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NATIONAL BUREAU OF STANDARDS REPORT

4697

EXPERIMENTAL RADIANT HEATING AND COOLING SYSTEM;
REFLECTION POINT, CINCINNATI, OHIO

by

R. S. Dill
P. R. Achenbach
O. N. McDorman

Report to

Federal Housing Administration
Washington 25, D. C.

NBS

U. S. DEPARTMENT OF COMMERCE
NATIONAL BUREAU OF STANDARDS

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Abstract

The performance of the heating and air conditioning system in a house of novel design at "Reflection Point," in Cincinnati, Ohio, was observed by members of the Bureau staff at the request of the Federal Housing Administration. A special electric radiant heating system in the same house was observed in 1953, and is covered in National Bureau of Standards Report No. 2929. During the tests of 1953, heat was supplied by means of electric resistance elements in "coves" around the walls of the rooms near the ceiling. Reflective wallpaper was used on the ceiling and walls, and reflective drapes were provided for the windows to conserve heat and to direct heat radiated from the elements to the contents and occupants of the rooms. No other heat insulating means was provided in the house.

Prior to the observations covered herein, the roof was built up to include an air space and insulated with a reflective material, the heating system was changed to a gas-burning type, and a cooling system was installed. Aluminum tubes in the coves, in lieu of the electric heaters, are used to warm the house in winter and cool it in summer. For this purpose, an ethylene-glycol solution is circulated through the tubes and the solution is warmed in winter by a conventional boiler and cooled in summer by a refrigerating machine. The reflective surfaces on walls, ceiling and windows were retained. Ventilating air, supplied by a fan and duct system, is "pre-conditioned" before entering the house by a coil and electrostaticc cleaner. This coil utilized about 60 percent of the total cooling effect of the refrigerating machine under the summer test condition and about 40 percent of the total boiler heat output under the winter test condition. The effect of the coil was not totally available for either heating

or cooling the house because the ducts between the coil and the ceiling grille were uninsulated. The heat transmission coefficient for the cove coils, when used for cooling, was found to be about 0.65 Btu per (hr., sq. ft.) of coil surface for each degree of temperature difference between coil surface and room air during summer operation. The heat transmission coefficient of these coils during winter heating was nearly twice as great, probably because a significant amount of convection heating occurred. The tests of the heating system showed that a satisfactory temperature distribution was produced in the house at outdoor temperatures near the winter design temperature in Cincinnati although water temperatures in excess of 200 F would have been required for adequate heating at design temperatures. The tests of the cooling system revealed that the condensing unit was loaded in excess of the rated full load of the three-horsepower motor for outdoor conditions less severe than design summer conditions in Cincinnati although the motor was not dangerously overheated. The amount of cooling surface provided and the method of heat transfer incorporated in the design did not permit the condensing unit to produce a ton of refrigeration per horsepower motor load as is conventionally claimed for air conditioning units.

The observations indicate that the arrangement functioned approximately in accord with the design concept in that heat exchange by radiation occurred between the coils in the coves and the floor and contents of the rooms, assisted by the reflective surfaces on the ceiling, walls, and over the windows. A small increase in heating and cooling capacity appears desirable and can be obtained by increasing the amount of heat transfer surface in the cove and preconditioning coil, or possibly by improving the cove coil arrangement to increase its heat transfer. Ultimate utilization of the design depends, of course, on desirability and economic factors.

1. INTRODUCTION

At the request of the Federal Housing Administration, the performance of the heating and air conditioning system was observed in a house of unusual design, at "Reflection Point," cincinnati, Ohio. The system in this house is unconventional in that aluminum tubes, in coves in the walls near the ceiling, are used for heating in winter and cooling in summer. Great reliance is placed on radiant heat transfer to or from the tubes in the design concept of the system. The tubes are blackened outside, and reflective wall and ceiling surfacing materials and reflective drapes over the windows are used to promote heat transfer by radiation between persons or contents of the house and the tubes. Ethylene-glycol solution, called "brine," is circulated through the tubes to serve as a heating medium in winter and a cooling medium in summer. Air for ventilation is drawn from outside the house, passes through an air cleaner and a tempering coil, and is delivered through the hall ceiling, which is a perforated type selected for the purpose. Data on the performance of the system were obtained under both winter and summer conditions. Results of previous work on this house, when it was equipped with an electric radiant heating system, are contained in NBS Report 2929, dated November 19, 1953.

2. DESCRIPTION OF THE HOUSE

The house is located on a bluff and overlooks the city of Cincinnati from the North. It is a one-story structure of about 2000 square feet floor area, with a flat roof, a nine-foot ceiling height, and with a concrete slab floor. The floor is on grade, except under the bedrooms and study which were built over a garage. Some dimensional data are given in table 1.

The coves containing the heating-and-cooling coils are gutter-like structures extending around the interior walls about one foot down from the ceiling. They support, partly enclose, and conceal the tubes or coils that serve for both heating and cooling the house. The surfaces facing the coils, a semi-cylinder of metal and a baffle, are polished, while the coils are blackened, to promote heat exchange with the rooms by radiation. The semi-cylinder or trough beneath the coils is insulated underneath and is concealed by a metal enclosure, of rectangular cross section, which can be seen in figure 3. The coils are composed of about 2300 feet of 3/4-inch O.D. aluminum tubing with an aggregate surface area of 451 square feet.

TABLE 1 - ROOM SIZES AND VOLUMES AT REFLECTION POINT

	Length of Cold Wall, Ft	Ceiling Height, Ft	Gross Cold Wall Area, Ft^2	Window Area, Ft^2	Door Area, Ft^2	Net Cold Wall Area, Ft^2
Living Room	39.6	9	356	100	0	256
Dining Room	27.8	9	250	78	0	172
Kitchen & Utility	29.0	8	232	68	18	146
Foyer	11.2	9	101	51	20	30
Interior Hall	5.5	7.9	43	8	20[a]	15
North Bedroom	26.2	9	236	58	0	178
South Bedroom	28.9	9	260	68	0	192
Study	14.6	9	131	53	0	78
Large Bath	7.0	8	56	6	0	50
Small Bath	0.0	10	21	3	0	18[c]
Whole House	189.8	-	1686	493	58	1135

	Other Heat Transmitting Partitions, Ft^2	Ceiling Area, Ft^2	Floor Area, Ft^2	Crack Length Ft	Room Volume, Ft^3
Living Room	0	377	377	0	3393
Dining Room	0	189	189	21	1701
Kitchen & Utility	0	254	254	51	2032
Foyer	0	176	176	19	1584
Interior Hall	113[a]	0[b]	233	37[a]	1841
North Bedroom	0	160	160	56	1440
South Bedroom	0	238	238	56	2142
Study	0	129	129	43	1161
Large Bath	49[a]	57	57	11	456
Small Bath	0	28	24	0	240
Whole House	162	1608	1837[d]	294	15990

a. Partitions adjacent to basement stairwell
b. Fresh air inlet
c. Walls for skylight
d. Total enclosed floor area for whole house was 1986 ft^2

Embossed aluminum foil paper, lacquered in various colors, was used for interior coverings on most of the wall and ceiling area. Previous measurements of the emissivities of samples of the various colors showed that the emissivity ranged from a low value of 0.26 for the bronze color to a high value of 0.40 for the red color with an average of 0.32 for eight samples of different papers.

Fabrics with moderate reflective properties were used for draperies over many of the large windows. The emissivities previously measured for these fabrics ranged from 0.58 to 0.69. Some of these draperies are visible in figure 3.

The floors are covered with aluminum foil with the reflective surface upward, a waffle-design foam rubber mat, and wall-to-wall nylon carpeting in the living room, dining room, den, and one bedroom. Cork floor covering is used for the portion of the floor over the basement and garage.

Forced ventilation with cleaned outdoor air is provided by means of a blower and an electrostatic precipitator to reduce random infiltration at doors and windows. A finned coil in the ventilation duct tempers the outdoor air in winter, and dehumidifies and cools it during the summer. The fresh air is introduced into the house through a perforated ceiling in the hall.

Large picture windows, in every room of the house, total about 1/3 of the gross exterior wall area. Most of the windows are of fixed plate glass with no provision for opening.

The heating system consists of a gas-fired boiler with a rated output of 166,000 Btu per hour, a $\frac{1}{2}$-horsepower circulating pump, a tempering coil in the inlet of the ventilation duct, and the series-parallel circuits in the ceiling coves. The tempering coil and cove coils are in parallel. There is a by-pass line between the inlet and outlet of the boiler and a three-way valve to control the mixing of heated boiler water and water by-passed around the boiler in response to an electronic control which senses the room temperature. An aquastat on the outlet piping of the boiler controls the operation of the gas burner.

The air conditioning system consists of a nominally rated 3-horsepower water chilling unit and circulating pump which forces water-glycol mixture through the same coil in the ventilation duct and the same cove coils that are employed

in the heating circuit. The coil in the ventilation duct is intended to do all of the dehumidification and a limited amount of sensible cooling while the cove coils are intended to do sensible cooling only. An evaporator pressure regulator is used to prevent excessively low temperatures in the water chiller. A cooling tower is employed for cooling the condenser water.

Inspection of the house construction, and published descriptions, indicate that the following materials were used:

Walls: Embossed aluminum foil wallpaper on 3/8-inch gypsum board on 2 x 4-inch studs with 3/8-inch plywood sheathing and 25/32-inch vertical redwood tongue-and-groove siding. There was no insulation in the stud spaces.

Roof: The original roof consisted of 7 x 12-inch wooden beams spaced eight feet on centers, carrying 1/4-inch plywood ceiling and 2 x 6-inch tongue-and-groove planking covered with six layers of roofing felt. The ceiling was covered with embossed aluminum foil paper. The beamed ceiling and embossed foil ceiling paper are visible in figure 4.

Prior to the observations reported herein, this original roof was "built-up" by laying joists on the top surface of the roof to support a layer of Infra 6S insulation, one-inch sheathing boards, and roofing felt. The joists were tapered to provide an air space of two inches at the center and ten inches at the edges, between the original roof and the reflective insulation. This air space was closed at the edges, but eight 6-inch gravity roof ventilators installed along the two higher edges permitted air exchange with the out-of-doors. These roof ventilators are shown in figure 5.

Floor: On grade, there was a layer of gravel, a concrete slab floor with aluminum foil covering, waffle-design sponge rubber mat, and nylon carpeting. Over the

basement and garage, a 5/16-inch layer of cork tile covers the concrete floor. Orlan tile is used in the kitchen, and quarry tile in the entry and hallway.

As shown in figure 1, there is a roof overhang to shade the south windows from the midday sun, and vertical extensions from the walls on the exterior of the south wall at two places shaded the south windows from the late afternoon sun. Vertically-slatted radiation shields were attached to the exterior of the east and west walls to decrease the solar absorption in the early morning and late afternoon. Those on the west wall are shown in figure 6.

3. TESTING EQUIPMENT

For measuring air temperatures, unshielded thermocouples with polished, soldered junctions were used. Such couples were installed at the 30-inch level at the middle of each room. In addition, an eight-inch black globe thermometer was used to evaluate the radiation-convection temperatures near the middle of the living room. Wall and ceiling surface temperatures were measured with 30-gage thermocouples inserted beneath the wall paper. Wall heat transfer was measured with three heat flow meters in the living room, two on the walls at the 60-inch level, and one at the middle of the ceiling, as shown in figure 3. Wet and dry bulb temperatures were measured with a sling psychrometer, and air velocities with a vane anemometer.

In addition, during the heating tests, the temperature of the ethylene-glycol solution was measured at the boiler inlet and outlet, at the tempering coil inlet and outlet, and the flow was measured by a water meter. Air temperature was measured at the inlet and outlet of the tempering coils and at the ceiling in the hall where the air entered the house. Gas consumption was measured with the house gas meter. Running time of the burner was observed and recorded in percent.

During the cooling tests the capacity of the water chiller was determined by measuring the rate of circulation of the glycol-water mixture with a water meter and its temperature change as it passed through the chiller. The enthalpy change of the ventilating air, as it passed through

the dehumidifying coil, was determined by measuring the entering wet bulb and dry bulb conditions, the change in dry bulb temperature, and the amount of condensate drained from the coil. The enthalpy of the ventilating air was also determined at the hall ceiling. Observations of the radiant energy emitted at various areas on the wall and ceiling were made with a radiometer. The power consumption of the condensing unit was observed with a watthour meter. The indoor and outdoor dry bulb and wet bulb temperatures were determined with a sling psychrometer.

4. SYSTEM PERFORMANCE; COOLING

The cooling equipment was observed in operation during three days in August 1954, and the cooling coil in the ventilating air duct was operated alone during one trial. This coil is the same that serves for tempering the ventilating air under winter conditions. In summer, its main function is considered to be dehumidification of the fresh or ventilating air drawn in from outside. In a second trial, cooling was effected by both the duct coil and the coils in the coves.

5. DUCT COIL; AIR CONDITIONING CAPACITY

During the first day of the tests, the dehumidifying coil in the ventilating duct was operated for about three hours, from 1:50 P. M. to 4:45 P. M. as the sole air conditioning means for the house. The outdoor conditions were, initial: 85.5 F, D. B.; 67 percent R. H., and final: 83 F, D. B.; 63 percent R. H. The sky was partly cloudy. Under this condition the coil reduced the inside temperature from 83.5 F to 81.7 F and the relative humidity from 70 percent to 58 percent in about three hours.

Performance data on the dehumidifying coil at 4:30 P.M. are summarized in table 2. The total cooling effect of the coil was about 17,000 Btu per hour, based on the coolant circulation rate and temperature change, or 18,600 Btu per hour, based on the air flow rate and enthalpy change. There is, of course, a discrepancy of about 10 percent between these figures, but inaccuracies of this order are expected in field tests of this kind. Corrected values of specific heat and density of the ethylene-glycol solution were used in the calculation.

TABLE 2

PERFORMANCE OF DEHUMIDIFYING COIL, 4:30 P.M.

Outdoor Conditions		
Dry Bulb Temp.	°F	83
Wet Bulb Temp.	°F	73
Relative Humidity	%	63
Enthalpy	Btu/lb	36.6
Sky		Partly cloudy
Indoor Conditions, Room Center 30-inch Level		
Living Room, Dry Bulb	°F	82.4
Living Room, 8-inch Globe	°F	82.6
Dining Room, Dry Bulb	°F	81.6
Kitchen, " "	°F	81.0
Foyer, " "	°F	82.1
Study, " "	°F	81.5
North Bedroom, Dry Bulb	°F	80.9
South Bedroom, " "	°F	82.1
Average, " "	°F	81.7
Living Room, R. H.	%	59
North Bedroom, R. H.	%	57
Coolant Circulation Rate	gph	285
Chiller Inlet Temp.	°F	46.2
Chiller Outlet Temp.	°F	37.9
Useful Cooling Capacity	Btu/hr	17,070
Power Used by Compressor	Watts	2,740
Ventilating Air Rate	cfm	244
Air Inlet D.B. Temp.	°F	84.8
D.B. Temp. at Dehumidifier Outlet	°F	49.6
D.B. Temp. at Hall Ceiling	°F	70.1
Moisture Removal Rate	lb/hr	9.27
Enthalpy Change in Vent. Air	Btu/hr	18,640
Sensible Heat Removal	Btu/hr	8,620
Latent Heat Removal	Btu/hr	10,020
Sensible Heat Gain Between Dehumidifier & Hall Ceiling	Btu/hr	5,140
Potential Sensible Cooling Capacity in Living Space	Btu/hr	2,910

The condensate collected from this duct coil was 9.27 pounds per hour, which accounts for about half of the total cooling effect of the coil. The resulting drop in absolute humidity was thus from 0.0152 to 0.0063 pounds of water vapor per pound of dry air. The average absolute humidity in the house was 0.0133 pounds of water per pound of air.

The air cooled by the duct coil warmed considerably in the duct and plenums on the way to the hall ceiling. The measured temperature in the coil discharge was about 50 F and at the hall ceiling it was about 70 F. Therefore the net cooling effect of the coil was somewhat less than 3000 Btu per hour under the existing condition.

The capacity of the refrigerating machine was greater than the coil could utilize under the existing condition. Evidence of this is the fact that the suction pressure was reduced considerably below the setting of the evaporator pressure regulator, which was 30 pounds per square inch.

6. PERFORMANCE OF COOLING SYSTEM; ALL COILS OPERATING

During the second day, the performance of the whole system, with both the duct coil and the cove coils in operation, was observed from 9 A. M. to 4 P. M. The observations from noon to 4 P. M., when the conditions were most severe, are reported in table 3.

In the forenoon, a dry bulb temperature of about 80 F was maintained, and the humidity was reduced from 72 to 61 percent in the house. A mangle and an electric oven were used part of the time.

In the afternoon, there were alternate periods of cloudiness and of bright sunshine. The outside temperature ranged from 84 to 88 F, and the relative humidity averaged about 71 percent. The difference between inside and outside temperatures was about 6 F. The indoor temperature increased about 1.2 F, while the relative humidity decreased about three percent in the four-hour period. The indoor dry bulb temperature averaged 79.5 F, and the relative humidity in the living room and one of the bedrooms averaged 60.5 percent and 58 percent, respectively.

TABLE 3

OBSERVED PERFORMANCE OF AIR CONDITIONING SYSTEM
ALL COILS OPERATING: 12 M. to 4:00 P.M.

Outdoor Conditions		
Dry Bulb Temp.	°F	85.4
Wet Bulb Temp.	°F	77.7
Relative Humidity	%	71
Enthalpy	Btu/lb	41.2
Sky		Partly Cloudy
Indoor Conditions, Room Center 30-inch Level		
Living Room, Dry Bulb	°F	79.4
Living Room, 8-in. Globe	°F	79.8
Dining Room, Dry Bulb	°F	79.3
Kitchen, " "	°F	80.0
Foyer, " "	°F	79.5
Study, " "	°F	79.0
North Bedroom, " "	°F	79.1
South Bedroom, " "	°F	80.2
Average, " "	°F	79.5
Living Room, R. H.	%	60.5
North Bedroom, R. H.	%	58.0
Coolant Circulation Rate	gph	313
Chiller Inlet Temp.	°F	52.1
Chiller Outlet Temp.	°F	40.6
Total Cooling Effect	Btu/hr	26,000
Power Used by Compressor	Watts	3,585
Ventilating Air Rate	cfm	244
Coolant Temp. at Dehumidifier Inlet	°F	40.4
Coolant Temp. at Dehumidifier Outlet	°F	52.1
Air Temp. at Dehumidifier Inlet	°F	85.4
Air Temp. at Dehumidifier Outlet	°F	56.2
Moisture Removal Rate of Dehumidifier	lb/hr	8.16

TABLE 3 (Cont'd)

Enthalpy Change in Vent. Air	Btu/lb of air	15.5
Enthalpy Change in Vent. Air	Btu/hr	16,140
Sensible Heat Removal in Vent. Air	Btu/hr	7,320
Latent Heat Removal in Vent. Air	Btu/hr	8,820
Sensible Heat Gain between Dehumidifier & Hall Ceiling	Btu/hr	3,460
Potential Sensible Cooling Capacity in Living Space (Vent. Air)	Btu/hr	2,330
Net Cooling from Cove Coils by Difference	Btu/hr	9,860
Coolant Temp. at Cove Coil Inlet	°F	41.6
Coolant Temp. at Cove Coil Outlet	°F	50.2
Temp. Diff. Room Air to Cove Coils	°F	33.6
Heat Transmission Factor of Cove Coils		
Btu/hr (°F) (linear ft)		0.13
Btu/hr (°F) (sq ft)		0.65
Ceiling Surface Temp. Room Center	°F	79.2
Wall Surface Temp. 60 in above Floor		
South Wall, Living Room	°F	81.8
West Wall, Living Room	°F	77.8

Table 3 shows that the brine chiller was producing a useful cooling effect of 26,000 Btu per hour under steady operation. The pressures on either side of the evaporator pressure regulator indicated that this valve was wide open, thus permitting the condensing unit to operate at maximum capacity for the existing conditions. The compressor motor power averaged 3,585 watts. This corresponds to about four horsepower at an estimated efficiency of 80 percent.

The condensate collected from the duct or dehumidifying coil and the inlet and outlet dry bulb temperatures of this coil indicated a total enthalpy change of 16,140 Btu per hour in the ventilating air. The average condensate rate was 8.16 pounds per hour corresponding to a latent heat removal of 8,820 Btu per hour, or 55 percent of the total cooling capacity of the coil. The ventilating air was warmed from 56 F to 70 F between the coil outlet and the perforated discharge panel in the hall ceiling. This corresponded to a sensible heat gain of 3,460 Btu per hour in the supply ducts and left a potential sensible cooling capacity in the ventilating air of 2,330 Btu per hour.

Subtracting the cooling capacity of the dehumidifying coil from the total cooling capacity of the chiller indicates that the cooling effect of the cove coils was probably about 9,860 Btu per hour. This capacity represents a heat transmission of 0.13 Btu per (hr., linear ft.) of coil or 0.65 Btu per (hr., sq. ft.) of coil surface for each degree F of temperature difference between brine and room air.

It should be noted in analyzing the performance of the duct coil that the dry bulb temperature observed at the coil outlet is not entirely consistent with the moisture content at the same station derived from the moisture content of the outdoor air and the condensate collected from the coil. This introduces an uncertainty of about ten percent in the capacity of this coil, but does not affect significantly the major conclusions derived from the tests.

7. COOLING LOAD COMPUTATIONS

Computed cooling loads for the house, summarized in table 4, are based on heat transmission data contained in the ASHAE "Guide." Table 4 indicates a cooling load of 25,680 Btu per hour for the conditions that prevailed indoors

TABLE 4

COMPUTED COOLING LOAD, BTU/HR AT
OBSERVED CONDITIONS AND DESIGN CONDITIONS

		Observed Conditions	Design Conditions
Outdoor D. B. Temp.	°F	87	95
Outdoor W. B. Temp.	°F	78.5	78
Wind Velocity	mph	Calm	6
Indoor D. B. Temp.	°F	80.1	80
Indoor W. B. Temp.	°F	69.4	67
Indoor R.H.	%	58.3	51
Cooling Loads			
Outside Walls	Btu/hr	3880	7970
Windows & Doors	Btu/hr	2790	7890
Inside Partitions	Btu/hr	0	600
Ceiling	Btu/hr	5890	7050
Floor over Basement	Btu/hr	1050	2250
Slab on Ground	Btu/hr	0	0
Infiltration	Btu/hr	9390	10860
Sub Total	Btu/hr	23000	36620
People, 4 (light activity)	Btu/hr	2000	2000
Electrical (estimated 200w.)	Btu/hr	680	680
Total Load	Btu/hr	25680	39300

Computed Transmission Coefficients, U,
Btu/hr (sq ft)(°F) for Air Conditioning

Wind Velocity	mph	0	6
Outside Walls	U=	0.246	0.264
Ceiling and Roof	U=	0.064	0.066
Windows and Doors	U=	0.75	0.99
Floor over Basement	U=	0.33	0.33
Inside Partitions	U=	0.37	0.37

and outdoors at 4 P. M. on the second day of the tests. It also indicates that the average cooling effect of 26,000 Btu per hour observed for the brine chiller should have been just adequate for the existing weather on the day of the tests. This equality of cooling effect and cooling load was corroborated, in general, by the nearly steady state conditions maintained indoors from noon to 4 P. M. while the condensing unit operated continuously. The computed cooling load for outdoor design conditions in Cincinnati was 39,300 Btu per hour, assuming indoor conditions of 80 F dry bulb and 67 F wet bulb, and outside conditions of 95 F dry bulb and 78 F wet bulb.

The equivalent temperature method described in the "Guide" was used for these computations. The shading of the exterior walls and windows by the roof overhangs, the wall extensions on the south side, and the slatted radiation shields on east and west sides were taken into account. The air space between the built-up roof and the original roof was considered to be a dead air space since the arrangement of the ventilation hoods did not assure positive air circulation through this space. The access of the chilled ventilating air to the underside of the hall and kitchen roof was taken into account in computing the roof load of these areas. The reflective surfaces of the paper on the walls and ceiling were taken into account in computing the heat transmission coefficients listed at the bottom of table 4. An occupancy of four people was assumed for both computations since four people were present in the house during the test of the air conditioning system. The air infiltration rate was assumed equal to the observed air delivery of the ventilating blower.

The computations in table 4 indicate that the outside walls, windows, ceiling, floor, and infiltration represented 15.1, 10.8, 22.9, 4.1, and 36.5 percent of the total cooling load, respectively, at the observed conditions. Observations of the heat exchange at the wall and ceiling surfaces made with heat flow meters and with a radiometer do not, in some cases, agree with the values obtained when the recommended ASHAE computation methods were applied, as will be seen later.

Table 5 summarizes the radiant heat transfer from selected areas of the walls, ceiling, and drapes, and through the undraped windows in the living room and dining

TABLE 5

RADIANT HEAT TRANSFER AS MEASURED WITH A RADIOMETER, 1:30 P.M., August 28

Source	Heat Transfer Rate Btu/hr (sq ft)
Heat Flow Meter, South Wall, Black	3.7
Heat Flow Meter, South Wall, Reflective	1.9
South Wall, After Washing	0.7
Ceiling, Reflective	0.4
West Wall, Reflective	0.4
Heat Flow Meter, West Wall, Reflective	0.4
West Wall Above Window, Reflective	2.0
West Drape	12.2
South Drape	12.2
North Drape	13.0
South Window Above Horizon	161.
North Window Below Horizon	31.
North Window Above Horizon	144.
Indoor-Outdoor Temp. Difference at 1:30 P. M., °F	(5.0)

room of the house at 1:30 P. M. on the second day of the tests as measured by a radiometer. These values do not include the convection component of heat transfer.

Figure 7 shows the heat transfer through the south and west walls and the ceiling of the living room as measured by heat flow meters attached to the inner wall surface at the 60-inch level. The surfaces of the meters exposed to the room were covered with the same reflective paper used on the walls and ceilings of the house except for one of the two meters on the south wall, which was left uncovered. Figure 7 shows that heat was being lost through all of these surfaces at 9 A. M. when the observations began. Heat gain commenced through the south wall soon after 9 A. M., through the west wall at noon, and through the ceiling between 1 and 2 P. M.

A few comparisons can be made between the data in figure 7 and table 5. Figure 7 shows that the total transfer between the south wall and room at 1:30 P. M. was 6.1 Btu per (hr., sq. ft.) for the black heat flow meter and 4.5 Btu per (hr., sq. ft.) for the reflective heat flow meter. Table 5 indicates that the radiant components were 3.7 and 1.9 Btu per (hr., sq. ft.), respectively, for these two surfaces, leaving convection components of 2.4 and 2.6 Btu per (hr., sq. ft.) in the two cases.

These comparisons of the heat transfer through the black and reflective heat flow meters do not mean that the heat transfer for the entire wall could be changed in the same ratio by changing the emissivity of the inner surface, but only that portion of the wall inside of the stud spaces. The stud space acted as a plenum chamber behind both heat flow meters because the heat flow meters were of small area. If large sections of the wall had been covered with wallpapers of different emissivities, the air temperatures in the stud spaces would be different behind the two coverings. The temperature in the stud space back of the reflective wallpaper would probably be higher than that behind the non-reflective wallpaper, and the difference in heat transfer through the entire wall would be less than the data in figure 7 and table 5 indicate. Table 5 shows that washing the reflective paper on the south wall significantly reduced the radiation heat transfer. The owner advised that this paper had been on the wall for several years.

Figure 7 shows that the total heat transfer of the west wall at 1:30 P. M. was 0.9 Btu per (hr., sq. ft.), whereas the radiometer measurement of the radiant component was 0.4 Btu per (hr., sq. ft.) at the same time and place. The radiant heat transfer of the west wall above the window where it was not externally shaded was five times as high as at the heat flow meter where the wall was shaded by the slatted radiation shield outdoors. The heat flow meter on the ceiling showed no heat transfer at 1:30 P. M., whereas the radiometer indicated a heat gain of 0.4 Btu per (hr., sq. ft.).

Some comparisons are also possible between the heat flow meter data in figure 7, and the computed heat transfer through the walls and ceiling of the living room, based on data in the ASHAE "Guide." Figure 7 shows that the heat transfer through the south and west walls of the living room at 4 P. M. was 5.2 and 2.1 Btu per (hr., sq. ft.), respectively. The "Guide" data indicate that the heat flow through a shaded wall at 4 P. M. would be about 1.48 Btu per (hr., sq. ft.), based on the computed heat transmission coefficients in table 4. This comparison indicates that the south and west walls were not the equivalent of a shaded wall. Even though they received no direct insolation, they were receiving some diffuse radiation from the sky and from the patio in the case of the south wall. Figure 7 shows a heat transfer through the ceiling at 4 P. M. of 0.75 Btu per (hr., sq. ft.), whereas the "Guide" data indicate a value of 2.94 Btu per (hr., sq. ft.), indicating that the ceiling was a better heat barrier for downward heat flow than the computed heat transmission coefficient indicates.

Table 5 shows that the drapes on the north, south, and west windows were radiating from 12 to 13 Btu per (hr., sq. ft.) to the rooms even though shaded from direct sunlight and, further, that sky radiation ranging from 144 to 161 Btu per (hr., sq. ft.) entered the north and south windows when they were uncovered. A completely shaded window would have transmitted only about 5 Btu per (hr., sq. ft.) for the existing temperature difference and air motion. The "Guide" does not list heat transmission data for windows shaded on the outside and covered with reflective draperies on the inside, but suggests that a shaded window should be treated as though its heat gain resulted primarily from the temperature difference existing on the two sides. The radiometer data in table 5 indicate that the heat transfer through the shaded, draped windows was more than twice that based on the air temperature difference alone.

The data obtained with the heat flow meters and radiometer, though grossly incomplete, indicate that the cooling load computations summarized in table 4 do not accurately represent the actual conditions that prevailed, and that it was perhaps fortuitous that the total computed cooling load was equal to the observed cooling capacity of the air conditioning system for the period under consideration. The foregoing analysis of the heat flow meter and radiometer data indicates that the actual heat flow through the south and west walls and through the draped windows was probably greater than the computed values shown in table 4, and that the actual heat flow through the ceiling was probably less than is shown by table 4.

The capacity of the refrigerating machine was shown, by the best indications, to be sufficient to maintain almost half the design inside-outside temperature difference usually assumed for Cincinnati when the ventilating air load was about 12 percent below that assumed for design conditions, and when the sun did not shine continuously. More system capacity would be desirable under the existing conditions and it may be considered inadequate for more severe conditions, including the design condition for Cincinnati.

8. SURFACE TEMPERATURES

Table 3 shows the average temperatures observed on the wall and ceiling surfaces in the living room during the period from noon to 4 P. M. The temperatures were measured with 30-gage thermocouples inserted just underneath the wallpaper, through openings made with a darning needle. The average surface temperature at the center of the ceiling in the living room was 79.2 F, differing 0.2 F from the average dry bulb temperature of the air at the 30-inch level in the room during the same period. The average surface temperatures of the south and west walls of the living room at the station 60 inches above the floor were 81.8 F and 77.8 F, respectively. The latter figure is probably inaccurate since it would indicate that the wall was gaining heat from both indoors and outdoors, whereas figure 7 indicates an outward heat flow during the afternoon. The temperature indicated by the globe thermometer in the living room was only 0.4 F higher than the air temperature measured with small thermocouple wire in the same room. This indicates that the globe was receiving a relatively small amount of radiated heat from the surrounding surfaces.

Figure 8 shows the ceiling surface temperatures measured in the living room from the center to the edge of the cooling cove by inserting a thermocouple underneath the reflective paper. These data reveal ceiling surface temperatures ranging from 79 F to 81 F at distances greater than 15 inches from the edge of the cove. There was a decrease in surface temperature near the cove which was caused, presumably, by the greater radiation to the cove coils. When the radiometer was sighted on the surface of the cove coils, it indicated a heat transfer to the coil surface of 20.7 Btu per (hr., sq. ft.). This agrees approximately with the value of 21.9 Btu per (hr., sq. ft.) that can be derived from the data in table 3 for total heat transfer to the cove coils and their total surface area.

9. SUMMER COMFORT CONDITIONS

The average indoor conditions maintained during the second day of tests were 79.5 F dry bulb and 59 percent relative humidity, corresponding to an effective temperature of about 74.8 F in the ASHAE effective temperature chart. It is recognized that the ASHAE chart does not necessarily reflect comfort conditions accurately when there is a greater or lesser than normal amount of radiation present in the environment. The reflective papers on walls and ceiling affect the radiant heat exchange between the occupants and these surfaces. The two observers engaged in these tests were not at optimum comfort. They were engaged in intermittent light activity while making observations; they noted slight perspiration on their foreheads and were aware of the low air motion in the living space.

10. SYSTEM PERFORMANCE; HEATING

The operation of the heating system was observed throughout one night while the outdoor temperature decreased gradually from 32 F at 9 P. M. to 9 F at 9 A. M. This decrease in temperature was accompanied by snowfall and a wind ranging in velocity from 0 to 9 miles per hour.

As the weather became colder during the night, the operating time of the gas burner increased, the by-pass valve around the boiler remained in a closed position more of the time, and finally, the aquastat setting of the boiler

had to be raised. Between 1 A. M. and 2 A. M., with an outdoor temperature of 26.5 F, the gas burner operated 35 percent of the time, the boiler outlet temperature averaged 163 F, and the by-pass valve cycled between 5/8 closed and fully closed. Between 8 A. M. and 9 A. M., when the outdoor temperature averaged 10 F, the gas burner operated 63.7 percent of the time, the boiler outlet temperature averaged 188 F, and the by-pass was closed at all times. During this latter period the aquastat setting of the boiler had to be raised to 200 F to provide satisfactory room temperatures. The differential of the aquastat was about 25 F. The average gas consumption rate between 8 A. M. and 9 A. M. was 134 cubic feet per hour, corresponding to a boiler output of 98,500 Btu per hour if the boiler were assumed to be 70 percent efficient.

The performance of the heating system is summarized in table 6 for the hour between 8 A. M. and 9 A. M. Air was passed through the preheater at a rate of 305 cubic feet per minute and was heated approximately 121 F above outdoor temperature. This preheater is the same coil that served to dehumidify the fresh air brought in for ventilation under summer conditions. The above figures indicate a heat transfer of 39,860 Btu per hour in the preheater coil. This value of heat transfer may be in error because the outlet air temperature increased and decreased as the boiler outlet temperature increased and decreased, so an accurate average could not readily be determined. The air decreased in temperature nearly 30 F between the preheater outlet and the discharge grille in the hall ceiling. This temperature drop corresponded to a heat loss of 9,350 Btu per hour in the ducts and hall plenum. Thus, the ventilating air at the hall ceiling could provide approximately 11,800 Btu per hour to offset transmission heat loss. By difference, the cove coils were computed to be supplying heat at a rate of about 53,600 Btu per hour after correcting the difference between boiler output and preheater output for the computed piping loss.

11. TEMPERATURE DISTRIBUTION

The air temperatures at the 30-inch level at the center of the several rooms averaged 66.7 F, whereas the black globe thermometer at the center of the living room indicated a temperature 4.7 F above air temperature at the same

TABLE 6

PERFORMANCE OF THE HEATING SYSTEM,
8:00 to 9:00 A.M., February 11

Outdoor Temp.	°F	10.0
Indoor Temp., Room Center, 30-in. Level		
Living Room, 8-in. Globe	°F	72.5
Living Room, Air	°F	67.8
Dining Room, Air	°F	66.6
Kitchen, Air	°F	66.0
Foyer, Air	°F	67.6
North Bedroom, Air	°F	65.8
Study, Air	°F	66.8
South Bedroom, Air	°F	66.2
Average, Air 30-in. Level	°F	66.7
Glycol-Water Circulation Rate	gal/hr	236.
Average Boiler Inlet Temp.	°F	151.6
Average Boiler Outlet Temp.	°F	186.3
Instantaneous Gas Consumption Rate	cu ft/hr	210.3
Percent Running Time		63.7
Average Gas Consumption Rate	cu ft/hr	134.0
Computed Heat Output Rate (70% eff.)	Btu/hr	98,490.
Ventilating Air Rate Through Preheater	cfm	305.
Air Temp. at Preheater Outlet	°F	131.1
Air Temp. at Hall Ceiling	°F	102.7
Heat Transfer in Preheater	Btu/hr	39,860.
Heat Loss Between Preheater & Hall Ceiling	Btu/hr	9,350.
Sensible Heat Available at Hall Ceiling for House Heating	Btu/hr	11,800.
Net Heat from Cove Coils, by Difference	Btu/hr	58,600.*
Ceiling Surface Temp., Center of Living Room	°F	81.5
Living Room Temp., 60 in. Above Floor		
South Wall Surface, Reflective	°F	63.7
South Wall Surface, Black	°F	66.6
Air 2 in. from South Wall	°F	68.4

*This value includes piping loss from the boiler, which was computed to be approximately 5,000 Btu/hr.

location. These temperatures are summarized in table 6. The maximum variation in air temperature at the 30-inch level for the seven room centers was 2 F. The difference between globe temperature and air temperature in the living room indicates a considerable radiant flux in that room and shows that the mean radiant temperature would be several degrees above the globe temperature. The observers were comfortable during most of the test, but some feeling of coolness was experienced about 6 A. M. This could have been the result of long hours of duty and lowered metabolism.

The surface temperature of a heat flow meter on the south wall covered with reflective wallpaper was 63.7 F, whereas the temperature of a similar heat flow meter with a black surface, also mounted 60 inches above the floor, was 66.6 F. The ceiling surface temperature at the center of the living room was 81.5 F during the coldest period of the test. The ceiling surface temperature at mid-length of the living room, from the edge of the south heating cove to the edge of the north heating cove, is shown graphically in Figure 9 for an outdoor temperature of 15 F. The rise in temperature near the cove edges indicates greater heat transfer from the coves in this area by radiation or convection, or both. The heat loss through the ceiling would also be greater near the cove edges. The arithmetical average ceiling temperature for the traverse shown in figure 9 was 87.7 F. The temperatures on the aluminum reflector shield in the cove ranged from 111 F to 112.5 F.

The air temperatures in a vertical line from floor to ceiling at the center of the living room, and at a station 36 inches from the south picture window in the living room, are plotted in figures 10 and 11, respectively. Figure 10 shows that the rug surface in the living room was 3 to 4 F warmer than the air one inch above the surface as a result of the radiant energy absorbed by the rug. The minimum air temperature was observed in the zone from 6 to 24 inches above the floor, and the floor surface temperature was approximately equal to the air temperature 60 inches above the floor. The air temperature increased rapidly in the zone from 80 to about 100 inches above the floor and dropped sharply in the last inch below the ceiling surface. The ceiling surface was 8 or 9 F cooler than the air one inch below the surface. The shape of the air temperature curve near the ceiling indicates considerable heat transfer from the coves by convection. If only radiant heat were striking the ceiling surface and being reflected, the air temperature

beneath the ceiling surface would be cooler than the surface. The coves were 12 inches high and open at the front face, so convection currents could be set up in the coves that would direct warm air outward over the ceiling surface as cooler air entered at the lower edge of the cove opening.

The presence of convection from the coves is also indicated by a comparison of the maximum air temperature shown in figure 11 with that for the corresponding outdoor temperature in figure 10. This comparison shows that the air temperature one inch below the ceiling was 95 F at a distance 24 inches from the cove edge, whereas the temperature at the same level was only 86 F at the room center. This indicates an outward flow of warm air near the ceiling from the cove toward the center of the room. This movement of warm air from the cove must contribute to the difference in ceiling surface temperature between perimeter and room center as shown in figure 9. These convection currents increase the amount of heat that can be transferred from the cove coils and do not adversely affect the comfort in the living zone because the stratum of warm air is above the 80-inch level.

Figure 10 shows that the minimum air temperature near the floor was lowered about 2 F and the maximum air temperature near the ceiling raised about 7 F, whereas the rug surface temperature was affected only one-half degree when the outdoor temperature decreased from 30 F to 10 F. A comparison of figures 10 and 11 shows little difference in air temperatures below the 80-inch level between the room center and a station 36 inches from the south picture window, which was only 15 inches in front of the bookcases beneath the window.

12. HEATING LOAD

The outdoor weather conditions and the computed heating loads of the various components of the house for the observed conditions are summarized in table 7. The heat transmission coefficients used for these computations are listed at the bottom of the table. Table 7 indicates that the heat loss of the house was 86,900 Btu per hour exclusive of duct losses and piping losses for an indoor-outdoor temperature difference of 57 F. Of this total, 20 percent is attributed to the exterior walls, 35.5 percent to the windows and doors,

TABLE 7

COMPUTED HEATING LOAD AT OBSERVED CONDITIONS, BTU/HR, 8:00 to 9:00 A.M., February 11

Outdoor Temp.	°F	10.0
Wind Velocity	mph	7.4
Indoor-Outdoor Temp. Diff.	°F	56.9

<u>Heating Loads</u>

Outside Walls	Btu/hr	17,400
Windows and Doors	Btu/hr	30,800
Inside Partitions	Btu/hr	900
Ceiling	Btu/hr	11,600
Floor over Basement	Btu/hr	4,160
Slab on Ground	Btu/hr	3,290
Infiltration	Btu/hr	18,740
Total Heating Load of Living Space	Btu/hr	86,890
Loss from Ventilating Air Ducts	Btu/hr	9,350
Total System Load (excluding Piping Loss)	Btu/hr	96,240

Computed Transmission Coefficients, Btu/hr (sq ft) (°F) for Heating

Ceiling and Roof	0.092
Outside Walls	0.270
Windows and Doors	1.02
Floors over Basement	0.33
Inside Partitions	0.37

13.4 percent to the ceiling, 8.6 percent to the floors, and 21.6 percent to infiltration of outdoor air. The air infiltration was assumed to be equal to the forced ventilation provided by the ventilation blower. It represented about 1.15 air changes per hour for the entire house.

The heat loss from the ventilating ducts was computed to be 9,350 Btu per hour, based on the observed temperature drop between the preheater outlet and the hall ceiling grille. Thus the total system load, exclusive of loss from the hot water piping, was computed to be 96,240 Btu per hour. If 5,000 Btu per hour were added for piping loss, the total boiler load would be 101,240 Btu per hour compared with a boiler output of 98,490 Btu per hour derived from the observed gas consumption and an assumed boiler efficiency of 70 percent.

Some inaccuracy in field observations is usually expected and the heat flow meter readings taken on the south and west walls and ceiling of the living room are not concordant with the computed heat transmission coefficients in table 7 in some respects. The heat flow indicated by the heat flow meters is plotted against time for the entire test in figure 12. Figure 12 shows an average heat flow for the south and west walls, using the value obtained with reflective wallpaper for the south wall, of about 10 Btu per (hr., sq. ft.) for the time between 8 A. M. and 9 A. M. The computed value, using the U-factor in table 7 and the observed temperature difference, would be 15.4 Btu per (hr., sq. ft.). The corresponding observed value for the ceiling averaged about 4.5 Btu per (hr., sq. ft.) in figure 12 for the same time interval, whereas the computed value was 7.1 Btu per (hr., sq. ft.). Probably the heat capacity of the walls and ceiling construction caused the instantaneous values observed with the heat flow meters to be too low, but the lag of the exterior walls was only one to two hours so heat lag would hardly account for the disparity between observed and computed values in the case of the walls.

Thus the heat flow meter data indicate that table 7 provides only an approximate accounting for the heat loss in the structure and they also create some uncertainties about the accuracy of the computed heat transmission coefficients. It has been shown by test and reported in the literature that the heat transfer through a cavity wall is not uniform from floor to ceiling.

Although the heat flow meters were near mid-height of the wall, they might not have been indicating an average heat flow.

The two heat flow meters on the south wall, with black and reflective surfaces, indicated a considerable difference in heat flow as shown in figure 12. As was the case for the air conditioning test, the stud space of the wall provided a plenum of equal temperature behind both meters. Thus the difference in indicated heat flow shown in figure 12 is the effect of placing a reflectively-covered and a black heat flow meter on a sheet of 3/8-inch gypsum board over this constant temperature plenum, and does not provide an adequate evaluation of the effect of using reflective wallpaper on the heat transfer through the whole wall. In this case, the reflectively-covered meter provides the better value of heat flow because the black meter was only about 16 square inches in area on an otherwise reflective wall.

13. DISCUSSION AND CONCLUSIONS

Accurate data are difficult to obtain under field conditions and some discrepancies are apparent in that contained in this report. However, conclusions concerning this type of house and its heating and cooling system are supported by the data as follows:

The results show that considerable heat can be transferred by radiation from cove coils to the floor and contents of a house when the wall and ceiling surfaces and window drapes are made of reflective materials in accord with the design concept. Under the observed winter condition, the ceiling was warmer than the floor so that heat reached the floor in part by radiation from the ceiling. However, the ceiling was not as warm nor as emissive as that ordinarily required by a ceiling radiant heating system, and the difference must be attributed to radiation by the coils. Use of more highly reflective material on ceiling, walls, and drapes would be expected to enhance the radiant transfer by the coils and lessen that from the ceiling.

Economy of heating equal to that expected of other typical systems probably can be attained with systems of the type investigated. A criterion for house heat requirements

in use by FHA is 55 Btu per hour for each square foot of floor area at design conditions. The observed total heat loss of the house investigated was about 100,000 Btu per hour for an indoor-outdoor temperature difference of about 57 F. This amounts to 50 Btu per hour for each square foot of floor area, since the house had a floor area of about 2000 square feet. At an indoor-outdoor difference of 70 F, the adjusted heat loss is

$(70/57)50 = 61$ Btu per hour for each square foot.

It is plausible to credit the reflective interior surfaces of the house with saving since the black bulb thermometer indicated a temperature nearly five degrees higher than the air temperature under the most severe test condition. Assuming an indoor-outdoor temperature of 65, instead of 70 F, the adjusted heat loss becomes

$(65/57)50 = 57$ Btu per hour for each square foot.

Both these adjusted heat losses exceed the criterion, but better economy can be attained by such changes as reducing the ventilating air, use of more highly reflective surfaces on walls and ceiling, insulation in exposed walls, insulation of the fresh air duct and plenum, etc.

The observed heat transfer coefficient for the cove coils when used for cooling was about 0.65 Btu per (hr., sq. ft.) for each degree F of temperature difference between coil surface and room air. It would be more logical to base the heat transmission coefficient for summer cooling on the difference between the mean radiant temperature of the room and the coil surface temperature since there was practically no convection during summer operation. However, the difference between mean radiant temperature and air temperature was so small during summer operation as to make this refinement unimportant.

The heat transfer coefficient of the cove coils for heating was about twice as great as for cooling. This difference is attributable to the more pronounced convection currents around these coils when they were used for heating. This convection is regarded as responsible for the relatively high air temperatures near the ceiling. When the coils were cold, the air probably stratified around them in the gutter-like structure used to conceal

them from below. This suggests that this structure might be slotted or perforated in future designs to permit passage of air and thus improve heat transfer by convection and increase their capacity for cooling.

The coil in the intake duct, used for tempering the outside air drawn in for ventilation in winter and for cooling and dehumidifying that air in summer, accounted for about 40 percent of the heating load in winter and about 60 percent of the cooling load in summer under the test conditions. This can be construed to mean that the ventilating rate is greater than necessary and that it can be reduced in future designs. It is possible to arrange for some recirculation through this coil to effect some economy of operation which is particularly desirable under the summer condition.

The cooling capacity required, 26,000 Btu per hour, agreed well with the computed cooling load, 25,680 Btu per hour, under the observed condition. This is regarded as fortuitous because the heat flow meters did not agree in all cases with heat transfer rates computed by conventional methods. Indications are that actual heat gains through the south and west walls are greater than computed and that the gain through the roof is less than that computed. This emphasized the complexity of the heat gain problem and the need for further research concerning it.

The observers reported a feeling of coolness during part of the most severe heating test and perspiration on the forehead during the most severe cooling test. Neither of these tests was conducted under as severe a condition as the design condition for Cincinnati. The tests showed that a satisfactory temperature distribution was produced in the house at an outdoor temperature of 10 F although it was indicated that boiler water temperatures in excess of 200 F would have been required to adequately warm the house at the design winter temperature of 0 F in Cincinnati.

Summer operation of the cooling system at conditions less severe than design summer conditions showed that the proportions of the components were such that the three

horsepower condensing unit was loaded to approximately four horsepower while producing a cooling capacity of about 26,000 Btu per hour. This performance was associated with an average dry bulb temperature of 79.5 F and an average relative humidity of about 59 percent in the house during the period from noon to 4 P. M. on the day of the test.

These data on the operation of the system and the comfort sensations cited above suggest that a greater heat transfer surface area in the cove coils would have been advantageous in both heating and cooling. However, the same result might possibly be obtained with the existing coil area by an arrangement that would increase the radiant and convection heat transfer rates.

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