EXPERIMENTAL STUDY OF DIRECT-EXPANSION GROUND COIL HEAT EXCHANGERS

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ABSTRACT

Direct-expansion ground coil (DXGC) ground-coupled heat pump (GCHP) systems have certain energy-efficiency advantages over conventional GCHPs. Principal among these are the elimination of the secondary heat transfer fluid heat exchanger and circulating pump. While the DXGC concept can have higher efficiencies, it also has more system design and operational problems (compressor starting and oil return problems). General design guidelines for DXGC systems are not well documented.

This study was designed to experimentally look into some of the advantages and disadvantages of DXGC designs. A laboratory study of a DXGC heat pump system was performed. The results of the laboratory tests indicated that a DXGC can absorb from, or dissipate to, the ground at least twice as much heat as conventional plastic tube ground coil heat exchangers. Some of the above-mentioned concerns were also studied during the tests. A set of qualitative general design guidelines was derived from the test results and observations and is presented in this report.

INTRODUCTION

Ground-coupled heat pump (GCHP) systems have long been recognized for their high energy-saving potential. Many types of ground coil designs are possible, but for heat pump systems, only two basic design concepts have been implemented so far. They are direct-expansion ground coils (DXGC) and secondary fluid circulating ground coils. A primary advantage of the DXGC over the secondary fluid systems is its inherently higher efficiency due to elimination of the secondary fluid heat exchanger and circulating pump. However, it has some significant application problems.

Early GCHP work (before 1970) dealt almost exclusively with heating. Most of the studies involved field tests using horizontal-type DXGCs (Ingersoll and Plass 1949; Guernse 1949; Coogan 1949; Smith 1950). Cooling mode operation was difficult due to cold initial ground temperature, which results in excessive condensing. The compressor would be starved of refrigerant and thus shut off by a low-pressure cut-off switch (Freund and Whitlow 1959). Corrosion of metal coils by the groundwater was also a major concern, since there is no easy way to repair a coil once it is installed. Later GCHP designs, with a secondary circulating fluid and an intermediate heat exchanger, were able to operate for both heating and cooling modes. The advent of plastic ground coil materials, such as polyethylene and polybutylene, practically eliminated coil leakage and corrosion problems for the secondary fluid systems. The plastic materials can easily be joined via heat fusion to form leakproof joints and last indefinitely underground.

Recently, the development of new technologies to handle the problems associated with DXGCs has led to renewed interest in this design. A recent study (Ratliff 1986) indicated that unless the local soil or groundwater contains highly oxidizing chemical compounds, there is practically no corrosion problem for buried copper coils. System-starting problems for cooling mode operation can be solved by two methods. One is to use shallow bores instead of deep ones. The compressor can lift refrigerant out of such short ground coils with much less difficulty. Another way is to use vertical tube-in-tube coils coupled with an oversized accumulator and a large refrigerant charge. During the compressor's off-period, the refrigerant will drain back to the ground coils from the accumulator. This will flood the ground coils with refrigerant and thus reduce the vertical lift required by the compressor.

While many field tests have been performed, a well-instrumented laboratory test of the concept has never been implemented before. In addition, there are no general guidelines available for DXGC designs.

The experimental results of a laboratory-scale DXGC heat pump system design with four shallow vertical wells are reported in this paper. Two types of coil—tube-in-tube and U-tube—were tested for both cooling and heating mode operation under various heat pump operating conditions. The coils were tested in parallel and with two sets of two coils in series.

The laboratory tests indicated that the DXGC concept is feasible with high system efficiency. The technical concerns associated with this concept were closely observed throughout the experiment. Results of the laboratory tests were reduced to produce general DXGC design guidelines.

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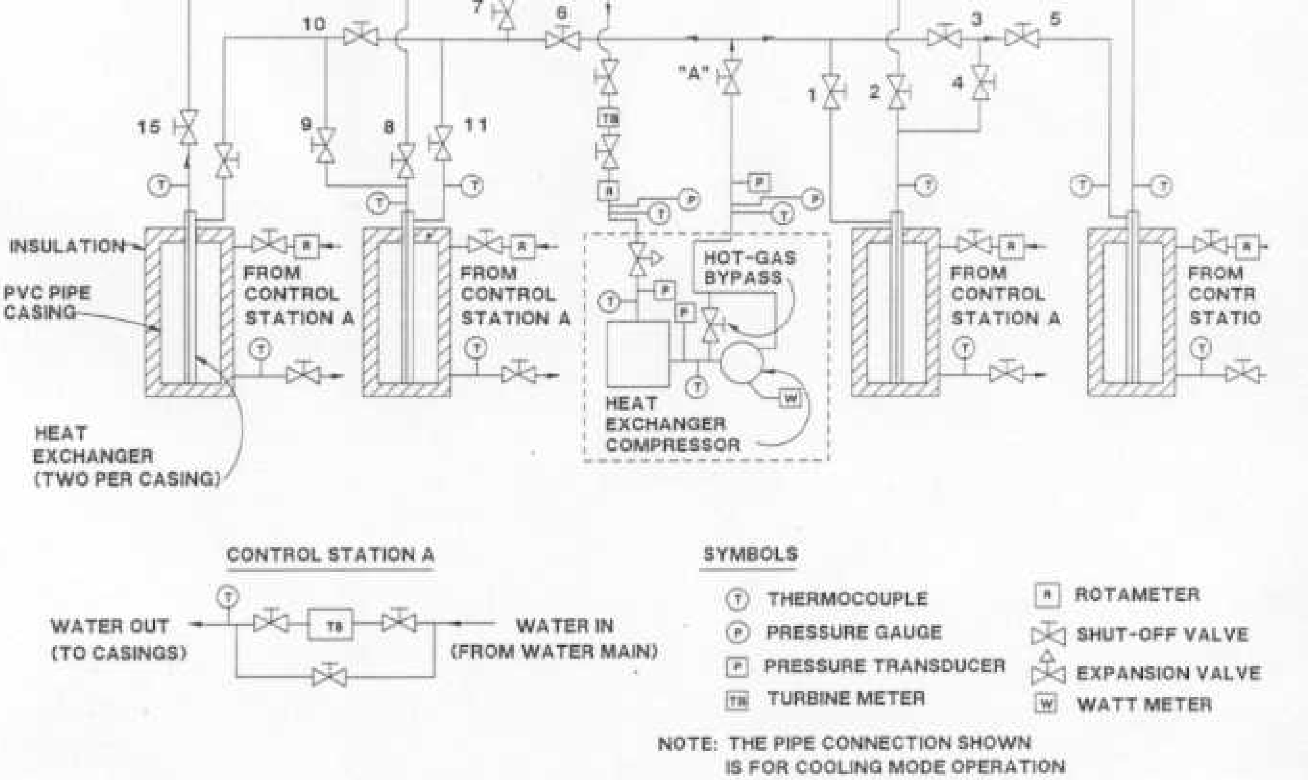

Figure 1 *Schematic of laboratory test setup*

DESCRIPTION OF LABORATORY DXGC HEAT PUMP SYSTEM DESIGN

The laboratory system was designed for a cooling capacity of 1/2- to 3/4-ton. The coil size selection was based on calculation of refrigerant velocity recommended by the *ASHRAE Handbook* (ASHRAE 1985) to ensure oil return. The system was designed in such a way that the four coils could be operated in parallel or with two sets of two coils in series. Tests were performed in two parts—cooling mode and heating mode. Cooling mode tests were performed first. The system piping was then altered to perform the heating mode tests. Figures 1 and 2 show the schematic and the actual test setup. The major system components and design features are as follows.

Heat Pump

The unit is a commercially available heat pump water heater with a 1-ton-capacity compressor. An accumulator was added to the system. The air-to-refrigerant heat exchanger was removed, and the refrigerant lines were connected directly to the test coils. A hot gas bypass line was installed to provide a quick equalization of the high- and low-side pressures. It also provided some capability for capacity modulation.

Ground Simulator

Four PVC tube columns, 15 ft (4.6 m) long and 4 in. (10 cm) in diameter, were used to simulate the ground, as shown in Figures 1 and 2. The tubes were insulated with 1-in. (2.5 cm) expanded rubber. Water flowed into the tubes from the top and drained from the bottom. By controlling the inlet water temperature and water flow rate, the system operation at different ground conditions could be simulated.

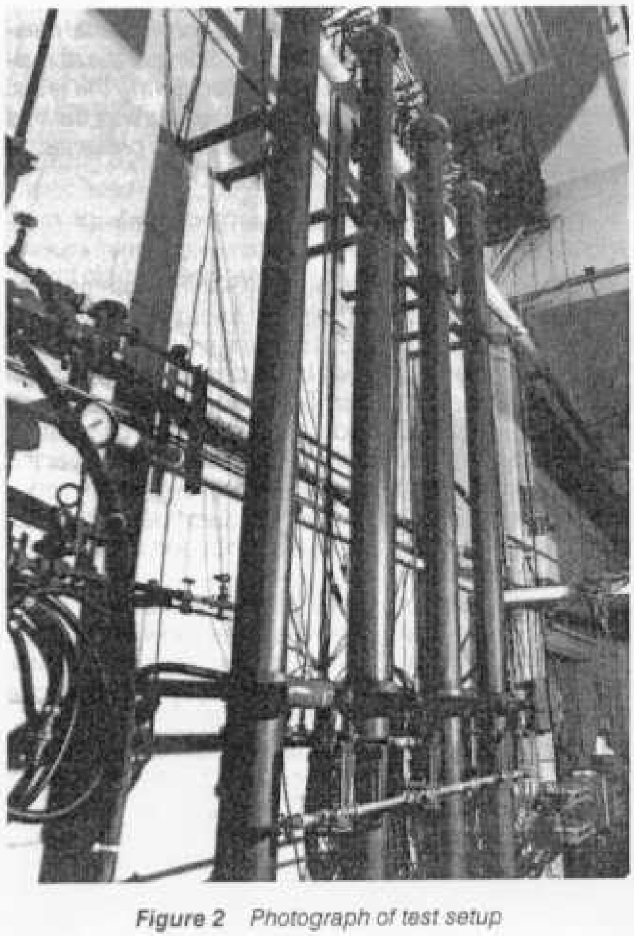

Figure 2 *Photograph of test setup*

Coils

Two sets of 15-ft (4.6 m) copper coils—one tube-in-tube and one U-tube—were installed in each column. The tube-in-tube coils were made of 1/4-in. (0.6 cm) inner tube and 3/8-in. (0.9 cm) outer tube. The U-tube was made of 1/4-in. (0.6 cm) tube. The coils could be operated in parallel or as two sets of two coils in series.

Water Supply

The water temperature and flow rate to both the indoor refrigerant-water heat exchanger and the columns were controlled separately. Process water with a temperature range from 60°F to 120°F (15.6° to 48.9°C) was available in the laboratory. System capacity could be adjusted by varying the water flow rate or water temperature, or both.

Flow Rate Measurement

Three turbine meters were used for refrigerant and water flow rate measurements. In addition, four rotameters were used for measuring the water flow rate to each column.

Pressure Measurement

Three pressure transducers were used to measure the pressures before and after the refrigerant expansion device and the compressor suction pressure. Each coil had additional pressure gauges installed to measure the pressure loss across the coil during operation.

Temperature Measurement

All temperatures were measured with type-T (copper-constantan) thermocouples. There were four thermocouples bonded on each coil. Each column, therefore, has eight thermocouples. During the test of the tube-in-tube coils, for example, the thermocouples on the U-tube provided the water temperature readings along the column and vice versa.

Data Acquisition System (DAS)

A DAS system with a computer was used for the experiment. The DAS collected and reduced the test data.

TEST PROCEDURES

Cooling Mode Tests

The inlet water temperature and flow rate to the heat pump indoor water-refrigerant heat exchanger were maintained at 62° to 65°F (16.7° to 18.3°C) and 3 gpm (0.19 L/h), respectively. Total water flow rate to the four columns was maintained at 3 gpm (0.19 L/h). However, the column inlet water temperature was varied from 65°F to 120°F (18.3°C to 48.9°C). At each temperature level, data were collected until the system had reached steady-state operation. The water temperature was then adjusted, and data collection started again at the new water temperature level. Column inlet water temperatures were used as the independent quantity on which other measured data were based.

Heating Mode Tests

The inlet water temperature and flow rate to the heat pump indoor water-refrigerant heat exchanger were maintained at 90 ±1°F (32.2 ±0.6°C) and 3 gpm (0.19 L/h). The inlet water temperature to the columns was maintained at 62° to 65°F (16.7° to 18.3°C). However, the total water flow rate varied gradually from 3 gpm (0.19 L/h) to less than 0.4 gpm (0.025 L/h). At each column water flow rate, the data were collected when the system started until the operation had reached steady state. The water flow rate was then adjusted and the data collection started again at the new water flow rate. Since the column water flow rates were changed for each test, the average column exit water temperature was used as the independent quantity on which the other measured data were based.

The procedures were repeated for both tube-in-tube and U-tube coils and for both parallel coil and two sets of two coils in series operation.

TEST RESULTS AND DISCUSSION

Cooling Mode Test Results: Parallel Refrigerant Flow

Figure 3 shows the ground coil heat dissipation rate per unit length of well for parallel refrigerant flow operation for both tube-in-tube and U-tube coils.

For tube-in-tube coil, parallel refrigerant flow operation, the heat dissipation rate ranged from 190 to 165 Btu/h•ft (183 to 159 W/m) for column inlet water temperatures of 62.7°F to 77.4°F (17.0°C to 25.2°C). The unit operated at very high discharge pressure. It was at 350 psia (2413 KPa) even with a column water inlet temperature as low as 62.5°F (16.9°C). When the water temperature reached 77°F (25.0°C), the discharge pressure increased to 400 psia (2758 KPa). Further increase of the inlet water temperature resulted in compressor cutoff by a high-pressure switch. For U-tube coil parallel flow operation, the coil heat dissipation rate ranged from 204 to 192 Btu/h•ft•well (196 to 185 W/m), higher than that of the tube-in-tube coils. The discharge pressure is about 30 psi (207 KPa) lower than the tube-in-tube coils.

There are two reasons for this difference. One is that the U-tube coils had a smaller coil diameter than the tube-in-tube coils (1/4 in. vs. 3/8 in. [0.6 cm vs. 0.9 cm]) but twice the length in the same well, which resulted in about 33% more heat transfer area than for the tube-in-tube coils. In addition, the tube-in-tube coil experienced a "short circuit"

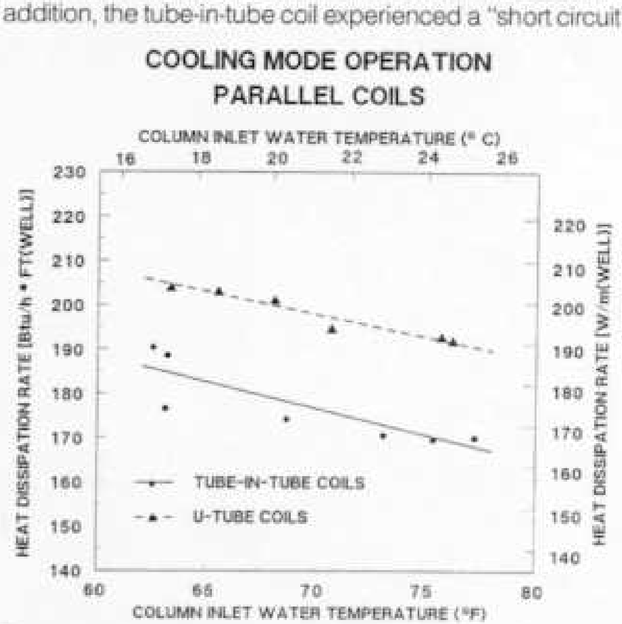

Figure 3 Cooling mode parallel coils—heat dissipation rate as a function of column inlet water temperature

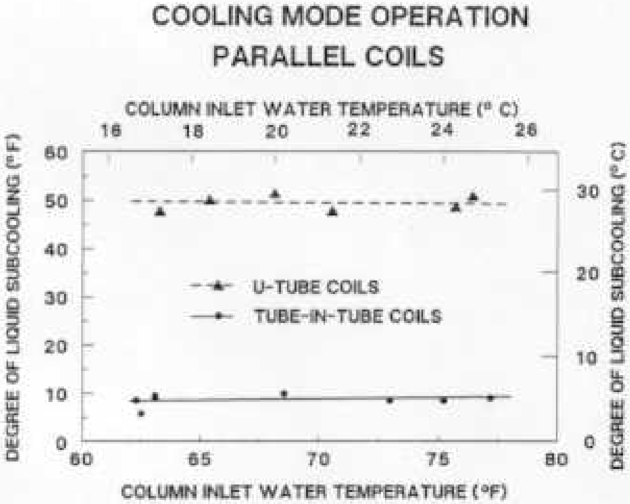

Figure 4 *Cooling mode parallel coils—degree of liquid subcooling as a function of inlet water temperature*

heat exchange between the refrigerant in the inner coil and in the annulus region. The much lower liquid subcooling level of the tube-in-tube coils, as shown in Figure 4, indicates that there was a serious heat transfer short circuit, which adversely affected the coil heat dissipation capability. The pressure drop across each set of coils was almost constant at 10 to 11 psi (69 to 76 KPa) for both tube-in-tube and U-tube coils.

Figure 5 shows the system coefficient of performance (COP) as a function of the column inlet water temperatures. As the column inlet water temperature increases (equivalent to the warming of the ground temperature), the COP drops linearly. The U-tube coils yielded higher system COPs than the tube-in-tube coils. Again, this was due to the heat transfer short-circuit and lower heat transfer area of the tube-in-tube coils. Use of an evacuated sheath around the inner tube (U.S. Patent 1986) could have resulted in much better tube-in-tube coil performance.

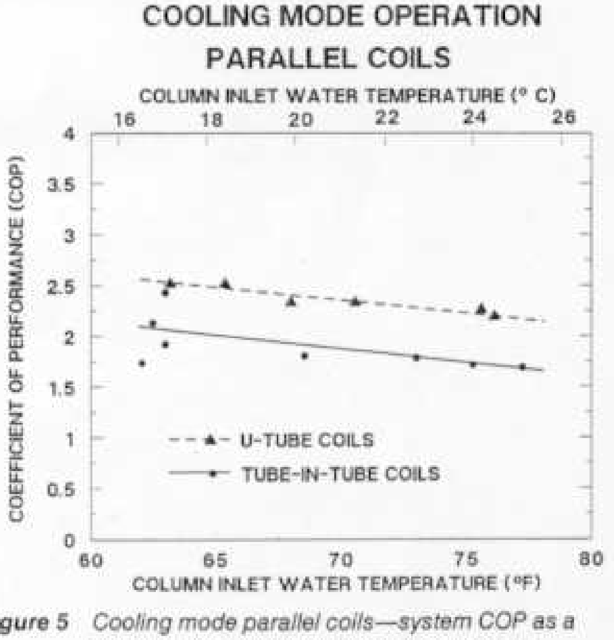

Figure 5 *Cooling mode parallel coils—system COP as a function of column inlet water temperature*

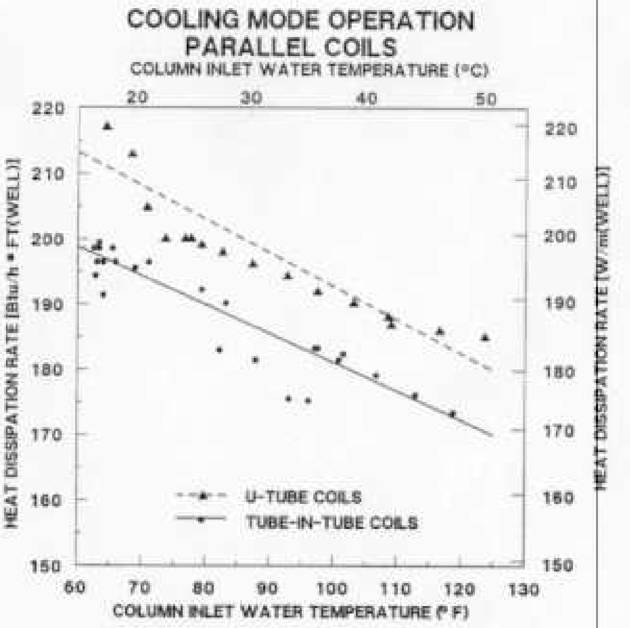

Figure 6 *Cooling mode two sets of two coils in series—heat dissipation rate as a function of column inlet water temperature*

Cooling Mode Test Results: Two Sets of Two Coils in Series

Figure 6 shows the heat dissipation rates for two coils connected in series. They are much higher than for parallel coil operation (see Figure 3). Every 20°F (11°C) column inlet water temperature increase results in about a 10 Btu/h•ft•well (9.6 W/m•well) heat dissipation capacity loss. For the tube-in-tube coils, the heat dissipation rates ranged from 165 to 205 Btu/h•ft•well (159 to 197 W/m•well) for column inlet water temperatures of 119.5°F to 63.5°F (48.6°C to 17.5°C). The system discharge pressures were much lower than for parallel coil operation at the same column inlet water temperatures. However, they had an equivalent range of pressure drop across the coils for the parallel operation. For the U-tube coils, the heat dissipation rates ranged from 180 to 215 Btu/h•ft•well (173 to 207 W/m•well) for column water temperatures from 124°F to 63°F (51°C to 17°C). The observed effect of heat transfer short-circuiting was not as serious as for parallel coil operation for the tube-in-tube coils.

Figure 7 shows the COP as a function of column inlet water temperatures. Again, U-tube coils have higher COPs, possibly due to their larger heat transfer area. For U-tube coils, a 10°F (5.5°C) increase in column water temperature results in a COP reduction of 0.25. For a good ground coil design, the ground temperature should not be more than 120°F, which means that a well-designed DXGC system should maintain steady-state cooling COPs of 2.0 or better.

Discussion of the Cooling Mode Test Results

It is clear that in this design the series coil connection can stand a much higher column inlet water temperature, 120°F (49°C) or higher, than the parallel coil arrangement. One reason could be that for series coil connection, the refrigerant velocity in the coils increased sharply, which resulted in a much higher refrigerant-side convective heat transfer coefficient, thus dissipating more heat.

COOLING MODE OPERATION
2-SETS OF TWO-COILS IN SERIES

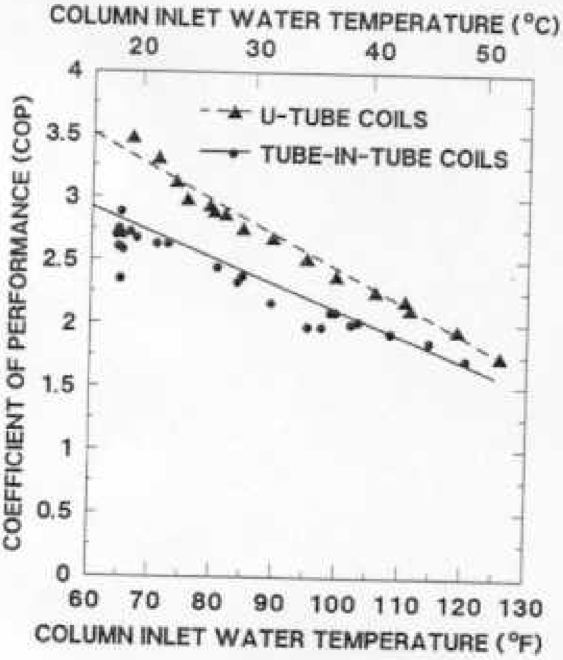

Figure 7 *Cooling mode two sets of two coils in series — system COP as a function of column inlet water temperature.*

Throughout the cooling mode experiment, there was no indication of oil return problems. However, during system start-up, it would take more than 10 minutes for the liquid line sight glass to show all liquid. System cycling losses could, therefore, be quite high.

It is interesting to note that in real-world secondary fluid circulation ground coil designs, ground temperatures adjacent to the coil are seldom more than 110°F (43.3°C). Otherwise compressor discharge pressures would be too high. Elimination of the refrigerant-fluid heat exchanger for DXGC design can result in a much higher ground temperature around the coil than for conventional GCHP systems. This is good in one sense because it increases the heat dissipation rate in the ground. Compared with the typical plastic ground coil design rule of thumb, 45 to 60 Btu/h•ft (43 to 58 W/m) (Bose 1980), DXGC coil heat dissipation rates observed in these tests are at least two to three times higher. However, one should be aware that potential ground dryout problems for soil around the coils, due to heat dissipation to the ground, will be aggravated by DXGC systems for the same reason. It is recommended that the great majority of ground coils for DXGC systems be under the groundwater table if efficient cooling mode operation is to be expected.

Heating Mode Test Results: Parallel Refrigerant Flow

In order to simulate low ground temperatures for heating mode tests, the water flow rate was reduced gradually. This enabled us to reduce the average column water temperature. The column water exit temperature is representative of the coldest spot of ground. Therefore, it is used as the independent variable for plotting the heating mode test data.

Figure 8 shows coil heat absorption rates per unit length of well for parallel refrigerant flow operation for both tube-in-tube and U-tube coils as a linear function of the column exit water temperature. Coil performance deteriorates rapidly as the water temperature drops (simulating cooling of the ground). However, even at 40°F (4.4°C), the heat absorption rate is still respectably high at around 100 Btu/h•ft•well (96 W/m•well).

It was found that frost started building on part of the suction line when the average column exit water temperature was less than 50°F (10°C). Therefore, the test was terminated when the average column exit water temperature was around 45°F in order to protect the compressor. Frost buildup at 45°F (7.2°C) indicated that the test system was not designed for low-temperature operation (the coils, i.e., were too short). This limited the test range for the system somewhat. However, it should not affect the generality of our test results for deriving the design guidelines.

For the tube-in-tube coil, parallel refrigerant flow operation, the heat absorption rate ranged from 100 to 145 Btu/h•ft•well (96 to 139 W/m•well) for column exit water temperatures of 44.5°F to 61°F (6.9°C to 16.1°C). The operating discharge pressure is much lower than for cooling-mode operation.

The heat absorption rate for U-tube coils ranged from 120 to 160 Btu/h•ft•well (115 to 154 W/m•well) for column exit water temperatures of 45° to 61°F (7.2° to 16.1°C), about 10% to 20% higher than that of tube-in-tube coils.

The average pressure drops across both type of coils are very small, only around 2 to 5 psi (14 to 35 KPa), which was much smaller than expected for cooling mode operation.

Compared with cooling mode operation, the effect of heat transfer short-circuiting for the tube-in-tube coils was much lower. The reason is that with evaporating refrigerant in the coils, there was almost no temperature difference between the inner tube and the annulus region. In fact, the refrigerant temperature in the inner tube could be slightly higher than the refrigerant in the annulus region (due to saturation pressure drop), thus slightly improving the coil performance. However, should the refrigerant vapor in the

HEATING MODE OPERATION
PARALLEL COILS

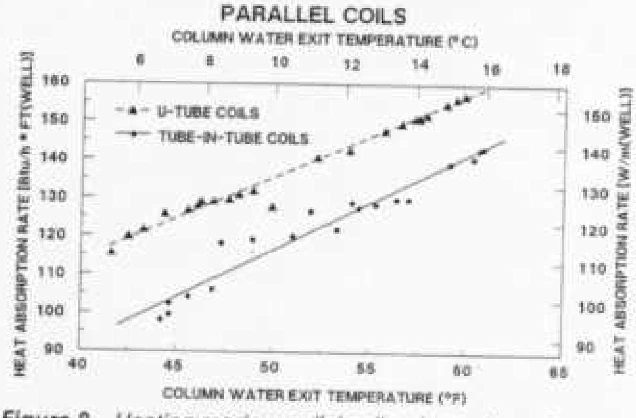

Figure 8 *Heating mode parallel coils—heat absorption rate as a function of column exit water temperature*

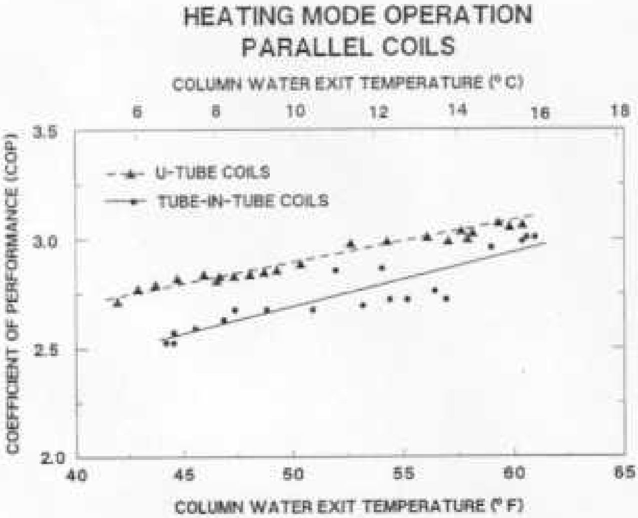

Figure 9 *Heating mode parallel coils—system COP as a function of column exit water temperature*

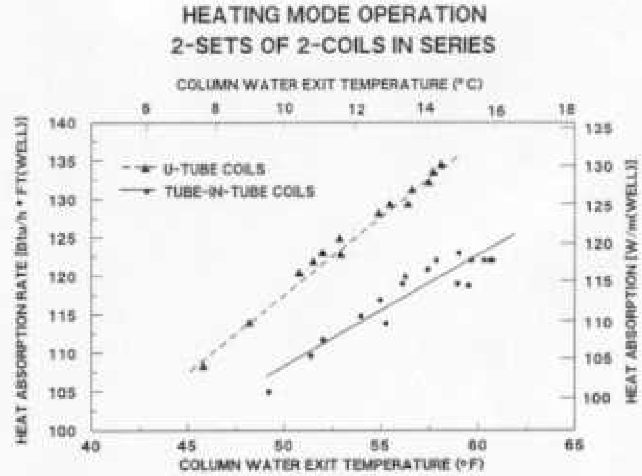

Figure 10 *Heating mode two sets of two coils in series—heat absorption rate as a function of column exit water temperature*

annulus region become highly superheated, it is still possible that heat transfer across the inner tube wall could somewhat adversely affect the performance of the tube-in-tube coil.

Figure 9 shows system COPs for both types of coils. The system COPs are actually fairly constant over the range of column water exit temperatures tested. This is due to the fact that when the temperature drops, the system suction pressure also drops, thus unloading the compressor and reducing compressor power consumption.

Heating Mode Test Results: Two Sets of Two Coils in Series

Figure 10 shows the heat absorption capacity per unit length of the column as a linear function of column water exit temperature for two sets of two coils connected in series. The test data suggest that the heat absorption rate is strongly affected by temperature. A 15°F (8.3°C) water temperature drop, in the U-tube case, will reduce the coil heat absorption capacity by 20 Btu/(h•ft•well) (19 W/m•well).

For tube-in-tube coils, the heat absorption rates varied from a high of 133 Btu/h•ft•well (118 W/m•well) to a low of 100 Btu/h•ft•well (101 W/m•well) for a water temperature range of 63°F to 49°F (17.2°C to 9.4°C). For the U-tube coils, heat absorption capacities ranged from 135 to 109 Btu/h•ft•well (130 to 105 W/m•well) for a water temperature range of 59°F to 46°F (15.0°C to 7.8°C). Again, the U-tube coil performed better by about 15 Btu/h•ft•well (14 W/m•well). The average pressure drop across the coils for this case was around 2 to 5 psi (13.8 to 34.5 KPa).

The tube-in-tube coils actually had slightly higher liquid subcooling levels than did the U-tube coils. This indicates that heat transfer short-circuiting for the tube-in-tube coil was not a factor in its performance in this case. Rather, heat transfer area became the dominating factor for coil performance.

Figure 11 shows system COPs as a function of the column exit water temperatures. Even though system operating conditions varied over a wide range, the system COPs remained fairly constant throughout the tests.

Discussion of Heating Mode Test Results

The test results indicate that there is no advantage for coils to be connected in series, because the system performed much better with parallel coil operation (compare Figures 8 and 10). The much higher pressure drop across the coils in the series arrangement adversely affected performance. Both arrangements yielded high levels of refrigerant liquid subcooling. However, the parallel coil operation had a much lower vapor superheat level than the series coil arrangement.

It was found that when the column water exit temperature fell below 50°F (10°C), frost started building around the suction line, indicating that the temperature differential between the refrigerant and the water (an indication of refrigerant-ground temperature differential) was very high. This points to an important advantage of DXGC over the conventional ground coil heat exchangers, namely, a high refrigerant-ground temperature differential will result in higher rates of moisture migration toward the coil in regions of unsaturated soil.

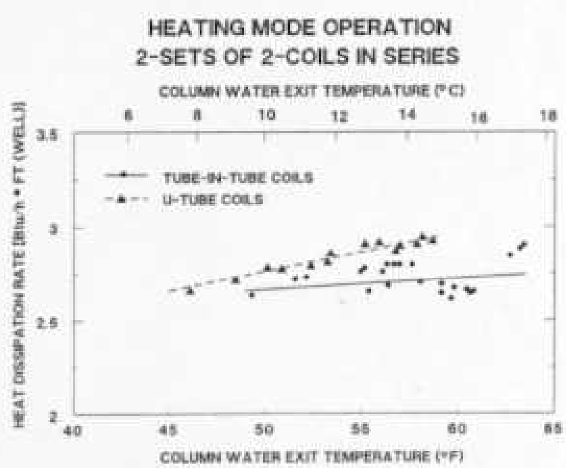

Figure 11 *Heating mode two sets of two coils in series—system COP as a function of column exit water temperature*

There was no starting problem for the heating mode operation. In addition, the refrigerant liquid line sight glass showed fully liquid flow a very short time after starting, indicating that system cycling losses should be lower for the heating mode than for the cooling mode.

All test results showed that the coil capacity is a linear function of column water exit temperature (or ground temperature), i.e., the coil-ground heat exchange capacity is a linear function of the ground temperature. The system COPs remained relatively constant due to reduced compressor power at lower water (ground) temperatures. Judging from the level of liquid subcooling, heat transfer short-circuiting between refrigerant in the inner tube and the annulus region for tube-in-tube coils is negligible for heating mode operation.

GENERAL DXGC DESIGN GUIDELINES FROM THE TEST RESULTS

Some general DXGC design guidelines were derived from the experimental results and observations.

1. DXGC coils can be at least twice as effective as conventional secondary fluid circulating coils. Bose (1980) had shown a 3 Btu/h•ft•°F (5.2 W/m•°C) heat dissipation rate for the cooling season for a conventional secondary fluid circulation GCHP. Assuming a 15°F (8.3°C) temperature differential between the secondary fluid and the ground, the heat dissipation rate will be around 45 Btu/h•ft (43 W/m), which is much lower than the DXGC performance measured in this project.

2. With the DXGC's much higher temperature differentials between the refrigerant and the ground, the potential to dry out the soil around the coil during summer operation is much higher than for a conventional GCHP. It is advisable to install DXGC coils vertically, with the majority of the coil below the groundwater table.

3. There will be no heat pump start-up or oil return problems for either cooling or heating mode operation if an array of shallow wells, less than 20 ft (6.0 m) deep, are used. Deep wells are feasible if a mechanism is provided to drain the refrigerant back to the ground coil to a level close to the ground surface.

4. Two shallow well coils connected in series will outperform two parallel coils for cooling mode operation. Two shallow well coils operated in parallel will outperform the same two coils connected in series for heating mode operation.

5. Simple tube-in-tube coil designs will have a severe annulus-to-inner-tube heat transfer short-circuiting problem for cooling mode operation. This problem will not appear, or becomes very minor, for heating mode operation. For areas where cooling loads are heavy, either U-tube coils or tube-in-tube coils with an evacuated third tube sheathing the inner tube to cut down the heat transfer short-circuiting will be needed.

6. Coil size selection based on refrigerant velocity specified by the *ASHRAE Handbook* to maintain oil return (ASHRAE 1985) seems adequate.

7. The data indicate that U-tube coils have higher heat absorption and rejection rates than the tube-in-tube coils. However, in this experiment, the U-tube coil had about 25% more heat transfer area. Thus, the tube-in-tube coils have a higher heat transfer rate per unit coil surface area.

8. In this experiment, heating and cooling mode operations were performed on the same coils without any major problems. Table 1 provides the range of ratios of the heating capacity to cooling capacity for the tests run under the project. It is expected that if the house heating and cooling load ratios are within the ranges shown in Table 1, then a single DXGC coil can be used for both heating and cooling mode applications. Otherwise, it is possible that either a two-coil system (two coils in the same well) must be employed, or some coils (if a multi-coil system) have to be shut off during operation of one mode. Here, it is assumed that the coils are designed for the heavier load (heating or cooling).

TABLE 1
Ratio of Heating Capacity to Cooling Capacity over Entire Test Range

Tube-in-Tube Parallel Refrigerant Flow	0.86 to 1.53
Tube-in-Tube 2 Sets of 2 Coils in Series	0.84 to 1.25
U-Tube Parallel Refrigerant Flow	0.96 to 1.37
U-Tube 2 Sets of 2 Coils in Series	0.79 to 1.30

9. When the ground temperature starts changing due to the operation of the ground coils, it is expected that the coil heating or cooling capacities will change linearly with the ground temperature. However, the system COPs will remain fairly constant over a wide range of ground temperatures. Power consumption (if any) for resistance heating is not considered in system COP calculation.

10. The refrigerant flow rate of DXGC can vary widely over the season. This, together with the additional tubing added to the system, makes it necessary to have a large accumulator to protect the compressor.

11. Expansion device design experience for a DXGC is rare. A slightly oversized TXV could cause valve hunting, as experienced by the author during the start of heating mode tests. This resulted in wide discharge pressure swings and caused compressor shutoff by a high-side pressure control. It may be possible to avoid the problem by using "orifice" plate or capillary-tube-type refrigerant expansion devices for DXGCs.

CONCLUSION

A DXGC heat pump system was assembled in the laboratory with a set of four 15-ft-long (4.6 m) insulated PVC columns used as the ground simulator. The DXGC heat pump system was well instrumented and tested under carefully controlled operating conditions.

Two sets of coils were installed in each column—a tube-in-tube coil and a U-tube coil. Both heating and cooling mode tests were performed for the coils connected in both parallel and series arrangements. The system worked as expected and the ground simulator was successful. It was found that for cooling mode operation, the series arrangement outperformed the coils in parallel arrangement. However, for heating mode operation, the situation was reversed. There appeared to be no oil return or system start-up problems for the laboratory system. It also seems that U-tube design is better than tube-in-tube design per unit length of the well because U-tube coils have about 33% more heat transfer area. However, tube-in-tube coils have a higher per-unit coil area heat transfer rate. One has to be aware that these results are more qualitative than quantitative.

A set of general design guidelines was derived from the experimental observations and test results. It is hoped that this information will be useful for those in the DXGC heat pump field.

The test results indicated that DXGCs have a heat transfer rate at least twice as high as conventional plastic coils, meaning that only half as much coil would be needed compared to conventional secondary-fluid-circulating GCHPs. In other words, a DXGC system can be installed in a much smaller lot than the conventional GCHP. With the additional saving of a refrigerant-fluid heat exchanger and a fluid circulating pump, the DXGC concept is not only technically superior, it may be economically viable compared to conventional GCHP designs.

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