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VERTICAL CONCENTRIC-TUBE GROUND-COUPLED HEAT EXCHANGERS*

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Abstract

An experimental and analytical project to study the design of vertical, concentric-tube ground-coupled heat exchangers for use in heat-pump applications is described. The experimental apparatus consists of a concentric configuration of two 47.7-m (155-ft) PVC pipes (sealed at both ends with connections so that hot or cold water could be pumped through the system) with instrumentation to measure heat transfer. This heat exchanger was placed in a 0.20-m (8-in.) inside-diameter well and backfilled with sand to establish good thermal contact. Heat transfer was measured for heat-exchanger operation under several sets of operating conditions. A mathematical model was developed and converted into an ANSI standard FORTRAN IV computer program to simulate the operation of the ground-coupled heat exchanger. Data collected using the experimental apparatus was used to validate the computer program, and the computer model was then used to study the effects of variations in heat-exchanger length, diameter, flow rate, and thermal conductivity and the thermal conductivity of the ground on the heat-exchanger performance. Results are presented.

1. INTRODUCTION

The realization that conventional energy resources are limited urges us to increase our efforts to use non-renewable sources more efficiently. One of several alternatives available is to search for energy saving modifications in the existing HVAC processes.

Heat pumps have long been considered as effective devices to extract heat from or reject heat to the environment. The majority of heat pumps use ambient air as the heat source and sink, but their energy efficiency is limited by the thermal capacity and temperature of the air. Ground water is a better heat source/sink for heat pump application (due to its superior thermal capacity and stable temperature), but it is limited because there are areas in the country where sufficient quality and quantity of ground water are either not available or the pumping and disposal costs and associated problems make it economically unattractive. Another alternative is to use the ground itself as the heat source and sink for a heat pump.

Extensive research on horizontal ground coupled heat pump systems was performed in the 1930's and 1940's. The ground coupled heat pump idea was set aside due to the low cost of energy at that period in spite of its high potential of energy conservation. At the present time, horizontal ground coupled heat pumps are popular in Europe where they are used for heating only applications. These types of systems have not been used very much in the United States where the heat exchanger must operate for both space heating and cooling. The

injection of hot water or glycol solution into a horizontal ground coupled heat exchanger, as occurs during the cooling season, dries out the soil around the tube and lowers the thermal conductivity of the ground, which in turn decreases the effectiveness of the heat exchanger.

Vertical, concentric tube-in-tube ground-coupled heat exchangers are an alternative to the horizontal configuration. They were first proposed by Ambrose [1] for heat pump applications in 1946. Such a configuration has advantages over a horizontal ground coil:

- it occupies a much smaller amount of surface area, and
- the ground is saturated with moisture a short distance below the surface and provides a good heat transfer medium for most of the length of the heat exchanger.

The moisture migration caused by heating the unsaturated soil around a shallow horizontal heat exchanger will affect only a small section of a deep well vertical heat exchanger.

Bose et al [2-4] described a geothermal well used for heat pump application in Oklahoma. While the performance of vertical ground coupled heat exchangers was shown in their work, little data could be found that applied to the design of a vertical heat exchanger. Oliver and Brandy [5] derived a closed form analytical solution for the concentric vertical ground coupled heat exchanger under steady state operation by assuming an isothermal ground surface 1.0 m (3.3 ft) away from the center of the heat exchanger. Their theory tends to give an overly conservative design, however, and in practical application, the

ground reaction to heat pump cyclic operation is highly transient and it is unrealistic to design the heat exchanger based on its steady state operation.

In this study, a 47.7 m (155 ft) vertical tube-in-tube ground coupled heat exchanger was installed and tested. A mathematical model describing the heat exchanger operation, as well as the ground reaction to the energy transfer, was developed and transformed into a computer code to simulate the heat exchanger performance. Four laboratory tests were performed with the ground coupled heat exchanger and also simulated with the computer program to validate the model. A parametric study was performed with the model to examine the effect of some design factors on the performance of the heat exchanger. This computer code is easy to use and is intended as a design tool for future installation of vertical ground coupled heat exchangers.

2. TEST APPARATUS AND PROCEDURE

A ground couple heat exchanger was installed for testing by the authors. The test site was core sampled to the depth of 76 m (250 ft) and the ground was found to be limestone to this depth, except for the first 5.5 m (18 ft), which was clay. The water table (the level at which the soil is saturated with water) at the test site was at 4.3 m (14 ft) below the surface. Two ground core samples, one each from the top and the bottom of the drilling, were tested for their thermal properties [6] (which were found to be almost identical):

$k = 3.03$ W/m°K (1.751 Btu/h ft°F),

$\rho = 2.7$ gm/cm^3 (168.5 lb/ft^3), and

$c_p = 1.065$ J/gm°K (0.255 Btu/lb°F).

The material selected for the ground coil itself was schedule 40 PVC pipe. The outer casing had a 0.127 m (5.0 in.) inside diameter and the inner tube was a 0.025 m (1.0 in.) inside diameter. The thermal conductivity of the PVC pipes was determined experimentally U6e by steady state heat transfer methods: [6] by steady state heat transfer methods:

$k = 0.175$ W/m°K (0.101 Btu/h ft°F) for the inner pipe,

$k = 0.190$ W/m°K (0.109 Btu/h ft°F) for the outer casing, and

$\rho = 1.23$ gm/cm^3 (77.0 lb/ft^3).

The specific heat of PVC was found to be about 1.51 J/gm°K (0.36 Btu/lb°F) [7].

Water was used as the working fluid throughout the tests; its flow rate was kept at 3.15×10^{-4} m^3/sec (5.0 gpm). This limit was established to avoid the transition region of fluid flow so that computer simulations could also be performed. The ground coil inlet water supply was connected to the laboratory process water supply mains so that either cold or hot water could be used at a relatively constant temperature. The inlet water supply was also connected to a 3.8 m^3 (1,000 gal.) ice tank so that water close to the freezing point, 0°C (32°F), could be pumped to the heat exchanger for low

temperature inlet water operation. A rotameter was used to measure the flow rate of the water.

Figure 1 shows a schematic of the ground coil used for the tests. The coil is 47.2 m (155 ft) in length and the annular region between the pipes has inner and outer diameters of 0.033 m (1.31 in.) and 0.127 m (5.0 in.) respectively. Circular spacers were bonded to the inner tube and temperature sensors were attached to the spacers. Figure 1 also shows the depths at which temperature sensors were located in the annular area between the two pipes, and the location of the sensors on the outside of the outer tube. Temperature sensors were also located in pilot holes near the heat exchanger. All the sensors were 4-wire platinum RTD's except those for the inlet and outlet water temperature measurements which were thermocouples. The RTD's and thermocouples were calibrated and have accuracies of ±0.3°C (±0.5°F) and ±0.5°C (+1°F), respectively.

The data acquisition system used in the test consisted of a digital computer with an 8 K word memory coupled with a floppy disk drive, an integrating digital voltmeter with an ohm converter, and a reed relay scanner. The programs used by the computer were written in the FOCAL language which has had some modifications made which facilitate data acquisition.

3. THEORETICAL DEVELOPMENT

The fluid from the heat pump is pumped down the center tube, as shown in Fig. 1, and flows upward in the annulus; it transfers energy to or from the ground as it rises. The energy transfer between the fluid and tube wall is by convection when the fluid is flowing and by conduction when the flow is stopped. The flow of heat within the walls of the tubes and ground is assumed to be radial conduction with no vertical heat transfer.

The following assumptions have been made to simplify the analysis:

1. the fluid is well mixed: its temperature is uniform in the center tube and in the annular area perpendicular to the axis of the flowing column,

2. the thermal and physical properties of the fluid, tube, and ground are constant, or independent of temperature and pressure variations,

3. heat transfer by radiation is neglected, and

4. the entire system has cylindrical symmetry about the axis of the fluid column.

The following mathematical model was constructed based on energy balances and the assumptions listed above.

Heat flow in the water within the inner tube and at the tube wall:

$$-v_1 \frac{\partial T_{f_1}}{\partial z} + \frac{2k_1}{\rho_f c_{p_f} r_1} \frac{\partial T_1}{\partial r} \Big|_{r=r_1} = \frac{\partial T_{f_1}}{\partial t} \qquad (1)$$

Heat flow within the inner pipe:

$$\frac{\partial^2 T_1}{\partial r^2} + \frac{1}{r}\frac{\partial T_1}{\partial r} = \frac{1}{\alpha_1}\frac{\partial T_1}{\partial t} \quad (r_1 < r < r_2, 0 < z) \tag{2}$$

Heat flow in the water in the annulus and at the tube walls:

$$v_2 \frac{\partial T_{f_2}}{\partial z} - \frac{2r_2 k_1}{(r_3^2 - r_2^2)\rho_f c_{p_f}} \frac{\partial T_1}{\partial r}\bigg|_{r=r_2} + \frac{2r_3 k_2}{(r_3^2 - r_2^2)\rho_f c_{p_f}} \frac{\partial T_2}{\partial r}\bigg|_{r=r_3} = \frac{\partial T_{f_2}}{\partial t} \tag{3}$$

Heat flow within the outer tube:

$$\frac{\partial^2 T_2}{\partial r^2} + \frac{1}{r}\frac{\partial T_2}{\partial r} = \frac{1}{\alpha_2}\frac{\partial T_2}{\partial t} \quad (r_3 < r < r_4, 0 < z) \tag{4}$$

Heat flow in the ground:

$$\frac{\partial^2 T_3}{\partial r^2} + \frac{1}{r}\frac{\partial T_3}{\partial r} = \frac{1}{\alpha_3}\frac{\partial T_3}{\partial r} \quad (r_4 < r \leq r_5, 0 \leq z) \tag{5}$$

Conditions at initial time (t=0):

$$T_1(z) = T_2(z) = T_3(z) = T_{f_1}(z) = T_{f_2}(z) \text{ are known} \tag{6}$$

Inlet condition at the surface (z=0):

$$T_{f_1}(t,0) = f(t) \text{ is a known function of time} \tag{7}$$

End condition at the bottom of the heat exchanger (z=L):

$$T_{f_1}(t,L) = T_{f_2}(t,L) \tag{8}$$

Boundary condition at the interface between the water and the inner pipe:

$$-k_1 \frac{\partial T_1}{\partial r} \Big|_{r=r_1} = h_1(T_1 - T_{f_1}) \tag{9}$$

Boundary condition at the interface between the water in the annulus and the inner pipe:

$$-k_1 \frac{\partial T_1}{\partial r} \Big|_{r=r_2} = h_2(T_{f_2} - T_1) \tag{10}$$

Boundary conditions at the interface between the water in the annulus and the outer pipe:

$$-k_2 \frac{\partial T_2}{\partial r} \Big|_{r=r_3} = h_3(T_{f_2} - T_2) \tag{11}$$

Boundary conditions at the interface between the outer pipe and the ground:

$$T_2 = T_3 \qquad (r=r_4) \tag{12}$$

$$k_2 \frac{\partial T_2}{\partial r} \Big|_{r=r_4} = k_3 \frac{\partial T_3}{\partial r} \Big|_{r=r_4} \tag{13}$$

Boundary condition at the assumed far field distance from the heat exchanger:

$$T_3 = T_3(z) \qquad (r < r_5) \tag{14}$$

Equations (9)-(11) are the boundary conditions for convective flow. When the fluid flow is stopped, such as between cycles, the heat transfer mechanism from fluid to tube walls changes from convection to conduction. The following conditions should be applied under these circumstances:

Heat flow at the interface between the water and the inner pipe:

$$\frac{\partial T_{f_1}}{\partial t} = \frac{2k_1}{\rho_f c_{p_f} r_1} \frac{\partial T_1}{\partial r} \Big|_{r=r_1} \tag{15}$$

Heat flow between the water in the annulus and the inner and outer pipes:

$$\frac{\partial T_{f_2}}{\partial t} = \frac{2r_3 k_2}{(r_3^2 - r_3^2)\rho_f c_{p_f}} \frac{\partial T_2}{\partial r} \Big|_{r=r_3} - \frac{2r_2 k_1}{(r_3^2 - r_2^2)\rho_f c_{p_f}} \frac{\partial T_1}{\partial r} \Big|_{r=r_2} \tag{16}$$

Boundary conditions at the interfaces between the water in the inner pipe and the annulus and the walls of the inner pipe:

$$T_1 = T_{f_1} \qquad (r=r_1) \tag{17}$$

$$T_1 = T_{f_2} \qquad (r=r_2) \tag{18}$$

Equations (15-18) assume that the temperatures of the fluid in the center and annular area are uniform and also do not account for a contact resistance between the fluid and center tube. Since the thermal conductivity of the center pipe should be kept as small as possible to minimize the energy short circuiting, the assumptions that there are no temperature gradients between the water and the walls of the inner pipe (Eqs. 17 and 18) should not cause any serious error. Equations (15-18) replace equations (1), (3), and (9) to (11) when the fluid is still.

The convective heat transfer coefficients for the water in the annulus need to be calculated depending on the vertical position in the heat exchanger. These computations are based on the work by Heaton et al [8] with the consideration of developing temperature and velocity fields. The convective heat transfer coefficient, h, is shown in Fig. 2 as a function of the distance below the surface for four different length heat exchangers. At the bottom of the heat exchanger, where the water in the annulus just starts to rise, the value of h is very high due to the fact that the velocity and temperature profiles are just starting to develop. The heat transfer coefficient then decreases as laminar flow is established as the water rises in the annulus.

4. COMPUTER MODEL

A finite difference computer program was written to solve the mathematical model numerically for either continuous or cyclic operation. It consists of the five partial differential equations which describe the temperature distribution in the water, the pipes, and the ground in terms of time, depth, and the distance from the heat exchanger. These are coupled radially by the boundary conditions imposed at the surface interfaces and vertically by the flow of the working fluid in the heat exchanger. The computer program uses an explicit solution of a finite difference approximation to this system of equations to calculate the temperature at fixed nodal points in the pipes, water, and ground. It employs a fixed time step and a fixed vertical spacing of nodes used for the solution while permitting a variable radial spacing of nodes in the ground. A check of the computations is made by calculating the amount of heat removed from the ground, pipes, and "resident" water (what is in the heat exchanger at start up) and comparing it to the heat removed from the system integrated over time (i.e., $\int \dot{m}C_p \Delta T dt$).

Variable h values were adopted even though the mathematical model assumes the uniform temperature and velocity profiles. Theoretically, the h values are different for the inner and outer tube walls of the annulus, but since the difference is small [8], the values calculated for the outer tube wall were used for inner tube wall as well. The fluid flow in the inner pipe is well within the region for turbulent flow and an appropriate heat transfer coefficient is used (3490

$W/m^2{}^\circ K$, 615 Btu/h $ft^2{}^\circ F$) [9].

The computer program has been written so that it is easy to use and does not require a familiarity with the partial differential equations. The input data consists of:

- the geometric specification of the heat exchanger (e.g., the length, ID's and OD's of the pipes),
- the physical properties of the materials used for the heat exchanger (e.g., the density, specific heat, thermal conductivity),
- the flow rate, density, and specific heat of the working fluid,
- the number of vertical nodes to use, and the number and spacing of nodes in the ground.

There are also options which allow a uniform initial temperature distribution in the ground or one that varies with depth. The user can also specify the inlet water temperature and flow rate as functions of time. The output from the program consists of a printed summary of the temperature distribution at fixed time intervals and also the average capacity of the heat exchanger.

The program is written in ANSI standard FORTRAN IV and requires 21K words of memory. It can require a lot of computer time to perform a long simulation since it must be able to model very rapidly changing transients. In most applications, however, the response of the ground to the heat exchanger approaches an asymptote and it should be possible to take advantage of this behavior in lieu of performing long simulations (in excess of 24 hours).

5. TEST RESULTS AND MODEL VALIDATION

A total of four tests were performed:

1. hot inlet water, about 43°C (110°F), continuous operation;

2. cold inlet water, about 4°C (40°F), continuous operation;

3. hot inlet water, about 43°C (110°F), cyclic operation, (2 cycles/hour with equal on and off times); and

4. cold inlet water, about 1°C (34°F), cyclic operation (2 cycles/hour with equal on and off times).

The temperatures were recorded every 2 minutes during the first, third, and fourth tests and were recorded every 10 minutes for the cold inlet water test with continuous operation. The flow rate was kept at 3.15×10^{-4} m^3/sec (5.0 gpm) for all the tests.

The ground temperature at the test site was expected to be around 13 to 14°C (55 to 57°F), however, an odd shape for the ground temperature profile was observed. All points within 6.1 m (20 ft) of the surface were found to be influenced by the ambient conditions. The temperature rises with the depth below 6.1 m (20 ft) until it reaches a maximum of 17.5°C (63.5°F) at 15.2 m (50 ft), and then decreases with further depth. The ground temperature at 47.2 m (155 ft) was around 16.7°C (62°F), which was also higher than expected. It is conjectured that the elevated and oddly shaped temperature profile was caused by the heat released from a large building 3.7 m (12 ft) from the heat exchanger.

6. VALIDATION FOR CONTINUOUS OPERATION

The ground temperature profile was measured before each test and was used as the initial conditions for each of the validation runs of the computer program. The simulations required specifications of convective heat transfer coefficients at the water-tube wall interfaces. The convective heat transfer coefficient for the inner tube wall was easily calculated [9] (since this flow is in the turbulent region) and was equal to 3,490 $W/m^2{}^\circ K$ (615 Btu/h $ft^2{}^\circ F$). The Reynolds number for fluid flow in the annular region for the hot inlet water tests was equal to 3,500, which was in the transition zone [10], although it is close to the laminar region. The Reynolds number for the water in the annulus for the cold water tests was in the laminar region (Re < 2,200). Consequently, we used the work of Heaton [8] to compute the heat transfer coefficients for the wells of the annular region, as previously mentioned.

Figure 3 shows the observed data and computer calculations for the two tests of continuous operation with hot and with cold inlet water. The four upper curves show, from the top to down,

- water temperature 46.6 m (153 ft) from the surface,
- water temperature 32.0 m (105 ft) from the surface,
- water temperature 16.8 m (55 ft) from the surface, and
- temperature at the outer edge of the casing 16.8 m (55 ft) from the surface.

The water and shell temperatures were chosen for validation of the model because they are the most fundamental measurements taken. These

curves show the test results for 6 hours of continuous operation with hot inlet water at 41.7°C (107°F) and then 4 hours of ground recovery. The solid lines are the calculated temperatures and the markers are used for the observed temperatures. The curves clearly show that the water temperature drops as it rises in the annulus between the tubes. After one hour of operation, the temperature difference between the inlet and outlet water was 5.4°C (9.8°F). This difference dropped to 4.2°C (7.5°F) after 6 hours, while the inlet water temperature was maintained constant, which indicates that the heat rejection to the ground was reduced by 23%. The ground temperature 0.34 m (1.1 ft) from the outside surface of the outer tube did not change at all during this time (measured but not shown on the figure). The shell outer edge temperature was much lower than the water temperature because the PVC pipe has a very low thermal conductivity coefficient, thus the tube wall acts as a layer of insulation.

After 6 hours of operation, the hot water was turned off in order to let the ground recover. After 4 hours of recovery, the water temperature in the annulus dropped from over 37.8°C (100°F) to an average of 25°C (77°F). The ground temperature at 0.34 m (1.1 ft) away from the coil rose by only 0.11°C (0.20°F) during the recovery period.

At the beginning of the heat exchanger operation, where the water and shell temperature changed abruptly, the computer program does not predict the data from the test very well. The model simulates the experimental results very well, however, after two hours of heat

exchanger operation as long as the water is flowing. When the water flow was stopped after 6 hours of operation, though, the computer model overpredicted the water temperature during the recovery period. When we added a condition to the program that eliminated the temperature gradient between the water and the inner wall of the outer casing, (by setting the water temperature equal to the outside wall temperature), the computer calculation matched the test results very well as shown by the dashed lines in Fig. 3. The added condition would underpredict the test result for the first half hour of water temperature recovery, though. This made the added condition undesirable for cyclic operation if the "off" cycle time were less than one hour.

It appears that when the water flow is stopped, a temperature differential between water and the inner wall of the casing persists but for a period of less than one hour. Since the heat exchanger was not adequately instrumented to allow further study of this question, we can only assume that this temperature differential is the source of the discrepancy between the observed and calculated temperatures.

Figure 3 also shows the data from 10 hours of continuous coil operation with cold inlet water at around 4.4°C (40°F). This test was continued for six days of operation and then six days of recovery, although not all of the data are shown. From the bottom up, these four curves are:

- water temperature 46.6 m (153 ft) from the surface,

- water temperature 32.0 m (105 ft) from the surface,
- water temperature 16.8 m (55 ft) from the surface, and
- temperature of the outer edge of the outer pipe 16.8 m (55 ft) from the surface.

It can be seen that after the first hour of operation, the temperature profile becomes very stable. The fluctuation of water temperature (as appears at 2 hours in Fig. 3) was due to minor changes in the inlet water temperature. The water inlet and outlet temperature differential was 1.6°C (2.8°F) at the end of 10 hours of operation and it was maintained even after six days of heat exchanger operation. The ground temperature, at 0.34 m (1.1 ft) away and 16.8 m (50 ft) below the surface, only dropped 0.6°C (1.0°F) after six days of continuous heat exchanger operation. The shell outer edge temperature was much higher than the water temperature at the same depth as shown in Fig. 3. Again, it shows that PVC has an insulating effect and is not a good material for this type of operation.

The model predicts the annulus water temperature at all levels within 0.6°C (1°F) of the test result. It also predicts the casing outer edge temperature within 0.9°C (1.3°F) of the measured temperature.

Figure 4 shows the observed shell and water temperature recovery when the water flow was stopped after 6 days of continuous coil operation for the cold inlet water test. Even though the water temperature recovered very rapidly at the beginning the temperatures did not recover to their original temperatures even after six days.

7. VALIDATION FOR CYCLIC OPERATION

Figure 5 shows the test results of heat exchanger cyclic operation with hot and cold inlet water. The water entering the ground coil was turned on during the first 30 minutes of each hour and then shut off to allow the ground to recover for 30 minutes. The top two curves are the water temperature in the annulus and the temperature at the outer edge of the shell 16.8 m (55 ft) from the surface for the test with hot inlet water. The bottom two curves are the temperatures at the same points for the cold inlet water test. Again, the solid lines are computed temperatures and the markers are observed values.

While the figure only shows the fluctuations of annulus water temperature 16.8 m (55 ft) below the surface, results indicate that the fluctuations of temperature occurred at all water levels. This figure shows that the peak temperature for the hot water test occurred at the end of each "on" cycle, and that the water temperature started dropping as soon as the water flow was stopped.

The shell outer edge temperature at 16.8 m (55 ft) below the surface had practically an asymptotic rising or falling trend for the two tests and only a slight temperature fluctuation was observed during the cycles. This was caused by two factors, one was the low thermal conductivity of the PVC material, and the other one was that when the water flow was stopped, the temperature of the water in the heat exchanger was still much different than the ground, which caused the ground temperature near the pipe to change continuously.

At the beginning of the first cycle, the model underpredicts the water temperature by almost 2.8°C (5.0°F) for the hot water test. After this cycle, however, the predictions are within 1.1°C (2.0°F) of the peak temperature. The model overpredicts the water temperature at the end of "off" cycle by as much as 1.4°C (2.5°F). The reason seems to be that the heat transfer mechanism, just after the water flow is stopped, is not well understood and consequently cannot be modeled accurately.

Except at the beginning of the first cycle of the cold water test, where the computer calculation mismatch the test results by 1.4°C (2.5°F), the calculated water temperature match the test results very well. After 6 hours of cyclic operation, the water flow was stopped. The model underpredicts the water temperature rise during this period (not shown in Fig. 5), again due to our lack of understanding of the heat transfer mechanism just after the water flow was stopped.

While the simulations were presented as the water and casing outer edge temperature 16.8 m (55 ft) below the surface, they match the annulus water temperature at almost all levels equally well. The exception to this is the top 3.0 m (10 ft) of the heat exchanger where the influence of the ambient temperature change and moisture migration in the unsaturated soil could not be handled by the model.

Based on the results given above, we conclude that the mathematical model predicts the test results very well when the heat exchanger

is operated continuously. By changing a boundary condition, the program also predicts the coil recovery temperature well if the coil was allowed to rest for at least one hour. For cyclic operation with a 30 minute recovery period, the model does not predict the coil recovery rate as accurately as we would like, particularly for hot inlet water. The calculation of water temperature at the beginning of each "off" cycles could be improved if we had a better understanding of the heat transfer mechanism between water and tube wall when the water flow is stopped abruptly.

The test results indicate that after the first couple of hours of coil operation the water and shell temperature change steadily, but slowly, due to the enormous amount of ground mass involved in energy transfer with the heat exchanger. It was felt that any computer simulation of tests longer than 10 hours of operation was unnecessary for validation of the model. However, for the design of the ground coupled heat exchangers, the involvement of the seasonal operation will require long periods of computer calculation of the heat exchanger performance. It should be possible to abbreviate these computations, though, by determining the asymptotic behavior of the system and extrapolating to a critical time, such as when the inlet water temperature or cycling behavior change dramatically, and then resuming the transient calculations.

8. PARAMETRIC ANALYSIS

A parametric analysis was performed to investigate the effect of variations in design parameters on system performance. The parameters involved in the actual system design which are of interest for this kind of analysis are the inner pipe and outer shell casing materials, the size of the outer casing pipe, the water flow rate, and the length of the heat exchanger. The model is limited in its application to some extent because as we increase the water flow rate, the Reynolds number will also increase and the flow can change from the laminar to the transition region (where heat transfer cannot be modelled accurately). Information about the convective heat transfer coefficient, h, in the transition region is extremely hard to find. The water flow rate involved in the analysis is therefore limited to 3.15×10^{-4} m^3/s (5.0 gpm) or less. The restriction on the maximum flow rate the model can handle will be higher for other working fluids (e.g. a glycol solution) than it is for water and may in fact allow all the flow rates that would be of interest to the user.

Figure 6 shows the effect of the the change of outer casing thermal conductivity values for continuous coil operation with hot inlet water, 42°C (107°F). It can be seen that when the casing thermal conductivity increases from 0.189 to 6.056 W/m°C (0.109 to 3.5 Btu/h ft°F), water temperature in the annulus 16.8 m (55 ft) below the surface shows an appreciable drop. In the first hour of operation it falls from 38.3 to 35.3°C (101 to 95.5°F), although the temperature differential becomes smaller as the coil operation continues. At the

end of 5 hours, the temperature varied from 38.7 to 37.3°C (101.6 to 99.2°F). The greatest improvement came from the change of thermal conductivity, k, value from 0.189 to 0.460 W/m°C (0.109 to 0.266 Btu/h ft°F), which corresponds to using high density polyethylene for the outer casing instead of PVC. Further increases in the casing thermal conductivity show a diminished effect on water temperature and hence heat exchanger capacity. The use of metal casings because of their superior thermal conductivity would not be justified based on this analysis because their performance is only slightly better while their cost is significantly higher.

Figure 7 shows the effect of the diameter of the outer casing on the heat exchanger capacity. These data were computed for heat exchangers made out of high density polyethylene (solid lines) and PVC (dashed lines) with inlet water at 42°C (107°F), a flow rate of 3.15 x 10^{-4} m^3/sec (5 gpm), and an inlet water to far field ΔT of 25C° (45F°). It indicates that larger casing diameter does improve the heat exchanger performance, although not in proportion to the increase in surface area. Table 1 shows this effect for PVC more directly; the second column is the ratio of the capacities of four points on the top dashed curve in Fig. 7 to the capacities of the corresponding points on the middle dashed line. Although the surface area of the heat exchanger has been increased by 60% the rate of heat rejection for the 0.20 m (8 in.) I.D. heat exchanger is at most 36% larger than that for the 0.13 m (5 in.) I.D. heat exchanger. The third and fourth columns

of Table 1 show similar results when a 0.13 m (5 in.) I.D. and a 0.20 m (8 in.) I.D. heat exchanger, respectively, are compared to a 0.076 m (3 in.) I.D. heat exchanger. The reason that the heat rejection rate does not move in direct proportion to the ratio of surface areas for the heat exchangers is the change in water velocities (resulting from the larger annular area), which affect the Reynolds numbers and hence the heat transfer coefficients.

Figure 8 shows the effect of ground thermal conductivity on the heat exchanger performance. Again, the solid lines are for a high density polyethylene heat exchanger and dashed lines for one made of PVC. It can be seen that by doubling the ground thermal conductivity from 3.03 to 6.05 W/m°C (1.75 to 3.50 Btu/h ft°F), the rate of heat rejection goes up by about 25% when the polyethylene heat exchanger is run continuously (and by lower percentages when the heat exchanger is cycled on and off). The corresponding increase for the PVC heat exchanger is only about 12%, and this again shows the insulating effects of the PVC pipe used for the outer casing.

Figure 9 has two sets of curves that show the calculated average capacity per unit length for tube-in-tube vertical ground coupled heat exchangers as functions of the length of the heat exchanger and of the operating time and volumetric flow rate. All six of these curves are for heat exchangers made out of PVC. The three solid lines are for a flow rate of 3.15×10^{-4} m^3/sec (5 gpm) for continuous operation, operation at 50% on time (and 3 cycles/hour), and for operation at 33%

on time (also at 3 cycles/hour). The dashed lines represent the corresponding capacities for a water flow rate of 1.89×10^{-4} m^3/sec (3 gpm). Several things can be observed about the capacity per unit length from this figure:

1. it is as much as 90% as high at 1.89×10^{-4} m^3/sec (3 gpm) as it is at 3.15×10^{-4} m^3/sec (5 gpm), and

2. under cyclic operation it does not decrease in proportion to the operating time (e.g. the heat exchanger rejects 48% as much heat to the ground when it runs 20 minutes out of each hour as it does when it runs continuously).

These computations were done for hot inlet water (42°C, 107°F), 16.7°C (62°F) ground temperature, and schedule 40 PVC for both pipes.

The decrease in heat exchanger performance per unit length is a result of the variations in the heat transfer coefficients at the interface between the water in the annulus and the outer casing. In our application the correlation that was used for the heat transfer in regions of developing laminar flow gives the heat transfer coefficients in terms of the distance from the bottom of the heat exchanger. Consequently, for example, the heat transfer coefficients for the bottom 61 m (200 ft) of a 122 m (400 ft) heat exchanger are the same as those for the entire length of a 61 m (200 ft) heat exchanger. The heat transfer coefficients for the top 61 m (200 ft), however, are significantly less since laminar flow is fully developed, and hence decrease the average per unit length performance of the heat exchanger.

Figure 10 is a plot of the average capacity of a 61 m (200 ft) heat exchanger with a 3.15 $x10^{-4}$ m^3/sec (5 gpm) flow rate at several inlet water temperature to far field temperature ΔT's and schedules of operation (continuous operation, 50% on time, and 33% on time). The solid lines are for a heat exchanger made out of high density polyethylene and the dashed lines for PVC. In each case the ambient ground temperature was assumed to be 16.7°C (62°F) and the water temperature into the coil was varied above or below this. The performance was found to be independent of the sign of the temperature differential and was nearly linear in the temperature differential.

9. DISCUSSION AND CONCLUSION

The mathematical model for the vertical ground coupled heat exchanger has been validated by laboratory tests for both continuous and cyclic operation. The parametric study shows that using high thermal conductivity material will increase the amount of energy exchange between the heat exchanger and ground, but the improvement becomes negligible when the casing thermal conductivity was close to, or higher than, that of the ground. In other words it would not be cost effective to use metal casing for its high thermal conductivity value.

The parametric study also shows several possible ways of designing a higher capacity heat exchanger:

1. increase the flow rate of the working fluid,
2. increase the diameter of the outer casing, and
3. increase the length of the heat exchanger.

There are other factors that need to be taken into account, however, when making any one of these changes independently.

Increasing the flow rate will increase the amount of heat transfer, due to the increase of convective heat transfer coefficients between the water and the wall of the casing, but the overall rate of heat rejection does not necessarily increase in proportion to the flow rate. The effect of changing the flow rates is greater for long heat exchangers than it is for short ones and is also more significant during cyclic operation than it is for continuous operation. This result has not been fully explained. A 67% increase in flow rate from 1.89 x 10^{-4} m^3/sec (3 gpm) to 3.15 m^3/sec (5 gpm) increases the heat transfer coefficient between the water in the annulus and the outer casing by from 15–24%. The increase in capacity, however, can be as small as 6% for a 30 m (100 ft) heat exchanger running continuously to as high as 56% for a 122 m (400 ft) heat exchanger cycling with a 33% on time. Heat exchanger operation appears to be limited by the thermal conductivity of the outer casing and the ground under some circumstances. The ground can only absorb or give up so much heat regardless of how great the flow rate of water through the heat exchanger. This limit is reached during continuous operation, even at low flow rates, and there is nothing to be gained by a higher flow

rate. Cyclic operation permits some time for the ground to recover in between periods of heat exchanger operation and consequently the limit is higher. This amount of heat may not be able to be absorbed or rejected at the lower flow rates and the flow rate can be the limiting feature instead of the ground thermal conductivity. Further study is needed to understand this interaction better.

The length of the heat exchanger affects its capacity and the effective length of a heat exchanger can be increased by either making a longer single tube or by putting several heat exchangers either in series or in parallel. The decreasing heat transfer rate per unit length of heat exchanger, as shown in Fig. 9, would suggest that several shorter heat exchangers would be better than a single long one. Unfortunately this conclusion assumes that each of the multiple heat exchangers have the same flow rate and entering water temperature as the single long one. Further analysis shows that the advantage of multiple heat exchangers over a longer one is very small when these factors are taken into account.

effects of percentage Δ's

Figures 9 and 10 can be used to compare two 61 m (200 ft) PVC heat exchangers with a single 122 m (400 ft) one (under continuous operation with a flow rate of 3.15×10^{-4} m^3/sec (5 gpm) and inlet water temperature of 42°C (107°F) and 25 C° (45 F°) inlet water to far field ΔT). If the two heat exchangers are connected in series the first one will have a capacity of 6.59 kW (22,500 Btu/h) and the exit water temperature can be computed to be 37°C (98°F). This gives a 20

C° (36 F°) inlet water to far field ΔT for the second heat exchanger, which would then have a capacity of 5.27 kW (18,000 Btu/h). The combined capacity is 11.86 kW (40,500 Btu/h) compared with the 11.72 kW (40,000 Btu/h) for a single 122 m (400 ft) heat exchanger under comparable conditions.

If the heat exchangers are connected in parallel, the flow rate into each would have to be one half of that for the single longer heat exchanger in order to be able to compare the two cases. Extrapolation from the two curves in Fig. 9 gives a capacity of 5.71 kW (19,500 Btu/h) for each of two 61 m (200 ft) heat exchangers at 1.58×10^{-4} m^3/sec (2.5 gpm) for a combined capacity of 11.43 kW (39,000 Btu/h). This is actually less than the capacity of the 122 m (400 ft) heat exchanger, although that may be due to the uncertainty resulting from the extrapolation. Consequently, it appears that if there is an advantage to multiple heat exchangers over a single long one, it is at most marginal.

There are several areas in which the computer model needs to be refined:

- the heat transfer between water in the annulus and the outer casing during cyclic operation needs to be improved,
- the convective heat transfer when the flow is in the transition region cannot be modelled as the program is written and a correlation for this region is needed, and
- the heat transfer in unsaturated soil near the surface (where moisture migration and the influence of the ambient air temperature affect performance) cannot be modelled accurately and this effect needs to be accounted for.

The program is easy to use, however, and the calculations are accurate as long as these limitations are not strained too much.

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Table 1. Comparison of relative heat rejection rates for different length and diameter heat exchangers.

Length of HX	Ratio of Surface Areas 1.60	1.67	2.67
100 ft	1.36	1.41	1.91
200	1.34	1.38	1.85
300	1.32	1.36	1.80
400	1.31	1.34	1.75

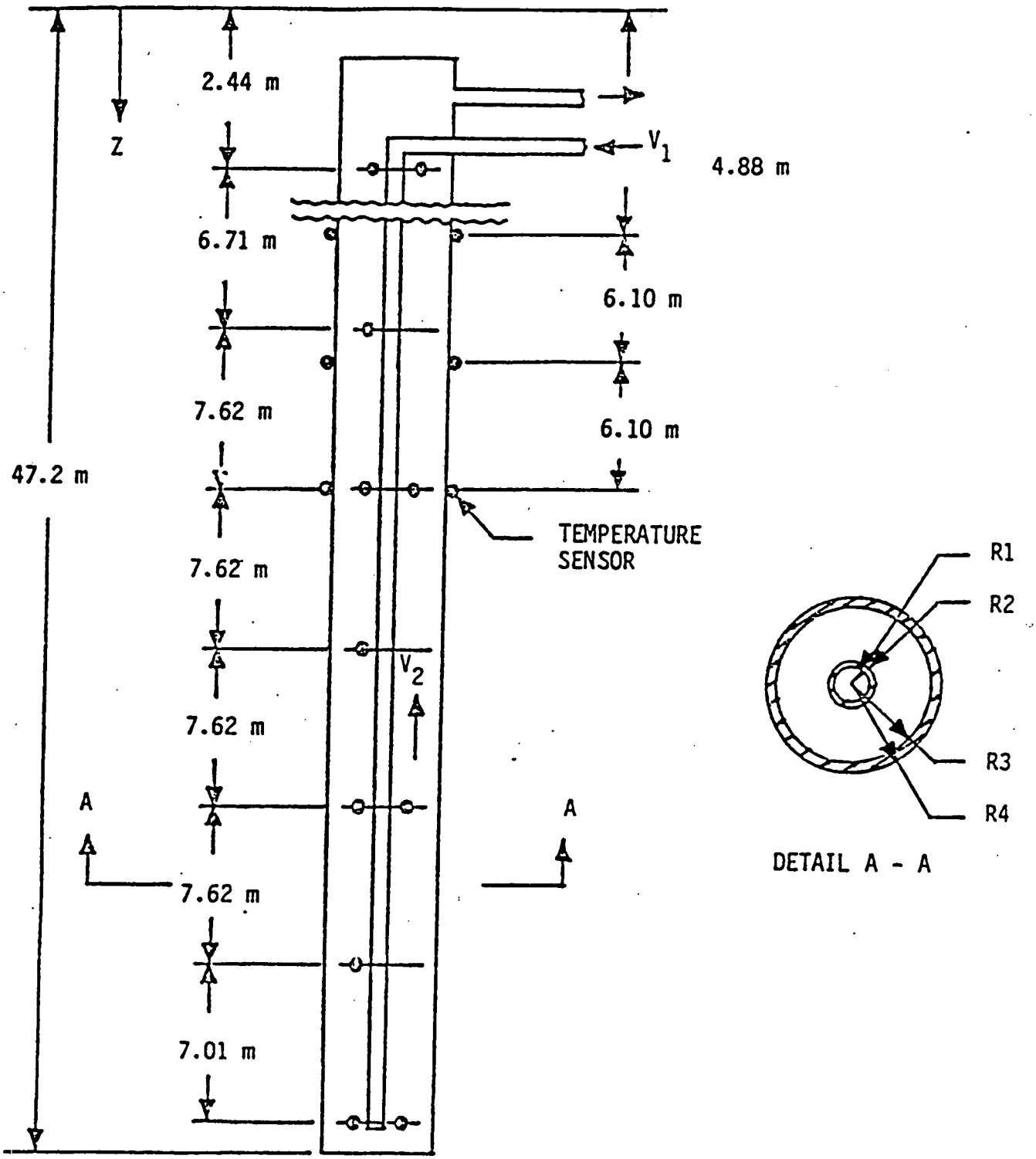

Figure 1. SCHEMATIC OF VERTICAL TUBE IN TUBE GROUND COIL

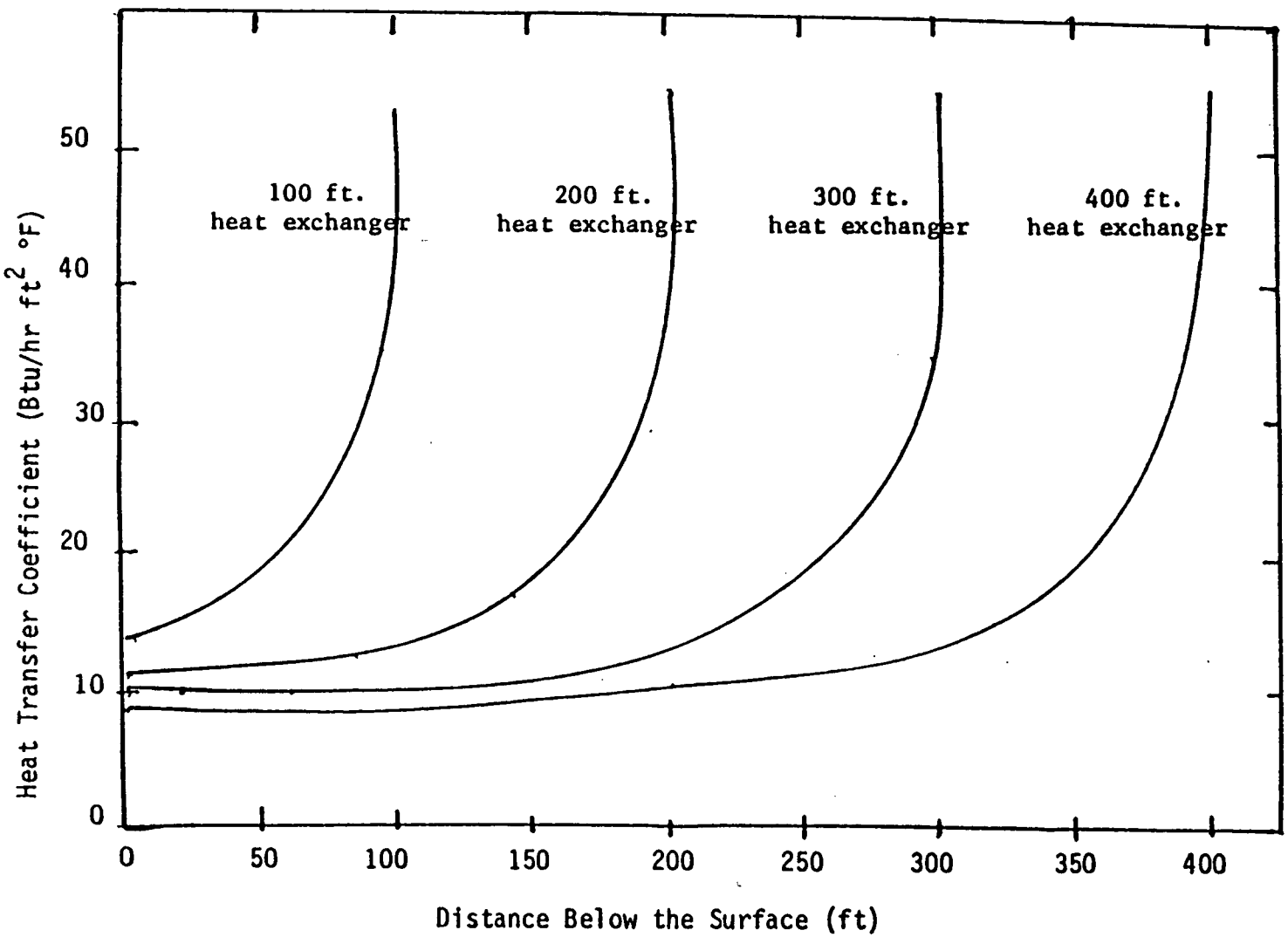

Fig. 2 Heat transfer coefficients in regions of laminar and deveolping laminar flow.

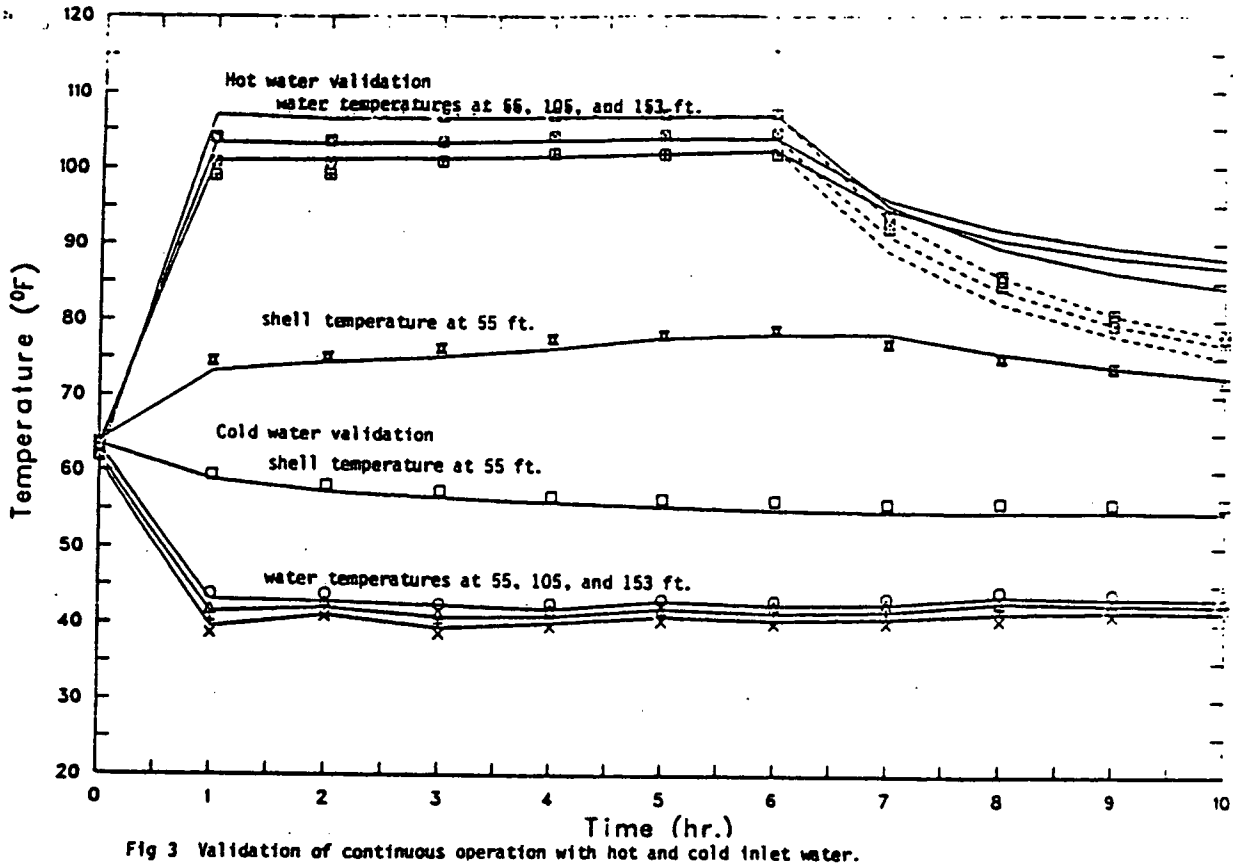

Fig 3 Validation of continuous operation with hot and cold inlet water.

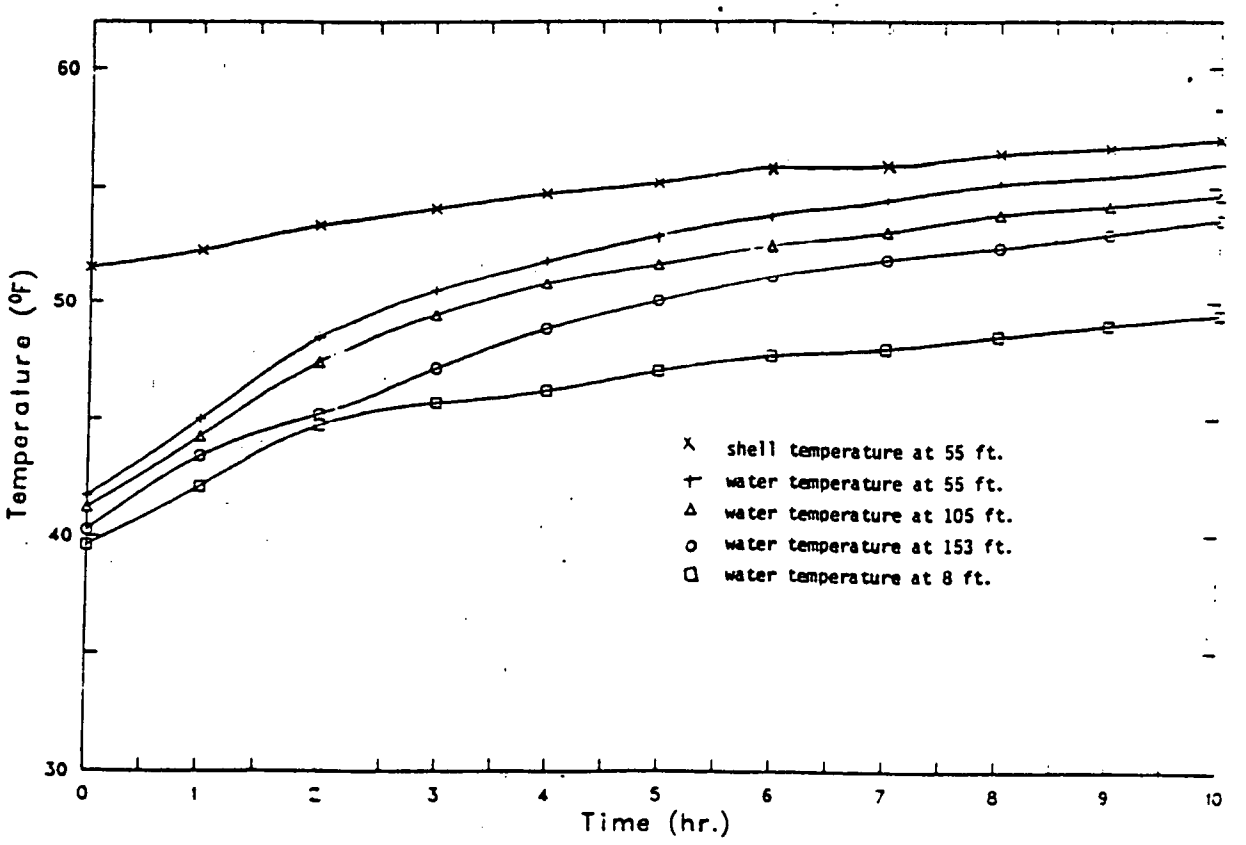

Fig. 4 Observed water and shell temperatures during ground recovery after six days of operation.

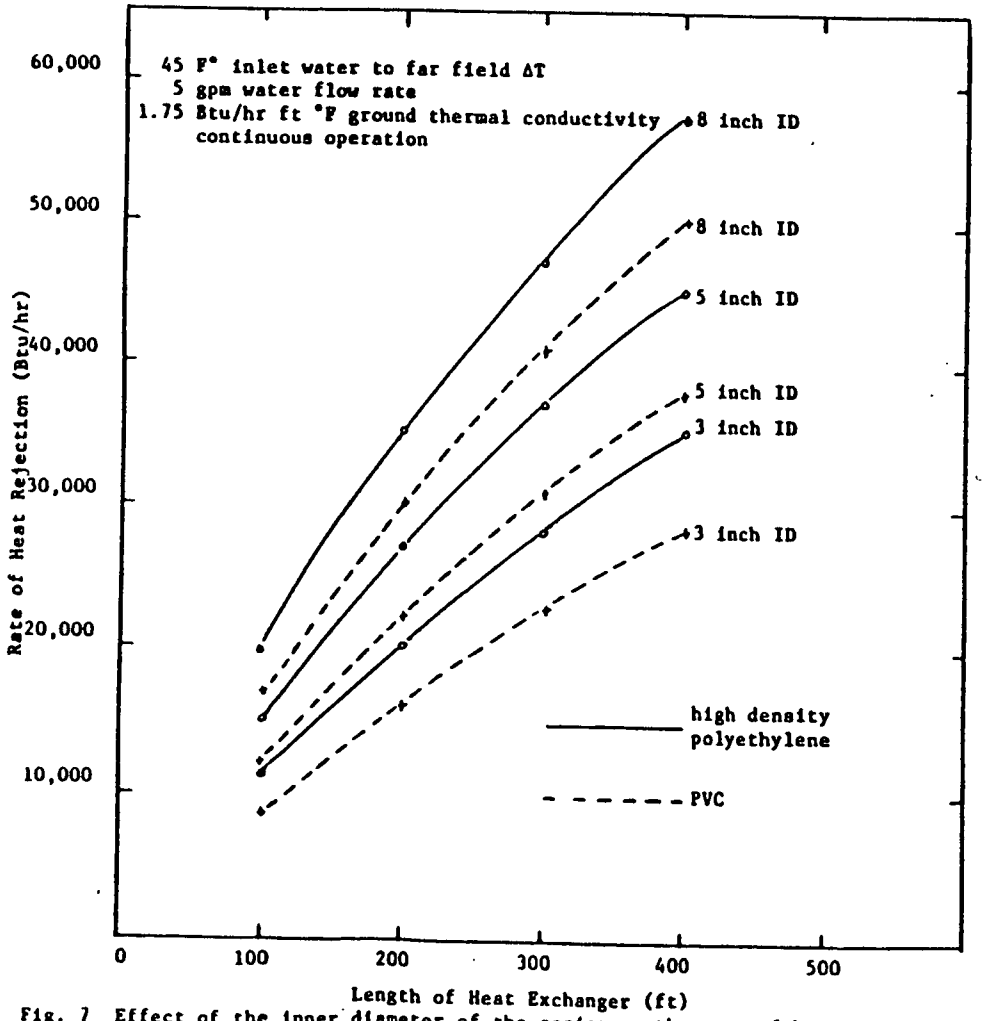

Fig. 7 Effect of the inner diameter of the casing on the rate of heat rejection to the ground.

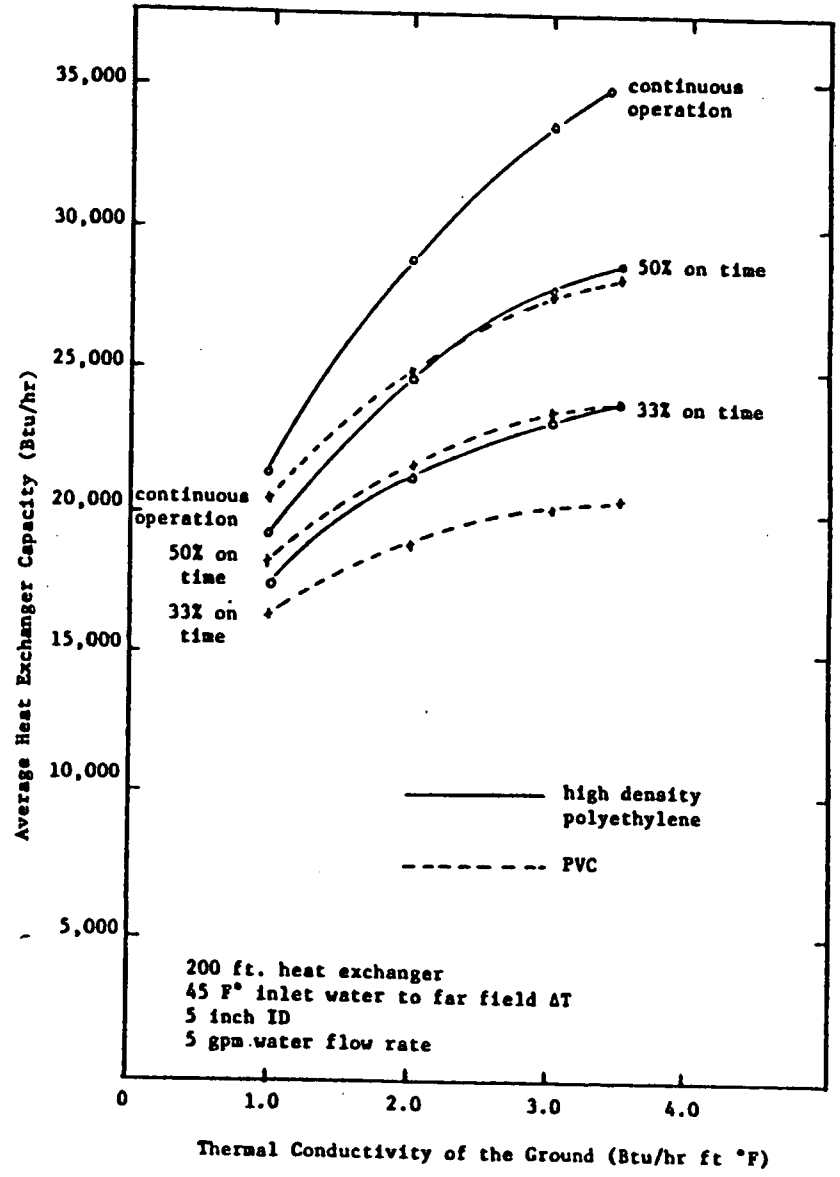

Fig. 8 Effect of the thermal conductivity of the ground on heat exchanger perform

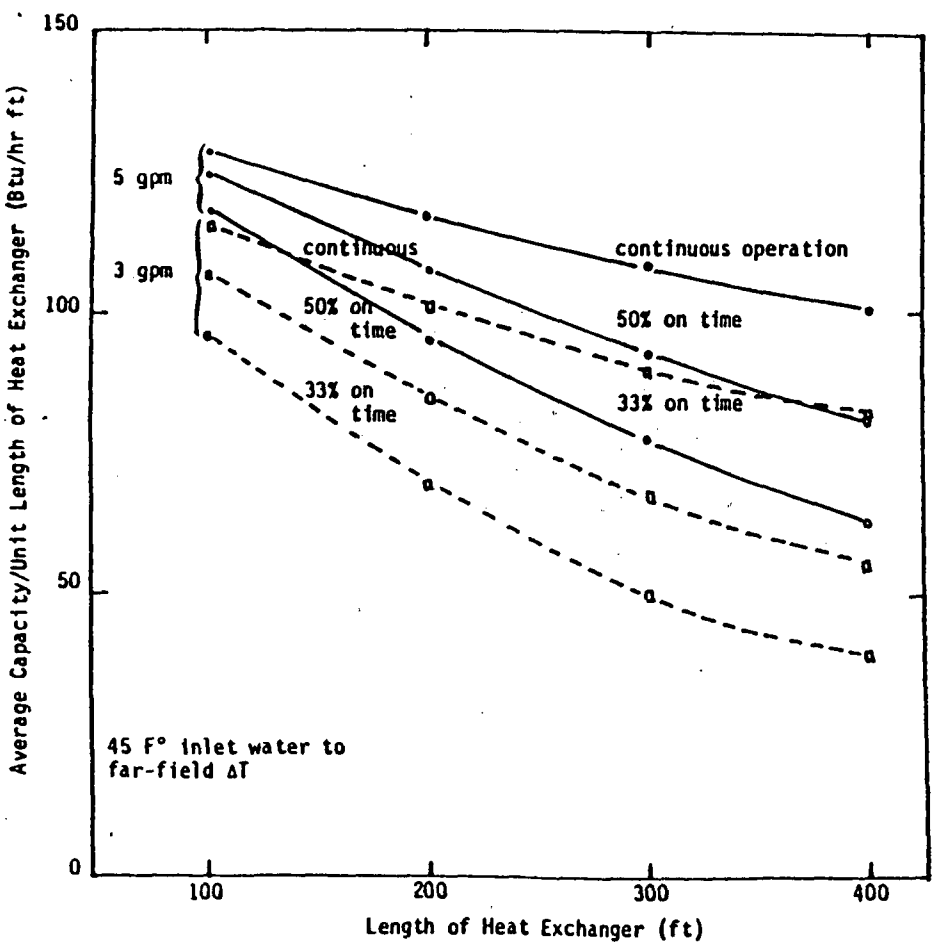

Fig. 9 Average capacity per unit of heat exchanger length for different lengths of heat exchangers, flow rates, and operating conditions.

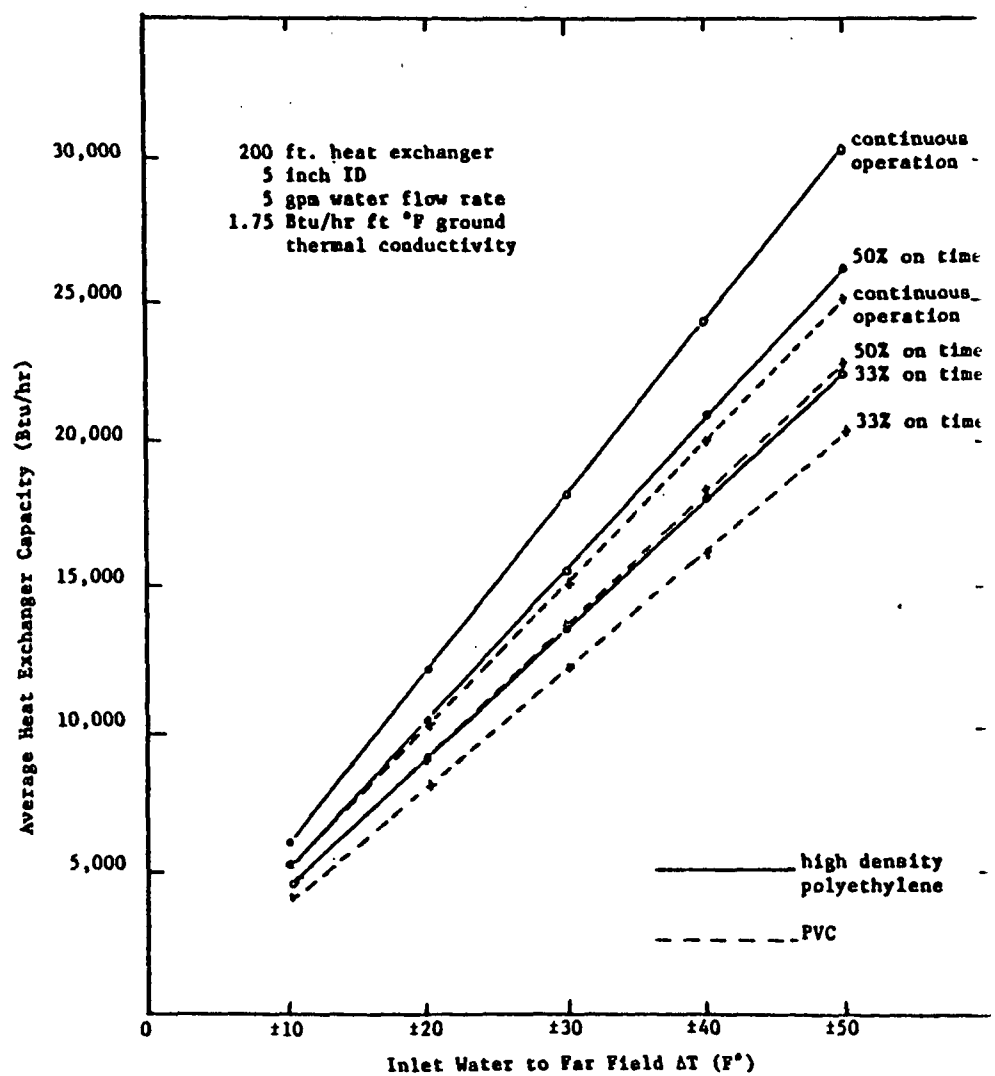

Fig. 10 Dependence of average heat exchanger capacity on the inlet water to far field temperature difference.

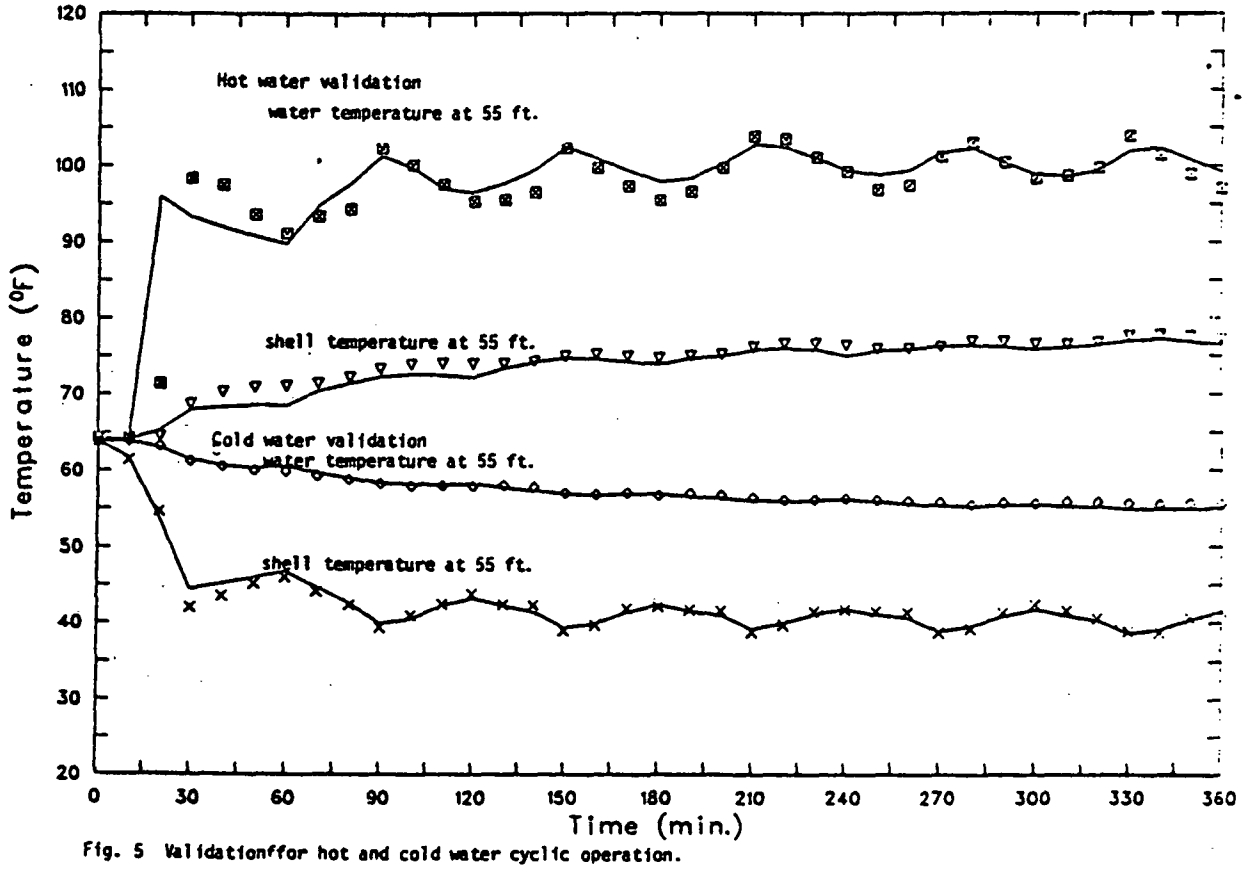

Fig. 5 Validationffor hot and cold water cyclic operation.

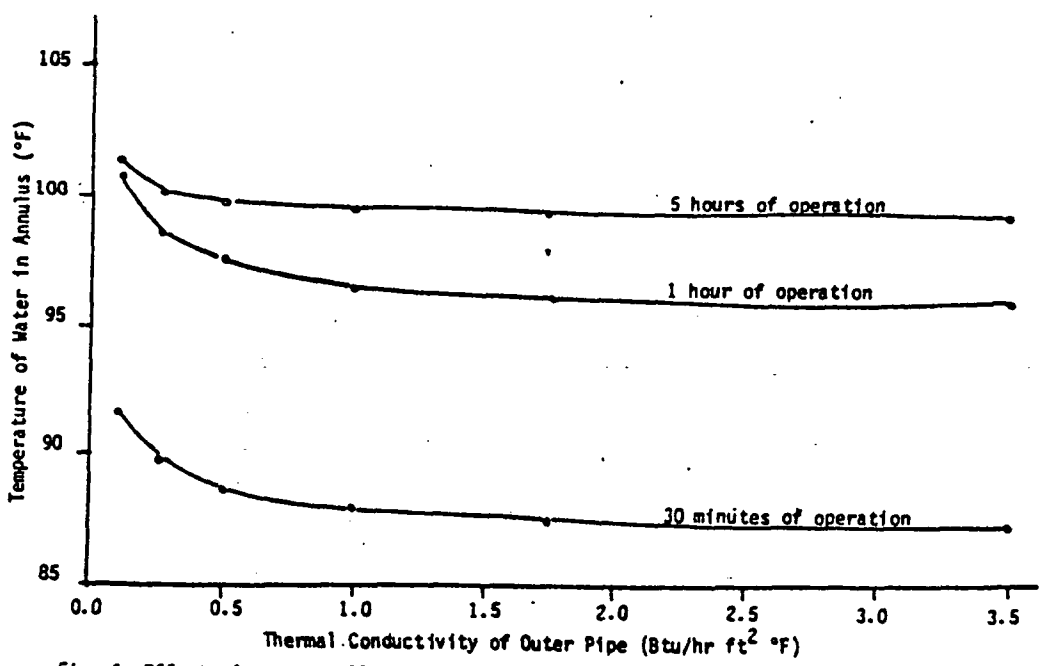

Fig. 6 Effect of outer shell thermal conductivity on the heat exchanger performance