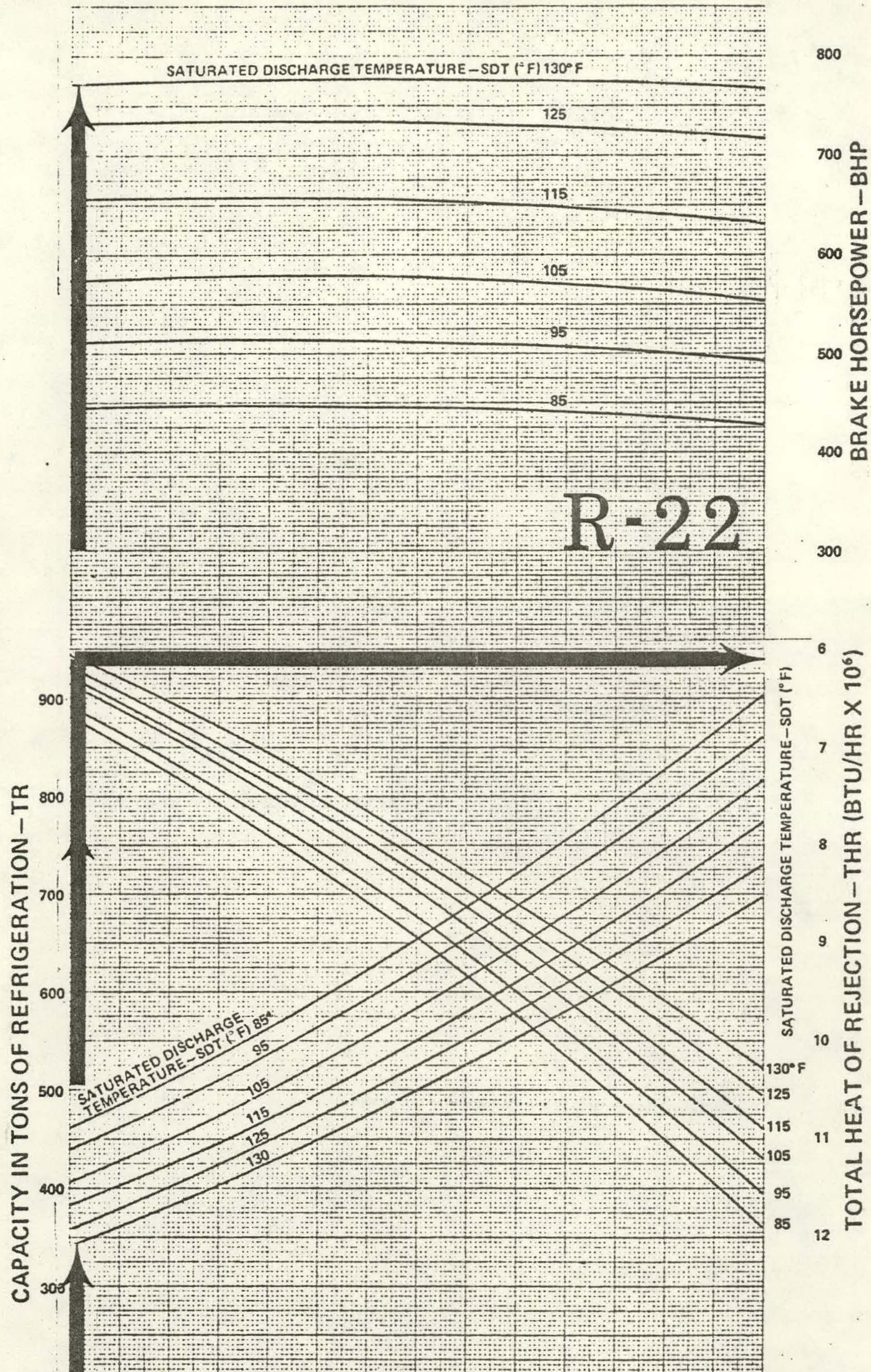
2.2 Equipment Sizing and Selection

2.2.1 Compressors

The compressors are selected according to the heating requirements in winter time when they operate at ADT = 130°F. The compressors should also be able to operate at standard air conditioning conditions that is at 35°F SST and 105°F SDT and their performance at these conditions should be comparable with the performance of a standard centrifugal compressors.

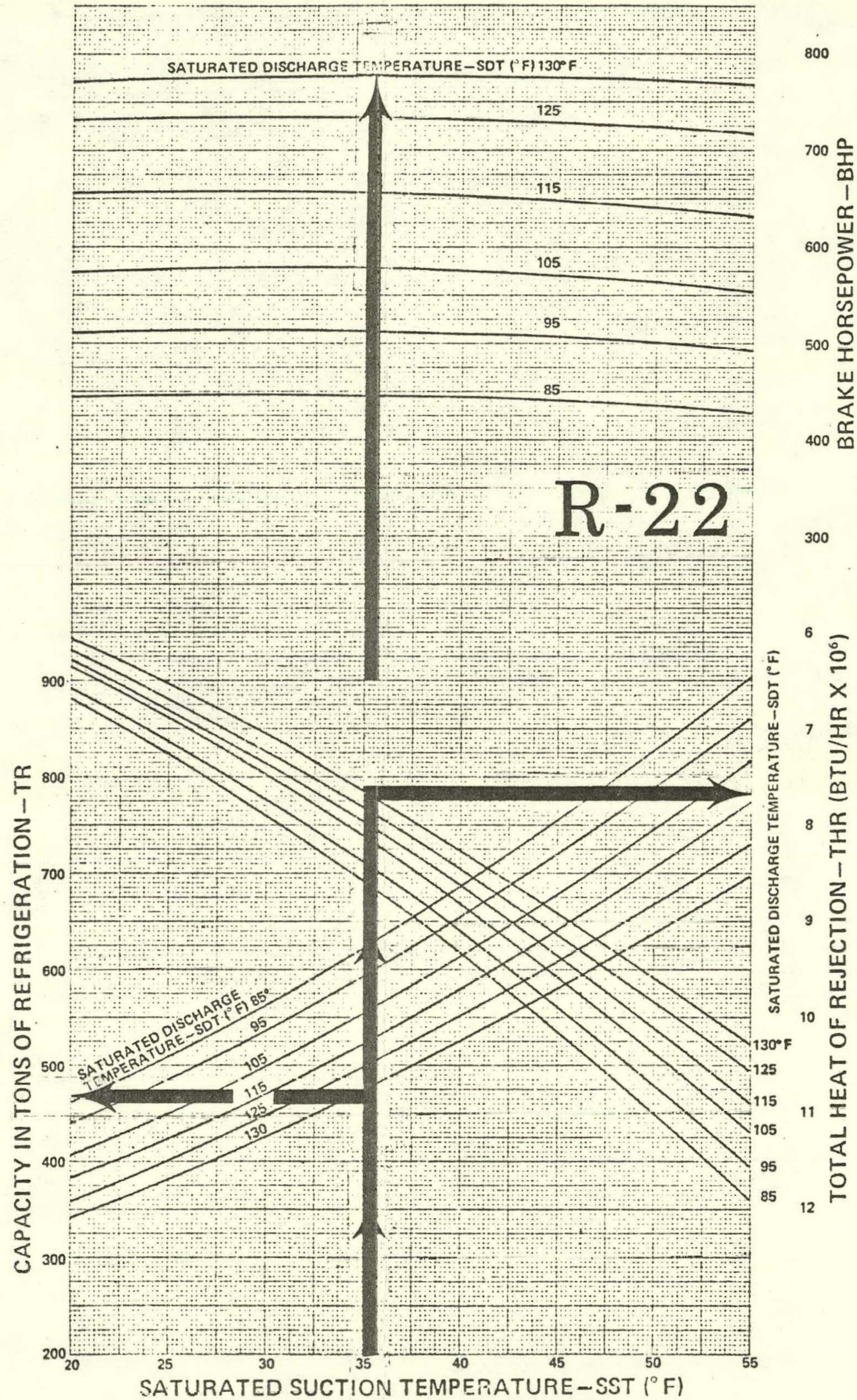
Select Dunham-Bush screw compressors, model F (800) hp x 2512

At 20°F SST and 130°F DST heat rejection is 6,069,000 Btu/hr, refrigerating effect 342 ton, power input 772 hp. Three units will satisfy heating and domestic hot water at 35°F SST and 130°F SDT. At these conditions the heat rejection is 7,672,000 BTU/hr, refrigeration effect 475 ton and power input 775 hp.



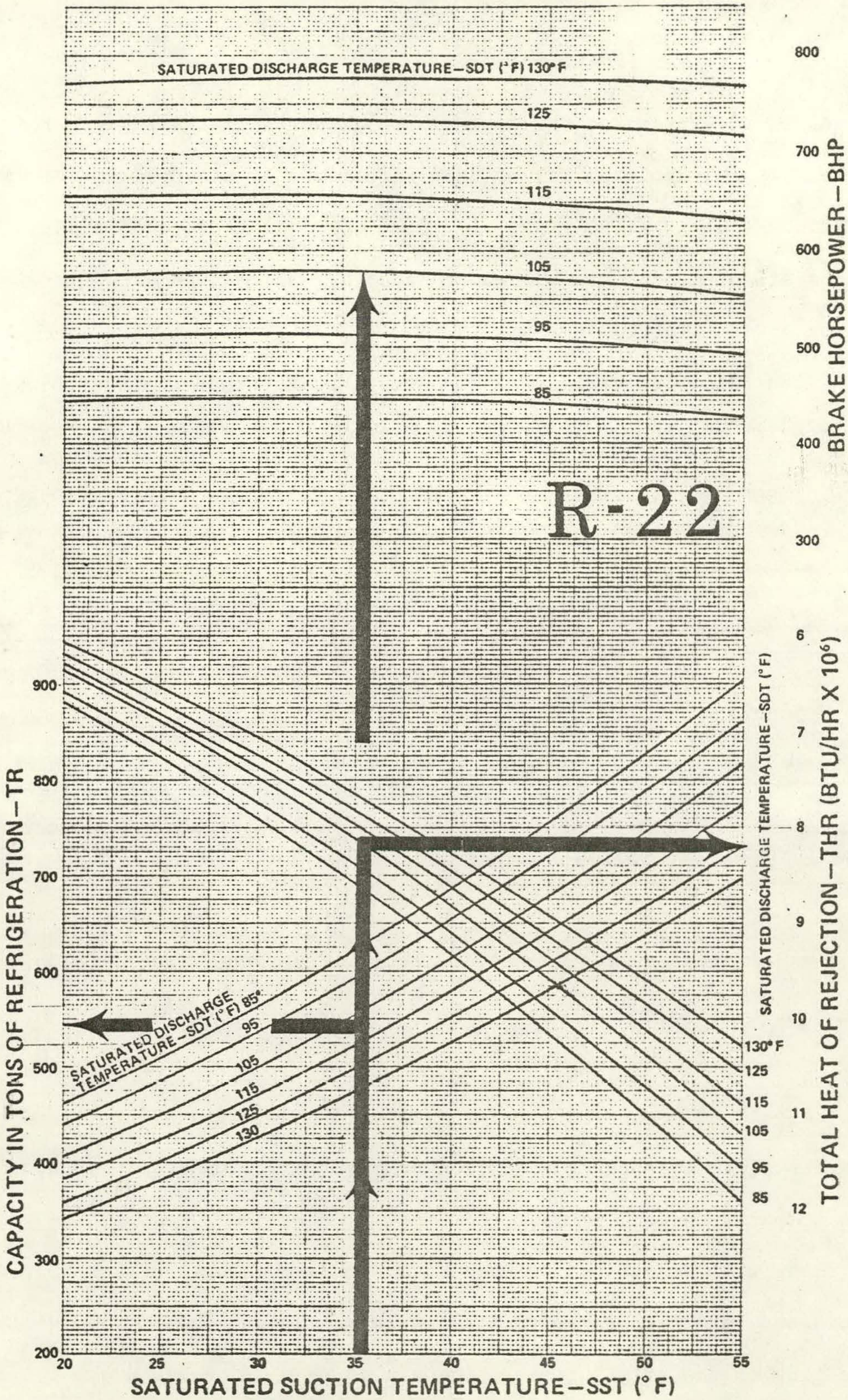
DUNHAM-BUSH SCREW COMPRESSOR E(HP)×2512

FIG. - A2.1 COMPRESSOR PERFORMANCE WINTER MODE



DUNHAM-BDUNHAM-BUSH SCREW COMPRESSOR E(HP)×2512

FIG.-A2.2 COMPRESSOR PERFORMANCE SUMMER DHW MODE



DUNHAM-BUSH SCREW COMPRESSOR E (HP) × 2512

FIG.-A2.3 COMPRESSOR PERFORMANCE SUMMER MODE

Also, to supplement the cooling requirements in the summer, the same compressors will again be called upon to perform at standard air conditioning conditions that is at 35^{o}F SST and 105^{o}F SDT. At these conditions, the cooling effect is 555 ton and power input 578 hp heat rejection by the cooling tower 8,131,000 BUT/hr.

The following table recapitulates the performance of E(800) x 2512 compressor at the stated conditions:

TABLE A 2.1 DUNHAM BUSH E(800) x 2512 SCREW COMPRESSOR PERFORMANCE

	SST = 20^{o}F SDT = 130^{o}F	SST = 35^{o}F SDT = 130^{o}F	SST = 35^{o}F SDT = 105^{o}F
Tonnage	342 ton	475 ton	555 ton
Power Input	772 hp	775 hp	578 hp
Heat Rejection	6,069,000 Btu/hr	7,672,000 Btu/hr	8,131,000 Btu/hr
hp/Ton	2.26	1.63	1.04
COP	3.09	3.89	----
COPC	2.09	2.89	4.53

The performance of the compressors is illustrated in the following charts (Figs. A 2.1, A 2.2 and A 2.3).

2.2.2 Ice Generating Equipment.

Ice will be generated by six units each consisting of 12 flat direct expansion plate evaporators and 12 expansion valves manufactured by the Turbo Company, Denton, Texas.

Water will be sprayed by evaporater pumps over the surface of the plates and frozen. The evaporaters will work in cycles and one of them will perform harvesting of ice at all times. The six units will generate a total of 648 tons of ice per day (24 hours).

2.2.3 Chillers

Three chillers will be selected of 555 ton capacity each to enable chilled water generation in the summer in conjunction with domestic hot water generation and also with heat rejection to the cooling towers.

2.2.4 Condensers

Condensers will be of the double bundle type capable of rejecting 8,131,000 BTU/hr in the summer while chilled water is generated. The temperature of the refrigerent at this condition should be 105°F, while the condenser water will be 95°F-85°F.

The condenser should be capable of operating also in the winter, rejecting 6,069,000 BTU/hr at a refrigerent temperature of 130°F, while the condenser water should be between 90° and 120°F.

2.2.5 Cooling Towers

Cooling towers will be selected for summer operation at night time. Three Marley cooling towers of 555 ton capacity each will be selected.

2.2.6 Pumps

The following table summarizes the selection of the Aurora pumps.

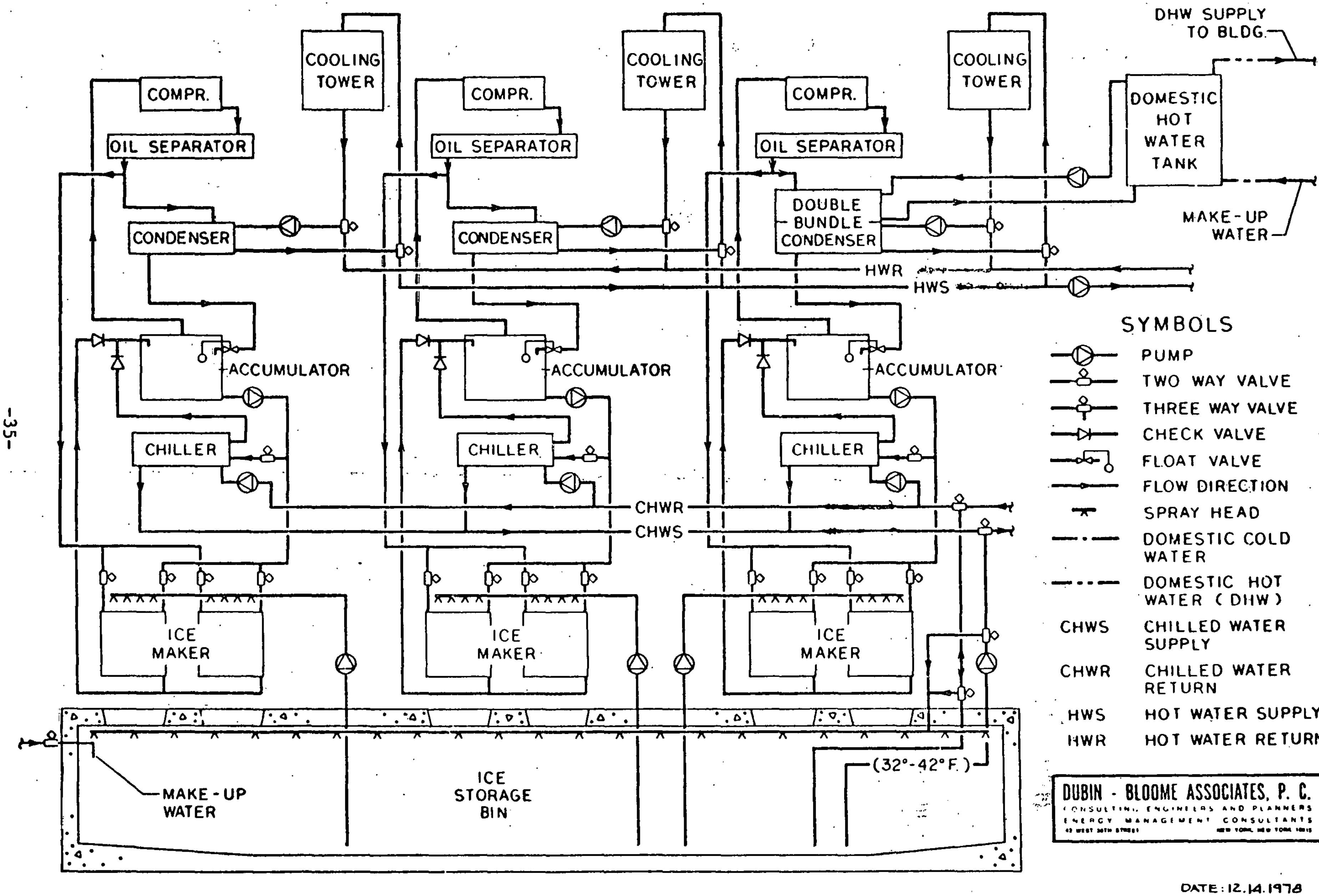
TABLE A-2.2 HP-ICES PUMP SELECTION

Designation	Service	GPM	Head Foot	HP	Efficiency %	Model & RPM
P-IA P-1B P-1C	Heating	367	150	22	68	3x4x14 Series 410 1750 RPM Double Suction
P-2A P-2B	Chilled water bin to primary loop	2250	110	70	89	8x10x1713 Series 410 1150 RPM Double Suction
P-3A P-3B P-3C	Chilled water chiller to bin	830	70	18	80	15x6x15 Series 410 1150 RPM Double Suction
P-4A P-4B P-4C	Condenser water cooling tower	1626	80	38	88	8x8x1113 Series 410 1750 RPM Double Suction

2.3 System Operation

Fig. A.2.4 illustrates the energy center for the Market Square complex. On the low pressure role of the refrigeration cycle there are chillers connected in parallel with the ice making evaporaters which enables the system to either generate ice or chilled water at a higher SST when simultaneous heating and cooling is required which results in a lesser power input. The control system will enable the three units to operate either in the same mode (ice or chilled water) or generating simultaneously ice and chilled water. However, the control system will minimize the generation and use of chilled water during the winter in order to produce as much ice

FIG.-2.4 ICE GENERATING HEAT PUMP ICES



HP-ICES PROJECT

as possible for the summer while in the winter the cooling requirements will be satisfied by enthalpy control.

The refrigeration cycle is designed to operate as a recirculating system allowing to connect alternately two different evaporaters with different flow rates to the same compressor preventing thereby liquid refrigerant from entering the compressor suction.

On the high pressure side of the refrigeration cycle there are double bundle condensers supplying either hot water for heating or condenser water to the cooling towers.

In the relatively mild climate of Washington, D. C., the ice generated by the heat pump during the winter is not sufficient to cover the summer cooling requirements.

The deficiency in "cooling" energy is made up in two ways. First, domestic hot water is generated in summer during night time to eliminate the electrical billing demand with simultaneous production of chilled water. The resulting COP is higher than for ice generation as the SST is 35°F while the SDT remains at 130°F. Storage tanks are provided to enable the use of the domestic hot water during the day. The remainder of the deficiency in "cooling " energy is produced by operating the chillers in conjunction with cooling towers. This is also done during the summer nights and results in a yet higher COP as the SDT is reduced to 105°F.

The chilled water, whether produced with domestic hot water or with cooling towers, is stored in one special compartment of the ice storage bin which is isolated from the remainder of the bin. The chilled water is used up the following day. The entire system will be integrated with the building to minimize additional load requirements. The ice bin will be located under the east side of the building at the lowest level and will occupy about 40% of the total boundary area of the complex. The three mechanical rooms which constitute the energy center will be located above the ice bin and all at the same level to enable the ice to drop directly into the bin below.

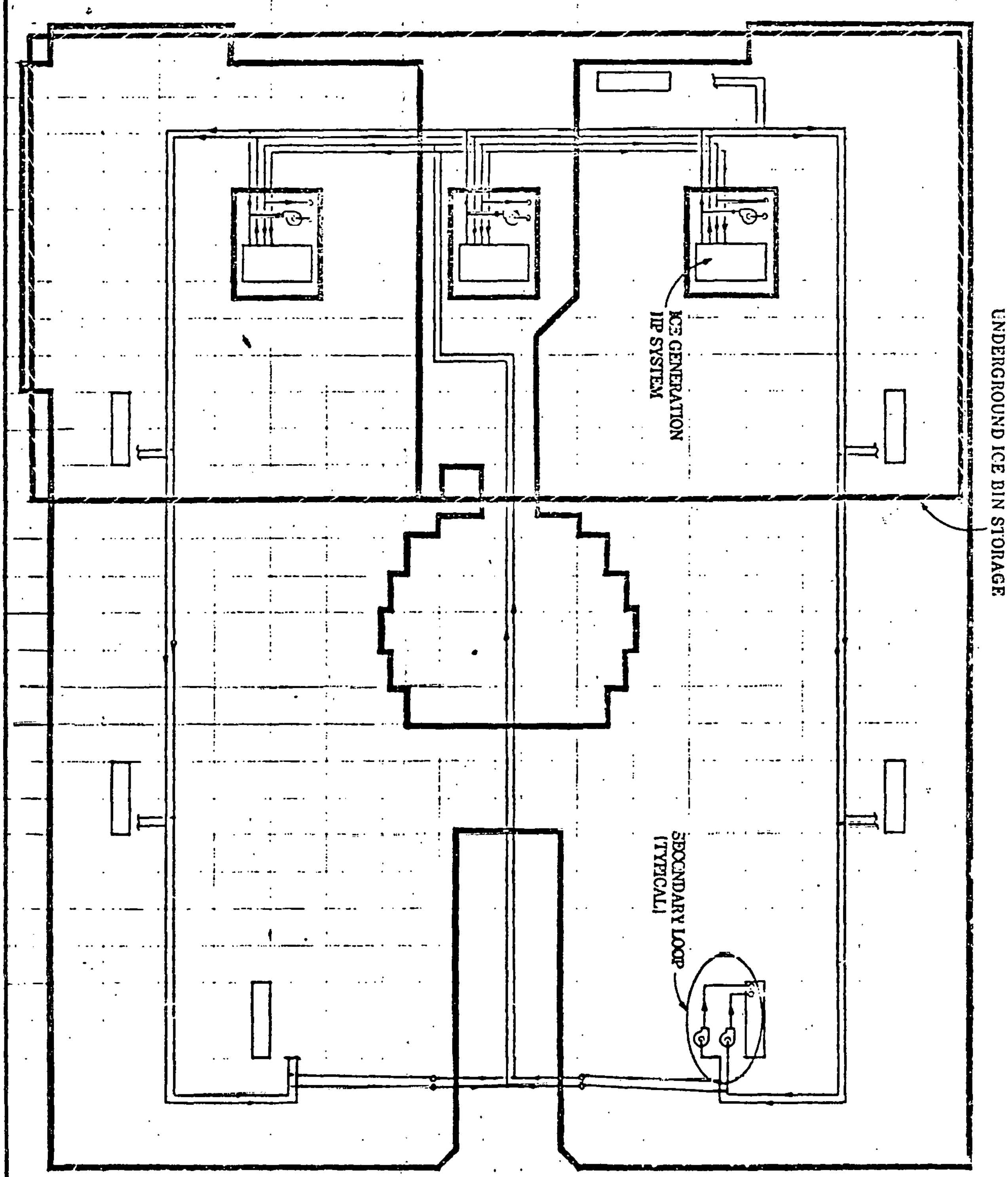
The chilled water distribution system will consist of a primary-secondary pumping arrangement with special pumps supplying chilled water from the bin to the primary and another set of pumps supplying chilled water from the chillers to the bin during summer operation.

The primary system will be on the same level as the mechanical rooms with the branches on the north and south sides with a common return. Secondary pumps will serve all the different buildings and sectors.

The temperature difference in the chilled water primary will be 16°F while that in the secondary loops will be determined by the end users enabling thus a considerable amount of flexibility.

A similar pumping arrangement will serve the heating distribution system, with a temperature difference in the primary of 30°F. The cooling towers will be located on the south-east roofs of the complex (See Figs. A2.5, A2.6, and A2.7).

FIG.- A2.5
MARKET SQUARE COMPLEX WASHINGTON, D.C.
HP-ICES SCHEMATIC DESIGN



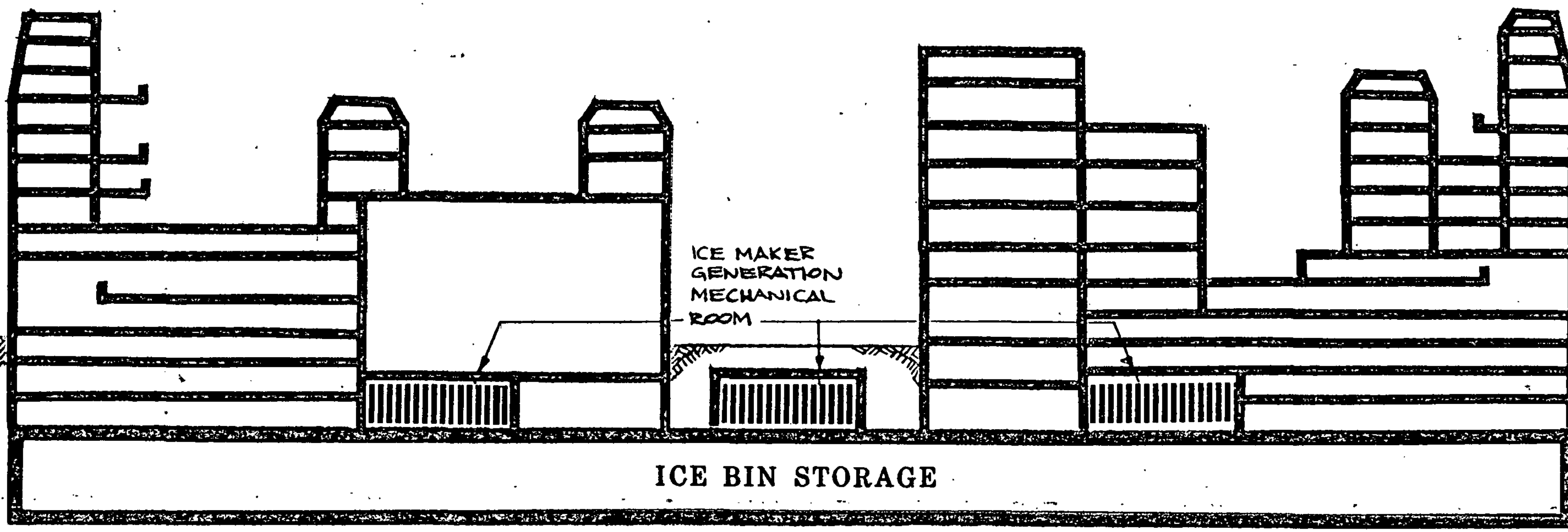


FIG. A2.6 MARKET SQUARE ICE BIN STORAGE
CROSS SECTION

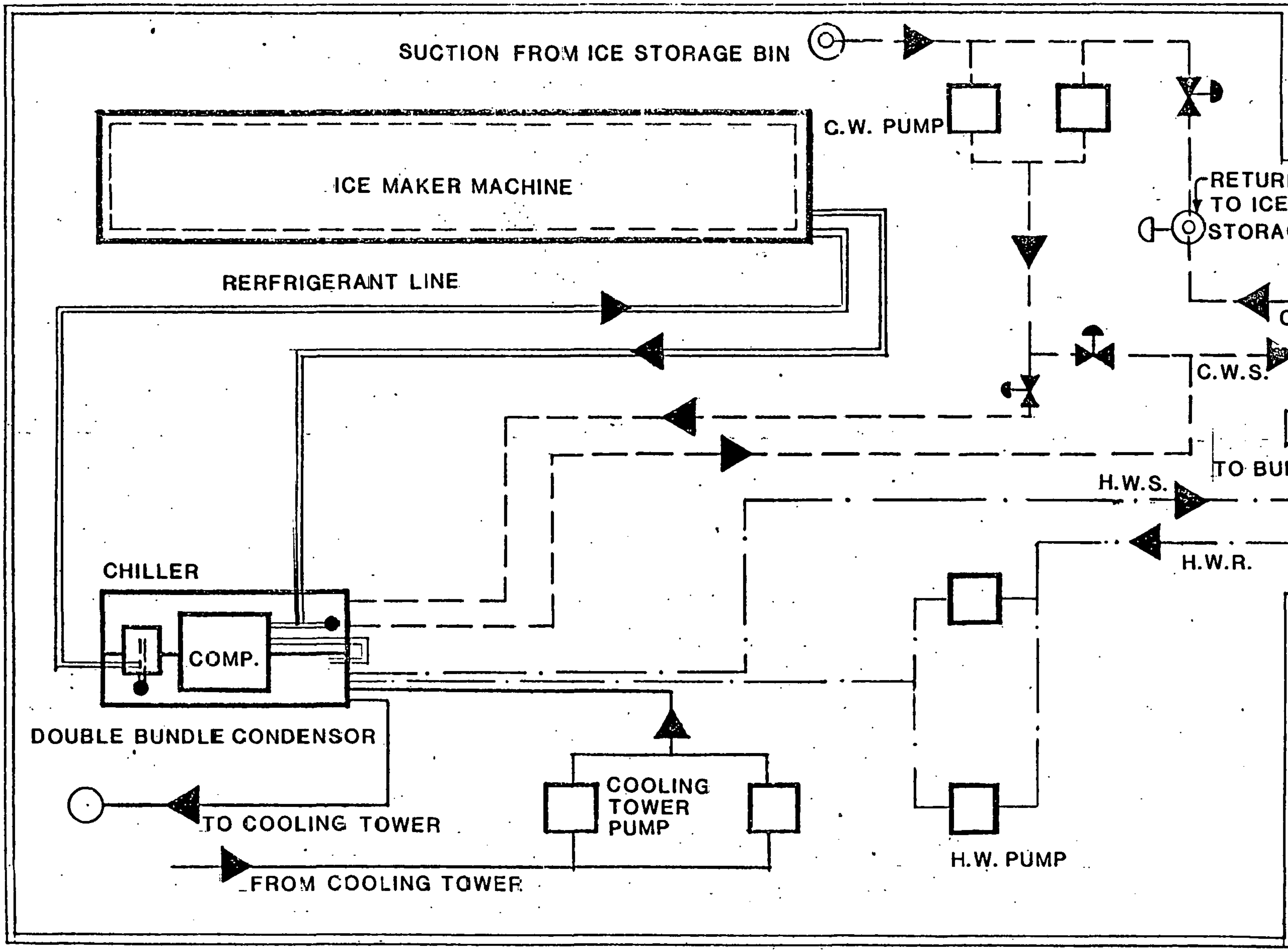


FIG. A.2.7 TYPICAL MECHANICAL ROOM FOR THE MARKET SQUARE

A.3 EXPECTED PERFORMANCE

3.1 General Performance

It is expected that the "cooling" energy generated during the winter in the form of ice, after deducting 8% losses due to heat leakage into the ice bin will be 18,878x10^6 BTU which is 52% of the summer cooling requirement of 36,108x10^6 BTU.

Domestic hot water generation accounts for 4380 x 10^6 BTU of "cooling" energy which is another 12% of the summer cooling requirement.

The remaining 36% of the summer cooling requirement, i.e. 12,850 x 10^6 BTU are supplemented by employing the compressor in conjunction with cooling towers and with chilled water generation at night time during August, September and October. The conditions of operation are 35°F SST and 105°F SDT which results in yet a higher COP.

Thus, the COP_H's of winter operation, summer DHW generation and summer operation with heat rejection to cooling towers are 3.09, 3.89 and 5.53 respectively.

The separate insulated compartment of the ice storage bin is so sized that it can supply a summer's peak day cooling requirement and can be charged during 9 hours of operation of all the three compressors operating at the highest COP which for cooling is 4.53.

The HP-ICES supplies 64% of "free" cooling while supplying all the heating and domestic hot water requirements for the Market Square complex.

Thus while supplying 36,239 X 10^6 BTU of heating and domestic hot water, the system also supplies 23, 258 x 10^6 BTU of useful "cooling" energy and 1,646 x 10^6 BTU of losses. All this at an energy expenduture of 3,320,000 kWh.

The overall annual COP is, therefore estimated at

$$COP_{overall} = \frac{(36,239 + 23,258) \times 10^6}{3,320,000 \times 3413} = 5.25$$

But taking into account also the cooling generated with heat rejection to the cooling towers the $COP_{overall} = \frac{(36,239 + 36,108) + 10^6}{4,151,000 \times 3413} = 5.11$

The section of Annual Energy Balance in this chapter will show the procedure followed in deriving the above cited figures and will illustrate the results with the help of tables and graphs.

3.2 ANNUAL ENERGY BALANCE

The first column of Table A-3.1 represents the annual energy requirements for heating and domestic hot water.

Thus, in November this requirement amounts to 4138 x 10^6 BTU and the heat pump generates 2799 x 10^6 BTU of cooling (ice). As there is no cooling requirement in November this "cooling" energy is disposed of in the ice bin.

This goes on until March with a maximum accumulation of 18,187 x 10^6 BTU after subtracting heat leakage into the bin. This maximum accumulation also determines the size of the ice bin.

In April there is already a cooling requirement of 1805 x 10^6 BTU which exceeds the cooling generated of 1790 x 10^6 BTU by 15 x 10^6 BTU and this deficiency comes out of storage.

The following month there is no ice generated but there is 901 x 10^6 BTU of cooling generated in the form of chilled water and to satisfy the cooling requirement of 3611 x 10^6 Btu, 2710 x 10^6 Btu are taken out of storage which together with 175 x 10^6 Btu losses (i.e. heat leakage into the storage) brings down the stored "cooling" energy to 15,113 x 10^6 Btu. A similar procedure is followed through June and July.

TABLE A-3.1 ANNUAL ENERGY BALANCE

MONTH	HEATING & DHW REQUIREMENT BTUx10^6	COOLING REQUIREMENT BTUx10^6	COOLING GENERATED BY HEAT PUMP - BTU x 10^6			STORAGE BALANCE BTU x 10^6			
			WINTER $COP_H=3.09$	SUMMER $COP_H=3.89$	SUMMER WITH COOLING TOWER $COP_C=4.53$	DEPOSIT (INTO STORAGE)	DEBIT (OUT OF STORAGE)	LOSSES	NET ACCUMULATION IN STORAGE END OF MONTH
Nov.	4138	0	2799	0	0	2799	0	0	2799
Dec.	6294	0	4257	0	0	4257	0	143	6913
Jan	6583	0	4453	0	0	4453	0	149	11217
Feb.	5795	0	3920	0	0	3920	0	134	15003
March	4887	0	3305	0	0	3305	0	121	18187*
April	2647	1805	1790	0	0	0	15	174	17998
May	1213	3611	0	901	0	0	2710	175	15113
June	666	4694	0	495	0	0	4199	237	10677
July	686	7944	0	510	0	0	7434	262	2981
Aug.	544	7944	0	404	4810	0	2730	251	0
Sept.	727	6860	0	540	6320	0	0	0	0
Oct.	2059	3250	0	1530	1720	0	0	0	0
Total	36239	36108	20524	4380	12850	18734	17073	1646	-

*Maximum volume of Ice Storage Bin = $\dfrac{18187 \times 10^6 \text{ BTU}}{144 + (50-32) \times 36} = 3118 \times 10^3$ cu. ft.

In August the losses of 251 x 10^6 Btu reduce the storage to 2730 x 10^6 Btu which is used up entirely together with 404 x 10^6 Btu of cooling in the form of chilled water, generated by the heat pump while generating domestic hot water. But, there is still a demand of 4810 x 10^6 Btu and this is supplemented by operating the screw compressor at conventional conditions of 35°F SST and 105°F SDT and with heat rejection to cooling towers. This will be done at night time with daily storage to eliminate demand.

Throughout September and October the compressor similarly operating in conjunction with cooling tower supplements the cooling generated in the form of chilled water by the heat pump while supplying some heating and generating domestic hot water.

Fig. A-3.4 depicts the annual energy balance. The upper side of the bargraph shows the heating and domestic hot water requirements, while below the abscissa axis the "cooling" energy generated at various times of the year. The thick contour lines depicts the actual cooling requirements.

The "cooling" energy generated during the winter (lower left) will cover the blank white area under the contour and the storage losses.

The winter cooling load shown in Fig. A-1.9 is satisfied using outside air with enthalpy controls and is therefore not included in the energy balance of the HP-ICES system in Fig. A-3.1.

3.3 DETERMINATION OF THE ICE STORAGE BIN VOLUME

From Table A 3.1 the maximum accumulation of "cooling" energy occurs in March and this energy will determine the size of the ice storage bin. To eliminate the electrical demand over the entire summer and to operate the heat

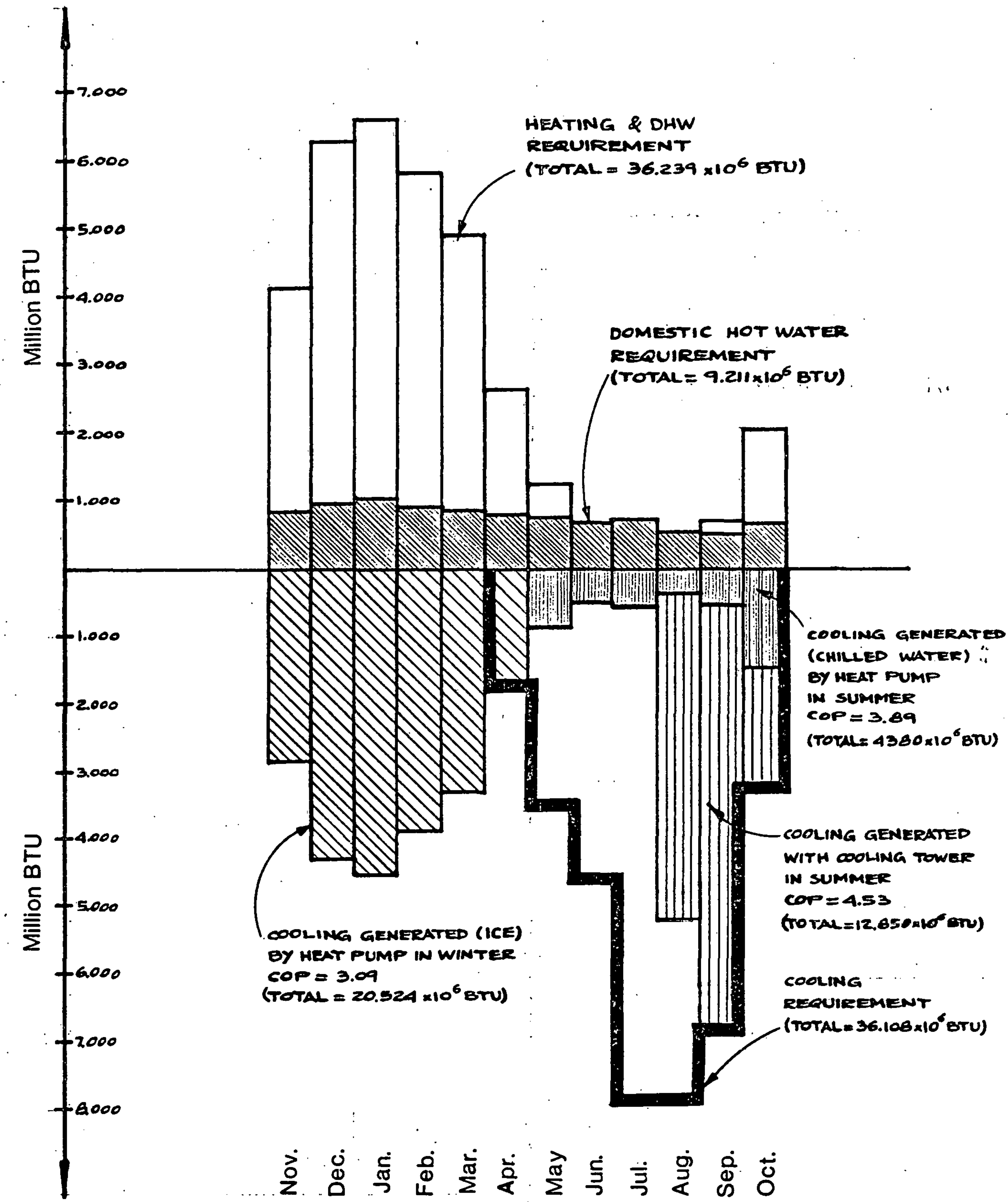


FIG.-A3.1
HP-ICES Annual Energy Balance

pump at higher COP's while generating chilled water, a daily chilled water storage is required. The chilled water storage will occupy a relatively small part of the ice stroage bin but will be separated from the remainder of the bin and insulated.

This arrangement will enable us the generation of domestic hot water at night time while using 56°F water after the ice in the compartment has melted as a source for the chiller and operating at a higher COP at a SST = 35°F and SDT = 130°F.

In August, September and October conventional operation at SST = 35°F and SDT = 105°F will be required with heat rejection to the cooling towers.

The volume of the daily chilled water storage should be able to accommodate the cooling required for a summer peak day which is

$$3000 \text{ ton} \times 5 \text{ EFLH} = 15000 \text{ ton-hours}$$

The volume of the water is

$$\frac{15000 \times 12000}{16 \times 62.4} = 180{,}300 \text{ cu. ft.}$$

To accommodate the ice this volume has to increase in the ratio of the densities of water to ice, i.e. $\frac{62.4}{36.0} = 1.73$

Thus, the volume of the chilled water compartment is

$$180{,}300 \times \frac{62.4}{36.0} = 312{,}500 \text{ cu. ft.}$$

* The density of ice is 56 lb./cu.ft. According to Stal Ice Making Equipment Company, the packing fraction of ice results in an ice-water-air mixture density of 36 lb./cu.ft.

The cooling effect of this ice is: $312{,}500 \times 36 \times 162 = 1822.5 \times 10^6$ Btu This is about 50% of the cooling requirement for May and this means that by the middle of May we already have available 180,000 cu. ft. of water at 52°F which can be cooled by the chiller (Note: Had we not used a separate compartment in the ice bin we would have to wait until sometime in July when all the ice melted and we would then have ice water at 32°F, not suitable to put through chiller).

The above arrangement will enable the operation of the chiller and storage through May, June and July as a heat source for DHW generation while in August, September and October we use the chilled water daily storage.

The ice storage bin will be approximately 400 ft x 400 ft and 20 ft deep. The top of the bin will be insulated by 4 inch polyurethane (R = 36.4) and 2 inch polyurethane (R = 18.2) on the sides and bottom. This will result in only 8% heat leakage losses in the bin which have been taken into account in the energy balance. The losses have been broken down by month (The calculations are based on results obtained in the First Report).

To satisfy the load on a summer peak day three compressors will have to operate during night time for nine hours at full load at 35°F SST and 105°F SDT at which conditions the cooling output will be 555 tons, yielding 15000 ton-hours.

3.4 ANNUAL ENERGY CONSUMPTION

Table A 3.2 shows the energy consumption for the compressor by month for three main modes of operation and also the total consumption by month.

Table A 3.3 shows the energy consumption by month for the electrically driven auxiliaries and Table A 3.4 shows the total consumption both for compressors and auxiliaries. This table is depicted by a bar graph in Fig. A 3.2.

TABLE A 3.2 ANNUAL ELECTRIC ENERGY CONSUMPTION FOR HEAT PUMP COMPRESSOR

MONTH	HEATING GENERATED BTU x 10^6	COOLING GENERATED BTU x 10^6			ELECTRIC ENERGY CONSUMPTION - KWH			
		COP_H=3.09	COP_H=3.89	COP_C=4.53	COP_H=3.09	COP_H=3.89	COP_C=4.53	TOTAL
November	4138	2799	0	0	392,000	-	-	392,000
December	6294	4257	0	0	597,000	-	-	597,000
January	6583	4453	0	0	624,000	-	-	624,000
February	5795	3920	0	0	549,000	-	-	549,000
March	4887	3305	0	0	463,000	-	-	463,000
April	2647	1790	0	0	251,000	-	-	251,000
May	1213	0	901	0	-	91,000	-	91,000
June	666	0	495	0	-	50,000	-	50,000
July	686	0	510	0	-	52,000	-	52,000
August	544	0	404	4810	-	41,000	311,000	352,000
September	727	0	540	6320	-	55,000	409,000	464,000
October	2053	0	1530	1720	-	155,000	111,000	266,000
Annual	36239	20524	4380	12850	2,876,000	444,000	831,000	4,151,000

TABLE A 3. 3 ANNUAL ELECTRIC ENERGY COMSUMPTION
For Auxiliaries - KWH

Month	Heating Pumps	Domestic Hot Water	Bin to Primary (Chilled W.)	Chiller to Bin	Condenser Water Pump & Cooling Tower Fans	Ice Maker Pump*	TOTAL
Nov.	17,483	2278	-	-	-	1736	21497
Dec.	28,078	2669	-	-	-	2788	33535
Jan.	29,286	2822	-	-	-	2908	35016
Feb.	25,660	2567	-	-	-	2548	30775
March	21,068	2465	-	--	-	2095	25628
April	9,708	2244	14080	-	-	964	26996
May	2,498	1632	21120	2212	-	248	27710
June	0	1292	31680	1218	-	-	34190
July	0	1343	31680	1246	-	-	34269
Aug.	0	1054	31680	11074	33840	-	77648
Sept.	1,128	1003	31680	14602	44556	113	93082
Oct.	7,291	1309	14080	7364	12126	724	42894
Total	142,200	22678	176000	37716	90522	14124	483240

* 2 pumps 5 hp or 4 kW, same number of hours as heating pumps.

The maximum energy consumption occurs during December, January and February, while the minimum occurs during May, June and July.

TABLE A 3.4 ELECTRICAL ENERGY CONSUMPTION

MONTH	ELECTRICAL ENERGY CONSUMPTION KWH
January	659016
February	579775
March	488628
April	277996
May	118710
June	84190
July	86269
August	429648
September	556969
October	308170
November	413497
December	630535
Total Annual	4633403

Winter (November through April)	=	3,049,447
Summer (May through October)	=	1,583,956
		4,633,403

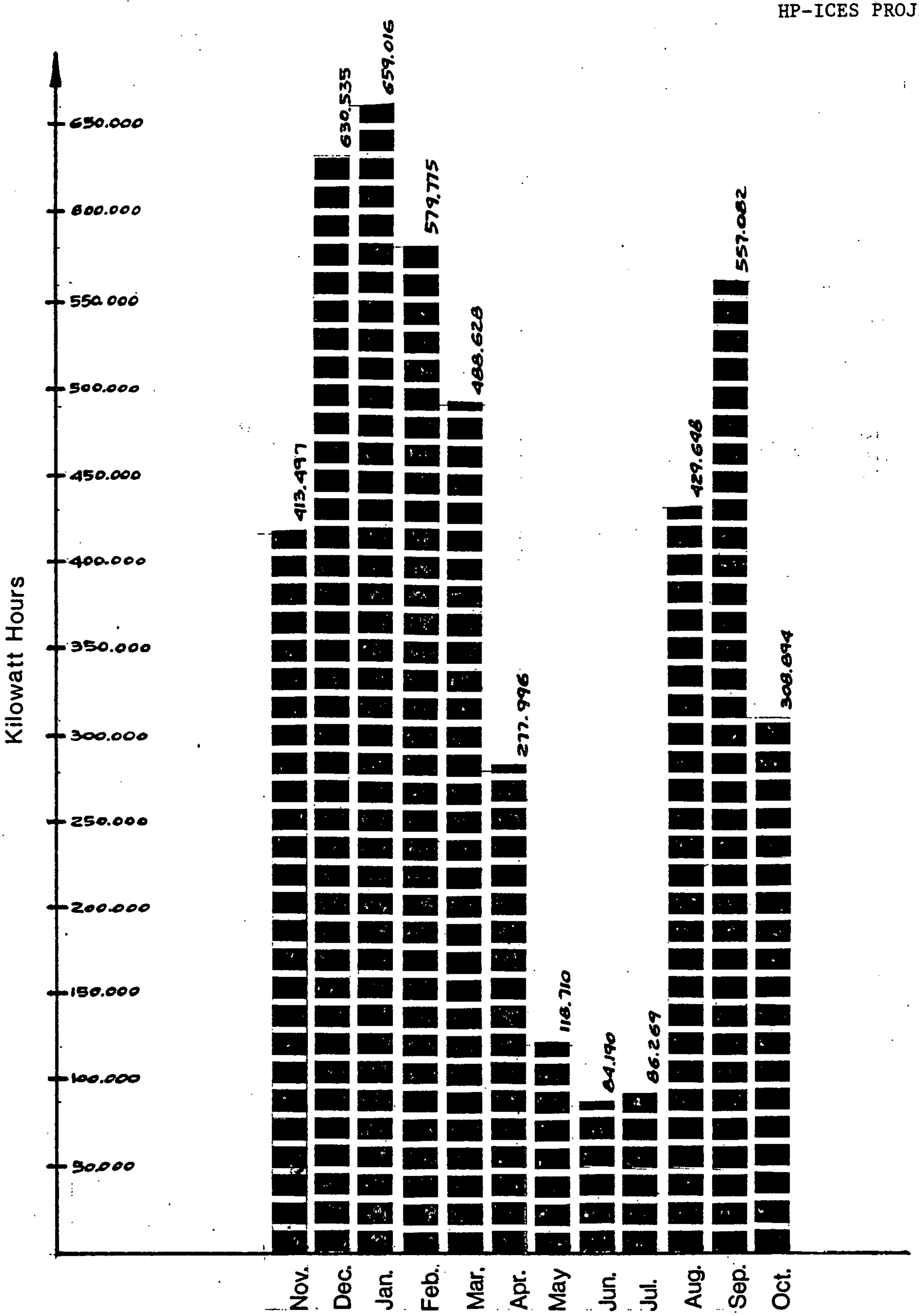


FIG.-A3.2 HP-ICES Annual Electric Energy Consumption

3.5 RESOURCES UTILIZATION

Table A 3.5 was completed according to ASHRAE 90-75, Section 12 requirements. The annual fuel and energy resources determination was made according to Table 12-4 in the above Section, for Washington, D .C. South Atlantic Retion. (See Table A.3.5)

Table A 3.5 HP-ICES

ANNUAL FUEL AND ENERGY CALCULATION FORM 12-2

Line	Annual Fuel and Energy Calculation Form 12-2 — Fuel and Energy Supplied to Site			C.O. Total From Form 12-1 Line 13	RUF From Supplier or From Tables	C1 S. Tons Coal	C2 MCF Nat'l	C3 BBL Crude Oil	C4 Grams U-235	C5 10³ KWH Hydro	C6 Other	C7 Other
14	Fuel Oil, Light											
15	Fuel Oil, Heavy											
16	Gas	Nat'l	MCF									
		Oil	BBL									
17	Coal											
18	Elec. Winter			3049.45								
		Coal	S. Tons		0.27	823.35						
		Gas	MCF		0.56		1707.69					
		Oil	BBL		0.53			1616.21				
		Nuc	Grams		3.84				11709.89			
		Hydro	10³ KWH		0.07					213.46		
		Other										
19	Elec. Summer			1583.96								
		Coal	S. Tons		0.26	411.83						
		Gas	MCF		1.04		1647.32					
		Oil	BBL		0.56			887.02				
		Nuc	Grams		3.84				6082.41			
		Hydro	10³ KWH		0.04					63.36		
		Other										
20	Elec. Annual											
		Coal	S. Tons									
		Gas	MCF									
		Oil	BBL									
		Nuc	Grams									
		Hydro	10³ KWH									
		Other										
21	(Other)											
22					Total Resources	1235.18	3355.01	2503.23	17792.30	276.82		

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3.6 SELECTION OF A COMPARATIVE CONVENTIONAL SYSTEM.

3.6.1 General Description

A conventional system of sufficient capacity to satisfy the heating and cooling demands was selected to serve as a basis of comparison with the HP-ICES system.

The chillers will operate in conjunction with a primary-secondary pumping arrangement with a temperature differential in the primary of $56^{\circ}-40^{\circ} = 16^{\circ}F$.

For heating, the primary loop will operate at temperatures $240^{\circ}F-160^{\circ}F$. The heating pump was selected to deliver:

$$\frac{13,000,000}{500 \times 80} = 326 \text{ gpm}$$

at a head of 125 ft.

The energy requirement for heating and domestic hot water is $36,234 \times 10^6$ Btu.

Fuel consumption $$\frac{36234 \times 10^6}{0.7 \times 1000 \times 1000} = 51,770 \text{ M cu.ft. of gas.}$$

Pumping power for heating pumps is calculated by assuming

$$\text{EFLH} \quad \frac{36,234 \times 10^6 \text{ BTU}}{16,000,000 \text{ BTU/hr}} = 2265 \text{ hours}$$

$$2265 \times 1.7 = 3850 \text{ hours for auxiliaries}$$

$$15 \times 0.746 \times 3850 = 43,082 \text{ kWh}$$

See Fig. A 3.3 for schematic diagram for the conventional system.

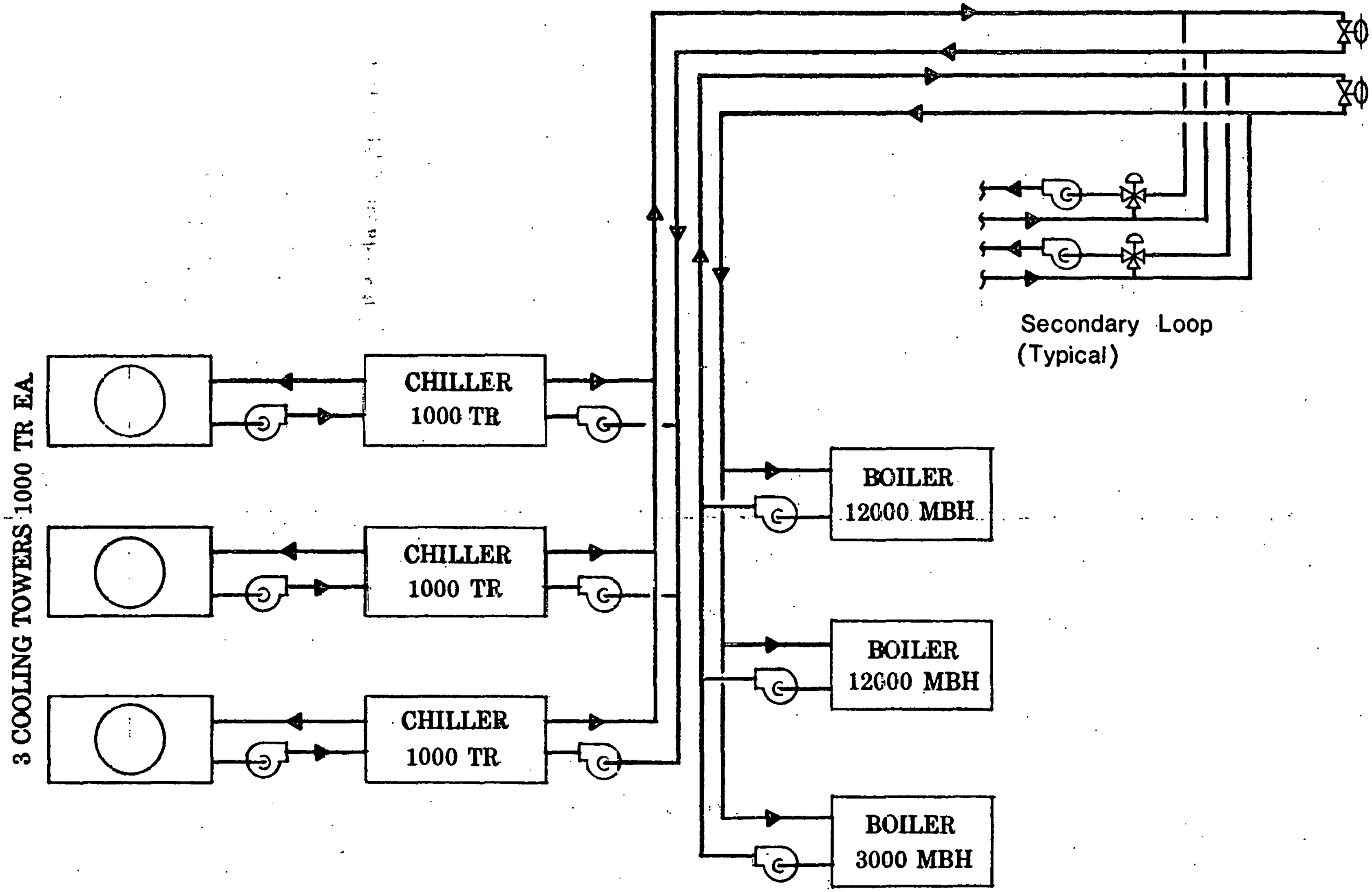

FIG. 3.3
MARKET SQUARE COMPLEX
WASHINGTON, D.C.
CONVENTIONAL SYSTEM
SCHEMATIC DESIGN

3.6.2 Equipment Selection

a. Chillers

Carrier centrifugal packaged hermetic, three units 1000 ton each, 772 KW input, 35°F SST, 105°F SDT model 19 EB 8983 DP.

b. Cooling Towers

Three Marley, 1000 ton each, 60 hp fan each

c. Boilers

Two 12,000,000 BUT/hr Cleaver Brooks hot water boilers, one 3,000,000 BTU/hr for domestic hot water.

d. Pumps

TABLE A 3.6 PUMPS FOR CONVENTIONAL SYSTEM

DESIGNATION	SERVICE	GPM	HEAD FEET	HP	EFFICIENCY	MODEL
P-1A P-1B P-1C	Chilled Water	1500	110	50	0.75	Aurora 8x10x15A Series 410
P-2A P-2B P-2C	Condenser Water	3000	80	75	0.82	Aurora 10x12x15C Series 410
P-3A P-3B P-36	Heating	326	125	15	0.68	Aurora 3x4x12 Series 360

See Tables A.3.7 and A.3.8 for energy consumption for the refrigeration compressors and refrigeration auxiliaries respectively.

3.7 RESOURCES UTILIZATION COMPARISON

For the resources utilization of the conventional system, see Table A 3.9. The bottom lines of Table A 3.5 and Table A 3.9 are compared in Table A 3.10 which shows that HP-ICES uses more crude oil, more hydroelectric power and more coal and Uranium (U-235) than the conventional system, but significantly less natural gas. It is difficult to see the improvement in

TABLE A 3.7 ENERGY CONSUMPTION FOR REFRIGERATION COMPRESSORS
(FOR THE CONVENTIONAL SYSTEM)

Month	Cooling Requirements Btu x 10^6	CONSUMPTION			DEMAND			
		%	KWH	Cost $	%	KW	Cost $	Total Cost &
January	0	0	0	0	0	0	0	0
February	0	0	0	0	0	0	0	0
March	0	0	0	0	0	0	0	0
April	1805	5	116122	1144	25	579	2,327	3,471
May	3611	10	232308	2288	50	1158	4,655	6,943
June	4694	13	301981	4279	100	2316	11,997	16,276
July	7944	22	511064	7242	100	2316	11,997	19,239
August	7944	22	511064	7242	100	2316	11,997	19,239
September	6860	19	441327	6254	50	1158	5,998	12,252
October	3250	9	209083	2963	25	579	3,000	5,963
November	0	0	0	0	0	0	0	0
December	0	0	0	0	0	0	0	0
Total Annual	36108	100	2,332,949	31,412			51,971	83,383

TABLE A-3. 8
ENERGY CONSUMPTION FOR ELECTRICALLY DRIVEN REFRIGERATION AUXILIARIES

Month	CONSUMPTION			DEMAND			
	%	KWH	Cost $	%	KW*	Cost $	
January	0	0	0	0	0	0	0
February	0	0	0	0	0	0	0
March	0	0	0	0	0	0	0
April	8	52893	521	50	207	832	1,353
May	12	79339	781	100	414	1,664	2,445
June	18	119008	1,686	100	414	2,145	3,831
July	18	119008	1,686	100	414	2,145	3,831
August	18	119008	1,686	100	414	2,145	3,831
September	18	119008	1,686	100	414	2,145	3,831
October	8	52893	750	50	207	1,072	1,822
November	0	0	0	0	0	0	0
December	0	0	0	0	0	0	0
Total	100	661,158**	8,796			12,148	20,944

*3 chilled water pumps = 150 hp
3 cond. water pumps = 225 hp
3 cooling tower fans = 180 hp
555 hp

$$\text{Auxilliaries Power Ratio} = \frac{414 \text{ kW}}{3000 \text{ ton}} = 0.138 \text{ kW/ton}$$

**Full load hours = 939 x 1.7 = 1597 Hr

kWh = 1597 x 414 = 661,158 KW

TABLE A 3.9 CONVENTIONAL SYSTEM

ANNUAL FUEL AND ENERGY CALCULATION FORM 12-2

Annual Fuel and Energy Calculation Form 12-2				C.O. Total	RUF From	Fuel and Energy Resources Used on Site and Off Site To Meet Energy Requirements of Building/Project						
Line	Fuel and Energy Supplied to Site			From Form 12-1 Line 13	Supplier or From Tables	C1 S. Tons Coal	C2 MCF Nat'l	C3 BBL Crude Oil	C4 Grams U-235	C5 10^3 KWH Hydro	C6 Other	C7 Other
14	Fuel Oil, Light											
15	Fuel Oil, Heavy											
16	Gas	Nat'l	MCF		1.16		60053.2					
		Oil	BBL		0.0053			247.38				
17	Coal											
18	Elec. Winter			43.082								
		Coal	S. Tons		0.27	11.63						
		Gas	MCF		0.56		24.13					
		Oil	BBL		0.53			22.83				
		Nuc	Grams		3.84				165.43			
		Hydro	10^3 KWH		0.07					3.02		
		Other										
19	Elec. Summer			2994.11								
		Coal	S. Tons		0.26	778.47						
		Gas	MCF		1.04		3113.87					
		Oil	BBL		0.56			1676.70				
		Nuc	Grams		3.84				11497.38			
		Hydro	10^3 KWH		0.04					119.76		
		Other										
20	Elec. Annual											
		Coal	S. Tons									
		Gas	MCF									
		Oil	BBL									
		Nuc	Grams									
		Hydro	10^3 KWH									
		Other										
21	(Other)											
22					Total Resources	790.10	63191.20	1973.91	11662.81	122.78		

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energy use from this table, therefore, an energy source comparison will be used with a heat rate of 11,000 BTU/kWh $\left(\frac{3413}{11000} = 0.31\right)$

TABLE A.3.10 ENERGY RESOURCES COMPARATIVE ANALYSIS

ANNUAL FUEL AND ENERGY RESOURCES ON SITE AND OFF SITE TO MEET ENERGY REQUIREMENTS OF THE PROJECT					
	C1	C2	C3	C4	C5
System	S Tons Coal	MCF Nat'l	BBL Crude Oil	Grams U-235	10^3 KWH Hydro
Conventional	790.10	63191.20	1973.91	11662.81	122.78
HP-ICES	1235.18	3355.01	2503.23	17792.30	276.82

The first column of tables A.3.5 and A.3.9 were used to compile Table A.3.11.

Thus from Table A.3.7 the kWh used for the HP-ICES were 3,049,450 plus 1,583,960 which adds up to 4,633,410 kWh. Thus

$$\frac{4,633,410 \times 3413}{0.31} = 51,012, \times 10^6 \text{ Btu which}$$

appears in Table A.3.11.

Similarly from Table A.3. 9, first column for the conventional system 51,770,000 cu. ft. of gas yields 51,770 x 10^6 Btu while 2,994,110 kWh and 43,082 kWh yield

$$\frac{3037,192 \times 3413}{0.31} = 33,439 \times 10^6 \text{ Btu}$$

Thus the total Btu's are $(51{,}770 + 33{,}439) \times 10^6$

$$= 85{,}209 \times 10^6 \text{ Btu}$$

The raw source energy comparison is given in Table A.3.11 below.

TABLE A.3.11 ANNUAL RAW SOURCE ENERGY UTILIZATION COMPARISON

System	Total Energy Used Btu x 10^6
Conventional	85,209
HP-ICES	51,012

For First and Second Law analyses see Section C.

A.4 EXPECTED ECONOMICS FOR THE MARKET SQUARE PROJECT

4.1 INTRODUCTION AND ASSUMPTIONS

The method used here for evaluating and comparing the economics of the HP-ICES system and a conventional system of the same heating and cooling capacity will be a life cycle cost analysis.

The following are the assumptions made in evaluating the economics.

(a) The Conventional System:

The conventional system will be a central system with conventional centrifugal chillers and gas-fired boilers. Although the methods of generation of heating and cooling differ in the two systems, the distribution systems were assumed to be the same.

(b) The First Cost:

The first cost of each system includes the generation equipment and distribution system, but does not include the end-users equipment (air handling units) or the cost of land.

The reason for this is that the main object of the economic evaluation is to compare the HP-ICES system with a conventional system and examine the relative costs.

The cost of the end-users equipment can vary greatly, depending on details of the system. However, it would be about the same for the HP-ICES system and the conventional system.

The HP-ICES system requires a large amount of space for the storage. Because of the difficulty in getting large areas of land in this project, it will be necessary to use underground bins which do not take up additional land area outside the building. Therefore, with proper design, the main difference in land use between the HP-ICES and conventional systems will not be in the land area used but in the cost of excavating the land for the ice bin and in the institutional considerations involved in pipes crossing roads, etc.

In the analysis, the cost of excavating the land, constructing the ice bin, etc. is taken into account. It is assumed that the area of the land used will be about the same for the two systems, and that the large storage will be under the buildings and will not take up additional land area.

(c) Equipment Prices:

These were obtained from the following manufacturers: Compressors from Dunham-Bush and Vilters; ice makers from Trubo Co., Texas, and Stal Co., Sweden; boilers from Cleaver-Brooks Co., and pumps from Aurora Co.

(d) Energy Prices:

Energy prices were calculated according to Potomac Electric Power Company Service schedule DC-GS,

Billing Demand	November - May	\$4.02/kW
	June - October	\$5.18/kW
Energy Charge	November - May	\$0.00985/kWh
	June - October	\$0.01417/kWh

Gas prices were those of Washington Gas Light Co. - \$0.303/therm. Details of the energy costs will be given later.

(e) System Life, Mortgage Period, Salvage Value, etc.

The life of the equipment is assumed to be 20 years and the life of the distribution system and ice storage bin 40 years. The mortgage period and the period of the economic analyses are both assumed to be 20 years. The depreciation life of the overall system (for the purpose of tax deductions) is assumed to be 20 years. For certain kinds of ownership of the system, it may be possible to use a smaller depreciation period and increase the tax savings. However, the results will be given here only for a depreciation life of 20 years.

The salvage value of the equipment after 20 years will be assumed to be 20% of the first cost. For the ice bin in the HP-ICES system, the salvage value will be assumed to be 40% of the initial cost of the ice bin (i.e. the cost of excavation and ice bin construction). The reason for assuming this high value is that after the useful life of the HP-ICES system, it is possible to put the bin to good use by converting it into a parking lot without too large an additional expense. For the complete HP-ICES system, including the equipment and the storage, the overall percent salvage value is found to be about 30%.

(f) Cost of Insurance, Maintenance, Labor, Administration, etc.

Insurance costs are assumed to be about 0.5% of the first cost. The costs of maintenance, labor, administration, etc. are estimated for each project separately.

(g) Discount Rate and Inflation Rate

The general inflation rate is assumed to be an average of 6% per year. The discount rate used in evaluating the present worth of money is assumed to be 2% more than the inflation rate, i.e. 8% per year.

(h) Taxes

The effective income tax bracket for the commercial system, assuming ownership by a private corporation, is assumed to be 46%. The tax deductions that a business can make are all allowed for.

The property tax is assumed to be 1% per year.

4.2 DETAILS OF COST ESTIMATES FOR THE HP-ICES SYSTEM

The following are the details of the cost estimates for the HP-ICES System.

a. <u>First Cost for the HP-ICES System</u>

1.	Ice Bin Excavation (150,000 cu.yard @ $0.50/cu.yd.)	=$ 75,000
2.	Ice Bin Construction (3,200,000 cu.ft. @ $0.50/cu.yd. incremental cost; ice bin is part of the underground structure)	= 1,600,000
	Ice Bin Sub-Total	$1,675,000
3.	Three Dunham-Bush screw compressors including condensers, chillers, oil separators and fluid accumulators (342 ton at 20°F SST and 130°F SDT)	= 300,000
4.	Six Ice Makers of total capacity 640 ton ice $\left(\frac{3 \times 342}{1.6} = 640\right)$	= 600,000
5.	Three Cooling Towers (of 600 ton each)	= 75,000
6.	Pumps	
	Three Pumps used for heating	= 7,500
	Two Chilled Water Pumps (Bin to Primary)	= 14,500
	Three (Chiller to Bin) Pumps	= 10,200
	Three Condenser Water Pumps	= 13,500
7.	Chilled Water Distribution System (Primary and Secondary based on 20,000 linear feet of 6" dia. pipe)	= 800,000
8.	Hot Water Distribution System (Based on 20,000 linear feet of 3" dia. pipe).	= 400,000
9.	Electrical Installation and Connection (2054 kW)	= 238,050
	Sub-Total First Cost	=$4,133,750
10.	Contingency 10%	= 413,380
	Total First Cost	=$4,547,130

b. Operating Cost for HP-ICES System

1.	Energy Cost	=$	100,568
2.	Maintenance (1% of Equipment First Cost)	=	30,000
3.	Labor, 8 station Engineers (at $20,000 each)	=	160,000
4.	Administration, 6 people (at $15,000 each)	=	90,000
	TOTAL OPERATING COST	$	380,568

Details of the Energy Cost, given above as part of the Operating Cost, are as follows:

Details of the Energy Cost:

1.	Compressors (see Table A 3.2)	=$	88,986
2.	Pumps used for heating	=	2,796
3.	Bin to Primary Loop Pumps	=	5,570
4.	Chiller to Bin Pumps	=	525
5.	Condenser Water Pump & Cooling Tower Fans	=	1,283
6.	Domestic Hot Water Pump	=	1,164
7.	Ice Maker Pumps (4 kW Demand charge during seven months and 132 kWh)	=	244
	Total Energy Cost	=$	100,568

(Note: For items 2 through 6 above, see Appendix I)

4.3 DETAILS OF COST ESTIMATES FOR THE CONVENTIONAL SYSTEM

The following are the details of the cost estimates for a conventional system of the same heating and cooling capacity as the HP-ICES system.

a. First Cost for Conventional System

1.	Three (1,000 ton each) conventional centrifugal chillers, Carrier Model No. 19EB 8983P packaged hermetic, including controls, starter with installation	=$	600,000
2.	Three cooling towers, 1,000 ton each	=	90,000
3.	Two 12,000,000 BTU/HR Boilers	=	160,000
4.	One 3,000,000 BTU/HR DHW Boiler	=	40,000
5.	Pumps		
	Three Chilled Water	=	15,000
	Three Condenser Water	=	22,500
	Two Pumps for Heating	=	6,000
6.	Chilled Water Distribution System (as for HP-ICES)	=	800,000
7.	Hot Water Distribution System (as for HP-ICES)	=	400,000
8.	Electrical Installation and Connection (2752 kW)	=	318,550
	Sub-Total First Cost	=$	2,452,500
9.	Contingency 10%	=	245,205
	TOTAL FIRST COST	=$	2,697,255

b. Operating Cost for Conventional System

1.	Energy Cost	=$	263,048
2.	Maintenance (1% of Equipment First Cost)	=	27,000
3.	Labor, 8 station engineers	=	160,000
4.	Administration - 6 people	=	90,000
	TOTAL OPERATING COST	=$	540,048

Details of the Energy Cost fot the conventional system, given as part of the Operating Cost, are as follows:

Details of Energy Cost for the Conventional System:

1.	Compressors (see Table A 3.7)	=$	83,383
2.	Refrigeration Auxiliaries (see Table A 3.8)	=	20,944
3.	Pumps used for heating		
	Demand 6 x 14 x 0.746 x 4.02	=	270
	Energy 43,082 x $0.00985	=	424
4.	Domestic Hot Water Pumps (as per HP-ICES)	=	1,164
	Sub-Total Electrical	=$	106,185

5. Heating & DHW

$$= \frac{36,234 \times 10^6 \text{ BTU}}{0.7 \times 10^5} = 517,700 \text{ Therms}$$

Cost @ $0.303/Therms	=	156,863
TOTAL ENERGY COST	=$	263,048

4.4 LIFE-CYCLE COST ANALYSIS OF THE MARKET SQUARE PROJECT

This section summarizes the steps and the results of the comparative life-cycle cost analysis for the Market Square HP-ICES system and a conventional system with the same capacity.

Table A4.1 gives the details of the life cycle analysis of the HP-ICES system and a conventional system with the same capacity.

The following is a summary of the results:

1. The HP-ICES system is more energy-efficient than the conventional system. In the example given here, the savings in the annual average energy cost savings achieved by using the HP-ICES system is about $217,000.
2. In the present example, the total annual average life-cycle cost is also less for the HP-ICES system than for the conventional system. However, the savings in the total life-cycle cost is a relatively small difference between two large numbers, and is sensitive to parameters such as the interest rate. When these parameters are varied, the life-cycle cost for the HP-ICES system is less than or comparable to that for a conventional system.
3. The life cycle capital cost for the HP-ICES system is about 70% greater than that for the conventional system.

   The total operating cost (including energy cost) for the HP-ICES system, minus the tax savings, is about 27% of the corresponding quantity for the conventional system.
4. The cost of money (i.e. the mortgage interest rate) and the energy price escalation rate are among the most important factors in determining the relative economics of the two systems.

5. The relative values of the life-cycle capital costs, energy costs, etc. are the most useful parameters in comparing the economics of the two types of systems.

6. In the example given here, the Return on Investment for the HP-ICES system, as compared to the conventional system is found to be about 15%. The Payback Period is therefore about 6.7 years. (The Return on Investment is found by determining the value of the discount rate for which the Net Life Cycle Savings becomes zero.)

TABLE A4.1 ECONOMIC ANALYSIS, MARKET SQUARE

		HP-ICES SYSTEM	CONVENTIONAL SYSTEM
FC	First Cost	$4,547,130	$2,697,255
DPF	Down Payment (% of First Cost)	10%	10%
MP	Mortgage Period	20 years	20 years
EP	Period of Economic Analysis	20 years	20 years
MIR	Mortgage Interest Rate (% per year)	15%	15%
DR	Discount Rate	8%	8%
IR	Inflation Rate	6%	6%
EC	Present Energy Costs per year	$ 100,568	$ 263,048
ER	Energy Price Escalation Rate	12%	12%
DPER	Depreciation Period	20 years	20 years
SV	Overall Salvage Value (as fraction of First Cost)	30%	20%
IT	Effective Income Tax Rate	46%	46%
C1	Insurance Costs (0.5%)	$ 22,736	$ 13,486
C2	Cost of Maintenance, Labor, Administration, etc.	$ 280,000	$ 277,000
PT	Property Tax Rate	1%	1%

TABLE A4.1, cont.

		HP-ICES SYSTEM	CONVENTIONAL SYSTEM	LIFE CYCLE SAVINGS BY USING HP-ICES SYSTEM	AVERAGE ANNUAL SAVINGS BY USING HP-ICES SYSTEM
DP	Initial Down Payment	$ 454,713	$ 269,726		
MPF	PWF For Mortgage Loan Payment	1.569	1.569		
MP	PW of Mortgage Payment over Life Cycle	6,419,208	3,807,730	-$2,611,478	
	Annual Av. Mortgage Payment	320,960	190,387		-$130,610
LI= (INT)x (FC-DP)	PW of Loan Interest Paid over Life Cycle	4,985,982	2,957,571		
CC =DP+MP	PW of Life Cycle Capital Costs Before Tax Deduction	6,873,921	4,077,455	-$2,797,195	
	PWF for Energy Prices	26.74	26.74		
ECL	PW of Life Cycle Energy Cost	2,689,175	7,033,870	$4,344,695	
	Annual Av. Energy Cost	134,459	351,694		$217,235
IMC	PW of Insur. & Maint. Cost	4,721,392	4,530,344		
PT	PW of Prop. Tax Payments	709,159	420,657		
OAC =(EC+ IMC+ PT)	PW of Total Operating and Admin. Costs	8,119,727	11,984,871	3,865,144	
	Annual Av. Op. & Admin. Cost	405,986	599,244		$193,257
SSV	PW of System Salvage Value	292,673	115,738		
DEP	PW of Depreciation	$1,562,554	$ 1,059,281		

TABLE A4.1

		HP-ICES SYSTEM	CONVENTIONAL SYSTEM	LIFE CYCLE SAVINGS BY USING HP-ICES SYSTEM	AVERAGE ANNUAL SAVINGS BY USING HP-ICES SYSTEM
TLC =(CC+ OAC- SSV)	PW of Total Life Cycle Cost Before Tax Deduction	$14,700,974	$15,946,588	$ 1,245,614	
ITS	PW of Income Tax Savings (inc. Deprec.) = (IT) (DEP + OAC + LI)	6,747,400	7,360,794		
(OAC- ITS)	(Op. & Admin. Cost - Tax Savings)	1,372,326	4,624,078	3,251,752	$162,588
NLC	Net Life Cycle Cost After Tax Deduction = TLC - ITS	$ 7,953,574	$ 8,585,795	$ 632,221	

4.5 COST TO THE USER AND BASIS FOR CHARGING IN THE MARKET SQUARE PROJECT

The price to be charged to each user is a very complex question, especially for a new type of system such as the HP-ICES system. According to conventional pricing policies, it would depend on the category of the user and the volume of energy usage. The price would also vary with time.

We shall use a very simplified method, and obtain an average price for energy over the period of economic analysis. The cost to the user will consist of two parts: a first part, the demand charge, that depends on the peak load for the user, which pays for the life cycle average capital cost of the system; and a second part, the energy charge, that pays for the total operating costs. Each of these includes a profit margin.

A price schedule for a user of the HP-ICES system would include different rates for different levels of energy usage. Also, for a commercial user, there would be a penalty for exceeding a predetermined peak load. Here, we do not consider such an elaborate price schedule, but merely give the average price that a user would pay for each MBTU/Hr of peak load, and for each MBTU of energy used, over a period of 20 years. (Note that MBTU denotes a million BTU.) We also give the actual price the user pays each year, with an assumed price escalation rate.

The price estimates will be based on the assumption that all the energy produced will be used. It is assumed that the capacity of the system remain the same, i.e. there is no expansion.

The results for the Market Square are as follows:

(a) Capital Costs

Present Worth of Life Cycle Capital Costs Before Tax Deduction

= \$6,873,921

Annual Average Life Cycle Capital Cost = \$343,700

Maximum Peak Load =

Maximum of Peak Heating Load and Peak Cooling Load

= 38.5 x 10^6 BTU/Hr. = 38.5 MBTU/Hr.

Average Cost to Owner for Each MBTU/Hr. Capacity =

\$8,927 per year per MBTUH capacity.

In units of kilowatts, this corresponds to \$30.5 per year per kW capacity, or \$2.5 per month per kW capacity.

This, together with a profit margin, would be recovered from the users over the life cycle of the system, e.g. through a schedule of demand charges.

For a 40% profit margin, the demand charges to the user would be an average of about \$1,041 per month per MBTUH capacity (or about \$3.55 per month per kW capacity). This is in terms of the present worth.

If the demand payments were constant over the life cycle, this would require a dollar payment of about \$1,335 per month per MBTUH capacity (or about \$4.56 per month per kW capacity). This assumes a discount rate of 8% and an inflation rate of 6% per year. For comparison, this is about 69% more than for the conventional system considered here. Also note that in the Potomac Electric Power Company Schedule used in determining the cost of the energy used by the HP-ICES system, the electricity demand charges were \$5.18 per month per kW between June and October, and \$4.02 per month per kW between November and May.

(b) Energy Costs

Average Annual Operating and Administration Costs over the Life Cycle (in terms of present worth) = $405,986

Total Energy Usage Paid for by the Users = 72,000 MBTU (This consists of about 27,000 MBTU for heating, 36,000 MBTU for cooling, and 9,000 for hot water.)

Average Energy Cost per MBTU = $5.64 per MBTU

Adding a profit margin of 40% results in the following energy cost to the user:

Average Energy Price Paid by the User Over The Life Cycle = $7.90 per MBTU, in terms of the Present Worth.

With the HP-ICES system, the energy price escalation rate would be less than the escalation rate for conventional fuel. Assuming that the conventional fuel price escalates at 12%, and that the other parts of the operating and administration costs increase at the general inflation rate, the above energy price implies a dollar payment of $8.63 per MBTU (equivalent to 2.95¢/kWh) initially, escalating at 8% per year.

Note that the electrical energy charge of the Potomac Electric Power Company is 1.417¢/kWh between June and October, and 0.985¢/kWh between November and May, while the natural gas price charged by Washington Gas Light Co. is $3.03 per MBTU. These prices are assumed to escalate at 12% per year.

For comparison, the same procedure applied to the conventional system results in an average energy price paid by the user over the life cycle (in terms of the Present Worth) of about $11.25 per MBTU.

The dollar payment would be \$10.61 per MBTU (equivalent to 3.62¢/kWh) initially, escalating at 9.5% per year.

The average life cycle energy price paid by the user of the HP-ICES system would be about 32% less than that paid by the user of the conventional system, for a constant volume of energy usage.

A.5 INSTITUTIONAL CONSIDERATIONS, MARKET SQUARE COMPLEX

A. ENVIRONMENTAL IMPACTS

1) Pollution and Noise

The use of an HP-ICES system in the Market Square Complex, with heat pumps driven by electricity, uses less energy overall than a conventional system with the same capacity, and therefore leads to less pollution.

Since the heat pump in this system would be electricity driven, there will be practically no pollution at the location of the heat pump. The pollution associated with the generation of the electricity would be localized at the power plant, and can therefore be isolated and controlled more easily.

The cooling system involves an ice bin, and would not create any air pollution or thermal pollution, in contrast to the large cooling towers in conventional systems, which can create both thermal pollution and chemical corrosion problems. The only cooling towers in the HP-ICES system would be smaller, and would be used only a small fraction of the time -- i.e. at night during about three months in the year, which would lead to minimum thermal pollution or other environmental problems.

Because the HP-ICES system operates with closed loops, there will be little or no thermal pollution or chemical water pollution, in contrast to what happens in a conventional system with water cooling.

Since the storage bin contains only water or ice, leakage from it would not cause any pollution problems, in contrast to systems using brine or other fluids.

Since the storage is large, care must be taken to drain it only into the storm sewer and not into the sanitary sewer.

The noise from the cooling tower operation would also be very small, because the cooling tower would be operated only a small part of the time.

2) Land Utilization

Since the large storage bin is under the building, it will not take up excessive additional land. The total space requirement for the HP-ICES system for this project would be less than that for a conventional system.

The piping network would be underground. Care should be taken to integrate it with the existing buildings and roads, so that it does not become necessary to open up roads or cross public areas.

3) Resource Consumption in Construction

This would be less than with a conventional system.

4) Retirement

The retirement of the system should not give rise to any environmental problems. The large storage bin can be converted into a parking area without major alterations. The mechanical rooms can be used for other purposes.

B. STATUTES, CODES, REGULATIONS, ETC.

Special questions can come up in the Market Square project because it is located in Washington, D.C.

The Height of Buildings Act in Washington, D.C. establishes a maximum height of 160 feet for buildings fronting on Pennsylvania Avenue between the 1st and 15th Streets, N.W. It may be necessary to seek an amendment to this Act to be able to develop the Market Square Complex. However, this would be a minor amendment.

Under the original zoning regulations, the development area included two separate zoning classifications. It would be desirable to amend the original zoning regulations so that the whole development area would be a single new zone, with a unified system of standards in the area.

The other proposed regulations include the following:

(a) Mandatory Standards, which would cover areas such as Energy Conservation, the overall conformity of the project with the environment, noise and air quality abatement during construction, and other questions such as historical and architectural preservation.

(b) Non-mandatory Standards for assisting developers in designing the projects; these would include guidelines in designing the buildings and facilities. These may become part of the building code by the time that the project is implemented.

Another important kind of regulation would be the tax laws, which may provide incentives for using the system that is the best available in terms of Energy Conservation.

C. POTENTIAL OWNERSHIP AND FINANCING APPROACHES FOR THE MARKET SQUARE HP-ICES PROJECT

The possible approaches are the following:

A. Ownership and financing by the federal government or an agency created by it.

B. Ownership and financing by the local municipality, or an agency created by it.

C. Financing by a third party, such as a developer or a private corporation (which could be a utility).

D. A combination of A, B and C above.

The main factors to be considered are the following:

1) The project requires a large financial backing and involves a financial risk.

2) The project will require a high level of technical ability and competent management.

3) Being in Washington, D.C., the project will be subject to special regulations.

4) The project will have high public visibility in the city and in the whole nation. It is meant to serve as a model for other projects in the nation.

The advantages of Approach A will be that the federal government will not have any difficulty in obtaining the large financial and management resources, and is in a position to accept the financial risk.

On the other hand, the project needs participation by a large number of private agencies, who would prefer the flexibility and efficiency that can be more easily obtained with private management.

If the financing of the project is done by a municipality, the financing mechanism will be either through bonds issued by the municipality, or through federal financial support obtained by the municipality, or through the formation of municipal joint action agencies. An important requirement on the financing mechanism will be that if the project fails financially, it should not create a serious risk for the city's finances.

Financing by a third party, such as a developer, or a private corporation (including a utility company) has the advantages that a private company may possess the technical expertise that a municipality normally would not have, and may be able to operate with greater flexibility and efficiency than the federal government or many municipalities. The main problems for a developer or private corporation are that most of them do not have the large financial resources required and cannot afford the risk involved in a large investment in a large and new type of system such as the HP-ICES system.

It appears that the best alternative for ownership and financing would involve a combination of the three approaches A, B and C, with financial backing by the federal government, and management by a private corporation or jointly by the municipality and a private corporation. Such an arrangement should be possible within the framework of the Pennsylvania Avenue Development Corporation.

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B. PARK PLAZA - BOSTON

SUMMARY

1. AREAS

Sector	Area ft^2	Per cent of Total
Residential (1600 dwelling units)	2,011,000	34.9
Commercial		
a. Retail and Galleria	567,000	9.9
b. Office	1,110,000	19.3
c. Hotel	900,000	15.7
d. Parking	1,076,000	18.7
c. Service	84,000	1.5
Total	5,748,000	100.0

2. LOADS

Heating		Domestic Hot Water		Cooling	
Demand	Annual Energy Requirement	Demand	Annual Energy Requirement	Demand	Annual Energy Requirement
165,000,000 BTU/hr	$186,380 \times 10^6$ BTU	35,800,000	$75,980 \times 10^6$ BTU	11680 Tons	10,056,000 ton-hrs

3. COOLING SAVED BY THE USE OF HP-ICES (BY-PRODUCT OF HEATING AND DHW)

$120,672 \times 10^6$ BTU or 100%

4. PERFORMANCE AND COSTS

		HP-ICES System	Conventional System
1.	Source Energy Consumption		
	Winter	$138{,}436\times10^6$ BTU	$333{,}285\times10^6$ BTU
	Summer	$40{,}044\times10^6$ BTU	$175{,}932\times10^6$ BTU
	Total	$178{,}480\times10^6$ BTU	$509{,}217\times10^6$ BTU
	Energy Saving BTU	$330{,}737\times10^6$ BTU	---
2.	Annual Energy Cost	$ 777,693	$2,546,707
	Energy Cost Saving	$1,787,014	---
3.	Annual Operating Cost	$1,171,943	$2,839,707
	Oper. Cost Saving	$1,667,764	---
4.	First Cost	$29,011,400	$6,397,655
	Incremental First Cost	$22,613,745	
5.	Pay-back	8 years	
6.	Rate of Return	13%	

B.1 COMMUNITY DESCRIPTION

1.1 COMMUNITY DESCRIPTION

Park Plaza is a proposed 5,750,000 square feet project, situated in downtown Boston between the Arlington and Stuart Streets and the Public Garden.

It was first conceived by Mayor Kevin White in 1970, and thereafter, sponsored by the Boston Redevelopment Authority (BRA) as a huge privately financed urban renewal project which was to transform the somewhat shabby and deteriorating Park Square area into a glittering complex of office buildings, residential and hotel high rise with plazas, promenades, entertainment facilities, shops and parking garages.

Park Plaza is intended to be a showplace and a link between the Back Bay and the downtown shopping area, but economic feasibility should be the key to the whole proposal.

The complex is distributed over three parcels, designated Parcel 1, 2 and 3. Parcel 1 comprises a residential tower of 546,000 square feet containing 420 dwelling units, 493,000 square feet of retail spaces and above the retail, 510,000 square feet of low rise offices and additional office space of 600,000 square feet in an office tower.

Parcel 2 contains a high rise hotel of 900,000 square feet out of which a thousand rooms occupy 460,000 square feet. Public spaces (lobbies, restaurants, shopping, etc.), occupy 340,000 square feet. The remaining 100,000 square feet being occupied by service.

Parcel 3 comprises a high rise tower of 1,030 luxury dwelling units of 1,315,000 square feet total area, and a small tower of 150 dwelling units which

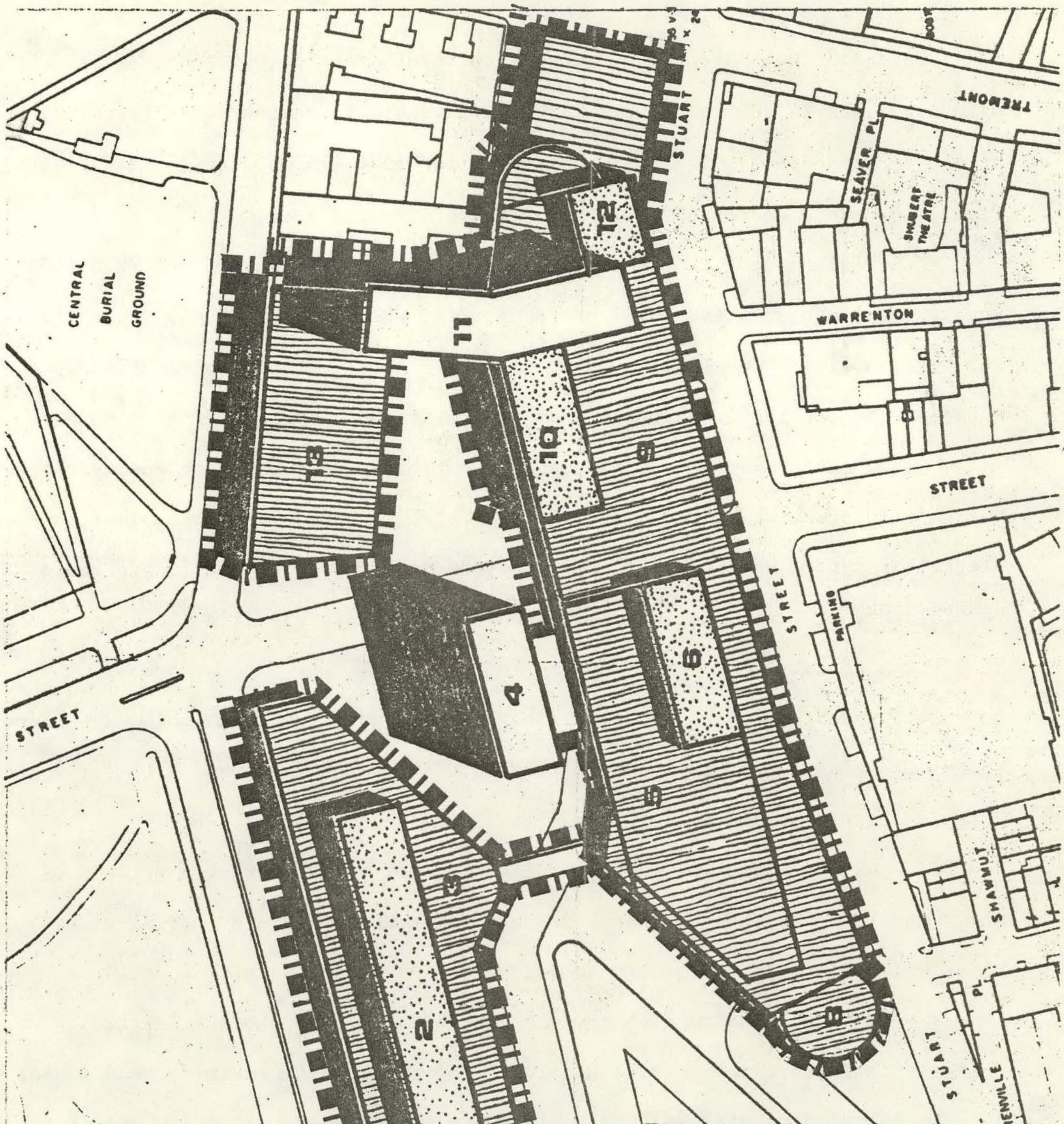


FIG.- B1.1
PARK PLAZA,
BOSTON PROJECT
SYSTEM DESCRIPTION

are subsidized and non-air conditioned, of total area 150,000 square feet. It also compreses 24,000 square feet of retail area and together with Parcel 2, it can accommodate 3,000 cars in a parking lot of 1,076,000 square feet. Also, Parcels.2, and 3 shares 84,000 square feet of service area.

Dividing the Park Plaza Complex by sectors, the residential (2,011,000 square feet) accounts for 35.0% of total area, the retail (567,000 square feet) for 9.9%, office space (1,110,000 square feet) for 19.3% and hotel (900,000 square feet) for 15.6%. Parking and service accounts for the rest.

The energy demand and consumption estimates were based on energy conservation standards, while the building materials, standards and mechanical and electrical system were based on ASHRAE Standards 90-75. The buildings would be of luxury type and insulated and the mechanical and electrical systems would be of high quality.

Exposed walls and roofs will have a high mass construction of 100 lb/cu.ft. and will be insulated to give a "u" factor of 0.06 for roofs and 0.08 for walls. Interior partition and floor slabs will be designed for 60 lbs/ft.

Where possible insulation will be located on the external surface of walls and roofs so that advantage is taken of the structure's thermal mass, which will sat as a heat reservoir and smooth out peak loads.

High mass construction gives the building a high thermal inertia characteristic that reduces the effect of rapidly changing outside conditions.

All windows will be double glazed to reduce heat conduction and sound transmission. Windows facing south will be provided with external shading devices designed to exclude high angle summer sun, but allow entry of low angle winter sun. All windows will be carefully caulked and weatherstripped to reduce infiltration of outside air.

Concerning energy conservation, spaces like corridors, wash rooms, utility rooms, vertical transport and other similar spaces that do not have such stringent temperature condition requirements should be located in those areas where heat loss is potentially the greatest (usually at the north wall). These spaces can then act as a buffer between outside and inside occupied areas.

Areas of common requirements such as lighting level - same occupied hours etc. should be grouped together so they can be served by one system designed for those conditions and operated only when required.

The loads and annual energy requiremnts are as follows:

a) Cooling:

Demand	=	11,680 tons
Annual Requirements	=	10,056,000 Ton-hours

b) Heating:

Demand	=	165,000,000 Btu/hr
Annual Requirement	=	$186,380 \times 10^6$ Btu

c) Domestic Hot Water:

Demand	=	35,800,000 Btu/hr
Annual Requirement	=	$75,980 \times 10^6$ Btu

d) Electrical:

Demand	=	23,300 KW
Annual Requirement Base Load	=	8,000,000 kWh

1.2 DESIGN CONDITIONS *

Latitude 42°

Winter design temperature 10°F

Summer DB - 88°F, WB - 74°F

Inside temperature, winter 68°F, night set back 58°F, summer 78°F DB, 50% RH

Daily temperature range during summer - 16°F

Equivalent full load hours cooling 400-1200

Heating degree-days 5634

* Ref. - ASHRAE - Fundamentals 23
Systems 43

1.3 BASIC ASSUMPTIONS

Based on square and cubic feet which in turn is based on standard requirements for different sectors in the project.

1. Cooling Demand
   a. Residential 1.8 ton per dwelling unit
   b. Retail 240 ft^2/ton; galleria = 200 ft^2/ton
   c. Office 300 ft^2/ton
   d. Hotel, public 150 ft^2/ton, rooms 5/8 ton per room
2. Heating Demand
   a. Residential 3.5 Btu/hr per ft^3
   b. Retail 2.4 Btu/hr per ft^3
   c. Office 3.8 Btu/hr per ft^3
   d. Hotel 3.5 Btu/hr per ft^3
3. Domestic Hot Water Demand
   a. Residential 1.0 Btu/hr per ft^3
   b. Retail 0.4 Btu/hr per ft^3
   c. Office 0.4 Btu/hr per ft^3
   d. Hotel 1.0 Btu/hr per ft^3
4. Energy Requirements for Heating (per year)
   a. Residential 4.85 M Btu per ft^3
   b. Retail 1.15 M Btu per ft^3
   c. Office 3.45 M Btu per ft^3
   d. Hotel 4.85 M Btu per ft^3
5. Energy Requirements for Domestic Hot Water (per year)
   a. Residential 2.5 M Btu per ft^3
   b. Retail 0.2 M Btu per ft^3
   c. Office o.5 M Btu per ft^3
   d. Hotel 2.5 M Btu per ft^3

1.4 AREA TABULATION

Residential

Parcel 1	Tower, 420 units	546,000 ft^2	
Parcel 3	Tower, 1030 units (luxury)	1,315,000 ft^2	
	Tower, 150 units (subsidized)	150,000 ft^2	
			2,011,000 ft^2

Retail Galleria

Parcel 1	Retail	385,000 ft^2	
	Galleria	108,000 ft^2	
Parcel 3	Retail	50,000 ft^2	
	Galleria	24,000 ft^2	
			567,000 ft^2

Office

Parcel 1	Tower	600,000 ft^2	
	Over Retail	510,000 ft^2	
			1,110,000 ft^2

Hotel

Parcel 2	Public	341,000 ft^2	
	Guest Rooms (1000 rooms)	459,000 ft^2	
	Service	100,000 ft^2	
			900,000 ft^2

Parking

Parcel 2	1600 cars	580,000 ft^2	
Paecel 3	1400 cars	496,000 ft^2	
			1,076,000 ft^2

1.4 AREA TABULATION

Residential			
Parcel 1	Tower, 420 units	546,000 ft^2	
Parcel 3	Tower, 1030 units (luxury)	1,315,000 ft^2	
	Tower, 150 units (subsidized)	150,000 ft^2	
			2,011,000 ft^2
Retail Galleria			
Parcel 1	Retail	385,000 ft^2	
	Galleria	108,000 ft^2	
Parcel 3	Retail	50,000 ft^2	
	Galleria	24,000 ft^2	
			567,000 ft^2
Office			
Parcel 1	Tower	600,000 ft^2	
	Over Retail	510,000 ft^2	
			1,110,000 ft^2
Hotel			
Parcel 2	Public	341,000 ft^2	
	Guest Rooms (1000 rooms)	459,000 ft^2	
	Service	100,000 ft^2	
			900,000 ft^2
Parking			
Parcel 2	1600 cars	580,000 ft^2	
Paecel 3	1400 cars	496,000 ft^2	
			1,076,000 ft^2

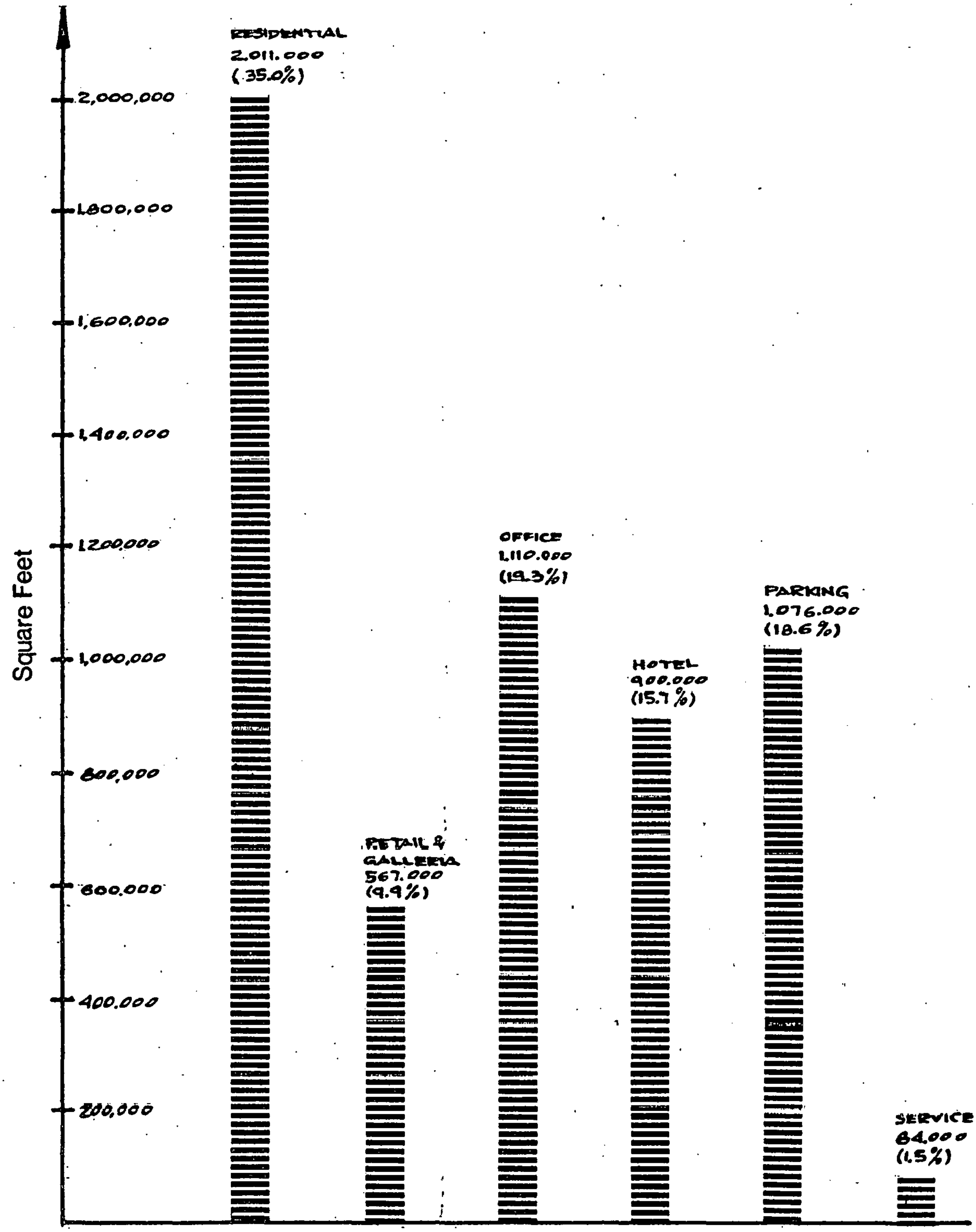


FIG.- B1.2

Area Distribution By Sectors

Service

Parcel 2		45,000 ft^2	
Parcel 3		39,000 ft^2	
			84,000 ft^2
	Total Gross Area		5,748,000 ft^2
Heated Basement, Parcel 1 Retail			45,000 ft^2
	Sub-total		5,793,000 ft^2

Non Heated Areas

Parking	1,076,000 ft^2
Service	84,000 ft^2
	1,160,000 ft^2
Total Gross Heated Area	4,633,000 ft^2

Non Air Conditioned Areas

Subsidized Apartments	150,000 ft^2
Hotel Service	100,000 ft^2
Service	84,000 ft^2
	334,000 ft^2
Total Gross Air Conditioned Area	4,338,000 ft^2

GROSS VOLUME SUMMARY

Residential	- Parcel 1	4,900,000 ft^3
	Parcel 3	13,200,000 ft^3
Retail	- Parcel 1	7,200,000 ft^3
	Parcel 3	1,200,000 ft^3
Office	- Parcel 1	13,200,000 ft^3
Hotel	- Parcel 2	9,000,000 ft^3
		48,700,000 ft^3

1.5 REFRIGERATION PEAK DEMAND ESTIMATE

1. Residential

Parcel 1	420 units at 1.8T/unit		=	756T
Parcel 3	1,030 untis at 1.8 T/unit		=	1,854T
		Sub-Total	=	2,610T

2. Retail and Galleria

Parcel 1	Retail	385,000 ft^2 : 240 ft^2/T	=	1,604T
	Galleria	108,000 ft^2 : 200 ft^2/T	=	540T
Parcel	Retail	50,000 ft^2 : 240 ft^2/T	=	208T
	Galleria	24,000 ft^2 : 200 ft^2/T	=	120T
		Sub-Total		2,472T

3. Office

Parcel 1	Tower	600,000 ft^2 : 300 ft^2/T	=	2,000T
	Over Retail	510,000 ft^2 : 300 ft^2/T	=	1,700T
		Sub-Total		3,700T

4. Hotel

Parcel 2	Public	341,000 ft^2 : 150 ft^2/T	=	2,273T
	Rooms	1,000 rooms at 5/8T/room	=	625T
		Sub-Total		2,898T
		TOTAL PEAK DEMAND	=	11,680T

REFRIGERATION PEAK DEMAND ESTIMATE (Cont'd)

TABLE B 1.1 COOLING PEAK LOAD DISTRIBUTION BY SECTORS

	Parcel 1	Parcel 2	Parcel 3	Total	% of Total
Residential	756T	-	1,854T	2,610T	22.3
Retail	1,604T	-	208T	1,812T	15.5
Galleria	540T	-	120T	660T	5.7
Office	3,700T	-	-	3,700T	31.7
Hotel	-	2,898T	-	2,898T	24.8
Total	6,600T	2,898T	2,182T	11,680T	100.0
% of Total	56.5	24.8	18.7	100.0	

Using diversity factor 0.82 *

Tonnage 11680 x 0.82 = 9577 tons

(say 10,000 tons)

1.6 ENERGY REQUIREMENT AND CONSUMPTION FOR COOLING

Equivalent full load hours:

Residential	-	1,000
Retail	-	1,000
Galleria	-	1,000
Office	-	1,000
Hotel	-	1,200

Use 1050 Equivalent Full Load Hours for entire system.

*The peak energy usages for the residential and commercial sectors do not coincide. Thus the diversity factor which is the ratio of the maximum total load to the sum of the residential peak load and the commercial peak load was estimated at 0.82.

TABLE B 1.2 ENERGY REQUIREMENT FOR COOLING*

Month	% Refrigeration Energy Requirement	Refrigeration Energy Requirement Btux10^6
January	0	0
February	0	0
March	0	0
April	5	6,034
May	10	12,067
June	13	15,687
July	22	26,548
August	22	26,548
September	19	22,928
October	9	10,860
November	0	0
December	0	0
Total Annual	100	120,672

* Only the mechanical refrigeration loads are considered. If there is a cooling load in the winter, enthalpy control will be applied which does not affect the energy balance.

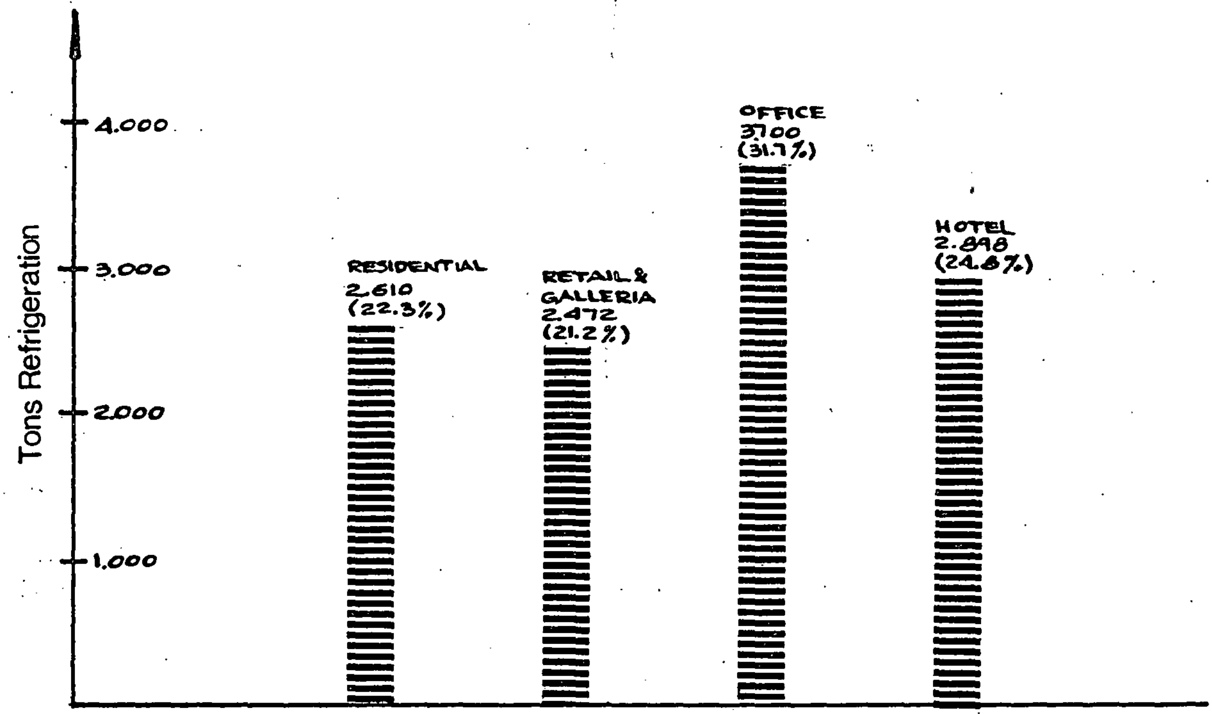


FIG.- B1.3

Tonnage Distribution By Sectors

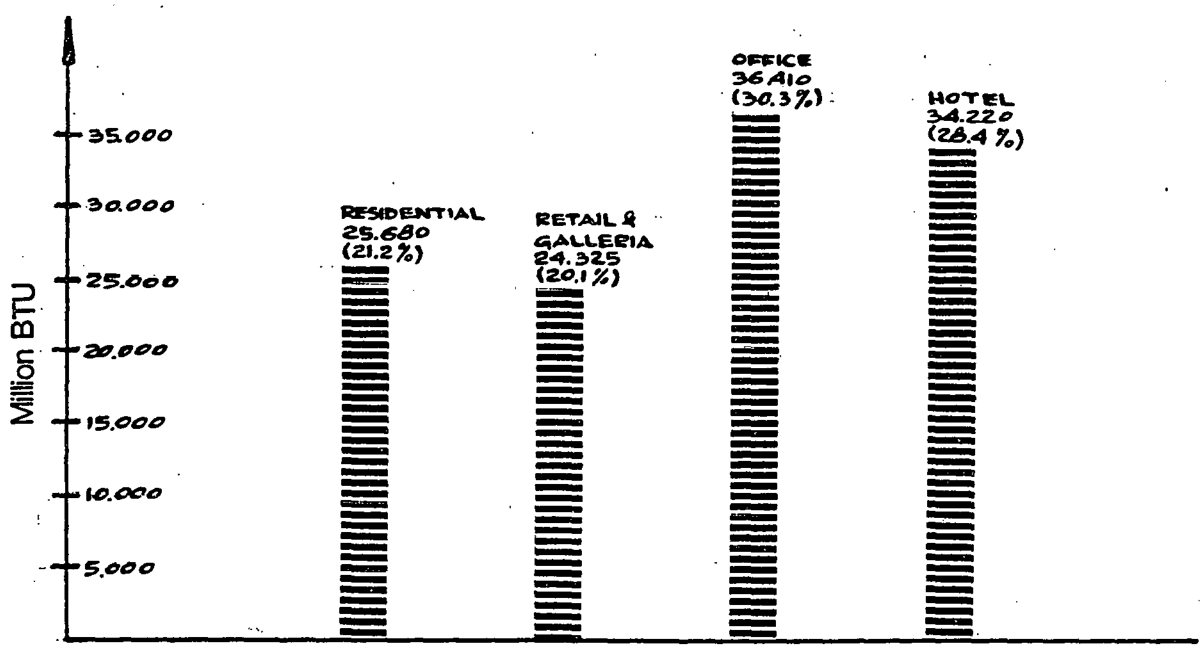


FIG.- B1.4

Annual Energy Requirement For Cooling

1.7 HEATING AND DHW PEAK DEMAND ESTIMATE

Residential					
Parcel 1	546,000 ft^2 x 9 ft	=	4,900,000 CF		
	Htg. 4,900,000 CF x 3.5 Btu/hr/CF	=	4,9	=	17,000,000 Btu/hr
	DHW 4,900,000 CF x 1.0 Btu/hr/CF			=	4,900,000 Btu/hr
Parcel 3	(1,315,000 + 150,000)ft^2 x 9 ft	=	13,200,000 CF		
	Htg. 13,200,000 CF x 3.5 Btu/hr/CF			=	46,000,000 Btu/hr.
	DHW 13,200,000 CF x 1.0 Btu/hr/CF			=	13,200,000 Btu/hr
Retail					
Parcel 1	385,000 + 108,000	=	493,000 ft^2		
	Basement	=	45,000 ft^2		
	448,000 ft^2 x 16 ft	=	7,200,000 CF		
	Htg. 7,200,000 CF x 2.4 Btu/hr/CF			=	17,300,000 Btu/hr
	DHW 7,200,000 CF x 0.4 Btu/hr/CF			=	2,900,000 Btu/hr
Parcel 3	50,000 + 24,000 = 74,000 ft^2 x 16ft	=	1,200,000 CF		
	Htg. 1,200,000 CF x 2.4 Btu/hr/CF			=	2,900,000 Btu/hr
	DHW 1,200,000 CF x 0.4 Btu/hr/CF			=	500,000 Btu/hr
Office					
Parcel 1	1,110,000 ft^2 x 12 ft	=	13,200,000 CF		
	Htg. 13,200,000 CF x 3.8 Btu/hr/CF			=	50,400,000 Btu/hr
	DHW 13,200,000 CF x 0.4 Btu/hr/CF			=	5,300,000 Btu/hr
Hotel					
Parcel 2	900,000 ft^2 x 10 ft	=	9,000,000 CF		
	Htg. 9,000,000 CF x 3.5 Btu/hr/CF			=	31,400,000 Btu/hr
	DHW 9,000,000 CF x 1.0 Btu/hr/CF			=	9,000,000 Btu/hr
TOTAL	Htg.			=	165,000,000 Btu/hr
	DHW			=	35,800,000 Btu/hr

TABLE B 1.3 HEATING AND DHW PEAK DEMAND SUMMARY BY SECTOR

MBH

Sector	Parcel 1			Parcel 2			Parcel 3			Total		
	Heating	DHW	Sub-Total	Heating	DHW	Sub-Total	Heating	DHW	Sub-Total	Heating	DHW	Grand Total
Residential	17,000	4,900	21,900	-	-	-	46,000	13,200	59,200	63,000	18,100	81,100
Retail	17,300	2,900	20,200	-	-	-	2,900	500	3,400	20,200	3,400	23,600
Office	50,400	5,300	55,700	-	-	-	-	-	-	50,400	5,300	55,700
Hotel	-	-	-	31,400	9,000	40,400	-	-	-	31,400	9,000	40,400
Total	84,700	13,100	97,800	31,400	9,000	40,400	48,900	13,700	62,600	165,000	35,800	200,800

1.8 ANNUAL HEATING AND DHW REQUIREMENT ESTIMATE

Residential

Parcel 1	Htg.	4,900,000 CF x 4.85 M Btu/CF	=	23,600,000 M Btu
	DHW	4,900,000 CF x 2.50 M Btu/CF	=	12,200,000 M Btu
Parcel 3	Htg.	13,200,000 CF x 4.85 M Btu/CF	=	64,000,000 M Btu
	DHW	13,200,000 XF x 2.50 M Btu/CF	=	33,000,000 M Btu

Retail

Parcel 1	Htg.	7,200,000 CF x 1.15 M Btu/CF	=	8,300,000 M Btu
	DHW	7,200,000 CF x 0.20 M Btu/CF	=	1,440,000 M Btu
3	DHW	1,200,000 CF x 0.20 M Btu/CF	=	240,000 M Btu

Office

Parcel 1	Htg.	13,200,000 CF x 3.45 M Btu/CF	=	45,500,000 M Btu
	DHW	13,200,000 CF x 0.50 M Btu/CF	=	6,600,000 M Btu

Hotel

Parcel 2	Htg.	9,000,000 CF x 4.85 M Btu/CF	=	43,600,000 M Btu
	DHW	9,000,000 CF x 2.50 M Btu/CF	=	22,500,000 M Btu

TABLE B 1.4 ANNUAL HEATING AND DHW REQUIREMENT SUMMARY BY SECTOR

BTU X 10^6

Sector	Parcel 1			Parcel 2			Parcel 3			Total		
	Heating	DHW	Sub-Total	Heating	DHW	Sub-Total	Heating	DHW	Sub-Total	Heating	DHW	Grand Total
Residential	23,600	12,200	35,800	-	-	-	64,000	33,000	97,000	87,600	45,200	132,800
Retail	8,300	1,440	9,740	-	-	-	1,380	240	1,620	9,680	1,680	11,360
Office	45,500	6,600	52,100	-	-	-	-	-	-	45,500	6,600	52,100
Hotel	-	-	-	43,600	22,500	66,100	-	-	-	43,600	22,500	66,100
Total	77,400	20,240	97,640	43,600	22,500	66,100	65,380	33,240	98,620	186,380	75,980	262,360

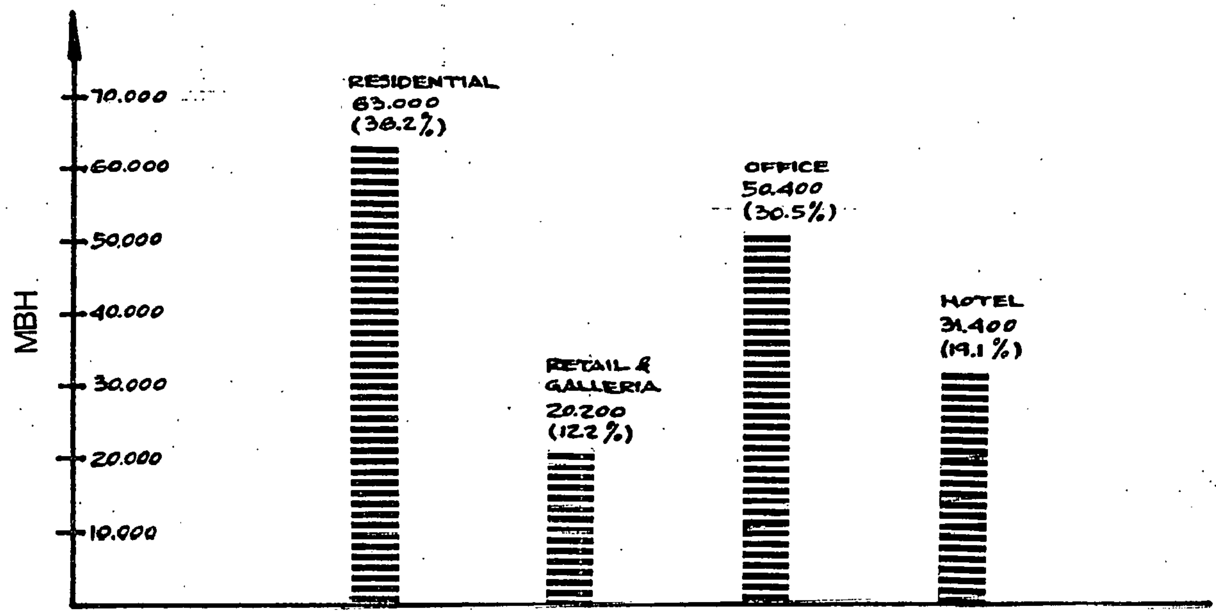


FIG.- B1.5

Space Heating Demand

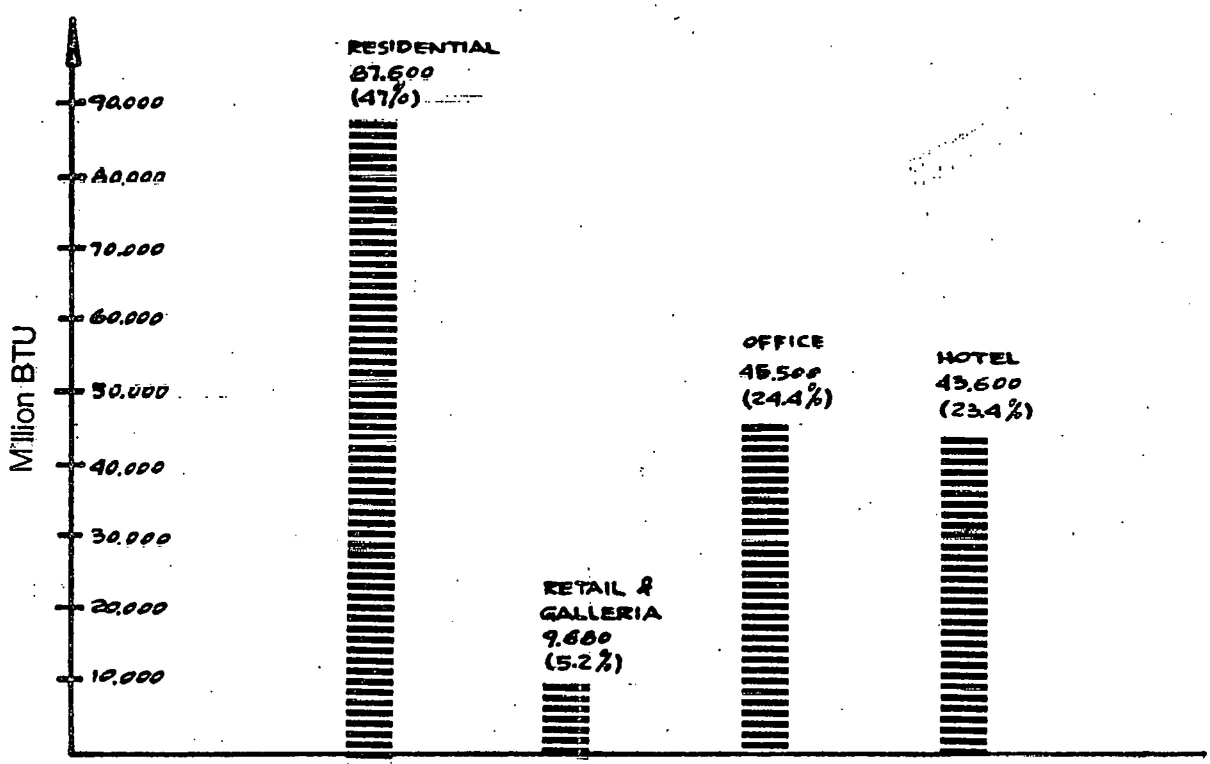


FIG.- B1.6

Annual Energy Requirement For Space Heating

TABLE B 1.5 ANNUAL HEATING AND DOMESTIC HOT WATER PROFILE

Month	DD	Heating Requirement Btu x 10^6	DHW Requirement Btu x 10^6	Heating & DHW Requirements Btu x 10^6
January	1,088	35,992	6,453	42,445
February	972	32,155	5,829	37,984
March	846	27,987	6,453	34,440
April	517	16,971	6,245	23,216
May	208	6,881	6,453	13,334
June	39	1,290	6,245	7,535
July	0	0	6,453	6,453
August	0	0	6,453	6,453
September	66	2,183	6,245	8,428
October	316	10,454	6,453	16,907
November	603	19,948	6,245	26,193
December	983	32,519	6,453	38,972
Annual	5,634	186,380	75,980	262,360

TABLE B 1.6 RECAPITULATION OF ANNUAL ENERGY REQUIREMENTS FOR COOLING, HEATING & DHW

	Residential	Commercial	Total
Cooling	25,689 x 10^6 BTU	94,983 x 10^6 BTU	120,672 x 10^6 BTU
Heating	87,600 x 10^6 BTU	98,780 x 10^6 BTU	186,380 x 10^6 BTU
DHW	45,200 x 10^6 BTU	30,780 x 10^6 BTU	75,980 x 10^6 BTU

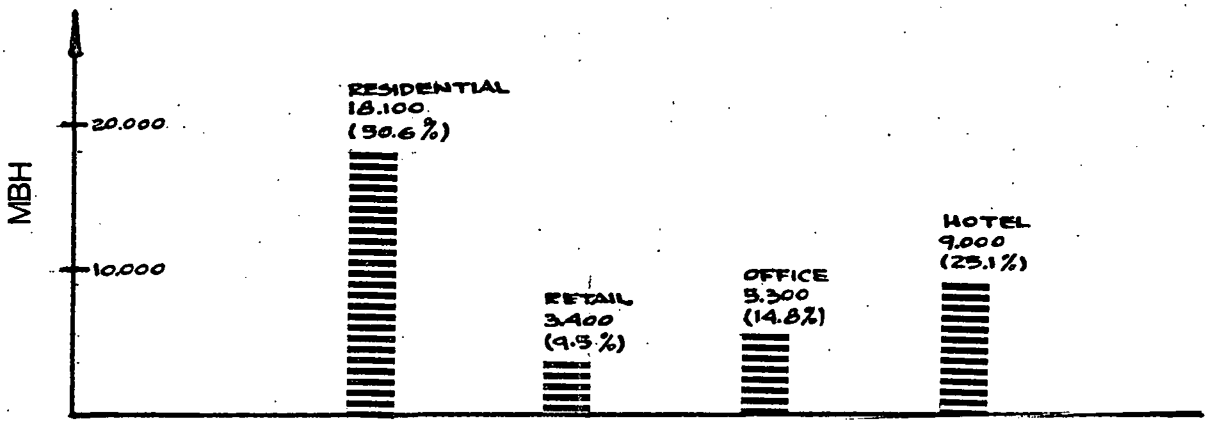


FIG.- B1.7

Domestic Hot Water Demand

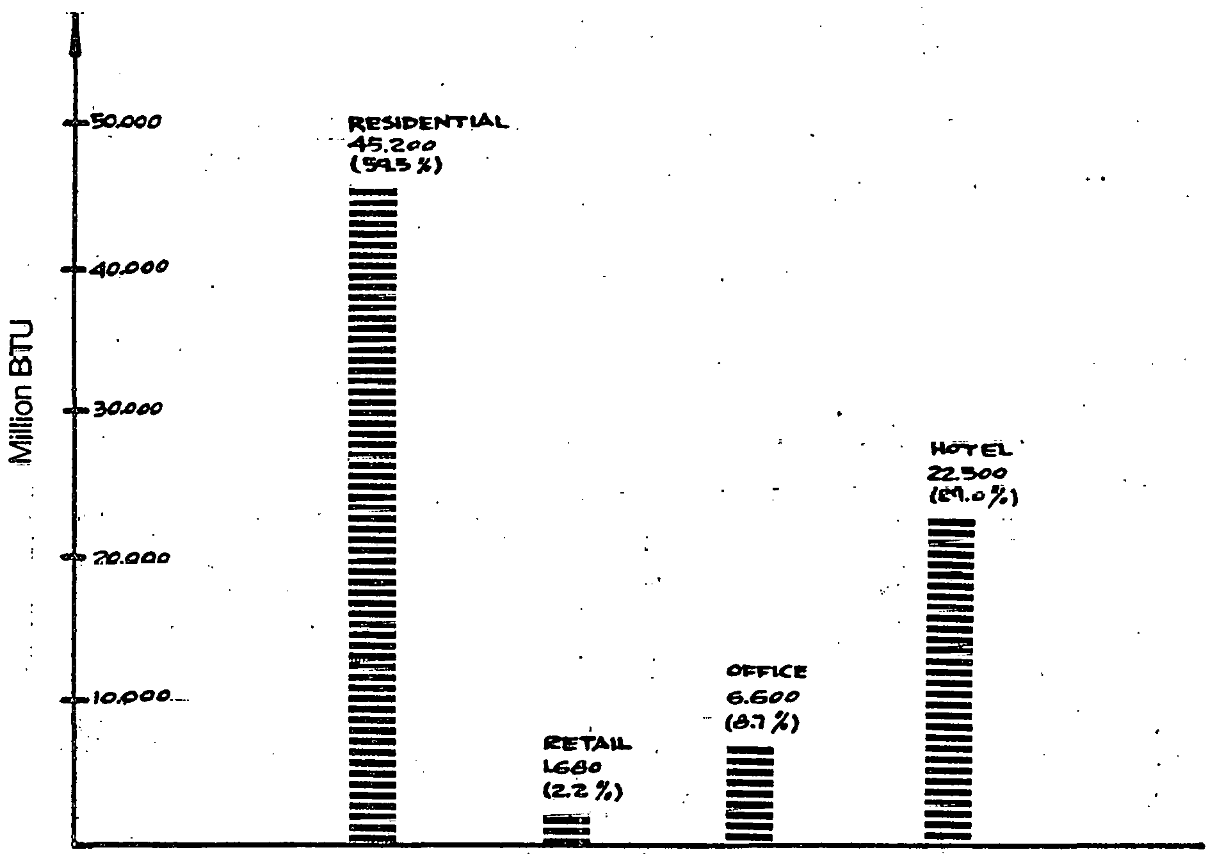


FIG.- B1.8

Annual Energy Requirement For Domestic Hot Water

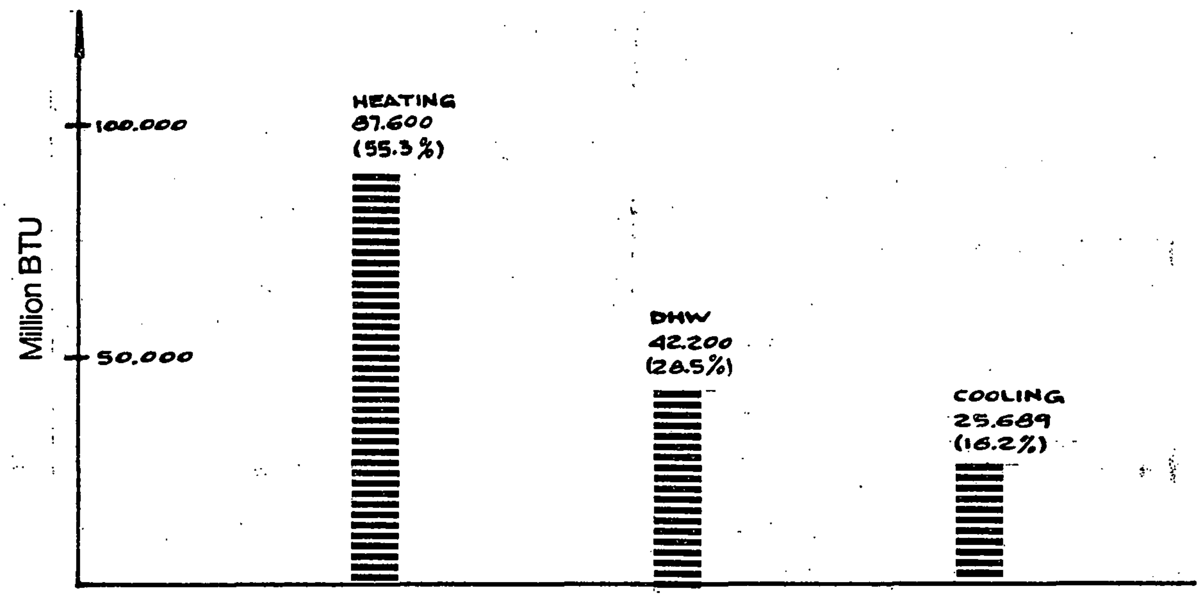


FIG.- B1.9

Residential Sector Comparison Of Heating,DHW & Cooling

Annual Energy Requirements.

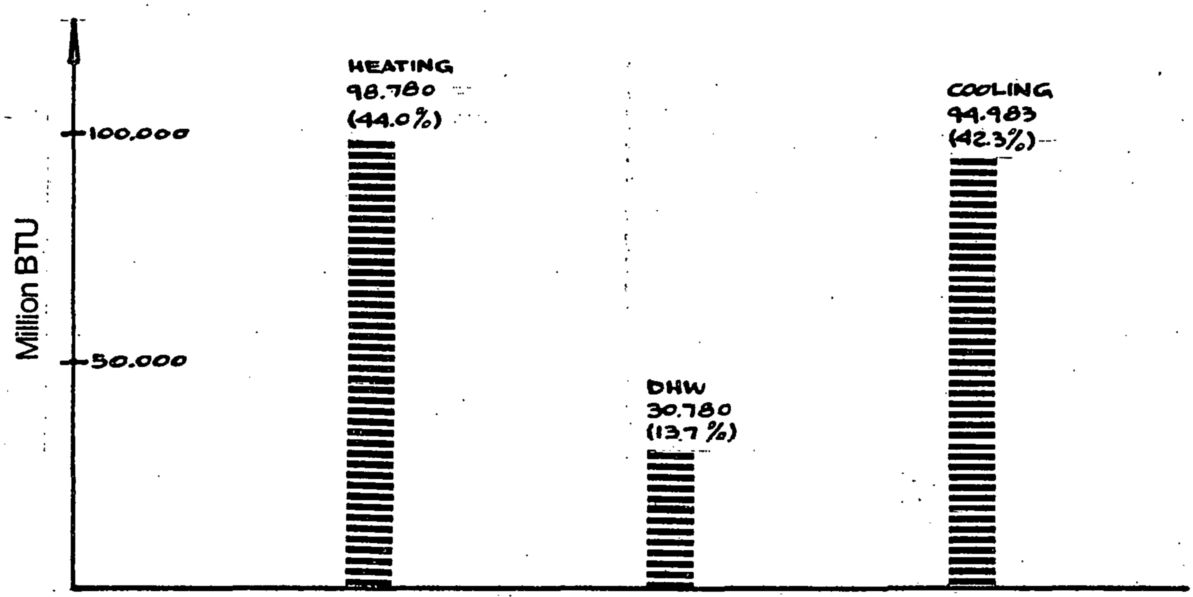


FIG.- B1.10

Commercial Sector „ Retail' Office & Hotel„ Comparison Of

Heating, DHW & Cooling — Annual Energy Requirements

B.2. SYSTEM DESCRIPTION

2.1 INTRODUCTION

The HP-ICES as applied to Park Plaza, Boston differs somewhat from the HP-ICES as applied to the Market Square Complex in Washington, DC.

The results of the First Report indicated that an attractive system could be obtained if gas or diesel engine drive could be used to drive the heat pump.

The heat recovered from the jacket water heat exchanger and exhaust silencer greatly enhance the efficiency of the heat pump cycle.

Not only additional heat is gained but also the discharge temperature of the heat pump compressor can now be reduced and the COP of the heat pump improved.

For the Boston, Mass. area the First Report indicated that the winters are rather severe and more than enough ice will be accumulated in winter time, which would have to be subsequently melted.

However, the heating requirement being partly satisfied by the heat recovered from the engine, the heat pump can afford to deliver less heat and therefore, less ice which results in a smaller ice storage bin, less insulation and less power input to the heat pump.

Moreover, the amount of ice could be reduced still further, in order to leave "room" for domestic hot water generation in the summer with the simultaneous generation of chilled water with a higher COP. In such cases the domestic hot water is generated by the heat, recovered from the engine, while the engine drives the heat pump.

2.2 SYSTEM COMPONENTS

The system will consist of energy centers, a distributions system and terminal units. The energy center will have seven dual fuel engine driven heat pumps with heat recovery from jacket water and exhaust gas, tube type ice-makers and ice makers circulating pumps, chilled water primary pumps between the storage bin and primary loop, chilled water pumps between the chillers and primary loop and hot water pumps.

The distribution system will have a primary loop and secondary station pumps for each building and sector.

Terminal units will be of two kinds: for the industrial areas, small offices and hotel guest rooms a 4-pipe fan coil unit system will be used, while for the commercial sector, like public assembly areas and landscape offices air handling units with enthalpy control, zoning system and distribution system will be used.

2.3 EQUIPMENT SIZE AND SELECTION

2.3.1 Engines

Seven dual fuel 12 cylinder engines Fairbanks Morse, opposed piston, 2400 hp each, 900 rpm, model 38 DD-8-1/8, with speed increasing gears.

2.3.2 Seven Vilters screw compressors VRS-130, with chillers of 1440 ton capacity each, condensers, valves and controls.

2.3.3 Fourteen units of ice makers Model FKE 705A, 525 ton ice per day capacity each. Manufactured by the Stal Company of Sweden, each consisting of five ice generators of the tube type including liquid accumulation tank and seven ice maker pumps 352 gpm, 7½ hp. each.

2.3.4 Pumps : The following table summarizes the pump selection.

TABLE B 2.1 PUMP SELECTION FOR HP-ICES

The Following Table Summarizes the Pump Selection ~~(TABLE B 3.5)~~

Pump Designation	No. of Pumps	Service	G.P.M	Head Pt.	BHP	Motor HP	%	Model Aurora
P-1	6	Heating	1817	150	63	75	82	6x8x15 Series 410
P-2	7	Chilled Water	2632	150	125	125	87	8x10x15 B Series 410
P-3	7	Condenser Water	5200	80	125	125	87	10x12x12 B Series 410
P-4	7	Ice Maker	352	50	6.5	7½	85	3x4x9 B Series 340
P-5	1	DHW	2387	150	125	125	73	8x10x15 B Series 410

2.4 SYSTEM OPERATION

Fig. B 2.1 is a schematic diagram of the system. Hot water returning from the community passes through the heat pump condenser where it is heated to 112°F then passes through the jacket water heat exchanger where its temperature is raised another 6°F and then goes to the exhaust silencer where it is heated up to 120°F.

Figures B 2.2 and B 2.3 describe the distribution system and the ice bin location.

The huge ice bin of approximately 15,000,000 cubic feet will be integrated with community buildings, but because of its size it will have to be built partly under the buildings and partly under the streets and public garden area as was done in the case of the public parking lots.

The basement level will be used for the seven main mechanical rooms and for the primary distribution loop.

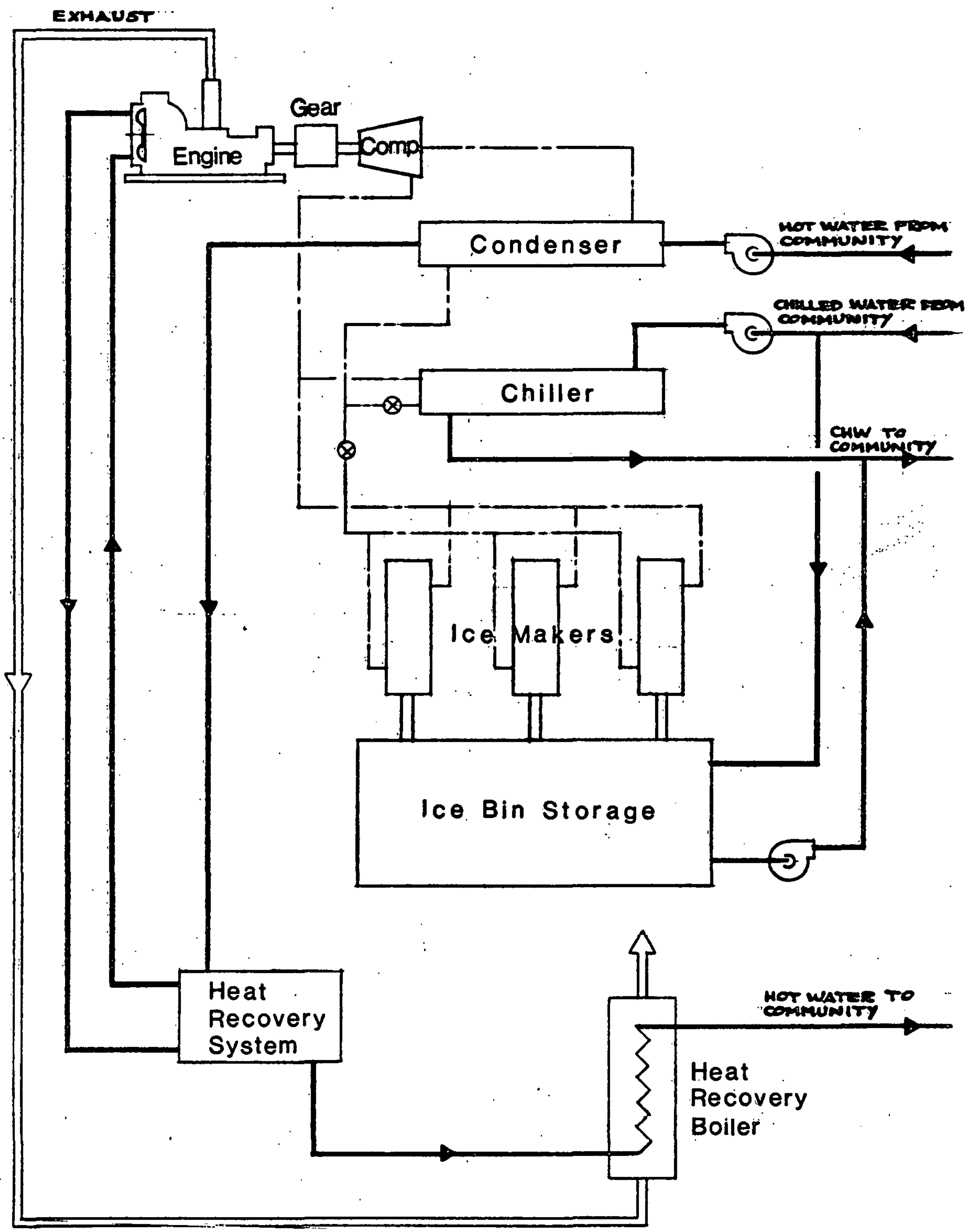


FIG.- B2.1

HP-ICES Ice Generating

Driven By Dual Fuel Engine

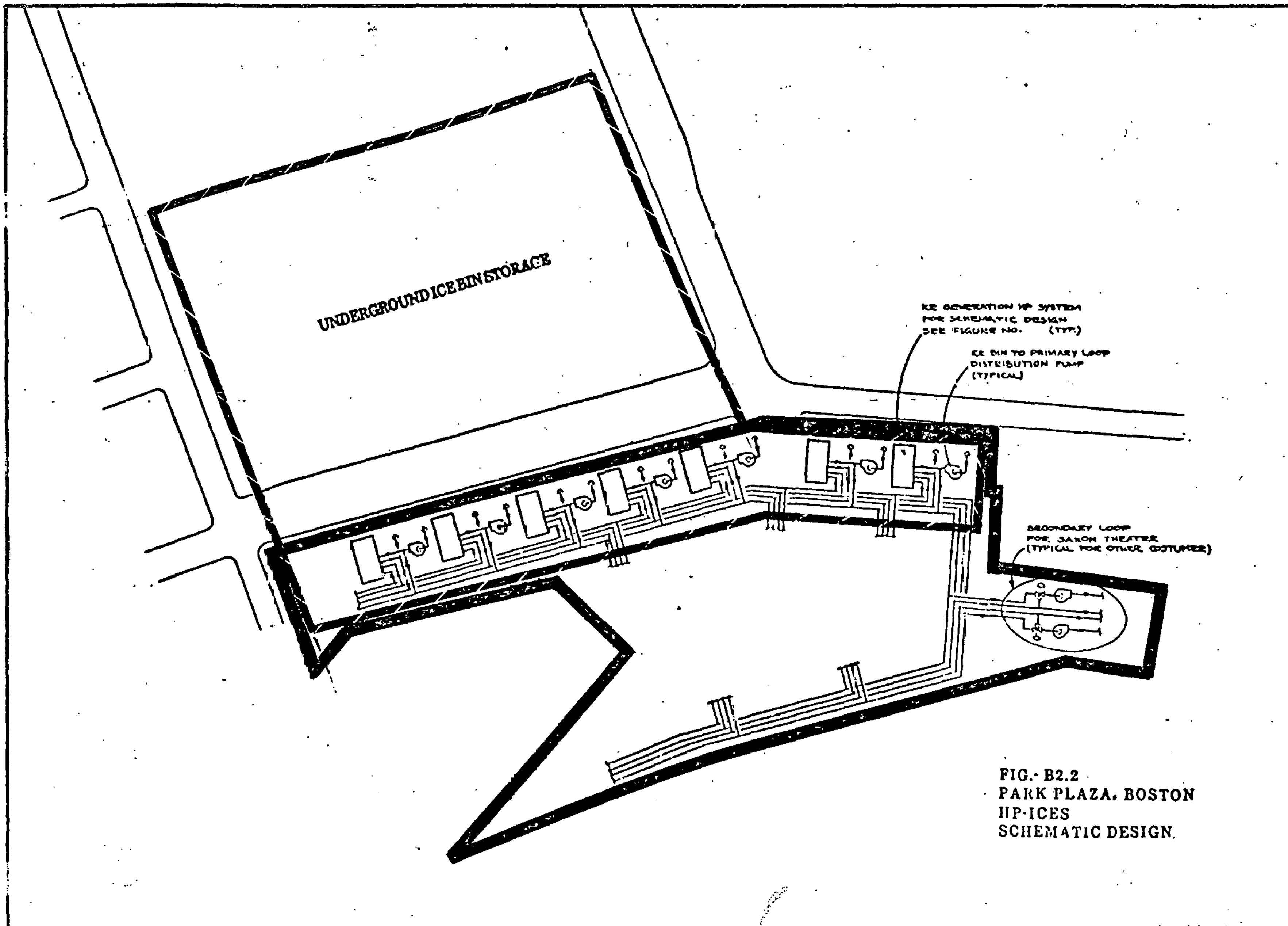


FIG.- B2.2
PARK PLAZA, BOSTON
HP-ICES
SCHEMATIC DESIGN.

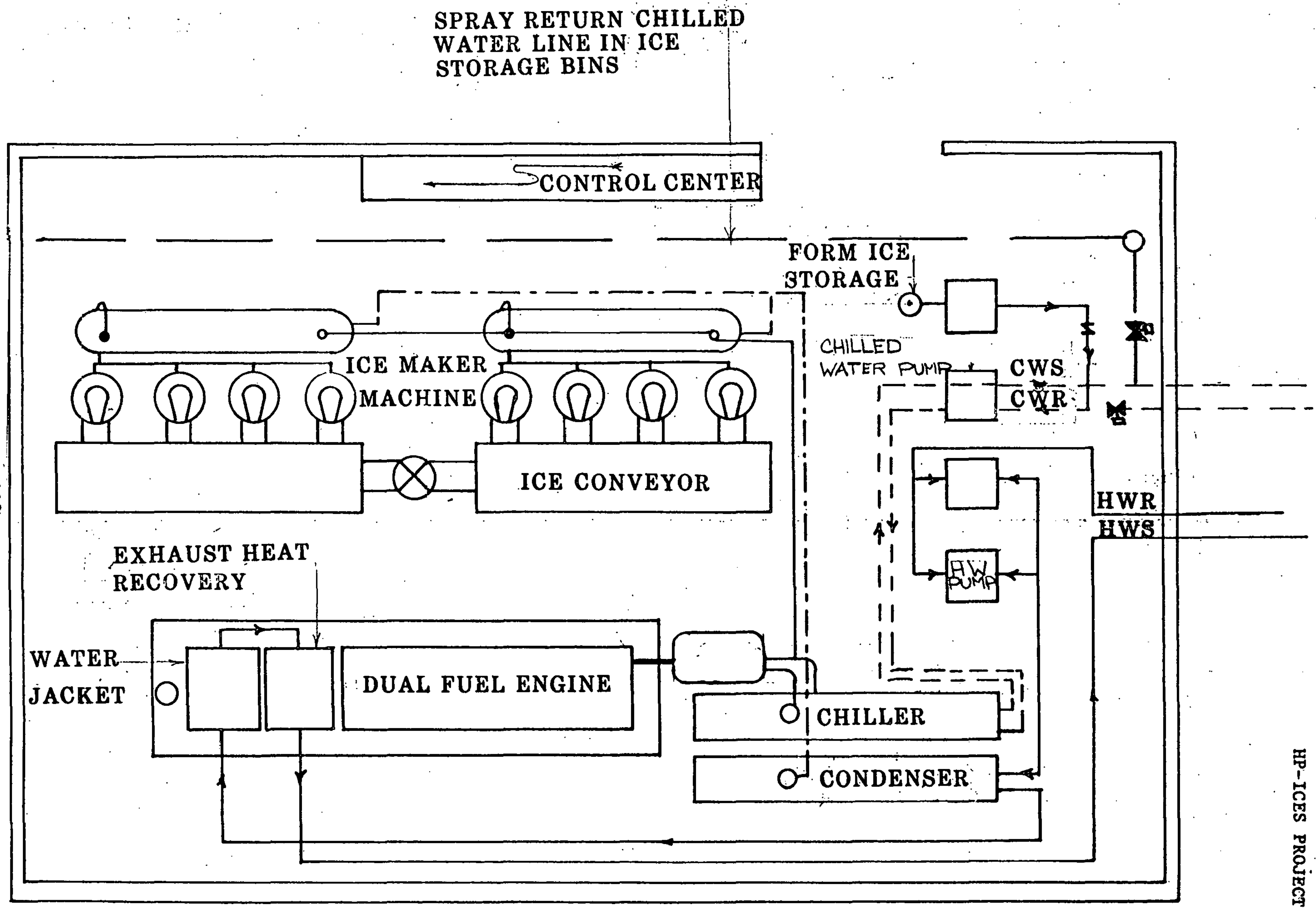


FIG. B.2.3 TYPICAL MECHANICAL ROOM FOR THE PARK PLAZA BOSTON

The ice making units will be located as close as possible to the ice storage bin to reduce problems associated with the conveying of the ice.

The ice is conveyed from the tube type ice maker to the storage bin by means of a conveyor.

The distribution system will be arranged in primary-secondary loops as for the Market Square Complex.

For both heating and cooling a primary-secondary loop system will be employed. The temperature difference in the cooling primary loop will vary between $\Delta T = 24^{o}F$ to $\Delta T = 16^{o}F$.

For heating the temperature difference in the primary will be $30^{o}F$. The primary-secondary system will reduce the power required for pumping and reduce the pipe sizes and enable the end user to select his own ΔT in the secondary system as required.

The operation of the compressors with heat rejection to cooling towers is not necessary in the Park Plaza case. All the cooling required is supplied by the heat pump.

To sum up, the overall heating, domestic hot water and cooling requirements of the Park Plaza community complex will thus be supplied with one system operating at a high efficiency both economically and energy-wise.

B.3 EXPECTED PERFORMANCE

3.1 INTRODUCTION

The ice generating heat pump as applied to Park Plaza Boston will be driven by a dual fuel engine.

The dual fuel engine will supply a considerable amount of recovered heat from both jacket water and exhaust and this will increase the overall amount of heat supplied by the heat pump and considerably increase the overall COP.

A series arrangement of the heat pump condenser, jacket water heat exchanger and exhaust silencer, in that order, will enable to reduce the condenser discharge temperature, improving thereby the heat pump performance, while the overall system supply temperature will remain the same.

The heat supplied by the heat pump condenser will amount to 60% of the overall heat supplied, while the jacket water and engine exhaust will contribute 40%. The condenser outlet temperature will be reduced by 10^{o}F and this will reduce the power input to the heat pump by 10% with an improvement of the COP by the same percentage.

On the other hand a deisel fuel engine driven heat pump while reducing the amount of heat handled by the heat pump, will also reduce the amount of ice generated by the heat pump to the point where the cooling generated will just be sufficient to cover the summer cooling requirement and ice bin losses.

This in turn will diminish the ice bin storage volume required by 20% relative to the storage required by an electrically driven heat pump.

The dual fuel engine driven heat pump will also eliminate the peak electrical demand both in winter and summer and will allow the generation of chilled water, if required, even during the summer peak load hours, eliminating the necessity of chilled water daily storage and associated with it additional investment costs.

The dual fuel engine drive also causes a reduction in energy consumption when compared with a conventional system.

3.2 GENERAL PERFORMANCE

The dual fuel engine selected is manufactured by Colt Industries, 12 cylinder Fairbanks Morse, Model 38DD-8-1/8. The performance of this engine is summarized in Table B 3.1.

TABLE B 3.1.

HEAT BALANCE OF A 12 CYLINDER FAIRBANKS MORSE DUAL FUEL ENGINE MODEL 38DD-8$\frac{1}{8}$

h.p. Required	Rated h.p.	h.p. Required as Percentage of Rated	Heat Input Btu/hp-hr	Heat Recovered Btu/hp-hr	New Work Output Btu/hp-hr	Total Recovered Heat Btu x 10^6
2295	2400	95.6	7500	3700	2545	8.5
2205	2400	92.0	7500	3700	2545	8.16
2115	2400	88.0	7500	3700	2545	7.82
1950	2400	81.0	7800	3988	2545	7.78

See Appendix for more information

Vilters Compressors VRS-130 selected utilizing R-22 and operating at 3540 RPM. The performance of these compressors at various conditions is summarized in Table B 3.2.

Column four of the table shows the performance of the compressors for

VILTER SCREW COMPRESSOR VRS-130 DRIVEN BY DUAL FUEL ENGINE - FAIRBANKS MORSE 38DD 8-1/8 12 CYL
(At Different Suction and Discharge Temperature)

TABLE B 3.2

SDT/ SST	130°F/20°F	125°F/20°F	120°F/20°F	115°F/20°F	130°F/35°F	120°F/35°F	105°F/35°F
Ton	1035	1090	1125	1193	1440	1575	1755
BHP	2430	2295	2205	2115	2610	2295	1950
Heat Rejection Btu/hr	18.6×10^6	18.92×10^6	19.1×10^6	19.7×10^6	23.9×10^6	24.74×10^6	26×10^6
COP_H	3.01	3.24	3.4	3.66	3.6	4.236	5.28
COP_C	2.01	2.24	2.4	2.66	2.6	3.236	4.28
Heat Recovered with Diesel Engine Btu/hr	9.0×10^6	8.5×10^6	8.16×10^6	7.82×10^6	-	8.5×10^6	7.78×10^6
Total Heat Rejection Btu/hr	27.6×10^6	27.42×10^6	27.26×10^6	27.52×10^6	-	33.24×10^6	33.78×10^6
Improved COP_{HI}	4.463	4.695	4.86	5.113	-	5.7	6.86

winter conditions at 20^oF SST and 120^oF SDT while column seven shows the performance of the compressors during the summer when the compressors operate at 35^oF SST and 120^oF SDT generating domestic hot water with the simultaneous generation of chilled water.

The table also shows the integrated performance of the compressors when operating in conjunction with the engines. The lower SDT is possible in view of the heat recovered from the jacket water and exhaust gases.

The table also gives the heat balance of the engine heat pump combination at various conditions giving the heat recovered and the COP. The minimum horsepower required is 2295 and occurs at 120^oF SDT and 35^oF SDT which condition will govern in selecting the size of the engine.

The COP_{HI} means "improved" COP. Thus from table B 3.3 the total heat rejected (by condenser and engine) is 33,240,000 Btu/hr and therefore

$$COP_{HI} = \frac{33,240,000}{2295 \times 2545} = 5.7$$

All the cooling requirements of 120672×10^6 Btu can be supplied by the heat pump while supplying heating and domestic hot water in winter time with a $COP_{HI} = 4.86$, heating in May and October with a COP = 5.7 and the domestic hot water in the summer with a COP = 6.86.

The above mentioned COP's are heating and "improved" COP's which means that they are obtained by dividing both the heat rejected by the heat pump and the heat recovered from the engine ~~by~~ the shaft work input to compressor.

Out of the 120672×10^6 Btu of cooling required 100370×10^6 Btu is generated in the form of ice in winter and after deducting $15,602 \times 10^6$ Btu losses, $84,768 \times 10^6$ Btu or 70% are available as "free" cooling.

Additional $9,841 \times 10^6$ Btu or 8% of cooling in the form of chilled water are generated in the months May and October while supplying heating required during these months.

The remainder 26,063 x 10^6 or 22% are generated during the summer while generating domestic hot water.

No daily storage system is required as the dual fuel engine incurs no electrical demand charge and the heat pump can be operated during peak summer days.

Thus 186,380 x 10^6 Btu of heating, 75,980 x 10^6 Btu of domestic hot water and 120,672 x 10^6 Btu of cooling are supplied annually at an expenditure of 142,480 x 10^6 Btu in the form of gas and 8387 x 10^6 Btu in the form of pilot oil. This yelds an annual overall COP of:

$$\frac{(186,380 + 75980 + 120672) \times 10^6}{(142480 + 8387) \times 10^6} = 2.54$$

as related to source energy. When related to shaft work the expenditure is 20,000,000 BHP-hr and the

$$COP_{overall} = \frac{383,032 \times 10^6}{20 \times 10^6 \times 2545} = 7.52$$

which is much higher than the already high COP of 5.12 for the electrically driven heat pump for the Market Square complex proving the desirability of the engine driven heat pump.

3.3 ANNUAL ENERGY BALANCE

Table B 3.3 summarizes the annual energy balance and shows the "cooling" energy generated both in the form of ice and chilled water.

During the winter months the system will operate at 120°F and 20°F SDT and SST respectively and with a COP_{HI} = 4.86. The heat pump will generate all the heating and domestic hot water required and will generate ice.

During the summer months the cooling required will be supplied mainly from the storage but also partly by the heat pump. The domestic hot water will be generated by the heat recovered from the engine allowing the operation of the compressor at 105°F SDT and 35°F SST for the generation of chilled water. The condenser water will be heated in the engine heat recovery system to the set point temperature.

The "cooling" energy deposited in the storage exceeds the energy withdrawn from the storage by about 16.5% which means that a considerable amount of ice generated will be allowed to melt by heat leakage into the ice storage bin permitting a considerable reduction in the ice bin insulation with a corresponding reduction in the initial investment. We can afford such an approach since ice will be generaged anyway as required by the heat pump on heating the buildings of the community.

It is also seen from the storage accumulation column of Table B 3.4 that the maximum accumulation of ice occurs in April and this month's accumulation will determine the volume of the ice storage bin which will be

$$\frac{88190 \times 10^6 \text{ Btu}}{162 \times 36} = 15,122,000 \text{ cu. ft.}$$

Fig. B 3.2 is a bar graph representation of the energy balance as given in Table B 3.4. We note that because of the large storage in the HP-ICES system, the performance will not be appreciably altered by the occurrence of peak heating, peak cooling, and simultaneous operation modes. Because of this, it is sufficient to consider typical average heating and cooling loads, and not consider in detail the peak loads.

The storage volume as determined above will take care of the cooling requirement during the summer with allowance for heat leakage of 15,602 x 10^6 Btu This allowance will determine the wall insulation of ice bin.

TABLE B 3.3

HP-ICES ANNUAL ENERGY BALANCE										
Month	Energy Required for Heating Btu x 10^6	Energy Required for DHW Btu x 10^6	Energy Required for Cooling Btu x 10^6	COOLING GENERATED Btu x 10^6			Deposit Supply to Storage Btu x 10^6	DEBIT Btu x 10^6		Storage Accumulation Btu x 10^6
				120/20° COP_{HI}=4.86	120/35° COP_{HI}=5.7	105/35° COP_{HI}=6.86		Taken Out of Storage	Losses	
November	19948	6245	0	12934.8	0	0	12934.8	0	1024.4	11910.4
December	32519	6453	0	19245.4	0	0	19245.4	0	1024.4	30131.4
January	35992	6453	0	20960.5	0	0	20960.5	0	1024.4	50067.5
February	32155	5829	0	18757.5	0	0	18757.5	0	1024.4	67800.6
March	27987	6453	0	17007.4	0	0	17007.4	0	1024.4	83783.6
April	16971	6245	6034	11464.7	0	0	5430.7	0	1024.4	88190
May	6881	6453	12067	0	3906.47	4026	0	4134	1576	82480
June	1290	6245	15687	0	0	4701	0	10986	1576	69918
July	0	6453	26548	0	0	4026	0	22522	1576	45820
August	0	6453	26548	0	0	4026	0	22522	1576	21722
September	2183	6245	22928	0	0	5258	0	17670	1576	2476
October	10454	6453	10860	0	5935	4026	0	900	1576	0
Annual Total	186,380	75,980	120,672	100,370.3	9,841.47	26,063	94,336.3	78,734	15602	–

1) Peak cooling load is 11,680 ton with diversity – 10,000 ton

2) Peak heating load 165,000,000 Btu/hr

3). Peak DHW load 35,800,000 Btu/hr.

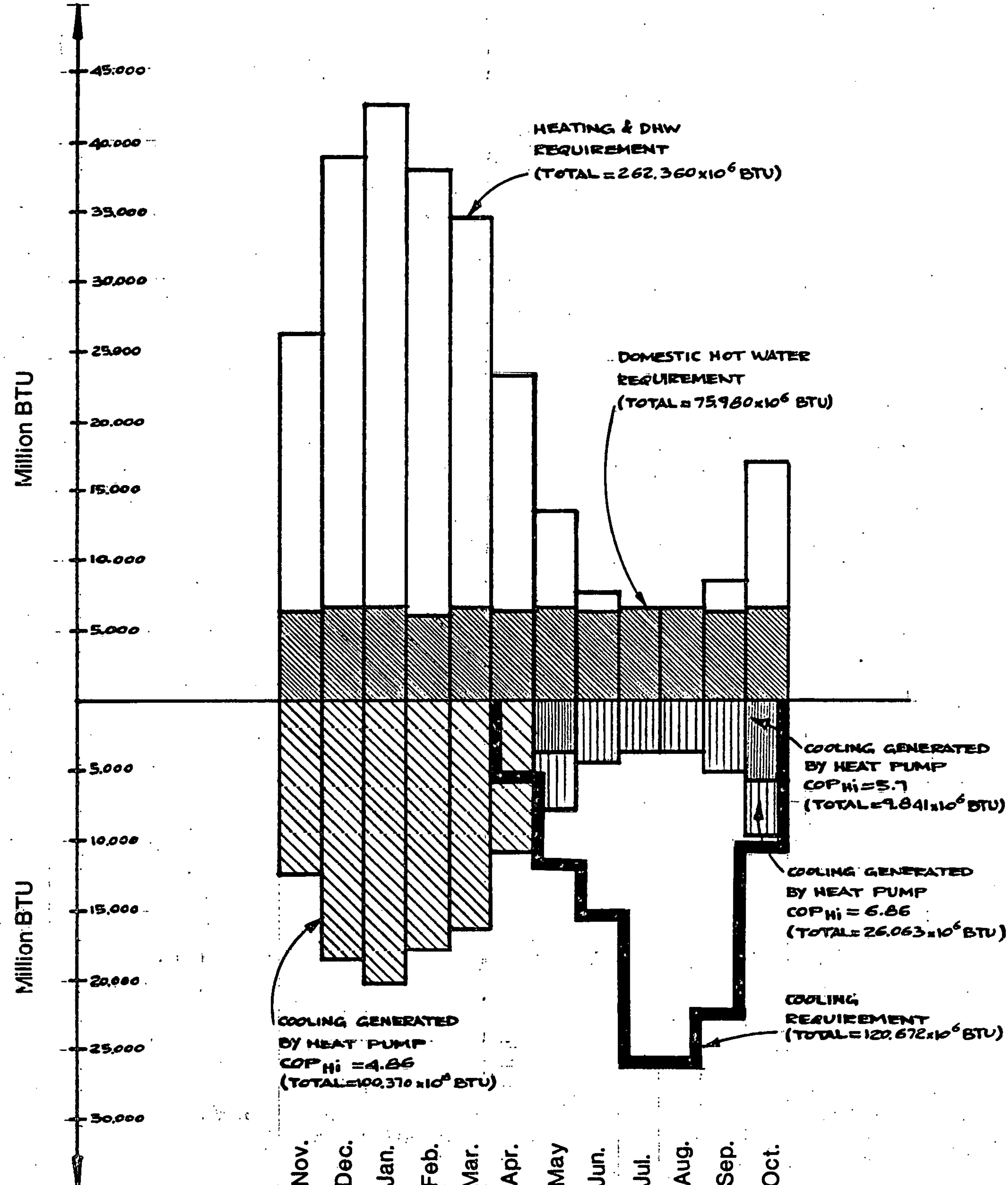


FIG.- B3.2

HP-ICES Annual Energy Balance

Let us assume that the ice bin storage will be located underground with a depth of 25 ft. and a square shaped base. That will lead to a length dimension of 777 feet and a surface area of a 1,285,158 square feet. The average ground temperature was assumed as 57^o and the resulting U-factor is 0.068 Btu/hr-^{o}F. Total monthly heat leakage is 15602×10^6 Btu. (Table B 3.3) Assuming 30 days in a month:

$$U = \frac{1576 \times 10^6 \text{ Btu/month}}{30 \text{ days} \times 24 \text{ hr/day} \times 1285158 \text{ ft}^2 \times 25^o \text{ (DT)}}$$

$$= 0.068 \text{ Btu/hr-}^o\text{F}$$

This means average R = 14.7

Concrete 12" thick has R = 0.5

According to ASHRAE Fundamentals, Chapter 24, the heat losses for an underground uninsulated wall below 7 ft. deep is 0.069 Btu/hr-ft^2-^{o}F.

At a depth of 25 feet the heat loss would be a lot lower. Therefore, it is reasonable to take as any average the value of 0.068 Btu/hr ft^2 ^{o}F for the storage bin losses.

3.4 ANNUAL ENERGY CONSUMPTION

Table B 3.4 summarizes the energy consumption for the dual fuel engine at different conditions of operation as specified in the same table.

The engine operates on natural gas and pilot oil and to satisfy the winter and summer load seven engine-compressor units are required.

In winter operation the heat rejection of one engine-compressor unit is 27.26×10^6 Btu/hr and seven units will reject 190.82×10^6 Btu/hr, enough to satisfy the heating and domestic hot water load of 200.8×10^6 Btu, taking into account a certain load diversity.

TABLE B 3.4

ANNUAL ENERGY CONSUMPTION											
	Heating Generation		Cooling Generation			Consumption BHP-H Btu x 10^6	Consumption Btu x 10^6	Gas Btu x 10^6	Pilot No. 2 Oil Btu x 10^6	Gal. of No. 2 Oil	Cu. Ft. of Gas x 10^6
	Space Heating	DHW	120/20° COP_{HI} = 4.36	120/35° COP_{HI} = 5.7	105/35° COP_{HI} = 6.86						
November	19948	6245	12934.8	0	0	2.1177	15882.6	15018.6	864.00	6171	15.0186
December	32513	6453	19245.4	0	0	3.151	23631.5	22345.9	1285.60	9183	22.3459
January	35992	6453	20960.5	0	0	3.432	25737.3	24337.04	1400.260	10002	24.33704
February	32155	5829	18757.2	0	0	3.071	23032.3	21779.33	1252.97	8950	21.77933
March	27987	6453	17007.4	0	0	2.7840	20883.4	19747.53	1135.870	8113	13.747.53
April	16971	6245	11464.7	0	0	1.877	14077.5	13311.684	765.816	5470	13.311684
May	6881	6453	C	3906.47	4026	0.8434	6440.5	6062.435	378.064	2700	6.062436
June	1293	6245	0	0	4701	0.4316	3366	3150.2	215.8	1542	3.1502
July	0	6453	0	0	4026	0.3631	2882	2697.45	184.55	1318	2.69745
August	0	6453	0	0	4026	0.3691	2882	2697.45	184.55	1318	2.69745
September	2183	6245	0	0	5258	0.4822	3765	3523.65	241.35	1725	3.52365
October	10434	6453	0	5935	4026	1.09	8287	7808.45	478.55	3418	7.80845
Total Annual	186380	75930	96,971	9841.47	26063	20.0183	150867.1	142480	8387.38	59910	142.47972

HP-ICES PROJECT

In the summer one unit will produce 1755 ton of refrigeration and seven units will supply 12,285 ton which exceed the required 10,000 ton.

Usually one unit will be sufficient to generate domestic hot water with a second unit assisting.

3.5 AUXILIARY ENERGY CONSUMPTION

Table B 3.5 shows the energy consumption of the auxiliaries. The boxed numbers in the upper left corners in the first three columns represent the hours of operation.

Table B 3.6 shows the annual energy consumption both for engines and auxiliaries.

Fig. B 3.2 is a bar graph representation of the total annual energy comsumption.

Tables B 3.7 and B 3. 8 show the resource energy utilization.

TABLE B 3.5 HP-ICES ENERGY CONSUMPTION FOR AUXILIARIES

Energy Input KWH								
Heating Pumps		DHW Pumps		Chilled Water Pumps		Ice Maker Pumps	Total Electrical Input
691	194862	175	16318		0	24197.6	235,377.6
742	209244	180	16785		0	25939.2	251,968.2
744	209808	180	16785		0	26006.4	252,599.4
672	189504	163	15200		0	23492	228,196
735	207270	180	16785		0	25704	249,759
217	61194	175	16318	117	76372	8271	162,155
147	41454	180	16785	176	114834	0	173,123
57	16074	175	6318	263.5	172000	0	204,392
0	0	180	16785	263.5	172000	0	188,785
0	0	180	16785	261.5	172000	0	188,785
115	32430	175	16318	263.2	172000	0	220,748
209	58938	180	16785	117	76372	0	152,095
4329	1220778		197967		955628	133610.9	2,507,983.2

The number on the corner is the operating hours

ANNUAL ENERGY CONSUMPTION
FOR ENGINE AND AUXILIARIES TO
SUPPLY HEATING, DHW AND COOLING

TABLE B 3.6

Month	Cu. Ft. of N. Gas	Pilot Oil Gal. Of No. 2 Oil	KWH Electric
November	15,018,600	6,171	235,378
December	22,345,900	9,183	251,968
January	24,337,040	10,002	252,600
February	21,772,330	8,950	228,196
March	19,747,530	8,113	249,759
April	13,311,684	5,470	162,155
May	6,062,436	2,700	173,123
June	3,150,200	1,542	204,392
July	2,692,450	1,318	188,785
August	2,697,450	1,318	188,785
September	3,523,650	1,725	220,748
October	7,808,450	3,418	152,095
Total	142,479,720	59,910	2,507,984

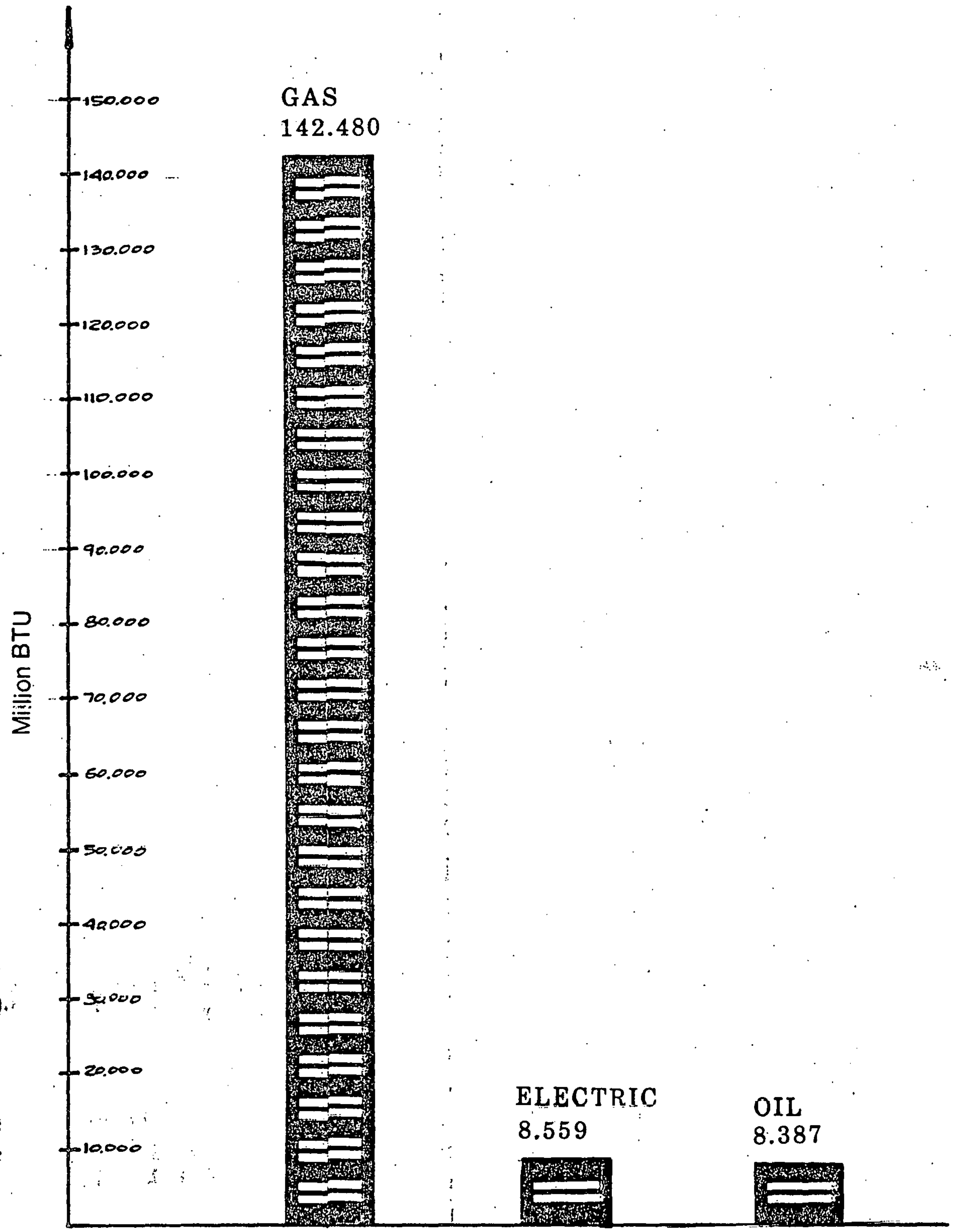


FIG.- B3.3

HP-ICES Annual Energy Consumption

Driven BY Dual Fuel Engine

TABLE B 3.7

HP-ICES

ANNUAL FUEL AND ENERGY CALCULATION FORM 12-1

		Building/Project Energy Req'ments		Fuel and Energy Supplied to Site								
Line	Column	A1	A2	B1	B2	B3	B4	B5	B6	B7	B8	B9
	Function	Thermal 10^6 Btu	Electric 10^3 KWH	Coal 10^6 Btu	Gas 10^6 Btu	Light Oil 10^6 Btu	Heavy Oil 10^6 Btu	Elec. Win. 10^3 KWH	Elec. Sum. 10^3 KWH	Elec. Ann. 10^3 KWH	Other	Other
1	Heating	186380						1380.06				
2	Cooling	120672							1127.93			
3	Water Heating	75980										
4	HVAC Auxiliaries											
5	Lighting											
6	Elevators											
7	Computers											
8	Cooking											
9	Process											
10	Other											
11	Other											
12	Other											
13	Total Carry Fwd. to Form 12-2				142480	8387.38		1380.06	1127.93			

TABLE B 3.8

ANNUAL FUEL AND ENERGY CALCULATION FORM 12-2

	Annual Fuel and Energy Calculation Form 12-2			C.O. Total	RUF From	Fuel and Energy Resources Used on Site and Off Site To Meet Energy Requirements of Building/Project						
Line	Fuel and Energy Supplied to Site			From Form 12-1 Line 13	Supplier or From Tables	C1 S. Tons Coal	C2 MCF Nat'l	C3 BBL Crude Oil	C4 Grams U-235	C5 10^3 KWH Hydro	C6 Other	C7 Other
14	Fuel Oil, Light			8387.38	0.198			1660.70				
15	Fuel Oil, Heavy											
16	Gas	Nat'l	MCF	142480	1.10		156728					
		Oil	BBL		0.0028			398.95				
17	Coal											
18	Elec. Winter			1380.06								
		Coal	S. Tons		0.03	41.40						
		Gas	MCF									
		Oil	BBL		1.19			1642.26				
		Nuc	Grams		11.96				16505.50			
		Hydro	10^3KWH		0.05					69.00		
		Other										
19	Elec. Summer			1127.93								
		Coal	S. Tons		0.03	33.84						
		Gas	MCF		0.62		699.30					
		Oil	BBL		1.29			1445.03				
		Nuc	Grams		8.11				9147.50			
		Hydro	10^3KWH		0.06					67.70		
		Other										
20	Elec. Annual											
		Coal	S. Tons									
		Gas	MCF									
		Oil	BBL									
		Nuc	Grams									
		Hydro	10^3KWH									
		Other										
21	(Other)											
22					Total Resources	75.24	157427.30	5146.94	25653	136.70		

3.6 COMPARATIVE CONVENTIONAL SYSTEM

A comparative conventional system was selected to meet the peak cooling and heating requirements of the entire community complex. (See Fig. B 3.3).

3.6.1 Cooling System

The peak cooling load of about 10,000 ton will be covered by four centrifugal carrier chillers operating with R-11 at SST = 35^{o}F and SDT = 105^{o}F and having the following capacities and power inputs:

a.	5000 ton	3996 kW
b.	1920 ton	1596 kW
c.	1920 ton	1596 kW
d.	1000 ton	772 kW
Total	9840 ton	7960 kW

These units of various sizes were selected to enable the overall system flexibility at full and part load and to improve performance by avoiding, as fas as possible, the operation of any one unit at part load.

The chilled water distribution system will consist of a primary-secondary loop operating with a DT = 16^{o} (56^{o}F - 40^{o}F) that will reduce the gpm and pumping power.

The following table summarizes the pump selection.

TABLE B 3.9 - PUMP SELECTION FOR CONVENTIONAL SYSTEM

Pump No.	No. of Pumps	Water Service	g.p.m	Head Pt.	Power Input BHP	Model Aurora
P-1	5	Chilled	3500	150	175	10 x 12 x 15B - 410
P-2	5	Condense	6000	80	150	10 x 12 x 12B - 410

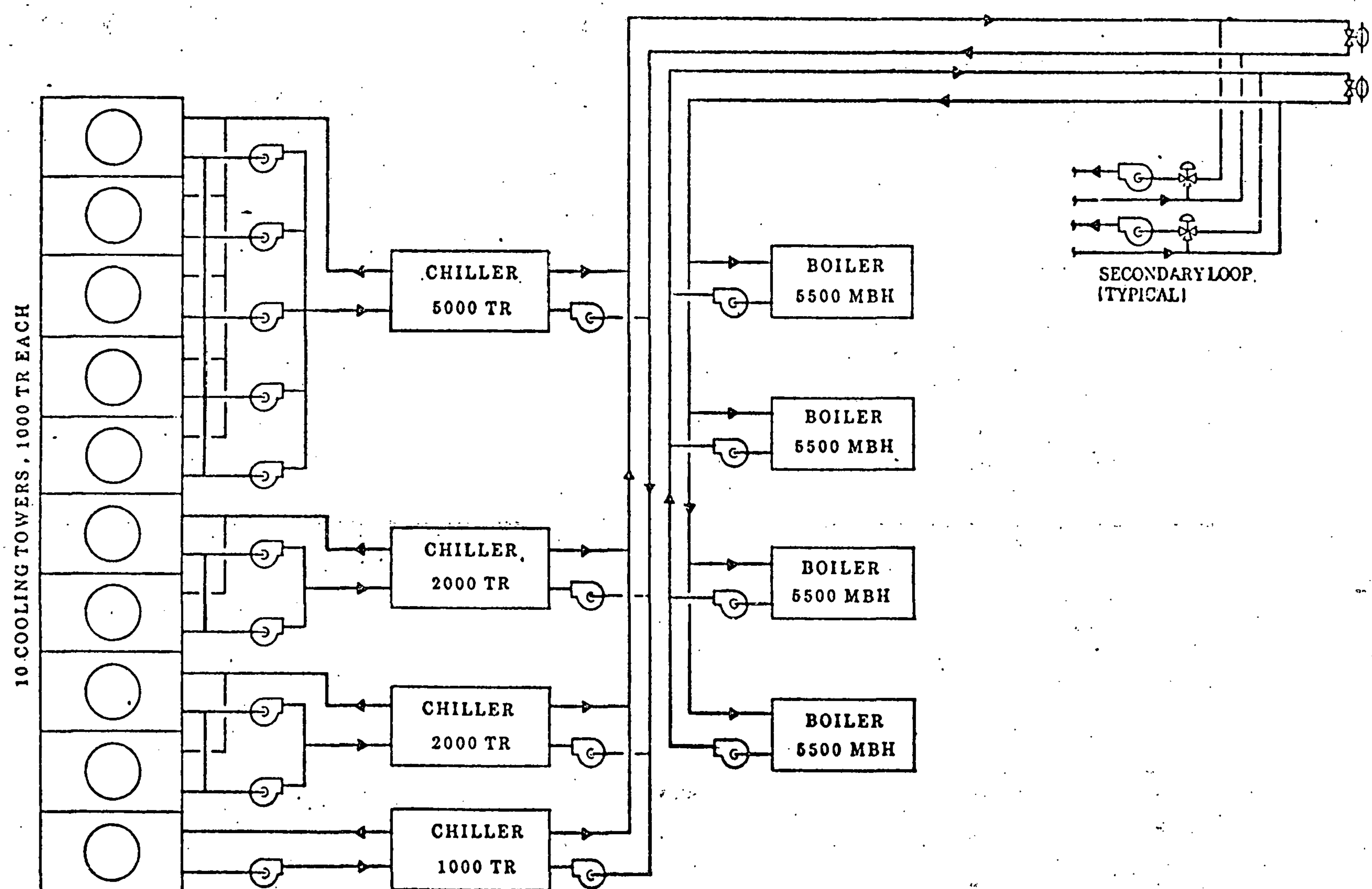


FIG.- B3.4
PARK PLAZA, BOSTON
CONVENTIONAL SYSTEM

COMPRESSOR DRIVE ELECTRICAL ENERGY CONSUMPTION

TABLE B 3.10

Month	CONSUMPTION %	CONSUMPTION KWH*	CONSUMPTION Cost $	DEMAND %	DEMAND Peak KW	DEMAND Cost $	Total Cost $
January	0	0	0	0	0	0	
February	0	0	0	0	0	0	
March	0	0	0	0	0	0	
April	5	402,500	21,340	50	4673	22,664	44,004
May	10	805,000	42,679	75	7010	33,998	76,672
June	13	1,046,500	55,482	100	9345	45,332	100,814
July	22	1,771,000	110,333	100	9345	54,575	164,908
August	22	1,771,000	110,333	100	9345	54,575	164,908
September	19	1,529,500	95,288	75	7010	40,938	136,226
October	9	724,500	45,136	50	4673	27,790	72,426
November	0	0	0	0	0	0	
December	0	0	0	0	0	0	
Total	100	8,050,000	$480,591			$279,372	$759,963

* 9577 TONS x 0.8 KW/TON x 1050 EFLH
= 8,050,000 KWH

TABLE B 3.11 REFRIGERATION AUXILIARIES ELECTRICAL ENERGY CONSUMPTION

Month	CONSUMPTION			DEMAND			TOTAL COST $
	%	KWH	Cost $	%	Peak KW	Cost $	
January	0	0		0	0		
February	0	0		0	0		
March	0	0		0	0		
April	8	240,000	12,724	50	830	4,025.5	167,695
May	12	300,000	19,086	100	1660	8,051	27,137
June	18	540,000	28,629	100	1660	8,051	36,680
July	18	540,000	33,642	100	1660	9,694.4	43,336.4
August	18	540,000	33,642	100	1660	9,694.4	43,336.4
September	18	540,000	33,642	100	1660	9,694.4	43,336.4
October	8	240,000	14,952	50	830	4,847.2	19,799.2
November	0	0		0	0		
December	0	0		0	0		
Tot. Annual		3,000,000	$76,317			$54,060	$230,377

Demand = 1660 KW

Use = 1050 x1.7 = 1785 say 1800

Equivalent Full Load Hours

Annual KWH = 1660 x 1800

= 2,988,000 KWH

Account for Miscellaneous say 3,000,000 KWH

3.6.2 Heating System

To meet the heating requirement of 165,000,000 Btu/hr and domestic hot water requirement of 35,800,000 Btu/hr, four boilers of 55,000,000 Btu/hr each were selected.

a. Heating Requirement $= 186,380 \times 10^6$ Btu

Energy Consumption:

Oil $= \dfrac{186,380 \times 10^6}{0.7 \times 140,000}$

$= 1,902,000$ gallons

Gas $= \dfrac{186,380 \times 10^6}{1000 \times 1000 \times 0.7}$

$= 266,260 \text{ M ft}^3$

b. DHW Requirement

Energy Consumption: $75,980 \times 10^6$ Btu

Oil $= \dfrac{75,980 \times 10^6}{0.7 \times 140,000}$

$= 775,310$ gallons

Gas $\dfrac{75,980 \times 10^6}{0.7 \times 1000 \times 1000}$

$= 108,550 \text{ M ft}^3$

The primary loop will have a temperature difference of $240^\circ - 160^\circ = 80^\circ$F and four primary pumps where selected.

The flow rate of each pump will be

$$\frac{220 \times 10^6}{4 \times 500 \times 80} = 1375 \text{ gpm, say } 1400 \text{ gpm}$$

From the project layout the estiamted head will be 125 feet. Thus, four pumps were selected having the following characteristics:

gpm = 1400; head = 125 ft; hp = 60; efficiency 71%;

rpm = 1150; Model = Aurora 8 x 10 x 17B, Series 410;

Total horse power = 240 hp = 240 x 0.746 = 179 kW

Power Consumption for Auxiliaries

a. Boiler auxiliaries, 73.3 kW per boiler, 4 x 73.3 = 293.3 kW

b. Primary heating pumps = 179.0 kW

c. Equivalent full load hours for boilers

$$= \frac{\text{Heating \& DHW requirements}}{\text{Heating \& DHW demand}}$$

$$= \frac{262{,}360 \times 10^6}{200{,}800{,}000} = 1306 \text{ hours.}$$

For the heating pumps the number of operating hours wi taken as the total number of hours when the temperature is below 57°F Electric power consumption is thus:

Boiler auxiliaries	= 293.3 kW x 1306 hours	= 383,050 kWh
Heating pumps	= 179.0 kW x 4329	= 775,064 kWh
	Total	= 1,158,114 kWh

See Tables B3.12 and B3.13 for resource energy utilization.

Table B 3.12

CONVENTIONAL SYSTEM

ANNUAL FUEL AND ENERGY CALCULATION FORM 12-1

		Building/Project Energy Req'ments		Fuel and Energy Supplied to Site								
Line	Column	A1	A2	B1	B2	B3	B4	B5	B6	B7	B8	B9
	Function	Thermal 10^6 Btu	Electric 10^3 KWH	Coal 10^6 Btu	Gas 10^4 Btu	Light Oil 10^6 Btu	Heavy Oil 10^6 Btu	Elec. Win. 10^3 KWH	Elec. Sum. 10^3 KWH	Elec. Ann. 10^3 KWH	Other	Other
1	Heating	186380			266260							
2	Cooling	120672							11050			
3	Water Heating	75980			108550							
4	HVAC Auxiliaries											
5	Lighting											
6	Elevators											
7	Computers											
8	Cooking											
9	Process											
10	Other											
11	Other											
12	Other											
13	Total Carry Fwd. to Form 12-2				374810				11050			

Table B 3.13

ANNUAL FUEL AND ENERGY CALCULATION FORM 12-2

Annual Fuel and Energy Calculation Form 12-2				C.O. Total	RUF From	Fuel and Energy Resources Used on Site and Off Site To Meet Energy Requirements of Building/Project						
Line	Fuel and Energy Supplied to Site			From Form 12-1 Line 13	Supplier or From Tables	C1 S. Tons Coal	C2 MCF Nat'l	C3 BBL Crude Oil	C4 Grams U-235	C5 10^3 KWH Hydro	C6 Other	C7 Other
14	Fuel Oil, Light											
15	Fuel Oil, Heavy											
16	Gas	Nat'l	MCF	374.81	1.10		412.29					
		Oil	BBL		0.0028		1.050					
17	Coal											
18	Elec. Winter			1158.12								
		Coal	S. Tons		0.03	34.75						
		Gas	MCF									
		Oil	BBL		1.19			1378				
		Nuc	Grams		11.96				13851			
		Hydro	10^3 KWH		0.05					57.9		
		Other										
19	Elec. Summer			11050								
		Coal	S. Tons		0.03	331.50						
		Gas	MCF		0.62		6851					
		Oil	BBL		1.29			14254.50				
		Nuc	Grams		8.11				89615.50			
		Hydro	10^3 KWH		0.06					663		
		Other										
20	Elec. Annual											
		Coal	S. Tons									
		Gas	MCF									
		Oil	BBL									
		Nuc	Grams									
		Hydro	10^3 KWH									
		Other										
21	(Other)											
22	Total Resources					366.25	7264.34	15632.50	103466.50	720.90		

3.7 RESOURCE ENERGY UTILIZATION

The resource energy utilization is summarized in Table B 3.14 below.

TABLE B 3.14 ENERGY RESOURCES COMPARATIVE ANALYSIS

	ANNUAL FUEL AND ENERGY RESOURCES ON SITE AND OFF SITE TO MEET ENERGY REQUIREMENT OF THE PROJECT				
	C1	C2	C3	C4	C5
System	S. tons coal	MCF Nat'l	BBL Crude Oil	Grams U-235	10^3 kwh Hydro
Conventional	366.25	7264.34	15632.50	103466.50	720.90
HP-ICES	75.24	157427.30	5146.94	25653	136.70

Thus the HP-ICES uses much less coal, **oil, uranium and hydroelectric** power than the conventional system, **but uses more gas.**

From Table B 3.8 (first column) representing the energy utilization of the HP-ICES, the system uses 8387.38 x 10^6 Btu in the form of pilot No. 2 oil (see also Table B 3.4) and 142,480 x 10^6 Btu in the form of gas. Also the system uses 2,508,000 kwh for auxiliaries (Table B 3.5).

Thus the raw source energy is

$$(8{,}387.38 + 142{,}480) \times 10^6 \text{ Btu} + \frac{2{,}508{,}000 \times 3413}{0.31} \text{ Btu} =$$

$$178{,}480 \times 10^6 \text{ Btu}$$

The conventional system uses 266,260 M cu. ft. of gas plus 108,550 M cu. ft., a total of 374,810 M cu. ft. of gas. It also uses 11,050,000 kwh for refrigeration and auxilaries drive (Tables B 3.10 and B 3.11) and 1,158,114 kwh for boiler auxiliaries and heating pumps. Its raw source energy is thus

$$374{,}810 \times 10^6 \text{ Btu} + \frac{12{,}208{,}114 \times 3413}{0.31}$$

$$= 509{,}217, \times 10^6 \text{ Btu}$$

These results are tabulated in Table B 3.15.

TABLE B 3.15 ANNUAL RAW SOURCE ENERGY UTILIZATION COMPARISON

System	Total Energy Used Btu x 10^6	
Conventional	509,217	
HP-ICES	178,480	

For First and Second Law analyses see Section C.

B.4 EXPECTED ECONOMICS FOR THE PARK PLAZA PROJECT

4.1 INTRODUCTION AND ASSUMPTIONS

The method used here for evaluating and comparing the economics of the HP-ICES system and a conventional system of the same heating and cooling capacity will be a life-cycle cost analysis.

The following are the assumptions made in evaluating the economics.

(a) The Conventional System:

The conventional system will be a central system with conventional centrifugal chillers and gas fired boilers. Although the methods of generation of heating and cooling differ in the two systems, the distribution systems were assumed to be the same.

(b) The First Cost:

The first cost of each system includes the generation equipment and distribution system, but does not include the end users equipment (air handling units) or the cost of land.

The reasons for this are similar to those discussed in detail for the Market Square project in an earlier section.

(c) Equipment Prices:

These were obtained from the following manufacturers: Compressors from Dunham-Bush and Vilters; ice makers from Trubo Co., Texas and Stal Co., Sweden; boilers from Cleaver-Brooks Co., and pumps from Aurora Co.

(d) Energy Prices:

Energy prices were calculated according to the Boston Edison Co. General Rate G-1:

Demand	November-June	\$4.85/kW
	July-October	\$5.84/kW
Energy	November-June	\$0.053/kWh
	June-October	\$0.063/kWh
Gas	Boston Gas	\$4.00/M cu.ft.
No. 2 Fuel Oil		\$0.57/gallon
Lubricating Oil		\$1.00/gallon

Details of the energy costs will be given later.

(e) System Life, Mortgage Period, Salvage Value, etc.

The first of the equipment is assumed to be 20 years and the life of the distribution system and ice storage bin 40 years. The mortgage period and the period of the economic analyses are both assumed to be 20 years. The depreciation life of the overall system (for the purpose of tax deductions) is assumed to be 20 years. For certain kinds of ownership of the system, it may be possible to use a smaller depreciation period and increase the tax savings. However, the results will be given here only for a depreciation life of 20 years.

The salvage value of the equipment after 20 years will be assumed to be 20% of the first cost. For the ice bin in the HP-ICES system, the salvage value will be assumed to be 40% of the initial cost of the ice bin (i.e. the cost of excavation and ice bin construction). The reason for assuming this high value is that after the useful life of the HP-ICES system, it is possible to put the bin to good use by converting it into a parking lot without too large an additional expense. For the complete HP-ICES system, including the equipment and the storage, the overall percent salvage value is found to be about 30%.

(f) Cost of Insurance, Maintenance, Labor, Administration, etc.

Insurance costs are assumed to be about 0.5% of the first cost. The costs of maintenance, labor, administration, etc. are estimated for each project separately.

(g) Discount Rate and Inflation Rate

The general inflation rate is assumed to be an average of 6% per year. The discount rate used in evaluating the present worth of money is assumed to be 2% more than the inflation rate, i.e. 8% per year.

(h) Taxes

The effective income tax bracket for the commercial system, assuming ownership by a private corporation, is assumed to be 46%. The tax deductions that a business can make are all allowed for.

The property tax is assumed to be 1% per year.

4.2 DETAILS OF COST ESTIMATES FOR THE HP-ICES SYSTEM IN THE PARK PLAZA PROJECT

The following are details of the cost estimates for the HP-ICES system.

a. First Cost for the HP-ICES System

1.	Seven units of dual fuel engine (12 cylinder Fairbanks Morse Model 38DD8 1/8, 900 rpm) mounted on common base coupled to a speed-increasing gear to drive a screw compressor, with engine auxiliaries and heat recovery equipment installation included: $800,000 for each of 7 units	=$ 5,600,000
2.	Seven units Vitler Screw Compressor VRS - 130 (without motor) coupled with oil separator, double-bundle condenser chiller and all valves and control required: $120,000 for each of 7 units	= 840,000
3.	14 units of Stal ice maker equipment FKE 705A including 5 ice generator each; including accumulation liquid tank and all auxiliaries required. $290,000 for each of 14 units	= 4,060,000
4.	Six pumps for heating; capacity 1817 gpm, 100 hp	= 45,000
5.	Seven chilled water pumps 2632 gpm, 125 hp	= 60,000
6.	1 DHW pump 2387 gpm, 125 hp	= 9,000
7.	Seven ice-maker pumps 352 gpm, 7-1/2 hp	= 10,000
8.	Excavation and construction of storage volume of 15,000,000 cu.ft., at $1 per cu.ft.	= 15,000,000
9.	Primary distribution pipes network for chilled water, bin system and heating system	= 650,000
10.	Electric installation and connection for 800kW service	= 100,000
11.	Sub Total	= 26,374,000
12.	Contingency 10%	= 2,637,400
	TOTAL FIRST COST	=$29,011,400

b. <u>Operating Cost for the HP-ICES System</u>

1.	Energy Cost	=	$ 777,693
2.	Maintenance cost for seven dual fuel engine @ $5/HP-Year	=	84,000
3.	Lubricating oil consumption and cost 0.5 gal/hr of 2400 hp engine Oil Consumption - 1500 EFLH x 0.5 gal/hr/engine x 7 engines= 5,250 gal/year Cost = 5,250 gal x $1/gal	=	5,250
4.	Maintenance cost for seven laborers @ $20,000 per person	=	140,000
5.	Station engineer $25,000 per year	=	25,000
6.	Administration -6 people @ $15,000 per person	=	90,000
7.	Maintenance (1% of first cost of equipment)	=	50,000
	TOTAL OPERATING COST FOR HP-ICES SYSTEM	=	$1,171,943

Details of the Energy Cost, given above as part of the Operating Cost, are as follows:

Details of the Energy Cost:

The cost estimates are based on the energy consumption described in Table B.3.4 and Fig. B.3.5.

a. Annual natural gas at 100 psi:142,480 MCF x \$4/MCF =\$ 569,920

b. Annual No. 2 fuel oil: 59,910 gal x \$0.57/gallon = 34,150

c. General electric service rate G-1

c-1 Winter energy charge (Nov.-June included)

1,757,566 kWh x 5.30171¢/kWh = 93,181

c-2 Winter demand

Heat pump 6 x 64 hp = 384

DHW pump 1 x 125 hp = 125

Ice maker 7 x 2.5 hp = 52.5

Total Winter Demand = 561.5 x 0.746 = 419 kW

Winter demand charge:

419 kW x \$4.85/kW/month x 8 months = 16,252

c-3 Summer energy charge (July-October):

750,417.2 kW x \$0.0623 = 46,763

c-4 Summer demand charge

Chilled water pumps 7 x 125 = 875 hp

DHW pump 1 x 125 = 125 hp

Total Summer Demand = 1000 hp

Total kW = 1000 hp x 0.746 kW/hp = 746 kW

Demand cost: 746 kW x \$5.84/kW/month x 4 months= 17,427

TOTAL ENERGY COST FOR

CONVENTIONAL SYSTEM =\$ 777,693

4.3 DETAILS OF COST ESTIMATES FOR THE CONVENTIONAL SYSTEM IN THE PARK PLAZA PROJECT

The following are the details of the cost estimates for a conventional system of the same heating and cooling capacity as the HP-ICES system.

a. First Cost for the Conventional System

1.	4 Cleaver Brooks Boilers (55,000 MBH each), including installation: $362,700 for each boiler	=$ 1,450,800
2.	One 5000 ton chiller (Carrier Model 17DA open 837384), including motor riser coupling base control and starter	= 960,000
3.	Two 2000 ton chillers (Carrier Model 193B886XX hermetic), including control and starter	= 840,000
4.	One 1000 ton chiller (Carrier Model 19 EB8983PP hermetic), including control and starter	= 200,000
5.	Five chilled water pumps (Capacity 35000 gpm, 175 hp power)	46,650
6.	Five condenser water pumps (Capacity 6000 gpm, 150 hp power)	= 65,600
7.	Ten cooling towers (Capacity 1000 tons each, 60 hp power)	= 300,000
8.	Four pumps for heating (Capacity 1375 gpm, 60 hp power)	= 20,000
9.	Primary chilled-water distribution network, with equivalent length of 3000 LF and 22 inch diameter pipes, including insulation	= 312,000
10.	Primary hot water distribution network, with equivalent length of 3000 LF and 12 inch diameter pipes, including insulation	= 205,000

11.	Condenser water piping with total equivalent length of 4000 LF and 14 ft diameter	=$ 265,000
12.	Electric installation and connection for 10,000 kW high and low voltage service	= 1,150,000
13.	Sub-Total	= 5,816,050
14.	Contingency 10%	= 581,605
	TOTAL FIRST COST FOR CONVENTIONAL SYSTEM	=$ 6,397,655

b. Operating Cost for Conventional System

1.	Energy Cost	=$ 2,564,707
2.	Maintenance cost for seven laborers $20,000 per person	= 140,000
3.	Station engineer $25,000 per person	= 25,000
4.	Administration 6 people @ $15,000	= 90,000
5.	Maintenance (1% of first cost of equipment)	= 20,000
	TOTAL OPERATING COST FOR CONVENTIONAL SYSTEM	=$ 2,839,707

Details of Energy Cost for the Conventional System:

The cost estimates are based on the energy consumption described in Tables B3.10 and B3.11.

a. Winter Energy Cost		
Natural gas for heating: 226,260 MCF x $4/MCF	=$	1,065,040
b. Natural gas for DHW: 108,550 MCF x $4/MCF	=	434,200
c. Electricity usage (General Electric Service Rate G-1)		
1) Energy charge: 1158114 kWh x $0.0530107/kWh	=	61,392
2) Demand charge 472 kW x $4.85/kW/month x 6 months	=	13,735
Summer Energy Cost for Cooling		
3) Electric energy charge	=	656,908
4) Electric energy demand for cooling	=	333,432
TOTAL ENERGY COST FOR CONVENTIONAL SYSTEM	=$	2,564,707

4.4 LIFE CYCLE COST ANALYSIS OF THE PARK PLAZA PROJECT

This section summarizes the steps and the results of the comparative Life Cycle Cost Analysis for the Park Plaza HP-ICES system and a conventional system with the same capacity.

Table B4.1 gives the details of the life cycle cost analysis of the HP-ICES system and a conventional system with the same capacity.

The following is a summary of the results:

1. The HP-ICES system is more energy efficient than the conventional system. In the example given here, the savings in the annual average energy cost savings achieved by using the HP-ICES system is about $2.4 million.

2. The total annual average life cycle cost is also less for the HP-ICES system than for the conventional system by about $4.2 million.

3. The life cycle capital cost for the HP-ICES system is about 4.5 times as large as that for the conventional system.

   The total operating cost (including energy cost) for the HP-ICES system is less than the tax savings. For the conventional system, the total operating cost, minus the tax savings, is about $36 million.

4. The cost of money (i.e. the mortgage interest rate) and the energy price escalation rate are among the most important factors in determing the relative economics of the two systems.

5. The relative values of the life cycle capital costs, energy costs, etc. are the most useful parameters in comparing the economics of the two types of systems.

6. In the example given here, the Return on Investment for the HP-ICES system, as compared to the conventional system is found to be about 12.7%. The Payback Period is therefore about 8 years.

TABLE B 4.1 ECONOMIC ANALYSIS PARK PLAZA, BOSTON

		HP-ICES SYSTEM	CONVENTIONAL SYSTEM
FC	First Cost	$ 29,011,400	$ 6,397,655
DPF	Down Payment (% of First Cost)	10%	10%
MP	Mortgage Period	20 years	20 years
EP	Period of Economic Analysis	20 years	20 years
MIR	Mortgage Interest Rate (% per year)	15%	15%
DR	Discount Rate	8%	8%
IR	Inflation Rate	6%	6%
EC	Present Energy Costs per year	$ 777,693	$ 2,564,707
ER	Energy Price Escalation Rate	12%	12%
DPER	Depreciation Period	20 years	20 years
SV	Overall Salvage Value (as fraction of First Cost)	30%	20%
IT	Effective Income Tax Rate	46%	46%
C1	Insurance Costs (0.5%)	$ 145,057	$ 31,988
C2	Cost of Maintenance, Labor, Administration, etc.	$ 394,250	$ 275,000
PT	Property Tax Rate	1%	1%

LE B 4.1, continued

		HP-ICES SYSTEM	CONVENTIONAL SYSTEM	LIVE CYCLE SAVINGS BY USING HP-ICES SYSTEM	AVERAGE ANNUAL SAVINGS BY USING HP-ICES SYSTEM
DP	Initial Down Payment	$ 2,901,140	$ 639,766		
MPF	PWF For Mortgage Loan Payment	1.569	1.569		
MP	PW of Mortgage Payment over Life Cycle	40,955,552	9,031,605	-$31,923,947	
	Annual Av. Mortgage Payment	2,047,778	451,580		-$1,596,198
LI= (INT)x (FC-DP)	PW of Loan Interest Paid over Life Cycle	31,811,342	7,015,104		
CC =DP+MP	PW of Life Cycle Capital Costs Before Tax Deduction	43,856,692	9,671,370		
	PWF for Energy Prices	26.74	26.74		
ECL	PW of Life Cycle Energy Cost	20,795,411	68,579,937	$ 47,784,526	
	Annual Av. Energy Cost	1,039,776	3,429,013		2,389,226
IMC	PW of Insur. & Maint. Cost	8,410,892	4,787,706		
PT	PW of Prop. Tax Payments	4,524,543	997,762		
OAC =(EC+ IMC+ PT)	PW of Total Operating and Admin. Costs	33,730,847	74,365,404	40,634,557	
	Annual Av. Op. & Admin. Cost	1,686,542	3,718,270		2,031,728
SSV	PW of System Salvage Value	1,867,303	411,782		
DEP	PW of Depreciation	$ 9,969,337	$ 2,198,459		

TABLE B 4.1, continued

		HP-ICES SYSTEM	CONVENTIONAL SYSTEM	LIFE CYCLE SAVINGS BY USING HP-ICES SYSTEM	AVERAGE ANNUAL SAVINGS BY USING HP-ICES SYSTEM
TLC =(CC+ OAC- SSV)	PW of Total Life Cycle Cost Before Tax Deduction	$75,720,237	$83,624,994		
ITS	PW of Income Tax Savings (inc. Deprec.) = (IT) (DEP + OAC + LI)	34,735,302	38,446,325		
(OAC- ITS)	(Op. & Admin. Cost - Tax Savings)	- 1,004,455	35,919,080	$36,923,535	$1,846,177
NLC	Net Life Cycle Cost After Tax Deduction = TLC - ITS	$40,984,935	$45,178,668	$ 4,193,733	

4.5 COST TO THE USER AND BASIS FOR CHARGING IN THE PARK PLAZA PROJECT

The price to be charged to each user is a very complex question, especially for a new type of system such as the HP-ICES system. According to conventional pricing policies, it would depend on the category of the user and the volume of energy usage. The price would also vary with time.

We shall use a very simplified method, and obtain an average price for energy over the period of economic analysis. The cost to the user will consist of two parts: a first part, the demand charge, that depends on the peak load for the user, which pays for the life cycle average capital cost of the system; and a second part, the energy charge, that pays for the total operating costs. Each of these includes a profit margin.

A price schedule for a user of the HP-ICES system would include different rates for different levels of energy usage. Also, for a commercial user, there would be a penalty for exceeding a predetermined peak load. Here, we do not consider such an elaborate price schedule, but merely give the average price that a user would pay for each MBTU/Hr. of peak load, and for each MBTU of energy used, over a period of 20 years. (Note that MBTU denotes a million BTU.)

We also give the actual price the user pays each year, with an assumed price escalation rate.

The price estimates will be based on the assumption that all the energy produced will be used. It is assumed that the capacity of the system remains the same, i.e. there is no expansion.

The results for the Park Plaza Project are as follows:

(a) <u>Capital Costs</u>

Present Worth of Life Cycle Capital Costs before Tax Deduction

= $43,856,692

Annual Average Life Cycle Capital Cost = $2,192,835

Maximum Peak Load =

Maximum of Peak Heating Load and Peak Cooling Load

= 165×10^6 BTU/Hr = 165 MBTU/Hr.

Average Cost <u>to Owner</u> for each MBTU/Hr Capacity

= $13,290 per year per MBTUH capacity.

In units of kilowatts, this corresponds to $45.36 per year per kW capacity, or $3.78 per month per kW capacity.

This, together with a profit margin, would be recovered from the users over the life cycle of the system, e.g. through a schedule of demand charges.

For a 40% profit margin, the demand charges to the user would be an average of about $1,551 per month per MBTUH capacity, (or about $5.29 per month per kW capacity). This is in terms of the Present Worth,

If the demand payments were constant over the life cycle, this would require a dollar payment of about $1988 per month per MBTUH capacity (or about $6.78 per month per kW capacity). This assumes an inflation rate of 6% per year, and a discount rate of 8%.

For comparison, this is larger by a factor of about 4.5 than the capital costs for the conventional system considered here. Also note that in the Boston Edison Schedule used in determining the cost of the energy used by the HP-ICES system, the electricity demand charges were $5.84 per month per kW between July and October, and $4.85 per month per kW between November and June.

(b) Energy Costs

Average Annual Operating and Administration Costs over the Life Cycle (in terms of present worth) = $1,686,542

Total Energy Usage Paid for by the Users = 383,030 MBTU (This consists of 186,380 MBTU for heating, 120,670 MBTU for cooling, and 75,980 MBTU for hot water.)

Average Energy Cost per MBTU = $4.40 per MBTU

Adding a profit margin of 40% results in the following energy cost to the user:

Average Energy Price Paid by the User over the Life Cycle = $6.16 per MBTU, in terms of the Present Worth.

With the HP-ICES system, the energy price escalation rate would be less than the escalation rate for conventional fuel. Assuming that the conventional fuel price escalates at 12%, and that the other parts of the operating and administration costs increase at the general inflation rate, the above energy price implies a dollar payment of $5.71 per MBTU (equivalent to 1.95¢/kWh) initially, escalating at 9.7% per year.

Note that the electrical energy charge of the Boston Edison Company is 6.3¢/kWh between July and October, and 5.3¢/kWh between November and June. The natural gas price charged by Boston Gas is $4 per thousand cubic feet or about $5.33 per MBTU of heating energy, assuming an efficiency of 75%. For No. 2 Fuel Oil, the price quoted is $0.57/gallon which gives a cost of about $5.70 per MBTU, assuming an efficiency of about 70% and an energy content of about 1.4 therms per gallon of NO. 2 Fuel Oil. These prices are assumed to escalate at 12% per year, on the average.

For comparison, the same procedure applied to the conventional system results in an average energy price paid by the User over the life cycle (in terms of the Present Worth) of about $9.71 per MBTU. The dollar payment would be $7.61 per MBTU (equivalent to 2.60¢/kWh) initially, escalating at 11.5% per year.

The average life cycle energy price paid by the user of the HP-ICES system would be only about 16% of that paid by the user of the conventional system, for a constant volume of energy usage.

B.5 INSTITUTIONAL CONSIDERATIONS PARK PLAZA, BOSTON

A. ENVIRONMENTAL IMPACTS

1) Pollution - Air Pollution, Water Pollution, Thermal Pollution

In contrast to the electricity driven heat pumps in the Market Square project, the heat pumps in the Park Plaza project would be driven by a natural-gas-fired engine. Therefore there would be a question of air pollution generated at the site because of the heat pump operation.

The mitigating factor will be that the HP-ICES system with a natural gas fired engine will use much less natural gas than a conventional system with a natural-gas-fired furnace or boiler. This would decrease the amount of air pollution generated.

With the type of gas fired engine suggested for use in this system, the emissions can be kept below the allowed limits, especially since the air-to-natural-gas ratio is designed to be large (about 40 to 1), which keeps the combustion temperature relatively low.

The cooling system with the ice storage bin will not create any air pollution or thermal pollution, and will avoid the thermal pollution and chemical corrosion problems that cooling towers in conventional systems can give rise to.

Because the system will operate with closed loops, there will be little or no thermal or chemical water pollution.

Since the storage bin contains only water or ice, leakage from it would not cause any pollution problems. Care must be taken to drain the storage bin only into the storm sewer and not into the sanitary sewer, because of the large volume of water involved.

2) <u>Noise</u>

A preliminary study of noise problems in this project shows the following:

(a) The projected increase in traffic volume will not lead to a significantly worse noise environment than exists now.

(b) The air conditioning system could lead to serious noise problems. To avoid these problems, the system and the occupied spaces should be acoustically treated.

(c) Construction noise can be serious in the initial stages of construction. The best way of limiting this would be to hasten the construction process, and try to limit the noise to day time working hours.

(d) Vibration noise arising from the traffic noise can be a problem. Care should be taken to ensure that the building mechanical systems do not make the problem worse.

Overall, the noise related to the traffic should not be perceptibly worse than current noise levels.

Building mechanical equipment would be the other source of noise; this would include intake and exhaust louvers, discharge stacks, cooling towers, etc. in a conventional system. The absence of cooling towers in the HP-ICES system would remove the main source of noise from building mechanical equipment.

3) Land Utilization

Since this project is planned in the middle of a city, land utilization is a matter of concern.

A park of the proposed size may give rise to land problems. The location of the large storage bin below the park would require the permission of the city. Piping would have to cross streets and other public areas, and needs to be carefully planned.

4) Retirement

The retirement of the system can be achieved without causing any major environmental problems, e.g. by converting the large storage bin into a parking lot and making use of the equipment and spaces for other applications.

B. <u>POTENTIAL OWNERSHIP AND FINANCING APPROACHES FOR THE PARK PLAZA HP-ICES PROJECT</u>

The available alternatives here are the following:

A. Ownership and financing by the local municipality, or an agency created by it.

B. Management and financing by a private corporation which may be a utility.

C. A combination of A and B above.

The main factors to be considered are the following:

1) The project requires large financial resources and can involve a large risk.

2) Such a large and complex project will require a high level of technical ability and competent management.

3) Since the project is in the middle of a city, questions relating to land utilization and land ownership become very important. The HP-ICES project will necessarily involve having pipes cross existing roads and may also involve locating the large storage bin underground under publicly owned land.

If approach A is followed, the local municipality or an agency created by it would own the system. The financing would be done either through bonds issued by the municipality, or through federal funding obtained by the municipality, or through the formation of municipal joint action agencies. In this project also, a requirement on the financing mechanism would be that if the project fails financially, it will not create a serious problem for the city's finances.

Financing by a private corporation would have the advantage of greater flexibility and efficiency, and greater ease in obtaining the required technical expertise. However, most private corporations would not have the large financial resources required or be able to handle the financial risk involved in a large, new type of system such as the HP-ICES system.

In this project also, the best approach to financing would probably involve a combination of the two approaches A and B, such as management by a private corporation and funding through bonds issued by the municipality. This should work well with the original conception of the overall project as a privately financed urban renewal project sponsored by the Boston Redevelopment Authority.

C. FIRST AND SECOND LAW ANAYSIS FOR BOTH APPLICATIONS

C-1 INTRODUCTION

This anaysis compares the availability losses and the energy losses for the two HP-ICES concepts and for a comparative conventional system. Main attention was concentrated on the major components that generated the heating and the cooling requirements. The analysis is based on the same output for heating of 1,00,000 Btu/hr and for cooling of 12,000 Btu/hr.

C-2 ANALYSIS

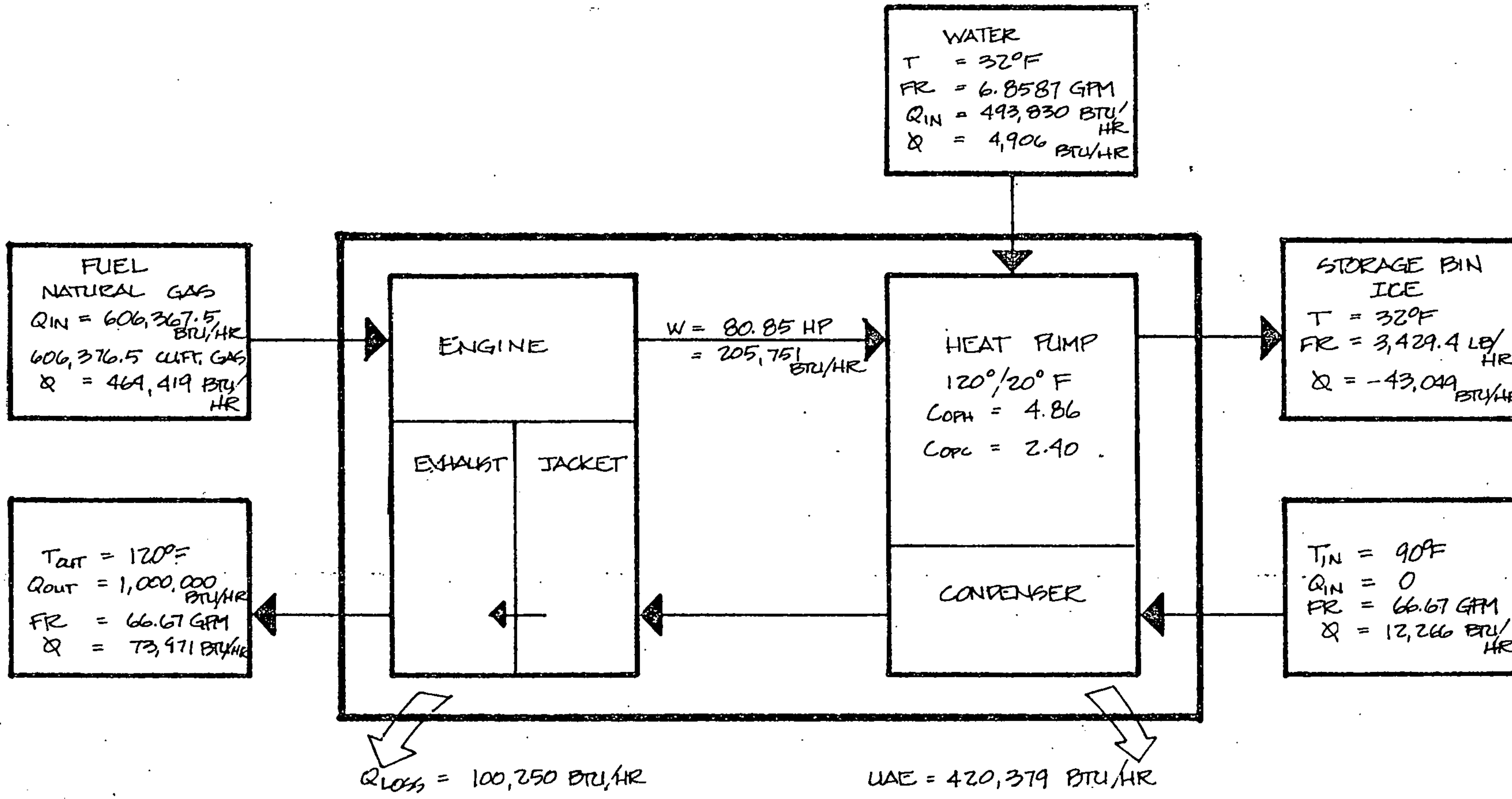
The first system investigated is the HP-ICES driven by a dual fuel engine. The standard conditions were taken as 70°F, 530°R and 14.7 psi. The Lower Heating Value of the gas is 23,600 Btu/lb, with the air weight in the cylinder taken as 30 lb/lb of gas. The exhaust temperature after recovering, is assumed to be 300°F, 750°R, and the compression ratio $V_1/V_2 = 13.562$. The compressor operates with 120°/20°F SDT and SST respectively. Figure C-2.1 shows the flow of energy and availability.

The second system investigated is the electric driven Hp-ICES. This system was considered on a raw source energy base with a power plant heat rate of 11,000 Btu/kW. The heat pump compressor in this case works with 130°/20°F SDT and SST respectively to meet the requirement for the 120°F outlet temperature. This results in a $COP_H = 3.01$ and $COP_C = 2.01$. The flow of energy and availability is shown on Figure No. C-2.2.

The third system investigated is the comparative conventional system using a gas fired boiler with an inlet temperature of 160°F, outlet temperature

FIG. C-2-1

ENERGY AND AVAILABILITY ANALYSIS OF HP-ICES DRIVEN BY DUAL FUEL ENGINE

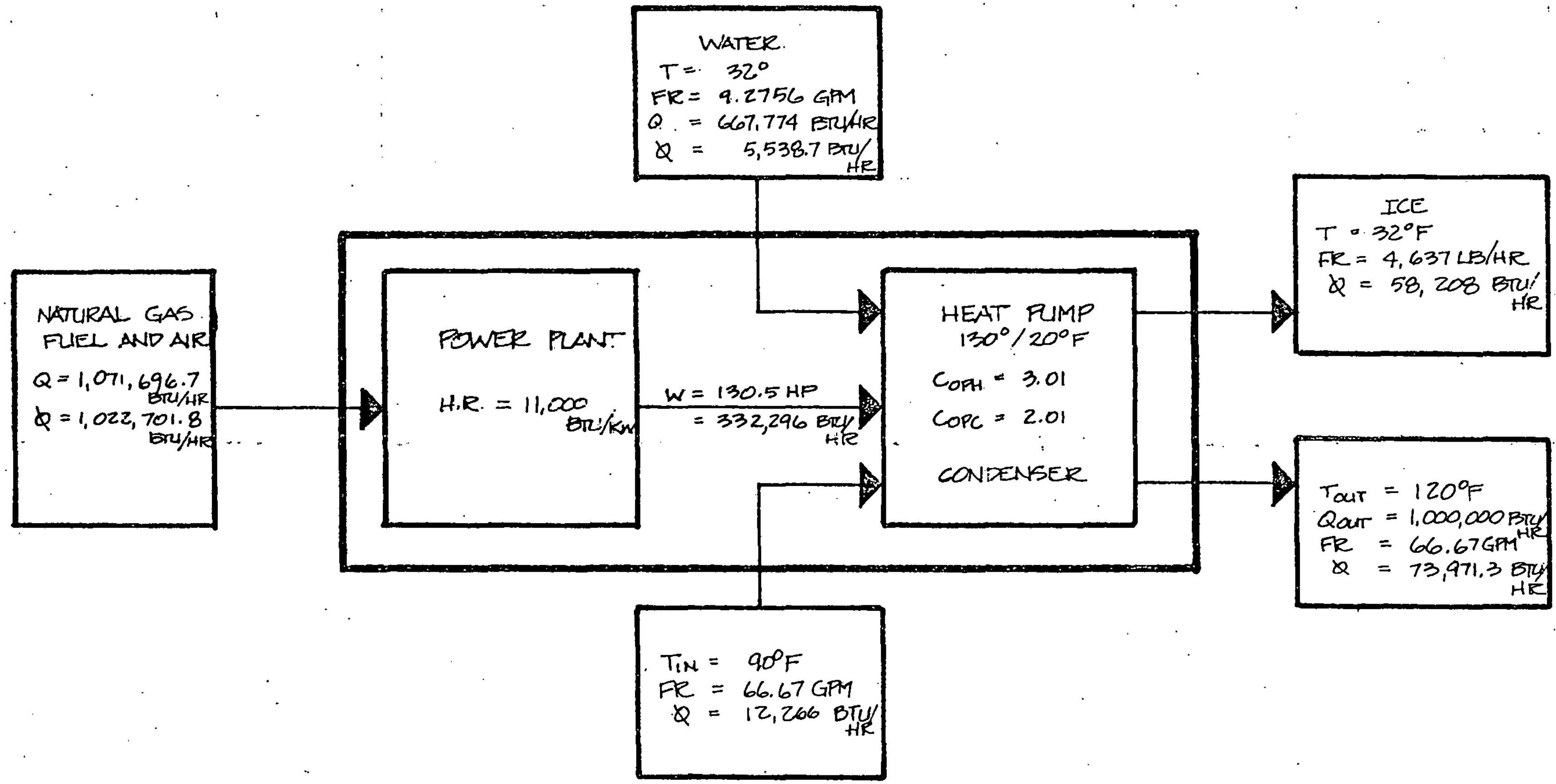


ENERGY LOSSES $= \Sigma Q_{OUT} - \Sigma Q_{IN} = 1{,}000{,}000 - 1{,}100{,}250$
$= 100{,}250$ BTU/HR

AVAILABILITY LOSSE $= \Sigma \dot{X}_{OUT} - \Sigma \dot{X}_{IN} = (73{,}971 - 43{,}049) - (464{,}419 + 4906 + 12{,}266)$
$= -450{,}669$ BTU/HR

FIG. C-2-2

ENERGY AND AVAILABILITY ANALYSIS OF ELECTRICALLY DRIVEN HP-ICES



ENERGY LOSSES $= \Sigma Q_{OUT} - \Sigma Q_{IN} = 1{,}000{,}000 - (1{,}071{,}696.7 + 667{,}774)$

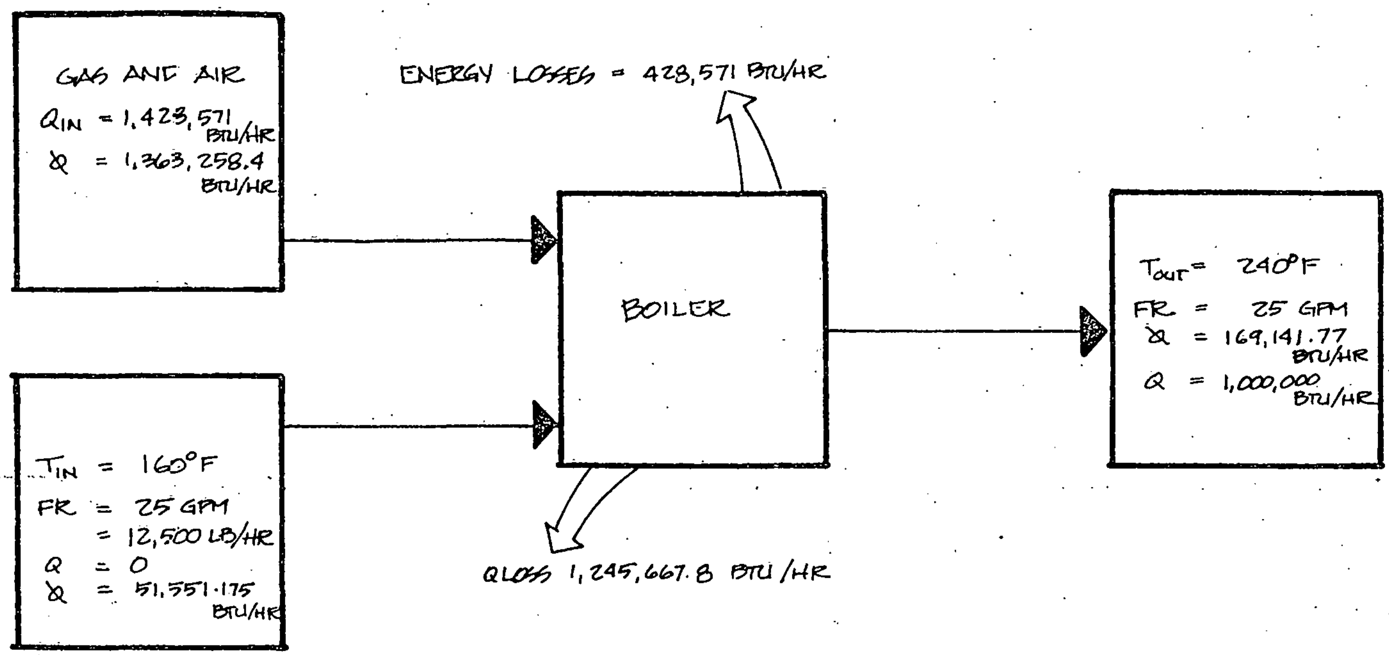
$= -739{,}470.7$ BTU/HR

AVAILABILITY LOSSES $= \Sigma \mathcal{Q}_{OUT} - \Sigma \mathcal{Q}_{IN} = (73{,}971.3 - 58{,}208) - (1{,}022{,}701.8 + 12{,}266 + 5{,}538.7)$

$= -1{,}024{,}743.2$ BTU/HR

FIG. C-2-3

ENERGY AND AVAILABILITY ANALYSIS FOR CONVENTIONAL BOILER



$$\text{ENERGY LOSSES} = \Sigma Q_{OUT} - \Sigma Q_{IN} = 1{,}000{,}000 - 1{,}428{,}571 = -428{,}571 \text{ BTU/HR}$$

$$\text{AVAILABILITY LOSSES} = \Sigma \mathcal{A}_{OUT} - \Sigma \mathcal{A}_{IN} = 169{,}141.77 - 1{,}363{,}258.4 - 51{,}551.175 = -1{,}245{,}667.8 \text{ BTU/HR}$$

of 240°F and boiler efficeincy of 70%. Figure No. C-2.3 shows the energy and availability flow of this system.

C.3 COMPARATIVE ANALYSIS FOR WINTER MODE

TABLE C3.1 COMPARATIVE ANALYSIS FOR WINTER MODE

	Dual Fuel Engine Driven HP-ICES	Electric Driven HP-ICES	Comparative Conventional Boiler
Output Heat Btu/hr	1,000,000 @ 120°F	1,000,000 @ 120°F	1,000,000 @ 240°F
Fuel Energy Input	606,367.5 Btu/hr	1,071,696.7 Btu/hr	1,428,571 Btu/hr
First Law Energy Input	1,100,250 Btu/hr	1,739,470 Btu/hr	1,428,571 Btu/hr
First Law Energy Losses	100,250 Btu/hr	739,470 Btu/hr	428,571 Btu/hr
Availability Losses	450,669 Btu/hr	1,024,743.2 Btu/hr	1,245,667.8 Btu/hr

The table above shows that dual fuel engine driven HP-ICES is the most energy efficient system. This system has minimum fuel energy input and the minimum losses in energy and availability, primarily due to the heat recovery system on the engine. As a result, it is most efficient for the consumer because of the lower fuel requirements and the "free" cooling available in the summer.

The electrically driven HP-ICES was second most efficient system. It appears from the table that while this system has more first law losses than the conventional system, from the viewpoint of the end user, this system is more efficient in fuel input and availability losses.

The last system considered, the conventional boiler, is most inefficient system from an energy and availability point of view. The energy availability of the conventional boiler could be improved a little by working with lower temperatures and a smaller temperature differential. This, however, causes more flow, more pumping horsepower and availability losses. Therefore, the boiler temperature and temperature differential have to be investigated together with the flow rate and the pumping power input to find the optimal condition with the minimum of availability losses.

It seems that the heat pump performance can be improved by keeping the saturated discharge temperature (SDT) as low as possible. This reduces the power input to the compressor and by that, the availability losses. All of this can be done up to the point where the temperature in the condenser will not satisfy the heating requirement. 110°F water temperature was found to be sufficient for the requirement load. It seemed that the SDT can be reduced as far as 110°F but this will require an infinite heat exchanger that is only academic and not economic. As a result, the temperature is set to be minimum point that will be sufficient for the load and for heating exchanger performance.

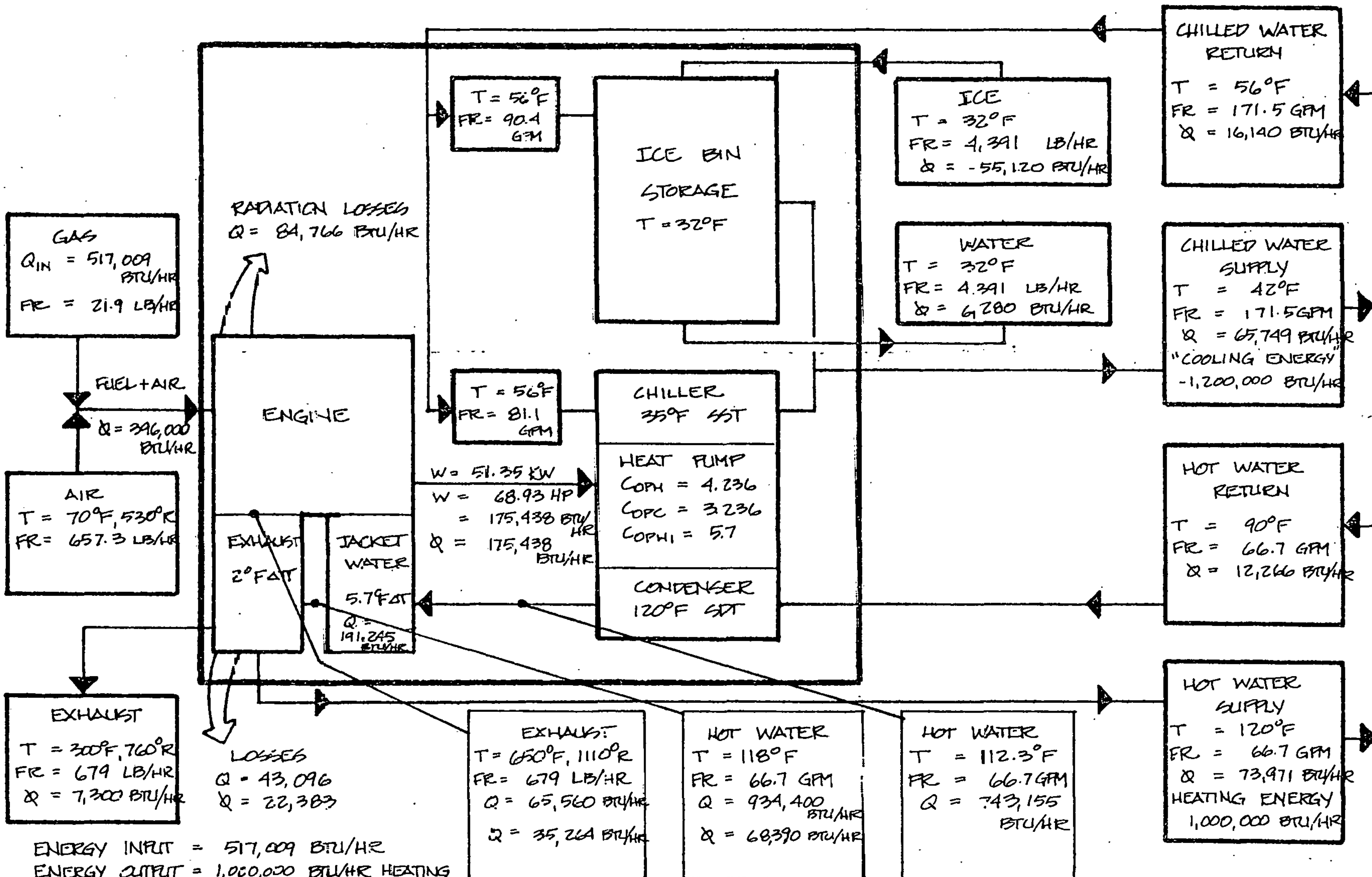
The dual fuel engine driven HP-ICES with its heat recovering system allows the reduction of 10°F in the SDT of the compressor and by that, it improves the overall performance of the system.

C-4 AVAILABILITY ANALYSIS OF SIMULTANEOUS HEATING AND COOLING MODE

In investigating the three system in the simultaneous heating and cooling mode the output is taken to be 1,000,000 Btu/hr for heating and 1,200,000 Btuh, for cooling with the standard condition of 70°F, 530°R, and 14.7 psi.

FIG. C-4-1

SIMULTANEOUS HEATING AND COOLING MODE FOR GAS ENGINE DRIVEN HP-ICES



ENERGY INPUT = 517,009 BTU/HR
ENERGY OUTPUT = 1,000,000 BTU/HR HEATING
1,200,000 BTU/HR COOLING

AVAILABILITY LOSS = $\sum \mathcal{Q}_{OUT} - \sum \mathcal{Q}_{IN}$ = (73,971 + 65,749 + 6,280 + 7,300) − (−55,120 + 16,140 + 12,266 + 396,000)
= − 215,986 BTU/HR

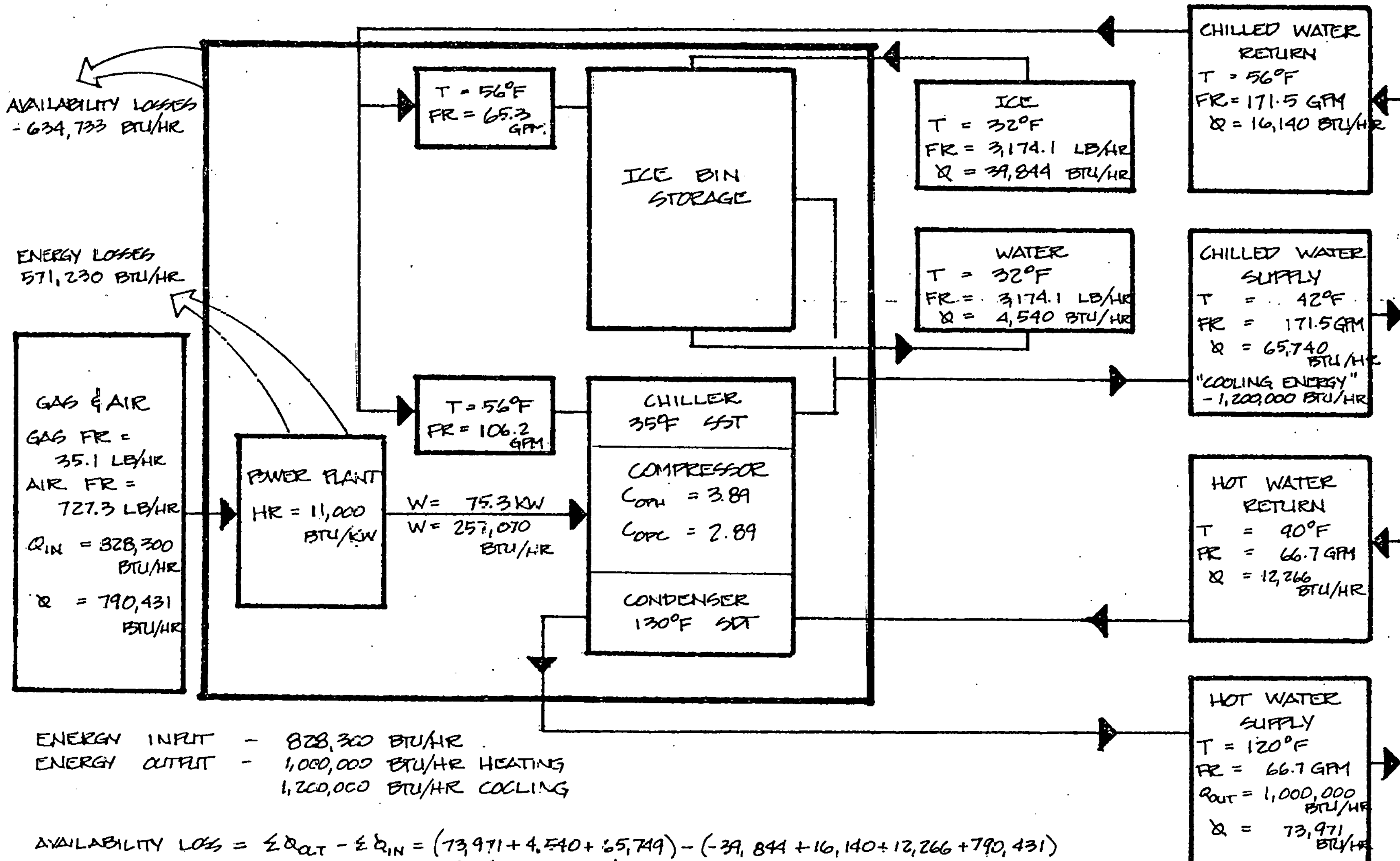
The dual fuel engine driven HP-ICES in this case generates 68.93 hp while the compressor condition is 35°/120°F SST and SDT respectively. The hot water passes through the condenser, picks up about 75% of the heat and leaves the condenser at 112.3°F. The heat recovery system of the engine will increase the temperature of the water to its utilization temperature of 120°F. The system utilizes the ice bin to satisfy the 52% deficiency in "cooling energy' requirement that is not produced by the heat pump while generating the heating requirement. As a result, the energy consumption and the availability losses of the system are reduced. (Figure C-4.1 shows the flow of energy and availability in this system.)

The large decrease in the available energy, about 26%, occurs by virtue of the irreversible flow of heat to the "heating water". Another 20% of the loss of available energy in the engine is due to the incomplete combustion, dissociation, radiation, friction and passage of heat to the lubricating oil. Only 44% of the available energy is converted to brake work. This work also converts to heat in the compressor and thereby add to the available energy losses of the system.

The electric driven HP-ICES is the second system considered. The system was evaluated with respect to raw source energy based on a power plant with a heat rate of 11,000 Btu/kW. The heat pump compressor in this case works on 130°/35°F SDT and SST respectively, to meet the requirement of 120°F outlet temperature for cooling. This condition will result in a $COP_H = 3.89$ and $COP_C = 2.89$. The flow of energy and availability are shown on Figure No. C-4.2. This system also utilizes the ice bin to convert the 38% deficiency in "cooling energy" that was not generated by the heat pump while producing the heating required. The large decrease of the available energy occurs in the power plant due to irreversible flow of heat to the cooling

FIG. C-4-2

SIMULTANEOUS HEATING AND COOLING MODE FOR ELECTRICALLY DRIVEN HP-ICES

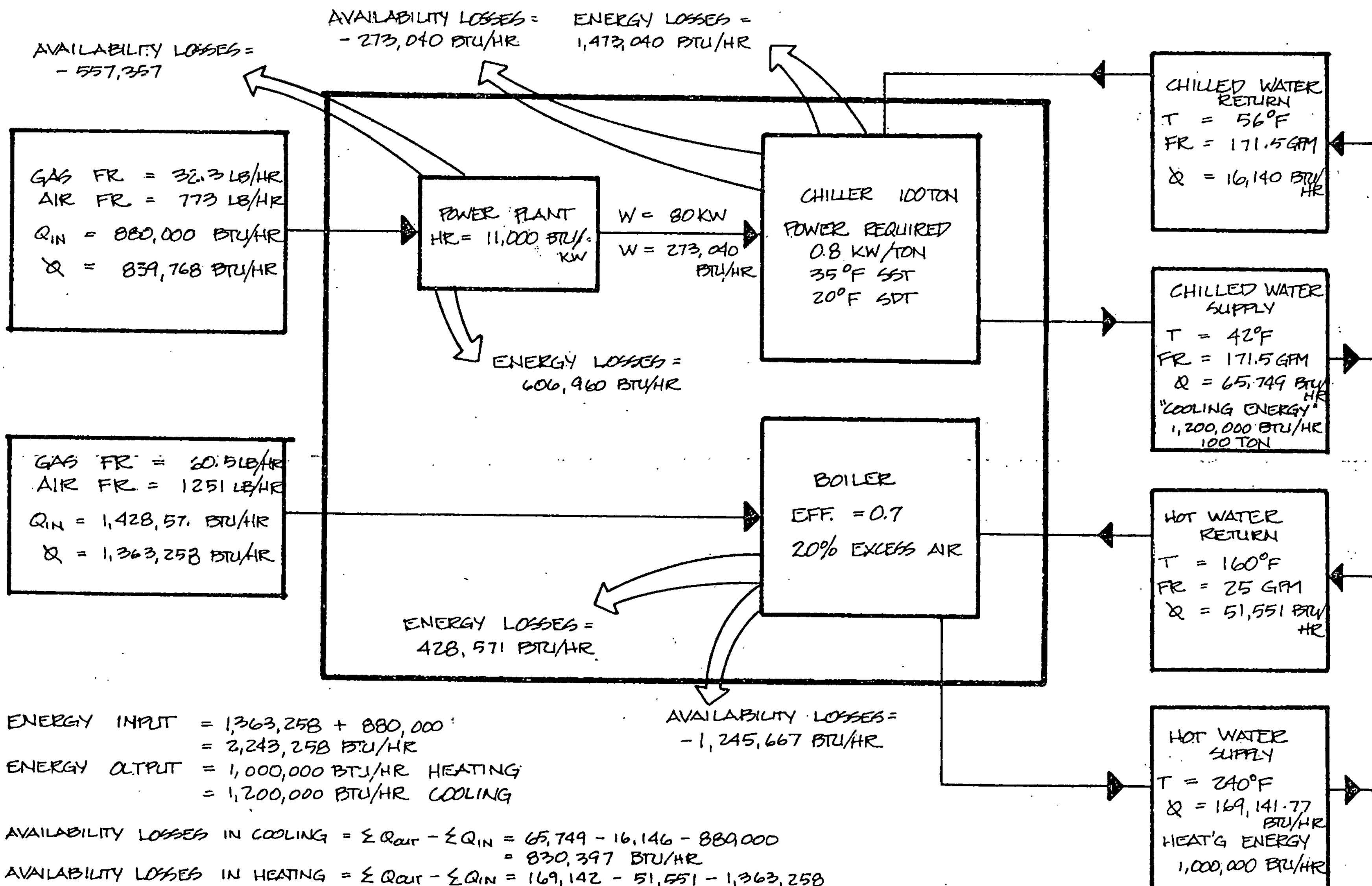


ENERGY INPUT - 828,300 BTU/HR
ENERGY OUTPUT - 1,000,000 BTU/HR HEATING
1,200,000 BTU/HR COOLING

$$\text{AVAILABILITY LOSS} = \Sigma Q_{OUT} - \Sigma Q_{IN} = (73,971 + 4,540 + 65,749) - (-39,844 + 16,140 + 12,266 + 790,431)$$
$$= -634,733 \text{ BTU/HR}$$

FIG. C-4-3

SIMULTANEOUS HEATING AND COOLING MODE FOR CONVENTIONAL SYSTEM



ENERGY INPUT = 1,363,258 + 880,000
= 2,243,258 BTU/HR

ENERGY OUTPUT = 1,000,000 BTU/HR HEATING
= 1,200,000 BTU/HR COOLING

AVAILABILITY LOSSES IN COOLING = $\Sigma Q_{OUT} - \Sigma Q_{IN}$ = 65,749 - 16,146 - 880,000
= 830,397 BTU/HR

AVAILABILITY LOSSES IN HEATING = $\Sigma Q_{OUT} - \Sigma Q_{IN}$ = 169,142 - 51,551 - 1,363,258
= 1,245,667 BTU/HR

UAE = 2,076,064 BTU/HR

water via the exhaust.

The third system investigated is a comparative conventional system using for heating a gas fired boiler with 70% efficiency and for cooling a centrifugal chiller with a 4.4 COP_C and working condition of $105^{\circ}/35^{\circ}F$ or SDT and SST respectively. The heating utilization temperature was increased to $240^{\circ}F$ while the return is $160^{\circ}F$. The chilled water temperature stays the same as with the other systems, $42^{\circ}F$.

This system was also considered on raw source energy base with power plant heat rate of 11,000 Btu/kW. The heating and cooling system work separately thus the energy rejected by the cooling cycle was not utilized by the heating system. Figure No. C-4.3 describes the availability and the energy flow and the major energy and availability losses. In the boiler the losses of available energy occurs due to the irreversible flow of heat to the "heating water" and energy losses via the chimney. In the cooling system the energy losses are via the cooling tower and in the power plant via the cooling water and the exhaust. The entire power generated by the power plant is converted to heat in the compressor and dumped into the cooling tower.

TABLE C-4.1
COMPARATIVE ANAYSIS FOR SIMULATNEOUS HEATING AND COOLING MODE

	Dual Fuel Engine Driven HP-ICES	Electric Driven HP-ICES	Comparative Conventional System	Percent HP-ICES Compared to Conventional	
				Dual Fuel Engine Driven HP-ICES	Electric Driven HP-ICES
Output Heat Btu/hr	1,000,000 @ 120°F	1,000,000 @ 120°G	1,000,000 @ 240°F	100%	100%
Output Cooling Ton	1,200,000 100 Ton	1,200,000 100 Ton	1,200,000 100 Ton	100%	100%
Fossil Fuel Energy Input Btu/hr	517,009	828,300	2,308,571	22.4%	36%
First Law Energy Input Btu/hr	1,717,009	2,028,300	3,508,571	49%	58%
Energy Stored	632,800	457,070	-	-	-
First Law Energy Losses	127,862	571,230	2,508,571	5%	23%
Availability Losses	-215,986	-634,733	-2,076,064	10%	30%

Table C-4.1 shows that in the simultaneous heating and cooling mode the HP-ICES and specifically the dual fuel engine driven HP-ICES are more efficient than the conventional system. The dual fuel engine driven HP-ICES required only 22.4% the energy consumption while losing only 5% energy and 10% availability compared to the conventional system. This is due to the transference of energy from the cooling zone to the heating zone and to the ice bin to be stored for winter use.

The electric driven HP-ICES has more losses in energy and availability due to the fact that there is no opportunity for heat recovery in the power plant as the heat rejected is wasted through a cooling tower.

The conventional system is inefficient from an energy and availability point of view, due to the fact that all heat rejection from the compressor is dumped in the cooling tower and there is no interaction between cooling and heating systems. It is the combination of all of these factors that result in the dual fuel engine driven HP-ICES having the lowest energy consumption, as well as the lowest losses in availability and energy of all the systems compared.

D. ANALYSES DESCRIPTION

Following is the description of the analyses for the ice generating HP-ICES which began with the evaluation of the building load profile.

The cooling, heating and domestic hot water loads were calculated by hand using the degree-days method for heating and the equivalent full load hours method for cooling.

Actually, a computer community energy analysis should be developed to enable to consider the load diversity between the different sectors and buildings in the community.

Next, the working temperatures of the heat pump were determined. The evaporator temperature was fixed by the ice making requirements while the condensing temperature was determined as the optimum temperature that will satisfy the heating load and at the same time require a minimum horse power input. The temperatures and the peak heating load were the deciding factor in the sizing of the heat pump.

Following the selection of the equipment, the COP for heating, cooling and the annual overall COP were determined. Consequently, the performance of the heat pump was evaluated at different evaoporator and condenser temperatures to suit the different conditions of heat pump operation throughout the year with the main concern for energy conservation and a minimum power input.

The focal point in the determination of the applicablility of the HP-ICES is the annual energy balance between the heating and cooling requirements and the heating and cooling generated by the heat pump. Any discrepancy between the cooling required and cooling generated by the heat pump was balanced out by the annual storage system.

The maximum accumulation of ice in the storage as obtained in the annual energy balance determined the ice storage bin size.

If the "cooling" energy generated by the heat pump exceeded the cooling requirements of the community, a system modification was made with an additional heat source for the system, as happened in the case of Park Plaza Boston. In this case a dual fuel engine driven heat pump with a heat recovery system was selected and an evaluation of the engine and heat pump performance was made to determine the new improved COP_H of the engine-compressor package as well as the cooling and the annual average overall COP's.

The heat recovered from the engine jacket water and exhaust gases reduced the amount of heat that the heat pump had to generate and consequently reduced also the amount of ice generated and the size of the ice bin. As the next step, the annual energy consumption was evaluated including the energy required for the primary auxiliaries.

A conventional comparative system was selected which would be able to satisfy the same heating and cooling requirements as for the HP-ICES system and the annual energy consumption for such a system was calculated.

To stress the outstanding benefits of the HP-ICES a comparative raw source energy consumption analysis was made in accordance with the ASHRAE Standard 90-75 section 12.

It would be advisable if a computer program were developed with a capability to determine the performance of the ice generating heat pump on a community scale using an annual storage system.

Feasibility of the HP-ICES from the energy point of view alone is, however, not sufficient for its wide acceptance and an economic feasibility study was performed to prove its overall desirability.

The method used for evaluating and comparing the economics of the HP-ICES system and a conventional system of the same heating and cooling capacity uses a life cycle cost analysis.

The first cost of each system includes the generation equipment and distribution system, but does not include the end-users equipment (air handling units) or the cost of land.

The HP-ICES system requires a large amount of space for the storage. In the analysis, the cost of excavating the land, constructing the ice bin, etc.,was taken into account. It was assumed that the area of the land used will be about the same for the two systems, and that the large storage will be under the buildings and will not take up additional land area.

Equipment prices and costs were obtained from the manufacturers and contractors.

Energy prices were calculated according to public utility services.

The life of the equipment was assumed to be 20 years and the life of the distribution system and ice storage bin 40 years. The mortgage period and the period of the economic analyses were both assumed to be 20 years. The depreciation life of the overall system (for the purpose of tax deductions) was assumed to be 20 years.

The salvage value of the equipment after 20 years was assumed to be 20% of the first cost. For the ice bin in the HP-ICES system, the salvage value was assumed to be 40% of the initial cost of the ice bin (i.e. the cost of excavation and ice bin construction).

Insurance costs were assumed to be about 0.5% of the first cost. The costs of maintenance, labor, administration, etc. were estimated for each project separately.

The general inflation rate was aseumed to be an average of 6% per year. The discount rate used in evaluating the present worth of money was assumed to be 2% more than the inflation rate, i.e., 8% per year while the fuel escalation rate was assumed to be 12%.

The effective income tax bracket for the commercial system, assuming ownership by a private corporation was assumed to be 46%. The tax deductions that a business can make were allowed for.

The property tax was assumed to be 1% per year.

An average price that the user would pay for each MBTU/HR (million Btu per hour) of peak load, and for each MBTU of energy consumption over a period of 20 years was developed. The price considers profits and energy escalation rates.

The price estimates were based on the assumption that all the energy produced will be used. It was assumed that the capacity of the system remains the same, i.e. there is no expansion.

For basic assumptions see the corresponding sections, for example, assumption on load were made before the load calculations, the assumption on economics were made before the economic section, etc.

APPENDIX 1

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APPENDIX 1. MARKET SQUARE

1. Pump Selection

a. Heating Pump

Peak heating load	=	13,064,000 Btu/hr
Domestic hot water	=	3,086,000 Btu/hr
Total		16,150,000 Btu/hr

For primary hot water loop select three pumps, one for each heat pump, $\Delta T = 30^{o}F$.

$$\frac{5{,}500{,}000 \text{ Btu/hr}}{500 \times 30} = 367 \text{ gpm}$$

Condenser pressure drop = 25 ft.

Assume 2500 ft. equivalent pipe length (including fittings) 5 ft per 100 ft.

Pressure drip of 2500 $= \frac{2500 \times 5}{100} = 125$ ft.

Total Pump head = 150 ft.

Select Aurora three pumps 3 x 4 x 14, Series 410, double suction, 22 hp each

68% efficiency, 1750 rpm, NPSH 8 ft.

b. Bin Chilled Water Pumps (pumping from bin to primary loop)

3000 ton x 1.5 gpm/ton ($\Delta T = 16^{o}F$) = 4500 gpm

Select two pumps to operate in parallel 2250 gpm each and one stand-by.

Assume 2000 equivalent linear feet of

Piping: $\frac{5 \times 2000}{100} = 100$ ft.

Static head: = 10 ft.

Total = 110 ft.

Select Aurora three pump 8 x 10 x 17B , Series 410 double suction 70 hp each.

89% efficiency, 1150 rpm, NPSH = 11 ft.

c. <u>Chilled Water Pumps (to operate in summer time between chiller and bin)</u>

Three pumps $\dfrac{555 \text{ ton} \times 12000 \text{ Btu/hr}}{16^{o}F(\Delta T) \times 500} = 830 \text{ gpm}$

Chiller pressure drop = 25 ft.

Assume 700 ft. of equivalent pipe length

$$\frac{700 \times 5}{100} = 35 \text{ ft.}$$

Static head = 10 ft.

Total head 70 ft.

Select Aurora three pumps 15 x 6 x 15, Series 410, double suction, 18 hp each, 80% efficiency, 1150 rpm, NPSH = 9 ft.

d. <u>Condenser Water Pump (for summer operation with cooling towers)</u>

Heat rejection of each 555 ton chiller 8,131,000 Btu/hr

$$\frac{8,131,000}{10^{o} \times 500} = 1626 \text{ gpm}$$

$\Delta T = 10^{o}F$

Condenser pressure drop = 25 ft.

Static head = 10 ft.

For a height of 250 ft. pressure drop = $\dfrac{500 \times 5}{100}$ = 25 ft.

Fittings = 20 ft.

Total pump head = 80 ft.

Select Aurora three pumps, 8 x 8 x 11B, Series 410, double suction, 38 hp, 88% efficiency, 1750 rpm, NPSH = 15 ft.

2. Pumping Energy Consumption

TABLE APP-1.1 HEATING PUMP PUMPING ENERGY

			CONSUMPTION		DEMAND		
Month	DD	Hours of Operation	KWH	Cost	Peak Demand	Cost	Total Cost
January	871	727	29286	288	40	161	449
February	762	637	25660	253	40	161	414
March	626	523	21068	208	40	161	369
April	288	241	9708	96	40	161	257
May	74	62	2498	25	20	81	106
June	0	0	0	0	0	0	0
July	0	0	0	0	0	0	0
August	0	0	0	0	0	0	0
September	33	28	1128	16	20	104	120
October	217	181	7291	103	40	207	310
November	519	434	17483	172	40	161	333
December	834	697	28078	277	40	161	438
Total	4224	3530	142200	$1438		$1358	$2,796

$$\text{Heating FL Hours} = \frac{\text{Heating Requirement}}{\text{Heating Load}} = \frac{27000 \times 10^6}{13 \times 10^6} = 2077 \text{ FLH}$$

Assume pumps operate during 2077 x 1.7 = 3530 hours.

NOTE: The pumps were sized to accommodate the heating and DHW Load i.e. for 22 hp each. Since, however, DHW will be calculated separately, reduce the pumping power in ratio:

$$\frac{13000000 \times 22 \times 3}{16{,}000{,}000} = 54 \text{ hp}$$

Thus, the pumping power consumption

= 3530 hours x 54 x 0.746 = 142,200 KWH

The kWh (or the operating hours) were distributed by month in the same ratios as the degree days.

TABLE APP. 1.2 - BIN TO PRIMARY CIRCUIT PUMP ENERGY CONSUMPTION

	CONSUMPTION			DEMAND			
Month	% KWH	Actual KWH	Cost $	%KW	Peak KW	Cost $	Total Cost $
January	0	0		0	0	0	0
February	0	0		0	0	0	0
March	0	0		0	0	0	0
April	8	14080	139	50	55	221	360
May	12	21120	208	100	110	442	650
June	18	31680	449	100	110	570	1,019
July	18	31680	449	100	110	570	1,019
August	18	31680	449	100	110	570	1,019
September	18	31680	449	100	110	570	1,019
October	8	14080	199	50	55	285	484
November	0	0		0	0	0	0
December	0	0		0	0	0	0
Annual		176000	$2342			$3228	$5,570

Equivalent Full Load Hours Cooling = 939

Assume Pumps operate 939 x 1.7 = 1596 (say 1600 hours)

Two pumps 70 hp each = 52.2 (say 55 KW)

Annual KWH = 110 x 1600 = 176000 KWH

TABLE APP. 1.3 CHILLER TO BIN PUMPING POWER
(With DHW Generation and Heat
Rejection to Cooling Tower)

	Domestic Hot Water Generation		Cooling Tower Operation			
Month	Hours of* Operation	KWH	Hours**	KWH	Total KWH	Cost $
January						
February						
March						
April						
May	158	2212			2212	22
June	87	1218			1218	17
July	89	1246			1246	18
August	71	994	240	10,080	11074	157
September	95	1330	316	13272	14602	207
October	268	3752	86	3612	7364	104
November						
December						
Total Annual		10752		26964	37,716	$525

*With one pump operating

$$\text{Hours} = \frac{\text{Cooling Requirement}}{\text{Cooling Capacity}} = \text{For May } \frac{901 \times 10^6 \text{ Btu}}{475 \text{ ton} \times 12000 \text{ Btu/hr/ton}}$$

$$= 158 \text{ hours; } 18 \text{ hp} = 14 \text{ KW}$$

**With three pumps operating

$$\text{Hours} = \frac{\text{Cooling Requirement}}{\text{Cooling Capacity}} = \text{For August } \frac{4810 \times 10^6 \text{ Btu}}{3 \times 555 \text{ ton} \times 12000 \text{ Btu/hr/ton}}$$

$$= 240 \text{ hours ; } 3 \times 14 = 42 \text{ KW}$$

TABLE APP 1.4 DOMESTIC HOT WATER PUMPING ENERGY

Month	Energy Requirement Btu x 10^6	Compressor Heat Rejection Btu/hr	Hours* of Pump Operation	Pumping** Demand KW	Energy Consumption KWH	Cost $	Total Cost
January	1010	6,069,000	166	17	2822	28	96
February	919	6,069,000	151	17	2567	25	93
March	881	6,069,000	145	17	2465	24	92
April	804	6,069,000	132	17	2244	22	90
May	738	7,672,000	96	17	1632	16	84
June	666	7,672,000	76	17	1292	18	106
July	686	7,672,000	79	17	1343	19	107
August	544	7,672,000	62	17	1054	15	103
September	515	7,672,000	59	17	1003	14	102
October	673	7,672,000	77	17	1309	19	107
November	817	6,069,000	134	17	2278	22	90
December	958	6,069,000	157	17	2669	26	94
Annual	9211				22678	$248	$1,164

$$* \quad \frac{\text{Energy Requirement}}{\text{Heat Rejection of one Heat Pump}}$$

For January $\dfrac{1010 \times 10^6 \text{ Btu}}{6{,}069{,}000 \text{ Btu/hr}} = 166$ hours

** Assume one heating pump 22 hp = 0.746 x 22 = 17 kW

TABLE APP 1.5 PUMPING POWER FOR CONDENSER WATER PUMPS AND COOLING TOWER FANS (KWH)

Month	Hours of Operation*	Pumps (85KW)	Tower (56KW)	Total	Cost$
August	240	20,400	13,440	33,840	480
September	316	26,860	17,696	44,556	631
October	86	7,310	4,816	12,126	172
Total		54,570	35,952	90,522	$1,283

*Same as for chiller to bin pumps.
Pumps 3 x 38 = 114 hp = 85 KW
Fans 3 cells at 555 tons, 25 hp each = 75 hp = 56 KW

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APPENDIX - 2

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Appendix 2

1. Pump Selections

a. Heating Pumps

Heating delivered = 27.26×10^6 Btu/HR

Hot water gpm = $\frac{27.26 \times 10^6 \text{ Btu/H8}}{500 \times 30}$

= 1817 gpm

Head from schematic 150 ft.

Hours of operation, (see table below)

TABLE APP 2.1
Number of Hours of Temperatures Below 57°F

Month	N	D	T	F	M	A	M	T	T	A	S	O	Total
Number of Hours	691	742	744	672	735	217	147	57	61	11	115	209	4401

We deduct the hours in July and August and come up with 4329 hours.

Ref. Engineering Weather Data.
AFM - 88-8 Chapter 6 June, 1967.

b. Domestic Hot Water

GPM = $\frac{35,800,000}{30 \times 500}$

= 2387

25 ft. pressure drop in condenser

25 ft. pressure drop in heat exchanger

100 ft. piping and fitting

Total = 150 ft.

No. of hours of operation

$\frac{\text{monthly load}}{\text{peak load}} = \frac{6245 \times 10^6}{35.8 \times 10^6}$

= 174.8 hours per month

c. Chilled Water Pumps

$$GPM = \frac{1755 \times 12{,}000}{500x(56-40)} = 2632$$

Head = 150 ft.

Hours distribution of percentage

d. Condenser Water Pumps

$$CPM = \frac{26.0x10^6}{10x500} = 5200$$

Head = 80 ft.

e. Cooling Tower Fans

1755 Ton - 110 HP

HEAT BALANCE—38DD8⅛ DUAL FUEL

Cylinders	rpm	Bhp/Cyl.	Temp. °F.		Per Cent Load Rated Bhp	Heat in BTU/Bhp/Hr.						
			Cooling Water	Lube Oil		Useful Work	Cooling Water	Lube Oil	Exhaust Sensible	Exhaust Latent	Rad. & Unacct.	Total Input
4 & 5	720	160	165	170	25	2546	2800	1930	4315	100	219	12000
					50	2546	1710	1140	2555	50	99	8100
					75	2546	1440	960	2155	33	66	7200
					100	2546	1350	900	2020	25	59	6900
					110	2546	1350	900	2020	23	61	6900
6 - 12	720	160	165	170	25	2546	2770	1850	3990	100	164	11400
					50	2546	1710	1140	2460	50	94	8000
					75	2546	1440	960	2070	33	51	7100
					100	2546	1350	900	1935	25	44	6800
					110	2546	1350	900	1935	23	44	6800
4 & 5	900	200	165	170	25	2546	3480	2340	5110	108	116	13700
					50	2546	2130	1440	3100	54	130	9400
					75	2546	1715	1155	2520	36	128	8100
					100	2546	1560	1050	2295	27	122	7600
					110	2546	1560	1050	2295	23	126	7600
6 - 12	900	200	165	170	25	2546	3290	2200	4700	108	56	12900
					50	2546	2130	1440	3080	54	50	9300
					75	2546	1715	1155	2490	36	58	8000
					100	2546	1560	1050	2250	27	67	7500
					110	2545	1560	1050	2250	23	71	7500
4 & 5	720	160	250	185	25	2546	1755	2630	4580	62	427	12000
					50	2546	1040	1560	2710	50	194	8100
					75	2546	875	1315	2280	33	151	7200
					100	2546	820	1230	2140	25	139	6900
					110	2546	820	1230	2140	23	141	6900
6 - 12	720	160	250	185	25	2546	1680	2520	4300	62	292	11400
					50	2546	1040	1560	2680	50	124	8000
					75	2546	875	1315	2240	33	91	7100
					100	2546	820	1230	2100	25	79	6800
					110	2546	820	1230	2100	23	81	6800
4 & 5	900	200	250	185	25	2546	2300	2970	5410	108	366	13700
					50	2546	1410	1820	3310	54	260	9400
					75	2546	1130	1460	2660	36	263	8100
					100	2546	1030	1330	2420	27	247	7600
					110	2546	1030	1330	2420	23	251	7600
6 - 12	900	200	250	185	25	2546	2180	2810	5050	108	206	12900
					50	2546	1410	1820	3270	54	200	9300
					75	2546	1130	1460	2640	36	188	8000
					100	2546	1030	1330	2390	27	177	7500
					110	2546	1030	1330	2390	23	181	7500

EXHAUST GAS

TABLE 7

ENGINE			TEMP. °F.		Blower	Exhaust	EXHAUST TEMP. °F. ENGINE LOAD			
Model	Cyl.	RPM	Oil	Water	CFM	Lb./Hr.	1.0	0.75	0.50	0.25
38D8⅛	4	720	170–185	165	2,450	11,250				
38DD8⅛	5				3,100	14,230		Model 38D8⅛		
	6				3,500	16,060	620	520	420	330
	8				4,550	20,880				
	9				5,130	23,550		Model 38DD8⅛		
	10				5,700	26,160	620	610	550	450
	12				7,500	34,420				
38D8⅛	4	900	170–185	165	3,060	14,050	650	540	440	340
38DD8⅛	5				3,875	17,790				
	6				4,800	22,030				
	8				6,160	28,270				
	9				7,000	32,130		Model 38DD8⅛		
	10				7,675	35,230				
	12				9,375	43,030	650	630	570	450
	4	720	185	250	2,450	11,250				
	5				3,100	14,230	640	540	440	350
	6				3,500	16,060				
	8				4,550	20,880		Model 38DD8⅛		
	9				5,130	23,550				
	10				5,700	26,160	640	630	570	470
	12				7,500	34,420				
	4	900	185	250	3,060	14,050				
	5				3,875	17,790	670	560	460	360
	6				4,800	22,030				
	3				6,160	28,270		Model 38DD8⅛		
	9				7,000	32,130				
	10				7,675	35,230	670	650	590	480
	12				9,375	43,030				
38DS8⅛	4	720	170–185	165	2,050	9,410	600	590	570	470
	5				2,560	11,750				
	6				3,070	14,100				
	8				4,100	18,820				
	9				4,610	21,160				
	10				5,120	23,500				
	12				6,140	28,180				
38DS8⅛	4	900	170–185	165	2,560	11,750	630	610	590	480
	5				3,200	14,690				
	6				3,840	17,630				
	8				5,130	23,550				
	9				5,760	26,440				
	10				6,400	29,380				
	12				7,680	35,250				
33TD8⅛—Int. Blower	6	720	170–185	165	4,400	20,200	680	680	680	620
4/4 Blower	9				6,600	30,300	700	670	550	400
Int. Blower	12				8,800	40,400	680	680	680	620
38TD8⅛—Int. Blower	6	900	170–185	165	5,500	25,250	700	700	700	640
4/4 Blower	9				8,250	37,870	730	730	600	430
Int. Blower	12				11,000	50,500	700	700	700	640
38TD8⅛—Int. Blower	6	720	185	250	4,400	20,200	700	700	700	640
4/4 Blower	9				6,600	30,300	720	690	570	420
Int. Blower	12				8,800	40,400	700	700	700	640
38TD8⅛—Int. Blower	6	900	185	250	5,500	25,250	720	720	720	660
4/4 Blower	9				8,250	37,870	750	750	620	450
Int. Blower	12				11,000	50,500	720	720	720	650
38TDD8⅛—2/3 Blow.	6	720	170–185	165	4,400	20,200	750	770	770	700
4/4 Blow.	9				6,600	30,300	700	690	650	600
2/3 Blow.	12				8,800	40,400	750	770	770	700
38TDD8⅛—2/3 Blow.	6	900	170–185	165	5,500	25,250	760	780	780	720
4/4 Blow.	9				8,250	37,870	730	720	690	630
2/3 Blow.	12				11,000	50,500	760	780	780	720
38TDD8⅛—2/3 Blow.	6	720	185	250	4,400	20,200	770	790	790	720
4/4 Blow.	9				6,600	30,300	720	710	680	620
2/3 Blow.	12				8,800	40,400	770	790	790	720
38TDD8⅛—2/3 Blow.	6	900	185	250	5,500	25,250	780	800	800	740
4/4 Blow.	9				8,250	37,870	750	740	710	660
2/3 Blow.	12				11,000	50,500	780	800	800	740

ENGINE PUMP DELIVERY TABLE 8

Engine			GPM Rated Pump Delivery	
Model	**Cyl.**	**RPM**	**Soft Water**	**Lube Oil**
38D8⅛	4	720	215	200
38DD8⅛	5	720	265	200
38DS8⅛	6	720	320	200
&	8	720	430	250
38TD8⅛	9	720	480	(1) 250
38TDD8⅛	10	720	530	300
(6, 9 & 12 Cyl. only)	12	720	640	(2) 360
	4	900	270	250
	5	900	330	250
	6	900	400	250
38D8⅛	8	900	540	310
	9	900	600	(3) 310
	10	900	680	375
	12	900	800	(4) 450

Maximum temperature differential through engine—
Water — 10° All engines.
Oil — 30° All engines.

(1) 300 on 38TD8⅛ & 38TDD8⅛
(2) 400 on 38TD8⅛ & 38TDD8⅛
(3) 375 on 38TD8⅛ & 38TDD8⅛
(4) 500 on 38TD8⅛ & 38TDD8⅛

Auxiliary systems are required by all engines. The following pages show diagrams of the various systems as applied to Fairbanks Morse engines. The systems covered are jacket coolant, fuel oil, lubricating oil, air cooler, air and gas fuel.

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ECONOMIC ANALYSIS PROGRAM

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APPENDIX 3 - ECONOMIC ANALYSIS PROGRAM

METHOD USED IN THE LIFE CYCLE COST ANALYSIS

The basic formula used is the following:

If a cost inflates at a rate r, and the discount rate is d, then the total Present Worth (P.W.) of payments of one dollar every year over a life cycle of n years is given by the Present Worth Factor (PWF):

$$PWF(n,r,d) = \frac{1}{(d-r)}\left[1-\left(\frac{1+r}{1+d}\right)^n\right] \quad , \text{ for } r \neq d$$

or

$$= \frac{n}{(1+d)^n} \quad , \text{ for } r = d$$

For energy costs, the inflation rate is the fuel price escalation rate. For other costs, it is taken as the general inflation rate.

For mortgage payments, the details are as follows. For an amount of 1 dollar borrowed for Y years at an interest rate of i with a discount rate d, the Present Worth of Y annual payments is the following:

MPF = Present Worth Factor For Total Mortgage Payments

$$= \quad \frac{i}{d} \quad \frac{\left[1+\frac{1}{(1+d)^y}\right]}{\left[1-\frac{1}{(1+i)^y}\right]}$$

Of this, the Present Worth of the loan interest payment as a fraction of the loan principal amounts to the following:

INT = (PW of Total Mortgage Loan Interest)/Loan Principal

$$= \frac{i}{d} \frac{\left[1 - \frac{1}{(1+d)^Y}\right]}{\left[1 - \frac{1}{(1+i)^Y}\right]} - \frac{1}{(d-i)}\left[1 - \left(\frac{1+i}{1+d}\right)^Y\right] \left\{ \frac{i}{\left[1 - \left(\frac{1}{1+i}\right)^Y\right]} - i \right\}$$

$$\equiv MPF - P.W.F.(Y,i,d) \left[\frac{1}{\left(\frac{1}{1+i}\right)^Y} - 1 \right]$$

The Present Worth of the capital cost over the life cycle is the following:

$$\left.\begin{array}{l}\text{P.W. of Capital Cost Before}\\ \text{Income Tax Deduction}\end{array}\right\} = DP + (FC - DP)\ (MPF).$$

In evaluating the Present Worth of the Insurance costs, the Maintenance Costs, and the Property Tax, the inflation rate assumed is the general inflation rate.

The depreciation is calculated using a straight-line depreciation method, over the assumed depreciation lifetime.

The net life cycle cost is given by the following

= [Capital Cost + Energy Cost + Insurance & Maintenance Cost +

Property Tax] - Salvage Value

- Income Tax Savings.

In the above, the Income Tax Savings are given by the following expression for a private corporation:

Income Tax Savings =

(Income Tax Rate) [Depreciation + Energy Cost + Loan Interest

+ Insurance & Maintenance Cost + Property Tax]

The Return on Investment is the value of the discount rate at which the life cycle savings become zero. The discounted Payback Period is defined as the reciprocal of the Return on Investment.

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