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Hydronic Heating and Cooling with Valance Units

by

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ENGINEERING EXPERIMENT STATION BULLETIN NO. 466
ABSTRACT

This bulletin, one of a series of Engineering Experiment Station publications reporting research conducted under a cooperative agreement between the Institute of Boiler and Radiator Manufacturers and the University of Illinois, describes tests made in the I=B=R Hydronic Research House during 1960 and 1961 on a system designed to both heat the house in winter and cool it in summer using the same units and piping for both seasons. Low installation cost would be an obvious advantage of such a system.

The I=B=R Hydronic Research House is a tri-level home having a total floor area of approximately 1,600 square feet. It has a minimum of insulation and is operated without storm sash. Under these conditions any potential weakness in the performance of a heating or cooling system is amplified. The design heating load for an indoor temperature of 70° F. and an outdoor temperature of −10° F. is 76.01 M.b.h. The total design cooling load for indoor and outdoor temperatures of 75° F. and 95° F. respectively is 32.22 M.b.h.

Whenever the same unit is used to both heat and cool a room, one is faced with the problem of deciding whether to select the unit on the basis of the calculated design heating load, calculated design cooling load, or on some combination of these two. Since, in Urbana, Illinois, the winter season is much longer and more severe than the summer cooling season, it was decided to select the room units on the basis of the design winter heating loads.

The room heating units consisted of a fin tube section, hanger, cover, and trough for collecting condensate removed from the air during summer operation. These units were designed to be installed along outside walls near the ceiling. Air circulation through these units was by gravity both summer and winter. In the summer chilled water was circulated through the tube whereas heated water was used in the winter season. The same piping system was used for both seasons except that provision was made to prevent circulation of chilled water through the boiler in summer and heated water through the chiller in winter.

The piping arrangement used was a three-zone, series-connected system. Zoning was by house levels. All thermostats were located thirty inches above the floor.

A 5-horsepower water chiller was used for summer operation. The compressor and evaporator sections of the chiller were located in the boiler room, while the air cooled condenser was located outdoors at the rear of the house.

During the winter a sectional cast iron boiler designed for gas firing was used. The boiler had a net I=B=R water rating of 90,000 B.t.u.h.

The design system water temperatures were 40° F. for summer operation and 215° F. for winter operation.

Four series of tests were made; two in the summer and the remaining two during the winter. Each series of tests was continued until data were obtained over a wide range of outdoor temperatures and general weather conditions. The following conditions were common to all four series:

All windows were closed at all times. Outside doors were closed except for periods when persons were in the act of entering or leaving the house. Room doors were in the open position at all times. Draperies were pulled to the side of the glass area as far as they would go and remained in this position at all times.

The two series made during summer weather were designated as Series B-60 and C-60. Test conditions for these two series over and above those conditions listed in the preceding paragraph were as follows:

Series B-60 — The three zone thermostats were set to maintain an average air temperature of 75° F. at a height of 30 inches above the floor in
the room in which the thermostat was located. Crawl space vents were open.

Series C-60 — All test conditions for this series were the same as for Series B-60 except that plastic curtains were installed at the top and bottom of the staircase. These curtains were sealed to the walls, floor, and ceiling so that no air could circulate from one house level to another through the staircase.

The two remaining test series represented winter conditions of operation. These series were designated as Series E-60 and F-60. In these series, the following test conditions prevailed in addition to those which were common to all four test series:

Series E-60 — Each of the three zone thermostats was set to maintain an average air temperature of 73°F. 30 inches above the floor in the room in which the thermostat was located. The high limit control in the boiler was set at 225°F. Crawl space vents were closed during all winter tests.

Series F-60 — All test conditions for this series were the same as for Series E-60 except that the plastic curtains were installed in the same manner as in Series C-60.

Basically the test procedure was the same for all four test series. Each test was 24 hours in length and the test day started and ended at 8:00 a.m.

Key water and refrigerant temperatures were constantly recorded by means of recording potentiometers. Recording instruments also provided continuous records of the heat meter readings; outdoor air temperatures; room air temperatures in the living room, bedroom Number 1, and the recreation room; air temperature in the attic above the dining room and in the crawl space; water flow rates through the different zones and static pressure in the system.

Many other observations were made manually four times per day; at 8:00 a.m., 1:00 p.m., 4:00 p.m., and 10:00 p.m. A few observations such as comfort votes, occupancy, CO₂ content, and the rate of flow of the flue gas were taken at less frequent periods of time.

The results of this study indicated that this system had excellent summer performance characteristics. In the winter its operating characteristics were as satisfactory as those of a ceiling panel system. It produced warmer floor surface temperatures than did a baseboard system; however, in some cases it did not prevent the movement of cool air from the windows across the floor, nor did it keep the air 3 inches above the floor as warm as did the baseboard system. This was particularly true on the second level of the house.

The results also indicated that while the ratio of summer and winter design loads for each level of the house was reasonably constant, air movement up and down the staircase and heat transmission through the floor of the third level caused the actual summer load on the third level to be well in excess of the calculated cooling load, while the actual winter load for this level was well below the calculated heating load. As a consequence, it was not possible to obtain good balance the year round when using the same units for both heating and cooling.
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I. INTRODUCTION

A. PRELIMINARY STATEMENT

This bulletin is one of a series of Engineering Experiment Station publications reporting research conducted under a cooperative agreement between the Institute of Boiler and Radiator Manufacturers and the University of Illinois. This agreement was originally approved in 1940, and under its terms the institute is represented by a research committee composed of engineers active in the heating industry. One function of this committee is to set forth problems for investigation which are of greatest concern to manufacturers and installers of steam and water heating and air conditioning equipment. The Engineering Experiment Station staff selects those problems which can best be studied with facilities available at the University. Funds for defraying a major part of the expense are provided by the Institute.

Studies of cooling systems adaptable to homes heated in winter with hot water systems have been included in this program since the summer of 1953. Results of investigations prior to 1960 have been reported by the University of Illinois(1, 2, 3, 4)* and in papers(5, 6, 7) appearing in technical journals. Earlier results have shown several satisfactory ways of accomplishing this but, in general, these involve the use of two separate systems. This bulletin describes tests made in the I=B=R Hydronic Research House during 1960 and 1961 on a system designed to both heat in winter and cool in summer using the same room units and piping for both seasons. Low installation cost would be an obvious advantage of such a system.

B. ACKNOWLEDGMENTS

This bulletin is a result of a cooperative investigation jointly sponsored by the University of Illinois Engineering Experiment Station and the Institute of Boiler and Radiator Manufacturers. The investigation was carried on as a project of the Department of Mechanical Engineering under the administrative direction of Professor N. A. Parker, department head. Acknowledgment is made to organizations which furnished equipment and materials, and to C. Fan, research assistant, and W. J. Graham, instrument technician, for their aid in setting up test equipment, conducting tests, and analyzing data. The assistance of Mr. G. R. Sward should also be acknowledged. Heating and cooling outputs of the valance assembly were determined under his direction in the I=B=R Laboratory.

C. OBJECT OF INVESTIGATION

The object of this investigation was to observe both heating and cooling characteristics of hydronic systems using room units located near the ceiling (hereafter referred to as valance systems) and to compare these with the performance characteristics of the baseboard heating systems and cooling systems which were previously investigated.

Performance characteristics included room air temperatures and variations, indoor relative humidity, and air motion created by the system, as well as installation and operating costs.
II. DESCRIPTION OF EQUIPMENT

A. I=B=R HYDRONIC RESEARCH HOUSE

Since this is the first bulletin to contain results obtained in the I=B=R Hydronic Research House, it is fitting that this report contain a description of the house and some of the reasons for building it.

The construction and style of a house have distinct effects on the performance of heating and air conditioning equipment. Therefore, it is important that houses used for heating and air conditioning research be representative of current trends. Furthermore, the design of a research house should be such that the more difficult heating and air conditioning problems will be encountered.

It was to meet these conditions that on September 5, 1958, ground was broken for the construction of a tri-level house to replace the I=B=R Research Home built in 1940 as the laboratory in which to study the performance of hydronic heating and cooling systems. Figure 1 shows the exterior of the house, and floor plans are shown in Figure 2.

The tri-level house design in itself presents some problems not encountered in conventional one- and two-story house constructions. Liberal use of glass was made in the living room. Several types of wall and ceiling constructions were made available for study. A minimum of insulation was used in the walls and ceilings, since the more poorly insulated houses are the more difficult to heat and cool satisfactorily. To increase the flexibility of investigation, plywood panels were substituted for the more conventional drywall or plaster construction in all second and third level rooms except kitchen and bathrooms so that they could be opened to make changes in the type and amount of insulation used. In all other respects, the construction of the house was representative of modern construction practice and quality of workmanship.

Information on general house construction and size follows:

<table>
<thead>
<tr>
<th>Description</th>
<th>Size</th>
</tr>
</thead>
<tbody>
<tr>
<td>Floor area*, first level</td>
<td>477 sq. ft.</td>
</tr>
<tr>
<td>Floor area*, second level</td>
<td>592 sq. ft.</td>
</tr>
<tr>
<td>Floor area*, third level</td>
<td>569 sq. ft.</td>
</tr>
<tr>
<td>Floor area*, garage</td>
<td>560 sq. ft.</td>
</tr>
<tr>
<td>Floor area*, equipment room</td>
<td>90 sq. ft.</td>
</tr>
<tr>
<td>Floor area*, total</td>
<td>2,288 sq. ft.</td>
</tr>
<tr>
<td>Number of rooms</td>
<td>10</td>
</tr>
<tr>
<td>Number of baths</td>
<td>2 1/2</td>
</tr>
</tbody>
</table>

*Based on inside room dimensions

Figure 1. I=B=R Hydronic Research House
II. DESCRIPTION OF EQUIPMENT

Second level

Figure 2. Valance System Layout
Types of Construction


Second and third level walls: 1-in. vertical wood siding stained dark brown on 3%-in. plywood sheathing. 2-in. by 4-in. studs with 1-in. insulation between. Interior finish, 1/-in. plywood paneling.

Roof: White asbestos shingles on 3%-in. plywood sheathing. 2-in. by 6-in. rafters with 2-in. insulation between. The 3½-in. air space between insulation and sheathing vented to outdoors through continuous slots 1-in. wide extending the full length of each soffit. The construction of the soffit vent is shown in Figure 3.

Bedroom ceilings: ½-in. acoustical tile on 2-in. by 2-in. furring strips and ½-in. masonite attached to under side of roof rafters. (Can be modified to provide attic space if desired.)

Entrance and living room ceilings: ½-in. acoustical tile on 2-in. by 2-in. furring strips attached to the bottom of the roof rafters.

Dining room ceiling: ½-in. acoustical tile on 2-in. by 2-in. furring strips with attic space above.

Kitchen and breakfast room ceilings: Lath and plaster with attic space above.

First level floor: Vinyl asbestos tile, 4 inches of concrete on 4 inches of gravel fill. A 4-mil polyethylene film between the concrete and fill to serve as a vapor barrier. One-in. styrofoam insulation located on the inside edge of the foundation walls extended from the floor surface to a depth of 24 inches below the floor surface.

Second level floor: Asphalt tile, ⅜-in. plywood over 1-in. by 6-in. wood subfloor on 2-in. by 10-in. joists. Crawl space below.

Crawl space: Five vents, with a total area of 4.13 square feet, were provided in the wall of the crawl space. These were equipped with dampers so that the crawl space could be either vented or unvented. A 4-mil polyethylene film over the surface of the ground to reduce the transfer of water vapor from the ground to the air in the crawl space.

A room-by-room summary of wall, glass, floor, and ceiling areas is presented in Table 1.

B. DESIGN LOADS

The design heating and cooling loads for the house were determined in accordance with the pro-
II. DESCRIPTION OF EQUIPMENT

Table 2A

Design Heating Loads

<table>
<thead>
<tr>
<th>Room</th>
<th>Glass</th>
<th>Walls</th>
<th>Ceilings</th>
<th>Floors</th>
<th>Infiltration</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>First Level</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Recreation</td>
<td>3.31</td>
<td>4.35</td>
<td></td>
<td>0.92</td>
<td>3.31</td>
<td>11.89</td>
</tr>
<tr>
<td>Den</td>
<td>2.52</td>
<td>1.94</td>
<td></td>
<td>0.34</td>
<td>1.29</td>
<td>6.06</td>
</tr>
<tr>
<td>Bath No. 1</td>
<td>0.85</td>
<td>0.94</td>
<td></td>
<td>0.13</td>
<td>0.27</td>
<td>2.29</td>
</tr>
<tr>
<td>Hall A</td>
<td></td>
<td></td>
<td></td>
<td>0.11</td>
<td></td>
<td>1.11</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>20.35</td>
</tr>
<tr>
<td>Second Level</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Living Room</td>
<td>2.86</td>
<td>0.37</td>
<td>1.60</td>
<td>2.17</td>
<td>2.29</td>
<td>13.49</td>
</tr>
<tr>
<td>Dining Room</td>
<td>1.71</td>
<td>1.37</td>
<td>0.69</td>
<td>0.92</td>
<td>0.80</td>
<td>5.49</td>
</tr>
<tr>
<td>Kitchen</td>
<td>2.51</td>
<td>1.26</td>
<td>0.80</td>
<td>0.92</td>
<td>1.14</td>
<td>6.63</td>
</tr>
<tr>
<td>Breakfast Room</td>
<td>0.62</td>
<td>0.57</td>
<td>0.37</td>
<td>0.57</td>
<td>1.26</td>
<td>3.21</td>
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<tr>
<td>Entry</td>
<td>2.74</td>
<td>0.23</td>
<td>0.46</td>
<td></td>
<td></td>
<td>3.42</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>33.49</td>
</tr>
<tr>
<td>Third Level</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bedroom No. 1</td>
<td>1.37</td>
<td>1.14</td>
<td>0.92</td>
<td>0.11</td>
<td>1.26</td>
<td>4.80</td>
</tr>
<tr>
<td>Bedroom No. 2</td>
<td>2.06</td>
<td>1.71</td>
<td>0.92</td>
<td>0.11</td>
<td>1.83</td>
<td>6.63</td>
</tr>
<tr>
<td>Bedroom No. 3</td>
<td>1.69</td>
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<td>1.03</td>
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<td>6.86</td>
</tr>
<tr>
<td>Entry</td>
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<td>0.11</td>
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<td>2.74</td>
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<tr>
<td>Bath No. 2</td>
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<td>0.41</td>
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<td>0.69</td>
</tr>
<tr>
<td>Lavatory A</td>
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<tr>
<td>Hall B</td>
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<td>0.23</td>
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</tr>
<tr>
<td>Total</td>
<td></td>
<td></td>
<td></td>
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</tr>
<tr>
<td>House total</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>76.01</td>
</tr>
</tbody>
</table>

C. BASIS OF DESIGN OF VALANCE SYSTEM

Whenever the same unit is used to both heat and cool a room, one is faced with the problem of deciding whether to select the unit on the basis of the calculated design heating load, calculated design cooling load or on some combination of these. Since heating demands are far more severe than cooling in Urbana, Illinois, the following procedure was used in selecting the valance units for the I=B=R Hydronic Research House.

Step 1. Both the design heating and cooling loads were calculated for each room using I=B=R Guides H-20 and C-30 respectively.

Step 2. The length of wall available for the location of valance units was determined for each room.

Step 3. The room having the highest calculated heat loss per foot of available wall (Step 2) was selected.

Step 4. Using the room selected in Step 3 and a chilled water temperature of 40° F., the length of valance assembly required to provide cooling equal to the calculated design cooling load of the room was determined.

Step 5. Using the length of valance assembly obtained in Step 4, an average water temperature was determined which would result in a heating output of the selected radiation equal to the calculated design heating load of the room. This was then used as the design water temperature for the entire system.

Step 6. Using the design water temperature determined in Step 5, the length of valance assembly required in each room of the house to satisfy the design heating load was determined.

The amounts of valance selected for each room of the house are shown in Table 3. The outputs shown in columns 4 and 7 are based on the results.

Table 2B

Design Cooling Loads

<table>
<thead>
<tr>
<th>Room</th>
<th>Glass Heat Gain</th>
<th>Walls Heat Gain</th>
<th>Ceilings</th>
<th>Floors Heat Gain</th>
<th>Infiltration Heat Gain</th>
<th>Occupancy Heat Gain</th>
<th>Total Heat Gain</th>
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<tr>
<td>First Level</td>
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<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Recreation</td>
<td>3.31</td>
<td>4.35</td>
<td></td>
<td>0.92</td>
<td>3.31</td>
<td>11.89</td>
<td>20.35</td>
</tr>
<tr>
<td>Den</td>
<td>2.52</td>
<td>1.94</td>
<td></td>
<td>0.34</td>
<td>1.29</td>
<td>6.06</td>
<td>7.06</td>
</tr>
<tr>
<td>Bath No. 1</td>
<td>0.85</td>
<td>0.94</td>
<td></td>
<td>0.13</td>
<td>0.27</td>
<td>2.29</td>
<td>2.52</td>
</tr>
<tr>
<td>Hall A</td>
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<td></td>
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<td>0.11</td>
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<td>1.11</td>
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<tr>
<td>Total</td>
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<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>Living Room</td>
<td>2.86</td>
<td>0.37</td>
<td>1.60</td>
<td>2.17</td>
<td>2.29</td>
<td>13.49</td>
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<td>Dining Room</td>
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<td>0.80</td>
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<td>Kitchen</td>
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<td>Breakfast Room</td>
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<td>33.49</td>
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<tr>
<td>Third Level</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bedroom No. 1</td>
<td>1.37</td>
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<td>0.92</td>
<td>0.11</td>
<td>1.26</td>
<td>4.80</td>
<td></td>
</tr>
<tr>
<td>Bedroom No. 2</td>
<td>2.06</td>
<td>1.71</td>
<td>0.92</td>
<td>0.11</td>
<td>1.83</td>
<td>6.63</td>
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</tr>
<tr>
<td>Bedroom No. 3</td>
<td>1.69</td>
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<td>1.03</td>
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<td>2.17</td>
<td>6.86</td>
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<tr>
<td>Entry</td>
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<td>0.45</td>
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<td>0.11</td>
<td>0.34</td>
<td>2.74</td>
<td>3.38</td>
</tr>
<tr>
<td>Bath No. 2</td>
<td></td>
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<td>0.41</td>
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</table>
of tests made in the I=B=R Laboratory as described in Chapter IV, Section A of this bulletin. Since the units were selected on the basis of the calculated heating loads, the ratio of installed "rated" heating capacity to design heating load is approximately 1.0 in all cases, with the exceptions of the bath room on the first level and lavatory B on the third level. In the first of these two cases a 4-foot unit was the longest that could be installed and in the second, valance was installed to meet the heat losses not only of lavatory B but also of lavatory A, bath 2, and the hall.

The design cooling loads of each room are shown in column 6, and the "rated" cooling capacity of the installed valance units are shown in column 7. The ratios of installed cooling capacity to design cooling load are shown in column 8. While the ratios of installed heating capacity to design heating load were approximately 1.0 in most cases, the ratios of installed cooling capacity to design cooling load ranged from 1.15 to 0.57. Thus, it is apparent that there is no fixed ratio between design heating and cooling loads for the rooms. A system designed to be in balance in one season will not necessarily be in balance in the other season.

D. THE VALANCE SYSTEM

The valance system was used for both summer and winter seasons and consisted of the valance units themselves plus a chiller, boiler, pumps, piping, valves, and controls. A schematic diagram of the complete system as installed in the house is shown in Figure 2. A cross sectional drawing of the valance unit, consisting of a finned tube section, hanger, cover, and trough for collecting condensate removed from the air during summer operation is shown in Figure 4. Drain connections were provided at one end of each assembly to allow the water in the trough to be removed. Figure 5 illustrates the appearance of the finished installation in the dining room.

Air circulation through the valance unit was by gravity both summer and winter. In the summer, chilled water was circulated through the tube...
II. DESCRIPTION OF EQUIPMENT

Figure 5. Valance Units Installed in Dining Room

whereas heated water was used in the winter season. The same piping system was used for both seasons, except that provision was made to prevent circulation of chilled water through the boiler in summer and heated water through the chiller in the winter. All sections of the piping system not located directly over the condensate collection trough of the valance units were insulated with a foamed plastic insulation approximately \( \frac{3}{4} \) inch thick. All joints in this insulation were sealed with a plastic cement to make the insulation vapor-tight throughout.

The piping arrangement used was a three-zone, series-connected system. The piping system was sized on the basis of circulating 1 g.p.m. for each 10,000 B.t.u.h. design heat loss of the area served. Zoning was by house levels. All thermostats were located 30 inches above the floor. Their positions are shown in Figure 2.

A 5-horsepower water chiller was used for summer operation. The compressor and evaporator sections of the chiller were located in the boiler room while the air cooled condenser was located outdoors at the rear of the house. Sixty-cycle, single-phase electrical energy was supplied to the compressor at 230 volts and to the condenser at 115 volts.

During the winter a sectional cast iron boiler designed for gas firing was used. This boiler was completely enclosed by an insulated sheet metal jacket. The boiler had a net I=B=R water rating of 90,000 B.t.u.h. Natural gas having a heating value of 976 B.t.u. per cubic foot was used as the fuel. The average gas burning rate was approximately 165 c.f.h., and the burner was adjusted to give a \( \text{CO}_2 \) content in the flue gas of approximately 8 percent.

E. SYSTEM CONTROL

1. Winter

In winter the controls consisted of three room thermostats, three relays, and three pumps which responded to the demand of the room thermostats. Gravity circulation of water was prevented by flow control valves located in the supply main of each zone just above the boiler. The boiler was equipped with a high limit control and pressure relief valve.

The operating sequence of the system was as follows: As soon as any one of the zone thermostats indicated the need of heating in that area, the circulating pump supplying water to that area was put into operation. At the same time the gas burner in the boiler was turned on. As heating was required in additional zones the thermostats put the appropriate pump into operation. As long as any circulating pump was in operation, the gas burner continued to operate until the temperature of the water in the boiler was raised to the setting of the high limit control (225° F.). When the setting of the high limit control was reached, the burner was turned off, but as long as heating was required in any area the pump remained in operation. In the event that the burner was turned off by action of the high limit control it would be restarted automatically as soon as the temperature of the water in the boiler dropped approximately 40° F. below the setting of the high limit control.

As sufficient heat was supplied to each zone the thermostat in that zone stopped the pump serving that zone. At the time the last zone received sufficient heat, the burner was also turned off. During these tests no provision was made to supply hot faucet water from the boiler by means of an indirect heater.

2. Summer

The controls used for summer operation of the valance system consisted of three room thermostats (one in each zone), three relays, and three motorized zone valves which controlled the flow of chilled water to each zone by responding to the demand of
the zone thermostats. A water temperature control located in the chiller outlet was used in addition to the usual high and low side controls supplied as a part of the water chiller package. The chilled water was circulated through the system by a single high head pump.

The operating sequence of the system was as follows: As soon as any one of the zone thermostats indicated need of cooling in that area, the circulating pump started in operation and the motorized valve for that zone moved to the open position. As additional zones required cooling, the zone valves of those circuits also moved to the open position. The pump continued to run as long as any thermostat demanded cooling. As long as the circulating pump was in operation, the chiller operated sufficiently to maintain the temperature of the water leaving the chiller between 40° F. and 45° F.

As soon as sufficient cooling effect had been supplied to any zone the thermostat closed the motorized zone valve controlling the flow of water in that part of the system. As soon as sufficient cooling had been supplied to all zones, the pump and chiller were also turned off and the entire system remained idle until cooling was again required in at least one zone.

The high and low side controls in the chiller acted as safety controls to prevent operation of the chiller if for any reason the refrigerant pressure on the high side became excessively high or the temperature in the evaporator section became too low.

F. INSTRUMENTATION

Approximately 250 copper-constantan thermocouples, made of 28 gauge wire, were installed in and around the house to provide for the measurement of temperatures. These temperatures can best be grouped in the following categories:

1. House
   a. Air temperatures at 3, 30, and 60 inches above the floor and 3 inches below the ceiling in each room of the house. Air temperature at 90 inches above the floor in rooms with ceiling height exceeding 9 feet.
   b. Surface temperatures of floors, walls, ceilings, and intermediate sections of building members.
   c. Air temperatures in the attic and crawl space.

2. Outdoor Air

3. Ground
   a. Temperature of the ground to depths of approximately 7 feet below grade level both under and to the side of the house.

4. Cooling System
   a. Temperatures of the water entering and leaving the chiller and each room unit used in the cooling system.
   b. Temperatures of the air entering and leaving the air-cooled condenser.
   c. Refrigerant temperatures entering and leaving the condenser.

5. Heating System
   a. Temperatures of the water entering and leaving the boiler and each room unit used in the heating system.
   b. Temperatures of the gasses leaving the boiler and at two locations in the chimney.

All thermocouples were connected to selector switches on a central switchboard. The e.m.f. produced by each thermocouple could be read to 0.001 mv on a precision potentiometer used with a highly sensitive galvanometer. A 10-point recording potentiometer, used with an auxiliary switchboard, made it possible to obtain either instantaneous or continuous printed records of the readings of any selected group of thermocouples.

Elbow meters (s) connected to differential pressure recorders or manometers were used to measure the rate of water flow in each zone of the cooling and heating systems. All flow meters were calibrated in place and were capable of measuring the existing flow rates with an error not to exceed 5 percent.

The operating times of all electrical equipment used in the systems under test were obtained by the use of self-starting electric clocks wired into the electrical circuits.

Watt-hour meters readable to 10 watt-hours were used to measure the power consumption of all electrical equipment included in the systems under test.

An Orsat apparatus graduated to read CO₂ content to 0.2 percent was used to measure the completeness of combustion.

Humidity indicators and recorders using sensing elements made of hair were used to determine the moisture content of the room air. The wet- and dry-bulb temperatures of the outdoor air were ob-
tained with a recording instrument in which outdoor air was continuously drawn over liquid filled temperature sensing elements by means of a fan. All humidity indicators and recorders were calibrated periodically with an aspirated psychrometer, shielded from radiation effects.

Other instruments included: heat meters, for measuring heat flow through building constructions, a specially designed Thomas Meter, for measuring the rate of gas flow up the chimney, and a micro-manometer, for measuring indoor-outdoor pressure differences across the walls of the house.
III. TEST CONDITIONS AND PROCEDURES

A. CONDITIONS COMMON TO ALL TESTS

Four series of tests were made; two in the summer and the remaining two during the winter. Each series of tests was continued until data were obtained over a wide range of outdoor temperatures and general weather conditions. The following conditions were common to all four series.

All windows were closed at all times. Outside doors were closed, except for periods when persons were in the act of entering or leaving the house. Room doors were in the open position at all times. All draperies, except those in bedrooms 2 and 3, were pulled to the side of the glass area as far as they would go and remained in this position at all times. The draperies in bedrooms 2 and 3 were drawn across the glass at night. The door to the equipment room was left in the closed position. The access from the equipment room to the crawl space was closed.

B. SUMMER TEST CONDITIONS

The two series made during summer weather were designated as Series B-60 and C-60. Test conditions for these two series over and above those conditions listed in the preceding paragraph were as follows:

Series B-60. The three zone thermostats were all set to maintain an average air temperature of 75°F. at a height of 30 inches above the floor in the room in which the thermostat was located. Crawl space vents were open.

Series C-60. All test conditions for this series were the same as for Series B-60, except that plastic curtains were installed at the top and bottom of the staircase. These curtains were sealed to the walls, floor, and ceiling so that no air could circulate from one house level to another through the staircase. Figure 6 shows one of these curtains installed.

C. WINTER TEST CONDITIONS

The two remaining test series represented winter conditions of operation. These series were designated as Series E-60 and F-60. In Series E-60, the following test conditions prevailed in addition to those which were common to all four test series.

Series E-60. Each of the three zone thermostats was set to maintain an average air temperature of 73°F. 30 inches above the floor in the room in which the thermostat was located. The high limit control in the boiler was set at 225°F. Crawl space vents were closed during all winter tests.

Series F-60. All test conditions for this series were the same as for Series E-60 except that the plastic curtains were installed in the same manner as in Series C-60.

In the remainder of this report it will be common practice to refer to Series C-60 and F-60 as operating with levels isolated, and to Series B-60 and E-60 as operating when levels were not isolated.

Figure 6. Plastic Curtain Installed in Staircase Between First and Second Levels
D. TEST PROCEDURES AND OBSERVATIONS

Basically the test procedure was the same for all four test series. Each test was 24 hours in length, and the test day started and ended at 8:00 a.m. Test observations common to both summer and winter tests included:

1. All temperatures included in groups 1, 2, and 3 as listed in Chapter II, Section F.
2. Water flow rates through the different zones.
3. Operating time and power consumption of each pump.
5. Sky conditions, wind speed and direction.
6. Relative humidity of indoor air.

Additional observations made during the summer tests included:

1. All temperatures included in group 4 as listed in Chapter II, Section F.
2. Operating time and power consumption of zone valves, compressor and condenser fan motors.
3. Occupancy of the house.

Additional observations made during winter tests included:

1. All temperatures included in group 5 as listed in Chapter II, Section F.
2. Operating time of the gas burner.
3. Carbon dioxide content of the flue gas.
4. Fuel consumption.

Key water and refrigerant temperatures were constantly recorded by means of recording potentiometers. Recording instruments also provided continuous records of the heat meter readings; outdoor air temperatures; room air temperatures in the living room, bedroom Number 1, and the recreation room; air temperature in the attic above the dining room, and in the crawl space; water flow rates through the different zones, and static pressure in the system.

Most other observations were made manually four times per day; at 8:00 a.m., 1:00 p.m., 4:00 p.m., and 10:00 p.m. A few observations such as comfort votes, occupancy, CO₂ content, and rate of flow of the flue gas were taken at less frequent intervals.
IV. VALANCE OUTPUTS

A. ESTABLISHMENT OF RATINGS IN THE I=B=R LABORATORY

Prior to selecting any valance for installation in the research house, an 8-foot section was delivered to the I=B=R Laboratory* to establish heating and cooling outputs to be used in selecting units for test in the research house. The unit was installed on the back wall of the warm wall test booth with the hanger nails located 2 inches below the ceiling. The method of installation in the test booth was the same as that proposed for use in the research house.

During the heating output tests, the inlet air temperature, measured 11 inches below the ceiling (at the bottom of the valance unit) and 3 inches in front of the unit, was maintained at 80° F. Heated water was circulated through the unit at a rate of 500 pounds per hour. Tests were made with water temperatures of 170° F., 186° F., 203° F., and 220° F. The results of these tests are represented by the curve in Figure 7.

During the cooling output tests the inlet air temperature (measured 1 1/2 inch below the ceiling and 12 inches in front of the valance unit) was maintained at 80° F. with a relative humidity of 50 percent. Chilled water at an average temperature of 40° F. was circulated through the valance unit at the rate of 500 pounds per hour (1 g.p.m.). The total cooling output (sensible and latent) thus determined was 150 B.t.u.h. per linear foot of finned section.

In this report, heating and cooling outputs as determined by the I=B=R Laboratory will be referred to as “rated” outputs to distinguish them from outputs determined by tests in the research house. These “rated” outputs were used in selecting the valance lengths for each room as described in Chapter II, Section C of this bulletin.

*B. MEASURED OUTPUTS IN THE I=B=R HYDRONIC RESEARCH HOUSE

1. Winter

In order to have the desired design output of 425 B.t.u.h./ft., Figure 7 indicates that the valance would have to be operated at a water–air temperature difference of 103° F. Assuming an inlet air temperature of 80° F. would require a water temperature of 183° F.

During the course of the heating season there was reason to believe that the output of the valance was lower than anticipated. This was particularly true for the lower level where the capacity of the units was insufficient to carry the load above an average water–air temp difference, deg F

Figure 7. Valance Water Heat Output

The I=B=R Laboratory was built and is operated by The Institute of Boiler and Radiator Manufacturers. It is fully equipped with warm wall test booths and necessary instruments for tests to determine both the heating and cooling outputs of radiation. The laboratory has adequate heating and air conditioning equipment to maintain desired air temperatures and humidities in the test room regardless of outdoor weather conditions.
IV. VALANCE OUTPUTS

indoor-outdoor temperature difference of 50°F, even though the water temperature was raised to 215°F. For this reason output tests were conducted in order to determine the valance output and to investigate the factors which may affect it.

Figure 7 also includes the points for output tests conducted on the valance in the research house. The group of points in the area of 300 to 350 B.t.u./ft. were obtained with essentially the same input to the boiler was reduced in order to obtain indoor-outdoor temperature difference of approximately 5°F. The points in the area of 200 B.t.u./ft. were obtained with essentially the same house and outdoor conditions; however, the gas input to the boiler was reduced in order to obtain lower average water temperatures. For these tests the total output of the valance was obtained for each level by measuring the water flow rate and the temperature drop through the units. The output per foot was then obtained by dividing the total output by the installed length. The air temperature used in arriving at the water–air temperature difference was the average of the temperatures measured in the center of the rooms at a height above the floor approximately level with the top of the valance units.

The agreement between tests in the laboratory and in the research house was very good, indicating that the air temperatures as measured 3 inches below the ceiling (or at a level corresponding to the location of the top of the valance units) were representative of the air temperature measured in the laboratory at the inlet of the test unit. However, the air temperatures measured in the research house were well above the 80°F. used in the laboratory tests and hence, the valance outputs in the research house were lower than expected.

In Figure 8 the average air temperatures for the first and second levels of the research house, as measured at the center of the rooms at a height approximately level with the top of the valance units, are plotted against indoor-outdoor temperature difference. At design conditions (indoor-outdoor temperature difference of 80°F.) the average air temperatures for the first and second levels were 127°F. and 138°F. respectively. Thus, to determine the output of valance for design purposes a temperature of approximately 130°F. instead of

![Figure 8. Room Air Temperature, 3 Inches Below Ceiling](image)

Table 4

<table>
<thead>
<tr>
<th>Room</th>
<th>Design Cooling Load</th>
<th>&quot;Rated&quot; Cooling Output of Valance</th>
<th>Average** Air Temperature 30 in. Above Floor</th>
<th>Average** Water Temperature</th>
<th>Change** in Water Temperature Through Unit</th>
<th>Measured** Cooling Output M.b.h.</th>
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<td>M.b.h.</td>
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<td>41.3</td>
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<td>43.6</td>
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*This does not include bath number 2, lavatory A, and hall B.
**Average for tests made on July 25, August 2, August 18, and September 1, 1960.
Average indoor relative humidity about 48%.
Water circulation rate through first level units = 1,360 lb./hr.
Water circulation rate through second level units = 1,640 lb./hr.
Water circulation rate through third level units = 1,400 lb./hr.
Water circulation rate through the chiller = 4,400 lb./hr.
80°F. should have been used for the inlet air temperature. A water temperature of 233°F. would be required to yield a valance output of 425 B.t.u.h. per linear foot.

2. Summer

On several occasions during the summer the cooling output of the valance installed in the research house was checked by observing the water flow rate and the water temperatures in and out of each assembly. The results of these tests together with “rated” cooling outputs are shown in Table 4. The values in Table 4 are averages for four tests in the research house made at times when all units were in periods of continuous operation so that steady state conditions could be approximated. Even so, there was not the precise control of room air and water temperatures that were possible in the laboratory. Also, the temperature drop through some of the units was less than 1°F. Because of these test limitations, a room by room comparison of test outputs and “rated” outputs shows wide variations. However, taking each level as a unit shows more consistent results with the test outputs ranging from a low of 0.98 times the “rated” output for the second level to a high of 1.09 for the first and third levels. The ratio of test output to “rated” output for the entire house was 1.04.

In the research house the system was operated to maintain the air temperature 30 inches above the floor at approximately 75°F. On the second and third levels, the temperature of the air 90 inches above the floor was about 2°F warmer than that of the air 30 inches above the floor. On the first level, it was about 3°F warmer. Therefore, the inlet air temperature in the research house was about 77°F as opposed to 80°F in the laboratory. Also, the average water temperatures were about 2°F to 3°F warmer in the research house than in the laboratory tests, and there were differences in the relative humidity of the inlet air. All of these have their effect on the output of valance units. Nevertheless, the agreement between the laboratory and the tests in the house was close enough to demonstrate that the general test procedure used to determine cooling outputs yielded results which approximated results obtained by actual use.
V. LOADS AND EQUIPMENT PERFORMANCE IN
THE I=B=R HYDRONIC RESEARCH HOUSE

A. MEASURED HEATING AND COOLING LOADS

Loads on the valance system were measured in the I=B=R Hydronic Research House for each of the four series of tests described in Chapter III of this report.

During the 1959-60 heating season, the heating system used was a series loop, cast iron, baseboard system, as contrasted to the series loop valance system used during the 1960-61 heating season. Both systems employed the same controls and method of operation as outlined in Chapter II, Section E 1, with the exception of the method used in controlling the boiler water temperature. As given in Chapter II, Section E 1, the operation of the burner when using the valance system was governed by the room thermostats with the setting of the high limit control determining the maximum boiler water temperature that could be attained. When using the baseboard system, the boiler water temperature was governed by an outdoor air temperature control.

Since the methods of operation and the components associated with the baseboard and valance systems were essentially the same, a comparison of the fuel inputs to the burner for the two systems is equivalent to a comparison of the total heating load of the house resulting from the use of each of the systems.

1. Total Heating Load

Figure 9 shows the effect of indoor-outdoor temperature difference on fuel input to the boiler for Series E-60. The curve drawn through these points was determined statistically and represents the best straight line fit of the data.

In Figure 9 it can be seen that there is a fair amount of spread in the data and the confidence intervals are quite large. It is not unreasonable to expect some dispersion in a correlation such as this, for there are variables other than indoor-outdoor temperature difference which influence fuel consumption. For the I=B=R Hydronic Research House, wind speed was found to have a great influence on fuel consumption. Using helium as a tracer gas, infiltration tests were conducted during the 1960-61 heating season. These tests indicated an air infiltration rate of approximately 3.0 air changes per hour at design conditions. Similar tests conducted in the former I=B=R Research Home, a structurally tight house, indicated air change rates in the order of 0.75 air changes per hour at design conditions. Thus, it is obvious that the I=B=R Hydronic Research House was not a structurally tight house and high infiltration rates caused by wind affected the daily fuel consumption.

The lighter line curves containing the fuel consumption data in Figure 9 give the variation in fuel consumption in the I=B=R Hydronic Research House for different wind speeds.
consumption to be expected for wind variations from 0 to 15 m.p.h. This band of variations in fuel consumption was derived by using the data obtained from the infiltration tests conducted. It can be seen that with the exception of nine points, the test data are confined within this limiting band. From the data plotted and the limiting curves, it can be concluded that based on the measured load at 0 m.p.h. wind the measured load increased approximately 2 percent per increase in wind speed of one m.p.h. The reasons for some points falling outside this band could be the effect of sun, wind speeds higher than 15 m.p.h., and wind direction.

The fuel consumption data plotted against indoor-outdoor temperature difference for the 1959-60 heating season showed a dispersion similar to that shown in Figure 9. For the 1959-60 season no measurements of infiltration rates were made. However, it was found that better correlation between fuel consumption and indoor-outdoor temperature difference existed if the data were grouped into several wind speed categories. When an analysis of this type was made it was found that based on the measured load at 0 m.p.h. wind the measured load increased approximately 3.7 percent per m.p.h. increase in wind speed.

Comparing the fuel consumption for the two heating seasons at a 7 m.p.h. wind (this is the average winter wind speed at Urbana, Illinois) revealed that the fuel consumption for the 1960 series was 16.5 percent higher than the 1959 series.

Since there was a difference in the relative placement of the baseboard and valance within the area to be heated, that is, the baseboard was placed at the lower extremities and the valance at the higher extremities of outside walls, it was initially believed that there would be a difference in the heat loss characteristics of the house between the two seasons. In particular, it was believed that the higher ceiling temperatures produced by the valance would lead to higher ceiling heat losses. The placement of heat meters on the interior of the second and third level ceilings substantiated this belief.

Furthermore, since the performance of a valance system is similar to a ceiling panel system, it is interesting to note that the increased fuel consumption is in fairly good agreement with the results obtained when investigating the performance of a hot-water ceiling panel system in 1953 and 1954. This study indicated that ceiling panel systems increased the heat loss of a one-story house by approximately 27 percent.

In addition to producing higher transmission losses through the ceiling, the valance also produced higher transmission losses through walls, windows, and floors. These losses are discussed in detail in Chapter V, Section A 5, and the results are tabulated in Table 5.

It can be seen in Table 5 that the changes in level loads were not the same for the baseboard and valance systems. Furthermore, in many cases the difference was caused by differences in heat flow from the house. Considering only the differences in heat flow from the house to be the cause for change in total load, Table 6 was formed. This table includes all the level losses in percent of the level load along with the building member through which the loss occurred. From this table it can be seen that 7.5 percent of the change in total load can be

### Table 5

<table>
<thead>
<tr>
<th>Total Change in Load</th>
<th>Change Due to Air Movement</th>
<th>Change Due to Ceiling Losses</th>
<th>Change Due to Heat Transfer from First to Third Level</th>
<th>Change Due to Increased Wall Losses</th>
<th>Change Due to Increased Glass Losses</th>
<th>Total Change Accounted for</th>
<th>Total Change Unaccounted for</th>
</tr>
</thead>
<tbody>
<tr>
<td>Baseboard Valance</td>
<td>Percent Change in First Level Load</td>
<td>Baseboard Valance</td>
<td>Percent Change in Second Level Load</td>
<td>Baseboard Valance</td>
<td>Percent Change in Third Level Load</td>
<td>Baseboard Valance</td>
<td>Valance</td>
</tr>
<tr>
<td>First Level Load</td>
<td>18.0 72.0</td>
<td>18.0 33.0</td>
<td>0.0 0.0</td>
<td>0.0 3.7</td>
<td>0.0 11.8</td>
<td>18.0 57.6</td>
<td>0.0 14.4</td>
</tr>
<tr>
<td>Second Level Load</td>
<td>7.0 21.0</td>
<td>7.0 0.0</td>
<td>0.0 9.6</td>
<td>0.0 5.0</td>
<td>0.0 0.0</td>
<td>0.0 -1.0</td>
<td>0.0 -53.8</td>
</tr>
<tr>
<td>Third Level Load</td>
<td>7.0 -25.0</td>
<td>25.0 -84.0</td>
<td>25.0 -97.0</td>
<td>0.0 -15.0</td>
<td>0.0 0.0</td>
<td>25.0 -40.2</td>
<td>0.0 -53.8</td>
</tr>
<tr>
<td>House Totals</td>
<td>76,010 5730</td>
<td>33,490 930</td>
<td>33,490 930</td>
<td>33,490 930</td>
<td>33,490 930</td>
<td>33,490 930</td>
<td>33,490 930</td>
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</tbody>
</table>

### Table 6

<table>
<thead>
<tr>
<th>Increased Winter Heat Loss Produced by Valance System</th>
</tr>
</thead>
<tbody>
<tr>
<td>Level</td>
</tr>
<tr>
<td>--------</td>
</tr>
<tr>
<td>First</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Second</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Third</td>
</tr>
<tr>
<td>House</td>
</tr>
</tbody>
</table>

Total Accounted for Increase in Load = 5730 / 76,010 = 7.5%
Figure 10. Measured Cooling Loads, September 1, 1960
accounted for. Thus, 9 percent remains unaccounted. This 9 percent represents a load of 6,840 B.t.u.h. at design conditions which (when considering the assumptions that were made in order to make the computation) represents good agreement.

2. Cooling Load on a Design Day

In order to determine the variation in load for each level of the house through the course of a day, special tests were made. In these tests hourly readings were taken of the operating times of the chiller and each zone, the rates of water circulation through the chiller and each zone, water temperatures in and out of the chiller and each zone, the air temperatures in each room, the outdoor air temperature, and the power consumption of all electric motors. The hourly cooling loads were calculated from the measured water flow rates and the water temperature changes.

The data obtained on total cooling outputs of the equipment during one of these tests are presented in Figure 10. The test was conducted with the levels isolated (Series C-60). The maximum outdoor temperature was about 95° F. at 4:00 p.m. (daylight saving time). The actual test results are represented by the solid lines. The third level zone operated continuously from 10:00 a.m. until about 9:00 p.m., indicating that at least for a part of this time the actual heat gain on the third level must have exceeded the output of the valance units installed. The broken line curves in Figure 10 are an attempt to estimate the probable loads had the valance units been adequate in capacity. These curves have been drawn so that the area under the broken line curve is equal to the area under the corresponding solid curve.

The first and third levels had windows on the NE, SE, and NW sides and hence received direct radiation all morning and late afternoon. Because of the morning sunshine, the maximum load for these levels occurred at about 1:00 to 2:00 p.m. The bulk of the glass on the second level was on the NW side of the house and therefore the maximum load on this level did not occur until about 6:00 p.m.

Since, when using the load calculation procedure in I=B=R Guide C-30, one calculates the external sensible heat gains and then makes allowances for internal and latent loads, it is desirable to have a breakdown of the measured maximum total loads in order to compare the different components with calculated values. The external sensible cooling load was determined by subtracting from the total measured load the latent load and the sensible heat gains within the house due to occupancy and to the use of power. The sensible load due to occupancy was assumed to be 250 B.t.u. per man-hour.

Table 7 contains a breakdown of the measured maximum total loads and time of occurrence for the September 1, 1960 test. For comparison, the calculated loads as obtained by the use of I=B=R Guide C-30 are also shown. The measured maximum loads for the first and second levels were about 75 percent of the calculated loads while for the third level the measured load was about 10 percent in excess of the calculated load. Because the maximum loads on the different levels did not

<table>
<thead>
<tr>
<th>Table 7 Cooling Loads</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Calculated Design Cooling Loads, Guide C-30</strong></td>
</tr>
<tr>
<td><strong>Maximum Outdoor Temperature = 95°F.</strong></td>
</tr>
<tr>
<td><strong>Total Load</strong></td>
</tr>
<tr>
<td><strong>M.b.h.</strong></td>
</tr>
<tr>
<td>First Level</td>
</tr>
<tr>
<td>Second Level</td>
</tr>
<tr>
<td>Third Level</td>
</tr>
<tr>
<td>House Totals</td>
</tr>
</tbody>
</table>

**Calculated Design Cooling Loads with Values Adjusted to Provide for Shading that Actually Existed**

| Maximum Outdoor Temperature = 95°F. |
| **Maximum Outdoor Temperature = 95°F.** |
| **September 1, 1960** |
| **First Level** | **Second Level** | **Third Level** | **House Totals** |
| **M.b.h.** | **M.b.h.** | **M.b.h.** | **M.b.h.** |
| First Level | 6.84 | 4.53 | 0.60 | 2.18 | 0.75 | 1:30 p.m. |
| Second Level | 12.34 | 8.43 | 0.60 | 3.32 | 0.75 | 1:30 p.m. |
| Third Level | 8.73 | 6.55 | 0.60 | 2.18 | 0.75 | 1:30 p.m. |
| House Totals | 28.91 | 20.49 | 1.20 | 7.22 | 0.75 | 1:30 p.m. |

**Measured Loads at Time of Maximum Total Load**

| Maximum Outdoor Temperature = 95°F. |
| **Maximum Outdoor Temperature = 95°F.** |
| **September 1, 1960** |
| **First Level** | **Second Level** | **Third Level** | **House** |
| **M.b.h.** | **M.b.h.** | **M.b.h.** | **M.b.h.** |
| First Level | 5.95 | 0.63 | 3.31 | 1.48 | 0.75 | 1:30 p.m. |
| Second Level | 11.70 | 0.92 | 0.60 | 2.28 | 0.81 | 2:00 p.m. |
| Third Level | 9.65 | 0.30 | 0.30 | 2.35 | 0.76 | 2:00 p.m. |
| House | 25.30 | 15.33 | 0.30 | 6.22 | 0.75 | 5:15 p.m. |
occur all at the same time, the maximum load on the house did not equal the sum of the measured maximum loads for the individual levels. The measured maximum load for the whole house occurred at about 5:15 p.m. and was about 78 percent as large as the estimated load using I=B=R Guide C-30.

In Guide C-30 no credit is given for shading resulting from overhand except on the south wall. Since the research house faces the southeast this means that no external shading was considered on any of the windows. Observations made during the summer revealed that much of the SE wall was shaded by the overhang. The windows in the kitchen, breakfast room, and entry were always in the shade, whereas partial shading resulted on the windows in the den, living room, and dining room.

In view of the above, the loads for a maximum outdoor temperature of 95° F. were recalculated by adjusting glass gains to correspond to shading conditions observed. For these conditions the calculated load for the first level was 6.84 M.b.h. and for the second level 13.34 M.b.h. No change was made in the calculated load for the third level. The total measured loads shown in Table 7 are in fair agreement with the above calculated loads.

Comparisons similar to the above may be made for design and measured external sensible loads at the time the measured maximum total load occurred. Such comparisons show even better agreement between design and measured values except for the first level. In the ease of the first level, when actual shading conditions of the house were assumed, the measured values were only about 15 percent of the calculated values. This level had a concrete floor on the ground. Guide C-30 assumes the heat gain through such a floor to be zero, but measured floor surface temperatures in the den and recreation room were from 0.5° to 1.7° F. lower than the air temperature 3 inches above the floor. Using the average difference of 1.1° F. and a film coefficient of 1.08, it would appear that the heat flow from the rooms to the ground was approximately 1.2 B.t.u.h. per square foot of floor area, or about 0.60 M.b.h. for the entire first level. The remainder of the difference between the measured and calculated values of external sensible heat gain for the first level was probably due to inaccurate assumptions as to the allotment of the heat equivalent of total power used in the house. The total amount of power was metered, but the amount assigned to each level had to be estimated.

Sensible heat ratios (sensible heat divided by total heat) at the time of maximum measured total load ranged from a low of 0.75 for the first level and the house as a whole to a high of 0.81 for the second level. These values also agree quite well with the value of 0.75 assumed in I=B=R Guide C-30.

While there were differences between the measured and calculated heat gain values, these differences were not of alarming proportions. In practically every case the measured loads were less than the calculated values, thus indicating that the cooling load procedure contained in I=B=R Guide C-30 is sufficiently conservative to use for estimating residential design cooling loads.

3. Effects of Method of Operation and Type of Cooling System Employed on Cooling Load

A comparison of the effects of the methods of operation and the system used on the daily cooling requirements could have been determined by comparing the total daily cooling loads. However, there were day-to-day variations in internal and latent loads that were not a function of either the method of operation or the system employed. For this reason it was concluded that the external sensible load would serve as a better index for such comparisons.

The total external sensible cooling load on the house was determined by subtracting from the measured total load the latent load and the sensible heat gains within the house due to occupancy and to the use of power. The sensible load due to occupancy was assumed to be 250 B.t.u. per man-hour. The measured total load for each zone was obtained from the hours of operation, the rate of water circulation, and the temperature rise. The sum of the measured zonal total loads represented the measured total load for the house.

Figure 11 is a plot of the daily external sensible loads for both Series B-60 and C-60. It will be noted that isolating the house levels from one another did not affect the external sensible load on the house. One curve represents the external sensible load on the house obtained during the summer of 1959 when the house was being cooled by fan-coil units, and levels were isolated by plastic curtains installed at the top and bottom of the staircase as in Series C-60. The external loads observed when using the valance system were significantly
lower than those observed when using fan-coil units.

This could mean that there was some loss of conditioned air from the ducts of the fan-coil system directly to the attics, or that there was a change in the natural infiltration rate to the house between the time that the fan-coil tests and the valance tests were made.

Also air temperatures near the ceiling were much warmer when using the valance system. This could mean that ceiling losses with the valance system were reduced.

Twenty-four hour average heat flow rates through the ceiling of bedroom Number 1, as measured by a heat flow meter located on the inside surface of the ceiling at the approximate center of the room, are given in Table 8. Two values are given for the summer of 1959, one with and the other without the fans in operation. It is apparent that fan operation had a marked effect on the rate of heat transfer through the ceiling. Since fans were not used in the valance system, one set of data is sufficient to represent the heat flow rates.

At maximum outdoor temperatures of 95°F and above, there was little, if any, significant difference in the heat flow rate through the ceiling when using the valance and the fan-coil systems (fans in operation). At a maximum outdoor temperature of 90°F, the heat flow rate obtained in Series C-60 was in agreement with that obtained in Series G-59 when the fans were not in operation. Since the percent of total time that the fans operated in the fan-coil system increased as the outdoor air temperature increased, it is apparent that the total heat flow through the ceiling for Series G-59 was represented by the values obtained with no fan operation in mild weather; they gradually increased to equal the values obtained with continuous fan operation as the outdoor temperature increased. Thus, over the total range of outdoor temperatures, the actual heat transfer rate through the ceiling when using the valance probably was not far different from that obtained when using the fan-coil system. At least, it was apparent that there was not enough difference to explain the large differences in measured loads for these two systems. Consequently, it would appear that the difference resulted from loss of conditioned air from the ducts of the fan-coil system used in 1959 to the attic or the outdoors, to duct transmission losses to the attic, or to high infiltration rates when using the fan-coils.

The analysis described in Appendix A indicates that the infiltration rate in the summer of 1959 was about 0.4 air changes per hour greater than during the summer of 1960, and that in all probability, this increase was the result of the design of the fan-coil cooling system used in the house during the summer of 1959. Furthermore, this increase in infiltration rate could easily account for the changes in cooling loads observed in the summers of 1959 and 1960. Thus, the increase in both external latent and sensible loads should be charged to the design and operation of the fan-coil system rather

Table 8

<table>
<thead>
<tr>
<th>Series</th>
<th>Average Heat Flow Rate (Mean ± 95 Percent Confidence Interval)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C-60 (Valance System)</td>
<td>0.1 ± 0.4, 1.0 ± 0.2, 2.0 ± 0.8</td>
</tr>
<tr>
<td>G-59 (Fan-Coil Units, Fans On)</td>
<td>1.2 ± 0.2, 1.2 ± 0.5, 1.2 ± 0.9</td>
</tr>
<tr>
<td>G-59 (Fan-Coil Units, Fans Off)</td>
<td>0.2 ± 0.1, 0.3 ± 0.2, 0.4 ± 0.3</td>
</tr>
</tbody>
</table>

Figure 11. External Sensible Cooling Loads on House
than to any possible changes in the house between the two cooling seasons.

4. Reserve Capacity of System; Series B-60

In Table 9 the operating times of each of the zones and the chiller are expressed in terms of percent of total elapsed time between the hours of 1:00 p.m. and 9:00 p.m. During this period, the third level zone started continuous operation when the maximum outdoor temperature reached 92°F, while at design outdoor temperature (95°F), zones 1 and 2 operated about 52 and 76 percent of the time respectively. During this same period the chiller operated 78 percent of the time. Thus, it appears that the entire system, with the exception of zone 3, had ample capacity to carry the total cooling loads on a design day.

It is interesting to note that according to Table 9, the lower the level in the house, the greater the reserve capacity of the zone. However, Table 3 indicates that almost the reverse should be expected because the cooling output of the valance on the first level represented about 86 percent of the calculated cooling load, and on the third level 91 percent.

5. Transfer of Load

One of the problems encountered in heating or cooling a multilevel home is the transference of load between levels. Calculation procedures are available from which the load for each level can be determined. However, these procedures do not take into account the transference of load which may occur. If it were possible to heat or cool a home in a manner which produced no vertical air temperature gradients there would be little or no problem of load transfer. However, since there is always an air temperature gradient in a typically heated or cooled home there exists the natural tendency for the warmer air near the ceiling of the lower level to rise to the upper level whereas the cooler air near the floor of the upper level settles to the lower level. This interchange of air results in higher than calculated heating loads on the lower level and lower than calculated loads on the upper level.

The effect of this air interchange on cooling loads is just the reverse of this, that is, it creates higher than calculated loads on upper levels and lower than calculated loads on the lower level. The rate of air interchange between levels of a multilevel home is obviously governed by the ease with which air can flow from level to level, which is basically a function of the physical manner in which the levels are interconnected and the temperature gradients produced by the system.

WINTER

The baseboard and valance systems tested for heating in the research house produced quite different air temperature gradients as shown in Figure 12. As far as air interchange between levels is concerned, the most significant difference between the two systems is the air temperature produced 3 inches below the ceiling. It can be seen in Figure
that the valance system produced much higher ceiling temperatures than did the baseboard system, and consequently it could be expected that the air interchange between levels would be higher for the valance system.

In Table 10 the percentage of the total calculated load is given for each level. Table 10 also shows the percentage of the total measured load for the baseboard and valance systems conducted with, and without, the levels isolated. From this table it can be seen that when operating with the baseboard system and the levels isolated there was excellent agreement in the distribution of the calculated and measured loads. Assuming that the calculated loads are accurate, this agreement would indicate that there was negligible transfer of load when the house was heated with a baseboard system under isolated conditions. Also, it would indicate that the transference of load when operating with the levels not isolated is the result of air movement only. This was true only for the baseboard system as shown in Table 10.

Continuing with the assumption that the measured loads obtained while heating the house with baseboard under isolated conditions were the actual level heat losses, Table 11 was derived. In Table 11 the total shift in load along with the shift in load due to air movement in the staircase is given. The total shift in load was obtained for both the baseboard and valance series by dividing the percent of total level load for the not isolated condition by the corresponding percent of total level load for the isolated baseboard series. Thus, the values given in this portion of the table represent the percentage difference in level loads for either the baseboard or valance not isolated series as compared to the baseboard isolated series. The portion of the table that contains the change in load due to air movement in the staircase was derived by dividing the difference in percent of total level load between the not isolated and isolated series for either the baseboard or valance series by the percent of total level load of the baseboard isolated series.

Due to the manner in which Table 11 was derived, it can be noted that the total change in load and the change in load due to air movement in the staircase are identical for the baseboard system. It can also be noted that both load changes are higher for the valance system. The higher change in load due to air movement in the staircase, as previously discussed, can be attributed to the larger air temperature gradient produced by the valance system.

The higher total load change experienced when heating with the valance system resulted from increases in ceiling, wall, and glass heat transmission. Before beginning test work with the valance system it was expected that the higher ceiling air temperatures that it would produce, as compared to the baseboard system, would result in higher heat flow rates through the second and third level ceilings. For this reason heat meters were installed on the interior of the second and third level ceilings. Figure 13 depicts the relationship of the heat flow through the second and third level ceilings to the indoor-outdoor temperature difference for both the baseboard and valance tests. The curves drawn through these points, which were determined statistically, show that the ceiling losses for both the second and third levels were higher for the valance test. At an indoor-outdoor temperature difference of 70° F. the loss through the second level ceiling was 78 percent higher, while for the third level the loss was 67 percent higher. If it is assumed that the calculated ceiling heat losses are correct for the baseboard series, the percentage changes indicated above applied to the calculated ceiling losses of 3,600 B.t.u/h and 3,400 B.t.u/h for the second and third level ceiling respectively result in increases of 2,810 B.t.u/h and 2,280 B.t.u/h. Percentage-wise these represent respective gains in the second and third level of 9.6 percent and 11.8 percent of the total level load. For the second level the gain is in the right direction. That is, it in part accounts for
V. LOADS AND EQUIPMENT PERFORMANCE

V. LOADS AND EQUIPMENT PERFORMANCE

Figure 13. Heat Flow Through Living Room and Bedroom No. 1 Ceilings

Figure 14. Heat Flow Through Bedroom No. 1 Floor

some of the difference between the total change in load and the change in load due to air movement in the staircase, as indicated in Table 11. For the third level the gain is not in the right direction. However, there is one major source of heat transfer to the third level which is yet to be considered.

The construction and physical arrangement of the research house is such that the third level is directly above the first level. In addition, the separation between the entire first level ceiling area and third level floor area has 2-in. by 10-in. joists to which the acoustical tile of first level ceiling and the flooring and floor tile of the third level floor are affixed. The "U" factor for this composite section is 0.20 B.t.u.h per square foot-F. At an indoor-outdoor temperature difference of 55° F, the first level ceiling air temperature was observed to be approximately 110° F, while the third level floor air temperature was 73° F. Under these conditions, a flow of heat of 7.5 B.t.u.h per square foot from the first to the third level could be expected. During operation with the valance system a heat meter was affixed to the floor in the center of bedroom Number 1. The heat meter was attached to a multipoint recorder to produce a continuous record of the heat flow through the floor. In Figure 14 the relationship of heat flow through the floor to indoor-outdoor temperature difference is given. The test points shown in Figure 14 were not obtained from instantaneous values of heat flow and indoor-outdoor temperature difference, but rather from daily averages of continuous recordings of the two variables.

It is interesting to note that the heat flow through the floor reached a peak value of approximately 5.3 B.t.u.h. per square foot at an indoor-outdoor temperature difference of 55° F. One might expect the heat flow rate to continue to increase throughout the range of indoor-outdoor temperature difference; however, scrutiny of the system and its performance as given in Appendix B substantiates the curve of Figure 14.

Due to the nature of heat flow through the third level floor, as shown in Figure 14, it is impossible to assign a value to the percentage of heat transferred by this means which will be constant over the entire range of indoor-outdoor temperature differences encountered. However, the winter in Urbana, Illinois, as shown in columns 1 and 2 of Table 20 is not distributed uniformly over the range of outdoor temperatures. In fact, approximately 94 percent of the winter occurs at temperatures above 20° F. (indoor-outdoor temperature difference less than 55° F.). Thus, in order to establish a value which is representative of the percentage of level load transfer it is necessary to examine the percentage of transfer over a range of outdoor temperatures. An analysis of the heat flow...
through the floor covering a range in outdoor temperature of 20° F. to 45° F. (approximately 57 percent of an Urbana, Illinois, winter) revealed that the percentage of level load transferred through the floor was relatively constant for both the first and third levels. The values obtained were a 17 percent increase in the first level load and a 15 percent decrease in the third level load.

From the data taken at the research house there was one other factor which materially changed the distribution of loads between the baseboard and valance series. This factor was the heat transfer through walls. Referring to Figure 15 it can be seen that at design conditions the valance system produced recreation room wall surface temperatures approximately 8.5° F. above those produced by the baseboard system. This elevated inside wall temperature would be responsible for an increase in wall transmission losses of 11.3 percent. Applying this 11.3 percent wall transmission increase to the first and second levels only (since they were the levels in operation) resulted in increases for the first and second level loads of 3.9 percent and 1.4 percent respectively. Assuming that the window losses of the first and second levels were increased by the same percentage as the wall losses, the resultant increases in the first and second level loads would be 3.7 percent and 5.0 percent respectively.

In Table 11 the total change in load and change in load due to air movement in staircase is given. It can be noted that for the valance system there is a difference between these two load changes, and in the discussion which just preceded, an attempt has been made to account for these differences. Taking into account all factors discussed, Table 5 was formed. Referring to Table 5 it can be seen that, after taking all known factors into consideration, there was good agreement between the measured and accounted for load changes on the first and second levels of the house. The agreement for the third level, however, was not very good. One point that should be noted in Table 5 is that the accounted for changes in loads for the first and second levels were less than the measured load changes while for the third level the table indicates that in some manner, still unaccounted for, heat was being transferred from the first and second levels to the third level.

A very good possibility of transference of load existed between the second and third levels through approximately 80 square feet of common wall. Since this common wall was between the upper part of the living room and entry and the lower part of bedroom Number 1, lavatory A and bath Number 2, and since it has already been shown that the air temperature in the upper portion of rooms heated with valance was as much as 30° F. warmer than the temperature of the air in the lower half of the rooms, it is apparent that the wall surface temperature on the living room and entry side of the common wall was considerably higher than on the other side. Therefore there was, in all probability, a flow of heat through the common wall from the second to the third level.

In addition to the preceding statement, it has been shown that the inside surface temperature of the outside walls for the first and second levels was higher when using the valance system than when using the baseboard system. It seems reasonable to assume that the inside surface temperatures of the inside walls were also higher and thus, with operation of the valance on the second level but not on the third, it seems reasonable to assume that this too would tend to promote a flow of heat through the common wall from the second to the third level of the house. The existence of a flow of heat from the second to the third level would tend to make the first and last lines of Table 5 to be in better agreement. Investigation of heat flow through the common wall was not made during either heating season.

Summer

To ascertain the effect of air movement on the distribution of cooling loads, the house levels were isolated in exactly the same manner as they were for determining the effect of air movement on heat-
ing loads. The results of this series of tests are shown in Table 12. With the plastic curtains in place, the zone for the third level started 100 percent operation at a maximum outdoor temperature of 96° F. as compared to 92° F. when the staircase was open. At a maximum outdoor temperature of 90° F. the operating time of the third zone was reduced from 92 percent of the total time to 77 percent. This represents a reduction in cooling load of about 16 percent. There was no significant change as far as the zone for the second level was concerned, but the operating time of the zone for the first level was increased from 42 to 50 percent of the time when the plastic curtains were in place. This represents a 19 percent increase in the cooling load of the first level. From this analysis, it is apparent that air movement was responsible for a sizeable transference of cooling load between the first and third levels.

B. ENERGY CONSUMPTION

1. Winter

Figure 9 shows the daily fuel and power consumption obtained over a range of indoor-outdoor temperature differences when using the valance system. For this particular heating system, these are the only two items chargeable to energy consumption.

<table>
<thead>
<tr>
<th>Table 12</th>
<th>Table 14</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Effect of Air Movement Between House Levels on Summer Load Distribution</strong></td>
<td><strong>Breakdown of Cooling Loads — Valance and Fan-Coil Systems</strong></td>
</tr>
<tr>
<td>Maximum Outdoor Temperature = 90° F.</td>
<td>Maximum Outdoor Temperature, Degrees F. 85 90 95</td>
</tr>
<tr>
<td>Open Staircase Series B-60 and C-60</td>
<td>Load in 100,000 B.t.u. Per Day</td>
</tr>
<tr>
<td>Closed Staircase Series C-60</td>
<td>(Mean ± 95 Percent Confidence Interval)</td>
</tr>
<tr>
<td>Elapsed Time from 9:00 a.m. to 9:00 p.m.</td>
<td>Series B-60 and C-60</td>
</tr>
<tr>
<td>First Level Zone 42</td>
<td>Total Load 1.79 ± 0.19 2.31 ± 0.17 2.82 ± 0.26</td>
</tr>
<tr>
<td>Second Level Zone 60</td>
<td>Latent Load 0.32 ± 0.07 0.53 ± 0.06 0.43 ± 0.09</td>
</tr>
<tr>
<td>Third Level Zone 93</td>
<td>Sensible Load 1.52 ± 0.16 1.98 ± 0.14 2.39 ± 0.24</td>
</tr>
<tr>
<td>Chiller 64</td>
<td>Internal Sensible Load 0.22 ± 0.05 0.50 ± 0.03 0.48 ± 0.05</td>
</tr>
<tr>
<td><strong>100 percent operation reached at a maximum outdoor temperature of 92° F.</strong></td>
<td><strong>Exterior Sensible Load 1.00 ± 0.17 1.46 ± 0.15 1.91 ± 0.23</strong></td>
</tr>
<tr>
<td><strong>100 percent operation reached at a maximum outdoor temperature of 96° F.</strong></td>
<td><strong>Sensible Heat Ratio 0.85 ± 0.83 ± 0.83</strong></td>
</tr>
<tr>
<td><strong>100 percent operation reached at a maximum outdoor temperature of 96° F.</strong></td>
<td><strong>Total Load 2.20 ± 0.30 3.33 ± 0.55 4.46 ± 0.87</strong></td>
</tr>
<tr>
<td><strong>100 percent operation reached at a maximum outdoor temperature of 96° F.</strong></td>
<td><strong>Latent Load 0.94 ± 0.10 0.78 ± 0.19 1.02 ± 0.26</strong></td>
</tr>
<tr>
<td><strong>100 percent operation reached at a maximum outdoor temperature of 96° F.</strong></td>
<td><strong>Sensible Load 1.69 ± 0.25 2.55 ± 0.45 3.44 ± 0.71</strong></td>
</tr>
<tr>
<td><strong>100 percent operation reached at a maximum outdoor temperature of 96° F.</strong></td>
<td><strong>Internal Sensible Load 0.40 ± 0.01 0.60 ± 0.08 0.90 ± 0.13</strong></td>
</tr>
<tr>
<td><strong>100 percent operation reached at a maximum outdoor temperature of 96° F.</strong></td>
<td><strong>Exterior Sensible Load 1.26 ± 0.22 2.05 ± 0.40 2.84 ± 0.94</strong></td>
</tr>
<tr>
<td><strong>100 percent operation reached at a maximum outdoor temperature of 96° F.</strong></td>
<td><strong>Sensible Heat Ratio 0.76 ± 0.77 ± 0.77</strong></td>
</tr>
</tbody>
</table>

2. Summer

Daily power consumptions for the valance and the fan-coil systems are shown in Table 13. The total daily cooling loads as measured in 1959 (fan-coil system) and 1960 (valance system) are shown in Table 14. Table 15 shows the power consumption per 100,000 B.t.u. total cooling load for each of the systems. These values were obtained by dividing the power consumption values in Table 13 by the corresponding daily total heat gain (in units of 100,000 B.t.u.) shown in Table 14. The rate of power consumption for the condenser was about twice as high in 1960 as in 1959. This unit was outdoors and it is probable that exposure to the weather throughout the year caused an increase in friction in both fan and motor bearings. In addition, the fan belt was tightened between the summers of 1959 and 1960, and it is quite possible that some slippage had been occurring prior to the tightening of the belt.

There was also a difference in the unit power consumption of the circulator. This resulted from the fact that the circulator operated continuously in the fan-coil system and only operated during the periods when at least one zone was requiring cooling in the valance system. On the basis of average values, there appeared to be a slight difference in the unit power consumption of the compressor for the valance and fan-coil systems. This difference proved to be statistically insignificant. Therefore an average value of 17.0 kilowatt-hours per 100,000 B.t.u. was assumed. Total power consumptions based on this assumption are shown in parentheses in Table 15.

Had there been no losses from the ducts of the fan-coil system, no changes in infiltration rate, no change in internal load, and no change in unit power consumption of equipment common to both systems, the total daily power consumption for the
fan-coil system would have exceeded that of the valance system by the power required by the fan motors in the fan-coil units and the additional power required for continuous operation of the pump. For design weather, this amounted to 0.8 and 0.2 kilowatt-hour per 100,000 B.t.u. cooling load for the fan-coils and the pump respectively, or a total of 2.94 kilowatt-hours per day.

However, there were losses from ducts and differences in infiltration rates, internal loads, and in the methods of control, which did affect the total cooling load as shown in Table 14.

In Table 16 the total daily power consumption for the valance system has been adjusted to represent conditions that would have prevailed had not the unit power consumption of the condenser changed between the time of the fan-coil and valance tests. In addition to the observed daily power consumption for the fan-coil system, two adjusted sets of values are shown. One represented the daily power consumption that would have been expected had there been no change in internal loads and the other was adjusted to represent operation of the fan-coil system assuming the same rates of infiltration as well as the same internal loads that existed when testing the valance system.

From Table 16 it will be noted that on an “as observed” basis (items a and b) the power consumption of the fan-coil system for a design day exceeded that of the valance system by 35.3 kilowatt-hours per day or by 67 percent. Correcting the power consumption of the fan-coil system to the same internal loads as existed when testing the valance system reduced this difference to 31.5 kilowatt-hours per day or 60 percent. This increase represents the additional power used to operate the fan motors and to take care of the additional external cooling load created by increased infiltration. Correcting the power consumption of the fan-coil system to both the same internal cooling load and the same infiltration rate reduced the difference in power consumption of the two systems to 2.9 kilowatt-hours per day or 5.5 percent.

The 2.9 kilowatt-hours per day represents power increases which are inherent to the design and method of operation of the fan-coil system. The difference between 2.9 and 31.5, or 28.6 kilowatt-hours per day represents an increase in power consumption which according to Chapter V, Section A 3, and Appendix A probably was the result of air leakage from the ducts of the fan-coil system and could have been eliminated had the ducts been airtight in construction or located entirely within the air-conditioned space.

C. COSTS

1. Installation

An accurate estimate of the installation cost of a heating and air conditioning system is difficult to obtain as there is no satisfactory way to estimate...
V. LOADS AND EQUIPMENT PERFORMANCE

Table 17B

<table>
<thead>
<tr>
<th>Nominal Pipe or Tube Size</th>
<th>Unit Installation Time, Hours</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Pipe</strong></td>
<td><strong>Copper</strong></td>
</tr>
<tr>
<td><strong>Pipe</strong></td>
<td><strong>Copper</strong></td>
</tr>
<tr>
<td><strong>Pipe</strong></td>
<td><strong>Man-hours per piece</strong></td>
</tr>
<tr>
<td>1⁄₂&quot;</td>
<td>0.06</td>
</tr>
<tr>
<td>¾&quot;</td>
<td>0.07</td>
</tr>
<tr>
<td>1&quot;</td>
<td>0.12</td>
</tr>
<tr>
<td>1½&quot;</td>
<td>0.19</td>
</tr>
<tr>
<td>2&quot;</td>
<td>0.32</td>
</tr>
</tbody>
</table>

The basis of labor estimates is shown in Tables 17A and 17B.

A summary of total installation costs is given in Table 18. This table indicates that for heating only, the baseboard system should cost slightly less to install than the valance system. However, for year-round operation the expected installation cost of a valance system is approximately 20 percent less than the installation cost of a combination baseboard heating and fan-coil cooling system. A year-around hydronic system had an estimated installation cost approximately $1,250 above the cost of a system designed for heating only. The chiller represented the major portion of the additional cost.

It is interesting to note that even though the unit installation times used as a basis of estimating labor costs all appeared to be liberal, the total direct labor costs for the installation of a hydronic system represented only about 15 percent of the total installation cost and therefore, if any appreciable reduction in installation cost is to be made, ways of reducing material cost must be considered.

2. Operating Costs

In Chapter V, Section B, daily fuel and power consumptions are related to indoor-outdoor temperature differences or to maximum outdoor temperatures. If the frequency of occurrence of different outdoor temperatures is known, these data may be used to estimate yearly fuel and power consumptions.

In Tables 19 and 20 the seasonal fuel and power consumptions for the several systems are estimated by summing the product of the corrected daily con-
Table 20  
Seasonal Fuel and Power Consumption — Winter Operation (Based on records of U.S. Weather Bureau Station at University of Illinois. Includes months of January, February, March, April, May, September, October, November, and December from September 1936 to May 1941).

<table>
<thead>
<tr>
<th>Average</th>
<th>Average Gas</th>
<th>Baseboard, Series N-59</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outdoor Temp. F.</td>
<td>No. Days Per Year</td>
<td>Valance, Series E-60</td>
</tr>
<tr>
<td></td>
<td></td>
<td>eu. ft. per season</td>
</tr>
<tr>
<td>1</td>
<td>2</td>
<td>3</td>
</tr>
<tr>
<td>-10 to 5</td>
<td>0.2</td>
<td>2400</td>
</tr>
<tr>
<td>-5 to 0</td>
<td>0.4</td>
<td>2267</td>
</tr>
<tr>
<td>0 to 5</td>
<td>0.8</td>
<td>2144</td>
</tr>
<tr>
<td>5 to 10</td>
<td>2.2</td>
<td>1561</td>
</tr>
<tr>
<td>10 to 15</td>
<td>4.6</td>
<td>1838</td>
</tr>
<tr>
<td>15 to 20</td>
<td>7.6</td>
<td>1684</td>
</tr>
<tr>
<td>20 to 25</td>
<td>13.6</td>
<td>1331</td>
</tr>
<tr>
<td>25 to 30</td>
<td>25.4</td>
<td>1378</td>
</tr>
<tr>
<td>30 to 35</td>
<td>33.8</td>
<td>1225</td>
</tr>
<tr>
<td>35 to 40</td>
<td>40.0</td>
<td>1072</td>
</tr>
<tr>
<td>40 to 45</td>
<td>23.4</td>
<td>1215</td>
</tr>
<tr>
<td>45 to 50</td>
<td>22.6</td>
<td>796</td>
</tr>
<tr>
<td>50 to 55</td>
<td>20.8</td>
<td>613</td>
</tr>
<tr>
<td>55 to 60</td>
<td>19.8</td>
<td>459</td>
</tr>
<tr>
<td>60 to 65</td>
<td>22.0</td>
<td>306</td>
</tr>
<tr>
<td>65 to 70</td>
<td>19.0</td>
<td>153</td>
</tr>
<tr>
<td>70 to 75</td>
<td>13.6</td>
<td>0</td>
</tr>
<tr>
<td>Seasonal Totals</td>
<td>209.8</td>
<td>229439</td>
</tr>
</tbody>
</table>

* Power consumption high because of insufficient radiation on first level of house.

sumptions (Table 16 and Figure 9), and the frequency of occurrence of each outdoor temperature for a typical winter or summer in Urbana, Illinois. These total power and fuel consumption estimates are for specific systems operating in a specific house at certain specified conditions and therefore would not apply to these same systems used in other houses or climates. However, they may be used as a basis of comparison of relative fuel and energy requirements for the several systems included in the study.

Seasonal fuel and energy requirements are all that are required to compare the relative efficiency of the various systems; however, operating cost is dependent not only on the total consumptions of fuel and power, but also on the price that must be paid for these sources of energy.

Table 21 shows the summer and winter operating costs for each of the systems based on the seasonal fuel and power consumptions in Tables 19 and 20 and the average unit cost of natural gas and electrical power prevailing in Urbana at the time of the test program. According to Table 21, the seasonal cost of operating the fan-coil system was $149.19. However, this value cannot be compared directly with the seasonal cost of operating the valance system in that, as pointed out in Chapter V, Section B, there were changes in infiltration and internal loads between the times of the tests on the fan-coil and valance systems which had an effect on operating cost. Had the fan-coil system been designed so that there could be no duct losses to unconditioned space and the internal heat gains been the same as when testing the valance system, the seasonal operating cost would have been reduced to $118.56. Since the seasonal cost of operating the valance system in summer was $110.13 it is evident that under the most ideal conditions, the cost of operating a fan-coil system would exceed that of the valance system by approximately $8.50 or 7.7 percent.

In summer, the seasonal cost of operation ranged from approximately $110.00 for the valance system to $150.00 for the fan-coil system as observed, whereas for winter the seasonal costs were about $170.00 and $138.00 for the valance and baseboard systems respectively. Summer operating costs in Table 21 are based on the assumption that the system would be operated on all days for which the maximum temperature was 76° F. or higher, regardless of the time of the year that these days occur. According to Table 19, about 35 percent of the total power consumption required for summer operation occurred on days for which the maximum outdoor temperature was between 76° F. and 85° F. When the maximum outdoor temperature is less
V. LOADS AND EQUIPMENT PERFORMANCE

than 85° F. the need for cooling is not very great. Since, in the average home, the cooling system normally would not be used during many of these days, actual summer operating costs probably would be as much as 25 percent less than those shown in Table 21.

In winter the higher ceiling, wall, and glass losses produced by the valance system (see Chapter V, Section A 5) were reflected as an increase of about $25.00 per season in the cost of the fuel used.

On a year-around basis, the cost of operating the valance system was about $11.00 less than the cost of operating the combination baseboard fan-coil system as installed. Had there been no duct losses from the fan-coil system, the year-around cost of operating the combination baseboard fan-coil system would have been about $23.00 less than that of the valance system.

D. CLEANLINESS OF OPERATION

The tests reported in this bulletin were not designed specifically to yield information regarding the cleanliness of operation of the valance system. A longer period of operation would be required to give definite answers. After one complete year of operation, no dirt patterns were observed on walls or ceilings of any of the rooms of the house, neither were odors observed in the rooms during either summer or winter operation. The summer tests were run immediately after the system was installed and it was observed that the condensate collected from this system had a yellow color. Also an oily deposit was left on the inside of the trough intended for the collection of condensate. Apparently, the condensate was removing an oil from the surface of the tube and fins which was left there from the manufacturing process. There was little or no evidence of such a deposit the second summer of operation. So apparently, had the system been operated through a heating season first, or had the units been more thoroughly cleaned following fabrication, no oily film would have been deposited on the trough.

During the one year of operation, there was no evidence of the collection of dirt or lint on the valance elements. But again, this testing period was too short for the results to be conclusive.
VI. COMFORT CONDITIONS — I=B=R HYDRONIC RESEARCH HOUSE

A. FACTORS TO BE CONSIDERED

Comfort is a human response and can neither be defined nor measured as precisely as such phenomena as temperature. However, it is known that some of the environmental factors influencing the comfort of the occupant of an air-conditioned space are the room air temperature and temperature variations, air movement, humidity, and surface temperatures of the occupied space. At the present time, it is generally agreed that within the occupied portion of the conditioned space, air temperature should be uniform at approximately 75° F., air movement should not be perceptible, and surface temperatures should approximate the room air temperature. These factors (as related to the baseboard and the valance systems for heating, and to the valance and fan-coil systems for cooling) will be discussed in the paragraphs which follow.

B. WINTER

1. Room Air and Surface Temperatures

In order to show the effects of indoor-outdoor temperature difference and position in the room on vertical temperature gradients, it is convenient to present the data in tabular form as in Table 22. It will be observed that as the indoor-outdoor temperature difference increased, all air temperature differences measured from the 30-inch level also increased, and in all cases the air temperature differences when using baseboard were smaller than when using the valance system. The high air temperature near the ceiling obtained with the valance system had no adverse effects on comfort conditions within the room as long as the bottom of the valance unit was at least 6 inches above head level. If the unit was lower than this, the layer of hot air would probably extend to a low enough level that it would be disagreeable to occupants of the room while standing. Floor surface temperatures at the center of the room were always warmer with valance than with baseboard.

Comparing temperatures in the center of the living room with corresponding temperatures 3 feet from the window shows that at 30 inches above the floor the air temperatures remained constant at 73° F. with either system. At an indoor-outdoor temperature difference of 80° F. the air temperature 3 inches above the floor when using the valance system was 66° F. in the center of the room and 63° F. 3 feet from the windows. With baseboard these two temperatures were 70° F. and 72° F. respectively.

At the center of the room and at an indoor-outdoor temperature difference of 80° F., the floor surface temperature when using valance was 73° F. while for baseboard it was only 65° F. On the other hand, 3 feet from the windows the floor surface temperature when using valance was 63° F., 10° F. cooler than at the center of the room while with baseboard it was 67° F., 2° F. warmer than at the center of the room and 4° F. warmer than that obtained with valance.

2. Effect of Room Construction on Air and Surface Temperatures

The discussion in the preceding section, as well as in all other sections of this report, is based on

<table>
<thead>
<tr>
<th>Room</th>
<th>30-In. Level Temperature Maintained at 73° F.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Valance &amp; Baseboard</td>
<td>Indoor-Outdoor Temperature Difference, °F.</td>
</tr>
<tr>
<td>Location</td>
<td>Valance</td>
</tr>
<tr>
<td>3° below ceiling</td>
<td>86</td>
</tr>
<tr>
<td>60° above floor</td>
<td>75</td>
</tr>
<tr>
<td>30° above floor</td>
<td>73</td>
</tr>
<tr>
<td>3° above floor</td>
<td>79</td>
</tr>
<tr>
<td>Floor surface</td>
<td>73</td>
</tr>
<tr>
<td>3° below ceiling-30°</td>
<td>13</td>
</tr>
<tr>
<td>60°-30°</td>
<td>2</td>
</tr>
<tr>
<td>3°-30°</td>
<td>-3</td>
</tr>
<tr>
<td>Floor surface-30°</td>
<td>0</td>
</tr>
</tbody>
</table>

* All temperatures rounded off to nearest degree F.
VI. COMFORT CONDITIONS

Comparing data collected at Pierce with that obtained at the University of Illinois makes it possible to demonstrate some of the effects house construction has on the performance of these systems. In Figure 16 the vertical temperature gradients in the center of the living room and near the windows of the living room of the I=B=R Hydronic Research House are compared with temperature gradients obtained in the test room in the Pierce Foundation studies. There were minor variations in indoor temperature and outdoor air temperature during the various tests and in order to make it convenient to compare results, temperature differences based on the observed indoor air temperature at the 30-inch level have been plotted rather than the actual temperatures themselves. It will be noted that, generally speaking, the air temperatures near the floor were lower in the I=B=R Research House than at Pierce Foundation. It will also be noted that the heat loss of the room per square foot of floor area apparently had more effect on air temperatures near the floor when the valance system was in use than when using the baseboard system. This was particularly true in the area 2 to 3 feet from the window.

With the exception of the area just in front of the windows, floor surface temperatures were always warmer with the valance system than with the baseboard system. It is quite possible that the low floor surface temperature observed 3 feet from the windows in the living room of the I=B=R Hydronic Research House was due to the location of furniture. A davenport was located just in front of this window, and it is quite possible that this shielded the area of the floor (at which the thermocouple was located) from some of the radiation from the ceiling area.

Data taken in the I=B=R Hydronic Research House indicated that the air temperature near the floor was too low and the air movement at this level was too high for comfort. However, examination of the curves of Figure 16 shows that this was not the case for data taken at the Pierce Foundation where the room was better insulated than the living room of the I=B=R Hydronic Research House. At the center of the test room at Pierce Foundation there was no significant difference between the air temperatures obtained with valance and baseboard at any level from the floor to 60 inches above the floor. The floor surface temperature was warmer with valance than with baseboard. Near the windows, the air temperature 3 inches...
above the floor was only slightly cooler with valance than with baseboard. These data would indicate that the performance of the valance system in a room or home which has a reasonable degree of insulation is entirely satisfactory. It is only in houses in which the insulation is poor that the performance of the valance system is questionable.

3. Room Temperature Balance

In Table 23 the maximum room temperature deviations between rooms for various wind conditions are given for both the baseboard and valance systems. The temperature difference indicated in these tables is the maximum average difference in the air temperature 30 inches above the floor which existed between the room in which the thermostat was located and any other room on the same house level. In all cases the air temperature in the room in which the thermostat was located was deducted from that in the other rooms. Thus, a positive difference indicates the room containing the thermostat to be the cooler, and vice versa. This table clearly shows that both systems were in reasonably good balance for conditions of no wind, and both were seriously affected by wind direction. The temperature balance of the third level was extremely dependent on wind direction for both systems. Referring to Table 23 it can be seen that for the baseboard series the maximum temperature deviations on the third level varied from +3.7° F. to -5.8° F., depending on whether the thermostat was on the windward or leeward side of the house. Thus, if the thermostat rooms were maintained at a given temperature the other rooms on the third level could be expected to experience a 9.5° F. variation in temperature with extreme variations in wind direction. A similar analysis of algebraic differences used in conjunction with Table 23 yields the maximum temperature variation experienced on the three levels with both systems.

In addition to wind direction, temperature balance was affected by the sun. From the data collected during both heating seasons it was not possible to make a quantitative analysis of the sun’s effect on temperature balance. However, observations of room air temperatures and system performance did give a general indication of the effect of the sun. On clear days, shortly after sunrise and throughout the morning, the air temperatures in the rooms in the eastern portion of the house would rise above those in the western portion, while in the afternoon the opposite occurred. The thermostat for the third level was located in the eastern portion of the house, and on clear sunny mornings the elevated temperature in that portion resulted in little or no circulator operation. Consequently, the temperatures of the rooms having a western exposure fell below the thermostat setting. Then in the afternoon, as the rooms in the eastern portion cooled and circulator operation was resumed, the sun effect caused the temperature of the rooms having a western exposure to approximate the thermostat setting. The thermostats for the first and second levels were located in the western portion of the house which resulted in the temperature of the rooms in the eastern portion of this level being above thermostat setting in the morning and below in the afternoon.

Considering the magnitude of the observed effects of both wind and sun on room temperature balance, it would appear that room by room zoning would be required to affect any real improvement in room temperature balance. However, it may be that the effects of wind on room temperature balance, as reported in this bulletin, are exaggerated by the house construction. When the I=B=R Hydronic Research House was built, no special instructions were given to the builder since it was desired that the house construction be representative of the usual construction practices of today. No vapor barrier other than the paper on the back of the insulating blanket was installed. Infiltration tests made during the winter of 1960 indicated that the infiltration rate for the house was expressed by the equation

\[ I = 0.227 + 0.0725 W + 0.0205 T \]

where
- \( I \) = infiltration rate in air changes per hour
- \( W \) = wind velocity in miles per hour
- \( T \) = indoor-outdoor temperature difference in degrees Fahrenheit

The above equation indicates an air change rate of
almost 3 per hour at design conditions of 80° F., indoor-outdoor temperature difference and a wind speed of 15 m.p.h. The generally accepted infiltration rate used in calculating design heating loads is from 0.7 to 1.0 air changes per hour.

Since the completion of the tests reported in this bulletin, a vapor barrier consisting of a 4 mil polyethylene film has been installed in the ceilings and sidewalls of the second and third levels of the research house and the infiltration tests are being repeated. Insufficient data are available at the time of writing this report to definitely establish the effectiveness of this film in reducing the infiltration rate, but preliminary results indicate that the effect may be quite pronounced. If this is borne out, it would appear that the effect of wind on room temperature balance would be greatly diminished by the proper installation of a vapor barrier.

4. Air Movement

Air movement in the living and dining room area is shown for both the baseboard and valance systems in Figure 17. The air velocities shown in
this figure were determined by measuring the time taken by a smoke formation laid in the room to travel a given distance. In portions of the living room-dining room area air movement was random in direction and below 10 f.p.m. In these areas no velocity indication has been made.

Referring to Figure 17, it can be seen that air velocities above 25 f.p.m. were observed in only one location across the floor of the living-dining room area while heating with baseboard. This observation was made near a doorway which led to the kitchen area. Along the outside walls a vertical downward movement of air at a velocity of approximately 35 f.p.m. existed from the ceiling to about the mid-height of the wall. At this level the downward movement of air was intercepted by an upward movement of warm air (produced by the baseboard) at a velocity of approximately 15 f.p.m. Thorough mixing of the two air currents took place with the result that no measurable air velocities were observed in the living zone.

In Figure 17b the air velocities observed while heating with valance are given. Referring to this figure it can be noted that air velocities ranging from 18 to 45 f.p.m. were observed just above the floor. In general all air movement along the floor was from the outside walls toward the staircase. This air movement originated at the outside walls and had essentially the same initial velocity as was observed while heating with baseboard. However, while heating with valance there was no source of heat along the lower extremities of the outside wall to intercept the cold air and it continued to settle and move across the floor.

Alfred Koestel and G. L. Tuve have defined a draft as any localized feeling of coolness or warmth of any portion of the body, due to both air movement and air temperature, with humidity and radiation considered constant.\(^{(10)}\) Data obtained by F. C. Houghten, Carl Gutberlet, and Edward Witkowski at the ASHVE laboratory\(^{(11)}\) indicate that almost no occupant of the room would object to the combination of air temperature and movement measured in the living room of the research house at a point 3 inches above the floor and 3 feet away from the windows when the room was being heated by a baseboard system and the outdoor temperature was 0° F., while more than 40 percent of the occupants would object to the combination of air temperature and movement occurring at this same location and at the same outdoor temperature when the room was being heated by a valance system. These same data indicate that for conditions measured at the center of the room, the incidence of complaints of cool ankles registered by the occupants would be approximately 9 percent for the baseboard system and 25 percent for the valance system.

5. Humidity

For both the 1959 and 1960 heating seasons no effort was made to control humidity in the research house. The humidity obtained with both heating systems at an outdoor temperature of 0° F. was approximately 20 percent, which is considered to be acceptable for winter operation.

C. SUMMER

1. Room Air Temperature and Temperature Balance

Table 24 shows that for all practical purposes the valance system maintained constant average room air temperatures 30 inches above the floor at all maximum outdoor air temperatures up to 95° F. This corresponds favorably with the air temperatures maintained with the fan-coil system during the summer of 1959.

Figure 18 shows the room air temperatures obtained during a special test on September 1, 1960. Maximum and minimum air temperatures were obtained for each cycle of operation and, during times of continuous operation, the air temperatures were read hourly. The maximum outdoor temperature during this test was 95° F. Cyclic temperature fluctuations in the three rooms in which the thermostats were located were about 1.5° F. on the first level and about 2.5° to 3° F. on the second and third levels. These were assumed to be representative of the operating differentials of the thermostats used during the tests.

Cyclic operation was obtained throughout the day on the first level. The average air temperature in the recreation room was held close to 76° F. from 9:00 a.m. to 9:00 p.m. dropping off slightly

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<td>Third Level</td>
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Noon
Central daylight time

Figure 18. Room Air Temperatures, September 1, 1960
preceding and following this period of time. The average air temperature in the den climbed to a maximum of about 77°F at 9:00 a.m. while the sun was on both walls of the room and then decreased slowly to a minimum of about 74°F at 6:00 p.m. when both exposures of the den were in the shade.

The large roof overhang on the second level shaded the walls of this level throughout most of the day and hence the effect of direct sunlight on room air temperature balance is not as well defined as on the other levels. Even so, the kitchen was a little warmer during the morning hours than in the afternoon.

The thermostat for the third level was located in bedroom Number 3, on the northeast side of the house. The air temperature in this room slowly increased throughout the morning and continued until about 2:00 p.m. The total increase was about 1.5°F. The zone valve for this level was in the open position from 11:00 a.m. until 7:00 p.m. indicating that the cooling load for the third level during this period was larger than the cooling output of the units installed. However, the increase in air temperature in the room in which the thermostat was located was too small to be of consequence.

On the other hand, the air temperature in bedroom Number 1 decreased from 75°F at 7:00 a.m. to about 72.5°F at 11:00 a.m. During this period, bedroom Number 1 was on the shaded side of the house. When bedroom Number 1 received direct sunlight in the afternoon, its air temperature increased from the low of 72.5°F to a high of about 75.5°F. This again demonstrates the effect of solar radiation on the hour by hour distribution of cooling load between the rooms.

To determine the uniformity of temperature from room to room, the air temperature 30 inches above the floor in each of the other rooms of that level. Table 25 shows the results of this analysis for two test days, September 1, 1960 (valance system) and August 13, 1959 (fan-coil system). Both systems exhibited the same characteristics with room temperatures ranging from 3.2°F lower than the temperature in the room in which the thermostat was located to 2.0°F higher. A good share of this spread was caused by direct solar radiation. Those rooms on the sunny side of the house always tended to be warmer than those on the shaded side.

While these differences in temperature were sufficient in some cases to be noticed as one went from one room to another, it is doubtful that better control could be obtained without the use of additional zones. Low temperatures occurred at the time when the load in the room in which the thermostat was located was at its maximum.

Figure 19 shows typical temperature gradients.
VI. COMFORT CONDITIONS

Figure 20. Summer Air Movement in Living and Dining Rooms — Valance System

in the living room and bedroom Number 3 for Series G-59 (fan-coil) and C-60 (valance) for a day in which the maximum outdoor temperature was 95° F. Both systems produced the same temperature conditions from the floor to a height of about 60 inches above the floor. Above this level the air temperatures were higher when using the valance system.

The air temperature 3 inches below the ceiling in bedroom Number 3 was only 73.5° F. when using fan-coils. The drop in temperature at the ceiling level was caused by cool air from the air conditioning system being blown across the ceiling. The corresponding air temperature obtained with the valance system was almost 81° F. Higher temperatures directly under the ceiling could reduce the total heat gains into the room; however, heat meter readings taken on the ceiling of the bedroom did not confirm this.

As far as could be determined, the higher ceiling temperatures obtained with the valance system were not sufficient to affect the degree of comfort experienced by occupants of the room.

2. Air Movement

Figure 20 shows the air movement in the living room as measured on August 23, 1960. This was a clear day with a maximum outdoor temperature of about 88° F. The valance units were operating at the time of the observations.

Along the walls below the valance units there was movement of cooled air toward the floor at a velocity of about 30 f.p.m. Air movement across the floor was from the dining room and the outside wall toward the living room and staircase. This movement was at speeds of 15 to 20 f.p.m.

Air movement near windows was upward at a speed of 28 to 30 f.p.m. This upward movement continued to a point just below the bottom of the valance unit. Here the upward current of warm air from the window and the downward current of cool air from the valance unit mixed to form a current of air projecting downward at an angle of about 45 degrees at a maximum velocity of about 30 f.p.m. Within a distance of 18 to 24 inches this current of air diffused with other air in the room and the velocity was dissipated. Air movements could not be detected without instruments except directly under the valance units located where there were no windows and within 12 inches of the wall.

3. Humidity

Figure 21 is a graphic log of indoor air temperatures and humidities for two test days, one with the fan-coil system in operation and the other with the valance. The difference in indoor temperature was due to the thermostat setting and not the type of cooling equipment in use. In both tests the variation in the indoor temperature was about the same.

There was a distinct difference in the indoor
relative humidity obtained with the two cooling systems. When using the fan-coil system, chilled water was constantly circulated through the coil and the operation of the fan was governed by the thermostat. With this method of operation there was no opportunity for re-evaporation of condensate from the coils. As a result the indoor relative humidity remained at approximately 50 percent at all times.

The only means of controlling the output of the valance system was to have the thermostat control the operation of the pump. This permitted the coil surfaces to warm up during the off periods. As a result re-evaporation took place during periods of light load and the relative humidity in the house increased. Thus each morning the relative humidity in the house rose to about 65 percent and, as the cooling load increased during the heat of the day, the relative humidity decreased to about 50 percent. Even during the hottest part of the day some re-evaporation did take place during the off-periods as is evidenced by the saw-toothed nature of the humidity curve.
VII. SUMMARY AND CONCLUSIONS

A. CONCLUSIONS

The results of this study indicated that the valance system had excellent summer performance characteristics. In the winter its operating characteristics were as satisfactory as those of a ceiling panel system. It produced warmer floor surface temperatures than did a baseboard system; however, it did not prevent the movement of cool air from the windows across the floor, nor did it keep the air 3 inches above the floor as warm as did the baseboard system. This was particularly true on the second level of the house.

The results also indicated that while the ratio of summer and winter design loads for each level of the house was reasonably constant, air movement up and down the staircase and heat transmission through the floor of the third level caused the actual summer load on the third level to be well in excess of the calculated cooling load while the actual winter load for this level was well below the calculated heating load. As a consequence, it was not possible to obtain good balance the year round when using the same units for both heating and cooling.

Other results of this investigation were as follows:

Winter Operation

1. Based on the measured load at 0 mile per hour wind, the measured load when using valance increased approximately 2 percent per 1 mile per hour increase in wind speed. When using baseboard, the rate of increase in load was about 3.7 percent per 1 mile per hour increase in wind speed.

2. Due to construction details, the infiltration rates for the I=B=R Hydronic Research House were high. By improved construction to reduce infiltration the effect of wind on room temperature balance would be diminished.

3. At an average wind speed of 7 miles per hour, the fuel consumption obtained with the valance system was about 16.5 percent higher than that obtained with the baseboard system.

4. When operating with baseboard and levels isolated there was excellent agreement between the ratios of the measured load for each level to the total measured load and the ratios of calculated load for each level to the total calculated load. This would seem to indicate that there was little or no transfer of load from one level to another when operating with baseboard and levels isolated.

5. The only shift in load when operating with baseboard was that due to movement of air up and down the staircase.

6. Air movement in the staircase when operating with baseboard increased the first and second level loads by 18 and 7 percent respectively and reduced the third level load by 25 percent.

7. Air movement in the staircase when operating with the valance system increased the first and second level loads by 33 and 9 percent respectively and reduced the third level load by 40 percent.

8. Shifts in level loads resulting from causes other than air movement in the staircase when operating with the valance system resulted in increases in the first and second level loads of 39 and 15 percent respectively.

9. The heat transfer rate through the floor from the first to the third levels reached a maximum value of 5.3 B.t.u.h. per square foot at an indoor-outdoor temperature difference of 50°F. This transfer represented a 17 percent increase in the first level load and a decrease of 15 percent in the third level load.

10. The valance system produced much higher ceiling air temperatures than did the baseboard system.

11. Inside wall surface temperatures when operating the valance system at design conditions were 8.5°F warmer than those obtained when operating the baseboard system.

12. Due to high ceiling temperatures, heat losses through the ceilings when operating with the valance system were 78 percent higher on the second level and 67 percent higher on the third level.
than those obtained when operating with the baseboard system. These gains represent an increase in the house heat loss of about 6.7 percent.

(13) The increases in the heat losses through the ceilings when operating with the valance system represented respective gains in the second and third level loads of 3.3 and 4.2 percent.

(14) The increase in inside surface temperature when using valance resulted in an increase in the heat loss through the walls. This increase applied to the first and second levels only since there was little or no operation of the units on the third level. The estimated increase in wall losses amounted to 3.9 and 1.4 percent of the first and second level loads respectively.

(15) If window losses were increased by the same percentage as the wall losses, the increased loss through glass would represent increases of 3.7 and 5.0 percent in the first and second level loads.

(16) The seasonal cost of heating the I=B=R Hydronic Research House with the baseboard system was $138.17 whereas for the valance system it was $169.62.

(17) In winter the higher ceiling, wall, and glass losses produced by the valance system were reflected as an increase of about $25.00 per season in the cost of the fuel used.

(18) The valance system produced cooler air temperatures below the 30-inch level and warmer air temperatures above the 30-inch level than did the baseboard system.

(19) The valance system produced warmer floor surface temperatures than did the baseboard system.

(20) For design weather, air temperature near the ceiling when using valance was about 130°F; however, this had no adverse effect on comfort conditions within the room as long as the bottom of the valance unit was at least 6 inches above head level.

(21) At an indoor-outdoor temperature difference of 80°F, the air temperature 3 inches above the floor was 66°F in the center of the living room and 63°F 3 feet from the windows when using the valance system. With baseboard, these temperatures were 70°F and 72°F respectively.

(22) At the center of the living room and at an indoor-outdoor temperature difference of 80°F, the floor surface temperature when using valance was 73°F while for baseboard it was only 65°F. On the other hand, 3 feet from the windows the floor surface temperature when using valance was 63°F whereas with baseboard it was 67°F.

(23) In general, all air movement across the floor of the living room when using valance was from the windows and outside walls toward the staircase. Velocities ranged from 18 to 45 feet per minute.

(24) When using baseboard, the air movement across the floor of the living room near the windows was in a direction toward the windows at a velocity of about 10 feet per minute. Elsewhere, air movement across the floor was too slow to measure.

(25) Vertical movement of cool air near the living room windows was downward at a velocity of about 35 feet per minute with both systems. This air movement extended all the way from the top of the window to the floor when using the valance system. With baseboard it extended from the top of the window to about mid-height of the window where it mixed with an upward current of warm air from the baseboard located at the bottom of the window.

(26) Indoor relative humidity was about 20 percent when the outdoor temperature was about 0°F. This was true no matter which heating system was in use.

(27) Room temperature balance was independent of the type of heating system. The balance was reasonably good for conditions of low wind.

(28) Wind had a pronounced effect on room temperature balance while the effect of sunshine was minor.

(29) Based on data obtained in the ASHVE laboratory, almost no occupant of the room would object to the combination of air temperature and movement measured in the living room of the research house at a point 3 inches above the floor and 3 feet away from the windows when the room was being heated by a baseboard system and the outdoor temperature was 0°F.

(30) More than 40 percent of the occupants could be expected to object to the combination of air temperature and movement occurring at this same location and at the same outdoor temperature when the room was being heated by a valance system.

(31) For conditions measured near the floor at the center of the living room, the expected incidence of complaints of cool ankles would be about 9 percent for the baseboard system and about 25 percent for the valance system.

Summer Operation

(1) The measured maximum loads for the first
and second levels were about 75 percent of the calculated loads while for the third level the measured load was about 10 percent in excess of the calculated load.

(2) The measured maximum loads for the different levels did not all occur at the same time, and as a result the maximum load for the house did not equal the sum of the maximum loads for the individual levels.

(3) The measured maximum load for the whole house was about 78 percent of the estimated load using I=B=R Guide C-30.

(4) In Guide C-30 no credit is given for shading resulting from roof overhang except on the south wall. Observations showed that extensive shading resulted from roof overhang on both southeast and southwest walls.

(5) If calculated loads were adjusted for shade conditions which actually existed the agreement between measured and calculated loads was even better than indicated in (3) above.

(6) Measured sensible heat ratios at the time of maximum load ranged from 0.75 to 0.81. The design procedure in Guide C-30 assumes a sensible heat ratio of 0.75.

(7) While there were differences between the measured and calculated heat gain values, these differences were not of alarming proportions, and in practically every case the measured loads were less than the calculated load.

(8) Isolating the house levels by plastic curtains at the top and bottom of the staircase did not affect the total load on the house.

(9) There was no significant difference in the rate of heat transmission through the ceiling when using valance or fan-coils.

(10) The increase in external load when using the fan-coil system resulted from high infiltration rates probably caused by the design of the fan-coil system.

(11) The entire system with the exception of the third level zone had ample capacity to take care of the total cooling loads on a design day. This zone started continuous operation from 1:00 p.m. to 9:00 p.m. at a maximum outdoor temperature of 92°F.

(12) Air movement in the staircase increased the cooling load on the third level by about 16 percent, and decreased the first level load by about 19 percent.

(13) Had there been no losses from ducts of the fan-coil system, no change in infiltration rate, no change in internal load and no change in unit power consumption of equipment common to both systems, the total daily power consumption for the fan-coil system would have exceeded that of the valance system by the power required by the fan motors in the fan-coil units and the additional power required for continuous operation of the pump. For design weather this would amount to 2.94 kilowatt-hours per day.

(14) After correcting the power consumption of the fan-coil system to the same internal loads as existed when testing the valance system, the power consumption of the fan-coil system on a design day exceeded that of the valance system by 31.5 kilowatt-hours per day.

(15) The difference between 2.9 and 31.5 or 28.6 kilowatt-hours per day represents an increase in power consumption which was probably the result of air leakage from the ducts of the fan-coil system and could have been eliminated had the ducts been air tight in construction or located entirely within the air conditioned space.

(16) A baseboard heating system should cost slightly less to install than a valance system, however, for year round operation the expected installation cost of a valance system is approximately 20 percent less than the installation cost of a combination baseboard heating and fan-coil cooling system.

(17) Labor represented about 15 percent of the total installation cost of either the heating or year round systems.

(18) The observed cost of operating the fan-coil system during the summer was $149.19. However, had the fan-coil system been designed so that there could be no duct losses to unconditioned space and had the internal heat gains been the same as when testing the valance system, the seasonal operating cost would have been reduced to $118.56.

(19) The seasonal cost of operating the valance system during the summer was $110.13.

(20) Both fan-coil and valance systems maintained constant average room air temperatures 30 inches above the floor at all maximum outdoor temperatures up to 95°F.

(21) Cyclic air temperature variations were 1½°F to 3°F. These were assumed to be representative of the operating differentials of the thermostats used during the tests.

(22) Solar effects were of sufficient magnitude
to cause an unbalance in the room air temperature of as much as 3°F.

(23) Both the fan-coil and valance systems produced about the same vertical air temperature gradient between the floor and a height of 60 inches above the floor (about 1°F). Above this level the air temperatures were higher when using the valance system.

(24) While the air temperatures near the ceiling were higher with the valance than with the fan-coil system, the difference was not enough to have an effect on one's feeling of comfort nor on heat gains through the ceilings.

(25) When using the fan-coil system, chilled water was continuously circulated through the coils and the fans were cycled by the thermostat. This method of operation produced an indoor humidity of approximately 50 percent irrespective of outdoor temperature.

(26) When using the valance system, it was necessary to control output by allowing the thermostat to control water circulation through the valance units. With this method of control the humidity indoors was about 50 percent during periods of heavy load, but tended to increase whenever the cooling load was reduced.

(27) Air movements in the rooms when using the valance system could not be detected without the use of instruments except directly under the valance units located where there were no windows and within 12 inches of the wall.
VIII. REFERENCES CITED


IX. APPENDICES

A. METHODS OF DETERMINING INFILTRATION RATES

Summer Operation

During the summer of 1961 direct measurements of the infiltration rate were made using the tracer gas technique. When using this procedure, a tracer gas (helium) was introduced into each room to a concentration of about 1 percent. During the period immediately following the introduction of the helium into the rooms the concentration of the helium was measured at regular intervals in order to obtain the rate of dilution. Assuming that the dilution of the helium resulted only from the replacement of helium-laden indoor air by helium-free outdoor air it can be shown that the rate of replacement (infiltration) in air changes per hour can be expressed by the equation:

\[ N = \frac{1}{\Delta t} \log_{e} \frac{C_1}{C_2} \]  

in which
- \( N \) = number of air changes per hour
- \( \Delta t \) = duration of test in hours
- \( C_1 \) = concentration of helium in the room air at the start of the test
- \( C_2 \) = concentration of helium in the room air at the end of the test

Since the apparatus required to determine infiltration rates by the use of a tracer gas was not available in the summers of 1959 and 1960, it was necessary to develop another method of estimating the infiltration rate in order to make it possible to compare infiltration rates for the house obtained when using each of the two types of cooling systems included in this study.

If all sources of moisture gains are known, the latent load may be used to determine the infiltration rate as follows. The amount of water vapor brought into the house with the infiltrating air may be expressed by the equation:

\[ W_o = (M_o - M_i) \times I \times d_i \times V \times \theta \]  

in which
- \( W_o \) = water vapor brought into the house by infiltrating air in pounds
- \( M_o \) = water vapor in outdoor air in pounds of water per pound of dry air
- \( M_i \) = water vapor in indoor air in pounds of water per pound of dry air
- \( I \) = infiltration rate in air changes per hour
- \( d_i \) = density of indoor air in pounds of dry air per cubic foot
- \( V \) = volume of house in cubic feet
- \( \theta \) = duration of test in hours

The total quantity of water added to the indoor air may be expressed by the equation:

\[ W = W_o + W_i \]  

in which
- \( W \) = total quantity of water vapor added to the indoor air in pounds
- \( W_i \) = water vapor added to indoor air from indoor sources in pounds
- \( W_o \) = same as in equation (2)

Since there was no cooking, washing, and drying of clothes or other such processes which would add water vapor to the air of the research house, the only significant source of water vapor within the house was the occupants. Their activity consisted mainly of desk work, and at such conditions the latent heat loss from an individual is approximately 150 B.t.u.h. or about 0.139 pounds of water vapor per hour.

By using test periods in which the indoor humidity remained constant, a condition of steady state was approximated so that

\[ W = c \]  

in which
- \( c \) = water removed from the air by the air conditioning system in pounds
- \( W \) = same as in equation (3)

Combining equations (2), (3), and (4) and solving for \( I \), the following equation is obtained.

\[ I = \frac{c - W_i}{(M_o - M_i) \times d_i \times V \times \theta} \]  

Table 26 shows the infiltration rates obtained simultaneously by equation (5) and by the tracer gas technique. In one case the difference in infiltration...
tion rate as obtained by the two methods was 0.17 air changes per hour. In all other cases the difference was no more than 0.07 air changes per hour. On the basis of this agreement it was concluded that summer infiltration rates for the Hydronic Research House could be determined with sufficient accuracy by the use of equation (5) and observed latent loads and the indoor and outdoor air conditions.

By the use of equation (5) the infiltration rates were calculated for those tests in Series G-59 for which the average outdoor temperature was about 85°F (design) and for all tests in Series C-60. The results of these calculations are shown in Table 26. Using the average values of infiltration rates for Series C-60 and G-59 as shown in Table 26, the difference is about 0.4 air changes per hour at an average outdoor temperature of 85°F.

The observed increase in infiltration could represent additional air leakage into the house due to different structural conditions or it could be increased in leakage induced by loss of conditioned air from the supply ducts located in the attic spaces. In the first case, the temperature of the exfiltrating air would be approximately 75°F. While in the second case, it would be about the same as the temperature of the air leaving the fan-coil units, or about 53°F. Obviously the temperature of the air leaving the house will have a direct effect on the sensible cooling load.

The external sensible load at an average outdoor temperature of 85°F was about 93,000 B.t.u. per day higher for Series G-59 than for Series C-60. Since there were no changes in house construction which would affect heat transfer rates through walls, floors, ceilings, or glass, this increase in sensible load must have resulted from the increase in infiltration rate noted in the preceding paragraphs.

At any instant the quantity of air entering the house from the outdoors (infiltration) must be equal to the quantity of air escaping to the outdoors (exfiltration). In analyzing the effects of air changes on cooling loads it is more convenient to consider exfiltrating air.

The sensible cooling load resulting from exfiltrating air is expressed by the equation:

\[ H_s = I \times V \times p \times C \times (t_o - t) \quad (6) \]

where

- \( H_s \) = sensible heat gain resulting from air leakage
- \( I \) = air leakage from house in air changes per hour
- \( V \) = volume of conditioned space = 13,630 cubic feet
- \( p \) = density of air in rooms = 0.0735 pound per cubic foot of dry air
- \( C \) = specific heat of air = 0.24 B.t.u. per pound dry air
- \( t_o \) = temperature of outdoor air, Fahrenheit (assumed to be 85°F)
- \( t \) = temperature of air exfiltrating from house (assumed to be 75°F if air escaped through cracks in the building construction and 53°F if exfiltrating air was from supply ducts of air conditioning system).
Using equation (6) it is found that if it is assumed the 0.4 air change per hour increase in infiltration resulted in the loss of an equal amount of air at room temperature the increase in sensible cooling load would be about 23,100 B.t.u. per day. On the other hand, if the increase in infiltration was the result of loss of conditioned air from the supply ducts of the air conditioning system the increase in sensible cooling load would be approximately 74,000 B.t.u. per day. Even this value is about 19,000 B.t.u. per day (790 B.t.u.h.) short of the measured difference in sensible cooling load of 93,000 B.t.u. per day.

While the difference between the calculated and the measured increase in sensible cooling loads appears to be large, it is not unreasonable when one considers the approximations that had to be made. The variations in the measured infiltration rates for both test series were large and the assumed average values of indoor, outdoor, and conditioned air temperatures were all subject to some error. In addition to this there would be some radiation and convection losses from the supply ducts to the air in the attic spaces even though these ducts were insulated.

No great degree of accuracy can be claimed for the preceding analysis; however, the evidence indicates that the exfiltration resulting from the increased infiltration rate during Series G-59 occurred from the supply ducts of the air conditioning system. This leakage is not unreasonable as it represents about 11 percent of the total quantity of air circulated through these ducts. While it is impossible to prove from the data collected that had there been no leakage of air from the supply ducts of the air conditioning system the infiltration rates for Series G-59 and C-60 would have been about the same, it is reasonable to make this assumption because of the capacity of the fans to create much larger pressure differentials between the inside of the duct and the outdoors than would normally be produced by wind and differences between indoor and outdoor temperatures. In fact, the leakage of air from the supply ducts would have the same effect as exhausting a like amount of air from the house by means of a kitchen ventilating fan.

B. HEAT FLOW THROUGH THIRD LEVEL FLOOR

The variation in heat flow through the third level floor with respect to indoor-outdoor temperature difference can be substantiated if the radiant heat transfer between the third level floor and ceiling and convective heat transfer from the third level floor are investigated over the range of indoor-outdoor temperature differences encountered.

In general, the operation of the third level began at an indoor-outdoor temperature difference of approximately 30° F., and as this temperature difference increased, the operating time along with the average ceiling temperature increased. While the ceiling temperature was increasing with increasing indoor-outdoor temperature difference, the third level floor surface temperature was also increasing because of the increasing temperature of the first level ceiling below it. As shown in Figure 22 the floor surface minus ceiling surface temperature difference of bedroom Number 1 reached a maximum value at an average indoor-outdoor temperature difference of 52° F. Thus, the radiant heat transfer from floor to ceiling could be expected to be greatest at this temperature difference.

Assuming a unit area, the radiant heat transfer between the floor and ceiling can be given by the equation

\[ q_{12} = \sigma (T_1^4 - T_2^4) f_{12} \]

where \( \sigma = \text{Stefan - Boltzmann Constant} = 0.1713 \times 10^{-8} \text{ B.t.u.h.}/(\text{sq ft} \cdot \text{R})^4 \)

\( R = \text{degree rankine} = F + 460 \)

\( T_1 = \text{floor temperature in R} \)

\( T_2 = \text{ceiling temperature in R} \)

\( f_{12} = \text{view factor for gray enclosures}^{(18)} \)

\[ f_{12} = \frac{1}{\left( \frac{1}{\Sigma_1} - 1 \right) + \frac{A_1}{A_2} \left( \frac{1}{\Sigma_2} - 1 \right) + \frac{1}{F_{12}}} \]

and for our case assuming \( A_1 = A_2 \)
\[ f_{12} = \frac{F_{12}}{F_{12} \left( \frac{1}{\Sigma_1} + \frac{1}{\Sigma_2} - 2 \right) + 1} \]

\[ F_{12} = \text{view factor for opposed parallel squares} \text{ for ratio smaller side distance between} = 1.1 \]
\[ \Sigma_1, \Sigma_2 = \text{emissivity of the ceiling and floor respectively} \]

If it is assumed that \( \Sigma_1 = \Sigma_2 = 0.9 \), and the walls are re-radiating, \( f_{12} \) can be found to be 0.498. Thus
\[ q_{1\rightarrow 2} = 0.498 \times 0.1713 \times 10^{-8} (T_1^4 - T_2^4) \]

Since the difference between \( T_1 \) and \( T_2 \) is small, little error will result in the equation if either \( T_1 \) or \( T_2 \) is considered to be constant while the other assumes a value different by the amount indicated in Figure 22. Assuming the floor to be at a constant temperature of 75° F. and assigning values to the ceiling temperature as indicated by Figure 22 the value obtained for the radiant heat transfer given in Figure 23 was derived.

Before a comparison can be made of the measured heat flow through the floor, the conductive and convective component of heat transfer must be considered. In Figure 24 the difference between the floor surface temperature and the air temperature 3 inches above the floor is given for the range of temperatures encountered.

The coefficient of heat transfer for heat loss from the floor by conduction and convection can be given as \(^{(20)}\)
\[ h_m = 0.38 (\Delta t_s)^{0.25} \]
where \( h_m = \text{coefficient in B.t.u.h./sq ft \ (F)} \)
\[ \Delta t_s = \text{temperature difference across the film in F} \]

By using this formula with \( \Delta t_s \) equal to \( \Delta t \) given in Figure 24 the conductive and convective component of heat transfer given in Figure 23 was derived.

In Figure 23, curve 3 represents the total calculated heat transfer rate through the floor. This value was obtained by summing curves 1 and 2. The measured heat flow rate through the floor is also included in this figure. By comparing the total calculated with the total measured heat flow rate it can be seen that the nature of the two curves is very similar. In fact, along the rising portions of the curves the agreement is very good. The lack of good agreement along the decreasing part of the curves may be due to the lack of data at large indoor-outdoor temperature differences. In any event, the nature of the curve given in Figure 14 is substantiated by this analysis, and the agreement between the calculated and measured heat flow rate through the floor is good considering all the simplifying assumptions made.
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