ANL/CNSV-TM-23

Heat-Pump-Centered Integrated Community Energy Systems

System Development

Georgia Institute of Technology Final Report

MASTER

ICES
INTEGRATED COMMUNITY ENERGY SYSTEMS
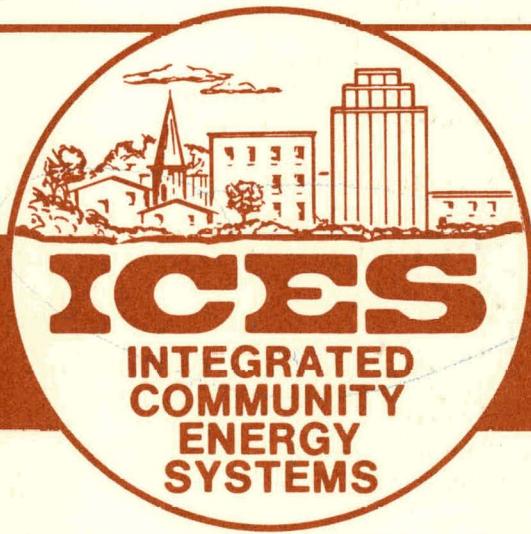
ENERGY AND ENVIRONMENTAL SYSTEMS DIVISION

ARGONNE NATIONAL LABORATORY

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HEAT-PUMP-CENTERED
INTEGRATED COMMUNITY ENERGY SYSTEMS

System Development.
Georgia Institute of Technology Final Report*

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

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

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

A specific ICES may consist of either a partial or complete integration of these approaches, as appropriate to strike the desired balance among a community's economic, social, environmental, and energy-conservation goals. An ICES can be applied to a total community as well as to portions of a community and the services provided need not be the same for all areas served. While ICES include a broad spectrum of technologies to meet energy service requirements, the ICES concept does not arbitrarily define either the energy services to be provided or the type, size, or function of the area to be served. As a result, an ICES is tailored for each application. The determination of the kind and number of energy services provided and the size of the service area for each such service is based on an optimum combination of energy efficiencies, indigenous resource and labor supplies, economics, and environmental conditions.

Heat-pump-centered integrated community energy systems (HP-ICES) are energy systems for communities which provide heating, cooling and/or

other thermal energy services through the use of heat pumps. Since heat pumps primarily transfer energy from existing and otherwise probably unused sources, rather than convert it from electrical or chemical to thermal form, HP-ICES offer significant potential for energy savings. By powering these heat pumps with nonscarce fuels, the use of which would be impractical in most conventional systems, less-abundant fuels including natural gas and oil can be conserved. Secondary benefits of HP-ICES include reduction of adverse environmental effects as compared to conventional systems, reliable production of services in contrast to the increasingly frequent utility curtailments and interruptions, and delivery of services to consumers at costs lower than those for conventional systems (including acquisition, operation, and maintenance costs).

The report which follows is a result of the System Development Phase of the HP-ICES Project. The objective of this multiphase project is development and demonstration of HP-ICES concepts leading to one or more operational systems by the end of 1984. The seven phases include System Development, Demonstration Design, Design Completion, HP-ICES Construction, Operation and Data Acquisition, HP-ICES Evaluation, and Upgraded Continuation.

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

The report which follows presents the findings in the development and analysis of one concept under investigation. This report was prepared by the Georgia Institute of Technology.

WASTEWATER HEAT RECOVERY
HEAT PUMP CENTERED-INTEGRATED COMMUNITY
ENERGY SYSTEMS

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ABBREVIATIONS AND SYMBOLS

SYMBOL	DEFINITION
A	area (orifice), ft^2 - (sec. 10.2.2)
A_s	heat transfer surface area, ft^2
ADF	<u>A</u>verage <u>D</u>aily <u>F</u>low
AMC_i	<u>A</u>nnual <u>M</u>aintenance <u>C</u>osts of the ith alternative
ADC_i	<u>A</u>nnual <u>O</u>perating <u>C</u>osts of the ith alternative
APDC	<u>A</u>rea <u>P</u>lanning and <u>D</u>evelopment <u>C</u>ommission
ARI	<u>A</u>ir Conditioning and <u>R</u>efrigeration <u>I</u>nstitute
ASHRAE	<u>A</u>merican <u>S</u>ociety of <u>H</u>eating, <u>R</u>efrigerating and <u>A</u>ir Conditioning <u>E</u>ngineers
avg.	average
BOD	<u>B</u>iochemical <u>O</u>xygen <u>D</u>emand
Btu	<u>B</u>ritish <u>t</u>hermal <u>u</u>nit
Btuh	<u>B</u>ritish <u>t</u>hermal <u>u</u>nits per <u>h</u>our
C	thermal conductance of pipe, $\frac{Btu}{hr-ft^2-^oF}$ (sec. 11.4.7)
C	specific heat of water, $\frac{Btu}{Lbm-^oF}$ (sec. 2.3.1)
C_v	specific heat of water, $\frac{Btu}{Lbm-^oF}$ (sec. 11.4.7)
c_p	specific heat of water, $\frac{Btu}{Lbm-^oF}$ (sec. 11.4.4)
c	weir or orifice constant (sec. 10.2.2)
CDD	<u>C</u>ooling <u>D</u>egree <u>D</u>ays
CFT	cubic feet
C.O.P.	<u>C</u>oefficient <u>o</u>f <u>P</u>erformance
cu.ft.	cubic feet

ABBREVIATIONS AND SYMBOLS (Cont'd)

SYMBOL	DEFINITION
$^{\circ}C$	degree Celsius
D	diameter, ft.
d	distance, ft.
d	delta or "change in" (sec. 5.3.1)
DD	heating degree days
E	energy
EER	Energy Efficiency Ratio
EPA	Environmental Protection Agency
F	fouling factor (sec. 11.4.8)
f	friction factor (sec. 11.4.7)
FC_i	First Cost of the ith alternative
FHA	Federal Housing Authority
FPM	Feet Per Minute
FPS	Feet Per Second
ft.	feet
ft/min	feet per minute
ft/sec	feet per second
g	acceleration due to gravity
gal	gallon
GPD	Gallons Per Day
gpm	gallons per minute
H	head, ft. (sec. 10.2.2)
h	head, ft. (sec. 10.2.2)
h_i	surface conductance of pipe interior $\frac{Btu}{hr\text{-}ft^2\text{-}^{\circ}F}$ (sec. 11.4.7)

SYMBOL	DEFINITION
h_1	enthalpy @ state 1, etc., $\frac{Btu}{Lbm}$
HDD	Heating Degree Days
HER	Heat Extraction Rate
HEW	Health, Education and Welfare
HHV	High Heating Value
HP	Horse Power
hp	horse power
HP-ICES	Heat Pump - Integrated Community Energy Systems
HP-WHR	Heat Pump-Wastewater Heat Recovery
hr	hour
HRF	Heat Rejection Factor
HRR	Heat Rejection Rate
HUD	Housing and Urban Development
HVAC	Heating, Ventilating and Air Conditioning
ICES	Integrated Community Energy Systems
k(K)	constant
K_p	heat transfer factor $\frac{Btu}{hr\text{-}ft\text{-}^oF}$ (sec. 11.4.7)
K_s	thermal conductivity of soil $\frac{Btu\text{-}inch}{hr\text{-}ft^2\text{-}^oF}$ (sec. 11.4.7)
K_w	thermal conductivity of pipe material $\frac{Btu\text{-}inch}{hr\text{-}ft^2\text{-}^oF}$
KW(kw)	KiloWatt
KWH(kwh)	KiloWatt Hour
L	BOD (sec. 5.3.1)

SYMBOL	DEFINITION
L	length, ft.
l	liter
lb	pound
lbm	pound-mass
LCWT	$\underline{L}$eaving $\underline{C}$ondenser $\underline{W}$ater $\underline{T}$emperature - ^{o}F
LEWT	$\underline{L}$eaving $\underline{E}$vaporator $\underline{W}$ater $\underline{T}$emperature - ^{o}F
LMTD	$\underline{L}$og $\underline{M}$ean $\underline{T}$emperature $\underline{D}$ifference - ^{o}F
ln	natural log
M	mass, Lbm
$\dot{M}(\dot{m})$	mass rate of flow, $\frac{Lbm}{hr}$
MBtu	Btu x 10^3
mg	milligram
MGD	$\underline{M}$illion $\underline{G}$allons per $\underline{D}$ay
mo.	month
n	exponent defined for each weir (sec. 10.2.2)
NIC	$\underline{N}$ot $\underline{I}$n$\underline{C}$luding
NWC	$\underline{N}$orth $\underline{W}$est $\underline{C}$entral region
P	design (peak heating load per hour, Btuh)
PSI	$\underline{P}$ounds per $\underline{S}$quare $\underline{I}$nch
PV_i	Present Value of the ith alternative
Q	total heat, Btu (sec. 2.3.1)
Q	free discharge (sec. 10.2.2)
$\dot{Q}$	rate of heat transfer, Btuh

SYMBOLS AND ABBREVIATIONS (Cont'd)

SUMBOL	DEFINITION
QUAD	Btu x 10^{15}
R	universal gas constant
r	radius, ft.
RTD	Resistance Temperature Device
SA	South Atlantic region
sq.ft.	square feet
T	temperature, ^{o}F
t	time, hrs.
therm	natural gas Btu x 10^{5}
TD	design temperature difference, ^{o}F
U_u	overall heat transfer coefficient, $\frac{Btu}{hr-ft^{2}-^{o}F}$ (sec. 11.4.7)
V	velocity, ft/sec (sec. 10.2.2)
$\bar{v}$	velocity, ft/sec (sec. 11.4.7)
VA	Veterans Administration
W	flow of wastewater, $\frac{Lbm}{hr}$ (sec. 2.3.1)
W_c	compressor work, KWH or
W_{losses}	work of thermal losses, Btuh
yr.	year
Δ	change in
η	efficiency
ρ	density $\frac{Lbm}{ft^{3}}$
θ	time, hrs

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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 $50^{\circ}F$ to $80^{\circ}F$. Primary energy source savings from the nationwide implementation of this system is projected to be 6.0 QUADS- fuel oil, or 8.5 QUADS - natural gas, or 29.7 QUADS - coal for the period 1980 - 2000, depending upon the type and mix of conventional space conditioning systems which could be displaced with the HP-WHR system. Site-specific HP-WHR system designs are presented for two application communities in Georgia. Performance analyses for these systems project Annual cycle system C.O.P.'s of 2.049 and 2.519. Economic analysis on the basis of a life cycle cost comparison shows one site-specific system design to be cost competitive in the immediate market with conventional residential and light commercial HVAC systems. The second site-specific system design is shown through a similar economic analysis to be more costly than conventional systems due mainly to the current low energy costs for natural gas. It is anticipated that, as energy costs escalate, this HP-WHR system will also approach the threshold of economic viability.

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1.0 SYSTEM CONCEPT

1.1 INTRODUCTION

The Heat Pump Centered-Integrated Community Energy System (HP-ICES) concept presented in this report is essentially a cascaded heat recovery scheme which extracts low-grade thermal energy from a community's wastewater effluent by means of electrically driven heat pumps. The extracted thermal energy is distributed into the community via a moderate temperature water piping system. At each point of end-use the thermal energy is extracted from the transport media by unitary heat pumps and utilized for space heating purposes. The system can also operate in a cooling mode by utilizing the community wastewater flow as a sink for heat rejected during space cooling. From an energy utilization standpoint, the concept represents a method of utilizing 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

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WASTEWATER EFFLUENT
FROM COMMUNITY

WASTEWATER TREATMENT
PLANT

PRIMARY CLARIFICATION

TEMPERED WATER RETURN

WASTEWATER
STORAGE

HEAT PUMP UNITS
THROUGHOUT
COMMUNITY

END
USERS

THERMAL
ENERGY
TO
END USERS

BYPASS
WASTEWATER

CENTRAL STATION
HEAT PUMP

WASTEWATER
USED AS
HEAT SOURCE

THERMAL
STORAGE

WARM WATER SUPPLY
PUMPED TO COMMUNITY

WASTEWATER FOR
FURTHER TREATMENT

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

are presented in subsequent sections.

1.2 SYSTEM ENERGY FLOW

Conventional space conditioning systems utilize high-grade energy sources, such as natural gas or oil, in direct combustion in building heating equipment or electricity generated at efficiencies of less than 40 percent. As depicted in Figure 1-2, both processes lose substantial 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 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.

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STACK LOSSES-10%

TRANSMISSION LOSSES-9%

ELECTRIC GENERATION PLANT

33%

ELECTRIC SYSTEMS

LOW-GRADE THERMAL EFFLUENT 57%

PRIMARY FUEL SOURCE 100%

STACK LOSSES-35%

TRANSMISSION LOSSES-8%

DIRECT-FIRED SYSTEMS
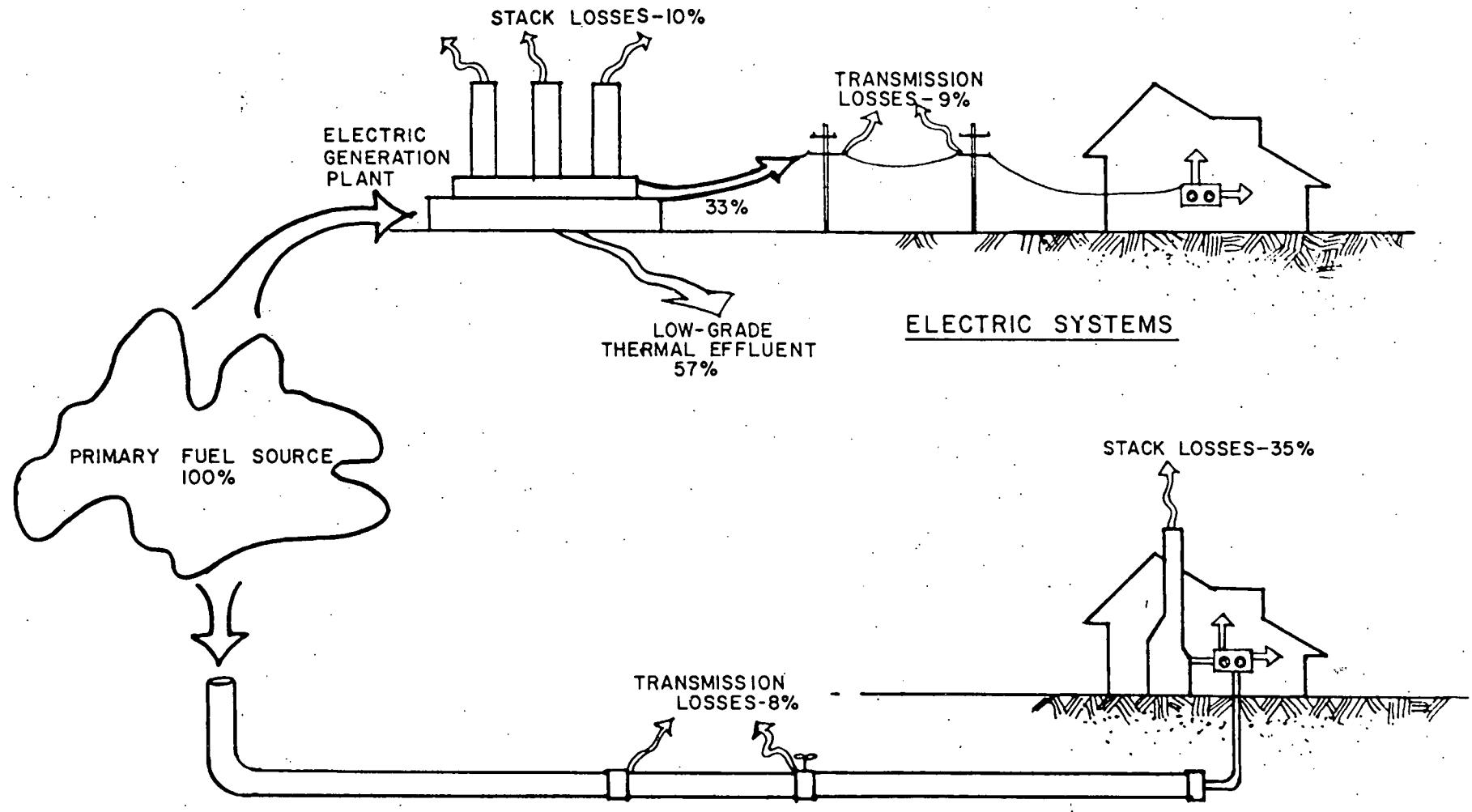
FIGURE 1-2
CONVENTIONAL SYSTEMS
ENERGY FLOW

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 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 waste heat source the HP-WHR scheme seeks to utilize is the community wastewater effluent. This source can include sanitary sewer discharges,

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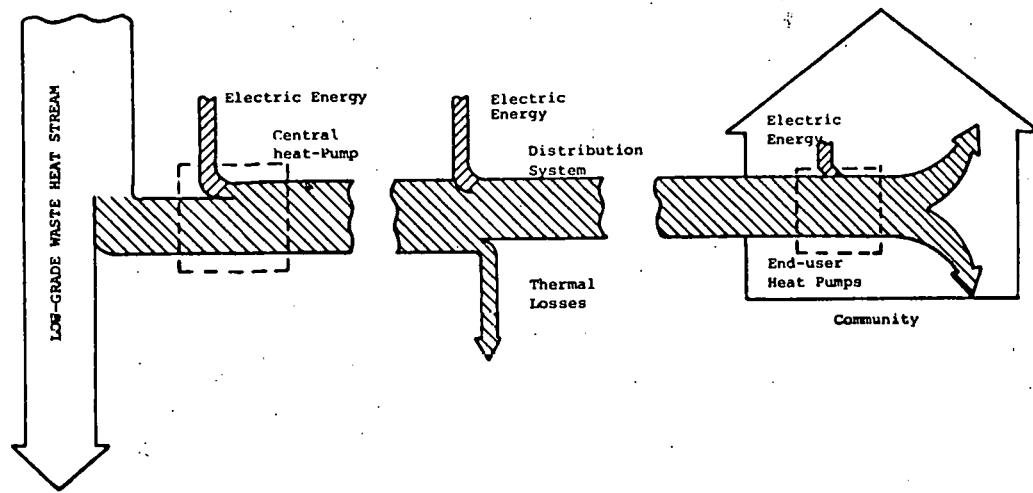
HP-ICES ENERGY FLOW
HEATING MODE

FIGURE I-3
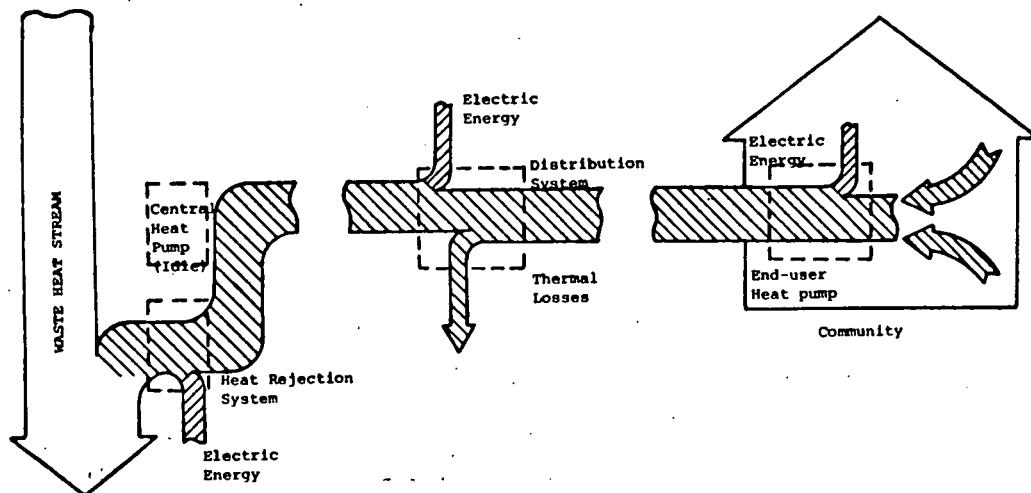
HP-ICES ENERGY FLOW
COOLING MODE

FIGURE I-4

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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 over weekend and holiday periods in areas where there are a large number of industrial discharges.
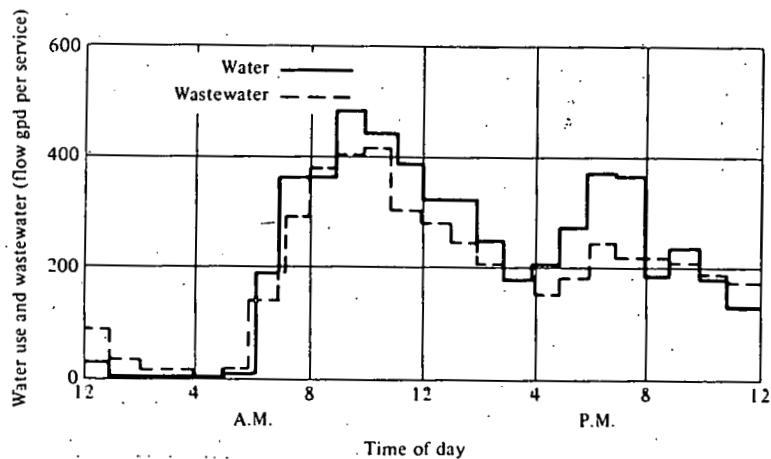
WASTEWATER FLOW CYCLE
FIG. 1-5
SOURCE: WATER SUPPLY & POLLUTION CONTROL[8]

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.

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In predominantly residential communities there will be some concern over the thermal content of the wastewater streams being tapped. Typically, wastewater treatment systems are designed to operate satisfactorily at the lowest wastewater temperatures that might be expected in the particular region in which the treatment facility is to be located. Indeed, a thorough search of literature on wastewater treatment revealed that actual plant influent temperatures are not generally considered or required in the design of treatment systems. Consequently, it is rare for existing treatment plants to record the temperature of the plant wastewater influent. The only regulations now existing deal with the thermal pollution of the body of water receiving a wastewater treatment plant's discharge and thus are meant to restrict the maximum temperature of discharge. Obviously, the operation of an HP-WHR system in conjunction with a wastewater treatment system could significantly affect the operating conditions and thus the treatment efficiency of the wastewater facility. The potential effects are discussed in more detail in Chapter 5.0 - ENVIRONMENTAL IMPACT. In general though, the change in operating temperatures must be given consideration during the design phase of new wastewater treatment plants to properly size biological treatment units or, when exisitng plants are considered for retrofit of an HP-WHR system, to predict lost treatment efficiencies. However, it is anticipated that, as long as the discharge quality and temperature of the effluent from wastewater facilities serving in conjunction with HP-WHR systems meet existing requirements, no further regulations regarding effluent temperature levels will be necessary.

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 8.0 - 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 waste-
water 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 dis-
charges will introduce a variety of contaminants 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 $1\mu M$,
dissolved impurities such as volatiles and minerals and, of course, a con-
siderable 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 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 turbe 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 reciprocat-
ing, 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 accom-
plished 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

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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)
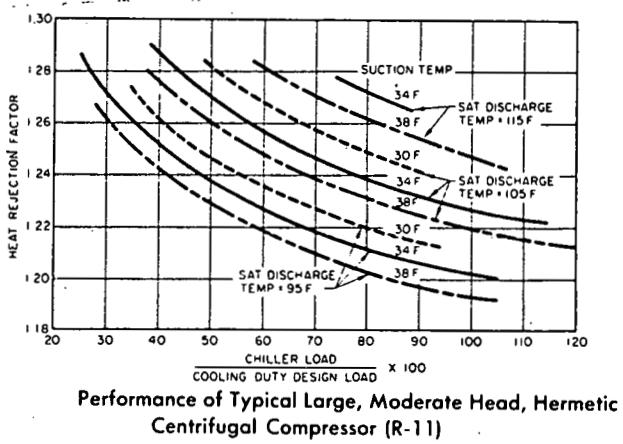
FIG. 1-6

SOURCE: ASHRAE SYSTEMS HANDBOOK[49]

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"; J.E. Christian, 1975.

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 or 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 7.0 - 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.

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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 termperature 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.

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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 1/2 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 reliable system

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COMPONENT DESCRIPTION

The water-to-air heat pump actually is a reverse-cycle, self-contained refrigeration system as shown in Fig. 2.1. 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.
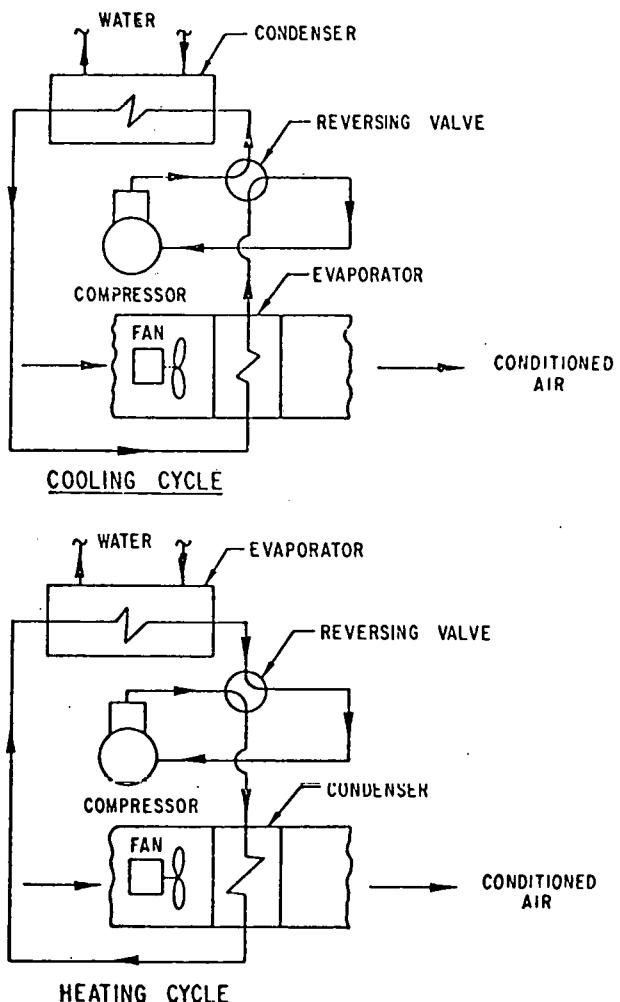
Fig. 2.1 Water-to-Air Heat Pump Showing Cooling and Heating Cycles

ICES TECHNOLOGY EVALUATION

FIG. 1-7
HEAT PUMP CYCLES
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operation. Improved heating performance is especially attractive when compared
to air-source heat pumps due to the absence of the attending defrost cycle 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 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 improve-
ments 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 pro-
cedures 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

20
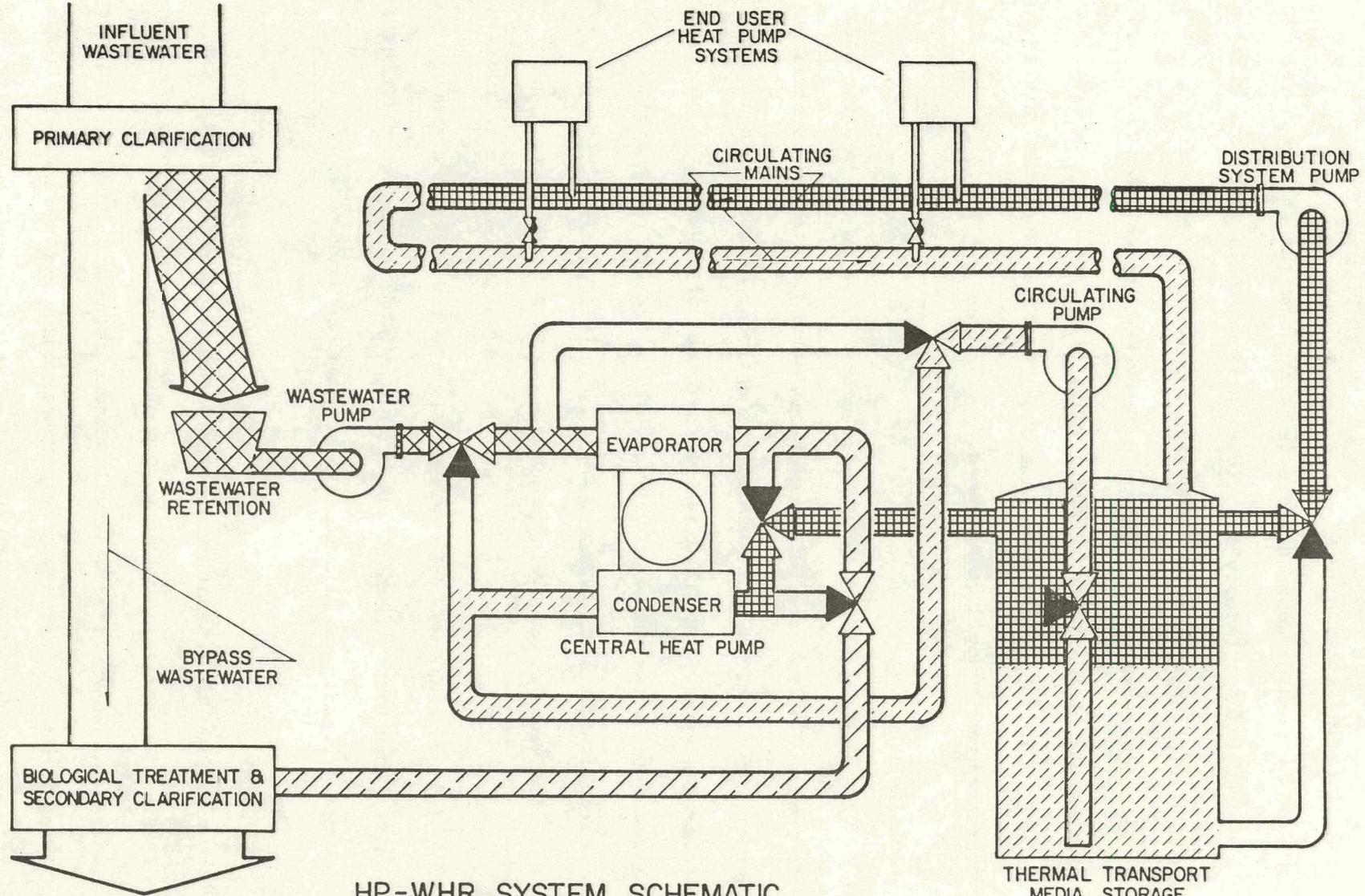
HP-WHR SYSTEM SCHEMATIC
HEATING MODE
FIG. 1-8

stratification will be taken.

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 individual build-
ing loads. Heat would be rejected by the pump to the transport media acting
as the heat sink. Spent 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

INFLUENT WASTEWATER

PRIMARY CLARIFICATION

END USER HEAT PUMP SYSTEMS

CIRCULATING MAINS

DISTRIBUTION SYSTEM PUMP

CIRCULATING PUMP

WASTEWATER PUMP

EVAPORATOR

WASTEWATER RETENTION

CONDENSER

CENTRAL HEAT PUMP

BYPASS WASTEWATER

THERMAL TRANSPORT MEDIA STORAGE

BIOLOGICAL TREATMENT & SECONDARY CLARIFICATION

23
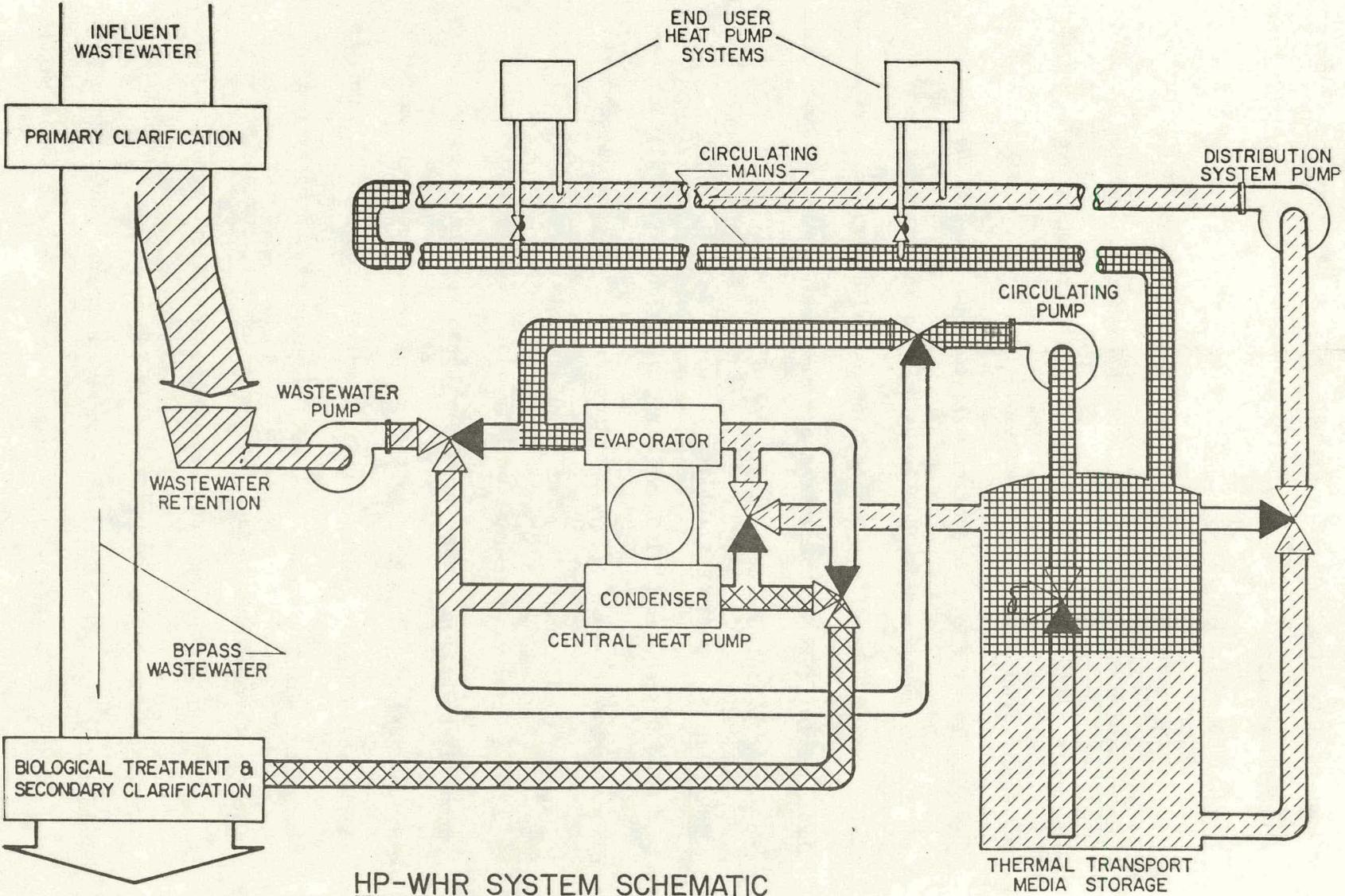
HP-WHR SYSTEM SCHEMATIC
COOLING MODE
FIG. 1-9

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 house-
hold 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 assump-
tion that the energy availability of this source hinges upon is that the tem-
perature 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 in-
vestigation seems to validate this assumption. A quick estimate of the
energy available from this source can be developed from flow and tempera-
ture 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 addi-
tional 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: $[(65° - T_{outside}) \times No. days]$. Figure 2-1 shows the heating degree-day regions in the continental U.S.
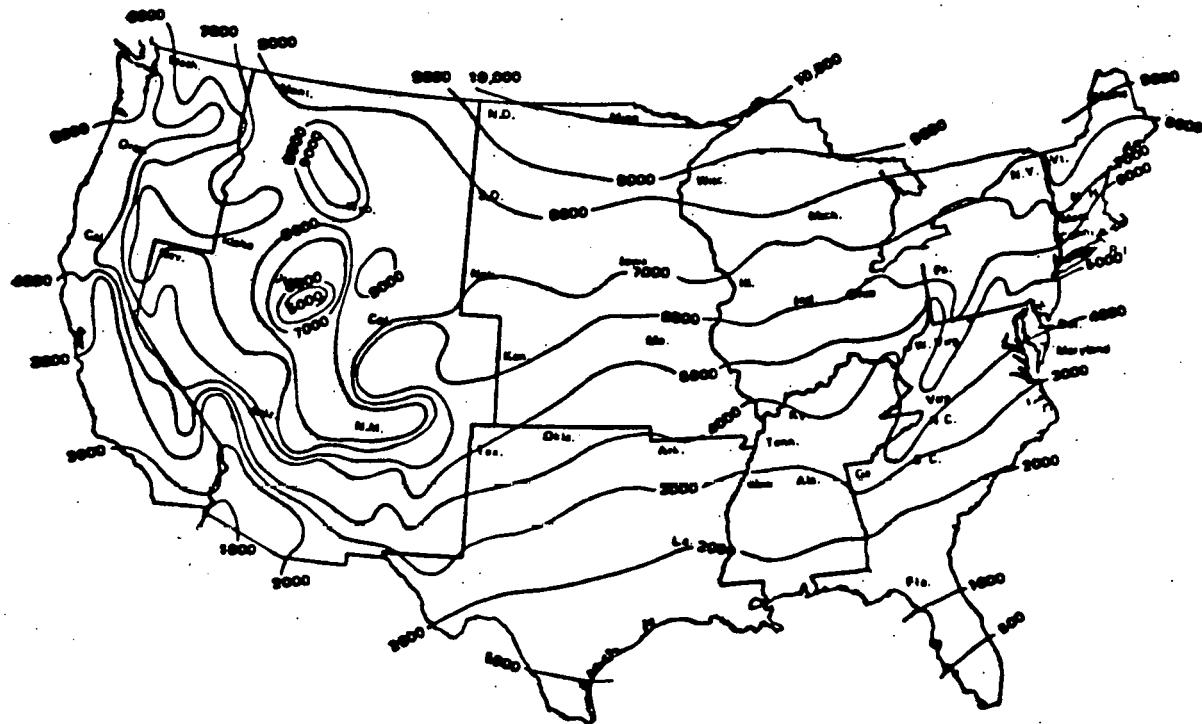
(Heating Degree-Day Map of The United States)

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 condi-

tions. (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 capa-

cities 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 con-

ditioned 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 feasi-

bility 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 ex-

pected 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:

Annualized Cost = (Installed Cost) x (Capital Recovery Factor)

+ (Annual Energy Use) x (Unit Price of Energy)

+ Annual Maintenance

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 central-

izes 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 in-
fluencing 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 com-
pared 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 re-
cent surveys (Arkansas Power & Light Co., 1974,[1] Professional Builder,
1976[26]) that lower operating and maintenance cost expectations were reason-
able justification for purchasing a heat pump, and that an additional in-
vestment of up to $600 for energy conservation would be acceptable if the

payout were within six years.

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 <u>INSTITUTIONAL/COMMUNITY CONSIDERATIONS</u>

Acceptance of an HP-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

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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 and 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 depen-

dent 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 avail-

ability 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, com-

mercial, 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.

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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 waste-

water 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 consump-
                      tion)] + 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

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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:

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

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:

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

Which reduces to:

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

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

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

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 $(\frac{gals}{home\text{-}day})$.

From a technical standpoint it would be possible to size, design, and

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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 waste-water 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 waste-water (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 po-tential for expanding system service area increases correspond-ingly. Thus the additional thermal energy made available by in-creased 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) re-veals that 59.53% of the total U.S. population resides in com-munities of 5000 or more and are connected to a municipal waste-water 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. There-fore, 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 pop-ulation figure in developing the estimate.

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• From the statistical breakdown in TABLE A-4, the number of po-
tential 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 segre-
gated 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, how-
ever, 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 per-
centage 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 pro-
jection 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 (1968) 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%

$$Q_{WASTE} = 3.040 \times 10^6 \ (221,848,000) \ (.5953) \ \frac{Btu}{year}$$
$$= 401.48 \times 10^{12} \ \frac{Btu}{year}$$

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 utilization of this amount of low-grade thermal energy will require an energy expenditure by the system which will be in indirect proportion to the overall system C.O.P. and in direct proportion to the delivered load. This expenditure is also, by definition of system C.O.P., viewed as an energy addition to the total thermal load which is to be delivered to the end-users and is defined by the equation below:

$$Q_{power} = \frac{Q_{WASTE}}{C.O.P._{SYSTEM}}$$

The total load delivered is then the sum of the energy reclaimed, Q_{WASTE}, and the energy expended in the system, Q_{power}, thus:

TABLE 2-1

HEAT EXTRACTED FROM WASTEWATER

YEAR	POPULATION (EST.)	Q/YR BTU x 10^{12}
1980	221,884,000	401
1981	223,661,000	405
1982	225,474,000	408
1983	227,287,000	411
1984	229,100,000	415
1985	230,913,000	418
1986	232,547,200	421
1987	234,181,400	424
1988	235,815,600	427
1989	237,449,800	430
1990	239,084,000	433
1991	240,385,400	435
1992	241,686,800	437
1993	242,988,200	440
1994	244,289,600	442
1995	245,591,000	444
1996	246,610,000	446
1997	247,629,000	448
1998	248,648,000	450
1999	249,667,000	452
2000	250,686,000	454
TOTAL		9041 x 10^{12} BTU

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

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$$Q_{TOTAL} = Q_{WASTE} + Q_{power}$$

$$= Q_{WASTE} + \frac{Q_{WASTE}}{C.O.P._{SYSTEM}}$$

$$+ Q_{WASTE} (1 + \frac{1}{C.O.P._{SYSTEM}}) \qquad (2-4)$$

Substituting the estimate of Q_{WASTE} from TABLE 2-1 and the overall HP-WHR system C.O.P. of 2.245 (from Chapter 3.0 - EXPECTED PERFORMANCE) yields:

$$Q_{TOTAL} = 9041 \times 10^{12} (1 + \frac{1}{2.245})$$

$$= 13,068 \times 10^{12} \text{ Btu}$$

Thus, the utilization of the previously estimated quantity of reclaimed thermal energy is tantamount to the delivery of $13,068 \times 10^{12}$ Btu at an on-site energy expenditure of $Q_{WASTE}/C.O.P._{SYSTEM}$, or 4027×10^{12} Btu (1.18×10^{12} KWH).

This on-site energy expenditure by the HP-WHR system can be translated to primary fuel resource consumption under the following conditions:

ELECTRIC GENERATION EFFICIENCY - Average coal-fired power plant heat

rate of 10,300 Btu/kwh is assumed. Transmission and distribution losses

total 9%. High heating value (HHV) of 12,000 Btu/lbm for coal is

assumed.

Primary fuel resource consumption in the form of coal is thus $13,248 \times 10^{12}$

Btu or 552 million (552×10^{6}) tons for the delivery of the estimated 13.1

QUAD thermal load.

The above HP-WHR system energy consumption can be compared to that required

for the delivery of a comparable thermal load via the conventional space

conditioning system options. The following assumptions are in effect for the

various options:

    ELECTRIC STRIP HEATING UNITS - Heating C.O.P. is 1.0, and the above

mentioned assumptions for electric generation efficiency are also in

effect.

    FUEL-OIL FIRED FURNACE - Combustion and thermal transfer efficiency

is taken as 65%; HHV for fuel oil is assumed at 140,000 Btu/gal; no

transmission or distribution losses.

    NATURAL GAS-FIRED FURNACE - Combustion and thermal transfer efficiency

is taken as 65%; HHV for natural gas is assumed constant at 1000 Btu/CFT;

8% transmission and distribution losses.

Under the above assumptions, the electric strip heat option would require

42,987 x 10^{12} coal Btu's, or 1791 million (1791 x 10^6) tons. The HP-WHR

system performance thus represents a 69% reduction in coal consumption in

meeting the equivalent thermal load.

For the operation of fuel oil-fired furnaces in meeting the approximate

13.1 QUAD thermal load, the primary fuel resource consumption would amount

to 20,105 x 10^{12} fuel oil Btu's, or 3.42 billion barrels. The HP-WHR system

performance in comparison to this option represents a 35% reduction (on a Btu-

to-Btu basis) in consumption of primary fuel resources. It should also be

noted that operation of the heat pump system would effectively force a shift

in consumption from the more scarce fuel oil resources to the relatively

abundant coal resources.

For the operation of natural gas-fired furnaces in meeting the estimated

13.1 QUAD thermal load, the primary fuel resource consumption would amount

to 21,713 x 10^{12} natural gas Btu's, or 21.7 trillion cubic feet. In

comparison, the HP-WHR system performance represents a 39% reduction (on a

Btu-to-Btu basis) in consumption of primary fuel resources. For this option,

as for the fuel oil-fired furnace option, the operation of the heat pump

system would force a significant shift away from consumption of scarce fuel resources.

In summary, the HP-WHR system exhibits the potential for conserving from 35% to 69% of the scarce fuel resources consumed in meeting the approximate 13.1 QUAD thermal load which potentially could be handled through wastewater heat reclaim. Primary fuel source savings can be quantified at:

	COAL	– 29.74 QUADS
or	FUEL OIL	– 6.86 QUADS
or	NATURAL GAS	– 8.47 QUADS

Fuel savings are of course dependent upon the type of conventional space heating system which would be displaced with the HP-WHR system.

It is duly noted that, while the estimate takes into account the full potential for wastewater heat reclaim for the 20 year time period between 1980 and 2000, it is highly improbable that HP-WHR systems could be installed and operational by the 1980 starting date of the study period and that wide-spread implementation of the system concept would be unlikely, even with a successful "fast-track" demonstration, before perhaps 1990. Obviously though, from the above 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.

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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 of the heating mode, 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 analysis of 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, and after delivering heat to the end-user heat pumps, the water is then returned to the central unit. A reversal of heat flow 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 a central heat pump are presented in Section 3.4 - ANALYSIS OF RESULTS along with the results of a distributed water-to-air heat pump system deleting the central heat pump. These 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

CLARIFIED WASTEWATER HEADER

RETURN TO TREATMENT PLANT

CENTRAL STATION HEAT PUMP

Q_e

EVAPORATOR

COMPRESSOR

W_c

EXPANSION VALVE

CONDENSER

Q_c

END USER HEAT PUMP

CONDENSER

RECIRCULATED COLD AIR

Q_c

WARM AIR

COMPRESSOR

W_c

EXPANSION VALVE

EVAPORATOR

Q_e

RETURN HEADER

THERMAL STORAGE

DISTRIBUTION HEADER

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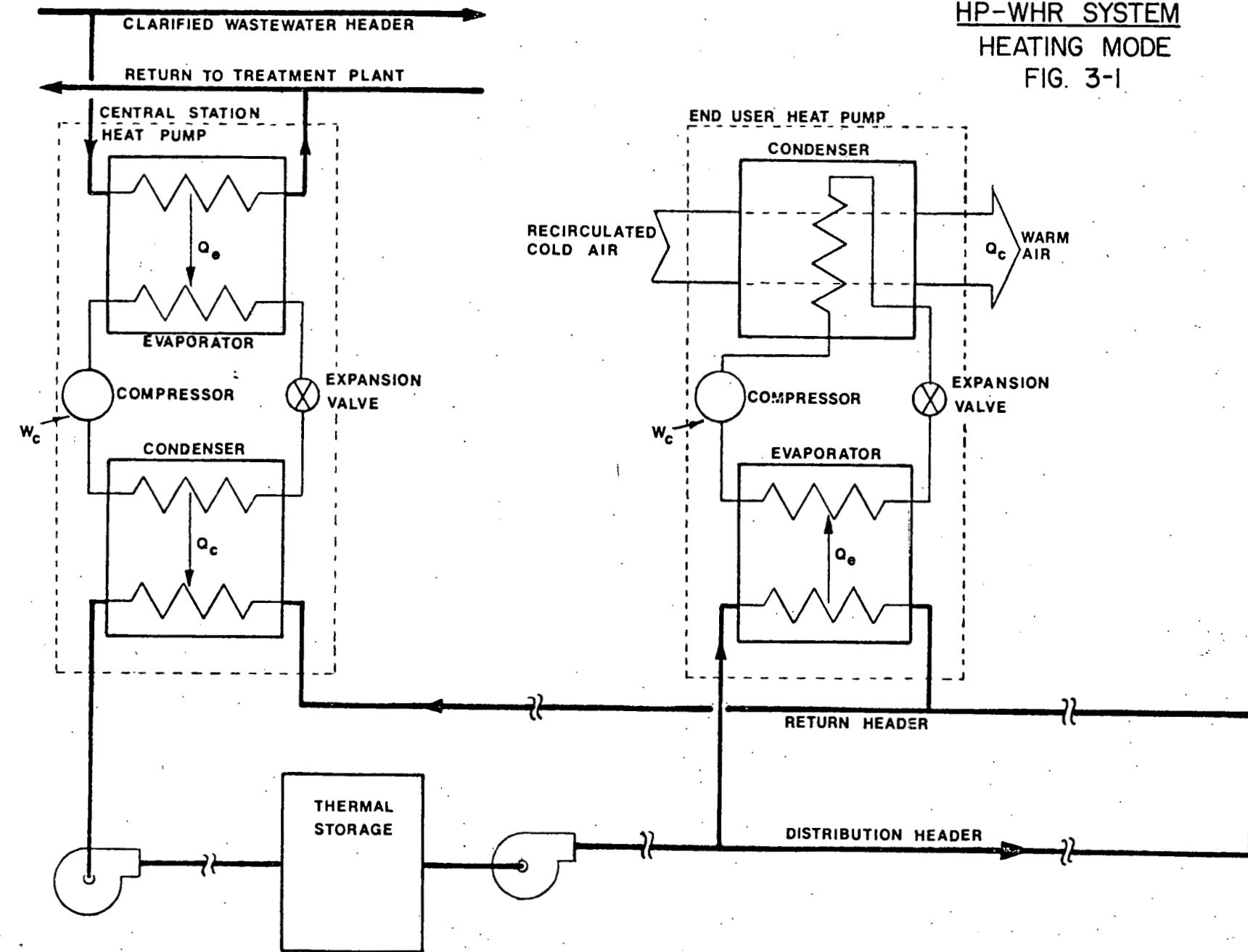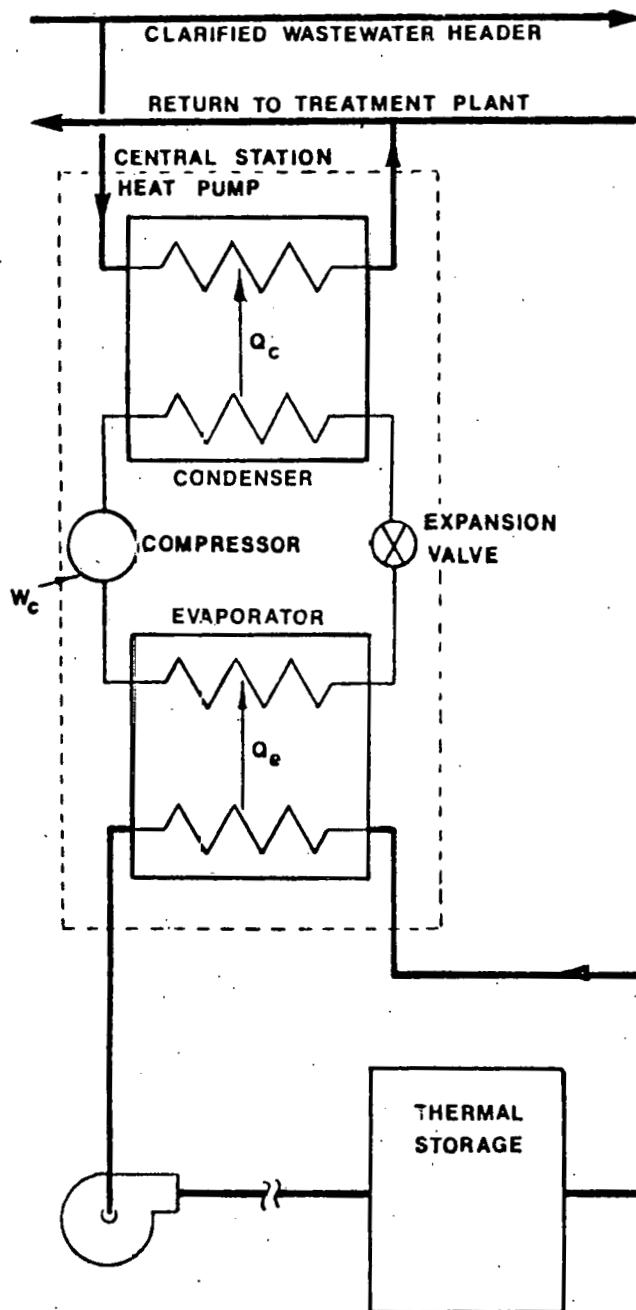
HP–WHR SYSTEM
COOLING MODE
FIG. 3-2
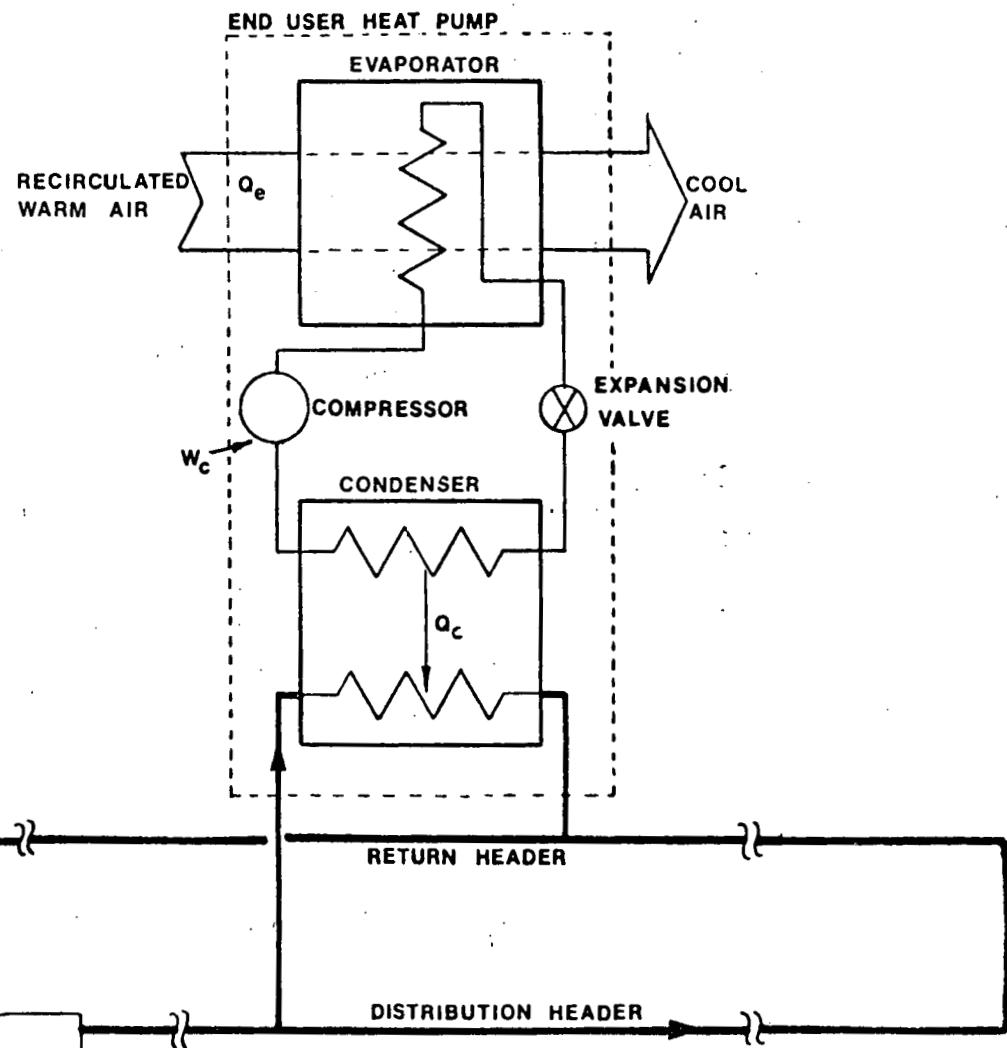
air temperature. The wastewater effluent temperature depends on factors such as the time of day, ground temperature, outside air temperature, and the presence of thermal discharges from industrial processes. There is also the time-lag associated with the temperature change of the reservoirs or lakes 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:

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

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

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.
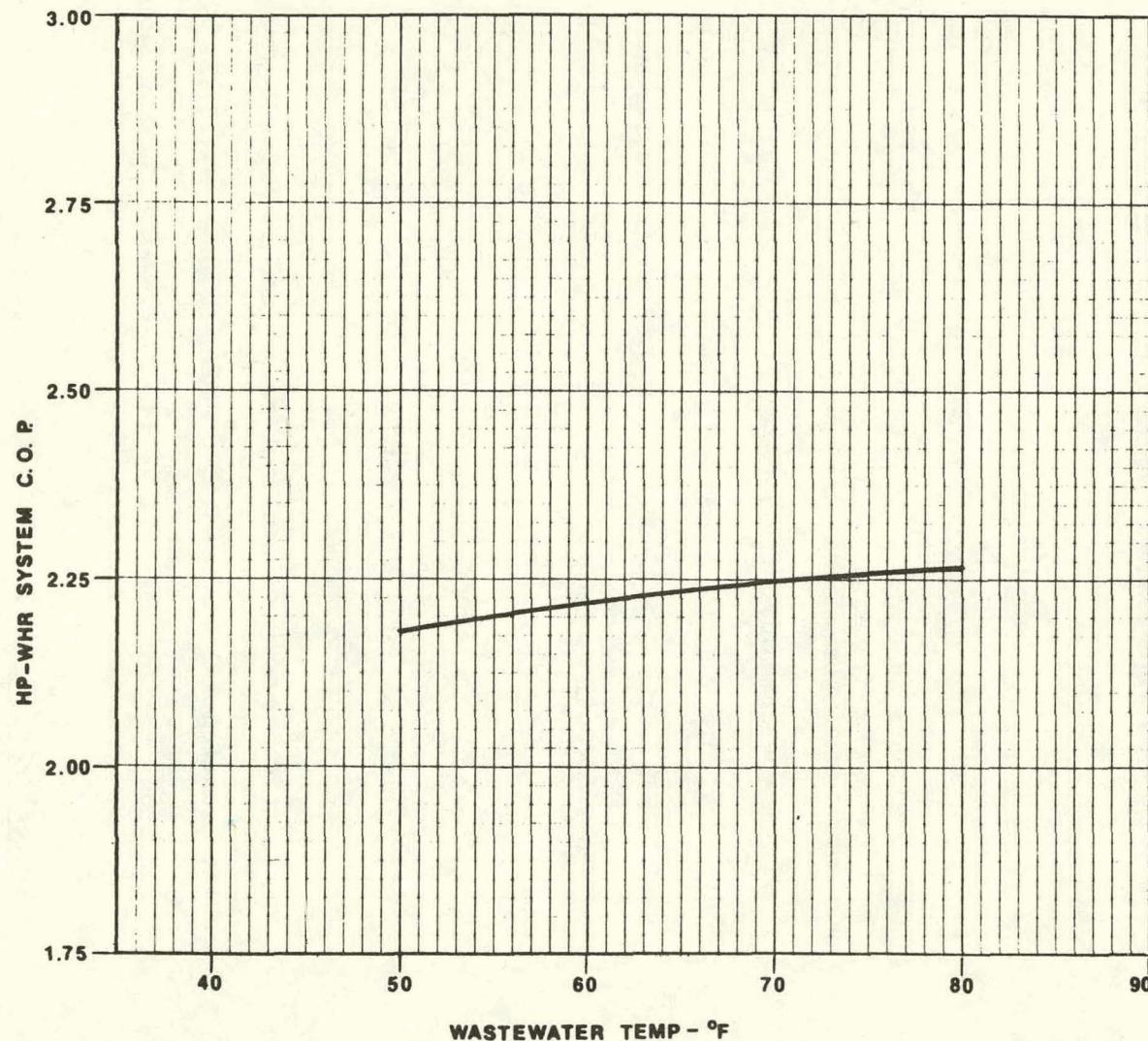
ENTERING WATER TEMPERATURE AT
END USER UNIT 80°F

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

60

FIG. 3-3

WASTEWATER TEMPERATURE vs HP-WHR SYSTEM C. O. P.
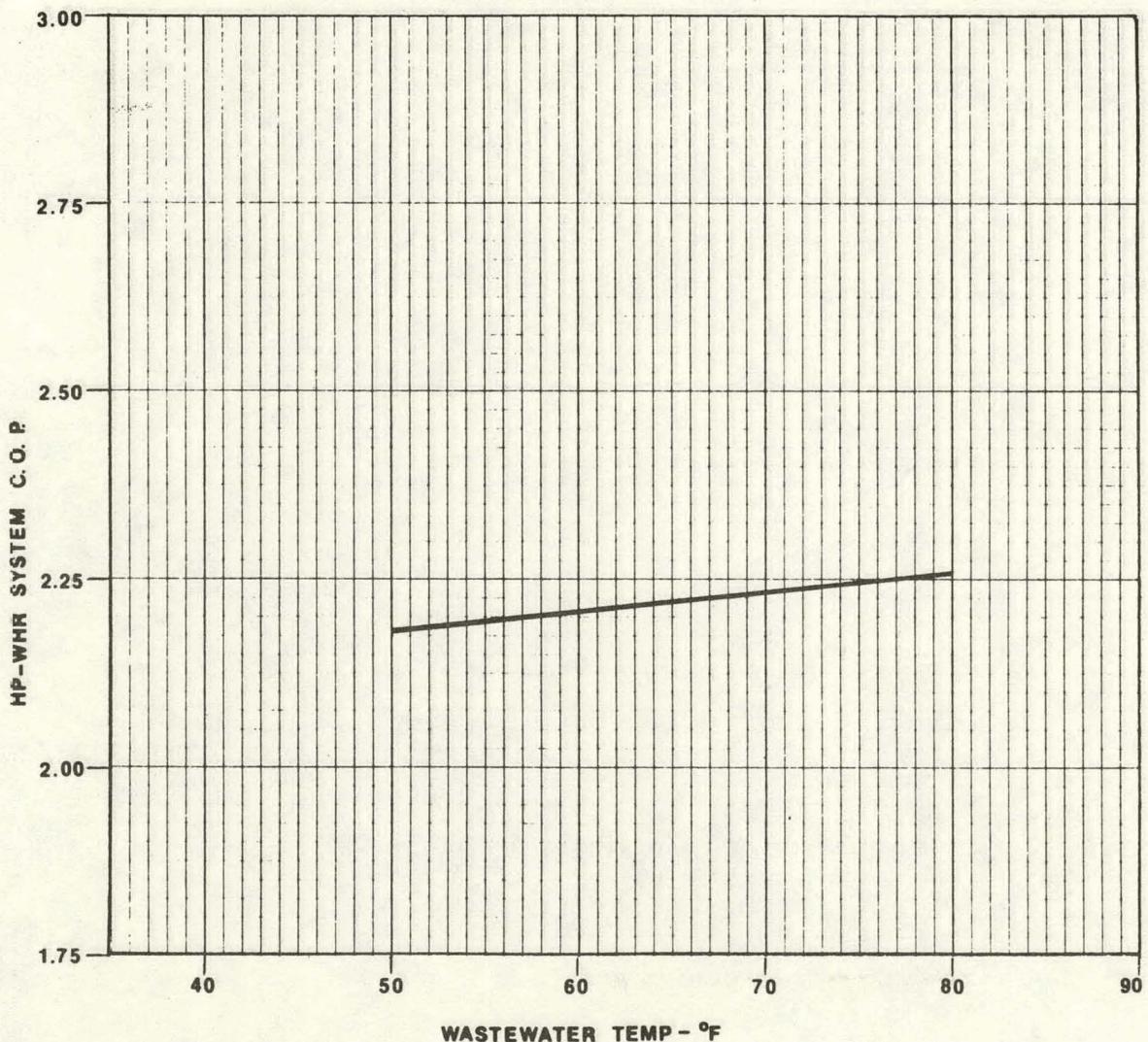
ENTERING WATER TEMPERATURE AT END USER UNIT 95 °F

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	$E_{System} \dfrac{Btu}{Btu}$	$E_{Primary\ Fuel\ Energy} \dfrac{Btu}{Btu}$
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 Unitary Air-to-Air Heat Pumps.[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	$E_{System} \dfrac{Btu}{Btu}$	$E_{Primary\ Fuel\ Energy} \dfrac{Btu}{Btu}$
60	40	1.913	0.523	1.686
80	40	1.874	0.534	1.721
80	50	1.843	0.543	1.750

65

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	NWC	47210	3861	51071	164745
HEAT PUMP	SA	16478	14077	30555	98564
CONVENTIONAL					
FUEL	NWC	155717	12454	168171	168171
GAS (WARM-AIR)	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.
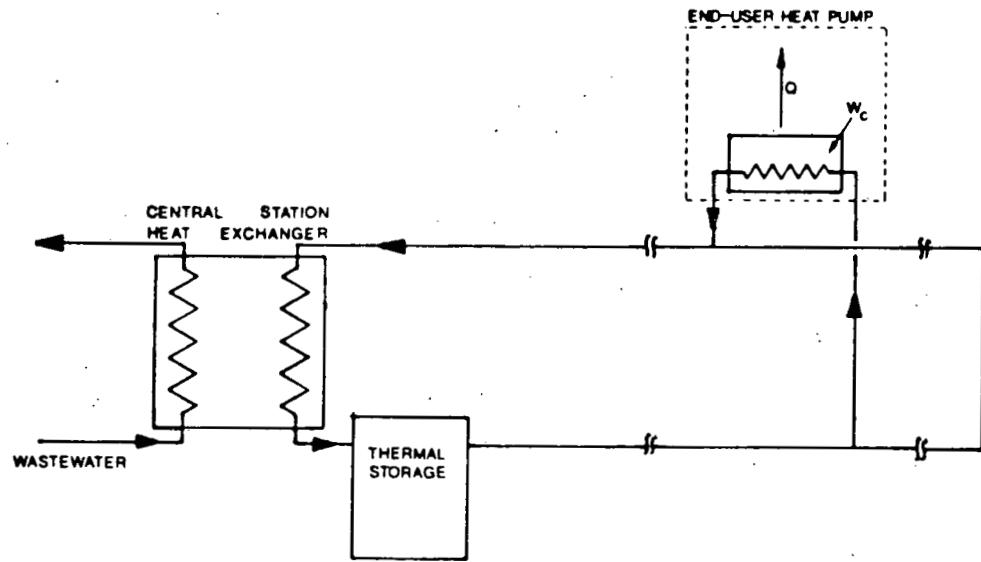
DISTRIBUTED WATER SOURCE HEAT PUMP SYSTEM

FIG. 3-5

TABLE 3-5
DISTRIBUTED HEAT PUMP SYSTEM
C.O.P. VS. EFFLUENT TEMPERATURE

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 SECOND LAW ANALYSIS

The previous performance analyses have used the first law of thermodynamics to determine the required system energy input. These analyses have been characterized by first law efficiencies defined as: Useful Energy Output divided by Required Energy Input. In the case of the HP-ICES system this analysis determines an overall system coefficient of performance. As expected, this analysis yields values greater than unity.

A second law thermodynamic analysis must be performed to determine the system's true thermodynamic efficiency. The second law recognizes that energy is characterized by quality as well as quantity. By comparing second law efficiencies a true determination of resource utilization efficiency can be made.

To recognize energy quality, the concept of availability is used. Availability is the thermodynamic property which measures the potential of a system to do work when restricted by surroundings at some ambient condition. Availability is not conserved. It is destroyed by friction and heat transfer through a finite temperature difference. This destruction of availability is the true thermodynamic loss which causes a process to be irreversible.

The availability analysis for the HP-ICES schemes utilizing wastewater heat recovery center on the system composed of a central heat pump or heat exchange device, distribution system and storage and end-user heat pumps. The energy and process streams crossing the boundaries of the system consist of the following:

1) The wastewater heat source entering at ambient conditions.

2) Electricity input to the central heat pump or heat exchange device.

3) Thermal losses from the distribution and storage system.

4) Electricity input to circulating water pumps on the distribution system.

5) Electricity input to the end-user heat pumps.

6) Heat or cooling energy in the form of cool or warm air to the end-user.

7) Miscellaneous parasitic electricity requirements.

The analysis of the system must also take into account the second law thermodynamic efficiency of the electric generating plant supplying electricity to the HP-ICES.

This second law analysis of the HP-ICES system is somewhat limited since the central heat pump and end-user heat pumps must be considered as "black-boxes". It is obvious from a review of the literature that the heat pump devices within the system inherently destroy availability at their component heat exchangers, compressors, and throttling valves. Significant improvement in the overall system performance can be achieved by optimizing the internal components used for end-user heat pumps and central station heat pumps. An optimization of individual heat pump components was not undertaken in this study since that would constitute a heat pump design rather than a system application.

An availability analysis of a conventional system consisting of a natural gas space heater and electric air conditioning system was also made for comparative purposes.

3.6.1 ANALYSIS PROCEDURE

For the system analyses it was recognized that the objective of the system was to provide heating or cooling for a space. This objective is

accomplished by providing an appropriate quantity of warm or cold air to the space. For practical reasons a supply air temperature of 120°F was selected for heating and 55°F was selected for cooling. The space temperature was assumed to be 75°F, which is also considered the reference temperature for energy availability.

For the systems considered, the available energy balance is:

Available Energy Transferred into the system	=	Available Energy Destruction Within the system	+	Available Energy Transfer Out of the system

The availability of the energy contained in the air stream to the space is considered the desired output of the system and an indicator of the minimum (theoretical) amount of energy needed for space conditioning.

The ratio of the desired availability to the availability of the energy inputs to the system is termed the second law efficiency.

3.6.2   CONVENTIONAL SYSTEMS

A conventional system schematic composed of a natural gas heater and electric air conditioner is presented in Figure 3-6. A second law analysis per unit input of primary fuel yields a second law heating efficiency of 2.6%, assuming a first law efficiency of 65 percent.

With a C.O.P. of 2.2 the electric air conditioning system has a second law efficiency of 1.38%.

3.6.3   CONVENTIONAL SYSTEM PERFORMANCE

HEATING

1st Law (efficiency - 65%)

PRIMARY
FUEL

NATURAL GAS
SPACE HEATER

AIR AT
120 °F

HEAT TO
END-USER

AVAILABILITY= 1.0 Btu/UNIT

AVAILABILITY = .0258 Btu/UNIT

HEATING MODE

AVAILABILITY = .0194 Btu/UNIT

HEAT REJECTED
TO ATMOSPHERE

AIR CONDITIONING
SYSTEM

AIR AT
55 °F

HEAT FROM
END USER

PRIMARY FUEL

ELECTRIC
GENERATING
PLANT

ELECTRICTY

AVAILABILITY =
1.0 Btu/UNIT

AVAILABILITY = .33 Btu/UNIT

AVAILABILITY = .45 Btu/UNIT

ELECTRICITY
INPUT

LOSSES

COOLING MODE

FIGURE 3-6

CONVENTIONAL
HEATING/COOLING SYSTEM
AVAILABILITY ANALYSIS

Primary fuel in $= 1$ Btu/hr

120°F air out $= .65$ Btu/hr

$$\dot{M}_{\text{air out}} = \frac{.65}{.24(120-75)} = .060 \text{ lb/hr}$$

2nd Law (Availability relative to 75°F)

Availability of fuel input $= 1$ Btu/hr

Availability of 120°F air out $= \dot{M}(C_p)[T-T_o-(T_o \ell n \frac{T}{T_o})]$

$$= .060(.24)[580-535-(535 \ell n \frac{580}{535})]$$

$$= .0258 \text{ Btu/hr}$$

Second Law Efficiency $= \dfrac{.0258}{1.00} = .0258$

COOLING

1st Law (C.O.P. $= 2.2$, Electric Gen. Eff $= 33\%$)

Electricity to heat pump $= .454$ Btu/hr

55°F air heat removed $= 1.0$ Btu/hr

$$\dot{M} \text{ of } 55^{\circ}\text{F air*} = \frac{1.0}{.24(75-55)} = .208 \text{ lb/hr}$$

* Assumes Dry Air

2nd Law (Availability relative to 75°F)

Availability of primary fuel input $= (.454) \div (.33) = 1.377$

Availability of 55°F air out $= .208(.24)[515-535-(535 \ell n \frac{515}{535})]$

$$= -.0191 \text{ Btu/hr}$$

Second Law Efficiency $= \dfrac{.0191}{1.377} = .0138$

6.4 HP-ICES WASTEWATER HEAT RECOVERY SCHEME

HP-WHR system heating and cooling schematics are presented in Figures 3-7 and 3-8. The second law analysis for the system presented yields a heating efficiency of 3.0% and a cooling efficiency of .94% at an overall system C.O.P. of 2.2.

Significant improvements can be made in the system if individual component performance is improved. This is especially true where the end-user heat pump is concerned.

HP-ICES WASTEWATER HEAT RECOVERY SCHEME

HEATING

1st Law Balance

Waste heat input source	=	1 Btu/hr
Electricity input to central heat pump	=	.1538 Btu/hr
Electricity input to pumps, etc. (assume 3% of input)	=	.0300 Btu/hr
Heat losses to surroundings from distribution system (assume 3% of input)	=	.0300 Btu/hr
Electricity input to end-user heat pump	=	.5769 Btu/hr
Output to space @ 120°F air	=	1.7307 Btu/hr

$$\dot{M} \text{ air out} = \frac{1.7307}{.24(120-75)} = .16025 \text{ lb/hr}$$

2nd Law (Availability relative to 75°F)

$$\text{Availability of } 120°F \text{ air} = .16025(.24)[580-535-(535\ln\frac{580}{535})]$$

$$= .06894 \text{ Btu/hr}$$

WASTE HEAT SOURCE

WASTEWATER AT 60 °F

CENTRAL HEAT PUMP
COP = 7.5

DISTRIBUTION AND STORAGE

END-USER HEAT PUMP
COP = 3.0

AIR AT 120 °F

HEAT TO END-USER

THERMAL LOSSES

HP-ICES SYSTEM BOUNDARY

ELECTRICITY INPUT

PUMP ELECTRICITY INPUT

ELECTRICITY INPUT

HEATING MODE

ELECTRIC GENERATING PLANT

PRIMARY FUEL

ELECTRICITY

LOSSES

AVAILABILITY = 1.0 Btu/UNIT

AVAILABILITY = .33 Btu/UNIT
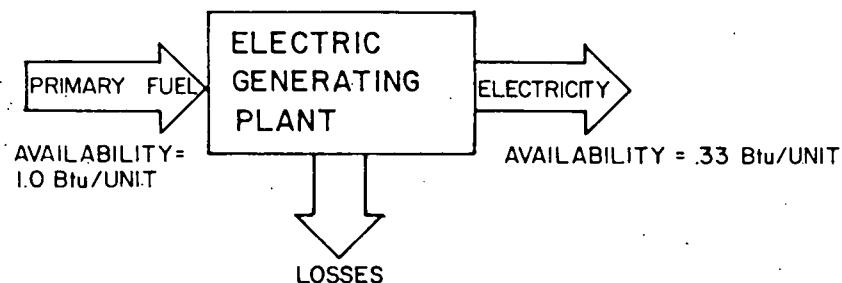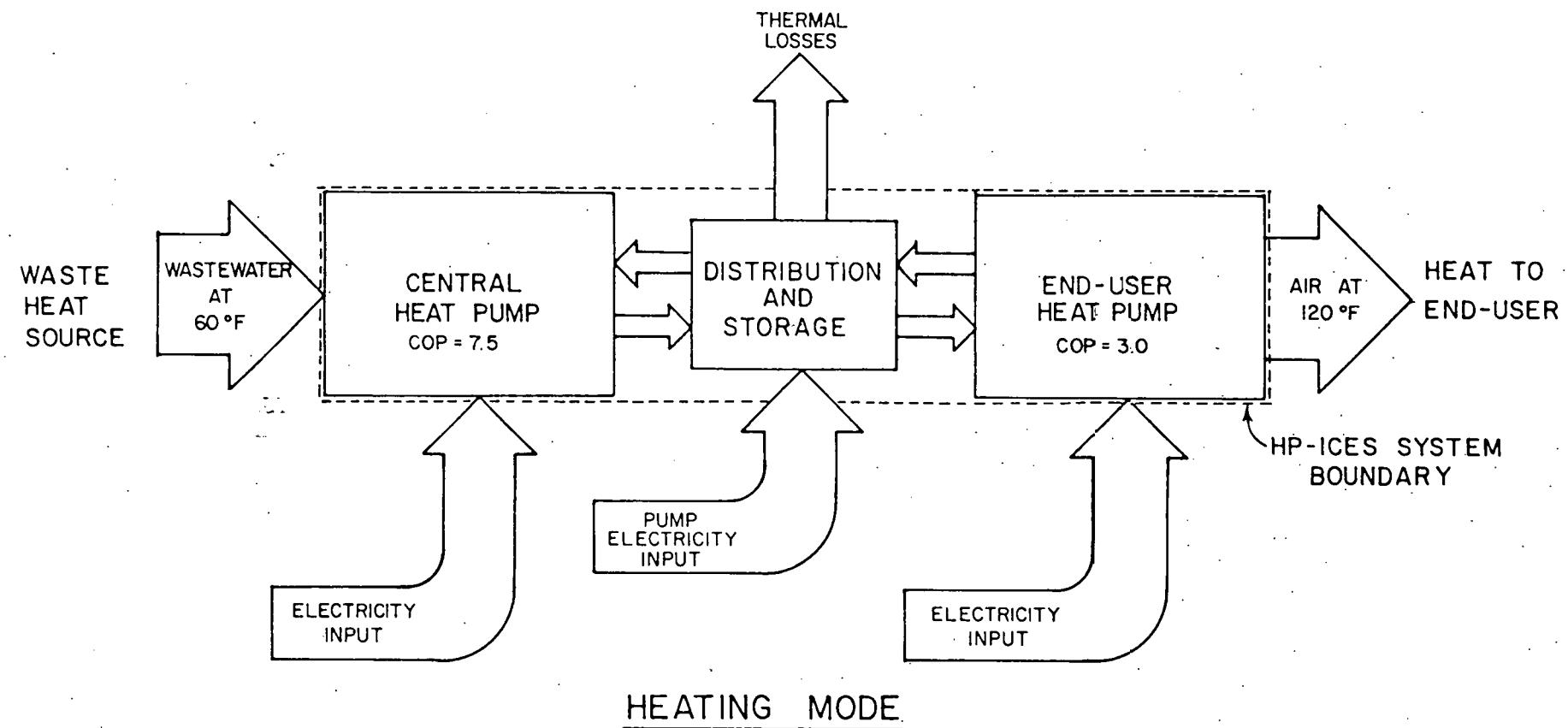
FIGURE 3-7

HP-ICES WASTEWATER HEAT RECOVERY SCHEME AVAILABILITY ANALYSIS

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COOLING MODE
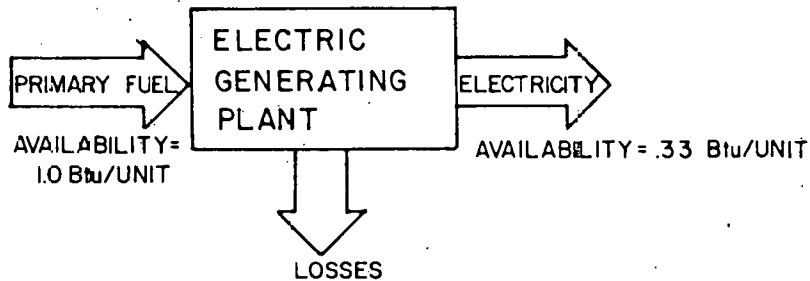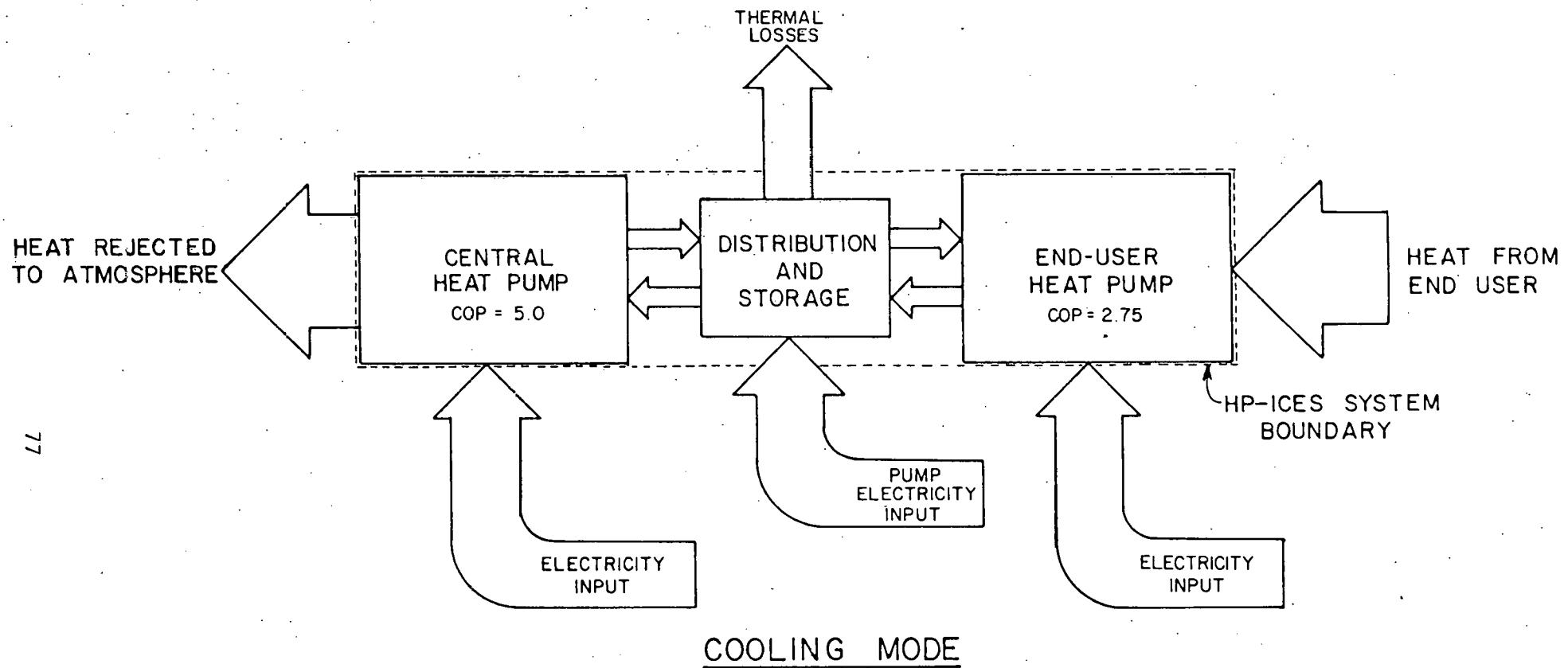
FIGURE 3-8

HP-ICES WASTEWATER
HEAT RECOVERY SCHEME
AVAILABILITY ANALYSIS

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Availability of energy inputs
(primary fuel)

      Electricity to Central Heat Pump   =   .1538

      Electricity to Pumps   =   .0300

      Electricity to End-User Heat Pump = $\dfrac{.5769}{.7607}$ ÷ .33 = 2.305 Btu/hr

Second Law Efficiency   = $\dfrac{.06894}{2.305}$ = .0299

COOLING

1st Law Balance

Heat from end-user   = 1 Btu/hr

Electricity input to   = .363 Btu/hr
end-user heat pump

Electricity input to   = .03 Btu/hr
pumps, etc.

Heat losses to surround-   = .03 Btu/hr
ings

Electricity input to   = .272 Btu/hr
central heat pump

$\dot{M}$ of air to space @ $55°F$ = $\dfrac{1}{.24(75-55)}$ = .20833 lb/hr

2nd Law (Availability relative to $75°F$)

Availability of $55°F$ air   = $.20833(.24)[515-535-(535\ln\frac{515}{535})]$

          = -.0191 Btu/hr

Availability of energy inputs
(primary fuel)

      Electricity to Central Heat Pump   =   .272 Btu/hr

      Electricity to Pumps   =   .03

      Electricity to End-User Heat Pump = $\dfrac{.363}{.665}$ ÷ (.33) = 2.015 Btu/hr

Second Law Efficiency   = $\dfrac{.0191}{2.015}$ = .0094

SUMMARY

In summary, it is the belief of the authors that this chapter demonstrates the HP-WHR scheme as a potentially viable energy supply alternative for some specific applications. The actual determination of feasibility and expected performance is highly dependent on site-specific physical characteristics and local economic conditions, and will be addressed in the application study of Chapters 8.0 and 9.0 of this report.

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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.[27] 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		Annual Energy Requirements*(MBtu)	
			Heating	Cooling	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

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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.

Northern Climate

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 (run-outs to end-users not included)	61,100 ft
Longest Piping Run (Supply & Return)	9,900 ft
Number of End-Users	470
Distribution Pump Horsepower	450 hp

Southern Climate

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

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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

Northern Climate

	Kwh/Yr.	Cost/Yr.
Cooling	1,915	$115
Heating	11,236	674
TOTAL	13,151	$789

Southern Climate

	Kwh/Yr.	Cost/Yr.
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

ELECTRICAL ENERGY COST PER END-USER

DOLLARS x 100

4¢ PER KWH

3¢ PER KWH

2¢ PER KWH

SOUTHERN CLIMATE

NORTHERN CLIMATE

FIG. 4-1

TOTAL HP-WHR UTILITY COST PER END-USER

DOLLARS x 100

ELECTRICAL ENERGY COST
COMMERCIAL/RESIDENTIAL

4¢/6¢ PER KWH

3¢/4.5¢ PER KWH

2¢/3¢ PER KWH

SOUTHERN CLIMATE

NORTHERN CLIMATE
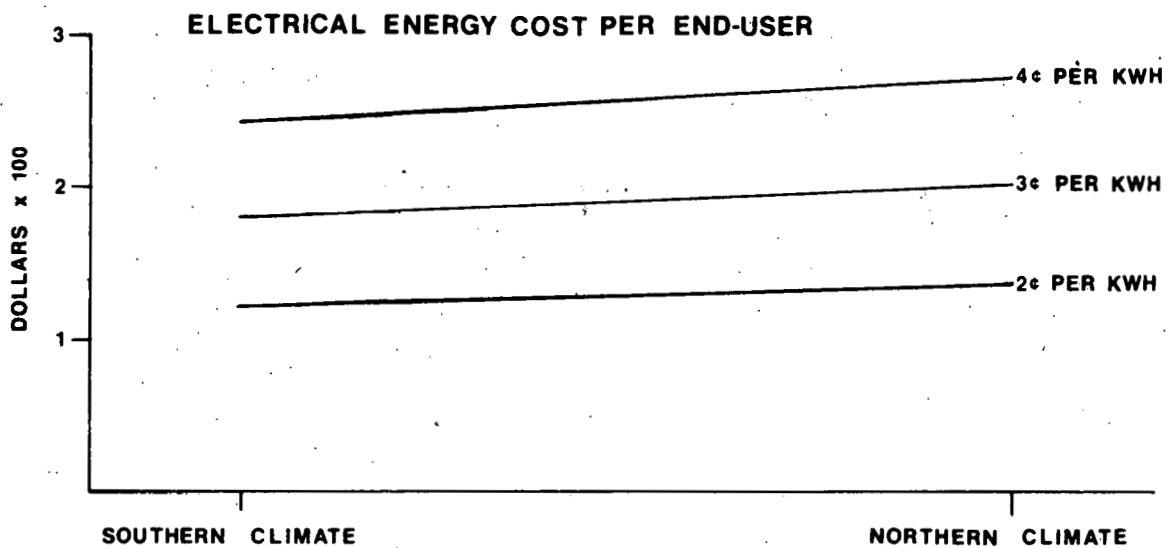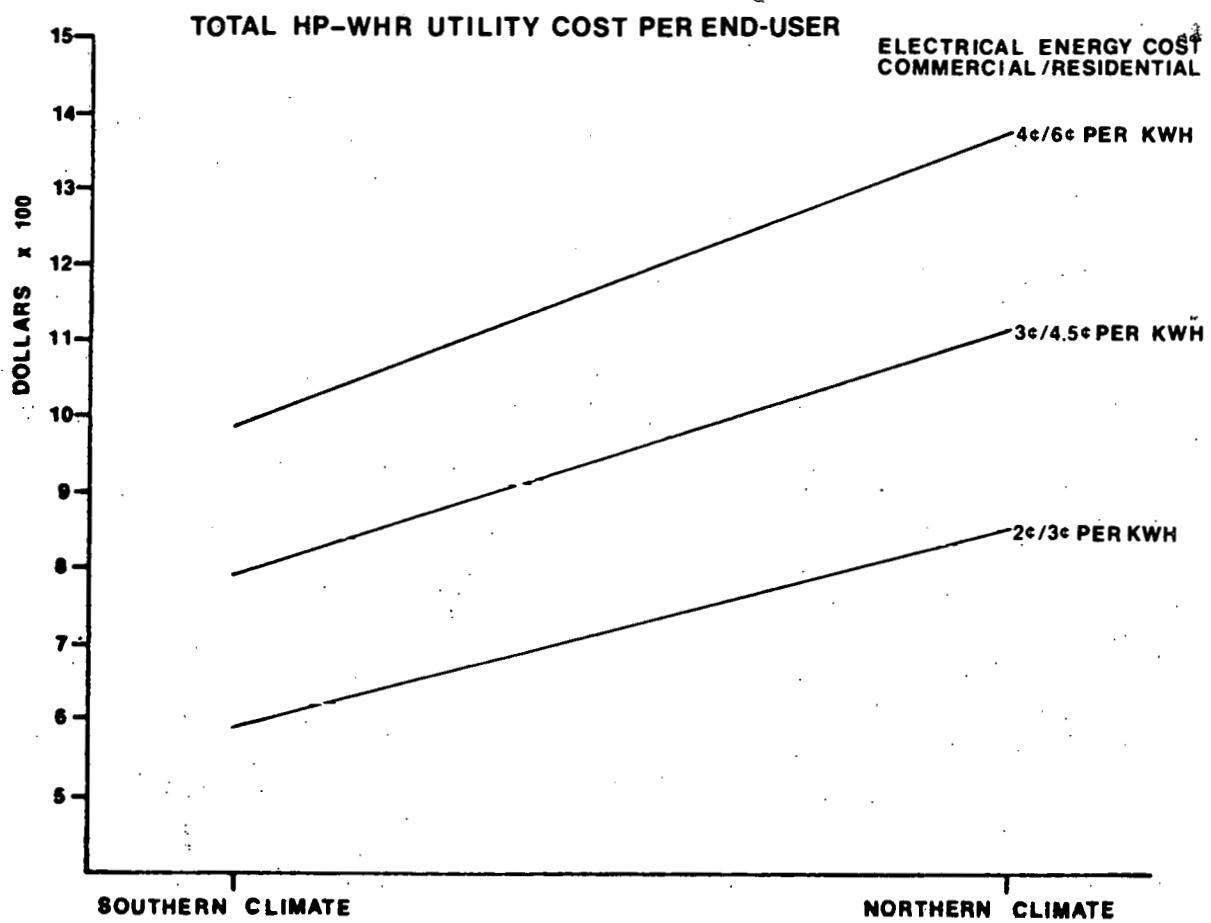
FIG. 4-2

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 | | COOLING | |
	Energy Consumption	Energy Cost	Energy Consumption	Energy Cost
NATGAS/Elect Cooling	1622.5 therms	$ 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

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

FIG. 4-3

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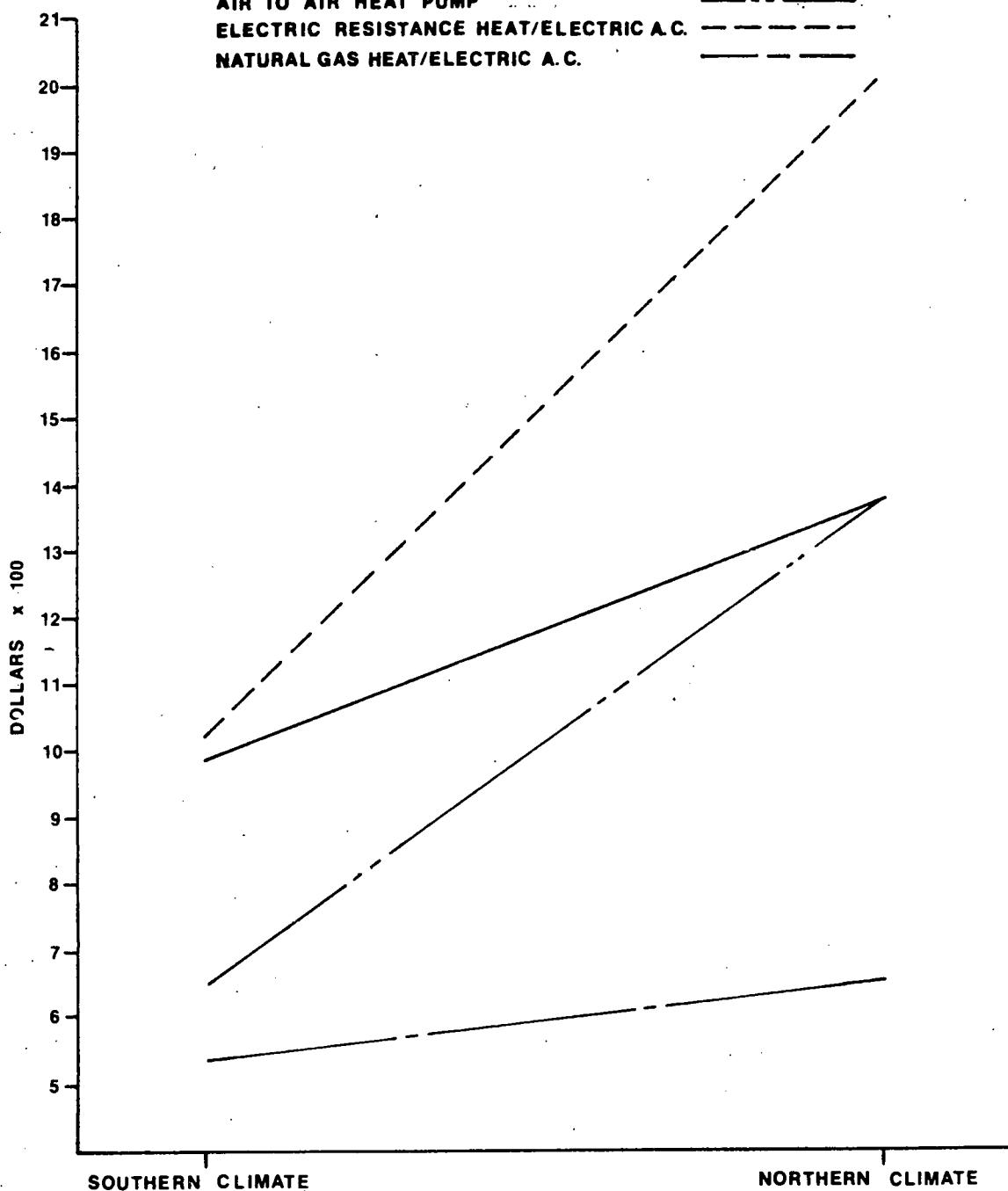
Table 4-4 (cont)

Southern Climate

| System | HEATING | | COOLING | |
	Energy Consumption	Energy Cost	Energy Consumption	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:

Northern Climate (serving 470 customers)

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

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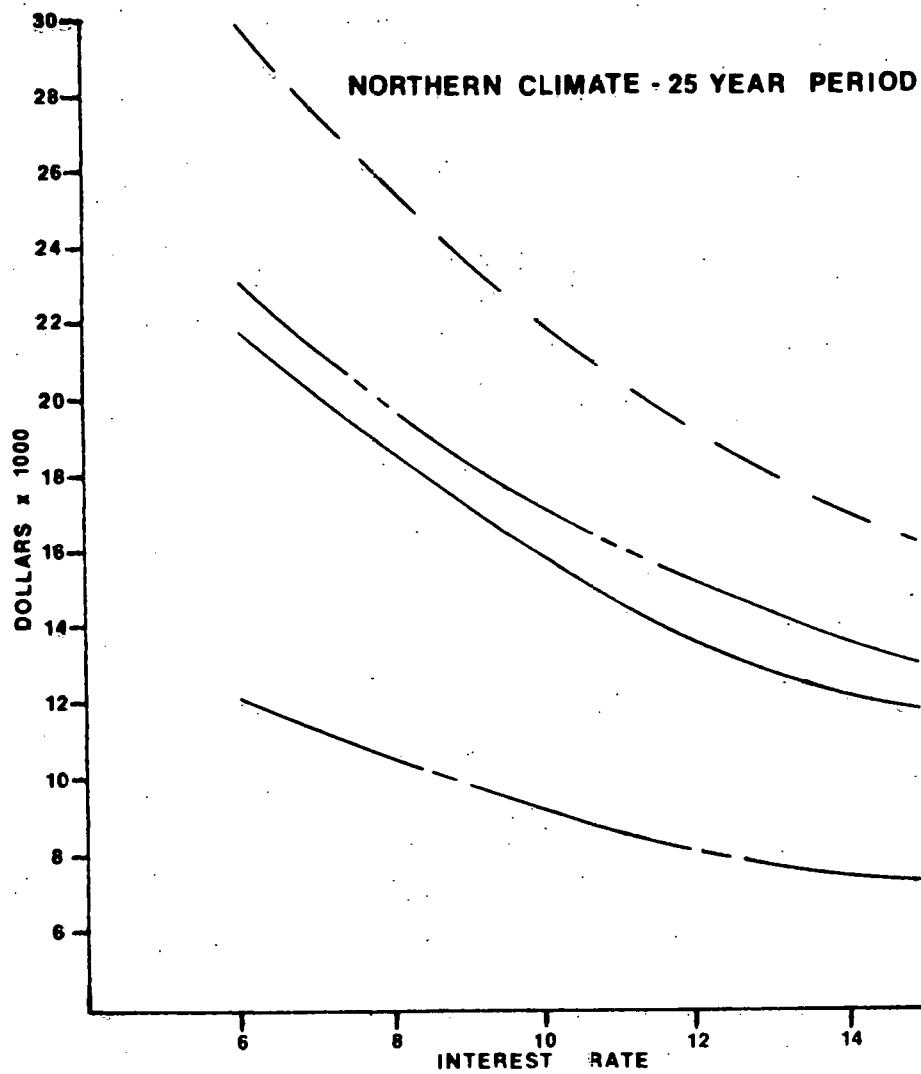
NOTE: COMMERCIAL ELECTRICITY AT 4¢/KWH
         RESIDENTIAL ELECTRICITY AT 6¢/KWH
         NATURAL GAS AT 30¢/THERM

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

NORTHERN CLIMATE - 25 YEAR PERIOD

SOUTHERN CLIMATE - 25 YEAR PERIOD

FIG. 4-4
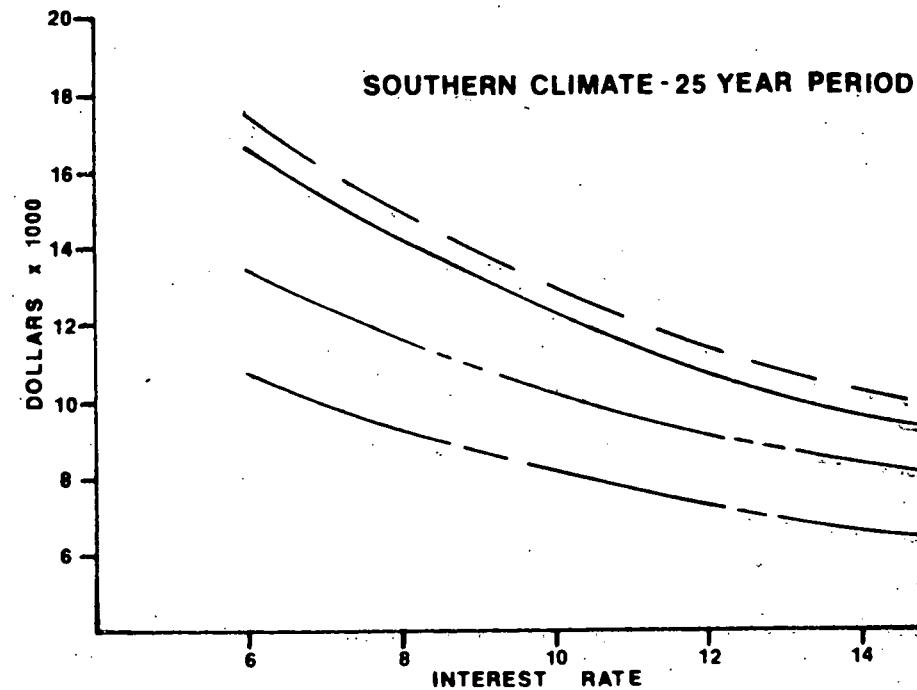
FIG. 4-5

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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 ith alternative

FC_i = First cost of the ith alternative

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

AOC_i = Annual operating costs of the ith alternative

AMC_i = Annual maintenance costs of the ith 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 quality, 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

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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

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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 committed 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 there 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. (Lowering the wastewater temperature will slow down the biological treatment process and require larger aeration basins to increase the time of treatment.)

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.

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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 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 to 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 upom 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 at which the wastewater is biologically treated. Typically, the colder the temperature, the slower the rate of biological treatment. Hence, as the rate of the biological reaction is decreased, the time required to provide the same degree of treatment must be increased. This means that with lower biological reaction rates the volume of the aeration basin must be increased to provide a longer detention time (and reaction time) to achieve the same degree of treatment as the higher reaction rates. Biochemical reactions, in general, follow the van't Hoff rule of a doubling of reaction rate for a $10^{\circ}C$ ($18^{\circ}F$) increase in temperature over a restricted temperature range. Studies with activated sludge 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 $d\ln k/dT$ 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

$$k_2 = k_1 e^{\left[\theta (T_2 - T_1)\right]}$$

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 empirically to vary from 0.056 in the temperature range between 20° and 30°C and 0.135 in the temperature range between 4° and 20°C[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. Using this equation and assuming a decrease in the biological reaction rate from 0.25 to 0.18 (55°F to 45°F) the detention time t would have to be increased 38.7% in order to achieve the same degree of treatment. Hence, the volume of the aeration basin would need to be increased 38.7% and the aeration equipment would need to be designed for the larger volume and longer retention times.

Another aspect of the wastewater treatment process that is influenced by temperature is the anaerobic digestion of sewage solids or sludge. As with the liquid wastewater treatment, the biological activity rates are temperature dependent. In some areas of the country anaerobic digesters must be heated during the winter months in order to maintain acceptable rates of microbiological activity. However, since the digestion of sludge

solids is separate from the liquid treatment process, the HPC-ICES scheme would not have any influence on the temperature in the anaerobic digester and hence would not affect the sludge digestion process.

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 stated generally that as wastewater effluent temperatures increase, dissolved oxygen saturation concentrations decrease, and predominant aquatic lifeforms 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 be 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 time, the permissible temperature increase is a matter of controversy. Detailed discussion of thermal pollution may be found in Krendel 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 proposed 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 needed there. Once a preliminary 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.

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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

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required that such independent population projections be developed. However, many of these independent population projections were later found to be in errc 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."[56] 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

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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?

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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

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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 communi-
ties 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 popu-
lation 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, waste-
water 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

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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 the central 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.

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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 eliminat 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 un-developed, 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.

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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.

SOLAR COLLECTOR FIELD

SOLID WASTE INCINERATION HEAT RECOVERY BOILER

CENTRAL STATION

COLD   HOT

PEAK HEATING / BACK-UP BOILER

COLD

HOT

HEAT   PUMPS

METHANE STORAGE

HOT

COLD

THERMAL STORAGE

ANAEROBIC SLUDGE DIGESTION

SEWAGE TREATMENT PLANT

CIRCULATING WATER RETURN FROM COMMUNITY

HOT CIRCULATING WATER TO COMMUNITY END USERS

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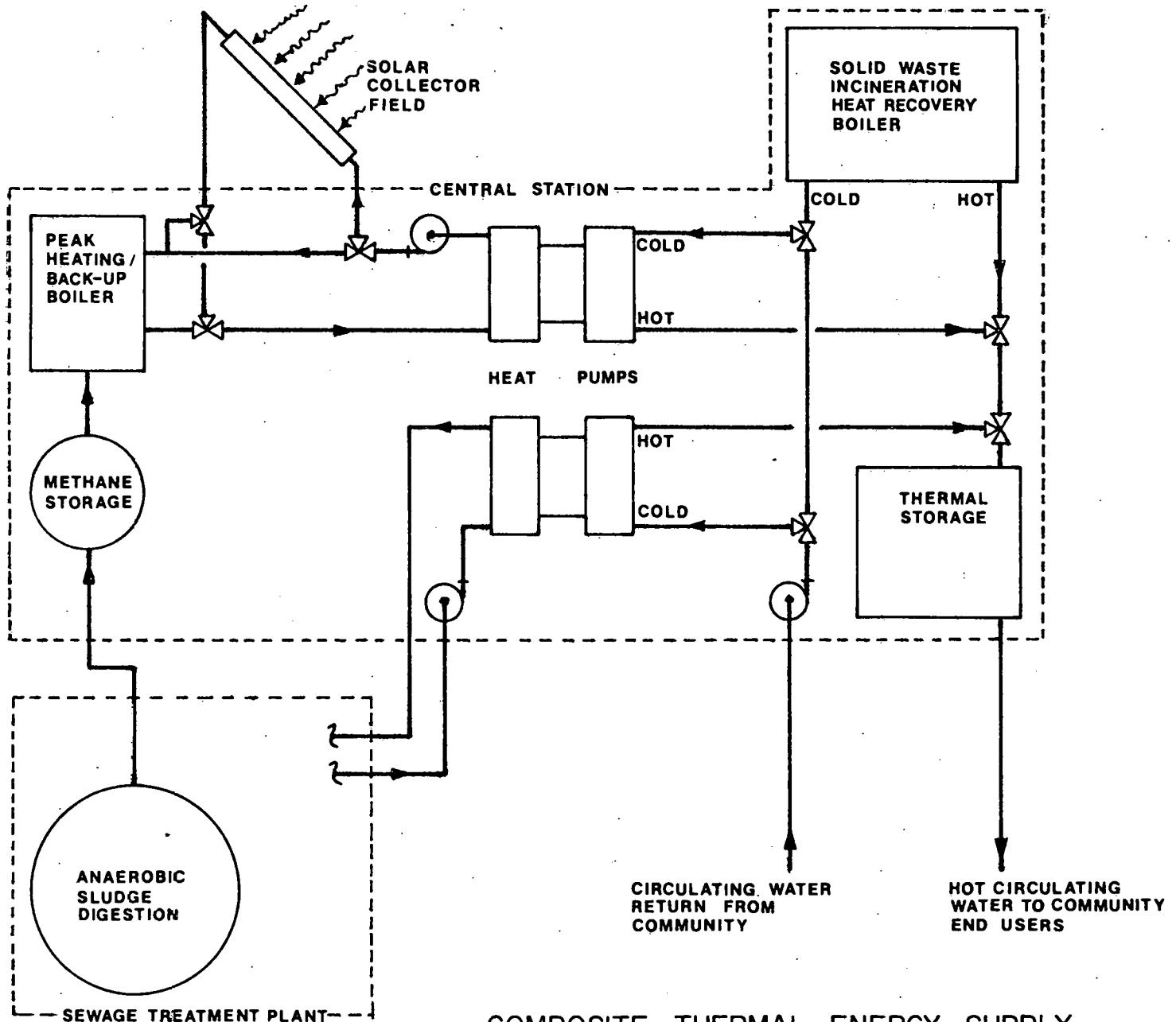
COMPOSITE THERMAL ENERGY SUPPLY
FOR   HPC-ICES
FIGURE   7-1

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 10.) 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 disad-
vantages 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 coin-
cidental 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 pro-
bable 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 drawbacks 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

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INDUSTRIAL HP-WHR SYSTEM SCHEMATIC

FIGURE 7-2

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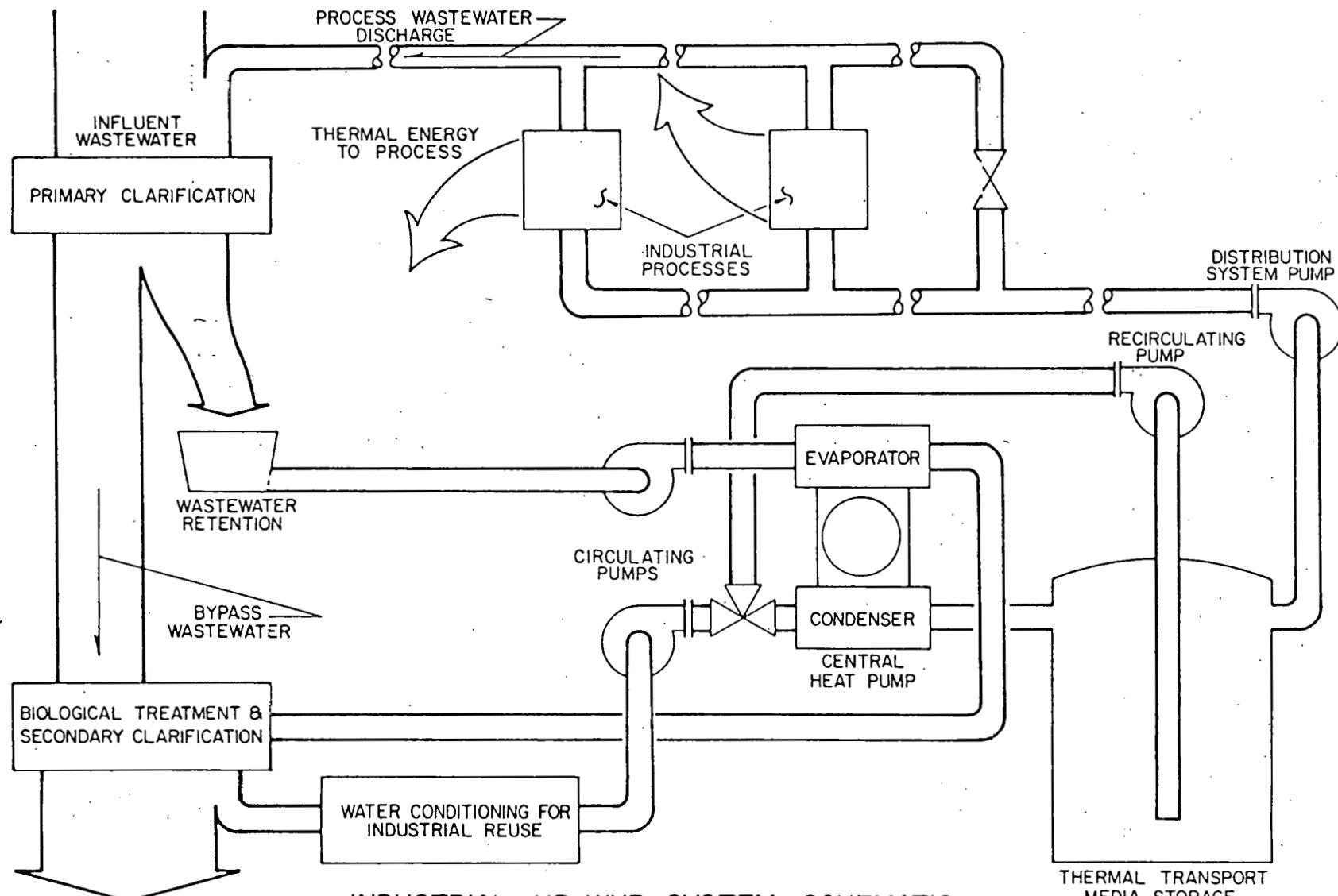
a system applied in this manner would be greatly simplified since the
thermal demand placed on the system would directly coincide with the supply
of available energy rejected through the industrial processes. This over-
all system concept is especially attractive to industrial communities
requiring large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

7.5 <u>SUMMARY</u>

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.

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8.0 SITE-SPECIFIC FEASIBILITY STUDY FOR CONYERS, GEORGIA

8.1 INTRODUCTION

The preceeding chapters have detailed the development of the Heat Pump Wastewater Heat Recovery (HP-WHR) system concept. Thermodynamic and economic analyses on a hypothetical system seem to indicate the system would be competitive, both performance-wise and cost-wise, with the other more standard options for space conditioning. However, such analyses and projections are necessarily based on a set of assumptions which may or may not be valid for a particular site of application. Therefore, in order to determine the effectiveness of such an HP-WHR system on the community level, a site-specific examination must be performed.

Chapters 8.0 and 9.0 of this report develop the methodology, via case studies of two application communities, of examining the HP-WHR concept on a community basis and validate the feasibility of the concept on the community level. The two application communities, Conyers, Georgia and Cornelia, Georgia (shown on the following location map), were selected for their varieties of community characteristics and configurations which would affect the application of an HP-WHR system. The applications of the HP-WHR system concept to each of the demonstration communities make note ot the ditterent community characteristics and take advantage of them to emphasize the wide range of application possibilities of this system concept.

The City of Conyers, Georgia, located in close proximity to the metropolitan Atlanta area, is in a high growth area of the state for both residential and industrial developments. Additionally, the segmentation of the sewer system in Rockdale County (in which Conyers is located) facilitates the application

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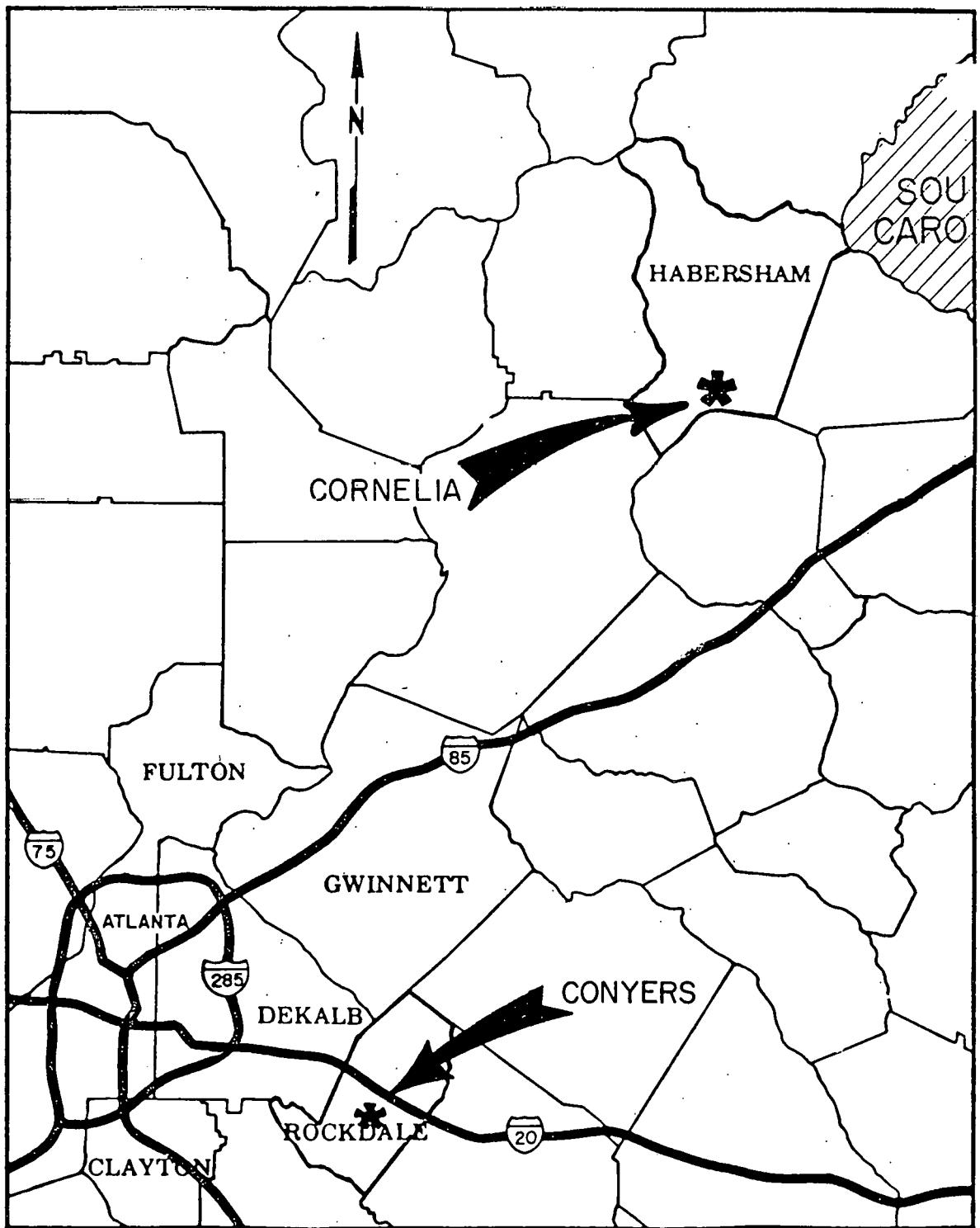
LOCATION MAP
APPLICATION COMMUNITIES
FIGURE 8-1

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of the HP-WHR system concept to a developing residential area in close proximity to one of the wastewater treatment plants on the outskirts of the Conyers area. Thus, the opportunity to examine strategies for phasing construction in high growth residential areas is presented.

The City of Cornelia, located in the mountains of northeast Georgia in Habersham County, is in a low growth area of the state. Because of the low growth rate, application of an HP-WHR system to Cornelia will involve primarily system retrofits to existing structures. Additionally, due to the layout of the community and its attending sewer system, the primary focus of the Cornelia system will be to provide service to customers in the commercial category. Service for one large industrial customer is also included in the system plan.

With the completion of these feasibility studies, the cost-and energy-effectiveness and the flexibility of application of the HP-WHR system concept should be successfully demonstrated.

8.2 COMMUNITY DESCRIPTION - CONYERS, GEORGIA

The City of Conyers, Georgia is located in the center of Rockdale County approximately 22 miles east of downtown Atlanta. The City and County are considered to be within the Atlanta Metropolitan area and have experienced significant commercial and residential growth as a result of the overall growth that has been occurring within the Metropolitan Atlanta Area. The building of Interstate Highway 20 through Rockdale County in the 1960's provided a corridor of rapid accessibility to Atlanta and to the later completed Interstate 285 Perimeter Highway around Atlanta. A large percentage of the population of Conyers and Rockdale County commute to the Atlanta area to work.

8.2.1 POPULATION GROWTH

Population is a dynamic factor in the development of a geographical entity and the Conyers-Rockdale County area is no exception. Conyers and Rockdale County have experienced significant growth in recent years, influenced by their location within the Atlanta Metropolitan Region. The tremendous growth of Rockdale County in the period of 1960 - 1970, approximately 72%, can be attributed largely to an in-migration of people caused by Atlanta's urban sprawl. Phenomenal residential development, principally in the form of detached, single-family dwelling subdivisions, has resulted. In addition, industrial development in Rockdale County has flourished in response to the railroads and major traffic arteries which provide easy access to Atlanta. The City has almost matched the County's growth rate, showing an approximate 70% increase in population during the 1960 - 1970 period. During the first 6 years of this decade, the growth rate has shown no signs of declining. The County has exhibited a 43% growth rate and the City a 31% growth rate, despite the recession of 1974 - 1975. Based on the observation of historical population trends, the Conyers-Rockdale County area is expected to continue its rapid rate of growth over the next 20 years.

8.2.2 ECONOMIC CHARACTERISTICS

Over the last 20 years the economy of the Conyers-Rockdale County area has changed from that of a sparsely populated rural and farming community to a rapidly growing suburban residential community within the Metropolitan Atlanta Area. This change has caused a significant decrease in the number of people who are employed in farming and agricultrual-type activities. Consequently, there has been an increase in the percentage of the population involved

in business, commercial and industrial-type jobs. In recent years the Conyers-Rockdale County area has experienced significant industrial growth which is located primarily in those areas served by Interstate Highway 20 and the Georgia Railroad. There has also been significant commercial growth in the form of shopping centers, banks, service stations, and other commercial activities needed to serve the increasing residential population. In the last ten years the Conyers-Rockdale County area has experienced a residential housing boom consisting of both apartments and single family dwellings. All indications are that this high level of residential construction activity will continue for some time into the future.

According to the 1970 Census, the median family income for the City of Conyers was $8,992 per year and the per capita income was $2,792 per year. Most of the population of Conyers could be classified in the low to moderate income level with only 13.4 percent earning $15,000 per year or more and 12.2 percent being below the poverty level.

8.2.3   LAND USE

The City of Conyers has within its corporate limits a total of 3572 acres. Due to a recent annexation of several land areas contiguous to the town core, the majority of the City's land is undeveloped. Some 54.2%, or 1935 acres, of the land area may be classified in this category. The second largest category is residential, with 924 acres or 25.9% of the land area. Streets and other public rights-of-way occupy approximately 372 acres or 10.4% of the land area. Public and semi-public land occupies 149 acres (4.2%); commercial usage requires 159 acres (4.4%), and industrial occupies 33 acres or 9% of the land area.

The City of Conyers and the contiguous area which will be considered for service with the HP-WHR system are shown on Figure 8-2.

8.2.4 CLIMATE

The climate of the Conyers-Rockdale area is humid and continental. The winters are mild, but they have very changeable temperatures. The prevailing wind during the winter is northerly. The weather is largely controlled by movement of areas of high and low barometric pressure and the accompanying winds. In winter these conditions cause frequent alteration of warm moist southerly winds and cold dry northerly winds.

The average winter temperature is 49.3°F. The temperature usually rises rapidly in March and April. The difference in the mid-winter (January) average and the mid-summer (July) average is 27.9°F, which is relatively small compared with a difference of 60°F in some of the more northern states. The summers are warm but are comparatively free from extreme heat due to the altitude and latitude of the area. The average summer temperature is 75.1°F. The average date of the first killing frost is November 8 and for the last is March 29.

Rainfall varies somewhat from year to year, but its seasonal distribution is generally favorable for crops. Serious drought is not likely to occur more than once in 10-15 years. The rainfall reaches a peak in the winter and again in mid-summer. Fall is the driest season of the year and about one-half of the rainfall comes in quantities of one inch or more in a 24-hour period.

The closest United States Climatological Station is located in Covington, Georgia in Newton County. Covington is eleven miles south of Conyers so the data should be applicable to the study area.

CITY OF CONYERS
HP-WHR SYSTEM SERVICE AREA
FIGURE 8-2

The climate of the region is influenced by the Atlantic Ocean, the Gulf of Mexico, and the Appalachian Mountains. The mountains shield the area from the heavy snow and extreme winter temperatures found in eastern Tennessee, northern Alabama, and Arkansas. Likewise, the ocean and gulf affect the seasonal temperatures in the summer, and "continentality" is not as noticeable as the areas located further inland.

Because of the geographical location, temperatures are mild in winter and summer, with summers moderately warm and humid. During the summer, some extended periods of hot weather are experienced, but such periods are not unbearable. The hottest weather comes in brief spells during the summer. Even though days are hot, nights are mild with temperatures dropping into the sixties and seventies.

Humidity ranges around eighty to ninety percent during the summer causing an occasional early morning fog, and normally the humidity falls in early afternoon to fifty to sixty percent. The average wind velocity comes from the northeast or northwest in the winter and from the south in the spring and summer.

Winters are generally mild with cold spells lasting only a few days even in mid-winter. Temperature drops to 32 degree Fahrenheit or below about fifty days each winter. The temperature reaches the teens or twenties at least once a month during the winter and ten degree temperatures or below occur once a year about every fourth year.

8.2.5 PROPOSED SERVICE AREA

The community area to be serviced by the Conyers Heat Pump-Wastewater Heat Recovery System is located relatively close to the thermal energy source

and will be composed of single family residential structures. The unique advantages of the Conyers service area are: 1) the proximity of the service area to the thermal energy source, and 2) the opportunity to service both existing structures through space conditioning system retrofit and new structures through initial system installation as the community grows.

The proposed service area of approximately 472 acres was subdivided into three zones for the purposes of estimating thermal energy consumption. The service zones, shown in Figure 8-3, vary in the degree of development. Zone 1, 149 acres, is approximately 40% complete with regard to its build-out potential of approximately 50 houses. Zone 2 is a presently undeveloped parcel of land, approximately 116 acres, which shows excellent potential for development during the later phases of the study period. Zone 3, 207 acres, is also a relatively new subdivision which is approximately 50% complete in its potential build-out of 150 houses. Obviously, there is significant room for expansion of the number of homes served from the HP-WHR system and for realizing the significant initial cost advantages of installing the system in new structures, as opposed to retrofitting existing structures and space conditioning systems.

The energy source analysis indicates that the energy reasonably available from the waste treatment plant will serve approximately 400 average homes in the Conyers area. This is 100 plus percent of the service area's projected requirements. However, energy augmentation by solar or other means might prove feasible if growth in the area exceeds projections.

The other significant point favoring the Conyers system is the proximity of the residential development to the waste treatment plant. While this situation is usually uncommon, the suburban nature of Conyers' recent growth

CITY OF CONYERS
2.75 MILES

ALMAND CREEK

CURRY CIR.

ALMAND BRANCH
TREATMENT PLANT

ZONE 3

ZONE 2
(UNDEVELOPED)

ALMAND BRANCH RD.

MORRIS DR.

ZONE 1

STANTON RD.

STOCKBRIDGE HWY. 138

EBENEZER RD.

POTTS RD.

0   300'   600'

CONYERS, GEORGIA HP-WHR SYSTEM
SERVICE AREA

FIGURE 8-3

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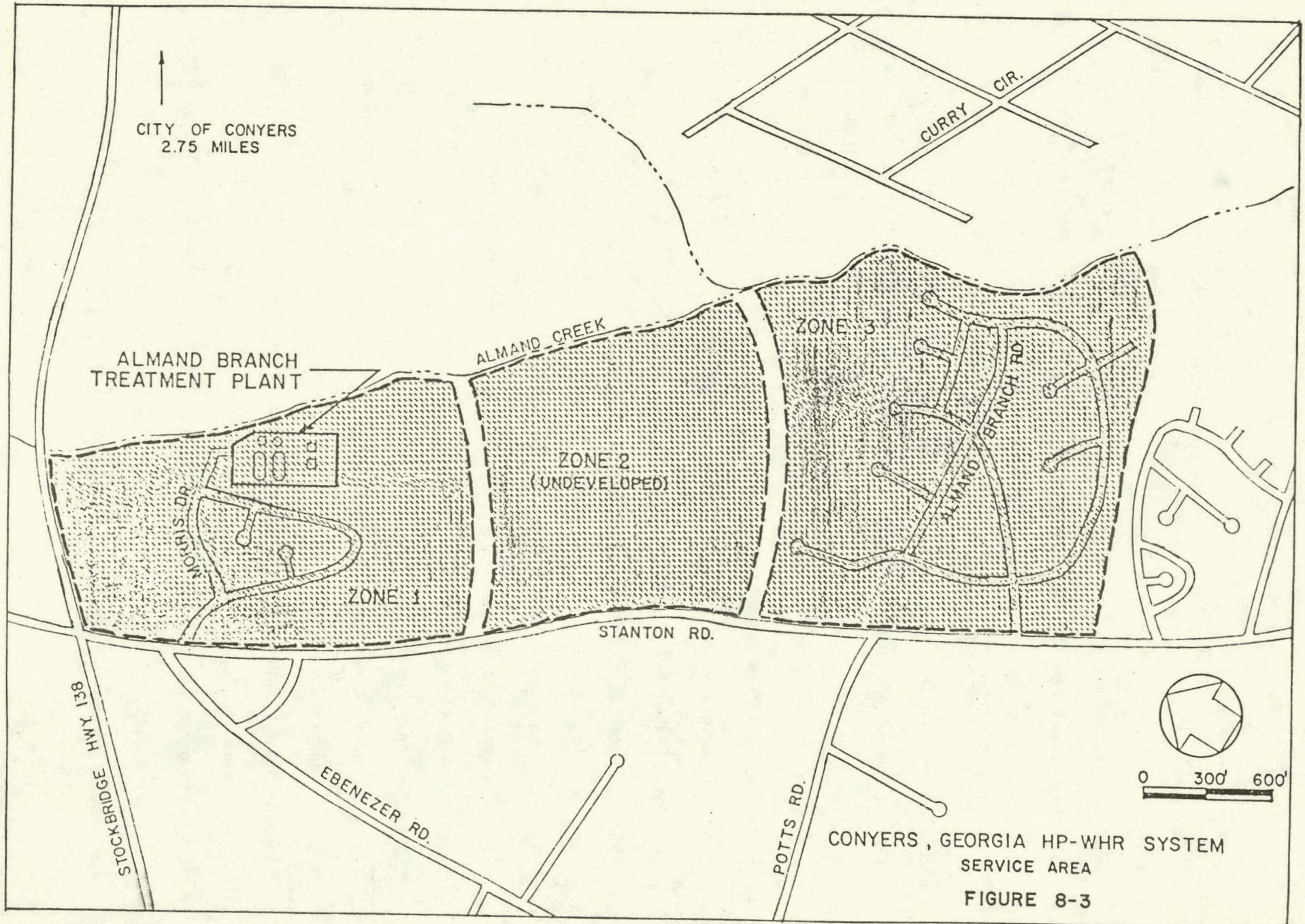
has promoted residential subdivision expansion in the outlying areas. This relatively dense growth in the area of the wastewater treatment facility will tend to decrease investment costs in the thermal distribution system and make the project more economically viable.

8.2.6 <u>BUILDING CONSTRUCTION</u>

The proposed service area was divided into three zones in order to facilitate estimating thermal energy consumption. All structures within the service area are detached, single family dwellings. Building construction is similar within and between zones.

The housing units within the proposed service areas range in size from 1,000 to 1,600 square feet and are generally less than five years old. Spatially, they are arranged on land lots of approximately 1/4 to 1/3 acre. The units are predominately split level units with three bedroom units. However, some ranch type designs are intermingled throughout the subdivisions. FHA minimum property standards or VA standards were adhered to in the majority of units. Dwelling unit walls are the 3 1/2" cavity wall construction consisting of gypsum board inside face, 2" x 4" stud supports, 3 1/2" fiberglass insulation, particle board outside sheathing and exterior siding or brick exterior. Attic insulation, 4" to 8" of fiberglass loose fill or 6" batting, is used in the majority of construction. Floors are generally uninsulated, but carpet is normally included as a floor covering. Window construction is the double-hung sliding type with single pane glass. Storm windows are not used on the majority of the houses.

Typically, each house is heated by a forced air, natural gas furnace. Most houses are air conditioned by means of duct mounted refrigerant coil and

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air cooled condensing units. System sizes range from 2 to 5 tons capacity air conditioning. Furnaces are usually 80,000 to 100,000 Btu input size.

One especially attractive feature of the Conyers service area is that the area is less than 30 percent complete with respect to residential development. The fewer number of units that must be retrofitted will make the scheme more attractive. Substantial first cost benefits can be obtained by installing units at the time of original construction.

8.2.7 HEATING AND COOLING LOADS

A descriptive statistic in determining the overall desirability of providing central heating and cooling media for use by a community is the total number of hours per year when some amount of heating or cooling is required by most residences. To derive this statistic, cumulative totals were developed from hourly temperature distribution data to indicate the number of hours per year when heating or cooling is required in Conyers. Heating was assumed to be needed when the outside air temperature was 65°F and below. This basis yielded an approximate 4593 hours per year when some form of heating energy was required by most buildings. Likewise, cooling was assumed to be required when the outside air temperature was above 75°F. For the Conyers area, this resulted in 1974 total hours per year which required some form of cooling. With such substantial heating and cooling seasons as the above, a central heating/cooling system shows obvious potential for service in the area.

Monthly Loads

Heating and cooling loads for the Conyers system were estimated on a monthly basis for those building structures within the projected service area. The primary emphasis in load development was placed on monthly figures since

no greater degree of detail could be justified in a preliminary study of this type. Peak loads were also developed using design conditions from the 1977 Fundamentals Volume of the ASHRAE Handbook and Product Directory.[50]

The estimate of monthly heating and cooling requirements was calculated by a computer program developed by the Community Energy Systems Branch of Georgia Tech. To estimate the total thermal load per service zone, the computer program, when appropriately input with the pertinent data, multiplies the number of building units of a particular type times the average floor area per unit for that building type times the monthly thermal integrity factor for that building type to obtain a total monthly BTU requirement for each building category. Total thermal requirements for all building categories are then summed to give an overall load for each zone. Finally, total zone loads are summed to give an overall system thermal load.

Input to the program included thermal integrity factors developed from the Rand Corporation's ENERGY USE AND CONSERVATION IN THE RESIDENTIAL SECTOR: A REGIONAL ANALYSIS.[27] Unitized energy consumption data was estimated on an annual basis. Heating energy required was 30 MBtu/sq.ft. annually. Cooling energy required was 20 MBtu/sq.ft. annually. Monthly heating and cooling loads were determined by apportioning the annual requirement according to the percentage of heating or cooling degree days occurring in a particular month. Heating and cooling degree days were taken from "Monthly Normals of Heating and Cooling Degree Days 1941-1970",[55] published by the National Climatic Center, Asheville, North Carolina. Table 8-1 is a summary of the percentage figures utilized in the load estimate.

TABLE 8-1
Annual Heating and Cooling Degree Days
Conyers, Georgia

Month	Heating		Cooling	
	Degree Days	% Total	Degree Days	% Total
Jan.	701	22.6	0	0.0
Feb.	560	18.1	0	0.0
March	443	14.3	12	.7
April	144	4.6	27	1.7
May	27	0.9	154	9.7
June	0	0	321	20.2
July	0	0	403	25.4
August	0	0	388	24.4
Sept.	8	0.3	227	14.3
Oct.	137	4.4	57	3.6
Nov.	408	13.2	0	0.0
Dec.	667	21.6	0	0.0
	3095	100.0	1589	100.0

The service area under consideration for Conyers contains only residential buildings. Building count and area were estimated from field surveys. Table 8-2 presents a month-by-month summary of the total projected thermal load on the HP-WHR system. The full results of the computerized load calculations and peak load determinations are presented in Appendix C.

TABLE 8-2
CONYERS HP-WHR SYSTEM LOADS

PERIOD	HEATING LOAD BTU x 10^9	COOLING LOAD BTU x 10^9
Jan.	2.915	0.0
Feb.	2.335	0.0
March	1.845	0.060
April	0.593	0.146
May	0.116	0.834
June	0.0	1.737
July	0.0	2.184
August	0.0	2.098
Sept.	0.039	1.230
Oct.	0.568	0.310
Nov.	1.703	0.0
Dec.	2.786	0.0

12.900×10^9 BTU $\qquad$ 8.599×10^9 BTU

Peak Heating Load

As mentioned above, peak loads were estimated using the procedures and design conditions from the ASHRAE Handbook.[50] The degree day formula, with terms rearranged to solve for the peak load, P, was utilized as shown below:

$$P = \frac{(E) \times (TD)}{(DD) \times (24)}$$

Where:

P — Design heating load per hour

E = Heat energy required

TD = Design temperature difference

DD = Heating degree days

For this estimate:

$E = 2.915 \times 10^9$ BTU – from the computer load estimate for January

$TD = 70^\circ - 17^\circ = 53^\circ F$ – desired inside temperature minus the 99% design temperature for Conyers

DD = 701 – from Table 8-1

Thus

$$P = \frac{(2.915 \times 10^9) \times (53^\circ)}{(701) \times (24)}$$

$$= 9.18 \times 10^6 \text{ Btu/hr}$$

Peak Cooling Load

Utilizing the design figure of an average 550 square feet per peak ton of refrigeration for residential structures, the peak cooling load was estimated as follows:

$$\text{Peak Cooling Load} = \frac{430,000 \text{ ft}^2 \text{(building area served)}}{550 \text{ ft}^2/\text{ton}}$$

$$= 782 \text{ tons}$$

8.2.8 THERMAL ENERGY RESOURCES

From earlier discussions on the general HP-WHR system concept and from
the descriptions of the specific system to be applied in the demonstration
community of Conyers, Georgia, it is apparent that the low-grade heat in
the community's wastewater will be utilized as the thermal energy source.
Due to the site specific variables to be considered in characterizing a
community's wastewater as the thermal energy source for this system, a
determination of the quantities of thermal energy available from this
source necessarily involves examination and analysis of the specific waste-
water collection and treatment system to be evaluated for the application of
this scheme. The following sections examine the wastewater facilities
of the Conyers, Georgia community and develop estimates of the thermal
energy available to an HP-WHR installation at a proposed site in Conyers.

CITY OF CONYERS, GEORGIA WASTEWATER FACILITIES

The City of Conyers and contiguous unincorporated areas of Rockdale
County are served by three sewage treatment facilities. Boar Tusk Creek
sewage treatment plant, approximately one mile outside the corporate city
limits off of East View Road, is a 500,000 GPD activated sludge facility
which handles approximately 75% of the domestic wastewater generated with-
in the city limits. A second facility, the Almand Branch sewage treatment
plant located approximately 2 miles outside the corporate limits off of
Stockbridge Highway is a 1.0 MGD, oxidation ditch type of treatment facility
which handles the remaining 25% of the City's domestic wastewater plus
significant amounts of industrial discharge. A third treatment facility,
the Atlanta Suburbia sewage treatment plant, located near the Rockdale
County line off of Underwood Road, is a 200,000 GPD activated sludge
facility which serves the Salem East, Salem Lake, Salem Woods, and Salem

Village residential subdivisions.

Because of advantages in system configuration and plant siting, and due to its proximity to a compact potential service area comprised of existing subdivisions, areas now under development, and several institutional structures (schools), the Almand Branch sewage treatment plant has been selected for evaluation of available thermal energy for the proposed Conyers HP-WHR system.

The Almand Branch plant, constructed in 1973, was designed for the treatment of domestic sewage of average pollutant concentrations and flow characteristics. The plant design is the oxidation ditch type, a modified form of activated sludge treatment, which may be classified in the extended aeration group. Average daily design flow is 1.0 MGD, and BOD removal efficiency was designed at 90% or better.

A flow sheet of the plant is provided in Figure 8-4. The principal components of the plant, along with pertinent design data, are as follows:

- Barminutor with manually cleaned by-pass bar screen.

- Raw Sewage Pumping Station with 2 variable speed vertical pumps and 1 constant speed vertical pump

- Aeration tanks (2) each at 194' long and 64' wide at the top with a 6' water depth. Approximate capacity 413,000 gallons each. Mechanical aerators are 26' rotor aerators. Theoretical retention time at design flow of 1.0 MGD is 20 hours.

- Final clarifiers (2) circular with 35'-0" diameter and 9"-0" side water depth. Approximate capacity of each tank is 64,000 gallons with a 3 hour retention time.

- Return sludge pumping station with 2 Midland Totally Submersible sewage pumps.

- Sludge drying beds (4) each at 26' wide and 49' long.

- Chlorine contact tank, 2 sections, each 18' wide and 20' long with a 7' water depth. Approximate capacity is 34,000 gallons with 50 minute retention time at design flow.

- Parshall flume, 9", with flow monitoring device.

- Control building containing laboratory and office, chlorine room, storage room and shop.

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CITY OF CONYERS, GA.

ALMAND BRANCH SEWAGE TREATMENT PLANT

FIGURE 8-4

WASTEWATER INFLUENT

PUMPING STATION

COMMINUTOR & BAR SCREEN

FLOW

FLOW

OPERATIONS AND MAINTENANCE

OXIDATION DITCHES

ACCESS ROAD

SLUDGE HOLDING TANK

CLARIFIERS

RETURN SLUDGE STRUCTURE & PUMPS

SLUDGE DRYING BEDS

CHLORINE CONTACT TANKS

PARSHALL FLUME

PUMPING STATION

EFFLUENT
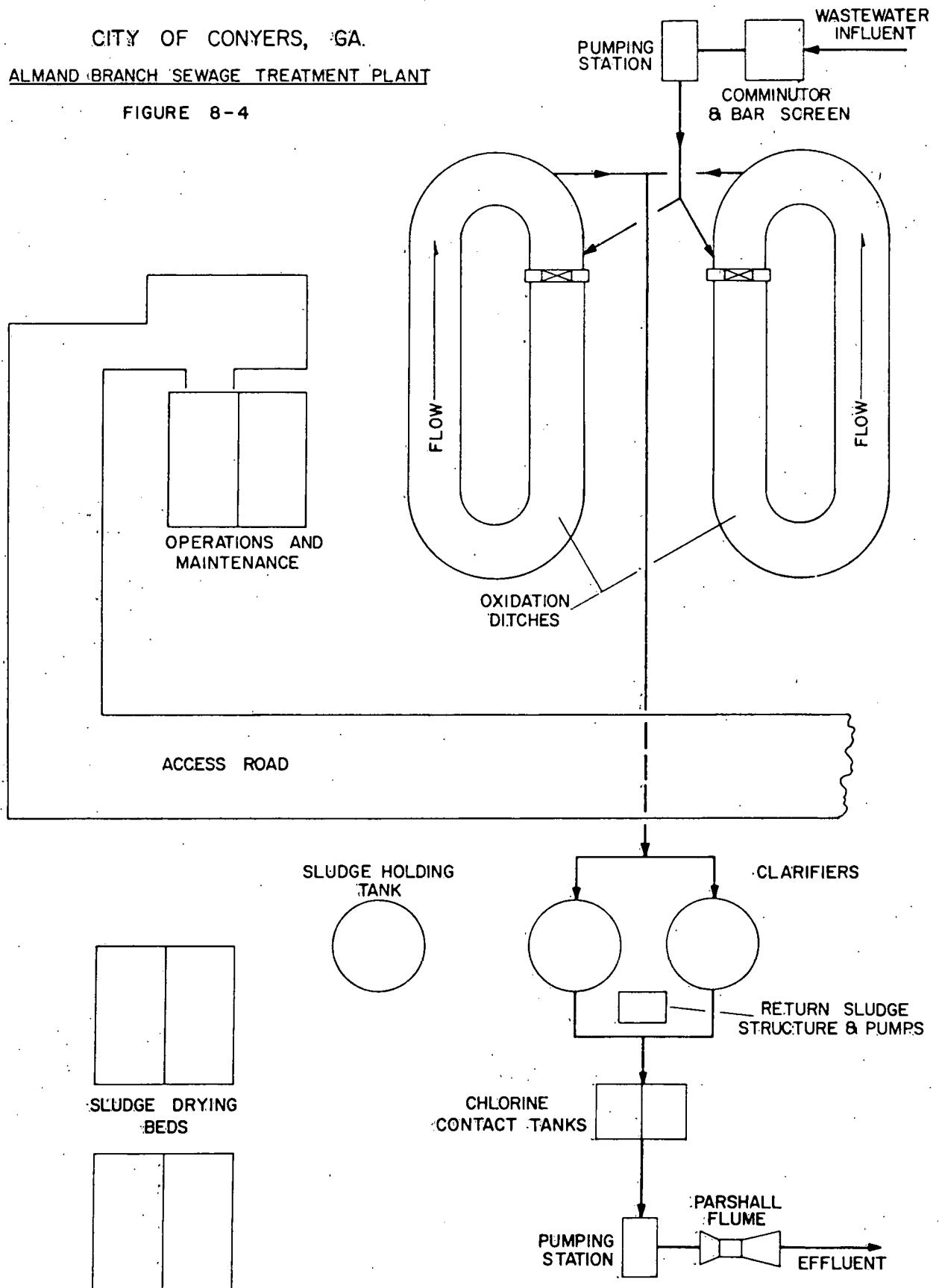
• Standby electric power system.

Due to the newness of the facility and a good operation and maintenance program by the City, very few operational problems with plant equipment have developed.

Sewers which are tributary to the Almand Branch Facility consist of 1390 lineal feet of 4", 3750 lineal feet of 6", 58,390 lineal feet of 8", 5940 lineal feet of 10", 7960 lineal feet of 12", 3080 lineal feet of 15", 7410 lineal feet of 18", 3890 lineal feet of 24", and 2730 lineal feet of 30" gravity sewers. There is one pumping station with 890 lineal feet of 6" force main.

FLOW AND TEMPERATURE DATA - CONYERS, GA.

As discussed in Chapter 10.0 - COMPONENT TESTING, site specific temperature and flow measurements are the necessary first step in developing a data base for determining the thermal energy available at any particular site of application. Plant effluent flow and temperature data from existing plant instrumentation was obtained for the period 10/14/78 to 11/16/78. Temperature sensors and flow instrumentation are located at the discharge of the treatment facility in the vicinity of the chlorine contact basin. Data for the period 11/8/78 through 11/14/78, selected as representative of plant conditions during the subject period, is presented graphically (Flow vs. Time, and Temperature vs. Time) in Figure 8-5 and summarized in Table 8-3.

Flow was noted to be on a daily cycle with miscellaneous peaks and dips during the weekdays. Weekend flow was relatively constant. Weekday flow rates averaged 0.662 MGD, and weekend rates averaged 0.534 MGD. Overall, flow rates did not vary from the average to the extent that might be expected from experience with wastewater flow cycles. Weighted average daily flow (ADF) for the seven day period was 0.623 MGD.

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CONYERS, GEORGIA

SEWAGE TREATMENT PLANT
11-8-78—11-14-78

FIGURE 8-5

157
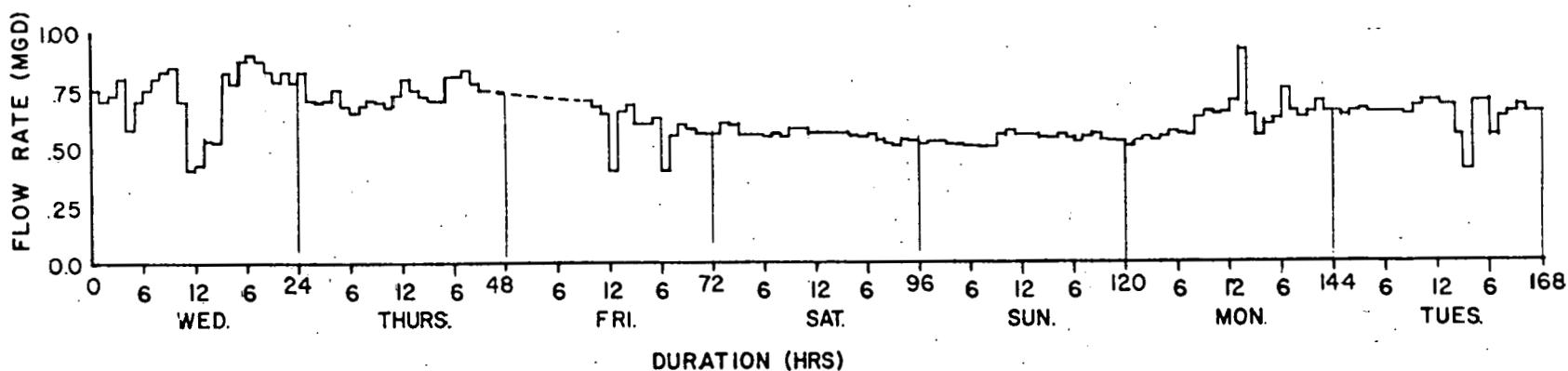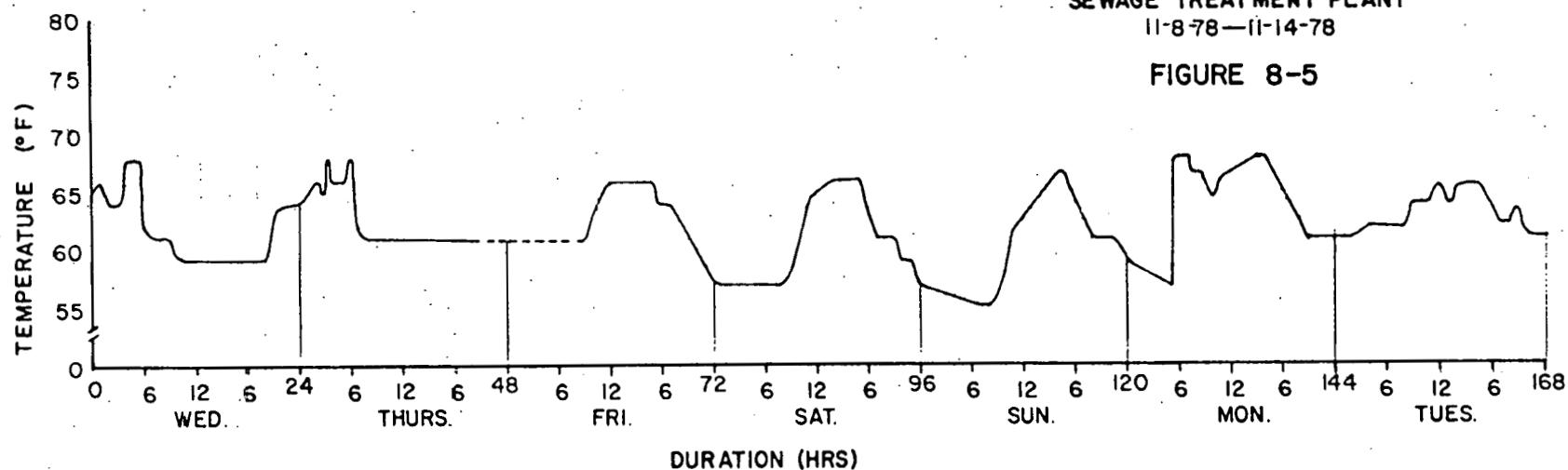
TABLE 8-3
CONYERS, GA. WASTEWATER SYSTEM
ALMAND BRANCH SEWAGE TREATMENT PLANT
TEMPERATURE AND FLOW DATA
11/8/78 TO 11/14/78

PLANT EFFLUENT TEMPERATURE

DAY		LOW °C/°F	HIGH °C/°F
WED	11/8	14/57	20/68
THUR	11/9	16/61	20/68
FRI	11/10	14/57	19/66
SAT	11/11	14/57	19/66
SUN	11/12	13/55	19/66
MON	11/13	14/57	20/68
TUES	11/14	16/61	19/66

FLOW

HIGH:	$41,667 \frac{gal}{hr}$ ($\simeq$ 1.0 MGD) 11/13 1:00pm
LOW:	$7083 \frac{gal}{hr}$ ($\simeq$ .17 MGD) 11/8 12:00 noon
AVG. WEEKDAY:	0.662 MGD
AVG. WEEKEND:	0.534 MGD
WEIGHTED AVG. DAILY FLOW:	.623 MGD

Plant effluent temperatures showed a relatively smooth plot, ranging only from 55°F to 68°F. There seemed to be a pattern of higher temperatures during the daytime hours, likely due to the long retention time - 20 hours - of the aeration tanks.

It is expected that, for this section of the Conyers/Rockdale County wastewater collection and treatment system, the operating conditions - flow and temperature - will continue to exhibit the trends illustrated in the foregoing graphs. Flow rates may show some escalation, to the extent that they reflect the growth in this area of the county. Extreme seasonal influences may reduce the operating temperatures somewhat, but will not significantly affect the availability of thermal energy at this source.

From the above characterization of the Conyers wastewater conditions, it is apparent that significant amounts of thermal energy are available for reclaim with a Conyers HP-WHR system. The next report section develops an estimate of the quantities of thermal energy available at the proposed plant location.

THERMAL ENERGY ESTIMATE

For the purposes of developing a nationwide projection of available thermal energy from wastewater, flow was defined as a function of national population, and industrial wastewater discharges (a significant source of rejected thermal energy) were ignored due to insufficient data. However, a more meaningful estimate can be developed on a site-specific basis because considerations of the local wastewater conditions may be taken into account. The temperature and flow data from the Conyers facility will serve as the basis for this energy estimate.

System design for the Conyers facility dictates that wastewater be extracted from the aeration basins and pumped to the central station (which would likely be sited at the treatment plant). Thus the combined capacity,

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826,000 gallons, of the two aeration basins would be utilized as system storage to absorb the diurnal fluctuations in flow and temperature. Therefore the figures for average plant effluent flow and temperature can be considered typical of the wastewater conditions which could be expected for the HP-WHR central station equipment. It is duly noted that the conditions documented herein are from a somewhat arbitrarily selected one-week period which may or may not have been representative of conditions to be expected over an annual cycle. However, inspection of the seven-day flow cycle reveals no characteristics which would indicate it to be atypical. Further, since there is adequate capacity within the system to "accumulate" the surges in influent wastewater flow and complete the treatment process at a more steady flow rate, it is anticipated that the documented weekly flow cycle can indeed be taken as typical. The temperature of the wastewater, on the other hand, will likely exhibit more significant variations due to seasonal influences. However, again due to the adequate storage capacity within the treatment system, it is anticipated that the affect of ambient temperature changes on the wastewater temperature will be moderated somewhat. The temperature and flow data presented here and utilized in the following estimate of available thermal energy are presented primarily to provide a firm base for this "ballpark" estimate. Figures 8-6, and 8-7, Temperature vs. Duration and Flow vs. Duration, respectively, for the period 11/8/78 through 11/14/78, illustrate that the average conditions, Flow .623 MGD and Temperature 62°F, are indeed indicative of typical conditions throughout the subject period (Note that 12 hours has been omitted from the one week period due to a blank in the temperature and flow data print outs.)
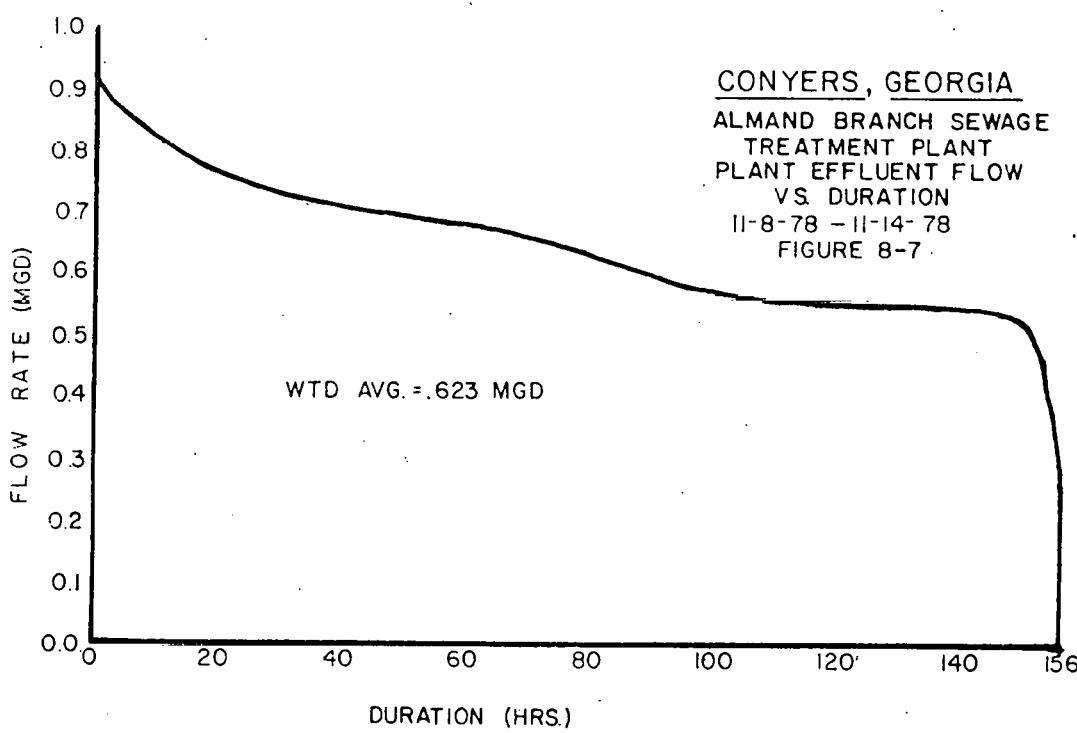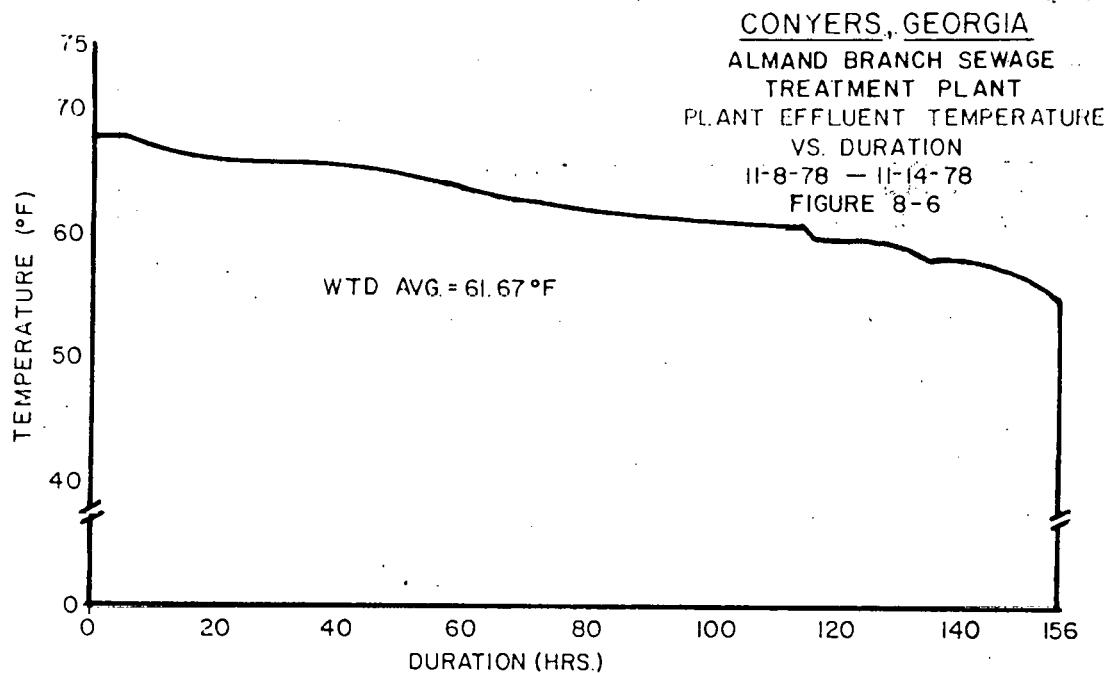
CONYERS, GEORGIA
ALMAND BRANCH SEWAGE
TREATMENT PLANT
PLANT EFFLUENT TEMPERATURE
VS. DURATION
11-8-78 — 11-14-78
FIGURE 8-6

WTD AVG. = 61.67°F

CONYERS, GEORGIA
ALMAND BRANCH SEWAGE
TREATMENT PLANT
PLANT EFFLUENT FLOW
VS. DURATION
11-8-78 — 11-14-78
FIGURE 8-7

WTD AVG. = .623 MGD

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From Chapter 2.0 - POTENTIAL APPLICATIONS, the total amount of heat

available from the thermal supply (wastewater) can be calculated from equation

(2-2), shown below:

$$Q = W \, C \, (\Delta T) \qquad (2\text{-}2)$$

Where:

Q = total heat, Btu

W - flow of wastewater, lbm

C - specific heat of wastewater (assumed to be 1.0 Btu/lbm-$^{\circ}$F)

ΔT - change in temperature in supply medium across the central station
equipment heat exchanger, $^{\circ}$F

Inspection of equation (2-2) reveals that for a given flow of wastewater,

the quantity of heat available is directly proportional to ΔT. Theoretically,

heat could be removed from a water source down to approximately 32°F.

However, due to equipment operating characteristics, a practical lower limit

of 40°F can be set. General HVAC system design practice dictates a

$\Delta T \simeq 10^{\circ}$F. However, for this particular application, larger ΔT's may be

necessary or desirable. Therefore, in order to bracket the operating conditions

which may be in effect at the central station, ΔT's of 10°F (typical of

HVAC operation), 15°F, and 20°F (the approximate maximum which could be

expected under the wastewater conditions at Almand Branch) will be considered.

Estimates of the thermal energy available for reclaim at Conyers' Almand

Branch sewage treatment plant can now be calculated.

As an example, and utilizing equation (2-2), the energy reclaimed from

community wastewater would be calculated thusly:

$$\text{AVG. DAILY FLOW} - 623{,}000 \text{ gallons}$$

$$\Delta T - 20^{\circ}$$

$$Q_{DAILY} = 623{,}000 \, \frac{gal}{day} \times (8.33 \, \frac{lbm}{gal}) \times (1.0 \, \frac{Btu}{lbm\text{-}^{\circ}F}) \times (20^{\circ}F)$$

$$= 103{,}791{,}800 \, \frac{Btu}{day}$$

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Derived through this estimative methodology, Table 8-4 summarizes

the quantities of thermal energy potentially available, on a daily and annual

basis, to an HP-WHR system installed at the proposed location in Conyers

and operating at the various conditions discussed in the foregoing paragraphs.

TABLE 8-4
HEAT AVAILABLE FROM WASTEWATER
CONYERS, GA.
ALMAND BRANCH WASTEWATER FACILITY

ΔT (°F)	WASTEWATER CONDITIONS		Q_{DAILY} AVG. DAILY HEAT RECOVERY (BTU/DAY)
	IN(°F)	OUT(°F)	
10	62	52	51.90×10^6
15	62	47	77.84×10^6
20	62	42	103.79×10^6

An important consideration in the application of an HP-WHR system to a

community is the balance between the portions of the community served by the

wastewater collection and treatment system and the portion of the community

proposed for service by the heat pump system. Based upon the above estimate

of thermal energy available at the Conyers plant and upon the assessment of

the proposed service area, the space conditioning requirements of approximately

400 houses could be served under this system application. From the measured

average flow rate of 623,000 gallons per day and utilizing the 100 gallon

per capita per day design figure for wastewater consumption (from Chapter 2.0),

an equivalent wastewater system service area of 6230 persons can be defined.

Further agregating the population served by the average 3.26 persons per

household in Conyers[56] defines an equivalent service area of 1911 houses.

The balance between the wastewater system service area (effectively the thermal

energy supply to the heat pump system) and the HP-WHR system service area is

thus 1911 houses to 400 houses, or approximately 4.8 to 1. It should be re-emphasized that, as discussed in Chapters 2.0 and 4.0, variations in climate can significantly affect the balance ratio, even to the extent of doubling the required number of houses served by the wastewater system for each house on the HP-WHR system. However, for this particular application and for communities exhibiting similar characteristics, the 4.8 to 1 balance ratio appears quite workable.

8.3 CONYERS SYSTEM DESCRIPTION

The Conyers community proposed as the system service area offers several unique characteristics which enhance the viability of application of the Heat Pump-Wastewater Heat Recovery System concept. First, the wastewater treatment plant proposed for service as the HP-WHR system thermal source is located in close proximity to the anticipated service area. Second, the anticipated service area is comprised of a mixture of existing residential units and land developed for residential construction in the immediate future. This mixture represents an excellent market potential for the heat pump system in new homes as well as for the retrofit of existing units. Third, the anticipated service area is projected, due to the historical high growth rate trend of the Conyers community, to reach its ultimate build-out potential within five to ten years. This rapid expansion will allow phasing of the central plant and distribution system to accommodate increases in thermal load and number of units served. Finally, the proposed service area is located outside the Conyers corporate limits and within the jurisdiction of the county government. With one less government entity to deal with, the creation of an authority-type ownership and operation arrangement whould be somewhat simplified.

The basic system concept for the Conyers HP-WHR facility is essentially the same as the cascaded heat pump concept presented in the initial system discussion of Chapter 1.0. One addition will be made to the system in the form of an evaporative cooling device for the summer (cooling) mode of operation. The evaporative cooling device (cooling tower) will be used in lieu of the central heat pump and will reject the heat gain of the housing units to the atmosphere rather than into the wastewater effluent. Figure 8-8 is a single

COOLING TOWER

WASTEWATER MAINS

WASTEWATER SUMPS

WASTEWATER PUMPS

CIRCULATING PUMPS

CENTRAL STATION HEAT PUMP

DISTRIBUTION PUMPS

END-USER HEAT PUMPS

THERMAL STORAGE
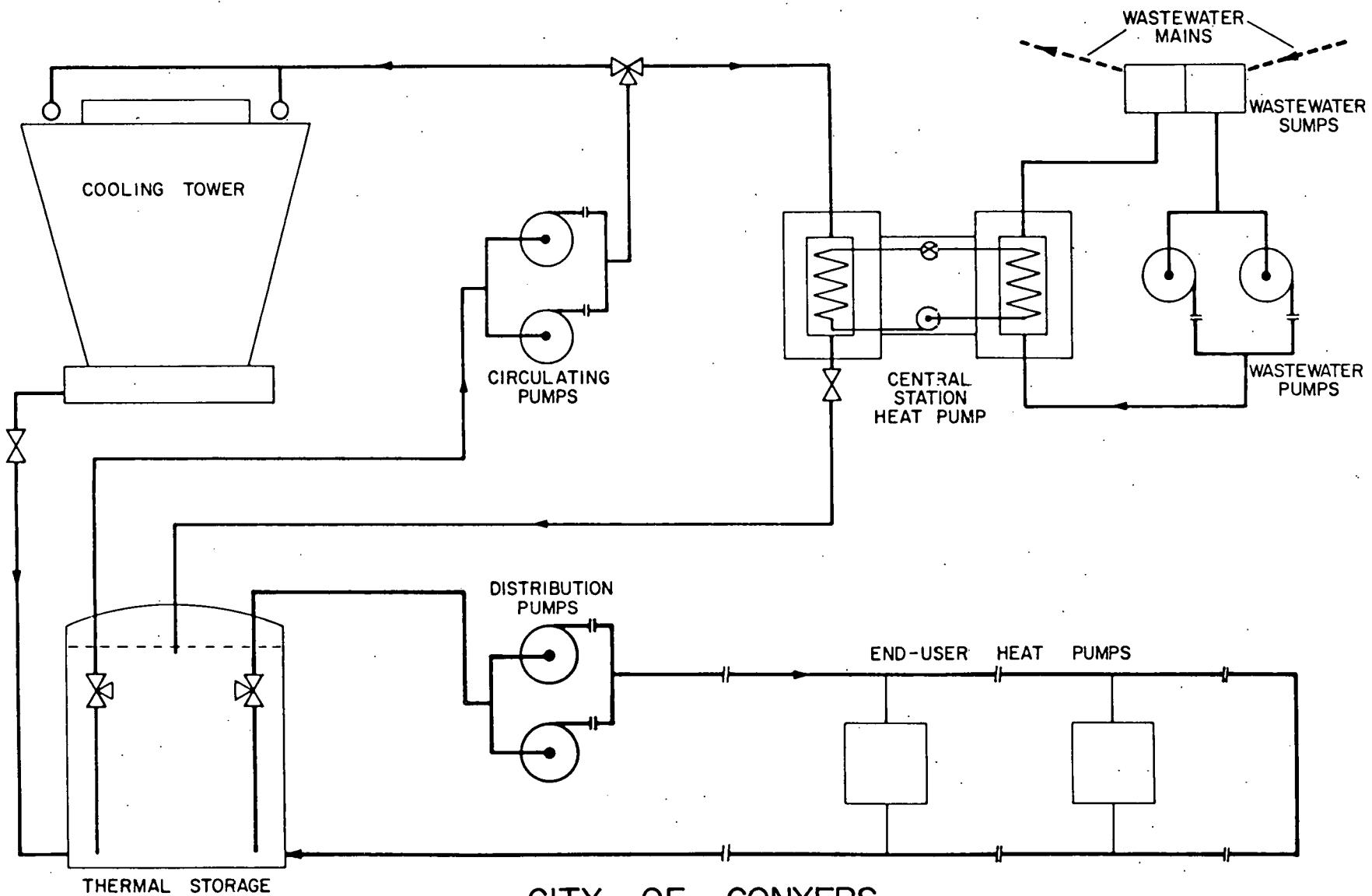
CITY OF CONYERS

HP-WHR SYSTEM SCHEMATIC

FIGURE 8-8

line schematic diagram of the basic system design.

The system is designed for operation in a heating or cooling mode. In the heating mode, wastewater is extracted from the treatment process immediately after primary clarification and pumped by the sewage pumps from the sump to the central heat pump. Low grade thermal energy is extracted from the wastewater in the heat pump evaporator, and the sewage is returned to the treatment process. The thermal energy is transferred, via the refrigerant cycle, into the thermal transport media (potable water) at the heat pump condenser. The transport media then flows to the thermal storage vessel. The valve and piping arrangement inside the vessel and on the suction line of the distribution pumps takes advantage of the temperature stratification within the tank to extract the warmest media at the uppermost fluid levels for distribution. Water is distributed into the community at approximately 90°F with return at approximately 80°F. The cooler transport media at the lower fluid levels within the vessel is extracted and recirculated through the circulating pumps and the central heat pump to be thermally "re-charged". The cooling tower circuit is effectively excluded from the system in the heating mode through proper valving.

In the cooling mode, the cooler transport media, approximately 80°F, is extracted from the lower levels of the storage vessel and distributed into the community. Warm water, at approximately 90°F, is returned from the community and dumped into the thermal storage vessel, where it immediately stratifies according to the fluid temperatures within the vessel. The warmer media is then extracted from the upper fluid levels and circulated via the circulating pumps through the cooling tower where the excess thermal energy is rejected into the atmosphere. The now-cool media is then returned from the cooling tower basin to the storage vessel and again made available for distribution

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into the community. In this mode, the central heat pump component is effectively excluded from the system through proper valving.

In either mode, the end-user heat pumps are cycled in response to individual building loads, utilizing the thermal transport media as a heat source or sink, as appropriate.

During mild winter conditions and during the transitional seasons, it is possible that simultaneous heating and cooling may occur at different locations on the system. This situation can be expected particularly when the system serves a composite service area of commercial and residential structures. Operation in this mode will tend to benefit overall system performance because the heat rejected into the thermal transport media during space cooling is an effective energy addition which can be utilized at other locations on the system. In effect, the system will act to transfer unwanted heat from one location to another where it is required.

The HP-WHR system central station will be sited on the grounds of the existing wastewater treatment facility, with the central station layout as illustrated on Figure 8-9. Piping interconnection with the existing treatment system will be made in such a manner so as to enable the HP-WHR system to utilize the full retention capacity of the two oxidation ditch components as effective warm wastewater storage.

The warm water distribution system into the community will be designed and routed as illustrated in Figure 8-10. One major distribution trunk line will be routed out of the southwest corner of the central plant/sewage treatment plant site and into the Zone 1 service area. A second distribution trunk line will be routed from the southeastern corner of the site, skirting the boundaries of the undeveloped Zone 2, and into the partially developed Zone 3.

THERMAL STORAGE

THERMAL STORAGE

COOLING TOWER

CENTRAL STATION

① ② ④ ③

⑤ ⑥

OPERATIONS AND MAINTENANCE

⑦

ACCESS ROAD

PUMPING STATION

WASTEWATER INFLUENT

COMMINUTOR & BAR SCREEN

FLOW

FLOW

OXIDATION DITCHES

CLARIFIERS
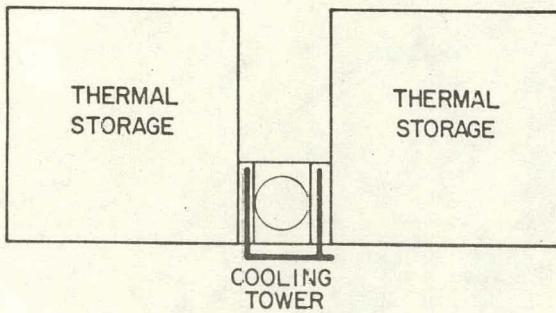
LEGEND

1. CONTROL CABINET
2. CIRCULATION PUMPS
3. DISTRIBUTION PUMPS
4. WATER TREATMENT
5. ELECTRICAL CABINET
6. HEAT PUMP
7. PUMPING STATION

CITY OF CONYERS, GA.

HP-WHR SYSTEM
PROPOSED CENTRAL STATION LAYOUT

FIGURE 8-9

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CITY OF CONYERS
2.75 MILES

HP-WHR
CENTRAL PLANT

ALMAND BRANCH
TREATMENT PLANT

ALMAND CREEK

CURRY CIR.

ZONE 3
207 ACRES

MANHOLE
(TYPICAL)

ALMAND BRANCH RD.

ZONE 2
(UNDEVELOPED)
116 ACRES

MORRIS DR.

5"

ZONE 1
149 ACRES

12"

10"

STANTON RD.

STOCKBRIDGE HWY. 138

EBENEZER RD.

POTTS RD.

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0   300'   600'

CONYERS, GEORGIA HP-WHR SYSTEM
SERVICE AREA AND DISTRIBUTION PIPING
FIGURE 8-10
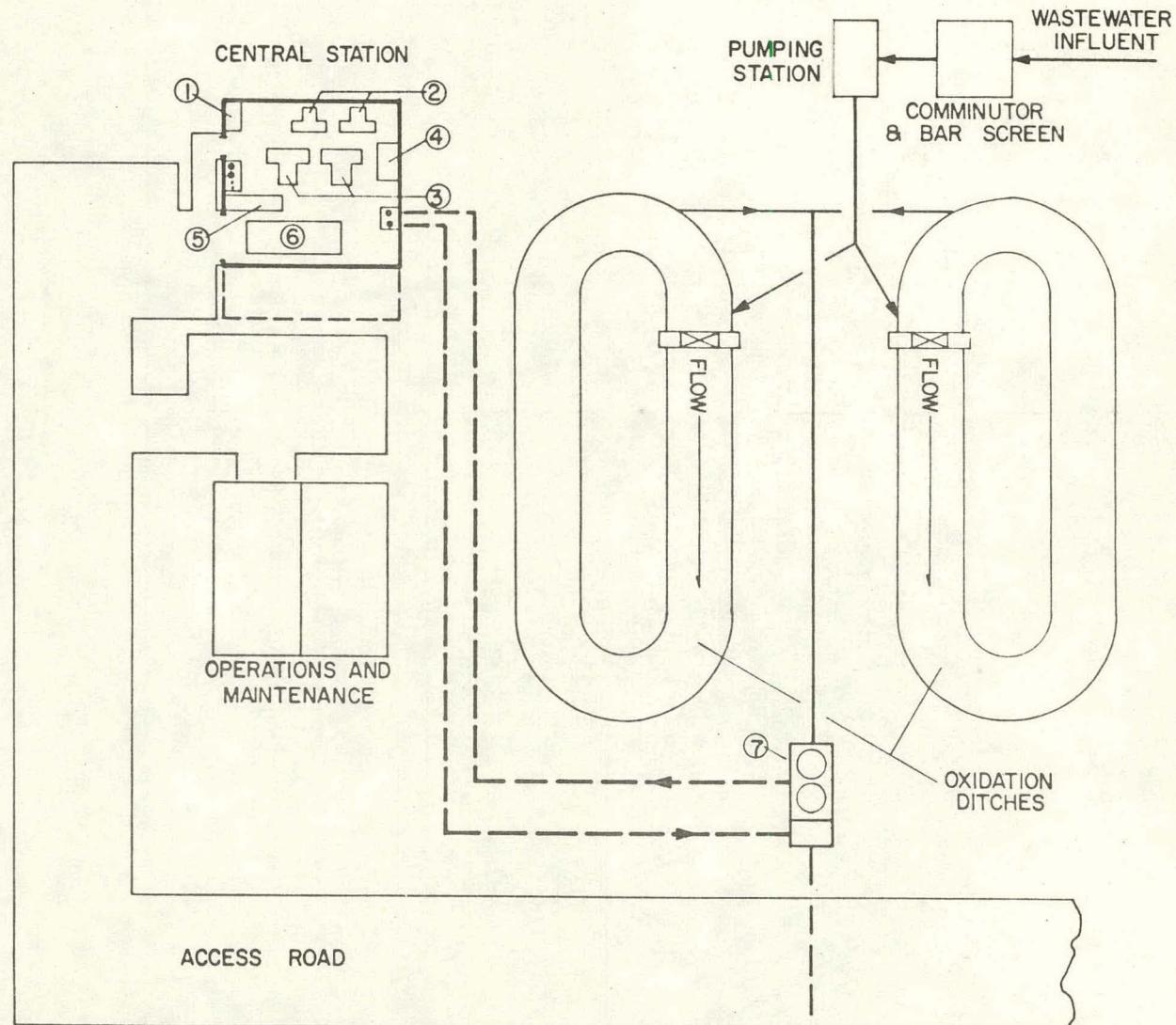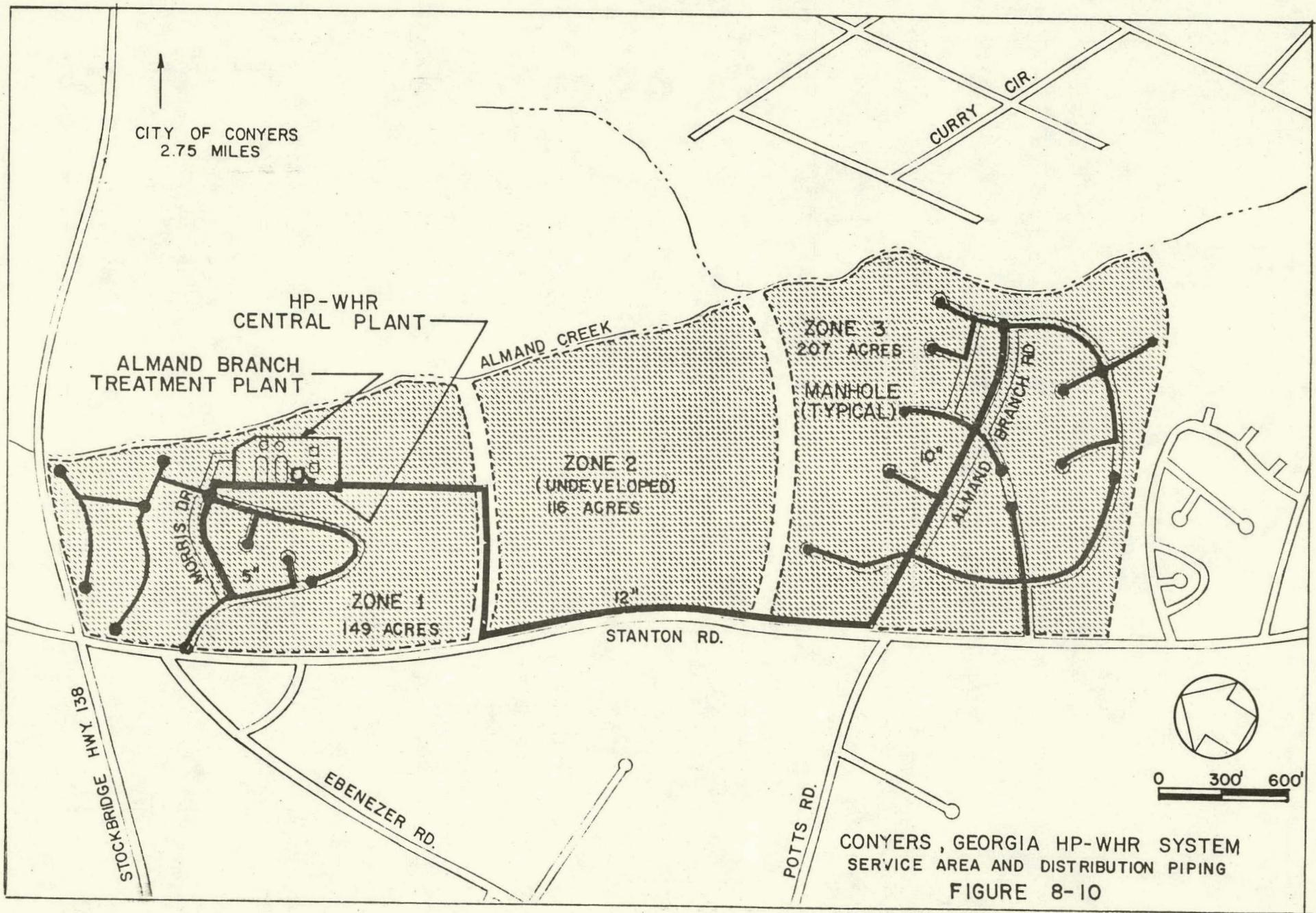
This line will be appropriately sized to handle the full build-out potential of both zones. Feeder lines for each zone are specified and routed as per typical water distribution system design practice.

System design and component specification for the Conyers HP-WHR system is presented in the following report sections.

8.3.1 SYSTEM DESIGN CALCULATIONS

The major components comprising the HP-WHR system which are to be specified in a system development study include the central heat pump, the thermal storage vessel, the cooling tower, auxiliary pumps, and end-user heat pumps. However, since for any system the end-user heat pump specification will be necessarily structure-specific, sizing of those components will not be addressed in this report section. The remainder of the major component design calculations are presented in the following paragraphs.

Central Heat Pump

The central heat pump will be specified to deliver the estimated average daily heating requirement during the peak heating month of January. The equivalent load on the central heat pump is determined by first calculating the heat to be extracted from the wastewater flow through the following equation:

$$Q_{wastewater} = \frac{\{[Q_{heat\ req'd}(1 - \frac{1}{C.O.P._{end-user}})] \times (1 - \frac{1}{C.O.P._{central\ heat\ pump}})\}}{DAYS\ IN\ PERIOD}$$

Substituting the following values:

$$Q_{heat\ req'd} = 2.915 \times 10^9 \text{ Btu-from Table 8-2}$$

$$C.O.P._{end-user} = 2.9 - \text{selected as a median value for the heating performance of the hydronic units assessed in Reference 7}$$

$$C.O.P._{central\ heat\ pump} = 4.5\text{-from the C.O.P. vs. Capacity Curve of Reference 5}$$

Thus,

$$Q_{wastewater} = \frac{[2.915 \times 10^9 (1- \frac{1}{2.9})\,](1- \frac{1}{4.5})}{31}$$

$$= 47.92 \times 10^6 \text{ Btu/day}$$

Thermal storage will be provided in the system in order to lower the nominal peak load seen by the central station equipment. Thus, the central heat pump will be sized to provide the above heating requirement over 18 hours, operating at nominal full load. Operation on this basis yields:

$$\text{Heat Extraction Rate} = \frac{47.92 \times 10^6}{18}$$

$$= 2.66 \times 10^6 \text{ Btu/hr}$$

Conditions at the central heat pump evaporator dictate wastewater intake at $62^\circ F$ (from section 8.2.8) and a ΔT drop of $10^\circ F$. Thus, the required waste-water flow rate is determined by the equation below:

$$\text{Flow Rate} = \frac{HER}{(8.33\ \frac{lbm}{gal})(60\ \frac{min}{hr})(10^\circ F)(1.0\ \frac{Btu}{lbm-^\circ F})}$$

$$= \frac{2.66 \times 10^6}{(8.33)(60)(10)(1.0)}$$

$$= 532 \text{ GPM}$$

At this point it may be noted that the required wastewater flow rate exceeds the _average_ instantaneous flow through the treatment plant - 432 GPM (0.623 MGD/(24)(60)). However, it should be reiterated that central heat pump operation is specified to provide the average daily load over an 18 hour period and that the resulting daily required flow is then 0.574 MGD (532 GPM x 60 x 18). The additional flow - 100 GPM average during central heat pump operation - can be obtained when necessary through pulldown of the thermal supply medium - wastewater - from the retention capacity of the oxidation ditch components of the wastewater treatment facility.

Two other options for central heat pump operation also exist which would alleviate the apparent flow rate mismatch. The first option would require supplying the estimated average daily heating requirement over a longer period of operation. If the design flow rate of wastewater through the central heat pump evaporator is reduced to the average flow figure of 432 GPM, the necessary heat extraction rate would be reduced to 2.16×10^6 Btu/hr, and the time of operation would be increased to approximately 22 hours.

The second option would involve specifying a slightly larger ΔT across the evaporator. At the calculated heat extraction rate of 2.66×10^6 Btu/hr (for 18 hour operation) and utilizing the average flow figure of 432 GPM, the required ΔT is found to be 12.3^oF, obviously within the range of reasonable operation for the type machine being considered for service.

It is expected, however, that specifying the central heat pump evaporator for the extraction rate of 2.66×10^6 Btu/hr and flow rate of 532 GPM will provide an appropriately conservative design.

The operating parameters at the condenser can be defined through similar calculations:

$$\text{Heat Rejection Rate} = \text{HER} \left(1 + \frac{1}{\text{C.O.P.}}\right)$$

$$= 2.66 \times 10^6 \left(1 + \frac{1}{4.5}\right)$$

$$= 3.25 \times 10^6 \text{ Btu/hr}$$

System circulating water intake to the condenser will be at 80°F and a rise of 10°F will be achieved across the heat exchanger.

$$\text{Flow Rate} = \frac{\text{HRR}}{(8.33)(60)(10)(1.0)}$$

$$= \frac{3.25 \times 10^6}{(8.33)(60)(10)(1.0)}$$

$$= 650 \text{ GPM}$$

The compressor for the central station unit will receive shaft power from an electric motor drive; performance will be on the following basis:

$$W_c = \frac{(\text{HER})\left(\frac{1}{\text{C.O.P.}}_{\text{central heat pump}}\right)}{3413 \frac{\text{Btu}}{\text{KWH}}}$$

$$= \frac{(2.66 \times 10^6)}{(4.5)(3413)}$$

$$= 173 \text{ KW}$$

The central heat pump will be constructed with shell-and-tube condenser and evaporator and a centrifugal compressor. The heat pump will have approximately the same physical dimensions and appearance as a nominal 250 ton centrifugal water chiller used in air conditioning applications. Due to the comparatively low capacity requirement for the central station machine, one unit will be specified to serve the full requirements of the HP-WHR system. The reliability of this type machine is typically high; however, thermal storag

will be appropriately sized to provide effective back-up capacity in the event of central heat pump failure.

Thermal Storage

Inspection of the wastewater flow cycle at the subject Conyers treatment plant (presented in section 8.2.8) reveals that the characteristic low flow period during the seven-day flow cycle had an approximate 60 hour duration. In order to provide adequate thermal energy supply capacity for the system during these low flow periods, the thermal storage component will be sized for a nominal 72 hour period (making an allowance for a + 20% variance in duration from the flow cycle documented in section 8.2.8) under January heating demand conditions. The nominal 72 hour capacity will also provide an adequate downtime "window" for the central heat pump in the event of a failure requiring emergency repairs or for necessary scheduled maintenance.

Allowable temperature pulldown within the vessel will be designed at $40^{\circ}F$. The primary consideration in defining this range is the effect on end-user heat pump performance. According to the equipment assessment of Reference 7, the performance of a hydronic heat pump, with nominal C.O.P. of 2.910 at Standard ARI rating conditions (the figure previously utilized in assessing central heat pump thermal loads) improves to a C.O.P. of 3.084 at the HP-WHR design condition of $90^{\circ}F$ supply water. Conversely, as supply water temperature drops to the design lower limit of $50^{\circ}F$, C.O.P. drops only to 2.852. Therefore, it is expected that the $40^{\circ}F$ pulldown will provide more than adequate thermal storage capacity, and that during all but the peak heating period, end-user heat pump performance will equal or exceed the nominal 2.91 C.O.P.

The physical capacity of the thermal storage component is computed as

follows:

$$Q_{\text{thermal storage}} = \frac{(2.915 \times 10^9 \text{ Btu})(1 - \frac{1}{2.9})}{744 \text{ hrs}} \times 72 \text{ hrs}$$

$$= 185.16 \times 10^6 \text{ Btu}$$

$$M_{\text{water storage}} = \frac{185.16 \times 10^6 \text{ Btu}}{(40^{\circ}\text{F})(1.0 \frac{\text{Btu}}{\text{Lbm}-^{\circ}\text{F}})}$$

$$= 4,628,908 \text{ Lbm}$$

$$= 555,691 \text{ GALLONS}$$

$$= 74,300 \text{ FT}^3$$

This necessary capacity will be provided in the form of two square tanks with dimensions of 45' x 45' x 20'(depth). The vessels will be field constructed of steel reinforced concrete. Installation will be partially below grade with the exposed vertical surfaces being earth bermed for improved heat retention.

Cooling Tower

The cooling tower component of the Conyers HP-WHR system will be specified to dissipate the average daily heat gain on the system during the peak cooling month of July, as determined below:

$$Q_{\text{avg clg load}} = \frac{Q_{\text{monthly}} (1 + \frac{1}{\text{C.O.P.}_{\text{end-user}_{\text{cooling}}}})}{\text{DAYS IN MONTH}}$$

$$= \frac{(2.184 \times 10^9)(1 + \frac{1}{2.5*})}{31}$$

$$= 98.28 \times 10^6 \text{ BTU/DAY}$$

*heating C.O.P. of subject hydronic unit = 2.91, cooling C.O.P. = 2.53; from Reference 7

The required heat rejection rate will be determined on the basis of 18 hours operation per day at nominal full load, thus:

$$HRR \ = \ \frac{98.28 \times 10^6}{18}$$

$$= \ 5.46 \times 10^6 \ \text{Btu/hr}$$

System operating parameters dictate a $10^\circ F$ temperature drop in the circulating water. Required cooling tower flow is then determined as:

$$\dot{M}_{\text{cooling tower}} \ = \ \frac{5.46 \times 10^6 \ \frac{\text{Btu}}{\text{hr}}}{(8.33 \ \frac{\text{LBM}}{\text{GAL}})(60 \ \frac{\text{MIN}}{\text{HR}})(10^\circ F)(1.0 \ \frac{\text{BTU}}{\text{LBM-}^\circ F})}$$

$$= \ 1092 \ \text{GPM}$$

A modular, pre-constructed tower with a nominal flow rating of 1200 GPM at the required $10^\circ F$ TD will be selected. Fan horsepower will be 30 HP.

Auxiliary Pumps

The three pump functions on the HP-WHR system will be specified as discussed below:

Wastewater pumps will be variable speed types, each capable of delivering total required flow at design pressure drops. The flow rate required for each pump is 535 GPM at an estimated dynamic head of 60 ft. of water. This head rating makes adequate allowance for piping and central station heat exchanger pressure drops.

Circulating pumps will each be sized for one half of the total required flow of 600 GPM. Pump head required will be approximately 90 ft. of water.

Distribution pumps will each be sized for one half of the total required

flow to the end-users. One pump will be a variable speed type. The total flow through the distribution system will be based on serving the heat dissipation requirements at peak conditions with a 10^OF temperature rise. Based on a nominal heat rejection requirement of 16,740 Btu per ton of air conditioning, the required flow rate for a 782 ton peak load is 2618 GPM. (Heating season peak flow requirement is approximately 1350 GPM.) Total pumping head at peak flow will be 500 ft. of water. This head includes allowances for end-user system and distribution system piping pressure drops.

Distribution Piping

The required circulating water flow rates at peak load will determine distribution pipeline sizing. Peak heating and cooling loads for each zone are first determined through the procedures outlined in section 8.2.7 and utilizing the appropriate zonal data from the computer analysis. The required flow rates per zone are then calculated and piping sizes are selected to yield proper line velocities and acceptable pressure drops.

ZONE R-1

$$\text{PEAK HEATING LOAD} = \frac{[(3.729 \times 10^6)(1-\frac{1}{2.9})](53)}{(701) \qquad (24)}$$

$$= 0.77 \times 10^6 \text{ Btu/hr}$$

$$\text{PEAK COOLING LOAD} = \frac{(55,000 \text{ ft}^2)(16,740 \text{ Btu/hr})*}{550 \text{ ft}^2/\text{PEAK A-C TON}}$$

$$= 1.67 \times 10^6 \text{ Btu/hr}$$

*Heat Rejection Rate based on the 2.53 cooling C.O.P. of the subject hydronic unit.

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$$\dot{M}_{heating} = \frac{(0.77 \times 10^6)}{(8.33)(60)(10)(1.0)}$$

$$= 154 \text{ GPM}$$

$$\dot{M}_{cooling} = \frac{1.67 \times 10^6}{4998}$$

$$= 334 \text{ GPM}$$

ZONE R-2

$$\text{PEAK HEATING LOAD} = \frac{[(1.22 \times 10^9)(1-\frac{1}{2.9})](53)}{(701) \quad (24)}$$

$$= 2.52 \times 10^6 \text{ Btu/hr}$$

$$\text{PEAK COOLING LOAD} = \frac{(180,000)(16,740)}{550}$$

$$= 4.58 \times 10^6 \text{ Btu/hr}$$

$$\dot{M}_{heating} = \frac{(2.52 \times 10^6)}{4998}$$

$$= 504 \text{ GPM}$$

$$\dot{M}_{cooling} = \frac{(5.48 \times 10^6)}{4998}$$

$$= 1096 \text{ GPM}$$

ZONE R-3

$$\text{PEAK HEATING LOAD} = \frac{[(1.32 \times 10^9)(1-\frac{1}{2.9})](53)}{(701) \quad (24)}$$

$$= 2.72 \times 10^6 \text{ Btu/hr}$$

$$\text{PEAK COOLING LOAD} = \frac{(195,000)(16,740)}{550}$$

$$= 5.94 \times 10^6 \text{ Btu/hr}$$

$$\dot{M}_{heating} = \frac{(2.72 \times 10^6)}{4998}$$

$$= 544 \text{ GPM}$$

$$\dot{M}_{cooling} = \frac{(5.94 \times 10^6)}{4998}$$

$$= 1188 \text{ GPM}$$

Obviously, the circulating water flow rates into each zone during the cooling mode will be the governing criteria in distribution pipe size selection. Pipe sizes and the corresponding flow rates, velocities, and pressure drops for each distribution system piping segment are presented in Table 8-5.

TABLE 8-5
DISTRIBUTION PIPING SPECIFICATIONS

LOCATION	FLOW RATE (GPM)	PIPE SIZE (INCHES)	VELOCITY FPS	PRESSURE DROP (PSI/100 FT)
Trunk Line	2618	14"	5.46	.25
Zone R-2 & R-3 Trunk	2284	12"	6.48	.40
Zone R-2 Feeder	1096	10"	4.47	.25
Zone R-3 Feeder	1188	10"	4.85	.30
Zone R-1 Feeder	334	5"	5.46	.90

8.3.2 SYSTEM COMPONENT DESCRIPTIONS

The Heat Pump Waste Heat Recovery Systems analyzed in this report are composed of standard HVAC components commonly used by the residential, commercial and industrial sectors. Some special testing and performance rating may be necessary where the intended HP-WHR application parameters exceed the normal

operating range of the specific equipment involved. A brief description of each component follows:

Central Heat Pump

The central heat pump is a package unit including compressor and electric drive, heat exchangers, operating and safety controls, interconnecting piping and refrigerant charge. Unit construction and performance will conform to the ARI Standard for centrifugal water chillers accepted by the air conditioning industry.

The evaporator and condenser will each be of shell and tube construction. Water box covers should be of the "marine" type so that removal for tube cleaning and replacement may be easily accomplished. The heat exchanger tubes should be seamless copper and will be individually replaceable. The compressor selected for service is of centrifugal design with hermetic electric motor drive. Variable inlet guide vanes at the compressor inlet are used to modulate unit capacity by varying refrigerant flow. Screw or reciprocating compressors may be considered for other applications. The motor used for the compressor drive is a squirrel-cage induction type operating at constant speed. The motor will be refrigerant cooled.

Cooling Tower

The cooling towers chosen are of the induced draft, prefabricated type. They will be mounted on field erected structural steel supports installed over the thermal storage basins. The tower design will include non-combustible casing and fill, lower collection basin, distribution headers and an electric-drive fan assembly.

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Pumps

Distribution system pumps will be the horizontal, split-case type. Pump components should be selected for long lasting service. Recommended materials are bronze and steel alloys commonly used for water pumping service. The pumps will be driven by variable speed electric motors.

Circulating pumps used within the central heat recovery plant will be the horizontal, split-case type or end-suction type depending on the manufacturer. Pumps for this service would be similar to pumps used for condenser water service in typical heating and air conditioning applications.

Pumps delivering wastewater effluent to the central heat recovery system should be of the vertical turbine type for sewage pumping service. A non-clogging impeller should be specified and the inlet to the suction of the pump should be protected by a suction strainer. In some cases the pump will require variable speed motor drive.

Thermal Storage

The thermal storage device will consist of a cylindrical or rectangular tank located schematically between the central heat pump and the distribution system. It will be constructed of steel reinforced concrete and be located partially below ground level. The portions above ground will be earth bermed for structural support and to retard heat loss. The vessel will be insulated with 3/4 inch styrofoam sheathing and will be covered to retard water surface heat loss and evaporation. A piping and valve arrangement within the vessel will serve to make use of temperature stratification within the vessel to provide appropriate temperature supply water to the distribution system or

return water to the central heat pump unit.

Control Systems

Automatic controls for the Heat Pump-Wastewater Heat Recovery Systems will be pneumatic or electric types commonly found in industrial and commercial use for temperature and pressure measurement and valve operation. The limited number of operator functions involved will probably make an electric control system the most economic choice.

Water Treatment System

A water treatment system to eliminate scale build-up and control bacteriological growth will be incorporated into the distribution water system. Packaged systems for proportional chemical feeding and periodic conductivity analysis are commonly used with refrigeration condenser systems and are similar to the type which will be used with the HP-WHR system.

Piping System (at Central Plant)

System piping within the central heat recovery plant will be seamless black steel. Connections to equipment will be made with flanges or removable ring type connectors. Valves and piping accessories will be iron-bodied types for the appropriate service classification.

Piping System (Distribution)

The distribution system piping will be constructed of PVC pressure pipe similar to the types used in potable water distribution systems. Joints will be made by solvent cementing or a mechanical bell and gasket connection. The

use of plastic piping material will eliminate galvanic corrosion problems and simplify installation. Additionally, the plastic system should offer lower pumping resistance than other materials.

8.3.3 <u>EXPECTED PERFORMANCE</u>

In order to develop an estimate of overall system performance which would also take into account the variances in C.O.P. which result from serving the continually changing thermal loads during an annual cycle, an estimate of system performance for each month of a typical year was made. The estimating procedure involved tallying all energy inputs to the system, including central heat pump power, pump work, cooling tower fan work, system auxiliaries power, and end-user heat pump power, and comparing that total (in Btu equivalents) to the net heating or cooling effect provided to the structures within the system service area. Estimating the monthly performance in this manner revealed system performance in the peak heating, peak cooling, and simultaneous heating and cooling modes. Estimates of instantaneous performance were not attempted since thermal storage within the system allows for averaging of absolute peak conditions. The energy consumption figures presented in the performance analysis are considered to be conservative since the level of analysis appropriate to a preliminary study could not take into account operating variables such as the effects of varying condenser water temperature below design conditions or higher than average temperatures at the thermal source during the heating season.

The general assumptions in effect for the performance analysis are as follows:

<u>Distribution Media Temperatures</u>

	HEATING MODE	COOLING MODE
Supply	$90^{\circ}F$	$80^{\circ}F$
Return	$80^{\circ}F$	$90^{\circ}F$

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Pump Work

Pump work for all subsystems was estimated as the work required to transfer the necessary quantities of thermal energy at the appropriate temperatures and pressures. Distribution pump work was estimated on the basis of supplying the required heating or cooling effects at the specified ΔT of $10^{o}F$ by maintaining a constant head on the system and varying flow proportionately with thermal load. All pump efficiencies were assumed to be 65%.

Cooling Tower Work

The cooling tower was assumed to operate at full load (as per design specifications) to dissipate the estimated monthly heat gain. No allowance was made for improved tower performance due to decreased ambient wet bulb temperature.

End-User Heat Pumps

End-user equipment was assumed to operate at the nominal heating and cooling C.O.P.'s of 2.91 and 2.53, respectively (from Reference 7).

The results of the monthly performance analysis are summarized in Table 8-6 below. The full analysis and calculational procedures are presented in Appendix C.

TABLE 8-6
CONYERS HP-WHR SYSTEM PERFORMANCE

	Heating Btu Delivered to Space X 10^9	Cooling Btu Delivered to Space X 10^9	Central Plant Energy Input KWH	End-User Energy Input KWH	SYSTEM C.O.P.
Jan.	2.915	0	125,411	294,512	2.034
Feb.	2.335	0	98,856	235,913	2.044
March	1.845	0.060	92,247	193.439	1.954
April	0.593	0.146	17,742	77,924	2.285
May	0.116	0.834	21,090	109,464	2.132
June	0	1.737	44,092	203,574	2.055

(Table 8-6 Cont'd)

TABLE 8-6 (Cont'd)

	Heating Btu Delivered to Space X 10^9	Cooling Btu Delivered to Space X 10^9	Central Plant Energy Input KWH	End-User Energy Input KWH	System C.O.P.
July	0	2.184	57,236	255,962	2.043
Aug.	0	2.098	80,560	245,883	1.883
Sept.	0.039	1.230	31,930	148,095	2.065
Oct.	0.568	0.310	3,679	93,719	2.641
Nov.	1.703	0	71,825	172,060	2.046
Dec.	2.786	0	118,260	281,479	2.042
TOTAL	12.900	8.599	762,928	2,311,124	

With the monthly system performance thus defined for each month during the annual cycle, the overall system C.O.P. can be easily derived. Considering the system as a "black-box," its performance can be determined by the expression

$$C.O.P. = \frac{Q_{out}}{W_{in} \times 3413}$$

where

Q_{out} = heating or cooling effect provided by the system, Btu

W_{in} = work input to the system, KWH

Thus, substituting from Table 8-6:

$$C.O.P. = \frac{(12.900 \times 10^9 + 8.599 \times 10^9)}{(762,928 + 2,311,124) \times 3413}$$

$$= 2.049$$

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COMPONENT ENERGY USE

In order to identify the areas of energy usage on the HP-WHR system which would benefit system performance the most through decreased consumption, the performance analysis also included a breakdown of individual component energy usage. Overall central plant power consumption was first analyzed and plotted as illustrated in Table 8-7 and Figure 8-11. Consumption was then further broken down according to individual components, as summarized in Table 8-8.

It is apparent from the component analysis and from the monthly consumption figures presented in Appendix C that system performance is severely penalized by auxiliaries such as pumps and cooling towers. Obviously then, this area of energy consumption exhibits the potential for improvement if equipment offering better than average operating efficiencies can be economically specified.

By far the largest energy user on the system is the collective end-user heat pump. Prototype hydronic heat pumps which offer nominal C.O.P.'s in the 4.0 - 5.0 range (as opposed to the 2.91 assumed for the analysis) are currently under development and show vast potential, if incorporated into the HP-WHR system, to improve system performance by decreasing end-user heat pump energy consumption while delivering the same thermal load.

Finally, in actual practice some improvement in performance may be gained through operating practices such as producing water at higher or lower than design temperatures at nominal central heat pump full loads during off-peak hours in order to "charge" the thermal storage component of the system.

TABLE 8-7

ELECTRICITY USAGE ANALYSIS
FOR CONYERS HP-ICES
IN COOPERATION WITH THE TECHNOLOGY DEVELOPMENT LAB
GEORGIA TECH ENGINEERING EXPERIMENT STATION
ATLANTA, GEORGIA 30332

I N P U T   D A T A

BILLING PERIOD	ACTUAL DEMAND	BILLING DEMAND	CONSUMPTION	FUEL ADJUSTMENT	LOAD FACTOR
JANUARY,1979	680	523	125411	.000	.25
FEBRUARY,1979	680	523	98856	.000	.22
MARCH,1979	680	523	92247	.000	.18
APRIL,1979	630	523	17742	.000	.04
MAY,1979	600	523	21090	.000	.05
JUNE,1979	550	550	44092	.000	.11
JULY,1979	550	550	57236	.000	.14
AUGUST,1979	550	550	80560	.000	.20
SEPTEMBER,1979	550	550	31930	.000	.08
OCTOBER,1979	630	523	3679	.000	.01
NOVEMBER,1979	680	523	71825	.000	.15
DECEMBER,1979	680	523	118260	.000	.23

LOCAL TAX IS $ .03

P R O J E C T E D   E L E C T R I C I T Y   C O S T

BILLING PERIOD	BILLING AMOUNT
JANUARY	5658.20
FEBRUARY	4994.70
MARCH	4670.23
APRIL	1012.41
MAY	1176.78
JUNE	2306.06
JULY	2951.37
AUGUST	4096.46
SEPTEMBER	1708.97
OCTOBER	250.37
NOVEMBER	3667.61
DECEMBER	5520.72
TOTAL	$ 38013.87

PLOT OF CONSUMPTION

FIG. 8-11

CONSUMPTON(KWH)

130000

98250

66500

34750

3000

J   F   M   A   M   J   J   A   S   O   N   D   J

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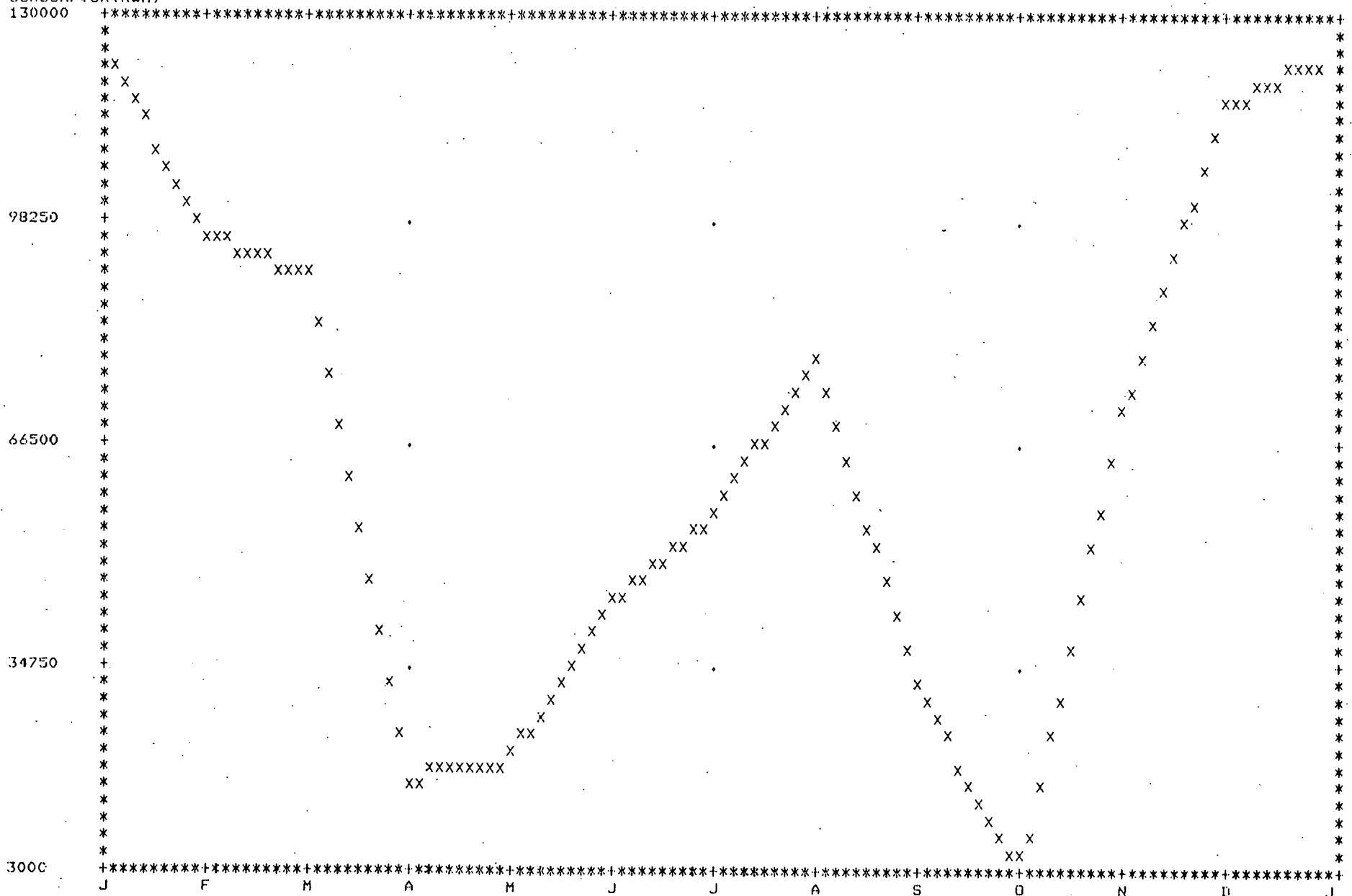
TABLE 8-8
CONYERS HP-WHR SYSTEM
ANNUAL COMPONENT ENERGY USE

ITEM	ENERGY USE KWH	PERCENT OF CENTRAL PLANT	PERCENT OF SYSTEM
Distribution Pumps	141,056	18.5	4.6
Circulating Pumps	128,812	16.9	4.2
Sewage Pumps	24,011	3.1	0.8
Cooling Tower	48,807	6.4	1.6
Misc. Electrical	12,000	1.6	0.4
Central Heat Pump	408,242	53.5	13.3
Central Plant Total	762,928	100.0	24.9
End-user Heat Pumps	2,311,124		75.1
System Total	3,074,052 KWH		100.0

ENERGY CONSERVATION POTENTIAL

The Heat Pump-Wastewater Heat Recovery System shows the potential to favorably alter the existing energy use patterns within a community and to promote the overall conservation of primary energy sources. In the case of the Conyers community, the existing energy use pattern shows a reliance on electric energy for cooling and natural gas for heating. Essentially all end-users within the proposed HP-WHR system service area now exhibit this energy usage pattern.

The energy consumed on-site by the conventional space conditioning systems in meeting the heating and cooling requirements projected by the computerized load estimate was first calculated. The assumptions were that heating requirements were met with natural gas fired furnaces operating at a 65% driven efficiency and that cooling requirements were met with electrically driven air-conditioning units with C.O.P.'s of 2.0. Based on these assumptions the conventional systems consume annually the following amounts of energy:

	Cooling	Heating
Zone R-1	161,148 KWH	25,385 THERMS
Zone R-2	527,395 KWH	83,077 THERMS
Zone R-3	571,345 KWH	90,000 THERMS
TOTAL	1,259,888 KWH	198,462 THERMS

The above totals compare to the annual total HP-WHR system energy requirement of 3,074,052 KWH (from Table 8-8). It is duly noted that the HP-WHR system consumption represents an approximate 144% increase in electrical energy consumption. However, on-site consumption of natural gas is reduced to zero. The rise in electrical energy consumption is attributable to two factors:

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1) the demand for natural gas in satisfying heating requirements is effectively displaced with the electrical demand of the heat pump system, and 2) the overall heat pump system operates at a slightly lower efficiency (lower C.O.P.) than the conventional electrically driven air conditioning units in meeting cooling requirements.

In order to compare the projected primary energy resource consumption of the HP-WHR system to that of the conventional space conditioning systems, the procedures for determining annual fuel and resource energy utilization, as outlined in ASHRAE Standard 90-75, ENERGY CONSERVATION IN NEW BUILDING DESIGN,[58] were followed. As can be seen from the results of this assessment as illustrated on the following pages, the primary effect of the HP-WHR system operation would be to displace the consumption of natural gas with that of coal and nuclear energy. A minor increase will also be noted in the consumption of hydro-power and crude oil. A secondary, but desirable, effect would be to free the additional quantities of natural gas for use in satisfying other, less flexible energy requirements.

Line	Column	Building/Project Energy Req'ments		Fuel and Energy Supplied to Site								
		A1	A2	B1	B2	B3	B4	B5	B6	B7	B8	B9
	Function	Thermal 10^6 Btu	Electric 10^3 KWH	Coal 10^6 Btu	Gas 10^6 Btu	Light Oil 10^6 Btu	Heavy Oil 10^6 Btu	Elec. Win. 10^3 KWH	Elec. Sum. 10^3 KWH	Elec. Ann. 10^3 KWH	Other	Other
1	Heating	12,900	/////		19,846				/////			
2	Cooling	8599	/////					/////	1259.8			
3	Water Heating		/////									
4	HVAC Auxiliaries											
5	Lighting	/////										
6	Elevators	/////										
7	Computers	/////										
8	Cooking											
9	Process											
10	Other											
11	Other											
12	Other											
13	Total Carry Fwd. to Form 12-2				19,846				1259.8			

CONYERS-CONVENTIONAL SYSTEMS

Line	Annual Fuel and Energy Calculation Form 12-2 Fuel and Energy Supplied to Site			C.O. Total From Form 12-1 Line 13	RUF From Supplier or From Tables	Fuel and Energy Resources Used on Site and Off Site To Meet Energy Requirements of Building/Project						
						C1 S. Tons Coal	C2 MCF Nat'l	C3 BBL Crude Oil	C4 Grams U-235	C5 10^3 KWH Hydro	C6 Other	C7 Other
14	Fuel Oil, Light											
15	Fuel Oil, Heavy											
16	Gas	Nat'l	MCF	19,846	1.16		23,021					
		Oil	BBL	/////								
17	Coal											
18	Elec. Winter			-0-	/////							
		Coal	S. Tons	/////	.27							
		Gas	MCF	/////	.56							
		Oil	BBL	/////	.53							
		Nuc	Grams	/////	3.84							
		Hydro	10^3 KWH	/////	.07							
		Other										
19	Elec. Summer			1259.8	/////							
		Coal	S. Tons	/////	.26	327.5						
		Gas	MCF	/////	1.04		1310.2					
		Oil	BBL	/////	.56			705.5				
		Nuc	Grams	/////	3.84				4837.6			
		Hydro	10^3 KWH	/////	.04					50.39		
		Other										
20	Elec. Annual			/////	/////							
		Coal	S. Tons	/////								
		Gas	MCF	/////								
		Oil	BBL	/////								
		Nuc	Grams	/////								
		Hydro	10^3 KWH	/////								
		Other										
21	(Other)											
22					Total Resources	327.5	24,331.2	705.5	4837.6	50.39		

ASHRAE STANDARD 90-75

Line	Column / Function	A1 Thermal 10⁶ Btu	A2 Electric 10³ KWH	B1 Coal 10⁶ Btu	B2 Gas 10⁶ Btu	B3 Light Oil 10⁶ Btu	B4 Heavy Oil 10⁶ Btu	B5 Elec. Win. 10³ KWH	B6 Elec. Sum. 10³ KWH	B7 Elec. Ann. 10³ KWH	B8 Other	B9 Oth
		Building/Project Energy Req'ments		Fuel and Energy Supplied to Site								
1	Heating	12,900	▨					1876.2	▨			
2	Cooling	8599	▨					▨	1197.9			
3	Water Heating		▨									
4	HVAC Auxiliaries											
5	Lighting	▨										
6	Elevators	▨										
7	Computers	▨										
8	Cooking											
9	Process											
10	Other											
11	Other											
12	Other											
13	Total Carry Fwd. to Form 12-2							1876.2	1197.9			

CONYERS-HP-WHR SYSTEM

Line	Fuel and Energy Supplied to Site			C.O. Total From Form 12-1 Line 13	RUF From Supplier or From Tables	C1 S. Tons Coal	C2 MCF Nat'l	C3 BBL Crude Oil	C4 Grams U-235	C5 10³ KWH Hydro	C6 Other	C7 Other
						Fuel and Energy Resources Used on Site and Off Site To Meet Energy Requirements of Building/Project						
14	Fuel Oil, Light											
15	Fuel Oil, Heavy											
16	Gas	Nat'l	MCF	–0–	1.16							
		Oil	BBL	▨								
17	Coal											
18	Elec. Winter			1876.2	▨							
		Coal	S. Tons	▨	.27	506.6						
		Gas	MCF	▨	.56		1050.7					
		Oil	BBL	▨	.53			994.4				
		Nuc	Grams	▨	3.84				7205			
		Hydro	10³ KWH	▨	.07					131.3		
		Other										
19	Elec. Summer			1197.9	▨							
		Coal	S. Tons	▨	.26	311.5						
		Gas	MCF	▨	1.04		1245.8					
		Oil	BBL	▨	.56			670.8				
		Nuc	Grams	▨	3.84				4600.			
		Hydro	10³ KWH	▨	.04					47.9		
		Other										
20	Elec. Annual				▨							
		Coal	S. Tons	▨								
		Gas	MCF	▨								
		Oil	BBL	▨								
		Nuc	Grams	▨								
		Hydro	10³ KWH	▨								
		Other										
21	(Other)											
22				Total Resource		818.1	2296.5	1665.2	11,805	179.2		

SECOND LAW PERFORMANCE ANALYSIS

A second law performance analysis was performed for the Conyers HP-WHR
system during peak heating season, peak cooling season and a period when both
heating and cooling was required. The analysis was conducted on a monthly
basis.

Heating was accomplished by heating air from $75^{\circ}F$ to $120^{\circ}F$ and cooling
was provided by cooling air from $75^{\circ}F$ to $55^{\circ}F$. The reference state for the
air stream availabilities was $75^{\circ}F$. In determining the primary resource
utilization of electric energy a second law efficiency of .33 was used. The
results are tabulated below.

CONYERS HP-WHR SYSTEM
SECOND LAW ANALYSIS

SEASON	HEATING SUPPLIED	COOLING SUPPLIED	AVAILABILITY OF HEATING & COOLING SUPPLIED	AVAILABILITY OF ENERGY SUPPLIED TO SYSTEM	SECOND LAW EFFICIENCY	AVAILABILITY OF ENERGY SUPPLIED TO SYSTEM	SECOND LAW EFFICIENCY
January (Heating)	2.915×10^9	-0-	1.161×10^8	4.34×10^9	.02670	4.484×10^9	.0258
July (Cooling)	-0-	2.184×10^9	4.186×10^7	3.23×10^9	.01296	3.008×10^9	.01391
October (Mixed)	$.568 \times 10^9$	$.310 \times 10^9$	2.854×10^7	9.69×10^8	.0294	1.3009×10^9	.0219

8.3.4 EXPECTED ECONOMICS

This report section evaluates the expected economic performance of the HP-WHR system as compared to conventional heating and cooling systems in the Conyers area. First costs, operating costs and maintenance costs for each of the compared systems are estimated in order to determine annualized costs to consumers. A life cycle cost comparison for the system is also made.

System design and performance has been presented in previous sections of this report. For the purposes of this analysis it is assumed that some form of government entity will own and operate the central utility and distribution system. End-users will own and operate their own heat pump systems and purchase thermal energy from the central plant. The purchase price for the thermal energy will be determined by the cost to produce and deliver the energy plus maintenance costs, personnel costs, administrative costs and debt service.

It is anticipated that the thermal utility would provide distribution and metering equipment to the property line of the customer. Piping and equipment installed on the user's property would be paid for and become the property of that user.

Debt service for the central plant was based on a 7 percent per year interest rate with payments over a 25 year period. This rate appeared to be to be approximately mid-range for rates that could be expected for Conyers.

The following cost analysis is broken into three parts: first costs, operating costs, life cycle costs. A summary of results is presented in the latter part of the section.

FIRST COSTS

First costs are categorized as costs incurred by the central utility or

costs incurred by the end-users. In Conyers' case, it is anticipated that the central utility's costs will be those associated with the central plant, distribution system and metering. End-users will be responsible for the retrofit or initial installation of their own HVAC system and for piping from their heat pump unit to the meter connection.

Central plant first costs were based on the equipment sizes and types presented in the system design section. A detailed cost breakdown is presented in Appendix C. A summary of these costs follows:

Central Plant	$ 635,870
Distribution System	361,520
Metering Equipment (100 - 1" meters, including installation)	35,000
Total	$1,032,390

End-user system costs were computed on the basis of 175 ft. of 1" pipe per end-user to connect to the distribution system and $600 per installed ton of air conditioning capacity for retrofit situations. The $600 per ton is for installation of hydronic heat pump units only, with connection to existing duct systems, and is assumed to include any salvage value of equipment removed from the end-user buildings. New installations were assumed to involve no differential cost over the standard HVAC system. For Conyers the end-user cost is approximately:

300 tons installed x $600		
Total cost of retrofit	=	$180,000
Piping 350 users x 175 ft.		
x $2.50 (installed)	=	153,125
Total retrofit cost	=	$333,125

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The retrofit costs are based on the individual prices each end-user would be required to pay if the work were contracted on an individual basis.

OPERATING COSTS

Operating costs were based on the results of the performance analysis presented in section 8.3.3. The cost of electric power for the central plant was computed from Georgia Power Company's PL-1 rate schedule. Power for end-users was assumed to cost 5.00¢/KWH. The cost of thermal energy delivered to end-users by the HP-WHR system will be discussed in a subsequent section. From Table 8-7 of the performance analysis, the annualized cost for power to the central plant will be approximately $38,015.

The end-users on the HP-WHR system will use approximately 2,311,124 KWH per year. This electricity will cost approximately $115,556 under individual billings.

Maintenance costs for the HP-WHR central station are estimated at 5 percent per year of the installed cost of the station and 0.5 percent per year for the distribution system. These maintenance costs are:

Item	Annual Maintenance Cost
Central Station Equipment	$31,793
Distribution System & Meters	1,982
Total	$33,775

End-user maintenance costs were assumed to be $20 higher per installed ton of air conditioning capacity than would comparable system maintenance on conventional systems. This figure is:

$$782 \quad x \quad 20 \quad = \quad \$15,640 \text{ per year}$$

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HP-WHR SYSTEM 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 the end-users will be 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 Cornelia system using the stated electric energy costs, an administrative cost of 10 percent of total owning and operating costs, and debt service based on an interest rate of 7 percent per year over 25 years. It is recognized that the seven percent interest rate is somewhat arbitrary and will vary with the type of financing arrangements actually implemented; however, it was selected as a compromise between typical 1977 yields for Municipal Class A bonds and Utility bonds issued by private companies.

The total annual cost to the end-users for thermal service was computed as follows:

CONYERS - HP-WHR SYSTEM
ANNUAL THERMAL UTILITY CHARGES
(COMPONENT COSTS IN 1979 DOLLARS)

Energy Cost	$ 38,015
Maintenance on Central Plant and Distribution System	33,775

(UTILITY CHARGES - cont'd)

ANNUAL THERMAL UTILITY CHARGES
(cont'd)

Debt Service on Total System Cost (7%, 25 years) ($1,032,390 x .0858)	88,580
Subtotal	$160,370
10% Administrative fee (Including Insurance & Contingencies)	16,035
Total	$176,405

Based on the above total, the necessary cash flow for the system will be as projected in Table 8-9.

LIFE CYCLE COSTS

A life cycle cost analysis was prepared for the Cornelia HP-WHR system and the existing conventional systems. 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.

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

$$PV_i = FC_i + \sum_{y=1}^{25} [f(j,y) \times (ADC_i + AMC_i)]$$

TABLE 8-9
CONYERS HP-WHR SYSTEM
THERMAL UTILITY CHARGES
CASH FLOW

```
             THE FOLLOWING ASSUMPTIONS WERE USED
ANALYSIS PERIOD OF 25 YEARS INTEREST RATE OF    .0 PERCENT
SAL/COST ESCALATION=.000 MAINT. ESCALATION= .080
AVERAGE UTILITY ESCALATION:E=.110 INCOME ESCALATION=.000
```

YR	DEBT S./ OR COST	ANNUAL:COSTS MAIN,ETC	UTILITIES	ADMIN.	D FACT	COST.
0	88580.	0.	0.	0.	*1.000	-88580.
1	88580.	36477.	42197.	16725.	*1.000	183979.
2	88580.	39395.	46838.	17481.	*1.000	192295.
3	88580.	42547.	51990.	18312.	*1.000	201429.
4	88580.	45951.	57709.	19224.	*1.000	211464.
5	88580.	49627.	64057.	20226.	*1.000	222490.
6	88580.	53597.	71104.	21328.	*1.000	234609.
7	88580.	57884.	78925.	22539.	*1.000	247929.
8	88580.	62515.	87607.	23870.	*1.000	262572.
9	88580.	67516.	97244.	25334.	*1.000	278674.
10	88580.	72918.	107941.	26944.	*1.000	296382.
11	88580.	78751.	119814.	28715.	*1.000	315860.
12	88580.	85051.	132994.	30662.	*1.000	337287.
13	88580.	91855.	147623.	32806.	*1.000	360864.
14	88580.	99204.	163861.	35165.	*1.000	386810.
15	88580.	107140.	181886.	37761.	*1.000	415367.
16	88580.	115711.	201894.	40618.	*1.000	446803.
17	88580.	124968.	224102.	43765.	*1.000	481415.
18	88580.	134966.	248753.	47230.	*1.000	519529.
19	88580.	145763.	276116.	51046.	*1.000	561505.
20	88580.	157424.	306489.	55249.	*1.000	607742.
21	88580.	170018.	340203.	59000.	*1.000	658680.
22	88580.	183619.	377625.	64982.	*1.000	714806.
23	88580.	198309.	419164.	70605.	*1.000	776657.
24	88580.	214173.	465272.	76802.	*1.000	844827.
25	88580.	231307.	516451.	83634.	*1.000	919973.

Where:

PV_i = Present value of the ith alternative

FC_i = First cost of the ith alternative

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

ADC_i = Annual operating costs of the ith alternative

AMC_i = Annual maintenance costs of the ith alternative

The results of the life cycle cost analysis are presented in Table 8-10 for the conventional systems as the base case and Table 8-11 for the HP-WHR system 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.

SUMMARY OF EXPECTED ECONOMICS

The results of the life cycle economic analysis indicate that the HP-WHR system as proposed for the Conyers community is more costly to the end-users than the conventional space conditioning system alternative. On the basis of the results presented in Tables 8-10 and 11, implementation of the HP-WHR system would result in a net additional cost to the end-users of $2,274,520. Several factors detract from the comparative system economics. The arduous retrofit costs for portions of the proposed service area, even though borne by the individual system subscribers, increase the capital investment necessary to implement the system and thus penalize the economic attractiveness from the end-users' point of view. System performance during the cooling mode is comparable, on a C.O.P. basis, to the standard electric air conditioning systems typically in residential service in the proposed service area.

TABLE 8-10
CONYERS CONVENTIONAL SYSTEMS (BASE CASE)
LIFE CYCLE COST ANALYSIS

YR	DEBT SERVICE OR COST	BASE MAINTENANCE	UTILITY COSTS N.GAS	ELECT.	TOTAL COSTS	DISCOUNT FACTOR	DISCOUNTED COSTS
0	0.	0.	0.	0.	0.	1.000	0.
1980 = 1	0.	0.	71,179.	71,595.	142,774.	.090	129,782.
2	0.	0.	80,432.	79,470.	159,902.	.826	132,079.
3	0.	0.	90,888.	88,212.	179,100.	.751	134,504.
4	0.	0.	102,704.	97,916.	200,620.	.683	137,023.
5	0.	0.	116,055.	108,686.	224,741.	.621	139,564.
6	0.	0.	131,142.	120,642.	251,784.	.564	142,006.
7	0.	0.	148,191.	133,912.	282,103.	.513	144,719.
8	0.	0.	167,455.	148,643.	316,098.	.467	147,617.
9	0.	0.	189,225.	164,993.	354,218.	.424	150,188.
10	0.	0.	213,824.	183,143.	396,967.	.386	153,229.
11	0.	0.	241,621.	203,288.	444,909.	.350	155,718.
12	0.	0.	273,032.	225,650.	498,682.	.319	159,080.
13	0.	0.	308,526.	250,472.	558,998.	.290	162,109.
14	0.	0.	348,634.	278,023.	626,657.	.263	164,811.
15	0.	0.	393,956.	308,606.	702,562.	.239	167,912.
16	0.	0.	445,171.	342,553.	787,724.	.218	171,724.
17	0.	0.	503,043.	380,233.	883,276.	.198	174,889.
18	0.	0.	568,439.	422,059.	990,498.	.180	178,290.
19	0.	0.	642,336.	468,486.	1,110,822.	.164	182,175.
20	0.	0.	725,839.	520,019.	1,245,858.	.149	185,633.
21	0.	0.	820,198.	577,221.	1,397,419.	.135	188,652.
22	0.	0.	926,824.	640,716.	1,567,540.	.123	192,807.
23	0.	0.	1,047,311.	711,194.	1,758,505.	.112	196,953.
24	0.	0.	1,183,462.	789,426.	1,972,888.	.102	201,235.
25	0.	0.	1,337,312.	876,262.	2,213,574.	.092	203,649.
TOTALS			11,076,799.	8,191,420.	19,268,219.		4,096,348.

ASSUMPTIONS:

1) Natural Gas Cost Escalation - 8%
2) Electric Energy Cost Escalation - 11% (3% Over Base Inflation Rate)
3) Discount Rate - 10%

TABLE 8-11
CONYERS HP-WHR SYSTEM
LIFE CYCLE COST ANALYSIS

YR	DEBT SERVICE OR COST	ADDITIONAL MAINTENANCE	UTILITY COSTS ELECT.	UTILITY COSTS THERMAL	TOTAL COSTS	DISCOUNT FACTOR	DISCOUNTED COSTS
0	0.	0.	0.	0.	0.	1.000	0.
1980 = 1	0.	16,891.	128,267.	183,979.	329,137.	.090	299,186.
2	0.	18,242.	142,377.	192,295.	352,914.	.826	291,508.
3	0.	19,702.	158,038.	201,429.	379,169.	.751	284,757.
4	0.	21,278.	175,422.	211,464.	408,164.	.683	278,776.
5	0.	22,980.	194,719.	222,490.	440,189.	.621	273,357.
6	0.	24,819.	216,138.	234,609.	475,566.	.564	268,219.
7	0.	26,804.	239,913.	247,929.	514,646.	.513	264,013.
8	0.	28,949.	266,303.	262,572.	557,824.	.467	260,505.
9	0.	31,264.	295,597.	278,674.	605,535.	.424	256,748.
10	0.	33,766.	328,112.	296,382.	658,260.	.386	254,088.
11	0.	36,467.	364,204.	315,860.	716,531.	.350	250,787.
12	0.	39,384.	404,267.	337,287.	780,938.	.319	249,119.
13	0.	42,535.	448,736.	360,864.	852,135.	.290	247,119.
14	0.	45,938.	498,097.	386,810.	930,845.	.263	244,812.
15	0.	49,613.	552,888.	415,367.	1,017,868.	.239	243,271.
16	0.	53,582.	613,706.	446,803.	1,114,091.	.218	242,873.
17	0.	57,868.	681,213.	481,415.	1,220,496.	.198	241,658.
18	0.	62,498.	756,147.	519,529.	1,338,174.	.180	240,871.
19	0.	67,498.	839,323.	561,505.	1,468,326.	.164	240,805.
20	0.	72,897.	931,648.	607,742.	1,612,287.	.149	240,232.
21	0.	78,729.	1,034,130.	658,680.	1,771,539.	.135	239,159.
22	0.	85,027.	1,147,884.	714,806.	1,947,717.	.123	239,569.
23	0.	91,830.	1,147,884.	776,647.	2,142,638.	.112	239,975.
24	0.	99,176.	1,414,308.	844,827.	2,358,311.	.102	240,549.
25	0.	107,110.	1,569,882.	919,973.	2,596,965.	.092	238,922.
TOTALS		1,234,847.	14,675,470.	10,679,948.	26,590,265.		6,370,868.

ASSUMPTIONS:

1) Maintenance and Replacement Cost Escalation - 8%
2) Electric Energy Cost Escalation - 11% (3% Over Base Inflation Rate)
3) Thermal Utility Charge Escalation - 11% (From Table 8-9)
4) Discount Rate - 10%

Thus no performance or economic benefit in favor of the HP-WHR system is presented. Finally, the current low energy costs for natural gas, the heating resource of choice in the proposed service area, make the gas-fired furnace option the most cost effective heating system alternative. This last factor, combined with the fact that the proposed system is essentially cooling dominated, is perhaps the most critical detriment to system economics. It is anticipated that, if the system were located at heating dominated sites-of-application, or if (and when) natural gas prices escalate, the system will approach the threshold of economic viability.

8.3.5 <u>PLAN FOR HP-WHR SYSTEM PHASING - CONYERS, GEORGIA</u>

An HP-ICES will operate in its community as a form of public utility which will provide a needed and potentially widespread (in relation to the overall community) service. Serving in this capacity, the system is subject to the need for long-range planning in order to provide its service in a manner most beneficial, service-wise and cost-wise, to the end-user. The primary concern is that the system be designed and constructed to fulfill the needs of the existing viable service areas. However, consideration must also be given to expanding the system in the most efficient manner and at the most advantageous time to accommodate future growth within the community. Chaper 6.0 - PROJECTED GROWTH discusses methods of projecting growth within a specific community and addresses construction phasing of central plant facilities and the distribution system for ranges of growth rate situations. The following paragraphs apply the general approaches outlined in the earlier chapter to the Conyers demonstration community situation.

The initial approach of the Conyers - based HP-WHR system will be to provide thermal utility services to a compact area of residential subdivisions just outside the corporate city limits on Stanton Road. The first candidate subdivision is located adjacent to the Almand Branch sewage treatment facility in the vicinity of the intersection of Stockbridge Highway 138 and Stanton Road. This subdivision is currently under development with several structures already completed and an ultimate build-out potential of 40 to 50 houses. A second subdivision approximately one mile down Stanton Road from the first is at a comparable degree of development with 60 to 75 houses completed and a build-out potential of 100 to 150 houses. Additionally, there is a parcel of land directly between the two subdivisions and roughly equivalent in land area to the second which is considered a likely candidate for development if the Conyers community continues its trend of rapid growth. Ultimate build-out

potential of this parcel of land, if developed, is approximately 150 houses.

Accounting for the total build-out potential of each of the three areas, this

compact residential community would show a viable service area of approximately

120 houses within a one and one half mile corridor around the sewage treat-

ment plant. Since the current building inventory is approximately 30% of

ultimate potential, and since it is estimated that the area, at its present

rate of growth, will reach its build-out potential within five to ten years,

this community will offer opportunities for phased system construction and

a combination of residential retrofits and new home system installation.

CENTRAL PLANT PHASING   -   In general, it is advisable to plan and construct

central plant facilities for a high growth area (such as this residential

community) in several stages over a given planning period (usually 20 years).

This procedure avoids committing large sums of money for excess capacity at

the central plant which may not be needed until much later in the planning

period. However, the Conyers demonstration community is an interesting mix

of aspects of high growth and low growth areas. The growth potential of the

subject area can be shown to be approximately 200% of the existing number of

structures in the area over a 20 year planning period. The limited land

area in the community, however, puts an effective ceiling on the ultimate

expansion in the immediate area. The combination of the above two factors

results in the estimate that the area may reach its potential in five to ten

years. Considering this estimate and considering the fact that central

station capacity may be limited somewhat by the relatively low capacity of

the sewage treatment plant, it is deemed advisable to plan for installation

of the bulk of central station capacity during the initial construction.

A minor plant expansion would then be planned to finalize central station

capacity in approximately five years. The pipe lines, sumps, and pumps which

serve on the sewer tic-in subsystem will be constructed initially to handle

100 to 110% of the ultimate wastewater flow which could be accommodated at the sewage treatment facility. The major building structure will be constructed to accommodate all equipment which would be necessary to serve the ultimate build-out potential of the system. Since the total quantity of thermal energy available from the wastewater could reasonably be handled by one central station heat pump, it is likely that full capacity will be installed. Installation of the auxiliary equipment (circulating pumps, distribution pumps, thermal storage, and cooling tower) will be based on estimated system requirements during the first five year period. At the time when additional capacity is required, a minor plant expansion would involve the installation of an additional circulating pump, distribution pump, cooling tower cell, and if required, thermal storage. All equipment installation could be handled with little or no major modification to the in-plant piping system or structure. Costs for system expansion within the central plant should thus be minimized.

DISTRIBUTION SYSTEM PHASING   -   There are several characteristics of the design and construction of high-volume water distribution (and return) lines which will influence the manner in which such systems will be constructed and expanded. First, water and sewer and, in this case, thermal utility lines have a useful life of approximately 50 years. Secondly, a major portion of the cost of constructing such lines is not in material costs but in manpower and equipment expenditures for the construction process. Hence, the most economical way to provide additional capacity generally is to construct a larger line initially rather than a parallel line at a later date.

The first phase of distribution system construction will involve installation of a distribution trunk line into the subdivision adjacent to the sewage treatment facility site. Due to the layout of the subdivision and to housing construction patterns, construction of the distribution system within this area can also be phased. The main trunk line would first be constructed along Morris Drive and Cindy Drive, with tributary

lines along Rodgers Drive and into the as yet unnamed northern section of the subdivision to be installed at later dates. In all cases, line sizing and construction would be based on estimated ultimate requirements for the area so that the expense and inconvenience of constructing parallel lines at different stages of development could be avoided.

The second phase of distribution system construction would involve installation of a separate distribution trunk line into the second subdivision now under development. The main trunk line of this leg would run in a generally southerly direction from the plant, over to Stanton Road, and into the subdivision along Almand Branch Road. Several structures could be served directly from this main trunk line, and as the degree of development in the area warrants, tributary lines could be constructed in a manner similar to the first pahse construction. It should be noted that construction of distribution lines for service into this area would quite likely coincide with the central station expansion discussed in the preceeding section.

The third phase of construction, involving the development of the vacant parcel of land between the two subdivisions, is very tentative. Discussion of this phase is only to identify the potential of the area if current housing and growth trends continue. There are two options for providing service into this area. Depending upon the layout and development pattern of this zone, a main distribution trunk could be constructed off of the bulk transmission trunk line located along Stanton Road, or a separate trunk line could be constructed from the central station for service to the area.

Due to the development patterns in the three relatively segregated zones in the community adjacent to the Conyers sewage treatment facility, it is apparent that there is ample opportunity to demonstrate the options for phased construction of the central plant, distribution system, and end-user systems in an HP-WHR system.

8.4 INSTITUTIONAL CONSIDERATIONS

As with any undertaking that proposes to provide a public service in some form to the members of a community, the implementation of a Heat Pump Centered-Integrated Community Energy System will encounter significant institutional barriers. In order to impose a priority for consideration, however, these barriers can be classified into three fairly definitive categories. The first category involves the institutional considerations typical in the implementation of any large scale venture such as ownership, operation, and financing alternatives. Categories two and three involve consideration of the effects upon implementation of the existing state and local laws, respectively. The following sections of this chapter discuss these institutional considerations with regard to their effect upon the implementation of the HP-WHR concept.

8.4.1 OWNERSHIP AND OPERATION OF A HEAT PUMP CENTERED-INTEGRATED COMMUNITY ENERGY SYSTEM (HP-ICES)

There would be several alternative approaches to the ownership and operation of a Heat Pump Centered-Integrated Community Energy System. Such a system would be very similar to a public utility that provides water supply, wastewater collection and treatment, electrical power, or natural gas. Consequently, the alternative methods for ownership and operation would be similar to the types of institutions which provide the public utilities just listed. The three basic types of institutions that could serve as the owner and operator of a Heat Pump Centered-Integrated Community Energy System are as follows:

1. A privately-owned public utility company

2. A municipal-owned public utility

3. A special purpose authority

The privately-owned public utility company would be similar to Georgia Power Company or the Atlanta Gas Light Company. Such companies are generally profit making corporations that are owned by the individual stockholders who have invested in the corporation.

An example of a municipal-owned public utility would be a city or county water system which is operated as a part of the city or county government. Some municipal governments also operate electrical distribution systems and natural gas distribution systems within their jurisdictions as a part of the utilities provided by the local government.

The third type of institution would be a special purpose authority created solely for the purpose of owning and operating a Heat Pump Centered-Integrated Community Energy System. In Georgia such authorities can be created by an act of the State Legislature. Such authorities are non-profit organizations created for a single purpose and having jurisdiction over a limited and clearly defined area. The most common special-purpose authorities in Georgia are water authorities whose jurisdiction usually covers all or part of a county. Speical purpose authorities can build, own and operate the specified facilities within their jurisdiction. Such authorities can enter into the necessary financial arrangements needed for constructing and operating their facilities. However, the revenues for the authority are limited strictly to those revenues that can be generated from the operation of the facilities which the authority has constructed. In most cases, these revenues come from the service charges billed to individual residential and commercial customers of the system. The main difference between a special-purpose authority and a municipal-owned public utility is found in the available means of financing. A special-purpose authority is generally a separate entity from

the municipal or county government, both financially and organizationally. The authority is generally independent and financially self-supporting from the municipal or county government. However, a municipal-owned public utility can be operated as a part of a municipal or county government and may not be financially self-supporting. This means that a municipal-owned public utility may be financed in part by general obligation bonds, ad valorem taxes, or other sources of revenue which the municipal or county government may have. The revenues that are derived from the operation of a municipal-owned public utility do not have to provide the total financing for the operation and maintenance of the utility.

The special-purpose authority is further defined in the Georgia Code as follows:

> "The 'authority system of finance' is a system whereby the
> state or a municipality can carry out a function of a public
> or charitable nature through an agency created by the
> legislature and designated by the creating statute as an
> 'authority.' Such an authority has the power to borrow
> funds for its establishment and early operation through the
> issuance of revenue bonds, which, for their retirement and
> for the payment of interest thereon, may only pledge funds
> arising from the operation of the authorized project. In
> no event may such bonds pledge the credit of the municipality,
> create any lien on public property, nor give the bondholder
> any right to compel the exercising of the taxing power."

8.4.2 FINANCING ALTERNATIVES FOR AN HP-ICES

The alternative means for financing a Heat Pump Centered-Integrated Communtiy Energy System can be best described by the different categories of ownership which have been previously presented.

A privately-owned public utility company is a profit making corporation and generally can use the same methods of financing that are available to other corporations and businesses. Short-term financing can generally be obtained from banks and other lending institutions and long-term financing can be obtained by issuing stocks, bonds, or warrants.

A municipal-owned public utility can usually obtain short-term financing from banks or other lending institutions. Long-term financing is generally obtained by selling revenue bonds or general obligation bonds to private investors, banks, or other financial institutions. Municipal-owned public utilities in recent years have been able to obtain significant money for improvements or expansions through Federal or State grant or loan programs. The Federal Environmental Protection Agency has a significant grant program to assist local governments in the construction of interceptor sewers and wastewater treatment facilities. The Farmers Home Administration within the Department of Agriculture has both a loan and grant program to help small communities in the construction of water and sewer systems. The Department of Housing and Urban Development has provided significant sums of grant monies to urban areas for the construction of water and sewer facilities, in addition to other public facilities that would not be categorized as utilities. The Economic Development Administration has also provided Federal grant funds to construct or expand utilities to serve industrial growth areas.

A special-purpose authority could obtain short-term financing from banks

or other lending institutions if sufficient collateral could be provided or adequate assurance that the loan could repaid. The principal means for long-term financing of a special-purpose authority would be through the sale of revenue bonds which would be repaid from the annual revenues of the authority. Therefore, the strength and potential for growth would determine the rating for the revenue bonds and the difficulty that might be encountered in selling the revenue bonds. Special-purpose authorities are generally eligible for Federal and State grant loan funds as previously described for municipal-owned public utilities.

8.4.3 COMPARISON OF ALTERNATIVE METHODS OF OWNERSHIP, OPERATION AND FINANCING OF

AN HP-ICES

The two previous sections have discussed the alternative methods available for the ownership, operation and financing of a Heat Pump Centered-Integrated Community Energy System. This section will provide a comparative discussion of the alternative means for ownership, operation and financing of a Heat Pump Centered-Integrated Community Energy System.

The alternative of a privately-owned public utility company has two significant drawbacks. The first drawback is that such companies would not generally desire to take on the responsibility for such a project until its feasibility and profit potential had been fully established. An exception to this might be that if massive Federal grant funds were involved in the initial demonstration and there was little financial risk to the utility company, then the company might be willing to be involved. The preceding statements assume that the public utility company concerned was already in existence and providing other services prior to being approached about involve-ment with a Heat Pump Centered-Integrated Community Energy System. At this

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stage in the development of the HP-ICES concept, it would be virtually impossi-

ble to consider creating a new public utility company solely for the purpose

of owning and operating the HP-ICES. Because of the uncertainties involved

with the initial capital cost, operating cost, and profit potential, it would

be highly unlikely that such a company could successfully sell the necessary

stock that would be required for initial financing and operation.

The alternative of a municipal-owned public utility offers several

advantages and disadvantages. One of the main advantages, as previously

mentioned, is that a municipal-owned public utility has several alternative

means of financing available to it. Long-term financing through the sale of

municipal bonds offers an attractive means of financing construction since

the interest rates on municipal bonds are often lower due to the tax-free

status which they offer investors. Since a municipal-owned public utility

does not have to make a profit, the utility rates which must be charged to

customers are generally lower than might be charged by privately-owned utility

companies. There are several disadvantages which are posed by municipal-owned

public utilites. One disadvantage is that municipal-owned public utilities

usually try to provide service to the entire community or municipality.

However, with the HP-ICES concept, it may be that only a portion of the

community can be served. Another disadvantage to a municipal-owned public

utility is that in recent years many small to medium sized towns have come to

depend heavily on Federal grant and loan funds to improve and expand their

public utilities (primarily water and sewer). Because of this trend, local

officials will often postpone needed construction or improvements until

massive Federal grant funds can be obtained which will pay, in some cases, 100

percent of the total cost involved. Because of this trend and because of the

uncertainties involved in the initial construction of a demonstration project for an HP-ICES it is highly unlikely that a municipality would undertake such a system without significant Federal grant funds being involved in both the initial construction and operation of the system.

The last alternative is the creation of a special-purpose authority. An advantage to the special-purpose authority is that the area covered by the authority could be limited solely to the area which would be served by the Heat Pump Centered-Integrated Community Energy System. Therefore, local officials could not be criticized for operating a municipal utility which served only a limited portion of the total community. The main disadvantage to a special-purpose authority are the inherent limitations in financing construction with local funds. Special-purpose authorities are limited almost totally to revenue bonds for long-term financing. It is also difficult to issue revenue bonds for new authorities who do not have established customers and a past history of revenue collections. For a new type of utility such as the HP-ICES which has not been fully proven and demonstrated, it would be even more difficult to issue revenue bonds to finance the construction of such a system. Consequently, a special-purpose authority would also require significant Federal grant funds to help pay for the initial construction and operation of the system.

Of the three alternative methods that have been considered for ownership, operation and financing of the HP-ICES, only two appear to be viable alternatives. These are the municipal-owned public utility and the special-purpose authority. The initial construction and operation of such a system under either of these alternatives would definitely require significant Federal grant funds and future availability of Federal grant or low interest loan funds.

STATE OF GEORGIA-REVIEW OF LEGAL AND REGULATORY FACTORS OF CONCERN IN THE

DEVELOPMENT OF AN HP-ICES

In the development and implementation of any new concept, it is important

that existing laws and regulations be reviewed to determine what legal con-

straints may impede or affect actual implementation. In this regard, state and

local laws, regulations, codes and ordinances have been reviewed to determine

what impact they might have on the construction and operation of a Heat Pump

Centered-Integrated Community Energy System. In order for this review to be

applicable to the two application communities, the review has been limited to

laws and regulations of the State of Georgia and to codes and ordinances for

the cities of Cornelia and Conyers, Georgia. Existing statutory law for the

State of Georgia was review in the form of The Code of Georgia, Unannotated,

1978 edition. Statutory laws for the State of Georgia are organized and

presented in this publication by subject matter which include Titles 1 through

114. A review of these titles indicated only three titles which might have

application to a Heat Pump Centered-Integrated Community Energy System. These

titles were "Title 17, Canals, Waters and Navigations," "Title 88, Public

Health," and "Title 92A, Public Safety".

In reviewing Title 17, Canals, Waters and Navigation, one chapter in

particular was found to have a potential effect on the construction of a Heat

Pump Centered-Integrated Community Energy System. This was Chapter 17-5,

"The Georgia Water Quality Control Act". This chapter establishes certain

laws and regulations concerning water quality of the streams, rivers and lakes

within the State of Georgia. The chapter designates the Environmental Protec-

tion Division of the Georgia Department of Natural Resources as the state

agency with the responsibility for implementing and enforcing the water quality

laws and regulations of the state. One of the requirements of this chapter is that municipalities and industries must obtain permits from the state in order to construct and operate wastewater treatment facilities. These permits are also required for expanding or modifying existing wastewater treatment facilities. The permits which are issued by the Environmental Protection Division constitute both a Federal and State permit under the Federal Water Pollution Control Act of 1972 and these permits establish treatment requirements and effluent limitations which must be achieved by the wastewater treatment facility. Consequently, the planning and design of a Heat Pump Centered-Integrated Community Energy System must evaluate what effect the operation of such a system will have on the municipal wastewater treatment plant and its ability to operate within the limits established by its permit.

Another chapter which may have indirect implication is Chapter 17-13, The Georgia Safe Drinking Water Act of 1977. This chapter creates the necessary laws and regulations needed for the establishment of a safe drinking water program for the State of Georgia. One aspect of such a program is to prevent improper connections being made between potable water systems and non-potable water sources or wastewater sources. One of the potential dangers of a Heat Pump Centered-Integrated Community Energy System is the possibility that someone will mistakenly make a connection to the HP-ICES distribution system, thinking that it is a potable water distribution main. For this reason, it will be important that the pipe network in the HP-ICES be adequately marked to avoid improper cross connections with potable water systems.

Title 88, Public Health, assigns specific responsibilities to the Georgia Department of Human Resources and to County Boards of Health for the protection of the public health. These responsibilities include protection

to the water quality public water supplies. This responsibility would include the responsibility to prevent and minimize the potential for cross connections between potable and non-potable water supplies.

8.4.5 <u>CITY OF CONYERS GEORGIA - REVIEW OF LOCAL ORDINANCES AND CODES</u>

The local codes and ordinances of the City of Conyers and Rockdale County were reviewed to determine what impacts or restrictions they might impose on the development of a. Heat Pump Centered-Integrated Community Energy System. This review indicated that there were several local building codes which would require an in depth review in conjunction with the design and construction of a Heat Pump Centered-Integrated Community Energy System. The local building codes which have been adopted for the Conyers-Rockdale County area are as follows:

1. Southern Standard Building Code - 1976 Edition

2. Southern Standard Plumbing Code - 1975 Edition

3. Southern Standard Mechanical Code - 1976 Edition With 1978 Amendments

4. National Electrical Code - 1978 Edition.

The first three (3) are published by the Southern Building Code Congress International, Inc., located in Birmingham, Alabama. These building codes are widely used by local governments both in Georgia and other parts of the United States. These codes establish minimum construction standards which must be followed in the construction of residential, commercial and industrial building. In reviewing the codes noted above, there were not requirements or specifications found that would severly restrict the development of an HP-ICES. There were, however, several aspects of the codes which would need to be considered in the planning and design of the system.

Some of the requirements of the codes are as follows:

1. Pipes used for transporting non-potable water must be clearly identified to avoid accidential tie-ins or connections to the potable water system of a home or commercial establishment.

2. The plumbing code establishes other restrictions that are needed to prevent cross-connections and backflow from the non-potable HP-ICES piping to the potable water system.

3. The plumbing code also establishes minimum criteria and specifications for piping materials and related appurtenances to be used within residential and commercial buildings.

4. The plumbing code may also include additional specifications or requirements for the HP-ICES piping as a result of the higher pressures which may be involved with this piping as compared with normal water distribution systems.

5. At the present time there is only limited reference to heat pump type heating systems within the various codes. However, as heat pumps become more widely used, future revisions of the codes will probably contain more specific provisions and criteria related to heat pump specifications and installation requirements.

There were no other local ordinances or codes that would affect or restrict the construction of a Heat Pump Centered-Integrated Community Energy System within the Conyers, Georgia-Rockdale County area.

9.0 SITE-SPECIFIC FEASIBILITY STUDY FOR CORNELIA, GEORGIA

9.1 COMMUNITY DESCRIPTION - CORNELIA, GEORGIA

The City of Cornelia is located in the southeastern corner of Habersham County, which lies in the Piedmont Plateau Region of northeast Georgia in the foothills of the Blue Ridge Mountains. Cornelia lies along U. S. Highway 441, a major north-south highway, which carries a high volume of tourist traffic. Cornelia is located 80 miles northeast of Atlanta, 27 miles northeast of Gainesville, 46 miles north of Athens and 84 miles west of Greenville, South Carolina.

The topography of Cornelia ranges from fairly broad, level land along streams and branches to gently rolling hills in the higher elevations. Elevation ranges from approximately 1,400 feet to 1,680 feet above sea level.

9.1.1 POPULATION GROWTH

According to published population data, Cornelia was the second fastest growing city in Habersham County for the decade 1950 - 1960, the slowest growing city in the county for the decade 1960 - 1970, and grew at approximately the same rate as the other six towns in the county for the period 1970 - 1977.

Population projections for Cornelia through 1995 are as shown in Table 9-1. These projections are based on the assumption that growth in Cornelia will represent a fixed percentage of the estimated population increase of Habersham County as included in the Georgia Mountains Future Land Use Plan, March 1973.

TABLE 9-1
POPULATION AND PROJECTIONS FOR CORNELIA (1970-1995)

Years	Population	Projected Population	Percent of Increase
1970	3,014		
1977		3,404	12.9
1980		3,571	4.9
1985		4,017	12.4
1990		4,463	11.1
1995		5,578	24.9

Source: Cornelia Community Development Concept Plan, Georgia Mountains
Planning and Development Commission, 1977.[57]

9.1.2 ECONOMIC CHARACTERISTICS

Accelerated growth is greatly changing the economy of Cornelia and Haber-
sham County. The region is in transition from an agricultural-oriented base
to a rapidly-growing manufacturing base. Cornelia has in the past received
most of its economic benefits from agricultural products. But in the past two
decades the city and county have received benefits from increasing industrial
development and commercial activity. Cornelia presently has a somewhat
balanced economy with respect to industrial employment, agricultural production
and retail sales volume.

Due to Cornelia's size, data conerning Cornelia's economy is not avail-
able except as a part of Habersham County. The economy of Habersham County
is diversified with the largest part of the labor force employed in manufactur-
ing, trade, finance, insurance, real estate, and services. Manufacturing, which
accounts for the largest number of jobs in the county, comprises 43.7% of the

county's employment. Finance, insurance, real estate, and services comprise
the second largest category of employment in the county with 26.1%. Jobs
in wholesale and retail trade comprise another 16.3% of the county work force.
The balance of employemnt is in the areas of construction with 8.1%, agriculture
with 5.2%, or transportation and public utilities employment with 3.5% of the
work force.

Projections of economic growth in Georgia from 1975 to 1985 have been
made by the Georgia Economic Forecasting Project under the direction of Dr.
John Legler at the University of Georgia. Using these projections as an
indication of Cornelia's economic growth for the next ten years, the town can
expect per capita income to increase faster than consumer prices, total
employment to increase approximately two and one-half percent annually, a steady
increase in manufacturing employment and earnings, and a concomitant increase
in the dollar volume of retail sales.

9.1.3 LAND USE

The City of Cornelia encompasses 1,158 acres within its corporate limits.
Residential acreage accounts for the bulk of the land usage with 62.7% or 726
acres. Agricultural usage and open space comprise the second largest land
usage category with 164 acres or 14.2%. Commercial land usage is a comparable
131 acres or 11.3%. Public and semi-public land usage and industrial usage
are the smallest categories with 7.8% (90 acres) and 4% (47 acres), respectively.
The average population density is 2.5 persons per acre.

The land use pattern in Cornelia is similar in most respects to that found
in cities of the same size. The core, the oldest section of the city, is
exclusively in retail stores, business establishments, offices, banks and
similar uses. Immediately surrounding the core area and along both sides of

the railroad tracks are located commercial and wholesale establishments includ hardware, gasoline stations, restaurants, farm sales and services and automobile dealers. Slightly to the southeast of the core, most of the local government functions are concentrated.

Away from the core area of Cornelia are found other commercial uses. Such businesses are mainly neighborhood oriented such as grocery stores, auto repair shops, service stations, and restaurants. Most of the highway oriented businesses which cater to automobile services are found along U. S. Highway 441, 23, and 123.

The majority of industrial development in Cornelia has occurred in the southwest section of the city, while residential development has occurred throughout with concentrations occurring to the northeast and southeast.

The City of Cornelia, its attending municipal wastewater treatment facility, and the city's contiguous areas to be considered for service with the proposed HP-WHR system are illustrated in Figure 9-1.

9.1.4 CLIMATE

The climate of Cornelia is relatively mild with mild summers and moderate winters. The average annual precipitation is approximately 50 inches. This precipitation is nearly always in the form of rain with snow falling only occasionally in winter. Temperature averages for summer months range from 70 to 80 degrees while averages for winter months range from 40 to 45 degrees Fahrenheit. Periodic blasts of wintry Arctic air are felt by the City with the average number of freezing days being 45 to 60. The frost-free season is long and usually averages around 215 days. Frosts can occur in late October and the danger of frosts normally passes by mid or late April.

PROPOSED
SERVICE AREAS

NORTH

0   1000   2000

FEET
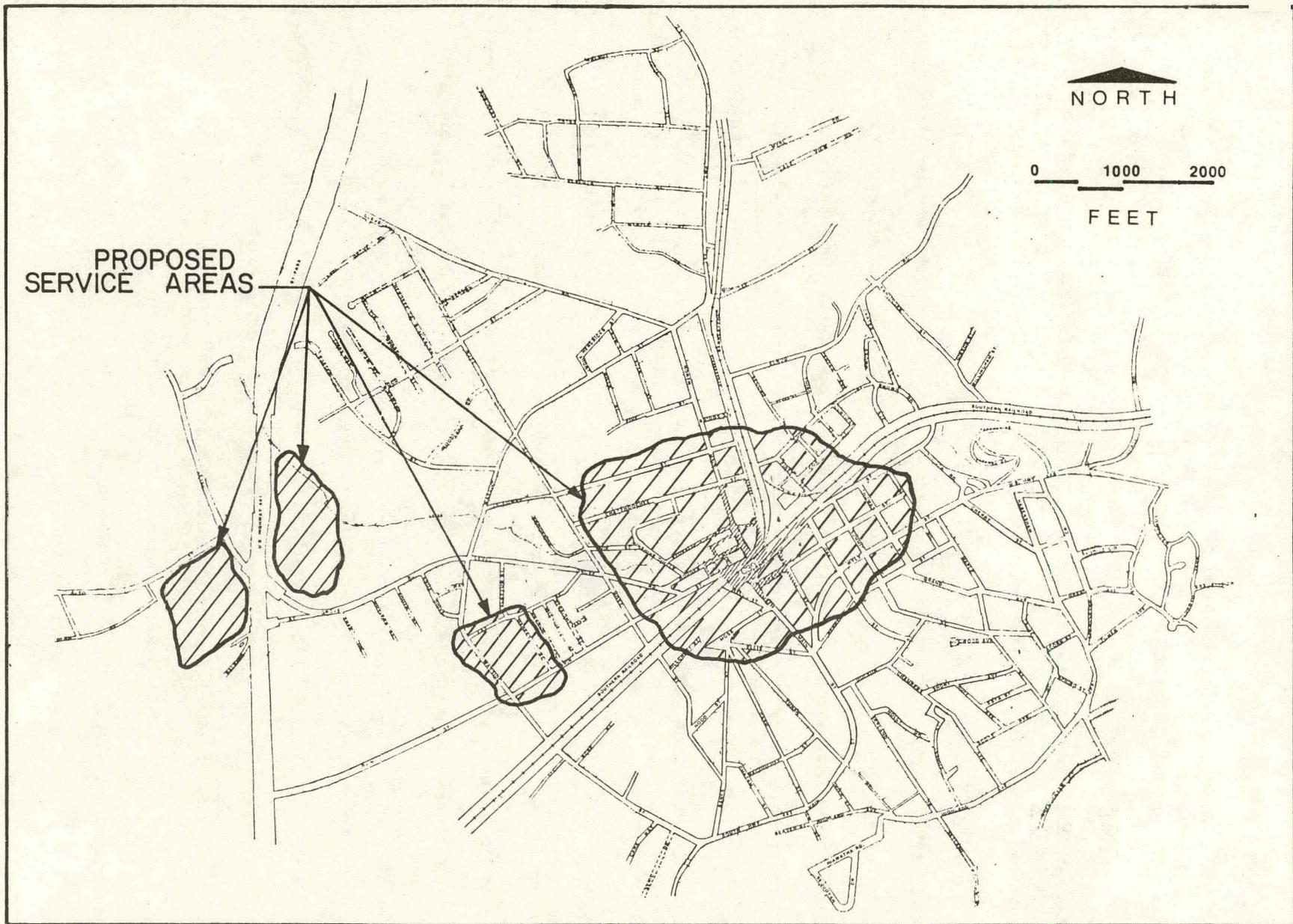
CITY OF CORNELIA
HP-WHR SYSTEM SERVICE AREAS
FIGURE 9-1

9.1.5 PROPOSED SERVICE AREA

The community area proposed for service by the Cornelia Heat Pump - Wastewater Heat Recovery System is restricted to a fairly close group of commercial establishments, an industrial plant and the central business district. This somewhat limited service group was chosen after consideration of the points presented in the following paragraphs.

The initial consideration when determining the Cornelia service area was proximity of thermal loads to the energy source. As is the case with many waste treatment facilities, Cornelia's is located remote from the town core and is significantly removed from most residential groupings. Residential growth has tended to move away from the treatment plant area due to more favorable access to Gainesville, Georgia (the nearest town of significant size) and because of the unpleasant side effects of the treatment plant itself - i.e. odors from incomplete waste treatment. For this reason, the newer, larger residences offering the greatest potential for using the thermal media provided by the system are furthest from the thermal source. Those residences nearest the treatment plant are 30 to 40 years old, small and generally in need of weatherization. These older houses are typically heated with natural gas and are not air conditioned.

Another significant factor influencing the decision not to deal specifically with residential service is a lack of predictable residential growth. Community development studies for Cornelia predict small population increases over the next 20 years with no trends which would indicate housing growth in a particular area convenient to serve from the HP-WHR system. Accordingly, it was determined that Cornelia's residential sector did not offer an attractive market for thermal energy services from a central wastewater heat recovery plant.

The commercial sector of Cornelia is primarily "Highway-oriented" or located in the central business district in the heart of town. This group would appear to offer the highest potential for thermal energy use in the shopping center area located west of town along the Rt. 441 Cornelia Bypass and in the central business district. Factors favoring the use of thermal energy through heat pump retrofit are: heavy use of space air conditioning, trend toward remodeling some of the central business district shops, and a significant number of commercial shops using electric resistance heating.

The only significant industrial customer present within the service area is the Fieldale Poultry Processing Plant. This plant is also the supplier of most of the thermal effluent contained in the wastewater stream. It is envisioned that a nearly continuous supply of $70 - 80^{\circ}F$ water could be utilized by Fieldale for their ice-making and process refrigeration requirements, thus eliminating their need for evaporative coolers. This arrangement would also benefit the system performance since the additional heat input during winter operation could be used by other buildings.

9.1.6 BUILDING CONSTRUCTION

As shown in Figure 9-2, three zones, categorized by locality and building type, were identified within the proposed service area.

Zone C-1 is the central business district located in the heart of the City of Cornelia. It is an extremely dense building area for this size town with an inventoried 73 commercial structures, composed of one and two story brick buildings, most of which have adjoining walls. In most cases the building shells are 30 to 40 years old. Many buildings have had interior remodeling to serve the current occupants. As is typical of older commercial structures, little or no insulation is present except where added during remodeling. There

are also some newer structures ranging in age from 3 to 20 years. Most buildings in this zone are heated and cooled with forced air systems. The heating requirements are provided by natural gas furnaces or electric resistance heaters. Cooling is accomplished by direct refrigerant expansion systems using air cooled condensers.

Zone C-2 is a new strip shopping center located on the Highway 441 Bypass. When ultimate build-out potential is reached, an estimated 10 commercial structures will be located on this site. The center is typical of current commercial construction and consists of masonry walls, glass storefront and a metal deck, insulating board and built up roof assembly. Insulation is included in the ceilings and walls. The stores within the shopping center will be heated by electric resistance heat and cooled by roof mounted packaged air conditioning units. Each store has its own individual HVAC system.

Zone C-3 is an existing strip type shopping center located on the Highway 441 Bypass. An estimated 8 commercial establishments will be located at this site, upon completion. The center is approximately three years old and is constructed with masonry walls, glass storefront and a metal deck, insulating board and built-up roof assembly. Typically, only the roof is thermally insulated. The stores within the shopping center are heated and cooled by roof mounted packaged air conditioning units which utilize electric resistance heat. Each store has its own individual HVAC system.

Total commercial structure square footage in the three proposed service zones is estimated at 434,709. The estimate is based upon actual inventories of the existing structures in the central business district (Zone C-1) and upon estimates of the completed square footage of the commercial establishments to be located in the two shopping centers (Zones C-2 and C-3).

TO SEWAGE
TREATMENT FACILITY

3500'

WOOD ST.

FOSTER RD.

HOYT ST.

WAYSIDE ST.

WAYSIDE ST.

ELROD RD.

NORTH MAIN ST.

GALLOWAY ST.

CITY LIMITS

EXISTING SHOPPING
CENTER
ZONE C-3

HP-WHR
CENTRAL
STATION

LEVEL GROVE RD.

OAK ST.

NEW
SHOPPING
CENTER
ZONE C-2

FIELDALE
POULTRY PLANT
ZONE I-4

CENTRAL BUSINESS
DISTRICT
ZONE C-1

U.S. HWY. 441

IRVIN ST.

CITY LIMITS

SOUTHERN RAILROAD

YONAH ST.

MAIN ST.

0   300' 600'

CITY OF CORNELIA
SERVICE AREAS

FIGURE 9-2

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9.1.7 HEATING AND COOLING LOADS

Monthly Loads

Heating and cooling loads for the Cornelia study area were estimated on
a monthly basis for those building structures served within the projected
service area. The primary emphasis in load development was placed on monthly
figures since no greater degree of detail could be justified in a preliminary
study of this type. Peak loads were also developed using design conditions
from the 1977 Fundamentals volume of ASHRAE Handbook and Product Directory.[50]

The estimation of monthly heating and cooling requirements was calculated
by the computer program discussed previously in Chapter 8.0 of this report.
The input to the calculation procedure takes into account building area served
and unit building energy requirements determined by the program user for build-
ing type and climatological zone.

The service area under consideration for Cornelia contains only commercial
establishments and an industrial plant located close to the proposed central
heat recovery plant. Building count and area were estimated from field surveys
and data excerpt from the Cornelia Community Development Plan[57] prepared by the
Georgia Mountains Area Planning and Development Commission. Experience with
commercial building energy use patterns in the southeast indicates that annual
heat losses will total approximately 50 MBtu/ft^2 and that cooling requirements
will be in the range of 32 MBtu/ft^2 for the type buildings found in the Cornelia
service area.

Monthly heating and cooling loads were determined by apportioning the
annual requirement according to the percentage of heating or cooling degree
days occurring in a particular month. Heating and cooling degree days were

taken from "Monthly Normals of Heating and Cooling Degree Days 1941 - 1970", published by the National Climatic Center, Asheville, North Carolina.[55] Table 9-2 is a summary of the percentage figures utilized in apportioning the annual load estimates.

TABLE 9-2
Annual Heating and Cooling Degree Days
Cornelia, Georgia

Month	Heating		Cooling	
	Degree Days	% Total	Degree Days	% Total
Jan.	732	21.0	0	0.0
Feb.	605	17.3	0	0.0
March	492	14.1	12	1.0
April	188	5.4	20	1.7
May	61	1.7	108	9.0
June	10	0.3	244	20.3
July	0	0.0	322	26.8
August	0	0.0	310	25.6
Sept.	29	0.8	155	12.9
Oct.	196	5.6	32	2.7
Nov.	468	13.3	0	0.0
Dec.	716	20.5	0	0.0
	3497	100.0	1203	100.0

Industrial energy requirements are significantly more difficult to estimate without a detailed analysis of process and environmental constraints. However, to account for a potential energy credit from the industrial concern included in the service area, a constant monthly cooling requirement of 230 MBtu/month was allowed. This cooling requirement represents a constant demand

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at the subject plant for condenser water service for ice-making equipment.

Table 9-3 presents a month-by-month summary of the total projected thermal load on the HP-WHR system. The results of the computerized load calculations and peak load determinations are presented in Appendix D.

TABLE 9-3
CORNELIA HP-WHR SYSTEM LOADS

PERIOD	HEATING BTU X 10^9	COOLING BTU X 10^9
Jan.	4.564	0.0002
Feb.	3.760	0.0002
March	3.065	0.139
April	1.174	0.235
May	0.370	1.252
June	0.065	2.826
July	0.0	3.730
August	0.0	3.574
Sept.	0.174	0.922
Oct.	1.217	0.374
Nov.	2.891	0.0002
Dec.	4.456	0.0002
	21.736×10^9 BTU	13.053×10^9 BTU

Peak Heating Load

As mentioned above, peak loads were estimated using the procedures and design conditions from the ASHRAE Handbook.[50] The degree day formula, with terms rearranged to solve for the peak load, P, was utilized as shown below:

$$P = \frac{(E) \times (TD)}{(DD) \times (24)}$$

Where:

P = Design heating load per hour

E = Heat energy required

TD = Design temperature difference

DD = Heating degree days

For this estimate:

E = 4.564×10^9 Btu – from the computer load estimate for January

TD = $70^\circ - 17^\circ = 53^\circ F$ – desired inside temperature minus the 99% design temperature for Cornelia.

DD = 732 – from Table 9-2.

Thus

$$P = \frac{(4.564 \times 10^9) \times (53^\circ)}{(732) \times (24)}$$

$$= 13.77 \times 10^6 \text{ Btu/hr}$$

Peak Cooling Load

Utilizing the design figures of an average 350 square feet per peak ton of refrigeration for commercial structures, the peak cooling load for the proposed Cornelia service area was estimated as follows:

$$\text{Peak Cooling Load} = \frac{434,709 \text{ ft}^2 \text{(building area served)}}{350 \text{ ft}^2/\text{ton}}$$

$$= 1242 \text{ Tons}$$

9.1.8 THERMAL ENERGY RESOURCES

As noted in earlier chapters of this report, the low-grade heat in a community's wastewater will be utilized as the thermal energy source in an HP-WHR based HP-ICES. Also as explained earlier, many site-specific variables affect the amounts and availability of thermal energy from this source at any particular point of application. To determine the quantities of thermal energy available at a specific demonstration community necessarily involves examination and analysis of the community's wastewater collection and treatment facilities along with any appropriate documentation of operating conditions, performance, etc. The following sections examine the wastewater facilities of the Cornelia, Georgia community and develop estimates of the thermal energy available to an HP-WHR installation at a proposed location in Cornelia.

CITY OF CORNELIA, GEORGIA WASTEWATER FACILITIES

The City of Cornelia completed in July 1977 a comprehensive review of its wastewater collection and treatment facilities in the form of a 201 Facilities Plan under E.P.A. Project # C130585-01-0.[60] Pertinent data and descriptions from this report will be adopted as the basis for the evaluation of thermal energy availability at the Cornelia community.

The layout of the wastewater system, detailing location of lines, sizes, manholes, pump stations, and force mains, is depicted on Figure 9-3, reproduced from the 201 Plan. Also shown on this layout are the major division lines between the flow metering areas employed for the 201 Study Analysis. Sections

of the system vary in age from one year to 65 years. The older portion of the system is composed of concrete pipes with concrete joints and is located principally in the downtown area, located in flow metering area 3 and the western portion of area 4. The most recently constructed sewers are located in flow metering areas 2 and 6, and the outskirts of area 7. Sewer construction in these areas incorporates terra cotta pipe with rubber O-ring sealed joints. The general condition of the entire sewer system is good relative to deterioration of existing pipes and manholes.

The original Cornelia Treatment Plant was built at its present location in 1963. The original plant consisted of coarse screening, grit removal, fine screening (1/4"), primary clarification, low rate trickling filter, polishing pond, anaerobic sludge digestion (uncovered), and sand drying beds. The facility was expanded in 1967 with the following additions: a second primary clarifier, a large high rate trickling filter, a secondary clarifier, and a second anaerobic sludge digestor (uncovered). More recently, and in response to the NPDES permit requirements for fecal coliform removal and post-treatment aeration, a chlorine feed system and contact basin with cascading outfall were added to the process. The chlorination system and cascading outfall became operational in mid-July 1976. The plant as it exists today is shown in Figure 9-4.

FLOW AND TEMPERATURE DATA - CORNELIA, GEORGIA

As discussed in Chapter 10.0 - COMPONENT TESTING, site-specific temperature and flow measurements are the necessary first step in developing a data base for determining the thermal energy available at any particular site of application. Measurement of these parameters at the Cornelia sewage treatment facility were performed during the period 10/13/78 to 10/20/78. Temperature measurement was with a Simpson Type 23103 portable strip-chart recorder with
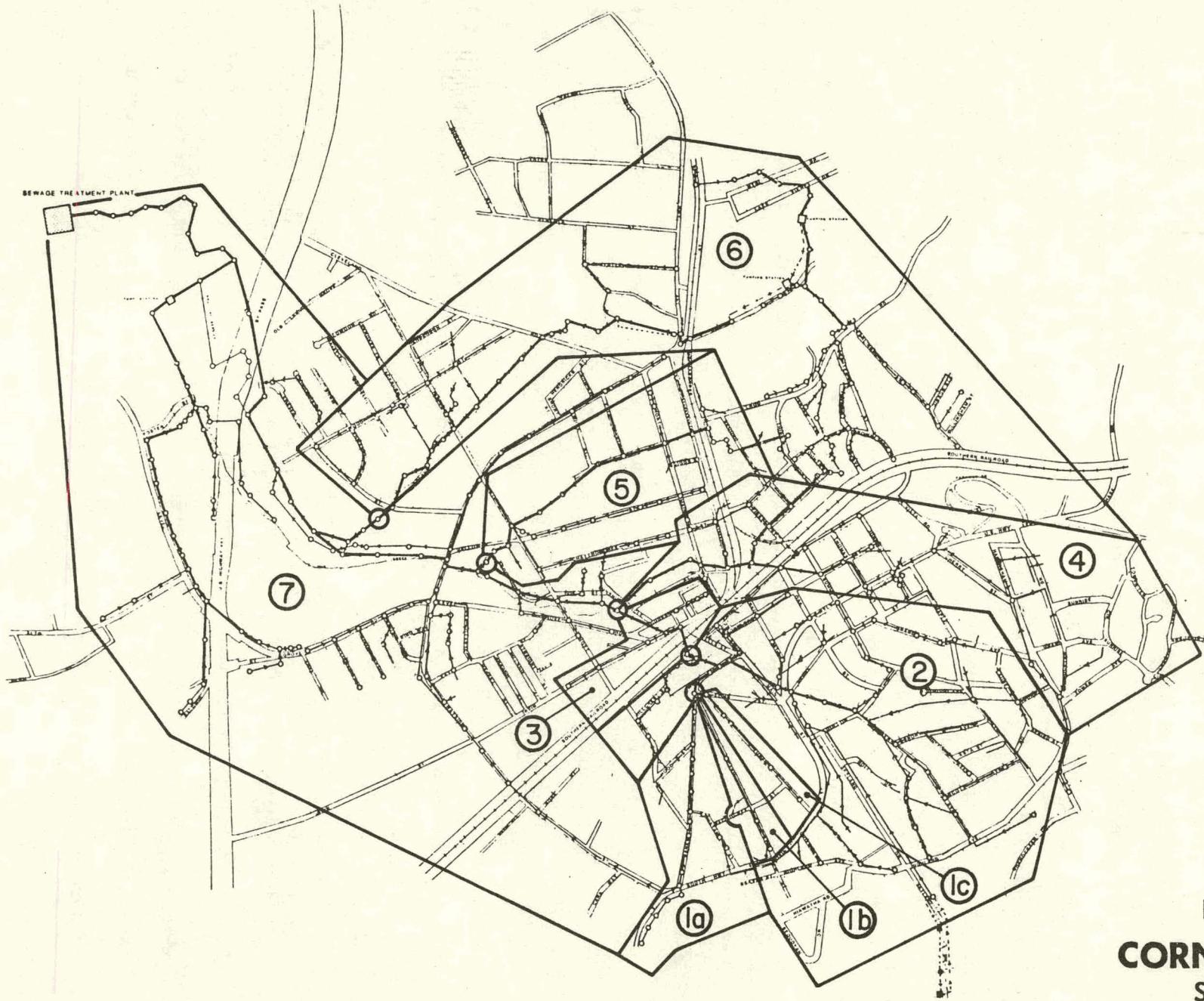
LEGEND

o—o	EXISTING LINES
●····●	PROPOSED LINES
o	MANHOLE
▢	PUMPING STATION
▢	TREATMENT PLANT
FM	FORCE MAINS

FIGURE 9-3

CORNELIA, GEORGIA
SEWER SYSTEM

LEGEND
SEWAGE FLOW
SLUDGE
BYPASS

CHLORINATOR &
STORAGE BLDG.

POTABLE WATER WELL

NOTE:
CHLORINE CONTACT BASIN IS
LOCATED AT N.W. EDGE OF POLISHING POND

EXISTING DIRT ROAD

POTABLE WATER LINE

OLD DIGESTER
(UNCOVERED)

NEW DIGESTER
(COVERED)

CHLORINE FEED LINE

FILTRATE OUTFALL

SAND DRYING BEDS

EXISTING DIRT ROAD

GREASE PUMP-OUT

SLUDGE PUMP STA.

DIGESTER OVERFLOW LINE

ROTARY SCREEN

GRIT CHAMBER

SLUDGE RETURN

PRIMARY CLARIFIER

OLD TRICKLING FILTER

RAW SLUDGE WET WELL

LAB

INFLUENT

PRIMARY CLARIFIER

SCREEN WASH LINE

SECONDARY CLARIFIER

MAIN FLOW BYPASS

RECIRC LINE

POLISHING POND

NEW TRICKLING FILTER

CITY OF CORNELIA
WASTEWATER TREATMENT PLANT
FLOW SCHEMATIC

FIGURE 9-4

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a type R-22, 3-SE100-120 resistance temperature device (RTD). Measurements were made at two-points in the sewage treatment plant: 1.) plant influent at the head of the grit chamber, and 2.) at the secondary clarifier (essentially the last point in the treatment process before discharge). Flow measurement was taken from existing plant instrumentation. Results are presented graphically (Flow vs. Time and Temperature vs. Time) in Figure 9-5 and summarized in Table 9-4.

Flow was found to be on a daily cycle, as expected, with significant peaks at mid-morning (9:30 - 11:00am) and mid-afternoon (1:30 - 3:00pm) during the weekdays (no doubt due to the Monday through Friday operation of the Fieldale poultry processing plant which produces approximately 72% of the wastewater processed at the Cornelia plant). Flow was noticed to be fairly constant and at a low level on the weekend with Saturday flows at a relatively higher level (and with several significant peaks) than on Sunday. Average Daily Flow (ADF) during the weekdays was 1,640,000 gallons per day (1.65 MGD). ADF during the weekend was 350,000 gallons per day (.35 MGD). Weighted ADF over the entire week is 1.25 MGD.

Plant influent temperature on the weekdays seemed to follow with the level of flow (again likely contingent on Fieldale's discharge) with high temperatures ranging from 95°F to 109°F, peaking usually in late afternoon (4:00pm or later) or into the evening hours. High temperatures on the weekend ranged from 82°F to 90°F. Low temperatures on plant influent occurred usually during the early morning hours (7:00 - 9:00am) and ranged from 70°F to 82°F.

Temperatures at the secondary clarifier remained relatively constant at all times during the day and week, ranging from 68°F to 82°F. Slightly higher temperatures seemed to prevail in the clarifier during the weekend (perhaps due

CORNELIA, GEORGIA
SEWAGE TREATMENT PLANT
10-14-78—10-19-78

FIGURE 9-5
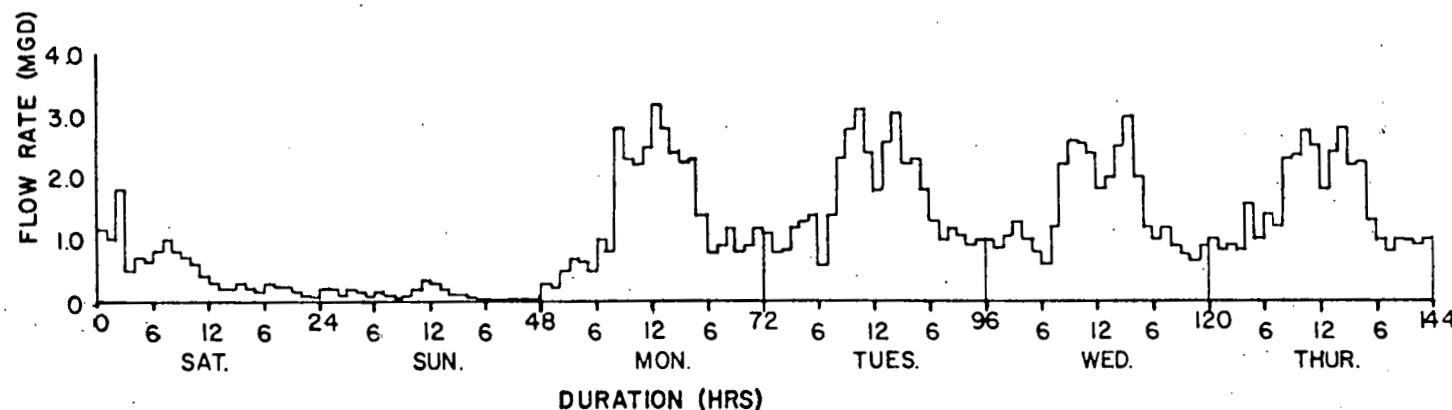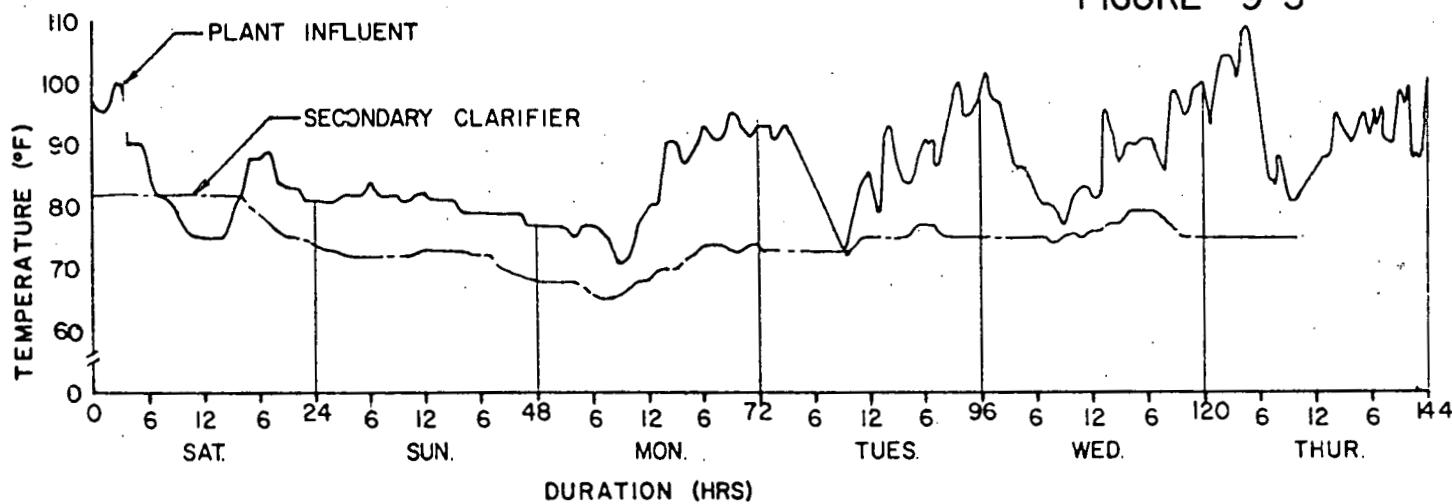
to the extended retention time - up to 15 hours - made necessary by the low
weekend flow levels).

It is expected that, for the Cornelia wastewater system, sewage flows will
continue to show a daily pattern of high flows during normal working hours due
to the continuing presence of the Fieldale plant, and lower flows during evening
and early morning hours. The weekend pattern of proportionately lower flows
will also likely continue.

It is also expected that the temperature of the plant influent will
continue to exhibit daily peaks of approximately 104°F during the late after-
noon hours of the weekdays, due to the relatively constant temperature discharge
of Fieldale's processes. Temperatures at the secondary clarifier will continue
on their relatively constant trend. Extreme seasonal influences may reduce the
operating temperatures somewhat, but will not significantly affect the avail-
ability of the thermal energy at this source.

From the above characterization of the Cornelia wastewater conditions it
is apparent that signficant amounts of thermal energy are available for reclaim
with an HP-WHR system. The next report section will discuss briefly the factors
pertinent to the wastewater system which will affect the location of the HP-WHR
central station and will develop estimates of the quantities of thermal energy
available at the proposed location.

CENTRAL PLANT LOCATION & THERMAL ENERGY ESTIMATE

As explained in the discussion of the proposed service area, primarily
commercial customers will be served from the initial system. The most
attractive service areas, therefore, will be the central business district and
the two shopping centers developing along the newly constructed Cornelia By-pass.
Inspection of the sewer system layout reveals that a central station sited in

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TABLE 9-4
CORNELIA, GA. WASTEWATER SYSTEM
SEWAGE TREATMENT PLANT DATA

TEMPERATURES

		PLANT INFLUENT		SECONDARY CLARIFIER	
Day		Low (°C/°F)	High (°C/°F)	Low (°C/°F)	High (°C/°F)
FRI	10/13	28/82	36/97	28/82	30/86
SAT	10/14	24/75	32/90	28/82	28/82
SUN	10/15	25/77	28/82	20/68	23/73
MON	10/16	21/70	35/95	19/66	23/73
TUE	10/17	23/73	39/102	22/72	25/77
WED	10/18	25/77	40/104	24/75	25/77
THUR	10/19	26/79	43/109	24/75	24/75
FRI	10/20	26/79	40/104	-----	-----

FLOW

Low: $4167 \frac{\text{gal}}{\text{hr}}$ ($\simeq$ 0.1 MGD) SUN 10/15 5:00pm

High: $154,167 \frac{\text{gal}}{\text{hr}}$ ($\simeq$ 3.7 MGD) MON 10/16 12:15pm

Weekday Avg: 1,640,000 GPD*

Weekend Avg: 350,000 GPD*

* FROM E.P.A. 201 FACILITIES PLAN[60]

the vacant area bordered by Foster Street, Wayside Street, and Level Grove

Road would have access to the parallel sewer mains which carry the bulk of the

community wastewater effluent, including the discharge from the Fieldale plant,

to the sewage treatment plant. Thus, the wastewater flows from flow metering

areas 1A-C, 2, 3, 4, 5, and Fieldale would be available for heat reclaim at the

central station. A station sited at this location would also be central to a

distribution system serving the two main service areas, being approximately

mid-way on a line between the central business district and the new shopping

centers.

From Chapter 2.0 - POTENTIAL APPLICATIONS, the total amount of heat avail-

able from the thermal supply (wastewater) can be calculated from equation (2-2)

shown below:

$$Q = W C (\Delta T)$$

Where:

Q   =   total heat, Btu
W   =   flow of wastewater, lbm
C   =   specific heat of wastewater (assumed to be 1.0 Btu/lbm-$^\circ$F)
ΔT =   change in temperature of supply medium across the central station
         equipment heat exchanger, $^\circ$F

For the purpose of making nationwide projections on available thermal

energy from wastewater, flow was defined as a function of national population,

and industrial wastewater discharges (a significant source of reject thermal

energy) were ignored due to insufficient data. However, due to the availability

of the 201 Facilities Plan for the Cornelia community and sewage system, actual

flow data for the various flow metering areas in the community will be used

as the basis for the estimates for available thermal energy. It should be noted

that these figures also include industrial discharges.

Flow data from the 201 study is presented in tabular form as Table 9-5,

TABLE 9-5
CORNELIA, GA. WASTEWATER SYSTEM
GENERAL FLOW METERING AREA DATA

FLOW METERING AREA	LINEAR FT. OF SEWER	SEWER DIAMETER (INCHES)	AVERAGE DAILY FLOW (GPD)
1A	5,600	8	9,000
1B	2,600	8	4,300
1C	1,900	6	4,300
2	30,200	10	60,000
3	3,300	10	14,000
4	17,800	10	46,000
5	15,600	12	38,000
6	26,300	10	83,000
7(W/O FIELDALE)	38,600	18	93,000
FIELDALE			920,000
TOTAL	141,900 FT.		1,271,600 GPD
*WEEKDAY			1,640,00
*WEEKEND			350,000

*SOURCE: EPA 201 FACILITIES PLAN

and schematically as Figure 9-6. Figure 9-6 also displays schematically the proposed location of the HP-WHR central station fiacility in relation to the overall sewer system. Thus, the central station, installed at this proposed location with the necessary storage capacities to absorb the diurnal fluctuations in flow and take advantage of the full available wastewater flow, would have access to an average flow (ADF) of 1,095,600 gallons per day (1.096 MGD – the sum of flows from metering areas 1A, 1B, 1C, 2, 3, 4, 5, and Fieldale). Note that this figure is a weighted daily average of flows over a one week period. Weekend flows may be significantly less than this weighted average, thus the necessity of incorporating some form of storage capacity into the central station design.

Temperature data from the sewage treatment facility will also be utilized in developing estimates of available thermal energy. It is duly noted that the conditions documented herein are from a somewhat arbitrarily selected one-week period which may or may not have been representative of conditions to be expected over an annual cycle. However, inspection of the seven-day flow cycle reveals no characteristics which would indicate it to be atypical. Further, since the flow pattern documented in this effort coincides quite accurately with that found during the E.P.A. 201 Facilities Plan analysis, it is anticipated that the documented weekly flow cycle can indeed be taken as typical. The temperatures of the wastewater, on the other hand, will likely exhibit more significant variations due to seasonal influences. However, due to the fact that the major controlling factor on wastewater temperature (at least during the five weekdays) is the temperature of Fieldale's process discharge, it is anticipated that the affect of ambient temperature changes on the wastewater temperature will be moderated somewhat. The temperature and

CORNELIA, GEORGIA
WASTEWATER SYSTEM
FLOW AREA SCHEMATIC
FIGURE 9-6
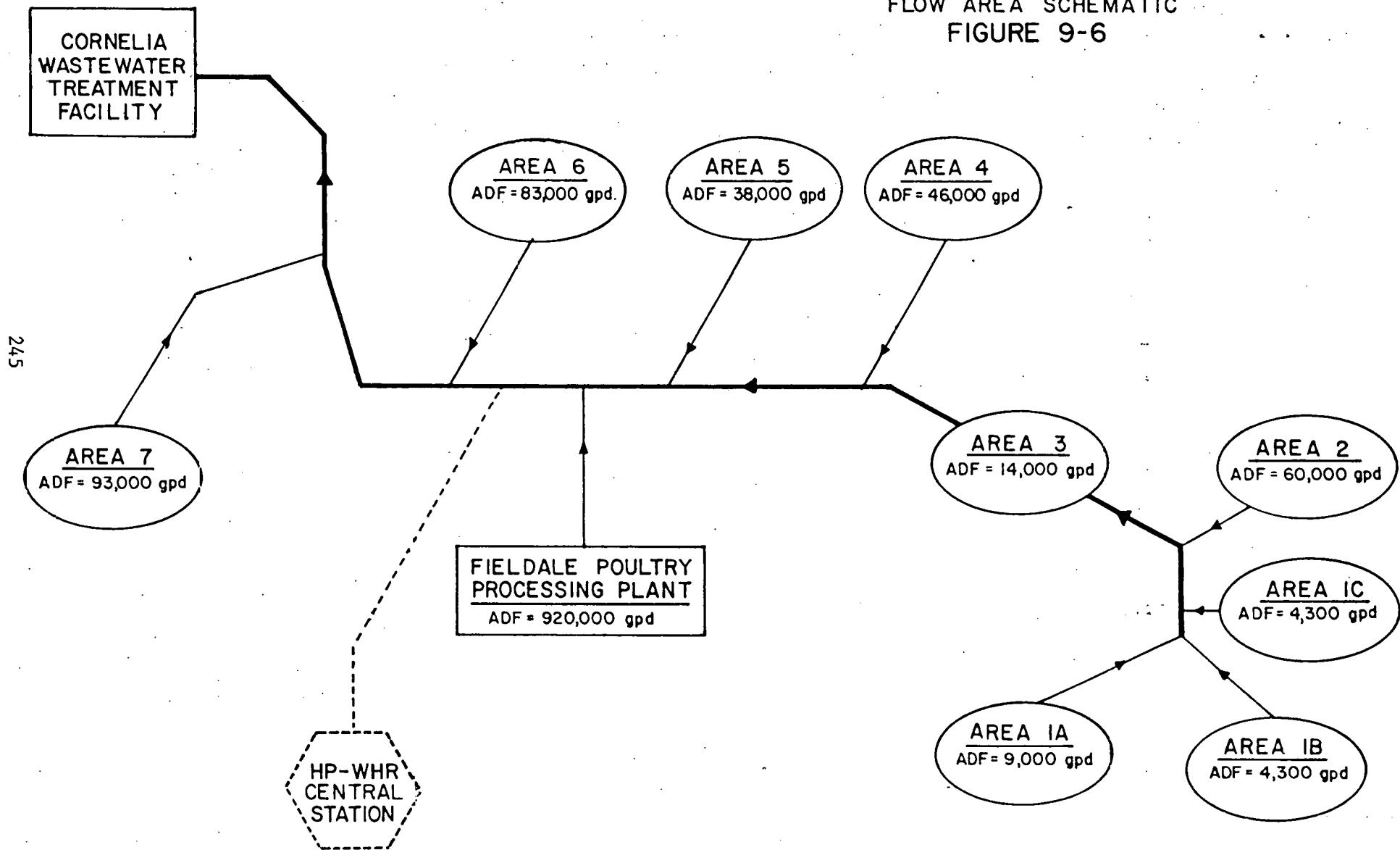
CORNELIA
WASTEWATER
TREATMENT
FACILITY

AREA 6
ADF = 83,000 gpd.

AREA 5
ADF = 38,000 gpd

AREA 4
ADF = 46,000 gpd

AREA 7
ADF = 93,000 gpd

AREA 3
ADF = 14,000 gpd

AREA 2
ADF = 60,000 gpd

AREA IC
ADF = 4,300 gpd

FIELDALE POULTRY
PROCESSING PLANT
ADF = 920,000 gpd

AREA IA
ADF = 9,000 gpd

AREA IB
ADF = 4,300 gpd

HP-WHR
CENTRAL
STATION

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flow data presented here and utilized in the following estimate of available
thermal energy are presented primarily to provide a firm base for this "ballpark"
estimate. Figure 9-7 is a Temperature vs. Duration curve for plant influent
temperature during the period 10/13/78 through 10/20/78, developed from the
strip-chart temperature plot. A weighted average of the data shows the average
temperature to be 86°F.

CORNELIA, GEORGIA
SEWAGE TREATMENT PLANT
PLANT EFFLUENT TEMPERATURE
VS. DURATION

WTD AVG. = 86°F
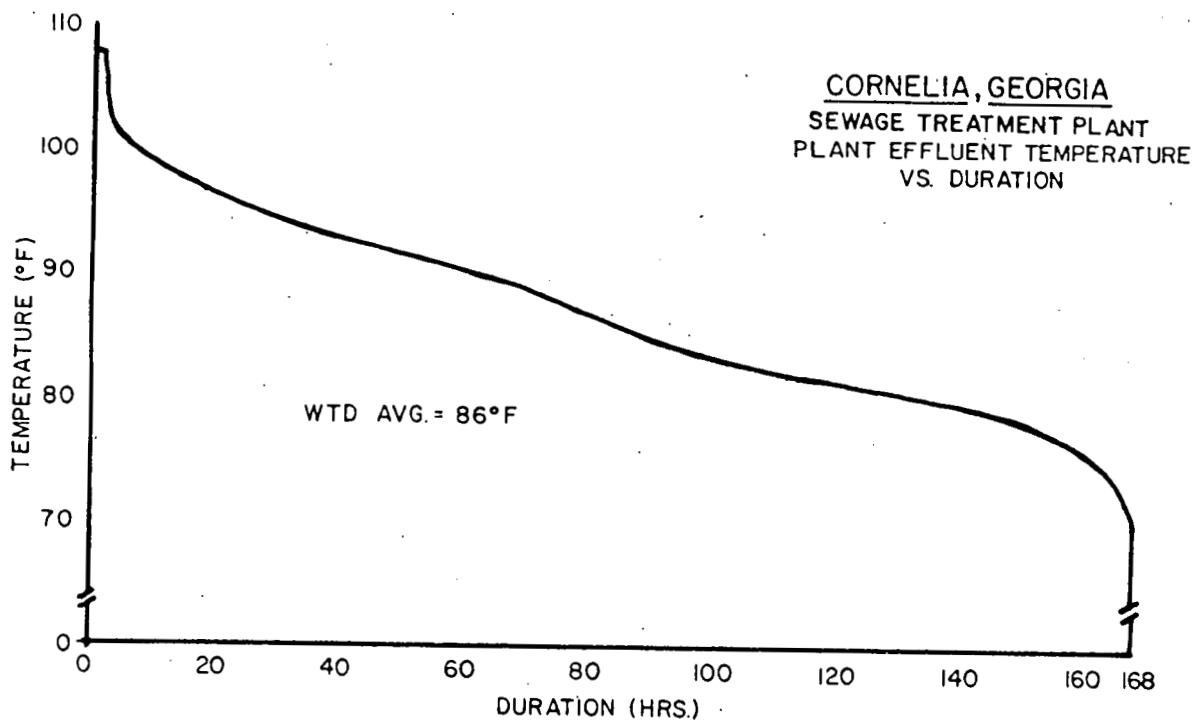
Figure 9-7

.A third condition to be defined for the thermal energy estimate is the change in temperature of the heat supply medium, ΔT in equation (2-2), which could be achieved across the central station equipment heat exchanger. Essentially this would represent the heat transferred out of the wastewater flow. Theoretically, heat could be removed from a water source down to approximately 32°F. However, due to equipment operating characteristics, a practical limit of 30°F can be set for ΔT.

Inspection of equation (2-2) reveals that the greatest quantity of heat would be reclaimed with the largest ΔT possible, in this case $\Delta T = 30$°F. General HVAC system design practice dictates a $\Delta T \simeq 10$°F. However, in this particular application, larger ΔT's may be necessary or desirable. Therefore, in order to bracket the operating conditions which may be in effect at the central station, ΔT's of 10°F, 20°F, and 30°F will be considered. Estimates of the thermal energy available for reclaim from Cornelia's wastewater can now be calculated.

As an example, and utilizing equation (2-2), the energy reclaimed from community wastewater would be calculated thusly:

$$\text{AVG. DAILY FLOW} - 1,095,600 \text{ gallons}$$

$$\Delta T - 30°F$$

$$Q_{DAILY} = 1,095,600 \frac{gal}{day} \times (8.33 \frac{lbm}{gal}) \times (1.0 \frac{Btu}{lbm-°F}) \times (30°F)$$

$$= 273,790,440 \frac{Btu}{day}$$

Derived through this estimative methodology, Table 9-6 summarizes the quantities of thermal energy potentially available, on a daily basis, to an HP-WHR system installed at the proposed location in Cornelia and operating at the various conditions discussed in the foregoing paragraphs.

TABLE 9-6
HEAT AVAILABLE FROM WASTEWATER
CORNELIA, GEORGIA

$\Delta T(^{\circ}F)$	WASTEWATER CONDITIONS IN($^{\circ}$F)	OUT($^{\circ}$F)	Q_{DAILY} AVG. DAILY HEAT RECOVERY (Btu/day)
10	86	76	91.26×10^{6}
20	86	66	182.53×10^{6}
30	86	56	273.79×10^{6}

An important consideration in the application of an HP-WHR system to a community is the balance between the portions of the community served by the wastewater collection and treatment system and the portion of the community proposed for service by the heat pump system. Based upon the above estimate of thermal energy available at the Cornelia HP-WHR central plant and upon the assessment of the proposed service areas, the space conditioning requirements of the 73 commercial structures in the central business district and of the approximate 18 structures which will ultimately be constructed in the two shopping centers, and a portion of Fieldale's process requirements can be served under this system application. Examination of the wastewater collection system layout and of existing land use patterns and building locations reveals that, due to the proposed location of the HP-WHR central plant and its attendant take-off from the sewer trunk lines, only a portion of the community will serve to supply wastewater/thermal energy to the heat pump system. A field inventory of Cornelia and contiguous areas performed for the Cornelia Community Development Concept Plan[57] indicates that 868 residential structures, 194 commercial and public structures, and 11 industrial structures are served by the wastewater collection system prior to the HP-WHR system tie-in. Of

primary consequence is the fact that the wastewater effluent from Fieldale, which comprises approximately 72% of the total average daily wastewater flow from the community, is available at the proposed central plant location. Obviously, with this composite service area (residential, commercial, and industrial discharges) on the wastewater system, a concise balance ratio cannot be drawn as could be for the purely residential service Conyers system. However, the above figures detailing the respective service areas of the wastewater and heat pump systems should serve to illustrate the balance between the two.

9.2 CORNELIA SYSTEM DESCRIPTION

The wastewater heat recovery system for the Cornelia community will intercept warm sewage effluent from community industries and the majority of community residential buildings and extract heat ultimately to be used by selected commercial and industrial customers. The Cornelia system concept, while not different from the concept of extracting low-grade heat from sewage, differs from the original Heat Pump-Wastewater Heat Recovery System concept of cascaded heat pumps in several areas. These areas are briefly outlined in following discussions.

The relatively high temperature of the sewage effluent encountered in the Cornelia community wastewater collection system (86°F) suggests that a water-to-water heat exchanger might be more practical for extracting the waste heat contained in the sewage effluent than using a centrifugal heat pump. With this system modification, distribution system water temperatures of approximately 70°F should be obtainable for use by end-user heat pump systems throughout the community. In the event that wastewater temperatures were lowered due to dilution or unforeseen curtailment of thermal discharges into the sewer system, the heat exchanger component could be easily replaced with a central heat pump of the type envisioned for the basic HP-WHR System.

Another variation from the original Heat Pump-Wastewater Heat Recovery System concept is the use of an evaporative cooling tower to dissipate building heat in the summer rather than direct heat rejection to the wastewater stream. As demonstrated in the earlier concepts, the evaporative cooling component offers a significant energy conservation potential especially when one considers the high temperature of the wastewater stream into which the central heat pump would have to reject building heat. Figure 9-8 is a single

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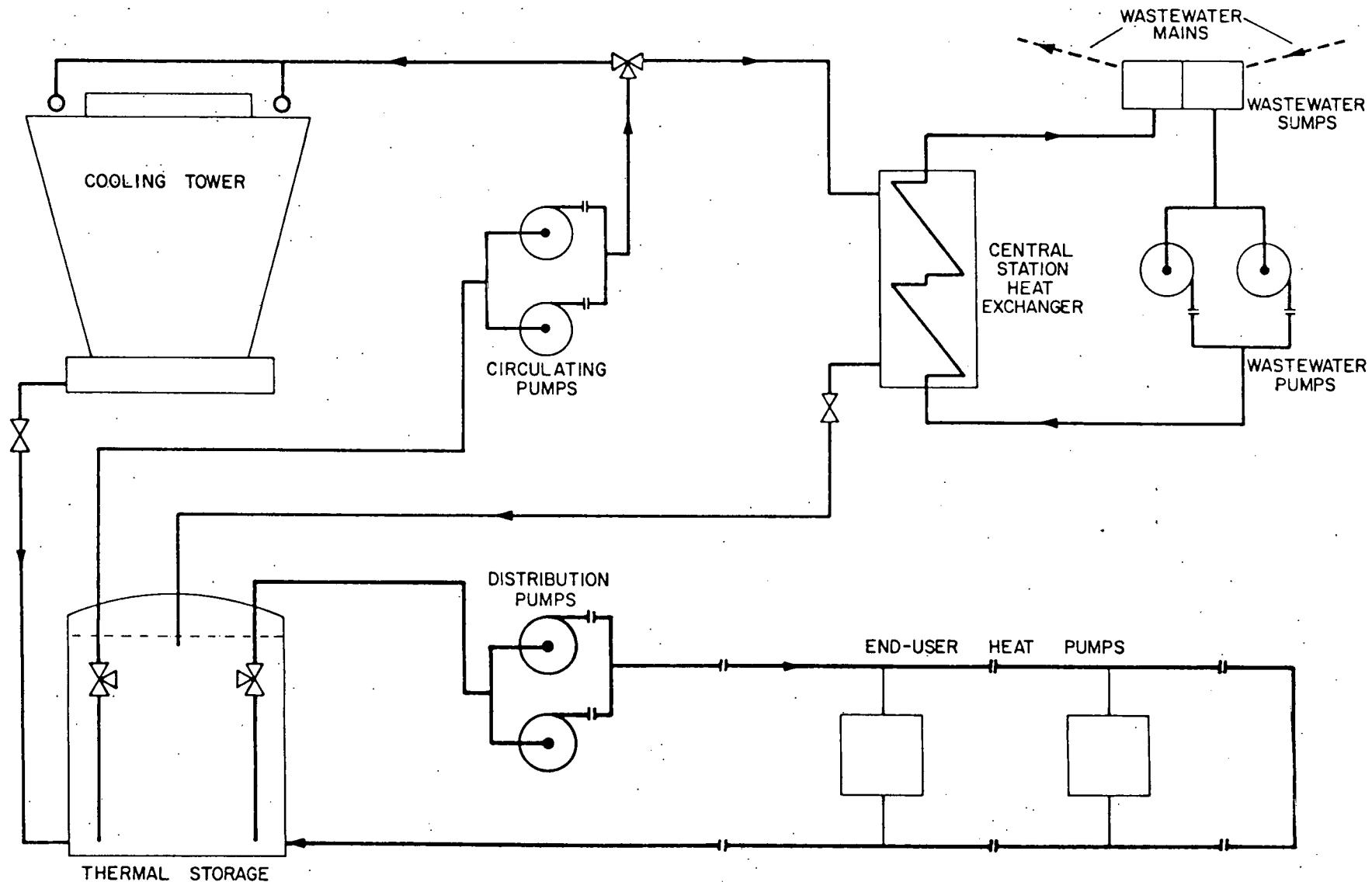
CITY OF CORNELIA

HP-WHR SYSTEM SCHEMATIC

FIGURE 9-8

line schematic diagram for the Cornelia HP-WHR system.

The end-users served by the Heat Pump-Wastewater Heat Recovery System will be a subset of the Cornelia community composed of selected industrial and commercial establishments and the central business district. This subset was chosen because of its proximity to the thermal energy source, its high dependence on electric resistance heating as a current energy source, and the perception that these segments of the community would better be able to retrofit their buildings and establishments to use the recovered heat than would the typical residential consumer in the City of Cornelia.

The HP-WHR central station will be sited on a parcel of land centrally located and in close proximity to the four proposed service zones. Interconnection with the existing wastewater collection system will be at a point on the main trunk line crossing the selected central plant site. The central plant layout and piping interconnection will be as illustrated in Figure 9-9. The warm water distribution system for the selected service areas will be designed and routed as illustrated in Figure 9-10.

The HP-WHR system application in the Cornelia community offers the unique opportunity to evaluate heat recovery from wastewater effluent through direct heat exchange techniques, thermal energy distribution to end-users, and the ultimate use of that low-grade thermal energy by commercial and industrial consumers. It is felt that the proposed system concept offers the greatest potential for implementation by the Cornelia community.

9.2.1 SYSTEM DESIGN CALCULATIONS

The major components comprising the HP-WHR system which are to be specified in a system development study include the central heat pump, the thermal storage vessel, the cooling tower, auxiliary pumps, and end-user heat pumps.

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12"

EXISTING SEWER MAINS

15"

NEW SEWER MAINS

WATER TRTMNT

SEWAGE PUMPS

CIRC. PUMPS

DIST PUMPS

BACK-UP BLR (IF REQ'D)

HEAT EXCHANGER

FUTURE EXPANSION

COOLING TOWER(S)

CNTRLS

ELECTRICAL

ROLL UP DOORS

THERMAL STORAGE

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WAYSIDE STREET

NORTH

CITY OF CORNELIA, GA.

HP-WHR SYSTEM
PROPOSED CENTRAL STATION LAYOUT

FIGURE   9-9

TO SEWAGE
TREATMENT FACILITY

3,500'

CITY LIMITS

WOOD ST.

FOSTER RD.

HOYT ST.

WAYSIDE ST.

GALLOWAY ST.

NORTH MAIN ST.

ELROD RD.

EXISTING SHOPPING
CENTER
ZONE C-3

HP-WHR
CENTRAL
STATION

MANHOLE
(TYPICAL)

18"

LEVEL GROVE RD.

3"

4"

4"

3"

OAK ST.

8"

8"

6"

3"

NEW
SHOPPING
CENTER
ZONE C-2

5"

4"

14"

12"

3"

3"

4"

US. HWY. 441

FIELDALE
POULTRY PLANT
ZONE I-4

CENTRAL BUSINESS
DISTRICT
ZONE C-I

SOUTHERN RAILROAD

YONAH ST.

MAIN ST.

IRVIN ST.

CITY LIMITS

0   300' 600'

CORNELIA, GEORGIA HP-WHR SYSTEM
SERVICE AREA AND DISTRIBUTION PIPING

FIGURE   9-10

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However, since for any system the end-user heat pump specification will be necessarily structure-specific, sizing of those components will not be addressed in this report section. The remainder of the major component design calculations are presented in the following paragraphs.

Central Heat Exchanger

The central heat exchanger will serve in lieu of a central heat pump to transfer the low-grade thermal energy in Cornelia's wastewater to the HP-WHR system transport media. Due to the fact that the central station is located on a site away from the sewage treatment facility, the potential for warm wastewater storage in the facilities clarifiers will not be available. Thus the central heat exchanger must be of sufficient size to take advantage of the peak wastewater flows during a daily cycle. Peak flow during the documented period was 154,167 gal/hr. The heat exchanger will be sized to handle approximately 80% of this peak flow, or 2100 GPM. An entering wastewater temperature of $80°F$ will be assumed with a ΔT drop of $20°F$ across the heat exchanger. ($80°F$ is assumed rather than the documented $86°F$ in order to produce an appropriately conservative design.) The circulating water system between the heat exchanger and thermal storage will consequently be designed to accommodate a flow rate of 4200 GPM entering at $55°F$ and leaving at $65°F$ (the $10°F$ ΔT rise which will be necessary for thermal energy distribution into the community). Operation will be on a 24 hour basis.

Based on the above noted operating conditions, the Log Mean Temperature Difference across the central heat exchanger would be:

$$\text{LMTD} = \frac{(60-55)-(80-65)}{\text{Ln}[\frac{(60-55)}{80-65}]}$$

$$= 9.10\,^\circ F$$

Assuming an overall heat transfer coefficient, U_o, of 250 Btu/hr-ft^2-$^\circ$F typical in heat exchangers of this type[59], the heat exchanger area required is determined as follows:

$$A = \frac{\text{GPM} \times (8.33\,\frac{\text{LBM}}{\text{Gal}})(60\,\frac{\text{MIN}}{\text{HR}}) \times \Delta T}{(U_o) \times \text{LMTD}}$$

$$= \frac{(2100)(8.33)(60)(20)}{(250)(9.1)}$$

$$= 9227\ \text{ft}^2$$

The external dimensions of a heat exchanger with this amount of heat transfer surface will be approximately 26' length by 8' diameter.

The applicability of the central heat exchanger is verified when the effective load on the central station is compared to its potential performance. Since the thermal storage component will allow the system to moderate somewhat the peak thermal demand, the quantity of heat which must be extracted from the wastewater flow during an average day of the peak heating month of January can be used as the basis of the comparison. The average daily heat requirement is calculated as follows:

$$Q_{\text{wastewater}} = \frac{Q_{\text{heat req'd}}\ (1-\frac{1}{\text{C.O.P.}_{\text{end-user}}})}{\text{DAYS IN PERIOD}}$$

$$= \frac{(4.564 \times 10^9)(1-\frac{1}{2.9})}{31}$$

$$= 96.46 \times 10^6\ \text{Btu/day}$$

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For comparison, the potential for heat reclaim at the central heat exchanger can be calculated, based on the available wastewater flow at the central station site. Inspection of the flow vs. time plots from section 9.1.8 reveals that the absolute peak flow at the sewage treatment facility exceeds the specified flow capacity of the central heat exchanger 3.02 MGD (2100 GPM x 60 x 24) — for just 2 hours of the 144 hour test period. Therefore, since no inordinate amount of the daily flow is received at rates exceeding the heat exchanger capacity, the average daily flow figures for the central station site can be utilized as the basis for the heat available calculation. Thus:

$$Q_{available} = 1,095,600 \ \frac{Gal}{day} \ x \ (8.33 \ \frac{lbm}{gal}) \ x \ (1.0 \ \frac{Btu}{lbm-^{o}F}) \ x \ (20^{o}F)$$

$$= 182.53 \ x \ 10^{6} \ \frac{Btu}{day}$$

The specified capacity of the central heat exchanger is thus compatible with the flow and temperature conditions to be expected on the Cornelia wastewater system and provides adequate capacity for serving the projected loads on the HP-WHR system.

Thermal Storage

Inspection of the wastewater flow cycle at the Cornelia treatment plant (presented in section 9.1.8) reveals that the characteristic low flow period during the seven-day flow cycle had an approximate 60 hour duration. In order to provide adequate thermal energy supply capacity for the system during these low flow periods, the thermal storage component will be sized for a nominal 72 hour period (making an allowance for a + 20% variance in duration

from the flow cycle documented in section 8.2.8) under January heating demand

conditions. The nominal 72 hour capacity will also provide an adequate down-

time "window" for the central heat exchanger for necessary scheduled maintenance

such as tube cleaning.

Allowable temperature pulldown within the vessel will be designed at 25°F.

As in the Conyers system design, the primary consideration in defining this

range is the effect on end-user heat pump performance. According to the equip-

ment assessment of Reference 7, the performance of the selected hydronic heat

pump at the upper design temperature of 65°F is very close to the nominal

C.O.P. of 2.91. Heating C.O.P. deteriorates to 2.79, however, as the lower

limit of 40°F is reached. This lower temperature also approaches what is

generally considered to be the lower acceptable limit on entering water

temperature for hydronic units.

The physical capacity of the thermal storage component is computed as

follows:

$$Q_{thermal\ storage} = \frac{(4.56 \times 10^9 \text{ Btu})(1 - \frac{1}{2.9}) \times 72 \text{ hrs}}{744 \text{ hrs}}$$

$$= 289.40 \times 10^6 \text{ Btu}$$

$$M_{water\ storage} = \frac{289.40 \times 10^6 \text{ Btu}}{(25^{\circ}F)(1.0 \frac{\text{Btu}}{\text{LBM-}^{\circ}F})}$$

$$= 11,576,009 \text{ LBM}$$

$$= 1,389,677 \text{ GALLONS}$$

$$= 185,811 \text{ FT}^3$$

This necessary capacity will be provided in the form of two rectangular

tanks with dimensions of 50' x 60' x 35' (depth). The vessels will be field

constructed of steel reinforced concrete. Installation will be partially

below grade with the exposed vertical surfaces being earth bermed for improved

heat retention.

Cooling Tower

The cooling tower component of the Conyers HP-WHR system will be specified

to dissipate the average daily heat gain on the system during the peak cooling

month of July, as determined below:

$$Q_{avg\ clg\ load} = \frac{Q_{monthly}(1 + \frac{1}{C.O.P._{end-user_{cooling}}})}{\text{DAYS IN MONTH}}$$

$$= \frac{(3.73 \times 10^9)(1 + \frac{1}{2.5*})}{31}$$

$$= 168.45 \times 10^6 \text{ Btu/day}$$

*heating C.O.P. of subject
hydronic unit = 2.91,
cooling C.O.P. = 2.53;
from Reference 7.

The required heat rejection rate will be determined on the basis of 18

hours operation per day at nominal full load, thus:

$$HRR = \frac{168.45 \times 10^6}{18}$$

$$= 9.36 \times 10^6 \text{ Btu/hr}$$

System operating parameters dictate a $10^{\circ}F$ temperature drop in the circu-

lating water. Required cooling tower flow is then determined as:

$$\dot{M}_{cooling\ tower} = \frac{9.36 \times 10^6 \frac{Btu}{hr}}{(8.33 \frac{Lbm}{gal})(60 \frac{min}{hr})(10^{\circ}F)(1.0 \frac{Btu}{Lbm-^{\circ}F}}$$

$$= 1873 \text{ GPM}$$

A modular, pre-constructed tower with a nominal flow rating of 2000 GPM at the required 10°F TD will be selected. Fan horsepower will be 50 HP.

Auxiliary Pumps

The three pump functions on the HP-WHR system will be specified as discussed below:

Wastewater pumps will be variable speed types, each capable of delivering 2/3 of required flow at design pressure drop. The flow rate required for each pump is 1400 GPM at an estimated dynamic head of 50 ft. of water. This head rating makes adequate allowance for piping and central heat exchanger pressure drops.

Circulating pumps will each be sized for one half of the total required flow of 4200 GPM. Pump head required will be approximately 80 ft. of water.

Distribution pumps will each be sized for one half of the total required flow to the end-users. One pump will be a variable speed type. The total flow through the distribution system will be based on serving the heat dissipation requirements at peak conditions with a 10°F temperature rise. Based on a nominal heat rejection requirement of 16,740 Btu per ton of air conditioning, the required flow rate for a 1250 ton peak load is 4187 GPM. (Heating season peak flow requirement is approximately 2400 GPM.) Total pumping head at peak flow will be 325 ft. of water. This head includes allowances for end-user system and distribution system piping pressure drops.

DISTRIBUTION PIPING

The required circulating water flow rates at peak load will determine distribution pipeline sizing. Peak heating and cooling loads for each zone

are first determined through the procedures outlined in section 9.1.7 and utilizing the appropriate zonal data from the computer analysis. The required flow rates per zone are then calculated and piping sizes are selected to yield proper line velocities and acceptable pressure drops.

ZONE C-1

$$\text{PEAK HEATING LOAD} = \frac{[(3.488 \times 10^9)(1-\frac{1}{2.9})](53)}{(732) \qquad\qquad (24)}$$

$$= 6.89 \times 10^6 \text{ Btu}$$

$$\text{PEAK COOLING LOAD} = \frac{(332,208 \text{ ft}^2)(16,740 \text{ Btu/hr*})}{350 \text{ ft}^2/\text{PEAK A-C ton}}$$

$$= 15.89 \times 10^6 \text{ Btu}$$

*Heat Rejection Rate based on the 2.53 cooling C.O.P. of the subject hydronic unit.

$$\dot{M}_{heating} = \frac{6.89 \times 10^6}{(8.33)(60)(10)(1.0)}$$

$$= 1378 \text{ GPM}$$

$$\dot{M}_{cooling} = \frac{15.89 \times 10^6}{4998}$$

$$= 3178 \text{ GPM}$$

ZONE C-2

$$\text{PEAK HEATING LOAD} = \frac{[(0.3675 \times 10^9)(1-\frac{1}{2.9})](53)}{(732) \qquad\qquad (24)}$$

$$= 0.73 \times 10^6 \text{ Btu}$$

$$\text{PEAK COOLING LOAD} = \frac{(35,000)(16,740)}{(350)}$$

$$= 1.67 \times 10^6 \text{ Btu}$$

$$\dot{M}_{heating} = \frac{0.73 \times 10^6}{4998}$$

$$= 146 \text{ GPM}$$

$$\dot{M}_{cooling} = \frac{1.67 \times 10^6}{4998}$$

$$= 334 \text{ GPM}$$

ZONE C-3

$$\text{PEAK HEATING LOAD} = \frac{[(0.708 \times 10^9)(1 - \frac{1}{2.9})](53)}{(732)} \frac{}{(24)}$$

$$= 1.40 \times 10^6 \text{ Btu}$$

$$\text{PEAK COOLING LOAD} = \frac{(67,500)(16,740)}{350}$$

$$= 3.23 \times 10^6 \text{ Btu}$$

$$\dot{M}_{heating} = \frac{1.40 \times 10^6}{4998}$$

$$= 280 \text{ GPM}$$

$$\dot{M}_{cooling} = \frac{3.23 \times 10^6}{4998}$$

$$= 646 \text{ GPM}$$

ZONE I-4

PEAK HEATING LOAD $= 0.0$

PEAK COOLING LOAD $= 230.0 \times 10^9$ Btu

$$\dot{M}_{cooling} = 150 \text{ GPM}$$

Obviously, the circulating water flow rates into each zone during the cooling mode will be the governing criteria in distribution pipe size selection. Pipe sizes and the corresponding flow rates, velocities, and

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pressure drops for each distribution system piping segment are presented in Table 9-7.

TABLE 9-7
DISTRIBUTION PIPING SPECIFICATIONS

LOCATION	FLOW RATE (GPM)	PIPE SIZE (INCHES)	VELOCITY FPS	PRESSURE DROP (PSI/100 FT)
Trunk Line	4309	18"	5.43	.20
Zone C-1 Trunk	3179	14"	6.63	.35
Zones C-2 & C-3 Trunk	980	8"	6.26	.60
Zone C-3 Feeder	646	8"	4.12	.30
Zone C-2 Feeder	334	5"	5.46	.85
Zone I-4 Feeder	150	4"	3.83	.55

9.2.2 SYSTEM COMPONENT DESCRIPTIONS

The components used in the Heat Pump Wastewater Heat Recovery Scheme for Cornelia are essentially identical to those used in the Conyers system. Sizing and capacity have of course been adjusted for the Cornelia application. One notable exception is the substitution of a heat exchanger for the central heat pump. The central heat exchanger will be a single shell-and-tube type unit. Water box covers should be of the "marine" type so that tube cleaning and replacement will be less difficult. The tubes should be seamless copper and individually replacable. The heat exchanger should be constructed in accordance with the A.S.M.E. Code for Unfired Pressure Vessels.

9.2.3 EXPECTED PERFORMANCE

A performance analysis on the Cornelia HP-WHR system was performed follow-
ing the same procedures and utilizing the same basic assumptions (with one excep-
tion, as noted below) as for the previous Conyers system analysis (presented
in section 8.3.3). All energy inputs to the system were tallied, including
pump work, cooling tower fan work, system auxiliaries power, and end-user heat
pump power. That total (in Btu equivalents) was then compared to the net heat-
ing or cooling effect provided to the structures within the system service area.
Estimating the monthly performance in this manner revealed system performance
in the peak heating, peak cooling, and simultaneous heating and cooling modes.

Since the Cornelia system, when operating in the heating mode, utilizes
a central heat exchanger rather than a central heat pump to extract heat from
the wastewater stream, there will necessarily be a change in the circulating
water conditions during the heating season. Cooling mode conditions will be
similar to those in effect on the Conyers system. Conditions are as noted
below:

Distribution Media Temperatures

	HEATING MODE	COOLING MODE
Supply	65°F	80°F
Return	55°F	90°F

The results of the monthly performance analysis are summarized in Table
9-8 below. The full analysis and calculational procedures are presented in
Appendix D.

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TABLE 9-8
CORNELIA HP-WHR SYSTEM PERFORMANCE

	Heating Btu Delivered to Space X 10^9	Cooling Btu Delivered to Space X 10^9	Central Plant Energy Input KWH	End-User Energy Input KWH	System C.O.P.
JAN.	4.6	0	25,906	449,262	2.836
FEB.	3.8	0	19,485	371,129	2.850
MARCH	3.1	.14	14,512	319,171	2.845
APRIL	1.2	.23	4,822	144,154	2.794
MAY	.37	1.30	25,374	188,494	2.287
JUNE	.06	2.80	57,721	334,017	2.139
JULY	0	3.70	77,660	433,636	2.120
AUG.	0	3.60	102,621	421,916	2.011
SEPT.	.17	.92	25,394	124,426	2.132
OCT.	1.20	.37	4,258	160,562	2.791
NOV.	2.90	0	14,219	283,230	2.857
DEC.	4.50	0	23,502	439,496	2.848
TOTAL	21.9	13.06	396,474	3,669,493	

As Table 9-8 indicates, heating season performance (C.O.P.) is better than would be expected with air-to-air heat pumps, and cooling system performance is comparable to air-to-air heat pumps or standard electric air-conditioning systems.

With the monthly system performance thus defined for each month during the annual cycle, the overall system C.O.P. can be easily derived. Considering the system as a "black-box," its performance can be determined by the expression

$$C.O.P. = \frac{Q_{out}}{W_{in} \times 3413}$$

where

Q_{out} =   heating or cooling effect provided by the system, Btu

W_{in} =   work input to the system, KWH

Thus, substituting from Table 8-6:

$$C.O.P. = \frac{(21.9 \times 10^9 + 13.06 \times 10^9)}{(396,474 + 3,669,493) \times 3413}$$

$$= 2.519$$

COMPONENT ENERGY USE

In order to identify the areas of energy usage on the HP-WHR system which would benefit system performance the most through decreased consumption, the performance analysis also included a breakdown of individual component energy consumption. Overall central plant power consumption was first analyzed and plotted as illustrated in Table 9-9 and Figure 9-11. Power use within the central plant was then further disaggregated according to individual components, as summarized in Table 9-10.

As with the Conyers system, the largest energy user on the system is the collective end-user heat pump. Therefore, hydronic units with improved C.O.P.'s, several of which are under development, offer the prospect of significantly reduced energy consumption in this area of system performance improvement.

Benefits similar to those identified for the Conyers system can also be gained through the judicious selection of energy efficient pumps and motors and through the development of innovative, energy conscious operating procedures.

TABLE 9-9

ELECTRICITY USAGE ANALYSIS
FOR CORNELIA HP-ICES
IN COOPERATION WITH THE TECHNOLOGY DEVELOPMENT LAB
GEORGIA TECH ENGINEERING EXPERIMENT STATION
ATLANTA, GEORGIA 30332

INPUT DATA

BILLING PERIOD	ACTUAL DEMAND	BILLING DEMAND	CONSUMPTION	FUEL ADJUSTMENT	LOAD FACTOR
JANUARY,1979	618	570	25906	.000	.06
FEBRUARY,1979	618	570	19485	.000	.05
MARCH,1979	618	570	14512	.000	.03
APRIL,1979	618	570	5822	.000	.01
MAY,1979	600	570	25374	.000	.06
JUNE,1979	655	600	57721	.000	.12
JULY,1979	600	600	77660	.000	.17
AUGUST,1979	600	600	102621	.000	.23
SEPTEMBER,1979	600	600	25394	.000	.06
OCTOBER,1979	618	570	4258	.000	.01
NOVEMBER,1979	618	570	14219	.000	.03
DECEMBER,1979	618	570	23502	.000	.05

LOCAL TAX IS $ .03

PROJECTED ELECTRICITY COST

BILLING PERIOD	BILLING AMOUNT
JANUARY	1413.22
FEBRUARY	1097.98
MARCH	853.83
APRIL	379.86
MAY	1387.10
JUNE	2975.18
JULY	3954.08
AUGUST	5179.54
SEPTEMBER	1388.08
OCTOBER	285.36
NOVEMBER	839.45
DECEMBER	1295.20
TOTAL	$ 21048.89

PLOT OF CONSUMPTION
FIG. 9-11
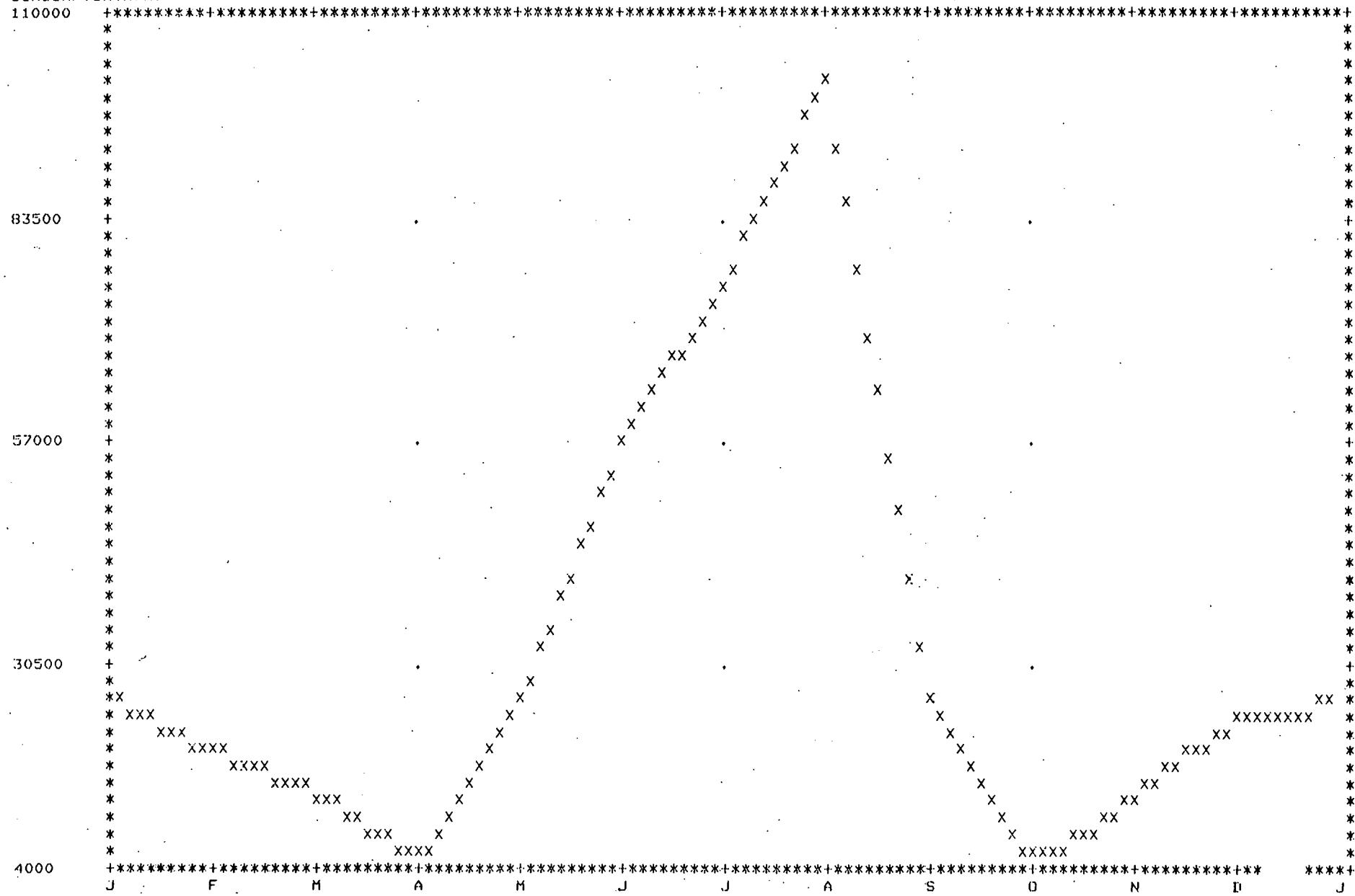
TABLE 9-10
CORNELIA HP-WHR SYSTEM
ANNUAL COMPONENT ENERGY USE

Item	Energy Use KWH	Percent Of Central Plant	Percent Of System
Distribution Pumps	149,540	37.7	3.7
Circulating Pumps	141,239	35.6	3.5
Sewage Pumps	25,192	6.4	0.6
Cooling Tower	70,903	17.0	1.7
Misc. Electrical	9,600	2.4	0.2
Central Plant Total	396,474	100.0	
End-Use Heat Pumps	3,669,493		90.3
System Total	4,065,967		100.0

ENERGY CONSERVATION POTENTIAL

The Heat Pump-Wastewater Heat Recovery System shows the potential to favorably alter the existing energy use patterns within a community and to promote the overall conservation of primary energy sources. In the case of the Cornelia community, the existing energy use pattern shows a reliance on electric energy for cooling and natural gas and electric energy for heating.

The energy consumed on-site by the conventional space conditioning systems in meeting the heating and cooling requirements projected by the computerized load estimate was first calculated. The assumptions were that heating require-ments were met with natural gas fired furnaces operating at a 65% thermal

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efficiency or with electric resistance heaters and that cooling requirements were met with electrically driven air-conditioning units with C.O.P.'s of 2.2. Each commercial service zone was treated independently since their source of energy for heating differs. Zone C-1 employs approximately 60 percent natural gas heating and 40 percent electric resistance heating. Zones C-2 and C-3 utilize electric resistance type heaters. Zone I-4, the industrial application, is assumed to require the same quantity of energy under either the conventional system scheme or the HP-WHR scheme.

Based on the above assumptions and observations, the conventional systems consume annually the following amount of energy:

	Cooling	Heating
Zone C-1	1,331,000 KWH	2,490,000 KWH + 130,769 THERMS
Zone C-2	146,498 KWH	527,395 KWH
Zone C-3	266,361 KWH	996,191 KWH
Zone I-4	35,000 KWH	-------
TOTAL	1,778,859 KWH	4,913,586 KWH + 130,769 THERMS

The total HP-WHR system energy requirement of 4,065,967 KWH (from Table 9-10) compares favorably with the above total of 4,792,445 KWH plus 130,769 therms for conventional system on-site consumption. It is noted that the comparison reflects a 15% reduction in electric energy consumption plus 100% reduction in natural gas consumption. These reductions are attributable to the effective displacement of electric resistance heating and natural gas heating by the heat pump system.

In order to compare the projected primary energy resource consumption of

the HP-WHR system to that of the conventional space conditioning systems, the procedures for determining annual fuel and resource energy utilization, as outlined in ASHRAE Standard 90-75, ENERGY CONSERVATION IN NEW BUILDING DESIGN,[58] were followed. As can be seen from the results of this assessment as illustrated on the following pages, the primary effect of HP-WHR system operation would be to conserve approximately 84% of the natural gas which would have been utilized in the Cornelia service area. Due to the reduction in overall electric energy requirements, an across-the-board reduction in the consumption of coal, crude oil, nuclear, and hydro power will also be noticed. A secondary, but desirable, effect would be to free the additional quantities of natural gas for use in satisfying other, less flexible energy requirements.

Line	Column	A1	A2	B1	B2	B3	B4	B5	B6	B7	B8	B9
		Building/Project Energy Req'ments		Fuel and Energy Supplied to Site								
	Function	Thermal 10⁶ Btu	Electric 10³ KWH	Coal 10⁶ Btu	Gas 10⁶ Btu	Light Oil 10⁶ Btu	Heavy Oil 10⁶ Btu	Elec. Win. 10³ KWH	Elec. Sum. 10³ KWH	Elec. Ann. 10³ KWH	Other	Oth
1	Heating		▨		13,077			4013.5	▨			
2	Cooling		▨					▨	1778.8			
3	Water Heating		▨									
4	HVAC Auxiliaries											
5	Lighting	▨										
6	Elevators	▨										
7	Computers	▨										
8	Cooking											
9	Process											
10	Other											
11	Other											
12	Other											
13	Total Carry Fwd. to Form 12-2				13,077			4013.5	1778.8			

CORNELIA–CONVENTIONAL SYSTEMS

Line	Fuel and Energy Supplied to Site			C.O. Total From Form 12-1 Line 13	RUF From Supplier or From Tables	C1 S. Tons Coal	C2 MCF Nat'l	C3 BBL Crude Oil	C4 Grams U-235	C5 10³ KWH Hydro	C6 Other	C7 Other
14	Fuel Oil, Light											
15	Fuel Oil, Heavy											
16	Gas	Nat'l	MCF	13,077	1.16		15,169					
		Oil	BBL	▨								
17	Coal											
18	Elec. Winter			4013.5	▨							
		Coal	S. Tons	▨	.27	1083.6						
		Gas	MCF	▨	.56		2247					
		Oil	BBL	▨	.53			2127				
		Nuc	Grams	▨	3.84				15,411			
		Hydro	10³ KWH	▨	.07					280.9		
		Other										
19	Elec. Summer			1778.8	▨							
		Coal	S. Tons	▨	.26	462.5						
		Gas	MCF	▨	1.04		1850					
		Oil	BBL	▨	.56			996				
		Nuc	Grams	▨	3.84				6830			
		Hydro	10³ KWH	▨	.04					71.1		
		Other										
20	Elec. Annual			▨	▨							
		Coal	S. Tons	▨								
		Gas	MCF	▨								
		Oil	BBL	▨								
		Nuc	Grams	▨								
		Hydro	10³ KWH	▨								
		Other										
21	(Other)											
22					Total Resources	1546.1	19,266	3123	22,241	352		

ANNUAL FUEL AND ENERGY CALCULATION FORM 12-1

Line	Column	A1	A2	B1	B2	B3	B4	B5	B6	B7	B8	B9
		Building/Project Energy Req'ments		Fuel and Energy Supplied to Site								
	Function	Thermal 10⁶ Btu	Electric 10³ KWH	Coal 10⁶ Btu	Gas 10⁶ Btu	Light Oil 10⁶ Btu	Heavy Oil 10⁶ Btu	Elec. Win. 10³ KWH	Elec. Sum. 10³ KWH	Elec. Ann. 10³ KWH	Other	Other
1	Heating	21,900						2274.7				
2	Cooling	13,060							1791.3			
3	Water Heating											
4	HVAC Auxiliaries											
5	Lighting											
6	Elevators											
7	Computers											
8	Cooking											
9	Process											
10	Other											
11	Other											
12	Other											
13	Total Carry Fwd. to Form 12-2								2274.7	1791.3		

CORNELIA—HP-WHR SYSTEM

ANNUAL FUEL AND ENERGY CALCULATION FORM 12-2

Line	Fuel and Energy Supplied to Site			C.O. Total From Form 12-1 Line 13	RUF From Supplier or From Tables	C1 S. Tons Coal	C2 MCF Nat'l	C3 BBL Crude Oil	C4 Grams U-235	C5 10³ KWH Hydro	C6 Other	C7 Other
	Annual Fuel and Energy Calculation Form 12-2					Fuel and Energy Resources Used on Site and Off Site To Meet Energy Requirements of Building/Project						
14	Fuel Oil, Light											
15	Fuel Oil, Heavy											
16	Gas	Nat'l	MCF	–0–	1.16							
		Oil	BBL									
17	Coal											
18	Elec. Winter			2274.7								
		Coal	S. Tons		.27	614.2						
		Gas	MCF		.56		1274					
		Oil	BBL		.53			1205.6				
		Nuc	Grams		3.84				8735			
		Hydro	10³ KWH		.07					159.2		
		Other										
19	Elec. Summer			1791.3								
		Coal	S. Tons		.26	465.7						
		Gas	MCF		1.04		1863					
		Oil	BBL		.56			1003.1				
		Nuc	Grams		3.84				6879			
		Hydro	10³ KWH		.04					71.7		
		Other										
20	Elec. Annual											
		Coal	S. Tons									
		Gas	MCF									
		Oil	BBL									
		Nuc	Grams									
		Hydro	10³ KWH									
		Other										
21	(Other)											
22			Total Resources			1079.9	3137	2208.7	15,614	230.9		

SECOND LAW PERFORMANCE ANALYSIS

A second law performance analysis was performed for the Cornelia HP-WHR system during the peak heating season, peak cooling season and a period when both heating and cooling was required. The analysis was conducted on a monthly basis.

The same set of assumptions used in the Conyers study are applied to the Cornelia analysis. The results of the analysis are tabulated below.

CORNELIA HP-WHR SYSTEM
SECOND LAW ANALYSIS

SEASON	HEATING SUPPLIED	COOLING SUPPLIED	AVAILABILITY OF HEATING & COOLING SUPPLIED	HP-ICES AVAILABILITY OF ENERGY SUPPLIED TO SYSTEM	SECOND LAW EFFICIENCY	CONVENTIONAL AVAILABILITY OF ENERGY SUPPLIED TO SYSTEM	SECOND LAW EFFICIENCY
January (Heating)	4.6×10^9	-0-	1.832×10^8	4.646×10^9	.03944	7.07×10^9	.02591
July (Cooling)	-0-	3.7×10^9	7.092×10^7	4.484×10^9	.01581	5.09×10^9	.01393
October (Mixed)	1.2×10^9	$.37 \times 10^9$	5.489×10^7	1.66×10^9	.0331	2.355×10^9	.0233

EXPECTED ECONOMICS

This section evaluates the expected economic performance of the HP-WHR

system as compared to conventional heating and cooling systems in the Cornelia

area. First costs, operating costs and maintenance costs for each of the

compared systems are estimated in order to determine annualized costs to con-

sumers. A net present value comparison for the system is also made.

System design and performance has been presented in previous sections of

this report. For the purposes of this analysis it is assumed that some form

of government entity will own and operate the central utility and distribution

system. End-users will own and operate their own heat pump systems and pur-

chase thermal energy from the central plant. The purchase price for the thermal

energy will be determined by the cost to produce and deliver the energy plus

maintenance costs, personnel costs, administrative costs, and debt service.

It is anticipated that the thermal utility would provide distribution and

metering equipment to the property line of the customer. Piping and equipment

installed on the user's property would be paid for and become the property of

that user.

Debt service for the central plant was based on a 7 percent per year

interest rate with payments over a 25 year period. This interest rate appeared

to be in the high-end of a range that would be expected for Cornelia.

The following cost analysis is broken into three parts: first costs,

operating costs, life cycle costs. A summary of results is presented in the

latter part of the section.

FIRST COSTS

First costs are categorized as costs incurred by the central utility or

costs incurred by the end-users. In Cornelia's case, it is anticipated that

the central utility's costs will be those associated with the central plant, distribution system and metering. End-users will be responsible for the retrofit of their own HVAC system and for piping from their heat pump unit to the meter connection.

Central plant first costs were based on the equipment sizes and types presented in the system design section. A detailed cost breakdown is presented in Appendix D. A summary of these costs follows:

Central Plant	$ 848,283
Distribution System	$ 275,020
Metering Equipment (100 - 2" meters, including installation)	$ 30,000
Total	$1,153,303

End-user system costs were computed on the basis of 100 ft. of 2" pipe per end-user to connect to the distribution system and $800 per installed ton of air conditioning capacity. The $800 per ton is assumed to include any salvage value of equipment removed from the end-user buildings. For Cornelia the retrofit cost is approximately:

1450 tons installed x $800

Total cost of retrofit	=	$1,160,000
Piping: 100 users x 100 ft. x $4.00 (installed)	=	$ 40,000
Total retrofit cost	=	$1,200,000

The retrofit costs are based on the individual prices each end-user would be required to pay if the work were contracted on an individual basis.

OPERATING COSTS

Operating costs were based on the results of the performance analysis

presented in section 9.2.3. The cost of electric power for the central plant

was computed from Georgia Power Company's PL-1 rate schedule. Power for end-

user systems was assumed to cost 5.75¢/KWH. The cost of thermal energy delivered

to end-users by the HP-WHR system will be discussed in a subsequent section.

From Table 9-9 of the performance analysis, the annualized cost for power to

the central plant will be approximately $21,050. End-users on the HP-WHR

system will use approximately 3,669,500 KWH per year. This electricity will

cost approximately $210,996 under individual billings.

Maintenance costs for the HP-WHR central station are estimated at 5 per-

cent per year of the cost of the station and 0.5 percent per year for the

distribution system. These maintenance costs are:

Item	Annual Maintenance Cost
Central Station Equipment	$42,414
Distribution System & Meters	$ 1,447
Total	$43,861

End-user maintenance costs were assumed to be $10 higher per installed

ton of air conditioning capacity than would comparable system maintenance on

conventional systems. This figure includes a provision for replacement costs

of units in order to perpetuate the system.

$$1450 \times \$10 = \$14,500 \text{ per year}$$

HP-WHR SYSTEM OPERATION AS A THERMAL UTILITY

In order to compare the HP-WHR scheme to conventional utility 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 anticipa-

ted that the system will be operated as a thermal utility. The charges to the

end-users will be 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 Cornelia system using the stated electric energy costs, an administrative cost of 10 percent of total owning and operating costs, and debt service based on an interest rate of 7 percent per year over 25 years. It is recognized that the seven percent interest rate is somewhat arbitrary and will vary with the type of financing arrangements actually implemented; however, it was selected as a compromise between typical 1977 yields for Municipal Class A bonds and Utility bonds issued by private companies.

The total annual cost to the end-users for thermal service was computed as follows:

<div align="center">

CORNELIA - HP-WHR SYSTEM
THERMAL UTILITY CHARGES
(COMPONENT COSTS IN 1979 DOLLARS)

</div>

Energy Cost	$ 21,050
Maintenance on Central Plant and Distribution System	$ 43,860
Debt Service on Total System Cost (7%, 25 years) ($1,153,303 x .0858)	$ 98,955
Subtotal	$163,865
10% Administrative fee (including Insurance and Contingencies)	$ 16,385
Total	$180,250

Based on the above total, the necessary cash flow for the system will be as projected in Table 9-11.

TABLE 9-11
CORNELIA HP-WHR SYSTEM
THERMAL UTILITY CHARGES
CASH FLOW

THE FOLLOWING ASSUMPTIONS WERE USED
ANALYSIS PERIOD OF 25 YEARS INTEREST RATE OF   .0 PERCENT
SAL/COST ESCALATION=.000 MAINT. ESCALATION= .080
AVERAGE UTILITY ESCALATION:E=.110 INCOME ESCALATION=.000

YR	DEBT S./ OR COST	ANNUAL:COSTS MAIN,ETC	COSTS UTILITIES	ADMIN.	D FACT	COST.
0	98955.	0.	0.	0.	*1.000	-98955.
1	98955.	47369.	23365.	16969.	*1.000	186658.
2	98955.	51158.	25936.	17605.	*1.000	193654.
3	98955.	55251.	28789.	18299.	*1.000	201294.
4	98955.	59671.	31955.	19058.	*1.000	209640.
5	98955.	64445.	35470.	19887.	*1.000	218757.
6	98955.	69600.	39372.	20793.	*1.000	229720.
7	98955.	75168.	43703.	21783.	*1.000	239609.
8	98955.	81182.	48511.	22865.	*1.000	251512.
9	98955.	87676.	53847.	24048.	*1.000	264526.
10	98955.	94690.	59770.	25342.	*1.000	278757.
11	98955.	102266.	66344.	26757.	*1.000	294322.
12	98955.	110447.	73642.	28304.	*1.000	311349.
13	98955.	119283.	81743.	29998.	*1.000	329979.
14	98955.	128825.	90735.	31852.	*1.000	350367.
15	98955.	139131.	100716.	33880.	*1.000	372682.
16	98955.	150262.	111794.	36101.	*1.000	397112.
17	98955.	162283.	124092.	38533.	*1.000	423862.
18	98955.	175265.	137742.	41196.	*1.000	453158.
19	98955.	189287.	152893.	44114.	*1.000	485249.
20	98955.	204430.	169712.	47310.	*1.000	520406.
21	98955.	220784.	188380.	50812.	*1.000	558931.
22	98955.	238447.	209102.	54650.	*1.000	601154.
23	98955.	257522.	232103.	58858.	*1.000	647438.
24	98955.	278124.	257634.	63471.	*1.000	698185.
25	98955.	300374.	285974.	68530.	*1.000	753833.

LIFE CYCLE COSTS

A life cycle cost analysis was prepared for the Cornelia HP-WHR system and the existing conventional systems. The analysis was performed from the end-user point of view, since the consumer will ulitmately 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.

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

$$PV_i = FC_i + \sum_{y=1}^{25} [f(j,y) \times (ADC_i + AMC_i)]$$

Where:

PV_i = Present value of the ith alternative

FC_i = First cost of the ith alternative

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

ADC_i = Annual operating costs of the ith alternative

AMC_i = Annual maintenance costs of the ith alternative

The results of the life cycle cost analysis are presented in Table 9-12 for the conventional systems as the base case and Table 9-13 for the HP-WHR system 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.

TABLE 9-12
CORNELIA CONVENTIONAL SYSTEMS (BASE CASE)
LIFE CYCLE COST ANALYSIS

	YR	DEBT SERVICE OR COST	BASE MAINTENANCE	UTILITY COSTS N.GAS	ELECT.	TOTAL COSTS	DISCOUNT FACTOR	DISCOUNTED COSTS
	0	0.	0.	0.	0.	0.	1.000	0.
1980 =	1	0.	0.	48,025.	369,702.	417,727.	.909	379,714.
	2	0.	0.	54,268.	410,369.	464,637.	.826	383,790.
	3	0.	0.	61,323.	455,510.	516,833.	.751	388,142.
	4	0.	0.	69,295.	505,616.	574,911.	.683	392,664.
	5	0.	0.	78,303.	561,234.	639,537.	.621	397,152.
	6	0.	0.	88,483.	622,970.	711,453.	.564	401,259.
	7	0.	0.	99,986.	691,496.	791,482.	.513	406,030.
	8	0.	0.	112,984.	767,561.	880,545.	.467	411,215.
	9	0.	0.	127,672.	851,993.	979,665.	.424	415,378.
	10	0.	0.	144,269.	945,712.	1,089,981.	.386	420,733.
	11	0.	0.	163,024.	1,049,740.	1,212,764.	.350	424,467.
	12	0.	0.	184,217.	1,165,211.	1,349,428.	.319	430,468.
	13	0.	0.	208,165.	1,293,385.	1,501,550.	.290	435,449.
	14	0.	0.	235,227.	1,435,657.	1,670,884.	.263	439,442.
	15	0.	0.	265,806.	1,593,579.	1,859,385.	.239	444,393.
	16	0.	0.	300,361.	1,768,873.	2,069,234.	.218	451,093.
	17	0.	0.	339,408.	1,963,449.	2,302,857.	.198	455,966.
	18	0.	0.	383,531.	2,179,428.	2,562,959.	.180	461,333.
	19	0.	0.	433,390.	2,419,166.	2,852,556.	.164	467,819.
	20	0.	0.	489,731.	2,685,274.	3,175,005.	.149	473,076.
	21	0.	0.	553,396.	2,980,654.	3,534,050.	.135	477,097.
	22	0.	0.	625,338.	3,308,526.	3,933,864.	.123	483,865.
	23	0.	0.	706,632.	3,672,464.	4,379,096.	.112	490,459.
	24	0.	0.	798,494.	4,076,435.	4,874,929.	.102	497,243.
	25	0.	0.	902,298.	4,524,843.	5,427,141.	.092	499,297.
TOTALS				7,473,626.	42,298,847.	49,772,473.		10,927,544.

ASSUMPTIONS:

1) Natural Gas Cost Escalation - 13%
2) Electric Cost Escalation - 11%
3) Discount Rate - 10%

TABLE 9-13
CORNELIA HP-WHR SYSTEM
LIFE CYCLE COST ANALYSIS

YR	DEBT SERVICE OR COST	ADDITIONAL MAINTENANCE	UTILITY COSTS ELECT.	THERMAL	TOTAL COSTS	DISCOUNT FACTOR	DISCOUNTED COSTS
0	$1,200,000.	0.	0.	0.	0.	1.000	$1,200,000.
1980 = 1	0.	15,660.	234,206.	186,658.	436,524.	.909	396,800.
2	0.	16,913.	259,968.	193,654.	470,535.	.826	388,662.
3	0.	18,266.	288,565.	201,294.	508,125	.751	381,602.
4	0.	19,727.	320,307.	209,640.	549,674.	.683	375,427.
5	0.	21,305.	355,541.	218,757.	595,603.	.621	369,869.
6	0.	23,010.	394,650.	228,720.	646,330.	.564	264,558.
7	0.	24,850.	438,061.	239,609	702,520.	.513	360,393.
8	0.	26,838.	486,248.	251,512.	764,598.	.467	357,067.
9	0.	28,986.	539,736.	264,526.	833,248.	.424	353,297.
10	0.	31,304.	599,106.	278,757.	909,167.	.386	350,938.
11	0.	33,809.	665,008.	294,322.	993,139.	.350	347,599.
12	0.	36,513.	738,159.	311,349.	1,085,021.	.319	346,441.
13	0.	39,435.	819,357.	329,979.	1,188,771.	.290	344,744.
14	0.	42,589.	909,486.	350,367.	1,302,442.	.263	342,542.
15	0.	45,996.	1,009,529.	372,682.	1,428,207.	.239	341,341.
16	0.	46,676.	1,120,577.	397,112.	1,567,365.	.218	341,686.
17	0.	53,650.	1,243,841.	423,862.	1,721,353.	.198	340,828.
18	0.	57,942.	1,380,663.	453,158.	1,891,763.	.180	340,517.
19	0.	62,578.	1,532,536.	485,249.	2,080,363.	.164	341,180.
20	0.	67,584.	1,701,115.	520,406.	2,289,105.	.149	341,077.
21	0.	72,991.	1,888,238.	558,931.	2,520,160.	.135	340,222.
22	0.	78,830.	2,095,944.	601,154.	2,775,928.	.123	341,439.
23	0.	85,136.	2,326,498.	647,438.	3,059,072.	.112	342,616.
24	0.	91,947.	2,582,413.	698,185.	3,372,545.	.102	344,000.
25	0.	99,303.	2,866,479.	753,833.	3,719,615.	.092	342,205.
TOTALS		1,144,838.	26,796,231.	9,471,154.	37,412,223.		10,037,049.

ASSUMPTIONS:

1) Maintenance and Replacement Cost Escalation - 8%
2) Electric Cost Escalation - 11% (3% Over Base Inflation Rate)
3) Thermal Utility Charge Escalation - 11% (From Table 9-11)
4) Discount Rate - 10%

SUMMARY OF EXPECTED ECONOMICS

The results of the life cycle economic analysis indicate that the HP-WHR system as proposed for the Cornelia community is cost competitive in the immediate market with the conventional space conditioning systems alternative. Indeed, on the basis of the results presented in Tables 9-12 and 13, implementation of the HP-WHR system would result in a net benefit to the end-users of $890,405. This favorable comparison results in part from the definition of the system service area and in part from the site-specific wastewater conditions in the community. By limiting the scope of the proposed service area to commercial installations, the type of conventional space conditioning systems which will be displaced by the HP-WHR system are limited. Specifically, in the Cornelia service zones a high percentage of the heating units which will be displaced are electric resistance types. With the HP-WHR system offering heating C.O.P.'s of approximately 2.8, the energy expenditures are greatly reduced while delivering a comparable thermal load. The relatively high wastewater temperatures experienced on the Cornelia sewage system also affect system economics by allowing the use of a direct heat exchanger during the heating season in lieu of a central heat pump. Energy expenditures for central plant power are thus reduced, and correspondingly system performance and economics are improved. It should be noted that this system, as was the Conyers HP-WHR system, is essentially cooling dominated. It is anticipated that, if the system concept were adapted for use in the colder climes, system economics would be even further improved.

9.2.5　PLAN FOR HP-WHR SYSTEM PHASING - CORNELIA, GEORGIA

An HP-ICES will operate in its community as a form of public utility
which will proivde a needed and potentially widespread (in relation to the
overall community) service. Serving in this capacity, the system is subject
to the need for long-range planning in order to provide its service in
a manner most beneficial, service-wise and cost-wise to the end-user. The
primary concern is that the system be designed and constructed to fulfill
the needs of the existing viable service areas. However, consideration
must also be given to expanding the system in the most efficient manner
and at the most advantageous time to accommodate future growth within the
community. Chaper 6.0 - PROJECTED GROWTH discusses methods of projecting
growth within a specific community and addresses construction phasing of
central plant facilities and the distribution system for ranges of growth
rate situations. The following paragraphs apply the general approaches
outlined in the earlier chapter to the Cornelia demonstration community
situation.

The initial approach of the Cornelia - based HP-WHR system will be to
provide thermal utility services to a fairly compact area of commercial
establishments (two shopping centers: one existing, one under construction)
along the newly constructed Cornelia bypass, an industrial plant in the
same general sector of town (southwest), and the central business district.
The central business district has realized its ultimate build out potential
(with regard to space availability and restrictions) and there now exists
a trend toward renovating or remodeling downtown area shops. The city's
major industrial installation, the Fieldale poultry processing plant, is
well established in their process needs with no major plant revisions or
expansions planned in the foreseeable future. The existing shopping center
at the intersection of Level Grove Road and the Cornelia bypass already
accommodates three tenants, two restaurants and one large grocery store,

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and exhibits potential for more commercial growth in the immediate future.

CENTRAL PLANT PHASING - In general, a central plant installation for a community (or service area) which expects a moderate rate of growth, 20 to 50%, during the planning period (the category into which Cornelia fits) can be handled with initial construction of the major portions of plant capacity, then minor expansions to the central facilities at appropriate times in the planning period. Minor expansions include installing larger pump motors, changing pump impellers, adding new heat exchangers, or adding new equipment in space already provided in the initial construction. The proposed central plant layout for the Cornelia system is based on the above mentioned approach. The major building structure will be constructed to accommodate all equipment which would be necessary to serve the ultimate build out potential of the system. The lines, sumps and pumps which serve on the sewer tie-in subsystem will be constructed initially to handle 100 to 110% of the ultimate flow which could be expected through the accessible section of the sewer mains by the end of the projection period. Initial equipment installation (circulating pumps, distribution pumps, thermal storage tanks, and cooling tower) will provide adequate system capacity to serve the needs of the existing service areas plus any growth projected to occur in these areas over the next five to ten years. At the time when additional capacity is required, a minor plant expansion would involve the installation of an additional circulating pump, distribution pump, cooling tower cell, and if required, thermal storage capacity. All equipment additions could be handled with little or no major modification to the in-plant piping systems or structure. Costs for system expansion within the central plant should thus be minimized.

DISTRIBUTION SYSTEM PHASING - There are several characteristics of the design and construction of high-volume water distribution (and return)

lines which will influence the manner in which such systems will be constructed and expanded. First, water and sewer and, in this case, thermal utility lines have a useful life of approximately 50 years. Secondly, a major portion of the cost of constructing such lines is not in material costs but in manpower and equipment expenditures for the construction process. Hence, the most economical way to provide additional capacity generally is to construct a larger line initially rather than a parallel line at a later date.

For the central business district, a distribution leg will be constructed initially with adequate capacity to serve all potential customers in this service area. Due to the degree of development of this area of the community, no extraordinary expansion of this commercial district is anticipated which could not be accommodated by simply allowing additional hook-ups to the system.

For the Fieldale industrial process, whose needs are well established and with little prospect of expansion in the near future, a separate distribution leg will be constructed with adequate initial capacity to provide their total requirements.

For the newly developing commercial district along the bypass, it is anticipated that fairly accurate projections of the ultimate build-out potential of the area can be made. A major consideration is that the somewhat limited availability of accessible acreage will restrict the commercial growth of the area to the moderate range.

Since it is likely that the ultimate potential would dictate only a small incremental increase in system capacity over what would be constructed to serve only the projected mid-range requirements, it is probable that a full capacity distribution leg would be constructed for this area also.

There are several residential areas peripheral to the potential commercial and industrial service areas which are not addressed in the

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initial development of the Cornelia HP-WHR system. If indeed these areas prove serviceable after the initial system is constructed and under development, they can be included in future system expansions, to the extent that thermal energy is still available from the community wastewater, with the construction of distribution trunk lines from the central plant into the potential service areas.

9.3 INSTITUTIONAL CONSIDERATIONS

9.3.1 CITY OF CORNELIA, GEORGIA REVIEW OF LOCAL ORDINANCES AND CODES

The local codes and ordinances of the City of Cornelia were reviewed to determine what impacts or restrictions they might impose on the development of a Heat Pump Centered-Integrated Community Energy System. This review indicated that there were several local building codes which would need to be reviewed in depth in conjunction with the design and construction of an HP-ICES. The local building codes which have been adopted for the City of Cornelia are as follows:

1. Southern Standard Building Code - 1970 Edition.

2. Southern Standard Plumbing Code - 1971 Edition.

Since these codes are some of the same codes used in the Conyers and Rockdale County area the same comments and discussion for Conyers would apply to Cornelia.

There were no other local ordinances or codes that would affect or restrict the construction of a Heat Pump Centered-Integrated Community Energy System within the City of Cornelia.

10.0 COMPONENT TESTING

10.1 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.

10.2 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.

10.2.1 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.

10.2.2 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 10.2.1.

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 10-1, 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 he used on pipes

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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]
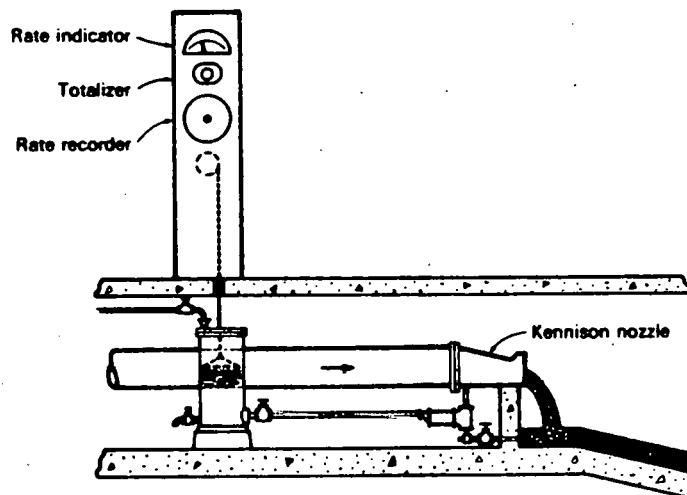
Kennison flow nozzle installation [from BIF].

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

FIGURE   10-1
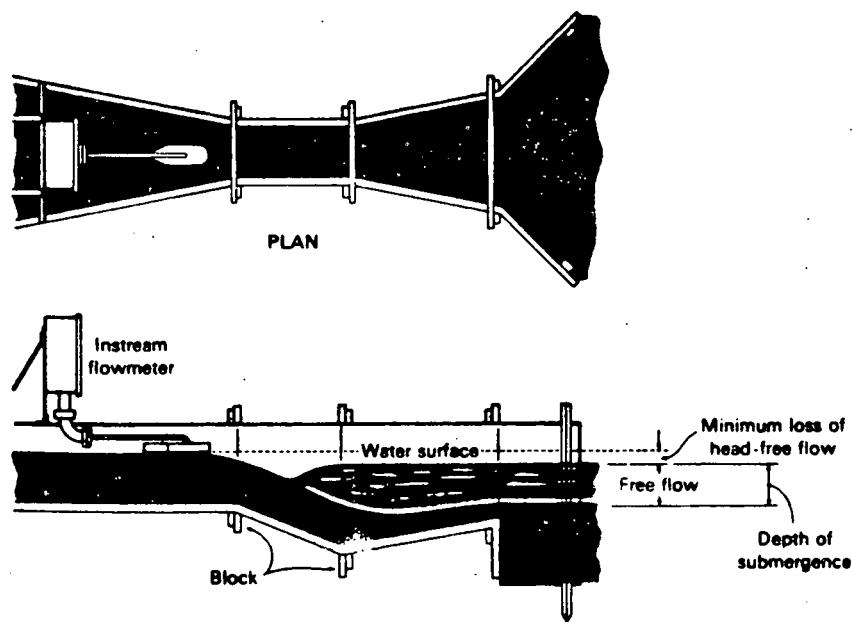
SECTION
Schematic of Parshall flume metering installation [from Fischer & Porter].

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

FIGURE   10-2

The Parshall Flume installation, pictured in Figure 10-2, 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 :

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

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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 sophisti-
cated methods.

10.2.3 SUMMARY

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

10.3 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 per-
formance 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 Sta-

tion 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.

10.3.1 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 per-

formance 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.

10.3.2 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 decompocable 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 engineer-

ing 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 mini-

mum retention times, etc. Temperature and flow data will be developed at

the clarifier as explained in sections 10.2.1 and 10.2.2 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 ex-

changer surfaces will prove to be the most troublesome problem associ-

ated 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 waste-

water to be circulated through the Central Station equipment heat exchangers

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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 10-3 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 waste-water 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.

COP = 105. - 9000 (FF)
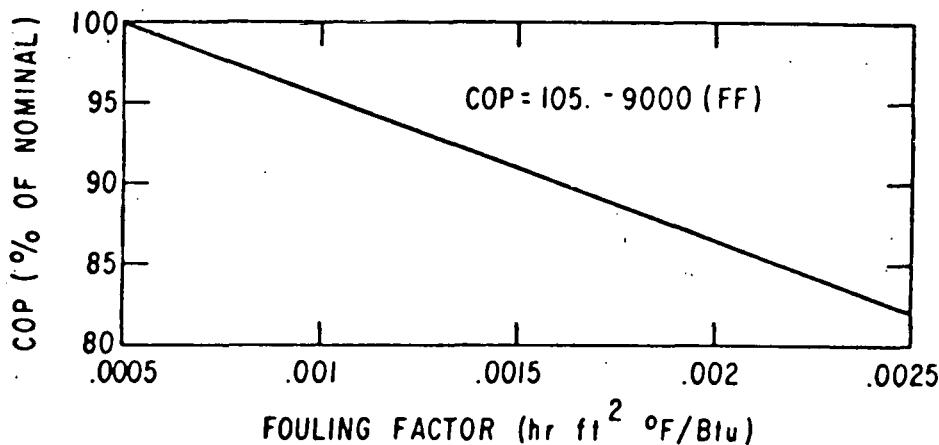
REDUCTION OF PACKAGE CHILLER C.O.P.
AS A FUNCTION OF FOULING FACTOR

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

FIGURE 10-3

10.4 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.

10.4.1 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.

.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.

10.5 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.

10.6 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.

11.0 ANALYSES DESCRIPTION

11.1 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.

11.2 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

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it will be necessary to develop a new primary data-base correlating waste-water effluents, thermal quality, and density of energy users.

11.3 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.

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.

11.4 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.

11.4.1 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

11.4.2 ASSUMPTIONS

a) The basic vapor compression cycle as described in section 11.4.3
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 11.4.9.

b) The water distribution lines are buried 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 11.4.8.

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 11.4.8.

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.

11.4.3 VAPOR COMPRESSION CYCLE

The heat pump used in the anlaysis of the HP-WHR system is based on the vapor compression refrigeration cycle (Figure 11-1). 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.
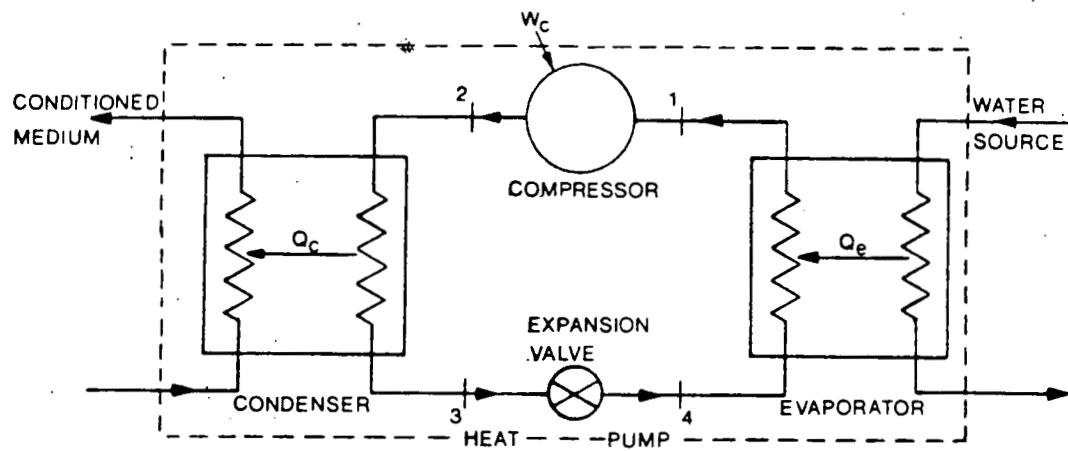
BASIC VAPOR COMPRESSION CYCLE

FIGURE II-I

The coefficient of performance of the heat pump is given by

$$\text{COP (ideal)} = \frac{Q_c}{\text{Energy input}} = \frac{h_2 - h_3}{h_2 - h_1} \tag{11-1}$$

$$\text{COP (actual)} = \frac{Q_c}{E_{th}} \tag{11-2}$$

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

$$\text{COP (refrigerator or chiller)} = \frac{Q_c}{E_{th}} \tag{11-3}$$

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

$$COP = \frac{[(HRF)E]}{[HRF-1]} \tag{11-4}$$

$$HRF = \frac{(COP)}{(COP-E)} \tag{11-5}$$

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

$$COP_{heat\ pump} = COP_{refrigerator} + E \tag{11-6}$$

11.4.4 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:

$$W_{ce} = \frac{Q_{ce}}{COP_{end-user}} \qquad (11-7)$$

Heat absorbed in the evaporator:

$$Q_{ee} = \frac{Q_{ce}}{HRF} \qquad (11-8)$$

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.

$$Q_e = \dot{m}_w \, c_p \, (T_{e1} - T_{e2}) \qquad (11-9)$$

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:

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

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:

$$W_c = \frac{Q_c \text{(central heat pump)}}{COP \text{ (central heat pump)}} \quad \text{(11-10)}$$

Total energy input to compressor:

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

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

$$= 0.05 \, \Sigma W_c \quad \text{(11-12)}$$

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:

$$COP_{system} = \frac{\text{(heating load)}}{E_{system}} \quad \text{(11-13)}$$

A similar calculation procedure is used for the cooling mode.

11.4.5 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 performance factor obtained from Reference 5.

and the heat extracted from the distribution system,

$$Q_E = \frac{Q_H}{HRF} \tag{11-14}$$

Where HRF heat rejection factor.

For the heat pump operating in the cooling mode

$$W_{c2} = \frac{Q_L}{COP_{cooling}} = \frac{Q_L}{(COP_{heating} - E)} \tag{11-15}$$

and the heat rejected $Q_c = Q_L \times HRF$ (11-16)

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

$$Q_L \, HRF = \frac{Q_H}{HRF_2} \tag{11-17}$$

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

$$\frac{Q_H}{Q_L} = HRF^2 \tag{11-18}$$

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

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

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

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HRF $\quad$ = 1.456

COP $\quad$ = 2.96, $\quad$ COP$_{\text{cooling mode}}$ $\quad$ = 2.04
heating mode

1) Q_H $\quad$ = 55,000 Btuh heating

2) Q_L $\quad$ = 25,943 Btuh cooling

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

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

TOTAL ΣW = 31298 Btuh

and heat delivered (in the condenser) to space = 37613 x HRF$_2$ = 55000

Total load = 55000 + 25943 = 80943

$\quad$ Total ΣW_c = 31,298 + 0.05 ΣW_c = 32862

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

11.4.6   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 11-2.
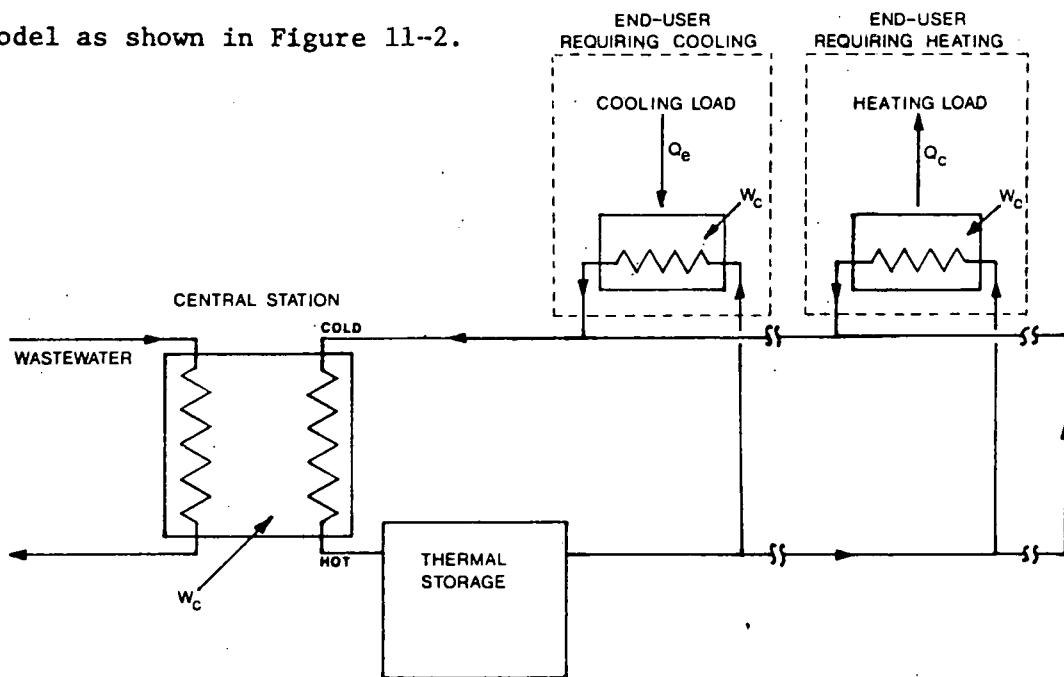
MODEL FOR SIMULTANEOUS HEATING AND COOLING
FIGURE II-2

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 11.4.3, we have

$$W_{c1} = \frac{Q_H}{COP}$$

end-user pump

$$(11-19)$$

11.4.7 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:

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

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.

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 con-

sequent 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 temper-

ature 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 con-

sideration. The thermal conductivity values taken from Reference 46 are

given in the TABLE 11-1 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:

$$Q = K_p(T_p - T_g), \frac{Btu}{hr\text{-}ft.} \tag{11-21}$$

Where

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

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$$T_p = \text{temperature of fluid in the pipe}$$

$$T_g = \text{undisturbed earth temperature}$$

K_p is given by

$$R = \frac{1}{K_p} = \frac{1}{C} + \frac{6}{\pi K_s} \ln \left\{ \frac{d}{r} + \left(\frac{d}{r}\right)^2 - 1 \right\} \qquad (11\text{-}22)$$

C = thermal conductance of pipe $\left(\frac{Btu}{hr\text{-}ft^2{}^\circ F}\right)$

When

d = distance of ground surface to center of pipe (ft.)

r_o = outside radius of pipe (inches)

K_s = thermal conductivity of soil $\left(\frac{Btu\text{-}inch}{hr.ft^2\text{-}^\circ F}\right)$

and

$$\frac{1}{C} = \frac{1}{h_i 2\pi r_i} + \frac{6}{\pi K_w} \ln \frac{r_o}{r_i} \qquad (11\text{-}23)$$

Where

r_i = inside radius of pipe (ft)

K_w = thermal conductivity of pipe material$\left(\frac{Btu\text{-}inch}{hr\text{-}ft^2\text{-}^\circ F}\right)$

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

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

TABLE 11-1

THERMAL CONDUCTIVITY OF SOIL

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

	Type Soil		
Moisture Content	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]

for water at 80°F was found to be $35.5 \frac{Btu}{hr \cdot ft}$. For dry and sandy soil, the heat loss would be $9.5 \frac{Btu}{hr \cdot ft}$

Temperature drop in pipe lines:

Heat loss $Q = \dot{m} C_{pw}(T_2 - T_1)$ (11-24)

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 $(35.5 \times 5280) \frac{Btu}{hr}$ 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. 11-3

$$\dot{Q} = \dot{m}(h_2 - h_1) + \dot{M}\frac{dE}{d\theta}$$ (11-25)

Where

E = the internal energy of water in the reservoir

= $C_v(T)$

(symbol legend cont'd)

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Θ = time

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

C_v = specific heat of water = $1.0 \dfrac{Btu}{lbm-^{o}R}$

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, $\dfrac{dE}{d\Theta}$ can be studied by neglecting $\dot{m}(h_2-h_1)$ for the given instant. Then

$$Q = UA(T-T_o) = M \frac{dE}{d\Theta} \tag{11-26}$$

Where

T_o is the temperature of surroundings

Assuming a "U" value of $0.3 \dfrac{Btu}{hr.ft^2-^{o}F}$ (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

$$\frac{dT}{d\Theta} = -0.0203 \ \frac{^{o}F}{hr}, \text{ since } C_v = 1.0$$

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.
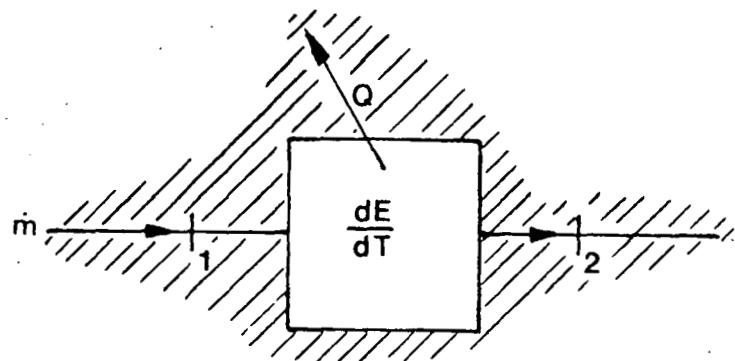
BLOCK DIAGRAM FOR THERMAL STORAGE ANALYSIS

FIGURE II-3

¹1.4.8 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 11-4 and 11-5. 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)

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

$$- 32.7860\,(LCWT) + 0.378447\,(LCWT^2) - 0.00142857\,(LCWT^3)$$

$$+ \frac{1092.31}{LEWT} + \frac{2071.02}{LEWT^2} \tag{11-27}$$

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.

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
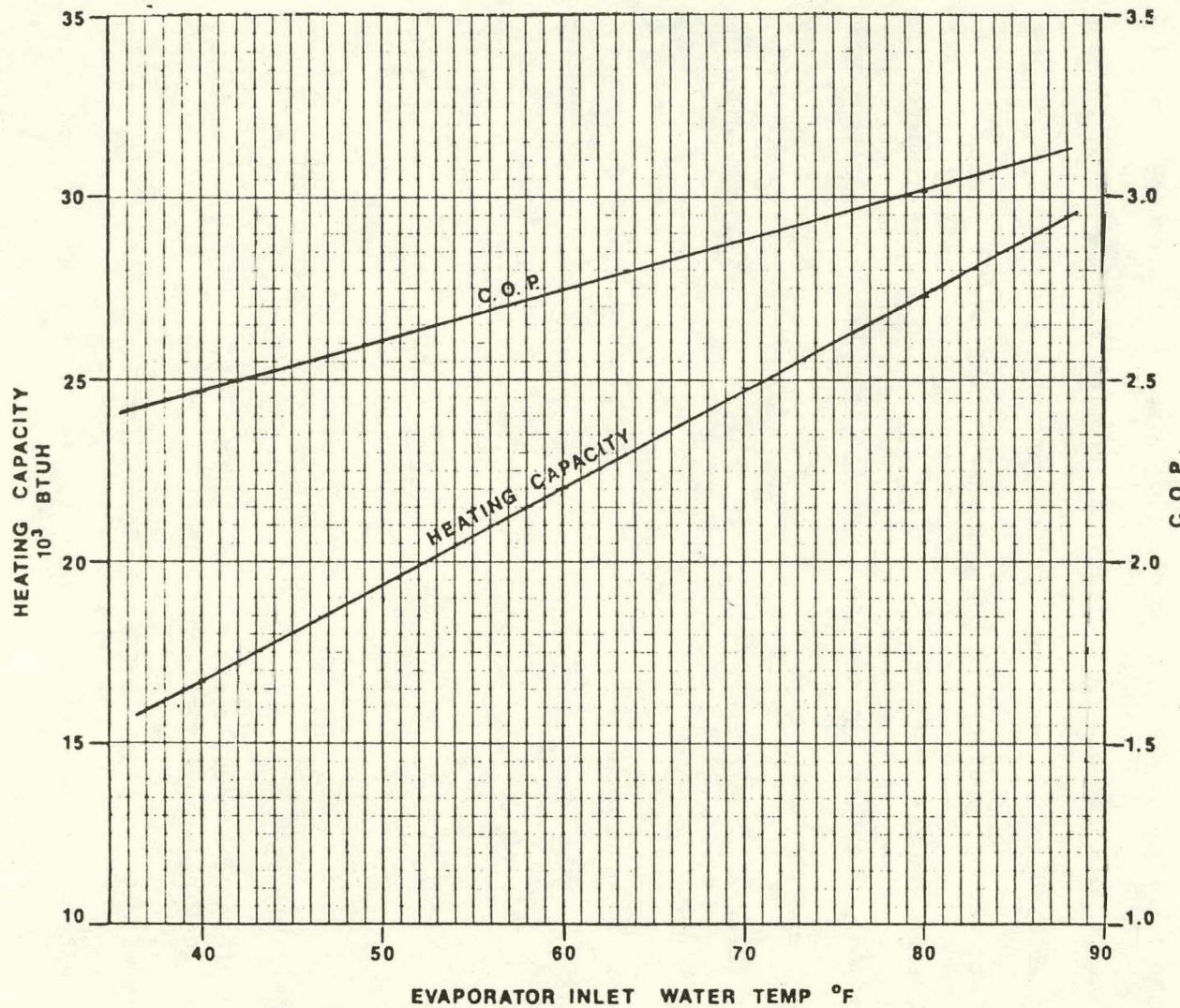
NOMINAL CAPACITY: 22,000 BTUH

NOMINAL C.O.P.: 2.740

WATER-TO-AIR HEAT PUMP
C.O.P. VS. INLET WATER TEMP.
FIGURE II-4
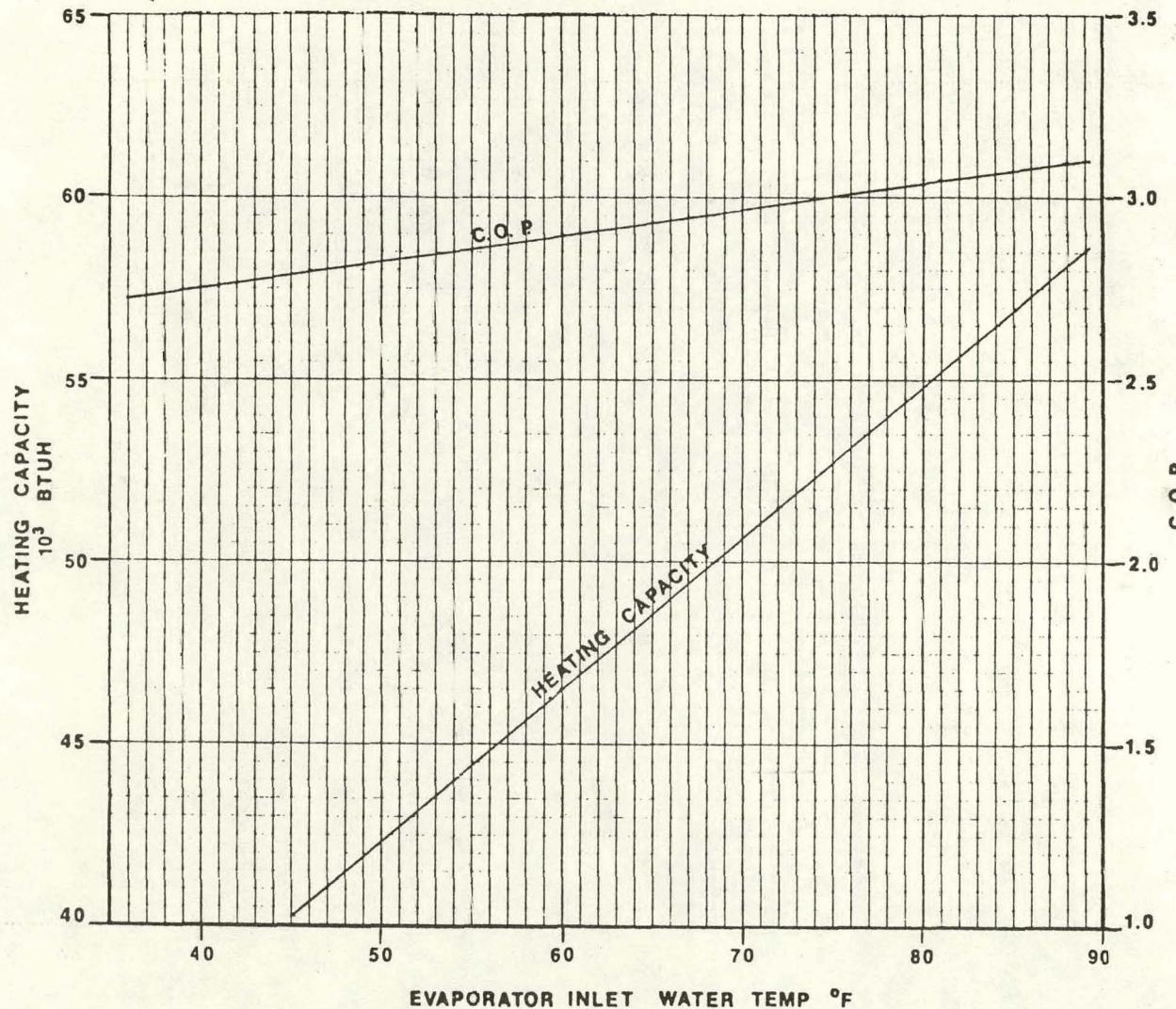
NOMINAL CAPACITY: 46,500

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

WATER-TO-AIR HEAT PUMP

C.O.P. VS. INLET WATER TEMP.

FIGURE II-5

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

$$\text{COP}_{\text{chiller}} = \frac{\text{COP}_{\text{nominal}}}{\% \text{ nominal work input expressed as a decimal}} \tag{11-28}$$

The COP chiller found for the given temperatures of water, is related to the COP heat pump by equation 11-6 given in section 11.4.3. The COP value, found by using equation 11-28 were reduced to allow for the fouling factors. Information on fouling factor correction presented in Reference 5 was used as a guide for this correction. The corrected values of COP used in our calculations were found by

$$\text{COP}_{\text{(corrected)}} = \text{COP}_{\substack{\text{chiller or} \\ \text{heat pump}}} \times F$$

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

TABLES 11-2 and 11-3 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 11-6.

TABLE 11-2

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 11-3

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
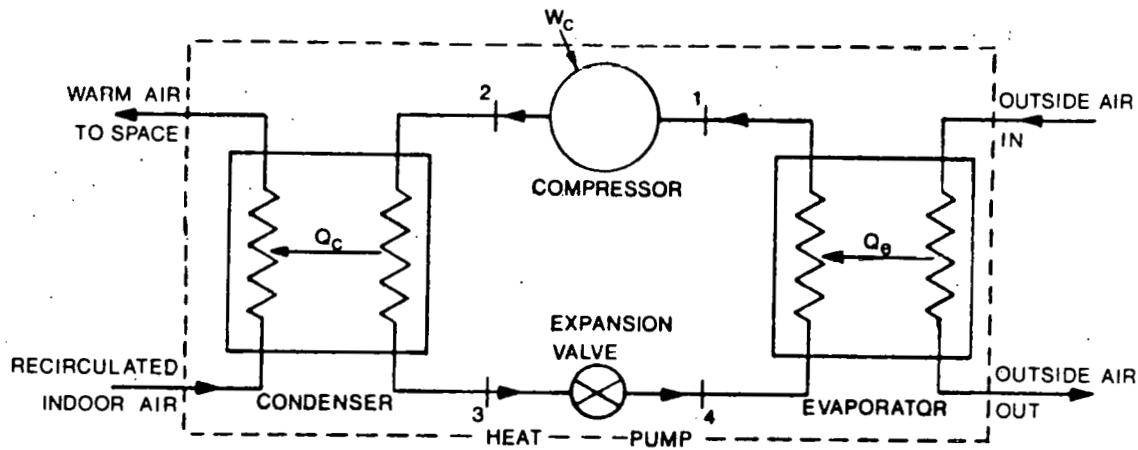
AIR-TO-AIR HEAT PUMP

FIGURE II-6

11.4.9 SITE-SPECIFIC SYSTEM DESIGNS

Calculations: Variable Pump Work

The distribution piping and pump system are designed and specified on the basis that a relatively constant head will be maintained on the system and flow will vary according to load. Pump work required will therefore also vary with load. The governing equation which calculates pump work as a function of flow is developed below:

System Pumping Coefficient

$$HEAD(ft) = C_v (GPM_{peak})^2$$

$$\therefore \quad C_v = \frac{HEAD}{(GPM_{peak})^2}$$

Pump Work (Electrical)

$$W_e = \left[\frac{C_v(\text{GPM})^3(8.33\ \frac{\text{Lbm}}{\text{gal}})}{(60\ \frac{\text{sec}}{\text{min}})(738\ \frac{\text{FT-Lbf}}{\text{sec-kw}})(\eta_{pump})(\eta_{motor})}\right] \times \text{hrs}$$

Assuming $\eta_{pump} = .65$ and $\eta_{motor} = .80$, the equation becomes:

$$W_e = \left[\frac{C_v(\text{GPM})^3(8.33)}{(60)(738)(.65)(.80)}\right] \times \text{hrs}$$

$$= (3.618 \times 10^{-4})(C_v)(\text{GPM})^3(\text{hrs})$$

In order to simulate in the calculational procedure the monthly load variations represented by the heating and cooling load duration curves, a percentage load factor can be inserted in conjunction with the flow factor (as shown below), and the hours at a particular percent of peak load can then be considered.

$$W_e = (3.618 \times 10^{-4})(C_v)(\text{LOAD FACTOR X GPM}_{peak})^3(\text{hrs})$$

For particular system applications, system pumping coefficients and peak flows can be inserted in the general equation, and the simplified equation then has only percent of peak load vs. time to consider as variables. Finally, the appropriate heating or cooling load duration curves can be broken down into percent and time factors by the finite integral technique and input into the equation. The sum of all integrals under the curve gives the total pump work required for any particular period.

CONYERS HP-WHR SYSTEM

PUMP HEAD	–	500 FT.
PEAK COOLING FLOW	–	2618 GPM
PEAK HEATING FLOW	–	1202 GPM

$$C_v = \frac{500}{(2618)^2} = 7.295 \times 10^{-5}$$

COOLING: $W_e = (3.618 \times 10^{-4})(7.295 \times 10^{-5})(L.F. \times 2618)^3$ hrs

$= 473.55 \ (L.F.)^3$ hrs

HEATING: $W_e = (3.618 \times 10^{-4})(7.295 \times 10^{-5})(L.F. \times 1202)^3$ hrs

$= 45.83 \ (L.F.)^3$ hrs

CORNELIA HP-WHR SYSTEM

PUMP HEAD	–	325 FT.
PEAK COOLING FLOW	–	4309 GPM
PEAK HEATING FLOW	–	1954 GPM

$$C_v = \frac{325}{(4309)^2} = 1.7504 \times 10^{-5}$$

COOLING: $W_e = (3.618 \times 10^{-4})(1.7504 \times 10^{-5})(L.F. \times 4309)^3$ hrs

$= 506.64 \ (L.F.)^3$ hrs

HEATING: $W_e = (3.618 \times 10^{-4})(1.7504 \times 10^{-5})(L.F. \times 1954)^3$ hrs

$47.24 \ (L.F.)^3$ hrs

In order to reduce the monthly heating and cooling load duration curves into a data set which could be input into the pump work equations, the bin

method of data reduction was employed. Peak heating load was assumed to occur
at the 99% design temperature and peak cooling at the 2% design temperature.
Bins were defined in 5°F increments from the design temperature to the appro-
priate datum (65°F for heating and 75°F for cooling), and the midpoint tem-
perature for each bin was assigned a percent of peak load figure. Duration
at each average load level was then interpolated from the load duration curve.
The data points, then in the proper form for use in the pump work equations,
were inserted in the appropriate heating or cooling equation and the work cal-
culation performed.

Several assumptions were necessary concerning system operation.

- During the heating season, if the estimated heating load dropped
  below 34%, the required pump work during that period was assumed to
  remain constant at the 34% level, representing the work requirements
  to maintain the system minimum flow. Operation during periods of
  marginal or no load was also assumed to be at the 34% load level
  for the same reason.

- During the cooling season, if the estimated cooling load dropped
  below 10%, the required pump work was assumed to remain constant
  at the 10% level to account for the minimum flow pump work. Like-
  wise, operation during periods of marginal or no cooling load was
  assumed to be at the 10% load level.

- During periods requiring both heating and cooling, heating loads
  above the 34% level were accounted for in the normal manner. Heat-
  ing loads of 1-34% were assumed to be met with operation at the
  34% load level. Cooling loads of 10% and up were also accounted
  for as explained above. However, cooling loads of 1-10% _and_ periods

with no apparant heating or cooling load were assumed to require

system operation at the 10% cooling load levels.

The results of the assessment of distribution pump power requirements are

summarized in Appendices C and D.

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REFERENCES

AUTHORS

1. Ambrose, E. R., <u>Heat Pumps and Electric Heating</u>, John Wiley
        and Sons, Inc., New York, 1966.

2. Arkansas Power and Light Company, Market Research Section,
        <u>Heat Pump Study</u>, Little Rock, Arkansas, 1974.

3. Binkley, C., B. Collins, <u>et al</u>, <u>Interceptor Sewers
        and Urban Sprawl</u>, Lexington Books, Washington, D.C.,
        1975.

4. Calm, James M., "Recovery of Wasted Heat With Central-
        ized and Distributed Heat Pump Systems," Presented
        at the ASME Winter Annual Meeting, 1978.

5. Christian, J. E., <u>Central Cooling - Compressive
        Chillers</u>, Argonne National Laboratory, ANL/CES/TE
        78-2, Argonne, Illinois, March 1978.

6. Christian, J. E., <u>Unitary Air-to-Air Heat Pumps</u>,
        Argonne National Laboratory, ANL/CES/TE 77-10,
        Argonne, Illinois, July 1977.

7. Christian, J. E., <u>Unitary Water-to-Air Heat Pumps</u>,
        Argonne National Laboratory, ANL/CES/TE 77-9,
        Argonne, Illinois, October 1977.

8. Clark, J.W., W. Viessman, and M. J. Hammer, <u>Water
        Supply and Pollution Control</u>, Dunn-Donnelley,
        New York, 1977.

9. Godfrey, R. S. (ed.), <u>Building Construction Cost
        Data</u>, Robert Snow Means Co., Inc., Massachusetts,
        1977.

10. Gordian Associates, <u>Heat Pump Technology, A Survey
        of Technical Development, Market Prospects and
        Research Needs</u>, Report HCP/M2121-01 for the
        Department of Energy, June 1978.

11. Holman, J. P., <u>Heat Transfer</u>, McGraw-Hill, Inc., U.S.A.,
        1972.

12. James, R. W., S. A. Marshall and S. N. Salerja, "The
        Heat Pump as a Means of Utilizing Low Grade Heat
        Energy," B.S.E., Vol. 43, January 1976.

13. Jennings, B. H., <u>Environmental Engineering, Analysis
        and Practice</u>, International Textbook Co.,
        Pennsylvania, 1970.

14. Kelly, G. E. and J. Bean, "Dynamic Performance of Residential Air-to-Air Heat Pumps," Proceedings of the 2nd Annual Heat Pump Technology Conference, Oklahoma State University, Stillwater, Oklahoma, October 18-19, 1976.

15. Kennedy, A. and C. Lee, Integrated Community Energy Systems (ICES) Commercialization Case Studies, Vol.1, Argonne National Laboratory, Energy and Environmental Systems Div., ANL/ICES-TM-4, Argonne, Illinois, 1978.

16. Kreith, F. J., Principles of Heat Transfer, International Textbook Co., Pennsylvania, 3rd Edition.

17. Kreith, F. J. and J. F. Kreidler, Solar Heating and Cooling, McGraw-Hill Book Company, New York, 1975.

18. Krenkel, P. A., and F. L. Parker (eds.), Biological Aspects of Thermal Pollution, Vanderbilt University Press, Nashville, Tenn., 1969.

19. Kusuda, T. and P. R. Achenback, "Earth Temperature and Thermal Diffusivity at Selected Stations in the United States," ASHRAE, January 25-28, 1965, Chicago, Illinois.

20. Lin, K. C. and G. W. Heinke, "Plant Data Analysis of Temperature Significance in the Activated Sludge Process," Journal WPCF, February 1977.

21. Metcalf and Eddy, Inc., Wastewater Engineering: Collection, Treatment, Disposal, McGraw-Hill Book Company, New York, 1972.

22. Pallio, F. S., "Energy Conservation and Heat Recovery in Wastewater Treatment Plants," Water and Sewage Works Magazine, February 1977.

23. Parker, W. H., R. W. Beausoliel and G. E. Kelly, "Factors Affecting the Performance of Residential Air-Air Heat Pumps," Semiannual meeting of the American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Chicago, Illinois, February 1977.

24. Phelps, E. B., Stream Sanitation, John Wiley and Sons, New York, 1944.

25. Phetteplace, G. and N. Labbe, Heat Distribution With Low Temperature Hot Water, Cold Regions Research and Engineering Laboratory, U. S. Corps of Engineers, Hanover, New Hampshire, August 1978.

26. Professional Builder: "Consumer/Builder Survey on Housing," January, 1976.

27. Rand Corporation, Energy Use and Conservation in the
      Residential Sector: A Regional Analysis, Report
      PB-254-468 for the U.S. Department of Commerce,
      June 1975.

28. Roensch, L. F., J.C. Grier and E. F. Klen, "The
      Achievement of Slime Control in Utility Surface
      Condensers Without Impairing Discharge Water
      Quality," Proceedings of the 40th Annual American
      Power Conference, April 26, 1978.

29. Rohsenow, W.M. and V.P. Hartnett, Handbook of Heat
      Transfer, McGraw-Hill, Inc., U.S.A., 1973.

30. Sawyer, C.N. and G. A. Rohlich, "Sewage Works Journal,"
      11:946, (1939).

31. Sawyer, C.N. and P.L. McCarty, Chemistry for Sanitary
      Engineers, McGraw-Hill Book Co., New York, 1967.

32. Schroepfer, G.J., M.L. Robins, and R. H. Susag,
      "The Research Program on the Mississippi River
      in the Vicinity of Minneapolis and St. Paul,"
      Advances in Water Pollution Research, Vol. 1,
      Peramm Press Ltd., London, 1964.

33. Sector, Peter, W., "Demonstration of Building Heating
      with a Heat Pump Using Thermal Effluent," CRREL
      Technical Report SR 77-11, AD A041024, May 1977.

34. Sonntag, R. E. and G. J. Van Wylen, Introduction to
      Thermodynamics: Classical and Statistical, John
      Wiley and Sons, Inc., New York, 1971.

35. Sussman, M. V., "Availability Analysis," Proceedings
      of the International Conference on Energy Use
      Management, Tucson, Arizona, October 1977.

36. Thurson, H.G., W.J. Fabrycky and G.J. Thurson, Engineering
      Economy, Prentice Hall, Inc., New Jersey, 1971.

37. Van Horne,J.C.,Financial Management and Policy, 2nd
      edition, Prentice-Hall, Inc., Englewood Cliffs, New
      Jersey, 1971.

38. Wagner, Tom, Population Projections for Georgia Counties,
      1980-2010, State Data Center, Office of Planning and
      Budget, Atlanta, Georgia, September 1977.

39. Wood, B.D., Applications of Thermodynamics, Addison-
      Wesley Publishing Company, Inc., Massachusetts, 1969.

TITLES

40. MOODY'S MUNICIPAL AND GOVERNMENT MANUAL, 1978,
    Volume 1
    Moody's Investor's Service, Inc.
    99 Church Street, New York, New York

41. Municipal Financing of Integrated Community Energy
    Systems, Argor... National Lagoratory, ANL/ICES-TM-3,
    November 1977

42. Municiral Waste Facilities - Statistical Summary, 1968
    Inventory, U.S. Department of the Interior, Federal
    Water Quality Administration, Division of Technical
    Support, Technical Data and Information Branch, 1969.

43. National Gas Survey, 1978, U.S.Department of Energy,
    Federal Energy Regulatory Commission DOE/FERC-008,
    Uc - 13; Washington, D.C., June 1978.

44. Population of the United States, U.S. Department of
    Commerce, Social and Economic Statistics Administra-
    tion, Bureau of the Census, Series P-23, No. 49,
    Washington, D.C., May 1974.

45. "Request for Proposals No. 78-437 - Heat Pump Centered
    Integrated Community Energy Systems, Phase I,"
    Argonne National Laboratory, Argonne, Illinois,
    March 1978.

46. Technical Guildelines for Energy Conservation, National
    Bureau of Standards, AD/A-041 668, U.S. Department
    of Commerce, National Technical Information Service,
    June 1977.

47. Water Quality Criteria, National Technology Advisory
    Committee, Federal Water Pollution Control
    Administration, Washington, D.C., 1968.

48. ASHRAE Equipment Handbook, American Society of Heating,
    Refrigerating, and Air-Conditioning Engineers, Inc.,
    New York, 1975.

49. ASHRAE Systems Handbook, American Society of Heating,
    Refrigerating, and Air-Conditioning Engineers,
    Inc., New York, 1976.

50. ASHRAE Handbook of Fundamentals, American Society of
    Heating, Refrigerating, and Air-Conditioning Engineers,
    Inc., New York, 1977.

51. ASHRAE Standard 14-67, "Methods of Testing for Rating
    Mechanical Condensing Units," American Society of
    Heating, Regrigeration, and Air-Conditioning Engineers,
    Inc., New York, 1967.

52. ASHRAE Standard 37-69, "Methods of Testing for Rating
    Unitary Air-Conditioning and Heat Pump Equipment,"
    American Society of Heating, Refrigeration, and
    Air Conditioning Engineers, Inc., New York, 1969.

53. ARI Standard 320-76, "Standard for Water Source Heat
    Pumps," Air-Conditioning and Refrigeration Institute,
    Arlington, Virginia, 1976.

54. ARI Standard 550-77, "Standard for Centrifugal Water-
    chilling Packages, "Air-Conditioning and Refrigeration
    Institute, Arlington, Virginia, 1966.

SUPPLEMENTAL REFERENCES

55. "Monthly Normals of Heating and Cooling Degree Days, 1941-
    1970," National Climatic Center, Asheville, North Carolina.

56. "Georgia - General Population Characteristics," U. S. Department
    of Commerce, Social and Economic Statistics Administration,
    Bureau of the Census, Washington, D.C., 1974.

57. Cornelia Community Development Concept Plan; Georgia Mountains
    Planning and Development Commission; Gainesville, Ga., 1978.

58. ASHRAE Standard 90-75, Energy Conservation in New Building Design;
    American Society of Heating, Refrigerating and Air Conditioning
    Engineers, Inc.; New York, 1975.

59. Potter, P. J., Power Plant Theory and Design, Ronald Press,
    New York, 1959.

60. City of Cornelia, 201 Facilities Plan; E.P.A. Project #C 130585-01-0;
    Hensley-Schmidt, Inc., Consultants; July 1977.

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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]

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	P5,151	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

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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]

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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 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	9,570,149	499	1,079,910	202	467,600	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	1,523,230	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	13,431,175	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	7,200,345	28	4,159,460	30	3,448,600	24	40,905	230	13,511,350	17	1,343,960
250,000–500,000	154	13,620,080	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	1,977,408
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	3	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	531	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	285,390	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	113	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	294,330
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	136	436,780	114	403,180	1	2,500	11	31,100			125	436,180	1	600
Nebraska	431	845,695	357	633,865	8	13,185	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	623	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	443	2,843,795	356	898,940	28	1,315,350	45	614,740	13	14,765	440	2,843,145	3	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	8,637,320	195	9,867,585	132	633,995	935	23,115,300	365	3,034,970
North Atlantic	893	14,170,086	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	27	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	93	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	3,292,795	12,365	130,684,771	1,558	9,541,278

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

SOURCE: <u>Municipal Waste Facilities – Statistical Summary, 1968 Inventory</u>.[41]

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]	26,136,577	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: <u>Municipal Waste Facilities - Statistical Summary, 1968 Inventory</u>. [41]

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 x 10^6 Btu/Yr
Cooling -- 18.96 x 10^6 Btu/Yr

Southern Climate

No. of End-Users -- 980

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

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

Northern Climate

Heating

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

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

$$3.32 \times 10^{10} \text{ Btu/Yr}$$

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

(4) Kwh input to central heat pump = 4.75 x 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 =

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

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

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

$5.8 \times 10^{10}/12 \times 10^6 = 4833$ 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 x (2370 + 975) x 75 hp x .745 kwh/hp = 224,282 kwh/Yr

Southern Climate

  1.2 x (1778 + 4833) x 75 hp x .745 kwh/hp = 594,990 kwh/Yr

Distribution Pumps

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

Northern Climate $\simeq$ 6%
Southern Climate $\simeq$ 5%

(1) Northern Climate Pumping Energy =

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

(2) Southern Climate Pumping Energy

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

Miscellaneous

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

APPENDIX   C

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CONYERS HP-WHR SYSTEM
COMPUTER PRINTOUT
THERMAL LOAD ASSESSMENT

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CONYERS HP-WHR SYSTEM

TYPE DEFINITIONS
****

HEATING DATA (* ....)
****

TYPE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
RES	6.780	5.430	4.290	1.380	.270	0.000	0.000	0.000	.090	1.320	3.960	6.480

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COOLING DATA (* ....)
****

TYPE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
RES	0.000	0.000	.140	.340	1.940	4.040	5.080	4.880	2.860	.720	0.000	0.000

* (Btu x 10^3/sq. ft.)

ZONE R-1

HEATING DATA(*...)

TYPE	NO. UNITS	MEAN SQ FT/UNIT	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
RES	50	1100.8	.3729	.2987	.2360	.0759	.0149	0.0000	0.0000	0.0000	.0050	.0726	.2178	.3564	1.6500
TOTAL			.37E+00	.30E+00	.24E+00	.76E-01	.15E-01	.0	.0	.0	.50E-02	.73E-01	.22E+00	.36E+00	
TOTAL HEAT.a(1.000)P. F.			.3729	.2987	.2360	.0759	.0149	0.0000	0.0000	0.0000	.0050	.0726	.2178	.3564	

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COOLING DATA(*...)

TYPE	NO. UNITS	MEAN SQ FT/UNIT	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
RES	50	1180.8	0.0000	0.0000	.0077	.0187	.1067	.2222	.2794	.2684	.1573	.0396	0.0000	0.0000	1.1000
TOTAL			0.0000	0.0000	.0077	.0187	.1067	.2222	.2794	.2684	.1573	.0396	0.0000	0.0000	
TOTAL COOL.a(1.000)P. F.			.0	.0	.77E-02	.19E-01	.11E+00	.22E+00	.28E+00	.27E+00	.16E+00	.40E-01	.0	.0	

$* \times 10^9$

```
**********************************************************************************************************************************

                                                      ZONE R-1
                                                      **********

                                                   SUMMARY  SHEET
                                                   **************


              MEAN                                        TOTAL HEATING        TOTAL COOLING         ADJUSTED             ADJUSTED
    TYPE    UNIT  AREA    NO. OF UNITS    TOTAL AREA          (BTU)                (BTU)          TOTAL HEATING        TOTAL COOLING
    ****    *********     ************    **********       ************         ************      *************        *************

    RES       1100.0          50          55000.0          .165E+07            .110E+07          .165E+07             .110E+07


         TOTAL ZONE HEATING IS ...............   .17E+07 BTU
         TOTAL ZONE COOLING IS ...............   .11E+07 BTU
         ZONE LAND AREA IS ...................              ACRES
         TOTAL ZONE HEATING PER ACRE IS .....              BTU/ACRE
         TOTAL ZONE COOLING PER ACRE IS .....              BTU/ACRE
         TOTAL BUILDING AREA IS .............   55000.00 SQUARE FEET
         TOTAL NUMBER OF UNITS IS ............      50
         HEATING PENETRATION FACTOR IS .......   01.000
         COOLING PENETRATION FACTOR IS .......   01.000
         ADJUSTED ZONE HEATING IS ...........   .17E+07 BTU
         ADJUSTED ZONE COOLING IS ...........   .11E+07 BTU
         ADJUSTED NO. OF UNITS IS ...........   50.0000
         ADJUSTED BUILDING AREA IS ..........   55000.0000 SQUARE FEET
         ADJUSTED HEATING PER ACRE IS .......              BTU/ACRE
         ADJUSTED COOLING PER ACRE IS .......              BTU/ACRE


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ZONE R-2

HEATING DATA(*)

TYPE	NO. UNITS	MEAN SQ FT/UNIT	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
RES	150	1200.0	1.2204	.9774	.7722	.2484	.0486	0.0000	0.0000	0.0000	.0162	.2376	.7128	1.1664	5.4000
TOTAL			.12E+01	.98E+00	.77E+00	.25E+00	.49E-01	.0	.0	.0	.16E-01	.24E+00	.71E+00	.12E+01	
TOTAL HEAT.a(1.000)P. F.			1.2204	.9774	.7722	.2484	.0486	0.0000	0.0000	0.0000	.0162	.2376	.7128	1.1664	

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COOLING DATA(*)

TYPE	NO. UNITS	MEAN SQ FT/UNIT	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
RES	150	1200.0	0.0000	0.0000	.0252	.0612	.3492	.7272	.9144	.8784	.5148	.1296	0.0000	0.0000	3.6000
TOTAL			0.0000	0.0000	.0252	.0612	.3492	.7272	.9144	.8784	.5148	.1296	0.0000	0.0000	
TOTAL COOL.a(1.000)P. F.			.0	.0	.25E-01	.61E-01	.35E+00	.73E+00	.91E+00	.88E+00	.51E+00	.13E+00	.0	.0	

$* \times 10^9$

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********************************************************************************************************

                                                    ZONE R-2
                                                    **********

                                                SUMMARY  SHEET
                                                **************


                        MEAN                                      TOTAL HEATING      TOTAL COOLING       ADJUSTED          ADJUSTED
            TYPE      UNIT  AREA      NO. OF UNITS     TOTAL AREA      (BTU)              (BTU)         TOTAL HEATING     TOTAL COOLING
            ****      *********      ************    *********   **************    ***************    **************   **************

            RES        1200.0          150           180000.0        .540E+07          .360E+07          .540E+07         .360E+07


            TOTAL ZONE HEATING IS ................  .54E+07 BTU
            TOTAL ZONE COOLING IS ...............   .36E+07 BTU
            ZONE LAND AREA IS ....................           ACRES
            TOTAL ZONE HEATING PER ACRE IS .....            BTU/ACRE
            TOTAL ZONE COOLING PER ACRE IS .....            BTU/ACRE
            TOTAL BUILDING AREA IS ..............  180000.00 SQUARE FEET
            TOTAL NUMBER OF UNITS IS ............      150
            HEATING PENETRATION FACTOR IS ......   01.000
            COOLING PENETRATION FACTOR IS ......   01.000
            ADJUSTED ZONE HEATING IS ...........   .54E+07 BTU
            ADJUSTED ZONE COOLING IS ...........   .36E+07 BTU
            ADJUSTED NO. OF UNITS IS ...........  150.0000
            ADJUSTED BUILDING AREA IS .......... 180000.0000 SQUARE FEET
            ADJUSTED HEATING PER ACRE IS .......            BTU/ACRE
            ADJUSTED COOLING PER ACRE IS .......            BTU/ACRE


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***************************************************************************************************************

                                                      ZONE R-3..
                                                      **********

                                                      HEATING DATA(*   )
                                                      *****************

          NO.    MEAN SQ
   TYPE  UNITS  FT/UNIT   JAN     FEB     MAR     APR     MAY     JUN     JUL     AUG     SEP     OCT     NOV     DEC     TOT
   ****  *****  *******   ***     ***     ***     ***     ***     ***     ***     ***     ***     ***     ***     ***     ***

   RES    150   1300.0   1.3221  1.0589   .8365   .2691   .0527  0.0000  0.0000  0.0000   .0176   .2574   .7722  1.2636  5.8500
                         -------  ------- ------- ------- ------- ------- ------- ------- ------- ------- ------- -------

   TOTAL                 .13E+01 .11E+01 .84E+00 .27E+00 .53E-01 .0      .0      .0      .18E-01 .26E+00 .77E+00 .13E+01

   TOTAL HEAT.a(1.000)P. F.  1.3221  1.0589   .8365   .2691   .0527  0.0000  0.0000  0.0000   .0176   .2574   .7722  1.2636
```

```
                                                      COOLING DATA(*   )
                                                      *****************

          NO.    MEAN SQ
   TYPE  UNITS  FT/UNIT   JAN     FEB     MAR     APR     MAY     JUN     JUL     AUG     SEP     OCT     NOV     DEC     TOT
   ****  *****  *******   ***     ***     ***     ***     ***     ***     ***     ***     ***     ***     ***     ***     ***

   RES    150   1300.0   0.0000  0.0000   .0273   .0663   .3783   .7878   .9906   .9516   .5577   .1404  0.0000  0.0000  3.9000
                         -------  ------- ------- ------- ------- ------- ------- ------- ------- ------- ------- -------

   TOTAL                 0.0000  0.0000   .0273   .0663   .3783   .7878   .9906   .9516   .5577   .1404  0.0000  0.0000

   TOTAL COOL.a(1.000)P. F.  .0      .0      .27E-01 .66E-01 .38E+00 .79E+00 .99E+00 .95E+00 .56E+00 .14E+00 .0              .0
```

$$* \times 10^9$$

ZONE R-3
****

SUMMARY SHEET
****

TYPE	MEAN UNIT AREA	NO. OF UNITS	TOTAL AREA	TOTAL HEATING (BTU)	TOTAL COOLING (BTU)	ADJUSTED TOTAL HEATING	ADJUSTED TOTAL COOLING
RES	1308.0	150	195000.0	.585E+07	.390E+07	.585E+07	.390E+07

```
TOTAL ZONE HEATING IS ................  .59E+07 BTU
TOTAL ZONE COOLING IS ................  .39E+07 BTU
ZONE LAND AREA IS ....................         ACRES
TOTAL ZONE HEATING PER ACRE IS .....          BTU/ACRE
TOTAL ZONE COOLING PER ACRE IS .....          BTU/ACRE
TOTAL BUILDING AREA IS .............. 195000.00 SQUARE FEET
TOTAL NUMBER OF UNITS IS .............     150
HEATING PENETRATION FACTOR IS .......   01.000
COOLING PENETRATION FACTOR IS .......   01.000
ADJUSTED ZONE HEATING IS ............  .59E+07 BTU
ADJUSTED ZONE COOLING IS ............  .39E+07 BTU
ADJUSTED NO. OF UNITS IS ............. 150.0000
ADJUSTED BUILDING AREA IS ........... 195000.0000 SQUARE FEET
ADJUSTED HEATING PER ACRE IS ........          BTU/ACRE
ADJUSTED COOLING PER ACRE IS ........          BTU/ACRE
```

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TOTAL SUMMARY SHEET

HEATING DATA (*)

ZONE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
R-1	.3729	.2987	.2360	.0759	.0149	0.0000	0.0000	0.0000	.0050	.0726	.2178	.3564	1.6500
R-2	1.2204	.9774	.7722	.2484	.0486	0.0000	0.0000	0.0000	.0162	.2376	.7128	1.1664	5.4000
R-3	1.3221	1.0589	.8365	.2691	.0527	0.0000	0.0000	0.0000	.0176	.2574	.7722	1.2636	5.8500
TOTAL	2.9154	2.3349	1.8447	.5934	.1161	0.0000	0.0000	0.0000	.0387	.5676	1.7028	2.7864	

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COOLING DATA (*)

ZONE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
R-1	0.0000	0.0000	.0077	.0187	.1067	.2222	.2794	.2684	.1573	.0396	0.0000	0.0000	1.1000
R-2	0.0000	0.0000	.0252	.0612	.3492	.7272	.9144	.8784	.5148	.1296	0.0000	0.0000	3.6000
R-3	0.0000	0.0000	.0273	.0663	.3783	.7878	.9906	.9516	.5577	.1404	0.0000	0.0000	3.9000
TOTAL	0.0000	0.0000	.0602	.1462	.8342	1.7372	2.1844	2.0984	1.2298	.3096	0.0000	0.0000	

$* \times 10^9$

ZONE	NO. OF UNITS	ADJUSTED NO. OF UNITS	HEATING PENET FACT	COOLING PENET FACT	TOTAL AREA	TOTAL ADJ AREA	TOTAL HEATING	TOTAL COOLING	ADJUSTED TOTAL HEATING	ADJUSTED TOTAL COOLING
R-1	50	50.0	1.000	1.000	.55E+05	.55E+05	1650000.00	1100000.00	1650000.00	1100000.00
R-2	150	150.0	1.000	1.000	.18E+06	.18E+06	5400000.00	3600000.00	5400000.00	3600000.00
R-3	150	150.0	1.000	1.000	.20E+06	.20E+06	5850000.00	3900000.00	5850000.00	3900000.00

```
TOTAL HEATING IS .......................... .13E+08 BTU
TOTAL COOLING IS .......................... .86E+07 BTU
TOTAL ADJUSTED HEATING IS ................. .13E+08 BTU
TOTAL ADJUSTED COOLING IS ................. .86E+07 BTU
TOTAL ACREAGE SERVED IS ...................        ACRES
ADJUSTED TOTAL HEATING PER ACRE IS .........        BTU/ACRE
TOTAL NUMBER OF UNITS ..................... 350.
ADJUSTED TOTAL NUMBER OF UNITS ............ 350.00
TOTAL BUILDING AREA IS .................... 430000.00 SQ. FT.
ADJUSTED TOTAL BUILDING AREA IS ........... 430000.00 SQ. FT.
ADJUSTED PENETRATION FACTOR IS ............ 1.000
```

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CONYERS HP-WHR SYSTEM PERFORMANCE ANALYSIS

1. Heating and cooling loads are determined from computerized load estimates.

2. Heat to distribution system (HDS)
   (assume End-User Heating C.O.P. = 2.9)

$$HDS = \text{Monthly Heat Required} \times (1- \frac{1}{2.9})$$

3. Heat required from sewage effluent (HSE)
   (assume End-User Cooling C.O.P. = 2.5
          Cent. Heat Pump C.O.P. = 4.5)

$$HSE = (HDS - 1.4 \text{ [Cooling Required]})(1- \frac{1}{4.5})$$

4. Heat dissipated at cooling tower (HDCT)
   (assume End-User Cooling C.O.P. = 2.5
                   Heating C.O.P. = 2.9)

$$HDCT = 1.4 \text{ (Cooling Required)} - HDS$$

5. Circulating pump energy consumption
   (assume 30 percent extra consumption)

$$KWH = \frac{1.3 \times [\frac{HSE + HDCT}{10}] \times 90ft.}{.65(\text{efficiency}) \times (2.65 \times 10^6 ft\text{-}lb/KWH)}$$

6. Sewage pump energy consumption
   (assume 10 percent extra consumption)

$$KWH = \frac{1.1 \times [\frac{HSE}{10}] \times 60ft.}{.65(\text{efficiency}) \times (2.65 \times 10^6 ft\text{-}lb/KWH)}$$

7. Distribution pump energy consumption

$$KWH = \frac{[\frac{HDS + 1.4 \text{ (cooling required)}}{10}] \times 500ft.}{.65(\text{efficiency}) \times (2.65 \times 10^6 ft\text{-}lb/KWH)}$$

8. Cooling tower energy consumption

$$KWH = \frac{HDCT \times .8 \text{ KWH/HP} \times 30 \text{ HP}}{5.558 \times 10^6 \text{ Btu/Hr}}$$

9. Central heat pump energy consumption (C.O.P. = 4.5)

$$\text{KWH} = \frac{[\text{HSE}] \times \frac{1}{4.5}}{3413 \text{ Btu/KWH}}$$

CONYERS HP-WHR SYSTEM
DISTRIBUTION PUMP POWER REQUIREMENTS

MONTH	COOLING		HEATING	
	HRS	KWH	HRS	KWH
JAN	---	---	744	9707.88
FEB	---	---	672	6016.11
MARCH	---	---	744	4349.97
APRIL	244	1105.29	476	1426.27
MAY	530	7515.68	214	442.49
JUNE	664	15,969.70	56	100.88
JULY	744	22,260.29	---	------
AUG.	744	46,930.92	---	------
SEPT.	563	11,508.57	157	290.61
OCT.	276	668.78	468	1321.16
NOV.	---	---	720	3779.37
DEC.	---	---	744	7662.83
TOTALS	3765 HRS	105,959.23 KWH	4995 HRS	35,097.57 KWH

TOTAL KWH REQUIREMENT - 141,056.80 KWH

CONYERS HP-WHR SYSTEM
MONTHLY PERFORMANCE ANALYSIS

	HEATING LOAD X 10⁹	COOLING LOAD X 10⁹	HEAT① FROM SEWAGE	HEAT① TO DIST. SYS.	HEAT① DISS. @ COOLING TOWER	CENT. H.P. KWH	DIST. PUMP KWH	CIR. PUMP KWH	SEWAGE PUMP KWH	COOLING TOWER KWH	MISC. AUX. KWH	TOTAL CENT. PL. KWH	END② USER KWH	TOTAL SYSTEM KWH	SYSTEM C.O.P.
JAN.	2.915	0	1.485	1.909	0	96,689	9,708	12,325	5,689	0	1,000	125,411	294,512	419,923	2.034
FEB.	2.335	0	1.189	1.529	0	77,416	6,016	9,868	4,556	0	1,000	98,856	235,913	334,769	2.044
MARCH	1.845	.060	1.125	1.209	0	73,249	4,350	9,338	4,310	0	1,000	92,247	193,439	285,686	1.954
APRIL	.593	.146	.184	.388	0	11,980	2,532	1,526	704	0	1,000	17,742	77,024	94,766	2.285
MAY	.116	.834	0	.076	1.092	0	7,958	7,417	0	4,715	1,000	21,090	109,464	130,554	2.132
JUNE	0	1.737	0	0	2.432	0	16,071	16,519	0	10,502	1,000	44,092	203,574	247,666	2.055
JULY	0	2.184	0	0	3.058	0	22,260	20,771	0	13,205	1,000	57,236	255,962	313,198	2.043
AUG.	0	2.098	0	0	2.937	0	46,930	19,948	0	12,682	1,000	80,560	245,883	326,443	1.883
SEPT.	.039	1.230	0	0	1.722	0	11,799	11,696	0	7,435	1,000	31,930	148,095	180,025	2.065
OCT.	.568	.310	0	.372	.062	0	1,990	421	0	268	1,000	3,679	93,719	97,398	2.641
NOV.	1.703	0	.868	1.116	0	56,516	3,779	7,205	3,325	0	1,000	71,825	172,060	243,885	2.046
DEC.	2.786	0	1.419	1.825	0	92,392	7,663	11,778	5,427	0	1,000	118,260	281,479	399,739	2.042
	12.900	8.599				408,242	141,056	128,812	24,011	48,807	12,000	762,928	2,311,124	3,074,052	

POWER REQUIREMENTS:

				CENT. H.P.	DIST. PUMP	CIR. PUMP	SEWAGE PUMP	COOLING TOWER	MISC. AUX.		END USER	
PERCENT OF CENTRAL PLANT –				53.5	18.5	16.9	3.1	6.4	1.6			
PERCENT OF SYSTEM –				13.3	4.6	4.2	0.8	1.6	0.4		75.2	

① Btu X 10⁹

② END-USERS:
  HTG C.O.P. = 2.91
  CLG C.O.P. = 2.53

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CONYERS HP-WHR SYSTEM
CONSTRUCTION COST ESTIMATES

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ENGINEERING EXPERIMENT STATION

GEORGIA INSTITUTE OF TECHNOLOGY ● ATLANTA, GEORGIA 30332

CONSTRUCTION COST ESTIMATE		DATE PREPARED			SHEET	OF

PROJECT

HP-WHR SYSTEM

LOCATION

CONYERS

CENTRAL PLANT

DRAWING NO.			ESTIMATOR		CHECKED BY	

Central Plant SUMMARY	QUANTITY		LABOR		MATERIAL		TOTAL COST
	NO. UNITS	UNIT MEAS.	PER UNIT	TOTAL	PER UNIT	TOTAL	
Central Heat Pump	1	ea.		18,500	ea.	35,000	53,500
Sewage Pumps	2	ea.	1750	3,500	2000	4,000	7,500
Circulating Pumps	2	ea.	600	1,200	3000	6,000	7,200
Distribution Pumps	2	ea.	3000	6,000	16000	32,000	38,000
Cooling Tower	1	ea.	5300	5,300	13500	13,500	18,800
Chemical Treatment System	1	Lump	500	500	3000	3,000	3,500
Control System		Lump	5000	5,000	9000	9,000	14,000
Piping, Valves, etc.		Lump	3000	3,000	11500	11,500	14,500
(inc. Hangers & Ftgs)							
Underground Piping	1000	ft	9.0	9,000	6	6,000	15,000
Electrical	Lump	Lump				55,000	55,000
Building	3000	SQ. FT.	4	12,000	16	48,000	60,000
Thermal Storage	2	ea.			75,000	150,000	221,300
Misc. Sitework		Lump		6,000			6,000
TOTALS							514,300
+ 8% Overhead							41,144
							555,444
+ 8% Profit							44,436
							599,880
+ 6% Engineering Fee							35,993
GRAND TOTAL							635,873

ENGINEERING EXPERIMENT STATION
GEORGIA INSTITUTE OF TECHNOLOGY • ATLANTA, GEORGIA 30332

CONSTRUCTION COST ESTIMATE		DATE PREPARED		SHEET	OF
PROJECT HP-WHR SYSTEM					
LOCATION CONYERS					
DISTRIBUTION SYSTEM					
DRAWING NO.		ESTIMATOR		CHECKED BY	

Distribution _____ SUMMARY	QUANTITY		LABOR		MATERIAL		TOTAL COST
	NO. UNITS	UNIT MEAS.	PER UNIT	TOTAL	PER UNIT	TOTAL	
Underground Piping[1]		L.F.					
Inc. Pipe, Installation							
Trench & Backfill							
14" Dia.	1,000		2.40	2,400	13.00	13,000	
12" Dia.	6,500		2.00	13,000	10.80	70,200	
10" Dia.	8,000		1.80	14,400	8.40	67,200	
8" Dia.	1,000		1.70	1,700	5.00	5,000	
6" Dia.	4,000		1.45	5,800	3.25	13,000	
5" Dia.	2,000		1.30	2,600	2.25	4,500	
4" Dia.	8,000		1.70	13,600	1.00	8,000	
3" Dia.	16,000		1.60	25,600	.60	9,600	
2" Dia.	12,000		1.40	16,800	.50	6,000	
Paving Patch		Lump				8,000	
TOTALS				$95,900		$196,500	$292,400
+ 8% Overhead							23,392
							315,792
+ 8% Profit							25,263
							341,055
+ 6% Engineering Fee							20,463
TOTAL							$361,518

[1] Price per foot based on 2 pipes per trench.

CONYERS HP-WHR SYSTEM
CONSTRUCTION COST ESTIMATE

DESCRIPTION: THERMAL STORAGE VESSELS - 2 EA.
45' x 45' x 20' (depth); NOMINAL CAPACITY - 600,000 GALLONS
REINFORCED CONCRETE, INSULATED, EARTH BERMED

ITEM	DESCRIPTION	UNIT	QUANTITY	UNIT PRICE	AMOUNT
1	CONCRETE (IN PLACE)*	C.Y.	694	$300.00	$208,200
2	COVER	S.F.	4050	2.00	81,000
3	INSULATION (IN PLACE)	S.F.		LUMP SUM	5,000
TOTAL - DIRECT COSTS					$221,300

* NOTE - GRADING, EXCAVATION AND BACKFILL WILL BE
INCLUDED IN CENTRAL PLANT COST ESTIMATE.

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APPENDIX   D

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CORNELIA HP-WHR SYSTEM
COMPUTER PRINTOUT
THERMAL LOAD ASSESSMENT

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CORNELIA HP-WHR SYSTEM

TYPE DEFINITIONS
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HEATING DATA (* ....)
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TYPE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
COM	10.500	8.650	7.650	2.700	.850	.150	0.000	0.000	.400	2.800	6.650	10.250
IND	0.000	0.000	0.000	0.000	0.000	0.008	0.000	0.000	0.000	0.000	0.000	0.000

COOLING DATA (* ....)
········•••••••••••••••••••••

TYPE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
COM	0.000	0.000	.320	.540	2.880	6.500	8.580	8.220	2.120	.860	0.000	0.000
IND	230.000	230.000	230.000	230.000	230.000	230.000	230.000	230.000	230.000	230.000	230.000	230.000

* (Btu x 10^3/sq.ft.)

ZONE C-1

HEATING DATA(*)

TYPE	NO. UNITS	MEAN SQ FT/UNIT	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
COM	72	4614.0	3.4882	2.8736	2.3421	.8970	.2824	.0498	0.0000	0.0000	.1329	.9302	2.2092	3.4051	16.6104
TOTAL			.35E+01	.29E+01	.23E+01	.90E+00	.28E+00	.50E-01	.0	.0	.13E+00	.93E+00	.22E+01	.34E+01	
TOTAL HEAT.@(1.000)P. F.			3.4882	2.8736	2.3421	.8970	.2824	.0498	0.0000	0.0000	.1329	.9302	2.2092	3.4051	

COOLING DATA(*)

TYPE	NO. UNITS	MEAN SQ FT/UNIT	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
COM	72	4614.0	0.0000	0.0000	.1063	.1794	.9568	2.1594	2.8503	2.7307	.7043	.2857	0.0000	0.0000	9.9729
TOTAL			0.0000	0.0000	.1063	.1794	.9568	2.1594	2.8503	2.7307	.7043	.2857	0.0000	0.0000	
TOTAL COOL.@(1.000)P. F.			.0	.0	.11E+00	.18E+00	.96E+00	.22E+01	.29E+01	.27E+01	.70E+00	.29E+00	.0	.0	

$$* \times 10^9$$

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**************************************************************************************************************

                                                    ZONE C-1
                                                    **********

                                                    SUMMARY  SHEET
                                                    ****************


                 MEAN
      TYPE      UNIT  AREA      NO. OF UNITS      TOTAL AREA      TOTAL HEATING      TOTAL COOLING      ADJUSTED          ADJUSTED
      ****      **********      ************      **********        (BTU)              (BTU)         TOTAL HEATING    TOTAL COOLING
                                                                 ************       ************     *************    *************

      COM        4614.0            72            332208.0         .166E+08           .997E+07         .166E+08         .997E+07


            TOTAL ZONE HEATING IS ................  .17E+08 BTU
            TOTAL ZONE COOLING IS ................  .10E+08 BTU
            ZONE LAND AREA IS ...................             ACRES
            TOTAL ZONE HEATING PER ACRE IS .....************ BTU/ACRE
            TOTAL ZONE COOLING PER ACRE IS .....         BTU/ACRE
            TOTAL BUILDING AREA IS .............         SQUARE FEET
            TOTAL NUMBER OF UNITS IS ...........     72
            HEATING PENETRATION FACTOR IS .......  01.000
            COOLING PENETRATION FACTOR IS .......  01.000
            ADJUSTED ZONE HEATING IS ...........  .17E+08 ETU
            ADJUSTED ZONE COOLING IS ...........  .10E+08 BTU
            ADJUSTED NO. OF UNITS IS ...........  72.0000
            ADJUSTED BUILDING AREA IS .........  332208.0000 SQUARE FEET
            ADJUSTED HEATING PER ACRE IS ....... ********** BTU/ACRE
            ADJUSTED COOLING PER ACRE IS .......         BTU/ACRE


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ZONE C-2

HEATING DATA(*)

TYPE	NO. UNITS	MEAN SQ FT/UNIT	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
COM	10	3500.0	.3675	.3028	.2468	.0945	.0298	.0053	0.0000	0.0000	.0140	.0980	.2328	.3588	1.7500
TOTAL			.37E+00	.30E+00	.25E+00	.95E-01	.30E-01	.53E-02	.0	.0	.14E-01	.98E-01	.23E+00	.36E+00	
TOTAL HEAT.ə(1.000)P. F.			.3675	.3028	.2468	.0945	.0298	.0053	0.0000	0.00001	.0140	.0980	.2328	.3588	

COOLING DATA(*)

TYPE	NO. UNITS	MEAN SQ FT/UNIT	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
COM	10	3500.0	0.0000	0.0000	.0112	.0189	.1008	.2275	.3003	.2877	.0742	.0301	0.0000	0.0000	1.0507
TOTAL			0.0000	0.0000	.0112	.0189	.1008	.2275	.3003	.2877	.0742	.0301	0.0000	0.0000	
TOTAL COOL.ə(1.000)P. F.			.0	.0	.11E-01	.19E-01	.10E+00	.23E+00	.30E+00	.29E+00	.74E-01	.30E-01	.0		.0

$* \times 10^9$

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                                                        ZONE C-2
                                                        **********

                                                     SUMMARY  SHEET
                                                     **************

                    MEAN                                    TOTAL HEATING       TOTAL COOLING        ADJUSTED          ADJUSTED
          TYPE    UNIT  AREA     NO. OF UNITS    TOTAL AREA      (BTU)               (BTU)         TOTAL HEATING    TOTAL COOLING
          ****    *********      ************    **********   ************        ************     *************    *************

          COM       3500.0           10           35000.0      .175E+07            .105E+07          .175E+07         .105E+07


                TOTAL ZONE HEATING IS ................   .18E+07 BTU
                TOTAL ZONE COOLING IS ................   .11E+07 BTU
                ZONE LAND AREA IS ....................          ACRES
                TOTAL ZONE HEATING PER ACRE IS .....          BTU/ACRE
                TOTAL ZONE COOLING PER ACRE IS .....          BTU/ACRE
                TOTAL BUILDING AREA IS .............   35000.00 SQUARE FEET
                TOTAL NUMBER OF UNITS IS ...........        10
                HEATING PENETRATION FACTOR IS ......   01.000
                COOLING PENETRATION FACTOR IS ......   01.000
                ADJUSTED ZONE HEATING IS ..........   .18E+07 BTU
                ADJUSTED ZONE COOLING IS ..........   .11E+07 BTU
                ADJUSTED NO. OF UNITS IS ...........   10.0000
                ADJUSTED BUILDING AREA IS ..........   35000.0000 SQUARE FEET
                ADJUSTED HEATING PER ACRE IS .......          BTU/ACRE
                ADJUSTED COOLING PER ACRE IS .......          BTU/ACRE


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ZONE C-3
••••••••••

HEATING DATA*
••••••••••••••••••••••

TYPE	NO. UNITS	MEAN SQ FT/UNIT	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
COM	9	7500.0	.7088	.5839	.4759	.1823	.0574	.0101	0.0000	0.0000	.0270	.1890	.4489	.6919	3.3750
TOTAL			.71E+00	.58E+00	.48E+00	.18E+00	.57E-01	.10E-01	.0	.0	.27E-01	.19E+00	.45E+00	.69E+00	
TOTAL HEAT.�(1.000)P. F.			.7088	.5839	.4759	.1823	.0574	.0101	0.0000	0.0000	.0270	.1890	.4489	.6919	

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COOLING DATA*
•••••••••••••••••••

TYPE	NO. UNITS	MEAN SQ FT/UNIT	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
COM	9	7500.0	0.0000	0.0000	.0216	.0365	.1944	.4388	.5792	.5549	.1431	.0581	0.0000	0.0000	2.0264
TOTAL			0.0000	0.0000	.0216	.0365	.1944	.4388	.5792	.5549	.1431	.0581	0.0000	0.0000	
TOTAL COOL.�(1.000)P. F.			.0	.0	.22E-01	.36E-01	.19E+00	.44E+00	.58E+00	.55E+00	.14E+00	.58E-01	.0	.0	

$* \times 10^9$

SUMMARY SHEET

TYPE	MEAN UNIT AREA	NO. OF UNITS	TOTAL AREA	TOTAL HEATING (BTU)	TOTAL COOLING (BTU)	ADJUSTED TOTAL HEATING	ADJUSTED TOTAL COOLING
COM	7500.0	9	67500.0	.338E+07	.203E+07	.338E+07	.203E+07

```
TOTAL ZONE HEATING IS ................   .34E+07 BTU
TOTAL ZONE COOLING IS ...............    .20E+07 BTU
ZONE LAND AREA IS ..................            ACRES
TOTAL ZONE HEATING PER ACRE IS .....           BTU/ACRE
TOTAL ZONE COOLING PER ACRE IS .....           BTU/ACRE
TOTAL BUILDING AREA IS .............     67500.00 SQUARE FEET
TOTAL NUMBER OF UNITS IS ...........        9
HEATING PENETRATION FACTOR IS ......    01.000
COOLING PENETRATION FACTOR IS ......    01.000
ADJUSTED ZONE HEATING IS ...........    .34E+07 BTU
ADJUSTED ZONE COOLING IS ...........    .20E+07 BTU
ADJUSTED NO. OF UNITS IS ...........    9.0000
ADJUSTED BUILDING AREA IS ..........    67500.0000 SQUARE FEET
ADJUSTED HEATING PER ACRE IS .......           BTU/ACRE
ADJUSTED COOLING PER ACRE IS .......           BTU/ACRE
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HEATING DATA *
****

TYPE	NO. UNITS	MEAN SQ FT/UNIT	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
COM	0	0.0	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
IND	1	1.0	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
TOTAL			.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	
TOTAL HEAT.@(1,000)P. F.			0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	

COOLING DATA *
****

TYPE	NO. UNITS	MEAN SQ FT/UNIT	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
COM	0	0.0	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
IND	1	1.0	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0028
TOTAL			.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	
TOTAL COOL.@(1,000)P. F.			.23E-03	.23E-03	.23E-03	.23E-03	.23E-03	.23E-03	.23E-03	.23E-03	.23E-03	.23E-03	.23E-03	.23E-03	

$* \times 10^9$

ZONE I-4
****

SUMMARY SHEET
****

TYPE	MEAN UNIT AREA	NO. OF UNITS	TOTAL AREA	TOTAL HEATING (BTU)	TOTAL COOLING (BTU)	ADJUSTED TOTAL HEATING	ADJUSTED TOTAL COOLING
COM	0.0	0	0.0	0.	0.	0.	0.
IND	1.0	1	1.0	0.	.276E+04	0.	.276E+04

```
TOTAL ZONE HEATING IS ................  .0      BTU
TOTAL ZONE COOLING IS ................  .28E+04 BTU
ZONE LAND AREA IS ....................          ACRES
TOTAL ZONE HEATING PER ACRE IS .....,           BTU/ACRE
TOTAL ZONE COOLING PER ACRE IS .....            BTU/ACRE
TOTAL BUILDING AREA IS ..............    1.00 SQUARE FEET
TOTAL NUMBER OF UNITS IS ...........        1
HEATING PENETRATION FACTOR IS .......   01.000
COOLING PENETRATION FACTOR IS .......   01.000
ADJUSTED ZONE HEATING IS ...........     .0     BTU
ADJUSTED ZONE COOLING IS ...........    .28E+04 BTU
ADJUSTED NO. OF UNITS IS ...........    1.0000
ADJUSTED BUILDING AREA IS ..........     1.0000 SQUARE FEET
ADJUSTED HEATING PER ACRE IS .......            BTU/ACRE
ADJUSTED COOLING PER ACRE IS .......            BTU/ACRE
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TOTAL SUMMARY SHEET

HEATING DATA (*)

ZONE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
C-1	3.4882	2.8736	2.3421	.8970	.2824	.0498	0.0000	0.0000	.1329	.9302	2.2092	3.4051	16.6104
C-2	.3675	.3028	.2468	.0945	.0298	.0053	0.0000	0.0000	.0140	.0980	.2328	.3588	1.7500
C-3	.7088	.5839	.4759	.1823	.0574	.0101	0.0000	0.0000	.0270	.1890	.4489	.6919	3.3750
I-4	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
TOTAL	4.5644	3.7602	3.0647	1.1737	.3695	.0652	0.0000	0.0000	.1739	1.2172	2.8908	4.4558	

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COOLING DATA (*)

ZONE	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOT
C-1	0.0000	0.0000	.1063	.1794	.9568	2.1594	2.8503	2.7307	.7043	.2857	0.0000	0.0000	9.9729
C-2	0.0000	0.0000	.0112	.0189	.1008	.2275	.3003	.2877	.0742	.0301	0.0000	0.0000	1.0507
C-3	0.0000	0.0000	.0216	.0365	.1944	.4388	.5792	.5549	.1431	.0581	0.0000	0.0000	2.0264
I-4	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0002	.0028
TOTAL	.0002	.0002	.1393	.2350	1.2522	2.8258	3.7300	3.5735	.9218	.3741	.0002	.0002	

$* \times 10^9$

ZONE	NO. OF UNITS	ADJUSTED NO. OF UNITS	HEATING PENET FACT	COOLING PENET FACT	TOTAL AREA	TOTAL ADJ AREA	TOTAL HEATING	TOTAL COOLING	ADJUSTED TOTAL HEATING	ADJUSTED TOTAL COOLING
C-1	72	72.0	1.000	1.000	.33E+06	.33E+06	16610400.00	9972884.16	16610400.00	9972884.16
C-2	10	10.0	1.000	1.000	.35E+05	.35E+05	1750000.00	1050700.00	1750000.00	1050700.00
C-3	9	9.0	1.000	1.000	.68E+05	.68E+05	3375000.00	2026350.00	3375000.00	2026350.00
I-4	1	1.0	1.000	1.000	.10E+01	.10E+01	0.00	2760.00	0.00	2760.00

TOTAL HEATING IS .... .22E+08 BTU

TOTAL COOLING IS .... .13E+08 BTU

TOTAL ADJUSTED HEATING IS .... .22E+08 BTU

TOTAL ADJUSTED COOLING IS .... .13E+08 BTU

TOTAL ACREAGE SERVED IS .... ACRES

ADJUSTED TOTAL HEATING PER ACRE IS .... BTU/ACRE

TOTAL NUMBER OF UNITS .... 92.

ADJUSTED TOTAL NUMBER OF UNITS 92.00

TOTAL BUILDING AREA IS .... 434709.00 SQ. FT.

ADJUSTED TOTAL BUILDING AREA IS .... 434709.00 SQ. FT.

ADJUSTED PENETRATION FACTOR IS 1.000

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CORNELIA HP-WHR SYSTEM PERFORMANCE ANALYSIS

1. Heating and cooling loads determined from computerized load estimates.

2. Heat to distribution system (HDS) (assume end-user heating C.O.P. = 3.0)

$$\text{HDS} = \text{Monthly Heat Required} \times (1 - \frac{1}{3.0})$$

3. Heat Required from sewage effluent (HSE) (assume end-user cooling C.O.P. = 2.5)

$$\text{HSE} = \text{HDS} - (1 + \frac{1}{2.5}) \times \text{Monthly Cooling Required}$$

4. Heat dissipated at cooling tower (HDCT)
   (assume end-user cooling C.O.P. = 2.5
                   heating C.O.P. = 3.0)

$$\text{HDCT} = 1.4 (\text{Monthly Cooling Required}) - \text{H.D.S.}$$

5. Circulating pump energy consumption: (assume 30 percent extra consumption)

$$\text{KWH} = \frac{1.3 \times [\frac{\text{HSE}}{20} + \frac{\text{HDCT}}{10}] \times 80 \text{ ft.}}{.65 \text{ efficiency} \times 2.65 \times 10^6 \text{ft-lb/KWH}}$$

6. Sewage pump energy consumption (assume 10 percent extra consumption)

$$\text{KWH} = \frac{1.1 \times [\frac{\text{HSE}}{20}] \times 60 \text{ ft.}}{.65 \text{ efficiency} \times 2.65 \times 10^6 \text{ft-lb/KWH}}$$

7. Distribution pump energy consumption:

$$\text{KWH} = \frac{[\frac{\text{HDS} + 1.4(\text{Cooling})}{10}] \times 325 \text{ ft.}}{.65 \text{ efficiency} \times 2.65 \times 10^6 \text{ft-lb/KWH}}$$

(Monthly)

8. Cooling tower energy consumption:

$$\text{KWH} = \frac{\text{HDCT} \times .8 \text{ KWH/HP} \times 50 \text{ HP}}{9.28 \times 10^6}$$

CORNELIA HP-WHR SYSTEM
DISTRIBUTION PUMP POWER REQUIREMENTS

MONTH	COOLING		HEATING	
	HRS	KWH	HRS	KWH
JAN.	---	---	744	10,006.53
FEB.	---	---	672	6,201.19
MARCH	---	---	744	4,483.79
APRIL	244	1,182.49	476	1,470.15
MAY	530	8,040.58	214	456.10
JUNE	664	17,085.05	56	103.98
JULY	744	23,814.98	---	---
AUG.	744	50,208.64	---	---
SEPT.	563	12,313.34	157	299.55
OCT.	276	715.49	468	1,361.80
NOV.	---	---	720	3,895.64
DEC.	---	---	744	7,898.57
TOTALS	3765 HRS	113,360.57 KWH	4995 HRS	36,177.3 KWH

TOTAL KWH REQUIREMENT - 149,537.87 KWH

CORNELIA HP-WHR SYSTEM
MONTHLY PERFORMANCE ANALYSIS

	HEATING LOAD X 10^9	COOLING LOAD X 10^9	HEAT[1] FROM SEWAGE	HEAT[1] TO DIST. SYS.	HEAT[1] DISS. @ COOLING TWR.	DIST. PUMP KWH	CIR. PUMP KWH	SEWAGE PUMP KWH	COOLING TOWER KWH	MISC. KWH (AUX.)	TOTAL CENT. PL. KWH	END[2] USER KWH	TOTAL SYSTEM KWH
JAN.	4.6	0	3.06	3.06	0	10,007	9,237	5,862	0	800	25,906	449,262	475,168
FEB.	3.8	0	2.53	2.53	0	6,201	7,637	4,847	0	800	19,485	371,129	390,614
MARCH	3.1	.14	1.87	2.06	0	4,484	5,645	3,583	0	800	14,512	319,171	333,683
APRIL	1.2	.23	.48	.80	0	2,653	1,449	920	0	800	5,822	144,154	149,976
MAY	.37	1.30	0	.25	1.57	8,497	9,467	0	6,610	800	25,374	188,494	213,868
JUNE	.06	2.80	0	.04	3.88	17,189	23,396	0	16,336	800	57,721	334,017	391,738
JULY	0	3.70	0	0	5.18	23,815	31,235	0	21,810	800	77,660	433,636	511,296
AUG.	0	3.60	0	0	5.04	50,209	30,391	0	21,221	800	102,621	421,916	524,537
SEPT.	.17	.92	0	.11	1.17	12,613	7,055	0	4,926	800	25,394	124,426	149,820
OCT.	1.2	.37	.28	.80	0	2,077	845	536	0	800	4,258	160,562	164,820
NOV.	2.9	0	1.93	1.93	0	3,896	5,826	3,697	0	800	14,219	283,230	297,449
DEC.	4.5	0	3.00	3.00	0	7,899	9,056	5,747	0	800	23,502	439,496	462,998
	21.9	13.06				149,540	141,239	25,192	70,903	9,600	396,474	3,669,493	4,065,967

POWER REQUIREMENTS:

PERCENT OF CENTRAL PLANT –	37.7	35.6	6.4	17.9	2.4
PERCENT OF SYSTEM –	3.7	3.5	0.6	1.7	0.2

PERCENT OF SYSTEM – ... 90.3

[1] Btu X 10^9

[2] END-USER HEAT PUMPS
HTG C.O.P. = 3.00
CLG C.O.P. = 2.53

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CORNELIA HP-WHR SYSTEM
CONSTRUCTION COST ESTIMATES

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ENGINEERING EXPERIMENT STATION
GEORGIA INSTITUTE OF TECHNOLOGY ● ATLANTA, GEORGIA 30332

CONSTRUCTION COST ESTIMATE		DATE PREPARED			SHEET	OF
PROJECT HP-WHR SYSTEM						
LOCATION CORNELIA						
CENTRAL PLANT						
DRAWING NO.		**ESTIMATOR**			**CHECKED BY**	

Central Plant ____ SUMMARY	QUANTITY		LABOR		MATERIAL		TOTAL COST
	NO. UNITS	UNIT MEAS.	PER UNIT	TOTAL	PER UNIT	TOTAL	
Central Heat Exch.	1	ea.	----	25,000	+	100,000	125,000
Sewage Pumps	2	ea.	2000	4,000	2,500	5,000	9,000
Circulating Pumps	2	ea.	600	1,200	4,000	8,000	9,200
Distribution Pumps	2	ea.	3000	6,000	14,000	28,000	34,000
Cooling Tower	1		----	9,500	-----	24,000	33,500
Chemical Treatment System	Lump		----	500	-----	3,000	3,500
Control System	Lump		----	4,000	-----	8,000	12,000
Piping, Valves, etc.	Lump			3,000		12,500	15,500
(Inc. Hangers & Ftgs)							
Underground Piping	800	ft	9.0	7,200	6.0	4,800	12,000
Electrical	Lump					45,000	45,000
Building 4000 sq.ft.	1	SQ. FT.	4	16,000	16.00	64,000	80,000
Land Acquisition	Lump					3,000	3,000
Thermal Storage	2	ea.					298,400
Misc. Sitework						6,000	6,000
TOTALS							686,100
+ 8% Overhead							54,888
SUBTOTAL							740,988
+ 8% Profit							59,279
SUBTOTAL							800,267
+ 6% Engineering Fee							48,016
GRAND TOTAL							848,283

CONSTRUCTION COST ESTIMATE		DATE PREPARED				SHEET	OF

PROJECT

   HP-WHR SYSTEM

LOCATION

   CORNELIA

   DISTRIBUTION SYSTEM

DRAWING NO.			ESTIMATOR			CHECKED BY	

Distribution _____ SUMMARY	QUANTITY		LABOR		MATERIAL		TOTAL COST
	NO. UNITS	UNIT MEAS.	PER UNIT	TOTAL	PER UNIT	TOTAL	
Underground Piping (1)		L.F.					
Inc. Pipe, Installation,							
Trench & Backfill							
18" Dia.	600		2.85	1,710	19.20	11,520	13,230
14" Dia.	5,700		2.40	13,680	13.00	74,100	87,780
12" Dia.	900		2.00	1,800	10.80	9,720	11,520
8" Dia.	4,680		1.70	7,956	5.00	23,400	31,356
6" Dia.	600		1.45	870	3.25	1,950	2,820
5" Dia.	3,420		1.30	4,446	2.25	7,695	12,141
4" Dia.	6,660		1.70	11,322	1.00	6,660	17,982
3" Dia.	2,550		1.60	4,080	.60	1,530	5,610
Paving Patch		SQ.YD. LUMP				40,000	40,000
TOTAL							222,439
+ 8% Overhead							17,795
							240,234
+ 8% Profit							19,218
							259,452
+ 6% Engineering Fee							15,567
							$275,020

(1) Price per foot based on 2 pipes per trench.

CORNELIA HP-WHR SYSTEM
CONSTRUCTION COST ESTIMATE

DESCRIPTION: THERMAL STORAGE VESSELS - 2 EA.
50' x 60' x 35' (depth); NOMINAL CAPACITY - 1,400,000 GALLONS
REINFORCED CONCRETE, INSULATED, EARTH BERMED

ITEM	DESCRIPTION	UNIT	QUANTITY	UNIT PRICE	AMOUNT
1	CONCRETE (IN PLACE)*	C.Y.	938	$300.00	$281,400
2	COVER	S.F.	6,000	2.00	12,000
3	INSULATION (IN PLACE)			LUMP SUM	5,000
	TOTAL - DIRECT COSTS				$298,400

* NOTE - GRADING, EXCAVATION, AND BACKFILL WILL BE
INCLUDED IN CENTRAL PLANT COST ESTIMATE.

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APPENDIX E

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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:

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

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

Where

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

$$\text{Evaporator heat} = \text{Flow rate (lb/hr)}$$
$$\text{x Specific heat of water}$$
$$(1.0\text{Btu/lb-}^{\text{o}}\text{F})$$

$$\text{x Change in water temperature } (^{\text{o}}\text{F})$$

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:

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

Where

W = flow rate of chilled water, lb per hr

T_e = temperature of water entering the unit, $^{\text{o}}\text{F}$

T_l = temperature of water leaving the unit, $^{\text{o}}\text{F}$

NOTE: The specific heat of water is implicit in the equation and is assumed to be 1.0 Btu/lbm-$^{\text{o}}$F

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:

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

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: <u>Heat Transfer</u>, J. P. Holman.[11]

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 F

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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_{el} = 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_{cl} = 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-user load of the house based on 75°F inside and -20°F outside.

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

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

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

$$HRF = 1.435$$

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: $\Sigma \dot{W}_c = \dot{W}_{ce} + \dot{W}_{cc} = 23603$ Btuh

PUMPING ENERGY:

Mass rate of flow of water through the evaporator of the end user

unit = $\dot{M}_w \, \Delta h$ (end-user heat pump) = $\dot{M}_w \, C_{pw}$ (80-70) = 38327.52

Where

$$C_{pw} = 1.00$$

$$\dot{M}_w = 3832.75 \text{ lbm/hr}$$

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

MASS RATE OF FLOW OF EFFLUENT:

HRF(central heat pump) $= \dfrac{6.991}{6.991-0.92} = 1.152$

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

$\dot{M}_{eff}$ = mass rate of flow of effluent $= \dfrac{33269.97}{1 \times (10)} = 3327.00$ 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: $\Sigma W_c + \Sigma W_p +$ 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

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

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

= 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 medi
(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, ineffi-
ciencies 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:

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

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

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

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

For example, for a pressure rise of

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

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

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

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 $0.425°F$ 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)) = $95^\circ F$

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

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

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 $C.O.P._{Season} = 1.382$

$$E_{Heating} = 47.2 \times 10^6 \text{ Btu/yr}$$

Distribution

ANL

J. Calm (50)
A. Kennedy (25)
K. Macal
T. Marciniak
J. Pascual (2)
V. Rabl
J. Tschanz
ANL Libraries (C. Archer)
ANL Washington (B. Graves) (10)
TIS Files (E.N. Pettitt)

DOE

S. Cavros
J. Kaminsky
G. Leighton
J. Millhone
J. Rodousakis (25)
I. Sewell
D. Walter
DOE-TIC (27)

External

C. Brett, University of Alabama (2)
P. Swenson, Consolidated Natural Gas Service Company (2)
F. Dubin, Dubin-Bloome Associates (2)
H. Lorsch, Franklin Research Center (2)
D. Wade, Georgia Tech Research Institute (15)
P. Anderson, Honeywell Energy Resources Center (2)
L. Katter, Rocket Research Company (2)

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