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EXPERIMENTAL ANALYSIS OF DIRECT-EXPANSION GROUND-COUPLED HEAT PUMP SYSTEMS

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1. ABSTRACT

Direct-expansion ground-coil-coupled (DXGC) heat pump systems have certain energy efficiency advantages over conventional ground-coupled heat pump (GCHP) systems. Principal among these advantages are that the secondary heat transfer fluid heat exchanger and circulating pump are eliminated. While the DXGC concept can produce higher efficiencies, it also produces more system design and environmental problems (e.g., compressor starting, oil return, possible ground pollution, and more refrigerant charging). Furthermore, general design guidelines for DXGC systems are not well documented.

A two-pronged approach was adopted for this study: 1) a literature survey, and 2) a laboratory study of a DXGC heat pump system with R-22 as the refrigerant, for both heating and cooling mode tests done in parallel and series tube connections. The results of each task are described in this paper. A set of general design guidelines was derived from the test results and is also presented.

2. KEYWORDS

Heat pump, ground coil, ground coupled heat pump, ground heat exchanger, and ground source heat pump.

3. INTRODUCTION

A primary advantage the DXGC system has over secondary fluid systems is its inherently higher efficiency due to elimination of both the secondary fluid heat exchanger and a circulating pump. However, the DXGC system has some significant application problems.

Early GCHP work (before 1970) dealt almost exclusively with heating. Most studies involved field tests using horizontal type DXGC systems. Cooling mode operation was difficult because of heat pump start-up problems. As refrigerant became trapped in the ground coils, the compressor was starved of refrigerant and eventually shut off by the low-pressure cut-off switch. Corrosion of buried metal coils was also a major concern, since there is no easy way to repair a leaky coil once it is installed. Later GCHP designs, using a secondary circulating fluid and an intermediate heat exchanger, were able to operate in 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 secondary fluid systems. The plastic materials can easily be butt-welded to form leak-proof joints and can last indefinitely under the ground.

Recent development of new technologies to handle the problems associated with DXGC systems has generated renewed interest in this design. A study by Ratcliff (1987) indicated that unless the local soil or water contains highly oxidative chemical compounds, practically no corrosion problem exists for copper coils. System starting problems in cooling mode operation can be solved by two methods. One uses shallow vertical wells instead of deep ones. With shallow wells, the compressor can lift refrigerant out of the ground coils without much difficulty. Another method uses vertical tube-in-tube coils coupled with an oversized accumulator and a large refrigerant charge. During the compressor "off" period, refrigerant drains back to the ground coils from the accumulator. The ground coils are thus flooded with refrigerant which reduces the vertical lift required of the compressor.

While many field tests have been performed, a literature search on DXGC has never been performed. Neither are there any general guidelines available for DXGC designs. The principal goal of this project was therefore to generate such guidelines.

A two-pronged approach toward the analysis of DXGC designs was taken in this study:

(1) a <u>literature search</u> to evaluate the results from previous work on DXGC systems; and

(2) a <u>laboratory Test</u> to report the findings of a laboratory-scale DXGC heat pump system design with four shallow vertical wells. Two types of coils, tube-in-tube and U-tube, were tested in both cooling and heating mode operation. The coils were tested both in parallel operation and with two sets of two coils in series.

The laboratory tests indicated that DXGC systems are feasible with high system efficiency. The technical concerns associated with DXGC designs were closely monitored throughout the experiment. Results of the laboratory tests were used to produce general DXGC design guidelines.

4. LITERATURE SEARCH

Earlier researchers knew that ground coil performance was closely related to soil conditions. Vestal (1949) summarized the problems involving soil thermal properties very well. Almost all DXGC work dealt with winter heating only. Line source theory was exclusively used for coil design (Ingersoll and Plass, 1949). Some interesting coil design observations can be derived from this work:

- Thermal penetration caused by ground coil operation is usually not more than three feet (Freund and Whitlow, 1959).
- Latent heat released by soil freezing is small compared with the total heat absorbed from the ground; however, the thermal conductivity increase due to soil freezing is very important (Coogan, 1949).
- Smaller tubes are preferable to large ones, with the optimum diameter somewhere between 1/2 and 3/4 inches (1.27 and 1.91 cm) (Coogan, 1949).

- Compressors would not operate properly in summer unless a large amount of additional refrigerant were added to the system (Freund and Whitlow, 1959).

GCHP work in European countries is basically for heating only, and Europeans are willing to accept a longer payback period than is typically sought in the U.S.. Fordsmand (1985) indicated that, under certain assumptions, the payback period is 11 years for heating-only operation in Denmark, which including the cost of solar collectors for additional heat input to the ground. For those areas with warmer climatic conditions requiring both heating and cooling, the payback period is reduced to six years. DXGC systems must be designed for both heating and cooling in the U.S. Summer compressor starting is still a major concern. Three novel design concepts that deal with this issue for DXGC heat pump systems have been identified.

The first uses two sets of coils in the same bore hole (Jungworth, 1987) (one set, with larger tubes, is used for winter operation, and the smaller set is used for summer operation). This design is intended to accommodate refrigerant velocity changes for different operating modes. The possible drawbacks are that the initial cost is higher, more refrigerant charge is needed, and heat pump capacity is lost because of leakage between the coil sets.

The second concept is a three-tube tube-in-tube design with a patented refrigerant reservoir-accumulator (U.S. Patent, 1986). During the "off" cycle period, refrigerant will be drained back to the ground coils by gravity. The refrigerant will fill the ground coils close to grade level so that the compressor can avoid a severe starting problem. This design concept is more technically advanced.

The third design involves a five-foot-wide by four-foot-deep trench with a polyethylene vapor seal. Refrigerant lines are submerged in a water bath formed by a semi cylindrical section of metal culvert. Gravel is placed around the culvert up to a depth of three feet (Edwards and Safemazandarani, 1985). Possible drawbacks are a higher initial cost and the possible loss of water from the bath.

It is always possible that a leak will develop in the ground coil, which could leak refrigerant and oil into the ground. The major heat pump refrigerant, R-22, has been proven safe to the environment, except for a small ozone depletion potential. As for compressor oils, most of which are mineral oils, discussion with an Oak Ridge National Laboratory (ORNL) chemist revealed that some compressor oils are treated to safeguard from any harmful effects to the human body. Untreated compressor oils are labeled as such. In general, this chemist felt that the small amount of oil involved would not cause any pollution problems (Sand, 1988). As for the effect on the ground temperature due to long-term heat pump operation, most previous work indicated a complete ground temperature recovery if the systems were operated year-round, which would be the typical operation pattern in the United States. This concern is much more serious in areas such as many European countries where the winter heating load is very heavy or where GCHPs are operated for winter heating only.

Although abundant field experimental data are available, it is difficult to derive general design guidelines, such as how long a coil should be under certain design conditions, etc., from the field data. Laboratory DXGC tests under controlled operating conditions are better suited for answering some of these questions. Mathematical models for DXGC coil design are available, but most of them are based on line source theory. Other models are based on a more detailed mathematical derivation, but will consume more computer time (Mei, 1986).

5. LABORATORY TEST

The laboratory system at ORNL was designed for a cooling capacity of 1/2 to 3/4 ton (1.76 to 2.64 kW). Coil size selection was based on the calculation of refrigerant velocity recommended by the ASHRAE Fundamentals Handbook (1989) in order to ensure oil return. The coils were reconnected in such a way that they could be operated either in parallel or as two sets of two coils in series. Tests were performed in both cooling mode and heating mode. Figures 1 and 2 show the schematic and the actual test setup.

5.1 Test Procedures

5.1.1 Cooling Mode Tests

The inlet water temperature and flow rate to the indoor water-refrigerant heat exchanger of the heat pump were maintained at 62 - 65°F (16.7 - 18.3°C) and 3 gpm (0.19 L/h) respectively. The total water supply to the four PVC columns was maintained at 3 gpm (0.19 L/h). However, the column inlet water temperature was varied from 65 to 120 °F. Data were collected at each temperature level 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. The column inlet water temperatures are used as the independent quantity on which other measured data are based.

5.1.2 Heating Mode Tests

The inlet water temperature and flow rate to the water-refrigerant heat exchanger of the heat pump were maintained at 90 ± 1°F (32.2 ± 0.6°C) and 3 gpm (0.19 L/h) respectively. The inlet water temperature to the columns was maintained at 62 - 65°F (16.7 - 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). Data were collected at each column water flow rate for the time 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 are changed for each test, the average column exit water temperature is used as the independent quantity on which the other measured data are based.

Both cooling and heating test procedures were repeated for both tube-in-tube and U-tube coils and for both parallel arrangements and arrangements using two sets of two coils in series.

5.2 Test Results and Discussion

5.2.1 Cooling Mode Test Results: Parallel Refrigerant Flow

Figure 3 shows the ground-coil heat dissipation rate per unit length of the 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 to 77.4°F (17.0 to 25.2°C). The unit operated at a very high discharge pressure (325 psia or 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 in the inlet water temperature resulted in compressor cut-off by the high-pressure safety switch. The coil heat dissipation rate ranged from 204 to 192 Btu/(h-ft) (196-185 W/m) for U-tube parallel operation, higher than that of the tube-in-tube coils. The discharge pressure is about 30 psi (207 kPa), lower than that of 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 were 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 thermal "short circuit" between the refrigerant in the inner coil and that 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.

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 tube-in-tube coils yielded lower system COPs than the U-tube coils. Again, this lower COPs were due to the heat transfer short-circuit and the 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.

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

Figure 6 shows the heat dissipation rates for two coils connected in series. The rates are much higher than those for the parallel coil operation (see Figure 3). Every 20°F increase in inlet column water temperature will result in about 10 Btu/(h-ft-well) (9.6 W/m-well) of heat dissipation capacity loss.

Figure 7 shows the COPs as a function of column inlet water temperatures. Again, U-tube coils have higher COPs probably because of their larger heat transfer area. For U-tube coils, a 10°F (5.5°C) increase in ground temperature results in a COP reduction of 0.25. For a good ground coil design, the ground temperature should not be above 120°F (49°C), which means that a well-designed DXGC system should maintain steady state cooling COPs of 2.0 or better.

5.2.3 Discussion of Cooling Mode Test Results

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

Throughout the cooling mode experiment, there was no indication of oil return problems. However, during the start-up, the liquid line did not show all the liquid for more than 10 minutes. 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 the compressor discharge pressures would be too high. Elimination of the refrigerant-fluid heat exchanger for DXGC design can result in a ground temperature around the coil that is much higher than that of conventional GCHP systems. This is good in one sense because it increases the heat dissipation rate to the ground. However, one should be aware of the potential for ground dry-out problems for soil around the coils, because of high heat dissipation rate, that could be aggravated by DXGC systems. 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.

5.2.4 Heating Mode Test Results: Parallel Refrigerant Flow

In order to simulate low ground temperature 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 coil for parallel refrigerant flow operation as a linear function of the column water exit 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 up 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 (7.2°C) in order to protect the compressor. Frost buildup at 45°F (7.2°C) indicates that the test system was not designed for low-temperature operation (i.e., the coils were too short). This problem should not affect the generality of our test results for deriving design guidelines however.

The heat absorption rate for U-tube coils ranged from 120 to 160 Btu/(h-ft-well) (115-154 W/m-well) for a column average exit temperature of 45 to 61°F (6.9 to 16.1°C), which is higher than that of tube-in-tube coils by about 10 to 20%.

The average pressure drops across both types of coils are very small, around 2 to 5 psi (14 - 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 that of the refrigerant in the annulus region (because of the saturation pressure drop), thus slightly improving the coil performance. However, should the refrigerant vapor in the 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 for the range of column water exit temperatures tested. This is because when the temperature drops, the system suction pressure also drops, thus unloading the compressor and reducing compressor power consumption.

5.2.5 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 the 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 the temperature. A drop of 15°F (8.3°C) in water temperature will reduce the coil heat absorption capacity by 20 and 15 Btu/(h-ft-well) (19 and 14.3 W/m-well) for U-tube and tube-in-tube coils respectively.

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

5.2.6 Discussion of Heating Mode Test Results

The test results indicate that no advantage exists for coils connected in series because the system performed much better with parallel coils (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, parallel coil operation yields a much lower vapor superheat level than does 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 using DXGCs rather than conventional plastic-tube 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.

No starting problem existed for heating mode operation. In addition, the sight glass for the refrigerant liquid line showed full 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 the column water exit temperature (or ground temperature), i.e., coil-ground heat exchange capacity is a linear function of ground temperature. System COPs remained relatively constant because of reduced compressor power at lower water (or ground) temperatures. Judging from the level of liquid subcooling, heat transfer short circuiting between refrigerant in the inner tube and that in the annulus region for tube-in-tube coils is negligible for heating mode operation.

6. GENERAL DXGC DESIGN GUIDELINES BASED ON TEST RESULTS

The following general DXGC design guidelines have been derived from the experimental results and observations.

(1) The higher temperature differentials between refrigerant and ground with a DXGC have a higher potential to dry out the soil around the coil due to coil heat dissipation during summer operation than do conventional systems. It is advisable to install DXGC coils vertically, so that the majority of the coil should be below the groundwater table.

(2) Using an array of shallow wells less than 20 feet (6.0 m) deep will minimize both heat pump startup and oil return problems for either cooling or heating mode operations. Deep wells are feasible if a proper mechanism is provided to drain the refrigerant back to the ground coil to a level close to the ground surface.

(3) Two shallow-well coils connected in series will outperform two parallel coils for cooling mode operation. For heating mode operation, two shallow-well coils operated in parallel will outperform the same two coils connected in series.

(4) 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 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.

(5) Selecting coil size based on the refrigerant velocity specified by the ASHRAE Handbook (1989) to maintain oil return seems adequate.

(6) The data indicate that U-tube coils have higher heat absorption and rejection rates than the tube-in-tube coils. In this experiment, however, the U-tube coil has about 33% more heat transfer area than does the tube-in-tube coil. If the heat transfer is proportional to the coil-ground heat transfer area, tube-in-tube coils have a higher heat transfer rate per unit coil surface area.

(7) Heating and cooling mode operations were performed on the same coils without any major problems in this experiment. Table 1 provides the range of the ratios of the heating capacity to

the cooling capacity observed during our tests. 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 operations. Otherwise, it is possible that either a two-coil system (two coils in the same well) must be employed or several coils (if a multi-coil system) must be shut off during operation in one mode. It is assumed here that the coils are designed for the heavier load (heating or cooling).

Table 1. Ratio of heating capacity over 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

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

(9) Refrigerant flow rate for a DXGC system can vary widely over the season. This variation, together with the additional tubing of the system, make it necessary to have a large accumulator to protect the compressor.

(10). Sizing expansion devices for the indoor coil is usually easy because ample design experience is available. Design experience is rare, however, for direct-expansion ground coils. A slightly oversized thermal expansion valve could cause "valve hunting" as experienced by the authors during the start of heating mode tests. This "valve hunting" resulted in wide discharge pressure swings and caused the high-side pressure control to shut off the compressor. The problem can be avoided by using an "orifice" plate or capillary tube refrigerant expansion device for DXGCs.

7. CONCLUSIONS

A DXGC heat pump system was assembled in the laboratory with a set of four 15-ft (4.6-m) long insulated polyvinylchloride columns designed 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 with the coils connected in both parallel and series arrangements. The system worked as expected and the ground simulator was successful. For cooling mode operation, the series arrangement outperformed the coils in parallel arrangement. For heating mode operation, however, the situation was reversed. There appears to be no oil return or system startup problem for shallow well systems (wells not deeper than 15 to 20 feet or 4.6 to 6 m). Also, the U-tube seems better than the tube-in-tube design per unit length of well because U-tube coils have about 33% more heat transfer area in this experiment. However, tube-in-tube coils have higher heat transfer rates per unit coil area.

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

The test results indicated that direct-expansion coils have a heat transfer rate at least twice as high as that of conventional plastic coils, meaning that only half as much coil is needed compared with the conventional secondary fluid circulating GCHP. With the additional savings gained by eliminating a refrigerant-fluid heat exchanger and a fluid circulating pump, the DXGC concept is not only more efficient, it may be economically viable as well.

8. ACKNOWLEDGMENT

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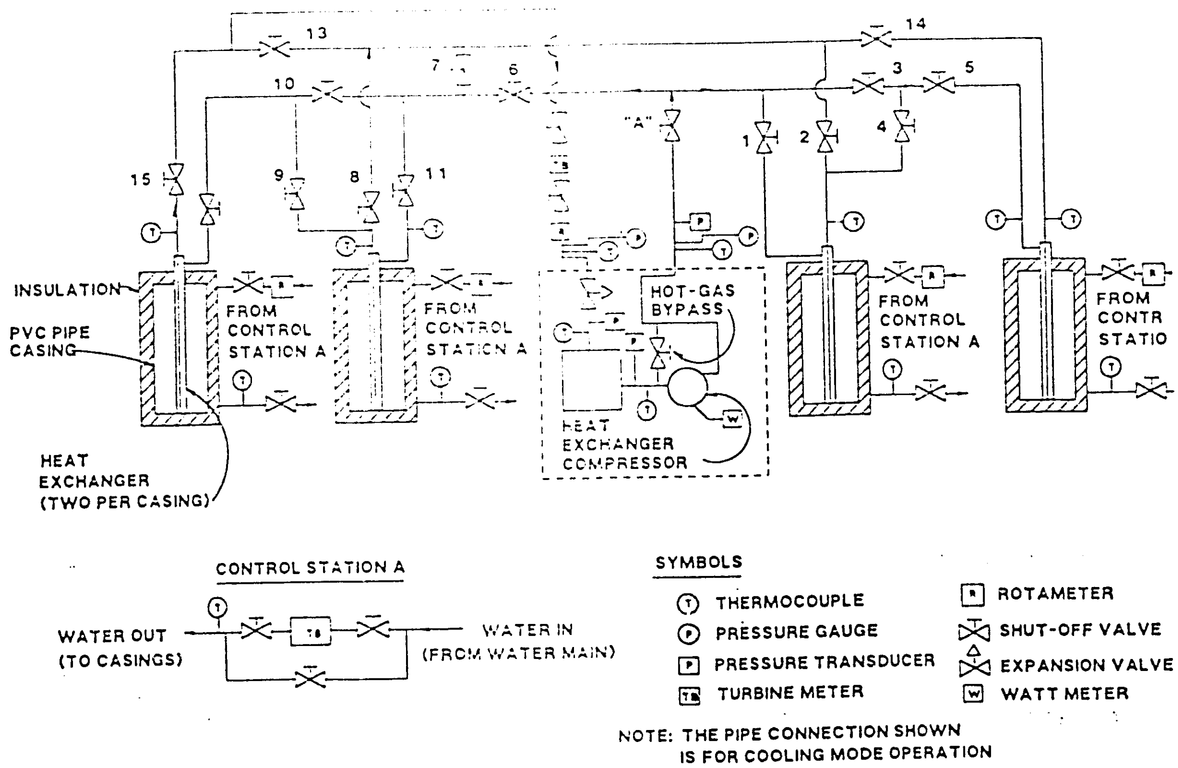

Figure 1 *Schematic of laboratory test setup*

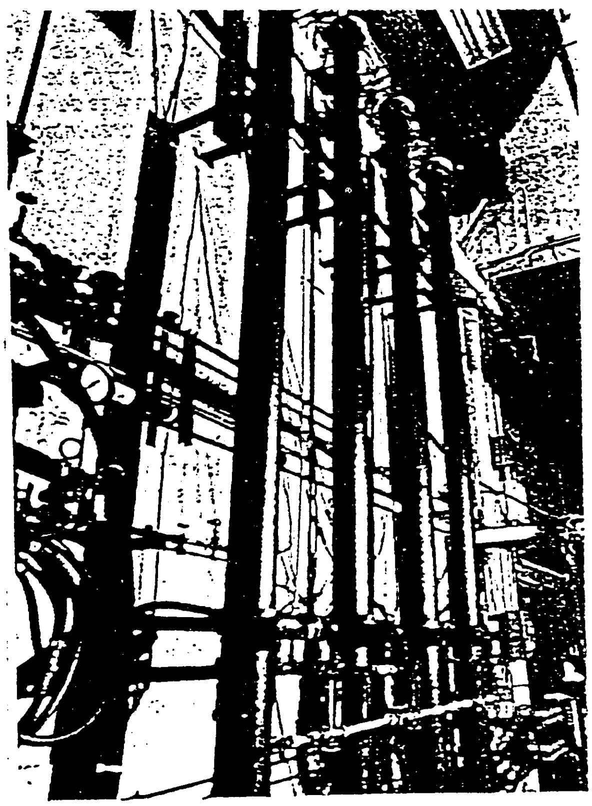

Figure 2 *Photograph of test setup*

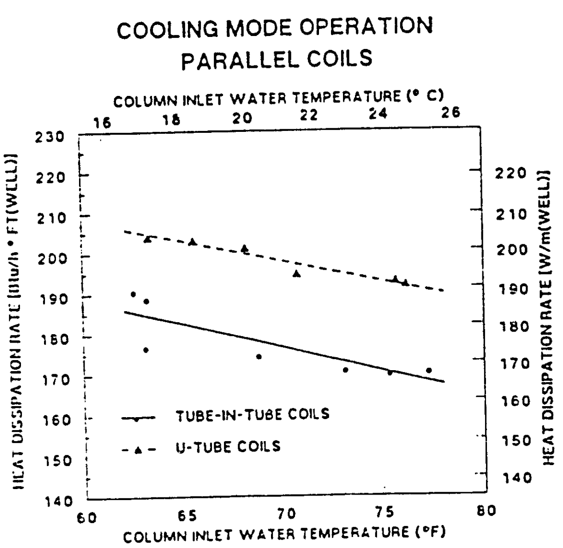

Figure 3 *Cooling mode parallel coils—heat dissipation rate as a function of column inlet water temperature*

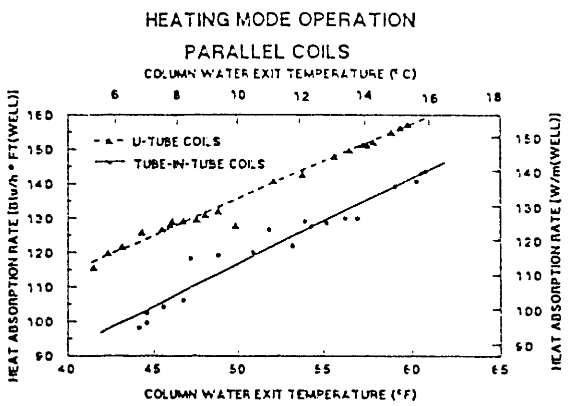

Figure 8 *Heating mode parallel coils—heat absorption rate as a function of column exit water temperature*

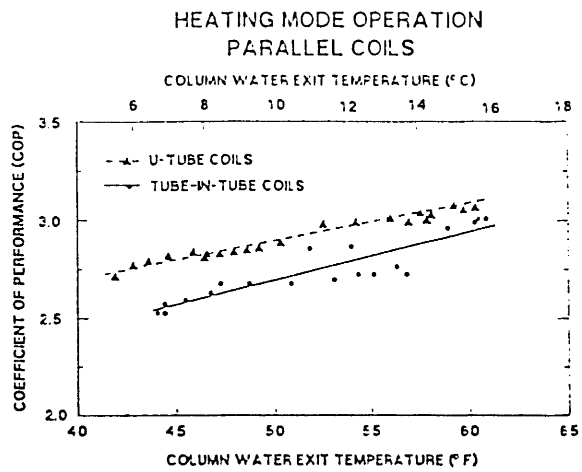

Figure 9 *Heating mode parallel coils—system COP as a function of column exit water temperature*

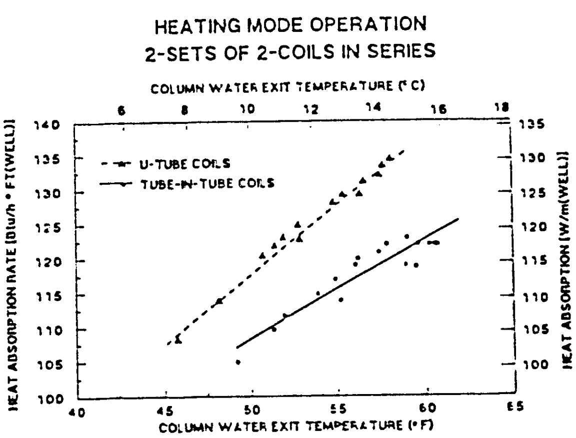

Figure 10 *Heating mode two sets of two coils in series—heat absorption rate as a function of column exit water temperature*

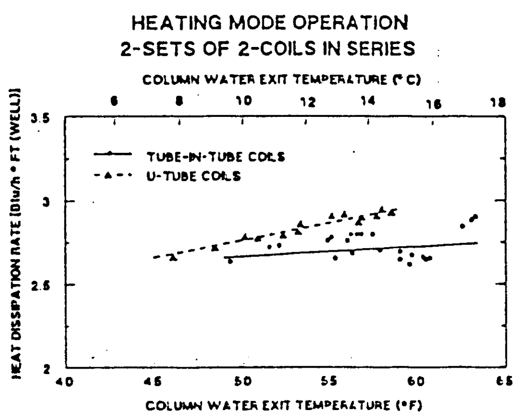

Figure 11 *Heating mode two sets of two coils in series—system COP as a function of column exit water temperature*

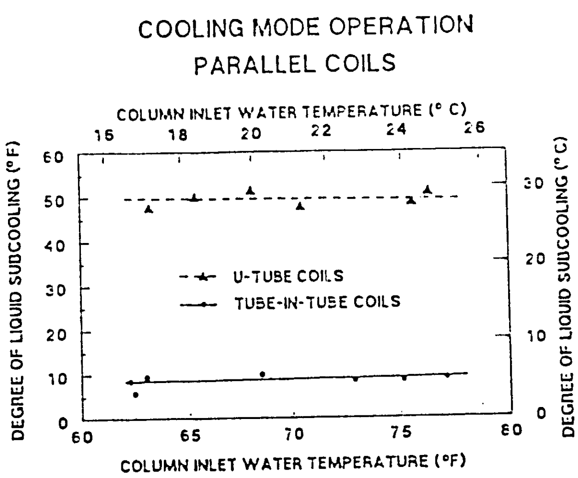

Figure 4 *Cooling mode parallel coils—degree of liquid subcooling as a function of inlet water temperature*

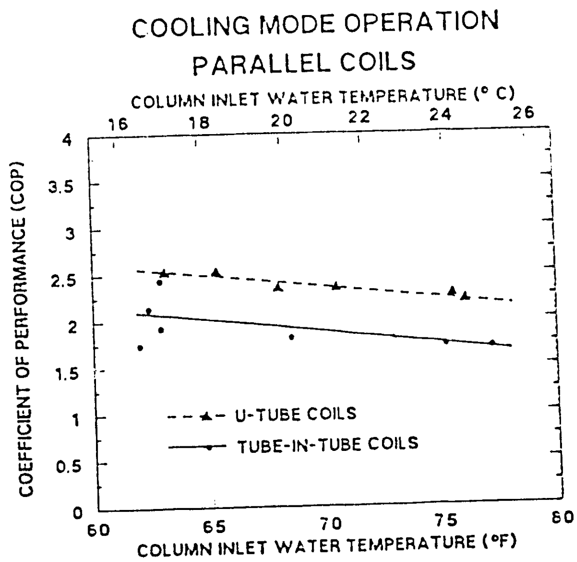

Figure 5 *Cooling mode parallel coils—system COP as a function of column inlet water temperature*

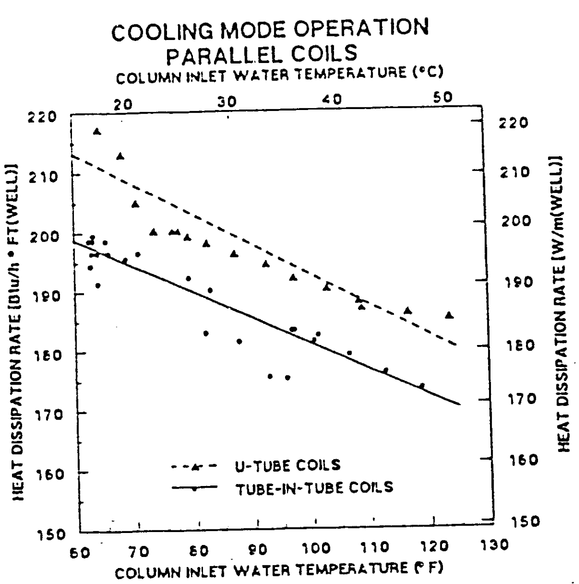

Figure 6 *Cooling mode two sets of two coils in series—heat dissipation rate as a function of column inlet water temperature*

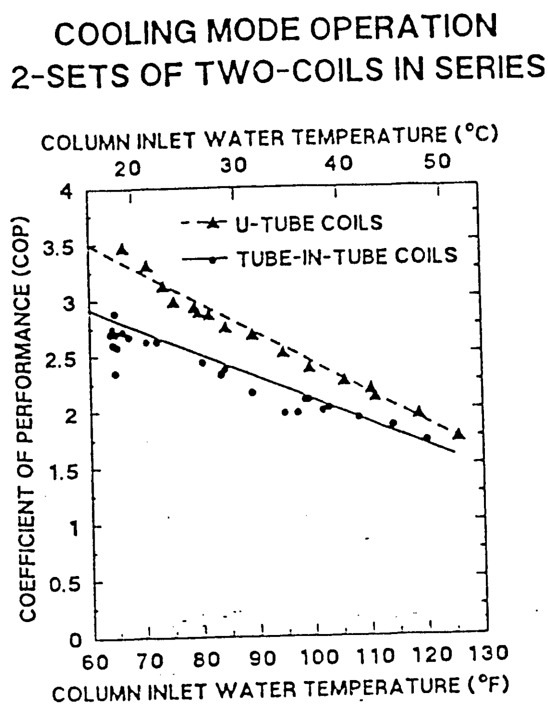

Figure 7 *Cooling mode two sets of two coils in series — system COP as a function of column inlet water temperature.*

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