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Performance of Hot Water Panel Heating Systems

by

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ENGINEERING EXPERIMENT STATION BULLETIN NO. 453
ABSTRACT

This Bulletin presents the results of tests made on floor panel and ceiling panel heating systems. The systems were installed in the Floor Slab Laboratory at the University of Illinois between the fall of 1951 and spring of 1954. The investigation was a part of a heating and air conditioning research program in the Department of Mechanical Engineering. The program was sponsored by the Institute of Boiler and Radiator Manufacturers.

The tests were designed to simulate current residential heating systems. Ceiling panels consisting of nominal 1/2-in. copper tube embedded in the plaster and floor panels consisting of 3/4-in. black iron pipe laid on 12-in. centers in a 5-in. concrete floor slab were studied. Two inches of concrete covered the coils. There were four test rooms, all of the same size. Test Rooms A, B, and C were exposed only to the north, while Room D had both a north and east exposure. Rooms A and B had one window each, which had an area equal to 10% of the floor area, while Rooms C and D each had two such windows. Cellular glass insulation, 1-in. in thickness was used under the entire floor of Room A, along the inside edge of the foundation wall from the level of the floor surface to a depth of 18 in. in Room B, and along the exposed edge of the floor slab in Rooms C and D. In Rooms C and D the insulation was extended under the floor slab for a distance of 24 in. from the outside edge as a border.

Heat meters and thermocouples were used to make direct measurements of heat flow rates from the panels as well as the temperatures of the water, ground, air, and unheated surfaces.

The following is a summary of the results obtained for the test conditions investigated:

A. Bare Panels

1. The air temperatures at the center of the rooms were very uniform with a variation between the floor and 60-in. level of 0.5 F. The temperatures of the air 3 in. below the ceiling and 3 in. above the floor were practically the same.

2. While the average floor surface temperature was uniform across the panel, the heat output from the surface of the panel was much higher near the exposed wall and window than at the center of the room. Thus the floor panel system automatically increases the heat output rate in areas near points of high heat loss.

3. Fuel savings resulting from the use of insulation under the entire floor slab as compared to the use of edge insulation only are too small to justify the additional cost.

4. The measured heat flow from panel to room ranged from 87% of the calculated heat loss exclusive of loss through the floor in Room A to 10% of the calculated above-floor loss in Room C.

5. Vertical insulation along the inside edge of the foundation wall is as effective as L-type edge insulation.

6. At design conditions the loss from the reverse side of panels with edge insulation amounted to 20 to 23% of the total panel output. The reverse loss was roughly twice as great as the estimated heat loss through unheated floor slabs.

7. Calculations based on observed heat losses through floors indicate that at an indoor-outdoor temperature difference of 80 F a floor panel heating system would burn 10% more fuel than a system using radiators or convectors.

8. At design conditions the measured heat transfer from the panel to Room A was essentially the same as the calculated output obtained by substituting actual measured air, panel surface, and average unheated surface temperatures in accepted equations for the radiant and convective heat outputs of floor panels.

9. The apparent thermal resistance of a bare concrete panel of the construction tested was about 1.05 (F per in.) per Btuh (sq ft).

B. Covered Floor Panels

1. Addition of floor coverings to bare floor panels reduced the ability of the system to maintain a constant room air temperature.

2. At design conditions of 80 F indoor-outdoor temperature difference the maximum difference in room air temperature between the levels 3 in. above the floor and 3 in. below the ceiling was 3.5 F.

3. When operating with covered floor panels, the exposed wall surface temperature was about 8 F lower and the average unheated surface temperature (AUST) was about 4 F lower than the room air temperature measured at the center of the room 30 in. above the floor.
4. The carpeting caused an increase in the floor surface temperature along a line toward the center of the room and smoothed out the heat flow profile from the panel to the carpet.

5. Due to heat storage in the ground at design conditions, covering the floor panels with any type of carpeting made pronounced increases in the reverse loss from the panel, the required water temperature, and the boiler size. In mild weather the opposite effects were observed with a net result that carpeting of floor panels caused no appreciable increase in seasonal fuel consumption.

6. Because of the large increase in water temperature required when carpeting is applied over floor panels, it may be impossible to balance floor panel systems in which carpeting is used in some rooms only, unless the piping is arranged to permit zoning with the use of more than one water temperature.

7. At design conditions the measured panel output was from 7 to 18% greater than the calculated panel output. For a given panel minus room-air temperature difference, the panel heat output to rooms which had more severe exposures were 15 to 20% greater than outputs in rooms with less glass area.

8. The thermal resistance of the combinations of carpeting and pads tested ranged from 0.204 (F per in.) per Btuh (sq ft) for the rubber pad alone to 0.96 (F per in.) per Btuh (sq ft) for the heavy carpet and 40 oz jute pad.

9. The thermal resistance of both the asphalt tile and the rubber tile was about 0.05 (F per in.) per Btuh (sq ft).

10. Floor coverings such as asphalt tile or rubber tile which have a thermal resistance of 0.2 (F per in.) per Btuh (sq ft) or less had a negligible effect on the performance of floor panel systems.

11. The relative humidity in the room which had a carpet and pad was consistently greater than the room floored with asphalt tile, indicating that there was probably more water vapor coming through the concrete floor slab in this room.

C. Ceiling Panels

1. While the ceiling panels were able to maintain warm floors over most of the room area, they could not adequately heat the floor along the exposed walls.

2. An indoor air temperature of 72 F could be maintained in Room B at all outdoor temperatures above -10 F. In Room A the system was unable to maintain an air temperature of 72 F at outdoor temperatures below about 25 F. This resulted from a poor bond between plaster and coils.

3. Tests in the room with the coil below the lath (Room B) showed that the water temperature gradually increased with decreasing outdoor temperatures. The average water temperature reached 140 F at an outdoor temperature just under -10 F, which agreed with design assumptions.

4. In the room with the coil above the lath (Room A) the water temperature increased to 140 F when the outdoor temperature was about 25 F.

5. At design conditions the energy input to the ceiling panel in Room B was about 8% higher than that required for the floor panel.

6. The seasonal fuel consumption of the Room B ceiling panel system was 27% higher than that of the floor panel system for the same room.

7. With the attic heated to 60 F, the reverse loss from the ceiling panel in Room B at design conditions was about 39% of the total heat input to the water. With attic temperature uncontrolled, the reverse loss at design conditions was about 48% of the total heat input.
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1. Project History

This is the fourteenth publication prepared under a cooperative agreement between the Institute of Boiler and Radiator Manufacturers and the University of Illinois which was approved January 2, 1940. Under the terms of the agreement the Institute is represented by a Research Committee consisting of engineers active in the heating industry. One of the functions of this committee is to propose problems for investigation that are of the greatest concern to the manufacturers and installers of steam and hot water heating equipment. From these problems the Engineering Experiment Station staff selects for study those which can best be investigated with the facilities and equipment available at the University. The Institute provides funds for a major part of the research work.

In 1951 the Institute of Boiler and Radiator Manufacturers agreed to include a panel heating project in the steam and water heating research program, provided the work would not duplicate that then being done by the American Society of Heating and Air-Conditioning Engineers. While the panel heating research at the University was never officially a part of the ASHAE program, both the proposed research outlines and the results obtained were submitted to the ASHAE Technical Advisory Committee on Panel Heating and Cooling for review. This practice was most helpful in developing the program and in coordinating this work with the work being done at the ASHAE Laboratory.

The authors wish to acknowledge the assistance of the Small Homes Council, a division of the University, which made available the laboratory and many of the instruments used in the tests. The authors also thank those companies which supplied equipment and instruments.

2. Floor Slab Laboratory

The Floor Slab Laboratory, shown in Figs. 1, 2, and 3, was constructed in 1947 to study methods of insulating unheated concrete floor slabs. In 1951

* Exponent numerals refer to corresponding entries in References.
the building was modified to facilitate the investigation of hot water panel heating using \( \frac{3}{4} \)-in. welded steel pipe embedded in concrete floor slabs and nominal \( \frac{3}{8} \)-in. copper tube embedded in the plaster ceilings. Rooms A, B, and C were exposed only to the north, while Room D had both north and east exposures. Rooms A and B had one window each, which had an area equal to 10% of the floor area, while Rooms C and D each had two such windows. All windows were located in the north wall; thus, the effect of solar radiation was minimized. A summary of the over-all heat transfer coefficients and design heat losses for each room is given in Table 1. The walls between the rooms were
Fig. 3. Floor Plans and Instrumentation Details of Floor Slab Laboratory
### Table 1
Heat Loss Data

<table>
<thead>
<tr>
<th>Construction</th>
<th>Thermal Conductance, (F per in.)</th>
<th>Design Heat Loss, Btuh (sq ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Floor</td>
<td>0.69*</td>
<td>800</td>
</tr>
<tr>
<td>Walls</td>
<td>0.16</td>
<td>1,230</td>
</tr>
<tr>
<td>Ceiling</td>
<td>0.08</td>
<td>1,080</td>
</tr>
<tr>
<td>Windows</td>
<td>1.13</td>
<td>1,810</td>
</tr>
<tr>
<td>Infiltration</td>
<td>0.61†</td>
<td>1,100</td>
</tr>
<tr>
<td>Total Heat Loss</td>
<td>6,020</td>
<td>6,020</td>
</tr>
</tbody>
</table>

* Btuh per lin ft of exposed edge.
† Air changes per hr.

The test room floors were separated by 6 in. of cellular glass insulation which extended from the floor surface into the ground to a depth of 4 ft. The insulation between the corridor and test room floors extended to a depth of 2 ft.

The three methods of insulating the floor slabs studied in this investigation are shown in Fig. 3. The insulation used consisted of cellular glass having a thermal conductivity of 0.42 Btuh per hr (sq ft) (F) per in. of thickness. A 4-in. layer of coarse gravel under the floor provided suitable drainage and minimized the capillary rise of moisture from the soil. A duplex kraft paper was placed over the gravel fill for a membrane dampproofing. All the joints were lapped and then sealed with cold tar.

The soil under and adjacent to the Laboratory contained 3 ft of fill which was primarily a silty clay composed of approximately 18% clay, 53% silt, 24% sand, and 5% gravel over approximately 1 ft of the original black silty clay top soil. The normal grade level around the building was 8 in. below the finished floor and sloped away from the building for proper drainage.

### 3. Heating System

A hot-water floor panel heating system was installed in each test room. The panels were designed in accordance with the I=B=R Installation Guide No. 6. They consisted of 3/4-in. welded steel pipe on 12 in. centers and were located so that the top of the pipe was 2 in. below the surface of the floor. In order to vary the output of the panel in conformity with the design heat loss of the room, various portions of the total coil length were by-passed by means of an arrangement of valves. The next to last line of Table 1 shows the required panel area in each room based on a heat emission rate of 50 Btuh per sq ft. The last line gives the active panel area used during the tests. In each room hot water was supplied to the panel at the outside wall. Each test room was provided with its own electric water heater, watt-hour meter, circulating pump, necessary piping, and control system. A schematic diagram of the heating system is shown in Fig. 2, while Fig. 4 is a photograph showing the portion of the test room heating equipment located in the corridor.

Test Rooms A and B were provided with ceiling panel heating systems as well as the floor panel systems just described. The ceiling panels also were designed in accordance with I=B=R Installation Guide No. 6 and consisted of nominal 1/2-in. copper tube located on 6-in. centers. The first coil was located 3 in. from the outside wall. I=B=R Installation Guide No. 6 assumes a heat emission rate of 60 Btuh per sq ft of ceiling panel area, and on that basis each room required an active panel area of 100 sq ft, a panel the width of the room and extending from the outside wall inward a distance of 7 ft. In order to reduce friction loss, the coils in
each panel were divided into parallel circuits. Details of the construction of the ceiling panels are shown in Figs. 5 and 6.

4. Floor Coverings

At the beginning of the 1952-53 heating season, rubber tile was installed on the floor of Room A and asphalt tile was installed in Room B. Both types of tile consisted of 9-in. by 9-in. by 1/8-in. squares. An experienced tile setter installed the tile according to the manufacturer's recommendations as to adhesives and methods of installation. After the tile was laid, a coat of wax was applied.

Two types of carpeting and three pads were used for the tests reported here; the manufacturer's specifications are given in Table 2. These carpets and pads were cut to room size and installed by competent tradesmen. The carpets and pads were selected in consultation with members of the Carpet Institute, Inc., as being representative of the types of carpeting most commonly used in modern residences.
In order to correlate panel performance with the type of floor covering used, the thermal conductance of the various floor coverings had to be determined. A survey of the literature indicated that there is no official standard test procedure for determining the thermal properties of textiles, but several methods of using the hot plate apparatus have been proposed.\(^{(3, 4, 5)}\)

The thermal conductances of the floor coverings used in these tests were determined in a standard guarded hot plate\(^{(6)}\) apparatus using the test procedure outlined by the American Society of Testing Materials. The distance between the hot and cold plates was made equal to the measured thickness of the sample as laid on the floor. The thermal conductivities of the asphalt tile, rubber tile, carpets, and pads were determined by hot plate tests both individually and in combinations. The results are shown in column 4, Table 2, while the thermal conductivity supplied by the manufacturer is given in column 5.

### 5. Instrumentation

Complete instrumentation for temperature measurement was provided, consisting of approximately 800 copper-constantan thermocouples which were used to measure the temperatures of room air, heated and unheated room surfaces, attic air, outdoor air, and water in the system. Along the center line of each room, perpendicular to the north wall, a large number of thermocouples were placed in the ground and under the surface of the concrete slab. Along these same center lines, thermocouples were located across the rooms at intervals of 6 in. on the top surface of the floor slabs. During tests in which floor coverings were used, thermocouples were located at the top surface of the floor covering at a sufficient number of points to obtain a representative measure of the surface temperature.

In Rooms A and B the thermocouples were installed in the top surface of the tile by cutting a very small groove and embedding the leads in the groove so that the top of the junction and the leads were flush with the tile surface. A coat of cellulose cement assured good thermal bond between the tile and the thermocouple junction. The thermocouples measuring the temperature of the top surface of the concrete floor slab were installed in the same way. When the slabs were covered with floor tile, the leads for the thermocouples located on the slab surface were brought into the room at joints between floor tiles. Surface couples on the carpeting were pressed slightly into the pile of the carpet.

Number 24 AWG wire was used for all thermocouples except those used to measure surface temperatures, which were made from 30 AWG copper.
and 28 AWG constantan wire. Surface couples were installed so that at least 3 in. of lead wire on each side of the junction was in contact with the surface whose temperature was to be measured. (7)

Forty-four heat flow meters of the type developed at the ASHAE Research Laboratory, (8) made and calibrated at the University of Illinois, were installed both on the top surface of the concrete floor slabs and under the gravel fill. Ten commercial heat flow meters were installed at the edges of the floor slabs in order to determine edge losses. Other instruments were provided to measure the relative humidity, electrical input to the immersion heaters and control circuits, water flow rates, and level of the water table under the building. Continuous records of the more important temperatures and heat flow rates were obtained by the use of two recording potentiometers.

When making tests on the ceiling panels in Rooms A and B, the heat meters were transferred from the floor surface to corresponding locations on the ceiling. Additional thermocouples were installed on the north-south center line of the ceilings to obtain representative temperatures and temperature variations.

Auxiliary equipment included exhaust fans, heaters, and thermostatic controls for the attic, ventilating fans in the corridor, and thermostatically controlled convection heaters in the corridor and instrument room.

6. Operating Conditions

For all the tests conducted with bare floor panels, the electric water heaters were adjusted to a heat input rate equal to 1.3 times the estimated design heat loss of the room it served. This was done to simulate the normal piping and pickup factors. It was necessary to make some increases in the heat input rate to floor panels which were covered with carpeting, and in some instances the heat input rate to ceiling panels had to be increased also. A high limit control was provided in each system. For the tests with bare floor panels it was adjusted to turn off the heater whenever the leaving water attained a temperature of approximately 130 F. Higher water temperatures were required for the tests in which either carpeting or ceiling panels were used. For these tests the setting of the limit control was adjusted accordingly. Continuous operation of the circulators was used while the operation of the electric water heaters was controlled by room thermostats set to maintain a constant air temperature of 72 F at the 30-in. level. Locations of room thermostats are shown in Fig. 2.

The corridor floor was maintained at the same temperature as that of the adjoining test room floor by manual regulation of the voltage supplied to electric cables embedded in the corridor floor. The temperature of the air in the corridor was kept approximately the same as that in the test rooms by means of thermostatically controlled convection heaters. Except for certain ceiling panel tests, the temperature of the air in the attic was maintained at a constant value of 60 F by thermostatic control.

Complete thermocouple and meter readings for each room were taken several times each day, and the more important data were recorded over 24-hr periods.

Several special tests were made in which the operating conditions differed from those just described. Operating conditions for these tests are described along with the discussion of the test results.
II. RESULTS OF FLOOR PANEL TESTS

7. Bare Floor Panels

a. Heat Flow from Panel vs. Energy Input to Water — Heat flows from a heated floor slab in three directions — upward into the room, downward to the ground or crawl space, and horizontally from the exposed edge to the outdoors. Therefore, heat meters were installed on the top surface, along the exposed edge, and under each floor slab in positions where the heat flow rates were believed to be representative of those for the total respective areas (Fig. 3). Thus the total heat flow in any direction could be determined by multiplying the respective area by the average heat flow rate in that direction. The total flow from the panel was the sum of the total flows in each of the three directions.

The energy input to the water from the heater and the pump motor was equal to the total heat output of the panel plus the heat loss of the connecting piping. Water temperatures were taken at the inlet and outlet of the heater and at the inlet and outlet connections to the panel. It was found that the heat loss from the insulated connecting piping was negligible. The heat loss through the walls of the heater could not be measured, but since the heater consisted of about 2 ft of well-insulated 2-in. pipe and since the average water temperatures ordinarily were less than 130°F, this heat loss should not exceed 30 Btuh, which is negligible. It was concluded that the total energy input to the water was also a good measurement of the total heat output of the panel.

Over a wide range of outdoor temperatures the total heat output of the floor panel in each room was determined from the heat meter readings, and the total energy input was obtained from the readings of the watt-hour meters in the heater and pump motor circuits. It was assumed that 60% of the energy input to the pump motor was utilized in raising the water temperature. Table 3 summarizes the observations made in each of the four test rooms.

With the exception of Room C, the two methods of measuring the total heat output of the panel agreed within 10%. One heat meter in Room C was located on the boundary of the active and inactive panel areas. This condition did not exist in the other rooms. The heat output rate changed rapidly along this line, and it is quite probable that the readings of this meter were not representative of the heat flow from the active panel area. Had this meter been located entirely within the active panel area in Room C, the heat meter and electric meter data might have been in better agreement.

Except at low indoor-outdoor temperature differences, the heat flow as measured by the heat meters was consistently lower than the heat equivalent of the energy input to the electric heaters and of the work done by the pumps. A slight difference in this direction was to be expected, since the energy input to the heaters included the small loss which occurred from the heater wall and piping between heater and panel. On the other hand, the heat meter readings covered only a small percentage of the total panel area. The assumption that the heat flow rate determined from this small sampling was representative of the total heat flow of the panel could result in inaccuracies.

Because of the lack of test points at high indoor-outdoor temperature differences, it was necessary to extrapolate data for all rooms to design conditions. This extrapolation could account for part of the discrepancies at high indoor-outdoor temperature differences.

<table>
<thead>
<tr>
<th>Indoor-Outdoor Temperature Difference, F</th>
<th>Panel Output, Room A Watt-Hour Meter, Btuh</th>
<th>Panel Output, Room B Watt-Hour Meter, Btuh</th>
<th>Panel Output, Room C Panel Output, Room D</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>1130</td>
<td>1250</td>
<td>1330</td>
</tr>
<tr>
<td>40</td>
<td>2280</td>
<td>2220</td>
<td>2770</td>
</tr>
<tr>
<td>60</td>
<td>3420</td>
<td>3200</td>
<td>4180</td>
</tr>
<tr>
<td>80</td>
<td>4540</td>
<td>4230</td>
<td>5390</td>
</tr>
</tbody>
</table>

* Extrapolated values.
In practically every instance the agreement was within 10% for the two methods of measurement, which indicated that instrumentation was adequate. Since there appeared to be fewer chances of error in the measurements of heat flow based upon the electrical input to the system, this method of measurement was used for the total heat flow; the heat meter data were used to proportion the flow from the panel in each direction.

b. Distribution of Heat Flow from Bare Floor Panels — Figure 7 shows the total heat flow from the panel of each room as measured by the electric meters in the pump motor and heater circuits. Figure 8 shows the percentage distribution of the total heat flow in each direction from each of the panels as measured by the heat meters. These curves are based on conditions of operation by which the rooms were maintained at a constant temperature of approximately 72 F at the 30-in. level.

From Fig. 8 it may be observed that for all indoor-outdoor temperature differences in excess of 30 F the division of heat flow from the panels in all four rooms was about the same, with the useful heat into the room being 3 to 5% higher in Room A than in the other rooms. At indoor-outdoor temperature differences of less than 30 F there were wide differences in the division of the heat flow from the different panels with Rooms A and B showing a greater proportion of the total heat being supplied to the rooms.

Actual heat flow in any direction from a panel may be obtained by multiplying the total heat flow at any given indoor-outdoor temperature difference as read from Fig. 7 by the corresponding percentage value read from Fig. 8. Table 4 has been con-
structed in this manner for a temperature difference of 80°F in order to compare the measured heat flow into the rooms with the calculated heat losses. The total measured heat flow from the panels ranged from 90% of the total calculated heat loss of Room A to 111% of the calculated loss of Room C. The measured upward heat flow from the panels ranged from 87% of the calculated above floor heat loss in Room A to 101% of the calculated above floor heat loss in Room C.

It should be pointed out that the construction of Room A and Room B was supposedly the same, and, therefore, the calculated heat losses were the same; however, the measured upward heat flow from the panel in Room B was about 13% higher than that of the panel in Room A. Later investigation revealed some torn insulation in the wall of Room B and one small wall area with no insulation. After this was repaired, the measured heat inputs to Rooms A and B were in better agreement. All of the tests on bare floor panels were made prior to the time the wall insulation in Room B was repaired. While the condition of the insulation had some effect on results obtained, the effect was not of sufficient magnitude to alter conclusions drawn.

While Fig. 8 indicated that there was a large difference in the heat flow downward and from the edge of the panels at low indoor-outdoor temperature differences, the difference in absolute values was small.

In order to make a true comparison of the effectiveness of the insulation, the data had to be integrated over the whole range of outdoor temperatures. The first two columns of Table 5 show the
average outdoor temperature conditions experienced in Urbana, Illinois, from 1936 to 1945 for the months of January, February, March, April, May, September, October, November, and December. The rest of the table indicates the total reverse losses on a heating season basis based on Figs. 7 and 8. These values were totaled and the differences taken. The reverse loss of the panel in Room B exceeded that of Room A by 623,120 Btu per heating season, while the reverse loss in Room C exceeded that of Room A by 1,609,930 Btu per season. Assuming a seasonal operating efficiency of 70%, these increases in the reverse losses represent yearly increases in the seasonal fuel consumption of 8.9 and 23.0 therms of gas or 6.4 and 16.4 gal of fuel oil (140,000 Btu per gal), respectively. This small saving obviously does not warrant the additional expense of providing insulation under the entire slab.

In order to make fair comparisons of subfloor losses for the two types of edge insulation used in Room B and Room C at the same indoor-outdoor temperature differentials, the following conditions should exist:

1. The average water temperature in the panels should be equal.
2. The panel areas should be equal.
3. The above floor room characteristics should be the same.
4. The ground conditions should be the same.

The same design water temperature was used in each room, and, therefore, for any operating condition the water temperatures were approximately the same for each room. Ground conditions were as uniform as it was possible to make them, and since normal variations in the film coefficient on the room side of the panel would produce only a very small change in the downward heat flow, only the difference in the panel area could have an appreciable effect upon its magnitude.

The panel area in Room C exceeded that used in Room B by 29 sq ft. This additional area was located on the side of the room opposite the exposed edge. A test at an average outdoor temperature of 26 F indicated that the increase in downward heat flow amounted to 1.9 Btu per hr for each sq ft of additional panel area. Assuming this to be a representative average value for a winter consisting of 246 days, the downward heat flow from the panel in Room C corrected to the panel area used in Room B would be 3,987,910 — (1.9 × 24 × 29 × 246) = 3,662,100 Btu per heating season as compared to 3,000,600 Btu per heating season for Room B. Thus, the insulation used for Room C appears to be about 3,000,600/3,662,100, or 82% as effective as the insulation used in Room B in reducing downward heat flow. The difference in the seasonal reverse loss was 3,662,100 — 3,000,600, or 661,500 Btu. At 70% efficiency this is equivalent to 9.2 therms of gas or 6.7 gal of oil. Obviously, this difference is of no economic value.

### Table 5

**Seasonal Reverse Losses from Panels**

(Edge and to Ground, based on electric meter data.)

<table>
<thead>
<tr>
<th>Avg. Outdoor Temp., F</th>
<th>Avg. No. Days per Heating Season</th>
<th>Type A</th>
<th>Total Loss, Btu per Heating Season</th>
</tr>
</thead>
<tbody>
<tr>
<td>-10 to -5</td>
<td>0.2</td>
<td>18,900</td>
<td>3,780</td>
</tr>
<tr>
<td>-5 to 0</td>
<td>0.4</td>
<td>18,130</td>
<td>7,250</td>
</tr>
<tr>
<td>0 to 5</td>
<td>0.8</td>
<td>17,200</td>
<td>13,750</td>
</tr>
<tr>
<td>5 to 10</td>
<td>2.2</td>
<td>16,420</td>
<td>36,200</td>
</tr>
<tr>
<td>10 to 15</td>
<td>4.6</td>
<td>15,360</td>
<td>71,500</td>
</tr>
<tr>
<td>15 to 20</td>
<td>7.6</td>
<td>14,400</td>
<td>119,200</td>
</tr>
<tr>
<td>20 to 25</td>
<td>13.6</td>
<td>13,440</td>
<td>160,800</td>
</tr>
<tr>
<td>25 to 30</td>
<td>25.4</td>
<td>12,480</td>
<td>263,600</td>
</tr>
<tr>
<td>30 to 35</td>
<td>33.8</td>
<td>11,520</td>
<td>414,000</td>
</tr>
<tr>
<td>35 to 40</td>
<td>39.0</td>
<td>10,560</td>
<td>543,000</td>
</tr>
<tr>
<td>40 to 45</td>
<td>45.4</td>
<td>9,600</td>
<td>682,000</td>
</tr>
<tr>
<td>45 to 50</td>
<td>51.6</td>
<td>8,640</td>
<td>821,000</td>
</tr>
<tr>
<td>50 to 55</td>
<td>57.8</td>
<td>7,680</td>
<td>960,000</td>
</tr>
<tr>
<td>55 to 60</td>
<td>64.0</td>
<td>6,720</td>
<td>110,000</td>
</tr>
<tr>
<td>60 to 65</td>
<td>70.8</td>
<td>5,760</td>
<td>124,000</td>
</tr>
<tr>
<td>65 to 70</td>
<td>77.6</td>
<td>4,800</td>
<td>138,000</td>
</tr>
<tr>
<td>Totals</td>
<td>246.2</td>
<td>3,987,410</td>
<td>3,986,810</td>
</tr>
</tbody>
</table>

**Difference from Room A**

- **Type A**
  - Subfloor Loss, Btu/day Season: 238,000
  - Total Loss, Btu per Heating Season: 3,000,600
- **Type B**
  - Subfloor Loss, Btu/day Season: 30,300
  - Total Loss, Btu per Heating Season: 6,050
- **Type C**
  - Subfloor Loss, Btu/day Season: 35,700
  - Total Loss, Btu per Heating Season: 7,130
the total output of the panel in Room B and 20.5% in Room C was lost directly to the ground or to the outdoors without being of value to offset the above floor heat losses of the rooms. This is commonly referred to as the reverse loss from the panel.

At the design condition, the total heat flow from the panels as determined from Fig. 7 was 5,600 Btuh in Room B, and 7,400 Btuh in Room C. The reverse losses were 23.0 and 20.5% of these values or 1,290 and 1,515 Btuh, respectively. After correcting for the difference in panel areas in Rooms B and C, the reverse losses were 1,340 and 1,515 Btuh, respectively, based on the panel area of Room C. These reverse losses are equivalent to 93 Btuh per lin ft of exposed edge for Room B and 104 Btuh per lin ft of edge in Room C. Table 6 shows the heat loss, expressed in Btuh per lin ft of exposed edge, from both unheated slabs and concrete floor panels of various types of construction. The losses from the unheated floor slabs were calculated from the data in the 1953 ASHVE Guide. A comparison of the 1-in. vertical type insulation shown in Column C indicates that the reverse loss for the concrete floor panel was 43 Btuh per lin ft of exposed edge for Room B and 104 Btuh per lin ft of edge in Room C. Table 6 shows the heat loss, expressed in Btuh per lin ft of exposed edge, from both unheated slabs and concrete floor panels of various types of construction. The losses from the unheated floor slabs were calculated from the data in the 1953 ASHVE Guide. A comparison of the 1-in. vertical type insulation shown in Column C indicates that the reverse loss for the concrete floor panel was 43 Btuh per lin ft of exposed edge in excess of that for an unheated slab. As applied to Room C, this would amount to

\[ \frac{43 \times 14.5}{7400 - 623} \times 100 \text{ or } 9.2\% \text{ greater than} \]

that required to heat with a more conventional system. The loss for an unheated slab using the 1-in. edge insulation shown in Column A would be somewhere between 45 and 55 Btuh per lin ft. Using a value of 50 and comparing it with 104 for a concrete floor panel with the edge insulation shown in Column A, the fuel required to heat Room C at design conditions by a concrete floor panel system would be

\[ \frac{783}{7400 - 783} \times 100 \text{ or } 11.8\% \text{ greater than that required to heat with a more conventional system utilizing the same edge insulation.} \]

d. Floor Surface Temperatures and Heat Flow — Floor surface temperatures were obtained by means of thermocouples installed on 6-in. centers on a line perpendicular to the north wall and passing through the center of the room. These thermocouples were arranged so that they were located alternately over and midway between coils.

Figure 9 shows a plot of the heat flows and floor surface temperatures for each of the test rooms at a time when the outdoor temperature was 26.7 F. For any one panel the maximum variation in floor surface temperature between adjoining active tubes was about 3 F, while the average surface temperature across the panel remained practically constant. Even though there was little change in the floor surface temperature across the panel, the upward heat flow as measured by the heat meters varied from about 15 Btuh per sq ft at the center of the room to about 45 Btuh per sq ft near the window. To determine how much of this increase might be assigned to increased radiation and how much to increased convection due to the presence of the window, a piece of insulation board 4 ft high by 8 ft long was placed in a vertical position between the windows in Room D and the heat meters nearest to the outer wall. With the bottom located 8 in. above the floor, the baffle offered very little resistance to normal convection currents over the floor, but it did shield the heat meters from the
radiation effects of the windows and most of the exposed wall. Dropping the baffle to the floor not only shielded against radiation effects, but also prevented the circulation of cool air from the window and exposed wall over the floor in the area of the heat meter location.

The effect of the baffle location on the heat meter readings is shown in Table 7. When the heat meters were exposed to the window (no baffle), the average upward heat flow rate was 26 Btuh per sq ft. With 8 in. clearance between the bottom of the baffle and the floor, the average upward heat flow was reduced to 21.6 Btuh per sq ft, and with the baffle located in contact with the floor, the upward heat flow was reduced to 16.1 Btuh per sq ft. In other words, increased radiation to windows and cold wall accounted for about 45% of the total increase in upward heat flow from the panel near the windows and convection currents accounted for about 55% of the total increase.

These readings indicate that a floor panel system has the desirable characteristic of automatically increasing the heat output rates in areas adjacent to points of high heat loss from the room. For a given floor panel surface temperature and room air temperature, one could expect a somewhat higher output per sq ft of panel area in an uninsulated room or one with large glass area than in a fully insulated room with limited glass area.

The average effective panel surface temperature for each room was obtained by averaging the readings of all floor surface thermocouples located on the heated portion of the floor. For each room this average temperature was plotted against the indoor-outdoor temperature difference corresponding to the time the readings were taken (Fig. 10). The maximum difference in the average panel surface temperature for all rooms was about 3 °F at any indoor-outdoor temperature difference. Some difference would be expected, since the installed panel area was based upon the calculated heat loss of the room and since the ratio of the actual heat loss to the calculated heat loss ranged from 0.90 for Room A to 1.11 for Room C. It is also true that the panel surface temperatures as measured on the north-south center line of the rooms were not necessarily the same as at other points in the rooms. This was especially true in Rooms A and B where the effect of the smaller window area on the floor temperatures was greater along the center line of the rooms than toward the sides of the rooms. It is probable that had the instrumentation in each room been adequate to obtain the true average panel surface temperature for the entire panel area, the difference between rooms would have been even less than that indicated in Fig. 10.

The panel surface temperatures as measured in Rooms B and C were accepted as being representa-
tive of the true average for each of the four test rooms, in that the curves for these rooms were the mean of all four rooms with only a 1.5 F deviation from the extremes.

The curves of Fig. 11 were constructed by dividing the upward heat flow from each panel (Figs. 7 and 8) at any given indoor-outdoor temperature difference by the panel area and plotting the quotient against the corresponding panel surface temperature as obtained from the mean of the curves in Fig. 10.

These curves show the relationship between panel output to the test room and panel surface temperature. At a design condition of 80 F indoor-outdoor temperature difference, the average panel surface temperature for all rooms was taken at 86 F, while the panel output varied from 32.4 Btuh per sq ft for Room A to 40.6 Btuh per sq ft for Room C.

The output of the panel in Room B was higher than that of Room A, while the calculated heat losses of the two rooms were the same. Due to faulty insulation, the actual heat loss of Room B was higher than that of Room A. Since the two rooms had the same panel area, it follows that the heat emission rate from Panel B must have been higher than that of Panel A, either as a result of higher operating temperature or of an increase in radiation because of the lower wall surface temperature over the poorly insulated area. Had the insulation in Room B been identical to that in Room A, the curve for Room B in Fig. 11 would have approached that for Room A.

It can be seen that at design conditions the output of the panels in Rooms C and D with two windows each was 20 to 25% greater than the output of the panel in Room A which had only one window. This is in agreement with the results of the special test showing the effect of windows on panel output.

The circle plotted at design conditions in Fig. 11
was obtained by using the following formulas as found on pages 555 and 556 in the 1953 ASHVE Guide.

\[ q_c = 0.81 (t_a - t_w)^{1.12} \]  

(1)*

and the radiation is given by

\[ q_r = 0.142 \left[ \left( \frac{T_s}{100} \right)^4 - \left( \frac{T_w}{100} \right)^4 \right] \]

(2)

and \( q_t = q_c + q_r \)

where \( q_c \) = heat transfer by convection in Btuh per sq ft

\( q_r \) = heat transfer by radiation in Btuh per sq ft

\( q_t \) = heat transfer, total, in Btuh per sq ft

\( t_s \) = panel surface temperature in F (86F)

\( t_a \) = air temperature in F (72F)

\( T_s \) = panel surface temperature in F (absolute) (546F)

\( T_w \) = average unheated surface temperature of the surrounding surfaces in F (absolute) (526.5F)

All temperatures correspond to observed conditions in Room A at an indoor-outdoor temperature difference of 80 F. It can be seen that the output of the panel in Room A was essentially the same as the calculated value, while the panel outputs in Rooms C and D were higher than the calculated value by almost 20% or about 6.5 Btuh per sq ft. Had the actual average unheated surface temperatures for Rooms C and D been known and used in calculating the panel output, the calculated output would have more nearly approached the test values for these rooms.

8. Covered Floor Panels

a. Water Temperatures — Water temperatures were measured in each test room at the inlet and outlet of the coils; the arithmetic mean of these temperatures was taken to be the average. For each room the average water temperatures for 24-hr test periods were plotted against the corresponding indoor-outdoor temperature difference. Figure 12 shows two typical curves which are representative of curves obtained from all the test rooms. In all cases it was observed that the greater the thermal resistance of the floor covering, the greater the increase in the average water temperature for a given increase in indoor-outdoor temperature difference.

Figure 13 shows the relationship between the average water temperature at an indoor-outdoor temperature difference of 50 F and the thermal resistance of the floor coverings used in Rooms C and D. Also indicated on the figure are the standard deviations of the data from the line of regression. With the exception of the tests conducted with the \( \frac{1}{4} \)-in. rubber pad where only a relatively small number of test points were obtained, the greater the thermal resistance, the larger the standard deviation.

The above floor heat output of a floor panel can be expressed in terms of an equivalent thermal transmittance and a temperature difference between water and room air:

\[ \frac{Q}{A} = U (t_w - t_a) \]

(3)

where \( Q \) = above floor heat flow in Btuh

\( A \) = panel area in sq ft

\( U \) = equivalent thermal transmittance from water to room air in Btuh per sq ft (F)

\( t_w \) = average water temperature in F

\( t_a \) = room air temperature in F

At a 50 F indoor-outdoor temperature difference, the above floor heat flow per sq ft of panel area in Room D was 22.8 Btuh. This value was obtained by determining the average panel surface temperature at 50 F indoor-outdoor temperature difference and the corresponding heat flow as described in Section 7d. The water temperature for tests with a bare floor panel in Room D was 96 F at a 50 F indoor-outdoor temperature difference (Fig. 12). Letting \( R \), the equivalent thermal resistance, equal \( 1/U \) and rearranging equation 3:

\[ R = \frac{(t_w - t_a)}{Q/A} \]

(4)

\[ R = \frac{(96 - 72)}{22.8} = 1.05 \]

For tests with floor coverings, the same equation applies, and \( R \) is approximated by the equivalent resistance of the bare floor plus the resistance of the floor covering. \( t_w \) becomes the water temperature obtained with the covered panel \((t_{wc})\). The air
temperature and heat flow remain the same. Equation 2 then becomes

$$R + R_{fe} = \frac{(t_{we} - t_a)}{Q/A}$$  \hspace{1cm} (5)

Substituting for R from equation 2 yields

$$t_{we} = (Q/A) R_{fe} + t_w$$  \hspace{1cm} (6)

or for Room D

$$t_{we} = 22.8 R_{fe} + 96$$  \hspace{1cm} (7)

Equation 7 is the equation of curve 1 in Fig. 13. Curve 2 has been fitted to the test points.

With the exception of the tests with the bare floor panel, all the points were located below the theoretical line, and the points for Room C were the farthest removed. It is evident that the water temperature required for a material with a given thermal resistance as determined in the guarded hot plate apparatus was less than that theoretically required. This may have been the result of inter-

Fig. 12. Average Water Temperature, Room D, Floor Panel
mittent operation of the heater and "nonsteady" conditions of testing. Equation 5 is based upon the assumption that the concrete slab with a coil can be replaced by a slab with a plane heat source such that one dimensional heat flow occurs. Also, differences in the surface conditions for the carpets as tested in the guarded hot plate and as installed in the test room could have resulted in differences in over-all thermal conductance.

It is necessary to know only the equivalent thermal resistance of the bare floor panel and the thermal properties of the floor covering used in order to predict the required water temperature for the covered panel. In order to make the results more general, the dimensionless ratio \( \frac{t_w - t_a}{t_w - t_u} \) was obtained which was plotted against the dimensionless ratio of the resistance of the floor covering to the equivalent resistance of the bare panel (Fig. 14).

The relationship between the water temperatures and resistance of floor coverings (Fig. 13) was obtained at a 50 F indoor-outdoor temperature difference. If straight line extrapolation to design conditions (80 F) can be assumed, it would be necessary to take the difference in water temperature for the bare and covered floor under consideration from Fig. 13 and multiply by 8/5 in order to obtain the water temperature difference at design conditions. The plot of Fig. 14 holds true regardless of the indoor-outdoor temperature difference, because for two different panels the ratio of water to room air temperature differences is equal to the ratio of over-all resistance of the panels, which is independent of outdoor temperature.

From Fig. 13, curve 2, the average water temperature required with a heavy weight carpet and 40 oz pad at an indoor-outdoor temperature difference of 50 F is 127 F. At 80 F indoor-outdoor temperature difference (design conditions in Ur-
bana, Illinois), the required water temperature would be $72 + \frac{8}{5}(127 - 72)$ or 160 F. Similarly, the water temperature at 80 F indoor-outdoor temperature difference would be 110 F for a bare panel. Thus a 50 F ($160 - 110$) increase in water temperature would be required when using the heavy weight carpet and 40 oz pad. If this required increase in water temperature were not taken into account in the design of a system, satisfactory performance could not be expected.

Rebalancing a system in which one circuit serves a floor panel which has been covered with a floor covering having a high thermal resistance may not be possible by increasing the water temperature in the entire system and throttling down the flow of water to uncovered panels. This practice would increase the water temperature drop and decrease the effective panel area for those circuits serving bare panels. Hot areas would be apt to appear at the point where the water enters the uncovered panels. This may be undesirable and could be avoided by proper design conditions. It may be necessary to utilize separate zones so that different water temperatures can be maintained in different circuits.

b. **Total Heat Input to Panels** — The total energy supplied to the water in the heating system of each test room by the water heater and the circulator was measured by means of watt-hour meters. This energy, minus the losses from the insulated heater and connecting piping, was transferred to the panel. Since the losses from the heater and piping were negligible in comparison with the total panel output, the total energy input to the water was assumed equal to the total energy supplied to the panel.

Figure 15 shows the total energy supplied to the floor panel in each test room plotted against the indoor-outdoor temperature difference. The curves were plotted by the least squares method. The scattering of the test points was due to the effects of uncontrollable variables, such as variations in the intensity of solar radiation and wind velocity and changes in the outdoor temperature. However, sufficient tests were run so that the curves fitted to the test points represent trends for average weather conditions.

Superimposed as dotted lines in Fig. 15 are curves fitted by the method of least squares to data obtained for bare panels. It can be seen that in Room A the rubber tile had practically no effect on the total energy input to the panel. No similar comparison was made in Room B between a bare panel and one covered with asphalt tile because changes were made in the insulation of Room B which decreased the heat losses for the tests with
covered floor panels from the heat losses for tests with bare floor panels. However, since the thermal conductivity of the asphalt tile in Room B was the same as that of the rubber tile in Room A, it may be assumed that the effect of the asphalt tile on the total energy input to the panel also was insignificant.

The curves for Room C show that at indoor-outdoor temperature differences greater than about 30 F, the total energy input to the panel was greater for the carpeted floor than for the bare floor; while at indoor-outdoor temperature differences less than about 30 F the reverse was true. The same characteristic was shown in Room D, although the differences in energy input were smaller.

The energy input to the floor panel in Room C at design conditions (80 F indoor-outdoor temperature difference) was about 40% higher when the panel was covered by the heavy carpet and 40 oz pad than when no floor covering was used. Thus the insulating effect of the heavy carpet and pad increased the reverse losses and heat storage within the panel and floor covering to such an extent that the boiler capacity had to be increased by about 40% in order to satisfy room heat losses at design conditions. The increase in required boiler capacity when the panel was covered by a light carpet and 1/4-in. rubber pad (Room D) was about 16%. No increase in boiler capacity was required for either the rubber or asphalt tile (Rooms A and B).

In order to make this comparison it was necessary to extrapolate the curves of Fig. 15 about 20 F beyond test points. Past experience has shown that these curves are ordinarily straight lines, and, therefore, it is believed that values obtained by these extrapolations are reasonably accurate.

If the distribution of outdoor temperatures for a typical winter is known, the curves of Fig. 15 may be used to estimate the effect of floor coverings on seasonal fuel consumption. Table 8 shows the estimated energy inputs (fuel consumptions) for Room C when heated by a bare panel and by a panel covered with a heavy carpet and 40 oz pad. When the bare panel was used, the total annual fuel consumption was about 12,900,000 Btu or approximately 129 therms. The annual fuel consumption for the same room using a combination of heavy carpet and 40 oz pad over the panel was about 13,525,000 Btu or approximately 135 therms. Thus the difference in fuel consumption was about 6 therms per year or an increase of approximately
The total heat flow from the panel as measured by heat meters (located on the floor surface, under the gravel fill, and at the exposed edge of the panel) was plotted in Fig. 16 against the indoor-outdoor temperature difference. For comparison the total energy input curves of Fig. 15 have been reproduced as dotted lines. With the exception of Room C, the total energy input and the total heat flow from the slab were in good agreement. At low indoor-outdoor temperature differences, the measured heat flow from the panel exceeded the measured energy input in Room C, while at high indoor-outdoor temperature differences the reverse was true.

Figure 16 represents results of tests which were made after the initial warm-up of the ground had taken place. Also, the majority of the tests were at indoor-outdoor temperature differences ranging from 30 to 50°F with only a few tests at higher or lower indoor-outdoor temperature differences. Obviously, as water temperatures are increased to compensate for the additional heat requirements of the room which result from a drop in outdoor temperature, the temperature of the concrete and gravel constituting the floor must increase. The higher the thermal resistance of the floor covering, the greater the required change in the average floor temperature per degree change in outdoor temper-
at ure will be. Also, floor coverings having high thermal resistance will retard the rate of change of heat flow upward into the room resulting from a change in panel temperature. Both of these conditions tend to accentuate differences in measured inputs and outputs and to increase the scatter of test points for Room C.

Figure 17 was constructed by totaling the measured heat flow from the floor panel in each room and calculating the percentage of the total heat flow in each direction. Superimposed on this set of curves are the data for bare concrete floor panels. The curves for Rooms A and B show that the percentage distribution of heat flow from the panels covered with either asphalt or rubber tile was practically the same as that for bare panels. However, the use of a heavy carpet and pad in Room C caused a large difference in the percentage distribution of heat flow from the panel. At indoor-outdoor temperature differences greater than about 33 F, the addition of the heavy carpet and pad decreased the percentage heat flow upward into the room and increased the percentage flowing downward into the ground. On the other hand, at indoor-outdoor temperature differences of less than about 33 F, the heavy carpet and pad had quite the opposite effect; the percentage of upward heat flow was increased and the percentage of downward heat flow was decreased.

It should be pointed out that the curves of Fig. 17 show comparisons of heat flow from the panel and not energy input to the panel. Because of heat storage in the panel, the two are not the same. The upward heat flow in Room C was approximately the same for both the covered and bare panel at an indoor-outdoor temperature difference of 80 F. This being the case, the increase in total heat flow from the carpeted panel in Room C may be expressed by the equation

$$\frac{H_t' - H_t}{H_t} = \frac{1}{H_u/H_t} - \frac{1}{H_u/H_t}$$

Substituting for $H_u/H_t$ and $H_u/H_t$, the values obtained from Fig. 17, equation (8) becomes

$$\frac{1}{0.63} - \frac{1}{0.79} = 25.4\%$$

Previously, it was shown that carpeting the floor in Room C increased the required energy input at design conditions by 40%. The difference between the increase in energy input and the increase in total heat flow from the panel represents increased heat storage in the panel itself. Since heat storage is a function of the change in mean panel temperature, which at design conditions was much less for the bare panel than for the covered panel, it follows that heat storage effects were also much smaller for the bare panel than for the covered panel. Increased heat storage for the covered panel was equal to the increase in energy input minus the increase in total heat flow from the panel. The increase in energy input was the difference between the two curves of Fig. 15. By assuming that the heat flow from the bare panel was equal to the energy input at design conditions, the increase in total heat flow for the covered panel is given by the difference between the total heat flow from Fig. 16 and the energy input for the bare panel from Fig. 15. By this method of analysis the increased heat storage for the covered panel in Room C was

$$\frac{(9550-6800) - (8550-6800)}{6800} = 1000 \text{ Btuh}$$

At design conditions the reverse losses in Room C were 37% of the total heat flow from the carpeted panel while for the bare panel these losses were 21%. Since the above floor heat flow was the same in both cases, the reverse losses for the covered panel were

$$\frac{.37 \times \text{upward heat flow}}{.63}$$

or about 2.2 times as large as those for the bare floor. The increase in reverse losses at design conditions was calculated for each combination of carpet and pad tested. Curve 1 of Fig. 18 shows a plot of the ratio of the reverse losses for the covered
panel to the reverse losses for the bare panel versus the thermal resistance of the floor coverings.

Table 9 shows the reverse heat loss for a floor slab utilizing L-type insulation. Values are tabulated for unheated and heated slabs with two types of floor covering. The heat losses for the heated, covered floor panels have been obtained by multiplying the heat loss for the bare panel by the value obtained from curve 1, Fig. 18. It can be seen that the heat loss for the bare, heated panel was about twice as great as that for a similar unheated slab. The heat loss for the heated slab with light weight carpet and rubber pad was about 3.3 times as great as that for the bare, unheated slab, while the heat loss for the heated slab with heavy carpet and pad was about 4.8 times as great.

The percentage increase in total panel output at design conditions resulting from the use of floor coverings is shown by curve 2, Fig. 18, while the percentage increase in total heat input is shown by curve 3. Since the floor covering did not affect the above floor heat loss of the room, the upward heat flow from the panel was unaffected by the covering. Therefore, curve 2, Fig. 18, is a measure of the increase in reverse losses from the panel, and curve 3 is a measure of the increase in reverse losses plus the increased heat storage in the panel. The effect of the floor coverings on heat storage within the panel is represented by the difference between curves 2 and 3.

To apply these results to design, it may be more convenient to express the increases in per cent of upward heat flow rather than in per cent of total heat flow from the panel. To do so requires that the values from curves 2 and 3 be divided by 0.79 minus the ratio of upward heat flow to total heat flow for the bare panel. Thus for a given structure, if the above floor heat loss is calculated by the conventional method in the Guide, the increase in heat storage, reverse losses, and energy input may be obtained for any thermal resistance of floor covering from curves 4 and 5.

In general, the floor coverings with high thermal resistance had the undesirable characteristic of increasing the reverse losses and heat storage in the panel in cold weather. This resulted in a lower percentage of useful heat to the room, but on a seasonal basis this was compensated for, at least in part, by an increase in the percentage of useful heat obtained during warmer weather. The heat input required to provide for heat storage in the panel is in effect a panel pick-up allowance which should be made when selecting a boiler. Also if the increase in reverse losses is not accounted for in the heat loss calculations, additional allowance should be made in boiler sizing to compensate for the increased reverse losses.

d. Application of Results—In order to summarize the observed effects of floor coverings on
the performance of a floor panel heating system, an illustrative example has been chosen. Three cases are presented: bare concrete slab, concrete slab with carpet and pad having a thermal resistance of 1.0, and concrete slab with carpet and pad having a thermal resistance of 2.0. For this example, Table 10 shows that the use of a floor covering having a thermal resistance of 1.0 requires a 26 F increase in water temperature, which results in an increase in reverse loss amounting to 16,400 - 10,400 = 6,000 Btuh. A floor covering having a thermal resistance of 2.0 requires a 55 F increase in water temperature, which results in an increase in reverse loss amounting to 25,100 - 10,400 = 14,700 Btuh. For the floor panel with a carpet and pad having a resistance of 1.0, an additional 2,720 Btuh must be provided; while for the floor panel with a carpet and pad having a resistance of 2.0, an additional 9,590 Btuh must be provided for heat storage, which is a pick-up allowance. Therefore, the minimum net output of the boiler for the panels with coverings having resistances of 1.0 and 2.0, respectively, exceeds that for the bare panel by 8,720 Btuh and 24,290 Btuh, or about 17 and 48%. These figures do not include the normal piping and pick-up factor. Another 30% is usually added to provide an allowance for piping and pick-up.

e. Room Air Temperature Control — In order to evaluate the ability of the heating system to control the room air temperature at a given value, the maximum and minimum air temperatures occurring at the location of the thermostat during 24-hr periods were recorded. Plotting indicated no correlation between the observed maximum—minimum temperature difference and the indoor—outdoor temperature difference; therefore, the data were grouped and averaged according to operating conditions (Table 11).

The ability of the thermostat to control the room air temperature at a given value depends upon the thermostat differential, the response of the panel (time required for the panel to heat and cool), the time lag for transient heat flow through the exposed walls and windows, and changes in the outdoor temperature. The thermostat differential setting and the outdoor temperature for any one test period were the same for all four test rooms. The time lags for transient heat flow through the north exposure in Rooms C and D were probably less than for Rooms A and B because of the larger window area. Because of this and the fact that the carpeted floor panels did not respond to load changes as quickly as did the bare panels, the room air temperatures in Rooms C and D were subjected to more variation than in Rooms A and B.

The effects of floor coverings and operating conditions on room temperature variations are best shown by a room-by-room analysis of Table 11. Columns 2 and 4 show that the thermostat in Room A was apparently set about 1.6 F higher during the tests in Case IV than in Case III. Column 6 shows that the average temperature variation in this room was about the same in both cases. Columns 3, 5, and 7 show the standard deviations (\( S_d \)) from the average values given in columns 2, 4, and 6, respectively. The standard deviation is a measure of the reproducibility of observations. Sixty-eight % of all observations will normally be within \( \pm \) one standard deviation of the average of all observations, and 95% will be within \( \pm \) two standard deviations. In Room A the standard deviations were about the same in Case IV as in Case III. Thus it is apparent that while there was about a 1.5 F change in the thermostat setting, the control of room air temperature was not affected. In Room A the daily range of variation in room air temperature at the thermostat was 0.5 F to 1.3 F for Case III and 0.6 F to 1.6 F for Case IV.

A similar analysis of Rooms B, C, and D shows that, as in Room A, the control of room air temperature was not affected by unavoidable changes in the setting of the thermostat. Also the average maximum minus minimum air temperature differences in Room B were higher than in Room A, indicating a difference in the thermostat operating differentials even though the differential settings of the thermostats were the same. This affected the
daily variation of room air temperature in Room B, which was 1.6 to 2.8 F for Case III and 2.1 to 3.5 F for Case IV (Col. 8).

The table shows that in Room C, Cases I and II, the standard deviation for the maximum room air temperature was 0.6 to 0.8 F, while that for the minimum temperature was 0.3 to 0.4 F. During this time no floor coverings were used in the room. The effect of the carpet and pad in Room C, as shown by Cases III and IV, was to increase the standard deviation to values of 1.0 to 1.3 F for the maximum and 0.5 to 0.6 F for the minimum. As shown in column 8, this resulted in a large daily variation in room air temperature.

The values for the standard deviations in Room D were very nearly the same as those in Room C. However, the average maximum minus minimum room air temperature was about 0.7 to 1.0 F higher, indicating a larger variation in daily room air temperature. The standard deviations from the average maximum room air temperature in Rooms A and B for these periods of time were only about one-half as large as those in Rooms C and D. The standard deviation from the average minimum room air temperatures in Rooms A and B were nearly the same as those in Rooms C and D.

Any change causing an increase in the values of column 5 of Table 11 will increase the variation in minimum air temperature occurring in the room at the end of the off-periods of the thermostat, thus making it necessary to increase the thermostat setting accordingly to insure that the room air temperature remains above a predetermined minimum value required for comfort. Changes causing an increase in the values of column 3, Table 11, will increase the normal variations in maximum room air temperature occurring at the end of thermostat on-periods. Such overruns in room air temperature are usually due to heat storage within the panel. However, the rate of heat loss from the room resulting from differences in construction or differences in outdoor temperature may be a contributing factor. For example, when the carpets were interchanged in Rooms C and D, the values in columns 3 and 7 for Room C were increased even though the carpet was lighter. This was due to the warmer weather during Case V as compared to that during Case IV.

One can conclude from this table that, other factors remaining constant, the addition of floor coverings to bare floor panels will reduce the ability of the system to maintain a constant room air temperature. The greater the thermal resistance of the floor covering, the greater will be the resulting room air temperature variation.

In order to show the effect of floor coverings on room air temperature control with large outdoor temperature changes, a period of time was chosen during which there was a sharp decrease in outdoor temperature. Figure 19 shows a plot of room

---

**Table 11**

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>(1)</td>
<td>(2)</td>
<td>(3)</td>
<td>(4)</td>
<td>(5)</td>
</tr>
<tr>
<td>Room A</td>
<td>72.2 ±0.5</td>
<td>71.3 ±0.6</td>
<td>0.9 ±0.4</td>
<td>0.5-1.3</td>
</tr>
<tr>
<td>Room B</td>
<td>75.6 ±0.8</td>
<td>75.6 ±0.5</td>
<td>1.6 ±1.4</td>
<td>1.0-2.2</td>
</tr>
<tr>
<td>Room C</td>
<td>76.0 ±1.0</td>
<td>75.0 ±0.6</td>
<td>2.0 ±1.4</td>
<td>1.2-2.8</td>
</tr>
<tr>
<td>Room D</td>
<td>75.8 ±1.0</td>
<td>75.5 ±0.5</td>
<td>2.5 ±1.2</td>
<td>1.3-3.7</td>
</tr>
</tbody>
</table>

* Col. 6 ± Col. 7.

**OPERATING CONDITIONS**

**Case I** - January 25-February 28, 1952
Room C — Bare Panel
Room II — March 1-30, 1952
Room C — Bare Panel
Room C — December 23-31, 1952
Room A — Rubber Tile
Room B — Asphalt Tile
Room C — Heavy Carpet and Pad
Room D — Light Carpet and Rubber Pad

**Case II** - January 25-February 28, 1952
Room A — Rubber Tile
Room B — Asphalt Tile
Room C — Heavy Carpet and Pad
Room D — Light Carpet and Rubber Pad

**Case III** - December 23-31, 1952
Room A — Rubber Tile
Room B — Asphalt Tile
Room C — Heavy Carpet and Pad
Room D — Light Carpet and Rubber Pad

**Case IV** - January 25-February 28, 1952
Room A — Rubber Tile
Room B — Asphalt Tile
Room C — Heavy Carpet and Pad
Room D — Light Carpet and Rubber Pad

**Case V** - February 27-March 29, 1953
Room C — Light Carpet and Rubber Pad
Room D — Heavy Carpet and Rubber Pad

---

* $S_d$ = Standard deviation from average.
air and outdoor temperature vs. time for Rooms A, C, and D for this period.

The room air temperature for Room A (rubber tile) remained relatively constant even during the rapid change in outdoor temperature. However, during the first rise in outdoor temperature, the air temperature in Rooms C and D (carpet and pad) rose a few degrees. When the large outdoor temperature drop occurred, the room air temperature began to decrease. The thermostats turned on the water heaters in the heating systems in Rooms C and D at about 6:30 p.m. Even so, the room air temperatures continued to decrease until about 11:00 p.m. At this time they leveled off at a value some 3 to 4°F lower than the thermostat setting and then began to climb. Heat was finally being supplied at a great enough rate to more than offset the increased room heat loss due to the outdoor temperature drop. The minimum room air temperature occurred in Room C about one hour later than in Room D.

By 7:30 a.m. the thermostat in Room D was satisfied, but the thermostat in Room C continued to call for heat until about 9:15 a.m. At 8:30 a.m. and 2:30 p.m. the air temperatures in Rooms C and D reached their respective maximum values. While the maximum room air temperature in Room D occurred one hour after the heater went off, the maximum room air temperature in Room C occurred about 5 hours after the heater went off due to the increased heat storage in the panel and the less severe room exposure. After the outdoor temperature became relatively constant, the room air temperature in all test rooms became more stable. Much better control of room air temperature was afforded in Room A than in Rooms C and D due
had a negligible effect on the control as compared to the effect of the carpet and pad combinations.

Since there was an extremely long heater operating period for Room C due to the rapid decrease in outdoor temperature, the overrun in room air temperature was exaggerated. Therefore, a study was made over one complete heater cycle in Room C at a time when the outdoor temperature was relatively constant at about 33°F. Temperatures of slab surface, carpet surface, room air, and water were measured during this test, along with heat flow from the slab to the carpet. The floor covering consisted of the heavy carpet and 40 oz pad. Figure 20 shows a plot of the measured temperatures and heat flows vs. time. At 12:30 a.m. the room air temperature at the thermostat was dropping. At approximately 12:50 a.m. the thermostat called for heat, and the water heater was turned on. At this time the room air temperature was 70.8°F. The panel inlet water temperature almost immediately increased from 106°F to 140°F, and the average slab surface temperature began to increase above 106°F. At the same time the average heat flow from the slab to the floor covering began to increase. The average carpet surface temperature continued to rise until about 10 min after the heater turned on, whereupon it started to increase. Meanwhile, the room air temperature at the thermostat continued to decrease and reached its minimum value at about 1:25 a.m., or about 35 min after the heater went on.

The panel inlet water temperature continued to rise until about 2:10 a.m. when the thermostat was satisfied and the heater went off. The room air temperature at this point was about 70.8°F, the same as at the time the thermostat turned the heater on. Five or 10 min after the heater turned off, the average slab surface temperature reached a maximum and thereafter decreased along with the heat flow from the slab to the floor covering. Even though the heat flow from the slab to the floor covering was decreasing, the average carpet surface temperature continued to rise until about 2:40 a.m. when it reached a maximum. The maximum carpet surface temperature occurred 30 min after the thermostat was satisfied. The room air temperature continued to rise for another 30 min making about an hour's delay between the times when the water temperature and the room air temperature reached their maximum values. There was a 25-min time delay between the time when the slab surface and the carpet surface reached their maximum values.

to the comparatively rapid response of the system with the tile floor covering.

These curves show the effects of the carpet and pad combinations on the control of the room air temperature. Part of the difference in the control characteristics of Rooms A, C, and D might be attributed to the differences in exposure. However, additional panel capacity was provided in design to offset this portion of the difference. Observations in Rooms A, C, and D, as reported in Table 11, verify the conclusion that the additional exposure
This test indicates that there was a relatively long period between the times when the thermostat called for heat and when the air temperature started to increase. The effect of the carpet and pad was to retard the flow of heat from the water to the room air and cause about a 0.3 F under and a 1.0 F overrun in room air temperature. Eighty min after the thermostat called for heat it was satisfied. Thus the response time for the heating system was very long.

f. Room Air Temperature and Velocity Distribution—Air temperatures were measured in each test room at the levels 3, 12, 30, 60, 84, and 93 in. above the floor. At the center of the room the maximum temperature difference between the various levels was obtained by subtracting the lowest temperature from the highest temperature, regardless of distance above the floor. These are correlated with indoor-outdoor temperature difference in Table 12. About 35 tests were made in each case with indoor-outdoor temperature differences ranging from 15 F to 63 F. A statistical analysis showed that at 95% confidence level the average difference in room air temperature between the levels of 3 in. above the floor and 3 in. below the ceiling in Rooms A and B would not exceed 2.3 F. For Rooms C and D this variation would not exceed 3.8 F. Interchanging the floor coverings in Rooms C and D did not affect the air temperature difference.

The correlation of room air temperatures as measured at the 30-in. level in the center of the room with the corresponding indoor-outdoor temperature differences also are shown for each room in Table 12. The trend was for the room air temperature to decrease slightly as the indoor-outdoor temperature increased. However, observations of the room air temperature near the thermostat indi-
cated that indoor-outdoor temperature difference had no effect on the room air temperature at this location. Apparently, the lower air temperature at the center of the room resulted from the circulation of cool air from the window which was not warmed to the normal room temperature.

Two days were selected in which average outdoor temperatures were about 37°F and 20°F, respectively, and the room air temperatures and gradients were very close to the average values in Table 12. Floor to ceiling temperatures at the center of the room were plotted for those days for all four test rooms (Fig. 21).

All room air temperatures were corrected to a condition of 70°F at the 30-in. level for comparative purposes. Variations in temperature from floor to ceiling were very small and, except for the floor surface temperature, were virtually independent of outdoor temperature. Since the thermocouple at the 30-in. level in Room A was within a few inches of the thermo-integrator, which had a surface temperature of about 80°F, the indicated temperature at this point was probably high. Had the thermo-integrator not been present, it is probable that the temperature curve in Room A would have more closely approximated that of Room B.

In Rooms C and D the temperature curves (Fig. 21) show a somewhat different shape than those in Rooms A and B. In Room C there was an increase in air temperature from the 3-in. level to the 30-in. level, while in Room D there was a slight decrease in temperature from the 3-in. level to the 12-in. level, and the temperature at the 30-in. level was 0.6°F to 1.2°F higher than that 12 in. above the floor. Above the 30-in. level in all test rooms, the room air temperature was very uniform for both moderate and low outdoor temperatures. When the outdoor temperature was 20°F, the room air temperature 3 in. above the floor in Rooms A and B at a location 2 ft from the north wall was about 3°F lower than the corresponding value at the center of the room, whereas in Rooms C and D the difference in air temperature at these locations was 1.5°F to 2.5°F lower. This was to be expected since a definite downward movement of cold air was noticeable at the north wall. This cool air continued to move across the floor toward the center of the room.

To determine the velocity of air movement in the rooms, a heated wire anemometer capable of measuring air velocities as low as 10 fpm was used as a probe. Figure 22 represents a cross section of Room C with air speed, direction, and temperature indicated. The air was cooled at the north wall and window, dropped to the floor, and then moved in the direction of the south wall with a comparatively high velocity. Passing along the floor the air was heated by the panel, whereupon it rose and drifted back slowly toward the north wall. Circulation of air occurred only as a result of air temperature differences. The air temperatures were very uniform throughout the room except for locations along the floor and north wall. Near the window and within 12 in. of the floor the velocity was high and the temperature was low. In extreme cases this combination of high velocity and low air temperature will result in undesirable drafts.

g. Floor Covering and Slab Surface Temperatures—Figure 23 shows a plot of floor covering surface (panel surface) minus room air temperature difference vs. indoor-outdoor temperature difference. Also plotted in the same figure are the slab surface minus room air temperature differences. The dotted curves of Fig. 23 show that at design conditions the panel surface temperatures in all test rooms were from 12 to 15°F above the room air temperature measured in the center of the room at the 30-in. level.

In Rooms A and B the temperature drop across the tile was very small as evidenced by the differences in the values of the two curves for each room. Such was not the case in Rooms C and D. At design conditions the temperature drop across the
carpet and pad in Room C was about 54 °F, while the temperature drop across the carpet and pad in Room D was about 38 °F. These relative values would be expected since the effective conductance of the carpet and pad in Room C was greater than that in Room D.

The equivalent thermal resistance of the floor covering in each room is given by equation 5 (Section 8a). The value of \( Q/A \) at design conditions for each room was obtained by multiplying the percent above-floor heat flow from Fig. 17 by the total heat flow from the panel and slab at 80 °F indoor-outdoor temperature difference and dividing by the panel area.

Since for Room C the upward heat flow (Section 8c) included that for the inactive panel area while the temperature difference across the floor covering was evaluated for only the active panel area, the upward heat flow had to be corrected to include the active panel only. Recorded data indicate that the total upward heat flow should be reduced by about 5% to correct for the inactive panel area. Therefore, the equivalent thermal resistance of the floor covering in Room C was

\[
\frac{54.0}{(0.95)(36.8)} = 1.55 \text{ hr (sq ft) (F) per Btu}
\]

while that in Room D was

\[
\frac{38.0}{40.5} = 0.94 \text{ hr (sq ft) (F) per Btu}
\]

The thermal resistances of these carpet and pad combinations in Rooms C and D were 1.85 and 0.99 hr (sq ft) (F) per Btu, respectively, as determined in the standard guarded hot plate. The two thermal resistances determined by these different methods were within 5% in Room D, while those in Room C differed by about 16%. The differences in thermal resistances of the floor coverings as determined by the two methods could be due to uncontrollable variables. The method of surface temperature determination in the two tests was different. Under actual operating conditions the heat flow from the slab to the room was not steady, and there was probably considerable heat storage within the carpet and pad. The moisture content of the floor coverings could have influenced the results to the extent that the thermal resistance would change and heat would be required to evaporate.

<table>
<thead>
<tr>
<th>Room</th>
<th>Outdoor Temperature, °F</th>
<th>SURFACE TEMPERATURES</th>
<th>Room A</th>
<th>Room C</th>
</tr>
</thead>
<tbody>
<tr>
<td>East Wall</td>
<td>Distance above Floor =</td>
<td>24 in.</td>
<td>71.4</td>
<td>72.9</td>
</tr>
<tr>
<td>West Wall</td>
<td>Distance above Floor =</td>
<td>24 in.</td>
<td>70.5</td>
<td>70.3</td>
</tr>
<tr>
<td>South Wall</td>
<td>Distance above Floor =</td>
<td>24 in.</td>
<td>70.2</td>
<td>69.4</td>
</tr>
<tr>
<td>North Wall</td>
<td>Distance above Floor =</td>
<td>24 in.</td>
<td>69.7</td>
<td>68.4</td>
</tr>
<tr>
<td>Window (North Wall)</td>
<td>Distance above Floor =</td>
<td>42 in.</td>
<td>32.0</td>
<td>39.7</td>
</tr>
<tr>
<td>Floor (On N-S Centerline of Room)</td>
<td>Distance from North Wall =</td>
<td>6 in.</td>
<td>77.4</td>
<td>82.1</td>
</tr>
<tr>
<td>Ceiling (On N-S Centerline of Room)</td>
<td>Distance from North Wall =</td>
<td>6 in.</td>
<td>67.6</td>
<td>64.5</td>
</tr>
</tbody>
</table>

Table 13
Room Surface Temperatures

![Fig. 23. Panel and Slab Surface Minus Room Air Temperatures](image-url)
any moisture present in the covering. No attempt was made to evaluate the moisture content of the floor covering for any test. Furthermore, the samples used for hot plate tests were not necessarily representative of the actual coverings used, since variations in materials and manufacturing techniques could result in differences in physical properties.

h. Glass Surface Temperatures — Figure 24 shows a plot of the inside window surface temperature for Room A vs. indoor-outdoor temperature difference. This curve was identical with those obtained for the other test rooms. Also indicated on this plot are the inside glass surface temperatures obtained in the I=B=R Research Home with a conventional hot water heating system using small tube radiators. There was no difference in the observed window surface temperatures in the two cases. The glass surface temperature dropped off rapidly with decreasing outdoor temperature. At design conditions of 80 F indoor-outdoor temperature difference, the glass surface temperature was about 26 F.

The dotted curve of Fig. 24 represents the relative humidity of 72 F air which will result in a dew point temperature equal to the measured surface temperature of single glazed windows.

i. Wall, Floor, and Ceiling Surface Temperatures — To show the distribution of surface temperature in the test rooms, two days were selected, one when the outdoor temperature was about 12 F and the other when the outdoor temperature was about 35 F. Table 13 shows the observed surface temperatures for various elements of Rooms A and C for these two days. The unheated surface temperatures were considerably less for the day when the outdoor temperature was 12 F than for the day when the outdoor temperature was 35 F.

Table 13 shows that at 24 in. above the floor in Rooms A and C the east wall surface temperatures were greater than the west. This was due to the presence of the supply piping in the east walls of
these rooms. The temperature of the lower half of the east wall also was greater in Room C than in Room A because of the increased water temperature in the supply piping in Room C. In both rooms the temperature of the lower half of the partitions was greater than that of the upper half, probably the result of conduction and radiation from the heated slab to the partitions.

The difference between the air temperature and the area weighted average temperature of all the unheated surfaces of the room (AUST), the average north wall surface temperature (NWST), and the average room air temperature. Even though Rooms C and D had more severe exposures, the AST in these rooms was slightly higher for any given indoor-outdoor temperature difference than in Rooms A and B. This was due to increased panel area and higher temperature of the lower partition surfaces because of the carpeting.

The AST has been established as a parameter which determines the total heat transfer from a panel for a given panel surface temperature and infiltration rate. Relationships between AST, panel surface temperature, infiltration, room air temperature, and total panel heat output have been found at the ASHAE Laboratory. These relationships apply to rooms whose geometry is similar to that which would be encountered in practice.

Since calculation of the AST does not include the heated panel area, the net radiant heat exchange between a body located in the room and its surroundings would depend not only upon the AST, but also the panel surface temperature. Furthermore, the location of the body with respect to the surrounding surfaces must be taken into consideration. “Body temperature depends upon the balance between heat production and heat loss.”

Heat production is determined by the metabolic rate, and heat is lost from the body by convection, evaporation, and radiation. Therefore, for a given degree of activity or metabolic rate, the body surface temperature is determined by the environmental factors affecting the heat exchange between the body and its surroundings. These factors are dry-bulb temperature, humidity, air motion, and surface temperature of the surroundings. If a person were sitting at rest in the center of the test room, his body temperature would then be
determined by these environmental factors. Upon moving away from the center of the test room toward the window, the net radiant heat exchange between the body and surrounding surfaces would change as a result of a change in configuration factor of the body with respect to the surrounding surfaces. The configuration factor of the body with respect to the exposed wall would increase while the configuration factor of the body with respect to the other room elements would decrease. If the air temperature, humidity, and motion were the same near the window as at the center of the room, a lower body surface temperature would result in the position near the window in order that the total heat loss from the body would remain equal to the metabolic rate. The magnitude of this change in body surface temperature would depend upon the change in net radiant heat transfer from the body. This is also affected largely by the type of clothing worn.

Since a change in the net radiant heat transfer from the body would result from a change in location within the room, one constant, such as the AUST or AST, could not represent the MRT (mean radiant temperature) which is an index of the amount of radiation taking place between the body and its environment, referred to a uniform environment. The MRT is a function of location of the body within the test room.

Several instruments have been employed for the measurement of MRT in an environment. These include the eupathoscope, globe thermometer, and thermo-integrator. A thermo-integrator was installed at the center of Room A with the center of the instrument located approximately 30 in. above the floor. Table 15 shows the relationship of the MRT, AUST, and AST for Room A to the indoor-outdoor temperature difference. The AST and MRT were practically the same for all values of indoor-outdoor temperature difference. The AST and MRT decreased slightly with increasing indoor-outdoor temperature difference, while the AUST dropped off rapidly. It is evident from these data that the AST closely approximated the MRT obtained at the center of Room A 30 in. above the floor. It should be pointed out that such may not be the case in other locations.

Since the AST approximated the MRT at the center of the room, the data of Table 15 suggest that a lower room air temperature might be tolerated in Rooms C and D than in Rooms A and B because of higher AST’s for a given indoor-outdoor temperature difference.

### Table 15

<table>
<thead>
<tr>
<th>Temperature Difference, °F</th>
<th>MRT at Center</th>
<th>AST</th>
<th>AUST</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>70.5 ± 0.1</td>
<td>70.1 ± 0.4</td>
<td>70.2 ± 0.2</td>
</tr>
<tr>
<td>40</td>
<td>70.0 ± 0.1</td>
<td>69.6 ± 0.1</td>
<td>68.9 ± 0.1</td>
</tr>
<tr>
<td>60</td>
<td>69.4 ± 0.1</td>
<td>69.6 ± 0.4</td>
<td>67.3 ± 0.2</td>
</tr>
<tr>
<td>80</td>
<td>68.8 ± 0.3</td>
<td>69.3 ± 0.7</td>
<td>66.2 ± 0.4</td>
</tr>
</tbody>
</table>

*Figures in table are for confidence limit of 95% (average values).*

### Table 16

<table>
<thead>
<tr>
<th>Room</th>
<th>Heat Flow to Room</th>
<th>Average Measured Heat Flow</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>1000</td>
<td>1010 ± 61</td>
</tr>
<tr>
<td></td>
<td>2000</td>
<td>2085 ± 131</td>
</tr>
<tr>
<td></td>
<td>3000</td>
<td>3160 ± 131</td>
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<tr>
<td></td>
<td>4000</td>
<td>4230 ± 239</td>
</tr>
<tr>
<td></td>
<td>5000</td>
<td>5305 ± 320</td>
</tr>
<tr>
<td>B</td>
<td>1000</td>
<td>1225 ± 125</td>
</tr>
<tr>
<td></td>
<td>2000</td>
<td>2365 ± 81</td>
</tr>
<tr>
<td></td>
<td>3000</td>
<td>3415 ± 141</td>
</tr>
<tr>
<td></td>
<td>4000</td>
<td>4495 ± 161</td>
</tr>
<tr>
<td></td>
<td>5000</td>
<td>5570 ± 239</td>
</tr>
</tbody>
</table>

*Figures in table are for confidence limit of 95%.*

The air temperature used in these calculations was measured at a distance of 3 in. above the floor at two locations — 2 ft from the north wall and at the center of the room. The air temperatures at these locations depended upon outdoor temperature and differed from one another 2 to 4 F. The AUST was used for the temperature of the surroundings as recommended in the Guide. Table 16 shows the measured panel output versus the calculated output for all four test rooms. In all cases the measured panel output was consistently higher than the calculated panel output. At design conditions of 80 F indoor-outdoor temperature difference, the per cent difference between the calculated and measured panel output was from 7 to 18% (based on the measured output). The per cent difference was greater for the rooms with more severe exposures.

The calculated convection heat transfer from the panel was based on the equation expressing the heat flow from a horizontal flat plate facing upward with no unusual end effects. The presence of the exposed wall in the test rooms caused a downward...
cold air current which passed over the surface of the panel as indicated in Fig. 22. This increased air velocity over the surface of the panel probably resulted in a greater heat flow from the panel by convection for a given surface to air temperature difference than that calculated by equation (1). This could account for part of the difference in the calculated and measured panel output. Also the panel surface temperatures measured in Rooms C and D may not have been accurate because of the difficulty in establishing the actual surface of the carpeting. The heat flow, as determined by heat flow meters in Rooms C and D, was actually the heat flow from the slab to the carpet and pad. Since there was some heat storage in the floor coverings, the heat flow from carpet to room at any instant was not necessarily the same as that measured by the heat flow meters. This may have been a factor influencing the difference in the measured and calculated panel outputs.

The relationship between panel output and panel surface minus room air temperature is shown in Table 17. In Rooms A and B, for which the exposures were similar, the panel outputs for a given panel minus room air temperature difference were approximately the same. The outputs for Rooms C and D, where the exposures were more severe, were 15 to 20% greater than in Rooms A and B at the same panel minus room air temperature difference. It should be pointed out that the same panel minus room air temperature difference in all rooms does not represent the same outdoor conditions since the variation in panel surface minus room air temperature with outdoor temperature was not the same in all test rooms. Variations in panel output were due to variations in exposure, infiltration, and panel area.

At design conditions the panel output for Room A was about 30.4 Btuh per sq ft, while the output of the panel in Room B was about 35.4 Btuh per sq ft—a difference of about 5.0 Btuh per sq ft. Based on the panel area, the difference in panel outputs represents about (116 X 5.0) or 580 Btuh. Calculations indicate that a difference of 0.3 air changes per hr could account for this difference in panel output, but no measure of the infiltration rate was made during these studies.

Data have been presented through the ASHAE(1) which correlated AUST, panel output, panel surface temperature, and infiltration. However, no comparison of results could be made because no attempt was made to control the air temperature in the ASHAE experiments. Because the air temperature was the independent variable in those experiments, the room air temperatures were all lower for a given test as compared to those reported in this paper. As a result, the panel outputs reported in this bulletin are greater for a given panel surface temperature and AUST than those reported in the ASHAE results.

k. Relative Humidity — The relative humidity in each test room was obtained with calibrated hair-type relative humidity indicators. Figure 25 shows a plot of the relative humidity vs. indoor-outdoor temperature difference. A comparison of the relative humidity in Rooms B and D for Test I
indicates that during mild weather (less than 20 F indoor-outdoor temperature difference) the relative humidity was about the same for both rooms. However, as the indoor-outdoor temperature difference increased, the relative humidity in Room B decreased more rapidly than that in Room D. The plot of relative humidity for Room D, Test II, shows that with the heavy weight carpet and rubber pad, the relative humidity was higher than for either room during Test I. Also there appeared to be more scattering of test points.

Throughout the entire test period it was observed, especially during moderately cold weather preceded by damp, warm weather, that condensation occurred on the windows in rooms with carpeting, while the windows in rooms which had no carpeting were dry. This may have been due to the tendency of the carpet to absorb moisture during the warm moist weather and to release it as the slab was heated when the outdoor temperature dropped. However, the fact that the relative humidity in rooms with carpeted panels was consistently higher than that in rooms with uncovered panels even during extended cold weather suggests that there may have been an actual increase in the rate of transfer of water vapor through the concrete floor slabs in the carpeted rooms.

Tests conducted with laboratory specimens of various construction at the Forest Products Laboratory have indicated that the rate of migration of water vapor through floor slabs is proportional to the vapor pressure difference across the slab. Furthermore, the tests indicated that the type of vapor barrier used under the concrete floors in the Floor Slab Laboratory was not very effective in reducing water vapor transfer rates.

In the test rooms with carpeting on the floor it was necessary to operate the heating system with higher slab temperatures than in rooms without carpets. This in turn increased the temperature of the gravel fill under the slab. Considering the air saturated in the voids between pieces of stone constituting the gravel fill, calculations were made for the vapor pressure difference across the slab. Figure 26 shows a plot of this vapor pressure difference for Rooms B and D. The vapor pressure difference in Room D was consistently greater than that in Room B. The smaller motive head for the transfer of vapor through the slab in Room B plus the presence of the asphalt tile which acted as an additional vapor barrier must have accounted for the difference in relative humidities between test rooms.

By establishing a mass balance on the water vapor in Rooms B and D, the resulting relative humidity in the test rooms was calculated for various outdoor temperatures. Condensation and absorption of water vapor by materials in the test rooms as well as the transfer of water vapor through interior partitions and ceilings were omitted from consideration. The resistance to transmission of water vapor by carpets and pads was considered negligible, and the room air temperature was taken to be constant at 72 F. Under steady-state conditions the rate of water vapor transferred through the slab plus the rate of water vapor brought into the room due to infiltration was equal to the rate of water vapor removal from the room due to exfiltration plus the rate of water vapor transferred through the exterior wall. In equation form this mass balance becomes:

\[ W_{in} + M_v(P_s - P_w) + M_w(P_w - P_{w0}) = W_{out} \]  

(9)

where \( W \) = infiltration rate, in lb per hr
\( w_i \) = vapor content of infiltrating air, in lbs of water vapor per lb of dry air
\( w \) = vapor content of exfiltrating air, in lb of water vapor per lb of dry air
\( P_s \) = saturation pressure of water vapor in the voids between pieces of stone constituting the gravel, in \(^*\text{Hg}\)
\( P_w \) = partial pressure of the water vapor in the test room, in \(^*\text{Hg}\)
\[ P_{\text{wo}} = \text{partial pressure of the water vapor in the outdoor air, in } ^\circ\text{Hg} \]

\[ M_s = \text{equivalent permeance of the concrete floor slab, in grains per hr (sq ft) (} ^\circ\text{Hg}) \]

\[ M_w = \text{equivalent permeance of the exterior wall, in grains per hr (sq ft) (} ^\circ\text{Hg}) \]

For similar slab construction the value of \( M_s \) has been given\(^{24}\) as \( 2.45 \text{ gal day} (1000 \text{ sq ft}) (0.345) \text{ psi} \). Converting this value to approximate units \( M_s = 8.2 \text{ grains per hr (sq ft) (} ^\circ\text{Hg}) \). The value of \( M_w \) was calculated from values reported in the \textit{Guide}, and \( M_w = 0.264 \text{ grains per hr (sq ft) (} ^\circ\text{Hg}) \). Since the infiltration rate was unknown, a value of \( \frac{1}{2} \text{ air change per hr} \) was assumed for both test rooms. This was equivalent to 676 cu ft per hr or 50.7 lb per hr standard air. The mass balance for Room D can then be written:

\[ 50.7w_i + (8.2) (169) (P_s - P_w) \]

\[ + (0.264) (81) (P_w - P_{\text{wo}}) \]

\[ = 4354w \frac{P_w}{29.9 - P_w} \]

(10)

where \( w = 4354 \frac{P_w}{29.9 - P_w} \)

In order to determine the condition of the outdoor air, the relative humidity at a given dry-bulb temperature was taken equal to that for a five-year average obtained in Urbana, Illinois.\(^{25}\) For a given outdoor temperature, the value of \( w_i \) was determined and the value of \( P_w \) was taken as the saturation pressure corresponding to the observed temperatures under the slab. Thus the equation was solved for \( P_{\text{wo}} \), and the relative humidity was determined for 72 \text{ F room air dry-bulb temperature.} \)

Figure 27 shows a plot of the indoor relative humidity thus determined vs. indoor-outdoor temperature difference. For Room B the relative humidity with floor panel heating was consistently higher than that obtained with ceiling panel heating. This was due to the increased vapor transfer rate through the floor slab when it was heated. Recorded data in Room B show that the measured relative humidity with ceiling panels was actually less than that obtained with a heated floor panel, which is in agreement with the calculated values of relative humidity. For Room D the calculated relative humidity was consistently much higher than that for Room B due to increased vapor transfer rates through the floor slab. These trends are in agreement with those indicated by Fig. 25. In Room D the relative humidity decreased with decreasing outdoor temperature to a minimum value and thereafter increased. Beyond the minimum value, the vapor transfer rate through the slab was great enough to more than offset the decreased vapor gain due to infiltration to such an extent that the relative humidity in the test room increased. Below the minimum value, the increased gain in vapor due to infiltration of outdoor air with high moisture content more than offset the decreased vapor transfer rate through the slab so that the net result was an increase in the relative humidity in the test room.

A comparison of Figs. 25 and 27 shows that the measured relative humidity was never as high as the calculated. This could have been due to the omission of condensation and absorption terms in the vapor mass balance. Condensation on cold windows definitely occurred during cold weather, as pointed out previously. Also the assumption of 100% relative humidity in the voids of the gravel fill could have been in error. Previous work\(^{25}\) has shown that the relative humidity depends upon the moisture content of the soil under the slab. All of these unaccounted for vapor losses would tend to reduce the relative humidity in the test rooms and bring the calculated and measured values into closer agreement.

The individual terms of equation (9) representing vapor gains and losses have been plotted in Fig. 28. Vapor loss due to exfiltration (which was the only vapor loss included in equation (9)) was equal to the sum of vapor gains due to infiltration and transmission through the slab and exposed
The transmission through the exposed wall is seen to be practically insignificant. Vapor gains due to infiltration decreased with decreasing outdoor temperature as a result of decreased moisture content of the outdoor air. Transmission through the slab increased with decreasing outdoor temperature as a result of increased vapor pressure differences across the slab. The sum of these three curves represents the vapor loss due to exfiltration.

In order to determine the infiltration rate, \( W \), in equation (9) at zero outdoor temperature, the value of relative humidity as obtained from Fig. 25, Test I, Room D, was used. Solving equation (10) for the infiltration rate resulted in a value of about 200 lb per hr or 2 air changes per hr. This value would seem high for infiltration at zero outdoor temperature. However, the \( W \) terms in equation (9) include all the losses in vapor from the test room. Therefore, in the actual case, these terms include condensation, absorption, etc. Using two air changes per hr, the values of each component of equation (9) were plotted in Fig. 28. Both vapor gains due to infiltration and transmission through the slab were higher than those obtained with \( \frac{1}{2} \) air change per hr. Also the vapor loss due to exfiltration was greater than the corresponding value with \( \frac{1}{2} \) air change per hr. The difference in vapor loss for the two cases was about 1050 grains per hr or 0.15 lb per hr. This amount of water vapor could possibly have been due to condensation, absorption, and unaccounted for losses in equation (9).

During the 1953-54 heating season the limit switch settings and the firing rates of heaters serving carpeted rooms were increased. This was done because the results of special pick-up studies indicated that the carpeted panels responded very slowly with limit switches set at 130 F and firing rates equal to 1.33 times the design heat loss. Also the attic temperature was uncontrolled for the 1953-54 studies. Under these conditions during extremely cold weather the vapor pressure difference across the slab with a heavy carpet and heavy pad reached as high as 5.1 in. of Hg. Thus a large motive force was provided for vapor transmission through the slab. This resulted in even higher relative humidity in rooms with carpeted panels than in rooms with bare or tiled panels.

9. Pick-Up Studies — Object and Operating Conditions

The object of these studies was to determine the ability of the floor panel heating systems to pick up the room heat load. The effects of floor coverings on the pick-up rate were studied, and required water temperatures and heat input rates were established for each room.

Two pick-up tests were made, one in November 1952, previous to any normal operation of the panel systems, and the other in January 1953 after the system had been in operation for some time. The attic temperature was controlled at about 60 F, and the corridor and instrument room air temperatures were controlled at 72 F. The heaters were turned off and the windows of all test rooms were left open all night preceding the start of the test in order to allow the room air temperature to drop.

At the start of the test the windows were closed and the water heaters turned on. The initial rate of energy input to the heaters, equivalent to the gross output of a conventional boiler, was equal to the calculated design heat loss of the room plus 33% which represented the normal piping and pick-up allowance as recommended by the Institute of Boiler and Radiator Manufacturers.\(^{(27, 28)}\) The thermostats were shorted out of the control circuits in order to provide continuous operation of the system. As the tests progressed, changes were made as required in the limit switch settings and in the energy input rates to the water heaters.

Since the outdoor temperature varied considerably (from about 14 F to 57 F) during the test, transient effects of this changing temperature had to be considered in the analysis of test results.
10. Pick-Up Studies — Discussion of Test Observations

a. Air, Water, and Floor Surface Temperatures — November 1952 Test — Data taken during the November 1952 test are tabulated in Table 18. Twelve hours after the start the inlet water temperatures in Rooms C and D were high enough (130 F to 135 F) to cause intermittent operation of the heaters due to limit control action. The inlet water temperatures in Rooms A and B did not level off until about 4 hrs later and reached values of 144 F and 132 F, respectively. Intermittent operation of the heaters by action of the limit control resulted in a decrease in the average inlet water temperature while the outlet water temperature leveled off. It also resulted in a decrease in the total rate of energy input to the water. Therefore, for a given water flow rate, the water temperature drop would be expected to decrease, as evidenced by Items 5, 21, 37, and 53 in Table 18.

For all practical purposes, the room air and floor surface temperatures in Rooms A and B became stabilized at their maximum values by the end of the first 24 hrs of operation. Variations occurring after this time could be attributed to variations in the temperature of the outdoor air. Since the water temperatures in Rooms C and D were lower than those in Rooms A and B and because the carpets and pads offered appreciable resistance to heat flow (which increased heat storage in the ground), heat was not transferred to Rooms C and D at rates great enough to maintain increasing room air temperatures after the test has been in progress for about 16 hrs. At this time the room air temperatures in Rooms C and D started to decrease and continued to decrease until about 28 hrs after the start of the test, when the outdoor temperature leveled off and the limit control settings in Rooms C and D were raised to 160 F.

Immediately following the change in the limit control settings in Rooms C and D, 28 hrs after the start of the test, the floor surface temperatures in these rooms began to increase, and the room air temperatures began to increase shortly thereafter. In order to bring the air temperatures in Rooms C and D to approximately the same values as those in Rooms A and B, it was necessary to increase the limit control setting in Room C to 170 F (59 hrs after the start of the test) and to increase the heat input rate to 12,000 Btu/h in both rooms (75 hrs after start of test). After these changes had been made the final room air temperature in Rooms B, C, and D were all about 85 F, while the temperature in Room A was 90 F.

Based on the calculated heat loss of Rooms C and D with heated attics, the final heat input rate of 12,000 Btu/h corresponded to piping and pick-up factors of 1.80 and 1.42 respectively. The normal piping and pick-up factor is 1.33. Thus it was necessary to increase both the piping and pick-up factors and the water temperatures in Rooms C and D in order to take care of the pick-up load and bring these rooms to the same final temperature as the rooms without carpeting on the floors. The required increase was greatest in Room C where the floor was covered by the heavy carpet and 40 oz pad.

b. Ground Temperature — November 1952 Test — At the start of the test the temperature of the ground 2 ft and 3 ft under the test room floors was about 63 F. The ground temperature dropped very slowly for the first 16 hrs of the test, after which the temperature 2 ft below the floor started to increase. The temperature at 3 ft below the floor did not start to increase until the test had been in progress for about 24 hrs. The ground temperature at both levels continued to rise at a very slow rate throughout the remainder of the test. At the end of 96 hrs of operation the temperature of the ground 2 ft below the floor was 67.6 F, 72.4 F, 77.3 F, and 81.7 F under Rooms A, B, C, and D, respectively. The corresponding temperatures at 3 ft below the floor were 64.2 F, 67.9 F, 69.3 F, and 71.6 F.

Because of the extra subfloor insulation in Room A, the ground temperatures under this room were lower than under Room B, even though the water temperature in the floor coils was higher in Room A. The higher water temperatures required in Rooms C and D because of the carpeting on the floor caused corresponding increases in the ground temperature under those rooms. The effect of these increases in ground temperature on operating efficiency is discussed in the following paragraph.

c. Heat Flow — November 1952 Test — At the start of the tests, ground, slab, and room air temperatures were such that there was upward heat flow from the ground to the slab and from the slab to the room air in each of the test rooms. Immediately after the start of the test, the temperature of...
| Time From Start of Test | 0 | 4 | 8 | 12 | 16 | 20 | 24 | 28(a) | 32 | 36 | 40 | 44 | 48 | 52 | 56 | 60 | 64 | 68 | 72 | 76 | 80 | 84 | 88 | 92 | 96 |
|-------------------------|---|---|---|---|----|----|----|------|----|----|----|----|----|----|----|----|----|----|----|----|----|----|----|----|----|---|
| Degrees F               | F | -5 | -1 | 1 | 5 | 9 | 13 | 17 | 21 | 25 | 29 | 33 | 37 | 41 | 45 | 49 | 53 | 57 | 61 | 65 | 69 | 73 | 77 | 81 | 85 |
| Barometer 30 in. Level  |   |   |   |   |   |   |   | 1016 | 1018 | 1020 | 1022 | 1024 | 1026 | 1028 | 1030 | 1032 | 1034 | 1036 | 1038 | 1040 | 1042 | 1044 | 1046 | 1048 | 1050 |
| Average Floor Surface Temp | F | -50 | -46 | -42 | -38 | -34 | -30 | -26 | -22 | -18 | -14 | -10 | -6 | -2 | 0 | 4 | 8 | 12 | 16 | 20 | 24 | 28 | 32 | 36 | 40 | 44 | 48 |
| Ground Temp, 7 ft Below Floor | F | -52 | -48 | -44 | -40 | -36 | -32 | -28 | -24 | -20 | -16 | -12 | -8 | -4 | 0 | 4 | 8 | 12 | 16 | 20 | 24 | 28 | 32 | 36 | 40 | 44 | 48 |
| Average Cold Wall Surface Temp. | F | -60 | -56 | -52 | -48 | -44 | -40 | -36 | -32 | -28 | -24 | -20 | -16 | -12 | -8 | -4 | 0 | 4 | 8 | 12 | 16 | 20 | 24 | 28 | 32 | 36 | 40 | 44 |
| Average Ceiling Surface Temp. | F | -60 | -56 | -52 | -48 | -44 | -40 | -36 | -32 | -28 | -24 | -20 | -16 | -12 | -8 | -4 | 0 | 4 | 8 | 12 | 16 | 20 | 24 | 28 | 32 | 36 | 40 | 44 |
| Heat Flow From Sides of Slab | Btu/hr ft² | 350 | 304 | 258 | 212 | 166 | 120 | 74 | 28 | 8 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Heat Flow From Bottom of Slab | Btu/hr ft² | 1080 | 974 | 868 | 762 | 656 | 550 | 444 | 338 | 232 | 126 | 120 | 114 | 108 | 102 | 96 | 90 | 84 | 78 | 72 | 66 | 60 | 54 | 48 | 42 | 36 | 30 | 24 |
| Heat Flow From Top of Slab | Btu/hr ft² | 393 | 355 | 317 | 279 | 241 | 203 | 165 | 127 | 90 | 53 | 40 | 28 | 15 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Heat Flow From Bottom of Slab | Btu/hr ft² | 217 | 199 | 181 | 163 | 145 | 127 | 109 | 91 | 73 | 55 | 37 | 19 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |

**NOTES:**
(a) 28 hours - Normal limit switch settings to 160 F in Room C and D.
(b) 59 hours - Normal limit switch setting to 180 F in Room C.

**FIGURE:**
- **Heat Flow From Slab:** Heat flow from slab calculated based on measured temperature differences.
- **Heat Flow From Sides of Slab:** Heat flow from sides of slab calculated using floor surface temperature differences.
- **Heat Flow From Bottom of Slab:** Heat flow from bottom of slab calculated using temperature differences.
- **Heat Flow From Top of Slab:** Heat flow from top of slab calculated using temperature differences.

- **Drop in Water Temperature:** Drop in water temperature calculated using room temperature differences.
- **Barometer:** Barometer reading in inches of mercury.

**TABLE 18**
- **Pick-up Test—Nov. 1952**
- **ILLINOIS ENGINEERING EXPERIMENT STATION**
the floor slabs started to increase. This resulted in an increased upward heat flow from the slab to the room air and a reversal in the flow of heat between the slab and ground. The rates of heat flow upward, downward, and from the edge of the slabs increased rapidly for the first 24 hrs of the test. During the remainder of the test the changes were less pronounced except for the effects of changes made in the setting of the limit controls and heat input rates to the water in Rooms C and D.

During the first 12 hrs of operation the percentage heat flow to the test rooms decreased for all rooms, but the decrease was much less pronounced in Room A. Thereafter, the percentage heat flow leveled off with the following approximate values at the end of the test:

<table>
<thead>
<tr>
<th>Direction of Heat Flow</th>
<th>Room A</th>
<th>Room B</th>
<th>Room C</th>
<th>Room D</th>
</tr>
</thead>
<tbody>
<tr>
<td>Up to Room</td>
<td>75</td>
<td>70</td>
<td>55</td>
<td>60</td>
</tr>
<tr>
<td>Down to Ground</td>
<td>20</td>
<td>25</td>
<td>40</td>
<td>35</td>
</tr>
<tr>
<td>Horizontal to Out-door Air (Edge)</td>
<td>5</td>
<td>5</td>
<td>5</td>
<td>5</td>
</tr>
</tbody>
</table>

A comparison of Items 16, 32, 48, and 64 in Table 18 indicates that for the first 24 hrs of operation, the L-type and vertical type of insulation (Rooms B, C, and D) were comparable, whereas the insulation under the entire floor (Room A) was 20 to 25% more effective than the others in reducing subfloor and edge losses. A reduction in subfloor losses must result in a corresponding increase in useful heat flow to the room. It should be noted that at the start of the test, the water temperatures in all rooms increased while the heat flow from the top of the slab in Rooms B, C, and D (expressed in percentage of total heat flow) decreased at a rate considerably greater than that in Room A. Thus, the insulating effect of the floor coverings and lack of insulation under the slab retarded the heat flow from the floor panel to the room air. At the end of the test the heat flow upward in Room B was within 5% of that in Room A. However, it was necessary to increase both water temperature and heat input rates in rooms with carpets in order to obtain sufficient above-floor heat flow.

It must be remembered that these values indicating the effectiveness of insulation under the entire floor in reducing subfloor losses and the retarding of upward heat flow by floor coverings apply only to pick-up operating conditions. In Section 7b it was shown that insulation under the entire floor increased normal operating efficiency by only about 3%, and in Section 8a it was shown that floor coverings had almost no effect on seasonal fuel costs when room temperature was maintained at a constant value.

d. January 1953 Test — Data taken during the January 1953 pick-up test are tabulated in Table 19. The ground temperatures at the start of this pick-up study at the 3 ft depth for Rooms A, B, C, and D were 63 F, 69 F, 79 F, and 75 F, respectively. The corresponding ground temperatures at the 2 ft depth were 65 F, 71 F, 83 F, and 81 F, respectively. The ground temperatures under Room C were higher than those in Room D because during the normal operation of the heating system prior to the pick-up studies, the floors were covered with a light weight carpet and rubber pad in Room D and a heavy weight carpet and heavy pad in Room C. Thus higher water temperatures were required in Room C. The ground temperatures were lower under Room A than under Room B because Room A had insulation under the entire floor, while Room B had the vertical type of insulation.

For the January test the ground temperature at the 3-ft depth under all rooms remained relatively constant, while the temperature at the 2-ft depth increased as the test progressed.

The room air temperature, water temperature, heat flow, and floor surface temperatures exhibited characteristics very similar to those for the November pick-up study. However, all of these increased at greater rates during studies in January, because the ground was warmer at the start of this test. The outdoor air temperature characteristic was very similar to that for the previous tests, yet the room air temperature increased much faster for this study than it did for the previous one.

It was found that for both Rooms A and C the above floor heat flow ranged from 1000 to 2000 Btu/h greater in the January test than in the November test. The below floor heat flows were about the same in both tests even though the ground temperatures were greater at the beginning of the January studies.

For both Rooms A and C the above floor heat flow during the first 24 hrs of operation was greater for the January test than for the November test by 10 to 15 percentage points. After about 60 hrs of operation, the per cent above floor heat flows for the two studies were almost equal. For the
### Table 19
Pick-up Test — Jan. 1953

<table>
<thead>
<tr>
<th>1. Time from Start of Test (Hours)</th>
<th>2. Outdoor Air Temperature (°F)</th>
<th>3. Inlet Water Temperature (°F)</th>
<th>4. Outlet Water Temperature (°F)</th>
<th>5. Drop In Water Temperature (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>6. Room Air Temp., 30 in. Level</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8. Ground Temp., 2 ft below floor</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>9. Ground Temp., 3 ft below floor</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>10. Average Cold Surf. Temp.</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>12. Heat Flow from Top of Slab</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>13. Heat Flow from Edge of Slab</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>14. Heat Flow from Bottom of Slab</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>15. Heat Flow from Slab, Total</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>16. Heat Flow from Top of Slab</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>17. Heat Flow from Edge of Slab</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>18. Heat Flow from Bottom of Slab</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Room A

<table>
<thead>
<tr>
<th>Time from Start of Test</th>
<th>Outdoor Air Temperature</th>
<th>Inlet Water Temperature</th>
<th>Outlet Water Temperature</th>
<th>Drop In Water Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>33.2</td>
<td>41.8</td>
<td>36.3</td>
<td>25.7</td>
</tr>
<tr>
<td>28</td>
<td>28.4</td>
<td>31.1</td>
<td>28.5</td>
<td>19.0</td>
</tr>
</tbody>
</table>

### Room B

<table>
<thead>
<tr>
<th>Time from Start of Test</th>
<th>Outdoor Air Temperature</th>
<th>Inlet Water Temperature</th>
<th>Outlet Water Temperature</th>
<th>Drop In Water Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>30</td>
<td>86.0</td>
<td>113.5</td>
<td>132.8</td>
<td>135.2</td>
</tr>
<tr>
<td>36</td>
<td>61.8</td>
<td>87.0</td>
<td>111.9</td>
<td>116.1</td>
</tr>
<tr>
<td>60</td>
<td>22.2</td>
<td>26.5</td>
<td>24.7</td>
<td>25.3</td>
</tr>
</tbody>
</table>

### Notes

(a) 24 hours from start of test — Baked limit switch settings in Rooms A and B
(b) 33 hours from start of test — Increased "firing rates" in Rooms C and D to 220 volts.
first 24 hrs of operation the below floor heat flow in Room C reached a maximum of about 43% in the November study, while the maximum during this period of time for the January study was only 32%. In Room A the difference in the maximum heat flows to the ground for the first 24 hrs of operation during November and January tests was about 7%.

The above floor heat flows in Rooms B, C, and D for the January test were all within 5 to 10 percentage points of each other, while the above floor heat flow for Room A was about 10 to 15 percentage points higher than for the other rooms after 12 hrs of operation. Thus the subfloor insulation in Room A was again shown to be more valuable during pick-up operation than the type used in Rooms B or C.
III. CEILING PANEL TESTS

11. Operating Conditions

The following two series of ceiling panel tests were conducted in Rooms A and B:

Series C-1 in which the attic temperature was maintained at a constant value of 60 F (1952-53 heating season).

Series C-2 in which the attic temperature was uncontrolled and was allowed to change with changes in the outdoor temperature and general weather conditions (1953-54 heating season). Figure 29 shows a plot of the average attic air temperature above Room D during Series C-2 which is representative of the attic air temperature above the other three test rooms.

The heating systems and test equipment used in ceiling panel tests have been described in Sections 2, 3, and 5. Except for the panels themselves, the heating equipment was the same as that used for the tests on the floor panels. Continuous circulation of water through the ceiling panels was employed, while the temperature of the water was varied according to need by intermittent operation of the water heaters. The heat input rate to the water heaters was made to equal 1.33 times the calculated design heat loss of the room, and the heaters were turned off and on as required by the same thermostats as were used for the floor panel tests. Corridor floor and corridor air temperatures were maintained approximately the same as the adjacent test room floor and air temperatures, respectively.

12. Energy Input (Fuel Consumption)

The total energy inputs to the ceiling panel systems in each room were obtained for 24-hr periods by watt-hour meters. In Fig. 30 the heat equivalents of these measured energy inputs (fuel consumption) are plotted against the corresponding indoor-outdoor temperature differences for both Rooms A and B. Curves from Fig. 15 representing the daily heat input rates required for floor panel heating in these rooms are shown as dotted lines for comparative purposes.

In Room A, in which the heating coils were located above the metal lath, there was no significant difference in the heat input rates for Series C-1 and C-2. However, at an indoor-outdoor temperature difference of approximately 45 to 50 F, the heat input rate tended to level off. The limit control was adjusted to limit the water temperature to 140 F. In Room A this temperature was reached at an indoor-outdoor temperature difference of approximately 45 F, and, therefore, at higher temperature differences the operation of the water heater was controlled by the limit switch rather than the thermostat. The panel output to the room under these conditions was not sufficient to offset the room heat loss, and the room temperature dropped below the setting of the room thermostat.

In Room B, in which the heating coils were located below the metal lath, there was no apparent "leveling off" of the heat input rate curve, since the water temperature did not reach the value of the limit switch setting (140 F) within the range.
of test conditions. Furthermore, the panel output was sufficient to offset the room heat loss and the room air temperature remained equal to the thermostat setting even during the coldest weather experienced (67°F indoor-outdoor temperature difference).

Extrapolating the curves for Series C-1 and C-2 in Room B indicates that at design conditions (80°F indoor-outdoor temperature difference), the heat input rate for Series C-1 was 5950 Btuh, while for Series C-2 it was 6950 Btuh, an increase of 17%. Since the room heat loss below the ceiling was the same for both series of tests, this increase was due entirely to increased heat losses from the ceiling to the attic.

The energy input for ceiling panel heating was always larger than that required for the floor panels with the difference in Room A being greater than that in Room B over the range of indoor-outdoor temperature differences from 0 to 45°F. Beyond this value of indoor-outdoor temperature difference no comparison in Room A can be made because the ceiling panel could no longer supply the total heat requirement of the room. For Room B the percentage increase in heat input rate at design conditions for the ceiling panel over that for the floor panel, both operating with heated attic, was 450 Btuh (5950 – 5500) or 8%.

In order to make a comparison of the annual fuel consumption for floor and ceiling panels in
Room B, the data for Series C-1 and floor panel studies were integrated over a representative heating season following the same procedure used in Section 7b to construct Table 5. The results are shown in Table 20.

The total annual fuel consumption for the ceiling panels was about 113.6 therms, while that for the floor panel studies was about 89.4 therms. The difference was 24.2 therms or about 27%, based on the fuel consumption with floor panels.

13. Distribution of Heat Flow and Ceiling Surface Temperature

The heat flow distribution from the panels to the room was obtained by plotting the instantaneous heat flow as measured by the heat meters against the position of the meters on the panel. A smooth curve was fitted to the test points. Observed ceiling surface temperatures were plotted in the same manner. Figure 31 shows a typical plot of this type for a test made when the heaters were on in both rooms.

A comparison of the ceiling surface temperatures shows that in Room B the variation in temperature from a point directly under a coil to one between coils was of the order of 10 F, while in Room A it was only 1 F to 2 F. Also the mean surface temperature of the ceiling was much higher in Room B than in Room A. These differences were due to the better embedment of the coil in the plaster in Room B.

The mean ceiling surface temperature for both Room A and Room B became greater near the center of the room than at the exposed wall and decreased rapidly beyond the middle of circuit "B." However, the variation in the mean temperature across the effective panel area was less in Room A than in Room B.

The heat flow characteristic for Room B was similar to the surface temperature characteristic for that room. However, in Room A the heat flow dropped off rapidly near the exposed wall, while the mean surface temperature decreased only slightly. The reason for the rapid reduction in the rate of heat flow from the panel to the room near the north wall in Room A is not apparent, but it is believed to be related to the relatively low effective panel surface temperature and possibly to the nature of the air circulation in the room.

14. Heat Flow from Panels

Figure 32 shows a plot of heat flow from panel to room vs. indoor-outdoor temperature difference. The distribution of test points representing Series C-1 and Series C-2 in Room A approximated a straight line for indoor-outdoor temperature differences of 45 F and less, while above 45 F indoor-outdoor temperature difference the heat flow "levelled off", at a maximum value. This characteristic
There was no appreciable difference in the test points for Series C-1 and C-2 in either Room A or Room B, indicating that the heat flow from panel to room was independent of the attic conditions.

The heat flow from the floor panel to the room included not only the heat lost by infiltration and transmission through the walls and glass, but also the heat lost by transmission through the unheated ceiling; the heat flow from the ceiling panel to the room included floor heat losses in addition to infiltration and wall and glass transmission losses. Therefore, assuming that the infiltration and transmission losses through the walls and windows were the same for both the floor and ceiling panel studies, the difference in the total heat flow from the panel to the room, as obtained for floor and ceiling panels, represented the difference in the ceiling losses when using the floor panel and the floor
losses when using the ceiling panel. For design conditions in Room B this difference was \((4260 - 3600) = 660\) Btuh. The calculated heat loss through the unheated ceiling was 1080 Btuh (Table 1). If the calculated loss through the unheated ceiling is assumed to be substantially correct, the actual floor loss in Room B at design conditions was only about 420 Btuh or about 53% of the calculated value.

Assuming the heat losses from the insulated piping and heater to be negligible, the difference in the heat supplied to the water and the heat output of the panel to the room was equal to the heat exchange between the ceiling panel and the attic. For Series C-2, total heat input to the water in Room B was 6800 Btuh, while the total heat flow from the panel to the room was 3600 Btuh. Thus \((6900 - 3600) = 3300\) Btuh represents the reverse heat loss from the panel to the uncontrolled attic air. The ratio of upward heat flow to downward heat flow from the panel was 3300/3600 = 0.92. At design conditions 3300/6800, or about 48%, of the heat input to the water was lost to the attic. For tests with controlled attic temperature, 5900 Btuh was the total heat input to the water for Room B at design conditions. Thus \((5900 - 3600) = 2300\) Btuh represents the reverse loss from the panel to the attic. The ratio of upward to downward heat flow was 2300/3600 = 0.64 and 2300/5900, or about 39% of the heat input to the water was lost to the controlled attic air.

An analysis similar to the one just made for Room B was not attempted for Room A since the design room heat loss could not be satisfied by the ceiling panel with uncontrolled attic temperature.

Figure 33 shows a plot of the heat flow from the panel to Room B expressed as percentage of the total panel output. At indoor-outdoor temperature differences greater than 30 F, the percentage of heat flow to the room was practically constant at 50% for Series C-2, while that for Series C-1 varied between 50 and 60%.

15. Water Temperatures

The water temperatures at the inlet and outlet of each coil were recorded continuously on recording potentiometers. The average temperature of the water in the coil was taken as the arithmetic mean of the average inlet and outlet water temperatures.

Figure 34 shows a plot of the average water temperature in Rooms A and B vs. the indoor-outdoor temperature difference for both Series C-1 and C-2. The plot representing Series C-1 in Room A shows that the water temperature increased as the outdoor temperature decreased and reached 140 F at 54 F indoor-outdoor temperature difference. Two tests with a water temperature in excess of 140 F were obtained because the limit switch temperature setting was quite high and allowed the water temperature to reach an average maximum value of 151 F. Normally, the water temperature was limited to 140 F. The plot of data for Series C-2 indicates that the water temperature in Room A reached the limit switch setting of 140 F at about 45 F indoor-outdoor temperature difference. Beyond this point the water temperature remained relatively constant.

The plot of average water temperature vs. indoor-outdoor temperature difference for Room B shows that for a given outdoor condition, the required water temperatures were lower than those in Room A. The average water temperature increased with increasing indoor-outdoor temperature difference. The curve fitted to the data of Series C-2 was slightly steeper than that fitted to the data of Series C-1, indicating that somewhat higher water temperatures were required with an uncontrolled attic temperature than without.

The more rapid increase in water temperatures in Series C-2 was due to the increased losses from the panel to the attic when the attic temperatures were uncontrolled. This is consistent with the observed effect of attic temperature on heat flow rates from, and energy input to, the ceiling panels.

16. Room Conditions

a. Room Air Temperatures — The room air temperature obtained with ceiling panels was not as uniform as that obtained with floor panels. In Fig. 35 the maximum temperature difference meas-
Fig. 34. Average Water Temperatures for Ceiling Panels

Ceiling panel, series C-1
Controlled attic temperature

Ceiling panel, series C-2
Uncontrolled attic temperature

Room A
Room B

Indoor-outdoor temperature difference, deg F
Average water temperature, deg F
ured at the center of Rooms A and B between the levels of 3 in. above the floor and 3 in. below the ceiling has been plotted against the indoor-outdoor temperature difference (Series C-1). Both curves show that as the indoor-outdoor temperature difference increased the maximum room air temperature difference became larger, and at an indoor-outdoor temperature difference of 70 F, the maximum difference in room air temperature measured between the levels of 3 in. above the floor and 3 in. below the ceiling was approximately 10 F.

Figure 36 shows a plot of the room air temperature gradient measured at the center of Rooms A and B for Series C-1. These have been corrected to a temperature of 70 F at the 30-in. level for comparative purposes. When the outdoor temperature was 17 F, there was over a 3 F difference in temperature between the 12-in. and the 60-in. levels above the floor. There was approximately a 4 F difference in temperature from the level of 60 in. above the floor to 3 in. below the ceiling. The room air temperature gradients for Series C-2 were similar to those in Fig. 36, except for Room A in which the room air temperature dropped in cold weather because of the inability of the ceiling panel to supply sufficient heat to offset the room heat loss.

The steep temperature gradient from the floor to the 30-in. level in Room A was the result of having the thermo-integrator located close to the thermocouple at the 30-in. level in that room. Heat from the thermo-integrator affected the reading of the 30-in level temperature.

In Fig. 37 the room air isotherms along the north-south centerlines of Rooms B and C have been plotted for a test conducted when the outdoor temperature was about 19 F. At the time of this test Room B was heated by a ceiling panel and Room C by a floor panel. The figure shows that in Room B, immediately under the effective ceiling panel area, the isotherms were very close together, suggesting that a layer of stagnant air occupied this space. Heat transferred from the ceiling panel to the air was accomplished primarily by conduction through this air film. Below the 7-ft 6-in. level the room air isotherms became farther apart. At the center of the room there was a 6 F difference in temperature from the floor to 1 ft below the ceiling. Nearer the exposed wall the isotherms again become close together as a result of the cooling effect this wall and window had on the room air.

In Room C, heated by a floor panel, the room air temperatures were much more uniform than those produced by the ceiling panel. The only place where there was a large temperature gradient was near the north wall and window. Beyond a distance
of 2 ft from the north wall the room air temperature was very uniform at 70°F.

b. Room Air Temperature Control — The room air temperature at the thermostat was recorded continuously, and the maximum and minimum temperatures occurring for 24-hr periods were determined.

Table 21 shows the maximum, minimum, and maximum minus minimum room air temperature, along with the respective standard deviations from the mean. A comparison of the data for Rooms A and B shows that the location of the coils with respect to the lath in the ceiling panel had no significant effect on the ability of the system to maintain constant room air temperatures. Moreover, by comparing data for the ceiling panels, as shown in Table 21, with those for floor panels with tile covering, as shown in Table 11, it is found that both types of heating systems displayed the same control characteristics (see Section 8e). Both provided good room air temperature control with temperature variations in the order of magnitude of 1 F for the ceiling panels and 1 to 2 F for the floor panels.

c. Surface Temperature — The AUST and AST were determined for Rooms A and B for each daily reading of surface temperature. Lines 2, 3, 4, and 5 of Table 22 show the difference between the average room air temperature and the average AUST in Rooms A and B at several indoor-outdoor temperature differences and for Series C-1 and C-2. In both rooms and for both test series, the difference between the AUST and room air temperature remained practically constant at all outdoor conditions. For Room B the AUST was approximately 3/16 F below the room air temperature, while for Room A it was about 1 to 1 1/4 F lower.

The fact that the panel surface temperature was lower in Room A than in Room B would explain the slightly lower observed AUST in Room A.

For floor panels the difference in room air temperature minus AUST (Table 14) increased with increasing indoor-outdoor temperature difference because the decrease in wall and window surface temperatures more than offset the relatively small increase in floor panel surface temperature. Such was not the case with ceiling panels. The increase in ceiling panel surface temperature was sufficient to offset the reduction in wall and window temperatures, and, therefore, the room air temperature minus the AUST remained constant.

The AST minus room air temperature is shown for both Series C-1 and C-2 in lines 6, 7, 8, and 9 of Table 22. In Room A this difference was practically zero for both test series, while in Room B it increased with increasing indoor-outdoor temperature difference. At 80°F indoor-outdoor temperature difference (design conditions in Urbana, Illinois) the AST in Room B was about 3 to 4 F greater than the room air temperature.

The coils were located above the metal lath in the ceiling panel used in Room A, and this construction resulted in lower panel surface temperature than that obtained in Room B where the coils...
were below the lath. This in turn resulted in both lower AST and room air temperature in Room A than in Room B.

Observations taken in Rooms A and B when operating with the floor panels showed that the room air temperature remained practically constant at from zero to 1 F above the AST (about the same as that obtained in Room A with the ceiling panel). The increase in the ceiling panel surface temperature in Room B and the accompanying increase in radiant heat exchange with surrounding surfaces resulted in a higher AST than obtained with the floor panel. Therefore, for the same heat balance on a body in the test room, it is probable that a lower room air temperature would have to be maintained in a room when using a ceiling panel such as that used in Room B than when using a floor panel.

During test Series C-1 a thermo-integrator was installed in a vertical position at the center of Room A with the center of the instrument approximately 30 in. above the floor. The instrument was read each day along with the room surface temperature readings. Figure 38 shows a plot of the MRT obtained with the thermo-integrator vs. the corresponding indoor-outdoor temperature difference. The MRT increased slightly with the ceiling panel with decreasing outdoor temperature and ranged from 72 F to 73 F (slightly higher than the air temperature).

With the floor panel the MRT decreased with decreasing outdoor temperature and ranged from 70.5 F to 69.0 F (slightly less than the room air temperature). Thus the MRT obtained with the ceiling panels in Room A was from 1.5 F to 3.5 F higher than that obtained with the floor panel. Since the ceiling panel surface temperature in Room A was limited because of the poor thermal characteristics of the ceiling, it is probable that in Room B the MRT obtained with the ceiling panel would exceed that in Room A, and the slope of the ceiling panel curve of Fig. 38 would be even greater.

It is evident that because of higher surface temperatures the ceiling panel transferred a greater quantity of heat by radiation than did the floor panel. The ceiling panel behaved less like a con-
vection system than did the floor panel. According to Raber and Hutchinson,\(^{(30)}\) the greater the value of MRT minus room air temperature, the greater the radiant heat transfer and the more a system behaves like a radiant panel.

The ability of a ceiling panel to produce comfortable room conditions in a room having a concrete slab floor is dependent upon the floor surface temperature produced. Average floor surface temperatures were obtained in Rooms A and B for both Series C-1 and Series C-2 by taking the arithmetic mean of the floor surface readings each day. Figure 39 shows a plot of these average floor surface temperatures vs. indoor-outdoor temperature differences. According to Raber and Hutchinson,\(^{(30)}\) the greater the value of MRT minus room air temperature, the greater the radiant heat transfer and the more a system behaves like a radiant panel.

Table 23 shows the floor surface temperatures measured along the center lines of Rooms A and B for Series C-1 and C-2. For all tests the floor surface temperature was lower near the north (exposed) wall than at the center of the room. In all tests except Series C-2, Room A, the outdoor temperature had little effect on the floor surface temperature at distances greater than about 3 ft from the north wall. In Room A, during Series C-2, the lower the outdoor temperature, the lower was the floor surface temperature at all locations along the center line of the room. In Room B the floor surface temperature was slightly higher during Series C-2 than during Series C-1. The increase in floor surface temperature may have been due to higher panel surface temperatures for Series C-2, or more likely the warmer floor during Series C-2 resulted from the fact that floor panel studies were made in this room prior to running Series C-2 and insufficient time may have been allowed for ground temperatures to return completely to normal.

At about 3 ft from the north wall the floor surface temperature was very nearly the same as the room air temperature at the 30-in. level, and at distances between 4 or 5 ft and 9 or 10 ft from the north wall, the floor surface temperature exceeded the room air temperature at the 30-in. level. However, the floor surface temperatures were rather low (68 F or less) near the exposed walls. While ceiling panels were able to maintain warm floors over most of the room area, they could not adequately heat the floor along the exposed walls as did the floor panels.

d. Relative Humidity — The relative humidity of all four test rooms was obtained for each reading.
time by means of hair-type relative humidity indicators. The relative humidity for covered floor panels was discussed in Section 8k. Figure 40 shows a plot of relative humidity vs. indoor-outdoor temperature difference for the ceiling panel heated rooms. In both Rooms A and B the relative humidity decreased with decreasing outdoor temperature, and during extremely cold weather the relative humidity dropped below 10% at any time. The reason for this was the differences in rate of vapor transfer from the ground through the floor slab. Since the ground was unheated under ceiling panel heated rooms, the motive head for the transfer of water vapor through the

slab was considerably less than for floor panel heated rooms. During moderate weather (30 F indoor-outdoor temperature difference) the relative humidity in ceiling panel heated rooms was only 5 to 10% lower than that in floor panel heated rooms.

During extremely cold weather, the low relative humidity in ceiling panel heated rooms, combined with large variations in dry bulb temperature from floor to ceiling, could result in rather uncomfortable conditions. This would be especially true in cases where the occupants were sensitive to low relative humidity (allergy, asthmatic, etc.). Also unless the moisture content of the wood used on the interior of a new house was low at the time of construction, damage could result from drying out and warping.
IV. REFERENCES


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