27
2-14-80
DNTIS

MASTER

ANL/CNSV-TM-25

ENERGY • EKISTICS • ENVIRONMENT • ECONOMY
4E

Heat-Pump-Centered Integrated Community Energy Systems

System Development

University of Alabama Final Report

ICES
INTEGRATED COMMUNITY ENERGY SYSTEMS

ENERGY AND ENVIRONMENTAL SYSTEMS DIVISION

ARGONNE NATIONAL LABORATORY

OPERATED FOR THE U. S. DEPARTMENT OF ENERGY
UNDER CONTRACT W-31-109-ENG-38

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

The facilities of Argonne National Laboratory are owned by the United States Government. Under the terms of a contract (W-31-109-Eng-38) among the U. S. Department of Energy, Argonne Universities Association and The University of Chicago, the University employs the staff and operates the Laboratory in accordance with policies and programs formulated, approved and reviewed by the Association.

MEMBERS OF ARGONNE UNIVERSITIES ASSOCIATION

The University of Arizona
Carnegie-Mellon University
Case Western Reserve University
The University of Chicago
University of Cincinnati
Illinois Institute of Technology
University of Illinois
Indiana University
The University of Iowa
Iowa State University
The University of Kansas
Kansas State University
Loyola University of Chicago
Marquette University
The University of Michigan
Michigan State University
University of Minnesota
University of Missouri
Northwestern University
University of Notre Dame
The Ohio State University
Ohio University
The Pennsylvania State University
Purdue University
Saint Louis University
Southern Illinois University
The University of Texas at Austin
Washington University
Wayne State University
The University of Wisconsin-Madison

NOTICE

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States nor any agency thereof, nor any of their employees, makes any warranty, expressed or implied, or assumes any legal liability or responsibility for any third party's use or the results of such use of any information, apparatus, product or process disclosed in this report, or represents that its use by such third party would not infringe privately owned rights. Mention of commercial products, their manufacturers, or their suppliers in this publication does not imply or connote approval or disapproval of the product by Argonne National Laboratory or the United States Government.

This informal report presents preliminary results of ongoing work or work that is more limited in scope and depth than that described in formal reports issued by the Energy and Environmental Systems Division.

Printed in the United States of America. Available from National Technical Information Service, U. S. Department of Commerce, 5285 Port Royal Road, Springfield, Virginia 22161

ANL/CNSV-TM-25

DISCLAIMER

This book was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

ARGONNE NATIONAL LABORATORY
9700 South Cass Avenue
Argonne, Illinois 60439

HEAT-PUMP-CENTERED INTEGRATED COMMUNITY ENERGY SYSTEMS

System Development
University of Alabama Final Report*

by

Walter J. Schaetzle
C. Everett Brett
and
Marvin S. Seppanen

Bureau of Engineering Research
and
Natural Resources Center
THE UNIVERSITY OF ALABAMA
University, Alabama 35486

December 1979

Prepared for

Energy and Environmental Systems Division
Argonne National Laboratory
Under Argonne Contract No. 31-109-38-4550

Argonne Project Manager: James M. Calm, PE

Work sponsored by

U.S. DEPARTMENT OF ENERGY
Assistant Secretary for Conservation and Solar Energy
Office of Buildings and Community Systems
Community Systems Division
Community Systems Technology Branch

*This report supersedes ANL/ICES-TM-30: *Heat-Pump-Centered Integrated Community Energy Systems, System Development, University of Alabama Interim Report.*

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

PREFACE

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

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

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

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

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

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

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

The report which follows presents the findings in the development and analysis of one concept under investigation. This report was prepared by the University of Alabama with technical support from Alabama Power Company and American Air Filter Corporation.

TABLE OF CONTENTS

TABLE OF CONTENTS (CON'T)

LIST OF FIGURES

LIST OF FIGURES (CON'T)

LIST OF FIGURES (CON'T)

LIST OF TABLES

LIST OF TABLES (CON'T)

LIST OF TABLES (CON'T)

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

ABSTRACT

The heat-pump-centered integrated community energy system unifies groups of controlled environments (homes, businesses, etc.) into energy modules. The energy rejected from air conditioning is stored and retrieved for heating. The energy sink created by heating is utilized for air conditioning. The storage system is provided by nature in underground porous formations filled with water aquifers. The energy is transported by a two-pipe system, one for warm water and one for cool water, between the aquifers and the controlled environments. Each energy module contains the controlled environments, an aquifer, wells for access to the aquifers, the two pipe water distribution system and water source heat pumps. The heat pumps upgrade the energy in the distribution system for use in the controlled environments. Since climate varies with location, some locations will require additional energy. This can be added as low quality waste heat, solar energy or other source of low cost energy. In turn, some locations will have excess energy. This can be dissipated with cooling towers or ponds during colder weather periods.

Heat pump performance 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 70^{O}F with an air-source heat pump using outside ambient temperature of -10^{O}F for a thermal energy source. If 80^{O}F 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 two wells is reversed, and relatively warm water is recovered and pumped to the heat pumps. The net usage of water is zero.

By using local aquifers, the distance water is transported is minimized. This allows smaller pipelines and requires less pumping power than the normal community system. The system will not be effective for an individual home, since the volume of injected water is too small and large thermal losses, up to 100%, will occur. The system uses the module concept for a group of controlled environments. Each module has a water distribution system, a pair of wells and an aquifer thermal energy storage system. By connecting the water distribution systems and some controls, modules can be combined. The combinations result in more reliability and lower maintenance costs.

Economically, the system shows improvement on both energy usage and capital costs. The system saves over 60% of the energy required for resistance heating; saves over 30% of the energy required for most air-source heat pumps (primary function of climatic location); and saves over 60% of the energy required for gas, coal, or oil heating, when comparing 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 listed 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 four 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.

Heat pump performance is being upgraded by utilizing improved compressors, improved blowers, and larger heat transfer areas. Major changes in these areas will be incorporated into water-source heat pumps over the next few years. Performance is expected to improve between 20% and 40%, making the proposed system 15% to 35% more efficient than the performance analysis contained in this report. These changes have been incorporated into some water source heat pumps being sold.

The proposed system has been analyzed as demonstration projects for a downtown portion of Louisville, Kentucky, and a section of Fort Rucker, Alabama. The downtown Louisville demonstration project is tied directly to major buildings while the Fort Rucker demonstration project is tied to a dispersed subdivision of homes. The Louisville project shows a payback of approximately three years, while Fort Rucker is approximately 30 years. The primary difference is that at Fort Rucker new heat pumps are charged to the system. In Louisville, either new construction requiring heating and cooling systems or existing chillers are utilized.

I. SYSTEM DESCRIPTION

The heat-pump-centered integrated community energy system uses a two pipe water distribution system to connect the controlled environments with underground energy storage in aquifers. The community is broken into energy modules. The energy modules can be connected by connecting the respective water distribution systems. 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 1-1 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 1-2 and 1-3. Figure 1-2 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 electric distribution or telephone service.

A number of alternative systems is analyzed. The primary difference between the systems is the water temperatures. In the basic system, 60°F cool water and 80°F warm water are circulated. In Alternative I some direct heat transfer to the environment from the water is utilized in conjunction with the heat pump. In Alternative II 40°F cold water and 60°F warm water are circulated. The cooling is by direct heat transfer from the cool water. The heating uses 60°F water which is cooled to 40°F. This system is similar to the ice producing heat pump system. Alternative III

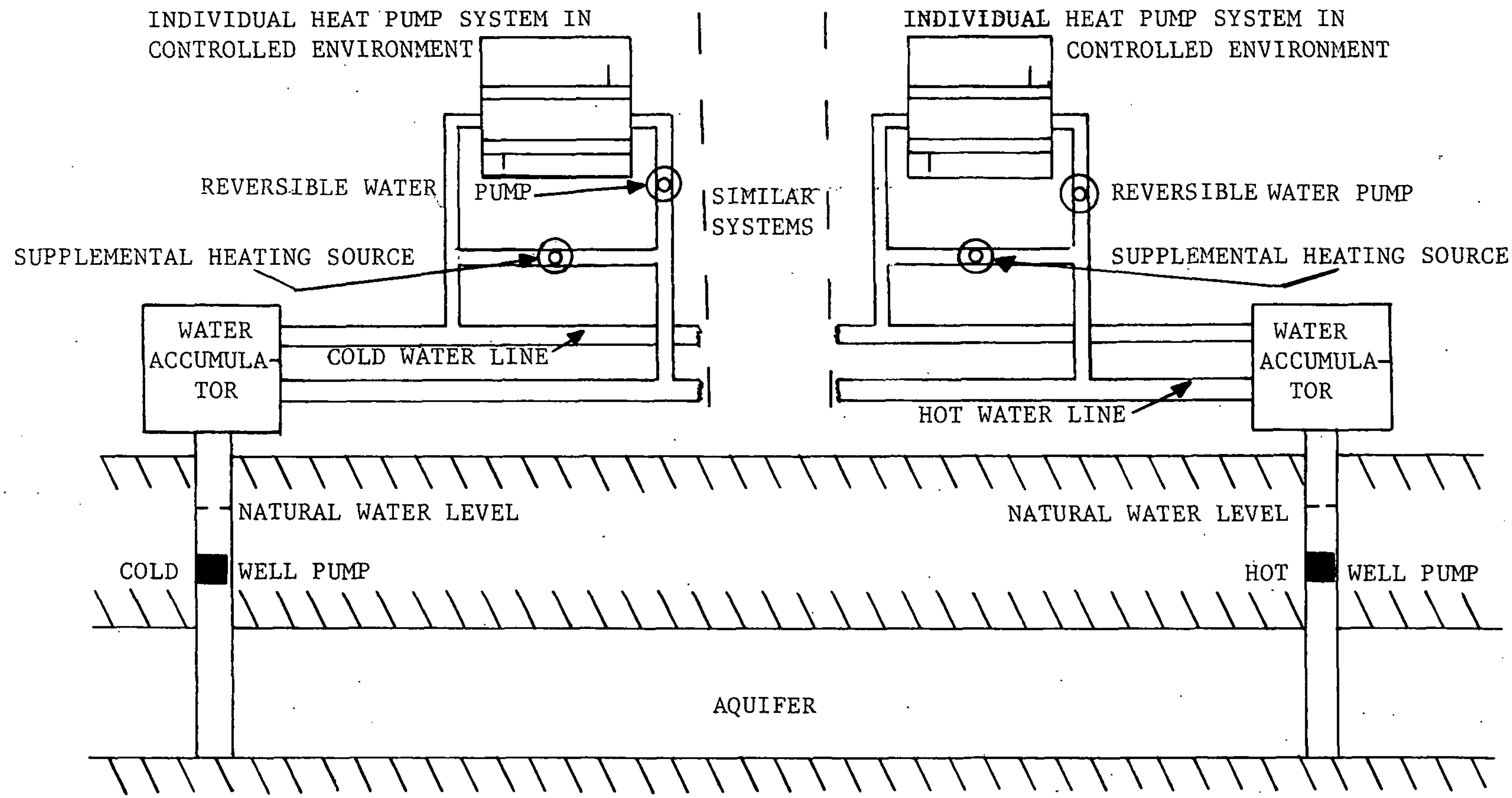


FIGURE 1-1. SCHEMATIC OF THE TECHNICAL COMPONENTS FOR THE PROPOSED HEAT PUMP CENTERED INTEGRATED COMMUNITY ENERGY SYSTEM

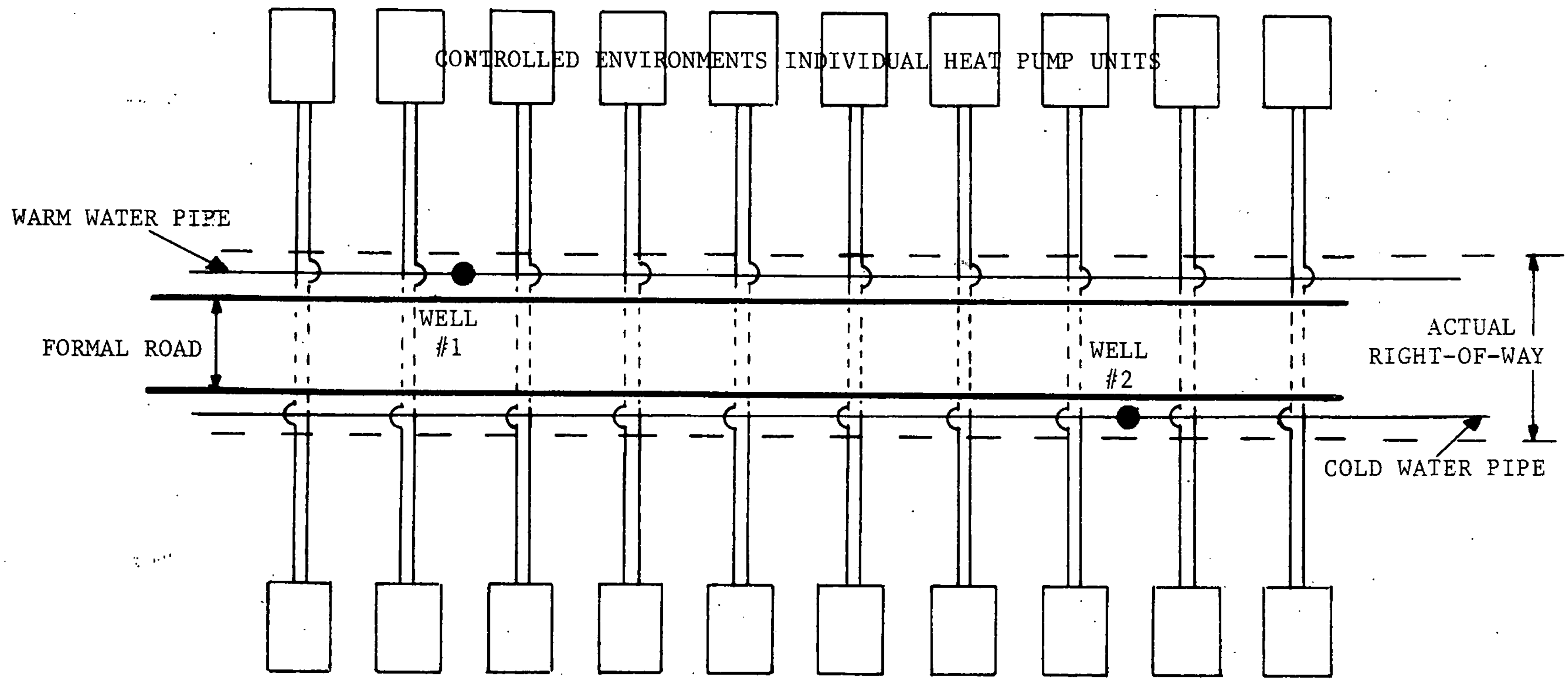


FIGURE 1-2. STANDARD HEAT PUMP SYSTEM MODULE IN HEAT PUMP INTEGRATED COMMUNITY ENERGY SYSTEM

2.12.78

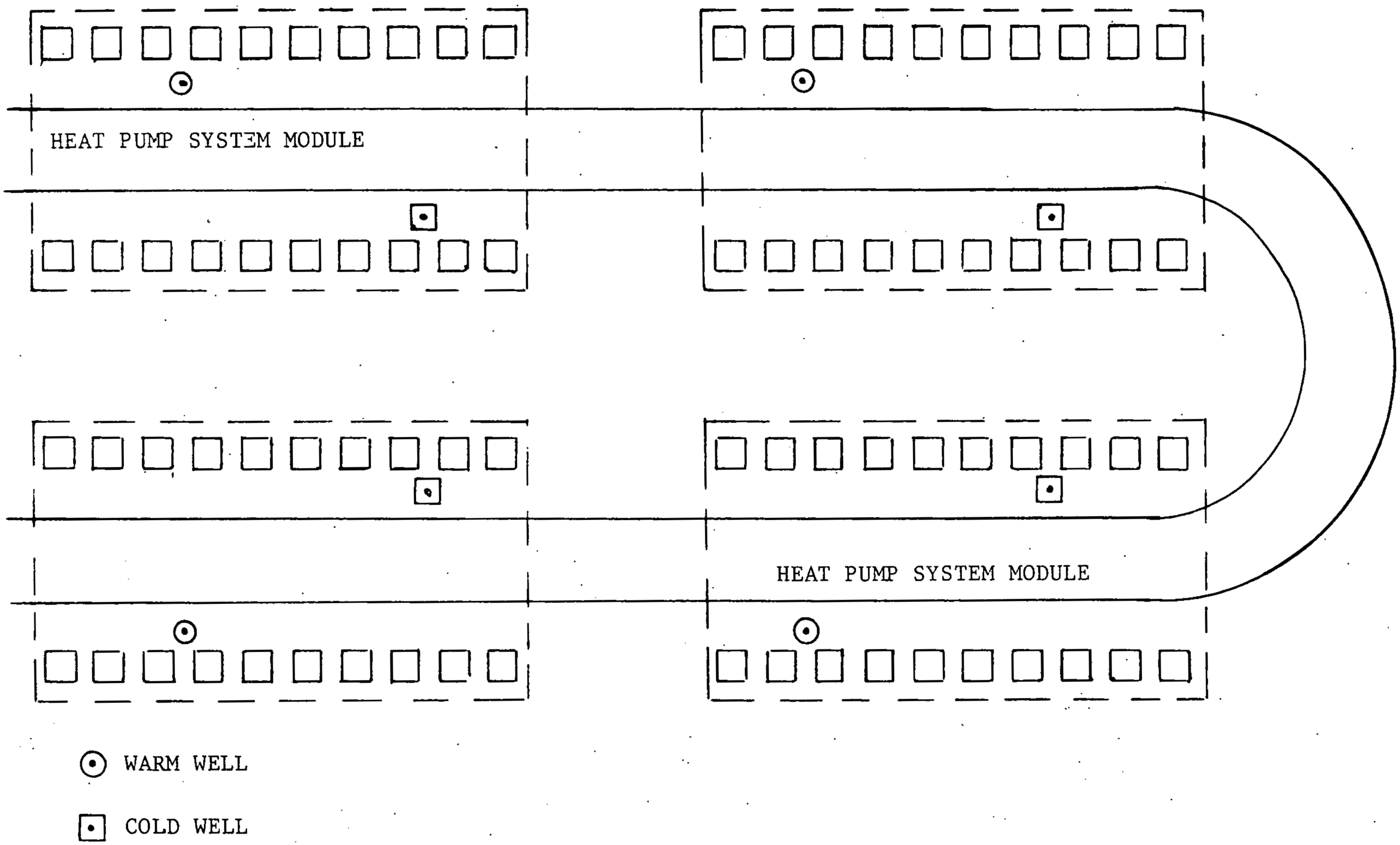


FIGURE 1-3. GROUPS OF HEAT PUMP SYSTEM MODULES IN HEAT PUMP INTEGRATED COMMUNITY ENERGY SYSTEM

2·12·78

uses 80^oF cool water and 100^oF warm water. The 100^oF water is used for direct heat transfer and the 80^oF water used for cooling is heated to 100^oF. It is the opposite concept to Alternative III or the ice producing heat pump.

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

B. The thermal energy storage module, or the aquifer: Aquifers are water-bearing rock formations 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 shallow aquifers adequate for utilization in this system. Many urban areas are concentrated in valleys and near rivers where extensive aquifers exist.

As a result, over 75% of the population is located near adequate aquifers. 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 four 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 captive air 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 water distribution systems provide sufficient water locally for fire protection. As a result, the peak loads for domestic water are reduced. The benefit to the controlled environment is higher heating and cooling system performance. The system performance will be over three times that of resistance heating, will approach two times the performance of air-source heat pumps in some locations, and will increase by more than 60% 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, similar 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. In addition, the owner pays for the electricity charges which will encourage conservation. Charging for water usage is another possibility.

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. Various possibilities for charging such as demand, peak loads, and usage are envisioned.

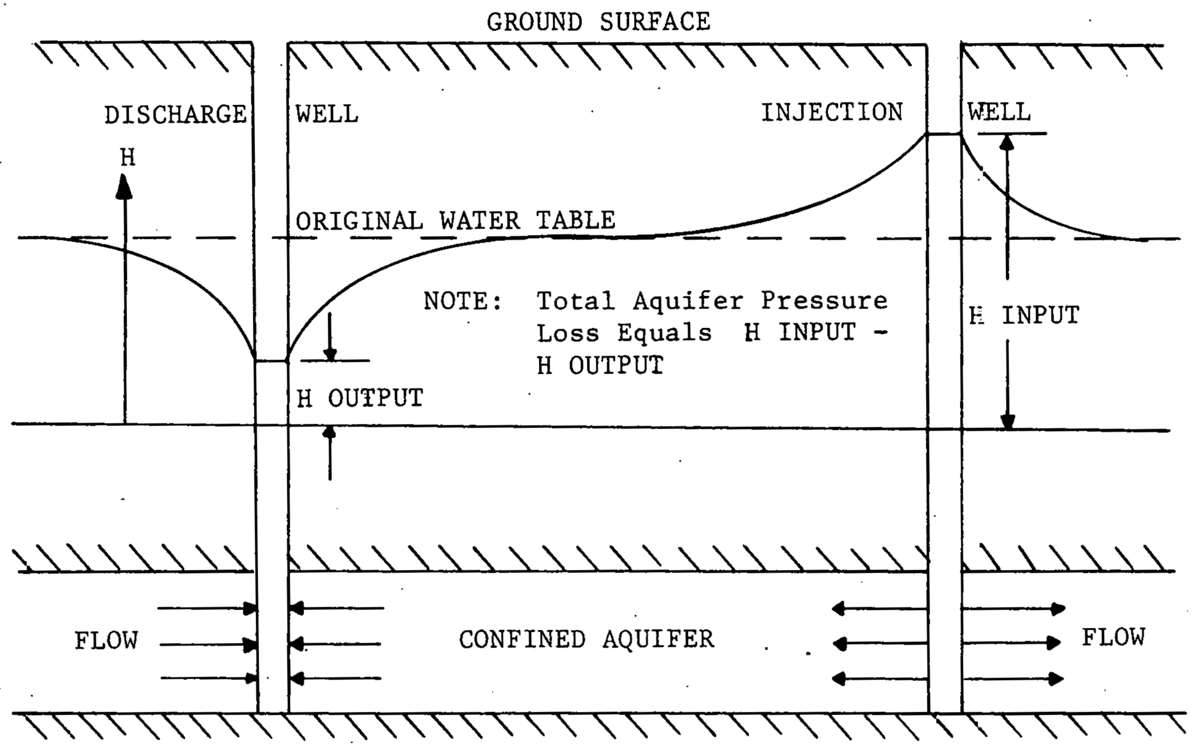


FIGURE 1-4. STANDARD WELL PAIR FOR 1 MODULE

20.11.78

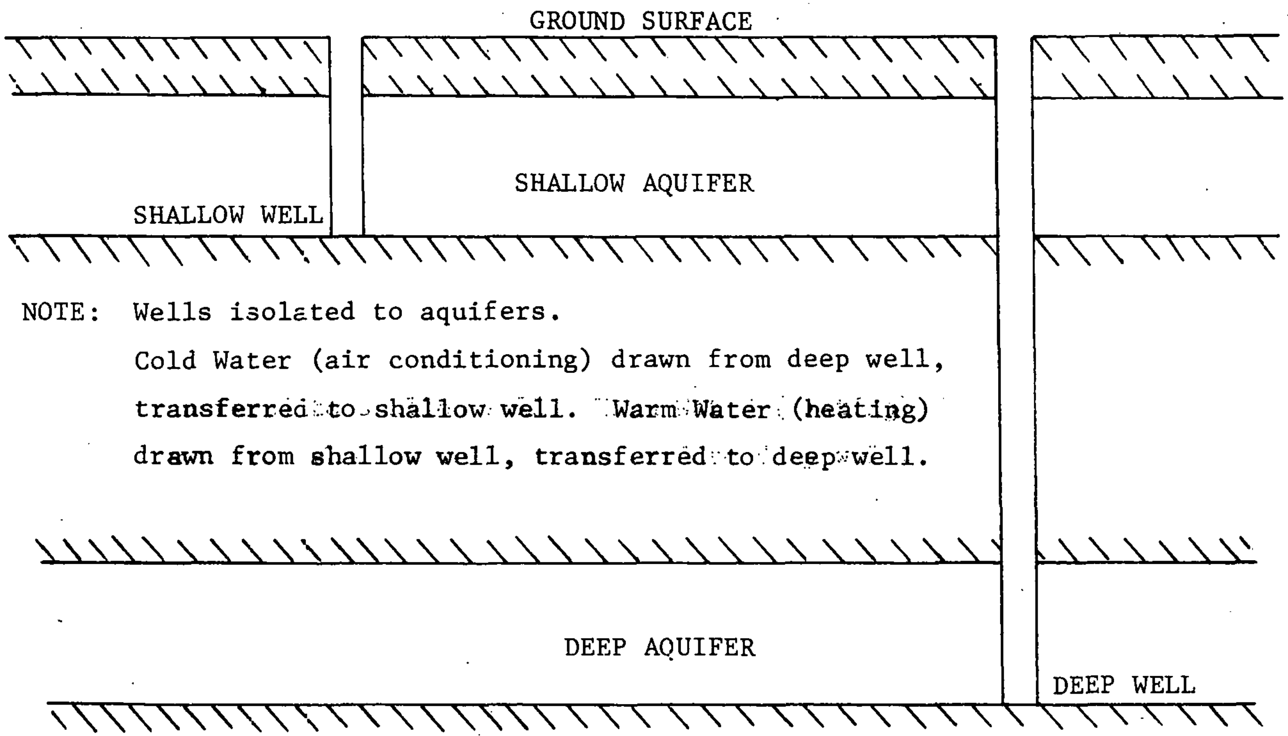


FIGURE 1-5. 2 AQUIFER SYSTEM FOR 1 MODULE SYSTEM

20.11.78

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 flow 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 six 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 1-4). During the injection and recovery process, a water front and a termperature front (at lower velocity) move between the wells. In the cooling cycle, warmer water is injected in one well, and cool water is withdrawn from the other and piped to the heat pump. For heating, the process is reversed and warm 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 six months. Depending on withdrawal and injection rates, a spacing of a few hundred feet 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 relative cold water (Figure 1-5). 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

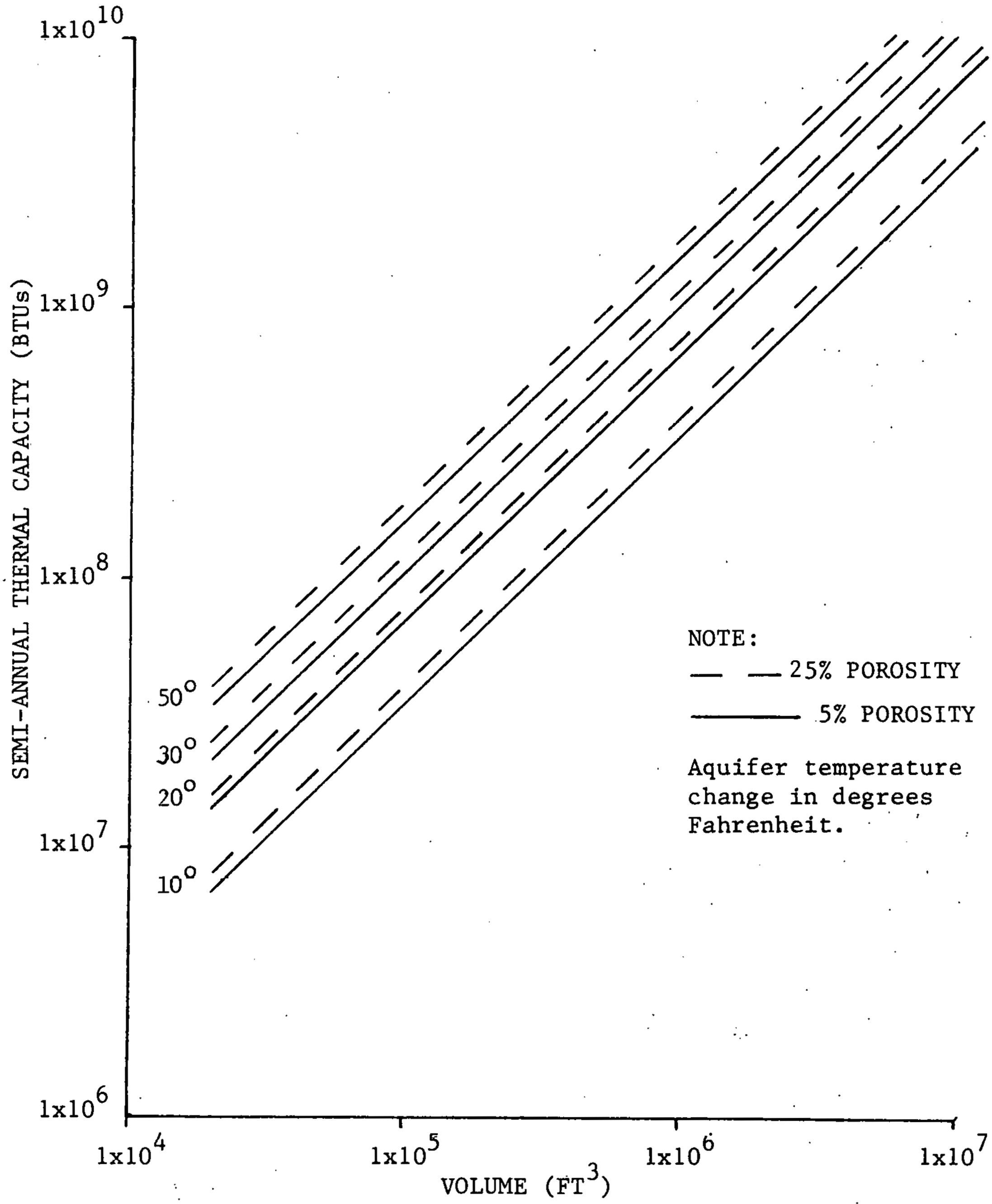


FIGURE 1-6. AQUIFER THERMAL ENERGY STORAGE CAPACITY

20-11-78

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 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 rocks have a specific gravity of approximately 2.6 and a specific heat of about 0.2 Btu/^{o}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, a rock volume of 100 ft. by 200 ft. by 30 ft. has a storage capability of 2.0 x 10^8 Btu with a 10^{o}F temperature change. General values for the storage capacity of aquifers as a function of size, porosity, and temperature variations are given in Figure 1-6.

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 four inches to 15 inches and depths from 20 to a few hundred feet. 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 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 1-1. Installed side-by-side with spacing between the two, 1 pipe will carry relatively warm water and the other relatively cold water. Each well will have a pumping system (Figure 1-7) 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 1-8 and 1-9 give pressure losses for homes separated by 100 ft. and 200 ft., respectively at various flow rates.

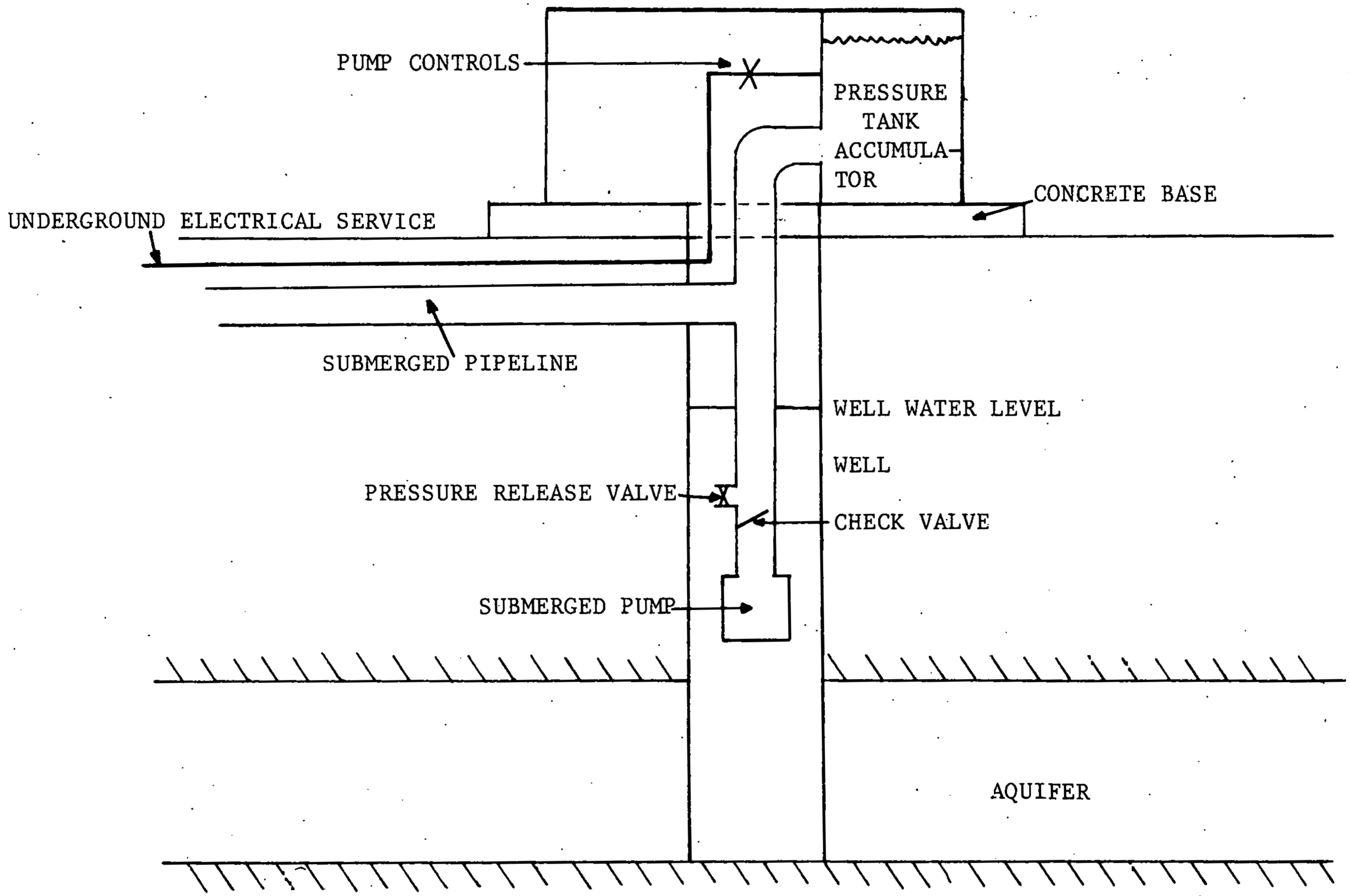


FIGURE 1-7. SCHEMATIC OF WELL PUMPING SYSTEM

2.12.78

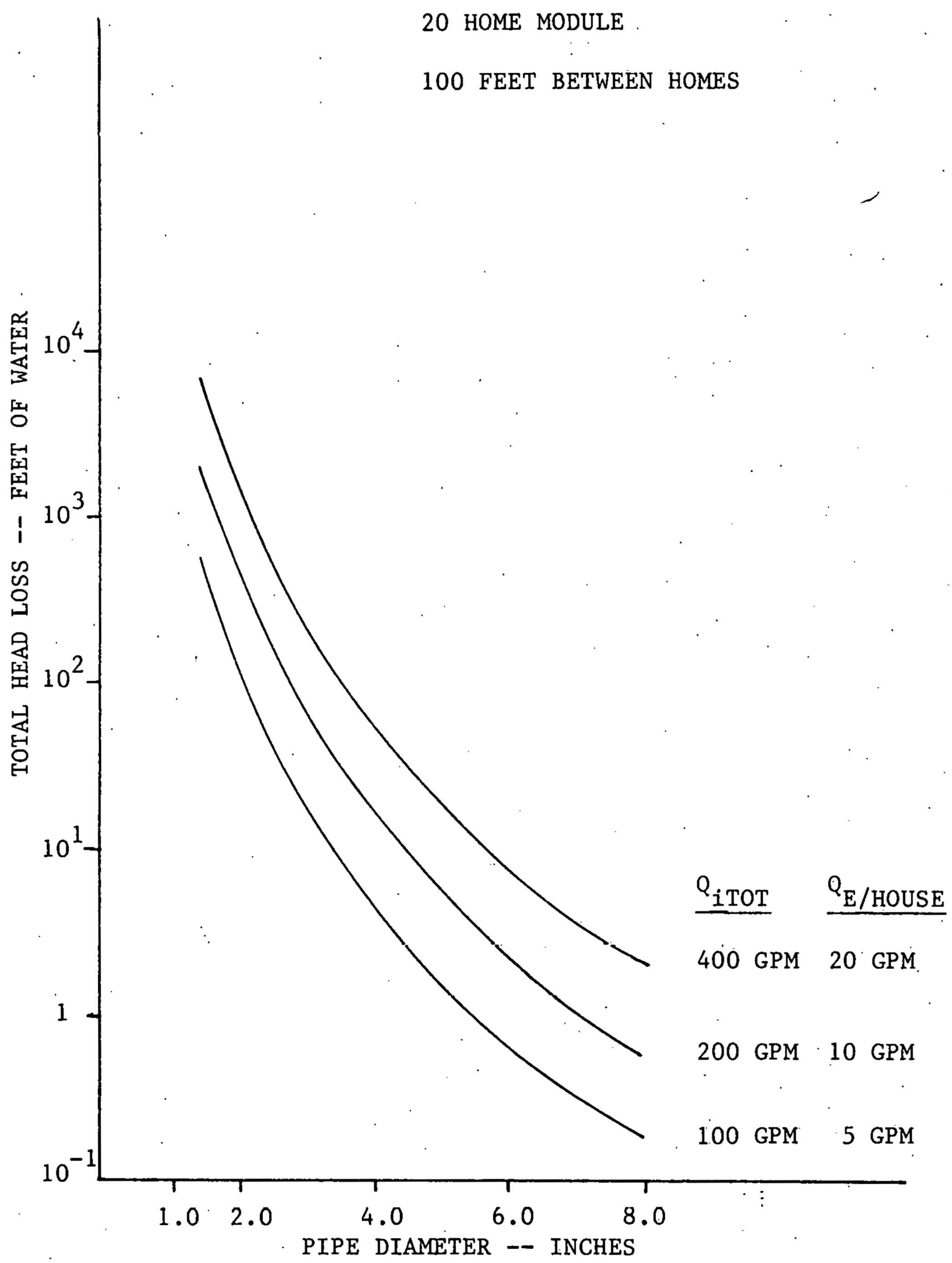


FIGURE 1-8. MODULE TOTAL DISTRIBUTION PIPE PRESSURE LOSS

20.11.78

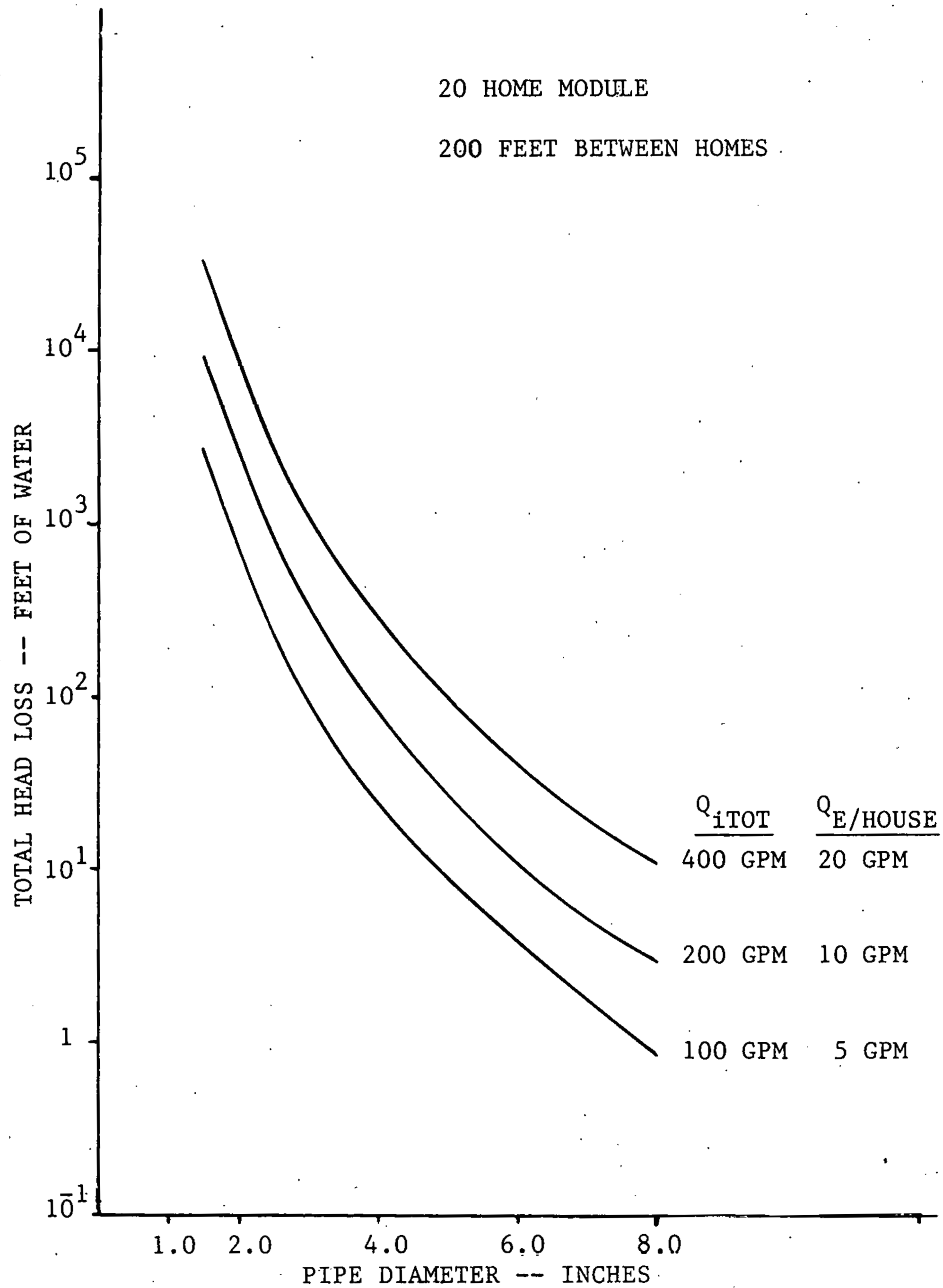


FIGURE 1-9. MODULE TOTAL DISTRIBUTION PIPE PRESSURE LOSS

20.11.78

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 one foot of water if two homes at opposite ends of a module heat and cool simultaneously (flow rate of 10 GPM). Pressure loss in the pipe lines to buildings from the primary pipeline is minimized by using large pipelines one inch or more in diameter. Since installation labor is a major cost, the utilization of larger lines does not linearly increase 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 one 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 required 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 (COP_c) for cooling where W is work input, Q_L is the thermal energy removed and Q_H is the thermal energy rejected or:

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

and the work is equal to:

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

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 or performance (COP_H) for heating or:

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

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

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:

$$\frac{Q_{Heating}}{Q_{Cooling}} = \frac{COP_c + 1}{COP_c} \times \frac{COP_H}{COP_H - 1}$$

The ratio of heating to cooling load is shown in Figure 1-10 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 buiding, 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:

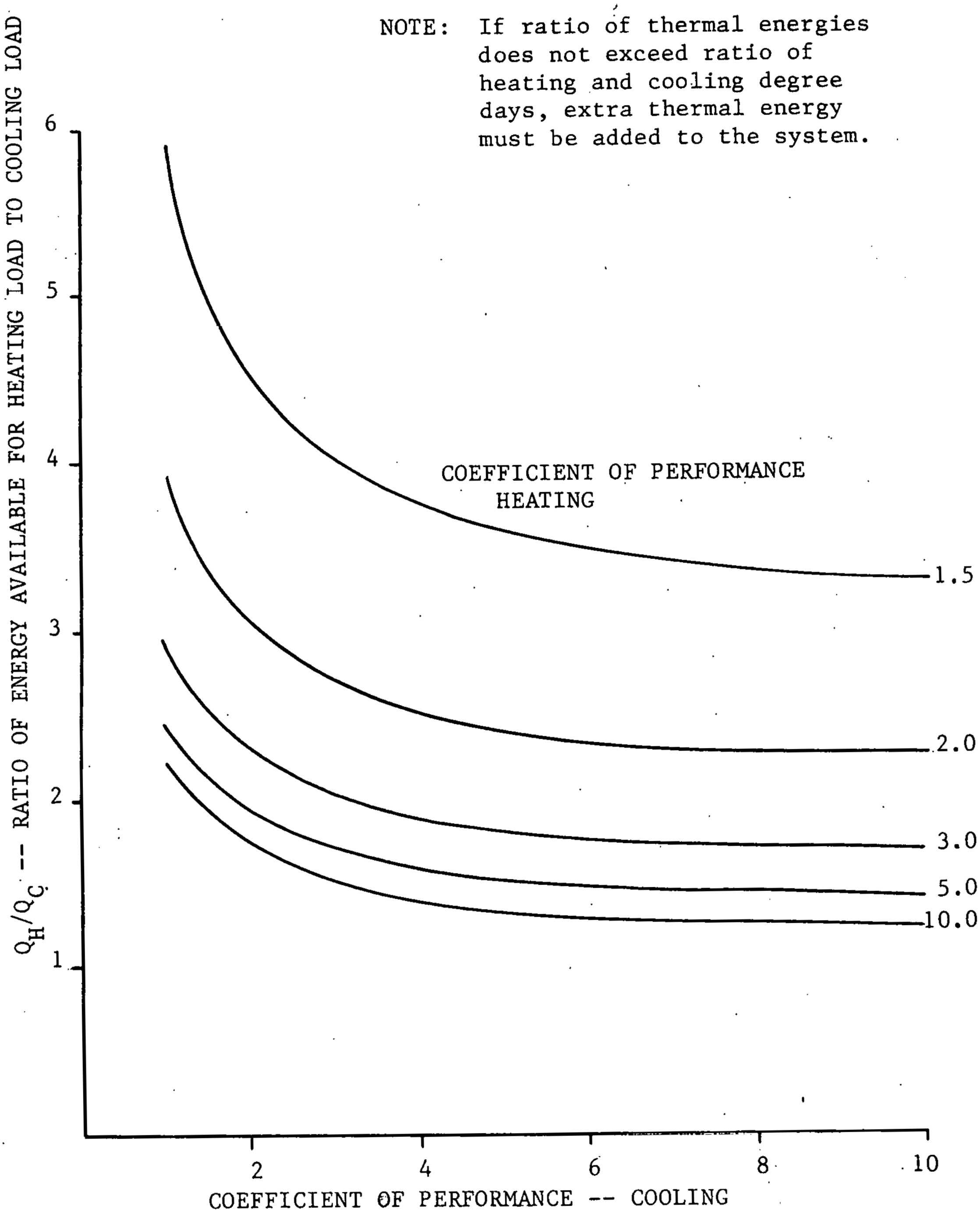


FIGURE 1-10. HEATING TO COOLING RATIO AVAILABILITY AS A FUNCTION OF HEAT PUMP PERFORMANCE

NK 4-12-78

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^{o}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 XI.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, multi-story buildings have no losses through common 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 three 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 based on water utilization, and 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 such as the power to operate equipment are still expected to encourage a conservation-type 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 approximately $1,000 per home in a typical subdivision and be on an assessment basis, such as water lines.

Service by the utility is expected to be minimal with less than one 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 eliminates 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 two 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. However, combining the local aquifers under roadways and parks should provide sufficient storage. 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 (Expected Economics, Section IV), the proposed system can be incorporated economically in essentially any new residential subdivision with spacing between homes of 500 ft. or less. The well system and distribution system can be incorporated in the capital price for the lots.

The system feasibility increases as a neighborhood becomes more dense. The limit is multistory buildings with minimum spacing. Here the length of the water distribution system is minimized, however pipeline construction per foot of length is increased. The Louisville demonstration analysis, Section IX, shows a 3 year return for construction costs of wells and water distribution.

Since most major buildings have a central heating and cooling system with water source chillers, retrofit is minor as major equipment does not require replacement. In some systems heat exchangers and other components must be modified or changed due to the higher pressures required in the heating cycle versus cooling cycle. Because of building inner cores and corresponding internal loads, the heating loads in large buildings are minimum. This helps the conversion of the chillers to alternate heat pumps. In turn this allows heat to be transferred to modules with homes and high heating loads. Geographically the system is economically feasible nationwide in areas which have aquifers, with the exception of southern Florida (Expected Economics, Section IV). In this area the best system is probably a ground water system with one 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.

For new systems in residential neighborhoods, downtown areas with major building and a mixture of the previous two systems, the capital costs are not capital intensive. The lower costs for installed water source heat pumps and lower capacity models (Expected Performance, Section III, and Expected Economics, Section IV) cancel part to total capital costs of the storage and distribution system. These systems are not capital intensive.

Retrofits for areas with larger buildings is feasible with a 3-year payback noted for the Louisville demonstration project (Section IX). Retrofits for residences is questionable when old heating and cooling systems must be replaced as in the Ft. Rucker demonstration project (Section X). It must be noted that these are on commercial rates. Changing to residential rates cuts payback by over 50%, however, the conversion is still too capital intensive to be economically feasible.

The expected economics of the proposed system versus current domestic heating and cooling alternatives presented in Section IV considers in detail the impact of fuel cost escalation. In general the economic performance of the proposed system improves versus the current alternatives with fuel cost escalation.

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 for retrofits without intensive capital.

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 2-1 is a map of the continental United States showing adequate porous aquifers for water systems[1]. Of major importance are the areas adjacent

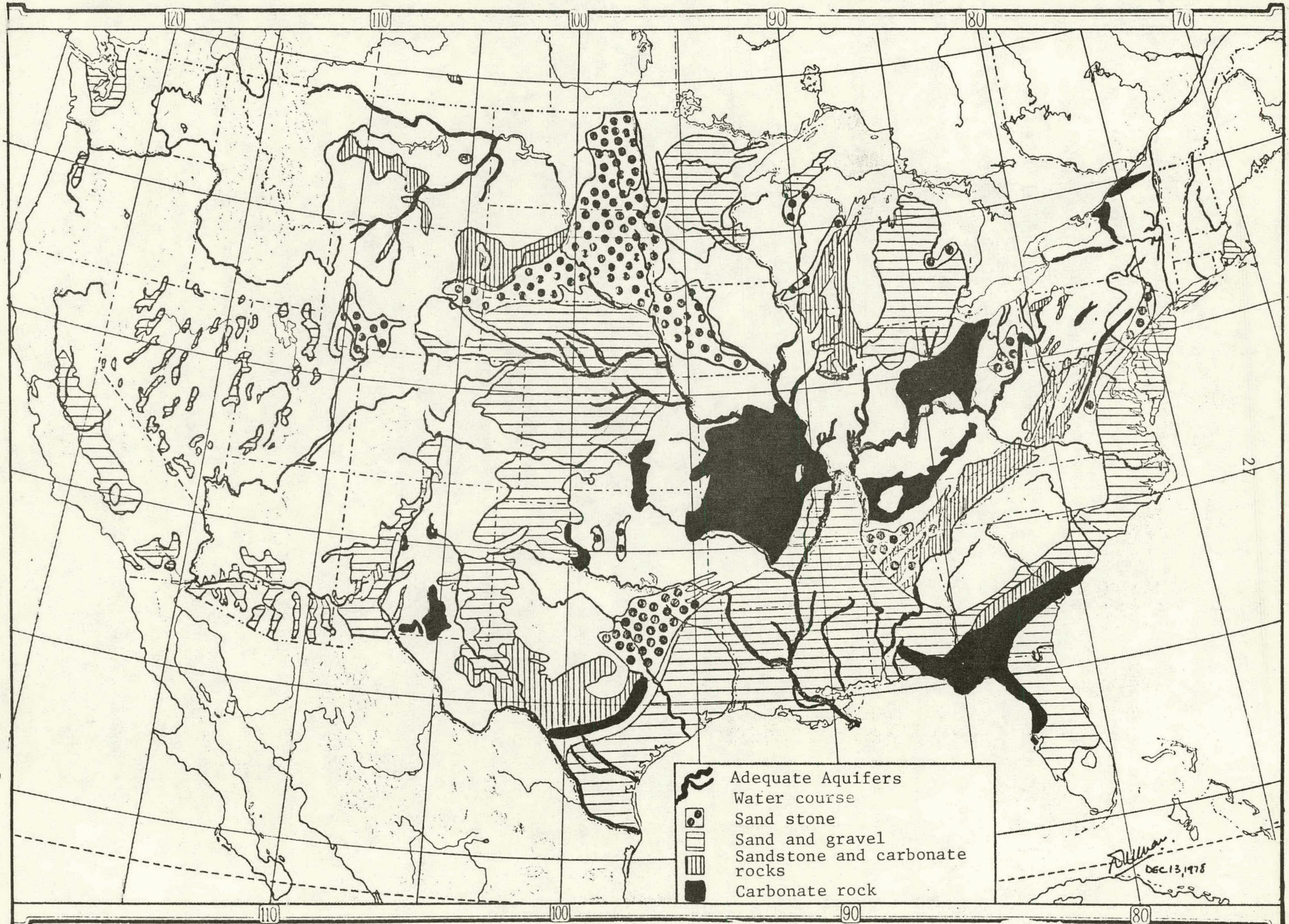


Figure 2-1. Adequate Aquifers for Proposed System in United States

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 aquifers shown, some fractured and porous metamorphic and igneous rock will serve as adequate aquifers. From the map over half of the continental United States land area has adequate aquifers available for operation of this system. This area is estimated at over 60% of the land area. Since the population centers are located in these areas, especially along the rivers, lakes and seacoast, over 75% of the population has adequate aquifers. 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 areas[2]. From the map (Figure 2-1) it is estimated 75% of the metropolitan areas and 60% of the non-metropolitan areas can use the proposed system. Combining numbers, 71% of the United States households can utilize the system. It is estimated 75% of all United States businesses, which are located primarily in metropolitan areas, can utilize the system. Total applications as of 1975 are[2]:

71% of 72,5000 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 in population, 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 over 5,000,000 modules.

For air-conditioning, use of electricity can be decreased by over 50% by using a combination of the basic system and first three alternatives to replace present systems. For heating, the end 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, and nuclear energy[2]. For heating, the energy requirement also decreases an average of over 50% over systems presently utilized.

An estimate is made of the source energy which can be saved if the system is employed where technically feasible without limits on economics. A major fallacy in this analysis is that with the installation of this system air-conditioning will be used in many cases where normally only heating is utilized. Energy prices could help correct this factor.

The basic data used for this analysis is from Residential and Commercial Energy Use Patterns 1970-1990[3]. The residential and commercial end use energy values for 1970 are used for the sections of the country analyzed. The percentage of heating and cooling for average residential and commercial end use energy is determined on a weighted basis. The cooling and heating end use energy values are calculated based on these numbers. An annual compound growth rate for residential and commercial heating and cooling is determined on a weighted basis for the period 1970 to 1990. These values are assumed to be valid for 1970-2000. The growth values for 30 years are applied to the 1970 energy values to determine the year 2000 energy usage. The heating values for 1970 are divided by a 0.35 power plant efficiency to convert to source energy. For energy saving, the 50% average energy savings is multiplied times 73% utilization for annual energy savings in the year 2000. The tabulations of these numbers are given in Table 2-1.

Table 2-1. National Heating and Cooling Energy Use - 1970-2000*

	Fraction End Use Energy 1970		End Use Energy 1970			Source Energy Use 2000			Annual Energy Savings
	Heating[1]	Cooling[2]	Total[3]	Heating[4]	Cooling[5]	Heating[6]	Cooling[7]	Total	Heating & Cooling[8]
North Central	0.75	0.0093	5.0	3.75	0.133	6.11	1.36	7.47	2.73
Northeast	0.76	0.0068	4.3	3.27	0.084	5.33	0.85	6.18	2.25
South	0.58	0.0546	3.3	1.91	0.515	3.11	5.25	8.36	3.05
West	0.62	0.0279	2.1	1.30	0.167	2.12	1.70	3.82	1.39

Annual Compound Growth Rate[9]		Growth for 30 Years	Overall Annual Total 9.4
Cooling	8.07%	10.2	
Heating	1.65%	1.63	

*Residential and Commercial Energy Use Patterns, FEA, Nov. 1974, v 1

1. Weight percentage of residential and commercial end energy uses for heating
2. Weight percentage of residential and commercial end energy uses for cooling
3. Total end energy use in quadrillion Btu for region
4. End energy use for heating in quadrillion Btu for region
5. End energy use for cooling in quadrillion Btu for region
6. Heating energy in quadrillion Btu multiplied by heating growth for 30 years (Assumes Source Energy 1970)
7. Cooling energy in quadrillion Btu multiplied by cooling growth for 30 years divided by 0.35 for average power plant efficiency (Assumes electricity in 1970)
8. Use 50% savings on 73% of heating and cooling energy in quadrillion Btu
9. Weight percentage of residential and commercial heating and cooling of annual growth factors 1970-1990, assumed continuous to 2000

The total annual savings results in an annual energy savings of

9.4 quadrillion Btu

which equals

4.6 million barrels of petroleum per day

which equals

9.4 trillion $ft.^3$ of natural gas per day

which equals

376 million tons of coal per year.

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

III. EXPECTED PERFORMANCE

The expected performance of the proposed system is evaluated by application to a typical home in a residential neighborhood. 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 heating and cooling system coefficient of performance, and total energy utilized, namely the source energy supplies. The proposed system shows an advantage over standard present day heating and cooling systems not only in performance, but the flexibility of any available energy source. These include hydro, coal, nuclear, gas, oil, and solar.

The energy balance diagram of the proposed system and present heating and cooling systems are shown in Figure 3-1. The new development is the economical annual thermal energy storage provided by nature. Energy rejected during cooling periods is stored and used during heating periods. The sequence is shown in Part A of Figure 3-1. This shows the energy going into the power plant being converted to work for use in the heat pump. The heat pump draws and rejects heat to thermal energy storage in the aquifer for heating and cooling cycles respectively. The normal air to air heat pump draws and rejects heat from the environment. The energy in the storage system is much more favorable in heat pump utilization. For example, when a normal system is pumping heat from 0^{O}F at ambient conditions to 70^{O}F at the controlled environment conditions, the proposed system is pumping heat from 80^{O}F in storage to the 70^{O}F controlled environment. These conditions are much more favorable and require less work input. The furnace type heating as shown in Part C also has waste heat losses. The power plant is included in the schematic to relate performance for all systems on the basis of paid-for-energy -- fuel into the system. In addition, the annual energy storage

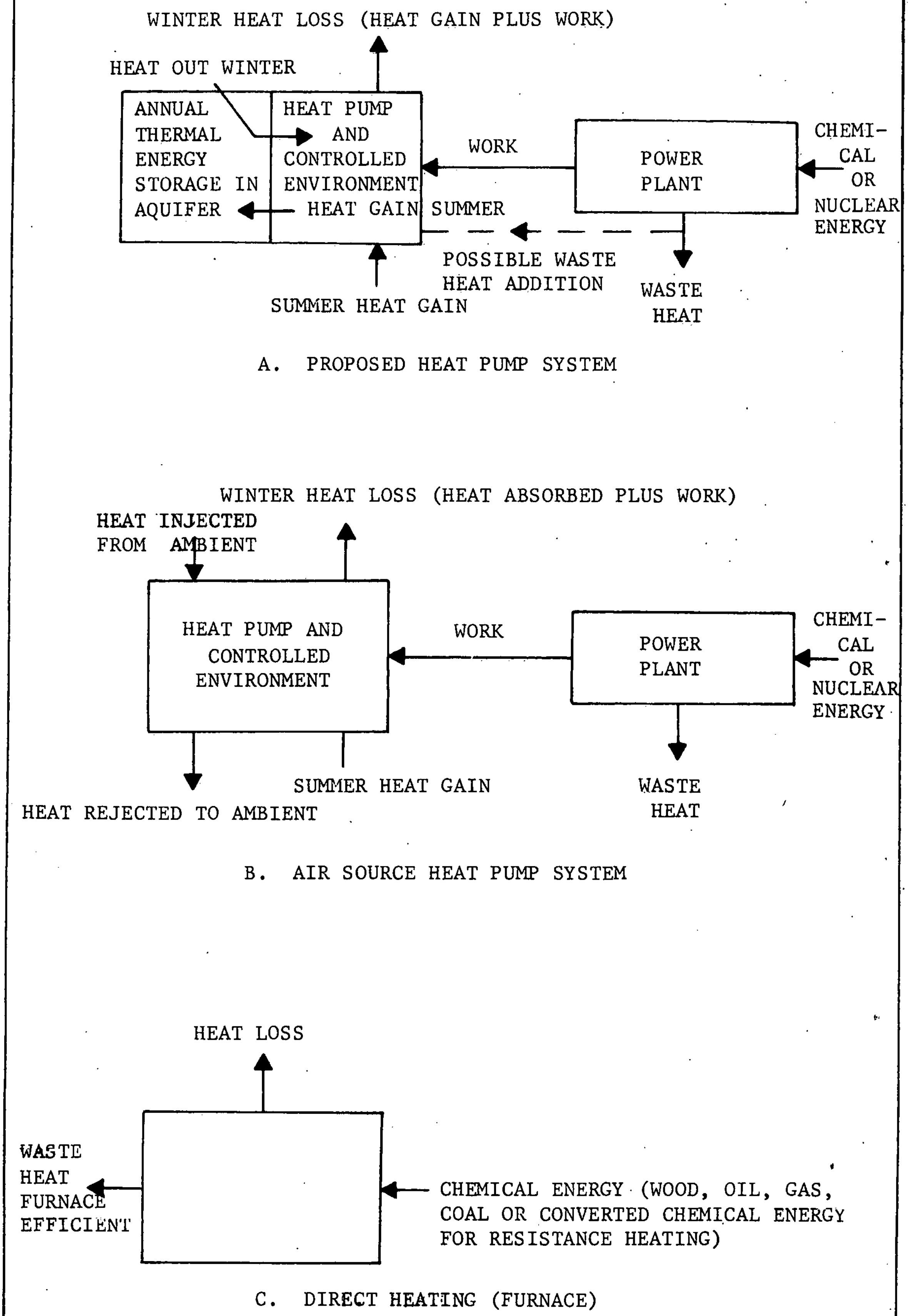


FIGURE 3-1. FIRST LAW OF THERMODYNAMICS BALANCE DIAGRAM.

3-6-79

allows energy addition or rejection to the system at all times of the year. Solar energy can be added during the summer at maximum insolation. Heat can be rejected in the winter.

The second law of thermodynamics analysis[4] analyzes 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 3-2, the total entropy generation equals $(Q_L + W)/T_H - Q_L/T_L$. The rejected energy is the energy absorbed from the lower temperature source plus the work into the system or $Q_L + W$. The thermal energy absorbed is Q_L. The entropy term assumes constant temperature heat rejection and absorption. This term must be greater than or equal to zero. As the temperatures approach each other, the work required to keep the entropy positive approaches zero. Minimizing the temperature difference minimizes the required work. This does not account for heat pump efficiency, but places the heat pump in a system where performance can be maximized. The highest performance heat pumps economically available are used in the system to keep the work as close as possible to the minimum required work. This project does not develop the heat pump but the system in which the heat pump operates. The second law of thermodynamics analysis indicates that minimizing the temperature difference over which the heat pump operates is a major step towards maximizing performance.

Applying the second law of thermodynamics analysis to the total proposed system, the heat pump becomes part of the system. In this case the thermal energy reservoir becomes part of the system or control volume. (Part A, Figure 3-1). As a result on an annual basis the thermal energy stored and recovered from the reservoir need not be included in the entropy generation assuming ideal injection and recovery. For heating utilization the

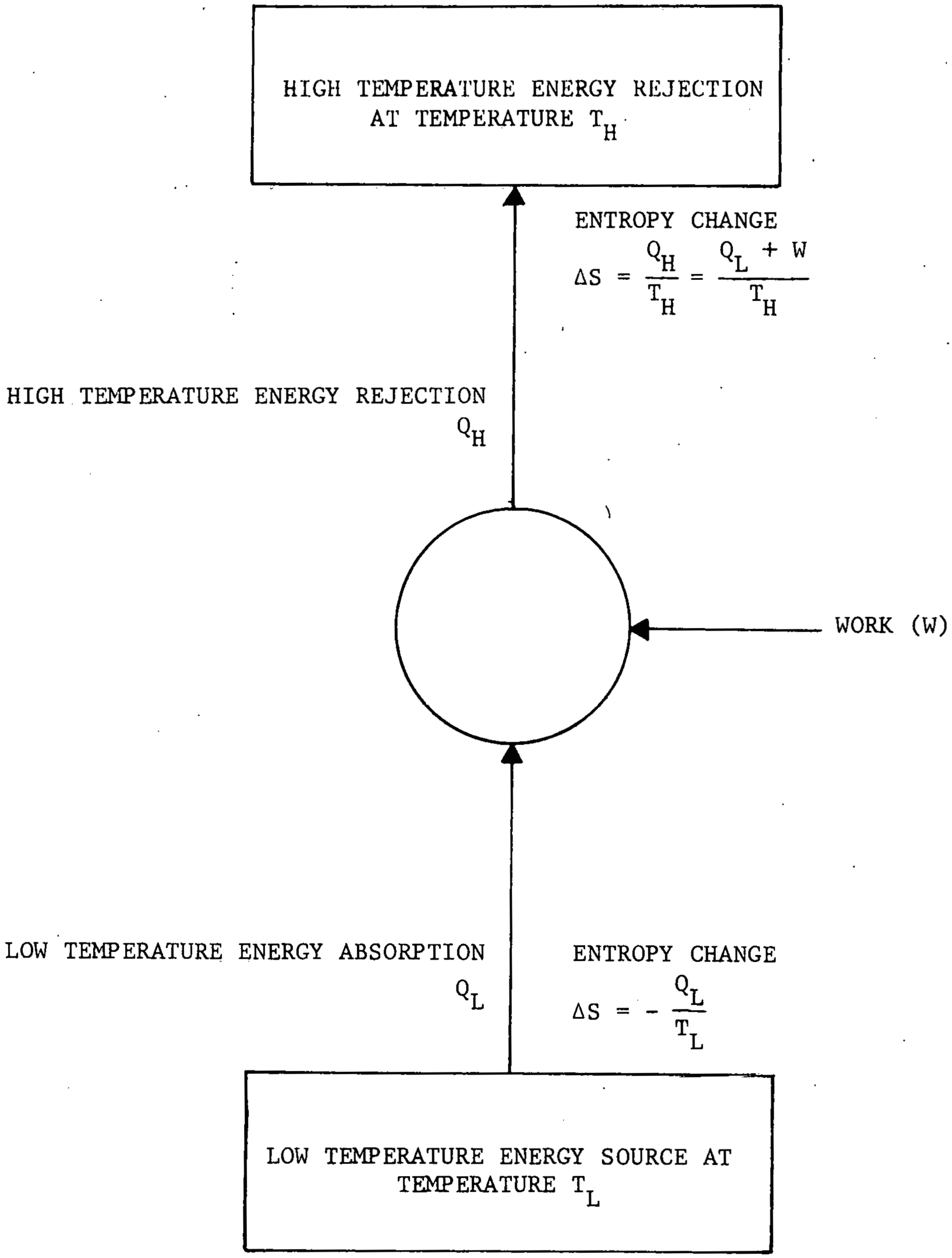


FIGURE 3-2. SECOND LAW OF THERMODYNAMICS BALANCE DIAGRAM

3-9-79

losses to environment, and for cooling the gains from the environment are included in the entropy analysis.

Since the performance based on a straight entropy analysis is difficult to visualize, number calculations are based on availability. The reference temperature utilized is $77^{\circ}F$ and the hydrocarbon combustion temperature is assumed to be $3500^{\circ}R$. For the reservoir's temperature, an average temperature of $70^{\circ}F$ is used for heating and cooling. The heating is $120^{\circ}F$ and the cooling output is $50^{\circ}F$. In reality the comparisons are made to the work which can be produced by the Carnot Cycle. A calculation of Carnot efficiencies follows in order to show the physical comparisons.

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

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

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

$$\eta_{Power\ Plant} = \frac{\text{Work Out (Electrical Energy)}}{\text{Thermal Energy Input}}$$

The ideal coefficients of performance and power plant efficiency according to Carnot cycles (absolute temperatures) are:

$$COP_{Cooling} = \frac{T_L}{T_H - T_L} = \frac{510^{\circ}R}{530^{\circ}R - 510^{\circ}R} = 25.5$$

$$COP_{Heating} = \frac{T_H}{T_H - T_L} = \frac{580^{\circ}R}{530^{\circ}R - 530^{\circ}R} = 11.6$$

$$\eta_{Power\ Plant} = \frac{T_H - T_L}{T_H} = \frac{3500^{\circ}R - 537^{\circ}R}{3500^{\circ}R} = 0.847$$

Overall system performance is COP times η or overall performance Carnot capability for

Cooling = 25.5 X 0.847 = 21.6

Heating = 11.6 X 0.847 = 9.83

These numbers represent the cooling or heating energy available for a unit of source energy. Ideally as energy levels are degraded the quantity of energy increases and availability remains constant. The numbers are very sensitive to the temperature differences. The proposed systems and air source heat pump system move in the direction of ideal performances. Using high quality energy for direct heating is a waste as an overall performance is limited to less than 1.0. The availabilities at the end of this section demonstrate this phenomena. The system performance is maximum, air-source heat pumps rank second, and hydrocarbon and resistance heating ranks last.

Numerous studies have been made using the earth and other types of storage to minimize these temperature differences.[5,6] This has not only been true in the United States, but also in Europe.[7,8] 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 work formations through which water can flow to transfer the thermal energy. A large, annual storage system is developed for a community energy system.

The coefficients of performance are applied directly to the heating and cooling sequence in the basic and alternative systems developed in this report. For comparison purposes, the listed performance for approximately three ton water source heat pumps are utilized. The performance of all three tons plus or minus ten percent ARI listed water-source heat pumps is averaged and used as the performance for water-source heat pump B. The high performance is used for water-source heat pump A. The basic performance is corrected for changes in water temperature, air temperature and humidity using detailed data from American Air Filter. General comparisons with data from other manufacturers showed these corrections are valid.

The performance of water-source heat pump A and water source heat pump B in the basic system and alternatives are given in Tables 3-1 to 3-6. Compressor and fan power are included in the ARI listings. A conservative estimate of 300 watts for water pump power is developed in Analysis Section XI. C. The systems are:

1. Basic: This system supplies water directly to the heat pump for cooling at 60°F and for heating at 80°F. The high performance water-source heat pump A and average performance water-source heat pump B operating characteristics are given in Tables 3-1 and 3-2. The percentage difference between the two heat pumps is small.

2. Alternative I: This alternative uses the same water temperatures as the basic system. However, a theoretical 50 efficient heat exchanger is placed between the air flow and water flow (Figure 3-3). As a result, some direct heat exchange occurs which does not require heat pump work. This becomes particularly effective with the new thermostat settings of 78°F for cooling and 65°F for heating. Performance shows a small finite step increase. Performance for water-source heat pump A and water-source heat pump B are given in Table 3-3 and 3-4 for this alternative.

3. Alternative II: This alternative uses a low water temperature range of 40°F to 60°F. As a result, the cooling is accomplished by direct heat exchange without heat pump work. The performance for cooling more than doubles. However, due to a lower temperature water for heating, the heating shows a small decrease in performance. This system is similar to the ice-making heat pump system. This system uses 40°F water in a nature provided aquifer rather than a man-produced storage system for ice. This system uses "free cooling."

4. Alternative IV: This alternative is the opposite of Alternative III as hot water in the temperature range of 80°F to 1000°F is stored in the

Table 3-1. System Performance of Water Source Heat Pump A

HEATING

Entering Water Temperature (°F)	Capacity[1] (KW)	Power[2] (KW)	COP[3]
59	8.00	2.84	2.82
70	9.50	3.22	2.96
81	10.99	3.52	3.13
90	12.51	3.95	3.16

COOLING (67°F WBT)

Entering Water Temperature (°F)	Capacity[1] (KW)	Power[2] (KW)	COP[3]
64	11.23	3.33	3.37
75	10.81	3.43	3.15
84	10.40	3.60	2.89
95	9.99	3.73	2.68

(1) Based on ARI listing-High Performance Unit

(2) Power for compressor, fan, and water pump

(3) COP = Capacity (KW)/input to compressor + fan + 0.3 KW for water pumping

Table 3-2. System Performance of Water Source Heat Pump B

HEATING

Entering Water Temperature (°F)	Capacity (MBTUH)	Power (KW)	COP
60	27.3	3.08	2.66
70	32.4	3.49	2.72
80	37.5	3.82	2.88
90	42.7	4.29	2.92

COOLING (67°F WBT)

Entering Water Temperature (°F)	Capacity (MBTUH)	Power KW	COP
65	38.3	3.78	2.97
75	36.9	3.90	2.78
85	35.5	4.09	2.54
95	34.1	4.24	2.36

(1) Based on ARI listing-Average Performance Unit

(2) Power for compressor, fan, and water pump

(3) COP = Capacity (KW)/input to compressor + fan + 0.3 KW for water pumping

Table 3-3. System Performance for Modified Water Source Heat Pump A with Water to Air Heat Exchanger -- Alternative I

HEATING

IDBT[4] (°F)	Pump Capacity[1] (KW)	Water Coil Capacity (KW)	Total Capacity[5] (KW)	Power[2] (KW)	COP[3]
61	9.50	4.10	13.60	3.22	4.22
64	9.88	3.05	12.92	3.29	3.92
70	11.75	2.02	13.77	3.74	3.69

COOLING(67°F WBT)

IDBT[4] (°F)	Pump Capacity[1] (KW)	Water Coil Capacity (KW)	Total Capacity[5] (KW)	Power[2] (KW)	COP[3]
70	11.23	2.02	13.25	3.33	3.98
75	11.14	3.05	14.19	3.35	4.23
81	11.02	4.04	15.06	3.38	4.46
84	10.93	5.07	16.00	3.40	4.71

(1) Based on ARI listing-High Performance Unit

(2) Power for compressor, fan, and water pump

(3) COP = Capacity (KW)/input to compressor + fan + 0.3 KW for water pumping

(4) Indoor dry bulb temperature

(5) Dry coil capacity based on inlet water temperatures of 60°F for cooling 80°F for heating

Table 3-4. System Performance of Water Source Heat Pump B with Water to Air Heat Exchanger -- Alternative I

HEATING

IDBT (^{o}F)	Pump Capacity (MBTUH)	Water Coil Capacity (MBTUH)	Total Capacity (MBTUH)	Power (KW)	COP
60	32.4	14.0	46.4	3.48	3.90
65	33.7	10.4	44.1	3.56	3.63
70	40.1	6.9	47.0	4.05	3.40

COOLING

IDBT (^{o}F)	Pump Capacity (MBTUH)	Water Coil Capacity (MBTUH)	Total Capacity (MBTUH)	Power (KW)	COP
70	38.3	6.9	45.2	3.78	3.5
75	38.0	10.4	48.4	3.80	3.73
80	37.6	13.8	51.4	3.84	3.93
85	37.3	17.3	54.6	3.86	4.14

(1) Based on ARI listing-Average Performance Unit

(2) Power for compressor, fan, and water pump

(3) COP = Capacity (KW)/input to compressor + fan + 0.3 KW for water pumping

(4) Indoor dry bulb temperature

(5) Dry coil capacity based on inlet water temperatures of 60^{o}F for cooling 80^{o}F for heating

Table 3-5. System Performance for Modified Water Source Heat Pump A with Water to Air Heat Exchanger -- Alternative II

Water temperature variation 40°F-60°F

HEATING

IDBT[2] (°F)	Air Flow (ccfm)	Water Flow (gpm)	Capacity[1] (KW)	Power[3] (KW)	COP[4]
16	1000	4	8.32	2.84	2.93
18	1000	4	8.18	2.84	2.88
21	1000	4	8.00	2.84	2.82

COOLING (67°F WBT)

IDBT[5] (°F)	Air Flow (m^3/min)	Water Flow (gpm)	Capacity[1] (KW)	Power[6] (KW)	COP[4]
61	28.3	2.42	5.92	.9	6.51
64	28.3	3.91	10.29	.9	11.0
66	28.3	4.88	13.28	.9	14.1
70	28.3	6.00	16.53	.9	18.0

(1) Based on ARI listing-High Performance Unit

(2) Indoor dry bulb temperature, °F

(3) Power for compressor + fan + 0.3 KW for water pumping

(4) COP = capacity (KW)/input power

(5) Indoor wet bulb temperature, °F

(6) Power input for fan = 0.3 KW for water pumping

Table 3-6. System Performance for Modified Water Source Heat Pump A with Water to Air Heat Exchanger -- Alternative III

Water temperature variation 80°F-100°F

HEATING

IDBT[1] (°F)	Air Flow (ccfm)	Water Flow (m^3/hr)	Capacity[2] (KW)	Power[3] (KW)	COP[4]
61	2000	4.79	15.12	1.05	14.40
64	2000	4.17	13.39	1.05	12.75
70	2000	3.60	11.64	1.05	11.09

COOLING (67°F WBT)

IWBT[5] (°F)	Air Flow (ccfm)	Water Flow (m^3/hr)	Capacity[6] (KW)	Power[7] (KW)	COP[4]
61	1000	4.48	8.47	3.6	2.35
64	1000	4.61	9.09	3.7	2.46
66	1000	4.79	9.79	3.8	2.58
70	1000	5.01	10.11	3.9	2.59

(1) Indoor dry bulb temperature, °F

(2) Water coil + fan work

(3) Power input for fan + 0.3 KW for water pumping

(4) COP = Capacity/input power

(5) Indoor wet bulb temperature

(6) Based on ARI listing-High Performance Unit

(7) Power for compressor + fan + 0.3 KW for water pumping

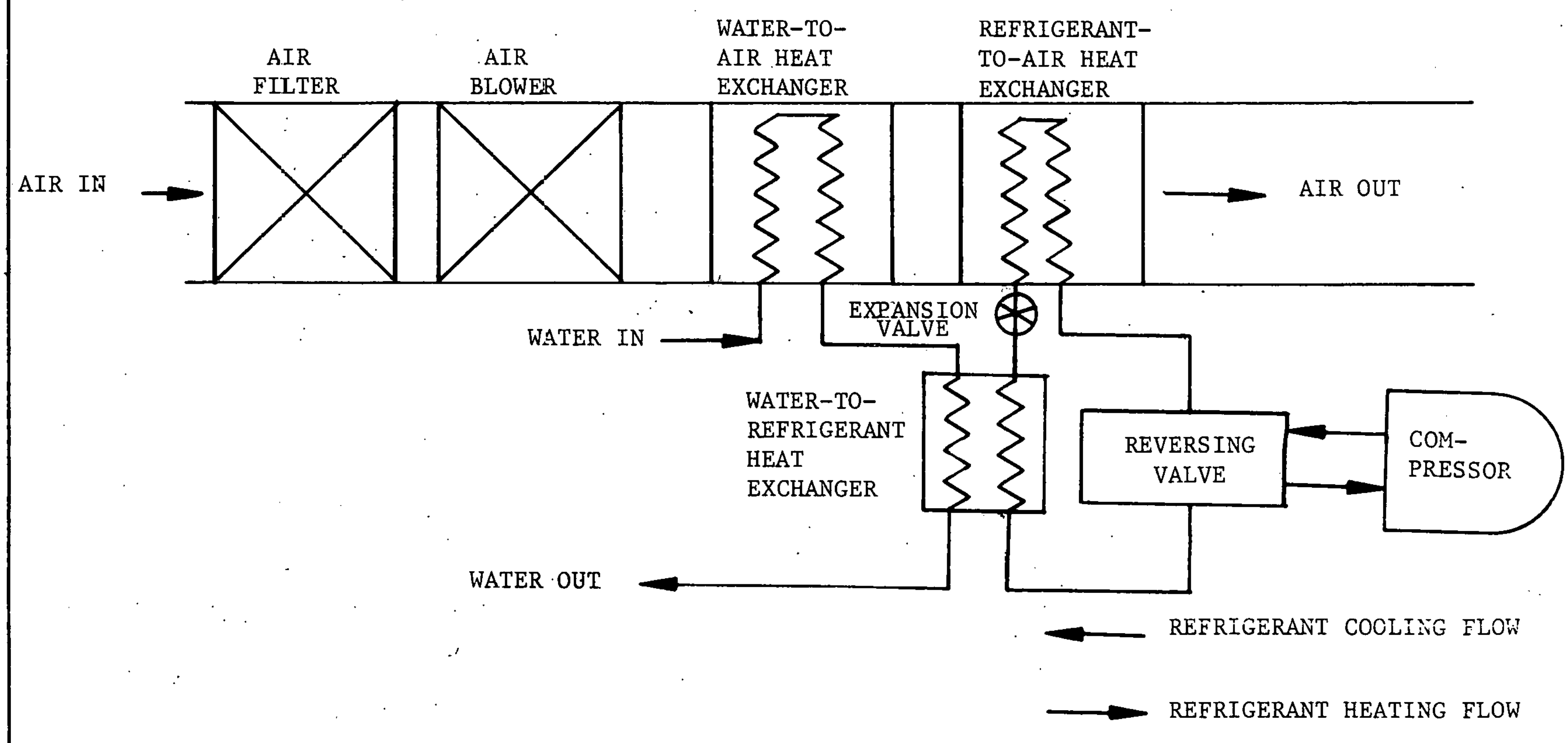


FIGURE 3-3. MODIFICATION OF BASIC WATER SOURCE HEAT PUMP SYSTEM

aquifer. The heating is by direct heat exchange and the heating performance more than doubles. However, the cooling performance shows a decrease due to the warmer water. This system uses "free heating." The performance for water-source heat pump A is given in Table 3-6.

The heat pumps and heat exchangers for these systems in the operating temperatures range are available as off-the-shelf items. The matching of components for the upper range temperature in large sized equipment must be analyzed for component and compressor capabilities in the entire pressure range.

The performance values for the proposed system and alternatives have higher potential in the future. The average ARI listed water source heat pump performance data have potential for improvement especially in small units. New compressors of higher efficiency have been introduced to the market for a few years. Basic data from Tecumseh support this fact.[9] In addition, motor efficiencies have improved and heat-exchanger areas have increased. Estimates for these changes will increase water source heat pump performance by 20 to 30%. This means the proposed system and alternatives have potential performance increases of the same magnitude.

Systems by WESCORP[10] have incorporated the new compressors, new motors, and improved heat exchangers. The basic performance in these references is over 40% higher than performance of present average ARI listing. However, capital costs are over 100% higher. At standard conditions without the work for water pumping, the COP for heating is over 4.2 and the COP for cooling is over 3.5 for units in the five to ten ton range. Variation in performance with water temperature is smaller than for the standard water-source heat pumps. Placing these units in the proposed system with the work for water pumping and applying the performance increase could result in future performance 20% greater than the present evaluations. It must be noted there is some question on the testing conditions for the WESCORP performance data. However, a performance evaluation is required to actuate the guarantee for each installation. These evaluations have matched predicted values.

In addition, heat exchangers are available for these units to heat hot water. The superheated refrigerant leaving the compressor is used for heating the hot water. The result is that hot water is basically energy-free during cooling periods and will be heated with a COP of over 4.2 during heating periods.

Accurate capital costs on heat pumps are hard to obtain; however general list price data by WESCORP indicate a capital cost of over two times the cost of standard water source heat pumps. With large usage and competition these prices are expected to decrease. Other manufacturers, including American Air Filter, are expected to have similar performance units within the next two years.

Table 3-7 gives the performance of larger reciprocating water source heat pumps. The performance is slightly higher than the smaller units. The percent change in performance and capacity as a function of temperature are extrapolated to the large centrifugal and screw type compressors, since general performance data for these units over wide temperature ranges have not been available. This assumption is based on the compressor pressure ratios being direct functions of temperature. The work input to the units, assuming compressor efficiency is constant, is a direct function of pressure ratio. Tables 3-8 and 3-9 give the predicted increases in performance and capacity for the large centrifugal and screw type compressors units when utilized in the proposed

TABLE 3-7. HEATING AND COOLING PERFORMANCE FOR LARGE* ROOF MOUNTED WATER SOURCE HEAT PUMPS

SYSTEM	ENTERING WATER TEMP. °F	WATER FLOW HEAT/COOL gpm	HEATING CAPACITY MBTUH	INPUT[1] POWER KW	HEATING[2] COP	COOLING CAPACITY MBTUH	INPUT[1] POWER KW	COOLING[2] COP
RCRM - 24@ 8,000 cfm	60	30/40	249	21.9	3.33	310	25.4	3.58
	70		273	23.3	3.45	299	27.0	3.24
	80		298	24.8	3.52	284	28.8	2.89
	90		320	26.6	3.53	267	30.4	2.58
RCRM - 33@ 10,000 cfm	60	42/56	331	30.6	3.17	415	33.5	3.64
	70		363	32.4	3.28	399	35.5	3.29
	80		396	34.6	3.36	380	38.0	2.93
	90		425	37.1	3.36	356	40.1	2.61
RCRM - 42@ 12,000 cfm	60	53/71	450	41.2	3.21	555	46.0	3.54
	70		495	43.7	3.32	535	48.9	3.20
	80		539	46.6	3.39	508	52.3	2.85
	90		579	49.9	3.40	477	55.2	2.54

*Based on specifications from American Air Filter catalog AHU-1-121-JAN-02 for heating temperature of 70°F and cooling wet bulb temperature 67°F, with water pump power rated at 40w/gpm.

[1]Power for compressor, fan, and water pump.

[2]COP = Capacity (Kw)/Kw (compressor + fan + water pumping).

Table 3-8. Screw Type Chillers Performance and Cost Data*

Open Chillers

System A Capacity Tons	System B Capacity Tons	System A COP	System B COP	System A Capital Cost ($)	System B Capital Cost ($)	System A Inst. Cost **	System B Inst. Cost **	System A Total Installed Cost ($)	System B Total Installed Cost ($)
500	454.55	4.70	5.97	54,000.00	50,708.11	3,825.00	3,540.84	57,825.00	54,248.95
750	681.82	4.70	5.97	70,569.00	66,266.74	5,312.09	4,917.43	75,881.09	71,184.17
Hermetic Chillers									
100	90.91	4.18	5.31	18,666.96	17,529.00	1,038.64	961.48	19,705.60	18,490.48
200	181.82	4.22	5.36	29,495.34	27,697.28	1,820.96	1,685.68	31,316.30	29,382.96
500	454.55	4.33	5.50	54,000.00	66,266.74	5,312.09	4,917.43	75,881.09	71,184.17

*System A Capacity is the capacity required for the building. System B is the standard capacity requirement which gives the identical output as System A operating with 60°F condensor water rather than the 80°F water produced by cooling towers.

Performance and capacity variations with temperature taken from American Air Filter Catalog AHU-1-121-JAN-02.

Performance (COP) is based on chiller power input. Values do not include water flow pumping or air flow distribution.

Unit cost data and installation man-hours taken from J.E. Christian, Central Cooling - Compressive Chillers, ANL/CES/TE78-2, 1978.

**$17 per man-hour is assumed.

Table 3-9. Centrifugal Chiller Performance and Cost Analysis*

Open Chillers

System A Capacity Tons	System B Capacity Tons	System A COP	System B COP	System A Capital Cost ($)	System B Capital Cost ($)	System A Inst. Cost **	System B Inst. Cost **	System A Total Installed Cost ($)	System B Total Installed Cost ($)
90	81.32	4.16	5.28	17,413.00	16,351.62	953.68	882.84	18,366.68	17,234.46
500	454.55	4.70	5.97	54,000.00	50,708.11	3,825.00	3,540.84	57,825.00	54,248.95
1000	909.09	4.90	6.22	85,324.46	80,122.43	6,706.03	6,207.79	92,030.49	86,330.22
1250	1,136.36	4.81	6.11	98,863.09	92,835.50	8,034.57	7,437.61	106,897.66	100,273.11

Hermetic Chillers

System A Capacity Tons	System B Capacity Tons	System A COP	System B COP	System A Capital Cost ($)	System B Capital Cost ($)	System A Inst. Cost **	System B Inst. Cost **	System A Total Installed Cost ($)	System B Total Installed Cost ($)
80	72.73	4.08	5.18	16,110.65	15,128.80	866.90	802.52	16,977.55	15,931.32
500	454.55	4.40	5.59	54,000.00	50,708.11	3,825.00	3,540.84	57,825.00	54,248.95
1000	909.09	4.60	5.84	85,324.46	80,122.43	6,706.03	6,207.73	92,030.49	86,330.22
2000	1,818.18	4.40	5.59	134,819.70	126,600.06	11,757.10	10,883.58	146,576.80	137,483.64

*System A Capacity is the capacity required for the building. System B is the standard capacity requirement which gives the identical output as System A operating with 60°F condensor water rather than the 80°F water produced by cooling towers.

Performance and capacity variations with temperature taken from American Air Filter Catalog AHU-1-121-JAN-02.

Performance (COP) is based on chiller thermal output divided by chiller power input. Values do not include water flow pumping or air flow distribution.

Unit cost data and installation man-hours taken from J.E. Christian, Central Cooling - Compressive Chillers, ANL/CES/TE78-2, 1978.

**$17 man-hour is assumed.

system. The performance and capacity show increases with a corresponding decrease in capital cost for identical output.

Figures 3-4 and 3-5 show a major advantage of the proposed system performance versus the air source heat pump's performance. Neither the coefficient of performance nor the capacity are affected by ambient conditions in the proposed system. The air source heat pumps at 0^{o}F have lost 80% of their 60^{o}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 will help 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%. Integrating some of these systems into an electrical system decreases power plant peak loads. 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 3-10 and 3-11 present the heating and cooling energy requirements of the various type systems for the well-insulated typical home described in Section XI.A for five cities. The basic system and alternatives are compared to the standard heating and cooling systems. These include gas and oil heating, resistance heating, standard air-source air conditioning and standard air-source heat pumps. System performance is based on the output in Tables 3-1 to 3-6. Air-source heat pump and air-conditioning performance is evaluated in Section XI.A. Oil and gas efficiencies are indicated at 62% which includes performance and combustion efficiency. The tables show the total energy in electrical units where applicable or as fuel heating values for

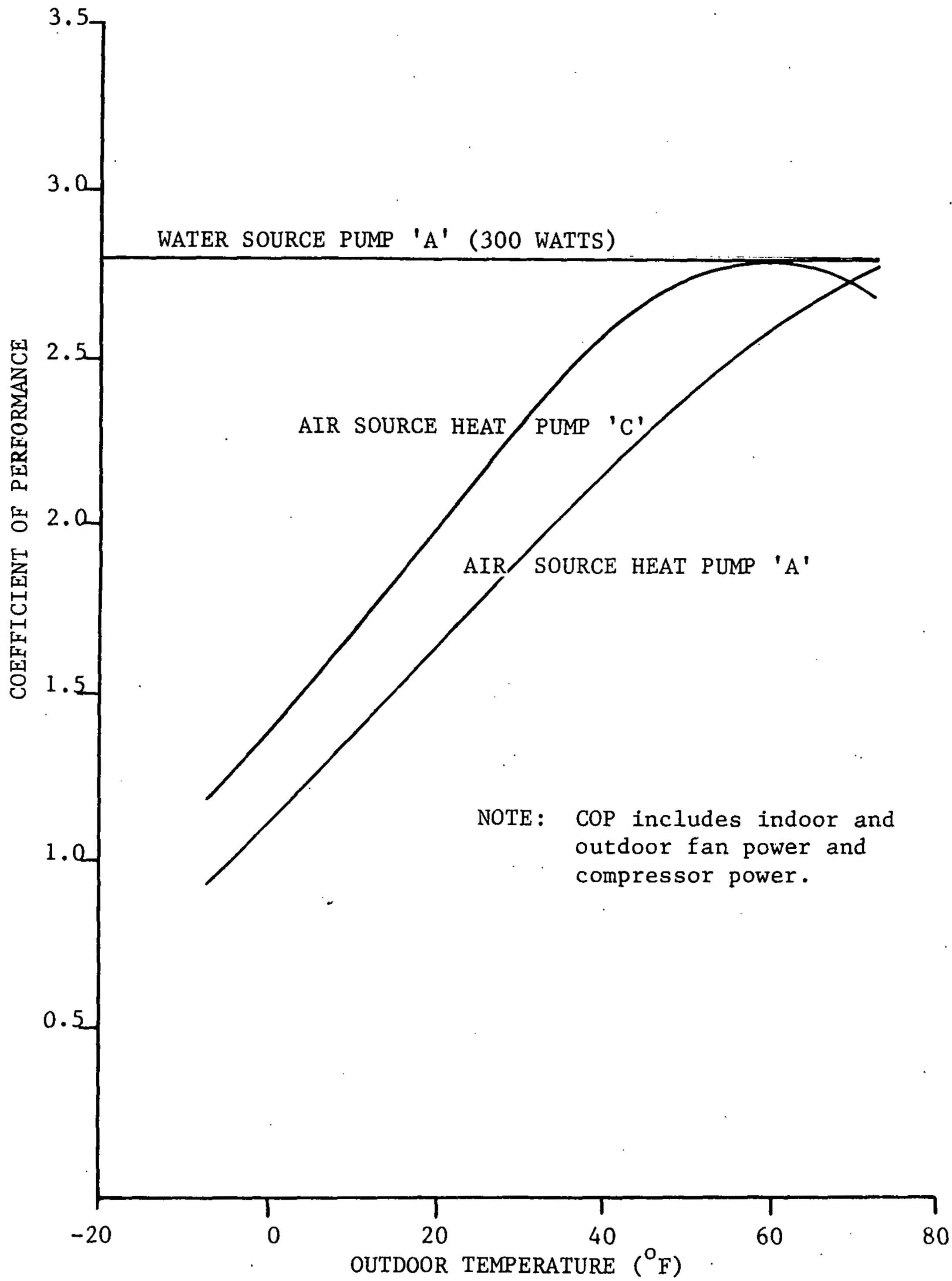


FIGURE 3-4. HEAT PUMP HEATING PERFORMANCE COMPARISON

2.12.78

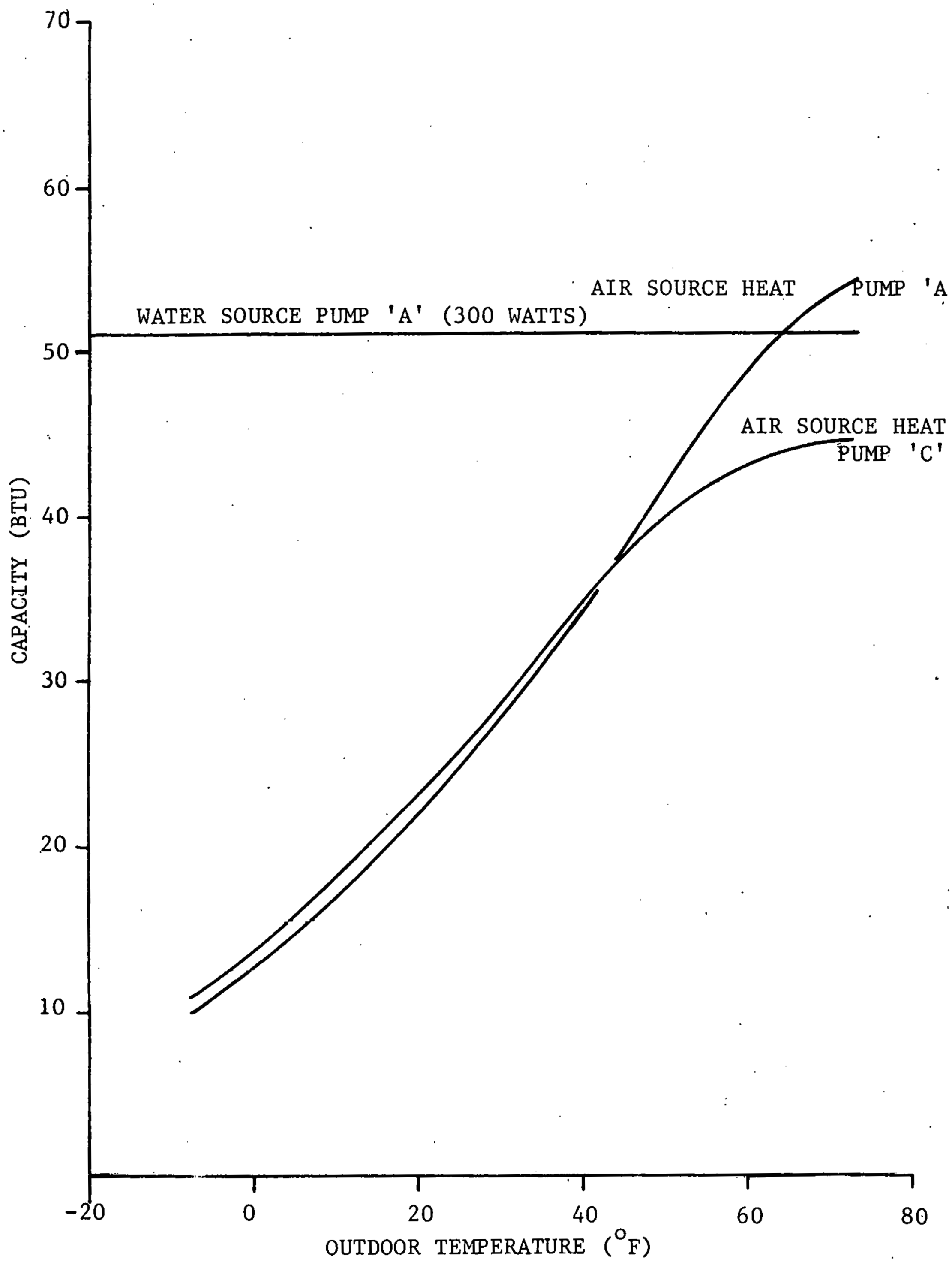


FIGURE 3-5. HEAT PUMP HEATING CAPACITY COMPARISON

4.12.78

Table 3-10. Energy Required for Heating Well Insulated Typical Home

	Birmingham	New York	Miami	Detroit	St. Louis
Annual Heating KWH	$21.6(10^3)$	$28.3(10^3)$	$1.05(10^3)$	37.3	28.1
Annual Heating BTU	$73.7(10^6)$	$96.6(10^6)$	$3.582(10^6)$	$127.1(10^6)$	95.9
Direct Electrical Heating					
Power Input KW-hr	$21.6(10^3)$	$28.3(10^3)$	$1.05(10^3)$	$37.3(10^3)$	$28.1(10^3)$
Power Plant Input BTU	$211(10^6)$	$276(10^6)$	$10.2(10^6)$	$364(10^6)$	$274(10^6)$
Gas And Oil Heating					
Furnace Input BTU	$119(10^6)$	$156(10^6)$	$5.78(10^6)$	$205(10^6)$	$155(10^6)$
Gas Therms	1189	1557	5778	2053	1546
Oil Gallons	820	1074	39.9	1415	1067
Air-Source Heat Pump A					
COP	2.43	2.33	2.70	2.16	2.20
Power Input KW-hr	$8.89(10^3)$	$11.6(10^3)$	$.43(10^3)$	$15.3(10^3)$	$11.6(10^3)$
Power Plant Input BTU	$86.6(10^6)$	$113(10^6)$	$4.19(10^6)$	$149(10^6)$	$113(10^6)$
Air-Source Heat Pump C					
COP	2.0	1.92	2.27	1.76	1.80
Power Input KW-hr	$10.8(10^3)$	$14.2(10^3)$	$.52(10^3)$	$18.6(10^3)$	$14.1(10^3)$
Power Plant Input BTU	$105(10^6)$	$138(10^6)$	$5.1(10^6)$	$182(10^6)$	$137(10^6)$
Twin Air-Source Heat Pump A					
COP	2.45	2.37	2.70	2.21	2.25
Power Input KW-hr	$8.81(10^3)$	$11.5(10^3)$	$.43(10^3)$	$15.2(10^3)$	$11.5(10^3)$
Power Plant Input BTU	$85.8(10^6)$	$112(10^6)$	$4.15(10^6)$	$148(10^6)$	$112(10^6)$
Water-Source Heat Pump A					
COP	3.13	3.13	3.13	3.13	3.13
Power Input KW-hr	$6.96(10^3)$	$9.15(10^3)$	$.34(10^3)$	$12.0(10^3)$	$9.06(10^3)$
Power Plant Input BTU	$67.2(10^6)$	$88.0(10^6)$	$3.25(10^6)$	$115(10^6)$	$87.5(10^6)$
Modified Water-Source Heat Pump A Alternative I					
COP	3.92	3.92	3.92	3.92	3.92
Power Input KW-hr	$5.51(10^3)$	$7.21(10^3)$	$.27(10^3)$	$9.51(10^3)$	$7.16(10^3)$
Power Plant Input BTU	$53.8(10^6)$	$70.4(10^6)$	$2.60(10^6)$	$92.5(10^6)$	$70.0(10^6)$
Alternative II					
COP	2.88	2.88	2.88	2.88	2.88
Power Input KW-hr	$7.49(10^3)$	$9.82(10^3)$	$.36(10^3)$	$12.9(10^3)$	$9.71(10^3)$
Power Plant Input BTU	$73.0(10^6)$	$95.4(10^6)$	$3.53(10^6)$	$125.9(10^6)$	$95.3(10^6)$
Alternative III					
COP	12.75	12.75	12.75	12.75	12.75
Power Input KW-hr	$1.70(10^3)$	$2.23(10^3)$	$.09(10^3)$	$2.93(10^3)$	$2.21(10^3)$
Power Plant Input BTU	$16.5(10^6)$	$21.6(10^6)$	$.80(10^6)$	$28.5(10^6)$	$21.5(10^6)$

Table 3-11. Energy Required for Cooling Well Insulated Typical Home

	Birmingham	New York	Miami	Detroit	St. Louis
Annual Cooling KW-hr	15.8(10^3)	6.97(10^3)	29.3(10^3)	6.95(10^3)	12.9(10^3)
Annual Cooling BTU	53.8(10^6)	23.8(10^6)	99.8(10^6)	23.7(10^6)	43.9(10^6)
Air-Source Heat Pump A					
COP	2.73	2.82	2.75	2.79	2.72
Power Input KW-hr	5.80(10^3)	2.56(10^3)	10.74(10^3)	2.55(10^3)	4.73(10^3)
Power Plant Input BTU	56.3(10^6)	24.9(10^6)	104(10^6)	24.8(10^6)	45.9(10^6)
Air-Source Heat Pump C					
COP	2.0	2.47	2.40	2.44	2.37
Power Input KW-hr	7.9(10^3)	3.49(10^3)	14.6(10^3)	3.48(10^3)	6.45(10^3)
Power Plant Input BTU	76.9(10^6)	34(10^6)	143(10^6)	33.8(10^6)	62.8(10^6)
Twin Air-Source Heat Pump A					
COP	2.72	2.81	2.75	2.79	2.72
Power Input KW-hr	5.82(10^3)	2.57(10^3)	10.8(10^3)	2.56(10^3)	4.75(10^3)
Power Plant Input BTU	56.5(10^6)	24.9(10^6)	105(10^6)	24.9(10^6)	46.1(10^6)
Water-Source Heat Pump A					
COP	3.37	3.37	3.37	3.37	3.37
Power Input KW-hr	4.7(10^3)	2.07(10^3)	8.67(10^3)	2.07(10^3)	3.83(10^3)
Power Plant Input BTU	45.6(10^6)	20.1(10^6)	84.1(10^6)	20.1(10^6)	37.2(10^6)
Modified Water-Source Heat Pump A — Alternative I					
COP	4.71	4.71	4.71	4.71	4.71
Power Input KW-hr	3.35(10^3)	1.48(10^3)	6.19(10^3)	1.47(10^3)	2.73(10^3)
Power Plant Input BTU	32.6(10^6)	14.4(10^6)	60.2(10^6)	14.4(10^6)	26.6(10^6)
Alternative II					
COP	18.0	18.0	18.0	18.0	18.0
Power Input KW-hr	.88(10^3)	.39(10^3)	1.62(10^3)	.39(10^3)	.72(10^3)
Power Plant Input BTU	8.55(10^6)	3.77(10^6)	15.8(10^6)	3.76(10^6)	6.97(10^6)
Alternative III					
COP	2.59	2.59	2.59	2.49	2.59
Power Input KW-hr	6.10(10^3)	2.69(10^3)	11.2(10^3)	2.68(10^3)	4.98(10^3)
Power Plant Input BTU	59.3(10^6)	26.2(10^6)	109(10^6)	26.1(10^6)	48.4(10^6)

furnaces. These quantities are at the point of consumption. Also, the electrical units are changed to total heating values for fuel entering the power plant. This allows energy consumption comparisons at the point of raw energy usage. A 40% efficient plant is utilized which represents a highly efficient fossil fuel plant.

The energy requirements are converted to energy source requirements per ASHRAE Standard 90-75 Section XII.[11] The results are given in Tables 3-12 to 3-14 for heating, cooling, and the combination of heating and cooling. The change of types of resource energies between the proposed systems and standard systems may be noted from the tables.

The comparisons indicate that in areas such as Miami with both mild, cold and warm temperatures, the advantages of the system are relatively small. 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 to 6. For example, Detroit shows approximately the same performance for Alternative I and II. Farther north Alternative III is most feasible if plus 100 $^{\circ}$F waste heat is available. Otherwise Alternative I shows better results.

For the vast majority of the country, with the exception of the lower half of Florida and part of the western coastal area, the system shows a sizable increase in performance. A summary of expected performance nationally is shown in Figures 3-6 and 3-7. The numbers for the water source heat pumps are for high performance units. The numbers are below the performance data published by WESCORP for units presently being sold. The data for the air to air heat pumps are averages obtained for the analysis of the 24 cities. The COP's utilized for the basic proposed system are 4.0 for

Table 3-12. Annual Resource Consumption for Heating

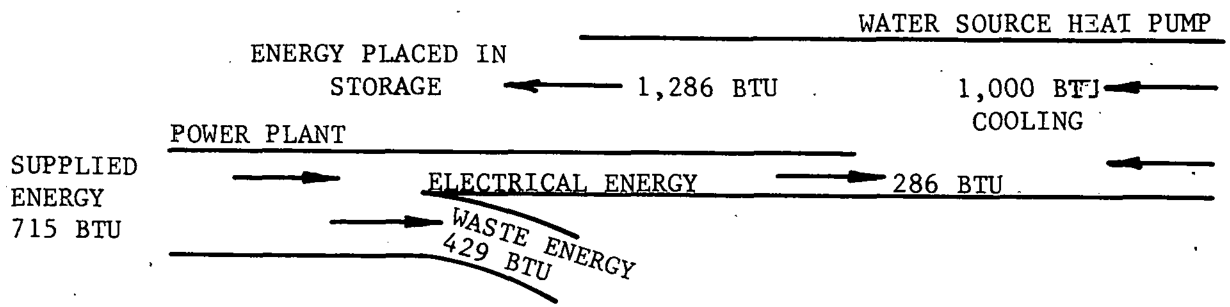
	Birmingham	New York	Miami	Detroit	St. Louis	Birmingham	New York	Miami	Detroit	St. Louis
End Use										
Energy 10^3 KWH	21.6	28.3	1.05	37.3	28.1					
Energy 10^6 BTU	73.7	96.9	3.58	127	95.9					
Resource	Direct Electric Heating					Alternative I				
COP						3.92	3.92	3.92	3.92	3.92
Coal S. Tons	7.34	5.66	.28	17.5	1.12	1.87	1.44	0.07	4.45	2.87
Gas MCF	6.26	.57	.59	12.7	280	1.60	0.14	0.15	3.22	9.31
Oil BBL	1.08	20.0	.56	6.71	2.53	0.28	5.34	0.14	1.71	0.93
Nuc. Grams	64.6	157	4.03	111	36.3	16.47	40.0	1.04	28.3	36.7
Hydro 10^3 KWH	4.75	3.68	.07	.37	1.41	1.21	0.94	0.02	0.09	0.64
	Air Source Heat Pump A					Alternative II				
COP	2.43	2.33	2.70	2.16	2.20	2.88	2.88	2.88	2.88	2.88
Coal S. Tons	3.02	2.32	.12	7.19	.46	2.55	1.96	0.10	6.09	3.89
Gas MCF	2.58	.23	.24	5.20	116	2.17	0.20	0.20	4.40	12.63
Oil BBL	.44	8.58	.23	2.75	1.04	0.37	7.25	0.19	2.33	1.26
Nuc. Grams	26.58	64.4	1.65	45.8	14.9	22.4	54.4	1.38	38.7	49.73
Hydro 10^3 KWH	1.96	1.51	.03	.15	.58	1.65	1.27	0.03	0.13	0.87
	Water Source Heat Pump A					Alternative III				
COP	3.13	3.13	3.13	3.13	3.13	12.75	12.75	12.75	12.75	12.75
Coal S. Tons	2.37	1.83	0.09	5.64	3.62	0.58	0.45	0.02	1.38	0.89
Gas MCF	2.02	0.18	0.19	4.08	11.78	0.49	0.04	0.05	1.00	2.87
Oil BBL	0.35	6.77	0.18	2.16	1.18	0.08	1.65	0.04	0.53	0.29
Nuc. Grams	20.8	50.8	1.30	35.8	46.4	5.08	12.4	0.33	8.77	11.31
Hydro 10^3 KWH	1.53	1.19	0.02	0.12	0.82	0.37	0.29	0.01	0.03	0.20
	Fossil Fuel at 62% Efficiency									
Oil Heat, #2 or Diesel										
Oil	23.6	30.9	1.14	40.6	30.7					
Gas Heat										
Gas MCF	28	168	6.70	223	170					
Oil EBL	0.655	0.472	0.031	0.434						

Table 3-13. Annual Resource Consumption for Cooling

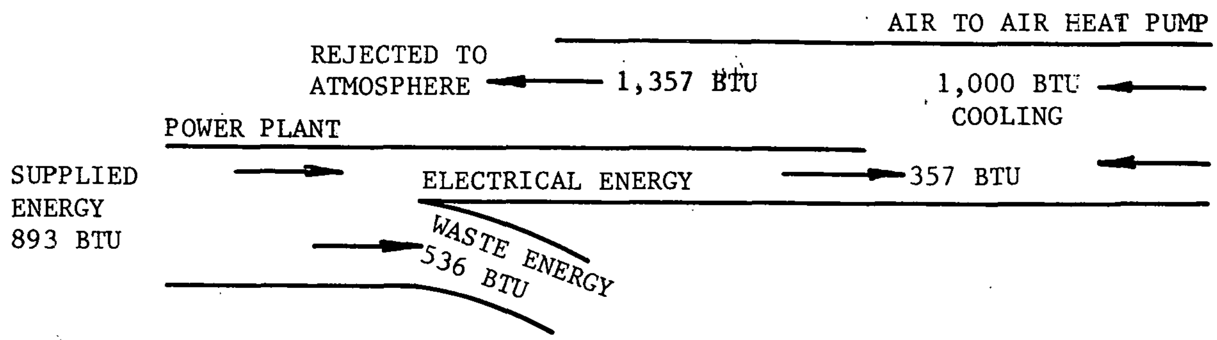
	Birmingham	New York	Miami	Detroit	St. Louis	Birmingham	New York	Miami	Detroit	St. Louis
End Use										
Energy 10^3 KWH	5.8	6.97	29.3	6.95	12.9					
Energy 10^6 BTU	53.8	23.8	99.8	23.7	43.9					
Resource	Air Source Heat Pump A					Alternative II				
COP	2.73	2.82	2.75	2.79	2.72	18.0	18.0	18.0	18.0	18.0
Coal S. Tons	2.37	.52	2.77	1.10	1.42	0.36	0.08	0.41	0.17	0.21
Gas MCF	2.03	1.46	11.1	1.87	23.5	0.31	0.23	1.69	0.29	3.55
Oil BBL	0.46	1.85	5.96	0.30	0.33	0.07	0.29	0.91	0.05	0.05
Nuc Grams	12.4	9.48	40.9	7.45	10.1	1.88	1.48	6.25	1.16	1.53
Hydro 10^3 KWH	0.64	0.30	0.43	0.02	0.05	0.10	0.04	0.07	0.01	0.01
	Water Source Heat Pump A					Alternative III				
COP	3.37	3.37	3.37	3.37	3.37	2.59	2.59	2.59	2.59	2.59
Coal S. Tons	1.92	0.43	2.26	0.91	1.15	2.50	0.57	2.93	1.88	1.49
Gas MCF	1.64	1.22	1.22	9.04	1.55	2.14	1.59	11.72	2.02	24.7
Oil BBL	0.38	1.55	4.87	0.25	0.27	0.49	2.02	6.31	0.32	0.35
Nuc Grams	10.1	7.93	33.4	6.17	8.18	13.1	10.3	43.3	8.02	10.7
Hydro 10^3 KWH	0.52	0.25	0.35	0.02	0.04	0.67	0.32	0.45	0.03	0.05
	Alternative I									
COP	4.71	4.71	4.71	4.71	4.71					
Coal S. Tons	1.37	0.44	2.27	0.91	1.00					
Gas MCF	1.17	1.23	9.09	1.55	16.56					
Oil BBL	0.27	1.56	4.88	0.24	0.24					
Nuc Grams	7.18	7.98	33.6	6.19	7.16					
Hydro 10^3 KWH	0.37	0.25	0.35	0.02	0.03					

Table 3-14. Annual Resource Consumption For Heating And Cooling

Resource	Birmingham	New York	Miami	Detroit	St. Louis	Birmingham	New York	Miami	Detroit	St. Louis
	Direct Electric Heat With Air Source Heat Pump A					Water Source Heat Pump A Heating and Cooling				
Coal S. Tons	9.71	6.18	3.05	18.6	2.54	4.29	2.26	2.35	6.55	4.23
Gas MCF	8.29	2.03	11.7	14.6	304	3.66	1.40	1.45	4.33	12.1
Oil BBL	1.54	22.7	6.52	7.0	2.86	0.73	8.32	5.05	2.41	1.35
Nuc Grams	77.0	166	45.2	118	46.4	30.9	58.7	34.7	42.0	54.6
Hydro 10^3 KWH	5.39	3.98	.50	.39	1.46	2.05	1.44	0.37	0.14	0.86
	Oil Heat With Air Source Heat Pump A					Alternative I				
Coal S. Tons	2.37	.52	2.77	1.10	1.42	3.24	1.88	2.34	5.38	3.87
Gas MCF	2.03	1.46	11.1	1.87	23.5	2.77	1.37	9.24	4.77	25.9
Oil BBL	24	33	7.1	41	31	0.55	6.90	5.02	1.95	1.17
Nuc Grams	12.4	9.48	40.9	7.45	10.1	23.7	48.0	34.6	34.5	43.9
Hydro 10^3 KWH	.64	.30	.43	.02	.05	1.58	1.19	0.37	0.11	0.67
	Gas Heat With Air Source Heat Pump A					Alternative II				
Coal S. Tons	2.37	.52	2.77	1.10	1.42	2.91	2.04	0.51	6.26	4.10
Gas MCF	130	169	17.8	225	194	2.48	0.43	1.89	4.57	12.8
Oil BBL	1.11	2.32	6.0	.87	.76	0.44	7.54	1.10	2.38	1.31
Nuc Grams	12.4	9.48	40.9	7.45	10.1	24.3	55.9	7.63	39.9	51.2
Hydro 10^3 KWH	.64	.30	.43	.02	.05	1.75	1.31	0.10	0.14	0.89
	Air Source Heat Pump A Heating and Cooling					Alternative III				
Coal S. Tons	5.39	2.84	2.89	8.29	1.88	3.08	1.02	2.95	3.16	2.38
Gas MCF	4.61	1.69	11.3	7.07	140	2.63	1.64	11.8	3.02	27.6
Oil BBL	0.90	10.4	6.19	3.05	1.37	0.57	3.67	6.35	0.85	0.64
Nuc Grams	39.0	74.0	42.6	53.3	25.0	18.2	22.7	43.6	16.8	22.0
Hydro 10^3 KWH	2.60	1.8	0.46	0.17	0.63	1.04	0.61	0.46	0.06	0.25



BASIC PROPOSED HEAT PUMP INTEGRATED COMMUNITY ENERGY SYSTEM



STANDARD AIR TO AIR CONDITIONER OR HEAT PUMP

FIGURE 3-6. GENERAL PERFORMANCE SUMMARY OF COOLING SYSTEMS

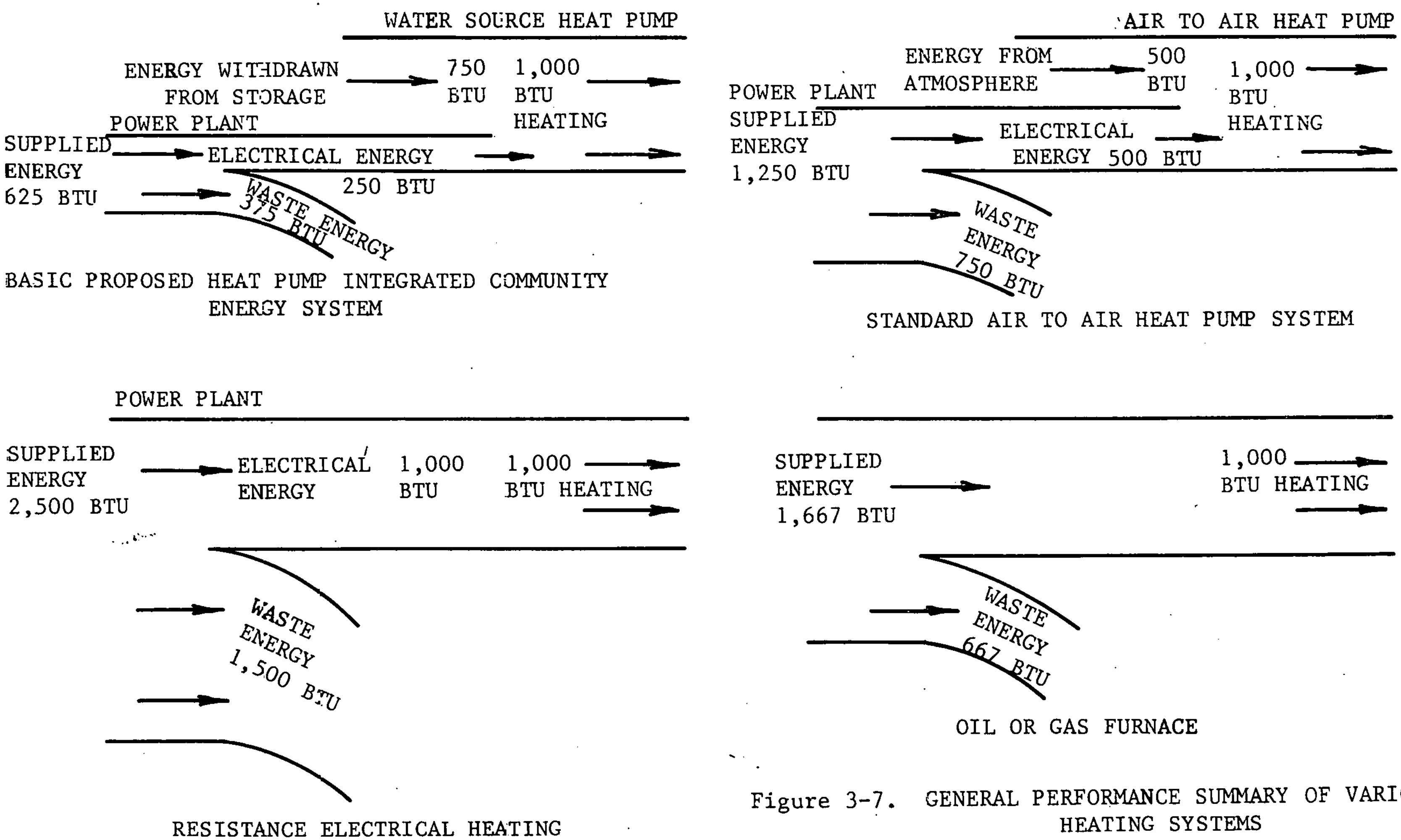


Figure 3-7. GENERAL PERFORMANCE SUMMARY OF VARIOUS HEATING SYSTEMS

heating and 3.5 for cooling. The alternative systems have higher performance in most cases. The values used for the air-source heat pump systems are 1.8 for heating and 2.8 for cooling. These correspond to the high performance units presently on the market.

Utilizing the values in Figures 3-6 and 3-7, a first and a second law analysis are made on the basic proposed system and various other heating and cooling systems for comparison. The analysis basically reduces to an availability analysis based on Carnot Cycle efficiency. The results are given in Tables 3-15 and 3-16. As energy is degraded the quantity of energy must increase according to availability analysis. The heat pump systems move in this direction and the resulting performance supports these results. The power plant efficiency is high, however, a percentage of the power is electric with a much higher conversion efficiency.

Using the same temperatures as utilized for Tables 3-15 and 3-16, the first law and second law performances for the alternatives are:

	First Law Performance	Second Law Performance	
	Alternative I		
Heating	1.57	0.128	Output 75°F
Cooling	1.88	0.140	Output 55°F
	Alternative II		
Heating	1.15	0.215	Output 120°F
Cooling	7.2	0.615	Output 50°F
	Alternative III		
Heating	5.1	0.48	Output 90°F
Cooling	1.04	0.099	Output 50°F

Table 3-15. First and Second Law Analysis Summary of Heating Systems for 1000 Btu Output*

Basic Proposed Heat Pump Integrated Community Energy System

To Power Plant		Losses at Power Plant	
Energy	625 Btu	Energy	375 Btu
Availability	536 Btu	Availability	286 Btu
From Storage		To Controlled Environment	
Energy	750 Btu	Energy	1000 Btu
Availability	56 Btu	Availability	160 Btu
First Law Performance		1000/625 = 1.6	
Second Law Performance		160/536 = 0.286	

Standard Air-Source Heat Pump System

To Power Plant		Losses at Power Plant	
Energy	1250 Btu	Energy	750 Btu
Availability	1071 Btu	Availability	571 Btu
From Atmosphere		To Controlled Environment	
Energy	500 Btu	Energy	1000 Btu
Availability	0	Availability	160 Btu
First Law Performance		1000/1250 = 0.80	
Second Law Performance		160/1071 = 0.15	

Oil or Gas Furnace

To Power Plant		To Controlled Environment	
Energy	1667 Btu	Energy	1000 Btu
Availability	1429 Btu	Availability	160 Btu
First Law Performance		1000/1667 = 0.60	
Second Law Performance		160/1429 = 0.11	

Electric Resistance Heating

To Power Plant		Losses at Power Plant	
Energy	2500 Btu	Energy	1500 Btu
Availability	2140 Btu	Availability	1140 Btu
To Controlled Environment			
Energy	1000 Btu		
Availability	160 Btu		
First Law Performance		1000/2500 = 0.40	
Second Law Performance		160/2140 = 0.075	

* Assumes Power plant efficiency equals 40% (8530 Btu heat rate)
Effective fuel temperature 3040°F (3500°R)
Reference temperature 40°F (500°R)
Air temperature to controlled environment 120°F (580°R)
Storage temperature 80°F (540°R)
Does not include mixing losses with room air

Table 3-16. First and Second Law Analysis Summary of Cooling Systems for 1000 Btu Output*

Basic Proposed Heat Pump Integrated Community Energy System

To Power Plant		Losses at Power Plant	
Energy	715 Btu	Energy	429 Btu
Availability	602 Btu	Availability	311 Btu
To Storage		From Controlled Environment	
Energy	1286 Btu	Energy	1000 Btu
Availability	70 Btu	Availability	73 Btu
First Law Performance		715/100 = 1.40	
Second Law Performance		73/602 = 0.12	

Standard Air-Source Heat Pump System

To Power Plant		Losses at Power Plant	
Energy	893 Btu	Energy	536 Btu
Availability	753 Btu	Availability	396 Btu
To Atmosphere		From Controlled Environment	
Energy	1357 Btu	Energy	1000 Btu
Availability	0 Btu	Availability	73 Btu
First Law Performance		1000/893 = 1.12	
Second Law Performance		73/753 = 0.097	

* Assumes
- Power plant efficiency equals 40% (8530 Btu Heat rate)
- Effective fuel temperature $3040^{\circ}F$ ($3500^{\circ}R$)
- Reference temperature $90^{\circ}F$ ($550^{\circ}R$)
- Temperature to controlled environment $50^{\circ}F$ ($510^{\circ}R$)
- Storage temperature $60^{\circ}F$ ($520^{\circ}R$)
- Does not include transfer and mixing losses with room air

The direct ("free") heating and cooling give high first law and second law performances; however, some of this must be charged to the opposite cycle. The results are very sensitive (especially second law) to output and ambient temperatures. The mixing of the output and controlled environment air reduces the second law performance by over a factor of 2.

IV. EXPECTED ECONOMICS

Evaluating the expected economics of a system in its conceptual design stage leaves numerous uncertainties. Fortunately, the proposed heat pump centered integrated community energy system uses existing hardware components with known operating and economic characteristics. Their performance as an integrated system has not been validated, yet looks technically feasible. Economically any new system must be expected to incur a front-end developmental cost. Following a typical learning curve, these costs can be expected to reduce as system design and management experience grows. The analysis presented here does not consider such front-end design costs, but rather looks at potential systems operating after suitable experience has been gained. The base case analysis further assumes the existing component performances to be unchanged.

Rapidly increasing energy costs have presented a problem in the analysis. First, energy costs vary geographically due to transportation, marketing and allocation differences. Second, the cost of liquid fuels has risen rapidly, in some cases doubling over the span of this research effort. Thus, energy costs have been left as a variable in the analysis, allowing the readers to draw their own conclusions about the expected economics of the proposed system.

This section deals in depth with the expected economics of the proposed system for new residential application. Also considered are residential retrofit and commercial applications and the organizational structure of the water supply and distribution subsystems.

IV.A Residential Applications, New Construction

New residential construction on a subdivision or community basis represents an attractive setting for the proposed system. The layout of the community can be designed with consideration given to potential well sites, the water distribution subsystem configuration and solar heat collection locations. The proposed system may eliminate the need for a natural gas distribution system and may serve as the fire protection water source, thus reducing the required size of the potable water mains. While these factors seem to favor the proposed system in new residential construction, they are difficult to quantify and are likely to vary in impact from community to community. In addition, the proposed system is likely to favor all-electric communities which in turn will impact the demand pattern for electricity. The impact of the proposed system on the electric power generating system has not been modeled in detail.

Analysis

A comparative approach has been taken in the expected economic analysis of the alternative systems for new residential heating and cooling. A total annual cost has been computed for a single home using each alternative system in each of five cities selected to demonstrate a range of heating and cooling loads. The total annual cost provides for capital recovery, system maintenance and energy costs. Because of the uncertainty associated with future energy costs, much of the analysis is based on a range of possible energy costs. Parametric analysis of other critical assumptions are also included.

Table 4-1 illustrates the development of the alternative systems, capital and maintenance costs. The capital costs for the installed heating and cooling units are from Christian.[12,13] The installed cost of the modified water source heat pump includes an additional $300 for the added controls and the pre-water heat exchangers.

The capital cost for the installed water supply and distribution subsystems used by the proposed water source heat pump system is developed in

Table 4-1. Annualized Capital and Maintenance Cost for Alternative Systems

System Alternatives	Capital Cost: Home Unit	Capital Cost: Water Supply & Distribution Subsystems[1]	Capital Cost: Total	Annual Equivalent Cost[2]	Annual Cost Maintenance[5]
Gas or Oil Furnace and Air Conditioner	$2,500	$ 0	$2,500	$294	$38
Resistance Heat and Air Conditioner	2,500	0	2,500	294	38
Air Source Heat Pump A (High Efficiency)	2,800	0	2,800	329	42
Air Source Heat Pump C (Moderate Efficiency)	2,500	0	2,500	294	38
Water Source Heat Pump A[3]	2,000	1,391	3,391	398	51
Modified Water Source Heat Pump A[3,4]	2,300	1,391	3,691	434	55

1. Assume a 20-home module size, total capital cost $27,820.
2. Based on a 20-year economic life for the entire system, no salvage value, no tax considerations, 10% interest rate using capital recovery factor = 0.11746.
3. 3-ton unit.
4. Includes a pre-water heat exchanger used for direct heat (cold climates) or direct heating (warm climates).
5. Annual Maintenance Cost = 1.5 % Total Capital Cost

Table 4-2. The water supply subsystem consists of a pair of wells, pumps and capacitance tanks with the necessary controls and solar collector for additional heat makeup. The pumps are sized to service a 20-home module using an aquifer with a 50 or more ft. of porous and highly permeable water-bearing section within 200 ft. of the surface. A less favorable aquifer could significantly increase the cost of the water supply subsystem.

The water distribution subsystem costs are based on the 20-home module. The primary water distribution is handled by a pair of 6-inch PVC pipes laid in a single trench parallel to the residential street. An average home spacing (lot size) of 100 ft. is assumed. With homes on both sides of the street, a 20-home module requires 1000 ft. of primary distribution. The secondary water distribution, from the primary to the home water source heat pump, is comprised of a pair of 1.5-inch PVC pipes. Assuming an average home setback of 50 ft. from the street center line a total of 1000 ft. of secondary distribution is required per 20-home module. A total of 2000 ft. of trenching is required per module. Because this is a new residential community, no disruption of streets or sidewalks is caused by the installation of the water distribution subsystem. In fact, the same trench could be used for the potable water, sewer, electric and telephone utilities. The total 20-home module water supply and distribution subsytem cost of $27,820 is divided equally, yielding a per home capital cost of $1,391.

The capital recovery method is used to compute the annual equivalent capital cost. The capital recovery factor, CRF, assures the recovery of the initial investment with interest over the economic life of the system. Assuming a 10 percent interest rate with a 20-year economic life, a capital recovery factor of 0.11746 is obtained using the expression

$$\mathrm{CRF} = \frac{i\,(1+i)^n}{(1+i)^n - 1}$$

Table 4-2. Water Supply and Distribution Subsystems Capital Cost

Water Supply System		
Well Construction (2 wells, less than 200 ft.)	$3,000	
Pumping System (pump, motor and controls)	$6,000	
Capacitance Tank (2 tanks)	$ 260	
Solar Collector (1000 $ft.^2$ at $3/$ft.^2$)	$3,000	$12,260
Water Distribution System[1]		
Double 6-inch, Primary (1000 ft. at $9.20/ft.)	$9,200	
Double 1.5-inch, Secondary (1000 ft. at $3.40/ft.)	$3,400	
Trench (2000 ft. at $1.48/ft.)	$2,960	$15,560
Total Water System Capital Cost		$27,820

1. Based on 100 ft. average home spacing on both sides of the street with a 50 ft. setback from street center line.

where i represents the annual interest rate and n the economic life in years. The annual system maintenance costs are assumed to be 1.5% of the original capital costs per Bezdek, et al.[14]

Table 4-1 illustrates the relatively high capital cost of the proposed water source heat pump system. While the water source heat pump is the least expensive home unit, the water supply and distribution subsystems contribute a substantial proportion of the total system cost and the total system cost is greatest. For the proposed system to be economically justified, it must offer sufficient savings in energy costs to afford the recovery of the high initial capital and annual maintenance costs. The energy saving potential of the proposed system was demonstrated in Section III. Tables 3-10 and 3-11 are combined in Table 4-3 to display the annual heating and cooling energy consumption for the alternative systems. The energy consumption data for the water source heat pump systems include the energy consumed by the water supply and distribution subsystems.

To construct a base case for analysis the cost of electricity and fuel is assumed to be \$0.04 per kw-hr and \$2.50 per million Btu (equivalent to \$0.35 per gallon of fuel oil). Table 4-4 illustrates the total annual cost computed for the alternative systems in each of the five cities assuming the base case energy costs.

These costs include: energy, capital recovery and maintenance. The best total cost system is without exception the modified water source heat pump with its best alternative (II for Miami with 40° F water for direct cooling and III for other cities with 100°F water for direct heating). The total annual cost of the modified water source heat pump system ranges from 13 to 30 percent lower than the next best alternative. Table 4-4 clearly demonstrates the economic potential of the modified water source heat pump system. However, for the next several analysis steps only the proposed water source heat pump system, not the modified system will be considered.

Table 4-3. Annual Heating and Cooling Energy Consumption for Alternative Systems

	BIRMINGHAM	DETROIT	MIAMI	NEW YORK	ST. LOUIS
Gas or Oil Heating/Air Source Heat Pump C Cooling					
Electric (10^3 Kw-hr.)	7.90	3.48	14.6	3.49	6.45
Fuel (10^6 Btu)	119	205	5.78	156	155
Resistance Heating/Air Source Heat Pump C Cooling					
Electric (10^3 Kw-hr.)	29.50	40.78	15.65	31.79	34.55
Air Source Heat Pump A (High Efficiency)					
Electric (10^3 Kw-hr.)	14.69	17.85	11.17	14.16	16.33
Air Source Heat Pump C (Moderate Efficiency)					
Electric (10^3 Kw-hr.)	18.70	22.08	15.12	17.69	20.55
Water Source Heat Pump A (Includes Water Supply and Distribution Subsystems)					
Electric (10^3 Kw-hr.)	11.66	14.07	9.01	11.22	12.89
Modified Water Source Heat Pump A - Alternative I (Includes Water Supply and Distribution Subsystems)					
Electric (10^3 Kw-hr.)	8.86	10.98	6.46	8.69	9.89
Modified Water Source Heat Pump A - Alternative II (Includes Water Supply and Distribution Subsystems)					
Electric (10^3 Kw-hr.)	8.37	13.29	1.98[1]	10.21	10.43
Modified Water Source Heat Pump A - Alternative III (Includes Water Supply and Distribution Subsystems)					
Electric (10^3 Kw-hr.)	7.80	5.61	11.29	4.92	7.19

[1] Best Modified Water Source Heat Pump Alternative underscored and used in subsequent analysis.

Table 4-4. Annual Costs for Alternative Systems, Base Case

BIRMINGHAM	DETROIT	MIAMI	NEW YORK	ST. LOUIS
Gas or Oil Heating/Air Source Heat Pump C Cooling				
$ 945	$ 983	$930	$ 861	$ 977
Resistance Heating/Air Source Heat Pump C Cooling				
$1,512	$1,963	$958	$1,603	$1,714
Air Source Heat Pump A (High Efficiency)				
$ 958	$1,085	$818	$ 937	$1,024
Air Source Heat Pump C (Moderate Efficiency)				
$1,080	$1,215	$936	$1,039	$1,154
Water Source Heat Pump A (Includes Water Supply and Distribution Subsystems)				
$ 916	$1,012	$810	$ 898	$ 965
Modified Water Source Heat Pump A Best Alternative (Includes Water Distribution System)				
$ 801	$ 713	$568	$ 685	$ 776

Base Case Assumptions:

Electricity cost $0.04/Kw-hr.

Fuel cost $2.50/10^6 Btu

20-home Module

20-year System Economic Life

10% interest rate

Perhaps the best way to further consider Table 4-4 is to next eliminate the resistance heating alternative system and the moderate efficiency air source heat pump alternative system due to consistently greater total annual cost than the remaining alternatives. The three remaining alternatives share a relatively close position in terms of total annual cost. In Miami, which has the greatest cooling load, the alternative systems rank.

Alternative System	Total Annual Cost	Percent of Best
Water Source Heat Pump	$810	100
Air Source Heat Pump	$818	101
Fuel Furnace/AC	$930	115

No significant cost difference exists between the proposed water source heat pump system and the high efficiency air source heat pump in Miami. In Birmingham and St. Louis, which have moderately high cooling loads, the alternative systems rank.

Alternative System	Total Annual Cost	Percent of Best
Water Source Heat Pump	$916-$965	100-100
Fuel Furnace/AC	$945-$977	103-101
Air Source Heat Pump	$958-$1,024	105-106

In Detroit and New York, which have relatively low cooling but high heating loads, the alternative systems rank.

Alternative System	Total Annual Cost	Percent of Best
Fuel Furnace/AC	$983-$861	100-100
Water Source Heat Pump	$1,012-$898	103-104
Air Source Heat Pump	$1,085-$937	110-109

Thus, at the base case energy costs the proposed system ranked best in total annual cost in Miami, Birmingham and St. Louis and within four percent of the best in Detroit and New York.

Because the base case energy costs are relatively low and because those costs are rapidly increasing it is of interest to study the impact of energy costs on the alternative systems. Certainly in many regions of the U.S. electricity is not available at \$0.04/Kw-hr. Liquid fuel prices, in particular fuel oil, have rapidly increased in price over the first half of 1979. Due to price deregulation and demand trends, it seems likely that over the next decade liquid fuel prices will increase at a rate greater than the general cost of living, i.e., the capital and maintenance costs. The rise in the cost of electricity is expected to be less rapid due to its in-place capital intensive generation and distribution systems and its utilization of several energy sources, including abundant coal.

To investigate the impact of rising energy costs Table 4-5 was constructed assuming a doubling of energy cost from the base case: the electricity cost from \$0.04 to \$0.08/Kw-hr. and fuel cost from \$2.50 to \$5.00/10^6 Btu. The capital recovery and maintenance costs were assumed to remain fixed at the levels indicated in Table 4-1. Again the modified water source heat pump system is the least cost alternative. Among the other systems, the proposed water source heat pump system consistently ranks first in lowest total annual cost with a 3 to 12 percent margin over the second best alternative (high efficiency air source heat pump in Birmingham and Miami and the fuel furnace with AC in the other cities). The total annual cost of the proposed system holds a 35 to 128 percent margin over the resistance heating with AC system, the most costly alternative.

Energy Cost Sensitivity Analysis

Energy cost is the key variable in the expected economics of the alternative systems. At base case energy costs the proposed system is least expensive except

Table 4-5. Annual Costs for Alternative Systems, Doubled Energy Cost
Electricity \$0.08/Kw-hr. - Fuel \$5.00/10^6 Btu

BIRMINGHAM	DETROIT	MIAMI	NEW YORK	ST. LOUIS
Gas or Oil Heating/Air Source Heat Pump C Cooling				
\$ 1,559	\$ 1,635	\$1,529	\$ 1,391	\$ 1,623
Resistance Heating/Air Source Heat Pump C Cooling				
\$ 2,692	\$ 3,594	\$1,584	\$ 2,875	\$ 3,096
Air Source Heat Pump A (High Efficiency)				
\$ 1,546	\$ 1,799	\$1,264	\$ 1,504	\$ 1,677
Air Source Heat Pump C (Moderate Efficiency)				
\$ 1,828	\$ 2,098	\$1,541	\$ 1,747	\$ 1,976
Water Source Heat Pump A (Includes Water Supply and Distribution Subsystems)				
\$ 1,382	\$ 1,575	\$1,170	\$ 1,347	\$ 1,481
Modified Water Source Heat Pump A Best Alternative (Includes Water Supply and Distribution Subsystems)				
\$ 1,113	\$ 937	\$ 647	\$ 882	\$ 1,064

in Detroit and New York where the oil or gas furnace with air source heat pump for cooling has the lowest total annual cost. The actual cost difference between the three top alternatives is small, at most 15 percent. With a doubling of energy costs, the superiority of the proposed system becomes clearly evident. The impact of increasing energy costs on the relative economics of alternative systems can be demonstrated graphically. To make such a graphical comparison, the total annual cost of two alternative systems is computed as a function of the energy cost. The ratio of these costs is calculated and plotted versus the energy cost. A single plot is sufficient for all alternative system comparisons except the gas or oil furnace with air conditioning which consumes both gas or oil as a fuel and electricity.

Figure 4-1 illustrates the impact of variable electricity cost on the relative cost of the gas or oil furnace with a moderate efficiency air source heat pump for cooling compared with the proposed water source heat pump system. For this analysis fuel costs are fixed at $2.50 per million Btu. The cost of electricity is varied from 0 to 20 cents per Kw-hr. The curves plot the ratio of the total annual cost of the alternative system to the total annual cost of the proposed system for each of the five cities. Where the curves lie above the break-even line (ratio = 1.00), the proposed system is least costly. On the other hand, where the curves lie below the break-even line, the alternative is superior. Two distinct patterns appear in Figure 4-1. In Miami, increasing electricity cost would favor the proposed system. In the more northerly cities, with their greater heating loads, the proposed system has the cost advanatge only with relatively low cost electricity.

It is of interest to note that the curves for all non-Miami cities cross break-even line at an electricity cost of about 3-5 cents/Kw-hr. This break-even cost is about the current minimum cost at which domestic electricity can be

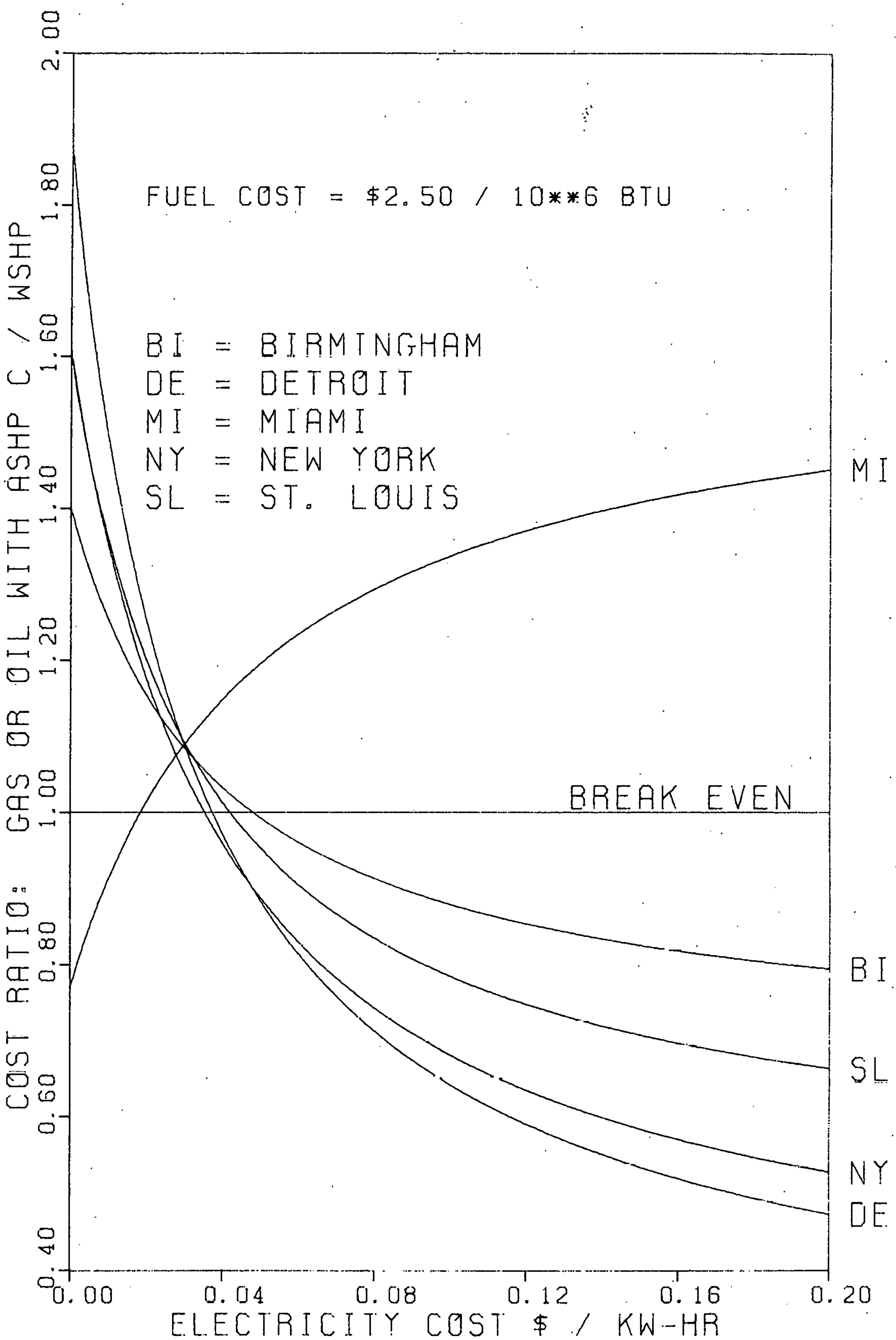


FIGURE 4-1 VARIABLE ELECTRIC COST -
GAS OR OIL WITH ASHP C / WSHP

obtained. Thus, with the exception of Miami, the proposed system is about par with the gas or oil furnace system with a fuel cost of $2.50/million Btu and 4 cent/Kw-hr. electricity. The proposed system loses its advantage if electricity costs rise relative to fuel costs. This event seems unlikely.

Figure 4-2 presents the same systems cost ratio with a fixed electricity cost ($0.04/Kw-hr.) and a variable fuel cost ($0 to $10/million Btu). Again two patterns exist. For Miami, the proposed system retains a 13 to 20 percent cost advantage regardless of the fuel cost. In the remaining cities the break-even point occurs at about $2.50/million Btu. The cost advantage of the proposed system becomes very significant as the cost of fuel increases. With $10/million Btu fuel and $0.04 electricity, the proposed system in Detroit would have 2.49 ($2521/$1022) cost advantage over the alternative system. The cost advantage is less extreme in New York (2.26), St. Louis($2.22) and Birmingham (2.01). Nonetheless, Figure 4-2 clearly demonstrates that increasing fuel costs could swing the economic advantage to the proposed system. However, it is unlikely that fuel costs will rise as dramatically as indicated while electricity costs remain fixed at $0.04/Kw-hr. The joint impact of variable electricity and fuel costs is demonstrated for the cities of Detroit and Birmingham in Figures 4-3 and 4-4. In these two figures, each curve represents a different electricity cost (4, 8, 12, 16 and 20 cents/Kw-hr.). In general, the curves follow the pattern of Figure 4-2, indeed, the curves for 4 cent electricity are identical.

As expected from Figure 4-1, with increasing electricity costs the proposed system's advantage decreases. The break-even fuel cost ($/million Btu) for the various city/electricity cost combinations are:

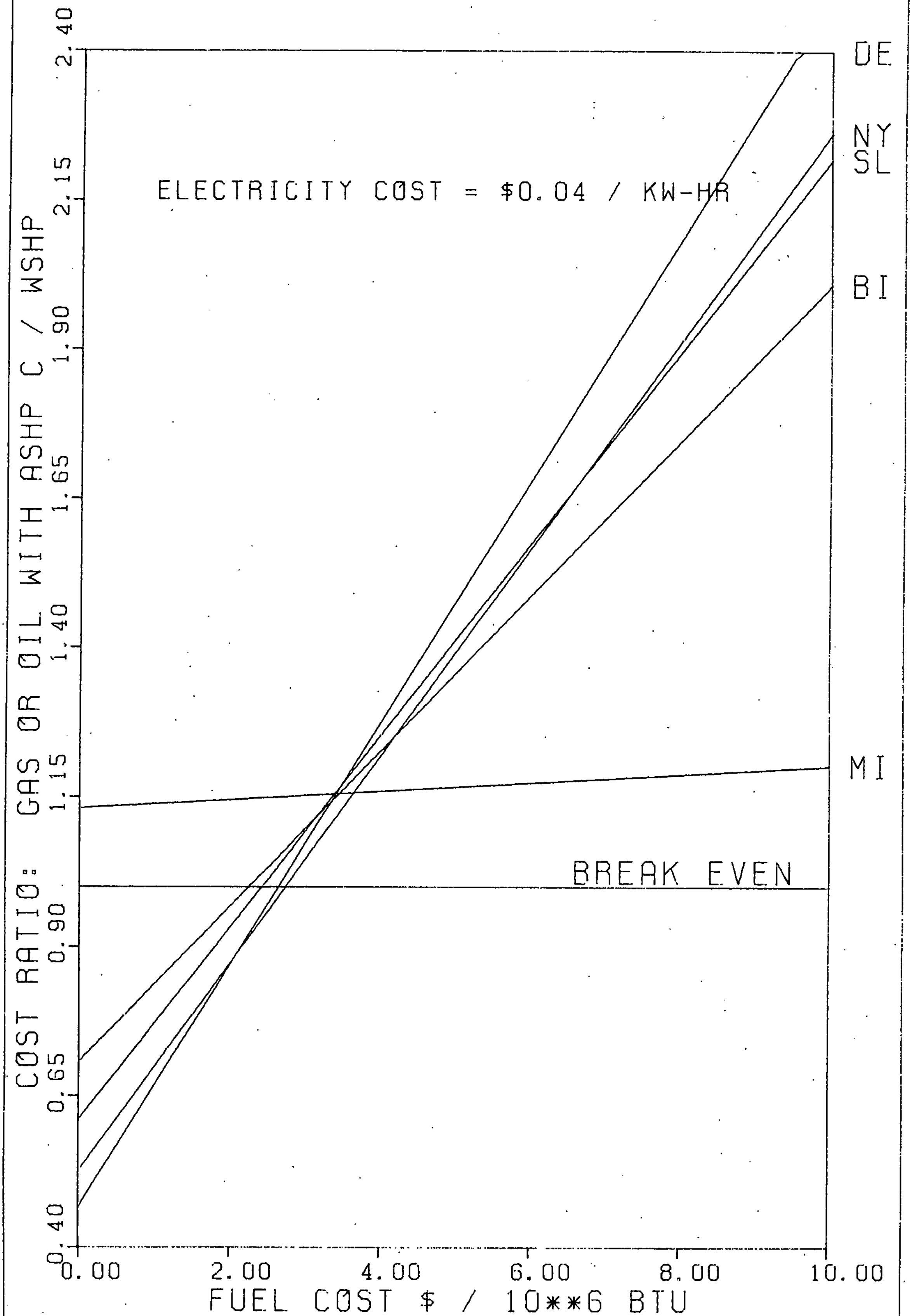


FIGURE 4-2 VARIABLE FUEL COST -
GAS OR OIL WITH ASHP C / WSHP

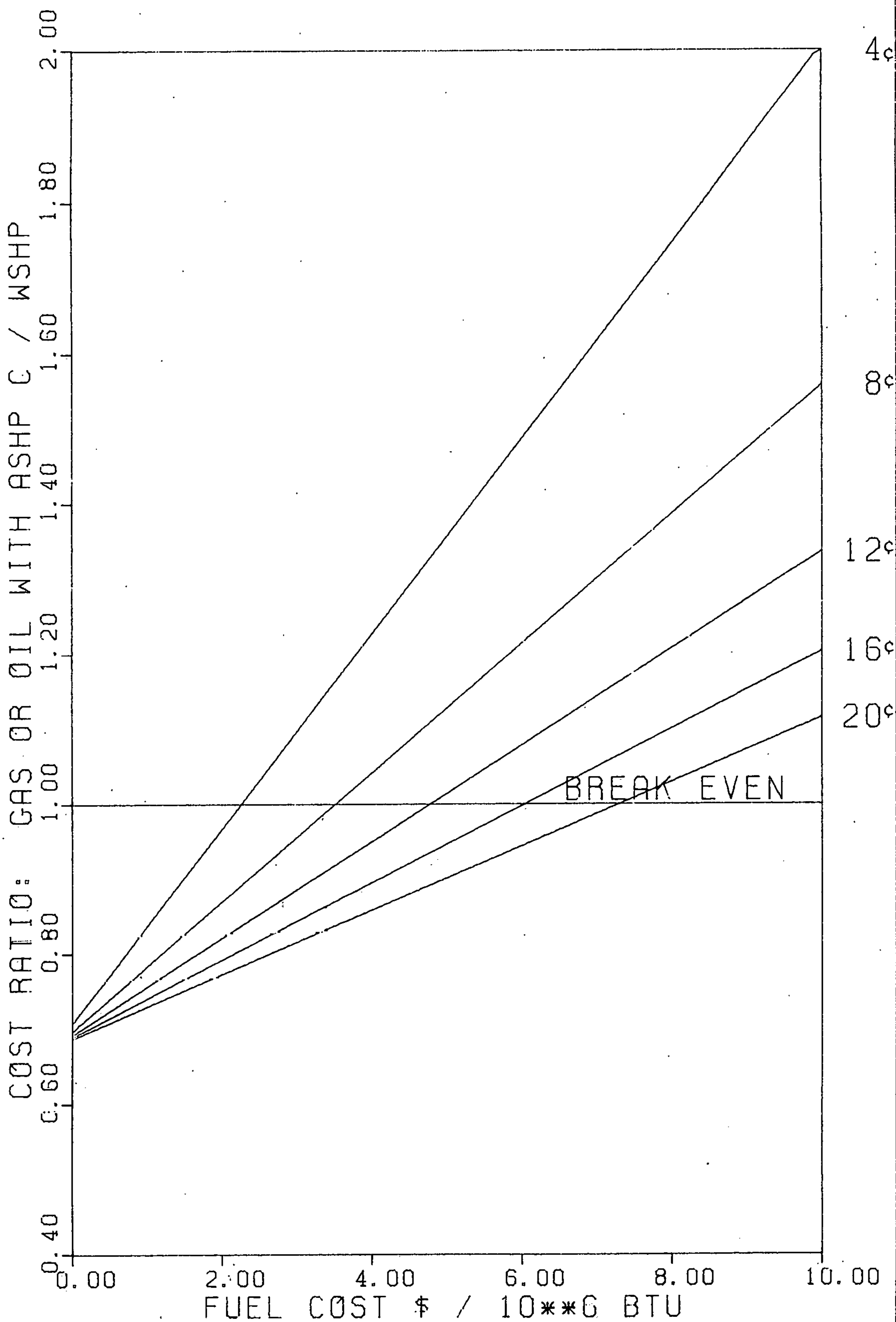


FIGURE 4-3 VARIABLE FUEL COST - BIRMINGHAM - GAS OR OIL WITH ASHP C / WSHP

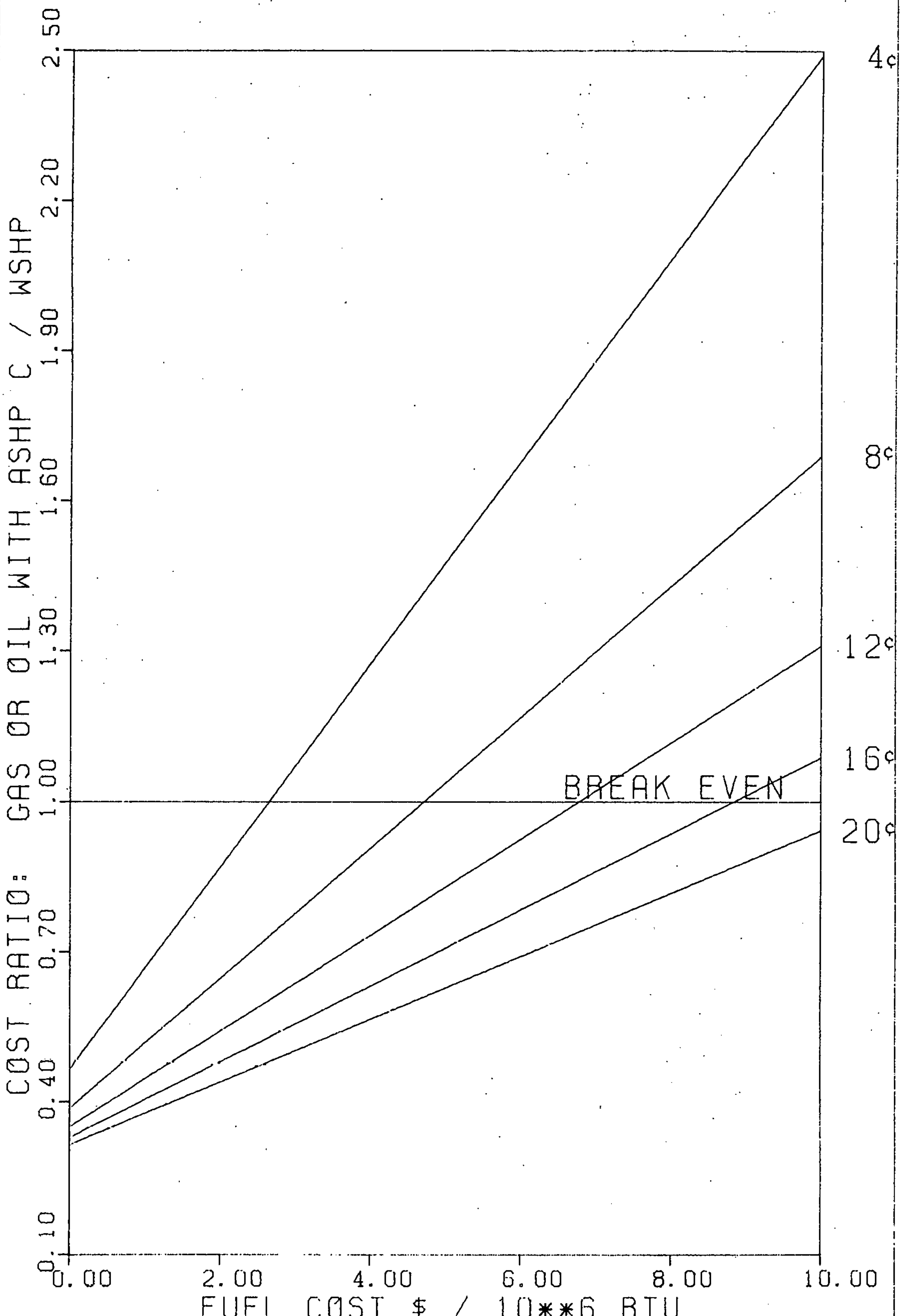


FIGURE 4-4 VARIABLE FUEL COST - DETROIT -
GAS OR OIL WITH ASHP C / WSHP

City	Electricity Cost (¢/Kw-hr.)				
	4	8	12	16	20
Birmingham	2.3	3.5	4.8	6.0	7.4
Detroit	2.7	4.8	6.8	8.9	>10
New York	2.7				
St. Louis	2.4				

Note that under conditions of very high electricity costs, the proposed system cost loses it advantages to the conventional gas or oil furnace system. It should be pointed out that if energy costs rise to such a level, the other system costs might also increase and invalidate the results of this analysis.

The comparison of the proposed system to the other system alternatives is more straight forward because electricity is the only energy source. Figure 4-5 compares the proposed system with an alternative system including resistance heating and air conditioning utilizing the moderate efficiency air source heat pump. The proposed system is consistently the most economical whenever electricity costs are above break-even values of 0.6 and 1.9 cents/Kw-hr. for the non-Miami cities and Miami respectively. The economic advantage of the proposed system increases dramatically with electrictiy costs. At 20 cents/ Kw-hr., the cost ratio ranges from 1.54 ($3462/$2251) in Miami to 2.60 ($8488/ $3263) in Detroit. Clearly the proposed system is economically superior to the resistance heating alternative.

The proposed system is compared to a system alternative using a high-efficiency air source heat pump in Figure 4-6. The proposed system holds the cost advantage for electricity cost above 3.7 cents/Kw-hr. Notice that the proposed system gains its greatest economic advantage in the colder cities (higher heating loads) as electricity costs increase. In no case does the proposed system have a cost advantage ratio greater than 1.20 ($3941/$3263). Notice the relative flatness of the curves above electricity costs of 10 cents/ Kw-hr.

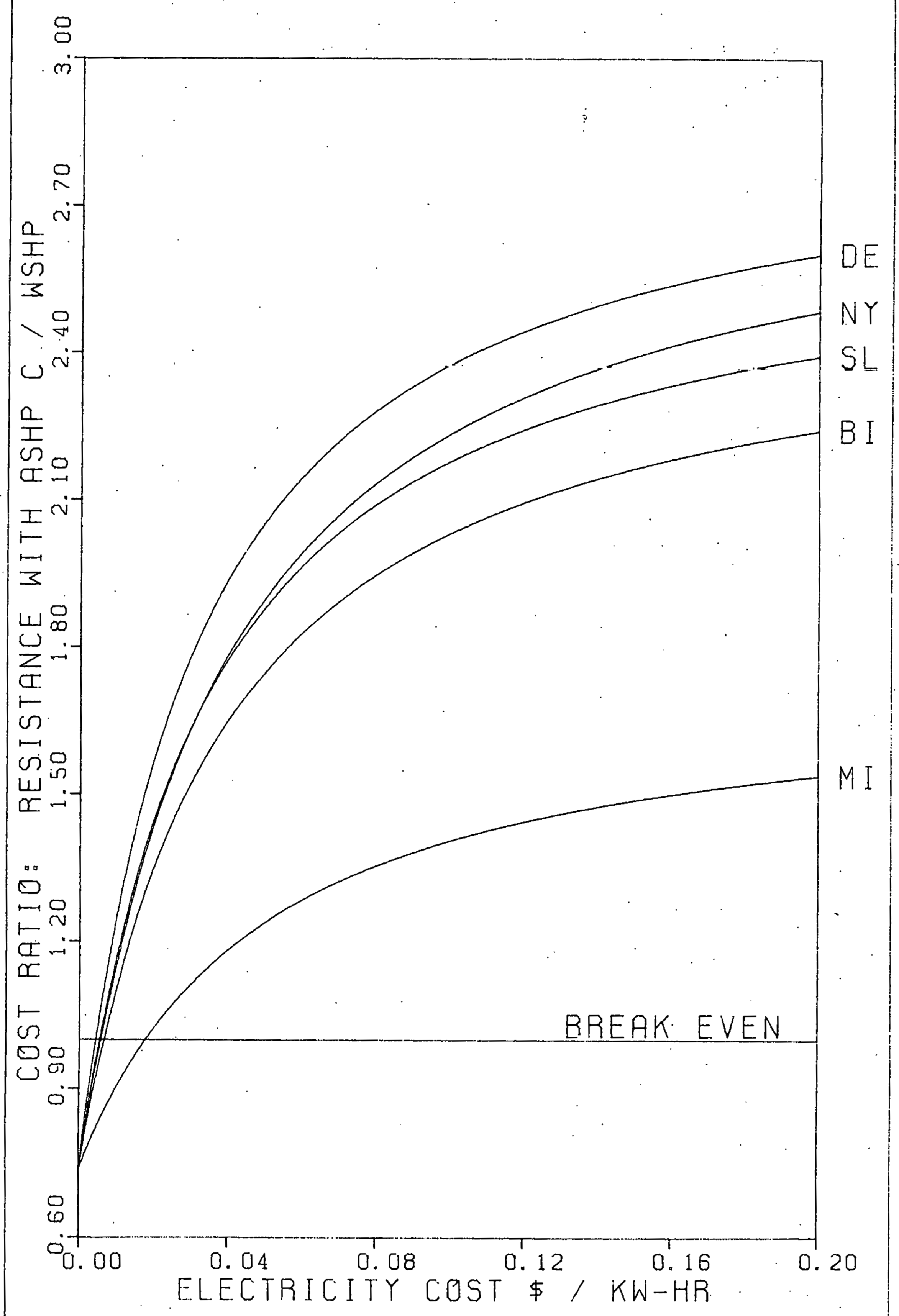


FIGURE 4-5 VARIABLE ELECTRIC COST - RESISTANCE WITH ASHP C / WSHP

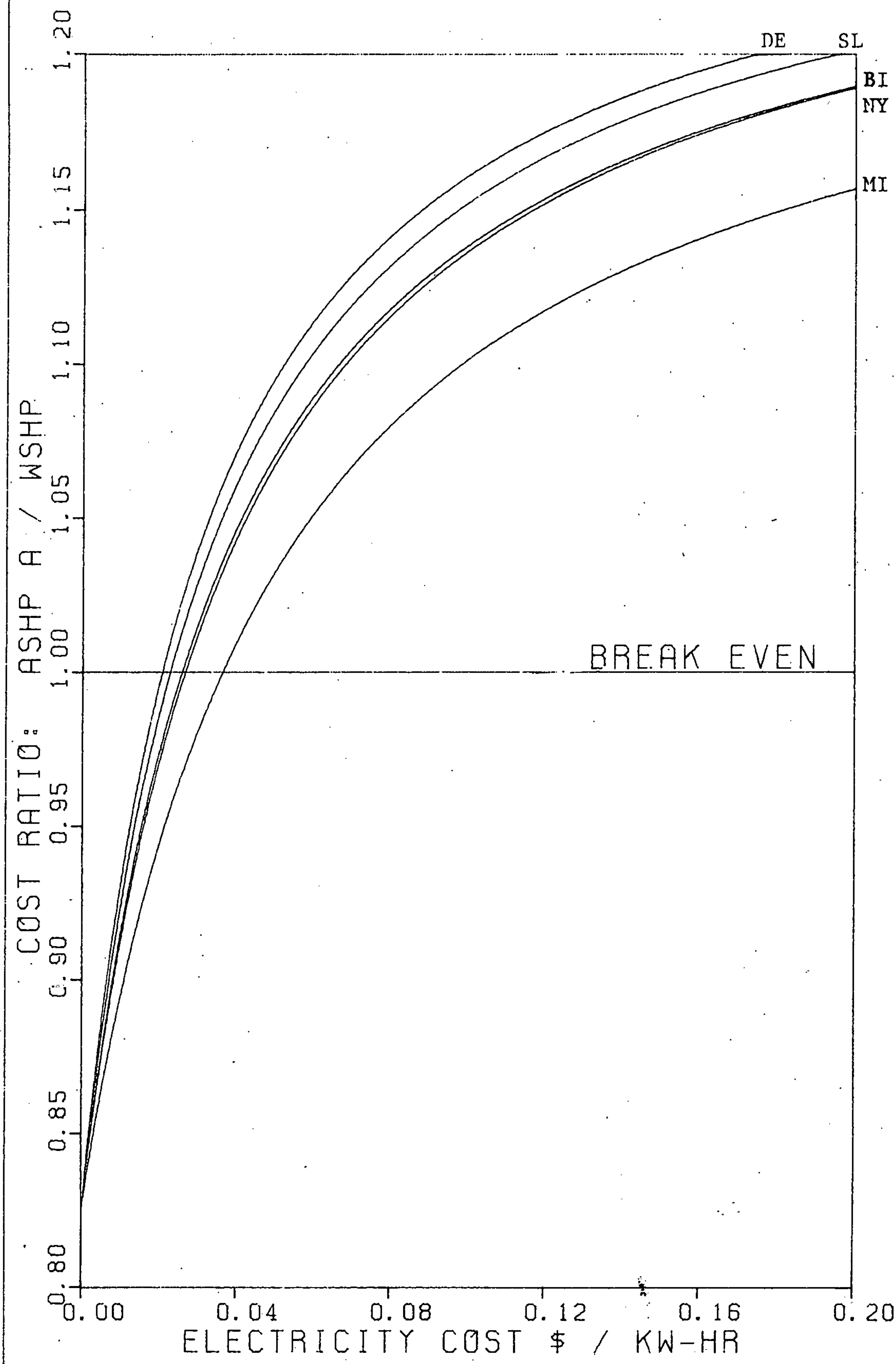


FIGURE 4-6 VARIABLE ELECTRIC COST - ASHP A (HIGH EFFICIENCY) / WSHP

Figure 4-7 compares the proposed system to the moderate efficiency air source heat pump. In this case, the proposed system is much more economical than the air source heat pump. Clearly the moderate efficiency air source heat pump should not be recommended for domestic heating and cooling. In all circumstances other than very inexpensive electricity (less than 2.0 cents/Kw-hr.), either the proposed system or the high efficiency air source heat pump is economically superior.

The final graph in this series, Figure 4-8, compares the proposed system with a similar system involving a water source heat pump modified with a pre-water heat exchanger for either direct heating or cooling. For this analysis the most favorable modification alternative (see Table 4-3) was selected for each city: Alternative II for Miami and Alternative III for the non-Miami cities. Figure 4-8 clearly indicates the economic superiority of the modified system with electrictiy costs above a breakeven point of about 1 cents/Kw hr. Increasing electricity costs to 10 cents/Kw-hr. gives the modified system cost advantages (inverses of the plotted values) of 1.27, 1.44, 1.60, 1.77 and 1.97 for Birmingham, St. Louis, New York, Detroit and Miami respectively. The advantage of the modified system continues to grow, but less rapidly, as electricity costs increase beyond the 10 cents/Kw-hr. level. Figure 4-8 dramatically demonstrates the potential of the properly selected modified water source heat pump system. If the previous analyses were repeated using the modified water source heat pump, the economic advantage of the proposed system would be significantly increased in all cases.

Basic Assumption Sensitivity Analysis

The graphical analysis technique used to compare the alternative systems under conditions of variable energy costs can be used to analyze the cost

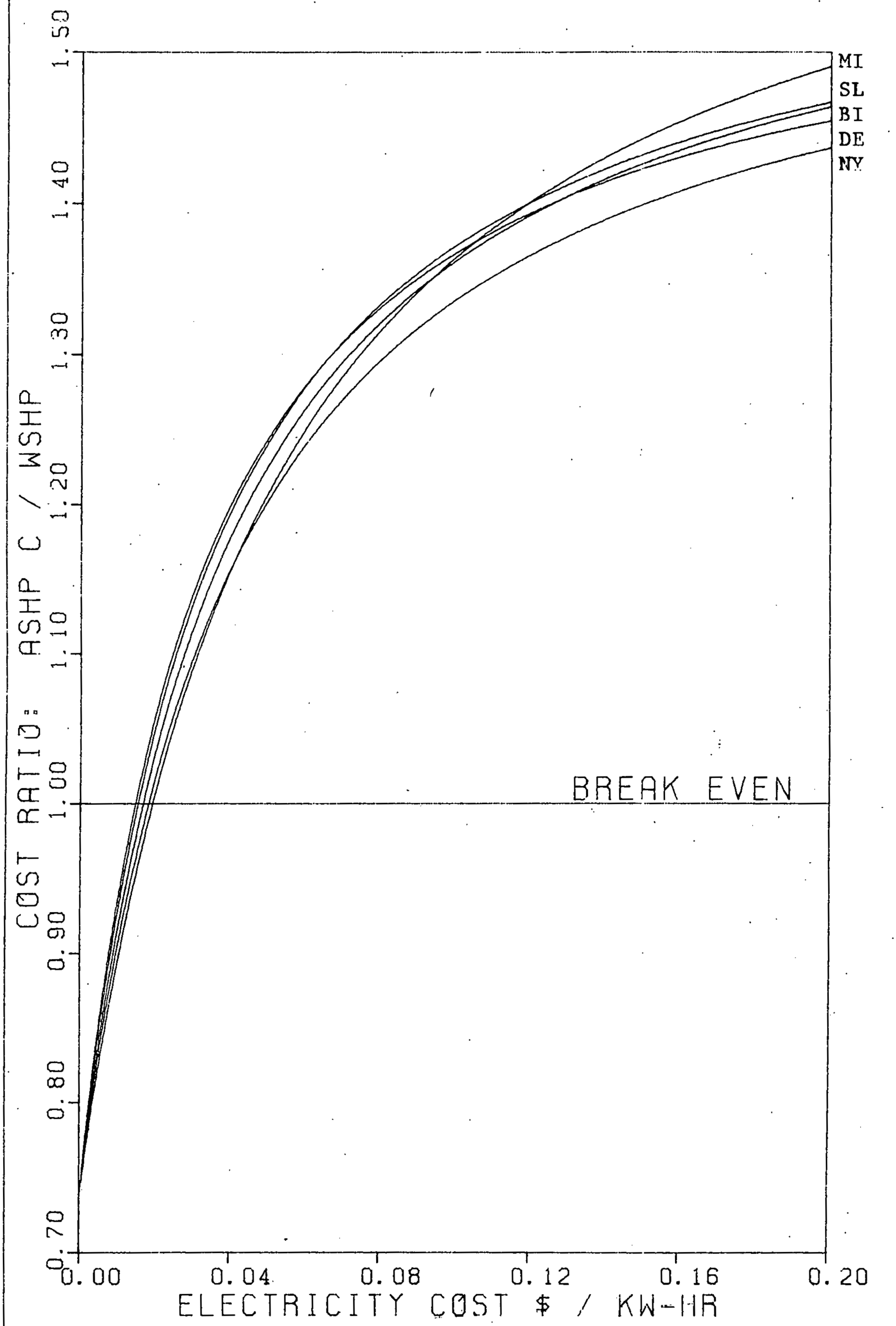


FIGURE 4-7 VARIABLE ELECTRIC COST - ASHP C (MODERATE EFFICIENCY) / WSHP

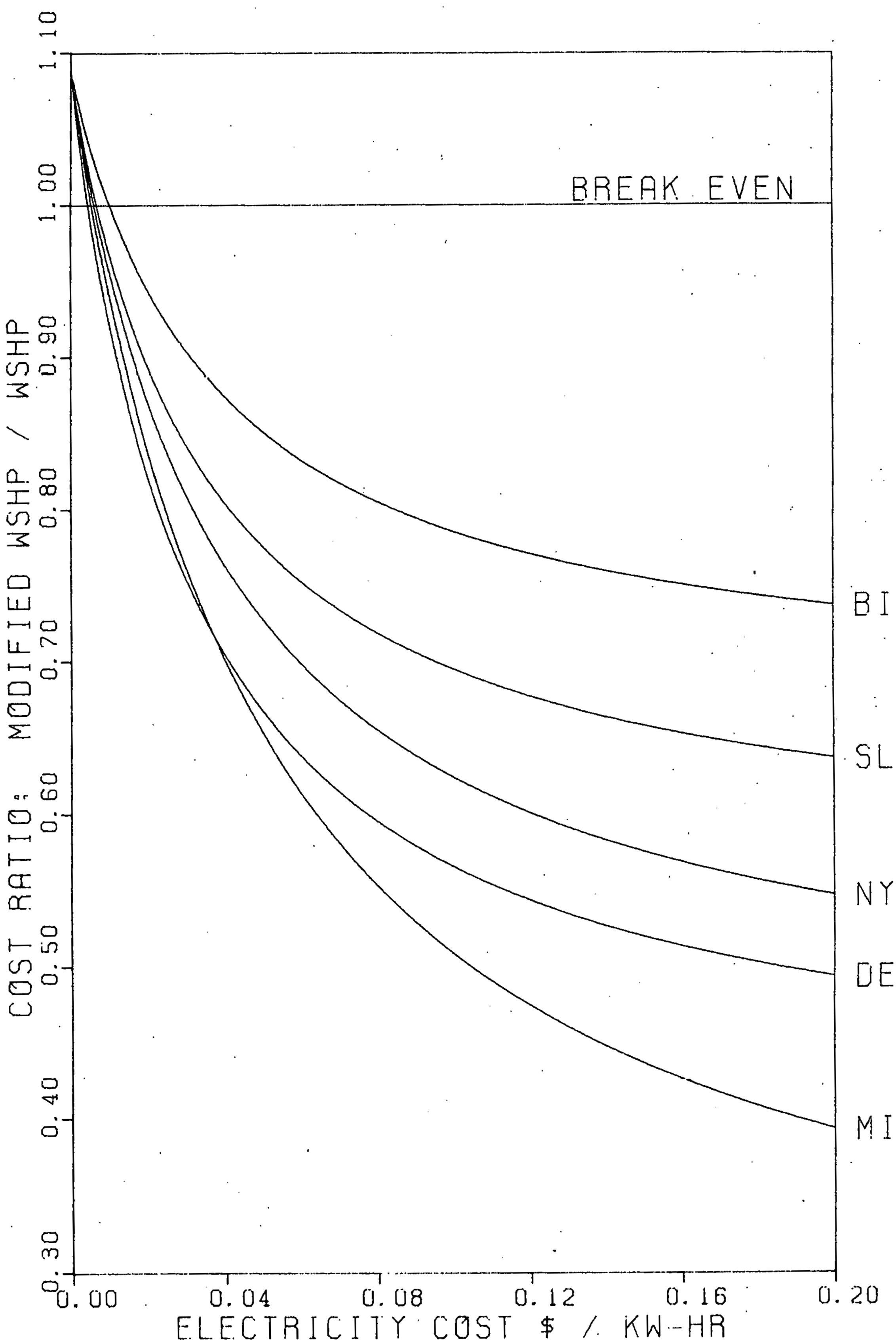


FIGURE 4-8 VARIABLE ELECTRIC COST - MODIFIED WSHP / WSHP

impact of several of the base case system modeling assumptions:

the number of units per module (base case-20);

the unit spacing (base case-100 ft.);

the interest rate (base case-10%); and the

system's economic life (base case-20 years).

For this analysis, the base case of the proposed system is used as the cost ratio denominator. The total annual cost of the proposed system generated by the variable parameter is used as the cost ratio numerator. Thus, curves above the breakeven line indicate a more expensive system than that used in the previous section's analysis. The interest here is to determine the magnitude of the cost impact a change in the base case assumptions has on the economic viability of the proposed system.

First consider the number of homes or units per module. The water supply and distribution subsystems are designed to accommodate a 20-home module. Obviously, these subsystems could handle a module with fewer homes and might be able to accommodate a few additional homes. The results of this analysis are plotted on Figure 4-9. Because the base case (cost ratio denominator) assumed a 20-home module size, all curves cross the breakeven line at 20. A larger module size spreads the cost of the water supply and distribution subsystems over more units and lowers the annual equivalent cost, i.e., the cost ratio is less than one. Small modules understandably cost more because fewer homes share those costs. Notice that the curves for the five cities are almost identical to one another. As a matter of identification, at a 30-home module size the cost ratios are: 0.976, 0.975, 0.974, 0.973 and 0.970 for Detroit, St. Louis, Birmingham, New York and Miami respectively. Two conclusions can be drawn from Figure 4-9. The cost impact of modules smaller than 20 homes is rather dramatic. A 10-home module would be 7 to 9 percent more expensive

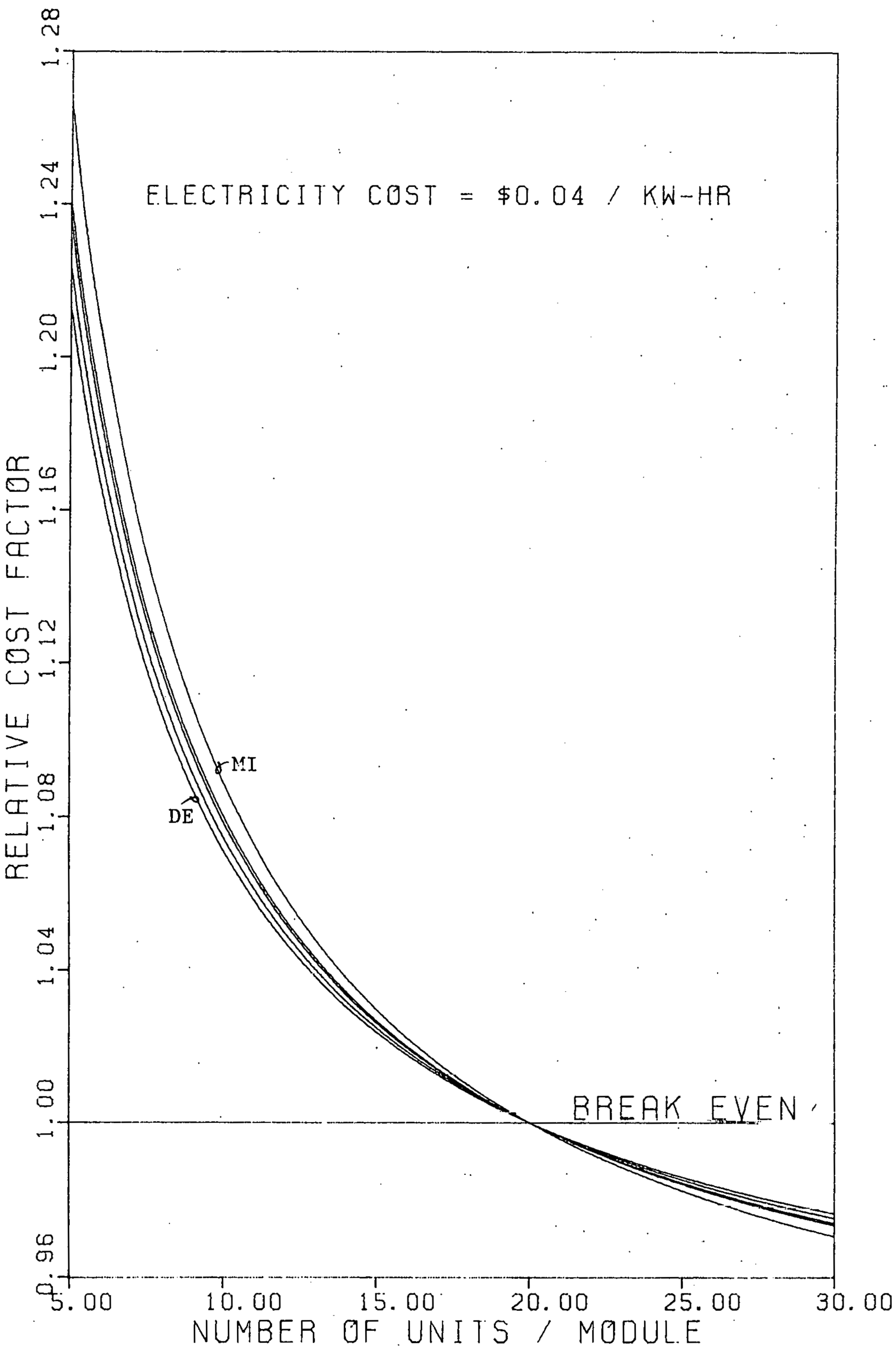


FIGURE 4-9 VARIABLE NUMBER OF UNITS PER MODULE VERSUS BASE CASE (20 UNITS)

than the base case, a 5-home module 21 to 26 percent more expensive. Module sizes less than 15-units appear economically unjustifiable. On the other hand, increasing the module size over 20 homes has a relatively small economic impact. In fact, the curves rapidly become flat. Thus, even if the water supply and distribution subsystems could physically handle 30 or more units, the economic impact would be marginal.

Figure 4-10 illustrates the impact of variable unit spacing. The base case assumed a spacing of 100 ft. with homes on both sides of the street. The unit spacing is likely to vary widely depending on the type of residential community. A garden apartment or townhouse community might have an effective unit spacing of 25 feet or less. On the other hand, single family homes on large lots, or on only one side of the street, or where all homes in the community do not utilize the system could easily increase the effective unit spacing to several hundred feet. Again the curves for the various cities are quite close to one another. Note the curves are linear, a more accurate analysis which included the additional energy consumed by the enlarged water distribution system would shift the curves slightly concave upward. The curves indicate that reducing unit spacing to 25 feet would reduce the total annual cost by about 5%. Increasing unit spacing to 500 feet would increase the total annual cost by 25 to 30 percent. Such a large cost increase may make the proposed system economically unfeasible in semi-rural residential communities or where less than 100 percent saturation must be expected.

Most energy related economic analyses have used an interest rate of 10% for the capital recovery calculation. Traditionally, this rate has been slightly above the first mortagage rate for private residences and about equal to the second mortgage rate for secured home improvements. The current double-digit inflation rate and the record Federal Reserve discount rates may at least

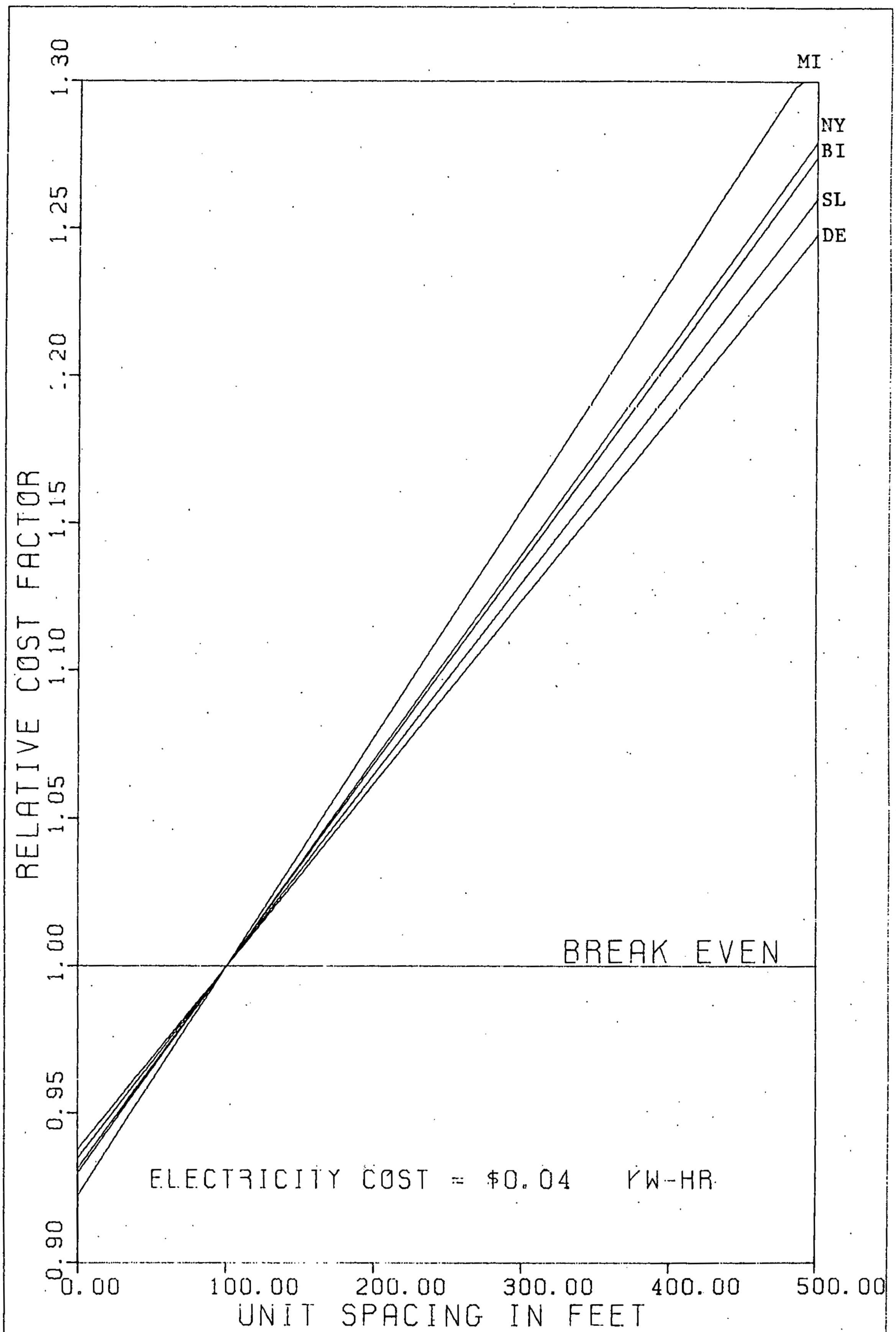


FIGURE 4-10 VARIABLE UNIT SPACING VERSUS BASE CASE (100 FOOT SPACING)

temporarily make the base case interest rate of 10% obsolete. Figure 4-11 graphically considers the impact of a variable interest rate. Because the proposed system involves a large capital investment, the interest rate does have significant impact. Even raising the interest rate to 20 percent (not out of line with current second mortgage rates) would increase the annual cost about as much as increasing the unit spacing to 500 ft. (Figure 4-10). Decreasing the interest rate to zero percent (government subsidized loan) would reduce the proposed system's annual cost to 75 percent of its base case level. This indicates the potential impact a government subsidy could have to encourage the development of community energy systems. Note, this analysis has not considered the impact of income tax and the deductibility of interest payments. Because interest payments are income tax deductible, the actual interest change impact could be less severe than illustrated in Figure 4-11.

The system's economic life is the final model parameter to be subjected to a graphical sensitive analysis. The base case assumed a 20-year for all system components. In actual practice the economic life of each component is likely to vary. For example, pumps may have an economic life of 10 years or less, while the PVC piping in the water distributor subsystem should last indefinitely.

Figure 4-12 varies the economic life from 0 to 50 years. Figure 4-12 clearly indicates that severe cost increases must be expected if the system life drops below 10 years. The cost advantage of extending the system life beyond 20 years in insignificant.

This graphical sensitivity analysis of the assumptions used to construct the base case can be summarized by highlighting the conditions which make the proposed system economically unfavorable, i.e., increasing the total annual cost 25 percent or more. These include:

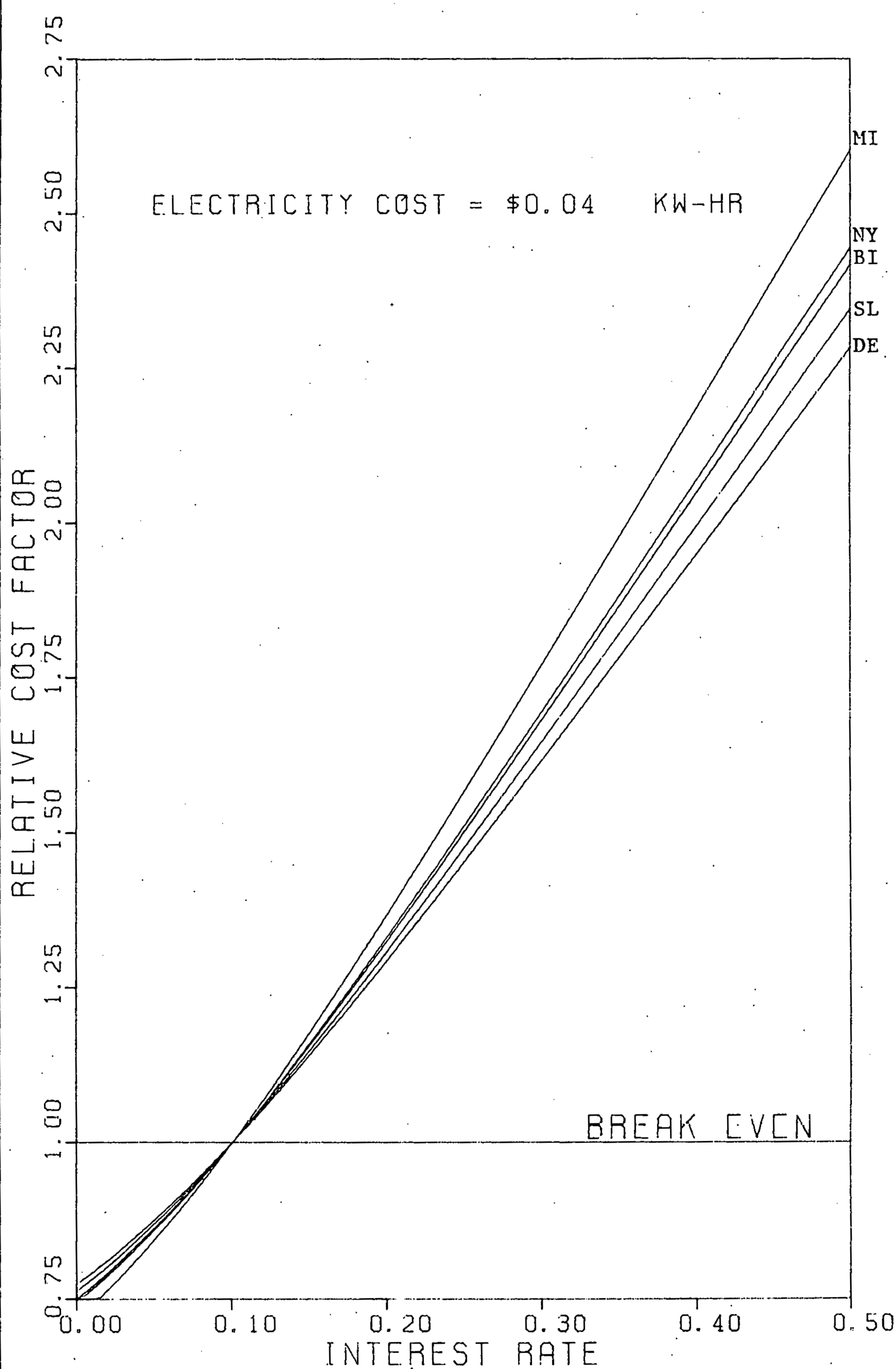


FIGURE 4-11 VARIABLE INTEREST RATE VERSUS BASE CASE (10 PERCENT)

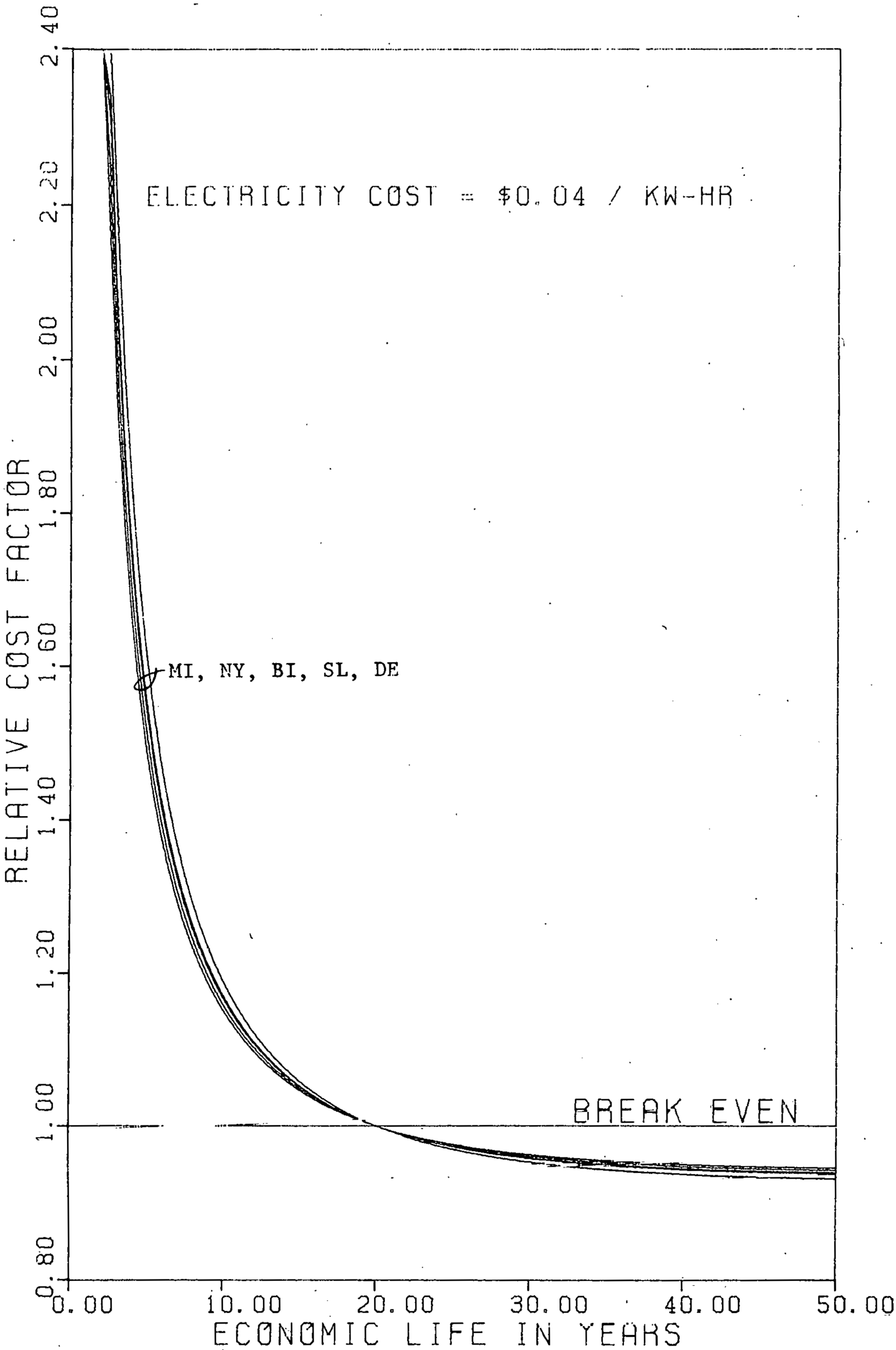


FIGURE 4-12 VARIABLE ECONOMIC LIFE VERSUS BASE CASE (20 YEARS)

1) reducing the module size to less than 5 units;

2) increasing the unit spacing to more than 500 feet;

3) increasing the interest rate to 20% or more;

4) decreasing the system life to 10 years or less.

Combinations of these impacts have not been analyzed, but are likely to further reduce the economic feasibility of the proposed system.

IV.B Residential Applications - Retrofit

The economic potential of the proposed system recommends it favorably for new residential construction. Several drawbacks could limit its retrofit potential:

1. Less than 100% saturation would increase the effective home spacing.

2. The water distribution subsystem installation costs would be increased due to disruption of streets and existing utilities.

3. Conversion may necessitate the removal of functional existing heating and cooling equipment.

4. Selling conversion to private home owners would be difficult. Low cost financing may have to be made available.

Certain residential communities have the potential to eliminate or minimize these problems. Section X proposes the retrofit of a residential community on a military base. Other governmental or institutionally owned housing offers a reasonable retrofit potential. Likewise, existing multi-family units such as apartment or townhouse complexes might be readily converted to the proposed system. Block renovation projects in inner-city neighborhoods also have retrofit possibilities.

IV.C Commercial Applications

Section IX includes a proposed system for a downtown commercial area. Such a system involves several property owners from both the private and public sector. Commercial applications such as shopping centers and industrial or office parks normally would involve a single owner. Small commercial applications are similar to the residential system from a basic equipment standpoint. In the small commercial establishment a furnace and air conditioner or 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 consultant, 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.

IV.D Water Supply and Distribution Subsystem Management

A significant open question in the adoption of the water source heat pump centered integrated community energy system relates to the management of the water supply and distribution subsystems. What type of organization is required to design, finance, install and maintain these subsystems? The creation of a new utility, like those currently providing water, electricity and natural gas, might be required. Some water source heat pump systems could be started without the development of a formal utility structure. The system would be initially designed, built and financed by the developer of the new residential community. The management and maintenance of the system would be assumed by the homeowners association. Independent contractors might enter into agreements with homeowners associations for system maintenance. System maintenance fees could be collected by the homeowners association and distributed to the contractor on a fixed price or per-call basis.

As water source heat pump systems become more prevalent, a more formal utility structure will become necessary. That structure could be part of an existing water, electric or gas utility or formed as a separate entity. The utility's ownership may be either private, as are many current suppliers of electrical power and natural gas, or public like most water systems. Public ownership may offer advantages in terms of seeking investment funds via tax-free bonds. Croke, et al, have described the financing of such systems.[15] The inertia associated with municipal organizations may make a private utility company the most expeditious alternative.

The water supply and distribution utility would have responsibilities beyond the servicing of the system. Some of its roles would include:

1. System planning
2. System maintenance
3. Emergency preparedness
4. Aquifer quality assurance
5. Heat balance control

The public health and welfare aspects of proper aquifer management demand careful monitoring and regulation of the water supply. The utility or a separate management board must be charged with that responsibility. With widespread adoption of the water source heat pump system concept, aquifer management will become a significant concern of local governments. A county water quality officer might be required.

With institutional application such as a military base, an individual in the base civil engineering office could be designated to manage the water supply and distribution system. That individual would direct the on-base maintenance activities and coordinate aquifer management information with off-base officials and other aquifer users.

V. EXPECTED ENVIRONMENTAL IMPACTS

All environmental impacts above the ground are expected to be either favorable or of minimal adverse proportions with respect to present heating and cooling systems. 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. Iron algae is a possibility which could clog the aquifer passages. However, 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.[16,17] 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 decrease of energy input through the reduced use of oil and gas more than offsets the 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 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°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:

$$CaCO_3 + H^+ \rightarrow Ca^{++} + HCO_3^-$$

and, limestone dissolves by the equation:

$$CaCO_3 + H_2CO_3 \rightleftharpoons Ca^{++} + 2HCO_3^-$$

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:

$$H_2O + CO_2 \rightleftharpoons H_2CO_3$$

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.[18]

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 $Ba(NO_3)_2$ dissolve in water to form neutral solutions. Others such as iron sulfate may hydrolize 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 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).[19] About 10 different injection tests with different samples were made. The duration of the tests ranged up to some 100 days in certain instances.

In summary, any change in aquifers under conditions of nominal increases in temperature of the return water--anything in the range of 25^{o}, 35^{o} or even 50^{o} 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 to require 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.

In downtown city areas with major buildings, growth is expected to progress more quickly than in the residential areas. Since retrofit is shown to be feasible (Section IX), economic pressure will encourage the growth. Since modules will include fewer buildings, administrative decisions are minimized. Modules will be sized to provide sufficient water (well capacity) for a building or buildings. Many major buildings will be one module with groups of wells or could have more than one module with independent heating and cooling systems. Integrating with the other modules allows aquifer control and increases reliability Connected modules need not necessarily be adjacent.

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 1-8 and 1-9. 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 the proposed heat pump system is local high recovery energy storage with a simple uninsulated 2-pipe system. A number of variations to the basic system are considered. The first three alternatives add a heat exchanger and new controls. These are evaluated in Section III, Expected Performance. A second set of variations involves variation in the design of basic water distribution systems between aquifers and the heat pump systems. The third set of variations employs abandoned deep mines, caverns, rivers or lakes for energy storage rather than aquifers. The systems are not analyzed due to lower performance and higher capital costs relative to the basic systems.

The first three 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 3-3.

The water temperature range in the first alternative is the same as that proposed for the basic system, 60^{o}F for cooling and 80^{o}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 are 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 3-3). 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. This system is similar to the ice producing heat pump systems, except the cold water replaces the ice. 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. This is the opposite to the ice producing heat pump systems. In this case the cooling performance with 80°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, four additional heat pump water distribution systems were considered.

The first two systems use individual water source heat pumps in the controlled environments. The second two systems use a central heat pump and transmit chilled and hot water. The piping sequences are shown in Figures 7-1 to 7-4.

The first two 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 reliaibility of the required insulation on pipes over a long period of time (in excess of 20 years).

2. Piping costs are appreciably higher (Table 11-22). 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, buildings, 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, when local aquifers are not available, 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 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. It loses the basic concept of local storage with relatively short water transfer distances. Overall performance will be lower.

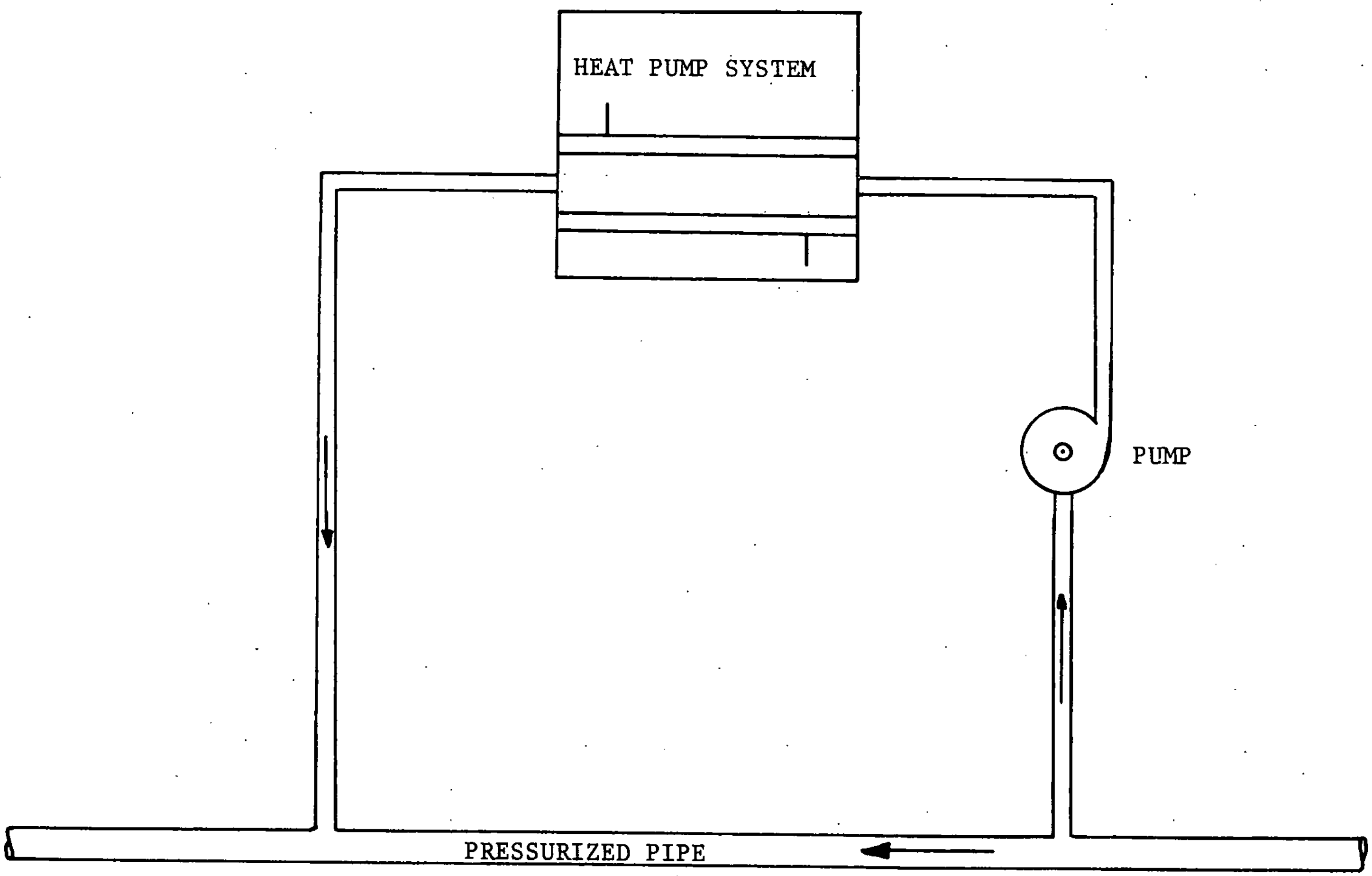


FIG. 7-1. 1 PIPE SYSTEM WITH PUMP CIRCULATING WATER THROUGH HEAT EXCHANGER
(NO INSULATION ON THE PIPES)

20.11.78

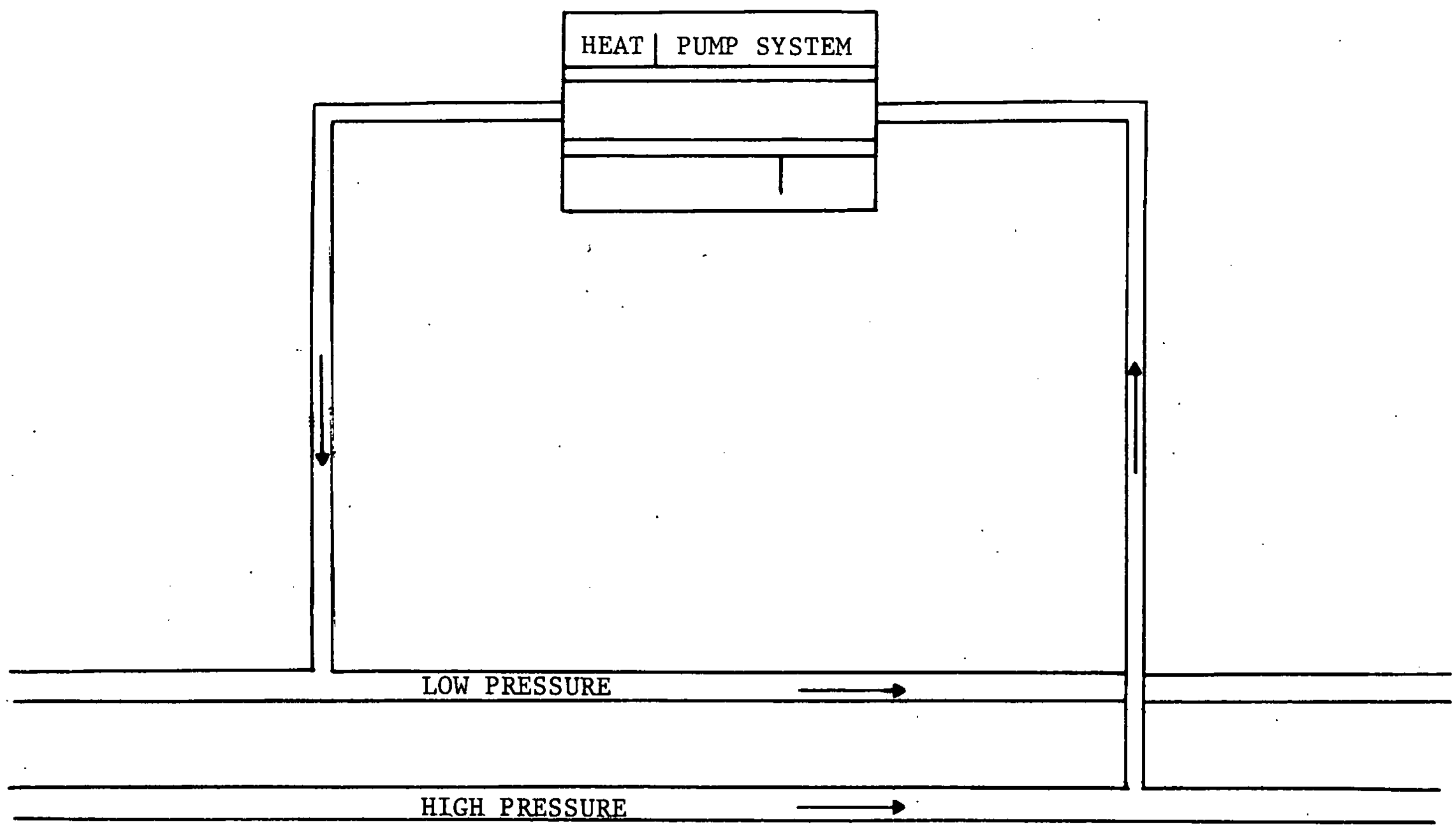


FIG. 7-2. 2 PIPE SYSTEM
(NO INSULATION ON THE PIPES)

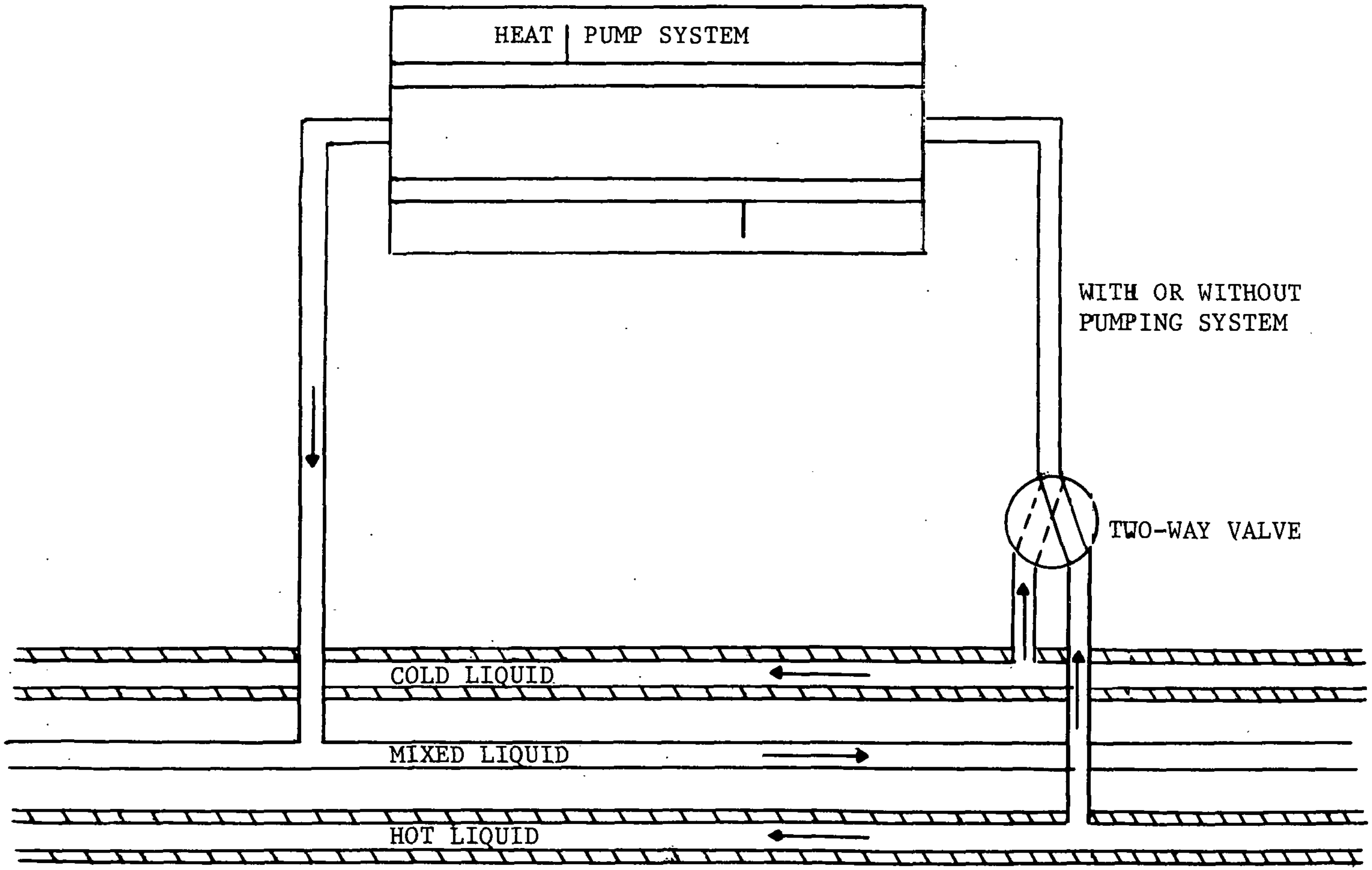


FIG. 7-3. 3 PIPE SYSTEM -- CENTRAL HEATING AND CHILLING
(ALL PIPES INSULATED)

20.11.78

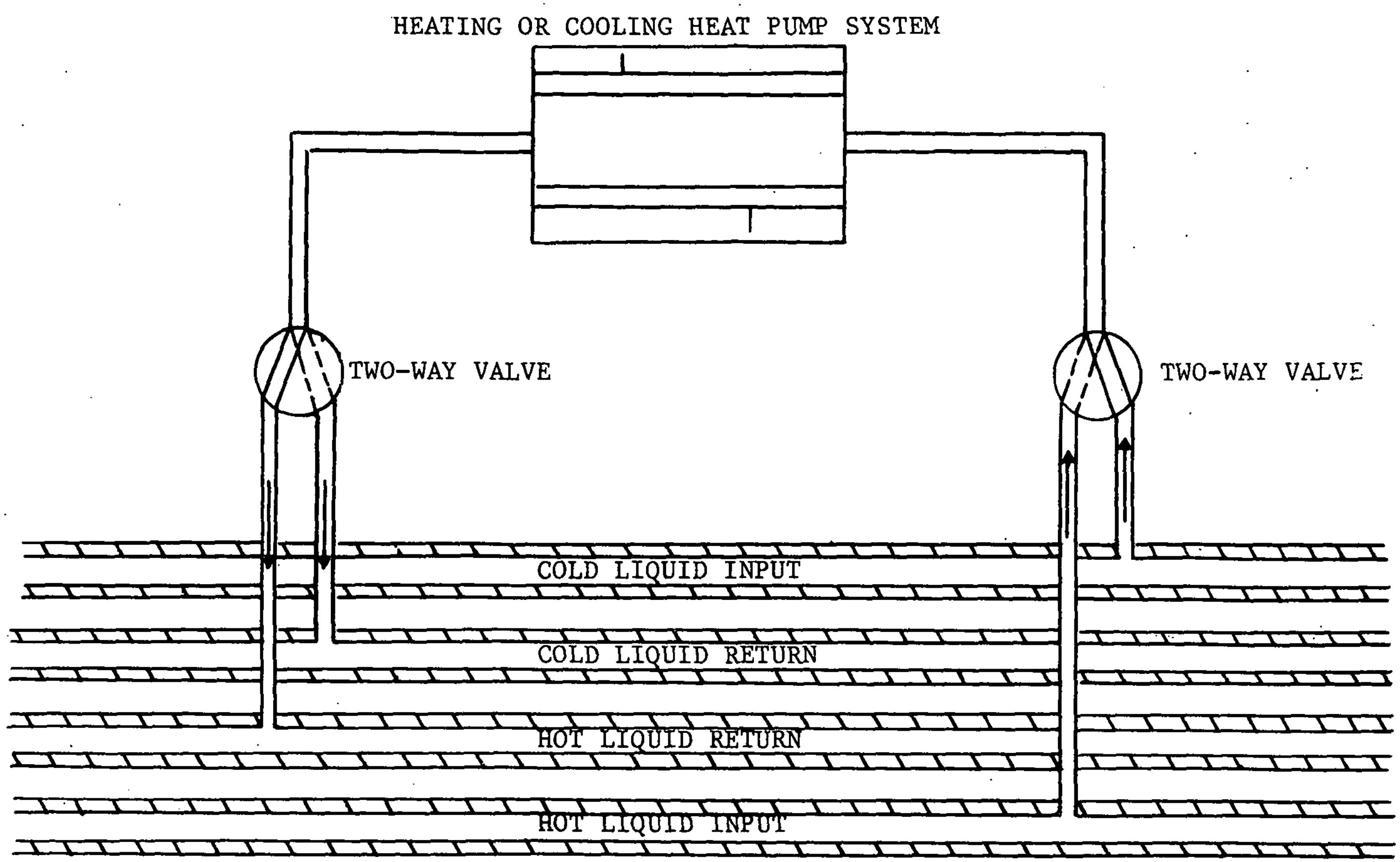


FIG. 7-4. 4 PIPE SYSTEM -- CENTRAL HEATING AND CHILLING
(ALL PIPES INSULATED)

20-11-78

A second alternative when local aquifers are not available is the use of nearby rivers and lakes. In the case of streams, water temperatures cannot be controlled. In the case of lakes the water temperature will be hard to control. In both cases a single pipe system is expected to be most economical, as distance transport is required. A large decrease in performance is indicated. This alternative would rarely be used, especially in the case of natural rivers and lakes as these are usually associated with aquifers.

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 auxiliary direct heat transfer systems in series or parallel with the heat pump show the best performance. Some evaluation tests producing performance data for the first three alternatives are 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. These two 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.

Actual operation of the pumps, captive air tanks and a control system will display problems in this area. Dynamic response with pumps starting and stopping can be valuable. It must be noted many portable water supply systems operate with this type system. However, there is no water return.

IX. LOUISVILLE DEMONSTRATION PROJECT

Louisville is one of the many cities located along rivers such as the Ohio, Mississippi, and Missouri where aquifers have developed from glacial outwash and riverine processes forming deposits of sand, gravel and silt. As a result, a fair aquifer for thermal storage exists at moderate depths below the surface. In the Louisville area, bedrock underlies the aquifer proposed for heating and cooling purposes. Water seeps through joints and bedding planes in the limestone and can provide large amounts of water for ideally located wells. However, the cracks and joints, along with a low probability of good wells, make this source questionable for heat transfer and thermal energy storage.

As with many river towns, a major downtown area has developed, surrounded by a larger sprawling metropolitan area. For the first phase of the proposed demonstration area, the system is being applied to a group of major downtown buildings. These consist of city-owned buildings, state-owned buildings and privately-owned buildings. Some buildings will be retrofitted and some buildings will have completely new systems. Most buildings will have large central heating and cooling systems whereas one building and parts of others will have small distributed heat pump units.

Many of the downtown areas have used water-source heat pumps in the past. The system was a once-through system with the water being discarded into the sewage system. The heating and cooling systems were converted to standard oil heating and water tower chiller systems for two primary reasons. First the demand for water became excessive and the aquifer water level fell drastically. Second, the treatment of sewage made discarding of water, economically prohibitive. The proposed system bypasses both these problems, and a large percentage of the water will be returned to the aquifer. The discharged water is a requirement to keep the aquifer below basement levels.

This secondary problem exists in the area. The land, originally similar to swampland, has a water table which is rising and threatening foundations and basements.[20] Excess water can be pumped to the Ohio River and used as a means to balance the annual energy cycle. If excess heat from air conditioning is available, warm water can be discarded and, if the heating load is larger, excess cold water can be discarded.

IX.A. General Description

The general area for the proposed demonstration is shown on the map in Figure 9-1. The names of the buildings in the proposed application are listed on the map. The building ownership includes city government, county government, state government, and private investors. Inclusion of some federal buildings is a distinct possibility.

The buildings listed in the first stage for the system are installing water source heat pumps in the immediate future and using ground water with provisions for water storage.[21,22] Time is critical in order to control the level of water in the aquifer. The water in the aquifer is rising and will flood basements in a few years. A 2-pipe system will be installed to distribute the water. The system can conveniently be converted to the proposed system in the future. Cost verifications derived from the original implementation will make future engineering estimates more accurate. The buildings in Phase I are Jefferson County buildings:

1. Fiscal Court
2. Court House
3. Court House Annex
4. Old First National Bank (converted into a Court House Annex)
5. Old Jail (to be renovated into offices)

All these buildings will be retrofits.

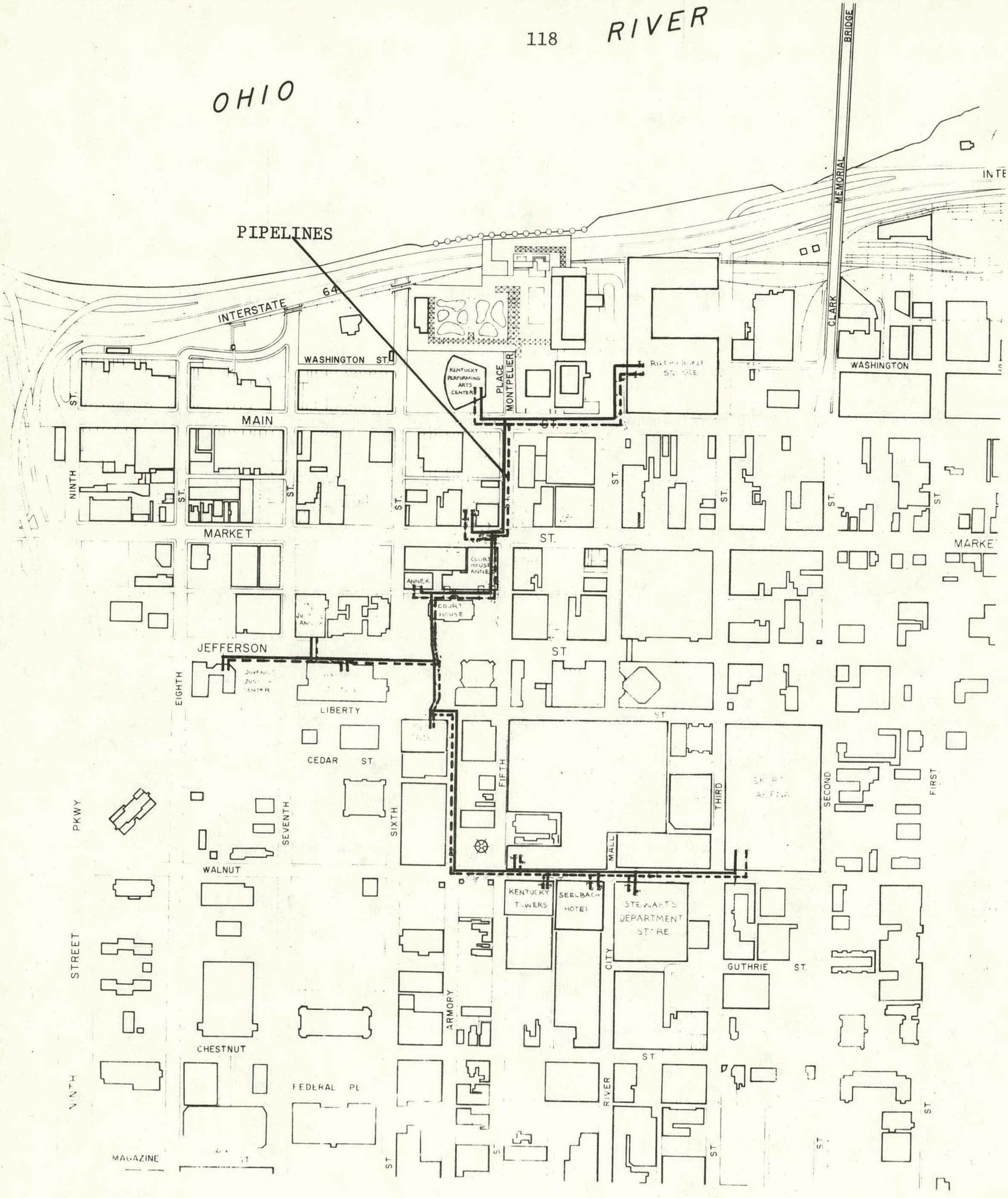


FIGURE 9-1. SECTION OF DOWNTOWN LOUISVILLE ANALYZED FOR DEMONSTRATION PROJECT

Phase II will include city and county buildings with some planned buildings. These buildings include:

1. Hall of Justice
2. County Juvenile Detention Center (under construction)
3. Performing Arts Center (planned)
4. Sports Arena (planned)

The Hall of Justice is a retrofit whereas the other buildings can include design of the heating and cooling systems to match the proposed system. Decisions in the near future are necessary for the County Juvenile Detention Center. The time table for the last 2 buildings is not critical. State support is being solicited for both these buildings.

The private sector can be considered Phase III even though some of the work will be done simultaneously. The buildings in this group are part of a plan of rejuvenation in the downtown area. A few speculative downtown buildings have experienced success, resulting in additional expansion. The major potential facility is the Galleria which consists of 3 office buildings, a major department store and about 150 small shops. A mall running between the office buildings and the department store and shops is to be enclosed, heated and cooled by the state. A summary of the buildings in this phase is:

1.	Galleria	350,000 ft.^{2} office space
		100,000 ft.^{2} department store
		150,000 ft.^{2} small shops
2.	Stewart's Department Store (minor renovation)	
3.	Seelback Hotel (major renovation)	
4.	Kentucky Towers (residential apartments)	

A number of other buildings such as the Galt House East & West in conjunction with a riverfront development have a possibility of being added to the demonstration proposal.

IX.B. Basic Heating and Cooling Loads

Estimated peak loads for heating and cooling are shown in Table 9-2 based on a local Louisville engineering study (unpublished). The peak loads for cooling are in general equal to or larger than the heating load due to internal loads except for the apartment building. Placing the apartment building and the hotel in the system help reduce average internal loads for the system.

Table 9-1 gives the summary weather conditions for Louisville. It should be noted the number of degree heating days is 3 times as large as the cooling days. The equilibrium temperatures for the major buildings are below 32°F during occupation. Some buildings require cooling at outdoor temperatures below 10°F. Adding homes or apartments in future expansions with 65°F, outdoor temperature equilibrium point will bring the system much closer to equilibrium and decrease annual water flow.

Peak water flow rates will be based on peak cooling loads as no heating for tradeoff exists under these conditions, so the design of the well, pump and distribution system is based on this criterion. Winter peak load will be decreased by simultaneous cooling in the inner cores of the building. Peak water flow rate is based on 8,155 tons of cooling. Based on an average cooling COP equal to 4.0 (large chillers, Tables 3-8 and 309) the energy rejected at peak load equals 1.25 times the cooling load or:

$$1.25 \times 8{,}155 \text{ tons} \times 12{,}000 \text{ BTUH/Ton} = 1.22 \times 10^{8} \text{ BTUH}$$

For the 20°F temperature difference, a peak flow rate of 12,700 gpm is required. At 800 gpm per well with a 50% safety factor, allowing for breakdowns, fires, etc., 24 pairs of wells are required. An overall system of 8 interconnected modules with 6 wells in a module is selected. Modules must be concentrated in areas of usage to minimize flow rates and pressure losses.

TABLE 9-1. WEATHER DESIGN DATA FOR LOUISVILLE, KENTUCKY

Average Annual Hours At Temperature And Humidity[2]

Temp. °F	Humidity {90} <30	30-49	50-69	70-79	80-89	90-100	Total[1]
102	4	3	0	0	0	0	7
97	9	32	2	0	0	0	43
92	19	113	59	0	0	0	193
87	21	143	183	9	0	0	355
82	25	161	244	97	18	0	541
77	19	155	223	159	152	37	742
72	22	110	209	144	189	195	869
67	20	90	174	142	170	161	758
62	19	103	137	117	168	148	693
57	14	120	126	99	131	165	654
52	8	109	169	83	116	135	619
47	3	87	195	97	120	131	634
42	2	71	198	128	126	125	649
37	0	44	196	151	156	155	703
32	0	30	190	148	160	103	631
27	0	16	113	92	78	35	332
22	0	13	66	52	28	11	169
17	0	7	44	23	21	4	97
12	0	5	24	12	5	0	45
7	0	0	15	8	2	0	25
2	0	0	6	2	1	0	8
-3	0	0	1	1	0	0	3
-8	0	0	0	0	1	0	1

Design Data

Latitude 38° 1'   Longitude 85° 4'   Elevation 474 ft. MSL

Winter

Design Dry Bulb Temperature[3], °F   99% - 5   97.5% - 10

Summer

	1%	2.5%	5%
Design Dry Bulb and Coincident Wet Bulb Temperature[3], °F	95/74	93/74	90/74
Design Wet Bulb Temperature[3], °F	79	77	76

Degree Days[4]		Mean	Maximum/Year	Minimum/Year
	Heating	4377	5023/76-77	3815/73-74
	Cooling	1361	1717/77	1055/74

1. May not equal sum of row due to rounding
2. Based on US Weather Bureau data published for Louisville, Kentucky
3. ASHRAE Fundamentals Handbook, 1977
4. Based on published NOAA data, 1969-1977, for Louisville, Kentucky

Internal loads are very significant in this study. Some of the buildings have a heating-cooling breakeven temperature around 10°F. At least 2 of the buildings have used air conditioning during a daytime workhour visit with outside temperatures near 0°F. These buildings will require heating at night.

For preliminary estimates, annual heating and cooling numbers are determined from peak loads, building functions and hourly temperature date in Table 9-1. All buildings in Phase I and the first 2 buildings in Phase II will have inner and outer cores with relatively low internal loads. All these buildings are essentially 100% distributed office space. Nightime heating is expected below 50°F temperatures (37% of annual hours). The last 2 buildings in Phase II will have peak hours during engagements, for cooling. Otherwise heating and cooling are expected to change over around 65°. The Galleria and Stewart's Department Store will have the largest peak cooling loads during peak business hours (4 to 6 hours during peak temperature days). It should be noted their peak loads for cooling are much greater than peak loads for heating. All other buildings have either larger peak heating loads or equal peak heating and cooling loads. The Seelback and Hotel Kentucky Towers have very small inner cores so their heating and cooling change areas occur near 60°F.

Based on the previous analysis and weather hours, it is estimated 40% of peak heating load (50% of annual hours is below 57°F) for four months equals average annual heating load. It is estimated 20% of peak cooling loads equals average cooling load for 12 months. This results with:

Annual Heating Load	1.0×10^{11} Btu
Annual Cooling Load	1.72×10^{11} Btu

Annual cooling load is 1.72 times greater than annual heating load due to internal loads. As a result, cooling towers will be required in the system to remove excess heat. The cooling towers must operate only in periods when $60^{o}F$ water temperatures can be obtained. Existing cooling towers with the present buildings are expected to have sufficient capacity. In addition, warm water ($80^{o}F$) can be discharged to maintain equilibrium and water-table levels. Figure 1-10 for heating and cooling COP's equal to 4.0 indicates the equilibrium ratio is approximately 1.75 times as much heating is available as compared with cooling. Part of this ratio can be reversed in the future by adding private homes and apartments to the system. Based on degree days these buildings will have 3.5 times as much heating as cooling.

During the periods which require some heating (over 50% of annual hours are below $57^{o}F$), simultaneous heating and cooling occur. During this period water will be circulated between buildings without moving in and out of the aquifers. At times there will be zero interaction with the aquifers. During these periods the aquifer pumping will be small so overall performance will be up to 7% greater.

For part load performance the water pumping efficiency will remain almost constant. With proper controls water demand can be met by operating a fraction of pumps at peak efficiency to meet demand. The smaller heat pump units will operate at peak efficiency at all times as control is an on-off sequence.

The large central units operate continuously at part load. According to J.E. Christian 61, COP at part load for screw type units is almost constant down to 40% load. The COP falls rapidly below 40% load. The centrifugal compressor units act similarly with a more gradual decrease below 50% load.

An additional technique which is expected to be utilized in the new buildings is "free cooling" with outside air. Any time outside temperature drops to below $57^{o}F$, outside air is circulated through the buildings. This will reduce the cooling load by approximately 30% and possibly eliminate the need for cooling towers.

IX.C. Aquifer Description

The geologic framework that controls the availability of water in the Louisville area is illustrated in Figure 9-2.[23] The upland areas are underlain by shale and limestone of Silurian, Devonian and Mississippian Ages. These rocks dip to the southwest at about 40 ft. per mile. The present valley of the Ohio River along the western and northwestern part of the area cut into the shale and limestone during glacial times. The rock valley is filled with alluvium of Quartenary Age which underlies the Ohio River flood plain to a maximum depth of 130 ft. The alluvium consists of glacial outwash, sand, and gravel with a blanket of recent silt and clay and is connected hydraulically with the Ohio River along much of its course in the area.

The glacial deposit of sandygravel in the flood plain has a vast water-storage capacity and high transmissivity and is the principal aquifer in the Louisville area. The limestone provides a secondary aquifer, particularly where solution openings occur along extensive joint systems and well-formed bedding planes. Limestone in the central part of Jefferson County yields water to many domestic wells, and the limestone bedrock beneath the glacial sand and gravel in the city yields large quantities of water to industrial wells. The clay and shale are not significant as aquifers but are important because they influence the flow of water to and from other formations.

Formations of Ordovician and Silurian ages are exposed in the eastern one-third of the county. Formations of Mississippian Age comprise the bedrock of the Knobs area in the southwestern part of the county. These formations, however, are not of hydrologic importance locally and are not defined in the local hydrologic system.

The aquifer to be used in this study is shown as a cross section in Figure 9-2. The aquifer consists of a 60 to 120 ft. bed of sand and gravel interspersed with silt layers at various locations and levels.[24,25] It is

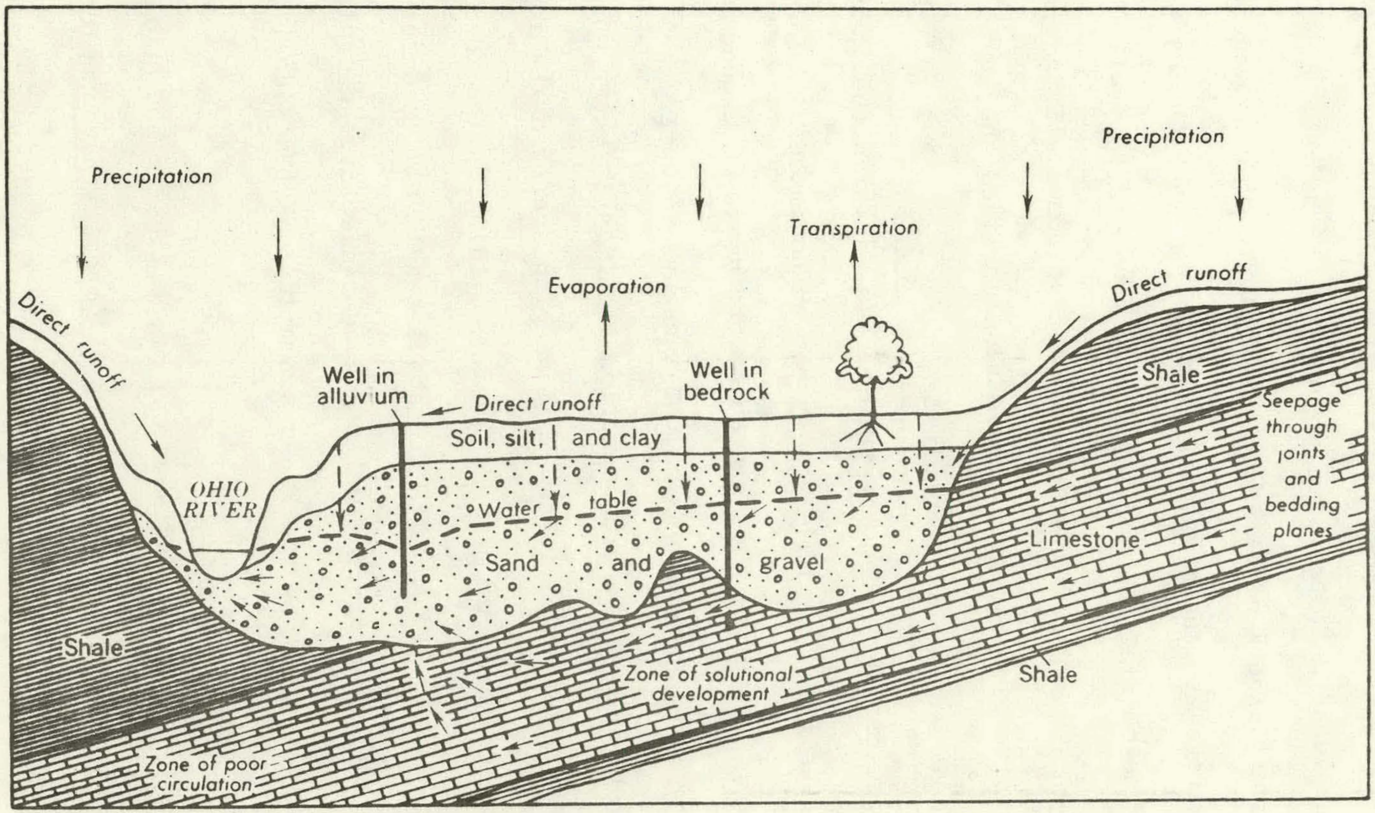


FIGURE 9-2. GEOLOGICAL CROSS SECTION UNDER LOUISVILLE. FROM SUMMARY OF HYDROLOGIC CONDITIONS OF THE LOUISVILLE AREA. GEOLOGICAL SURVEY WATER SUPPLY PAPER 1819-C.

covered with a layer of silt, minimizing the flow of water into the aquifer.

The limestone directly below this aquifer has some potential for producing water. However, the water moves through joints and bedding planes which conflict with the ideal heat transfer properties of the pebbly bed heat exchanger and the water distribution pattern characteristic of a homeogenous aquifer. No deeper aquifers are known to be available for the proposed system.

Much of the heating and cooling in the downtown Louisville area since the 1940's has been done with water source heat pumps. In addition, the process industries along the river have also utilized the aquifer as a source of process water. As a result the water table has been lowered to the point of questionable economic return. In addition the charge for disposal of water in sewer lines has made the system uneconomical. Consequently, heating and cooling systems have abandoned the usage of ground water.

Figure 9-3 shows projections of the increase in water-table level for the years 1962 to 1982. The cross-section location is shown as AA in Figure 9-4. The water level is now beginning to approach the level of foundations and basements of major downtown buildings. The ideal water-table level is between 410 and 412 ft. above sea level. A water-table level of 412 ft. is used for projected wells and well discharge calculations. This provides a saturated aquifer thickness somewhat less than ideal for the proposed system.

Information with respect to physical properties of the aquifer is, for the most part, not available. Values of porosity have apparently never been recorded, although the nature of the aquifer suggests that a range of 25-30% is reasonable. Some data on permeability of the aquifer are available, but not with respect to the downtown area. The range of permeabilities shows

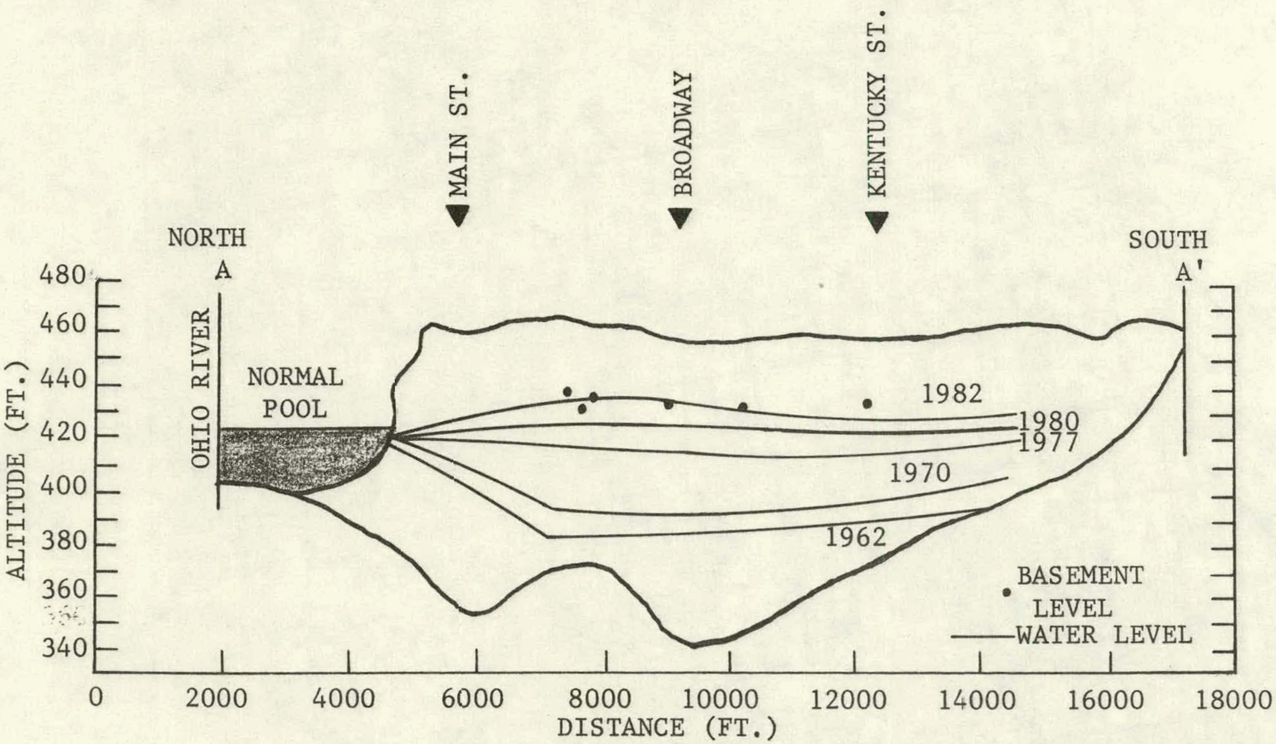
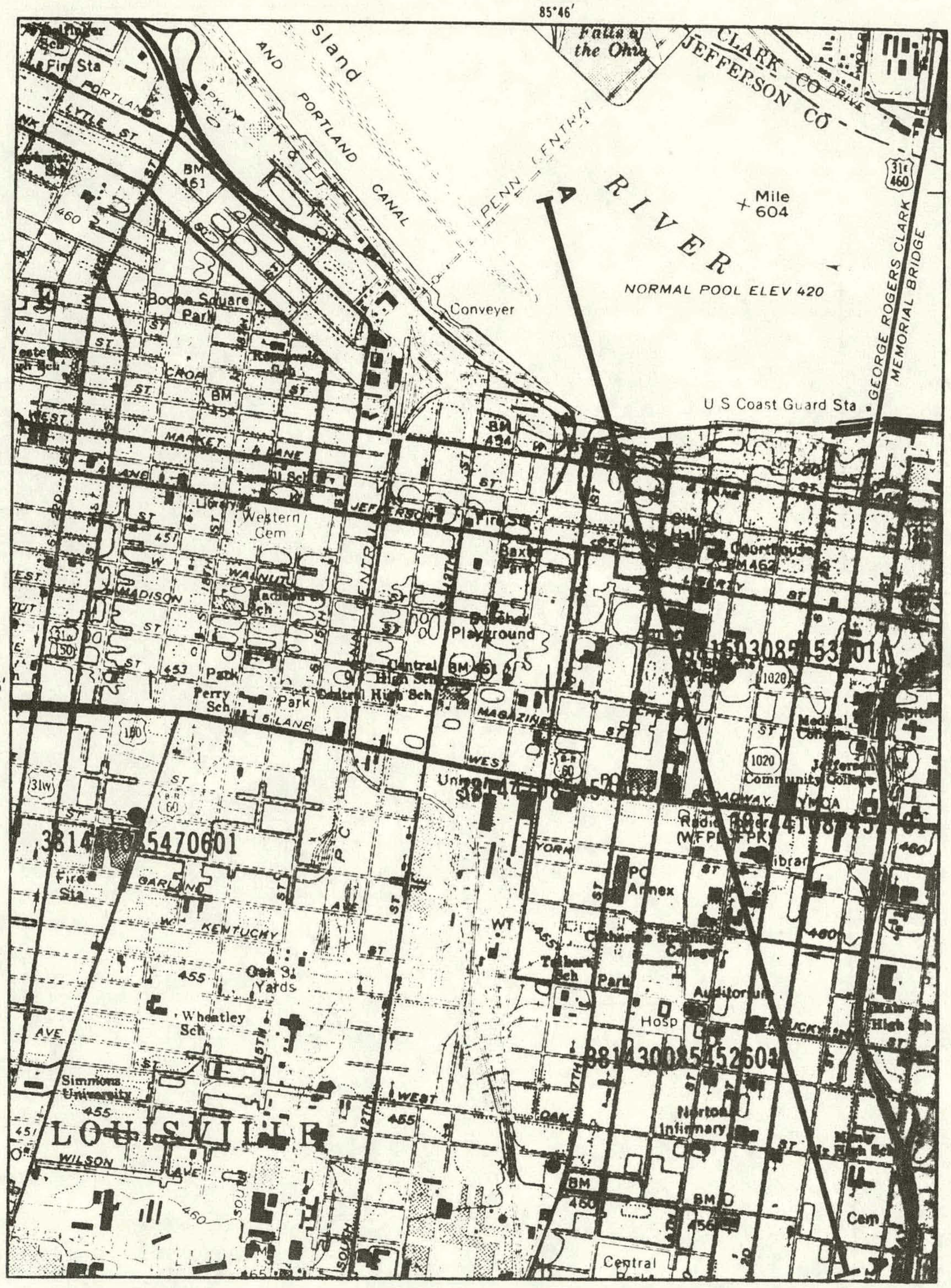


Figure 9-3. GENERALIZED PROFILE SHOWING PAST, PRESENT, AND PROJECTED WATER LEVELS BENEATH DOWNTOWN LOUISVILLE, KENTUCKY.

FROM RISING GROUND WATER LEVEL IN DOWNTOWN LOUISVILLE, KENTUCKY, 1972-1977. U.S. GEOLOGICAL SURVEY. WATER-RESOURCES INVESTIGATIONS 77-92.

3.2.79



SCALE 1:24,000

State Number

Well Collection Data

0 1000 2000 3000 4000 5000 FEET

0 500 1000 1500 METERS

Figure 9-4. Downtown Louisville, Kentucky, showing locations of section line and observation wells.

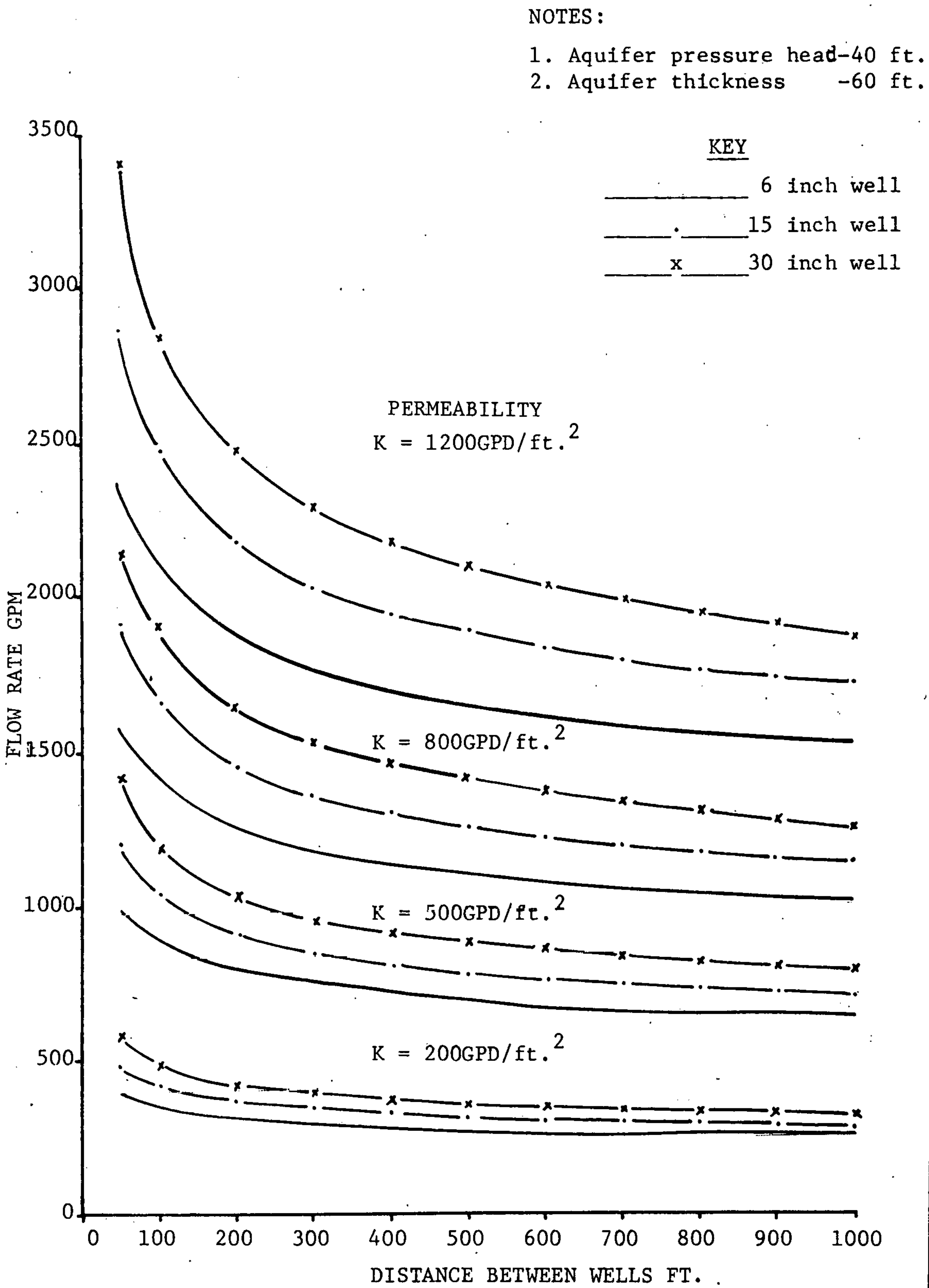


Figure 9-5. FLOW RATES FOR LOUISVILLE, KENTUCKY, AQUIFER

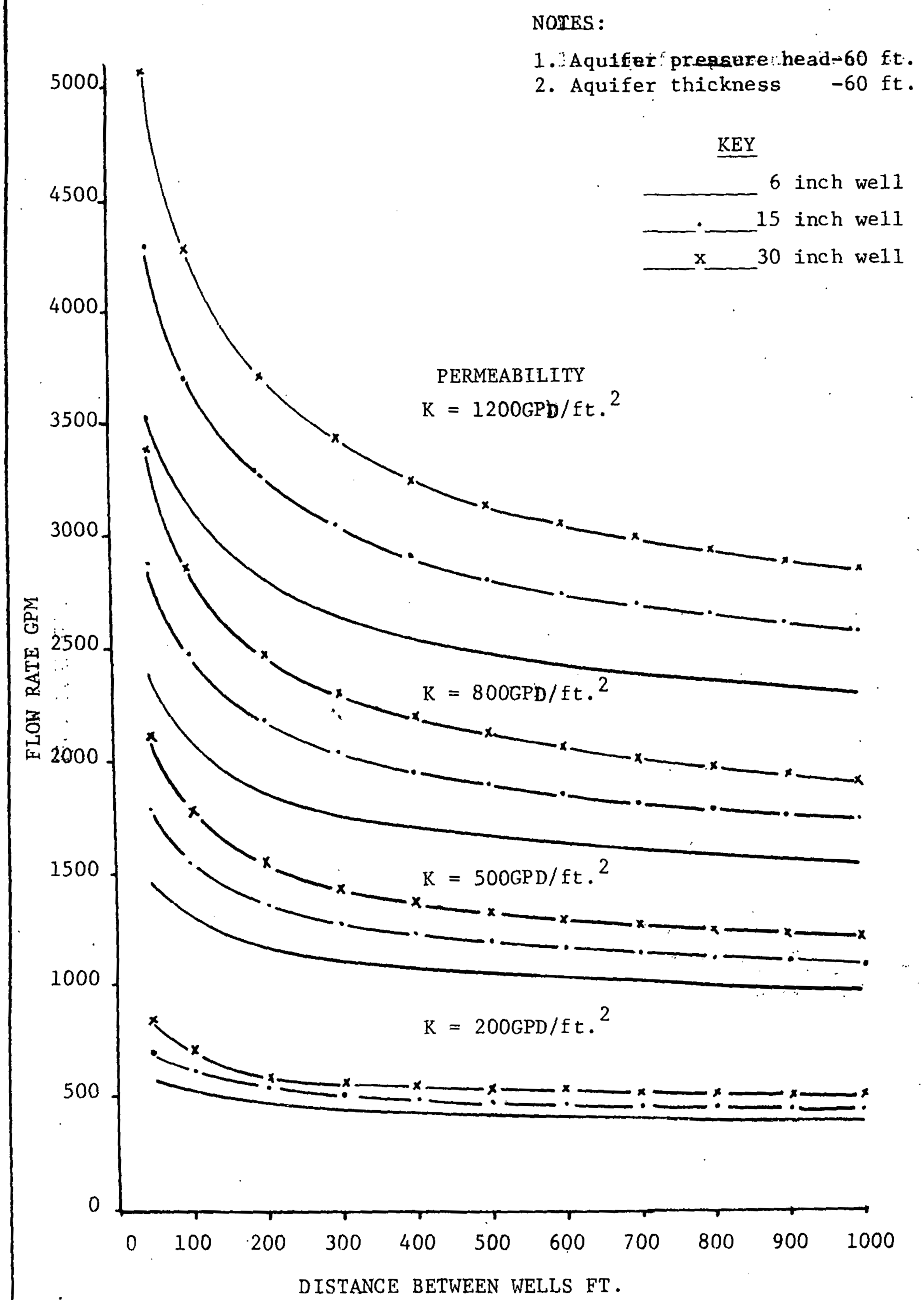


Figure 9-6. FLOW RATES FOR LOUISVILLE, KENTUCKY, AQUIFER

large variations, from 150 to 1,800 gal./day/ft.^{2}. The values are medians of data derived from 30 to 50 wells.

A water-quality summary for the aquifer is given in Table 9-2. With the overall concentrations of impurities between 150 and 334 ppm, the water can be directly used in large and small heating and cooling systems.

A series of calculations has been made for yields from various well sizes, separations and permeabilities in the Louisville aquifer. The assumed depth of the aquifer is 60 ft. The yields are given in Figures 9-5 and 9-6. The large range of possible yields makes well placement important.

Over 330 recorded wells have been drilled in the area. Less than 50 of these are still active. Cross sections of the aquifers have been constructed from driller's logs showing the extent of gravel, sand and silt. From a study of these cross sections and known values of permeability, well locations with prospects for high permeabilities can be projected with some degree of confidence. Wells should be located in areas with minimum amounts of silt which would indicate the aquifer will be of high permeability. In addition, wells should be located in areas of maximum aquifer thickness. Based on this analysis, 15-inch wells capable of yielding over 800 gpm comprise one of the assumptions for this study. A small test well will be drilled at each location to assure the location for a production well. The aquifer variations made this a requirement. The wells will range in depth from 80 to 110 ft. for purposes of estimating costs.

IX.D. Proposed System

The system for the demonstration project will be limited to establishing and maintaining the well system and the water distribution system. The heating and cooling systems in the buildings (all major buildings in this project) will be the responsibility of the owners. The advantage to the owners will be lower

Table 9-2. Water Quality Summary from Sand and Gravel Aquifer Below Louisville*

Constituent	Concentration (ppm)		
	Maximum	Minimum	Average
Silica (SiO_2)	7.0	2.4	5.6
Iron (Fe)	2.2	.2	.8
Calcium (Ca)	56	28	39
Magnesium (Mg)	15	7.6	10
Sodium (Na)	31	6.3	15
Potassium (K)	3.2	.9	2.1
Sulfate (SO_4)	131	38	74
Chloride (Cl)	59	8.0	24
Fluoride (F)	.4	.1	.2
Nitrate (NO_3)	6.0	2.0	4.0
Dissolved solids (total)	334	150	214
Total hardness (as $CaCO_3$)	197	93	137
Alkalinity (as $CaCO_3$)	107	49	74
Specific conductance — micromhos at 25°C	543	243	351
pH	8.8	6.7	---------
Temperature — °F	81	35	60

*From Summary of Hydrologic Conditions of the Louisville Area. Kentucky, Geological Survey Water-Supply Paper, 1819-C nameplate capacity chillers which can be used for cooling and heating attached to major energy reductions. The substitution of electricity for fuel oil also has major price effects. The lower nameplate capacity results in a decrease of capital cost and installation cost. The basic system is shown schematically in Figure 1-1. A layman's schematic is shown in Figure 9-7.

The system will be divided into 8 modules with 6 wells in a module. A total of 8 modules is required to complete the system. The modules are designed to match the various phases. Phase I and Phase II can be developed as independent modules.

The wells will be sand wells 120 ft. deep with 12 in. casing and a 15 in. bore hole. Each well will have a pumping system capable of pumping 800 gpm. Water flow will be controlled by a computer system operating variable speed pumps as a function of demand. This is considered feasible for this system as a continuous water flow will be required and the number of pumps operating can be limited. This allows the pumps to operate near maximum efficiency with a minimum of captive air water tanks.

The distribution system is indicated by solid and dashed lines representing pipelines on the map in Figure 9-1. Pipelines with solid and dashed lines will be placed in available tunnels. All lines are uninsulated 12 in. cast iron pipes. This allows flow rates of over 1,000 gpm with pressure drops of less than 30 psi in the most extreme case. In general, pressure drops will be less than 10 psi. Wells will be distributed to minimize the flow rates in pipelines. The well locations will require test drillings due to aquifer variations for optimum locations, so they have not been placed.[24,25] Well locations will be within 10 ft. of pipelines. Placing fire plug outlets between pipelines will provide water flow rates in excess of 2000 gpm at all

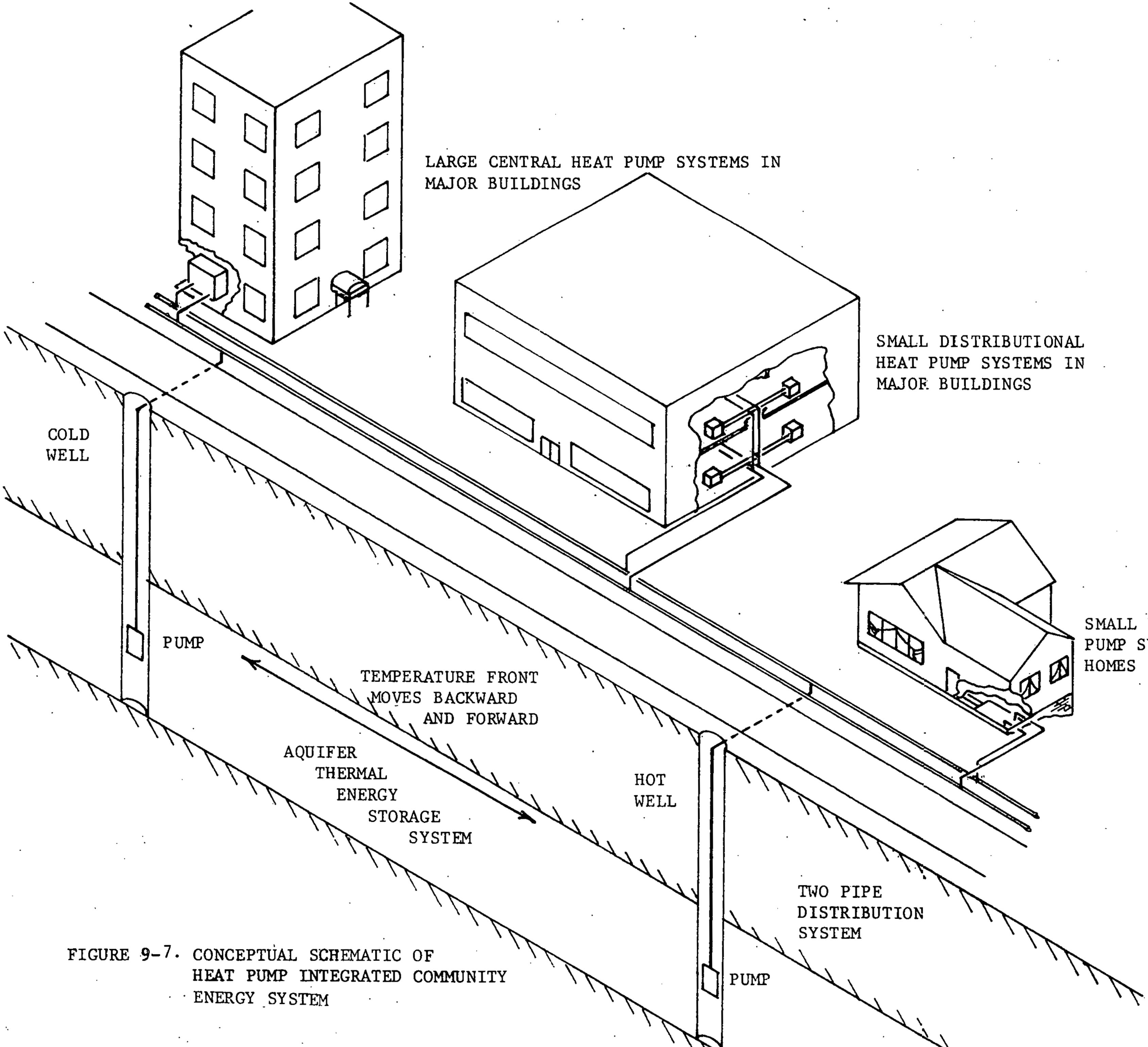


FIGURE 9-7. CONCEPTUAL SCHEMATIC OF HEAT PUMP INTEGRATED COMMUNITY ENERGY SYSTEM

locations and in excess of 500 gpm along 50% or more of the pipelines. This is sufficient water for all fire requirements in the area.

IX.E. Expected System Performance

Since the system will operate basically with large units, the performance will be based on 500 ton centrifugal hermetic chillers listed in Table 3-9. These values do not include primary, secondary distribution or the air circulation. Since basic data for heating does not exist for these large units, the coefficient of performance is assumed to equal the coefficient of performance for cooling. This is a conservative assumption as heating coefficient of performance is normally higher than the cooling performance with the given water input temperatures. The basic coefficient of performance utilized is 5.59.

For the analysis, this value is corrected for the proposed system water supply at the rate of 40 watts/gpm. A temperature difference of 20^{o}F is assumed for the water flow through the heat pump. The flow rate for a 500 ton hermetic cooling output is 750 gpm with a pump input of 30 Kw. This is approximately a 10% decrease in performance or an effective COP of 5.0. This value does not include internal water or air distribution. For cooling, the value does not subtract the work involved in water or air distribution.

Using this value the energy requirements for the proposed system per year are:

Cooling

$$1.72 \times 10^{11} \text{ Btu}/5.0 = 10^{10} \text{ Btu} = 1.00 \times 10^{7} \text{ Kw-hr}$$

At the power plant this requires:

$$8.6 \times 10^{10} \text{ Btu}$$

Heating

$$1.00 \times 10^{11} \text{ Btu}/5.0 = 2.0 \times 10^{10} \text{ Btu} = 5.9 \times 10^{6} \text{ Kw-hr}$$

At the power plant this requires:

$$5.0 \times 10^{10} \text{ Btu}$$

If the energy storage system is not installed and 60°F ground water is utilized, the cooling performance will not change. However, the required groundwater is not expected to be available. If cooling towers are used with 80°F water, the energy use will increase by approximately 33% to:

$$4.57 \times 10^{10} \text{ Btu} = 1.34 \times 10^{7} \text{ Kw-hr}$$

At the power plant this requires:

$$1.14 \times 10^{11} \text{ Btu}$$

For heating if ground water at 60°F is used the pumping rate must increase and chiller efficiency also decreases. The total increase is estimated at 36% or the heating energy will be:

$$2.72 \times 10^{10} \text{ Btu} = 7.97 \times 10^{6} \text{ Kw-hr}$$

At the power plant this requires:

$$6.8 \times 10^{10} \text{ Btu}$$

If oil or gas heat is utilized with a 60% efficiency the required energy increases by 330% to:

$$1.66 \times 10^{11} \text{ Btu}$$

Table 9-3 summarizes the performance for the proposed HP-ICES system versus the standard system for heating and cooling. The performance is given on a first law basis of required loads, user energy, user resource energy and resource energy. A second law analysis assuming chemical energy availability at 3500°R is included. The annual resource requirements are given.

Table 9-3. Heating Performance Summary for Louisville Project

	Annual Heating Load 1.00×10^{11} Btu		Annual Cooling Load 1.72×10^{11} Btu	
FIRST LAW ANALYSIS				
	HP-ICES	Oil Furnace	HP-ICES	Cooling Tower-Chiller
Energy, User Input	5.9×10^{6} Kw hr	1.66×10^{11} Btu	1.00×10^{7} Kw hr	1.34×10^{7} Kw hr
Resource Energy-User Input	5.0×10^{10} Btu	1.66×10^{11} Btu	8.6×10^{10} Btu	11.4×10^{10} Btu
First Law Performance	2.0	0.6	2.0	1.5
Resource Energy[1]				
Coal (10^3 S Tons)	2.01	-	3.66	4.87
Gas (MMCF)	1.71	-	4.69	6.23
Oil (Bbls)	0.30	31.1	1.07	1.43
Nuclear (10^3 Grams)	17.6	-	28.6	38.1
Hydro (10^6 Kw hr)	1.30	-	1.48	1.96
SECOND LAW ANALYSIS[2]	(40 °F Reference Temperature)		(90 °F Reference Temperature)	
Input (Chemical) Availability	4.28×10^{10} Btu	1.42×10^{11} Btu	7.24×10^{10} Btu	9.6×10^{10} Btu
Power Plant Losses, Availability	2.28×10^{10} Btu	-	3.8×10^{10} Btu	5.04×10^{10} Btu
Final Availability (120°F input)	1.37×10^{10} Btu	1.37×10^{10} Btu	1.25×10^{10} Btu	1.25×10^{10} Btu
Second Law Performance	0.32	0.096	0.17	0.13
ANNUAL HEATING AND COOLING				
Resource Energy User Input	13.6×10^{10} Btu	28.0×10^{10} Btu		
Resource Energy				
Coal (10^3 S. Tons)	5.67	4.87		
Gas (MMCF)	6.40	6.23		
Oil (10^3 Bbls)	1.37	32.5		
Nuclear (10^3 Grams)	46.2	38.1		
Hydro (10^5 Kw hr)	2.78	1.96		

1. Per ASHRAE Standard 90-75 Section XII.
2. After mixing with controlled environmental air, Second Law Performance is appreciably reduced.

IX.F. Expected Economics

The economics are governed by energy savings cost, capital cost and maintenance. The energy cost comparison in Louisville is against low cost TVA electricity at \$0.029/Kw-hr for the period to June 1979. Present oil cost is \$0.35/gal. (January 1979). Both these prices are expected to increase, so the energy savings cost in reality should be multiplied by an appreciable factor. By the fall of 1979, oil prices have essentially doubled so an oil price of \$0.70 is utilized. Depending on the system comparison, capital costs for the proposed system are somewhat higher. Figures have not been calculated for maintenance, but maintenance costs will be lower for the proposed system. The chillers will operate over lower and constant temperature differences. The oil furnaces and oil supply system will be discarded. A standard cost of 5% of capital will be utilized for maintenance and operating costs.

Using the comparisons in the last section for cooling, energy costs are:

Proposed system

$$1.00 \times 10^7 \text{ Kw-hr/yr.} \times \$0.029 = \$290,000/\text{yr.}$$

Versus:

Cooling Tower System

$$1.34 \times 10^7 \text{ Kw-hr/yr.} \times \$0.029 = \$389,000/\text{yr.}$$

The annual energy cost savings for cooling equals \$99,000/yr.

Using the comparisons in the last section for heating:

Proposed system

$$5.9 \times 10^6 \text{ Kw-hr/yr.} \times \$0.029 = \$171,000/\text{yr.}$$

Versus oil heat at \$0.70/gal., 145,000 Btu/gal, or \$4.82 per 10^6 Btu:

$$1.66 \times 10^{11} \text{ Btu} \times \$4.82/10^6 \text{ Btu} = \$800,000/\text{yr.}$$

The annual energy cost savings for heating equals \$629,000. The total energy savings equals \$728,000/yr. The comparison is not valid nationally as TVA economical power is compared to the normal price of oil.

The capital cost summary for installation of the water distribution system is given in Table 9-4. The cost data compiled from R.S. Means[26] assume each type of job is in excess of \$150,000. As a result some of the installation costs such as sidewalk repair must be multiplied by a factor to account for the small work orders. The overall cost of the water supply system is \$1,238,390 as calculated in Table 9-4. An estimated capital cost of the installed heat pump systems is given in Table 9-5. The total costs for wells, water distribution and heat pumps is:

Wells and water distribution system	\$1,238,390
Installed heat pumps	2,574,600
Additional piping and installation (20% of heat pumps)	514,920
Total estimated cost	\$4,327,910

The standard system would include the heat pumps (chillers) and additional piping. The standard system cost is estimated at:

Installed chillers	\$2,574,600
Additional piping and installation (20% of chillers)	514,920
Basic estimated cost	\$3,089,520

Additional capital costs for the base system include:

1. Cooling towers,
2. Oil furnaces and
3. A major potable water supply system which provides fire production. This includes purifying facilities.

The HP-ICES total cost is expected to be less than the standard system. However, the wells and water distribution must be developed and maintained. Depreciating by sinking fund on a 20 year basis with an annual 8% interest

TABLE 9-4. PROPOSED SYSTEM COSTS NOT INCLUDING HEAT PUMPS

Louisville, Kentucky

System Components	Statistics Phases I	Statistics Phases II	Statistics Phases III	Costs x $1,000 Phases I	Costs x $1,000 Phases II	Costs x $1,000 Phases III	Totals Statistics	Totals Costs
Water Supply Subsystem								
Wells (120 ft.x12 in.) @ $10,200 each	6	6	36	61.2	61.2	367.2	48	$ 489,600
Pumps + Controls @ $10,000 each	6	6	36		60.0	360.0	48	$ 480,000
Water Distribution Subsystem*								
Double 12 inDuctile iron @ $38/ft.	3,000 ft.	1,000 ft.	2,000 ft.	114.0	38.0	76.0	6,000	$ 228,000
Gatevalvaes, 12 in. @ $980/each	3	6	8	2.94	5.88	7.84	17	$ 16,660
Trenches*								
Excavation and Backfill @ $1.48/ft. 4 ft. wide x 3 ft. deep	1,840 ft.	840 ft.	1,700 ft.	2.72	1.24	2.52	4,380	$ 6,480
Pavement Patching 3 in. thick asphalt @ $4.00/ft.	1,800 ft.	800 ft.	1,650 ft.	7.02	3.02	6.06	4,250	$ 17,000
Curb Replacement Concrete 6 in.x12 in. @ $25/each	3	3	4	0.075	0.075	0.01	10	$ 250
Sidewalk Repair Concrete, reinforced, 4 in. thick @ $40/each	3	3	4	0.012	0.012	0.016	10	$ 400
							Overall Total	$1,238,390

*Based on R.S. Means, Building Construction Cost Data, Robert Snow Means Co., Inc., Duxbury, Mass. (1978).

TABLE 9-5. Estimated Heat Pump Capital Costs*

	No.	Standard Size (Tons)	Cost Unit	Installation Cost	Installed Cost/Unit	Total Cost	
A. <u>Jefferson County</u> <u>Phase I</u>							
1. Fiscal Court	1	250	$35,000	$ 1,800	$ 36,800	$ 36,800	
2. Court House	1	250	35,000	1,800	36,800	36,800	
3. Court House Annex	1	100	20,000	1,000	21,000	21,000	
4. Old First National Bank	1	200	32,000	1,700	33,700	33,700	
5. Old Jail	1	200	32,000	1,700	33,700	33,700	
							$ 162,000
B. <u>Jefferson County</u> <u>Phase II</u>							
1. County Hall of Justice	1	250	35,000	1,800	36,800	36,800	
2. Juvenile Justice Detention Center	1	250	35,000	1,800	36,800	36,800	
3. Performing Arts Center	1	250	35,000	1,800	36,800	36,800	
4. Sports Arena	1	250	35,000	1,800	36,800	36,800	
							309,200
C. <u>Louisville Central Area</u> <u>Phase III</u>							
1. Galleria+	16	250	35,000	1,800	36,800	588,800	
	100	5			2,850	285,000	
2. Seelback Hotel	210	4			2,440	512,400	
3. Kentucky Towers	200	5			2,850	570,000	
4. Stewarts' Dept. Store	4	250	35,000	1,800	36,800	147,200	
							2,103,400
					Overall Total		$ 2,574,600

*Cost are based on Christian, J.E., <u>Central Cooling- Compressive Chillers</u>, ANL/CES/TE78-2, (1978), and <u>Unitary Water-to-Air Heat Pumps</u>, ANL/CES/TE77-9, (1977).

Man-hour costs where estimated at $17/hr.

+To be established in steps

rate (tax free bonds) on $1,238,390 or an annual rate of 0.027 plus .08 equals an annual capital cost rate of 0.107. Add 5% for maintenance and operation. Total costs are:

	Annual Rate	Cost
Capital cost	0.107	$132,251
Maintenance and operation	0.05	61,920
	Total annual cost	$194,171

Energy costs for the water system operation are included in energy costs. The annual energy savings were $728,000/yr. or an annual cash flow savings of $533,829.

IX.G. Institutional Considerations

In the late 1940's and early 1950's Louisville used water source heat pumps for heating and cooling. From the excess water usage the aquifer water level decreased to a non-usable level. The use of heat pumps was halted and the aquifer level has been rising until basements have become threatened. The breweries in the area still use aquifer water. Numerous wells, most not being used, still exist with most of the wells in major building basements. As a result many city, county and state regulations controlling wells and their use have developed. The legal implications are impossible to implement.

City, county, state and private organizations have combined to establish a board to produce a controlling body and a set of regulations exclusively for Jefferson County. The first step has been to develop an Inner Local Agreement. This has the support of city and county officials, the governor and the state legislature. The goal, believed to be attainable, is to pass a bill through the legislature cancelling all previous statutes and codes applying to the Jefferson aquifer and establishing a Ground Water Management Authority. This authority will be composed of city, county and state officials. They will have the power to establish the statutes and codes for wells and ground water utilization. This will include controlling the aquifer level. Work in this direction has been on-going for the last two years. The board is expected to be established within the next few months. All regulations will be established by the board.

The water distribution system and wells will be established and operated by the county. Underground tunnels through which some of the pipelines will traverse are presently owned and operated by the county. Part of the system is expected to be established as soon as the new codes and statutes are established.

Billing is expected to be by the county on quantity of water utilized. As homes and apartments are added, a fixed method of determining water bills is expected to be established. The cost of the water is expected to be minor (about 10%) to power costs.

All internal heat pump systems, central and distributed, will be designed, purchased and operated by the building owners. The cutoff point will be the water meter.

IX.H. Project Growth

The proposed demonstration project includes less than 1 square mile. The proposed system can easily be extended to cover a 50 square mile area composed of commercial buildings, industry and residences. A 30:1 increase in energy applications for heating and cooling is extimated.

The major downtown buildings should naturally phase into the system. To initiate general homes, a major public building housing project borders the downtown area. It consists of 1 section of 22 three-story, buildings, with 6 apartments to a building. Over 100 three-story buildings are in the adjoining section with eight to 16 apartment in each building. Residential neighborhoods are adjacent to these housing projects. The building and aquifer structures are conducive to expansion of the system.

X. FORT RUCKER DEMONSTRATION PROJECT

Fort Rucker, Alabama, is a military installation operated by the U.S. Army. Figure 10-1 illustrates the major physical facilities of Fort Rucker. The fort consists of training, support and residential facilities. Many of these buildings have been constructed within the past 25 years and are air conditioned. Table 10-1 indicates the weather conditions experienced at Fort Rucker and demonstrates the large cooling loads which must be handled. Most of the classroom and other central buildings are served by centralized steam boilers. These facilities offer a good potential for aquifer storage with chiller units. In addition, Fort Rucker has three family housing communities.

X.A. General Description

The Allen Heights residential community of Fort Rucker, Alabama, has been proposed as a retrofit residential demonstration site. Allen Heights consists of 279 duplex buildings comprising 558 family housing units. The buildings were constructed within the 1958-1962 time frame. Each wood-frame building rests on a concrete slab. The exteriors are combinations of brick and wood siding with roofs of asphalt shingles. Each unit contains 1,376 square ft. and is heated with a gas furnace and cooled by a central air conditioner. The buildings are separated roughly by a 200 ft. spacing with a 50 ft. setback from the street centerline.

X.B. Aquifer Description

Fort Rucker is situated on lands straddling the boundary between Coffee and Dale counties which lie within what is referred to as the Gulf Coastal Plain Province.[26,27] The area is underlain by geologic formations comprising parts of two major age groups, the Tertiary and Cretaceous Systems. Formations of the Coastal Plain crop out in east-west trending belts and dip gently to the south toward the Gulf of Mexico at the rate of 25 to 40 ft. per mile. Figure 10-2

FIGURE 10-1. OVERALL MAP OF FORT RUCKER INDICATING RESIDENTIAL AREA SELECTED FOR DEMONSTRATION

TABLE 10-1. WEATHER DESIGN DATA FOR DOTHAN, ALABAMA

Average Annual Hours at Temperature and Humidity[2]

Temp. °F	Humidity (%) ≤30	30-49	50-69	70-79	80-89	90-100	Total[1]
102	1	1	0	0	0	0	2
97	2	25	2	0	0	0	30
92	5	95	108	0	0	0	208
87	9	102	368	55	3	0	536
82	10	94	249	243	142	11	750
77	12	69	185	182	401	493	1341
72	27	84	143	128	270	748	1411
67	32	91	131	118	185	484	1038
62	26	100	133	96	147	379	882
57	24	94	136	94	125	225	698
52	21	83	144	99	109	151	609
47	12	68	137	97	102	92	506
42	5	52	109	79	85	45	377
37	2	27	67	50	45	25	214
32	0	16	45	24	15	8	109
27	0	5	25	12	6	0	49
22	0	0	4	1	1	0	7
17	0	0	1	1	0	0	3

Design Data

Latitude 31° 2'   Longitude 85° 2'   Elevation 321' MSL

Winter

Design Dry Bulb Temperature[3], °F   99% - 23   97.5% - 27

Summer

	1%	2.5%	5%
Design Dry Bulb and Coincident Wet Bulb Temperature[3], °F	94/76	92/76	91/76
Design Wet Bulb Temperature[3], °F	80	79	78

Degree Days[4]		Mean	Maximum/Year	Minimum/Year
	Heating	1546	2361/76-77	1037/73-74
	Cooling	2774	2981/73	2405/76

1. May not equal sum of row due to rounding.
2. Based on U.S. Weather Bureau data published for Mobile, Alabama.
3. ASHRAE Fundamentals Handbook, 1977.
4. Based on published NOAA data, 1969-1977, for Mobile, Alabama.

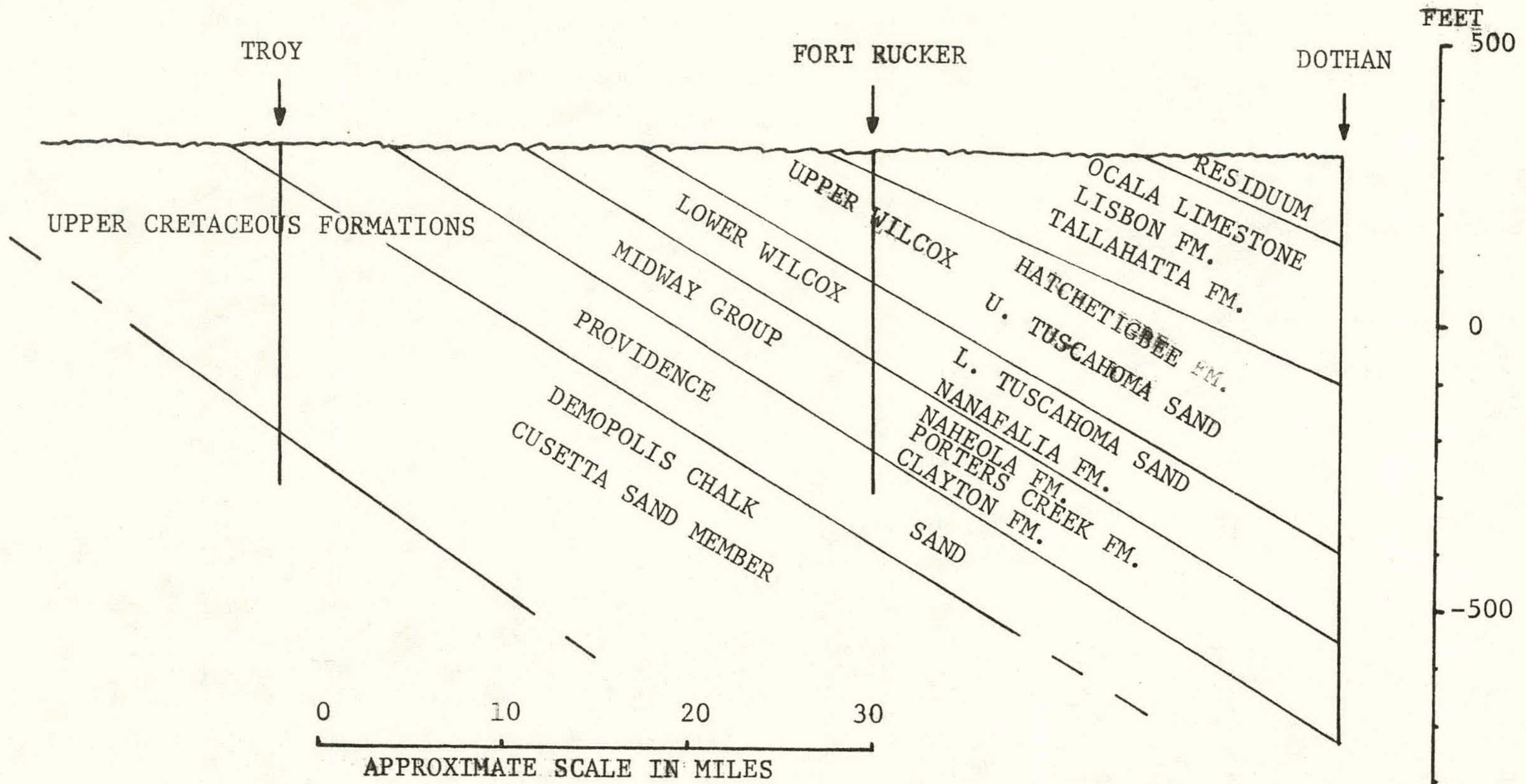


FIGURE 10-2. GENERALIZED CROSS SECTION SHOWING THE STRATIGRAPHIC SEQUENCE IN THE VICINITY OF FORT RUCKER, ALABAMA

3·6·79

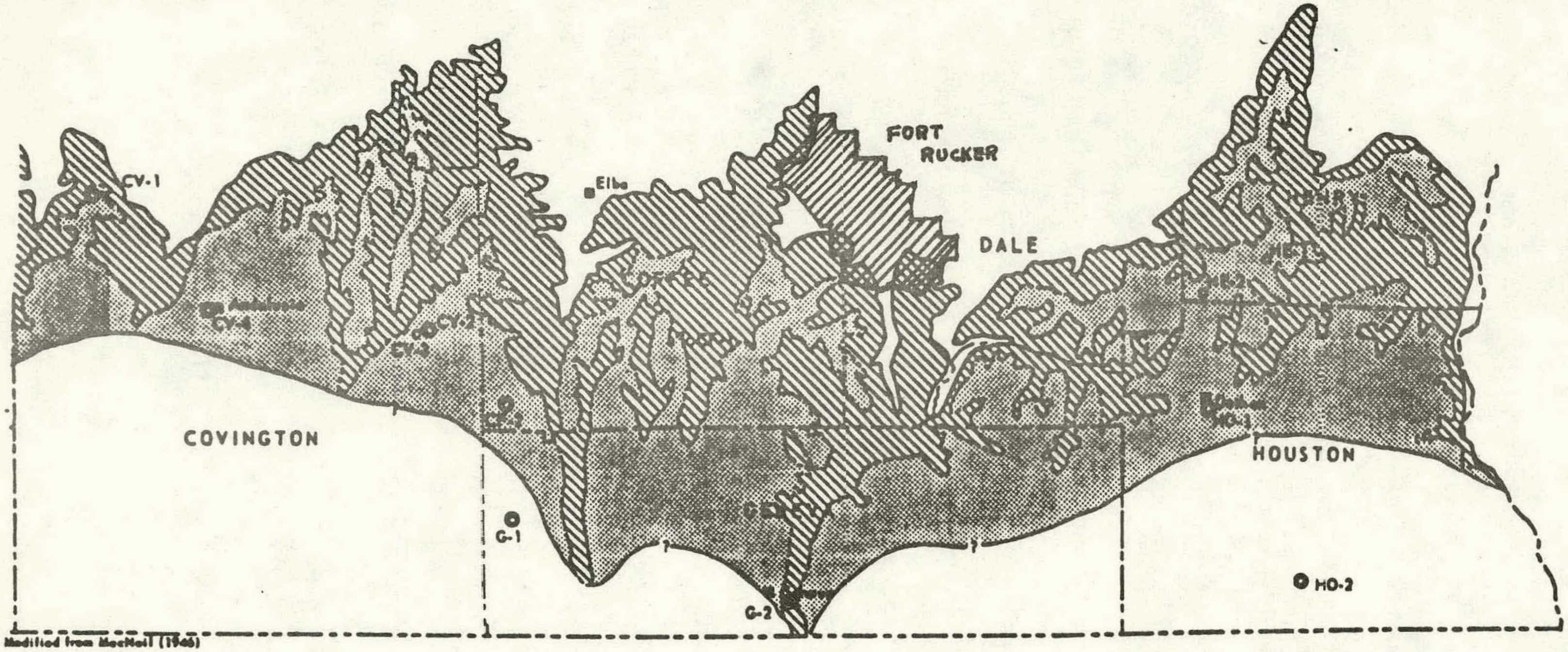
is a generalized cross section showing the stratigraphic sequences of geologic formations in the vicinity of Fort Rucker.

The geologic formations that crop out in southeastern Alabama differ widely in porosity and permeability such as well-sorted sands and gravels and cavernous limestones. Unsorted sands and sandstones have somewhat lower porosities and yield a moderate amount of water to wells.

Three major aquifers underlie Fort Rucker. These are the Lisbon Aquifer, the Upper Wilcox Aquifer and the Midway-Lower Wilcox Aquifer. The Lisbon Aquifer is of limited thickness in the Fort Rucker area, and therefore affords a questionable source of supply. The other two, however, provide adequate potential for the needs of the proposed project. Values for transmissivity, permeability, coefficient, of storage and porosity of the aquifers are not available, but reliable estimates of these physical parameters have been made from comparison with data obtained from recognized sources.

The Lisbon Aquifer is comprised of the Hatchetigbee, Tallahatta, Lisbon, and Moodys Branch formation and the Ocala Limestone, all of Tertiary Age. The massive glauconitic sand beds within these formations are important sources of ground water. Figure 10-3 shows the outcrop of the Lisbon Aquifer and Figures 10-4 and 10-5, respectively, are contour maps of the top and bottom of the aquifer.

In general, water from the Lisbon Aquifer is soft to hard and low in total dissolved solids. Locally high concentrations of iron, chloride and bicarbonate may limit utilization of water from this aquifer. Chemical analyses of water from selected wells tapping the Lisbon Aquifer are given in Table 10-2. The downdip freshwater-saltwater contact is well south of the Fort Rucker area.



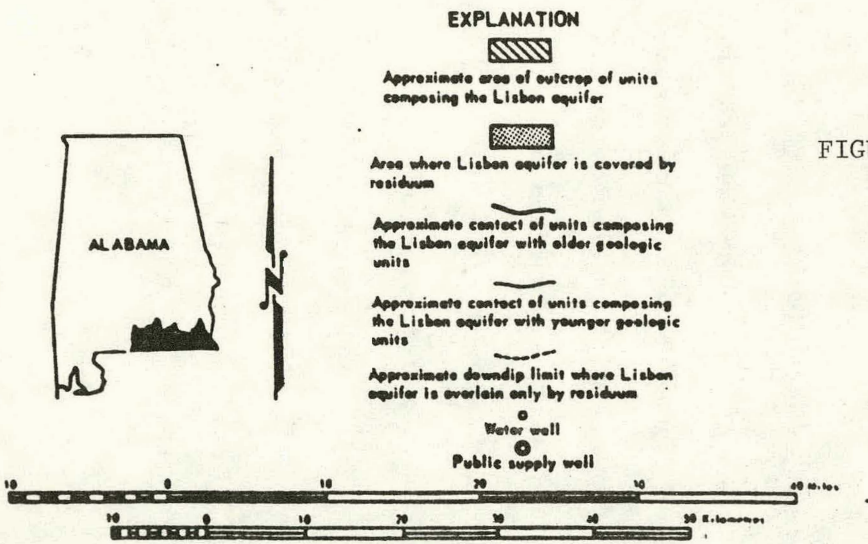
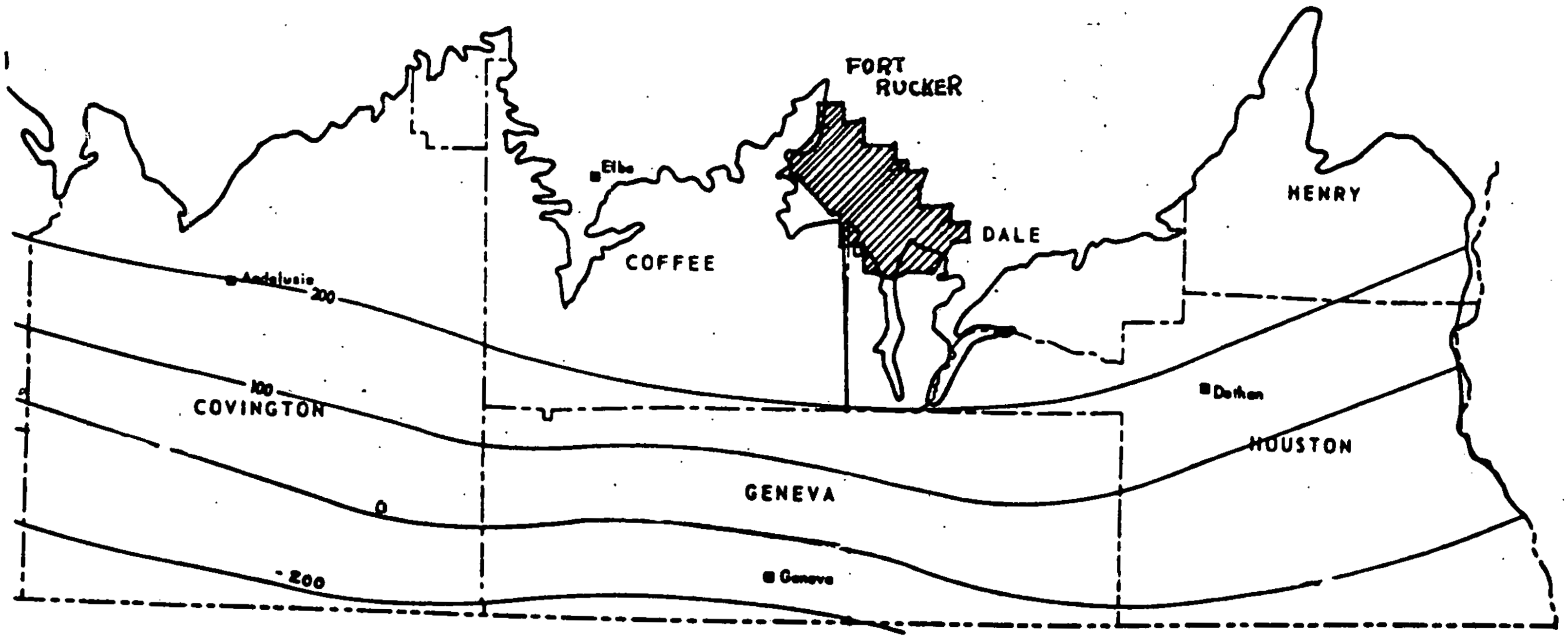
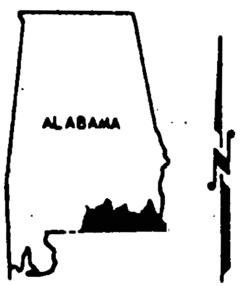


FIGURE 10-3. **MAP OF RECHARGE AREA OF THE LISBON AQUIFER AND SELECTED WELL LOCATIONS.**





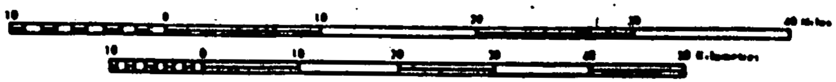
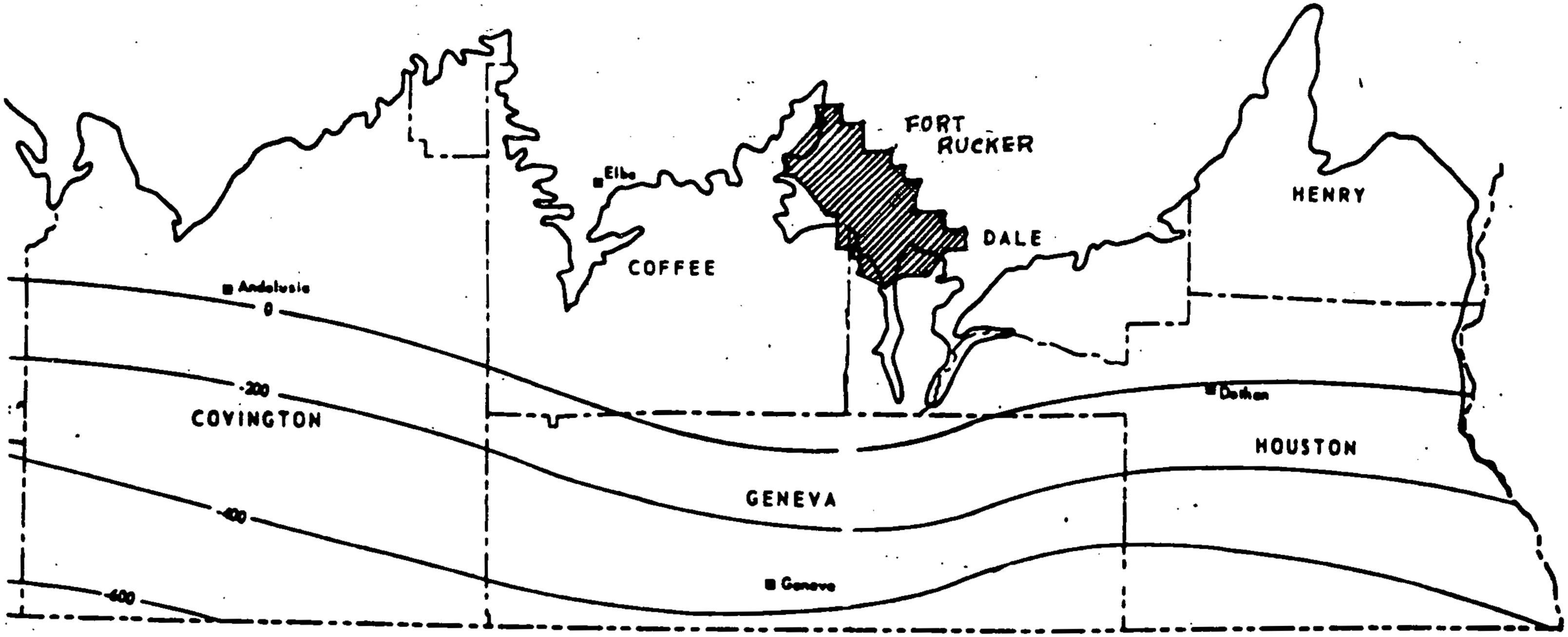


FIGURE 10-4. CONTOUR MAP OF ESTIMATED TOP OF LISBON AQUIFER.



EXPLANATION

Contact of the Lisbon aquifer with underlying sediments

--200--
Approximate elevation of base of Lisbon aquifer relative to sea level

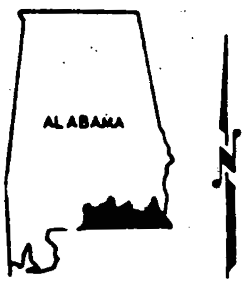


FIGURE 10-5. CONTOUR MAP OF ESTIMATED BOTTOM OF THE LISBON AQUIFER.

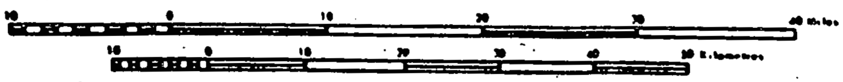


TABLE 10-2. Water Quality of the Lisbon Aquifer from Selected Wells (in ppm).

Well Number	Water-Bearing Unit	Collection Date	Well Depth (feet)	Iron	Calcium	Magnesium	Sodium	Bicarbonate	Sulfate	Chloride	Fluoride	Nitrate	Dissolved Solids	pH
CV - 1	Tl	1965	---	.02	28	1.5	7.6	82	2.6	12.0	0	8.6	114	7.4
CV - 2	Tl	1965	191	.01	24	.7	.9	71	0	2.1	0	.6	74	6.9
CV - 3	Tl	1965	279	.01	30	2.9	2.3	104	3.8	4.0	0	.2	108	7.1
CV - 4	Tmb, Tl, Tth	1965	331	0	34	7.6	2.8	133	6.4	4.8	0	.1	140	7.4
CF - 1	Tl	1964	52	.30	--	---	---	20	---	4.8	--	---	---	6.9
CF - 2	Tl	1964	185	.20	--	---	---	127	---	2.0	--	---	---	7.9
G - 1	Tl	1964	290	.08	--	---	---	.35	---	4.0	--	---	---	8.0
G - 2	Tl	1964	150	.02	--	---	---	122	---	10.0	--	---	---	7.5
HE - 1	Tl	1965	90	.10	4.0	1.9	4.0	10	2.2	4.0	.0	15.0	44	6.4
HE - 2	Tth	1965	183	.06	34	.0	1.4	96	7.8	.4	.1	.1	104	6.5
HO - 1	Tl, Tth	1951	335	.01	7.8	---	---	27	1.4	3.8	.0	2.8	44	6.3
HO - 2	Tl	1955	250	.0	109.6	---	---	116.4	75.3	17.1	.1	.1	---	6.9

Water-Bearing Units: Tmb - Moodys Branch Formation; Tl - Lisbon Formation; Tt - Tallahatta Formation; Tth - Tallahatta and Hatchetigbee Undifferentiated

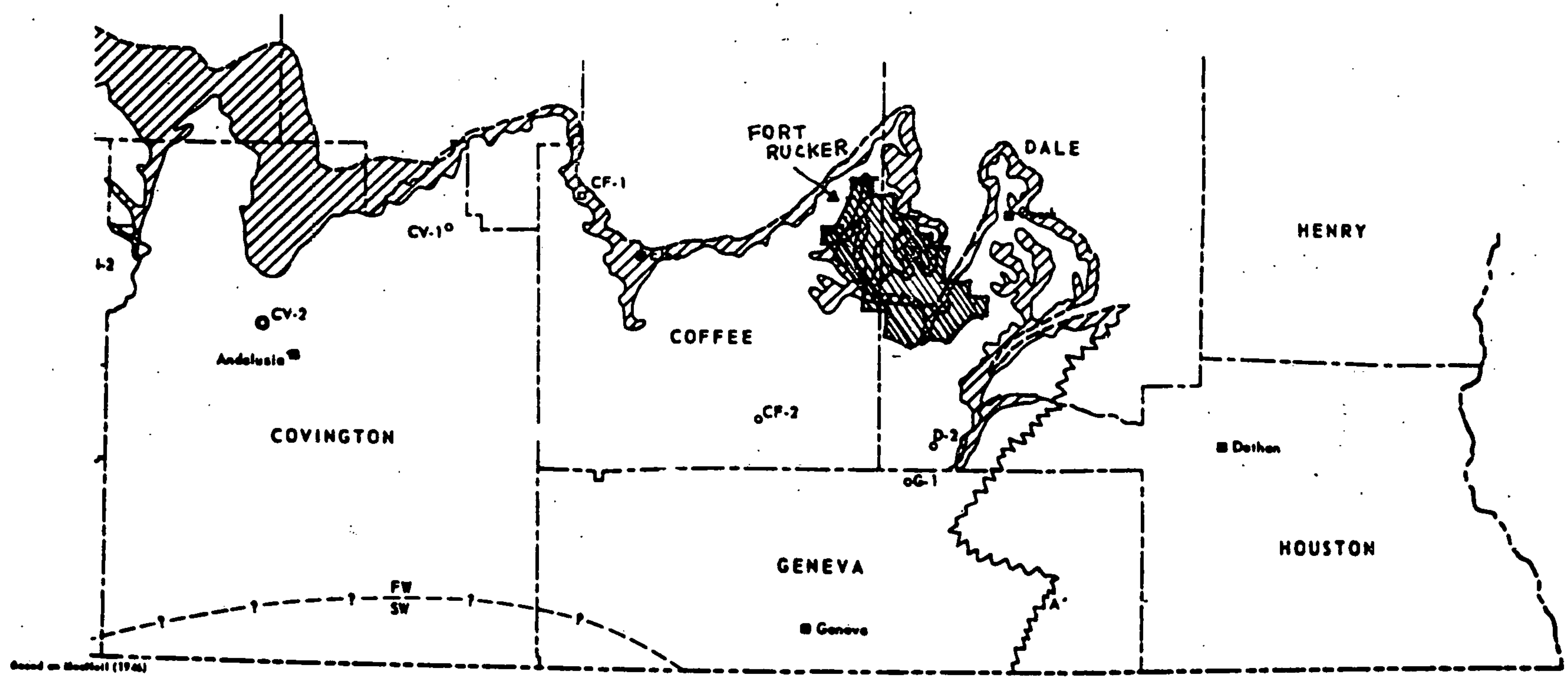
The Lisbon Aquifer is comprised of several individual hydraulically interconnected aquifers. Average yields of wells withdrawing water from the Lisbon Aquifer range from 0.5 to 1.0 million gallons per day.

The Upper Wilcox Aquifer is comprised of Coastal Plain sediments of the lower and middle Eocene series of Tertiary Age. In Covington, Coffee and western Geneva counties, sands of the Tuscahoma, Hatchetigbee and Tallahatta strata are undifferentiated. The Upper Wilcox and Lisbon Aquifers are indistinguishable in Dale, Henry, Houston and eastern Geneva counties. The thickness of the Upper Wilcox Aquifer is approximately 200 ft. in the vicinity of Fort Rucker. Massive sand beds within these strata are important sources of ground water.

Figure 10-6 shows the outcrop, recharge area and location of the freshwater-saltwater interface of the Upper Wilcox Aquifer. Figures 10-7 and 10-8, respectively, are contour maps of the top and bottom of the aquifer. Producing zones are lenticular in nature and cannot be traced for any great distance. Average porosity for the aquifer is estimated at 30%.

In general, water from the Upper Wilcox Aquifer is soft to very hard and low to moderate in dissolved solids, with locally high iron, chloride and bicarbonate content. Table 10-3 shows chemical analyses of water from selected wells completed in the Upper Wilcox Aquifer.

The Midway-Lower Wilcox Aquifer is comprised of the Clayton, Porters Creek and Naheola Formations of the Midway Group and the Nanafalia Formation and Tuscahoma Sand of the Wilcox Group, all of Tertiary Age. As shown on Figure 10-9, the outcrop and recharge area of the Midway-Lower Wilcox Aquifer is north of the Fort Rucker area. The thickness of the combined formations is approximately 200 ft. and the dip is toward the south and southwest at approximately 40 ft. per mile.



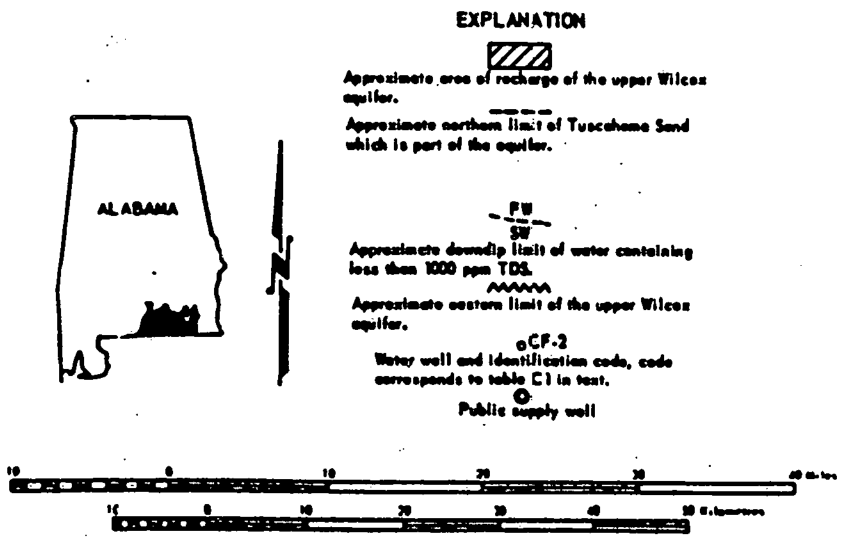
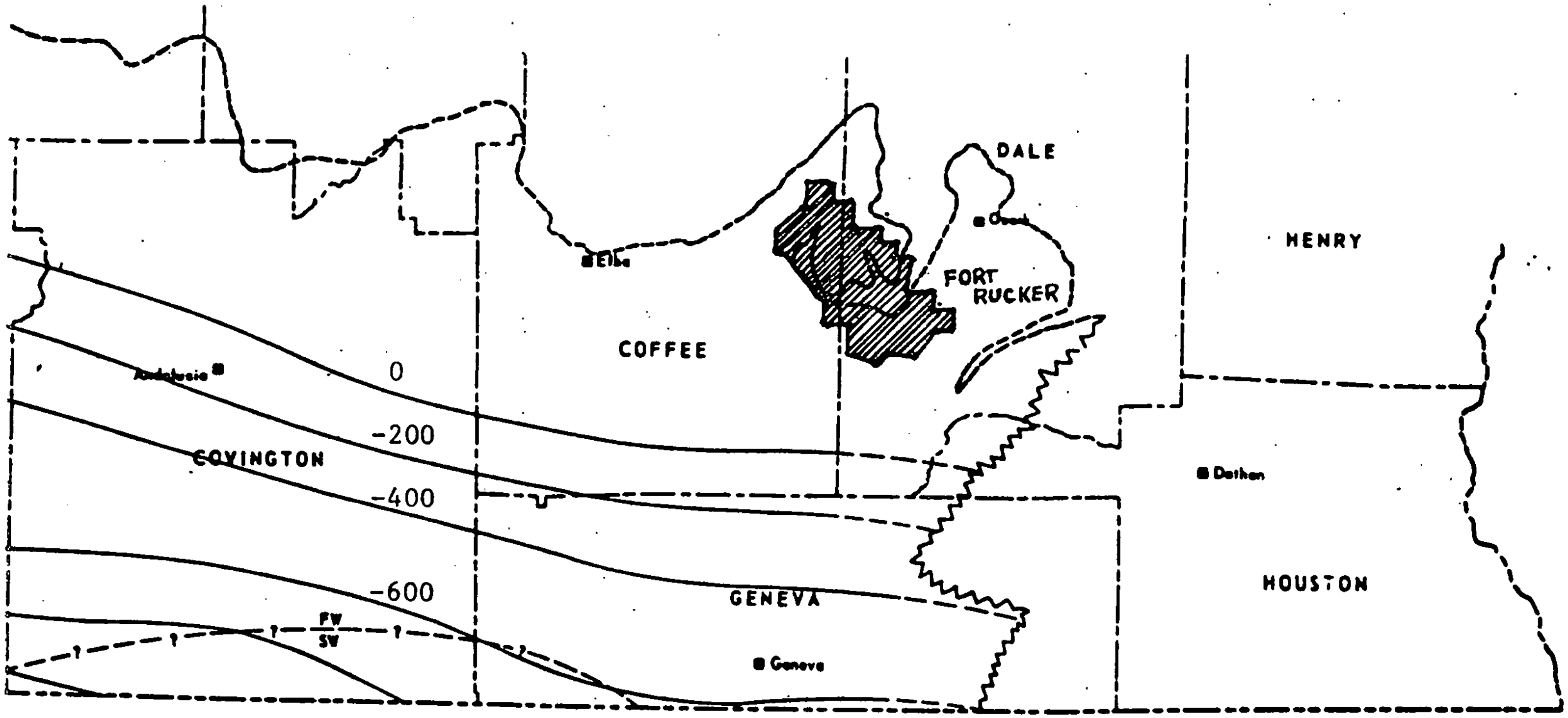


FIGURE 10-6. APPROXIMATE AREA OF RECHARGE OF THE UPPER WILCOX AQUIFER AND SELECTED WELL LOCATIONS.



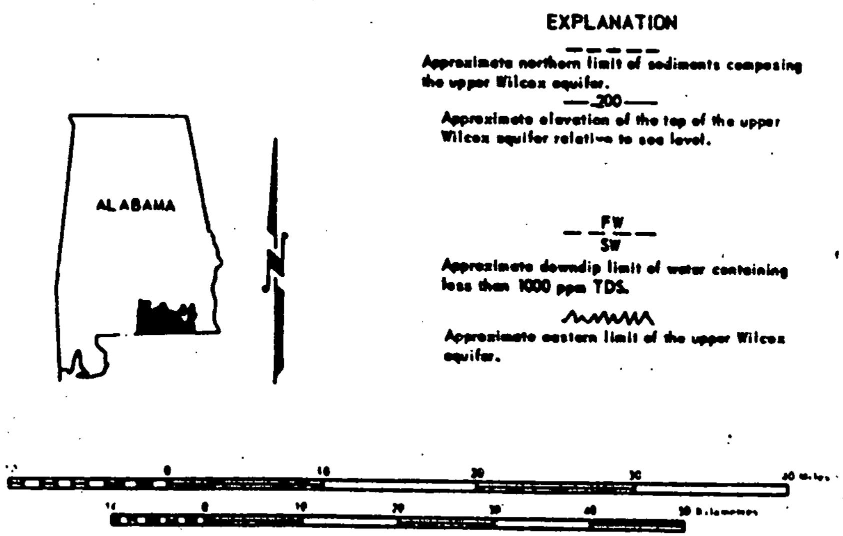
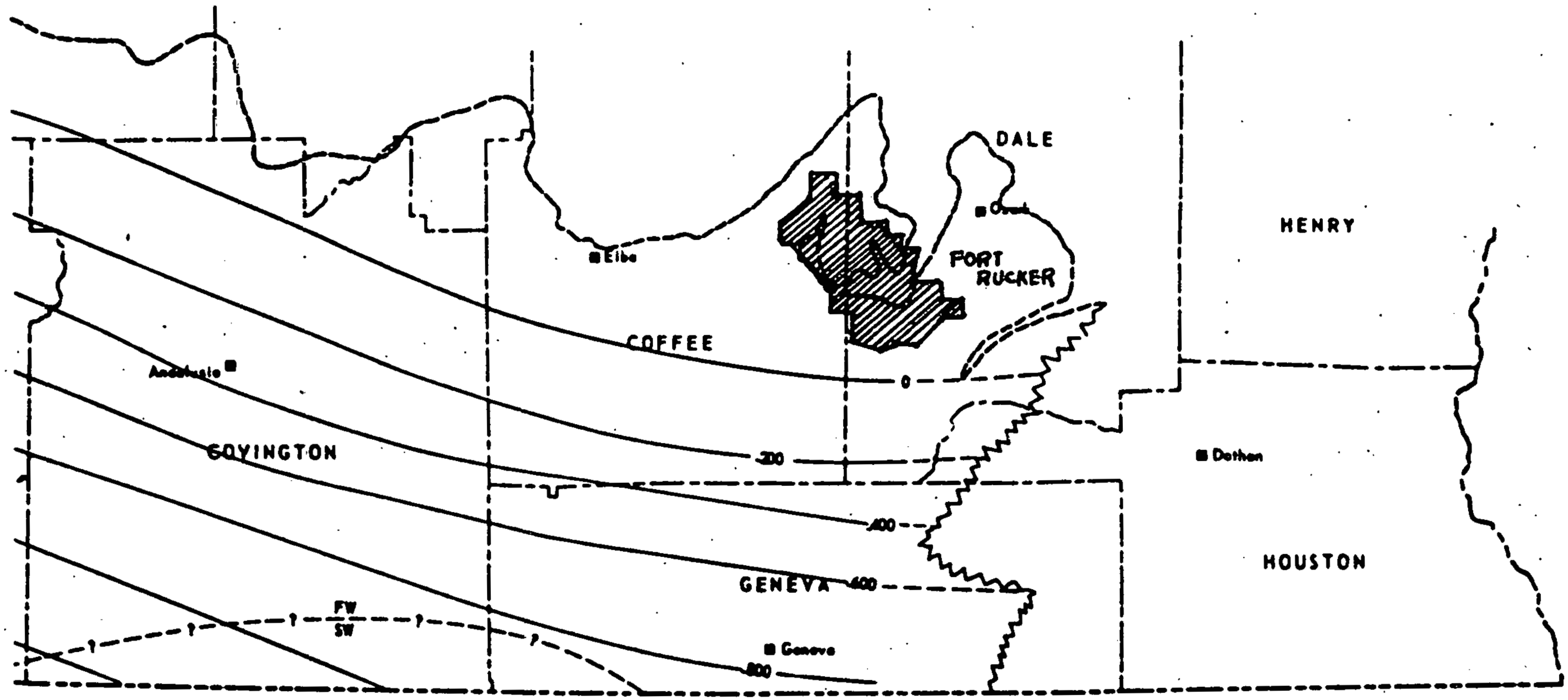


FIGURE 10-7. **CONTOUR MAP OF THE ESTIMATED TOP OF THE UPPER WILCOX AQUIFER.**



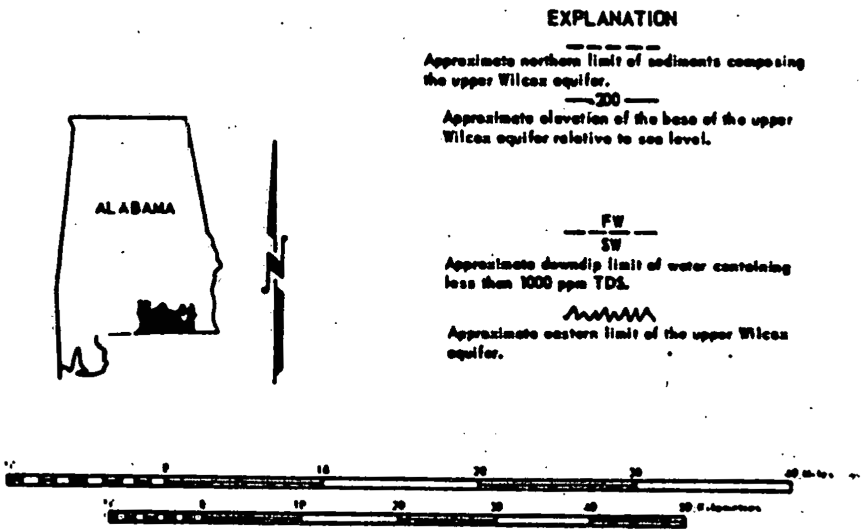
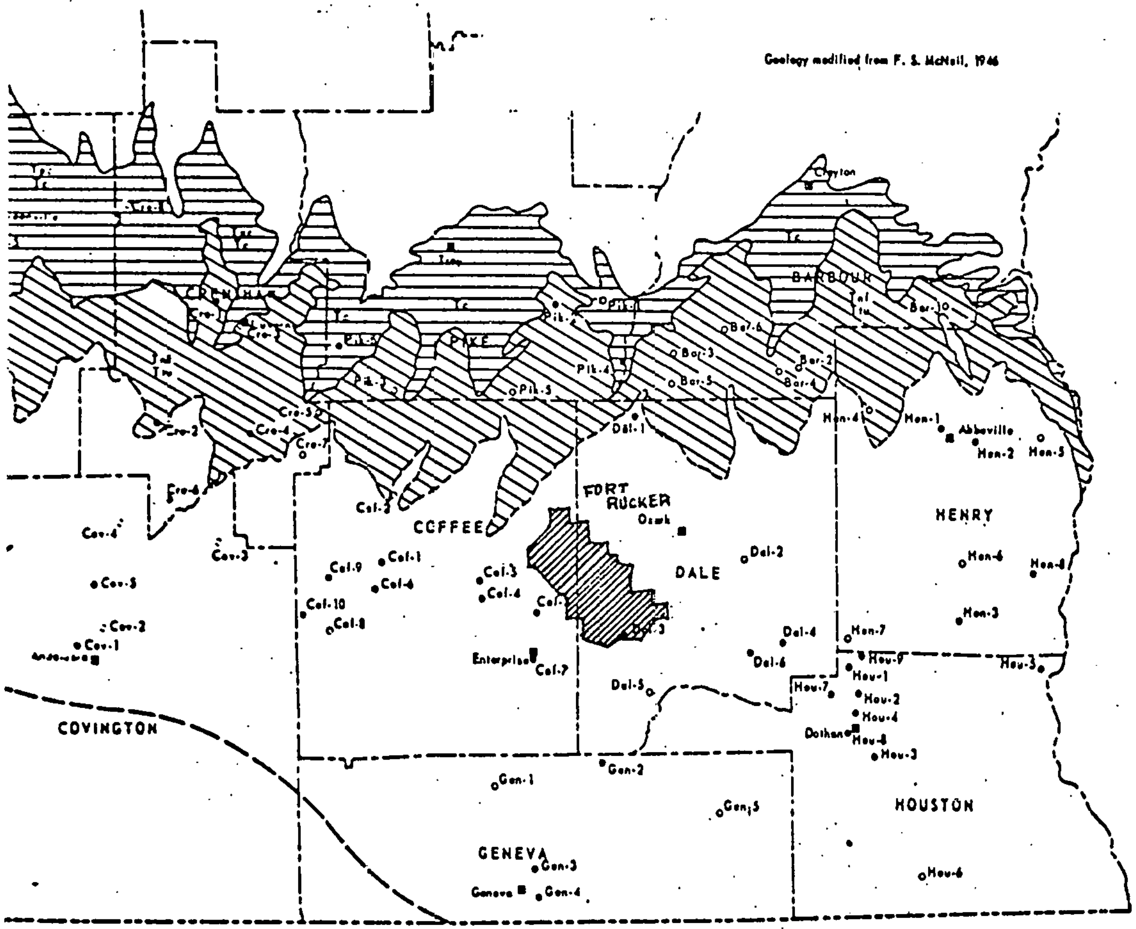


FIGURE 10-8. CONTOUR MAP OF ESTIMATED BASE OF THE UPPER WILCOX AQUIFER.

TABLE 10-3. Water Quality from Selected Wells Tapping the Upper Wilcox Aquifer.

Well Number	Water-Bearing Unit	Collection Date	Well Depth (feet)	Iron	Calcium	Magnesium	Sodium	Bicarbonate	Sulfate	Chloride	Fluoride	Nitrate	Dissolved Solids	pH
CV - 1	Ttu	1965	330	.10	---	---	---	110	---	5.8	--	--	---	7.3
CV - 2	Tth, Ttu	1965	485	.09	7.0	1.0	104	260	2.4	19.0	.2	.3	273	7.4
CF - 1	Ttu	1964	182	.13	---	---	---	172	---	3.4	--	--	---	7.7
CF - 2	Tth, Ttu	1964	430	.08	---	---	---	113	---	4.6	--	--	---	8.0
G - 1	Tth	1965	409	.09	---	---	---	172	---	7.0	--	--	---	7.7
D - 2	Ttu	1965	230	.07	---	---	---	172	---	11.0	--	--	---	8.2

Water-Bearing Units: Ttu - Tuscahoma Sand
Tth - Tallahatta and Hatchetigbee Formation Undifferentiated



EXPLANATION

		Formation(s)	Map Symbol	Geohydrologic description of area
TERTIARY SYSTEM–Eocene Series	Midway and Lower Wilcox Groups	Clayton, Porters Creek, Naheola	Tc, Tpc Tna	Approximate area underlain by strata of one or more of these formations that are predominantly clay; probably insignificant as a recharge area.
		Naheola	Tna	Approximate outcrop area of predominantly sandy strata of the Coal Bluff marl member of the Naheola; a potential area for storage and recharge to aquifer.
		Clayton, Porters Creek	Tc Tpc	Approximate outcrop area where formations contain significant amounts of permeable materials. Porters Creek strata, if present, are included in Clayton Formation east of Crenshaw County because of similar lithologies; a significant area for storage and recharge to aquifer.
		Nanafalia Formation and basal Tuscahoma Sand, undifferentiated	Ttu, Tnf	Outcrop area of Nanafalia Formation and approximate outcrop of basal, sandy strata of Tuscahoma Sand. Dashed boundary is approximate southern limit of basal Tuscahoma sands; a significant area for storage and recharge to aquifer.

Index to symbols:

Tc, Clayton Formation

Tpc, Porters Creek Formation

Tna, Naheola Formation

Tnf, Nanafalia Formation

Ttu, Tuscahoma Sand

Cov-1 Well for which a chemical analysis and capacity or yield data are shown in Tables 4 and 5.

Cov-2 Well for which a chemical analysis is shown in table ; yield data unavailable.

Line delineating the approximate southern extent of water in aquifer containing 1000 mg/l dissolved solids or less.

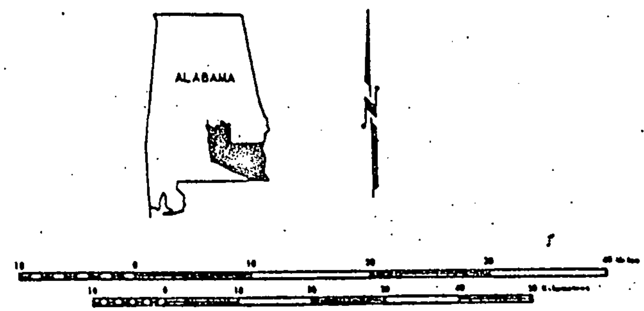


FIGURE 10-9. OUTCROP AREA OF MIDWAY-LOWER WILCOX AQUIFER AND LOCATIONS OF SELECTED WELLS.

In general, the quality of water from the Midway-Lower Wilcox Aquifer is similar to that of water of the Upper Wilcox and Lisbon Aquifers. Chemical analyses from selected wells producing from the Midway-Lower Wilcox Aquifer are listed in Table 10-4. The freshwater-saltwater contact is shown on Figure 10-9.

Yields of wells tapping the Midway-Lower Wilcox Aquifer range from 20 to 1,000 gallons per minute. Specific capacity averages 8.0 gallons per minute per ft. of drawdown. Table 10-5 is a listing of specific capacity and yield data for selected wells. Transmissivity values are estimated at 5,000 to 14,000 gallons per day per ft.

Data shown in Table 10-5 indicate that wells 750 ft. in depth should yield 800 gpm. These wells penetrate aquifers slightly over 250 ft. in thickness and should be capable of yielding a flow rate of 3 gpm/ft. or 4.3×10^3 gal./day/ft. The corrected flow rate as shown in Figure 9-1 is 2.9×10^4 gal./day/ft. The ratio of $t_{thermal}$ to t_{water} is 4:0. A time of three months is required for a 1-year transient temperature front to move between wells as full flow rate. Figure 11-8 indicates a spacing of 500 ft. is required between wells.

In summary, calculations for the Fort Rucker project employ wells 750 ft. in depth producing 800 gpm each. Estimating K between 100 and 500 gal./day/ft.^{2}, the head loss between wells is 20 to 40 ft. as shown in Figure 11-3.

X.C. Proposed System

The roads which serve Allen Heights community have the effect of grouping the community into 3 more or less equal divisions. These divisions have been labeled A, B and C on Figure 10-10. The divisions consist respectively of 101, 75 and 103 buildings. As a demonstration site, any one or all three divisions could be converted to the proposed system. An alternative would be to convert two divisions to different alternatives of the proposed system

TABLE 10-4. Chemical Analyses of Water from Wells Tapping the Midway-Lower Wilcox Aquifer (in ppm).

Well Number	Date of Collection	Well Depth (feet)	Water-Bearing Units	Iron	Calcium	Magne-sium	Bicar-bonate	Sulfate	Chlo-ride	Fluo-ride	Dis-solved Solids	pH
COF - 1	04/02/64	731	Tc	0.23	--	--	228	--	6.0	--	---	7.7
COF - 2	07/08/64	245	Tnf,Tc	0.19	--	--	248	--	5.6	--	---	7.9
COF - 3	07/21/64	640	Tc	0.70	--	--	168	--	6.4	--	---	8.2
COF - 4	04/02/64	537	Tc	0.29	--	--	178	--	4.0	--	---	7.7
COF - 5	04/02/64	358	Tnf(?)	0.28	--	--	183	--	4.0	--	---	7.8
COF - 6	06/21/64	262	Ttu,Tnf,Tc(?)	0.22	--	--	176	--	3.8	--	---	7.9
COF - 7	02/01/64	775	Ttu,Tnf,Tc	0.20	--	--	168	--	3.0	--	---	7.8
DAL - 1	03/23/51	668	Tnf(?)	0.40	62.0	2.0	---	12.0	--	.00	209	7.8
DAL - 2	07/12/65	450	Ttu,Tnf,Tc	0.58	--	--	164	--	3.8	--	---	8.1
DAL - 3	07/16/65	624	Ttu,Tnf,Tc,Kp(?)	0.03	--	--	154	--	6.6	--	---	8.1
DAL - 4	04/05/51	552	Tnf	0.20	56.0	00	---	9.0	--	00	188	8.3
DAL - 5	07/18/65	394	Ttu(?),Tnf	0.09	--	--	152	28.0	--	--	---	7.9
DAL - 6	07/30/65	765	Tnf(?),Tc	0.20	20.0	5.0	---	10.0	5.0	0.20	221	8.5
GEN - 1	01/06/65	790	Tnf	1.70	--	--	147	--	6.6	--	---	8.4
GEN - 2	07/19/65	409	Ttu	0.09	--	--	172	--	7.0	--	---	7.7

Water-Bearing Units: Kp - Providence Aquifers; Tc - Clayton Formation; Tpc - Porters Creek Formation; Tna - Naheola Formation; Tnf - Nanafalia Formation; Ttu - Tuscahoma Sand

TABLE 10-5. Specific Capacity and Yield Data for Selected Wells Tapping the Midway-Lower Wilcox Aquifer.

Well Number	Total Depth (feet)	Productive Interval	Water-Bearing Unit	Reported Yield (gpm)	Specific Capacity
COF - 1	731	308-534	Tc	500	7.6
COF - 3	640	540-640	Tc	--	17.0
COF - 4	537	488-534	Tc	100	6.8
COF - 5	358	299-358	Tnf(?)	100	--
COF - 6	262	---	Ttu,Tnf,Tc(?)	*20	--
COF - 7	775	410-765	Tnf,Tc	250	9.0
COF - 9	787	597-787	Tnf,Tc	250	9.0
COF -10	913	684-913	Tnf(?),Tc	1,000	14.0
DAL - 1	668	272-668	Tnf(?),Tc	90	7.5
DAL - 3	624	105-620	Ttu,Tnf,Tc	892	16.0
DAL - 4	552	506-547	Tnf	80	1.8
DAL - 6	765	705-755	Tnf(?),Tc	100	3.8
GEN - 2	409	180-409	Ttu	*7	--
GEN - 3	1,300	---	Tnf	*1,160	--
GEN - 9	1,050	890-1,040	Tnf	500	11.1

*Flowing well

Water-Bearing Units: Tc - Clayton Formation
Tpc - Porters Creek Formation
Tnf - Nanafalia Formation
Ttu - Tuscahoma Formation

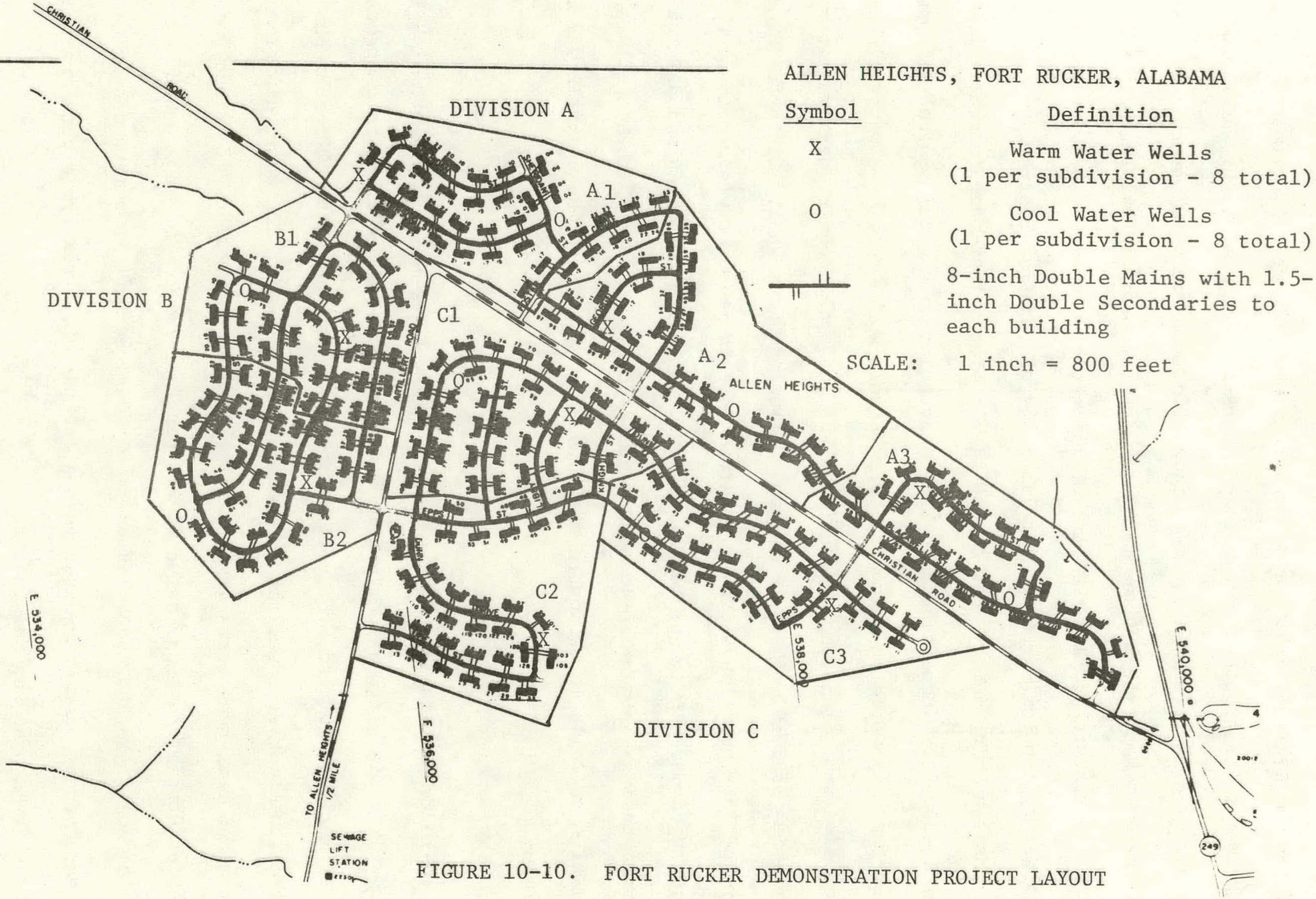


FIGURE 10-10. FORT RUCKER DEMONSTRATION PROJECT LAYOUT

and use the third division as a control group. Figure 1-1 serves as the schematic diagram for the proposed system at Ft. Rucker. Each building, which contains two units, will be served by one pair of 1.5 inch secondary pipes. The estimates displayed in Table 10-6 illustrates the cost of retrofitting each division (A, B and C) separately, plus the cost of interconnecting (x) the divisions to form a larger system for purposes of reliability and aquifer control.

Each housing unit is now cooled with a single four-ton air source heat pump (central air conditioner). With conversion to a water source heat pump, a three-ton unit would be adequate. Such a unit would require a water flow rate of up to five gpm. Thus the respective divisions would require maximum water flow capacities of 1010, 750 and 1030 gpm. Allowing for a safety factor of 2, three pairs of 750-gpm wells would be required in each of the divisions A and C and two pairs in division B. Each well is assumed to be 800 feet deep and drilled for 12-inch casing. Figure 10-10 shows the proposed location of those wells. The cost of such wells is estimated at \$20,600 per well based on Campbell and Lehr's equation[28] of:

$$\text{W. C.} = 850 \times \text{Depth}^{0.373}$$

for 12-15 inch wells in sand or gravel. Since that equation was applicable to 1966 well costs, figures derived from its use have been doubled to compensate for inflation. Further, the cost of the pump (750 gpm with a 275-foot head) and control unit for each well is estimated to be \$10,000. The water distribution system and heat pump costs were the same as used in section IV, except for an added street crossing charge of \$10 per foot for 1.5 in. pipe and \$15 per foot for 8 in. pipe. No heat addition or reject devices have been included in the estimated costs, nor do costs estimated include the necessary expense for engineering design and supervision.

Table 10-6. Residential Retrofit Costs

Allen Heights, Fort Rucker, Alabama

SYSTEM COMPONENTS	STATISTICS Divisions A	B	C	X	Total	COSTS X $ 1,000 Divisions A	B	C	X	Total
Water Supply Subsystem										
Wells (800 ft. x 12 in) @ $20,600	6	4	6	-	16	123.6	82.4	123.6	-	$ 329.6
Pumps + Contols @ $10,000	6	4	6	-	16	60.0	40.0	60.0	-	$ 160.0
Water Distribution Subsystem										
Double 8 in. PVC @ $15.98/ft.	9,500 ft.	7,250 ft.	10,250 ft.	1,600 ft	28,700 ft.	151.8	115.9	165.4	25.6	$ 458.7
Street Crossings @ $15/ft	9 275 ft.	4 100 ft.	8 200 ft.	4 160 ft.	685 ft.	3.4	1.5	3.0	2.4	$ 10.3
Double 1.5 in. PVC @ $4.88/ft.	5,100 ft	3,750 ft.	5,050 ft.	-	13,900 ft.	28.9	18.3	24.6	-	$ 71.8
Street Crossings @ $10/ft.	41 1,024 ft.	33 825 ft.	45 1,125 ft.	-	2,975 ft.	10.3	8.3	11.3	-	$ 29.9
Heat Pump Subsystem										
3-ton water source heat pump $2,000 installed	202	150	206		558	404.0	300.0	412.0	-	$1,116.0
Total Costs						782.0	566.4	799.9	28.0	$ 2,176.3

X.D. Expected Performance

The current energy usage per unit is 126×10^6 Btu per year,[29] of which 63.5×10^6 Btu are supplied by natural gas and used for space heating, hot water heating, cooking and clothes drying. The 62.5×10^6 Btu energy balance is supplied by electricity which is used for both lighting and air conditioning. The buildings' (two units) peak transfer loads are: gain 20.6×10^3 Btu/hr. and loss 64.1×10^3 Btu/hr. The peak building electrical consumption is 15.54 Kw. Table 10-7 was constructed assuming that 75% of the current energy consumption is used for space heating and cooling, an efficiency of 50% for the present gas furnaces and a COP of 2.0 for the air conditioners. The proposed system eliminates all on-site natural gas use for heating and reduces the electricity consumption from 13.7 to 10.4×10^3 Kw-hr. or 32% for heating and cooling. Table 10-7 also compares the energy usage by original sources based on the East South Central distribution.

X.E. Expected Economics

The latest available energy costs paid by Fort Rucker are:

Electric: \$0.01489 + 0.00318 (fuel adjustment) = \$0.01807 per Kw/hr.

Electric Demand Charge: \$4.60/Kw

Gas: \$1.051/mcf = 1.051/1.028 = $\$1.022/10^6$ Btu

Neglecting the electric demand charge, the annual per unit heating and cooling costs for the current and the proposed systems are \$297 and \$188. This affords the proposed system an annual savings of \$109 per unit or \$60,800 for the 558 unit community of Allen Heights. This annual savings is

$$\frac{\$\ 60{,}800}{\$2{,}176{,}300} \cdot 100\% = 2.79\%$$

of the original investment which yields a payback period of

$$1/2.79\% = 36 \text{ years.}$$

TABLE 10-7. Annual Energy Consumption per Unit Residential Retrofit, Allen Heights, Fort Rucker, Alabama

	CURRENT SYSTEM			PROPOSED SYSTEM		
	Heating	Cooling	Total	Heating	Cooling	Total
Load (10^6 Btu)	23.8	93.8	117.6	23.8	93.8	117.6
COP	0.5	2.0		3.13	3.37	
Required Energy (10^6 Btu)	47.6	46.9	94.5	7.6	27.8	35.4
Energy by end use source						
Natural Gas (10^6 Btu)			47.6			0.0
Electricity (10^3 Kw-hr)			13.7			10.4
Energy by original source						
Coal (5 tons)			5.2			4.0
Gas (MCF)			55.8			3.3
Oil (Bbls)			1.1			0.6
NUC (Grams)			35.1			26.6
Hydro (10^3 Kw-hr)			2.2			1.7

A payback period of 36 years is unreasonably long for the recommendation of the proposed system. However, several additional points should be considered:

1. The current energy costs are extremely low, but are expected to rise significantly.

2. A more efficient cooling cycle system could be designed using the modified water source heat pump.

3. The calculations do not consider the replacement or maintenance costs associated with the present system.

4. Electricity demand charges have been excluded.

5. Improved building insulation and temperature control have not been considered.

X.F. Institutional Considerations

All land and ground water use rights on Fort Rucker are property of the United States government. Funding for major modification of the Fort Rucker environmental control system requires congressional approval.

X.G. Projected Growth

The proposed demonstration project includes only one of three residential communities and occupies only about one-tenth of the buildup area of Fort Rucker. System expansion to the other residential communities and the other areas of Fort Rucker seems natural. A combination of residential communities presents an opportunity to include one or more schools in the system. The inclusion of schools, which are closed during the summer, reducing their cooling load would help balance the annual heating and cooling loads for the system.

XI. 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. A large part of the performance data normally listed in the Appendix is also included.

XI.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 values normally associated with standard construction. The walls contain 3½ in. of insulation. The ceiling contains 6 in. of insulation, and the floor contains 3½ in. of insulation. The basic insulation for this package is:

R-21 for ceiling insulation
R-14 for walls
R-27 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.[30] These are listed in Tables 3-10 and 3-11 in the Performance Section III. The degree-days are based on an equilibrium temperature of 65°F, allowing for a normal internal load. For larger buildings, equilibrium temperature, 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 since most chilling systems are water-source units, little direct analysis is made of these buildings. It is assumed the new variations in water temperatures will show a similar increase in performance in comparison to the basic 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 ambient temperature variations throughout the country (Appendix I). The input to the program for 24 cities is NOAA weather data showing average annual hours in 5^{o} F temperature increments (Table 11-1); 99% design temperatures, internal load and heat pump performance data (Tables 11-2 to 11-6); and the heating and cooling loads at ASHRAE design temperatures. Air-source heat pumps, A, B and C are respectively high, medium and low performance units. A variation in capital cost is associated with the performance. 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 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 11-5 and 11-6). It should be noted that energy for compressor heating and defrosting are not included in these figures. As a result, actual heating performance is slightly below these values. The use of twin units doubles pump capacities of air source heat pumps A and C which have been employed to indicate the performance

Table 11-1. Annual Hours in Five Degree Fahrenhiet 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
Tuscon, 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 11-2. Performance of Air-Source Heat Pump A*

COOLING

DBT[1]	WBT[2]	TOTAL CAP MBTUH	SENSIBLE CAP @ IDBT[3] 72	76	80	INPUT POWER[4] KW	COP[5]
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	60		70		75		80		
ODT	CAP MBTUH	KW[4]	CAP MBTUH	KW[4]	CAP MBTUH	KW[4]	CAP MBTUH	KW[4]	COP[5] @ 70°
-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

[1]Outdoor Dry Bulb Temperature, °F

[2]Indoor Wet Bulb Temperature, °F

[3]Indoor Dry Bulb Temperature, °F

[4]Power requirement for compressor, outdoor fan, and indoor fan, KW

[5]Coefficient of Performance = Capacity (Kw)/Input Power[4] (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 11-3. Performance of Air-Source Heat Pump B*

COOLING

DBT[1]	WBT[2]	TOTAL CAP MBTUH	SENSIBLE CAP @ IDBT[3] 72	76	80	INPUT POWER[4] KW	COP[5]
	59	29.4	29.4	29.4	--	4.05	2.13
85	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
	59	28.4	28.4	--	--	4.36	1.91
95	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
	59	25.6	25.6	--	--	4.54	1.65
105	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
	59	22.7	--	--	--	4.64	1.43
115	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	60		70		75		80		
ODT	CAP MBTUH	KW[4]	CAP MBTUH	KW[4]	CAP MBTUH	KW[4]	CAP MBTUH	KW[4]	COP[5] @ 70°
-18	10.7	2.46	10.4	2.46	10.4	2.46	9.4	2.46	1.24
-08	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

[1]Outdoor Dry Bulb Temperature, °F

[2]Indoor Wet Bulb Temperature, °F

[3]Indoor Dry Bulb Temperature, °F

[4]Power requirement for compressor, outdoor fan, and indoor fan, KW

[5]Coefficient of Performance = Capacity (Kw)/Input Power[4] (Kw)

*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 11-4. Performance of Air-Source Heat Pump C*

COOLING

DBT[1]	WBT[2]	TOTAL CAP MBTUH	SENSIBLE CAP @ IDBT[3] 72	76	80	INPUT POWER[4] KW	COP[5]
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	60		70		75		80		
ODT	CAP MBTUH	KW[4]	CAP MBTUH	KW[4]	CAP MBTUH	KW[4]	CAP MBTUH	KW[4]	COP[5] @ 70°
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

[1]Outdoor Dry Bulb Temperature, °F

[2]Indoor Wet Bulb Temperature, °F

[3]Indoor Dry Bulb Temperature, °F

[4]Power requirement for compressor, outdoor fan, and indoor fan, KW

[5]Coefficient of Performance =Capacity (Kw)/Input Power[4](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 11-5. Annual Performance for Temperature Variation in Twenty-four Cities Utilizing High Performance Air-Source Heat Pump

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]	Cooling COP[3]	Heating KWH[1]	Heating KWH[2]	Heating COP[3]
Birmingham, AL	96	17	295	831	8928	2.73	3496	66	2.43
Tucson, 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 11-6. Annual Performance for Temperature Variation in Twenty-four Cities Utilizing Low Performance Air-Source Heat Pump

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]	Cooling COP[3]	Heating KWH[1]	Heating KWH[2]	Heating 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)

effects of decreased resistance heating (Tables 11-7 and 11-8). 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 the capital cost for these combinations are not economically feasible and they have been eliminated from further study. The next 2 studies use the twin unit A with increased internal load. These studies (Tables 11-9 and 11-10) give the performance effect of larger buildings where the ratio of internal load to maximum external load becomes larger. This has the effect of increasing the cooling load and decreasing the heating load. The variation can be used to match buildings in modules to obtain equal cooling and heating water requirements.

Individual run sheets are shown in Tables 11-11 to 11-16. Tables 11-11 to 11-13 demonstrate the variations for air source heat pump A at different locations using a standard internal load of 5000 Btu. Table 11-14 represents a run with the water source heat pump A. Tables 11-15 and 11-17 give the performance with larger internal loads to represent the performance for larger buildings where the ratio of internal load to maximum external load becomes larger. This has the effect of increasing the cooling load and decreasing the heating load. The 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 a [31] specific gravity of 0.9, results in higher heating value of the fuel or

$$\mathrm{HHV} = 19{,}000\ \mathrm{Btu/lb}$$

$$= 145{,}000\ \mathrm{Btu/gal}$$

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 11-7. 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]	Cooling COP[3]	Heating KWH[1]	Heating KWH[2]	Heating 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 11-8. 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 $70^{o}F$

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 ^{o}F	Heating Design Temp ^{o}F	Annual Heating Load $BTUx10^{-5}$	Annual Cooling Load $BTUx10^{-5}$	Cooling KWH[1]	Cooling COP[3]	Heating KWH[1]	Heating KWH[2]	Heating 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 11- 9. 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 Design Temp °F	Heating Design Temp °F	Annual Heating Load BTUx10^{-5}	Annual Cooling Load BTUx10^{-5}	Cooling KWH[1]	Cooling COP[3]	Heating KWH[1]	Heating KWH[2]	Heating COP[3]
Birmingham, AL	96	17	161	1152	12205	2.77	1986	0	2.37
Tucson, 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 11-10. 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]	Cooling COP[3]	Heating KWH[1]	Heating KWH[2]	Heating COP[3]
Birmingham, AL	96	17	25	1893	19676	2.82	347	0	2.14
Tucson, 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 11-11. Air-Source Heat Pump A
Performance Computer Run for Birmingham, Alabama

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 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= COEFICIENT 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.	2976019.	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.	6635385.	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.	488.	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	2.54	COOLING
107.	5.	172500.	34.5	4.53	100.0	23.	.0	.0	2.23	COOLING

ANNUAL HEATING LOAD, BTU = 29486508.
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 11-12. Air-Source Heat Pump A Performance Computer Run for Tucson, Arizona

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. OF104. 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= COEFICIENT 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.	689286.	27.1	3.60	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	660.	.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.	781.	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	79.8	1756.	.0	.0	2.50	COOLING
102.	357.	11665000.	35.4	4.43	93.9	1465.	.0	.0	2.34	COOLING
107.	152.	5244000.	34.5	4.53	100.0	689.	.0	.0	2.23	COOLING
112.	31.	1035500.	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 11-13. Air-Source Heat Pump A
Performance Computer Run for Milwaukee, Wisconsin

THIS PROGRAM WILL ANALYZE PERFORMANCE AT LOCATION 28,MILWAUKEE, WISCONSINFOR AIR 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 , 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= COEFICIENT 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 62% EFFICIENCY WOULD REQUIRE 819. THERMS.
FUEL OIL T AT 62% EFFICIENCY WOULD REQUIRE 565. GALLONS

Table 11-14. Water-Source Heat Pump A
Performance Computer Run for Milwaukee, Wisconsin

THIS PROGRAM WILL ANALYZE PERFORMANCE AT LOCATION 28,MILWAUKEE, WISCONSINFOR 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 , 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= COEFICIENT 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

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 11-15. Double Capacity Air-Source Heat Pump A
Performance Computer Run for Milwaukee, Wisconsin

THIS PROGRAM WILL ANALYZE PERFORMANCE AT LOCATION 28,MILWAUKEE, WISCONSINFOR 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 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= COEFICIENT 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

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 A[illegible]% EFFICIENCY WOULD REQUIRE 819. THERMS.
FUEL OIL H[illegible] AT 62% EFFICIENCY WOULD REQUIRE 565. GALLONS

Table 11-16. Double Capacity Air Source Heat Pump A Performance Computer Run for Milwaukee, Wisconsin with Increased Internal Load

THIS PROGRAM WILL ANALYZE PERFORMANCE AT LOCATION 28,MILWAUKEE, WISCONSINFOR TWIN AIR A HEATING/COOLING SYSTEM
CALCULATIONS ARE FOR INDOOR TEMPERATURE MAINTAINED AT 70. F AND INTERNAL LOAD OF 10000. 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 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= COEFICIENT 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 11-17. Double Capacity Air-Source Heat Pump A Performance Computer Run for Milwaukee Wisconsin with Large Internal Load

THIS PROGRAM WILL ANALYZE PERFORMANCE AT LOCATION 28,MILWAUKEE, WISCONSINFOR 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 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= COEFICIENT 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

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 3-1 and Tables 11-2 to 11-4. 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 heat pumps compared to air source heat pumps on the basis of ARI capacity are appreciably less.[12,13] 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. As a result, this heating deficit must be eliminated by resistance heating.

Exception is taken to Christian's estimate of maintenance cost.[12,13] Since 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 3 service calls were required the first year. All 3 calls were for burned-out fan motors. With over 700 units installed, service calls are averaging around 100 per year. This is less than 1 call per 7 units per year.

XI.B. Aquifer Analysis

The movement, occurrence and distribution of water below the surface is known as groundwater hydrology.[32,33] 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 1-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.[32] 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 primary 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 11-1.[34] 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.[32,33] 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 indicate that the flow is in the direction of the decreasing 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 well bore, is laminar flow.

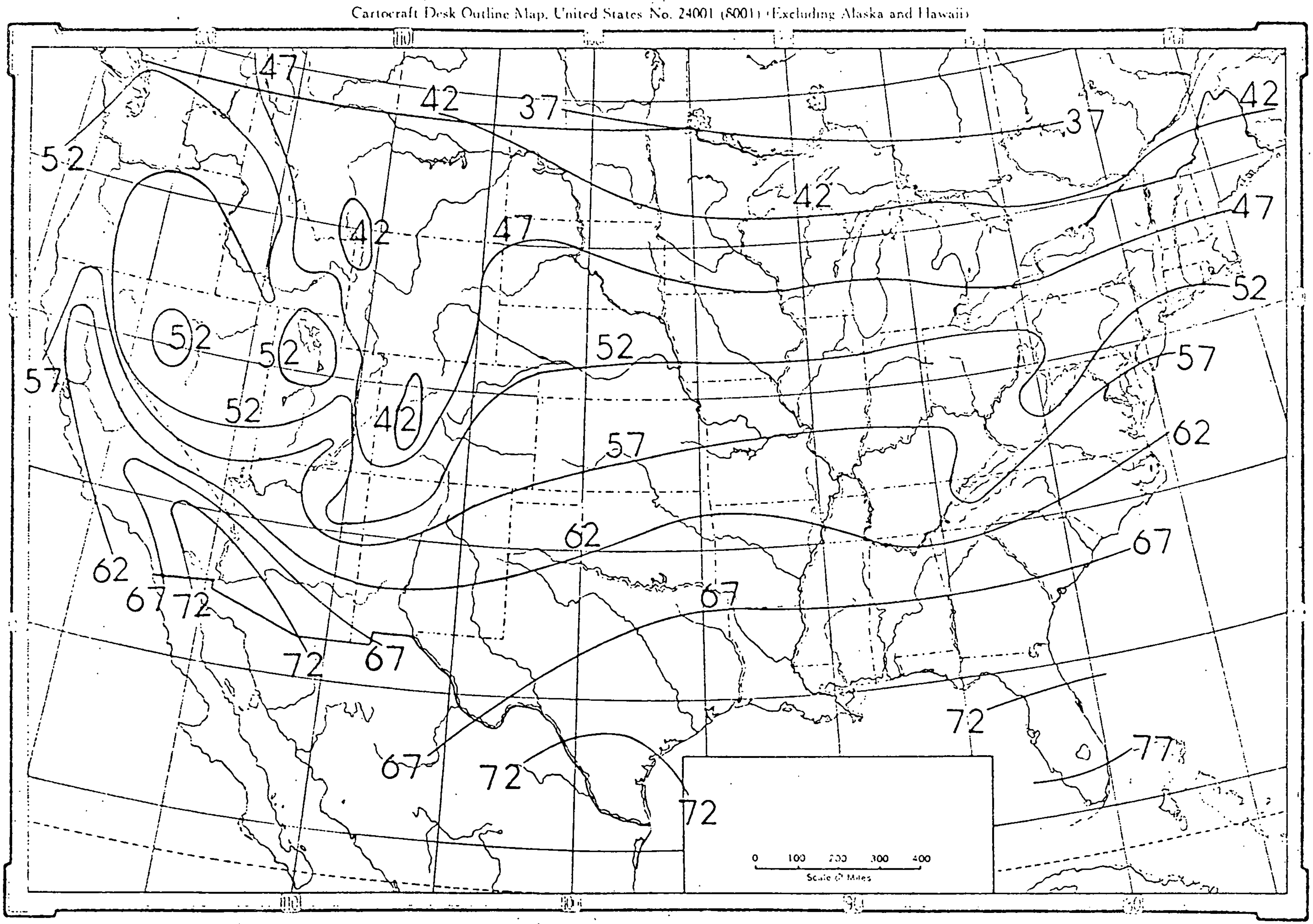


Figure 11-1. AVERAGE TEMPERATURES OF SHALLOW GROUND-WATER IN DEGREES FAHRENHEIT

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

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

where μ is fluid viscosity, and

λ is specific weight

The relationship between the specific permeability and the laboratory coefficient of permeability for water at 60°F is:

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

The values of permeability for different types of aquifers are shown in Figure 11-2.[32]

The rate of flow of groundwater is governed by permeability of the aquifer and the hydraulic gradient. Natural flow rates range between 5 ft./year to 5 ft./wk. Most aquifers considered for thermal storage will have natural flow rates of less than 15 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,

$$Q = AK\frac{dh}{dr} = 2\pi rbK\frac{dh}{dr}$$

where b is thickness of the water-bearing portion of well.[32] Rearranging and integrating for boundary conditions of $r = r_w$ and $h = h_w$ for conditions

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

10^6 10^5 10^4 10^3 10^2 10 1 10^{-1} 10^{-2} 10^{-3} 10^{-4}

Figure 11-2. LABORATORY COEFFICIENT OF PERMEABILITY, K_s, GAL./DAY ft.^{2} AT A HYDRAULIC GRADIENT OF 1 ft./ft. (Modified after Todd, 1959)

20.11.78

at the well and $r = r_o$ and $h = h_o$ at a distance from the well yields:

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

The head "h" varies linearally with the logarithm of the distance r for all flow rates. This equation is limited to a confined homogenous aquifer of uniform thickness. It is used as an approximate model of real aquifers. Figure 1-4 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 11-3 shows the pressure as related to head loss between a pair of wells based on these relationships for 6 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 x (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.

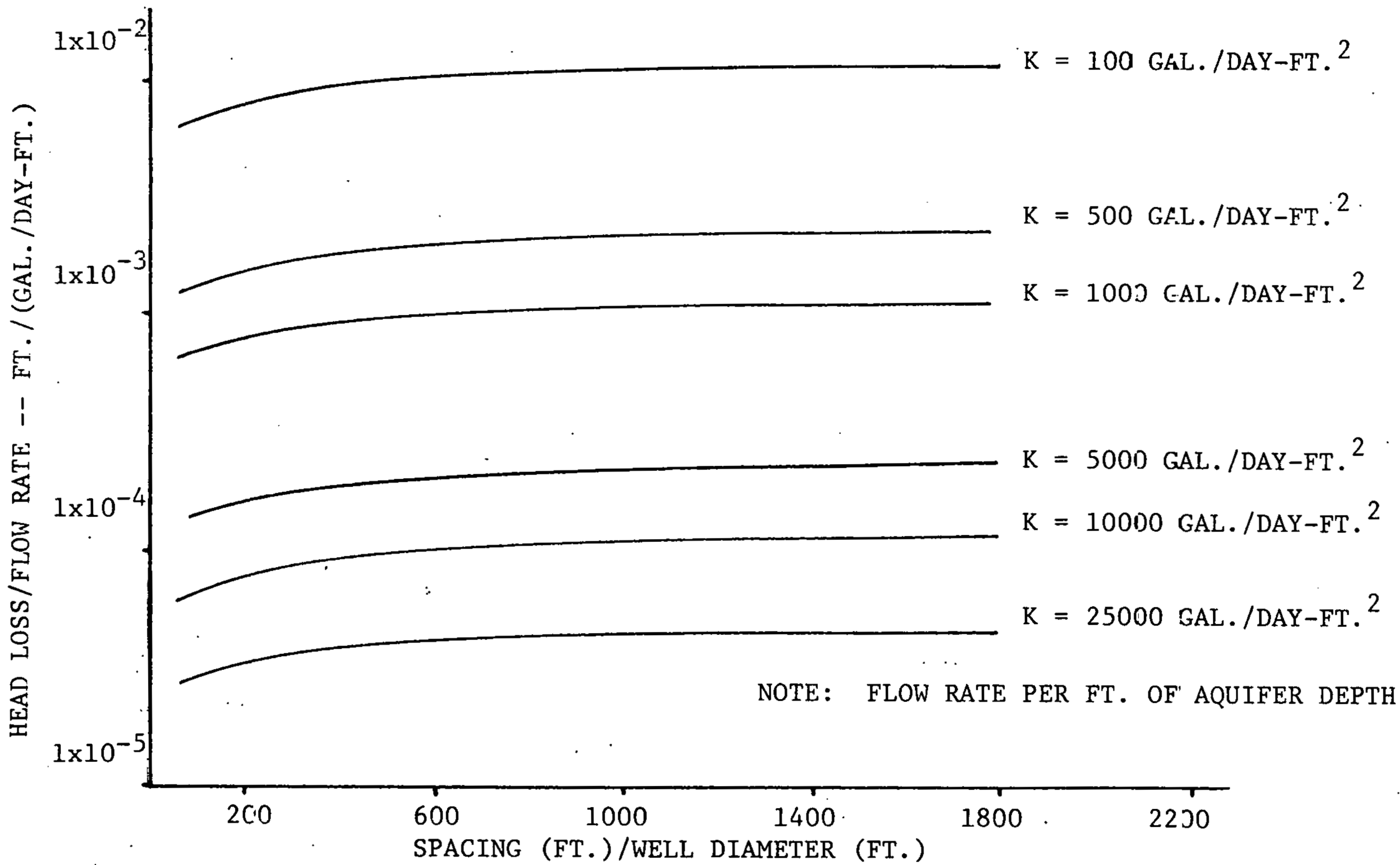


Figure 11-3. WELL PRESSURE DROP AS A FUNCTION OF SPACING AND PERMEABILITY

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 and field tests.[35,36,37] Flow patterns for a pair of wells, one injection and one withdrawal, are shown in Figures 11-4 to 11-7. These flow patterns have been developed assuming potential flow and matching constant stream function and potential function values.[38,39] The curves show visually the patterns of groundwater velocities under various conditions. Figure 11-4 includes only the flow from the wells. Figures 11-5 to 11-7 have imposed natural flows. The natural ground water flow velocity has been arbitrarily chosen so its magnitude is equal to one-half the pre-existing velocity from water injection and withdrawal at the centerpoint between the wells. 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. 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 11-9. By integrating the velocity (function of flow rate) with respect to time along the center lines between the 2 wells, a relationship between distance, time and flow rate can be established. Using the coordinates in Figure 11-9 the velocity at any point along the x axis is:

$$V = \frac{Q}{2\pi(r_o+x)} + \frac{Q}{2\pi(r_o-x)}$$

$$= \frac{Q}{\pi}\ \frac{r_o}{r_o^2-x^2}$$

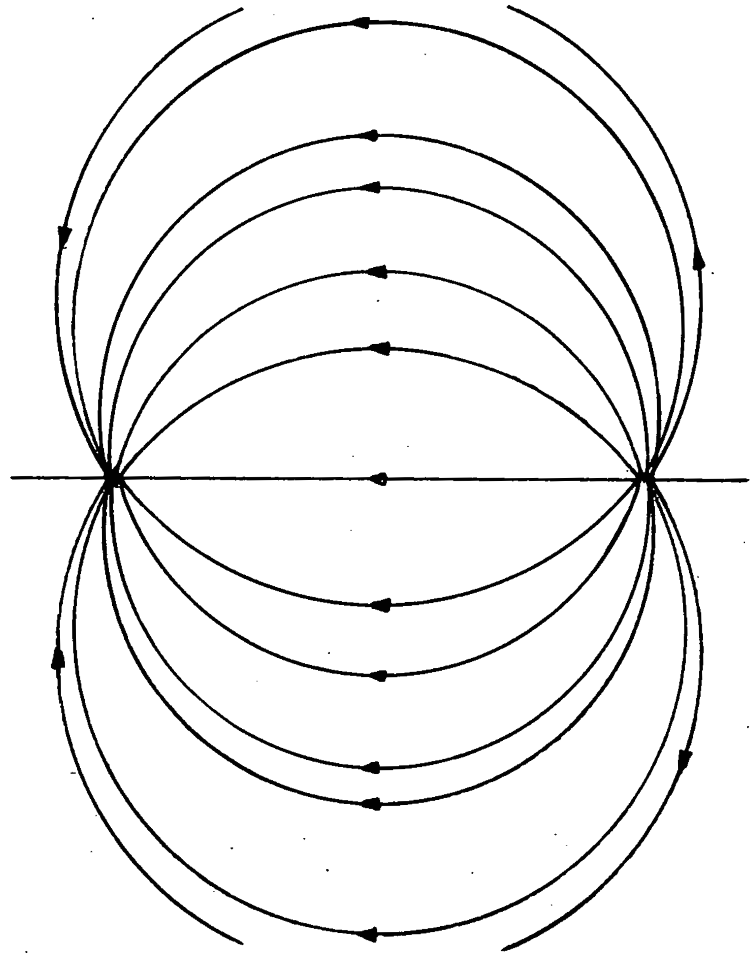


Figure 11-4. IDEAL STREAMLINE FOR INJECTION WELL AND DISCHARGE WELL WITH EQUAL FLOW RATES WITHOUT GROUND FLOW

4.12.78

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

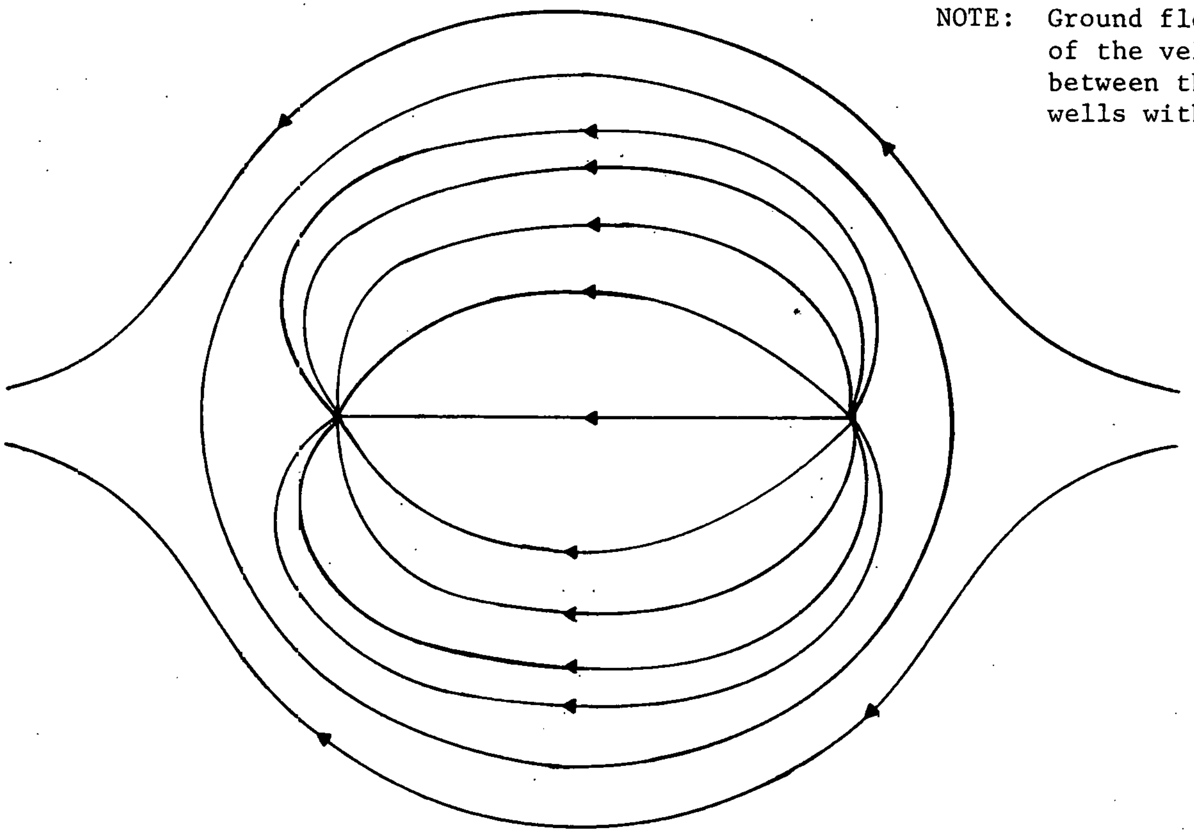


Figure 11-5. STREAMLINES FOR INJECTION AND DISCHARGE WELLS WITH EQUAL FLOW AND GROUND FLOW IN THE SAME DIRECTION

20-11-78

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

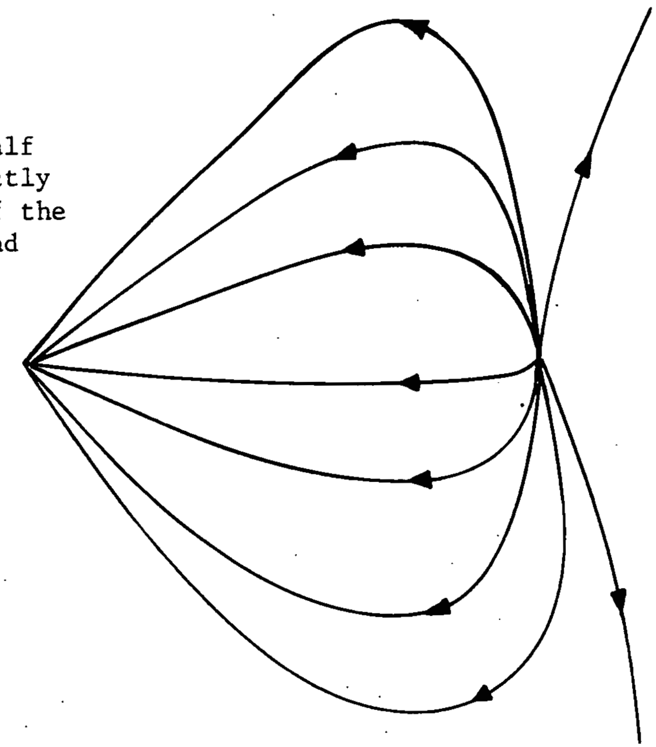


Figure 11-6. STREAMLINES FOR INJECTION AND DISCHARGE WELLS WITH EQUAL FLOW AND GROUND FLOW PERPENDICULAR TO THE FLOW

20.11.78

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

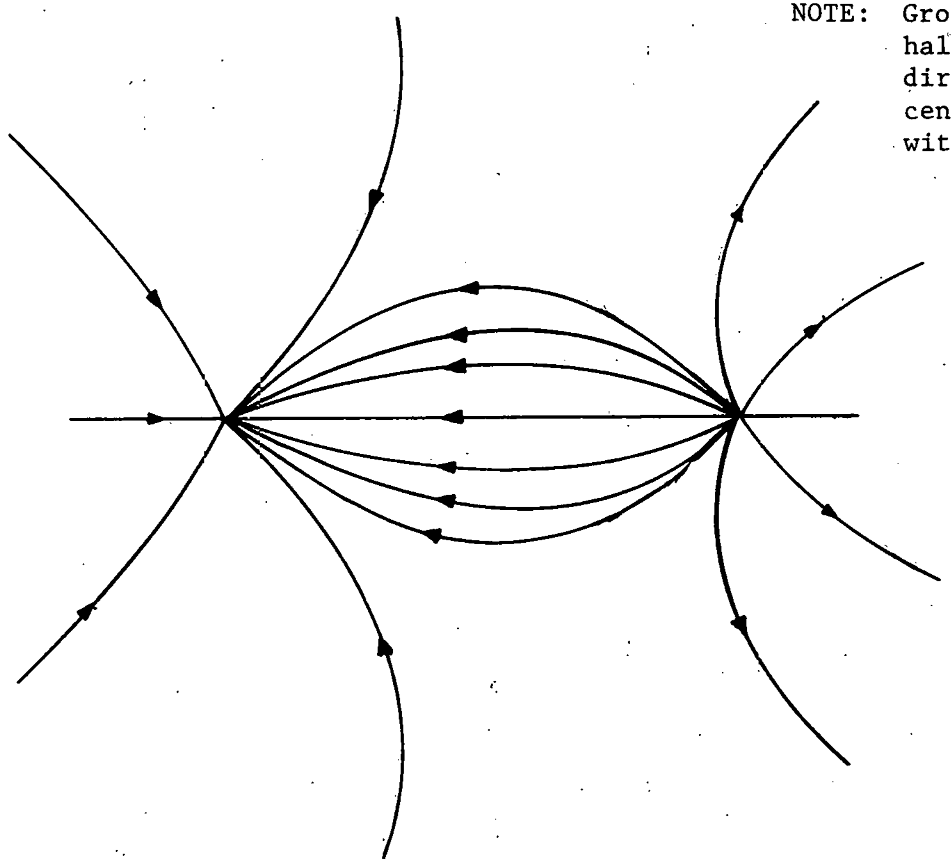


Figure 11-7. STREAMLINES FOR INJECTION AND DISCHARGE WELLS WITH EQUAL FLOW AND GROUND FLOW OPPOSING

20·11·78

The time required to traverse from $-r_o$ to r_o is:

$$t = {}^{r_o}\frac{dx}{v} = \frac{4\pi r_o^2}{3Q} = \frac{4\pi R^2}{3Q}$$

Figures 11-8 and 11-9 give 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 appreciably greater. Velocity will increase by the ratio of void per unit of cross-sectional area. It is assumed that the ratio which relates to porosity defined on a volume basis is equal to the ratio on a cross-sectional 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. through 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:

Corrected Flow Rate		
$2.88(10^3)$ 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 $\frac{1}{2}^{\circ}$F 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 as the 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:

$$V_{Thermal} = \frac{\rho_{H_2O} C_{p_{H_2O}} \phi}{\rho_{H_2O} C_{p_{H_2O}} \phi + \rho_{rock} C_{p_{rock}} (1-\phi)} V_{H_2O}$$

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 Figures 11-8 and 11-9 change as:

$$t_{Thermal} = \frac{\rho_{H_2O} C_{p_{H_2O}} \phi + \rho_{rock} C_{p_{rock}} (1-\phi)}{\rho_{H_2O} C_{p_{H_2O}} \phi} t_{H_2O}$$

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 time.

The ratio of $t_{Thermal}$ to t_{Water} 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
3.08 for a porosity of 0.2
2.21 for a porosity of 0.3
1.76 for a porosity of 0.4

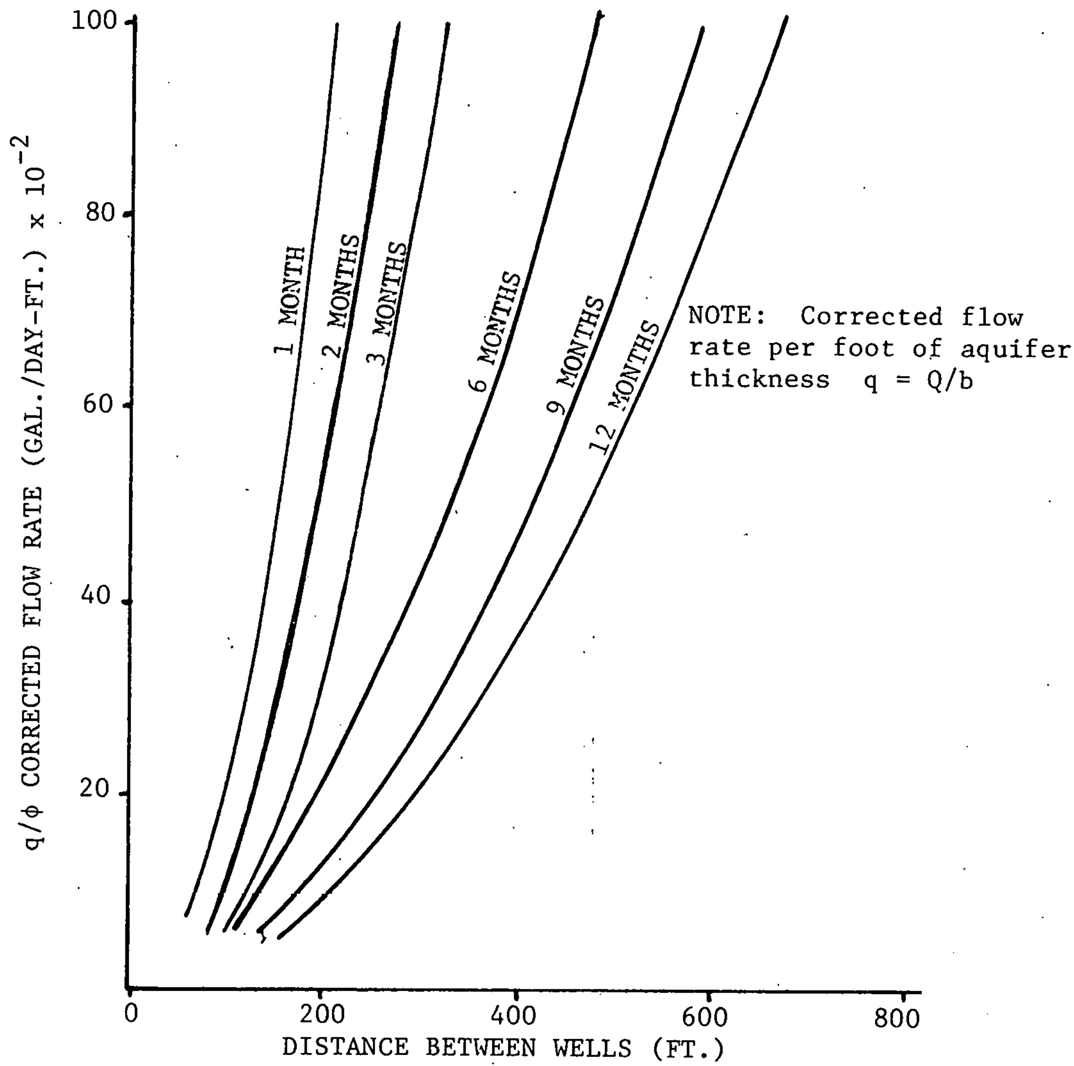


Figure 11-8. TIME REQUIRED FOR WATER FRONT TO PASS BETWEEN WELLS

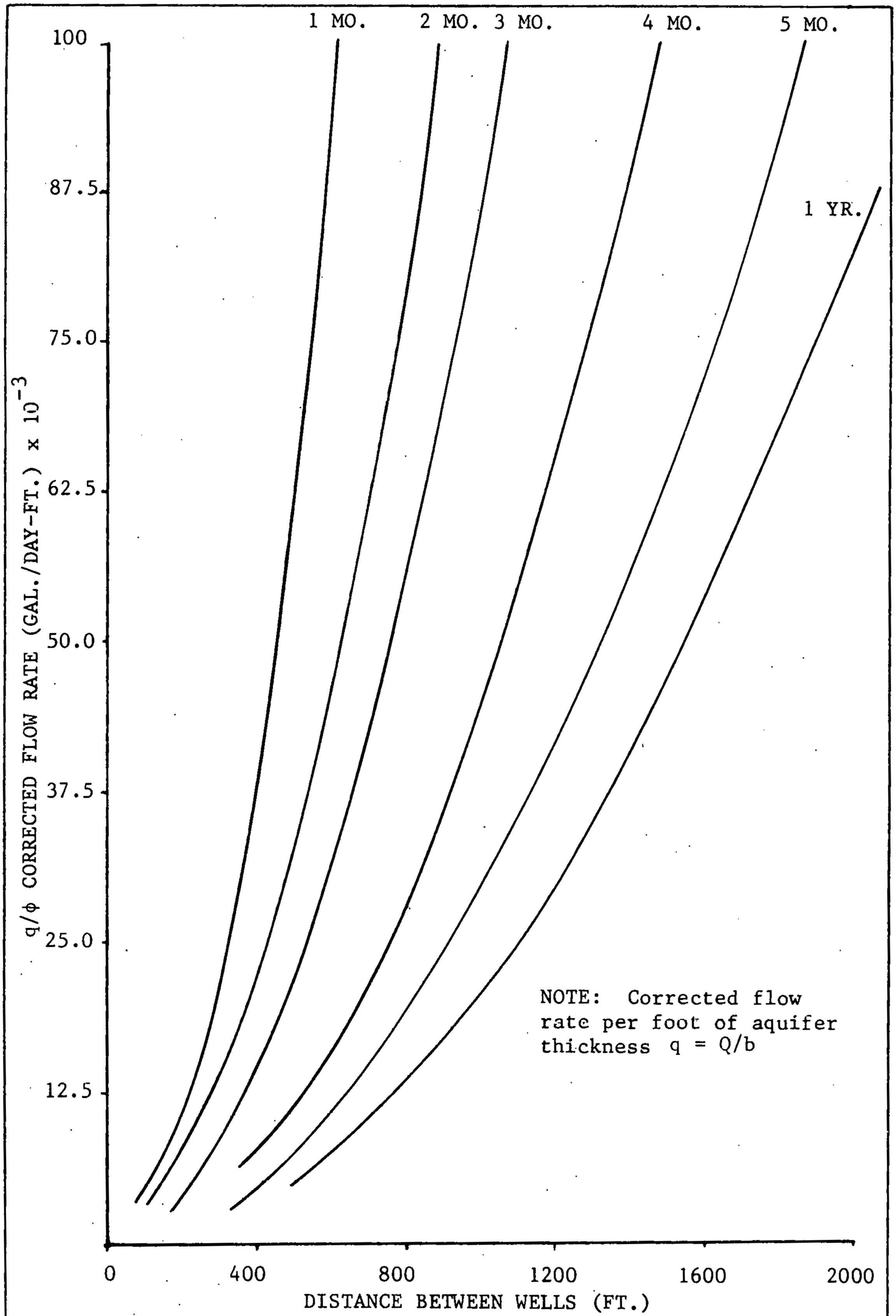


Figure 11-9. TIME REQUIRED FOR WATER FRONT TO PASS BETWEEN WELLS

or a 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.

Figure 11-10 shows the streamlines leaving the source and the location of the thermal front at the time the front reaches the sink. The computer program which follows the streamlines and calculates the times is described in Appendix III. The area inside this curve equals the area available for thermal storage. Integrating the area under the curve results:

$$A = 1.05\ R^2$$

where R is equal to the distance between wells. The volume available for thermal storage equals the above area times the aquifer thickness (b) or:

$$Vol = A\ b$$

Figure 11-11 gives the area available for thermal storage as a function of well spacing. For the example wells 500 ft. apart in a 100 ft. thick aquifer the volume is:

$$Vol = 1.05\ (500^2)\ ft.^2 \times 100\ ft. = 2.6 \times 10^7\ ft.^3$$

The storage capacity assuming 10% porosity and a 20°F temperature change is 9.9×10^9 Btu.

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 11-12 is a map of Alabama depicting 5 formations suited for supply wells

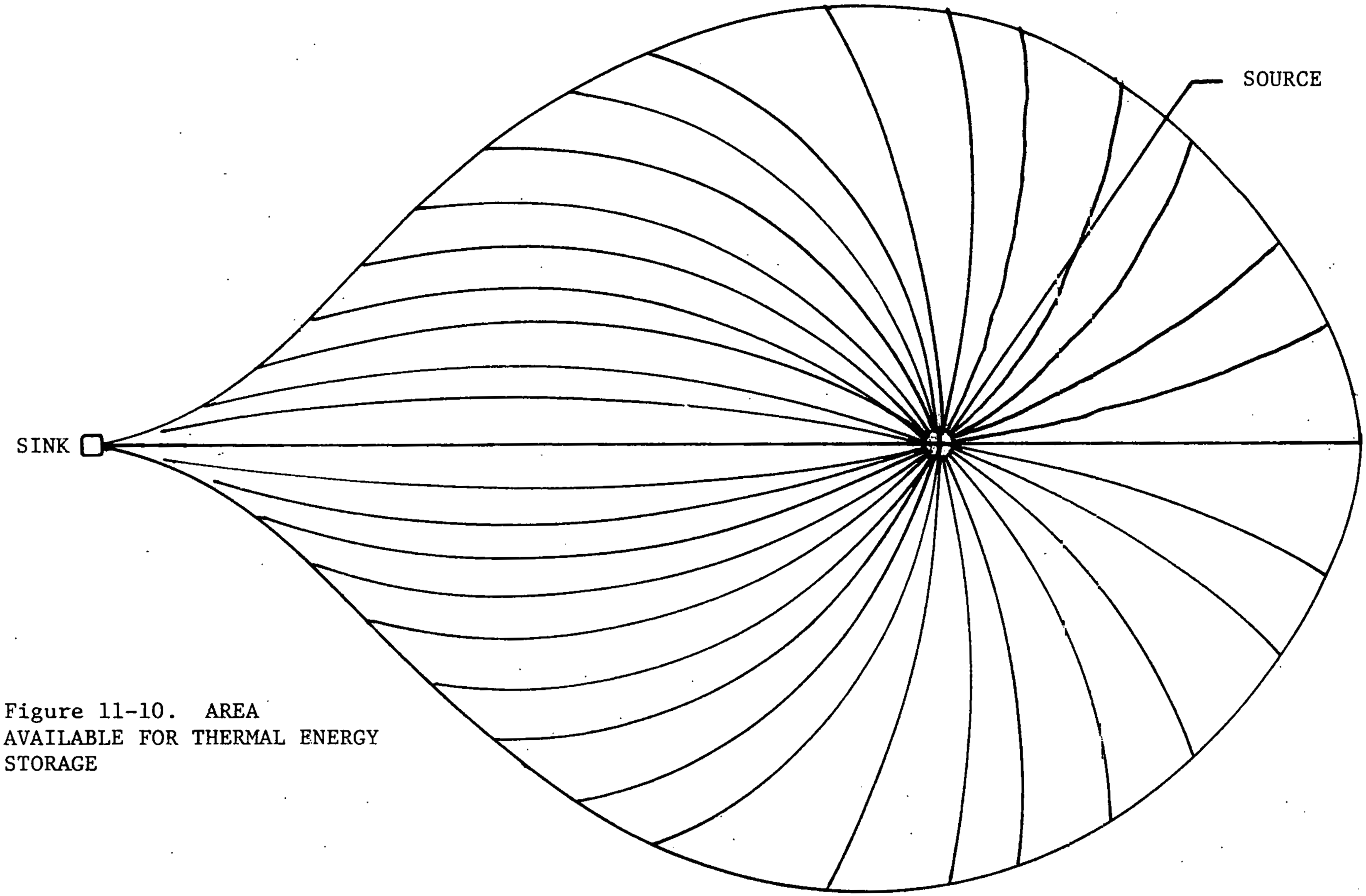


Figure 11-10. AREA AVAILABLE FOR THERMAL ENERGY STORAGE

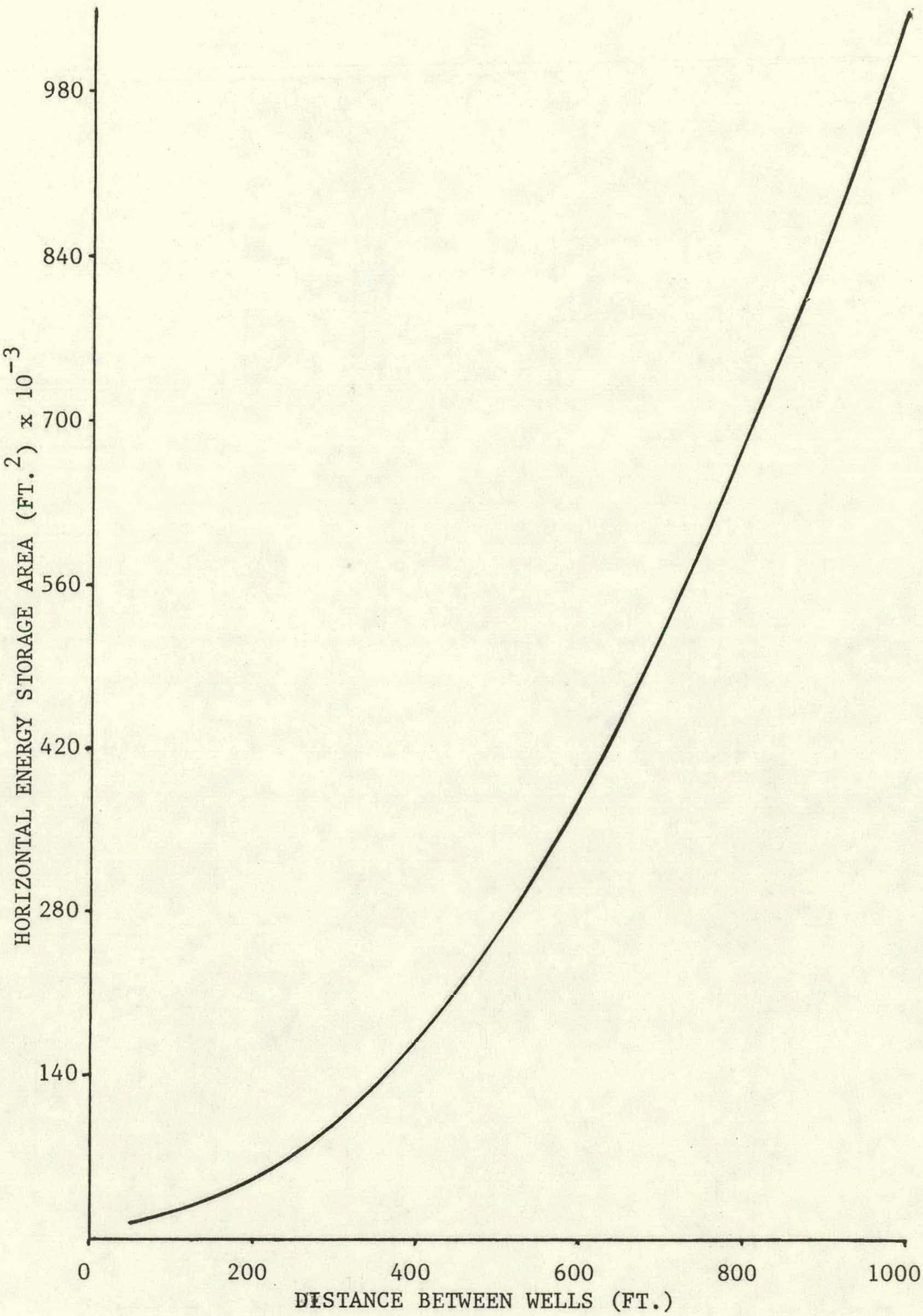


Figure 11-11. HORIZONTAL ENERGY STORAGE AREA

3.2.79

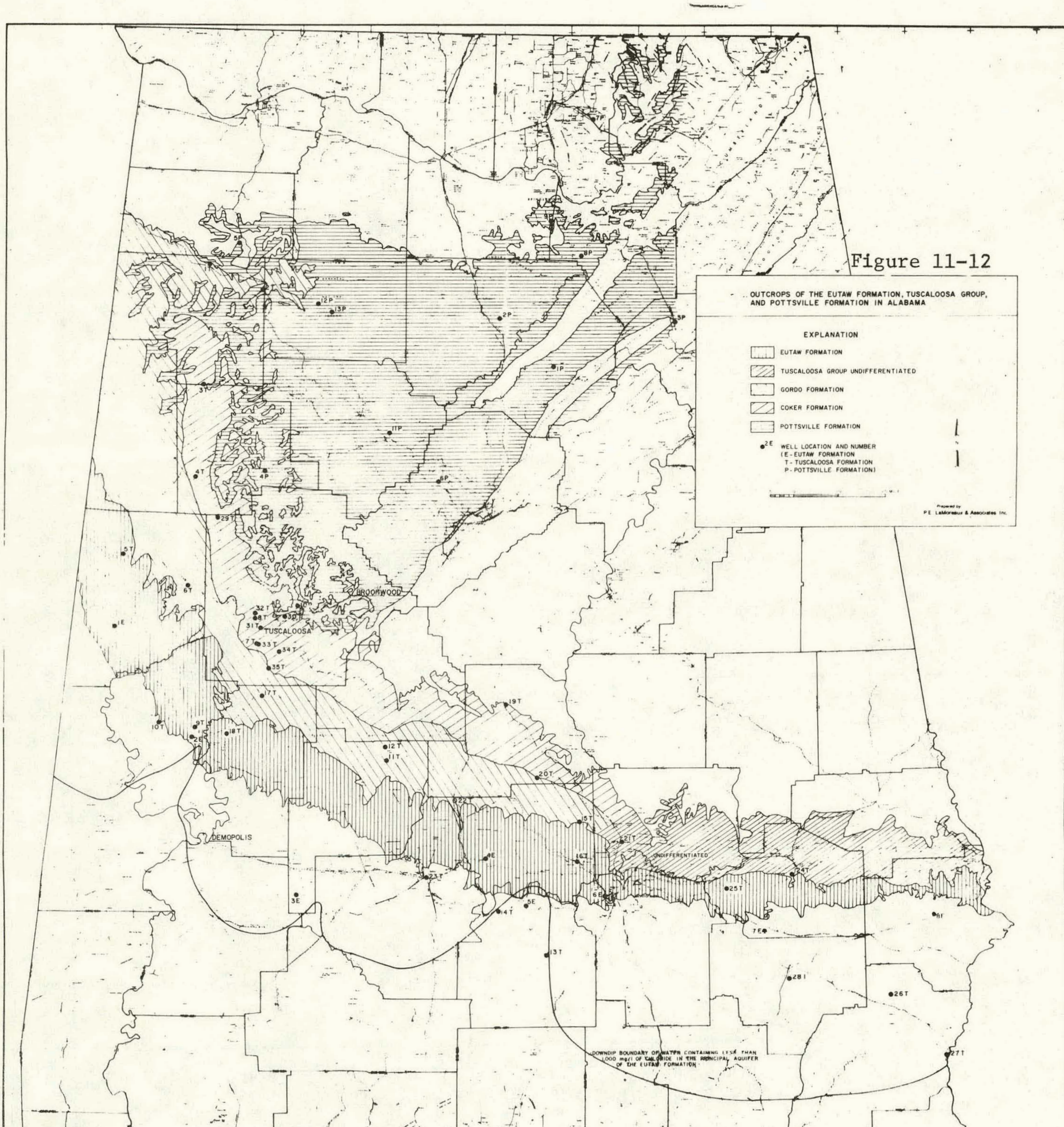


Figure 11-12
OUTCROPS OF THE EUTAW FORMATION, TUSCALOOSA GROUP, AND POTTSVILLE FORMATION IN ALABAMA
EXPLANATION
EUTAW FORMATION
TUSCALOOSA GROUP UNDIFFERENTIATED
GORDO FORMATION
COKER FORMATION
POTTSVILLE FORMATION
2E WELL LOCATION AND NUMBER (E-EUTAW FORMATION T-TUSCALOOSA FORMATION P-POTTSVILLE FORMATION)
Prepared by P.E. LaMoreaux & Associates Inc.
BROOKWOOD
TUSCALOOSA
DEMOPOLIS
UNDIFFERENTIATED
DOWNDIP BOUNDARY OF WATER CONTAINING LESS THAN 1,000 mg/l OF CHLORIDE IN THE PRINCIPAL AQUIFER OF THE EUTAW FORMATION

in the proposed system. It should be noted that over 80% of the surface area is suited for 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 areas. Figure 11-13 shows the extent of these formations in a portion of the United States.

XI.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. Well type is primarily a function of rock between surface and aquifer.[28]

Two broad categories of well types are generally recognized in the United States -- first, wells which penetrate relatively "soft" soil and rock 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 igenous 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 11-18 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

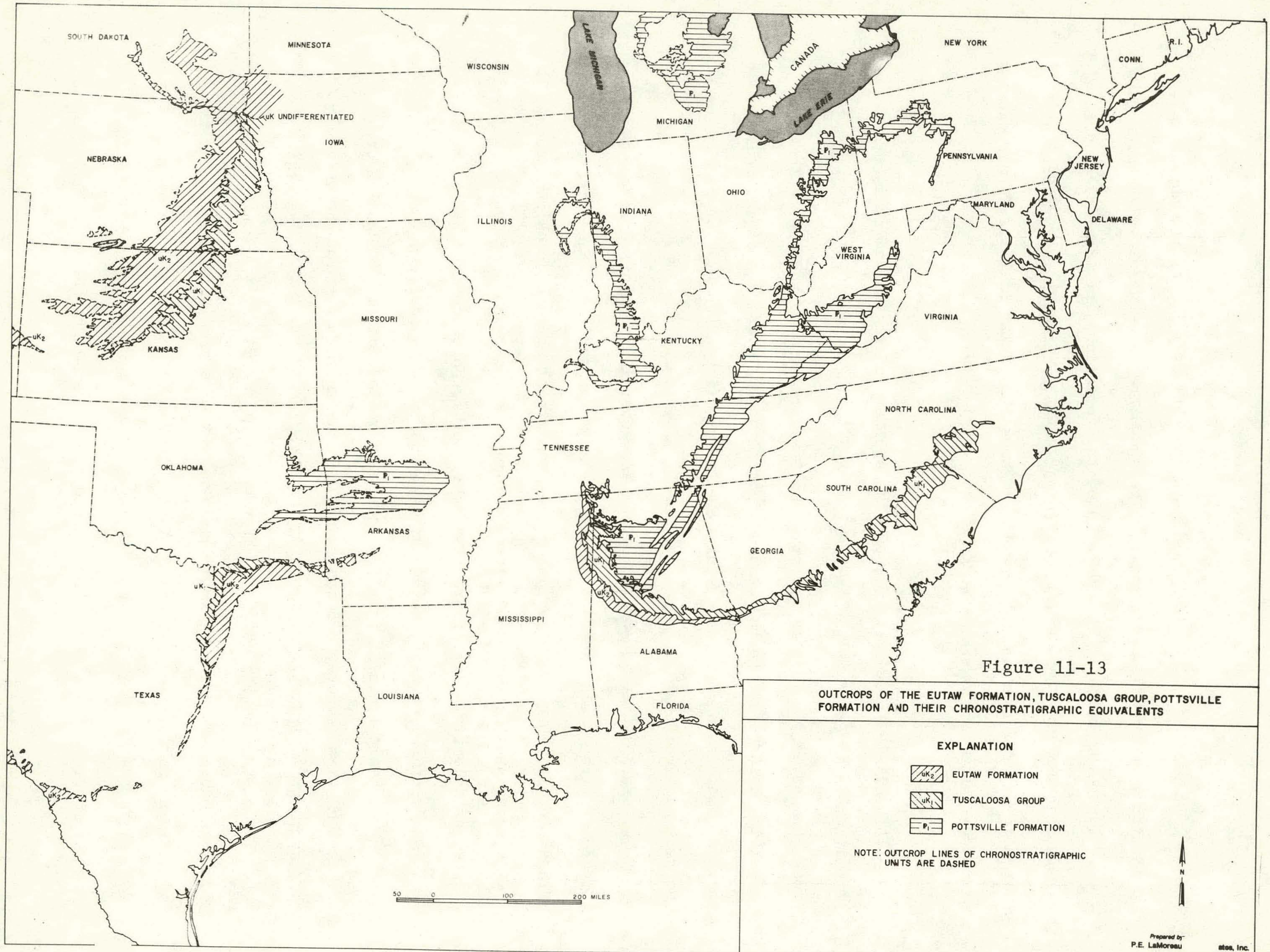


Figure 11-13

OUTCROPS OF THE EUTAW FORMATION, TUSCALOOSA GROUP, POTTSVILLE FORMATION AND THEIR CHRONOSTRATIGRAPHIC EQUIVALENTS

Table 11-18. Average Water-Well Drilling Costs in the United States, 1978[a]

Casing Diameter, Inches	Aquifer Type (b)	Drilling Method (c)	Depth, Feet	Cost Range Dollars (d)
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, Urbana, 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.

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 well construction 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 11-12. 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 the 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 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 1-8 and 1-9) and 30 psig pipeline pressure of 30 psi (60-ft. head) is:

Power = 5 gpm x Total Head = 5 gpm (8 lb./gal.) x (50+10+60) ft.

= 4,800 ft. lb./min. = 100 watts

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.

Performance and cost data are given in Table 11-19 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	1,007.00
Control System (pressure equated)	195.00
Total	$ 1,677.00

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.

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:

	Plain Tank	Diaphragm Tank
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.

Table 11-19. Performance Tables for Submersible Pumps[1]

Pump Cost	Stage No.	Pressure Psia	Depth to Water in Feet: 50	75	100	125	150	175	200
			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

XI.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 capatible 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.[40]

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 11-20.

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^{o}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^{o}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^{o}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.

Table 11-20. Normal Physical Properties of PVC Piping Materials[40]

Properties	Materials: Type I	Type II	ASTM test no.
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. F 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

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 11-21.

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

Table 11-21. Commercial Sizes (IPS) and Weights of Polyvinyl Chloride (PVC)[40] Pipe (Abstracted from ASTM Specification D1785-64T)

Nominal size, in.	Schedule	Wall thick-ness,**in.	OD, in.	ID, in.	Theoretical 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.

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, 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 11-14.

Type I is being considered for the piping system.

Cost analysis has been made for the various system alternatives shown in Section VII (Figures 7-1 to 7-4). The results are listed in Table 11-22. 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 1-8 and 1-9. 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

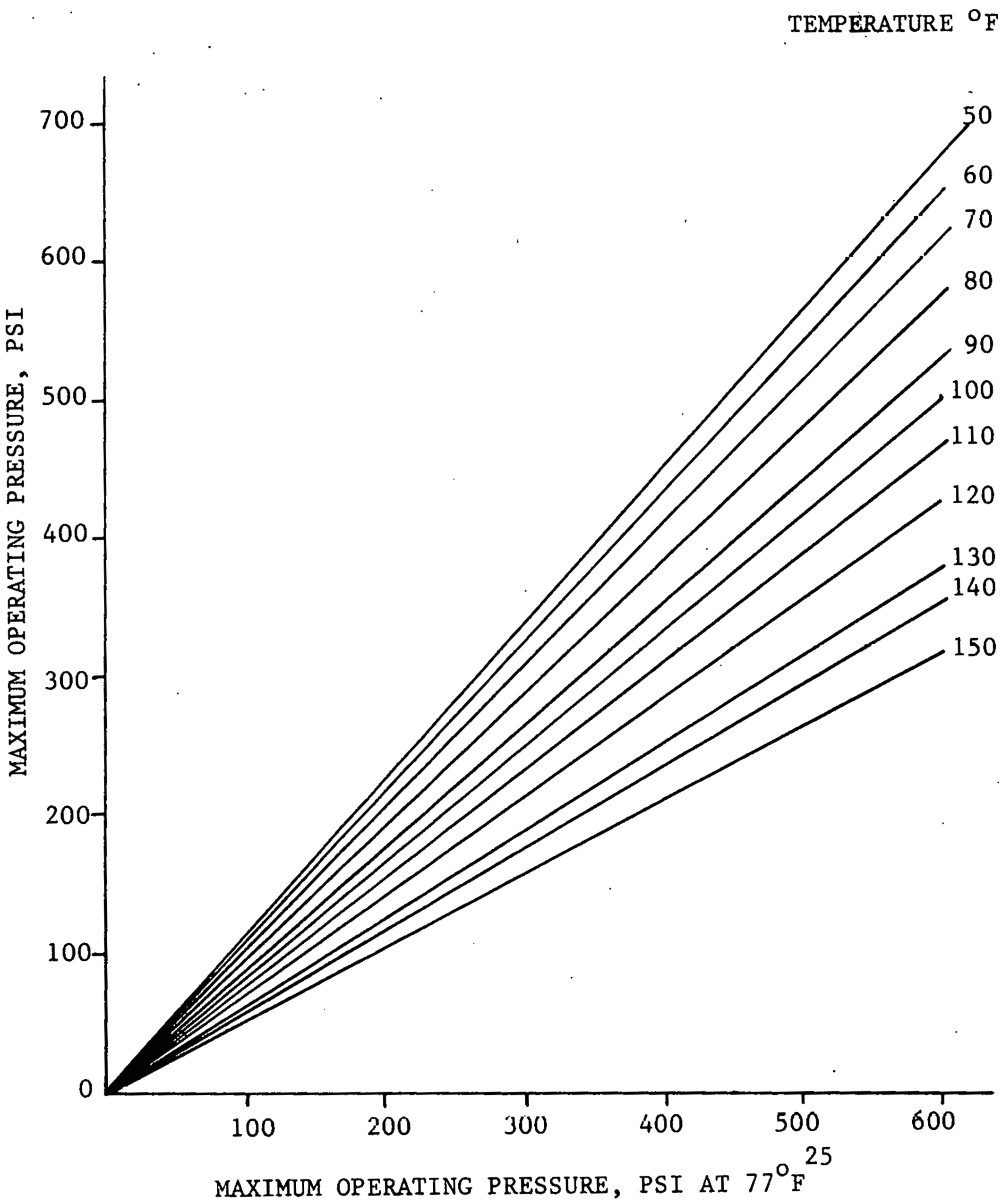


Figure 11-14. MAXIMUM OPERATING PRESSURE OF NORMAL-IMPACT (TYPE I) PVC AT ANY TEMPERATURE

20-11-78

Table 11-22.[6] COST OF DISTRIBUTION PIPE[1], INSTALLATION[2], AND TRENCHING[3]

DIAMETER (INCHES)	PIPE COST PER LINEAR FOOT: UNINSULATED PIPE 1	2	3	2 INSULATED; 1 UNINSULATED	4 INSULATED PIPES	INSULATION COST PER LINEAR FOOT (7)	TRENCHING COST PER LINEAR FOOT: 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 accommodate 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.[41]

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.

of over 4 inches keeps the pressure losses at negligible values and allows flow both ways between the pipes. Change in elevation will not affect 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 gpm below 15 ft. of water. A ½-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.

XI.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.[42,43,44,45] His basic premise is that mineral buildup 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 11-15.

Knudson's series of papers have shown the variations for a number of minerals. Table 11-23 shows a typical mineral content for a series of runs with various salts and changing pH. Tables 11-24 and 11-25 show typical mineral contents in ground water in certain formations in Alabama. Table 11-24 shows relatively low levels and Table 11-25 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.

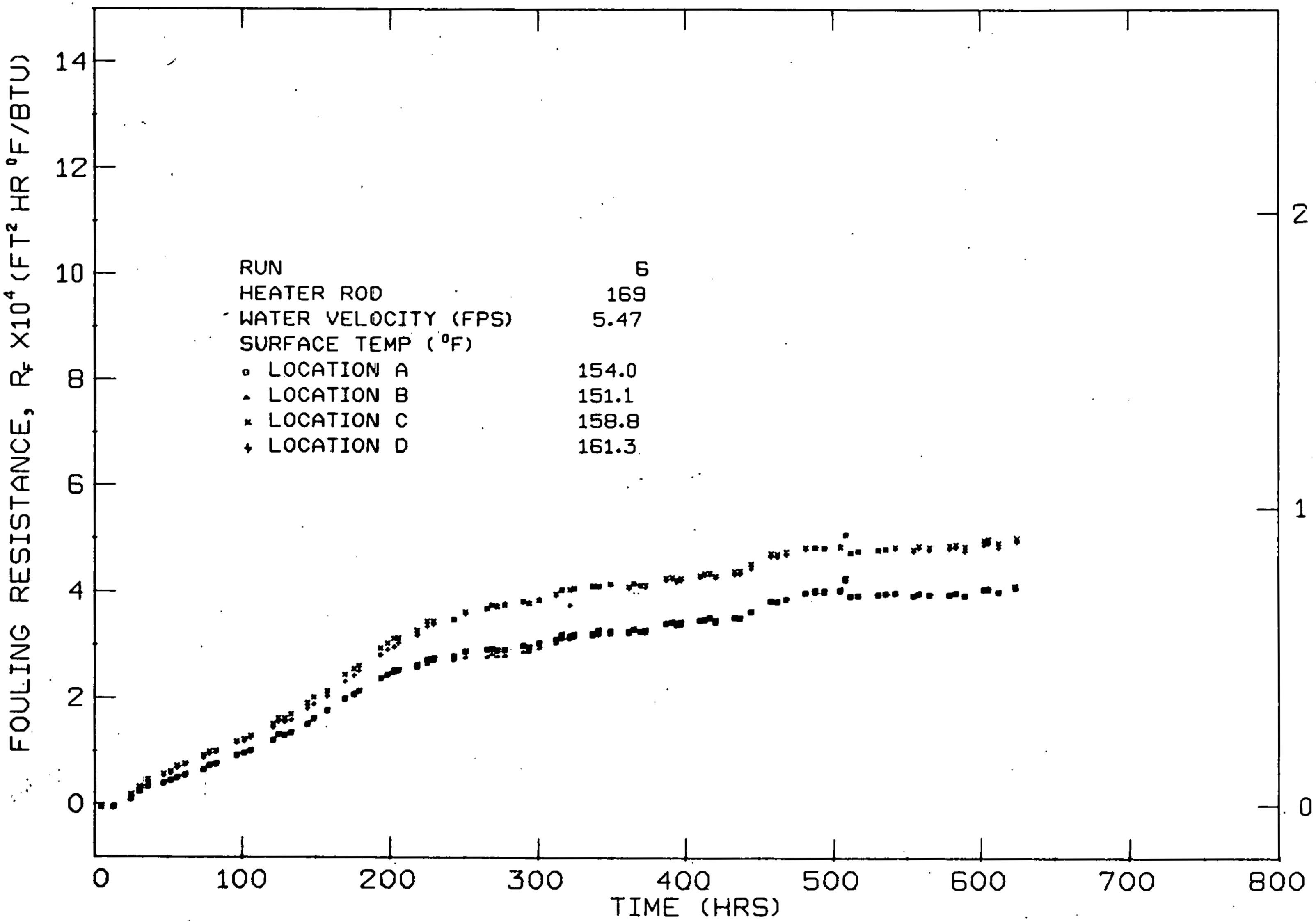


Figure 11-15. TIME EFFECT ON HEAT EXCHANGER FOULING BY LEE AND KNUDSON[45]

Table 11-23. Average Cooling Tower Water Quality, Runs 6-20[45]

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 11-24. Chemical Analyses of Water Samples From the Tuscaloosa Aquifers.[46]
(Kg-Gordo Formation, Kck-Coker Formation)

Map number	Well	Dates of Collection	Water-bearing Formation	Miligrams per liter: 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	Hardness as $CaCO_3$: Total	Noncarbonate	Specific conductance in Micromhos 25°C	pH
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. Woodard	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

Table 11-25. Representative Chemical Analyses of Water from Wells Tapping the Principal Aquifer of the Eutaw Formation in Alabama[46]

Well Number in this Report	County and Well Number in County Report	Date of Collection	Milligrams per liter														Dissolved Solids	Total	Noncarbonate	Specific conductance (micromhos at 25°C)	pH
			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)					
1E	Picken X-2	2-10-64	24	2.2		16	3.7	15	4.5	108	0	0	1.8	0.1	0.5	119	55	0	181	6.9
2E	Greene R-14	5-14-52	12	.12		5.2	1.4	214	4.5	232	0	1.4	206	1.2	.3	2/ 562	19	0	997	7.0
3E	Perry W-1	7-24-63	12			4.5	1.2	1/ 147		332	20	4.0	15	2.0	.4	369	16	0	616	8.4
4E	Autauga 0-2	12-1-59		3.4				2.3		34	0	7.2	2.0	.6	.1		27	0	79	6.7
5E	Lowndes D-6	12-12-55		.01						34	0	1.0	1330	0	.9		675	647	4,150	7.0
6E	Montgomery K-53	10-20-51	37	1.0	.08	28	2.0	9.0	1.7	106	0	6.2	3.0	.1	.2	141	78	0	183	6.9
7E	Macon V-9	3-31-58		.33		71	1.7	36		224	0	78	19	.1	0		184	0	536	7.6
8E	Russell L-6	5-10-62	24	1.1		40	10	6.4	8.6	197		5.6	1.1	.1	0	194	143	0	322	7.4

1/ Calculated Na + K reported as Na.

2/ Calculated

Table 11-26. Solar Energy Collections on 100-Square Foot Horizontal Collector*

Energy Collected - 10^3 Btu

CITIES	APRIL	MAY	JUNE	JULY	AUGUST	SEPTEMBER	JUNE-AUGUST OVERALL
Birmingham, AL	3301	3708	3803	3782	3337	2870	10922
Tucson, 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 manufactors test data
Average efficiency 0.65
No glazing

In his papers, Knudson's maximum value of resistance fouling is of the order of magnitude of 0.0004 ft.^{2} hr.$^\circ$F/Btu. The value for sea water is given as 0.0005 ft.^{2} hr $^\circ$F/Btu. The effect of this level is shown to be minor.[44,45]

The general range of heat transfer coefficients for a Freon 12 condensor is 80 to 150 Btu/ft.^{2} hr. $^\circ$F. The reciprocal or resistance therefore varies from 0.012 to 0.00667 ft.^{2} hr. $^\circ$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 form biological growth must be evaluated in the laboratory or in demonstration projects. These variables can be controlled to an extent, if necessary, by the addition of chemical inhibitors. Another possibility is to eliminate all air-water interfaces in the system.

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.

XI.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 practical, 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 11-26 gives the horizontal solar energy collection quantities for the summer months per 100 ft.^{2} of collector area.[47,48] Collection efficiencies are based on Figure 11-16 where the June through August solar energy

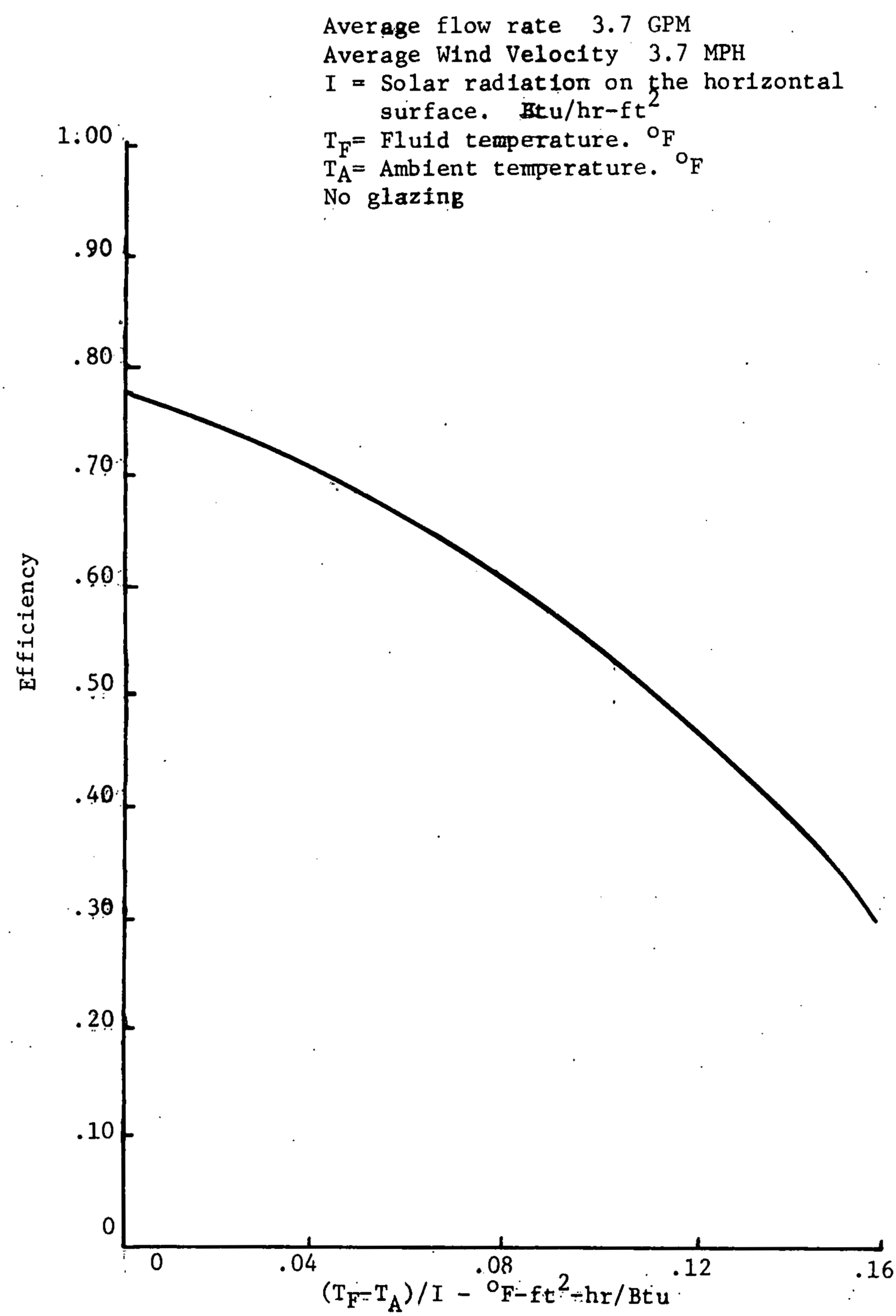


Figure 11-16. MEASURED EFFICIENCY OF SOLAR COLLECTOR

total is also shown.[49] An increase in performance will be made by inclining the collectors. Optimum angle of inclination is shown in Figure 11-17 for various time periods.[50]

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 11-18 shows a typical installation.

Figure 11-19 shows schematically a hard rubber solar collector without glazing, and Figure 11-20 shows schematically a black bag solar collector.[50,51] Both collectors retail between \$2.50 and \$3.50 per ft.^{2} and are designed for roof collection. Figure 11-21 shows a proposed solar pond collector and an estimation of installed costs with required pump and controls.[52,53,54,55]

For the economic analysis, an installed 100 ft.^{2} panel is included for energy addition at a cost of \$3.00/ft.^{2} or \$3,000 total. It must be emphasized that this would be used only if waste heat is not available at a competitive cost.

XI.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 normally 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.

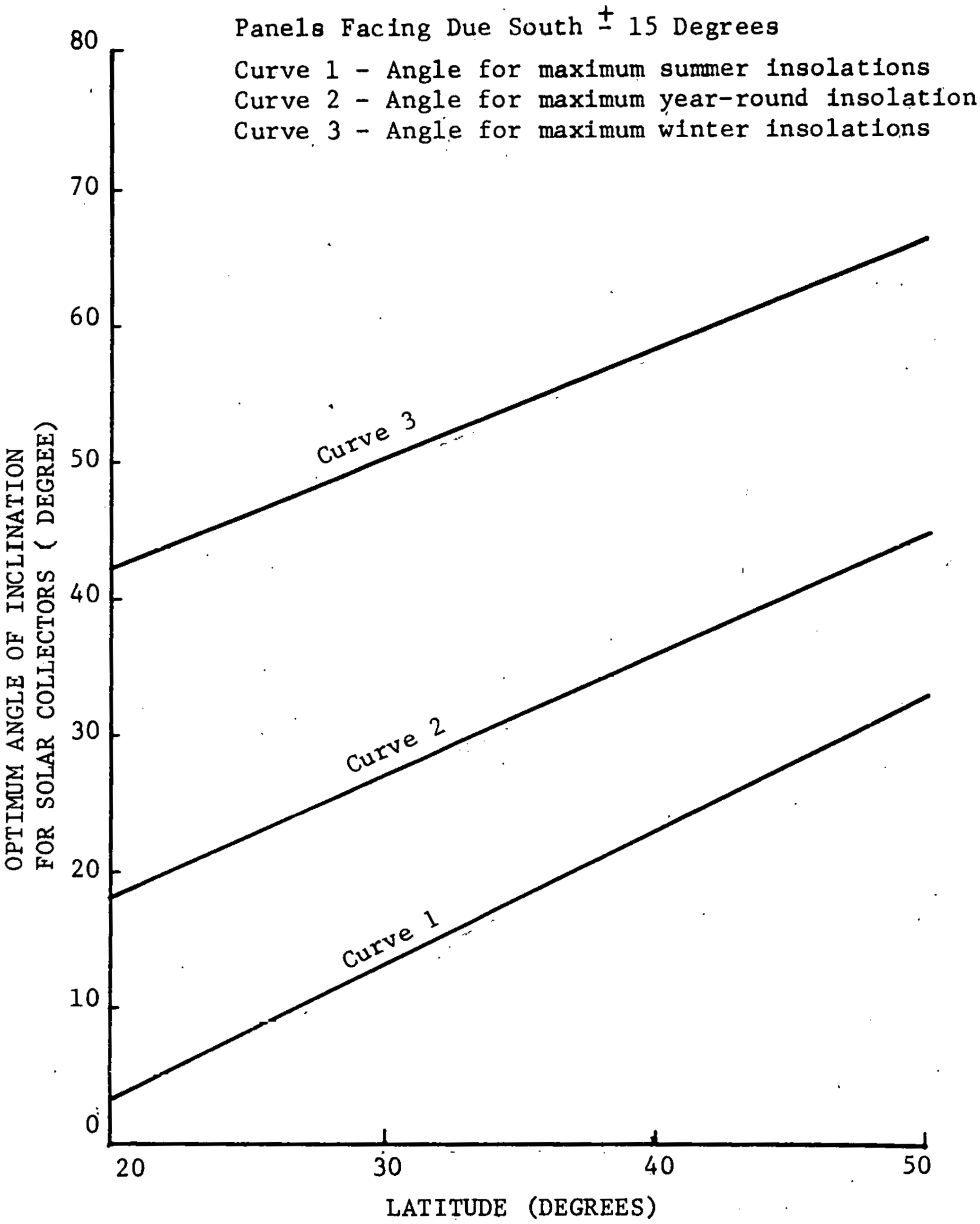


Figure 11-17. OPTIMUM COLLECTOR INCLINATION vs. LATITUDES[39]

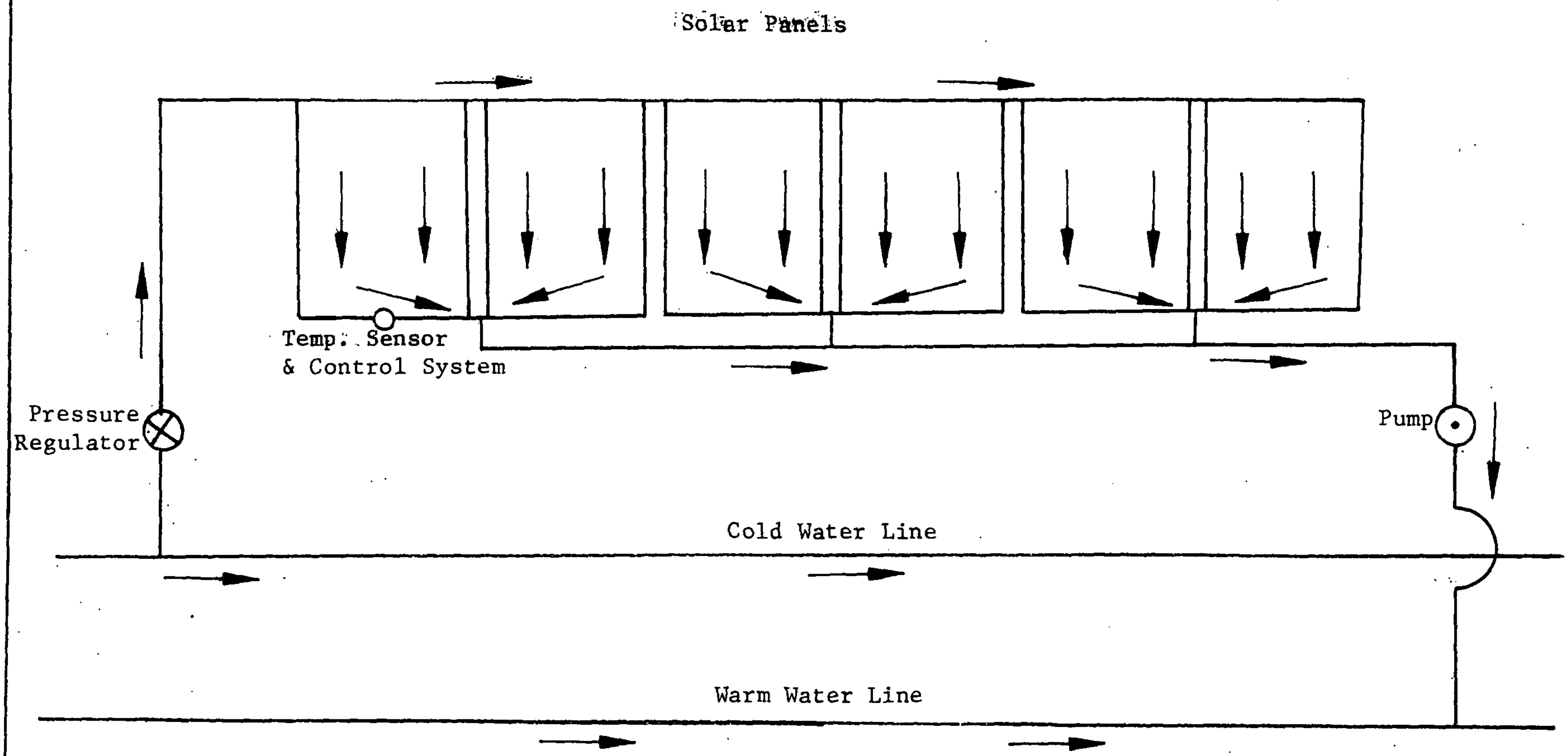


Figure 11-18. STANDARD SOLAR COLLECTOR SYSTEM WITH UNGLAZED PANELS

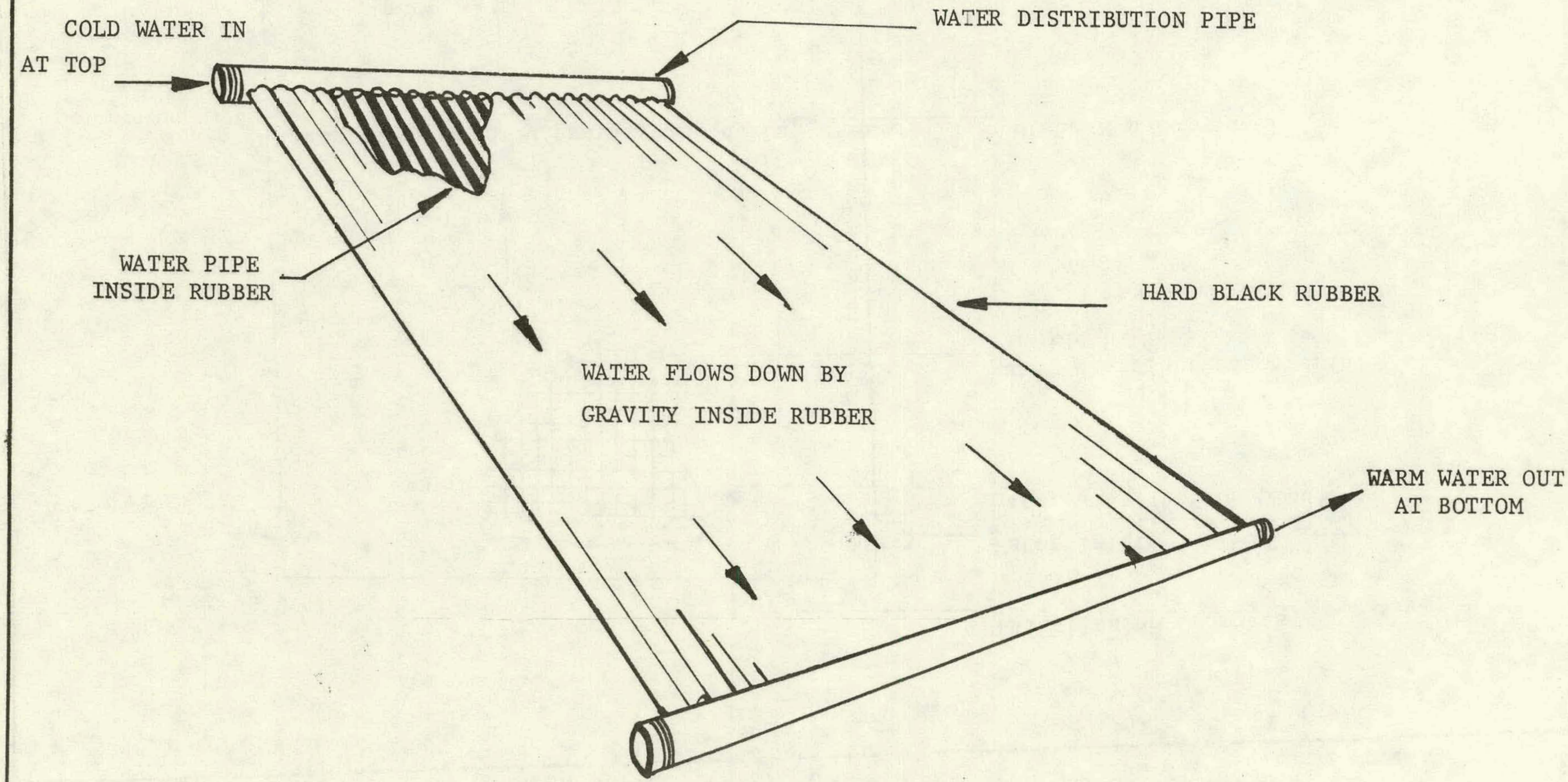


Figure 11-19. SKETCH OF HARD RUBBER MOLDED SOLAR PANEL WITHOUT GLAZING

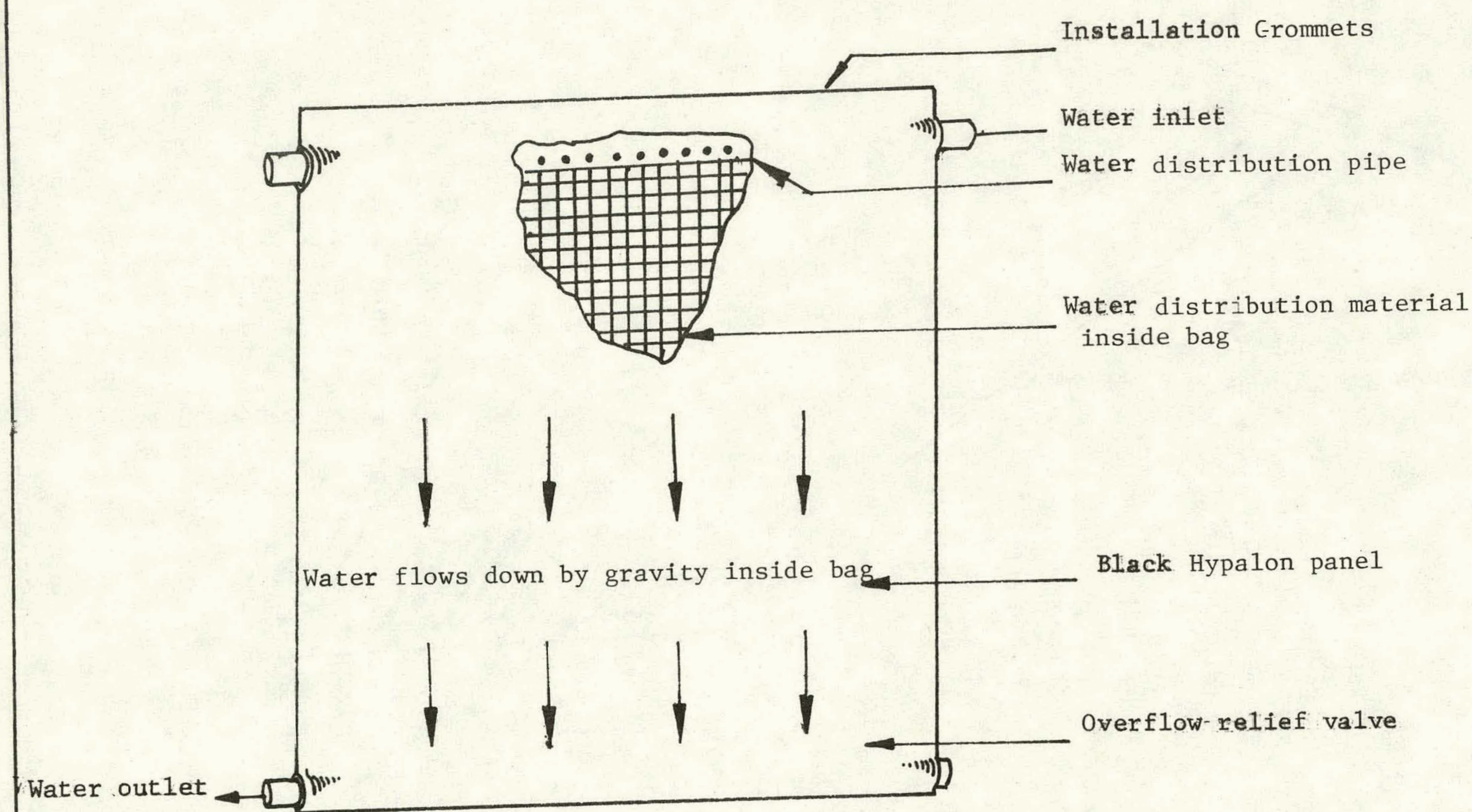
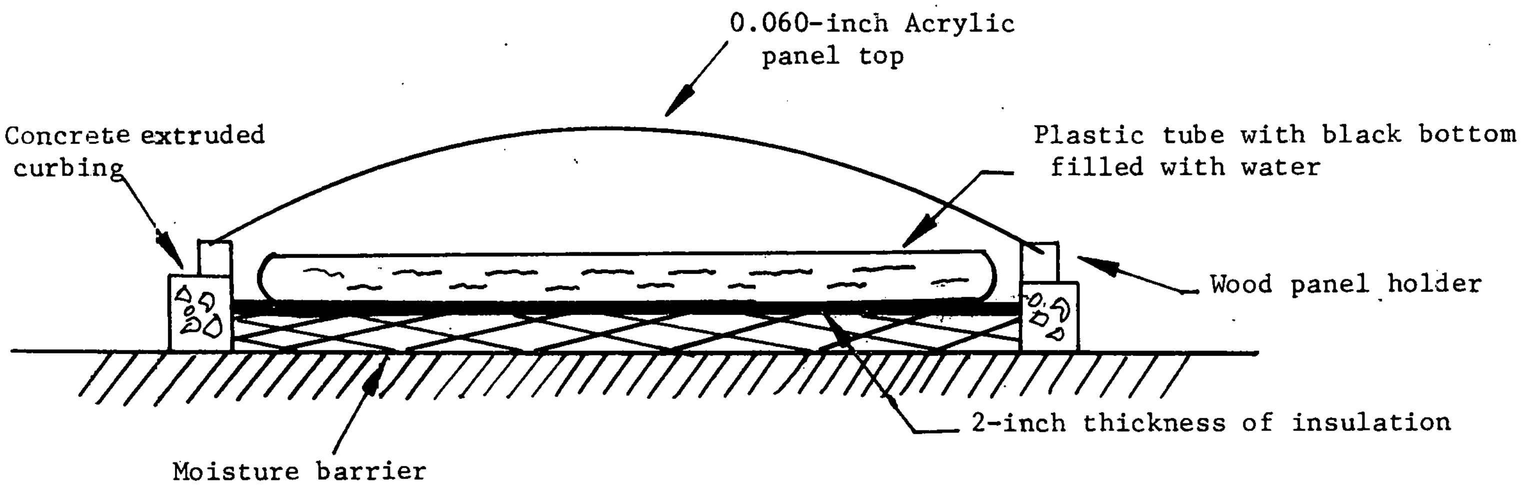


Figure 11-20. PANEL CONFIGURATION FOR SOLAR BAG COLLECTOR

FIGURE 11-21. 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	\$2.87 per ft.^{2}	2.24 per ft.^{2}

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.

XII. 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 2 data cooling and heating 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:

Total Heating Load = (Heat Load Due to Heat Transfer) - Internal Load

Total Cooling Load = (Cooling Load Due to Heat Transfer) + 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 calculation, 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 $5^{o}F$ 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,

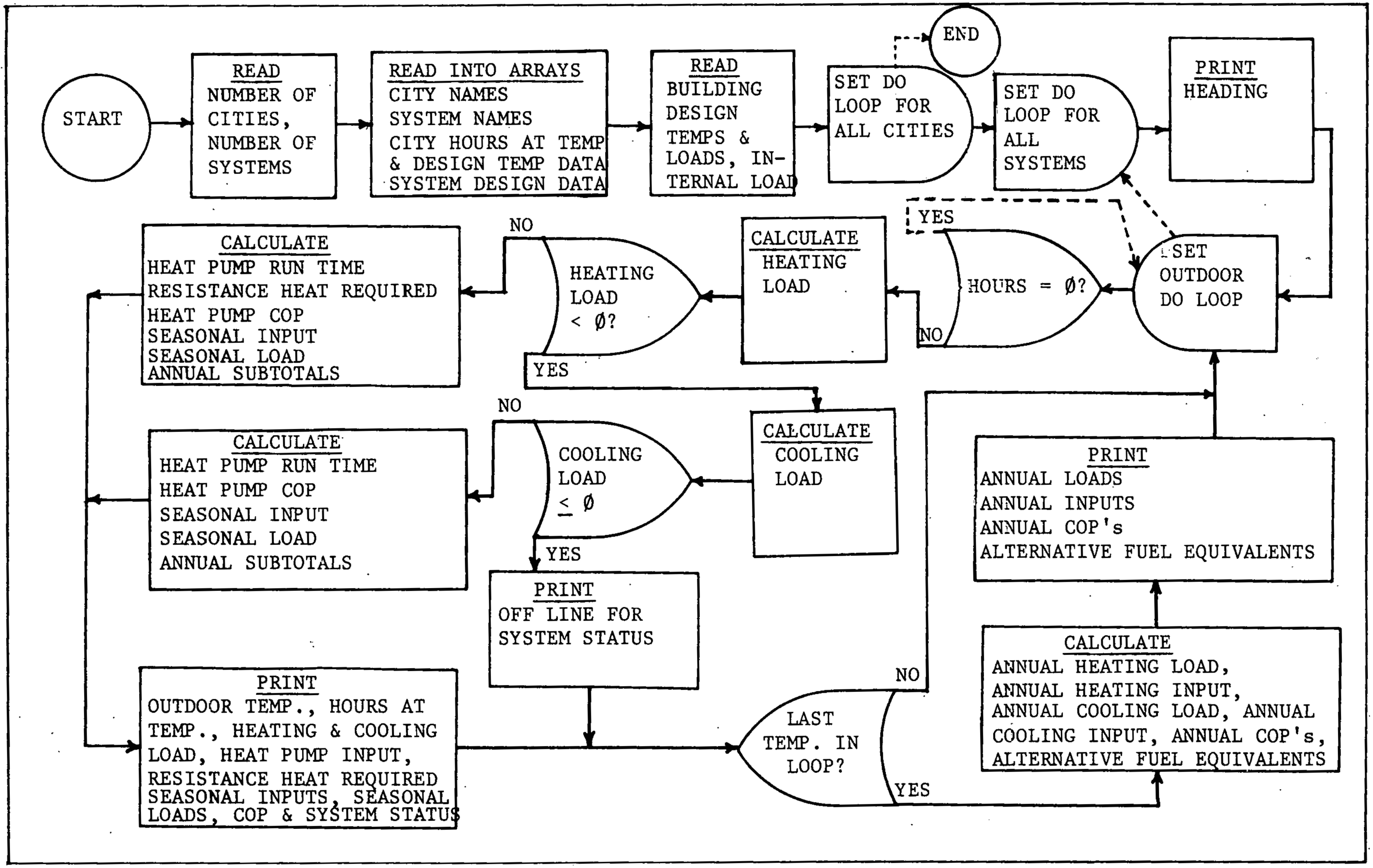
$$\text{COP} = \frac{\text{Total Load}}{\text{Total Power Input}}$$

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

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

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.



Flow Chart for Heating and Cooling Performance Computer Programs

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, ^{o}F
CIDT -	Cooling indoor design temperature, ^{o}F
LOAD -	Heating design load, BTUH
CLOAD -	Cooling design load, BTUH
ILOAD -	Internal load, BTUH
TT -	Thermostat temperature in building, ^{o}F
ODT -	Heating outdoor design temperature, ^{o}F
CODT -	Cooling outdoor design temperature, ^{o}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^{o}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, ^{o}F
B -	System heating/cooling capacity, MBTUH, at outdoor temperature A
C -	Heat pump compressor run time in 5^{o} increment, percent
D -	Input for heat pump at temperature A, KW
E -	Seasonal heating hours in $5^{o}F$ increment about A ($A \pm 2.5^{o}F$)
F -	Seasonal heat pump input in 5^{o} increment about A
G -	Resistance heat required in $5^{o}F$ increment about A
H -	Seasonal resistance heat needed in $5^{o}F$ increment about A
QT -	Seasonal heating/cooling load in $5^{o}F$ increment about A

```
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 ()
      CODT=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) CODT,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',
```

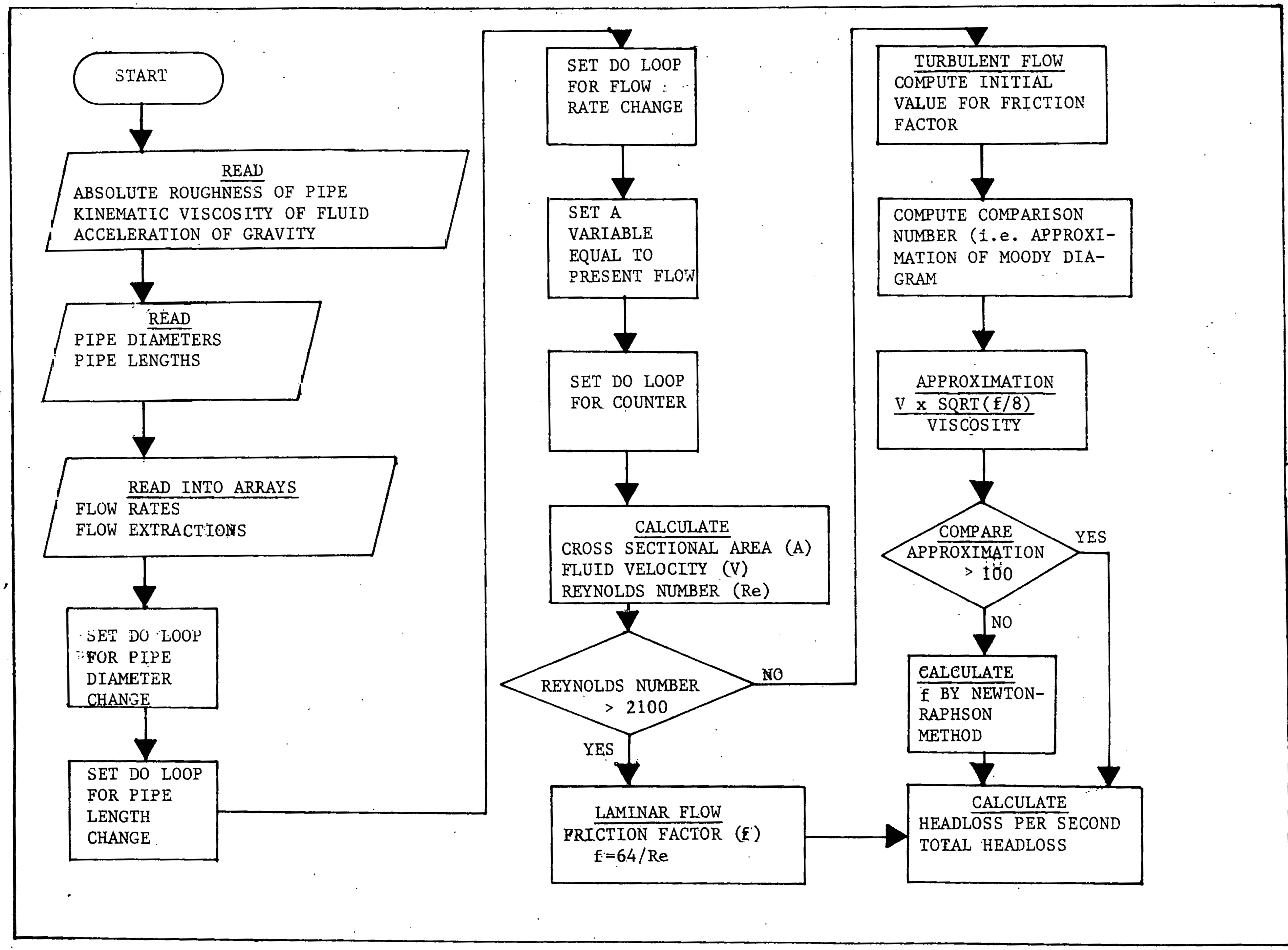
```
&                                          /,' ','D= HEAT PUMP CAPACITY
&AT OUTDOOR TEMPERATURE A, MBTUH',
&                                 /,' ','E= HEAT PUMP SYSTEM INPUT, KW, IN
&CLUDING AIR HANDLER REQUIREMENT, KW',
&                                /,' ','F= HEAT PUMP COMPRESSOR RUN TIME I
&NPERCENT 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= COEFICIEN
&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/(CODT-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
```

XIII. 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.[56] 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.


START
READ
ABSOLUTE ROUGHNESS OF PIPE
KINEMATIC VISCOSITY OF FLUID
ACCELERATION OF GRAVITY
READ
PIPE DIAMETERS
PIPE LENGTHS
READ INTO ARRAYS
FLOW RATES
FLOW EXTRACTIONS
SET DO LOOP FOR PIPE DIAMETER CHANGE
SET DO LOOP FOR PIPE LENGTH CHANGE
SET DO LOOP FOR FLOW RATE CHANGE
SET A VARIABLE EQUAL TO PRESENT FLOW
SET DO LOOP FOR COUNTER
CALCULATE
CROSS SECTIONAL AREA (A)
FLUID VELOCITY (V)
REYNOLDS NUMBER (Re)
REYNOLDS NUMBER > 2100
NO
YES
LAMINAR FLOW
FRICTION FACTOR (f)
f=64/Re
TURBULENT FLOW
COMPUTE INITIAL VALUE FOR FRICTION FACTOR
COMPUTE COMPARISON NUMBER (i.e. APPROXIMATION OF MOODY DIAGRAM
APPROXIMATION
V x SQRT(f/8)
VISCOSITY
COMPARE APPROXIMATION > 100
YES
NO
CALCULATE
f BY NEWTON-RAPHSON METHOD
CALCULATE
HEADLOSS PER SECOND
TOTAL HEADLOSS

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 ($^e/D$) 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)*D(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*SQRT(F/8.)*EVIS
      IF(PAR.GT.100.)GO TO 1
      NCT=0
2     FS=SQRT(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
```

XIV. APPENDIX III. MULTIWELL PROGRAM DESCRIPTION

This program iteratively defines streamlines for flow from a source well in arbitrary system of source and sink wells. This is accomplished by providing an initial set of starting points at a radius about the well under study which is analagous to the radius of the well in the physical system. Velocities impressed on a particle at that point by all of the wells in the system plus any natural drift velocities are then superimposed to give a resultant velocity. The resultant velocity taken over a finite time increment allows calculation of the next point along the streamline. This process is repeated until the particle enters the radius of a sink well or the input limit time is reached.

Provision is made for the user to input aquifer data of porosity and applied drift velocities, and well data to describe the position and flow rate of each well in the system. In addition the user inputs a limit time at which final positions of streamline particles will be printed, and a print increment time for output of intermediate positions of particles along their streamlines. The program contains a loop to compute streamlines beginning in twenty degrees increments about the source well, thus describing the entire system of streamlines generated by the well.

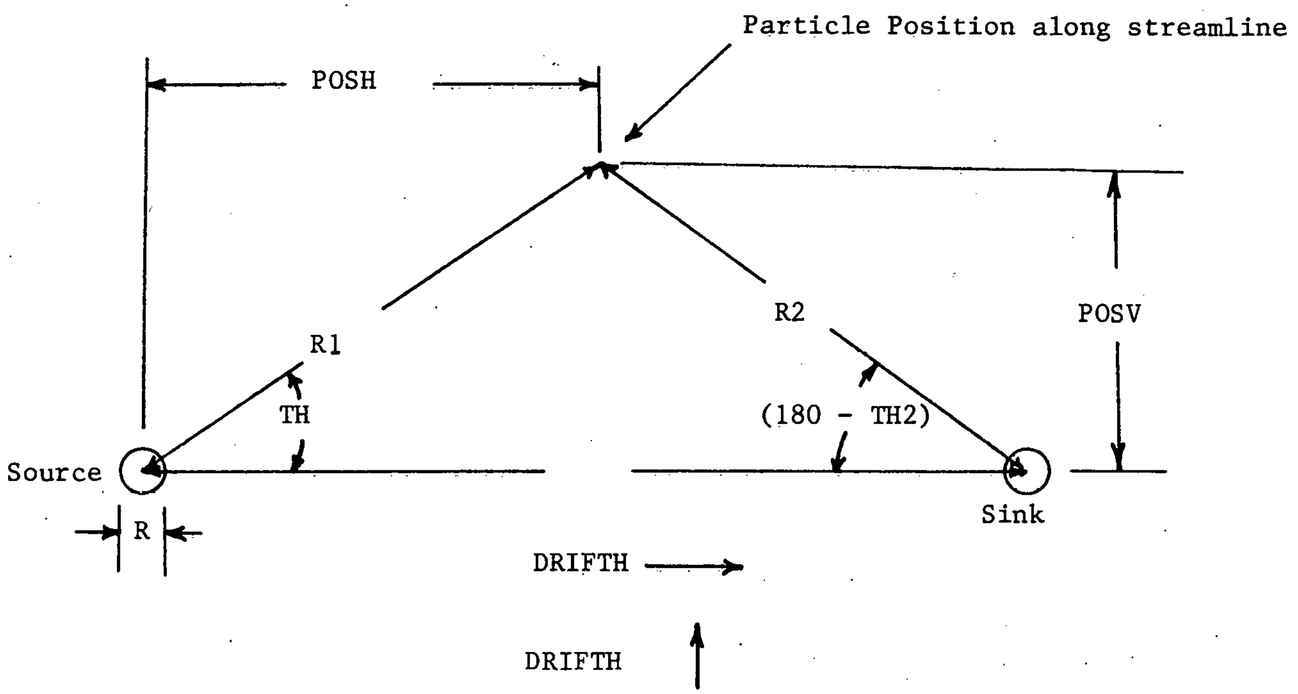
Equations

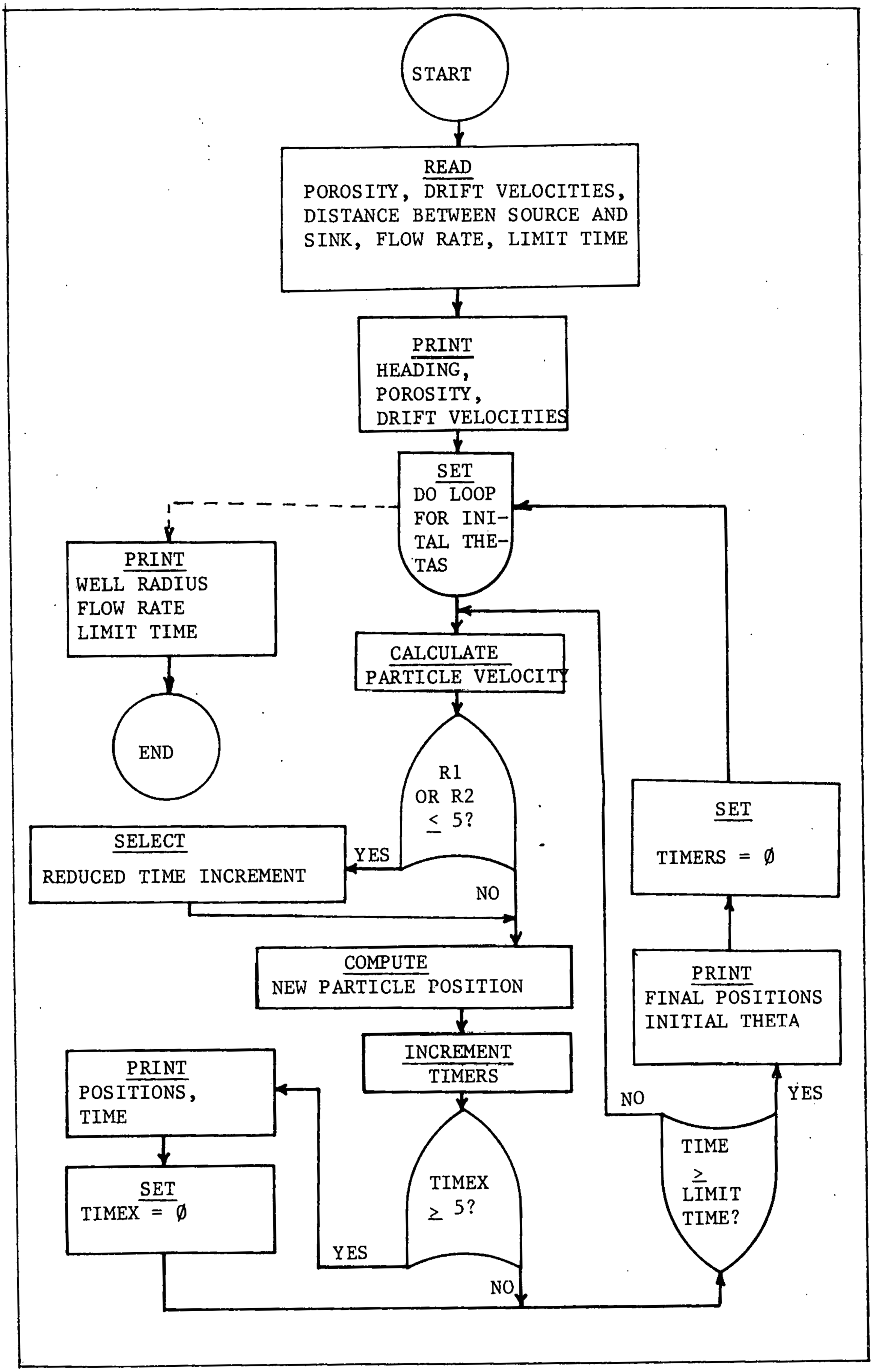
$$\text{Velocity due to well} = \frac{\text{flow rate (area/time)}}{\text{(radius from well)} \times \text{(porosity)}}$$

$$\text{Particle velocity} = \text{Velocities due to wells} + \text{drift velocities}$$

Note: All the above velocities are vector quantities. Positions and velocities are described in a standard X-Y coordinate field.

Horizontal and vertical drift velocities are input as data and assumed constant over field.





START
READ
POROSITY, DRIFT VELOCITIES,
DISTANCE BETWEEN SOURCE AND
SINK, FLOW RATE, LIMIT TIME
PRINT
HEADING,
POROSITY,
DRIFT VELOCITIES
SET
DO LOOP
FOR INI-
TAL THE-
TAS
PRINT
WELL RADIUS
FLOW RATE
LIMIT TIME
END
CALCULATE
PARTICLE VELOCITY
R1
OR R2
≤ 5?
YES
NO
SELECT
REDUCED TIME INCREMENT
SET
TIMERS = Ø
COMPUTE
NEW PARTICLE POSITION
PRINT
FINAL POSITIONS
INITIAL THETA
INCREMENT
TIMERS
PRINT
POSITIONS,
TIME
TIMEX
≥ 5?
YES
NO
NO
YES
TIME
≥
LIMIT
TIME?
SET
TIMEX = Ø

MULTIWELL PROGRAM VARIABLES

DELT	Time increment
DRIFTX	Drift velocity along X axis
DRIFTY	Drift velocity along Y axis
DX	X displacement of particle during DELT
DY	Y displacement of particle during DELT
I,J,K	Integer counters
N	Number of wells in system
N1	Number of source well for streamlines desribed
PHI	Starting angle from well, in degrees
POROS	Porosity of aquifer
POSX	X position of particle
POSY	Y position of particle
PRINT	Print increment time for intermediate particle positions
R	Well radius
RK	Distance from particle to well
RX	X distance from particle to well
RY	Y distance from particle to well
STARTX	X position of source well
STARTY	Y position of source well
T	Elapsed time for streamline
THETA	angle from well to particle
THETIN	streamline starting angle from well, radians
TIME	Final time for streamline description
TIMEX	Elapsed time since last intermediate print
VK	Magnitude of velocity due to well

VTX	Sum of superimposed X velocities
VTY	Sum of superimposed Y velocities
VXK	X component of velocity due to well
VTK	Y component of velocity due to well
WELL (a,b)	Matrix variable for well data

a = well number in system
b- 1 X position of well
2 Y position of well
3 flow rate at well, (volume/aquifer thickness x time)

```
      DIMENSION WELL(10,3)
      WRITE(6,800)
  800 FORMAT(' INPUT NUMBER OF WELLS, STARTING WELL NUMBER,',/,
     &' POROSITY, X DRIFT, Y DRIFT, LIMIT TIME, & PRINT TIME.')
      READ(5,100) N,N1,POROS,DRIFTX,DRIFTY,TIME,PRINT
      DO 10 I=1,N
      WRITE(6,900) I
  900 FORMAT(' INPUT X POSITION, Y POSITION, & FLOW RATE FOR WELL',I4)
      READ(5,100) (WELL(I,J),J=1,3)
   10 CONTINUE
  100 FORMAT ()
      STARTX=WELL(N1,1)
      STARTY=WELL(N1,2)
      R=1.
      WRITE(6,400)N1,N,POROS,DRIFTX,DRIFTY,TIME,PRINT
  400 FORMAT('1',10X,'*** WELL SYSTEM SIMULATION ***',
     &/,'0',20X,'BY J. F. LECROY',/,'0',
     &'THIS PROGRAM OUTPUT DESCRIBES A SYSTEM OF STREAMLINES FROM WELL'
     &,I3,' IN A SYSTEM OF ',I3,' WELLS',/,' ',
     &'POROSITY= ',F5.4,'  DRIFTX= ',F5.3,' DRIFTY= ',F5.3,/,' ',
     &'FINAL TIME= ',F5.1,' PRINT INCREMENT TIME= ',F5.1,/,'0')
      WRITE(6,700)
  700 FORMAT(' ',3X,'WELL NUMBER',5X,'X POSITION',10X,'Y POSITION',
     &10X,'FLOW RATE')
      DO 70 M=1,N
      WRITE(6,600) M,(WELL(M,K),K=1,3)
  600 FORMAT(' ',6X,I3,15X,F5.0,15X,F5.0,15X,F5.0)
   70 CONTINUE
      DO 20 I=1,341,20
      PHI=FLOAT(I)-1.
      THETIN=PHI/57.29578
      POSX=STARTX+R*COS(THETIN)
      POSY=STARTY+R*SIN(THETIN)
      TIMEX=0.
      T=0.
   50 DO 30 K=1,N
      IF (K.EQ.1) VTX=0.
      IF (K.EQ.1) VTY=0.
      IF (K.EQ.1) DELT=.1
      RX=POSX-WELL(K,1)
      RY=POSY-WELL(K,2)
      RK=SQRT((RX*RX)+(RY*RY))
      IF((K.NE.N1).AND.(RK.LE.R)) GO TO 99
      VK=WELL(K,3)/(6.2831853*RK)
      THETA=ATAN2(RY,RX)
      VXK=VK*COS(THETA)
      VYK=VK*SIN(THETA)
      VTX=VTX+VXK
      VTY=VTY+VYK
      IF(RK.LE.5.) DELT=.01
   30 CONTINUE
      DX=DELT*((VTX/POROS)+DRIFTX)
      DY=DELT*((VTY/POROS)+DRIFTY)
      POSX=POSX+DX
      POSY=POSY+DY
      IF (T.GE.TIME) GO TO 1000
```

```
      IF(TIMEX.GE.PRINT)GO TO 2000
   60 T=T+DELT
      TIMEX=TIMEX+DELT
      GO TO 50
 1000 WRITE(6,200) PHI,N1,POSX,POSY
  200 FORMAT('0','FINAL POSITION FOR PARTICLE STARTING AT ',F4.0,
     &'DEGREES FROM WELL ',I3,/,' ',10X,'X POSITION = ',F5.1,5X,
     &' Y POSITION = ',F5.1,/,'0',/,'0')
      GO TO 20
 2000 WRITE(6,300) PHI,N1,POSX,POSY,T
  300 FORMAT(' ','INTERMEDIATE POSITION FOR PARTICLE STARTING AT ',F4.0,
     &'FROM WELL ',I3,' POSX= ',F5.1,' POSY= ',F5.1,' AT',F3.0,' DAYS')
      TIMEX=DELT
      GO TO 60
   99 WRITE(6,500) PHI,K,T
  500 FORMAT(' ','STREAMLINE BEGINNING AT ',F5.1,'DEGREES ENDS AT WELL',
     &I3,' AT TIME= ',F5.1,/,'0',/,'0')
 20   CONTINUE
      STOP
      END
```

XV. REFERENCES

1. Natural Atlas of the United States of America, U.S. Department of Interior, Geological Survey (1970).

2. Statistical Abstract of the United States, 1977, Bureau of Census, U.S. Department of Commerce (1977).

3. Residential and Commercial Energy Use Patterns, 1970-1990, FEA, Project Independence, Vol. 1, (1974).

4. Van Wylen, G.J. and Sonntag, R.E., Fundamentals of Classical Thermodynamics, John Wiley & Sons, Inc., New York (1973).

5. Cropsey, M.G., "Analysis of Soil as Heat Source for Heat Pump System," Transactions of the ASAE, pp. 846-848, (1966) ASAE.

6. Groethe, S.P., Sutton, G.E. and Liffler, W.A., "Earth as a Heat Source and Sink for Heat Pumps," Engineering Progress at the University of FLorida, Vol. IV, No. 6, (June, 1950) Florida Engineering and Industrial Experiment Station.

7. Krumme, V.W., "Luft-Wasser-Warmepumpe in Einfamilienhaus," Electrowarme International 32 (January, 1974) A1.

8. Michels, V.F., "Probleme einer Luft-Wasser-Warmepumpe bei tiefen Aussentemperaturen," Bull. SEV/VSE 68 (April, 1977) 8,16.

9. Compressor Performance curves and tables by Tecumseh, transfered through American Air Filter, (November, 1978).

10. WESCORP Solargy Systems Catalog, including Typical Specification Sheets, WESCORP Inc. 1979-019 and WESCORP Inc. 1978-098. Includes numerous design specifications and configuration, from WESCORP Inc., 15 Stevens Street, Andover, Ma., 01810.

11. ASHRAE Standard 90-75, Section 12, (1977).

12. Christian, J.E., Unitary Water-to-Air Heat Pumps, Argonne National Laboratory Report ANL/CES/TE/77-10, Argonne, Illinois (July, 1977).

13. Christian, J.E., Unitary Water-to-Air Heat Pumps, Argonne National Laboratory Report ANL/CES/TE/77-9, Argonne, Illinois (October, 1977).

14. Bezdek, R.H., Hirshberg, A.S., and Babcock, W.H., "Economic Feasibility of Solar Water and Space Heating," Science, pp. 1214-1220, Vol. 203, 23 (March, 1979).

15. Croke, K., Baum, J., and Rosenberg, R., Municipal Financing of Integrated Community Energy Systems, ANL/ICES-TM-3, Argonne National Laboratory, Argonne, Illinois (1977).

16. Huff, F.A., and Changnon, S.A., "Precipitation Modification by Major Urban Areas," Bulletin American Meterological Society, Vol., 54, No. 12, (December, 1973).

17. Changnon, S.A., "Urban-Industrial Effects on Clouds and Preciptation," Proceedings on Inadvertant Weather Modification, Utah State University, (1973).

18. Krauskopf, K.B., Introduction to Geochemistry, McGraw-Hill Book Company, (1967).

19. Grubbs, D.M., Haynes, C.D., Hughes, T.H., and Stow, S.H., "Compatibility of Subsurface Reservoirs with Injected Liquid Wastes," NRC Report 721, The University of Alabama (June, 1972).

20. Rising Ground-Water Level in Downtown Louisville, Kentucky, 1972-1977, United States Geological Survey (September, 1977).

21. Susemichel, A.H., Louisville: Groundwater for Energy, By Necessity - Not Choice, Hughes & Susemichel, Inc., Louisville, Kentucky (January 31, 1979).

22. Fry, W.E., Louisville Central Area Ground Water Utilization for Energy Efficient Heating and Air Conditioning of Downtown Louisville Buildings, Present and Future, American Air Filter Co., Inc. (June 20, 1978).

23. Bell, E.A., Summary of Hydrologic Conditions of the Louisville Area Kentucky, United States Government Printing Office, Washington (1966).

24. Price, Jr., W.E., Hydrolic Investigations, Atlas HA - 130, United States Geological Survey (1964).

25. Price, Jr., W.E., Hydrolic Investigations, Atlas HA - 111, United States Geological Survey (1964).

26. Barksdale, H.C. and Moore, J.D., et al, unpublished, Water Content and Potential Yield of Significant Aquifers, Geological Survey of Alabama (1976).

27. Geological Survey of Alabama, 1949, Special Report 20, Water Resources and Hydrology of Southeastern Alabama, (1949).

28. Campbell, M.D. and Lehr, J.H., Water Well Technology, McGraw-Hill, (1977).

29. Basewide Energy Systems Plan, 80 Percent Energy Use Surveys, Vol. 1 Narrative, Fort Rucker, Alabama, Black and Veatch, Consulting Engineers, (November, 1978), Contract DACA01-77-C-0094.

30. ASHRAE 1977 Handbook of Fundamentals, American Society of Heating Refrigeration, and Air Conditioning Engineers, New York (1977).

31. Skortzke, B.G.A., Vapat, W.A., Power Station Engineering and Economy, McGraw-Hill, (1960).

32. Todd, D.K., Groundwater Hydrology, John Wiley & Sons, Inc., New York (1963).

33. Craft, B.C. and Hawkins, M.F., Applied Petroleum Reservoir Engineering, Prentice - Hall, Inc., Englewood Cliffs, New Jersey (1959).

34. Gannon, R., "Ground-water Heat Pumps: Home Heating and Cooling From Your Own Well," Popular Science, pp. 78-82 (February, 1978).

35. Henry, H.R., McDonald, J.R., and Alverson, R.M., Aquifer Performance Tests Under Two-Phase Flow Conditions, BER Report 131-118, The University of Alabama (March, 1971).

36. Cormary, Y., et. al., "Heat Storage in a Phreatic Aquifer: Campuget Experiment," Electhricite'de France (19).

37. Moly, F.J., Warman, J.C., and Thomas, E.J., Aquifer Storage of Heated Water: Experimental Study, Auburn University (1978).

38. Hunsaker, J.C. and Rightmire, B.G., Engineering Applications of Fluid Mechanics, McGraw-Hill, New York (1947).

39. Rauscher, M., Introduction to Aeronautical Dynamics, John Wiley & Sons, Inc., New York (1953).

40. Crocker, S., Piping Handbook, McGraw-Hill, New York (1945).

41. Means, R.S., Building Construction Cost Data, Robert Snow Means Co., Inc. Duxbury, Massachusetts (1978).

42. Knudson, J.G., Personal Communication, Oregon State University, (June, 1978).

43. Morse, R.W., and Knudson, J.G., "Effect of Alkalinity on the Scaling of Simulated Cooling Tower Water," The Canadian Journal of Chemical Engineering, Vol. 55 (June, 1977).

44. Knudson, J.G., and Story, M., "The Effect of Heat Transfer Surface Temperature on the Scaling Behavior of Simulated Cooling Tower Water," AICHE Symposium Series, No. 174, Vol. 74 (1977).

45. Lee, S.H. and Knudson, J.G., Scaling Characteristics of Cooling Tower Water, Department of Chemical Engineering Oregon State, (August, 1978).

46. Lamoreaux, P.E., Personal Communication, Data compiled from Alabama State Geological Survey Data, (October, 1978).

47. Bennett, I., "Monthly Maps of Mean Daily Insolation for the United States," Solar Energy, Vol. 9 (1965).

48. Liu, B.Y. and Jordan, R.C., "A Rational Procedure for Predicting The Long-Term Average Performance of Flat-Plate Solar-Energy Collectors," Solar Energy, Vol. 7, No. 2 (1963).

49. Streed, E.R., Hell, J.E., Thomas, W.C., and Dawson, A.D., "Results and Analysis of a Round Robin Test Program for Liquid-Heating Flat-Plate Collectors," Solar Energy, Vol. 22, No. 3 (1978).

50. Burke Solar Heater, Burke Rubber Company, San Jose, California (1978).

51. Energy From the Sun in Your Back Yard, Fayco, Inc., Menlo Park, California (1977).

52. Day, J.A., Clarke, A.F., Dickenson, W.C., and Iontuono, A., Industrial Process Heat from Solar Energy, IECEC75 Record (1975).

53. Labor, II., "Solar Ponds: Large-Area Solar Collectors for Power Production," Solar Energy, Vol. 7, No. 4, pp. 189-194.

54. Weinberger, H., "The Physics of the Solar Pond," Solar Energy Vol, 8, No. 2, pp. 45-56 (1964).

55. Dickinson, W.C., Clark, A.F., Day, J.A., and Wouters, L.F., "The Shallow Solar Pond Energy Conversion System," Solar Energy, Vol. 18, pp. 3-10 (1976).

56. Jeppson, R.W., Analysis of Flow in Pipe Networks, Ann Arbor Science, Ann Arbor (1976).

Additional References Used For Data And General Background Information

57. ASHRAE 1975 Handbook of Equipment, American Society of Heating Refrigeration, and Air Conditioning Engineers, New York (1975).

58. ASHRAE 1976 Handbook of Systems, American Society of Heating Refrigeration, and Air Conditioning Engineers, New York (1976).

59. ASHRAE 1978 Handbook of Applications, American Society of Heating Refrigeration, and Air Conditioning Engineers, New York (1978).

60. Strock, C., Handbook of Air Conditioning, Heating and Ventilating, The Industrial Press, New York (1959).

61. Christian, J.E., Central Cooling - Compressive Chillers, Argonne National Laboratory Report ANL/CES/TE 78-2, Argonne, Illinois (March, 1978).

62. Baumeister, T., and Marks, L.S., Standard Handbook for Mechanical Engineers, McGraw-Hill, New York (1967).

63. Fink, D.G. and Carroll, J.M., Standard Handbook for Electrical Engineers, McGraw-Hill, New York (1968).

64. Meyer, C.F., "Status Report on Heat Storage Wells," Water Resources Bulletin, American Water Resources Association, (April, 1976).

65. Kormi, L and Keenan, J.D., "Energy Recovery in Brewery Industry," Journal of the Environmental Engineering Division, pp. 445-459, (June 1977).

66. Comly, J.B., Jaster, H. and Quaile, J.P., Heat Pumps-Limitations and Potential, GE Report 75 CRD185, General Electric Company, Schenectady, New York (September, 1975).

67. Kennedy, A.S., and Tschanz, J.F., Factors that Influence the Acceptance of Integrated Community Systems, Argonne National Laboratory Report ANL/ICES-TM-6, Argonne, Illinois (January, 1978).

68. "Heat Pump System Combines Features of Unitary and Central Approches," Architectural Record, (October, 1969).

69. Olivieri, J.B., "Internal Source Heat Pumps and Heat Recovery," ASHRAE Journal (October, 1967).

70. Cavalleri, G. and Galigno, G., "Proposal for the Production and Seasonal Storage of Hot Water to Heat a City," Solar Energy, Vol. 19 (1976).

71. Whitehead, E.R., "Using Heat Pumps for Energy Recovery," Plant Engineering, (April 14, 1977).

72. Hise, E.C. and Wilson, J.V., "A Heat Pump Cycle with an Air-Water Working Fluid," 12th IECEC, Vol. 1, (1977)

73. Heat Pump Technology: A Survey of Technical Developments, Market Prospects and Research Needs, HCP/M2121-01, U.S. Government Printing Office, Washington, D.C. (June, 1978).

74. Becker, H.P., "Energy Conservation Analysis of Pumping Systems," ASHRAE Journal, (April, 1975).

75. Tracor JITCO, Feasibility of a District Cooling System Using Natural Cold Water, Argonne National Laboratory ANL/ICES-TM-10, Argonne, Illinois (December, 1977).

76. Proceedings of Thermal Energy Storage in Aquifers Workshop, Lawrence Berkeley Laboratory, University of California, (December, 1978).

XVI. NOMENCLATURE

AEC	Annual Electric Consumption
AFC	Annual Fuel Consumption
ASC	Annual System Cost, \$/System
^{o}C	Degrees Celsius
COP_C	Coefficient of Performance for Cooling
COP_H	Coefficient of Performance for Heating
C_P	Specific Heat, Btu/lb-^{o}F
CRF	Capital Recovery Factor, percent
ELECT	Electric Utility Cost, \$
^{o}F	Degrees Farenheit
FUEL	Fuel Cost
i	Annual Interest Rate
I	Incident Solar radiation, Btu/ft^2
IDBT	Indoor Dry Buld Temperature
IPS	Iron Pipe Sizes
K	Permability of Aquifer, Gallons per Day per Square Foot
n	Years for Depreciation and Interest
NUM	Number of Units per Module
q	Flow Rate, gal/day-ft
Q	Flow Rate, ft^3/min or gpm
$Q_{E/House}$	Water Flow into House System, gpm
Q_H	Thermal Energy Rejected by Heat Pump
Q_L	Thermal Energy Absorbed by Heat Pump
ΔS	Change in Entropy
t	Time
T_A	Ambient Temperature, ^{o}F

T_F	Fluid Temperature for Solar Collector, ^{o}F
T_H	Temperature at Hot End of Heat Pump System
T_L	Temperature at Cool End of Heat Pump System
UAOC	Annual Unit Operating Cost, $/year
UCC	Unit Capital Cost, $/unit
V	Length/units Time ft/sec, ft/day
W	Work Input Btu or KWh
WBT	Indoor Wet Bulb Temperature
γ	Fluid Specific Weight
η	Efficiency
μ	Fluid Viscosity
ρ	Density
ϕ	Porosity Percent
ϕ	Availability, Btu

Distribution

ANL

J. Calm (50)
A. Kennedy (25)
K. Macal
T. Marciniak
J. Pascual (2)
V. Rabl
J. Tschanz
ANL Libraries (C. Archer)
ANL Washington (B. Graves) (10)
TIS Files (E.N. Pettitt)

DOE

S. Cavros
J. Kaminsky
G. Leighton
J. Millhone
J. Rodousakis (25)
I. Sewell
D. Walter
DOE-TIC (27)

External

C. Brett, University of Alabama (15)
P. Swenson, Consolidated Natural Gas Service Company (2)
F. Dubin, Dubin-Bloome Associates (2)
H. Lorsch, Franklin Research Center (2)
D. Wade, Georgia Tech Research Institute (2)
P. Anderson, Honeywell Energy Resources Center (2)
L. Katter, Rocket Research Company (2)

THIS REPORT SUPERSEDES ANL/ICES-TM-30