The United Nations University

GEOTHERMAL TRAINING PROGRAMME
Orkustofnun, Grensásvegur 9,
IS-108 Reykjavík, Iceland

Reports 1996
Number 1

LECTURES ON DIRECT UTILIZATION OF GEOTHERMAL ENERGY

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ISBN 9979-827-95-5

PREFACE

Since the foundation of the UNU Geothermal Training Programme in 1979, it has been customary to invite annually one internationally renowned geothermal expert to come to Iceland as the UNU Visiting Lecturer. This has been in addition to various foreign lecturers who have given lectures at the Training Programme from year to year. The UNU Visiting Lecturers have been in residence in Reykjavík from one to eight weeks. They have given a series of lectures on their speciality and held discussion sessions with the UNU Fellows attending the Training Programme. The lectures are open both to the UNU Fellows at the Training Programme, staff members of Orkustofnun (National Energy Authority), and the geothermal community in Iceland.

It is the good fortune of the UNU Geothermal Training Programme that so many distinguished geothermal specialists have found time to visit us. Following is a list of the UNU Visiting Lecturers during 1979-1996:

1979	Donald E. White	United States
1980	Christopher Armstead	United Kingdom
1981	Derek H. Freeston	New Zealand
1982	Stanley H. Ward	United States
1983	Patrick Browne	New Zealand
1984	Enrico Barbier	Italy
1985	Bernardo Tolentino	Philippines
1986	C. Russel James	New Zealand
1987	Robert Harrison	UK
1988	Robert O. Fournier	United States
1989	Peter Ottlik	Hungary
1990	Andre Menjoz	France
1991	Wang Ji-yang	China
1992	Patrick Muffler	USA
1993	Zosimo F. Sarmiento	Philippines
1994	Ladislaus Rybach	Switzerland
1995	Gudmundur Bödvarsson	United States
1996	John Lund	United States

Professor John Lund of the Oregon Institute of Technology (USA) was the UNU Visiting Lecturer in 1996. He and his colleagues at the OIT and the Geo-Heat Center in Klamath Falls (Oregon) have for the last two decades made a great contribution to geothermal development in the world through their numerous publications, text books, and the Quarterly Bulletin of the Geo-Heat Center. John Lund gave a series of lectures on the direct use of geothermal energy at the UNU Geothermal Training Programme in 1987. It was a great pleasure to have him with us again in 1996 in the capacity of the UNU Visiting Lecturer. We are very grateful to him for writing up his lecture notes and thus make the lectures available to a very much larger audience than those who were so fortunate in attending his lectures in Reykjavík.

With warmest wishes from Iceland,

Ingvar B. Fridleifsson, director,
United Nations University
Geothermal Training Programme

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COMMENTS BY THE LECTURER

I felt extremely privileged to have been invited to lecture to the United National University Geothermal Training Program students in the summer of 1996. These lectures were based mainly on research and publication from the Geo-Heat Center, a program funded by the U.S. Department of Engineer, Office of Geothermal Technologies, Washington, D.C.

The lectures were, in part, based on publications and research that I have undertaken, however, I must emphasize that much of the lecture material came from other sources. This material from other sources was either summerized or directly quoted from other publications, as I felt this was the best and most current material available. All of the material used in each lecture is referenced at the end of each section. Unfortunately, it may not be clear from the printed material which material was the work of others and which is mine. I am thus attempting to correct this oversight.

Lecture 1 - *Direct Heat Utilization of Geothermal Resources* is based mainly on work written and published by Derek Freeston of the Geothermal Institute, University of Auckland, Jon Steinar Gudmundsson, then of Stanford University and now with the Norwegian Technical Institute, and myself.

Lecture 2 - *Downhole Heat Exchangers* is based on field work and research work by Gene Culver of the Geo-Heat Center, Gordon Reistad of Oregon State University and myself. The New Zealand Experience comes mainly from work by members of the staff at the Geothermal Institute, University of Auckland.

Lecture 3 - *Greenhouses*, for the most part, is almost a direct quote from work by Kevin Rafferty of the Geo-Heat Center from our publication: Geothermal Direct-Use Engineering and Design Guidebook. The Introduction and Examples comes from my experiences, and I also did the metric conversions for Rafferty's section on Greenhouse Construction.

Lecture 4 - *Aquaculture* is again, for the most part, a direct quote of work by Kevin Rafferty of the Geo-Heat Center from the Geothermal Direct-Use Engineering and Design Guidebook. The Introduction, Examples and General Design Considerations are based on my own work, and I also did the metric conversions for Rafferty's section on Specific Design Considerations.

Lecture 5 - *Ground-Source (Geothermal) Heat Pumps* is a summary of material from a number of different authors, included Stephen Kavanaugh of the University of Alabama, Kevin Rafferty and Paul Lienau of the Geo-Heat Center, Harry Braud et al. of Louisiana, and my own work.

Lecture 6 - *Industrial Applications* is mainly based on my own work and on work by Paul Lienau of the Geo-Heat Center published in the Geothermal Direct-Use Engineering and Design Guidebook.

Finally, as an additional source of information on the direct utilization of geothermal energy, I would recommend the Geo-Heat Center's 454-page book on Geothermal Direct-Use Engineering and Design Guidebook. This publication can be ordered through our webpage: <www.oit.edu/~geoheat>.

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LECTURE 1

DIRECT HEAT UTILIZATION OF GEOTHERMAL RESOURCES

ABSTRACT

Direct utilization of geothermal energy consists of various forms for heating and cooling instead of converting the energy for electric power generation. The major areas of direct utilization are (1) swimming, bathing and balneology, (2) space heating and cooling including district heating, (3) agriculture applications, (4) aquaculture applications, (5) industrial processes, and (6) heat pumps. Major direct utilization projects exploiting geothermal energy exist in about 38 countries, and the estimated installed thermal power is almost 9,000 MWt utilizing 37,000 kg/s of fluid. The world-wide thermal energy used is estimated to be at least 108,100 TJ/yr (30,000 GWh/yr) - saving 3.65 million TOE/yr. The majority of this energy use is for space heating (33%), and swimming and bathing (19%). In the USA the installed thermal power is 1874 MWt, and the annual energy use is 13,890 TJ (3,860 GWh). The majority of the use (59 %) is for heat pumps (both ground coupled and water source), with space heating, bathing and swimming, and fish and animal farming each supplying about 10%.

KEYWORDS

Geothermal, direct use, balneology, space heating, district heating, greenhouses, aquaculture, industrial processes, heat pumps.

1.1 INTRODUCTION

Direct or non-electric utilization of geothermal energy refers to the immediate use of the heat energy rather than to its conversion to some other form such as electrical energy. The primary forms of direct use include swimming, bathing and balneology (therapeutic use), space heating and cooling including district heating, agriculture (mainly greenhouse heating and some animal husbandry), aquaculture (mainly fish pond and raceway heating), industrial processes, and heat pumps (for both heating and cooling). In general, the geothermal fluid temperatures required for direct heat use are lower than those for economic electric power generation.

Most direct use applications use geothermal fluids in the low- to moderate-temperature range between 50° and 150°C, and in general, the reservoir can be exploited by conventional water well drilling equipment. Low-temperature systems are also more widespread than high-temperature systems (above 150°C), so they are more likely to be located near potential users. In the US, for example, of the 1,350 known or identified geothermal systems, 5% are above 150°C, and 85% are below 90°C (Muffler, 1979). In fact, almost every country in the world has some low-temperature systems, while only a few have accessible high-temperature systems.

1.2 UTILIZATION

Traditionally, direct use of geothermal energy has been on a small scale by individuals. More recent developments involve large-scale projects, such as district heating (Iceland and France), greenhouse complexes (Hungary and Russia), or major industrial use (New Zealand and the US). Heat exchangers are also becoming more efficient and better adapted to geothermal projects, allowing use of lower temperature water and highly saline fluids. Heat pumps utilizing very low-temperature fluids have extended geothermal developments into traditionally non-geothermal countries such as France, Switzerland and Sweden, as well as areas of the midwestern and eastern US. Most equipment used in this projects are of standard, off-the-shelf design and need only slight modifications to handle geothermal fluids (Gudmundsson and Lund, 1985).

Worldwide, the installed capacity of direct geothermal utilization is almost 9,000 MWt and the energy use is about 108,100 TJ/yr (30,000 GWh/yr) distributed among 38 countries (Figure 1.1). This amounts to saving an equivalent 3.65 million tonnes of fuel oil per year (TOE). The distribution of the energy use among the various types of use is shown in Figure 1.2 for the entire world and, for comparison, the US. The installed capacity in the US is 1,875 MWt and the annual energy use is 13,890 TJ (3,860 GWh), saving 0.47 million TOE (Lienau et al., 1995). Internationally, the largest uses are for space heating (33%) (3/4 of which is due to district heating), and for swimming, bathing and balneology (19%); whereas, in the US the largest use is for geothermal heat pumps (59%). In comparison, Iceland's largest geothermal energy use is 74% for space heating (4,530 GWh/yr) - primarily with district heating systems (Ragnarsson, 1995). The world wide use data is based on Freeston (1996), but have been modified according to Freeston (1990), country update reports from the World Geothermal Congress (1995), and the author's personal experience.

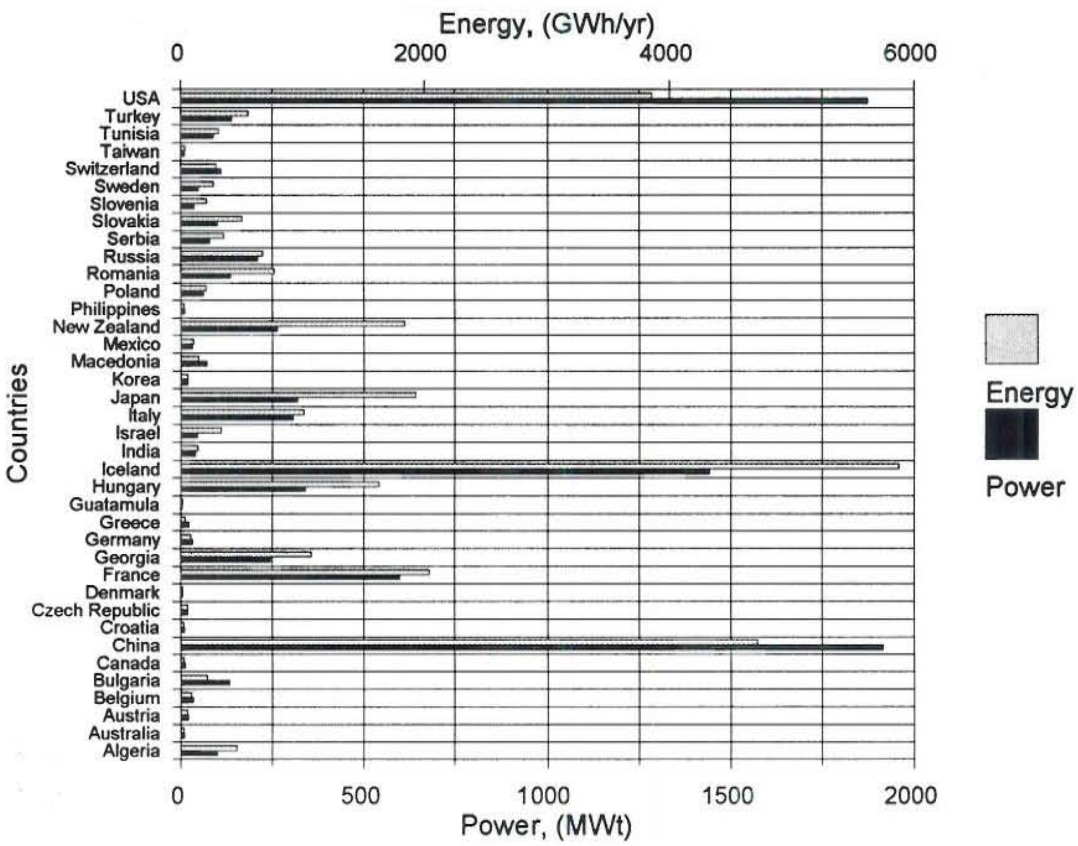

FIGURE 1.1: Direct use geothermal capacity and utilization by country

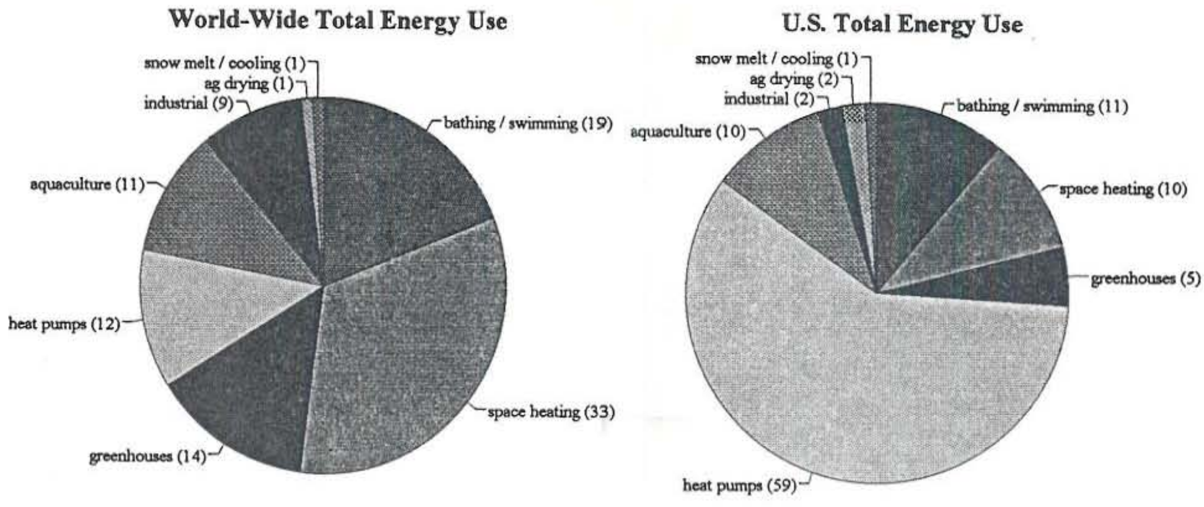

FIGURE 1.2: Distribution of geothermal energy use in the world and the US

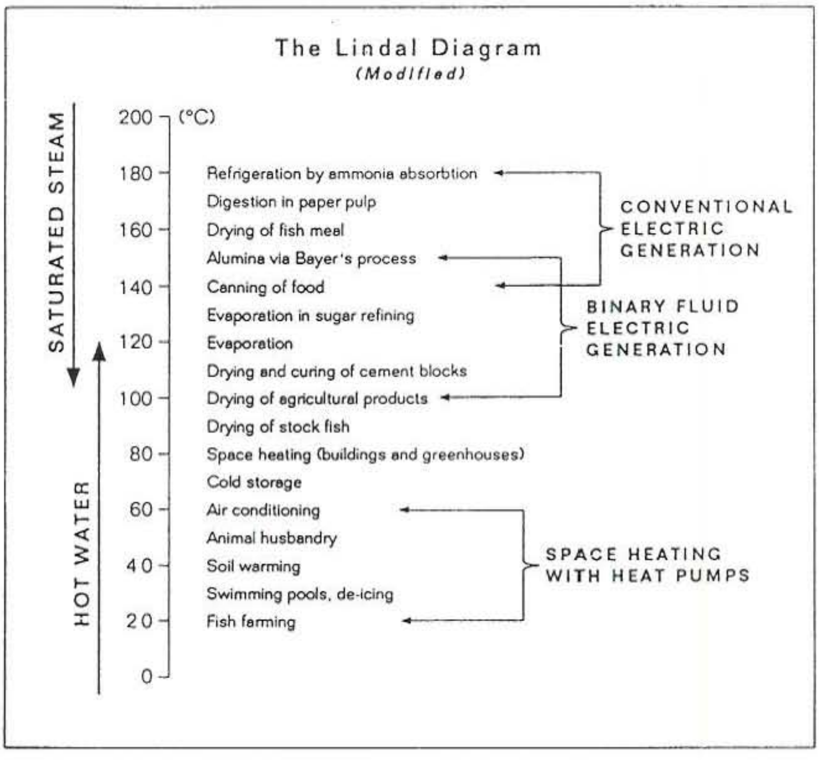

FIGURE 1.3: Lindal diagram

The Lindal diagram (Gudmundsson and Lienau, 1985), named for Baldur Lindal, the Icelandic engineer who first proposed it, indicates the temperature range suitable for various direct use activities (Figure 1.3). Typically, the agricultural and aquacultural uses require the lowest temperatures, with values from 25° to 90°C. The amounts and types of chemicals such as arsenic and dissolved gases such as boron, are a major problem with plants and animals, thus heat exchangers are often necessary. Space heating requires temperatures in the range of 50° to 100°C, with 40°C useful in some marginal cases and ground source heat pumps extending the range down to 4°C. Cooling and industrial processing normally require temperatures over 100°C. The leading user of geothermal energy, in terms of market penetration, is Iceland, where more than 85% of the population enjoys geothermal heat in their homes from 27 municipal district heating services, and 44% of the country's total energy use is supplied by direct heat and electrical energy derived from geothermal resources (Ragnarsson, 1995).

1.2.1 Swimming, bathing and balneology

Romans, Chinese, Ottomans, Japanese and central Europeans have bathed in geothermal waters for centuries. Today, more than 2,200 hot spring resorts in Japan draw 100 million guests every year, and the "return-to-nature" movement in the US has revitalized many hot spring resorts.

The geothermal water at Xiaotangshan Sanitarium, northwest of Beijing, China, has been used for medical purposes for over 500 years. Today, the 50°C water is used to treat high blood pressure, rheumatism, skin disease, diseases of the nervous system, ulcers and generally for recuperation after surgery. In Rotorua, New Zealand at the centre of the Taupo Volcanic Zone of North Island, the Queen Elizabeth Hospital was built during World War II for US Servicemen and later became the national hospital for the treatment of rheumatic disease. The hospital has 200 beds, and outpatient service, and a cerebral palsy unit. Both acidic and basic heated mud baths treat rheumatic diseases.

In Beppu on the southern island of Kyushu, Japan, the hot water and steam meet many needs: heating, bathing, cooking, industrial operations, agriculture research, physical therapy, recreational bathing, and even a small zoo (Taguchi et al., 1996). The waters are promoted for " digestive system troubles, nervous troubles, and skin troubles." Many sick and crippled people come to Beppu for rehabilitation and physical therapy. There are also eight Jigokus ("burning hells") in town showing various geothermal phenomena, used as tourist attractions.

In the former Czechoslovakia, the use of thermal waters has been traced back before the occupation of the Romans and has had a recorded use of almost 1,000 years. Today, there are 60 spa resorts located mainly in Slovakia, visited by 460,000 patients usually for an average of three weeks each. These spas have old and well-established therapeutic traditions. Depending on the chemical composition of the mineral waters and spring gas, availability of peat and sulfurous mud, and climatic conditions, each sanitarium is designated for the treatment of specific diseases. The therapeutic successes of these spas are based on centuries of healing tradition (balneology), systematically supplemented by the latest discoveries of modern medical science (Lund, 1990).

Bathing and therapeutic sites in the US include: Saratoga Springs, New York; Warm Springs, Georgia; Hot Springs, Virginia, White Sulfur Springs, West Virginia, Hot Springs Arkansas; Thermopolis, Wyoming; and Calistoga, California. The original use of these sites were by Indians, where they bathed and recuperated from battle. There are over 115 major geothermal spas in the US with an annual energy use of 1,500 TJ (Lund, 1996).

1.2.2 Space conditioning

Space conditioning includes both heating and cooling. Space heating with geothermal energy has widespread application, especially on an individual bases. Buildings heated from individual wells are popular in Klamath Falls, Oregon, Reno, Nevada, and Taupo and Rotorua, New Zealand. Absorption space cooling with geothermal energy has not been popular because of the high temperature requirements and low efficiency. Geothermal heat pumps (ground water and ground coupled) have become popular in the US, Canada, Germany and Switzerland, used for both heating and cooling.

An example of space heating and cooling with low- to moderate-temperature geothermal energy is the Oregon Institute of Technology in Klamath Falls, Oregon (Figure 1.4). Here, eleven buildings, approximately 62,000 square meters of floor space, are heated with water from three wells at 89°C. Up to 62 L/s of fluid can be provided to the campus, with the average heat utilization rate over 0.53 MWt and the peak at 5.6 MWt. In addition, a 541 kW (154 tons) chiller requiring up to 38 L/s of geothermal fluid produces 23 L/s of chilled fluid at 7°C to meet the campus cooling base load.

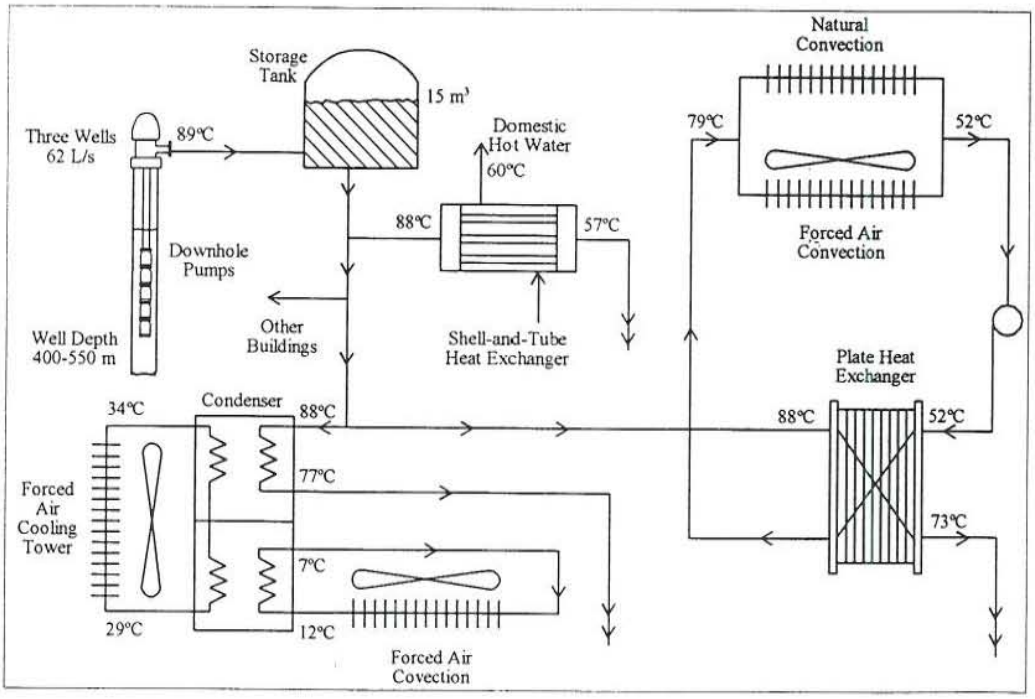

FIGURE 1.4: Oregon Institute of Technology heating and cooling system

1.2.3 District heating

District heating originates from a central location, and supplied hot water or steam through a network of pipes to individual dwellings or blocks of buildings. The heat is used for space heating and cooling, domestic water heating, and industrial process heat. A geothermal well field is the primary source of heat; however, depending on the temperature, the district may be a hybrid system, which would include fossil fuel and/or heat pump peaking.

Geothermal district heating systems are in operation in at least 12 countries, including Iceland, France, Poland, Hungary, Turkey, Japan and the US. The Warm Springs Avenue project in Boise, Idaho, dating back to 1892 and originally heating more than 400 homes, is the earliest formal project in the US. The Reykjavik, Iceland, district heating system (Hitaveita Reykjavikur) (Figure 1.5), established in 1930, is

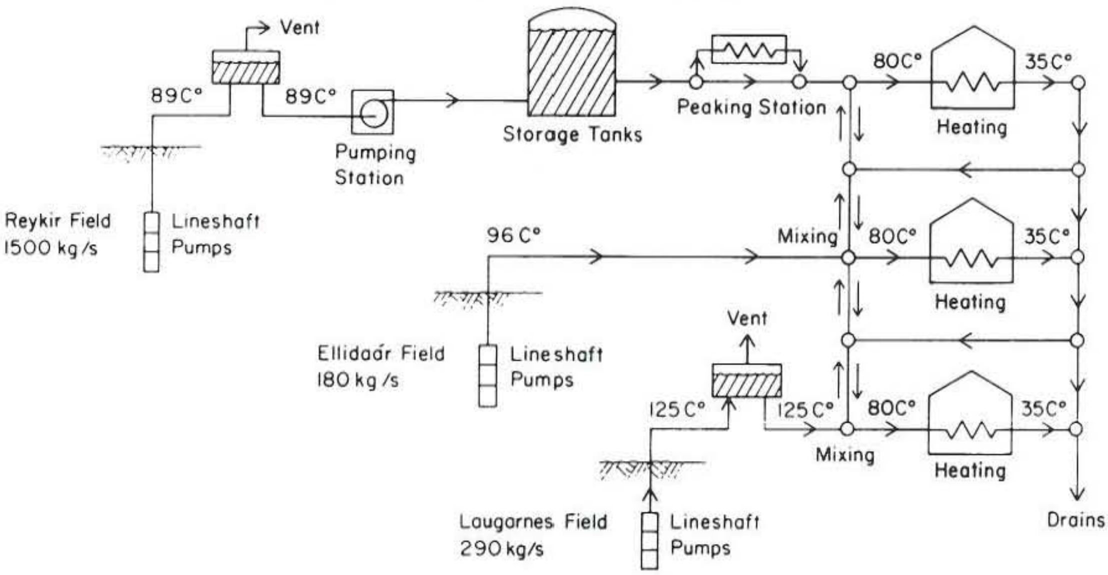

FIGURE 1.5: Reykjavik district heating system (prior to the Nesjavellir connection)

probably the most famous (Frimannsson, 1991). This system supplies heat for a population of around 145,000 people. The installed capacity of 640 MWt is designed to meet the heating load to about -10°C, however, during colder periods the increased load is met by large storage tanks and an oil fired booster station.

In France, production wells in sedimentary basins provide direct heat to more than 500,000 people from 40 projects. These wells provide from 40 to 100 L/s of 60° to 100°C water from depths of 1,500 to 2,000 m. In the Paris basin, a doublet system (one production and one injection well) provides 70°C water, with the peak load met by heat pumps and conventional fossil fuel burners (Figure 1.6).

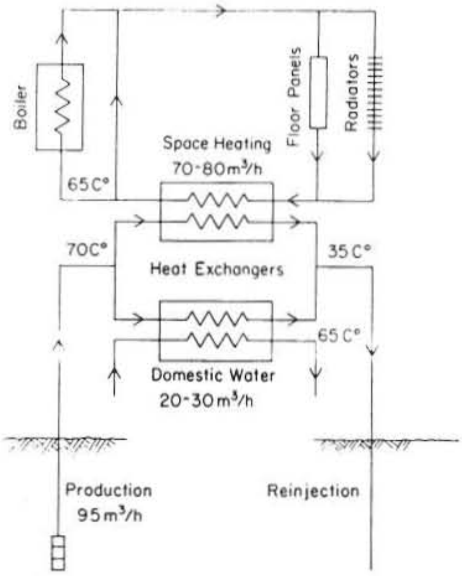

FIGURE 1.6: Melun l'Almont (Paris) doublet heating system

1.2.4 Agribusiness applications

Agribusiness applications (agriculture and aquaculture) are particularly attractive because they require heating at the lower end of the temperature range where there is an abundance of geothermal resources. Use of waste heat or the cascading of geothermal energy also has excellent possibilities. A number of agribusiness applications can be considered: greenhouse heating, aquaculture, animal husbandry, soil warming and irrigation, mushroom culture, and biogas generation.

Numerous commercially marketable crops have been raised in geothermally heated greenhouses in Hungary, Russia, New Zealand, Japan, Iceland, China, and the US. These include vegetables, such as cucumbers and tomatoes, flowers (both potted and bedded), house plants, tree seedlings, and cacti. Using geothermal energy for heating reduces operating costs (which can account for 35% of the product cost) and allows operation in colder climates where commercial greenhouses would not normally be economical.

The use of geothermal energy for raising catfish, shrimp, tilapia, eels, and tropical fish has produced crops faster than by conventional solar heating. Using geothermal heat allows better control of pond temperature, thus optimizing growth (Figure 1.7). Fish breeding has been successful in Japan, China, and the US. A very successful prawn raising

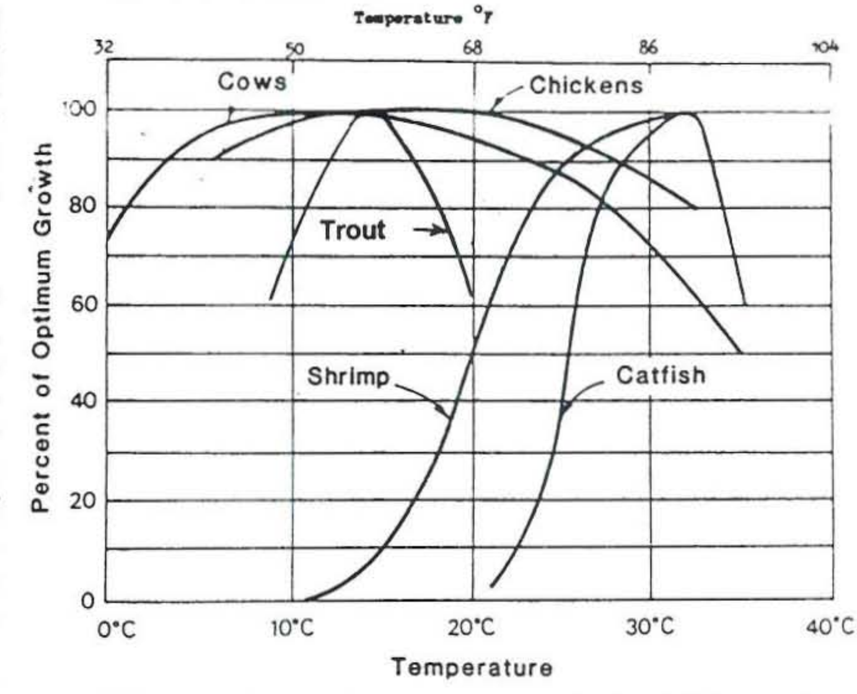

FIGURE 1.7: Effect of temperature on animal and fish growth

operation, producing 400 tonnes of Giant Malaysian Freshwater Prawns per year at US$ 17 to 27/kg, has been developed near the Wairakei geothermal field in New Zealand (Lund and Klein, 1995). The most important factors to consider are the quality of the water and disease. If geothermal water is used directly, concentrations of dissolved heavy metals, fluorides, chlorides, arsenic, and boron must be considered.

Livestock raising facilities can encourage the growth of domestic animals by a controlled heating and cooling environment. An indoor facility can lower mortality rate of newborn, enhance growth rates, control disease, increase litter size, make waste management and collection easier, and in most cases improve the quality of the product. Geothermal fluids can also be used for cleaning, sanitizing and drying of animal shelters and waste, as well as assisting in the production of biogas from the waste.

1.2.5 Industrial applications

Although the Lindal diagram shows many potential industrial and process applications of geothermal energy, the world's uses are relatively few. The oldest industrial use is at Larderello, Italy, where boric acid and other borate compounds have been extracted from geothermal brines since 1790. Today, the two largest industrial uses are the diatomaceous earth drying plant in northern Iceland and a pulp, paper and wood processing plant at Kawerau, New Zealand. Notable US examples are two onion dehydration plants in northern Nevada (Lund, 1995), and a sewage digestion facility in San Bernardino, California. Alcohol fuel production has been attempted in the US; however, the economics were marginal and thus this industry has not been successful.

Drying and dehydration are important moderate-temperature uses of geothermal energy. Various vegetable and fruit products are feasible with continuous belt conveyors (Figure 1.8) or batch (truck) dryers with air temperatures from 40° to 100°C (Lund and Rangel, 1995). Geothermally drying alfalfa, onions, pears, apples and seaweed are examples of this type of direct use. A new development in the use of geothermal fluids is for enhanced heap leaching of precious metals in Nevada by applying heat to the cyanide process (Trexler et al., 1990). Using geothermal energy increases the efficiency of the process and extends the production into the winter months.

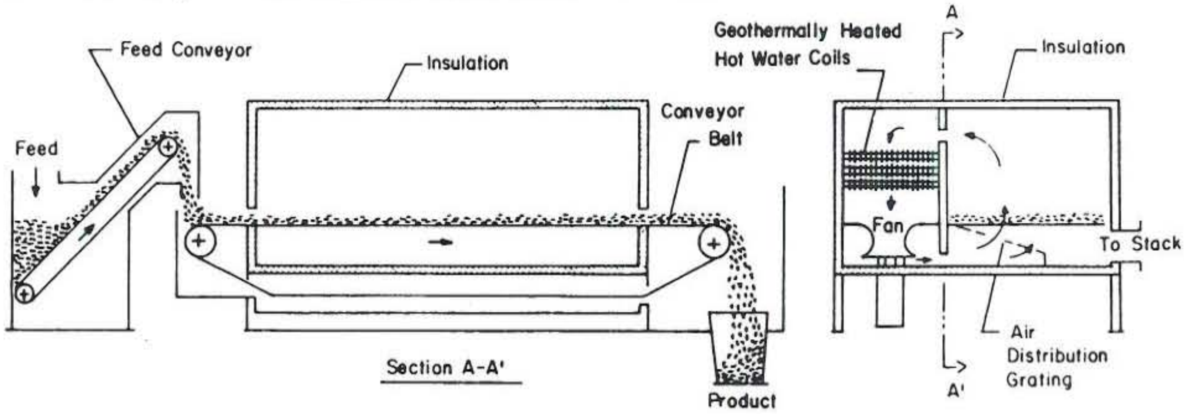

FIGURE 1.8: Continuous belt dehydration plant, schematic

1.3 EQUIPMENT

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 consideration, 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 used side of the system as shown in Figure 1.9.

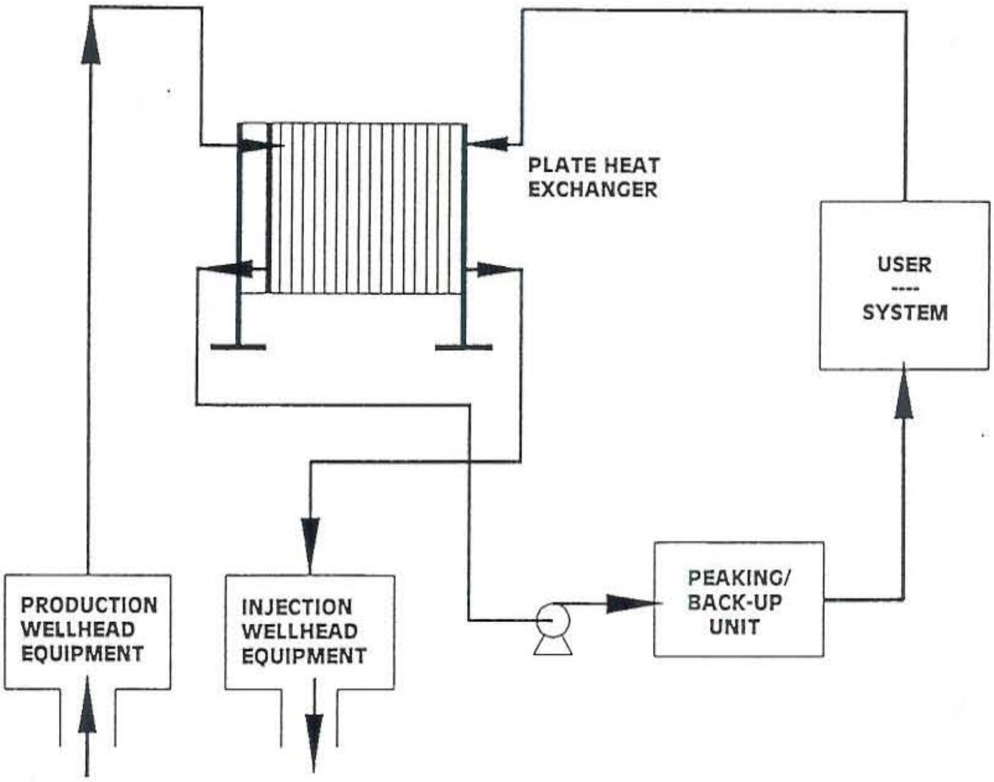

FIGURE 1.9: Geothermal direct utilization system using a heat exchanger

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 1.9). 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 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. Detailed discussion of equipment design and use can be found in Lienau and Lunis (1991).

1.3.1 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: line shaft pump systems and submersible pump systems.

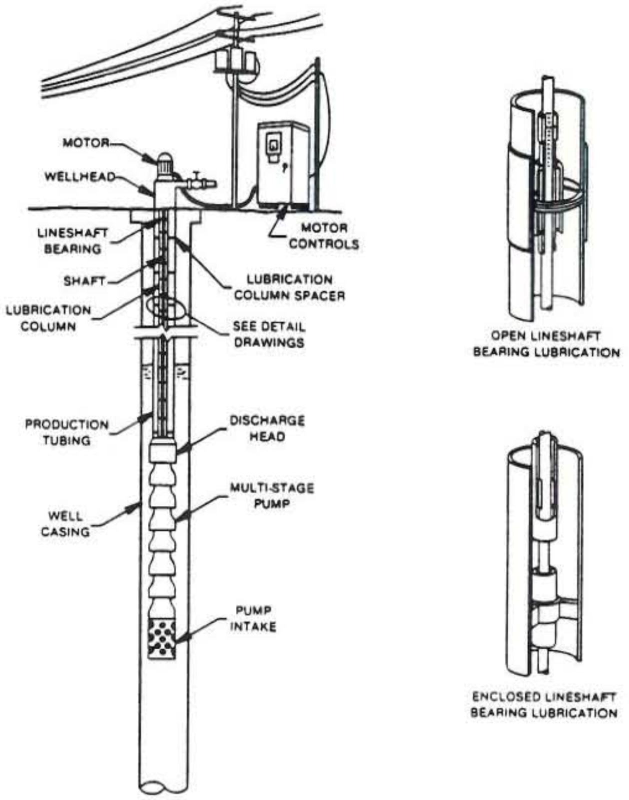

FIGURE 1.10: Line shaft pump

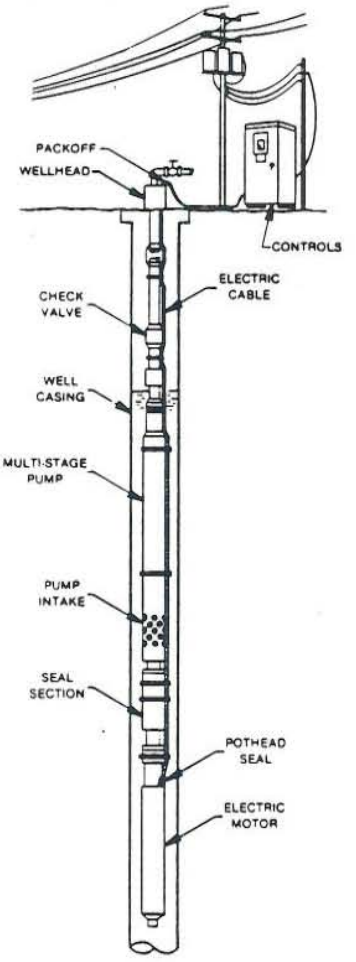

FIGURE 1.11: Submersible pump

The line shaft pump system (Figure 1.10) 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 centred 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 electric submersible pump system (Figure 1.11) 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 (line shaft have been used on the Oregon Institute of Technology campus in 89°C water for 45 years). If a line shaft pump is used, special allowances must be made for the thermal expansion of various components and for oil lubrication of the bearings. The line shaft pumps are preferred over the submersible pump in conventional geothermal applications for two main reasons: the line shaft pump cost less, and it has a proven track record. However, for setting depths exceeding about 250 m a submersible pump is required.

1.3.2 Piping

The fluid state in transmission lines of direct use projects can be liquid water, steam vapour 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 50 to 200°C) causes stress that must be accommodated by careful engineering design.

The cost of transmission lines and the distribution networks in direct use projects is significant. This is especially true when the geothermal resource is located at great distance from the main load centre; however, transmission distances of up to 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 1.13 later).

Carbon steel is now the most widely used material for geothermal transmission lines and distribution networks, especially if the fluid temperature is over 100°C. Other common types of piping material are fibreglass 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 not 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 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 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 ground water 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 US$300 per meter of length), 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. Above ground 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 above ground 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 fibreglass. Below ground such pipes should be protected with Polyvinyl chloride (PVC) jacket; above ground aluminum can be used. Generally, 2.5 to 10 cm of insulation is adequate. In two-pipe systems the supply and return line are usually insulated, whereas in single-pipe system only the supply line is insulated.

At flowing conditions the temperature loss in insulated pipelines is in the range of 0.1 to 1.0°C/km, and in uninsulated lines the loss is 2 to 5°C/km (in the approximate range of 5 to 15 L/s flow for 15 cm diameter pipe) (Ryan, 1981). It is less for larger diameter pipes, i.e. less than 2°C loss is experienced in the new above ground 29 km long and 80 and 90 cm wide line (with 10 cm of rock wool insulation) from Nesjavellir to Reykjavik in Iceland. The flow rate is around 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 1.12 shows fluid temperatures, as a function of distance, in a 45 cm diameter pipeline, insulated with 5 cm of urethane.

Several examples of above ground and buried pipelines installations are shown in Figure 1.13.

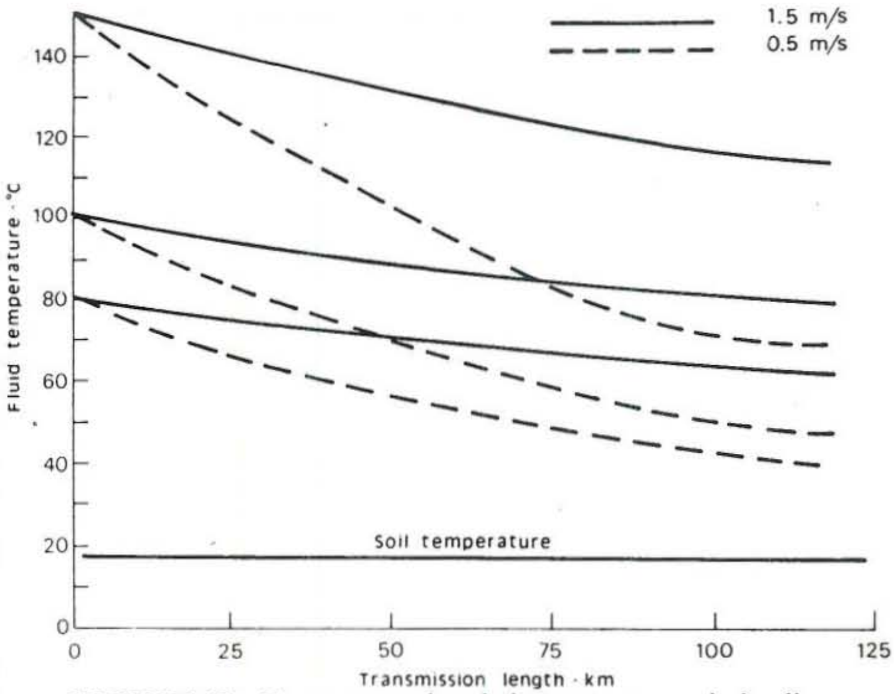

FIGURE 1.12: Temperature drop in hot water transmission line

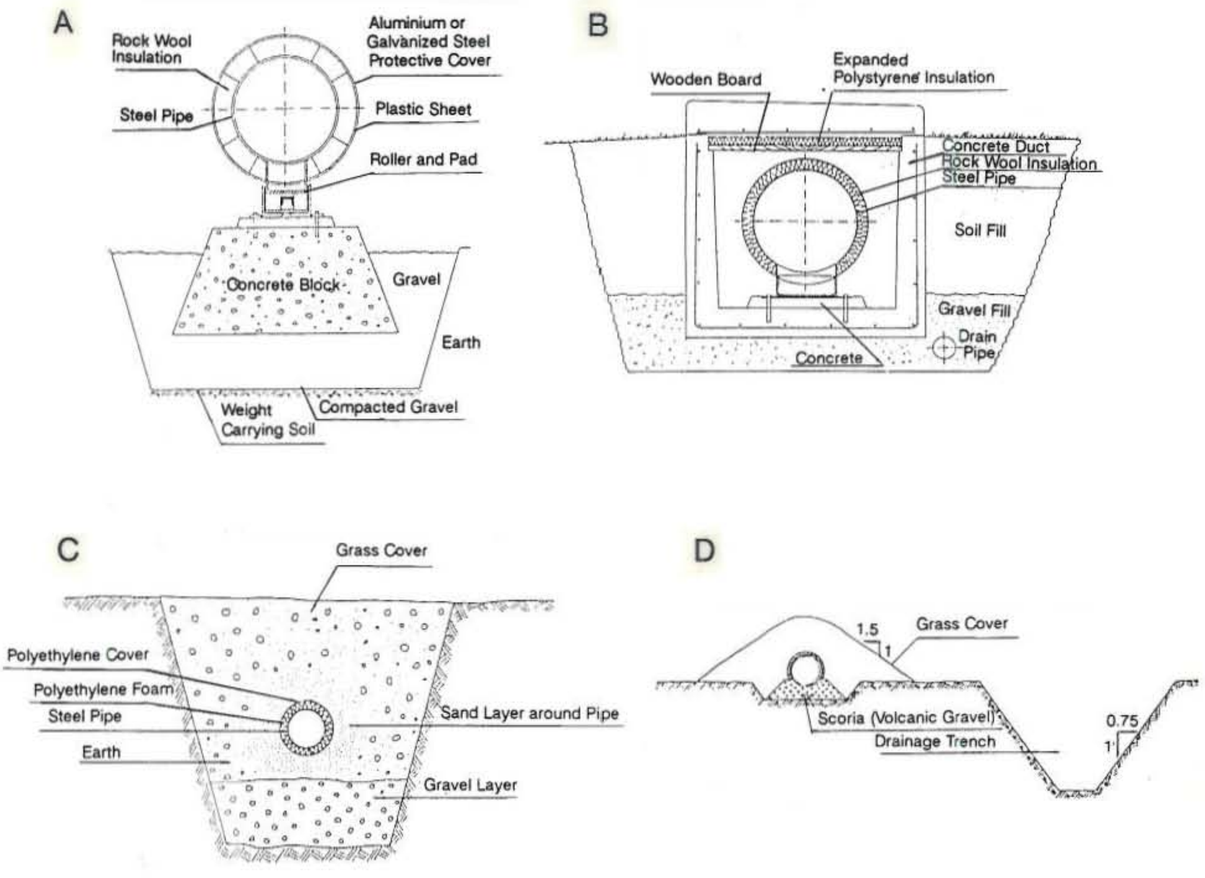

FIGURE 1.13: Examples of above and below ground pipelines, a) Above ground pipeline with sheet metal cover; b) Steel pipe in concrete tunnel; c) Steel pipe with polyurethane insulation and polyethylene cover; d) Asbestos cement pipe with earth and grass cover

Steel piping is shown in most case, but FRP or PVC can be used in low temperature applications. Above ground pipelines have been used extensively in Iceland where excavation in lava rock is expensive and difficult; however, in the USA below ground installations are more 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).

1.3.3 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 1.14). 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 addition 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 world-wide.

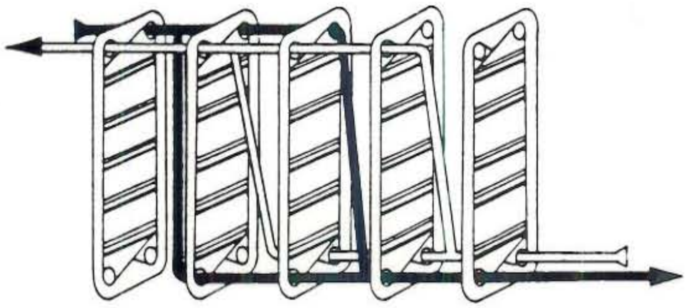

FIGURE 1.14: Plate heat exchanger

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.

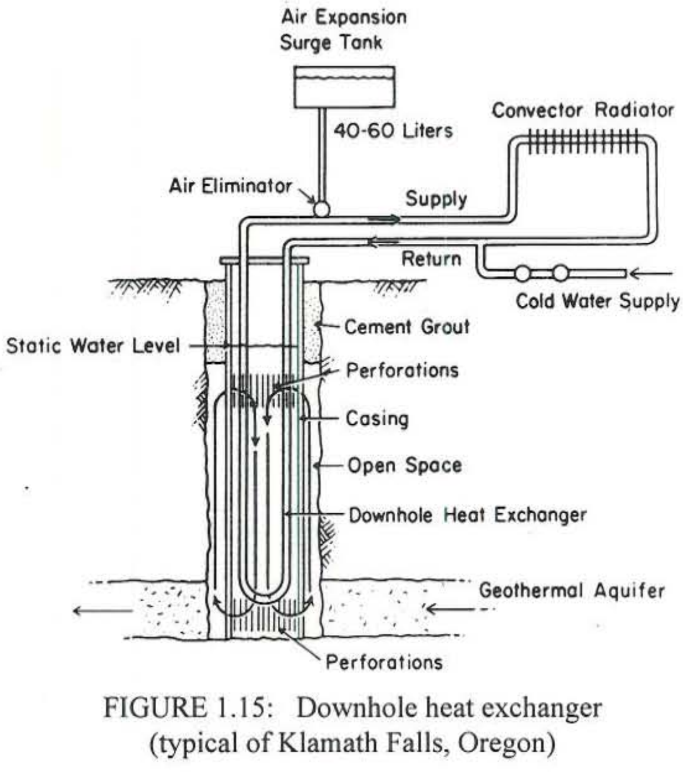

FIGURE 1.15: Downhole heat exchanger
(typical of Klamath Falls, Oregon)

Downhole 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 1.15). 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).

1.3.4 Heat pumps

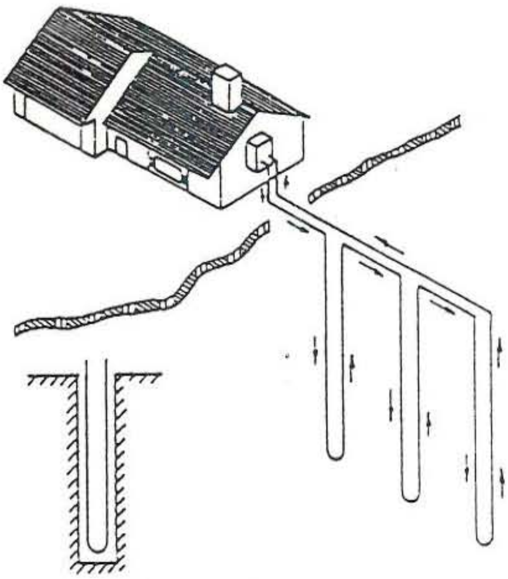

FIGURE 1.16: Typical ground-source heat pump installations

At the present time, ground-coupled and ground-water (often called ground-source or geothermal) heat pump systems are being installed in great numbers in the United States, Switzerland and Germany (Kavanaugh, 1991; Rybach and Hopkirk, 1995). Ground water aquifers and soil temperatures in the range of 5 to 30°C are being used in these systems. Ground-source heat pumps (GSHP) utilize ground water in wells or by direct ground coupling with vertical heat exchangers (Figure 1.16). Just about every state in the USA, especially in the midwestern and eastern states are utilizing these system, in part subsidized by public and private utilities. It is estimated that almost 60,000 Ground water systems and more than 50,000 closed loop vertical systems are already in use.

Like refrigerators, heat pumps operate on the basic principle that fluid absorbs heat when it evaporates into a gas, and likewise gives off heat when it condenses back into a liquid. A geothermal heat pump system can be used for both heating and cooling. The types of heat pumps that are adaptable to geothermal energy are the water-to-air and the water-to-water. Heat pumps are available with heating capacities of less than 3 kW to over 1,500 kW.

1.3.5 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. All three can be adapted directly to geothermal energy or converted by retrofitting existing systems.

1.3.6 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 temperatures (about 160°C), but they also have high efficiency. The single stage units can be driven with hot water at temperatures as low as 77°C (such as at Oregon Institute of Technology - see Figure 1.4). 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 120°C for a reasonable performance. Figure 1.17 illustrates how the geothermal absorption process works.

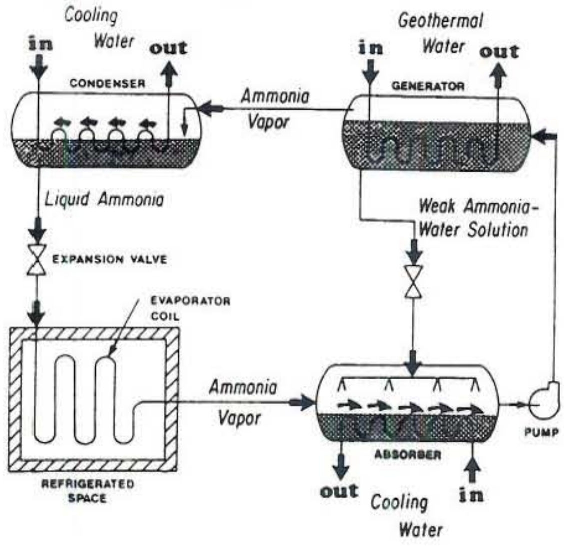

FIGURE 1.17: Geothermal absorption refrigeration cycle

1.4 ECONOMIC CONSIDERATIONS

Geothermal projects require a relatively large initial capital investment, with small annual operating costs thereafter. Thus, a district heating project, including production wells, pipelines, heat exchangers, and injection wells, may cost several million dollars. By contrast, the initial investment in a fossil fuel system includes only the cost of a central boiler and distribution lines. The annual operation and maintenance costs for the two systems are similar, except that the fossil fuel system may continue to pay for fuel at an every-increasing rate, while the cost of geothermal fuel is stable. The two systems, one with a high initial capital cost and the other with high annual costs, must be compared.

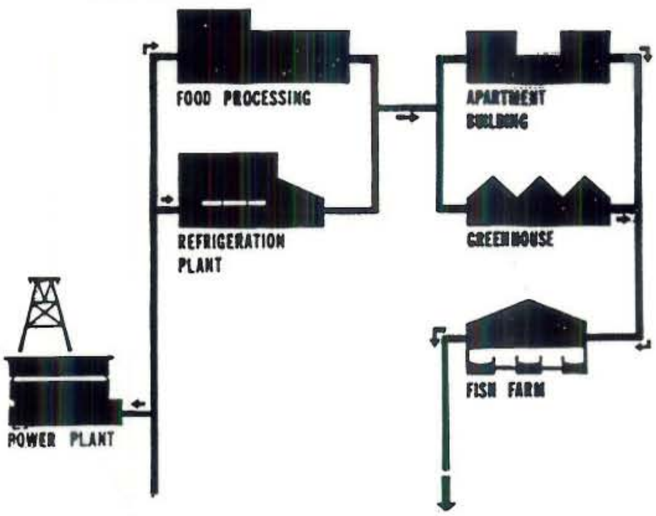

FIGURE 1.18: An example of cascading

Geothermal resources fill many needs; power generation, space heating, greenhouse heating, industrial processing, and bathing to name a few. Considered individually, however, some of the uses may not promise an attractive return on investment because of the high initial capital cost. Thus, we may have to consider using a geothermal fluid several times to maximize benefits. This multistage utilization, where lower and lower water temperatures are used in successive steps, is called cascading or waste heat utilization. A simple form of cascading employs waste heat from a power plant for direct use projects (Figure 1.18).

Geothermal cascading has been proposed and successfully attempted on a limited scale throughout the world. In Rotorua, New Zealand, for example, after geothermal water and steam heat a home, the owner will often use the waste heat for a backyard swimming pool and steam cooker. At the Otake geothermal power plant in Japan, about 165 tonnes per hour of hot water flows to downstream communities for space heating, greenhouses, baths and cooking. In Sapporo, Hokkaido, Japan, the waste water from the pavement snow melting system is retained at 65°C and reused for bathing.

Recent estimates (1990 data) of the capital cost for various direct use project in the US are as follows:

Space Heating (individual):	US$ 463/kW of installed capacity
District Heating:	US$ 386/kW of installed capacity
Greenhouses:	US$ 120/kW of installed capacity
Aquaculture:	US$ 26/kW of installed capacity

Recent international data (Freeston, 1996) gives US$ 270/kW of installed capacity for all projects reported, with a range from US$ 40 to US$ 1880/kW. In the US, the annual operation and maintenance cost is estimated at 5% of the installed cost.

1.5 FUTURE DEVELOPMENTS

There appears to be a large potential for the development of low- to moderate-enthalpy geothermal direct use across the world which is not currently being exploited due to financial constraints and the low price of competing energy sources. Given the right environment, and as gas and oil supplies dwindle, the use of geothermal energy will provide a competitive, viable and economic alternative source of renewable energy.

Future development will most likely occur under the following conditions:

1. Collocated resource and uses (within 10 km apart),
2. Sites with high heat and cooling load density (>36 MWt/sq. km),
3. Food and grain dehydration (especially in tropical countries where spoilage is common),
4. Greenhouses in colder climates,
5. Aquaculture to optimize growth - even in warm climates, and
6. Ground-coupled and ground water heat pump installation (both for heating and cooling).

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LECTURE 2

DOWNHOLE HEAT EXCHANGER

2.1 INTRODUCTION (Lund et al., 1975)

The downhole heat exchanger (DHE) eliminates the problem of disposal of geothermal fluid, since only heat is taken from the well. The exchanger consists of a system of pipes or tubes suspended in the well through which "clean" secondary water is pumped or allowed to circulate by natural convection. These systems offer substantial economic savings over surface heat exchangers where a single-well system is adequate (typically less than 0.8 MWt, with well depths up to about 500 ft [150 m]) and may be economical under certain condition at well depths to 1500 ft (450 m).

Several designs have proven successful, but the most popular are a simple hairpin loop or multiple loops of iron pipe (similar to the tubes in a U-Tube and shell exchanger) extending near the well bottom (Figure 2.1). An experimental design consisting of multiple small tubes with "headers" at each end suspended just below the water surface appears to offer economic and heating capacity advantages.

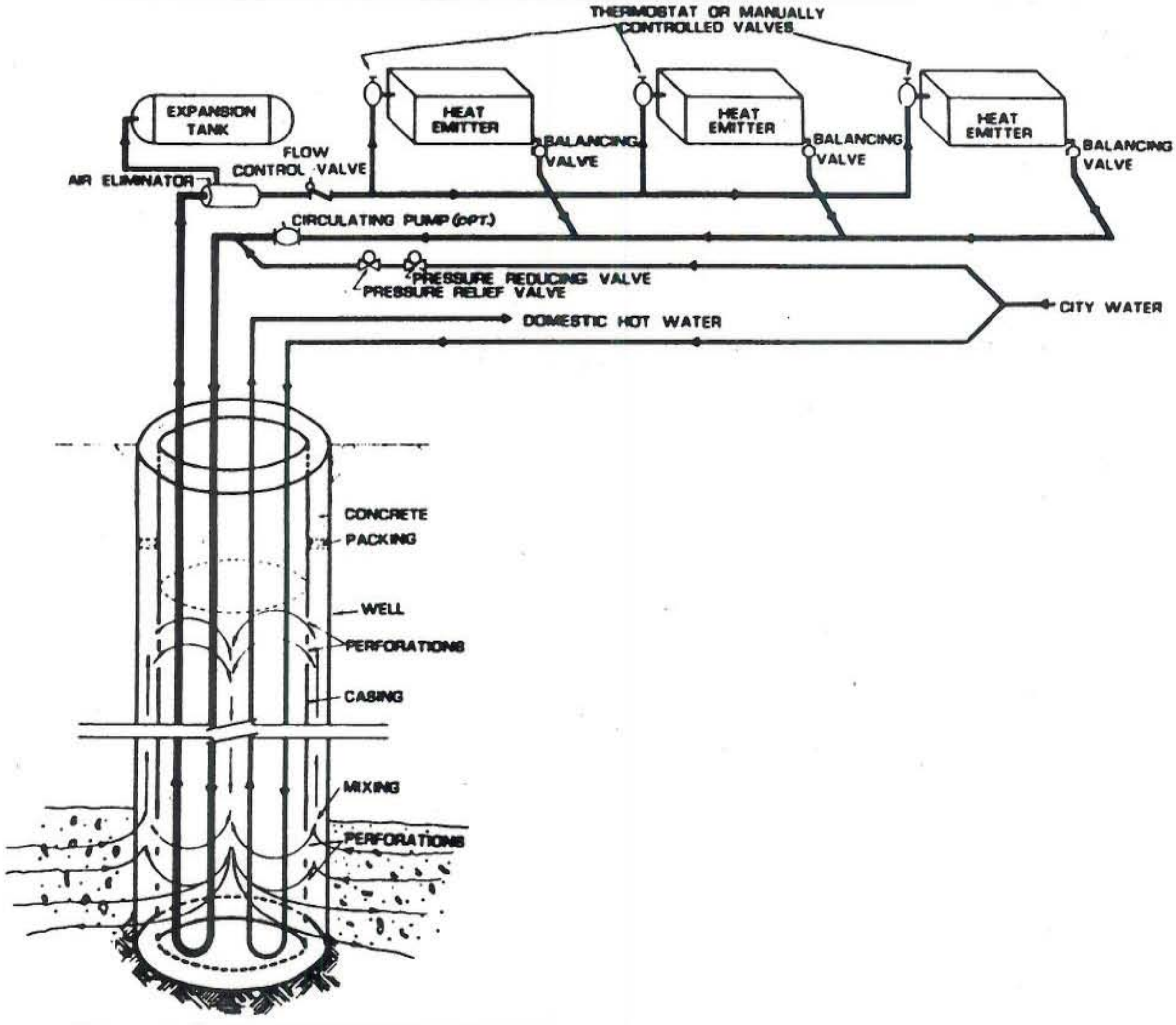

FIGURE 2.1: Typical downhole heat exchanger system (Klamath Falls, OR)

In order to obtain maximum output, the well must be designed to have an open annulus between the wellbore and the casing and perforations above and below the heat exchange 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. If the design parameters of bore diameter, casing diameter, heat exchanger length, tube diameter, number of loops, flow rate and inlet temperature are carefully selected, the velocity and mass flow of the natural convection cell in the well may approach those of a conventional shell-and-tube heat exchanger.

The interaction between the fluid in the aquifer and that in the well is not fully understood; but, it appears that outputs are higher where there is a high degree of mixing indicating that somewhat permeable formations are preferred.

Considering life and replacement costs, materials should be selected to provide economical protection from corrosion. Attention must be given to the anodic-cathodic relationship between the exchanger and the casing since it is relatively expensive to replace the well casing. Experience in the approximately 600 downhole exchangers in use indicates that corrosion is most severe at the air-water interface at static water level and that stray electrical currents can accelerate corrosion. Insulating unions should be used to isolate the exchanger from stray currents in building and city water lines. Capping the top of the casing will also reduce the air-water interface corrosion.

2.2 DESIGN AND CONSTRUCTION DETAILS (Culver, 1987)

DHE outputs range from supplying domestic hot water for a single family from a 40 foot, 140°F (12 m, 60°C) well at Jemenez Springs, New Mexico, to over 1 MWt at Ponderosa High School from a 560 foot, 202°F, 16 inch (170 m, 94°C, 40 cm) diameter well in Klamath Falls, Oregon. DHEs are also in use in New Zealand, Turkey, Hungary, Iceland, Russia and others. A well producing 6 MWt has been reported in use in Turkey.

The wells in Klamath Falls are 10 or 12 inch (25 or 30 cm) diameter drilled 20 or more feet (6 m) into "live water" and an 8 inch (20 cm) casing is installed. A packer is placed around the casing below any cold water or unconsolidated rock, usually at depths of 20-50 feet (6-15 m), and the well cemented from the packer to the surface. The casing is torch perforated (½ inch x 6 inch [1 x 15 cm]) in the live water area and just below the lowest static water level. Perforated sections are usually 15-30 feet (4-9 m) long and the total cross-sectional area of the perforations should be at least one-and-a-half to two times the casing cross section. Since water levels fluctuate summer to winter, the upper perforations should start below the lowest expected level. A 3/4 or 1 inch (2 or 2.5 cm) diameter pipe may be welded to the casing and extended from surface to below the packer permits sounding and temperature measurements in the annulus and is very useful in diagnosing well problems.

"Live water" is locally described as a hot water aquifer with sufficient flow and permeability to wash away the fines produced in a cable tool drilling operation or major lost circulation in rotary drilling.

The space heating DHE is usually 1 ½ or 2 inch (4 or 5 cm) diameter black iron pipe with a return U at the bottom. The domestic water DHE is 3/4 or 1 inch (2 or 2.5 cm) diameter pipe. The return U usually has a 3-5 foot (1-2 m) section of pipe welded on the bottom to act as a trap for corrosion products that may fill the U preventing free circulation. Couplings should be malleable rather than cast to facilitate removal (Figure 2.2).

Other DHE types in use are short multiple tubes with headers at each end and straight pipes extending to near the well bottom with coils of copper or steel pipe at the ends. In Reno, Nevada, many DHE wells are pumped by small submersible pumps to induce hot water to flow into the well. Systems for use with heat pumps circulate refrigerant in the DHE pipes. A 20 kWt, 16 foot (5 m) prototype heat pipe system was successfully tested at least several months in the Agnano geothermal field in southern Italy (Figure 2.3) (Cannaviello et al., 1982).

The first downhole heat exchanger, locally known as a coil, was installed in a geothermal well in Klamath Falls about 1930. The temperature of the well water and the predicted heat load determine the length of pipe required. Based on experience, local heating system contractors estimate approximately 1 ft of coil per 1500 Btu per hr (1.4 kW/m) required as an average for the year. The "thermo-syphon" (or gravity) feed in standard hot-water systems) process circulates the domestic water, picking up heat in the well and releasing the heat in the radiators. Circulation pumps are required in cooler wells or in larger systems to increase the flow rate. Thermo-syphon circulation will provide 3-5 psi (0.2-0.35 bar) pressure difference in the supply and return lines to circulate 15-25 gal/min (1-1.5 L/sec) with a 10-20°F (5-11°C) temperature changed.

There are several older or cooler wells that are pumped directly into the storm sewers or canal. In most cases, the well is pumped in order to increase the flow of geothermal waters and to raise the temperature

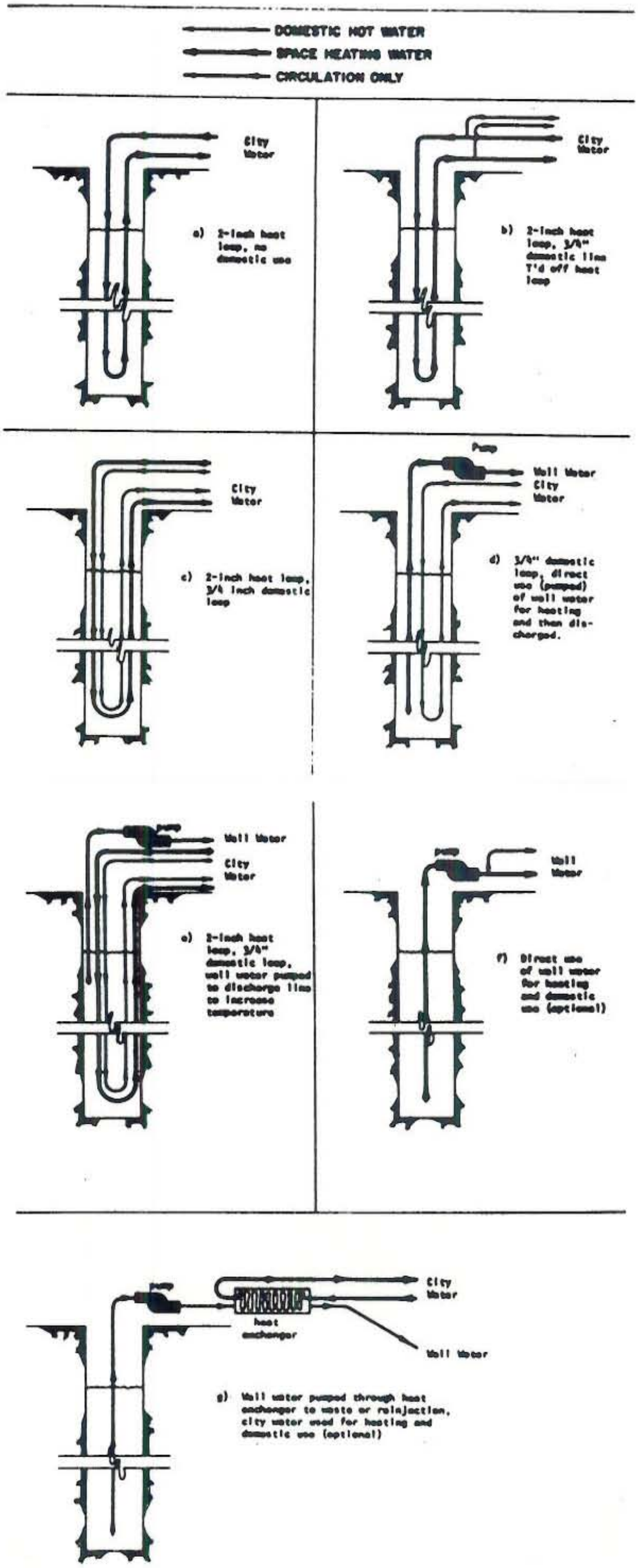

FIGURE 2.2: Downhole heat exchanger systems (Klamath Falls, OR)

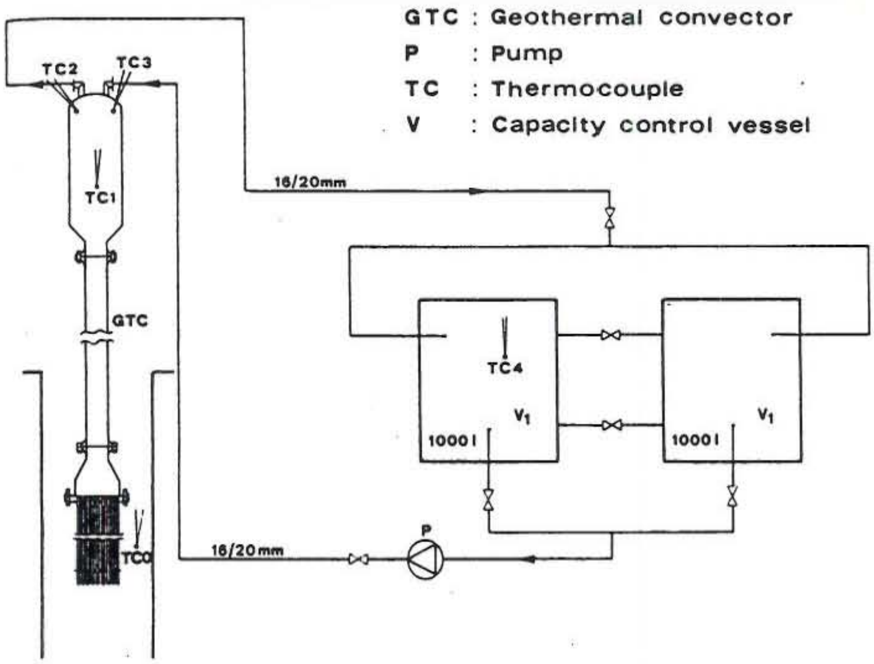

FIGURE 2.3: Experimental loop in Agnano, Italy

of the well to a level locally considered satisfactory for use in space heating, about 140°F (60°C) (see Figures 2.2d and 2.2e). In a few instances, mostly in the artesian area, well water is pumped directly through the heating system.

Considering life and replacement costs, materials should be selected to provide economical protection from corrosion. Attention must be given to the galvanic cell action between the groundwater and well casing since the casing is an expensive replacement. As indicated earlier, experience indicates that general corrosion is most severe at the air-water interface at the static water level and that 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 building and city water lines. Galvanized pipe is to be avoided since many geothermal waters leach zinc.

Considerable success has been realized with non-metallic pipe, both fiberglass reinforced epoxy and polybutylene. Approximately 100,000 feet (30,000 m) of fiberglass reportedly has been installed in Reno at bottom hole temperature up to 325°F (163°C). The oldest installations have been in about 10 years. The only problem noted has been National Pipe Taper Threads (NPT) thread failure in some pipe that was attributed to poor quality resin. The manufacturer has warranted the pipe including labor costs.

Although the thermal conductivity for non-metallic pipes is much lower, the overall heat transfer coefficient is a combination of the pipe thermal conductivity, film coefficients of any scale or corrosion products on both sides. Since the non-metallic pipe is smooth, does not corrode and scale does not stick to it, the overall heat transfer can be nearly as good.

Average DHE life is difficult to predict. For the 500 or so black iron DHEs installed in Klamath Falls, average life has been estimated to be 14 years; however, in some instances, regular replacement in 3-5 years has been required (Lund et al., 1975). 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 milliamps have been measured. In others, examination of DHEs after removal reveals long, deeply corroded lines along one side of a DHE. This may be due to continual thermal expansion and contraction while laying against the side of an uncased well. Constant movement would 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 oil, preferably turbine oil (because of environmental acceptability) as is used in enclosed tube lineshaft pumps, or paraffin in the well appears to help somewhat, but is difficult to accurately evaluate.

For some reason, DHE wells are typically left open at the top. There appears to be no good reason they could not be sealed air tight. Once the initial charge of oxygen was used up in forming corrosion products, there would be no more available since there is essentially no dissolved oxygen in the water. Closed wells appear to extend the life of the DHE (Swisher and Wright, 1990).

2.2.1 Convection cells

Although the interaction between the water in the well, water 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. Also, there must be some degree of mixing (i.e., water from the aquifer) continuously entering the well, mixing the well water, and water leaving the well to the aquifer. There are two methods of inducing convection.

When a well is drilled in a competent formation and will stand open without casing, an undersized casing can be installed. If the casing is perforated just below the lowest 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 (Figures 2.4 and 2.5). Cold surface water and unstable formations near the surfaces are cemented off above a packer. If a DHE is then installed and heat extracted, a convection cell, flowing down inside the casing and up in the annulus between the well wall and casing, is induced. The driving force is the density difference between the water surrounding the DHE and water in the annulus. The more heat extracted, the higher the velocity. Velocities of 2 feet per second (0.6 m/s) have been measured with very high heat extraction rates; but, the usual velocities are between 0.04 and 0.4 feet per second (0.01-0.1 m/s).

In Klamath Falls, it has been experimentally verified that when a well is drilled there is no flow in the wellbore. When the undersized perforated casing is installed, a convection cell is set up flowing up the inside of the casing and down the annulus between casing and well wall. When a DHE is installed and heat is extracted, the convection cell reverses flowing down in the casing (around the DHE) and up the annulus. Similar circulation patterns were noted in New Zealand using convection promoters.

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 2.6) (Allis and James, 1979). The

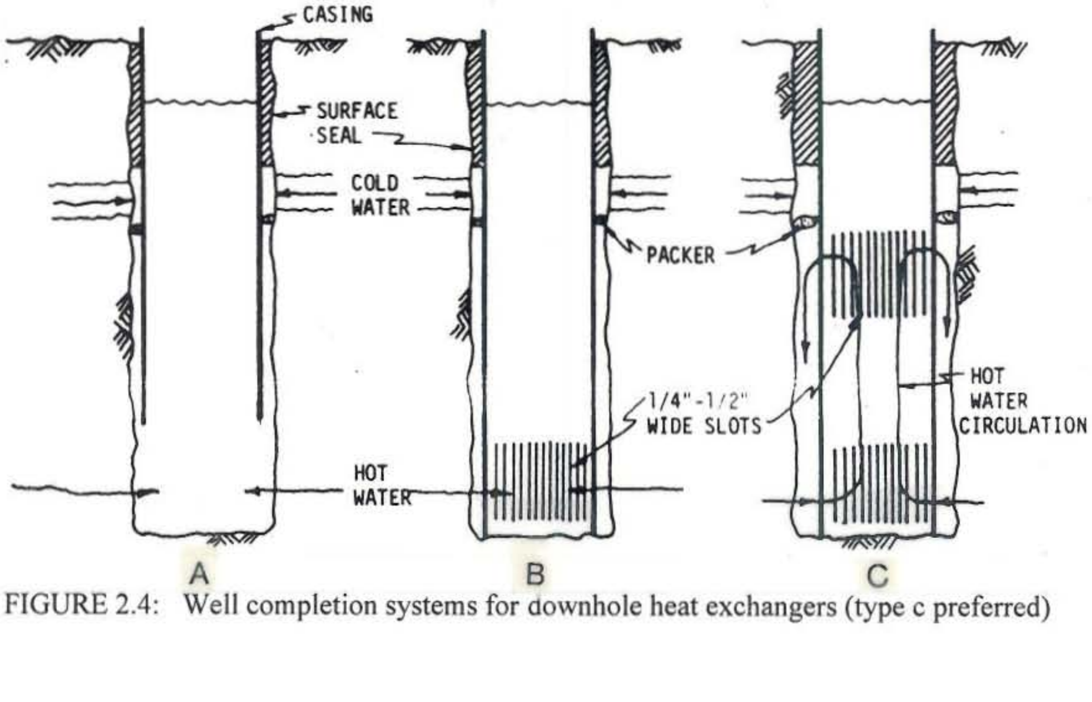

FIGURE 2.4: Well completion systems for downhole heat exchangers (type c preferred)

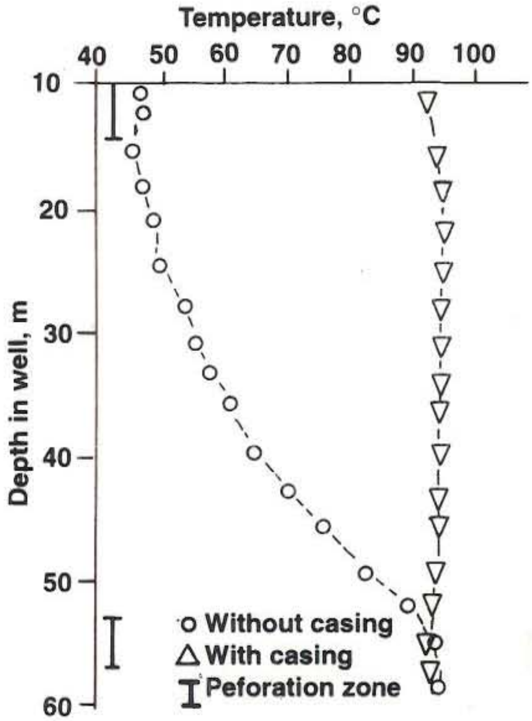

FIGURE 2.5: Temperature vs. depth for a geothermal well (with and without perforations)

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 sits on the bottom and has 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 a smaller convector pipe is used. Both lab and field tests indicate that the convection cell velocities are about the same in optimized designs and are similar to those measured in the undersized casing system. A summary of the New Zealand research is provided at the end of this section.

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

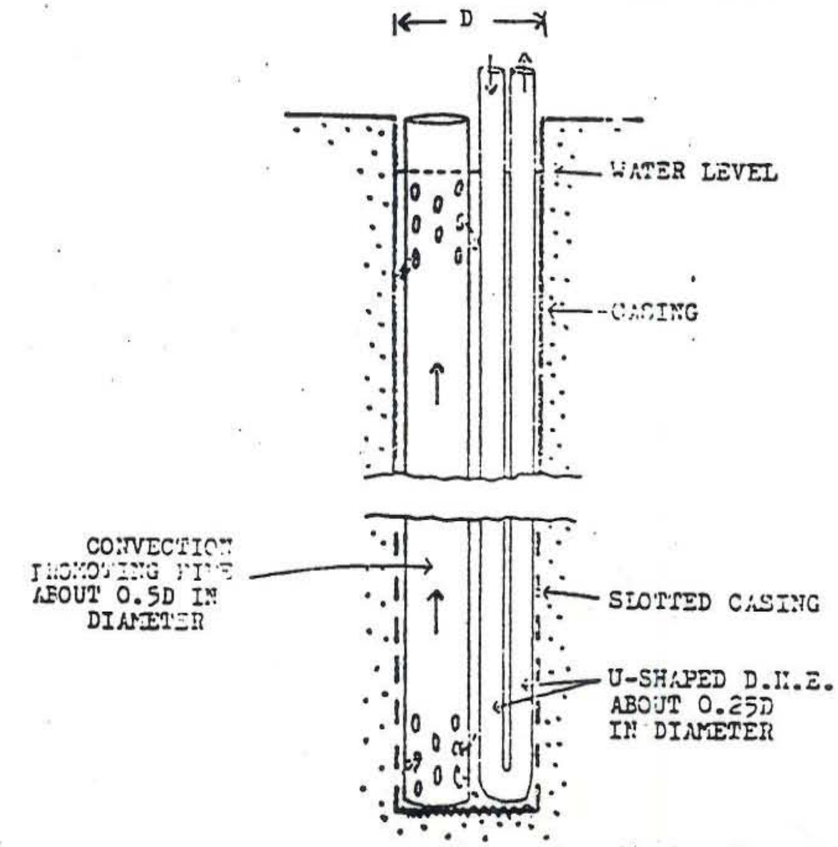

FIGURE 2.6: Convector promotion and DHE (New Zealand type)

Maximum convection rates are obtained when the casing or convector pipe are insulated from each other. 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 since the casing or convector are the same temperature as the water.

2.2.2 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 - 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

$$Rm = 1 - \frac{m\ add}{m\ total}$$

where

Rm = mixing ratio,
m add = mass flow of new water,
m total = total mass flow of convecting water.

Note that a large number indicates a smaller proportion of new water in the convection cell.

Mixing ratios vary widely between wells in the same aquifer and apparently depend on aquifer permeability. Also, 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 permeability allowing "new" hot water to enter the well or, more probably, allowing "used" cool water 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 measured and usually range from about 0.5-0.94 indicating little mixing. The theoretical maximum steady-state can be estimated if one knows the hydraulic conductivity and hydraulic gradient, and assumes some temperature drop of the water.

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

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

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

$$vAd = KAd\Delta h/\Delta l$$

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

$$Q = KAdc(T_o - T_1)\Delta h/\Delta l$$

where

A = Cross section of well in the aquifer or the perforated section,
d = Density of water,
c = Specific heat,
T_0 = Aquifer temperature,
T_1 = Temperature of water returning to the aquifer.

Multiplying the above by *Rm* -1, or about 0.4 to 0.5, one can determine the expected steady-state DHE output. The most important factor in the equation is *K*. This value can vary by many orders of magnitude - even in the same aquifer - depending on whether major fractures are intersected, 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 about 3 feet per year (0.3-1.0 m/yr) (Allis, 1981). The hot aquifer is generally encountered in mixed or inter-bedded layer of fine sand and silt stone. In Klamath Falls, on the other hand, where the hot aquifer is in highly fractured basalt and coarse cinders, specific velocity is estimated at 20 to 150 feet per day (6 to 46 m/day), perhaps higher in localized areas. Values of K in seven wells in Moana were estimated at 3×10^{-4} ft per second (1×10^{-7} meters per second). This implies a factor of 10 thousand to 10 million difference in the steady-state output. Indeed 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 gallons per day (6800 L/day) and the pump is thermostatically controlled. This is designed to switch on the pump if the DHE temperature drops below some predetermined level, usually about 120°F (49°C). 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 good design procedure. Culver and Reistad (1978) presented a computer program that appears to predict DHE output to within 10-15% 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), and later modified by Lienau and Culver as documented in (Culver, 1990). The program enables optimum geometric parameters to be chosen to match a DHE to a load if one assumes a mixing ratio.

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 and that with low permeability, a convection promoter can promote thermal storage and, thereby, increase non-steady state output.

Permeability can be rather accurately estimated with relatively simple Hvorslev plots used in well testing. Relating the permeability, thus obtained to mixing ratios typical in other permeabilities, could give an estimate of the mixing ratio one could use in the computer program. The problem is, there seems to be no middle ground data available, only very high and very low permeabilities, and precious little of that.

2.3 NEW ZEALAND EXPERIENCE WITH DOWNHOLE HEAT EXCHANGERS

2.3.1 Early work by R.G. Allis and R. James (Allis and James, 1979)

The research of Allis and James was into the use of domestic wells, which were used for low-grade direct heating, and potentially powerful steam-water wells, which often had relatively cool water over most of their depth. Thermal convection was inhibited by the large aspect ratio (length to diameter ratio) of the well. The domestic wells could not use the Klamath Falls types downhole heat exchanger since the well had already been cased without perforations, and thus heat could only be extracted over a very short length near the bottom. In a similar manner, deep high-temperature geothermal wells are sometimes difficult to discharge because of the great depth of cool water overlying the hot zone.

In their laboratory research, they found that, if a pipe (promoter) is inserted into their model of a well, natural convection will occur and the hot water will flow to the top of the well (Figure 7). The diameter of the pipe determined whether the hot water would flow up the pipe and down the annulus, or the reverse. The promoter pipe should at least be slotted on the lower end, especially if is rests on the bottom of the well.

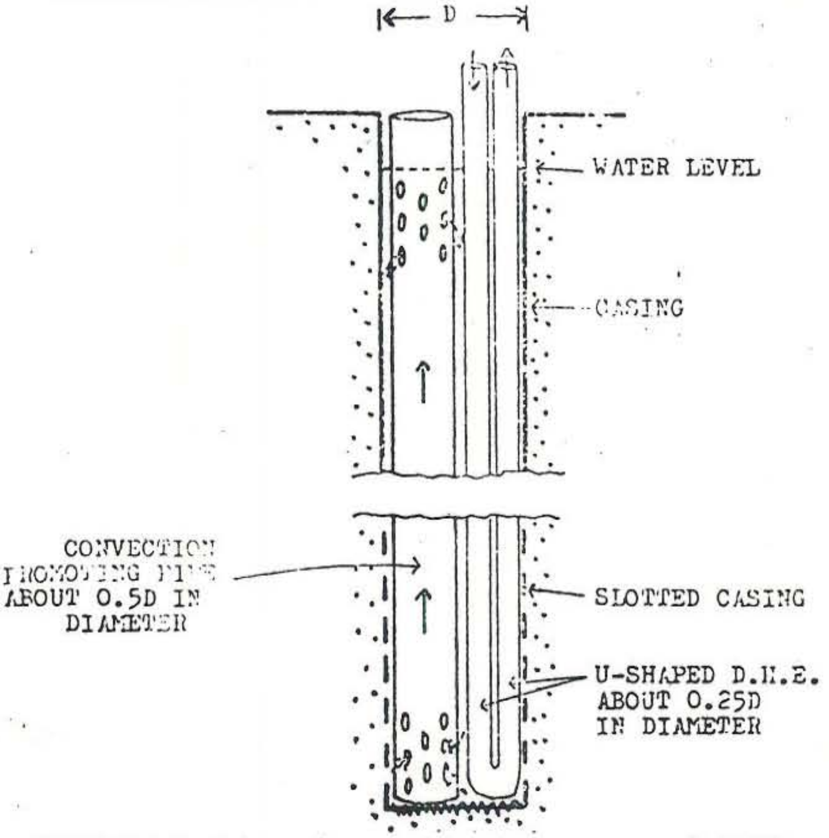

FIGURE 2.7: Scheme for optimizing the heat output of a DHE

The research also found that in domestic wells, the promoter pipe improved heat output by 60 to 120%, depending upon the diameter. The DHE is the annulus gave higher heat output from the DHE than if placed inside the promoter (Figures 8 and 9). The maximum flow (vertical) in the well occurs when the frictional pressure-drop in the promoter pipe equals that of the annulus. Thus, the promoter pipe should equal 0.5 the well diameter if the DHE is placed in the annulus and, should equal 0.7 the well diameter if place inside the promoter. The DHE pipe should be 0.25 the well diameter (Figure 7). Using these recommended dimensions, the quantity of thermal energy available to the DHE is limited mainly by the existing bottom hole permeability. If the temperature of the well water is less than boiling (100°C), then stiff plastic pipe can be used since it is a poor thermal conductor, and is also comparatively smooth.

For potentially powerful (high-temperature) geothermal wells which are difficult to discharge, a 2-inch (5-cm) diameter pipe, positioned beneath the water level (Figure 10), should raise wellhead pressure (by promoting internal convection) to the point where controlled, spontaneous discharge is possible. This promoter pipe is place approximately 160 feet (50 m) below the water surface and is slotted on both ends. Once the well head valve is opened and production conditions exist, the pipe should only slightly restrict vertical mass discharge which takes place both in the annulus and in the pipe.

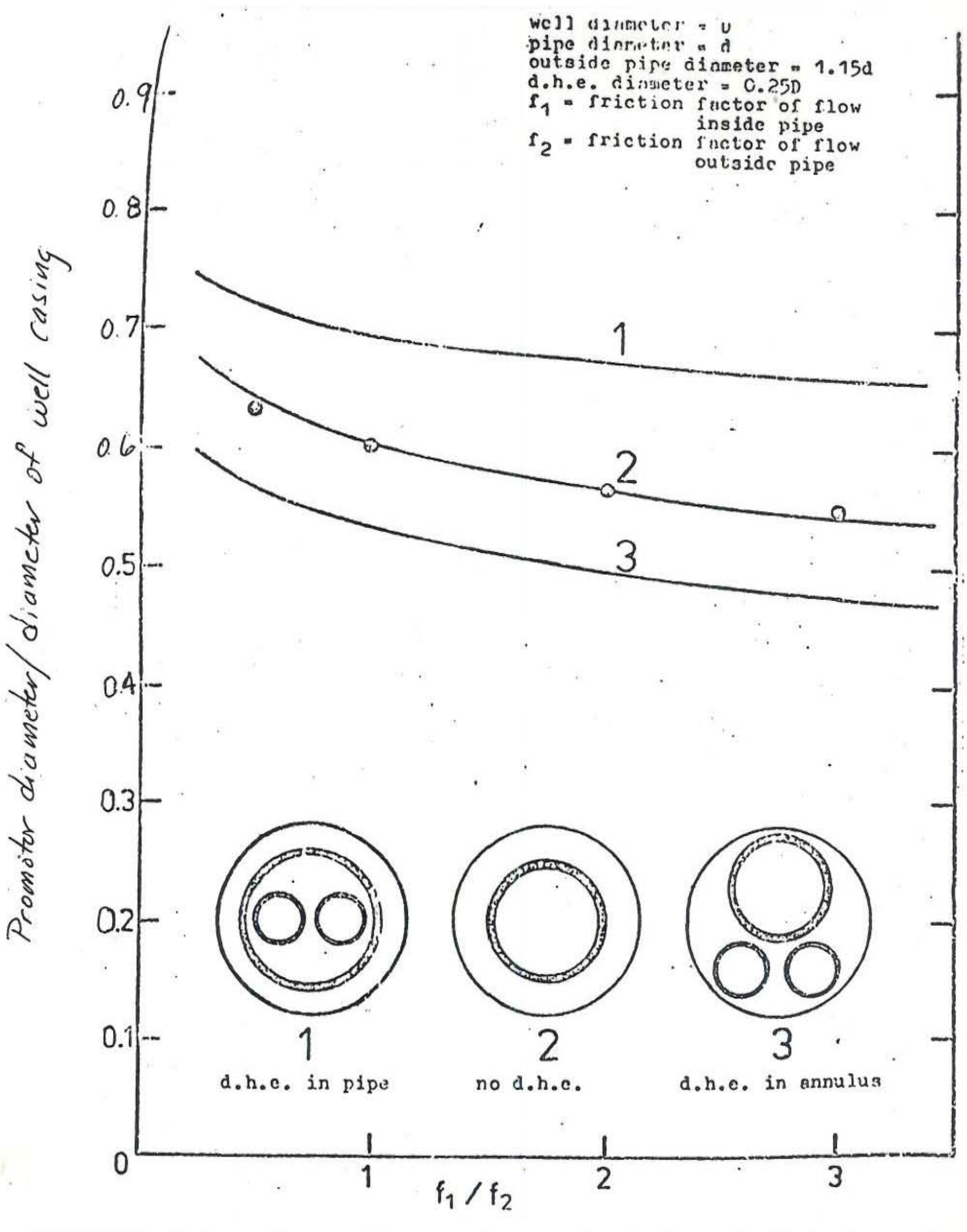

FIGURE 2.8: Optimum diameter of the convection promoting pipe for varying friction factor ratio; in the three cross-sections, $f_2/f_1 = 2$. Refer to text for derivation of the 4 points and the curves

This method of stimulating vertical convection, and thus promoting uniform high temperature throughout the well column, is preferred instead of using a downhole pump or airlifting with the associated environmental problems of fluid disposal.

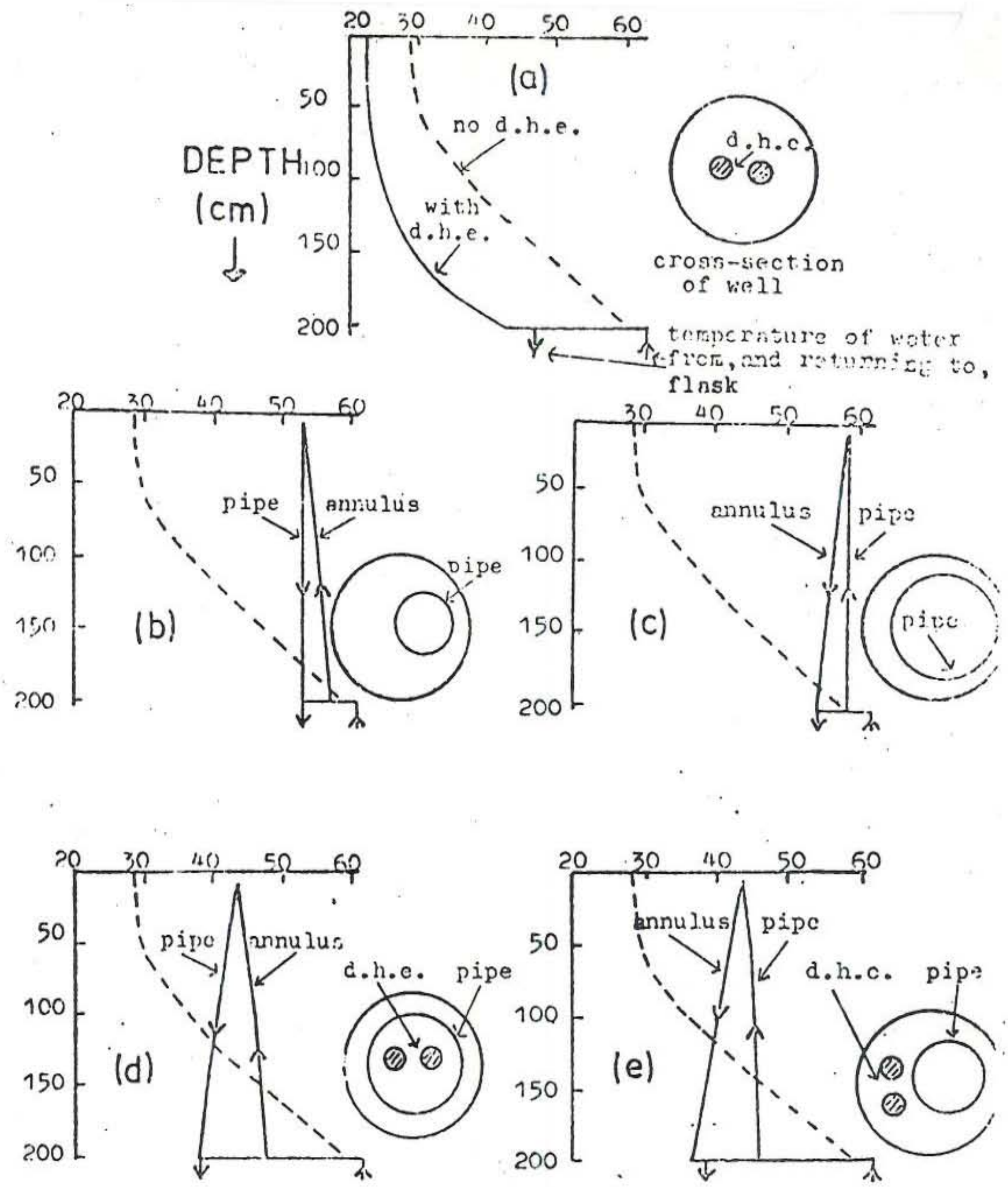

FIGURE 2.9: Characteristic temperature and flow regimes observed in a laboratory model of a well; configuration of convection promoting pipe and/or DHE is shown in the cross-section in each case

2.3.2 Experimental work at the University of Auckland (Freeston and Pan, 1983)

The investigations of Freeston and Pan are based on both laboratory and field work in the Taupo area of New Zealand, and using results of work in the Moana area of Reno, Nevada (Allis, 1981). They looked at the heat transfer and flow mechanisms in the vertical convection cell of wells with DHE's. Conclusions were drawn from computer analysis and subsequent field testing. The work by Allis (1981) showed that in order to obtain 10 kW continuously for 24 hours (peak of 20 to 30 kW) from a DHE to supply a household from a 20 cm diameter well 50 m deep in a reservoir where the hydraulic gradient was 1%, a permeability of about 50 darcies (5×10^{-4} m/s) is necessary. Above 50 darcies a promoter can be utilized; below, a small pump will be necessary. However, Allis did note that in wells with permeability below 50 darcies, the stored heat in the form of hot rock adjacent to the well may provide

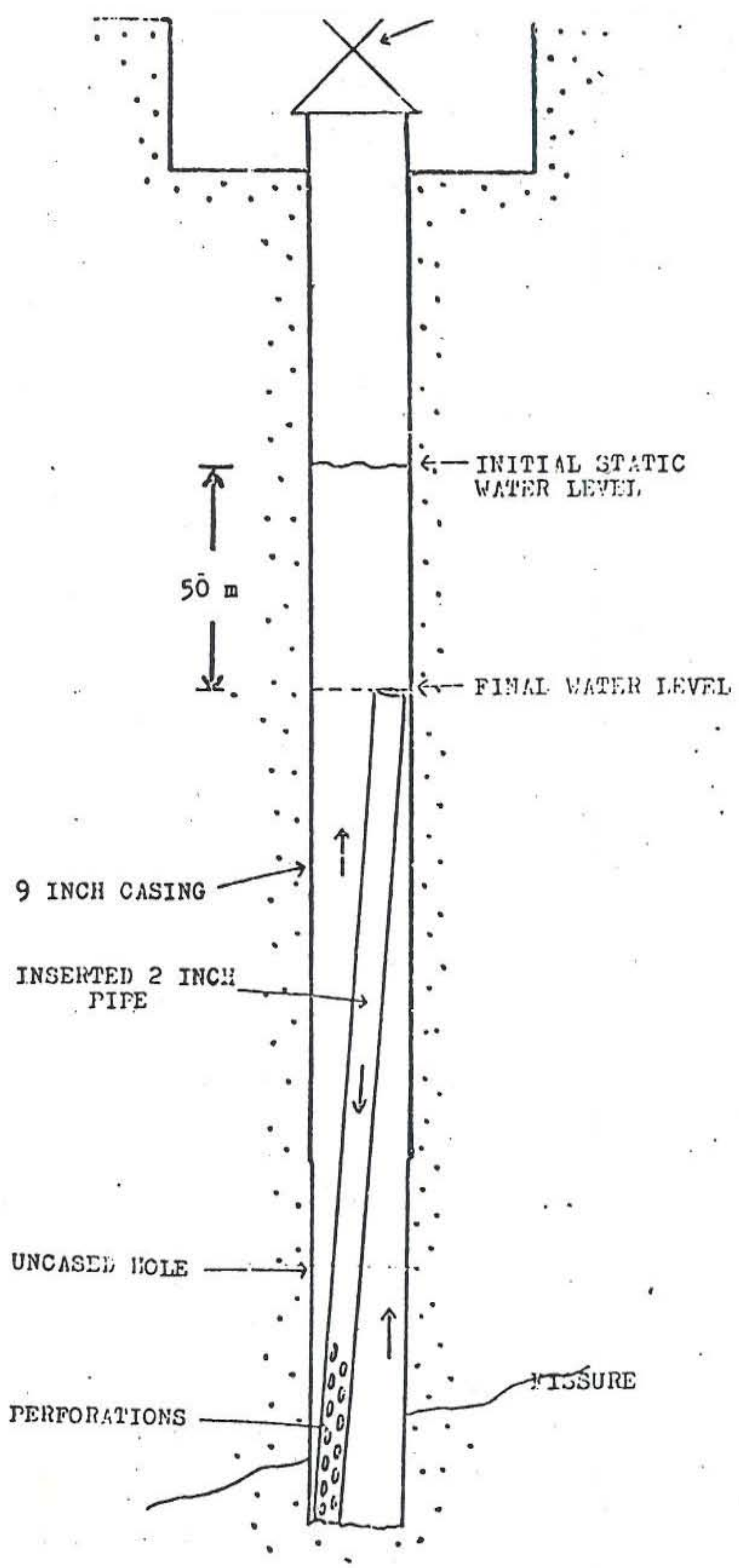

FIGURE 2.10: Scheme for promoting convection in a geothermal power well

sufficient heat for days or even weeks. However, in the long-term, the well will cool off. The use of a convector pipe will not improve the long-term heat output of the DHE. A convector pipe only keeps the well water mixed and does not draw in fresh hot water. As mentioned above, these wells require some form of pumping to induced the required heat flow.

Based on their work and that of Allis, it was concluded that convective (vertical) flow must be favoured instead of conductive (horizontal) flow between the promoters and the annulus, in order to maximize the output of DHEs. The relationship that best defined these conditions is

$$D^2/L$$

where
D = diameter of the promoter pipe in cm,
L = length of the pipe in m.

For $D^2/L < 1$, then conductive heat flow dominates.

For $D^2/L > 1$, then convective heat flow dominates.

Since heat flow should be convective, (1) long and small diameter promoters should be avoided as they do not generated significant circulation of the well fluid, and (2) it is best to have promoters with as large a diameter as possible, and use a low-conductivity material to get maximum vertical convection.

2.3.3 Field work in Rotorua (Dunstall and Freeston 1990)

A series of test were conducted on a U-tube DHE installed in a 100 mm diameter well which previously provided a steam/water mixture to heat a building in Rotorua. The field work was conducted to study the fluid temperatures inside the heat exchanger tubes resulting in a better understanding of the heat transfer processes involved in a typical Rotorua DHE/well system.

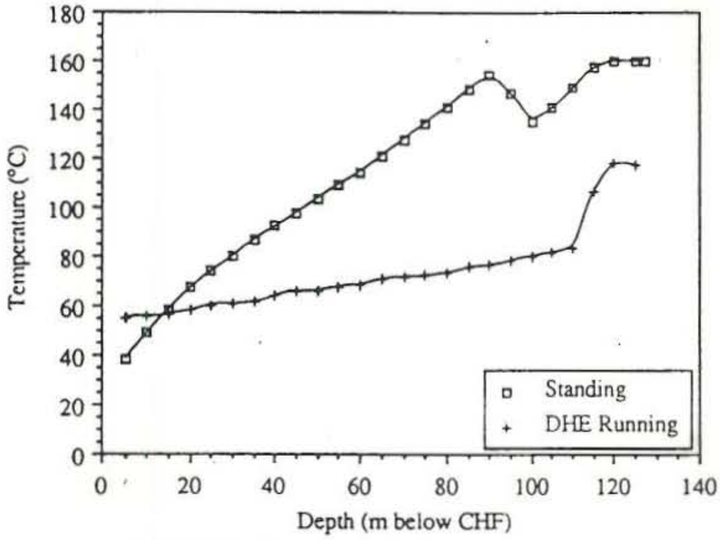

FIGURE 2.11: Downhole temperature

The DHE/well system consisted of a 123 meter deep well cased to 112 meters with a bottom hole temperature of 160° C. A 25-mm diameter U-tube produced a maximum output of 150 kW. The standing vs. DHE running (in operation) temperature profile of the well is shown in Figure 11.

The results of the test showed that the temperature increased below the uncased portion and thus almost all heat exchange occurred at the feed zone. In the cased portion, heat was lost to the return leg (cold leg) of the DHE from the supply leg (hot leg) (Figure 12).

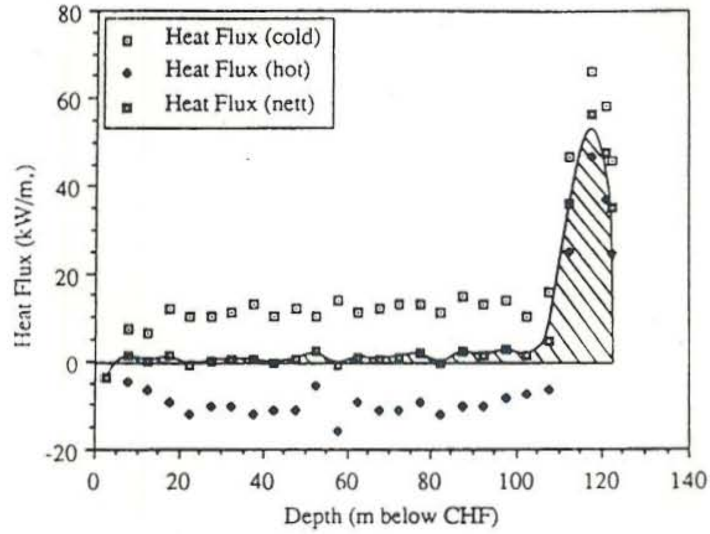

FIGURE 2.12: Heat flux (1.2 l/s)

In order to prevent this heat loss they recommended that either (1) the return leg from the casing bottom to the surface be insulated, or (2) that a smaller diameter return pipe be used, thus producing higher velocities inside it resulting in less heat loss.

The flow rate in the DHE was also varied from just above 0.4 L/s to 1.2 L/s (Figure 13). As was expected, the heat output in kW increased with flow rate, however the typical output curve will flatten as flow rates become "high" (Figure 14). At "high" flow rates the return temperature has a tendency to fall off, negating some of the gain, and pumping costs also increase. The likely increase in heat output would probably more than cover

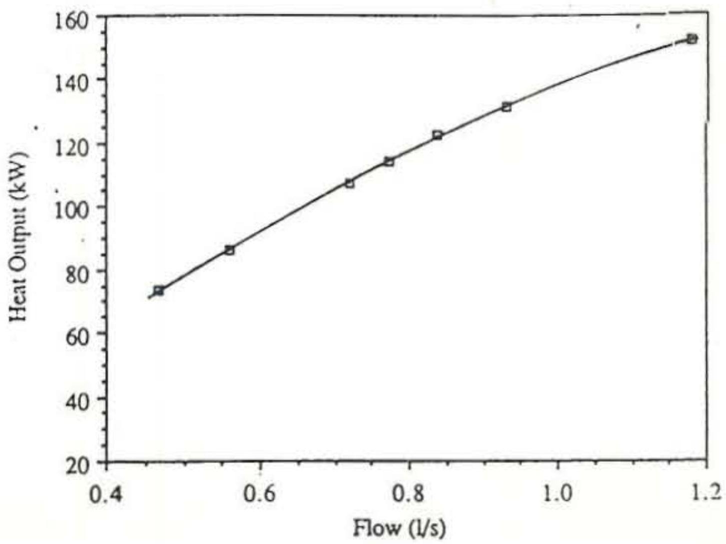

FIGURE 2.13: Heat output vs. flow rate

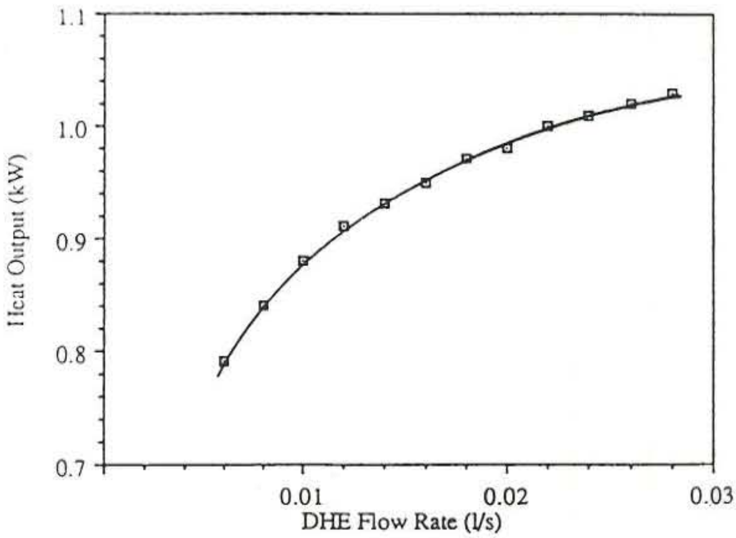

FIGURE 2.14: Typical output curve for a DHE (taken from numerical data by Pan, 1983)

CIRCULATION DIRECTION

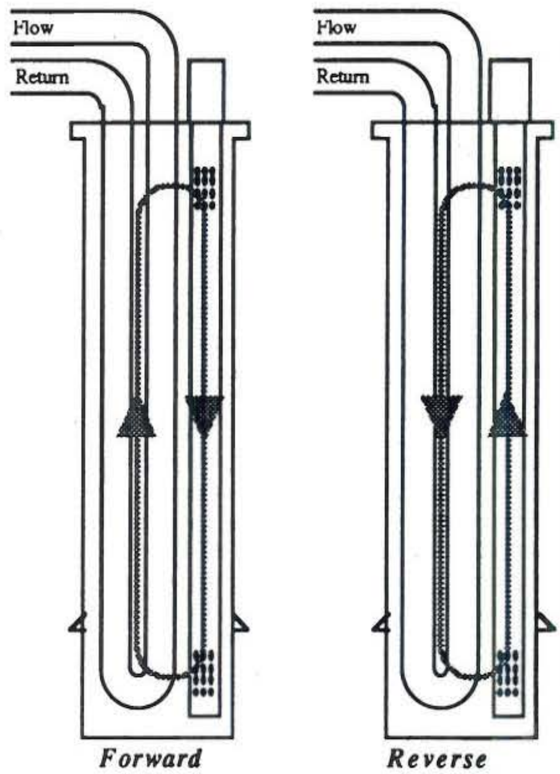

FIGURE 2.15: Circulation flow directions

the increase pumping cost for moderate increases in flow rate. They concluded that the use of promoters would greatly enhance the well heat output.

2.3.4 Experimental work at the University of Auckland
(Hailer and Dunstall, 1992)

Two different materials, copper and PVC, were used to construct two identical U-tube DHEs, which were tested over a range of DHE flow and well aquifer cross flow rates. During most tests a PVC convection promoter pipe was fitted in the well, to allow a bulk circulation of the well fluid. Various combinations of the DHE pipes inside and outside the promoter pipe were investigated. Some comparisons were made to results obtained during full scale testing in a shallow Rotorua well.

The basic conclusion, as found by others, was that the output from the DHE increased with increasing cross flow in the aquifer at the well bottom and with increasing DHE flow rates. Both relationships appear nearly linear at low flow rates but the performance improvement tapers off as the flow rate increases (as was seen in the work by previous work by others discussed earlier). In larger diameter wells, the DHE supply (hot leg) temperature was less than the well temperature, therefore, the was no heat loss by conduction.

When a promoter pipe was installed the bulk well circulation can be obtain in either a forward or reverse direction (Figure 15), with the forward direction (up the annulus and down inside the promoter pipe) yielding a higher heat output of the DHE by 10 to 20% (Figures 16 and 17). Thus, they recommended that the DHE return (cold) leg be placed inside the promoter pipe with the supply (hot) leg in the annulus to produce forward flow. Otherwise, forward flow may have to be stimulated by use of an airlift pump. It was also found that the heat output of the PVC DHE is quite high compared to the copper DHE, considering its thermal conductivity. This was due to the low percentage of the total heat transfer resistance represented by the tube and the fact that no heat was lost in the return leg. Thus, they recommended that a hybrid DHE consisting of a copper supply and a PVC return leg be used to provide higher heat transfer rates, as compared to either of the single material DHEs.

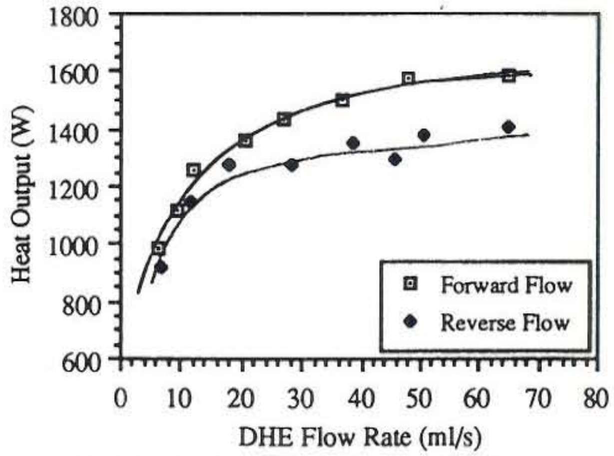

FIGURE 2.16: Heat output vs. DHE flow (32 mm promoter) (Cross flow rate 25.8 ml/s)

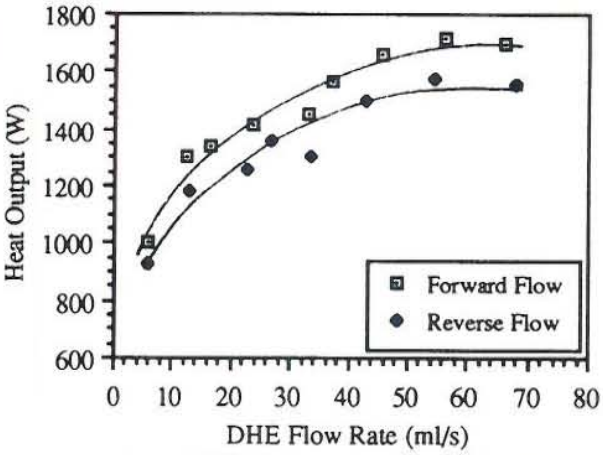

FIGURE 2.17: Heat output vs. DHE flow (32 mm promoter) (Cross flow rate 31.4 ml/s)

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LECTURE 3

GREENHOUSES

3.1 INTRODUCTION

A number of commercial crops can be raised in greenhouses, making geothermal resources in cold climates particularly attractive; however, growth can be optimized in warmer climates. These include vegetables, flowers (potted and cut), house plants, and tree seedlings. The optimum growth temperature of cucumbers, tomatoes, and lettuce is shown in Figure 3.1 below (Barbier and Fanelli, 1977). Cucumbers grow best in the temperature range 77°-86°F (25°-30°C), tomatoes near 68°F (20°C), and lettuce at 59°F (15°C), and below. The growing time for cucumbers is usually 90 to 100 days; while, the growing cycle for tomatoes is longer, in the range 9 to 12 months. The use of geothermal energy for heating can reduce operating costs (which can account for up to 35 percent of the product cost) and allows operation in colder climates where commercial greenhouses would not normally be economical. In addition, greenhouses can be suited to large quantities of relatively low-grade heat. Furthermore, better humidity control can be derived to prevent condensation (mildew), botritis, and other problems related to disease control (Schmitt, 1981).

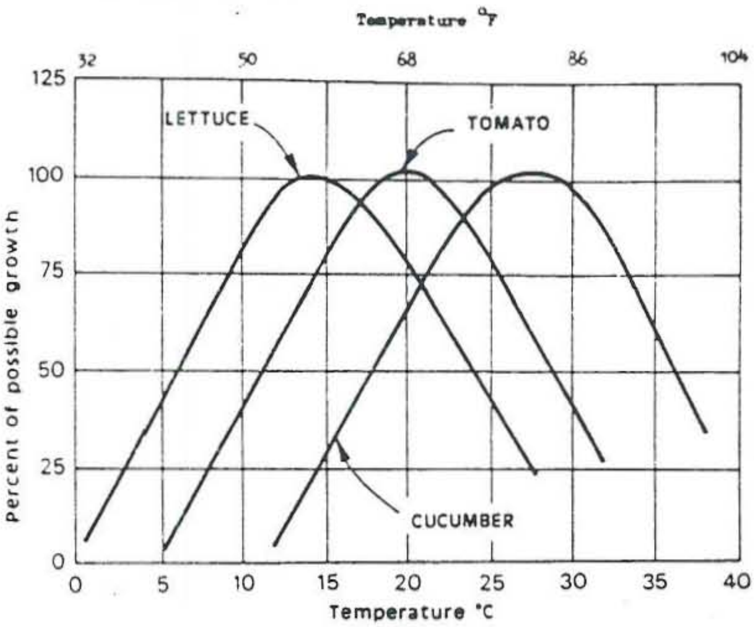

FIGURE 3.1: Optimum growing temperature for selected agricultural products

3.2 EXAMPLES OF GEOTHERMALLY HEATED GREENHOUSES

There are numerous uses of geothermal energy for greenhouse heating throughout the world, estimated at 12,000 TJ/year (1995). In the USSR, it is reported that over 6,200 acres (2,500 ha) of agricultural land are heated by geothermal of which 25 acres (10 ha) are covered by greenhouses. In Hungary, over 300 acres (120 ha) of greenhouses are heated geothermally. Many of these greenhouses are built on rollers, so they can be pulled from their location by tractors, the ground cultivated with large equipment, and then the greenhouse returned to its location. In addition, to minimize the cost, much of the building structure pipe supporting system also acts as the supply and radiation system for the geothermal fluid. Greenhouses cover about 20 acres (8 ha) in Japan where a variety of vegetables and flowers are grown. Individual greenhouses, operated by farmers and covering 3,200-16,000 ft^2 (300-1,500 m^2) use 158°-212°F (70°-100°C) geothermal water. Many large greenhouses totalling about one acre (0.4 ha), are operated as tropical gardens for sightseeing purposes. New Zealand has numerous greenhouses using geothermal hot water and steam. At the Land Survey Nursery in Taupo, greenhouses are heated by geothermal steam and soil is sterilized (pasteurized) at 140°F (60°C) to kill insects, fungus, worms, and some bacteria. In Iceland, over 35 acres (14 ha) are heated, including a greenhouse, restaurant, and horticulture college at Hveragerdi. Everything from bananas, coffee beans, cacti, and tropical flowers to the standard tomatoes and cucumbers are grown in these greenhouses. Studies of the economic feasibility of greenhouses in Iceland have been based on theoretical 82 acres (33.5 ha) facility, which would grown asparagus on 25 acres (10 ha), flower seedlings on 2 acres (1 ha), and cucumbers on 1 acre (0.5 ha). Projected profit on the initial investment would amount to 11 percent before taxes, and the greenhouses would provide jobs for 250 persons (Hansen, 1981).

Numerous geothermally heated greenhouses exist in the U.S.; several examples are described as follows. In Salt Lake City, Utah, a 250,000 ft^2 (23,000 m^2) greenhouse is using 200 gpm (12.6 L/s) of 120°F (49°C) water for heating. Utah Roses, Inc., is producing cut roses for a national floral market. The 4,000 ft (1,200 m) geothermal well has replaced a natural gas/oil heating system. Fifteen miles (24 km) south of Klamath Falls, Oregon, on the Liskey ranch, approximately 50,000 ft^2 (4,600 m^2) of greenhouses are heated with 195°F (90°C) water from a 270 ft (82 m) deep well. One of the greenhouses consists of four 42 ft by 150 ft (13 m by 46 m) buildings connected to form one large complex. Initially, seedlings were raised for federal and private agencies. More recently, succulents, cacti, and potted plants are raised. All plants are grown in trays on raised tables, with the heat supplied by pipes under each table (Laskin, 1978; Lund, 1994) At Honey Lake, California, near Susanville, over thirty 30 ft by 124 ft (9 m by 38 m) quonset-design greenhouses were used to raise cucumbers and tomatoes. The vegetables are raised by hydroponics, with the heat being supplied by forced air heaters. Production rates were about 1,500 pounds (680 kg) of cucumbers per unit per week and 850 pounds (358 kg) of tomatoes per unit per week. The cover of each greenhouse consisted of two layers of 6 mil sheeting (plastic). A small electric air blower continually inflated the area between the two layers and maintained an air space of about 6 inches (15 cm), resulted in heat savings of approximately 40 percent over conventional coverings. The savings, using geothermal heat as compared to conventional fuel, averages $4,500 per acre per year ($11,100/ha/year)(Boren and Johnson, 1978). A similar analysis has been made for a greenhouse provided in La Grande, Oregon. The double 6 mil polyethylene covering required 45 percent less heating than single layer (Higbee and Ryan, 1981) (Figure 3.2).

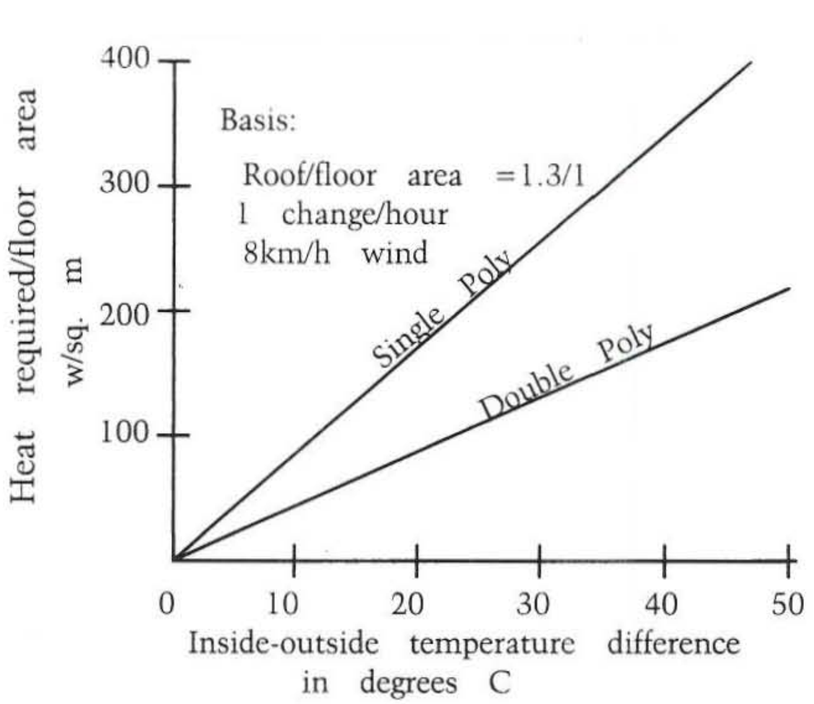

FIGURE 3.2: Example of heat requirement for greenhouses in La Grande, Oregon (outside design temperature = 1°F [-17°C]

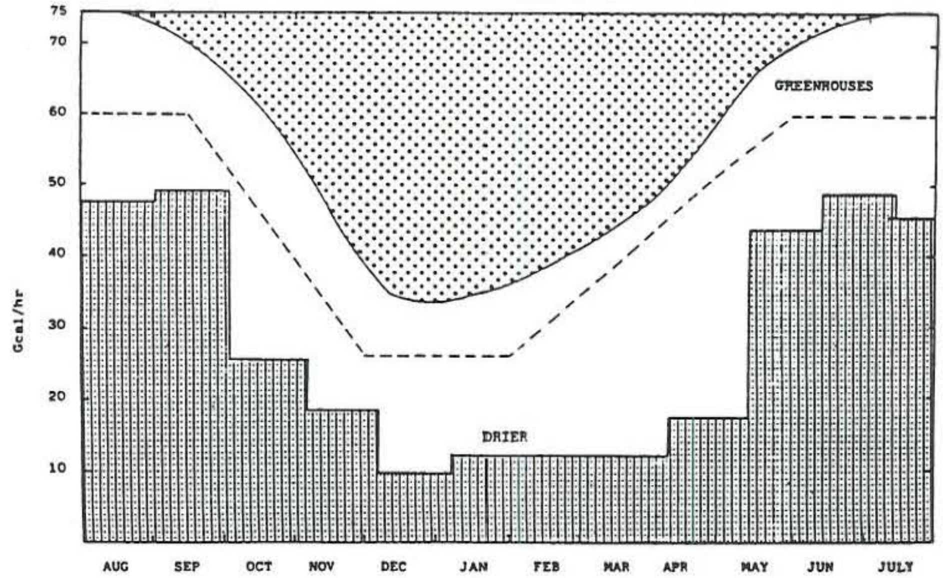

FIGURE 3.3: Geothermal energy consumed by greenhouses and drier at Mt. Amiata, Italy

One of the world's largest geothermally-heated greenhouse operations is near Mt. Amiata, Italy. Approximately 54 acres (22 ha) of greenhouses are used to produce potted plants and flowers. Waste heat is supplied from a 15 MWe power plant. The greenhouse is operated in conjunction with an experimental drier that operates during the summer months when greenhouse heating is low. This combination maximizes the utilization of the geothermal heat as shown in Figure 3.3 (Lund, 1987).

Approximately 50 acres (20 ha) of greenhouses are heated geothermally in the United States. The largest single greenhouse operation is at Animas, in southwestern New Mexico, where 4 hectars are used for raising cut roses. The five leading greenhouse locations are (Table 3.1):

TABLE 3.1: The five leading greenhouse locations in the United States

Location	**Annual energy (TJ)**	**Capacity (MWt)**	**Load factor**	**Area (Acres / ha)**	**Product**
Animas, NM	75.8	(6)	40.0%	10.8 / 4.4	Cut roses and bedding plants
Buhl, ID	63.0	(5)	40.0%	5.7 /2.3	Potted and bedding plants
Sandy, UT	47.1	(4)	25.0%	4.7 / 1.9	Cut roses
Wendel/Susanville, CA	43.4	(3)	43.2%	3.7 /1.5	Vegetables
Helena, MT	25.0	(2)	45.0%	2.0 /0.8	Cut roses

3.3 GENERAL DESIGN CRITERIA

Greenhouse heating can be accomplished by (1) circulation of air over finned-coil heat exchangers carrying hot water, often with the use of perforated plastic tubes running the length of the greenhouse in order to maintain uniform heat distribution, (2) hot-water circulating pipes or ducts located in (or on) the floor, (3) finned units located along the walls and under benches, or (4) a combination of these methods. A fifth approach is using hot water for surface heating. Surface-heated greenhouses were

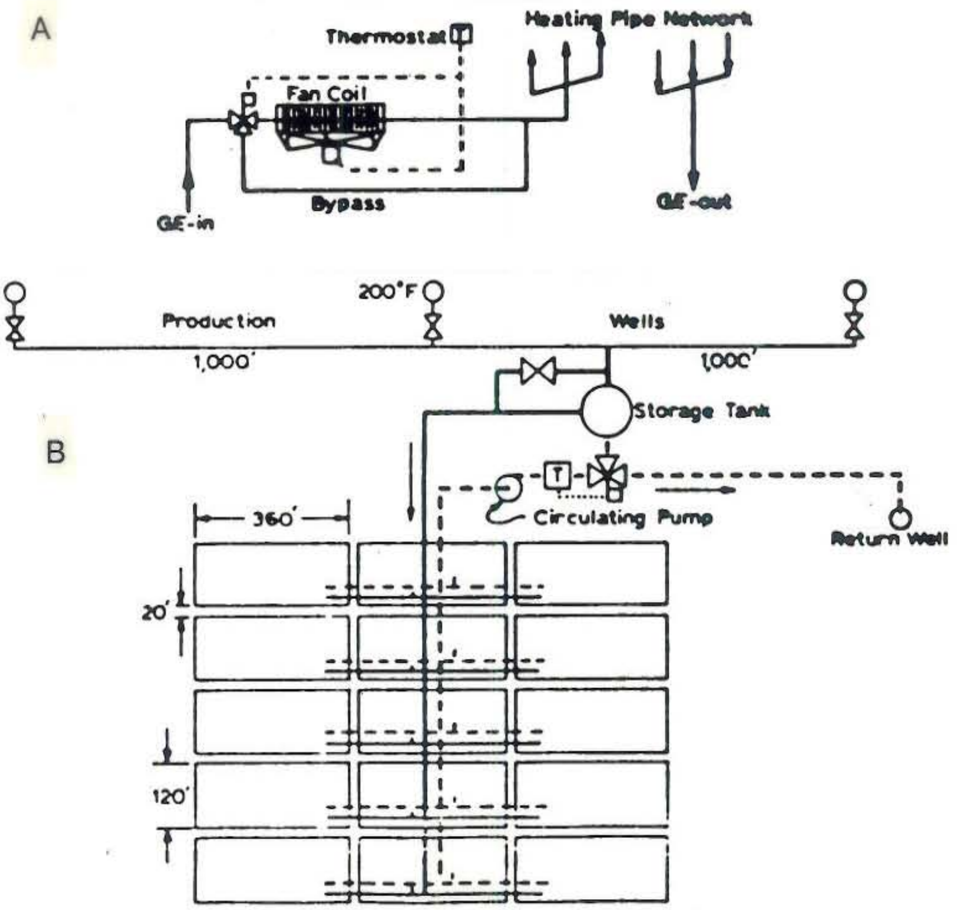

FIGURE 3.4: a) Unit heating system design (three per house); b) Six-hectar greenhouse complex

developed several decades ago in the USSR. The application of a flowing layer of warm water to the outside surface of the greenhouse can provide 80-90 % of the energy needed. The flowing layers of warm water prevent snow and ice from accumulating.

The most efficient and economical greenhouse development consists of large structures covering one-half to a full acre (0.2-0.4 ha) (Figure 3.4). A typical size would be 120 to 360 ft (36 by 110 m), constructed of fiberglass with furrow-connected gables. Heating would be from a combination of fan coils connected in series with a network of horizontal pipes installed on outside walls and under benches. A storage tank would be required to meet peak demand and for recirculation of the geothermal water to obtain the maximum temperature drop. Approximately 100 gpm (6.3 l/s) of 140°-180°F (60°-82°C) water will be required for peak heating. The average is much less. Fortunately, most crops require lower nighttime than daytime temperatures. Greenhouse construction and outfittings will run from \$5 to \$10 per square foot (\$54 to \$108 per m^2).

3.4 GREENHOUSE CONSTRUCTION (Rafferty, 1985 & 1991)

3.4.1 Construction materials

In order to make an evaluation of geothermal heating systems for greenhouses, it is first necessary to examine the different heating requirements imposed by various construction methods.

At one time, greenhouses were constructed exclusively of cypress wood frames and single glass panels. Recent years have seen substantial changes in construction techniques and materials. In general, construction may be considered to fall into one of the following four categories:

1. Glass,
2. Plastic film,
3. Fiberglass or similar rigid plastics, and
4. Combination of two and three.

All of the above are generally constructed of steel or aluminum frames. See Figure 3.5 for common greenhouse shapes.

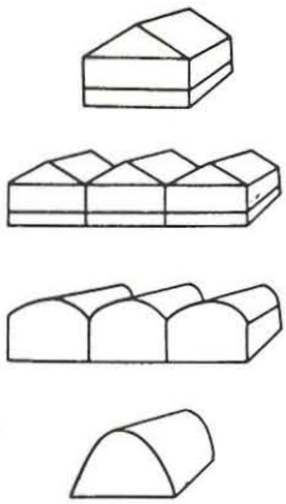

FIGURE 3.5: Structural shapes of commercial greenhouses

Glass greenhouses are the most expensive to construct due to both the cost of the glazing material and the requirement for a stronger framework to support the glass. In many cases, fiberglass panels are employed on the side and end walls of the structure. Building profile is generally of peaked design, with 36 and 42 ft (11 and 13 m) widths, and lengths in 20 ft (6 m) increments most common. This type of greenhouse is preferred by growers whose plants require superior light transmission qualities. In addition to offering the highest light quality, the glass greenhouse also has the poorest energy efficiency. Heating costs are high due to the poor insulating quality of single glazing and the high infiltration of cold air through the many "cracks" in the construction. The issue of high transmission loss has been addressed in recent years through the introduction of new, double glazing panels for glass houses. However, due to the expense of these panels and their effect upon light transmission, most glass greenhouses remain single layer.

Plastic film greenhouses are the newest variation in greenhouse construction techniques. This type of structure is almost always of the arched roof or "quonset-hut" design. The roof can come all the way down to the ground or can be fitted with side walls. The side walls, if employed, and end walls are generally of fiberglass construction. Maintenance requirements for the plastic film are high in that it generally requires replacement on 3 year intervals or less, depending on the quality of the material. Most plastic film houses employ a double layer of film separated by air space. The air space is maintained by a small blower which pressurizes the volume between the layers. This "double poly" design is a very energy efficient approach to greenhouse design. It not only reduces transmission losses (losses through the walls and roof) by 30-40% (see Figure 3.2), but also substantially reduces infiltration (in leakage of cold air). Although the plastic film tends to lose more heat than glass through radiation, the net effect is a reduction in heating requirements compared to glass construction. Infiltration is reduced because the "cracks" present in other types of construction are eliminated through the use of the continuous plastic film. As a result, there is less opportunity for the cold outside air to penetrate the structure. The superior energy efficiency on the film construction comes at the price of reduced light transmission, however. As a result, highly light-sensitive crops cannot be grown in the double-poly greenhouse as successfully as in other constructions. These greenhouses are generally constructed in 30 ft (9 m) widths with 100 and 150 ft (30 and 46 m) lengths.

Fiberglass greenhouses are similar in construction to the glass houses described above. They are generally of peaked roof design, but require less structural support as a result of the lower weight of the fiberglass. Heat loss of the fiberglass house is about the same as the glass house. Although the fiberglass material has a lower conductivity than glass, when considered in the overall building heat loss, this has little effect.

3.4.2 Heating requirements

In order to select a heating system for a greenhouse, the first step is to determine the peak heating requirement for the structure (Rafferty, 1994). Heat loss for a greenhouse is composed of two components: (1) transmission loss through the walls and roof, and (2) infiltration and ventilation losses due to the heating of cold outside air.

To evaluate transmission loss, the first step is to calculate the surface area of the structure. This surface area should be subdivided into the various materials employed (i.e., square feet (square m) of double plastic, fiberglass, etc.).

After determining the total surface area (SA) of the various construction materials, this value is then combined with a design temperature difference (DTD) and a heat loss factor (HLF) for each, to calculate the total transmission heat loss:

$$Transmission\ heat\ loss = (SA_1 \times DTD_1 \times HLF_1) + (SA_2 \times DTD_2 \times HLF_2)$$

The design temperature difference is a function of two values: design inside temperature and design outside temperature. The inside design value is simply the temperature to be maintained inside the space (usually in the 50°-65°F [10°-18°C] range). The design outdoor temperature is <u>not</u> the coldest outdoor temperature recorded at the site. It is generally considered to be a temperature which is valid for all but 22 hours per year during the heating season. Acceptable values for various locations are generally available from state energy offices or organizations such as American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE).

The final value in the transmission heat loss equation is the heat loss factor (HLF or U-Factor). Acceptable values for various materials are shown in Table 3.2.

TABLE 3.2: Glazing material U-values (based on 24 km/hr wind speed)

Material	kJ/m² hr °C	W/m² °C
Glass	22.5	6.2
Fibreglass	20.4	5.7
Single poly	23.5	6.5
Double poly	14.3	4.0

The heat loss factor is also influenced by wind speed. The above values are based upon a wind speed of 15 miles per hour (mph) (24 km/h). If other wind speeds are expected to occur at the design outside condition, then allowances should be made for this by adjusting the HLF according to Table 3.3.

TABLE 3.3: U-values at various wind velocities (kJ/m² hr °C)

	Wind velocity (km/hr - m/s)					
Material	**0**	**8 - 2.24**	**16 - 4.47**	**32 - 8.94**	**40 - 11.2**	**48 - 13.4**
Glass	15.6	19.4	21.2	23.3	23.7	24.1
Fibreglass	14.2	17.7	19.4	21.1	21.6	22.0
Single poly	16.5	20.4	22.3	24.3	24.7	25.1
Double poly	10.9	12.9	13.8	14.6	14.9	15.0

As mentioned previously, total heat loss is a function of two components: (1) transmission heat loss, and (2) infiltration. For greenhouse design, infiltration is generally analyzed via the "air change" method. This method is based upon the number of times per hour that the air in the greenhouse is replaced by cold air leaking in from outside. The number of air changes which occurs is a function of wind speed, greenhouse construction, and inside and outside temperatures. Table 3.4 outlines general values for different types of greenhouse construction.

TABLE 3.4: Air change data for various glazing materials

Material	Air Changes/hr
Single glass	2.5-3.5
Double glass	1.0-1.5
Fiberglass	2.0-3.0
Single poly	0.5-1.0
Double poly	0.0-0.5
Single poly w/low fiberglass sides	1.0-1.5
Double poly w/low fiberglass sides	0.5-1.0
Single poly w/high fiberglass sides	1.5-2.0
Double poly w/high fiberglass sides	1.0-1.5

As the number of air changes is related to the volume of the greenhouse, after selecting the appropriate figure from above, it is necessary to calculate the volume of the structure. Calculations do not include ventilation.

$$Heat\ loss\ due\ to\ infiltration = AC/H \times volume \times DTD \times 0.018 \quad (Btu/hr)$$

where volume is in ft^2 and DTD is in °F, or

$$= AC/H \times volume \times DTD \times 0.102 \quad (kW)$$

where volume is in m^2 and DTD is in °C (for kcal/ hr the multiplication coefficient is 0.088).

$$Total\ greenhouse\ heating\ requirement = transmission\ loss + infiltration\ loss$$

Note: The above is usually converted to energy loss / unit of floor area, as most greenhouses operators understand this relationship better.

This is the "peak" or design heating load for the greenhouse. The heating equipment selected for the structure would have to be capable of meeting this requirement. Formulas are provided at the end of this article.

3.4.3 Annual heating load

In many cases, the quantity of energy required to heat a greenhouse over an entire year is of interest. For a conventionally heated greenhouse, this figure would allow one to calculate annual fuel costs. In a geothermal application, the annual energy requirement would be related to pumping costs or heating expenses if energy is to be purchased from an outside entity. The annual energy determination is a complex one involving many repetitious heat loss and solar heat gain calculations. Actual heating requirements are influenced by the local climate, solar weather, set point temperatures and greenhouse construction.

3.4.4 Heat exchangers

In most geothermal applications, a heat exchanger is required to separate actual heating equipment from the geothermal fluid. This is because of the scaling and corrosion associated with most geothermal fluids. Generally, the heat exchanger is placed between two circulating loops, the geothermal loop and the clean loop, as shown in Figure 3.6.

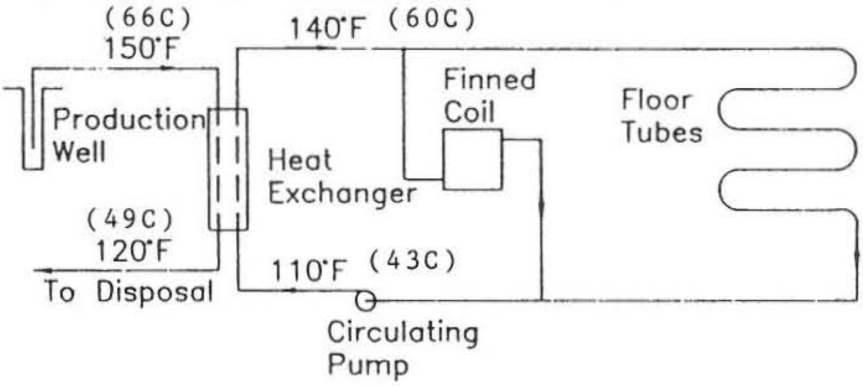

FIGURE 3.6: Heat exchanger schematic

As a result of this heat exchanger, there is some loss in the temperature available for use in the actual heating equipment. The temperature loss depends upon the type of heat exchanger used. For plate-type heat exchangers, a temperature of 5-10°F (3-6°C) should be applied, for shell-and-tube heat exchangers 15-20°F (8-11°C), and for homemade configurations 20-40°F (11-22°C). For example, assuming a geothermal resource temperature of 150°F (66°C) is available, use of a plate-heat exchanger could result in 140°F (60°C) supply water, as shown in Figure 3.6.

Now that the heating requirement and supply water temperature have been established, various heating systems can be evaluated with respect to their ability to meet this demand. For geothermal applications, the available geothermal resource temperature has a large impact upon the system chosen. This is a result of the fact that certain types of heating methods yield better results with low-temperature fluid than others.

3.4.5 Greenhouse heating systems

There are basically eight different geothermal heating systems which are applied to greenhouses:

1. Finned pipe,
2. Unit heaters,
3. Fan coil units,
4. Soil heating,
5. Cascading,
6. Plastic tubing,
7. Bare pipe, and
8. Combination of the above.

Often the choice of heating system type is not dictated by engineering considerations such as maximum use of the available geothermal resource or even the most economical system, but on grower preference. Grower preference may be based strictly on past experience and familiarity with growing crops with that system. It may also be influenced by factors such as the type of crop, or potential disease problems. Some crops, such as roses and mums, require closely controlled humidity and a considerable amount of air circulation to prevent leaf mildew. If a radiant floor system is used, auxiliary circulating fans will be required. Tropical and subtropical potted plants, on the other hand, may require high humidity and higher soil temperatures. In this case, a radiant, under the bench system will be preferred, perhaps combined with an overhead air system for snow melting, and to get maximum sunlight during winter months in areas of high snow fall. Certain flowering plants may require shading to control blooming, thereby enabling the grower to market at the most opportune time. The type and location of the shading cover can affect the placement of heating and air handling equipment and, perhaps, the type of heating.

All these things should be taken into consideration and the heating system designer should maintain close communication with the grower in the selection of type and the placement of heating devices. The following paragraphs outline the performance of the heating system mentioned above.

Finned pipe

As the name implies, finned pipe is usually constructed of steel or copper pipe with steel or aluminum fins attached to the outside. These fins can either be circular, square or rectangular in shape. In the size range employed in greenhouses, the steel pipe with steel fins is most common.

Since most finned-type heating equipment used in geothermal projects was originally designed for standard hot water use, heating capacity is generally based upon 200°F (93°C) or higher average water temperature and 65°F (18°C) entering air temperature. If the available supply temperature from the geothermal system is less than the 200°F (93°C) value, the capacity of the heating equipment, in this case finned pipes, will be less than the rated value. In addition, heating capacity of finned pipe, usually expressed in Btu/hr per lineal foot (W/m), is influenced by fin size, pipe size and flow velocity. Table 3.5 shows one manufacturer's rating for equipment and Table 3.6 shows the appropriate de-rating factors to be applied for average water temperatures of <200°F (93°C).

TABLE 3.5: Hot water ratings of heating elements
(average water temp. 200°F [93°C]) - Base Case

Base heating element	Rows	kJ/hr/m	W/m
108 fins/m - 33 fins/ft	1	3873	1076
	2	6709	1864
	3	8714	2421
131 fins/m - 40 fins/ft	1	4219	1172
	2	6950	1932

TABLE 3.6: De-rating factors for heating elements (<93°C)

Average water (°C)	Factor
88	0.90
82	0.80
71	0.62
60	0.47
49	0.30
38	0.17

It is important to note that the capacity of this equipment is indexed to <u>average</u> water temperature, not supply water temperature.

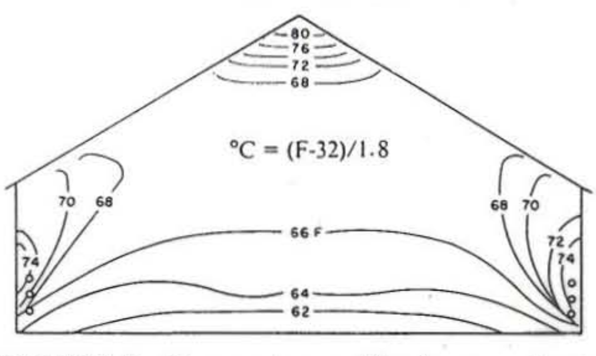

FIGURE 3.7: Temperature profiles in a greenhouse heated with radiation piping along the side walls

Finned elements are generally installed along the long dimension of the greenhouse adjacent to the outside wall. Improved heat distribution is achieved if about one-third of the total required length is installed in an evenly spaced pattern across the greenhouse floor (ASHRAE, 1978) (Figure 3.7). This system has the disadvantage of using precious floor space that would otherwise be available for plants. In addition, it is less capable of dealing effectively with ventilation if it is required. Maintenance requirements are low, particularly if a heat exchanger is used. In addition, the natural convection nature of the finned pipe system does not increase electrical costs as a result of the fan operation.

Standard unit heaters
Unit heaters consist of a finned coil and small propeller fan contained in a predesigned unit. These units are available in either horizontal or vertical configurations and are generally hung from the greenhouse structure at roof level. Air is discharged either directly into the greenhouse or into a perforated plastic distribution tube (see Figures 3.8 and 3.9).

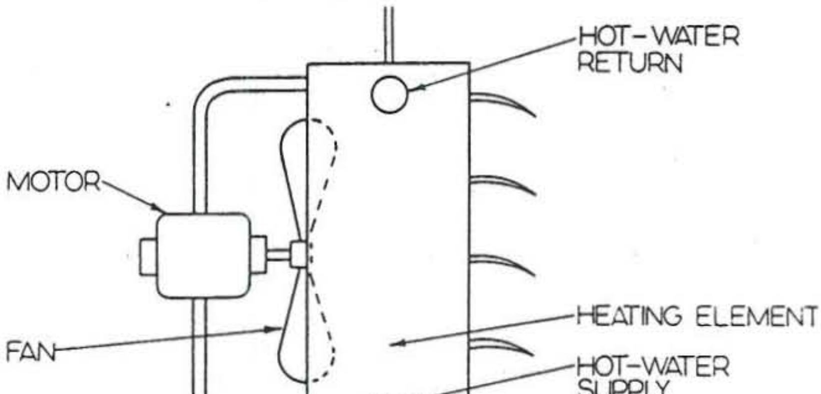

FIGURE 3.8: Horizontal hot water unit heater

As with the finned pipe equipment, unit heaters are generally rated at 200°F (93°C) entering water temperature (EWT) and 60°F (16°C) entering air temperature (EAT). Changes in either of these two parameters will affect unit capacity (usually expressed in Btu/hr or w). Since most geothermal resources applied to greenhouses are <200°F (93°C), some adjustment of unit capacity is necessary. Table 3.7 shows a typical set of manufacturer's performance data for unit heaters at standard conditions (200°F EWT/60°F EAT [93°C/15.6°C]). To adjust for other conditions, Table 3.8 values are employed. It is important that the gpm (L/s) values shown in Table 3.7 and met. Providing a unit with a flow less

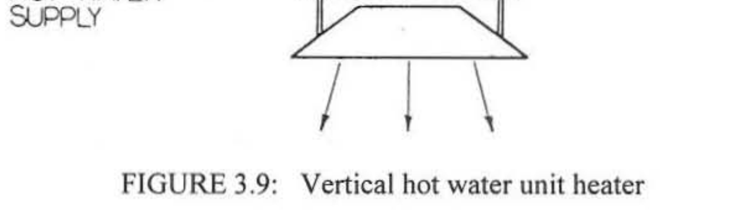

FIGURE 3.9: Vertical hot water unit heater

than that shown will decrease capacity.

Because these units are generally constructed with copper tubes, even very small concentrations of dissolved hydrogen sulphide (H_2S) or ammonia (NH_3) will result in rapid failure. In addition, the long path through which the water must flow in the unit heater can result in scaling if the fluid has this tendency. As a result, a unit heater system should not be applied without an isolation heat exchanger.

TABLE 3.7: Hot water unit heater ratings (Modine)
(93°C EWT and 15.6°C EAT)

Model	kJ/hr	L/s	m³/min (air)	Final air temperature (°C)	hp - W
A	95,000	0.57	50	43	1/6 - 125
B	140,000	0.84	92	38	1/3 - 250
C	146,000	0.88	82	42	1/3 - 250
D	209,000	1.26	129	39	1/2 - 375
E	236,000	1.39	130	42	1/2 - 375
F	288,000	1.70	145	42	1/2 - 375

TABLE 3.8: Unit heater correction factors (Modine)
(EWT = entering water temperature, EAT = entering air temperature)

EWT (°C)	EAT (°C)			
	4.4	15.6	26.7	37.8
27	0.293	0.143	0	0
38	0.439	0.286	0.140	0.069
49	0.585	0.429	0.279	0.137
60	0.731	0.571	0.419	0.273
71	0.878	0.714	0.559	0.410
82	1.024	0.857	0.699	0.547
93	1.170	1.000	0.833	0.684

Fan coil units

These units are very similar to the standard unit heater discussed previously. They consist of a finned coil and a centrifugal blower in a single cabinet. A few manufacturers offer units in an off-the-shelf line for low-temperature greenhouse heating. It is much more common that they are custom selected. The difference between the fan coil unit and the hot-water unit heater is primarily in the coil itself. In the fan coil system, the coil is much thicker and usually has closer fin spacing than the coil in a unit heater. Unit heaters generally have only a one or two row coil. A custom designed coil can have as many as six or eight rows. The additional rows of tubes create more surface area. The added surface area allows for more effective heat transfer, resulting in the ability to extract more heat from the water.

Soil heating

This system generally involves using the floor of the greenhouse as a large radiator. Tubes, through which warm water is circulated, are buried in the floor of the greenhouse. Heat from warm water is transferred through the tube to the soil and, eventually, to the air in the greenhouse.

In the past, tube materials were generally copper or steel. Because of corrosion and expansion problems with these materials, non-metallic materials have seen increasing application in recent years. The most popular of these is polybutylene. This material is able to withstand relatively high temperature (up to ~180°F [82°C]) and is available in roll form for easy installation. PVC piping is only available in rigid form and is limited with respect to temperature. Polyethylene and similar materials are available in flexible roll form, but are (as PVC) generally limited in terms of temperature handling ability.

A soil heating system is preferred by many operators because it results in very even temperature distribution from floor to ceiling, and does not obstruct floor space or cause shadows. However, its ability to supply 100% of the heating requirements of a greenhouse necessitates a rather mild climate and a low-inside design temperature. This is caused by the nature of heat transfer in the system. As heating requirements are increased, the required heat output from the floor is increased. In order to produce more heat, the floor surface temperature must be increased. Very quickly a point is reached at which it is difficult to spend extended periods on such a hot floor. In addition, if plant are grown on or near the floor (including benches), heat transfer to the plants may be excessive with a radiant floor system. This problem is illustrated in Figure 3.10. As a result, this system is generally employed in conjunction with another system such as unit heaters. The floor system supplies the base load for the greenhouse and the secondary system is used for occasional peaking purposes.

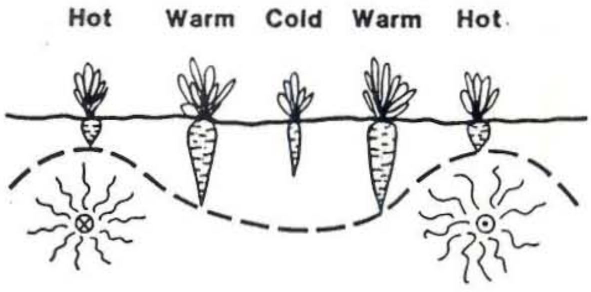

FIGURE 3.10: Idealized effect of temperature on carrots

The procedure for designing a floor system consists of:

1. Determining the heat load for the greenhouse,
2. Calculating the required floor temperature to meet the load, and
3. Calculating the required size, depth and spacing of the tubes.

The next step is to determine the depth and spacing of the tubes supplying the heat. Tube spacing and size is dependent upon the available water temperature. Generally, depth is more a function of protecting the tubes from surface activity than system design, and a figure of 2 to 6 in. (5 to 15 cm) below the surface is common.

Since it is the purpose of the floor panel system to use the floor as a large radiator, it follows that the installation of the tubing should result in an uniform a floor surface temperature as possible. This is accomplished by two general approaches: (a) placing smaller diameter tubes at close spacing near the surface of the floor, or (b) placing larger tubes spaced further apart at a greater burial depth. The theory behind this approach is to reduce the difference between the distance heat must travel vertically (from the tube to the surface directly above it) and laterally (from each tube to the surface between the tubes) (Adlam, 1947).

The depth at which the tubes are to be buried is often a function of protecting them from surface activity. For burial in the soil floor of a greenhouse, a depth of at least 2 to 3 in. (5-8 cm) should be employed. If crops are to be grown directly in the soil, depth requirements are such that this type of system becomes impractical.

Tubing size is a function of heating requirements. Common sizes are 1/2 in., 3/4 in., and 1 in. (12, 19 and 25 mm) with the smaller sizes used generally in the 2 to 4 in. (5-10 cm) depth and the larger lines for depths of 5 in. (13 cm), and greater.

The final determination of the size and spacing is a function of heat output required, mean-water temperature, soil conductivity, and burial depth.

The required heat loss is fixed by the type of greenhouse construction used. Soil conductivity is also fixed by site characteristics. As mentioned earlier, the minimum burial depth is fixed by surface activity. As a result, the choice of size and spacing is balanced against mean-water temperature, the single parameter over which the designer has some control. Table 3.9 lists some maximum mean-water temperatures above these values will result in floor surface temperatures >90°F (32°C). If workers are to spend extended periods in the greenhouse, floor surface temperatures above this value would be unacceptable.

TABLE 3.9: Maximum recommended mean water temperature (°C)

Burial depth (mm)	Steel pipe		Polybutylene tupe	
	k = 6	k = 9	k = 6	k = 9 *
25	44	41	51	44
50	47	43	55	49
75	50	46	59	53
100	52	47	62	55
125	53	49	64	57
150	57	52	69	61

* k = 6 Btu in/hr ft2 °F = 0.864 W/m °C
k = 9 Btu in/hr ft2 °F = 1.30 W/m °C

In addition to the maximum mean-water temperature, it is also important when making this calculation to be aware of system ΔT (supply temperature minus return water temperature) and its impact upon system design. Temperatures drops above ~15°F (8°C) should employ a double serpentine to balance the circuit output. For ΔT below 15°F (8°C), a single serpentine can be used as shown in Figure 3.11. The heat output per lineal foot (m) of tube can be determined from Figure 3.12.

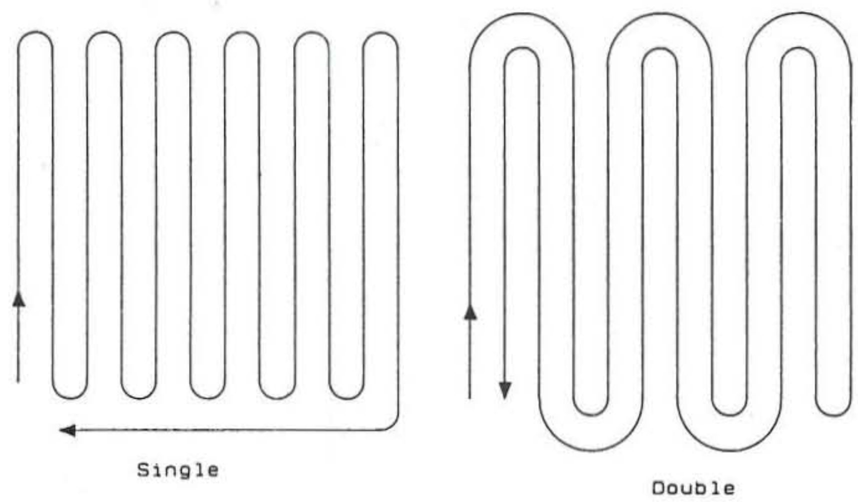

FIGURE 3.11: Single and double serpentine piping layout

It is suggested that a heat exchanger be used for buried systems. This is for two reasons: protection from scaling and control of

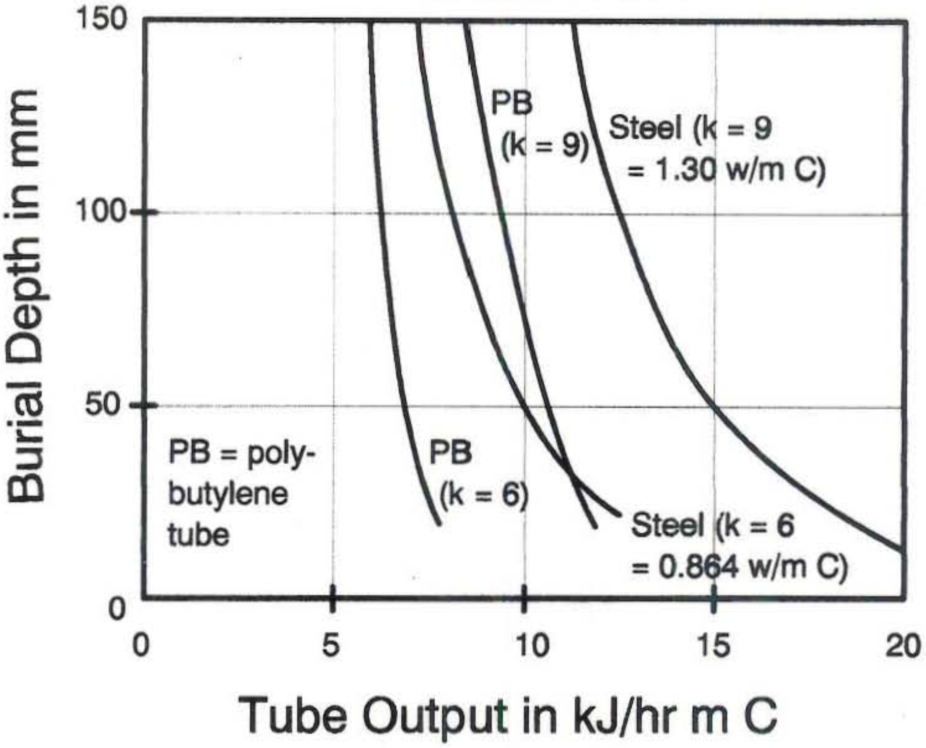

FIGURE 3.12: Heat output for radiant floor system

temperature. Control of temperature is the most critical. The only method of controlling the output of floor system is by controlling the water temperature in the tubes. The use of a heat exchanger allows this control to be carried out more easily. The flow of geothermal fluid to the exchanger is regulated to maintain a given supply temperature to the heating loop as shown earlier in Figure 3.5.

A great deal of piping material is required to supply just 60% of the peak requirement of a greenhouse in a cold location. In addition, the inability to grown directly in or on the soil surface also restricts the wide acceptance of this type of system.

An example of soil heating in Iceland is found in Gudmundsson (1983), where these systems are frequently used outside with or without a plastic cover.

Cascading

This method, which was developed by the Soviets for waste heat applications, involves distributing water over the outside of the greenhouse in a thin "sheet" of flow. Although this is a very effective method of heating a greenhouse, there are some disadvantages that would limit its use in geothermal applications.

Distributing large quantities of warm water over a surface exposed to the atmosphere results in substantial energy losses. These losses exceed by many times the requirements of the greenhouse. As a result of the large heat losses from the cascaded fluid, a great deal of evaporation takes place. Because of the many chemical species contained in geothermal fluids, evaporation would tend to cause concentration and subsequent deposits of these constituents on greenhouse surface.

Because of these disadvantages, it is unlikely that such a system would be applied to any great extent in the U.S. Therefore, it will not be discussed here.

Bare tube system

This system involves the use of bare tubing, usually small diameter polybutylene or similar material. The tubing is installed either on the floor or suspended under benches. It is preferable for the tubing to be located low in the greenhouse; although, a portion may be located overhead. Regardless of the installation location, it is very important that the tubing be arranged such that each tube is separated from the others. If the tubes are bunched together, the effective surface area of each is reduced, thus lowering heating capacity.

In colder regions, this system encounters the same problem as the floor panel system in that large quantities of tubing are required to meet the design requirement. Control of the system in many cases has been manual by way of gate valves. However, as with the floor panel system, the use of a heat exchanger can allow accurate control of temperature and, hence, output. Design of a system is based upon the average water temperature of the heating loop.

Bare pipe system

In many locations such as Iceland, China, Hungary, Russia and Italy, bare steel pipes are commonly used, placed along the outer walls of the greenhouse. These are typically 2 in (5 cm) in diameter and consist of upto six or eight runs hung on each side (and even the ends) with U-bend reversals in a continuous loop. Geothermal water and even two phase flow is circulated directly through these pipes with the heat being transferred into the space by natural convection. "A rule of thumb" used in Icelandic greenhouse operations is to use "2 m of pipe for every 2 m^2 of floor space" (Sverrir Thorhallsson, pers. comm., 1996). Small shell-and-tube heat exchangers are connected to the system to provide warm water for cleaning and plant watering (cold water may cause thermal root shock). This appears to be a very successful heating system and has an advantage over finned tube systems where the fins may clog with dirt and plant material. Appendix 3.1B provides the necessary formulas to use for this design method.

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APPENDIX 3.1: GREENHOUSE DESIGN FORMULAS

A. Soil heating

$$\frac{q}{A} = 1.70\left(\left(\frac{1.8\,T_p + 492}{100}\right)^4 - \left(\frac{1.8AUST + 492}{100}\right)^4\right) + 7.87\,(T_p - T_a)^{1.32}$$

where q/A = heat/area in kJ/hr•m^2,
T_p = floor surface temperature in °C,
T_a = indoor air temperature in °C,
AUST = average temperature of unheated surfaces in the greenhouse (walls and roof) in °C,

and,

$$IST = IDT - (0.0291 \times U \times \Delta T)$$

where IST = inside surface temperature in °C,
IDT = inside design temperature in °C,
U = glazing material heat loss factor in kJ/m^2•hr•°C,
ΔT = design temperature difference (inside - outside) in °C,

and,

$$AUST = \frac{A_1 \times IST_1 + A_2 \times IST_2 .... A_n \times IST_n}{A_1 + A_2 .... A_n}$$

where A = surface area of glazing material in m^2,
IST = inside surface temperature of material in °C.

B. Bare pipe system

$$\frac{q}{l} = \left([4.422 \times (\frac{1}{D})^{0.2} \times (\frac{1}{1.8T_{ave} + 32})^{0.181} \times (\Delta T)^{1.266} + (15.7\,10^{-10})[(1.8T_1 + 32)^4 - (1.8T_2 + 32)^4] \right) 11.345A$$

where q/l = Heat output of tubing per length in kJ/hr•m,
D = Outside diameter of tubing in mm,
T_{ave} = 255.6 + (AWT + T_{air})/2 in °C,
AWT = T_s - ΔT/2 in °C,
T_s = Greenhouse water supply temperature in °C,
T_{air} = Greenhouse design air temperature in °C,
ΔT = AWT - (T_{air} + 3) in °C,
T_1 = 255.6 + AWT in °C,
T_3 = (AUST + T_{air})/2 in °C,
T_2 = 255.6 + T_3 in °C,
A = Outside surface area of pipe/unit length in m_2/m.

Total length of pipe required for the greenhouse is

$$l = q/(q/l)$$

where q = heat load of greenhouse in kJ/hr,
q/l as determined above in kJ/hr•m.

LECTURE 4

AQUACULTURE

4.1 INTRODUCTION

Aquaculture involves the raising of freshwater or marine organisms in a controlled environment to enhance production rates. The principal species that are typically raised are aquatic animals such as carp, catfish, bass, tilapia, frogs, mullet, eels, salmon, sturgeon, shrimp, lobster, crayfish, crabs, oysters, clams, scallops, mussels, and abalone.

The use of geothermal energy for aquaculture rather than water dependent upon the sun for its heat has demonstrated that more fish can be produced in a shorter period of time. When the water temperature is below the optimal range, the fish loses its ability to feed because the basic body metabolism is affected (Johnson, 1981). Thus, a good geothermal supply, due to its constant temperature, can "out perform" a natural mild climate.

Ambient temperature is generally more important for aquatic species than land animals. This suggest that the potential use of geothermal energy for aquaculture may be greater than for animal husbandry, such as pig and chicken rearing. Figure 4.1 shows the growth trends for a few land and aquatic species (Barbier and Fanelli, 1977). Land animals grow best in a wide temperature range, from just under 50°F (10°C) and up to about 68°F (20°C). Aquatic species such as shrimp and catfish have a narrower range of optimum production at a higher temperature, approaching 86°F (30°C). Trout and salmon, however, have a lower optimum temperature no higher than 59°F (15°C).

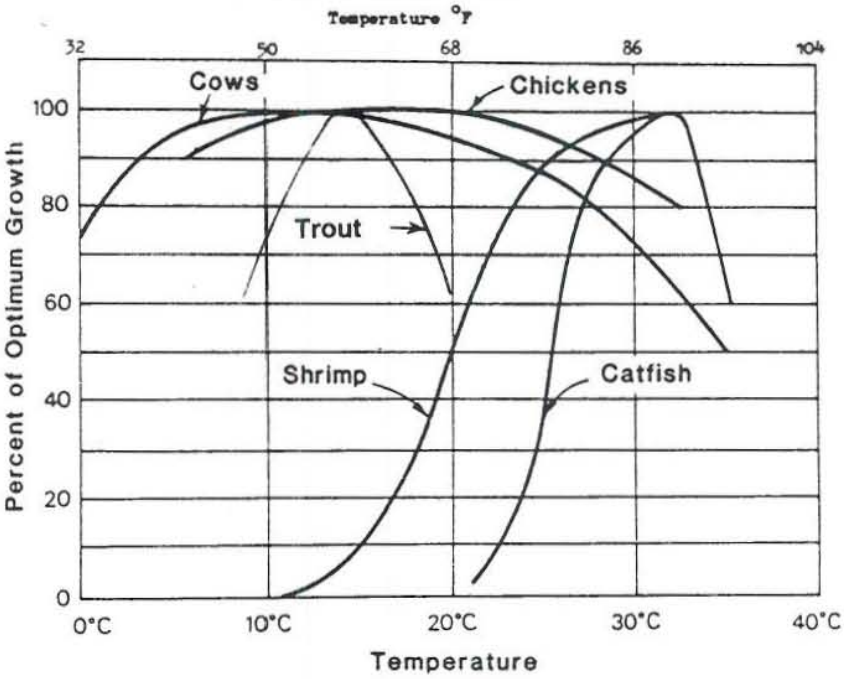

FIGURE 4.1: Optimum growing temperatures for selected animal and aquatic species

4.2 EXAMPLES OF GEOTHERMAL PROJECTS

Fish breeding is a successful business in Japan where carp and eels are bred and raised. The eels are the most profitable and are raised in 10 in. (25 cm) diameter by 3 ft (0.9 m) long earthenware pipes. Water in the pipes is held at 73°F (23°C) by mixing hot spring water with river water. The adult eels weigh from 3.5 to 5 oz. (100 to 150 grams), with a total annual production of 8,400 lbs (3,800 kg). Alligators and crocodiles are also raised in geothermal water. These reptiles are being bred purely for sightseeing purposes. In combination with greenhouses offering tropical flora, alligator farms are offering increasingly large inducements to the local growth of the tourist industry (Japan G.E.A., 1974). Icelandic fish hatcheries raise 610,000 salmon and trout fingerlings annually in geothermal water. A total of 10 fish hatcheries existed around the country - a new and fast-growing industry (Hansen, 1981).

In the U.S., aquaculture projects using geothermal water exist in Idaho, Oregon, and California. Fish Breeders of Idaho, Inc., located near Buhl, has been raising channel catfish in high-density concrete raceways for over 15 years. The water is supplied by artesian geothermal wells flowing at 6,000 gpm (380 L/s) at 90°F (32 °C). Cold water from springs and streams is used to cool the hot water to 80-85°F (27-29°C) for the best production temperature. Normal stocking densities are from 50 to 10 pounds of fish per cubic foot of space (80 to 160 kg/m^3). The maximum recommended inventory for commercial production is about 10,000 to 15,000 pounds per second foot of water (1.6 to 2.4 x 10^5 kg/m^3/s). Yearly production will usually be three to four times the carrying capacity. Oxygen and ammonia are the principal factors limiting production (Ray, 1979).

Giant freshwater prawns (Macro brachium rosenbergii) were raised at Oregon Institute of Technology from 1975 to 1988. Some work has also been done in trout culture and mosquito fish (Gambuzia affinis). This work has provided data demonstrating that a tropical crustacean can be grown in a cold climate (as low as -20°F [-7°C]) where the water temperature is maintained at the optimal growing temperature for this species of 81-86°F (27-30°C). Initially, two smaller outdoor ponds 4 ft (1.2 m) deep were used, and then two half-acre (0.2 ha) were built. A selected brood stock was held in a small spawning building where larvae were hatched in artificial saltwater and reared to the post-larva stage which made the facility self-supporting. Growth rates of 7/8 in. (2 cm) per month were maintained (twice that obtained in tropical climates) with a 1 ft^2 (900 cm^2) of surface area per animal maximum density. The plumbing system of the ponds consisted of perforated diffuser pipes, control valves and thermostats to maintain an optimum temperature of the pond. This provided an even distribution of geothermal energy throughout the pond (Johnson, 1978 and 1981; Smith, 1981).

A very successful catfish raising operation has been started by the Indian community at Fort Bidwell in northeastern California. Geothermal well water at 105°F (40°C) is mixed with cold water to produce 80°F (27°C) water which is then piped into 25 ft long by 8 ft wide by 4 ft deep (7.6 m x 2.4 m x 1.2 m) raceways. Two sets of parallel raceways use 900-1,000 gpm (57 to 63 L/s). A 1 ft (0.3 m) drop between raceways is used to aerate the water. One ounce (28 g) fish at 3,000 per raceway are initially stocked, producing a surviving 2,000 fish at 2 pounds (0.9 kg) each in five months. Construction of the raceways and well cost $100,000. The fish are sold live at the source for $1.40 per pound ($3.09/kg) and delivered live to San Francisco where they wholesale for $2 per pound ($4.41/kg) and retail for $3 to $4 per pound ($6.60 to $8.80 per kg). Production cost at Fort Bidwell is approximately $0.60 per pound ($1.32/kg) (personal communication with William Johnson).

An example of a tropical fish raising operation is near Klamath Falls, Oregon (Lund, 1994). Effluent water from a greenhouse operation is used to heat 37 shallow tropical fish ponds. These ponds are 100 ft (30 m) long and 13 ft (4 m) wide, and vary from 3 to 4.5 ft (1.0 to 1.4 m) deep. They are kept at a constant 74°F (23°C) temperature. At present, the owner raises 85 varieties of cichlid fish for pet stores in San Francisco and Portland. Approximately 1,000 fish, 3 to 4 in. (7.5 to 10 cm) long, are shipped each week from the local airport. The geothermal heat is a real advantage, as the greatest demand for the fish is during the winter months.

A summary of the leading concentrations of fish farms in the United States is given in Table 4.1:

TABLE 4.1: The leading concentration of fish farms in the United States

Location	Annual use TJ - MW_t	Load factor	Product
Buhl, ID	310.2 - 12	80%	Catfish
Mecca, CA	75.0 - 10	25%	Prawns
Wabushka, NV	13.8 - 2	25%	Catfish/Tropicals
Ft. Bidwell, CA	12.1 - 1	80%	Catfish
Paso Robles, CA	11.8 - 1	50%	Catfish

One of the largest and successful freshwater prawn farms was established in New Zealand in 1987 to take advantage of geothermal waste heat from the Wairakei power generating field on North Island. At present, they have 19 ponds which vary in size from 0.5 to 0.9 acres (0.2-0.35 ha) and have depth of 3.3 to 3.9 ft (1.0-1.2 m). The ponds are kept at a temperature of 75°F (24°C) -heat supply limitations keeps the temperature from the ideal of 82°F (28°C) -with a variation from one end of the pond to the other of 2°F (1°C). Presently, the farm is capable of producing up to 33 tons (30 tonnes) of prawns per year. The adult prawns are harvested after nine months, averaging 14 to 18 per pound (30 to 40 per kg) and sold at US$8/lb wholesale, and US$12/lb retail (US$17/kg and US$27/kg). Ninety percent of the harvested prawns are sold to a restaurant on the property, which caters to about 25,000 tourists each year. In the near future, another 99 acres (40 ha) will be added on the other side of the Wairakei power plant using waste cooling water from a proposed binary power generator. This would create a unit capable of becoming the third largest freshwater prawn producer in the world, producing 440 tons (400 tonnes) a year, which would mean an income of more than US$6.7 million annually (Lund and Klein, 1995).

4.3 GENERAL DESIGN CONSIDERATIONS

Aquaculture ponds are best constructed with 1/4 acre (0.1 ha) of surface area. A size of 50 ft by 200 ft (15 by 61 m) is ideal for harvesting. A minimum-sized commercial operation should have 7 to 10 acres (3 to 4 ha) under development (water surface area), or about 30 to 40 ponds. The maximum surface area that should be considered for a single pond is one-half an acre (0.2 ha). Figure 4. 2 illustrates OIT's geothermal pond design.

The most important items to consider are quality of the water and disease. If geothermal water is to be used directly, evaluation of heavy metals such as fluorides, chlorides, etc., must be undertaken to determine if the fish or prawns can survive. A small test program is often a wise first step. An aeration pond preceding the stocked ponds will often solve the chemical problem.
Crops that are a good candidate for aquaculture are listed in Table 4.2.

TABLE 4.2: Crops that are a good candidate for aquaculture

Specie	Growth period (months)	Water temperature (°F / °C)
Tropical fish	2-3	74-90 / 23-27
Catfish	4-6	80-85 /27-29
Trout	4-6	55-65 / 13-18
Prawns	6-9	81-86 / 27-30

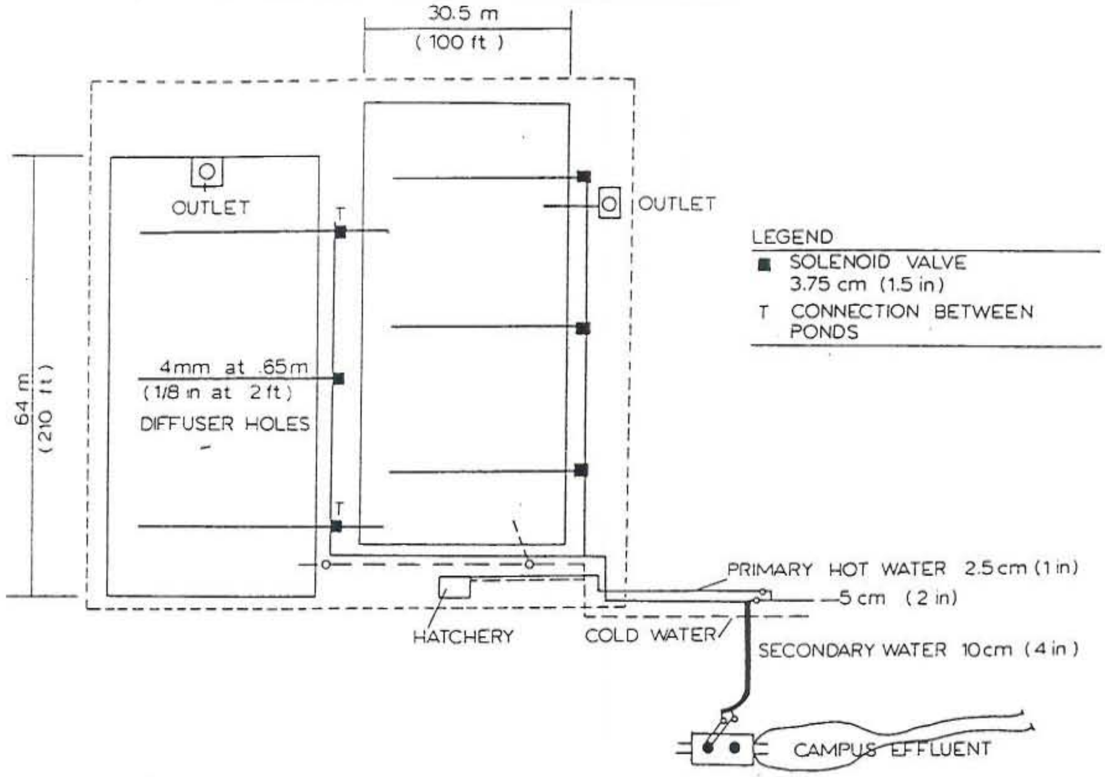

FIGURE 4.2: OIT's geothermal aquaculture research project

A more detailed list of species and temperature requirements is found in Table 4.3.

TABLE 4.3: Temperature requirements and growth periods for selected aquaculture species

Species	**Tolerable extremes**		**Optimum growth**		**Growth period to market size**
	(°F)	**(°C)**	**(°F)**	**(°C)**	**(Months)**
Oysters	31-97	0-36	76-78	24-26	24
Lobsters	32-88	0-31	72-75	22-24	24
Penaeid shrimp					
Kuruma	40-?	4-?	77-87	25-31	6-8
Pink	52-104	11-40	75-85	22-29	6-8
Salmon (Pacific)	40-77	4-25	59	15	6-12
Freshwater prawns	75-90	24-32	83-87	28-31	6-12
Catfish	35-95	17-35	82-87	28-31	6
Eels	32-97	0-36	73-86	23-30	12-24
Tilapia	47-106	8-41	72-86	22-30	-
Carp	40-100	4-38	68-90	20-32	-
Trout	32-89	0-32	63	17	6-8
Yellow perch	32-86	0-30	72-82	22-28	10
Striped bass	?-86	?-30	61-66	16-19	6-8

Tropical fish (goldfish) are generally the easiest to raise and have a low investment and high yield. Smaller ponds can also be used. An average of 150,000 fish per year can be raised from one acre (0.4 ha), requiring the lowest temperature water; thus, they can better use low-temperature resources of cascaded water. Freshwater prawns generally have a high market value, with marketable sizes being 16 to 20 tails to the pound (35 to 44 per kg). Channel catfish are also popular, especially as fillets. Production rates depend upon water quality and flow rates.

Ponds require geothermal water of 100-150°F (38-66°C) and a peak flow of 300 gpm (19 L/s) for one acre (0.4 ha) of uncovered surface area in colder climates. The long axis of the pond should be constructed perpendicular to prevailing winds to minimize wave action and temperature loss. The ponds are normally constructed of excavated earth and lined with clay or plastic where necessary to prevent seepage loss. Temperature loss can be reduced, thus reducing the required geothermal flow, by covering the pond with a plastic bubble. Construction cost, exclusive of geothermal wells and pipelines, will run $30,000-50,000 per acre ($75,000-125,000 per hectare).

4.4 SPECIFIC DESIGN CONSIDERATIONS (Rafferty, 1991)

A non-covered body of water, exposed to the elements, exchanges heat with the atmosphere via four mechanisms: evaporation, convection, radiation and conduction. Each of these is influenced by different parameters and will be discussed separately in the paragraphs below.

4.4.1 Evaporative loss

Evaporation is generally the largest component of the total heat loss from the pond. When water is evaporated from the surface of the pond, the heat is taken from the remaining water. As a result, as each pound of water evaporates from the surface, 1,000 Btu (478 kJ/kg) is lost with the escaping vapour. Losses can occur by evaporation even when the water temperature is at or below the surrounding air temperature. This is because water evaporates from the surface of the pond at the wet bulb temperature. At 100% relative humidity, the wet bulb temperature is the same as the dry bulb temperature (dry bulb is the temperature which is given by a standard thermometer). At anything less than 100% relative humidity, the wet bulb temperature is less than the dry bulb temperature and, as a result, evaporation loss can occur below the air temperature.

The rate at which evaporation occurs is a function of air velocity and the pressure difference between the pond water and the water vapour in the air (vapour pressure difference). In simple terms, as the temperature of the pond water is increased or the relative humidity of the air is decreased, evaporation rate increases. The equation which describes the rate of evaporation (of a 150 ft [50 m] long pond) is shown below (Eckert, 1959).

$$Wp = \frac{2930\,v}{1.8\,Ts + 492}\,(Pw - Pa)\,A$$

where

- Wp = Rate of evaporation (kg/hr)
- A = Pond surface area (m^2)
- v = Air velocity (m/s)
- Pw = Saturation vapour pressure of the pond water (bars) (1.00 bar = 1.02 kg/cm^2)
- Pa = Saturation pressure air dew point (bars)
  = saturation pressure at air temperature x relative humidity
- Ts = Surface temperature (°C)

For enclosed ponds or indoor swimming pools, this equation can be reduced to (ASHRAE, 1978):

$$Wp = 14.46 \times A \times (Pw - Pa)$$

where

- Wp = Rate of evaporation (kg/hr)
- A = Pond area (m^2)
- Pw = Saturation pressure of the pond water (bar)
- Pa = Saturation pressure of air dew point (bar)

Following are some common values for v, Pw and Pa:

For v: @ 8 km/hr, 2.23 m/s
@ 16 km/hr, 4.47 m/s
@24 km/hr, 6.70 m/s

For Pw: @ 15.6°C, 0.0176 bar
@ 21.1°C, 0.0250 bar
@ 26.7°C, 0.0349 bar
@ 32.2°C, 0.0481 bar

For Pa: For outdoor locations with a design dry bulb air temperature of below 0°C, Pa can be taken as 0.0051 bar.
For indoor locations with a design of approximately 24°C and 50% relative humidity, Pa can be taken as 0.0145 bar.

To obtain the heat loss in kJ/hr simply multiply the kg/hr loss by the specific enthalpy of the pond water at temperature Ts:

$$q_{ev} = Wp \times h_{fg} \quad (kJ/hr)$$

where

- h_{fg} = specific enthalpy of the pond water at temperature Ts (kJ/kg)

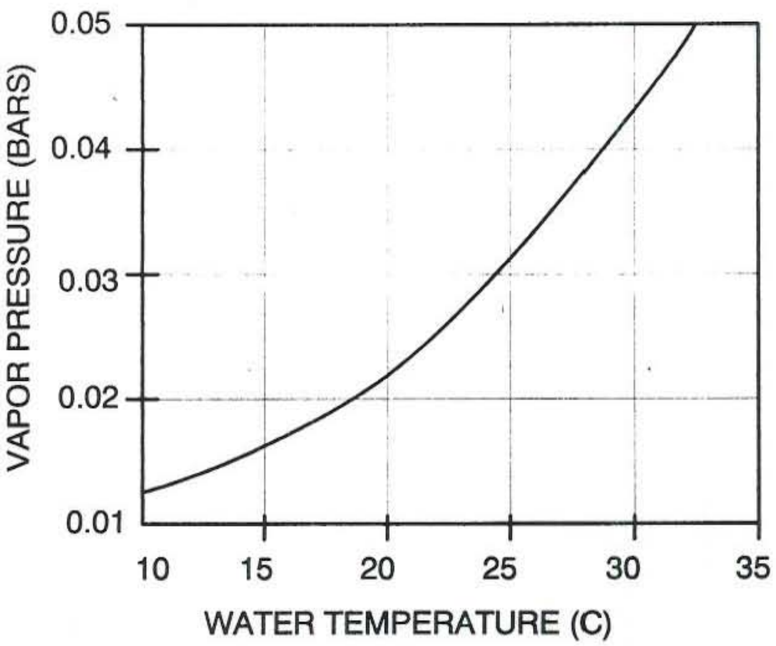

FIGURE 4. 3: Plot of pond water vapour pressure vs. temperature

This is the peak or design heat loss. It is important to note that the example values given above are for the design (worst) case. At higher outdoor air temperatures and different relative humidities, this value would be less. As mentioned earlier, the rate of evaporation loss is influenced by the vapour pressure difference between the pond water and the water vapour in the air. Figure 4.3 illustrates the effect of increased pond water temperature on vapour pressure (Pw). It is obvious that reduced water temperature would reduce the vapour pressure difference and hence, the rate of evaporation.

4.4.2 Convective loss

The next major mechanisms of loss from the pond surface is that of convection. This is the mode associated with the heat losses to cold air passing over the pond surface. The two most important influences on the magnitude of convective heat loss are wind velocity and temperature difference between the pond surface and the air. This is evidenced in the following equation (Wolf, 1983):

$$q_{cv} = 9.045 \times v \times A \times (Tw - Ta)$$

where:

q_{cv} = Convection heat loss (kJ/hr)
v = Air velocity (m/s)
A = Pond area (m^2)
Tw = Water temperature (°C)
Ta = Air temperature (°C)

For an indoor pool, this equation would be (Lauer, undated)

$$q_{cv} = 9.00(Tw - Ta)1.25 \times A$$

Figure 4. 4 illustrates the importance of air velocity on convective heat loss. The shape of this curve would also be similar for evaporation loss.

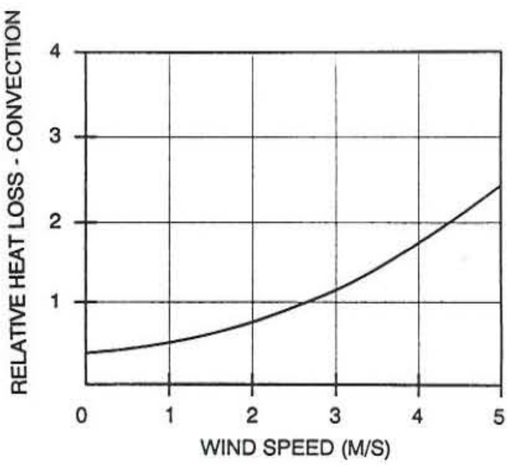

FIGURE 4.4: Plot of relative convective heat loss vs. wind speed

4.4.3 Radiant loss

Radiant heat loss, the third largest component of the total heat loss, is dependent primarily on the temperature difference between the pond surface temperature and the surrounding air temperature. Under normal circumstances, radiant heat exchange is assumed to occur between solid bodies with little or no gain to the air in between the bodies. However, due to the evaporative losses near the pond surface, the air tends to contain a large quantity of water vapour. When this is the case, the pond surface radiates to the water vapour in the air, which is assumed to be at the temperature of the air itself. The equation which describes this process is as follows (Stoever, 1941):

$$q_{RD} = 1.836 \times 10^{-8} ([492 + 1.8\,Tw]^4 - [492 + 1.8\,Ta]^4) \times A$$

where

q_{RD} = Radiant heat loss (kJ/hr)
Tw = Pond water temperature (°C)
Ta = Air temperature (°C)
A = Pond surface area (m^2)

4.4.4 Conductive loss

The final mode of heat loss is that of conduction. This is the loss associated with the walls of the pond. Of the four losses, conduction is by far the smallest and in many calculations is simply omitted. The method on the following page (ASHRAE, 1985) is valid for a pond depth of 1 to 2 m.

$$q_{CD} = ([L+W]\,12.45 + [L+W].4084)\,(Tw - Ta - 8.33))$$

where

q_{CD}	= Conductive heat loss (kJ/hr)
L	= Length of pond (m)
W	= Width of pond (m)
Tw	= Design water temperature (°C)
Ta	= Design outside air temperature (°C)

4.4.5 Summary

$$q_{Total} = q_{EV} + q_{CV} + q_{RD} + q_{CD}$$

It must be noted that these losses are the peak or maximum heat loss. At any given time during the year, other than the design case, the heat loss would be less than this value. The annual heating requirement cannot be determined from simply multiplying the peak heating requirement by 8,760 hours/year. Because of the need for consideration of varying temperature, wind, humidity and solar heat gain, methods for calculating the annual heating requirement are beyond the scope of this article.

However, in general, if the load factor (LF) can be estimated, the annual heat requirement is

$$q_A = q_T \times 24 \times 365 \times LF$$

where

q_A	= Total heat requirement (kJ/yr)
LF	= 0.40 (Worldwide average)

4.4.6 Surface cover

As mentioned earlier, heat losses from the pond surface are most heavily influenced by wind velocity and the temperature difference between the pond and the surrounding air. Any method which can be employed to reduce either of these values would substantially reduce heating requirements.

For outdoor pools, a floating cover is an excellent example. The use of a 1.2 cm floating foam cover (on the pool surface) would reduce the peak loss. This reduction is in large measure, a result of the floating-type cover. Unfortunately, a floating cover is generally not considered practical for commercial aquaculture applications.

4.4.7 Pond enclosure

A pond enclosure is another, though much more expensive, option for reducing heat loss. The advantages provided by an enclosure depend to a large extent upon the construction techniques employed (covering material, degree of enclosure, pressure or absence of ventilation). The variety of construction methods and materials available are too numerous to cover here. The basic advantages of an enclosure are reduced air velocity, reduced temperature difference between the pond and surrounding air, and reduced vapour pressure difference between the pond water and air (increased relative humidity). These effects reduce the losses associated with evaporation, convection and radiation.

4.4.8 Thermal mass

One final method for reducing peak heating requirements for pond or pool heating lies in the use of the large thermal mass supplied by the water itself. Water is an excellent heat storage medium. This stored heating capacity can be used to reduce the peak heating requirements on the heating system.

As a result, the pond will cool by several degrees during the night. The heating system would then bring the pond back up to the temperature during the day when higher temperatures and solar gain would reduce heating requirements.

The degree to which thermal storage can be incorporated into the heating system design is a complex issue of environmental factors, pond characteristics, and the species being raised. Some species, such as prawns, are particularly sensitive to temperature fluctuations (Johnson, 1978).

4.4.9 Flow requirements

The rate of flow required to meet the peak heating demand of a particular pond is a function of the temperature difference between the pond water and the resource temperature. The following equation can be used to determine the flow requirement:

$$Q = \frac{q_{Total}}{15{,}038 \times (Tr - Tw)}$$

where

Q	= Resource flow requirements (L/s)
q_{Total}	= Total calculated pond heat loss (kJ/hr)
	$= q_{EV} + q_{CV} + q_{RD} + q_{CD}$
Tw	= Pond temperature (°C)
Tr	= Resource temperature (°C)

Again, the point is made that this is the peak requirement. The required flow at any other time would be a smaller value. This approach is valid for aquaculture projects and resource temperatures up to levels which would prove harmful if supplied directly to the pond. Above this temperature (which varies according to species), the heating water would have to be mixed with cooler water to reduce its temperature. Two methods are possible for mixing. If a sufficient supply of cold water is available, the hot water could be mixed with the cold water prior to introduction in the pond. A second approach, which would apply in the absence of cold water, would be to recirculate pond water for mixing purposes. The recirculation could be combined with an aeration scheme to increase its beneficial effect. In both cases, the quantity of cold or recirculated water could be determined by the following formula:

$$Q_c = \frac{Q_h(T_h - T_m)}{T_m - T_c}$$

where

Q_c = Required cold flow rate (L/s)
Q_h = Hot water flow rate (L/s)
T_h = Temperature of hot water (°C)
T_c = Temperature of cold water (°C)
T_m = Temperature of desired mixed water (°C)

The above methods are presented to provide an introduction to the subject of heat losses from ponds. The equations provided are simplifications of very complex relationships and should be employed only for initial calculations. In addition, losses which can occur from various aeration schemes and other activities have not been addressed. It is strongly recommended that a competent engineer be enlisted for final design purposes.

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LECTURE 5

GROUND-SOURCE (GEOTHERMAL) HEAT PUMPS

5.1 INTRODUCTION

When a geothermal resource temperature falls below the 100 to 120°F (40-50°C) range, it is frequently impractical to use the fluid directly for most applications. A heat pump can be used to transfer heat from a low-temperature resource to that of a high-temperature reservoir, thus providing the higher temperature needed for space heating. The process can also be reversed by removing heat from a high-temperature resource and rejecting it to a lower temperature reservoir, thus providing cooling to a space.

Air-source heat pumps have been used for many years for both space heating and cooling; however, their efficiency is influenced by the variation in outside air temperature. When heat is most needed, the outside air is cooler, thus often requiring backup electric resistance heating during the coldest days. Similarly, cooling is most needed during the hottest days, requiring the equipment to work at low efficiencies.

Ground-source heat pumps, often referred to as geothermal heat pumps, overcome the problem of resource variations, as ground temperatures remain fairly constant throughout the year. Depending upon the soil type and moisture conditions, ground (and groundwater) temperatures experience little if any seasonal variations below about 30 ft (10 m).

The ground-source or geothermal heat pumps (GSHP or GHP), thus have several advantages over air-source heat pumps. These are: (1) they consume less energy to operate, (2) they tap the earth or groundwater, a more stable energy source than air, (3) they do not require supplemental heat during extreme low outside temperature, (4) they use less refrigerant, (5) they have a simpler design and consequently less maintenance, and (6) do not require the unit to be located where it is exposed to weathering.

The main disadvantage is the higher initial capital cost, being about 30 to 50% more expensive than air source units. This is due to the extra expense and effort to bury heat exchangers in the earth or providing a well for the energy source. However, once installed, the annual cost is less over the life of the system, resulting in a net savings. The savings is due to the coefficient of performance (COP), averaging over 3 for GSHP as compared to 2 for air-source heat pumps. A corresponding improvement is obtained in the cooling mode, as measured by the energy efficiency ratio (EER). These terms are defined later in this paper.

5.2 NOMENCLATURE (Kavanaugh, undated)

Two basic types of ground-source heat pumps exist: ground-coupled and water source. There are a variety of names for the ground-coupled heat pumps. These include ground-source heat pumps, earth-coupled heat pumps, earth energy systems, ground source systems, geothermal heat pumps, closed-loop heat pumps, and solar energy heat pumps. Much of the confusion arises from local marketing needs. Sales people may wish to connect GCHPs to renewable energy sources (solar, geothermal), dissociate them from air hear pumps (GS systems), or connect them to environmental awareness (earth energy). A generally (although not universally) recognized nomenclature is shown in Figure 5.1.

Ground-coupled heat pumps are a subset of ground-source heat pumps (GSHPs). GSHPs also include groundwater and lake water heat pumps (water source). The distinguishing feature of GCHPs is that they are connected to a closed-loop network of tubing that is buried in the ground. The most common method of ground-coupling is to bury thermally-fused plastic pipe either vertically or horizontally. A water or antifreeze solution is circulated through the inside of the tubing and heat is released to or absorbed from the ground. No water enters the system from the ground. Water-to-air heat pumps are located inside the building and are connected to the water loop with a circulator pump. This type of system is referred to as a secondary fluid GCHP since there is an intermediate liquid between the refrigerant and the ground.

A less frequently used system is referred to as a direct expansion (DX) GCHP. Refrigerant lines are buried in the ground in either a vertical or horizontal arrangement. Thus, the intermediate heat exchanger and fluid are eliminated. The possibility of higher efficiency than secondary fluid GCHPs does exist. However, larger charges of refrigerant are required and system reliability is compromised. Therefore, the future of DX GCHP is not clear because of environmental concerns.

A variety of ground-coupled heat

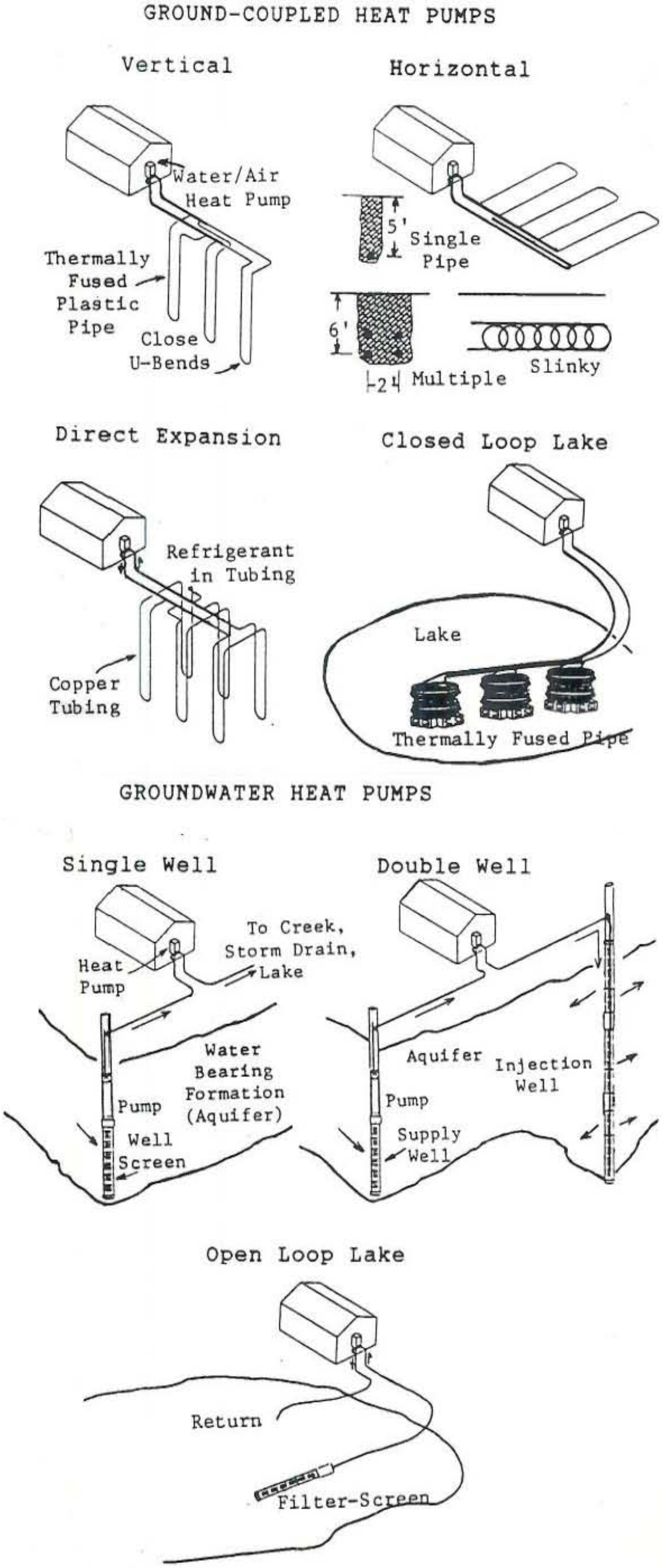

FIGURE 5.1: Ground-source heat pump types

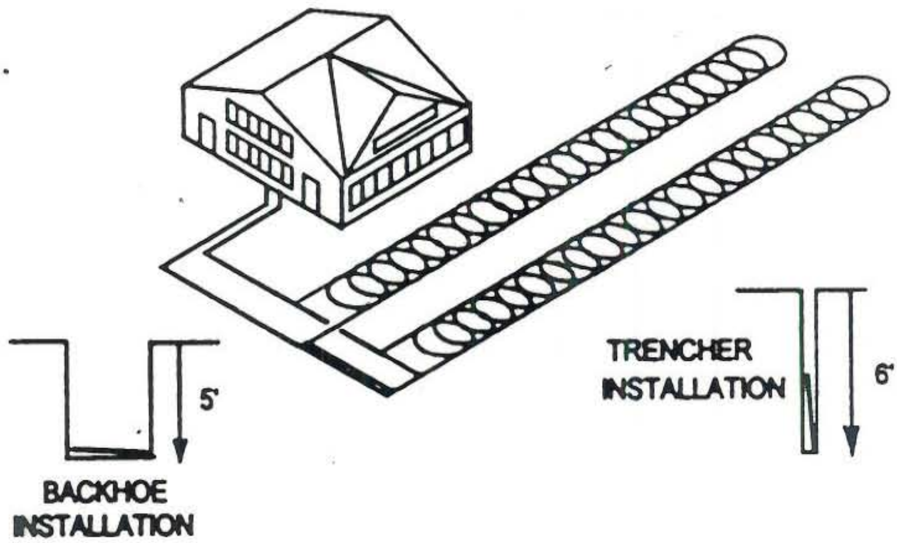

FIGURE 5.2: Slinky coil ground-coupled heat pump system

exchanger designs have recently been proposed and demonstrated as cost-cutting alternatives. These include the Geo-Bag (a large plastic bag buried in the ground and filled with water), a large diameter borehole with spiral coils, and the horizontal-placed slinky coil (Figure 5.2).

5.3 DETAILS ON THE COMMON GROUND-SOURCE HEAT PUMPS SYSTEMS (Lund, 1989)

Two major types exist: ground-coupled (closed loop) or water source (open loop). The ground coupled uses a buried earth coil with circulating fluid in a closed loop of horizontal or vertical pipes to transfer thermal energy to and from the earth. The water source uses a well or an open pond to provide an energy source or sink. Ground-coupled systems have been used in northern Europe for many years, but were not used on a commercial scale in the U.S. until 1980. Ground coupling is used where insufficient well water is available, where the quality of the well water is a problem, where drilling and casing of wells are expensive, or where disposal of well water is restricted.

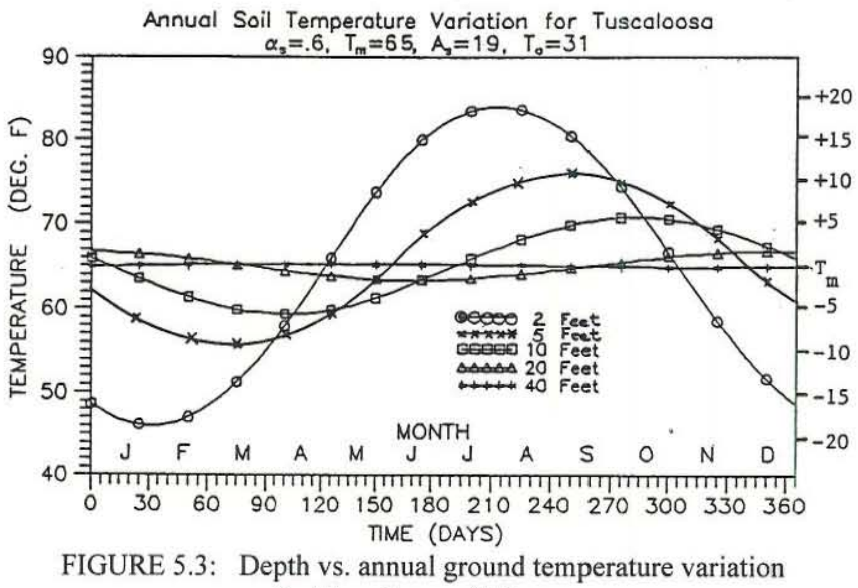

FIGURE 5.3: Depth vs. annual ground temperature variation for Tuscaloosa, Alabama

In the horizontal mode of the ground-coupled system, pipes are buried in trenches spaced a minimum of 5 ft (1.5 m) apart and from 4 to 6 ft (1.2 to 1.8 m) deep. This allows for minimum thermal interference between pipes; however, this system is affected by solar radiation. Solar radiation will cause a cycling of soil temperatures, that lags in time and decreases with depth due to the insulating properties of the soil as shown in Figure 5.3); however, the temperature is much more stable than for air source units. Moist soil will have greater temperature swings than dry soil. The loops can be placed in a double layer as shown in Figure 5.4. Vertical installation (Figure 5.5) of the coils are used where land space is limited or trenching would disturb the surface landscape, and drilling costs are reasonable. Holes are drilled approximately 150 ft (45 m) deep and 15 to 20 ft (4.5 to 6.0 m) apart.

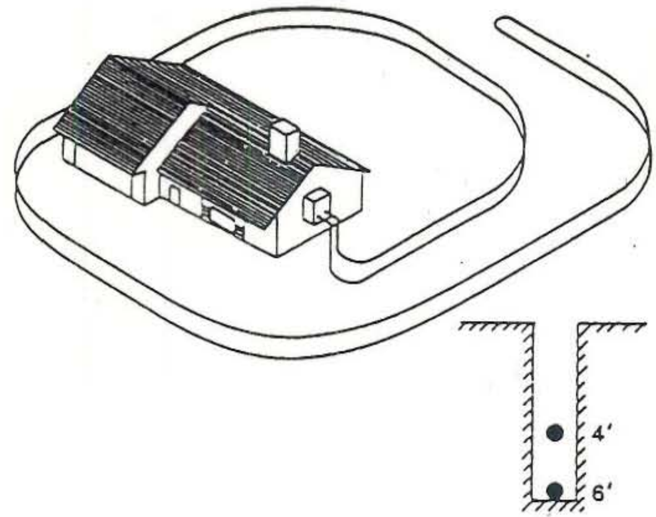

FIGURE 5.4: Two-pipe horizontal ground heat exchanger (source: Oklahoma State University):
- Earth coil type: Horizontal - two-layer;
- Water flow: Series;
- Typical pipe size: 1 ½ -2". (3.8-5.1 cm);
- Practical length
  - Double loop: 210-300 ft/ton (18.2-26.0 m/kW);
  - Single loop: 420-600 ft/ton (36.4-52.0 m/kW);
- Burial depth: 4-6 ft (1.2-1.8 m)

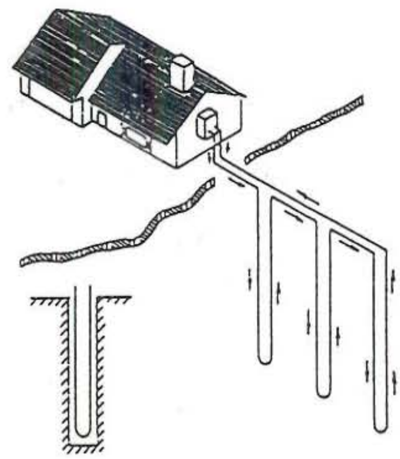

FIGURE 5.5: Series vertical ground heat exchanger (source: Oklahoma State University):
- Earth coil type: Vertical - Single U-Bend;
- Water flow: Series;
- Pipe sizes: 1, 1 ½ & 2" (2.5, 3.8 & 5.1 cm);
- Bore length: 165-200 ft/ton (14.3-17.3 m/kW);
- Pipe length (single pipe): 230-400 ft/ton (19.9-34.7 m/kW)

Computer programs have been developed (Dexheimer, 1985) to calculate the length of horizontal earth coils for heating and cooling. Polyethylene pipes are the most popular in use, and along with socket-fusion joining, are usually guaranteed for over 50 years.

Whereas horizontal loops are affected by solar radiation, rain and wind, the vertical loops are controlled by the mean-annual temperature of the area and the geothermal gradient, and thus have a more stable temperature environment. Water wells are usually used where one is already available, such as for domestic water supply. Normally, a minimum diameter of 6 in. (15 cm) and a production of about three gallons per minute per ton (3.23 L/min/kW) of heat pump capacity is required. Three tons (10.5 kW), a typical residential load, requires about nine gallons per minute (34.1 L/min). The 6 in. (15 cm) diameter well casing is required to place the pump and return line (Figure 5.6). The fluid can either be returned to the well by the return line, placed in an injection well, or disposed on the surface such as for irrigation. Pipes have also been anchored to the bottom

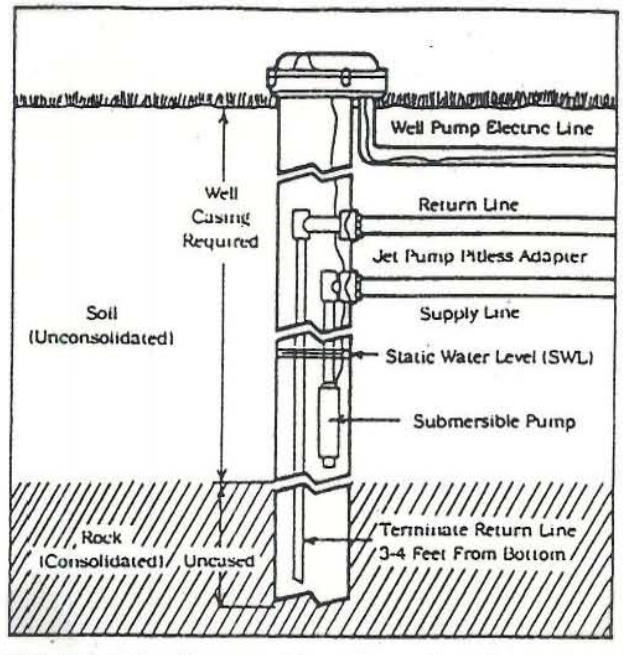

FIGURE 5.6: Cross-section view of geothermal well (source: Water-source Heat Pump Book)

of surface ponds (minimum depth of 6 ft (1.8 m)); however, the heating and cooling capacities are affected by solar radiation and other surface weather factors similar to the horizontal loops. Installation is cheaper and heat transfer is more efficient; however, ponds do not maintain a constant temperature as wells do and the pipes are more vulnerable to accidental damage.

5.4 HEAT PUMP OPERATION (Kavanaugh, 1991)

The operation of the heat pump unit is the same for air-source and ground-source configuration. The main difference is that the air-source requires an outside unit (accumulator and fan) which may frost up in cold weather, requiring frequent defrosting. They also require a backup heating source (electric or gas) when outside temperatures are too low for efficient operation. The operation and cycle in both cooling and heating mode of the heat pump are shown in Figures 5.7 and 5.8 (Oklahoma State University, 1988).

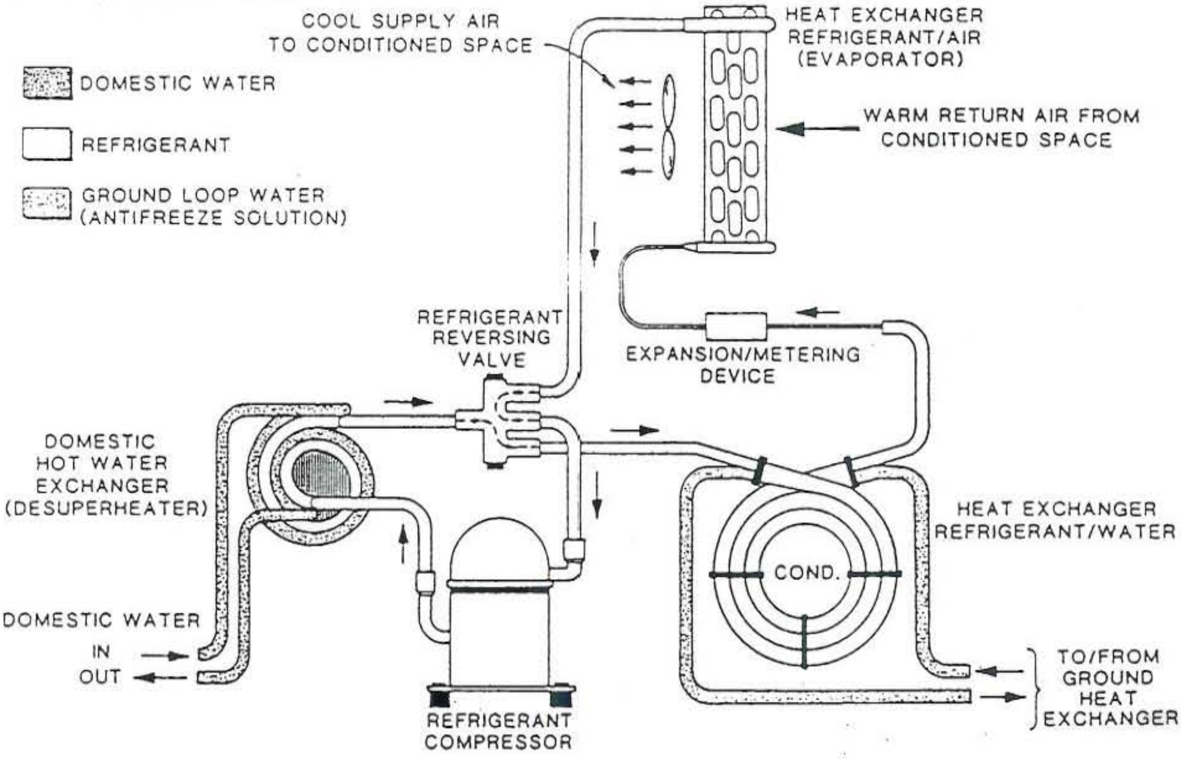

FIGURE 5.7: Cooling cycle (source: Oklahoma State University)

As seen from these figures, the basic components of a standard heat pump are an electric motor-driven compressor, a reversing valve, an expansion device, and two heat exchangers. A desuperheater can be added to heat domestic hot water as shown on the left side of the figures. One heat exchanger transfers heat between the heat pump and the environment, and the second transfers heat to and from the interior of the building, referred to as the condenser or evaporator (depending on which mode is used).

Refrigerant enters the compressor shell as a low-temperature, low-pressure gas. It passes around the motor and is heated before entering the intake of the compression chambers. The compression process elevates both the pressure and temperature of the refrigerant gas. In the cooling mode (Figure 5.7), this gas enters the reversing valve and is routed to the heat exchanger in contact with the environment (ground heat exchanger). Since the gas is at a high temperature, relatively cool air or water from the environment (geothermal source) can be used to remove heat from the refrigerant in the heat exchanger. Removal of heat results in the cooling and <u>condensing</u> of the refrigerant. Pressure loss is usually small in the condenser; therefore, the refrigerant exits the condenser as a liquid with a temperature slightly above the environment's.

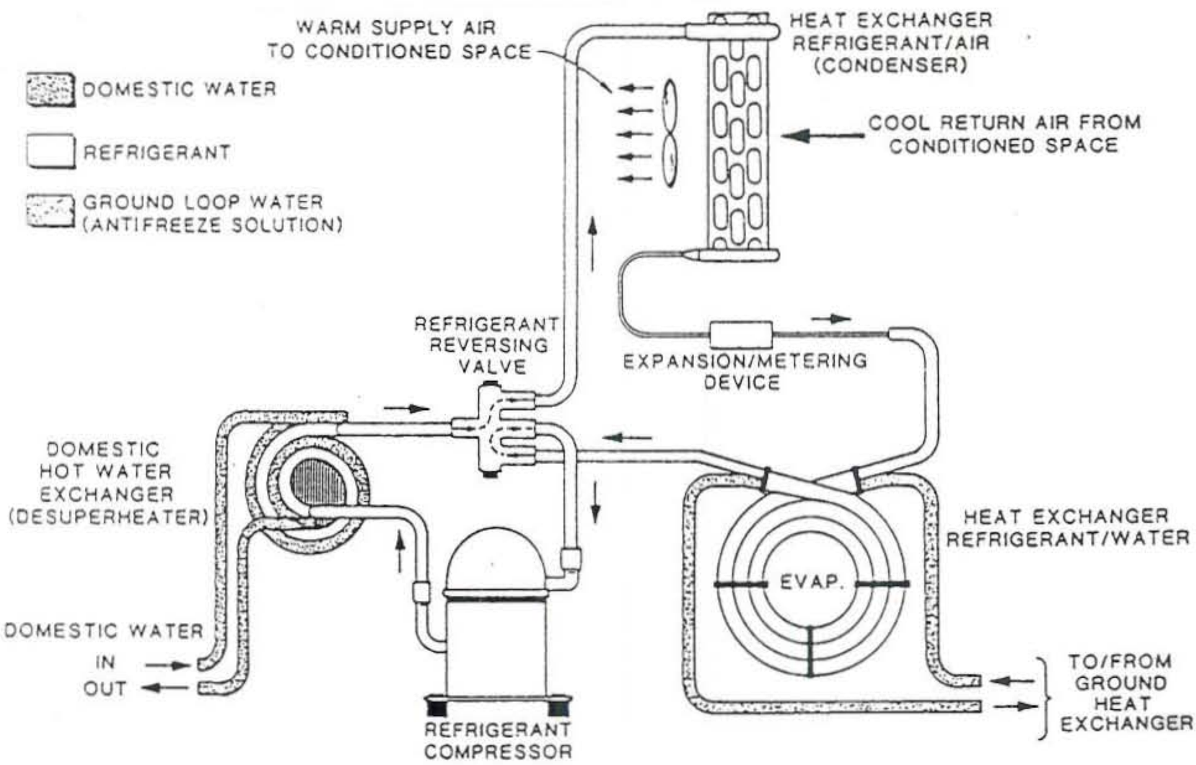

FIGURE 5.8: Heating cycle (source: Oklahoma State University)

The liquid then experiences a drop in pressure across the restriction in the expansion device. This causes a rapid decrease in temperature. The temperature inside the building is much warmer than the refrigerant entering the indoor heat exchanger. Therefore, the liquid is evaporated, and in the process, heat is removed from the building air in the evaporator. Thus, we have the desired cooling effect. The evaporated gas is then passed through the reversing valve before returning to the compressor.

In the heating mode (Figure 5.8), the solenoid in the reversing valve is switched so that the hot compressor discharge gas is routed to the indoor heat exchanger. This exchanger is now used to condense the hot refrigerant with the relatively cool indoor air; therefore, the desired heating effect is carried out on the building. The condensed liquid enters the expansion device in a reverse direction. The pressure loss results in a temperature decline so that the environment (ground) transfers heat to the cold refrigerant. This causes the refrigerant to evaporate. The low-temperature, low-pressure gas returns to the compressor through ports in the reversing valve.

Actual heat pumps may have additional components such as a multiple expansion devices, fans, pumps, additional heat exchangers, auxiliary heat sources, accumulators, and control and safety mechanisms. However, the basic means of "pumping heat" between the building and the environment (geothermal source) is essentially the process shown in Figures 5.7 and 5.8.

5.5 EXAMPLES OF GROUND AND WATER-SOURCE HEAT PUMP TYPES (Kavanaugh,1991)

Figure 5.9 shows four common ground and water-source heat pumps. The home on the far left utilizes a vertical closed-loop and ground-coupled heat pump. Water is circulated through a water-to-refrigerant heat exchanger (condenser in cooling, evaporator in heating) in the heat pump. Upon leaving the heat pump, the water passes out of the house and into under-ground headers buried 2-4 ft (0.6-1.2 m) below

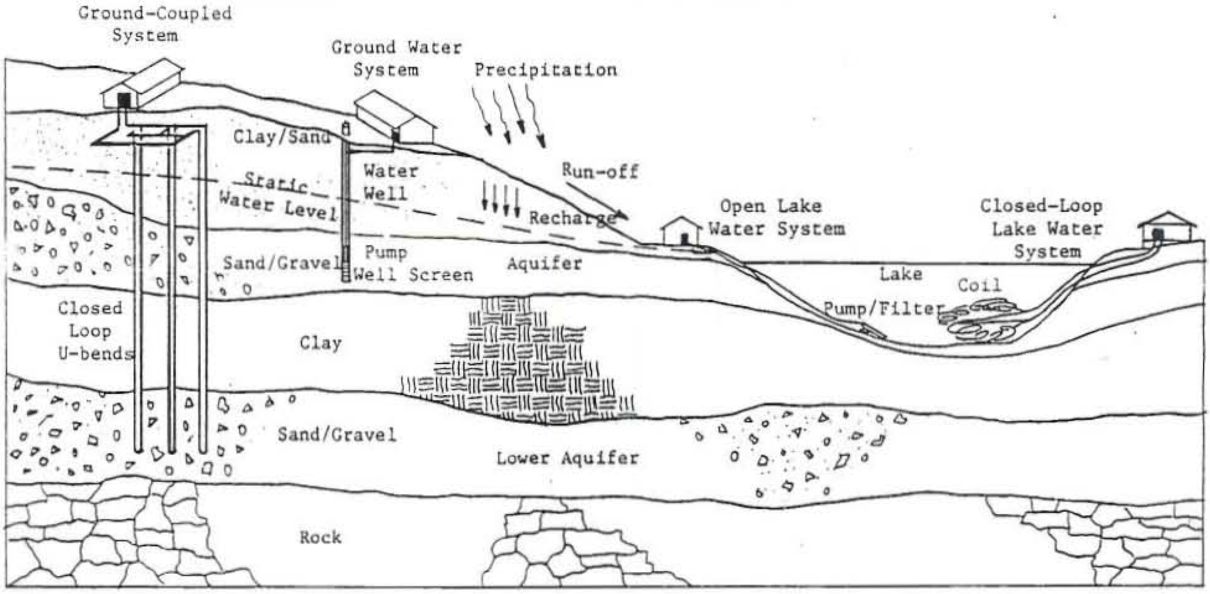

FIGURE 5.9: Ground-coupled, groundwater and two lake-water heat pump systems in a typical hydrogeological formation

the surface. In this installation, the flow is split into three vertical U-Bend ground couplings. Bore depth is typically 150-250 ft per ton (13.0-21.7 m/kW) of heat pump, pipe size of 3/4-1 ½ in. (1.9-3.8 cm), and bore diameter of 4-6 in. (10-15 cm). Drilling depth is determined by geological considerations. Obviously, the driller in this installation stopped when the rock strata was encountered. Deeper or more shallow bore depths are possible as long as the required bore per ton requirement is maintained. Note also that water flow is split into three circuits. This is often necessary to minimize water pump size.

The next house is heated and cooled by a groundwater heat pump system. A well (4 in. [10 cm] casing minimum) is drilled into the shallow aquifer. A well screen is placed on the bottom section of the casing through which groundwater is extracted. A submersible pump located slightly above the screen delivers water to the heat pump in the house. In this installation, the water is obviously disposed of at the surface; however, it could also be reinjected if aquifer water levels are critical. Two important requirements in these systems are a clean, well-developed well, and that the pump should be located several feet (~ 1 m) below the lowest seasonal static water level. It would not be uncommon to drill to a deeper aquifer; although, the shallow aquifer is preferred in terms of operating and first cost.

Lakes can be used in two manners. The home on the left shore utilizes an open-loop heat pump system. Water is pumped from a submersible pump near (but not on) the lake bottom and through the heat pump, and is returned near the surface of the lake. Above surface pumps can also be used. They require greater power input, freeze precautions and some method of preventing loss of suction during off-periods.

The home on the right lake front is heated and cooled with a closed-loop heat pump. Much like the ground-coupled system, water (or a water/antifreeze mixture) is circulated through a copper or thermally-fused plastic pipe coil and the heat pump. These systems do not require water filtration like the open systems. However, they often do require freeze protection and protection from damage, and performance improvements are moderate in <u>shallow</u> lakes.

Specific examples of GCHP use

Worldwide, GSHP account for 12% of the geothermal energy used for direct applications, amounting to approximately 13,000 TJ (3,600 GWh) annually. Present estimates indicate that there are 60,000 groundwater and 90,000 ground-coupled (55% vertical) heat pump installations in the USA (updated from Ellis, 1989). The annual growth rate is estimated at 17%. Ground-source heat pumps account for 59% of the geothermal direct use in the USA, amounting to over 8,000 TJ (almost 2,300 GWh) annually. A typical installation, which would be for a single-family residence, consists of a 3 ton unit (10.5 kW) using about 8 gals/min (30 L/min) with a 10°F (6°C) temperature drop in the circulating fluid. This would shave about 5 kW off winter-peak heating demand and about 2.5 kW from summer demand. Thus, 200,000 homes using GSHPs would avoid constructing a new 1,000 MW power plant. Although the incremental cost of the ground-coupled closed loop adds about US$ 3,000 to the cost of a residential heating system, payback occurs in 3-5 years from money saved on utility bills. Currently, the main GCHP uses is in the mid-western and southeastern states, where many utilities are offering rebates of US$ 500-2000 to homeowners to install GSHP in order to take advantage of the peak shaving (by increasing load leveling for the utility, referred to as demand side management) (Lund, 1988; Lienau and Lund, 1992).

The largest GSHP installation in the United States is the Galt House East Hotel and Waterfront Office Building in Louisville, Kentucky (Pinckley, 1995). Heating and air conditioning is provided for 600 hotel rooms, 100 apartments, and 960,000 square feet (89,000 square meters) of office space, totaling 1,740,000 square feet (161,650 square meters). The system can extract 2,800 gals/min (177 L/s) of groundwater from four wells at 58°F (14°C) and can either remove energy from the well water for heating or add heat to the well water from the air conditioning. The water is then discharged into a storm-water system. The system provides 4,500 tons (15.8 MW) of cooling and 67 million Btu/hr (19.6 MW) of heating. The hotel complex energy use is approximately 53% of a similar non-GSHP system in an adjacent unit, for a monthly savings of approximately US $25,000. The emission of CO_2, SO_2, HO_x and particulates are also reduced.

There have also been increased utilization of GSHP in Europe, especially in Germany, France and Switzerland. In Switzerland, almost 6,000 units using 10,000 vertical borehole heat exchangers (BHE) have been installed and have been operating reliably for decades (Rybach and Hopkirk, 1995). A typical single-dwelling house has a capacity demand of about 10 kW; however, the BHE system is 30 to 40% higher in cost in comparison with a conventional oil-fired- system. Environmental awareness, enforced by a governmental subsidy (US$ 200 per heat pump kW), is the main incentive for the BHE installations.

5.6 EQUIPMENT (Rafferty, 1991)

In the commercially available size range, equipment is in two basic configurations: positive displacement and centrifugal. Centrifugal machines are used for the largest applications with positive displacement equipment for smaller capacities. The following sections briefly discuss each of these types of equipment.

5.6.1 Positive displacement

Reciprocating compressor heat pumps, the most common positive displacement type, are available as standard units in sizes generally below 3 x 10^6 Btu/h (0.88 MW) heating output (McQuay, 1986). This equipment employs a 1, 4, 6, or 8-cylinder compressor on smaller equipment and multiple 4 or 6-cylinder compressors on larger units (Carrier Corp., 1981). These units are also used for the smaller residential applications of around 3 tons or 36,000 Btu/hr (10.5 kW).

Capacity control is accomplished by suction cut-off type cylinder unloading down to ~15-20% capacity below which hot gas by-pass must be employed. As a result, it is important, particularly for space heating applications, that equipment selection considers off-peak operation. The number of control steps is dependent upon the number of compressor cylinders, with 4-step control available on the smaller units and up to 8-step control on the large units.

Table 5.1 illustrates the off-peak performance of small, medium and large (with respect to the 3 x 10^6 Btu/hr (0.88 MW) capacity) reciprocating heat pumps.

TABLE 5.1: Off-peak performance of reciprocating heat pumps (McQuay, 1986)

Small		**Medium**		**Large**	
Capacity (%)	**kW (%)**	**Capacity (%)**	**kW (%)**	**Capacity (%)**	**kW (%)**
100	100	100	100	100	100
86	81	92	88	94	90
68	60	84	76	87	80
24	20	72	64	75	62
-	-	60	54	59	46
-	-	31	27	51	37
-	-	-	-	42	29
-	-	-	-	23	14

The increase in efficiency at part load is because of the nature of capacity control employed by the manufacturer from which data were taken. This is a result of a special unloading arrangement and the part load operation of two separate refrigerant circuits on the heat pump. This increases the amount of heat transfer area available in the evaporator and condenser relative to load requirements, thus increasing efficiency.

The refrigerant employed is a function of the temperatures between which the machine is working. Table 5.2 presents a summary of refrigerant temperature limitations.

TABLE 5.2: Reciprocating heat pump refrigerant temperature limitations (McQuay, 1986; Carrier Corp., 1981)

Refrigerant	**Maximum condenser leaving water temperature**		**Minimum/maximum evaporation leaving water temperature**	
	(°F)	**(°C)**	**(°F)**	**(°C)**
R-22	130	54	42/90	6/32
R-500	150	66	40/100	4/38
R-12	170	77	40/100	4/38
R-114	220	104	70/120	21/49

Evaporators are the shell-and-tube type with water generally on the shell side. However, one major manufacturer produces equipment with water on the tube side. Condensers are also shell-and-tube with water on the tube side. Reciprocating machines do not generally include a separate liquid sub-cooling heat exchanger, though sub-cooling is addressed in condenser circuitry (Carrier Corp., 1987).

Packaged reciprocating heat pumps are supplied from the factory with all safety and operating controls for the machine including, in most cases, compressor starters. The machines need only to be interfaced with system controls and a power source.

Other positive displacement models are rotary (rolling piston, rotary valve, single screw and twin screw), and orbital (scroll and trochoidal). Their performance is similar to the reciprocating compressor. See the ASHRAE Equipment Guide for more details.

5.6.2 Centrifugal

Centrifugal heat pumps are available in capacities ranging from ~ 1 x 10^6 to 25 x 10^6 Btu/hr (0.29-7.32 MW) in a single unit (McQuay, 1983). The equipment features a single or dual compressor, depending upon the size. One large manufacturer of this equipment in the U.S. employs a high-speed wheel, driven by a hermetically-sealed squirrel cage motor through a single helical-gear couple. Motor cooling is provided by controlled liquid refrigerant injection. A second manufacturer employs a 3-stage compressor operating at motor speed. Refrigerant temperature limitations are similar to those shown in Table 5.2 for reciprocating equipment.

Stable part-load operation is maintained by inlet guide vanes with the assistance of an adjustable diffuser block at the wheel exhaust (McQuay, 1983). Construction of the balance of the machine is similar to that of the reciprocating machine with the exception that the source water in the evaporator flows through the tubes, rather than the shell as in reciprocating equipment. This configuration permits the use of alternate tube construction materials to accommodate (without the use of a heat exchanger loop) aggressive fluids in certain applications.

5.7 **PERFORMANCE** (Lienau, et al., 1995)

The energy performance of a GSHP system can be influenced by three primary factors: the heat pump machine, the circulating pump or well pumps, and the ground-coupling or groundwater characteristics.

The heat pump is the largest single energy consumer in the system. Its performance is a function of two things: the rated efficiency of the machine and the water temperature produced by the ground-coupling (either in the heating or cooling mode). The most important strategy in assembling an efficient system is to start with an efficient heat pump. It is difficult and expensive to enlarge a ground-coupling to improve the performance of an inefficient heat pump.

Water-source heat pumps are currently rated under one of three standards by the American Refrigeration Institute (ARI). These standards are ARI-320, ARI-325, and ARI-330. The standard intended for ground-coupled systems is ARI-330 entitled "Ground Source Closed Loop Heat Pump Equipment." Under the standard, ratings for cooling EER and heating COP are published. It's important to consider that these are single-point ratings rather than seasonal values as in the case of the air-source equipment. Cooling EER values are based on an inlet water temperature of 77°F (25°C). Heating COP values are based on a heating inlet temperature of 32°F (0°C). These values are characterizations of a northern climate.

The current ARI directory contains equipment with EER ratings of less than 10 to a high of 18.6. COP values range from 2.8 to 3.6. It is evident that there is a wide range of equipment performance at the standard rating conditions. Based on these values, it is evident that the performance of the equipment can vary by as much as 100% according to the quality of heat pump purchased.

In recent years, there has been a substantial increase in the efficiency of GSHP equipment. Based on performance reported in the ARI directory for 1987 and 1994, the increase in EER ranges from 26 to 56 percent, and in COP from 35 to 50 percent depending on the entering water temperature. Figures 5.10 and 5.11 show this increase in performance for a typical machine based on average values of EER and COP as a function of entering water temperature. Based on improvements in performance of GSHPs from 1987 to 1994, the date of a GSHP installation should be noted.

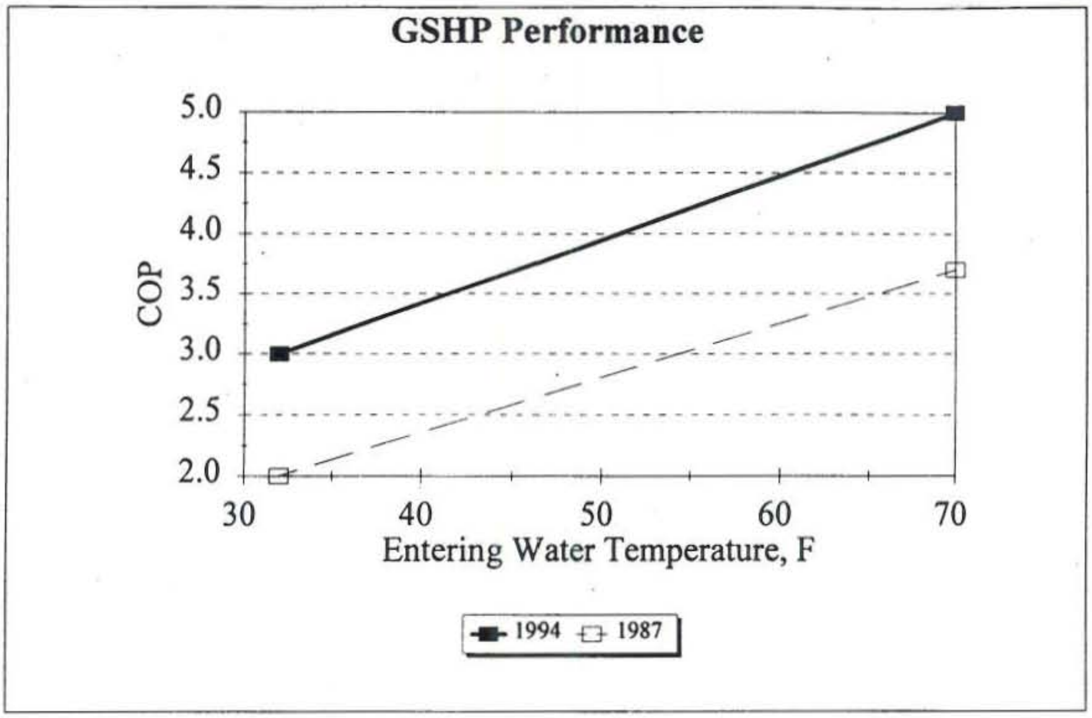

FIGURE 5.10: GSHP performance improvements from 1987 to 1994 for heating mode

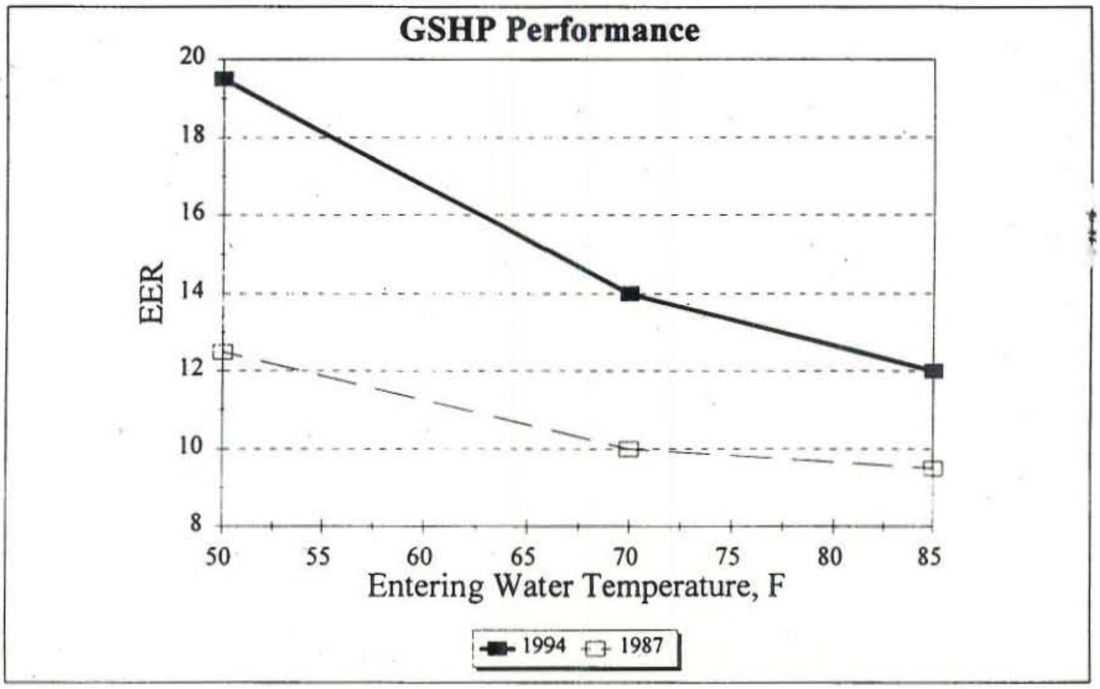

FIGURE 5.11: GSHP performance improvements from 1987 to 1994 for cooling mode

The actual performance of the equipment is a function of the water temperature produced by the ground-coupling. The values discussed above are based on standard rating conditions (77°F cooling and 32°F heating [25°C and 0°C]). The actual temperatures are a function of the local ground temperatures and the design of the ground-coupling. For example, in a region where the local ground temperature is 60°F (16°C) and the ground loop is designed for the customary 20-25°F (-7 to -4°C) above-ground temperature, a heat pump rated at an EER of 16.8 would actually operate at an EER of 14.2 under peak-load conditions. A poorly designed loop, which forces the unit to operate at 30°F (-1°C) aboveground temperature, would reduce the value to less than 13.0. These examples are for cooling operation which is the dominant load in commercial applications. The same relationship holds for heating operations, however.

Figures 5.12 and 5.13 show EER and COP as a function of inlet temperatures for a 3 ton (10.5 kW) machine designed for ground-coupled systems suggests the following guidelines for pumping power for commercial ground-coupled systems.

The system energy performance is also influenced by the pumping energy required to circulate the fluid through the heat pump and the ground loop. One author (Kavanaugh, 1991) in the design of ground-coupled systems suggests the following guidelines for pumping power for commercial ground-coupled systems:

1. Efficient systems: <50 watts/ton (<14 watts/kW)
2. Acceptable systems: 50-100 watts/ton (14-28 watts/kW)
3. Inefficient: >100 watts/ton (>28 watts/kW)

To put these values into perspective, consider an office building with a 50 ton (175 kW) cooling load and heat pump units selected to operate at an EER of 14 under peak conditions.

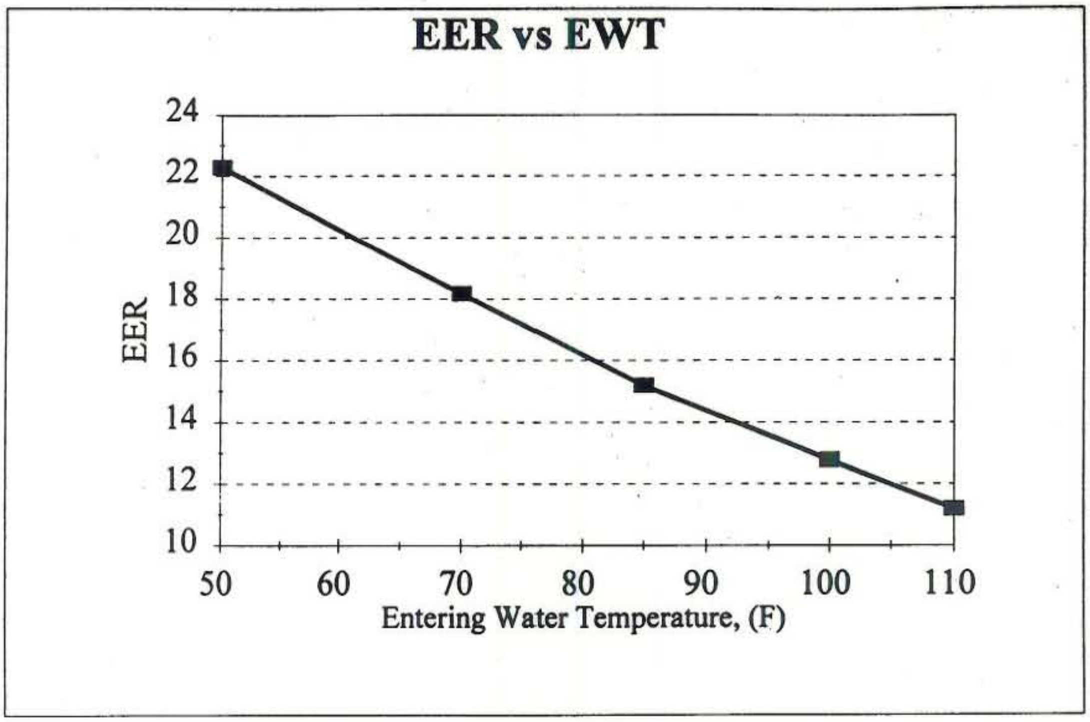

FIGURE 5.12: EER for a 3 ton (10.5 kW) GSHP

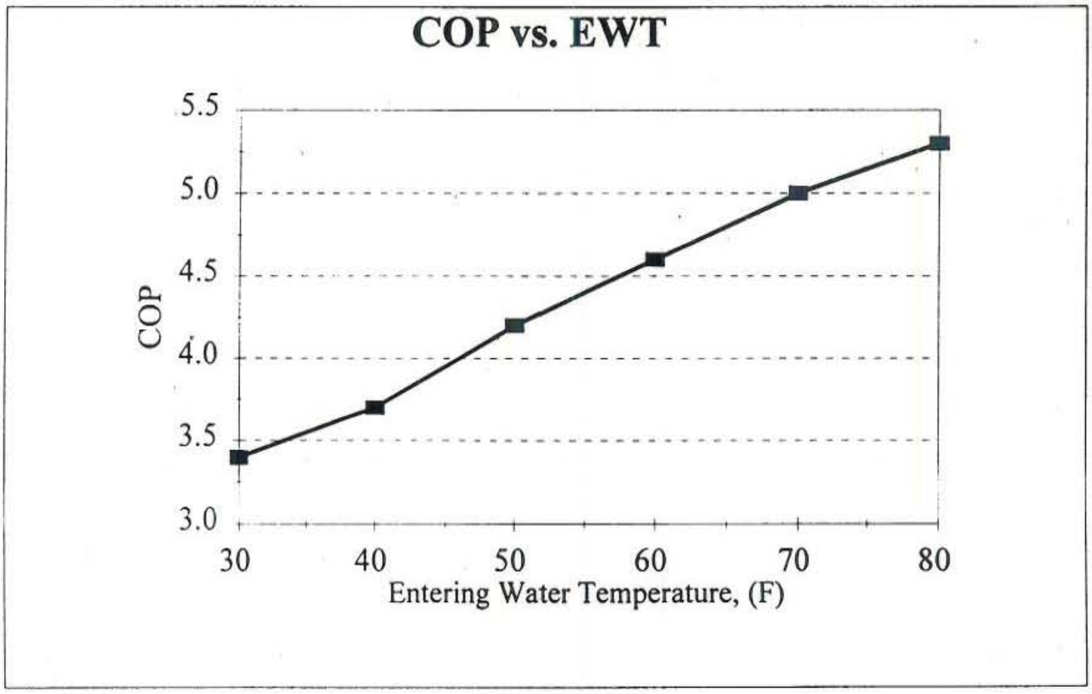

FIGURE 5.13: COP for a 3 ton (10.5 kW) GSHP

With an efficient circulating pump design (35 watts/ton [10 watts/kW]}, the energy demand of the circulating pump would amount to (50 tons) *x* (35 watts/ton) = 1750 watts [175 kW *x* 10 watts/kW = 1750 watts]. Combining the pump demand with the heat pump unit demand results in a system EER of 13.5.

The same building and equipment coupled to a poorly designed pumping system consuming 120 watts (6,000 watts pumping power) per ton would yield a system EER of only 12.2; thus, compromising the premium paid for the higher-efficiency equipment. As indicated above, coupling this system to an inadequate ground-coupling could easily reduce the system EER to between 10 and 11.

In summary, it is necessary when evaluating a ground-coupled system to consider the efficiency of the machine, the adequacy of the ground-coupling and the nature of the pumping design to fully understand the efficiency of the system.

Basic formulas

Energy Efficiency Ratio (cooling mode):

$$EER = \frac{Qc}{Qe} = \frac{cooling\ capacity\ (Btu/hr\ [kW])}{electric\ power\ input\ (compressor)(kW)}$$

Coefficient of Performance (heating mode):

$$COP = \frac{Qh}{Qe} = \frac{heating\ capacity\ (Btu/hr\ [kW])}{electric\ power\ input\ (compresser)(kW)}$$

Usually the heat extraction rate for heating is less than the heat rejection for cooling; thus, the GCHP should be designed to accommodate the larger value.

5.8 HEAT PUMP UNITS (Kavanaugh, 1991)

Currently in the U.S., the water-source heat pump unit most widely used is the packaged water-to-air system. Split systems, air-to-water and water-to-water are offered on a more limited scale. Component variations in water-to-air units occur primarily in the type of expansion device and water-to-refrigerant coils. Before 1975, most units had copper (or copper-nickel) tube-in-tube coils with capillary tube expansion devices that are intended for use with groundwater with temperatures above 55°F (13°C). The market today includes units capable of handling inlet solution temperatures between 25 and 100°F (-4 and 38°C). This is accomplished with better heat exchangers, expansion devices, and compressors. Two commonly used water-to-refrigerant exchangers are modified tube-in-shell (water in tube side) and coaxial tube-in-tube (water on inner tube refrigerant side in the annulus). Both have extended surfaces on the refrigerant side to compensate for the lower film coefficients. Manufacturers are selecting heat exchangers with lower water-side pressure losses to minimize pumping requirements.

Expansion valves permit a much wider acceptable range of refrigerant evaporation and condensation temperatures. This device is especially suited to ground-coupled, lake water, and closed-loop water systems in which temperature fluctuations are experienced.

In hot, humid climates, the addition of a heat exchanger in the high side of the refrigerant loop for heating domestic water is almost always recommended. This device is typically a desuperheater that uses compressor heat to generate hot water either in the cooling mode or with excess heating capacity (available in southern climates) in the heating mode. Units are now available that have larger heat exchangers and control mechanisms that permit the full condensing capacity of the refrigerant circuit to be used for heating water.

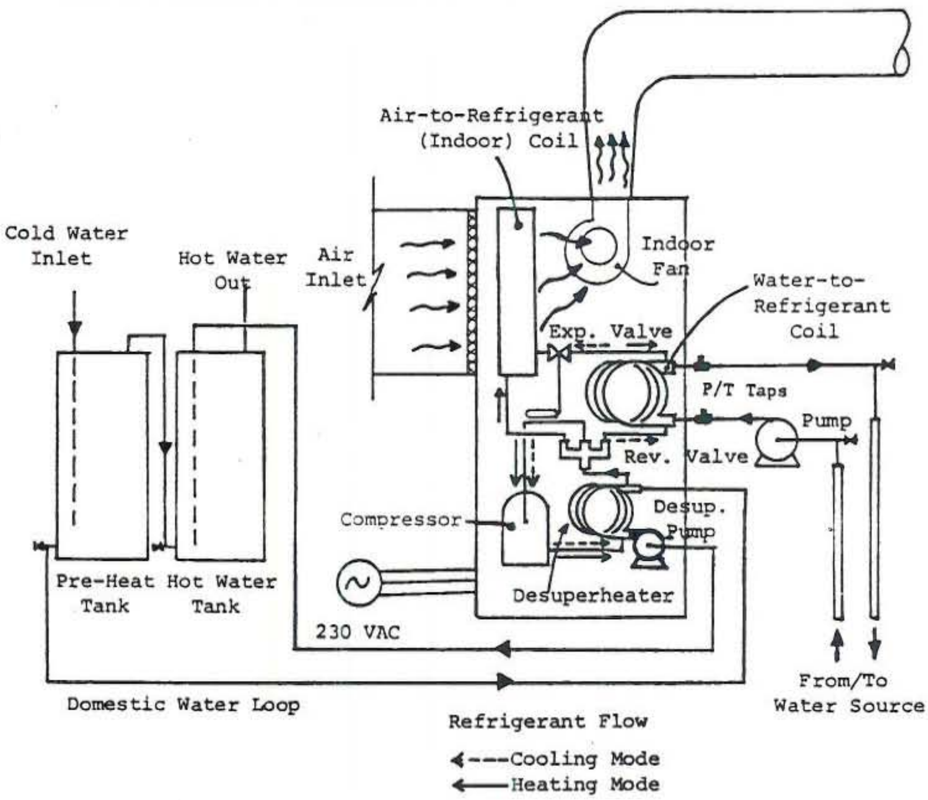

FIGURE 5.14: Typical water-to-air heat pump arrangement

Figures 5.14 and 5.15 show a typical arrangement of a packaged water-to-air heat pump. The desuperheater water heater and pump are typically available either as part of the package or as a field-installed option. The pump for the primary water-to-refrigerant coil is usually not part of the package because its size and type vary significantly.

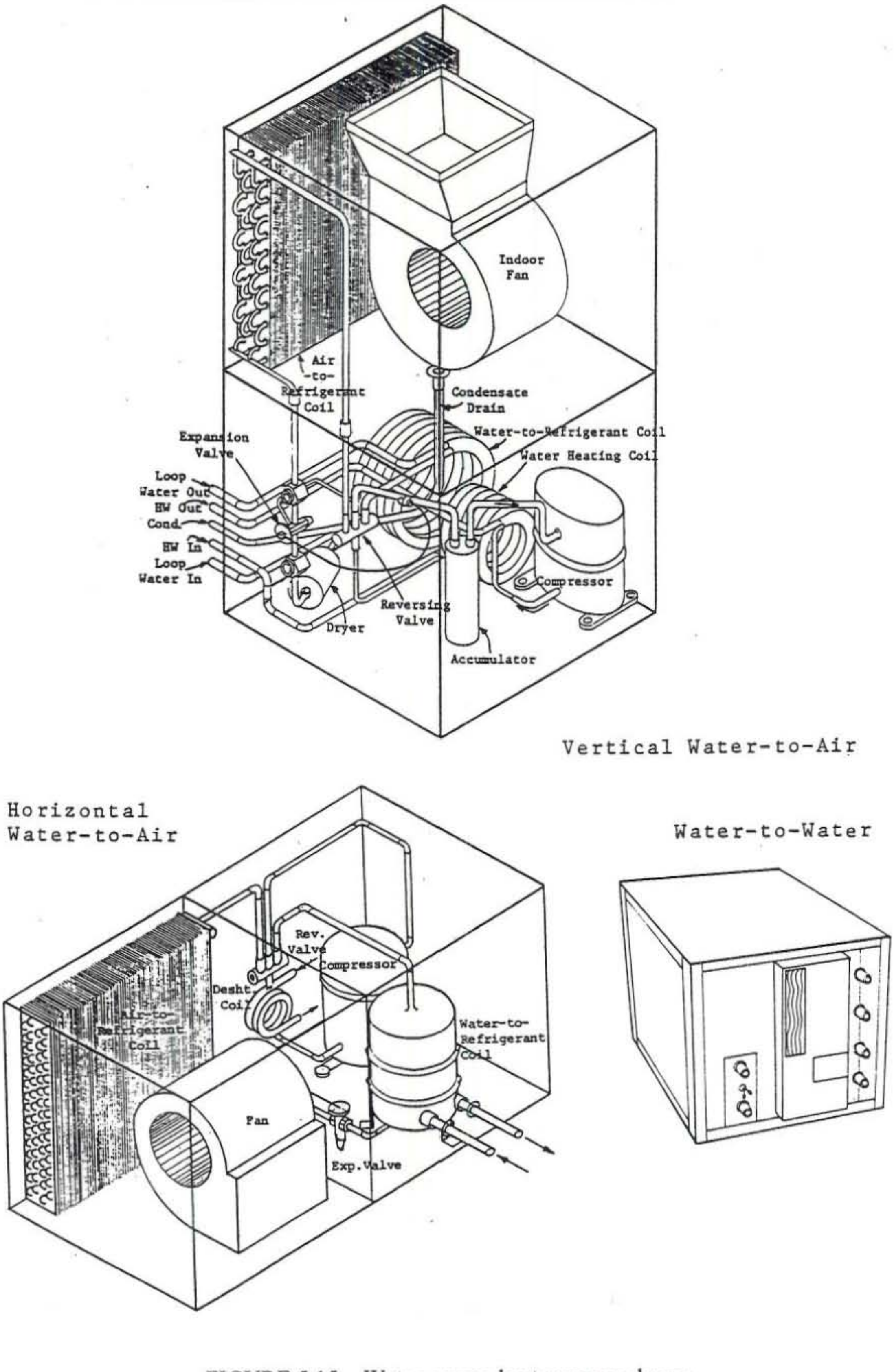

FIGURE 5.15: Water-source heat pump packages

Unit performance

Not all "water-source" heat pumps are alike. Some are optimized for heating in colder climates and some are not intended for low water temperatures. Many have quality, well-designed components; but, some contain "bottom-of-the-line" equipment. The quality of the various units can not be elaborated upon in this paper. However, the rated performance and resulting characteristics can be discussed. Table 5.3 is included to show the wide variation in the performance of four similarly sized water-to-air heat pumps. Total cooling (TC) and unit energy efficiency ratios (EER) are given at two different entering water temperatures (EWT). Note that the EERs are comparable at 85°F (29°C) for the non-scroll compressor models; but, Brand X has a substantially lower value at 65°F (18°C) EWT. The most recent advances have come with the use of scroll compressors (Brand C). They have outstanding heating and cooling efficiencies. However, work remains to be done in order to enhance heating capacities with lower entering water temperatures.

TABLE 5.3: Water-to-air heat pump capacities and efficiencies
9 gpm, 1200 cfm, 80°/67°F EAT (34 L/min, 34 m^3/min, 27°/19°C)

	Cooling				**Heating**				**Water coil head loss**
	TC (1000 Btu/h)		**EER***		**TH (1000 Btu/h)**		**COP***		
EWT (°F)	**65**	**85**	**65**	**85**	**45**	**65**	**45**	**65**	**(ft H_2O)**
Brand A	42	37	14.6	11.2	43	54	4.1	4.5	14
Brand B	38	34	14.6	11.5	32	44	3.3	3.9	15
Brand C (scroll)	42	39	18.7	14.8	34	43	4.1	4.7	7.4
Brand X	37	36	13.0	11.3	21	29	2.3	2.9	20

* If it includes compressor and fan power, deduct 6 to 8% for closed-loop **system** efficiency and 15 to 25% for open-loop **system** efficiency

The differences are even more important in the heating mode. Heating capacity in MBtu/hr and COP are shown for two values of EWT. Note that Brand A has substantially greater capacity and COP with a 45°F (7°C) EWT than either Brands B or X. While Brand X performance may be acceptable with groundwater, its performance will be poor with a ground-coupled or lake water system. Brand C costs more; but, the improvements in cooling efficiency warrant use in cooling-dominated climates. In heating-dominated climates, Brand A may be a better choice.

Another important aspect of water-source heat pumps is pressure drop (or head loss) across the water-to-refrigerant coil. When this value is high, additional or larger pumps will be required. The recommended 6 to 8 percent efficiency penalty may substantially increase in closed-loop systems. Brands A and B have marginally high losses and Brand X's is unacceptable. Brand C has very good head loss.

5.9 COMMERCIAL (LARGE-SCALE) GROUND-SOURCE HEAT PUMP SYSTEMS
(Rafferty and Knipe, 1988; Rafferty, 1995)

Unitary ground-source heat pump systems for commercial buildings can be installed in a variety of configurations. The oldest and, until recently, most widely used approach was the groundwater system. In this design (Figure 5.16), ground-water from a well or wells is delivered to a heat exchanger installed

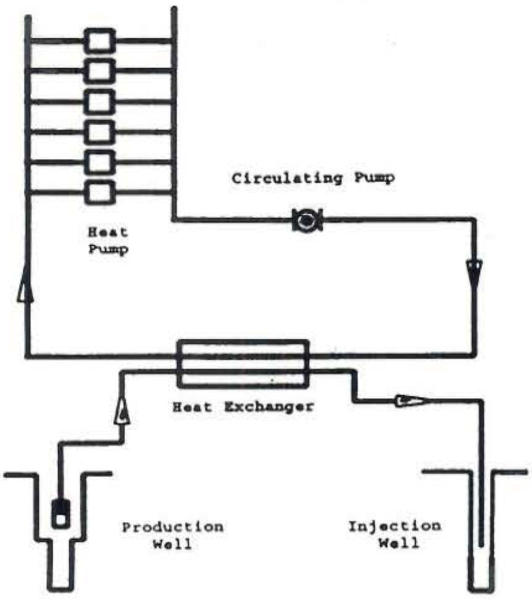

FIGURE 5.16: Groundwater heat pump system

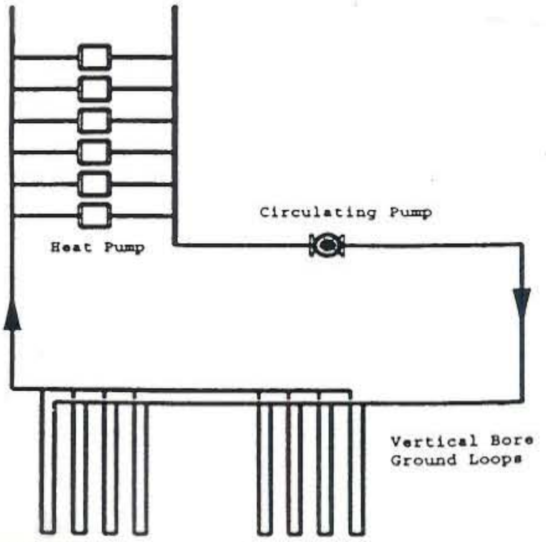

FIGURE 5.17: Ground-coupled heat pump system

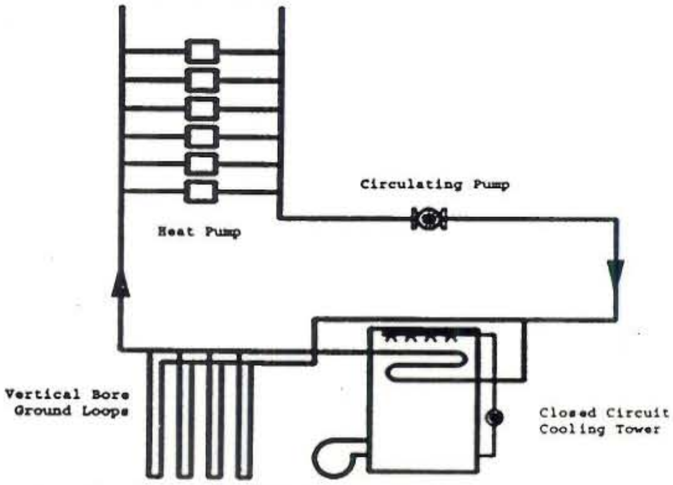

FIGURE 5:18: Hybrid ground-coupled heat pump system

in the heat pump loop. After passing through the heat exchanger (where it absorbs heat from or delivers heat to the loop), the groundwater is disposed of on the surface or in an injection well. The use of an injection well is desirable in order to conserve the groundwater resource.

A second and increasingly popular design is the ground-coupled heat pump system. In this approach (Figure 5.17), a closed loop of buried piping is connected to the building loop. For most larger commercial applications, the buried piping is installed in a grid of vertical boreholes 100 to 300 ft (30-90 m) deep. Heat pump loop water is circulated through the buried piping network absorbing heat from or delivering heat to the soil. The quantity of buried piping varies with climate, soil properties and building characteristics, but is generally in the range of 150 to 250 ft (of borehole) per ton (13-22 m/kW) of system capacity. Borehole length requirements are almost always dictated by heat rejection (cooling mode) duty for commercial buildings.

A third design for ground-source systems in commercial buildings is the "hybrid" system. This approach (Figure 5.18) may also be considered a variation of the ground-coupled design. Due to the high cost associated with installing a ground loop to meet the peak cooling load, the hybrid system includes a cooling tower. The use of the tower allows the designer to size the ground loop for the heating load and use it in combination with the tower to meet the peak cooling load. The tower preserves some of the energy efficiency of the system, but reduces the capital cost associated with the ground loop installation.

In addition to the three designs discussed above, ground-source systems can also be installed using lake water, standing column wells and horizontal ground-coupled approaches. The first three schemes have wide use and broad potential application.

It is common in the ground-source heat pump industry to refer to the costs for the ground-source portion of the system on cost per ton (3.5 kW) basis. In keeping with this practice, most cost data presented is expressed in terms of cost per ton. It is important to note, however, that the cost per ton refers to the actual load imposed on the ground-source portion of the system. This is not the same as the installed capacity of the equipment. Due to load diversity, the peak load imposed upon the heat rejection equipment is always less than the total installed capacity. The load used for cost calculations is frequently referred to by engineers as the block load.

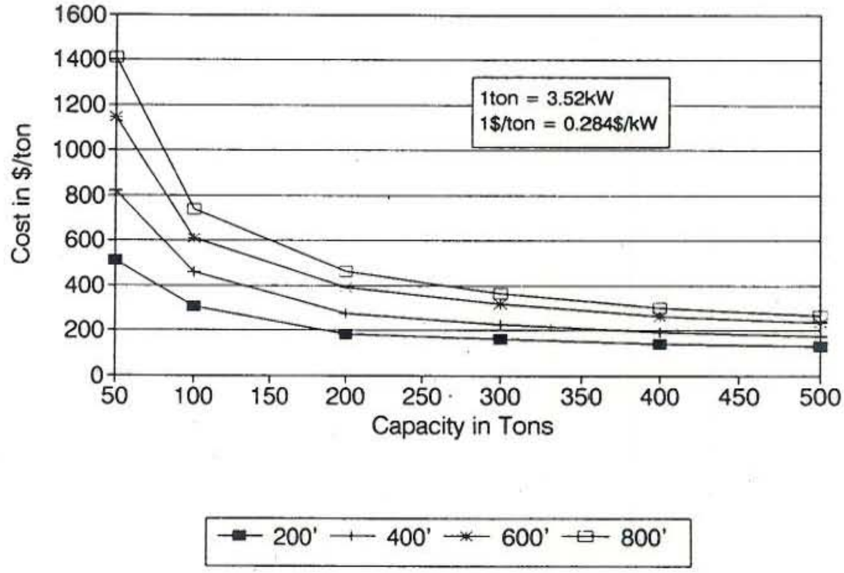

FIGURE 5.19: Groundwater system cost
(1 well - 60°F (15.5°C) water)

Results

Figure 5.19 presents the results for the 60°F (16°C) groundwater case assuming the use of a single production/ injection well pair to serve the system. The four curves shown indicate cost (in $/ton) for four different ground-water well depths: 200, 400, 600 and 800 ft (60, 120, 180 and 240 m). In all cases, the values shown include costs for the production wells, well flow testing, production well pump, pump variable-speed drive, buried piping for transport of the groundwater to the building, heat exchanger to isolate the groundwater from the building loop, heat exchanger controls, injection well, injection well flow testing, and a 15% contingency factor. As indicated, the depth requirement for the wells has a substantial impact upon the installed cost. In addition, the unit cost for small systems (50-100 tons [176-352 kW]) is often higher by a factor of 3 compared to costs for larger systems (300-500 tons [1055-1758 kW]).

For ground-coupled systems, actual project costs rather than calculations were used. Costs for these systems are a function of two values: the number of feet of borehole necessary per ton of heat rejection, and the cost per foot for completing the vertical bore and installing the piping. To arrive at a cost per ton, a value of US $5 per foot (US $16/m) of bore has been used. Although some recent projects have been the beneficiary of cost as low as US $3.75 per ft (US $12/m), and one as low as US $3 per ft (US $10/m), many areas of the country are still reporting costs of as much as US $15 per ft (US $50/m). Cost estimates for this comparison are based on a line cost of US $5 per ft and 200 ft/ton = US $1000/ton.

Hybrid systems include both a ground loop and a cooling tower. The ground loop is sized to meet the heating load and, it along with the tower, is used to meet the cooling heat rejection load. As a result, hybrid system costs are a combination of ground loop costs and cooling tower costs. Using the US $5 per ft (US $16/m) value for the hybrid ground loop portion and vendor quotes for the cooling tower, Figure 5.20 shows the cost per ton for the hybrid system based on 60°F (16°C) soil temperature. The four curves shown for the hybrid system reflect costs for different ratios of heating loop length versus cooling loop length. As indicated, hybrid systems enjoy more favorable economics as the heating ground loop length decreases as percentage of the cooling ground loop length requirement. This is because the cost per ton of the cooling tower is less than the cost per ton of the ground loop.

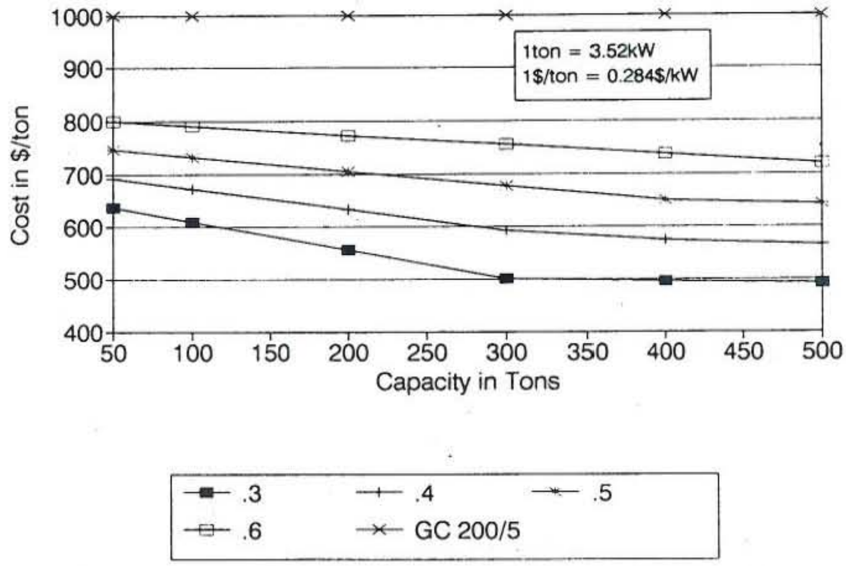

FIGURE 5.20: Hybrid system cost - 60°F (15.5°C) soil

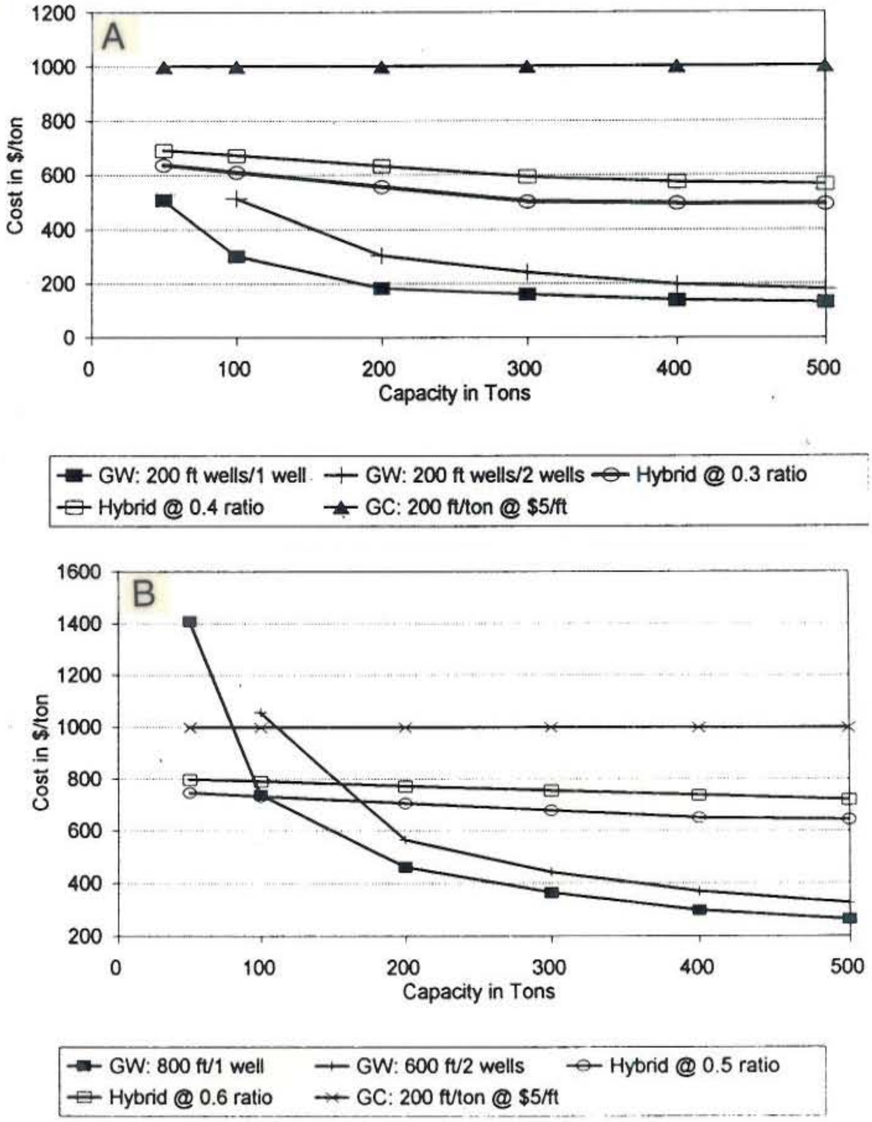

FIGURE 5.21: Ground-source system costs, 60°F water or soil, a) Low case; b) High case

Generally, the hybrid system is attractive in situations where ground loop costs per ton are high, and where the heating loop length requirement is low relative to the cooling loop length requirement.

Figure 5.21 presents a comparison of the three types of systems for 60°F (16°C) soil. The ground-coupled system cost line is based upon US $5 per ft (US $16/m) and 200 ft per ton (17.3 m/kW) (US $1000 per ton $285/kW). The two hybrid system curves in Figure 5.21a are based upon loop length ratios (heating ÷ cooling) of 0.30 and 0.40, whereas Figure 5.21b used loop ratio lengths 0.5 and 0.6. The former are the most favourable conditions for hybrid systems. The two groundwater curves are based upon 200 ft (60 m) wells and one production/ injection well pairs (upper curve). Again, these are the most favorable conditions calculated for groundwater systems in this paper. It is clear that, based on these conditions, the groundwater system enjoys substantial capital cost advantage over the remaining two systems over the entire range of capacity covered. The loop ratios in Figure 5.21b are the least favourable for the hybrid system.

As indicated (Figure 5.21b) at system capacities of 100-175 tons (350-610 kW) and above, the groundwater system has the capital cost advantage over hybrid and ground-coupled systems. Below this range, the hybrid system is the most attractive. It is only under conditions of less than 100 tons (350 kW) with well depths of 800 ft (240 m), that the groundwater system capital cost exceeds that of the ground-coupled system.

This discussion addresses only system capital cost. In the process of system selection, other issues should be considered as well. These would include operating costs such as electricity for pumps and fans, water treatment costs (tower), and regulatory issues with respect to groundwater. As a result, system capital cost provides only a portion of the information required for informed decision making.

5.10 GROUND-COUPLED HEAT EXCHANGER DESIGN (Braud et al., 1988)

5.10.1 Heat exchanger configurations

Concentric Pipes. The concentric pipe heat exchanger consists of a closed pipe casing with an inner pipe for return flow, Figure 5.22. The pipes are at the axis of an earth cylinder of radius equal to the radius of thermal influence of the heat effect. Heat transfer between the circulating fluid, as it flows down the annual space, and the surrounding earth is the useful heat transfer. As the fluid returns up the inner pipe, it experiences heat gain due to heat flow across the inner pipe wall, and crossover heat flow is detrimental to the heat exchange process. It can be reduced to practical low values by proper selection of material for the inner pipe.

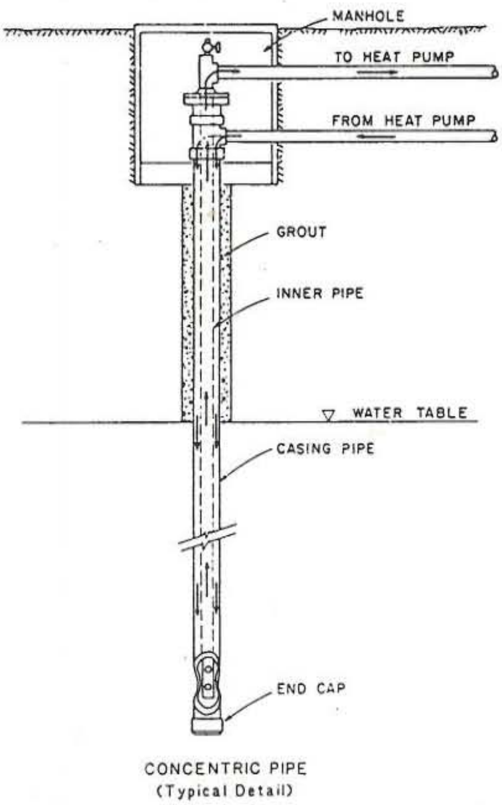

FIGURE 5.22: Concentric pipe configuration for vertical heat exchanger

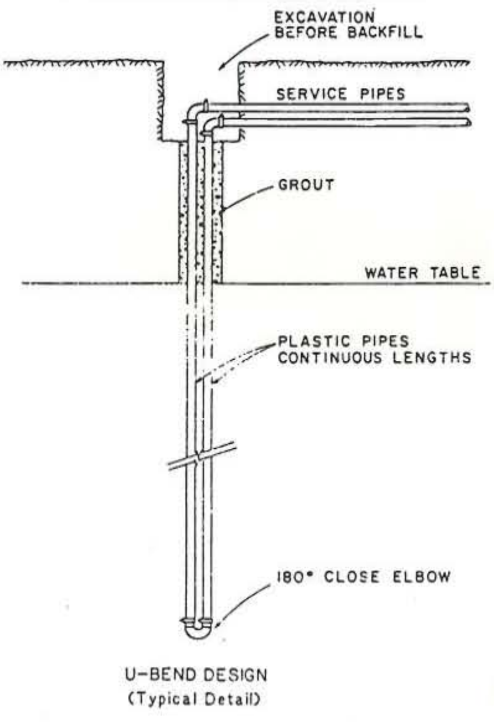

FIGURE 5.23: Two pipe side-by-side configuration for vertical heat exchanger

U-Bend. The U-bend configuration consists of two pipes side-by-side and connected to each other with a 180° fitting at the bottom (Figure 5.23). Fluid flows down one pipe and returns up the other. As the fluid descends, the temperature difference in the fluid and the earth mass causes heat flow. Because of temperature differences in the two fluid streams, there is also some deterious crossover heat flow. In this configuration, there are two walls of pipe material to impede the crossover heat flow; but, the presence of each pipe interferes with the heat loss to earth of the other.

Equation for Heat Flow. The equation for rate of heat transfer from the fluid in the heat exchanger to the earth mass is

$$\frac{Q}{L} = U \times \Delta T \tag{1}$$

where

- Q = Rate of heat transfer, Btu/hr (W) for the whole heat exchanger length,
- L = Length of the heat exchanger, m (ft),
- U = Conductance rate for heat transfer from the circulating fluid to the earth, Btu/hr °F (temperature difference per ft of length) (W/°C/m) for the operating conditions,
- ΔT = $(T_2 + T_1)/2 - T_o$, the difference in average fluid temperature in the pipes $(T_2 + T_1)/2$, and the earth temperature T,
- T_2 = Fluid exit temperature, °F (°C),
- T_1 = Fluid entry temperature, °F (°C),
- T_0 = Earth temperature, °F (°C).

As given in many heat transfer texts, the conductance term U for heat flow from fluid in the heat exchanger to the earth can be estimated with the conductance coefficient for composite cylinders. The impedance to heat flow is caused by the thermal resistance of the pipe wall and the soil cylinder around the casing. Fluid surface resistance films are small relative to the other terms and are encompassed in the two resistance terms. We can express U as

$$U = \frac{2\pi}{\textit{Soil resistance} + \textit{Pipe resistance}} = \frac{2\pi}{R_s + R_p} \tag{2}$$

The thermal influence of the pipe walls which separate the two fluid streams affects the temperature change in the fluid passing through the heat exchanger for given operating conditions, and it is manifest in the magnitude of the U-value as defined first equation. The casing wall resistance R_p can be calculated with sufficient accuracy from handbook values. The earth thermal resistance term R_s values can be solved for in the second equation. In fact, the purpose of field testing is to quantity the R_s value under different operating conditions and heat exchanger designs. Field testing was thus performed for the two heat exchanger configerations in Louisiana.

5.10.2 Results

Earth Conductance. The rate at which the earth would absorb heat was relatively high at the beginning of a run and declined as time went on. With on-off injection of heat, the instantaneous conductance values were always higher than with constant-on rate as shown in Figure 5.24, where conductance values for 25%, 50% and 100% on-time are given. Asymptote values are the best estimates of the conductance (U-value) derived from regression curve fits to the test data, Figure 5.25. The constant-on

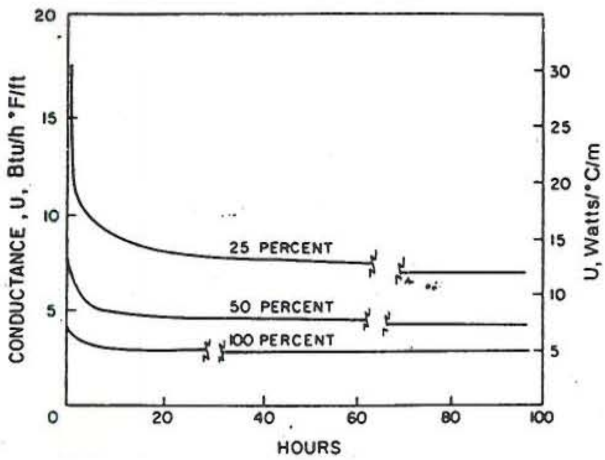

Figure 5.24: Typical runs of heat injection to earth with three duty cycle

conductance rate (i.e., 100% value) (continuous heat injection for 48 or more hours) was found to be U = 4.86 W/°C/m (2.81 Btu/h°F/ft of pipe length); for 50% on time, the effective U-value was 7.51 W/°C/m (4.34); and for 25% duty cycle, U was 11.88 (6.87), see Table 5.4. By use of Equation 2, the earth resistance term R_s was also calculated, Table 5.4. With the steel casing, the thermal resistance of the steel was negligible relative to the surrounding earth resistance term ($R_p \cong 0$).

Most of the runs were made with SCH 40 PVC inner return pipe. Runs with thin wall SDR 26 inner pipe exhibited less temperature drop of the circulating water. This effect was manifest in effective conductance values of U = 4.24 W/°C/m (2.45 Btu/h °F/ft) in 100% duty cycle. Wall thickness increased the heat transfer rate by only 5% over SCH 40.

Measured comparable heat exchange rates to earth with a steel casing with U-values range from 5.19 W/°C/m (3.0 Btu/h °F/ft) to 3.46 (2.0).

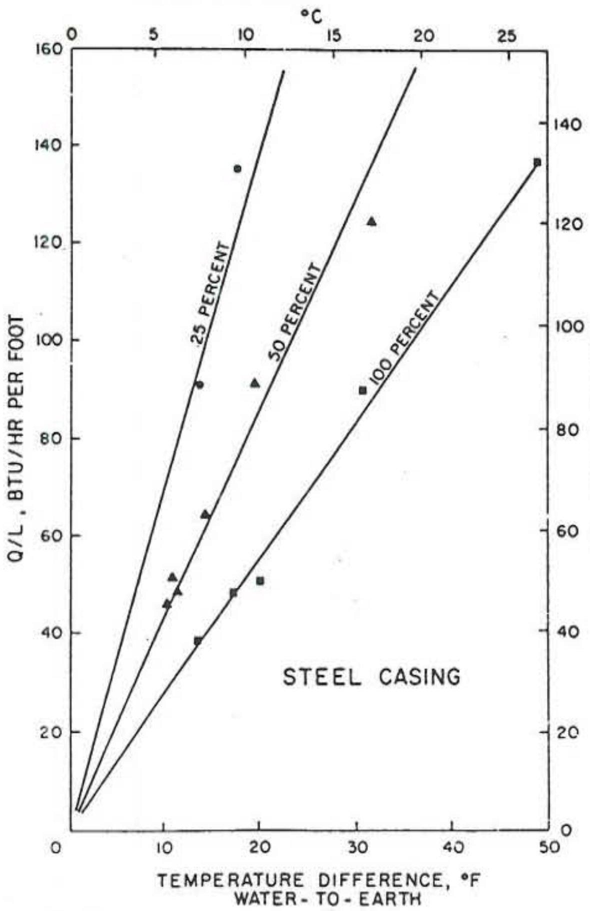

FIGURE 5.25: Earth absorption rate, Q/L plotted vs. average water to earth temperature difference for 25, 50 and 100% run time (duty cycle)

PVC Pipe Casing. Present day pipe costs dictate that plastic pipe is cheaper pipe than metal. Besides, PVC is widely used for water wells. Earth heat transfer rates with PVC plastic pipe can be calculated with second equation, the earth resistance term, R_s, Table 5.4, and the known thermal conductivity and wall thickness for PVC pipe. The U-value is given in Table 5.5.

TABLE 5.4: Earth heat exchange rate with steel casing and SCH 40 PVC inner pipe

Percent run time	100	50	25
U: W/°C/m	4.86	7.51	11.88
Btu/h/ °F/ft	2.81	4.34	6.87
Earth resistance R_s: C m/W	1.29	0.889	9.526
h °F ft /Btu	2.24	1.45	0.91

TABLE 5.5: Heat exchanger parameters for PVC casing and inner pipe

Percent run time	100	50	25
U: W/°C/m	3.58	4.85	6.37
Btu/h/ °F/ft	2.07	2.80	3.68
PVC pipe resistance R_p: C m/W	0.461	0.461	0.461
h °F ft /Btu	0.797	0.797	0.797

Thin wall PVC casing and SCH 40 PVC inner pipe are the most cost-effective combination
A residential heat pump using a PVC concentric pipe heat exchanger was monitored with good agreement to the experimental values. U-values ranged from 2.9 to 5.1 (2.0 to 3.4) depending on run-time for the particular test day in June and July, 1981. Heating mode operation during severe cold weather in January, 1982, caused circulation water to drop to a minimum of 15°C (58°F). This value is a safe temperature and indicates that circulation water does not need antifreeze protection in South Louisiana.

Conductance values in Table 5.4 agree with other field tests of vertical earth-coupled heat exchangers. Bose, et al. (1980) determined the long-term conductance to earth in a 12.7 cm (5 in.) PVC casing with 3.2 cm (1 ¼ in.) SCH 40 PVC inner pipe. The U-value was 2.95 (1.71).

U-Bend. Twelve runs of heat injection were made with the 81 m (265 ft) polyethylene pipe U-bend heat exchanger. Values of earth conductance are given in Table 5.6.

These values are close to those for PVC pipe except for 25% duty cycle. Because the earth resistance to heat flow is so much greater than the pipe wall resistance, the two designs (concentric pipe vs. U-bend) provide about the same performance.

TABLE 5.6: Conductance to earth with polyethylene U-bend heat exchanger

Percent run time	100	50	25
U: W/°C/m	3.46	4.71	11.60
Btu/h/ °F/ft	2.0	2.72	6.70

5.10.3 Heat exchanger size (length)

Besides the energy parameters for the heat pump, one must know the highest supply water temperature acceptable for cooling mode and the lowest temperature acceptable for heating mode. These values along with local earth temperature establish the design temperatures for sizing an earth-coupled heat exchanger.

Example: Find the heat exchanger length needed for a heat pump with 7,032 W (24,000 Btu/h) cooling capacity at 35°C (95°F) in an area with earth temperature T = 21°C (70°F). The heat pump duty cycle is estimated to be 50% run time during warmest summer days. Manufacturer specifications give a high temperature limit of 35°C (95°F) for entering water. The heat pump discharge water will be 5.6°C (10°F) warmer than entry. Total heat rejection of the heat pump = 4,747 W (16,200 Btu/hr) per ton or 9,493 W (32,400 Btu/hr). In the heating mode, the heat pump has a heat of absorption value of 7,325 W (25,000 Btu/hr) at 7.2°C (45°F). Discharge water will be 3.23°C (6°F) cooler than entry. Low temperature limit for entering water is 7.2°C (45°F). Design with PVC pipe, concentric pipe configurations.

Solution:

Cooling mode:

1. Find the design water-to-earth temperature difference, ΔT:

$$\Delta T = \frac{35 + (35 + 5.6)}{2} - 21 = 16.8°C$$

$$\Delta T = \frac{95 + (95 + 10)}{2} - 70 = 30°F$$

2. In Table 5.5 read the effective conductance rate for PVC pipe casing with 50% duty cycle, U = 4.85 W/°C/m (2.80 Btu/h °F/ft).

3. Solve for L in the Equation 1:

$$L = \frac{9{,}483\ W}{4.85\ W/°C/m \times 16.8°C} = 117\ m$$

$$L = \frac{32{,}400\ Btu/h}{2.80\ Btu/h\ °F/ft \times 30\ °F} = 385\ ft$$

Heating mode:

1. Find design water-to-earth temperature difference, ΔT:

$$\Delta T = 21.1 - \frac{(7.2 - 3.3) + 7.2}{2} = 15.6\ °C$$

$$\Delta T = 70 - \frac{(45 - 6) + 45}{2} = 28\ °F$$

2. Assume that the heat pump will run 12 h in 24 h in coldest weather. The 50% duty cycle value for U then applies.

U = 4.85 W/°C/m (2.80 Btu/h °F/ft)

3. Solve for L in the Equation 1:

$$L = \frac{7{,}325\ W}{4.85\ W/\ °C/m \times 15.6\ °C} = 97\ m$$

$$L = \frac{25{,}000\ Btu/h}{2.80(Btu/h\ °F/ft) \times 28\ °F} = 318\ ft$$

As this example shows, the heat exchanger length needed is the larger value which is about 117 m (385 ft) of PVC casing for the heat pump in cooling mode.

5.11 ADDITIONAL INFORMATION ON GROUND-COUPLED INSTALLATIONS

- Ground-loop installed cost currently represents a large portion of system cost. The initial key to simulating market interest in GSHPs is bringing down those costs, according to an Electric Power

Research Institute (EPRI) report, "Ground-Source and Hydronic Heat Pump Market Study." The report provides detailed information on the residential GSHP market, emphasizing the requirements for performance improvement and cost reduction.

- The size and installed cost of the ground loop depends on the thermal conductivity of surrounding soil or rock. See Figure 5.26, in which thermal resistivity (the inverse of conductivity) is plotted versus moisture content with soil type as the parameter. Thermal conductivity is a sensitive function of the type of soil or rock and its moisture content. The ground-loop designer must be able to identify the soil or rock, choose a design value for the minimum moisture content, and arrive at a design value for thermal conductivity. Until recently, there was no rational approach to this task. EPRI research shows that four soil texture classes (sand, silt, clay and loam) may be used to characterize soils and provides design values of thermal conductivity for each. The range of values for identified rock types has also been developed. This information and its rationale are presented in a report, "Soil and Rock Classification According to Thermal Conductivity."

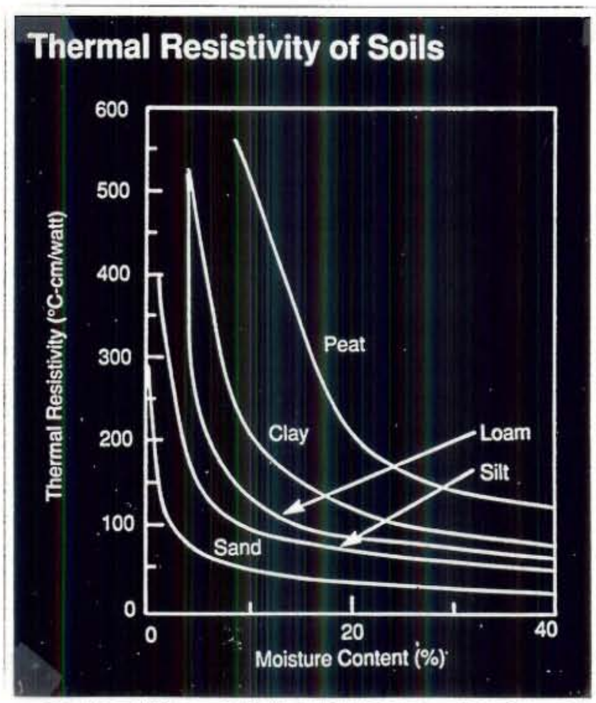

FIGURE 5.26: Ground loops function more efficiently in wetter soils where thermal resistivity is lower

- A companion volume, "Soil and Rock Classification for the Design of Ground-Coupled Heat Pump System: Field Manual," published in cooperation with the National Rural Electric Cooperative Association (NRECA), Oklahoma State University (OSU), and the International Ground-Source Heat Pump Association (IGSHPA), describes simple field procedures for identifying soil and rock types, and provides appropriate thermal property design values. The soil and rock field manual is scheduled for incorporation in future editions of the authoritative, "Closed-Loop/Ground-Source Heat Pump Systems Installation Guide," published by NRECA, OSU and IGSHPA.

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LECTURE 6

INDUSTRIAL APPLICATIONS

6.1 INTRODUCTION

Industrial uses of geothermal energy can involve many applications over a wide range of temperatures. Categories of energy uses include process steam, direct heat and electricity generation. Geothermal energy can contribute substantially to these industrial needs, as illustrated by the Tasman pulp and paper plant in New Zealand where geothermal steam provides 55 % of the energy for all three categories listed. The most important consideration is matching the temperature of an available geothermal resource with a compatible industrial process. In some situations, low-temperature resources can be boosted through heat pumps and temperature amplifier to satisfy higher temperature demands.

Potential industrial applications of geothermal energy include process heating, evaporation, drying, distillation, refrigeration (absorption), sterilization, washing, de-icing (mining operations), salt and chemical extraction. A summary of some of the various uses is shown in Figure 6.1 (Lindal, 1973).

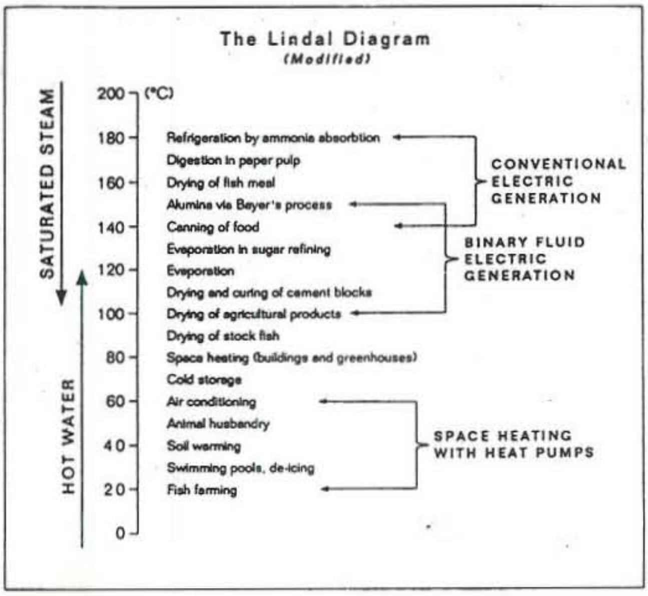

FIGURE 6.1: The Lindal diagram on typical fluid temperatures for direct applications of geothermal resources (modified from Lindal, 1973)

While there are many potential industrial uses of geothermal energy, the number of worldwide applications is relatively small. They do, however, represent a fairly wide range of uses, from fish, grain, mineral and timber drying, to pulp and paper processing, and to chemical recovery. The oldest known use of geothermal energy for industrial processing is in Italy. This early industrial application included the use by the Etruscans of boric acid deposited by the steam and hot water at Larderello, Italy. They used the deposits to make enamels to decorate their vases. Commercial extraction of the acid started in 1818, and by 1835, nine factories had been constructed in the region. Originally, the boric acid was obtained by boiling off the geothermal water using firewood as a heat source. From 1827 onwards, geothermal steam was used as the energy source. With increase in production, growth in trade, and refinement of the process, a wide range of boron and ammonium compounds were produced in the early 1900s. This process continued until World War II, where a total of 6,500 tonnes had been produced. After the war, the plant was put into operation again, only this time the raw product was imported from Turkey, and geothermal steam used as the drying source. Approximately 30 tonnes of steam per hour are used in the process.

Other well-known examples of industrial uses of geothermal energy are in New Zealand. At Kawerau, the Tasman Pulp and Paper Company uses high-temperature steam for timber drying, black liquor evaporation, pulp and paper drying, and electric power generation. Approximately 400,000 tonnes of newsprint, 200,000 tonnes of kraft pulp and 190,000 m^3 of timber are produced annually. Geothermal energy produces 26% of the process steam requirements and 6% of the mill's electric load (Hotson, 1995). Six wells are presently being used to provide 180 tonnes/hr of steam (100-125 MW_t). One of the largest wells in the world is located here, estimated to produce over 25 MW of thermal energy (170 tonnes/hr). The total investment cost for geothermal is US$6.8 million, the majority of which is for well development. This amounts to approximately US$70 per kWt and will reduce the price of energy to 70% that of conventional fuels for an annual savings of US$1.3 million. The annual maintenance costs are 2% of the capital cost. Tasman is developing a process to extract silica from the bore water to be used in place of imported calcined clay as a filler in the production of newsprint.

In the vicinity of the Broadlands field of New Zealand are several unique applications of geothermal energy (Lund, 1976a). At the Lands Survey Nursery in Taupo, greenhouses are heated by geothermal steam, and soil is sterilized (pasteurized) at 6°C to kill insects, fungus, worms and some bacteria. At Lake Rotokaua, an estimated 20 million tonnes of sulfur lie within 60 meters of the surface having a purity of up to 80 %. Originally, the sulfur was extracted by the Frasch process using geothermal steam injected in four boreholes. Presently, they are strip mining the low-grade surface deposits and using geothermal steam to extract the sulfur. The sulfur is then combined with cold water in a slurry and shipped by tanker truck for use in fertilizer production. At Broadlands, adjacent to the Ohaaki geothermal power stations, the Taupo Lucerne Limited (NZ) uses geothermal steam and hot water as the heat source for drying of lucerne (alfalfa) into "De-Hi" produced from the fibrous part of the plant, and "LPC" (lucerne protein concentrate) which is a high protein product produced from extracted juice (Pirrit and Dunstall, 1995). A number of contract growers provide feed stock for the plant, whereas several years ago it was a cooperative of 12 farms using different operation (Van de Wydeven and Freeston, 1979). The fibrous lucerne material is dried in an air stream heated by geothermal hot water which is passed through finned tubular type heat exchanger units. Initially, only dry separated geothermal steam was used in the plant at approximately 175°C, but in 1993 the system was converted by the installation of additional heating coils to use the hot water which remains after flash steam has been removed for generating electricity. Separated hot water is now used at 148°C. The system is steam purged to prevent silica build-up in the heat transfer coils. What was originally a pilot plant operation has been developed into full scale production plant producing approximately 3,000 tonnes per annum of dried product for the New Zealand and Australian markets. After crushing the plants to remove juice, the fibrous part of the crop (stem and leaves) are passed to the heated air transport drying system. This reduces the moisture content from an average of 68% to 10% by weight. The LPC is the dried extracted juice.This product is very high in protein (48% by weight) and is used in the egg industry as a feed supplement for chickens. Dried timber is also produced on the site in a separate batch operation. Up to 1000 m^3 of dried fenceposts, poles and sawn timber, mainly Radiata Pine, are produced each month.

In northern Iceland, a diatomaceous earth drying plant uses high-temperature steam to remove about 80% of the moisture from a diatomaceous slurry dredged from nearby Lake Myvatn. Hot water keeps the beneficiated diatomite slurry reservoir from freezing during operation in winter. Steam is used for keeping the slurry tank from freezing, for space and fuel oil heating and, for the most part, in drying the diatomite slurry in steam tubes. Figure 6.2 illustrates the flow

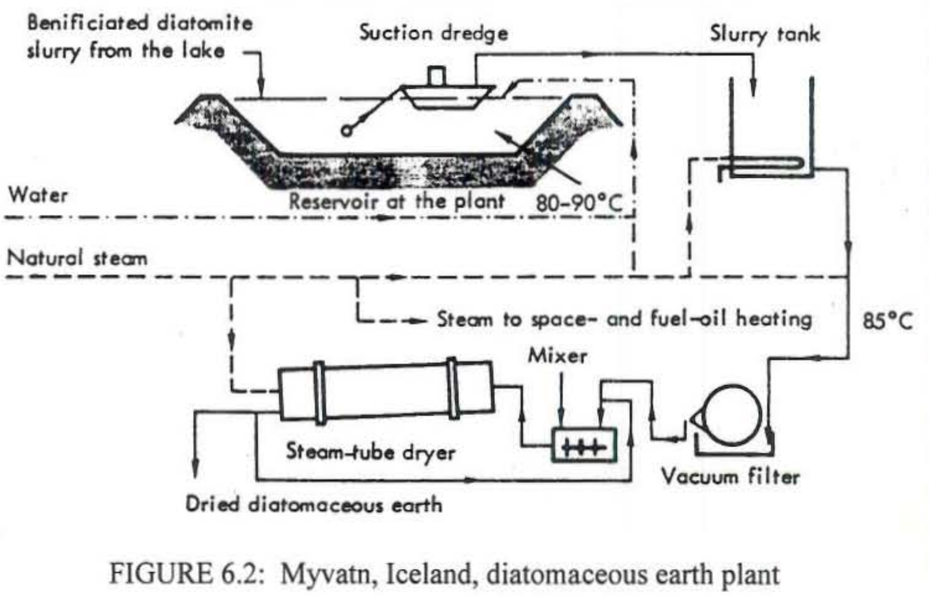

FIGURE 6.2: Myvatn, Iceland, diatomaceous earth plant

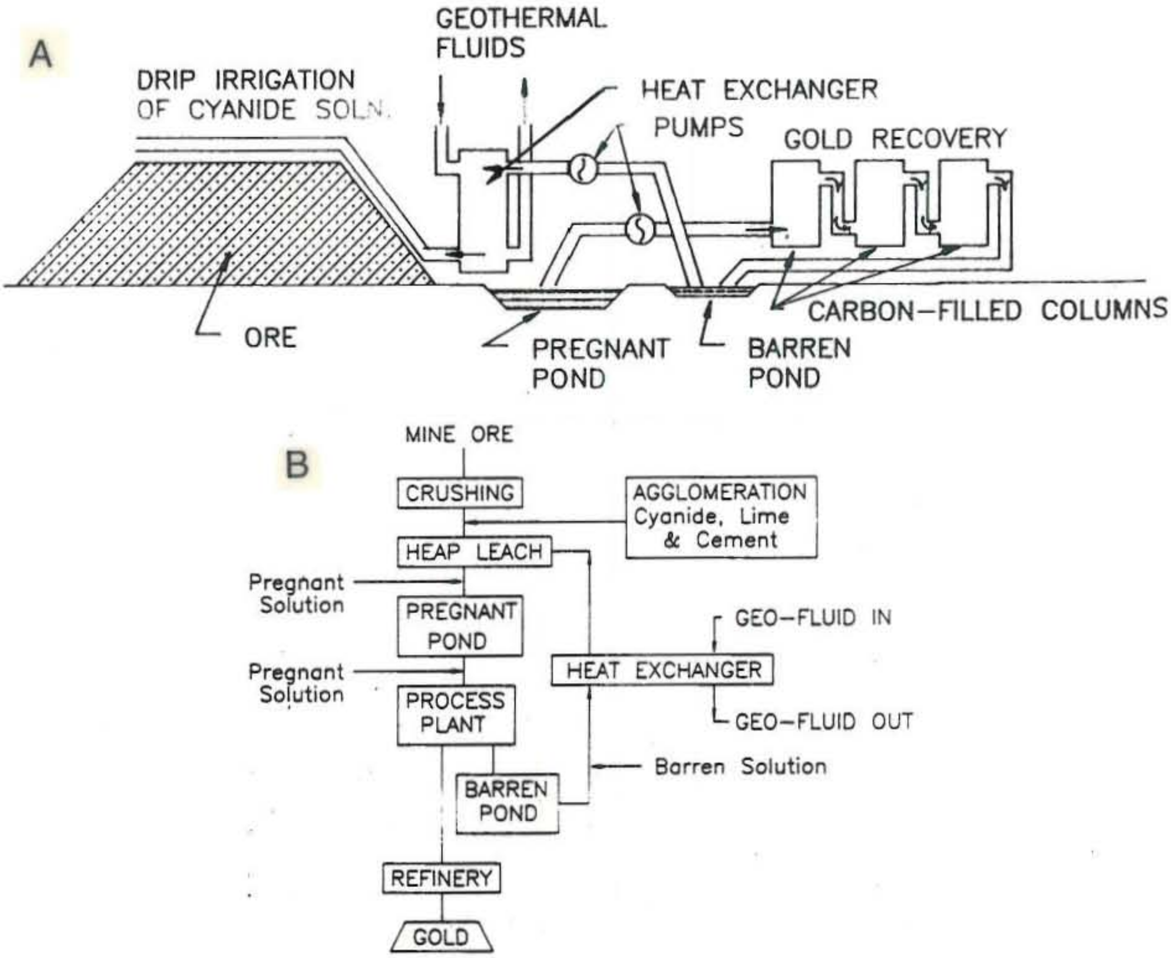

FIGURE 6.3: a) Idealized thermally enhanced heap leach (Trexler et al., 1990);
b) Heap leach process flow

diagram of the diatomaceous drying process (Howard, 1975). In addition, Iceland has developed several experimental projects for the drying of wool, fish, seaweed, hay and other grains (Lindal, 1995).

Heap leaching in Nevada, USA, uses geothermal water to enhance gold and silver recovery (Trexler et al., 1990). Heating the cyanide leach solution with geothermal energy provide for year-round operation and increases precious metal recovery (estimated at 5-17%, experimentally). Figure 6.3 illustrates a geothermally enhanced heap leaching process typical of central Nevada.

6.2 APPLICATIONS AND POTENTIAL USES

Direct utilization of geothermal energy for process heating, for the most part, utilizes known technology. Basically, hot water is hot water whether from a boiler or from the earth. The utilization of geothermal energy requires only straightforward engineering design rather than revolutionary advances and major scientific discoveries. The technology, reliability, economics and environmental acceptability have been demonstrated throughout the world.

It must be remembered that each resource is different and the systems must be designed accordingly. Granted, there are problems with corrosion and scaling, generally confined to the higher temperature resources; but, most of these problems can be surmounted by materials selection and proper engineering designs. For some resources, standard engineering materials can be used if particular attention is given

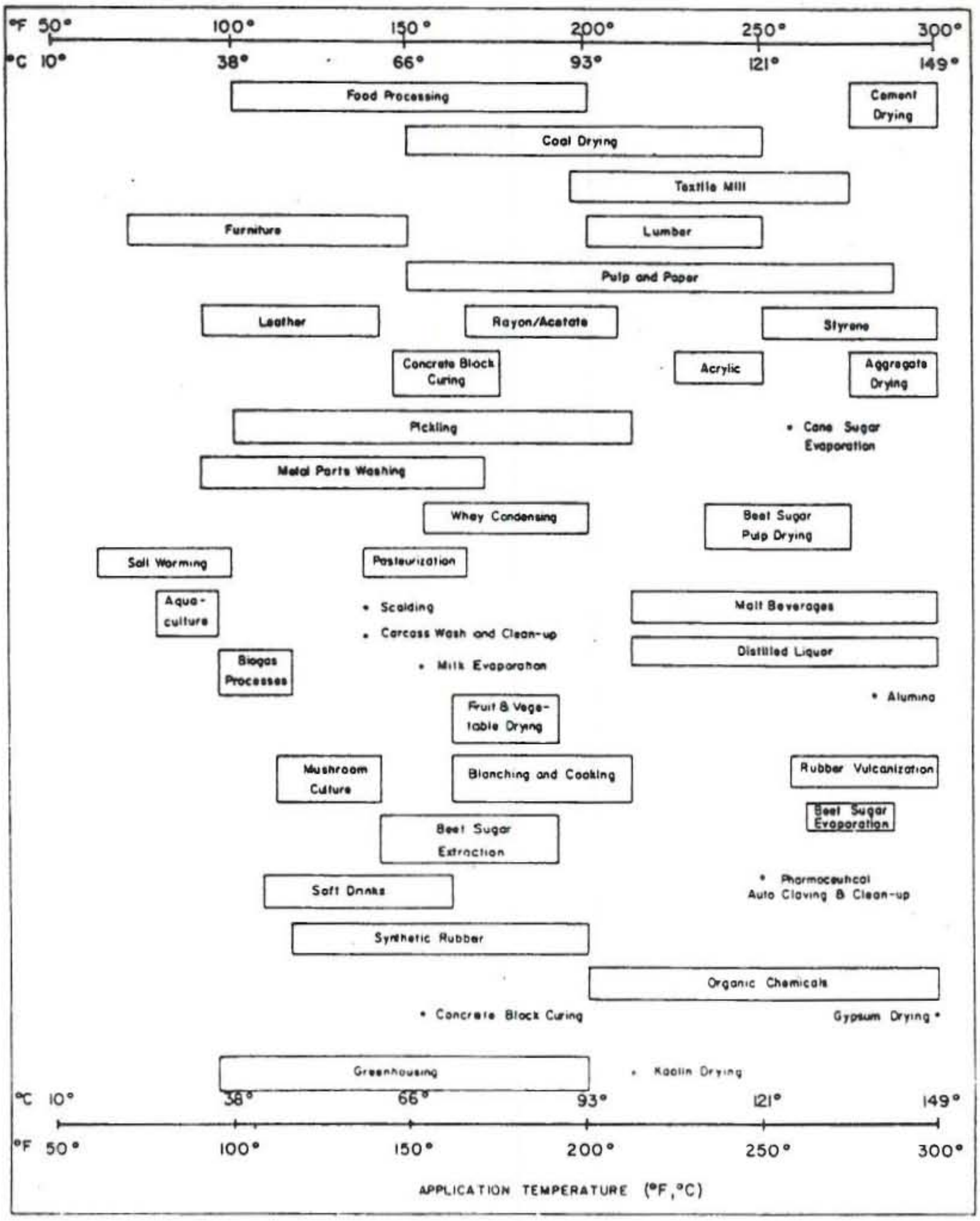

FIGURE 6.4: Application temperature range for some industrial processes and agricultural applications

to the exclusion and/or removal of atmospheric and geothermally generated gases. For others, economical designs are possible which limit geothermal water to a small portion of the overall system by utilizing highly efficient heat exchangers and corrosion resistant materials in the primary side of the system.

Industrial processing typically requires the highest temperatures, using both steam and super-heated water. Temperatures up to 150°C are normally desired; however, lower temperatures can be used in some cases, especially for drying of various agricultural products. Some experimental work is being performed with grain, hay, tobacco, fruit and paprika drying. In these cases, hot water supplies heat to forced-air heat exchangers and 49-60°C air is blown over the product to be dried (Lienau and Lund, 1974). A graphic representation of application temperatures is shown in Figure 6.4. Table 6.1 gives the percentage of various process heat requirements met by various temperature ranges (Lund et al., 1980).

Reports by Lindal (1973), Reistad (1975), Howard (1975), and Lienau (1991) survey industrial applications and the potential for geothermal use in a number of the industries. The following is an outline of the basic processes and several of the more recently considered applications.

TABLE 6.1: Industrial process heat requirements

Industry	40-60°C	60-80°C	80-100°C	100-120°C	120-140°C	140-160°C	160-180°C	180-200°C	200°C	250°C
Meat packing	NA	99%	100%							
Prepared meats	NA	46.2%	61.5%	100%						
Natural cheese	23%	100%								
Fluid milk	NA	NA	100%							
Canned fruit and vegetables	NA	NA	22.7%	67.6%	100%					
Dehydrated fruits and veget.	NA	100%								
Potato dehydration										
granules	NA	19.9%	40%	53%					100%	
flakes	NA	19.9%	40%	53%				100%		
Frozen fruit and vegetables	NA	NA	30%	100%						
Wet corn milling	21.5%			36.4%	46%		84.1%		100%	
Prepared feeds										
pellet conditioning	NA	NA	100%							
alfalfa drying	NA	NA	NA	NA	NA	NA	NA	NA	100%	
Beet sugar	NA	7.4%	22.4%		95.4%					100%
Soft drinks	60.9%	100%								
Sawmills and planing mills	NA	NA	NA	NA	NA	100%				
Alumina	NA	NA	NA	NA	76.2%					100%
Soaps	NA	NA	0.6%						100%	
Detergents	NA	NA	52.2%				99.9%		100%	
Concrete blocks										
low pressure	NA	100%								
autoclaving	NA	NA	NA	NA	NA	NA	NA	100%		
Ready mix	100%									

6.2.1 Basic processes

In industrial applications, thermal energy in the temperature range being considered here (up to 150°C) is used in the basic processes of:

- Preheating
- Cooking, blanching, peeling
- Sterilizing
- Drying
- Washing
- Evaporating
- Distilling and separating
- Refrigeration

6.2.2 Preheating

Geothermal energy can be effectively used to preheat boiler and other process-feed water in a wide range of industries. Many manufacturing industries utilize boilers distributing steam throughout the plants. For a variety of reasons, much of the condensate is not returned. This imposes a considerable load on the boiler for feed-water heating of incoming water at typically 10-16°C up to the temperature at which it is introduced into the boiler, typically 90-150°C, depending on the system. The geothermal resource can often be used to off-load the boiler of some or all of this preheating load.

A wide variety of industries use, for various processes, large quantities of feed water which can be preheated or heated geothermally to the use temperature. Some of these applications also use heat-reclaim methods which must be analyzed when evaluating the potential for geothermal use.

6.2.3 Washing

Large amounts of low-temperature energy (35-90°C) is consumed in several industries for washing and cleanup. One principal consumer is food processing, with major uses in meat packing for scalding, carcass wash and cleanup (60°C), in soft drink container and returnable bottle washing (77°C), in poultry dressing as well as canning and other food processes. Textile industry finishing plants are another large consumer of wash water at 90°C. Smaller amounts are used in plastics (88-93°C) and leather (50°C). Most of these are consumptive uses.

Sizable amounts of hot water and other hot fluids at temperatures under 90°C are used in the several metal-fabricating industries (fabricated metal products, machinery and transportation equipment) for parts degreasing, bonderizing and washing processes. Most of these are non-consumptive uses with a 6-14°C range in the fluid and reheating to the use temperature.

6.2.4 Peeling and blanching

Many food-processing operations require produce peeling. In the typical peeling operation, the produce is introduced into a hot bath (which may be caustic) and the skin or outer layer, after softening, is mechanically scrubbed or washed off. Peeling equipment is usually a continuous-flow type in which the steam or hot water is applied directly to the produce stream or indirectly by heating a produce bath. In most instances, produce contact time is short. Blanching operations are similar to peeling.

Produce is usually introduced into a blancher to inhibit enzyme action, provide produce coating, or for cooking. Blanching may be either a continuous or batch operation. Typical blanching fluids require closely controlled properties. Thus, it is unlikely that geothermal fluids could be used directly in blanchers and peelers because of the water quality. Geothermal fluids could, however, provide the energy through heat exchangers.

The temperature range for most of the peeling and blanching systems is 77-104°C. These heating requirements are readily adaptable to geothermal resources.

6.2.5 Evaporation and distillation

Evaporators and distillators are routinely found in many processing plants to aid in concentrating a product or separating products by distillation. Most frequently, the evaporator will operate as a batch process in which a quantity of product is introduced and maintained at some given temperature for a period of time. The source temperature requirements vary with the product being evaporated. However, in a majority of agricultural processes, water is being driven off, and in these cases, operating temperatures of 80-120°C are typical. In some circumstances, the evaporators operate at reduced pressures which decrease temperature needs and improve product quality. Evaporators are commonly found in sugar processing, mint distilling and organic liquor processes. Evaporators, depending upon temperature and flow-rate requirements, can be readily adapted to geothermal energy as the primary heat source. The energy can be transferred through secondary heat exchangers to the working fluids or, in some instances, used directly at the evaporator, depending upon existing plant designs or adaptations to new plant expansions.

6.2.6 Sterilizing

Sterilizers are used extensively in a wide range of industries and include applications such as equipment sterilization in the meat-packing and food-processing industries, and sterilization for the canning and bottling industry. Most sterilizers operate at temperatures of 105-120°C and would utilize geothermal energy with the use of heat exchangers to heat the potable sterilizer water. Many sterilizers operate in a continuous mode. Equipment wash down and sterilization, however, may occur periodically or at shift changes.

6.2.7 Drying

Many industries utilize heat at temperatures under 150°C for evaporating water or to dry the product, material or part. The largest consumers are pulp and paper drying, and textile product drying - mostly in the 90-150°C range.

Other large consumers of energy for drying are in beet-pulp drying, malt-beverage and distilled-liquor grain drying, and cement drying. Additional large energy consumers in the drying application area are onions (discussed later in this report), grain, lumber kiln, plywood and veneer drying. Smaller consuming industries having drying applications include coal, sugar, furniture, rubber, leather, copper concentrate, potash, soybean meal, tobacco, pharmaceutical tablet and capsule, explosives and paving-aggregate drying.

6.2.8 Refrigeration

Cooling can be accomplished from geothermal energy through lithium-bromide and ammonia absorption refrigeration systems.

The lithium-bromide system is the most common because it has water as the refrigerant; however, it is limited to cooling above the freezing point of water and has as its major application the delivery of chilled water for comfort or process cooling and dehumidification. These units may be either one- or two-stage. The two-stage units require higher temperatures (about 163°C), but also have a higher COP (cooling output/source energy input), being about 1 to 1.1. The single-stage units are currently receiving substantial research emphasis in regard to use with solar energy and can be driven with hot water a temperatures somewhat below 90°C and will typically have a COP of 0.65.

For geothermally-driven refrigeration at temperatures below the freezing point of water, the ammonia absorption system must be considered. These can operate down to about -40°C evaporator temperature. However, these systems are normally only applied in very large tonnage capacities (100 tons [350 kW] and above) and have seen limited use. For the lower temperature refrigeration, the driving temperature must be at or above about 120°C for a reasonable performance.

At Oregon Institute of Technology, geothermal hot water provides for the heating needs of the campus

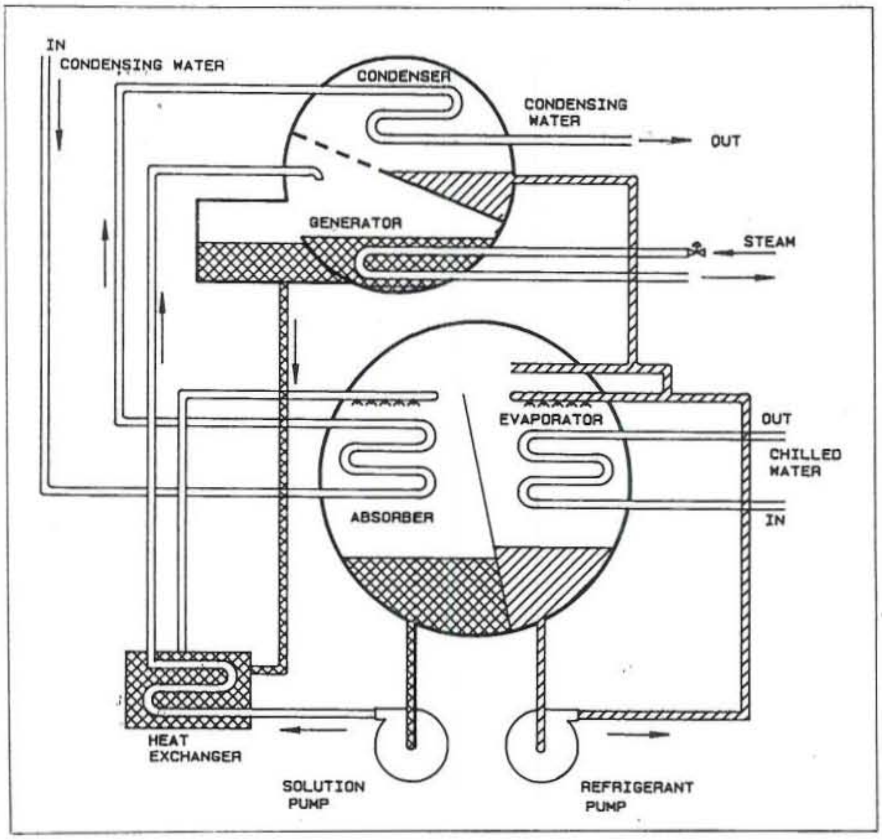

FIGURE 6.5: Lithium-bromide absorption refrigeration

and in conjunction with an absorption chiller installed in 1980, supplies a portion of the cooling needs as well (Lienau, 1996).

The chiller, which works on the same principle as a gas-fired refrigerator supplies the base-load cooling needs of approximately 26,000 m^2 of buildings area. To accomplish this task, it requires 38 L/s of 88°C geothermal fluid. In comparison to a standard application, this machine produces only about 50% of its cooling capacity. This is due to the fact that absorption chillers are designed to operate on a 116°C heat input. The OIT resource only produces 89°C fluid and as a result, the machine capacity is reduced.

Figure 6.5 gives a flow scheme for a typical lithium-bromide/water absorption cycle.

6.3 DETAILS OF SELECTED INDUSTRIAL APPLICATIONS

Three different industrial applications of geothermal energy will be described in detail. The three are milk pasteurization, vegetable dehydration and fruit drying. All of these processes are presently in operation in the world using geothermal energy: two in the U.S. and one in Mexico. In addition, research in the utilization of silica from waste geothermal brine will be discussed.

6.3.1 Milk Pasteurization (Lund, 1976b; Belcastro, 1979)

Medo-Bel Creamery, in Klamath Falls, Oregon, is the only creamery at least in the US known to have used geothermal heat in the milk pasteurization process. But another application has been reported in

Iceland (Thorhallsson, 1988) The geothermal well located at the corner of Spring and Esplanade Streets was first drilled in 1945 by the Lost River Dairy. The well was designed by a New York engineering firm to insure maximum heat at the wellhead with a minimum of pumping time. The facility is no longer in use.

A 233 m deep well was cased to 190 m with 20 cm diameter casing and to 149 m with 15 cm diameter casing. The original well had an artesian flow of around 119 L/s at the surface of 82°C water. Based on a recent profile (1974), the well water varied from 80°C at the surface to 98°C at a depth of 137 meters, with the artesian surface at one meter below the ground level. The geothermal hot water was pumped directly from the well to the building approximately 15 m away through an overhead line. This overhead line allowed easy maintenance and prevented freezing during cold weather since it was self-draining.

Rather than using downhole heat exchangers as is common in Klamath Falls, the water was used directly in air handling units in each room and in the plate-type pasteurizing heat exchanger (Figure 6.6). The used hot water was then emptied into the storm sewer where it was later used by industry in the south end of the town.

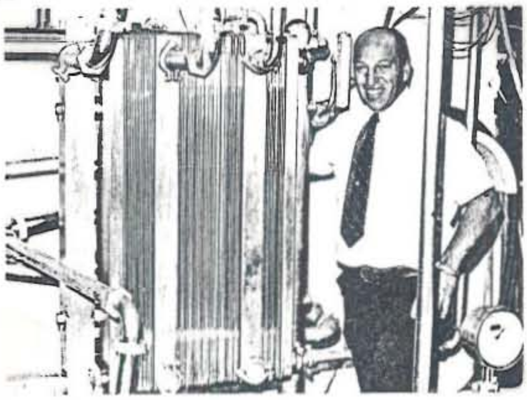

FIGURE 6.6: Owner, Elmer Belcastro, standing next to the plate heat exchanger (pasteurizer)

The pasteurization process involved pumping up to 6 L/s of geothermal fluid into the building and through a short-time pasteurizer (Cherry Burrell plate heat exchanger of stainless steel construction) (Figure 6.7). The geothermal water was pumped from the well at 87°C and passed through one section of the plate heat exchanger where the incoming milk was heated from 3°C to 71°C. The milk is then passed through the homogenizer and to the second section of the plate heat exchanger where the geothermal fluid heated the milk to a minimum temperature of 77°C for 15 seconds in the short-time pasteurizer. If the milk temperature dropped below 74°C, the short-time pasteurizer automatically recirculated the milk until the required exposure was obtained. Once the milk was properly pasteurized it was pumped back through the other side of the first section of the plate heat exchanger where it is cooled to 12°C. It was finally chilled to 3°C by cold water in the third section of the plate heat exchanger, where the milk went into the cartons with no chance of cook on. This insured both flavour and longer shelf life. As an added bonus, the outgoing heated milk was cooled somewhat by passing it by the incoming cold milk and the cold milk was in turn heated slightly by the outgoing milk. Milk was processed at a rate of 0.8 L/s, and a total of 225,000 kg were processed each month. Some steam was necessary in the process to operate equipment; thus, geothermal water was heated by natural gas to obtain the required temperature. Geothermal hot water was also used for other types of cleaning.

In addition to the milk pasteurizing, some batch pasteurizing of ice cream mix was carried out by geothermal heat. A 950 liter storage tank was used to mix geothermal hot water and process steam to a temperature of 121°C. This heat was then used to pasteurize the ice cream mix at 63°C for 30 minutes. This method was the original pasteurizing method used at the creamery.

The geothermal water had slightly over 800 ppm (mg/L) dissolved solids of which approximately half were sulfate, a quarter sodium and a tenth silica. The pH of the water was 8.8. Minimum corrosion was evident in the well, requiring the jet pump to be replaced only once in the 30 year period (1974). The original pump was rated at one hp (0.7kW) and a new pump was rated at 7 ½ hp (5.6 kW). The corrosion had also been minimum in the area heaters and did not affect the stainless steel plate heat exchanger. Corrosion was substantial in the pipelines.

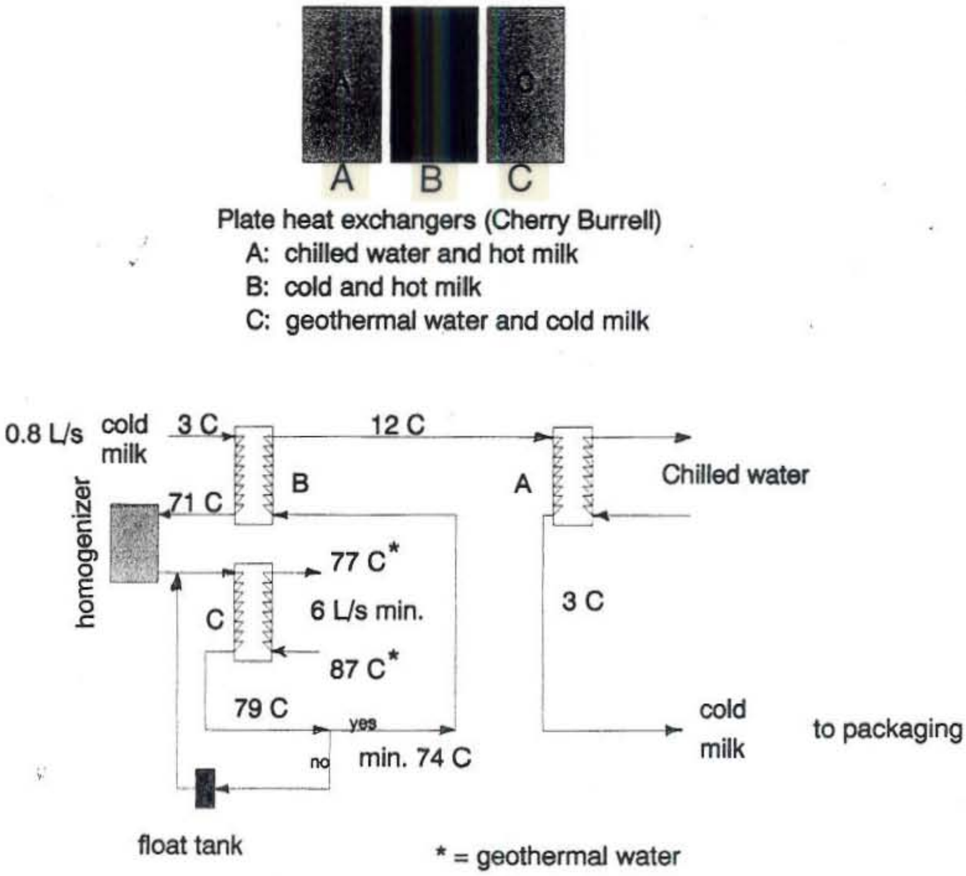

FIGURE 6.7: Medo-Bel milk pasteurization process

The annual operational cost of the system was negligible. However, the savings amounted to approximately $1,000 per month as compared to conventional energy costs. Geothermal hot water was also used to heat the 2,800 m^2 building, which amounted to a substantial savings during the winter months.

6.3.2 Onion dehydration

See attached papers in Appendix 6.1 on onion dehydration (Lund, 1995).

6.3.3 Fruit dehydration

See attached paper in Appendix 6.2 on pilot fruit driers for the Los Azufres geothermal field in Mexico (Lund and Rangel, 1995).

6.3.4 Use of silica waste

See attached paper in Appendix 6.3 on the use of silica waste from the Cerro Prieto geothermal field in Mexico (Lund and Boyd, 1996).

6.4 SPECIAL CONSIDERATIONS

In addition to the aspects discussed above, process-heating applications involve several additional factors that can seriously impact the design and feasibility of using the geothermal resource. This section considers a number of these factors.

6.4.1 Retrofit vs. new installations

In many of the large and complex industrial operations, most of the potential application in the very near future will be of a retrofit type. For these, the geothermal system design will be largely the supply of the hot fluid to the system or building boundary, and extensive internal equipment modifications will be essentially absent for several reasons: expense, process disruption and the noted agri-business practice of maintaining proprietary process secrecy.

New facilities offer the advantage of much greater potential geothermal heat applications: base loading levels can be established, equipment designs can be modified to accommodate the hot fluids (heat transfer surfaces, for example, could be enlarged to provide the same amount of heat from hot liquids as compared to, say, high-pressure steam), and all suitable plant aspects can be designed in view of the rapidly deteriorating fossil-fuel situation.

6.4.2 Applicability of heat pump

In a number of instances, the situation may arise where the geothermal fluid temperature is lower than the required application temperature and/or the flow rate of geothermal fluid is not sufficient to directly meet the needs of the application. In such circumstances, the use of a heat pump to allow additional energy to be extracted from the geothermal fluid (lowering the disposal temperature) and raise the thermal-energy output temperature may be desirable. At the present time, units are commercially available with output temperatures up to about 110°C. A combination of heat pump and heat exchanger(s) may prove beneficial, in various situations, to obtain a greater energy extraction (larger temperature drop) from the geothermal resource. The economic feasibility of such installations varies with the specifics of the resource and the application. However, two major considerations are that: (1) the temperature lift of the heat pump (for COP $\leq$ about 3) should be less than about 44-50°C (the smaller the lift, the better the feasibility), and (2) auxiliary energy, usually in the form of electricity, is required.

6.4.3 Direct and indirect application of geothermal fluids in processing

Several factors should be considered by the designer in using a geothermal fluid directly in a process stream. In most instances, use of the geothermal fluid directly will result in the elimination of additional heat exchangers, pumping and piping. However, the economic savings may be overshadowed by consideration for peaking, product contamination and environmental concerns.

Direct use may not be practical in many cases. If the process has or is required to have a standby or peaking capability provided by an auxiliary boiler, it may not permit use of the geothermal fluid in the boiler as feed water. In cases where the process loop has special water treatment requirements, introduction of geothermal water complicates such treatment and may prove uneconomical.

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APPENDIX 6.1

ONION DEHYDRATION

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ABSTRACT

Onion dehydration consists of a continuous operation, belt conveyor using fairly low-temperature hot air from 38 - 104°C (100 to 200°F). Typical processing plants will handle 4500 kg (10,000 pounds) of raw product per hour (single line), reducing the moisture from around 83% to 4% (680 to 820 kg - 1,500 to 1,800 pounds finished product). An example of a geothermal processing plant is Integrate Ingredients at Empire, Nevada, in the San Emidio Desert. A total of 6.3 million kg (14 million pounds) of dry product are produced annually: 60% onion and 40% garlic. A 130°C (266°F) well provide the necessary heat for the plant.

GENERAL DESCRIPTION

All onions for processing are grown from specific varieties best suited for dehydration. Specific strains of the Creole Onion, Southport Globe Onion, and the Hybrid Southport Globe were developed by the dehydration industry. They are white in color and process a higher solid content which yields a more flavorful and pungent onion.

Onion dehydration involves the use of a continuous operation, belt conveyor using fairly low-temperature hot air from 38 - 104°C (100 to 220°F). The heat originally was generated from steam coils, but now natural gas is more popular. Typical processing plants will handle 4,500 kg (10,000 pounds) of raw product per hour (single line), reducing the moisture from around 83 percent to 4 percent 680 - 820 kg (1,500 to 1,800 pounds finished product). These plants produce 2.3 million kg (5 million pounds) of dry product per year using from 3,500 - 4,600 kJ per dry kg (0.15 to 0.20 therms per dry pound) produced +630 kJ (-0.06 therms of electrical energy), or 9,300 kJ per kg (4,000 Btu per pound) of water evaporated.

An example of one type of processing equipment, the Proctor dehydrator, is a single-line unit (64.6 m long and 3.8 m wide (212 ft long and 12.5 ft wide), requiring 2450 m^3 (86,500 ft^3) of air per minute and up to 42 million kJ per hour (40 million Btu per hour). Due to the moisture removal, the air can in some cases only be used once, and thus is exhausted. Special silica gel--Bryair, desiccations units are required in the final stage. Approximately $200,000 in fuel are thus used in a single-line dryer in a year's operation (180 days).

PROCESSING STEPS

Onion dehydration using a continuous conveyor dryer involves the following basic steps: a) harvesting, b) transporting to the plant, c) curing, d) washing, e) slicing, f) dehydration in three to four stages, g) milling, and h) packaging. Each of these steps is discussed in detail for a Proctor (Proctor and Schwartz, Inc. of Philadelphia) dehydrator. A diagram of a typical dryer is shown in Figure 1.

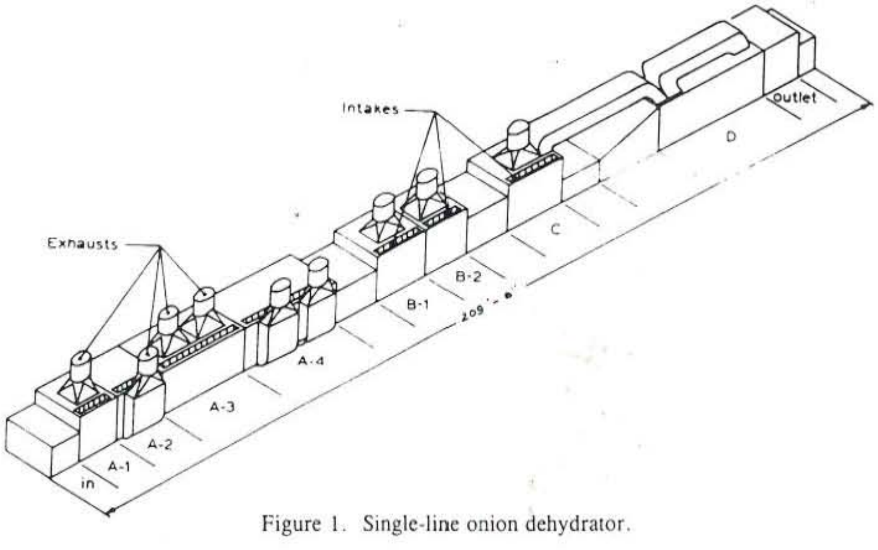

Figure 1. Single-line onion dehydrator.

Harvesting is accomplished mechanically by specialized equipment that is designed and fabricated by the processing industry. Harvesting is accomplished by a small] crew of 20 to 30 people used to inspect the onions and to operate the equipment. The onions are topped, dug, inspected, and loaded into bulk trucks holding about 22,700 kg (50,000 pounds) each.

The trucks loaded with onions are taken directly to the plant. They are loaded into large curing bins where excess moisture is removed by passing large volumes of heated air 38°C (100°F) through the onions. Curing conditions the onions so that peeling and processing can be accomplished successfully.

After curing for 48 to 72 hours, the onions are passed into the processing line. The earlier method of scooping up the onions with a tractor has been replaced with an automatic conveyor system that gently carries them to the preparation line. Machines automatically remove any tops that may remain attached to the onions. They are than inspected, washed in a high-pressure washer, soaked in a stainless steel tank to remove sediment, washed again in a high-pressure washer, and resoaked in a bath of highly chlorinated water in order to reduce bacteria to the lowest possible level. The onions are then reinspected and placed in stainless steel surge tanks. Two large stainless steel tanks are used so that one can be washed as the other is being used. The onions are fed out of the surge tanks into the slicers. Razor-sharp rotating knives cut the onions into uniform slices, which are then passed to the dryer.

From the slicers, a continuous and uniform flow of onions is conveyed to the wiper feed that carries the sliced product laterally across the open-feed end extension. Here the onions are carefully transferred to the dryer conveyor for the first stage of drying. This is the most critical stage; where, under high-volume air flow conditions and with moderately high temperatures, the bulk of the water is rapidly removed from the onion. The moisture content of the onion is reduced from an initial 83 percent to 25 percent. This is called the "A" stage; where, onion loading depth is approximately 10 cm (4 inches).

Absolute uniformity and controlled depth of loading on the dryers is necessary to prevent "pinking", an enzymatic dis-coloration that can take place in the onion slice if proper drying conditions are not maintained. The pure white color of the discharged product from this drying stage is a test of the high quality of the product. Normal drying temperature for stage "A" is around 104°C (220°F); however, temperatures as low as 82°C (180°F) can be used. The lower the temperature will increase the processing time; however, the quality will be improved.

High-powered blowers and exhaust fans move the air over natural gas burners (or geothermal heat exchangers), and through the beds of onions on the dryer conveyor, to evaporate the necessary tons of water removed from the product each hour. Close air volume and pressure control must be maintained in all parts of this drying stage as the air moves up and down through the bed to obtain product drying uniformity. Automatic temperature controllers and a long list of safety devices control the continuous operation.

At the proper point in the drying process, the onions are automatically transferred to the second stage ("B" stage) of drying; where, under reduced temperature conditions and deeper bed loadings (approximately 30 cm [12 inches]), the difficult to remove diffused water is slowly withdrawn. Here, moisture content is reduced 10 percent. At the special transfer zone, the onions are gently handled by rotary devices that assure full removal from the first-stage dryer and separation removal of clumps for uniform second-stage loading.

The second stage of drying transfers to the third stage ("C" stage) with even deeper loading (approximately 75 - 100 cm [30 to 40 inches] deep), as the deeply diffused water becomes even more difficult to remove. Moderating temperatures and air flows are used to maintain close product temperature control as a steady evaporation of water is reduced from each onion slice and the evaporative cooling effect can no longer be counted on to maintain the low product temperature required for maxi-mum product quality. After leaving the "C" stage, moisture content is down to 6 percent.

A special unloader takes the now nearly dry onions off the third-stage conveyor, transferring them to the elevating conveyer for the fourth and final stage of dehydration (if necessary). Here, conveyor loading depths up to 1.8 m (6 ft) are used for final moisture reduction and equilibration. Dehumidified air from a two-stage desiccation unit is counter-flowed through this deep layer to bring the finished onions to the point (about 4 percent moisture) where milling can best be accomplished and shelf life maintained.

After drying, the onions are passed over a long stainless steel vibrating conveyor that gently carries them to the milling area. In the mill, skin is removed by aspirators from the onion pieces. The onions are then milled into sliced, large chopped, chopped, ground, granulated and powdered onions.

In general, four stages (A through D) are preferred; however, if the ambient air humidity is below about 10 percent, stage D can be eliminated. Also temperature and number of compartments in each stage may vary. Figure 2 shows the typical energy requirements for a drier.

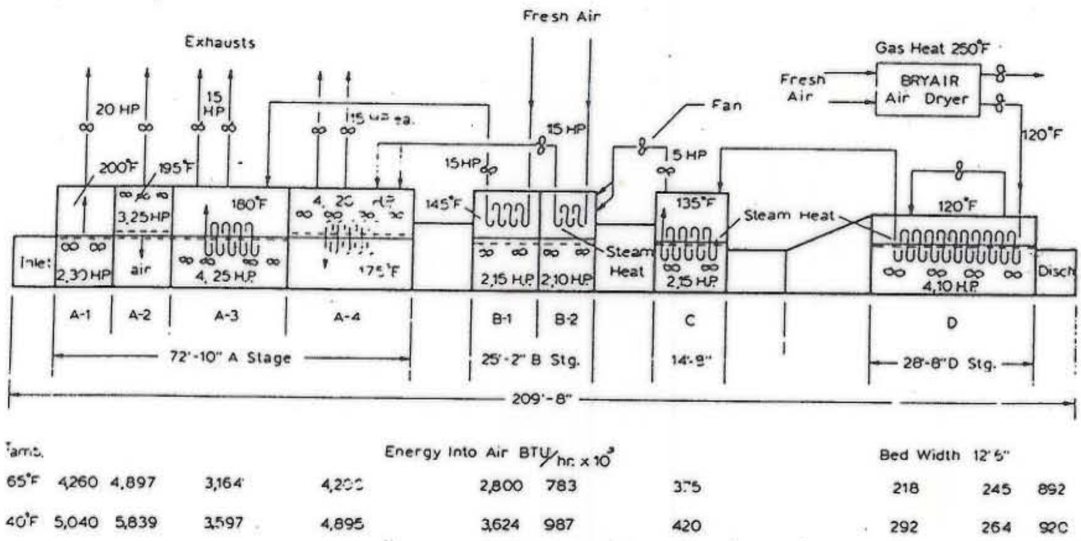

Figure 2. Temperature and energy requirements for each compartment of a single-line onion dehydrator.

Stage D, supplying desiccated air with a Bryair unit, reduces the moisture content of the product to a point below that of the ambient air. The unit is divided into two sides: the process side, which supplies desiccated air to the dryer after it has been passed through silica gel beds; and the reactor side in which heated air is passed over the silica gel beds in order to remove the moisture which had been absorbed in the process side.

The process air is drawn in from the outside under ambient conditions of temperature and humidity, passed through a filter and a cooling coil, and then is circulated through a dry silica gel bed where some of the moisture is absorbed. The process air then is drawn out by a fan and directed to the D2 stage of the dryer. This process air leaves the Bryair unit at a temperature of about 50°C (120°F) with a moisture content of about 30 grams per pound (4 g per kg).

On the reactor side, ambient air is drawn into the intake and passed over a gas burner which heats the air to about 120°C (250°F), after which the air is circulated through the silica gel beds so that the moisture which had been absorbed in the process side is removed.

A suction fan on the discharge side then exhausts the moisture-laden reactor air to the atmosphere at temperatures of from 65° - 107°C (150° to 225°F). A slight pressure differential is main-tained between the process and reactor sides so that air is pre-vented from leaking to the process side from the reactor side. For additional details see Lienau, et. al, (1978), and Lund and Lienau (1994).

GEOTHERMAL EXAMPLE

Integrated Ingredients dedicated their new onion and garlic processing plant on May 25, 1994 (Figure 3). "Grunion" as the new community of 72 employees has been labeled, is located just south of Empire and Gerlach and about 160 km (100 miles) north of Reno, Nevada. The plant, run by Integrated Ingredients (based in Alameda, CA), is a division of Burns Philp Food, Inc., which owns brands such as Spice Islands, Durkee-French and Fleischmann's. This plant gives the company the ability to produce its own products for industrial and consumer markets instead of purchasing them.

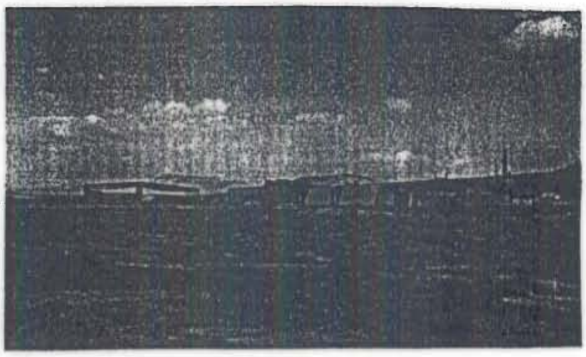

Figure 3. Overview of dehydration plant.

The plant was located in the San Emidio Desert at the edge of the vast Black Rock Desert and the Great Basin to take advantage of the high-temperature geothermal resource (approximately 132°C [270°F]). The resource is also used by the OESI/AMOR II 3.6 MW binary plant about a mile south of the dehydration plant and a gold heap leaching operation just to the north of the plant (Wind Mt. mine operated by AMAX). In addition to the geothermal energy, the high desert is an ideal location for onion and garlic processing because the cold winters kill damaging microbes. Dry winters and summers also help.

The raw product is grown throughout California and northern Nevada, with the local Empire Farms providing 11 million kg (24 million pounds) of onions per year. The processing plant operates 24-hours a day, seven days a week all year in three shifts. The company even provides daily shuttle service for workers from Fernly, about a hour's drive to the south. When onions are hauled from as far away as El Centro, California, six double-trailer trucks per day provide 136,000 kg (300,000 pounds) of wet product for processing. A cold storage warehouse, kept at -0.6°C (31°F), can store as much as 21,800 tonnes (24,000 tons) of product, which provides for the year around operation.

The 136,000 kg (300,000 pounds) of wet onions will produce 22,700 kg (50,000 pounds) of dry product at about 5% moisture. A total of 6.3 million kg (14 million pounds) of dry product are produced annually: 60% onion and 40% garlic. The finished product goes into barrels that weigh from 57 kg (125 pounds) for coarsely sliced onion, up to 136 kg (300 pounds) for finely finished garlic or onion powder. Product size can be powdered, granulated, ground, minced, chopped or sliced. The final produced is sold for seasoning in soups, cheezes, crackers, sauces, salad dressing and snack food.

PLANT OPERATION

The plant covers approximately 9,300 m² (100,000 ft²) and presently has one drier line (Figures 4 - 8). A second line is planned to be constructed in this space for increased future

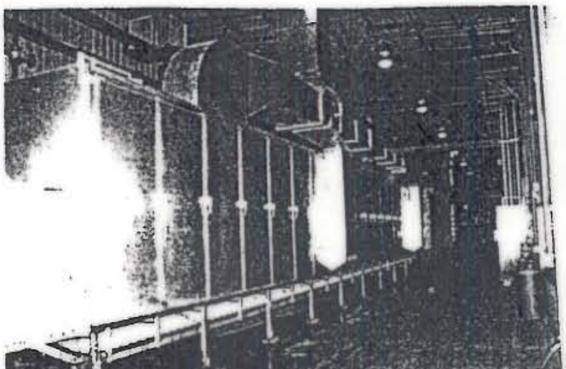

Figure 4. A-stage with B- and C-stages in the distance.

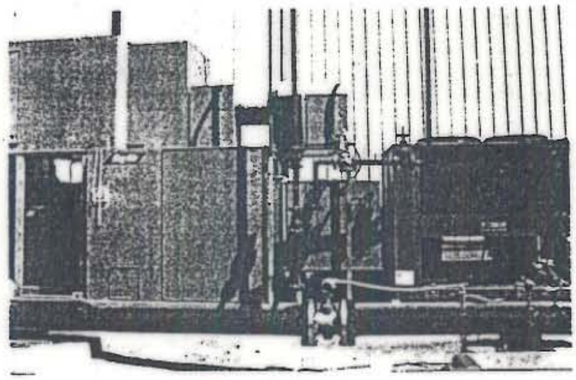

Figure 5. Bry-Air drier.

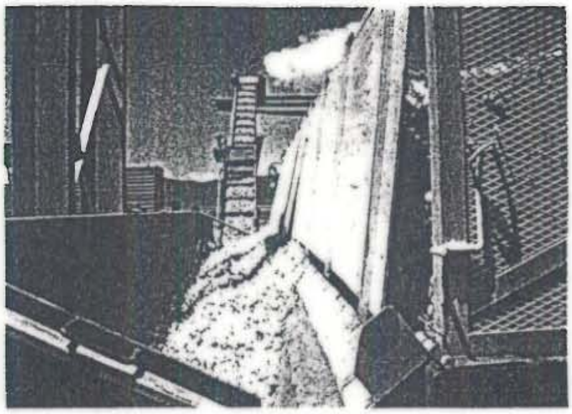

Figure 6. Raw onions being loaded onto the conveyor belt from a side-dump truck.

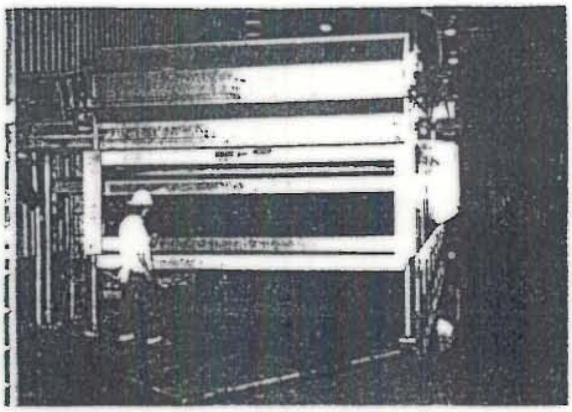

Figure 7. Front of A-stage, onion loading is from the top.

production. The stainless steel, drier line manufactured by The National Drying Machinery, Co. of Philadelphia, Pennsylvania, is approximately 3.6 m wide and 61 m long (12 feet wide and 200 feet long) consisting of three stages (A, B and C) and a Bry-Air drier to remove the final moisture from the product. This latter piece of equipment is used mainly in the winter months when the humidity is higher and can be boosted in temperature with electric energy.

In the case of onions, the product is washed, sliced and augered on the A-stage at about 5-cm (2-inch) depth where 96°C (205°F) air is used. The depth increases to 15 cm (6 inches) on the B-stage and the air temperature is lowered to around 85°C (185°F). On the final C-stage the depth is around 35 cm (14 inches) and the temperature 74°C (165°F). The moisture is around 85% at the start, 30 to 10% on the B-stage, and 4 to 6% in the dried product.

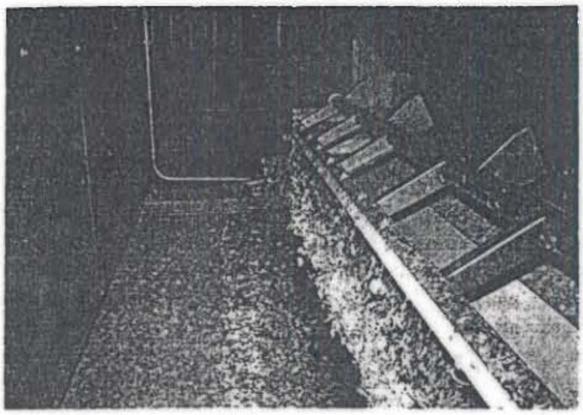

Figure 8. A-stage loading with sliced onions.

GEOTHERMAL ENERGY

The geothermal resource originates about 670 m (2200 feet) below the plant and is sealed by a silica cap. A 107-m (350-foot) deep well just south of the plant produces 130°C (266°F) water. Up to 3400 L/min. (900 gpm) are supplied to the plant with a 26 kW (75-hp) pump through an 25-cm (10-inch) insulated steel pipeline. Ball-joint expansion units are used on the pipeline at changes in direction (Figure 9). The discharge temperature varies depending upon the outside weather and plant operation, but can be as low as 71°C (160°F). Thus, the maximum energy use is around 47 million kJ/hr (45 million Btu/hr). The uninsulated discharge line, has a flow control valve operated by solar energy, with the waste fluid injected back into the ground south of the plant. A second well is presently being drilled just east of the plant to supply heat to the planned second line.

The San Emidio Desert or Mud Flat geothermal resource is expressed on the surface as an altered zone up to 30 m wide and 3 km long (100 ft wide and two miles long) along the east side of the desert between the Fox Range and

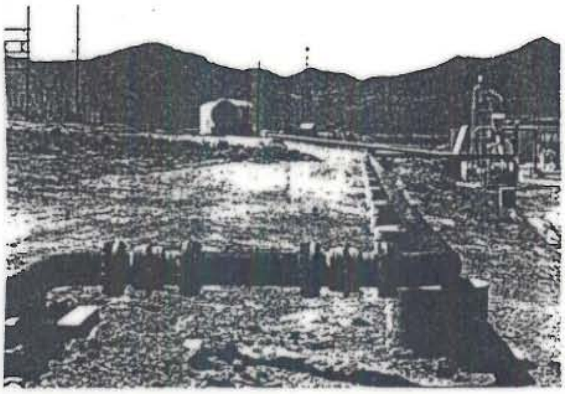

Figure 9. Ball-joint expansion device on pipe line.

Selenite Range. Cinnabar, sulfur, gypsum, siliceous sinter, opal, chalcedony, quartz, kaolinite and other alteration minerals occur in sands and gravels of Pleistocene age along the north-south zone. These altered deposits are covered by younger, unaltered alluvial and lacustrine deposits. The alteration and mineralization represent the deposits of hot springs which were probably more active in the past. The zone is near the high-water level of glacial Lake Lahontan (Pyramid Lake just to the south is a remanent of this lake), to which the mineralization may be related in some way.

The zone is still thermally active, and the ground is often warm 0.6 to 0.9 m (2 to 3 feet) below the surface. Water standing in shallow boreholes is up to 53°C (128°F) one m (3 feet) below the ground surface. A drill hole in 1955 encountered boiling water at 26 m (87 ft), and Chevron Oil Co. drilled a 1223-m (4013-ft) geothermal test well in 1975. The San Emidio Springs (T32N, R23E, S16) is reported at 75°C (167°F) flowing at 30 L/min. (8 gpm) with a pH of 6.7 and TDS of 4478. The water is mainly sodium-chloride with 2300 ppm chloride, 1400 ppm calcium, 125 ppm magnesium, 110 ppm potassium, 93 ppm bicarbonate, 6.3 ppm boron, 5 ppm fluoride and 0.13 ppm iron (Garside and Schilling, 1979).

ACKNOWLEDGEMENTS

We would like to thank the following people at Integrated Ingredients for the opportunity to visit the plant: Dave Wilson, Bill Lanning, Mark Fredrickson and Linda Lumm.

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APPENDIX 6.2

PILOT FRUIT DRIER FOR THE LOS AZUFRES GEOTHERMAL FIELD, MEXICO

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ABSTRACT

The region around the Los Azufres geothermal field is a large fruit producer, thus the local growers may be interested in constructing dehydration facilities at Los Azufres and to buy surplus heat from CFE. CFE is installing a small fruit drier to demonstrate the feasibility of drying fruit with geothermal water, and thus encourage private development on a larger scale. The pilot drier will be enclosed in a small building with room for two trucks (hand carts) each holding 30 one-meter square trays with a total capacity of approximately one tonne of wet fruit. The forced air heat exchanger and fan unit were designed for the high elevation (3000 m). Operation will begin in the summer of 1995.

Key words: dehyration, fruit drying, direct use, agriculture, truck drier.

1.0 INTRODUCTION

Comision Federal de Electricidad (CFE) has a division in charge of the exploration of geothermal reservoir located in Los Azufres, State of Michoacan (about 250 km west of Mexico City). At present, CFE is only using the steam from the wells and rejecting the hot water that comes off associated with the steam.

CFE is promoting the use of the geothermal hot water in industries with high consumption of heat. Several local industries are interested in constructing facilities at Los Azufres and to buy heat from CFE. So far, CFE has installed a chamber for drying lumber with very good results. They have also constructed a pilot greenhouse to grow cut flowers in winter and to produce gladiolus bulbs.

Since the region of Los Azufres is mainly a fruit producer (pears, peaches, guava, etc.), they propose to construct a small fruit dehydrator for demonstration purposes. They are confident that if they succeed in showing the feasibility of drying fruit with geothermal heat, they will have a demand for larger drying installations.

As a result of this interest, the Geo-Heat Center in cooperation with CFE, designed a pilot geothermal fruit drier that is presently under construction. The details of the design are presented in this paper.

2.0 LOS AZUFRES GEOTHERMAL FIELD

The Los Azufres geothermal field, started in 1981, is located approximately 80 km east of Morelia (Fig. 1). The elevation of the field varies from 2000 to 3000 m. The mean annual air temperature is 12° C with the temperatures ranging from 31° C to -4° C. The annual precipitation is 1,171 mm, the average relative humidity is 63% and the atmospheric pressure 0.73 bars (based on average conditions from 1983 to 1990).

The field has 63 geothermal wells at an average depth of 2,100 m. The total steam production is 1,550 t/h with a noncondensible gas content of 3% by weight, which is composed of 97% CO_2 and 3% H_2S. The field's brine production is 1,600 t/h at a separation

Figure 1. Location of the Los Azufres geothermal field, Mexico.

temperature of 170° C. The average production from a well is 50 t/h, with the brine at a pH of 7.2 and a chemical composition of (in mg/L):

Cl	4,399
Na	2,321
SiO_2	1,050
K	628
B	365
HCO_3	83
Li	32
As	28
SO_4	23
Ca	16
Rb	5
Cs	4
Mg	trace

The well to be used for the fruit drier is AZ-22 located on the main administration building compound at Los Azufres, at an elevation of 2864 m. The well produces 110 t/h at 170° C. The production from this well will be shared with the timber drier and used for future space heating of the administration buildings. Cold water is available from a nearby stream.

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3.0 FRUIT PRODUCTION AND DRYING

At present, the annual fruit production of the surrounding area is as follows (in tonnes/year):

Pears	15,000
Prunes	5,000
Peaches	1,350
Guavas	900
Apples	140

Based on recent studies, it is estimated that there is a demand for drying 40% of the annual fruit production. This would allow the growers to sell the fruit over a longer time period and to ship to more distant markets without worrying about spoilage.

Since two thirds of the fruit grown in the area is pears, the drier was designed to handle this fruit; however, other types can also be dehydrated in the same drier.

The preparation and processing of the fruit is based on a 1946 report prepared by the California Agricultural Experiment Station (Perry, et al., 1946). No new publications are available; however, the information is still valid according to Dr. James Thompson of the Agricultural Engineering Station at the University of California at Davis (Thompson, 1992). These details will not be discussed in this paper.

The actual design will handle about one tonne of fruit (wet) per drying cycle. Cutting, storing and packaging of the fruit will be done on site in a separate building. A cold storage facility may be designed to keep fresh fruit when harvest exceeds the capacity of the drier.

4.0 FRUIT DRIER DESIGN

Since the drier is a demonstration project, the size will be minimal to expedite construction and minimize cost. The design is based on preliminary work reported by Herman Guillen (1987).

4.1 Building Design

The drier building will be about 4.00 m long, 1.35 m wide and 3.2 m high (Fig. 2). The actual dimensions will depend upon the size of local building materials.

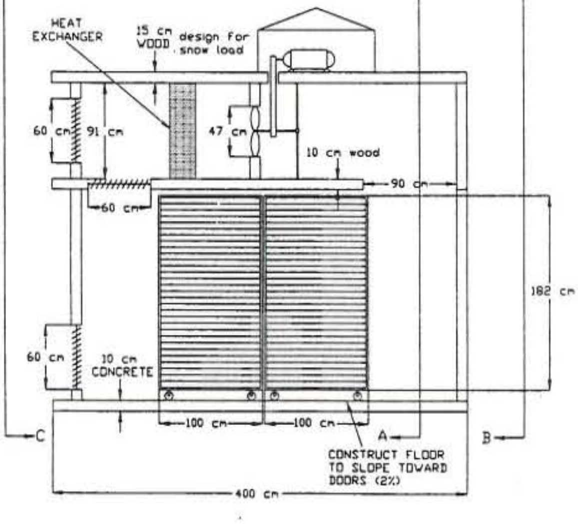

Figure 2a. Detailed drawing of tunnel dehydrator.

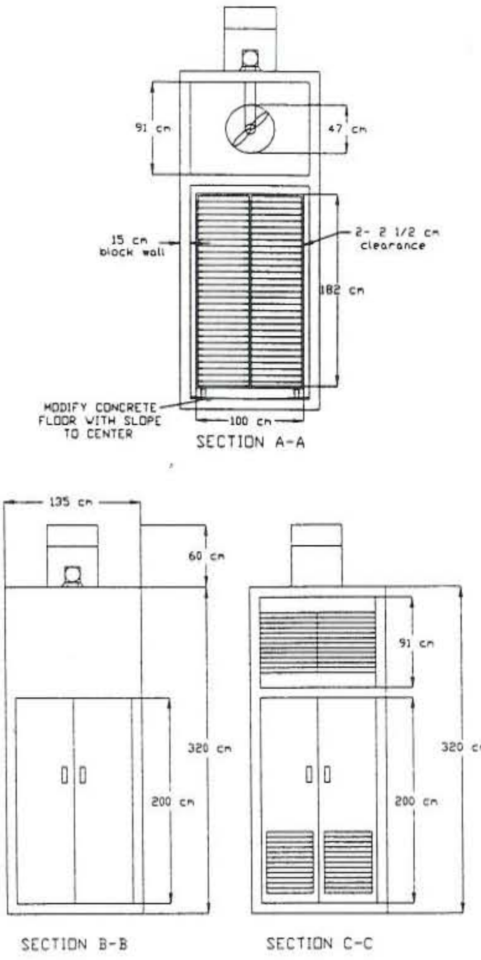

Figure 2b. Cross sections of dehydrator.

The walls will be constructed of concrete block, the ceiling and roof of timber and the floor of placed concrete. The floor will have a slight depression down the middle and slope toward the front doors to drain any juices from the drying fruit and for ease of cleaning. The heat exchangers and fan motor will be housed on the roof so that the latter is away from the hot air stream.

The trucks and walls are designed so that there is no more than about 2 to 2.5 cm of clearance on either side and at the top. This will maximize the air velocity and efficiency of the system.

Louvered doors will be provided for entering, recirculating and leaving air. The louvers will be manually set, but could be set automatically as controlled by temperature and humidity sensors.

4.2 Truck and Drying Tray Design

Two trucks will be used; each with a base of 1.00 m by 1.00 m and 1.82 m high when loaded with trays (Fig. 3). The truck base will have four casters (pivot wheels) and a detachable handle that can be attached at either end. This will allow the trucks to be reversed when halfway through the drying process time. The base will be constructed of plywood approximately 2 cm thick. Each truck will carry 30 trays. Each tray will carry approximately 15 kg of fruit (wet) for a total of 450 kg per truck and almost one tonne per drying cycle for the two trucks. During the drying operation the moisture content of the fruit will change from about 80% to 20% by wet weight. Drying time is approximately 24 hours (Thompson, 1994).

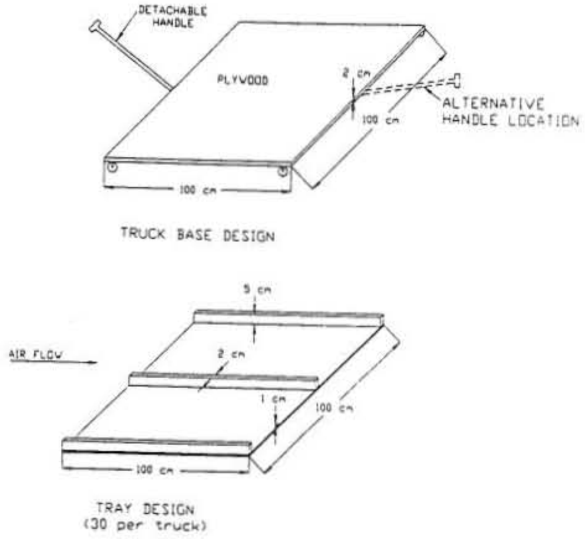

Figure 3. Truck and tray design.

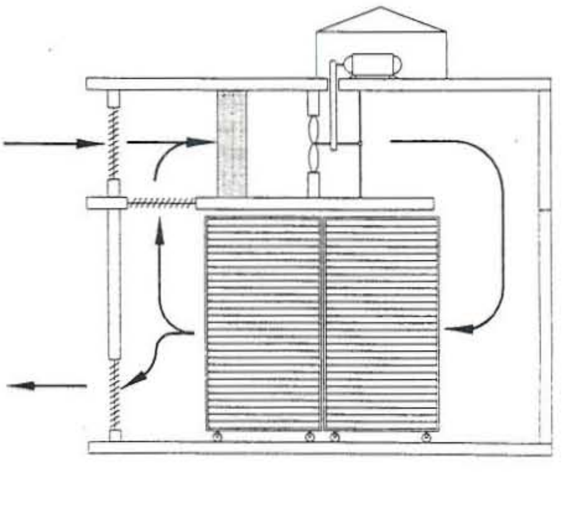

Figure 4. Tunnel dehydrator air flow pattern.

The trays will be constructed of 1-cm thick plywood and have 5-cm high by 2-cm wide wood strip attached to either edge, along with one down the center (parallel to the air flow) for strength and stacking. The plywood trays will have 1-cm diameter holes drilled in them for drainage of fruit juice produced during drying.

4.3 Heat Exchanger Design

The required air speed for fruit drying is high; ideally about 240 to 300 m/min. with a minimum of 150 m/min (Thompson, 1992). Estimating that the trays and fruit block 50% of the tunnel, then the cross section for air flow will be 1.00 m x 2.00 m x 0.50 = 1.00 m^2. Thus, a minimum capacity of 150 m^3/min will be needed (240 to 300 m^3/min. ideal). Converting this requirement to 2864 m elevation (air density ratio equals approximately 0.70), a minimum capacity of 215 m^3/min. will be necessary at Los Azufres to produce the same drying capacity. Fortunately, the evaporation rate will also be increased at this elevation due to the reduction in outside pressure relative to the vapor pressure in the fruit, thus allowing the use of the minimum design air flow.

A minimum of 0° C outside entering air temperature and a maximum of 70° C drying temperature was assumed. (The ideal temperature for pear drying is 60° C and the maximum is 74° C.) The geothermal resource was assumed to enter at 120° C and exit at 100° C. Based on these assumptions, the required heat exchanger will need two rows of 8 finned tubes at 91 cm by 91 cm cross section (Rayner, 1992).

This is the design for the most severe conditions. The geothermal flow rate can be adjusted by a valve to compensate for changing outside air temperature. A three-way valve with a temperature sensor in the air stream could be used for automatic control. The air flow will enter through a 60 cm x 100 cm louver, through a 91 cm x 91 cm duct in the top of the building, and then flow down through the trucks (Fig. 4). The air can then be exhausted or it can be recycled if the outside air temperature is very low. In many dehydrators, at least 90% of the air is recycled to conserve input energy.

The actual temperature and air flow rates will have to be adjusted by trial-and-error to achieve the proper final product in terms of moisture, texture and color.

A second heat exchanger of the water-to-water type may be necessary to reduce the possible effects of corrosion or scaling from the geothermal water. This would consist of a small plate heat exchanger with a secondary loop supplying passive water to the water-air heat exchanger in the drier building. The plate heat exchanger would be sized to handle the future heating load from the administration buildings.

4.4 Fan Unit Design

The tube axial fan was designed for 215 m^3/min and 2 cm of water pressure head loss (air flow friction loss) at 0.722 g/L air (1.20 g/L at sea level). This will require 1.05 BHP or a 1.5 hp motor (1.12 kW). The fan will be 61 cm in diameter and have 5 blades with a 10.5° blade tip pitch. Due to the high temperature of the air flow, the fan motor will have to be located on top of the building outside of the hot air stream. Details of the fan and housing are shown in Figure 5 (Rayner, 1992).

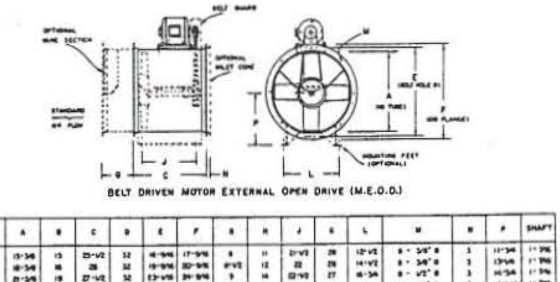

Figure 5. Details of fan unit.

4.5 Estimated Costs

The estimated costs are as follows:

Building	$2000
Truck and trays	500
Heat exchanger	800
Fan unit	1700
Controls/piping	1000
TOTAL	$6000

The use of local materials and labor may reduce the above costs.

5.0 CONCLUSIONS

The building is presently under construction and will be completed in early 1995. Operation will begin for the summer/fall 1995 harvest season. Through this project and others, CFE anticipates increased interest and use of the waste geothermal brine. Cascading will increase the efficiency and reduce the cost of producing and utilizing the geothermal resource.

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APPENDIX 6.3

RESEARCH ON THE USE OF WASTE SILICA FROM THE CERRO PRIETO GEOTHERMAL FIELD, MEXICO

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ABSTRACT

The Geo-Heat Center has been investigating the utilization of waste silica from the Cerro Prieto geothermal field for several years. The main objectives of the research were to combine silica with various additives to (1) form bricks for low cost housing, and (2) to produce a suitable road surfacing material. The various additives that were tested included hydrated lime, portland cement, plastic fibers, asphalt cement and emulsified asphalt. The silica-cement combination produced the strongest bricks and had the best weather resistance, whereas, the silica-lime combination produced the bricks with the lowest thermal conductivity and specific gravity density. The addition of plastic fibers to the silica-lime mixture improved both strength and weather resistance. The combination of asphalt and silica is not suitable as a road surfacing material, however, silica-cement appears promising.

INTRODUCTION

The Geo-Heat Center has been investigating the utilization of waste silica from the Cerro Prieto geothermal field for several years (Lund et al., 1994, 1995a, and 1995b). The main objectives of the research was to combine silica with various additives to (1) form bricks for low cost housing, and (2) to produce a suitable road surfacing material.

The impetus behind this project was the large quantities of silica being produced from waste brines at the power plants in the Imperial valley of Mexico and California, and a cooperative agreement between the U.S. Department of Energy (USDOE) and Comision Federal de Electricidad (CFE) of Mexico.

Of specific interest was the Cerro Prieto geothermal field in Mexico which has an installed capacity of 620 MW, and in the process generates 6,400 tonnes/hr (7,000 tons/hr) of brine consisting of about 6 tonnes/hr (6.6 tons/hr) of silica (927 ppm average). Since the geothermal fields of the area extend into the Imperial Valley of California where waste silica is produced from an additional 420 MW of geothermal power generation, it is hoped that this research would also be applicable to the U.S. side of the border.

The residual waste brine, after evaporation is reduced to 5,600 tonnes/hr (6,200 tons/hr) at Cerro Prieto. It is then disposed of into large surface evaporation ponds covering 18.6 square km (4,600 acres) in area. The volume of silica in these ponds is unknown, however the field has been operating since 1973, and thus there should be approximately half a million tonnes of silica in the ponds.

Some attempts have been made by UNOCAL at their Imperial Valley plant (now owned by Magma Power) to use their waste silica stabilized with cement for roads and dikes around the plant. However, concern over low levels of radioactivity, has curtailed this work. They are now disposing of the waste, extracted by a crystallizer-clarifier system to control scaling, to a separate disposal site.

CFE has done testing on various mixtures of silica and additive for building blocks and roofing tiles. Samples of their results are displaced at the museum at Cerro Prieto. Unfortunately, no documentation of this testing was every prepared, thus the results and many of the additives are unknown.

RESEARCH OBJECTIVES

The objectives of the research were to:

1. Produce low specific gravity bricks that were suitable for low-cost building construction using the waste silica with various cementing additives (i.e. have adequate strength, low thermal conductivity and high resistance to weathering).

2. Produce a mixture of silica with either cement or asphalt that would be suitable for a low-volume road surfacing (i.e. has adequate strength and stability, and resistance to traffic abrasion).

The testing procedure would include:

1. Mixing the silica with lime, cement, pozzolan and fibers to mold bricks and cubes, and then cure them under various conditions of temperature and moisture.

2. Test molded specimens after various curing times (7, 14 and 28 days) in flexure (bricks) and compression (cubes).

3. Test dried samples for thermal conductivity and weathering.

4. Test silica-asphalt mixtures by Marshall stability and immersion-compression.

SILICA CHARACTERISTICS

The term "silica" is used here to describe material that is mainly silica, but does contain other chemical species. Three separate samples of silica waste were taken and shipped from Cerro Prieto during the two years of the study. The initial sample, unknown to us, was from an evaporite deposit at a silencer, whereas the later two samples were actually taken from the evaporation ponds. The evaporite deposit had a specific gravity of 2.29 and was extremely fine grained (over 90% passed the #200 sieve (0.075 mm). The two pond samples had specific gravities of 2.27 and 2.18, and were much coarser with visible amorphous particles (Figure 1). These latter samples had approximately 75% and 30% passing the #200 sieve (see Figure 2 for the complete mechanical analysis). Since the initial sample results were not typical of what could be obtained from the larger source in the evaporation ponds, the results were not considered significant, but are documented in (Lund, et al., 1995a).

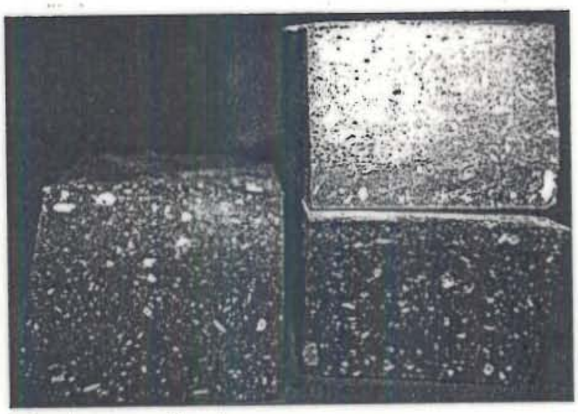

Figure 1. Cross-section of silica-cement bricks showing silica gradation.

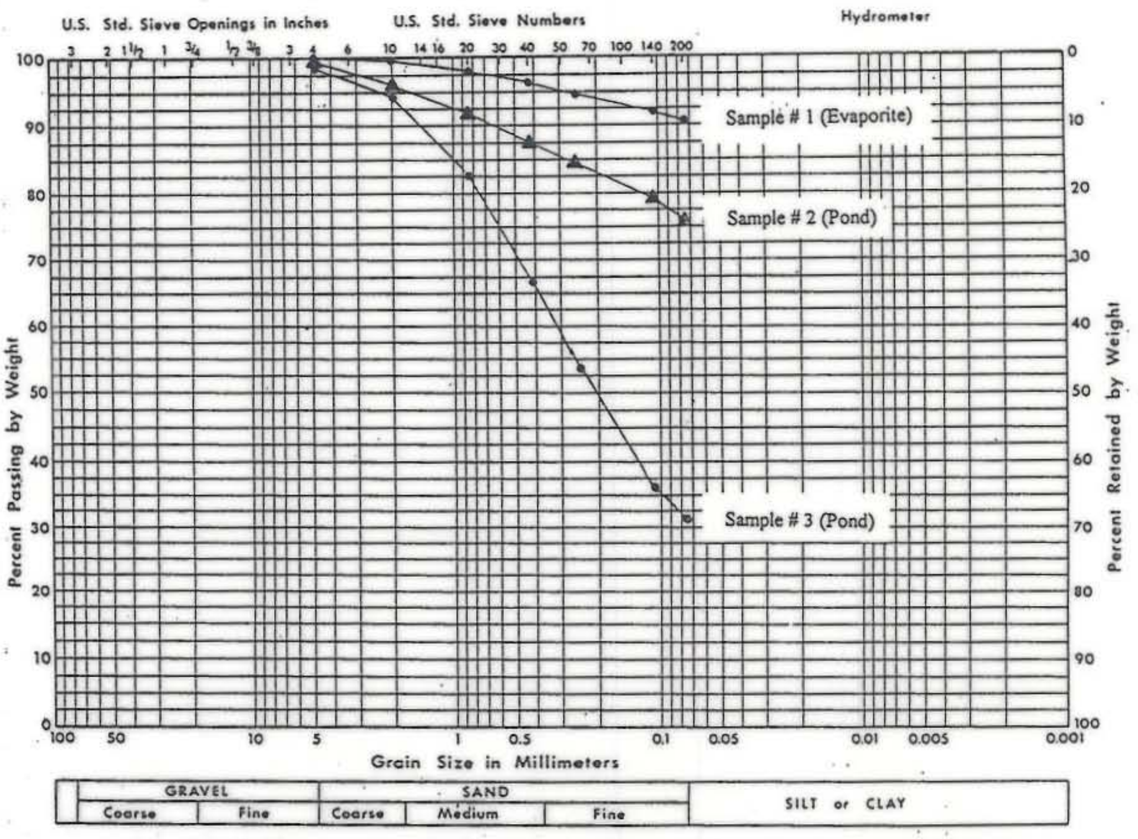

Figure 2. Mechanical analysis of Cerro Prieto silica waste.

According to work done at Cerro Prieto in 1993 (Residencia General de Cerro Prieto, 1994), the typical chemical analysis of the brine is shown in Table 1.

TABLE 1
Chemical Analysis of the Brine (mg/l - ppm)

Total dissolved solids	28,286
Chloride	15,638
Sodium	8,510
Potassium	1,971
Silica	927
Calcium	388

Work done for us by Brookhaven National Laboratory (personal communication, Dr. Eugene T. Premuzic, 1996) on the two pond waste silica samples are shown in Table 2 (with over 100 ppm concentration). The two sample appeared to be composed primarily of silica (over 80%) and varying amounts of potassium, calcium and chloride. They were very low in barium and no thorium or radium were detected by Energy Dispersive Spectroscopy (EDS) or the counting procedure. The results were obtained by both EDS x-ray analysis and by Scanning Electron Microscopy (SEM).

TABLE 2
Chemical Analysis of Waste Silica (ppm)

Isotope	Sample #2	Sample #3
Silicon	3745.4	4308.5
Iron	1521.8	1749.2
Calcium	815.4	823.2
Aluminum	294,8	645.9
Zinc	390.9	89.8
Phosphorus	321.0	0
Boron	230.5	229.4
Manganese	156.1	241.2
Magnesium	20.2	120.6

TESTING PROCEDURE

Bricks and Cubes

The bricks were formed in 7.60 cm wide by 5.10 cm high by 15.2 cm long (2 in. x 3 in. x 6 in.) molds with removable sides. These would then be cured under various conditions of moisture and heat and finally tested in bending (flexure) by three point loading (Figure 3). The test procedure closely followed ASTM C 293-79 (Standard Test Method for Flexural Strength of Concrete), and were tested after 7, 14 and 28 days of curing. This was later modified and only 7 days of curing was used.

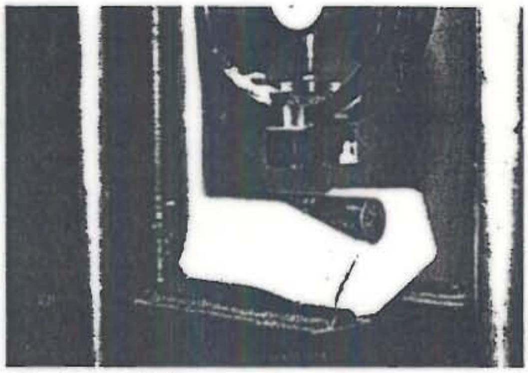

Figure 3. Flexure testing of silica-mixture bricks.

Cubes were formed in 5.10 cm (2.00 in.) square molds and then tested in unconfined compression following ASTM C 109-90 (Test Method for Compressive Strength of Hydraulic Cement Mortars). Some difficulty was experienced in determining the maximum strength of these specimens, producing variable results, thus this test procedure was later suspended.

Asphalt Mixtures

The asphalt cement (AR-4000) mixtures were compacted into 10.2-cm (4.00-in.) diameter by 5.1-cm (2.00-in.) high specimens and then heat cured. They were then tested in compression according to the Marshall Stability Test (ASTM D 1559-82). This test procedure would determine if the material was suitable for use as asphalt concrete structural pavement surfacing material. Emulsified asphalt (CRS-2H) mixtures were tested in immersion-compression (ASTM D 1074-83 and D 1075-81) to determine its suitability for surface treatment of roads in the form of a slurry seal.

Thermal Conductivity

Samples of all the bricks were mailed to USGS in Menlo Park for thermal conductivity testing. In conjunction with these test, the dry specific gravity of each brick was determined to see if there was a significant correlation between the two measurements. The thermal conductivity was determined using the conventional needle probe in a half-space mode (Sass, et al., 1984).

Weathering

The more promising mixtures for the bricks were subjected to a weathering test. Since time was not available for an extended outdoors test, an accelerated laboratory test procedure was developed. This involved a wet-dry test where a dried brick was first sprayed with water, then soaked overnight (about 12 hours), then oven dried at 60°C (140°F) for 12 hours, before repeating the cycle. A total of 10 cycles were performed and the initial dry weight was compared to the final dry weight to determine a percentage loss. The greater the loss, the less suitable the mixture is for construction use where it will be exposed to weathering.

TESTING RESULTS

A summary of the flexural strength, specific gravity, thermal conductivity and weather percent loss are show in Table 3. The type of sample indicates the weight proportions of silica to cementing material. The sample numbers indicate which sample of silica was used (1 = original silencer sample, 2 and 3 = pond samples).

The silica-hydrated lime mixtures produced the lowest specific gravity, thus indicating that they would have the best insulating values (low thermal conductivity). These mixtures also produced the lowest strengths of all the various additive combinations. Initially the samples were cured in a water bath with poor results, and then heat cured in an oven at 60°C (140°F) for 7, 14 and 28 days. The heat curing was to simulate accelerated curing in the field. Flexural and compression testing produced lower strengths with increased curing time, contrary to what was expected. Upon a detailed investigation, it appeared that the samples were drying out in the oven which prevented adequate curing and produced minute thermal cracks in the bricks (Figure 4). The longer the curing time the more thermal micro-cracks that were produced, since the curing water in the bricks was evaporating. The samples then failed in flexure along these thermal micro-cracks.

Since, the strength of lime-stabilized mixtures is both time and temperature dependent, it was found that curing temperatures above 50°C (122°F) should be avoided, with 40°C (104°F) recommended without introducing pozzolanic reactive products that significantly differ from those expected during field curing (Transportation Research Board, 1987). Research reveals that the lower curing temperature is equivalent to producing 28-day strength in about 69 hours (Biswas, 1972 and Townsend and Donaghe, 1976). Thus, we felt that 7-days curing was more than adequate to simulate field curing time.

Table 3
Summary of Test Results Hydrated Lime Mixtures

Sample Name	Type of Sample	7-day Flex (KPa)	14-day Flex (KPa)	28-day Flex (KPa)	Specific Gravity	Thermal Conductivity, W/mK	Weathering Percent Loss
1I	1-Silica/1-Lime	387.80	258.60	86.20	0.64	0.36	73.10
2A	1-Silica/1-Lime	1,034.20	861.80	560.20	0.72	0.30	7.60
3HA	1-Silica/1-Lime	732.60			0.96	0.32	2.00
1J	2-Silica/1-Lime	129.30	86.20	86.20	0.58	0.35	100.00
2B	2-Silica/1-Lime	517.10	430.90	344.70	0.67	0.30	47.60
2P	2-Silica/1-Lime	1,465.10	1,465.10	1,335.90	0.65	0.31	6.00
2Q	2-Silica/1-Lime	1,637.50	1,982.20	1,637.50	0.67	0.31	6.00
3IA	2-Silica/1-Lime	517.10			0.96	0.34	2.80
1N	3-Silica/1-Lime	86.20		43.10	0.49	0.29	100.00
2C	3-Silica/1-Lime	430.90	344.70	258.60	0.65	0.31	12.30
3JA	3-Silica/1-Lime	3,447.40			0.99	0.35	3.20
3KA	4-Silica/1-Lime	2,542.40			0.96	0.30	3.10
3LA	5.67-Silica/1-Lime	301.60			0.95	0.36	5.00
3MA	9-Silica/1-Lime	172.40			0.89	0.34	17.30
3NA	19-Silica/1-Lime	129.30			0.89	0.32	100.00
1K	1-Silica/1-Cement	2,930.30	2,973.40	2,844.10	0.81	0.36	3.70
2D	1-Silica/1-Cement	6,334.60	5,515.80	5,774.40	1.08	0.34	2.30
3T	1-Silica/1-Cement	5,946.70			1.47	0.44	10.20
1F	2-Silica/1-Cement	1,508.20	1,465.10	1,809.90	0.73	0.36	9.50
2E	2-Silica/1-Cement	3,231.90	3,705.90	4,438.50	0.88	0.33	6.20
2N	2-Silica/1-Cement	3,792.10			0.79	0.31	4.40
2O	2-Silica/1-Cement	3,921.40			0.82	0.30	4.30
3U	2-Silica/1-Cement	4,912.50			1.24	0.38	6.50
1H	3-Silica/1-Cement	861.80	861.80	818.80	0.57	0.34	7.70
2F	3-Silica/1-Cement	3,231.90	3,878.30	4,524.70	0.80	0.30	7.20
3V	3-Silica/1-Cement	4,179.90			1.24	0.39	1.70
3DA	4-Silica/1-Cement	2,154.60			1.24	0.40	12.20
3EA	5.67-Silica/1-Cement	1,034.20			1.15	0.34	12.20
3FA	9-Silica/1-Cement	517.10			1.04	0.32	10.10
3GA	19-Silica/1-Cement	129.30			0.92	0.29	11.70
1L	1-Silica/1-Lime/1-Cement	3,016.50	3,059.50	3,447.40	0.84	0.36	3.50
2G	1-Silica/1-Lime/1-Cement	3,447.00	4,524.70	3,361.20	0.96	0.35	6.60
1D	2-Silica/1-Lime/1-Cement	2,154.60	1,465.10	1,637.50	0.81	0.42	17.60
2H	2-Silica/1-Lime/1-Cement	3,705.90	4,309.20	4,697.10	0.84	0.35	10.90
1G	3-Silica/1-Lime/1-Cement	1,508.20	1,077.30	1,206.60	0.67	0.34	6.80
2I	3-Silica/1-Lime/1-Cement	3,188.80	2,973.40	3,533.60	0.80	0.30	4.90
2M	4-Silica/1-Lime/1-Cement	2,628.60	4,093.80	2,844.10	0.71	0.32	5.70
1S	1-Silica/1-Lime/1-Fiber	430.90	711.00		0.57	0.34	8.40
2J	1-Silica/1-Lime/1-Fiber	1,809.90	1,508.20	1,249.70	0.67	0.27	9.40
1YA	2-Silica/1-Lime/1-Fiber	86.20			0.49		100.00
2K	2-Silica/1-Lime/1-Fiber	1,465.10	1,120.40	1,508.20	0.59	0.28	11.00
1ZA	3-Silica/1-Lime/1-Fiber	86.20			0.44		100.00
2L	3-Silica/1-Lime/1-Fiber	1,766.80	1,335.90	1,292.80	0.62	0.29	12.00

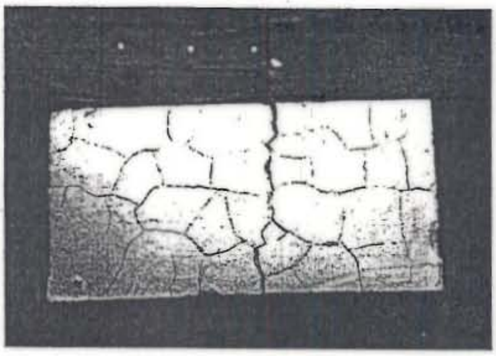

Figure 4. Top view of a silica-lime brick showing micro-fractures (enhanced with a ball-point pen).

Based on the above findings, two changes in our procedure were introduced (1) curing at 40°C instead of 60°C, and (2) curing in moisture-proof plastic bags. As a results, almost no moisture was lost from the bricks and higher strength were produced and these increased with curing time. The results of 7-day flexural strengths are shown in Figure 5. The silica from sample site #3 were mixed with a silica to lime proportion from 1:1 all the way to 19:1 (50% to 5% lime by total dry weight of mix), and cured using the revised procedure. Moisture content of these latter samples varied from 63% to 71% by total dry weight of the mix. These strengths are more indicative of what can be produced in the field.

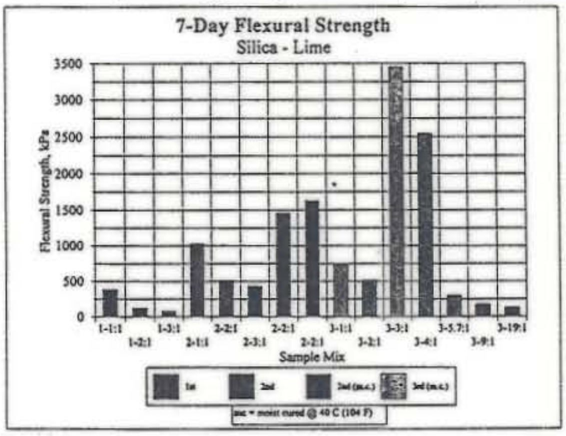

Figure 5. 7-day flexural strength of selected silica-lime mixtures.

Portland Cement

Portland Cement mixtures (using Type II cement) produced flexural strengths that were approximately twice that produced by lime stabilization. The flexural strengths are dependent upon the amount of mixing water used, as the lower water/cement ratios produce high strengths. Samples were tested using a silica-cement ratio ranging from 1:1 to 19:1 (50% to 5% cement by total dry weight of mix). Specific gravities and thermal conductivities were slightly higher for the cement mixtures as compared to the lime mixtures. Moisture contents varied from 52% to 70% by total dry weight of the mix. The flexural strength results of cement mixtures are shown in Figure 6.

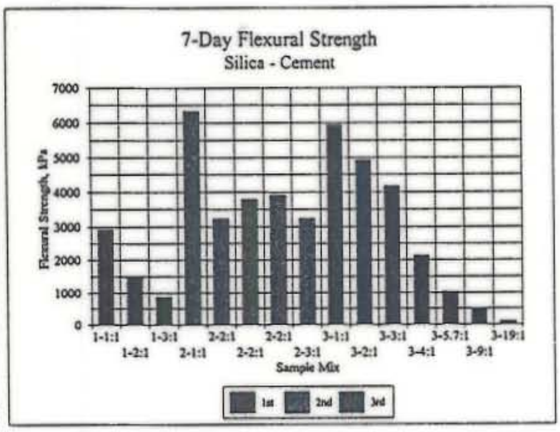

Figure 6. Flexural strength of selected silica-cement mixtures.

Portland Cement and Hydrated Lime

Results from the combined cement and lime stabilization produced strengths between those obtained from just lime and cement alone. There appears to be no strong advantage to using this combination of additives, unless the cost of lime is considerably less than cement, and the strengths higher than those obtained from just lime stabilization are desired.

Hydrated Lime and Plastic Fibers

Approximately eight grams (0.3 oz) of plastic fibers, varying between 1.4 and 2.7 percent by dry weight of sample, were used to provide additional flexural strength to the lime stabilized samples. This produced significantly higher strengths than those samples without fibers cured at 60°C (140°F) and only slightly higher strength when compared with those cured at 40°C (104°F).

Thermal Conductivity

Thermal conductivity was determined for various dry weight samples of bricks using the conventional needle probe in a half-space mode at the USGS laboratory in Menlo Park, California (person communication with Colin Williams). The thermal conductivities varied from 0.27 to 0.44 W/mK. In general, the lower the specific gravity of the mixture, the lower the thermal conductivity. Also, for a particular sample of silica, the thermal conductivity of the bricks decreased with increasing silica content. Specific gravities of the silica-lime samples varied from 0.635 to 0.991 and for the silica-cement samples from 0.571 to 1.244. The silica from sample site #3 produced the highest specific gravities and the highest thermal conductivities. The thermal conductivities compare with values for common brick at 0.72, gypsum or plaster board at 0.17, glass fiber insulation at 0.043 and urethane foam at 0.026 W/mK. Figures 7 and 8 are a plot of specific gravity vs thermal conductivity for selected silica-lime and silica-cement samples.

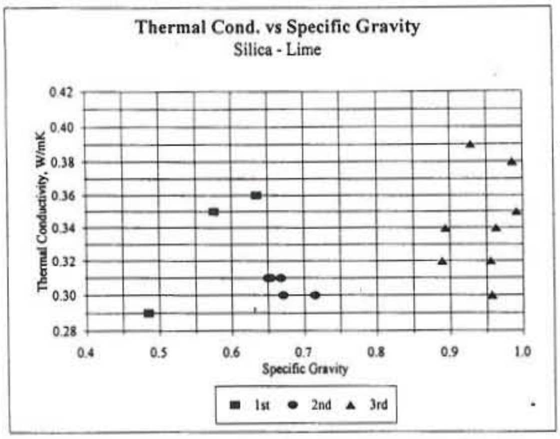

Figure 7. Specific gravity vs. thermal conductivity of selected silica-lime samples.

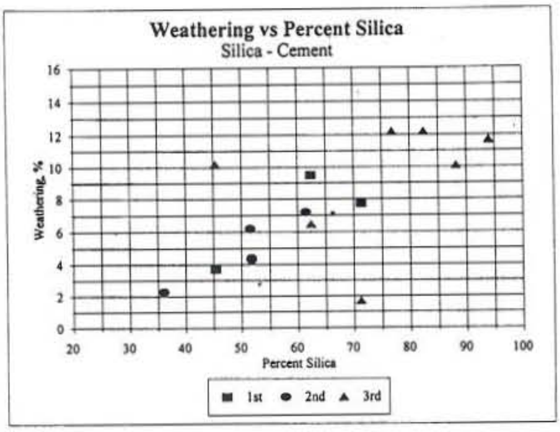

Figure 8. Specific gravity vs. thermal conductivity of selected silica-cement samples

Weathering

In general, the higher the silica content the greater the percentage weight loss due to the simulated weathering cycles. Most of the silica-lime mixtures cured at 60°C (140°F) completely failed (100% loss) before the end of the test period. Silica-lime samples with plastic fibers held together much better, usually with only a 10% weight loss. The silica-cement and silica-lime-cement mixtures fared well, all except one, with less than 12% loss. The silica-lime samples cured at 40°C (104°F) in sealed plastic bags, had less than 12% loss, except for the 10% and 5% lime content samples. Figures 9 and 10 show the relationship between silica content and the percent of weathering for the three silica sources.

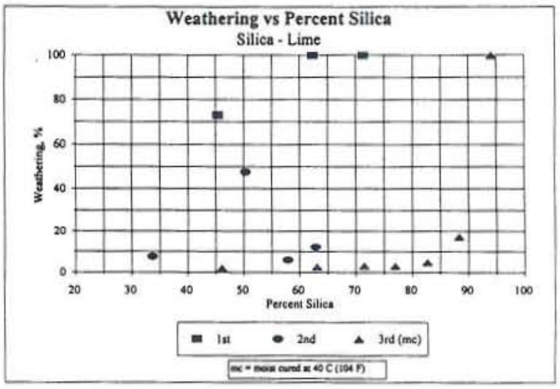

Figure 9. Weathering vs. silica content for selected silica-lime samples.

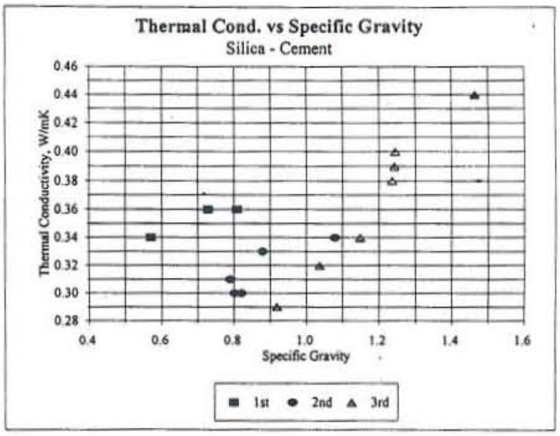

Figure 10. Weathering vs. silica content for selected silica-cement samples.

Asphalt Cement

The Marshall mix design method (ASTM D 1559) was used to evaluate the suitability of the asphalt cement (AR-4000) as an additive for a structural pavement. Various combination of aggregate, sand and silica were investigated with the silica content at 10%. Asphalt contents from 4% to almost 20% by weight of mix were used. The higher percentages were necessary to hold the mix together, as the lower percentages did not provide enough cohesion. In all cases the stability was extremely low and the flow was extremely high. Based on these results, this mix combination was not considered acceptable for use in the field.

Asphalt Emulsion

Immersion-compression tests (ASTM D 1074 and D 1075) were performed on mixtures of silica and emulsion (CRS-2H) to determine their suitability as a road surface treatment. Ten to 18% by weight of emulsion was used. All samples disintegrated during testing and thus failed the test. This use was also rejected for field testing.

CONCLUSIONS

The main conclusions from the testing are:

1. Silica-lime mixtures have low strength and weather resistance. However, they have high insulating properties. With controlled curing conditions, at ambient temperatures up to 40°C (104°F) and without loss of moisture, the strength and weather resistance improves considerably. The addition of fibers to the mixtures increases the strength and weather resistance.

2. Silica-cement mixtures have high strength and weather resistance. However, they have slightly lower insulating properties. These mixtures can better be used in load bearing wall.

3. Asphalt mixtures are not suitable using silica and thus should not be considered for any field construction.

4. Silica-cement mixtures also appear to have application as road surfacing material with the addition of an asphaltic chip seal for erosion protection.

FUTURE INVESTIGATIONS

It is proposed to test several walls constructed of silica-lime and silica-cement mixtures in the Imperial Valley area. This will provide long term field testing of the various types of bricks and determine if they need protective coatings, reinforcing, etc.

During the course of the investigation it was determined that a lightweight roofing tile using portland cement, silica and cellulose fibers is presently being manufactured in Mexico City and sold through outlets in the U.S. under the brand name "Maxitile." Their advertised advantage is that they are lighter weight (60 percent lighter than clay or concrete tile at 20 kg/m^2 [4 lbs/ft^2]). CFE is presently investigating the potential for use of the Cerro Prieto waste silica by this manufacturer.

ACKNOWLEDGMENTS

We would like to thank Hector Gutierrez Puente of CFE in Mexicali, Colin Williams of USGS in Menlo Park, CA, and Eugene T. Premuzic of Brookhaven National Laboratory in Upton, NY for their support and assistance.

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