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HEAT PUMP CENTERED INTEGRATED COMMUNITY ENERGY SYSTEMS

System Development

Georgia Institute of Technology Interim Report

ICES INTEGRATED COMMUNITY ENERGY SYSTEMS

ARGONNE NATIONAL LABORATORY

ENERGY AND ENVIRONMENTAL SYSTEMS DIVISION

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ARGONNE NATIONAL LABORATORY
9700 South Cass Avenue
Argonne, Illinois 60439

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System Development
Georgia Institute of Technology Interim Report

by

D. W. Wade, PE, B. C. Trammel, and
B. S. Dixit, Ph.D, PE

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Georgia Tech Research Institute

and

D. C. McCurry, PE and B. A. Rindt, PE
McCurry and Associates, Inc.

950934?

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PREFACE

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

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

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

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

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

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

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

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

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

Nine parallel concepts are under development in the initial, System Development, phase. The report which follows presents the interim findings in the development and analysis of a concept being developed by Georgia Institute of Technology.

ABSTRACT

Heat Pump Centered-Integrated Community Energy Systems (HP-ICES) show the promise of utilizing low-grade thermal energy for low-quality energy requirements such as space heating and cooling. The Heat Pump - Wastewater Heat Recovery (HP-WHR) scheme is one approach to an HP-ICES that proposes to reclaim low-grade thermal energy from a community's wastewater effluent. This report develops the concept of an HP-WHR system, evaluates the potential performance and economics of such a system, and examines the potential for application. A thermodynamic performance analysis of a hypothetical system projects an overall system Coefficient of Performance (C.O.P.) of from 2.181 to 2.264 for wastewater temperatures varying from 50° F to 80° F. Primary energy source savings from the implementation of this system is projected to be 5.014 QUADS, or the energy equivalent of 687 millions tons of coal, from 1980 to the year 2000. Economic analysis shows the HP-WHR scheme to be cost-competitive, on the basis of a net present value life cycle cost comparison, with conventional residential and light commercial HVAC systems.

1.0 SYSTEM CONCEPT

1.1 INTRODUCTION

The Heat Pump Centered-Integrated Community Energy System (HPC-ICES) concept presented in this report is essentially a cascaded heat recovery scheme which extracts low-grade thermal energy from wastewater effluent by means of electrically driven heat pumps. From an energy utilization standpoint, the concept represents a method of using portions of the lowest quality energy available in the community energy degradation cycle.

Figure No. 1-1 is a simplified schematic of the basic system elements. These elements consist of: a primary heat source, a central station heat pump, a thermal distribution system, and individual end-user heat pumps. Detailed discussions of each element will be presented in the following sections.

The technical feasibility of wastewater heat recovery appears well within the realm of believability. Institutional acceptance barriers, economic viability and the efficiency of resource utilization by such a scheme are somewhat in doubt and will be addressed by various sections of this report and by detailed application analyses in the Final Report.

The remainder of this section will describe each of the system elements, component configurations, system operating modes, applications and potential benefits from adoption of the scheme. In-depth analyses of system characteristics are presented in subsequent sections.

1.2 SYSTEM ENERGY FLOW

Conventional community heating and cooling systems use high grade energy sources such as natural gas or oil burned directly in home heating equipment or rural use of electricity generated at efficiencies less than 40 percent. As depicted in Figure 1-2, both processes lose substantial

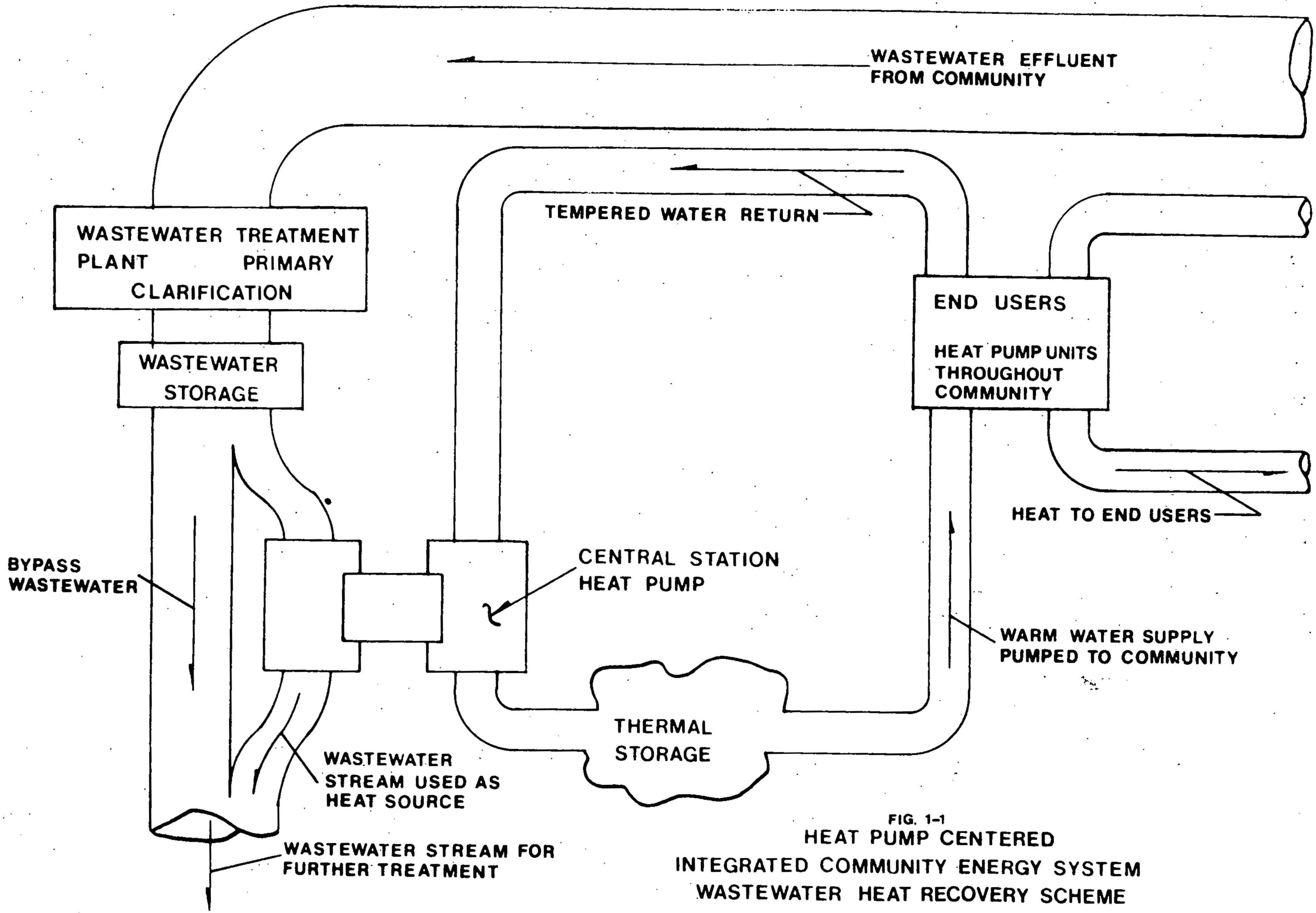


FIG. 1-1
HEAT PUMP CENTERED
INTEGRATED COMMUNITY ENERGY SYSTEM
WASTEWATER HEAT RECOVERY SCHEME

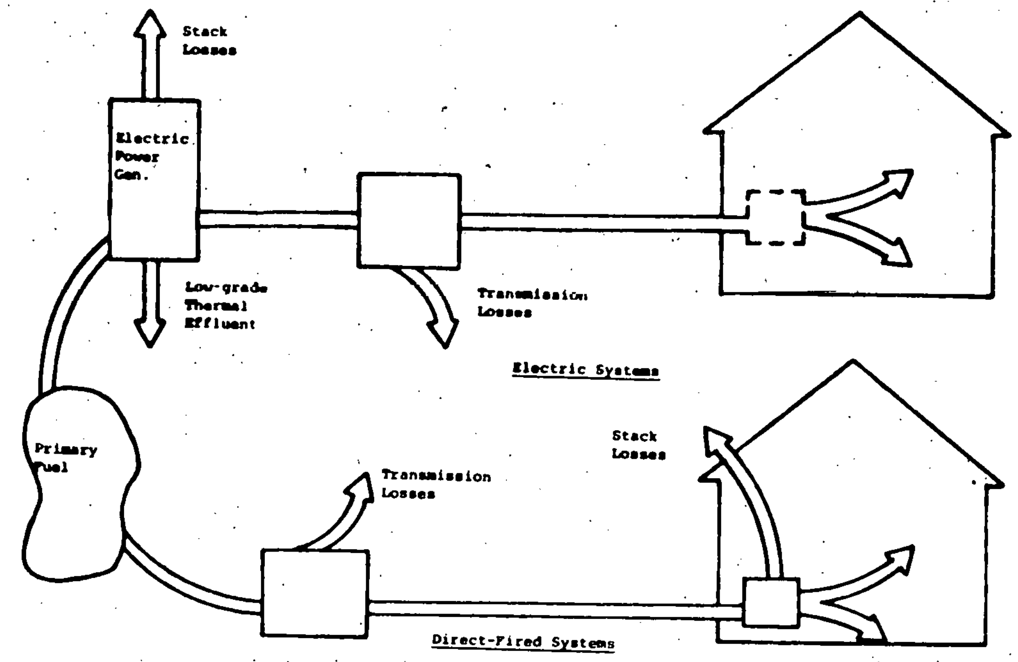
amounts of energy in the conversion process from primary fuel source to end use.

One way to improve the utilization efficiency of high grade energy sources is to use waste heat from high quality energy processes to satisfy lower quality requirements. The effect is to degrade energy quality in a cascading arrangement so that the highest quality energy is not used to satisfy low quality needs.

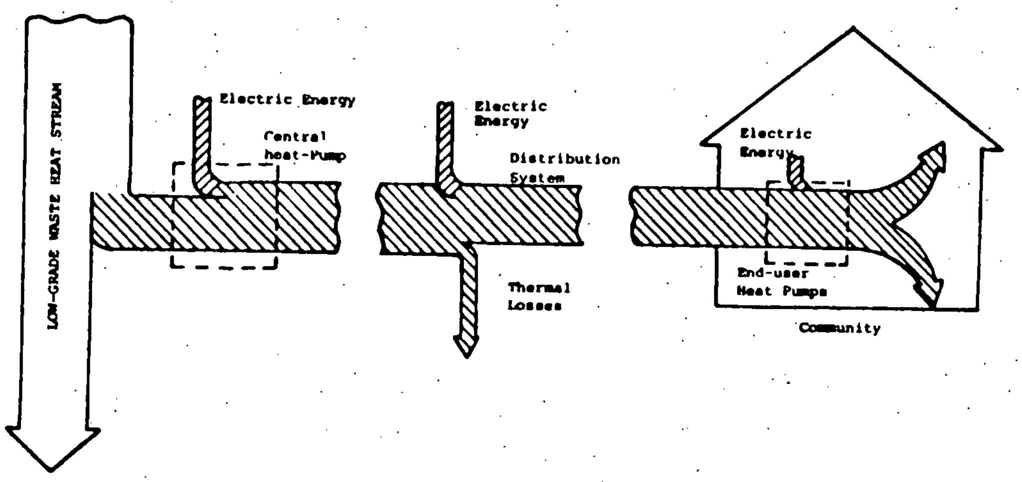
In the case of our community energy system discussed previously, it may be possible to utilize waste energy streams from a community's industries, electric generating plants and other high-grade energy users to satisfy portions of the community low-grade energy needs such as space heating. Waste heat utilization schemes such as this would lessen the requirements for new high-grade energy sources and improve the energy utilization efficiency of the resources currently consumed by high quality energy users.

One virtually untapped low-grade energy source is the wastewater effluent streams generated by industry, power generation, residences and the commercial sector. Admittedly, the desirability of this energy source is low due to its low-energy quality and contaminated state, however, the amount of energy contained is significant when one considers the quantities involved. If we are to fully integrate energy use by the community, we must consider every available energy source.

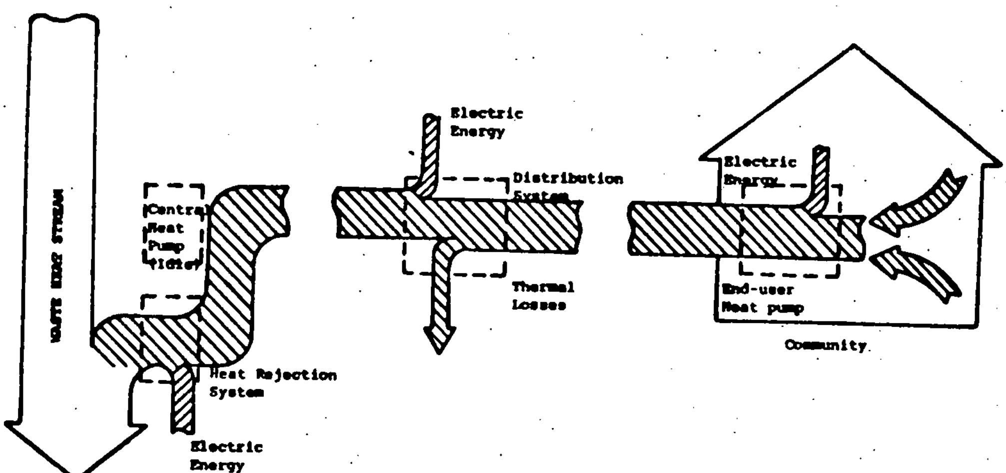
The energy flow scheme presented by Figure 1-3 is a representation of how low-grade heat from a community's wastewater stream could be utilized for building space heating. Heat could be extracted from industrial and residential sanitary sewer effluent and from cooling water streams discarded from power plants and industrial users by means of heat



CONVENTIONAL SYSTEMS
FIG. 1-2



HPC-ICES
HEATING MODE
ENERGY FLOW
FIG. 1-3



HPC-ICES
COOLING MODE
ENERGY FLOW
FIG. 1-4

pumps. The heat pumps would be used to elevate the temperature of the heat sufficiently for use in space heating. The Heat Pump Wastewater Heat Recovery Scheme (HP-WHR) applied to a community would require elements which would:

- Insure adequate energy supply to the Community.
- Distribute the thermal energy to the Community.
- Allow the end-users in the Community to utilize the thermal energy.

Additionally, the distribution system and end-user heat pumps could dissipate heat extracted from buildings during the cooling season. This process is depicted by Figure 1-4.

1.3 SYSTEM DESCRIPTION

A complete description of the HP-WHR scheme requires a discussion of each of the system's major components, their characteristics, and how they interact. Referring again to Figure 1-1, these components are:

- Primary Heat Source
- Central Station Heat Pump
- Thermal Distribution System
- Individual End-User Heat Pumps

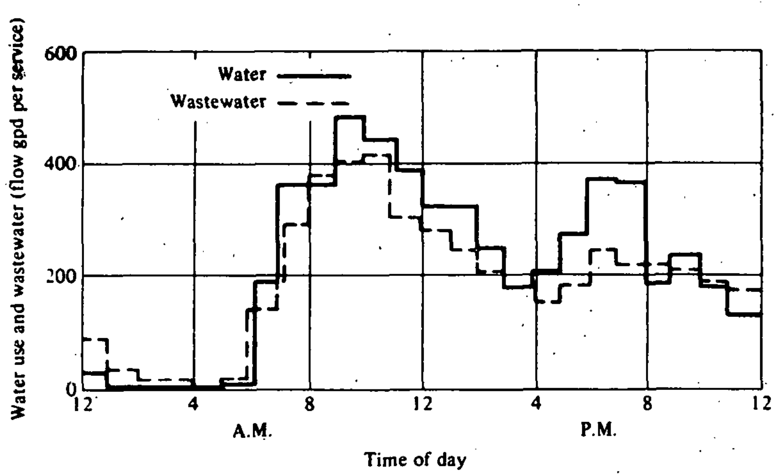
1.3.1 PRIMARY HEAT SOURCE

The primary source of waste heat the HP-WHR scheme seeks to make use of is community wastewater effluents. This can include sanitary sewer discharges, storm sewer discharges and cooling water discharges. However, only sanitary sewer discharges will be addressed in this report.

We are all aware of the numerous warm water discharges into the sanitary sewer from our own residences. Bathing, cooking, washing and many other home activities use hot or warm water which is eventually discharged to the sanitary sewer system. Similarly, but on a much larger scale, industrial and commercial activities create large volumes of warm water which is discharged to the sanitary sewer system. Common industrial sources are cooling water, spent process water, wash water and blowdown from heating and cooling systems.

Many disadvantages associated with using sewage effluent are apparent due to the dirty nature of wastewater. The incentives which must be kept in mind when approaching the problem are that the collection system for the energy contained in the stream is already installed and new capital is not required to build the collection system. Secondly, the potential for energy recovery will be greatest where it is needed most, i.e. in heavily populated, dense metropolitan areas.

The most important characteristics of the sewage effluent will be its quantity, thermal quality and physical properties. These characteristics will of course, be site specific and vary somewhat on a daily, weekly and seasonal basis. It is possible to make several general comments about the nature of each characteristic. The quantity of wastewater from a community is generally expressed by sanitary engineers in terms of gallons per day. On an hourly basis the flow can vary substantially from the average daily flow. It is usually characterized by two distinct flow peaks as shown in Figure 1-5. Flows can also drop substantially over weekend and holiday periods in areas where there are a large number of industrial discharges.



SOURCE: <u>WATER SUPPLY AND POLLUTION CONTROL</u>[8]
J. W. Clark, <u>et al.</u>
Fig 1-5

One other significant factor that can influence the quantity of wastewater is ground and surface water infiltration. Infiltration of ground and surface water into the sewer system will vary seasonally and with the age and design of the sewer system. In older systems where storm and sanitary sewers are combined, an extreme variation in flow of several hundred percent can occur during heavy rains. Additionally, the influx of surface water during the winter months will tend to lower sewage temperatures. This could be significant, especially where melting snow infiltrates into the sanitary sewer system.

Thermal quality of the wastewater will vary according to site specific conditions. Industrial waste streams tend to be at higher temperatures than residential discharges. For that reason, communities with high industrial concentrations may offer more potential for energy recovery than strictly residential communities. It is even conceivable that some wastewater effluents may be utilized without the need for an initial temperature boost by the central station heat pump.

In predominantly residential communities there will be some concern about the thermal content of the wastewater streams being tapped. A thorough search of the literature concerned with wastewater treatment revealed that effluent temperatures are not generally required or considered in the design of treatment plants. Similarly, it is rare for existing treatment plants to record the temperature of the community effluent entering the plant.

The lack of temperature data will hinder initial HP-WHR system designs. However, if the concept appears feasible, temperature measurements are relatively simple to obtain with recording thermometers. The necessary measurements will be discussed further in Chapter 8 - COMPONENT TESTING.

A lower temperature limit approaching ground temperature seems appropriate for most sewer collection systems. Since ground temperature is relatively constant throughout the year and is considerably above ambient air temperatures in the winter months, it would appear that even in the worst case the wastewater stream would be preferable as a heat source to the lower temperature ambient air.

Physical properties of wastewater streams deserve special attention on a site specific basis. Like the other properties of effluents, industrial discharges will introduce a variety of contaminents into the sanitary sewer system. In some cases the corrosive or fouling nature of industrial discharges may render the wastewater stream unusable.

As a rule, the sanitary sewer effluent entering a sewage treatment plant will have the viscosity and density which is approximately the same as pure water. The major impurities in the effluent are suspended particles which may be removed by settling or filtration, colloidal particles smaller than 1μM, dissolved impurities such as volatiles and minerals and, of course, a considerable microbiological population. A typical analysis of impurities in sanitary sewer effluent will be as follows:

300 mg/liter Suspended Solids

500 mg/liter Dissolved Solids

The remainder of the effluent is essentially water.

The impurities normally found in municipal sewage are routinely handled by conventional treatment practices. The highly polluted nature of the effluent has caused designers to avoid using it whenever possible due to expected fouling problems.

In the actual design of an HP-WHR scheme, considerable attention will be required to provide adequate fouling allowances for heat transfer surfaces and provisions for cleaning and inspecting components subjected to contact with effluent streams.

1.3.2 CENTRAL STATION HEAT PUMP

The Central Station heat pump equipment element of the system will serve to extract energy from the primary heat source and elevate its temperature sufficiently for distribution and use throughout the community. Although only one unit is shown throughout the schematics in this report, it may be advantageous to combine multiple units in series or parallel arrangements to achieve the desired temperature elevation or improve part load performance characteristics.

Central station heat pumps have been used extensively over the past thirty years in business and industry. The most notable example of this is "water chilling" machines used by the air conditioning industry in large scale building air conditioning. The central station heat pump used in the HP-WHR scheme will be similar to the machine used for building air conditioning systems, except selection procedures will be modified for the required service conditions.

The central station heat pump operates on a closed refrigerant cycle consisting of working fluid compression, condensation and evaporation. Typically, the working fluid will be one of the freon refrigerants selected for characteristics suitable for the temperature rise required and the machinery's compressor characteristics.

The evaporator and condenser heat transfer components are commonly shell and tube heat exchangers comprised of steel shell and non-ferrous tubes. Depending on temperature rises and flow quantities, the water in the tubes may make multiple passes. Tube material and thickness is of special importance in the HP-WHR scheme due to the abusive nature of wastewater effluent. Some experience may be required before an optimum heat exchanger design can be settled on for this application.

The compressor of the central station heat pump can be of the reciprocating, screw or centrifugal type driven by an electric motor. In practice, it is anticipated that the reciprocating compressor will have little applicability due to its limited capacity and somewhat lower operating C.O.P. than centrifugal or screw type compressors. For the purposes of this study, a centrifugal compressor is used as the central heat pump since it is widely used by the air conditioning industry for central station air conditioning applications in the temperature ranges suggested.

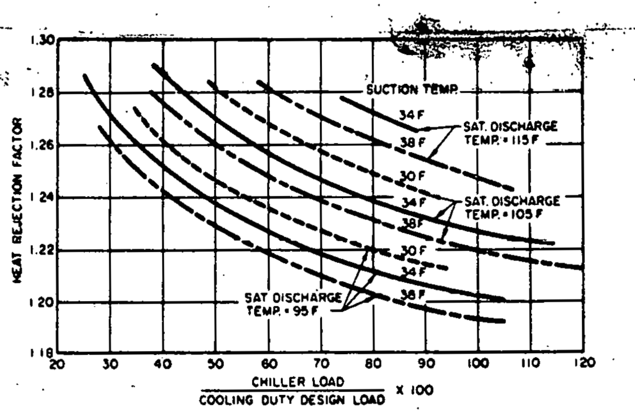
A screw compressor will have similar performance characteristics to the centrifugal type. However, commercially available units have somewhat lower capacity than their centrifugal counterparts. In some cases a screw compressor may be advantageous due to its ability to operate at higher temperature and pressure differentials.

The compressor of the central station heat pump is thus assumed to be a centrifugal type driven by an electric motor. Capacity control will be accomplished by inlet vanes to the centrifugal impeller. It is also anticipated that the machine will be a hermetic design, thus allowing the refrigerant to absorb the heat given off by the electric motor.

Figure 1-6 is a performance curve for a typical central station water chilling machine used in heat recovery application. The data is presented in terms of Heat Rejection Factor (HRF), which is a ratio of the heat rejected at the condenser to heat absorbed by the evaporator. For refrigerant-gas-cooled hermetic compressors the Coefficient of Performance is defined approximately by the following formula:

$$COP = [HRF \times .99]/[HRF - 1]$$

Note that the on-site COP of the central machine is quite high, ranging from 5 to 7 for the machine shown in Figure 1-6. In actual practice machine COP could be maximized within economic constraints by judicious selection of central station heat pump components. The overall coefficient of performance of the central station will be reduced somewhat by power requirements for auxiliary equipment such as pumps, controls, etc.



Performance of Typical Large, Moderate Head, Hermetic Centrifugal Compressor (R-11)

FIGURE 1-6

SOURCE: ASHRAE Handbook and Product Directory[49] 1976 Systems Volume

Similar performance data for cooling COP's are available from equipment manufacturers or may be derived from the ICES Technology Evaluation Number ANL/CES/78-2 entitled, "Central Cooling-Compressive Chiller"; V.E. Christian, 1975.[5]

1.3.3 THERMAL DISTRIBUTION SYSTEM

Two types of thermal distribution systems were considered for use with the HP-WHR system. One system would utilize portions of the wastewater which has been clarified to transport thermal energy from the central station to the end-users. Once the energy was extracted from the transport media it would be discarded to the sanitary sewer system and returned to the waste treatment plant.

The other distribution system would be a closed loop circulating system utilizing supply and return piping from the central heat source. The closed loop circulating system is commonly used in heating and air conditioning systems for thermal energy transport.

Both thermal distribution systems would incorporate some form of storage component to provide for optimum storage of heat extracted from the primary heat source and for adequate peaking capacity during maximum energy demand periods.

The numerous problems associated with the proposed utilization of clarified wastewater for a thermal transport media lead us to adopt a more conventional circulating water system as a basis for analysis. The clarified wastewater scheme is presented in Chapter 7 - IDENTIFIED VARIATIONS for consideration in specialized cases.

1.3.4 CLOSED LOOP SYSTEM

A closed loop thermal distribution system would incorporate supply and return piping mains throughout the community. This would allow the transport media to be returned to the central station to be regenerated.

Closed loop circulating water systems are commonly used in heating and air conditioning systems and industrial process applications. The advantage of the closed loop system is primarily its freedom from external contamination. Its disadvantages are higher installed cost and greater pumping friction loss due to the increased distance of pressure-flow piping.

It is anticipated that the energy requirements for thermal transport media circulation could be minimized by proper design of the pumping system to incorporate variable pumping or parallel pump staging. Likewise, two pipe circulating systems will enjoy some installation economies since trenching and associated construction costs for one or two pipes are essentially the same.

One final paragraph concerning either system is appropriate on the subject of piping materials. Recent advances in piping materials have made plastic-type piping almost an industry standard for municipal water distribution piping systems. The low temperature application discussed here will be constructed with plastic or fiberglass piping, taking advantage of the latest cost reduction techniques.

1.3.5 THERMAL STORAGE

Regardless of the type distribution system selected, it will generally be desirable to include a device for storing the thermal energy extracted from the wastewater stream and apportioning it to the community as required. Consideration of the storage volume will be of prime importance when actual design is undertaken.

In most cases it is anticipated that the quantity of transport media will dictate some form of underground, covered vessel. To be cost-effective this may take the form of a plastic-lined earthen reservoir with a floating or air-supported canopy.

In the case of a clarified wastewater system, it may be advantageous to split the storage requirements between underground and elevated vessels. This would allow optimum pump sizing since peak flows could be supplied from the elevated tank, as in potable water systems.

1.3.6 SYSTEM INSULATION

Insulation for piping and thermal storage systems will have to be evaluated on a case-by-case basis. The systems presented in this report assume that distribution system thermal losses will be kept to a small percentage of the heat content of the transport media.

Calculations presented in the analysis section of this report for the thermal losses from buried plastic pipe at low temperature differentials are expected to indicate that underground pipe insulation may not be cost effective. For that reason, the analyses presented in this report assume no thermal insulation on the underground piping system.

1.3.7 END-USER HEAT PUMPS

The end-user heat pump system utilized in the HP-WHR scheme would be the type commercially available in the ½ to 25 ton cooling range. No substantially different application techniques from common practices currently employed are anticipated for the end-user systems.

An excellent technology evaluation of unitary water-to-air heat pumps is available in a report by the same name prepared by J.E. Christian of the Oak Ridge National Laboratory ANL/CES/TE 77-9 (Reference 7). Figure 2.1 of that report is included here as Figure 1-7 and serves to describe the operation of the unitary water-to-air heat pumps which will be utilized by this scheme.

Performance characteristics vary substantially with manufacturer and size of the specific equipment. Typical manufacturer's COP's at standard rating conditions range from 2.5 to 3.4 on the heating cycle and from 2.44 to 2.66 on the cooling cycle. Part load and seasonal performance factors will also vary substantially.

The use of a relatively constant temperature water source offers substantial energy performance improvement and the promise of more relaible system operation. Improved heating performance is especially attractive when compared to air-source heat pumps due to the absence of the attending defrost cycles and supplemental electric strip heat output.

Heat pump technology has advanced considerably over the last few years. Further improvements in new heat pump design can be expected in

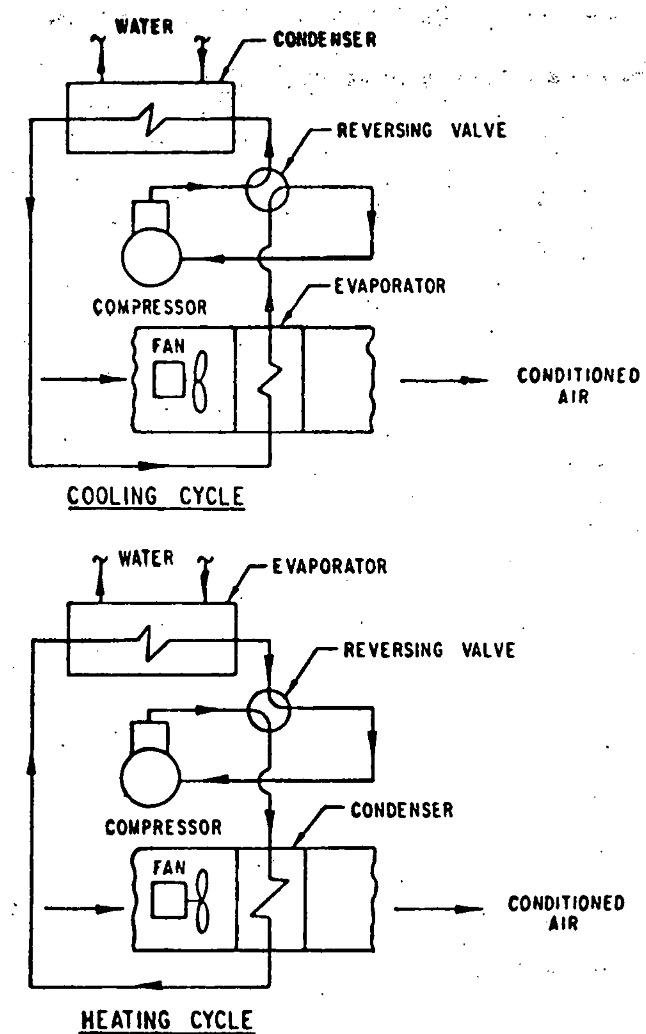


FIGURE 1-7. Water-to-Air Heat Pump Showing Cooling and Heating Cycles
SOURCE: Unitary Water-to-Air Heat Pumps [7]
ANL/CES/TE 77-9

COMPONENT DESCRIPTION

The water-to-air heat pump actually is a reverse-cycle, self-contained refrigeration system as shown in Fig. 1-7. In the cooling cycle, heat is absorbed through the air coil from the conditioned space and transferred through the refrigeration process, to water circulating through a water-to-refrigerant heat exchanger. In the heating cycle, heat is absorbed from water circulating through the water coil and transferred to the air coil by the refrigeration process where it is then supplied to the space being heated.

the areas of compressor capacity control and prime mover types. Widespread interest in water-to-air heat pumps for residential use should produce even further improvements and decreased unit costs associated with increased quantity production.

1.4 SYSTEM OPERATION

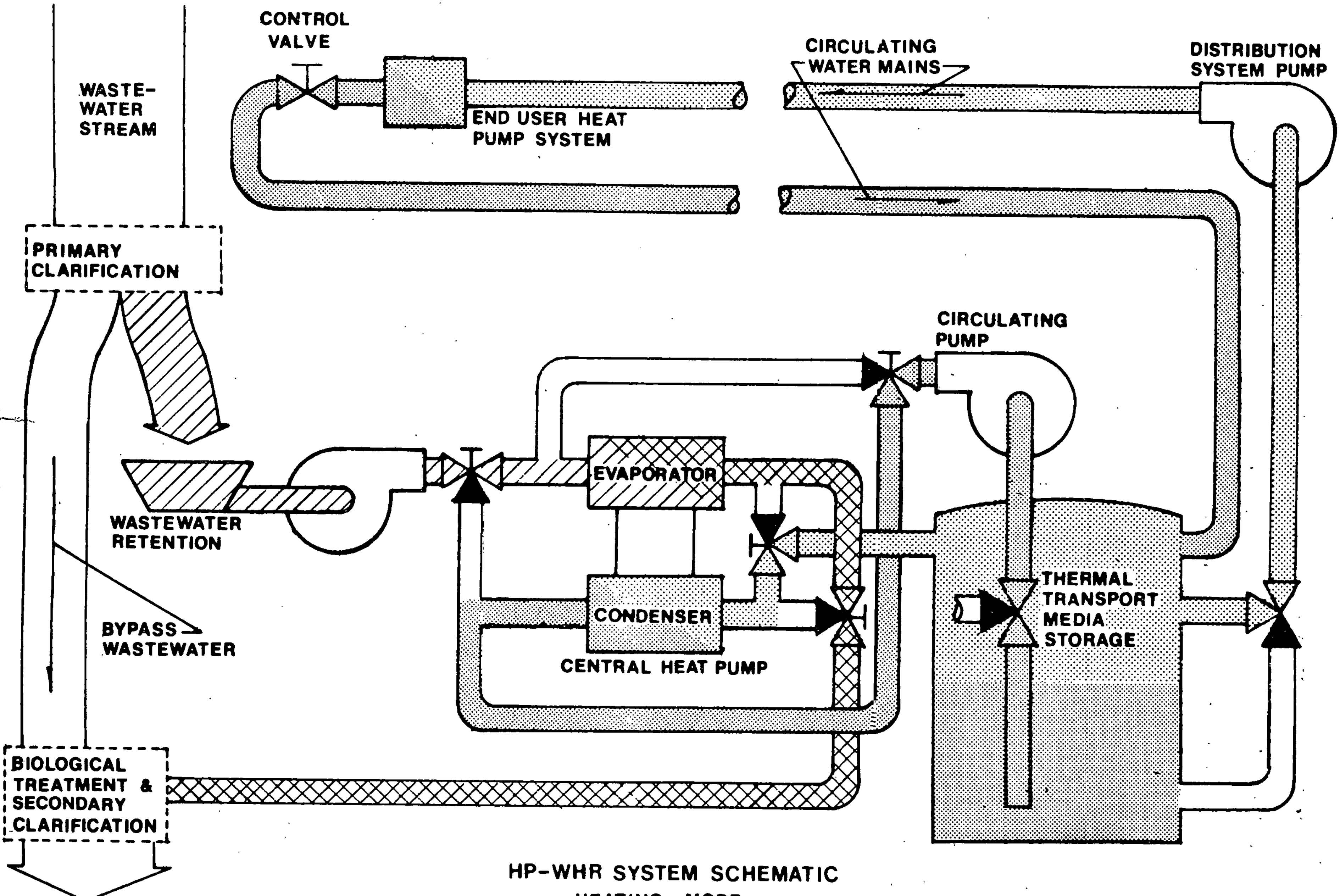
The Heat Pump Wastewater Heat Recovery Scheme presented in this report is intended to operate in a heating and cooling mode. Actual operating procedures will vary with the size and complexity of the system so for simplicity's sake the description presented here will consider only a minimal number of functional components.

1.4.1 HEATING MODE

A schematic of the HP-WHR system operating in the heating mode is presented in Figure 1-8. A clarified wastewater thermal transport media is utilized.

In the heating mode, wastewater effluent enters the treatment plant and goes through a primary clarification process, then passes through the evaporator of the central station heat pump. A retention pond is shown immediately after the primary clarification process to level out diurnal peaks in the sewage flow. After passing through the central heat pump, the wastewater effluent continues through the sewage treatment process.

The thermal transport media is circulated between the central heat pump condenser, where it picks up heat, and the storage vessel. A mixing valve is provided in the storage vessel to indicate that full advantage of thermal stratification will be taken.



HP-WHR SYSTEM SCHEMATIC
HEATING MODE
FIG. 1-8

The thermal transport media is drawn from the storage vessel in response to system loads. Primary distribution lines would have some recirculation or pressure relief to insure a constant supply temperature during low load periods.

Each end-user heat pump would be cycled in response to a space mounted thermostat. Used transport media would be returned to the distribution system according to individual heat pump requirements. When the unit is inactive, no transport media would be consumed.

1.4.2 COOLING MODE

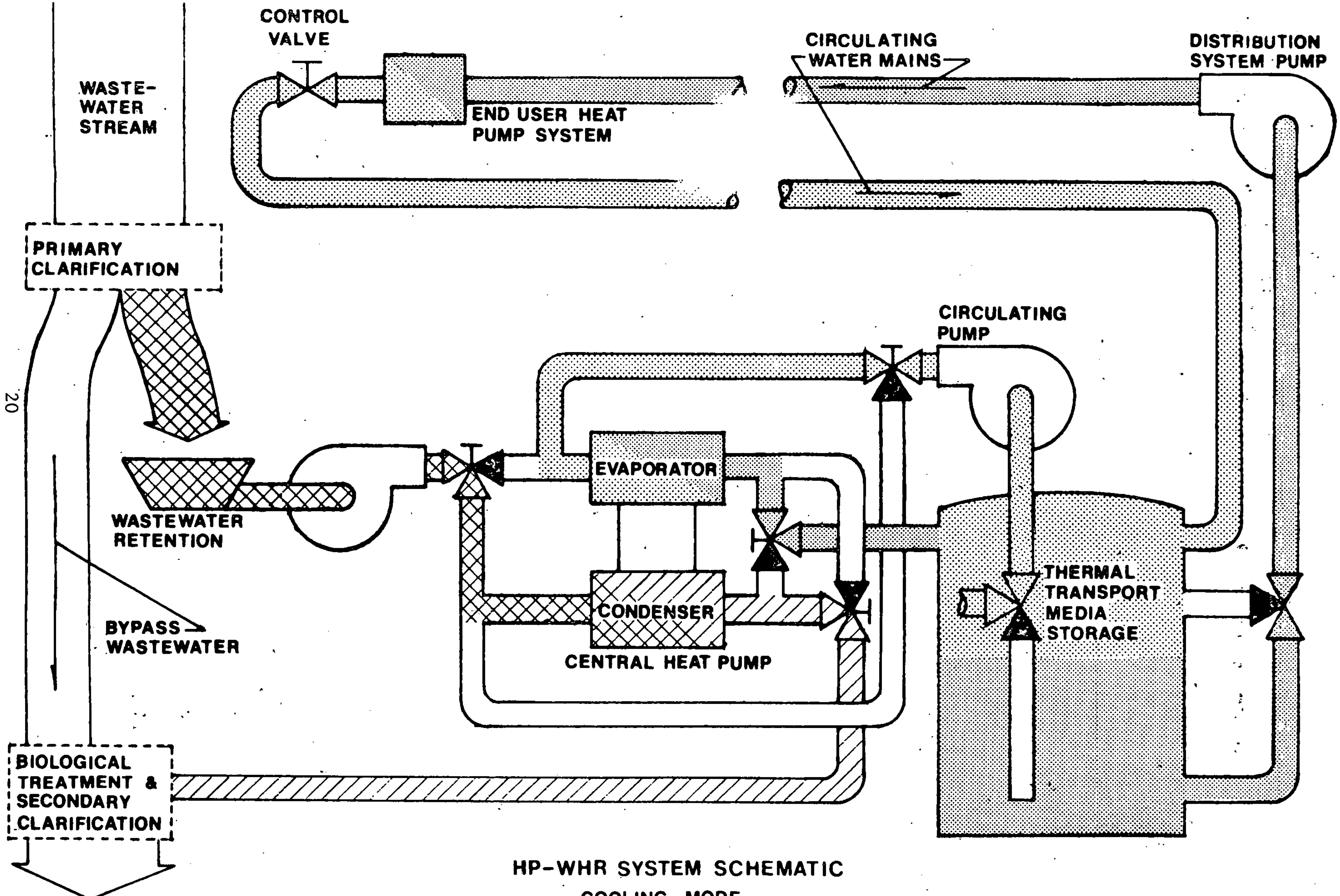
A schematic of the HP-WHR system operating in the cooling mode is presented in Figure 1-9. A recirculating thermal transport media is utilized.

In the cooling mode, wastewater effluent enters the condenser side of the central heat pump and then passes on to normal sewage treatment practice. Warm thermal transport media is extracted from the storage vessel, cooled and then recirculated to the storage vessel. The thermal transport media is then drawn from the storage vessel in response to system loads.

Each end-user heat pump would be cycled in response to a space mounted thermostat. Heat would be rejected by the pump to the transport media acting as the heat sink. Used transport media would be returned to the distribution system according to individual heat pump requirements.

1.4.3 PART LOAD OPERATION

The HP-WHR schematics presented in this section offer considerable flexibility for full and part load operation. Leveling wastewater flow over an 18-24 hour period will allow the central station heat pumps to operate under relatively constant high load factors. This will also



HP-WHR SYSTEM SCHEMATIC

COOLING MODE

FIG. 1-9

allow the most cost-effective sizing of the central station equipment since peak community thermal demands will be met from the transport media storage vessel.

1.4.4 SIMULTANEOUS HEATING AND COOLING

During mild weather conditions it is probable that simultaneous heating and cooling will take place throughout the system at different end-users. This will be especially true where there are commercial buildings mixed with residential construction.

In general, this operation will tend to benefit the heating performance of the system since less heat will need to be extracted from the waste-water stream. In effect,the heat pumps and distribution system will be transferring unwanted heat from the air conditioning process to heating needs throughout the community.

1.5 OPERATION AS A THERMAL UTILITY

The wastewater HP-ICES was originally conceived without regard to how the system would be owned and operated. In general, however, it appears that the system could be most advantageously operated as a thermal utility.

Operation as a thermal utility would require that the central heat pump, distribution pumps, and distribution system be owned and operated by the utility. Energy supplied to the end-users would be metered at the point the end-user connects to the distribution system. Each end-user would own and operate his own water-source heat pump system in order to use the thermal energy supplied by the utility company.

It is conceivable that any number of utility arrangements ranging from landlord owned and operated systems in a building complex to metro-politan systems owned and operated by municipal governments would be possible. A private utility company scheme could also be initiated.

An especially attractive arrangement would be a municipally owned and operated thermal utility serving the community's residential and commercial customers. Under this scheme the central heat pump and distribution system could be operated similar to or as a division of the water and sewer departments. Administrative matters pertaining to finance and revenue collection could be handled in a manner similar to water and sewer services.

The municipally operated utility scheme would enjoy a number of advantages including: overall responsibility and control of water, sewer and thermal energy distribution systems, the ability to finance the system through municipal bonds, administrative, operating and maintenance personnel familiar with utility operations and water distribution through piping systems. Municipal governments will also be better able to accept the risks associated with long term ownership and operation of the HP-ICES concept.

1.6 SYSTEM APPLICATIONS

The wastewater HP-ICES concept may be applied to both new construction areas and retrofit situations. In evaluating an application of the concept, consideration must be given to both new and retrofit conditions.

In general, it is anticipated that each end-user will own and operate his own water-source heat pump system. In operating his system the end-user would purchase thermal energy from the thermal utility either on a Btu basis or water flow rate basis. There are, of course, special cases such as a building complex of rental units where ownership may be retained by the landlord and thermal energy may not be metered.

It is apparent that new construction will offer the most potential for economic inclusion of the wastewater HP-ICES concept. The advantages seen in new construction projects include lower initial cost for end-use systems, a minimum amount of prejudice associated with changing existing systems, and overall distribution system installation economies associated with initial infrastructure construction (such as common trenches for thermal, water, and sewer utility systems).

The more difficult case is the retrofit situation where the thermal utility must be installed in existing areas requiring relatively expensive unitary installation costs and the retrofit of conventional end-use systems. In any event, the retrofit case will be the more expensive per unit and will make the HP-ICES more difficult to economically justify.

2.0 POTENTIAL APPLICATIONS

2.1 INTRODUCTION

The goal of the developing Integrated Community Energy System (ICES) technology is to optimize the available resources in meeting the energy needs of various communities through new and innovative methods. While the primary products of a Heat Pump Centered-Integrated Community Energy System (HP-ICES) are low-level thermal energy services for space heating and/or cooling needs, the ultimate determination of the kind, number and amount of energy services provided should be based on the optimum combination of energy efficiencies, economics, and environmental considerations. In the sections of this chapter, factors affecting the potential for application of this particular HP-ICES, including availability of resources, climate, economic considerations, institutional/community factors, system reliability, and unique application conditions, will be discussed. From these considerations, a projection of the potential for application (through the year 2000) will be developed.

2.2 FACTORS AFFECTING APPLICATION

The aim of an ICES is to optimize energy usage in a community, taking into account the many factors that may affect that energy usage. The focus of this section of Chapter 2 will be to catalogue and discuss the more important factors that will determine the applicability of this HP-ICES scheme to various communities.

2.2.1 AVAILABILITY OF RESOURCES

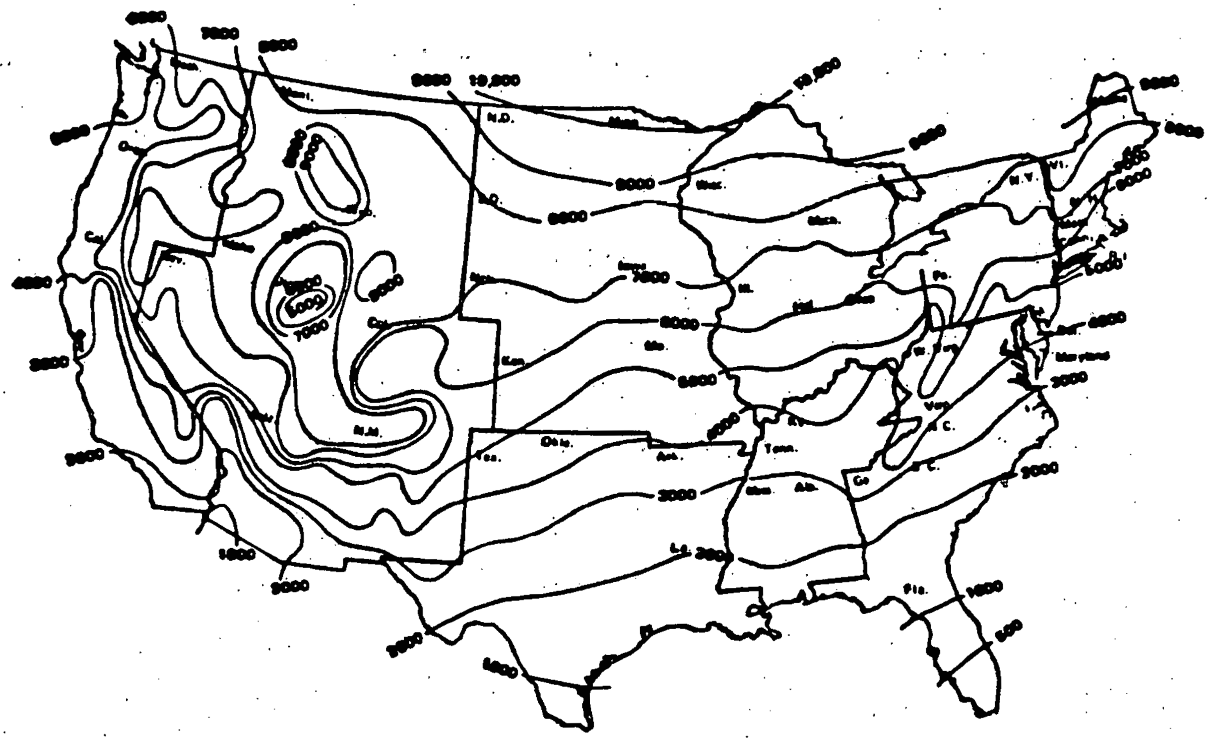
A viable but presently untapped source of low-grade thermal energy is the heat discharged during domestic energy usage and rejected heat from industrial processes. Very low-grade thermal energy, degraded from higher quality sources through domestic chores such as cooking, washing, and bathing, is regularly discharged from the individual household because of its low thermodynamic availability. Likewise, waste heat from industrial processes and heavy machinery cooling systems is discharged because of its low thermodynymic availability. A common "sump" for this low-grade thermal energy is the sanitary sewer collection system in the community. Depending on the size and make-up of the community, and the degree of development of the attending sanitary sewer system, billions of Btu's per day may be available from this thermal source. The assumption that the energy availability of this source hinges upon is that the temperature of the community wastewater effluent will be higher than or equal to ground temperature. Several site-specific variables may affect the effluent temperature by a few plus or minus degrees, but preliminary investigation seems to validate this assumption. A quick estimate of the energy available from this source can be developed from flow and temperature data for each site of application. The principal advantage of utilizing this source is that it would be almost universally available; a community large enough to consider an integrated energy system would in all probability have an interceptor sewer system and/or treatment facility capable of supplying enough thermal energy to make this scheme technically feasible. A second major advantage is that the thermal source would require no additional interface with the Central Station Heat Pump equipment(i.e. the effluent could be piped directly to the central heat pumps).

Of course, several other thermal energy sources, such as solar, combustion of solid or gaseous wastes, or composite energy sources could be considered for easy adaption into an HP-ICES scheme. However, since a community with a thermal energy consumption density sufficient to justify an HP-ICES would likely have a well-developed wastewater collection system, wastewater heat recovery will be evaluated as the primary thermal energy source for this particular ICES scheme. Other variations in energy source are discussed in Chapter 7 - IDENTIFIED VARIATIONS.

2.2.2 CLIMATE

It was mentioned in the preceding section that the base assumption for application of this scheme was that the minimum temperature of the wastewater effluent would be approximately equal to ground temperature. It is obvious from this assumption that climate will have a significant influence upon the operating conditions of the system by affecting the conditions of the thermal energy source, as well as affecting the capacity of the main heat pump components, and the heating/cooling loads at each end-user location. The conditions at the thermal source are site-specific (with climatological influences) and should be documented according to the recommended measurements discussed in Chapter 8 of this report. The addition of auxiliary thermal energy sources may also significantly affect the initial conditions. Climatological areas of the U.S., and the resultant typical heating loads and system capacities, have been well documented and will be discussed in this section.

For the purposes of analyzing heating and cooling loads, the United States can be divided into climatological sections according to the number of degree-days in a heating or cooling season. (A degree-day is based on the difference between actual average outside dry bulb temperature and a 65°F comfort index. As an example, one day with an average temperature of 50°F equals 15 degree-days heating: $[(65° - T_{outside}) \times \text{No. days}]$. Figure 2-1 shows the heating degree-day regions in the continental U.S.



SOURCE: <u>Energy Use and Conservation In The Residential Sector: A Regional Analysis</u>, Rand Corporation[27]

(Heating Degree-Day Map of The United States)

FIGURE 2-1

In a study entitled "Heat Pump Technology" by Gordian Associates[10] a comprehensive computer simulation of the heating and cooling load of a typical residence and commercial building was developed. As input, the program required various parameters such as building construction details and materials, orientation, geographic location, and local weather conditions. (Building details for the simulation model are contained in Appendix A .) With the validated computer model, the subject buildings could be artificially moved to any climatic region in the country to obtain typical heating and cooling loads. Programs were run for 9 cities in the continental U.S. (Weather data for the test cities is shown in Table A-1 of Appendix A.) The results show loads in the commercial building ranging from 166,510 Btuh heating in Concord, N.H. to 206,119 Btuh cooling in Houston, Tx. and in the residential building ranging from 52,088 Btuh heating in Concord to 38,351 Btuh cooling in Tulsa, Okla. The tabulated results of the simulation can be found in Tables A-2 and A-3 in Appendix A. In general, the simulation showed that the heating and cooling load and the system heating or cooling capacity is dependent on outdoor temperature. This generalization from the computer simulation agrees with the performance data available from the equipment manufacturers - i.e., sensible and total (sensible + latent) cooling capacities are functions of indoor wet bulb and indoor and outdoor dry bulb temperatures. While the practice of artificially relocating the simulated buildings to different climatic regions may not give an entirely accurate indication of the community heating load to be placed on a proposed HP-ICES because of regional discrepancies in building construction and community composition and layout, it is effective to illustrate strictly the climatic effect on heating and cooling loads.

It must be explained at this point that the type heat pumps proposed for use in the HP-WHR scheme, water-source heat pumps, offer an advantage over the air-to-air heat pumps modeled in the computer program, especially in the colder regions. In the heating mode, with ambient temperatures lower than 47°F and relative humidities higher than 60%, standard air-to-air heat pumps show the tendency to frost on the outside (evaporator) coil. To remove the frost, the "defrost cycle" is automatically engaged and heat is taken from the conditioned space and circulated to the outdoor coil to melt the ice build-up. Essentially the unit is cooling the conditioned space during the duration of the defrost cycle, and the inside air must be tempered with auxiliary electric strip heat. The overall result is a lowered Coefficient of Performance (C.O.P.) for the air-to-air heat pump unit. Since, for a water-source heat pump, the heat exchange takes place with a water supply that is kept at temperatures well above the freezing point, no defrost cycle is needed. Therefore, the water source unit offers a higher C.O.P.

Because of the above-mentioned problem with icing on the outdoor coil and the required defrost cycle, (air-to-air) heat pumps have had higher acceptance in the regions of the U.S. with a small-to-moderate heating requirement and a proportionately larger cooling requirement (notably the South). However, with the thermal energy supplied in a circulating water medium and transferred (in the heating modes) into the conditioned space via a water source heat pump, and with a stable thermal energy supply (such as wastewater effluent), it is expected that system performance (in terms of overall C.O.P.) would be acceptable in any of the climatic regions of the U.S.

Climatic differences will obviously affect system sizing. From data developed in the subsequent chapters detailing performance and economics, it appears that an HP-WHR system sized at 1000 tons (12,000,000 Btuh extraction rate) and requiring a 3.0 MGD wastewater flow as the thermal source would serve approximately 470 typical homes in a New England clime (6820 Degree-days heating and approximate peak load of 41 MBtu/home) compared to 980 typical homes in a Southern clime (2350 Degree-days heating and approximate peak load of 21 MBtu/home). To generalize the climatic effect on system sizing, systems located in colder climes will serve proportionately less end-users than systems in the warmer climes for a given size of central station installation. As a corollary, service to a given number of end-users will require a larger central station heat pump and proportionately more wastewater flow, in the colder climes.

Climatic influences may also affect the applicability of the HP-ICES concept in an indirect manner. Building design and construction and community composition and layout vary noticeably from climatic region to region. For example, in the colder northeast regions, housing patterns tend to be more dense, the structures more well insulated, and typically there is a higher degree of industrial development for a given size community. These characteristics make the application of an HP-ICES desirable in these areas for the following reasons: the denser housing patterns make installation of a thermal energy distribution system more cost effective; better insulation decreases somewhat the heat loss from structures; and waste heat rejected from industrial processes into the community's wastewater collection system increases the amount of thermal energy available for extraction with an HP-WHR system. However, indirect effects such as these are difficult, if not impossible, to quantify through calculations in order to determine their effect on the feasibility of the HP-ICES concept. They will figure significantly, though, into the technical and economic viability of a system as determined in the site-specific performance and economic analyses.

Unfortunately, it is beyond the scope of this study to develop detailed guidelines for determining the applicability of an HP - ICES on the basis of climatic factors. Instead, the methodology for evaluating the climatic influences on the feasibility of such system on a site-specific basis will be developed in the Part II analysis of the demonstration communities. However, to summarize the previous discussions: 1.) Climate will directly affect the thermal demand related to space heating and cooling that will be placed on an HP-ICES. 2.) System design and sizing (including determining the number of end-users to be served by the system) will directly depend on the thermal demand on the system and thus will be affected, in turn, by climate. 3.) Indirect climatic effects seen in building construction and community composition will influence the feasibility of an HP-ICES, albeit through the procedures for performance and economic analyses. 4.) System performance (in terms of overall energy effectiveness) is expected to be acceptable in any of the climatic regions of the U.S.

2.2.3 ECONOMIC CONSIDERATIONS

A critical factor in judging the acceptability of an HP-ICES to the community is cost-effectiveness. "It should be stressed that, although the saving of fuel may be an acknowledged goal for the country, in a free market the individual consumer is expected only to pay for efficiency to the extent that the added investment could potentially be recovered through the savings in fuel consumption."[10] Thus, consumer acceptance can be expected to hinge upon a favorable comparison of savings in energy costs vs. additional cost of the individual heat pump unit plus the thermal utility expense. The factors entering into this comparison can be expressed in an equation for annualized cost:

$$\begin{aligned}\text{Annualized Cost} &= (\text{Installed Cost}) \times (\text{Capital Recovery Factor})\\ &+ (\text{Annual Energy Use}) \times (\text{Unit Price of Energy})\\ &+ \text{Annual Maintenance}\end{aligned}$$

An HP - ICES can be categorized as a capital intensive venture by virtue of the large first-cost associated with constructing the Central Station and distribution system. Such an installation effectively centralizes a major portion of the energy-related expenses and offers the economies of scale available in the form of municipal or institutional energy rates. Being a capital intensive venture, though, the economic viability of such a system will be significantly affected by the availability and cost of capital. Inflationary trends will increase the installed cost of an ICES (note the importance of installed cost in the equation for annualized cost), thus adversely affecting the availability of capital for such a system. The current cost of capital would also figure prominently by way of influencing the capital recovery factor. As costs and interest rates rise, either independently or jointly, the result will be a higher system annualized cost which must be apportioned to the end-users by means of the thermal utility charge.

Individual end-user space conditioning systems, on the other hand, can be categorized as energy intensive systems. The affects of inflation on system installed costs and on availability and cost of capital are rarely noticed because of the proportionately small expense of the system as compared to the overall cost of a residence, especially when the system costs are amortized through a typical new home mortgage. Escalating energy costs (which translate into higher operating costs for space conditioning) are immediately noticed by the consumer, however, because this portion of the annualized system cost is paid "out-of-pocket." Because consumers have become so energy conscious of late, the rising trend of energy costs may be regarded as a major accelerating economic factor favoring an HP-ICES. In fact, purchasers of space conditioning equipment have indicated in recent surveys (Arkansas Power & Light Co., 1974,[1] Professional Builder, 1976[26]) that lower operating and maintenance cost expectations were reasonable justification for purchasing a heat pump, and that an additional investment of up to $600 for energy conservation would be acceptable if the payout were within six years.[10]

It must be explained at this point that there is a trade-off to be optimized according to the end-users' best interest. If the consumer is to be expected to finance the incremental cost of a heat pump over conventional residential space conditioning alternatives, the Central Station and distribution system of an HP - ICES must be financed at the most advantageous costs and rates so that, when the annualized cost of the system is apportioned to the end-users through the thermal utility charge, the cost of the thermal utility plus the nominal charge for energy (electricity) required to run his system will compare favorably with the energy charge that could be expected for running a more conventional residential space conditioning system so that the expected payback period on the end-users' equipment would fall within an acceptable range.

It is expected that installation costs for an HP-ICES will remain well within the reasonable range. The major effect of high costs and interest rates would be to make the system less cost-competitive with other individual end-user system alternatives and thus slow the rate at which new customers would be attracted to tie-in to the system.

2.2.4 INSTITUTIONAL/COMMUNITY CONSIDERATIONS

Acceptance of an HPC-ICES in the community will depend heavily upon the acceptance from the individual consumer. Heat pump equipment marketed in the 1950's and early 1960's had serious problems with performance and reliability, in addition to substantial problems with installation and servicing. The net result of these compounding problems was a general rejection of heat pump equipment by the public. However, recent strides in improving performance and the current focus on energy conservation seem to be changing the attitude of the consumer. Recent surveys have shown that consumers would be willing to invest the additional first cost of a heat pump (up to approximately $600) for the improved performance and lower operating and maintenance costs. A brief look at consumer demographics supports this contention; the 1976 average new home cost of $48,600 indicates that the housing market has shifted to a position where the heat pump could be considered an alternative for the average new residential building. Non-residential consumers seem receptive to the heat pump market if an economic analysis of system owning and operating costs projects a payback period of from two (2) to five (5) years.

One potential stumbling block to community acceptance may be requirements set forth in state and local building codes. There are no specific provisions in the model building codes recognized by HUD and HEW (codes: BOCA, ICBO, National Board of Fire Underwriters, Southern Building Code Congress International) that cover heat pumps, although various related sections that cover mechanical refrigeration, electric strip heat, and general electrical requirements will apply. Generally, though, there are no provisions in the model building codes that are seen as overly restrictive.

A provision in the California Energy Code stating that electric resistance heating cannot exceed 10 percent of the building space heating requirement (for non-residential buildings) may prove troublesome. The electric strip heat of an air-to-air heat pump would fall under this restriction. However, it should be noted that, since the thermal energy in this HP-ICES is supplied in a circulating water medium whose temperature will not fall below the (water source) heat pump balance point, the use of auxiliary electric strip heat is not anticipated.

Separate performance codes for heat pumps based on the ASHRAE 90-75 performance standard (American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc.) are now being developed by the model code organizations. However, since the codes are being developed from standards already generally accepted in the industry, these new provisions, whose requirements are presented below, are not seen as inhibiting factors.

MINIMUM C.O.P. REQUIREMENT

OUTDOOR TEMP. °F	BY 1/1/77	BY 1/1/80
47 Dry Bulb, 43 Wet Bulb	2.2	2.5
17 Dry Bulb, 15 Wet Bulb	1.2	1.5

A last major question in the community acceptance of an HP-ICES is whether the community, as an entity, will accept the idea of an additional municipal utility, to be billed in the same manner as electricity, water usage or natural gas. The growing acceptance of the public to heat pumps, plus the growing concern for energy conservation seem to indicate a receptiveness on the part of the community. However, the ultimate acceptance will depend upon the understanding of the benefits of an HP-ICES: its improved performance in the task of space conditioning (as compared to the more conventional end-user systems) and the resulting cost-effectiveness of such a system. Some form of community education may be necessary to facilitate community acceptance on this basis.

2.2.5 RELIABILITY

Overall system reliability will depend upon two factors: reliability of the energy supply and its associated conversion/delivery equipment, and the reliability of the Central Station heat pump equipment. In the residential sector, a reliability bonus is offered with this system in that the end-user will utilize a water source heat pump.

An additional consideration in specifying an energy source for a particular site of application is the degree of reliability that could be expected from each available source and its associated equipment. One principal reason, aside from availability, that solar energy or waste heat recovery boilers are not utilized as primary energy sources is the interruptable nature of the energy supply. A solar supply has the obvious disadvantage of overcast skies. A waste heat recovery boiler has a two-fold disadvantage in its dependence on a supply of prepared fuels such as trash or wood scraps (difficulties at the processing station or in transportation would effectively shut down the boiler station), in addition to the need for periodic maintenance (as per insurance regulations and various pressurized boiler codes) and the potential for forced outages. Wastewater effluent, on the other hand, is a continuous process flow that should provide a fairly constant energy supply. Flow quantities through the sewage treatment plant are cyclic, on daily and weekly patterns, but most treatment plants have a recirculation provision for the minimum required flow through the plant that may be utilized as a second tie-in point on the energy source. Some allowance can be made for down time on the energy supply subsystem in sizing the thermal storage capacity in the system. However, engineering judgement dictates that the least complicated energy supply subsystem be specified to improve system reliability and to avoid compounding problems in other segments of the system.

A second major consideration in optimizing system reliability is the expected reliability of the equipment to be utilized at the Central Station. All major brands of heavy HVAC equipment reflect state-of-the-art design and component selection for performance <u>and</u> reliability, and as such, could be expected to run for extended periods with little or no maintenance. However, in looking ahead to the inevitable maintenance outage, there are provisions in the design of the Central Station that will minimize the consequences of an equipment outage and thus enhance the system reliability.

All major manufacturers offer guidelines for selecting and installing multiple units. Generally, this type application involves splitting the load of the station between two pumps (or more), each sized at approximately 60% (proportionately less if more units are utilized) of the estimated load. Depending on site-specific conditions and requirements such as temperature supply and required temperature of the distribution medium, the units may be connected in a parallel or series piping configuration. Such a design would lessen the load on both units under all but peak conditions and thus extend the life of the equipment. Necessary crossover and bypass piping at the station would also enable each unit to serve as a back-up to the other in the event of a forced or maintenance outage.

System reliability will be judged in the end-user segment not only by the reliability of the thermal service provided by the Central Station but also by the reliability of the equipment required to utilize it. This HP - ICES offers the advantages of utilizing water source heat pumps at the end-user. Water temperature is expected to range from 50° to 90°F, whereas air temperature may range from -10° to 95°F. The water-to-air systems eliminate the need to operate at low heat source temperature conditions, while the air-to-air units operating at low ambient temperatures require a high compressor discharge gas temperature because of the high pressure ratio across which the refrigerant must be pumped. The effect is to place a stress on the air-to-air system compressor and reduce its reliability. Thus, it is believed that the reliability of a water-to-air heat pump unit would be greater than that of an air-to-air unit of similar compressor design.[7]

2.2.6 UNIQUE APPLICATION CONDITIONS

It is not expected that this HP -ICES scheme will be overly dependent on any unique application conditions. With the wastewater effluent from a community providing the thermal energy input to the system, there will not be a dependence on an energy source that may be particular to a specific region or community configuration. Virtually any community with a thermal load demand and population density high enough to make an HP-ICES cost effective will have a sewage treatment facility that could be utilized in this scheme. In areas where additional resources are available, the wastewater thermal source may be augmented with one or several alternative energy sources discussed in Chapter 7 - IDENTIFIED VARIATIONS. Industrial installations within the community that discharge waste heat (such as a power plant) may dictate that the Central Station be located near that particular facility, or that a modular Central Station/distribution system (one with several thermal energy supplies located in different areas of the community) be adapted to this scheme. Generally, the universal availability of wastewater as the primary energy source will allow a broad scope of potential applications and the availability of additional resources will enhance the effectiveness of such a system.

2.2.7 SUMMARY

The following generalizations are offered in summary of the foregoing discussions of criteria that will determine the applicability of an ICES. A community is broadly defined as "a complex of facilities (and open spaces) that are employed in human activities and that are connected by networks for moving people, messages, goods, and services in the residential, commercial, industrial, agricultural, recreational, or institutional sectors or combination of sectors" (from Request for Proposal 78-4327).[45] Thus any number of community configurations may be considered a candidate for some form of ICES. The common denominator requirement is that the candidate community show a thermal load demand and a population (user) density high enough to make a central facility for heat reclaim and its associated distribution system cost-effective. Waste heat in the community's wastewater effluent is expected to be the most universally available thermal energy source. However, the primary energy supply may be augmented with one or several other energy supply options, depending on their availability. Climate will affect significantly the heating/cooling load demanded from an ICES, and thus system sizing, although it will not significantly affect the general applicability of such a system. Economic conditions are favorable for the implementation of an HP-ICES if the system proves cost-competitive with the options now available in residential and commercial HVAC. State and local building codes should not prove to be overly restrictive to such an HP-ICES. However, some form of community education may be necessary to facilitate acceptance of an ICES. And finally, reliability will be a major consideration in the specification and design of the Central Station and its associated equipment. With this general categorization in mind, it is obvious that a broad range of communities will qualify for consideration for an HP-ICES based on the HP-WHR scheme.

2.3 PROJECTIONS

In order to assess the desirability of promoting the HP-ICES concept, it is necessary to quantify the applicability of any particular scheme by developing estimates of the possible points of application and of the energy savings that would result from implementation of such a scheme in these communities. Projections for evaluating the applicability of the HP-WHR scheme will be developed by examining the potential for thermal energy supply (from wastewater effluent) and the coincident community space conditioning load that could be handled by an HP-WHR system.

2.3.1 THERMAL ENERGY SUPPLY

From the discussion in previous sections, it can be seen that the potential for application of this HP-WHR scheme in an Integrated Community Energy System (ICES) will be based on the availability of community wastewater effluent as the primary thermal energy source. The availability of this supply will, in turn, be based upon the population that is serviced by wastewater collection systems. The total flow of wastewater available as a thermal energy supply is defined by the following equation:

Available Flow = [(Population at year T) x(% Sewered Population) x(Per capita water consumption)] + Industrial Discharge (2-1)

With the variables explained below.

Population - ultimate population of service area at the target date, year T. This projection will be made on a nationwide basis and thus will utilize national projections for total population.

Percent Sewered Population - the portion of the total population that is served by wastewater collection systems and attending sewage treatment facilities which could be considered for retrofit of an HP-WHR system. A maximum percentage figure of 70% (.70) of the total U.S. population connected to sewage systems is taken from the Statistical Summary of the 1968 Municipal Waste Facility Inventory.[41] The ultimate figure used in the estimate will be somewhat lower, as explained in later discussion.

Per Capita Water Consumption - total water use (and discharge) is estimated using the typical figure for average consumption of 100 gallons per capita per day (gpcd).[3] Variations in flow range from 50% to 170% of the average on a daily cycle. Thus, the actual instantaneous flow available for heat transfer may vary above or below the average (in gpm - prorated from the average daily flow figure). However, it is expected that, over a 24-hour cycle, and accounting for the averaging effect of the thermal storage capacity incorporated into the system, the average figure of 100 gpcd will be available as the thermal energy supply.

Industrial Discharge - water discharged (in gallons per day average) from industrial processes into the wastewater collection system. This figure will vary with each site of application and thus will be used only when developing a site-specific estimate of available wastewater flow.

With the total flow of wastewater available as the thermal energy source defined, the total amount of heat available from the thermal supply can be calculated from the following equation:

$$Q = WC\ (\Delta T) \qquad (2\text{-}2)$$

Where

Q = total heat, Btu
W = flow of wastewater, lbs.
C = specific heat of water (assumed to be 1.0 Btu/lbs.-°F)
ΔT = change in temperature in supply medium, °F (essentially this is the heat transferred out of the wastewater flow). A ΔT of 10°F, widely used in HVAC system design, will be assumed as the temperature drop across the central station equipment heat exchanger.

The two equations can be combined to define the total thermal energy available, Q (in Btu's), from wastewater effluent as a function of total population. Thus:

$$Q = \{[(\text{Population at year T}) \times (\%\ \text{Sewered Population}) \times (\text{Per Capita Water Consumption})] + \text{Industrial Discharge}\ (\frac{\text{gal}}{\text{day}})\} \times \{8.33\ \frac{\text{lbs.}}{\text{gal}}\}\ \{1.0\ \frac{\text{Btu}}{\text{lb-°F}}\}\ \{\Delta T\ (°F)\}\quad \frac{\text{Btu}}{\text{day}}$$

Which reduces to:

$$Q = (\text{Population at year T}) \times (\%\ \text{Sewered Population}) \times (100) \times (8.33) \times (1.0) \times (10) \quad \frac{\text{Btu}}{\text{day}}$$

$$= 8330\ (\text{Population})\ (\%\ \text{Sewered Population}) \quad \frac{\text{Btu}}{\text{day}}$$

$$Q = 3.040 \times 10^6\ (\text{Population})\ (\%\ \text{Sewered Population}) \quad \frac{\text{Btu}}{\text{year}} \qquad (2\text{-}3)$$

Note that Industrial Discharge is deleted from equation 2-3 and will be used only in site-specific analyses.

2.3.2 THERMAL ENERGY DEMAND

With the quantities of thermal energy available from wastewater defined, some consideration must be given to the balance between energy supply and demand in order to determine the number and size of communities which may be considered for a HP-WHR system application.

From data developed in the subsequent performance and economic analyses (and discussed briefly in section 2.2.2 - CLIMATE), a central station installation of 1000 ton capacity would require 3. 0 MGD wastewater flow to supply the heating requirements of 470 homes in a New England clime with approximately 6820 degree-days per heating season. (Note: System performance in a New England clime is used as the basis for this evaluation because performance during the heating season is seen as the limiting criterion in optimizing the balance between thermal energy supply and demand on the system. Thus, defining flow quantities on the basis of satisfactory performance in a cold climate will throw the ultimate estimate of the number of communities to be considered for such a system to the conservative side.) Prorating the flow requirements to each residence gives a minimum flow per home of 6383 gallons per home per day $(\frac{\text{gals}}{\text{home-day}})$.

From a technical standpoint it would be possible to size, design, and install an HP-WHR system for perhaps as few as 5 homes, although the chances for economic viability of such a small system would be very slim. Therefore, to facilitate system design and construction and to accelerate the economic recovery of the system capital costs, the smallest system to consider would cover an approximate 100 home service area. The wastewater flow required for this service area would then be 638,300 gallons per day. This figure, divided by the 100 gallon per capita per day average water consumption figure, gives the population of the theoretic minimum size community that could be considered as a candidate for an HP-WHR system: 6383 persons. With allowances for minor variations in heating loads, service areas, and industrial discharges, and to facilitate the derivation of community and population estimates from available statistics, communities with populations of 5000 persons or more will be considered candidates for an HP-WHR system. It should be noted here that an ICES (and in this case an HP-WHR system) can be applied to a total community or just to portions of it. The HP-WHR concept depends upon the idea of reclaiming waste heat from the total flow of community wastewater and distributing it into certain viable service areas, not necessarily the full community. Thus, an HP-WHR heating system serving a small area integrated with a wastewater collection system serving a larger area does not contradict the ICES concept. However, defining a miminum population limit for community size is necessary when considering the balance between thermal energy supply and demand on the minimum acceptable size system. With the considerations of thermal energy supply outlined in the preceeding section and the above mentioned guidelines on community size, estimates of the number of potential sites of application and the potential energy savings can be developed.

2.3.3 ESTIMATES

Estimates for points of application and for energy conservation to be made on a nationwide basis will necessarily be based on a set of assumptions. The following discussions detail the assumptions pertinent to the estimates to be developed in this section:

- A community of 5000 persons or more would discharge enough wastewater (and thus enough thermal energy) to service the minimum acceptable system size of 100 homes. As community population increases above 5000, a proportionate increase in thermal energy becomes available through the community wastewater, and the potential for expanding system service area increases correspondingly. Thus the additional thermal energy made available by increased populations will be utilized at some point in the HP-WHR system.
- The statistical breakdown of community sizes and attending sewage treatment facilities from the 1968 Municipal Waste Facilities Inventory[41] (reproduced in Appendix A as TABLES A-4 and A-5) reveals that 59.53% of the total U.S. population resides in communities of 5000 or more <u>and</u> are connected to a municipal wastewater collection system with attending treatment plant. Deriving an estimate for the most optimistic market penetration, barring any acceptance barriers except technological, dictates that the full potential for wastewater heat reclaim be realized. Therefore, the full 59.53% (.5953) of the total population, who reside in communities of sufficient size to be considered as candidates for an HP-WHR system, will be used as the percent sewered population figure in developing the estimate.

- From the statistical breakdown in TABLE A-4, the number of potential sites of application (i.e. communities with populations of over 5000) is 3795. Extremely large communities (over 100,000) are included in this number because it is expected that, in these communities, wastewater collection and treatment will be segregated among several treatment facilities, thus providing the opportunity to install either segregated systems or modular central station facilities. The number of acceptable sites is not expected to change during the projection period due to the fact that new or planned communities would necessarily have to undergo several years of growth before arriving at a stage of development when an HP-WHR system would become a viable option. These communities should not be dropped from consideration, however, for such a system since new community developments would provide an ideal site of application for an ICES, designed from scratch and optimized for the particular community configuration. They simply cannot be included in this projection because of insufficient data.
- The population increase within each potential site of application will be assumed to be directly proportional to the national percentage increase in population during the projection period. Existing facilities or facilities now under construction will be assumed to supply sufficient sewage treatment capacity to handle the proportionate population increase during the projection period.
- It should be noted that the number of satisfactory sites and the estimate of percent sewered population is conservative due

to the fact that they are based on somewhat outdated (1978) data.

Bearing in mind the above assumptions and utilizing the thermal energy supply approximation shown as equation 2-3, the energy reclaimed from community wastewater, on a nationwide basis by community energy systems based on the HP-WHR concept, would be calculated thusly:

1980 U.S. Population - 221,848,000 (est.)

% Sewered Population - 59.53%

$$Q = 3.040 \times 10^6 \, (221{,}848{,}000) \, (.5953) \; \frac{\text{Btu}}{\text{year}}$$

$$= 401.48 \times 10^{12} \; \frac{\text{Btu}}{\text{year}}$$

Derived through this approximative method, TABLE 2-1 summarizes the amount of thermal energy potentially available for space conditioning needs, from a starting date of 1980 to the year 2000 (inclusive), extracted from community wastewater through HP-WHR systems installed at the 3795 potential sites of application.

The net energy savings attributable to utilization of the HP-WHR concept is directly proportional to the optimized system C.O.P., as shown in the following general equation:

$$Q_{net} = Q_{Total} - \frac{Q_{Total}}{C.O.P.} \tag{2-4}$$

Thus, the delivery of 9040.48×10^{12} Btu at an overall system C.O.P. of 2.245 (from Chapter 3 - EXPECTED PERFORMANCE, with a nominal wastewater temperature of 70°F) requires the energy expenditure of 4026.94×10^{12} Btu (or 1.179×10^{12} KWH). This expenditure leaves a net energy savings of 5013.55×10^{12} Btu, over the period 1980 - 2000.

Utilization of this amount of thermal energy form a previously wasted source also implies a savings in other forms of primary energy. The comparative savings are developed from the following assumptions (from Reference 10):

TABLE 2-1

HEAT EXTRACTED FROM WASTEWATER

YEAR	POPULATION (EST.)	Q/YR BTU x 10^{12}
1980	221,848,000	401.48
1981	223,661,000	404.76
1982	225,474,000	408.04
1983	227,287,000	411.32
1984	229,100,000	414.61
1985	230,913,000	417.89
1986	232,547,200	420.84
1987	234,181,400	423.80
1988	235,815,600	426.76
1989	237,449,800	429.72
1990	239,084,000	432.67
1991	240,385,400	435.03
1992	241,686,800	437.38
1993	242,988,200	439.74
1994	244,289,600	442.09
1995	245,591,000	444.45
1996	246,610,000	446.29
1997	247,629,000	448.14
1998	248,648,000	449.98
1999	249,667,000	451.83
2000	250,686,000	453.67
TOTAL		9040.49 x 10^{12} Btu

POPULATION ESTIMATES: "Population of the United States"[44]

Electric Energy: An average plant heat rate of 10,300 Btu/kwh is used to convert on-site consumption to resource energy (coal) consumption by a generating station. Transmission and distribution losses of 9% are assumed. High heating value (HHV) of 12,000 Btu/lb for coal is also assumed.

Fuel Oil: High heating value is assumed to be 139,000 Btu/gal; no transmission or distribution losses.

Natural Gas: High heating value of natural gas was assumed to be constant at 1,000 Btu/CFT; 8% transmission and distribution losses are also assumed.

Note: Sample calculations are contained in Appendix A.

Table 2-2 summarizes the net savings in alternate primary energy sources, based on the utilization of the thermal energy reclaimed by the HP-WHR system. Total primary energy source savings at the year 2000 represent 687 million (687×10^6) tons of coal, or 859 million (859×10^6) barrels of fuel oil, or 5.42 trillion (5.42×10^{12}) cubic feet of natural gas. Obviously, from this comparison, a vast potential for primary energy resource conservation can be realized by the development and implementation of an HP-ICES based on the HP-WHR concept.

TABLE 2-2

PRIMARY ENERGY SOURCE SAVINGS

PERIOD	NET THERMAL ENERGY SAVINGS	PRIMARY ENERGY SOURCE EQUIVALENCES		
		COAL	OIL	GAS
	Btu x 10^{12}	LB x 10^6	GAL x 10^6	CFT x 10^{12}
1980-85	1363.18	373,680	9807	1.472
1986-90	1183.33	324,379	8513	1.278
1991-95	1219.32	334,244	8772	1.317
1996-2000	1247.72	342,029	8976	1.348
TOTALS	5013.55 x 10^{12} Btu	1,374,332 x 10^6 LBS	36,068 x 10^6 GALS	5.415 x 10^{12} CFT

5.014 QUADS - WHICH IS

EQUIVALENT TO 687 x 10^6 LBS or 859 x 10^6 BARRELS or 5.42 TRILLION CFT

1.) NOTE: NET THERMAL ENERGY SAVINGS IS COMPUTED FROM EQN. 2-4

3.0 EXPECTED PERFORMANCE

3.1 INTRODUCTION

The scheme of HP-WHR system for a community is explained in the section on system description. For purposes of thermodynamic analysis, a schematic of the HP-WHR system that includes: one end-user heat pump, one central station heat pump at the treatment plant, a water storage reservoir, and distribution lines is shown in Figure 3-1. A similar system for the cooling mode is shown in Figure 3-2. It is possible that a number of central station heat pumps may be used depending on the application.

In the thermodynamic model for the heating mode, wastewater effluent at constant temperature from the treatment plant enters the central heat pump where its thermal energy (heat) is extracted and delivered to the distribution medium (water). The distribution water enters a thermal storage tank to provide a water supply at constant temperature to the end-user heat pumps. After delivering heat to the heat pumps, the water is returned to the central unit. The reverse process takes place for the cooling mode in which the energy is ultimately rejected to the effluent. Calculations of the energy requirements of the HP-WHR system using central heat pumps are presented in Section 3.4 Analysis of Results. The results of a distributed water-to-air heat pump system deleting central heat pumps are also presented. The performance results are compared with those of the conventional systems.

3.2 SELECTION OF CONDITIONS

In the selection of wastewater effluent temperatures for estimating the peak load and normal load performances of the HP-WHR system, efforts were made to correlate wastewater effluent temperature with the outside

HP-WHR SYSTEM
HEATING MODE

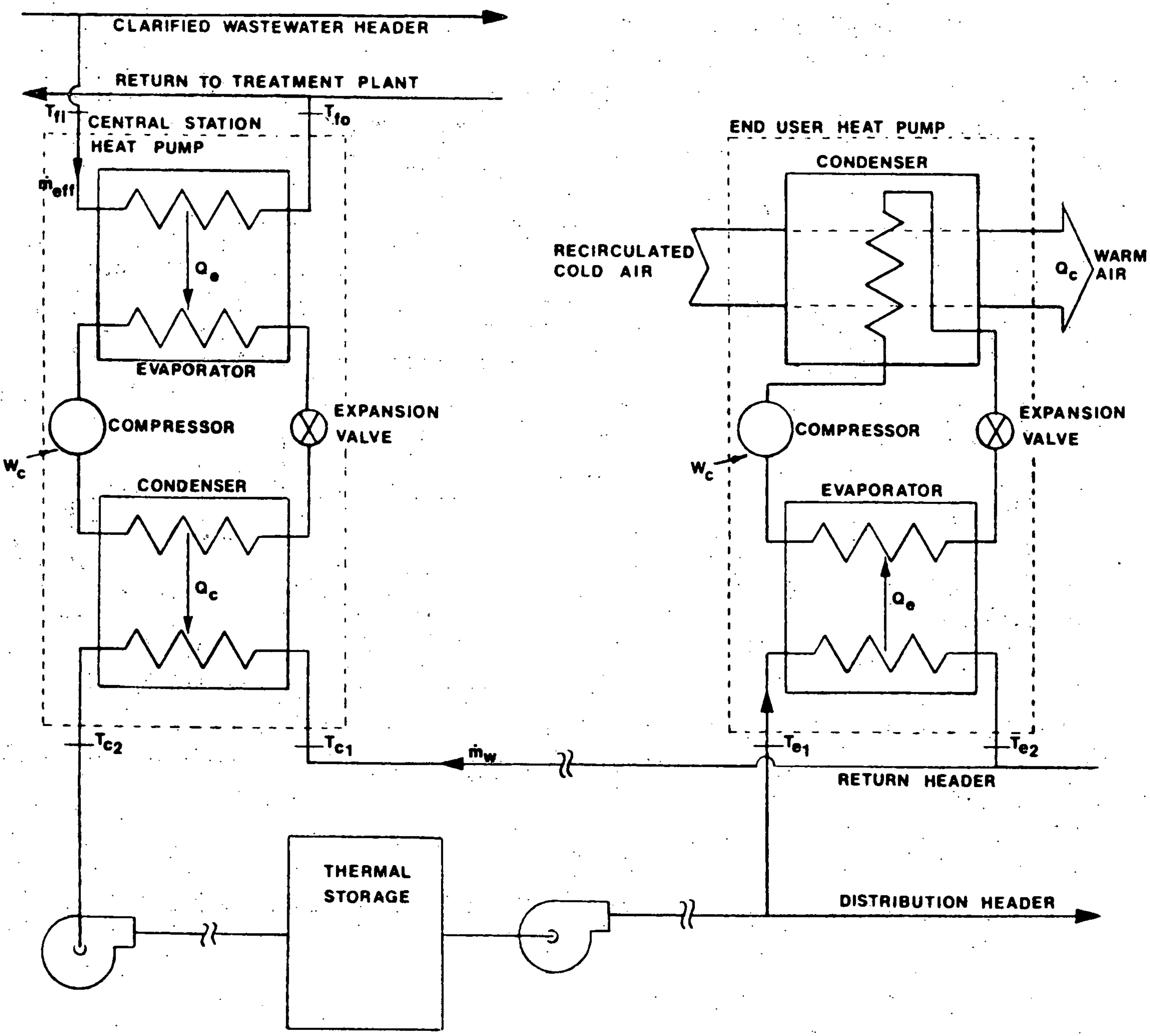


FIG. 3-1

HP-WHR SYSTEM
COOLING MODE

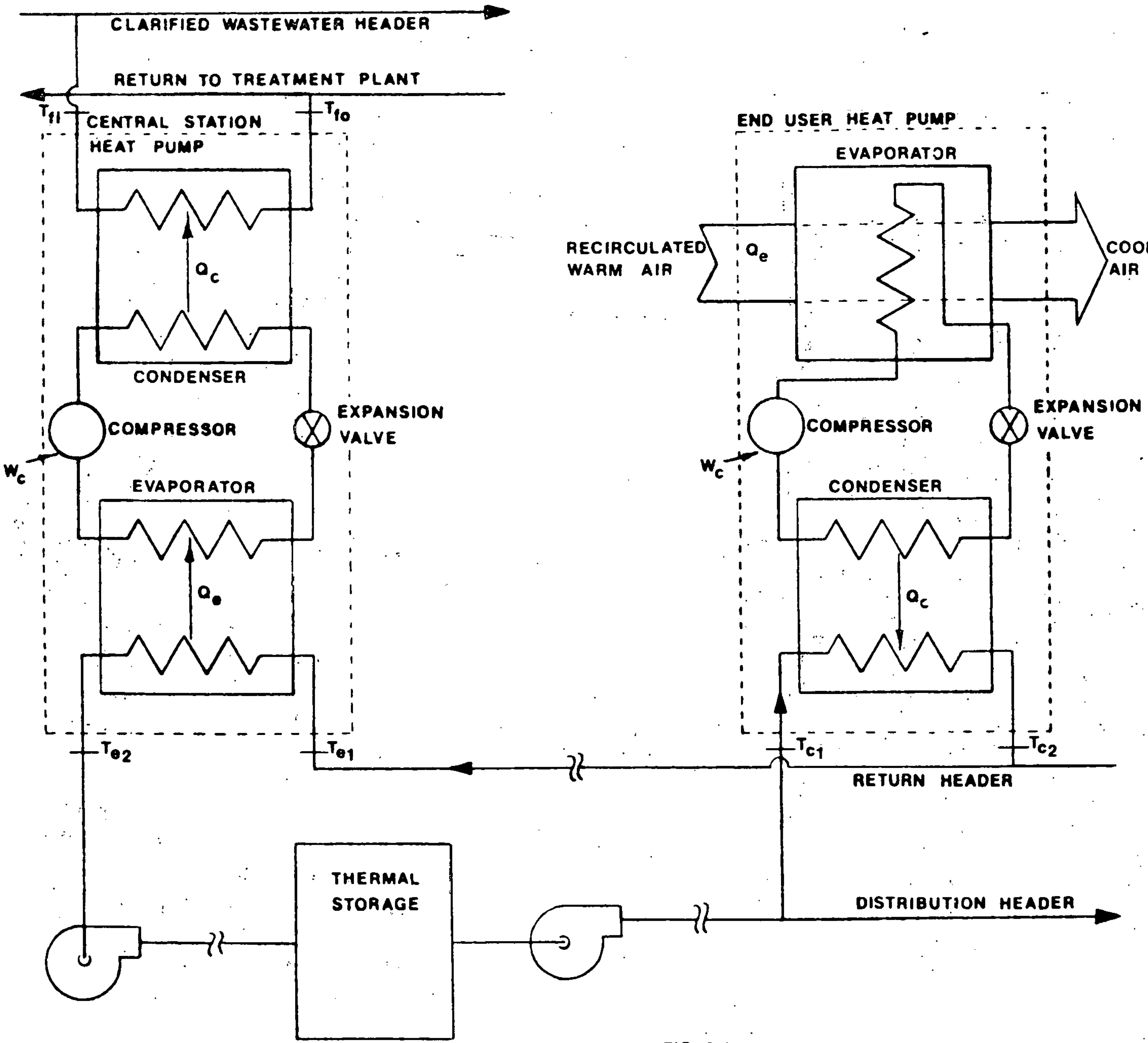


FIG. 3-2

air temperature. The wastewater effluent temperature depends on factors such as the time of day, ground temperature, outside air temperature, and the presence of thermal discharges from industrial processes. There is also the time-lag associated with the temperature change of the reservoirs or lakes as the water supply is drawn into community systems when seasonal changes in air temperatures occur. It is therefore believed that there can be wide variations in wastewater temperatures even within the same geographical region. Based on site-specific observations and study of various reports it is observed that the wastewater effluent temperatures (in parts of Canada) have ranged from lower 40's (°F) to upper 90's (°F).[20,22] One of the studies reports the average annual wastewater temperature at a plant in the North Chicago area to be between 50°F and 60°F.[20] It is also observed that wastewater effluents undergo temperature changes during the treatment processes.

In this study, effluent temperatures of 50°F, 60°F, 70°F, and 80°F have been chosen for analysis. It is believed that these temperatures are representative of a number of locations in the United States.

3.3 SYSTEM COP AND ENERGY CONSUMPTION

The coefficient of performance of the HP-WHR system (or air-to-air system) is defined as the ratio of the heating or cooling effect provided to the space at the end-user location divided by the energy equivalent of all energy inputs including compressors, pumps, and losses. This provides a measure for comparing the energy consumption of the HP-WHR system with that of conventional systems.

The system energy consumption in providing heating or cooling is:

$$E_{system} = \frac{1}{COP_{system}}$$

$$\text{Primary Fuel Energy Consumption (based on fossil fuels)} = \frac{E_{system}}{0.31} \quad (3\text{-}2)$$

Where

$$0.31 = \eta_{generation} \times \eta_{transmission} \times \eta_{distribution}$$

NOTE: The value of 0.31, which is the overall efficiency of conversion of primary fuel energy to useful electrical energy and transmission to application, is taken from Reference 4. This value is assumed for calculation, but is subject to variation.

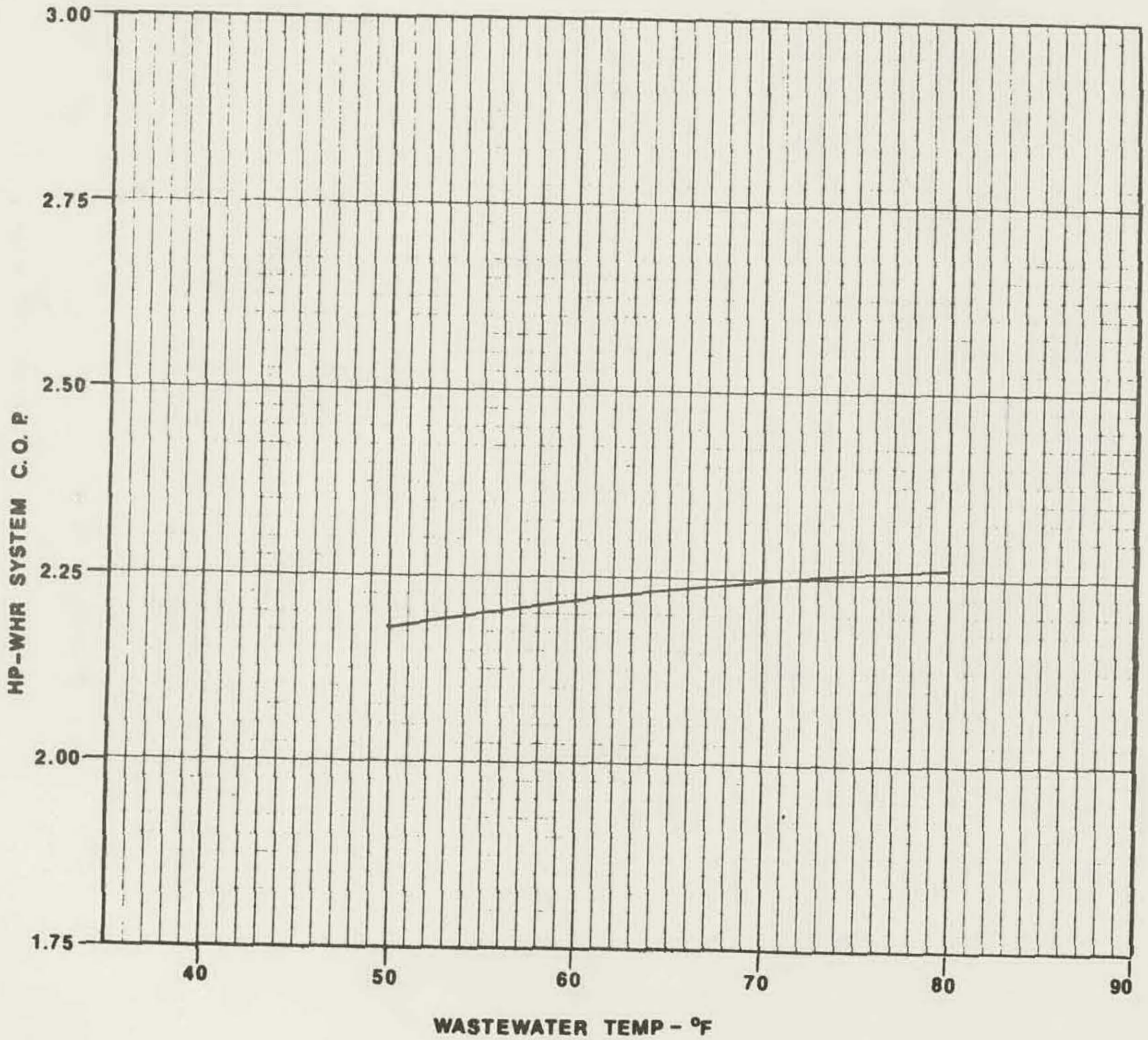
3.4 ANALYSIS OF RESULTS

3.4.1 HEATING MODE

Peak Load - The system COP values were calculated for the given design loads (peak) as a function of the temperature of the effluent as it enters the evaporator of the central heat pump. The results are presented as Figures 3-3 and 3-4 and summarized in Table 3-1. The temperature of water entering the end-user unit was held constant at 80°F. The Coefficient of Performance of the HP-WHR system increases by only 3.8% as the temperature of the effluent increases from 50°F to 80°F. This is to be expected because the COP of the end-user unit is held at constant value (relatively high) at the entering water temperature of 80°F, and the end-user unit consumes a greater share of the total energy input. Using equation 3-1, the energy required by the HP-WHR system to deliver 1 Btu of heating load is also listed in Table 3-1. These electrical energy values were translated to primary energy requirements by using equation 3-2 and are also shown in Table 3-1. For the purposes of comparison, similar data for an air-to-air heat pump is presented as Table 3-2.

Typical Heating Load - At heating loads lower than design heating

WASTEWATER TEMPERATURE vs HP-WHR SYSTEM C. O. P.

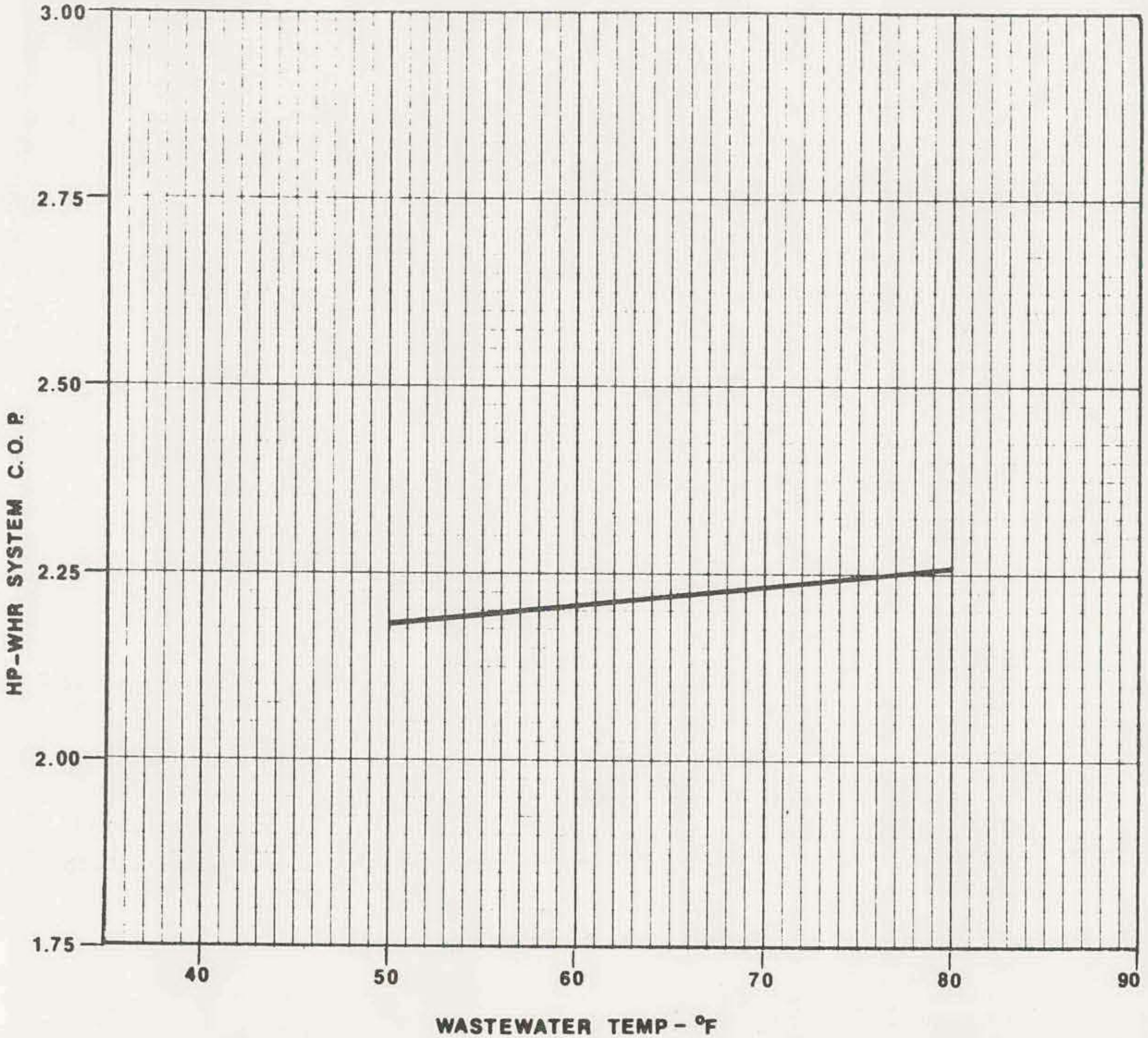


ENTERING WATER TEMPERATURE AT END USER UNIT 80°F

ENTERING WASTEWATER TEMP – °F	HP-WHR SYSTEM C.O.P.
50	2.181
60	2.219
70	2.245
80	2.264

FIG. 3-3

WASTEWATER TEMPERATURE vs HP-WHR SYSTEM C. O. P.



ENTERING WATER TEMPERATURE AT END USER UNIT 95 °F

ENTERING WASTEWATER TEMP – °F	HP-WHR SYSTEM C.O.P.
60	2.209
70	2.241
80	2.260

FIG. 3-4

TABLE 3-1

HP-WHR SYSTEM

Design Heating Load = 55000 Btuh
(Based on -20°F outside air temperature and 75°F inside air temperature)

Effluent Temperature °F	Temperature of Water Entering End-user-°F	COP System	E_{System} $\frac{Btu}{Btu}$	$E_{Primary\ Fuel\ Energy}$ $\frac{Btu}{Btu}$
50	80	2.181	0.4585	1.479
60	80	2.219	0.4506	1.454
70	80	2.249	0.4446	1.434
80	80	2.264	0.4417	1.425
		2.228 Avg.		
60	95	2.209	0.4527	1.460
70	95	2.241	0.4460	1.440
80	95	2.260	0.4425	1.427
		2.233 Avg.		

TABLE 3-2

AIR-TO-AIR HEAT PUMP

Nominal COP = 2.8 Heating Mode

Outside Air Temperature-°F	COP Heat Pump	$E_{Required}$ $\frac{Btu}{Btu}$	$E_{Primary}$ Fuel Energy
-20	0.672	1.488	4.800
0	1.652	0.605	1.953
10	1.932	0.5175	1.669
20	2.016	0.496	1.600
25	2.219	0.4506	1.453
47	2.753	0.363	1.170

NOTE: The COP values of air-to-air heat pumps were computed using the information given in <u>Unitary Air-to-Air Heat Pumps</u>.[6]

the equipment would be operating at part load resulting in cyclic operation. Cyclic operation results in a loss of efficiency. Information in the form of an equation for the part load performance of water-to-water heat pumps is available (Reference 5) but precise information on the part load performance of water-to-air heat pumps is not available. Information from Reference 10 on part load performance of air-to-air heat pumps indicates that, at heat loads of 15% of rated capacity, the cyclic operation of a heat pump can result in a decreased COP of approximately 30% of the full load, steady state value. At loads on the order of 80% heating, the decrease in COP is about 4 to 5% of its steady state value. These figures apply to air-to-air heat pumps and are based on the experimental work by Kelly & Bean[14], and Parken & Beausoliel.[23]

Cooling Mode - The results of the cooling mode analysis are presented in Table 3-3 for wastewater temperatures of 60°F and 80°F, and the COP values are used in the calculation of annual energy requirements.

3.4.2 SIMULTANEOUS HEATING AND COOLING

The need for simultaneous heating and cooling arises where in certain buildings of the community, the internal heat gains exceed the heat loss from the building. This may happen in the early periods of the Fall and Spring seasons. Since the heat rejected to the distribution water by the end-user heat pumps operating in cooling mode acts as a heat source for the heat pumps operating on the heating mode, the work input to the central heat pump will be reduced. The extent of the energy saved during this type of operation will depend on the relative distribution of heating and cooling loads. In the special case, when the total heat rejected

TABLE 3-3

HP-WHR SYSTEM PERFORMANCE COOLING MODE

Wastewater Temperature Entering Central Unit-°F	Water Temperature Entering End-User Unit-°F	COP System	E_{System} $\frac{Btu}{Btu}$	$E_{Primary\ Fuel\ Energy}$ $\frac{Btu}{Btu}$
60	40	1.913	0.523	1.686
80	40	1.874	0.534	1.721
80	50	1.843	0.543	1.750

by the end-user heat pumps operating in the cooling mode is absorbed by the heat pumps operating in the heating mode, the central heat pump will require no work input. Such a situation can be obtained by several combinations of heating and cooling loads, units with different COP's, etc. In the section on analysis, a model consisting of two end-user units is analyzed by means of an example. It is shown that, for two end-user units of identical COP values, the work input to the central unit is zero when the ratio of heating load/cooling load = HRF^2, where HRF is called the heat rejection factor defined in the section on analysis.

3.4.3 ANNUAL ENERGY CONSUMPTION

The annual energy consumption by the HP-WHR system and other conventional systems are calculated and the total energy required by residences in the Northwest Central region and the South Atlantic region are listed in the Table 3-4. In the calculation of the annual energy consumptions by the HP-WHR system and other conventional systems, the chapters on energy estimating methods of the ASHRAE Handbook and Product Directory, Systems Volumes 1973 and 1976[49] were consulted. The use of the bin method of calculation could not be used because data on part load performance of water-to-air heat pumps was not available. However, the dynamic efficiency loss factors were considered in accordance with the data derived from Reference 10.

3.4.4 OBSERVATIONS

The results of this study indicate that HP-WHR systems perform better than the air-to-air heat pumps in Northern locations, at outside air temperatures down to 25°F in the heating mode.

The Coefficient of Performance of the HP-WHR system varies with

TABLE 3-4

ANNUAL ENERGY CONSUMPTION*

SYSTEM	GEOGRAPHICAL REGION	HEATING SEASON MBTU/SEASON	COOLING SEASON MBTU/SEASON	E ANNUAL MBTU/YR	ANNUAL PRIMARY FUEL CONSUMPTION MBTU/YR
HP-WHR	NWC	32096	4398	36934	119142
	SA	17898	15306	32942	106264
AIR-TO-AIR HEAT PUMP	NWC	47210	3861	51071	164745
	SA	16478	14077	30555	98564
CONVENTIONAL FUEL GAS (WARM-AIR)	NWC	155717	12454	168171	168171
	SA	87068	45409	132477	132477
ELECTRIC	NWC	65244	3861	69105	222919
	SA	36481	14077	50558	163090

*The annual energy consumption listed in this table is based on design loads.

NWC - North West Central Region
SA - South Atlantic Region

effluent temperatures, but no to a great degree. As seen in Table 3-1 and Figure 3-1, the system COP increased by only 3.8% when the effluent temperature changed from 50°F to 80°F.

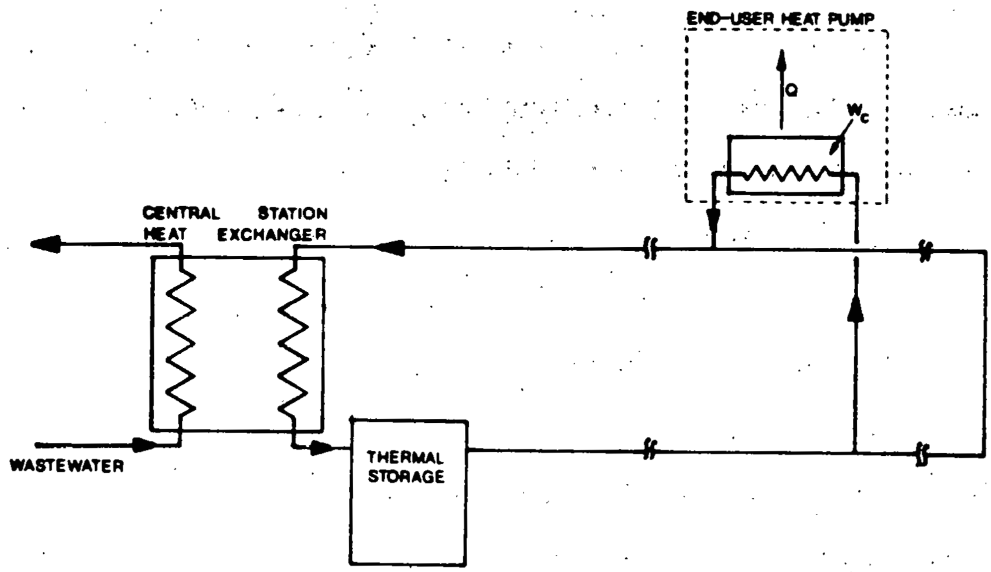
In the cooling mode, the air-to-air heat pump performs better than the HP-WHR system. Compared to a conventional heating system (gas warm-air furnace), the HP-WHR system consumes a greater amount of primary fuel both in the Northern and Southern locations.

Distributed water-to-air heat pumps consume less energy compared to the HP-WHR systems using central heat pumps. The results in Table 3-4 indicate that the system COP is high, and in general, the system performs better than conventional heating systems, such as air-to-air and gas furnace-warm air heating systems on the basis of primary fuel consumption. A variation of the HP-WHR scheme is discussed in the next section. As shown in the example in Section 9.4.6, the simultaneous heating and cooling mode of operation improves the performance of the HP-WHR system.

3.5 HP-WHR SYSTEMS USING ONLY DISTRIBUTED WATER-TO-AIR HEAT PUMPS

The use of distributed heat pumps (with no central heat pump) using heat from wastewater effluent as the source of energy is possible if the wastewater is at a sufficiently high temperature to allow for heat transfer to the distribution water by means of a heat exchanger. If such conditions are obtainable, the scheme would be as shown in Figure 3-5. This system COP is found to be higher than the HP-WHR system using central heat pumps. But at lower effluent temperatures, the size of the heat exchangers would have to be very large to allow for sufficient quantities of heat transfer, and the pumping losses resulting from increased flow rates of water would be high.

The COP system and energy requirements are listed in Table 3-5.



DISTRIBUTED WATER SOURCE HEAT PUMP SYSTEM

FIG. 3-5

TABLE 3-5

Effluent Temperature	Distribution System Supply Temperature	COP Distributed System	E Systems	E Primary Fuel Energy
80°F	60°F	2.779	0.360	1.16
70°F	50°F	2.562	0.390	1.258
65°F	45°F	2.528	0.395	1.274

3.6 SUMMARY

In summary, it is the belief of the authors that this report demonstrates the HP-WHR scheme as a potentially viable energy supply alternative for some specific applications. The actual determination of feasibility and expected performance are highly dependent on site-specific physical haracteristics and local economic conditions, and will be addressed in the Demonstration Community Application Study in Part II of this report.

4.0 EXPECTED ECONOMICS

4.1 INTRODUCTION

The consumer will ultimately pay the costs for either a conventional heating and cooling system or a Heat Pump - Wastewater Heat Recovery System. For that reason, any economic advantages the HP-WHR scheme may have to offer must be fully explored and documented.

This report section is concerned with the expected economic performance of the HP-WHR system and with conventional natural gas, air-to-air heat pump, and electric resistance heating systems. First costs, operating costs, and maintenance costs of the various systems will be estimated and used in a net present value comparison of the considered alternative.

4.2 SYSTEM DESIGN

It is demonstrated in the expected performance section that overall system efficiency is dependent on effluent source temperatures and end-user loads placed on the system. For the purpose of economic evaluation, two geographic locations were chosen as representatives of extremes in climatological differences encountered throughout the country. Effluent temperatures were assumed to average 60°F in the winter months and 80°F in the summer months.

4.2.1 END-USER LOADS

End-user heating and cooling loads were developed using the Rand Corporation's Energy Use and Conservation in the Residential Sector: A Regional Analysis , Report Number PB-254-468, June 1975. Data for the northern type climate was selected from the New England Area statistics and data for the southern type climate was selected from the West South Central Area. The size of a typical residence was taken as the national average of 1185 square feet. Table 4-1 is a summary of the data used.

Table 4-1

Typical End-User Heating and Cooling Load Data

CLIMATE	Annual HDD	Annual CDD	Approx. Peak Load MBtu/Hr Heating	Cooling	Annual Energy Requirements*(MBtu) Heating	Cooling
Northern	6820	1200	41	25	105,465	18,960
Southern	2350	2750	21	32	37,920	45,030

HDD = Heating Degree Days
CDD = Cooling Degree Days

*Figure 7, Reference 27

4.2.2 DESIGN ASSUMPTIONS

In performing the economic analysis, several assumptions about the configuration of the HP-WHR system were made in order to develop unitized costs. These assumptions are not, in any way, intended to illustrate an optimum system design or limit the thinking of a perspective system designer when applying this concept.

The assumptions used were as follows:

1. The heat extraction by the central heat pump is 12,000 MBtu/Hr for 20 hrs per day, 5 days per week. Total heat extraction per month of 5.2×10^9 Btu.

2. Average system heating C.O.P. of 2.2.

3. Number of end-users served is determined by dividing the monthly heating energy deliverable from the system by the typical end-user requirements in a January month.

4. Thermal storage capacity determined by the energy required by all end-users in 4 consecutive average heating days

during the month of January. Temperature pull-down of storage of 30°F.

5. Variable flow pumping, resulting in a constant percentage of pumping energy to system delivered energy.

6. Negligible thermal losses.

Using these assumptions, the following system configurations were used.

<u>Northern Climate</u>	
Central Heat Pump Extraction Rate	5.2×10^9 Btu/Mo.
Circulation Pump Horsepower	75 hp
Thermal Storage Vessel	5×10^6 gal
Distribution System Length (NIC run-outs to end-users)	61,100 ft
Longest Piping Run (Supply & Return)	9,900 ft
Number of End-Users	470
Distribution Pump Horsepower	450 hp
<u>Southern Climate</u>	
Central Heat Pump Extraction Rate	5.2×10^9 Btu/Mo.
Circulation Pump Horsepower	75 hp
Thermal Storage Vessel	5×10^6 gal
Distribution System Length (NIC run-outs to end-users)	127,400 ft
Longest Piping Run (Supply & Return)	13,950 ft
Number of End-Users	980
Distribution Pump Horsepower	625 hp

4.3 FIRST COSTS

First costs for the various system components were developed with the aid of R.S. Means, "Building Construction Cost Data 1978."[9] Unitary equipment costs were taken from the ICES technology evaluation reports; ANL/CES/TE 77-9, "Unitary Water to Air Heat Pumps" and ANL/CES/TE 77-10, "Unitary Air-to-Air Heat Pumps". Supplemental information was also obtained from local equipment distributors.

4.3.1 HEAT PUMP-WASTEWATER HEAT RECOVERY SYSTEM

The HP-WHR system cost estimate is broken into three sections: Central Plant, Distribution System, and End-User Systems. An approximation for each system was made on a unitized basis assuming a Central Heat Pump heat extraction rate of 12,000,000 Btu/hr and end-user heat pump systems with an installed heating capacity of 46.5 MBtu/hr and cooling capacity of 40 MBtu/hr at ARI standard 240-75 conditions. The distribution system was assumed to serve customers at an average piping length of 130 ft per customer, and an additional 150' ft of run-out piping from the distribution system to the end-user.

The following costs were developed for the Central Plant and the northern and southern climate distribution system schemes:

CENTRAL PLANT

Item	Material & Installation
Central Heat Pump	$144,200
Pump for Circulation	30,000
Interconnecting Piping and Valves	120,000
Control Systems	18,000
Interface Modifications with Sewage Treatment Plant	80,000
Building to House Components (3,000 sq. ft.x $20 sq. ft)	60,000
SUBTOTAL	$452,200
25% Contractor & Engineering Fees	113,050
TOTAL	$565,250

Northern Climate

Distribution System

	Material & Installation
Central Storage Vessel 5,000,000 gal x $.01 per gal =	$ 50,000
Variable Speed - Distribution System Pump	70,000
Distribution System Piping 3,050 ft (10" pipe) @ 12.80/ft 6,100 ft (8" pipe) @ 10.10/ft 6,100 ft (6" pipe) @ 7.45/ft 33,500 ft (4" pipe) @ 5.65/ft 12,200 ft (2" pipe) @ 3.15/ft	374,085
SUBTOTAL	$494,085
25% Contractor & Engineering Fees	123,520
TOTAL	$617,605

Southern Climate

Distribution System

	Material & Installation
Central Storage Vessel 5,000,000 gallons @ $.01 per gal	$ 50,000
Variable Speed - Distribution System Pump	80,000
Distribution System Piping 6,370 ft (10" pipe) @ 12.80/ft 12,740 ft (8" pipe) @ 10.10/ft 12,740 ft (6" pipe) @ 7.45/ft 70,070 ft (4" pipe) @ 5.65/ft 25,480 ft (2" pipe) @ 3.15/ft	781,280
SUBTOTAL	$ 911,280
25% Contractor & Engineering Fees	227,820
TOTAL	$1,139,100

Unit Costs Associated with Each End-User:

Water Metering Equipment	$200
150'-3/4" Run-Out Piping @ $1.5 per ft	225
TOTAL	$425

4.3.2 END-USER SYSTEM COST

The installed cost of an end-user "water-to-air" heat pump system was taken to be $2,000. Since all conventional systems comparisons will assume a forced air distribution system, the installed costs presented herein do not make any allowance for ductwork, insulation, grilles, and associated installation. These costs also refer to new installation and are not necessarily representative of retrofit conditions.

4.3.3 CONVENTIONAL SYSTEM COSTS

Only end-user costs were developed for conventional systems since accounting for energy-supply equipment costs and distribution system costs are included in the energy charges paid by the consumer. The equipment costs for each of the conventional alternatives is as follows:

Gas Heat/3 Ton Elect. Cooling:	$2,200
Air-to-Air Heat Pump:	$2,900
Resistance Heating/Electric Cooling:	$2,250

4.4 OPERATING COSTS

Operating costs were developed from the efficiency analyses presented in Section 3 of this report. It was assumed that electrical power for central plant equipment is purchased from a utility at a reduced cost per kwh due to the tiered rate schedule applied to industrial/commercial type customers. Residential electric power would be purchased on an individual basis. Where combined energy costs are given for the HP-WHR scheme, it is assumed that residential electric energy cost per kwh is 1.5 times the central energy plant cost. The cost of thermal energy from the HP-WHR scheme to the end-user is discussed in a subsequent section.

4.4.1 CENTRAL STATION AND DISTRIBUTION SYSTEM OPERATING COSTS

The input energy for delivery of heating energy and dissipation of cooling energy by the central plant were developed from the performance analyses of Section 3. Appendix B gives a complete breakdown of how energy quantities were derived.

A summary of the electrical energy required for each of the central plant components is presented in Table 4-2.

Table 4-2

CENTRAL PLANT

Electrical Energy Requirements

Northern Climate		kwh/yr
Central Heat Pump		
Heating		1,392,905
Cooling		632,485
Circulating Pump		224,282
Distribution Pump		1,027,858
Auxiliaries		12,000
	TOTAL	3,289,530
Southern Climate		
Central Heat Pump		
Heating		1,043,600
Cooling		3,131,060
Circulating Pump		594,990
Distribution Pump		1,190,615
Auxiliaries		12,000
	TOTAL	5,972,265

Figure 4-1 presents the annual electricity costs for the central plant under different average energy costs ranging from 2¢ to 4¢ per kwh.

4.4.2 END-USER OPERATING COSTS

The typical end-user is assumed to have the heating and cooling load requirements presented in Table 4-1. Based of those requirements and seasonal coefficients of performance of 2.75 for heating and 2.9 for cooling, the following end-user electric energy requirements were determined:

Table 4-3

HP-WHR System
End-User Electric Energy Requirements

	Kwh/Yr.	Cost/Yr.
Northern Climate		
Cooling	1,915	$115
Heating	11,236	674
TOTAL	13,151	$789
Southern Climate		
Cooling	4,549	$273
Heating	4,040	242
TOTAL	8,589	$515

The cost per unit for electrical energy was estimated at 6¢ per kwh.

In addition to the electric energy purchased by the end-user, an additional charge would be made for the thermal utility service from the central plant. These charges are explained in Section 4.6, and for the examples given, would be as follows:

Northern Climate - $590 annually

Southern Climate - $469 annually

ANNUAL HP-WHR SCHEME CENTRAL PLANT COSTS

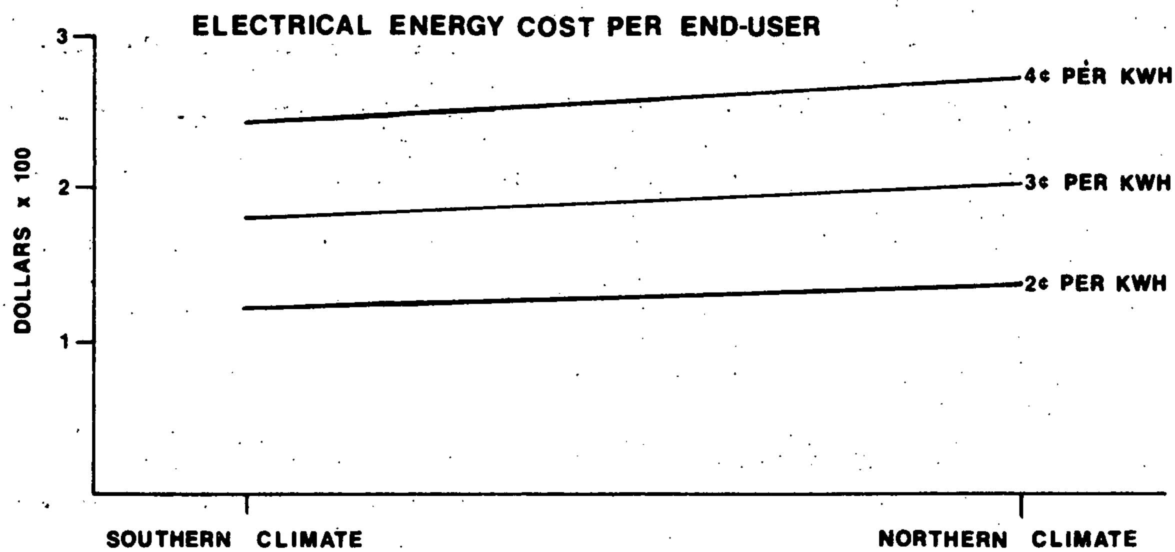


FIG. 4-1

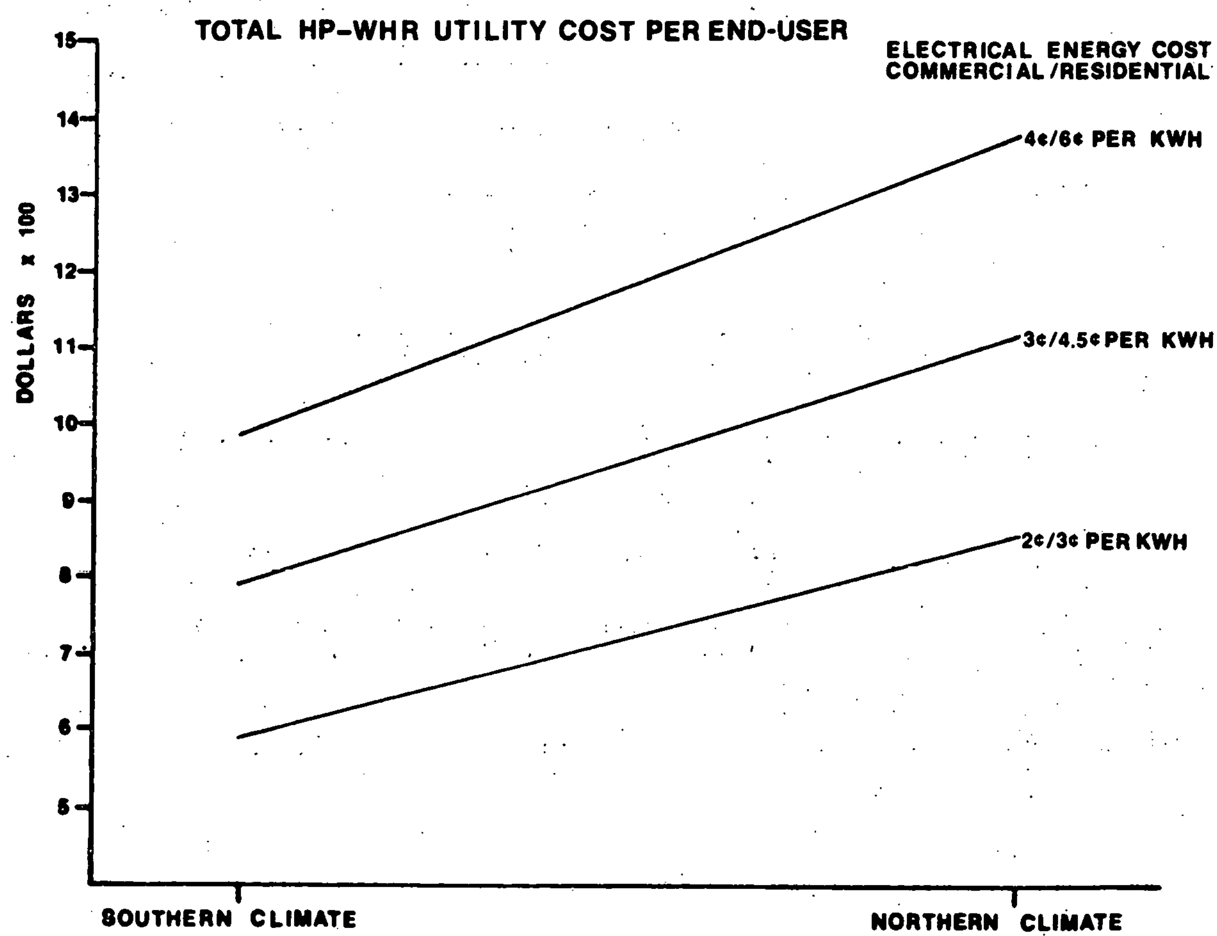


FIG. 4-2

4.4.3 TOTAL HP-WHR SYSTEM OPERATING COSTS

Figure 4-2 is a graph of total system energy costs to the end-user at several electric energy rates. This graph was developed by adding the end-user's cost of electric and thermal energy. Comparisons with conventional systems presented in Figure 4-3 are made at an electric energy rate of 6¢ per kwh.

4.4.4 CONVENTIONAL SYSTEM OPERATING COSTS

Based on the load assumption stated in the previous discussion on end-user heating and cooling requirements, operating costs were estimated for each conventional system. Direct fired heating was assumed to be 65 percent efficient, Air-to-Air heat pump seasonal performance factor was assumed to be 1.5 for northern climate and 2.3 for southern climates, and electric resistance heating was assumed to have a COP of 1.0 All systems for air conditioning were assumed to have energy efficiency ratios of 7.5. The results are presented in Table 4-4.

Table 4-4
Conventional System Energy Costs
Northern Climate

System	HEATING Energy Consumption	HEATING Energy Cost	COOLING Energy Consumption	COOLING Energy Cost
NATGAS/Elect Cooling	1622.5 ths	$ 486	2528 kwh	$152
Elect Air-to-Air hp	20,600 kwh	$1,236	2528 kwh	$152
Resistance Heat/Elect A.C.	30,900	$1,854	2528 kwh	$152

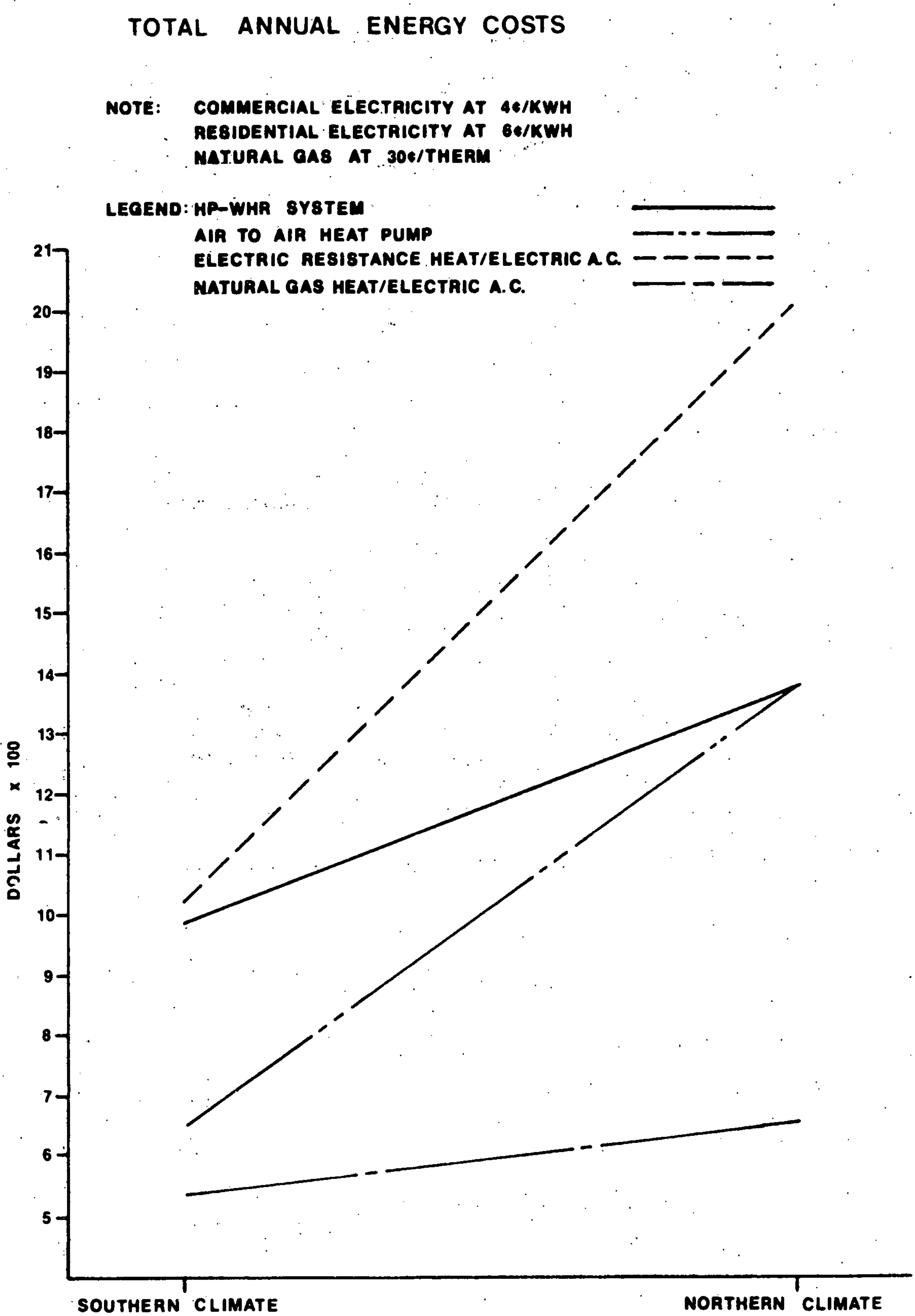

TOTAL ANNUAL ENERGY COSTS
NOTE: COMMERCIAL ELECTRICITY AT 4¢/KWH
RESIDENTIAL ELECTRICITY AT 6¢/KWH
NATURAL GAS AT 30¢/THERM
LEGEND: HP-WHR SYSTEM
AIR TO AIR HEAT PUMP
ELECTRIC RESISTANCE HEAT/ELECTRIC A.C.
NATURAL GAS HEAT/ELECTRIC A.C.
DOLLARS x 100
21
20
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
SOUTHERN CLIMATE
NORTHERN CLIMATE

FIG. 4-3

Table 4-4 (cont)

Southern Climate

System	HEATING Energy Consumption	HEATING Energy Cost	COOLING Energy Consumption	COOLING Energy Cost
NATGAS/Elect Cooling	584 th	$175	6,000 kwh	$360
Elect Air-to-Air hp	4830 kwh	$290	6,000 kwh	$360
Resistance Heat/Elect A.C.	11,110 kwh	$667	6,000 kwh	$360

The cost per unit for energy was estimated at 6¢/kwh for electricity and $.30 per therm for natural gas.

4.4.5 MAINTENANCE COSTS

Maintenance costs for unitary equipment were estimated with the aid of the ICES Technology Evaluation Documents.[6,7] Maintenance of the central heat pump equipment was estimated at 2 percent of installed costs per year for the central station equipment and 0.5 percent per year for the distribution system. The following maintenance costs were used:

HP-WHR SYSTEM Item	ANNUAL MAINTENANCE COST
Central Station Equipment	$11,305
Distribution System	No. $3,090 So. $5,695
Unitary H.P. (Water to Air)	$165 per unit
CONVENTIONAL SYSTEMS	
Gas/Electric Cooling	$130 per unit
Air-to-Air Heat Pump	$180 per unit
Resistance/Elect. Cooling	$160 per unit

Annual maintenance costs for the unitary end-user equipment was developed from references

4.5 HP-WHR SCHEME OPERATION AS A THERMAL UTILITY

In order to compare the HP-WHR scheme to conventional utility supplied schemes, it is necessary to perform an analysis from the end-user's viewpoint. To account for thermal energy distributed from the central plant, it is anticipated that the system will be operated as a thermal utility. The charges to end-users will be made based on meter readings and will be sufficient to cover the central plant's operating and maintenance costs, debt service, and administrative costs associated with operation of the system.

The cost figures contained in this report were generated for the northern and southern locations using the stated electric energy costs, an administrative cost of five percent of total owning and operating costs, and debt service based on an interest rate of 7 percent per year. It is recognized that the seven percent interest rate is somewhat arbitrary and will vary with the application and type of financing arrangements actually implemented; however, it was selected to be between typical 1977 yields for Municipal Class A bonds and Utility bonds issued by private companies.

The cost per customer for thermal service was computed as follows:

<u>Northern Climate (serving 470 customers)</u>

Energy Cost @ 4¢/kwh	$131,581
Debt Service	
Central Plant and Distribution System $1,182,855 x .0858	101,489
End-User Connections $199,750 x .0858	17,138

Northern Climate (Continued)

Maintenance on Central Plant & Dist. System	$ 14,395
SUBTOTAL	$264,603
5 percent administration	13,230
TOTAL	$277,833

Annual Cost Per Customer = $591

Southern Climate (serving 980 customers)

Energy Cost @ 4¢/kwh	$238,891
Debt Service Central Plant and Distribution System $1,704,350 x .0858	146,233
End-User Connections $416,500 x .0858	35,736
Maintenance on Central Plant & Dist. System	17,000
SUBTOTAL	$437,860
5 percent administration	21,893
TOTAL	$459,753

Annual Cost Per Customer = $469

4.6 LIFE CYCLE COSTS

A net present value life cycle cost analysis was prepared for the HP-WHR system and each of the conventional system alternatives. The analysis was performed from the end-user point of view, since the consumer will ultimately pay all costs associated with each system. In the case of thermal energy distributed by the HP-WHR system, central plant capital and energy costs are embedded in the cost of service to the end-user. A range of interest rates was used for discounting future expenditures. The results are presented in Figures 4-4 and 4-5.

PRESENT VALUE OF EXPENDITURES

NOTE: COMMERCIAL ELECTRICITY AT 4¢/KWH
RESIDENTIAL ELECTRICITY AT 6¢/KWH
NATURAL GAS AT 30¢/THERM

LEGEND: HP-WHR SYSTEM
AIR TO AIR HEAT PUMP
ELECTRIC RESISTANCE HEAT/ELECTRIC A.C.
NATURAL GAS HEAT/ELECTRIC A.C.

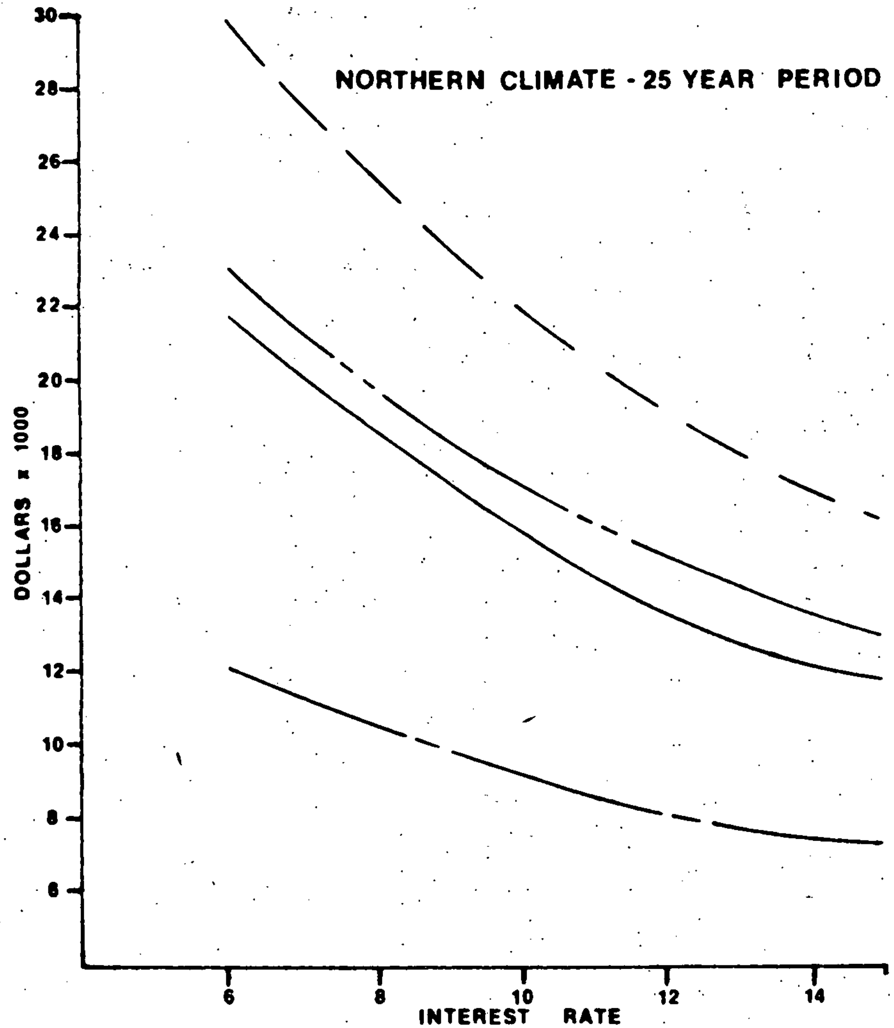


FIG. 4-4

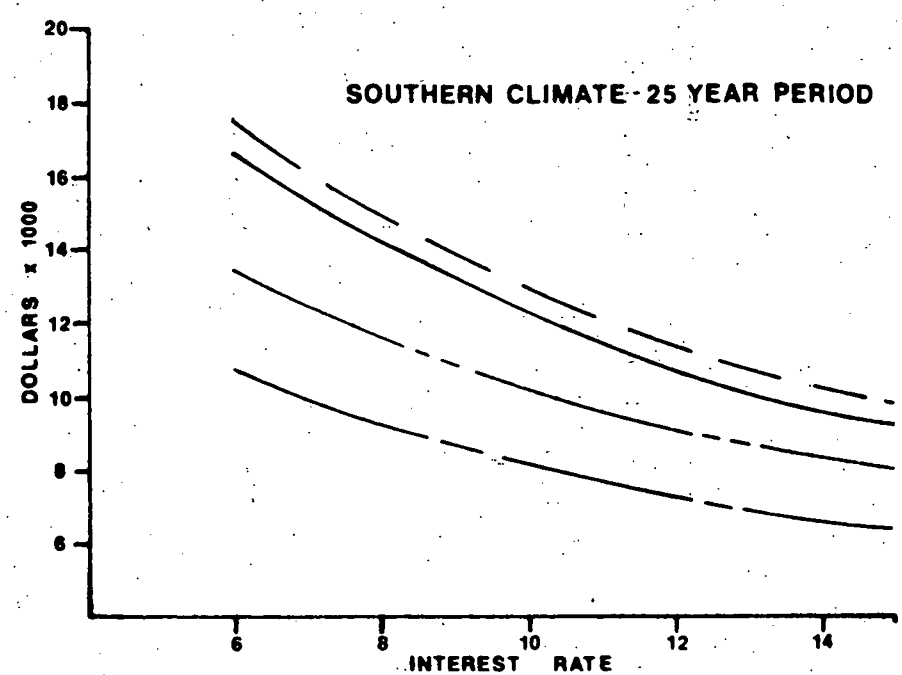


FIG. 4-5

4.6.1 DETERMINATION OF PRESENT VALUE

The present value of each alternative was determined for a twenty-five year period at the stated discount factor. The following general formula was used:

$$PV_i = FC_i + f(j,25) \times (AOC_i + AMC_i)$$

where

PV_i = Present value of the i^{th} alternative

FC_i = First cost of the i^{th} alternative

$f(j,25)$ = The present worth factor at j percent interest for 25 years

AOC_i = Annual operating costs of the i^{th} alternative

AMC_i = Annual maintenance costs of the i^{th} alternative

The present value figure is given as a positive number; however, in actuality it should be realized that the number is a present value of present and future costs.

Figures 4-4, 4-5 present a comparison of the HP-WHR system and several conventional systems at 8 percent, 10 percent, and 12 percent present worth factors.

4.7 SUMMARY OF EXPECTED ECONOMIC PERFORMANCE

The net present value analysis of the HP-WHR scheme and the conventional schemes indicates that the HP-WHR scheme can be cost competitive with air-to-air heat pump systems and electric resistance heating systems depending on comparative energy costs and geographic location. As expected, the conventional natural gas/electric scheme is much more cost effective at today's natural gas prices.

5.0 ENVIRONMENTAL IMPACT

5.1 INTRODUCTION

During the last 15 years there has been increasing concern within the United States for protection of all aspects of the environment. This concern has resulted in the passage of numerous Federal laws and the subsequent development of implementing regulations to protect various aspects of the environment. These laws have covered such diverse areas as air qualtiy, water quality, noise control, archaeological and historic preservation, and protection of endangered species of plants and wildlife. The most significant of the Federal laws and Executive Orders affecting the environment are listed below:

1. The Archaeological and Historic Preservation Act of 1974;
2. The Clean Air Act;
3. The Coastal Zone Management Act of 1972;
4. The Endangered Species Act of 1973;
5. The Federal Water Pollution Control Act, as amended;
6. The Fish and Wildlife Coordination Act of 1958;
7. The Flood Disaster Protection Act of 1973;
8. The Marine Protection Research and Sanctuaries Act of 1972;
9. The National Environmental Policy Act of 1969;
10. The National Historic Preservation Act of 1966; Executive Order 11593 ("Protection and Enhancement of Cultural Environment," May 13, 1971): and 36 CFR Part 800 ("Procedures for the Protection of Historic and Cultural Property," January 25, 1974);
11. The Rivers and Harbors Act of 1899;
12. The Safe Drinking Water Act of 1974;

13. The Solid Waste Disposal Act;
14. The Water Resources Planning Act of 1965;
15. The Wild and Scenic Rivers Act of 1968;
16. Executive Order 11296 ("Evaluation of Flood Hazards in Locating Federally Owned or Financed Buildings, Roads, and Other Facilities, and in Disposing of Federal Lands and Properties," August 10, 1966);
17. Federal Insecticide, Fungicide, and Rodenticide Act as amended, and
18. The Noise Control Act of 1972.

Some of the laws noted above are concerned with one specific area of the environment, while others have a much broader area of concern. One of the most significant laws is the National Environmental Policy Act of 1969 which requires all Federal agencies to prepare detailed Environmental Impact Statements on major Federal actions that significantly affect the quality of the human environment. The National Environmental Policy Act requires that agencies include in their decision making process an appropriate and careful consideration of all environmental aspects of proposed actions, an explanation of potential environmental effects of proposed actions and their alternatives for public understanding, a discussion of ways to avoid or minimize adverse effects of proposed actions, and a discussion of how to restore or enhance environmental quality as much as possible. Facilities such as a Heat Pump Centered-Integrated Community Energy System (HP-ICES) which may be constructed in part with the assistance of Federal grant of loan funds, must be subjected to an environmental impact analysis to determine whether or not significant environmental impacts will result either directly or indirectly from the proposed project. This analysis is reviewed by appropriate Federal and State agencies and interested citizens or organizations and is submitted to the

President's Council on Environmental Quality. If significant environmental impacts will result from the proposed project, then a full Environment Impact Statement must be developed along with the related reviews and public hearings required.

Some of the major points which must be addressed in the preparation of an environmental impact appraisal are as follows:

1. Brief description of project;
2. Probable impact of the project on the environment;
3. Any probable adverse environmental effects which cannot be avoided;
4. Alternatives considered with evaluation of each;
5. Relationship between local short-term uses of man's environment and maintenance and enhancement of long-term productivity;
6. Steps to minimize harm to the environment;
7. Any irreversible and irretrievable commitment of resources;
8. Public objections to project, if any, and their resolution; and
9. Agencies consulted about the project.

The environmental impacts of facilities such as AN HP-ICES can be divided into two broad categories. These categories are:

1. The environmental impacts of construction, and
2. The environmental impacts of operation.

The next two sections will address the environmental impacts in each of these categories for heat pump centered-integrated community energy systems.

5.2 ENVIRONMENTAL IMPACTS OF CONSTRUCTION

5.2.1 TEMPORARY IMPACTS

The environmental impacts of construction can be further divided into those temporary impacts that exist only during the construction phase and the permanent impacts of the facilities' construction and location. The temporary environmental impacts caused by construction activities are as follows:

1. Noise and dust generated by construction activities;
2. Removal of vegetation and pavement causing soil erosion and siltation during construction;
3. Disruption of normal traffic patterns caused by construction in streets;
4. Potential for broken gas and water mains and sewer lines; and
5. Temporary inconveniences for businesses and residences while individual connections are being made

Most of the temporary impacts are the nuisance conditions that are usually associated with construction operations. Most of the impacts can be minimized by the proper planning and execution of various construction activities. For instance, the impact of noise in a residential neighborhood can be minimized by limiting the contractor's operations to normal daytime hours when many people are away from home at work or school. Since much of the distribution and return system will be constructed in existing streets and rights-of-way, some disruption of normal traffic patterns is unavoidable. However, by proper planning of construction operations, the actual time that a street must be closed can be minimized. One of the most significant potential impacts during the construction operations is the potential for breaking gas mains, water mains, and existing sewer lines while constructing the HP-ICES distribution and return system. In particular, a broken gas main or sewer line could create a significant environmental hazard. A broken gas main can create either an immediate or delayed explosion of disastrous proportions. A broken sewer line can result in the discharge of raw and untreated sewage to nearby drainage ditches, creeks and streams, and create serious public health hazards and water pollution problems. For those reasons it is important that the initial design carefully locate existing water and gas mains and sewer lines. In addition, it is important that the contractor take certain precautions during construction in the event that all such facilities are not properly and adequately located on the construction plans.

5.2.2 PERMANENT IMPACTS

The principal impacts of construction operations that are of a permanent nature are as follows:

1. Construction materials required for construction;
2. Energy utilized in the construction process;
3. Land occupied by proposed facilities; and
4. Disruption of previously undisturbed terrain.

The concrete, steel, and other materials required during construction are essentially irreversible and irretrievable commitments of these resources. That is, once they are used in the proposed construction, it is highly unlikely that they will ever be used again for any other purpose. The energy utilized during the construction process wil be primarily gas, oil and diesel fuel that is required for the operation of trucks, bulldozers, ditch diggers, and other construction equipment. Some electrical energy will also be utilized for electrical tools and equipment used during the construction of buildings and other structures.

The land on which the proposed facilities are to be located will be commiteed to this use for a minimum of 20 years and possibly for as much as 40 to 50 years or longer. However, this commitment is not permanent, in that, if the facilities were to be abandoned in future years, they could be torn down and the land could be utilized for other purposes. The major land requirement will be associated with the central plant facility and related storage facilities and cooling ponds that will be constructed. Land required for the distribution and return system will only be a narrow easement that is only 10 to 15 feet wide. In most cases, the easement for the distribution and return system will be located in existing streets and highway rights-of-way. In most cases, the presence of the distribution and return system will not prevent the land itself from being utilized for other purposes. However, in some cases ther may be a restriction in building a structure or building within the permanent easement.

One of the major environmental impacts of construction operations is the disruption of previously undisturbed terrain. During construction of the transmission and return system, rights-of-way must be cleared of vegetation and smoothly graded so that construction equipment can move easily along the right-of-way while digging the trench for the transmission main. Clearing and grading is also required at sites where buildings or other structures must be constructed. If the construction takes place in an area that has been previously undisturbed, then there is a danger that unique vegetation or wildlife habitat or previously undiscovered archaeological sites may be damaged or destroyed by the construction operations. For the Heat Pump Centered-Integrated Community Energy System, most of the construction for the transmission and distribution system is expected to be along existing streets and highways and therefore will not involve much construction in areas that have been previously undisturbed. The environmental impacts of construction that do occur in undisturbed terrain can be minimized by careful site selection and routing of transmission mains to avoid unique vegetation, wildlife, or archaeological sites.

5.3 ENVIRONMENTAL IMPACT OF OPERATION

5.3.1 IMPACTS ON SEWERS AND WASTEWATER TREATMENT FACILITIES

One of the most significant potential impacts of the proposed Heat Pump Centered-Integrated Community Energy System is on the existing sewers and wastewater treatment facilities within the community. There are several aspects of the project that could create an impact on existing wastewater facilities. These impacts are:

1. The additional hydraulic loading on the sewer system if the sewer system is used as the means of returning the heat pump water supply to the central facility.
2. The additional hydraulic loading on the wastewater treatment facility itself if the sewer system is used as the means of returning the heat pump water supply to the central plant.
3. The effect on the wastewater treatment process if the temperature of the untreated wastewater is raised or lowered as a result of the operation of the heat pump system.
4. The environmental impact on water quality and aquatic life in the stream where the treated wastewater is discharged as a result of a change in temperature of the wastewater discharge.

Because of the significant nature of these impacts, a separate discussion of each will be presented.

One of the schemes to be evaluated includes using the existing sewer lines to transport the used water from the heat pump system back to the municipal wastewater treatment plant and central heat exchangers and pumping facilities. This additional discharge to the sewer system would constitute an additional hydraulic loading on the sewer system over and above the estimated wastewater flows that were used to design the sewers initially. Consequently, if the existing sewer system were to be utilized as a part of the return system, it would be necessary to evaluate the impact of this additional hydraulic flow on the existing sewers. Several questions would need to be addressed. These are:

1. Can the existing sewer lines carry the peak hydraulic flows that will result when the spent water from the heat pump system is discharged into the sewer system?
2. Which sewer lines will be overloaded and how much additional sewer capacity must be provided?
3. Even if the existing sewer lines have sufficient capacity to carry the additional hydraulic flows, the ultimate capacity of the sewers will be reached sooner than originally anticipated without the water from the heat pump system. How soon will the capacity of existing sewer lines be reached?
4. If parallel or replacement sewer lines must be constructed to provide additional capacity, then the environmental impacts for this construction must be evaluated.

The hydraulic loading at the wastewater treatment plant would also be a point of concern in the HP-ICES scheme described above. The pumping facilities for the influent wastewater, the bar screen and grit removal facilities, and the primary clarifier would all have be sized to handle the additional hydraulic loading. If these facilities do not have adequate capacity to handle the additional hydraulic flow, their performance will be significantly impaired and the treatment facility will not achieve the degree of treatment for which it was designed. This could result in the discharge of raw or improperly treated wastewater to the stream.

A heat pump centered energy recovery system utilizing wastewater as the heat source and sink will raise or lower the temperature of the wastewater relative to the normally encountered temperatures. The temperature of the wastewater will be raised in the summer and lowered in the winter. Since the treatment of domestic wastewater usually depends upon some form of microbiological activity (e.g., activated sludge, trickling filters, etc.), and since temperature is an important factor affecting the biological activity rates, changing the temperature of wastewater will alter the rate in which the wastewater is biologically treated. Typically, the colder the temperature, the slower the rate of biological treatment. Biochemical reactions, in general, follow the van't Hoff rule of a doubling of reaction rate for a 10°C (18°F) increase in temperature over a restricted temperature range. Studies with activated slidge have shown the reaction rate to be more than doubled for a 10°C rise in temperature[30].

Typically, biological wastewater treatment systems are designed to operate at the coldest wastewater temperature that might be expected in the particular region in which the treatment facility is to be located. Typical minimum design temperatures may range from 45°F to 55°F. Since biological treatment rates would be slowed as a result of the lowered temperature in the winter, the lowered temperature would need to be considered during the design phase to size biological treatment units or to predict lost treatment efficiencies for existing facilities.

The change in biological activity rates may be expressed mathematically by the Arrhenius equation[31],

$$\frac{d\ln k}{dT} = \frac{E_a}{RT^2}$$

where dlnk/dT represents the change in the natural log of the biological activity rate constant with temperature, R is the universal gas constant, and E_a is a constant for the reaction termed the "activation energy." Integrating between the limits gives

$$\ln \frac{k_2}{k_1} = E_a \frac{(T_2 - T_1)}{RT_2T_1}$$

where k_2 and k_1 are rate constants at temperatures T_2 and T_1 respectively. Temperature is expressed in degrees Kelvin. By substituting the constant Θ for E_a/RT_2T_1, the equation becomes

$$\ln \frac{k_2}{k_1} = \Theta\,(T_2 - T_1)$$

or

$$k_2 = k_1\, e\, \Theta\,(T_2 - T_1)$$

By using the expanded form of e^x, the equation can be approximated as

$$k_2 = k_1 \left[1 + \Theta(T_2 - T_1)\right].$$

The constant Θ has been shown emperically to vary from 0.056 in the temperature range between 20^o and 30^oC and 0.135 in the temperature range between 4^o and 20^oC[32]. A value for Θ often quoted in the literature for biological wastewater treatment reaction rates is 0.047[24].

Therefore, a decrease in the minimum wastewater temperature from 55°F to 50°F would theoretically decrease a typical domestic wastewater biological reaction rate constant from 0.25 to 0.22 (12% decrease) and a decrease to 45°F would decrease the reaction rate constant to 0.18 (28% decrease).

Once adjusted for temperature, the reaction rate constant, k, which also varies for varying wastes and treatment conditions, can be utilized in the following equation to size biological treatment units:

$$L_t = L\ (10^{-kt})$$

where L is the initial BOD(biochemical oxygen demand, a measure of the organic strength of wastewater) and L_t is the treated BOD at retention time, t.

Additionally, wastewater temperature changes affect oxygen transfer rates, disinfection rates, and under extreme conditions the aquatic biota in the receiving stream. Both oxygen transfer and disinfection rates follow the van't Hoff-Arrhenius relationship, but in the opposite manner. Elevated temperatures decrease oxygen transfer rates while increasing disinfection rates. Therefore, consideration would likely need to be given to the decreased disinfection kill rates in the winter and decreased oxygen transfer rates in the summer.

A detailed discussion of thermal pollution and the effects of temperature changes on aquatic life is beyond the scope of the report. However, it may be state generally that as wastewater effluent temperatures increase, dissolved oxygen saturation concentrations decrease, and predominant aquatic lifeform become less desirable in character. Heated water discharges, after initial mixing, should not increase the temperature of the main body of the receiving waters above 95°F if fish life is to preserved. An excessive increase in the temperature of the wastewater-diluting water mixture above that of the diluting water should not be permitted. Increases should be limited to 3°F to 5°F, according to the report of the National Technical Advisory

Committee[47]. At the present tme, the permissible temperature increase is a matter of controversy, Detailed discussion of thermal pollution may be found in Krenkel and Parker[18].

5.3.2 IMPACTS ON MUNICIPAL WATER SYSTEM

The alternative scheme which utilizes primary clarified wastewater in the distribution system to each individual heat pump presents a potential environmental impact to the municipal water system. With this type of system there is always a danger that someone will make a mistake and connect the heat pump water system to the municipal water system and thereby cause contamination of the municipal water system. Even if the primary clarified wastewater is chlorinated before going into the heat pump distribution system, a potential for contamination would exist if cross connections were made between the heat pump water distribution system and the potable water distribution system. Another potential danger is that a plumber or homeowner might by mistake connect into the heat pump water distribution system for the water supply for a home or business. Such a mistake would, of course, be a serious public health hazard.

5.3.3 IMPACTS ON ENERGY SOURCES

One of the most significant positive impacts of the proposed Heat Pump Centered-Integrated Community Energy System is a more efficient utilization of available energy for the heating and cooling of residential homes and commercial establishments. In addition, the heat pump system will utilize low-grade heat found in wastewater discharges which is now wasted. The primary energy requirements for the porposed heat pump system will be for the pumping facilities which will be a part of the distribution system, the electrical power for each individual heat pump and the energy required at the central plant facility for the pumps, heat exchangers, and other equipment and other equipment needed there. Once a prelimnary design and layout has been developed for a particular system, it will be possible to estimate the total energy that will be required in the system's operation and compare this energy consumption with the amount of energy that would have been required with a conventional heating and cooling system.

6.0 PROJECTED GROWTH

6.1 INTRODUCTION

A Heat Pump Centered-Integrated Community Energy System will, in a sense, be a public utility that will be designed to serve all or part of the existing population of a community and all or part of future increases in the community's population. Consequently, the planning and design for such a system must carefully evaluate both the existing population of the community and future changes that are anticipated in the community's population. The purpose of this section is to review briefly the methods that are available to planners and engineers for determining future population changes and to describe how AN HP-ICES would be expanded to meet various rates of increasing population.

6.2 METHODOLOGY FOR PROJECTING GROWTH

6.2.1 POPULATION PROJECTIONS

There are numerous methods available to engineers and planners for projecting the future population of a given area. The most commonly used methods may be classified as:

1. graphical;
2. decreasing rate of growth (increase);
3. mathematical or logistic;
4. ratio and correlation;
5. component;
6. employment forecasts.[21]

Using one or more of these methods, an individual researcher can gather the appropriate data for a particular area and develop a totally independent population projection for that area. The method utilized can be a simple graphical extension of population based on past population changes or it can be a complex and mathematically sophisticated method which requires considerable time and an involved analysis. For many years the preliminary planning that was done for public facilities required that such independent population projections be developed. However, many of these independent population projections were later found to be in error, due in part to the assumptions made by the investigators and in part to the personal bias of the investigators concerning the area in question. In recent years Federal and State agencies have become actively involved in developing State and National population projections which in turn are disaggregated to the local level. For most public facilities that are being financed in part with Federal grants or loans, it is usually required that population projections used in designing these facilities be in agreement with State and Federal projections that have been developed for the particular area in question.

In Georgia, the State Office of Planning and Budget has the responsibility for developing population projections in the State. A recent report by that office was entitled "Population Projections for Georgia Counties, 1980-2010."[38] The population projections described in this publication are termed "baseline projections." The term "baseline" denotes an estimate of what can be expected if there are no substantial changes that would impact population growth. The population projections for Georgia counties utilized data from each individual county along with data from each Area Planning and Development Commission. In Georgia each of the 159 counties is located in one of 18 Area Planning and Development Commissions (APDC). One of the main functions of an APDC is to maintain accurate and complete information on population changes and population trends for each of the counties within the APDC. In addition, most of the APDC's have developed population projections for individual communities and cities within their jurisdiction.

Within the Federal government, the Water Resources Council has in recent years coordinated the work of several Federal agencies in the area of population projections. Under a cooperative agreement with the Water Resources

Council, the Bureau of Economic Analysis, U. S. Department of Commerce, has worked jointly with the Economic Research Service, U. S. Department of Agriculture, to develop unified population projections for each state in the nation along with the counties within each state. Population projections developed under this program have been commonly referred to by the acronym of "OBERS." A concerted effort has been made to assure cooperation between the Federal agencies involved, each responsible State agency, and the local Area Planning and Development Commissions to assure uniformity and agreement among the population projections published by each of these agencies.

The most complete data available on existing population is the Federal Census of Population which is conducted every 10 years. The Census of Population contains extensive population data for each state and the counties and municipalities within each state. Between censuses, the Bureau of the Census in the U. S. Department of Commerce in conjunction with the State agency that has responsibility for population projections publishes a yearly report which gives the estimated population of each state and the counties within the state for that year. These reports are referred to as "Current Population Reports Series P-25." For individual communities and municipalities within a county, it may be necessary to utilize other sources of information to estimate existing population between censuses. Changes in population since the last census can be estimated by utilizing school enrollment records, records of births and deaths, records of building permits for residential construction and automobile registration records. Fairly accurate estimates of existing population can be made by properly utilizing such information as noted above.

6.2.2 ECONOMIC ACTIVITY PROJECTIONS

A Heat Pump Centered-Integrated Community Energy System would serve not only residential areas but commercial and industrial areas as well. Hence the planning and design for such a system would need to consider growth of commercial and industrial activities of the area. Projections of future economic activity within an area most frequently give consideration to past commercial and industrial growth that has occurred and the potential for future commercial and industrial growth based on the conditions generally considered necessary and desirable for economic growth to occur.

Area Planning and Development Commissions are frequently the best source of information concerning past commercial and industrial growth and projections of future commercial and industrial growth for a particular area. These agencies frequently conduct studies to analyze the type of economic growth which has occurred in a particular area and at the same time analyze the potential for future economic growth. Such studies often include estimates on the expansion of existing businesses that can be anticipated and projections of the type and size of new businesses and industry that may locate within the area during a particular time frame. In addition to the studies and reports published by the APDC's, the State government may have an office or department whose primary responsibility is promotion of commercial and industrial growth within the state. Data available from this office or department may be valuable in projecting the future economic growth of an area. However, it may be limited by the depth of coverage for a particular geographical area.

Other sources of information concerning the future economic growth of an area would include the planning departments in the larger cities and Chambers of Commerce. However, it should be noted that the economic growth which may be projected by a Chamber of Commerce is likely to be the ideal growth which the Chamber of Commerce desires to see and not necessarily the amount of economic growth that can be realistically anticipated.

6.2.3 FUTURE LAND USE PLANS AND EXISTING LAND USE PATTERNS

Another important aspect in the design of a public facility such as AN HP-ICES is determining where within the study area the residential, commercial and industrial growth will occur. The previous two sections have presented the data sources and methodology that can be utilized in projecting population and economic growth for a particular area. However, it is equally important that the specific areas within the community be delineated where residential, commercial and industrial growth is anticipated to occur. The most frequently used method for determining where future growth will occur is by analyzing future land use plans that have been adopted for the community. Future land use plans are generally developed by a city planning commission or an Area Planning and Development Commission and have taken into consideration existing land use, existing public facilities such as water and sewer lines, streets and major thoroughfares, and physical features such as flood plains, topography, soil and geological characteristics and other factors which might influence ultimate land use and development. Taking into consideration the factors that have been noted, future land use plans generally propose those land uses which appear to be most desirable for each particular tract of land. Before utilizing future land use plans in the planning process, it is important that planners and engineers determine the validity of a particular land use plan. In particular the following questions should be addressed:

1. Has the land use plan been formally adopted by the local government?
2. Are the goals and objectives of the land use plan being implemented through a program of planning and zoning regulations?
3. Is the land use plan used as a guide for decision making by local governmental officials?

If the land use plan has been formally adopted and is being implemented through planning and zoning regulations, then it can be a valuable tool in determining where future growth within the community will occur. However, if the land use plan is not being implemented, then it may be more desirable to use other sources of information to project where growth will occur. In some cases future land use and growth can be more accurately projected by analyzing existing land uses and recent trends and changes in land use. In many cases, the past trends in land use and land use changes for the last five to ten years may continue for the next five to ten years. That is, a portion of town which was previously undeveloped but has seen heavy residential development in recent years is likely to continue to experience residential development in the coming years unless some limiting factor develops. In the same sense, areas of town which have recently experienced commercial or industrial development are likely to continue to experience such development if suitable land is available. Data and information on existing land use and land use patterns is available from several sources. Planning commmissions and Area Planning and Development Commissions often have published reports that contain extensive data and maps on existing land uses. In addition, maps that show existing zoning often accurately reflect existing land uses. Changes in land uses and changing land use patterns can be detected by reviewing zoning changes in recent years, location of new residential and commercial construction in recent years, aerial photographs taken of the same area over a period of years and maps of the community that were developed at several different time periods.

6.2.4 ACCURACY OF GROWTH PROJECTIONS

In projecting future population and economic growth for an area, it is important for planners and engineers to have an understanding of the degree of accuracy that can be anticipated with these projections. There are numerous factors which influence the population and economic growth of an area and unexpected events or changes in existing conditions can have a significant influence on population and economic growth. As a general rule, the accuracy of population estimates decreases as

(1) the time period of the forecast increases,

(2) the population of the area decreases, and

(3) the population rate of change increases.[21]

Consequently, population projections for the next three to five years will generally be much more accurate than population projections for the next ten to 20 years. Also, population projections for a large city are more likely to be accurate than population projections for a small city. In small communities population growth is significantly influenced by unexpected events that may occur during the planning period. For small communities such things as the opening of a new industry or the closing of an existing industry or changes to a nearby military installation can significantly affect the population of the community.

For the reasons that have been noted, projections of future growth may not be highly accurate and, hence, this fact must be considered by planners and engineers in the planning and design of public facilities such as the Heat Pump Centered-Integrated Community Energy System.

6.3 CENTRAL PLANT AND MAIN PUMPING FACILITIES

6.3.1 INITIAL DESIGN AND SIZING

In the planning of public facilities such as water treatment plants, wastewater treatment plants or the central portion of AN HP-ICES, one of the most important decisions to be made is the initial size and capacity of the facility. Since the design of such facilities often requires six to twelve months and the construction frequently requires a minimum of one to two years, these facilities cannot be rapidly expanded in a short time frame. Hence, considerable thought has to be given to the initial design and capacity that is to be provided. The initial design should also include provision for future expansions to be made in the most economical way possible. The layout of the central plant should show where future expansions will be located in relation to the initial facilities constructed. Under some circumstances, it may be more economical in the first phase of construction to go ahead and construct larger piping that will be sufficient to handle future flows during later expansions. In some cases, it is desirable to allow extra space in major structures such as buildings or pump stations so that additional facilities such as a pump or heat exchanger could be added at a later date with a minimum of additional cost involved.

For each community in which it is proposed to construct AN HP-ICES, it will be necessary to conduct a study to determine what portions of the community can be economically served by such a system. This study would also project the anticipated growth within the community that would be served by the system in future years. The design and sizing of the central plant station would be based on the capacity needed to serve the existing population and future growth that is anticipated to occur within the next five to ten years and possibly longer. The next two sections will discuss the sizing of the central plant facilities to handle various rates of growth within different communities.

6.3.2 PHASING TO HANDLE LOW TO MODERATE GROWTH SITUATION

In recent years, the most common time frame that has been utilized in planning public facilities such as water and wastewater systems has been 20 years. By definition, the start of the planning period is the date that the initial facilities which are constructed begin operation. Hence, the time required for engineering study, design and construction is not included as a part of the planning period. There were a number of reasons that a 20-year planning period was chosen. One of the main reasons was that it is very difficult to accurately project population and economic growth beyond 20 years. For the purpose of this discussion, a 20-year planning period will be utilized to describe how a Heat Pump Centered-Integrated Community Energy System will be expanded to handle various rates of growth.

There are a number of different ways that can be used to describe the capacity of AN HP-ICES. However, for this discussion, the capacity of the system will be discussed in terms of the number of customers served. To further simplify the discussion, it will be assumed that there are no commercial customers and that all customers are single family residential users. The purpose of this section is to explain how the central plant portion of an HP-ICES would be expanded to handle low to moderate growth that occurs within the community during the 20-year planning period. Low to moderate growth is defined here as being an increase of 0 to 50% over and above the initial number of customers served at the beginning of the planning period. By this definition, a system that has 1,000 customers at the beginning of the planning period might grow to 1,500 customers by the end of the planning period. The initial sizing of facilities and later expansion of facilities to serve a community will depend not only on the amount of growth that occurs during the planning period, but also on when the growth occurs during the planning period. In most cases, unless better data is available, it will be assumed that the growth will occur at a uniform rate over the entire 20-year planning period. However, in some cases, it may be possible to show that the maximum growth will take place in the early, middle or latter portions of the planning period. For communities that experience a fairly low rate of growth (0 to 20%) during the 20-year planning period, it is generally more desirable to construct sufficient capacity initially to handle the anticipated growth rather than to attempt small scale expansions of the central plant. Moderate rates of growth (20 to

50%) can generally be handled by minor expansions to the central facilities during the 20-year planning period. Minor expansions at the central facilities would include such items as providing larger pump motors, changing pump impellers, adding a new heat exchanger, or a new pump in space that had been provided during the initial design and construction. The initial design and layout of thecentral facility plant would have made provisions for these minor expansions and, hence, major construction of expensive structures or buildings would not be required.

6.3.3 PHASING TO HANDLE HIGH GROWTH SITUATION

In this discussion a high rate of growth is defined as an increase in capacity of more than 50% over the initial capacity that is required. Hence, if a system initially serves 1,000 residential customers, by the end of the planning period the number of customers served will have increased to more than 1,500 customers. For communities that experience a high rate of growth, the central plant facilities may have to be expanded as much as 200 to 300% over their original capacity during the 20-year planning period. In situations such as this, there are several reasons that it is usually desirable to construct the total capacity needed at the central plant in several stages over the 20-year planning period. Because of the inaccuracies involved with growth projections, it would be very risky to construct the total capacity required in the initial stage of construction. From a monetary standpoint, it is not desirable to spend large sums of money to construct excess capacity at the central facility that will not be needed until much later in the planning period. It is more desirable to postpone major capital expenditures until later in the planning period if possible. However, it should be pointed out that the monetary benefits of a program of phased expansion have been offset considerably in recent years by the rising costs of construction due to inflation.

For communities that are expected to experience a high rate of growth during the planning period, the central plant and related facilities would be designed and constructed to provide sufficient capacity for the first five to ten years with one or two planned expansions to provide the total capacity needed at the end of the 20-year planning period. The design and layout of the initial facility construction would take into consideration the future expansions that are proposed and to the maximum extent possible the initial design and construction would attempt to minimize the cost of future expansion.

6.4 DISTRIBUTION AND RETURN SYSTEM

6.4.1 INITIAL DESIGN AND SIZING

The distribution and return system for a Heat Pump Centered-Integrated Community Energy System will consist of pressure distribution mains very similar to a water system and either a closed loop, pressure return system or a gravity return system utilizing the existing sewer system. In addition to the distribution and return system, other related facilities include the high pressure pumps and storage reservoirs that will be required. There are several characteristics about the design and construction of water and sewer lines that influence the manner in which distribution and return systems will be constructed and expanded. First of all, water and sewer lines generally have a useful life of from 40 to 50 years. Second, a major portion of the cost of constructing water and sewer lines is not for the pipe and material involved, but for the manpower and equipment required in the construction process. Hence, the most economical way to provide additional capacity is generally to construct a larger line initially rather than a parallel line at a later date.

6.4.2 INITIAL CONSTRUCTION PHASING

Since construction of the distribution system may require from 12 to 36 months or more, a phased approach to construction may be desirable. The distribution system would be divided into operable segments each of which could be placed into operation upon completion. This procedure would eliminate the need for the total construction to be complete before the system was placed into operation.

6.4.3 PHASING TO HANDLE LOW TO MODERATE GROWTH SITUATION

Once it has been determined which areas of the community are to be served by the HP-ICES, a study should be made to determine which portions of the distribution and return system will be constructed immediately and which portions will be constructed later in the planning period. For those communities that are expected to experience low to moderate growth, the entire distribution and return system may be constructed initially. When the distribution and return system is constructed for a particular geographical area, the distribution and return mains are sized to handle the anticipated growth that will occur in that geographical area during the planning period. This means that if a subdivision has homes constructed on 70% of the lots, then the distribution and return system will be sized to handle the existing homes and also anticipated homes to be constructed on the remaining lots in the subdivision. In most cases, low to moderate rates of growth can be accommodated within the geographical area of the initial distribution and return system construction and new customers can be served by merely connecting onto the distribution and return system. In other cases, minor extensions of the distribution and return system may be sufficient to serve new areas of growth not served by the original system.

6.4.4 PHASING TO HANDLE HIGH GROWTH SITUATION

As previously mentioned, a high rate of growth is defined here as an increase of 50% or more in the number of customers served initially by the system and may, in some cases, be an increase of 200 to 300% over the initial size of the system. For a community that experiences a high rate of growth, some of this growth will occur on undeveloped land that has already been served by the original distribution and return system. However, for communities that experience very high rates of growth, it is likely that significant residential and commercial development will occur in areas that have been previously undeveloped, but adjacent to the community. In such situations, completely new distribution and return systems must be constructed as these areas develop. An advantage to new residential and commercial development is that a distribution and return system can be constructed more economically if it is planned and constructed as a part of the initial development, as opposed to being constructed after the streets and other utilities have been constructed.

In order to serve new areas that are expected to develop during the planning period, it may be necessary to construct new transmission trunk lines from the central plant facility to the facility of the developing area. In some cases these new lines may parallel trunk lines that were built initially. In other cases it may be more economical to construct larger mains initially to handle the later growth in order to avoid the expense of constructing a parallel line at a later time.

7.0 IDENTIFIED VARIATIONS

7.1 INTRODUCTION

In any system that utilizes the number of separate components that are required in the HP-WHR system, the possibilities for component variation and system mutation become virtually unlimited. However, several areas of system design show promise for performance and/or system economics improvement through innovative systems engineering. These areas include: possibilities of utilizing other sources of thermal energy, component selection and specification from the extensive varieties of available equipment, and variations in system design and utilization. Several of the obvious variations in each category are discussed in the sections that follow.

7.2 THERMAL ENERGY SOURCES

While it is expected that the most promising and most easily accessible source of low-grade thermal energy will be a community's wastewater effluent, a site-specific energy resource inventory may dictate that other sources be considered. Other promising alternatives include solar energy, waste heat from industrial processes (such as power generation), waste heat recovery through solid (or gaseous) waste incineration, or composite energy sources.

Solar technology has produced active solar energy collectors, such as the General Electric model TC-100, that are proving effective in harnessing solar thermal energy. Another workable variation of collector-type is the solar pond collector design. This collector variation may prove economic and advantageous when large demand is placed on the HP-ICES system and when large land areas are available for siting the Central Station. The principal advantage to utilizing solar energy as the primary energy source in an HP-ICES scheme is harnessing a previously untapped, universally available thermal energy source. There are several major disadvantages, however, to utilizing solar energy in this scheme (at least as the primary energy source). First are the typical disadvantages associated with active solar systems - size and cost. A central solar facility proposed for a middle Georgia location required 3323 ft.^{2} of solar collectors at an estimated system cost of $246,298 to supply the space heating and cooling needs for 15 houses (955 net (heated)ft.^{2} average). A second major disadvantage to utilizing solar energy is that a backup thermal energy source must be supplied for periods when solar energy is not available. The third disadvantage is that an additional interface is required between the thermal source and the Central Station heat pump equipment.

Waste heat rejected from industrial processes may be reclaimed in essentially the same manner as from community wastewater. Depending upon the process and the location of the industry in relation to the community, low-grade thermal energy may be available in amounts significant enough to warrant a modular system with several central station installations. A central station co-located on the site of a utility power plant and extracting thermal energy from the condenser circulating water is an attractive option as an alternative energy source. The advantages are similar to those of the wastewater heat recovery option; i.e. utilizing a previously untapped source of (low-grade) thermal energy at an easily accessible location. An additional advantage is that, as the hourly demand on the HP-ICES from the industrial sectors increases, a proportional increase in thermal energy rejected from the industrial processes becomes available. This inherent demand-tracking supply will simplify somewhat the load balance of an HP-ICES.

A third major energy source for consideration is the heat of combustion from various waste materials. One or several waste-heat-type boilers could be utilized in the system to reclaim heat in the form of trash (deposited at sanitary landfills), wood scraps (available as waste from various wood processing operations), or combustible gasses (generated as by-products in certain processes). For example, methane gas is typically generated in anaerobic sludge digesting processes at sewage treatment plants and is sometimes utilized as an auxiliary fuel in the digestor heating furnaces. The principal advantage of these sources is that the primary fuels, where available, would be very inexpensive. However, several major disadvantages weigh against utilizing combustion heat as a primary thermal energy source. First, the fuels that are available in bulk (trash, wood scraps) would require transportation and preparation before firing in the heat reclaim boiler. Secondly, the additional first cost of the heat reclaim boiler(s) would extend the payback period of the initial installation of the Central Station and make it less attractive economically. Finally, the additional interface (the heat recovery boiler) between the thermal source and the central station equipment is again required.

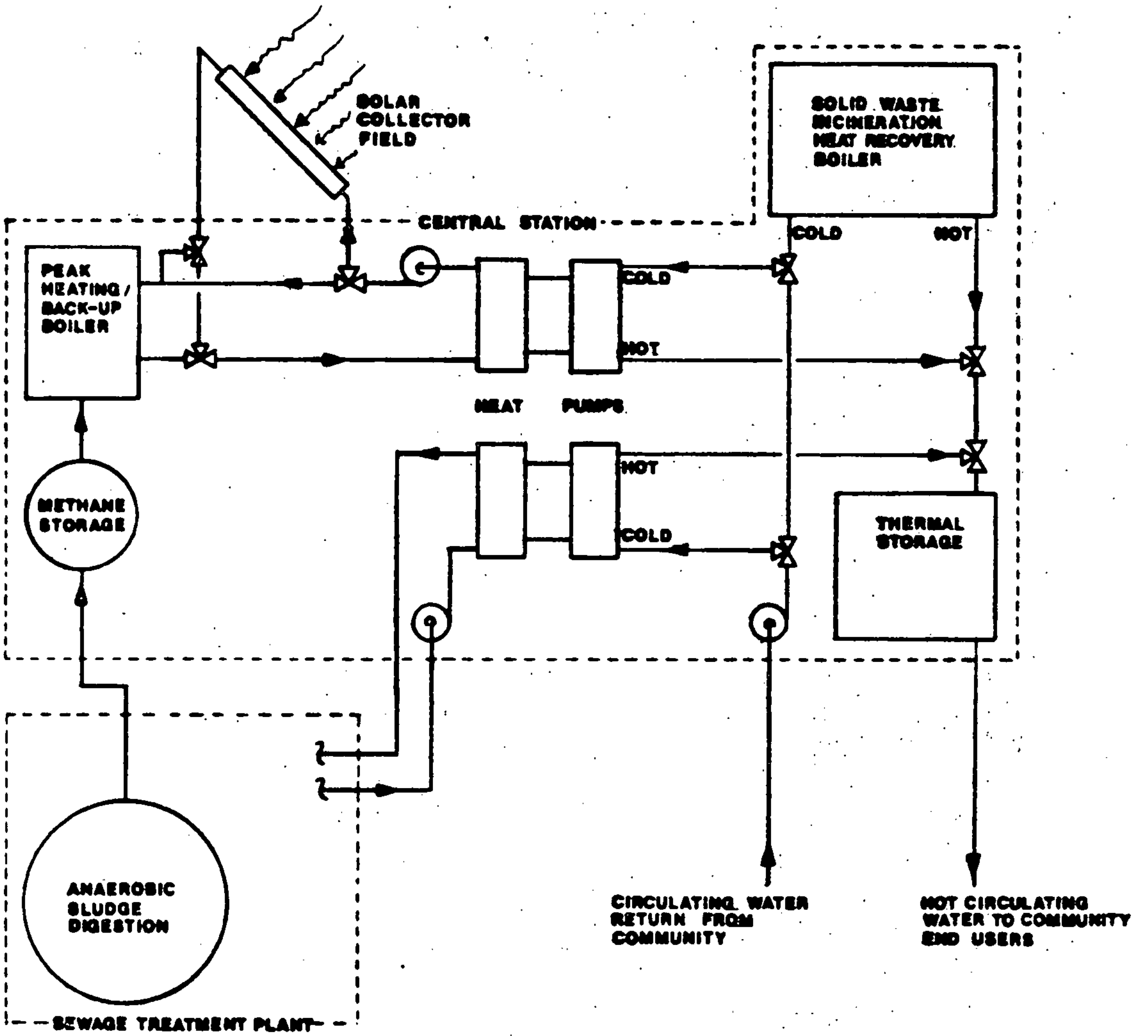
Depending on a site-specific inventory of available resources, a composite thermal-source ICES may prove economically feasible and energy effective. The basic design of the overall system, detailed in Chapter 1, can be adapted very easily to multi-thermal source form. An evaluation of the feasibility of utilizing one or more of the above mentioned sources will depend on the particular needs, characteristics, and resource inventories at each site of application. For a hypothetical example, though, let us consider a community with a sewage treatment plant of limited capacity. The entire thermal demand on the HP-ICES may not be supplied by the wastewater effluent. However, the primary energy source may be supplemented with a field of solar collectors that could be expected to supply a certain percentage of the thermal load (optimized with regard to size and costs). Also available for peak demand periods or as a back-up thermal source for the solar segment of the thermal supply would be a limited supply of methane gas from the treatment plant's anaerobic sludge digestion system, to be converted in a small heat-reclaim boiler. In addition, a second large-scale solid waste utilization-type boiler could be incorporated for trash incineration (trash that would normally be disposed of at a sanitary landfill typically located at or near the sewage treatment facility). Such a system would provide a site-efficient composite thermal source for the HP-ICES, optimized for the particular energy supplies and demands of the community. Figure 7-1 details one possible configuration of such an installation.

SUMMARY

It is anticipated that the most promising source of thermal energy will be the low-grade heat available in the community's wastewater effluent. However, there is the possibility of augmenting the primary energy source with any combination of the other alternative sources. The overall consideration is that enough thermal energy must be available, through any combination of sources, to provide the community, or portions of it, with a continuous, consistent, and reliable thermal service, via the HP-ICES. For each particular site of application, an energy resource inventory must be performed according to the First and Second laws of Thermodynamics and the procedures of "Availability Analysis" (References 34 and 35).

7.3 COMPONENTS

System detail design and component specification is often a headache because of the wide variety and quality of components on the market.



COMPOSITE THERMAL ENERGY SUPPLY
FOR HPC-ICES

FIG. 7-1

However, this same variety provides the engineer with extensive opportunity to improve system performance through judicious component selection and innovative subsystem design.

The most obvious candidate for component variation is the heat pump, either the Central Station pump or the end-user pump. The most widely available Central Station equipment is based on a fairly standard design with variations mainly in compressor design. (The most popular compressor designs are centrifugal, reciprocating, and screw-type.) Available water source heat pumps (for residential use) are even more similar. While most pumps employ electric induction motors to drive the Rankine cycle compressor, there are obviously other means of providing power to the compressor. Innovations in the field of thermal engine heat pumps include several composite heat pump cycles. Stirling - Rankine, Rankine - Rankine, Brayton - Rankine, Otto - Rankine, and the Ericsson - Ericsson. (Discussion of each of these cycles are found in Reference 3.) Refinement of the devices based on these cycles promises to make them competitive, both performance-wise and cost-wise, with equipment now available.

The distribution system will be designed according to standard water distribution system design practice. However, there is considerable latitude left for the design engineer in the specification of the piping materials to be used. There may be significant advantages to utilizing the new piping systems based on plastics such as Polyvinyl Chloride (PVC) or high density Polyethylene. Advantages that are immediately obvious are reductions in materials costs, lower coefficient of thermal conductivity of the material, and ease of installation of the integrated design piping systems.

Water quality of the thermal energy source/sink, i.e. wastewater, may cause problems that could be remedied by judicious component selection. The increased fouling factor of the water to be circulated through the Central Station equipment heat exchanger may dictate one of several options. Option one is the standard HVAC design technique of specifying a larger size heat exchanger to accomodate the increased resistance to heat transfer contributed by the high fouling factor of the circulated wastewater. Option two would be the specification of a plate-type heat exchanger which would allow easy disassembly and cleaning and thus a speedy turnaround time for scheduled maintenance. Option three would be the specification of a mechanical cleaning system for the heat exchanger such as reverse-flow actuated brush scrubbers or a continuous cycle spherical rubber scrubber system such as an Amertap. Which, if any, of the above options should be selected will depend on the quality of circulated water at each specific site. General guidelines covering expected wastewater fouling factors and the corresponding appropriate subsystem modifications may be developed in the component testing program outlined in Chapter 8. However, the final judgement is best left to the system engineer performing the detail design and component specification.

7.4 SYSTEM VARIATION

The basic Heat Pump Wastewater Heat Recovery (HP-WHR) system concept lends itself nicely to variations within the system or even to variations in the utilization of the end-product thermal service.

One particular variation that shows promise is the addition to the system of an evaporative cooling component (cooling tower). Instead of sacrificing the power necessary to reject heat to the wastewater heat sink through the Central Station heat pump, the community circulating water could be cooled by means of a mechanical draft cooling tower. Some sacrifice would be necessary in the form of pump and fan horsepower, but it is expected that the combined operating and maintenance costs of the cooling tower during the cooling season would be less than those associated with running the Central Station equipment. Also, if necessary or advantageous, the distribution system could be segmented with several cooling towers located throughout the community to handle cooling of the circulating medium.

Another variation that may be considered is the clarified Wastewater System. This system involves utilizing an open distribution loop to deliver the reclaimed thermal energy. Delivery of thermal energy to the end-user is accomplished by utilizing partially clarified wastewater as the transport medium in a one pipe distribution network. However, several major disadvantages may eliminate this variation from consideration.

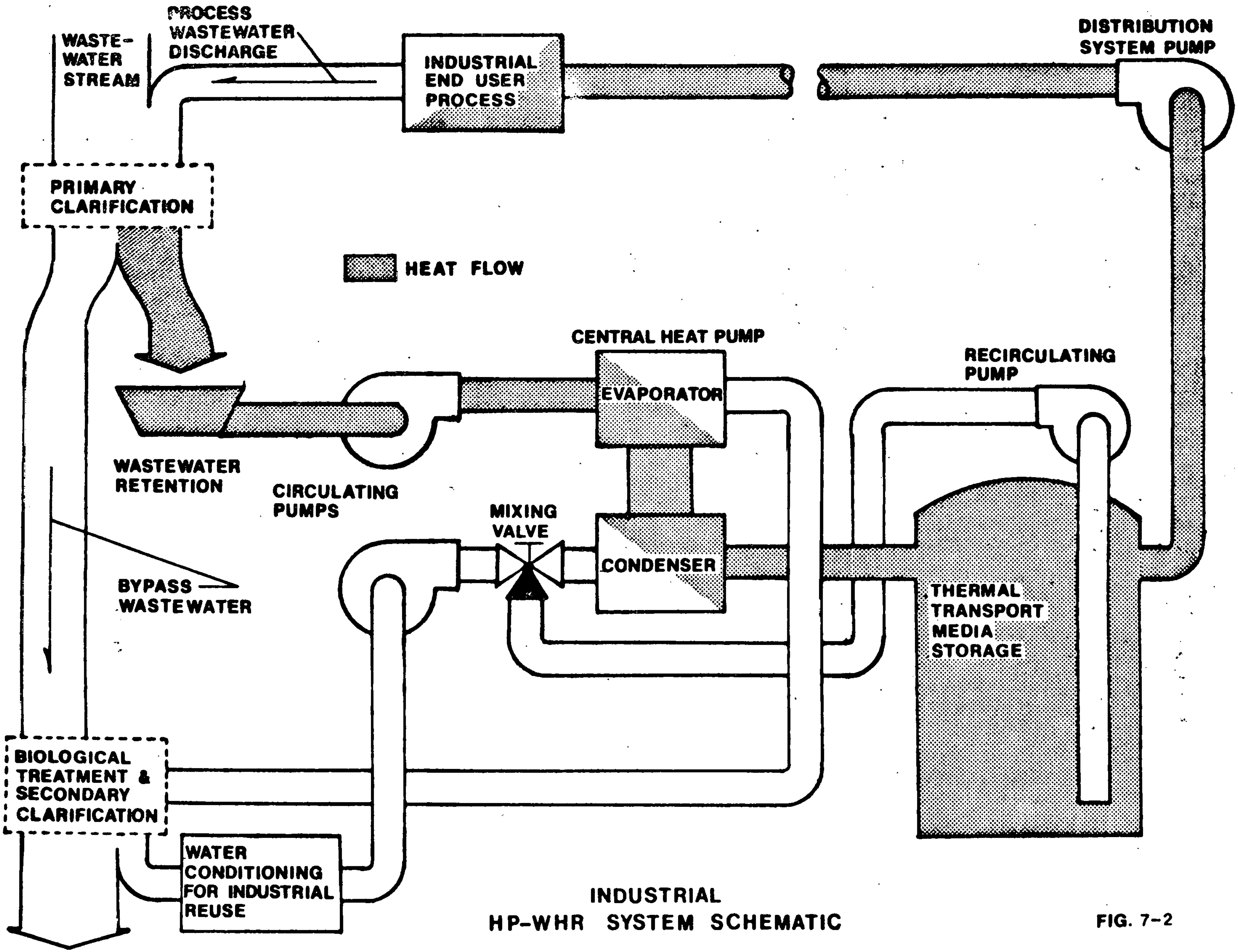
First, the demand for thermal energy in the heating mode will tend to coincide with the morning peak of wastewater effluent flow. Wastewater flow typically is on a 24-hour cycle with the major peak at mid-morning (9:00 - 11:00am) and a second, lesser peak at late afternoon. Peak thermal demand typically is early-to mid-morning due to the early morning heating load and cold start-up of many businesses and industrial processes. This coincidental peak may pose serious hydraulic overloads on existing sewage collection systems where thermal media flow at peak conditions might be a large percentage of the normal wastewater flow. No general rule is applicable since hydraulic loading of existing sewer systems is dependent on the specific application and will change with a number of variables such as sewer age and location in the sewage collection network.

Another serious problem is the impurity and bacteriological content of the clarified effluent. Severe fouling of heat transfer surfaces is probable due to the high suspended solids (BOD) content of the effluent. A relatively serious public health hazard might also be introduced by accidental leakage of the contaminated effluent.

One further detrimental effect the clarified wastewater system would impose when the transport medium is returned to the sanitary sewer system would be to lessen the overall thermal quality of the primary heat source by lowering its temperature. The amount of available thermal energy would be decreased by the mixing process.

Even with all the negatives aforementioned, the clarified wastewater concept may be viable in specific instances and should not be wholly discarded without consideration. In general application, however, it would seem that the scheme poses an unacceptable number of problems which require specialized solutions. It is doubtful that the distribution system cost advantage can outweigh the penalties.

A third system variation (pictured in Figure 7-2), perhaps the most promising, proposes to supply the thermal services in a form suitable for use strictly in industrial processes, and to utilize wastewater treated and conditioned for industrial re-use as the thermal energy distribution medium. (This latter provision in the HP-WHR system variation is becoming more and more attractive as current Environmental Protection Agency 201 Facilities Studies require an evaluation of the feasibility of re-using all or portions of the wastewater treated at sewage treatment facilities. Industrial re-use of treated wastewater is a logical first-step in implementing water re-use technology.) As shown in Figure 7-2, an open distribution system is proposed. However, since the amounts of water conditioned by this system will directly displace the amounts of (make-up) water discharged through industrial processes, the danger of hydraulicly overloading existing wastewater collection systems is avoided. Also, the load balance on



INDUSTRIAL
HP-WHR SYSTEM SCHEMATIC

FIG. 7-2

a system applied in this manner would be greatly simplified since the thermal demand placed on the system would directly coincide with the supply of available energy rejected through the industrial processes. This overall system concept is especially attractive to industrial communities requiring large quantities of high temperature water for any number of process uses.

7.5 SUMMARY

The energy source, component, and system variations discussed in the preceeding sections just begin to address the possibilities. The basic HP-WHR system concept is flexible enough to allow easy adaptation to a number of different applications. Technically speaking, the possibilities are limited only by the ingenuity of the design engineer.

8.0 COMPONENT TESTING

8.1 INTRODUCTION

The equipment to be utilized in this HP-ICES scheme will, for the most part, be standard, "off-the-shelf" components that are readily available from the major HVAC equipment manufacturers. However, due to the novel application of this equipment, it is expected that Application Ratings will not be available for the range of operating conditions that will be in effect for this system. Since it is the intention of this proposal to provide a system that offers improved performance over standard Heat Pump systems, performance data specific to this application will be developed through component testing. Three areas will be considered: 1.) baseline data on the thermal energy source(s), 2.) Central Station heat pump equipment performance, and 3.) end-user heat pump equipment performance. With this specific performance data in hand, the detail design of the system can be optimized according to energy supply and demand.

8.2 THERMAL ENERGY SOURCE(S) - BASELINE DATA

Each application of this scheme may utilize one or more of several potential thermal energy sources. Obviously, each will require heat reclaim equipment particular to the source. However, the low grade energy that is available from a community's wastewater effluent will, in all probability, be the most widely accessible source. Therefore, the focus of this section will be to determine the baseline data, considering wastewater effluent as the primary energy source, that the HP-WHR system performance will depend upon. The parameters that will be considered are temperature and flow. Discussions of the approach to be taken in obtaining these data follow.

8.2.1 TEMPERATURE

One assumption that the performance of this HP-ICES scheme is based upon is that community wastewater effluent temperature will be greater than or equal to ground temperature. Several factors, such as soil type and thermal diffusivity, and factors particular to the design and operation of individual wastewater collection and treatment systems, will affect the effluent temperature. However, this base assumption appears to be generally valid.

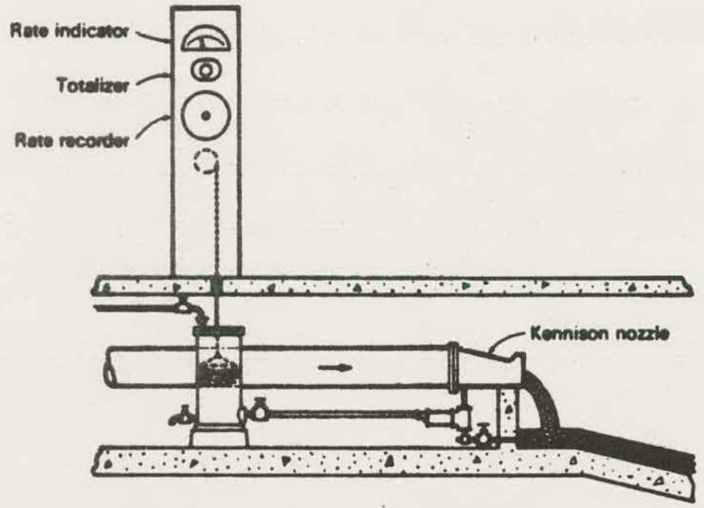
A cursory look at estimated operating temperatures can be found in a tabulation of ground temperatures contained in Technical Guidelines for Energy Conservation.[46] Also, predictions of estimated operating temperatures (i.e. earth temperatures) can be derived from the procedures developed by Kusuda and Achenback.[19] In general, this type survey and estimate will be a suitable first step in determining applicability to a specific region or location. However, to obtain a data base from which an overall system Coefficient of Performance (C.O.P.) can be derived, specific data must be gathered at the point of application. Temperature measurements should be taken at the appropriate points in the sewage treatment plants serving the demonstration communities including raw sewage influent, plant effluent, proposed supply tie-in point(s) (if different from plant effluents), and ground temperature. Measurements should be made on an hourly basis for a period of one week (168 hours) during each of the following months: January, April, July, and October. The preferred method of data taking would be temperature transducers (i.e. thermocouples or resistance temperature devices) in conjunction with a multipoint strip chart recorder. Additionally, available plant operating records should be surveyed to verify the correlation between the empirical data and typical conditions.

8.2.2 FLOW RATES

The amount of thermal energy available at the sewage treatment plant will also depend upon the mass flow rate of the energy supply medium (plant influent/effluent). Flow rates may be available in the form of plant influent flow recordings and/or E.P.A. 201 Facilities Plans. However, the mass flow rate available as a thermal energy source may, depending on the design and/or operation of each particular sewage treatment plant, differ from the rates documented in available reports or records. Therefore, the following flow measurements are recommended: raw sewage influent (possibly available from existing instrumentation), plant effluent, and flow at the proposed supply tie-in point(s) (if different from plant effluent). Again, measurements should be made on an hourly basis for a period of one week (168 hours) during the months of January, April, July, and October. Flow data collection periods should correspond with the periods of temperature data collection mentioned in section 8.2.1.

Existing automatic flow monitoring instrumentation should be utilized where possible. Two principal types of flow measurement instrumentation, the Kennison Flow Nozzle-type installation, and the Parshall Flume installation, are in widespread use in modern sewage treatment plants and will yield flow rate information of sufficient accuracy for this study.

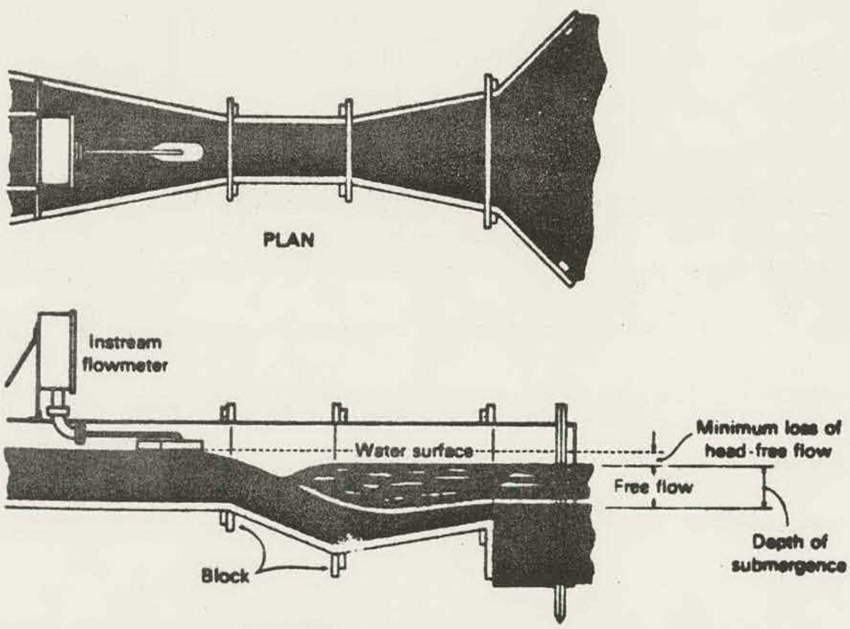
The Kennison Nozzle installation, pictured in Figure 8-1, makes use of the Venturi principle but typically employs a nozzle, instead of the Venturi tube, inserted in (or at the end of) a pipe. The throat of the nozzle is considerably larger in diameter than the throat of a Venturi tube sized for the same range of flow, and the resulting total loss of head is approximately the same. These nozzles can be used on pipes varying in size from 6 to 36 inches in diameter and can be used for metering flows up to 18 million gallons per day (MGD).[21]



Kennison flow nozzle installation [from BIF].

SOURCE: <u>Wastewater Engineering</u>, Metcalf and Eddy, Inc.[21]

FIGURE 8-1



Schematic of Parshall flume metering installation [from Fischer & Porter].

SOURCE: <u>Wastewater Engineering</u>, Metcalf and Eddy, Inc. [21]

FIGURE 8-2

The Parshall Flume installation, pictured in Figure 8-2, utilizes the Venturi principle for the measurement of water flow in open channels. Because the throat width of the flume is constant, the discharge can be obtained from a single upstream measurement of depth.[21]

Temporary flow measurement methods can be utilized where necessary. A simple and accurate method of flow measurement in open channels is the use of one of several weir configurations: rectangular, triangular (or V-notched), or trapezoidal. The rate of discharge over a weir can be directly related to the head of the discharge over the weir by the general equation:

$$Q_1 = ch^n$$

Where

Q_1 = free discharge, cfs

c = constant, defined for each weir configuration

h = measured head, ft.

n = exponent, defined for each particular weir

For the widely used triangular (V-notch) weir the equation is defined as follows:

$$Q = 2.5\ h^{5/2}$$

SOURCE: Wastewater Engineering, Metcalf and Eddy, Inc.[21]

A preferred method of flow measurement in closed pipes is the use of an orifice plate and the attendant differential pressure instrumentation. In accordance with Torricelli's theorem that the velocity of flow through an orifice is equal to the velocity acquired by a freely falling body in a space corresponding to the head over the orifice, the discharge through an orifice is :

$$Q = cAV = cA\sqrt{2gH}$$

Where

Q	=	discharge, cfs
c	=	constant, depending on orifice configuration
A	=	orifice area, sq. ft.
V	=	velocity, fps
g	=	acceleration due to gravity, ft/sec^2
H	=	heat, ft.

SOURCE: Wastewater Engineering, Metcalf and Eddy, Inc.[21]

An acceptable alternative method to orifice plates would be the use of an Annubar-type device which also relates flow to the differential pressure across the device.

Depending on the specific point of application, the above mentioned methods may be impractical or impossible, in which case reasonable estimates may be developed through computation. This method involves applying standard flow formulas to field-measured data such as depth of flow and slope of discharge pipes. In addition, a coefficient of roughness must be selected. The method is, at best, an approximation dependent upon the steadiness of flow at the time of observation.[21] Nevertheless, this method may serve in place of, or as a "ballpark" verification of, more sophisticated methods.

8.2.3 SUMMARY

With the above mentioned data in hand, a detailed estimate of the quantities of thermal energy available at the system-to-treatment plant interface can be developed. This data will also form the basis for the performance evaluation of the overall HP-ICES scheme.

8.3 CENTRAL STATION HEAT PUMP EQUIPMENT

Once a data base is established for the conditions and amounts of thermal energy that are available at the source(s), a comprehensive performance analysis of the HP-WHR scheme can be undertaken. Obviously, the key component in this scheme is the Central Station Heat Pump Equipment and its performance as the interface between the energy source(s) and the distribution medium. Since performance data for the Central Station equipment will likely not be available for the specific range of conditions of the thermal energy source, a significant effort should be directed toward developing this data to be used in the detailed performance analysis.

8.3.1 PERFORMANCE TESTING

Test procedures for Centrifugal Water-Chilling Packages are outlined in ARI Standard 550-77 (Air Conditioning and Refrigeration Institute)[54] and will be adopted for the performance testing to be done in conjunction with this analysis.

Testing should be performed in a laboratory environment, similar to that utilized for industry rating tests, where all variables may be controlled to the tolerances set forth in the ARI 550-77 standard. It may be desirable to contract an outside agency to perform the actual testing.

At least four sets of performance tests should be performed; each set to reflect the specific initial conditions documented in one of the four baseline data tests. The following test data, relevant to the performance of the unit, will be taken:

A. Temperature of water entering cooler, °F

B. Temperature of water leaving cooler, °F

C. Chilled water flow rate, GPM or lb. per hr.

D. Temperature of entering condenser water, °F

E. Temperature leaving condenser water, °F

F. Condenser water flow rate, GPM or lb. per hr.

G. Power input to compressor motor, KW

From this data, a heat balance will be performed (to the prescribed closure of 7.5 per cent) to substantiate the results of the test. With validated test results, an accurate C.O.P. can be derived for each set of baseline and operating conditions. Likewise, actual capacity in tons can be derived. General formulas for the heat balance, C.O.P., and capacity are included in Appendix D.

8.3.2 WATER QUALITY

A significant factor that will affect the performance of the Central Station equipment is the quality of water available from a sewage treatment system to be utilized as the thermal energy source. Due to the presence of suspended solids in the form of BOD (or decomposable organic matter) in raw sewage, there is the possibility of methane gas generation and retention in some segments of the HP-WHR system. The most significant problem involved in utilizing sewage as the thermal energy source, though, is that of an increased fouling factor on the heat exchanger surfaces.

Methane gas generation occurs in a sewage treatment system when the organic suspended solids (BOD), settled and collected at various points in the system, are reduced by biochemical digestion (typically in a covered, anaerobic sludge digestor). Such sludge digestors may utilize portions of the methane gas for temperature regulation within the digestor, while other designs may simply dispose of the excess gas with an open-flame flare. In order to avoid unnecessary methane generation in the Central Station equipment and its associated piping, the organic suspended solid content in the portion of the wastewater circulated through the Central Station equipment heat exchanger will be minimized by extracting it from the sewage treatment system at points following sludge removal (providing that the wastewater at these points is of sufficient temperature). It is expected that the remaining organic suspended solids will be minimal and that any appreciable build-up in the heat exchanger can be avoided by optimizing the wastewater flow rate through the Central Station equipment.

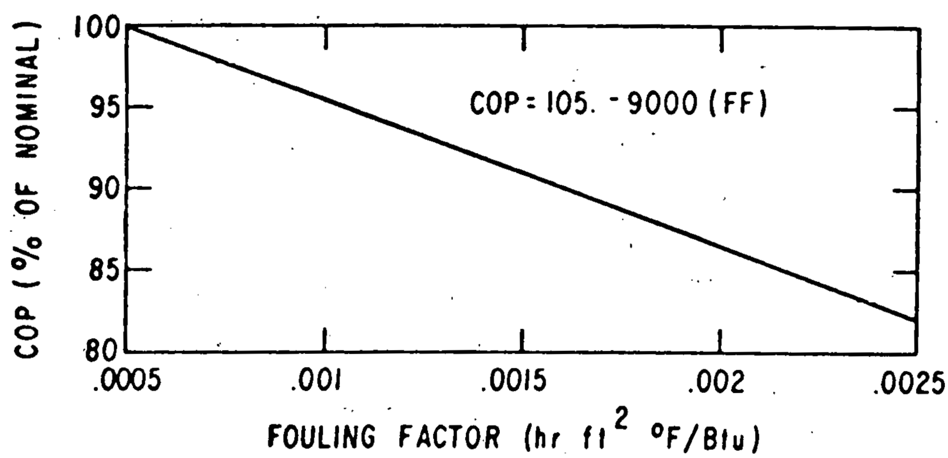
The design of most treatment plants will allow existing, open-type clarifiers to serve as accumulators for the thermal supply medium. Thus, the need for an enclosed, thermal supply medium storage vessel, along with the potential for methane gas generation/retention in the HP-WHR system, is avoided. If, in fact, such a vessel is required by the design of a particular sewage treatment system, provisions for methane disposal may be included in the design in accordance with standard wastewater engineering practice. In regard to clarifier performance as utilized in the HP-WHR scheme, standard manufacturers data is available in reference to storage capacity (volume), surface area, surface settling rates, maximum and minimum retention times, etc. Temperature and flow data will be developed at the clarifier as explained in sections 8.2.1 and 8.2.2 if it, in fact, proves to be the most advantageous tie-in point for the Central Station equipment.

Increased fouling factors on the Central Station equipment heat exchanger surfaces will prove to be the most troublesome problem associated with water quality in the HP-WHR system. The ARI standard for water-chiller testing assumes a fluid with a fouling factor of .0005 (also widely used as the rule-of-thumb in industry). However, water quality data surveyed from various sewage treatment plants indicates that fouling factors may range to several times that figure. The fouling factor for water samples taken from the Chicago Sanitary Canal has been quantified at .006 to .008. (a table of typical fouling factors from Reference 7 is reproduced in Appendix D as Table D-1.)

The first step in defining a fouling factor for the quality of wastewater to be circulated through the Central Station equipment heat exchangers should be collecting wastewater samples, representative of the water that will be circulated, at the point in the sewage treatment process where the thermal energy supply medium will be extracted. A full water quality analysis should be performed by a qualified testing agency or laboratory, and the analysis compared to typical analyses for the types of fluids for which a fouling factor has been defined. This comparison should give a "ball-park figure" for the expected fouling factors.

In order to obtain better defined data, though, more testing would be required. An additional performance test may be desirable in which the average baseline conditions are duplicated and the water supply to the evaporator simulates the water quality of the thermal energy source. Such a test would reveal the immediate discernable effects (if any) of water quality on C.O.P. and give an indirect indication of the fouling factor. (The decrease in performance resulting from a higher fouling factor is roughly equivalent to raising the leaving condenser water temperature 2°F for every .0005 increase in the fouling factor.) Figure 8-3 shows the estimated percentage reduction of the nominal C.O.P. at fouling factors ranging from .0005 to .0025 for either the evaporator or condenser. Continued testing under the above specified conditions will quantify the long term effects on the Central Station equipment C.O.P. and allow calculation of the actual fouling factor. (Formulas and definitions concerning fouling factors are included in Appendix D.) Moderately high fouling factors (.0005 to .0025) may dictate simply that a larger evaporator heat exchanger be specified for the Central Station equipment. Very high fouling factors (over .0025) may require that a mechanical cleaning system for the heat exchanger be specified. Several methods are acceptable, ranging from manual inspection and flushing, to reverse-flow activated brush scrubbers, to a continuous cycle, spherical rubber scrubber system such as an Amertap.

Once a range of values for the expected fouling factor of the wastewater is determined, a final equipment test (or set of tests), incorporating the appropriate mechanical cleaning system(s), is recommended to develop the baseline information necessary to evaluate the performance of these systems in this particular application. From this performance evaluation, general guidelines for design specifications and operating procedures in the HP-WHR system application can be developed.



REDUCTION OF PACKAGE CHILLER C.O.P.
AS A FUNCTION OF FOULING FACTOR

SOURCE: Central Cooling - Compressive Chillers, Christian[5]

FIGURE 8-3

8.4 END-USER HEAT PUMP EQUIPMENT

The ultimate community acceptance of this HP-ICES scheme will depend on the energy efficiency of the heat pump equipment in use in homes and places of business. This scheme offers the initial advantage of utilizing in the end-user segment water source or hydronic heat pumps. However, it is expected that the performance of the end-user equipment will be even more improved by the effect of cascading the heat gain of the hydronic heat pump with that of the Central Station equipment. Obviously, the performance of the end-user equipment will depend upon the conditions in the distribution medium supplied by the Central Station equipment. Again it is expected that application ratings for this specific set of conditions will not be available. To complete the data needed for the detailed performance analysis, tests on the end-user heat pump equipment must be performed.

8.4.1 PERFORMANCE TESTING

Test procedures for Water-source heat pumps are outlined in ARI standard 320-76 (Air-Conditioning and Refrigeration Institute)[53] and will be adopted for the performance testing to be done in conjunction with this analysis.

As with the components selected for use at the Central Station, end-user equipment should be tested in a laboratory environment where all variables may be controlled to the tolerances set forth in the ARI 320-76 standard. Again, it may be desirable to contact an outside agency to perform the actual testing. Four sets of performance tests should be performed; each set to reflect the distribution medium conditions documented in one of the four Central Station equipment tests. Test conditions set forth in the procedures for the standard rating test should be adhered to, with the exception of the water temperature entering the refrigerant-to-water heat exchanger, which will correspond to the output temperature of the distribution medium from the Central Station equipment.

From the performance data gathered in these tests, accurate C.O.P.'s (or Energy Efficiency Ratios, EER's, as appropriate) can be derived for each set of seasonal operating conditions.

8.4.2 PROCESS HEAT RECLAIM EQUIPMENT

In addition to residential and commercial space heating and cooling needs, it is expected that portions of the thermal load from this HP-ICES may be applied in industrial processes. However, due to the wide variety of heat exchangers that may be utilized in industry, it is impractical to outline a specific performance testing procedure for each. Instead, procedures will be outlined and performance data developed as specific applications (and the required equipment) are identified.

8.5 SUMMARY

With all pertinent performance data documented, typical operating conditions can be predicted and analyzed according to the procedures developed in Chapter 3. The result of the comprehensive performance analysis will be an overall system coefficient of performance (C.O.P.) which can be used in economic analyses, energy effectiveness studies, and other similar performance comparisons with other systems. Satisfactory performance, considering all the parameters discussed in previous sections, will allow the utilization in a community system of heretofore wasted heat in sufficient quantities to make such an HP-WHR system practicable.

The ultimate success of this HP-ICES scheme will depend on its economic feasibility, either now or in the foreseeable future. Quite obviously, the economic feasibility will directly relate to optimizing the overall system C.O.P. for the specific operating conditions for each site of application.

8.6 METHODOLOGY

Since the performance testing of the heat pumps (either Central Station or end-user) dictates a change in magnitude of only one parameter, temperature of the thermal source, over the standard test procedures, the test methodology set forth in the applicable ARI standards will be adopted for this performance analysis. Specific standards that apply are as follows:

ARI 320-76	Standard for Water-Source Heat Pumps[53]
ARI 550-77	Standard for Centrifugal Water-Chilled Packages[54]
ASHRAE 14-67	Methods of Testing for Rating Mechanical[51] Condensing Units
ASHRAE 37-69	Methods of Testing for Rating Unitary Air[52] Conditioning and Heat Pump Equipment

Additionally, recommendations set forth in Chapter 12 of the ASHRAE Handbook of Fundamentals, "Measurement and Instruments",[50] will be followed in the test procedures.

Unfortunately, it is beyond the scope of this discussion to outline detailed test procedures for each specific piece of equipment that will be utilized in this HP-ICES scheme. Accordingly, detailed test procedures will be developed at the outset of the actual testing program. A procedure for testing and analysis was developed in "Demonstration of Building Heating with a Heat Pump Using Thermal Effluent" by Peter W. Sector,[33] and will be used as a model in developing the detailed test procedures for this program.

9.0 ANALYSES DESCRIPTION

9.1 INTRODUCTION

The specific assumptions and methodology pertaining to the various analyses made in this report are described in the body of the report within each section. However, this section provides additional insight into the assumptions and calculations used in the Potential Applications Expected Economics and Performance sections of the report.

The tendency in most cases was to be conservative in the analyses and make what was considered to be reasonable assumptions. Practical engineering approximations and common industry practices were used where necessary. It was the intent of the authors to provide engineering calculations which would be familiar to the practicing engineering professional who might wish to utilize the concept in a specific application.

9.2 POTENTIAL APPLICATIONS

The potential applications section of the report is concerned with estimating the energy saving potential of the HP-WHR scheme on a national basis. The estimate was developed using the best wastewater effluent flow statistics available. Unfortunately, the wastewater effluent statistics are based on domestic water consumption figures and fail to include numerous thermally attractive sources such as industrial discharges directly to surface water streams and power plant waste heat.

The application of the HP-WHR concept is also dependent on population (user) density, since the distribution system will apparently be the dominant capital cost. High user densities will tend to increase the heat recovery application factor and enhance the energy effectiveness and thus the desirability of the scheme.

In order to develop a more accurate picture of potential applications it will be necessary to develop a new primary data-base correlating waste-water effluents, thermal quality and density of energy users.

9.3 EXPECTED ECONOMICS

The expected economics of the HP-WHR scheme is ultimately the determining factor in whether or not the scheme will be pursued seriously and implemented. Unfortunately, generalized economic performance is not accurate enough for a thorough evaluation of the system's performance in all cases.

It is possible to make some specific observations about the effects certain economic assumptions impose on the overall attractiveness of the system. These areas are discussed here.

First, the capital cost of the central plant and distribution system seriously impact the cost of energy supplied through thermal transport media. It appears that the distribution system is by far the most important of these capital costs. For that reason, areas with high densities of thermal users will prove more economical.

Financing is another area critical to the cost of the capital equipment. The interest rate and mortgage time could penalize the system during the initial years of operation. Of course, as time goes on, the fixed costs will remain constant and will become less important as rising energy costs and inflation affect variable operating costs.

Escalation rates for various fuel sources and the electric rate differential between residential and large municipal customers has an impact on when the HP-WHR scheme will become competitive with conventional energy sources such as natural gas. This study does not specifically assume escalation rates for the various fuels, but instead, considers that risk to be accounted for in the present worth factor of the present worth analysis.

The present worth factor is another variable affecting the outcome of the analysis. In the case of the factor and that of the fuel and inflation escalation estimates, an accurate analysis would require consideration of the best site-specific information available at the time the analysis is made.

Finally, the relative cost of alternative heating and cooling systems for the residential customer will influence the HP-WHR system reception by the ultimate consumer. Future community developments will certainly be more receptive to such a scheme than would retrofit candidates since the incremental cost over standard alternatives associated with the heat pump system will be small.

9.4 PERFORMANCE ANALYSIS

The objective of our performance analysis is to determine the energy requirements of the HP-WHR system delivering a given amount of heating or cooling effect for a community when using wastewater effluent as a heat source or sink. Additionally, the HP-WHR system will be compared with conventional systems serving the same function. The HP-WHR is modelled as shown in Figures 3-1 and 3-2 for the heating and cooling modes. As indicated in the figures, the system's principal components are: a central heat pump, distribution pipe lines, water pumps, storage reservoirs and end-user heat pumps. In actual practice, the system may consist of a battery of central heat pumps, one or more storage reservoirs, booster pumps, etc. For purposes of our analysis, we have included one end-user heat pump and a single central heat pump unit. The total energy required to deliver a unit of heating or cooling affect to one residential end-user involves proportional energy inputs to the central heat pump unit, the end-user unit and the associated pumping energy involved in water flow through evaporators, condensers, distribution pipe lines and provision for thermal losses. Each component will be discussed separately and a procedure for calculating the total energy (electrical or mechanical) input to the system is outlined utilizing available performance data. Sample calculations are included in the appendices.

9.4.1 SELECTION OF DESIGN LOADS

Two geographical regions in the United States were selected for the study of the HP-WHR system. The details of the houses and the design loads are given below. The selection of the design heating and cooling loads was based on the information presented in Reference 27.

Location: Northwest Central Region

House 1:
Design heating load = 55,000 Btuh
Design outside temperature = 20°F (heating)
Design inside temperature = 75°F
(65°F) degree-days = 8300
Design cooling loads = 11,500
Design outside air temperature = 89°F (cooling)
Design inside air temperature = 78°F (cooling)
Summer equivalent full load cooling hrs = 600

Location: South Atlantic Region

House 2:
Design heating load = 31,000 Btuh
Design outside temperature = 22°F (heating)
Design inside temperature = 75°F
(65°F) degree-days = 2920
Design cooling loads = 25,150 Btu/hr
Design outside air temperature = 95°F (cooling)
Design inside air temperature = 78°F (cooling)
Summer equivalent full load cooling hrs = 1000

9.4.2 ASSUMPTIONS

a) The basic vapor compression cycle as described in section 9.4.3 is used as a basis for the definition of terms used in the analysis. The heat pumps that are used in this analysis are selected from manufacturers' models listed for study in References 5,6, and 7, sized to suit the design loads used in our analysis. Brand names of the models are omitted. Heat pumps and their characteristics are presented in section 9.4.9.

b) The water distribution lines are burried approximately six feet underground to minimize heat loss. Under these conditions, the temperature change of the water due to heat loss or gain would be very small since the operating water temperatures are low. This assumption is justified by an example in section 9.4.8.

c) The thermal storage is designed so that the heat losses or gains would be negligible. An approximate analysis of the temperature change of water in the reservoir is discussed in section 9.4.8.

d) The thermal properties of the clarified effluent are assumed to be the same as those of water.

3) Apart from the heat exchange between the refrigerant and the heat transfer medium (water-to-air), heat losses to the surroundings are negligible.

Other assumptions will be stated as they are made in the course of the analysis and calculations.

9.4.3 VAPOR COMPRESSION CYCLE

The heat pump used in the anlaysis of the HP-WHR system is based on the vapor compression refrigeration cycle (Figure 9-1). The ideal cycle consists of the following processes:

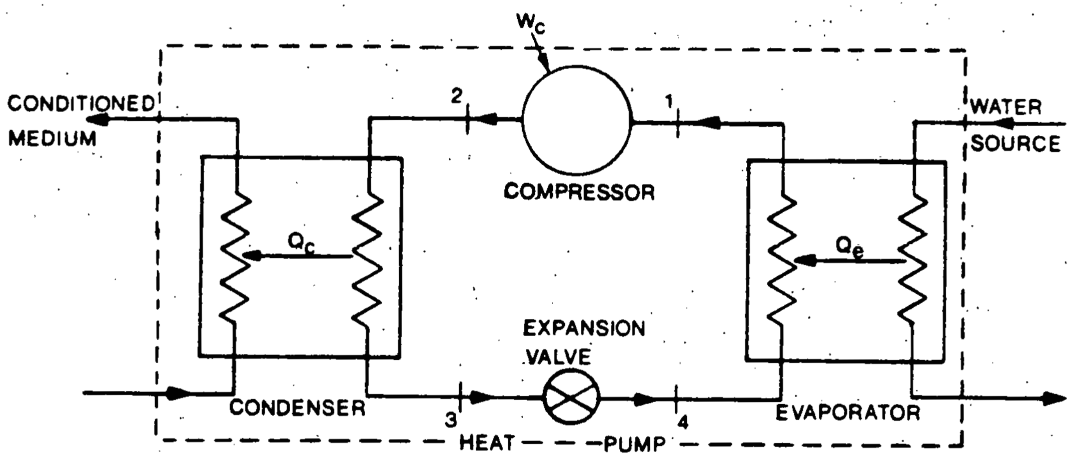
1.) isentropic compression of the vapor

2.) constant pressure heat rejection from the refrigerant in the condenser to the surroundings

3.) irreversible adiabatic expansion (throttling) in the expansion valve

4.) constant pressure heat absorption by the refrigerant in the evaporator from the surroundings

In actual cycles, there are losses due to the pressure drops in the flow tubes, heat loss from the tubes, inefficiency of the compression process, etc.



BASIC VAPOR COMPRESSION CYCLE

FIG. 9–1

The coefficient of performance of the heat pump is given by

$$\underset{\text{(ideal)}}{\text{COP}} = \frac{Q_c}{\text{Energy input}} = \frac{h_2-h_3}{h_2-h_1} \qquad (9\text{-}1)$$

$$\underset{\text{(actual)}}{\text{COP}} = \frac{Q_c}{E_{th}} \qquad (9\text{-}2)$$

Where E_{th} is the total energy input to the unit including motor losses and Q_c is the heat rejected in the condenser.

The coefficient of performance of the chiller (refrigerator) is given by

$$\underset{\text{(refrigerator or chiller)}}{\text{COP}} = \frac{Q_c}{E_{th}} \qquad (9\text{-}3)$$

Where Q_c is called the refrigerating effect or cooling.

Heat Rejection Factor (HRF):

This is the ratio of heat exchanged in the condenser to that exchanged in the evaporator and it is related to the coefficient of performance COP as follows. Using the development in Reference 49, we have

$$COP = \frac{[(HRF)E]}{[HRF-1]} \tag{9-4}$$

$$HRF = \frac{(COP)}{(COP-E)} \tag{9-5}$$

Where

E is 1.0 minus the proportion of input power not delivered to the condenser, expressed as a decimal, and its value depends on the manner in which the compressor is cooled. Throughout our analysis we have assumed that E = 0.92.

Also it can be shown that

$$COP_{heat\ pump} = COP_{refrigerator} + E \tag{9-6}$$

9.4.4 CALCULATION PROCEDURE

Calculation procedure for the energy requirement of the HP-WHR system to deliver a given amount of heating effect to the end-user:

Heating mode - Reference is made to Figure 3-1 and it is assumed that the heat extracted from the distribution water flowing through the evaporator of the end-user heat pump equals the heat rejected to the water in the condenser of the central heat pump.

The total energy requirements include the work input to the compressors of the end-user heat pump, the central heat pump and the energy required to overcome the distribution losses.

End-user heat pump:

$$Q_{ce} = \text{heating load}$$

work input to end-user heat pump:

$$W_{ce} = \frac{Q_{ce}}{COP_{end\text{-}user}} \tag{9-7}$$

Heat absorbed in the evaporator:

$$Q_{ee} = \frac{Q_{ce}}{HRF} \tag{9-8}$$

Where

HRF is called the heat rejection factor given by

$$HRF = \frac{COP}{(COP-E)}$$

Where

COP is the Coefficient of Performance of the heat pump.

$$Q_e = \dot{m}_w \, c_p \, (T_{e1}-T_{e2}) \tag{9-9}$$

Where

T_{e1} and T_{e2} are the entering and leaving temperatures of water flowing through the evaporator, $\dot{m}_w$ the mass rate of flow of water and c_p its specific heat.

Using the above equation the mass rate of flow of water is calculated and used for the computation of pressure drops as explained in the analysis section.

Central heat pump:

$$\underset{\text{(central heat pump)}}{Q_c} = \underset{\text{(end-user heat pump)}}{Q_e}$$

Where Q_c = heat rejected in the central station heat pump

Q_e = heat absorbed in the end-user heat pump

Work input to compressor of central heat pump is defined by the following equation:

$$W_c = \frac{Q_c\text{(central heat pump)}}{\text{COP (central heat pump)}} \tag{9-10}$$

Total energy input to compressor:

$$\Sigma W_c = \underset{\text{(central heat pump)}}{W_c} + \underset{\text{(end-user heat pump)}}{W_c} \tag{9-11}$$

$$\Sigma W_{losses} = W_p + \text{losses unaccounted}$$

$$= 0.05\ \Sigma W_c \tag{9-12}$$

Note: In the section on analysis it is shown that the pump energy associated with one end-user is approximately 5% of the total energy input to the compressors. However, as indicated in the sample calculations, the reduction in system COP would be 4-5% if the losses increased to 10%.

Total energy input to the HP-WHR system is defined by the following equation:

$$E_{system} = \Sigma W_c + \Sigma W_{losses}$$

and therefore:

$$COP_{system} = \frac{\text{(heating load)}}{E_{system}} \tag{9-13}$$

A similar calculation procedure is used for the cooling mode.

9.4.5 ANNUAL ENERGY CONSUMPTION

The calculation of the annual energy consumption of the HP-WHR system to deliver a given amount of heating or cooling effect involves the use of the part load performance data on the end-user heat pump (water-to-air), the central unit (water-to-water), the pumps and information on the transient performance of the thermal storage reservoir. Information on the part load performance of water-to-water compressive chiller is available (Reference 7) but no precise data on part load performance of water-to-air heat pumps is reported in technical literature at the present time. The end-user consumes about twice the energy consumed by the central heat pump in the HP-WHR system.

As discussed in section 3.5.2, heat pumps suffer dynamic efficiency losses due to part load operation during the year. Assuming that the dynamic efficiency losses of the HP-WHR system are similar to those of conventional heat pumps, its annual steady state energy consumption is adjusted as follows:

Annual energy consumption = steady state energy consumption x F

Where the factor F allows for the dynamic efficiency losses. According to Reference 10 the values of F are:

Heating season	F	=	1.096
Cooling season	F	=	1.164
Annual	F	=	1.118

Conventional systems: The procedure explained in the chapter on Energy Estimating Methods in Reference 49, is used as the basis for computing the annual energy consumption of gas warm air heating systems. The annual energy consumption by the air-to-air heat pump was calculated by using the seasonal formance factor obtained from Reference 5.

and the heat extracted from the distribution system,

$$Q_E = \frac{Q_H}{HRF} \tag{9-15}$$

Where HRF heat rejection factor.

For the heat pump operating in the cooling mode

$$W_{c2} = \frac{Q_L}{COP_{cooling}} = \frac{Q_L}{(COP_{heating} - E)} \tag{9-16}$$

and the heat rejected $Q_c = Q_L \times HRF$ (9-17)

If the heat rejected Q_c by end-user pump 2 (cooling mode unit) equals Q_E, the heat extracted from the distribution water by end-user pump 1, then there will be no net change in the energy of the distribution water and no work will be required by the central heat pump.

That is

$$Q_L\ HRF = \frac{Q_H}{HRF_2} \tag{9-18}$$

For the special case $HRF_1 = HRF_2 = HRF$:

$$\frac{Q_H}{Q_L} = HRF^2 \tag{9-19}$$

For other loads, the net energy (heat) added to or extracted from the effluent will depend on the net demand(heating $\pm$ cooling) of the load from the community.

EXAMPLE: Consider the case of two heat pumps whose COP's are the same, that is 2.96, and E = 0.92, then

$$HRF_1 = \frac{2.96}{2.96-.92} = 1.456 = HRF_2$$

$$\frac{Q_H}{Q_L} = (1.456) \times (1.456) = 2.12$$

$HRF = 1.456$

$COP_{\text{heating mode}} = 2.96$, $COP_{\text{cooling mode}} = 2.04$

1) Q_H = 55,000 Btuh heating

2) Q_L = 25,943 Btuh cooling

$$W_{c1} = \frac{55000}{2.96} = 18581$$

$$W_{c2} = \frac{25943}{2.04} = 12717$$

$$\text{TOTAL } \Sigma W = 31298 \text{ Btuh}$$

and heat delivered (in the condenser) to space = 37613 x HRF_2 = 55000

Total load = 55000 + 25943 = 80943

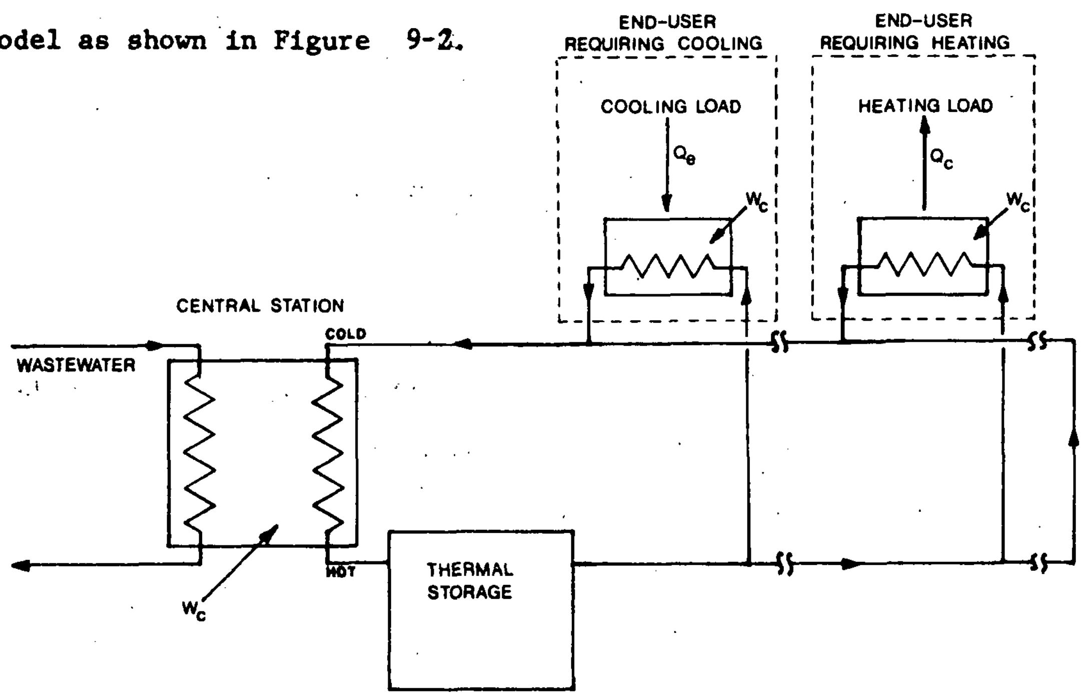
$$\text{Total } \Sigma W_c = 31,298 + 0.05\ \Sigma W_c = 32862$$

$$\text{Net COP system} = \frac{80,943}{32,862} = 2.463$$

9.4.6 SIMULTANEOUS HEATING AND COOLING ANALYSIS

The need for simultaneous heating and cooling will probably arise during the early parts of Fall and Spring seasons when the internal heat gains in some buildings exceed the heat loss. The total energy requirements of HP-WHR system in such a situation will depend on the distribution of heating and cooling loads and the water distribution system will have to be designed to handle the simultaneous demands of heating and cooling loads.

The energy requirements of the HP-WHR system can be calculated by assuming the model as shown in Figure 9-2.



MODEL FOR SIMULTANEOUS HEATING AND COOLING

FIG. 9-2

In this system, the heat rejected by the heat pumps acting in the cooling mode can be used as a heat source for the heat pumps acting in the heating mode, and the net effect is a reduction in the work of the compressor of the central unit. As an example, let Q_H and Q_L be the heating and cooling loads respectively of and in heat pumps 1 and 2. Using the equations in section 9.4.3, we have

$$\underset{\text{end-user pump}}{W_{c1}} = \frac{Q_H}{COP} \qquad (9\text{-}14)$$

For a mass rate of flow of 3832.75 $\frac{lbm}{hr}$ and a total head loss,

$\Delta H = (12 + 25 + 25 + 22) = 84'$, the work of the pump assuming an efficiency of 50% becomes 827.6 $\frac{Btu}{hr}$. In the sample calculations for the heating mode the pumping loss is shown to be 3.45% of the energy input to the compressors. In order to allow for unaccounted losses, the thermal and distribution losses are assumed to be 5% of the total energy input to the compressors in all performance calculations. However, increasing the distribution losses to 10% would reduce the system COP by about 4-5%.

The work required to pressurize the water by 100 psi and the consequent temperature change is discussed in Appendix D.

Heat Loss and Temperature Drop in the Distribution Piping:

The heat loss from pipes buried underground depends on the thermal conductivity of the soil, the pipe size and material, the ground temperature and inside water flow conditions. The thermal conductivity K of the soil varies widely depending on whether the soil is dry and sandy, wet and clay, etc., and has to be investigated for the particular area under consideration. The thermal conductivity values taken from Reference 46 are given in the TABLE 9-1 and, as suggested by the author, to be used only when precise data are unavailable from a site-specific soil survey.

Heat Loss from Pipe:

The heat loss per foot length of buried pipe is given by Reference 46 and 7 is used as a guide:

$$Q = K_p(T_p - T_g), \frac{Btu}{hr\text{-}ft.} \qquad (9\text{-}21)$$

Where

K_p = heat transfer factor, $\frac{Btu}{hr.ft.F}$

TABLE 9-1

THERMAL CONDUCTIVITY OF SOIL

$$k_s^* - \frac{\text{Btu - in}}{\text{hr-ft}^2\text{-°F}}$$

Moisture Content	Type Soil: Sandy	Silty	Clay
Dry	2	1	1
Medium	13	9	7
Wet	15	15	15

*These data are approximate values and should be used only when precise data are unavailable from the site.

SOURCE: Technical Guidelines for Energy Conservation, National Bureau of Standards.[46]

9.4.7 DISTRIBUTION SYSTEM LOSSES

Distribution losses consist of pressure losses due to friction in the pipe lines, thermal losses in pipes and the storage. Since the energy input to overcome these losses is part of the total energy input to the system, the order of magnitude of such losses is discussed in this section.

Frictional loss in a pipe:

Frictional losses in a pipe depend on the diameter of the pipe, the velocity of water and the roughness of pipe. For a velocity of in the range of 6 ft/sec, and a friction factor value of f = 0.04, the head loss for a 1000 ft. length of pipe is given by

$$H = f \frac{L}{D} \left(\frac{V^2}{2g}\right) = 0.04 \times \frac{1000}{1} \times \frac{6^2}{2(32.2)} = 22'$$

It is common practice to allow a frictional loss of about 25 ft. per 1000 ft. length of pipe.

Pump work:

The pressure drop due to flow of water in the condenser and evaporator tubes is usually of the order of 12 ft. to 25 ft. (Reference 5 & 7).

The work of the pump W required to overcome a total frictional head loss ΔH distribution pipeline, the evaporator and the condenser tubes for a mass rate of flow of $\dot{m}$ lbm/hr per one end-user can be estimated as follows:

$$W = \frac{\dot{m}\,\Delta H}{778 \times \eta_p} \quad \text{Btu/hr} \tag{9-20}$$

Where η_p is the efficiency of the pump. In addition, it may be necessary to pressurize the water for distribution to residences at higher elevations.

T_p = temperature of fluid in the pipe

T_g = undisturbed earth temperature

K_p is given by

$$R = \frac{1}{K_p} = \frac{1}{C} + \frac{6}{\pi K_s} \ln \{\frac{d}{r} + (\frac{d}{r})^2 - 1\} \qquad (9\text{-}22)$$

C = thermal conductance of pipe, $\frac{\text{Btu}}{\text{hr-ft}^2\text{°F}}$

When

d = distance of ground surface to center of pipe

r_o = outside radius of pipe, inches

K_s = thermal conductivity of soil, Btu-inch

and

$$\frac{1}{C} = \frac{1}{h_i 2\pi r_i} + \frac{6}{\pi K_w} \ln \frac{r_o}{r_i} \qquad (9\text{-}23)$$

Where

r_i = inside radius of pipe

K_w = thermal conductivity of pipe material $\frac{\text{Btu-inch}}{\text{hr-ft}^2\text{-°F}}$

h_i = surface conductance at the inside surface of pipe (water side), $\frac{\text{Btu}}{\text{hr-ft}^2\text{-°F}}$

Additional thermal resistance can be added to the pipe if insulation is used.

K_p factors can be calculated for various other systems, such as two pipe running in a tunnel, a single pipe with different kinds of insulations, are presented in the form of graphs in reference 46. Heat loss factors can also be obtained from manufacturers.

For the case of a single plastic bare pipe (O.D. = 12"), buried six feet deep in the ground at 55°F (medium dry, sand and clay soil), the K_p factor was found to be 1.422. The heat loss per foot length of pipe for water at 80°F was found to be 35.5 $\frac{Btu}{hr.ft.}$. For dry and sandy soil, the heat loss would be 9.5 $\frac{Btu}{hr.ft}$

Temperature drop in pipe lines:

$$\text{Heat loss } Q = \dot{m}\, C_{pw}(T_2 - T_1) \qquad (9\text{-}24)$$

Where

$\dot{m}$ is the mass rate of flow of water and is given by

$$\dot{m} = \rho A_c \bar{V}$$

T_1, T_2 = inlet and inlet temperatures of water flowing inside a pipe °F
A_c = cross sectional area of flow, ft.^{2}
ρ = density of water, $\frac{lbm}{ft^3}$
$\bar{V}$ = velocity of water in pipe, $\frac{ft}{hr}$

Using a heat loss of (35.5 x 5280) $\frac{Btu}{hr}$ for 1 mile length pipe and a flow velocity of 6 ft/sec, the temperature drop $(T_1 - T_2)$ is found to be less than a degree. This will be offset by the temperature rise due to pressurization in the pipe and frictional losses. The temperature rise due to a pressure increase of 100 psi is shown in the section on pump analysis.

Thermal storage:

The size of the thermal storage tank should have sufficient capacity to maintain a constant supply of water for several hours. It should have sufficient thickness and should be dimensioned for minimum surface area to minimize heat losses. The rate of change of temperature of water in the reservoir at any instant can be estimated by an energy balance. Fig. 9-3

$$\dot{Q} = \dot{m}\,(h_2 - h_1) + \frac{dE}{d\theta} \qquad (9\text{-}25)$$

Where

E = the internal energy of water in the reservoir
= $M\, C_v(T)$

(symbol legend cont'd)

Θ = time

$\dot{Q}$ = rate of heat loss from the storage reservoir and

C_v = specific heat of water = $1.0 \frac{Btu}{lbm}$

T = temperature of water

$\dot{m}$ = mass rate of flow

M = mass of water in the storage

$\dot{m}(h_2-h_1)$ = total enthalpy change of water entering and leaving the reservoir.

The volume of the tank containing say, 5×10^6 gallons would be 6.67×10^5 ft^3. For minimum surface areas, the diameter and height would be:

$$D = 94.7 \text{ ft.} = \text{diameter}$$

$$L = 94.7 \text{ ft.} = \text{height}$$

The exposed surface area $A_s = 35217.6$ ft^2

The effect of the heat transfer Q on the rate of change of internal energy, $\frac{dE}{d\Theta}$ can be studied by neglecting $\dot{m}(h_2-h_1)$ for the given instant. Then

$$Q = UA(T-T_o) = M \frac{dE}{d\Theta} \tag{9-26}$$

Where

T_o is the temperature of surroundings

Assuming a "U" value of 0.3 $\frac{Btu}{hr.ft^2F}$ (based on inside area) for wall structure composed of thick concrete and exposed to outside air temperature of 0°F, and inside water temperature of 80°F, we have

$$\frac{dE}{d\Theta} = \frac{-0.3 \times 35217.6 \ (80-0)}{5 \times 10^6 \times 8.33 \times 1.0}$$

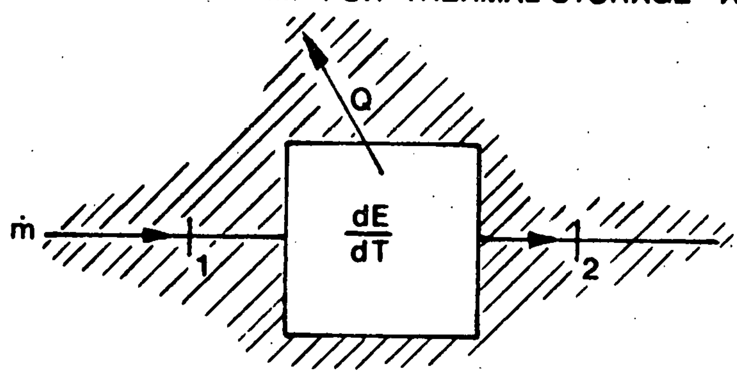
or

$$\frac{dT}{d\Theta} = -0.0203 \ \frac{°F}{hr} \quad \text{since } C_v = 1.0$$

This is the rate of change of temperature of water in the storage at the instant when its temperature is 80°F. This analysis assumes that there are no temperature gradients within the reservoir. However, in the actual system temperature gradients may exist.

FIG. 9-3

BLOCK DIAGRAM FOR THERMAL STORAGE ANALYSIS



9.4.8 HEAT PUMP PERFORMANCE DATA

The discussion presented here illustrates the procedure for computing COP's from the information on actual heat pumps given in References 5,6,7. The TABLES and figures for the performance of heat pumps are presented only for illustration. They do not cover the full range of data used in our calculations.

END-USER HEAT PUMPS - actual performance data on water-to-air end-user heat pump is obtained from the information given in References 5 and 7 in which the parameters such as heating capacity, cooling capacity, COP and EER are related to the entering water temperature by means of equations. Values of heating capacity and COP obtained by the use of these equations are shown in Figures 9-4 and 9-5. The COP values used for the cooling mode of the heat pump were converted from EER values given in References 6 and 7.

CENTRAL CHILLER - the performance of the central chiller is given by the relation (Reference 5)

$$\% \text{ of nominal power input} = 916.347 + 0.532633\,(X) - 0.000559686\,(X^2) + 0.0000230630\,(x^3)$$

$$- 32.7860\,(LCWT) + 0.378447\,(LCWT^2) - 0.00142857(LCWT^3) + \frac{1092.31}{LEWT} + \frac{2071.02}{LEWT^2} \qquad (9\text{-}27)$$

Where:

X = % of full load

LCWT = leaving condenser water temperature (°F)
LEWT = leaving evaporator temperature (°F)

which permits us to find the COP of the chiller for various values of temperatures of water or effluent entering and leaving the central chiller.

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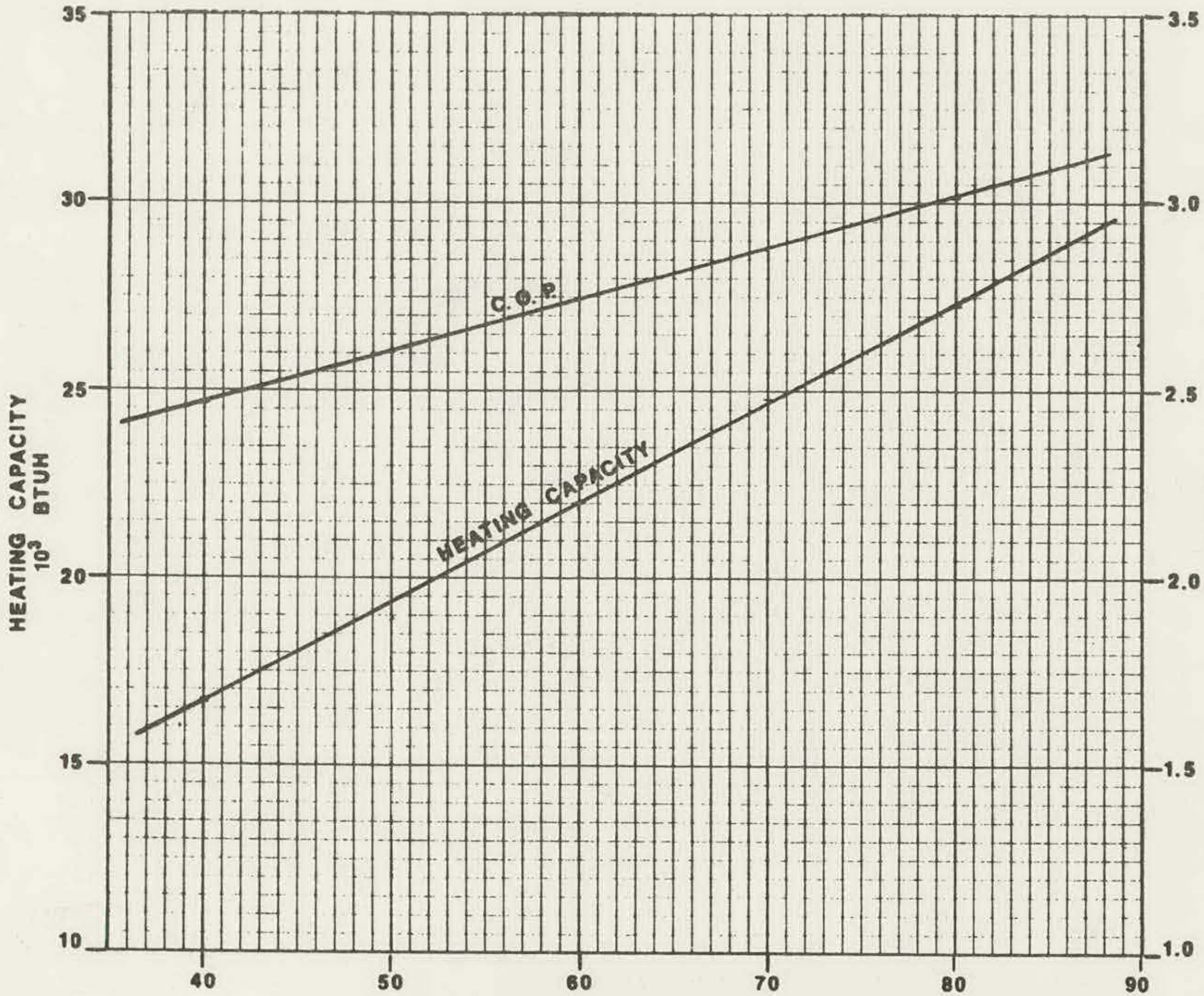
WATER-TO-AIR HEAT PUMP



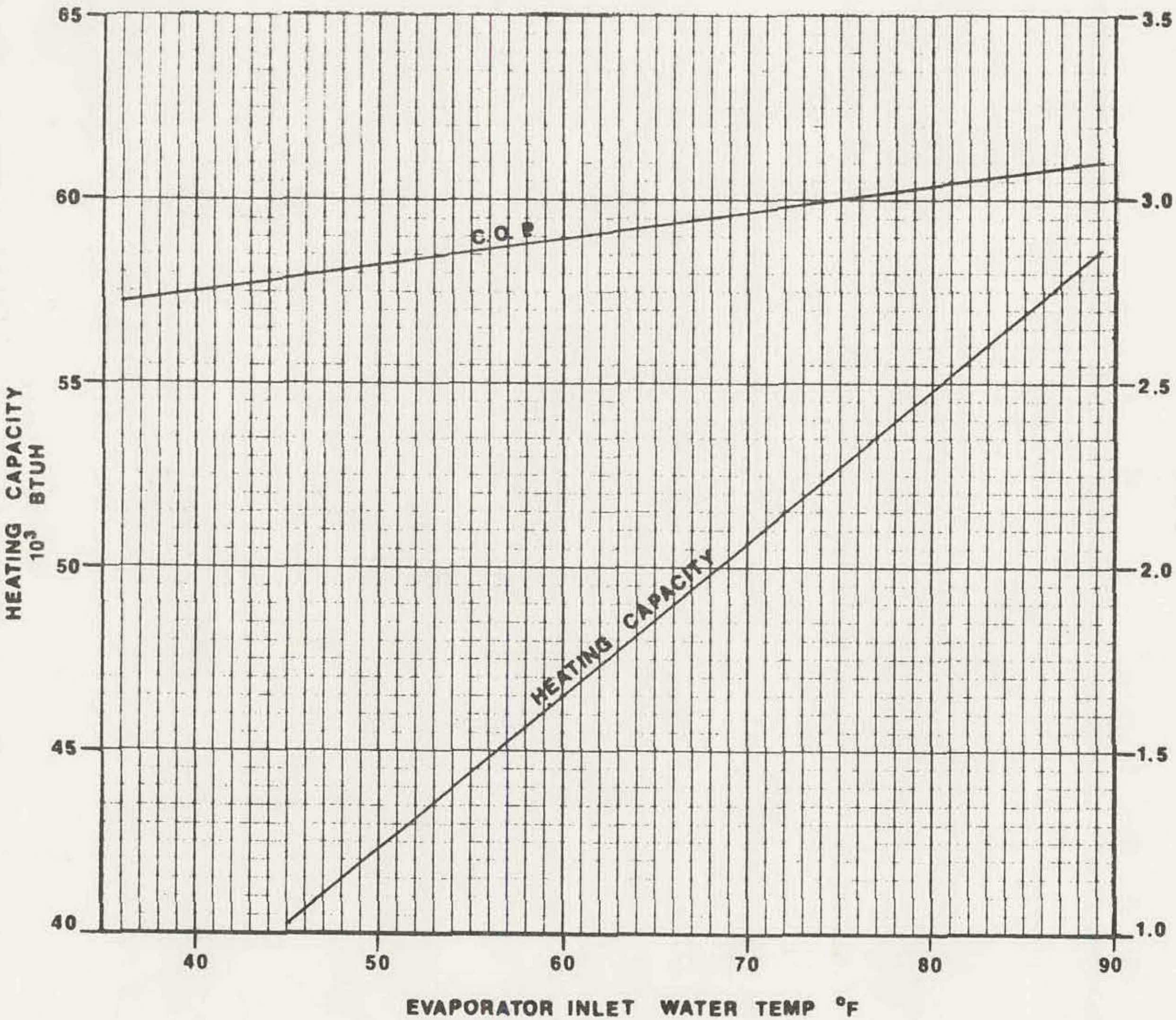
NOMINAL CAPACITY: 22,000 BTUH

NOMINAL C.O.P.: 2.740

WATER TEMP T_{ew} AT EVAPORATOR INLET - °F	HEATING CAPACITY BTUH	C.O.P.
40	16,720	2.466
50	19,360	2.603
60	22,000	2.740
70	24,640	2.877
80	27,280	3.014
95	31,240	3.219

FIG. 9-4

WATER-TO-AIR HEAT PUMP



NOMINAL CAPACITY: 46.500

NOMINAL C.O.P.: 2.890

WATER TEMP T_{ew} AT EVAPORATOR INLET – °F	HEATING CAPACITY BTUH	C.O.P.
45	40,222	2.780
50	42,315	2.818
60	46,500	2 890
70	50,685	2.962
80	54,870	3.035
95	61,147	3.142

FIG. 9–5

The COP of the chiller for a given set of Water temperatures other than the rated conditions is given by

$$COP_{chiller} = \frac{COP_{nominal}}{\text{\% nominal work input expressed as a decimal}} \quad (9\text{-}28)$$

The COP chiller found for the given temperatures of water, is related to the COP heat pump by equation 9-6 given in section 9.4.3. The COP value, found by using equation 9-28 were reduced to allow for the fouling factors. Information on fouling factor coorection presented in Reference 5 was used as a guide for this correction. The corrected values of COP used in our calculations were found by

$$COP_{(corrected)} = COP_{\text{chiller or heat pump}} \times F$$

Where F = fouling correction factor; a value of 0.931 for F was assumed in one calculation.

TABLES 9-2 and 9-3 are included to indicate the values of COP for representative water temperatures ranges.

Air-to-air heat pumps - The COP and EER values were computed using the information given in Reference 6. The computed values of the COP's for air-to-air heat pumps for various outside air temperatures are listed in Chapter 3. For the purposes of illustration, a typical air-to-air heat pump cycle is pictured in Figure 9-6.

TABLE 9-2

CENTRAL CHILLER

LCWT	LEWT	COP (Chiller)*
90°F	50°F	5.492
80°F	50°F	5.8864
70°F	50°F	6.023

LCWT = Leaving condenser water temperature

LEWT = Leaving evaporator water temperature

*COP of Chiller for Corrected for Fouling Factors

TABLE 9-3

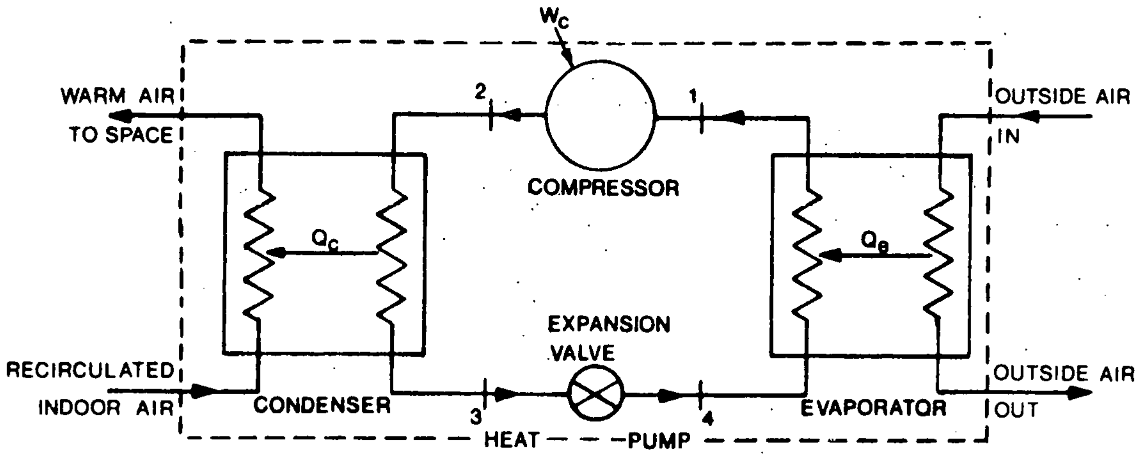
CENTRAL HEAT PUMP

LCWT	LEWT	COP (Heat Pump)*
80°F	40°F	6.494
80°F	50°F	6.991
80°F	60°F	7.361
80°F	70°F	7.647

LCWT = Leaving condenser water temperature

LEWT = Leaving evaporator water temperature

*COP of Heat Pump Corrected for Fouling Factors



AIR-TO-AIR HEAT PUMP

FIG. 9-6

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APPENDIX A

TABLE A-1

WEATHER DATA FOR TEST CITIES

City	ASHRAE[a] Heating Degree Days	Representative Heating Degree Day Range	WEATHER DATA USED IN THIS STUDY			
			Heating Degree Days	Cooling Degree Days	Year of Weather Data	Deviation Level
Houston	1278	500 - 2000	1290	2339	1955	3%
Birmingham	2551	2000 - 3000	2844	1928	1955	7%
Atlanta	2961	2500 - 3500	2821	1589	1959	6%
Tulsa	3680	3000 - 4000	3504	1949	1950	6%
Philadelphia	4486	4000 - 5000	4508	1104	1951	8%
Seattle	4424	--	4407	183	1960	2%
Columbus	5560	5000 - 6000	5467	809	1964	10%
Cleveland	6351	6000 - 7000	6097	613	1964	10%
Concord	7383	7000 - 8000	7377	349	1964	10%

(a) Heating degree days from ASHRAE 1973 Systems Handbook, Chapter 43.

SOURCE: "HEAT PUMP TECHNOLOGY," GORDIAN ASSOCIATES (1978)[10]

TABLE A-2

MOUNTAINSIDE COMMERCIAL BUILDING HEATING AND COOLING DESIGN LOADS

	Heating		Cooling			
	97½% Design Dry Bulb Temperature (°F)	Design Heat Load (Btuh)	2½% Design Dry Bulb Temperature (°F)	2½% Design Wet Bulb Temperature (°F)	Design Sensible Cooling Load (Btuh)	Design Total Cooling Load (Btuh)
Houston	32	85,151	94	80	143,447	206,119
Birmingham	22	104,613	94	78	143,447	195,667
Atlanta	23	106,664	92	77	139,380	189,992
Tulsa	16	118,971	99	78	155,656	203,852
Philadelphia	15	121,022	90	77	135,315	187,535
Seattle	32	86,152	79	65	112,937	124,957
Columbus	7	139,484	88	76	131,244	178,640
Cleveland	7	139,484	89	75	133,277	176,653
Concord	-7	166,150	88	73	131,244	164,972

SOURCE: "HEAT PUMP TECHNOLOGY," GORDIAN ASSOCIATES (1978)[10]

TABLE A-3

TEST HOUSE
DESIGN TEMPERATURES AND HEAT LOAD FOR HEATING AND COOLING EQUIPMENT

	HEATING		COOLING			
	97-1/2% DESIGN DRY BULB TEMPERATURE	DESIGN HEAT LOAD (BTUH)	2-1/2% DESIGN DRY BULB TEMPERATURE	2-1/2% DESIGN WET BULB TEMPERATURE	DESIGN SENSIBLE COOLING LOAD (BTUH)	DESIGN TOTAL COOLING LOAD (BTUH)
Houston	32	27,030	94	80	24,069	36,844
Birmingham	22	33,915	94	78	25,335	35,803
Atlanta	23	33,288	92	77	23,126	32,994
Tulsa	16	37,671	99	78	29,383	38,351
Philadelphia	15	38,242	90	77	24,880	35,302
Seattle	32	27,621	79	65	18,757	20,365
Columbus	7	42,913	88	76	23,468	33,090
Cleveland	7	42,913	89	75	24,454	32,845
Concord	-7	52,088	88	73	23,837	30,420

SOURCE: "HEAT PUMP TECHNOLOGY," GORDIAN ASSOCIATES (1978)[10]

MOUNTAINSIDE COMMERCIAL BUILDING GENERAL CHARACTERISTICS

Building Style	Two stories, masonry construction
Gross Floor Area	5580 square feet
Exposure	Front of building faces north
Occupancy	40 persons (maximum)
Lighting and Equipment Baseload	14 KW (maximum)
Construction Walls	Stucco surface over 12" concrete blocks (38# light, filled), 1-1/4" furring (16" on centers) and 1/2" styrofoam. Inside surface is 1/2" gypsum board. Net wall area: 4724 square feet.
Roof	3/8" asphalt built up roof over 1/2" plyscore sheathing, 2" x 10" rafters (16" on centers), fiberglass insulation (6") above 1/2" acoustical tile. Roof area: 2790 square feet.
Windows and Doors	Safety glass, 1/2" double glazed. Total glass area: 732 square feet.
Floor Slab	Concrete slab, on grade, 24" insulation
Ceilings	Hung ceiling between floors carries ducts and electric and water services. Ceiling height is 11 feet for each floor
Stairwell	Center staircase located just inside front entrance, double back design with turnaround landing.

TEST HOUSE GENERAL CHARACTERISTICS

House Style	Two stories, wood frame, 2 car garage
Living Area	1850 sq. ft. (excluding basement).
Room Complement	
First Floor	Foyer, living room, dining room, family room, kitchen, powder room.
Second Floor	Master bedroom, large bedroom, 2 small bedrooms, 2 baths.
Exposure	Front of house faces north.
Exterior Surface	Aluminum siding.
Doors	One - wood panel construction.
Roof	Black asphalt shingles over building paper and 5/8" plywood deck.
Glass Areas	16 glass areas, including patio sliding door. (For total glass area see bldg. plans.) All windows are aluminum cased, double hung, weather stripped, double pave insulating glass. Basement windows (2) are single glazed, 1/8" sheet glass.
Basement	
Walls	11 course, 8 inch concrete block
Floor	4 inch concrete slab over 4 inch gravel bed, 662 sq. ft.
Crawl Space	4 inch gravel bed over plastic vapor barrier.

TABLE A-4

Table 5A. *Summary of facilities having sewer systems and sewage disposal by population groups, states, and drainage basins*

	Total		Separate		Combined		Both		Not stated		Treated		Untreated	
	Number	Population served	Number	Population served	Number	Population served	Number	Population served	Number	Population served	Number	Population served	Number	Population served
Population Size Groups														
Under 500	1,975	693,874	1,711	613,599	54	18,020	15	5,605	195	56,650	1,723	621,959	252	71,915
500-1,000	2,392	1,828,753	2,015	1,550,566	149	119,830	44	30,850	184	127,507	2,057	1,599,828	335	228,925
1,000-5,000	5,666	12,385,893	4,713	384,413	499	1,079,910	202	467,400	252	454,170	4,990	11,193,008	676	1,192,885
5,000-10,000	1,573	9,570,149	1,242	7,706,256	120	766,020	117	788,700	94	309,173	1,437	8,890,051	136	680,098
10,000-25,000	1,248	15,504,150	984	1,881,310	117	1,749,125	88	1,523,280	59	350,485	1,164	14,467,295	84	1,036,855
25,000-50,000	502	12,697,700	371	8,903,795	47	1,662,470	57	1,944,745	27	186,690	473	12,058,065	29	639,635
50,000-100,000	288	13,421,175	201	8,280,485	38	2,694,985	31	1,918,290	18	527,415	269	12,673,695	19	747,480
100,000-250,000	247	14,855,310	165	7,206,345	28	4,159,460	30	3,448,600	24	40,905	230	13,511,350	17	1,343,960
250,000-500,000	154	13,620,000	129	7,112,805	11	2,898,975	10	3,118,300	4	490,000	149	11,997,955	5	1,622,125
Over 500,000	78	45,648,965	31	1,960,305	20	16,717,275	26	16,221,385	1	750,000	73	43,671,565	5	1,977,400
States														
Alabama	255	1,988,965	247	1,981,190	...	...	...	...	8	7,775	204	1,738,770	51	250,265
Alaska	25	62,365	22	28,465	...	...	3	33,900	...	...	6	11,505	19	50,860
Arizona	71	709,970	70	709,055	...	...	...	...	1	915	69	688,770	2	21,200
Arkansas	206	874,125	203	868,910	...	...	...	...	3	5,215	203	797,895	3	76,230
California	555	16,974,465	526	4,771,000	16	2,170,980	...	...	13	32,485	546	16,953,835	9	20,630
Colorado	218	2,420,398	204	1,558,088	2	8,500	2	101,655	10	752,155	204	2,413,203	14	7,195
Connecticut	88	1,581,155	61	714,145	15	482,950	8	360,200	4	23,860	83	1,554,535	5	26,620
Delaware	18	387,610	11	35,885	1	2,495	4	14,775	2	334,455	18	387,610	...	...
District of Columbia	1	1,750,000	...	...			1	1,750,000	...	...	1	1,750,000	...	...
Florida	521	2,939,837	437	2,788,725	2	33,770	1	30,000	81	87,342	510	2,875,207	11	64,630
Georgia	346	2,356,340	329	1,272,655	6	772,995	5	285,290	6	25,400	307	2,159,260	39	197,080
Hawaii	28	498,640	27	482,700	...	...	...	...	1	15,940	21	132,940	7	365,700
Idaho	105	363,870	88	287,860	9	49,405	5	21,455	3	5,150	93	348,880	12	14,990
Illinois	644	8,536,420	336	1,533,245	103	4,883,850	20	1,758,465	185	360,860	629	8,377,710	15	158,710
Indiana	390	3,603,760	167	486,895	192	3,002,830	3	44,340	28	69,695	302	3,493,210	88	110,550
Iowa	517	1,643,470	482	1,042,960	17	184,560	9	409,050	9	6,900	499	1,625,280	18	18,190
Kansas	452	1,896,600	429	1,493,800	5	351,400	2	44,500	16	6,900	435	1,664,345	17	232,255
Kentucky	252	1,382,930	239	657,670	3	171,780	9	553,085	1	475	239	1,370,705	13	12,225
Louisiana	197	2,322,520	195	2,284,500	...	...	...	...	2	38,020	173	1,393,945	24	928,575
Maine	112	553,115	35	78,755	40	199,785	34	236,500	3	38,075	33	180,005	79	373,110
Maryland	100	1,594,605	79	1,544,280	6	11,300	3	2,500	12	36,525	92	1,585,930	8	8,675
Massachusetts	162	3,726,010	91	634,910	35	876,840	7	2,102,955	29	111,305	94	2,886,575	68	839,435
Michigan	314	6,668,935	157	1,049,895	58	4,188,495	53	1,302,610	46	127,935	267	6,574,205	47	94,730
Minnesota	443	2,089,770	416	912,760	23	1,157,270	3	19,690	1	50	441	2,067,140	32	22,630
Mississippi	249	995,930	247	975,730	...	...	1	20,000	1	200	215	701,610	34	294,320
Missouri	564	3,074,615	494	1,390,960	14	613,745	19	923,310	37	146,600	490	1,953,920	74	1,120,695
Montana	126	436,780	114	403,180	1	2,500	11	31,100	...	...	125	436,180	1	600
Nebraska	431	845,695	357	623,865	8	12,185	4	186,630	62	23,015	384	824,030	47	21,665
Nevada	31	491,295	29	378,895	1	12,500	1	99,900	...	...	29	489,780	2	1,515
New Hampshire	79	300,800	30	93,350	23	82,650	24	119,300	2	5,500	27	132,850	52	167,950
New Jersey	318	5,827,710	224	3,718,950	7	289,500	4	1,287,355	83	531,905	318	5,827,710	...	...
New Mexico	80	673,970	80	673,970	...	...	...	...	...	...	80	673,970	...	...
New York	620	15,104,360	461	2,754,500	65	6,836,585	79	5,497,405	15	15,870	507	13,501,395	113	1,602,965
North Carolina	382	1,980,410	379	1,976,925	1	1,600	...	...	2	1,885	349	1,847,325	33	133,085
North Dakota	221	372,890	196	249,760	18	108,520	3	13,480	4	1,130	211	365,195	10	7,695
Ohio	609	7,822,259	428	2,850,514	111	1,337,780	61	3,623,265	9	10,700	537	7,707,459	72	114,800
Oklahoma	374	1,660,890	371	1,659,715	...	...	...	...	3	1,175	370	1,656,800	4	4,090
Oregon	176	1,139,292	141	471,612	17	195,200	15	463,930	3	8,550	167	1,102,642	9	36,650
Pennsylvania	746	10,237,071	508	3,532,566	106	514,010	84	6,084,570	48	105,925	486	9,309,041	260	928,030
Puerto Rico	98	826,825	95	816,125	...	...	...	...	3	10,700	91	716,160	7	110,665
Rhode Island	22	549,025	19	184,505	...	...	3	364,520	...	...	16	545,070	6	3,955
South Carolina	261	961,505	256	959,820	...	...	...	...	5	1,685	224	703,240	37	258,265
South Dakota	207	393,520	180	295,820	17	77,995	7	18,065	3	1,640	190	378,525	17	14,995
Tennessee	213	1,947,325	190	1,491,670	5	428,400	...	...	18	27,255	200	1,429,830	13	517,495
Texas	911	8,425,180	886	8,145,830	1	55,000	2	400	22	223,950	911	8,425,180	...	...
Utah	78	890,885	77	890,825	1	60	...	...	...	...	64	878,770	14	12,115
Vermont	94	208,300	18	27,210	10	13,780	46	160,455	20	6,855	38	163,180	56	45,120
Virginia	273	2,286,969	252	1,627,709	3	1,675	8	648,325	10	9,260	253	2,268,944	20	18,025
Virgin Islands	3	31,768	...	...	...	...	...	...	3	31,768	...	...	3	31,768
Washington	221	1,865,590	152	798,000	38	891,515	12	157,605	19	18,470	208	1,798,525	13	67,065
West Virginia	193	849,715	94	236,740	75	526,315	19	81,860	5	4,800	118	717,855	75	131,860
Wisconsin	442	2,843,795	356	898,940	28	1,315,350	45	614,740	13	14,765	440	2,843,145	2	650
Wyoming	81	255,805	77	253,845	...	...	...	...	4	1,960	78	255,050	3	755
Major Drainage Basins														
Northeast	1,300	26,150,270	786	7,111,370	187	8,537,320	195	9,867,585	132	633,995	935	23,115,300	365	3,034,970
North Atlantic	882	14,120,086	679	6,146,821	82	643,715	63	6,811,820	58	517,730	730	13,454,831	152	665,255
Southeast	1,877	10,701,976	1,761	9,453,179	8	806,765	7	317,990	101	124,042	1,708	9,663,351	169	1,038,625
Tennessee River	203	1,360,320	178	1,162,260	5	171,050	1	2,845	19	24,165	176	1,328,955	27	31,365
Ohio River	1,599	12,843,855	1,078	4,995,040	323	3,883,705	110	3,813,990	88	151,120	1,266	12,264,260	333	579,595
Lake Erie	273	4,141,429	164	944,364	74	750,795	27	2,440,355	8	5,915	227	4,064,009	46	77,420
Upper Mississippi	1,942	13,757,860	1,512	3,977,450	164	6,364,825	62	3,012,885	204	402,700	1,818	12,535,365	124	1,222,495
Western Great Lakes	545	9,796,845	311	1,508,565	104	6,299,815	78	1,837,510	52	150,955	491	9,695,175	54	101,670
Missouri River	1,693	6,782,690	1,518	4,444,070	48	1,015,720	29	444,745	98	878,155	1,582	6,403,240	111	379,450
Southwest-Lower Mississippi	1,448	8,049,208	1,416	7,861,878	3	9,200	4	122,035	25	56,095	1,377	6,377,188	71	1,672,020
Colorado River	194	1,284,353	185	1,272,033	2	8,500	1	1,655	6	2,165	180	1,250,273	14	34,080
Western Gulf	860	8,239,197	836	7,961,447	1	55,000	2	400	21	222,350	859	8,238,477	1	720
Pacific Northwest	513	3,563,072	388	1,738,792	64	1,136,120	36	655,990	25	32,170	478	3,443,767	35	119,305
California	512	16,737,670	484	4,534,705	16	2,170,980	...	...	12	31,985	504	16,718,180	8	19,490
Great Basin	128	1,277,620	122	1,160,615	2	12,560	2	103,400	2	1,045	116	1,271,795	12	5,825
Alaska	25	62,365	22	28,465	...	...	3	33,900	...	...	6	11,505	19	50,860
Hawaii	28	498,640	27	482,700	...	...	...	...	1	15,940	21	132,940	7	365,700
Puerto Rico	98	826,825	95	816,125	...	...	...	...	3	10,700	91	716,160	7	110,665
Virgin Islands	3	31,768	...	...	...	...	...	...	3	31,768	...	...	3	31,768
Total	14,123	140,226,049	11,562	75,599,879	1,083	31,866,070	620	29,467,105	858	3,292,995	12,565	130,684,771	1,558	9,541,278

NOTE: Total U.S. Population in 1967 - 197,000,000 (Est.)

SOURCE: Municipal Waste Facilities - Statistical Summary, 1968 Inventory.[41]

TABLE A-5

Table 11. *Population served by sewers and sewage treatment—by geographical areas*

Geographic areas	1960 census population of sewered communities	Estimated population connected to sewers	Percent of census population connected to sewers	Percent of national total population connected to sewers	Percent of connected population—sewage discharged	
					Treated	Raw
New England [1]	8,976,318	6,918,405	77.1	4.9	79.0	21.0
Middle Atlantic [2]	28,823,298	31,169,141	108.1	22.2	91.9	8.1
South Atlantic [3]	13,167,414	15,106,991	114.7	10.8	94.6	5.4
East North Central [4]	26,136,377	29,475,169	112.8	21.0	98.4	1.6
East South Central [5]	5,660,886	6,315,150	111.6	4.5	83.0	17.0
West North Central [6]	10,237,528	10,316,560	100.8	7.4	86.1	13.9
West South Central [7]	11,618,812	13,282,715	114.3	9.5	92.4	7.6
Mountain [8]	4,624,927	6,242,973	135.0	4.5	99.1	.9
Pacific [9]	13,075,283	19,979,347	152.8	14.2	99.4	.6
Alaska, Hawaii, Puerto Rico and Virgin Islands	1,522,064	1,419,598	93.3	1.0	60.6	39.4
Total	123,843,107	140,226,049	113.2	100.0	93.2	6.8

[1] Maine, New Hampshire, Vermont, Massachusetts, Rhode Island, Connecticut.
[2] New York, New Jersey, Pennsylvania.
[3] Delaware, Maryland, District of Columbia, Virginia, West Virginia, North Carolina, South Carolina, Georgia, Florida.
[4] Ohio, Indiana, Illinois, Michigan, Wisconsin.
[5] Kentucky, Tennessee, Alabama, Mississippi.
[6] Minnesota, Iowa, Missouri, North Dakota, South Dakota, Nebraska, Kansas.
[7] Arkansas, Louisiana, Oklahoma, Texas.
[8] Montana, Idaho, Wyoming, Colorado, New Mexico, Arizona, Utah, Nevada.
[9] Washington, Oregon, California.

Table 13. *Percentage data for sewer systems and raw and treated discharge—by population groups*

Population size groups	Percent of total number of sewered communities	Percent of 1960 census population of sewered communities	Percent of total population connected—				Percent of total number of treatment plants
			To sewer systems	To raw discharge or treatment facilities [1]	To raw sewage discharge facilities	To sewage treatment facilities	
Under 500	13.9	.5	.5	.5	.8	.5	13.7
500–1,000	17.5	1.4	1.4	1.3	2.4	1.2	16.4
1,000–5,000	41.6	10.2	9.6	8.8	12.5	8.6	39.7
5,000–10,000	11.7	8.6	7.9	6.8	7.1	6.8	11.4
10,000–25,000	9.3	14.8	13.8	11.1	10.9	11.1	9.3
25,000–50,000	3.3	11.7	11.3	9.1	6.7	9.2	3.8
50,000–100,000	1.6	11.3	11.5	9.6	7.8	9.7	2.1
100,000–250,000	.7	9.8	9.7	10.6	14.1	10.3	1.8
250,000–500,000	.3	10.4	11.4	9.7	17.0	9.2	1.2
Over 500,000	.2	21.3	22.7	32.6	20.7	33.4	.6
Total	100.0	100.0	100.0	100.0	100.0	100.0	100.0

[1] Population served by facilities in population size groups shown.

SOURCE: Municipal Waste Facilities - Statistical Summary, 1968 Inventory.[41]

SAMPLE CALCULATION: CONVERSION TO PRIMARY ENERGY SOURCE EQUIVALENTS

BTU TO COAL BURNED AT ELECTRIC GENERATING STATION:

$$\text{LBS OF COAL} = \frac{(Q_{net})\ \text{Btu}\ \left(\frac{1}{3413}\ \frac{\text{kwh}}{\text{Btu}}\right)\ \left(10{,}300\ \frac{\text{Btu}}{\text{kwh}}\right)\ (1.09)}{12{,}000\ \frac{\text{Btu}}{\text{LB}}}$$

FOR 1980 - 2000: $Q_{net} = 5013.55 \times 10^{12}$ Btu

$$\text{EQUIVALENT LBS OF COAL} = \frac{(5013.55 \times 10^{12})\ \frac{1}{(3413)}\ (10{,}300)\ (1.09)}{12{,}000}$$

$$= 1{,}374{,}332 \times 10^{6}\ \text{LBS}$$

BTU TO OIL BURNED IN END-USER SYSTEM:

$$\text{GALS OF OIL} = \frac{(Q_{net})\ \text{Btu}}{139{,}000\ \frac{\text{Btu}}{\text{Gal}}}$$

FOR 1980 - 2000: $Q_{net} = 5013.55 \times 10^{12}$

$$\text{EQUIVALENT GALS OF OIL} = \frac{5013.55 \times 10^{12}}{139{,}000}$$

$$= 36{,}069 \times 10^{6}\ \text{GALS}$$

NOTE: Q_{net} is derived from Q_{total} (TABLE 2-1) through EQN. 2-4.

BTU TO GAS BURNED IN END-USER SYSTEM:

$$\text{CUBIC FEET OF NATURAL GAS} = \frac{(Q_{net})\ \text{Btu}}{1{,}000\ \frac{\text{Btu}}{\text{CFT}}} \times 1.08$$

$$\text{FOR 1980 - 2000:}\quad Q_{net} = 5013.55 \times 10^{12}$$

$$\text{EQUIVALENT CFT OF NAT. GAS} = \frac{5013.55 \times 10^{12}}{1{,}000} \times 1.08$$

$$= 5.415 \times 10^{12}\ \text{CFT}$$

APPENDIX B

APPENDIX B

The following calculations were made to determine the annual energy inputs of the Central Plant and End User Schemes for the HP-WHR scheme.

Basic Assumptions

Northern Climate

No. of End-Users -- 470

Annual H.V.A.C. Load Requirements Per End-User

Heating -- 105.47×10^6 Btu/Yr
Cooling -- 18.96×10^6 Btu/Yr

Southern Climate

No. of End-Users -- 980

Annual H.V.A.C. Load Requirements Per End-User

Heating -- 37.92×10^6 Btu/Yr
Cooling -- 45.03×10^6 Btu/Yr

Northern Climate

Heating

(1) Total heat required by end users = $470 \times 105.5 \times 10^6 = 4.96 \times 10^{10}$ Btu/Yr

(2) Energy from distribution system = $4.96 \times 10^{10} \times (1 - \frac{1}{3.03}) =$

3.32×10^{10} Btu/Yr

(3) Energy input to central heat pump = $3.32 \times 10^{10}/6.99 = 4.75 \times 10^9$ Btu/Yr

(4) Kwh input to central heat pump = $4.75 \times 10^9/3413 = 1{,}392{,}905$ kwh/Yr

(5) Central heat pump equivalent full load hours =

$$3.32 \times 10^{10} \times (1 - \frac{1}{6.99})/12 \times 10^6 = 2370 \text{ hr/Yr}$$

Northern Climate (Continued)

Cooling

(1) Total heat rejection to distribution by end-users =

$$470 \times 18.96 \times 10^6 \times (1 + \frac{1}{3.2}) = 1.17 \times 10^{10} \text{ Btu/Yr}$$

(2) Energy input to central heat pump = $1.17 \times 10^{10}/5.42 = 2.16 \times 10^9$ Btu/Yr

(3) Kwh input to central heat pump = $2.16 \times 10^9/3413 = 632{,}485$ kwh/Yr

(4) Central heat pump equivalent full load hours =

$$1.17 \times 10^{10}/12 \times 10^6 = 975 \text{ hr/Yr}$$

Southern Climate

Heating

(1) Total heat required by end-users = $980 \times 37.92 \times 10^6 = 3.72 \times 10^{10}$

(2) Energy from distribution system = $3.72 \times 10^{10} \times (1 - \frac{1}{3.03}) = 2.49 \times 10^{10}$

(3) Energy input to central heat pump = $2.49 \times 10^{10}/6.99 = 3.56 \times 10^9$ Btu/Yr

(4) Kw input to central heat pump = $3.56 \times 10^9/3413 = 1{,}043{,}500$ kwh/Yr

(5) Central heat pump equivalent full load hours =

$$2.49 \times 10^{10} \times (1 - \frac{1}{6.99})/12 \times 10^6 = 1778 \text{ hr/Yr}$$

Cooling

(1) Total heat rejection to distribution by end-users =

$$980 \times (45.03 \times 10^6) \times (1 + \frac{1}{3.2}) = 5.8 \times 10^{10}$$

(2) Energy input to central heat pump = $5.8 \times 10^{10}/5.42 = 1.07 \times 10^{10}$ Btu/Yr

(3) Kwh input to central heat pump = $1.07 \times 10^{10}/3413 = 3{,}131{,}062$ kwh/Yr

(4) Central heat pump equivalent full load hours =

$$5.8 \times 10^{10}/12 \times 10^6 = 4833 \text{ hr/Yr}$$

Circulating Pumps

Assume circulating pumps run

20 percent more hours than the central heat pump equivalent full load hours.

Northern Climate

$$1.2 \times (2370 + 975) \times 75 \text{ hp} \times .745 \text{ kwh/hp} = 224{,}282 \text{ kwh/Yr}$$

Southern Climate

$$1.2 \times (1778 + 4833) \times 75 \text{ hp} \times .745 \text{ kwh/hp} = 594{,}990 \text{ kwh/Yr}$$

Distribution Pumps

Assume distribution pumps consume the same proportion of energy annually as they do at full load.

Northern Climate ≈ 6%
Southern Climate ≈ 5%

(1) Northern Climate Pumping Energy =

$$470 \times (105.5 \times 10^6 + 18.9 \times 10^6) \times .06/3413 = 1{,}027{,}858 \text{ kwh/Yr}$$

(2) Southern Climate Pumping Energy

$$980 \times (37.9 \times 10^6 + 45.03 \times 10^6) \times .05/3413 = 1{,}190{,}615 \text{ kwh/Yr}$$

Miscellaneous

Auxiliary equipment such as controls, security lighting, etc., is estimated to consume an additional 12,000 kwh annually.

APPENDIX C

HEAT BALANCE

In most cases, heat losses or heat gain caused by radiation, convection, bearing friction, oil coolers, etc., are relatively small and may not be considered in the overall heat balance, but may be compensated for in the heat balance closure allowance.

In a hermetic package, where the motor is cooled by refrigerant gas, chilled water, or condenser water, the motor cooling load will be included in the measured condenser load, hence:

$$q_{EV} + q_{KW}\ \text{INPUT} = q_c$$

Where

q_{EV} = net cooling capacity of liquid cooler, Btuh

q_{KW}INPUT = electric energy input to the compressor motor, expressed in Btuh

q_c = net heat rejected to the condenser, Btuh

CLOSURE: THE HEAT BALANCE FOR THE TEST RUN SHOULD BE WITHIN 7.5 PER CENT.

SOURCE: ARI STANDARD 550-77 [54]

COEFFICIENT OF PERFORMANCE (C.O.P.)

The Coefficient of Performance can be calculated, for any specific set of operating conditions, from the following formula:

$$\text{C.O.P.} = \frac{\text{Total Heat Out}}{\text{Heat In}}$$

$$= \frac{\text{Compressor heat + Evaporator heat}}{\text{Compressor heat}}$$

Where

Compressor heat = Power supplied to unit (KWH) x 3413 (BTU/KWH) (includes fan power)

Evaporator heat = Flow rate (lb/hr)
x Specific heat of water
($1.0 Btu/lb-^oF$)

x Change in water temperature (^oF)

SOURCE: "Demonstration of Building Heating with a Heat Pump Using Thermal Effluent," Sector.[33]

ACTUAL CAPACITY

The actual capacity, in tons, can be determined by the following formula:

$$\text{tons} = W(T_e - T_1)/12{,}000$$

Where

W = flow rate of chilled water, lb per hr

T_e = temperature of water entering the unit, ^oF

T_1 = temperature of water leaving the unit, ^oF

NOTE: The specific heat of water is implicit in the equation and is assumed to be 1.0 $Btu/lbm-^oF$

SOURCE: ARI Standard 550-77.[54]

FOULING FACTORS

Fouling factors must be obtained experimentally by determining the values of U(overall heat transfer coefficient) for both clean and dirty conditions in the heat exchanger. The fouling factor is thus defined as:

$$R_f = \frac{1}{U_{Dirty}} - \frac{1}{U_{Clean}} \qquad \frac{(HR - Ft^2 - {}^oF)}{Btu}$$

and the overall heat transfer coefficient, U, is defined for a simple double-pipe heat exchanger as:

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{A_i \ln (r_o/r_i)}{2\pi KL} + \frac{A_i}{A_o}\left(\frac{1}{h_o}\right)} \qquad \left(\frac{Btu}{hr\text{-}Ft^2\ -{}^oF}\right)$$

Where

h_i,h_o = convection heat transfer coefficient

A_i,A_o = surface areas of inside tube

r_o,r_i = inner and outer radii of inside tube

K = conduction heat transfer coefficient of tube material

L = length of exchanger tubes

SOURCE: Heat Transfer, J. P. Holman.[11]

TABLE C-1

TYPICAL FOULING FACTORS

TABLE 7. Fouling Resistances

Typical fouling resistances are referred to the surface on which they occur. In the absence of specific data for setting proper resistances the user may be guided by the values tabulated below.

Water

Temperature of Heating Medium	Up to 240°F		240-400°F	
Temperature of Water	125°F or less		Over 125°F	
Types of Water	Water velocity ft/sec		Water velocity ft/sec	
	3 ft and less	Over 3 ft	3 ft and less	Over 3 ft
Sea water	0.0005	0.0005	0.001	0.001
Brackish water	0.002	0.001	0.003	0.002
Cooling tower and artificial spray pond				
Treated makeup	0.001	0.001	0.002	0.002
Untreated	0.003	0.003	0.005	0.004
City or well water (such as Great Lakes)	0.001	0.001	0.002	0.002
Great Lakes	0.001	0.001	0.002	0.002
River water				
Minimum	0.002	0.001	0.003	0.002
Mississippi	0.003	0.002	0.004	0.003
Delaware, Schuylkill	0.003	0.002	0.004	0.003
East River and New York Bay	0.003	0.002	0.004	0.003
Chicago Sanitary Canal	0.008	0.006	0.010	0.008
Muddy or silty	0.003	0.002	0.004	0.003
Hard (over 15 grains/gal)	0.003	0.003	0.005	0.005
Engine jacket	0.001	0.001	0.001	0.001
Distilled	0.0005	0.0005	0.0005	0.0005
Treated boiler feedwater	0.001	0.0005	0.001	0.001
Boiler blowdown	0.002	0.002	0.002	0.002

*Ratings in columns 3 and 4 are based on a temperature of the heating medium of 240-400°F. If the heating medium temperature is over 400°F and the cooling medium is known to scale, these ratings should be modified accordingly.

SOURCE: Handbook of Heat Transfer, W.M. Rohsenow and J.P. Hartnett.[29]

APPENDIX D

SAMPLE CALCULATION: HEATING MODE

Reference is made to Figure 3-1 and the equations and procedure outlined in the analysis section (See 9.4.4).

DATA:

T_{fi} = temperature of effluent entering the evaporator of the central heat pump = 60°F

T_{fo} = temperature of effluent leaving the evaporator of the central heat pump = 50°F

T_{e1} = temperature of water entering the evaporator of the end user heat pump = 80°F

T_{e2} = temperature of water leaving the evaporator of the end user heat pump = 70°F

T_{c1} = temperature of water entering the condenser of the central heat pump = 70°F

T_{c2} = temperature of water leaving the condenser of central unit = 80°F

COP of the end user heat pump = 3.035 (from Fig. 9-5)

COP of the central heat pump = 6.991 (from Table 9-3)

Q = design heating load = 55,000 Btuh = Q_c = end-use of the house based on 75°F inside and -20°F outside.

W_{ce} = energy input to compressor of the end user heat pump =

$$\frac{Q_{c,\,(\text{end user unit})}}{\text{COP}} = \frac{55{,}000}{3.035}\ \text{Btuh} = 18121.9\ \text{Btuh}$$

Heat rejection factor HRF = $\dfrac{\text{COP}}{\text{COP}-0.92} = \dfrac{3.035}{3.035-0.92}$

$$\text{HRF} = 1.435$$

Heat absorbed in the evaporator of the end user heat pump Q_e (end user heat pump) = $\dfrac{55{,}000}{1.435}$ = 38327.52 Btuh

Heat rejected in the condenser of the central station heat pump = Q_e (end user heat pump) = 38327.52

Work input to compressor of the central station heat pump: $W_{cc} = \dfrac{38327.52}{6.991}$ = 5482.41

Total compressor energy input: $\Sigma W_c = \dot{W}_{ce} + \dot{W}_{cc} = 23603$ Btuh

PUMPING ENERGY:

Mass rate of flow of water through the evaporator of the end user unit $= \dot{M}_w \dot{Q}_e$ (end user heat pump) $= \dot{M}_w C_{pw} (80-70) = 38327.52$

Where

$C_{pw} = 1.00$

$\dot{M}_w = 3832.75$ lbm/hr (2.4gpm/12Btu)

The same mass flows through the condenser of the central heat pump.

MASS RATE OF FLOW OF EFFLUENT:

$$\text{HRF(central heat pump)} = \frac{6.991}{6.991-0.02} = 1.152$$

$$\text{Heat extracted from the effluent in the central station} = \frac{38327}{1.152} = 33269.97$$

$$\dot{M}_{eff} = \text{mass rate of flow of effluent} = \frac{33269.97}{1 \times (10)} = 3327.00 \text{ lbm/hr}$$

$$\dot{M}_{eff} = 3327.0 \text{ lbm/hr}$$

For these mass rates of flow the pumping energies for water flow through the condenser of the central pump, evaporator of the end user heat pump and the effluent pump were calculated, using the procedure outlined in the analysis section and the information given in References 5 and 7.

E_{System} = Total energy input to the system: $2\Sigma W_c + \Sigma W_p +$ other losses (Leakage, thermal, etc.

It is noted that the pumping energy (827. Btu/hr) is 3.5% of the compressor input. Increasing the pumping energy by 40% (to allow for other losses) would bring it to 4.9% (or 5% approximate).

Total energy input to compressor = 23,603.00 Btuh.

5% of compressor energy = 1180.00

Total energy input to system = 24,783.00 Btu/hr = E_{System}

$$\text{COP system} = \frac{55,000}{24,783.}$$

$$= 2.219$$

NOTE: If in an actual system the pumping energy and other losses is 10% of the total energy input to compressor, then system COP would become 2.118 - a reduction of 4.55% in the COP system.

SAMPLE CALCULATION: PUMP ANALYSIS

In the process of raising the pressure (or head) of the distribution medium (water or clarified effluent), energy is consumed by the pump. The energy added to increase the pressure increases the enthalpy of water resulting in an increase of temperature. Since heat transfer from the pump casing is negligible, inefficiencies due to frictional losses in the pumping process tend to increase the temperature of the fluid. Pumping energy and the temperature increase of water during the pumping process is relatively small as shown in the following analysis:

$$W_p = \frac{V_f(P_2-P_1)}{\eta} \times \frac{144}{778} \frac{Btu}{lbm} \qquad \text{(D-1)}$$

Where

V_f = Specific volume of water at its temperature (which is almost a constant over the temperature range)

P_1, P_2 = Inlet and exit pressures respectively, $\frac{lbm}{IN^2}$

W_p = Work input to pump

η_p = Efficiency of the pump

Also energy balance around the pump yields

$$q + h_1 = W_p + h_2 \qquad \text{(D-2)}$$

$$-(W_p) = h_2 - h_1$$

For example, for a pressure rise of

P_2-P_1 = 100 psi; water entering at 80^oF, V_f = 0.01607 $\frac{ft^3}{lbm}$

and assuming pump efficiency of 70%, we have from equation (D-1):

$$-W_p = \frac{0.01607(100)(144)}{.70} = 330.58 \ \frac{ft\text{-}lbf}{lbm}$$

or

$$-W_p = \frac{330.58}{778} = 0.425 \ \frac{Btu}{lbm}$$

Also, the enthalpy increase, $(h_2 - h_1) = (W_p) = 0.425 \ \frac{Btu}{lbm}$ (D-4)

The temperature rise of water $= C_p \ (T_2 - T_1) - h_2 - h_1 = 0.425$

Where C_p = specific heat of water $= 1.00 \ \frac{Btu}{lbm \ F}$.

The temperature rise of water during the pumping process is therefore approximately $0.425^{o}F$ for a pressure increase of 100 psi.

SAMPLE CALCULATION OF ANNUAL ENERGY CONSUMPTION

Reference is made to the ASHRAE Systems Volume, 1976. (Ref. 49).

a. Conventional Fuel:

For 40% oversized furnace, $C_f = 1.79$

Assumed rated efficiency; $Eff = 0.75$

H_L = Heating Load = 55,000 Btuh

C_D, For Northwest Central Region

= 0.57 (Degree Days = 8238,

TD = (75 - (-20)) = 95°F

$$E_{Heating} = \left(\frac{H_L \times DD \times 24}{TD \times Eff.}\right) C_D \times C_F$$

$$= 155.717 \times 10^6 \text{ Btu}$$

b. Air to Air Heat Pump:

$C_F = 1.0$

$$E_{Heating} = \left(\frac{H_L \times DD \times 24}{TD \times COP_{Seasonal}}\right) C_D$$

$COP_{Seasonal}$ for N.W. Central Region = 1,382

$COP_{Seasonal}$ for S. Atlantic Region = 2,220

For the Northwest Central region with $C.O.P._{Season}$ = 1.382

$E_{Heating}$ = 4,7210 MBtu/yr

Distribution

ANL

P. T. Bauer
J. M. Calm (50)
A. S. Kennedy
T. J. Marciniak
J. Martusek
M. Pacl (12)
J. Pascual
V. A. Rabl
J. Tschanz
ANL Libraries (C. Archer)
TIS Files (E. N. Pettitt)

DOE

G. Anderson
S. Cavros
J. Kaminsky
J. C. Rodousakis (25)
I. Sewell
DOE-TIC (65)

External

C. E. Brett, University of Alabama (3)
V. R. Anderson, Billings Energy Corporation (3)
P. F. Swenson, Consolidated Natural Gas Service Company (3)
A. Halfon, Dubin-Bloome Associates (3)
H. G. Lorsch, Franklin Research Center (3)
D. W. Wade, Georgia Tech Research Institute (25)
P. A. Anderson, Honeywell Energy Resources Center (3)
L. L. Northrup, Northrup Incorporated (3)
L. B. Katter, Rocket Research Company (3)