THE THERMAL PROPERTIES OF SOIL IN RELATION TO THE HEAT PUMP

by

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INTRODUCTION

The heating of living spaces has always been a problem of human comfort. There are many methods of heating and each method has its advantages and disadvantages. Electric resistance heating has been recognized as ideal but is unable to compete with other methods of heating on a cost basis. By use of a heat pump, it may be possible to get 3 to 5 kilowatt hours of equivalent heat for each kwh of energy supplied and thus enable electricity to compete with other fuels. The heat pump also possesses the additional advantage of being able to provide summer cooling with the same equipment.

There is considerable interest in the heat pump both from the users' and manufacturers' standpoints, and much research in various phases of heat pump development is being carried on by both public and private agencies. The heat pump offers possibilities of developing new types of heating and cooling equipment for year-round comfort. One of the most important and rapidly developing markets for the heat pump is in the rural areas. In these areas the heat pump offers year-round air conditioning which is not now available.

The heat pump offers a means of increasing the electrical load in the rural areas. In the low density population regions where the farms are mainly grain farms, the uses of electricity are mainly centered in the homes. That means that the cooking and water heater loads are basically important. The widespread use of such fuels, as liquified petroleum gas and petroleum for home heating, means a wider application of them for cooking and water heating. Therefore, electricity must be able to compete economically in the space heating field, if it is to maintain the cooking and water heating loads. Without these two loads, the central station rural electric service in the low density regions may not be on a sound economic basis. Therefore, the heat pump represents more than just a new load to the power supplier; it is a factor in providing a sound economic foundation for supplying electric service to the sparsely settled farm areas.

HEAT PUMP PRINCIPLES

The heat pump employs the standard compression refrigeration system. In this system there are four component events in the cycle. They are compression, condensation, expansion and evaporation. The refrigerant in the vapor form is drawn into the compressor where it is compressed and discharged to the condenser. This high pressure vapor has a temperature in the vicinity of 100° F. The air in the living space is warmed by heat transferred to the air from the condenser when the refrigerant changes to liquid on the inside of the condenser. The high pressure liquid refrigerant then passes through the expansion valve into the evaporator. The evaporator is connected to the suction side of the compressor and a low pressure is maintained in the evaporator where the refrigerant boils and extracts heat from the medium surrounding the evaporator. This medium is the heat source for the heat pump.

The evaporator may be placed in air, water, or earth, which are the general sources of heat for the heat pump. In many places the air is too cold to be an economical source of heat. Water is a good source except that it is unavailable in many places. Ground water at depths of 100 feet or more is not economical because of high pumping costs. The earth, however, is available as a heat source on a more universal basis.

The heat that is put into the living space is the heat taken from the source plus the heat of the work put into the system. For summer cooling the evaporator and condenser are usually interchanged so that the heat is pumped out of the living space.

The coefficient of performance, usually abbreviated COP, of a heat pump for winter heating is defined as a ratio of the quantity of heating produced to the quantity of energy supplied. A COP of 3 to 1 is usually considered necessary to make the heat pump economical as a heating method. The COP as a cooling machine for summer cooling is defined as the ratio of the quantity of cooling produced to the quantity of energy supplied. The COP for cooling is less than the COP for heating because the energy supplied is not useful in the living space and must be dissipated as is the heat removed from the living space.

REVIEW OF LITERATURE

According to Penrod (10) the principles of operation of the heat pump were first proposed by William Thomason (Lord Kelvin) in a paper presented at a meeting of the Glasgow Philosophical Society in December, 1852. Although this machine was never built, refrigeration machines operating on the Kelvin air cycle did come into common use.

According to Ambrose (1) an ideal source of heat for the heat pump is one which is abundant and inexpensive with an average temperature of 40° F. to 80° F. throughout the year. There are three common sources which may meet those requirements. They are the air, water, and soil. The air would be a satisfactory source of heat when the outdoor temperature exceeds 35° F. according to Sporn, Ambrose and Baumeister (12). However, in many areas the air temperature will be lower than that and remain lower for extended periods of time. According to Stringfield (13) the temperature of the groundwater in non-thermal wells at a depth of 30 to 60 feet is satisfactory as a heat source in nearly all of the United States. In Kansas the ground water temperature at those depths ranges from 54° F. in the northern part of the state to 59° F. along the southern border.

Kemler (8) says that, if the earth were used as a heat source at depths of 30 to 60 feet, very good operating conditions would exist because the average ground temperature would be near the yearly average air temperature. Under those conditions a reasonably high COP could be attained. Although, at all reasonable depths for a horizontal ground coil, the soil temperature is not uniform throughout the year, the soil temperature peaks lag behind the air temperature peaks. He further shows that at greater depths, there is more lag and the ground temperature variation throughout the year is less. Coogan (4) by methods of extrapolation determined that at a depth of 40 feet at Storrs, Connecticut, a non-varying isotherm of 54° F. would exist. Kemler (8) shows that at a depth of 6 feet the coldest temperature of the year occurs in March whereas the warmest temperature occurs in September. Both the high and the low peak temperatures occur at a time of the year when the demand on a heat pump is usually very small. At shallower depths than 6 feet the lag is less and a peak demand might occur at the same time as the lowest output of heat.

According to Kemler (8) there is no question of the capacity of the earth to supply the heat. The big problem is how to obtain this heat.

Dana (5) shows that it is possible to extract heat from the earth with either a horizontal or a vertical pipe buried in the soil. In each case the heat flow to the pipe is dependent upon the soil conditions and the temperature gradient established in the soil.

The soil is nonhomogenous, and any particular volume of it will contain several substances. The three substances concerned are soil particles, soil air, and soil moisture. Considering the transfer of heat through soils, Patten (9) states that heat will pass from soil grain to soil water 150 times faster than it will pass from soil grain to soil air. Smith and Byers (11) state that for dry soils the thermal conductivity of the actual soil materials varies little from one soil to another. This would indicate that the thermal conductivity is a function of the density and the moisture content of the soil. Values of thermal conductivity as reported by Hogentogler (6) and Kemler (7) are shown in Tables 1 and 2 respectively.

Table 1. Thermal conductivity of various soils.

Material	K, $\frac{(\text{Btu})\ (\text{in})}{(\text{sq ft})\ (\text{hr})\ (^{\circ}\text{F.})}$	K, $\frac{(\text{Btu})\ (\text{ft})}{(\text{sq ft})\ (\text{hr})\ (^{\circ}\text{F.})}$
Soil in earth's crust, dry	10.7	0.891
Sand, white dry	2.70	0.225
Diatomaceous earth	0.377	0.031
Mica	5.22	0.435
Chalk	6.38	0.53
Peat, dry	0.348	0.029

Table 2. Thermal conductivities of soil and related materials.

Material	K, $\frac{(\text{Btu})\ (\text{ft})}{(\text{sq ft})\ (\text{hr})\ (^{\circ}\text{F.})}$
Very light and dry soil	0.21
Clay soil (65.7)	0.14
Clay soil (78)	0.26
Clay soil (96)	0.38
Dry soil	0.08
Wet (maximum) soil	2.62
Wet soil	0.39
Moist soil	0.83
Dry clay soil	0.50
Moist clay soil	0.90
Well drained soil	0.55 to 0.57
Wet sub soil	1.34

Table 2. (concl.).

Material	K, $\frac{(Btu)(ft)}{(sq\ ft)(hr)(^{\circ}F.)}$
Dry sandy soil	0.45 to 0.65
Moist sandy soil	1.1
Soaked sandy soil	2.4
Dry sand	0.19 to 0.22
Moist sand	0.67
Sawdust	0.93

(Figures in parentheses are densities in pounds per cubic foot)

According to Dana (5), when heat is absorbed from the earth by a buried ground coil, the moisture in the soil will migrate toward the cold pipe. A water vapor pressure gradient will exist in the same direction as the temperature gradient. He further states that in winter heat pump operation, this is an advantage because the thermal conductivity of the soil is increased by this additional moisture. He also shows that the thermal conductivity of ice is higher than that of water, indicating that a region of frozen soil would be a better thermal conductor than wet soil. Coogan (4) found this to be true. This moisture movement characteristic makes it hard to predict the thermal conductivity which would occur under operating conditions for a heat pump.

Budenholzer (2) using a ground coil evaporator temperature of 20° F. obtained 30 $\frac{Btu}{(hr)(ft)}$ with 3/4 inch copper pipe coils buried 6 feet deep and spaced 6 feet apart. In that test he found that the ground temperature near the pipe had to be reduced

1° F. for each Btu obtained per hr/ft of pipe.

Dana (5) says that spacing the heat exchanger pipes of the ground coil system too close together will result in a reduced capacity for each of the adjacent pipes. Coogan (3) presents a mathematical analysis for determining the quantity of heat that can be absorbed from the soil by closely spaced pipes. He obtained reasonable agreement between the mathematical analysis and experimental results. He assumed that the undisturbed earth temperature is equal to the average undisturbed earth temperature between the surface and 12 feet down. He shows this equation:

$$q = \frac{2\pi k\,(T_g - T_c)}{\ln\left[\frac{4D}{d} - 1\right] + \frac{1}{2}\ln\left[\frac{4D^2 - 2\,Dd}{z^2 + \frac{d^2}{4}} + 1\right]}$$

where

q = heat absorbed Btu/hr/ft of pipe

T_g = average undisturbed earth temperature, $^{\circ}$F.

T_c = surface temperature of coil

D = depth of pipe below surface, ft

d = pipe diameter, ft

k = thermal conductivity of soil $\frac{\text{Btu (ft)}}{\text{(hr)(sq ft) }(^{\circ}\text{F.})}$

z = pipe spacing, ft

ln = natural logarithm

Coogan (3) also presents a comparison between the heat available from one tube of a closely spaced pair of tubes and the heat available from a single tube. He makes the basic assumptions of equal temperature differences and equal soil thermal conductivity for the two conditions. The symbols are the same as above with the addition of F, which is defined as the ratio between the heat absorbed per linear foot of one tube, when there are two tubes in the ground, and the heat absorbed per linear foot of tube when there is only one tube in the ground.

$$F = \frac{1}{1 + \dfrac{\dfrac{1}{2}\ln\left[\dfrac{4D^2 - 2Dd}{z^2 + d^2/4} + 1\right]}{\ln\left[\dfrac{4D}{d} - 1\right]}}$$

The heat available from two different spacings a 10-foot and a 2-foot spacing of 1 inch copper pipe (1.125 inch o.d.) at a depth of 6 feet will be compared:

$$F_{10} = 0.925$$

$$F_2 = 0.754$$

This comparison shows that the 10-foot spacing would produce 18 percent (based on 10-foot spacing) more heat per foot under the same conditions than the 2-foot spacing.

THE INVESTIGATION

The object of this investigation was to determine the thermal energy flow characteristics around a ground coil for two different spacings of the coil and for different operating conditions.

The flow of thermal energy within the soil was to be caused by circulation of a controlled temperature liquid in the ground coil at some temperature other than the undisturbed soil temperature at that depth. There were two ground coils used in this investigation. A ground coil consisted of a 100 ft length of 1 inch i.d. (1.125 inch o.d.) copper pipe placed in a horizontal U shape at a depth of six feet in sandy soil.

In one trench the legs of the U were 10 feet apart and in the other they were 2 feet apart. Suitable connections to surface equipment were allowed for at the ends of each of the 100 feet long test section.

As actual operating conditions of a heat pump would have been difficult to duplicate, it was decided to use a continuous operation heat pump to try to establish equilibrium soil temperature conditions. It was thought that the equilibrium conditions would not be reached in a time shorter than the duration of the worst cold spells during a Kansas winter. The range of operating conditions in the ground coil was to duplicate both winter and summer continuous operating conditions. As nearly as possible, the individual tests of this investigation were to occur at the time of year which might most nearly duplicate the worst heat pump operating conditions as far as the soil was concerned.

In order to know what the rate of heat flow was within the test section, it was necessary to know both the rate of fluid flow and the temperature change of the fluid in the test section. It was also necessary to know the temperature gradient within the soil. The most practical means of obtaining temperature data both in the soil and on the pipe seemed to be by the use of thermocouples. These were to be used with a potentiometer so that direct temperature readings could be obtained without reference to tabular data.

LAYOUT OF TEST EQUIPMENT

Temperature Measuring Equipment

The measurement of soil temperature was made by the use of copper-constantan thermocouples. This wire had an enamel finish and was wrapped with nylon. In order to waterproof the thermocouple electrical circuit, the completed thermocouples were immersed in an emulsified asphalt solution. This solution dried upon exposure to air and left the thermocouples waterproof. Because of the long length of thermocouple wire required, a thermocouple circuit employing one common wire was used to reduce the length of the constantan wire purchased. In this method the bare copper wire was wrapped around the bare constantan wire at the point where the temperature was to be measured. The junction was then soldered previous to being dipped in the emulsified asphalt solution. The several copper wires were connected to a selector switch so that the circuit containing the desired thermocouple could be located through the switch arrangement. The individual wires were made long enough to reach to a switch panel located inside the housing which was used to protect the surface equipment. For most of the first test, a millivoltmeter was used to determine the temperature at the various thermocouple locations. A potentiometer with the scale graduated for a copper constantan thermocouple was used for the remainder of the tests.

The thermocouples were located in the ground as is shown in Figures 2 and 4. These cross sections were located at a point 20 feet-in a direction along the pipe-away from the beginning of the test section. The temperatures at the entrance and exit of the test sections were determined by means of a thermocouple soldered onto the copper pipe at each of those points.

Heat Flow Measuring Equipment

The measurement of the quantity of heat flow required the measurement of the quantity of fluid circulated and the temperature difference between the entering and leaving fluid. The temperature difference was measured by thermocouples at the exit and entrance of the test section whereas the rate of fluid flow was measured by means of a calibrated water meter. Two meters were required, one for each test section. Both of these meters were equipped with hot water discs and were graduated in gallons. In the test using calcium chloride brine, it was necessary to determine the specific gravity of the solution. Therefore, the following formula was used to determine rate of heat flow in Btu/ (hr) (ft) and the symbols are explained immediately following:

$$Btu/(hr)\,(ft) = Q_m \times K_m \times W \times \frac{S}{L} \times dt$$

Btu/(hr) (ft) = British thermal units per hour per foot of pipe.
Q_m = fluid flow rate through meter in gallons per hour.
K_m = meter correction factor.
W = pounds of fluid per gallon.
S = specific heat of fluid, Btu/pound.
L = length of test section in feet.
dt = temperature difference of entering and leaving water, degrees Fahrenheit.

The fluid was circulated by means of a pump and the rate was controlled by means of valves in the line. On two of the tests a pipe circuit with control valve was used to by-pass the pump. This by-pass line connected the discharge side of the pump to the suction side and afforded a means of excellent control on a centrifugal pump. With this by-pass valve fully closed the entire discharge of the pump had to go through the test section. This method also allowed positive shut-off of the fluid flow by means of another valve without adjustment disturbance of the control valve when it was necessary to cease operation for needed repairs. This method was used with both a centrifugal pump and positive displacement pump.

On the other tests a single suction centrifugal pump was used as a circulator. The rate of flow was controlled by the use of restricting valves in the line.

Temperature Control Equipment

For the tests in extraction of heat from the soil, it was necessary to have refrigeration machinery. A milk cooler tank with about 300 gallons liquid capacity was obtained for use as a space to immerse the evaporator coil of the condensing unit in the fluid. The tank had a lid and the necessary connections for piping and refrigeration control. In the first test, two small condensing units were used and they later proved to be inadequate due primarily to the large heat gain within the building. These units were a 3/4 horsepower and a 1/3 horsepower condensing units. For the two tests with calcium chloride brine as the circulated fluid, a 3 horsepower condensing unit was used to remove the heat from the brine in the tank. The tank was provided with baffles so that the brine went under one baffle and over the next. There were 10 baffles in all. An extra circulation pump was provided to keep the brine at nearly uniform temperature conditions throughout the tank. The temperature controls on the condensing unit were set at a differential temperature of 1° F. This control included a mercury switch which was rather sensitive to vibration. The part of the controls containing the mercury switch was mounted on the wall, and whenever the compressor started, the vibrations were enough to cause a break of contact with the mercury column. This resulted in some blown fuses and one abandoned test. The situation was corrected by mounting that part of the controls on a post just outside the housing.

The method used to simulate summer cooling with a heat pump was to circulate hot water in the ground coil. The water was heated by electric immersion heaters placed near the bottom of an insulated tank. The liquid immersion thermostat mounted on the side of the tank had a range of 60° to 160° F. with a 1° F. differential. The tank had a liquid capacity of 100 gallons. The electrical demand of the immersion heaters was 4,200 watts.

Housing

With as much equipment as was necessary for this investigation, some housing was mandatory. A war surplus bus body served as the housing for the equipment. It served the need quite well except for the previously mentioned vibration. All surface equipment was housed in this old bus body. Because of the length of time required for personnel to be in the enclosure, some supplementary heat was necessary during the wintertime. All the thermocouple leads were brought inside to a switch panel so that the temperature data could readily be observed and recorded. The time required for that operation varied from 45 minutes to 2 hours.

During the late spring of 1949 it became necessary to move the housing to make room for the new college fieldhouse. However, it was still possible to carry on the investigation with the remaining equipment. A smaller shelter was set up for the pump, and covers were placed over the electrical controls. A tractor umbrella was used to eliminate the glare of the sunlight upon the potentiometer.

TEST PROCEDURE

Two tests were run on each spacing of the horizontal pipe. Each of the tests was run for a length of time until equilibrium or as nearly as possible equilibrium conditions were established. Test number one was continued after the establishment of apparently equilibrium conditions. Equilibrium conditions were impossible to establish during test number four. The time of each test during the year was such that the soil conditions were nearly the same as they might be with an operating heat pump installation.

At the start of each test the temperature difference between the entrance and exit of the section was determined once every two to four hours. In about three days the number of readings was reduced to two complete sets of temperature readings each day. Toward the last of each test, the number of readings was reduced to one per day. Sufficient time was required for making these readings, so that it was also possible to obtain the information on the rate of fluid circulation. The water meter was read at the beginning and at the end of the time required for making the temperature readings. The specific gravity of the calcium chloride brine solution was determined in order to find out the specific heat of the brine solution. As nearly as possible, a constant pumping rate through each test section was maintained.

The tests with different fluids and spacings are listed with the occurrence and duration of each test.

Test	Spacing	Fluid	Started	Ended	Hours
1	2 ft	Cold water	Dec. 13, 1948	Jan. 13, 1949	750
2	10 ft	Brine	Mar. 26, 1949	Apr. 13, 1949	422
3	2 ft	Brine	Apr. 13, 1949	Apr. 27, 1949	339
4	10 ft	Hot water	Aug. 23, 1949	Sept. 20, 1949	648

Test One

Test number one consisted of circulating cold water in the 100 feet of test section of the 1 inch i. d. (1.125 inch o.d.) copper pipe buried 6 feet deep in a horizontal U shape with the legs of the U 2 feet apart. This test was started on December 13, 1948 and was run continuously until January 13, 1949. This covered a time period of 750 hours. The cold water entered the test section at a temperature near 35° F. Table 3 shows the water flow rate and temperature data for this test. The heat transfer rate as shown in Fig. 1 was high for the first few hours of the test and gradually decreased until after 150 hours a fairly stable condition existed in the soil. The temperature difference between the pipe and the thermally undistrubed soil at 6 feet deep was relatively small, being about 8° F. or 9° F. Over the entire period of continuous operation under those conditions, an average of 21.5 Btu/(hr) (ft) was removed from the soil. Since there was no further decrease in the heat flow rate after the first 150 hours of operation, an average heat flow rate based on 400 hours of continuous operation should be safe to use for design with a cycling heat pump. If the temperature difference between the pipe and the thermally undisturbed soil had been larger, undoubtedly a larger quantity of heat per unit time

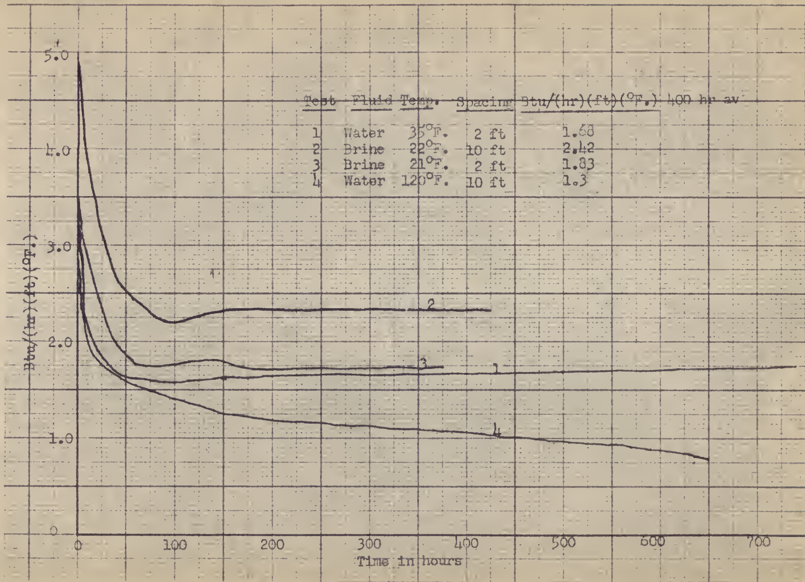


Fig. 1. Rate of heat flow for the various tests.

Table 3. Water flow data for test 1. Two-foot spacing.

Date	Time	Accumulated time, hours	Gal/hr corrected	Temperature difference	Btu/hr	Btu / (hr) (ft) (°F.)
Dec., 1948						
13	10:00					
13	10:30	0.5	152.8	4.55	5870	
13	11:08	1.13	151.5	3.5	4440	
13	1:40	3.67	152.8	3.46	4620	
13	3:10	5.17	152.8	2.72	3460	2.88
13	5:40	7.67	152.8	2.72	3460	2.47
13	8:08	10.13	152.8	2.04	2600	1.86
13	12:17	14.17	152.8		3460	2.47
14	9:10	23.17	152.8	2.04	2600	1.86
14	11:55	25.08	142.0	1.00	1190	0.92
14	2:30	28.50	143.0	1.8	2165	1.55
14	5:00	31.00	137.6	2.27	2565	1.83
14	8:30	34.50	135.5	2.27	2560	1.83
15	9:11	47.28	127.0	1.5	1586	
15	3:20	53.33	124.7	1.82	1890	
15	12:10	50.17	126.0	1.82	1910	1.27
15	4:56	54.93	124.7	1.82	1890	
15	7:30	57.50	120.5			
15	8:30 pm	58.50	166.0	1.82	2510	
16	8:45 am	70.75	132.0			
16	9:45	71.75	124.7	2.27	2355	1.81
16	6:00	80.00	146.0	1.82	2215	
16	8:45	82.75	147.2	1.82	2230	
17	9:30	95.50	150.8	1.36	1710	1.42
17	9:20	119.33	148.4	2.27	2810	2.32
18	2:30	100.50	124.7	1.82	1890	1.58
20	10:05	168.08	129.0	1.82	1960	1.63
20	3:00	173.00	137.6	1.82	2090	1.74
21	3:45	197.75	118.0	1.82	1795	1.62
24	11:00 am	265.00	112.0	1.0	837	0.69

Table 3. (concl.).

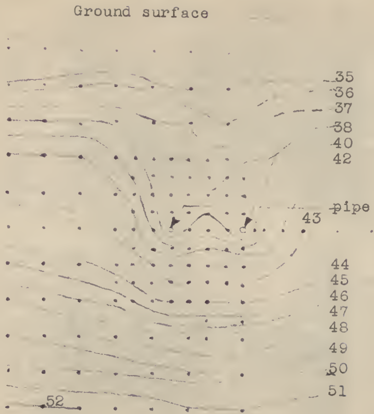
Date	Time	Accumulated time, hours	Gal/hr corrected	Temperature difference	Btu/hr	Btu / (hr) (ft) (°F.)
Jan., 1949						
11	3:00 pm	701.0	132.0	1.5	1650	1.83
12	2:40 pm	724.67	129.0	1.5	1615	1.79
13	9:00 pm	753.0	133.3	1.0	1110	1.31

could have been removed. The average rate of heat flow for 400 hours was 1.68 Btu/ (hr) (ft) (^{o}F.). The temperature difference was that difference between the pipe and thermally undisturbed soil at 6 feet deep. The isotherm curves in Fig. 2 for this test show that the distance from a pipe to the thermally undisturbed soil at 6 feet was 3.5 feet. This indicates that in this region a spacing, between horizontal 1 inch copper pipes, of 7 feet is wide enough that adjacent pipes will not influence each other, if water is circulated in the pipes at a depth of 6 feet.

The lowest temperature recorded for the thermally undisturbed soil at 6 feet deep was 43^{o} F. Since the circulated water temperature had to be above the freezing point of water, the total temperature differential was necessarily of small magnitude and that limited the quantity of heat available per unit length of the pipe. For that reason it was thought that the circulation of water in a closed ground coil circuit did not represent a very satisfactory method of extracting heat from the earth. Therefore, it seemed that if the earth is to serve as a source of heat for the heat pump, lower temperatures in the ground coil must be obtained. Calcium chloride was added to the water to make a brine solution which had a freezing point at less than 10^{o} F.

Test Two

Test number two consisted of circulating a cold calcium



Weather, clear
Time since start, 725 hours
Rate of water flow, 129 gph
Water temperature: tank, 33° F.
inlet, $34\frac{1}{2}^{\circ}$ F.
outlet, 36° F.
Temperature drop in coil, $1\frac{1}{2}^{\circ}$ F.
Heat transferred, 1500 Btu/hr
Dashed lines are isotherm extrapolations
Dots represent thermocouple locations
Scale, 3/8" = 1' 0".

Fig. 2. Typical soil isotherm curve for test 1, cold water in 2 foot spaced pipes.

chloride brine solution in the test section with the two pipes (one U shaped pipe) spaced 10 feet apart. The test was started on March 26, 1949 and ended on April 13, 1949. The continuous running time was 422 hours. Temperature conditions were observed after the cold brine circulation was stopped. Figure 3 shows the rate of heat absorption versus time for this test. The brine flow data is given in Table 4. A typical isotherm curve for this test is shown in Fig. 4. Over the entire period of the test, an average of 56.8 Btu/(hr) (ft) was obtained. The average heat flow rate for 400 hours was 2.42 $\frac{\text{Btu}}{(\text{hr})(\text{ft})(^{\circ}\text{F.})}$. The temperature difference was from pipe to undisturbed sandy soil at 6 feet deep.

The pipe temperature at a point along the pipe 20 feet from the entrance averaged 21° F. The refrigeration controls had a 1° F. differential temperature and were set at 15° F. Some heat was gained by the brine in its path from the supply tank to the test section, but as the pipe temperature was measured at the entrance and exit of the test section, the gain was of no significance.

During the 422 hours of continuous operation a condition near equilibrium was established in the soil surrounding the ground coil. Approximately 250 hours were required to establish that condition. Figure 5 shows the temperature versus time curves for various distances from the pipe for this test. The steady state or equilibrium condition could not be attained until all freezing of soil moisture had taken place. There seemed to be no freezing after the initial 150 to 200 hour period

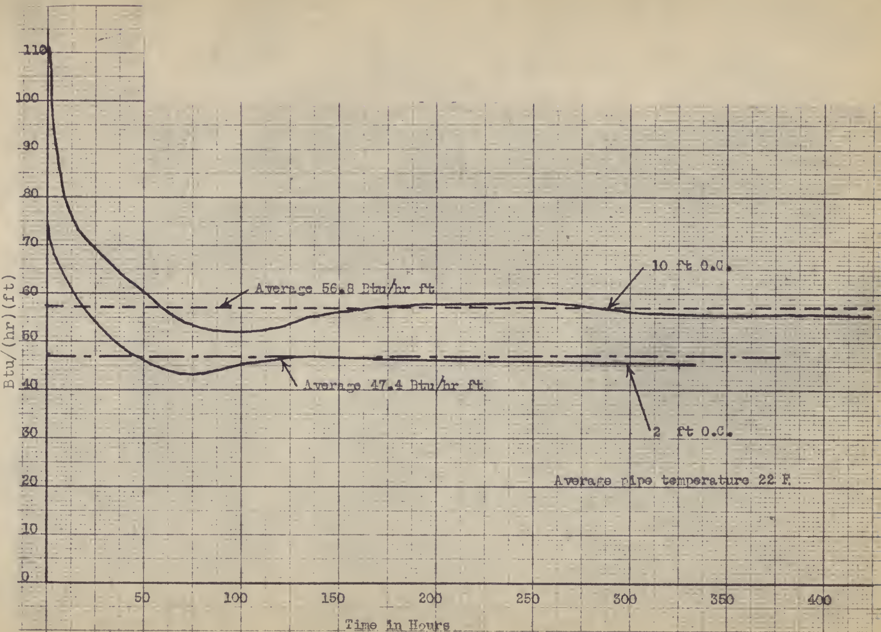


Fig. 3. Heat flow rate for the brine tests, 2 and 3.

Table 4. Brine flow data for test 2. Ten-foot spacing.

Date	Time	Accumulated time, hours	Meter reading	Gal/hr corrected	Temperature difference	Btu/hr	Btu (hr) (ft) (°F.)
March							
26	2345		90850		8		
27	0030		91000	206.2	7.5	11120	
27	0535	5.8	91760	154.2	7.2	7970	4.3
27	0618	6.5	91865	151.0	7	7600	4.1
27	1003	10.3	92404	148.0	7	7450	
27	1053	11.1	92527	152.0	7	7650	
27	1412	14.3	93001	155.0	7	7800	3.52
27	1447	14.9	93084	146.8	7	7380	
27	1658	17.1	93400	149.0	7	7500	
27	1743	17.8	93515	158.0	7	7950	
27	2145	21.8	94073	143.0	7	7200	
27	2225	22.5	94167	145.5	7	7320	3.4
28	0545	29.8	95202	145.5	6.5	6800	
28	0637	30.7	95300	147.0	6	6330	2.7
28	0914	33.3	96598	147.0	6		
28	1019	34.4	95848	143.0		7200	
28	1622	40.5	96660	138.4	2.5	2486	1.3
28	1712	41.3	96761	125.0	2.5	2245	
28	1900	43.0	97014	145.0	5	5210	2.48
28	2000	44.1	97146	136.0	6.5	6350	2.89
29	0717	55.4	98620	135.0	6	5810	2.64
29	0806	56.2	98726	134.0	6	5775	
29	1850	66.8	100107	133.0	6	5730	2.61
29	1952	68.0	100232	124.0	6	5390	
30	0815	80.4	101726	124.5	6	5360	
30	0835	80.7	101760	105.0	6	4520	
30	0954	82.0	101910	117.4	6	5060	2.25
30	1119	83.3	102072	118.0	6	5080	2.26
30	1931	91.6	102994	116.0	6	5000	2.22
30	2041	92.8	103119	110.7	6	4770	
31	0819	104.4	104264	101.5	6.5	4740	2.16

Table 4. (cont.).

Date	Time	Accumulated time, hours	Meter reading	Gal/hr corrected	Temperature difference	Btu/hr	Btu (hr) (ft) (°F.)
31	0910	105.6	104353	91.8	7	4620	
31	1843	114.8	105463	122.0	5.5	4820	2.1
31	1959	119.0	105652	156.0	5.5	6160	2.4
April							
01	0915	128.3	107497	155.0	5.5	6120	2.50
01	0847	128.9	107576	153.0	5.5	6050	2.47
01	1851	138.9	109067	153.0	5	5510	2.30
01	2004	140.1	109249	154.0	5	5540	2.31
02	0832	152.6	111104	153.4	5.5	6030	2.51
02	0915	153.3	111210	152.7	5.5	6060	2.52
02	1746	161.8	112428	147.5	5.5	5830	2.43
02	1904	163.1	112612	146.0	5.5	5760	2.40
03	0855	177.0	114485	139.7	5.5	5520	2.30
03	0935	177.6	114576	141.0	5.5	5570	2.32
03	2000	188.1	115970	138.0	5.5	5450	2.22
03	2108	189.2	116121	137.5	5.5	5430	2.22
04	0829	200.6	117640	138.0	5	4450	1.85
04	0915	201.3	117743	138.5	4.5	4470	1.86
04	0915	201.3	117743	138.5	4.5	4470	1.86
04	1926.5	211.5	119100	135.4	6	5825	2.38
04	2009.5	212.2	119195	136.6	6	5880	2.40
05	0845	224.8	120874	137.5	5.5	5430	2.26
05	0925	225.5	120964	139.2	5.5	5500	2.29
05	1900	234.9	122234	139.2	6	6000	2.45
05	2018	236.2	122418	146.0	6	6290	2.56
06	1048	250.7	124336	136.5	6	5880	2.26
06	1138	251.5	124457	150.0	6	6455	2.48
06	1848	258.7	125480	137.0	6	5900	2.46
06	1958	259.9	125563	137.0	6	5900	2.46
07	0811	272.1	127175	136.0	5.5	5370	2.24
07	0901	272.9	127285	136.0	5.5	5370	2.24

Table 4. (concl.).

Date	Time	Accumulated time, hours	Meter reading	Gal/hr corrected	Temperature difference	Btu/hr	Btu (hr) (ft) (°F.)
07	1839	282.6	128578	137.2	6	5920	2.46
07	1946	283.7	128717	138.5	6	5920	2.46
08	0724	298.6	130256	136.4	5.5	5390	2.24
08	0827	299.7	130393	134.6	5.5	5400	2.25
08	1827	307.4	131714	136.2	6	5860	2.3
08	1931	307.4	131855	136.4	6	5870	2.3
09	1748	319.7	133477	136.4	6	5870	2.39
09	0854	320.8	133622	136.0	5.5	5370	2.19
10	1100	346.9	137223	142.5	5.5	5630	2.17
10	1145	347.6	137336	155.3	5.5	6140	2.41
11	0910	369.1	140423	148.6	4.5	4870	1.84
11	0940	369.6	140545	252.0	4.5		
13	1311	421.1	148073	150.8		6490	2.24
13	1445	422.6	148202	150.8		6490	2.24

Weather, overcast
Time since start, 347 hours
Rate of brine flow, 142.5 gph
Brine temperature: tank, 15° F.
inlet, $18\frac{1}{2}$° F.
outlet, 24° F.
Temperature drop in coil, $5\frac{1}{2}$° F.
Heat transferred, 5630 Btu/hr
Dashed lines are isotherm extrapolations
Dots represent thermocouple locations
Scale, 3/8" = 1' 0".

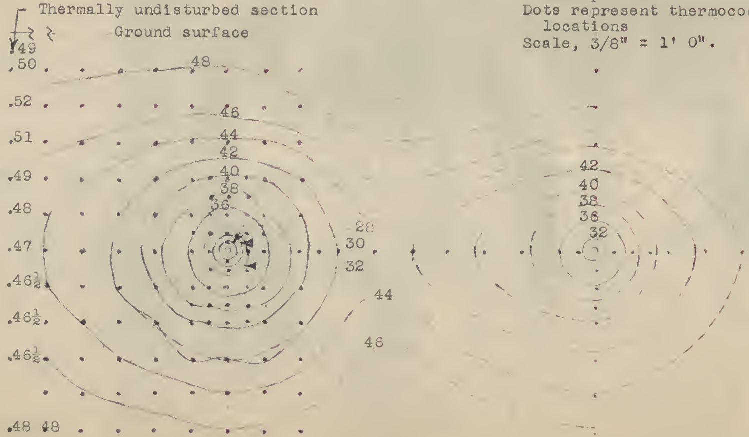


Fig. 4. Typical soil isotherm curve for test 2, brine in 10 foot spaced pipes.

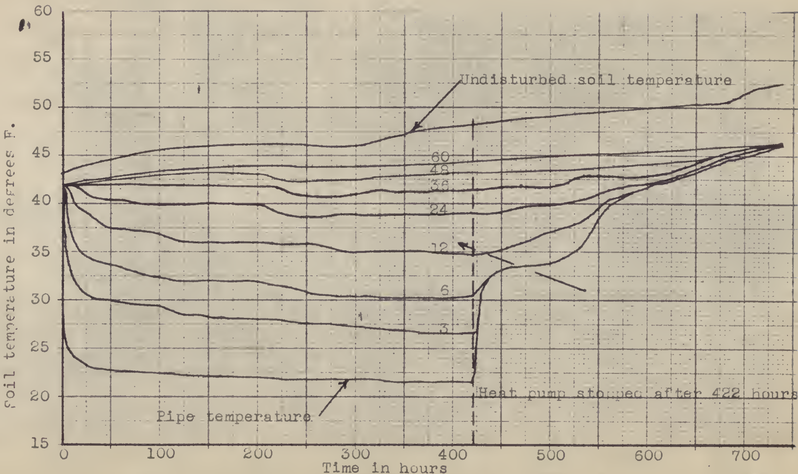


Fig. 5. Soil temperature for horizontal distances from pipe during test 2, brine in 10 foot spaced pipes.

of operation. Coogan (4, p. 7) found that 200 hours were required before the heat of fusion became unimportant under operating conditions similar to test 2. Since the frozen soil extended in each direction for only slightly more than 6 inches, the heat available from the heat of fusion was of small importance because of the 200 hours required to obtain all of it. However, ice is a better conductor of heat than either water or dry soil, so the freezing of the soil was of some importance from the conduction standpoint. The heat of fusion would be more important to a cycling heat pump than to a continuously operated heat pump, if both were operated slightly below the freezing point of water. The cycling heat pump would have the benefit of freezing and thawing. The thawing of the frozen soil in this test was quite rapid after thermal recovery was started.

Although the frozen zone thawed rapidly, complete recovery of the thermally disturbed zone was rather slow. Even after 300 hours of recovery, the affected soil was still slightly cooler than the undisturbed soil.

A comparison of the soil temperatures at a point midway between the pipes and 6 feet deep with the points at the same depth and distance from the pipe, but located outside rather than between pipes, is shown in Fig. 6. Although the temperature difference was only 2° F., it shows that they did not behave as individual pipes. A point located 5 feet away from the pipe in both the horizontal and vertical (deeper) directions was not affected by the pipe. This would indicate that a pipe

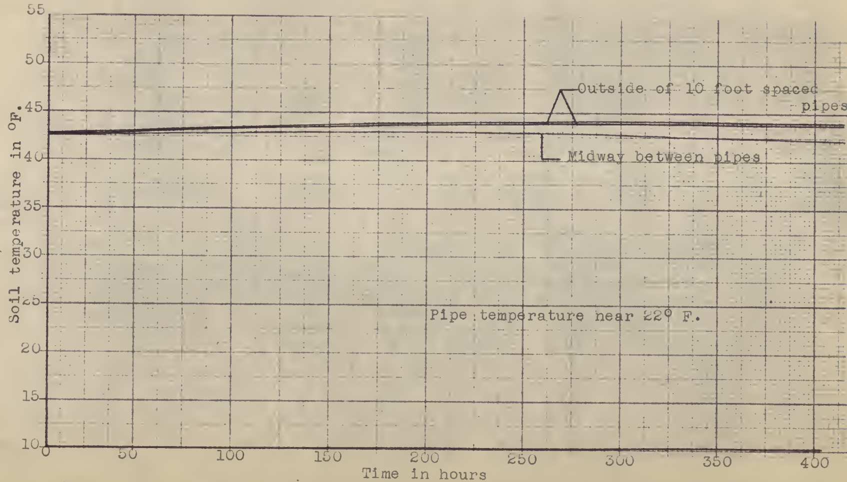


Fig. 6. Soil temperatures at 6 feet deep and 5 feet from pipe during test 2.

spacing of 14 feet would nearly eliminate the effect of one pipe on the other under the temperature and depth conditions of this test.

After the steady state condition was obtained, there was no further decrease in the heat flow rate. Probably this was because there was moisture migration toward the cold pipe. Wet soil has a higher thermal conductivity than dry soil. The soil near the pipe was much wetter than the more distant soil at the same depth. The following soil moisture data were taken three days after the heat pump was stopped.

Table 5. Soil moisture content at 6 feet deep after test 2.

Distance from pipe: inches	Moisture content: percent
6	14.35
12	9.33
18	9.08
24	7.89
36	7.87

Figure 5 shows that nearly one-third of the total temperature drop in the soil occurred in the first 6 inches around the pipe and that nearly one-half occurred in the first foot. Apparently a pipe spacing of two feet would be adequate to obtain a large percentage of the heat which could be obtained by much wider spacings. A pipe spacing of only one foot would apparently be satisfactory. A large proportion of the installation cost for a ground coil is for digging the trench. If a hairpin looped ground coil is placed in a single trench, a considerable saving in the length of trench required will be achieved. It would be necessary to dig in under the walls of the trench in order to get as wide a spacing as possible.

Test Three

Test three consisted of the circulation of cold brine in the 1 inch copper pipe buried 6 feet deep and spaced 2 feet apart. It was run with the temperature controls set the same as in test two and was started on the same day that test two was stopped, April 13, 1949. The failure of the circulation pump caused the stopping of this test on April 27, 1949, so that the total running time was 339 hours. During this time it was necessary to make almost daily adjustments of the by-pass valve which controlled the rate of flow of the brine. Table 6 shows the brine flow data for test 3. The average heat flow rate for the test was 47.4 $\frac{\text{Btu}}{\text{(hr) (ft)}}$. By extrapolating the heat flow rate versus time curve from 339 to 400 hours, the average rate of heat flow was found to be 1.83 $\frac{\text{Btu}}{\text{(hr) (ft) (}^{\circ}\text{F.)}}$ for 400 hours. The temperature difference was between the pipe and the thermally undisturbed soil. The soil was frozen for almost 6 inches from the pipe. Figure 3 shows a comparison of the heat flow rate of test two and test three. A typical isotherm curve for this test is shown in Fig. 7.

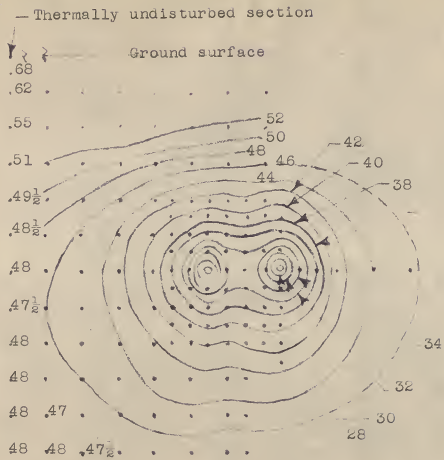
The 400-hour average heat flow rate of 1.83 $\frac{\text{Btu}}{\text{(hr) (ft) (}^{\circ}\text{F.)}}$ for test three was about 6 percent higher than the similar rate

Table 6. Brine flow data for test 3. Two-foot spacing.

Date	Time	Accumulated time, hours	Meter reading	Gal/hr corrected	Temperature difference	Btu/hr	Btu / (hr)(ft) (°F.)
April							
13	1455		182030				
13	1854	4.0	182494	126.4	7.5	6780	2.88
13	2043	5.8	182704	125.1	7.5	6740	
14	0050	9.9	183160	119.0	7.5	6410	2.98
14	0250	11.9	183373	114.4	7.5	6170	2.87
14	0714	16.3	183786	99.5	8.0	5710	2.66
14	0851	17.9	183921	89.8	8.0	5160	2.40
14	1340	22.7	184550	140.2	7.0		
14	1502	24.1	184730	141.5	7.0		
14	1511	pump stopped					
14	1926	28.4	185333	152.6	6.0	6570	2.68
14	2155	30.8	185682	151.0	6.0	6510	
15	0805	41.0	187080	148.0	4.5	4780	1.77
15	1002	42.9	187347	147.2	4.5	4760	
		pump off 38 min.					
15	1957	52.2	188629	148.5	4.5	4800	2.04
15	2157	52.4	188855	121.3	5.0	4355	
		current off 41 min.					
16	0905	64.7	189975	115.1	5.5	4550	1.75
16	1005	25.4	190074	106.4	6.0	4570	
16	2100	76.6	191045.5	95.7	6.5	4460	1.86
16	2245	78.4	191185.0	85.7	6.5	4000	1.67
17	0900	88.6					
17	1001	89.6		133.6	5.0	4800	1.78
17	2115	100.9	193964	132.8	5.0	4770	1.77
17	2225	102.0	194101	126.2	5.5	4980	1.99
18	0825	112.0	195258	124.2	5.5	4910	2.00
18	1933	123.2	196721	145.0	4.5	4690	1.80
18	0948	113.4	195410	118.0	5.5	4660	1.90
18	2157	125.5	197050	147.0	4.5	4750	1.83
19	1922	146.9	199792	137.6	5.5	5440	1.98
19	2031	148.1	199932	131.0	6.0	5650	2.05

Table 6. (concl.).

Date	Time	Accumulated time, hours	Meter reading	Gal/hr corrected	Temperature difference	Btu/hr	Btu / (hr) (ft) (°F.)
19	2031	148.1	199932	131.0	6.0	5650	2.05
20	1850	170.4	202268	112.6	6.0	4850	1.98
20	2050	172.0	202372	55.8	5.0	2210	
21	2021	pump stopped 70 min.					
21	2021	194.8	203918	74.3	5.5	2930	1.07
22	0821	206.8	205763	165.1	4.0	4750	1.70
22	1833	217.0	207074	137.2	4.5	4440	1.58
22	2002	218.5	207271	143.0	4.5	4620	1.65
23	1502	237.5	209958	152.0	4.5	4910	1.76
23	1629	238.9	210190	174.0	5.0	6250	2.19
24	1842	265.1	214870	192.0	4.0	5520	1.94
24	1942	266.1	215049	192.5	4.0	5540	1.95
25	1842	289.1	217924	134.2	5.0	4820	1.72
25	2151	291.3	218167	121.3	5.0	4360	1.56
26	1851	312.3	220454	117.0	5.5	4630	1.85
26	2008	330.3	220538	70.4	6.0	3030	
27	1630	333.9					



Weather, clear
Time since start, 289 hours
Rate of brine flow, 134.2 gph
Brine temperature: tank, 13° F.
inlet, 18° F.
outlet, 23° F.
Temperature drop in coil, 5° F.
Heat transferred, 4820 Btu/hr
Dashed lines are isotherm extrapolations
Dots represent thermocouple locations
Scale, 3/8" = 1' 0".

Fig. 7. Typical soil isotherm curve for test 3, brine in 2 foot spaced pipes.

of 1.68 $\frac{\text{Btu}}{(\text{hr})\ (\text{ft})\ (^{\circ}\text{F.})}$ for test one. This was probably due to the heat of fusion when ice was formed and to the increased conductivity of frozen soil over wet soil.

A comparison of test two and test three showed that the 2-foot spaced ground coil extracted only 75.8 percent as many $\frac{\text{Btu}}{(\text{hr})\ (\text{ft})\ (^{\circ}\text{F.})}$ as did the 10-foot spaced ground coil when operated under the same conditions. Therefore, with a 2-foot spacing 25 percent more pipe would be necessary to do the same job as with a 10-foot spacing. However, since the 2-foot spaced pipe could be placed in a single trench, there would be a saving of 37.5 percent on the length of trench necessary. If space is not a limiting factor, the relative cost of pipe and trench digging will determine the pipe spacing used for a heat pump using the earth as a heat source.

Coogan (3, p. 14) defined F as the ratio of the heat absorbed per linear foot of pipe, when there were two pipes in the ground, to the heat absorbed per linear foot with only one pipe in the ground. F for the 10-foot spacing in this test was 0.925. F for the 2-foot spacing was 0.754. A comparison of these two ratios shows that the 2-foot spacing would absorb 81.5 percent as much heat as the 10-foot spacing. Considering the many assumptions necessary, this value of the ratio of $F_2/F_{10} = 0.815$, compares favorably with the ratio of Q_2/Q_{10} which is 0.758. However, since the theoretical ratio, F_2/F_{10} is greater than the actual ratio, Q_2/Q_{10}, the theoretical ratio was not on the safe side for design purposes.

Although these tests were performed using a circulated liquid in the ground coil to bring heat to the heat pump, a system which works just as well for heat absorption is direct expansion. This system uses the refrigerant in the ground coil, and the ground coil becomes the evaporator of the refrigeration system. One heat transfer is eliminated, and the heat pump should then operate under better conditions as far as evaporator temperature is concerned.

Test Four

In summer when the heat pump would be used for cooling of living space, the problem becomes one of dissipation of thermal energy. The energy which must be dissipated is the heat removed from the space plus the work of compression. The problem is complicated by the tendency of the water vapor pressure gradient to be in the same direction as the temperature gradient.

Test four simulated heat pump summer conditions as far as the ground coil was concerned by having hot water circulated in it. A temperature of 120° F. in the hot water tank was selected as a temperature which might represent the condenser temperature of the refrigeration unit. The water was heated by electric immersion heaters.

On July 12, 1949, a hot water test on the 10-foot spacing was started but had to be discontinued on July 31 because of irregularity of the power supply.

Apparent recovery was allowed and a new test was started on August 23, 1949. The test ran for 648 hours and was stopped on September 20. Figure 8 shows that the average rate of heat dissipation for 648 hours was 54 $\frac{\text{Btu}}{\text{(hr) (ft)}}$. Table 7 shows the water flow data for this test. The heat dissipation rate decreased steadily throughout the time of continuous operation. Steady state conditions not only were not achieved but they were not even approached. The average rate of heat dissipation for 400 hours was 1.3 $\frac{\text{Btu}}{\text{(hr) (ft) (}^{\circ}\text{F.)}}$ with the temperature difference taken between the pipe and thermally undisturbed soil, which was more than 5 feet away horizontally as shown in Fig. 11. For the same pipe spacing and time, the average $\frac{\text{Btu}}{\text{(hr) (ft) (}^{\circ}\text{F.)}}$ for dissipation was considerably less than the same rate for heat absorption. The actual dissipation rate at the end of the test was 0.83 $\frac{\text{Btu}}{\text{(hr) (ft) (}^{\circ}\text{F.)}}$. The use of the 400-hour average is not a safe criterion for heat dissipation because the dissipation rate continually decreased. A typical isotherm curve for this test is shown in Fig. 9.

At the conclusion of test four an attempt was made to obtain soil samples for moisture determinations. In the immediate vicinity of the pipe the sandy soil was too dry and powdery to stick to the soil auger. The drying of the soil apparently was the cause of the decrease in the heat dissipation rate. Figure 10 shows that the temperature gradient in the soil changed very little after the first 200 hours of continuous operation. However, the heat dissipation rate continued to decrease after that time, indicating that the drying out of the soil was the most

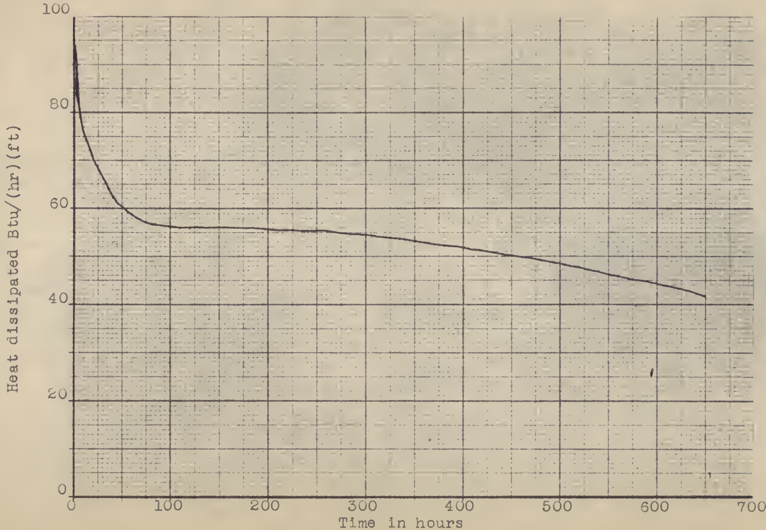


Fig. 8. Heat dissipated during test 4.

Table 7. Water flow data test 4. Ten-foot spacing.

Date	Time	Accumulated time, hours	Meter reading	Gal/hr uncorrected	Temperature difference	Btu/hr	Btu / (ft) (hr) (°F)
Aug.							
24	0807		215200				
24	0857	0.83		145.0	7.5	9350	2.99
24	1604	7.90	216374	148.0	6.0	7640	2.01
24	1646	8.60	216478	148.6	6.0	7670	2.02
24	2300	14.9	217403	148.4	5.5	7020	1.81
24	2348	15.7	217522	148.8	5.5	7030	1.81
25	0812	24.1	218750	146.0	5.5	6910	1.76
25	0853	24.7	218850	146.2	5.5	6920	1.76
25	1613	28.2	219920	148.0	6.0	7640	1.98
25	1658	32.7	220031	148.0	6.0	7640	1.98
25	2308	38.9	220921	144.4	5.0	6210	
26	0002	39.8	221051	144.5	5.0	6210	
26	0820	48.1	222247	144.2	4.5	5590	
26	0906	48.9	222357	143.5	4.5	5550	
26	1612	55.9	223365	142.0	4.0	4890	
26	1701	56.8	223491	154.3	4.0	5310	
27	0850	72.6	225716	140.8	5.5	6650	1.52
27	0938	73.4	225828	140.0	5.5	6630	
27	2022	84.1	227336	140.6	5.5	6650	
27	2100	84.8	227426	142.0	5.5	6710	
28	0923	97.2	229170	140.8	6.0	7260	
28	1007	97.9	229273	140.5	6.0	7250	
28	2020	108.1	230694	139.0	4.0	4780	
28	2058	108.7	230781	137.1	4.0	4710	
29	0834	120.3	232352	135.3	4.0	4750	
29	0915	121.0	232447	139.0	4.0	4780	
30	0812	143.8	235543	135.6	5.0	5820	
30			235665		5.0		
31	0829	168.1	238741	132.0	5.0	5680	1.23
31	0842	168.3	238770	133.8	5.0	5750	1.25
Sept.							
1	0828	192.1	241892	131.5	4.0	4520	1.00
1	0908	192.8	241980	132.0	4.0	4540	1.01

Table 7. (concl.).

Date	Time	Accumulated time, hours	Meter reading	Gal/hr uncorrected	Temperature difference	Btu/hr	Btu / (ft)(hr) (°F.)
2	0839	216.3	245055	130.8	5.0	5620	1.25
2	0917	216.9	245138	131.0	5.0	5640	1.25
3	0854	240.5	248216	130.3	5.0	5600	1.20
3	1008	241.8	248376	129.8		5580	1.24
6	0916	312.9	257638	130.4	4.5	5050	
6	1026	314.1	257791	131.2	4.5	5080	
7	0912	336.8	260768	130.8	5.0	5620	1.14
7	1004	337.4	260880	129.2	5.0	5560	1.12
8	0844	360.4	263827	130.0	5.0	5590	1.16
8	0932	361.2	263931	121.0	5.0	5210	1.09
9	0821	385.6	265551		6.0		
10	0857	408.7	268473	126.5	5.5	5980	1.29
10	0939	409.4	268568	135.8	5.5	6410	1.39
12	0822	456.1	274918	135.8	4.0	4670	0.99
12	0842	456.4	274963	135.0	4.0	4650	0.99
13	0806	479.8	278254	140.6	3.5	4230	0.825
13	0849	480.5	278356	139.5	3.5	4200	0.82
15	0813	527.9	285085	141.6	4.0	4870	0.98
15	0828	648.2	302123	141.6	4.0	4870	0.98
20	0828	648.2	302123	141.8	3.5	4260	0.81
20	0908	468.8	302218	142.4	3.5	4280	0.82
22	0928	697.1	315760				

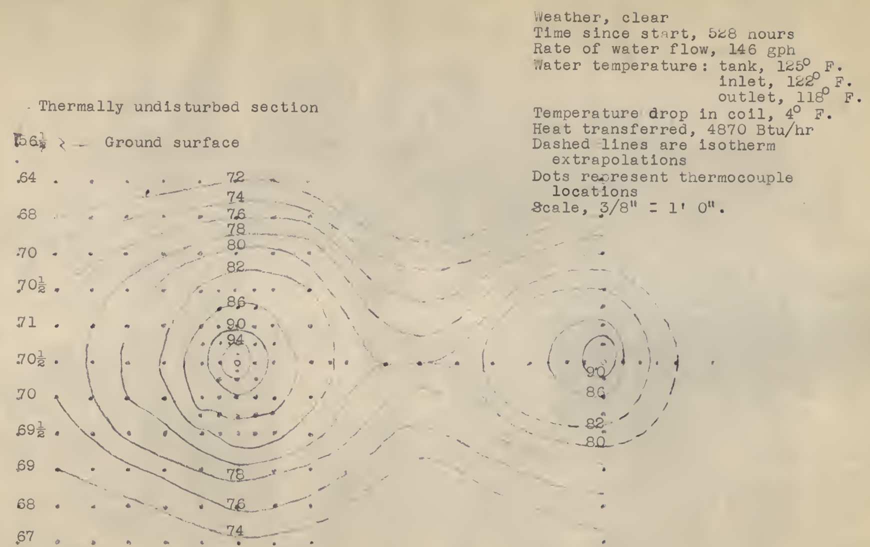


Fig. 9. Typical soil isotherm curve for test 4, hot water in 10 foot spaced pipes

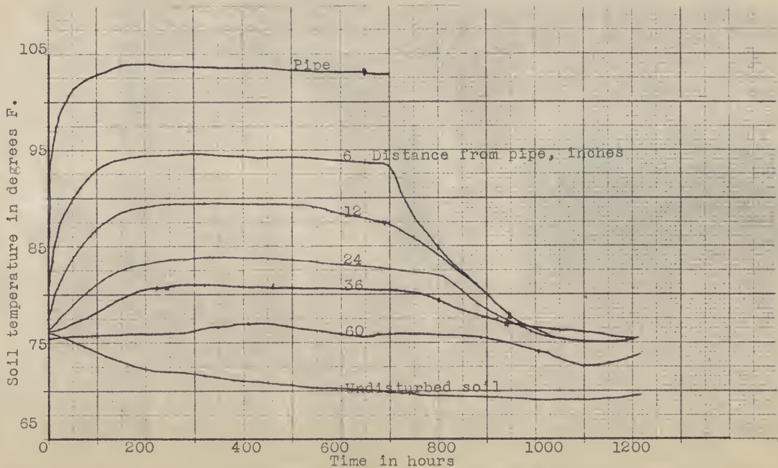


Fig. 10. Soil temperatures for horizontal distances from pipe during test 4.

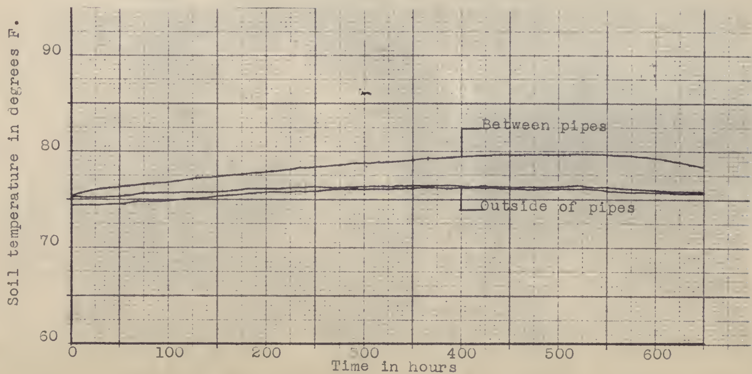


Fig. 11. Soil temperatures at 6 feet deep and 5 feet from pipe during test 4.

influential factor in the continued decrease of the heat dissipation rate.

If the heat pump which uses a ground coil is to be used for summer cooling, some method for wetting the soil around the pipe must be planned unless the cooling load is a very small portion of the total annual load.

SUMMARY

1. The earth is the only economical source of heat for a heat pump for all regions where a domestic heat pump might be desired.

2. The rate of heat absorption from the earth was high when the heat pump was first started but was gradually reduced until a relatively steady state existed. After approaching the steady state, no further reduction in the heat absorption rate occurred regardless of the length of time of operation.

3. The rate of heat dissipation into the earth was high at first and decreased with time of continuous operation. Steady state conditions were not achieved even with a 30-day period of continuous operation.

4. When heat was being absorbed from the earth, the soil moisture moved toward the cold pipe and the soil near the pipe was considerably wetter than the soil a few feet away from the pipe.

5. When heat was being dissipated into the earth, the soil moisture moved away from the hot pipe and the soil near the pipe became very dry.

6. The heat absorption rate was higher than the heat dissipation rate.

7. For successful dissipation of heat to the earth from a horizontal ground coil for any extended period of time, a supplemental supply of water to the soil around the pipe apparently would be necessary.

8. A horizontal pipe spacing of 10 feet would apparently be a greater spacing than is necessary when cold water is circulated in the ground coil during winter heat pump operation.

9. If a direct expansion ground coil is used at a depth of 6 feet with an evaporator temperature of 20° F., apparently a pipe spacing of 12 to 14 feet would be sufficient to get the maximum from each unit of pipe.

10. When a hairpin ground coil with the pipes spaced 2 feet apart at a depth of 6 feet in a single wide trench is used, about 25 percent (based on 10-foot spacing) more pipe would be required than if a 10-foot spacing under the same conditions were to be used. However, under those same conditions 37 percent less length of trench would be required. Additional space between the pipes of the hairpin coil can be obtained by digging under the trench wall at the bottom.

11. The thermal energy available due to the heat of fusion of the soil moisture was of minor importance to a continuously operated heat pump because a period of about 200 hours was required to freeze the soil for about 6 inches around the pipe with a pipe temperature of 20° F.

12. More research on this subject is needed especially in regard to design values for heat flow in soil under the conditions of a cycling heat pump.

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LITERATURE CITED

(1) Ambrose, E. R.
Progress report on the heat pump. Heating and Ventilating. Dec., 1946.

(2) Budenholzer, R. A.
Says heat pump requires 1.5¢ rate to compete on cost. Heating, Piping and Air Conditioning. Apr., 1947.

(3) Coogan, C. H., Jr.
Some aspects of heat absorption by a ground coil. Conn. Engg. Expt. Sta. Bul. 3. June, 1948.

(4) Coogan, C. H., Jr.
Summary of heat absorption rates for an experimental ground coil system. Conn. Engg. Expt. Sta. Bul. 4. March, 1949.

(5) Dana, H. J.
Typical design problems on the residential heat pump. Washington Engg. Expt. Sta. Bul. 73, July, 1949.

(6) Hogentogler, C. A.
Engineering properties of soil. New York: McGraw-Hill, 1937. p. 129.

(7) Kemler, E. N.
Progress report on the heat pump--economic possibilities. Heating and Ventilating. Dec., 1946.

(8) Kemler, E. N.
Heat sources for the heat pump. Heating, Piping and Air Conditioning. Jan., 1947.

(9) Patten, H. E.
Heat transference in soils. U. S. Dept. Agr. Bur. of Soils Bul. 59, 1909.

(10) Penrod, E. B.
Development of the heat pump. Kentucky Engg. Expt. Sta. Bul. 4. June, 1947.

(11) Smith, W. O., and H. G. Byers.
The thermal conductivity of dry soils of certain of the great soil groups. Soil Sci. Soc. Proc., 1938, p. 18.

(12) Sporn, Philip, E. R. Ambrose, and Theodore Baumeister. Heat pumps. New York: John Wiley and Sons, 1947. p. 39.

(13) Stringfield, V. T. Effect of air conditioning demand on well water availability. Heating and Ventilating. June, 1945.