GROUND AND WATER SOURCE HEAT PUMP PERFORMANCE AND DESIGN FOR SOUTHERN CLIMATES

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ABSTRACT

Ground and water source heat pump systems have very attractive performance characteristics when properly designed and installed. These systems typically consist of a water-to-air or water-to-water heat pump linked to a closed loop vertical or horizontal ground-coupling, an open groundwater loop, or a surface water loop. This paper discusses system performance characteristics, component selection procedures presently being used, improvements currently being considered and future possibilities for improved efficiency and reliability.

Optimum designs require proper matching of the heat pump unit to the water circulation system, the building space heating/cooling load and water heating requirements. General trends resulting from system and component choices will be discussed. Water heating methods with these heat pumps will be considered.

INTRODUCTION

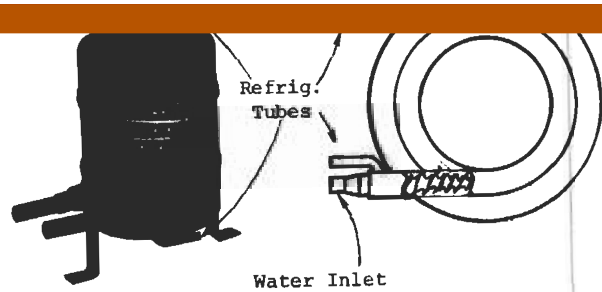
Ground and water source heat pumps are now a viable alternative to conventional cooling and heating systems. The installation and design integrity currently available has transformed these units from novel alternatives into reliable, efficient and well-packaged systems. The

The improvements in efficiency in air source cooling have caused many skeptics to question the need for ground and water source heat pumps. However, the potential for additional improvement in ground and water systems is even greater. Units currently available on the market have high dehumidification capabilities and efficiency in cooling. Heating performance in the climates of the southern United States permits the delivery of air at a comfortable temperature. Sufficient capacity is available to provide domestic hot water and auxiliary heat is typically unnecessary when the proper heat pump unit is selected.

This paper will be restricted to a discussion of the three major water loop options acceptable to the demands of systems operating in hot and humid climates: vertical ground-coupled, groundwater and lakewater. Closed loop commercial systems will not be discussed. The ground-coupling discussion is restricted to vertical because horizontal systems should generally only be used in special cases in cooling load dominated situations. These would include those installations in which deep trenching (5 feet or more) is very inexpensive, the soil remains saturated throughout the cooling season and trench lengths can be very long.

HEAT PUMP UNITS

Currently in the USA, the heat pump unit most widely used is the packaged water-to-air. Packaged water-to-water, and split systems 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 used copper (or copper-nickel) tube-in-tube coils with capillary tube expansion devices that are intended for use with groundwater with temperatures above 55° (13°C). The market today includes units capable of handling inlet solution temperature between 25°F(-4°C) to 100°F(38°C). This is accomplished with better heat exchangers, expansion devices and compressors. Two commonly used water-to-refrigerant exchangers are shown in Figure 1. Both have extended surfaces on the refrigerant side to compensate for the lower film coefficients. Figure 1(a) is a modified tube-in-shell (water on tube side) and Figure 1(b) is a coaxial tube-in-tube (water on inner tube, refrigerant in the annulus). Manufacturers are selecting heat exchangers with lower water side pressure losses to minimize pumping requirements.



a) Tube-in-Shell c) Tube-in-Tube (Co-axial)

Figure 1. Water-to-Refrigerant Heat Exchangers

Expansion valves permit a much wider acceptable range of refrigerant evaporation and condensation temperatures. This device is especially suited to ground-coupled, lakewater, 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 waste heat to generate hot water in the cooling mode or with excess heating capacity (which is 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. All these heat exchangers require vent spaces between the refrigerant and water to prevent contamination of the domestic water.

Figure 2 is included to show a typical arrangement of a packaged water-to-air heat pump. The desuperheater water heater and pump are typically available as part of the package or 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.

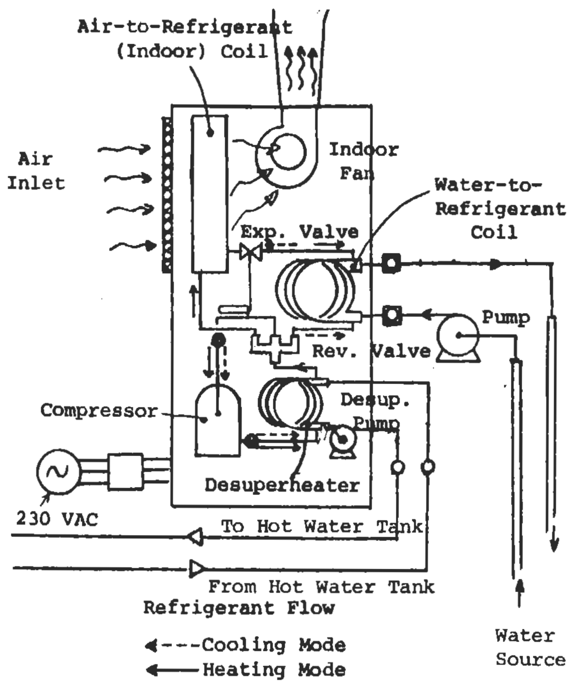


Figure 2. Water-to-Air Heat Pump with Hot Water Generator Desuperheater

PERFORMANCE

Performance characteristics of three 3.5 nominal ton (12.3 kW) water-to-air heat pumps are shown in Figure 3 and Tables 1,2 and 3. Figure 3 is a set of plots of entering water temperature vs. total capacity in thousands of Btuh. The solid line represents the performance of a unit with a tube-in-shell water-to-refrigerant coil and a bi-directional expansion valve. The dash/dot line represents a unit with a tube-in-tube coil and expansion valve. The dashed lines show the performance of a tube-in-tube unit with a capillary tube expansion device. Typically these units are not recommended for water temperatures below 55°F(13°C). Although differences in heating capacities do exist between the units, the Coefficients of Performance (COPS) are similar for a given water temperature.

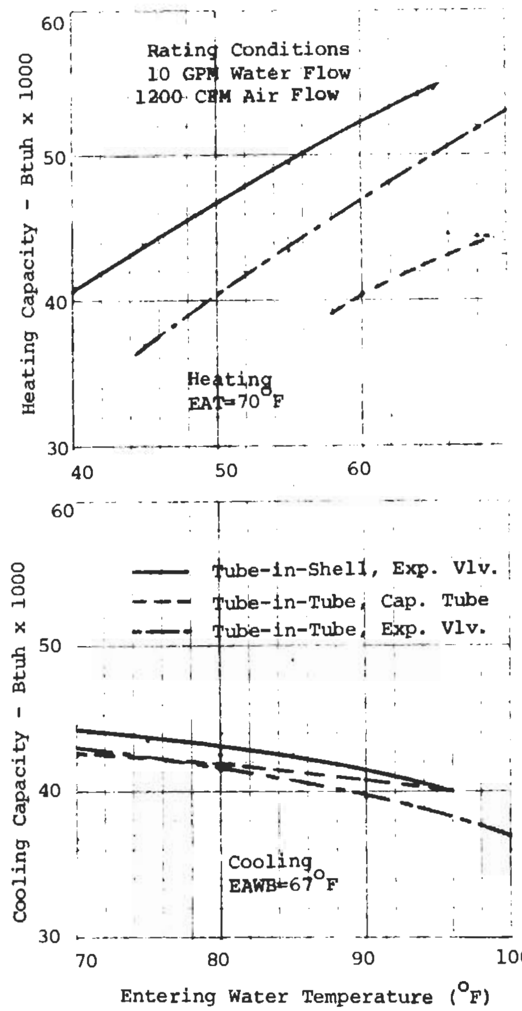


Figure 3. Heating and Cooling Capacities of Three Water-to-Air Heat Pumps

Cooling capacities of all three units are similar as shown. Cooling efficiencies, expressed as Energy Efficiency Ratio (EER), are very similar. Although the cooling capacity curve is very flat, the EER declines rapidly with increasing entering water temperature (EWT) as shown in Table 1. The inverse is true of heating efficiency. Although the heating capacity falls quickly with entering water temperature, the COP declines only slightly.

Table 1. Water-to-Air Heat Pump Efficiences

Heating		Cooling	
EWT(°F)	COP	EWT	EER(BTU/WHR)
40	3.3	70	13.7
50	3.5	80	12.2
60	3.8	90	10.8

EWT - Entering Water Temperature

The curves of Figure 3 were generated by assuming a water flow rate of 10 GPM, an air flow rate of 1400 CFM, no water heating with a desuperheater and entering air temperatures of

70°F (dry bulb) in heating and 67°F (wet bulb) in cooling. Table 2 is included to indicate trends resulting from changes in these variables. Total capacities found in Figure 3 should be multiplied by the value found in Table 2. Water flow rates below the rated value have significant negative effects on heating capacity and moderate negative effects on cooling capacity. Increasing flow rate above 10 GPM is of little value in either heating or cooling. The correction factor multipliers for varying entering air temperature are similar to air source units as shown in Table 2(b). However, the correction factors are very near unity for air flow rates 15% above and below the rated value of 1400CFM. This is quite unlike high efficiency air source units that require a minimum of 400 CFM/Ton. This permits the design of systems that can provide air at a comfortable temperature without significant penalties in capacity or efficiency.

Table 3 provides an indication of sensible capacity as a function of indoor dry bulb and wet bulb temperature. These values are lower than air source units. Thus these machines are typically well suited to humid climates.

Table 2. Capacity Correction Factors for a Water-to-Air Heat Pump

STANDARD CONDITIONS ARE UNDERLINED

Heating		Cooling	
(a) Water Flow Rate			
GPM	C.F.	GPM	C.F.
5.0	0.90	5.0	0.96
7.5	0.96	7.5	0.99
<u>10.0</u>	1.00	<u>10.0</u>	1.00
12.5	1.01	12.5	1.00
(b) Entering Air Temperature			
°F(db)	C.F.	°F(wb)	C.F.
60	1.04	61	0.90
65	1.03	64	0.95
<u>70</u>	1.00	<u>67</u>	1.00
75	0.97	70	1.04
(c) Air Flow Rate			
CFM	C.F.	CFM	C.F.
1250	0.98	1250	0.98
1325	0.99	1325	0.99
<u>1400</u>	1.00	<u>1400</u>	1.00
1475	1.01	1475	1.01

Table 3. Sensible Heat Factors for Various Entering Air Temperatures

		Dry Bulb Temperature (°F) 70	75	80	85
Wet Bulb Temp. (°F)	61	.63	.79	--	--
	64	.52	.68	.84	.92
	67	.42	.57	<u>.73</u>	.89
	70	--	.46	.62	.78

<u>HEAT PUMP SELECTION</u>

Heat pump performance is a result of a variety of factors and they should be selected based on the characteristics of each installation. Manufacturer's data must be available at the expected conditions. Conditions that are especially critical are entering water temperature and available system water flow rate.

In order to select the proper heat pump unit, a building heating and cooling load calculation must be performed. In hot and humid climates, select a unit that will meet the cooling load at manufacturer's suggested air flow rate (adjust within recommended ranges to meet sensible/latent requirements) and entering water conditions.

Entering water conditions (temperature and flow rate) can be estimated for vertical ground-couplings by finding the deep earth temperature at the site. Add 20 to 25°F to this temperature to estimate the worse case cooling mode entering water temperature (you must later size the coupling to meet these conditions). Flow rate should be a minimum of 2.5 GPM/Ton, but 3.0 GPM is recommended. The unit must meet the cooling load at the above water temperature and flow rate. Determine the heating capacity by subtracting 15 to 20°F from the deep earth temperature to estimate the entering water temperature. Water flow rate will be equal to or slightly less than cooling mode flow. If the unit does not meet the heating load, you may add auxiliary heat or select another unit that more closely matches the cooling/heating load combination. If excess heating capacity is available, it can be used to meet domestic hot water needs.

For groundwater units, local groundwater temperatures are widely available from county agents, state geological surveys or the National Water Well Association in Worthington, Ohio. Groundwater temperature will not vary significantly seasonally, unless a return injection well is placed near the heat pump supply well. A separation distance of 50 feet minimum will prevent this variation. The heat pump can then be sized to meet the cooling and heating loads using local groundwater temperatures. Water flow rates should range between 1.5 to 3.0 GPM/ton. The design rate is selected by optimizing the trade off between heat pump capacity and pumping power requirements.

For lakewater units, in southern climates, water in lakes near the surface approaches the outdoor air temperature, while water between 5 and 15 feet remains 10 to 15°F below maximum outdoor air temperature. Deep water in lakes in Alabama often remain below 50°F throughout the cooling season. Once the design lakewater temperatures are determined, three water loop options are available: open loop, closed loop with metal (copper) coils or closed loop plastic coils. As a first estimate for entering water temperature add 1°F to the lake water temperature for open loop systems with well insulated supply lines, 3 to 5°F for open loops with uninsulated lines, and 8 to 10°F for closed loop systems. Care must also be taken to properly size the heat pump unit in heating since thermal stratification may have a negative effect on lakewater temperature in southern climates. Lakes in Alabama typically range between 42°F and 50°F when undisturbed.

Additional important considerations in selecting heat pumps are head loss through the water-to-refrigerant coil, efficiency, water-to-refrigerant coil material and typical considerations such as price and arrangement.

GROUND-COUPLED HEAT PUMPS

Figure 4 depicts a residential vertical ground-coupled heat pump system. Water is circulated from the heat pump through a shallow trench (2 to 4 feet deep) into a series of vertical U-bend tubes buried in small diameter holes (3 to 5 inch). Total bore lengths are determined by accepted design methods. However, bore depths of individual loops are determined by economics resulting from drilling conditions at each site. Series flow paths are preferred in terms of thermal performance and resistance to flow imbalances or blockage. However, parallel patterns are typically installed in order to reduce pumping power requirements. Pumping power can be kept below 100 w/Ton. Installation methods have evolved and details can be found in References 1,2 and 3, and in manuals produced by many water-to-air heat pump manufacturers. Thermally fused plastic pipe is the generally accepted material.

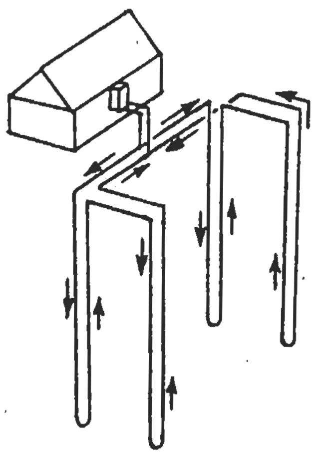


Figure 4. Vertical Ground Coupling with Series/Parallel Water Flow

SYSTEM PERFORMANCE

The performance of ground-coupled heat pump systems is difficult to predict because of the added complexity of modeling the ground-coupling. The building cooling and heating load dictate the load the heat pump places on the ground-coupling, which is a primary factor in determining the performance of the heat pump. (The longer the heat pump stays on, the more the return water temperature and capacity degrade). The result is that the heat pump performance model must be linked to the building load and ground-coupling models in an iterative loop. The process is further complicated because the large thermal mass of the ground requires that the system be simulated over a period of several months before design conditions are reached.

General daily and seasonal performance trends can be seen in Figure 5. This figure indicates typical entering water temperatures of a vertical 600 ft. (180m) ground-coupling linked to a 45,000 Btu/hr (13 kW) water-to-air heat pump in central Oklahoma. The figure shows daily variation in temperature during an extended period (7 weeks) of operation in excess of design cooling load. Note the gradual increase of peak temperature with time and the pattern of daily variation. The normal ground temperature at the site is 62°F (17°C). Heat pump performance can be deduced using the water temperatures in conjunction with manufacturer's data similar to Figure 3.

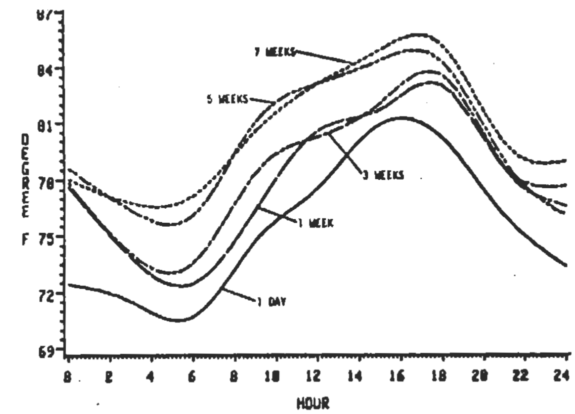


Figure 5. Heat Pump Entering Water Temperatures during Peak Cooling Load - Oklahoma

Figure 6 indicates the effectiveness of ground-coupled heat pump in the heating mode. This system is a 19,000 Btu/hr (5.6 kW) heat pump linked to a 250 ft. (75m) vertical ground-coupling in Oklahoma. While the air temperature is 2°F (-17°C), the water temperature entering the heat pump is 48°F (9°C). The unit will operate with a system COP above 3.4.

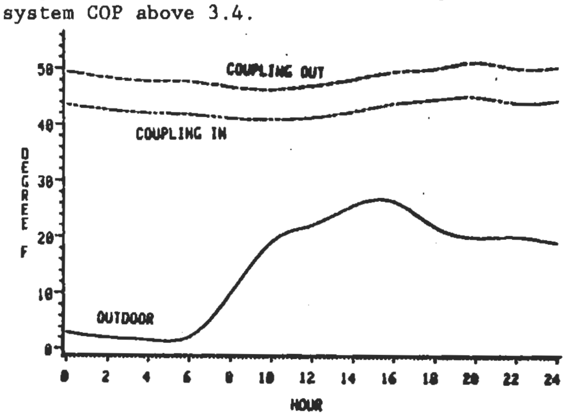


Figure 6. Heat Pump Water Temperatures During Peak Heating Load - Oklahoma

Similar performance characteristics have been observed in installations in Alabama and the Florida panhandle. A three-ton unit in

Tallahassee with a 600 ft.-3/4 inch ground-coupling operates approximately four degrees warmer than the installation in Oklahoma in the cooling mode. A similar size one inch ground-coupling installation in Tuscaloosa, Alabama operates at approximately the same temperature as shown in Figure 5 in cooling and 5°F warmer than those of Figure 6 during peak winter conditions.

In addition to being more efficient than air source heat pump systems, ground-coupled units provide a greater level of comfort. Air delivery is warmer in the winter and dehumidification capacity is higher in cooling. Part load performance and efficiency is high because of the natural temperature lag in a ground-coupling. During off periods heat transferred to the soil continues at a high rate. When the unit restarts entering water temperature is closer to the deep soil temperature for the amount of time that it takes a "lump" of fluid to circulate through the ground-coupling. Typical values are 5 to 15 minutes.

SYSTEM DESIGN METHODS

Methods of system design are summarized by Ball, Fischer, and Hodgett [4] and by Bose, Parker, and McQuiston [5]. Three levels are identified as empirically derived standards, modified analytical solutions and numerical methods using finite element or difference models of the ground-coupling and surrounding soil. Empirically derived standards (i.e. 175 ft/ton) have proven to be successful in areas and applications in which ground-couplings have been widely used. However, more detailed analysis is often needed when new climates, geological conditions and applications are encountered.

Several numerical models are listed in Reference 4. A good model is necessarily complex. Any model is seriously compromised by the uncertainty in ground properties (thermal conductivity, density, specific heat, moisture content, water velocity.) Therefore, in order to design a system within the accuracy expected of these numerical methods, it is necessary to sample ground conditions extensively. The opinion of this author is that numerical methods of design are an "overkill".

A compromise is utilized by the authors of Reference 6. Use of the Kelvin Line Source solution of a constant heat rate line source in an infinite medium was proposed by Ingersoll [3]. The solution allows the temperature difference between the outside pipe wall and the undisturbed soil to be approximated by:

$$t_r - t_u = \frac{q_{gc}}{2\pi k_g L} \int_{r/2\sqrt{\alpha T}}^{\infty} \exp(-\beta)/\beta \, d\beta \qquad (1)$$

which is simplified to:

$$t_r - t_u = \frac{q_{gc}}{2\pi k_g L} I(X) = \frac{RF \times q_{gc} \times R_g}{L} \qquad (2)$$

Where RF is the fraction of time the heat pump is running, q_{gc} is the heat being transferred in the coupling (heat rejected [HR] in cooling and heat extracted [HE] in heating), k_g is the ground thermal conductivity, R_g is the thermal resistance of the ground and L is the coupling bore length.

In addition to the temperature difference between the outside pipe radius and ground (t_r - t_u), a temperature difference exists between the water and outside pipe. Table 3 lists the pipe thermal resistance (R_p) for several size and types of pipes used with U-bend ground-couplings. This assumes the water flow is sufficient to cause transition or turbulent flow (mixing) and internal resistance is negligible. When flow is not laminar the temperature difference between the water inside the coupling and the pipe wall becomes:

$$t_w - t_r = q_{gc} x R_p / L \qquad (3)$$

Equations 2 and 3 can be combined and rearranged so that for cooling,

$$L_c = HR\ (R_p + RF \times R_g)/(t_w - t_u) \qquad (4)$$

and for heating,

$$L_h = HE(R_p + RF \times R_g)/(t_w - t_u) \qquad (5)$$

HR and HE should be provided by the heat pump manufacturer and RF is found in Reference 3 or estimated from monthly bin or degree-day data. The value for R_g is:

$$R_g = I(X)/(2\pi k_g) \qquad (6)$$

I(X) can then be found by calculating the dimensionless value

$$X = D_{eq}/(4\sqrt{\alpha T}) \qquad (7)$$

and entering this value on Figure 7 to find I(X). Values for D_{eq} are found in Table 4; α and k_g are found in Table 5. T is the time in days (if α is in ft^2/day) during which peak heating and cooling loads occurs. Thirty or thirty-one days are typical values used for design.

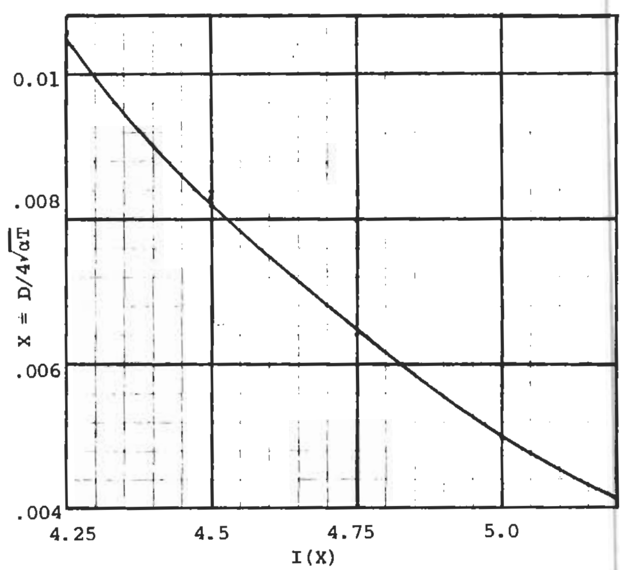


Figure 7. Graphical Results of Line Source Integral for Ground-Coupling Design

Table 4. Equivalent Pipe Thermal Resistances and Diameters for U-Bend Ground-Couplings

Diameter (in.)	SDR(Sch) (OD/t_{wall})	Eq. Dia. (ft.)	Eq. Therm. Res (hr-ft°F/Btu)
High Density Polyethylene - PE 3408			
3/4	9	.12	.12
"	11	"	.096
"	(Sch 40)	"	.116
1	9	.15	.12
"	11	"	.096
"	(Sch 40)	"	.109
1-1/2	9	.22	.12
"	11	"	.096
"	(Sch 40)	"	.08
Polybutylene - PB 2110			
3/4	11	.12	.16
"	13.5	"	.12
1	11	.15	.16
"	13.5	"	.12

Table 5. Equivalent Soil Thermal Properties

Material	Conductivity (k_g) (Btu/hr-ft°F)	Diffusivity (α_g) ft^2/day
Dense Rock	2.0	1.2
Rock	1.6	1.0
Wet Clay	1.4	0.75
Wet Sand*	1.2	0.7
Damp Clay	.8-1.0	.5-.6
Damp Soil	.5-.8	.4-.5

*Water movement will substantially increase heat transfer capability

The procedure followed would be to select a heat pump unit that meets the building cooling load with an entering water temperature (EWT) 20° to 25°F above the local deep earth temperature (t_u) which also meets the heating load with an EWT 15° to 18°F below t_u. The value of EWT is substituted for t_w in equations 4 and 5. A ground-coupling type is selected based on contractor preference. The required lengths (L_c, L_h) can be found with the greater of the two being the actual length needed.

THIS METHOD IS NOT EXACT BECAUSE OF THE COMPLEXITY OF DETERMINING THE VALUES GIVEN IN TABLE 4. THIS METHOD ASSUMES THERE IS A COMBINATION OF HEATING AND COOLING REQUIREMENTS ON THE COUPLING SO THAT A THERMAL REGENERATION OCCURS. IT SHOULD NOT BE USED FOR COOLING ONLY OR HEATING ONLY APPLICATIONS. MORE DETAILED SIMULATIONS ARE REQUIRED.

If you do not wish to consult a designer with coupling length calculation, the above method is acceptable but usually conservative. A suggestion would be to install a coupling in your geographic location, monitor water temperatures, heat rejection and extraction rates, and run fractions. You may then be able to determine if shorter coupling lengths are acceptable in residential and light commercial applications.

Often the greatest concern of ground-coupling design is the piping system layout and pump selection. It is possible to install a system that provides the recommended flow rate of 3 GPM/Ton with a power consumption less than 10% of the heat pump unit total. Standard methods of piping head pressure loss apply.

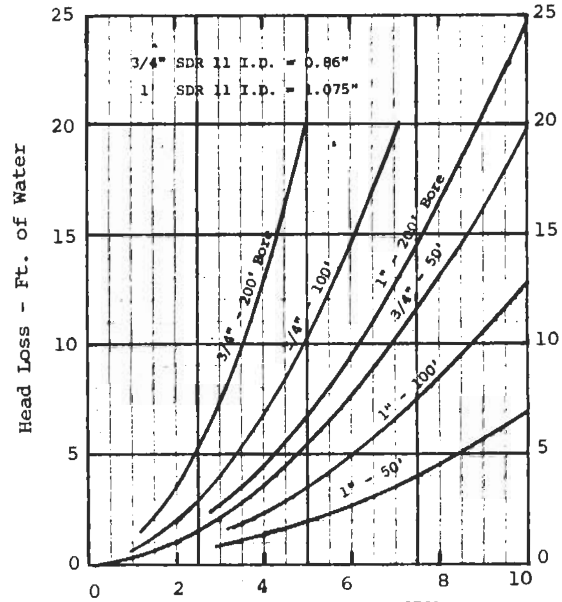


Figure 8. Head Loss Through Vertical U-bend Ground-Couplings

GROUNDWATER HEAT PUMPS

Until recently, groundwater heat pumps were the most common water source system. Figure 9 shows the major components to consist of a supply water well, a submersible or jet pump, a lightning arrestor, a water control scheme, a water-to-air heat pump, and a water disposal system. Water disposal may be accomplished at the surface (lakes, streams, irrigation systems) or

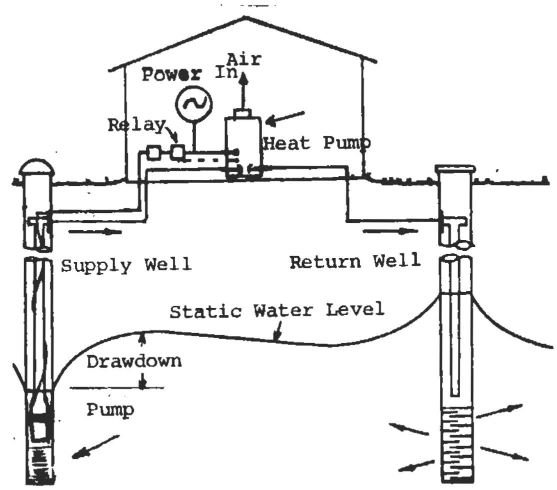


Figure 9.Groundwater Heat Pump System

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sub-surface (injection well, field lines, into the supply well in a different aquifer).

In terms of cooling and heating capacity, groundwater systems are clearly the best alternative. This results from the stable water temperatures supplied to the heat pump. Cooling capacity can be boosted further by pre-cooling return air directly with the groundwater. However, benefits are not significant with groundwater temperatures above 65°F (18°C) thus eliminating this possibility in most hot and humid climates.

Groundwater use for water source heat pumps is regulated in some states and municipalities, especially when injection wells are used. Some states have investigated the systems and determined them to be environmentally acceptable for residential applications (7). Some municipalities prohibit private water well installations. Additionally, groundwater systems have a reputation of requiring frequent maintenance because of fouling and control system malfunction. A third problem that often results from poor design is low efficiency due to massive pump oversizing.

SYSTEM DESIGN

The attractiveness of groundwater heat pump systems can be improved by design that effectively deals with the previously mentioned problems. Efficient and low maintenance systems require the installation of simple (and low cost) control schemes, elimination of unnecessary head losses, proper heat pump and water pump selection and, above all, proper water well development. Good development can reduce fouling, required pump size and control system failures.

Proper water well installation consists of drilling to the shallowest acceptable aquifer, proper well screen selection, installation of coarse sand around the well screen (gravel packing) where necessary, and well development to remove fine particles near the well screen. If these "fines" are not removed, water flow into the well will be restricted. Thus the drawdown on the well (Figure 9) will increase. This will limit the capacity of the well and result in a larger required pump size to overcome the loss in available head at the pump suction. "Fines" may also cause heat exchange fouling, component wear and injection well plugging. It is necessary to specify that the well driller develop the well by continued water flow reversals through the well screen. This surging action removes "fines" that may be bridged in the gravel pack. Pumping in only one direction will not adequately develop water wells. Development is typically accomplished by alternately pressurizing and pumping the well with air or water.

Water flow control schemes should be simple in open loop systems to minimize maintenance. The majority of the water loops installed in the U.S. are identical to domestic water well systems. The pump discharges into a bladder tank and with a solenoid valve located downstream. When the heat pump cycles on the valve opens and the tank pressure falls below the set point of a pressure switch. The pressure switch "makes" and the pump is cycled on. A slight modification of this would be to replace the solenoid valve with two water regulating valves. A direct acting valve will open with increasing compressor discharge pressure and the reverse acting opens with decreasing suction pressure. This system is recommended when capillary tube expansion devices are used in the heat pump.

Figure 9 includes a schematic of a system that is an improvement over the above systems. The water pump is activated through a relay that closes with the compressor relay. The advantages are low cost, no control valves to malfunction (primary cause of system failure), reduced number of pump starts (primary cause of shortened pump life), reduced water head losses (lower required pump size) and quiet operation (no water surges).

Proper water pump selection for efficiency and long life can be accomplished if drawdown is limited, pipes are properly sized, and no unnecessary restrictions are placed in the circulation loop. Submersible pumps are recommended over jet pumps because of their higher efficiency and reliability. These pumps consist of a series of impellers of a given flow rate stacked until proper head is available. A 1/2 horsepower (0.37 kW) pump could be 13 stages of 5GPM (19LPM) impellers stacked to deliver 280 ft.(85 m) of head or 6 stages of 12 GPM (45LPM) impellers that will deliver 140 ft. (43m). Proper design requires that pump be selected to provide 2 to 3 GPM/Ton and only enough head to circulate water through the system. Regulation of flow with valves is an unnecessary waste of energy. However, pump flow should not exceed manufacturer's recommended values. This will cause motor overload. In some cases stages may be removed from standard pumps if excess head is available. This is a good way to unload the pump motor and reduce energy consumption.

LAKEWATER HEAT PUMPS

Lakewater heat pump systems are an attractive alternative to individuals living near acceptable water sources. Design and installation information is much more limited than with ground-coupled and groundwater systems. As mentioned previously, several options are available and the thermal behavior of water bodies should be understood before design alternatives are discussed.

THERMAL PATTERNS IN RESERVOIRS

Figure 10 shows the results of a thermal study of a small lake in central Alabama (9). The climate is hot and humid with a moderate heating requirement (2500°F-days). Note that the upper portion of the lake is near 85°F (29°C) during the primary cooling period. However, the temperature remains below 54°F(12°C) during this period below the thermocline (10-25 feet). The temperature during the heating season remains above 46°F (8°C) in the entire lake. This pattern is common to lakes deeper than 30 feet (9m) with moderate inflows. However, shallower lakes tend to approach the average daily outdoor air temperature as do rivers and streams with high water velocities.

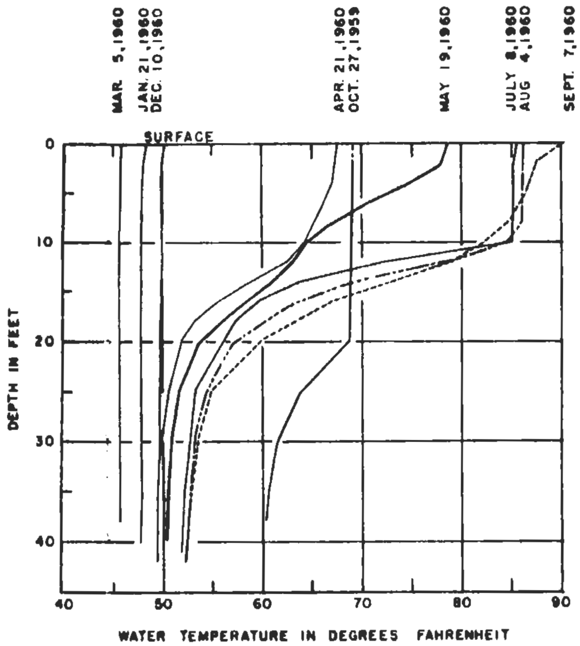

Figure 10. Seasonal/Depth Temperature Variations in Central Alabama Lake

Figure 11 shows the basic components of an open loop lakewater system. Several water pump arrangements are also shown. Filtration is a primary concern in these systems and extensive studies concerning acceptable methods have not been located. However, systems have been observed operating several seasons in Alabama with a minimum of precaution. An acceptable practice is to incorporate a fine slot well screen (0.012 inch or smaller) as a pump suction filter. Care must be exercised in keeping the screen off the bottom of the lake. As shown in Figure 11, possible pumping arrangements are a centrifugal pump with sufficient lift capacity, a submersible pump located on the bottom of the lake (it must be rated for non-vertical operation), a submersible pump near the surface with or without a sufficiently sized suction line to prevent cavitation, or and a similarly arranged vertical pump. Supply lines to the heat pump should be insulated if exposed or if direct or pre-cooling is attempted.

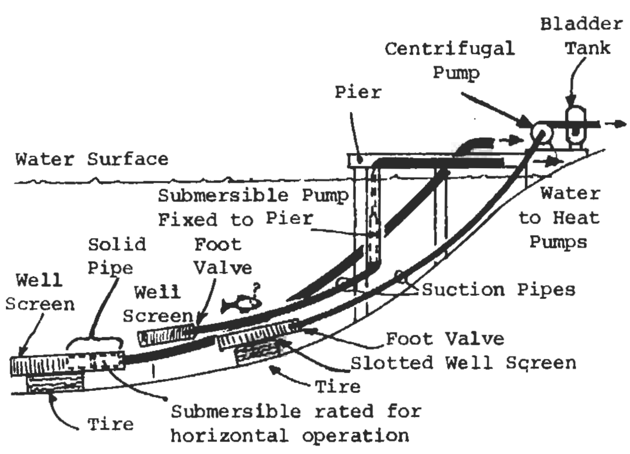

Figure 11. Three Pumping Arrangements for Open Loop Lakewater Heat Pump Systems

Closed loop systems are also an excellent alternative. They are chosen over open systems to reduce (or prevent) heat exchanger fouling and lower pump power requirements. The penalties for selecting these systems are the greater possibility of damage when used in public lakes and an 8^{o}F to 12^{o}F temperature source/sink degradation. A survey was performed by contacting individuals involved in closed loop systems. The consensus guidelines are 60 ft. of copper pipe or 250-300 ft. of plastic pipe per ton of heat pump, 3/4 inch (2.0 cm) to 1.5 inch (4 cm) diameter pipes, 10 to 12 ft. (3 to 4 m) minimum lake depth and a minimum lake surface area of 1 acre per residence. Systems installed in Alabama according to these guidelines appear to be operating properly when circulation systems are able to provide 2.5 to 3.0 GPM/Ton. Table 6 lists the results of tests performed on 3 shallow lake open-and closed-loop systems in Central Alabama. Deep lakewater systems are currently being monitored.

Table 6. Performance of Shallow Lakewater Heat Pumps Systems (8)

System Description	OAT(^{o}F)	EWT(^{o}F)	LWT(^{o}F)	COP (EER)
3 Ton Closed Loop 1000' PB Coil	25	44-49	40-44	2.9
8-10' Below Surface	97	90-95	99-105	2.7 (9.2)
4 Ton Open Loop 1/2 Sub, Pump	20	44	40	2.6
2 acre x 20' deep lake - 18'suction	93	62	70	3.3 (11.3)
4 Units on Open Loop 2,2.5,2.5 & 3.5 Ton One 1 hp cent. pump	30	44-48	39-42	3.2
25 acre x 10' deep lake - 6' suction	94	82	93	2.8 (9.6)

OAT - Outdoor Air Temperature
EWT - Water Temperature Entering Heat Pump
LWT - Water Temperature Leaving Heat Pump

LAKEWATER SYSTEM DESIGN

Open loop system design primarily involves heat pump selection as discussed previously, pump specification, determining the need for pipe insulation, and control system design. Above surface centrifugal pumps require three primary considerations: providing sufficient capacity to overcome system head loss, preventing loss of pump suction during off periods, and preventing cavitation. Head losses are found by adding piping and fitting losses, coil losses and the elevation between the lake and point of discharge. The point of discharge (below the heat pump) should be limited so as to prevent the system from operating at excessively high vacuums. Suction prime can be maintained in above surface pumps by placing a foot valve in the lake and a bladder tank and solenoid valve on the pump discharge. Control is accomplished identically to the domestic ground water system described previously. Should the foot valve leak when the pump is off, water from the tank will be forced from the tank into the suction line. The loss of pressure in the tank will cause the pump to cycle on before suction is lost. Excessive foot valve leaking will be indicated by pump short cycling.

Pump cavitation can be prevented by insuring the Net Positive Suction Head (NPSH) available exceeds NPSH required. NPSH available is:

$$NPSH(ft.H_2O) = 1.13BP(in.Hg) - h_l - \Delta h_{suc} \quad (8)$$

where BP is barometric pressure, h_l is the height between the pump suction and lake surface and h_{suc} is the piping head loss across the suction piping, foot valve and screen. NPSH required is supplied by the pump manufacturer as a function of water flow rate.

The precautions concerning suction loss are not as critical when submersible or vertical pumps are used. However, if suction lines are used, they must be sufficiently large to insure NPSH requirements are satisfied. Bladder tanks are unnecessary and pump can be cycled on with a relay.

Closed loop design involves the calculation of the overall heat transfer coefficent between the lakewater and the fluid circulating in the loop. The relative magnitude of the components limiting the lake coil heat transfer rates can best be seen by incorporating the thermal resistance concept as shown in Figure 12. The heat transfer in a horizontal pipe is given by:

$$q = (t_w - t_\infty)/(R_i+R_p+R_o), \quad (9)$$

where

$$R_i = 1/(2\pi r_i L h_i)$$
$$R_p = \ln(r_o/r_i)/(2\pi k_p L)$$
$$R_o = 1/(2\pi r_o L h_o).$$

Calculation of the internal coefficent, h_i, is somewhat inaccurate since closed loop design guidelines dictate flow rates in the transition flow regime (3000<Re<8000). However, the internal thermal resistance (R_i) is small compared to R_o (and R_p for plastic pipe) and the resulting error in the heat transfer rate, q, is small. Acceptable accuracy can be obtained by curve fitting the results of Sieder and Tate [10] in the transition regime.

Calculation of R_p is simple if the thermal conductivity of the pipe (k_p) is known. Calculation of the outside coefficent can be obtained by using the simplified natural convection coefficent formula:

$$h_o = (k_w/2r_o)C(GrPr)^m \quad (10)$$

For water and pipe sizes characteristic to lake coils, C = 0.53 and m = 0.25. The water thermal conductivity can be found in tables or calculated

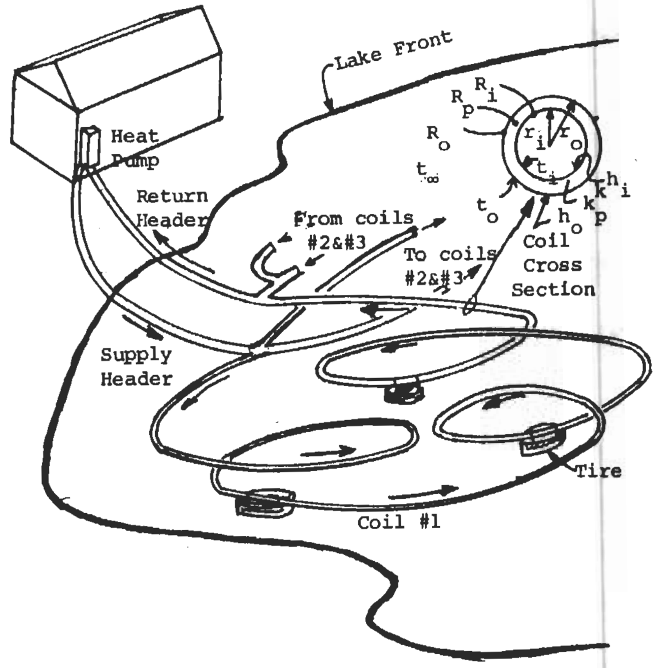

Figure 12. Closed Loop Lakewater Heat Pump System with Three Parallel Loops (One Shown)

from the curve fit equations for 35<t<100°F:

$$k_w = .304 + .781 \times 10^{-3}\, t(°F) - .177 \times 10^{-5}\, t(°F)^2 \quad (11)$$

Units for k_w are Btu/hr-ft-F.

The product of the Grashof-Prandtl numbers can also be calculated at these temperatures by,

$$GrPr = (3.61\times10^7 + .163\times10^6 t - 85.1\, t^2)(2r_o)^3(t_o - t_\infty) \quad (12)$$

The temperature (t) in the above equations is the average between the lakewater temperature (t_∞) and the outside coil surface temperature (t_o) in °F. The value for GrPr must be found in an iterative manner since t_o is unknown. Furthermore, the value for t_w changes as heat is transferred to or from the coil. The calculation is readily accomplished with a microcomputer. Figure 13 shows results from a Basic code using the above equations. The program permits the selection of a

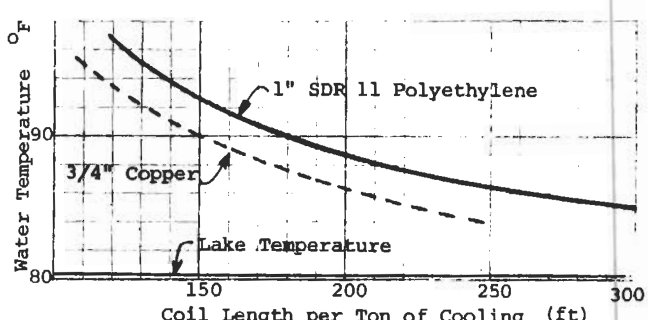

Figure 13. Lake Heat Pump Entering Water Temperatures

coil length by inputing heat pump performance, lake temperatures and pipe characteristics.

Installations observed in Alabama typically have sufficient coil length. A reoccurring flaw observed is insuffticent water flow. This problem can be remedied by increasing the number of loops in parallel. However, velocity must be sufficient to prevent laminar flow and air entrapment.

CONCLUSIONS

Water source systems are efficient and reliable systems when properly installed. Flexibility, performance, and economics are attractive when compared with systems currently on the market. The potential for improvements in these areas is very promising. The greatest need is for development of water loop contractors who can properly select, design, and install systems economically and reliably. A second need is for increased research and development with water source heat pump units.

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