A STUDY OF CYCLIC AND CONTINUOUS HEAT PUMP OPERATION AS IT AFFECTS HEAT TRANSFER RATES FOR TWO SOIL TYPES

by

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INTRODUCTION

The period since World War II has been a period of extensive activity in heat pump research. Interest has been shown by the electric utility industry, manufacturers of air conditioning equipment, and by the public in the possibilities of a practical method to heat the home electrically. Some of the outstanding features of electric heating are cleanliness, absence of combustion, ease of operation, and general comfort and convenience. It is becoming more apparent that the only practical method for heating electrically is through some device which will multiply the effectiveness of electricity for heating purposes by several times its direct energy conversion value. The heat pump is such a device. Research results have shown that a heat pump will multiply between two and three times the effectiveness of electricity for heating purposes.

Many successful installations, both commercial and industrial, have been made in Europe where fuel was scarce and expensive, and where hydroelectric power was available at very low rates. A fuel shortage has not yet become a serious factor in this country, but the other advantages of the heat pump have prompted many installations in the United States.

Recent figures released by the Edison Electric Institute list about 600 heat pump installations in 39 of the 48 states. It is possible that there are as many more units that have not been reported. All but 50 of these have been installed since 1945. Heat sources for these units are divided about as follows: water 53 per cent, air 40 per cent and ground coils 7 per cent.

The basic problem in the development of the heat pump is to find a comparatively cheap and reliable heat source and heat sink. At the present time there are at least 50 separate organizations doing research work on the availability of a universal source and sink. The three media to receive the most attention are — atmospheric air, ground water, and the earth. Of the three, least is known about the earth.

Present knowledge of the heat pump indicates that atmospheric air is not suitable for many areas in the United States unless an auxiliary source or storage of thermal energy is provided for periods of low temperature.

Ground water has proved to be a good heat source and sink in those areas where an abundant supply is available relatively close to the earth's surface. The cost of installing and operating such a system, including the disposal of the water, present difficulties that will eliminate ground water as a heat source and sink except in limited areas.

Of the three major heat sources, the earth is the only one left for universal application. The majority of the data available on the earth has been obtained from short time continuous operations. Such tests provide data on the ability of the earth to act as a heat source or sink but not for the conditions under which a heat pump operates. This lack of knowledge has prompted the research work to determine the heat transfer characteristics of Kansas soils under conditions similar to actual heat pump operation.

REVIEW OF LITERATURE

The interest in heat pump development after World War II prompted the Kansas Committee on the Relation of Electricity to Agriculture to initiate a research project to determine the feasibility of the heat pump for use on Kansas farms. The Committee has been active in this work since 1947; this committee has viewed the heat pump as another potential application of electricity which contributes to successful farming and farm living.

Funk (6) states that the first research work was done using buried copper tubing as heat exchangers for the purpose of studying the suitability of the earth to act as a heat source and as a heat sink for the heat pump. A cold brine solution and hot water were used in these tests. One phase of these tests was to investigate the effects of pipe spacing on the amount of heat available from the earth. One heat exchanger consisted of two tubes spaced two feet on centers and in the other one the tubes were spaced ten feet on centers.

In a test to simulate winter operation of the heat pump, cold brine at about 21° F. was circulated in the buried heat exchangers. The tests were continued until the rate of heat transfer from the soil to the tubing reached a practically constant value. The temperature change in the soil and the rate of heat flow were observed throughout the test. At the start of the test the rate of heat transfer from the soil to the tubing was 3.3 Btu/hour/foot of tubing/degree F. difference in temperature between the tubing and the temperature of the earth at a remote spot. This value decreased to 1.8 Btu after 100 hours of continuous operation.

A test to simulate summer operation was conducted in which hot water at about 120° F. was circulated in the buried tubing. At the start of this test the rate of heat transfer was two Btu/hour/foot of tubing for each degree F. difference between the temperature of the heat exchanger and that of the remote earth. This value decreased rapidly to one Btu at the end of 100 hours. The rate of heat transfer continued to decrease slowly as the soil dried around the pipe until it reached a value of 0.83 Btu at the end of 650 hours.

The following conclusions were drawn from the data obtained from these first tests:

1. When operating a ground coil at 20° F., it would be desirable to space the heat exchangers 10 to 12 feet apart and buried six feet deep.

2. About 25 per cent more tubing is required when two tubes are spaced two feet apart in the same trench than when spaced ten feet apart.

3. The heat dissipation rate is considerably less than the heat absorption rate.

4. The thermal energy available due to the heat of fusion of the soil moisture is of minor importance.

Data from these tests indicated that, under the same conditions, it would require approximately 830 feet of such buried tubing to heat and cool an average well insulated six room house with a design heating load of 60,000 Btu per hour.

Lipper (15) presented data on the operation of five "package type" heat pumps using well water as their heat source. It is significant to note that these units were located in river valleys where an abundant supply of ground water was available and at comparatively shallow depths (approximately 20 feet below the surface).

It was found that the cost of electric energy for operating the heat pumps varied from $1.07 to $1.25 per million Btu with an electric energy rate of $0.0125 per kwhr. If engineering principles had been applied to the installation of each unit, it is highly probable that the energy cost would not have exceeded $1.05 per million Btu. The energy cost for heating with these units was competitive with the cost of heating with fuel oil or liquified petroleum gas at the prices that existed during the heating season.

The highest monthly consumption for the "water" type heat pumps was 2,845 kwhr and 249,650 gallons of water for one unit during January of 1950. During this month it was estimated that 32,500,000 Btu were supplied to the house.

The heating coefficient of performance calculated for these units varied from 2.95 to 3.42. All units were identical except the motors used to drive the refrigeration compressors. The variation was a result of one or more of the following factors: differences in air duct design, differences in water system design and differences in evaporator loading. A lack of engineering experience in installing heat pumps caused the wide variation in operating characteristics.

From the viewpoint of the power supplier, it is evident that the energy usage of the heat pump is potentially great. One unit with a 7.5 horsepower motor on the compressor used 17,000 kwhr in 1950. A total of 24,000 kwhr of electric energy was used on the farm. The average annual electric energy consumption for this area was about 2,500 kwhr per farm consumer. Approximately 90 per cent of the energy consumption occurred over a period of six months during the heating season.

The Committee expressed the opinion that the results of the preliminary investigation on the earth heat exchangers would not apply to all soils or to all conditions. A laboratory test procedure for the purpose of predicting the capacity of any soil to act as a heat source or sink was formulated. A new project, employing direct expansion type ground coil heat exchangers, was planned to provide a field check up on the laboratory tests as well as to explore new phases of the problem.

Two extremes of soil types were used as backfill around the buried heat exchangers. The two backfills are: a fine blow sand and a heavy clay. Six separate heat exchangers are buried six feet deep and separated from each other a distance to prevent any interaction in the soil temperatures surrounding them. Direct expansion or condensation of the refrigeration occurs in the buried heat exchanger depending on which cycle the system is operating.

A summary of the tests conducted during the 1950-1951 heating season and the 1951 cooling season is presented in Table 1 as reported by King (13) and Lyman (16).

Kelly (11) states that the soil, unlike most solids, is a porous substance varying in both density and moisture. The thermal properties of soils not only vary with moisture content and density, but each soil type is a different material with a different grain structure. Therefore, the problem of the thermal properties of soils is in reality a study of many different materials.

In Kelly's laboratory investigation, a heating element to simulate the condenser of a heat pump system is placed at the center of a 24 inch cylindrical soil specimen. In conducting a test, the soil is mixed to obtain the desired moisture content. Then the soil specimen is packed into the soil chamber. A water jacket surrounds the soil chamber. During the test the heating element and the water jacket are maintained at a constant preselected temperature. The quantity of heat passing through the test section is determined by measuring the electric energy supplied to the heating element.

The following are observations Kelly made after completing 19 tests:

Table 1. Summary of test results obtained while operating on the heating cycle.

Test No.[1]	Trench No. and backfill material	Pipe temp. 25' from end °F.	Heat flow Btu/hr/ft. of coil	Heat flow Btu/hr/ft/°F.	Conductivity Btu/hr/ft/°F.
1	I (sand)	16.4	25.0	0.88	1.38
1	II (clay)	17.2	41.8	1.50	1.51
2	I (sand)	15.8	30.4	0.96	1.44
2	II (clay)	17.3	40.0	1.31	1.46
2	III(sand)	20.8	31.0	1.16	1.48
2	VI (clay)	20.0	38.8	1.41	0.98
3	I (sand)	Test discontinued after 72 hours due to flood.			
3	II (clay)				
4	III(sand)	110.4	36.6	0.79	1.46
4	VI (clay)	110.0	52.4	1.14	1.05
5	I (sand)	Test discontinued after 48 hours — this test meant only as a check on test 3.			
5	II (clay)				
6	II (clay)	111.4	34.0	0.77	1.30
6	III(sand)	110.0	29.4	0.70	1.44
6	IV (clay)	110.0	54.0	1.23	1.01
6	V (sand)	110.5	54.4	1.22	1.50
7	I (sand)	100.0	26.5	0.75	1.06
		110.2	32.2	0.70	1.16
		119.9	35.7	0.64	1.17
7	II (clay)	100.0	25.2	0.72	1.34
		109.8	32.6	0.71	1.43
		120.0	38.5	0.69	1.50

[1]The operating times of the tests were as follows: Test 1 506.5 hours; Test 2 670.5 hours; Test 4 223 hours; Test 6 195 hours; Test 7 243 hours.

1. An excellent method of measuring thermal conductivity of soil under various conditions of moisture and density is provided by this equipment.

2. A steady state condition is reached after a relatively short time of heat transfer operation.

3. It appears that moisture movement away from a cylindrical heat source is one of the most important factors governing the rate of heat transfer in soils.

4. No direct relationship has been established between moisture content and thermal conductivity in the region immediately surrounding a hot heat exchanger buried in soil. However, it was found that the thermal conductivity increased with an increase in moisture content in this region.

5. It appears that further investigation of the drying effect in soil, near a hot heat exchanger, is necessary in order to predict the performance of an earth imbedded heat exchanger of this type.

Hadley (8) concludes from test data that the moisture in soils has a significant effect on conductivity and its possible effects by gravitational transfer and thermal transfer. In one test approximately 80 per cent of the heat flow could be attributed to moisture migration.

According to Parkerson (17) the moisture migration in soils led to the following conclusions:

1. (a) The influx of moisture into the region around a cold heat exchanger swells and compacts the soil and thus improves the heat transfer characteristics.

(b) On the other hand the efflux of moisture from this region due to the hot heat exchanger causes the soil to loosen, shrink and in some cases contract sufficiently to leave an insulating air gap around the tubing.

2. (a) Heat transfer characteristics for a saturated soil may be used in the design of a ground coil for heat absorption.

(b) Until future experience justifies a different method, caution dictates the use of dry-soil characteristics for the design of a ground coil for heat dissipation.

Coogan (3) presents a mathematical analysis for determining the quantity of heat that can be absorbed from the soil by closely spaced heat exchangers. A reasonable agreement between the mathematical analysis and the experimental results was obtained. He assumed that the remote earth temperature was equal to the average earth temperature between the surface and twelve feet down. He shows this equation:

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

where

q = heat absorbed Btu/hr/ft of tubing

Tg = average remote earth temperature °F.

Tc = surface temperature of coil

D = depth of pipe below surface, ft

d = pipe diameter, ft

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

Z = pipe spacing, ft

ln = natural logarithm

Freberg and Pappas (5) compared heat transfer characteristics of clay and sand. The results indicate that the heat transfer varies with temperature and time. They conclude that sand appears to aid in the movement of moisture to and from the buried heat exchangers. It is pointed out that these results are from tests on one pipe size in one soil type and in one particular locality, Birmingham, Alabama.

Larson (14) presented data on the operation of a ground coil heat pump for the heating season of 1947-48. The daily operating time was determined by the degree days occurring during the preceding day. This plan provided a method of determining with accuracy the actual heat absorption rate of ground coils under practical conditions.

In regular operation, a heat pump may operate several times an hour to maintain a uniform temperature in the conditioned space. Such a plan of operation normally would not be recommended for test purposes since the system required several minutes to stabilize after the start of each cycle. Larson states that to his knowledge this test is the first of its kind in simulating and measuring the heating requirements day by day throughout an entire heating season.

Conclusions from his test data are:

1. The heat pump can be used successfully as a means of space heating in the Chicago area.

2. Conclusions are based on tests in which the periods of operation are proportioned to the day by day heating requirements throughout the heating season.

3. There is a great need for additional data on soil characteristics and the movement of moisture through the soil.

Sporn, Ambrose, and Baumeister (21) have presented an explanation of the fundamental thermodynamic principles involved in any discussion of the heat pump. They have also discussed certain basic heat pump designs, design factors, equipment, and applications. A study of this or any standard text on elementary thermodynamics is desirable as an aid to understanding the fundamental theory of heat pump operation.

THE INVESTIGATION

The primary objective of the research work conducted at Kansas State College has been to obtain fundamental data pertaining to heat pump operation under Kansas conditions. The investigation has been divided into three parts.

1. Operating characteristics of a ground coil when operating on a short time continuous cycle basis.

2. Operating characteristics of a ground coil when operating on a daily schedule of time cycles.

3. Optimum operating temperature of an earth heat exchanger.

Detailed tests were conducted on two types of backfill material. The plan was to obtain heat transfer characteristics, under similated operating conditions, for two extreme types of soil that may be found in Kansas. The soils tested were a heavy clay and a fine blowsand. When the characteristics of these soils are known it is expected that all other Kansas soils will fall between the extremes. The soils were prepared for testing by using them as backfill around the copper coils. The soil was put in place, puddled and leveled to obtain a well packed condition. The backfill material extends for one foot in all directions from the coil.

A total of nine thermal conductivity tests have been completed on the two soils. The tests to be discussed in this thesis will be limited to the two tests on cyclic operation of the heat pump. The test work also included an investigation of soil characteristics. With such data available, it would be possible to calculate how much heat could be extracted from the various soils and thereby design a heat collecting system.

The on-off type test was chosen to determine the effects of cycling on heat transfer rates and soil temperature distribution. In any practical installation, the heat pump will not operate continuously. A study of the data obtained from the operation of the "water type" heat pumps shows that they operated about one-third of the time during January of 1951 under Kansas conditions. They would operate a correspondingly less time in less severe weather. This was true because the design conditions for which the heat pump was installed did not exist for an extended length of time during any month. Since the heat flow was quite high at the start of any test but decreased quite rapidly for the first 45 to 90 minutes, it was decided the length of the operating cycle should be at least four hours. With this in mind, plus reasons of convenience in operation, the test cycles were set up on the basis of six hours of operation and six hours of recovery.

The six heat exchangers used in these tests were identical and were made up of a 50-foot section of ½ inch copper tubing placed in a 1 1/8 inch O.D. copper tube.

LAYOUT OF TEST EQUIPMENT

A schematic diagram of the equipment used in conducting the earth heat exchanger investigation is shown in Fig. 1. The heat exchanger spacing and location is shown in Fig. 1. Also shown is a cross section of the trenches as the heat exchangers were installed.

Temperature Measuring System

A detailed study of the heat transfer characteristics of these soils called for numerous temperature measurements on the buried heat exchanger and in the surrounding soil. The most practical method of obtaining these temperatures with a minimum of earth disturbance was by means of thermocouples.

Figure 1B shows a schematic diagram of the thermocouples in one trench, the arrangement in the other trenches was similar. Twenty gauge copper constantan wire was used for all thermocouple leads. A total of 346 thermocouples, including 16 used to measure the temperature of the earth at a remote location, were used to determine the temperatures at the various points. All leads were treated with a water resistant asphalt compound to lengthen their life. A code system is used in identifying the 346 thermocouples. The code included (1) the radial distance that it lies from the outside diameter of the tubing, (2) the letter of the branch that it is on, and (3) the Roman numeral of the trench.

Four switches, with seven banks of 24 connections per bank, were used with a 16 point automatic Brown recorder to record the temperatures as required.

Refrigerant Control System

The system with its instruments and controls is shown in Fig. 2. Valves that are open on the heating cycle are marked with an "H", and valves that are open on the cooling cycle are marked "C".

The Cooling Cycle. Tracing of the path of the refrigerant through the system on the cooling cycle will explain the plan for determining the rate of heat transfer from the refrigerant to the soil when the heat exchanger is used as a condenser.

Starting with liquid refrigerant leaving the receiver, the path of flow is through the drier and the flowrator. It then goes to a water loaded evaporator coil equipped with a manually controlled expansion valve. The expansion valve is used to regulate the rate of flow so that the condensed refrigerant emerges from the condenser with only a few degrees of subcooling.

From the evaporator, the vapor goes to the compressor inlet. The high temperature, high pressure vapor emerging from the compressor, is directed to the control pit and thence through a super-heater and then the heat exchanger. A standard condenser water regulating valve with its capillary tube connected to the heat exchanger is installed on the outlet of the heat exchanger to maintain a constant predetermined pressure at the outlet. As the earth is warmed and the condenser capacity is thus decreased the refrigerant flow is decreased to maintain the constant subcooled liquid temperature.

The refrigerant vapor is passed through a superheater just before it enters the buried tubing. Although the lines running from the compressor to the pit are insulated, all the superheat has been removed while the vapor passes through the intricate piping system. The superheater is a coil in an electrically heated and thermostatically controlled oil bath. The superheater re-evaporates any refrigerant that may have condensed and imparts to it a few degrees of superheat to prepare it for entering the condenser. It is necessary that the refrigerant be a slightly superheated vapor when it enters the condenser in order that its state may be identified and its enthalpy determined.

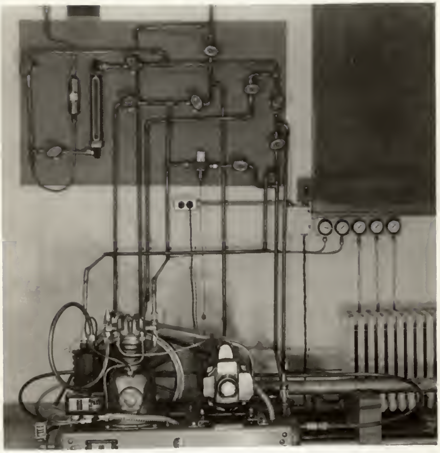
The superheated vapor is admitted to the heat exchanger through the ½ inch tubing down the center of the 1 1/8 inch copper tube. The space between the 5/16 inch tube and the 7/8 inch tube minimizes the heat transfer between the entering vapor and the emerging liquid. As stated before, the rate of flow of the refrigerant is controlled at the expansion valve so that the liquid leaving the condenser is slightly sub-cooled. Thus, its state can be identified and its enthalpy determined.

The liquid is returned from the condenser to the receiver. Its enthalpy

EXPLANATION OF PLATE I

The part of the heat pump system located inside the laboratory including the compressor unit, receiver, flowrator, pressure gages, valves, and lines.

PLATE I



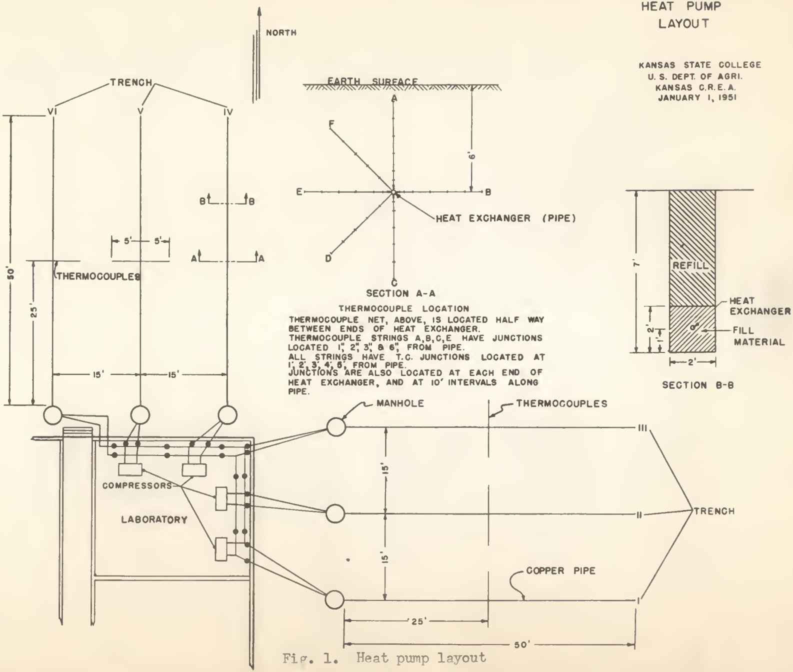


Fig. 1. Heat pump layout

1 EXPANSION VALVE, MANUAL CONTROL
2 OIL SEPARATOR
3 EVAPORATOR PRESSURE
4 FREON COMPRESSOR
5 WATER-COOLED CONDENSER
6 RECEIVER
7 EVAPORATOR (WATER COIL)
8 HEAD PRESSURE REGULATING VALVE
9 SUPERHEATER
10 STRAINER AND DRIER
11 FLOW METER
12 BURIED HEAT EXCHANGER
13 ELECTRIC MOTOR

H INDICATES VALVE OPEN FOR HEATING
C INDICATES VALVE OPEN FOR COOLING

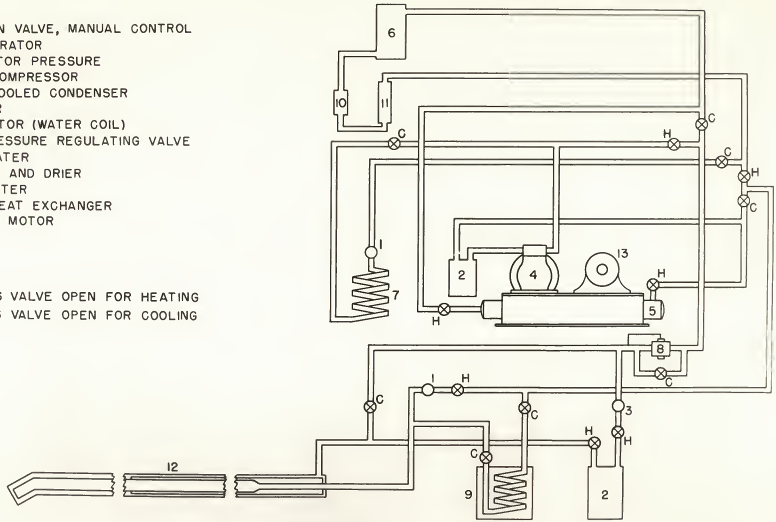


FIG. 2. SCHEMATIC DIAGRAM OF THE REFRIGERATION SYSTEM

change in the condenser is known and the rate of flow is known. From these data it is possible to determine the rate of heat transfer from the heat exchanger to the soil.

The Heating Cycle. Tracing of the flow for the heating cycle in Fig. 2 will show how the system is designed to accomplish its objectives when operating on that cycle. Starting with the receiver, the path of flow is first through the drier and then through the flowrater which is calibrated to measure the flow of liquid Freon-12 in pounds per hour. From the flowrater, the liquid passes to the control pit and enters the manually controlled expansion valve. The liquid refrigerant passes out through the 5/16 and 1/2 inch tube located inside the 1 1/8 inch tube. The liquid refrigerant is evaporated by heat flowing in from the soil. The refrigerant vapor is returned through the space between the large and small tubes.

The rate of flow of the refrigerant is controlled by the expansion valve so that it is slightly superheated when it leaves the 1 1/8 inch tube. The superheating starts approximately four feet back from the outlet of the evaporator. Thus, all but the last four feet of the tube surface are at the same temperature.

By knowing the temperature and pressure at which the liquid enters the expansion valve, the evaporator pressure, and the temperature of the superheated vapor leaving the evaporator, it is possible to determine precisely the enthalpy change of the refrigerant in passing through the evaporator. By knowing the pounds of refrigerant that flow through the evaporator during any time interval, it is possible to determine with accuracy the rate of heat transfer from the soil to the refrigerant during that time.

Several thermostatically controlled expansion valves were tried in an attempt to find a method to control the units automatically. Preliminary tests indicated that accurate control could not be obtained; therefore manual expansion valves were used.

TEST PROCEDURE

The temperature distribution around the buried heat exchangers was determined as a preliminary step before any heat transfer test could be started. If the temperatures in the soil around the tubing were the same as the temperatures measured in the soil at a remote point, the soil was considered to be in a normal state and tests could be started. A typical temperature distribution prior to testing is shown in Tables 2 and 3. These temperatures were recorded on September 25, 1951, just before the summer cycling was started. Similar temperature distribution values for the winter cycling tests are shown in Tables 4 through 9.

In starting a heat dissipation test, the automatic condenser regulating valves were set to maintain a 110° F. condensing temperature in the buried heat exchanger. Refrigerant flow was varied to maintain a constant amount of subcooling at the outlet of the heat exchanger. Any change made in the operating process was recorded to provide a continuous record of the test results.

During the first one or two hours of each cycle, it was necessary to make frequent manual adjustments of the refrigerant flow in order to maintain the proper subcooling. From then until the end of the six hour cycle, period adjustments were made less frequently. The following readings were taken at 30 minute intervals throughout the length of the cycle: refrigerant flow and refrigerant temperature and pressure readings at the inlet and exit of the test section. With these data the enthalpy change and heat transfer were computed.

As the test proceeded the total amount of heat dissipated or absorbed by the heat exchanger would decrease slightly after each cycle. The decrease was quite rapid for the first few days, subsequent cycles showed an increasingly smaller decrease after each cycle. After the heat dissipation or absorption value per cycle reached a relatively constant value, the heat pump was operated on a continuous basis to determine the relationship between heat transfer rates for the two methods.

The temperature distribution data were recorded every three hours to determine temperature rise and recovery during the on and off periods. An operator was on duty 24 hours a day to conduct the tests and make all readings and calculations.

Temperatures along the condenser averaged 110° F. The superheater was set to maintain an average of 115° F. in the superheated vapor supplied to the condenser. The rate of refrigerant flow was varied to maintain an average of 104° F. in the subcooled liquid as it left the buried heat exchanger. The enthalpy change for these conditions was 58 Btu per pound of refrigerant.

The same procedure was used when conducting a heat absorption test except that the temperatures were different. In this case the buried heat exchanger, acting as an evaporator, was held at a constant temperature by setting an automatic back pressure regulating valve. Three evaporator temperatures were used in this winter's heating tests — 10, 20, and 30° F. The average temperature of the refrigerant as it entered the expansion valve was 55° F. The rate of refrigerant flow was varied to maintain an average of 5° F. superheat in the refrigerant vapor as it leaves the heat exchanger.

Table 2. Temperature distribution around the heat exchanger in Trench IV when operating on 12 hour cycles and at 110° F.

Distance from thermocouple to coil (inches)	Vertically up				Vertically down				Horizontally, L.				Horizontally, R.			
	Elapsed time (hours)															
	0	180	186	372	0	180	186	372	0	180	186	372	0	180	186	372
	Temperature degrees F.															
1	67	78	83	87	67	79	99	101	68	78	94	96	67	79	96	98
2	67	77	81	86	67	79	93	97	68	78	90	93	67	79	91	94
3	67	77	79	84	67	79	90	94	68	78	85	90	67	79	87	91
6	67	76	77	82	67	77	82	88	68	75	80	85	67	77	81	86
12	67	73	74	76	67	75	77	80	68	73	74	78	67	74	75	80
24	67	68	70	72	67	69	72	74	68	68	70	72	67	70	72	73
36	67	67	68	69	67	66	69	70	68	66	68	70	67	68	70	70
48	66	68	68	66	67	64	67	68	68	66	67	68	67	66	69	68
60	64	68	70	64	67	64	66	66	68	65	66	66	67	66	68	68

Table 3. Temperature distribution around the heat exchanger in Trench V when operating on 12 hour cycles and at 110° F.

Distance from thermocouple to coil (inches)	Vertically up				Vertically down				Horizontally, L.				Horizontally, R.			
	Elapsed time (hours)															
	0	180	186	372	0	180	186	372	0	180	186	372	0	180	186	372
	Temperature degrees F.															
1	68	74	95	97	68	74	85	86	-	-	-	-	68	74	86	87
2	68	74	85	88	68	74	81	83	68	74	82	84	68	74	81	84
3	68	74	82	84	68	74	79	82	68	74	80	82	68	74	79	81
6	68	74	77	79	66	73	75	78	68	74	76	78	68	74	76	78
12	68	72	73	75	67	71	72	74	68	72	72	74	68	72	72	74
24	68	70	70	71	66	68	68	70	68	69	70	70	68	69	69	70
36	68	68	69	69	64	66	66	66	68	68	69	68	67	68	68	68
48	67	68	68	67	63	64	64	65	68	67	67	68	67	66	66	66
60	65	69	70	64	62	63	64	63	68	67	67	67	67	66	66	66

Table 4. Temperature distribution around the heat exchanger in Trench I, when operating on 12 hour cycles and at 30° F.

Distance from thermocouple to coil (inches)	Vertically up				Vertically down				Horizontally, L.				Horizontally, R.			
	Elapsed time (hours)															
	0	240	246	398	0	240	246	398	0	240	246	398	0	240	246	398
	Temperature degrees F.															
1	45	42	36	33	46	43	38	35	46	42	36	34	46	41	36	34
2	45	42	38	35	46	43	40	36	46	43	38	36	46	42	39	35
3	45	42	39	36	46	44	41	38	46	43	40	37	46	42	40	37
6	44	42	41	38	47	44	44	40	46	44	42	40	46	43	42	39
12	43	41	42	39	48	46	46	43	46	45	44	42	46	44	44	41
24	42	40	41	39	50	48	48	44	46	47	45	43	46	45	45	43
36	40	39	39	39	51	50	50	48	46	47	46	44	46	45	46	44
48	38	36	37	37	52	52	52	50	46	46	46	44	46	46	46	44
60	35	35	37	37	53	53	53	50	46	46	46	44	46	46	46	44

Table 5. Temperature distribution around the heat exchanger in Trench II, when operating on 12 hour cycles and at 30° F.

Distance from thermocouple to coil (inches)	Direction of thermocouples from Coil: Vertically up				Vertically down				Horizontally, L.				Horizontally, R.			
	Elapsed time (hours)															
	0	240	246	398	0	240	246	398	0	240	246	398	0	240	246	398
	Temperature degrees F.															
1	46	40	34	32	43	41	40	37	44	41	39	36	45	43	40	37
2	46	42	37	35	43	40	40	38	44	42	41	38	45	43	42	39
3	45	42	39	37	44	40	41	38	44	43	42	39	45	43	43	40
6	44	43	42	40	44	42	42	40	44	43	44	41	45	44	44	42
12	44	43	43	41	44	43	43	41	44	44	44	42	45	44	44	42
24	42	42	41	41	46	45	44	41	44	44	45	42	45	44	45	42
36	40	40	40	40	48	47	47	45	44	44	45	42	45	44	45	43
48	38	37	38	38	49	48	48	46	44	44	45	42	45	44	45	43
60	36	36	37	38	50	48	48	46	44	44	45	42	45	44	45	43

Table 6. Temperature distribution around the heat exchanger in Trench I, when operating on 12 hour cycles and at 20° F.

Distance from thermocouple to coil (inches)	Vertically up				Vertically down				Horizontally, L.				Horizontally, R.			
	Elapsed time (hours)															
	742	748	877	961	742	748	877	961	742	748	877	961	742	748	877	961
	Temperature degrees F.															
1	38	31	28	41	39	34	32	42	38	32	30	42	38	32	30	42
2	39	34	32	41	39	36	34	42	40	34	32	42	39	34	32	42
3	40	36	33	41	40	38	35	42	40	36	34	42	40	36	34	42
6	40	39	36	40	41	41	38	42	41	39	36	42	40	39	36	42
12	41	41	39	40	43	43	42	43	42	42	41	42	42	42	40	42
24	41	42	40	40	46	46	45	45	44	44	42	43	44	44	42	42
36	41	42	40	40	48	48	47	47	45	44	44	44	44	44	44	44
48	40	40	39	39	49	49	49	49	45	44	44	44	44	44	44	44
60	37	38	39	37	51	51	50	50	45	45	44	45	45	45	45	45

Table 7. Temperature distribution around the heat exchanger in Trench II, when operating on 12 hour cycles and at 20° F.

Distance from thermocouple to coil (inches)	Direction of thermocouples from coil: Vertically up				Vertically down				Horizontally, L.				Horizontally, R.			
	Elapsed time (hours)															
	742	748	877	961	742	748	877	961	742	748	877	961	742	748	877	961
	Temperature degrees F.															
1	31	24	23	32	36	35	33	36	36	33	30	36	36	33	31	36
2	32	29	26	32	37	36	34	36	37	36	34	36	38	36	34	36
3	34	33	30	34	37	37	35	36	39	38	36	38	39	38	34	38
6	39	39	36	38	39	39	36	38	40	40	38	39	40	40	38	40
12	41	41	40	40	40	40	39	40	41	41	40	40	42	42	40	40
24	41	42	40	40	42	42	41	42	42	42	42	42	42	43	42	42
36	41	41	40	40	44	44	43	42	43	43	42	42	43	43	42	42
48	40	40	39	39	45	45	45	45	43	43	42	43	43	44	43	43
60	38	38	39	37	47	47	46	47	44	44	43	44	44	44	44	44

Table 8. Temperature distribution around the heat exchanger in Trench I, when operating on an eight hour cycle at 20° F.

Distance from thermocouple to coil (inches)	Vertically up				Vertically down				Horizontally, L.				Horizontally, R.			
	Elapsed time (hours)															
	1009	1011	1013	1017	1009	1011	1013	1017	1009	1011	1013	1017	1009	1011	1013	1017
	Temperature degrees F.															
1	36	32	31	36	37	34	34	37	36	32	32	36	36	32	31	36
2	36	34	34	36	37	36	35	37	36	34	34	37	36	34	33	36
3	36	36	36	37	38	37	37	38	36	36	36	37	36	36	35	37
6	38	38	38	38	39	40	40	39	38	39	39	38	38	38	38	38
12	40	40	40	40	42	42	42	42	41	42	42	41	40	40	41	40
24	40	41	40	40	45	44	45	45	42	44	44	42	42	42	42	42
36	40	38	40	40	46	47	47	46	44	44	44	44	42	43	43	43
48	37	36	37	38	48	48	49	48	44	44	44	44	44	44	44	44
60	35	35	35	35	50	50	50	50	44	44	44	44	44	44	44	44

Table 9. Temperature distribution around the heat exchanger in Trench II, when operating on an eight hour cycle at 20° F.

Distance from thermocouple to coil (inches)	Direction of thermocouples from coil: Vertically up				Vertically down				Horizontally, L.				Horizontally, R.			
	Elapsed time (hours)															
	1009	1011	1013	1017	1009	1011	1013	1017	1009	1011	1013	1017	1009	1011	1013	1017
	Temperature degrees F.															
1	31	24	24	31	35	35	34	37	36	34	33	35	35	33	33	35
2	31	29	29	31	36	36	35	36	36	36	35	36	36	36	35	36
3	33	32	32	33	36	36	35	36	37	37	37	37	38	37	37	38
6	38	38	38	38	38	38	37	37	39	39	39	39	39	39	39	39
12	40	40	40	40	39	39	39	39	40	40	40	40	40	40	40	40
24	40	40	40	40	42	41	41	41	41	42	42	42	41	41	41	41
36	39	39	40	39	43	43	43	43	42	42	42	42	42	42	42	42
48	37	37	38	37	45	45	45	45	43	43	43	43	43	43	43	43
60	35	35	35	35	46	46	46	46	43	43	43	43	43	43	43	43

The enthalpy change for the 20° F. test was 61 Btu per pound of refrigerant.

TEST RESULTS

A considerable volume of data has been obtained from the two cycling tests covering both summer and winter operation of the ground coil system. Since much of the data were taken only to provide information necessary to compute the heat flow and conductivity, all of the data will not be presented here.

It was believed that the information would be more readily understood and available for further study if presented in curve form. This has been done except for a few tables of data. The curves also provide a ready means for correcting or eliminating values that are obviously in error due to the difficulties that occur during the tests.

Summer Cycling Tests

This test was conducted on Trenches IV and V to compare the effects of cycling on the ability of sand and clay to absorb thermal energy from the heat exchanger. The test was started September 25, 1951, and continued until October 11, 1951. The duration of the test was 384 hours. This date may appear to be late for conducting a summer test, but in checking the soil temperatures in Table 10 it is noted that the soil temperature at six feet has just reached its maximum point. Although the amount of heat to be dissipated for an air conditioning load at this time would not be as high as earlier in the season, the earth's capacity to absorb heat was at its lowest point. After 240 hours of cyclic operation, it was observed that the average rate of heat dissipation per cycle had reached a relatively constant value;

Table 10. Earth temperatures[1] occurring at a location that is thermally undisturbed by operation of the buried heat exchangers.

Date	Depth (ft) 4	6	8	Date	Depth (ft) 4	6	8
	Temperature				Temperature		
Feb 1	42.8	46.1	49.7	May 27	55.3	52.4	51.0
6	41.7	45.9	49.5	June 1	57.5	54.7	52.0
11	41.2	45.1	48.9	6	59.2	56.0	53.8
16	40.9	44.8	48.6	11	59.5	56.5	54.4
21	41.2	44.8	48.5	16	60.4	57.4	55.4
26	41.8	45.0	48.6	21	61.6	58.2	56.0
Mar 3	42.0	44.6	47.6	26	62.4	59.0	56.4
8	43.1	44.7	47.4	July 1	63.0	59.6	57.0
13	43.4	45.6	48.0	6	63.0	60.0	57.8
18	42.9	45.3	48.0	11	64.0	60.0	58.2
23	42.6	45.2	48.0	16	64.4	61.6	59.0
28	43.4	45.0	48.0	21	64.6	61.4	58.9
Apr 2	44.3	45.0	47.5	26	65.5	62.2	59.7
7	44.4	45.7	47.4	31	67.3	63.5	60.4
12	45.4	46.6	47.5	Aug 5	68.9	65.0	62.2
17	44.5	46.6	47.1	10	70.0	65.5	62.0
22	47.4	47.7	49.0	15	70.0	66.0	62.1
27	48.8	48.4	49.5	20	70.0	66.3	63.0
May 2	49.9	48.8	49.8	25	70.0	66.7	64.0
7	51.1	49.4	49.4	30	69.4	66.7	64.1
12	52.9	50.6	50.0	Sept 4	69.2	66.4	63.6
17	53.9	51.7	50.6	9	68.7	66.5	63.7
22	54.7	52.4	50.9	14	67.5	66.0	63.8

Table 10. (concl.).

Date	Depth (ft) 4	6	8	Date	Depth (ft) 4	6	8
	Temperature				Temperature		
Sept 19	67.5	65.9	64.1	Dec 18	48.3	51.1	53.6
24	66.5	65.4	64.0	23	45.6	48.8	52.9
29	65.1	64.4	63.6	28	42.4	44.3	48.3
Oct 4	64.7	64.6	63.6	Jan 2	41.6	46.4	49.6
9	64.0	63.8	63.4	7	41.4	46.0	50.6
14	64.0	63.7	62.6	12	41.6	46.2	50.0
19	63.7	63.7	63.7	17	42.2	46.2	50.2
24	61.9	62.2	62.2	22	42.0	46.0	49.6
29	60.6	61.3	62.0	27	42.3	45.5	48.9
Nov 3	59.0	60.9	61.5	Feb 1	42.1	45.0	48.5
8	55.9	58.7	60.2	6	41.4	43.4	47.5
13	54.6	57.9	59.9	11	42.1	44.0	48.1
18	53.1	56.1	58.2	16	43.0	44.8	48.5
23	52.1	55.4	57.8	21	43.5	45.2	48.7
28	50.5	54.0	56.5	26	44.0	46.0	49.0
Dec 3	50.0	54.0	56.3	Mar 2	43.7	45.9	48.4
8	50.0	52.5	55.5	7	44.0	46.0	48.0
13	50.0	52.3	54.7				

[1]The temperatures recorded are average values occurring during the five day period.

from this time until the end of the test the units were operated continuously.

The extremely variable heat flow is shown in Fig. 3. These data are for two cycles occurring between 180 and 204 hours after the start of the test. Also shown in Fig. 3 is the effect of cycling on the soil temperature. Temperature values plotted are those occurring one inch from the heat exchanger in a horizontal direction. The variation in heat flow for the entire test is plotted in Fig. 4. The value plotted during the cycling portion of the test is the average value for the six hour test.

Rate of Heat Dissipation. Figure 3, showing the variation in heat flow, was of most immediate interest. The minimum value for the clay soil is about 66 Btu per hour per foot of tubing and a little more than 80 Btu per hour for the sand. These values were reached after about 180 hours of cyclic operation, and remained practically constant until the end of the test. The average heat flow for these tests on the clay and sand were 76 and 88 Btu per hour per foot of tubing, respectively.

Variation in Soil Temperature. These differences in heat flow on the two trenches may be analyzed by examining the temperature distribution curves of Fig. 3, which show cycles in the earth temperatures. These are average values that occurred during the last one fourth of the cycling test. The curves clearly show that the temperature of the clay soil builds up to a higher value during the cycle than does the sand. This fact is important in that the lower the earth temperature, other things equal, the higher will be the heat flow. It is not clear what factors are responsible for the better heat transfer characteristics of the sand as compared to the clay.

Winter Cycling Tests

These tests were conducted on Trenches I and II to compare the effects of cycling on the ability of sand and clay to provide thermal energy to a ground coil heat exchanger. The tests were started January 22, 1952, and continued until March 9, 1952. The duration of the test was 1,128 hours. The remote earth temperature was 45.9. The variation in earth temperature as the test progressed is shown in Table 10.

A major portion of the heating load for a conditioned space occurs during the time covered by this test. Also the minimum earth temperature occurs within this period. This means that this test provides a comparison at a time that is the most critical for ground coil installations.

The variable heat flow from the soil to the heat exchanger, operating at 30° F., is shown in Fig. 5. These data are for two cycles occurring between 240 and 264 hours after the start of the test. The effect of heat pump cycling on the change in soil temperatures is also shown in Fig. 5. Temperature values plotted are those occurring one inch from the heat exchanger in a horizontal direction. Data obtained while operating the ground coil at 20° F. for two different time cycles are shown in Figs. 6 and 7.

The variation in heat flow for the entire test is plotted in Fig. 8. The values plotted during the cycling portion of the test are the average values for the "on portion" of the cycle.

Rate of Heat Absorption. The minimum average value of heat transfer during the "on portion" of the cycle is the value of importance for design of a ground coil heat exchanger. It is apparent from Figs. 5, 6, and 7 that the values vary considerably depending upon both temperature of the coil and type of backfill around the coil. These values are tabulated in Table 11.

Table 11. Heat transfer rates for different temperatures and soil types.

Backfill material	Temperature along coil, °F.	Time of cycle (hours)	Heat flow for Operation	
			Cycle[1]	Continuous
Sand	30	12	27	15
	20	12	31	23
	20	8	37	23
Clay	30	12	20	11
	20	12	30	24
	20	8	27	24

[1]The heat flow reported for the cycle portion are average values obtained while the unit was operating. Units of heat flow are Btu/hr/ft of tubing.

Table 11 shows that the heat flow for continuous operation was more than doubled after the ground coil was operated at 20° F. rather than 30° F. in the clay backfill and the heat flow was increased about 50 per cent through the sand backfill. It is also observed that the heat transfer values for cyclic operation are about 50 per cent greater than the value for continuous operation for both backfills and at the two different temperatures.

Variation of Soil Temperatures. Variation of earth temperatures for these soils are shown in Figs. 5, 6, and 7. These are average values that occurred during the latter portion of each test when the heat flow was relatively constant for each successive cycle. These curves show that the temperatures in the sand backfill reach lower values at the end of the on cycle and did not recover to a temperature that is as high as those in the clay backfill.

DISCUSSION OF TEST RESULTS

The values of heat flow from these tests are affected by numerous factors. A few of the more important factors are as follows: Transient effects during operation; temperature of the refrigerant, soil characteristics; and the effect of adjacent pipes on one another.

The transient effects on heat flow are readily apparent from Figs. 3, 5, 6, and 7. In each case the heat flow is relatively high at the beginning of each cycle; it decreases rapidly during the first two hours and then more slowly until the end of the test run.

The temperature curves in the lower halves of Figs. 3, 5, 6, and 7 show that the heat flow decreases in proportion to the change of temperature in the backfill around the heat exchanger. The earth temperatures in the sand backfill recovered quicker than the clay and therefore it was capable of providing a better heat transfer medium.

The effect of the fast recovery characteristic of the sand backfill is shown in Table 11 and Figs. 6 and 7. The heat flow through the sand soil increased about 20 per cent when the cycle time was reduced from 12 to 8 hours. The reverse trend was true for the clay soil, the heat flow decreased about 10 per cent when the cycle time was reduced.

The temperature difference between the refrigerant in the heat exchanger and that of the soil is called the driving force for heat transfer. The greater the temperature difference the higher the heat flow—other things being equal. The temperature of the refrigerant should be kept within certain limits. Operating efficiencies of Freon-12 condensing units are adversely affected when the condensing temperature exceeds 120° F. or the evaporator temperature falls below 20° F.

These data indicated that the soil type had a definite relation to the thermal energy transfer capacity. Data from these tests were not inclusive enough to provide a definite explanation to the variation in heat flow through different soil types. It was apparent that part of the variations were due to the following differences: density, conductivity, moisture movement, and the film effect at the heat exchanger.

The pipe spacing in these tests was 15 feet. Test results indicated that there was not any inter-reaction between the coils. Previous tests here at Kansas State College compared the effects of coils spaced two and ten feet apart on heat transfer.

Storage of heat in chemicals is currently being proposed for use in connection with the heat pump and particularly for improving the capacity and operating economy of air-to-air type units under design conditions. It appears from these data that it would be profitable to investigate the use of heat storage with heat pumps using the earth as a heat source. For example, if a given heat pump unit was designed to operate continuously under design conditions, that same earth heat exchanger would supply 3/4 of the design heating load when operating only 1/2 the time (Table 11). This means that the operating time could be reduced by 50 per cent with a sacrifice of only 25 per cent in the total heat delivered. If this 25 per cent of design heating capacity could be provided by heat storage for a period of time determined by the probable duration of design conditions, it appears that operating economy could be improved. The improvement would be derived from the increased average rate of heat transfer to the ground coil and the higher evaporator pressure that would result where thermostatic expansion valves are used in the refrigerant system. Providing the necessary heat storage seems to be a possibility that is within reason. A study of Kansas weather data has shown that for this area, 35 per cent of the design heating capacity is required for only 1.42 per cent of the heating season degree days. This means that a house with a design heating load of 80,000 Btu per hour would require approximately 600,000 Btu in heat storage.

In results of the nine tests to determine heat transfer characteristics of soil types the clay soil transferred more thermal energy on a continuous operation basis than did a sand soil. On the other hand, the sand soil transferred more thermal energy on a cycling basis than did the clay soil. These test results also indicate it might be advantageous to use a fine blow sand for the backfill rather than the original earth occurring at the site.

SUMMARY

1. A heat exchanger buried in the soil is capable of providing a heat source and a heat sink for the heat pump.

2. The rate of heat transfer to or from the soil is primarily affected by the soil type and condition and by the basis of heat pump operation.

3. Results of the tests so far show that clay provides a higher rate of heat transfer under continuous operation, whereas sand provides a higher rate of heat transfer when operating with "on" and "off" cycles.

4. Moisture content and moisture migration are apparently very important factors affecting the heat absorption rate. Moisture moves towards the cold coil in winter and away from the hot coil in summer operation.

5. In the summer cycling tests any difference in moisture migration in the two soil types was not noticeable in the heat flows obtained. Moisture movement is appreciable in only the first one or two inches and so far there

is not any method available to make accurate moisture measurements for these tests.

6. These tests indicate that a ground coil could be used effectively as a supplemental heat source for an air to air heat pump installation or could be used advantageously in conjunction with heat storage.

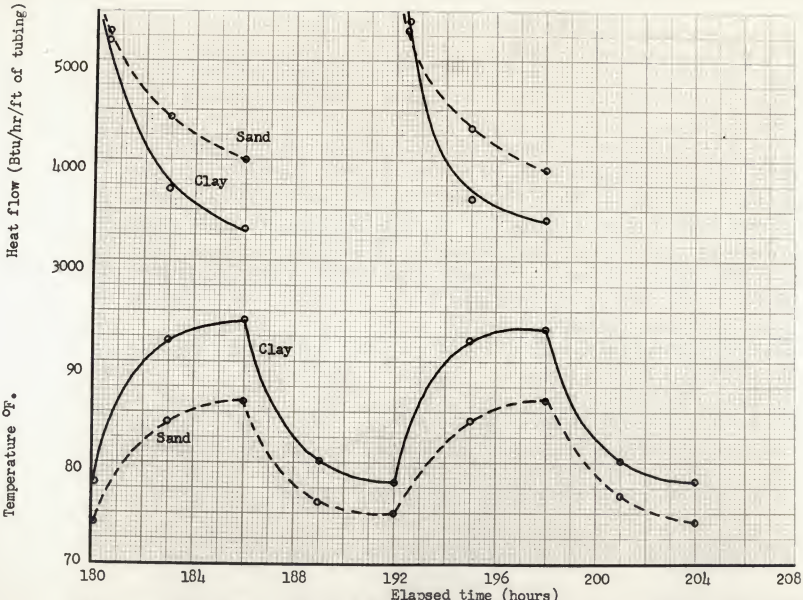


Fig. 3. Variations of heat flow and temperature distribution for summer cycling tests on Trenches IV and V, operating on 12 hour cycles and at 110° F.

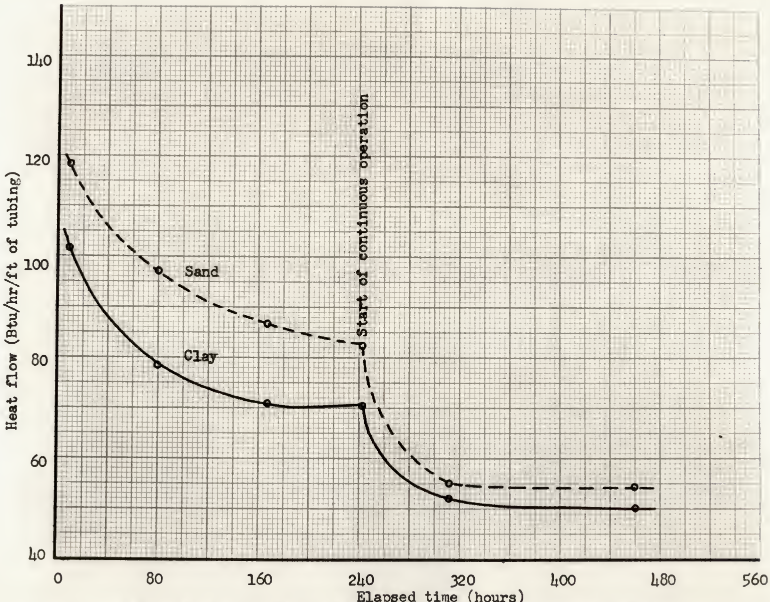


Fig. 4. A comparison of heat flows during cycling and continuous operation on Trenches IV and V when operating the condenser at 110° F.

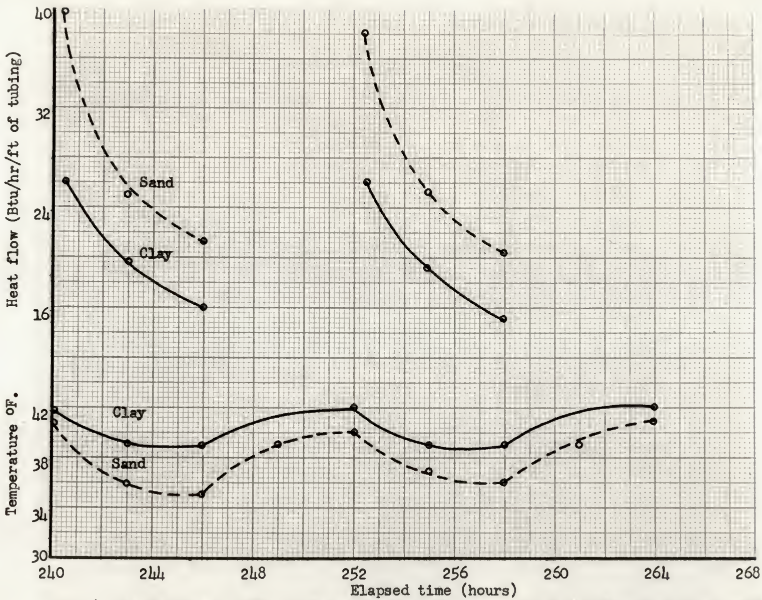


Fig. 5. Variations of heat flow and temperature distribution for winter cycling tests on Trenches I and II, operating on 12 hour cycles and at 30° F.

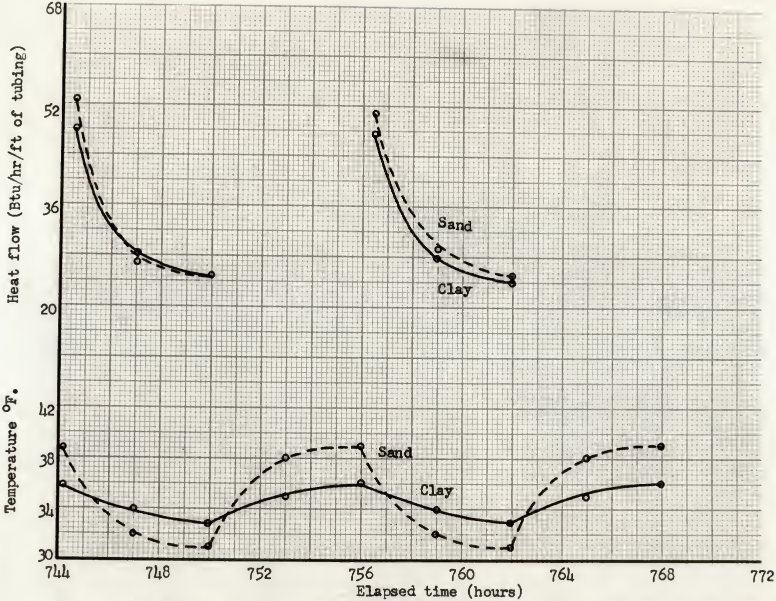


Fig. 6. Variations of heat flow and temperature distribution for winter cycling tests on Trenches I and II, operating on 12 hour cycles and at 20° F.

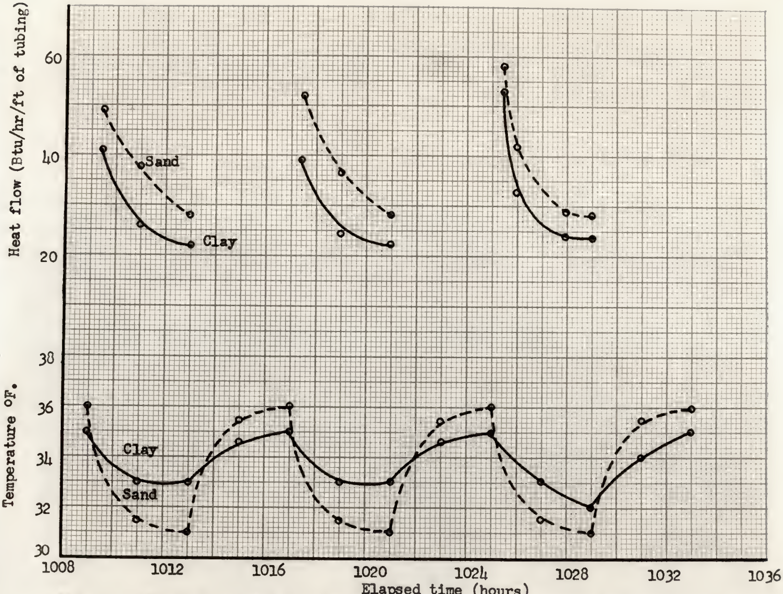


Fig. 7. Variation in heat flow and temperature distribution for winter cycling tests on Trenches I and II, operating on eight hour cycles and at 20° F.

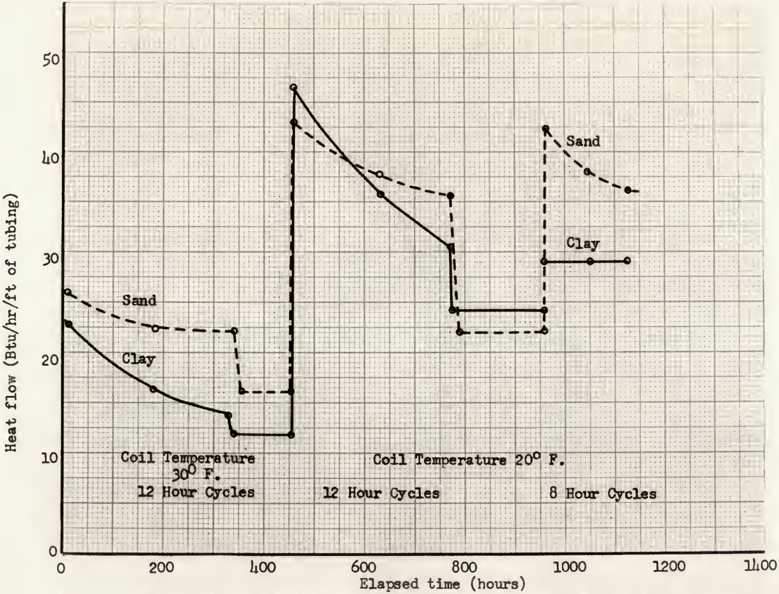


Fig. 8. Variations in heat flow for Trenches I and II under various winter operating conditions.

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A STUDY OF CYCLIC AND CONTINUOUS HEAT PUMP OPERATION AS IT AFFECTS HEAT TRANSFER RATES FOR TWO SOIL TYPES

by

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INTRODUCTION

It is becoming more apparent that the only practical method for heating electrically is through some device which will multiply the effectiveness of electricity for heating purposes by several times its direct energy conversion value. The heat pump is such a device.

Recent figures released by the Edison Electric Institute list about 600 heat pump installations in 39 of the 48 states. It is possible that there are as many more units that have not been reported. All but 50 of these have been installed since 1945.

The basic problem in the development of the heat pump is to find a comparatively cheap and reliable heat source and heat sink. At the present time there are at least 50 separate organizations doing research work on the availability of a universal source and sink. The three media to receive the most attention are--atmospheric air, ground water, and the earth. Of the three, least is known about the earth.

The majority of the data available on the earth has been obtained from short time continuous operations. Such tests provide data on the ability of the earth to act as a heat source or sink but not for the conditions under which a heat pump operates. This lack of knowledge has prompted the research work to determine the heat transfer characteristics of Kansas soils under operating conditions similar to actual heat pump use.

THE INVESTIGATION

The primary objective of the research work conducted at Kansas State College has been to obtain fundamental data pertaining to heat pump operation under Kansas conditions. The investigation has been divided into three parts:

1. Operating characteristics of a ground coil when operating on a short time continuous basis.

2. Operating characteristics of a ground coil when operating on a daily schedule of time cycles.

3. Optimum operating temperature of an earth heat exchanger.

Detailed tests were conducted on two types of backfill material—a heavy clay soil and a fine blowsand. The soils were prepared for testing by using them as backfill around the copper heat exchangers. The soil was put in place, leveled and puddled to obtain a well packed condition. The backfill material extended for one foot in all directions from the heat exchanger.

The on-off type test was chosen to determine the effects of cycling on heat transfer rates and soil temperature distribution. In any practical installation the heat pump will not operate continuously. A study of the data obtained from the operation of the "water type" heat pumps shows that they operated about one-third of the time during January of 1951. This was true because the design conditions for which the heat pump was installed did not exist for an extended length of time.

Six identical heat exchangers were used in these tests. They were made by placing a 50-foot section of 1/2 inch copper tubing inside of a 1 1/8 inch O. D. copper tube. This method was used to eliminate end losses as much as possible. The clay soil was used as a backfill around three of the heat exchangers and the sand was used on the remaining three. Direct expansion or condensation of the refrigerant occurred in the buried heat exchanger. Heat transfer calculations were made after the enthalpy changes in the refrigerant were determined.

TEST PROCEDURE

The tests on the two soil types were run simultaneously to compare the heat transfer rates under similar conditions. In conducting a heat dissipation test, automatic condenser regulating valves were set to maintain a 110° F. condensing temperature in the buried heat exchanger. Refrigerant flow was varied to maintain a constant amount of subcooling (approximately 5° F.) at the outlet of the heat exchanger. Any change made in the operating process was recorded to provide a continuous record of the test data. Refrigerant flow and refrigerant temperature and pressure readings at the inlet and exit of the test section were taken at 30 minute intervals. This was the data required to determine the enthalpy change of the refrigerant as it passed through the ground coil. After the heat dissipation rate per cycle reached a relatively constant value the heat pump was operated on a continuous basis to determine the relationship between heat transfer rates for the two methods.

The same procedure was used when conducting a heat absorption test except that the temperatures were different. Automatic evaporator pressure regulating valves were used to maintain a constant temperature along the heat exchanger. The rate of refrigerant flow was varied to maintain an average of 5° F. superheat in the refrigerant vapor as it left the heat exchanger.

TEST RESULTS

The values of heat flow from these tests are affected by numerous factors. A few of the more important factors are as follows: Transient effects during operation; temperature of the refrigerant, soil characteristics; and the effect of adjacent coils on one another.

In each case—dissipation or absorption—the heat flow was relatively high at the beginning of each cycle; it decreased rapidly during the first two hours and then more slowly until the end of the test run.

A summary of the test results obtained from the summer and winter tests including the cycling and continuous operation and at the several temperature conditions is presented in Table 1.

Table 1. A summary of the tests conducted on the two soil types.

Backfill material	Temperature along coil, °F.	Length of test (hrs)	Length of cycle (hrs)	Heat flow for operation	
				Cycle[1]	Continuous
		Summer Tests			
Sand	110	384	12	88	60
Clay	110	384	12	76	54
		Winter Tests			
Sand	30	456	12	27	15
Sand	20	505	12	31	23
Sand	20	167	8	37	23
Clay	30	456	12	20	11
Clay	20	505	12	30	24
Clay	20	167	8	27	24

[1]The heat flow reported for the cycle portion are average values obtained while the unit was operating. Units of heat flow are Btu/hr/ft of tubing.

It appeared from these data that it would be profitable to investigate the use of heat storage in chemicals for heat pumps using the earth as a heat source. For example, if a given heat pump unit is designed to operate continuously under design conditions, that same earth heat exchanger will supply

3/4 of the design heating load when operating only 1/2 the time. This means that the operating time can be reduced by 50 per cent with a sacrifice of only 25 per cent in the total heat delivered. Providing the necessary heat storage seems to be a possibility that is within reason. A study of Kansas weather data has shown that for this area, 35 per cent of the design heating capacity was required for only 1.42 per cent of the heating season degree days. This means that a house with a design heating load of 80,000 Btu per hour would require approximately 600,000 Btu in heat storage.

In results of the nine tests to determine heat transfer characteristics of soil types, the clay soil transferred more thermal energy on a continuous operation basis than did a sand soil. On the other hand, the sand soil transferred more thermal energy on a cycling basis than did the clay soil. These test results also indicated it might be advantageous to use a fine blow sand for the backfill rather than the original earth occurring at the site.

SUMMARY

1. A heat exchanger buried in the soil is capable of providing a heat source and a heat sink for the heat pump.

2. The heat transfer to or from the soil is primarily affected by the soil type and condition and by the basis of heat pump operation.

3. Results of the tests so far show that clay provides a higher rate of heat transfer under continuous operation, whereas sand provides a higher rate of heat transfer when operating with "on-off" cycles.

4. Moisture content and moisture migration are apparently very important factors affecting the heat absorption rate. Moisture moves in the direction of the temperature gradient. Under winter operation the moisture

moves towards the coil and away from it in summer operation.

5. In the summer cycling tests any difference in moisture migration in the two soil types was not noticeable in the heat flows obtained. Moisture movement is appreciable in only the first one or two inches and so far there is not any method available to make accurate moisture measurements for these tests.

6. These tests indicate that a ground coil could be used effectively as a supplemental heat source for an air heat pump installation or could be used advantageously in conjunction with heat storage.