MASTER

**HEAT PUMP CENTERED
INTEGRATED COMMUNITY ENERGY SYSTEMS**

System Development

University of Alabama Interim Report

ARGONNE NATIONAL LABORATORY

**ENERGY AND ENVIRONMENTAL SYSTEMS
DIVISION**

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PREFACE

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

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

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

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

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

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

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

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

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

Nine parallel concepts are under development in the initial, System Development, phase. The report which follows presents the interim findings in the development and analysis of a concept being developed by The University of Alabama with technical support from Alabama Power Company and American Air Filter.

ABSTRACT

The heat pump centered integrated community energy system unifies groups of controlled environments (homes, businesses, etc.) into energy modules. Each module has an annual thermal energy storage system and water distribution system; and, in the controlled environments, each module has a water-source heat pump system. The modules can function independently or as part of a larger system composed of groups of modules. The components of the system outside the home and commercial units are serviced by a public utility. Large commercial units can economically install and operate independent energy modules. Original planning sets module size arbitrarily at 20 homes or environment control spaces or a large commercial building. In actuality, module size will vary with location and heating and cooling loads.

Technically, the heat pump system uses water-source heat pumps paralleled with thermal energy storage in underground aquifers. The aquifer is a natural underground reservoir such as sand and gravel or porous rock in which water can be stored and transmitted. A central water distribution system circulates water between the aquifer and the heat pumps. In the summer, the thermal energy rejected by the heat pumps is stored in aquifers for use during the winter season. During the winter, the stored thermal energy is retrieved by the heat pump system. The rejected cooler water from the heat pump during heating periods is used to form a temperature heat sink in the aquifer for use during the cooling cycle. To equate heating and cooling, supplementary thermal energy must be added or removed from the system on an annual basis. Heat addition is accomplished by utilizing waste heat or solar energy addition. A limited amount of heat

removal is accomplished by rejecting warm water from the aquifer or by placing a cooling tower in the system. Balancing the requirements of various types of buildings can be a partial solution.

Heat pump performance capability is primarily a function of the temperature difference at which the system must operate. If the difference between the source temperature and the controlled environment is large, the heat pump performance is poor. The smaller the temperature difference, the less entropy generated. An example of a large differential would be keeping a home at 72°F with an air-source heat pump using outside ambient temperature of -10°F for a thermal energy source. If 80° water, the minimum goal of this project, is used as a thermal energy source, performance of the heat pump system is improved. As a result, energy input to the pump is reduced. The situation is reversed during the cooling season with water temperatures below air temperatures.

Thermal energy storage is accomplished by using an underground aquifer as a reservoir. Two (or more) wells, penetrating the aquifer and spaced a predetermined distance apart, furnish water to the integrated energy system. During the cooling season, rejected warm water is injected into one well in the aquifer and relatively cold water is pumped from the aquifer through the second well to the heat pumps. The well spacing as related to aquifer storage capacity is determined so that a period of 6 months, or preferably longer, is required for the temperature gradient to move from one well to the second well. During the heating cycle, the water flow direction with respect to the 2 wells is reversed, and relatively warm water is recovered and pumped to the heat pumps.

Economically, the system shows improvement on both energy usage and capital costs. The system saves two-thirds of the energy required for

resistance heating; saves 30% of the energy required for most air-source heat pumps (primary function of climatic location); and saves one-third of the energy required for gas, coal, or oil heating, compared to energy input required at the power plant for heat pump usage. Energy costs are less than the cost of gas or oil in some parts of the country. Capital costs also show an advantage when compared to a standard air-source heat pump system or to a gas, oil, or coal heating system plus an air conditioner. In general, for an individual residence or small commercial system, an air-source heat pump capital cost is approximately 100% more than an equivalent water-source heat pump with the same ARI capacity. For large commercial applications, most chillers already are water-source units connected to cooling towers. Under severe winter weather conditions, however, a water-source heat pump in this system will provide over 4 times more heating capacity than an ARI equally rated air-source heat pump. The difference in capital cost between the heat pump systems is offset by the cost of wells and the water distribution system.

The high performance air-source heat pump utilizes an improved compressor, improved blower and external fan motors, and larger heat transfer areas. These changes will be incorporated into water-source heat pumps over the next 2 years. Performance will improve between 20% and 30%, making the proposed system 15% to 25% more efficient than the performance analysis contained in this report.

Two identified modifications may be used to add a water-to-air heat exchanger to the water-source heat pump. The first increases performance by approximately 10%; the second increases performance, on an annual basis, by 50% in some areas of the country, and eliminates the peaks in summer power plant output.

I. SYSTEM DESCRIPTION

The heat pump centered integrated community energy system is designed in groups of integrated energy modules. The controlled environments in each module are served by individual water-source heat pumps. The heat pump could be a small unit in an individual room, a central heat pump for a residence, or an integrated heating and cooling system for a major building. Each module is served by a set of wells, an aquifer or aquifers, a pumping system, and a water distribution system. Figure 1 is a schematic of the general technical concept of the system module. The modules are designed so they can be combined to make larger integrated systems and thereby improve reliability, minimize emergency maintenance, minimize pressure losses, and allow for ordered expansion. Schematics of this module system are shown in Figures 2 and 3. Figure 2 shows wells and pumping stations located in the right-of-way adjacent to roads, eliminating formal land use for these systems. The well outposts are expected to be no more conspicuous than the service outposts for underground distribution or telephone service.

The integrated system is divided into the following major components:

A. The controlled environmental system. This includes the controlled environment (house, apartment, small commercial building) and the individual water-source heat pumps. The controlled environment can be an individual room, apartment, or home served by a small individual heat pump or heat pump system. It can also be a large element such as a complete commercial or apartment building using a central water-source heat pump to heat or cool the individually controlled environments. The heat pumps can be water-to-water or water-to-air and may use one or more stage compressors

FIGURE 1: SCHEMATIC OF THE TECHNICAL COMPONENTS FOR THE PROPOSED HEAT PUMP CENTERED INTEGRATED COMMUNITY ENERGY SYSTEM

FIGURE 2: STANDARD HEAT PUMP SYSTEM MODULE IN HEAT PUMP INTEGRATED COMMUNITY ENERGY SYSTEM

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of reciprocating, centrifugal, or screw configuration. Systems are expected to be off-the-shelf items.

B. The thermal energy storage module, or the aquifer. Aquifers are water-bearing rocks found near or at nominal depths below the soil zone of the earth's surface. Such rocks have physical properties of porosity and permeability, i.e. the presence of intergranular voids, fractures, solution channels, or other interconnected openings capable of receiving, storing, and transmitting water from precipitation that infiltrates from the ground's surface. Commonly aquifers consist of rock of sedimentary origin, such as limestones, dolomites, siltstones, sandstones and gravels; although, in some instances, metamorphic and igneous rocks may contain water in sufficient quantities to serve as a source of fresh water. Approximately 60% of the surface of the continental United States is underlain by aquifers adequate for utilization in this system. Many urban areas are concentrated in valleys and near rivers where extensive aquifers exist. Thermal energy transfer is accomplished by transferring warm and cool water to and from the aquifers.

C. The wells which provide access to the aquifers. Wells consist of drilled holes, normally between 4 and 15 inches in diameter; or in some instances the holes may be hand-dug in which case diameters are somewhat larger. Each well will contain a submerged pump, controls, and a water tank with a capacity that may range between 20 and 200 gallons.

D. The water distribution system. The water distribution system includes water lines and the controls between the aquifers and user locations. The lines provide the water flow to heat pumps and provide the water flow return to the aquifers.

E. The auxiliary heat addition. To balance heating and cooling loads, additional energy is required in most parts of the country. This energy is applied by heat exchangers to source water, by waste energy sources, and/or by direct energy addition from solar collectors.

F. The public service utility. The public service utility will provide and service the wells, pumping systems, and water distribution system. In addition, some waste energy heat exchangers, solar systems, and heat pump exchangers could be provided and serviced by the utility.

G. Other minor system uses, such as providing water for fire protection. The benefit to the controlled environment is higher heating and cooling system performance. The system performance will be over 3 times that of resistance heating, will approach 2 times the performance of air-source heat pumps in some locations, and will increase by more than 30% the performance of oil, gas, or coal heating systems. These numbers are discussed in detail under Section III, Expected Performance.

I.A. Controlled Environment System

Each controlled environment or section of the controlled environment has its own water source heat pump system. In the home, the general heating and cooling system is no different from that of any other home (controlled environment). The heat pump unit is bought and installed by the home owner. The only difference is the water source which is required for the heat pump. A public utility will provide the water source. The water is expected to be purchased for a monthly lump sum cost, similiar to electrical demand charge, based on the capacity of the heat pump system in the home. This charge rate, which is minor compared to heat pump energy usage cost, is expected to encourage energy conservation.

For commercial buildings, water can also be provided for individual units or central units. The owner and consulting engineer must determine the best heating, ventilating, and air-conditioning system for the particular building. Any type water-source heat pump can be utilized with the system. The heat pump can be water-to-air or water-to-water with compression supplied by a reciprocating, screw, or centrifugal compressor. The use of cool intake water or of warm intake water supplied to the heat pump systems provides the increase in performance. The charge for water in large buildings could be economically measured as a function of energy or water flow rate. Flexible methods for charging are envisioned.

I.B. Thermal Energy Storage

Aquifers are used for energy storage and as energy sinks. The aquifer may be comprised of sand, gravel, or porous rock which will store and allow the transmissivity of water. A suitable aquifer for this system will allow water recovery at a rate of one gal./min. per ft. of thickness. Optimum diameter of a well is considered to be 6 inches. As noted previously, aquifers of this quality are encountered in a large portion of the United States. Thermal energy is stored in the water-soil-rock combination. Two primary methods are used to store and retrieve the thermal energy. These are:

1. Two wells in the same aquifer where water is injected into one well and recovered in the second well (Figure 4). During the injection and recovery process, a water front and a temperature front (at lower velocity) move between the wells. In the cooling cycle, warmer water is injected in 1 well, and cool water is withdrawn from the other and piped to the heat pump. For heating, the process is reversed and warm

FIGURE 4: STANDARD WELL PAIR FOR 1 MODULE

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water is withdrawn and cold water injected. The optimum spacing between wells in the aquifer must be sufficiently large so the time for the thermal front (wave) to pass between the wells is greater than 6 months. Depending on withdrawal and injection rates, a spacing of a few hundred ft. between wells is considered sufficient. More than a single pair of wells in the same aquifer may be utilized. A more detailed analysis is given in Section IX.B.

2. Two aquifers are utilized with one aquifer storing relatively warm water and the second storing relatively cold water (Figure 5). In this case, cold water is recovered from the cold storage aquifer, pumped through the condenser of the heat pump for cooling, and injected as relatively warm water in the second aquifer. For heating, the process is reversed with warm water being recovered from the warm reservoir, pumped through the evaporator of the heat pump for heating, and injected as relatively cold water into the cold storage aquifer. An example would be isolated aquifers at depths of 80 ft. and 300 ft. where the aquifer at 80 ft. is originally lacking in sufficient water.

In both of the 2 examples of aquifer systems, water loss or gain on an annual basis would be negligible since recovered and injected water quantities are equal.

Thermal storage in aquifers is a function of porosity and type of rock. Most rock has a specific gravity of approximately 2.6 and a specific heat of about 0.2 Btu/°F/pound mass. Porosity can range from less than 5% to more than 30% with 10% to 20% as general averages for most aquifers. The thermal storage capacity per degree temperature change is just over 30 Btu/ft.³, varying with porosity. At this value,

FIGURE 5: 2 AQUIFER SYSTEM FOR 1 MODULE SYSTEM

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a rock volume of 100 ft. by 200 ft. by 30 ft. has a storage capability of 2.0×10^8 Btu with a 10°F temperature change. General values for the storage capacity of aquifers as a function of size, porosity, and temperature variations are given in Figure 6.

Water velocities outside the immediate well vicinity will be of the order of magnitude of one ft./day, and the temperature front velocity between wells is approximately one-half that of water velocity. Differences in the hydrostatic head between wells will be less than 50 ft. More detailed values are given in the analysis section IX.C.

I.C. Well System

A system of wells is required as a conduit for water between the aquifer and ground surface. Well diameters may range from 4 inches to 15 inches and depths from 20 to a few hundred ft. Well size and number of wells will be a function of water demand and aquifer characteristics, primarily those of permeability, porosity, and thickness. Water yield increases with diameter (logarithmic function). Well design for a particular aquifer will be established by a trade-off between well size and number of wells. Water per unit cost is the controlling factor. System reliability and legal and environmental requirements must be considered.

I.D. Water Distribution System

Water will be distributed from the wells to the controlled environments through an uninsulated 2-pipe pressurized water system. This is shown schematically in Figure 1. Installed side-by-side with spacing between the 2, 1 pipe will carry relatively warm water and the other

relatively cold water. Each well will have a pumping system (Figure 7) consisting of a submerged pump, an accumulator system, and controls to provide a specific pressure in the pipelines. The accumulators at each well head allow the pumps to operate at capacity for limited periods. The on-off sequences can be minimized in a group of modules by sequencing the pumps. A 2-pipe system from the primary pipe lines to the controlled environments (home, etc.) will include a separate small pump (on the order of 0.1 HP) to draw warm water or cold water as required from 1 primary line and reject the water to the other primary line. This allows the heat pumps to operate at peak performance at all times, with the capability of certain pumps cooling and other pumps heating simultaneously. At all times warm water is available for heating, and cold water is available for cooling, eliminating poor performance in a unit for heating during the cooling season or vice versa. The system also minimizes well pumping by thermal demand diversity. By using the module system, pressure losses in the main water lines are minimized as a result of short travel distances. Calculated hydrostatic heads for designed systems will be less than 30 ft. of water. Figures 8 and 9 give pressure losses for homes separated by 100 ft. and 200 ft., respectively, at various flow rates. Average maximum flow rate to a home is 6.0 GPM. Six-inch lines will be utilized as main water lines to provide adequate water for emergency purposes such as fire protection systems. It may also be noted the pressure loss in the main lines is well below 1 foot of water if 2 homes at opposite ends of a module heat and cool simultaneously (flow rate of 10 GPM). Pressure loss in

FIGURE 8: MODULE TOTAL DISTRIBUTION PIPE PRESSURE LOSS

20 HOME MODULE

100 FEET BETWEEN HOMES

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the pipe lines to buildings from the primary pipeline is minimized by using pipelines 1-inch or more in diameter as installation labor is a major cost.

If suitable aquifers are not in the immediate area of the module, primary lines can be constructed to nearby areas. For this case additional pumping power is required and performance is decreased.

Major buildings would be 1 or more independent modules. An ideal situation would be where an aquifer is situated under the building or under adjacent roads, parks, parking areas, etc. In this case pressure losses will be appreciably less than in the distributed home module.

All pipe utilized is Polyvinyl Chloride (PVC). The PVC pipe will:

1. Carry the pressure requirement,
2. Satisfy the temperature requirements,
3. Minimize pipe cost and installation cost, and
4. Eliminate the possibility of dissimilar metal to metal corrosion from galvanic reactions.

A more complete discussion is included in the analysis section IX.D.

I.E. Additions of Ground Water, Waste Heat, and Solar Energy

In as much as heating and cooling loads will not match at most locations, additional thermal energy must be added to or removed from the system annually. The work input for the cooling and heating must also be considered. An example of this equilibrium phenomenon follows:

Assume a 30,000 Btu annual cooling load and determine the energy available for heating. Assume coefficients of performance equal to 3.0 for both heating and cooling. This requires that 30,000 Btu must be removed from the controlled environment. The work required to remove the energy can be determined by the definition of coefficient of performance (β) for cooling where W is work input, Q_L is the thermal energy removed and Q_H is the thermal energy rejected or:

$$\beta = 3.0 = \frac{Q_L}{W} = \frac{Q_H}{Q_H - Q_L}$$

and the work is equal to:

$$W = \frac{Q_L}{\beta} = \frac{30,000}{3.0} = 10,000 \text{ Btu}$$

The total thermal energy injected into the well is Q_H or Q_L plus the work or $30,000 + 10,000$ which equals $40,000$ Btu. For heating the $40,000$ Btu becomes Q_L in the definition of coefficient of performance (β') for heating or:

$$\beta' = 3.0 = \frac{Q_H}{Q_H - Q_L} = \frac{Q_H}{Q_H - 40,000}$$

$$Q_H = \frac{Q_L \beta'}{\beta' - 1} = 40,000 \times \frac{3}{3 - 1} = 60,000 \text{ Btu}$$

This means $60,000$ Btu is available for heating in this case compared to the $30,000$ Btu for cooling or a 2:1 ratio. This indicates that in an area where the heating load equals twice the cooling load, the energy storage system breaks even on cooling and heating if both COPs equal 3. Where the ratio of heating to cooling is greater than 2, extra heating is required; and where the ratio of heating to cooling is less than 2, extra cooling is required.

The extra heat must be supplied by waste heat or low cost solar energy and the cooling by a cooling tower or aquifer drawdown. Controls to make the system operate must be developed. Analytically, the relationship between heating and cooling develops from an algebraic combination of coefficients of performance as follows:

$$\frac{Q_{\text{Heating}}}{Q_{\text{Cooling}}} = \frac{\beta + 1}{\beta} \times \frac{\beta'}{\beta' - 1}$$

The ratio of heating to cooling load is shown in Figure 10 as a function of the heating and cooling coefficients of performance.

In most of the country extra heat must be added to the system. This is especially true with some of the alternative systems which show higher coefficients of performance.

In the South and Southwest and for major buildings, supplementary cooling is required. The following are possible solutions:

1. Place a cooling tower in the system. Since annual storage exists, a small unit operating continuously can be utilized. In a major building, the unit is expected to be owner owned; and, in a residential system, the public utility is expected to own and maintain the units. This system is recommended for large building modules.
2. Place a heat exchanger or dry tower in the system. With annual storage the system can be operated during cold weather to provide heat release.
3. Discard water after heating. This method uses cold groundwater to replace the heated water which is then discarded. This is the system presently utilized in many water-source heat pump installations and is the system presently recommended for scattered modules.

In northern latitudes, large buildings (large internal loads) can be combined with residential and small commercial buildings to remove the excess heat. The totally residential modules will require heat addition.

The basic methods to add energy to systems requiring additional heat are:

FIGURE 10: HEATING TO COOLING RATIO AVAILABILITY AS A
FUNCTION OF HEAT PUMP PERFORMANCE

1. Waste Heat Sources -- This requires placing a heat exchanger between the source and the water distribution system. Any source such as power plant cooling water, waste heat from laundry runoff, etc., can be utilized. In the home the waste water from showers, sinks, dishwashers, washing machines, etc., can be run through a heat exchanger. When the temperature exceeds the required temperature (presently 80°F), a control system will open a valve and pump water from the cold water line (well) to the warm water line (well) through the heat exchanger. These heat exchangers can operate on an annual basis with the available annual storage in the system. Energy discarded in the summer can be utilized in the winter.

2. Solar Energy -- Low temperature, low cost solar panels, primarily swimming pool panels, can be used for supplemental energy. Three types of solar systems are analyzed in Section IX.E. These are a hard molded rubber solar panel, a solar panel made from polyethylene bags, and a solar panel system using polyethylene bags as shallow pond collectors. These systems can be used only during the summer months to preclude freezing problems. Controls would be similar to the heat exchanger controls. When the temperature reaches the proper value, valves open and a pump transfers water from the cold line (well) to the warm line (well) through the solar panels. The utility company is expected to own and maintain these systems. It should be noted that for roof installations ideally located buildings may be used. Each building does not need to have an individual system in a community integrated system.

3. Matching Buildings in Modules or Groups of Modules -- Large buildings normally have large internal heat loads and lower heat losses

per square foot. For example, multistory buildings have no losses through ceilings and floors. By matching the modules with large buildings and modules with smaller buildings, the heating load and cooling load combinations can be matched.

In general, heat addition can be a combination of all 3 methods. The 6-inch lines are sufficiently large to allow transport of water. A set of controls will have to be designed to regulate the variations. These are expected to be implemented and change with time over a period of years.

I.F. The Public Service Utility

A utility is recommended to drill the wells, install pumping equipment, controls, primary piping, etc., and maintain the systems. The controlled environment area owner is expected to be responsible for secondary pipe lines from the main distribution system to the individual heat pump systems.

Charges for utility involvement should be on a first-rate basis for water, based on heating and cooling equipment capacity. Minimizing equipment capacity will encourage energy efficient buildings. It should be noted that monthly water charges are expected to be less than 20% of heating and cooling costs. Direct energy costs are still expected to encourage a conservative energy lifestyle. Installation and monitoring of energy meters are nonessential costs with little benefit except to the utility. Meter reading and installation are expected to cost more than total monthly charges. Rates from \$2.00 to \$5.00 per thousand square ft. of floor area per month seem to be reasonable charges which cover both power costs and servicing costs. Capital costs for the distribution system will be in the amount of \$1,000 per home and be on

an assessment basis, such as water lines.

Service by the utility is expected to be minimal with less than 1 service call per pump well unit per year. If modules are joined, emergency type calls at odd hours should be almost nonexistent as adjacent modules will have short time (up to a few weeks) capability to provide water for an outage module. The proposed system eliminated the need for an onsite operator.

I.G. Minor Related Systems

The water system can supply large amounts of water for short period emergencies. Each module will have a water pumping capacity in excess of 400 gallons per minute (using warm and cold wells). A few interconnected modules can provide more than sufficient water for fire fighting units. In case of a major disaster with complete loss of power, the use of auxiliary power to the module system can provide emergency water.

I.H. Summary

The proposed heat pump centered integrated energy system combines businesses and neighborhoods into a module structure which provides warm and cold water through the use of thermal energy storage in aquifers for heat pumps at minimum capital costs. Annual thermal energy storage allows the excess (unwanted) energy in the summer to be stored for winter use using naturally available storage.

II. POTENTIAL APPLICATIONS

The proposed system has applications for basically any heating and cooling system in the United States located near an aquifer with sufficient yield capacity. Over 60% of inhabited areas fall in this category, excluding the Alaska permafrost area. Applications can include:

1. Residential neighborhoods,
2. Commercial buildings, multistory office buildings, etc.,
3. Apartment complexes, and
4. Various combinations of the above.

Limitations exist in 2 cases.

1. For small isolated loads, the cost of well drilling, pipeline distribution, and maintenance could make the system uneconomical; and
2. A major multistory building in a populated area might not have a sufficiently large aquifer for thermal storage to satisfy heating and cooling capacity of the building. Combination of this module with modules of other buildings or modules of a residential area can correct this situation. Also, auxiliary heating or waste heat and/or cooling tower can make up the difference in required aquifer capacity.

As demonstrated by the economic analysis for a 20-home module, the proposed system can be incorporated economically in essentially any new residential subdivision. The well system and distribution system can be incorporated in the capital price for the lots. Geographically the system is economically feasible nationwide in areas which have aquifers, with the exception of southern Florida. In this area the best system is probably a ground water system with 1 well and single piping for a neighborhood.

Farther north, Alternative II is ideal for the required ratio of heating to cooling. This region has the largest peak power plant load during the summer. These peaks can be reduced by 60% if the total system is implemented. Cooling towers operating during winter months in these regions can provide energy equilibrium for the required Alternative II 45°F-water.

As one moves farther north, extra heat is required in the system. Matching large buildings which have large internal loads with residential neighborhoods could balance the energy requirements. The second option is to use sources of available waste heat. The third option is to use low-cost swimming pool type solar energy collection during summer months, stored on an annual basis in an aquifer.

The system becomes much more feasible as gas, oil, and electric prices increase. Similar applicability is anticipated as gas and oil become scarce or rationed.

Conversion of large buildings, mostly with water source chillers, is the most likely application. The capital costs are minimized as most equipment can still be utilized. This is more feasible as interruptible gas service becomes more prevalent.

As central air conditioning is added in homes, presently less than 30% saturation, the capital cost of installing the systems is offset by the air conditioner cost. Social pressure for air conditioning can accelerate implementation of the proposed system.

The social pressure for solar energy can be an additional incentive. The system with solar supplemental energy is a solar-assisted heat pump system and may be eligible for income tax credits. Additional tax credits for implementation could accelerate the process.

An economic incentive could be reduced electric rates (lower peak loads) for communities incorporating the system.

Figure 11 is a map of the continental United States showing adequate porous aquifers for the proposed system¹. Of major importance are the areas adjacent to main river basins which generally afford suitable aquifers. The Mississippi, Missouri, Ohio, and Allegheny are examples of such basins. Major population centers are located along these stream basins. In addition to the aquifer shown, some fractured and porous metamorphic and igneous rock will serve as adequate aquifers. Essentially over 60% of the continental United States land area and over 70% of the continental United States population has adequate aquifers available for operation of this system. Population statistics for 1975 show that 155,021,000 (75%) of the population is concentrated in metropolitan areas and 58,000,000 (25%) in non-metropolitan areas². From the map it is assumed 75% of the metropolitan areas and 60% of the non-metropolitan areas can use the proposed system. Combining numbers, 71% of United States households can utilize the system. It is assumed 75% of all United States businesses can utilize the system. Total applications as of 1975 are²:

71% of 72,500,000 households = 51,475,000 households

75% of 13,900,000 active businesses = 10,400,000 businesses

Extrapolating these numbers to the year 2000, which anticipates a 40% increase, results in possible applications for the year 2000 as

72,000,000 households

14,600,000 businesses

Arbitrarily using 20 households and 10 businesses per module results in the installation of 5,060,000 modules.

Figure 11. Adequate Aquifers for Proposed System in United States

For air conditioning, use of electricity can be decreased by approximately 50% by using a combination of the proposed system and alternatives to replace present systems. For heating the use of natural gas (60% in 1975) and fuel oil (22% in 1975) is replaced by electricity which can be generated with less-critical fuels such as coal, hydropower, or nuclear energy². For heating the energy requirement also decreased an average of 50% over systems presently utilized.

Energy usage per year in the United States is estimated as²:

	<u>Total Energy Usage</u> <u>Quadrillion Btu</u>	<u>Commercial and Residential Energy Usage</u> <u>%</u>	<u>Quadrillion Btu</u>
1975	70.6	17.8	12.5
2000	163	34.6	56.4

Statistics for 1970 show that 70% of residential energy usage and 76% of commercial energy usage is for heating and cooling³. Applying an average 73% to the above numbers, energy usage in the United States for heating and cooling is:

1975	9.5 quadrillion Btu
2000	41.2 quadrillion Btu

The proposed systems with the present state-of-the-art water source heat pumps can save 50% of the energy required in 2000 when compared to energy consumption rates of present day heating and cooling systems or:

20.6 quadrillion Btu per year

which equals 10 million barrels of petroleum per day

which equals 20.6 trillion ft.³ of natural gas per day

which equals 824 million tons of coal per year

The proposed system and alternatives have the potential for economically feasible implementation with a corresponding potential to reduce energy consumption.

III. EXPECTED PERFORMANCE

The expected performance of the proposed system and first alternative shows a major increase in performance with respect to energy usage. The second 2 alternatives show larger increases in performance. The performance of the proposed system is compared to present systems, including resistance heating, oil and gas heat, and air source heat pumps. The comparison is made in energy utilized, namely thermal energy at the furnace or thermal energy in the power plant generating electricity. The proposed system shows an advantage over all other systems.

An energy analysis of the proposed system and present heating and cooling systems is shown in Figure 12. The new development is the economical annual thermal energy storage provided by nature. The energy distribution system (water distribution) allows the feasibility of injecting waste heat or solar energy into the system at any time of the year. The power plant is included in the schematic as performance for all systems has a final comparison on the basis of paid-for-energy -- fuel into the system.

A Second Law of Thermodynamics Analysis indicates the effect of decreasing the temperature difference over which the heat pump operates. Efficiency in the universe is defined as minimizing the rate of entropy generation. Per Figure 13, the total entropy generation equals $(Q+W)/T_H - Q/T_L$. This term must be greater than or equal to zero. As the temperature approach each other, the work required to keep ΔS positive approaches zero. This does not account for inefficiencies in the heat pumps, but places the heat pump in a system where performance can be maximized. The highest performance heat pumps available are used in the system to keep the work as close as possible to the minimum required work.

FIGURE 12: FIRST LAW OF THERMODYNAMICS ANALYSIS

FIG. 13 SECOND LAW OF THERMODYNAMICS ANALYSIS

This project does not develop the heat pump but the system in which the heat pump operates. The Second Law of Thermodynamics Analysis indicates minimizing the temperature difference over which the heat pump operates is a major step towards maximizing performance.

Numerous studies have been made using the earth and other types of storage to minimize these temperature differences.^{6,7} This has not only been true in the United States, but also in Europe.^{8,9} In many of the systems, the heat transfer is by pipes in the earth. In this study nature is allowed to provide the heat transfer mechanism, porous underground aquifers through which water can flow to transfer the thermal energy.

The proposed systems are first evaluated by coefficient of performance (COP) which is defined as:

$$\text{COP}_{\text{Cooling}} = \frac{\text{Energy Removed from Controlled Environment}}{\text{Work Required to Remove Energy}}$$

$$\text{COP}_{\text{Heating}} = \frac{\text{Energy Added to Controlled Environment}}{\text{Work Required to Add Energy}}$$

These definitions are applied directly and independently to the heating and cooling sequence (Tables 1 to 4). In Alternatives II and III, where direct heat exchanger is used for cooling or heating, very high coefficients of performance result. These values are somewhat artificial as the opposite sequence, heating and cooling, provides the proper temperature water to allow the direct thermal energy transfer. Part of this energy could be charged to the direct heat transfer. As a result, the annual energy usage must be applied to evaluate these systems. The energy for heating and cooling must be properly balanced.

The performance values for the proposed system and alternatives should be appreciably higher in the near future. The water source heat pumps produced by American Air Filter from which basic performance data

have been taken are somewhat obsolete. New compressors of higher efficiency have been on the market for a few years. Basic data from Tescumseh support this fact.⁴ In addition, motor efficiencies have improved and heat-exchanger areas have increased. American Air Filter is incorporating these changes (all components already on the market) in all of their units over the next 2 years. They estimate these changes will increase their water source heat pump performance by a minimum of 20 to 30 percent. This means the proposed system and alternatives will increase performance by the same magnitude. Air source heat pump A already has incorporated these changes and is utilized as a comparison with these systems. The best available heat pump data have been used for comparisons. The comparisons should indicate larger performance gains. Equipment in the proposed system has not been updated with components on the market, whereas the compared system performance already has incorporated these items. Very few of these high-performance heat pumps have been installed.

Performance of the proposed heat pump systems and alternative systems are shown in Tables 1 to 4. The proposed system performance (Table 1) is that of a basic water source heat pump using 300 watts for water pumping power, 60°F water for cooling, and 80°F water for heating. Alternative I, Figure 14, uses a 50% efficiency heat exchanger between the incoming air and water to transfer part of the energy by natural processes. A step gain is recorded in performance. Tables 3 and 4 use natural heat transfer for either cooling or heating. In Alternative II Table 3 65°F water is used for heating. Removing the energy from the water reduces water temperature to 45°F. This water is then used as

TABLE 1: PERFORMANCE OF WATER SOURCE HEAT PUMP A*
IN PROPOSED SYSTEM

COOLING							
ENTERING WATER TEMP °F	ID ¹ WBT	TOTAL CAP MBTUH	SENSIBLE CAP @ IDBT ²			POWER ³ KW	COP ⁴
			75	80	85		
65	61	31.8	26.3	29.2	31.8	3.90	2.39
	64	34.1	24.4	28.6	31.7	3.99	2.50
	67	36.8	21.2	25.2	30.7	4.06	2.66
	70	37.9	18.0	21.4	25.5	4.14	2.68
75	61	30.9	25.5	28.4	30.9	4.05	2.24
	64	33.15	23.7	27.7	30.8	4.15	2.34
	67	35.7	20.6	24.5	29.8	4.22	2.48
	70	36.8	17.4	20.8	24.8	4.30	2.51
85	61	30.3	25.0	27.8	30.3	4.13	2.15
	64	32.5	23.2	27.2	30.2	4.23	2.25
	67	35.0	20.2	24.0	29.2	4.30	2.39
	70	36.1	17.1	20.4	24.3	4.38	2.42
HEATING							
ENTERING WATER TEMP °F	CAP @ IDBT °F, MBTUH					POWER KW	COP @ 70°
	60	65	70	75	80		
60	44.5	43.7	42.8	41.9	41.5	4.36	2.88
70	48.2	47.2	46.3	45.4	44.9	4.63	2.93
80	54.1	53.0	52.0	50.9	50.4	5.02	3.04
90	59.1	57.9	56.8	55.7	55.1	5.45	3.05

1. Indoor Wet Bulb Temperature, °F
2. Indoor Dry Bulb Temperature, °F
3. Power for compressor, fan, and water pump
4. COP = Capacity (Kw)/Kw (compressor + fan + 300 watts for water pumping)

*Based on manufacturer's data at flow rate of 6 GPM for American Air Filter model HW33, taken from manufacturer's catalog specification page ENER-1-118-NOV-01.

Cost \$1,300 installed per J. E. Christian, Unitary Water to Air Heat Pumps, ANL/CES/TE77-9.

TABLE 2: SYSTEM PERFORMANCE FOR MODIFIED WATER SOURCE HEAT PUMP A
WITH PRE-WATER TO AIR HEAT EXCHANGER -- ALTERNATIVE I

HEATING										
Basic Water-Source Heat Pump System					Water-Source Heat Pump with Water-Air Exchanger					
IDBT ³ °F	Cap. ² MBTUH	Input ¹ Power KW	COP ⁴	Water Flow GPM	Pump Cap. MBTUH	Heat Exchanger Cap. ² MBTUH	Total ² Cap. MBTUH	Input ¹ Power KW	COP ⁴	Water Flow GPM
60	53.6	5.44	2.89	4	46.3	14.0	59.3	5.44	3.19	4
65	52.6	5.44	2.83	4	48.0	10.4	58.4	5.44	3.15	4
70	51.6	5.44	2.78	4	49.2	6.9	56.1	5.44	3.02	4

COOLING										
Basic Water-Source Heat Pump System					Water-Source Heat Pump With Water-Air Exchanger					
IDBT ³ °F	Cap. ² MBTUH	Input ¹ Power KW	COP ⁴	Water Flow GPM	Pump Cap. MBTUH	Heat Exchanger Cap. ² MBTUH	Total ² Cap. MBTUH	Input ¹ Power KW	COP ⁴	Water Flow GPM
70	37.1	3.98	2.73	5	37.5	6.9	44.4	3.98	3.27	6
75	37.1	3.98	2.73	5	37.1	10.4	47.5	3.98	3.50	6
80	37.1	3.98	2.73	5	36.4	13.8	50.2	4.10	3.59	6
85	37.1	3.98	2.73	5	36.0	17.3	53.3	4.20	3.72	6

¹Includes input for compressor, air handler, and water pump. Water pumping equals 300 watts.

²Total capacity (dry coil). Calculations based on survey of manufacturer's data at inlet water temperatures of 60°F for cooling and 80°F for heating.

³Indoor Dry Bulb Temperature.

⁴Coefficient of Performance = $\frac{\text{Capacity, MBTUH}}{3.412 \text{ MBTU/KW} \times \text{Power Input}}$

*Based on manufacturer's data for American Air Filter model HW33, specification page ENER-1-118

TABLE 3: SYSTEM PERFORMANCE FOR MODIFIED WATER SOURCE HEAT PUMP A
WITH PRE-WATER TO AIR HEAT EXCHANGER - ALTERNATIVE II

Water Temperature Variation 40°F to 60°F

HEATING

IDBT ¹ (°F)	AIRFLOW (SCFM)	WATERFLOW (GPM)	CAPACITY (MBTUH)	POWER ² (KW)	COP ³
60	1000	4.1	40.6	4.36	2.73
65	1000	4.0	39.0	4.36	2.62
70	1000	3.9	38.6	4.36	2.59

COOLING

IWBT ⁴ (°F)	AIRFLOW (SCFM)	WATERFLOW (GPM)	CAPACITY (MBTUH)	POWER ⁶ (WATTS)	COP ³
61	1000	2.4	20.2	900	6.52
64	1000	3.9	35.1	900	11.04
67	1000	4.9	45.3	900	14.08
70	1000	6.0	56.4	900	18.04

1. Indoor Dry Bulb Temperature, °F
2. Power Required for Compressor, Fan, and Water Pumping
3. $COP = \frac{CAPACITY}{(Power(KW) \times 3.412 (MBTUH/KW))}$
4. Indoor Wet Bulb Temperature At 70°F Dry Bulb Temperature
5. Net Capacity = Total Capacity - Fan Input (600W = 2.05 MBTUH)
6. Fan 0.6KW, Pumping 0.3KW

Heat Pump Data Extracted From American Air Filter Specification Sheet
 ENER - 1 - 118 - NOV. - 01 For HW-33 Unit

TABLE 4: SYSTEM PERFORMANCE FOR MODIFIED WATER SOURCE HEAT PUMP A
WITH PRE-WATER TO AIR HEAT EXCHANGER - ALTERNATIVE III

Water Temperature Variation 80°F to 100°F

HEATING					
IDBT ¹ (°F)	AIRFLOW (SCFM)	WATERFLOW (GPM)	CAPACITY ² (MBTUH)	POWER ⁶ (KW)	COP ⁴
60	2000	4.8	51.6	1.05	10.01
65	2000	4.2	45.7	1.05	8.93
70	2000	3.6	39.7	1.05	7.76

COOLING					
IWBT ⁵ (°F)	AIRFLOW (SCFM)	WATERFLOW (GPM)	CAPACITY (MBTUH)	INPUT ³ (KW)	COP ⁴
61	1000	4.5	29.7	4.21	2.07
64	1000	4.6	31.8	4.31	2.17
67	1000	4.8	34.3	4.38	2.30
70	1000	5.0	35.4	4.46	2.32

1. Indoor Dry Bulb Temperature, °F
2. Net Capacity, Heat Exchanger + Fan Input
3. Total Power For Fan, Pump Work and Compressor
4.
$$\text{COP} = \frac{\text{NET CAPACITY (MBTUH)}}{(\text{Power (KW)} \times 3.412 \text{ (MBTUH/KW)})}$$
5. Indoor Wet Bult Temperature
6. Fan 1.2KW, Pumping 0.3KW

Heat Pump Data Extracted From American Air Filter Specification Sheet
 ENER - 1 - 118 - NOV. - 01 For HW33 Unit With Twin Blower Unit For Heating

direct cooling for air conditioning. As a result, the heating COP decreases but the cooling COP shows a very large increase. The inverse occurs for Alternative III (Table 4). Water at 80°F is used for cooling which results in heating the water to 100°F. The 100°F water is then used to heat directly. The cooling COP decreases; however, the heating COP shows a large increase. The increase is not as large as for natural cooling as an increase in air flow is required. No change in water flow rate is necessary.

Figures 15 and 16 show a major advantage of the system performance versus the air source heat pumps performance. Neither the coefficient of performance nor the capacity are effected by ambient conditions. The air source heat pumps at 0°F have lost 80% of their 60°F capacity and 60% of their performance.

A major benefit is a reduction in power plant peak loads. Using Alternative II in parts of the South and the East Coast could eliminate peak summer loads. Matching loads to system alternatives, to types of buildings, and to power supplies is a performance possibility. The ratio of peak loads with Alternative II to Air Source Heat Pump C for air conditioning is proportional to the ratio of COP's or a decrease of over 60%. Energy comparisons show a load decrease of 80%. Integrating some of these systems in power plant operation decreases peak load.

A second major benefit is that oil and gas are not necessary. The power plants can operate on coal or nuclear energy saving critical fuels.

Tables 5 and 6 present the heating and cooling energy requirements for the various type systems for the well-insulated typical home listed in Section IX.A. for 5 cities. This home is an energy-conservation home and uses much less energy³ than over 99% of the present homes. A

Table 5: Energy Requirements for Heating Well Insulated Typical Home

		BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
	Units					
Annual Heating Load	Kw-hr	11.3(10 ³)	14.8(10 ³)	547	19.5(10 ³)	14.7(10 ³)
Annual Heating Load		38.55(10 ⁶)	50.5(10 ⁶)	18.7(10 ⁵)	66.5(10 ⁶)	50.2(10 ⁶)
Direct Electrical Heating						
Resistance Heating ¹	Kw-hr	14.8(10 ³)	14.8(10 ³)	547	19.5(10 ³)	14.7(10 ³)
Power Plant Input ²	Btu	110(10 ⁶)	144(10 ⁶)	53.4(10 ⁵)	190(10 ⁶)	143(10 ⁶)
Gas and Oil Heating						
Furnace Input ³	Btu	62.2(10 ⁶)	81.5(10 ⁶)	30.2(10 ⁵)	107(10 ⁶)	81.0(10 ⁶)
Gas	Therms	622	815	30.2	1070	810
Oil	Gallons	429	562	20.8	738	559
Air Source Heat Pump A						
COP	--	2.43	2.33	2.70	2.16	2.20
Power Input	Kw-hr	4.65(10 ³)	6.35(10 ³)	203	9.03(10 ³)	6.68(10 ³)
Power Plant Input ²	Btu	45.3(10 ⁶)	61.9(10 ⁶)	1.98(10 ⁶)	88(10 ⁶)	65.2(10 ⁶)
Air Source Heat Pump C						
COP	--	2.0	1.92	2.27	1.76	1.80
Power Input	Kw-hr	5.65(10 ³)	7.71(10 ³)	241	11.1(10 ³)	8.17(10 ³)
Power Plant Input ²	Btu	55.1(10 ⁶)	75.5(10 ⁶)	2.35(10 ⁶)	108(10 ⁶)	79.7(10 ⁶)
Air Source Twin Heat Pump A						
COP	--	2.45	2.37	2.70	2.21	2.25
Power Input	Kw-hr	4.61(10 ³)	6.24(10 ³)	203	8.82(10 ³)	6.53(10 ³)
Power Plant Input ²	Btu	44.9(10 ⁶)	50.9(10 ⁶)	1.98(10 ⁶)	86(10 ⁶)	53.7(10 ⁶)
Water Source Heat Pump A						
COP	--	2.78	2.78	2.78	2.78	2.78
Power Input	Kw-hr	4.1(10 ³)	5.32(10 ³)	197	7.01(10 ³)	5.29(10 ³)
Power Plant Input ²	Btu	39.6(10 ⁶)	51.9(10 ⁶)	1.92(10 ⁶)	68.3(10 ⁶)	51.6(10 ⁶)
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative I						
COP	--	3.02	3.02	3.02	3.02	3.02
Power Input	Kw-hr	3.74(10 ³)	4.90(10 ³)	181	6.46(10 ³)	4.87(10 ³)
Power Plant Input ²	Btu	36.5(10 ⁶)	47.8(10 ⁶)	1.77(10 ⁶)	62.9(10 ⁶)	47.5(10 ⁶)
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative II						
COP	--	2.59	2.59	2.59	2.59	2.59
Power Input	Kw-hr	4.36(10 ³)	5.7(10 ³)	2.11(10 ²)	7.53(10 ³)	5.68(10 ³)
Power Plant Input ²	Btu	4.25(10 ⁷)	5.57(10 ⁷)	2.06(10 ⁷)	7.34(10 ⁷)	5.54(10 ⁷)
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative III						
COP	--	7.76	7.76	7.76	7.76	7.76
Power Input	Kw-hr	1.46(10 ³)	1.9(10 ³)	70.49	2.51(10 ³)	1.89(10 ³)
Power Plant Input ²	Btu	1.42(10 ⁷)	1.86(10 ⁷)	6.89(10 ⁵)	2.45(10 ⁷)	1.85(10 ⁷)

1. COP = 1.0

2. Combined Power Plant and Distribution Efficiency Equals 0.35 Based on Higher Heating Value of Fuel

3. Combined Furnace and Combustion Efficiency Equals 0.62 Based on Higher Heating Value of Fuel

Table 6: Energy Requirements for Cooling Well Insulated Typical Home

		BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
	Units					
Annual Cooling Load	Kw-hr	7.66(10 ³)	3.38(10 ³)	14.2(10 ³)	3.37(10 ³)	6.25(10 ³)
Annual Cooling Load	Btu	26.1(10 ⁶)	11.5(10 ⁶)	48.5(10 ⁶)	11.5(10 ⁶)	21.3(10 ⁶)
Air Source Heat Pump A						
COP	--	2.73	2.82	2.75	2.79	2.72
Power Input	Kw-hr	2.81(10 ³)	1.19(10 ³)	5.16(10 ³)	1.21(10 ³)	2.30(10 ³)
Power Plant Input ²	Btu	27.3(10 ⁶)	11.7(10 ⁶)	50.4(10 ⁶)	11.8(10 ⁶)	22.4(10 ⁶)
Air Source Heat Pump C						
COP	--	2.0	2.47	2.40	2.44	2.37
Power Input	Kw-hr	3.83(10 ³)	1.37(10 ³)	5.92(10 ³)	1.38(10 ³)	2.64(10 ³)
Power Plant Input ²	Btu	37.3(10 ⁶)	13.3(10 ⁶)	57.7(10 ⁶)	13.5(10 ⁶)	25.7(10 ⁶)
Air Source Twin Heat Pump A						
COP	--	2.72	2.81	2.75	2.79	2.72
Power Input	Kw-hr	2.82(10 ³)	1.20(10 ³)	5.16(10 ³)	1.21(10 ³)	2.31(10 ³)
Power Plant Input ²	Btu	27.4(10 ⁶)	11.7(10 ⁶)	50.4(10 ⁶)	11.8(10 ⁶)	22.4(10 ⁶)
Water Source Heat Pump A						
COP	--	2.73	2.73	2.73	2.73	2.73
Power Input	Kw-hr	2.81(10 ³)	1.24(10 ³)	5.02(10 ³)	1.23(10 ³)	2.30(10 ³)
Power Plant Input ²	Btu	27.3(10 ⁶)	12.0(10 ⁶)	50.8(10 ⁶)	12.0(10 ⁶)	22.3(10 ⁶)
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative I						
COP	--	3.27	3.27	3.27	3.27	3.27
Power Input	Kw-hr	2.34(10 ³)	1.03(10 ³)	4.34(10 ³)	1.03(10 ³)	1.91(10 ³)
Power Plant Input ²	Btu	22.8(10 ⁶)	10.0(10 ⁶)	42.4(10 ⁶)	10.0(10 ⁶)	18.6(10 ⁶)
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative II						
COP	--	14.8	14.8	14.8	14.8	14.8
Power Input	Kw-hr	5.18(10 ²)	2.28(10 ²)	9.59(10 ²)	2.28(10 ²)	4.22(10 ²)
Power Plant Input ²	Btu	5.04(10 ⁶)	2.22(10 ⁶)	9.36(10 ⁶)	2.22(10 ⁶)	4.11(10 ⁶)
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative III						
COP	--	2.30	2.30	2.30	2.30	2.30
Power Input	Kw-hr	3.33(10 ³)	1.47(10 ³)	6.17(10 ³)	1.47(10 ³)	2.72(10 ³)
Power Plant Input ²	Btu	3.24(10 ⁷)	1.43(10 ⁷)	6.0(10 ⁷)	1.43(10 ⁷)	2.65(10 ⁷)

1. COP = 1.0
2. Combined Power Plant and Distribution Efficiency Equals 0.35 Based On Higher Heating Value Of Fuel
3. Combined Furnace and Combustion Efficiency Equals 0.62 Based On Higher Heating Value Of Fuel

3-ton heat pump capacity is basically required for either heating or cooling. The tables show the total energy in electrical units where applicable or as fuel heating values for furnaces. Also, the electrical units are changed to total heating values for fuel entering the power plant. This allows total energy comparisons.

Tables 7 to 18 give heating and cooling energy input comparisons of the proposed system and three alternatives for present day systems for heating, cooling, and heating and cooling. These are basic energy inputs, fuel at the furnace or power plant. The values are for the well-insulated typical home.

The comparisons indicate that in areas such as Miami with both mild, cold, and warm temperatures, the advantages of the system are questionable. Maximum performance is obtained with Alternative I in the area from central Florida to northern Georgia, Alabama, and Mississippi westward. Farther north Alternative II gives the best performance until heating loads overshadow cooling loads in the range of a ratio of 3:6. For example, Detroit shows approximately the same performance for Alternative I and II. Farther north Alternative III is most feasible if plus 100°F waste heat is available. Otherwise Alternative I shows better results.

For the vast majority of the country, with the exception of the lower one-half of Florida and part of the western coastal area, the system shows a sizable increase in performance.

Table 7: Energy*Comparison for Heating and Cooling Well Insulated Typical Home
Percentage Savings Utilizing Proposed Water Source Heat Pump A

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
Proposed Water Source Heat Pump System					
Total Energy Input Btu	6.69(10 ⁷)	6.39(10 ⁷)	5.27(10 ⁷)	8.03(10 ⁷)	7.39(10 ⁷)
Resistance Electric Heating and Air Source Heat Pump C Cooling					
Total Energy Input Btu	1.47(10 ⁸)	1.5(10 ⁸)	6.3(10 ⁷)	2.04(10 ⁸)	1.69(10 ⁸)
Relative Savings	54%	57.4%	16.3%	60.6%	56%
Gas and Oil Heating and Air Source Heat Pump C Cooling					
Total Energy Input Btu	9.95(10 ⁷)	9.48(10 ⁷)	6.07(10 ⁷)	1.2(10 ⁸)	1.07(10 ⁸)
Relative Savings	32.8%	32.6%	13.2%	33%	31%
Air Source Heat Pump A					
Total Energy Input Btu	7.26(10 ⁷)	7.36(10 ⁷)	5.24(10 ⁷)	9.98(10 ⁷)	8.76(10 ⁷)
Relative Savings	7.85%	13.2%	--	19.5%	15.6%
Air Source Heat Pump C					
Total Energy Input Btu	9.24(10 ⁷)	8.88(10 ⁷)	6.0(10 ⁷)	1.22(10 ⁸)	1.05(10 ⁸)
Relative Savings	27.6%	28%	12.2%	34%	29.6%

*Energy represents energy input at power plant for electrical power plus higher heating value of fuel for furnace input.

Table 8: Energy*Comparison for Heating Well Insulated Typical Home
Percentage Savings Utilizing Proposed Water Source Heat Pump A

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
Proposed Water Source Heat Pump System					
Total Energy Input Btu	39.6(10 ⁶)	51.9(10 ⁶)	1.92(10 ⁶)	68.3(10 ⁶)	51.6(10 ⁶)
Electric Resistance Heating					
Total Energy Input Btu	110(10 ⁶)	144(10 ⁶)	53.4(10 ⁵)	190(10 ⁶)	143(10 ⁶)
Relative Savings	64%	64%	64%	64%	64%
Gas and Oil Heating					
Total Energy Input Btu	62.2(10 ⁶)	81.5(10 ⁶)	30.2(10 ⁵)	107(10 ⁶)	81.0(10 ⁶)
Relative Savings	36%	36%	36.4%	36%	36%
Air Source Heat Pump A					
Total Energy Input Btu	45.3(10 ⁶)	61.9(10 ⁶)	1.98(10 ⁶)	88(10 ⁶)	65.2(10 ⁶)
Relative Savings	12.6%	16%	3%	22%	20.8%
Air Source Heat Pump C					
Total Energy Input Btu	55.1(10 ⁶)	75.5(10 ⁶)	2.35(10 ⁶)	108(10 ⁶)	79.7(10 ⁶)
Relative Savings	28.1%	31%	18%	36.7%	35.2%

*Energy represents energy input at power plant for electrical power plus higher heating value of fuel for furnace input.

Table 9: Energy* Comparison for Cooling Well Insulated Typical Home
Percentage Savings Utilizing Proposed Water Source Heat Pump A

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
Proposed Water Source Heat Pump System					
Total Energy Input Btu	27.3(10 ⁶)	12.0(10 ⁶)	50.8(10 ⁶)	12.0(10 ⁶)	22.3(10 ⁶)
Air Source Heat Pump A					
Total Energy Input Btu	27.3(10 ⁶)	11.7(10 ⁶)	50.4(10 ⁶)	11.8(10 ⁶)	22.4(10 ⁶)
Relative Savings	--	--	--	--	.4%
Air Source Heat Pump C					
Total Energy Input Btu	37.3(10 ⁶)	13.3(10 ⁶)	57.7(10 ⁶)	13.5(10 ⁶)	25.7(10 ⁶)
Relative Savings	26.8%	9.7%	11.9%	11%	13.2%

*Energy represents energy input at power plant for electrical power.

Table 10: Energy* Comparison for Heating and Cooling Well Insulated Typical Home
Percentage Savings Utilizing Proposed Modified Water Source Heat Pump A
With Pre-Water Heat Exchanger -- Alternative I

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
Proposed Water Source Heat Pump System					
Total Energy Input Btu	5.93(10 ⁷)	5.78(10 ⁷)	4.4(10 ⁷)	7.29(10 ⁷)	6.61(10 ⁷)
Resistance Electric Heating and Air Source Heat Pump C Cooling					
Total Energy Input Btu	14.7(10 ⁷)	1.5(10 ⁸)	6.3(10 ⁷)	2.04(10 ⁸)	1.69(10 ⁸)
Relative Savings	59.7%	61.5%	30.2%	64.3%	60.9%
Gas and Oil Heating and Air Source Heat Pump C Cooling					
Total Energy Input Btu	9.95(10 ⁷)	9.48(10 ⁷)	6.07(10 ⁷)	1.2(10 ⁸)	1.07(10 ⁸)
Relative Savings	40.4%	39%	27.5%	39.3%	38.2%
Air Source Heat Pump A					
Total Energy Input Btu	7.26(10 ⁷)	7.36(10 ⁷)	5.24(10 ⁷)	9.98(10 ⁷)	8.76(10 ⁷)
Relative Savings	18.3%	21.5%	16%	27%	24.5%
Air Source Heat Pump C					
Total Energy Input Btu	9.24(10 ⁷)	8.88(10 ⁷)	6.0(10 ⁷)	1.22(10 ⁸)	1.05(10 ⁸)
Relative Savings	35.8%	34.9%	26.7%	40.2%	37%

*Energy represents energy input at power plant for electrical power plus higher heating value of fuel for furnace input.

Table 11: Energy* Comparison for Heating Well Insulated Typical Home
Percentage Savings Utilizing Proposed Modified Water Source
Heat Pump A With Pre-Water Heat Exchanger -- Alternative I

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
Proposed Water Source Heat Pump System					
Total Energy Input Btu	36.5(10 ⁶)	47.8(10 ⁶)	1.77(10 ⁶)	62.9(10 ⁶)	47.5(10 ⁶)
Electric Resistance Heating					
Total Energy Input Btu	110(10 ⁶)	144(10 ⁶)	53.4(10 ⁵)	190(10 ⁶)	143(10 ⁶)
Relative Savings	66.8%	66.8%	66.8%	66.9%	66.8%
Gas and Oil Heating					
Total Energy Input Btu	62.2(10 ⁶)	81.5(10 ⁶)	30.2(10 ⁵)	107(10 ⁶)	81(10 ⁶)
Relative Savings	41.3%	41.3%	41.3%	41.2%	41.3%
Air Source Heat Pump A					
Total Energy Input Btu	45.3(10 ⁶)	61.9(10 ⁶)	1.98(10 ⁶)	88(10 ⁶)	65.2(10 ⁶)
Relative Savings	19.4%	22.8%	10.6%	28.5%	27.1%
Air Source Heat Pump C					
Total Energy Input Btu	55.1(10 ⁶)	75.5(10 ⁶)	2.35(10 ⁶)	108(10 ⁶)	79.7(10 ⁶)
Relative Savings	33.8%	36.7%	24.7%	41.8%	40.4%

*Energy represents energy input at power plant for electrical power plus higher heating value of fuel for furnace input.

Table 12: Energy* Comparison for Cooling Well Insulated Typical Home
Percentage Savings Utilizing Proposed Modified Water Source
Heat Pump A With Pre-Water Heat Exchanger -- Alternative I

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
Proposed Water Source Heat Pump System					
Total Energy Input Btu	22.8(10 ⁶)	10.0(10 ⁶)	42.4(10 ⁶)	10.0(10 ⁶)	18.6(10 ⁶)
Air Source Heat Pump A					
Total Energy Input Btu	27.3(10 ⁶)	11.7(10 ⁶)	50.4(10 ⁶)	11.8(10 ⁶)	22.4(10 ⁶)
Relative Savings	16.5%	14.5%	15.9%	15.3%	17%
Air Source Heat Pump C					
Total Energy Input Btu	37.3(10 ⁶)	13.3(10 ⁶)	57.7(10 ⁶)	13.5(10 ⁶)	25.7(10 ⁶)
Relative Savings	38.9%	24.8%	26.5%	25.9%	27.6%

*Energy represents energy input at power plant for electrical power.

Table 13: Energy* Comparison for Heating and Cooling Well Insulated Typical Home
Percentage Savings Utilizing Proposed Modified Water Source Heat Pump A
With Pre-Water Heat Exchanger -- Alternative II

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
Proposed Water Source Heat Pump System					
Total Energy Input Btu	4.75(10 ⁷)	5.79(10 ⁷)	1.15(10 ⁷)	7.56(10 ⁷)	5.95(10 ⁷)
Resistance Electric Heating and Air Source Heat Pump C Cooling					
Total Energy Input Btu	1.47(10 ⁸)	1.5(10 ⁸)	6.3(10 ⁷)	2.04(10 ⁸)	1.69(10 ⁸)
Relative Savings	67.7%	61.4%	81.7%	62.5%	64.8%
Gas and Oil Heating and Air Source Heat Pump C Cooling					
Total Energy Input Btu	9.95(10 ⁷)	9.48(10 ⁷)	6.07(10 ⁷)	1.2(10 ⁸)	1.07(10 ⁸)
Relative Savings	52.3%	38.9%	81%	37%	44.4%
Air Source Heat Pump A					
Total Energy Input Btu	7.26(10 ⁷)	7.36(10 ⁷)	5.24(10 ⁷)	9.98(10 ⁷)	8.76(10 ⁷)
Relative Savings	34.6%	21.3%	78.1%	24.2%	32.1%
Air Source Heat Pump C					
Total Energy Input Btu	9.24(10 ⁷)	8.88(10 ⁷)	6.0(10 ⁷)	1.22(10 ⁸)	1.05(10 ⁸)
Relative Savings	48.6%	34.8%	80.8%	38%	43%

*Energy represents energy input at power plant for electrical power plus higher heating value of fuel for furnace input.

Table 14: Energy* Comparison for Heating Well Insulated Typical Home
Percentage Savings Utilizing Proposed Modified Water Source
Heat Pump A With Pre-Water Heat Exchanger -- Alternative II

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
Proposed Water Source Heat Pump System					
Total Energy Input Btu	42.5(10 ⁶)	55.7(10 ⁶)	2.06(10 ⁶)	73.4(10 ⁶)	55.4(10 ⁶)
Electric Resistance Heating					
Total Energy Input Btu	110(10 ⁶)	144(10 ⁶)	5.34(10 ⁶)	190(10 ⁶)	143(10 ⁶)
Relative Savings	61.4%	61.4%	61.4%	61.4%	61.3%
Gas and Oil Heating					
Total Energy Input Btu	62.2(10 ⁶)	81.5(10 ⁶)	3.0(10 ⁶)	107(10 ⁶)	81.0(10 ⁶)
Relative Savings	31.7%	31.7%	31.4%	31.4%	31.6%
Air Source Heat Pump A					
Total Energy Input Btu	45.3(10 ⁶)	61.9(10 ⁶)	1.98(10 ⁶)	88(10 ⁶)	65.2(10 ⁶)
Relative Savings	6.2%	10%	--	16.6%	15%
Air Source Heat Pump C					
Total Energy Input Btu	55.1(10 ⁶)	75.5(10 ⁶)	2.35(10 ⁶)	108(10 ⁶)	79.7(10 ⁶)
Relative Savings	22.9%	26.2%	12.3%	32%	30.5%

*Energy represents energy input at power plant for electrical power plus higher heating value of fuel for furnace input.

Table 15: Energy* Comparison for Cooling Well Insulated Typical Home
Percentage Savings Utilizing Proposed Modified Water Source
Heat Pump A With Pre-Water Heat Exchanger -- Alternative II

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
	Proposed Water Source Heat Pump System				
Total Energy Input Btu	5.0(10 ⁶)	2.2(10 ⁶)	9.4(10 ⁶)	2.2(10 ⁶)	4.1(10 ⁶)
	Air Source Heat Pump A				
Total Energy Input Btu	27.3(10 ⁶)	11.7(10 ⁶)	50.4(10 ⁶)	11.8(10 ⁶)	22.4(10 ⁶)
Relative Savings	81.7%	81.2%	81.3%	81.4%	81.7%
	Air Source Heat Pump C				
Total Energy Input Btu	37.3(10 ⁶)	13.3(10 ⁶)	57.7(10 ⁶)	13.5(10 ⁶)	25.7(10 ⁶)
Relative Savings	86.6%	83.5%	83.7%	83.7%	84%

*Energy represents energy input at power plant for electrical power.

Table 16: Energy* Comparison for Heating and Cooling Well Insulated Typical Home
Percentage Savings Utilizing Proposed Modified Water Source Heat Pump A
With Pre-Water Heat Exchanger -- Alternative III

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
Proposed Water Source Heat Pump System					
Total Energy Input Btu	4.66(10 ⁷)	3.29(10 ⁷)	6.07(10 ⁷)	3.88(10 ⁷)	4.5(10 ⁷)
Resistance Electric Heating and Air Source Heat Pump C Cooling					
Total Energy Input Btu	1.47(10 ⁸)	1.5(10 ⁸)	6.3(10 ⁷)	2.04(10 ⁸)	1.69(10 ⁸)
Relative Savings	68.3%	78%	3.6%	81%	73.4%
Gas and Oil Heating and Air Source Heat Pump C Cooling					
Total Energy Input Btu	9.95(10 ⁷)	9.48(10 ⁷)	6.07(10 ⁷)	1.2(10 ⁸)	1.07(10 ⁸)
Relative Savings	53.2%	65.3%	--	67.7%	57.9%
Air Source Heat Pump A					
Total Energy Input Btu	7.26(10 ⁷)	7.36(10 ⁷)	5.24(10 ⁷)	9.98(10 ⁷)	8.76(10 ⁷)
Relative Savings	35.8%	55.3%	-15.8%	61.1%	48.6%
Air Source Heat Pump C					
Total Energy Input Btu	9.27(10 ⁷)	8.88(10 ⁷)	6.0(10 ⁷)	1.22(10 ⁸)	1.05(10 ⁸)
Relative Savings	49.7%	62.9%	-1.2%	68.2%	57.1%

*Energy represents energy input at power plant for electrical power plus higher heating value of fuel for furnace input.

Table 17: Energy* Comparison for Heating Well Insulated Typical Home
Percentage Savings Utilizing Proposed Modified Water Source
Heat Pump A With Pre-Water Heat Exchanger -- Alternative III

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
Proposed Water Source Heat Pump System					
Total Energy Input Btu	14.2(10 ⁶)	18.6(10 ⁶)	6.9(10 ⁵)	24.5(10 ⁶)	18.5(10 ⁶)
Electric Resistance Heating					
Total Energy Input Btu	110(10 ⁶)	144(10 ⁶)	53.4(10 ⁵)	190(10 ⁶)	143(10 ⁶)
Relative Savings	87%	87%	87%	87%	87%
Gas and Oil Heating					
Total Energy Input Btu	62.2(10 ⁶)	81.5(10 ⁶)	30.2(10 ⁵)	107(10 ⁶)	81(10 ⁶)
Relative Savings	77.2%	77.2%	77.2%	77.1%	77.2%
Air Source Heat Pump A					
Total Energy Input Btu	45.3(10 ⁶)	61.9(10 ⁶)	19.8(10 ⁵)	88(10 ⁶)	65.2(10 ⁶)
Relative Savings	68.7%	70%	65.2%	72.2%	71.6%
Air Source Heat Pump C					
Total Energy Input Btu	55.1(10 ⁶)	75.5(10 ⁶)	23.5(10 ⁵)	108(10 ⁶)	79.7(10 ⁶)
Relative Savings	74.2%	75.4%	70.6%	77.3%	76.8%

55

*Energy represents energy input at power plant for electrical power plus higher heating value of fuel for furnace input.

Table 18: Energy* Comparison for Cooling Well Insulated Typical Home
 Percentage Savings Utilizing Proposed Modified Water Source
 Heat Pump A With Pre-Water Heat Exchanger -- Alternative III

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
	Proposed Water Source Heat Pump System				
Total Energy Input Btu	32.4(10 ⁶)	14.3(10 ⁶)	60(10 ⁶)	14.3(10 ⁶)	26.5(10 ⁶)
	Air Source Heat Pump A				
Total Energy Input Btu	27.3(10 ⁶)	11.7(10 ⁶)	50.4(10 ⁶)	11.8(10 ⁶)	22.4(10 ⁶)
Relative Savings	-18.7%	-22.2%	-19%	-21.2%	-18.3%
	Air Source Heat Pump C				
Total Energy Input Btu	37.3(10 ⁶)	13.3(10 ⁶)	57.7(10 ⁶)	13.5(10 ⁶)	25.7(10 ⁶)
Relative Savings	13.1%	-7.5%	-3.9%	-5.9%	-3.1%

*Energy represents energy input at power plant for electrical power.

IV. EXPECTED ECONOMICS

In general, economics for the heat pump centered integrated community energy system are expected to be superior to present systems. Comparisons are made with gas, oil, and electric systems. For comparisons with electric systems (air source heat pumps or resistance heating plus air conditioning) both operating costs and capital costs are favorable. In comparison with oil or gas heating and air conditioning, operating costs vary with fuel costs as a function of location. All comparisons include air conditioning as the proposed system utilizes air conditioning. Some parts of the country, especially with respect to residential buildings, have not experienced 30% saturation yet on air conditioning.² No comparisons have been made with heating alone.

The economic effect of lower capacity requirements and load leveling on power plants has not been included. Alternative II (direct cooling) in particular can cut summer peak loads by over 30% (reducing air conditioning loads by 60%) in belts across the Midsouth. As a result capital outlays are reduced and plant productivity and efficiency increase. These are major favorable economic factors for the proposed system and alternatives.

The proposed system has different capital costs for the heating and cooling system in the home as compared with small business and commercial installations. In the home or small commercial establishment a furnace and air conditioner or an air source heat pump is replaced by a water source heat pump. The basic unit is a complete package with an integrated tube-in-tube heat exchanger. This system is expected to be owner-owned. In commercial systems designed by owner and the engineering con-

sultant, the present units, mainly chillers, are already water-source and can be converted to the proposed water supplies. In addition, a redesign of the system can include heating. The need for a furnace is eliminated. The requirement for cooling towers is eliminated, but they could be used to reject excess heat in southern parts of the country or in areas where buildings have large internal loads. Again, the internal system is privately owned. Based on these variations, it is assumed that, for economic analysis, capital costs are proportional to cost of the 3-ton units used in small buildings. Some errors will arise, especially in larger buildings where numerous combinations of internal loads and HVAC systems exist. A 3-ton water source unit operating in this system provides well over 4 tons of heating and 3 tons of cooling. It will meet the load capacities for the well-insulated building for both heating and cooling in over 90% of the country. In addition, the installed price for furnace plus central air conditioner is assumed to be the same as the installed price for air source heat pump C.

Capital costs of installed units for heating and cooling are therefore taken from Christian.^{10,11} It should be noted the installed price of a water source heat pump is basically one-half that of an installed air source heat pump. This results from the use of 1 packaged unit rather than 2 packaged units. The basic water source heat pump is also simpler and smaller than the air source heat pump. A surcharge of \$300 has been added to the air source heat pump A for exceptionally high quality. Few of these units are in service. The capital cost for the installed heating and cooling equipment used for the economic analysis is:

Furnace and Air-Conditioner	\$2500
Air Source Heat Pump A	2800
Air Source Heat Pump C	2500
Water Source Heat Pump A	1300
Modified Water Source Heat Pump A (all 3 alternatives)	1600

In addition, for the proposed system wells, pumping systems, energy addition, and a water distribution system are required. The installed capital costs for each of these components are included in the analysis section IX. The capital costs per module are:

Well Construction (2 wells)	\$ 3000	
Pumping System (pump, motor and controls)(2 systems)	6000	
Capacitance Tank (2 tanks)	260	
Solar Collector (1000 ft ² at \$3/ft ²)	3000	
Water Distribution System (primary and secondary)		
1000 ft. of Double 6-inch Main	\$9200	
1000 ft. of Double 1½-inch Secondary	3400	
Trench (2000 ft. at \$1.47/ft.)	<u>2940</u>	<u>15540</u>
Total Water System Capital Cost		\$27800
Total Cost per Home in a 20-Home Module		\$ 1390

Annual capital cost is based on an effective 10% interest rate plus
12
a sinking-fund depreciation over 20 years at the same 10% annual rate.

It must be noted interest rates vary and that public utilities and private utilities have access to different rates of interest. All equipment except solar collectors is expected to have lifetimes in excess of 20 years. The relatively small cost of energy addition makes the 20-year depreciation rate reasonable. Utilizing this criterion, the annual capital cost is 10% interest plus 1.7% depreciation for a total of 11.7% of the total capital cost. The annual capital cost for the system per home is:

	<u>Total Capital</u>	<u>Annual Cost</u>
Furnace and Air Conditioner	\$2500	\$292.50
Air Source Heat Pump A	2800	327.60
Air Source Heat Pump C	2500	292.50
Water Source Heat Pump A	2690	314.73
in Proposed System		
Modified Water Source Heat Pump	2990	349.83
A Plus System (Includes all 3 alternatives)		

Annual maintenance costs based on Christián are \$165 for both the air source and water source heat pumps for the first 5 years.^{10,11} After this period, maintenance costs are expected to increase at the rate of 7% per year. Due to simplicity and a low pressure ratio, the water source heat pumps should be expected to have a lower maintenance cost. Based on this assumption, maintenance costs for both the air source heat pumps and the proposed system are assumed to be an average of 25% higher than the \$165 over the 20 years. An average annual maintenance cost of \$205 is used in the economic analysis. Assuming water source heat pump maintenance is 25% less than that of an air source unit allows the utility \$1020 per year maintenance for 20 homes for the water distribution system and wells for one module. Resistance heating and furnace plus the air conditioner is given an annual maintenance cost of \$150 as compared with heat pump maintenance. The energy required for water pumping is included in the water source heat pump energy requirements.

Tables 19 to 21 show the annual energy cost and annual total cost for various heating and cooling systems in 5 cities. Total cost is energy cost, capital cost, and maintenance cost combined. Electricity costs vary from \$.04 to \$0.12 per Kw/hr and fuel cost is set at \$2.50 per 10^6 Btu.

A gas and oil cost of \$3.50 to \$4.50 per 10^6 Btu is reasonable in some parts of the country which represents a much higher cost for gas and oil heating energy in the comparisons. In turn the energy cost for gas and oil heating is much higher than the tables indicate.

In general, the costs for the proposed systems and alternatives are favorable on both a fuel and a capital cost basis.

Table 19: Economic Analysis for Annual Heating and Cooling for Well Insulated Typical Home
(\$0.04 Per Kw-hr Electric Cost)

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
Resistance Electric Heating and Air Source Heat Pump C Cooling					
Power Input Kw-hr	18.6(10 ³)	16.2(10 ³)	6.5(10 ³)	20.6(10 ³)	17.3(10 ³)
Annual Electric Cost	\$744	\$648	\$260	\$836	\$692
Total Annual Cost	\$1187	\$1091	\$703	\$1279	\$1135
Gas and Oil Heating and Air Source Heat Pump C Cooling (\$2.50 Per 10 ⁶ Btu Fuel Cost)					
Combustion Energy	62.2(10 ⁶)	81.5(10 ⁶)	30.2(10 ⁵)	107(10 ⁶)	81(10 ⁶)
Combustion Energy Cost	\$156	\$204	\$76	\$268	\$203
Power Input Kw-hr	3.83(10 ³)	1.37(10 ³)	5.92(10 ³)	1.38(10 ³)	2.64(10 ³)
Electric Cost	\$153	\$55	\$237	\$55	\$106
Annual Energy Cost	\$309	\$259	\$313	\$323	\$309
Total Annual Cost	\$752	\$702	\$756	\$766	\$752
Air Source Heat Pump A					
Power Input Kw-hr	7.5(10 ³)	7.54(10 ³)	5.4(10 ³)	10.2(10 ³)	9.3(10 ³)
Annual Electric Cost	\$300	\$302	\$216	\$408	\$373
Total Annual Cost	\$833	\$835	\$749	\$941	\$906
Air Source Heat Pump C					
Power Input Kw-hr	9.5(10 ³)	9.1(10 ³)	6.2(10 ³)	12.5(10 ³)	10.8(10 ³)
Annual Electric Cost	\$380	\$363	\$248	\$500	\$432
Total Annual Cost	\$878	\$861	\$746	\$998	\$930
Water Source Heat Pump A					
Power Input Kw-hr	6.9(10 ³)	6.6(10 ³)	5.2(10 ³)	8.3(10 ³)	7.6(10 ³)
Annual Electric Cost	\$276	\$264	\$208	\$332	\$304
Total Annual Cost	\$796	\$784	\$728	\$852	\$824
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative I					
Power Input Kw-hr	6.1(10 ³)	5.9(10 ³)	4.5(10 ³)	7.5(10 ³)	6.8(10 ³)
Annual Electric Cost	\$244	\$236	\$180	\$300	\$272
Total Annual Cost	\$799	\$791	\$735	\$855	\$827
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative II					
Power Input Kw-hr	4.9(10 ³)	5.9(10 ³)	1.2(10 ³)	7.8(10 ³)	6.1(10 ³)
Annual Electric Cost	\$196	\$236	\$48	\$312	\$244
Total Annual Cost	\$751	\$791	\$603	\$867	\$799
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative III					
Power Input Kw-hr	4.8(10 ³)	3.4(10 ³)	6.2(10 ³)	3.9(10 ³)	4.6(10 ³)
Annual Electric Cost	\$192	\$136	\$248	\$156	\$184
Total Annual Cost	\$747	\$691	\$803	\$911	\$739

Table 20: Economic Analysis for Annual Heating and Cooling for Well Insulated Typical Home
(\$0.08 Per Kw-hr Electric Cost)

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
Resistance Electric Heating and Air Source Heat Pump C Cooling					
Power Input Kw-hr	18.6(10 ³)	16.2(10 ³)	6.5(10 ³)	20.9(10 ³)	17.3(10 ³)
Annual Electric Cost	\$1488	\$1296	\$520	\$1672	\$1384
Total Annual Cost	\$1931	\$1739	\$963	\$2115	\$1827
Gas and Oil Heating and Air Source Heat Pump C Cooling (\$2.50 Per 10 ⁶ Btu Fuel Cost)					
Combustion Energy	62.2(10 ⁶)	81.5(10 ⁶)	30.2(10 ⁵)	107(10 ⁶)	81(10 ⁶)
Combustion Energy Cost	\$156	\$204	\$76	\$268	\$203
Power Input Kw-hr	3.83(10 ³)	1.37(10 ³)	5.92(10 ³)	1.38(10 ³)	2.64(10 ³)
Electric Cost	\$306	\$110	\$474	\$110	\$211
Annual Energy Cost	\$462	\$314	\$550	\$378	\$414
Total Annual Cost	\$905	\$757	\$993	\$821	\$857
Air Source Heat Pump A					
Power Input Kw-hr	7.5(10 ³)	7.54(10 ³)	5.4(10 ³)	10.2(10 ³)	9.3(10 ³)
Annual Electric Cost	\$600	\$603	\$432	\$816	\$744
Total Annual Cost	\$1133	\$1136	\$965	\$1349	\$1277
Air Source Heat Pump C					
Power Input Kw-hr	9.5(10 ³)	9.1(10 ³)	6.2(10 ³)	12.5(10 ³)	10.8(10 ³)
Annual Electric Cost	\$760	\$728	\$496	\$1000	\$864
Total Annual Cost	\$1258	\$1226	\$994	\$1498	\$1362
Water Source Heat Pump A					
Power Input Kw-hr	6.9(10 ³)	6.6(10 ³)	5.2(10 ³)	8.3(10 ³)	7.6(10 ³)
Annual Electric Cost	\$552	\$528	\$416	\$664	\$608
Total Annual Cost	\$1072	\$1048	\$936	\$1184	\$1128
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative I					
Power Input Kw-hr	6.1(10 ³)	5.9(10 ³)	4.5(10 ³)	7.5(10 ³)	6.8(10 ³)
Annual Electric Cost	\$488	\$472	\$360	\$600	\$544
Total Annual Cost	\$1043	\$1027	\$915	\$1155	\$1099
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative II					
Power Input Kw-hr	4.9(10 ³)	5.9(10 ³)	1.2(10 ³)	7.8(10 ³)	6.1(10 ³)
Annual Electric Cost	\$392	\$472	\$96	\$624	\$488
Total Annual Cost	\$947	\$1027	\$651	\$1179	\$1043
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative III					
Power Input Kw-hr	4.8(10 ³)	3.4(10 ³)	6.2(10 ³)	3.9(10 ³)	4.6(10 ³)
Annual Electric Cost	\$384	\$272	\$496	\$312	\$368
Total Annual Cost	\$939	\$827	\$1051	\$867	\$923

Table 21: Economic Analysis for Annual Heating and Cooling for Well Insulated Typical Home (\$0.12 Per Kw-hr Electric Cost)

	BIRMINGHAM	NEW YORK	MIAMI	DETROIT	ST. LOUIS
Resistance Electric Heating and Air Source Heat Pump C Cooling					
Power Input Kw-hr	18.6(10 ³)	16.2(10 ³)	6.5(10 ³)	20.9(10 ³)	17.3(10 ³)
Annual Electric Cost	\$2232	\$1944	\$780	\$2508	\$2076
Total Annual Cost	\$2675	\$2387	\$1223	\$2951	\$2519
Gas and Oil Heating and Air Source Heat Pump C Cooling (\$2.50 Per 10 ⁶ Btu Fuel Cost)					
Combustion Energy	62.2(10 ⁶)	81.5(10 ⁶)	30.2(10 ⁵)	107(10 ⁶)	81(10 ⁶)
Combustion Energy Cost	\$156	\$204	\$76	\$268	\$203
Power Input Kw-hr	3.83(10 ³)	1.37(10 ³)	5.92(10 ³)	1.38(10 ³)	2.64(10 ³)
Electric Cost	\$460	\$164	\$710	\$165	\$317
Annual Energy Cost	\$616	\$368	\$786	\$433	\$520
Total Annual Cost	\$1059	\$811	\$1229	\$876	\$963
Air Source Heat Pump A					
Power Input Kw-hr	7.5(10 ³)	7.54(10 ³)	5.4(10 ³)	10.2(10 ³)	9.3(10 ³)
Annual Electric Cost	\$900	\$905	\$648	\$1224	\$1116
Total Annual Cost	\$1433	\$1438	\$1181	\$1757	\$1649
Air Source Heat Pump C					
Power Input Kw-hr	9.5(10 ³)	9.1(10 ³)	6.2(10 ³)	12.5(10 ³)	10.8(10 ³)
Annual Electric Cost	\$1140	\$1092	\$744	\$1500	\$1296
Total Annual Cost	\$1638	\$1590	\$1242	\$1998	\$1794
Water Source Heat Pump A					
Power Input Kw-hr	6.9(10 ³)	6.6(10 ³)	5.2(10 ³)	8.3(10 ³)	7.6(10 ³)
Annual Electric Cost	\$828	\$792	\$624	\$996	\$912
Total Annual Cost	\$1348	\$1312	\$1144	\$1516	\$1432
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative I					
Power Input Kw-hr	6.1(10 ³)	5.9(10 ³)	4.5(10 ³)	7.5(10 ³)	6.8(10 ³)
Annual Electric Cost	\$732	\$708	\$540	\$900	\$816
Total Annual Cost	\$1287	\$1263	\$1095	\$1455	\$1371
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative II					
Power Input Kw-hr	4.9(10 ³)	5.9(10 ³)	1.2(10 ³)	7.8(10 ³)	6.1(10 ³)
Annual Electric Cost	\$588	\$708	\$144	\$936	\$732
Total Annual Cost	\$1143	\$1263	\$699	\$1491	\$1287
Modified Water Source Heat Pump A With Pre-Water Heat Exchanger -- Alternative III					
Power Input Kw-hr	4.8(10 ³)	3.4(10 ³)	6.2(10 ³)	3.9(10 ³)	4.6(10 ³)
Annual Electric Cost	\$576	\$408	\$744	\$468	\$552
Total Annual Cost	\$1131	\$963	\$1299	\$1023	\$1107

V. EXPECTED ENVIRONMENTAL IMPACTS

All environmental impacts above the ground are expected to be either favorable or of minimal adverse proportions. The largest impact is the decrease in air pollution resulting from the decrease in use of fossil fuels. Some environmental changes may occur in aquifers. Thermal cycling may affect interstitial minerals in aquifers, contaminate ground water, and change ground water temperature thereby accelerating biological growth. None of these effects is expected to be of major consequence.

Since the pumping stations and well housings are designed for the right-of-way next to roads in rural areas similar to electric transformer stations, use of sizable quantities of land surface is not required. In metropolitan areas pumping stations are expected to be under sidewalks with pipelines following the same underground tunnels as other utilities. Thus the loss of the land resource would be inconsequential.

Resources consumed in construction would be lost, yet the savings in energy requirements for the system would provide an offsetting benefit. Electrical energy consumed in the operation of the system would, again, result in a favorable benefit-to-cost ratio when compared with other systems. Failures in the system should be primarily mechanical in nature, although leakages could occur and cause temporary adverse effects. The eventual disposal of system components would have little effect on the environment. Much of the equipment would be salvable.

The greatest effect is to conserve energy and reduce associated environmental effects proportionally. Industrial and urban areas contribute atmospheric contamination not only in the form of particulate and gaseous substances, but also in the form of temperature changes. These

effects have been shown to modify weather by alteration of rainfall and temperature patterns.^{13,14} Reducing energy input and storing thermal energy during warm periods could decrease the adverse effects caused by heat rejection in urban areas. The same would apply to decreased energy usage during cold periods. The reduced use of oil and gas would more than offset the resources lost in retrieval of energy from aquifers.

Contamination of aquifers used for potable water can be avoided. It is anticipated the aquifer system will be confined to a site removed a considerable distance from any other private or public source. In Alabama, health codes allow septic tanks to be placed within 150 ft. of potable wells. In addition, the closed system will minimize oxygen intake. Diaphragm-capacitance tanks are recommended to eliminate water-to-air contact. However, the injection of warm water and some air could cause biological growth. Testing biological contamination in the first demonstration projects should be considered to provide a basis for treatment to eliminate this potential problem.

The water table is not significantly affected inasmuch as the same quantity of water is injected as withdrawn. During cooling cycles some water may be discarded to maintain thermal equilibrium.

Of importance could be the precipitation of salts which could clog aquifers and piping. Some salts increase in solubility as temperature drops (mainly carbonates), but others increase in solubility as temperature increases. The temperature range employed in the proposed systems ($\pm 32.5^{\circ}\text{F}$) is here considered to be insignificant from a geochemical standpoint. The impact of changing solubility of salts in aquifers, mostly Ca and Mg salts, at the anticipated temperature ranges could be measured only in terms of geologic time.

The carbonate of calcium is the most predominant carbonate in nature and, obviously, is the primary constituent in limestone or dolomite aquifers. Also, carbonate may be a predominant substance in interstitial spaces in other aquifers, notably sandstones. These considerations should apply to most aquifers.

Calcium carbonate is dissolved by weak acids by the following:

and, limestone dissolves by the equation:

which is an equilibrium situation where the reaction can go in either direction.

pH is a controlling factor inasmuch as the hydrogen ion concentration affects the equation first noted above. At low pH, the forward action results in solution, whereas at high pH the reverse action would occur leading to precipitation.

A more dominant situation relates to the pressure of CO_2 above the solution. This determines the concentration of dissolved H_2CO_3 by the relationship:

Any process increasing the available CO_2 promotes dissolution; decreasing the amount of CO_2 promotes precipitation.

The above has a bearing on the effects of temperature on the solubility of CaCO_3 in natural processes. The solubility of CaCO_3 in pure water "...decreases somewhat as the temperature rises." This is opposite to the behavior of most salts.

CO_2 , like any other gas, is less soluble in hot water than in cold

water. For instance, in oceans CaCO_3 at depth is dissolved, whereas at warm surface temperatures there may be precipitation.

It is considered that a temperature increase of possibly 25 to 35°F in return water to an aquifer would not result in any change in the chemical makeup of the circulation water, nor would it affect the physical parameters of the aquifer. The pH of "pure" water is 7.0, but if exposed to the atmosphere for a short time will be near 5.7 as a result of the intake of CO_2 from the atmosphere (unless neutralized by other things). Any process in the pumping and circulation of water in a heat pump system that affects pH might affect the solubility of both carbonates and salts. This should be minor.

Salts such as NaCl , K_2SO_4 , and $\text{Ba}(\text{NO}_3)_2$ dissolve in water to form neutral solutions. Others such as iron sulfate may hydrolyze leaving an acidic solution. Granted these changes will occur in nature, the introduction of a minimal temperature escalation should have no significant effect on the nature of the aquifer nor that of the circulating water. Silica, the constituent of sandstones, is dissolved by a process of forming a gel--in nature this may require thousands of years to be perceptible.

Laboratory studies of the solubility of chemical substances have shown a requirement for high temperature increments over extended periods to realize any effects of temperature. In studies involving the injection of liquid wastes into core samples under simulated reservoir conditions of temperature and pressure--i.e. 200°F and 600 psi, no difference in the permeability to flow was noted when compared with normal temperatures and pressures (surface conditions).

In summary, any change in aquifers under conditions of nominal increases in temperature of the return water--anything in the range of 25°, 35° or even 50° is expected to be negligible.

For the most part, environmental changes are expected to be positive; minor adverse effects can be dealt with or justified in light of the beneficial aspects of the system performance.

VI. PROJECTED GROWTH

Using the module concept, this version of the heat pump centered integrated community energy system can start in any populated area and expand to cover the entire community. The system is applicable to homes, businesses, major buildings, etc.

In new construction areas, codes can be developed requiring primary water distribution systems and wells. As there is additional population expansion in an area, additional wells can be used to increase capacity. An additional set of wells will not only provide the increased requirements for one module, but also can be used to provide the increased capacity requirements for a number of modules. This type growth results from the specification of 6-inch lines in the primary water distribution system.

As fuel prices and electricity prices increase, local populated areas can individually convert to this system. Conversion of internal centrally controlled cooling and heating systems is minor. In a home with a furnace and air conditioning, or heat pump, the heat exchangers and blowers in the duct work can be replaced by a water source heat pump. The conversion includes plumbing, pump installation, and minor ducting changes. In larger buildings, most chillers already use water sources from cooling towers. This system replaces the water source with water of more favorable temperature. It also permits some chiller systems to be converted directly to heating, eliminating gas, coal or oil furnaces, or boilers. This conversion can be of appreciable significance.

Additional wells can be added in a number of modules to provide water to areas not located above or near adequate aquifers. Additional pumps would be required in the primary distribution system for the transfer

of water. Based on the capacity of present urban water distribution systems, the minimum 6-inch lines would be sufficient for most areas. This can be deduced from pressure-loss curves shown in Figures 8 and 9. Heavily populated areas will require larger, separate mains.

Since the water system will be available to buildings, the development of water-source heat pumps for hot water heating is a possibility. This demand can arise once sufficient systems have been installed. Additional supplementary heating will have to be provided to the overall system to meet the hot water demand. Annual storage allows energy addition at any time of the year. The community energy concept allows other additions such as refrigeration in food and dairy stores, etc.

Growth would be expected to increase as water-source heat pumps become more efficient and some of the identified alternatives are evaluated. The basic system provides for these modifications without major changes.

The system has been devised so the projected growth can be similar to expansion in use following the introduction of natural gas into the American economy. Increased natural gas prices and other energy prices could accelerate this process.

VII. IDENTIFIED VARIATIONS

A basic concept in using the proposed heat pump system is that essentially 100% of the system is comprised of off-the-shelf components. A number of variations may be used to upgrade the performance of the basic system through heat pump system changes and through different control variations. The first 3 alternatives add a heat exchanger and new controls. A second set of variations involves the design of basic water distribution systems between aquifers and the heat pump systems. These systems are not analyzed for performance due to either loss in performance or higher capital and operating costs. The third set of variations employs abandoned deep mines or caverns for energy storage rather than aquifers.

The first 3 alternatives use a water-to-air (water-to-water) heat exchanger before the water enters the water source heat pump. A schematic of the system is shown in Figure 14.

The water temperature range in the first alternative is the same as that proposed for the basic system, 60°F for cooling and 80°F for heating. It should be noted that incoming water temperature is below room temperature for cooling and warmer than room temperature for heating. Therefore the air can be either precooled or preheated before reaching the heat pump. As a result some heat transfer occurs without compressor input thereby increasing overall efficiency. The change in air temperature and water temperature reaching the heat pump decreases heat pump efficiency. However, overall system capacity and performance is increased.

Two methods of control are envisioned for this system:

1. The heat pump and water-to-air (water-to-water) heat exchanger operate simultaneously at all times, and

2. The heat exchanger operates individually for heating until outside temperature reaches a set point and then the water-to-air (water-to-water) heat exchanger and heat pump both operate.

This mode of operation is not effective for cooling since temperatures sufficiently low for dehumidification are required.

Performance tables for this alternative are all based on simultaneous operation. This system is evaluated for performance and economics in the respective sections (Alternative I).

The second alternative uses a different type of control but uses the same hardware as the first alternative (Figure 14). The water range is 60°F to 40°F rather than the 80°F to 60°F range. For heating the performance drops by approximately 10% as a result of the lower-temperature water. However, for cooling direct heat exchange is available, and the performance more than doubles. Only fan power and pumping power are required; neither the heat pump cycle nor compressor is utilized for cooling except as a possible backup. This system on a long-term basis would conserve the most energy (Alternative II).

A third alternative is to reverse the temperature process and to use 100°F to 80°F water. In this case the cooling performance with 90°F water is decreased; however the performance for heating is appreciably higher. Since all the heating is sensible, higher airflow is required thereby increasing fan power to provide sufficient airflow. The higher temperatures also make waste heat addition more difficult and partially limit the system to areas where cooling load to heating load is sufficient to eliminate waste heat addition (Alternative III).

In addition to the proposed heat pump system, 4 additional heat pump water distribution systems were considered.

The first 2 systems use individual water source heat pumps in the

controlled environments. The second 2 systems use a central heat pump and transmit chilled and hot water. The piping sequences are shown in Figures 17 to 20.

The first 2 systems were eliminated due to performance losses associated with some loss of temperature control of the water. Also, successive heat pumps would receive degraded-temperature water. Separate provisions for heating and cooling are not available.

The central heat pump systems are eliminated for the following reasons:

1. There is some question on the reliability of the required insulation on pipes over a long period of time (in excess of 20 years).
2. Piping costs are appreciably higher (Table 43). Also the average distance water travels is higher requiring larger pipes and/or more pumping power.
3. Operating personnel are required at extra expense.
4. Land must be acquired for central units.
5. Land, building, and central units require more capital than individual units in per-unit output.
6. Performance of central units is not much higher than that of individual units.

An alternative to aquifer storage is storage in abandoned mines or caverns. In these systems separate mines or caverns are required for cold and warm water to prevent energy degradation. In addition, the water probably must be transported greater distances, requiring more pump power, since mines are not available locally in most urbanized areas. However, this is a viable alternative in areas of sub-standard aquifers.

FIG. 17 1. PIPE SYSTEM WITH PUMP CIRCULATING WATER THROUGH HEAT EXCHANGER
(NO INSULATION ON THE PIPES)

Dillon
20-11-78

FIG. 18

2 PIPE SYSTEM
(NO INSULATION ON THE PIPES)

Stidman
204-78

FIG. 19 3 PIPE SYSTEM -- CENTRAL HEATING AND CHILLING
(ALL PIPES INSULATED)

Thurman
20-11-78

FIG. 20 4 PIPE SYSTEM -- CENTRAL HEATING AND CHILLING
(ALL PIPES INSULATED)

D. J. Jones
20-11-78

VIII. COMPONENT TESTING REQUIREMENTS

Testing of actual components in the proposed system is not required as all components are off-the-shelf items. Testing of heat-pump systems identified as alternatives and experimentally evaluating a dual-well aquifer storage system in the laboratory and on a full-scale basis is feasible. Some tests could be conducted on well-water effects on heat exchangers, however appreciable research in this area is already being conducted. An option is to try to influence the direction of some of these projects.

The auxillary direct heat transfer systems in series or parallel with the heat pump show the best performances. Some evaluation tests producing performance data for the first three alternatives is desired.

Some testing should be applied to the aquifer system to determine actual temperature distributions and efficiency of the energy (both hot and cold) return. 2 real possibilities are considered.

1. Tests can be made on a scaled-down sand bed in the laboratory. Dimensionless numbers relating velocities, temperature, energy, porosity, permeability, specific heats, and mass can be developed and tested for applicability.
2. A number of probes should be included in a 2-well demonstration project to confirm temperature distribution in actual utilization. The buoyancy effect of the warm water is a major question. The results of injection and return from one well system near Mobile, Alabama, has shown an energy efficiency of 59%. The proposed 2-well system is expected to have a higher efficiency which will improve as the system attains annual equilibrium.

A small water source heat pump could be utilized to provide hot and cold water circulation in both laboratory and field testing.

IX. ANALYSIS DESCRIPTIONS

The following techniques were used to make the analysis in the preceding sections. The sequence corresponds with the sequences in the previous sections.

IX.A. Controlled Environmental System - Heat Pump Performance

The controlled environment includes homes, apartments, businesses, malls, major buildings, etc. A study has been made to show the variations of loads in typical neighborhoods and apartments. In general the loads are directly related to floor area with variations associated with insulation, infiltration, usage, location such as inside apartments, etc. A typical space of 2000 ft.² has been chosen as a basis for energy evaluations around the country. The insulation values are based on the TVA superinsulator package which also corresponds to the ASHRAE recommendations for most of the northern parts of the country. TVA recommends use of the insulation values throughout Alabama. The basic insulation for this package is:

- R-33 for ceiling insulation
- R-19 for walls
- R-19 for floors
- Double pane glass on all windows
- One infiltration change per hour

Using the basic ASHRAE load calculation techniques and converting to an annual load using degree-days, the annual heating and cooling loads were calculated for selected cities throughout the country. These are listed in Tables 5 and 6 in the performance section III. The degree-days are based on an equilibrium temperature of 65°F, allowing for a normal internal load. The degree-days can be changed to any equilibrium temperature by multiplying the degree change by 365 and subtracting and adding to the corresponding heating and cooling degree-days. For larger buildings equilibrium tempera-

tures, zero heating or cooling, often move down to the 40°F to 50°F range. Values are possible in the freezing range. In these type buildings, the heating and cooling systems are individually designed and can use outside ambient air for cooling at the lower temperatures. Due to variation in systems in each building and most chilling systems are already water-source units, little direct analysis is made of these buildings. It is assumed the new variations in water temperatures will show an increase in performance for most of these systems. In the heating mode most of these buildings can be converted from oil, coal, or gas to heat pumps using much of the same equipment used for cooling. The design change will be of major proportions.

For performance evaluation, not loads, a computer program has been developed to predict annual performance as a function of temperature (Appendix I). The input to the program for 24 cities is NOAA weather data showing average annual hours in 5°F temperature increments (Table 22); 99% design temperatures, internal load and heat pump performance data (Tables 23 to 25); and the heating and cooling loads at ASHRAE design temperatures. The loads are matched to heat pump capacities rather than actual loads. The output is a measure of performance only. The internal load allows the program to be used for larger buildings. Heating loads above heat-pump capacity are met with the addition of resistance heating. Runs were made for a variation of heat pumps. The first 2 studies with the standard internal load of 5000 Btu use air source heat pumps A and C to indicate performance with high and normal performance pumps with additional resistance heating (Tables 26 and 27). The use of twin units doubles pump capacities of air source heat pumps A and C which have been

Table 22

ANNUAL HOURS IN FIVE DEGREE FAHRENHEIT TEMPERATURE INCREMENTS FOR TWENTY-FOUR CITIES*

Cities, States	°F	-18	-13	-8	-3	2	7	12	17	22	27	32	37	42	47	52	57	62	67	72	77	82	87	92	97	102	107	112	117
Birmingham, AL		0	0	0	0	0	0	3	6	17	69	143	292	433	528	614	668	742	805	908	1138	924	658	488	264	62	5	0	0
Tucson, AZ		0	0	0	0	0	0	0	0	0	2	25	98	248	417	598	716	800	763	781	870	959	777	656	520	357	152	31	0
Los Angeles, CA		0	0	0	0	0	0	0	0	0	0	0	0	10	107	428	1054	1904	2193	1654	881	380	117	28	7	4	1	0	0
Denver, CO		1	0	1	6	22	36	78	119	216	359	553	721	717	692	704	678	731	783	684	549	437	332	236	103	10	1	0	0
Miami, FL		0	0	0	0	0	0	0	0	0	0	0	0	4	26	71	147	277	452	810	1708	2463	1795	888	125	2	0	0	0
Atlanta, GA		0	0	0	0	0	0	2	8	19	44	112	271	468	598	676	735	784	823	926	1185	883	625	401	177	29	2	0	0
Hilo, HI		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	11	361	2272	3001	2126	975	24	0	0	0	0	0
Chicago, IL		0	0	3	12	25	59	85	117	196	335	551	822	800	591	543	569	592	653	769	762	563	370	220	105	27	2	0	0
New Orleans, LA		0	0	0	0	0	0	0	0	0	2	9	47	128	282	449	621	692	850	987	1189	1671	979	620	229	12	0	0	0
Boston, MA		0	0	0	1	3	11	35	75	149	254	431	669	851	831	756	765	783	805	819	676	433	245	127	39	10	0	0	0
Detroit, MI		0	0	0	1	4	17	61	131	248	377	618	884	808	595	566	592	633	695	783	721	516	314	148	47	9	0	0	0
Saint Louis, MO		0	0	0	1	7	15	40	77	134	212	411	650	671	620	578	585	575	646	728	823	763	579	376	198	65	13	1	1
New York, NY		0	0	0	0	0	1	10	26	96	188	330	603	858	838	796	722	745	754	877	926	604	263	96	28	5	0	0	0
Akron, OH		0	0	0	5	11	32	67	153	226	429	646	867	657	630	616	633	686	778	839	675	437	275	112	24	3	0	0	0
Tulsa, OK		0	0	0	0	1	2	12	29	75	159	265	438	535	611	637	622	636	671	752	838	816	649	481	333	149	43	10	1
Portland, OR		0	0	0	0	0	0	1	4	10	40	123	343	772	1238	1271	1274	1316	1001	581	373	211	126	54	23	6	1	0	0
Philadelphia, PA		0	0	0	0	0	0	9	32	100	189	335	654	818	758	701	663	710	735	809	863	655	420	225	74	17	1	0	0
Providence, RI		0	0	0	1	4	9	39	84	163	275	491	729	829	790	763	762	764	822	799	668	434	229	86	25	2	0	0	0
Columbia, SC		0	0	0	0	0	0	0	2	19	56	138	293	411	523	623	673	722	808	895	1113	940	657	489	301	92	11	1	0
Nashville, TN		0	0	0	1	1	3	9	28	67	132	263	463	565	627	619	637	697	738	838	933	814	582	443	227	66	11	1	0
Houston, TX		0	0	0	0	0	0	0	0	2	4	18	64	141	291	452	570	681	772	980	1172	1611	949	677	327	56	1	0	0
Salt Lake City, UT		0	0	0	0	2	16	41	80	158	328	564	798	831	755	685	682	635	614	615	569	447	368	309	196	66	8	0	0
Seattle, WA		0	0	0	0	0	0	0	3	20	39	104	427	914	1408	1445	1462	1272	750	448	258	123	62	24	6	2	0	0	0
Milwaukee, WI		1	2	4	18	47	83	116	176	285	421	659	913	774	611	591	585	634	749	753	597	390	226	96	32	6	0	0	0

* From Summary of Hourly Observations, NOAA, Separate Number For Each City, 10 Year Average

TABLE 23. PERFORMANCE OF AIR-SOURCE HEAT PUMP A*

COOLING									
DBT ¹	WBT ²	TOTAL CAP MBTUH	SENSIBLE CAP @ IDBT ³			INPUT POWER ⁴ KW	COP ⁵		
			72	76	80				
85	59	33.1	28.6	33.2	34.2	3.93	2.47		
	63	34.9	23.1	27.5	32.0	4.03	2.54		
	67	36.7	18.0	22.2	26.4	4.13	2.61		
	71	38.5	13.3	17.2	21.2	4.23	2.67		
95	59	31.4	28.3	31.8	33.1	4.23	2.19		
	63	33.4	22.8	27.4	32.1	4.33	2.26		
	67	35.4	17.6	22.0	26.4	4.43	2.34		
	71	37.4	12.9	17.0	21.1	4.43	2.48		
105	59	29.1	27.6	30.1	31.5	4.43	1.93		
	63	31.3	22.2	27.0	31.5	4.53	2.03		
	67	33.5	17.1	21.7	26.3	4.63	2.12		
	71	35.7	12.3	16.6	21.0	4.73	2.21		
115	59	26.4	26.5	28.0	29.5	4.73	1.64		
	63	28.7	21.4	26.4	29.5	4.83	1.74		
	67	31.1	16.4	21.1	25.9	4.93	1.85		
	71	33.4	11.5	16.0	20.7	5.03	1.95		
HEATING									
IDBT ODT	60		70		75		80		COP ⁵ @ 70°
	CAP MBTUH	KW ⁴	CAP MBTUH	KW ⁴	CAP MBTUH	KW ⁴	CAP MBTUH	KW ⁴	
-8	12.0	2.6	11.0	2.7	10.5	2.7	10.0	2.8	1.19
2	16.0	2.8	14.7	2.9	14.1	3.0	13.5	3.1	1.49
12	20.7	3.1	19.2	3.2	18.4	3.3	17.7	3.3	1.76
22	26.1	3.3	24.4	3.5	23.5	3.6	22.6	3.6	2.04
32	32.0	3.6	30.1	3.7	29.1	3.8	28.1	3.9	2.38
42	37.9	3.8	35.9	4.0	34.8	4.1	33.8	4.2	2.63
52	42.9	4.1	40.8	4.3	39.7	4.4	38.7	4.5	2.78
62	45.8	4.3	43.9	4.6	42.9	4.7	41.9	4.8	2.80
72	46.3	4.5	44.6	4.8	43.8	4.9	43.0	5.0	2.72

¹Outdoor Dry Bulb Temperature, °F²Indoor Wet Bulb Temperature, °F³Indoor Dry Bulb Temperature, °F⁴Power requirement for compressor, outdoor fan, and indoor fan, KW⁵Coefficient of Performance = Capacity (KW)/Input Power⁴ (KW)

*Based on G.E. Pub. No. 22-1009-6, April 1977 for BWR936A1-A @ 1200 cfm
 Cost \$2500 installed per J.E. Christian, Unitary Air-to-Air Heat Pumps,
 ANL/CES/TE77-10 (\$300 surcharge added for high performance system making
 system installed cost \$2800 for cost comparison)

TABLE 24. PERFORMANCE OF AIR-SOURCE HEAT PUMP B*

COOLING							
DBT ¹	WBT ²	TOTAL CAP MBTUH	SENSIBLE CAP @ IDBT ³			INPUT POWER ⁴ KW	COP ⁵
			72	76	80		
85	59	29.4	29.4	29.4	--	4.05	2.13
	63	31.1	23.1	27.4	29.6	4.10	2.22
	67	33.9	14.6	18.9	23.6	4.18	2.38
	71	35.7	--	--	15.2	4.26	2.46
95	59	28.4	28.4	--	--	4.36	1.91
	63	30.2	23.7	28.1	30.2	4.46	1.98
	67	33.2	15.1	19.5	24.2	4.60	2.12
	71	35.8	--	--	15.6	4.70	2.23
105	59	25.6	25.6	--	--	4.54	1.65
	63	27.5	24.2	26.5	27.5	4.62	1.74
	67	30.8	15.7	10.3	25.2	4.82	1.87
	71	31.2	--	--	16.2	5.02	1.82
115	59	22.7	--	--	--	4.64	1.43
	63	24.8	24.8	24.8	24.8	4.78	1.52
	67	28.3	16.2	21.0	26.2	5.04	1.65
	71	31.2	--	--	16.7	5.33	1.72

HEATING									
IDBT ODT	60		70		75		80		COP @ 70°
	CAP	KW	CAP	KW	CAP	KW	CAP	KW	
-18	10.7	2.46	10.4	2.46	10.4	2.46	9.4	2.46	1.24
- 8	12.0	2.66	11.5	2.66	10.9	2.66	10.5	2.66	1.27
2	15.5	2.86	14.7	2.86	14.0	2.88	13.5	2.91	1.51
12	19.0	3.06	18.2	3.06	17.5	3.11	17.0	3.16	1.74
22	23.5	3.26	22.0	3.31	21.5	3.34	21.0	3.36	1.95
32	27.5	3.41	26.6	3.51	25.7	3.56	25.0	3.61	2.22
42	34.0	3.66	32.2	3.76	31.3	3.79	30.5	3.86	2.51
52	40.5	3.86	39.0	3.96	36.3	4.04	37.5	4.11	2.89
62	44.5	4.01	43.7	4.21	42.8	4.28	42.0	4.36	3.04
72	--	--	45.9	4.31	45.7	4.41	45.5	4.51	3.12

*Based on York Application Data Form 515.21-AD2 For CHPO-36 @ 1200 CFM

Cost \$2,500 installed per J.E. Christian, Unitary Air to Air Heat Pumps,
ANL/CSE/TE77-10

TABLE 25. PERFORMANCE OF AIR-SOURCE HEAT PUMP C*

COOLING							
DBT ¹	WBT ²	TOTAL CAP MBTUH	SENSIBLE CAP @ IDBT ³			INPUT POWER ⁴ KW	COP ⁵
			72	76	80		
85	59	34.6	29.2	33.8	35.5	4.97	2.04
	63	36.5	23.6	28.0	32.4	5.07	2.11
	67	38.4	18.6	22.7	26.8	5.17	2.18
	71	40.2	14.0	17.8	21.7	5.27	2.24
95	59	32.8	28.6	33.0	34.2	5.27	1.82
	63	34.9	23.2	27.7	32.2	5.37	1.91
	67	37.0	18.2	22.4	26.7	5.47	1.98
	71	39.1	13.5	17.5	21.5	5.67	2.02
105	59	30.4	27.8	31.1	32.5	5.57	1.60
	63	32.7	22.5	27.1	31.7	5.77	1.66
	67	35.0	17.5	21.8	26.2	5.87	1.75
	71	37.3	12.9	17.0	21.2	5.97	1.83
115	59	27.6	26.5	28.8	30.3	5.97	1.36
	63	30.0	21.5	26.2	30.3	6.07	1.45
	67	32.5	16.6	21.1	25.6	6.27	1.52
	71	35.0	12.0	16.3	20.6	6.37	1.61

HEATING									
IDBT ³ ODT	60		70		75		80		COP ⁵ @ 70°
	CAP MBTUH	KW ⁴	CAP MBTUH	KW ⁴	CAP MBTUH	KW ⁴	CAP MBTUH	KW ⁴	
2	15.3	3.3	14.0	3.5	13.4	3.6	12.8	3.6	1.17
12	19.6	3.6	18.2	3.8	17.5	3.9	16.8	4.0	1.40
22	25.2	3.9	23.5	4.1	22.7	4.2	21.8	4.3	1.68
32	31.6	4.3	29.7	4.5	28.7	4.6	27.8	4.7	1.93
42	38.7	4.6	36.5	4.8	35.5	5.0	34.4	5.1	2.23
52	45.9	4.9	42.7	5.2	42.6	5.3	41.5	5.5	2.46
62	53.0	5.3	50.8	5.6	49.7	5.7	48.6	5.8	2.66
72	56.8	5.5	54.8	5.8	53.8	6.0	52.8	6.1	2.77

¹Outdoor Dry Bulb Temperature, °F²Indoor Wet Bulb Temperature, °F³Indoor Dry Bulb Temperature, °F⁴Power requirement for compressor, outdoor fan, and indoor fan, KW⁵Coefficient of Performance = Capacity (Kw)/Input Power⁴(Kw)

*Based on G.E. Pub. No. 22-1028-6, Oct. 1977, for BWC036B @ 1200 cfm

Cost \$2,500 installed per J. E. Christian, Unitary Air-to-Air Heat Pumps,
ANL/CES/TE77-10

Table 26. Annual Performance for Temperature Variation in Twenty-four Cities

Air Source Heat Pump A

Internal Load of 5,000 BTUH

Indoor Design Temp. of 70°F

Cooling Design Load of 30,000 BTUH at Cooling Design Temperature

Heating Design Load of 36,000 BTUH at Heating Design Temperature

City, State	Cooling Design Temp °F	Heating Design Temp °F	Annual Heating Load BTUx10 ⁻⁵	Annual Cooling Load BTUx10 ⁻⁵	Cooling		Heating		
					KWH ¹	COP ³	KWH ¹	KWH ²	COP ³
Birmingham, AL	96	17	295	831	8928	2.73	3496	66	2.43
Tuscon, AZ	104	28	260	964	10791	2.62	2946	8	2.58
Los Angeles, CA	83	41	314	528	5378	2.88	3388	0	2.71
Denver, CO	89	-5	442	484	5155	2.75	5701	438	2.11
Miami, FL	91	44	58	1763	18816	2.75	633	1	2.70
Atlanta, GA	94	17	299	789	8400	2.75	3510	71	2.45
Hilo, HI	84	61	28	1835	18984	2.83	302	0	2.74
Chicago, IL	94	-5	450	500	5307	2.76	5770	569	2.08
New Orleans, LA	93	29	185	1159	12382	2.74	2070	8	2.61
Boston, MA	91	6	490	403	4219	2.80	6083	323	2.24
Detroit, MI	91	3	544	455	4778	2.79	6958	429	2.16
Saint Louis, MO	97	2	391	655	7054	2.72	4906	295	2.20
New York, NY	90	12	485	484	5031	2.82	5899	200	2.33
Akron, OH	89	1	533	422	4404	2.81	6837	532	2.12
Tulsa, OK	101	8	330	741	8110	2.68	4049	139	2.31
Portland, OR	89	17	485	231	2408	2.81	5631	36	2.51
Philadelphia, PA	93	10	453	550	5807	2.77	5552	167	2.32
Providence, RI	89	5	501	393	4088	2.82	6265	314	2.23
Columbia, SC	97	20	313	836	9016	2.72	3693	78	2.44
Nashville, TN	97	9	338	718	7726	2.72	4118	152	2.32
Houston, TX	96	27	179	1117	12017	2.73	2024	16	2.58
Salt Lake City, UT	97	3	500	483	5232	2.71	6307	289	2.22
Seattle, WA	84	21	629	186	1906	2.87	7266	88	2.51
Milwaukee, WI	90	-8	507	395	4103	2.83	6633	744	2.02

1. Annual energy input to heat pump, includes compressor, outdoor fan and indoor fan

2. Annual energy input to resistance heater

3. Annual cooling or heating load/annual energy input (dimensionless)

Table 27. Annual Performance for Temperature Variation in Twenty-four Cities

Air-Source Heat Pump C
Internal Load of 5,000 BTUH
Indoor Design Temp. of 70 F

Cooling Design Load of 30,000 BTUH at Cooling Design Temperature
Heating Design Load of 36,000 BTUH at Heating Design Temperature

City, State	Cooling	Heating	Annual	Annual	Cooling		Heating		
	Design Temp °F	Design Temp °F	Heating Load BTUx10 ⁻⁵	Cooling Load BTUx10 ⁻⁵	KWH ¹	COP ³	KWH ¹	KWH ²	COP ³
Birmingham, AL	96	17	295	828	10223	2.37	4200	128	2.00
Tucson, AZ	104	28	260	963	12409	2.27	3526	29	2.14
Los Angeles, CA	83	41	314	527	6114	2.53	4023	0	2.29
Denver, CO	89	-5	443	482	5880	2.40	6874	610	1.73
Miami, FL	91	44	58	1759	21516	2.40	751	4	2.27
Atlanta, GA	94	17	299	787	9616	2.40	4219	119	2.02
Hilo, HI	84	61	28	1831	21611	2.48	345	0	2.40
Chicago, IL	94	-5	450	499	6072	2.41	6923	783	1.71
New Orleans, LA	93	29	185	1158	14187	2.39	2474	17	2.17
Boston, MA	91	6	490	402	4814	2.45	7312	509	1.84
Detroit, MI	91	3	544	454	5455	2.44	8375	675	1.76
Saint Louis, MO	97	2	391	654	8093	2.37	5915	428	1.80
New York, NY	90	12	485	483	5738	2.47	7127	293	1.92
Akron, OH	89	1	533	421	5023	2.46	8235	759	1.74
Tulsa, OK	101	8	330	739	9303	2.33	4884	221	1.89
Portland, OR	89	17	485	231	2745	2.46	6774	72	2.08
Philadelphia, PA	93	10	453	549	6643	2.42	6706	265	1.91
Providence, RI	89	5	501	392	4662	2.46	7533	513	1.82
Columbia, SC	97	20	314	835	10348	2.36	4439	132	2.01
Nashville, TN	97	9	338	717	8865	2.37	4967	229	1.91
Houston, TX	96	27	179	1115	13768	2.37	2418	33	2.14
Salt Lake City, UT	97	3	500	482	6001	2.35	7618	445	1.82
Seattle, WA	84	21	629	186	2170	2.51	8734	132	2.08
Milwaukee, WI	90	-8	507	394	4683	2.41	7920	1058	1.66

1. Annual energy input to heat pump, includes compressor, outdoor fan and indoor fan
2. Annual energy input to resistance heater
3. Annual cooling or heating load/annual energy input (dimensionless)

employed to indicate the performance effects of decreased resistance heating (Tables 28 and 29). Since the resistance heat is used a very small percentage of the time, it may be noted that differences between single and double units are small. Obviously these combinations are not economically feasible and have been eliminated from further study. The next 2 studies use the twin unit A with increased internal load. These studies (Tables 30 and 31) give the performance effect of larger buildings where the ratio of internal load to maximum external load becomes larger. It has the effect of increasing the cooling load and decreasing the heating load. This variation can be used to match buildings in modules to obtain equal cooling and heating water requirements.

Individual run sheets are shown in Tables 32 to 38. Tables 32 to 34 demonstrate the variations for air source heat pump A at different locations using a standard internal load of 5000 Btu. Table 35 represents a sun with the water source heat pump A. Tables 36 and 37 give the performance with larger internal loads to represent the performance for larger buildings where the ratio of internal load to maximum internal load becomes larger. It has the effect of increasing the cooling load and decreasing the heating load. This variation can be used to match buildings in modules to obtain equal cooling and heating energy requirements.

Use of heating oil with an API gravity of 26, which corresponds to a¹² specific gravity of 0.9, results in higher heating value of the fuel or

$$\begin{aligned}\text{HHV} &= 19,000 \text{ Btu/lb} \\ &= 145,000 \text{ Btu/gal}\end{aligned}$$

These figures are used in the evaluation of gallons of oil required for heating.

The major difference in price between the small unit water source

Table 28. Annual Performance for Temperature Variation in Twenty-four Cities

Twin Air Source Heat Pump A
Internal Load of 5,000 BTUH
Indoor Design Temp. of 70°F

Cooling Design Load of 30,000 BTUH at Cooling Design Temperature
Heating Design Load of 36,000 BTUH at Heating Design Temperature

City, State	Cooling Design Temp °F	Heating Design Temp °F	Annual Heating Load BTUx10 ⁻⁵	Annual Cooling Load BTUx10 ⁻⁵	Cooling KWH ¹	COP ³	KWH ¹	Heating KWH ²	COP ³
Birmingham, AL	96	17	295	836	8989	2.72	3533	1	2.45
Tucson, AZ	104	28	260	972	10891	2.62	2950	0	2.58
Los Angeles, CA	83	41	312	545	5575	2.87	3388	0	2.71
Denver, CO	89	-5	443	505	5403	2.74	5966	43	2.16
Miami, FL	91	8	58	1773	18929	2.75	633	0	2.70
Atlanta, GA	94	17	299	796	8494	2.75	3549	0	2.47
Hilo, HI	84	61	28	1881	19473	2.83	302	0	2.74
Chicago, IL	94	-5	450	506	5377	2.76	6120	57	2.13
New Orleans, LA	93	29	185	1170	12510	2.74	2074	0	2.61
Boston, MA	91	6	490	407	4272	2.80	6272	14	2.28
Detroit, MI	91	3	544	460	4836	2.79	7217	11	2.21
Saint Louis, MO	97	2	391	660	7123	2.72	5083	15	2.25
New York, NY	90	12	485	488	5087	2.81	6011	1	2.37
Akron, OH	89	1	533	429	4483	2.80	7155	33	2.17
Tulsa, OK	101	8	330	746	8176	2.67	4129	4	2.34
Portland, OR	89	17	485	237	2477	2.80	5651	0	2.52
Philadelphia, PA	93	10	453	555	5869	2.77	5646	0	2.35
Providence, RI	89	5	501	399	4157	2.81	6451	14	2.27
Columbia, SC	97	20	314	842	9095	2.71	3733	0	2.46
Nashville, TN	97	9	338	723	7787	2.72	4202	10	2.35
Houston, TX	96	27	179	1122	12067	2.72	2032	0	2.59
Salt Lake City, UT	97	3	500	487	5287	2.70	6483	4	2.26
Seattle, WA	84	21	629	195	2083	2.86	7311	0	2.52
Milwaukee, WI	90	-8	507	400	4165	2.82	7088	93	2.07

1. Annual energy input to heat pump, includes compressor, outdoor fan and indoor fan
2. Annual energy input to resistance heater
3. Annual cooling or heating load/annual energy input (dimensionless)

Table 29. Annual Performance for Temperature Variation in Twenty-four Cities

Twin Air Source Heat Pump C
Internal Load of 5,000 BTUH
Indoor Design Temp. of 70°F

Cooling Design Load of 30,000 BTUH at Cooling Design Temperature
Heating Design Load of 36,000 BTUH at Heating Design Temperature

City, State	Cooling Design Temp °F	Heating Design Temp °F	Annual Heating Load BTUx10 ⁻⁵	Annual Cooling Load BTUx10 ⁻⁵	Cooling KWH ¹	COP ³	Heating KWH ¹	KWH ²	COP ³
Birmingham, AL	96	17	295	836	10328	2.37	4281	4	2.02
Tucson, AZ	104	28	260	972	12554	2.27	3543	0	2.15
Los Angeles, CA	83	41	314	545	6351	2.52	4023	0	2.29
Denver, CO	89	-5	443	505	6203	2.39	7282	110	1.76
Miami, FL	91	44	58	1773	21711	2.39	753	0	2.27
Atlanta, GA	94	17	299	796	9748	2.39	4296	3	2.04
Hilo, HI	84	61	28	1881	22236	2.48	345	0	2.40
Chicago, IL	94	-5	450	506	6167	2.40	7424	175	1.73
New Orleans, LA	93	29	185	1170	14361	2.39	2484	0	2.18
Boston, MA	91	6	490	407	4888	2.44	7659	35	1.87
Detroit, MI	91	3	544	460	5537	2.43	8850	36	1.79
Saint Louis, MO	97	2	391	660	8183	2.36	6212	40	1.83
New York, NY	90	12	485	488	5815	2.46	7325	7	1.94
Akron, OH	89	1	533	429	5126	2.45	8746	95	1.77
Tulsa, OK	101	8	330	746	9410	2.32	5036	7	1.92
Portland, OR	89	17	485	237	2833	2.45	6820	1	2.09
Philadelphia, PA	93	10	453	555	6725	2.42	6888	1	1.93
Providence, RI	89	5	501	399	4752	2.46	7885	34	1.85
Columbia, SC	97	20	314	842	10454	2.36	4522	0	2.03
Nashville, TN	97	9	338	723	8948	2.37	5115	21	1.93
Houston, TX	96	27	179	1122	13865	2.37	2438	0	2.16
Salt Lake City, UT	97	3	500	487	6081	2.35	7937	20	1.84
Seattle, WA	84	21	629	195	2289	2.50	8818	0	2.09
Milwaukee, WI	90	-8	509	400	4769	2.46	8575	278	1.68

1. Annual energy input to heat pump, includes compressor, outdoor fan and indoor fan
2. Annual energy input to resistance heater
3. Annual cooling or heating load/annual energy input (dimensionless)

Table 30. Annual Performance for Temperature Variation in Twenty-Four Cities

Twin Air Source Heat Pump A
Internal Load of 10,000 BTUH
Indoor Design Temp. of 70°F

Cooling Design Load of 30,000 BTUH at Cooling Design Temperature
Heating Design Load of 36,000 BTUH at Heating Design Temperature

City, State	Cooling	Heating	Annual	Annual	Cooling		Heating		
	Design Temp °F	Design Temp °F	Heating Load BTUx10 ⁻⁵	Cooling Load BTUx10 ⁻⁵	KWH ¹	COP ³	KWH ¹	KWH ²	COP ³
Birmingham, AL	96	17	161	1152	12205	2.77	1986	0	2.37
Tuscon, AZ	104	28	140	1291	14258	2.65	1618	0	2.54
Los Angeles, CA	83	41	140	838	8428	2.91	1530	0	2.68
Denver, CO	89	-5	233	768	7999	2.81	3325	17	2.04
Miami, FL	91	44	32	2186	23242	2.76	352	0	2.68
Atlanta, GA	94	17	158	1107	11639	2.79	1928	0	2.40
Hilo, HI	84	61	10	2300	23754	2.84	103	0	2.75
Chicago, IL	94	-5	243	760	7907	2.82	3509	18	2.02
New Orleans, LA	93	29	94	1528	16189	2.77	1074	0	2.56
Boston, MA	91	6	269	650	6657	2.86	3595	5	2.19
Detroit, MI	91	3	319	696	7175	2.84	4390	3	2.13
Saint Louis, MO	97	2	212	935	9911	2.77	2886	3	2.15
New York, NY	90	12	276	737	7549	2.86	3524	0	2.29
Akron, OH	89	1	308	675	6901	2.87	4328	9	2.08
Tulsa, OK	101	8	176	1040	11199	2.72	2293	1	2.25
Portland, OR	89	17	241	458	4618	2.91	2876	0	2.45
Philadelphia, PA	93	10	255	812	8436	2.82	3276	0	2.28
Providence, RI	89	5	275	641	6537	2.88	3706	5	2.18
Columbia, SC	97	20	176	1152	12263	2.75	2149	0	2.40
Nashville, TN	97	9	182	1018	10789	2.77	2351	5	2.26
Houston, TX	96	27	91	1479	15751	2.75	1051	0	2.53
Salt Lake City, UT	97	3	276	716	7594	2.77	3726	1	2.17
Seattle, WA	84	21	334	337	3382	2.93	3967	0	2.47
Milwaukee, WI	90	-8	276	608	6210	2.87	4112	43	1.95

1. Annual energy input to heat pump, includes compressor, outdoor fan and indoor fan
2. Annual energy input to resistance heater
3. Annual cooling or heating load/annual energy input (dimensionless)

Table 31. Annual Performance for Temperature Variation in Twenty-Four Cities

Twin Air Source Heat Pump A
Internal Load of 20,000 BTUH
Indoor Design Temp. of 70°F

Cooling Design Load of 30,000 BTUH at Cooling Design Temperature
Heating Design Load of 36,000 BTUH at Heating Design Temperature

City, State	Cooling Design Temp °F	Heating Design Temp °F	Annual Heating Load BTUx10 ⁻⁵	Annual Cooling Load BTUx10 ⁻⁵	Cooling KWH ¹	COP ³	KWH ¹	Heating KWH ²	COP ³
Birmingham, AL	96	17	25	1893	19676	2.82	347	0	2.14
Tuscon, AZ	104	28	22	2049	22136	2.71	264	0	2.39
Los Angeles, CA	83	41	21	1594	15762	2.96	231	0	2.62
Denver, CO	89	-5	29	1442	14593	2.90	514	7	1.65
Miami, FL	91	44	7	3038	32119	2.77	82	0	2.61
Atlanta, GA	94	17	21	1848	19050	2.84	290	0	2.14
Hilo, HI	84	61	1	3169	32590	2.85	14	0	2.78
Chicago, IL	94	-5	33	1426	14476	2.89	597	0	1.60
New Orleans, LA	93	29	12	2323	24290	2.80	149	0	2.41
Boston, MA	91	6	41	1299	12989	2.93	644	0	1.88
Detroit, MI	91	3	50	1304	13139	2.91	796	0	1.83
Saint Louis, MO	97	2	27	1628	16839	2.83	452	0	1.78
New York, NY	90	12	42	1380	13846	2.92	605	0	2.03
Akron, OH	89	1	47	1293	12941	2.93	785	0	1.76
Tulsa, OK	101	8	25	1765	18542	2.79	369	0	1.95
Portland, OR	89	17	21	1115	10941	2.99	283	0	2.20
Philadelphia, PA	93	10	34	1468	14917	2.88	503	0	2.00
Providence, RI	89	5	41	1284	12789	2.94	647	0	1.87
Columbia, SC	97	20	31	1883	19666	2.81	414	0	2.20
Nashville, TN	97	9	24	1737	18002	2.83	368	0	1.95
Houston, TX	96	27	13	2278	23925	2.79	161	0	2.35
Salt Lake City, UT	97	3	36	1353	13895	2.85	565	0	1.85
Seattle, WA	84	21	39	919	8952	3.01	515	0	2.26
Milwaukee, WI	90	-8	38	1246	12417	2.94	713	17	1.54

1. Annual energy input to heat pump, includes compressor, outdoor fan and indoor fan
2. Annual energy input to resistance heater
3. Annual cooling or heating load/annual energy input (dimensionless)

Table 32. Performance Computer Run

THIS PROGRAM WILL ANALYZE PERFORMANCE AT LOCATION 1, BIRMINGHAM, ALABAMA FOR AIR HEAT PUMP A HEATING/COOLING SYSTEM
 CALCULATIONS ARE FOR INDOOR TEMPERATURE MAINTAINED AT 70. F AND INTERNAL LOAD OF 5000. BTUH

HEATING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF 17. F AND LOAD OF 36000. BTUH AT DESIGN INDOOR TEMP. OF 70. F
 COOLING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF 96. F AND LOAD OF 30000. BTUH AT DESIGN INDOOR TEMP. OF 70. F.

A= OUTDOOR TEMPERATURE, FAHRENHEIT
 B= SEASONAL HEATING HOURS IN 5 DEGREE INCREMENT
 C= SEASONAL HEATING/COOLING LOAD IN 5 DEGREE INCREMENT, BTU
 D= HEAT PUMP CAPACITY AT OUTDOOR TEMPERATURE A, MBTUH
 E= HEAT PUMP SYSTEM INPUT, KW, INCLUDING AIR HANDLER REQUIREMENT, KW
 F= HEAT PUMP COMPRESSOR RUN TIME IN PERCENT AT TEMPERATURE A
 G= SEASONAL HEAT PUMP INPUT, KWH IN 5 DEGREE TEMPERATURE INCREMENT
 H= RESISTANCE HEAT REQUIRED IN 5 DEGREE INCREMENT, KW
 I= SEASONAL RESISTANCE HEAT NEEDED IN 5 DEGREE INCREMENT
 J= COEFFICIENT OF PERFORMANCE, DIMENSIONLESS
 K= SYSTEM STATUS

STANDARD CALCULATION

A	B	C	D	E	F	G	H	I	J	K
12.	3.	103189.	16.9	3.10	100.0	9.	5.1	15.4	1.23	HEATING
17.	6.	186000.	19.2	3.20	100.0	19.	3.5	20.8	1.36	HEATING
22.	17.	469264.	21.7	3.30	100.0	56.	1.7	29.4	1.61	HEATING
27.	69.	1570321.	24.4	3.50	99.2	240.	.0	.0	2.04	HEATING
32.	143.	2975019.	27.1	3.60	76.8	395.	.0	.0	2.21	HEATING
37.	292.	5085207.	30.1	3.70	57.9	625.	.0	.0	2.38	HEATING
42.	433.	6070170.	33.0	3.90	42.5	717.	.0	.0	2.48	HEATING
47.	528.	5608755.	35.9	4.10	29.6	641.	.0	.0	2.57	HEATING
52.	614.	4437019.	38.5	4.20	18.8	484.	.0	.0	2.69	HEATING
57.	668.	2558566.	40.8	4.40	9.4	276.	.0	.0	2.72	HEATING
62.	742.	322000.	42.7	4.50	1.0	34.	.0	.0	2.78	HEATING
67.	805.	1238462.	38.3	3.63	4.0	117.	.0	.0	3.09	COOLING
72.	908.	6635365.	38.1	3.73	19.2	650.	.0	.0	2.99	COOLING
77.	1138.	14881538.	37.9	3.83	34.5	1504.	.0	.0	2.90	COOLING
82.	924.	17413846.	37.6	3.93	50.1	1820.	.0	.0	2.80	COOLING
87.	658.	16196923.	37.2	4.03	66.2	1755.	.0	.0	2.71	COOLING
92.	458.	14827692.	36.7	4.13	82.8	1669.	.0	.0	2.60	COOLING
97.	264.	9530400.	36.1	4.23	100.0	1117.	.0	.0	2.50	COOLING
102.	62.	2194800.	35.4	4.43	100.0	275.	.0	.0	2.34	COOLING
107.	5.	172500.	34.5	4.53	100.0	23.	.0	.0	2.23	COOLING

ANNUAL HEATING LOAD, BTU = 29466508.

ANNUAL HEATING HEAT PUMP INPUT, KWH = 3496.

ANNUAL RESISTANCE HEAT REQUIRED, KWH = 66.

ANNUAL COOLING LOAD, BTU = 83091543.

ANNUAL COOLING HEAT PUMP INPUT, KWH = 8928.

ANNUAL COOLING COP = 2.73

ANNUAL HEATING COP = 2.43

GAS HEAT AT 62% EFFICIENCY WOULD REQUIRE 476. THERMS.

FUEL OIL HEAT AT 62% EFFICIENCY WOULD REQUIRE 328. GALLONS

Table 33. Performance Computer Run

THIS PROGRAM WILL ANALYZE PERFORMANCE AT LOCATION 5, TUCSON, ARIZONA FOR AIR HEAT PUMP A HEATING/COOLING SYSTEM
 CALCULATIONS ARE FOR INDOOR TEMPERATURE MAINTAINED AT 70. F AND INTERNAL LOAD OF 5000. BTUH

HEATING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF 28. F AND LOAD OF 36000. BTUH AT DESIGN INDOOR TEMP. OF 70. F
 COOLING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF 104. F AND LOAD OF 30000. BTUH AT DESIGN INDOOR TEMP. OF 70. F.

A= OUTDOOR TEMPERATURE, FAHRENHEIT
 B= SEASONAL HEATING HOURS IN 5 DEGREE INCREMENT
 C= SEASONAL HEATING/COOLING LOAD IN 5 DEGREE INCREMENT, BTU
 D= HEAT PUMP CAPACITY AT OUTDOOR TEMPERATURE A, MBTUH
 E= HEAT PUMP SYSTEM INPUT, KW, INCLUDING AIR HANDLER REQUIREMENT, KW
 F= HEAT PUMP COMPRESSOR RUN TIME IN PERCENT AT TEMPERATURE A
 G= SEASONAL HEAT PUMP INPUT, KWH IN 5 DEGREE TEMPERATURE INCREMENT
 H= RESISTANCE HEAT REQUIRED IN 5 DEGREE INCREMENT, KW
 I= SEASONAL RESISTANCE HEAT NEEDED IN 5 DEGREE INCREMENT
 J= COEFFICIENT OF PERFORMANCE, DIMENSIONLESS
 K= SYSTEM STATUS

STANDARD CALCULATION

A	B	C	D	E	F	G	H	I	J	K
27.	2.	63714.	24.4	3.50	100.0	7.	2.2	4.4	1.64	HEATING
32.	25.	689236.	27.1	3.50	100.0	90.	.1	3.5	2.16	HEATING
37.	98.	2282000.	30.1	3.70	77.4	281.	.0	.0	2.38	HEATING
42.	248.	4712000.	33.0	3.90	57.6	557.	.0	.0	2.48	HEATING
47.	417.	6135857.	35.9	4.10	41.0	701.	.0	.0	2.57	HEATING
52.	598.	6236286.	38.5	4.20	27.1	680.	.0	.0	2.69	HEATING
57.	716.	4398286.	40.8	4.40	15.1	474.	.0	.0	2.72	HEATING
62.	800.	1485714.	42.7	4.50	4.3	157.	.0	.0	2.78	HEATING
67.	763.	1795294.	38.3	3.63	6.1	170.	.0	.0	3.09	COOLING
72.	761.	5283235.	38.1	3.73	17.8	517.	.0	.0	2.99	COOLING
77.	870.	9723529.	37.9	3.83	29.5	983.	.0	.0	2.90	COOLING
82.	959.	14949117.	37.6	3.93	41.5	1563.	.0	.0	2.80	COOLING
87.	777.	15540000.	37.2	4.03	53.8	1683.	.0	.0	2.71	COOLING
92.	656.	16014117.	36.7	4.13	66.5	1802.	.0	.0	2.60	COOLING
97.	520.	14988235.	36.1	4.23	79.6	1756.	.0	.0	2.50	COOLING
102.	357.	11865000.	35.4	4.43	93.9	1485.	.0	.0	2.34	COOLING
107.	152.	5244000.	34.5	4.53	100.0	689.	.0	.0	2.23	COOLING
112.	31.	1038500.	33.5	4.63	100.0	144.	.0	.0	2.12	COOLING

ANNUAL HEATING LOAD, BTU = 26003142.

ANNUAL HEATING HEAT PUMP INPUT, KWH = 2946.

ANNUAL RESISTANCE HEAT REQUIRED, KWH = 8.

ANNUAL COOLING LOAD, BTU = 96441026.

ANNUAL COOLING HEAT PUMP INPUT, KWH = 10791.

ANNUAL COOLING COP = 2.62

ANNUAL HEATING COP = 2.58

GAS HEAT AT 62% EFFICIENCY WOULD REQUIRE 419. THERMS.

FUEL OIL HEAT AT 62% EFFICIENCY WOULD REQUIRE 289. GALLONS

Table 34. Performance Computer Run

THIS PROGRAM WILL ANALYZE PERFORMANCE AT LOCATION 28, MILWAUKEE, WISCONSIN FOR AIR SOURCE A
CALCULATIONS ARE FOR INDOOR TEMPERATURE MAINTAINED AT 70. F AND INTERNAL LOAD OF 5000. BTUH

HEATING/COOLING SYSTEM

HEATING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF -8. F AND LOAD OF 36000. BTUH AT DESIGN INDOOR TEMP. OF 70. F
COOLING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF 90. F AND LOAD OF 30000. BTUH AT DESIGN INDOOR TEMP. OF 70. F.

A= OUTDOOR TEMPERATURE, FAHRENHEIT
B= SEASONAL HEATING HOURS IN 5 DEGREE INCREMENT
C= SEASONAL HEATING/COOLING LOAD IN 5 DEGREE INCREMENT , BTU
D= HEAT PUMP CAPACITY AT OUTDOOR TEMPERATURE A, MBTUH
E= HEAT PUMP SYSTEM INPUT, KW, INCLUDING AIR HANDLER REQUIREMENT, KW
F= HEAT PUMP COMPRESSOR RUN TIME INPERCENT AT TEMPERATURE A
G= SEASONAL HEAT PUMP INPUT, KWH IN 5 DEGREE TEMPERATURE INCREMENT
H= RESISTANCE HEAT REQUIRED IN 5 DEGREE INCREMENT, KW
I= SEASONAL RESISTANCE HEAT NEEDED IN 5 DEGREE INCREMENT
J= COEFFICIENT OF PERFORMANCE, DIMENSIONLESS
K= SYSTEM STATUS

STANDARD CALCULATION

A	B	C	D	E	F	G	H	I	J	K
-18.	1.	35615.	.0	.00	.0	0.	10.4	10.4	1.00	HEATING
-13.	2.	66615.	.0	.00	.0	0.	9.8	19.5	1.00	HEATING
-8.	4.	124000.	9.5	2.60	100.0	10.	6.3	25.2	1.02	HEATING
-3.	18.	516462.	11.0	2.70	100.0	49.	5.2	93.3	1.07	HEATING
2.	47.	1240077.	12.7	2.80	100.0	132.	4.0	188.5	1.14	HEATING
7.	83.	1998385.	14.7	2.90	100.0	241.	2.7	228.1	1.25	HEATING
12.	116.	2525231.	16.9	3.10	100.0	360.	1.4	165.5	1.41	HEATING
17.	176.	3425231.	19.2	3.20	100.0	563.	.1	13.5	1.74	HEATING
22.	285.	4888846.	21.7	3.30	79.0	743.	.0	.0	1.93	HEATING
27.	421.	6250231.	24.4	3.50	60.8	897.	.0	.0	2.04	HEATING
32.	659.	8262846.	27.1	3.60	46.3	1098.	.0	.0	2.21	HEATING
37.	913.	9340692.	30.1	3.70	34.0	1148.	.0	.0	2.38	HEATING
42.	774.	6132461.	33.0	3.90	24.0	725.	.0	.0	2.48	HEATING
47.	611.	3431000.	35.9	4.10	15.6	392.	.0	.0	2.57	HEATING
52.	591.	1954846.	38.5	4.20	8.6	213.	.0	.0	2.69	HEATING
57.	585.	585000.	40.8	4.40	2.5	63.	.0	.0	2.72	HEATING
62.	634.	829077.	38.3	3.63	3.4	79.	.0	.0	3.09	COOLING
67.	749.	2707923.	38.3	3.63	9.4	257.	.0	.0	3.09	COOLING
72.	753.	6024000.	38.1	3.73	21.0	590.	.0	.0	2.99	COOLING
77.	597.	9253500.	37.9	3.83	40.9	935.	.0	.0	2.90	COOLING
82.	390.	8970000.	37.6	3.93	61.2	938.	.0	.0	2.80	COOLING
87.	226.	6893000.	37.2	4.03	82.0	747.	.0	.0	2.71	COOLING
92.	96.	3523200.	36.7	4.13	100.0	396.	.0	.0	2.60	COOLING
97.	32.	1155200.	36.1	4.23	100.0	135.	.0	.0	2.50	COOLING
102.	6.	212400.	35.4	4.43	100.0	27.	.0	.0	2.34	COOLING

ANNUAL HEATING LOAD, BTU = 50777535.

ANNUAL HEATING HEAT PUMP INPUT, KWH = 6633.

ANNUAL RESISTANCE HEAT REQUIRED, KWH = 744.

ANNUAL COOLING LOAD, BTU = 39568299.

ANNUAL COOLING HEAT PUMP INPUT, KWH = 4103.

ANNUAL COOLING COP = 2.83

ANNUAL HEATING COP = 2.02

GAS HEAT AT 62% EFFICIENCY WOULD REQUIRE 819. THERMS.

FUEL OIL HEAT AT 62% EFFICIENCY WOULD REQUIRE 565. GALLONS

Table 35. Performance Computer Run

THIS PROGRAM WILL ANALYZE PERFORMANCE AT LOCATION 28, MILWAUKEE, WISCONSIN FOR WATER SOURCE A HEATING/COOLING SYSTEM
 CALCULATIONS ARE FOR INDOOR TEMPERATURE MAINTAINED AT 70. F AND INTERNAL LOAD OF 5000. BTUH

HEATING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF -8. F AND LOAD OF 36000. BTUH AT DESIGN INDOOR TEMP. OF 70. F
 COOLING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF 90. F AND LOAD OF 30000. BTUH AT DESIGN INDOOR TEMP. OF 70. F.

A= OUTDOOR TEMPERATURE, FAHRENHEIT
 B= SEASONAL HEATING HOURS IN 5 DEGREE INCREMENT
 C= SEASONAL HEATING/COOLING LOAD IN 5 DEGREE INCREMENT, BTUH
 D= HEAT PUMP CAPACITY AT OUTDOOR TEMPERATURE A, MBTUH
 E= HEAT PUMP SYSTEM INPUT, KW, INCLUDING AIR HANDLER REQUIREMENT, KW
 F= HEAT PUMP COMPRESSOR RUN TIME IN PERCENT AT TEMPERATURE A
 G= SEASONAL HEAT PUMP INPUT, KWH IN 5 DEGREE TEMPERATURE INCREMENT
 H= RESISTANCE HEAT REQUIRED IN 5 DEGREE INCREMENT, KW
 I= SEASONAL RESISTANCE HEAT NEEDED IN 5 DEGREE INCREMENT
 J= COEFFICIENT OF PERFORMANCE, DIMENSIONLESS
 K= SYSTEM STATUS

STANDARD CALCULATION

A	B	C	D	E	F	G	H	I	J	K
-18.	1.	35615.	52.0	5.62	68.5	4.	.0	.0	2.71	HEATING
-13.	2.	66615.	52.0	5.62	64.1	7.	.0	.0	2.71	HEATING
-8.	4.	124000.	52.0	5.62	59.6	13.	.0	.0	2.71	HEATING
-3.	18.	516462.	52.0	5.62	55.2	56.	.0	.0	2.71	HEATING
2.	47.	1240077.	52.0	5.62	50.7	134.	.0	.0	2.71	HEATING
7.	83.	1998385.	52.0	5.62	46.3	216.	.0	.0	2.71	HEATING
12.	116.	2525231.	52.0	5.62	41.9	273.	.0	.0	2.71	HEATING
17.	176.	3425231.	52.0	5.62	37.4	370.	.0	.0	2.71	HEATING
22.	285.	4888846.	52.0	5.62	33.0	528.	.0	.0	2.71	HEATING
27.	421.	6250231.	52.0	5.62	28.6	676.	.0	.0	2.71	HEATING
32.	659.	8262846.	52.0	5.62	24.1	893.	.0	.0	2.71	HEATING
37.	913.	9340692.	52.0	5.62	19.7	1010.	.0	.0	2.71	HEATING
42.	774.	6132461.	52.0	5.62	15.2	663.	.0	.0	2.71	HEATING
47.	611.	3431000.	52.0	5.62	10.8	371.	.0	.0	2.71	HEATING
52.	591.	1954846.	52.0	5.62	6.4	211.	.0	.0	2.71	HEATING
57.	585.	585000.	52.0	5.62	1.9	63.	.0	.0	2.71	HEATING
62.	634.	829077.	37.4	4.66	3.5	103.	.0	.0	2.35	COOLING
67.	749.	2707923.	37.4	4.66	9.7	337.	.0	.0	2.35	COOLING
72.	753.	6024000.	37.4	4.66	21.4	751.	.0	.0	2.35	COOLING
77.	597.	9253500.	37.4	4.66	41.4	1153.	.0	.0	2.35	COOLING
82.	390.	8970000.	37.4	4.66	61.5	1118.	.0	.0	2.35	COOLING
87.	226.	6893000.	37.4	4.66	81.6	859.	.0	.0	2.35	COOLING
92.	96.	3590400.	37.4	4.66	100.0	447.	.0	.0	2.35	COOLING
97.	32.	1196800.	37.4	4.66	100.0	149.	.0	.0	2.35	COOLING
102.	6.	224400.	37.4	4.66	100.0	28.	.0	.0	2.35	COOLING

95

ANNUAL HEATING LOAD, BTU = 50777535.

ANNUAL HEATING HEAT PUMP INPUT, KWH = 5488.

ANNUAL RESISTANCE HEAT REQUIRED, KWH = 0.

ANNUAL COOLING LOAD, BTU = 39689100.

ANNUAL COOLING HEAT PUMP INPUT, KWH = 4945.

ANNUAL COOLING COP = 2.35

ANNUAL HEATING COP = 2.71

GAS HEAT AT 62% EFFICIENCY WOULD REQUIRE 819. THERMS.

FUEL OIL HEAT AT 62% EFFICIENCY WOULD REQUIRE 565. GALLONS

Table 36. Performance Computer Run

THIS PROGRAM WILL ANALYZE PERFORMANCE AT LOCATION 28, MILWAUKEE, WISCONSIN FOR TWIN AIR A HEATING/COOLING SYSTEM
 CALCULATIONS ARE FOR INDOOR TEMPERATURE MAINTAINED AT 70. F AND INTERNAL LOAD OF 5000. BTUH

HEATING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF -8. F AND LOAD OF 36000. BTUH AT DESIGN INDOOR TEMP. OF 70. F
 COOLING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF 90. F AND LOAD OF 30000. BTUH AT DESIGN INDOOR TEMP. OF 70. F.

A= OUTDOOR TEMPERATURE, FAHRENHEIT
 B= SEASONAL HEATING HOURS IN 5 DEGREE INCREMENT
 C= SEASONAL HEATING/COOLING LOAD IN 5 DEGREE INCREMENT, BTU
 D= HEAT PUMP CAPACITY AT OUTDOOR TEMPERATURE A, MBTUH
 E= HEAT PUMP SYSTEM INPUT, KW, INCLUDING AIR HANDLER REQUIREMENT, KW
 F= HEAT PUMP COMPRESSOR RUN TIME IN PERCENT AT TEMPERATURE A
 G= SEASONAL HEAT PUMP INPUT, KWH IN 5 DEGREE TEMPERATURE INCREMENT
 H= RESISTANCE HEAT REQUIRED IN 5 DEGREE INCREMENT, KW
 I= SEASONAL RESISTANCE HEAT NEEDED IN 5 DEGREE INCREMENT
 J= COEFFICIENT OF PERFORMANCE, DIMENSIONLESS
 K= SYSTEM STATUS

STANDARD CALCULATION

A	B	C	D	E	F	G	H	I	J	K
-18.	1.	35615.	.0	.00	.0	0.	10.4	10.4	1.00	HEATING
-13.	2.	66615.	.0	.00	.0	0.	9.8	19.5	1.00	HEATING
-8.	4.	124000.	19.0	5.20	100.0	21.	3.5	14.1	1.04	HEATING
-3.	18.	516462.	22.0	5.40	100.0	97.	2.0	35.3	1.14	HEATING
2.	47.	1240077.	25.4	5.60	100.0	263.	.3	13.6	1.31	HEATING
7.	83.	1998385.	29.4	5.80	81.9	394.	.0	.0	1.49	HEATING
12.	116.	2525231.	33.8	6.20	64.4	463.	.0	.0	1.60	HEATING
17.	176.	3425231.	38.4	6.40	50.7	571.	.0	.0	1.76	HEATING
22.	285.	4888846.	43.4	6.60	39.5	743.	.0	.0	1.93	HEATING
27.	421.	6250231.	48.8	7.00	30.4	897.	.0	.0	2.04	HEATING
32.	659.	8262846.	54.2	7.20	23.1	1098.	.0	.0	2.21	HEATING
37.	913.	9340692.	60.2	7.40	17.0	1148.	.0	.0	2.38	HEATING
42.	774.	6132461.	66.0	7.80	12.0	725.	.0	.0	2.48	HEATING
47.	611.	3431000.	71.8	8.20	7.8	392.	.0	.0	2.57	HEATING
52.	591.	1954846.	77.0	8.40	4.3	213.	.0	.0	2.69	HEATING
57.	585.	585000.	81.6	8.80	1.2	63.	.0	.0	2.72	HEATING
62.	634.	829077.	76.6	7.26	1.7	79.	.0	.0	3.09	COOLING
67.	749.	2707923.	76.6	7.26	4.7	257.	.0	.0	3.09	COOLING
72.	753.	6024000.	76.2	7.46	10.5	590.	.0	.0	2.99	COOLING
77.	597.	9253500.	75.8	7.66	20.4	935.	.0	.0	2.90	COOLING
82.	390.	8970000.	75.2	7.86	30.6	938.	.0	.0	2.80	COOLING
87.	226.	6893000.	74.4	8.06	41.0	747.	.0	.0	2.71	COOLING
92.	96.	3648000.	73.4	8.26	51.8	411.	.0	.0	2.60	COOLING
97.	32.	1456000.	72.2	8.46	63.0	171.	.0	.0	2.50	COOLING
102.	6.	318000.	70.8	8.86	74.9	40.	.0	.0	2.34	COOLING

96

ANNUAL HEATING LOAD, BTU = 50777535.

ANNUAL HEATING HEAT PUMP INPUT, KWH = 7088.

ANNUAL RESISTANCE HEAT REQUIRED, KWH = 93.

ANNUAL COOLING LOAD, BTU = 40099500.

ANNUAL COOLING HEAT PUMP INPUT, KWH = 4165.

ANNUAL COOLING COP = 2.82

ANNUAL HEATING COP = 2.07

GAS HEAT AT 62% EFFICIENCY WOULD REQUIRE 819. THERMS.

FUEL OIL HEAT AT 62% EFFICIENCY WOULD REQUIRE 565. GALLONS

Table 37. Performance Computer Run

THIS PROGRAM WILL ANALYZE PERFORMANCE AT LOCATION 28, MILWAUKEE, WISCONSIN FOR TWIN AIR A
 CALCULATIONS ARE FOR INDOOR TEMPERATURE MAINTAINED AT 70. F AND INTERNAL LOAD OF 10000. BTUH

HEATING/COOLING SYSTEM

HEATING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF -8. F AND LOAD OF 36000. BTUH AT DESIGN INDOOR TEMP. OF 70. F
 COOLING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF 90. F AND LOAD OF 30000. BTUH AT DESIGN INDOOR TEMP. OF 70. F.

A= OUTDOOR TEMPERATURE, FAHRENHEIT
 B= SEASONAL HEATING HOURS IN 5 DEGREE INCREMENT
 C= SEASONAL HEATING/COOLING LOAD IN 5 DEGREE INCREMENT, BTU
 D= HEAT PUMP CAPACITY AT OUTDOOR TEMPERATURE A, MBTUH
 E= HEAT PUMP SYSTEM INPUT, KW, INCLUDING AIR HANDLER REQUIREMENT, KW
 F= HEAT PUMP COMPRESSOR RUN TIME IN PERCENT AT TEMPERATURE A
 G= SEASONAL HEAT PUMP INPUT, KWH IN 5 DEGREE TEMPERATURE INCREMENT
 H= RESISTANCE HEAT REQUIRED IN 5 DEGREE INCREMENT, KW
 I= SEASONAL RESISTANCE HEAT NEEDED IN 5 DEGREE INCREMENT
 J= COEFFICIENT OF PERFORMANCE, DIMENSIONLESS
 K= SYSTEM STATUS

STANDARD CALCULATION

A	B	C	D	E	F	G	H	I	J	K
-18.	1.	30615.	.0	.00	.0	0.	9.0	9.0	1.00	HEATING
-13.	2.	56615.	.0	.00	.0	0.	8.3	16.6	1.00	HEATING
-8.	4.	104000.	19.0	5.20	100.0	21.	2.1	8.2	1.05	HEATING
-3.	18.	426462.	22.0	5.40	100.0	97.	.5	8.9	1.18	HEATING
2.	47.	1005077.	25.4	5.60	84.2	222.	.0	.0	1.33	HEATING
7.	83.	1583385.	29.4	5.80	64.9	312.	.0	.0	1.49	HEATING
12.	116.	1945231.	33.8	6.20	49.6	357.	.0	.0	1.60	HEATING
17.	176.	2545231.	38.4	6.40	37.7	424.	.0	.0	1.76	HEATING
22.	285.	3463846.	43.4	6.60	28.0	527.	.0	.0	1.93	HEATING
27.	421.	4145231.	48.8	7.00	20.2	595.	.0	.0	2.04	HEATING
32.	659.	4967846.	54.2	7.20	13.9	660.	.0	.0	2.21	HEATING
37.	913.	4775692.	60.2	7.40	8.7	587.	.0	.0	2.38	HEATING
42.	774.	2262461.	66.0	7.80	4.4	267.	.0	.0	2.48	HEATING
47.	611.	376000.	71.8	8.20	.9	43.	.0	.0	2.57	HEATING
52.	591.	1000154.	76.6	7.26	2.2	95.	.0	.0	3.09	COOLING
57.	585.	2340000.	76.6	7.26	5.2	222.	.0	.0	3.09	COOLING
62.	634.	3999077.	76.6	7.26	8.2	379.	.0	.0	3.09	COOLING
67.	749.	6452923.	76.6	7.26	11.2	612.	.0	.0	3.09	COOLING
72.	753.	9789000.	76.2	7.46	17.1	958.	.0	.0	2.99	COOLING
77.	597.	12238500.	75.8	7.66	27.0	1237.	.0	.0	2.90	COOLING
82.	390.	10920000.	75.2	7.86	37.2	1141.	.0	.0	2.80	COOLING
87.	226.	8023000.	74.4	8.06	47.7	869.	.0	.0	2.71	COOLING
92.	96.	4128000.	73.4	8.26	58.6	465.	.0	.0	2.60	COOLING
97.	32.	1616000.	72.2	8.46	69.9	189.	.0	.0	2.50	COOLING
102.	6.	348000.	70.8	8.86	81.9	44.	.0	.0	2.34	COOLING

ANNUAL HEATING LOAD, BTU = 27687691.

ANNUAL HEATING HEAT PUMP INPUT, KWH = 4112.

ANNUAL RESISTANCE HEAT REQUIRED, KWH = 43.

ANNUAL COOLING LOAD, BTU = 60854653.

ANNUAL COOLING HEAT PUMP INPUT, KWH = 6210.

ANNUAL COOLING COP = 2.87

ANNUAL HEATING COP = 1.95

GAS HEAT AT 62% EFFICIENCY WOULD REQUIRE 447. THERMS.

FUEL OIL HEAT AT 62% EFFICIENCY WOULD REQUIRE 308. GALLONS

Table 38. Performance Computer Run

THIS PROGRAM WILL ANALYZE PERFORMANCE AT LOCATION 28, MILWAUKEE, WISCONSIN FOR TWIN AIR A HEATING/COOLING SYSTEM
 CALCULATIONS ARE FOR INDOOR TEMPERATURE MAINTAINED AT 70. F AND INTERNAL LOAD OF 20000. BTUH

HEATING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF -8. F AND LOAD OF 36000. BTUH AT DESIGN INDOOR TEMP. OF 70. F
 COOLING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF 90. F AND LOAD OF 30000. BTUH AT DESIGN INDOOR TEMP. OF 70. F.

A= OUTDOOR TEMPERATURE, FAHRENHEIT
 B= SEASONAL HEATING HOURS IN 5 DEGREE INCREMENT
 C= SEASONAL HEATING/COOLING LOAD IN 5 DEGREE INCREMENT, BTU
 D= HEAT PUMP CAPACITY AT OUTDOOR TEMPERATURE A, MBTUH
 E= HEAT PUMP SYSTEM INPUT, KW, INCLUDING AIR HANDLER REQUIREMENT, KW
 F= HEAT PUMP COMPRESSOR RUN TIME IN PERCENT AT TEMPERATURE A
 G= SEASONAL HEAT PUMP INPUT, KWH IN 5 DEGREE TEMPERATURE INCREMENT
 H= RESISTANCE HEAT REQUIRED IN 5 DEGREE INCREMENT, KW
 I= SEASONAL RESISTANCE HEAT NEEDED IN 5 DEGREE INCREMENT
 J= COEFFICIENT OF PERFORMANCE, DIMENSIONLESS
 K= SYSTEM STATUS

STANDARD CALCULATION

A	B	C	D	E	F	G	H	I	J	K
-18.	1.	20615.	.0	.00	.0	0.	6.0	6.0	1.00	HEATING
-13.	2.	36615.	.0	.00	.0	0.	5.4	10.7	1.00	HEATING
-8.	4.	64000.	19.0	5.20	84.2	18.	.0	.0	1.07	HEATING
-3.	18.	246462.	22.0	5.40	62.2	60.	.0	.0	1.19	HEATING
2.	47.	535077.	25.4	5.60	44.8	118.	.0	.0	1.33	HEATING
7.	83.	753385.	29.4	5.80	30.9	149.	.0	.0	1.49	HEATING
12.	116.	785231.	33.8	6.20	20.0	144.	.0	.0	1.60	HEATING
17.	176.	785231.	38.4	6.40	11.6	131.	.0	.0	1.76	HEATING
22.	285.	613846.	43.4	6.60	5.0	93.	.0	.0	1.93	HEATING
27.	421.	64769.	76.6	7.26	.2	6.	.0	.0	3.09	COOLING
32.	659.	1622154.	76.6	7.26	3.2	154.	.0	.0	3.09	COOLING
37.	913.	4354308.	76.6	7.26	6.2	413.	.0	.0	3.09	COOLING
42.	774.	5477539.	76.6	7.26	9.2	519.	.0	.0	3.09	COOLING
47.	611.	5734000.	76.6	7.26	12.3	543.	.0	.0	3.09	COOLING
52.	591.	6910154.	76.6	7.26	15.3	655.	.0	.0	3.09	COOLING
57.	585.	8190000.	76.6	7.26	18.3	776.	.0	.0	3.09	COOLING
62.	634.	10339077.	76.6	7.26	21.3	980.	.0	.0	3.09	COOLING
67.	749.	13942923.	76.6	7.26	24.3	1321.	.0	.0	3.09	COOLING
72.	753.	17319000.	76.2	7.46	30.2	1696.	.0	.0	2.99	COOLING
77.	597.	18208500.	75.8	7.66	40.2	1840.	.0	.0	2.90	COOLING
82.	390.	14820000.	75.2	7.86	50.5	1549.	.0	.0	2.80	COOLING
87.	226.	10283000.	74.4	8.06	61.2	1114.	.0	.0	2.71	COOLING
92.	96.	5088000.	73.4	8.26	72.2	573.	.0	.0	2.60	COOLING
97.	32.	1936000.	72.2	8.46	83.8	227.	.0	.0	2.50	COOLING
102.	6.	408000.	70.8	8.86	96.0	51.	.0	.0	2.34	COOLING

98

ANNUAL HEATING LOAD, BTU = 3840461.
 ANNUAL HEATING HEAT PUMP INPUT, KWH = 713.
 ANNUAL RESISTANCE HEAT REQUIRED, KWH = 17.
 ANNUAL COOLING LOAD, BTU = 124697423.
 ANNUAL COOLING HEAT PUMP INPUT, KWH = 12417.
 ANNUAL COOLING COP = 2.94
 ANNUAL HEATING COP = 1.54

GAS HEAT AT 62% EFFICIENCY WOULD REQUIRE 62. THERMS.
 FUEL OIL HEAT AT 62% EFFICIENCY WOULD REQUIRE 43. GALLONS

heat pumps and air source heat pumps is in packaging and controls. The water source heat pump is available with both heat exchangers, compressor, and air handler in one packaged unit. Physically the water heat exchanger is smaller than the air source heat exchanger. The air source unit requires one external and one internal package. Also, due to external exposure the air source units require compressor crankcase heating and heat exchanger fin defrost. The controls to operate the system must be included. As a result the installed cost of the air source units is appreciably more than that of the water-source unit. The installed cost per Christian is listed in the pump performance Table 1 and Tables 23 to 25. A surcharge of \$300 has been added to the high performance air source heat pump A, and a \$300 surcharge has been added to the modified revisions of water source heat pumps with the additional heat exchanger. This charge is similar to a duct A frame heat exchanger for a 3-ton heat pump. It must be noted that fixed prices for heat pumps do not exist as they vary from region to region and dealer to dealer.

According to Christian, the prices for water source pumps compared to air source heat pumps on the basis of ARI capacity are one-half as much.^{10,11} However, the water-source heat pump provides essentially a constant heat output, a function of water temperature; whereas the air-source unit shows a rapid decline in heating output at low ambient temperatures. Comparisons of costs at low ambient temperatures may disclose a price ratio of as much as 4:8. As a result, this heating deficit must be eliminated by resistance heating.

Exception is taken to Christian's estimate of maintenance cost.^{10,11}

In as much as the water source heat pump is a much simpler system and operates over a smaller temperature range, less maintenance and service should be required. Due to the smaller temperature range resulting in lower compressor pressure ratios, a comparison with refrigerator maintenance and depreciation is possible. Of 150 units installed in the Tuscaloosa VA Hospital, only three service calls were required the first year. All three calls were for burned-out fan motors.

IX.B. Aquifer Analysis

The movement, occurrence, and distribution of water below the surface is known as groundwater hydrology.^{17,18} The groundwater referred to without further specification is commonly understood to mean water occupying all the voids within a geological stratum or a portion thereof. The saturated zone is to be distinguished from an unsaturated, or aeration, zone where voids are filled with water and air. This study is confined to the saturated zones for use as energy sinks and sources. However, the unsaturated zones may contain soil moisture and interact with the saturated zones, especially at shallow depths. The study is also limited to one-phase flow, water only. Combination of water and gas or water and oil are not considered.

Groundwater plays a critical role in the United States with over 50,000,000,000 gallons per day being required for public supplies, rural use, irrigation, and industrial supplies. As a result, the hydrology of groundwater is a well-defined science. The storage of thermal energy is basically undefined on a scientific basis. Numerous experiments or attempts at storage on small scales were carried out in the late 40's and 50's in conjunction with system development for heating and cooling.

The basic problems with the systems were unreliable equipment and poorly

engineered storage systems. One of the prime problems was using a finite volume as an infinite heat sink and source. General studies have shown the earth to be an extremely poor conductor of thermal energy.

The average temperatures of shallow groundwater as a function of location in the United States are shown in Figure 21.¹⁹ These are temperatures of water near the surface; temperature normally increases with depth on a uniform basis. Temperature of water in all cases noted on the map is suitable for heating purposes; however, use of water below 50°F is questionable. To improve heat pump performance, this project proposes to employ water with an initial temperature of 45°F or higher and to use return water that has been injected at temperatures as high as 100°F. Much of the cold water, below 55°F, can be used directly for cooling without the utilization of a heat pump. However, large scale use may lower the water table and reduce the availability of water.

The flow of water in an aquifer is a function of Darcy's law.^{17,18} Basically the flow rate can be given as:

$$Q = KA \frac{dh}{dL}$$

where Q is the flow rate,

$\frac{dh}{dL}$ is the hydraulic gradient

A is the cross-sectional area, and

K is the constant of proportionality known as the coefficient of permeability

The equation is sometimes written with a negative sign to show flow in the opposite direction from the gradient of the hydraulic head. The equation's validity is limited to laminar flow. Essentially all aquifer flow, except for underground streams or the area immediately adjacent to the

Figure 21. AVERAGE TEMPERATURES OF SHALLOW GROUND-WATER IN DEGREES FAHRENHEIT

well bore, is laminar flow.

The laboratory coefficient of permeability K_s is defined as a flow of 1 gal./day through a cross-sectional area of 1 ft.² with a hydraulic gradient of 1 ft./ft. and at a temperature of 60°F. A specific permeability "k" known as the darcy may also be used by separating out the properties of the fluid. A darcy is expressed as:

$$k = \frac{\mu Q}{\gamma A (dh/dL)}$$

where μ is fluid viscosity, and
 γ is fluid specific weight

The relationship between the darcy and the laboratory coefficient of permeability is:

$$1 \text{ darcy} = 18.2 K_s$$

The values of permeability for different types of aquifers are shown in Figure 22.¹⁷

The rate of flow of ground water is governed by permeability of the aquifer and the hydraulic gradient. Normal flow rates range between 5 ft./year to 5 ft./day. Most aquifers considered for thermal storage will have flow rates of less than 100 ft./year.

As water is removed by a well from an aquifer, the velocity increases from essentially zero at a relatively long distance from the well, to maximum velocity at the well. Analyzing the flow capacity as a function of distance from a well with radius r using Darcy's law,

$$Q = AK \frac{dh}{dr} = 2\pi r b K \frac{dh}{dr}$$

where b is thickness of the water-bearing portion of well.¹⁷ Rearranging and integrating for boundary conditions of $r = r_w$ and $h = h_w$ for conditions at the well and $r = r_o$ and $h = h_o$ at a distance from the well yields:

Aquifer type	Clean gravel	Clean sands; mixtures of clean sands and gravels; limestone, dolomite and certain igneous rocks	Very fine sands; silts mixtures of sand, silt, and clay; glacial till; stratified clays; etc.; sandstones and carbonates with low porosity and permeability	Unweathered clays; certain igneous and metamorphic rocks
Flow characteristics	Good aquifers		Poor aquifers	Impervious

Figure 22 LABORATORY COEFFICIENT OF PERMEABILITY, K_s , GAL./DAY ft.² AT A HYDRAULIC GRADIENT OF 1 ft./ft. (Modified after Todd, 1959)

William
20-11-78

$$Q = 2\pi K b \frac{h_o - h_w}{\ln(r_o/r_w)}$$

The head "h" varies linearly with the logarithm of the distance r for all flow rates. Figure 4, included earlier, shows a schematic of the fluctuations in h for a confined aquifer for a system with injection and withdrawal wells. As a result of this relationship, a good approximation of the head loss between a withdrawal and an injection well is twice the sum of the head differential from a well to a point halfway between the wells. Figure 23 shows the pressure as related to head loss between a pair of wells based on these relationships for three values of permeability. The values are aquifer-permeability ranges in the United States.

Aquifers of adequate permeability are prerequisite to the use of water-source heat pump systems. As an example, a pair of 6-inch diameter wells with a flow of 200 gal./min. (2.88×10^5) gal./day) between the wells spaced 500 ft. apart in an aquifer 100 ft. thick would produce a head differential of approximately:

0.04 ft.	at a permeability of 25,000 gal./day ft. ²
0.09 ft.	at a permeability of 10,000 gal./day ft. ²
0.90 ft.	at a permeability of 1,000 gal./day ft. ²
9.00 ft.	at a permeability of 100 gal./day ft. ²
90.00 ft.	at a permeability of 10 gal./day ft. ²
900.00 ft.	at a permeability of 1 gal./day ft. ²

Based upon the above calculations, a source that would serve the system design proposed in this study is limited to aquifers with permeabilities somewhat greater than 1 gal./day-ft.². A minimum of more than 30 gal./day-ft.² would be more realistic.

The flow patterns in the aquifers, resulting from low velocities, i.e. less than 1 ft./sec. are essentially potential flow except in the immediate vicinity of a stream or well. This has been verified by laboratory

FIG. 23 WELL PRESSURE DROP AS A FUNCTION
SPACING AND PERMEABILITY

Dillon
20-11-78

and field tests.^{20,21,22} Flow patterns for a pair of wells, one injection and one withdrawal, are shown in Figures 24 to 27. These flow patterns have been developed assuming potential flow and matching constant stream function and potential function values.^{23,29} The curves show visually the patterns of ground water velocities under various conditions. Figure 24 includes only the flow from the wells. Figures 25 to 27 have imposed ground flows. The ground-flow velocity has been arbitrarily chosen so its magnitude is equal to one-half the pre-existing velocity. The flow patterns indicate that either well could be used as the injection or withdrawal well. Upon withdrawal of water, however, a mixing of aquifer water and injected water will occur after a period of time. The result will be a decrease in system performance of a few percent. In turn, the mixed water will provide a buffer zone where the supply water is not completely cold or warm. This effect will be more noticeable with the second and third alternative systems where some direct heating and cooling occur with the water.

A potential flow field with equal sources and sinks (withdrawal wells and injection wells) has the velocity profile shown in Figure 29. By integrating the velocity (function of flow rate) with respect to time along the center lines between the two wells, a relationship between distance, time, and flow rate can be established. Figure 28 gives the time required for the water to traverse between the 2 wells where flow rate is a function of flow rate per unit thickness of the aquifer.

Since only a portion of the volume (porous part of the aquifer) is water-bearing, the actual velocity, based on the concept of continuity, is

FIG. 24 IDEAL STREAMLINE FOR INJECTION WELL AND DISCHARGE WELL
WITH EQUAL FLOW RATES WITHOUT GROUND FLOW

Rehan
4-12-78

NOTE: Ground flow equals half of the velocity directly between the center of the wells with zero ground flow.

FIG. 25

STREAMLINES FOR INJECTION AND DISCHARGE WELLS WITH EQUAL FLOW AND GROUND FLOW IN THE SAME DIRECTION

Dillon
20-11-78

NOTE: Ground flow equals half
of the velocity directly
between the center of the
wells with zero ground
velocity.

FIG. 26 STREAMLINES FOR INJECTION AND DISCHARGE WELLS WITH EQUAL
FLOW AND GROUND FLOW PERPENDICULAR TO THE FLOW

Sullivan
20-11-78

NOTE: Ground flow equals
half of the velocity
directly between the
center of the wells
with zero ground flow.

FIG. 27 STREAMLINES FOR INJECTION AND DISCHARGE WELLS WITH EQUAL FLOW
AND GROUND FLOW OPPOSING

Dellman
20-11-78

FIG. 28

Dellman
20-11-78

appreciably greater. Velocity will increase by the ratio of void per unit of cross-sectional area. It is assumed that this ratio relates to porosity as defined on a volume basis rather than an area basis. To determine the flow rate in an aquifer, the actual flow rate must be divided by the porosity. This means a flow rate of 100 gal./day per ft. of aquifer thickness with 10% porosity requires the same time to traverse between 2 wells as 1,000 gal./day per ft. though 100% porosity, or unrestricted flow.

For the previous example of 200 gal./min. ($2.88(10^5)$ gal./day) between 2 wells 500 ft. apart in a 100 ft.-thick aquifer, the corrected flow rate and the time for water to move between the wells are:

<u>Corrected Flow Rate</u>	<u>Traverse Time</u>	<u>Porosity</u>
$2.88(10^3)$ gal./day-ft.	Over 1 year	100%
$5.76(10^3)$ gal./day-ft.	Over 1 year	50%
$2.88(10^4)$ gal./day-ft.	About 3 months	10%

For the proposed system, the time for the water front to pass between the 2 wells is unimportant, but the time for the temperature front to pass between the 2 wells is critical.

To determine the velocity of the temperature front, 2 assumptions are made. First, the water temperature and rock temperature at any location in the aquifer are assumed to be identical. This is valid if the contact area between the water and rock is very large for the amount of heat transfer between the masses, which condition is true where the flow channels are numerous and very small, such as the passages for water in a bucket of sand. Second, the thermal conductivity through aquifers is very small or basically negligible. This is supported by the fact that relatively small temperature changes take place underground. Measurements at The University of Alabama have detected less than a $1/2^{\circ}\text{F}$ temperature change at 8 ft.

below the surface over an extended period of time. In the proposed systems well spacing is expected to be in the range of a few hundred feet (100-1,000 ft.), further supporting frontal-movement assumptions utilized herein.

Based on these assumptions, a temperature front follows the injected water front through the aquifer as a step function. This also means the temperature-wave process is reversed when the water is withdrawn. The result is that the energy stored in the aquifer is retrieved at the same temperature level as when injected. Or, stated otherwise--80°F water injected into a 50°F aquifer will theoretically be recovered at 80°F. The situation in practice will be very close to the theoretical assumption. The phenomenon is demonstrated physically every day in the home. When the hot water valve of a mixing spigot is turned on, hot water begins to move through the pipe. As the water moves through the pipe, the pipe acquires the same temperature as the water. Once the water begins to be warm at the spigot, it is hot almost instantly. This demonstrates injection. The recovery is demonstrated as the cold water valve is opened. The water remains warm while the pipe is warm but turns cold almost instantly with the flow of cold water. The temperature function moves as a step function at a velocity lower than water velocity. The aquifer process will be much closer to the theoretical case. The time required for the water to change temperature at the spigot always seems excessively long. This occurs because the temperature wave moves slower than the velocity wave or at a relatively low velocity compared to the water front. In an aquifer, the temperature-front velocity is proportional to the ratio of the fluid thermal capacity divided by fluid-plus-rock thermal capacity times the water front velocity or:

$$V_{\text{Thermal}} = \frac{\rho_{\text{H}_2\text{O}} C_{p_{\text{H}_2\text{O}}} \phi}{\rho_{\text{H}_2\text{O}} C_{p_{\text{H}_2\text{O}}} \phi + \rho_{\text{rock}} C_{p_{\text{rock}}} (1-\phi)} V_{\text{H}_2\text{O}}$$

where ρ equals density

C_p equals specific heat, and

ϕ is aquifer porosity

To determine the time required for the temperature wave to move between wells, the time for the water must be multiplied by the reciprocal of the same thermal capacity ratio as the velocities. The times, given in Figure 28, change as:

$$t_{\text{Thermal}} = \frac{\rho_{\text{H}_2\text{O}} C_{p_{\text{H}_2\text{O}}} \phi + \rho_{\text{rock}} C_{p_{\text{rock}}} (1-\phi)}{\rho_{\text{H}_2\text{O}} C_{p_{\text{H}_2\text{O}}} \phi} t_{\text{Water}}$$

The time for the temperature front to move through the aquifer is the critical time, not the time of water-front movement. Well spacing and water flow rates must be considered on the basis of thermal times.

The ratio of t_{Thermal} to t_{Velocity} equals

10.88 for a porosity of 0.05

5.68 for a porosity of 0.10

3.99 for a porosity of 0.15

or the temperature frontal-movement time for 200 gal./min. in an aquifer 100 ft. in thickness with 10% porosity is 5.68 times 3 months, or about 17 months, as indicated in the early calculations. This gives a "safety factor" of approximately 3 in determining well spacing. A 100 gal./min. well in an aquifer 50 ft. thick with 10% porosity would result in the same thermal travel time. In all the states, information is available with respect to ground water availability. Data collected by state geological surveys, the U. S. Geological Survey, and other water-resources organizations include aquifer depth, nature, thickness, and potential yield. Water-quality

analyses are available in many instances. In Alabama many of the wells drilled, both private and public, are logged, tested for water flow, and analyzed for mineral content in the water. Figure 29 is a map of Alabama depicting 5 formations suited for supply wells in the proposed system. Also shown are the several wells which have been logged and tested. It may be noted the formations are identified by local geographic names. These equivalent formations are referred to by different names in different states. Figure 30 shows the extent of these formations in a portion of the United States.

IX.C. Wells

The well is the conduit between the aquifer and the surface. Well construction is governed by a number of factors such as depth, type of soil and rock penetrated, aquifer characteristics, expected yield, and cost. These variables also relate to the method of drilling employed, the diameter of the hole, and the casing program.

Two broad categories of well types are generally recognized in the United States--first, wells which penetrate relatively "soft" soil and rock and are completed in water-bearing alluvium or sand and gravel above bedrock and second, wells which penetrate are completed in indurated rock such as sandstone, limestone, or dolomite. In certain instances, metamorphic and igneous rocks containing fractures or other voids may serve as suitable aquifers.

Three basic types of drilling equipment are employed, percussion or cable-tool rigs, augers, and rotary tools. The choice of equipment is governed primarily by the factors noted above. Of these, cost is a major consideration.

The cost of construction of water wells in the United States falls within a rather broad range for wells of a given depth, related primarily to the type of geologic formation penetrated. Table 39 gives average price estimates as of 1978 for construction of wells in the depth categories of 100 and 200 ft. Wells within this depth range and diameters shown in the table will generally provide adequate water for domestic or light industrial use, although there is no means other than experience and intimate knowledge of aquifers in an area to predict the yield of a particular well. Other things being equal, well diameter, a controllable factor, is one determinant of the volume of flow from an aquifer into the well.

In considering cost estimates for the construction of wells, provision should be made for contingencies that may be of considerable significance. As an example, a particular project may have to absorb the cost of test holes or unsuccessful wells. Enlarging the diameter of a well by reaming may be required to increase yield. Other unforeseen problems may arise that add materially to the cost of a supply system.

The ideal aquifer for this system might be one in the depth range of 100 to 150 ft. encountering 50 or more ft. of porous and highly permeable water-bearing section. In the study area, an example would be sands and gravels within the Tuscaloosa group; the areal distribution of these formations is shown in Figure 29. Wells of 6-inch diameter constructed in aquifers of this type should be expected to yield in excess of 100 GPM, adequate to serve a 20-home module.

The well system used for this study consists of a 6-inch diameter well with an assumed water-head depth of 30 ft. Lowering of head in the withdrawal well by 20 ft. raises the head in the injection well by 20 ft., assuming the aquifer properties are constant and flow rates are

Table 39. Average Water-Well Drilling Costs in the United States, 1978^(a)

<u>Casing diameter, inches</u>	<u>Aquifer type(b)</u>	<u>Drilling Method(c)</u>	<u>Depth, Feet</u>	<u>Cost Range Dollars(d)</u>
4	Sand and Gravel	Cable tools	200	1500-1750
6	Sand and Gravel	Cable tools	200	2400-2800
6	Sand and Gravel	Rotary	100	1500-1750
24	Sand and Gravel	Auger	100	1200-1575
6	Sandstone and Limestone	Rotary	100	900-1200
6	Sandstone and Limestone	Rotary	200	1550-1900

- (a) Personal communication, Mr. Michael Everly, National Water Well Association, Worthington, Ohio 43085. Additional references: Gibb, J.P., Cost of Domestic Wells As Water Treatment In Illinois, Circular 104, Illinois State Water Survey, Urbans, 1971; and Cederstrom, D.J., Cost Analysis Of Ground-Water Supplies In The North Atlantic Region, U.S. Geological Survey Water-Supply Paper 2034, 1973.
- (b) Refers to (1) generally unconsolidated sand, gravel and siltstones occurring above bedrock as opposed to (2) indurated sandstones, limestones and dolomites, or "bedrock".
- (c) The percussion or cable-tool method of drilling, used extensively in early years, is still applicable in areas where geological formations are competent. Rotary drilling may employ either air or hydraulic systems for removal of drill cuttings.
- (d) Costs based on 1970 data adjusted to 1978 economic level, including installed curbing or casing and screens as required.

In many cases, a considerable part of drilling costs involves the expense of moving in, setting up, transporting supplies (principally water), and moving out. The proposed system plans to locate wells along right-of-ways or other accessible places, to space wells no more than 1,000 feet apart, and to enter into multiple-well contracts, thus minimizing moving and related costs. In addition, rock types will be known, which will serve to reduce contingencies that add to contract prices.

equal. The ideal work required for primary pumping at a flow rate of 5 GPM, 50-ft. head in the well, 10-ft. head in pipelines (well below values for 100 GPM shown in Figures 8 and 9) and 30 psig pipeline pressure (60-ft. head) is:

$$\begin{aligned}\text{Power} &= 5 \text{ GPM} \times \text{Total Head} = 5 \text{ GPM} (8 \text{ lb./gal.}) \times (50+10+60) \text{ ft.} \\ &= 4800 \text{ ft. lb./min.} = 100 \text{ watts}\end{aligned}$$

At a pump efficiency of 50%, primary pumping per 3-ton water source heat pump is 200 watts. Secondary pumping has a much smaller head loss (less than 50 ft.), therefore pumping power input is assumed as 100 watts. The total pumping power is the total or 300 watts. This is the value used for performance and economic evaluations. Some evaluations also use 500 watts. To illustrate the comparison, water source heat pumps are designated as water source A(300) and water source A(500).

Performance and cost data are given in Table 40 for Meyer 6-inch SC submersible pumps. This is a commercially-rated submersible pump. For the economic evaluation, the 5-horsepower, 4-stage pump is utilized.

The cost data are:

Pump cost, 4-stage	\$ 475.00
5-horsepower, 3 phase, 230 volt	\$1007.00
Control system (pressure aquated)	<u>\$ 195.00</u>
Total	\$1677.00

This is the suggested list price for a single unit delivered to an individual. Actual prices installed can be 30% to 50% under this figure. For the economic section, assuming numerous simultaneous installations, a cost of \$1,500 per well including piping, check valves, pressure release valve, grounding, etc., is used for the installed system.

Table 40. Performance Tables for Submersible Pumps¹

Depth to Water in Feet			50	75	100	125	150	175	200
Pump Cost	Stage No.	Pressure Psia	Capacities in U.S. Gallons per Minute						
5 Horsepower-Motor Cost ² \$1007.00, Control Cost ² \$195.00									
\$475.00	4	30	117	106	91*	72			
	4	60	75						
\$616.00	8	30	90	85	81	77	71	65*	57
	8	60	78	72	66*	58	49	35	
7.5 Horsepower-Motor Cost ² \$1258.00, Control Cost ² \$195.00									
\$475.00	4	30	163	147	126*	108	80		
\$599.00	6	60	113	105	95*	84*	70	52	
10 Horsepower-Motor Cost ² \$1549.00, Control Cost ² \$249.75									
\$475.00	4	30	199	180	158*	129	90		
	4	60	141*	106					
\$536.00	5	30	180	169	157	144	130*	114*	91
	5	60	150	136	120*	97	66		
\$722.00	8	30	135	129	124	119	113	107	101
	8	60	122	116	110	104	97	89*	82
Prices of Basic Components of Larger Systems									
	15 HP	Motor \$1771.00		Controls \$369.00		6 Stage Pump \$599.00			
	20 HP	Motor \$2635.00		Controls \$478.00		8 Stage Pump \$722.00			

* Flow at peak efficiency

1. Data based on Meyer 6-inch SC submersible pumps (Price data for Fall 1978)

2. For 3 phase 230 volt service

Wells can be constructed to satisfy this system. As noted earlier, well costs will vary from area to area and can not be determined until the quality and extent of aquifers are known.

The capacitance tank at each well head will be in the 40 gal. to 200 gal. range. The 1978 price ranges on these tanks as provided by the Meyer Pump distributor are:

	<u>Plain Tank</u>	<u>Diaphragm Tank</u>
42 gal.	\$ 55	\$--
120 gal.	130	129
160 gal.	--	165

A diaphragm tank is recommended since a larger water volume can be utilized and an interface between air and water is eliminated. A cost of \$130 is used for economic analysis.

IX.D. Water Distribution System

The water distribution system transfers the water from the aquifers to the individual heat pumps in the home. The piping system must be both economical and compatible to the heat pump. Metal pipes such as steel or cast iron were eliminated on the basis of corrosion of dissimilar metals as well as economics. Copper and brass were eliminated primarily on an economic basis. As a result, polyvinyl chloride pipe is proposed as the distribution pipe.

The following specifications of polyvinyl chloride piping is a summary of material from the Piping Handbook.²⁵

Rigid, unplasticized polyvinyl chloride (PVC) is a thermoplastic material; it is tough and exceptionally resistant to chemical attack. Plastic pipe is generally available in iron pipe sizes (IPS). Typical sizes and weights for PVC are given in Table 41.

Table 41. Normal Physical Properties of PVC Piping Materials ²⁵

Properties	Materials		ASTM test no.
	Type I	Type II	
Tensile strength, room temperature, psi	7,000	6,000	D638
Modulus of elasticity in tension, psi	415,000	350,000	
Flexural strength, psi	14,500	11,500	D790
Izod impact strength, ft.-lb/in. of notch, notched, room temperature	0.5-1.0	10.0-19.0	D256
Izod impact strength, ft.-lb/in. of specimen unnotched, room temperature	45	55	D256
Specific gravity	1.38	1.35	D792
Hardness, Shore "D"	78/82	76/80	D785
Heat distortion, temperature, F, at 264 psi	165	155	D648
Coefficient of thermal conductivity, BTU/(sec) ft. ² x 10 ⁴	3.5	4.5	C-177
Specific heat, cal/(g.C)	0.25	0.25	
Coefficient of linear thermal expansion per C x 10 ⁵	5	10	D696
Water absorption, % in 24 hr. at 25 C	0.07	0.07	D570

PVC pipe is extruded. Fittings, flanges, and valves are manufactured by the injection-molding method which results in high density and complete homogeneity of the material. Three types of PVC piping are available.

Type I in the past has generally been marketed as "normal-impact" grade. It is now produced to a hydrostatic design stress of 2,000 psi for water at 73.4°F. Two grades are recognized under the designations PVC 1120 and PVC 1220.

Type II has been marketed as "high-impact" grade. Type II, Grade 1 is produced to a hydrostatic design stress of 1,000 psi for water at 73.4°F and is designated as PVC 2110. It is also produced to a hydrostatic design stress of 1,250 psi and is designated as PVC 2112; material produced to a hydrostatic design stress of 1,600 psi is designated PVC 2116.

Type IV is a newer grade produced to a hydrostatic design stress of 1,600 psi for water at 73.4°F. It is designated as PVC 4116.

Fittings and valves of Types I and II are readily available with threaded, solvent weld (socket), and flanged ends.

The 2 Type I PVC grades have greater strength properties over a wider temperature range. They can be used at temperatures up to 160°F and, in addition, at higher working pressures than can Type II PVC. These grades have superior chemical resistance throughout their temperature range.

Type I PVC grades should be specified in applications where greater strength, temperature resistance, or extreme chemical resistance is necessary. PVC is normally straw-colored without pigmentation. Industrial-process piping is usually colored dark gray for identification purposes but also may be produced as white, brown, red, or any other color.

PVC pipe is covered by ASTM Standard D1785, Tentative Specifications for Poly (Vinyl Chloride) (PVC) Plastic Pipe. Voluntary industry standards have also been issued by the U.S. Department of Commerce: Commercial Standard CS256, Polyvinyl Chloride (PVC) Plastic Pipe (SDR-PR and Class T) and Commercial Standard CS207, Rigid Unplasticized Polyvinyl Chloride Pipe. The normal physical properties of PVC are summarized in Table 42.

PVC piping is extensively used in highly corrosive applications involving acids, alkalies, salt solutions, alcohols, and many types of chemicals. PVC piping is also used in oil fields because it can carry sour crude oil to which PVC is chemically inert and because paraffin build-up is minimum on the smooth inside surfaces of this pipe. Other applications include salt-water disposal in oil fields and gas transmission service. PVC will handle most chemicals up to 150°F. Still other applications include the piping of cold water in industrial plants because PVC is nontoxic and will not impart odor or taste to the water, and also vent piping for the removal of acid fumes and corrosive gases from industrial plants.

PVC pipe and fittings experience little or no physical deterioration when exposed to direct sunlight; sunlight causes deterioration in several other plastic piping materials. Like other plastic materials, PVC will not produce sparks when struck. PVC is safe to use around explosives or flammable vapors, and it does not support combustion. Also, water contaminants do not build up on the smooth walls of PVC pipe or fittings.

Because polyvinyl chloride, like most other plastic materials, is somewhat notch-sensitive, maximum allowable working pressures for threaded pipe are considerably lower than those for unthreaded pipe. The strength of polyvinyl chloride decreases as the operating temperature increases,

Table 42. Commercial Sizes (IPS) and Weights of Polyvinyl Chloride (PVC)
Pipe (Abstracted from ASTM Specification D1785-64T)

Nominal size, in.	Schedule	Wall thick- ness,**in.	OD, in.	ID, in.	Theoreti- cal weight,* lb/ft.	Calculated min bursting pressure, psi	
						Note 1	Note 2
1/4	40	0.088	0.540	0.364	0.076	2,490	1,950
	80	0.119	0.540	0.302	0.096	3,620	2,830
1/2	40	0.109	0.840	0.622	0.153	1,910	1,490
	80	0.147	0.840	0.546	0.195	2,720	2,120
3/4	40	0.113	1.050	0.824	0.203	1,540	1,210
	80	0.154	1.050	0.742	0.265	2,200	1,720
1	40	0.133	1.315	1.049	0.305	1,440	1,130
	80	0.179	1.315	0.957	0.385	2,020	1,580
1 1/4	40	0.140	1.660	1.380	0.409	1,180	920
	80	0.191	1.660	1.278	0.550	1,660	1,300
1 1/2	40	0.145	1.900	1.610	0.489	1,060	830
	80	0.200	1,900	1.500	0.653	1,510	1,180
2	40	0.154	2.375	2.067	0.640	800	690
	80	0.218	2.375	1.939	0.910	1,290	1,010
3	40	0.216	3.500	3.068	1.380	840	660
	80	0.300	3.500	2.900	1.845	1,200	940
4	40	0.237	4.500	4.026	1.965	710	560
	80	0.337	4.500	3.826	2.710	1,040	810

*These representative values are not specified in ASTM D1785-64T.

**Thicknesses listed are minimum values. Tolerance is generally -0 + 10 per cent.

Note 1. Materials are PVC 1120, 1220, and 4116. A fiber stress of 6,400 psi was used in bursting pressure calculations.

Note 2. Materials are PVC 2112, 2116, and 2120. A fiber stress of 5,000 psi was used in bursting pressure calculations.

and allowable working pressures must be decreased at higher temperatures. Working pressures at 75°F are approximately 20% of the bursting pressures. When operating temperatures exceed 75°F, the maximum operating pressure for Type I pipe may be determined from Figure 31.

Type I is being considered for the piping system.

Cost analysis has been made for the various system alternatives shown in Section VII (Figures 17 to 20). The results are listed in Table 43. The prime costs are for pipes, pipe installation, and insulation.

The 2-pipe system has been selected on the basis of simplicity, performance, and cost. The 1-pipe system, though more economical, penalizes customers down the line with reduced performance. It also reduces performance with simultaneous heating and cooling.

The pressure losses for various pipe diameters and flow rates are shown in Figures 8 and 9. A modified computer code (Appendix II) for calculating pipe-pressure loads was used for the analysis. Laminar flow is assumed in the piping which is reasonable for the smooth pipe with limited flow. The system assumes each home receives the allotted water flow rate, thereby reducing the flow. This occurs at the given separation distances noted on the curve for 2 homes at each cutoff. The total pressure loss is the pressure loss through the 10 increments. Using pipe diameter of over 4 inches keeps the pressure losses at negligible values and allows flow both ways between the pipes. Change in elevation will not effect the pressure between pipes as they run parallel and adjacent.

Connections between the main lines and homes is to be made with 1½-inch or 2-inch pipe. This will keep the pressure loss for flows up to 20

FIG. 31 MAXIMUM OPERATING PRESSURE OF NORMAL-IMPACT (TYPE I)
PVC AT ANY TEMPERATURE

William
20-78

Table 43.⁶ COST OF DISTRIBUTION PIPE¹, INSTALLATION², AND TRENCHING³

DIAMETER (INCHES)	UNINSULATED PIPE			PIPE COST PER LINEAR FOOT		INSULATION COST PER LINEAR FOOT (7)	TRENCHING COST PER LINEAR FOOT	
	1	2	3	2 INSULATED; 1 UNINSULATED	4 INSULATED PIPES		EXCAVATION (4)	BACKFILL (5)
1½	1.70	3.40	5.10	8.10 + 1.70 = 9.80	16.20	2.35	1.11	0.37
2	2.05	4.10	6.15	8.90 + 2.05 = 10.95	17.80	2.40	1.11	0.37
4	3.20	6.40	9.60	14.30 + 3.20 = 17.50	28.60	3.95	1.11	0.37
6	4.60	9.20	13.80	23.70 + 4.60 = 28.30	47.40	7.25	1.11	0.37
8	7.25	14.50	21.75	32.10 + 7.25 = 39.35	64.20	8.80	1.11	0.37

1. The costs stated above are for Polyvinyl Chloride (PVC) pipe, class 160, SDR 26.
2. The pipe cost stated above includes material, installation, and subcontractors overhead and profit.
3. The trenching costs are stated for a 4 foot wide, 3 foot deep trench which would accomodate any combination of pipes.
4. This cost is for the use of a ½ cubic yard tractor backhoe.
5. This figure is made on the basis of using a dozer with up to a 300 foot hard distance and no compaction.
6. All the figures stated above were taken from and derived from those given in Means' Building Construction Cost Data.
7. The costs for the 1½" and 2" insulation is for ½" thick foam rubber sleeves and the cost of the 4" to 8" sizes is for 2" thick fiberglass insulation.

GPM below 15 ft. of water. A 1/12-horsepower pump (100 watts) can be used to circulate the water between the main water lines and the individual heat pumps. This type system is standard on numerous solar systems.

IX.E. Heat Exchanger Fouling

In many present day water source heat pump systems, the water is circulated in a closed loop. The water is cooled in a cooling tower and heated by waste heat or in some cases directly from other energy sources. Mineral content and pH are closely controlled. In the proposed system, water in the aquifers and mineral additions during water circulation must be evaluated.

Knudson has been active in condenser water fouling research for many years.^{27, 28, 29, 30} His basic premise is that mineral buildup occurs continuously, but erosion of the mineral buildup also occurs continuously. After a period of time the mineral buildup and erosion reach equilibrium. This is demonstrated by plotting fouling resistance " R_f ", the change in the heat transfer coefficient versus time. In general the time to equilibrium varies from 150 to 600 hours. Typical curves are shown in Figure 32.

Knudson's series of papers have shown the variations for a number of minerals. Table 44 shows a typical mineral content for a series of runs with various salts and changing pH. Tables 45 and 46 show typical mineral contents in ground water in certain formations in Alabama. Table 45 shows relatively low levels, and Table 46 shows relatively high levels. Knudson's test data and that for many aquifers cannot be compared directly, however, maximum hardness values may be of similar magnitude.

In his papers, Knudson's maximum value of resistance fouling is of the order of magnitude of $0.0004 \text{ ft.}^2 \text{ hr. } ^\circ\text{F/Btu}$. The value for sea water

FIGURE 32. TIME EFFECT ON HEAT EXCHANGER FOULING
BY LEE AND KNUDSON

Table 44. Average Cooling Tower Water Quality, Runs 6-20

Run	Total Hardness (ppm CaCO ₃)	Ca Hardness (ppm CaCO ₃)	m-alkalinity (ppm CaCO ₃)	k-alkalinity (ppm CaCO ₃)	Cl (ppm NaCl)	pH	TS (ppm)
6	219 (7.8)	152 (4.2)	190 (16.1)	23 (2.2)	243 (22.3)	8.97 (0.10)	613 (47)
7	217 (4.9)	151 (2.4)	191 (6.9)	20 (1.5)	216 (6.3)	8.90 (0.05)	568 (25)
8	221 (9.3)	154 (5.5)	206 (15.1)	21 (1.3)	221 (10.1)	8.89 (0.05)	575 (35)
9,10	216 (8.3)	150 (2.7)	183 (8.9)	23 (1.2)	243 (16.3)	9.01 (0.10)	604 (32)
11,12	218 (5.6)	151 (3.3)	182 (10.9)	25 (1.5)	266 (8.8)	9.03 (0.07)	657 (28)
13,14	218 (7.8)	149 (6.1)	194 (8.5)	25 (1.8)	297 (15.9)	9.00 (0.04)	677 (22)
15	218 (9.1)	148 (5.6)	210 (20.4)	27 (2.4)	299 (15.5)	9.02 (0.05)	683 (45)
16,17	218 (10.3)	148 (4.9)	228 (15.8)	29 (2.5)	301 (14.9)	9.05 (0.06)	691 (59)
18,19	222 (9.7)	152 (4.6)	253 (14.7)	35 (1.2)	320 (9.8)	9.09 (0.05)	745 (68)
20	223 (8.5)	151 (2.3)	251 (12.7)	35 (1.2)	316 (10.4)	9.09 (0.03)	725 (47)

Table 45

Map number	Well	Dates of Collection	Water-bearing Formation	Milligrams per liter														Specific conductance in micromns at 25oC	pH		
				Silica (SiO ₂) (ppm)	Iron (Fe) (ppm)	Manganese (Mn) (ppm)	Calcium(Ca)	Magnesium(Mg)	Sodium (Na)	Potassium (K) (ppm)	Bicarbonate(HCO ₃) (ppm)	Carbonate(CO ₃) (ppm)	Sulfate (SO ₄) (ppm)	Chloride (Cl) (ppm)	Fluoride (F) (ppm)	Nitrate (NO ₃) (ppm)	Dissolved Solids			Total	Hardness as CaCO ₃
1T	Marion Co. Hackleburg	12/10/64	Kg	7.7	0.20	--	2.3	0.3	1.6	0.7	1	0	0.2	3.3	0.1	6.0	26	7	6	34	4.8
2T	Marion Co. Craft Sch.	3/11/64	Kck	--	0.13	--	--	--	--	--	8	0	--	3.0	--	--	--	26	19	--	5.9
3T	Auburn	12/15/66	Kck	9.6	0.33	--	2.0	0.7	1.5	1.2	5	0	0.4	0.1	0	6.3	28	8	4	--	5.3
4T	Harold Wallace	9/21/66	Kg	--	0.19	--	--	--	--	--	8	0	--	4.6	--	--	--	8	1	--	7.6
5T	T. B. Woodward	9/25/62	Kg	8.7	1.6	0.0	9.2	1.9	3.0	4.4	46	0	0	2.3	0.2	0.01	55	31	0	--	7.0
6T	Adams	11/22/60	Kck	17.2	0.21	--	2.3	1.0	3.1		22.5	0	2.8	3.6	0.2	--	35	10.9	--	--	5.4
7T	J. Chaney	9/16/29	Kck	--	1.4	--	14.0	--	--	--	91.0	0	1.0	2.8	--	.05	--	48	--	--	--
8T	U.S.G.S.	11/07/61	Kck	15	.20	--	1.4	0.6	2.3	1.1	3	0	5.6	1.0	0.1	0.0	--	6	4	28	5.5
9T	M. F. Roebuck	8/16/65	Kg	11.0	.33	--	12	2.4	3.2	4.7	104	0	2.6	16	0.1	0.4	135	40	0	261	6.9
10T	E. F. King	8/16/65	Kck	12.0	.15	.04	18	3.7	7.8		91	0		1.3	0		103	60	0	183	7.1

Chemical Analyses of Water Samples From the Tuscaloosa Aquifers.
(Kg-Gordo Formation, Kck-Coker Formation)

31

Table 46. Representative Chemical Analyses of Water from
Wells Tapping the Principle Aquifer of the Eutaw Formation in Alabama 31

Well Number in this Report			County and Well Number in County Report																		Date of Collection																		Silica (SiO ₂)																		Iron (Fe)																		Manganese (Mn)																		Calcium (Ca)																		Magnesium (Mg)																		Sodium (Na)																		Potassium (K)																		Bicarbonate (HCO ₃)																		Carbonate (CO ₃)																		Sulfate (SO ₄)																		Chloride (Cl)																		Fluoride (F)																		Nitrate (NO ₃)																		Dissolved Solids																		Total																		Noncarbonate																		Specific conductance (micromhos at 25°C)																		pH																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																							
1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E	2E	3E	4E	5E	6E	7E	8E	1E</

1/ Calculated Na + K reported as Na.

2/ Calculated

is given as $0.0005 \text{ ft.}^2 \text{ hr } ^\circ\text{F/Btu.}$ The effect of this level is shown to be minor.

The general range of heat transfer coefficients for a Freon 12 condensor is 80 to 150 Btu/ft.² hr. °F. The reciprocal or resistance therefore varies from 0.012 to 0.00667 ft.² hr. °F/Btu. The change due to mineral fouling is approximately 1%. This is not a major problem.

Possibilities of heat exchanger fouling and aquifer clogging resulting from biological growth must be evaluated in the laboratory or in demonstration projects. These variables can be controlled, if necessary, by the addition of chemical inhibitors.

The problem of aquifer clogging resulting from solution and precipitation of minerals in small openings can be minimized by controlling the pH level. The velocity levels in most aquifers is very small. Therefore the fouling of heat exchangers is expected to be minimal.

IX.F. Solar Energy Addition

The system makes possible an economical method of utilizing solar energy to help heat buildings. A large portion of the continental United States requires much larger quantities of energy for heating than cooling. If waste heat is not available or the proper mixing of buildings is not practicable, solar energy is an alternative. With annual thermal storage and the relatively low collection temperatures, basic swimming pool heaters are a possibility. Summer usage eliminates the freezing problem and the requirement for glazing thereby allows direct system water circulation.

Table 47 gives the horizontal solar energy collection quantities ^{32,33} for the summer months per 100 ft.² of collector area. Collection efficiencies are based on Figure 33 where the June through August solar energy

TABLE 47 SOLAR ENERGY COLLECTIONS ON 100-SQUARE FOOT HORIZONTAL COLLECTOR*

CITIES	Energy Collected - 10^3 Btu						JUNE-AUGUST OVER ALL
	APRIL	MAY	JUNE	JULY	AUGUST	SEPTEMBER	
Birmingham, AL	3301	3708	3803	3782	3337	2870	10922
Tuscon, AZ	4306	5191	5095	4746	4227	3588	14068
Los Angeles, CA	3588	4078	3947	4078	3708	3229	11733
Denver, CO	3660	4449	4593	4449	4004	3229	13046
Miami, FL	3660	3782	3660	3708	3337	2870	13575
Atlanta, GA	3373	3708	3732	3856	3337	2870	10925
Chicago, IL	2942	3708	3947	4078	3337	2512	11362
New Orleans, LA	3157	3708	3660	3337	3337	2870	10334
Boston, MA	2942	3559	3947	3782	3337	2583	11066
Detroit, MI	2727	3337	3732	3856	3337	2153	10925
Saint Louis, MO	3086	3782	4019	4078	3634	2799	11731
New York, NY	3014	3708	3947	3708	3337	2655	10992
Akron, OH	2799	3337	3947	4078	3337	2511	11362
Tulsa, OK	3516	4004	4234	4449	3856	3086	12539
Portland, OR	2942	3708	3947	3708	2966	2512	10621
Philadelphia, PA	3014	3708	3947	3708	3337	2655	10992
Providence, RI	2942	3708	3947	3782	3337	2583	11066
Columbus, SC	3445	3708	3588	3708	3337	2870	10633
Nashville, TN	3157	3708	3947	4004	3337	2870	11288
Houston, TX	3373	3708	3732	3708	3708	3014	11148
Salt Lake City, UT	3588	4449	4664	4597	4078	3229	13339
Seattle, WA	2870	3708	3732	3782	2966	2512	10480
Milwaukee, WI	2870	3559	3588	3856	3337	2153	10781

*Based on manufacturers test data
Average efficiency 0.65
No glazing

FIGURE 33 Measured Efficiency Of Solar Collector 39

Figure 34

OPTIMUM COLLECTOR INCLINATION vs. LATITUDES

39

total is also shown. An increase in performance will be made by inclining the collectors. Optimum angle of inclination is shown in Figure 34 for various time periods.³⁴

In a community energy system, solar collectors are not required on each building or surface area. They can be installed, operated, and serviced on a building with optimum locations. Placing the panels on a roof helps the occupant by decreasing the heating and cooling loads without involving any inconvenience. Figure 35 shows a typical installation.

Figure 36 shows schematically a hard rubber solar collector without glazing, and Figure 37 shows schematically a black bag solar collector.^{34,35} Both collectors retail between \$2.50 and \$3.50 per ft.² and are designed for roof collection. Figure 38 shows a proposed solar pond collector and an estimation of installed costs with required pump and controls.³⁶

For the economic analysis, an installed 100⁰ft.² panel is included for energy addition at a cost of \$3.00/ft.² or \$3000 total. It must be emphasized that this would be used only if waste heat is not available at a competitive cost.

IX.G. Minor Related Systems

Providing water for fires is a definite potential of the system. In general, water requirements are determined by insurance underwriters and vary neighborhood-by-neighborhood with no set standards. Commonly residential requirements vary from 900 GPM to 2500 GPM. A mall or major building demands about 4000 GPM. City water lines in Tuscaloosa, Alabama, are predominately 8-inch and 6-inch lines for distribution to provide fire protection. Without fire protection requirements these would nor-

Figure 37. PANEL CONFIGURATION FOR SOLAR BAG COLLECTOR

FIGURE 38. TYPICAL SOLAR POND INSTALLATION WITH RELATIVE COSTS

Solar Pond Costs - ERDA/Sohio Project.

	Low cost - Air Inflated (PVC)	Long life - Panel top (Acrylic)
1. Site preparation	\$0.10 per ft. ²	\$0.10 per ft. ²
2. Moisture barrier	0.04	0.04
3. Bottom insulation	0.37	0.37
4. Water bag and glazing assembly	0.70	0.77
5. Curbing and mounting		0.33
6. Assembly hardware and installation	0.08	0.05
7. Piping and fittings	0.05	0.05
8. Pumps and motors	0.06	0.06
9. Reservoirs - hot	0.22	0.22
10. Reservoirs - cold	0.20	0.20
11. Instruments and controls	0.05	0.05
Total	1.84 per ft. ²	2.24 per ft. ²

mally be 4-inch and 2-inch lines. The larger system costs double the minor system costs using cast iron pipes. Some PVC pipe is being used in 2, 4, 6, and 8-inch diameters. Pumping capacity could be cut by 40% if fire protection is not included.

The connected module integrated community energy system has the capacity to provide the required water for short periods (time measured in days). Primary advantages are shorter pumping distances and relatively lower pumping requirements. Fire plugs utilizing both lines of the 2-pipe system could be installed. An additional check valve to each line would be required to eliminate cross flow. By reducing pressure in the lines with the fire engine pump, the submerged pumps will more than double the flow capacity. The 2 wells in a module will provide over 800 GPM under these conditions. Five connected modules will provide water for most any emergency.

Small neighborhoods without city water have the potential for fire protection water with this system. Such systems would reduce fire insurance on the average home by over \$100 per year, more than paying the utility fee for heating and cooling water usage. The fire protection possibility shows at least one fringe benefit available from the system.

Long term water withdrawal such as for irrigation has not been considered because of the large requirement. Using the water as potable water is not considered inasmuch as treating and monitoring installations would be required for each module. However, treated potable water could be provided using a separate but parallel supply system, eliminating the need for additional wells for drinking water.

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XI. APPENDIX I. ANNUAL HEATING AND COOLING
PERFORMANCE COMPUTER PROGRAM SUMMARY

This program is designed to compute annual heating and cooling efficiencies and energy requirements for any specific combination of heating/cooling unit, building, and city. It accomplishes this using input data describing heating/cooling system performance, building design parameters, and detailed weather data for specific cities. It can represent a home or major building by changing internal energy input.

Weather data used in the program have been abstracted from published NOAA reports (hours in 5°F temperature increments) and ASHRAE tables (99% design temperatures) to give, for each city, 30 parameters. These consist of 28 values for the number of hours per year in which temperature falls within 5 degree increments, starting at -18°F; the last two items are cooling and heating design outdoor temperatures, respectively. Heating and cooling design temperatures are 99% values taken from ASHRAE Fundamentals Handbook.

System parameters are abstracted from manufacturer's data for production systems, interpolated as necessary to fill the same 5°F increments used for weather data. Information stored for each 5°F temperature increment includes:

1. COOLING CAPACITY - MBTUH
2. COOLING POWER INPUT - KW
3. HEATING CAPACITY - MBTUH
4. HEATING POWER INPUT - KW

For maximum flexibility 5 inputs are required to specify building design conditions. These are:

1. Design Cooling Load - heat gain due to convection, radiation, and conduction at given design conditions
2. Design Cooling Indoor Temperature - the indoor temperature to be maintained during cooling plant operation
3. Design Heating Load - heat loss from building due to conduction, convection, and radiation under given design conditions
4. Design Heating Indoor Temperature - indoor temperature to be maintained during heating mode
5. Internal Load - heat gain in building due to occupants, machinery, lights, etc.

It was felt that these represent the minimum information necessary for a satisfactory working description of building loads.

The program's basic calculations assume that during both heating and cooling, heat transfer between a building and its environment is directly proportional to the temperature difference between indoors and the environment, so that:

$$\text{Heat Load Due to Heat Transfer} = (\text{Design Load}) \frac{(\text{Thermostat temp.} - \text{Outdoor temp.})}{(\text{Design indoor temp.} - \text{Design outdoor temp.})}$$

and

$$\text{Cooling Load Due to Heat Transfer} = (\text{Design Load}) \frac{(\text{Thermostat temp.} - \text{Design indoor temp.})}{(\text{Outdoor design temp.} - \text{Design indoor temp.})}$$

Then:

$$\text{Total Heating Load} = (\text{Heat Load Due to Heat Transfer}) - \text{Internal Load}$$

$$\text{Total Cooling Load} = (\text{Cooling Load Due to Heat Transfer}) + \text{Internal Load}$$

The program combines this information along with system performance data to calculate system run times as follows. First it computes heating load. If negative, it computes cooling load. If that too is negative then the program assumes the heating/cooling unit will be off. Otherwise it divides the load by system capacity to get run time in percent. If run time computed is over 100% for heating mode, resistance heat is added to reduce it to 100%. If 100% run time is exceeded in the cooling calcula-

tion, run time is reduced to 100% and cooling load is restricted to system capacity. Run time is multiplied by system power requirements and the number of hours in each 5°F increment to give seasonal power input at that temperature. Seasonal temperature hours at temperature are also multiplied by heating/cooling loads to give seasonal loads. Seasonal values are summed for computation of annual values of heating load, cooling load, annual heating input, and annual cooling input. Coefficients of performance are also calculated,

$$\text{COP} = \frac{\text{Total Load}}{\text{Total Power Input}}$$

as are equivalent heating fuel values, which assume 62% furnace efficiency for fossil fuel furnaces.

$$\text{Equivalent fuel required} = \frac{\text{Total Heating Load}}{\text{Heat Content per unit fuel} \times 62\%}$$

The program also incorporates provisions for loops to accomplish serial solutions for varying cities, systems, and design conditions.

A table defining the variables in the computer programs, a flow chart, and a program listing follow.

PROGRAM VARIABLES

CITY - MATRIX VARIABLE, contains encoded city design data

CITNAME - A/N names for cities in matrix

NO CIT - Integer number of cities in data bank

SYSTEM - MATRIX VARIABLE, containing performance data (loads and capacities) for all systems

SYSNAM - A/N names of systems in matrix

NO SYS - Integer number of systems in data bank

CODE - Integer code number of individual city

SYSNO - Integer code number of individual system

I,J,K,L - Integer variables used Sy do loops

IDT - Heating indoor design temperature, °F

CIDT - Cooling indoor design temperature, °F

LOAD - Heating design load, BTUH

CLOAD - Cooling design load, BTUH

ILOAD - Internal load, BTUH

TT - Thermostat temperature in building, °F

ODT - Heating outdoor design temperature, °F

CODT - Cooling outdoor design temperature, °F

SRT - Summation register for annual run time

SH - Summation register for annual resistance heat

SQT - Summation register for annual heating load

SCQT - Summation register for annual cooling load

SF - Summation register for annual heating energy input

SCF - Summation register for annual cooling energy input

COP - Coefficient of performance in 5°F increment

CCOP - Annual cooling coefficient of performance

- COPH - Annual heating coefficient of performance
- EGAS - Equivalent annual gas heat required
- EOIL - Equivalent annual oil heat required
- A - Outdoor temperature, °F
- B - System heating/cooling capacity, MBTUH, at outdoor temperature A
- C - Heat pump compressor run time in 5° increment, percent
- D - Input for heat pump at temperature A, KW
- E - Seasonal heating hours in 5°F increment about A ($A \pm 2.5^\circ\text{F}$)
- F - Seasonal heat pump input in 5° increment about A
- G - Resistance heat required in 5°F increment about A
- H - Seasonal resistance heat needed in 5°F increment about A
- QT - Seasonal heating/cooling load in 5°F increment about A

Flow Chart for Heating and Cooling Performance Computer Programs

```

C  HEATING & COOLING SYSTEM PERFORMANCE
    DIMENSION CITY(28,30),CITNAM(28,5),SYSTEM(5,4,30),SYSNAM(5,5)
    INTEGER CODE,SYSNO
    REAL LOAD,A,B,C,D,E,F,G,H,Q,  QT,IDT,ILOAD
    READ(5,100) NOCIT,NOSYS
    DO 5 CODE=1,NOCIT
      READ(5,119) (CITNAM(CODE,K),K=1,5)
5    CONTINUE
      DO 6 I=1,NOSYS
        READ(5,119) (SYSNAM(I,J),J=1,5)
6    CONTINUE
      DO 10 I=1,NOCIT
        READ(5,100) (CITY(I,J),J=1,30)
10   CONTINUE
      DO 20 I=1,NOSYS
        DO 90 J=1,4
          READ(5,100) (SYSTEM(I,J,K),K=1,28)
90   CONTINUE
20   CONTINUE
      READ(5,100) IDT,CIDT,LOAD,CLOAD,TT,ILOAD
      WRITE(6,200)
200  FORMAT('0',20X,'UNIVERSITY OF ALA. ',/,',',',15X,'HEATING AND AIRCO
&NDITIONING PERFORMANCE ANALYSIS',/,',',',9X,'SCHAFTZLE & LECROY')
      DO 210 CODE=1,NOCIT
        DO 220 SYSNO=1,NOSYS
          WRITE(6,300) CODE,(CITNAM(CODE,L),L=1,5),(SYSNAM(SYSNO,M),M=1,5)
300  FORMAT('1',5X,'THIS PROGRAM WILL ANALYZE PERFORMANCE AT LOCATION',
&I3,',',',5A4,'FOR ',5A4,'HEATING/COOLING SYSTEM')
          WRITE(6,301) TT,ILOAD
301  FORMAT(' ',,'CALCULATIONS ARE FOR INDOOR TEMPERATURE MAINTAINED AT
&',F4.0,' F AND INTERNAL LOAD OF ',F6.0,' BTUH')
100  FORMAT ( )
      COUT=CITY(CODE,30)
      ODT=CITY(CODE,29)
      WRITE (6,700) ODT,LOAD,IDT
700  FORMAT('0',,'HEATING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF',
&F4.0,' F AND LOAD OF ', F8.0,' BTUH AT DESIGN INDOOR TEMP. OF',F4.
&0,' F')
      WRITE(6,1100) COUT,CLOAD,CIDT
1100 FORMAT(' ',,'COOLING CALCULATIONS ARE FOR OUTDOOR DESIGN TEMP. OF'
&F4.0,' F AND LOAD OF',F8.0,'BTUH AT DESIGN INDOOR TEMP. OF',F4.0,
&' F.')
      WRITE(6,800)
800  FORMAT('0',,'A= OUTDOOR TEMPERATURE,FAHRENHEIT',/,',',',B= SEASONAL
& HEATING HOURS IN 5 DEGREE INCREMENT ',/,',',',C= SEASONAL HEATING/
&COOLING LOAD IN 5 DEGREE INCREMENT , BTU',

```

/' ' 'D= HEAT PUMP CAPACITY

& AT OUTDOOR TEMPERATURE A, MBTUH',
 & /' ' 'E= HEAT PUMP SYSTEM INPUT, KW, IN
 & CLUDING AIR HANDLER REQUIREMENT, KW',
 & /' ' 'F= HEAT PUMP COMPRESSOR RUN TIME I
 & NPERCENT AT TEMPERATURE A'
 & /' ' 'G= SEASONAL HEAT PUMP INPUT, KWH IN 5 DEGREE TEM
 & PERATURE INCREMENT',/' ' 'H= RESI
 & STANCE HEAT REQUIRED IN 5 DEGREE INCREMENT, KW',/' ' 'I= SEASONAL
 & RESISTANCE HEAT NEEDED IN 5 DEGREE INCREMENT',/' ' 'J= COEFFICIEN
 & T OF PERFORMANCE, DIMENSIONLESS',/' ' 'K= SYSTEM STATUS',/' '0',40
 & X, 'STANDARD CALCULATION',/' '0',9X, 'A B C D
 & E F G H I J K')

SRT=0.

SH=0.

SDG=0.

SF=0.

SQT=0.

SCF=0.

SCQT=0.

A=-18.

DO 50 M=1,28

115 E= CITY(CODE,M)

IF(E.LE.0.) GO TO 49

LEST=0

IF(A-TT) 111,112,113

111 B=SYSTEM(SYSNO,3,M)

D=SYSTEM(SYSNO,4,M)

LEST=LEST+1

IF(LEST.GT.2) GO TO 112

Q=(LOAD/(IDT-ODT))*(TT -A)-ILOAD

IF(Q.LT.0.) GO TO 113

IF(Q.EQ.0.) GO TO 112

C=(Q/B)/10.

IF (C.GE.100.) C=100.

QT=E*Q

G=(Q-(B*10.*C))/3412.

IF((Q-(B*1000.)).LE.0.) G=0.

H=E*G

F=E*D*C/100.

SF=Sf+F

SQT=SQT+QT

SH=SH+H

COP=(B+3.412*G)/((D+G)*3.412)

WRITE(6,900)A,E,QT,B,D,C,F,G,H,COP

900 FORMAT(' ',2X,2F9.0,F10.0 ,F9.1,F9.2,F9.1,F9.0,F9.1,F9.1,F9.2,3X

&,'HEATING')

GO TO 49

```

112 B=0.
    COP=0.
    C=0.
    D=0.
    QT=0.
    F=0.
    G=0.
    H=0.
    WRITE(6,910) A,E,QT,B,D,C,F,G,H,COP
910 FORMAT(' ',2X,2F9.0,F10.0 ,F9.1,F9.2,F9.1,F9.0,F9.1,F9.1,F9.2,3X
&,'OFF')
    GO TO 49
113 B=SYSTEM(SYSNO,1,M)
    LEST=LEST+1
    D=SYSTEM(SYSNO,2,M)
    IF(LEST.GT.2) GO TO 112
    Q=(CLOAD/(CDDT-CIDT))*(A-TT) +ILOAD
    IF(Q.LT.0.) GO TO 111
    IF (Q.EQ.0.) GO TO 112
    C=(Q/B)*.1
    IF (C.GE.100.) C=100.
    IF (Q.GE.(B*1000.)) Q=B*1000.
    CQT=E*Q
    CF=E*D*C*.01
    G=0.
    H=0.
    SCF=SCF+CF
    SCQT=SCQT+CQT
    COP=B/(D*3.412)
    WRITE(6,920) A,E,CQT,B,D,C,CF,G,H,COP
920 FORMAT(' ',2X,2F9.0,F10.0 ,F9.1,F9.2,F9.1,F9.0,F9.1,F9.1,F9.2,3X
&,'COOLING')
49 A=A+5.
50 CONTINUE
    COPH=(SQT/(SH+SF))/3412.
    CCOP=SCQT/(SCF*3412.)
    EGAS=SQT/(62000.)
    EOIL=SQT/(145000.*.62)
    WRITE(6,1000) SQT,SF,SH, SCQT,SCF,CCOP,COPH
1000 FORMAT('0','ANNUAL HEATING LOAD, BTU = ',F10.0,/,', ', 'ANNUAL HEATI
&NG HEAT PUMP INPUT, KWH = ',F10.0,/,', ', 'ANNUAL RESISTANCE HEAT RE
&QUIRED, KWH = ',F10.0,/,', ', 'ANNUAL COOLING LOAD, BTU = ',F10.0,/,
&' ', 'ANNUAL COOLING HEAT PUMP INPUT, KWH = ',F10.0,/,', ', 'ANNUAL C
&OOLING COP = ',F6.2,/,', ', 'ANNUAL HEATING COP = ',F6.2)
    WRITE(6,211) EGAS,EOIL
211 FORMAT(' ', 'GAS HEAT AT 62% EFFICIENCY WOULD REQUIRE',F10.0,' THER
&MS. ',/,', ', 'FUEL OIL HEAT AT 62% EFFICIENCY WOULD REQUIRE ',F6.0,
&' GALLONS')
119 FORMAT(5A4)
220 CONTINUE
210 CONTINUE

```

XII. APPENDIX II. COMPUTER PROGRAM FOR DETERMINING
PRESSURE LOSS IN PIPING SYSTEM DUE TO FRICTION

A computer program has been developed to analyze pipe friction losses in the piping system. The program uses the Darcy-Weisback equation as a basis. The variables utilized in the program are pipe diameters, pipe lengths, flow rates, and extraction flow rates. The pipe roughness for PVC pipe and viscosity for water are included as constants.

The co-owned program by Jepperson performs a friction factor analysis by approximating the Moody diagram for pipe flow.³⁷ The program determines if the flow is turbulent or laminar, determines the friction factor, and then calculates pressure losses. A flow chart, program variables, and listing of the program follow.

PROGRAM VARIABLES

D - Pipe diameter

FL - Pipe length

R & Q - Flow rate

EXT - Extraction rates at points along pipeline

HLTOT - Total head loss over length of pipeline

A - Pipe cross sectional area

V - Fluid velocity

RE - Reynolds Number

VIS - Absolute kinematic viscosity

F - Friction factor

E - Pipe roughness

ED - Parameter (ϵ/D) in Moody diagram, relative roughness

PAR - Approximation of Moody; allows check into laminar or turbulent flow

F - Friction factor symbol

FF - When set equal to zero gives friction factor equation for transition between hydraulically smooth and wholly rough

DF - Derivative of FF with respect to the friction factor (F)

DIF - Iterative factor used in approximating F

NCT - Iteration counter

```

DIMENSION D(5),FL(2),R(3),EXT(3)
READ(5,*)VIS,E,G
READ(5,*)(D(I),I=1,5),(FL(J),J=1,2)
READ(5,*)(R(K),K=1,3),(EXT(L),L=1,3)
DO 20 I=1,5
DO 20 J=1,2
DO 20 K=1,2
Q=R(K)
HLTOT=0.
DO 20 L=1,10
C D-PIPE DIAMETER,Q-FLOW RATE,FL-LENGTH OF PIPE,EXT-EXTRACTION RATE
C VIS-KINEMATIC VISCOSITY OF FLUID X 10**5,
C E-ABSOLUTE ROUGHNESS OF PIPE, G-ACCELERATION OF GRAVITY
A=.7839816*D(I)*L(I)
V=Q/A
RE=V*D(I)/VIS
IF (RE.GT.2100) GO TO 3
F=64./RE
3 EVIS=E/VIS
ELOG=9.35*ALOG10(2.71828183)
ED=E/D(I)
F=1./(1.14-2.*ALOG10(ED))**2
PAR=V*SQR(T(F/8.))*EVIS
IF(PAR.GT.100.)GO TO 1
NCT=0
2 FS=SQR(T(F))
FZ=.5/(F*FS)
ARG=ED+9.35/(RE*FS)
FF=1./FS-1.14+2.*ALOG10(ARG)
DF=FZ+ELOG*FZ/(ARG*RE)
DIF=FF/DF
IF (ABS(DIF).GT..00001.AND.NCT.LT.15) GO TO 2
F=F+DF
NCT=NCT+1
1 HL=F*FL(J)*V*V/(2.*G*D(I))
WRITE(6,101)Q,D,FL(J),F,HL
101 FORMAT('0',,Q=' ,F10.4,5X,'D=' ,F10.4,5X,'L=' ,F10.2,5X,'F=' ,
&F10.5,5X,'HEADLOSS=' ,F10.4)
HLTOT=HLTOT+HL
IF(L.LT.10) GO TO 20
WRITE(6,102) HLTOT
102 FORMAT('0',,10X,E10.6)
Q=Q-EXT(K)
20 CONTINUE
99 STOP
END

```

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