GROUNDWATER HEAT-PUMP HVAC DEMONSTRATION PROJECT.

PHASE I DESIGN DEVELOPMENT

By

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GROUNDWATER HEAT PUMP HVAC DEMONSTRATION PROJECT

PHASE I - DESIGN DEVELOPMENT

EXECUTIVE SUMMARY

Groundwater heat pumps have significantly higher efficiencies in the Texas Gulf Coast Region than conventional Heating, Ventilating, and Air Conditioning systems (HVAC). With extensive groundwater sources on the Gulf Coast, consumers can derive potential energy savings between 30 and 50 percent by switching to groundwater heat pumps. At present utility rates (1978 figures), payback periods are typically six years or less compared to gas heating and three years compared to electric heating.

Because the design of HVAC systems depends on the size and function of individual buildings, savings in energy and cost can best be determined at the present time by case studies focusing on typical structures.

This study reports the investigation of heat pump technology and application in the Gulf Coast area and provides the following information:

(1) Introduction to energy needs

(2) Basic heat pump technology

(3) History of the development of groundwater heat pump systems

(4) Conceptual design of groundwater heat pump systems

(5) Regulatory and environmental considerations

(6) References

This comprises the Final Report of the Groundwater Heat Pump HVAC Demonstration Project [Project #78-C-6-1], Phase I - Design Development, to the Energy Development Fund of the Texas Energy Advisory Council which authorized this study.

1. Introduction

Heat pumps work and they are efficient. A heat pump is a device that transports heat from one place to another and is capable of reversing the flow of heat on demand. A heat pump can therefore heat and cool a building. In the Gulf Coast area, groundwater is a stable source of heat and a "sink" for holding return heat. At a steady temperature of about 19.4° C (67° F), groundwater in the Texas Gulf Coast is nearest the desired space temperature for most heating and cooling and improves the efficiency of the heat pump.

In times of cheap energy, there was little incentive to use alternate energy systems. With the rapid rise in the cost of energy, there is a need to investigate and use efficient methods of supplying energy. Because of a limited market, few manufacturers have offered groundwater heat pump packages. With relatively inexpensive groundwater sources in the area and improved reliability of equipment, manufacturers may have sufficient encouragement to produce groundwater heat pumps.

2. Basic Heat Pump Technology

Heat pumps are cyclic machines that transfer heat from a low temperature source to a higher temperature sink. The window-installed air conditioner is a typical heat pump. It is not possible to convert the conventional air conditioner to a room heater. But heat pumps can be used throughout the year if a four-way valve is inserted after the compressor so that the flow of refrigerant can be reversed and the evaporator made to function as the condenser. Most commercial year round systems utilize this reversing valve concept.

The key to improved efficiency is the coefficient of performance (COP), which is affected by the varying source/sink temperatures. In general high COPs are obtained if the sources and sinks have temperatures close to the controlled temperature desired for the facility being heated or cooled.

Although there are various choices for the source and sink of heat pump air conditioning cycles, viable alternatives in the Gulf Coast area are air and groundwater. Air heat pumps have the disadvantage of having their COPs vary in inverse proportion to the heating or cooling need. During the summer, for example, maximum cooling is required when air temperatures are highest. The COP drops appreciably as the temperature increases. There is a similar effect in winter. The COP decreases as the outside air temperature decreases while the heating requirements increase. If temperatures drop below 0° C, the heat pump coils tend to frost over with a detrimental effect on heat pump performance.

A better alternative has a temperature consistently close to 22° C (72° F) to generate high COPs. These criteria are best satisfied by groundwater, which maintains a constant temperature of about 19.4° C (67° F) and is available extensively at shallow depths (less than 100 feet) so that deep wells are not required for non-potable water.

In the Gulf Coast area, heating is generally provided by gas or electric furnaces. These furnaces have a maximum COP of 1, since heat is delivered directly into the room (the actual COP may vary between 0.5 and 1.0). Compared with these furnaces, groundwater heat pumps show about a 4-to-1 advantage in energy efficiency. Thus, there is a significant potential for energy savings if gas or electric heating is replaced by groundwater heat pumps.

3. History of the Development of Groundwater Heat Pump Systems

The first groundwater heat pump system was built in 1948 by physicist Carl Nielsen of Ohio State University; his second system, installed in 1955, is still in operation. Engineers were attracted to these systems in the 1950's because they offered a simple means of heating and cooling with advantages in areas having an extreme temperature range.

In 1957, Battelle Memorial Institute in Columbus, Ohio, installed a groundwater heat pump as an alternate to coal and oil heating; over 300,000 square feet of space is cooled and heated with this system. In 1959, a Houston builder installed 40 3-ton groundwater heat pumps in new homes in a subdivision offering buyers a revolutionary new heating and cooling system. Five of these units are operational today and show heating and cooling costs 50 to 70 percent lower than homes that abandoned the original system and installed conventional air-to-air units.

Systems in the Houston project were designed for homes of about 1,300 square feet. Each home had a shallow water well less than 80 feet in depth and a discharge well at a different depth. Various aquifers were used between 30 and 80 feet. Wells had various lateral separations, some as close at 15 feet with the average being about 30 feet. These wells - one inch in diameter - had small pumps delivering four to five gallons of water per minute.

The 50-ton size seems to be the upper limit for single packaged systems; larger capacities can be designed but involve custom sizing of individual pieces of equipment, such as compressors and heat exchangers.

Because groundwater heat pumps were introduced when the technology was not fully reliable, widespread dissatisfaction with many units hindered development of the industry; today few manufacturers produce this equipment. Presently, costs of equipment and installation are not standardized. But those systems that were diligently maintained by the owner had few problems and proved long-lasting.

4. Conceptual Design of Groundwater HVAC Systems

To evaluate potential for groundwater heat pumps, investigators at the University of Houston studied four types of buildings that are in planning, under construction, or in existence:

(1) Residential Home

(2) Lab/Research Building

(3) School Building

(4) Office/Manufacturing Facility

Each building is divided into zones. The residential home has separate A/C units for the bedroom area and the living room/kitchen area. The lab/research building is split into three zones with separate air handlers for each zone. Instead of a refrigerant reversing system, a groundwater reversing system is conceived for the lab/research and school buildings. In the office-manufacturing facility, two 8-ton water source heat pumps replace a roof top 16-ton air cooled condensing unit. Heating and cooling loads for the buildings were obtained by standard ASHRAE techniques calculating the heat lost or gained through the building walls.

Economics of the groundwater heat pump is based on seasonal energy requirements. Formulas computing the economics require an appropriate EER (energy efficiency ratio) or COP (coefficient of performance) and the corresponding cost of primary energy. The assessed efficiency of groundwater heat pumps for cooling is an EER of 11 compared to 8 for air-to-air heat pumps. Heating COP for groundwater heat pumps is 4.0 compared to 2.5 for air-to-air heat pumps and 1.0 for electric furnaces.

The following findings demonstrate the economy of groundwater heat pumps in the Gulf Coast area:

(1) Net energy savings are a minimum of 30 percent and may be as high as 50 percent annually.

(2) Payback periods are shorter using groundwater heat pumps to replace conventional electrical resistance heating than to replace gas heating. If tax rebates

are considered, the payback period may be as short as three years; six years in the extreme.

If a groundwater heat pump is installed, yearly savings in the four cases show economies ranging from 30 to 50 percent. Savings in cooling costs are from 20 to 27 percent; in heating, from 38 to 75 percent.

CASE	ANNUAL $ SAVINGS (1978 dollars)	ANNUAL % SAVINGS (1978 dollars)
Residential Home	338	30
Lab/Research Building	2,276	50
School Building	12,122	47
Office-Warehouse	879	31

Placement of the supply and injection wells of the groundwater heat pump is a major design consideration. The size of the project determines whether several wells will be needed or one well. The ideal is to have two aquifers at different depths with groundwater cycling between the aquifers. The same aquifer may be used for both wells provided the wells are distant enough from each other to prevent temperature migration. During the operation of a groundwater heat pump system in the heating mode, the injected water will be cooler than the supply. In the cooling mode, the injected water will be warmer than the supply water. A well spacing of 40 feet is sufficient for a 5-ton unit, for example. A spacing of 50 feet would be conservative, yet convenient, for a typical residential lot.

For most applications up to 25 tons, adequate water can be found in the Gulf Coast at 100 feet or less. The COP of groundwater heat pumps assumed in this study takes into consideration the pumping power needed for the wells up to 100 feet depth, for which pumping power constitutes only 7.7 percent of the total power consumption. Deep wells necessary for large systems may require up to 20 percent of the energy; however, several shallow supply wells can usually be used instead of one deep well to reduce the energy requirements. For example, the Battelle Memorial Institute's system uses several 50-foot wells and one 400-foot well.

5. Regulatory and Environmental Considerations

The Harris-Galveston Coastal Subsidence District is the authority governing wells not drilled for the purpose of supplying potable water. The Subsidence District operates by House Bill No. 552, passed 12 May 1975, as amended by House Bill 390, passed 19 May 1977 (Vernon's Annotated Texas Statutes, Chapter 284, Water Auxiliary Laws). Although water is extracted from the ground through one well and injected in another, the groundwater heat pump is a closed system. It does not "use" or deplete the water supply, it cannot introduce foreign matter into the water supply, and does not give rise to health problems. The law exempts wells with a case diameter under five inches from the requirement of a permit. The law is not clear about injection wells, but an informal response from the Subsidence District indicates that exemptions would be readily granted for the installation of groundwater heat pumps.

The U. S. Energy Act of 1978 provides a "residential energy credit" available for expenditures after 20 April 1977 for equipment which uses a "renewable energy source" to heat, cool, or provide hot water for a taxpayer's principal residence. This amounts to 30 percent of the first $2,000 and 20 percent of the next $8,000 of the taxpayer's expenditure. Labor costs are included in deductibles, but they are subject to interpretation by the Internal Revenue Service.

There are few environmental problems in the installation of groundwater heat pumps. Water is not, in effect, removed from the source but recirculated. Temperature differences in the cooling and heating modes are not considered significant problems. In separate aquifers, temperature migration is not pertinent.

Although the study determined that there is unlikely to be any effect upon pressure when both the supply and injection wells are in the same aquifer, the use of different aquifers would be more likely to produce a measurable effect. Geologists surveyed were of the opinion that any pressure effect would be negligible in the Texas Gulf Coast region due to the size of the aquifers involved.

The corrosion of materials presents little problem even in large systems. Relatively simple fixes like magnesium anodes are sufficient to eliminate corrosion completely.

6. References

There is relatively little reference material on groundwater heat pump systems as a total HVAC package. Investigators provide a selective bibliography of information culled mainly from professional journals such as the *ASHRAE Journal*, *Heating & Ventilating News*, and *Building Systems Design*.

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PROJECT SUMMARY

The applicability, availability and economics of groundwater heat pump HVAC systems were investigated with respect to their potential for use in the Texas Gulf Coast Region. Groundwater heat pumps have significantly higher efficiencies than conventional HVAC systems. This fact, coupled with the extensive groundwater resources of the Gulf Coast area, suggests potential savings in energy use in this area.

A survey of heat pump manufacturers indicated that packaged water source heat pumps usable with groundwater are available up to about the 50 ton size though the number of companies with such systems are very few. Large tonnage systems also exist though the units are custom manufactured. A system at Battelle (Columbus) Laboratories is the largest to date.

Systems were designed for four building types to determine typical equipment and energy costs and payback times for groundwater systems when compared to conventional systems. It was determined that energy savings range between 30 and 50 percent and payback periods are typically six years or less with present utility rates. Generally, the payback periods for the additional capital cost of groundwater heat pump systems are about six years when compared with gas heating and three years when compared with electrical resistance heating.

The environmental and other implications of the mode of groundwater usage (i.e. between the same or different aquifers) were addressed. Though no general theoretical conclusions are

possible due to the site specific nature of the problem, data from existing groundwater systems and discussions with reputed geologists indicate that no significant temperature or pressure effects are likely throughout the lifetime of these systems. A crude thermal model of the temperature migration problem indicates that for typical homes well spacings need not be greater than 50 feet if the same aquifer is used for supply and discharge.

The legal and regulatory issues as they pertain to groundwater usage for heat pumps are confusing. However, it is felt that since the water is mainly being recirculated, not used, and is being reinjected, no special permits are necessary. Builders in this area already installing these systems have not faced any problems in this regard.

The Energy Tax Act of 1978 provides tax credit incentives to encourage certain energy conservation investments. It is the opinion of the National Water Well Association that groundwater heat pump systems are contemplated in the language of the law and therefore tax credits of up to $2,200 may be available to homeowners installing such systems.

GROUNDWATER HEAT PUMP HVAC DEMONSTRATION PROJECT
PHASE I - DESIGN DEVELOPMENT

1.0 INTRODUCTION

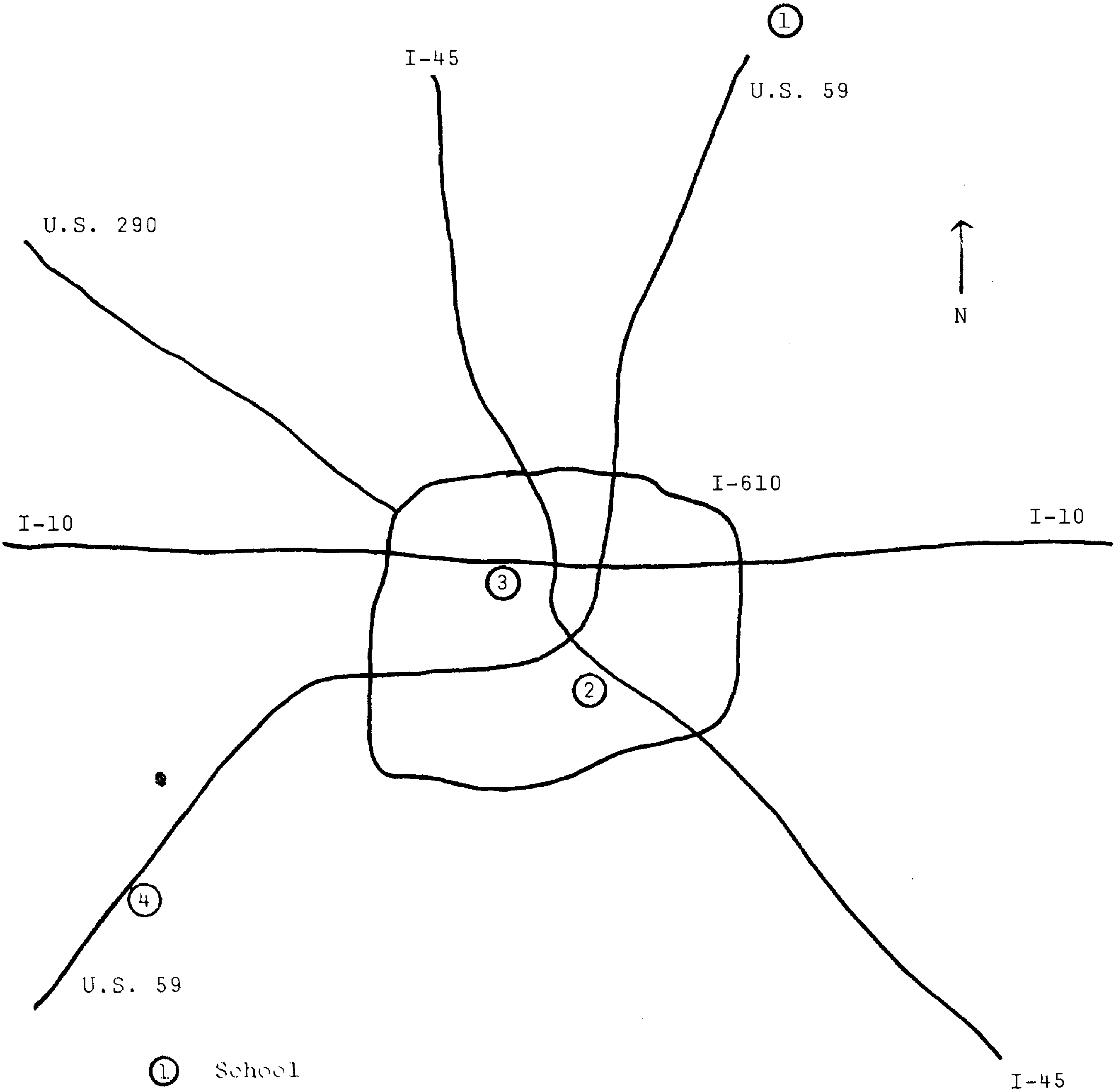
There is an urgent need to reduce the energy required for space heating and cooling. As one response to this need, the Texas Energy Advisory Council funded this investigation of the use of groundwater heat pumps in the Texas Gulf Coast Region.

Any device which can transport heat from one place to another can properly be called a heat pump. For the purposes of this study, a heat pump is a device which can not only transport heat, but is also capable of reversing the flow of heat upon demand. This system presupposes a heat source and a means of receiving heat where it is desired, the heat sink. The common heat pump used in space heating and cooling extracts heat from the air and transports it into a space to be heated or, in the cooling cycle, extracts heat from the space and "dumps" it into the air. Thus, air is both the heat source for heating and heat sink for cooling. The quality of air as a heat source/sink varies considerably with its ambient temperature. At the upper and lower extremes it is inadequate - heat pumps cannot efficiently "dump" heat into air when the temperature is much over 29.5° C (85° F). It is when the heat source/sink is nearest the desired space temperature that heat pump efficiencies are the highest.

One stable heat source/sink within the desired temperature range is groundwater, which has a steady temperature of about 19.4° C (67° F) in the Texas Gulf Coast Region. Groundwater is plentiful at relatively shallow depths in this area and should be developed as an energy source. What is sought in this effort is a statement regarding the impact groundwater heat pumps could have upon energy utilization in this area.

The concept of using groundwater as the heat source/sink for a heat pump is not new. In fact, this type of system has been in limited use at least since 1948. There is no question about whether or not they work, nor is there any debate about their potential efficiency. However, as the market for them has been rather limited, very few manufacturers have offered groundwater heat pump packages. One major drawback has been the need for a relatively inexpensive groundwater source. This problem, coupled with a lack of information and the complexity of the system, served to deter its development in times of cheap energy, when there was simply insufficient incentive to overcome these difficulties. Also, heat pumps of any type have only recently become popular, having been plagued by poor reliability and high cost for a number of years.

The rapid rise in the cost of energy has made more efficient systems an imperative. Higher initial capital cost and complexity are no longer the barriers they formerly were. The U.S. Department of Energy's new slogan, "If it saves energy, it'll pay for itself," is well taken. This study is limited to gathering information for the purpose of analyzing the operational characteristics, economics, and potential environmental impact of groundwater heat pump systems. An additional task is to provide gross designs of such systems for a variety of buildings in order to obtain some general information on system applications. These buildings, a school, a laboratory, an office-warehouse facility, and a house, are located as shown in Figure 1-1.



① School

② Laboratory

③ Office

④ House

2.0 BASIC HEAT PUMP TECHNOLOGY

2.1 Introduction

The heating and cooling of any enclosed space involves the transference of heat from or into that space in order to maintain the temperature within that space at a preset or desired level. The physical processes involved are illustrated in Figures 2-1a and 1b. Thus, when T_1, the outside temperature, exceeds T_2, the room temperature, heat tends to leak into the room and raise its temperature. To maintain the room temperature at the preset value, an equivalent amount of heat, Q, has to be removed from the room and rejected to a heat "sink." Conversely, if T_2 exceeds T_1, heat has to be added to the room from a heat "source" in order to compensate for heat loss. In either case the physical processes are the same with directions reversed only. These basic characteristics are summarized in Table 2-1.

Table 2-1

Fundamental Aspects of Heating and Cooling

	Cooling	Heating
1.	The surrounding temperature is higher than space temperature.	The surrounding temperature is lower than space temperature.
2.	Therefore, heat flows spontaneously from the surroundings to the space.	Therefore, heat flows spontaneously from the space to the surroundings.
3.	To maintain space temperature an equivalent amount of heat has to be rejected from the space.	To maintain space temperature an equivalent amount of heat has to be added to the space.

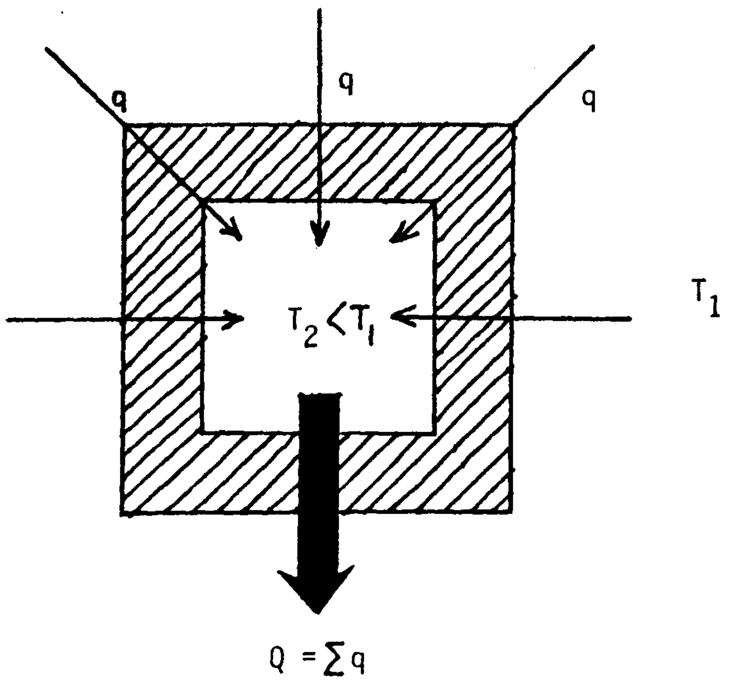


FIG. 2-1a HEAT FLOWS IN COOLING

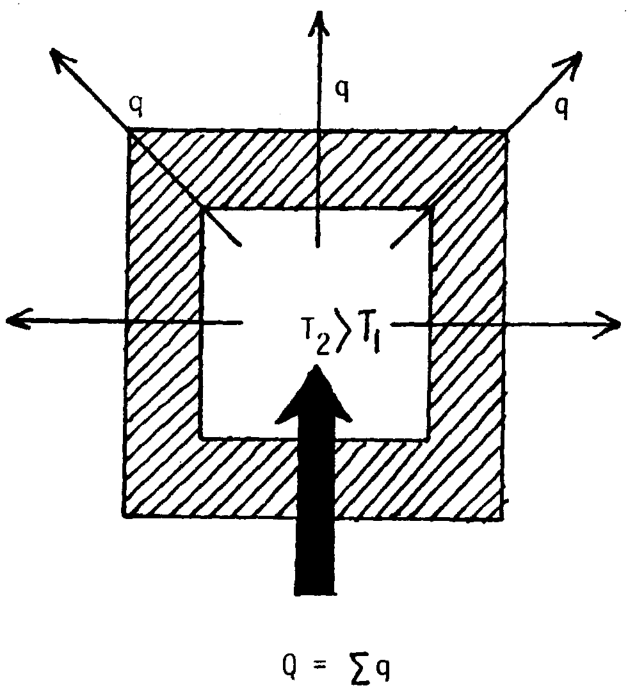


FIG. 2-1b HEAT FLOWS IN HEATING

q = heat leaks
Q = compensating heat flows

"Air conditioning" in the general sense is, therefore, concerned only with removal or addition of heat from or to an enclosed space to maintain its temperature at the desired level. From a mechanical viewpoint, air conditioning is concerned with heat sources and sinks and efficient mechanisms for removing and adding the desired amounts of heat. For example, at present we see the use of gas and electric furnaces (which are heat sources) in which simple convective flow of room air is used to transfer heat from the furnace to the room. On the other hand, air conditioners produce cooling by absorbing heat from the room air and then rejecting the heat to the outside air which works as the sink. We have lately also seen the popularization of "all season" heat pumps which use outside air both as the heat source (for heating) and the heat sink (for cooling). Generally speaking, mechanical heat pumps (as opposed to chemical heat pumps) are the most efficient devices for moving heat. However, there are several options for the heat source/sink besides air. As will be seen below, groundwater is the most appropriate choice as the heat source/sink, especially in the Texas Gulf Coast Region.

2.2 Heat Pumps

Heat pumps are cyclic machines that transfer heat from a low temperature source to a higher temperature sink. The operating cycle of a typical heat pump, e.g., a conventional, window-type air conditioner, is illustrated in Figure 2-2. Heat pumps generally utilize a circulating fluid, called a refrigerant, which alternately absorbs and rejects heat at

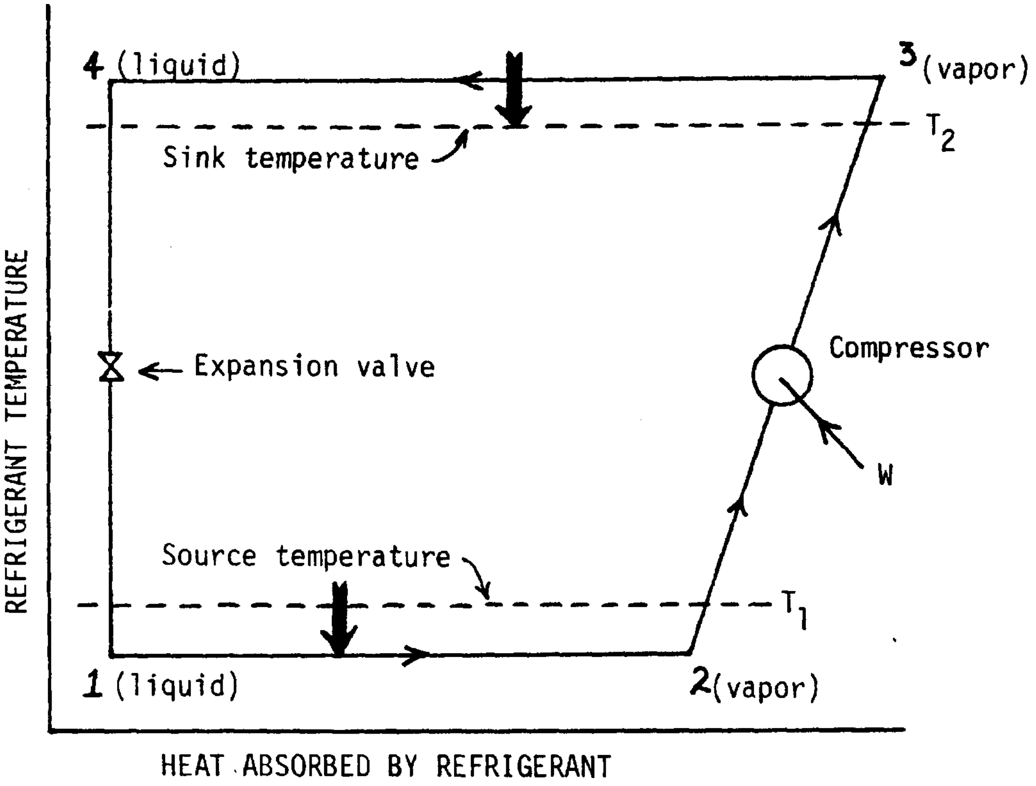


FIG. 2-2 OPERATING CYCLE OF A TYPICAL HEAT PUMP

different parts of the cycle. Thus, as in Figure 2-2, cold refrigerant liquid (at "1") absorbs heat from the low temperature source (at T_1) and becomes vapor. The vapor temperature is then increased to a level greater than the sink temperature (T_2) by the addition of energy (W) from an external source. The hot vapors then reject heat to the sink and in the process condense to a liquid ("4"). The hot liquid is then expanded through an expansion valve to cool it back to the temperature at the start of the cycle. This cycle is automatically repeated until the required amount of heat has been transferred.

The extra energy used to elevate the refrigerant from state "1" to "2" is supplied by either a compressor (in a mechanical heat pump) or simply heat (in a chemical heat pump).

2.3 Applicability of Heat Pumps for Heating and Cooling

The simplified description of the heat pump cycle shows that in essence a mechanical heat pump consists of a low temperature evaporator (for absorbing heat), a high temperature condenser (for rejecting heat), a compressor, and an expansion valve. It is evident that if, for example, during summer, the evaporator is placed in the room and the condenser is placed outside, then the heat pump can in principle transfer heat from the room air to the outside air and thereby maintain the room temperature at the desired level. Conventional air conditioners are actually heat pumps working in this, the cooling mode.

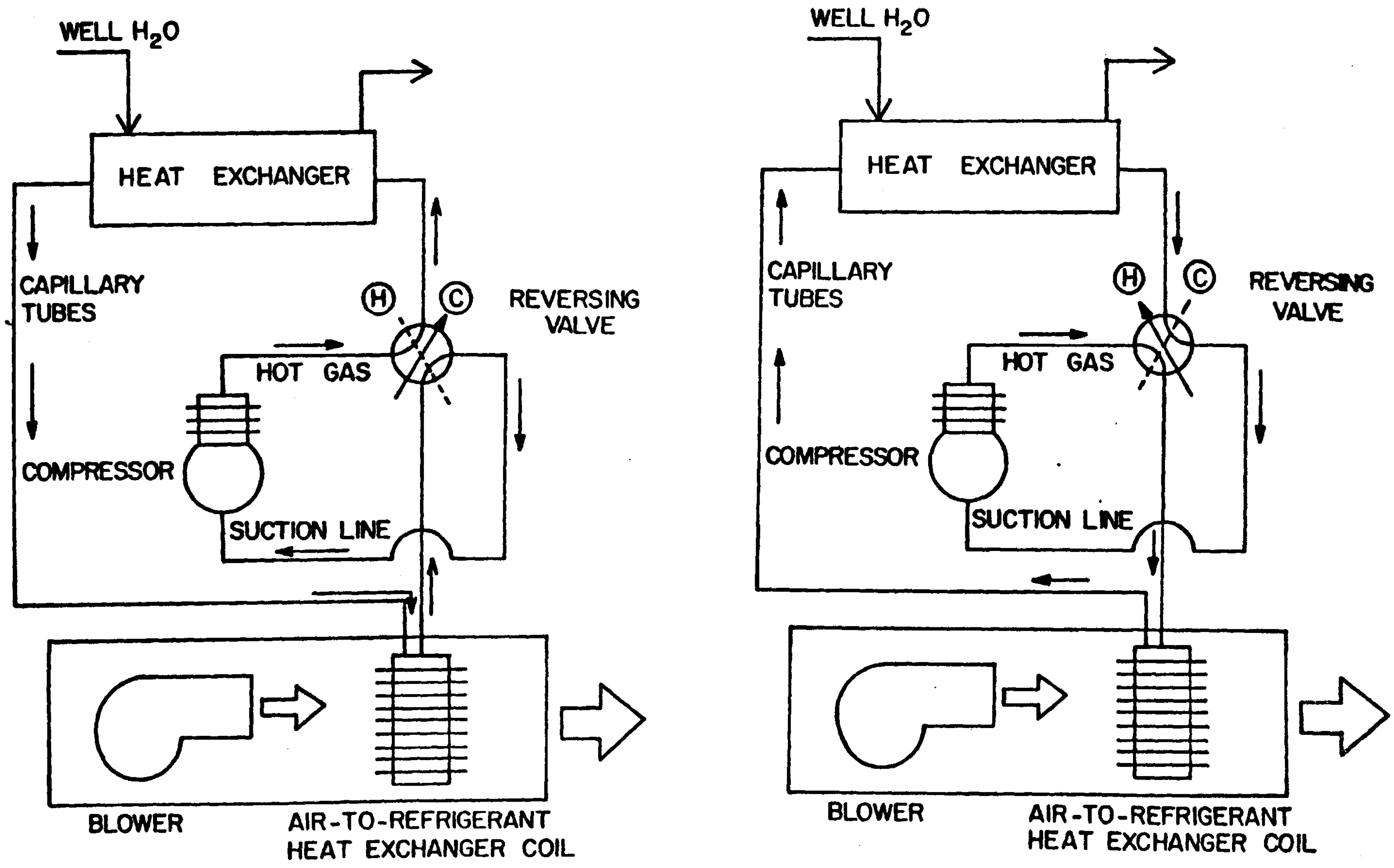
During winter, however, one needs to add heat to the room. If the same heat pump is to be used then the location of the evaporator and the condenser will need to be reversed so that heat from the outside air can be transferred to the room. Such a reversal is a practical impossibility. However, if a four-way valve is inserted after the compressor then the flow of refrigerant can be reversed and the evaporator can be made to function as the condenser, or vice versa, depending on the requirement. Thus, the same heat pump can be used year round. Most commercial year round systems utilize this reversing valve concept. Figure 2-3 shows the two operating modes of such systems.

2.4 Efficiency of Heat Pumps

As described in the previous sections, a heat pump operates between a low temperature source and a high temperature sink. Thus, as illustrated in Figure 2-4, the heat pump accepts some energy, Q_1, from a source at temperature T_1 and then rejects Q_2 to a sink at temperature T_2 where $T_2 > T_1$. This transference, however, is aided by some energy input, e.g., power to the compressor, denoted as W. Since energy is always conserved, it is seen that the energy rejected, Q_2, is actually the sum of the energy absorbed, Q_1, and the energy input, W; i.e.,

$$Q_2 = Q_1 + W \tag{1}$$

If the heat pump is in the heating mode, then Q_2 is net heat produced. Conversely, if it is working in the cooling mode, then Q_1 is the energy absorbed or the cooling produced. Thus, a useful measure of the efficiency of a heat pump is simply the ratio of the heating or cooling produced to the input energy. This ratio is generally greater than 1 (see Table 2-2); therefore, it is instead called the coefficient of performance (COP).



COOLING MODE

HEATING MODE

Fig. 2-3

OPERATING MODES OF REVERSIBLE HEAT PUMPS

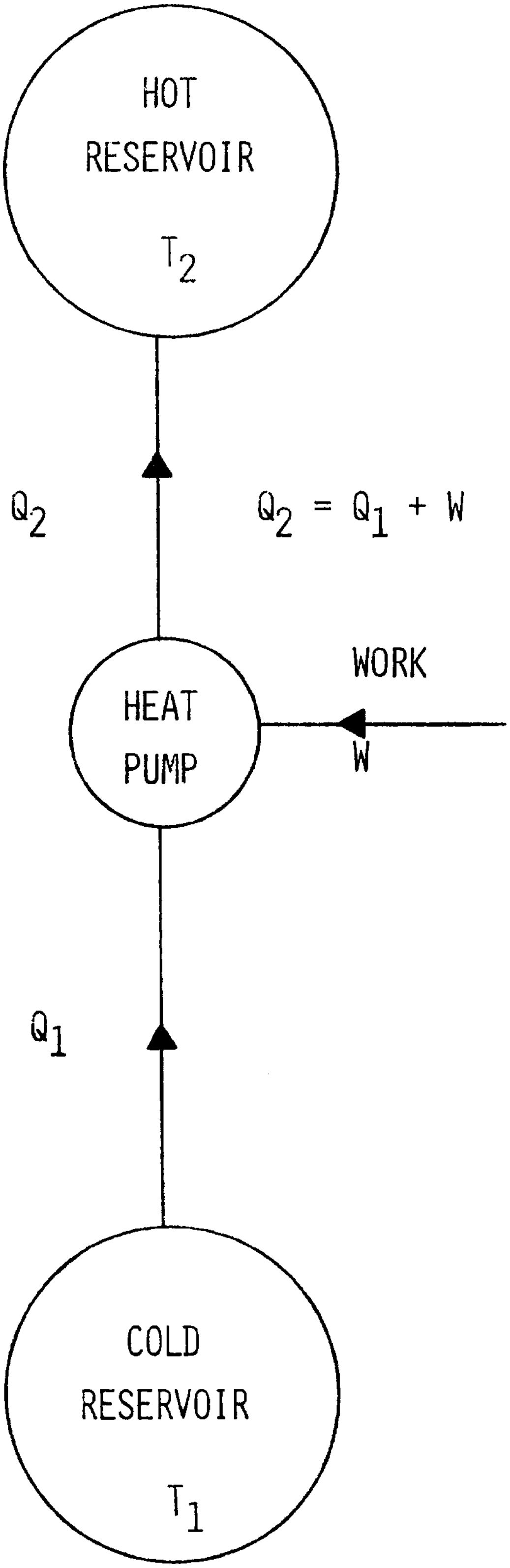


Fig. 2-4
CONCEPTUAL HEAT PUMP

Thus,

$$COP_{heating} = Q_2/W = Q_2/(Q_2 - Q_1) \quad (2)$$

and

$$COP_{cooling} = Q_1/W = Q_1/(Q_2 - Q_1). \quad (3)$$

A priori, it is not possible to estimate what the actual operating COPs might be for a particular set of source/sink temperatures. However, it is possible to estimate the maximum theoretical efficiency from thermodynamic theory.

It is a well known principle of thermodynamics that the most efficient way to convert thermal energy to mechanical energy is by the use of a Carnot engine (1). A corollary to this is the principle that a reversed Carnot engine is also the most efficient heat pump cycle. The COP of a heat pump operating in the reversed Carnot mode is given by

$$COP_{cooling} = 1/(T_2/T_1 - 1) \quad (4)$$

$$COP_{heating} = 1/(1 - T_1/T_2) \quad (5)$$

where T_1 and T_2 are the temperatures in Figure 2-4. <u>It should be noted that T_1 and T_2 are the absolute temperatures (i.e. Kelvin or Rankine).</u>

Equations 4 and 5 estimate the maximum theoretical efficiencies. Actual efficiencies are much lower due to mechanical inefficiencies, heat transfer limitations, and frictional resistance. However, equations 4 and 5 are important in that they show the effect of varying source/sink temperatures on the COP. Table 2-2 shows the cooling COPs obtained for three different sink temperatures (T_2) for a 23° C (74° F) (T_1) cooling application.

Table 2-2

Variation of Ideal COP with Sink Temperatures

$T_1 = 23° C (74° F)$

T_2(°C)	30	40	50
$COP_{cooling}$	42.3	17.4	11.0

It is observed that the COP is maximized as T_2, the sink temperature, approaches T_1, the controlled temperature. In fact, the COP is infinity if $T_2 = T_1$. A similar conclusion results on the heating side.

In general then, high COPs are obtained if the sources and sinks have temperatures close to the controlled temperature.

2.5 Source/Sink Combinations

In the previous section we have indicated that a reversed Carnot cycle is the most efficient means of transferring heat between a heat source and a sink. However, there are various choices for the source and sink of heat pump air conditioning cycles. Ambrose (2) discusses some of the various combinations. However, the viable alternatives in the Gulf Coast area are air and groundwater.

2.5.1 Air as a Source and Sink

The most popular choice so far has been air as both source and sink due to its easy availability and its infinite extent. However, air heat pumps have the peculiar disadvantage that their COPs vary in inverse proportion to the heating or cooling need. For example, during summer maximum cooling is required when the air temperatures are the highest. However, as is evident from

Table 2-2, the COP drops off appreciably as the temperature increases. There is a similar effect in winter. The COP decreases as the outside air temperature decreases while the heating requirements increase. In fact, if temperatures drop below 0° C, the heat pump coils tend to get frosted over, thereby having a detrimental effect on the heat pump performance. The winter problem is illustrated in Figure 2-5, which shows the heating requirements for a typical residence and the capacity of typical air-to-air heat pumps as a function of the outdoor temperature.

It is observed that as one sizes the pump to handle lower outside temperatures one ends up with a greater excess capacity. Thus, for localities with extreme winter temperatures, the heat pumps have to be specified to handle some fraction of the total heating load, the balance being supplied from supplemental sources like electric resistance heating. This problem is not too severe in the Gulf Coast area since summer cooling requirements are generally greater than winter heating requirements. Thus, some excess capacity is a built-in feature. But for localities in the panhandle of Texas, for example, the problem is crucial and air-to-air heat pumps have to be backed up by supplemental electric strip heaters, which inherently have a lower COP. Quoting Harris and Conde (3):

> Present day (air-to-air) heat pumps are designed to operate down to 0° F outdoor temperature on the heating cycle. Below this point...the system must heat entirely by supplemental electric heat at a COP of 1.0.... Overall COP can average 2.2 to 2.3

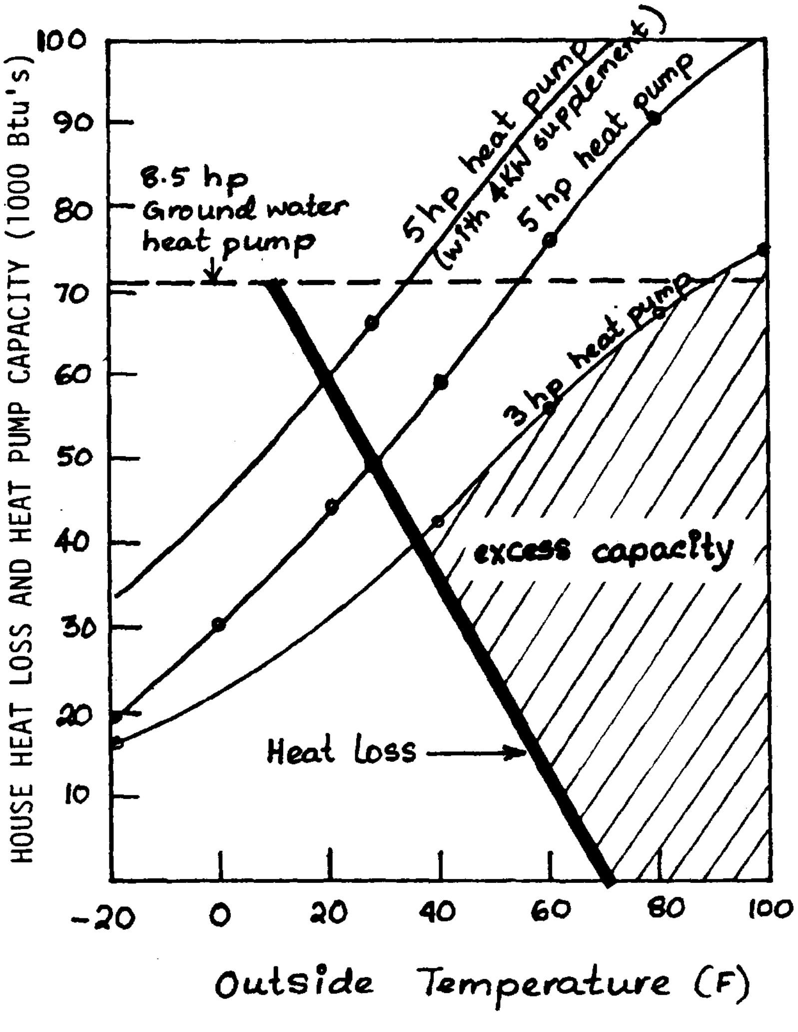


Fig. 2-5

VARIATION OF HEATING LOAD AND
HEAT PUMP CAPACITY VS. OUTSIDE AIR TEMPERATURE (3)

for Midwestern states and can approach 3.0 for the entire season in areas such as California and Florida. Critical unbalance, which causes compressor shutdown and straight electric heating, should not exceed 5 to 10 percent of the total operating hours for heat pump feasibility.

2.5.2 Groundwater as a Source and Sink

While air is a common and convenient source/sink, it obviously has certain disadvantages. A better alternative should be similarly extensive and also have a temperature consistently close to 22° C (72° F) to generate high COPs. These criteria are best satisfied by groundwater, which is both extensive and maintains a constant temperature between 8° C (47° F) and 22° C (72° F) over a majority of the United States (Figure 2-6). The groundwater in the Texas Gulf Coast Region exhibits a constant temperature of about 19.4° C (67° F) and is available extensively at shallow depths (less than 100 feet) so that deep wells are not required.

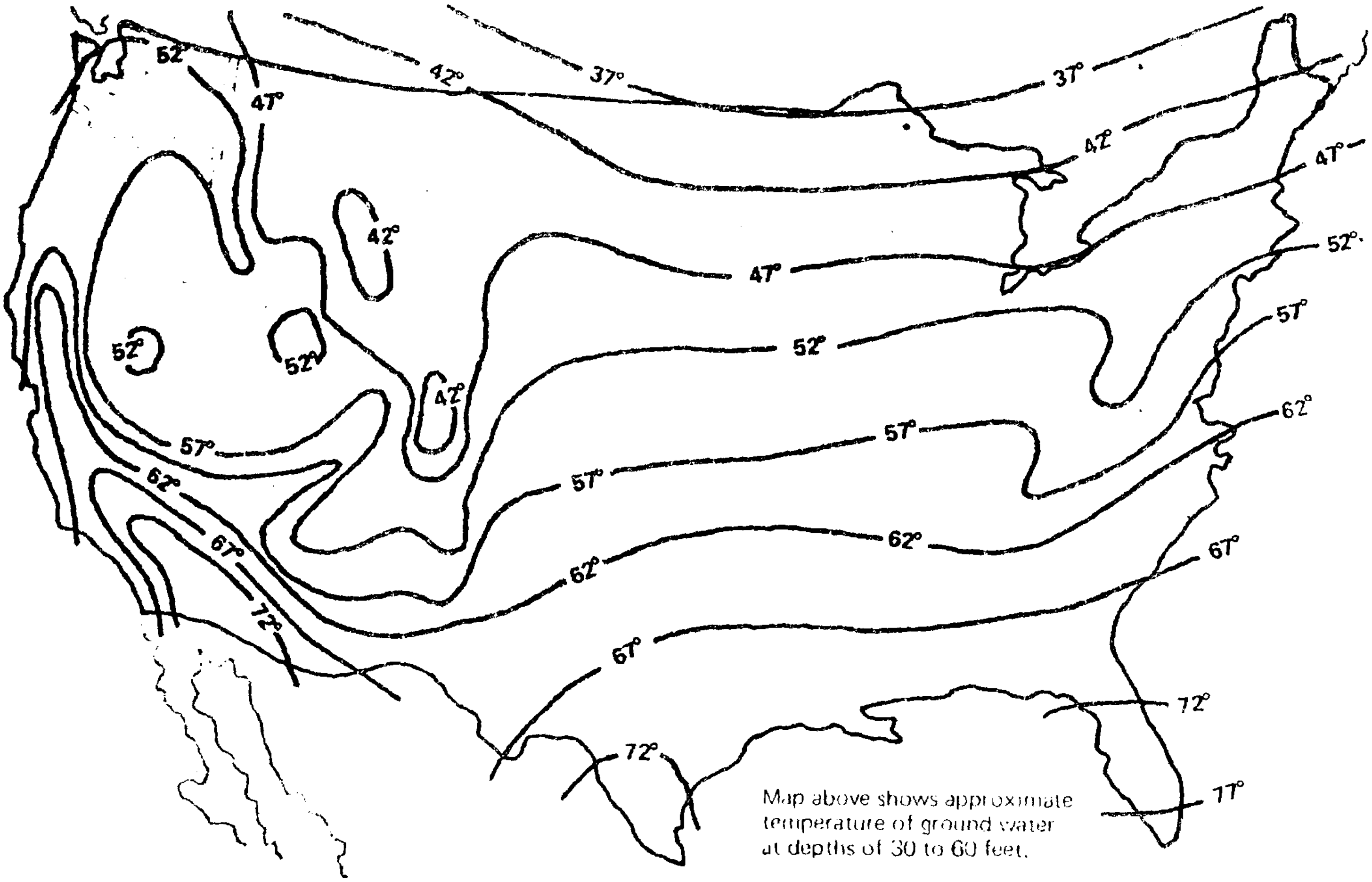
2.6 Comparison of COPs

2.6.1 Groundwater vs. Air

The advantage gained by using groundwater instead of air is easily seen by comparing the probable COPs for the two cases. For this comparison we will use a modified form of the COP, rather than the ideal form (equations 4 and 5), in order to better estimate actual COPs. Appendix 1 gives the necessary modifications to equations 4 and 5. The modified equations are

$$COP_{cooling} = 0.5/\{(T_2 + 15)/(T_1 - 15) - 1\} \qquad (6)$$

$$COP_{heating} = 0.5/\{1 - (T_2 - 15)/(T_1 + 15)\} \qquad (7)$$



Map above shows approximate temperature of ground water at depths of 30 to 60 feet.

Fig. 2-6

GROUNDWATER AVAILABILITY IN UNITED STATES

Table 2-3 compares the heating COPs (i.e. equation 7) for groundwater and air.

Table 2-3

Heating COPs with Groundwater and Air

Room Temp (T_1) - 23° C (74°F)

Air temp, T_2(°C)	5	10	15	20
Air COP	3.2	3.6	4.1	4.7
Groundwater COP (18°C)	4.4	4.4	4.4	4.4

It should be noted that the Table 2-3 COPs are only estimates, not manufacturers' specifications.

The results are also plotted in Figure 2-7 to show the approximate seasonal variation of averaged COPs for the two systems. Since the groundwater temperature is constant, its COP also remains constant, whereas the air COP varies in direct proportion to the air temperature. The shaded area between the curves is therefore an indication of the energy savings possible using groundwater when compared with air. The savings are actually greater when the problems of air heat pumps, mentioned in Section 2.5.1, are also considered. A similar analysis can also be done for cooling applications and will lead to the same general conclusions.

Another important advantage of groundwater systems is illustrated in Figure 2-5. It is seen that a groundwater heat pump can be sized to handle the peak winter load without the excess capacity becoming unreasonable. Thus, for severe winter climates, a groundwater system is a definite boon.

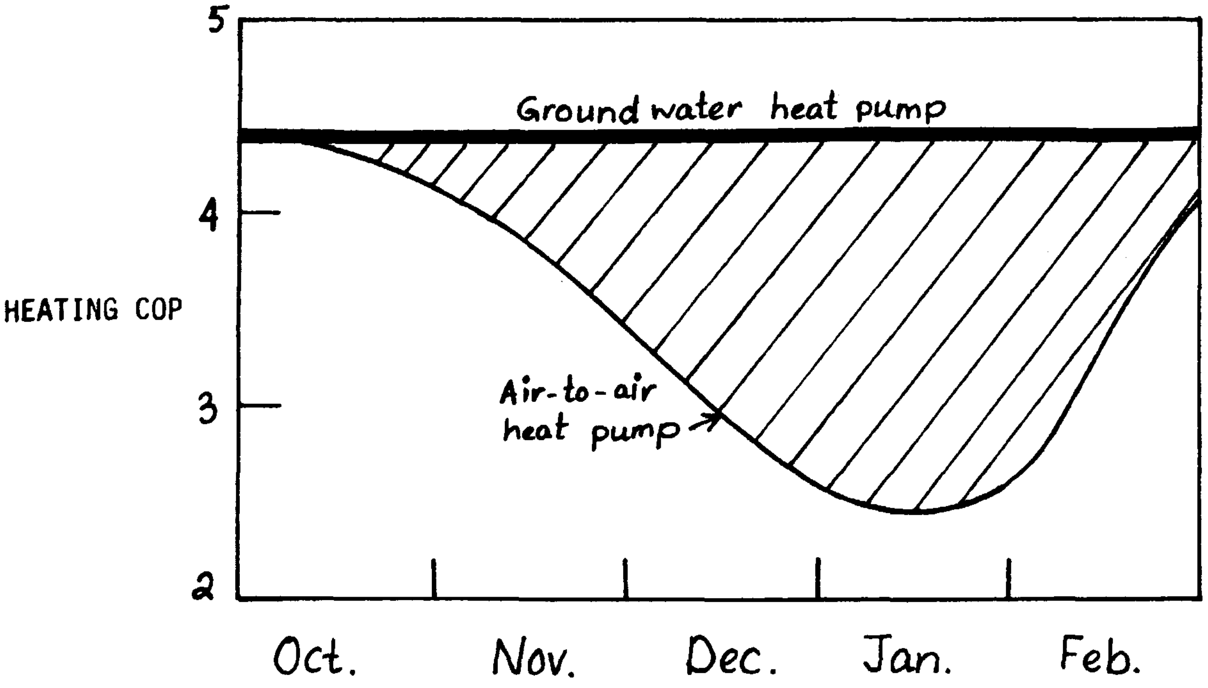


Fig. 2-7

SEASONAL VARIATION OF COP

2.6.2 Groundwater Systems vs. Gas/Electric Furnaces

In the Gulf Coast area heating is generally provided by gas or electric furnaces. These furnaces have a maximum COP of 1, since heat is delivered directly into the room (the actual COP may vary between 0.5 and 1.0). Therefore, when compared with these furnaces the groundwater heat pumps show about a 4 to 1 advantage in energy efficiency. In other words, there is a significant potential for energy savings if gas or electric heating is replaced by groundwater heat pumps.

3.0 HISTORY OF THE DEVELOPMENT OF GROUNDWATER HEAT PUMP SYSTEMS

The literature specific to groundwater heat pump systems is not as extensive as one might think, considering the fact that this is a 30-year-old technology. The bibliography (Appendix 2) presented contains references which range in scope from the highly technical to material from popular journals for the layman. Notably absent is any one compendium of application design data for wide use by builders and architects.

A professor in the Department of Physics at Ohio State University, Dr. Carl Nielsen, is generally recognized as the pioneer in the development of groundwater heat pump systems. He built his first system in 1948. His second system, installed in 1955, is still in operation.

During the 1950's, the use of heat pumps expanded rapidly and, in the commercial, industrial market, so did the use of groundwater as the heat source/sink. Engineers were attracted to these systems because they were a revolutionary and simple means of heating and cooling. Energy costs were of secondary importance.

The early major applications were in areas having an extreme temperature range. To overcome the inherent limitations in the use of air as the heat source/sink, groundwater was a natural choice. Battelle Memorial Institute in Columbus, Ohio, for example, adopted the groundwater heat pump concept in 1957 as an alternate to coal and oil heating. The groundwater there is 13° C (55° F). They currently condition over 300,000 square feet of space with this system.

In spite of the numerous examples of successful commercial operations, the use of these systems in homes was restricted due to lack of general knowledge, underdeveloped technology and the lack of a strong economic incentive. The commercial systems were custom designed and built to utilize groundwater of less than 16° C (60° F). Building a heavy duty residential unit that could use water below 16° C (60° F) involved extra cost that made for an unattractive market.

However, in the lower third of the country, with groundwater temperatures above 16° C (60° F), several manufacturers began offering units for the home. These systems have been rather common in Florida for over 20 years.

Recent developments have provided a strong economic incentive and this, along with the availability of more reliable heat pumps, has significantly increased interest in these systems. A number of articles promoting the use of these systems has appeared in popular journals within the past year and the restraints due to lack of general knowledge will rapidly fade. The problem of higher cost is no longer significant with the "pay-back" period being shortened with each rise in the price of energy.

A tremendous increase in the installation of groundwater heat pumps can be expected throughout the country, but especially in the Gulf Coast areas where the groundwater is not only relatively warm, but also plentiful.

The general application of heat pumps was stifled for a period of time because they were introduced for use without sufficient qualifications. As a result, there was widespread dissatisfaction and they were considered unreliable for a number of years. It has taken a major effort to overcome this stigma. The potential for a similar story is present in the application of heat pumps using groundwater as the heat source/sink. That potential lies in the fact that such systems will require more attention and better maintenance procedures than do conventional systems.

A case history to illustrate this point exists in the Houston area and is well worth relating.

3.1 Case History

A Houston builder needing a sales "spark plug" for a new subdivision project offered a revolutionary new heating and cooling system. This was in 1959 and the system was a groundwater heat pump unit manufactured by the Typhoon Heat Pump Company of Tampa, Florida. This company subsequently stopped making heat pumps and is now a part of the Hupp Corporation.

The builder installed these units in 40 new homes of about 1,300 square feet. Each home had a shallow (less than 80') water well and a discharge well of different depth. Various aquifers were used between 30 and 80 feet. These wells have various lateral separations, some being as close as 15 feet with the average being about 30 feet. These one-inch wells had small pumps delivering four to five gallons of water per minute.

The water was pumped through a heat exchanger and back into the ground.

These units were of three ton capacity and very compact. They were installed in closet-like spaces with return air entering through the louvered closet door. The water/freon heat exchanger and heat pump were in a lower section with the freon/air heat exchanger and air handler above. The entire unit measured about 18" x 18" x 6'. A specification sheet on these units was obtained and is reproduced as Figure 3-1.

Most of the original owners have moved and a majority of these homes are now tenant occupied. However, a survey was made which, though limited, permits confidence in the following observations:

1. Those systems which are still operational (no more than five) are more economical than conventional units. In fact, those homes which have replaced the groundwater system have heating and cooling costs between 50-70% higher than similar homes with the original system.
2. The primary reason for replacing the original system was continuing difficulties in obtaining repairs. Apparently, the air conditioning servicemen, being unfamiliar with the well system and the necessary controls, would recommend replacement when there was a problem beyond their normal experience. The water well servicemen were also not knowledgeable from a

system standpoint. After the initial installation and checkout, there was no overall knowledgeable source for maintenance and repair.

3. The major cause of failure in the systems was most likely the reversing valve in the heat pump. Whether this was due to inherent design or caused by water fluctations, corrosion-restricted heat exchangers or some other factor could not be determined.
4. Other failures were rather uncomplicated and infrequent. These were worn out water pumps, fan motors, and switches.
5. Many of the wells are still in use for watering lawns, washing cars, and other purposes even though they are no longer used in the HVAC system. This strongly indicates that well sufficiency was not a problem.
6. THOSE SYSTEMS WHICH WERE DILIGENTLY MAINTAINED BY THE OWNER HAD FEW PROBLEMS AND WERE LONG-LASTING.

3.2 Availability of Water Source Heat Pumps

While the concept of water source heat pumps is not new, we were able to identify only a handful of companies that sell them as standard systems. The standard sizes that are manufactured are also limited. The 50 ton size seems to be the upper limit for single packaged systems. Larger capacities can be designed, but involve custom sizing of each individual component, e.g., the compressor, heat exchangers, etc. Table 3-1 gives a partial list of manufacturers of small to medium sized packaged water source heat pumps.

TYPHOON PROP-R-TEMP PACKAGED HEAT PUMPS

MODEL 32AH

COOLING CAPACITIES

ENTERING W.B. TEMP.	ENTERING AIR D. B. TEMPERATURE							
	75°		80°		85°		90°	
	BTU/HR	SHR	BTU/HR	SHR	BTU/HR	SHR	BTU/HR	SHR
62	25,700	.78						
64	27,000	.68	27,000	.80				
66	28,300	.59	28,300	.71				
67	29,000	.55	*29,000	.67	29,000	.76		
68			29,700	.62	29,700	.73		
70			31,000	.55	31,000	.65	31,000	.75
72					32,600	.57	32,600	.66
74					33,500	.51	33,500	.59
76					35,000	.46	35,000	.54

NOTES:

1. SHR - Sensible Heat Ratio
2. Conditions applied to the above data
   - A. Capacities representing total BTU/HR output
   - B. Condensing Temp. 105°F
   - C. Air quantity 900 CFM

CONDENSING TEMPERATURE MULTIPLIER

Cooling capacities table based on condens. temperature of 105°. To obtain capacities for other conditions multiply above capacities by the following factor:

CONDENSING TEMPERATURE	TOTAL CAPACITY MULTIPLIER
95°	1.065
100°	1.035
* 105°	1.000
110°	.960
115°	.930
120°	.900

AIR QUANTITY MULTIPLIER

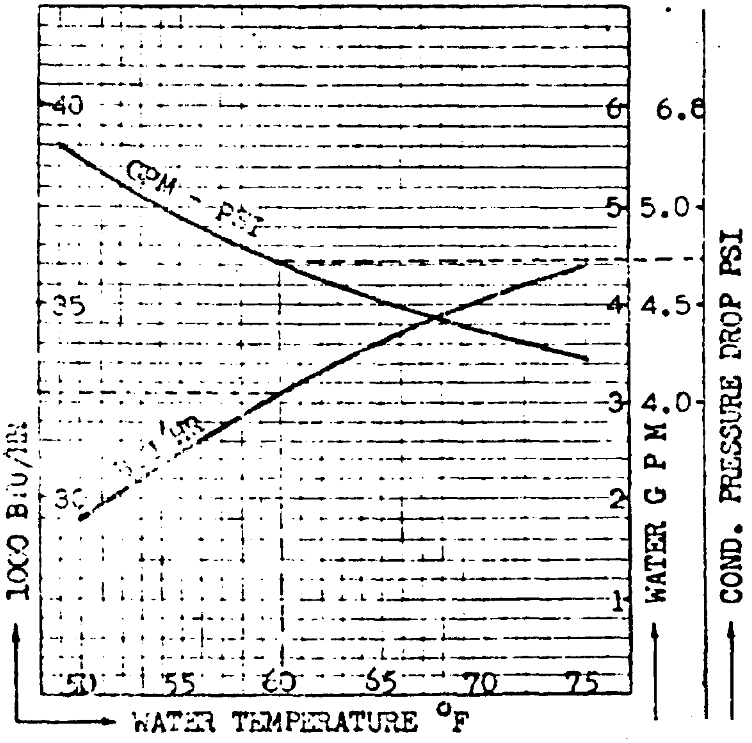
Above rating tables based on rated air quantity. To obtain capacities at other air flow rates, multiply above capacities by the following factors:

% OF RATED AIR QUANTITY	MULTIPLIER
80%	.94
90%	.98
* 100%	1.00
110%	1.03
120%	1.05
125%	1.06

* Standard Conditions

HEATING CAPACITIES (Based on 75° Water)

ENTERING D.B. TEMP.	BTU/HR	LEAVING D.B. TEMP.	POWER CONSUMPTI KW
80°	34,100	115	3.40
75°	35,100	111	3.25
* 70°	36,000	107	3.10
65°	36,900	103.0	3.00
60°	37,900	99	2.94



Example @ 60° Water

Heating Capacity- 32,700 BTU/H
GPM Required - 4.45
Cond. PSI - 4.75

Table 3-1

Manufacturers of Water Source Heat Pumps and Standard Sizes (Tons of Cooling)

1. Vanguard Energy Systems
   9133 Chesapeake Drive
   San Diego, CA 92123
   713/292-1433

   Sizes: 1½, 2, 3, 4,
   5, 6

2. Solargy Systems
   Division of Wescorp, Inc.
   15 Stevens Street
   Andover, MA 01810
   617/470-0520

   Sizes: 1, 1½, 2, 2½, 3,
   3½, 4, 5, 6, 7

3. Comfort Aire
   Heat Controller Inc.
   Jackson, MI 49203
   517/787-2100

   Sizes: 3, 3½, 4, 5, 8, 10,
   15, 20, 25, 30

4. Carrier
   Carrier Parkway
   Syracuse, NY 12301

   Sizes (1000 Btu/H): 14, 18
   22, 27, 33, 42

5. American Solar King Corp.
   6801 New McGregor Hwy.
   Waco, TX 76710
   817/776-3860

   Sizes: 3, 5, 10, 15, 20,
   25, 35, 45

6. Florida Heat Pump Co.
   610 SW 12th Avenue
   Pompano Beach, FL 33060
   305/781-0830

   Sizes: (1000 Btu/H): 10, 14,
   20, 27, 34, 42, 52, 62, 84,
   100, 120, 200, 240

(This is only a partial list representing manufacturers contacted and from whom we have received literature.)

Each of the above manufacturers was also contacted regarding the prices of its heat pump systems. It quickly became evident that no standardization, as far as prices are concerned, has been achieved to date. Table 3-2 lists the prices of some standard sizes as obtained from the manufacturers. All prices are exclusive of installation. The prices are plotted in Figure 3-2 to illustrate the wide variation mentioned above. Interestingly enough, however, all the systems apparently have some slope in the cost curve.

Table 3-2

List Prices* of Standard Water Source Heat Pumps

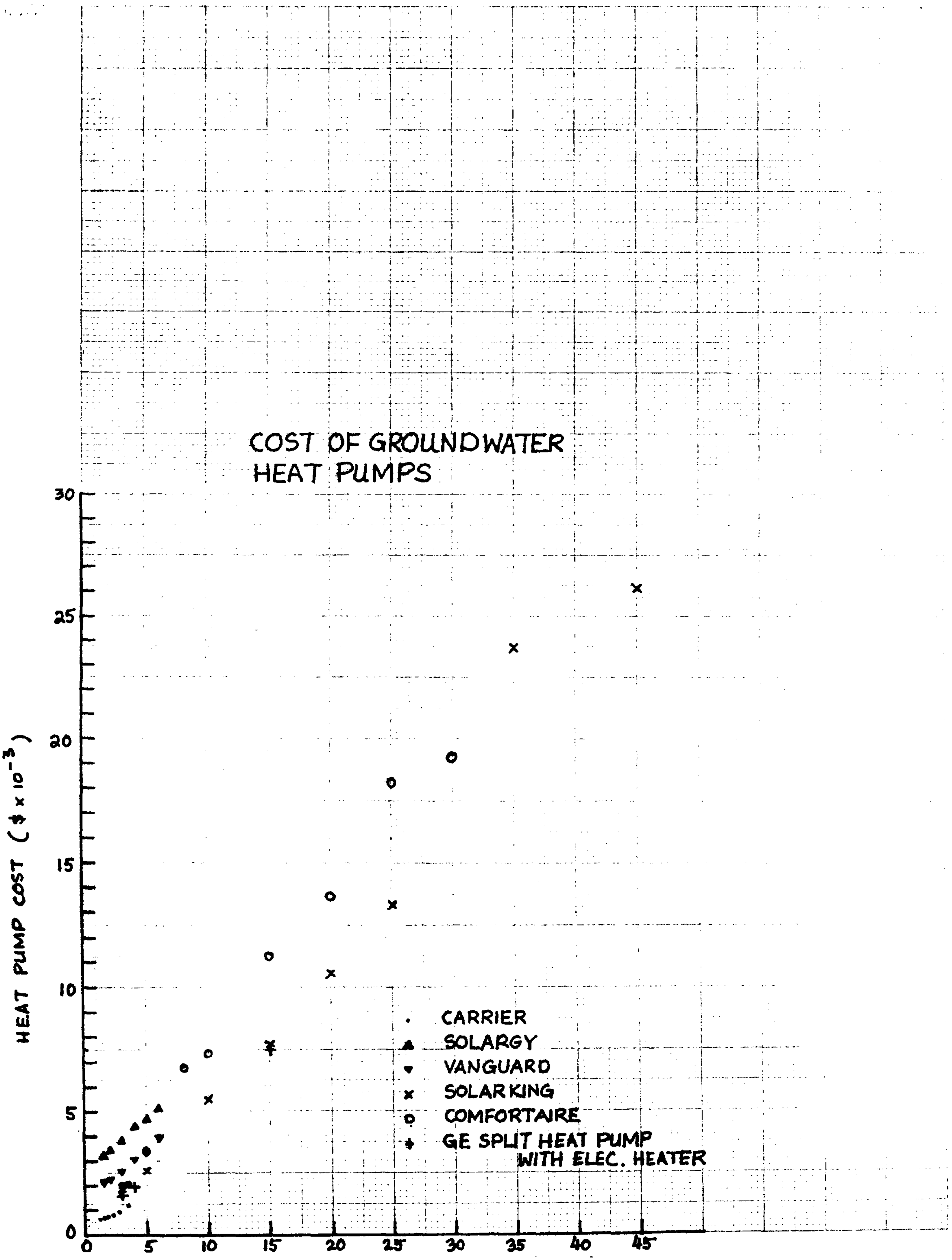
Size**	Solargy	Vanguard	Solar King	Comfort Aire	Florida HPC	Carrier
3	4500	2670	1860	1965	1465	1710
3½	--	--	--	2067	1724	2040
5	5600	3318	2606	3342	2741	--
10	--	--	5454	7302	5444	--
15	--	--	7702	11281	--	--
20	--	--	10650	13687	10660	--

*All prices in 1978 dollars, exclusive of installation and distributor discounts.

**Sizes are in tons (12,000 Btu) of cooling.

-- not a standard size

Fig. 3-2


COST OF GROUNDWATER HEAT PUMPS
HEAT PUMP COST ($ x 10^-3)
0
5
10
15
20
25
30
0
5
10
15
20
25
30
35
40
45
• CARRIER
▲ SOLARGY
▼ VANGUARD
× SOLAR KING
o COMFORTAIRE
+ GE SPLIT HEAT PUMP WITH ELEC. HEATER

4.0 CONCEPTUAL DESIGN OF GROUNDWATER HVAC SYSTEMS

In order to make a realistic evaluation of the potential for groundwater heat pumps, four case studies were performed. The case studies involved actual, planned (or in construction) buildings with each building being different in function and floor space. Heating and cooling load estimates were made where applicable and a conceptual groundwater heat pump system was designed for each case. Where no HVAC presently exists, a reasonable ducting scheme is also indicated. If a HVAC system exists (or has been designed) we have changed only the cooling and heating generation equipments specified by groundwater based equipments consistent with the design data. The equipment and energy cost differentials between the existing (or conventional) systems and the groundwater systems were also calculated for each case, though the equipment cost comparisons were limited to the main items, e.g., compressor, boiler, etc. In all cases it has been assumed that the heating and cooling distribution equipments are (or can be) the same. All prices and cost data used are either manufacturer quoted or estimated from Khashab (4).

4.1 Calculation Procedure for Heating and Cooling Loads

Three out of the four case studies involved a determination of the design heating and cooling loads since none were available. To retain some degree of realism, a somewhat detailed load calculation procedure was used for these cases. Standard ASHRAE techniques, e.g. as in 1977 ASHRAE Fundamentals (5), were used for the calculations.

In brief, the technique involves calculating the heat lost or gained through the building walls from the equation

$$q = U.\ A.\ \Delta T \tag{8}$$

where

q = Btu/H lost or gained

A = area of wall (or window, roof, ceiling)

ΔT = design temperature difference between indoor and outdoor temperature

U = overall heat transfer coefficient.

Procedures for calculating U are given (5). Figure 4-1 shows a sample load calculation sheet. Appendix 3 contains a summary of the load estimates and also the completed load calculation sheets. The calculated loads were then used to specify the sizes of the necessary equipments.

4.2 Sizing and Design

Following the load calculations, conventional and groundwater systems were sized to handle the design heating and cooling load and the supply air CFM requirements. The conceptual designs have considered the practicability of installing a single large unit as against a number of small units and the problem of zone control. We have also tried to examine the effect of using part air and part groundwater instead of a total groundwater system. In the larger systems, e.g., the lab/research building and high school building, we have also attempted to include a heat conservation scheme wherein if heating and cooling are required at the same time in different zones, it is done simply by transferring the required heats between the zones. These aspects are described further in the following design summaries.

Fig. 4-1

HEAT GAIN CALCULATION

Name ________________ **Room Used For** ________________ **Job #** ________
City ________________ **Room Number** ________________ **Sheet #** ________
Calculated By: ________________ **Date:** ________________ **Zone #** ________

DESIGN CONDITIONS

Outdoor DB Temp ______ °F **WB Temp** ______ °F **Abs. Hum.** ______ Gr/lb **Cal For** ______ AM ______ PM
Indoor DB Temp ______ °F **WB Temp** ______ °F **Abs. Hum.** ______ Gr/lb. **Room Volume** ________
Temp. Diff. ______ °F **Hum. Diff.** ______ Gr/lb

COL. I	II	III	IV	V	VI	VII	VIII	IX	X
SIDE FACING	WALL/PART. DIM. WINDOW & DOOR NO. & DIM. OR AREA	GROSS WALL AREA SQ/FT TOTAL GLASS AREA SQ/FT NET WALL AREA SQ/FT	$\frac{U_G}{U_W}$	$\frac{D_B}{T_D}$	CONDUC. FR. TEMP. DIFF. BTU/H	FACTOR $\frac{R_G}{R_W}$	SUN EFFECT BTU/H INCLUDING SHADING	FACTOR	HEAT LOSS BTU/H
CLG.									
FLR.									
			TOTAL						HEAT LOSS

TOTAL CONDUCTION: BTU/H . . . . ________
TOTAL SOLAR: . . . . ________
PEOPLE: NO. OF PEOPLE IN RM. (_____) X 220 . . . . ________
ELECTRICAL APPLIANCES: WATTS (_____) X 3.4 . . . . ________
OTHER SOURCES: BTU/H . . . . ________

TOTAL INTERNAL SENSIBLE HEAT GAIN . . . . * ________

VENTILATION AIR CFM. (_____) X TEMP. DIFF. (_____) X 1.08 . . . . ________

TOTAL SENSIBLE HEAT GAIN . . . . H_S ________

LATENT HEAT GAIN

PEOPLE: NO. IN RM. (____) X (____) BTU/H . . . . ________
OTHER SOURCES: LB. OF WATER PER/HR. (____) X 1060 . . . . ________
VENTILATION AIR: CFM (____) X HUM. DIFF. (____) X 0.66 . . . . ________
CFM SUPPLY AIR: ________ *

TOTAL LATENT GAIN H_L ________
TOTAL HEAT GAIN H_T ________
TONS ________

4.2.1 Residential Home

Figure 4-2 illustrates the house plan. The most energy conservative design for this case is achieved by treating the bedroom and the living room/kitchen areas as separate zones and requiring two separate A/C units to handle the individual loads. This design therefore allows one zone to be completely shut down if not in use, e.g., during the night, the living room/kitchen zone can remain without any air conditioning.

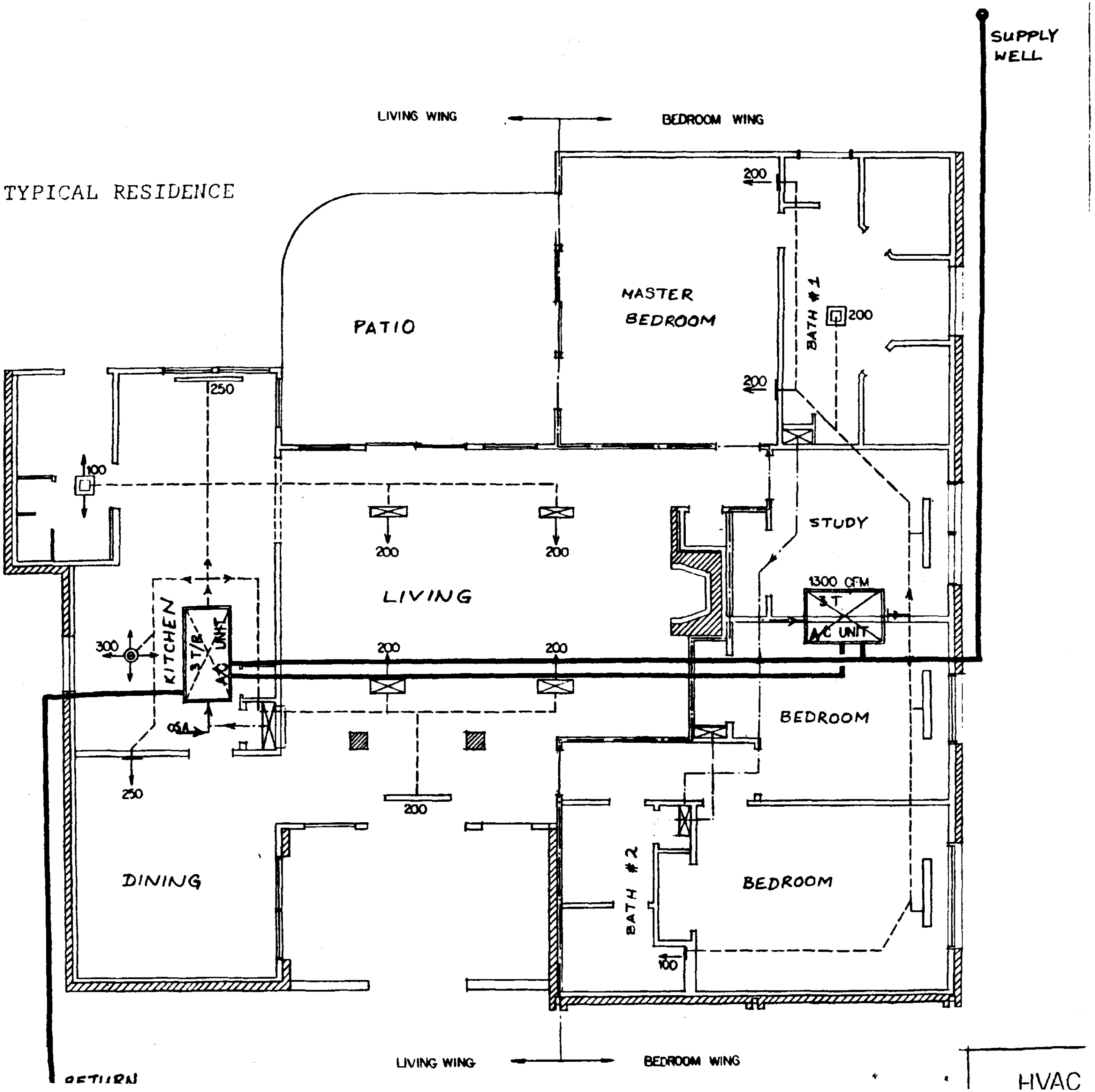
In the conventional case, this system will have two air-to-air heat pumps with the air handling units being placed in the attic and the condensing units (including the compressors) outside the house. Each heat pump will be equipped with resistance backup heating strips to handle extreme cold or "frosty" weather.

In the groundwater system, the air-to-air heat pumps will be replaced with water-to-air heat pumps of the same capacity. The groundwater will need to be piped up to the heat pumps and split up as shown. The inclusion of a surge tank at the inlet to the system may be beneficial to prevent short cycling of the water pumps, though experience with existing groundwater installations shows it to be unnecessary.

4.2.2 Lab/Research Building

The space in this building is conveniently split into three zones with separate air handlers for each zone. A four-pipe chilled and hot water circuit is used to provide heating or cooling whenever required. Figure 4-3 shows the placement of

TYPICAL RESIDENCE



HVAC

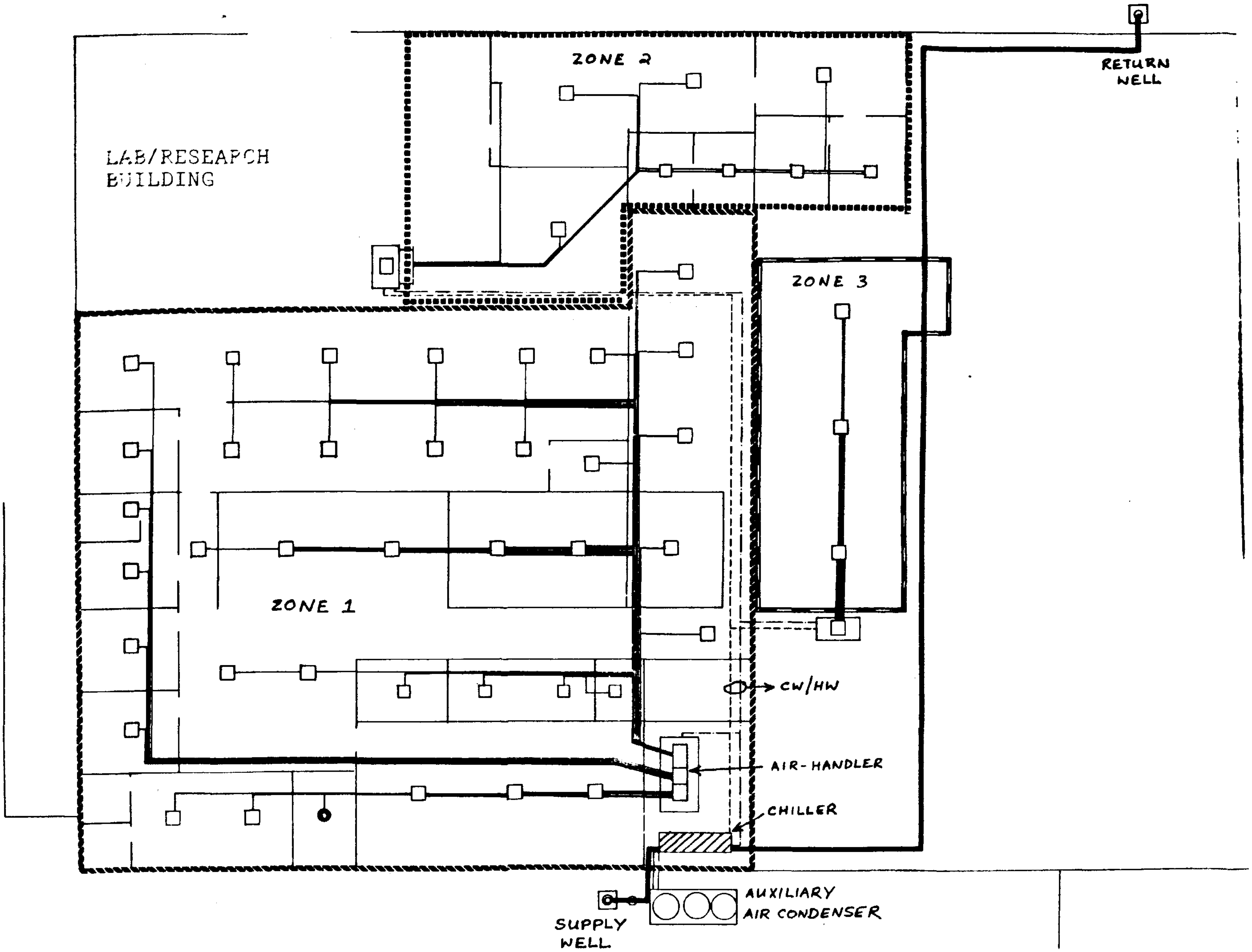

ZONE 2
RETURN WELL
LAB/RESEARCH BUILDING
ZONE 3
ZONE 1
CW/HW
AIR-HANDLER
CHILLER
AUXILIARY AIR CONDENSER
SUPPLY WELL

Fig. 4-3

the HVAC units and the zone separations. These placements remain the same irrespective of the system (i.e., air or groundwater).

The conventional HVAC design utilizes a packaged air-cooled water chiller and an electric furnace, which are standard off-the-shelf items. In this design the energy conservation feature alluded to in Section 4.2 can be easily implemented by routing the hot water circuit through an auxiliary heat exchanger which absorbs some of the heat of condensation of the refrigerant used in the water chiller. This can not only reduce the electric power requirement of the furnace, but also reduce the power to the air-cooled condenser.

The groundwater system design is illustrated in Figure 4-4. This design has the feature that instead of reversing refrigerant flow, the groundwater is routed either to the condenser or the evaporator, depending on the load requirements. For example, during the heating season, the groundwater will be piped through the evaporators (or chillers) to evaporate the refrigerant. The refrigerant vapors would then be compressed and thereby heated to a higher temperature. The hot vapors then give up sensible and latent heat to the hot water circuit. The energy conservation feature is easily implemented in this design since any heat picked up by the chilled water circuit (in the process of cooling) can be transferred, with a 3 or 4 fold amplification, to the hot water circuit. Thus, for each unit of energy conserved, or transferred between different zones in the building, an equivalent saving in the well pump power is also achieved.

Fig. 4-4

CONCEPTUAL SPLIT WATER/AIR HVAC SYSTEM (LAB/RESEARCH BLDG.)

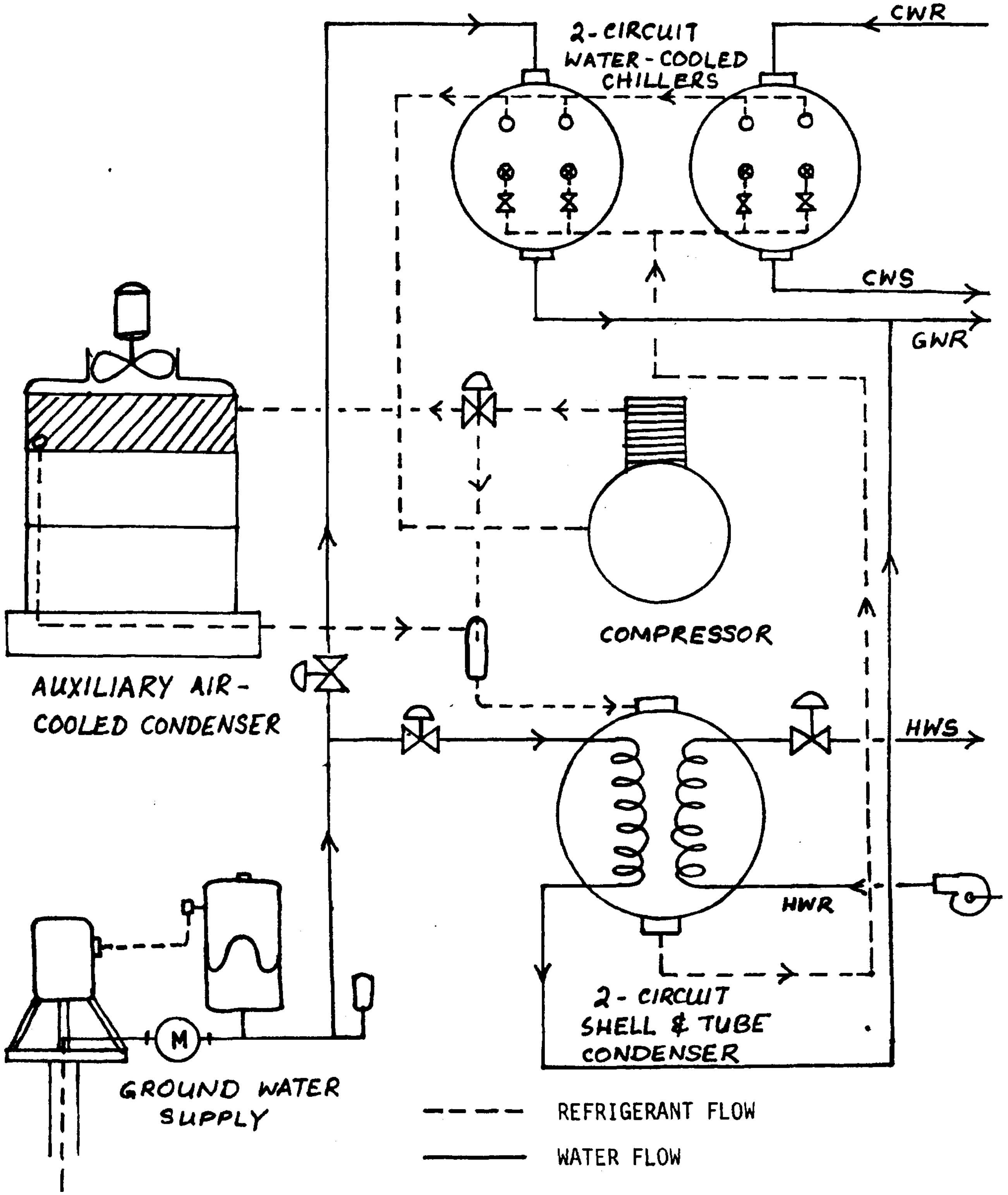


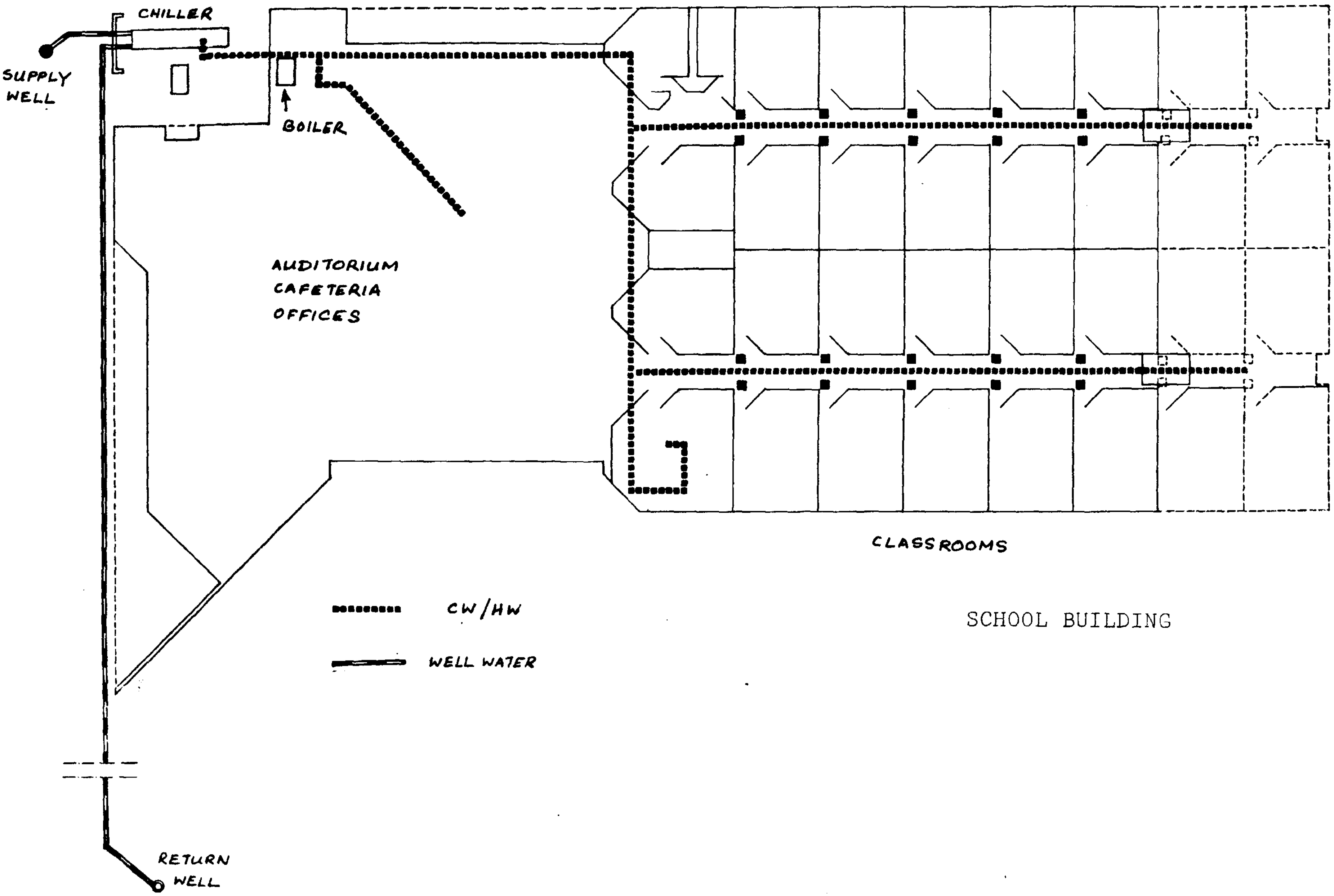
Figure 4-4 also shows an auxiliary air-cooled condenser. While the practicality and economics of this modification need to be demonstrated by more extensive modelling and experimentation, conceptually, some savings in the power to the well pump can be realized by using air, instead of groundwater, in the temperature range of 18° - 27° C (65° - 80° F). In this temperature range, the COPs for the two systems are about the same (see Table 2-3). Also, in this temperature range, the heating (or the cooling) load is minimal. Thus, at those sites where groundwater is not available at shallow depths (e.g., 100 feet), it may be more efficient to use the air circuit rather than the water circuit.

4.2.3 School Building

A conventional HVAC system for this building has already been designed by the consulting engineers on the project (see Figure 4-5). The designed system consists of a 126 ton packaged chiller and a 180 kW electric boiler providing hot and chilled water to a two-pipe system.

In the groundwater modified design we have replaced the chiller and boiler by a custom designed heat pump similar in concept to the one designed for the lab/research building. Thus, the custom unit consists of two heat exchangers for chilling, a compressor and a condenser.

A second iteration on this design was also considered which involved the downsizing of the groundwater heat pump to 70 tons and including a 60 ton packaged air cooled chiller in



SCHOOL BUILDING

Fig. 4-5

an attempt to reduce the power requirements of the well pumps. The difference in equipment cost between the only groundwater case and part air - part groundwater case is small (see Appendix 4). Thus, a more vigorous energy analysis will be necessary to determine the optimum system. However, only the groundwater case is used in this study to determine energy savings and payback periods shown in Table 4-3.

A further modification to the present system that will allow greater flexibility in operation is the replacement of the two-pipe system with a four-pipe system. This four-pipe system allows separate areas to demand heating and cooling simultaneously which cannot be met with the two-pipe system. However, since this modification is not a necessary feature of groundwater heat pumps, no cost and energy analysis was performed.

4.2.4 Office-Manufacturing Facility

We have designed the HVAC system only for the office area (see Figure 4-6) since it was felt that the warehouse and workshop areas cannot be profitably air conditioned and that no particular advantage will be gained by using groundwater in any other way.

For this case, the conventional design is represented by a roof top 16 ton air cooled condensing unit combined with a two-zone air handler for cooling and in-line gas fired duct heaters for heating.

For the groundwater case, two 8 ton water source heat pumps have been specified and the approximate locations of the units and the wells are shown in Figure 4-6.

OFFICE-MANUFACTURING FACILITY

APPROXIMATE LOCATIONS OF HVAC UNITS

1. Roof-top air-to-air heat pump (conventional case)
2. Two-zone air handler for air distribution (conventional case)
3. Air distribution ducting to 2nd. fl. offices
4. Air distribution ducting to 1st. fl. offices
5. Return air intakes (conventional case) and approximate location of water-to-air heat pumps (ground-water case)
6. Supply line from supply well (ground-water case)
7. Return line to reinjection well (ground-water case)

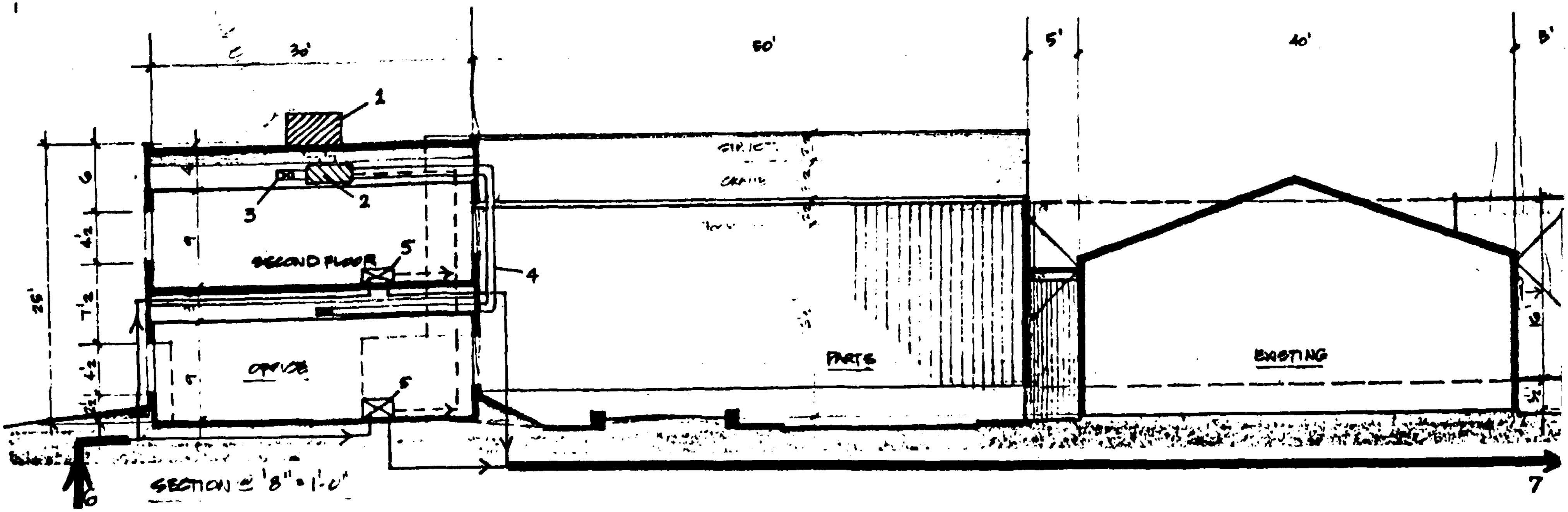
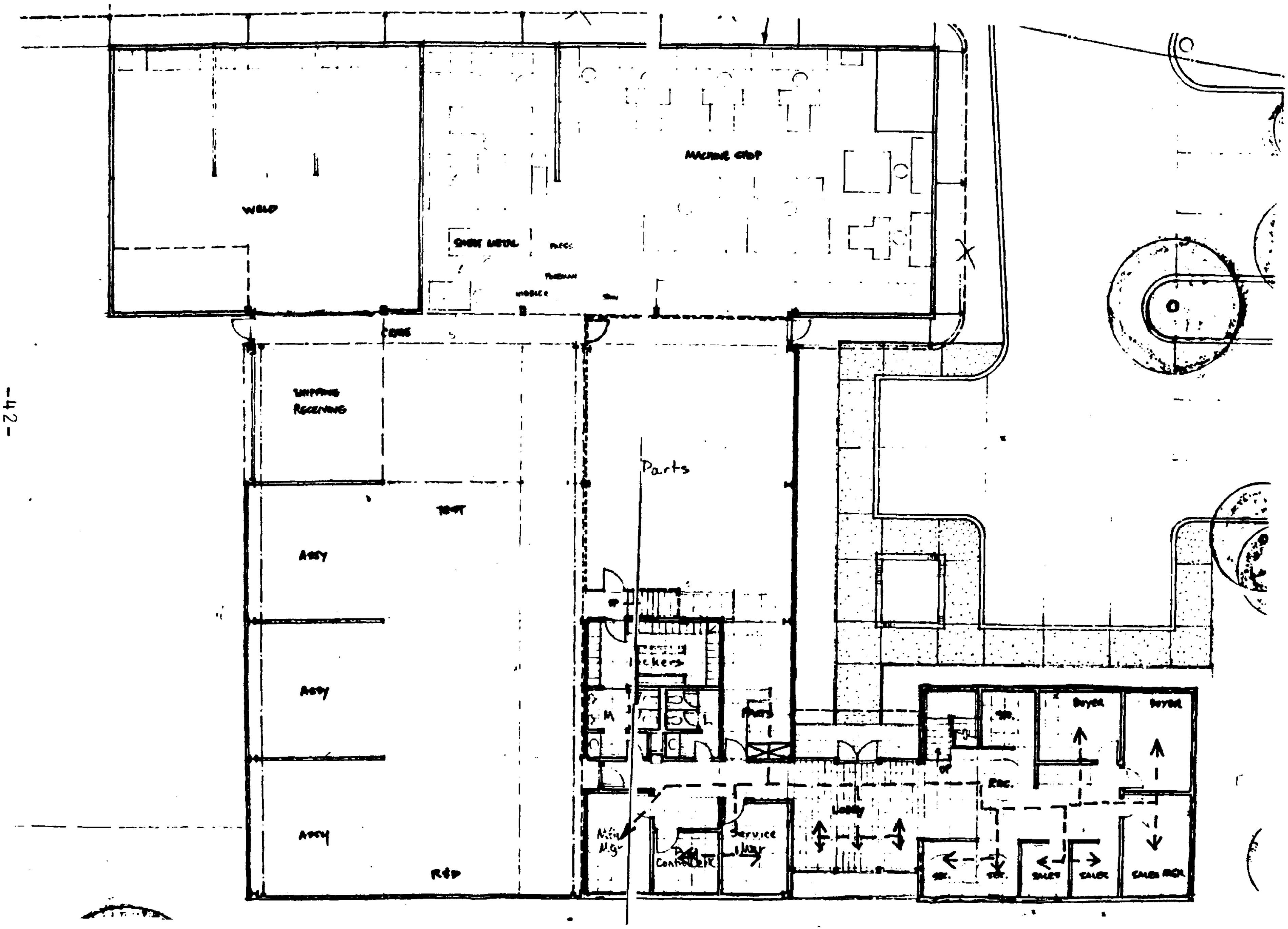


Fig. 4-6


MACHINE SHOP
WELD
SHEET METAL
PARTS
FOREMAN
MOBILE
CRIB
SHIPPING
RECEIVING
Parts
TEST
ASSY
ASSY
ASSY
R&D
UP
Lockers
M
L
Parts
UP
STOR.
BUYER
BUYER
REC.
LOBBY
Mfg Mgr
Prod Control Clerk
Service Mgr
SEC.
STOR.
SALES
SALES
SALES MGR

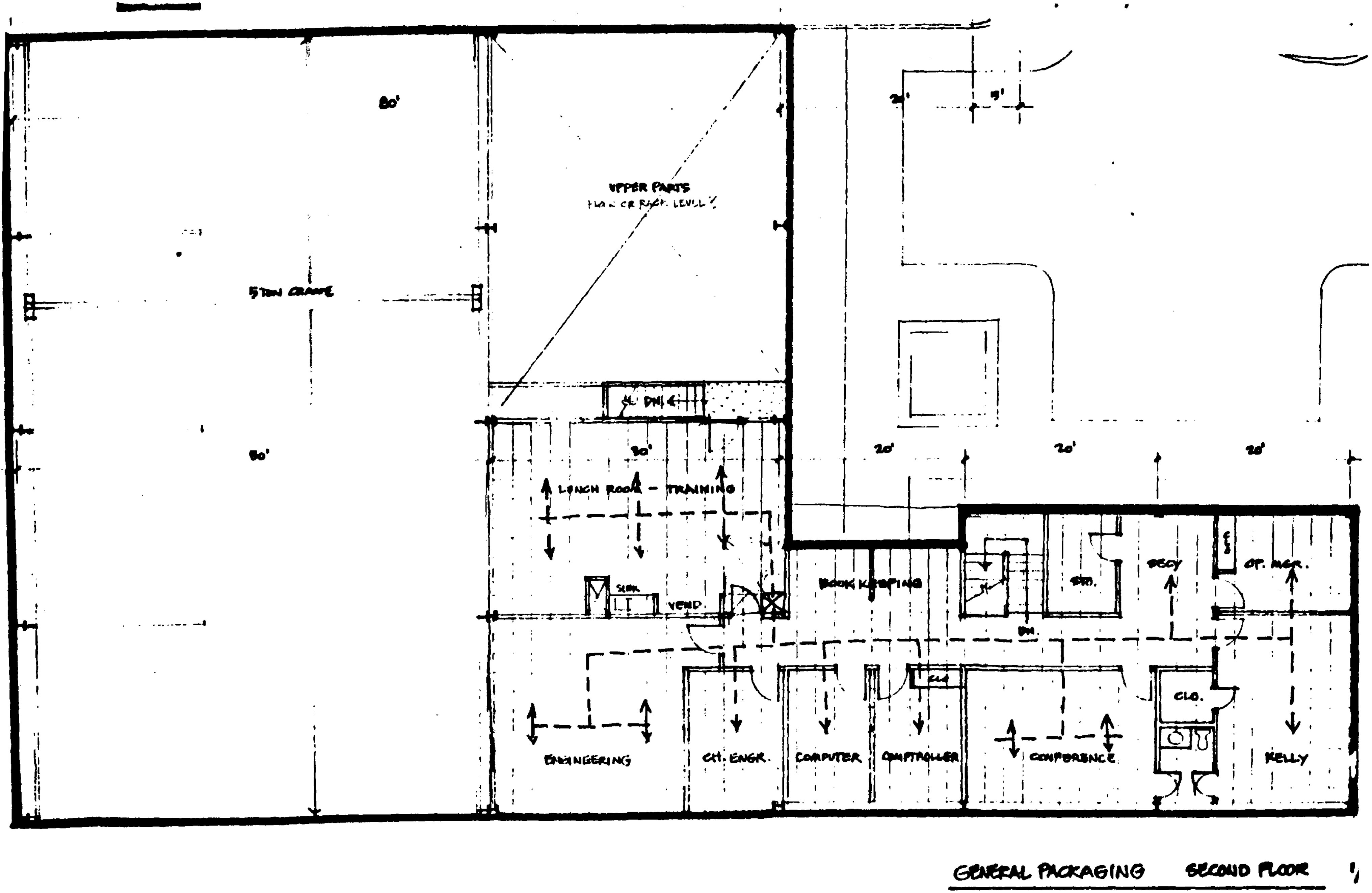

80'
UPPER PARTS
FLOOR OR RACK LEVEL ?
20'
5'
5 TON CRANE
UP DN
80'
30'
20'
20'
20'
LUNCH ROOM - TRAINING
SINK
VEND.
BOOKKEEPING
STR.
SECY
OP. MGR.
DN.
CLO.
ENGINEERING
CH. ENGR.
COMPUTER
COMPTROLLER
CONFERENCE
KELLY
GENERAL PACKAGING SECOND FLOOR

Fig. 4-6 cont'd

4.3 Economics

4.3.1 Seasonal Energy Requirements

The previous section detailed the analysis of the peak heating and cooling loads to be used for equipment sizing. However, on a seasonal basis, the loads will vary and the cumulative seasonal loads will be different from the peak loads. The net energy costs are proportional to the seasonal (rather than peak) loads and therefore an estimate of the seasonal energy requirements is necessary.

The seasonal requirement for heating is generally calculated using the "degree-day" method (4). For a 75° F indoor design temperature, the degree day formula gives

$$H = 24h\ (10d + D)/\Delta T \qquad (9)$$

where

H = seasonal heat loss, Btu

h = design heat loss, Btu/H

d = number of days in heating season

D = number of degree days in heating season

ΔT = design difference for heating between inside and outside air temperatures

For each building analyzed, h is the heat loss calculated in Section 4.1. Reasonable assumptions for d and D for Houston are

d = 180 days

D = 1,300 degree-days

and ΔT = 50° F (design value)

Thus,

$$H = 1,488 \times h, \text{ Btu} \tag{10}$$

A similar method does not exist for the seasonal cooling load. However, a reasonable estimate can be obtained by assuming that the seasonal load is the equivalent of 2,000 hours of peak cooling load; i.e.,

$$C = 2,000 \times c, \text{ Btu} \tag{11}$$

where

c = peak cooling load, Btu/H

Equations 10 and 11 can now be used to determine the energy cost of heating and cooling. Thus, for cooling,

$$\text{E.C.} = \{2,000 \times c/(\text{EER} \times 10^3)\} \times \$/\text{kWh} \tag{12}$$

where

E.C. = energy cost in dollars

EER = Energy Efficiency Ratio

= Cooling Capacity (Btu)/Electric Energy Input (watts)

\$/kWh = cost of electric energy

Thus,

$$\text{E.C.} = 2 \times c \times (\$/\text{kWh})/\text{EER} \tag{13}$$

For heating by a heat pump, we have

$$\text{E.C.} = \{1,488 \times h/(\text{COP} \times 3,412)\} \times \$/\text{kWh}$$

$$= 0.436 \times h \times (\$/\text{kWh})/\text{COP} \tag{14}$$

Equation 14 can also be used for electric heating if the proper COP is substituted into the equation.

For gas heating, we have

$$E.C. = \{1,488 \times h/(HV \times COP \times 10^3)\} \times \$/1,000 \text{ cu. ft.}$$

where

$$HV = \text{heating value of gas} = 1,000 \text{ Btu/cu.ft., for natural gas}$$

or

$$E.C. = 1.488 \times 10^{-3} \times h \times (\$/1,000 \text{ cu.ft.})/COP \qquad (15)$$

It is seen that each formula requires the substitution of an appropriate EER or COP and the corresponding cost of energy. Tables 4-1a and 1b list the efficiencies assumed for these calculations and the cost of primary energy.

Table 4-1a

Assumed Efficiencies of Different Heating and Cooling Methods

	Cooling EER	Heating COP
Air-to-air heat pump (with electric backup)	8	2.5
Groundwater heat pump	11	4.0
Air conditioner	8	
Gas furnace		0.6
Electric furnace		1.0

Table 4-1b

Assumed Utility Rates

Electricity	\$0.04/kWh
Gas	\$3.50/1,000 cu.ft.

4.3.2 Equipment Costs

The costs of both the conventional system and the groundwater system were determined either from direct manufacturer's quotes on specified equipments or estimated using cost curves in (4) (including a 21 percent inflation factor in refrigeration equipment costs between 1976 and 1978). The cost data are presented in Appendix 4 where the cost of each system is determined on a separate data sheet.

4.3.3 Cost and Energy Differentials Obtained with Groundwater Heat Pumps

The equipment and yearly energy cost differentials are summarized in Table 4-3 where the simple payback time (or the ratio of the equipment cost difference to the yearly operating cost difference) is also estimated. Two things are immediately clear on reviewing Tables 4-2 and 4-3:

1. Net energy savings are a minimum of 30 percent and may be as high as 50 percent on an annualized basis, and
2. Payback periods are shorter (therefore groundwater heat pumps are more attractive) when electrical resistance heating is used in the conventional case.

In any case, the payback period is only six years for the worst case. If tax rebates are considered, the payback period may be as short as three years. Thus, we can say unequivocally that groundwater heat pumps provide the most effective means for the heating and cooling of any type of building in the Texas Gulf Coast Region.

Table 4-2

Heating and Cooling[*] Costs for the Four Case Studies

	Case Study No. 1 (residential house)		Case Study No. 2 (lab/research)		Case Study No. 3 (school building)		Case Study No. 4 (Office/whse.)	
	Cool'g	Heat'g	Cool'g	Heat'g	Cool'g	Heat'g	Cool'g	Heat'g
1. Conventional	760	349	2,760	2,721	15,000	10,708	1,800	1,042
2. Groundwater	553	218	2,008	747	10,909	2,677	1,440	523
3. Seasonal savings (%)	27	38	27	73	27	75	20	50
4. Yearly savings ($)	338		2,726		12,122		879	
(%)	30		50		47		31	

---

* All costs are in 1978 dollars.

Table 4-3

Equipment and Operating Cost Differentials

	Residential House	Lab/Research Bldg.	School Bldg.	Office/whs
1. Equipment Cost Difference	+ $2,255	+ $7,975	+ $16,070	+ $5,546
2. Energy Cost Difference Per Year	- 338	- 2,726	- 12,122	- 879
3. Payback Period (years)	6.7	3	1.3	6.3

4.3.4 Effect of Well Depth on Pumping Power

The COP of groundwater heat pumps assumed in Section 4.3 (see Table 4-1a) took into consideration that the pumping power would constitute an average of eight percent of the total power consumed by the heat pump. The actual fraction is a site specific quantity since it will vary (linearly) with the level at which adequate groundwater is available. Table 4-4 illustrates this variation for the four case studies.

Table 4-4

Variation of Pumping Power with Well Depth

Case No.	Cooling Load (tons)	Well Depth (feet)	Pump Power (%)
1	6.3	100	7.7
(home)		50	3.9
2	23	100	7.7
(lab)		50	3.9
3	125	300	23
(school)		200	15
		100	8
4	16	100	7.4
(office/plant)		50	3.7

It is observed that at 100 feet depth, pumping power constitutes only 7.7 percent of the total power consumption. For most applications up to 25 tons, adequate water can be found in the Gulf Coast at 100 feet or less. Larger systems could use multiple supply wells located at shallow levels instead of one deep well. In extremely large systems, a single well most likely will not be able to supply all the water. Thus, multiple wells will be a necessity and they can again be located at more shallow levels. For example, the Battelle Memorial Institute's sytem (at Columbus, Ohio) uses several 50-foot wells and one 400-foot well. Thus, even though the single deep well consumes a lot of energy, the combined system uses only a small fraction of the total energy, 12 percent.

The decision between a single deep well or several more shallow wells, however, should be decided on the basis of a cost optimization analysis minimizing the net payback period. As an indication of the worst case possibility, we recalculated the payback period for the school (Table 4-3) assuming that the pumping power would constitute 23 percent of the total consumption. This increased the payback period to a little over three years. The payback periods for the other three case studies increased only marginally (one year or less) for reasonable variations of their pumping power. Thus, as a general rule one can assume that pumping power will constitute between seven and eight percent in most cases and that increases in payback periods are only marginal if only deep wells are available.

4.4 Well Placement and Temperature Migration

An important aspect in the design of groundwater heat pump systems is the placement of the supply and injections wells. An important advantage that can be utilized by the groundwater systems is the possibility of using the warm water discharged during the cooling cycle as supply water during the heating cycle to further augment the COPs. Similarly, if an improvement in the cooling cycle performance is desired then the cooled water from the heating cycle can be used during the cooling cycle. However, in order to effectively use the above advantage, well placements should be such that the discharged water does not migrate back to the supply well before the change of cycles.

There are basically two ways to ensure that the warmed (or cooled) water that is reinjected is not drawn back in during the same cycle. The first, and the most obvious, way is to use two different aquifers at different depths (Figure 4-7). Alternatively, the same aquifer can be used, with the wells spaced distant enough apart to prevent temperature migration back from injection to the supply well (Figure 4-8). There are obvious cost advantages in using the same aquifer and therefore it is necessary to estimate the minimum spacing necessary to avoid temperature migration.

4.4.1 Modelling of Temperature Migration

The minimum distance between wells is actually a site specific quantity since it depends on aquifer properties and dimensions. However, the logic for determining the distance can be developed by assuming an aquifer as shown in Figure 4-9. For this analysis it is assumed that the reinjected water occupies a cylindrical volume centered around the well axis. A thermal energy balance can be written for an elemental aquifer volume using established mathematical techniques for transient heat transfer phenomena (6).

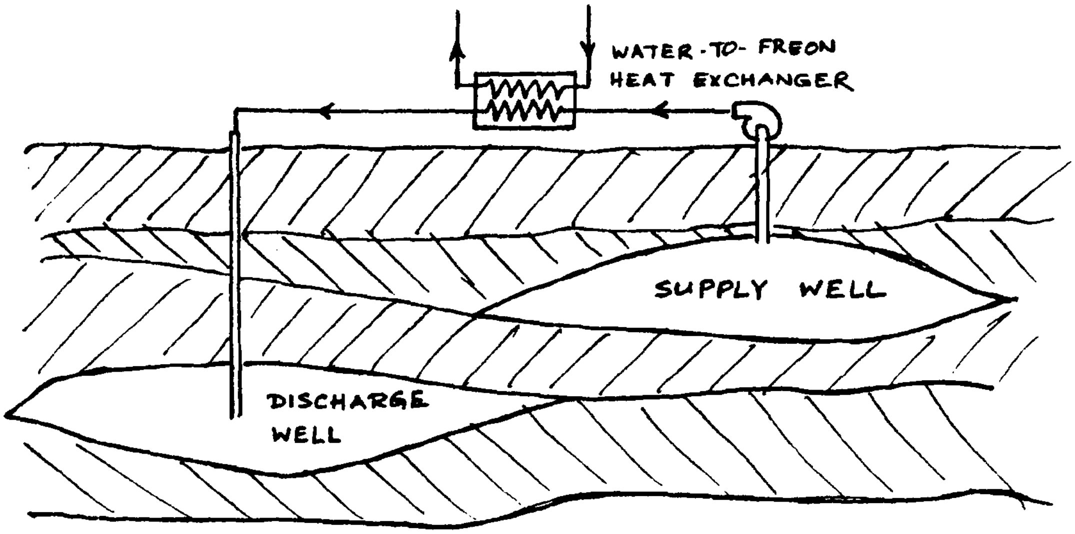

WATER-TO-FREON
HEAT EXCHANGER
SUPPLY WELL
DISCHARGE
WELL

Fig. 4-7

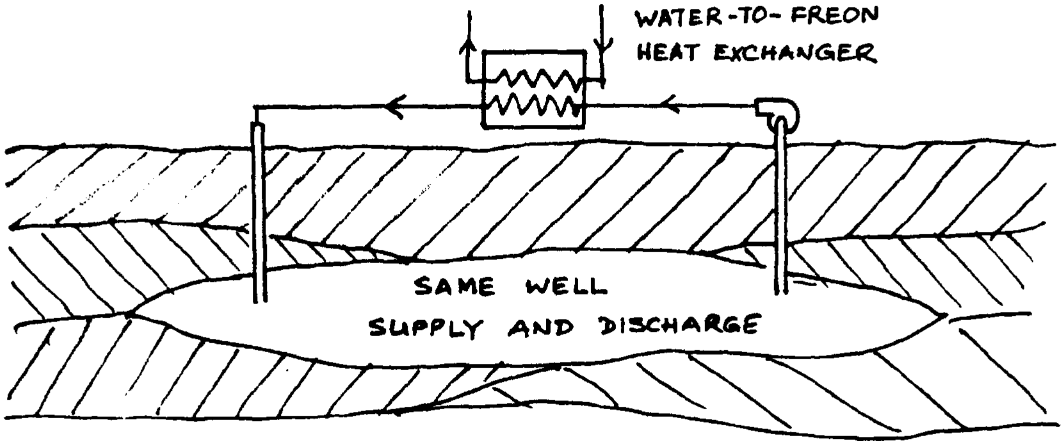

WATER-TO-FREON
HEAT EXCHANGER
SAME WELL
SUPPLY AND DISCHARGE

Fig. 4-8

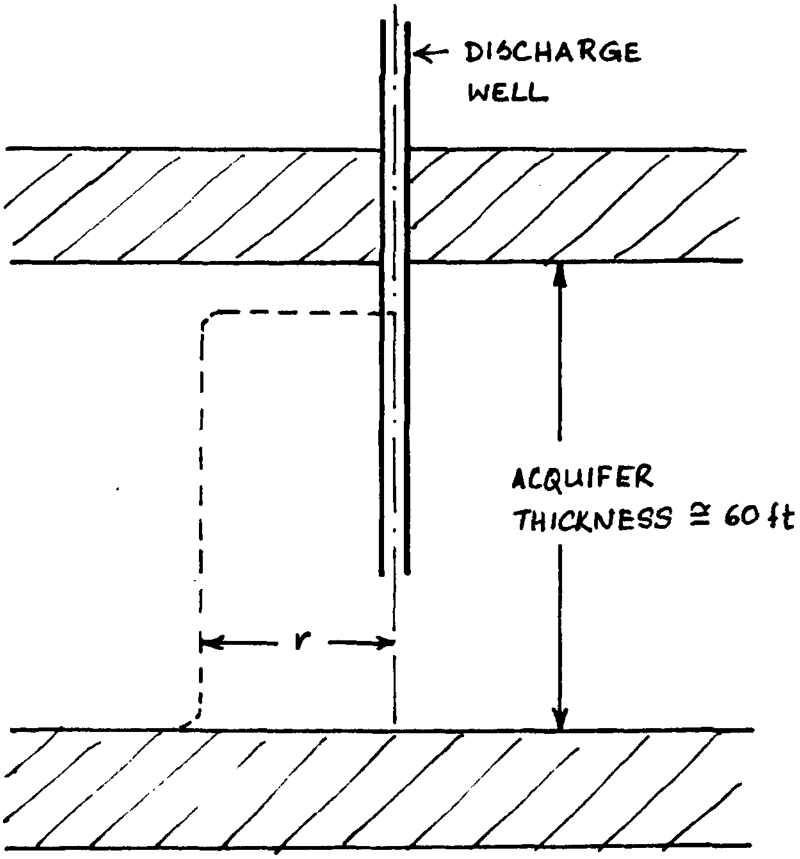


Fig. 4-9

WELL MODEL

Neglecting heat transport due to conduction, the temperature migration problem can be mathematically represented as:

$$\{(1-\varepsilon) \cdot \rho_a \cdot C_{P,a} + \varepsilon \cdot \rho_w \cdot C_{P,w}\} \cdot dT/d\Gamma$$
$$= -\dot{v} \cdot \rho_w \cdot C_{P,w} \cdot dT/dr \qquad (16)$$

where,

ρ_a , ρ_w = densities of aquifer material and water, respectively

$C_{P,a}$, $C_{P,w}$ = specific heats of aquifer material and water, respectively

ε = porosity of aquifer

$\dot{v}$ = velocity of water flow

$dT/d\Gamma$ = rate of change of temperature with time at a particular radial distance from the well axis

dT/dr = variation of temperature with radial distance from injection well at a particular time.

Representatives values of the aquifer properties are

ρ_a = 2.5 g/cc

$C_{P,a}$ = 0.24 cal/g.°C

and ε = 0.28

The water flow velocity is given by

$$\dot{v} = 0.19 \times (\text{TONS})/r, \text{ ft/hr} \qquad (17)$$

where

TONS = heat pump capacity in tons.

Equation 17 is derived considering 2,000 hours of operation (cooling cycle), a water flow of 2.5 gpm per ton and an aquifer thickness of 60 feet.

Substituting equation 17 into equation 16 and simplifying gives

$$dT/Dr + 0.267 \cdot (TONS/r) \cdot dT/dr = 0 \qquad (18)$$

Now equation 18 is the expression for the "Substantial Derivative" of the temperature (see Ref. 6). Thus, the velocity of the temperature "front," $\dot{v}_T$, is given by,

$$\dot{v}_T = 0.267 \cdot (TONS/r) \qquad (19)$$

Equation 19 represents the velocity at which the temperature "wave" or "front" travels and, therefore, is the migration speed. Conceptually, the temperature profile can be thought of as a horizontal "thermocline." The migration distance is obtained simply by integrating equation 19. Integration and simplification yields,

$$r_T = 32.67 \text{ x } TONS^{\frac{1}{2}}, \text{ ft} \qquad (20)$$

where,

r_T = migration distance or minimum well spacing.

It should be noted that equation 20 is derived for a particular aquifer, neglecting conduction effects. The main effect of conduction is to "smear the temperature profile" only (7). However, conduction is the primary mode of heat transfer for heat losses from the aquifer which are insignificant compared to the net heat injected (8).

Table 4-5 lists some results from equation 20. Also indicated are the well spacings for the case studies.

Table 4-5

Well Spacing vs. Heat Pump Tonnage

Tons	Spacing (r_T)	Remarks
5	74	residence
10	104	
20	148	
25	165	lab/research bldg.
50	233	
100	330	
150	404	school bldg.

Thus, for a typical residential unit, a well spacing of 80 feet is sufficient from equation 20. A spacing of 100 feet would be conservative, yet convenient for a typical residential lot.

5.0 REGULATORY AND ENVIRONMENTAL CONSIDERATIONS

5.1 Regulatory Aspects

Two wells are needed for the water portion of groundwater heat pump systems. There is an extraction well and an injection well. The water will be pumped out of the ground, through a heat exchanger and back into the ground. There will be no "use" of the water nor will any foreign material be introduced. It is a closed system. These wells do not give rise to any health implications and, therefore, fall outside the scope of health regulations.

The more pertinent regulatory consideration is that of subsidence, which is a sensitive matter in the Gulf Coast Region. The Harris-Galveston Coastal Subsidence District is the authority governing wells not drilled for the purpose of supplying potable water.

The Subsidence District operates by authority of House Bill No. 552, passed 12 May 1975, as amended by House Bill 390, passed 19 May 1977 (Vernon's Annotated Texas Statutes, Chapter 284, Water Auxiliary Laws). A general statement of the law is that a permit is required for any well located within the district which is to be used for the purpose of withdrawing groundwater. Certain exclusions to this general statement are made and the Act does not apply to:

1. Wells regulated under the provisions of Chapter 22, Water Code.
2. Shallow wells, commonly known as relief wells, producing water solely to prevent hazardous sand boils, de-water surface construction sites, or relieve hydrostatic

uplift on permanent structures and not used to provide a water supply for human consumption, agricultural use, manufacturing or industrial use or water injection.

3. Those persons owning only one well within the district, which well has a casing diameter of five inches or less; and

4. Such wells with a casing diameter of five inches or less which serve a single-family dwelling and which have a negligible effect upon subsidence within the district, provided that an exemption under this subdivision shall be allowed only upon application therefor in the manner and according to the form prescribed by the Board of Applications.

Clearly, any well with a case diameter over five inches will require a permit. Also, it is clear that one single well of less than five inches in diameter may be used without a permit or application for exemption. However, those groundwater heat pump systems described in this report which will need a well of less than five inches in diameter (all but very large commercial systems) also have an injection well. This set of circumstances is not so clear. A formal ruling from the Subsidence District was impractical for the purposes of this study. The safe course of action would be to make application for an exemption from the permit requirement. An informal response from the Subsidence District indicates that such exemptions would be readily granted.

5.2 Tax Considerations

The Energy Tax Act of 1978 provides a "residential energy credit" which is available for expenditures made after 20 April 1977 for equipment which uses a "renewable energy source" to heat, cool, or provide hot water for a taxpayer's principal residence.

The Energy Tax Act of 1978 added Section 44c to the Internal Revenue Code. This section provides that the amount of the credit (called the "residential energy credit") is 30 percent of the first $2,000 plus 20 percent of the next $8,000 of the expenditure the taxpayer makes on "renewable energy source property" which is defined as "property...which when installed in connection with a dwelling...transmits or uses...energy derived from geothermal deposits...for the purpose of heating or cooling such dwelling or providing hot water for use within such dwelling."

In another section of the Code [613(e)(3)], "geothermal deposits" is defined as "a geothermal reservoir consisting of natural heat which is stored in rocks or in an aqueous liquid or vapor (whether or not under pressure)." Groundwater would fit this definition.

The Act specifically makes deductible the labor costs incurred in installing the equipment which utilizes the renewable energy source. How far this goes is yet to be seen. This Act only recently became law and there are no indications as to how the IRS will interpret its provisions. The National Water Well Association takes the position that the credit covers groundwater heat pump systems, including well construction costs.

A very strong argument can be made to support the NWWA's position. It would be much easier to determine if one knew that Congress contemplated groundwater heat pump systems or even was aware of them.

5.3 Environmental Aspects

The following statements concerning any environmental impact resulting from the use of groundwater heat pump systems are of an empirical nature. This investigation, being limited to a conceptual design, could not produce data sufficient to support unqualified conclusions. These statements are also restricted to systems using an injection well, as opposed to those discharging the water into a river, lake or elsewhere.

The environmental aspects specific to groundwater heat pump use have been largely ignored. Of the two items involved, temperature and pressure, only the former has been treated and then solely in regard to heat recovery, not the environment.

5.3.1 Temperature Effects

During the operation of a groundwater heat pump system in the heating mode, the injected water will be cooler than the supply. In the cooling mode, the injected water will be warmer than the supply water. This change in temperature will vary with the flow rate as shown in Table 5-1.

Table 5-1

Effect of Flow Rate on Temperature Differentials

Gallons/Minute/Ton	ΔT(°F)
4.8	5
2.4	10
1.6	15
1.2	20
1.0	25

Thus, the units described in Section 3.1, a case history, which require a flow rate of 4.5 gallons/minute for the three-ton unit would increase or decrease the water temperature by 15° F. The groundwater would be cooled from 19.4° C (67° F) to 11.1° C (52° F) during the heating cycle (the heating being extracted from the space) and heated from 19.4° C (67° F) to 27.8° C (82° F) during the cooling cycle (the space heat being "dumped" into the water). It is estimated that the typical ΔT is going to be about 10° F with present design heat pumps.

If the injection well is in the same aquifer as the supply well, the spacing between the two becomes important. Spacing and temperature migration between wells in the same aquifer is treated in Section 4.4.

Under these conditions, there would be merit in systems having a reversing capability. However, if the spacing is proper, there should be no need to reverse the flow.

In practice, all of the documented systems currently in use have the supply well at a different depth and in a different aquifer than the injection well. Under these conditions, temperature migration is not pertinent.

Space conditioning in the Texas Gulf Coast Region requires slightly more cooling than heating. As a result, there will be a net heating of the groundwater at the injection point.

The true, specific amount of temperature change within an aquifer cannot be meaningfully addressed on a general basis. There are simply too many variables--size of the aquifer, soil porosity, flow rate, etc. However, there is no evidence that temperature change is a problem in areas that have had these systems in use for 20 years. Furthermore, based on discussions with reputable geologists (7), any effect would be negligible in the Texas Gulf Coast Region because of the near balance of heating and cooling.

5.3.2 Pressure Effects

The environmental impact of pressure change in an aquifer is, as with temperature change, highly site specific. As a practical matter, there is unlikely to be any effect when both the supply and injection wells are in the same aquifer. The use of different aquifers would be more likely to produce a measurable effect.

The shallow wells in use in the referenced case history have not resulted in any noticeable pressure effects. Some of these injection wells are as shallow as 30 feet. Likewise, there is no evidence of a problem from the rather wide use of these systems in Florida. Conceivably, the continual addition of water to an aquifer of this depth could lead to a change in ground level through hydrostatic pressure.

The geologists surveyed were of the opinion that any pressure effect would be negligible in the Texas Gulf Coast Region due to the size of the aquifers involved.

5.3.3 Corrosion Aspects

Experience with existing groundwater systems indicates that corrosion is not a very severe problem in most cases, especially due to the almost universal use of cupro-nickel tubing in the water-side heat exchangers of groundwater heat pumps. One manufacturer's representative (Vanguard Energy Systems, San Diego, California) stated that their tubing also has a built-in cleaning feature by virtue of its high coefficient of thermal expansion which promotes self-peeling of any scale that may build up. Electrochemical effects due to differing qualities of water from different aquifers are also suppressed in present day heat pumps by using "coil-in-coil" exchangers wherein the outer coil (for the groundwater) is made of plastic, while the inner coil (for freon or whichever refrigerant is being used) is made of cupro-nickel.

Large scale systems may pose a slightly more difficult problem and may require careful material selection. However, Battelle has found it beneficial to simply use sacrificial anodes to combat corrosion. According to Tom Molloy at Battelle Laboratories, sacrificial magnesium anodes were all that were needed to completely eliminate corrosion. No pre-treatment of any kind was necessary.(9)

Thus, it can be assumed that corrosion has a negligible detrimental effect on the groundwater systems.

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APPENDIX 1

OPERATING COPs OF HEAT PUMPS

The ideal efficiency (or COP) of a heat pump was shown to be a function of the source and sink temperatures in Section 2.6.1. It was also stated that the operating COPs are much lower than the ideal COPs due to various inefficiencies. Mechanical inefficiencies in the compressor and motor contribute one part to this COP decrease. Frictional resistance to refrigerant flow is the second contributing factor. A third factor is due to heat transfer limitations.

A general principle of heat transfer states that heat can be exchanged only between fluids at different temperatures. Thus if, for example, heat is being absorbed from room air into a refrigerant (see Figure 2-2), then the refrigerant must be at a lower temperature. On the other hand, the refrigerant must be at a higher temperature if it is to reject heat to some sink. Thus, the temperatures in equations 4 and 5 (section 2.4) have to be suitably modified to take into account this requirement.

The ideal cooling COP is

$$COP_{cooling} = 1/(T_2/T_1 - 1)$$

where T_2 is the sink temperature and T_1 is the source temperature. In operating systems, therefore, the refrigerant temperature will be higher than T_2 when rejecting heat, and lower than T_1 when absorbing heat. Thus, this first modification gives

$$COP_{cooling} = 1/\{(T_2 + \Delta T)/(T_1 - \Delta T) - 1\}$$

where ΔT represents the temperature differentials that exist across the heat exchangers.

The compressor inefficiency and the frictional pressure drop can be accounted for by including an efficiency factor (η); i.e.

$$COP_{cooling} = \eta/\{(T_2 + \Delta T)/(T_1 - \Delta T) - 1\}$$

Reasonable values for η and ΔT are 0.5 and 15 K, respectively. Thus, operating cooling COPs can be estimated by

$$COP_{cooling} = 0.5/\{(T_2 + 15)/(T_1 - 15) - 1\}$$

Similar modifications can be made to the heating COP giving

$$COP_{heating} = 0.5/\{1 - (T_2 - 15)/(T_1 + 15)\}$$

It should be mentioned that this procedure for estimating operating COPs (as opposed to ideal COPs) is entirely empirical. We have tried to include, in a simple way, the most important contributers to heat pump inefficiency. Detailed computer simulations involving dynamic operating characteristics have been performed by various authors that estimate actual COPs more accurately. The results of such simulations indicate that actual COPs may be as much as 25% lower than the COPs calculated here.

APPENDIX 2

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APPENDIX 3

GROUNDWATER HVAC SYSTEMS

Summary of Load Estimates
and
Completed Load Calculation Sheets

Case Study No. 1

Building type: single family residence

Location: Sugarland, Texas

Floor area: 1,972 square feet

Heating Load (Btu/H)

Heat loss by conduction = 36,502

Heat loss by leakage/ventilation = 13,500

Total heating load = 50,002

Cooling Load (Btu/H)

Heat gain by conduction = 14,601

Direct solar gain = 15,508

Internal heat generation (people and appliances) = 28,005

Sensible heat from leakage/ventilation = 5,400

Total latent heat gain = 12,520

Total cooling load = 76,034

Air supply CFM = 2,700

Tons/1,000 sq. ft. (approx.) = 3.2

Case Study No. 2

Building type:	office-cum-laboratory
Location:	Houston, Texas
Floor area:	7,843 square feet

Heating Load (Btu/H)

Heat loss by conduction	=	97,550
Heat loss by leakage/ventilation	=	56,160
Total heating load	=	153,710

Cooling Load (Btu/H)

Heat gain by conduction	=	71,632
Direct solar gain	=	2,384
Internal heat generation (people and appliances)	=	123,224
Leakage/ventilation	=	22,464
Latent heat gain	=	60,824
Total cooling load	=	280,528

Air supply CFM	=	9,572
Tons/1,000 sq. ft.	=	3.0

Case Study No. 3

Building type: school building

Location: Montgomery County, Texas

Floor area: 56,250 square feet

Total heating load: 615,000 Btu/H

Total cooling load: 1,512,000 Btu/H

No load calculations were performed for this building since the design specifications had already been finalized by the contractors.

Case Study No. 4

Building type: Two-story office building-cum-machine shop

Location: Katy, Texas

Floor area: 5,000 square feet (office space only)

(Air conditioning for the machine shop area is not provided due to its intimate contact with outside environment.)

Heading Load (Btu/H)

1st floor load = 53,080

2nd floor load = 63,000

Cooling Load (Btu/H)

1st floor load = 92,682

2nd floor load = 102,682

Supply Air

1st floor = 3,100 CFM

2nd floor = 3,500 CFM

DALE S. COOPER & ASSOCIATES

HEAT GAIN CALCULATION

Name Plan #21-150 A

~~Room~~ XXXX Used For residence

Job #

City Sugarland, Texas

Room Number

Sheet #

Calculated By: Dale S. Cooper

Date: 12/12/78

Zone #

DESIGN CONDITIONS

Outdoor DB Temp	95 °F	WB Temp	80	Abs. Hum.	130 Gr/lb	Cal For	AM 4 PM Aug 2
Indoor DB Temp	75 °F	WB Temp	62.5 °F	Abs. Hum.	66 Gr/lb	Room Volume	15,776 cu. ft.
Temp. Diff.	20 °F	Winter ΔT = 50°F		Hum. Diff.	64 Gr/lb	1972 x 8' = 15,776 cu. ft.	

COL. I	II	III	IV	V	VI	VII	VIII	IX	X
SIDE FACING	WALL/PART. DIM. WINDOW & DOOR NO. & DIM. OR AREA	GROSS WALL AREA SQ/FT TOTAL GLASS AREA SQ/FT NET WALL AREA SQ/FT	U_G U_W	D_B T_D	CONDUC. FR. TEMP. DIFF. BTU/H	FACTOR R_G R_W	SUN EFFECT BTU/H INCLUDING SHADING	FACTOR	HEAT LOSS BTU/H
	56' x 8'	448							
N	1-3x7 = 21	108	1.1	20	2376	25	2700		5,940
1 - 3x3=9	2-3x16 = 36 1-6x7 = 42	340	.07		476			2.5	1,190
	50' x 8'	400							
E	1-2x4 = 8 2-6x7 = 84	104	1.1	20	2288	23	2392	2.5	5,720
	1-3x4 = 12	296	.07		414			2.5	1,035
	42' x 8'	336							
S	Brickwall	-	-	20	-	-	-		
		336	.07		470			2.5	1,175
	54' x 8'	432							
W	1-3x3 = 9 2-3x7 = 42	87	1.1	20	1914	219	19053	2.5	4,785
	2-3x6 = 36	345	.07		483			2.5	1,207
E-Exposure	15' x 8'	120							
W-Wing	3 - 3x7=63	63	1.1	20	1386	23	1449	2.5	3,465
		57	.07		80			2.5	200
S-Front	16' x 10'	160							
XXXX	4 - 3x7 =84	84	1.1	20	1848	31	2604	2.5	4,620
Entrance		76	.07		106			2.5	265
CLG. XXXX					11841				
	(see take off)	1972	.07		2760			2.5	6,900
			TOTAL		14601		28,198		36,502

TOTAL CONDUCTION: BTU H	14,601	HEAT LOSS
TOTAL SOLAR: shading - single glass - light ven. blinds 55%	15,508	13,500
PEOPLE: NO. OF PEOPLE IN RM. (5) X 220	1,100	50,000
ELECTRICAL APPLIANCES: WATTS (7 kw) X 3.4	23,905	x2.5
OTHER SOURCES: BTU H	3,000	
TOTAL INTERNAL SENSIBLE HEAT GAIN	58,114	
VENTILATION AIR CFM. (250) X TEMP. DIFF. (20) X 1.08	5,400	
TOTAL SENSIBLE HEAT GAIN H_S		63,514

LATENT HEAT GAIN

PEOPLE: NO. IN RM. (5) X (180) BTU H	900	
OTHER SOURCES: LB. OF WATER PER/HR. (1) X 1060	1,060	
VENTILATION AIR: CFM (250) X HUM. DIFF. (64) X 0.66	10,560	
TOTAL LATENT GAIN H_L	12,520	12,520
TOTAL HEAT GAIN H_T		76,034
TONS	6.34	

CFM SUPPLY AIR: 2700 (min)

$\frac{58114}{1.08x20}$ =

DALE S. COOPER & ASSOCIATES **HEAT GAIN CALCULATION**

Name Lab / Research Room Used For offices Job #

City Houston, Texas Room Number Sheet # 1 of 3

Calculated By Dale S. Cooper Date: 2/7/79 Zone # 1

DESIGN CONDITIONS

Outdoor DB Temp 95 F WB Temp 80 F Abs. Hum. 130 Gr/lb Cal For AM 4 PM

Indoor DB Temp 75 F WB Temp 62.5 F Abs. Hum. 66 Gr/lb Room Volume 12,200 cu. ft.

Temp. Diff. 20 F Winter ΔT = 50°F Hum. Diff. 64 Gr/lb 1-A/C = 203 CFM

COL. I SIDE FACING	II WALL/PART. DIM. WINDOW & DOOR NO. & DIM. OR AREA	III GROSS WALL AREA SQ/FT TOTAL GLASS AREA SQ/FT NET WALL AREA SQ/FT	IV U_G U_W	V D_B T_D	VI CONDUC. FR. TEMP. DIFF. BTU/H	VII FACTOR R_G R_W	VIII SUN EFFECT BTU/H INCLUDING SHADING	IX FACTOR	X HEAT LOSS BTU/H
N	32 x 8	256							
to loading area	2 - 3x7	42	1.1	10	462			2.5	1155
		214	.1		214			2.5	535
E	60 x 8	480							
exposed	no windows or doors	- 480	.08	20	768	.08x15	576	2.5	1,920
S	20 x 8	160							
in whse	1 - 3x7	21	1.1	15	346			1.7	577
		131	.25		521			"	868
W	19 x 8	152							
exp. to whse.	1 - 3x7	21	1.1	15	346			"	577
		131	.25		491			"	818
	54 x 8	432							
W interior	Adjacent A/C space	- -		0	- -				- -
CLG.	33x21 = 693 32x26 = 832	1525							
	4" Batts		.08	50	6100			1	6,100
FLR.	same as CLG.	1525							
			.6	5	4575		576	1	4,575
			TOTAL		13,824				17,125

TOTAL CONDUCTION: BTU/H 13,824

TOTAL SOLAR: 576

PEOPLE: NO. OF PEOPLE IN RM. (15) x 220 3,300

ELECTRICAL APPLIANCES: WATTS (6100) x 3.4 20,740

OTHER SOURCES: BTU/H . . . . NONE. . . . . --

TOTAL INTERNAL SENSIBLE HEAT GAIN 38,440

VENTILATION AIR CFM. (200) x TEMP. DIFF. (20) x 1.08 4,320

TOTAL SENSIBLE HEAT GAIN . . . . H_S Cooling 42,760

HEAT LOSS

17,125

10,800

27,925

x2.5

LATENT HEAT GAIN

PEOPLE: NO. IN RM. (15) x (200) BTU/H 3,000

OTHER SOURCES: LB. OF WATER PER HR. (1) x 1060 1,060

VENTILATION AIR: CFM (200) x HUM. DIFF. (60) x 0.66 7,920

CFM SUPPLY AIR: 2572 TOTAL LATENT GAIN H_L Cooling 11,980

$\frac{50,000}{1.08 \times 18}$ = TOTAL HEAT GAIN H_T 54,740

TONS 4.5 Cooling

DALE S. COOPER & ASSOCIATES **HEAT GAIN CALCULATION**

Name Lab/Research
Room Used For offices
Job #
City Houston, Texas
Room Number
Sheet # 2 of 3
Calculated By: Dale S. Cooper
Date: 2/7/79
Zone # 2 w/o #3

DESIGN CONDITIONS

Outdoor DB Temp 95 °F WB Temp 80 °F Abs. Hum. 130 Gr/lb Cal For AM 4 PM Aug 21
Indoor DB Temp 75 °F WB Temp 62.5 °F Abs. Hum. 66 Gr/lb Room Volume 44,320 cu. ft.
Temp. Diff. 20 °F Winter ΔT = 50°F Hum. Diff. 64 Gr/lb 1 - A/C - 740 CFM

COL. I	II	III	IV	V	VI	VII	VIII	IX	X
SIDE FACING	WALL, PART. DIM. WINDOW & DOOR NO. & DIM. OR AREA	GROSS WALL AREA SQ/FT TOTAL GLASS AREA SQ FT NET WALL AREA SQ/FT	U_G U_W	D_B T_D	CONDUC. FR. TEMP. DIFF. BTU/H	FACTOR ~~R_G R_W~~	SUN EFFECT BTU/H INCLUDING SHADING	FACTOR	HEAT LOSS BTU/H
	67 x 8	536							
N exposed	1 - 7x8	56	1.1	20	1232			2.5	3,080
	front door	480	.08		768			"	1,920
	38 x 8	304							
E adj.loading	1 - 7x7	49	1.1	10	539			5	2,695
	door	255	.25		637			"	3,185
	79 x 8	632							
S exp. whse.	2 - 3x7	42	1.1	15	693			3.3	2,310
									7,373
	80 x 8	640							
W	1 - 4x3	12	1.1	20	264			2.5	660
		628	.08		1005	.08x36	1808	"	2,512
	80 x 67 =	5540							
CLG.	15 x 12 =		-					1	
	4" Batts	5540	.08	50	22160				22,160
	same as CLG.	5540							
FLR.	C on C slab on GND	5540	.6	5	16,620				16,620
			TOTAL		46,130				62,515 HEAT LOSS

TOTAL CONDUCTION: BTU/H 46,130
TOTAL SOLAR: 1,808
PEOPLE: NO. OF PEOPLE IN RM. (55) x 220 12,100
ELECTRICAL APPLIANCES: WATTS 22,160) x 3.4 75,344
OTHER SOURCES: BTU/H .... NONE --
TOTAL INTERNAL SENSIBLE HEAT GAIN 135,382
VENTILATION AIR CFM. (740) x TEMP. DIFF. (20) x 1.08 15,984
TOTAL SENSIBLE HEAT GAIN .... H_S Cooling 151,366

39,960
102,475
x2.5

LATENT HEAT GAIN

PEOPLE: NO. IN RM. (55) x (200) BTU/H 11,000
OTHER SOURCES: LB. OF WATER PER HR. (2) x 1060 2,120
VENTILATION AIR: CFM (740) x HUM. DIFF. (60) x 0.66 29,304
CFM SUPPLY AIR: 7000
TOTAL LATENT GAIN H_L Cooling 42,424
TOTAL HEAT GAIN H_T 103,790
TONS 16

$$\frac{135380}{1.08x18}$$

DALE S. COOPER & ASSOCIATES **HEAT GAIN CALCULATION**

Name I Lab/ Research Room Used For offices Job #
City Houston, Texas Room Number Energy Laboratory Sheet # 3 of 3
Calculated By: Dale S. Cooper Date: 2/8/79 Zone # 3 - add to #2

DESIGN CONDITIONS

Outdoor DB Temp 95 °F WB Temp 80 °F Abs. Hum. 130 Gr/lb Cal For AM 4 PM Aug 21
Indoor DB Temp 75 °F WB Temp 62.5 °F Abs. Hum. 66 Gr/lb Room Volume 6,224
Temp. Diff. 20 °F Winter ΔT - 50°F Hum. Diff. 64 Gr/lb 1 - A/C - 104 CFM

COL. I SIDE FACING	II WALL/PART. DIM. WINDOW & DOOR NO. & DIM. OR AREA	III GROSS WALL AREA SQ/FT TOTAL GLASS AREA SQ/FT NET WALL AREA SQ/FT	IV U_G U_W	V D_B T_D	VI CONDUC. FR. TEMP. DIFF. BTU/H	VII FACTOR R_G R_W	VIII SUN EFFECT BTU/H INCLUDING SHADING	IX FACTOR	X HEAT LOSS BTU/H
N adj. Z#2	Adjacent A/C Space	-- --		0					
E adj. whse.	23 x 8	207 -- 207	-- .3	25	1552			2.5	3104
S adj. whse.	46 x 8 3 - 3x7	368 42	1.1	25	1155			2.5	2310
W adj. whse.	18 x 8	144 -- 144	-- .3	25	1080				2160
CLG.	5 x 8 = 40 41 x 18 = 738	778 -- 778	.08	50	3112			1	3112
FLR.	same as CLG.	778	.6	5	2334			1	2334
			TOTAL		11678				17,910

TOTAL CONDUCTION: BTU/H .... 11,678 HEAT LOSS
TOTAL SOLAR: ...no exposure to sun .... -- 5,400
PEOPLE: NO. OF PEOPLE IN RM. (7) x 220 .... 1,540 23,310
ELECTRICAL APPLIANCES: WATTS (3000) x 3.4 10,200
OTHER SOURCES: BTU/H ..... NONE .... --
TOTAL INTERNAL SENSIBLE HEAT GAIN 23,418 x2.5
VENTILATION AIR CFM. (100) x TEMP. DIFF. (20) x 1.08 2,160
TOTAL SENSIBLE HEAT GAIN .... H_S Cooling 25,578

LATENT HEAT GAIN

PEOPLE: NO. IN RM. (7) x (200) BTU/H 1,400
OTHER SOURCES: LB. OF WATER PER/HR. (1) x 1060 1,060
VENTILATION AIR: CFM (100) x HUM. DIFF. (60) x 0.66 3,960
CFM SUPPLY AIR
TOTAL LATENT GAIN H_L 6,420
TOTAL HEAT GAIN H_T 31,998
TONS 3 Cooling

DALE S. COOPER & ASSOCIATES **HEAT GAIN CALCULATION**

Name office-warehouse Room Used For shops - offices Job #
City Houston Room Number 2nd floor Sheet #
Calculated By: Dale S. Cooper Date: 3/13/79 Zone #

DESIGN CONDITIONS

Outdoor DB Temp 95 F WB Temp 80 F Abs. Hum. 130 Gr/lb Cal For AM 4 PM Aug 21
Indoor DB Temp 75 F WB Temp 62.5 F Abs. Hum. 66 Gr/lb Room Volume 22,500 cu. ft.
Temp. Diff. 20 F Winter ΔT = 50°F Hum. Diff. 64 Gr/lb

COL. I SIDE FACING	II WALL, PART. DIM. WINDOW & DOOR NO. & DIM. OR AREA	III GROSS WALL AREA SQ FT TOTAL GLASS AREA SQ FT NET WALL AREA SQ/FT	IV U_G U_W	V D_D T_D	VI CONDUC. FR. TEMP. DIFF. BTU/H	VII FACTOR R_G R_W	VIII SUN EFFECT BTU/H INCLUDING SHADING	IX FACTOR	X HEAT LOSS BTU/H
		990							
N		220	1.1	20	4884			2.5	12,210
		768	0.08		1228			2.5	3,070
		605							
E		-		20					
		605	0.08		968			2.5	2,420
		990							
S		180	1.1	20	3960			2.5	9,900
		605	0.08		1296			2.5	3,240
		605							
W		21	1.1	20	462			2.5	1,155
		584	0.08		934			2.5	2,335
CLG.		2,500	0.08	50	10,000			1	10,000
FLR.									
				TOTAL	23,752				44,330

HEAT LOSS

TOTAL CONDUCTION: BTU/H 23,732
TOTAL SOLAR: .... -- → 20,250
PEOPLE: NO. OF PEOPLE IN RM. (15) x 220 3,300
ELECTRICAL APPLIANCES: WATTS (12,500) x 3.4 .... 42,500 64,580
OTHER SOURCES: BTU/H .... -- x2.5
TOTAL INTERNAL SENSIBLE HEAT GAIN 69,532
VENTILATION AIR CFM. (375) x TEMP. DIFF. (20) x 1.08 8,100
TOTAL SENSIBLE HEAT GAIN .... H_S 77,632

LATENT HEAT GAIN

PEOPLE: NO. IN RM. (15) x (180) BTU/H 2,700
OTHER SOURCES: LB. OF WATER PER/HR. ( ) x 1060 .... --
VENTILATION AIR: CFM (375) x HUM. DIFF. (60) x 0.66 14,850
CFM SUPPLY AIR: 3220 TOTAL LATENT GAIN H_L 17,550

$\frac{69532}{1.08x20}$ = ______↑ TOTAL HEAT GAIN H_T 95,182
TONS 7.9

DALE S. COOPER & ASSOCIATES **HEAT GAIN CALCULATION**

Name office-warehouse
City Houston
Calculated By: Dale S. Cooper
Room Used For shops - offices
Room Number 1st floor
Date: 3/13/79
Job #
Sheet #
Zone #

DESIGN CONDITIONS

Outdoor DB Temp 95 °F WB Temp 80 °F Abs. Hum. 130 Gr/lb Cal For AM 4 PM Aug
Indoor DB Temp 75 °F WB Temp 62.5 °F Abs. Hum. 66 Gr/lb Room Volume 22,500 cu. ft
Temp. Diff. 20 °F Winter ΔT = 50°F Hum. Diff. 64 Gr/lb

COL. I SIDE FACING	II WALL PART. DIM. WINDOW & DOOR NO. & DIM. OR AREA	III GROSS WALL AREA SQ/FT TOTAL GLASS AREA SQ FT NET WALL AREA SQ/FT	IV U_G U_W	V D_B T_D	VI CONDUC. FR. TEMP. DIFF. BTU/H	VII FACTOR R_G R_W	VIII SUN EFFECT BTU/H INCLUDING SHADING	IX FACTOR	X HEAT LOSS BTU/H
		990							
N		220	1.1	20	4884			2.5	12,210
		768	0.08		1228			2.5	3,070
		605							
E		-		20					
		605	0.08		968			2.5	2,420
		990							
S		180	1.1	20	3960			2.5	9,900
		605	0.08		1296			2.5	3,240
		605							
W		21	1.1	20	462			2.5	1,155
		584	0.08		934			2.5	2,335
CLG.									
FLR.				5					
		2500	0.6		7500			1	7,500
			TOTAL		21,232				41,830

TOTAL CONDUCTION: BTU/H 21,232
TOTAL SOLAR: . . . . --
PEOPLE: NO. OF PEOPLE IN RM. (15) x 220 3,300
ELECTRICAL APPLIANCES: WATTS (12,500) x 3.4 42,500
OTHER SOURCES: BTU/H . . . . --
TOTAL INTERNAL SENSIBLE HEAT GAIN 67,032
VENTILATION AIR CFM. (375) x TEMP. DIFF. (20) x 1.08 8,100
TOTAL SENSIBLE HEAT GAIN . . . . H_S 75,132

HEAT LOSS
20,250
62,080
x2.5

LATENT HEAT GAIN

PEOPLE: NO. IN RM. (15) x (180) BTU/H 2,700
OTHER SOURCES: LB. OF WATER PER/HR. ( ) x 1060 . . . . --
VENTILATION AIR: CFM (375) x HUM. DIFF. (60) x 0.66 14,850
CFM SUPPLY AIR: 3100
TOTAL LATENT GAIN H_L 17,550
TOTAL HEAT GAIN H_T 92,682
TONS 7.7

$$\frac{67032}{1.08 \times 20} =$$

APPENDIX 4

EQUIPMENT COST DATA SHEETS
FOR
CASE STUDIES

EQUIPMENT COST DATA

Case Study No. 1 Building Type residential HVAC System conventional

Cooling CAP 76,000 Btu/H Heating CAP 50,000 Btu/H Air CFM 2,700

Average EER 8.0 (cooling) Average COP 2.5 (heating)

ITEM	DESCRIPTION	MODEL	UNIT PRICE	QTY.	TOTAL PRICE
1	Air-to-air heat pumps	Sears #42A82203N2	1,280*	2	2,560
2	Backup electric furnace	Sears #42A58711N	250*	2	500
					$3,060

* Prices from Sears 1978-79 Catalogue.

EQUIPMENT COST DATA

Case Study No. 1

Building Type residential

HVAC System groundwater

Cooling CAP 76,000 Btu/H

Heating CAP 50,000 Btu/H

Air CFM 2,700

Average EER 11 (cooling)

Average COP 4.0 (heating)

ITEM	DESCRIPTION	MODEL	UNIT PRICE	QTY.	TOTAL PRICE
1	Groundwater heat pumps	Florida Heat Pump Co. #EM-42	1,775	2	$3,550
2	2", 100 ft. deep supply well with 1 h.p. motor		1,000*	1	1,000
3	2", 100 ft. injection well		670*	1	670
4	36 gal. surge tank	Sears #42A2921N	95**	1	95
					$5,315

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* Prices obtained from Almeda Water Well Service, P. O. Box 45474, Houston, TX 77045.

** Prices obtained from Sears 1978-79 Catalogue.

EQUIPMENT COST DATA

Case Study No. 2
Building Type lab/research
HVAC System conventional
Cooling CAP 23 tons
Heating CAP 13 tons (45 kw)
Air CFM
Average EER 8.0 (cooling)
Average COP 1.0 (heating)

ITEM	DESCRIPTION	MODEL	UNIT PRICE	QTY.	TOTAL PRICE
1	23 ton air cooled packaged water chiller (includes insulated chiller, condenser, compressor, motor, heat interchanger, normal basic controls and complete piping)		$470/ton *	1	$10,810
2	45 kw packaged electric boilers (jacket, circulator and controls)		30/kw *	1	1,350
					$11,160

* Prices estimated from Khashab (   )

EQUIPMENT COST DATA

Case Study No. 2
Building Type lab/research
HVAC System groundwater #1

Cooling CAP 23 tons
Heating CAP 45 kw
Air CFM

Average EER 11 (cooling)
Average COP 4.0 (heating)

ITEM	DESCRIPTION	MODEL	UNIT PRICE	QTY.	TOTAL PRICE
1	25 ton water source heat pump	Solar King	$13,332	1	$13,332
2	4", 100 ft. supply well		1,500 *	1	1,500
3	4", 100 ft. injection well		1,000 *	1	1,000
4	50 gpm (5 hp) submersible pump	Aermotor #SE50-5001	1,563	1	1,563
					17,395
				contingency (10%)	1,740
					$19,135

* Price estimates from Almeda Water Well Service, P. O. Box 45474, Houston, Texas 77045.

EQUIPMENT COST DATA

Case Study No. 2 Building Type lab/research HVAC System groundwater #2

Cooling CAP 23 tons Heating CAP 12.8 tons Air CFM

Average EER 11 (cooling) Average COP 4.0 (heating)

ITEM	DESCRIPTION	MODEL	UNIT PRICE	QTY.	TOTAL PRICE
1	25 ton water chiller assembly (includes a 2-circuit condenser, two 2-circuit chillers, a compressor and 4 expansion valves)	Carrier	$13,175	1	$13,175
2	4", 100 ft. supply well		1,500	1	1,500
3	4", 100 ft. injection well		1,000	1	1,000
4	50 gpm (5 hp) submersible pump		1,563	1	1,563
					17,238
				contingency (10%)	1,724
					$18,962

EQUIPMENT COST DATA

Case Study No. 3

Building Type school building

HVAC System conventional

Cooling CAP 125 tons

Heating CAP 180 kw (614 M Btu/H)

Air CFM

Average EER 8 (cooling)

Average COP 1.00 (heating)

ITEM	DESCRIPTION	MODEL	UNIT PRICE	QTY.	TOTAL PRICE
1	125 T packaged air cooled water chiller		$33,275	1	$33,275
2	180 kw electric boiler		4,356	1	4,356
					$37,621

EQUIPMENT COST DATA

Case Study No. 3 Building Type school building HVAC System groundwater #1

Cooling CAP 125 tons Heating CAP 180 kw (614 M Btu/H) Air CFM

Average EER 11 (cooling) Average COP 4 (heating)

ITEM	DESCRIPTION	MODEL	UNIT PRICE	QTY.	TOTAL PRICE
1	125 T assembled reciprocating water chiller		$37,500	1	$37,500
2	6", 300 ft. supply well		3,800	1	3,800
3	6", 300 ft. injection well		2,500	1	2,500
4	300 gpm, 40 hp submersible pump	Pioneer #P300, 40 hp	5,010	1	5,010
					48,810
				contingency (10%)	4,881
					$53,691

EQUIPMENT COST DATA

Case Study No. 3

Building Type school building

HVAC System groundwater #2

Cooling CAP 125 tons

Heating CAP 180 kw (614 M Btu/H)

Air CFM

Average EER (cooling)

Average COP (heating)

ITEM	DESCRIPTION	MODEL	UNIT PRICE	QTY.	TOTAL PRICE
1	60 T air cooled packaged chiller, plus 70 T assembled water cooled chiller	Carrier	$50,250	1	$50,250
2	6", 240 ft. supply well		3,600	1	3,600
3	6", 240 ft. return well		2,400	1	2,400
4	25 hp submersible pump	Pioneer #P230, 25 hp	2,770	1	2,770
					$59,020

EQUIPMENT COST DATA

Case Study No. 4 Building Type office/manufacturing HVAC System conventional

Cooling CAP 16 tons Heating CAP 10 tons Air CFM 6300

Average EER 8 (cooling) Average COP 0.6 (heating)

ITEM	DESCRIPTION	MODEL	UNIT PRICE	QTY.	TOTAL PRICE
1	15 T roof top air cooled condensing unit		$3,880	1	$3,880
2	Two zone, multizone blower coil unit (D-X coils, flat filters		4,650	1	4,650
3	120 MBH package gas furnace		300	1	300
					$8,830

EQUIPMENT COST DATA

Case Study No. 4 Building Type office/warehouse HVAC System groundwater

Cooling CAP 16 tons Heating CAP 10 tons Air CFM

Average EER 10 (cooling) Average COP 4 (heating)

ITEM	DESCRIPTION	MODEL	UNIT PRICE	QTY.	TOTAL PRICE
1	8 T water source reversible heat pump	Florida Heat Pump #HE1000	$5,003	2	$10,006
2	4", 100 ft. supply well		1,500	1	1,500
3	4", 100 ft. discharge well		1,000	1	1,000
4	50 gpm, 5 hp submersible pump	Aermotor #SE50-5001	1,563	1	1,563
					$13,069
				Contingency (10%)	1,307
					$14,376