Referring again to Figure 11.12, 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.

A good design procedure is currently lacking. Culver and Reistad (1978) presented a computer program that predicts DHE output to within 10 to 15% if the mixing ratio is known. The problem is, there is no way of predicting mixing ratio except by experience in a specific aquifer and then probably only over a fairly wide range as noted above. The procedure was written in FORTRAN but has been converted to HP-85 Basic by Pan (1983). The program enables the user to choose optimum geometric parameters to match a DHE to an energy load if a mixing ratio is assumed. The program does not include a permeability variable, nor does it take thermal storage into account. In wells with good permeability, thermal storage may not be a significant factor. Experience in Reno indicates that for low permeability wells, thermal storage is very important. With low permeability, a convection promoter can promote thermal storage and, thereby, increase non-steady-state output.

Permeability can be rather accurately estimated using relatively simple Hvorslev plots as described in Chapter 7, "Well Testing." Relating the permeability thus obtained to mixing ratios typical in other permeabilities could give an estimate of the mixing ratio that could be used in the computer program. Data are limited to very-high and very-low permeability situations. Middle ground data are not available.

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The mass flow through an area, A, perpendicular to the flow is therefore shown as:

$$v\ A\ d = K\ A\ d\ \Delta h/\Delta l$$

where d is the density of the water. The steady-state heat flow can be found by:

$$q = K\ A\ d\ c\ (T_o - T_l)\ \Delta h/\Delta l$$

where:

- A = cross section of well in the aquifer or the perforated section
- d = density of water
- c = specific heat
- T_o = aquifer temperature
- T_l = temperature of water returning to the aquifer.

Multiplying the above by (1 - Rm), or • 0.5 to 0.6, the expected steady-state DHE output can be determined.

The most important factor in the above equation is K. This value can vary by many orders of magnitude (even in the same aquifer) depending on whether major fractures are intersected, or drilling mud or debris partially clogs the aquifer, etc. The variation between aquifers can be even greater.

Based on short-term pump tests to determine hydraulic conductivity and an estimated 1% hydraulic gradient, the specific velocity in the Moana area of Reno is estimated at 1 to • 3 ft/y. The hot aquifer is generally encountered in mixed or interbedded layers of fine sand and silt stone. In Klamath Falls, where the hot aquifer is in highly fractured basalt and coarse pumice, specific velocity is estimated at 20 to 150 ft/day, perhaps higher in localized areas. Values of K in seven wells in Moana were estimated at 3.28×10^{-7} to 3.28×10^{-4} ft/s. The K in Klamath Falls is estimated to be at least 4.9 ft/s. This implies a factor of 10 thousand to 10 million difference in the steady-state output. Differences by a factor of 100 have been measured, and some wells in Moana have been abandoned because they could not provide enough heat, even for domestic hot water.

Many DHE wells in Moana 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 thermo-statically controlled. The system is designed to switch on the pump if the DHE temperature drops below some predetermined level, usually approximately 120°F. 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. This has been experimentally verified using a model well in a laboratory at the University of Auckland in New Zealand. With an electric heater at the model well bottom, and a convector pipe and DHE installed, a convection cell was induced and an apparent mixing ratio of 0.954 was calculated, similar to what might be expected in a very low permeability aquifer (Freeston and Pan, 1983).

As mentioned at the first of this section, the second method of using a DHE is extracting heat stored in surrounding rocks.

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.

It is now presumed, but not verified, that if a DHE were operating and then turned off, the convection cell would reverse in the case of the undersized casing and when the DHE is inside the convection promoter. If the DHE is outside the promoter, the cell direction would remain the same as when the DHE is extracting heat.

DHEs are principally used in space and domestic water heating applications: homes, schools, small commercial buildings and greenhouses, with the resulting intermittent 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.

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).

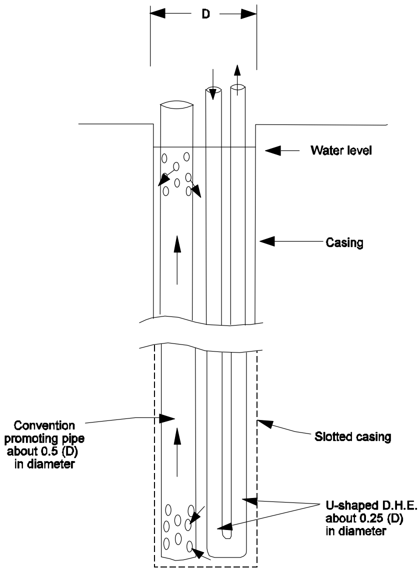

Figure 11.13 Convection promoter pipe with DHE (Allis and James, 1979).

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).

Maximum convection rates are obtained when the casing or convector pipe are insulated. This maintains the temperature and density difference between the cell legs. Non-metallic pipe is preferred. Although corrosion products help insulate the pipe, scaling does not normally occur to any great degree because the casing or convector are the same temperature as the water.

11.4.4 <u>Design Considerations</u>

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 <u>new</u> hot fluid to enter the well or, more probably, allowing <u>used</u> 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.

If K is the hydraulic conductivity and $\Delta h/\Delta l$ is the hydraulic gradient, by Darcy's Law, the specific velocity through the aquifer is given by:

$$v = K\ \Delta h/\Delta l.$$

$$U_{frp} = \frac{1}{\frac{1}{250} + \frac{0.080}{2.5} + \frac{1}{250}}$$

= 25 Btu/h ft^2 °F (no scale).

While the difference is significant, it is less than might be anticipated by comparing only the thermal conductivity of the two pipe materials. Usually, the depth of the well to reach geothermal fluids will accommodate the additional pipe required and the long life justifies the use of non-metallic pipe.

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 thermal 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.

11.4.3 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 11.12). 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.

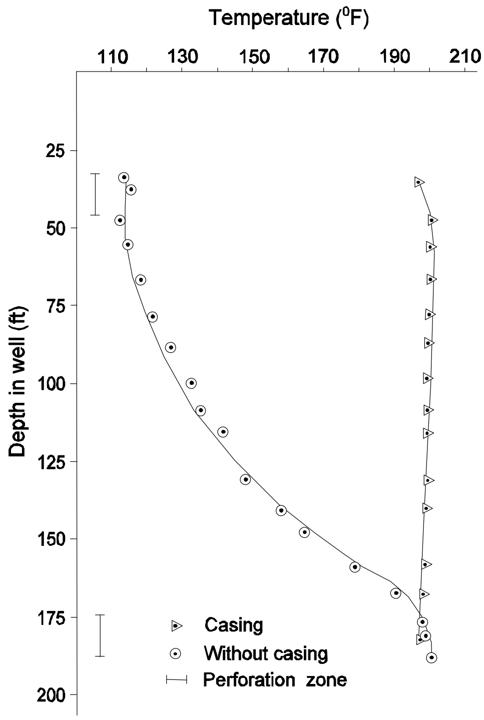

Figure 11.12 **Temperature vs. depth with and without casing (Culver and Reistad, 1978).**

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 11.13). The convector pipe is simply a pipe open at both ends, suspended in the well above the bottom and below the

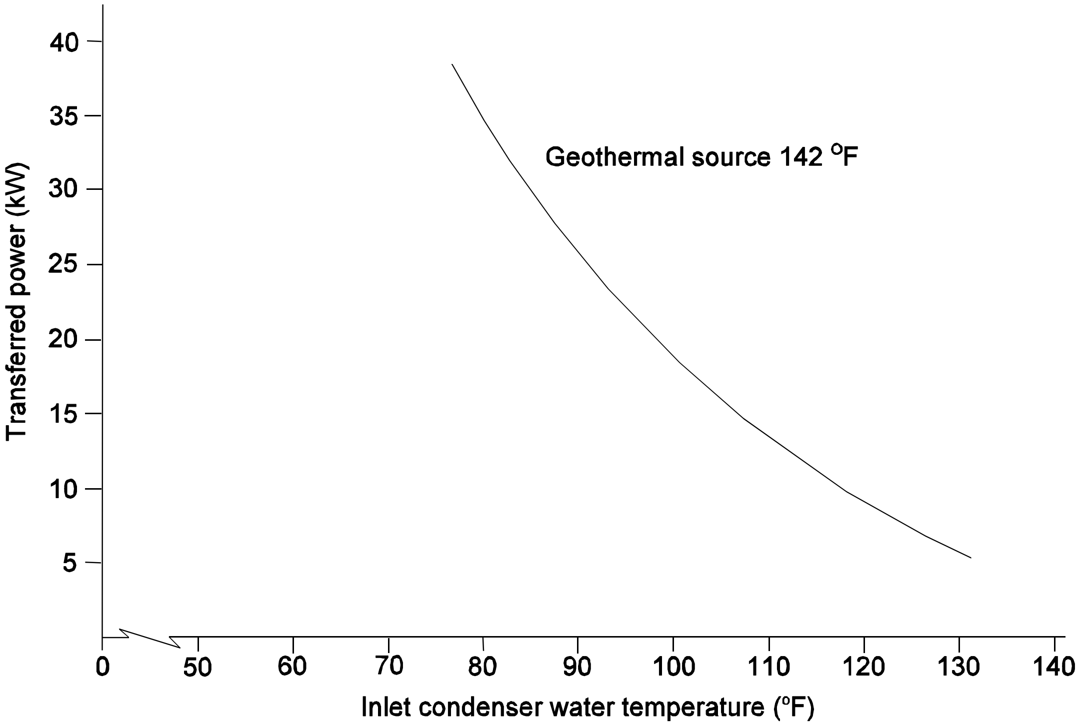

Figure 11.11 Performance of 20-kW geothermal convector (Cannaviello, et al., 1982).

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.

Fiberglass pipe is available in both NPT and a coarse 4 thread/in. male/female configuration in sizes at least down to 1-1/2-in. pipe.

DHE pipes should be approved for potable water, since it may heat domestic water and many geothermal re-sources supply water for drinking and spas. Although there are no known installations, chlorinated polyvinyl (CPVC) or irradiation cross-linked polyethylene are good candidates.

Thermal conductivity is much lower for non-metallic pipe than for metallic pipe. The value for fiberglass epoxy pipe is 2.5 Btu/h ft^2 °F/in. And polybutylene is 1.5 Btu/h ft^2°F/in; while, steel is 460 Btu/h ft^2°F/in. Also, non-metallic pipe is typically thicker than metallic. However, the overall thermal conductivity is a function of the conductivity of the pipe, the film coefficients on both the inside and outside, and of the scale. Scaling is an important consideration because in many geothermal fluids, significant scaling on metallic DHE pipes occurs; but, the scale will not build up on the non-metallic pipes. It is not unusual to see 1/16 in. of scale buildup on the geothermal side of DHE pipes, at least in Klamath Falls geothermal fluids. Scale can be assumed to have a thermal conductivity similar to limestone and concrete or about 7 Btu/h ft^2°F/in.

The overall heat transfer coefficient can be calculated from:

$$U = \frac{1}{\frac{1}{ho} + \frac{t}{kp} + \frac{t}{ks} + \frac{1}{hi}}$$

where:

U = overall thermal conductivity (Btu/h ft^2°F/in.)
ho = outside film coefficient
hi = inside film coefficient
t = pipe or scale thickness (in.)
kp = thermal conductivity of pipe material (Btu/h ft^2 °F/in.)
ks = thermal conductivity of scale.

Assuming film coefficients of 250 and using values cited earlier for steel, fiberglass and scale, the thermal conduc-tivities of typical pipes are:

$$Usteel = \frac{1}{\frac{1}{250} + \frac{0.154}{460} + \frac{0.0625}{7} + \frac{1}{250}}$$

= 58 Btu/h ft^2°F (with scale).

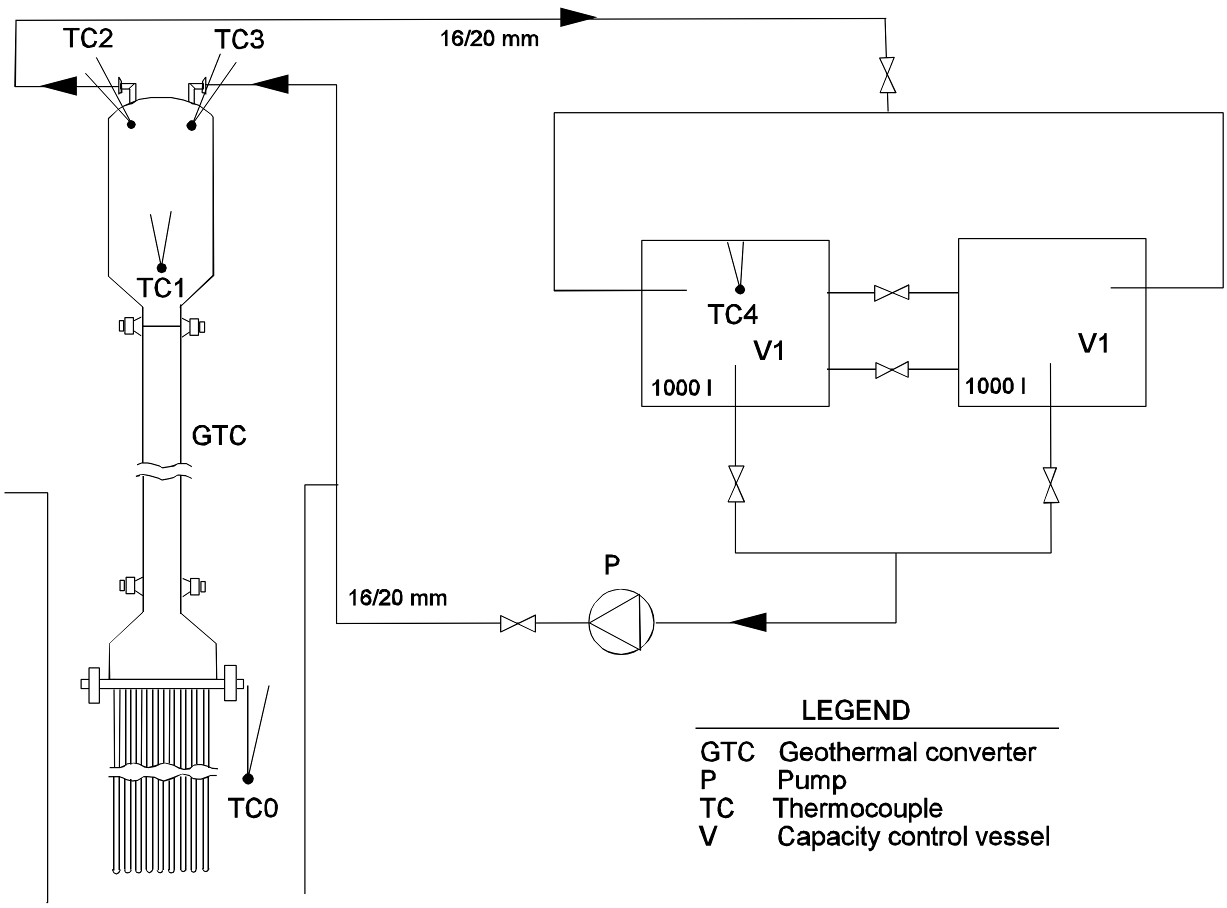

Figure 11.10 Schematic of experimental loop (Cannaviello, et al., 1982).

3. DHEs for heat pump applications. These will be covered in a later section.

4. Combined pump and DHE in a single well.

5. A 20-kWt, 16-ft prototype heat pipe system was successfully tested in the Agnano geothermal field in southern Italy (Connaviello, et al., 1982) (Figures 11.10 and 11.11).

6. A co-axial DHE was described and analyzed for conduction heat transfer only (Horne, 1980). Pan (1983) reported that heat output from a co-axial and U-tube exchanger were very nearly equal in a laboratory well simulation test that included convection. There are no known co-axial DHE installations except for heat pumps.

7. Research and field testing of promoters with downhole heat exchanger have been carried out in New Zealand (Allis and James, 1979; Freeston and Pan, 1983; Dunstall and Freeston, 1990). The promoter, a small perforated pipe inserted in the well, is used where the casing is not perforated.

11.4.2 <u>Materials</u>

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 nonmetallic pipe, both fiberglass-reinforced epoxy and polybutylene. 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 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, 202°F, 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.

11.4.1 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 11.9 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 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.

Other types of DHEs in use are:

1. Short multiple tubes with headers at each end. These are somewhat similar to a tube bundle in a shell and tube exchanger, but more open to allow for natural circulation rather than forced circulation. The tubes are suspended just below the upper perforations.

2. Straight pipes extending to near the well bottom with coils of copper or steel pipe at the ends.

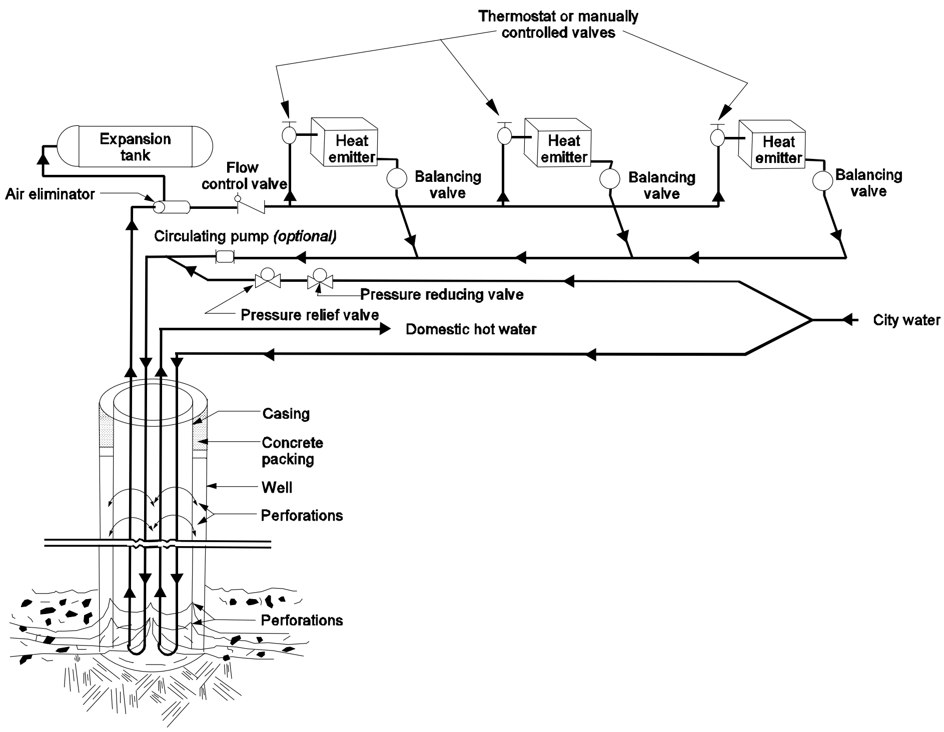

Figure 11.9 Typical hot-water distribution system using a downhole heat exchanger (Culver and Reistad, 1978).

Costs for both types of exchangers are combined on Figure 11.8 for units of less than 100 ft^2 heat transfer area. It is apparent that brazed plate units offer a significant savings for exchangers in the 2 - 30 ft^2 size range.

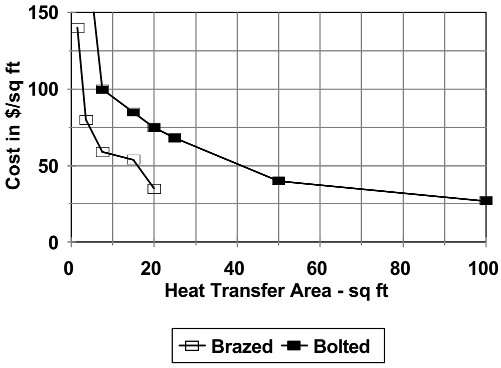

Figure 11.8 Plate heat exchanger cost comparison.

11.3.4 Brazed Plate Heat Exchanger Performance in Geothermal Fluids

A key factor in the determination of the economics of brazed plate heat exchangers is their expected service life in geothermal fluids. In order to evaluate this issue, plate heat exchangers were placed in service in three different geothermal fluids. The three locations for the installations (Boise, ID; Pagosa Springs, CO and Klamath Falls, OR) were chosen specifically due to the previous experiences with copper in geothermal fluids at these sites. Fluid chemistry for the three locations are detailed in Table 11.6.

Table 11.6 Test Site Fluid Chemistry*

	Klamath Falls, OR	Boise, ID	Pagosa Springs, CO
H_2S	0.5 - 1.5	0.3	5.0
Temp.	193°	176°	140°
TDS	795	290	3160
pH	8.6	8.2	6.7
Ca	26.0	2.0	240.0
F	1.50	14.0	N/A
Cl	51.0	10.0	160.0
CO_3	15.0	4.0	0
HCO_3	20.0	70.0	810.0
Na	205.0	90.0	640.0
K	1.50	1.6	87.0
SO_4	330.0	23.0	1520.0
SiO_2	48.0	160.0	61.4

* All values in mg/L except temperature (°F) and pH

In the past, the performance of copper tubing in Boise geothermal fluids has been good with water-to-air heating coils (with copper tubes) lasting as long as 10 years (Griffiths, 1990). In Klamath Falls, failure of copper tubing has occurred in approximately half this time with leaks reported in as little as 5 to 7 years. Pagosa Springs fluids have demonstrated the most aggressive reaction to copper with some failures as early as 2 years of service (Martinez, 1990). In all cases, these failures have been traced to corrosion promoted largely by hydrogen sulphide (H_2S). H_2S is present to some extent in virtually all geothermal fluids.

In order to evaluate the influence of fluid chemistry on the braze material, a test program involving four heat exchangers was developed. Three of the units were exposed to the geothermal fluid and a fourth was used as a control. In each location, the heat exchanger was connected to a continuous source of geothermal fluid with a flow rate of approximately 1 gpm. The Boise unit remained in place for 46 weeks, the Klamath Falls unit for 55 weeks and the Pagosa Springs exchanger for 26 weeks. All four heat exchangers were then forwarded to an engineering firm specializing in materials analysis.

Based on this limited testing, brazed plate heat exchangers of the design similar to these 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.

11.4 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

Consider a heat exchanger in which geothermal fluid enters the hot side at 180º and cools to 140º. Process water enters the cold side at 100º and is raised to 150º.

For this case:

$$\Delta Tm = 150^o - 100^o = 50^o$$

$$LMTD = \frac{(140 - 100) - (180 - 150)}{\ln\frac{(140 - 100)}{(180 - 150)}}$$

$$= 34.8^o$$

$$NTU = 50^o/34.8$$
$$= 1.44$$

Consider a second case in which we wish to heat the process water to a temperature closer to the geothermal fluid:

Geothermal (hot) side 180º - 140º
Process (cold) side 175º - 125º

For this case:

$$\Delta Tm = 175 - 125 = 50^o$$

$$LMTD = \frac{(140 - 125) - (180 - 175)}{\ln\frac{(140 - 125)}{(180 - 175)}}$$

$$= 9.1^o$$

$$NTU = 50/9.1$$
$$= 5.49$$

The importance of the NTU value lies in the fact that heat exchangers are capable of generating a given NTU for each fluid pass. The value is dependent upon their specific construction. For plate heat exchangers, depending upon plate design, an NTU of 0.6 to 4 per pass is generally possible.

Using a conservative value of 3, this would place a upper limit on the type of geothermal application to which single-pass brazed plate heat exchangers could be applied. Of our two examples, only the first would be within the capabilities of a brazed plate heat exchanger

Table 11.5 provides a broader view of the affect of this limitation in single-pass performance.

The line indicates the limits of the brazed plate units based on an NTU of 3.0 per pass. Applications which fall above the line would be within the capabilities of brazed plate units; while, applications below the line would require a multiple pass heat exchanger.

Table 11.5 Brazed Plate Heat Exchanger Application Limitations (Based on an NTU of 3.0 per pass)

	LMTD				
Δtm	**2**	**5**	**10**	**15**	**20**
10	**5**	**2**	**1**	**0.67**	**0.5**
20	**10**	**4**	**2**	**1.33**	**1.0**
30	**15**	**6**	**3**	**2.0**	**1.5**
40	**20**	**8**	**4**	**2.67**	**2.0**
50	**25**	**10**	**5**	**3.33**	**2.5**
60	**30**	**12**	**6**	**4.0**	**3.0**

In summary, brazed plate heat exchangers would, in most cases, be limited to applications characterized by greater than 10º log-mean temperature differences, flow of less than 100 gpm and heat transfer area of less than 200 ft^2.

11.3.3 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 11. 7 presents 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.

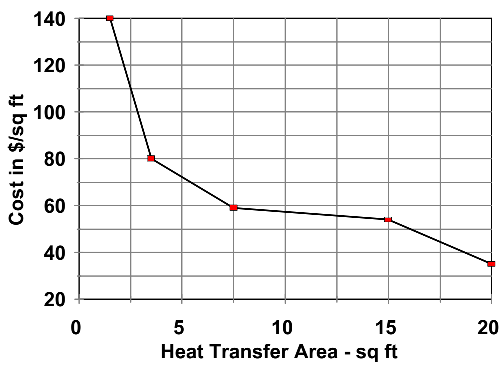

Figure 11.7 Brazed plate heat exchanger.

Protective shields for heat exchangers are typically included in quotes from vendors. These shields are in-stalled over the plate pack and are intended to protect the plates from damage. Cost of the shields is approximately 3% to 5% of exchanger cost. Unless there is regular activ-ity around the exchanger by personnel not associated with the mechanical equipment, these shields are not necessary.

Piping and location of the heat exchanger should be designed to allow easy access to the unit for disassembly and cleaning. If piping must be attached to the movable end plates (sometimes necessary in multi-pass designs), the piping should be of flanged or grooved end material which allows removal for maintenance purposes. Sufficient clear-ance should be allowed for plate removal from the frame.

It is useful to specify provision of at least one extra of each type of plate in the exchanger (with gaskets) for use by the owner during maintenance operations. Although it is not necessary to replace gaskets each time the exchanger is opened, should damage occur, the availability of the spare plates and gaskets would eliminate unneccsary downtime. For exchangers which use glue to attach the gaskets, stand-by plates should have the gaskets already glued in place. Most gasket glues require an 18- to 24-hour curing period before use. Pre-gluing the gaskets on the plates eliminates delay.

Maintenance personnel should be carefully instructed regarding plate cleaning and retorquing procedures. Metal brushes should be avoided for plate cleaning. Scratching of the plate surface can result in premature failure. If a metal brush is absolutely necessary, it should be of the same alloy as the plate (stainless brush for stainless plates). Retorqu-ing the exchanger should proceed carefully adhering to the manufacturer's tightening sequence and plate pack dimensions. Over torquing can cause gasket failture.

11.3 BRAZED PLATE HEAT EXCHANGERS

11.3.1 Construction

The brazed plate unit as shown in Figure 11.6 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.

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

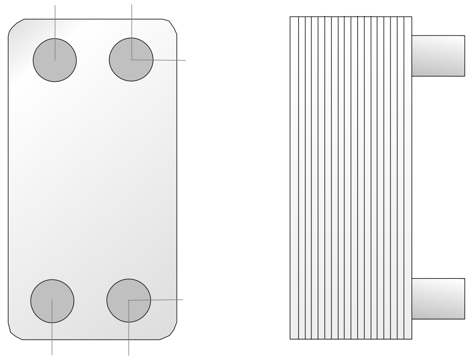

Figure 11. 6 Brazed plate heat exchanger.

alloys are generally avoided in geothermal system construc-tion. 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.

11.3.2 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 trans-fer 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 °F in most applications (SWEP, 1980) (ITT, 1988).

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.

This limitation is best illustrated through the Number of Transfer Units (NTU) approach to heat exchanger anal-ysis. The NTU is a dimensionless value which charac-terizes the performance of a heat exchanger based upon the log-mean temperature difference and the temperature change occurring in the unit. It can be expressed as follows:

$$NTU = \Delta tm/LMTD$$

where: Δtm = the largest temperature change occurring to a fluid in the heat exchanger

LMTD = log-mean temperature difference

An example best illustrates the use of these values.

Table 11.4 Vendor Quote[1] Results for Heat Exchanger Example (1997)

Case	Geothermal In/Out	Building In/Out	Flow Geo/Bldg.	Pressure Drop Geo/Bldg.	"U" Btu/hr · ft^2 · °F	Area ft^2	Cost $
1	170/130.6	115/132	375/900	4.5/12.2	1129	264	4800
2	170/130.4	122/138.5	375/900	3.0/15.1	1151	369	5660
3	170/130.1	125/141.6	375/900	3.9/12.0	1096	503	7100
4	170/130.2	115/131.6	375/900	2.3/7.1	729	410	6000
5	170/130.4	122/138.5	375/900	1.9/7.0	909	468	6700
6	170/130.1	125/141.6	375/900	2.9/7.0	925	598	7670

1. All selections based on 7,500,000 Btu/hr load.

The first conclusions to be drawn from these values is that the cost of the heat exchanger is relatively modest in the context of the system as a whole. The most expensive exchanger (Case #6) is $7670.

The trends in overall "U" value with respect to pressure drop are also evident in this table. Cases 1, 2, and 3 were selected for somewhat higher pressure drop values (12 - 15 psi on high-flow side); while, Cases 4, 5, and 6 were selected for lower values (approximately 7.0 psi on high-flow side). The first three selections result in an average U-value of 1100; whereas, the last three average about 860 Btu/hr ft^2 °F.

Comparing Cases 1 and 4, the use of the lower pressure drop heat exchanger would reduce pump energy by approximately 10,000 kWh assuming 1700 hr/yr operation for the well pump and 2500 hr/yr for the building loop pump. This would result in a simple payback on the added capital cost (for the lower pressure drop heat exchanger) of less than 2.5 years at $0.05/kWh.

11.2.5 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 11.5 presents a plot of plate heat exchanger costs in 1996 dollars/ft^2 of heat transfer area. Since heat transfer area takes into account duty, temperature difference and fouling, it is the most useful index for preliminary costing.

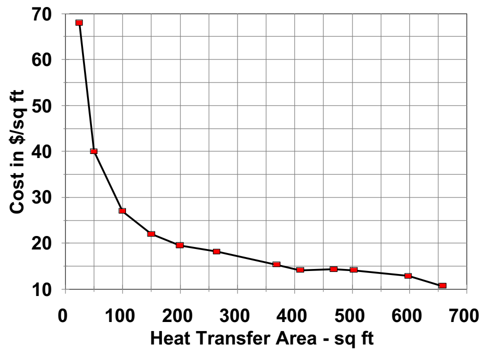

Figure 11.5 Plate heat exchanger cost for Buna-N gaskets and 316 stainless steel plates (1996).

The data used to generate Figure 11.5 are from a number of manufacturer's quotes for various geothermal applications. The costs are based on 316 stainless steel plate construction and medium nitrile gaskets.

11.2.6 Installation and Maintenance

The question of whether to use multiple heat exchangers instead of a single unit is more a function of the building use than system design. Going to a multiple heat exchanger design always increases costs. In general, two exchangers should only be necessary in applications where system downtime cannot be tolerated (detention facilities, hospitals, computer facilities, etc.). The time required to disassemble and clean plate heat exchangers is a function of size and number of plates; but, in most applications, the work can be accomplished in less than 8 hours by two workers. Small units require less time and labor. This work can easily be accomplished during off-hours in a building used less than 24 hours per day.

energy consumption associated with the well pump (on the geothermal side) and the loop pump (on the building side). As an example, consider a system with the loop operating at 250 gpm and the well pump selected for 150 gpm. Assuming the well pump operates at 1500 hours per year and the loop pump 2500 hours per year (constant flow), the annual pumping energy consumption (loop and well) would be 6558 kWh for a heat exchanger with a 12.5 psi pressure drop and 3908 kWh for an exchanger with a 7.5 psi pressure drop. At \$0.08 per kWh, this amounts to a cost difference of \$212/year. A heat exchanger sized for average conditions at a 2,500,000 Btu/hr load would require about 25% (35 ft^2) additional surface area at 7.5 psi compared to 12.5 psi. At a value of \$20 per ft^2, this amounts to an additional heat exchanger cost of approximately \$700. As a result, the simple payback on the added cost of the heat exchanger (selected for lower pressure drop) would amount to about 3.3 years.

It is clear from this example, that plate heat exchangers should be selected for pressure drop values toward the low end of the range shown in Figure 11.4. Although this results in higher first cost of the heat exchangers, this cost is rapidly repaid through pumping energy savings.

In summary, preliminary calculations for plate heat exchangers can be made using the guidelines in Table 11.3:

Table 11.3 Plate Heat Exchanger Design - Parameters

Fouling factor (total)	0.0001 to 0.0002
Pressure drop	Less than 10 psi
Resulting overall heat transfer coefficient	750 to 900 Btu/hr·ft^2 °F
Plate materials	316 SS or Titanium
Gaskets	Medium nitrile rubber (NBR)

Although the final selection of plate heat exchangers is made by the vendor using proprietary software, in order to reduce the number of alternatives submitted to the vendor, preliminary calculations are generally made to refine the selection. The general procedure is to:

- Calculate heat exchanger surface requirements for the heating load based on available geothermal flow and temperature,
- Compare with desired operating conditions, and
- Re-select if necessary based on alternate pressure drop, approach temperature, etc.

Example selection:

Application - radiant floor heating in 350,000 ft^2 factory
Resource temperature - 170°F
Available geothermal flow - 375 gpm
Load - 7,500,000 Btu/hr
Building loop supply water temperature - 135°F max.

Geothermal temperature drop:

= 7,500,000 Btu/hr ÷ (500 · 375 gpm)
= 40°F.

Geothermal exit temperature:

= 170° - 40°
= 130°F.

Note: In order to provide uniform heat output, floor heating systems normally operate a low system Δt (10 - 25°). For this example, a Δt of 15°F will be assumed.

Heat exchanger temperatures:

Geothermal side	170° • 130°F
Building side	135° • 120°F

$\Delta t_1 = 170° - 135° = 35°$
$\Delta t_2 = 130° - 120° = 10°$

$$LMTD = \frac{35 - 10}{\ln(\frac{35}{10})}$$

= 19.9°F

Assuming a 10 psi pressure drop and a fouling factor of 0.00015, Figure 11.4 indicates an overall U-value of approximately 950 Btu/hr·ft^2 °F.

Using a LMTD convection factor of 0.90, required heat exchanger area is:

Area = 7,500,000 Btu/hr ÷ (19.9°F · 0.90 · 950 Btu/hr·ft^2 °F)

= 441 ft^2

Table 11.4 presents a summary of actual vendor quotes for this application under various temperature and pressure drop conditions.

These alternate selections assume constant loads which due to variations in floor output would not strictly be the case; but, these differences would be small.

Table 11.2 presents some recommended ranges for fouling factors in plate heat exchangers.

Table 11.2 Fouling Factors for Plate Heat Exchangers

Fluid	Fouling Factor ($ft^2 \cdot {}^oF \cdot hr/Btu$)
Distilled water	0.00005
Soft water	0.0001
Hard water	0.00025
Treated cooling tower water	0.0002
Sea water - ocean	0.00015
River water	0.00025
Engine jacket	0.0003

It is important to bear in mind the impact of the fouling factor on a heat exchanger size. For a nominal clean heat transfer coefficient of 1000 $Btu/hr \cdot ft^{2\,o}F$, a fouling allowance of 0.0001 would result in an additional surface requirement of roughly 10%. A fouling allowance of 0.0005 would require a 50% larger heat exchanger. Overly conservative fouling factor selection has a much larger impact upon plate heat exchanger surface area compared to shell-and-tube equipment due to the higher overall heat transfer of plate heat exchangers.

11.2.4 Selection

Final selection of plate heat exchangers is done by the vendor using proprietary selection software. There are some general rules of thumb, however, which allow the designer to refine the flows and temperatures prior to submitting values to the vendor.

Prior to discussing the selection of plate heat exchangers for geothermal applications, it is useful to review heat exchanger calculations in general. Heat exchangers, regardless of the type, are selected to transfer a specific quantity of heat under a specific set of conditions. The key parameter in the selection process is the heat transfer area required to accomplish this task. The general formula below describes this situation.

$$Q = U \times A \times LMTD \times C_f$$

where:
- Q = Heat load in Btu/hr
- U = Overall heat transfer coefficient in $Btu/hr \cdot ft^{2\,o}F$
- A = Area (ft^2)
- LMTD = Log mean temperature difference (^oF)
- C_f = LMTD correction factor (0.85 - 1.0 for most geothermal applications).

The log mean temperature difference is calculated using the difference between the entering and leaving temperatures of the two fluids according to the following relationship:

Fluid 1 t_{out1} • t_{in1}
Fluid 2 t_{in2} • t_{out2}

$$LMTD = \frac{\Delta t_1 - \Delta t_2}{\ln(\frac{\Delta t_1}{\Delta t_2})}$$

where: $\Delta t_1 = t_{out1} - t_{in2}$
$\Delta t_2 = t_{in1} - t_{out2}$

Example LMTD calculation:

Geothermal side	170° • 130°F
Building loop side	150° • 115°F

$\Delta t_1 = 20^o$ $\Delta t_2 = 15^o$

$$LMTD = \frac{20 - 15}{\ln(\frac{20}{15})} = 17.4^o$$

The overall heat transfer coefficient can be selected from Figure 11.4. For good quality geothermal fluid in applications where competent operating personnel maintain the system, there is no reason to specify fouling factors any higher than 0.00015. In fact, some designers use no fouling allowances. For systems which will be less carefully maintained, a value of 0.0002 may be used.

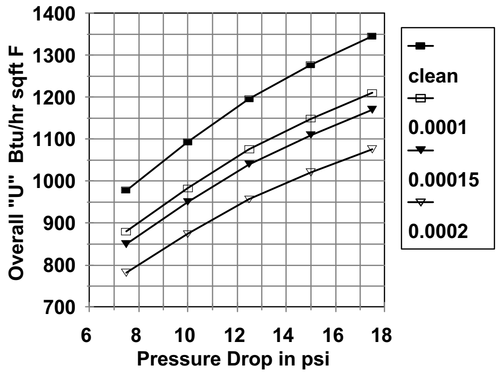

Figure 11.4 Performance of plate heat exchangers.

The question of permissible pressure drop is one which should be viewed in the context of the project. Obviously, the higher the pressure drop, the greater the

Austenitic stainless alloys with higher chromium and molybdenum contents could be recommended for this application also. These alloys, however, are generally not available as standard plate materials as is titanium (Ellis and Conover, 1981).

A typical application in which titanium has been employed is in geothermal systems that serve loads in which the secondary fluid is heavily chlorinated. The most common of these is swimming pools. The nature of swimming pools is such that the pool water is both high in chloride and oxygen content. As a result, titanium is the alloy generally selected. Plates made of 316 stainless steel, in the heat exchanger serving the swimming pool at Oregon Institute of Technology, Klamath Falls, Oregon, failed in less than 2 years as a result of localized corrosion.

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 11.1.

Table 11.1 Plate Heat Exchanger Gasket Materials (APV, Alfa-Laval, Tranter)

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.

The discussion of materials selection for both the plates and gaskets is based primarily upon the geothermal fluid characteristics. This assumes that the secondary fluid is of a relatively non-aggressive nature. Should the secondary fluid be a chemical process or other than treated water, additional materials selection considerations would apply.

Frame, Tie Bolts, and Fluid Connections

The frame of most plate heat exchangers is constructed of carbon steel. This is generally painted with an epoxy based material.

Tie bolts are of nickel-plated carbon steel. Alternative materials are available (stainless steel), though these are generally unnecessary for geothermal applications.

Standard connections are 150-lb flange-type of carbon steel construction (2-1/2 in. and larger). Connections of 1-1/2 in. and smaller are generally threaded. Alternate materials and configurations (threaded, grooved end, plain end, etc.) are available.

11.2.3 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 some-times useful in geothermal applications. This high thermal performance does come at the expense of a somewhat higher pressure drop. Figure 11.4 presents a generalized relationship for overall heat transfer and pressure drop in plate exchangers, based on several different total fouling factors .

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

11.2.2 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 offer the alloys listed below:

1. 304 Stainless Steel
2. 316 Stainless Steel
3. 317 Stainless Steel
4. Titanium
5. Tantalum
6. Incaloy 825
7. Hastelloy
8. Inconel
9. Aluminum Bronze
10. Monel.

In addition to these, a larger number of optional alloys are available by special order.

Most manufacturers will quote either 304 or 316 stainless steel as 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. This is illustrated in Figure 11.3. This figure contains two curves, one for 304 and one for 316. At temperature/chloride concentrations which fall into the region below the curve, the particular alloy in question is considered safe to use. Combinations of temperature and chloride content that are located above the curve offer the potential for localized pitting and crevice corrosion. Fluid characteristics above the curve for a particular alloy do not guarantee that corrosion will absolutely occur. However, this curve, based on oxygen-free environments, does provide a useful guide for plate selection. 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.

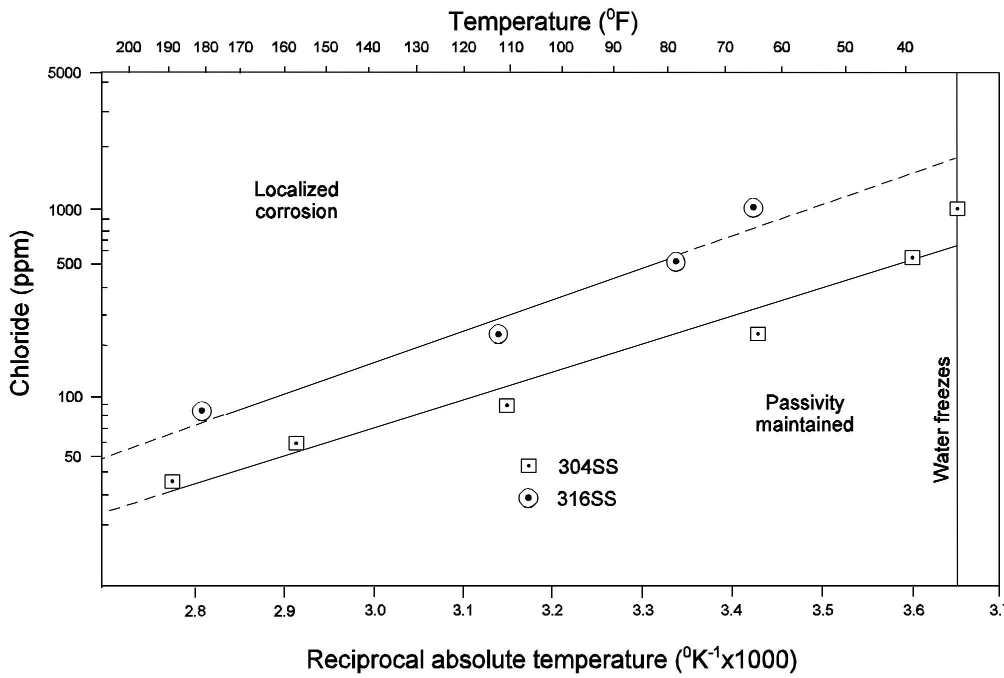

Figure 11.3 Chloride required to produce localized corrosion of Type 304 and Type 316 as a function of temperature (Efird and Moller, 1978).

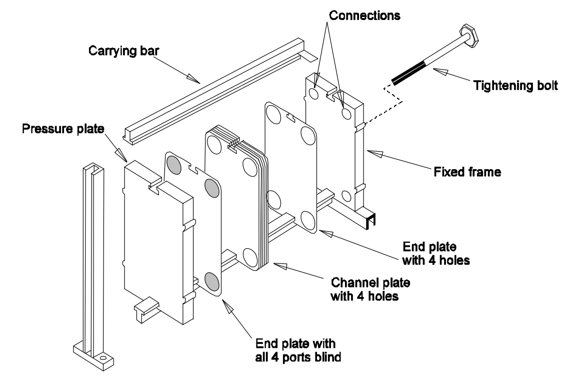

Figure 11.1 The plate heat exchanger.

Figure 11.2 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.

11.2.1 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, 400ºF, and 300 psig (Tranter, undated).

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.

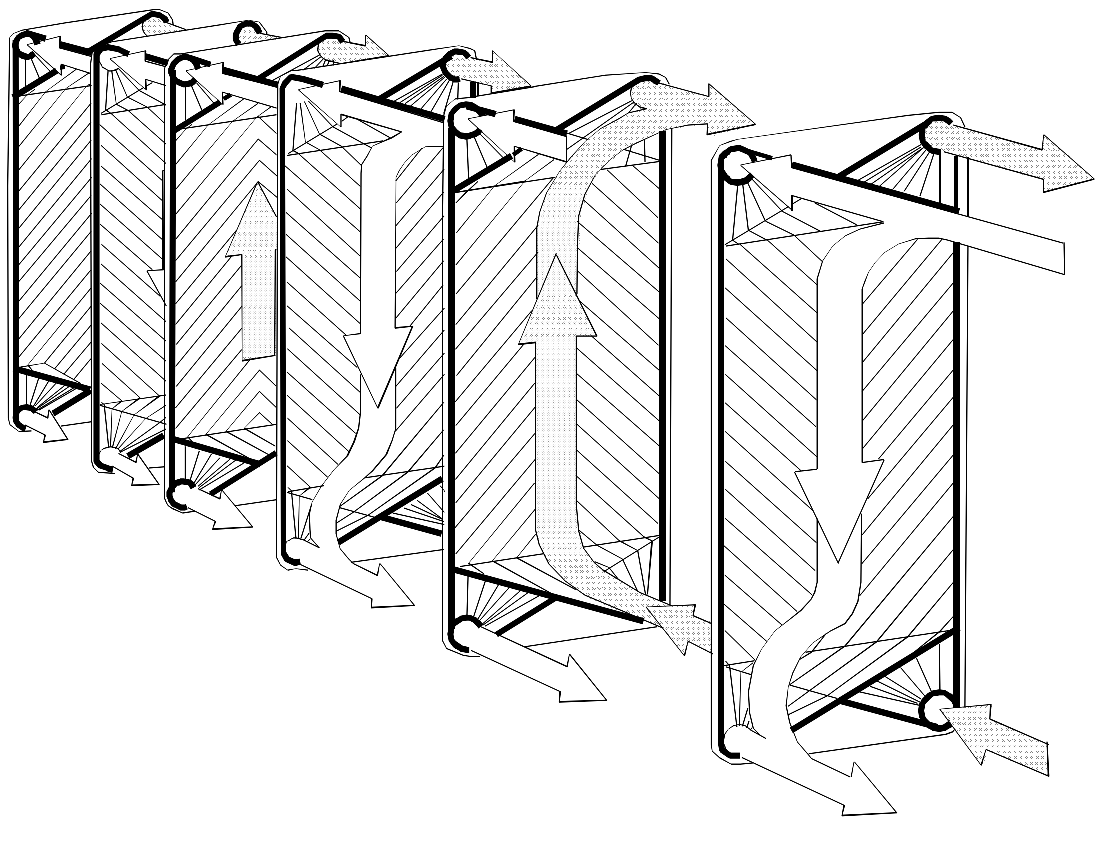

Figure 11.2 Nature of fluid flow through the plate heat exchanger.

CHAPTER 11
HEAT EXCHANGERS

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11.1 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.

Shell and tube heat exchangers play only a minor role in low-temperature, direct-use systems. These units have been in common use in industrial applications for many years and, as a result, are well understood. For these reasons, shell and tube heat exchangers will not be covered in this chapter.

11.2 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.

   Plate heat exchangers are capable of nominal approach temperatures of 10ºF compared to a nominal 20ºF for shell and tube units. In addition, overall heat transfer coefficients (U) for plate type exchangers are three to four times those of shell and tube units.

2. Availability of a wide variety of corrosion resistant alloys.

   Since the heat transfer area is constructed of thin plates, stainless steel or other high alloy construction is significantly less costly than for a shell and tube exchanger of similar material.

3. Ease of maintenance.

   The construction of the heat exchanger is such that, upon disassembly, all heat transfer areas are available for inspection and cleaning. Disassembly consists only of loosening a small number of tie bolts.

4. Expandability and multiplex capability.

   The nature of the plate heat exchanger construction permits expansion of the unit should heat transfer requirements increase after installation. In addition, two or more heat exchangers can be housed in a single frame, thus reducing space requirements and capital costs.

5. Compact design.

   The superior thermal performance of the plate heat exchanger and the space efficient design of the plate arrangement results in a very compact piece of equipment. Space requirements for the plate heat exchanger generally run 10% to 50% that of a shell and tube unit for equivalent duty. In addition, tube cleaning and replacing clearances are eliminated.

Figure 11.1 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 11.1, 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.