Space Heating by Electric Radiant Panels and by Reverse-Cycle

By
LOUIS SLEGEL
Professor of Mechanical Engineering

Bulletin Series No. 24

July 1948

Issued under a cooperative research grant from the Bonneville Power Administration.

Engineering Experiment Station
Oregon State System of Higher Education
Oregon State College

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PART A. HEATING BY ELECTRIC RADIANT PANELS

I. INTRODUCTION

1. **Purpose and scope of electric radiant panel project.** The art of space heating by means of low temperature radiant surfaces has been known for some time, and this method of heating has been used in Europe, especially in England, for a number of years. It has been only recently, however, that radiant heating has been used to any appreciable extent in the United States.

During recent years, a great deal of research and development work has been carried through on this type of heating in this country. A large number of radiant heating installations of all kinds have been made during these years. Today a great variety of equipment designed and manufactured expressly for radiant heating is available. Research and development in this field are still being carried on.

Because of their many desirable and advantageous features, prefabricated parts, grids, and even complete panels have been produced and are now available for hot water radiant heating installations. It seemed highly desirable at the time this project was initiated that a prefabricated panel for electric radiant heating be developed. At the time the research under this project was started, much of the design procedure then in use or proposed was largely based on certain assumed empirical factors. Through practical experience and research by others in many quarters, this situation has since been improved to some extent.

Inasmuch as few controls had as yet been developed specifically for electric panel heating, it seemed desirable also to make a study of several types of controls which were commercially available, to determine their suitability for this purpose and to develop new ones if the experience with present controls indicated the necessity or desirability of doing so. The purpose of this project has been, therefore, threefold: first, to design, construct, and obtain operating data on electric panels of a prefabricated type; second, while operating these panels, to obtain data which might be used basically for design procedure generally; and third, by actually using controls of various kinds, to determine their suitability and the requirements of controls for this type of heating in general. In the following pages is given a detailed description of the panels as constructed as well as detailed data on their performance under varying conditions.

2. **Acknowledgments.** This project has been made possible through a cooperative research grant from the Bonneville Power Administration. This is the second publication to be issued in connection with this project. (Circular 9, *Electric and Other Types of House Heating Systems*, 1946, was the first.) The author acknowledges his indebtedness and expresses his sincere thanks to the following concerns, all of whom generously furnished materials of some kind which made possible the construction of the laboratory at a time when most materials were extremely difficult to obtain; or who contributed, either outright or as a loan, instruments which were necessary in the carrying out of the research work involved:

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Driver-Harris Co., Harrison, New Jersey
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II. PANEL CONSTRUCTION

1. **Mechanical construction.** As has already been indicated, one of the purposes of this project was the construction of an electric radiant panel which could be manufactured commercially, and which would be suitable and satisfactory, from the standpoints of construc-

tion and performance as well as installation and cost, as a prefabricated panel. These requirements were kept in mind in the design and construction of the panels.

Three sizes of panels were constructed; two panels 2 ft x 8 ft; two 2 ft x 6 ft; two 4 ft x 8 ft. For commercial purposes, possibly both the lengths and widths of panels of this type should be multiples of 16 inches in order to accommodate their installation on studs at 16-inch center distance. However, construction could be the same regardless of size.

The construction of all panels was the same, with one exception, which will be explained later. The panels consisted of two layers of ¼-inch cement-asbestos board bolted together with the resistance wire embedded in grooves in one layer of the board and lying between the two layers. This construction is shown in Figure 1.

The grooves were routed in the panel by means of a power saw and were about ⅛ inch wide and 3/32 inch deep. All grooves were parallel in the panels and ran full length of the panels, except for about 2 inches at each end. Groove spacing in all panels was 2 inches.

2. **Resistance wire.** The resistance wire was then placed in these grooves and held in place by running small wires through the panel, these small wires being removed after the two boards of the panels were bolted together. The resistance wire used in these panels was No. 18 bare, high strength, 30 per cent conductivity Copperweld wire. This wire has a resistance of 21.70 ohms per thousand feet and was used primarily for two reasons, the principal reason being the fact that it has almost exactly the conductivity which was calculated as being required, and secondly, because of its relatively low cost as compared to certain alloy wires. Each panel contained only one

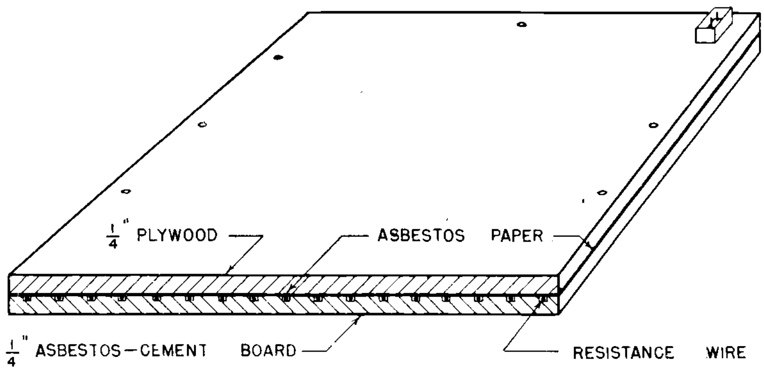

Figure 1. Sectional View Showing Construction Details of Electric Radiant Panels.

resistance wire, this being placed in the panel as explained, in a continuous length, going from one parallel groove to the next by means of a groove routed at each end of the panel at about 2 inches from the edge, as indicated previously. After the wire was secured in place in the grooves, the two boards of the panel were placed together with the joining surfaces coated with an adhesive compound, and the boards were bolted together; the small wires holding the resistance wire in position were then removed.

The panels were completed by bringing the two ends of the resistance wire out and fastening each end to a bolt running through both boards of the panel. These bolts ended in an outlet box on the back of the panel, thus serving both as anchors for the ends of the resistance wires as well as connecting terminals for the power lines.

One exception to this construction has been mentioned. This exception is that in four of the panels, one layer of $\frac{1}{4}$-inch fir plywood was substituted for one of the layers of cement-asbestos board. In these panels the wires were embedded in grooves in the asbestos-cement board as before. A sheet of asbestos paper was glued to this part of the panel, covering the entire grooved surface; the plywood board was then bolted to the grooved board, sandwiching the asbestos paper between, the asbestos paper thus serving to insulate the resistance wire from the plywood.

One other exception in the construction of the plywood backed panels, a detail which might be of importance, as will be pointed out in the discussion of performance data, is the relative location of the wires in the panels. In the panels made of two boards of asbestos-cement board, the wire was laid in grooves in the back panel while in the panels made of one layer of asbestos-cement board with a board of plywood on the back, the wires were laid in grooves in the asbestos-cement board which was, of course, the front or surface board.

In order to determine, if possible, the difference in the operating temperature of the resistance wire in the panel caused by the small air space surrounding the wire in the grooves, in some cases, noted specifically later, some of the grooves were filled with an insulating compound after the resistance wire was in position.

3. **Thermocouples.** During the construction of all these panels, thermocouples were placed in them in various locations in order to determine the temperatures at these locations during operation. When couples were placed in the wire grooves for determining the temperatures of the resistance wires, they were insulated electrically from the resistance wires by means of small porcelain insulators, into which the couples were inserted.

The junction boxes on these panels, in which the connection posts were located, were placed far enough from the edge of the panels so that they did not interfere with joists when the panels were hung.

The electrical resistance of each panel of course depended on the length of resistance wire in it. A 4 ft x 8 ft panel contained approximately 180 feet of resistance wire, giving it a total resistance of about 3.90 ohms. The total resistance of all of the eight panels was 17.60 ohms.

III. THE LABORATORY

1. **Dimensions and arrangement.** The laboratory in which all tests on these panels were made was a frame structure 20 ft x 30 ft x 16 ft in height. The cold room measured 20 ft x 20 ft x 16 ft in height; the rest of the building consisting of two floors, provided instrument and apparatus rooms.

The test room itself was 12 ft x 12 ft x 8 ft in height. This room, which will be called the warm room, is thus surrounded at top, bottom, and the four sides by 4 feet of air space, this air space being maintained at any required temperature by means of a cold-blast evaporator coil hung from the ceiling and in one corner of the cold room.

All joists, both ceiling and floor, and all studs in the warm room were 2 in. x 4 in. Walls consisted of ½ inch of fiber board on the inside of the studs, a layer of tar sheathing paper, 2 inches of cotton between studs fastened to the inside surface of the studs, paper side in, and ¼-inch fir plywood on the outside of the studs. The floor consisted of ½-inch plywood with 2 inches of cotton between joists and heavy tar paper tacked to the under side of the joists to enclose completely the joist space. This tar paper under the joists was used for two purposes: to retain any of the cotton insulation which might peel off or sag, and to cut down the air currents between joists, air currents which would otherwise very probably be set up in the air circulated by the fans in the cooling unit.

2. **Radiant panels.** The ceiling consisted of the radiant panels fastened to the under side of the ceiling joists by wood screws, and located as shown in Figure 2, with 3½ inches of crepe paper insulation laid on top of these panels in the joist spaces, and either ½-inch plywood or ½-inch insulating board laid on top of the ceiling joists. Panels 7, 8, 5, and 1 were of asbestos-cement board and plywood construction; the other four of asbestos-cement board entirely.

Care was taken throughout the construction of the warm room to make it as completely air tight as practicable. The one opening into the warm room, the doorway, was weatherstripped.

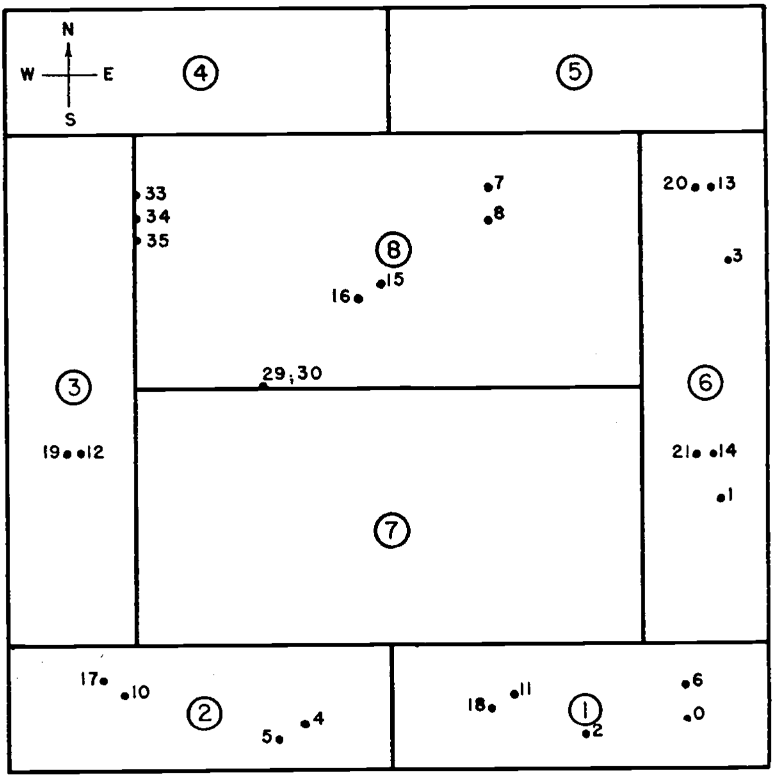

Figure 2. RADIANT PANEL AND THERMOCOUPLE LOCATIONS.

IV. THERMOCOUPLES AND THEIR LOCATION

1. **Panels and couple locations.** Figure 2 shows the arrangement of the heating panels as they were placed in the ceiling, as well as the location of the thermocouples located in these panels. Numbers in circles are panel numbers. The numbers not circled are thermocouple numbers. Table 1 indicates the location of all thermocouples used during the course of the tests.

Couples 0 to 3 inclusive were placed on the top or joist side of the panels, Couples 0 and 1 directly over resistance wires, and Couples 2 and 3 midway between two adjacent runs of the resistance wire, as indicated.

Couples 4 to 8 inclusive were buried couples. Those buried at a wire were in the wire groove, insulated from the resistance wire by a small porcelain insulator, as has already been stated; Couple 4 was midway between two adjacent runs of resistance wire and on joining surface of the two boards making up the panel.

Couples 10 to 21 were located as indicated. The thermocouple indicated as 22 is actually three couples connected in parallel in order to obtain average readings. These couples were placed on corner-to-corner diagonals, 7 inches from the two adjoining walls, in each of three corners. They, as well as Couples 23 and 24, were embedded in the floor, flush with the floor surface.

Table 1. Identification and Location of Thermocouples

Couple No.	Location
0	Top surface of Panel 1, over wire
1	Top surface of Panel 6, over wire
2	Top surface of Panel 1, over midpoint
3	Top surface of Panel 6, over midpoint
4	Buried couple of Panel 2, at midpoint
5	Buried couple of Panel 2, at wire
6	Buried couple of Panel 1, at wire
7	Buried couple of Panel 8, at wire, unfilled groove
8	Buried couple of Panel 8, at wire, filled groove
9	Spare for calibration
10	Room surface ceiling of Panel 2, at wire
11	Room surface ceiling of Panel 1, at wire
12	Room surface ceiling of Panel 3, at wire
13	Room surface ceiling of Panel 6, at wire (north)
14	Room surface ceiling of Panel 6, at wire (south)
15	Room surface ceiling of Panel 8, at wire
16	Room surface ceiling of Panel 8, at midpoint
17	Room surface ceiling of Panel 2, at midpoint
18	Room surface ceiling of Panel 1, at midpoint
19	Room surface ceiling of Panel 3, at midpoint
20	Room surface ceiling of Panel 6, at midpoint (north)
21	Room surface ceiling of Panel 6, at midpoint (south)
22	Floor surface corner 7 in., 7 in.; three points average
23	Floor surface 4 ft, 4 in.; three points average
24	Floor surface center; two points average
25	North wall, 1 ft, 4 in. from ceiling; average of 5 couples
26	North wall center row; average of 5 couples
27	North wall 1 ft, 4 in. from floor; average of 5 couples
28	South wall, 1 ft, 4 in. from ceiling
29	Globe temperature; average of 4 couples
30	Air temperature at globe; average of 4 couples
33	Air temperature of room 6 in. from ceiling
34	Air temperature of room midway
35	Air temperature of room 6 in. from floor
36	Cold room
37	North wall, outside, center; average of 5 couples

The couple indicated as 23 is, likewise, three couples connected in parallel, each of the three couples being located 4 ft 4 in. from a corner of the room on corner-to-corner diagonals.

Couples 25, 26, and 27 are each five couples connected in parallel and located as indicated. These couples were located in three rows, each couple being 24 inches from the next couple in the same row and the couples on the end of each row were 24 inches from the adjoining wall, thus dividing the twelve foot wall space into six equal parts of two feet each.

2. **Globe thermometer.** Couple 29 is actually a quadruple couple. The individual couples were soldered at equal intervals in a horizontal diametral plane of a 6-inch copper sphere. The sphere was of light gage copper and had no openings. After the thermocouples were soldered to it, the sphere was painted dull black with a light coat of paint. This device was used as a modified globe thermometer. This globe was located as indicated, midway between floor and ceiling.

Couple 30 was also quadruple as indicated. Individual couples were suspended at equal spacing about the periphery of the globe and approximately 2 inches from the globe, and were connected in parallel.

Couples 33, 34, and 35 were located as indicated. These couples were shielded by 2 concentric cylindrically shaped foil shields and by a circular shield above and below each couple. All foil was bright on both sides. Couple 33 had 2 circular shields above it; couple 35 had 2 circular shields below it. The temperature in the warm room was also recorded by a recording thermometer as well as checked regularly by mercury-glass thermometers, both shielded and unshielded.

The temperature in the cold room was recorded by a recording thermometer as well as checked by Thermocouple 36 and by a mercury-glass thermometer.

Five thermocouples, 37, were placed on the outer side of the warm room wall exactly opposite to Couples 26.

All thermocouples used were 24 gage copper-constantan. All junctions were kept as small as practicable, particularly those used for indicating air temperatures, and all junctions were arc fused.

3. **Details of couple installation.** All couples used for indicating surface temperature were very carefully fixed in place. In the ceiling panels a shallow groove was cut; the couples were cemented in place and covered with a very thin layer of Portland cement. The couples were flush with the panel surfaces. The couples in the floor were treated in the same way except here they

were covered with plastic wood. In the walls the couples were barely covered with fibers from the wall material itself. All leads followed the surfaces for at least 3 inches from the thermocouple junction. All thermocouples were connected to a common constantan lead. The cold junction was placed in ice water kept in a thermos bottle. Two thermocouples were calibrated before the tests were begun. A close check was made and a calibration curve was drawn from which all conversions from millivolts to degrees Fahrenheit were made.

V. CONTROLS

1. **Types investigated.** Two types of thermostatic controls were used during the course of these tests. One type was of the line-load design which could be connected directly across the power lines and needed no separate switch or relay arrangement, the power switches being contained in the thermostats themselves. Thermostats of the other type were of low capacity and required relays to open or close the power circuits. Some details of the operating characteristics of these thermostats will be included in the discussion of the test data.

While all of the data given in this report were taken using Thermostat A, all of the thermostats used in these tests will be identified here, as their performance will be discussed in a later section.

2. **Thermostat descriptions.** Thermostat A was a control of the low capacity type, requiring a relay to control the power to the heating circuits. The temperature-sensitive element of this control was of bi-metal. This thermostat was of the heat-anticipating type, with adjustable temperature differential. The temperature differential was listed by the manufacturer as being from ½ to 3 degrees. All data taken with this control were with the differential set close to the low side or at approximately ½ degree.

The relay used with this control and all other thermostats not of the line-load type was of self-energizing design and was placed in the circuit between the power source and the variable transformer.

Thermostat B was of the line-load type, with a vapor-filled temperature sensitive element. This control was nonanticipating and was nonadjustable as to temperature differential.

Thermostat C was of the same general design and was made by the same manufacturer as Thermostat A, but was of an older type without heat anticipation.

Thermostat D was of design entirely different from any other thermostat in the group. This control was of extremely low electrical capacity and, judging from the experience in the laboratory, was very sensitive to temperature. The sensitive element of this control was the case or cover, and this instrument was stated to be sensitive to radiant energy as well as to the ambient air temperature. It was not designed to be used as it was in these tests, but with and without modifications in the electric circuit a considerable amount of data was obtained with this control.

Thermostat E was made in the laboratory. It consisted primarily of a wafer type, vapor-filled thermostatic bellows mounted to actuate a microswitch. This control could be used either with or without a relay.

All thermostats were mounted in the middle of one wall, 5 feet from the floor.

VI. TEST PROCEDURE

The procedure used in all tests was as follows. The cold room was maintained as closely as possible at constant temperature, about 34 F, and with the warm room at some low temperature level, usually at about 42 F, and the control thermostat set at a fixed temperature, the power was thrown into the heating circuit.

Both voltage and wattage in the heating circuit were continuously recorded by means of a recording voltmeter and a recording wattmeter. An integrating wattmeter was also used. A Variac was used in the power circuit for varying the voltage for different tests. Both the recording meters and the integrating meter were in the circuit between the Variac and the heating elements in order not to include losses in the Variac.

All thermocouple circuit electro-motive forces were read at frequent intervals throughout each test by means of a potentiometer. The intervals between potentiometer readings varied to some extent in order to assure taking readings at critical periods during the tests, these intervals varying from 10 to 30 minutes.

After the power circuit was opened for the first time in each run by the thermostat, indicating that the thermostat had been satisfied, the recording of all data was continued until sufficient data had been recorded for each test condition to give a complete picture of the characteristics of that run.

When the above data had been recorded for most runs, a small fan was turned on in the warm room and the test procedure repeated. This small circulating fan was placed in the center of the floor and was tilted upward at about 30 degrees from horizontal so that the air passing through the fan swept the east wall and then passed over the ceiling surface to some extent.

The entire series of tests on these panels included a run, as outlined above, for each of the following conditions: Thermostats A, B, C, D, and E unshielded, all panels operating, fan on and fan off; Thermostat A, shielded by a sheet of bright aluminum foil placed around it in such a way as to shield it from direct radiation, but allowing a natural circulation of air around it, all panels operating; Thermostat A, unshielded, with center Panels 7 and 8 only operating, fan on and fan off; Thermostat A, unshielded with border Panels 1, 2, 3, 4, 5, and 6 operating, fan on and fan off. All data of these tests are not included in this report, but data which indicate the general operating characteristics of the panels for different conditions have been plotted and are shown in a later section of this bulletin. Shorter runs for various other conditions were also included in the complete test procedure.

VII. PERFORMANCE DATA AND RESULTS

1. **Warm-up requirements.** The data for a warming-up period are not detailed in this report, but may be summarized as follows. With all panels operating at 220 volts and a total resistance of 17.60 ohms in the heating coils, making a total of approximately 2,750 watts in the circuit, it required 3.5 hours to heat the room from 38 F to 72 F, the temperature at which the thermostat was set. At the end of the 3½-hour period, the thermostat, being satisfied, opened the circuit and turned off the power to the heating panels. Temperature outside of the warm room was approximately 35 F during this time.

The 2,750 watts energy demand mentioned above is the result of theoretical calculation, assuming a line voltage of 220 volts and a resistance of 17.6 ohms. The resistance of the circuits was determined when the panels were at a temperature of about 70 F, with very low voltage. Based on the integrating watt-hour meter during the 3½-hour warming-up period, there was a load of 2,440 watts on the panel circuits at 220 volts. Based on the recording wattmeter, the load was 2,480 watts. The discrepancy between the actual and the theoretical load demand is due largely, no doubt, to a variation in the resistance of the heating elements with a variation in temperature but probably also in part to variations in line voltage. According to manufacturers' data on this wire, the temperature coefficient of resistance is 0.0021 per degree Fahrenheit.

With all panels operating at 160 volts, it required seven hours to warm the room from 39 F to 72 F. Based on the wattmeter data, the load on the resistance circuit was 1,520 watts during this run. As compared to the 3½-hour warming-up period at 220 volts, the warming-up period at 160 volts required just twice the period of time at about 62½ per cent of the power load.

2. **Performance data plotted.** Performance data for most of the tests made during the entire investigation have been plotted. The plotted data for three distinct conditions are included in this report. These three conditions were selected as representing the three ways in which panels of this type might be used, as ceiling panels, and also because they reveal the temperature variations of most of the thermocouple points that would be of interest or of value.

These three test conditions are as follows: with all panels operating at 220 volts, designated as Run M, both with fan on and fan off; with border panels operating at 177 volts, designated as Run N, with fan on and fan off; with center panels only operating at 157 volts with fan on and fan off, designated as Run Q.

The three voltages of 220, 177, and 157 were arrived at in attempting to maintain the same power loads on the heating circuits in all of these three runs. The resulting power loads were approximately 2,460, 2,510, and 2,470 watts respectively at 220, 177, and 157 volts. Because of external loads on the same power lines which served the laboratory, it was impossible to maintain the loads on the heating circuits constant because of voltage fluctuations. These fluctuations caused some discrepancies in the resulting data as will be discussed more in detail later.

In order to make the data more comparable, all three runs M, N, and Q were made with the same thermostat, A, set at the same temperature and with the same differential setting.

In the graphs which represent the plotted data of temperature variations at the thermocouples, each curve is identified by a number at each end of the curve. These are the numbers of the thermocouples the temperature variations of which are represented by that curve.

3. **Room temperature, all panels operating.** Referring to the curves for Run M, fan off, during the regular power on-off cycle, the room air temperature, Couple 30, varied about 3 degrees or from 72 F to 75 F. The largest temperature gradient between the air at six inches from the floor, Couple 35, and six inches from the ceiling, Couple 33, was about 9 degrees or from 69 F to 78 F. The globe temperature, Couple 29, varied only slightly from the air temperature at the globe, Couple 30.

4. **Floor temperatures.** The floor surface temperature at the corners, Couple 22, is consistently lower than the air temperature six inches from the floor, Couple 35, while floor surface temperatures, Couples 23 and 24, are consistently higher than Couple 22.

5. **Wall surface temperatures.** Wall surface temperatures of the north wall, Couples 25, 26, 27, show a consistent gradient down-

ward from upper to lower levels in the wall. There is slight discrepancy between Couples 28 and 25, most of which is undoubtedly due to the fact that while Couple 28 is a single couple, 25 is five couples in parallel as has been pointed out previously.

6. **Ceiling panel temperatures.** With reference to ceiling panel temperatures, it is quite noticeable that Couple 15 room surface ceiling Panel 8 over the wire and Couple 11, room surface ceiling Panel 1 over the wire, are consistently higher at the high points on the curves than are corresponding points on Panel 3, 2, or Panel 6, Couples 12, 10, and 13, respectively, for example. Also couples buried at the wire in Panel 8, Couples 7 and 8, and in Panel 1, Couple 6, are consistently higher than Couples 5, Panel 2. Couple 18, room surface of Panel 1 midpoint between resistance wires, is consistently higher than Couples 19, 20, and 21, or 17, corresponding points on Panels 3, 6, and 2, respectively.

On the top surface of the panels, the temperature of Couple 0 over the wire on Panel 1 is consistently a little higher at highest points on the curves than that of Couple 1, Panel 6. The temperatures of Couple 2, top surface of Panel 1, at midpoint are slightly lower than those of Couple 3, at a corresponding location on Panel 6, even though the temperatures of Couple 6 are higher than those of Couple 5. Couple 0, top surface of Panel 1, over wire is slightly higher than Couple 1, Panel 6. However, the temperature at the wire of Panel 1 was considerably higher than the temperature at the wire of Panel 6.

7. **Panel types compared.** All of these facts seem to indicate that the room surface temperatures and temperatures at the wires of the asbestos-cement-plywood panels are consistently higher at peak temperatures than the temperatures of corresponding points on the asbestos-cement board panels and, also, that the temperatures of the top surface of the plywood panels are lower than the asbestos-cement panels. These data are not conclusive in this respect and they would be improved in this regard if all couples had been multiple, parallel couples which would have given a better indication of temperatures as more thermocouple positions would have been represented.

8. **Effect of fan.** Referring to the curves for Run M, fan on, the most apparent factor here is that all room air temperatures have been brought noticeably closer together. With the fan on, for example, the largest difference between the temperature of Couple 33, six inches from ceiling, and Couple 35, six inches from floor, is close to three degrees, while with the fan off it was nine degrees. The temperature variation at Couple 30 is about three degrees, or from

73.5 F to 76.5 F. Here again the temperatures of the floor toward the center, Couples 23 and 24, are higher than the temperature of the air six inches from the floor, Couple 35. The temperatures at different levels on the wall surface have been brought closer together and the gradient here is much less than with the fan off.

All other temperatures for Run M, fan on, follow the same general relative pattern as they did for Run M, fan off, except that, for example, the surface temperature of Panel 6 at midpoint, Couples 20 and 21, is now slightly lower than the corresponding points on Panel 2, Couple 17. This is very probably because the air circulated by the fan swept across Panel 6 more than it did across Panel 2. The temperatures at Couple 13 are consistently lower than those at Couple 12 undoubtedly for the same reason; likewise, temperatures of Couple 3 are lower with the fan on than those of Couple 4 with the fan off; the reverse of this is true with the fan on; all ceiling panel temperatures are lowered as compared with the corresponding temperatures with the fan off. The reason for this of course is obvious, since the convection coefficient at the surface of the panels was increased appreciably with the fan operating even though, as has already been pointed out, the fan was small and was on the floor, not directed toward the ceiling. The fan served primarily only to circulate the air in the room; in doing so it wiped the layer of warm air away from the ceiling and mixed it with the cooler air that was lower in the room.

9. **Border panels only operating.** Referring to Run N, fan off, this series being with only the border panels operating, it is seen that the general and relative trends of all curves are the same as for Run M, with all panels operating. There are several minor exceptions to this, and one quite noticeable difference in the general trend. For example, during Run N, the temperatures of Couples 28 and 25 are relatively higher than those of Couple 34 than they are with all panels operating. This is because the temperatures of all couples in the operating panels are higher when only part of the panels are operating than they are when all panels are operating, and the temperatures of the couples on the walls close to the border panels, which are operating in this case, will consequently be higher also.

When only part of the panels were operating, those panels were furnishing as much heat to the room as were all of the panels when all were operating, since the power load in both cases was practically the same. This means, of course, that the temperature of the radiant panels when only part are operating must be higher than when they are all operating. This accounts for the temperatures of all couples in the border panels for Run N being higher than the corresponding temperatures for Run M, especially at the tops of the curves, or in the region of the peak temperatures.

For Run N, fan off, the maximum difference between the temperature at Couple 33 and that at Couple 35 was about 6½ degrees, or from 71.5 F to 78.0 F. The maximum variation at Couples 30 was about 2 degrees, or from 72.5 F to 74.5 F.

Referring to the data for Run N, fan on, the graphs show that the maximum difference between the temperature at Couple 33 and Couple 35 is less than 2 degrees. The variation in temperatures at Couples 30 is less than 3 degrees, or from 74 F to about 77 F. Both of these conditions are better than with all panels operating; that is, they vary within closer limits. During this run, for the first time the peak temperatures of Couple 5 exceed those of Couple 11, although they have been very close throughout all of the data.

Referring to the data for Run Q, fan off, it is seen that all relative and general trends are very much like those for Run N, fan off, as well as for Run M, fan off. The peak temperatures of buried and surface couples in the panels are all consistently higher than for any other run. This is due to the fact that the panel area now heating the room is the smallest of these three runs, resulting in higher panel temperatures being required. This undoubtedly accounts for the higher variation in temperatures at Couple 30, for example, the maximum variation here being about 4 degrees, or from 72 F to 76 F. The greatest difference in temperature between Couple 33 and 35 is about the same, 4 degrees. These curves show a very slight but perceptible difference between the temperature at Couple 34 and those at Couple 30. These differences are no doubt due to the fact that Couple 34 is not as close to the center of the operating panels as is Couple 30, as is shown in Figure 2.

10. **Center panels only operating.** The data from Run Q, fan on, show a maximum difference in the temperatures of Couples 33 and 35 to be less than 3 degrees, or from about 75 F to slightly over 77 F. The maximum variation in the temperature of Couple 30 is less than 3 degrees, from about 69 F to about 72 F.

These data show the temperatures of Couples 30 to be slightly higher than those at Couple 34. This difference is almost certainly due to the fact that Couples 30 were closer to the center of the radiating panels than was Couple 34, as has already been mentioned. All other data and trends in the group are relatively the same as those in the other runs so far as comparison can be made.

11. **Thermostat response.** It is noticeable, particularly in Run M, fan off, and Run N, fan off, that the temperatures of nearly all couples start at higher levels than they reach at any other phase of the cycle. In these two cases, the first points plotted on the curves are the temperature readings taken after the warming-up period; these were taken just at the peaks, either immediately before or immediately after the power was shut off by action of the thermostat at the end of the warming-up period. The subsequent peak points of nearly all curves in these runs are lower than the temperatures at which the curves start. This condition is due to the lag in the action of the thermostat, allowing the air and other temperatures to overrun before the thermostat acts. Once the second phase of the cycling of the system has started, the thermostat appears to control at the same temperatures in succeeding phases of the cycle.

In Run M, fan off, the temperatures of Couples 34 and 30 at the beginning of the regular cycling overran the temperatures of the couples at the peaks during the regular cycle by about 2.5 degrees, reaching the peak temperature of 77.5 F, while the peaks during the regular cycle were all at about 75 F.

VIII. CONCLUSIONS

1. **Temperatures in panels.** It is of interest to note that the highest temperature recorded, that of Couple 7, buried alongside the resistance wire in Panel 8, was in the neighborhood of 146 F. The temperature at any point in these panels, while they are operating, is peculiar, of course, not only to the particular construction of the panels themselves, but to the construction, size, and shape of the room, et cetera. However, in this particular test, 64 square feet of panel were transmitting some 2,500 watts of electricity in the form of heat, making about 40 watts per square foot of panel. At this rate a 12 ft x 20 ft area, for example, would receive 9,600 watts. This is more than probably ever would be required in a normally constructed house. These facts indicate that from the standpoint of operating temperatures, panels of this type are perfectly safe and satisfactory.

2. **Failures experienced.** Twice during the course of these tests failure occurred in the panels. In both instances the failure was at a terminal post in an asbestos-cement-plywood panel and resulted in the burning in two of the resistance wires. These failures were due to construction defects and were in no way due to defects in the component parts.

Where the resistance wire connects with a terminal post, a loop at the end of the wire is turned around the post and held in place by a washer and nut. In both instances of failure referred to, this connection at the post had worked loose, arcing occurred, and the resistance wire was burned completely in two. The plywood was charred and burned around the terminal post, under the terminal box. No further damage resulted.

This defect in construction could be remedied either by placing a jam nut on the terminal post or by hard soldering or brazing this connection to assure permanent contact.

Because of somewhat faulty workmanship in hanging the panels, one of them cracked slightly during the course of the tests. This could easily have been prevented by proper mounting of the panels in the first place. While no precise record was kept, these panels probably operated for a total period of approximately 1,500 to 2,000 hours. At the end of the tests one of the panels was disassembled. There was no discernible deterioration of the wire or other component parts of the panel. Aside from the two failures due to arcing at the terminals and the avoidable cracking of one panel, these panels performed with entire satisfaction throughout the tests and appeared to be serviceable for an indefinite period of use.

3. **Cycle periods and power requirements.** Inasmuch as the amount of power furnished to the resistance elements was practically the same in all three of these tests, with all panels operating, center panels only, and border panels only, it is of interest to note that the cycle periods are practically the same in all three runs, as would be expected. The typical cycle consisted of a phase which averaged about forty-seven minutes in which the power was off, and an average period of twenty-seven minutes in which the power was on. Individual time intervals varied, partly because of the line voltage fluctuations, which have been mentioned.

Based on these data, a load of about 900 watts would be required to maintain the same temperature conditions within the room if this load were applied continuously. On this basis, a load of approximately 0.78 watt per cubic foot of room space would be required continuously to heat the room while power at the rate of about 2.15 watts per cubic foot was supplied with intermittent operation.

Naturally, this rate would vary with the size and construction of the space to be heated. Good prediction for any other structure could be made from these data by considering the construction and size of the laboratory room, along with its heat losses, which could be calculated theoretically as well as checked with the actual power requirements. The data could then be modified or corrected to arrive at the calculated requirements of the proposed structure.

All data indicate that two of the properties of the panels which were felt to be desirable were obtained; these are low thermal capacity and good thermal conductivity. Both of these properties should result in panels having quick response and little lag. Panels having little lag and which are quick to respond to the heat source or heating medium should decrease the tendency to overheat the room at the times when the thermostat turns off the heat and to underheat the room at the times when the thermostat turns on the heat. In other words, panels having these characteristics would tend to eliminate overruns and underruns, or would keep the room at a more uniform temperature. The fact that these panels have this characteristic is indicated by the curves which show a rapid rise and fall of temperatures within the panels themselves when the power is turned on and off, respectively.

The thermal capacity and the thermal response of a panel are directly related. A panel consisting of a large mass of material of high specific heat will have a large thermal capacity. Such a panel will require a long period of time to warm up and, likewise, will require a long period of time to cool off. Thus it will have a definite tendency to overrun and underrun the thermostat, and will, accordingly, have slow response to the control.

On the other hand, a panel which consists of a small amount of material having a low specific heat will have low thermal capacity and will be quick to respond to the control, eliminating or at least decreasing the tendency to overrun or underrun the thermostat, giving quick response both in heating and cooling. Any consideration of the problem of thermal response of a heating system must, of course, include a study of the control of the system, specifically the thermostat.

4. **Conditions controlling lag.** In order that the entire panel heating system shall possess minimum lag or the tendency to overrun or underrun the demands of the thermostat, and also that the temperature variations in the heated space be kept at a minimum, two requirements are apparent. First, the panels themselves must be of low thermal capacity and high thermal conductivity. Second, the thermostat must have a low temperature differential and should preferably be sensitive to radiant energy as well as to ambient air temperatures. The closeness of control or the temperature range within which the thermostat will maintain a heated space depends on its temperature differential. This differential is adjustable in some thermostats; in others it is fixed, or non-adjustable. In general, thermostats of high electrical capacity have a differential of about 3 degrees; that is, they will actuate on and off at temperatures 3 degrees apart. For example,

a thermostat having a temperature differential of 3 degrees and controlling the temperature of the ambient air in a room will open the circuit when the air surrounding the thermostat is 73 F, for instance, and will close the circuit when the air reaches 70 F.

Because of the operating and design principles of a thermostat, those designed to operate at low electrical capacity may have a smaller temperature differential than those of larger capacity. The load capacity of a thermostat can of course be increased by means of a relay.

5. **Thermostats.** All plotted data of this report were obtained using Thermostat A, with temperature differential set at about $\frac{1}{2}$ degree. This thermostat gave the best and most consistent results of all the controls used in the course of these tests. While the temperature variations of the air in the room at any fixed level were the smallest with Thermostat A, even these variations were outside of the set range of the differential of this control in some cases.

Tests made with this thermostat as well as with Thermostats B and C in which these controls were operated with their covers removed and also with these thermostats shielded from direct radiation from the panels by means of aluminum foil gave data only slightly different from the data when operated normally, indicating that these controls are only slightly sensitive to radiant energy and are affected almost entirely by the temperature of the surrounding air.

Thermostat B which was used without relay gave somewhat wider variations of both air and surface temperatures than any other thermostat used. While this thermostat was represented to be particularly sensitive to radiant energy, the results of the laboratory tests did not support this claim.

Thermostat C performed very much like Thermostat A, except for a slightly greater temperature differential.

Thermostat D, as already stated, was not designed to be used as it was in the laboratory. Because both the voltage and the current in the circuit which this thermostat was controlling were both much greater than it was designed for, the performance of this control was extremely erratic. However, the laboratory tests indicated that this control was the most sensitive not only to ambient air temperaures but was second only to Thermostat E in being sensitive to radiant energy.

Thermostat E operated with about the same temperature differential as Thermostat C. As indicated by operating it shielded and unshielded, this control was more sensitive to radiant energy than any other control in the group. This thermostat was designed and made as a result of the apparent need of a control of this type; that is, one that is sensitive to radiant energy, of fairly high electrical capacity, and not excessive in cost. Laboratory tests indicate that the first two requirements were met. Judging from the costs of the components of this control, the last requirement, that of low cost, should not be difficult to accomplish.

6. **Effects of fan.** It has already been noted that when the air in the room being heated by the radiant panels in the ceiling was circulated by means of a small fan, the temperature gradient of the air in the room as well as the gradient of the wall surfaces at different levels was considerably reduced. Circulating air in the panel-heated room results in lowering the proportion of heat dissipated from the panel by radiation and increases the proportion dissipated by convection. This, then, tends to bring the system closer to being a convection rather than a radiation system.

The laboratory test data did not appear fully to justify the claims often made that a radiant ceiling panel system produces a low floor to ceiling temperature gradient within the heated space. It must be borne in mind, however, that the test room was practically airtight, a condition not generally met in normal house construction. This fact alone does not account for the fairly high air temperature gradients recorded for Run M. In a house, however, the movement of the occupants in walking through or within the rooms, as well as the air movements caused by opening and closing doors and so on, would very probably be sufficient to circulate the air within the room enough to bring the otherwise relatively high air temperature gradient down to a not undesirable level.

Except for Run M, that is, for Runs N and Q, the data showed only a moderately high temperature gradient within the room. In a room with eight foot ceiling height, a temperature gradient from six inches below the ceiling to six inches above the floor of approximately 5 to 6 degrees in still air is not unreasonable. Very little air circulation would be required to bring this temperature gradient down to a very acceptable level.

Even without forced circulation of the air within the room, the test data show that the floor surface temperatures were consistently higher than the temperature of the air six inches from the floor. Warm floors are recognized as a considerable comfort factor in a house. Even relatively higher floor temperatures would probably be obtained in a house than were obtained in the laboratory since there was considerably more air circulation under the room floor than would be encountered in any properly constructed house.

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CHARTS FOR TEMPERATURE VARIATION DATA

The following six charts represent the plotted data to which reference has been made. As will be noted, there are two charts for each of the three operating conditions; run M, with all panels operating at 220 volts; run N, with border panels only operating at 177 volts; run Q, with center panels only operating at 157 volts. For each of these three conditions, data are plotted with still air in the warm room, designated as "fan off," and with the small fan operating in the warm room, designated as "fan on."

Figure 3. Temperature Variations, Run M, Fan Off.

Figure 4. Temperature Variations, Run M, Fan On.

Figure 5. Temperature Variations, Run N, Fan Off.

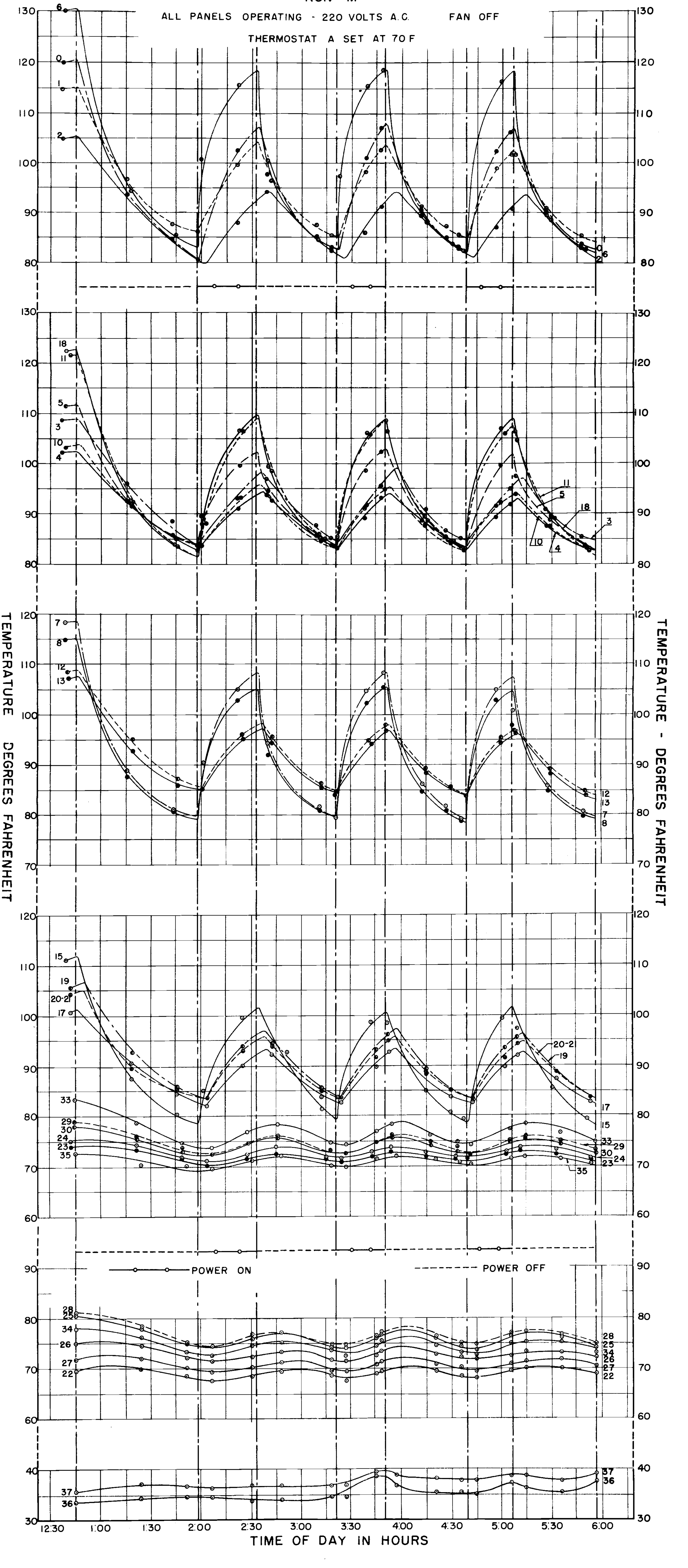
Figure 6. Temperature Variations, Run N, Fan On.

Figure 7. Temperature Variations, Run Q, Fan Off.

Figure 8. Temperature Variations, Run Q, Fan On.

TEMPERATURE VARIATIONS

RUN "M"

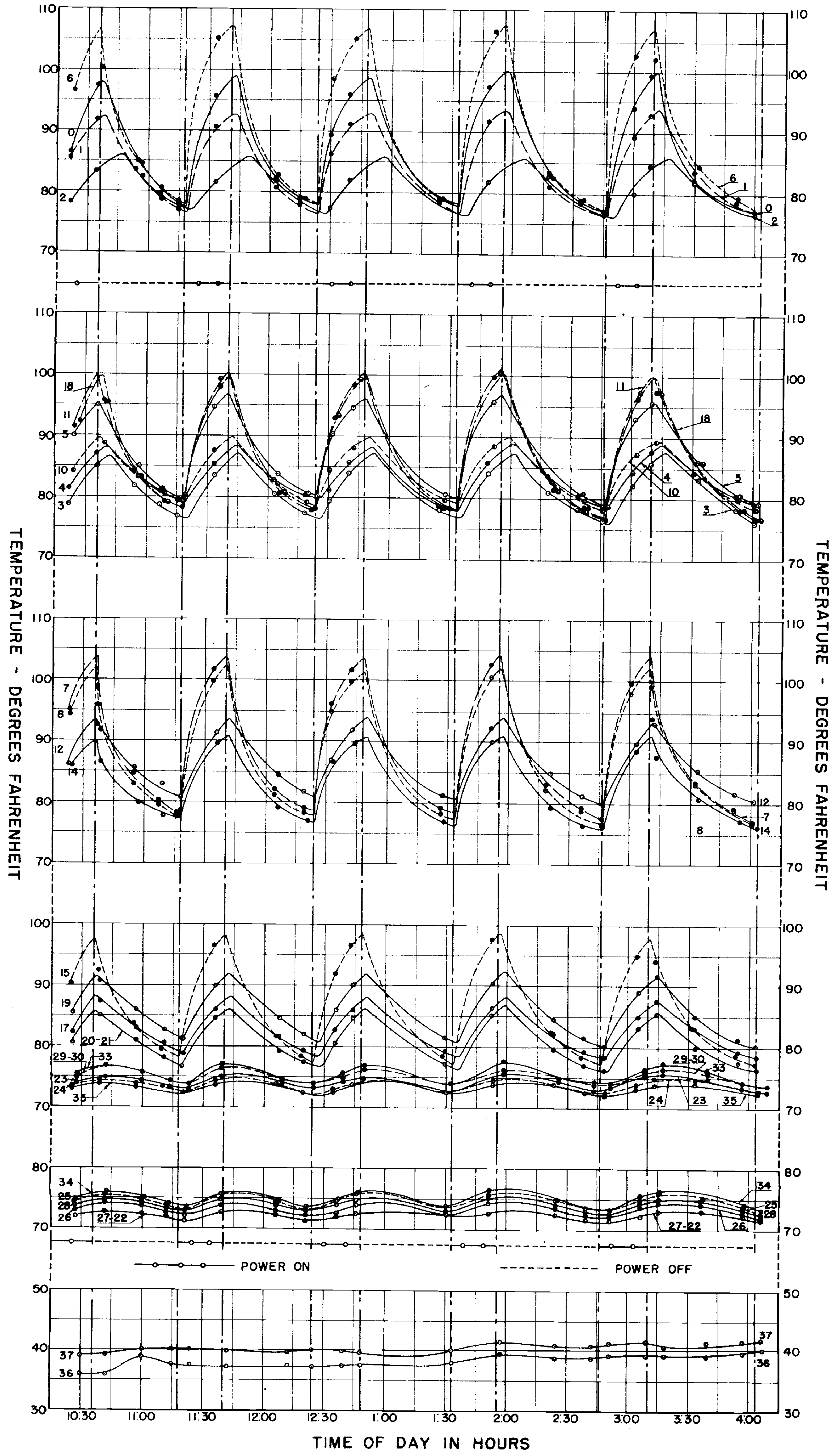


TEMPERATURE VARIATIONS

RUN "M"

ALL PANELS OPERATING - 220 VOLTS A.C. FAN ON

THERMOSTAT A SET AT 70 F

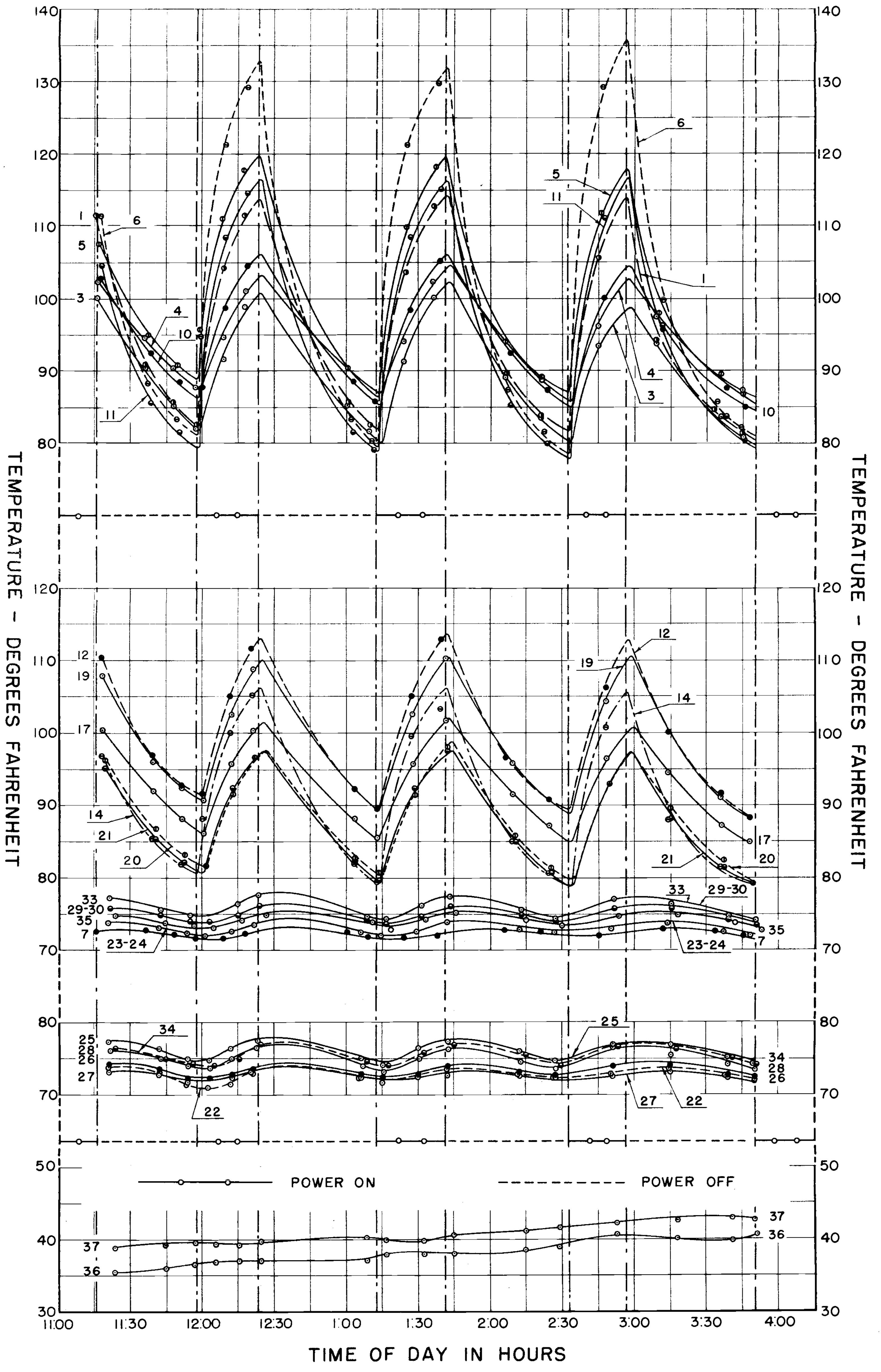


TEMPERATURE VARIATIONS

RUN "N"

BORDER PANELS OPERATING — 177 VOLTS A.C. FAN ON

THERMOSTAT A SET AT 70 F

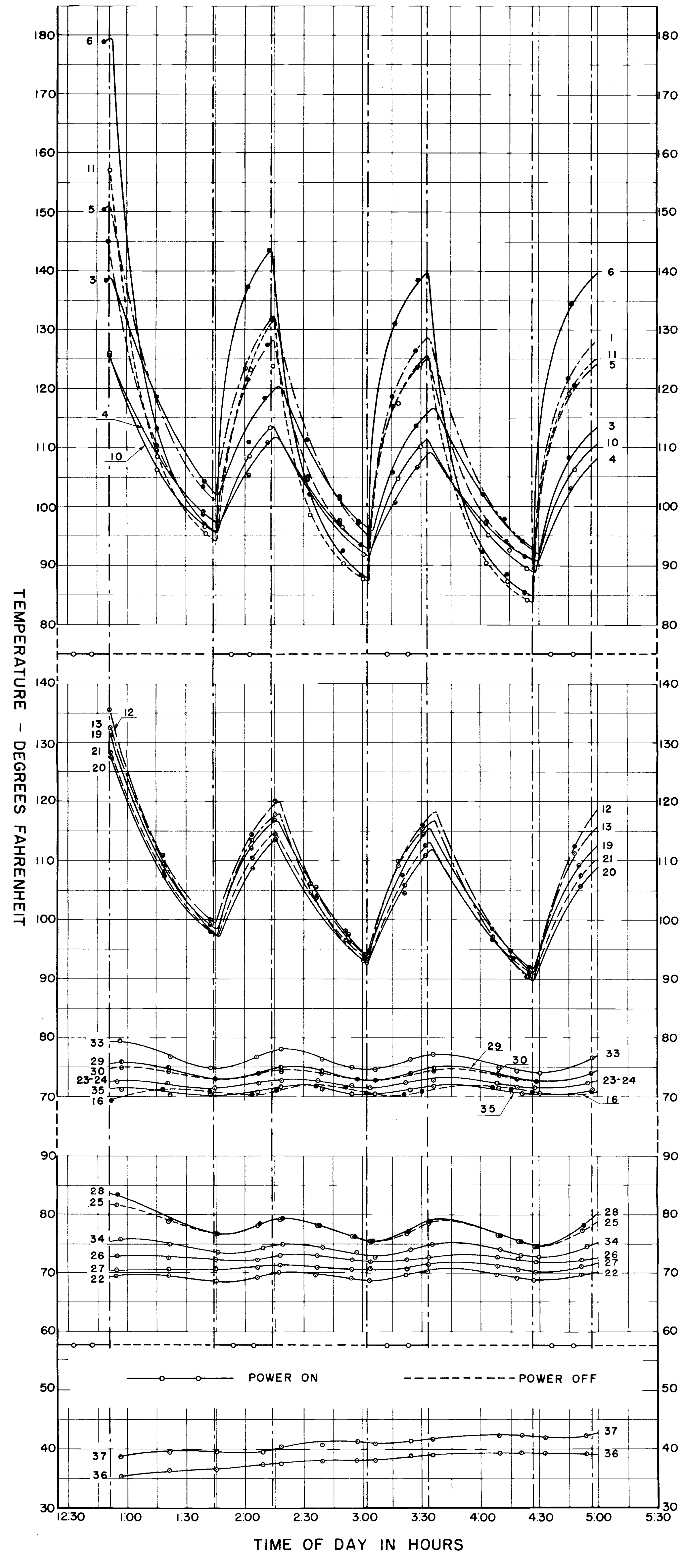


TEMPERATURE VARIATIONS

RUN "N"

BORDER PANELS OPERATING – 177 VOLTS A.C. FAN OFF

THERMOSTAT A SET AT 70 F

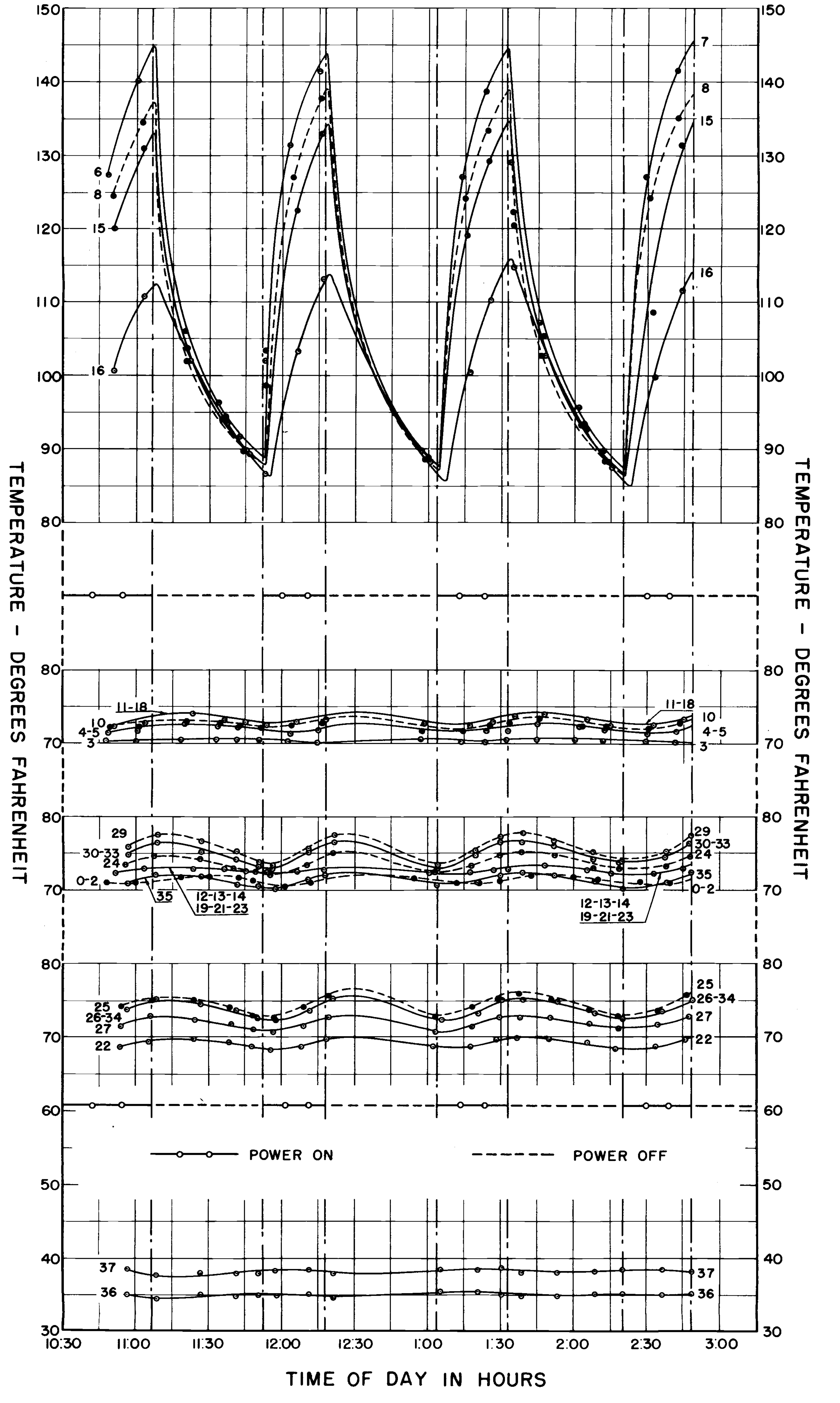


TEMPERATURE VARIATIONS

RUN "Q"

CENTER PANELS OPERATING — 157 VOLTS A.C. FAN OFF

THERMOSTAT A SET AT 70 F

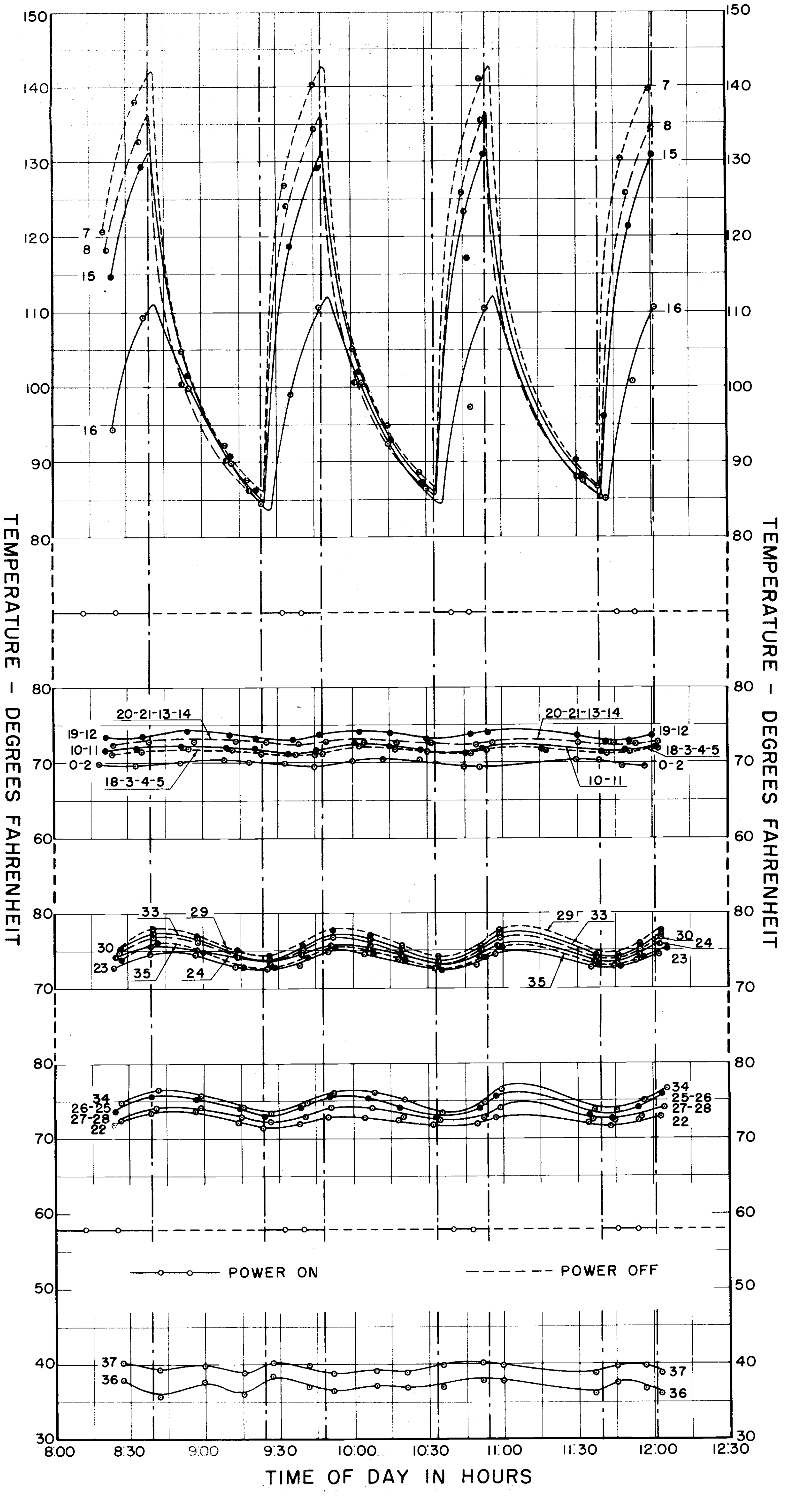


TEMPERATURE VARIATIONS

RUN "Q"

CENTER PANELS OPERATING 157 VOLTS A.C. FAN ON

THERMOSTAT A SET AT 70 F



PART B. REVERSE-CYCLE HEATING

I. INTRODUCTION: PURPOSE AND SCOPE OF REVERSE-CYCLE PROJECT

As with the research on radiant heating, dealt with in the first section of this report, the complete project was to include an investigation of the possibilities of reverse-cycle heating, particularly for house heating. This investigation could have been a theoretical one, but it was felt that more could be accomplished by actually operating a reverse-cycle installation. As is true of radiant panel heating, there has also been much development during recent years in the field of reverse-cycle heating. And, as with panel heating, there are still a number of "unknowns" in this field.

The investigation here reported was carried out primarily for the purposes of determining, if possible, in the time permitted, some of these "unknowns" and making available practical test data to those interested.

II. A REVERSE-CYCLE INSTALLATION

1. **Description of test plant.** Figure 9 illustrates schematically a reverse-cycle installation such as was used in this investigation. The ground coil consisted of 150 feet of $\frac{3}{4}$-inch copper tubing. This tubing was buried 30 inches below the ground surface, in a coil of irregular shape, and contained within a ground area of about 320 square feet. Every part of the coil was separated from any other part of the coil as far as possible in the space available. However, for a distance of about ten feet, one part of the coil ran parallel to another section of the coil and was only about 16 inches from it, and for a distance of about fifteen feet two sections of coil ran parallel about four feet from each other. The soil in which the coil was buried is principally a rather dense yellow clay.

A two-cylinder compressor having a bore of $1\frac{1}{2}$ inches, a stroke of $1\frac{3}{8}$ inches, and a displacement of 4.86 cu in. was used in this installation. This compressor operated at about 490 rpm. The motor used for driving the compressor was rated at $\frac{3}{4}$ hp. It was a single phase, 60 cycle, 115/230 volt, ac motor designed to operate at 1,725 rpm. The motor was operated at 110 volts and the power input to the motor was obtained by means of a recording wattmeter and a recording voltmeter, as well as an integrating watt-hour meter.

The condenser used was of shell-and-tube design in common use commercially in which the water circulates inside of finned tubes enclosed within a tank. The refrigerant is condensed on the outside

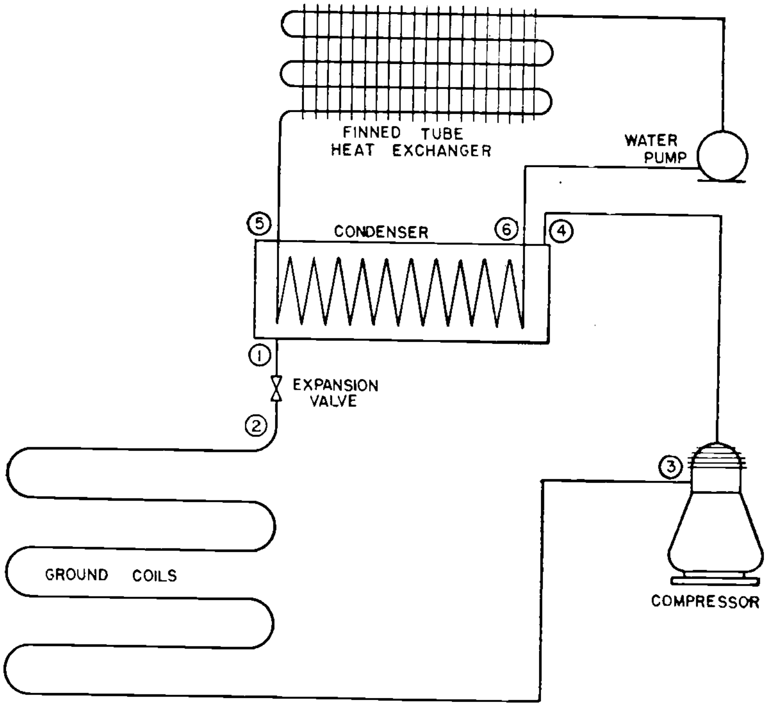


Figure 9. SCHEMATIC DIAGRAM FOR REVERSE-CYCLE HEATING UNIT.

of the tubes and collects in the bottom of the tank. No receiver tank was used, since the condenser served as a receiver as well as a condenser.

The water was circulated in a closed system by means of a centrifugal pump. This water, which was heated in the condenser, was cooled in a finned tube heat exchanger which was placed outside the laboratory. Air was circulated across the heat exchanger tubes by means of two fans mounted as an integral part of the heat exchanger. Between the pump and this heat exchanger was a water flow meter of the rotating vane type.

2. **Thermometer and pressure gage location.** Mercury-glass thermometers were placed in thermometer wells at the Points 1 and 3 in Figure 9 and at Points 1′, 1″, 3′, and 3″ in Figure 10. Pressure gages were located at Points 1, 2, and 3 in Figure 9 and at Points 1′, 2′, 3′, and 3″ in Figure 10. All gages were calibrated before the tests were begun.

Temperatures at Points 5 and 6 were obtained by recording thermometers of the extended bulb type, the bulbs being placed in wells at Points 4 and 5.

3. **Refrigerant and heat exchangers.** The entire system contained about thirteen pounds of the refrigerant, Freon 12. Figure 10 illustrates a schematic diagram for a reverse-cycle unit which is the same as that shown in Figure 9 except for the heat exchanger which has been added to this unit. As is indicated in the illustration, this heat exchanger is placed in the refrigerant circuit in order to subcool the liquid refrigerant on the high side and to superheat the cold gas on the low side. A more detailed discussion of the functioning of this exchanger will be presented later.

This subcooling superheating heat exchanger was of the Mickelaus type in which the liquid Freon, in this case, enters the exchanger through a header at one end, flows through tubes to the end of those tubes, then returns through tubes which surround, individually, the

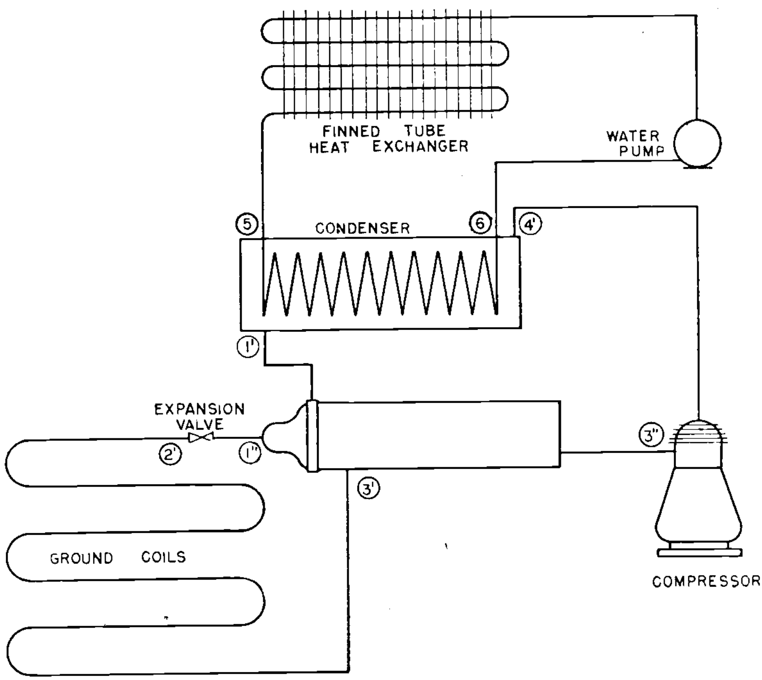


Figure 10. SCHEMATIC DIAGRAM WITH SUPERHEATING, SUBCOOLING HEAT EXCHANGER FOR REVERSE-CYCLE HEATING UNIT.

first set of tubes, and passes on out of the exchanger. The Freon gas enters the exchanger at the top and toward one end, and by means of baffles makes several passes across the bundle of tubes through which the liquid Freon is flowing, and then passes out of the exchanger at the bottom and at the end opposite to that at which it enters. Since the average temperature of the low pressure refrigerant in the exchanger was very near the surrounding room temperature, it was considered unnecessary to insulate this heat exchanger.

The condenser was thoroughly insulated, as were the tubing from the compressor to the condenser and the thermometer wells.

All other components of the unit when operating both with and without the heat exchanger were exactly the same except for the expansion valve. When operating without the heat exchanger a thermostatic expansion valve was used. When operating with the heat exchanger an automatic expansion valve was used. All data are comparable, however, since the same pressure on the low side was maintained in both cases.

4. **Ground thermocouples.** Nine thermocouples were placed in the ground at or near the ground coil for the purpose of obtaining ground temperatures at various locations with respect to the coil. All couples were of 24-gage copper-constantan and were well insulated, electrically, with a tar compound. These couples were about fifteen feet from the expansion valve.

Couple 1 was 24 inches, horizontally, from the coil. Couple 2 was 12 inches, horizontally, from the coil on the same side as Couple 1. Couple 6 was located similarly to Couple 2, but on the other side of the coil, while Couple 5 was located similarly to Couple 1, but on the same side of the coil as Couple 6. Couple 4 was 24 inches directly above the coil and about 6 inches from the ground surface. Couple 3 was 12 inches directly above the coil; Couple 8 was 12 inches directly below the coil, while Couple 7 was 24 inches directly below the coil. Couple 9 was on the coil surface, and was held in place by friction tape.

These couples, located as they were, provided the means for obtaining the temperature gradient about the coil for a radius of 2 feet.

III. OPERATING PROCEDURE

1. **Water control.** With the compressor operating, the amount of water circulated through the condenser and the outside heat exchanger was regulated by means of a valve. The regulation of the quantity of water circulated was necessary in order to keep the quantity as constant as possible, once the proper quantity was determined.

The proper quantity was the amount of water required to keep the temperature of the water at the inlet and outlet of the heat exchanger between limits fixed by the recording thermometers, and to maintain the pressure of the refrigerant in the condenser within desired limits.

2. **Heat exchanger fans.** Inasmuch as the heat exchanger in the water circuit was placed on the outside of the laboratory, the amount of heat transferred in this exchanger depended on the temperature of the outside air. This made it somewhat difficult to control not only the temperature of the water both in and out of the exchanger, but consequently the pressure of the refrigerant on the high side also. This difficulty was overcome to some extent by placing two sizes of fans in the heat exchanger. When the temperature of the outside air was high, both fans were operated; when very low, only the small fan was run; when the temperature was moderate, only the larger fan was used.

All data obtained from the gages, meters, thermometers, and thermocouples were recorded at various intervals, usually every twelve hours, although every twenty-four hours was found to be sufficient when conditions were steady.

IV. DATA AND RESULTS

1. **Operating log and results.** Starting on March 7, 1948, the reverse-cycle unit was operated continuously for a period of 44 days or a total of 1,054 hours without the heat exchanger in the refrigerant circuit. During this period the pressure of the refrigerant on the low side varied between $23\frac{1}{2}$ psig to 25 psig, the weighted average being 23.95 psig. This pressure was obtained from the gages at Points 1 and 3, Figure 9. The average temperature at Point 3 was 58.1 F; this temperature varied from 56 F to 63 F.

The weighted average pressure of the refrigerant on the high side, at Point 4, was 99.7 psig; this pressure varied from 85 psig to 110 psig. The average temperature of the refrigerant on the high side, Point 4, was 119.8 F; this temperature varied from 115 F to 128 F. The average temperature of the refrigerant leaving the condenser, Point 1, was 83.9 F; this temperature varied from 77 F to 92 F.

The temperature of the water leaving the condenser, Point 6, varied from a low of 72 F to a high of 109 F; the weighted average temperature at this point was 87.2 F. The weighted average temperature of the water entering the condenser, Point 5, was 66.5 F; this temperature varied from 49 F to 95 F. The weighted average temperature difference between these two points, 5 and 6, was 20.7 F.

The average water flow through the condenser and the heat exchanger was 24.4 gph.

During the test period of 1,054 hours, a total of 419.3 kw-hr of electricity was used in driving the compressor. This is the equivalent of 1,430,000 Btu. During the same time, a total of 4,450,000 Btu was obtained by the water in the condenser and dissipated by means of the heat exchanger on the outside of the laboratory. Of these 4,450,000 Btu, 3,840,000 Btu have been calculated as having entered the system from the ground and through the ground coil, and 610,000 Btu through the compressor. The bases for these calculations are indicated in the following discussion of the performance of the system.

The amount of heat obtained in the condenser was arrived at as follows. For each twelve or twenty-four hour period, the difference in degrees between the average temperature of the water leaving the condenser and the average temperature of the water entering the condenser was multiplied by the number of gallons per hour circulated during the period. This product was then multiplied by 8.35, the number of pounds of water per gallon, and then by either 12 or 24, depending on the length of the time period for which the calculations were being made. The final product of these multiplications was then the total number of Btu obtained from the condenser during that period.

It should be noted here that the method of arriving at the quantity of heat obtained from the system did not credit the system with any recovery of heat lost from the motor, compressor, or fans.

As already indicated, the temperatures of the water entering and leaving the condenser were obtained by means of recording thermometers. To obtain the average temperature from a thermometer chart for a twenty-four hour period, a radial planimeter was used, the chart having substantially equal divisions over the range involved.

2. **Coefficient of performance.** The COP, or coefficient of performance, of a reverse-cycle heating unit is the ratio of the heat or energy obtained from the system to that which was put into the system. The COP of this unit, for the total period operated, was 4,450,000 divided by 1,430,000, or 3.12. This means, of course, that 3.12 times as much heat, or energy in the form of heat, was obtained from the system as was put into the system in the form of electricity to drive the compressor.

According to the manufacturers' data, the motor driving the compressor was estimated as operating at about 57 per cent efficiency and the compressor at about 75 per cent efficiency. Consequently, 610,000 Btu of the total 1,430,000 Btu furnished to the motor in the form of electricity went into work compressing the refrigerant; the remaining 57 per cent or 820,000 Btu was lost. In a well-designed unit a large part of these 820,000 Btu could be recovered by sealing the motor and compressor together in an enclosing jacket. Most of the heat given off by the motor and compressor could thus be used to heat the refrigerant and could be recovered in the condenser. Assuming that 90 per cent of the heat lost by the motor and compressor could be recovered, allowing 10 per cent to be lost by radiation and convection, 738,000 Btu of this heat loss could then have been recovered and a total of 5,188,000 Btu obtained in the system. This would give a COP of 3.62 for the 1,054 hour run. It should be noted here that only the electricity used for driving the compressor has so far been considered.

If the motor, compressor, and pump and fans, if used, were located in the space to be heated then, of course, all of the energy supplied to the system could be recovered in the form of heat which would be used in warming the heated space. All of the energy supplied to the system plus that which the system obtained from the heat source could be utilized.

The electrical energy supplied to the fans in the outside heat exchanger and the motor driving the circulating pump were not metered during the operation of the system. However, after all the other data had been obtained, these fans and pump were operated continuously for a period of 36 hours, during which time 7.85 kw-hr of electricity were supplied to them. Of this energy total one half was supplied to the pump, the other half to the fans. On this basis about 230 kw-hr would have been used in 1,054 hours. This is the equivalent of 2,215,000 Btu, giving a COP for the entire system of 2.05. On the basis of recovering 90 per cent of the heat lost by the compressor-motor, an overall COP of 2.45 could have been obtained for this period.

It is quite apparent that the amount of electrical energy supplied to the fans and pump as compared to that supplied to the compressor-motor is very much greater than it would be in a typical reverse-cycle heating system. While the compressor-motor was of only $\frac{3}{4}$ hp, the water pump and heat exchanger were of large enough capacity to be used with a much larger compressor. The amount of electricity supplied to the fan and pump in the experimental unit was more than half of that supplied to the compressor-motor; in a typical installation this proportion would be more like one-tenth to one-fifteenth. Thus the COP of 2.05 arrived at above is considerably on the conservative side since it includes an allocation of entirely too large a proportion of energy to fans and pump.

Unquestionably, in an actual house heating system of this kind, all of the energy supplied to the fans in the heat exchanger would be recovered, assuming a forced warm air system were used. Assuming this to be true and that say 40 per cent of the energy supplied to the pump and motor would be recovered, it is reasonable to assume that 70 per cent of the total energy supplied to the fans and pump would be recovered in the system. On this basis, a total of 649.3 kw-hr of electrical energy, or 2,215,000 Btu would be supplied to the entire system during the 1,054 hours, and 5,738,000 Btu would be recovered, giving an overall COP of 2.59 for the entire period, assuming as before that 90 per cent of the compressor-motor heat loss were recovered. An even better coefficient of performance than this might well be expected from a well designed, properly installed reverse-cycle system, as already indicated.

3. **Cycles plotted on Mollier diagram.** Figure 11 is a Mollier diagram or a pressure-enthalpy chart for Freon 12 on which the diagrams for two cycles have been plotted, one cycle without the heat exchanger in the refrigerant circuit, and one with. The Points 1, 2, 3, and 4 are for the cycle without the heat exchanger, and correspond to the points of the same numbers in Figure 9. These points were plotted from the actual data obtained during the 1,054 hour test period. The average pressure during this period at Point 1, which is immediately after the refrigerant leaves the condenser, was 99.7 psig or 114.4 pounds per square inch absolute. The average temperature at this point was 83.9 F. The averages at the other points were as follows: Point 2, 23.95 psig or 38.7 psia; Point 3, 73.95 psig or 38.7 psia, 58.0 F; Point 4, 99.7 psig or 114.4 psia, 119.8 F.

Since the satisfactory performance of a heat pump depends to some extent on the pressure drop in the evaporator, or ground coil (and in other parts of the system, as well), it is of interest to note that during the course of the tests no appreciable difference between the pressure at Point 2, immediately after the expansion valve, and at Point 3, at the compressor, was recorded at any time. It is not unreasonable to assume that there actually was some pressure drop in the refrigerant in the ground coil. Assuming the gages to be accurate to within $\frac{1}{2}$ psi there could have been, by gage readings, a drop of 1 psi in this coil. However, since the cycles plotted in Figure 11 were from actual data, no pressure drop is indicated between Points 2 and 3 on the diagrams.

Considering a cycle as plotted on the diagram, the starting point may be taken anywhere on the cycle, say at Point 1. At this point the refrigerant has just left the condenser. It then passes through the expansion valve and enters the ground coil, which acts as the

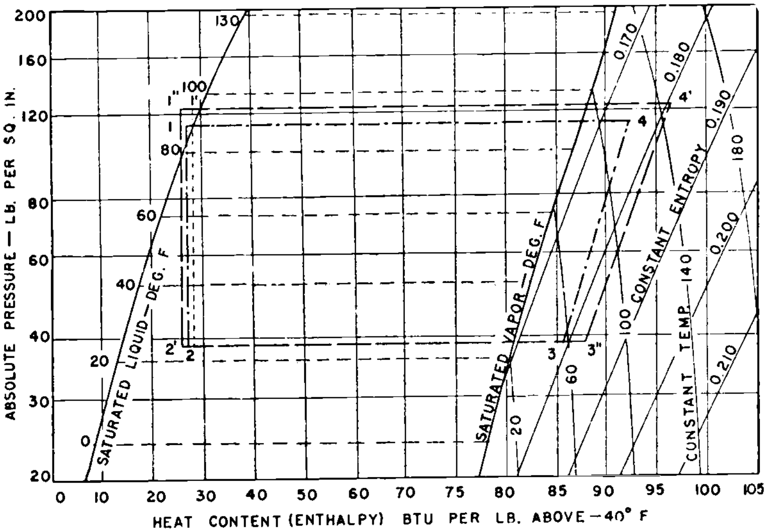


Figure 11. MOLLIER DIAGRAM FOR DICHLORODIFLUOROMETHANE (FREON).

evaporator in this case, entering the ground coil at conditions indicated by Point 2. In the ground coil the refrigerant picks up heat from the ground. This heat evaporates the liquid refrigerant at a constant pressure until it is completely evaporated or even superheated, as indicated at Point 3, at which conditions it enters the compressor and is compressed to Point 4. From Point 4 to Point 1 the high pressure vapor is condensed in the condenser, arriving back at the expansion valve at conditions indicated at Point 1.

4. **Theoretical COP.** The theoretical COP can now be calculated from the diagram of Figure 11. The difference in enthalpy between Point 4 and Point 1 represents the heat removed from the refrigerant in the condenser. The difference in the enthalpy between Points 2 and 3 represents the heat taken from the ground by the refrigerant as it passes through the ground coil. The difference in enthalpy between Points 3 and 4 represents the energy required to compress the refrigerant in the compressor.

Since the COP is the ratio of the amount of heat removed from the system to the amount put into the system, the COP is now $(h_4 - h_1)/(h_4 - h_3)$, where h with a subscript represents the enthalpy per pound of refrigerant at the point indicated by the subscript.

Thus, from the chart, Figure 11, $h_4 = 92.5$B, $h_1 = 27.0$B, $h_3 = 86.0$B. The COP is then $(92.5 - 27)/(92.5 - 86) = 10.08$. This is, of course, the theoretical COP for the cycle. Assuming, again, that the efficiency of the motor-compressor is 42.75 per cent, or 0.57×0.75, we must now divide the quantity 6.5, or $h_4 - h_3$ by 0.4275, obtaining 15.2 Btu. This is now theoretically the amount of energy per pound actually supplied to the motor-compressor. Thus the theoretical actual COP becomes 65.5 divided by 15.2 or 4.3. This corresponds to the actual COP of 3.12 obtained from the operating data.

5. **Operation with heat exchanger.** At the end of the 44-day period during which the system was operated without the heat exchanger, the system was shut down and the heat exchanger was placed in the refrigerant circuit. The system was operated for several days, but with somewhat unsatisfactory results because oil from the compressor accumulated in the heat exchanger. The system was then purged and the oil drawn from the heat exchanger. It was interesting to note that until the accumulation of oil in this piece of equipment became too great, a difference of 1 degree was obtained between the temperature of the liquid refrigerant leaving the exchanger and the refrigerant vapor leaving the exchanger, as indicated by the thermometers in wells at Points 1″ and 3″, Figure 10, respectively.

The system was then operated for a period of eleven days, or 264.5 hours. The average conditions during this period were as follows: Point 1′, 89.0 F, 110.7 psig; Point 1″, 76.9 F; Point 2′, 24.14 psig; Point 3′, 24.14 psig, 58.1 F; Point 3″, 73.6 F; Point 4′, 110.7 psig, 146.9 F.

The weighted average temperature of the water entering the condenser, Point 5, was 72.2 F; at Point 6 it was 91.2 F, making an average difference of 19 degrees. The weighted average flow of water through the condenser was 24.6 gph. For the entire 264.5-hour period that the system was operated, a total of 1,033,000 Btu were obtained as useful heat from the condenser. During this same period a total of 120.1 kw-hr was used to drive the compressor.

The actual COP can now be calculated as before. The Btu equivalent of the kw-hr of electrical energy supplied to the compressor was 410,000. The actual COP was 1,033,000 divided by 410,000 or 2.52. This is the actual COP neglecting the energy supplied to the water pump and the fans in the heat exchanger in the water circuit. The overall COP considering this power could be calculated as before.

This cycle is charted in Figure 11 with the primed numbers, as 1′, 2′, 3′, et cetera. Theoretically the heat exchanger should increase the coefficient of performance of the cycle. The object is, of course, to subcool the high pressure refrigerant, moving Point 1 farther to the left, to Point 1″, on the chart; assuming expansion without change of enthalpy in the expansion valve, this also moves Point 2 farther to the left, to Point 2′. By superheating the low pressure refrigerant in the heat exchanger, Point 3 is moved farther to the right, to Point 3″. Point 4 would then be moved farther to the right, to Point 4′. This means, then, that the distance between Points 4′ and 1′ is greater than the distance between Points 4 and 1. Since this length is a measure of the heat given up in the condenser, it is apparent that with this distance being greater, the amount of heat obtained from the system is greater than without the added heat exchanger. But the difference in enthalpy between Points 3″ and 4′ has also increased. In other words, while the amount of heat obtained from the system has increased, the amount of energy required to obtain this heat has also increased. But, theoretically, the increase in work put into the system should not be as great in proportion as the increase in heat obtained from the system; hence, the COP should increase. Actually, as will be seen, this was not experienced in our case.

The COP with heat exchanger is now expressed by the fraction $(h_4'—h_1'')/(h_4'—h_3'')$ or $(96.6—25.4)/(96.6—88.3)=8.58$. To get the theoretical actual COP, however, the difference $h'_4—h'_3$ must be divided by the combined efficiency of the motor-compressor. Assuming a combined motor-compressor efficiency of 42.75 per cent as before gives a COP of 3.67. It is apparent that the COP with the heat exchanger was not as high as it was without it. Since the COP of a reverse-cycle system will decrease as the difference between the pressures on the low and high sides increases, the decrease in COP with the heat exchanger was at least partly due to the fact that the high side pressure was higher with the heat exchanger than it was without it. This can be shown by assuming that the high side pressure with the heat exchanger is the same as without it and that the compression line 3″-4′ follows the same path up to the 114.4 psia line. The refrigerant, on reaching the 114.4 psia line would then have reached a temperature of 142 F and its enthalpy at these conditions would be 96.2 Btu per pound. The COP would then be $(96.1—25.4)/(96.1—88.3)=9.06$. By maintaining the same pressure on the high side, the theoretical COP would be increased from 8.58 to 9.06.

As can be seen on the diagram of Figure 11, there was a second factor which appreciably influenced the coefficient of performance of the system to make this coefficient lower. This was the fact that the compression process was not as efficient with the heat exchanger as it was without it; consequently proportionately more power was required for compression and the COP was lowered.

In actual practice, the practical purpose of using a heat exchanger is not necessarily to increase the COP but to increase the capacity of the evaporator. By subcooling the liquid refrigerant before it passes through the expansion valve, the amount of so-called flash gas produced at the valve is reduced and the quality of the vapor as it enters the evaporator is somewhat lower. The reduced quality of the vapor entering the evaporator may increase the heat transfer characteristics of the evaporator to some extent by making the surface of the tube wetter. The decrease in the amount of flash gas passing through the valve also is of advantage in reducing the erosion of valve parts.

The superheating of the low pressure vapor in the heat exchanger also tends to increase the capacity of the ground coil. Without the heat exchanger, a part of the evaporator, or ground coil, would be used to superheat the vapor before it enters the compressor. Inasmuch as this required superheating is done in the heat exchanger, that part of the evaporator coil which would be used for superheating could be used for evaporating, thus increasing the capacity of the evaporator coil. Except for very special circumstances, it is doubtful if the expense of providing a heat exchanger in the refrigerant circuit is justified in a reverse-cycle system of this kind.

6. **Ground temperatures.** Of some interest is the effect on the ground temperature due to the withdrawing of heat from it by means of the ground coil. Obviously the temperature at different depths in the ground will vary, depending on the temperatures of the atmosphere, amount of radiation from the sun, amount of rainfall, et cetera. No attempt was made to measure or determine these various factors, except the outside air temperatures. During the 1,054-hour period in which the system was operating without the heat exchanger, the outdoor temperature varied from 31 F to 71 F; the average temperature during this period was 44.3 F.

The average temperatures at Thermocouples 1 to 9 for the three days prior to starting the reverse-cycle system were: Couple 1, 44 F; Couple 2, 44 F; Couple 3, 43 F; Couple 4, 42 F; Couple 5, 44 F; Couple 6, 44 F; Couple 7, 46 F; Couple 8, 45 F; and Couple 9, 44 F.

As has been indicated, the temperatures at these thermocouples varied to some extent throughout the test period. The temperature at Couple 9, for example, varied from 22 F to 25 F; the temperature at Couple 4 varied from 37 to 48 F. The temperatures at the couples at the end of the test period were: Couple 1, 42 F; Couple 2, 40 F; Couple 3, 41 F; Couple 4, 44 F; Couple 5, 41 F; Couple 6, 39 F; Couple 7, 41 F; Couple 8, 41 F; and Couple 9, 25 F.

Before the tests began, there was a gradient between Couple 4 and Couple 7 of 4 degrees, from 42 F to 46 F, the temperature increasing at a rate of 1 degree per foot of depth. At the end of the test period, the gradient was from 44 F at Couple 4 to 41 F at Couple 7. Because of the variations in ground temperatures from natural causes, this reversed gradient at the end of the test period cannot be attributed entirely to the effects of the ground coil, however.

Two days after the system had been turned off, the temperatures at the couples were as follows: Couple 1, 44 F; Couple 2, 43 F; Couple 3, 43 F; Couple 4, 44 F; Couple 5, 42 F; Couple 6, 42 F; Couple 7, 42 F; Couple 8, 42 F; and Couple 9, 42 F. These data indicate the rapidity with which the soil surrounding the ground coil reached almost the same temperature at the various levels that were recorded before the tests were begun.

There was no indication, judging from these data, that there was any long-time frozen condition in the soil around the ground coil. During the course of the tests and while the reverse-cycle unit was operating there was definite evidence that freezing in the ground was taking place. Where the ground coil entered the ground it was surrounded by ice of about 1½ to 2 inches in thickness. The location of the ground coil was easily determined for most of its length by a noticeable swelling and rising of the ground above the coil. Soon after the system was turned off, this rise above the coil changed to a depression of from 2 to 3 inches in depth. Quite probably if the coil had been placed at a greater depth in the ground these effects would not have been so noticeable.

During the 1,054-hour test period, approximately 3,640 Btu per hour were withdrawn from the ground. Yet, as indicated by the readings of the thermocouples in the ground, the temperature of the soil at a radius of two feet from the coil was not appreciably affected. And, as already indicated, there was certainly no long-time effect on the soil in evidence.

V. CONCLUSIONS

1. **Heat sources discussed.** Before coming to conclusions with regard to the performance of the experimental installation or the probable performance of a typical reverse-cycle heating system using a ground coil, some consideration should be given to reverse-cycle systems using other sources of heat. Most of these sources have both advantages and disadvantages or some limitations as compared to the earth as a heat source. Likewise, the use of the earth as a heat source has its limitations.

There are three heat sources commonly considered for use with a reverse-cycle installation; these are the earth, the outside air, and water.

The principal advantages of the earth as a heat source are its universal, even if not always practical, availability and the fact that it is a relatively stable source of heat which is not appreciably affected by sudden changes in local weather conditions.

The means by which the heat may be removed from the earth might take any one of or a combination of several of many arrangements of tubes, coils, tanks, et cetera. Many schemes have been tried; some have been found to be better than others. There are as yet, however, not enough dependable data available to justify any definite conclusions as to which method is the best. In fact, the best method for one locality may not be the best for another. The principal limitations of the earth as a heat source seem to be, for the present at least, the amount of ground space available, the type of soil encountered, the specific heat, the density, the moisture content of the soil, and, of course, the natural temperatures of the ground at various depths, et cetera. The cost of installing the ground coil in whatever form it may take may of course be a predominating or even a governing factor.

Generally speaking, water is an excellent source of heat for a system of this kind. The water may be that of a well, a lake or stream, or city water. Here the limitations are obvious. Well water is not available or not economically practicable to obtain in many cases; lakes or streams are even more unavailable in most cases. The amount of water required would probably prohibit the use of city water for this purpose in most cases, not only from the standpoint of cost to the user but from the standpoint of willingness or ability to furnish it on the part of the utility company.

The principal advantages of water, when available, are its almost constant temperature, within limits of only a few degrees, and the fact that the water would probably be at a fairly high temperature, relatively. Well water, for example, might be at approximately 55 F to 70 F; lake and stream and city water might be within this same temperature range.

The relatively high and constant temperature level of available water sources has two particular advantages. One advantage is that the relatively high temperature makes possible the maintaining of a high pressure on the suction side of the compressor; the fact that the water temperature is relatively uniform makes possible the maintaining of the same degree of constancy in the low side pressure of the refrigerant. The maintaining of a high pressure on the suction side of course is a factor in increasing the COP of the system. The maintaining of this higher pressure at a constant or uniform level increases the ease of operation of the system by eliminating the necessity of making what otherwise might be frequent changes in the control settings.

Another advantage is the wide range of heat exchangers which are available for this purpose. Heat exchangers suitable for this specific purpose have been developed and widely used for other similar purposes for some time and consequently are highly efficient and available in a wide range of capacities.

Because of its universal availability, air might seem on first thought to be an ideal source of heat for reverse-cycle systems. While air is used for this purpose to some extent, its use is, for the present at least, limited to only certain localities where the outdoor air temperature varies between more or less relatively narrow limits. Two principal factors limit the use of air as a heat source. One is the temperature of the air; the other is the humidity.

It is quite obvious that as the temperature of the air decreases, with a resultant decrease in the difference between the temperature of the air and the temperature of the refrigerant in the evaporator, the amount of heat available from the air will decrease. This means, of course, that as the outdoor temperature drops and the house or space being heated requires more heat to keep its temperature at a fixed level, the amount of heat available from the air decreases. Not only does the amount of heat available from the air decrease as the temperature of the air decreases, but also the temperature of the refrigerant in the evaporator must decrease if a fixed temperature differential is to be maintained. Everything else remaining the same, a lowering of the refrigerant temperature decreases the COP of the system.

Another factor which has limited the use of air as a heat source is the accumulation of ice or frost on the evaporator or heat exchanger coils by condensation and freezing of the moisture contained in the air. Such frosting of the coils may impair the efficiency of the coils considerably. While the removal of this frost from the coils can be accomplished by one of several methods, or by a combination of such methods, such removal may be both expensive and inconvenient.

2. **Practical application of experimental results.** In the following remarks in which some comparison is made between the experimental installation and any assumed house heating reverse-cycle system, the discussion will be confined to a system using a ground coil, or to one which uses the earth as its heat source.

In predicting the performance of a reverse-cycle installation for heating an industrial building or dwelling or in comparing such an installation with the systems used in this project, a number of factors or differences which would influence such prediction or comparison should be considered. Some of these factors will be mentioned or discussed here.

First, the average temperature of the warm water obtained in the experimental system is probably too low for a practical heating system. While water temperatures required in radiant heating systems are generally considerably lower than those required in competing systems (which fact alone recommends the use of a radiant heating system in conjunction with a reverse-cycle system) even radiant heating systems would in general require higher water temperatures than were obtained in the experimental unit. If warmer water were required, this would necessitate a higher pressure on the high side of the compressor which in turn would tend to lower the COP of the system.

In order to maintain a given panel temperature of an embedded coil radiant panel it is sometimes possible to do so by increasing the amount of coil in the panel and decreasing the water temperature in the coil. This possibility should not be overlooked when a reverse-cycle system is contemplated as the heat source in a panel heating system in view of the fact that the water temperature is such a factor in determining the COP of the reverse-cycle system.

In the design of any such system, then, this factor should be considered and if it is possible to lower the temperature of the water in the radiant panel system and still maintain the required panel temperature, the cost of increasing the length of coil in the panel should be weighed against the savings in operating costs resulting from the lowering of the water temperature with its consequent lowering of the pressure on the high side of the compressor. The increased cost of pumping the larger volume of lower temperature water through the system should also be considered in an analysis of this kind; both these would probably be very minor factors in the overall design calculations.

Second, in a typical house-heating reverse-cycle system, which would require a much larger heating capacity than the experimental unit, the quantity of refrigerant passing through the ground coil (as well as the rest of the system) would be very much greater. Not only would this quantity be larger, but the length of coil through which it would have to pass would be longer. Both of these conditions would result in a larger pressure drop in the ground coil, as well as other parts of the system, all of which would tend to reduce the efficiency of the system with a resultant lowering of the COP of the system.

Third, the larger size motor and compressor required in a reverse-cycle house-heating system would definitely be an influencing factor in favor of this system. While the two factors discussed above might tend to lower the COP of the house-heating system as compared with the experimental unit, the increased size of the motor and compressor in the house-heating system would raise its COP.

While the motor used in the experimental unit was of only $\frac{3}{4}$ hp and was not loaded to anywhere near its rated capacity, a motor used in a house-heating reverse-cycle system would quite probably be of at least 3 hp and would normally operate at full load capacity. Since the efficiency of electric motors increases, generally, with size and also increases as load approaches rated capacity, the efficiency of the motor in the house-heating system would be considerably higher than that in the experimental unit.

While the heat lost from both the motor and compressor in either system could in a large part be recovered, the heat so recovered would not be as effective in a system of this kind as the energy that goes into the actual compression of the refrigerant. In other words, the heat recovered from an electric motor, for example, could be used and would be charged for on the same basis as the energy obtained in a direct electric heating system; it has not increased in value within the system. On the other hand, for every unit of energy which goes into compressing the refrigerant, approximately 3 or more units of heat or energy can be obtained from the system for heating purposes. Consequently, the system which utilizes the largest proportion of energy supplied to the system in compressing the refrigerant will have the highest COP. In this respect, then, the house-heating system with the larger, more efficiently operating motor should have a considerably better coefficient of performance than the smaller experimental unit.

It should be mentioned that a larger motor and compressor would have been used in the experimental installation in order to obtain data which would have been more comparable to a typical house-heating unit, but the very limited ground space available made it impossible to lay more ground coil than that which was used in the experimental unit. This condition precluded the use of a larger motor-compressor unit.

The short length of ground coil also restricted considerably the loading of the compressor motor. According to the test data, this motor was loaded to only approximately one-third of its rated full load capacity, or, say, approximately the equivalent of $\frac{1}{4}$ hp. On this basis about 600 feet of ground coil should be provided for each horsepower of compressor motor, assuming that the compressor is to be operated between the same pressure limits as was the experimental unit.

However, as has already been mentioned, most reverse-cycle heating systems would probably require a higher pressure on the high side of the compressor than was maintained in the experimental unit. Increasing this pressure would decrease the amount of refrigerant circulated, assuming the same power input to the motor, and this would, consequently, decrease the amount of ground coil needed. In view of this, probably 500 feet of ground coil would be sufficient for each horsepower of compressor motor.

The length of ground coil required in any case would depend on a number of factors. Among these factors are the operating pressures in the systems, the depth to which the coil is buried, the type of soil in which it is buried, the moisture content of the soil, its thermal conductivity, the climate in which the system is operated, et cetera. The exact extent to which these various factors influence the design of a reverse-cycle installation is as yet unknown. Consequently, no fixed rules for such designs can be made at this time.

With a unit having a much greater capacity than the experimental unit, a much greater freezing effect in the ground may be experienced than was found with the test unit, particularly since most heating systems would be operated for a longer period of time than was the test unit. In view of the limited amount of data obtained pertaining to this phase of the study, no definite predictions can be made in this connection. However, if both ground coil and ground space are provided in proportion to the heating demands of a system, compared to the test unit, the test data indicate that there is little probability of any damaging effects resulting from the freezing of the ground. Here, again, the results in any case would depend on the specific conditions at the installation.

The test data indicate that for the locality in which the experimental installation was operated, at least, the ground is a satisfactory source of heat for a reverse-cycle unit and, properly designed, such a unit should give a coefficient of performance of at least 3. Whether or not the operating costs of such a unit would be economically feasible would depend on the relative costs of the required electrical energy and the fuels available in the locality.

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