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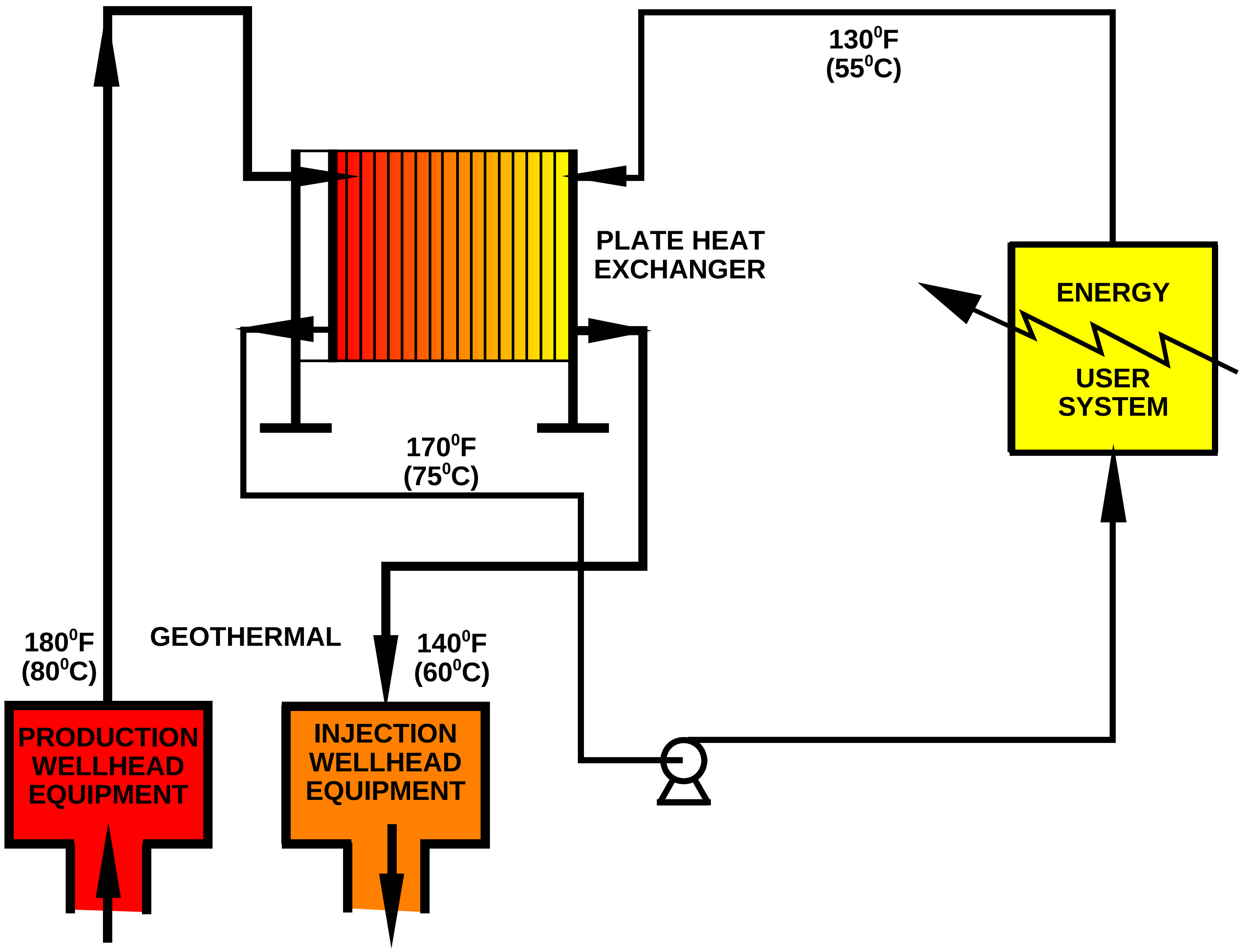
GEOTHERMAL DIRECT-USE EQUIPMENT



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GEO-HEAT CENTER QUARTERLY BULLETIN

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A Quarterly Progress and Development Report on the Direct Utilization of Geothermal Resources

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Cover: Typical geothermal direct-use equipment arrangement using a plate heat exchanger to isolate the potentially corrosive or scaling geothermal fluid from the equipment by using "clean" fluid in the secondary loop.

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GEOTHERMAL DIRECT-USE EQUIPMENT OVERVIEW

John W. Lund, P.E.
Geo-Heat Center

This article provides an overview of the various equipment components that are used in most geothermal direct-use project. Following, are articles describing in more detail five major types of equipment: well pumps, piping, heat exchangers, space heating equipment and absorption refrigeration equipment. These five articles are condensations of chapters written by Kevin Rafferty and Gene Culver, mechanical engineers with the Geo-Heat Center, that appear in the 3rd edition of our "Geothermal Direct-Use Engineering and Design Guidebook" (1998). Additional specifications and design information on these five major equipment items appear in this book as Chapters 9 through 13. Since these articles and chapters address only items used in direct heat projects (generally above about 100°F or 40°C), geothermal or ground-source heat pumps are not discussed. For information on the specifications, design and use of geothermal heat pumps used in commercial and institutional buildings, see Kavanaugh and Rafferty (1998)

INTRODUCTION

Standard equipment is used in most direct-use projects, provided allowances are made for the nature of geothermal water and steam. Temperature is an important considerations; so is water quality. Corrosion and scaling caused by the sometimes unique chemistry of geothermal fluids, may lead to operating problems with equipment components exposed to flowing water and steam. In many instances, fluid problems can be designed out of the system. One such example concerns dissolved oxygen, which is absent in most geothermal waters, except perhaps the lowest temperature waters. Care should be taken to prevent atmospheric oxygen from entering district heating waters; for example, by proper design of storage tanks. The isolation of geothermal water by installing a heat exchanger may also solve this and similar water quality derived problems. In this case, a clean secondary fluid is then circulated through the user side of the system as shown in Figure 1.

The primary components of most low-temperature direct-use systems are downhole and circulation pumps, transmission and distribution pipelines, peaking or back-up plants, and various forms of heat extraction equipment (Figure 1). Fluid disposal is either surface or subsurface (injection). A peaking system may be necessary to meet maximum load. Thus can be done by increasing the water temperature or by providing tank storage (such as done in most of the Icelandic district heating systems). Both options mean that fewer wells need to be drilled. When the geothermal water temperature is warm (below 120°F or 50°C), heat pumps are often used. The equipment used in direct-use projects represent several units of operations. The major units will now be described in the same order as seen by geothermal waters produced for district heating.

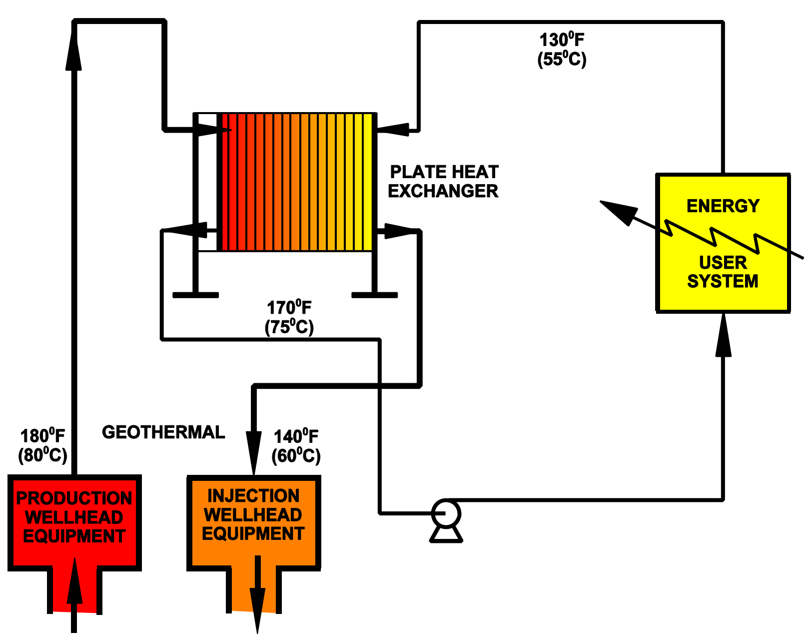

Figure 1. Geothermal direct utilization system using a heat exchanger.

DOWNHOLE PUMPS

Unless the well is artesian, downhole pumps are needed, especially in large-scale direct utilization systems. Downhole pumps may be installed not only to lift fluid to the surface, but also to prevent the release of gas and the resultant scale formation. The two most common types are: lineshaft pump systems and submersible pump systems.

The lineshaft pump system (Figure 2a) consists of a multi-stage downhole centrifugal pump, a surface mounted motor and a long drive shaft assembly extending from the motor to the pump. Most are enclosed, with the shaft rotating within a lubrication column which is centered in the production tubing. This assembly allows the bearings to be lubricated by oil, as hot water may not provide adequate lubrication. A variable-speed drive set just below the motor on the surface, can be used to regulate flow instead of just turning the pump on and off.

The electrical submersible pump system (Figure 2b) consists of a multi-stage downhole centrifugal pump, a downhole motor, a seal section (also called a protector) between the pump and motor, and electric cable extending from the motor to the surface electricity supply.

Both types of downhole pumps have been used for many years for cold water pumping and more recently in geothermal wells (lineshafts have been used on the Oregon Institute of Technology campus in 192ºF [89ºC] water for 36 years). If a lineshaft pump is used, special allowances must be made for the thermal expansion of various components and for oil lubrication of the bearings. The lineshaft pumps are preferred over the submersible pump in conventional geothermal applications for two main reasons: the lineshaft pump cost less, and it has a proven track record. However, for setting depths exceeding about 800 ft (250 m), a submersible pump is required.

PIPING

The fluid state in transmission lines of direct-use projects can be liquid water, steam vapor or a two-phase mixture. These pipelines carry fluids from the wellhead to either a site of application, or a steam-water separator. Thermal expansion of pipelines heated rapidly from ambient to geothermal fluid temperatures (which could vary from 120 to 400ºF [50 to 200&#176C]) causes stress that must be accommodated by careful engineering design.

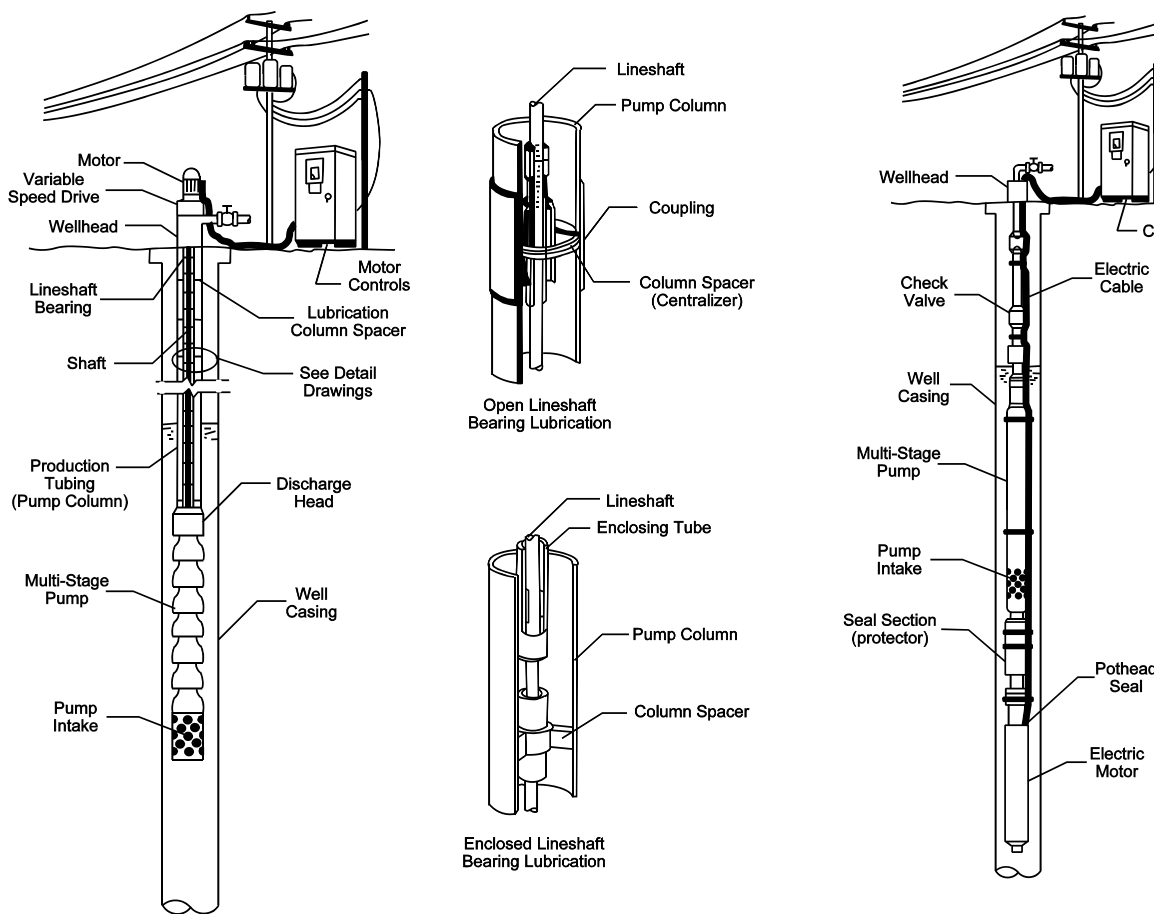

Figure 2. Downhole pumps: (a) lineshaft pump details, and (b) submersible pump details.

The cost of transmission lines and the distribution networks in direct-use projects is significant. This is especially true when the geothermal resources is located at great distance from the main load center; however, transmission distances of up to 37 miles (60 km) have proven economical for hot water (i.e., the Akranes Project in Iceland - Georgsson, et al., 1981), where asbestos cement covered with earth has been successful (see Figure 4 later).

Carbon steel is now the most widely used material for geothermal transmission lines and distribution networks, especially if the fluid temperature is over 212°F (100°C). Other common types of piping material are fiberglass reinforced plastic (FRP) and asbestos cement (AC). The latter material, used widely in the past, cannot be used in many systems today due to environmental concerns; thus, it is no longer available in many locations. Polyvinyl chloride (PVC) piping is often used for the distribution network, and for uninsulated waste disposal lines where temperatures are well below 212°F (100°C). Conventional steel piping requires expansion provisions, either bellows arrangements or by loops. A typical piping installation would have fixed points and expansion points about every 300 ft (100 m). In addition, the piping would have to be placed on rollers or slip plates between points. When hot water pipelines are buried, they can be subjected to external corrosion from groundwater and electrolysis. They must be protected by coatings and wrappings. Concrete tunnels or trenches have been used to protect steel pipes in many geothermal district heating systems. Although expensive (generally over $100 per ft ($300/m), tunnels and trenches have the advantage of easing future expansion, providing access for maintenance, and a corridor for other utilities such as domestic water, waste water, electrical cables, phone lines, etc.

Supply and distribution systems can consist of either a single-pipe or a two-pipe system. The single-pipe is a once-through system where the fluid is disposed of after use. This distribution system is generally preferred when the geothermal energy is abundant and the water is pure enough to be circulated through the distribution system. In a two-pipe system, the fluid is recirculated so the fluid and residual heat are conserved. A two-pipe system must be used when mixing of spent fluids is called for, and when the spent cold fluids need to be injected into the reservoir. Two-pipe distribution systems cost typically 20 to 30 percent more than single-piped systems.

The quantity of thermal insulation of transmission lines and distribution networks will depend on many factors. In addition to minimize the heat loss of the fluid, the insulation must be waterproof and water tight. Moisture can destroy the value of any thermal insulation, and cause rapid external corrosion. Aboveground and overhead pipeline installations can be considered in special cases. Considerable insulation is achieved by burying hot water pipelines. For example, burying bare steel pipe results in a reduction in heat loss of about one-third as compared to aboveground in still air. If the soil around the buried pipe can be kept dry, then the insulation value can be retained. Carbon steel piping can be insulated with polyurethane foam, rock wool or fiberglass. Below ground, such pipes should be protected with polyvinyl (PVC) jacket; aboveground aluminum can be used. Generally 1 to 4 inches (2.5 to 10 cm) of insulation is adequate. In two-pipe systems, the supply and return lines are usually insulated; whereas, in single-pass systems, only the supply line is insulated.

At flowing conditions, the temperature loss in insulated pipelines is in the range of 0.3 to 3°F/mile (0.1 to 1°C/km), and in uninsulated lines, the loss is 6 to 15°F/mile (2 to 5°C/km) in the approximate range of 80 to 240 gpm flow for a 6-in. diameter pipe (5 to 15 L/s for a 15-cm pipe)(Ryan, 1981). It is less for larger diameter pipes and for higher flows. As an example, less than 3°F (2°C) loss is experienced in the new aboveground 18-mile (29-km) long and 31- and 35-in. (80 - and 90-cm) wide line with 4 inches (10 cm) of rock wool insulation that runs from Nesjavellir to Reykjavik in Iceland. The flow rate is around 8,900 gpm (560 L/s) and takes seven hours to cover the distance. Uninsulated pipe costs about half of insulated pipe and thus, is used where temperature loss is not critical. Pipe material does not have a significant effect on heat loss; however, the flow rate does. At low flow rates (off peak), the heat loss is higher than at greater flows. Figure 3 (Gudmudsson and Lund, 1985) shows fluid temperature as function of distance, in a 18-in. (45-cm) diameter pipeline, insulated with 2 inches (5 cm) of urethane.

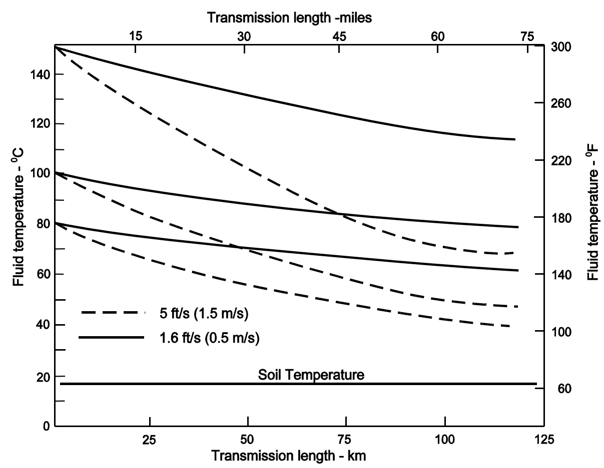

Figure 3. Temperature drop in hot water transmission line.

Several examples of aboveground and buried pipeline installations are shown in Figure 4.

Steel piping is shown in most case; but, FRP or PVC can be used in low-temperature applications. Aboveground pipelines have been used extensively in Iceland, where excavation in lava rock is expensive and difficult; however, in the USA, below ground installations are most common to protect the line from vandalism and to eliminate traffic barriers. A detailed discussion of these various installations can be found in Gudmundsson and Lund (1985).

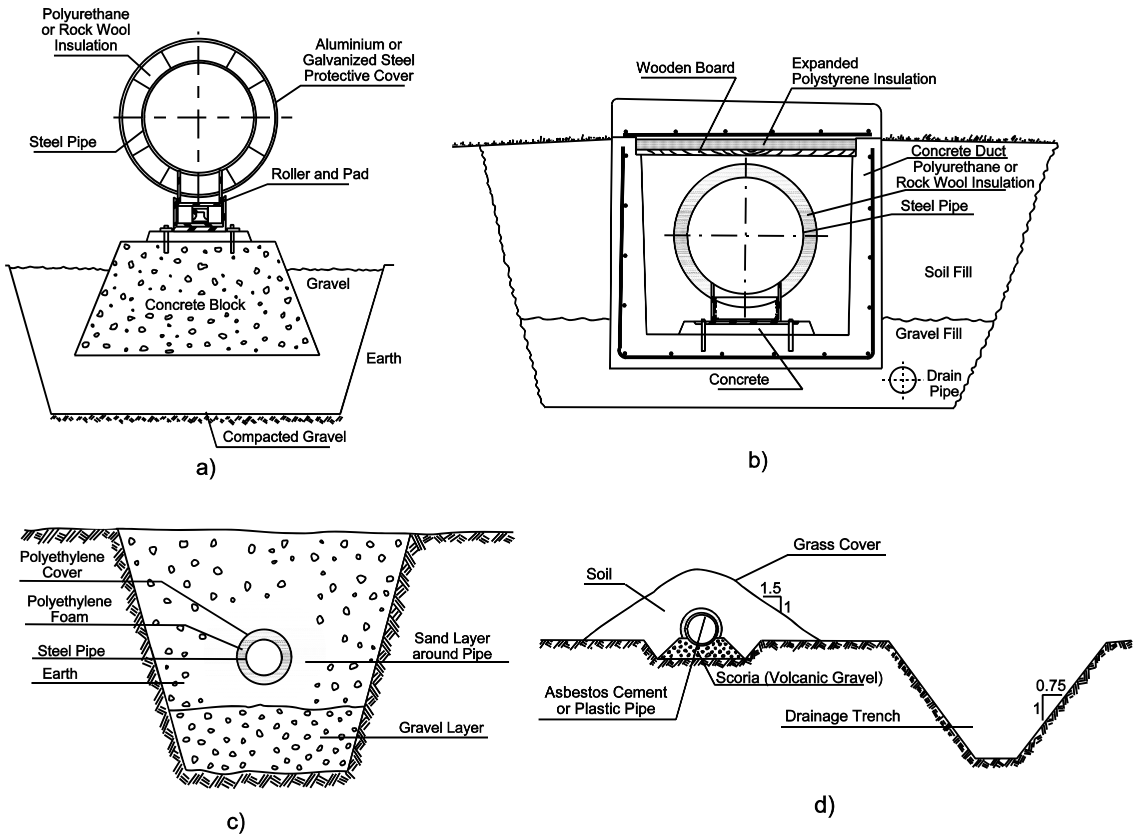

Figure 4. Examples of above and below ground pipelines: a) aboveground pipeline with sheet metal cover, b) steel pipe in concrete tunnel, c) steel pipe with polyurethane insulation and polyethylene cover, and d) asbestos cement pipe with earth and grass cover.

HEAT EXCHANGERS

The principal heat exchangers used in geothermal systems are the plate, shell-and-tube, and downhole types. The plate heat exchanger consists of a series of plates with gaskets held in a frame by clamping rods (Figure 5). The counter-current flow and high turbulence achieved in plate heat exchangers provide for efficient thermal exchange in a small volume. In addition, they have the advantage when compared to shell-and-tube exchangers, of occupying less space, can easily be expanded when additional load is added, and cost about 40% less. The plates are usually made of stainless steel; although, titanium is used when the fluids are especially corrosive. Plate heat exchangers are commonly used in geothermal heating situations worldwide.

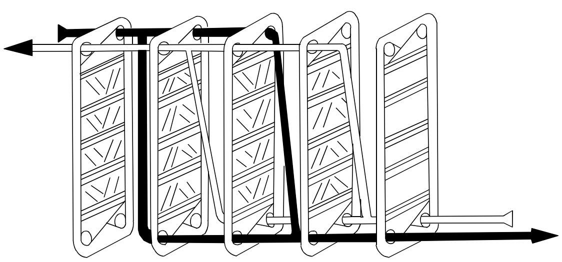

Figure 5. Plate heat exchanger.

Shell-and-tube heat exchangers may be used for geothermal applications, but are less popular due to problems with fouling, greater approach temperature (difference between incoming and outgoing fluid temperature), and the larger size.

Downhole be heat exchangers eliminate the problem of disposal of geothermal fluid, since only heat is taken from the well. However, their use is limited to small heating loads such as the heating of individual homes, a small apartment house or business. The exchanger consists of a system of pipes or tubes suspended in the well through which secondary water is pumped or allowed to circulate by natural convection (Figure 6). In order to obtain maximum output, the well must be designed to have an open annulus between the wellbore and casing, and perforations above and below the heat exchanger surface. Natural convection circulates the water down inside the casing, through the lower perforations, up in the annulus, and back inside the casing through the upper perforations (Culver and Reistad, 1978). The use of a separate pipe or promotor has proven successful in older wells in New Zealand to increase the vertical circulation (Dunstall and Freeston, 1990).

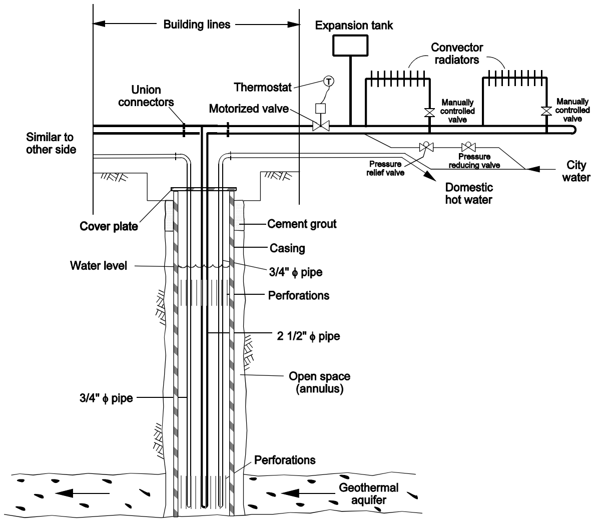

Figure 6. Downhole heat exchanger (typical of Klamath Falls, OR).

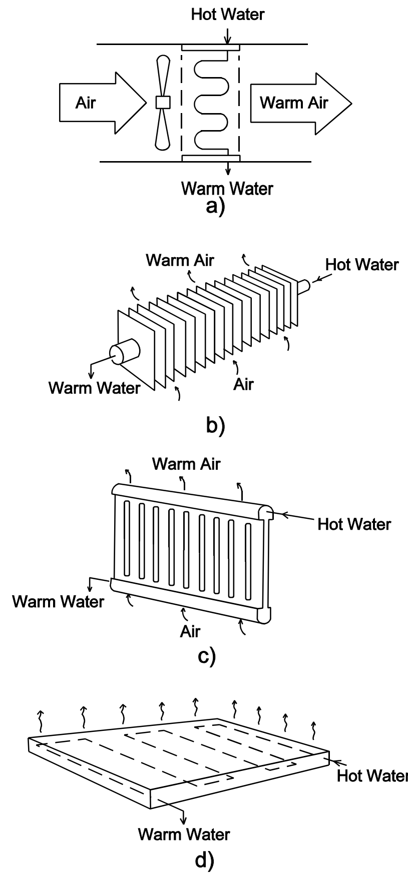

Figure 7. Convectors: a) forced air, b) material convection (finned tube), c) natural convection (radiator), and d) floor panel.

CONVECTORS

Heating of individual rooms and buildings is achieved by passing geothermal water (or a heated secondary fluid) through heat convectors (or emitters) located in each room. The method is similar to that used in conventional space heating systems. Three major types of heat convectors are used for space heating: 1) forced air, 2) natural air flow using hot water or finned tube radiators, and 3) radiant panels (Figure 7). All three can be adapted directly to geothermal energy or converted by retrofitting existing systems.

REFRIGERATION

Cooling can be accomplished from geothermal energy using lithium bromide and ammonia absorption refrigeration systems (Rafferty, 1983). The lithium bromide system is the most common because it uses water as the refrigerant. However, it is limited to cooling above the freezing point of water. The major application of lithium bromide units is for the supply of chilled water for space and process cooling. They may be either one-or two-stage units. The two-stage units require higher temperature (about 320°F - 160°C); but they also have high efficiency. The single-stage units can be driven with hot water at temperatures as low as 170°F (77°C)(such as at Oregon Institute of Technology). The lower the temperature of the geothermal water, the higher the flow rate required and the lower the efficiency. Generally, a condensing (cooling) tower is required, which will add to the cost and space requirements.

For geothermally-driven refrigeration below the freezing point of water, the ammonia absorption system must be considered. However, these systems are normally applied in very large capacities and have seen limited use. For the lower temperature refrigeration, the driving temperature must be at or above about 250°F (120°C) for a reasonable performance. Figure 8 illustrates how the geothermal absorption process works.

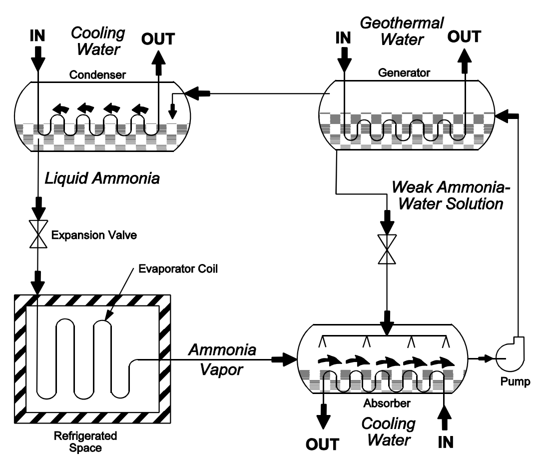

Figure 8. Geothermal absorption refrigeration cycle.

METERING (K. Rafferty)

For district heating systems (where heat is distributed to a large number of buildings from a central source), some means of energy use measurement is necessary to accommodate customer billing. Several approaches are available to accomplish this; but, the three most common approaches are: energy metering, volume metering and flat rate.

Energy (sometimes called Btu) metering involves the measurement of the water flow rate and the temperatures of the water entering the building (supply) and the water leaving the building (return). From the three values, the rate of energy use (Btu/min) can be calculated. Integrating these values over a longer period (a month) results in a value that can be used for customer billing. Energy metering requires a water flow meter, two temperature sensors and an electronic "integrator" to make the calculations (Figure 9). It provides the most accurate method of energy measurement, but at a cost much higher than the other methods. The cost of installing an energy meter in a small commercial customer would be in the range of $1000 to $1500 for moderate quality components.

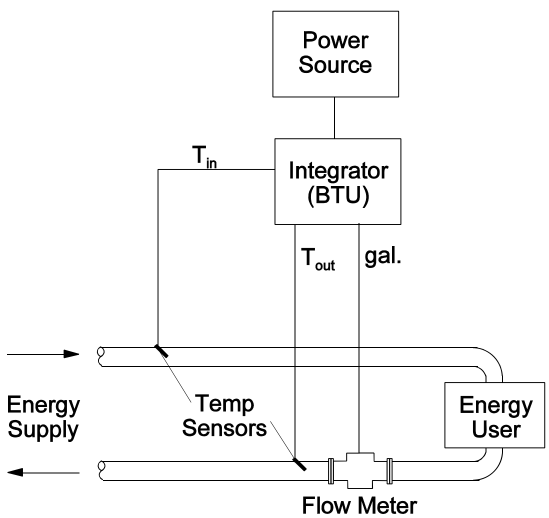

Figure 9. Energy metering.

Volume metering involves the measurement of only the water flow very much the same as in municipal water system operations. The volume of water over the period (gallons per month for example) is read from the meter and the customer's energy use is determined by multiplying the water volume used by an assumed heat content per volume (Btu per gallon for example). The equipment to accomplish this consists only of a water meter suitable for use in hot water. The cost of this for a small commercial customer would be in the range of $300. Because the customer's cost is determined only by the volume of water used, and the energy content of a given volume is directly related to the temperature difference, it is in his best interest to design and operate his system in such a way as to achieve a high temperature difference between the supply and return. This is also important to the district system operator since the capacity of any system is related to temperature difference.

Flat rate is the least sophisticated of the methods for customer billing. It simply consists of an agreement between the customer and the system operator that a flat sum ($/month or $/year) will be paid for the hot water service provided. In most systems that use a flat rate, there is some mechanical device installed to limit flow to the customer or regulate temperature. One of the primary advantages of flat rate is simpler marketing. There is no question in the customer's mind as to the savings, meter accuracy of impact of his current system efficiency. This approach works well with existing, small, simple customers for which there is a history of previous heating energy use.

In states where district heating is considered a regulated utility, the Public Utility Commission may have specific requirements for customer metering.

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WELL PUMPS

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PUMPING GEOTHERMAL FLUIDS

Introduction

Pumping is often necessary in order to bring geothermal fluid to the surface. For direct-use applications, there are primarily two types of production well pumps; (a) lineshaft turbine pumps and (b) submersible pumps - the difference being the location of the driver. In a lineshaft pump, the driver, usually a vertical shaft electric motor, is mounted above the wellhead and drives the pump, which may be located as much as 2,000 ft below the ground surface, by means of a lineshaft. In a submersible pump, the driver (a long, small diameter electric motor) is usually located below the pump itself and drives the pump through a relatively short shaft with a seal section to protect the motor from the well fluid.

Lineshaft pumps have two definite limitations: (a) they must be installed in relatively straight wells and (b) they are economically limited to settings of #2000 ft. For direct heat applications, the economic setting depth limit is probably closer to 800 ft. A general comparison of lineshaft and submersible pumps appears below in Table 1.

Table 1. Comparison of Lineshaft and Submersible Pumps

Lineshaft	Submersible
Pump stage efficiencies of 68 to 78%. Lower head/stage and flow/unit diameter. Higher motor efficiency. Little loss in power cable. Mechanical losses in shaft bearings.	Pump stage efficiencies of 68 to 78%. Generally, higher flow/unit diameter. Lower motor efficiency--operates in oil at elevated temperature. Higher losses in power cable. Cable at least partially submerged and attached to hot tubing.
Motor, thrust bearing and seal accessible at surface.	Motor, thrust bearings, seal, and power cable in well--less accessible.
Usually lower speed (1,750 rpm or less). Usually lower wear rate.	Usually higher speeds (3,600 rpm). Usually higher wear rate.
Higher temperature capability, up to 400°F+.	Lower temperature capability but sufficient for most direct heat and some binary power applications, assuming the use of special high-temperature motors.
Shallower settings, 2,000 ft maximum.	Deeper settings. Up to 12,000 ft in oil wells.
Longer installation and pump pull time.	Less installation and pump pull time.
Well must be relatively straight or oversized to accommodate stiff pump and column.	Can be installed in crooked wells up to 4 degrees deviation per 100 ft. Up to 75 degrees off vertical. If it can be cased, it can be pumped.
Impeller position must be adjusted at initial startup.	Impeller position set.
Generally lower purchase price at direct use temperatures and depths	Generally higher purchase price at direct use temperatures and depths.

In some installations, selection of a pump type will be dictated by setting depth, well size, well deviation, or temperature. If not restricted by these, the engineer or developer should select a pump based on lowest life cycle costs, including important factors such as expected life, repair costs, availability of parts, and downtime costs. Power consumption costs and wire-to-water efficiency, although certainly worth evaluating, may not be nearly as important as others factors, such as those above. For most direct heat applications, the lineshaft pump has been the preferred selection.

There are many factors that can affect the relative efficiencies of lineshaft versus submersible pumps: i.e. temperature, power cable length, specific design of impeller and bowl, column length and friction losses. The wire-to-water efficiency in the particular application is the import-ant factor. The bowl efficiency of a pump with extra lateral will be less than for standard lateral (discussed in the subsection on Relative Elongation) and clearances. The bowl efficiency of a submersible will be higher than a line-shaft of similar design because extra lateral is not required in the submersible. Motor efficiency generally favors the lineshaft design.

Lineshaft Turbine Pump

To understand the potential problems and solutions in lineshaft pumping, it is necessary to understand how the pumps are constructed. Figure 1 shows a typical lineshaft turbine pump with an enclosed oil-lubricated shaft. Enclosed shaft water lubricated pumps are also manufactured. The discharge head supports the column and shaft enclosing tube which, in turn, supports the multi-stage pump bowls and intake arrangement. The column is usually in 20 ft sections with either screwed or flanged connections. A shaft enclosing tube support "spider" is provided at intervals along the column. The enclosing tube is usually in 5 ft sections with a shaft bearing at each joint, although high speed pumps may have closer spacing. The lineshaft sections are the same length as the corresponding column. The enclosing tube is connected at the top of the bowl assembly to the discharge bowl where lubricating oil outlet ports are located. At the surface, it is connected to the discharge head with a tube tensioning assembly. The en-closing tube is tensioned after installation to help maintain bearing alignment. The enclosing tube provides a water-proof enclosure for the shaft and a path for gravity feed or pressure lubrication.

In an enclosed lineshaft oil lubricated pump, only the shaft bearings are oil lubricated. The pump shaft bearings (in the bowls between each impeller) are water lubricated. The oil is discharged into the well fluid outside the pump through the pump discharge case.

Open lineshaft pumps have seen limited success in geothermal applications. Most successful applications have been characterized by very high static water levels or flowing artesian conditions. Because the bearings are lubricated by geothermal hot water, bearings tend to heat and wear faster. Many of the more common bearing

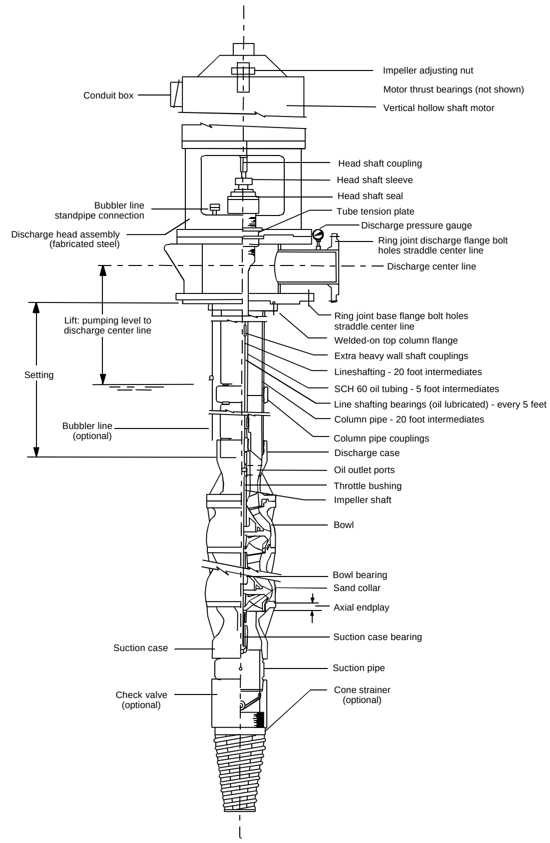

Figure 1. Typical lineshaft turbine pump with an enclosed oil-lubricated shaft.

materials are subject to corrosion or de-alloying by geothermal water and special bearing materials increase costs. If an open lineshaft design is used, the shaft should be of stainless steel to resist corrosion, again at a higher cost. As a result of the added costs for special materials and, likely shorter service life, the enclosed shaft design is preferred except for very clean, relatively cool (<140ºF) fluid.

The pump impellers are connected to the shaft by a collet or collet and key with locking screws. The shaft and all rotating parts are supported by the thrust bearings of the hollow shaft motor or a separate thrust bearing assembly. There is an impeller adjusting nut at the top in hollow shaft motor assemblies, or a coupling with adjusting nut for solid shaft driver arrangements.

When a vertical turbine pump stops, water flowing back down the column causes the pump to back spin. Because the pump is acting as the driver, there is very little danger of unscrewing shafting, but if the pump is started during back spin, it is likely to break shafting, loosen collets, or damage the motor. This could occur during momentary power failure or when a control signals a pump to start before the column fully drains. Foot valves, non-reversing ratchets, time delay switches, and rotation sensing switches can prevent this. Of these, non-reversing ratchet and time delays are the most common.

There are some advantages in allowing back spin. The free back spin indicates that nothing is dragging or binding and gives an indication of bearing conditions. It also permits the pump to be started with low load, reducing shock loads on shafting and bearings. A non-reversing ratchet also permits the column to drain, but it takes more time because the water flows backward through the bowls and impellers that are not rotating.

Foot valves prevent back spin and keep the column full of water, reducing the entrance of air and associated corrosion and scaling. They are, however, difficult to maintain in good condition because of scaling and corrosion properties of many geothermal fluids. Also, the pump always starts under a high load condition. Foot valves are recommended only for pumping levels <50 ft and when exclusion of air is mandatory.

Relative Elongation and Axial End Play or Lateral

A vertical turbine pump can be thought of as two concentric systems. The outer system consists of the column, impeller housings (bowls) and shaft enclosing tube. The inner system consists of the shaft and impellers. Forces resulting from dead weight, hydraulic thrust and thermal expansion result in different changes in length of these two systems. If not adequately allowed for in the design and operation of the pump, interference can occur resulting in damage to the pump.

In geothermal applications, an additional consideration is thermal expansion. Because of their differences in thickness, material and mass, the column, shaft enclosing tube, and shaft will all expand at different rates and reach thermal equilibrium at different times after initial startup. Additionally, the shaft in an enclosed lineshaft pump is somewhat thermally isolated from the water in the column by the space between the shaft and the inside diameter of the tube. Once thermal equilibrium is reached, thermal expansion has no direct affect on relative shaft elongation, but it must be compensated for as it occurs, either by adjusting the impellers or by allowing extra lateral. Obviously, in a system that cycles on and off, it must be allowed for in extra lateral.

Axial end play or lateral is accommodated through the vertical seal betwen the impeller and the bowl (shown in Figure 2). This is a kind of extended skirt on the bottom of the impeller and matching bore in the lower end of each bowl. These areas may have wear rings on the bowls, impeller or both. Standard cold water axial end play typically varies from 3/16 in. in a 4-in. diameter pump to 1-3/8 in. in a 30-in. diameter high head/stage pump. Corresponding maximum axial end play using standard castings is 1/4 to 1-3/4 in. This is obtained by additional matching of the bowls. Thermal expansion alone for a 400-ft static water level, 20°F well could be 4-3/4 in., which far surpasses the maximum axial end play for standard pumps. This illustrates why standard pumps are sometimes unsuitable for geothermal service, especially in a cycling situation. Failure to consider this has led to premature wear of impellers, bowls and bearings, broken lineshafts, and burned out electric motors. Proper end play and lineshaft sizing requires experience and understanding of relative shaft stretch, and knowledge of the range of operation on the lead versus flow curves.

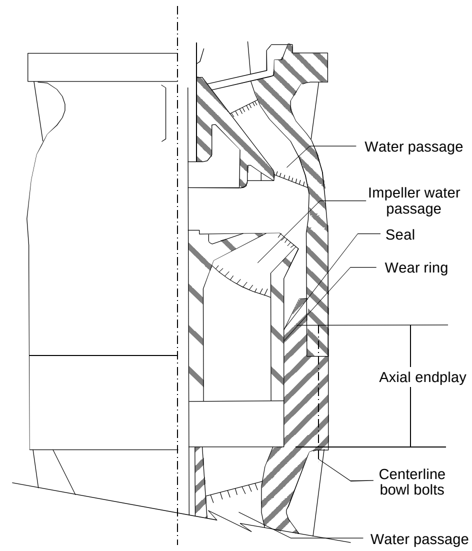

Figure 2. Cross-sectional of a pump bowl (Johnston Pump Company).

Extra lateral is accomplished by modifying the patterns from which the impeller and bowl castings are made and approximately matching the increased length. This entails extra cost and some manufacturers may not offer this option.

There is no real temperature break point for lineshaft pumps. For many applications up to 140°F, standard pumps, perhaps with machining up to maximum axial end play, will operate satisfactorily, particularly where the pump is operated continuously. For intermittent operation, thermal expansion and relative shaft elongation should be carefully checked.

A regular maintenance schedule is highly recommended. This includes lubrication of motor bearings (and thrust bearing if separate) and pump packing glands at specified intervals. Oil for shaft lubrication is usually gravity flow with a valve and sight glass to check the required number of drops per minute. This should be checked daily. Turbine oil 68 is the normally recommended lubricant.

Pump manufacturers can suggest a reasonable inspection frequency. It is usually more economic to pull a pump, inspect it and repair or replace parts as needed in the off season than to wait until it fails.

Submersible Pumps

A submersible pump is one in which the driver, or electric motor, is located in the well below the surface of the fluid being pumped and is usually below the pump itself. Submersible pumps, therefore, do not have the problems related to relative shaft elongation that lineshaft pumps do.

Submersible pumps can be separated into low temperature or standard pumps and high temperature pumps. The temperature limit is set primarily by the allowable temperature of the motor.

Low-Temperature Submersibles

Almost without exception, standard submersible pump motors are warranted to 90ºF or below. The allowable temperature is limited by the motor winding insulation and the heat dissipation available. Many standard submersible pump motors can be operated at 120 to 130ºF if proper allowances are made.

There are three basic types of submersible pump motors: wet winding, oil filled, and hermetically sealed.

In the wet winding motor, the motor is filled with water. Water proofing is achieved by special insulation on the stator winding wire, usually plastic, and because the wire and its insulation are bulkier, the motors are larger for a given rating. The motor is carefully filled at the surface to ensure there are no air bubbles and a filter installed in the fill port to ensure that the motor operates in clean water. Some brands are pre-filled and have an expansion diaphragm to allow for expansion and contraction of the filling solution and motor. Rotating seals and a sand slinger at the upper end prevent free circulation of well fluid in and out of the motor and reduce seal and spline wear by abrasive particles. Bearings are water lubricated.

Oil filled motors are prefilled with a dielectric oil. A rotating shaft seal (with sand slinger) is utilized to keep the oil in and water out. Because water has a higher density than oil, the motors have an oil reservoir with expansion bladder at the bottom. Any water that leaks through the seal in time migrates to the bottom of the reservoir. However, if the seal leaks there is probably always a small amount of water mixed with the oil surrounding the windings. Bearings are oil lubricated giving them higher capacity.

Hermetically sealed motors have the winding encased in a welded can, usually stainless steel. The windings may be similar to a surface motor with air inside the can but usually are embedded in a thermo-setting resin to provide better heat dissipation and reduce the possibility of water leaking in. The rest of the motor is similar to the wet type described above with the bearings water lubricated.

All small submersible motors have a thrust bearing at the lower end to carry pump downthrust and a small thrust bearing at the upper end to carry the momentary upthrust during pump startup. Some larger motors intended primarily for deep settings have a separate seal section providing for sealing and expansion. The seal section is located between the motor and the pump and contains the main thrust bearings.

High-Temperature Submersibles

High-temperature submersible pumps were developed for deep settings in oil fields. They are almost universally rated in barrels per day (bpd) rather than gallons per minute (gpm = bpd/34.3). For elevated temper-atures in both geothermal and oil fields, better elastomers for seals, higher temperature insulating materials for cable, and improved oils and bearings have been developed. Satisfactory operation has been attained in oil wells up to 290ºF. Figure 3 shows a submersible installa-tion. The gas separator shown is primarily used in oil field production. The function of the separator is to remove free gas from the fluid before it enters the pump where it would expand in the low-pressure suction area, possibly cause cavitation, and prevent proper pump operation.

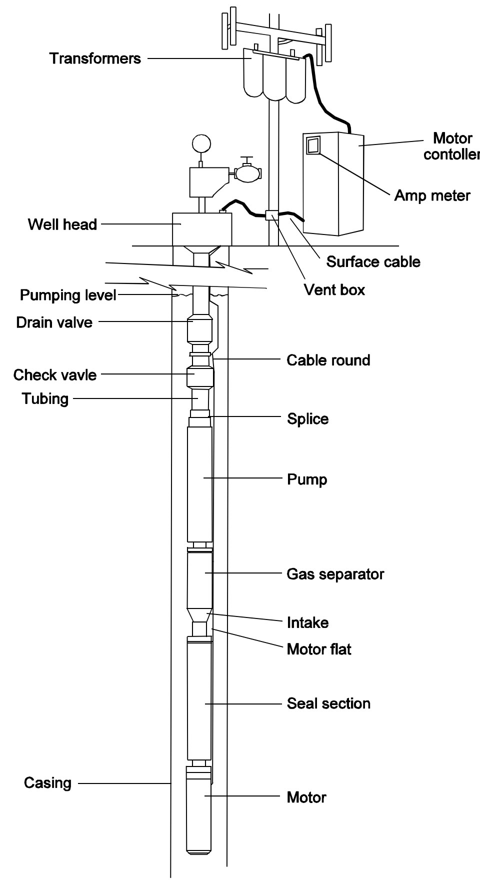

Figure 3. Submersible pump installation (Centrilift Hughes).

The pump section of a submersible is similar to a lineshaft in that it is a multi-stage centrifugal. Pump rpm is usually 3,475, which is higher than most lineshafts. Impellers are usually of the balanced or floating type to offset hydraulic thrust, because space for thrust bearings is limited

The seal section between the pump and motor provides for equalization of well fluid and internal motor pressure, allows for expansion and contraction of dielectric motor oil, provides a seal between the well fluid and motor oil and houses the thrust bearings. Separation of the well fluid and motor oil is accomplished by two or more mechanical shaft seals, elastomer expansion chamber and backup labyrinth.

Impellers are designed for balancing at peak efficiency. Operation at higher than design capacity results in upthrust, and lower than design capacity results in downthrust. Bearings are usually of the multiple tilting pad type; there are two, one for upthrust and one for downthrust.

Motors used in high-temperature submersibles are oil-filled, two-pole, three-phase, squirrel cage, induction type. Design voltages range from 230 to 5000 V.

In deep setting applications, motors are run at high voltages in order to reduce current flow. Voltages often are not the common values used in aboveground motors. In deep settings, there can be significant voltage drops in the downhole power cable. Submersibles, therefore, can require special above ground equipment, transformers and controllers, which are supplied by the manufacturers to match existing conditions.

Motors are built in 3-1/2 in. to 7-1/2 in. outside diameters to fit inside standard American Petroleum Institute (API) casing sizes. Rotors are generally 12 to 24 in. long and hp is increased by adding rotors. Single-motor lengths may reach 30 ft producing 400 hp and tandem motors 90 ft producing 750 hp. Motors have bearings designed to carry the rotor loads but not any additional pump loads.

Motor cooling is critical, and at least 1 ft/s flow past the motor is recommended. Flow inducer sleeves can increase flow velocity as described above for standard submersibles, and centralizers are often used to ensure even flows completely around the motors. Centralizers are required in deviated wells.

The cable providing electrical connection between the pump and surface is an important part of a submersible system. The cable is connected to the motor at a waterproof pothead that is usually a plug in type. Waterproof integrity is essential, and special EPDM elastomers are used for sealing. Pothead leaks were a continuing source of trouble in early submersibles for geothermal use, but the new designs have somewhat alleviated the problems. A flat motor lead extension cable is usually installed from the pothead to above the pumps. A cable guard is installed over the cable along the seal and pump sections to prevent mechanical damage during installation and removal. Either round or flat cable is spliced above the pump and run to the surface through the wellhead and to a junction box. Cable is available for several operating temperatures. Up to 180 to 200°F polypropylene insulation with nitrile jacket is used. At temperatures above 200°F, insulation and jacket are EPDM. Various configurations with or without tape and braid and lead sheathing are available for temperatures up to 450°F. Most cable has an interlocking armor of galvanized steel or monel. Galvanized steel will have a very short life in most geothermal fluids. Monel metals generally have longer expected life depending on the alloy and amount of hydrogen sulfide (H_2S) present.

Because all the submersible equipment is in the well, there is no maintenance that can be performed except scheduled pulling and inspection. Large submersibles may be equipped with recording ammeters that can help determine causes of failures and give an indication of pump and well performance. Pump wear, for instance, is indicated by decreasing motor output and current draw.

Excessive current in one or more legs might indicate motor or cable problems. If recording ammeters are installed, they should be checked regularly and the records analyzed.

VARIABLE-SPEED DRIVES FOR GEOTHERMAL APPLICATION

Introduction

Energy costs associated with the operation of production well pumps constitute a large expense for many geothermal systems. In direct use systems, particularly those serving predominantly space heating loads, there is a wide variation in flow requirements. As a result, an efficient means of controlling flow should be an integral part of these systems.

Because most systems utilize centrifugal lineshaft-driven or submersible well pumps, there are three methods available for controlling flow:

1. Throttling pump output
2. Varying the speed of the pump
3. Intermittent pump operation with storage tank.

Throttling the output of any fluid handling device is simply dissipating energy through the addition of friction. This is an inherently inefficient approach to flow control.

Intermittent pump operation can impose serious shock loads in the pumping system, particularly at bearings and impeller connections. This has, in several projects, led to pump failures. Storage tanks can serve as a point of entrance for oxygen, thus aggravating corrosion problems. The results of these combined effects has been unreasonably high maintenance costs.

Use of variable speed drives can significantly increase pump life. A slow speed pump will outlive a faster pump with identical installations and pump construction. The wear rate is proportional to somewhere between the square and cube of the speed ratio; as a result, a pump rotating twice as fast will wear at four to eight times the rate (Frost, 1988).

A review of the response of a basic pumping system suggests that pump speed control is a much more energy efficient approach to controlling flow rate. In a closed piping loop, flow varies directly with pump speed, pressure drop with the square of the pump speed and horsepower requirement with the cube of the pump speed.

One must realize that the above relationships are based upon a situation in which the pump head is composed entirely of friction head. In a geothermal system, much of the pump head may be composed of static head. Static head is, of course, independent of flow. As a result, for a pump operating against a 100% static head, the system response is one in which flow is directly related to speed, pressure drop is in-dependent of speed and horsepower varies directly with speed.

The savings to be achieved through speed control of a centrifugal fluid handling device under a 100% static head situation are then significantly less than the savings achieved in a 100% friction head situation over the same speed range. In addition, there is a limit imposed by a large static head upon the minimum pump speed. This minimum speed is a function of the ability of the pump to develop sufficient head to move the water out of the well.

Geothermal systems will fall somewhere between these two extremes (100% static head and 100% friction head) depending upon static level, drawdown and surface head requirements.

If the control strategy is based upon a constant wellhead pressure, the system very nearly approaches the 100% static head situation. In general, large surface pressure requirements (which vary with flow) relative to static head requirements tend to make speed controls more cost effective.

Most geothermal applications involve the use of a squirrel cage induction motor. The results in two basic approaches to pump speed control:

1. Motor oriented control
   a. Multi-speed motor
   b. Variable frequency drive (AC).

2. Shaft oriented control through the use of a fluid coupling.

The choice among the above techniques should consider: capital cost, duty cycle, hp, speed/torque relation-ship, efficiency, and maintenance requirement.

Conclusion

Among the various drive technologies available, the choice is a function of a host of project specific parameters. The information presented here, along with pump and well information from your project, should permit an accurate analysis to be carried out. The results of this analysis can then be employed in the decision process. Table 2 summarizes the various characteristics of the speed control techniques outlined herein.

LESSONS LEARNED

Listed below are a number of factors relating to pumps that can lead to premature failure of pumps and other components. Many of these have been noted or alluded to elsewhere, but are restated here. Some seem obvious, but the obvious is often overlooked (Culver, 1994).

1. Pump suppliers/manufacturers should be provided with complete data on all foreseen operating conditions and complete chemical analyses. Standard potable water analysis is not adequate, because they do not test for important constituents, such as dissolved gases.

2. In general, continuous or nearly continuous operation of well pumps is preferred. Short cycle start/stop operation should be avoided. This is particularly true for open lineshaft pumps. When the column drains, bearings and the inside of the column are exposed to oxygen, leading to corrosion.

Table 2. Summary of Speed Control Techniques

Method	Efficiency	Capital Cost	Maint. Required	Over Speed Capacity	Effect on Motor Life[e]	Turn Down	Auto Control	Size Range
Adjustable[a] Frequency (AC)	High	Moderate	Low	Y	Lowers	Inf.	Y	Franctional to several hundred
Fluid[b] Coupling	Moderate	High	Moderate	N	None	4:1	Y	5 - 10,000 hp
Multi-speed[c] Motors	Moderate	Low	Low	N	None	2:1	Y	Fractional to several hundred
Throttling[d]	Very low	Low	Low	N	None	No limit	Y	No limit

a. Allows motor operation in failure mode. Should use high-temperature rise motors. Minimum ambient temperature 50°F.
b. Poor efficiency at low output speeds.
c. Poor efficiety at low output speeds.
d. Stopped output speed in 2 or 4 increments, must throttle in between, possible problems with shaft and bearings.
e. Refers to older motors--depends on application.

Start/stop operations often necessitate a storage tank. This is often a source of air in-leakage. Parts per billion (ppb) of oxygen (O_2) in combination with ppb hydrogen sulfide (H_2S) can lead to early failure of copper and copper alloys, dezincification of brass and bronze and soldering alloys used in valves, fan coils, and piping.

As noted in Chapter 8, almost without exception, geothermal fluid contains some H_2S. If a start/stopmode of operation is used, air is drawn into the system when fluid drains down the column after the pump stops. This can cause a greatly accelerated rate of pitting corrosion in carbon steels, formation of cuprous sulfide films, and crevice corrosion of copper, brass and bronze (except leaded brass and bronze), de-alloying of lead/tin solders and dissolution of silver solder.

Start/stop operation imposes high shaft and coupling torque loads. It is believed this has led to early failure of lineshafts and lineshaft to motor couplings.

3. Records of pressure and flow versus rpm or power should be kept on a regular basis. Decreases in flow or pressure indicate something is wrong and is a portent of more drastic trouble that could occur later on.

4. Pumps should be pulled and inspected on a regular basis, based on experience or as recommended by the manufacturer.

5. Some minimum flow must be maintained in variable-speed applications. Relatively short periods of operation at shutoff will overheat pumps and motors.

6. Motors should be well ventilated. Although this seems obvious, several motors have been installed in below ground unventilated pits. With hot water piping in close proximity, the motor is near its upper operating temperature even when not in operation.

7. Packing glands should be well maintained. All above surface centrifugal pumps tend to in-leak air through packing glands, especially if starting at low suction pressure. Air in-leakage leads to corrosion. Leaks around lineshaft packing lead to corrosion/scaling of the shaft, making sealing progressively more difficult.

8. Enclosed lineshaft pumps require that lubricant (water or oil) be supplied before the pump is started. It has been observed that in installations where the lubricant flow started and stopped simultaneously with the pump motor, pumps failed prematurely.

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PIPING

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INTRODUCTION

The source of geothermal fluid for a direct use appli-cation is often located some distance away from the user. This requires a transmission pipeline to transport the geo-thermal fluid. Even in the absence of transmission line requirements, it is frequently advisable to employ other than standard piping materials for in-building or aboveground piping. Geothermal fluid for direct use applications is usually transported in the liquid phase and has some of the same design considerations as water distribution systems. Several factors including pipe material, dissolved chemical components, size, installation method, head loss and pumping requirements, temperature, insulation, pipe expan-sion and service taps should be considered before final specification.

In several installations, long transmission pipelines appear to be economically feasible. Geothermal fluids are being transported up to 38 miles in Iceland. In the U.S., distances greater than 5 miles, are generally considered uneconomical; however, the distance is dependent on the size of the heat load and the load factor.

Piping materials for geothermal heating systems have been of numerous types with great variation in cost and durability. Some of the materials which can be used in geothermal applications include: asbestos cement (AC), ductile iron (DI), slip-joint steel (STL-S), welded steel (STL-W), gasketed polyvinyl chloride (PVC-G), solvent welded PVC (PVC-S), chlorinated polyvinyl chloride (CPVC), polyethylene (PE), cross-linked polyethylene (PEX), mechanical joint fiberglass reinforced plastic (FRP-M), FRP epoxy adhesive joint-military (FRP-EM), FRP epoxy adhesive joint (FRP-E), FRP gasketed joint (FRP-S), and threaded joint FRP (FRP-T). The temperature and chemical quality of the geothermal fluids, in addition to cost, usually determines the type of pipeline material used. Figures 1 and 2 introduce the temperature limitations and relative costs of the materials covered in this article. Generally, the various pipe materials are more expensive the higher the temperature rating. Figure 2 includes 15% overhead and profit (O&P). Cost data in this article are based on Means 1996 Mechanical Cost Data.

Installation costs are very much a function of the type of joining method employed and the piping material. The light weight of most nonmetallic piping makes handling labor significantly less than that of steel and ductile iron in sizes greater than 3 in.

A recent report (Rafferty, 1996) evaluated some of the cost associated with geothermal distribution piping in the context of the applications in which it is often applied in the western U.S. The work involved characterizing the various components of the cost of installing distribution piping in developed areas and the potential for reducing these costs in an effort to serve single-family homes with geothermal district heating.

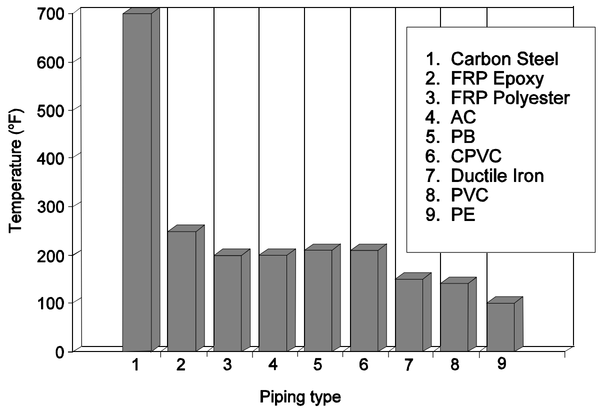

Figure 1 Maximum service temperature for pipe materials.

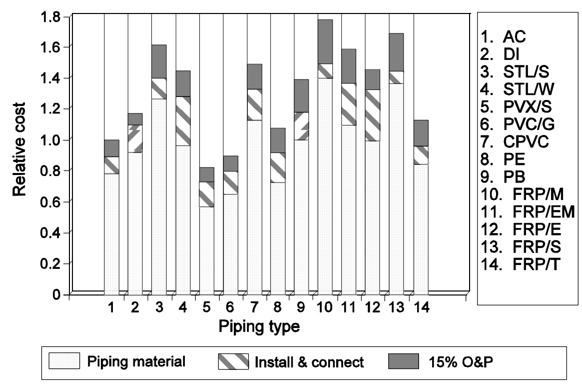

Figure 2 Relative cost of piping by type.

PIPE MATERIALS

Both metallic and nonmetallic piping can be considered for geothermal applications. Carbon steel is the most widely used metallic pipe and has an acceptable service life if properly applied. Ductile iron has seen limited application. Asbestos cement (AC) material has been the most widely applied product; however, enviornmental concerns have limited its use and availability. A discussion of piping material currently in use in U.S. district heating systems appears in Rafferty (1989).

The attractiveness of metallic piping is primarily related to its ability to handle high temperature fluids. In addition, its properties and installation requirements are familiar to most installation crews. The advantage of non-metallic materials is that they are virtually impervious to most chemicals found in geothermal fluids. However, the installation procedures, particularly for fiberglass and polyethylene are, in many cases, outside the experience of typical laborers and local code officials. This is particularly true in rural areas. The following sections review some specifics of each material and cover some problems encountered in existing geothermal systems.

Carbon Steel

Available in almost all areas, steel pipe is manufactured in sizes ranging from 1/4 to over 72 in. Steel is the material most familiar to pipe fitters and installation crews. The joining method for small sizes (<2-1/2 in.) is usually threading, with welding used for sizes above this level. For underground installations, all joints are typically welded when unlined piping is used. For epoxy-lined piping, some form of mechanical joint should be employed so that welding does not interfere with the integrity of the lining material. Commonly used steel pipe ratings are Schedule 40 (standard) and Schedule 80 (extra strong). In most cases, in the U.S., Schedule 40 piping is used for heating applications, although, in Europe and for some newer non-geothermal district systems in the U.S., lighter weights (approximately Schedule 20) are now used. Schedule 80 is employed for high pressure applications or in cases where higher than normal corrosion rates are expected.

Corrosion is a major concern with steel piping, particularly in geothermal applications. As mentioned above, some allowance can be made by using the thicker-walled Schedule 80 piping. However, this approach is valid only for uniform corrosion rates. In many geother-mal fluids, there are various concentrations of dissolved chemicals or gases that can result primarily in pitting or crevice corrosion. If the potential exists for this type of attack, or if the fluid has been exposed to the air before entering the system, carbon steel should be the material of last resort

Steel piping is used primarily on the clean loop side of the isolation heat exchanger, although in a few cases it has been employed as the geothermal transmission line material.

A distinct disadvantage in using steel pipe is that the buried pipe is also subject to external corrosion unless protected with a suitable wrapping or cathodic protection. The potential for external corrosion of metallic pipe systems should be considered for all direct buried installations. Various soil types, presence of groundwater, and induced current fields from power lines may accelerate external pipe corrosion and early system failure.

In at least two geothermal systems, unlined steel piping has performed well in normal operation but has suffered severe internal pitting corrosion during system shutdowns. In one case in which a system was down for approximately 6 months, carbon steel piping exhibited pitting corrosion rates of 70 to 200 mils/y (mpy). If unlined steel piping is employed on the geothermal side of the system, it is most critical to assure a complete internal drying of the material for extended shutdowns.

In both buried and aboveground installations, allowances for expansion must be made in the form of expansion joints or loops. These considerations have the effect of increasing both the labor and material costs of the piping system.

Galvanized steel has been employed with mixed success in geothermal applications. Some geothermal fluids have demonstrated the ability to leach zinc from solder and other alloys. Selective removal of the zinc from galvanized pipe could result in severe pitting corrosion. In addition, consideration should be given to the fact that the protective nature of the zinc coating is generally not effective above 135ºF.

An indication of the costs for steel piping is shown in Table 1.

Table 1. Steel Pipe Costs, Material Only

Size (in.)	Schedule 40 ($/lf)	Schedule 80 ($/lf)
2	3.46	4.73
4	10.16	14.12
6	17.71	34.76
8	43.39	65.56

Ductile Iron

Ductile iron is similar to cast iron with the exception of the form of the carbon component. In cast iron, the carbon (graphite) is in a flake-like structure. In ductile iron, the structure is more spherical or nodular. This small difference results in the greater strength, flexibility, and machinability from which the product derives its name. Ductile iron has been described as more corrosion resistant than cast iron. However, the slight difference in corrosion resistance would not be of any substantive meaning in most geothermal applications. Cast iron piping was employed for over 80 years in the Warm Springs geothermal system (Boise, ID).

As an iron material, ductile iron is susceptible to corrosion from both external and internal sources. External protection generally involves a moisture barrier. For a pre-insulated product, special moisture protection would only be required at the joints and other fittings.

Internal corrosion protection is usually provided by a lining. The two most common materials are cement mortar and coal tar epoxy. Coal tar epoxy is limited to a temperature of approximately 120ºF. Mortar lining, according to the Ductile Iron Pipe Producers Research Association, is suitable to a service temperature of 150ºF with a protective seal coat. Without the seal coat, maximum service tempera-ture is 212ºF. In some applications with very soft water, a leaching of the mortar lining has been observed when a seal coat is omitted. As a result, a special high temperature epoxy coating would be required. Unfortunately, quotes received by the San Bernardino Water District (operators of the San Bernardino, California, Geothermal District Heating

System) for linings of this type for 130ºF application would add $5.00 to $8.00/ lineal foot to the price of the pipe. In applications where water chemistry is such that bare cement lining is accept-able, ductile iron could be an economical piping choice.

Ductile iron is a much-thicker-walled product than standard carbon steel and, for uniform corrosion applica-tions, offers the probability of longer life. In geothermal applications, corrosion occurs by both uniform and pitting modes. Pitting corrosion rates of 70 to 200 mpy in carbon steel have been observed in one low-temperature (<150ºF) system during shutdown periods.

Ductile iron piping is cost competitive with asbestos cement material. In addition, its common use in water supply systems results in wider familiarity with its install-ation practices. However, ductile iron pipe is the heaviest material of those covered in this article. As a result, it would incur additional handing costs in comparison to the lighter weight materials. Table 2 outlines costs for ductile iron piping.

Table 2. Costs For - Ductile Iron Piping (Uninsulated - Tyton Joint)

Size (in.)	Cost ($/lf)
4	8.35
6	9.45
8	12.50
10	16.95
12	21.00

The most common method of joining ductile iron piping is through the use of a push-on or Tyton type joint. This is a bell and spigot gasketed joint. In addition, several versions of mechanical joints are available, although these are characterized by higher cost than the push-on joints.

It is important to specify gasket materials suitable for the application temperature when using this product. Most suppliers offer EPDM gaskets which are suitable for use to 200ºF.

Fiberglass (RTRP)

Fiberglass piping, commonly referred to as RTRP (reinforced thermosetting resin pipe) or FRP (fiberglass reinforced plastic), is available in a wide vareity of configurations. Two materials are epoxy resin and polyester resin. In addition, the piping is available in lined and unlined versions. The epoxy resin piping with an epoxy liner is generally selected for geothermal applications. Both epoxy resin and polyester resin systems can be compounded to be serviceable to temperatures of 300ºF. Regardless of the type of fiberglass material used, care must be taken to maintain operating pressure high enough to prevent flashing of hot fluids. At high temperatures (>boiling point), the RTRP systems are susceptible to damage when fluid flashes to vapor. The forces associated with the flashing may spall the fibers at the interior of the pipe surface.

Fiberglass piping is available from a number of manufacturers but, at the distributor and dealer level, it is considerably less common than steel. Most manufacturers produce sizes 2 in. and larger. As a result, if fiberglass is to be employed, another material would have to be used for branch and small diameter piping of <2 in.

As with all nonmetallic piping, the method of joining is a large consideration with respect to both installation time and expense. With FRP piping, a variety of methods are available, including mechanical (keyed, threaded and flanged) and adhesive type jointing. Of these, the bell and spigot/adhesive has seen the widest application in geothermal systems.

In making the choice between the mechanical and ad-hesive type of joining, consideration should include piping cost, fitting cost, contractor familiarity, and probable installation temperature.

The cost of the keyed joint piping is approximately 10% more than the bell and spigot/adhesive joint in the 6 in. size. Alternate versions of mechanical joining are some-what more expensive. The added cost of the keyed-type joint can be compensated for by the reduced labor necessary to complete the joint. Fitting cost should be carefully weighed with any mechanical joining system. If a large number of fittings are required, the labor savings can be quickly overshadowed by fitting material cost. In addition to the amount of labor required, the adhesive joint also demands a greater technical skill on the part of the installer. The epoxy adhesive must be properly mixed and applied to the joint under acceptable conditions to ensure a reliable set. One of the most important of these conditions is tempera-ture.

Below approximately 75ºF, curing time is substant-ially increased. As a result, if installation is to occur in a reasonable length of time, a special heating blanket must be applied to each joint after makeup to ensure proper curing. As with most other piping systems, the mechanical draw method is preferred for joint assembly.

Two recent developments which may be considera-tions are gasketed slip joint and integral thread joining. The slip joint approach provides for installation very similar to Tyton joint ductile iron or AC pressure pipe. Integral thread (with a double "O" ring) piping is also less labor intensive and low cost.

The axial expansion of FRP is approximately twice that of steel. However, because of the relatively low axial modulus, forces developed as a result of this expansion are only 3 to 5% that of steel under the same conditions. As a result, for buried installations with at least 3 ft of cover, sufficient restraint is provided by the overlying soil and no special precautions need be made for expansion other than adequate thrust blocking. For aboveground installations (on hangers), changes in direction are the most economical method of allowing for expansion.

Fittings are available from most manufacturers in a wide variety of configurations. In general, the bell and spigot/ epoxy joint system offers a greater number of fittings than the keyed joint system. In fact, it is likely that some field made adhesive joints will be required even if a keyed joint system is selected. Fittings are available to convert from the fiber-glass connections system to standard flange connections. Saddle fittings of fiberglass construction are available for service connections. Standard piping lengths are 20, 30, and 40 ft.

Cost for fiberglass piping systems are shown in Table 3. It should be noted that fitting costs can constitute a substantial portion of the total cost for a piping system.

Table 3. Cost for Fiberglass Piping (epoxy lined/ adhesive type joint)

		Fittings		
Size (in.)	Pipe ($/lf)	Ell ($/ea)	Tee ($/ea)	Joint Kit ($/ea)
2	6.70	38	53	11
3	9.21	45	63	14
4	11.37	97	81	17
6	17.76	150	217	21
8	28.64	215	250	25
10	38.79	260	400	28
12	49.34	345	435	31

Polyvinyl Chloride (PVC) and Chlorinated Polyvinyl Chloride (CPVC)

PVC is a low-temperature (maximum service temperature is 140ºF) rigid thermoplastic material. It is manufactured in 0.5 to over 12 in. in diameter and is, next to steel, the most commonly available piping material. Common ratings used for plumbing applications are Schedule 40 and Schedule 80. In most applications, the Schedule 40 would suffice. For higher temperature suspended applications, the Schedule 80 material would require slightly less support. The most common method of joining PVC is by solvent welding. Schedule 80 material can also be threaded. Most types of fittings and some valves are available in PVC up to approximately 12 in.

Table 4. Costs for PVC and CPVC Pipe and Fittings

Size (in.)	PVC Sch. 40 ($/lf)	CPVC Sch. 40 ($/lf)	90 Degree Ell PVC ($ ea)
2	1.42	4.27	2.45
3	2.08	8.22	5.25
4	2.68	11.06	9.40
6	4.68	22.36	30
8	7.70	--	77
10	15.04	--	--

CPVC is a higher temperature rated material with a maximum temperature rating of 210ºF. Pressure handling ability at this temperature is very low (as is PVC at its maximum temperature) and support requirements are almost continuous.

Costs for these piping materials are presented in Table 4. As a result of the high costs for CPVC, it has seen little application in geothermal systems.

Polyethylene (PE)

Polyethylene is in the same chemical family (poly-olefin) as polybutylene and is similar in physical characteristics. It is a flexible material available in a wide variety of sizes from 0.5 to 42 in. diameter. To date, this material has seen little application in direct-use geothermal systems, primarily because of its maximum service temperature of 140 to 150ºF. The piping is recommended only for gravity flow applications above this temperature. Very high molecular weight/high density PE can be employed for low pressure applications up to temperatures as high as 175ºF. The SDR (standard dimension ratio--a wall thickness description) requirements under these conditions, however, greatly reduce the cost advantages normally found in polyethylene. Use of the material in geothermal applications has been limited to small diameter (0.5 to 1 in.) tubing employed for bare tube heating systems in greenhouses and snow melting.

Some European district heating systems are using a cross-linked PE product for branch lines of 4 in. and under. This material is servicable to 194ºF at a pressure of approx-imately 85 psi. This product is currently available only in a pre-insulated configuration as discussed later in this article.

Joining is limited to thermal fusion of polyethylene pipe. The pressure ratings of polyethylene piping are a function of SDR and temperature. Costs for polyethylene piping are shown in Table 5.

Table 5. Costs for Polyethylene Pipe

Size (in.)	SDR	Cost ($/lf)
1/2	11	0.17
3/4	11	0.25
1	11	0.38
1-1/4	11	0.69
1-1/2	11	0.85
2	11	1.92
3	11	2.27
4	11	3.91
6	11	3.95
8	11	14.90

Copper

Copper piping, one of the most common materials in standard construction, is generally not acceptable for geo-thermal applications. Most resources contain very small quantities of hydrogen sulfide (H_2S), the dissolved gas that results in a rotten egg odor. This constituent is very aggres-sive toward copper and copper alloys. In addition, the solder used to join copper has also been subject to attack in even very low total dissolved solids (TDS) fluids. For these reasons, copper is not recommended for use in systems where it is exposed to the geothermal fluid.

Crosslinked Polyethylene (PEX)

Crosslinked polyethylene is a high-density poly-ethylene material in which the individual molecules are "crosslinked" during the production of the material. The affect of the crosslinking imparts physical qualities to the piping which allow it to meet the requirements of much higher temperature/pressure applications than standard polyethylene material. PEX piping carries a nominal rating of 100 psi @ 180°F.

Joining the piping is accomplished through the use of specially designed, conversion fittings which are generally of brass construction. Since the piping is designed primar-ily for use in hydronic radiant floor, sidewalk and street (snow melting) heating systems, a variety of specialty manifolds and control valves specific to these systems are available.

The tubing itself is available generally in sizes of 4 in. and less with the 3/4 in. and 1 in. diameter most com-mon. Piping with and without an oxygen diffusion barrier is available. The oxygen barrier prevents the diffusion of oxygen through the piping wall and into the water. This is a necessary corrosion prevention for closed systems in which ferrous materials are included.

Larger sizes of the PEX material are available as either bare or pre-insulated. The pre-insulated product is sold in rolls and includes a corrugated polyethylene jacket and a closed-cell polyethylene insulation. Rubber end caps are used to protect the exposed insulation at fittings. The flexible nature of the pre-insulated product offers an attrac-tive option for small-diameter distribution and customer service lines in applications where it is necessary to route the piping around existing utility obstacles.

Table 6 presents cost information for bare PEX piping. This information does not include fittings.

Table 6. PEX Piping Costs ($/ft)

Size	With O_2 Barrier	Without O_2 Barrier
3/4	1.65	1.35
1	3.30	2.50
1 1/4	4.25	3.10
1 ½	5.75	4.21
2	9.10	5.40
2 ½	--	7.65
3	--	10.60

PRE-INSULATED PIPING SYSTEMS

Most district heating systems or long transmission lines carrying warm geothermal fluid will require some form of insulation. This insulation can be provided by selected backfill methods, field applied insulation or, more commonly, a pre-insulated piping system.

The pre-insulated system consists of a carrier pipe, through which the fluid is transported, an insulation layer, and a jacket material.

There is a wide variety of combinations available in terms of jacket and carrier pipe materials. The only com-mon factor among most products is the use of polyurethane for the insulation layer. This insulation is generally foamed in place using a density of approximately 2 lb/ft^3 and a com-pressive strength of 25 psi. Thermal conductivity of the polyurethane varies, but a mean value of 0.18 Btu in./h ft^2 °F at 150°F is generally specified.

AC pre-insulated systems generally employ AC materials for both the carrier pipe and the jacket. Carrier piping is as described in the AC section above. The jacket material is usually a class 1500 sewer pipe product (ASTM C 428).

For steel, FRP, PB, PE, DUC, and PVC a variety of jacket materials are available. These include polyethylene, PVC, and fiberglass. The most common material is PVC. High impact type piping is employed for this service with a minimum thickness of 120 mil. Polyethylene jacketing material is commonly found on the European steel district heating lines and is generally a minimum of 125 mil. It is also used in corrugated form for the jacketing on pre-insulated PEX pipe. Fiberglass jacketing is used primarily with fiberglass and steel carrier material. Most jacketed systems (except fiberglass) employ a rubber end seal to protect the insulation from exposure to moisture. On fiberglass systems, the jacketing material is tapered at the end of each length to meet the carrier pipe, thereby forming a complete encasement of the insulation. Most systems employ a 1- to 2-in. insulation layer, with fittings often left uninsulated. Tables 7 and 8 presents cost data for selected examples of pre-insulated piping systems.

Table 7. Cost Data Pre-insulated Piping System

	Size			
Carrier Jacket	3 in. ($/lf)	4 in. ($/lf)	6 in. ($/lf)	8 in. ($/lf)
Steel/PVC	13.18	17.50	29.50	32.75
FRP/PVC (adhesive)	13.50	17.50	25.25	40.50
FRP/PVC (mechanical)	17.50	21.75	31.25	40.00
PVC/PVC (Schedule 40)	5.75	8.25	11.50	15.75
DUC/PVC	16.00	16.75	18.75	25.00

Table 8. Cost for Flexible Small Diameter Pre-insulated Tubing - PEX Carrier/PE Jacket

Size (in.)	Single Tube ($/lf)	Double Tube ($/lf)
3/4	18	25
1	21	31
1 1/4	27	39
1 ½	33	58
2	42	__
2 ½	55	__
3	60	__

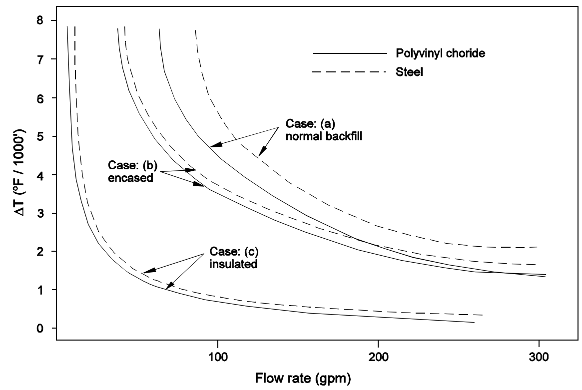

Figure 3. Buried pipeline temperature loss versus flow rate (Ryan, 1981).

UNINSULATED PIPING

High initial capital costs are one reason development has lagged in the area of district heating. Much of this cost (40 to 60%) is associated with the installation of the distribution piping network. The use of uninsulated piping for a portion of the distribution offers the prospect of reducing the piping material costs by more than 50%.

Although the uninsulated piping would have much higher heat loss than insulated lines, this could be compensated for by increasing system flow rates. The additional pumping costs to maintain these rates would be offset by reduced system capital costs. Preliminary analysis indicates that it would be most beneficial to use uninsulated lines in sizes above about 6 in. in certain applications.

It is important before discussing the specifics of uninsulated piping to draw a clear distinction between heat loss (measured in Btu/hr lf) and temperature loss (measured in °F/lf). Heat loss from a buried pipeline is driven largely by the temperature difference between water in the pipe and the ambient air or soil. The temperature loss which results from the heat loss is a function of the water flow in the line. As a result, for a line operating at a given temperature, the greater the flow rate the lower the temperature drop. In geothermal systems, the cost of energy is primarily related to pumping; this results in a low energy cost relative to con-ventional district systems and the ability to sustain higher energy losses (of the uninsulated piping) more economic-ally.

Figure 3 illustrates the relationship of flow rate and temperature loss. The figure is based upon 6 in. pre-insulated (1.8 in. insulation, PVC jacket, FRP carrier pipe) and a 6-in. uninsulated pipe buried 4 ft below the ground and operating at 170°F inlet temperature. Temperature loss per 1,000 ft is plotted against flow rate. As discussed above, the graph indicates the substantial increase in temperature loss at low flow rates.

The nature of the relationship shown in Figure 3 suggests that it may be possible in some applications to adequately boost flow through a line to compensate for tem-perature loss in an uninsulated line. A temperature control valve could be placed at the end of line which could direct some flow to disposal to maintain acceptable temperature.

The prospect for the use of uninsulated piping is greatest for larger sizes (>6 in.). This is related to the fact that in larger sizes the ratio of the exposed surface area (pipe outside surface area) compared to the volume (flow capacity) is reduced. This relationship reduces the heat lost per gallon of water passed through the line.

If the use of uninsulated piping is to be economically attractive, a high load factor (total annual flow divided by peak flow) is required. In many district systems, initial customer flow requirements amount to only a small fraction of the distribution capability. Many years are required for the system to approach full capacity. Under these condi-tions, the system is operated at very low load factor initially and the economics of uninsulated piping would likely not prove to be favorable.

Systems designed for an existing group of buildings or those which serve process loads are more likely candi-dates for the use of uninsulated piping.

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HEAT EXCHANGERS

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INTRODUCTION

Most geothermal fluids, because of their elevated temperature, contain a variety of dissolved chemicals. These chemicals are frequently corrosive toward standard materials of construction. As a result, it is advisable in most cases to isolate the geothermal fluid from the process to which heat is being transferred.

The task of heat transfer from the geothermal fluid to a closed process loop is most often handled by a plate heat exchanger. The two most common types used in geothermal applications are: bolted and brazed.

For smaller systems, in geothermal resource areas of a specific character, downhole heat exchangers (DHEs) provide a unique means of heat extraction. These devices eliminate the requirement for physical removal of fluid from the well. For this reason, DHE-based systems avoid entirely the environmental and practical problems associated with fluid disposal.

GASKETED PLATE HEAT EXCHANGERS

The plate heat exchanger is the most widely used configuration in geothermal systems of recent design. A number of characteristics particularly attractive to geothermal applications are responsible for this. Among these are:

1. Superior thermal performance.
2. Availability of a wide variety of corrosion resistant alloys.
3. Ease of maintenance.
4. Expandability and multiplex capability.
5. Compact design.

Figure 1 presents an introduction to the terminology of the plate heat exchanger. Plate heat exchanger, as it is used in this section, refers to the gasketed plate and frame variety of heat exchanger. Other types of plate heat exchangers are available; though among these, only the brazed plate heat exchanger has found application in geothermal systems.

As shown in Figure 1, the plate heat exchanger is basically a series of individual plates pressed between two heavy end covers. The entire assembly is held together by the tie bolts. Individual plates are hung from the top carrying bar and are guided by the bottom carrying bar. For single-pass circuiting, hot and cold side fluid connections are usually located on the fixed end cover. Multi-pass circuiting results in fluid connections on both fixed and moveable end covers.

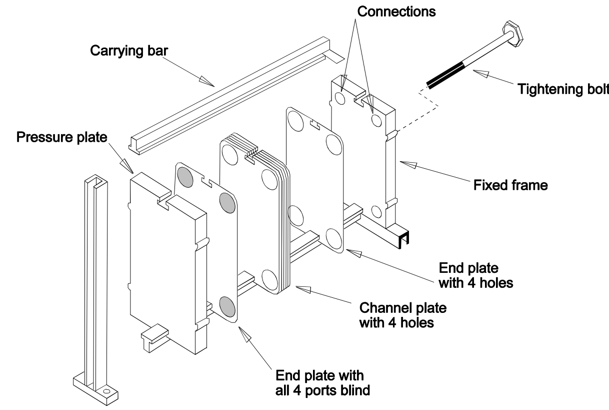

Figure 1. The plate heat exchanger.

Figure 2 illustrates the nature of fluid flow through the plate heat exchanger. The primary and secondary fluids flow in opposite directions on either side of the plates. Water flow and circuiting are controlled by the placement of the plate gaskets. By varying the position of the gasket, water can be channeled over a plate or past it. Gaskets are installed in such a way that a gasket failure cannot result in a mixing of the fluids. In addition, the outer circumference of all gaskets is exposed to the atmosphere. As a result, should a leak occur, a visual indication is provided.

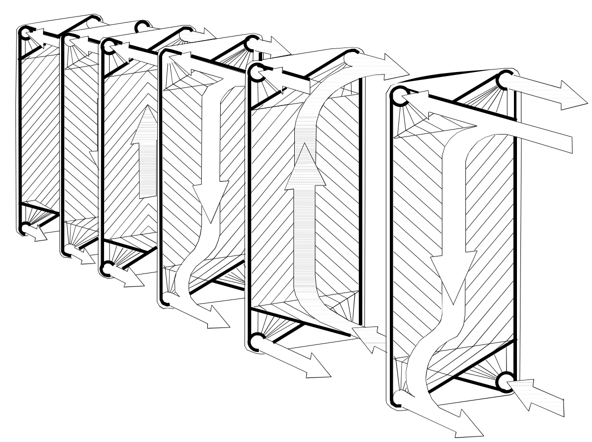

Figure 2. Nature of fluid flow through the plate heat exchanger.

General Capabilities

In comparison to shell and tube units, plate and frame heat exchangers are a relatively low pressure/low temperature device. Current maximum design ratings for most manufacturers are: temperature, 400°F, and 300 psig.

Above these values, an alternate type of heat exchanger would have to be selected. The actual limitations for a particular heat exchanger are a function of the materials selected for the gaskets and plates; these will be discussed later.

Individual plate area varies from about 0.3 to 21.5 ft^2 with a maximum heat transfer area for a single heat exchanger currently in the range of 13,000 ft^2. The minimum plate size does place a lower limit on applications of plate heat exchangers. For geothermal applications, this limit generally affects selections for loads such as residential and small commercial space heating and domestic hot water.

The largest units are capable of handling flow rates of 6000 gallons per minute (gpm) and the smallest units serviceable down to flows of approximately 5 gpm. Connection sizes are available from 3/4 to 14 in. to accommodate these flows.

Materials

Materials selection for plate heat exchangers focuses primarily upon the plates and gaskets. Since these items significantly effect first cost and equipment life, this procedure should receive special attention.

Plates

One of the features which makes plate-type heat exchangers so attractive for geothermal applications is the availability of a wide variety of corrosion-resistant alloys for construction of the heat transfer surfaces. Most manufacturers will quote either 304 or 316 stainless steel a the basic material.

For direct use geothermal applications, the choice of materials is generally a selection between 304 stainless, 316 stainless, and titanium. The selection between 304 and 316 is most often based upon a combination of temperature and chloride content of the geothermal fluid. Should oxygen be present in as little as parts per billion (ppb) concentrations, the rates of localized corrosion would be significantly increased (Ellis and Conover, 1981). Should the system for which the heat exchanger is being selected offer the potential for oxygen entering the circuit, a more conservative approach to materials selection is recommended.

Titanium is only rarely required for direct use applications. In applications where the temperature/chloride requirements are in excess of the capabilities of 316 stainless steel, titanium generally offers the least cost alternative.

The first cost premium for titanium over stainless steel plates is approximately 50%.

Gaskets

As with plate materials, a variety of gasket materials are available. Among the most common are those shown in Table 1.

Table 1. Plate Heat Exchanger Gasket Materials

Material	Common Name	Temperature Limit (°F)
Styrene-Butadiene	Buna-S	185
Neoprene	Neoprene	250
Acrylonitrile- Butadiene	Buna-N	275
Ethylene/Propylene	EPDM	300
Fluorocarbon	Viton	300
Resin-Cured Butyl	Resin-Cured Butyl	300
Compressed Asbestos	Compressed Asbestos	500

Testing by Radian Corporation has revealed that Viton shows the best performance in geothermal applications, followed by Buna-N. Test results revealed that neoprene developed an extreme compression set and Buna-S and natural rubber also performed poorly (Ellis and Conover, 1981).

Although Viton demonstrates the best performance, its high cost generally eliminates it from consideration unless its specific characteristics are required. Buna-N, generally the basic material quoted by most manufacturers, and the slightly more expensive EPDM material are generally acceptable for geothermal applications.

Performance

Superior thermal performance is the hallmark of plate heat exchangers. Compared to shell-and-tube units, plate heat exchangers offer overall heat transfer coefficients 3 to 4 times higher. These values, typically 800 to 1200 Btu/-hr·ft^2 °F (clean), result in very compact equipment. This high performance also allows the specification of very small approach temperature (as low as 2 to 5°F) which is sometimes useful in geothermal applications. This high thermal performance does come at the expense of a somewhat higher pressure drop. Selection of a plate heat exchanger is a trade-off between U-value (which influences surface area and hence, capital cost) and pressure drop (which influences pump head and hence, operating cost). Increasing U-value comes at the expense of increasing pressure drop.

Fouling considerations for plate heat exchangers are considered differently than for shell-and-tube equipment. There are a variety of reasons for this; but, the most important is the ease with which plate heat exchangers can be disassembled and cleaned. As a result, the units need not be over-designed to operate in a fouled condition. Beyond this, the nature of plate heat exchanger equipment tends to reduce fouling due to:

- High turbulence,
- Narrow high-velocity flow channels which eliminate low flow areas found in shell-and-tube equipment, and
- Stainless steel surfaces that are impervious to corrosion in most groundwater applications

Costs

For most geothermal systems, the plate heat exchanger can constitute a large portion of the mechanical room equipment cost. For this reason, it is useful to have a method of evaluating the capital cost of this component when considering the system design.

Final heat exchanger cost is a function of materials, frame size and plate configuration.

Figure 3 presents a plot of plate heat exchanger costs in 1996 dollars/ft^2 of heat transfer area based on a number of manufacturer's quotes for various geothermal applications. Since heat transfer area takes into account duty, temperature difference and fouling, it is the most useful index for preliminary costing.

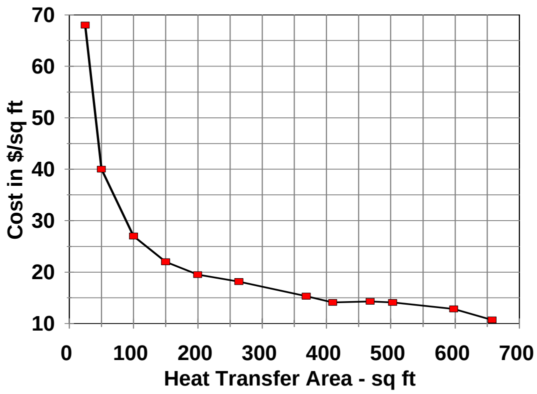

Figure 3. Plate heat exchanger cost for Buna-N gaskets and 316 stainless steel plates (1996).

BRAZED PLATE HEAT EXCHANGERS

Construction

The brazed plate unit as shown in Figure 4 eliminates the end plates, bolts, and gaskets from the design. Instead, the plates are held together by brazing with copper. This results in a much less complicated, lighter weight and more compact heat exchanger. The simpler design also results in greatly reduced cost.

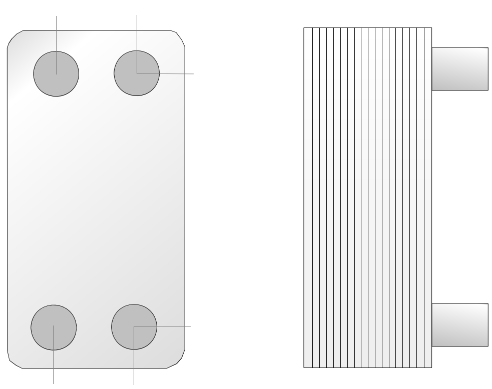

Figure 4. Brazed plate heat exchanger.

On the negative side, the brazed plate approach eliminates some of the advantages of the plate-and-frame design. In terms of maintenance, the brazed plate units cannot be disassembled for cleaning or for the addition of heat transfer plates as bolted units can.

Most importantly, however, the brazing material is copper. Since most geothermal fluids contain hydrogen sulphide (H_2S) or ammonia (NH_3), copper and copper alloys are generally avoided in geothermal system construction. The situation with brazed plate heat exchangers is especially critical due to the braze material and length (a few tenths of an inch) of the brazed joints.

Application Considerations

In addition to the material related questions, there are also issues related to the standard configuration of brazed plate heat exchangers.

Physical size of the exchangers limits application flow rates to approximately 100 gpm (athough one manufacturer produces units capable of 200 gpm). Maximum heat transfer area is limited to 200 ft^2. Heat transfer rates are similar to those of plate-and-frame heat exchangers and range from 800 - 1300 Btu/hr ft^2 ^{o}F in most applications.

The major design considerations for brazed plate exchangers is that standard units are manufactured in only single-pass flow arrangements for both hot and cold fluids. This influences the ability of the exchanger to achieve close approach temperatures in certain applications.

Heat Exchanger Equipment Cost

As discussed above, the low cost of the brazed plate heat exchanger is its most attractive feature. Since heat exchanger cost is influenced by a host of factors including hot- and cold-side fluid flows and temperatures, it is most useful to discuss costs in terms of heat transfer area.

Figure 5presents cost data for brazed plate heat exchangers. As indicated, a similar curve to the one shown earlier for plate-and-frame, holds for these units; however, it is offset toward lower costs.

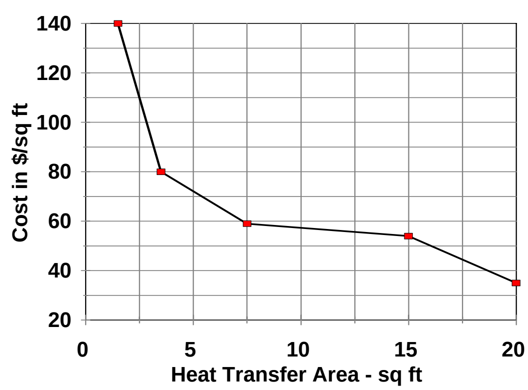

Figure 5. Brazed plate heat exchanger.

Based on limited testing, brazed plate heat exchangers should demonstrate a minimum service life of 12 years in fluids of less than 1 ppm H_2S and 10 years in fluids of 1 to 5 ppm H_2S.

Based on calculations of capital cost, service life, maintenance and installation cost our study (Rafferty, 1993) suggests that the selection of the brazed plate exchanger is valid when the capital cost is 50% or less of the plate-and-frame exchanger. This relationship was determined for fluids of < 5ppm H_2S.

DOWNHOLE HEAT EXCHANGERS

The downhole heat exchanger (DHE) is of a design that eliminates the problems associated with disposal of geothermal water since only heat is taken from the well. These systems can offer significant savings over surface heat exchangers where available heat loads are low and geologic and ground water conditions permit their use.

The use of a DHE for domestic or commercial geothermal space and domestic water heating has several appealing features when compared to the alternative geothermal heat extraction techniques. It is essentially a passive means of exploiting the geothermal energy because, in marked contrast to the alternative techniques, no water is extracted or flows from the well. Environmental and institutional restrictions generally require geothermal water to be returned to the aquifer from which it was obtained. Therefore, techniques involving removal of water from a well require a second well to dispose of the water. This can be a costly addition to a small geothermal heating project. The cost of keeping a pump operating in the sometimes corrosive geothermal fluid is usually far greater than that involved with the maintenance of a DHE.

The principal disadvantage with the DHE technique is its dependence on the natural heat flow in the part of the hot aquifer penetrated by the well. A pumped well draws in hot water and the resultant heat output is normally many times the natural value. This limitation on the potential heat output of a DHE makes it most suitable for small to moderate-sized thermal applications.

DHE outputs range from supplying domestic hot water for a single family at Jemez Springs, New Mexico to Ponderosa High School in Klamath Falls, Oregon. The single family is supplied from a 40 ft well and the school at over one MWt from a 560 ft, 202ºF, 16 in. diameter well. The DHE's are also in use in New Zealand, Austria, Turkey, the USSR and others. A DHE producing 6 MWt has been reported in use in Turkey.

Typical Designs

The most common DHE consists of a system of pipes or tubes suspended in the well through which clean water is pumped or allowed to circulate by natural convection. Figure 6 shows a U tube system typical of some 500 installations in Klamath Falls, Oregon. The wells are 10 or 12 in. diameter drilled 20 or more ft into geothermal fluids and an 8 in. casing is installed. A packer is placed around the casing below any cold water or unconsolidated rock, usually

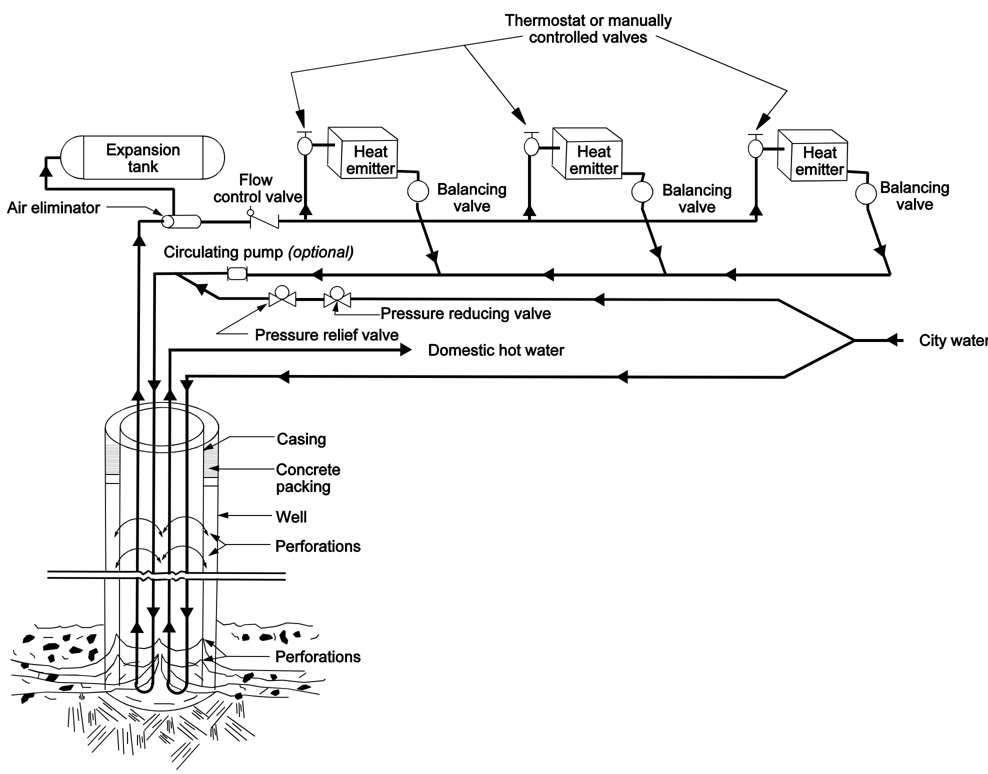

Figure 6. Typical hot-water distribution system using a downhole heat exchanger (Culver and Reistad, 1978).

20 to 50 ft, and the well cemented from the packer to the surface. The casing is torch perforated (0.5 x 6 in.) in the live water area and just below the static water level. Perforated sections are usually 15 to 30 ft long and the total cross-sectional area of the perforations should be at least 1-1/2 to 2 times the casing cross section. Because fluid levels fluctuate summer to winter the upper perforations should start below the lowest expected level. A 3/4 or 1 in. pipe welded to the outside of the casing and extending from ground surface to below the packer permits sounding and temperature measurements in the annulus and is very useful in diagnosing well problems.

The space heating DHE is usually 1-1/2 or 2 in. black iron pipe with a return U-bend at the bottom. The domestic water DHE is 3/4 or 1 in. pipe. The return U bend usually has a 3 to 5 ft section of pipe welded on the bottom to act as a trap for corrosion products that otherwise could fill the U-bend, preventing free circulation. Couplings should be malleable rather than cast iron to facilitate removal.

Materials

Considering life and replacement costs, materials should be selected to provide economical protection from corrosion. Attention should be given to the galvanic cell action between the DHE and the well casing, since the casing could be an expensive replacement item. Experience indicates that general corrosion of the DHE is most severe at the air-water interface at the static water level. Stray electrical currents can cause extreme localized corrosion below the water. Insulated unions should be used at the wellhead to isolate the DHE from stray currents in the buildings and city water lines. Galvanized pipe is to be avoided; since, many geothermal waters leach zinc and usually above 135°F, galvanizing loses its protective ability.

Considerable success has been realized with non-metallic pipe, both fiberglass-reinforced epoxy and poly-butylene. Approximately 100,000 ft of fiberglass reportedly has been installed in Reno at bottom-hole temperatures up to 325°F. The The only problem noted has been national pipe taper (NPT) thread failure that was attributed to poor quality resin in some pipe. Another manufacturer's pipe, with epoxied joints, performed satisfactorily. Before installing any FRP pipe, check with the manufacturer giving them temperature, water chemistry, and details of installation. Also check on warranties for the specific conditions.

Average DHE life is difficult to predict. For the approximately 500 black iron DHEs installed in Klamath Falls, the average life has been estimated to be 14 years. In some instances, however, regular replacement in 3 to 5 years has been required. In other cases, installations have been in service over 30 years with no problems. Stray electrical currents, as noted above, have undoubtedly been a contributing factor in some early failures. Currents of several tens of milli-amps have been measured. In others, examination of the DHEs after removal reveals long, deeply corroded lines along one side. This may be caused by themal expansion and contraction of the DHE against the side of the well bore where the constant movement could scrub off protective scale, exposing clean surface for further corrosion.

Corrosion at the air-water interface is by far the most common cause of failure. Putting clean turbine oil or paraffin in the well appears to help somewhat, but is difficult to accurately evaluate. Use of oil or paraffin is frowned on by the Enviornmental Protection Agency since geothermal water often commingles with fresh water.

DHE wells are typically left open at the top; but, there appears to be no reason they could not be sealed air-tight. Once the initial charge of oxygen is used up in forming corrosion products, there would be no more oxygen available because there is essentially no dissolved oxygen in the geothermal fluid. Swisher and Wright (1986) measured corrosion rates of mild steel in geothermal water under aerobic and anerobic conditions in the lab. They found aerobic corrosion rates of 260-280 micrometer/year with completely emersed specimens with paraffin on the water, 830 micrometer/year above the paraffin on partially emersed specimens and only 11 micrometer/year under anerobic conditions.

Convection Cells

Although the interaction between the fluid in the well, fluid in the aquifer, and the rock surrounding the well is poorly understood, it is known that the heat output can be significantly increased if a convection cell can be set up in the well. There is probably some degree of natural mixing (i.e., water from the aquifer continuously enters the well, mixes with the well fluid, and fluid leaves the well to the aquifer). There are two commonly used methods of inducing convection.

The first method may be used when a well is drilled in a stable formation, and will stand open without a casing. This allows an undersized casing to be installed. If the casing is perforated just below the minimum static water level and near the bottom or at the hot aquifer level, a convection cell is induced and the well becomes very nearly isothermal between the perforations (Figure 7). Cold surface water and unstable formations near the surface are cemented off above a packer. If a DHE is then installed and heat extracted, a convection cell is established with flow down the inside of the casing and up the annulus between the well wall and casing. The driving force is the density difference between the fluid surrounding the DHE and fluid in the annulus. The more heat extracted, the higher the fluid velocity. Velocities of 2 ft/s have been measured with very high heat extraction rates; but, the usual velocities are between 0.04 and 0.4 ft/s.

The second method is used where a different situation exists. In New Zealand where wells do not stand open and several layers of cold water must be cased off, a system using a convection promoter pipe was developed (Figure 8). The convector pipe is simply a pipe open at both ends, suspended in the well above the bottom and below the static water level. An alternate design involves the pipe resting on the bottom, and having perforations at the bottom and below static water level. The DHE can be installed either in the convector or outside the convector; the latter being more economical since smaller convector pipe is used (Freeston and Pan, 1983; Dunstall and Freeston, 1990).

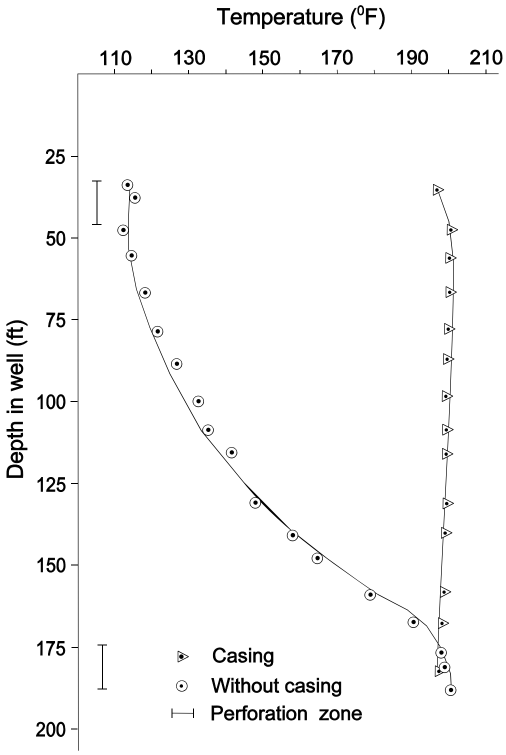

Figure 7. Temperature vs. depth with and without casing (Culver and Reistad, 1978).

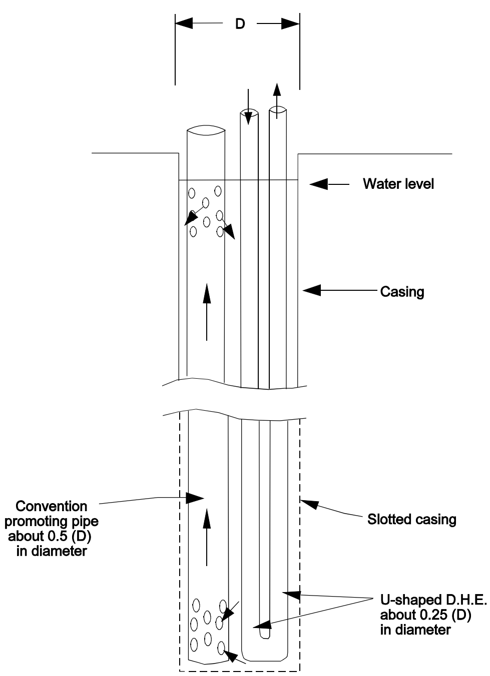

Figure 8. Convection promoter pipe with DHE (Allis and James, 1979).

Both lab and field tests indicate that the convection cell velocities are about the same in undersized casing systems and convector pipe systems.

Optimum conditions exist when frictional resistance because of wetted surfaces (hydraulic radius) is equal to both legs of the cell and DHE surface area is maximized, providing maximum heat transfer. For designs using undersized casing and DHE inside the convector, this occurs when the casing or convector is 0.7 times the well diameter. When the DHE is outisde the convector, the convector should be 0.5 times the well diameter. The full length U-tube DHE diameter is 0.25 times the well diameter in all cases. Partial length or multi-tube exchangers will have different ratios (Allis, 1979; Allis and James, 1979).

Design Considerations

Downhole heat exchangers extract heat by two methods: extracting heat from water flowing through the aquifer, and extracting stored heat from the rocks surrounding the well.

Once the DHE is extracting heat and a convection cell is established, a portion of the convecting water is new water entering the well from the aquifer, the same amount of cooled water leaves the well and enters the aquifer.

The ratio of convecting water to new water has been termed the mixing ratio and is defined as:

$$Rm = 1 - \frac{m_a}{m_t}$$

where:

Rm = mixing ratio
m_a = mass flow of new water
m_t = total mass flow of convecting water.

Note that a larger mixing ratio indicates a smaller proportion of new water in the convection cell.

Mixing ratios vary widely between wells even in the same aquifer and apparently depend on permeability. As more heat is extracted, the mass flow rate in the convection cell increases; but, the mixing ratio appears to remain relatively constant up to some point, then increases with further DHE loading. This is interpreted as the permeability, allowing new hot fluid to enter the well or, more probably, allowing used cool fluid to sink into the aquifer near the well bottom. At some combination of density difference and permeability, the ability to conduct flow is exceeded and the well rapidly cools with increasing load.

The theoretical maximum steady state amount of heat that could be extracted from the aquifer would be when the mixing ratio equals zero. That is, when all the water makes a single pass through the convection cell and out the well bottom. Mixing ratios lower than 0.5 have never been observed and usually range from about 0.5 to 0.94. The theoretical maximum steady heat extraction rate can be estimated if the hydraulic conductivity and hydraulic gradient are known and it is assumed there is some temperature drop of the water.

Many DHE wells in the Moana area of Reno are pumped to increase hot water flow into the well. Pumping rates for residential use is limited to 1800 gal/day and the pump is thermostatically controlled. The system is designed to switch on the pump if the DHE temperature drops below some predetermined level, usually approximately 120ºF. This method permits use of a well that would not supply enough heat using a DHE alone, yet minimizes pumped fluid and pumping costs. It is, however, limited to temperatures at which an economical submersible or other pump can be used.

Unfortunately, at the present time, there is no way to relate mixing ratio and permeability. With good permeability similar to well-fractured basalt, the mixing ratio might be approximately 0.5, in coarse sand approximately 0.8, and in clayey sand 0.9 to 0.94.

At the time the term mixing ratio was introduced, it seemed to be a logical hypothesis because all known DHE wells had (and most still have) perforations, at least in the hot aquifer zone. Some new fluid could enter the well, mix with fluid in the well and some used water exit the well. The mixing ratio is really a term for energy input into the well. Although perforations undoubtedly help, a solidly cased well with a DHE will provide heat. The energy output is then limited by the conduction of the rock and casing, allowing energy to flow into the well.

In Klamath Falls, it has been experimentally verified that when a well is drilled, there is negligible convective flow in the well bore. When undersized perforated casing is installed, a convection cell is set up, causing flow up the inside of the casing and down the annulus between the casing and well wall. When a DHE is installed and heat is extracted, the convection cell reverses with the flow downward in the casing (around the DHE) and up the annulus. Similar circulation patterns were noted in New Zealand using convection promoters.

DHEs are principally used in space and domestic water heating applications: homes, schools, small com-mercial buildings and greenhouses, with the resulting inter-mittent operation. When the heating system is not calling for heat, and if a convection cell can exist, it functions to store heat in rocks surrounding the well; especially those cooler rocks nearer the surface that would normally be at the natural temperature gradient for the locale. The under-sized casing or convection promoter then acts to increase thermal storage.

Referring again to Figure 7, it can be seen that up to the upper perforations, the well becomes very nearly isothermal, with the upper portion approaching the aquifer temperature and the rock temperature increasing significantly. When a DHE is turned on, the water in the well cools rather rapidly; the rate depending on the mixing ratio. As the water continues to cool, the convection cell extracts heat from the surrounding rocks.

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SPACE HEATING EQUIPMENT

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INTRODUCTION

The performance evaluation of space heating equipment for a geothermal application is generally considered from either of two perspectives: (a) selecting equipment for installation in new construction, or (b) evaluating the performance and retrofit requirements of an existing system.

With regard to new construction, the procedure is relatively straightforward. Once the heating requirements are determined, the process need only involve the selection of appropriately sized hot water heating equipment based on the available water temperature.

It is important to remember that space heating equipment for geothermal applications is the same equipment used in non-geothermal applications. What makes geothermal applications unique is that the equipment is generally applied at temperatures and flow rates that depart significantly from traditional heating system design.

HEATING EQUIPMENT PERFORMANCE AT NON-STANDARD CONDITIONS

For about the past 40 years, heating systems have been designed for a hot water supply temperature of 180 to 200°F with a 20°F temperature drop (ΔT). These tempera-tures were chosen largely to result in equipment require-ments similar to those of the older steam systems. Equip-ment manufacturer's selection data are indexed to these temperatures as are the practices of many design professionals.

Geothermal resources, of the variety frequently applied to space heating applications, are generally characterized by temperatures less than the standard 180 to 200°F range. Because well pumping costs constitute a sizable portion of the operating costs of a geothermal system, it is in the best interest of the designer to minimize flow requirements. This requires higher system ΔT than conventional designs. Accordingly, it is beneficial to examine the performance of heating equipment at low flow or temperature conditions.

Heat output is relatively unaffected by water side velocity above a certain critical value. It is important when designing for larger than normal ΔT (low flow rate) that the critical velocity for the heating equipment in question be avoided, as capacity falls off asymtotically in this region. For the piping diameters and temperatures generally encountered in heating equipment, velocities of 0.25 ft/s or less should be avoided for 1 to 2 in. lines (typical of finned tube radiators) and 0.50 ft/s or less for 1 in. and smaller lines (typical of finned coil equipment). For the 5/8 in. tubes commonly found in finned coil equipment, this velocity corresponds to a flow rate of approximately 0.6 gal/min gpm. In most cases, this very low velocity would only become a factor in applications of very low capacity (<15,000 Btu/h) using a ΔT of 40°F or more.

Figure 1 illustrates the average effect of reduced water temperature on hot water heating equipment performance. Individual terminal equipment types respond differently. Consult ASHRAE (1996) for exact performance.

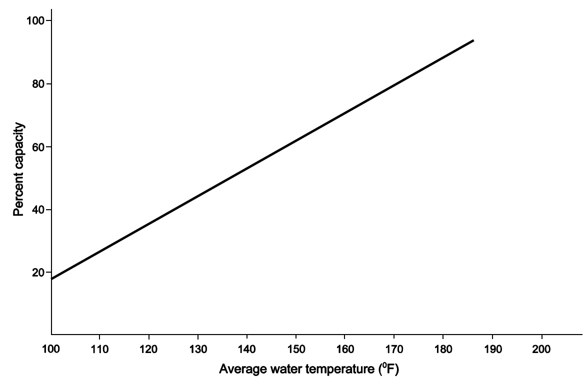

Figure 1. Average capacity at reduced water temperature (ASHRAE, 1996).

Figure 1, as for most heating equipment, is indexed to a temperature of 215°F. The percent capacity shown on the vertical axis is the percent of the 215°F rated capacity at the temperature in question. For example, the output of a particular piece of heating equipment at 150°F would be approximately 45% of its capacity at 215°F. This relationship holds for equipment such as finned tube radiators, unit heaters, cast iron radiators, and convectors.

For finned coils, the considerations are somewhat more complex with respect to low temperature service. For other types of equipment, compensation for low temperature operation is primarily in terms of additional length, larger individual units, or a greater number of units. For finned coils, the physical size (in terms of face area) can remain unchanged and the configuration of the coil (number of rows and fins/in. or both) adjusted to accommodate low temperature operation.

12.3 USE OF HEAT EXCHANGERS

Most geothermal systems will employ a heat exchanger to isolate the building heating loop from the geothermal fluid. As a result, the supply water temperature available to the heating system will be less than the geothermal resource temperature. In most cases, an allowance of a 10ºF loss through the heat exchanger will be sufficient for the selection of heating equipment. Economical heat exchanger selections generally fall between the 5 and 10ºF approach to the geothermal temperature.

CONTROLS CONSIDERATIONS

Certain control strategies enhance the effectiveness of using a geothermal resource in a building HVAC system. Some of the more important of these are discussed in the paragraphs below.

Main Heat Exchanger Control

Most geothermal systems use a plate type heat exchanger to isolate the building's circulating loop from exposure to the geothermal fluid. A variety of options can be used for control of this heat exchanger.

A method that can be utilized when the user has little or no control over the resource temperature and flow rate is shown in Figure 2. Under this design condition, the primary side of the heat exchanger is permitted to run wild (operate without temperature control) and temperature control is accomplished on the secondary side. This approach may be used for applications that involve cascaded resources, or when a constant resource flow must be maintained. A three-way valve on the secondary side of the heat exchanger is used for supply water temperature control.

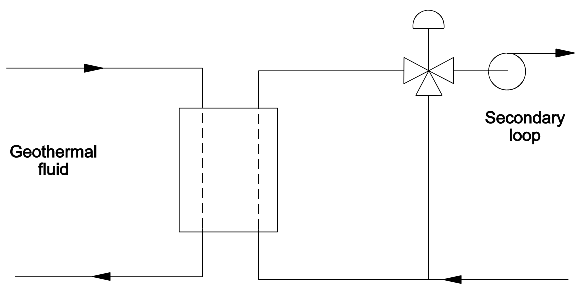

Figure 2. Heat exchanger used to isolate building heating loop from geothermal fluid.

Because most larger geothermal systems produce fluid from a well, there is adequate control of the source. As a result, control is applied to the primary side of the heat exchanger. In most cases, it is desirable to use a two-way control valve at the heat exchanger. The two-way valve allows for either throttling control of the production well pump or, when used in conjunction with a variable speed drive, allows control of the drive through production line pressure. In this way, only the quantity of geothermal fluid necessary to meet the load is pumped.

For temperature control, it is acceptable to place the control valve on either the inlet or outlet of the heat exchanger. Because of the very small fluid volume in the plate heat exchanger, there is little thermal mass to interfere with response to load changes as can sometimes be the case with downstream control valve locations in other applications.

In some cases, it may be desirable to place the control valve at the heat exchanger outlet. This location is preferred when the geothermal fluid contains a high percentage of dissolved gases, particularly CO_2. It is common for such gases to come out of solution when the fluid pressure is reduced (such as at a control valve) below the gas saturation pressure. Release of CO_2 can change the fluid pH to allow other species to precipitate out on nearby surfaces. However, downstream location for the control valve maintains the pressure on the heat exchanger to prevent such an occurrence.

Under most circumstances, the valve is controlled to maintain a particular supply fluid temperature. This set point can be reset by a discriminator control or by outdoor air temperature, depending upon the design of the system.

Supply Water Reset Control

Using a supply water reset control on the building loop, if possible, is desirable because this type of control results in a reduced supply water temperature with reduced load. Assuming a constant geothermal fluid temperature, such control allows for an increasing ΔT on the geothermal side of the heat exchanger as load decreases. This, in turn, allows for reduced fluid flow requirements from the production well. Reduced flow rates are always desirable in a geothermal system from both an economic standpoint and for aquifer conservation purposes.

Lower Supply Water Temperatures

Designing for the lowest secondary supply water temperature that is economically feasible reduces geothermal flow requirements. At a constant resource temperature, progressively lower supply water temperatures (on the building side of the heat exchanger) result in correspondingly lower geothermal flow requirements assuming a constant approach to the return water temperature.

Design for Higher System ΔT

For purposes of reduced geothermal flow, it is desirable to design for larger than the standard 20ºF temperature difference. Depending upon the specific design, a ΔT of 30 to 40ºF or more is desirable.

Use of Two-Way Control Valves

Two-way control valves are the preferred method of control for a geothermal space heating system. In addition to their superior control characteristics in general, they provide additional benefits for geothermal systems. With two-way valve control, the system responds to load reductions at a relatively constant ΔT. This contrasts with the three-way valve or "constant" flow control under which the system ΔT decreases with the load. The ability to maintain higher system ΔT is desirable with geothermal systems. Two-way control provides this feature (Haines,1983)(ASHRAE, 1996).

RETROFIT OF EXISTING SYSTEMS

Certain types of heating systems are more amenable to geothermal retrofit than others. For existing hot water systems, adequate operation at lower supply water temperatures may have to be verified. For non-hot water systems, it is likely that new hot water equipment will need to be installed adjacent to or in place of the existing equipment. Over the years, the Oregon Institute of Technology's Geo-Heat Center has gained considerable experience in evaluating heating systems for retrofit. Table 1 summarizes the results of this experience with regard to some of the systems that may be encountered. It is important to note that the retrofit suitability of these systems as indicated in the table is not absolute. Site specific considerations can easily alter the ability (either positively or negatively) of a system to accommodate hot water use.

Table 1. Retrofit Suitability Values[a,b] of Selected Heating Systems (Rafferty, 1986)

	Retrofit Suitability	
	Single Air Handler	Multiple Air Handler
Air System		
Low temperature hot water (<150°F)		
Single zone, multi-zone, dual duct	10	8
Terminal reheat, variable volume, induction	8	6
Standard hot water (180-200°F)		
Single zone, multi-zone, dual duct	8	7
Terminal reheat, variable volume, induction	7	6
Steam		
Single zone, multi-zone, dual duct	6	6
Terminal reheat, variable volume, induction	5	4
Electric resistance forced air	6	5
Air-to-air split system heat pump	4	3
Fossil fuel fired furnace	5	4
Roof top packaged equipment	4	3
Fossil fuel fired unit heaters	4	3
Water Systems		
Loop heat pump	--	10
Radiant panel	--	10
Fan coil/unit ventilator		
2 Pipe	--	9
4 Pipe single coil	--	9
4 Pipe	--	7
Unit heaters	--	7
Finned tube/convector	--	6
Steam Systems		
Finned tube radiation	--	3
Unit ventilator	--	3
Two pipe cast iron radiator	--	2
One pipe cast iron radiator	--	1
Perimeter Electric Systems		
Electric resistance baseboard	--	2
Through-the-wall units	--	1

a. Suitability values shown above are average. Site specific conditions frequently influence suitability in positive or negative ways. The table addresses only the mechanical considerations of the retrofit. The relative energy efficiency of the existing system also heavily influences retrofit suitability.

b. A value of 10 is best, 1 is worst.

The most important considerations is the degree of excess capacity present in the existing system. This excess capacity, present in most systems, is the result of a number of factors, the most important of which is conservative design practice. In addition, manual methods of equipment selection used in the past resulted in conservative results compared to present automated methods. It is not unusual to find a heating system with an over-design factor of 50% or more. This is a result of the nature of the system design. First, the peak heat loss is calculated, sometimes using unrealistically low outside design temperature that artificially increases the load by approximately 10%. Then a 10 to 30% safety factor is added, a 5% duct loss factor, and a 25% pickup factor (for regain after night set back). When equipment is selected, the capacity may be anywhere from 5 to 20% over the requirements because of equipment availability. When the results of all this are considered together, 1.10 x 1.10 x 1.05 x 1.25 x 1.05 = 1.68, the system can be grossly oversized. As a result, it can be operated at significantly reduced capacity and still meet heating requirements with no difficulty. In many cases, over-designed hot water systems have been operated at much reduced supply water temperatures (lower capacity) and actually provided improved performance through better part-load valve control.

The existing equipment capacity does not always reflect the actual heating requirements of the building. The presence of excess capacity in the existing system generally offers some advantage in the retrofit process.

Air Systems

Air systems involve the delivery of heated air from a central source, generally through a ducted distribution system, to the space to be heated. This group can generally be split into two classifications: (a) large building systems, and (b) small building systems.

Small building air systems are distinct from large building systems in terms of complexity and heat source. In smaller buildings, a separate boiler to supply hot water or steam to the air handlers is generally not included. As a result, individual equipment serves as both the heat source and the air handler as in the case of air-to-air heat pumps, roof top gas/electric units, and fossil fuel fired and electric resistance furnaces. The duct distribution system, if any, is generally much less sophisticated than in the large air systems. Retrofit costs for small building air systems are as much a function of the number of individual units as of the type of unit.

Many of the small building systems involve installation of a hot water coil for retrofit purposes. In many cases, the access to, and the sizing of, the return air duct would result in a much easier retrofit than the supply air duct location. Return air hot water coil retrofits should be avoided. Locating the heating coil in the return air stream results in two primary difficulties because of the elevated return air temperature: (a) reduced fan motor cooling, and b) reduced fan capacity.

Most small equipment is designed for return air cooling of the fan motor. Raising the temperature of the air stream (with the new coil) results in motor overheating. In addition, an increased return air temperature increases the specific volume of the air, thus reducing fan capacity. Placing the coil in the return air stream should be used only when full consideration has been given to these issues.

Water Systems

Water systems can be variously configured, but each will have a main hot water circulating loop that serves a number of individual heating units. Though individual terminal units may use small duct-type distribution systems within their respective zones, water systems do not have a central duct distribution system. Systems included: (a) loop heat pump, (b) radiant panel, (c) fan coil/unit ventilator, (d) hot water unit heater, and (e) finned tube/convection.

The loop heat pump system provides for one of the simplest retrofits to geothermal. This type of system, as depicted in Figure 3, uses a very low temperature water loop serving a large number of individual heat pump units throughout the building.

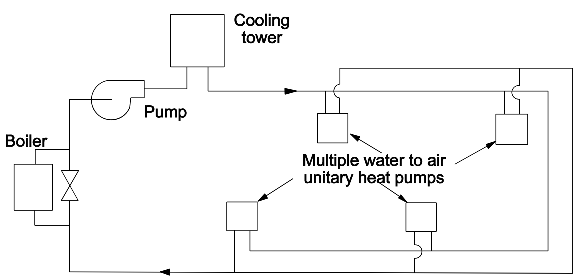

Figure 3. Water loop heat pump system flow schematic (Bloomquist, 1987).

During the cooling season, heat is rejected from the cooling tower to cool the circuit. During the colder periods of the year, heat is added to the loop by a boiler. The attractiveness of the loop heat pump system lies in the fact that the water circuit serving the heat pumps is generally maintained at 60 to 90ºF, depending upon the season.

Radiant panel systems are rarely used today, but were fairly common in construction of the 1950s. Applications that lend themselves well to this type of system are automotive repair shops, large high ceiling manufacturing structures, and schools. Radiant panel systems involve the circulation of warm water (90 to 130ºF) through piping that is embedded in the floor of the building. Older systems were constructed with copper or steel piping. Leaks that developed because of expansion and contraction, and corrosion resulted in expensive repair requirements. As a result, the panel system fell into disuse for many years. With the advent of new, nonmetallic piping products (primarily polybutylene), radiant panel systems have begun to reappear.

Fan coil (FC) and unit ventilator (UV) systems or both are found primarily in hotel/motel chains and schools. The system consists of a main hot water loop that serves a large number of terminal units located throughout the building. Coil units, as shown in Figures 4 and 5, consist of a sheet metal box containing a fan, air filter, and one or two coils. A unit ventilator is similar to a fan coil unit with the exception that it contains accommodations for the supply of outdoor air for ventilation.

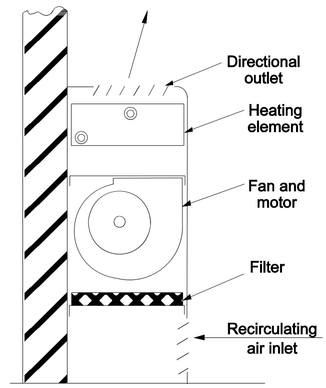

Figure 4. Vertical fan coil unit (Bloomquist, 1987).

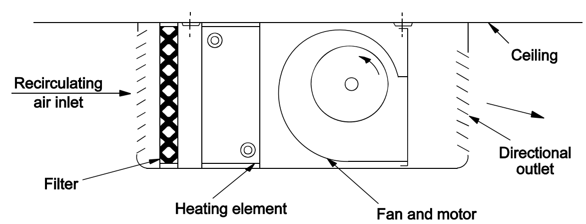

Figure 5. Horizontal fan coil unit (Bloomquist, 1987).

Two types of FC/UV systems are available: two pipe and four pipe. The two or four pipe designation refers to the water distribution system serving the terminal equipment. A two-pipe system includes only one supply line and one return line. As a result, it can supply only heating or cooling to the building at any particular time. Fan coil units and unit ventilators served by a two pipe system contain only one coil that serves as heating or cooling coil, depending upon the season.

The four pipe system includes a distribution system that contains both hot water supply and return lines and chilled water supply and return lines. As a result, either heating or cooling can be delivered to any zone at any time. Heating coils in these units generally require much higher water temperature than two pipe system units

Hot water unit heaters are a simpler version of the system described above. This equipment is found in applications in which noise generation and aesthetics are less of a consideration, such as automotive repair shops, warehouses, supermarkets, and small retail stores.

Unit heaters are available in two basic configurations: horizontal (Figure 6) and vertical (Figure 7), with horizontal units the most common. Assuming that the supply fluid temperature after connection to the geothermal system will be equal to or greater than the present supply temperature, this system would be a good candidate. If the expected supply fluid temperature will be less than the existing system, retrofit of the terminal equipment or peaking may be required. When operated on lower than originally designed water temperature, unit heaters produce correspondingly lower supply air temperatures. This can result in a drafty sensation for occupants. However, because of the application in which these units are normally found, a greater latitude can be taken with respect to performance.

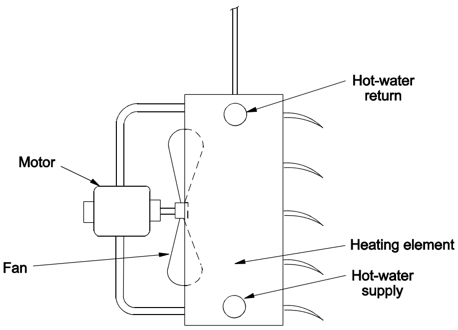

Figure 6. Horizontal hot water unit heater (Bloomquist, 1987).

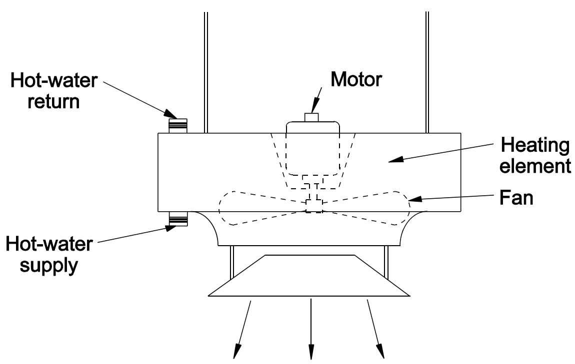

Figure 7. Vertical hot water unit heater (Bloomquist, 1987).

Finned tube/convector systems require the highest temperature of all hot water systems. This equipment is found in many types of buildings and frequently in conjunction with an air system in larger buildings. Because this system uses no fans for circulating, it relies entirely on elevated temperature to promote the air convection by which it operates. As a result, it does not perform well at temperatures less than that for which it was designed.

As with most other water systems, retrofit of the equipment is generally less economical than occasional peaking with the conventional boiler. The design philosophy for finned tube systems involves using a relatively low output/ft (Btu/h· lf) of element so as to result in a large length requirement, thus covering most of the inside perimeter of the building. As a result, it is difficult to compensate for lower temperature operation by installing additional heating elements.

Steam Systems

As with the water systems, steam systems may take a variety of configurations. Those included under this classification are: (a) finned tube radiation/convector, (b) unit heater/unit ventilator, (c) two pipe cast iron radiation, and (d) one pipe cast iron radiation.

The principle characteristic which distinguishes these from other systems is the use of a steam heating medium directly in the terminal equipment. Many buildings contain a steam boiler but use a convector to produce hot water for heating purposes. The system described in this section delivers the steam directly to the heating equipment. This is generally low pressure steam at 15 lb/in.^{2} (psi) or less. As a result, it has a temperature of approximately 200 to 240°F. This illustrates the primary disadvantage of steam equipment for geothermal heating operations. It is unlikely that most geothermal systems will be capable of delivering water that is hot enough to generate steam for the existing building's steam system. Because the supply water temperature for a hot water system will likely be less than 200°F, the steam equipment will operate at a much reduced capacity because it was designed for 200 to 240°F. If the system does not contain sufficient excess capacity to accommodate this, much of the terminal equipment will have to be replaced or significant peaking will be needed.

A second difficulty with steam systems that must be converted to hot water lies in the piping arrangement. Steam systems produce heat by allowing the steam to condense in the heating equipment. For each pound of steam condensed, 1000 Btu is supplied to the space. When the steam condenses, a large volume reduction occurs (1 lb of condensate or water is much smaller than 1 lb of steam). To accommodate this volume reduction, steam systems employ very large lines to deliver the steam to the heating equipment and very small ones to carry away the condensate. When the system is converted to hot water, the steam piping is usually much larger than required, which does not present a problem. The condensate lines, however, are frequently much smaller than required for the hot water flow. These lines, in some cases, must be replaced with adequately sized piping.

Steam controls are rarely acceptable for hot water operation. As shown in Figure 8, steam systems include not only a valve to control the flow of steam to the heating equipment, but a trap for regulating condensate flow out of the equipment. In a conversion to a hot water system the steam control valve should be replaced with a hot water control valve and the trap removed from the line.

The difficulty of the replacement of this equipment is compounded by the fact that most steam systems are at least 25 years old, and many are closer to 50 years old.

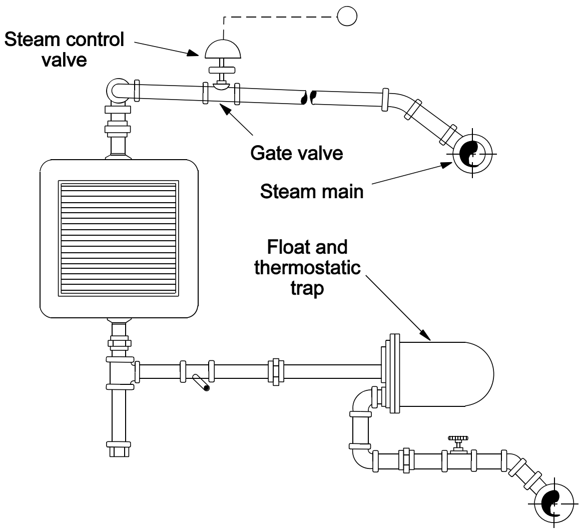

Figure 12.31 Steam unit heater (Bloomquist, 1987).

In summary, the magnitude of the retrofit requirements for steam systems frequently causes them to be uneconomical to connect to geothermal systems.

Finned tube radiation/convector systems operated on steam are very much the same as those described under the hot water systems above. The difficulty associated with installing additional elements discussed above is compounded by a large capacity reduction that is experienced in converting from steam to hot water. When piping and controls replacement are considered, this system is rated as only a 3 in terms of retrofit suitability.

Cast iron radiation systems are divided into two groups: one pipe and two pipe. Neither of these are particularly suitable for hot water operation.

Domestic Hot Water Heating

Domestic hot water heating is frequently served by retrofit heating systems. One of the early determinations to be made in a geothermal feasibility study is whether or not to connect a particular building's domestic hot water system to the retrofit heating system. The decision should be based primarily upon the volume of hot water used in the building. In general, hotels, motels, apartment buildings, high schools, restaurants, hospitals, and health clubs will be characterized by sufficient domestic hot water consumption to warrant retrofit of the existing system. Buildings such as offices, retail stores, theaters, and elementary schools are unlikely to be attractive domestic hot water candidates.

The preferred arrangement for domestic hot water heating is shown in Figure 9. Under this design, water exiting from the space heating heat exchanger is directed to the domestic hot water heat exchanger. This scheme provides for larger temperature drop in both the end user building and in the retrofit heating system. Larger temperature drops reduce system flow rates and required piping sizes.

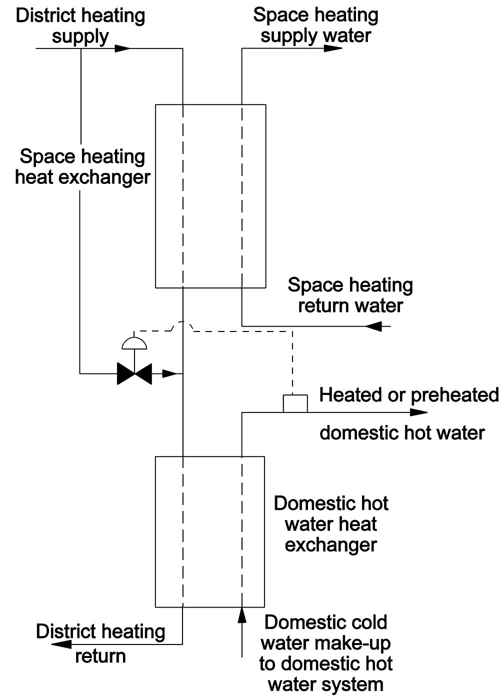

Figure 9. Typical domestic hot water heating flow scheme (Bloomquist, 1987).

Sizing procedures for this type of instantaneous heating arrangement are found in the ASHRAE, 1984 Systems Volume, Chapter 34. Basically, hot water demand in fixture units is determined and the required hot water flow rate for the building in question is found using the Modified Hunter Curves.

Under some conditions, the flow rate from the space heat exchanger will not be sufficient to raise the domestic hot water to the required temperature. In this case, a second circuit connected to the primary hot water supply can be added to the domestic hot water heating heat exchanger. This second circuit would provide the additional boosting of the domestic hot water to the required temperature.

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Rafferty, K., 1986. "Geothermal Retrofit of Existing Space Heating Systems," Geo-Heat Center Quarterly Bulletin, Vol. 9, No. 3, Oregon Institute of Technology, Geo-Heat Center, Klamath Falls, OR.

ABSORPTION REFRIGERATION

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INTRODUCTION

The absorption cycle is a process by which refrigeration effect is produced through the use of two fluids and some quantity of heat input, rather than electrical input as in the more familiar vapor compression cycle. Both vapor compression and absorption refrigeration cycles accomplish the removal of heat through the evaporation of a refrigerant at a low pressure and the rejection of heat through the condensation of the refrigerant at a higher pressure. The method of creating the pressure difference and circulating the refrigerant is the primary difference between the two cycles. The vapor compression cycle employs a mechanical compressor to create the pressure differences necessary to circulate the refrigerant. In the absorption system, a secondary fluid or absorbent is used to circulate the refrigerant. Because the temperature requirements for the cycle fall into the low-to-moderate temperature range, and there is significant potential for electrical energy savings, absorption would seem to be a good prospect for geothermal application.

Absorption machines are commercially available today in two basic configurations. For applications above 32°F (primarily air conditioning), the cycle uses lithium bromide as the absorbent and water as the refrigerant. For applications below 32°F, an ammonia/water cycle is employed with ammonia as the refrigerant and water as the absorbent.

LITHIUM BROMIDE/WATER CYCLE MACHINES

Figure 1 shows a diagram of a typical lithium bromide/water machine (Li Br/H_2O). The process occurs in two vessels or shells. The upper shell contains the generator and condenser; the lower shell, the absorber and evaporator.

Heat supplied in the generator section is added to a solution of Li Br/H_2O. This heat causes the refrigerant, in this case water, to be boiled out of the solution in a distillation process. The water vapor that results passes into the condenser section where a cooling medium is used to condense the vapor back to a liquid state. The water then flows down to the evaporator section where it passes over tubes containing the fluid to be cooled. By maintaining a very low pressure in the absorber-evaporator shell, the water boils at a very low temperature. This boiling causes the water to absorb heat from the medium to be cooled, thus, lowering its temperature. Evaporated water then passes into the absorber section where it is mixed with a Li Br/H_2O solution that is very low in water content. This strong solution (strong in Li Br) tends to absorb the vapor from the evaporator section to form a weaker solution. This is the absorption process that gives the cycle its name. The weak solution is then pumped to the generator section to repeat the cycle.

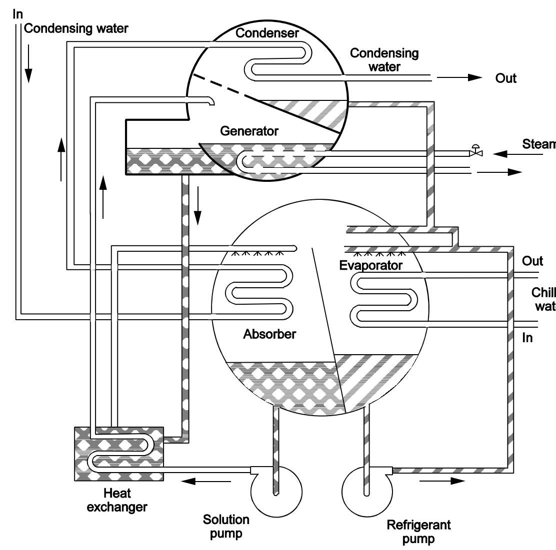

Figure 13.1 Diagram of two-shell lithium bromide cycle water chiller (ASHRAE, 1983).

As shown in Figure 1, there are three fluid circuits that have external connections: a) generator heat input, b) cooling water, and c) chilled water. Associated with each of these circuits is a specific temperature at which the machines are rated. For single-stage units, these tempera-tures are : 12 psi steam (or equivalent hot water) entering the generator, 85°F cooling water, and 44°F leaving chilled water (ASHRAE, 1983). Under these conditions, a coeffic-ient of performance (COP) of approximately 0.65 to 0.70 could be expected (ASHRAE, 1983). The COP can be thought of as a sort of index of the efficiency of the machine. It is calculated by dividing the cooling output by the required heat input. For example, a 500-ton absorption chiller operating at a COP of 0.70 would require: (500 x 12,000 Btu/h) divided by 0.70 = 8,571,429 Btu/h heat input. This heat input suggests a flow of 9,022 lbs/h of 12 psi steam, or 1,008 gpm of 240°F water with a 17°F ΔT.

Two-stage machines with significantly higher COPs are available (ASHRAE, 1983). However, temperature requirements for these are well into the power generation temperature range (350°F). As a result, two-stage machines would probably not be applied to geothermal applications.

PERFORMANCE

Based on equations that have been developed (Christen, 1977) to describe the performance of a single-stage absorption machine, Figure 2 shows the effect on COP and capacity (cooling output) versus input hot-water temperature. Entering hot water temperatures of less than 220°F result in substantial reduction in equipment capacity. The reason for the steep drop off in capacity with temperature is related to the nature of the heat input to the absorption cycle. In the generator, heat input causes boiling to occur in the absorbent/refrigerant mixture. Because the pressure is fairly constant in the generator, this fixes the boiling temperature. As a result, a reduction in the en-tering hot water temperature causes a reduction in the temperature difference between the hot fluid and the boiling mixture. Because heat transfer varies directly with temper-ature difference, there is a nearly linear drop off in absorp-tion refrigeration capacity with entering hot water tempera-ture. In the past few years, one manufacturer (Yazaki, undated) has modified small capacity units (2 to 10 ton) for increased performance at lower inlet temperature. How-ever, low-temperature modified machines are not yet avail-able in large outputs, which would be applicable to institutional- and industrial-type projects. Although COP and capacity are also affected by other variables such as condenser and chilled water temperatures and flow rates, generator heat input conditions have the largest impact on performance. This is a particularly important consideration with regard to geothermal applications.

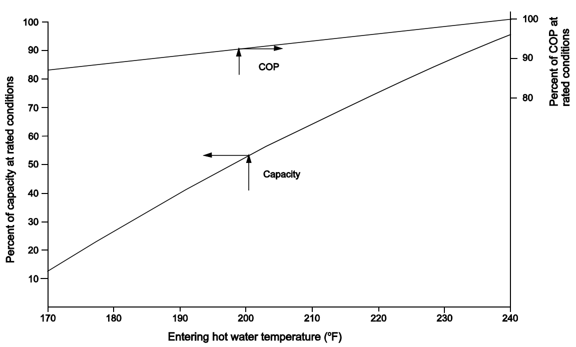

Figure 2. Capacity of a lithium bromide absorption chiller (Christen, 1977).

Because many geothermal resources in the 240°F and above temperature range are being investigated for power generation using organic Rankine cycle (ORC) schemes, it is likely that space conditioning applications would see temperatures below this value. As a result, chillers operating in the 180 to 230°F range would (according to Figure 2) have to be (depending on resource temperature) between 400 and 20% oversized respectively for a particular application. This would tend to increase capital cost and decrease payback when compared to a conventional system.

An additional increase in capital cost would arise from the larger cooling tower costs that result from the low COP of absorption equipment. The COP of singe effect equipment is approximately 0.7. The COP of a vapor compression machine under the same conditions may be 3.0 or higher. As a result, for each unit of refrigeration, a vapor compression system would have to reject 1.33 units of heat at the cooling tower. For an absorption system, at a COP of 0.7, 2.43 units of heat must be rejected at the cooling tower. This results in a significant cost penalty for the absorption system with regard to the cooling tower and accessories.

In order to maintain good heat transfer in the generator section, only small ΔTs can be tolerated in the hot water flow stream. This is a result of the fact that the machines were originally designed for steam input to the generator. Heat transfer from the condensing steam is a constant temperature process. As a result, in order to have equal performance, the entering hot water temperature would have to be above the saturated temperature corresponding to the inlet steam pressure at rated conditions. This is to allow for some ΔT in the hot water flow circuit. In boiler coupled operation, this is of little consequence to operating cost. However, because ΔT directly affects flow rate, and thus pumping energy, this is a major consideration in geothermal applications.

For example, assuming a COP of 0.54 and 15°F ΔT on the geothermal fluid, 250 ft pump head and 65% wire-to-water efficiency at the well pump, approximately 0.20 kW/t pumping power would be required. This compares to approximately 0.50 - 0.60 kW/t for a large centrifugal machine (compressor consumption only).

The small ΔT and high flow rates also point out another consideration with regard to absorption chiller use in space conditioning applications. Assume a geothermal system is to be designed for heating and cooling a new building. Because the heating system can be designed for rather large ΔTs in comparison to the chiller, the incremental cost of the absorption approach would have to include the higher well and/or pump costs to accommodate its requirements. A second approach would be to design the well for space heating requirements and use a smaller absorption machine for base load duty. In this approach, a second electric chiller would be used for peaking. In either case, capital cost would be increased.

LARGE TONNAGE EQUIPMENT COSTS

Figure 3 presents some more general cost information on large tonnage (>100 tons) cooling equipment for space conditioning applications. The plot shows the installed costs for both absorption chillers (Abs. chlr.), centrifugal chillers (Elec. chlr.), and auxilliary condenser equipment (cooling tower, cooling water pumps and cooling water piping) for both absorption chillers (Abs. twr.) And centrifugal chillers (Elec. twr.). As shown, both the chiller itself and its auxilliary condenser equipment costs are much higher for the absorption design than for electric-driven chillers. These are the primary capital cost differences that a geothermal operation would have to compensate for in savings.

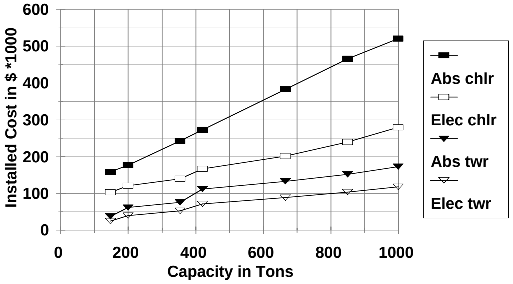

Figure 13.3 Chiller and auxiliary equipment costs - electric and absorption (Means, 1996).

SMALL TONNAGE EQUIPMENT

To our knowledge, there is only one company (Yazaki, undated) currently manufacturing small tonnage (<20 tons) lithium bromide refrigeration equipment. This firm, located in Japan, produces equipment primarily for solar applications. Currently, units are available in 1.3, 2, 3, 5, 7.5, and 10 ton capacities. These units can be manifolded together to provide capacities of up to 50 tons.

Because the units are water cooled chillers, they require considerably more mechanical equipment for a given capacity than the conventional electric vapor compression equipment usually applied in this size range. In addition to the absorption chiller itself, a cooling tower is required. The cooling tower, which is installed outside, requires interconnecting piping and a circulation pump. Because the absorption machine produces chilled water, a cooling coil and fan are required to deliver the cooling capacity to the space. Insulated piping is required to connect the machine to the cooling coil. Another circulating pump is required for the chilled water circuit. Finally, hot water must be supplied to the absorption machine. This requires a third piping loop.

In order to evaluate the economic merit of small absorption equipment compared to conventional electric cooling, Figure 4 was developed. This plot compares the savings achieved through the use of the absorption equipment to its incremental capital costs over a conventional cooling system. Specifically, the figure plots cost of electricity against simple payback in years for the five different size units. In each case, the annual electric cost savings of the absorption system (at 2,000 full load hours per year) is compared to the incremental capital cost of the system to arrive at a simple payback value. The conventional system to which absorption is compared in this case is a rooftop package unit. This is the least expensive conventional system available. A comparison of the absorption approach to more sophisticated cooling systems (VAV, 4-pipe chilled water, etc.) would yield much more attractive payback periods.

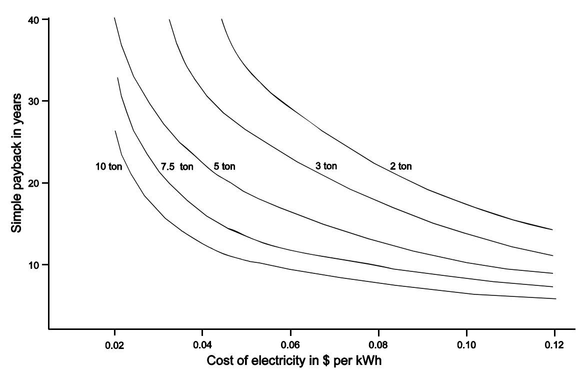

Figure 4. Simple payback on small absorption equipment compared to conventional rooftop equipment.

The plot is based on the availability of geothermal fluid of sufficient temperature to allow operation at rated capacity (190°F or above). In addition, other than piping, no costs for geothermal well or pumping are incorporated. Only cooling equipment related costs are considered. As a result, the payback values in Figure 4 are valid only for a situation in which a geothermal resource has already been developed for some other purpose (space heating and aquaculture), and the only decision at hand is that of choosing between electric and absorption cooling options.

Figure 4 also shows that the economics of small tonnage absorption cooling are attractive only in cases of 5 to 10 ton capacity requirements and more than $0.10 kW/h electrical costs. Figure 4 is based on an annual cooling requirement of 2,000 full load hours per year. This is on the upper end of requirements for most geographical areas. To adjust for other annual cooling requirements, simply multiply the simple payback from Figure 4 by actual full load hours and divide by 2,000.

The performance of the absorption cooling machine was based on nominal conditions in order to develop Figure 4. It should be noted that, as with the larger machines, performance is heavily dependent upon entering hot water temperature and entering cooling water temperature. Ratings are based on 190°F entering hot water, 85 °F entering cooling water and 48°F leaving chilled water. Flow rates for all three loops are based upon a 9°F ΔT.

Figure 4 illustrates the effect of entering hot water temperature and entering cooling water temperature on small machine performance. At entering hot water temperatures of less than 180°F, substantial derating is necessary. For preliminary evaluation, the 85°F cooling water curve should be employed.

COMMERCIAL REFRIGERATION

Most commercial and industrial refrigeration applications involve process temperatures of less than 32°F and many are 0°F. As a result, the lithium bromide/water cycle is no longer able to meet the requirements, because water is used for the refrigerant. As a result, a fluid which is not subject to freezing at these temperatures is required. The most common type of absorption cycle employed for these applications is the water/ammonia cycle. In this case, water is the absorbent and ammonia is the refrigerant.

Use of water/ammonia equipment in conjunction with geothermal resources for commercial refrigeration applications is influenced by some of the same considerations as space cooling applications. Figure 13.5 illustrates the most important of these. As refrigeration temperature is reduced, the required hot water input temperature is increased. Because most commercial and industrial refrigeration applications occur at temperatures below 32°F, required heat input temperatures must be at least 230°F. It should also be remembered that the required evaporation temperature is 10 to 15°F below the process temperature. For example, for a +20°F cold storage application, a 5°F evaporation temperature would be required.

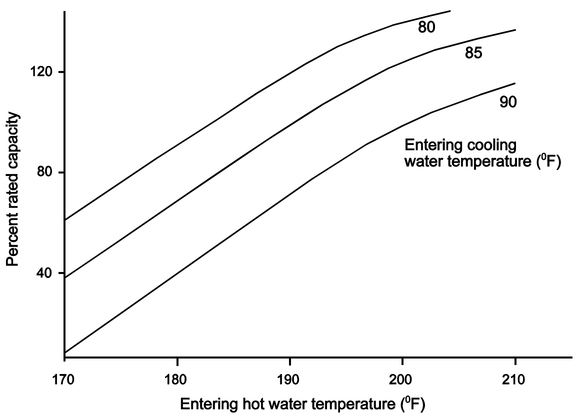

Figure 5. Small tonnage absorption equipment performance.

Research suggests a minimum hot water temperature of 275°F would be required. There is not a large number of geothermal resources in this temperature range. For geothermal resources that produce temperatures in this range, it is likely that small scale power generation would be competing consideration unless cascaded uses are employed.

Another consideration for refrigeration applications. That is the COP for most applications is likely to be less than 0.55. As a result, hot water flow requirements are substantial. In addition, the cooling tower requirements, as discussed above, are much larger than for equivalently sized vapor compression equipment.

CONCLUSION

In conclusion, it is necessary to evaluate the following factors when considering a geothermal/absorption cooling application for space conditioning.

Resource temperature

Substantial derating factors must be applied to equipment at temperatures less than 220°F. Very high resource temperatures or two-stage are required for low-temperature refrigeration.

Absorption machine hot water requirements compared to space heating flow requirements

Incremental well and pumping costs should be applied to the absorption machine.

Refrigeration capacity required

Larger machines have lower incremental capital costs on a $/ton basis. Coupled with the larger displaced energy, this result in a more positive economic picture.

Annual cooling load for space conditioning, in full load hours or for process cooling, in terms of load factor

Obviously higher utilization of the equipment results in more rapid payout.

Pumping power for resources with unusually low static water levels or drawdowns

Pumping power may approach 50% of high efficiency electric chiller consumption.

Utility rates

As with any conservation project, high utility rates for both consumption and demand result in better system economics.

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Means, R. S., 1996. "1996 Means Mechanical Cost Data," R. S. Means, Inc., Kingston, MA.

Yazaki Corporation, undated. Yazaki Gas & Solar Air Conditioning Equipment - Cat. No. 15.3 AME, Yazaki Corporation, Tokyo, Japan.

GEOTHERMAL PIPELINE

Progress and Development Update
Geothermal Progress Monitor

BOOK REVIEWS

Geothermal Direct-Use Engineering and Design Guidebook, 3rd Edition (Lund, J. W.; Lienau, P. J and B. C. Lunis, editors). This is an update of the popular direct-use guidebook, last published in 1991. All of the 19 chapters have been revised based on technical assistance experience at the Geo-Heat Center and reflecting current trends in industry. The 470-page book covers material on Lessons Learned, Nature of Geothermal Resources, Exploration for Direct-Heat Resources, Geothermal Fluid Sampling Techniques, Drilling and Well Construction, Well Testing and Reservoir Evaluation, Materials Selection Guidelines, Well Pumps, Piping, Heat Exchangers, Space Heating Equipment, Absorption Refrigeration, Greenhouses, Aquaculture, Industrial Applications, Engineering Cost Analysis, Regulatory and Commercial Aspects, and Environmental Considerations. The Guidebook can be ordered from the Geo-Heat Center, Oregon Institute of Technology, Klamath Falls, OR 97601 (Ph: 541-885-1750) for a cost of $49.00.

The Geysers Album: Five Eras of Geothermal History. This is a beautiful new 52-page book form the California Division of Oil, Gas and Geothermal Resources, authored by Susan Hodgson (1997). It presents a panorama of geothermal events at The Geysers in northern California, covering five historical eras overlapping in a mosaic of time. It is an intriguing story about man's interaction with geothermal energy in this unique resource area - including a wealth of photographs and information never before published. The eras covered are (1) untouched wilderness, (2) the time when Indians in the region first found the area 12,000 years ago, (2) organized tourism that began around 1848 and ended in 1980 when the last remnants of The Geysers Resort were razed, (3) the age of electric power development to light The Geysers Resort starting in 1921 and ending in the early 1930's, and (5) modern power development which began in 1955, when the first modern steam well was drilled in the area. Copies are available for only $5.00 each from the California Division of Oil, Gas and Geothermal Resources, 801 K Street, MS-20-20, Sacramento, CA 95814-3530, telephone: 916-445-9686.

Charging Ahead - The Business of Renewable Energy and What It Means for America, by John J. Berger. A John Macrae Book, Henry Holt and Company, NY (1997) - $30.00. This is 399-page book covering alternative energy, including nine chapters on solar power, five on wind power, two on bioenergy, one on geothermal energy, and several general chapters. Chapter 20 - Underground Power - covers "Geothermal's Rocky Road." The chapter is primarily devoted to geothermal power generation, covering the early history of development in New Zealand, Italy and The Geysers. The Geysers section describes work by B.C. McCabe, Joseph Aidlin and Carel Otte. "The Salton Sea Challenge", "Foreign Development Opportunities" and "Hot Rocks" are also discussed. Geothermal is described as "An Underutilized Resource", as ". . . the use has been slight nationwide, due in part to relatively low levels of federal research-and-development support and to low-cost fossil fuels....Globally, geothermal energy exists in vast quantities, including many locations throughout the United States, especially in the West and abroad. However, only a small part of the resource base can be developed economically because of current technology limitations. If one day the limitations are overcome, American would have an energy supply far beyond its needs." The chapter also address the environmental benefits of geothermal power as compared to fossil fuel sources, and that it can provide base load as well as peaking power, "...an important feature relative to other non-base load renewables." "Yet with all these attractive features and the potential for expanded capacity, geothermal energy has not been a major national research priority." This is what we have been preaching for years!!!

Mineral and Thermal Groundwater Resources by Marius Albu of Romania, David Banks of Norway and Harriet Nash of the UK, Chapman & Hall, London and New York (1997). This is a 447-page book with an exorbitant price of $150. Part One covers the history, use, and origins of mineral and thermal waters, thus it is not limited to geothermal fluids. Several chapters are devoted to exploring for, modeling and exploiting mineral water sources, along with concerns for the environmental issues and conservation of the resource. Part Two deals with case studies from Iceland, England, Lithuania, the north Caucasus area of CIS (the former Soviet Union), Norway, The Czech Republic and two areas of Romania. All of these areas have made use of mineral waters and some have developed spas based on the therapeutic use of these fluids. It does have geothermal applications in two chapters on "Thermal Water Systems," which address the problems of scaling and corrosion, drilling and borehole construction, transport, heat recoverability, heat regeneration, and resource management. It also discusses geophysical and geochemical investigations and analysis, and the estimation of the potential yield of and quantity of heat stored in a resource. The section on Iceland discusses the Reykjavik and Svartsengi district heating systems in detail, and would be of interest to geothermal readers, however, there is very little engineering design information elsewhere in the book. A great number of geothermal experts are referenced in the book, both U.S. and international scientists. Of course, the chapter on Norway deals with cold water resources, but they do bottle and export the natural spring water. Many of the chapters use fairly complicated mathematical analysis such as numerical modeling of groundwater systems and there are some fairly involved chemistry discussions, including geothermometry. This is an excellent book for those interested in developing a bottled mineral water source or therapeutic spa for commercial purposes, for either cold and hot water resources, however the reader does need a technical background in order to appreciate some of the material. It is a well written book, and covers both the historical perspective as well as providing assistance for the future developer and entrepreneur.

MEETINGS

New Zealand Geothermal Association Direct Heat Development Seminar, Quality Inn Conference Centre, Taupo, New Zealand, July 2-3, 1998. Contact: Ian Thain (email: i.a.thain@xtra.co.nz), Geothermal & Energy Technical Services, Ltd., 19 Cherry Lane, Acacia Bay, Taupo, New Zealand (phone: 64 (7) 378-1707).

33rd Intersociety Energy Conversion Engineering Conference (IECEC), Sheraton Colorado Springs Hotel, Colorado Springs, CO, August 2-6, 1998. “Meeting Global Energy and Environmental Needs.” Contact: ANS Proceedings Office (email: www.inspi.ufl.edu/IECED98), IECEC ‘98, 555 North Kensington Ave., La Grange Park, IL 60526. (Phone: (708) 579-8253).

17th World Energy Council Congress, George R. Brown Convention Center, Houston, TX, September 13-17, 1998. Contact: World Energy Congress, Inc., 1620 Eye Street, N.W., Suite 1050, Washington, D.C. 20006.

International Summer School, Ponta Delgada and Ribeira Grande, Azores, Portugal, September 14-19, 1998. International Workshop on Heating Greenhouses with Geothermal Energy (14 Sept.), International Seminar on Electricity Production of Geothermal Energy (15 Sept.), and International Course on Economy of Integrated Geothermal Projects (16-18 Sept.). Contact: Dr. Kiril Popovski, International Summer School on Direct Application of Geothermal Energy, ul. Dame Gruev br.1-III/16, 91000 Skopje, Macedonia, or Marnell Dickson, IIRG (email: marnell@iirg.pi.cnr.it), Piazza Solferino 2, 56126 Pisa, Italy (phone: 39 (50) 46-069

Geothermal Resources Council Annual Meeting, Town & Country Hotel, San Diego, CA, September 20-23, 1998. Contact: GRC (email: earth307@concentric.net or www.geothermal.org), PO Box 1350, Davis, CA 95617 (phone: (530) 758-2360).

Geothermal Village Power Workshop, Eldorado Hotel, Reno, NV, November 4-6, 1998. (USDOE, CEC, Sandia Nat. Lab., and GRC). Contact: Geothermal Resources Council, Davis, CA or John T. Finger, Sandia National Laboratories, PO Box 5800, MS 1033, Albuquerque, NM 87185 (phone: (505) 844-8089).

Energex ‘98 Conference, Bahrain Conference Centre, Holiday Inn Hotel, Myoneme, Bahrain, November 19-21, 1998. Contact: Conference Secretariat, (email: info@hfasin.com). Dr. W. E. Alnaser, Dean of Scientific Research, University of Bahrain, PO Box 32028, State of Bahrain (Ph: 973-683278/688396/682582).