UNIVERSITY OF ILLINOIS
BULLETIN

Vol. 41 January 4, 1944 No. 20

ENGINEERING EXPERIMENT STATION
BULLETIN SERIES No. 349

PERFORMANCE OF A HOT-WATER HEATING SYSTEM IN THE I=B=R RESEARCH HOME AT THE UNIVERSITY OF ILLINOIS

A REPORT OF AN INVESTIGATION

CONDUCTED BY

THE ENGINEERING EXPERIMENT STATION
UNIVERSITY OF ILLINOIS

IN COOPERATION WITH

THE INSTITUTE OF BOILER AND RADIATOR
MANUFACTURERS

BY

ALONZO P. KRATZ
WARREN S. HARRIS
MAURICE K. FAHNESTOCK
AND
ROSS J. MARTIN

PRICE: SEVENTY-FIVE CENTS

PUBLISHED BY THE UNIVERSITY OF ILLINOIS
URBANA

[Issued weekly. Entered as second-class matter at the post office at Urbana, Illinois, under the Act of August 24, 1912. Office of Publication, 358 Administration Building, Urbana, Illinois. Acceptance for mailing at the special rate of postage provided for in Section 1103, Act of October 3, 1917, authorized July 31, 1918.]

THE Engineering Experiment Station was established by act of the Board of Trustees of the University of Illinois on December 8, 1903. It is the purpose of the Station to conduct investigations and make studies of importance to the engineering, manufacturing, railway, mining, and other industrial interests of the State.

The management of the Engineering Experiment Station is vested in an Executive Staff composed of the Director and his Assistant, the Heads of the several Departments in the College of Engineering, and the Professor of Chemical Engineering. This Staff is responsible for the establishment of general policies governing the work of the Station, including the approval of material for publication. All members of the teaching staff of the College are encouraged to engage in scientific research, either directly or in coöperation with the Research Corps, composed of full-time research assistants, research graduate assistants, and special investigators.

To render the results of its scientific investigations available to the public, the Engineering Experiment Station publishes and distributes a series of bulletins. Occasionally it publishes circulars of timely interest, presenting information of importance, compiled from various sources which may not readily be accessible to the clientele of the Station, and reprints of articles appearing in the technical press written by members of the staff and others.

The volume and number at the top of the front cover page are merely arbitrary numbers and refer to the general publications of the University. *Above the title on the cover* is given the number of the Engineering Experiment Station bulletin, circular, or reprint which should be used in referring to these publications.

For copies of publications or for other information address

THE ENGINEERING EXPERIMENT STATION,
UNIVERSITY OF ILLINOIS,
URBANA, ILLINOIS

UNIVERSITY OF ILLINOIS
ENGINEERING EXPERIMENT STATION
BULLETIN SERIES No. 349

PERFORMANCE OF A HOT-WATER HEATING SYSTEM IN THE I=B=R RESEARCH HOME AT THE UNIVERSITY OF ILLINOIS

A REPORT OF AN INVESTIGATION

CONDUCTED BY

THE ENGINEERING EXPERIMENT STATION
UNIVERSITY OF ILLINOIS

IN COOPERATION WITH

THE INSTITUTE OF BOILER AND RADIATOR MANUFACTURERS

BY

ALONZO P. KRATZ
RESEARCH PROFESSOR OF MECHANICAL ENGINEERING

WARREN S. HARRIS
SPECIAL RESEARCH ASSOCIATE IN MECHANICAL ENGINEERING

MAURICE K. FAHNESTOCK
RESEARCH PROFESSOR OF MECHANICAL ENGINEERING

AND

ROSS J. MARTIN
SPECIAL RESEARCH ASSISTANT IN MECHANICAL ENGINEERING

PUBLISHED BY THE UNIVERSITY OF ILLINOIS

PRICE: SEVENTY-FIVE CENTS

4000—11-43—26897

UNIVERSITY
OF ILLINOIS
:: PRESS ::

ABSTRACT

This bulletin is the first to be published under a cooperative agreement between the Institute of Boiler and Radiator Manufacturers and the University of Illinois, which was formally approved January 2, 1940. A description of the I=B=R Research Home, which was built in Urbana, Illinois, furnished and completely equipped specifically for research work in steam and hot water heating, is given, as well as a discussion of the results of tests made during the 1940-41 and 1941-42 heating seasons. In all tests an oil-fired, cast-iron boiler supplying hot water to a one-pipe, forced-circulation, hot-water heating system with small-tube cast-iron radiators was used in the Research Home. The principal objectives of the tests were to compare the operating characteristics of the heating system when operated (1) with a control system permitting the boiler to be used to heat domestic hot water as well as to supply heat to rooms, and (2) with a control system permitting the boiler to supply heat to the rooms only. Fuel and power savings and room temperature conditions resulting from operating with reduced room air temperatures at night were determined for both methods of control. Incidental studies were made of the heat transmission through walls of the house and radiator recesses, and the effect of radiator location on the temperature distribution and air movement within the rooms. All windows remained closed throughout all tests.

Operating the system with a flow-control valve and low-limit aquastat maintaining a minimum boiler water temperature of 165 deg. F. resulted in an average increase in fuel consumption of approximately 0.5 gal. per day as compared with operation without a flow-control valve and low-limit aquastat. Fuel and power savings resulting from operation with reduced room air temperatures at night ranged from zero up to about 10 per cent, depending upon the method of control, and, in average winter weather, the indoor air cooled not more than 6 deg. F. during the 7.5 hours that the thermostat was set back at night.

In zero deg. F. weather the average difference between the temperature of the air 3 in. below the ceiling and that 3 in. above the floor was of the order of 4.5 deg. F. for all methods of operation. A difference of only 0.5 deg. F. between the maximum and minimum air temperatures at the 30-in. level was obtained during a cycle of operation.

When the radiators were located along exposed walls there was no noticeable movement of cool air across the floor in any of the rooms. With the radiators located along inside walls definite cool drafts were observed, and the floor to ceiling air temperature difference was increased by 3 to 5 deg. F.

In zero weather the ratio of the rate of heat loss through an uninsulated brick veneer recess to the heat loss rate through the fully-insulated brick veneer wall was about 16:1. One reflective surface located back of the radiator, or 1 in. of rigid insulation, reduced this ratio to about 9:1. The use of one reflective surface plus 1 in. of rigid insulation reduced the rate of heat loss until it was practically equal to that obtained back of a free standing radiator.

Less than 50 per cent of the heat required to heat the rooms in the Research Home was supplied by the radiators. The remainder of the heat was supplied by other sources, such as the chimney and uninsulated heating pipes, with a small amount supplied by lights and occupants. About 80 per cent of the heat contained in the fuel burned was made available as useful heat in the house.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

PERFORMANCE OF A HOT-WATER HEATING SYSTEM IN THE I=B=R RESEARCH HOME AT THE UNIVERSITY OF ILLINOIS

I. Introduction

1. *Preliminary Statement.*—This bulletin is the first to be published under a cooperative agreement between the Institute of Boiler and Radiator Manufacturers and the University of Illinois. This agreement was formally approved January 2, 1940. Early in June, 1940, ground was broken for the I=B=R Research Home, and the house was completed and all furniture and instruments installed by the latter part of the year. About the first of January, 1941, a test program was started in which the operation of a one-pipe, forced-circulation, hot-water heating system using recessed radiators and an oil-fired boiler was studied.

Under the terms of the agreement, the Institute is represented by an Advisory Research Committee consisting of five members. Since the inception of the program the following members of the Institute have served on this Committee:

J. P. Magos,* Crane Co., Chicago, Illinois

L. N. Hunter,† National Radiator Company, Johnstown, Pennsylvania

J. F. McIntire, United States Radiator Corporation, Detroit, Michigan

H. F. Randolph, International Heater Company, Utica, New York

S. K. Smith, H. B. Smith Company, Westfield, Massachusetts.

It is the function of this committee to propose such problems for investigation as are of the greatest interest to the manufacturers and installers of steam and hot-water heating equipment. Of these problems, the Engineering Experiment Station staff selects for study those which can best be investigated with the facilities and equipment available at the University. The Institute provides funds for defraying a major part of the expense of this research work.

2. *Acknowledgments.*—This investigation has been carried on as a part of the work of the Engineering Experiment Station of the University of Illinois and as a project of the Department of Mechanical Engineering. The investigation was conducted under the general ad-

*Chairman of Committee, 1941 and 1942.
†Chairman of Committee, 1940.

ministrative direction of DEAN M. L. ENGER, Director of the Engineering Experiment Station, and of PROFESSOR O. A. LEUTWILER, Head of the Department of Mechanical Engineering. Acknowledgment is hereby made to the various manufacturers who cooperated by furnishing materials and equipment used in the investigation.

3. *Objects of Investigation.*—In the broadest sense the object of these investigations is to study, under actual service conditions, the different types of steam and hot-water heating systems, and various changes therein, from the standpoint of the performance of the systems and their component parts and the comfort conditions produced within the building. Supplementary studies of the factors related to the performance of the structure are also to be undertaken.

Tests conducted during the 1940-41 and 1941-42 heating seasons are discussed in this bulletin. The principal objective of the tests made during the 1940-41 heating season was to compare the operating characteristics of an oil-fired, one-pipe, forced-circulation, hot-water heating system operated (1) with a control system which would permit the boiler to be used to heat domestic hot water as well as to supply heat to the rooms, and (2) with a control system such that the boiler could supply heat to the rooms only.

The principal object of the tests made during the 1941-42 heating season was to determine the fuel and power savings and room temperature conditions resulting from operating with reduced room air temperatures at night.

Incidental studies were made of the heat transmission through walls and radiator recesses, and the effect of radiator location on the temperature distribution and air movement within the room.

II. DESCRIPTION OF EQUIPMENT

4. *Research Home.*—The I=B=R Research Home, shown in Fig. 1, is a two-story building, typical of the small, well built American Home. The construction is brick veneer on frame, and all of the outside walls and the second story ceiling are insulated with mineral wool bats 3⅝ in. thick. A vapor barrier offers resistance to the flow of water vapor from the heated spaces into the insulation or to the outdoors. This vapor barrier consists of a glossy-surfaced, asphalt-impregnated sheathing paper weighing 50 lb. per roll of 500 sq. ft., applied with overlapping joints, and as continuous as possible, to the inside of the studs of all exterior walls, between the house and the garage, and on

FIG. 1. I=B=R RESEARCH HOME

the bottom of the second story ceiling joists. The exterior walls are constructed as follows: one course of face brick, an air space, building paper, ship lap sheathing on 2-in. by 4-in. studs, insulation in the form of 3⅝-in. mineral wool bats, vapor barrier, rock lath and plaster with trowled finish. The calculated coefficient of heat transmission, U, for this wall section is 0.074 B.t.u. per sq. ft. per hr. per deg. F. temperature difference. All windows and outside doors are weather stripped, and during the past winter two storm doors were used. Following standard practice the heat losses from the house were calculated using the methods and coefficients published in the 1940 ASHVE Guide. As recommended in the Guide, the infiltration loss was based on a wind velocity of 15 mi. per hr., and the lineal feet of crack existing around doors and windows, rather than on any assumed number of air changes per hour. The total calculated heat loss under design conditions for the house excluding the basement was 43 370 B.t.u. per hour with temperatures of −10 deg. F. outdoors and 70 deg. F. indoors.

The basement windows and door were almost entirely below grade and were protected by areaways. Under these conditions exposure to a wind velocity of 15 mi. per hr. seemed excessive, hence in deter-

TABLE 1
DATA ON RESEARCH HOME AND HEATING SYSTEM

Room	Dimensions	Heated Space cu. ft.	Calculated Heat Loss, B.t.u. per hr. (without storm sash)	Installed Radiation		
				Number of Radiators	Number of Sections	Square* Feet Equivalent Direct Radiation (E.D.R.)
First Floor						
Living Room	24 ft.-0 in. x 13 ft.-4 in.	2641	5 749	2	22†	35.2†
Dining Room	13 ft.-1 in. x 11 ft.-3 in.	1183	8 742	2	34	54.4
Kitchen	10 ft.-5 in. x 11 ft.-3 in.	799	3 199	1	14	22.4
Lavatory	7 ft.-0 in. x 2 ft.-8 in.	152	1 484	1	8	12.8
Vestibule	7 ft.-5 in. x 5 ft.-4 in.	284	4 848	1	16	25.6
Vestibule Closet	....	54	.....	..	..	....
Total (1st Floor)	....	5113	24 022	7	94	150.4
Second Floor						
N.E. Bedroom	10 ft.-7 in. x 9 ft.-9 in.	800	4 393	1	16	25.6
N.W. Bedroom	10 ft.-6 in. x 13 ft.-4 in.	1148	4 944	1	18	28.8
S.W. Bedroom	13 ft.-0 in. x 11 ft.-4 in.	1108	5 250	1	20†	32.0†
Bath	6 ft.-6 in. x 7 ft.-6 in.	374	2 606	1	10	16.0
Stair Landing and Hall	....	505	2 155	1	8	12.8
Closets	....	345	.....	..	..	....
Total (2nd Floor)	....	4280	19 348	5	72	115.2
Total (1st and 2nd Floors)		9393	43 370	12	166	265.6
Basement	24 ft.-0 in. x 25 ft.-2 in.	5084	32 810	1‡	50	80.0

Net I=B=R Boiler Rating = 63 000 B.t.u. per hr. Oil nozzle 1.20 gal. per hr.

*Based on a rating of 1.6 sq. ft. per section of free standing radiation. For recessed radiators this rating was reduced by 12½ per cent to compensate for recessing. Mean water temperature 195 deg. F., drop of 10 deg. F., heat emission rate 200 B.t.u. per sq. ft. per hr.

†During summer of 1941, 2 sections (3.2 sq. ft.) were added to radiation in living room and 5 sections (8.0 sq. ft.) were removed from the radiation in the S.W. bedroom.

‡This radiator was not used during either the 1940-41 or the 1941-42 heating season.

mining the outside film coefficient and the infiltration, a wind velocity of 5 mi. per hr. was assumed. Based on these assumptions, with an indoor temperature of 70 deg. F. and a ground temperature of 50 deg. F., the calculated heat loss from the basement alone was 32 810.

The total floor area of the heated space is 1174 sq. ft., and the volume is 9393 cu. ft. Table 1 gives a summary of the calculated heat losses, the volumes and the installed radiation for each room in the house.

In order to make the house easily adaptable to future test work it was necessary to provide some special structural features, particularly in the details of the plaster construction. Removable panels of sheet rock instead of plaster were used on sections of the walls over all hot water pipes in order to insure accessibility. However, care has been taken to insure that, so far as is possible, the house remains

FIG. 2. VIEW OF LIVING ROOM IN I=B=R RESEARCH HOME

a typical home, complete with furniture, rugs, window shades, drapes, and curtains. Figure 2 shows the character of the furnishings. The daily occupants of the home consisted of members of the research staff only. No cooking or other domestic processes requiring the application of heat were carried on.

5. *Radiator Recesses.*—The radiator recesses are so constructed that radiators with recess panels having inlet and outlet grilles can be used. In order to accommodate radiators of different sizes, provision has also been made for changing the size of the recesses with minimum marring of the interior finish. Since it was impossible to use full thickness mineral wool insulation in the walls back of the radiators, in each recess the surface of the sheathing facing the radiator was covered with reflective insulation. With the exception of one, all recesses were further insulated by applying pieces of rigid insulation the same size as the recesses on the outside of the sheathing. One-inch rigid insulation was used on ten recesses, half-inch on one, and none on another. Details of all the types of recesses are shown in Fig. 3. During the

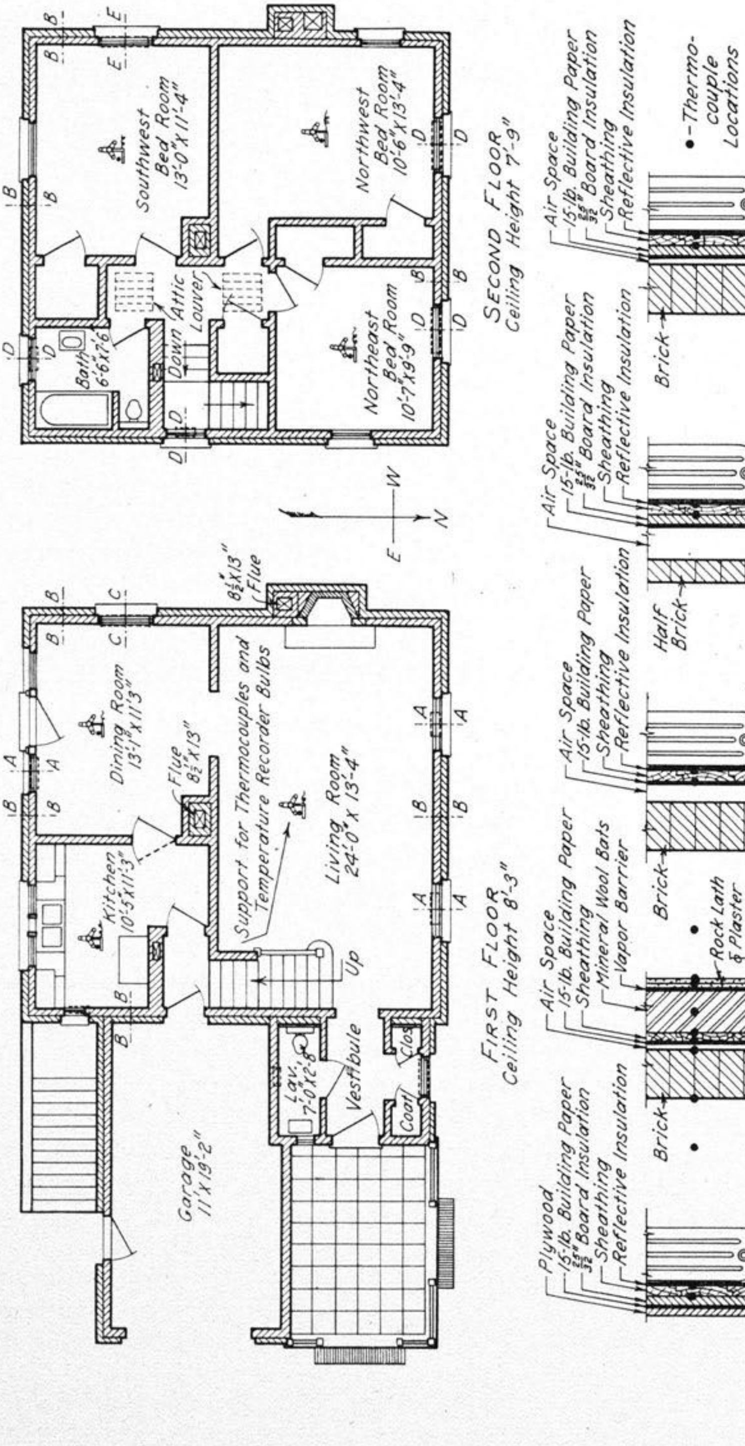

FIG. 3. FIRST AND SECOND STORY PLANS SHOWING RADIATOR LOCATIONS, TEMPERATURE MEASURING STATIONS, AND DETAILS OF RADIATOR RECESSES

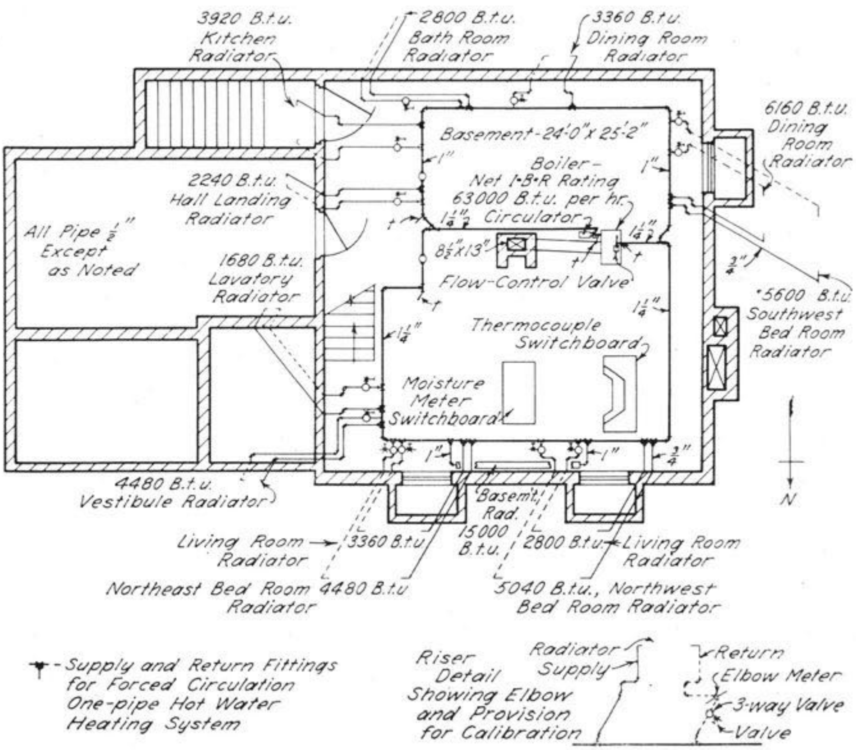

Fig. 4. Basement Plan With Locations of Heating and Testing Equipment

1941-42 heating season the reflective insulation was removed from the recesses in the dining room, living room, and S.W. bedroom in order that the effectiveness of this type of insulation when located back of radiators might be evaluated.

6. *Heating System.*—An oil-fired, cast-iron boiler used in connection with a one-pipe, forced-circulation, hot-water heating system with small-tube type cast-iron radiators was used in the Research Home during the heating seasons of 1940-41 and 1941-42. Some of the factors influencing the selection of this system for the initial installation were the flexibility, the relatively low cost, and the increasing popularity of this type of system.

Small-tube type, 19-in., 4-tube, cast-iron radiators set in open recesses below windows were used to transfer the heat from the water to the room air. In each case there was about 4-in. clearance between the ends of the radiator and the sides of the recess, 2¼-in. clearance

TABLE 2
PIPE AND FITTINGS USED IN INITIAL HEATING INSTALLATION

Section	Pipe Size in.	Length of Pipe ft.	Elbows		Tees		One Pipe Supply Fittings Number	One Pipe Return Fittings Number
			Size in.	Number	Size in.	Number		
Basement Main								
North Loop	1½	47.7	1¼	6	1¼ x 1¼ x ¾	1		
					1½ x 1¼ x ½	4	7	2
South Loop	1¼	1.7	1¼-45 deg.	3				
	1	31.2	1-45 deg.	1	1 x 1 x ¾	1		
			1	3	1 x 1 x ½	5	6	
Supply Trunk	1¼	6.3	1¼	2	1¼ x 1¼ x 1	1		
Return Trunk	1¼	15.8	1¼	2	1¼ x 1¼ x 1	1		
	2½	0.6			2½ x 2½ x 1¼	1		
Total		103.3		17		14	13	2
Risers								
S.W. Bedroom	¾	30.5	¾	11			1	
N.W. Bedroom	¾	33.0	¾	9			1	
N.E. Bedroom	½	33.9	½	11			1	1
Bath	½	44.8	½	15			1	
Stairway	½	39.2	½	13			1	
Kitchen	½	19.6	½	11			1	
Dining Room, West	½	11.8	½	11			1	
Dining Room, South	½	9.0	½	11			1	
Living Room, West	½	14.0	½	9			1	
Living Room, East	½	13.8	½	9			1	
Vestibule	½	27.7	½	14			1	
Lavatory	½	30.7	½	12			1	
Basement	1	14.0	1	4			1	1
Total		280.0		140			13	2

Main and Riser Totals

2½-in. Pipe	0.6 ft.
1¼-in. Pipe	71.5 ft.
1-in. Pipe	45.2 ft.
¾-in. Pipe	63.5 ft.
½-in. Pipe	235.6 ft.
Total	416.4 ft.

Specialities

1¼-in. Angle Flow Control Valve	1
1¼-in. Circulator	1

between the top of the radiator and the top of the recess, and ¼-in. between the back of the radiator and the wall of the recess. The front of the radiator was approximately flush with the wall of the room. The radiators were connected to two supply loops located in the basement as shown in Fig. 4. A typical radiator installation is shown in Fig. 2, and the location of all radiators in each room of the house is shown by the broken line rectangles in Figs. 3 and 4.

The amount of radiation installed in each room was based on a heat emission rate of 200 B.t.u. per sq. ft. of equivalent direct radiation with a mean water temperature of 195 deg. F. and a 10-deg. F. drop through the radiator, and was just sufficient to offset the calculated heat loss at an outdoor temperature of −10 deg. F. with the circulator operating continuously. In selecting the radiators the rating of 1.6 sq. ft. E.D.R. per section of free standing radiation, as listed in the manufacturers catalog, was reduced by 12.5 per cent in order to compensate for the anticipated reduction in heat output of the radiators when installed in recesses. The radiator installed in the basement was arbitrarily made large enough to offset only one-half of the calculated heat loss on the assumption that some heat would be available from the boiler jacket, smoke pipe and basement piping. The size, amount, and arrangement of the basement mains and piping used in connection with the radiators are shown in Table 2 and Fig. 4. These sizes were selected in accordance with existing standard practice.

Soon after putting the plant into operation it was discovered that the basement radiator and 12 sections of radiation in the dining room were not necessary. Accordingly the basement radiator and the south radiator in the dining room were blanked off and were never used in any of the subsequent tests. During the 1940-41 heating season it was observed that the S.W. bedroom was overheating and that the living room had a tendency to underheat. To correct this condition 5 sections were removed from the S.W. bedroom radiator, and 2 sections were added to one of the radiators in the living room during the summer of 1941. The heating system used in the 1941-42 season was identical with that used during the previous season except for these changes in the radiators in the S.W. bedroom and living room.

The cast-iron, hot-water boiler was composed of three 4-in. sections insulated on the front, back, sides, and top by a mineral wool blanket 1 in. in thickness, and was completely enclosed with an enameled sheet-metal jacket. All cracks around the base of the boiler and between sections were sealed with asbestos cement to prevent leakage of air into, or products of combustion out of, the boiler. The boiler had a net I=B=R rating of 63 000 or a gross rating of 95 000 B.t.u. per hr. when fired at the rate of 1.0 gal. of oil per hr. The oil burner was of the pressure atomizing, conversion type, and the refractory combustion chamber was 11 in. wide by 12 in. long by 12½ in. high, inside dimensions. The system was equipped with a 1¼-in. circulator and an angle flow-control valve. The latter prevented circulation of water through the system during off-periods of the circulator.

The operation of the burner and circulator was regulated by means of a conventional set of controls consisting of a room thermostat, located in the living room, an electric clock with a self-contained time switch, a low-limit aquastat, a high-limit aquastat, a stack switch, and a relay. The stack switch was connected in the burner circuit, and served as a safety control in case of failure of the ignition or oil supply. The room thermostat was of the heat-anticipating type, and provision was made through the action of the clock and time switch to reduce the temperature setting automatically at night and restore it to normal in the morning. The direct-acting low-limit aquastat was of the immersion type, and was located in the rear section of the boiler approximately 26 in. above the bottom of the water leg. This aquastat started the burner whenever the temperature of the water in the boiler reached a prescribed minimum. The high-limit aquastat, attached to the surface of the supply trunk immediately above the top of the boiler, served as an additional safety control to prevent overheating the water in the boiler. The particular arrangement of the controls used with each of the series of tests, and the resulting sequence of operations, are described in Chapter III.

7. *Testing Apparatus.*—While the house was being constructed, approximately 100 copper-constantan thermocouples made of No. 22 B and S gauge wire were permanently installed in the walls and ceilings in order to measure temperatures at important points in the structure under various operating conditions. At each of eight locations, one on each exposure of each story, nine thermocouples were installed to provide for establishing complete temperature gradients through the walls. Other thermocouples were installed in each recess to measure the temperature of the inside and outside surfaces of the sheathing directly back of the radiator. Another group of approximately 50 thermocouples provided for the measurement of air temperatures at various levels in the center of each room, in the attic, and in the basement. A fourth group made it possible to study the performance of the component parts of the heating system. Provision was made for measuring the temperature of the water entering and leaving each radiator, as well as the temperature of the water entering and leaving the boiler.

A central switchboard was located in the basement, and all of the thermocouples were connected to selector switches on this board. In this way the electromotive force of each thermocouple could be read quickly on a precision potentiometer used in connection with a highly sensitive galvanometer. A ten-point recording potentiometer used in connection with an auxiliary switchboard made it possible to obtain either instantaneous readings or continuous printed records of the electromotive force given by the thermocouples in any selected group.

Provision for measuring the rate of flow of water in the mains, and through each radiator in the heating system, was made by installing sixteen elbow meters to be used in connection with a sensitive differential pressure gauge, or manometer.* Use is made of the principle that a fluid flowing through an elbow creates a difference in pressure between one point on the inside radius and another point on the outside radius of the elbow. This difference in pressure is a measure of the rate of flow. This type of meter introduces no additional resistance to the flow of water.

Recording thermometers were used to make continuous records of the air temperatures in each of the six rooms, the outdoor air, and the flue gas temperatures. Continuous records of the CO_2 in the flue gases were obtained by means of a thermal-conductivity type recorder calibrated against an Orsat apparatus. The moisture content of the air was measured by means of four humidity indicators, one recording hygrometer, and one wet- and dry-bulb recorder which were checked periodically with an aspirated psychrometer. Drafts were measured by means of either inclined manometers or a recording draft gauge as the occasion demanded. The electrical input to circulator and burner motors was measured by means of integrating watt-hour meters reading directly to 10 watt-hours. Self starting electric clocks were wired into the burner and circulator motor circuits in such a way as to indicate the total time of operation.

III. Test Procedure

8. *General Procedures.*—During all of the tests the operation of the heating plant was controlled by a heat-anticipating thermostat located 30 in. above the floor in the living room. An oil burning rate of 1.1 gal. per hr. was used. This rate was the minimum obtainable with a clean fire and with a CO_2 content of the flue gases of not less than 8 per cent at the smoke outlet of the boiler. The oil used weighed 7 lb. per gal. and had a calorific value of 19 550 B.t.u. per lb.

The doors between rooms were open at all times. At 7:00 A.M., 11:00 A.M., 3:00 P.M., 7:00 P.M. observations were recorded of the room air temperatures as determined by the thermocouples located 3 in., 30 in., and 60 in. above the floor and 3 in. below the ceiling. The air temperature in the basement, the garage, and the attic, the relative

*"The Use of an Elbow in a Pipe Line for Determining the Rate of Flow in the Pipe." Univ. of Ill. Eng. Exp. Sta. Bul. No. 289.

humidity in the heated portion of the house, the temperature inside and outside the sheathing back of each radiator, and the temperature gradient through the walls of the house were also observed at these times. Complete daily records were made of the operating time, the number of cycles and the power consumption of the oil burner and circulator, and the weight of oil consumed. Recording instruments made continuous records of the stack temperature and draft, the CO_2, the temperature of the water at the boiler outlet and return, the outdoor air temperature, the indoor relative humidity, and the air temperature in each room 3 in. and 30 in. above the floor and 3 in. below the ceiling. Other daily observations included the total amount of electricity, gas, and water used in the house, the number of occupants, and general weather conditions.

Four principal series of tests were conducted. The operating conditions and test procedures for each are given in connection with the discussion of the respective test series. The procedure for all special tests is presented in connection with the discussion of the respective test results.

9. *Series A.*—Tests included in series A were conducted during the heating season of 1940-41. The average room air temperature at the 30-in. level was maintained constant at approximately 72 deg. F. both day and night. The wiring diagram of the arrangement of the control system as used for the tests in series A is shown in Fig. 5. When heat was required, both the oil burner and the circulator were started and continued to operate until stopped by the room thermostat, unless the period of operation was of sufficient duration to heat the boiler water to a temperature of 200 deg. F., corresponding to the setting of the high-limit aquastat. If this occurred, the high-limit aquastat would stop the burner, but would permit the circulator to continue operating.

The low-limit aquastat was so connected electrically that it would operate the burner independently of the room thermostat and circulator at any time the temperature of the water in the boiler dropped below 165 deg. F. The angle flow-control valve, located in the hot-water supply pipe leaving the boiler, prevented any circulation of water between the boiler and the radiators except while the circulator was in operation. This method of control is in common usage, and is necessary where the boiler is to be used both to heat the house and to heat domestic hot water.

10. *Series B.*—Tests included in series B were also conducted during the heating season of 1940-41. This series differed from series A

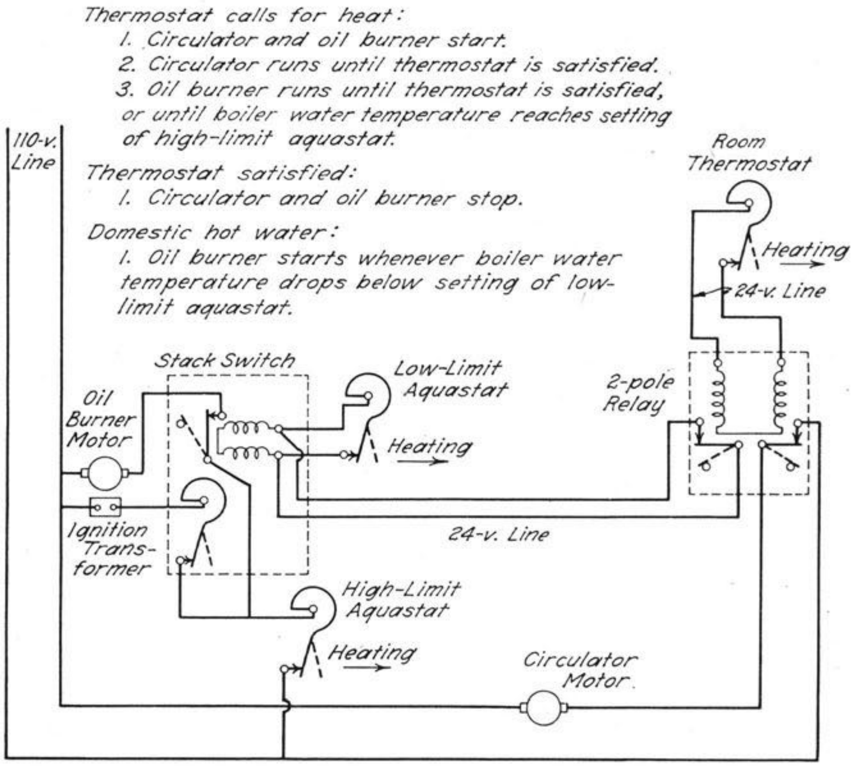

FIG. 5. CONTROL SYSTEM

only in that for series B the operation simulated that of a plant without a flow-control valve or low-limit aquastat. Such a plant is adapted to conditions under which no provision is made for heating water for domestic purposes by means of the boiler. The low-limit aquastat, shown in Fig. 5, was eliminated and the flow-control valve was locked open so that it could not prevent the circulation of water in the heating system at any time. Under these conditions the radiators could continue to cool all of the water in the system as long as sufficient temperature difference persisted to effect any circulation by gravity, and the temperature of the water in the boiler could vary anywhere between that of the room and 200 deg. F. The operation of the burner and circulator was controlled by the room thermostat alone.

11. *Series C.*—Tests included in series C were conducted during the heating season of 1941-42. The same heating system was used in

series A, B, and C, except that for series C the changes indicated in Section 6 were made in the amount of installed radiation in the living room and S.W. bedroom. The test procedure for series C was identical with that for series A, except that the setting of the room thermostat was reduced to 66 deg. F. at 10:00 P.M. and restored to 72 deg. F. at 5:30 A.M.

12. *Series D.*—Tests included in series D were conducted during the heating season of 1941-42, making use of the same heating system as that used for series C, operated without the flow-control valve and low-limit aquastat as in series B. The test procedure also was the same as that for series B, except that the setting of the room thermostat was reduced to 66 deg. F. at 10:00 P.M. and restored to 72 deg. F. at 5:30 A.M.

IV. Fuel Consumption and Operating Cycles

13. *With and Without Flow-Control Valve and Low-Limit Aquastat.*—A comparison of the burner performance curves for series A and B, shown in Fig. 6a, indicates that in mild weather, or at low indoor-outdoor temperature differences, operation on series A, with the low-limit aquastat and the flow-control valve operative, resulted in increased burner operating time, burner power, and fuel consumption. These increases were accompanied by higher flue gas temperatures and larger heat losses from the chimney gases resulting from increased burner operating time occasioned by the higher average boiler water temperature maintained in mild weather in series A as compared with that maintained in series B, for which the system was operated without a low-limit aquastat and flow-control valve. The increase in fuel consumption is further discussed in Section 26. As the indoor-outdoor temperature difference became larger, the average boiler water temperature for series B automatically approached that for series A as a result of the more frequent burner operation required to maintain the normal room temperature of 72 deg. F. At approximately 70 deg. F. indoor-outdoor temperature difference, the curves for series A became tangent to those for series B, and at all temperature differences above this the operating characteristics for both series were the same. This indicated that for indoor-outdoor temperature differences above 70 deg. F. the amount of burner operation required in series A to maintain normal room temperature was sufficient to maintain a minimum water temperature of approximately 165 deg. F. in the boiler at the location of the aquastat. In milder weather the room thermostat did not oper-

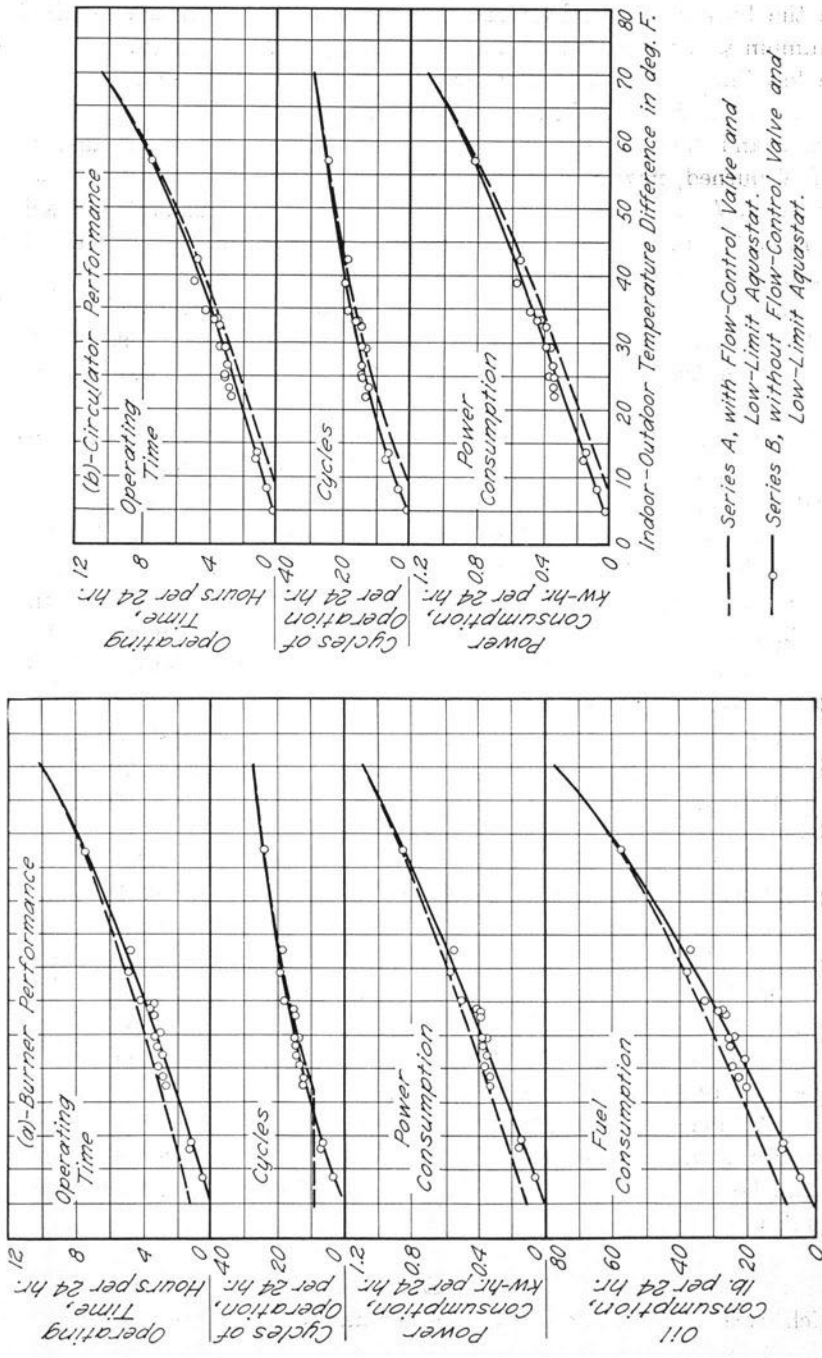

FIG. 6. BURNER AND CIRCULATOR PERFORMANCE CURVES, SERIES A AND B

ate the burner over a long enough period to maintain the required minimum water temperature in the boiler, and, as a result, in series A the low-limit aquastat continued the operation of the burner for a short time at the end of each cycle after the room temperature became normal and the circulator ceased to operate. The excess in the amount of fuel burned, power consumed, and time of burner operation shown for series A over that shown for series B is the amount of each required to maintain sufficient temperature in the boiler to make it possible to use the latter as a means of heating water for domestic purposes. Since the internal water heater was not connected to the hot-water storage tank during these tests, the results do not include any data on the standby loss from the storage tank, or on the requirements for the actual heating of water for domestic uses.

In Urbana, Illinois, the average outdoor temperature during the heating season is approximately 38 deg. F. Hence, if the house temperature is maintained at 72 deg. F. the average heating day is one in which the indoor-outdoor temperature difference is 34 deg. F. The average length of the heating season is 204 days. Comparisons based on this temperature difference may be regarded as indicative of the results to be secured from the entire heating season. From Fig. 6 it is evident that 33.5 and 30 lb. of oil per 24 hr. were required in series A and B, respectively, at an indoor-outdoor temperature difference of 34 deg. F., or, since the fuel oil used weighed 7 lb. per gal., operation on series A required an additional ½ gal. of oil per average heating day, or 102 gal. per heating season over that required for operation on series B.

This increase in the fuel consumption is directly related to the average temperature of the water in the boiler, and undoubtedly would be affected by any changes in the setting of the low-limit aquastat. It is possible that the results of further work now in progress will prove that a setting of the aquastat lower than 165 deg. F. may be sufficient to provide adequate hot water for domestic use, and in that case some reduction in the fuel consumption over that shown for series A may be expected. Furthermore, the excess amount of fuel required to operate the boiler so that it can be used to heat water for domestic service may vary with the size of plant and the total load on the boiler. Hence a figure representing the percentage fuel consumption is of doubtful significance.

No significant difference existed in the number of burner operating cycles required for series A and B except in very mild weather, in which case a minimum of 9 cycles per day was required in series A in order to maintain a minimum boiler water temperature of approximately 165 deg. F. at the location of the aquastat.

As shown by the curves in Fig. 6b, at a given indoor-outdoor temperature difference the circulator operating time, power, and number of cycles were slightly less in series A than in series B. This condition resulted from the fact that higher average radiator temperatures, and consequently higher rates of heat input to the rooms, were obtained during the on-periods of the circulator in series A than in series B.

14. *Reduced Room Air Temperatures at Night.*—Curves comparing the burner performance for series A in which the house was maintained at a uniform temperature of 72 deg. F. 24 hr. per day, and series C, in which the setting of the thermostat was reduced to 66 deg. F. at 10:00 P.M. and restored to 72 deg. F. at 5:30 A.M., are shown in Fig. 7a. With a given average outdoor temperature over a period of 24 hours any saving in fuel consumption resulting from reducing the thermostat setting at night is brought about by the fact that the average indoor temperature, and therefore the average indoor-outdoor temperature difference is reduced as compared with that existing when the room air temperature is maintained at 72 deg. F. for 24 hours. With a given heating system and house, the average indoor-outdoor temperature difference for the 24 hours determines the heat loss from the house irrespective as to whether this difference was obtained with a constant indoor temperature for the 24 hours as in series A, or with a variable one, as in series C. Hence these two series cannot be compared on the basis of a 24-hr. average for the indoor-outdoor temperature difference as applied to both series, because at a given difference the same fuel consumption would be obtained, and the result would be one curve instead of two. In order to determine the fuel saving it was necessary to make comparisons for days that had the same average outdoor temperature over the 24 hours, with the further condition that the same indoor temperature was maintained during the daytime for both series of tests. These conditions were satisfied by basing the indoor-outdoor temperature differences for series C and D on the average indoor temperature between 11:00 A.M. and 10:00 P.M. and the average outdoor temperature for 24 hours, and for series A and B on the 24-hr. averages for both the indoor and outdoor temperatures. All of the curves in Figs. 7 and 8 have therefore been plotted on this basis.

For both series A and C the heating system was equipped with a low-limit aquastat and flow-control valve. Operation with the thermo-

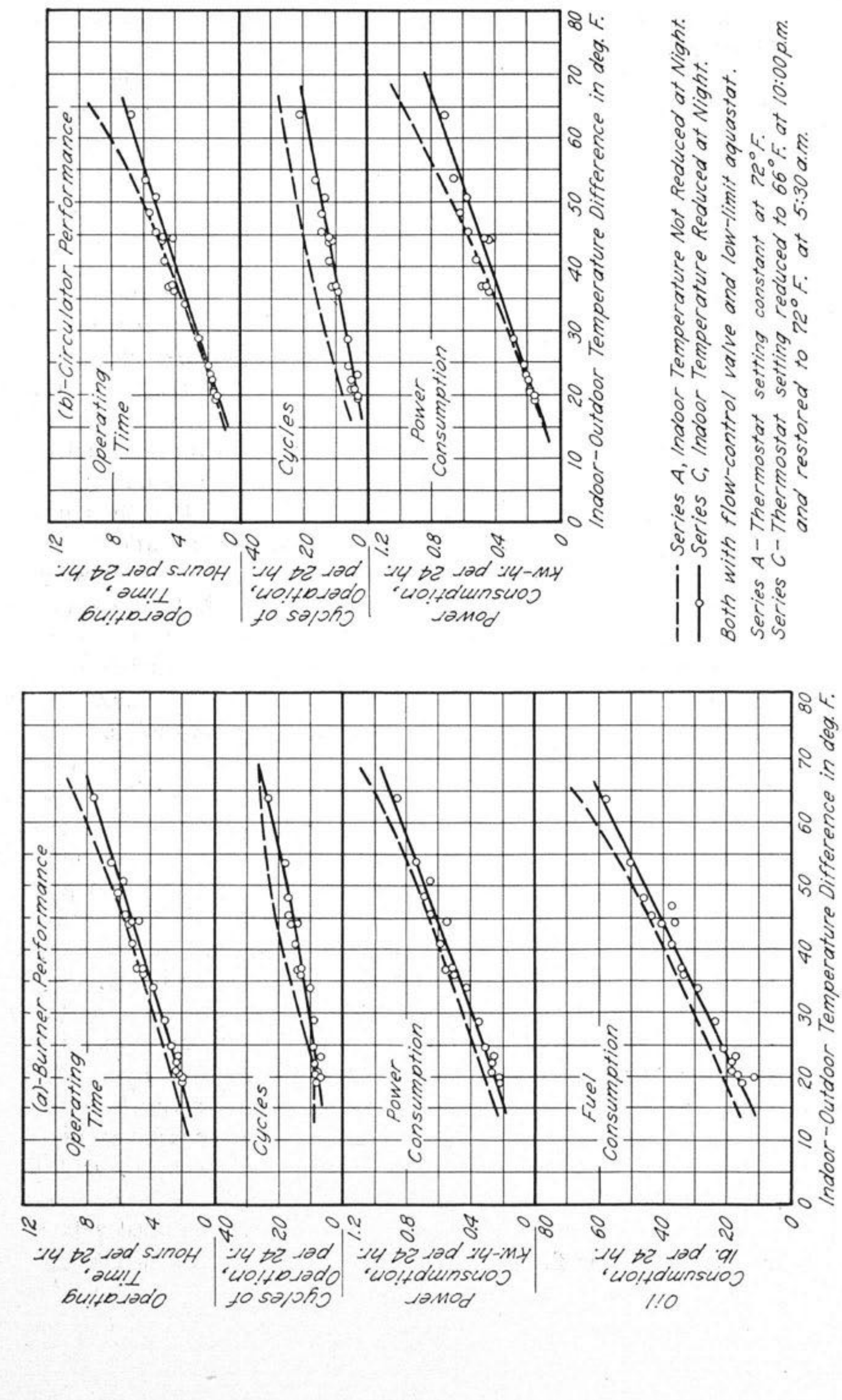

FIG. 7. BURNER AND CIRCULATOR PERFORMANCE CURVES, SERIES A AND C

stat setting lowered to 66 deg. F. at night resulted in reductions in the burner operating time, cycles of burner operation, burner power, and fuel consumption throughout the complete range of outdoor temperatures occurring during the tests. Savings effected at an indoor-outdoor temperature difference of 34 deg. F. are also approximately representative of seasonal savings. At an indoor-outdoor temperature difference of 34 deg. F. the actual saving in fuel amounted to 10.4 per cent.

At any given indoor-outdoor temperature difference occurring within the range of weather conditions encountered, fewer cycles of burner operation were obtained in series C than in series A. The greatest difference in the number of cycles accompanied temperature differences of from 40 to 50 deg. F. With temperature differences of less than 40 deg. F. the rate of cooling of the air in the house was such that the thermostat would operate the burner not more than once or twice during the night. When the indoor-outdoor temperature difference was greater than 50 deg. F. the rate of cooling increased, and sufficient time was required to raise the air temperature to 72 deg. F. in the morning so that the high-limit aquastat caused the burner to cycle several times. As a result, the number of burner cycles in series C approached that obtained in series A as the indoor-outdoor temperature difference became greater than 50 deg. F.

Figure 7b shows the circulator performance curves for series A and C, plotted on the same basis as the burner performance curves in Fig. 7a. At a 34 deg. F. indoor-outdoor temperature difference the operating time of the circulator was 5 per cent less for series C than it was for series A. The curves in Fig. 7 indicate that, as the indoor-outdoor temperature difference increased, an increase in the percentage of fuel saving and circulator and burner operating time occurred.

Operating under the conditions of series C resulted in a slightly greater reduction in the number of cycles of operation for the circulator, than for the oil burner. By comparing the curves representing the burner and circulator cycles for series C, in Fig. 7, it may be observed that at all indoor-outdoor temperature differences two or three more cycles of operation were obtained for the burner than for the circulator. During the night, in order to keep the temperature of the water in the boiler above that corresponding to the setting of the low-limit aquastat, several operations of the burner were required at low indoor-outdoor temperature differences, while no operation of the circulator was necessary. On the other hand, at high temperature differences, the time required in the morning to raise the indoor temperature from 66 deg. F. to 72 deg. F. was sufficient to allow the burner to

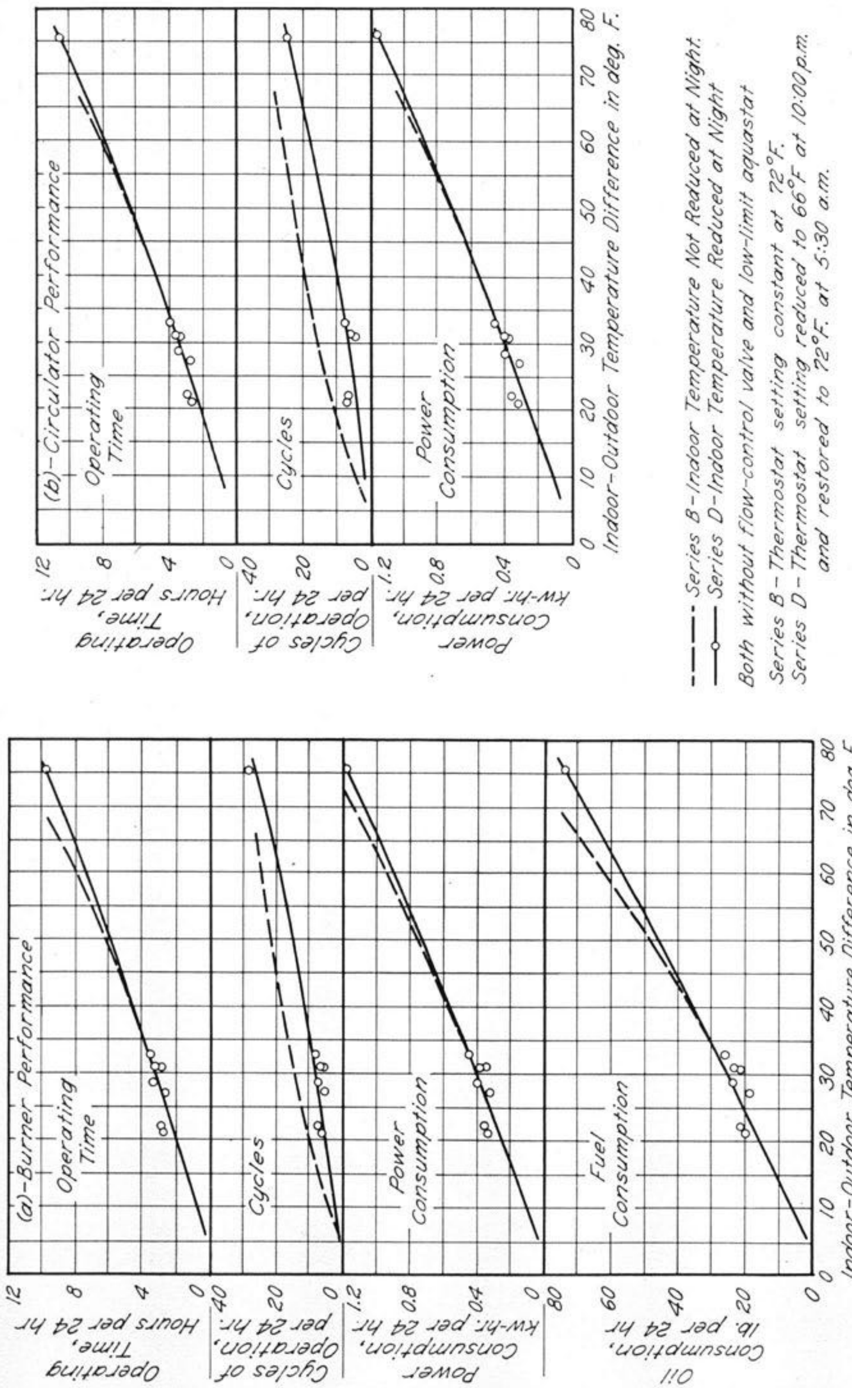

FIG. 8. BURNER AND CIRCULATOR PERFORMANCE CURVES, SERIES B AND D

cycle several times by action of the high-limit aquastat. During this time, since the circulator was controlled only by the room thermostat, it operated continuously.

Comparisons of the burner and circulator performances obtained for series B with those obtained for series D are shown in Fig. 8. During both of these series the heating system was operated without the flow-control valve and low-limit aquastat, and for series D the setting of the room thermostat was reduced at night.

The curves show that on days having an indoor-outdoor temperature difference of 34 deg. F., or less, reducing the indoor temperature at night resulted in no saving in the fuel used, or in the operating time and power consumption for the burner and circulator. The fuel saving indicated at an indoor-outdoor temperature difference of 34 deg. F. is only approximately representative of the seasonal saving, and since the curves do indicate some saving at indoor-outdoor temperature differences greater than 45 deg. F., it is reasonable to assume that some *seasonal* saving would be effected. However, since only a relatively small portion of the total heating season is represented by indoor-outdoor temperature differences greater than 45 deg. F., it is also evident that in the case of operation without the flow-control valve and low-limit aquastat in the system no material seasonal saving resulted when the indoor temperature was reduced at night. On the other hand, over the whole range of indoor-outdoor temperature differences, a reduction in the number of operating cycles for both the burner and the circulator was obtained. This reduction was approximately the same as that exhibited by the results for series C as compared with those for series A.

Over the whole range of outdoor temperatures, the time required to raise the average house temperature from 66 deg. F. to 72 deg. F., as shown in Fig. 9, was approximately 30 minutes less in series C than it was in series D. In the discussion of water temperatures in Section 16, it is shown that this reduction of 30 minutes in the time for warming up in the morning is a direct result of the additional heat stored in the water in the boiler when operating the system with a flow-control valve and low-limit aquastat. The additional 30 minutes of operation during the warming-up period in series D was almost equal to the difference in total saving in burner operating time resulting from the reduction of indoor temperature at night in series C as compared with series D, and was sufficient to account for the fact that the saving effected in series D was less than that effected in series C.

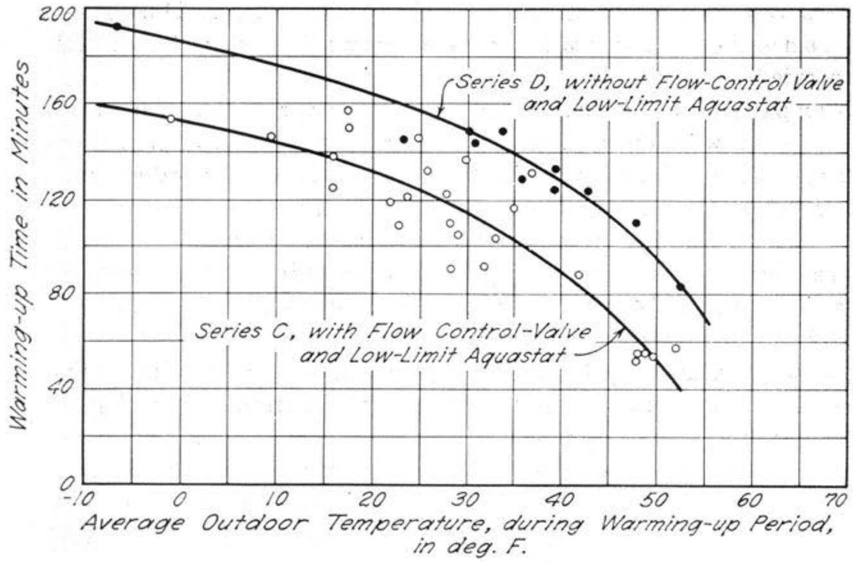

FIG. 9. TIME REQUIRED FOR MORNING WARMING-UP PERIOD, SERIES C AND D

At an average indoor-outdoor temperature difference of 34 deg. F., the burner operating time and fuel consumption was approximately the same in series B, C, and D.

V. WATER TEMPERATURES IN BOILER AND RADIATORS

15. *With and Without Flow-Control Valve and Low-Limit Aquastat.*—Figures 10 and 11 show the temperature variations of the water in the boiler and the radiators for a complete operating cycle representative of series A and series B. In series A, water was circulated through the heating system at a rate of about 13 gal. per min. during the time of circulator operation. When the circulator was not operating, the flow-control valve prevented any circulation of water taking place between the boiler and the radiators. As indicated by the curves in Fig. 10, the water in the boiler was still hot at the end of the off-period, while that remaining in the piping and radiators had cooled down to approximately 90 or 95 deg. F., depending on the length of the off-period. When the room thermostat closed the circuit the circulator and burner were started simultaneously. Within the time of one minute the hot water which was stored in the boiler was distributed to the radiators in the house, quickly supplying heat to prevent further cooling of the rooms. At the same time, the cooler water contained in

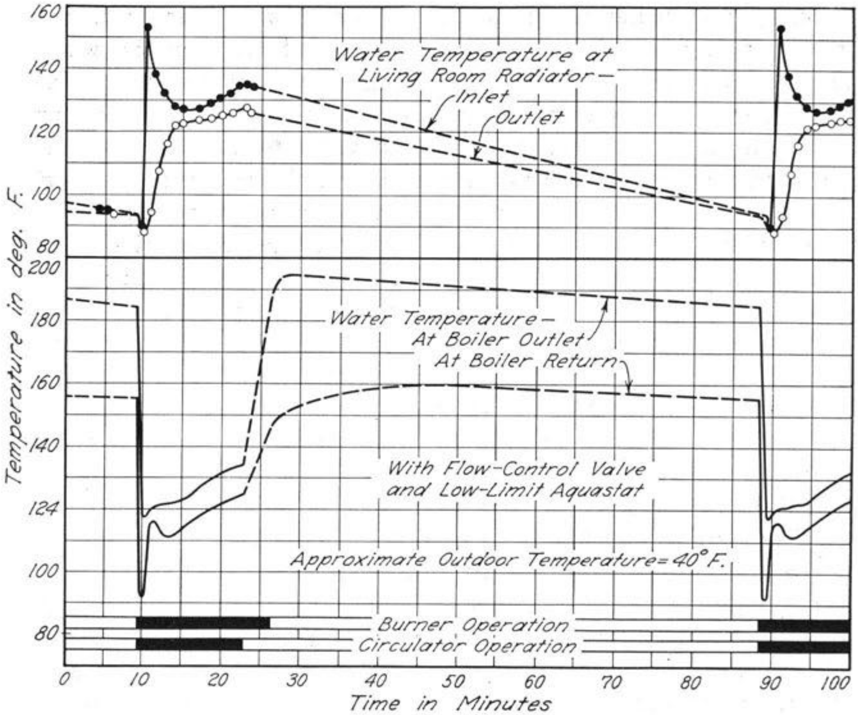

FIG. 10. WATER TEMPERATURES FOR SERIES A

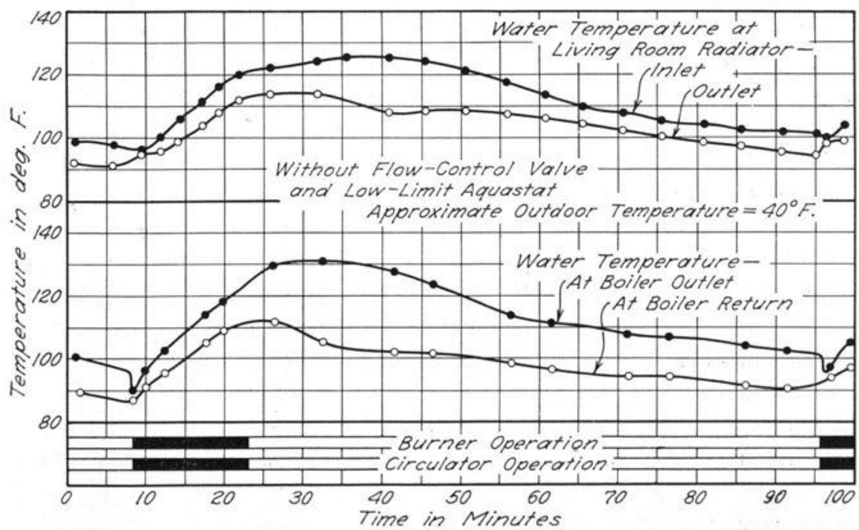

FIG. 11. WATER TEMPERATURES FOR SERIES B

the piping and radiators during the off-period was returned to the boiler to be reheated. After the first surge of heat from the water stored in the boiler had been expended, the water entering the radiators dropped in temperature owing to the fact that the burner, during the interim, could not heat the returned water to a temperature as high as that of the water originally stored in the boiler at the start of the on-period. The period represented by decreasing temperature of the water entering the radiators continued until all of the water contained in the piping and radiators during the off-period had passed through the boiler. After the period of dropping temperature, and extending throughout the rest of the on-period, there was a gradual increase in the temperature of both the water in the boiler and that supplied to the radiators, since the burner supplied heat to the water in the boiler more rapidly than it was removed from that in the piping and radiators.

In 13½ minutes the room thermostat stopped the circulator. At this time the average water temperature in the boiler was only 130 deg. F., and the low-limit aquastat continued to operate the burner for an additional 3½ minutes, heating the boiler water to approximately 175 deg. F., or 10 deg. F. above the aquastat setting. During the off-period of the circulator, with the flow-control valve in the system, the water in the radiators and uninsulated piping cooled about 40 deg. F., while that in the insulated boiler dropped approximately 10 deg. F.

As shown in Fig. 11 the variations in water temperature when operating on series B, without the flow-control valve and low-limit aquastat, were somewhat different. In this case the boiler and radiators cooled at about the same rate during the off-period, since there was nothing to prevent water circulation by gravity action. When the rooms required heat, the burner and circulator were started simultaneously by the thermostat, and a gradual increase in water temperatures followed. After 14½ minutes of operation the room thermostat stopped both the circulator and the burner, even though the temperature of the water in the boiler averaged only about 118 deg. F. As a result of heat absorbed, largely from the residual heat stored in the refractory fire box, the temperature of the water leaving the boiler continued to rise from 123 deg. F. at the time the burner shut off to 131 deg. F. some 9 or 10 minutes later. During the time of circulator operation, the water was forced through the heating system at a rate of 13 gal. per min., the same rate as in series A. When the circulator was not running some gravity circulation of water occurred, but at a rate too low to measure.

16. *Reduced Room Air Temperatures at Night.*—The same characteristic differences in the temperature cycles for the water in the boiler and the radiators obtained in series A and B, and shown in Figs. 10 and 11, were also obtained when the air temperature was reduced at night and the system operated with and without flow-control valve in series C and D respectively.

When the thermostat setting was reduced at night there was always a long off-period following the time at which the change was made. With the flow-control valve and low-limit aquastat in the system, series C, the water in the radiators and uninsulated piping cooled to room temperature during this long off-period, while that in the boiler cooled at an average rate of about 10 deg. F. per hr. until the temperature was approximately 165 deg. F. at the location of the low-limit aquastat. When this occurred, the burner operated for a sufficient length of time to raise the temperature of the water in the boiler to approximately 175 deg. F.

When operating without the flow-control valve and low-limit aquastat in series D, the water in the entire system cooled to room temperature during the long off-period following the reduction of the thermostat setting. The rate at which the air in the house cooled at night was such that, for the average winter day, the thermostat would not start the burner during the entire period of from 10:00 P.M. to 5:30 A.M. Under these conditions, in both series C and D, at the start of the morning warming-up cycle the temperature of the water in the radiators and piping was about the same as the indoor air temperature. In series D the water in the boiler also had cooled to room temperature, while in series C it could never cool below approximately 165 deg. F.

For both methods of operation, with and without flow-control valve and low-limit aquastat, the curves in Fig. 12 show the variations in temperature of the water at the outlet and inlet connections of the boiler during the entire warming-up period occurring after the thermostat had been reset to 72 deg. F. The outdoor temperature was approximately 42 deg. F. at 6:00 A.M. in both cases. The effect of the presence of the hot water in the boiler at the start of the warming-up period in series C is shown by the full-line curves. The temperature of the water at the boiler outlet was 158 deg. F. during the first 4 minutes of the period, then it dropped rapidly to about 108 deg. F., and then gradually increased until it attained a value of 203 deg. F. at the end of 85 minutes. At this time the indoor air temperature reached 72 deg. F. and the room thermostat stopped both the burner and the circulator. For series D, when the outdoor temperature was 42 deg. F., the tem-

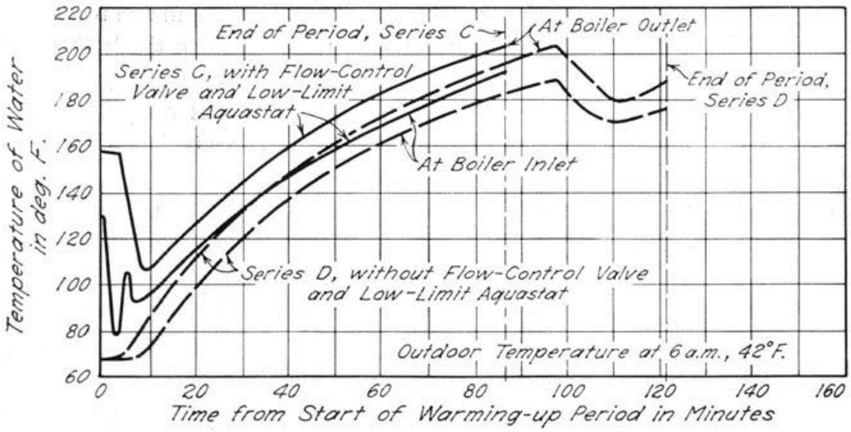

FIG. 12. BOILER WATER TEMPERATURES DURING WARMING-UP PERIOD, SERIES C AND D

perature of the water at the boiler outlet, represented by the upper broken line, was only 68 deg. F. at the start of the warming-up period. This temperature steadily increased until it reached 204 deg. F. at 97 minutes after the start of the period. At this time the high-limit aquastat stopped the burner, but not the circulator. The outlet water then cooled to 180 deg. F., and at this time the burner was restarted and continued to operate for the remainder of the warming-up period. In series D the time required to raise the indoor air temperature to 72 deg. F. was 121 minutes, or 36 minutes longer than that required with operation on series C at the same outdoor temperature.

Over the total time of the warming-up period, the heat released in the house by the radiators and piping is directly proportional to the area between the portions of the inlet and outlet boiler water temperature curves. Since the rate of circulation of the water through the heating system during the on-periods of the circulator was the same in both series, the proportionality factor would also be the same, and the heat releases for the two series can be directly compared by comparing areas included between the respective inlet and outlet temperature curves. Over the total time included in the warming-up period for each series, there is less than 2 per cent difference between the total areas obtained from the inlet and outlet water temperature curves representing series C and those representing series D. This indicates that, at the given outdoor temperature, the same quantity of heat was supplied to the house during the warming-up period by both methods of operation. However, in series D not only the water in the

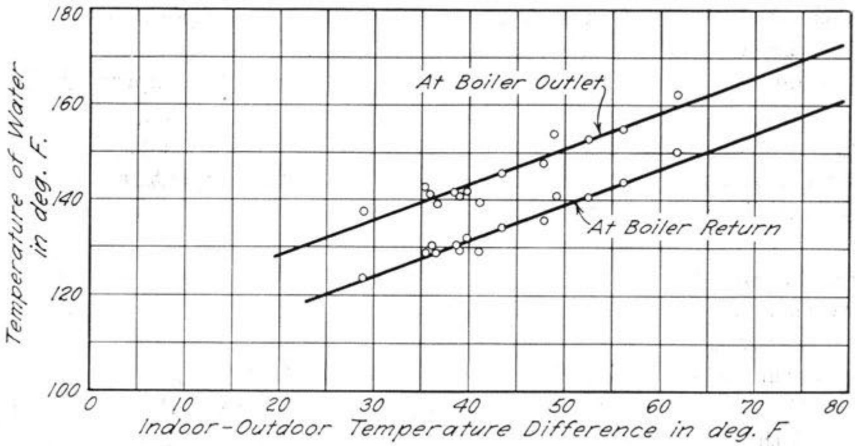

FIG. 13. AVERAGE TEMPERATURE OF WATER ENTERING AND LEAVING BOILER AT END OF CIRCULATOR ON-PERIOD, SERIES A

radiators and piping, but also the boiler castings, refractories, and the water in the boiler cooled to room temperature at night. As a result, during the additional 36 minutes required to bring the air temperature to 72 deg. F., the burner operated 22 minutes in order to supply the extra heat necessary to bring the boiler castings, refractories, and the water in the boiler back to normal temperature in the morning. Since, at a given outdoor temperature, the same amount of heat was supplied to the house during the warming-up period for both series C and D, but 22 minutes more of burner operation were required for series D than for series C, it seems evident that a lower overall house efficiency was obtained in series D than in series C. Part of the heat represented by the 22 minutes of additional burner operation was used to bring the boiler casting, refractories, and the water in the boiler back to normal temperature, and the remainder appeared as increased chimney losses resulting from the higher flue gas temperatures accompanying the longer period of burner operation occurring in series D.

17. *Relation Between Size of Radiators and Water Temperature.*—The curves in Fig. 13 show the water temperatures obtained at the outlet and return connections of the boiler for operation with series A. Just sufficient radiation was installed in the house to offset the calculated heat loss at an outdoor temperature of −10 deg. F. when the radiators were constantly supplied with water entering at 200 deg. F. and leaving at 190 deg. F. Under these conditions the temperature of the water leaving the boiler would be somewhat higher than 200 deg. F.,

and the circulator would operate continuously. By extrapolating the water temperature curves in Fig. 13 it may be observed that, when the indoor-outdoor temperature difference was 80 deg. F. the temperature of the water leaving the boiler was not greater than 175 deg. F., or approximately 30 deg. F. lower than that selected for the design of the system. By a similar extrapolation of the circulator operating time curve, shown in Fig. 6, a total operating time of only 12 hr. per day instead of 24 was obtained when the indoor-outdoor temperature difference was 80 deg. F. This seems to indicate that the radiators were considerably oversized for the house. In Section 26 it is shown that the actual heat loss from the house, including the basement, was 44 per cent greater than the calculated heat loss excluding the basement. It is further shown that only about one half the heat required to offset the actual heat loss from the house at the design temperature was supplied by the radiators. The balance was supplied from sources of heat other than the radiators. These sources consisted of the lights, the occupants, the power used, the supply mains and risers. In addition, some heat was regained from the chimney and the boiler and smoke pipe.

VI. Room Temperature Conditions

18. *Variations Between Rooms.*—The maximum difference between the average temperatures of the air at the 30-in. level in any two rooms was 1.1 deg. F., while the maximum differences observed on the first and second stories separately were 0.7 deg. F. and 1.1 deg. F. respectively. During the 1941-42 season the average temperature on the second story was about 0.3 deg. F. lower than that on the first story. The method of operating the heating system had no material effect upon the temperature variation between the rooms.

19. *Variations Within Rooms, With and Without Flow-Control Valve and Low-Limit Aquastat.*—There was some possibility that the elimination of the flow-control valve might cause overruns in the air temperatures within the house, but that this was not the case is indicated by the data given in Table 3. The values given in this table represent the conditions which existed in the living room, but they are also representative of the conditions in the other rooms of the house. A difference of only 0.5 deg. F. between the maximum and minimum air temperatures at the 30-in. level was obtained during a cycle of operation. The maximum variation at any level during a cycle was only 0.9 deg. F., and this occurred near the ceiling. These values are

TABLE 3
TEMPERATURE OF AIR IN LIVING ROOM* FOR SERIES B

Location	Maximum Temperature deg. F.	Minimum Temperature deg. F.	Difference deg. F.
Ceiling	72.9	73.8	0.9
30-in. level	71.3	71.8	0.5
Floor	69.8	70.6	0.8
Difference, Ceiling-Floor	3.1	3.2	

*Approximate outdoor temperature = 40 deg. F.

all well within the limits of control to be expected from the average room thermostat available today.

Two reasons may be assigned to explain why an overrun did not occur when operating under the conditions existing in series B: (1) For given outdoor temperatures above zero deg. F., the average water temperatures in the radiators were lower in series B than they were in series A, and, therefore, the rate of heat output was more nearly equal to the rate of heat loss from the house in series B than it was in series A. For given outdoor temperatures below zero deg. F. the water temperatures and rates of heat output were approximately the same for both series. (2) The boiler used in these tests had a water capacity of 13.5 gallons, while the radiators and piping had one of approximately 38 gallons. With either method of operation, the heat contained in the 38 gallons of water in the piping and the radiators, at the time the room thermostat stopped the circulator, was released to the house during the off-periods. Thus, the only additional heat released during the off-periods in series B, over that released in series A, was the heat contained in the 13.5 gallons of water in the boiler and that which the water could absorb from the heated boiler and setting.

The average differences in temperature between the air at the 30-in. level and that 3 in. above the floor, 60 in. above the floor, and 3 in. below the ceiling are shown plotted against the indoor-outdoor temperature difference in Fig. 14. In all cases the air at the 30-in. level was maintained constant at 72 deg. F. during the 24 hours of the day. The temperature difference from floor to ceiling increased with descreasing outdoor air temperatures, and the method of operating the plant had no material effect on these differences. In the coldest weather experienced, a difference of only approximately 4.5 deg. F. was observed between the average temperature of the air 3 in. above the floor and that 3 in. below the ceiling.

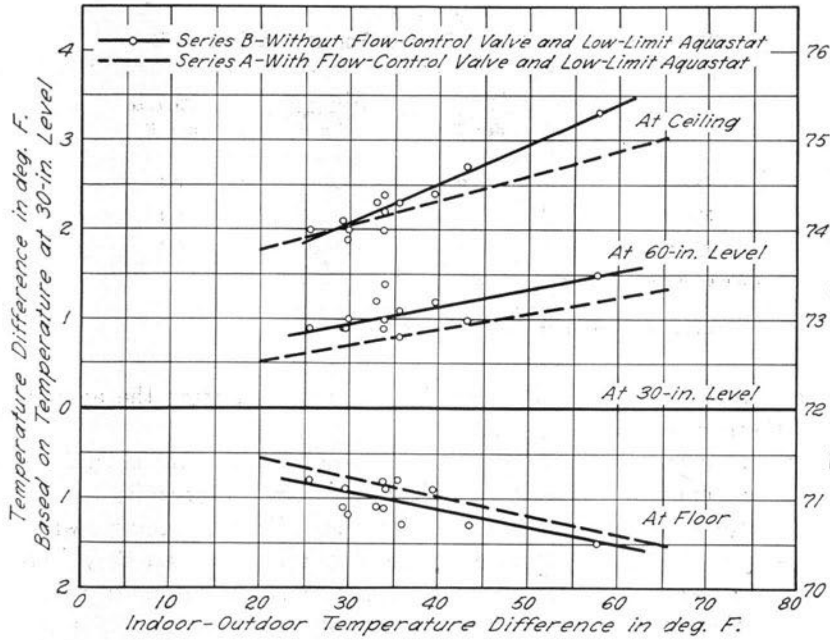

Fig. 14. Indoor Air Temperature Differentials, Series A and B

20. *Variations Within Rooms, With Reduced Indoor Temperature at Night.*—The indoor air temperature differentials obtained with series C, operating with reduced indoor air temperature at night, and series A, operating with a constant indoor air temperature of 72 deg. F. maintained for 24 hours, are shown in Fig. 15. The temperature differentials were based on the averages of the temperatures in all the rooms. For the purpose of determining whether the reduction of indoor temperature at night and the subsequent morning pickup in series C had any effect on the indoor air temperature differentials obtained with normal day time operation, only the temperatures prevailing during the period over which the house was maintained at 72 deg. F. were considered as having any significance. Hence, for series C in Fig. 15, the room temperature differentials obtained from the average indoor temperatures from 11:00 A.M. to 10:00 P.M. for a given day were plotted against the indoor-outdoor temperature differences obtained by subtracting the average temperature at the 30-in. level between the hours of 11:00 A.M. and 10:00 P.M. from the average outdoor temperature for the 24 hours. From Fig. 15 it may be observed that the reduction in the indoor temperature at night in series C had no significant effect on the indoor air temperature differentials.

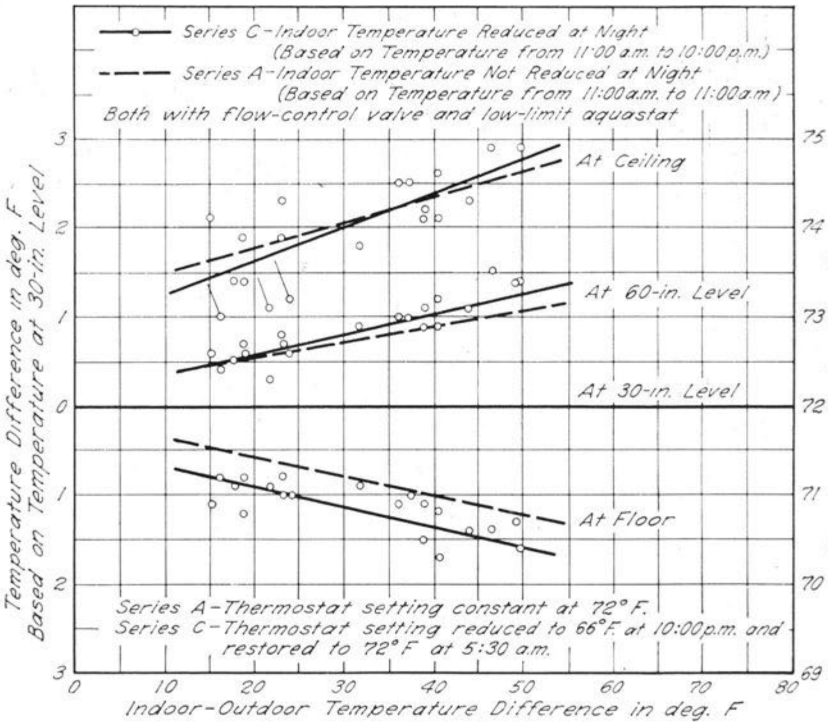

FIG. 15. INDOOR AIR TEMPERATURE DIFFERENTIALS, SERIES A AND C

At all indoor-outdoor temperature differences the temperature at the floor was about 0.3 deg. F. lower in series C than in series A. Between the running of series A and C two sections were added to one of the radiators in the living room and five sections were removed from the radiator in the S.W. bedroom. After these changes were made in the plant, a few check tests, run under the conditions of series A, indicated that the slight change in the temperature of the air at the floor should be attributed to causes other than effects resulting from the reduction in room air temperature at night. The curves shown in Fig. 15 also proved to be characteristic of the operation without the flow-control valve and low-limit aquastat in series B and D.

A typical graphic log showing air temperature variations and times of burner and circulator operation in the Research Home when operating with reduced indoor temperatures at night is presented in Fig. 16. At 10:00 P.M. the thermostat was set at 66 deg. F. From 10:00 P.M. to 12:30 A.M. neither the burner nor the circulator operated, while the air temperature at the 30-in. level decreased from 72 deg. F. to 66 deg.

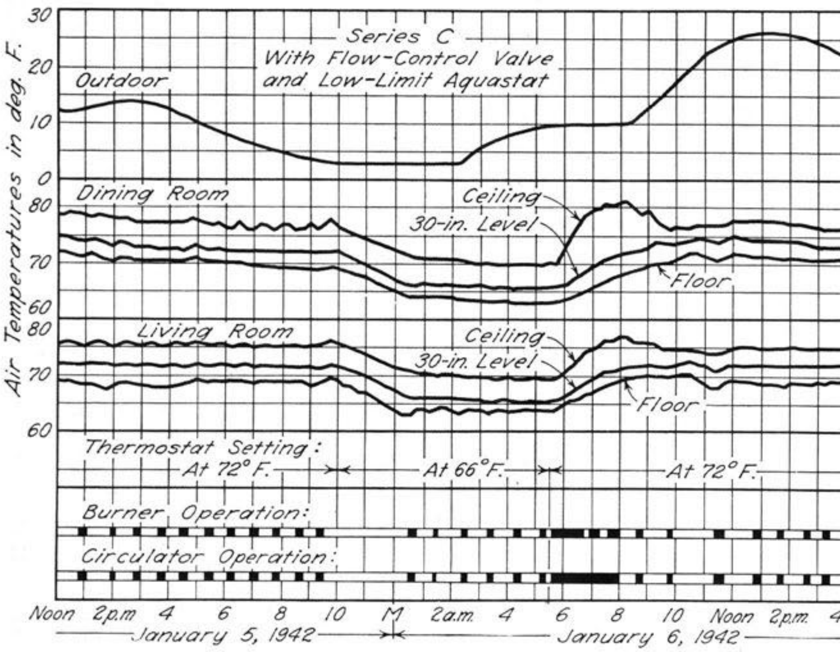

FIG. 16. GRAPHIC LOG FOR OPERATION WITH REDUCED INDOOR TEMPERATURES AT NIGHT, SERIES C

F. From 12:30 A.M. to 5:30 A.M. the burner and circulator operated intermittently to maintain the air temperature at 66 deg. F. At the latter time the thermostat was set at 72 deg. F., and the burner and circulator started. The circulator operated continuously for 2.5 hours and during this time the house temperature increased from 66 deg. F. to 72 deg. F. In the course of this 2.5-hr. warming-up period the oil burner operated continuously for the first 75 minutes, and the temperature of the water in the system increased until the high-limit aquastat stopped the burner. In about 10 minutes the water cooled enough to permit the aquastat to start the burner again and then, under control of the high-limit aquastat, the burner operated intermittently until the end of the 2.5-hr. warming-up period.

The air temperatures for both the dining room and the living room have been included in Fig. 16 since the overruns in temperature obtained in these two rooms during the warming-up period were the maximum and minimum observed in any of the rooms of the house. At the end of the warming-up period at 8:00 A.M., the temperature at the 30-in. level was just 72 deg. F. in both the dining room and the living

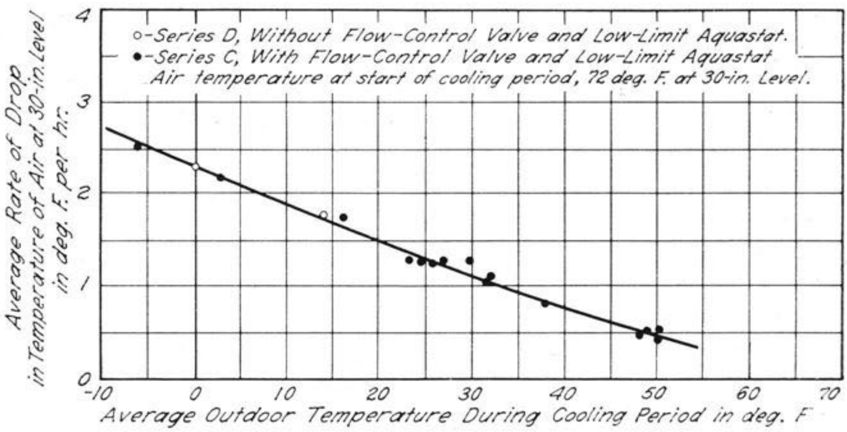

FIG. 17. RATE OF COOLING IN LIVING ROOM

room. At the same time, the temperature of the air 3 in. below the ceiling in the dining room reached a maximum of 81 deg. F., while that in the living room attained a maximum of only 77 deg. F. On the other hand the temperature of the air 3 in. above the floor in the dining room was 68 deg. F. and that in the living room was 69 deg. F. Thus it is evident that, while some overrun was exhibited in the temperature of the air 3 in. below the ceiling when the burner and circulator were stopped at the end of the warming-up period, no overrun in temperature occurred at the control level, 30 in. above the floor.

After the warming-up period, the natural circulation of air tended to equalize the air temperature at the different levels, decreasing the temperature of the air at the ceiling and slightly increasing the temperature at the floor and control levels, so that by about 10:00 A.M. the normal temperature differentials had been established in the rooms. Records similar to those shown in Fig. 16 were made of the temperature variations in each room of the house. These records indicated that there was a rough correlation between the amount of overrun in the temperature of the air near the ceiling and the ratio of the room volume to the amount of radiation installed. As this ratio increased the amount of overrun decreased.

After changing the setting of the thermostat from 72 deg. F. to 66 deg. F. at night, the rate at which the temperature of the air in the rooms dropped was primarily dependent upon the outdoor temperature prevailing during the cooling period. The average rate of temperature drop for the living room, which is also representative of that for the house as a whole, is shown in Fig. 17. An indoor temperature of 66 deg.

F. was never attained during the 7.5 hours between 10:00 P.M. and 5:30 A.M. unless the average cooling rate was equal to or greater than 6/7.5, or 0.8 deg. F. per hr. From Fig. 17 it may be observed that this occurred only when the average outdoor temperature at night was 39.5 deg. F. or less. It is therefore evident that at outdoor temperatures above 39.5 deg. F., setting the thermostat back more than 6 deg. F. would not result in any additional saving in fuel over that which could be obtained by the 6-deg. F. setback.

The cooling rate is a function of house construction and the indoor-outdoor temperature difference, and the total fuel saving that can be effected by operating with reduced indoor air temperatures at night is largely a function of three factors: (1) The number of degrees the thermostat is set back at night, (2) the number of hours of operation on reduced temperature at night, and (3) the rate at which the house cools. Taking into consideration the length of time required to warm the house in the morning and the time required to overcome the effect of cold walls and furniture, it is probable that, except in the most severe climates, the practical limits for night setback of the thermostat lie somewhere between 6 and 10* deg. F.

VII. Radiator Locations Relative to Exposed and Warm Walls

21. *General Statement.*—In order to determine whether the location and arrangement of the radiators had any effect on the comfort conditions maintained within a heated room, a series of tests was run in which five different locations and arrangements of radiators were used in the living room. In three of the installations single radiators consisting of 24 sections of 19-in. 4-tube, small-tube type radiation were used. In the other two installations the same amount and type of radiation was divided into two 12-section radiators. When the radiator was changed to the inside wall in the living room, corresponding changes were made in the location of those in the dining room and kitchen in order to minimize differences between rooms. With each of the radiator locations and arrangements the test procedure and method of operation was the same as those for series B, described in Section 10. In addition to the routine readings with each radiator installation, smoke studies were made in the living room in order to determine the nature of the air movement.

22. *Room Temperature Gradients.*—Figure 18 shows the location and arrangement of the radiators used in each of the five installations

*"Investigation of Oil-Fired Forced-Air Furnace Systems in the Research Residence," Univ. of Ill. Eng. Exp. Sta. Bul. No. 318, pp. 49-57.

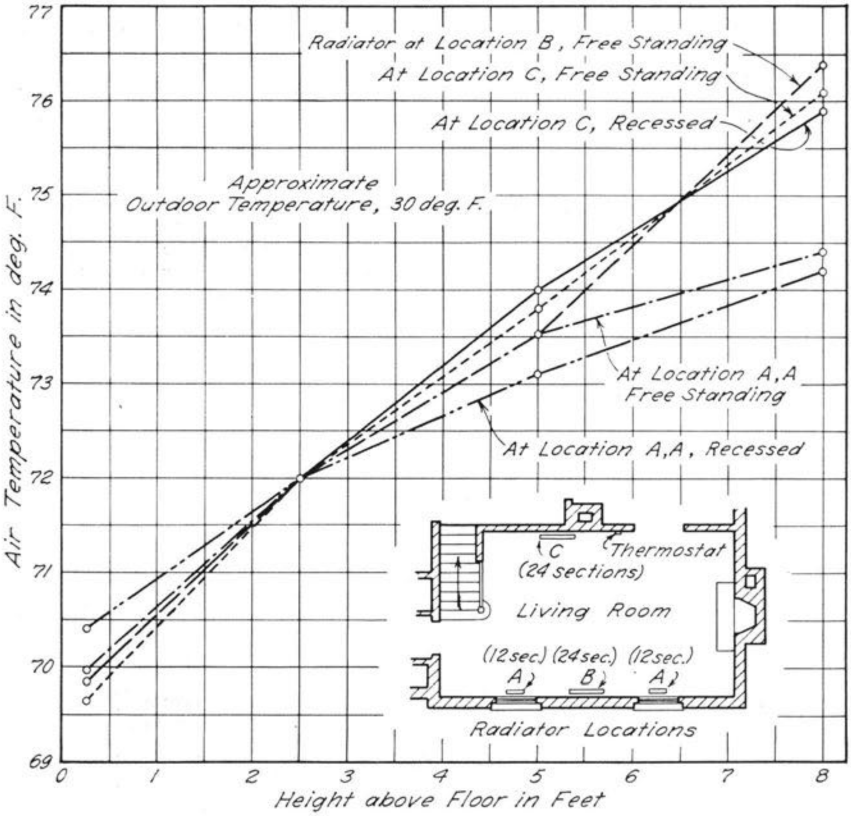

FIG. 18. AIR TEMPERATURE GRADIENT IN LIVING ROOM WITH FIVE DIFFERENT RADIATOR INSTALLATIONS

tested and the air temperature gradient obtained in the living room with each installation. These curves indicate that with a given radiator location, similar temperature gradients were obtained within the room with both the free-standing and recessed arrangements. All five of the radiator installations produced approximately the same temperature gradient in the living zone, or that portion of the room below the 60-in. level. However, at the floor and at the 60-in. level slightly higher and lower temperatures, respectively, were obtained with the recessed radiators under the windows than were obtained with the other installations.

The lowest ceiling temperature, approximately 74.5 deg. F., resulted when the radiators were placed under the windows in position A, slightly more than 76 deg. F. was obtained when they were located adjacent to the walls in positions B or C. Normally, high temperatures

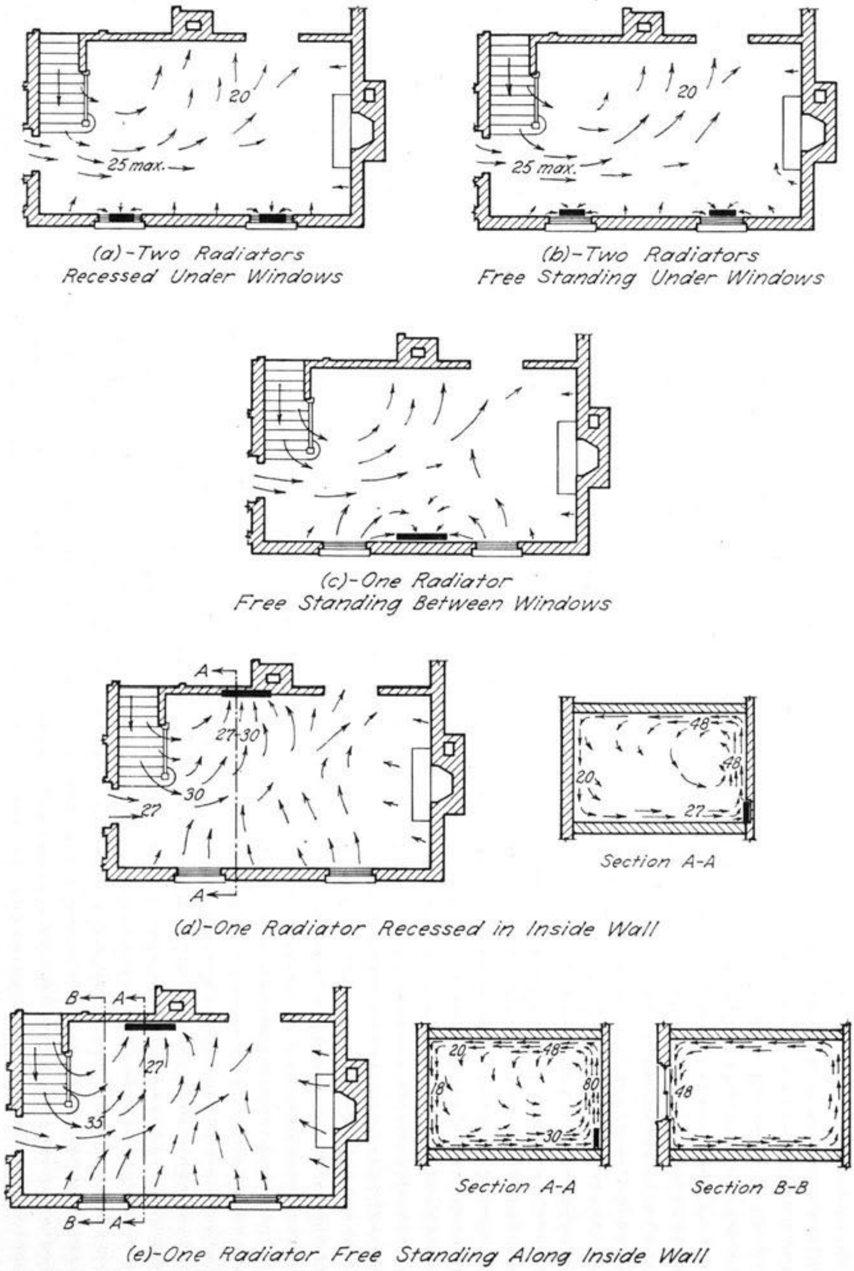

FIG. 19. AIR MOVEMENTS IN LIVING ROOM WITH FIVE DIFFERENT RADIATOR INSTALLATIONS

at the ceiling have little bearing upon the comfort of occupants, but they do result in higher heat losses from the structure, particularly in the case of rooms having ceilings exposed to the outdoors or to unheated attic spaces.

The temperature gradients shown in Fig. 18 were obtained for days when the average outdoor temperature was approximately 30 deg. F. These curves are also characteristic of those obtained for colder weather, but in the later case the gradients were steeper and the differences in the temperatures produced by the various installations were accentuated. At an outdoor temperature of 15 deg. F. the temperatures at the floor were approximately 1 deg. F. lower and the temperatures at the ceiling were from 1 to 1.5 deg. F. higher than those shown in Fig. 18.

23. *Air Movement.*—From the consideration of temperature gradients, it would appear that just as good operating results may be obtained from a radiator located along an inside wall as were obtained from one located in any other position except under a window. However, since both dry-bulb temperature and air movement are factors influencing comfort conditions, the use of air temperature gradients alone as a basis for comparing the performance of different radiator installations in a given room may be misleading. Smoke studies, therefore, were made in the living room to determine the amount and direction of the air movement resulting from the use of each of the five radiator installations. Results are shown by the diagrams in Fig. 19. These diagrams show the locations in which high or critical air velocities occurred, and the numbers indicate the magnitudes of the velocities in ft. per min.

When the radiators were located at A or B, along the exposed wall, cool air from the vestibule and stairway entered the room and the main current moved in a large arc through the middle of the room towards the unexposed wall and archway between the living room and dining room. The maximum velocity was approximately 25 ft. per min. When the two radiators were located in the positions shown as A, up currents of heated air from the radiators intercepted any down currents of cold air from the windows and thus prevented cold drafts across the floor. At the same time the cold air coming down over the windows mixed with the heated air rising from the radiators and thus lowered the temperature of the air traveling towards the ceiling. As a result the temperature of the air at the ceiling was moderated. Since even in very cold weather, the temperature of the inside surface of exposed walls was only about 3 deg. F. lower than that of the air in the room there was no flow of cold air down these walls and across the floor.

When the two small radiators, located under the windows at A, were replaced by a single radiator located between the windows at B, the air currents across the floor of the room remained practically the same, except for a slight movement of cool air from the windows toward the center of the room. The greater part of the cool air from the windows, however, was drawn directly into the radiator. In the absence of a window or other cold surface above the radiator the heated air rose to the ceiling, resulting in comparatively high temperature, as shown in Fig. 18. It then settled uniformly throughout the room. No objectionable drafts were observed at any time during the operation of the heating system with the radiators located at either A or B, along the exposed wall.

A marked increase in the movement of air over the floor was observed as soon as the radiator was placed along the inside wall at position C. In this case the main air current originating in the vestibule and at the stairway was augmented by cold air from the windows and exposed walls. Air velocities across the floor as high as 35 ft. per min. were observed. Under these conditions, cold drafts were noticeable around the ankles and the room was less comfortable than it was when the radiators were located along the exposed wall, in spite of the fact that the temperature of the air at the floor remained unchanged at 69 deg. F.

Increasing the velocity of 69 deg. F. air from 20 ft. per min. to 35 ft. per min. lowers the effective temperature* by approximately one-half degree. With a given air movement this is equivalent to lowering the dry-bulb temperature 1.5 deg. F., or from 69 deg. F. to 67.5 deg. F. Such a change is of sufficient magnitude as to be distinctly noticeable, particularly when occurring near the lower limit of comfort.

On studying the air currents in vertical sections of the room it was observed that the maximum air currents all occurred within a distance of one foot from the walls, floor, and ceiling. Thus, from the standpoint of comfort, the most significant air movement was that which took place across the floor.

VIII. Performance Characteristics of House

24. *Wall Temperature Gradients.*—By means of thermocouples located in the walls, as shown in section "B-B", Fig. 3, it was possible to obtain the temperature gradient through the north wall on a

*For a discussion of the implications of effective temperature see Chapter 2 of the American Society of Heating and Ventilating Engineers Guide for 1942.

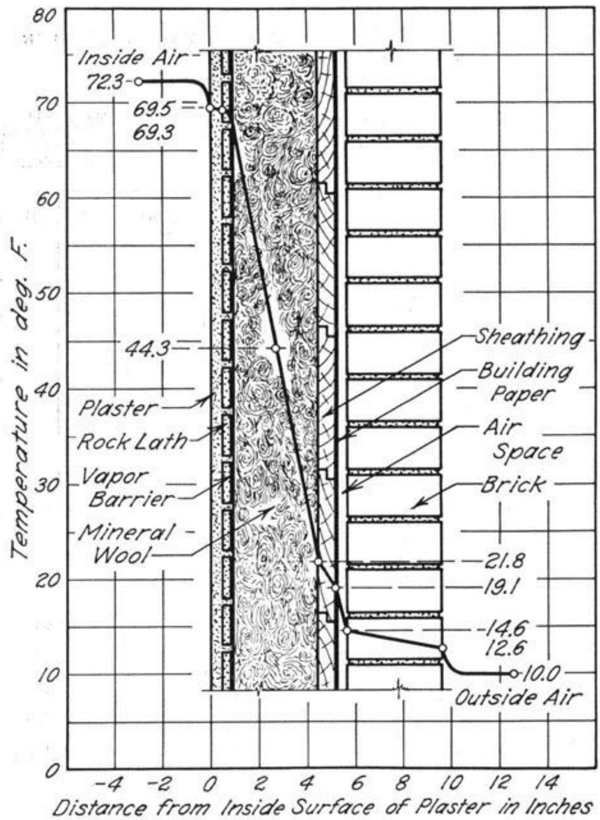

Fig. 20. Typical Temperature Gradient Through North Wall

day when the outdoor temperature averaged 10 deg. F. On the assumption that a condition of steady flow of heat through the wall exists in this case, an assumption consistent with those made in the calculation of heat losses from buildings, Fig. 20 offers a means of comparing the actual insulating value of any of the building materials used in the wall with the theoretical value. The insulating value of any given construction may be expressed in terms of thickness of an hypothetical characteristic insulation having the same conductivity as that of combined mineral wool and wood studs that would be required to have a resistance to heat flow equal to that of the given construction. If the observed temperature drops are accepted as being representative of the actual conductances of the corresponding constructions, and the drop through the 3⅝-in. combined mineral wool and studs is used as a base, then the insulating values computed from the ratios of the

TABLE 4
EQUIVALENT INSULATING VALUES OF MATERIALS AS USED IN WALLS IN RESEARCH HOME

Arrangement of Materials as Used in Walls	Conductance for Construction Stated in Col. 1 C	Observed Temperature Drops Through Construction deg. F. ΔT	Thickness of Characteristic Insulation* Equivalent to Construction in Col. 1 in inches	
			Based on C	Based on ΔT
1	2	3	4	5
4-in. brick	2.30	2.0	0.141	0.153
½-in. air space†	1.10	4.5	0.295	0.345
25/32-in. wood sheathing	1.02	2.7	0.318	0.207
3⅝-in. combined mineral wool and wood studs	0.089‡	47.3	3.625	3.625
¾-in. rock lath and plaster	4.40	0.4	0.074	0.031
9.66-in. wall as constructed	0.079	56.9	4.114	4.360

*Characteristic insulation assumed as one having a conductivity of 0.325 conforming with conductivity of combined mineral wool and wood studs.
†Not included in computed conductance for wall as constructed.
‡Conductivity of combined mineral wool and wood studs = 0.325.

different temperature drops to the base drop may be defined as actual insulating values. Those computed from the ratios of the commonly-accepted nominal values of the conductances may be defined as theoretical insulating values. Table 4 shows equivalent insulating values computed by both methods. Column 4 gives the insulating values which would be equivalent to each of the constructions listed in Column 1. These values are based on nominal conductance values, C, given in Column 2. Column 5 gives the actual thickness of the characteristic insulation equivalent to the same constructions, based on the observed temperature drops given in Column 3. The actual and theoretical thicknesses were in close agreement for the brick, the insulated studding space, and the total wall. Some disagreement was exhibited in the cases of the air space, the sheathing, and the rock lath and plaster. These discrepancies can probably be accounted for by deviations in the actual physical dimensions and moisture contents from those assumed for the computations. Following common practice, no air space between the brick and sheathing was allowed for in the computation of the overall conductance for the wall. However, the data indicate that in the actual wall sufficient benefit was derived from the space between the brick and the sheathing to offset the deficiencies shown in the cases of the sheathing and the lath and plaster, so that the actual and the theoretical insulating values for the total wall were in close agreement.

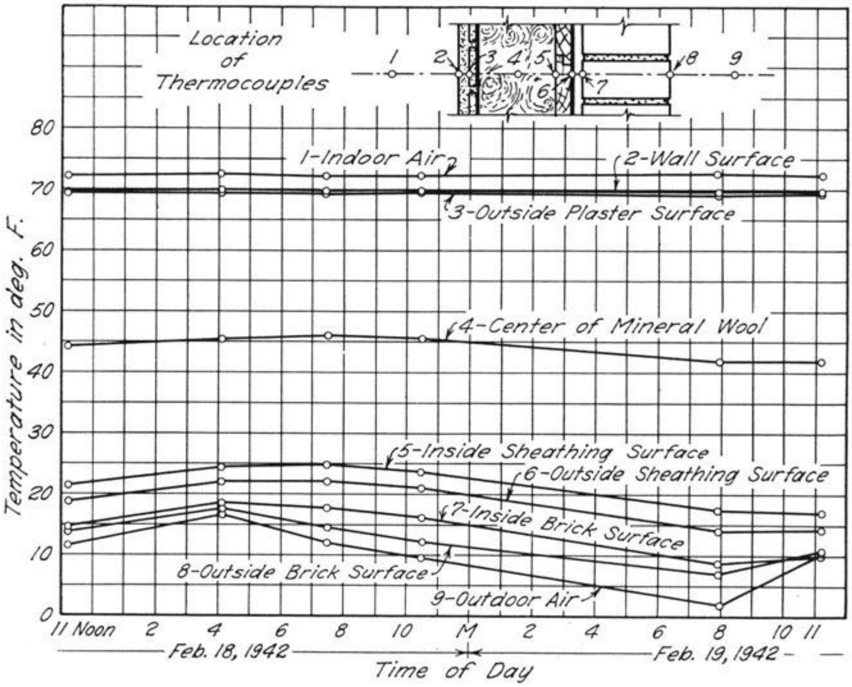

FIG. 21. TEMPERATURE VARIATIONS IN NORTH WALL DURING A 24-HOUR TEST PERIOD

Figure 21 shows the variations over a period of a day in the temperatures observed in a cross-section of the north wall of the house. Over the period of a daily cycle the temperature of inside surfaces remained practically constant irrespective of the changes occurring in the outdoor temperature and in the temperature of the outer layers of the wall.

Figure 22 shows the observed temperatures of the air at the 60-in. level in the living room, and of the inside surface of the north exposed wall at the same level, plotted against the indoor-outdoor temperature difference. Curves showing calculated temperatures of the inside surfaces of the actual insulated wall and a similar wall without insulation are also included. It may be observed that the actual temperature of the inside surface of the exposed wall decreased somewhat as the outdoor temperature decreased, and that at an indoor-outdoor temperature difference of 72 deg. F., corresponding to an average outdoor temperature of zero, it was only 3.8 deg. F. lower than that of the air in the room. Furthermore, a very close agreement was obtained between the observed and calculated temperatures of the inside surface

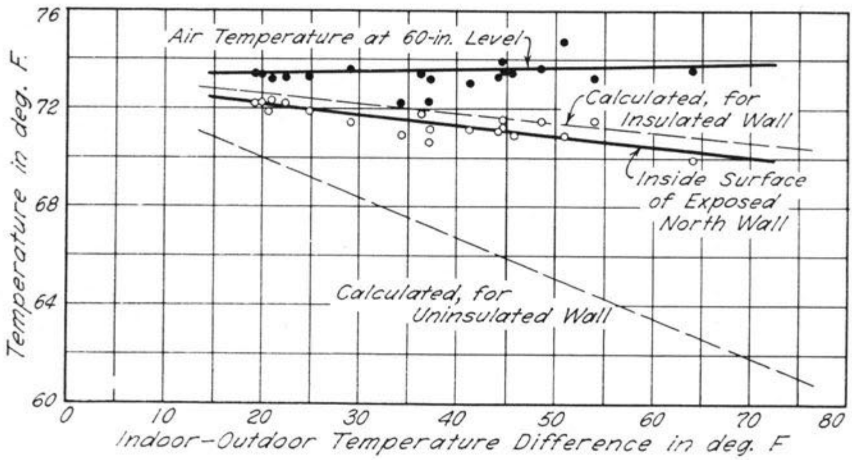

FIG. 22. AIR AND INSIDE WALL SURFACE TEMPERATURES IN LIVING ROOM

of the actual insulated wall. The curve of calculated temperatures for the uninsulated wall shows the temperatures that might have been expected if the insulation had been omitted from the walls of the Research Home. Under these conditions, the probable temperature of the inside surface in zero weather would have been 61.5 deg. F., or 12.2 deg. F. below the temperature of the air in the room. That is, the presence of the insulation apparently increased the temperature of the inside surface of the exposed wall an amount of the order of 8.5 deg. F. From the standpoint of comfort, the higher temperature of the inside surface of the insulated wall is of considerable significance in that it reduced the amount of heat lost from the body by radiation. Furthermore, it tended to minimize the floor-to-ceiling temperature difference, and thus minimize any tendency for cold floors brought about by downward currents of cold air resulting from the presence of cold wall surfaces.

25. *Heat Transmission Through Radiator Recesses.*—Throughout both heating seasons the temperatures of the inside and outside surfaces of the sheathing directly back of the radiators in the dining room and the S.W. bedroom were observed. These records are of interest in that they show how the use of insulation in the radiator recesses may affect the heat loss from the house. The details of each of the wall sections used back of the radiator in these two rooms are given in Table 5. In Fig. 23 the observed temperature drops through the sheathing for each of the wall sections described in Table 5 are plotted against the indoor-outdoor temperature differences. The method of

TABLE 5

NORMAL WALL AND RADIATOR RECESS CONSTRUCTION

Normal Wall Section	Wall Section Back of Radiator				
	Recessed Radiator				Free Standing Radiator
	Dining Room		South West Bedroom		Dining Room
	Number 1	Number 2	Number 3	Number 4	Number 5
Lath and plaster					Lath and plaster equivalent
Vapor barrier					
$3\frac{5}{8}$-in. mineral wool insulation					$3\frac{5}{8}$-in. mineral wool insulation
		Reflective insulation		Reflective insulation	
$\frac{7}{8}$-in. wood sheathing	$\frac{7}{8}$-in. wood sheathing	$\frac{7}{8}$-in. wood sheathing	$\frac{7}{8}$-in. wood sheathing	$\frac{7}{8}$-in. wood sheathing	$\frac{7}{8}$-in. wood sheathing
			$\frac{25}{32}$-in. rigid insulation	$\frac{25}{32}$-in. rigid insulation	
15-lb. felt building paper	15-lb. felt building paper	15-lb. felt building paper	15-lb. felt building paper	15-lb. felt building paper	15-lb. felt building paper
$\frac{1}{2}$-in. air space	1-in. air space	1-in. air space	$\frac{1}{4}$-in. air space	$\frac{1}{4}$-in. air space	1-in. air space
Full brick	Full brick	Full brick	Full brick	Full brick	Full brick

operating the heating plant had no apparent effect on these temperature drops. Since the conductance of the sheathing was the same in each case, the ratio of the temperature drops through the sheathing for any two sections is also representative of the ratio of the actual heat transmission through the sections. Therefore, for any given indoor-outdoor temperature difference, if the temperature drop through the sheathing for any given recess is divided by the temperature drop through the sheathing for the normal wall, a value which may be designated as the actual heat transmission ratio for that recess is obtained. This figure is also equivalent to the square feet of normal wall required to transmit the same quantity of heat per hour as is transmitted by one square foot of recess area. These actual heat transmission ratios have been calculated for each recess, and are shown as full-line curves in Fig. 24. The broken-line curves represent the ratio of the calculated overall heat transmission coefficient for each recess or wall section to that for the normal wall. The discrepancies between values read from the full-line curves and those read from the cor-

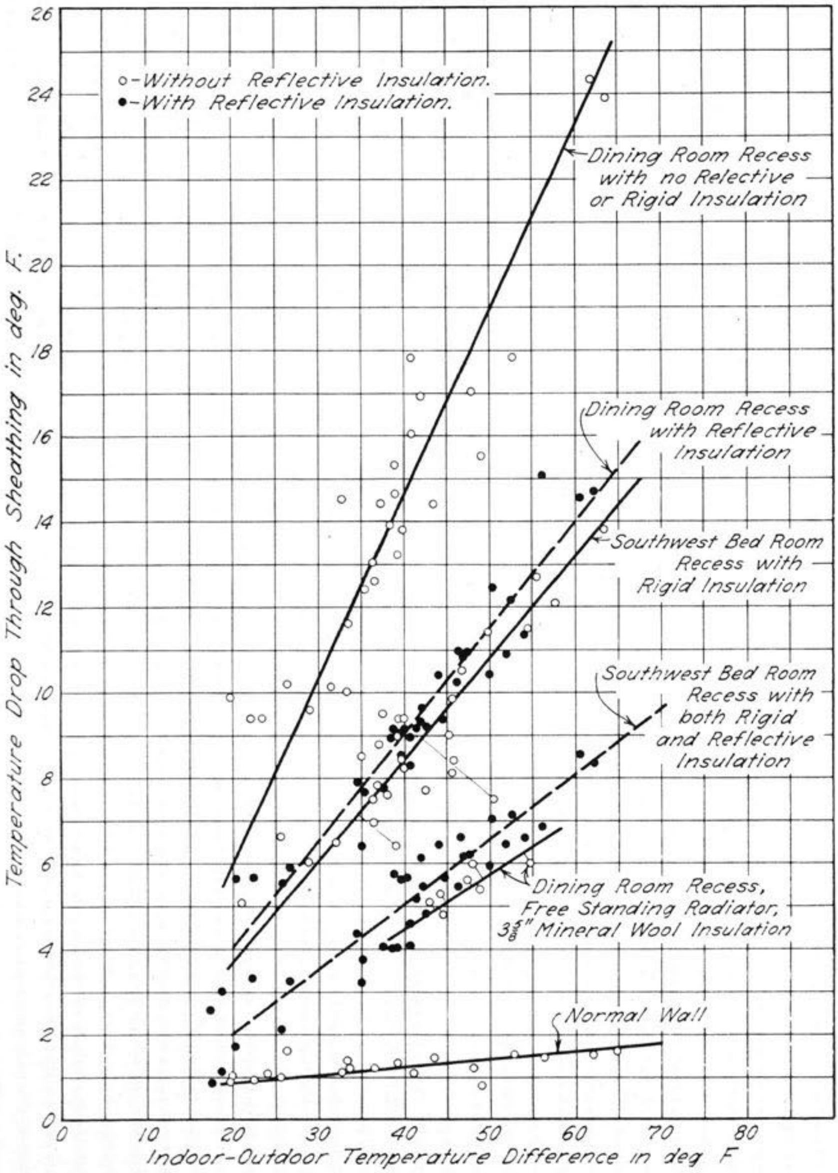

Fig. 23. Temperature Drop Through Sheathing in Radiator Recess and Normal Wall

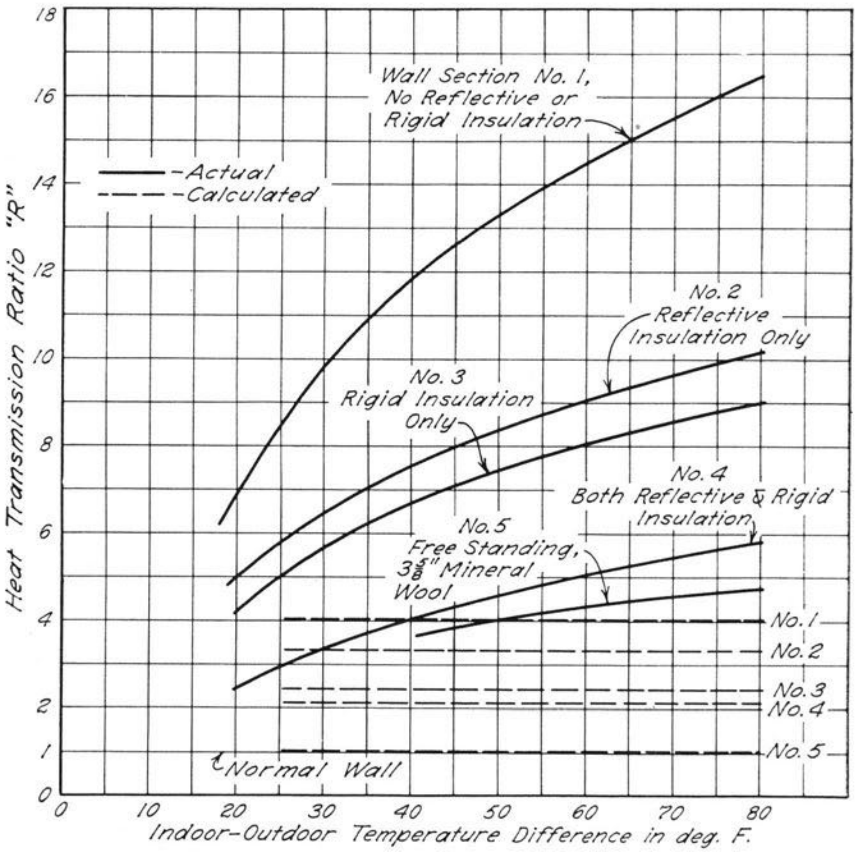

FIG. 24. HEAT TRANSMISSION RATES FOR RADIATOR RECESSES COMPARED WITH THAT FOR NORMAL WALL

responding broken-line curves at given indoor-outdoor temperature differences result from the fact that the temperature back of the radiators was higher than that of the air in the room. The increase in the actual heat transmission ratios as the indoor-outdoor temperature difference increased was undoubtedly brought about by the fact that the higher average radiator temperatures required as the weather became colder resulted in greater differences between the temperatures back of the radiators and that in the room.

It is evident from Fig. 24 that considerably more heat was lost through an uninsulated recess than through a wall back of a free standing radiator and that a large portion of this heat was saved by adequate insulation of the recess. It is also of interest to observe that reflective insulation on the surface of the recess was practically

TABLE 6

HEAT LOSS INCREASES RESULTING FROM RECESSES

Method of Installing Radiator	Overall Heat Transmission Coefficient B.t.u. per hr. per sq. ft. per deg. F.	Actual Heat Transmission Ratio*		Increase in Heat Loss From House B.t.u. per Hour		Sections of Radiation Equivalent to Increase in Heat Loss Shown in Column 5	Increase in Heat Loss From House in Per Cent of Calculated Heat Loss†	
		80 deg. F. indoor-outdoor temp. diff.	34 deg. F. indoor-outdoor temp. diff.	80 deg. F. indoor-outdoor temp. diff.	34 deg. F. indoor-outdoor temp. diff.		80 deg. F. indoor-outdoor temp. diff.	34 deg. F. indoor-outdoor temp. diff.
1	2	3	4	5	6	7	8	9
Normal Wall	0.074	1	1	0.0	0.0	0	0.0	0.0
Free standing (No. 5)	0.074	4.8	3.3	1195	308	4	2.8	1.68
Reflective and 25/32-in. rigid insulation in recess (No. 4)	0.161	5.8	3.6	1510	349	5	3.5	1.90
25/32-in. rigid insulation in recess (No. 3)	0.182	9.0	6.1	2520	685	9	5.8	3.72
Reflective insulation in recess (No. 2)	0.245	10.2	6.9	2895	791	10	6.7	4.30
No insulation in recess (No. 1)	0.299	16.5	10.7	4880	1300	17	11.3	7.08

*Square feet of normal wall equivalent to construction shown in Col. 1; values taken from Fig. 24.
†Calculated heat loss of house at 80 deg. F. indoor-outdoor temperature difference = 43 370 B.t.u. per hr.
Calculated heat loss of house at 34 deg. F. indoor-outdoor temperature difference = 18 400 B.t.u. per hr.

equivalent to $^{25}/_{32}$ in. of rigid insulation, and furthermore, that the heat lost through a recess insulated with a combination of the reflective and the rigid insulation was only slightly greater than that lost through the fully insulated brick veneer wall back of a free standing radiator.

Certain values read from the curves in Fig. 24 are of particular significance from two standpoints. On one hand, since an indoor-outdoor temperature difference of 80 deg. F. corresponds to the design temperature, values read at this temperature difference are directly related to the amount of radiation and the size of the boiler required. On the other hand, since an indoor-outdoor temperature difference of 34 deg. F. represents average winter weather in Urbana, Illinois, values read at this temperature difference can be correlated with the seasonal fuel consumption, and hence have a bearing on seasonal operating costs.

In the original calculations for the heat loss from the house the 53.25 sq. ft. represented by the recesses were considered as normal wall and no additional allowance was made for individual peculiarities in the construction of the recesses themselves nor for the effect of the presence of the radiator. Columns 5 and 6 in Table 6 show the increases in total heat loss that would have occurred at indoor-outdoor temperature differences of 80 deg. F. and 34 deg. F., respectively, due to the presence of the radiator and the type of wall construction, on the assumption that in each case all of the radiators used were installed in accordance with the designations in Column 1. The values in Columns 3 and 4 were read directly from the full-line curves in Fig. 24 at indoor-outdoor temperature differences of 80 deg. F. and 34 deg. F. The values in Columns 5 and 6 were then computed from the following formula:

$$h = AUt_d\ (R - 1)$$

in which

h = increase in heat loss from house, B.t.u. per hr.
A = total area of recesses, (53.25) sq. ft.
U = overall heat transmission coefficient for normal wall, (0.074) B.t.u. per hr. per sq. ft. per deg. F.
t_d = indoor-outdoor temperature difference, deg. F.
R = actual heat transmission ratio for the given recess, at the temperature difference t_d.

Column 7 shows the number of sections of 19-in., 4-tube, small-tube type radiation which are equivalent to the additional heat losses given in Column 5. It may be observed that, with recesses of the types designated in Column 1, additional allowances of from 4 to 17 sections would have to be made in the installed radiation in order to compensate for the losses inherent in these recesses. Column 8 shows that percentage increases of from 2.8 to 11.3 would have to be made in the amount of radiation installed. The increase in the size of the boiler would be even greater because of the additional piping and pick-up allowances. The percentages given in Column 7, at an indoor-outdoor temperature difference of 34 deg. F., are roughly correlated with the probable increases in seasonal fuel consumption, and show that due to the presence of the radiator and the type of wall construction, increases ranging from 1.68 to 7.08 per cent might be expected in operating costs with the different types of recesses.

The ratio of the actual heat transmission through a recess or a wall back of a radiator to that through the normal wall, or the wall not influenced by the presence of a radiator, is generally assumed to be numerically equal to the ratio of the calculated heat transmission coefficient of the recess or wall back of the radiator to that of the normal wall. This, however, is true only if the actual conditions of heat flow and the actual details of the constructions exactly conform to those assumed in the calculations for the heat transmission coefficients. In making such calculations it is commonly assumed (1) that steady flow actually exists, (2) that the air close to the wall has the same temperature as that in the main body of the room, and (3) that the inside surface of the wall is not subjected to radiation from any sources at a temperature higher than that of the air in the room. In the case of recesses, and walls back of radiators, subjected to cyclic action of both indoor and outdoor temperatures, none of these conditions is exactly satisfied. The heat flow is not steady, the temperature of the air close to the surface of the wall is not the same as that in the main body of the room, and the inside film coefficient is directly influenced by the close proximity of the hot radiator.

In order to derive a general relation between the actual heat transmission and the calculated values of the heat transmission coefficients, U, for recesses and walls back of radiators the actual heat transmission ratios have been plotted against the ratios of coefficients of the recesses, U_r, to the coefficients of the normal wall, U_w, as shown in Fig. 25. These curves are probably sufficiently general to permit the prediction of the approximate rate of heat transmission at the design indoor-outdoor temperature difference of 80 deg. F. for different types of recesses located in any given wall, and with mean water temperatures of approximately 170 deg. F. in the radiators. It may be observed

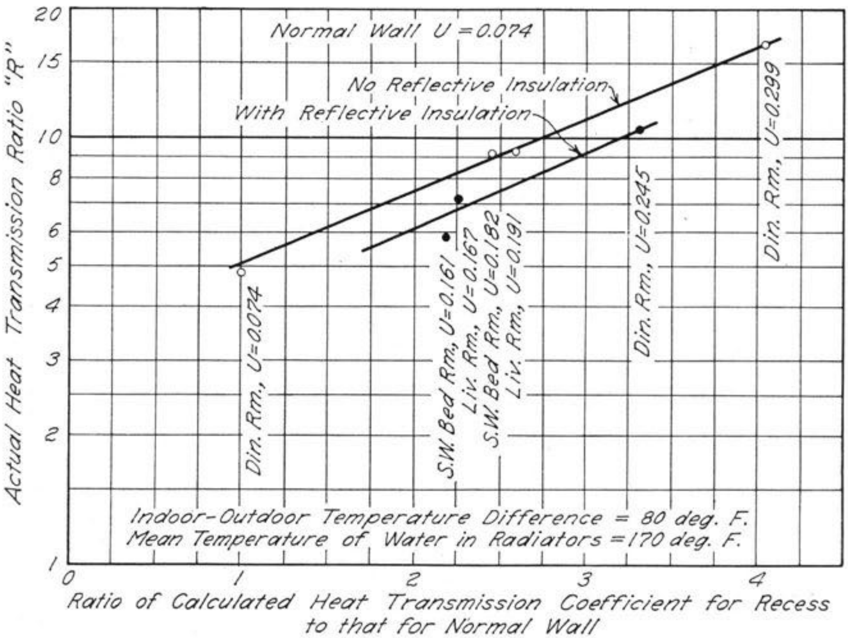

FIG. 25. RELATION BETWEEN ACTUAL HEAT TRANSMISSION RATES AND CALCULATED OVERALL HEAT TRANSMISSION COEFFICIENTS

that the relation between the actual rate of heat transmission and the calculated value of U was influenced by the nature of the surface directly exposed to the heat from the radiator. When this surface was covered with aluminum foil the actual rate of heat transmission with a given value of U was approximately 18 per cent less than that obtained when aluminum foil was not used. Apparently the close proximity of the hot radiator had a less disturbing effect on the accepted values of the inside film coefficient in the case of a reflective surface than it did in the case of a non-reflective surface. Furthermore, it should be emphasized that the actual rates of heat transmission through walls back of radiators are much greater than the calculated rates obtained by using an indoor-outdoor temperature difference in connection with the commonly-accepted values of the overall heat transmission coefficients.

26. *Heat Utilization and Overall House Efficiency.*—When a central heating system is used in connection with an inside chimney, as was done in the case of the Research Home, the only part of the total heat input to the house that is not actually utilized in offsetting heat

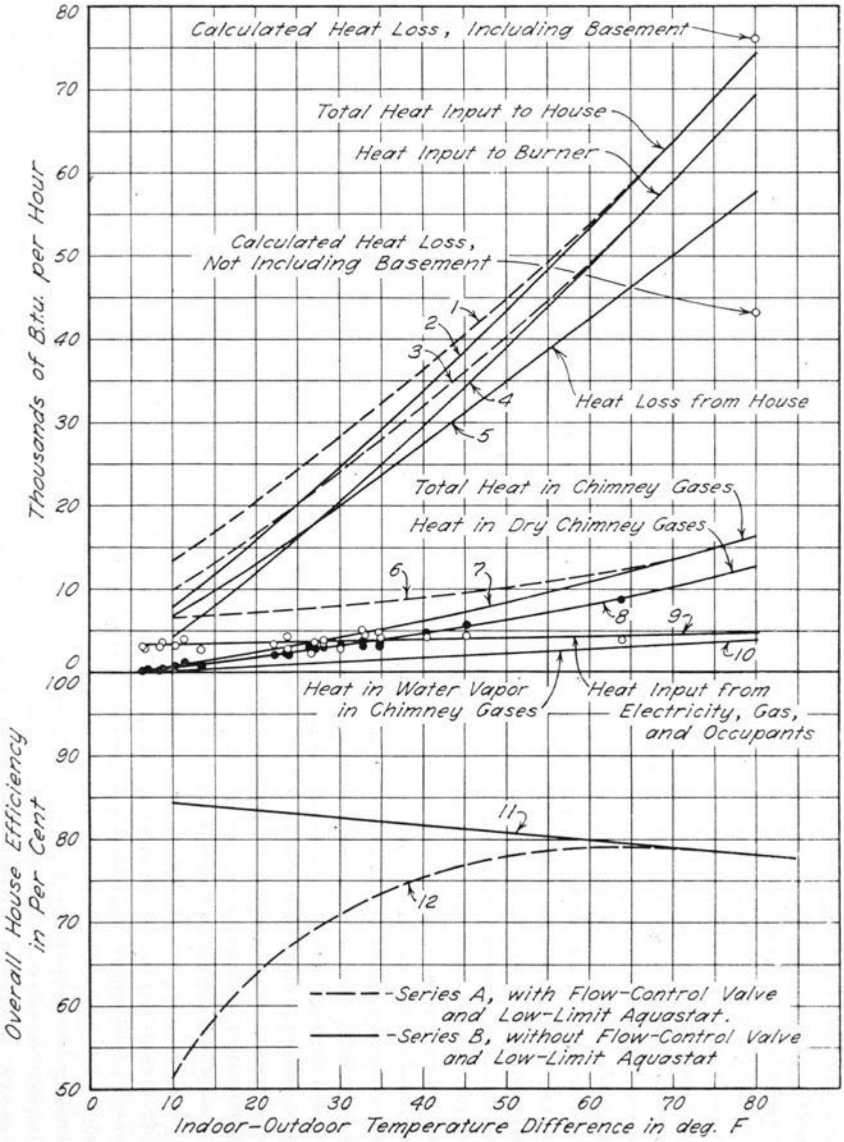

FIG. 26. HEAT UTILIZATION AND OVERALL HOUSE EFFICIENCY

losses is the heat lost in the chimney gases at the level of the upper boundary of the heated space. This boundary usually consists of the attic floor. One hundred times the ratio of total input minus the heat appearing in the chimney gases at the level of the attic floor to the total heat input has been defined as the "overall house efficiency." This overall house efficiency is an index of the effectiveness of heat utilization, and serves as a convenient basis in making comparisons between different heating installations.

In order to determine the total heat input to the house daily records were made of the pounds of oil used by the burner, the kilowatt hours of electricity used in lights and power, the cubic feet of gas burned in an auxiliary water heater and the man-hours of occupancy. The heat equivalents used for all of these sources of energy were as follows:

Oil—19 550 B.t.u. per lb.
Electricity—3415 B.t.u. per kilowatt hr.
Gas—1000 B.t.u. per cu. ft.
People—500 B.t.u. per man hr. of occupancy.

It was assumed that all of the heat represented by the oil burned, and the electricity used, and that supplied by the occupants was available as useful heat. The only appliance using gas was an auxiliary water heater. This heater was connected to the outside chimney, and it was estimated that 25 per cent of the total heat equivalent of the gas used was lost in the flue gases. Hence, only 75 per cent of the heat equivalent of the gas burned was considered available as useful heat. The assumption relative to the availability of the heat generated by the gas burned and the selection of 500 B.t.u. per man hr. as representative of the degree of activity of the occupants are somewhat open to question. However, the total amount of heat supplied by both of these sources was relatively small, and even if deviations of as much as 50 per cent from the assumed values occurred they would not cause variations outside of the normal limits of experimental error in the total amount of heat supplied.

The sum of the amounts of heat supplied to the house by electricity, gas, and occupants is shown as curve No. 9 in Fig. 26. Curves Nos. 3 and 4 were derived from the fuel consumption curves in Fig. 6, and show the heat inputs to the burner for series A and B respectively. The total heat inputs to the house when operating on series A and B were obtained by adding values read from curve No. 9 at given indoor-outdoor temperature differences to corresponding values read from curves Nos. 3 and 4, respectively, and are shown as curves Nos. 1 and 2.

Under any condition the actual heat loss from the house is equal to the difference between the total heat input and the heat lost in the chimney gases at the level of the attic floor. Since the actual heat loss from the house is a function of only the indoor-outdoor temperature difference and the type of construction, it is independent of the type and method of operation of the heating system. Hence, a single curve showing the relation between the actual heat loss from the house and the indoor-outdoor temperature difference constitutes the characteristic curve for the house. In order to establish such a curve it is not necessary to use the results from more than one series of tests covering one method of operation. For this purpose series B was selected as being most adaptable to analysis.

The heat loss in the chimney gases during the on-period consisted of the heat lost in the dry flue gases and that lost in water vapor carried in by air and water vapor formed by the combustion of the hydrogen in the fuel. During the off-period the loss consisted only of the heat lost in the dry flue gases and in water vapor carried in by air. The hourly heat losses in the dry gases and in water vapor for series B are shown by curves Nos. 8 and 10, respectively, in Fig. 26, and the sum, or the total hourly heat loss* in the chimney gases, is shown by curve No. 7. In each case the curve represents the total of the hourly losses for both the on- and the off-periods. Curve No. 5 shows the actual heat loss from the house, and represents the difference between curves Nos. 7 and 2.

For series B, the overall house efficiency, or the ratio of the actual heat loss from the house to the total heat input, as calculated from curves Nos. 5 and 2, is shown by curve No. 11. Inasmuch as the actual heat loss from the house remains the same for a given indoor-outdoor temperature difference irrespective of the method of operating the heating system, the overall house efficiency for any given method of operation can be calculated from curve No. 5 and the curve showing the total heat input to the house for that particular method of operation. The overall house efficiency for series A, calculated from curves Nos. 5 and 1, is shown by curve No. 12.

It may be observed, as mentioned in Section 13, that the overall house efficiency was lower in the case of series A than it was in the case of series B. At the same time the chimney losses shown by curve No. 6, obtained by difference from curves Nos. 5 and 1, were greater for series A than they were for series B.

The calculated heat losses from the house, both including and excluding the basement, are shown as isolated points in Fig. 26. It may

*A complete discussion of the method of computing the total chimney loss and curves from which those of Fig. 26 were derived are presented in the Appendix.

be observed from curve No. 5 that at the design indoor-outdoor temperature difference of 80 deg. F. the actual heat loss from the house, including the basement, was 57 500 B.t.u. per hr. as compared with the calculated heat loss of 43 370 B.t.u. per hr. for the first and second stories alone, and 76 180 B.t.u. per hr. for the house as a whole, including basement.

The amount of radiation installed in the house was just sufficient to offset the calculated heat loss of 43 370 B.t.u. per hr. predicated on continuous operation of the radiators when supplied with water entering at 200 deg. F. and leaving at 190 deg. F., and surrounded by air at 70 deg. F. In Section 17 it was shown that at the design indoor-outdoor temperature difference of 80 deg. F. the circulator did not operate more than approximately 12 hours per day, and, on the average, the maximum temperature of the water entering the radiators did not exceed 175 deg. F. Under these conditions, the heat supplied to the house by the radiators was probably less than 21 000 B.t.u. per hr., or approximately one-half of the calculated 43 370. The remainder of the actual heat loss of 57 750 B.t.u. per hr., or 36 750 B.t.u. per hr., was obviously supplied from other sources. About 5000 B.t.u. per hr. of this amount of heat was supplied by gas, electricity, and occupants, and the balance undoubtedly was furnished by the boiler jacket, the smoke pipe and the chimney.

If it is conceded that the calculated 43 370 B.t.u. per hr. is approximately correct for the heat loss from the structure above the basement, then the difference between this and the actual loss of 57 750 or 14 380 B.t.u. per hr. represents heat that was lost from the basement itself. The difference between 76 180 and 43 370 or 32 810 B.t.u. per hr. represents the calculated heat loss from the basement based on an assumed ground temperature of 50 deg. F. Obviously the commonly-accepted methods of calculation give values that are much too high for the heat losses from basements. The preponderance of the evidence indicates that most of the error in the commonly-accepted methods occurs in the calculation of the heat loss in the portion in contact with the ground. Of the total calculated heat loss of 32 810 B.t.u. per hr. the loss through the portion in contact with the ground amounted to 22 435 B.t.u. per hr., and the balance amounted to 10 375 B.t.u. per hr. Assuming that the latter is approximately correct, the difference between 14 380 and 10 375, or 4 005 B.t.u. per hr., probably represents the actual heat loss through the floor and the portion of the walls in contact with the ground. This corresponds to a heat loss of 3.21 B.t.u. per sq. ft. per hr. as compared with a value of 17.9 cor-

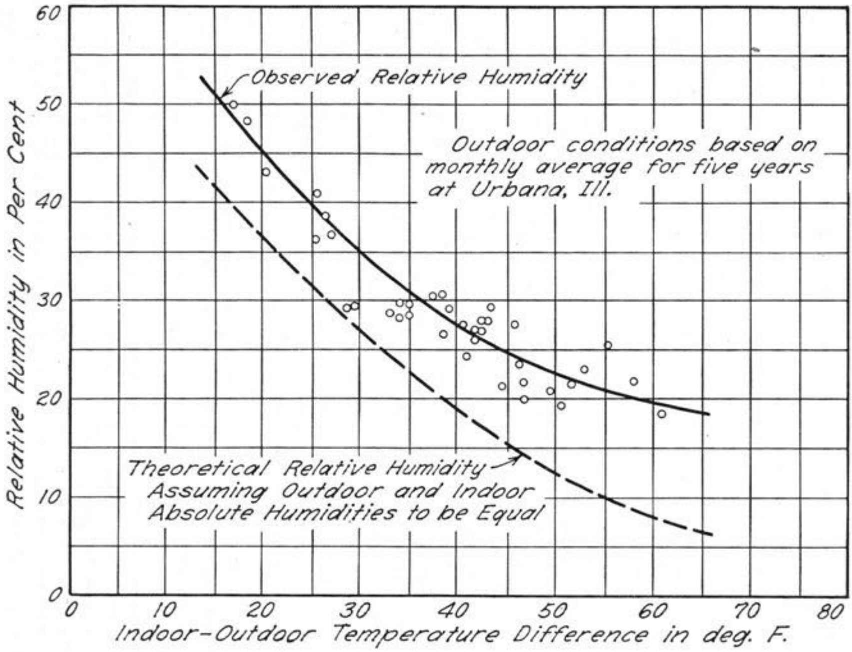

FIG. 27. RELATIVE HUMIDITY IN I=B=R RESEARCH HOME WITH NO HUMIDIFICATION

responding to the 22 435 B.t.u. per hr. obtained by the usual methods of calculation.

27. *Humidity Conditions.*—Humidity observations were made in the Research Home during the seasons of 1940-41 and 1941-42 although no humidifier was used and no attempt was made to control the indoor relative humidity. The full-line curve in Fig. 27 shows the observed indoor relative humidity and the broken-line curve shows the relative humidity which would have existed in the house if the absolute humidity indoors had been the same as that outdoors. The difference between these curves represents moisture added to the indoor air by occupants, by evaporation from building materials and house furnishings, and by evaporation from water normally used for general purposes in the house. It may be observed from the full-line curves that at indoor-outdoor temperature differences above 60 deg. F. the indoor relative humidity was below 20 per cent, and hence too low for comfort. No family cooking nor washing was done in the house. If these processes had been carried on it is probable that the indoor relative humidity would have been raised to 25 per cent or more even in the

coldest weather without any further form of humidification. This amount is sufficient for reasonable comfort, and is about as high as can be maintained in a building without storm sash without excessive condensation occurring on the windows.

Appendix A

METHOD OF COMPUTING CHIMNEY GAS LOSSES

1. *General Statement.*—The hourly loss in the chimney gases at the upper boundary of the heated space, that is at the level of the attic floor, may be divided into three parts consisting of, (1) the heat lost in the dry gases, (2) the heat lost in water vapor formed by the combustion of hydrogen in the fuel and (3) the heat lost in water vapor carried in by the air for combustion.

The heat lost in the dry gases during any given period of time may be computed from the equation:

$$H = W\,C_p\,(t_c - t_i)\,\Phi \tag{1}$$

in which

H = loss in dry chimney gas, B.t.u.
W = average rate of flow of dry chimney gas, lb. per hr.
C_p = specific heat of chimney gas, 0.24 B.t.u. per lb.
t_c = average temperature of the chimney gas at the level of the attic floor, deg. F.
t_i = average temperature of the indoor air, deg. F.
Φ = time, hours.

The heat lost in the water vapor formed by the combustion of hydrogen in the fuel during any given period of time may be computed from the equation:

$$H_c = 0.09 W_o \mathrm{H}_2\,(h_c - h_{fi})\,\Phi \tag{2}$$

in which

H_c = loss in water vapor formed by the combustion of hydrogen in the fuel, B.t.u.
W_o = weight of oil burned, lb. per hr.
H_2 = hydrogen in the fuel, per cent by weight
h_c = enthalpy of superheated steam at temperature t_c and the partial pressure of the vapor in the chimney (approximately 1.0 lb. per sq. in. absolute), B.t.u. per lb.
h_{fi} = enthalpy of liquid at temperature t_i, B.t.u. per lb.
Φ = time, hours.

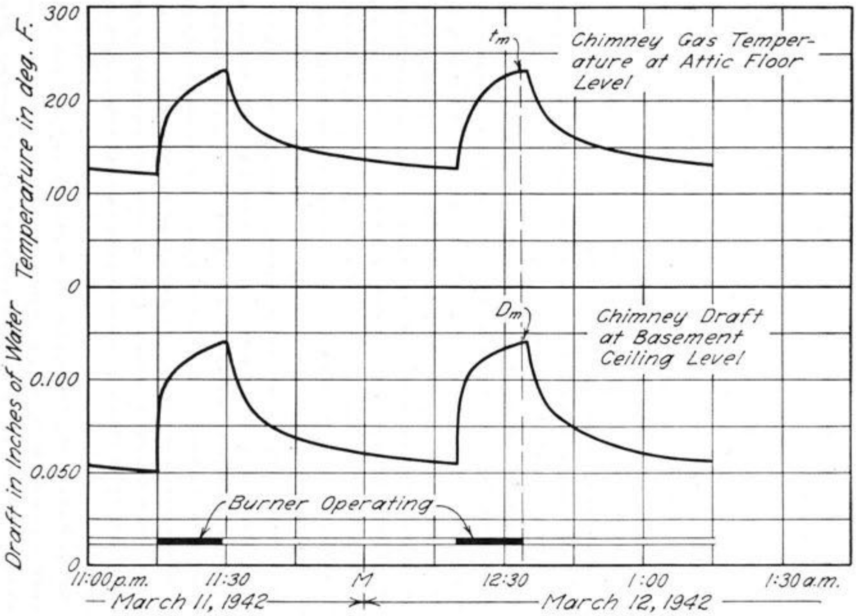

FIG. 28. CHARACTERISTIC CHIMNEY GAS TEMPERATURE AND DRAFT RECORDS

Each pound of air supplied for combustion contained only about 0.005 pounds of water vapor, and the loss from this source was very small as compared with the other losses, hence the heat lost in this water vapor was neglected in the calculations.

The rate of flow of chimney gases was not continuous and uniform, and it was not possible to calculate the rate of flow during the off-periods from the analyses of the fuel and chimney gases as was possible for the on-periods. Hence, it was immediately evident that in order to obtain the chimney losses over the period of a day the on-periods and the off-periods would have to be treated separately.

Continuous records were made of the available draft in the chimney at the level of the basement ceiling, and the temperature of the chimney gases at the level of the attic floor. The characteristic shapes of the temperature and draft records are shown in Fig. 28. The shapes of these curves for both the on- and the off-periods were such that no direct method of averaging could be used, however, by integrating a large number of characteristic curves, it was found that the average temperature and average draft for a given on- or off-period could be expressed in terms of the maximum temperature or draft shown by the recorder chart for that period. Furthermore, since it was found that the variation in the rate of flow of the chimney gases was similar to the variation in the draft, the average rate of flow could be correlated with the average draft. These methods were therefore used to obtain the separate averages for the on-periods and off-periods from which the heat losses in the dry chimney gases for these periods were computed. The total loss for the day was then obtained by summing the heat losses computed for the individual cycles.

2. *Average Temperature and Draft During On-Periods.*—For a large number of characteristic on-periods the temperatures of the chimney gases at the level of the attic floor as read from the recorder charts were plotted against time on logarithmic paper. The resulting curves were straight lines all having practically the same slope. The average slope was 0.14 and the characteristic equation was of the form:

$$t = C\varphi^{0.14} \tag{3}$$

in which

φ = time from start of on-period, minutes

t = instantaneous temperature of the chimney gases at the level of the attic floor at the time φ, deg. F.

C = a constant.

Integrating Equation (3) between the limits of zero and φ_2, and dividing by φ_2:

$$t_c = \frac{C\varphi_2^{0.14}}{1.14} \tag{4}$$

in which

t_c = average temperature of the chimney gases at the level of the attic floor, deg. F.

φ_2 = length of on-period, minutes.

From Equation (3) the temperature, t_m, at the end of the on-period is

$$t_m = C\varphi_2^{0.14}. \tag{5}$$

Dividing Equation (4) by Equation (5)

$$t_c = \frac{t_m}{1.14} = 0.877t_m\,. \tag{6}$$

Equation (6) gives the relation between the average temperature for the on-period and the temperature attained at the end of the period.

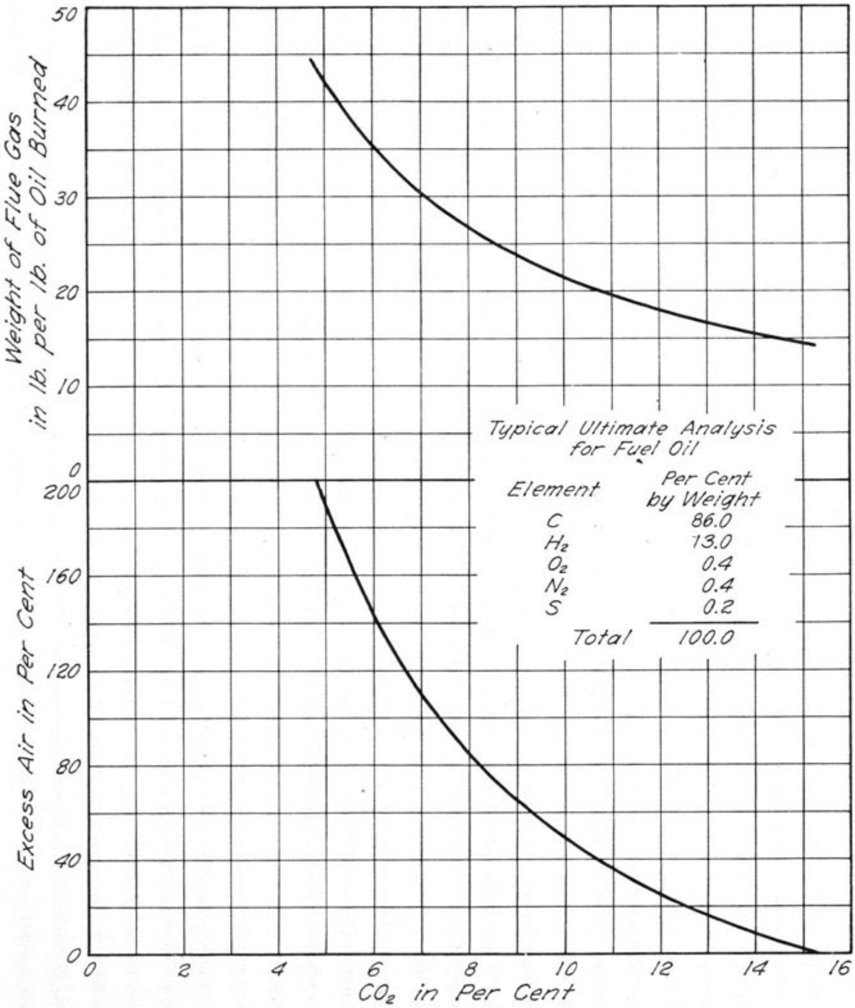

FIG. 29. DRY FLUE GAS PER POUND OF OIL BURNED

Since the latter was always the maximum temperature attained in the given cycle, it could be read from the chart with considerable accuracy.

By a similar analysis it was found that the average draft could be obtained from the recorder charts by making use of the equation:

$$D_c = 0.917 D_m \tag{7}$$

in which

D_c = average draft, in. of water
D_m = draft at end of on-period, in. of water.

3. *Average Rate of Flow During On-Periods.*—For a large number of on-periods, and over a wide range of chimney temperatures, a series of observations were made of the percentage of CO_2 in the chimney gases at the level of the attic floor and the draft in the chimney just above the point at which the smoke pipe entered the chimney. From these CO_2 readings, made by means of an Orsat apparatus, the corresponding rates of flow of the chimney gases in pounds per hour were computed from the curves shown in Fig. 29. These curves were obtained by assuming different percentages of excess air and calculating* the resulting CO_2 and weights of the products of combustion from the analysis of an oil typical of that used. From the weight rates of flow, the velocities of the gases in the chimney were computed and plotted on log paper against the corresponding observed drafts. The resulting curve was a straight line represented by the equation:

$$V = 1.71\sqrt{\frac{D}{\rho}} \tag{8}$$

in which

V = velocity, ft. per sec.
D = observed draft, in. of water
ρ = density of chimney gases at the level of the attic floor, lb. per cu. ft.

This was regarded as the characteristic flow equation applying to the chimney as used and to the measurements made at the points previously mentioned. Expressing the velocity in terms of weight of

*"Combustion Efficiencies as Related to Performance of Domestic Heating Plants," Univ. of Ill. Eng. Exp. Sta. Circ. 44, 1942, pp. 21-23.

gases flowing in pounds per hour and the density in terms of the temperature of the gases, Equation (8) reduces to

$$W = 20\,200\sqrt{\frac{D}{t + 460}} \tag{9}$$

in which

W = weight of chimney gas, lb. per hr.
D = observed draft, in. of water
t = temperature of chimney gases at the level of the attic floor, deg. F.

In order to determine the rate of flow for any given on-period the average temperature and draft were first obtained by using Equations (6) and (7), respectively, in connection with values read from the recorder charts. The average temperature and draft were then substituted in Equation (9).

4. *Average Temperature and Draft During Off-Periods.*—By an analysis similar to that discussed in Section 2, it was found that the average temperature during any given off-period could be expressed by the equation:

$$t_c = \frac{t_m}{0.814\varphi_2^{0.145}} \tag{10}$$

in which

t_c = average temperature of the chimney gases at the level of the attic floor, deg. F.
t_m = temperature of the chimney gases at the level of the attic floor at the start of the off-period, deg. F.
φ_2 = length of off-period, minutes.

Draft curves were plotted similar to the temperature curves discussed in Section 2. The characteristic equation for these curves was of the form:

$$D = D_m\varphi^n \tag{11}$$

in which

φ = time from start of off-period, minutes
D = instantaneous draft at time, in. of water
D_m = draft at start of off-period, in. of water.

The exponent, n, in Equation (11) is a variable, the value of which is dependent on D_m, as shown in Table 7.

TABLE 7
VALUES OF EXPONENT n TO BE USED IN EQUATIONS (11) AND (12)

D_m in. of water	n	D_m in. of water	n
0.100	−0.297	0.140	−0.151
0.110	−0.230	0.150	−0.136
0.120	−0.193	0.160	−0.123
0.130	−0.170	0.170	−0.114

Integrating Equation (11) between the limits of zero and φ_2 and dividing by φ_2:

$$D_c = \frac{\varphi_2{}^n}{n+1} D_m \tag{12}$$

in which

D_c = average draft, in. of water
φ_2 = length of the off-period, minutes
D_m = draft at start of off-period, in. of water.

5. *Average Rate of Flow During Off-Periods.*—During the off-periods no combustion occurred and air was drawn through the burner and over the heating surfaces in the boiler by the action of the draft in the chimney. Since there was no primary heat source, this draft was created entirely by the residual heat in the combustion chamber, boiler, and chimney. Furthermore, since no CO_2 was generated, it was not possible to determine the weight of gases flowing from the analyses of the gases and fuel as it was for the on-period. Therefore, some other method had to be employed.

The method employed was to introduce CO_2 into the chimney at a predetermined constant rate, and to measure the concentration of CO_2 appearing at the level of the attic floor. For this purpose pure CO_2 was stored in a gasometer and introduced into the base of the chimney through a short length of horizontal pipe extending across the chimney and perforated on both sides. The gasometer afforded a means of both measuring and controlling the rate of flow of CO_2. The perforated pipe introduced the CO_2 horizontally and at a sufficiently low velocity to permit diffusion across the whole cross-sectional area of the chimney. It also eliminated the possibility of any injection action interfering with the normal draft. Sufficient CO_2 was introduced to produce a concentration of about 2 per cent at the level of the attic floor, as measured by means of an Orsat apparatus. From this concentration the

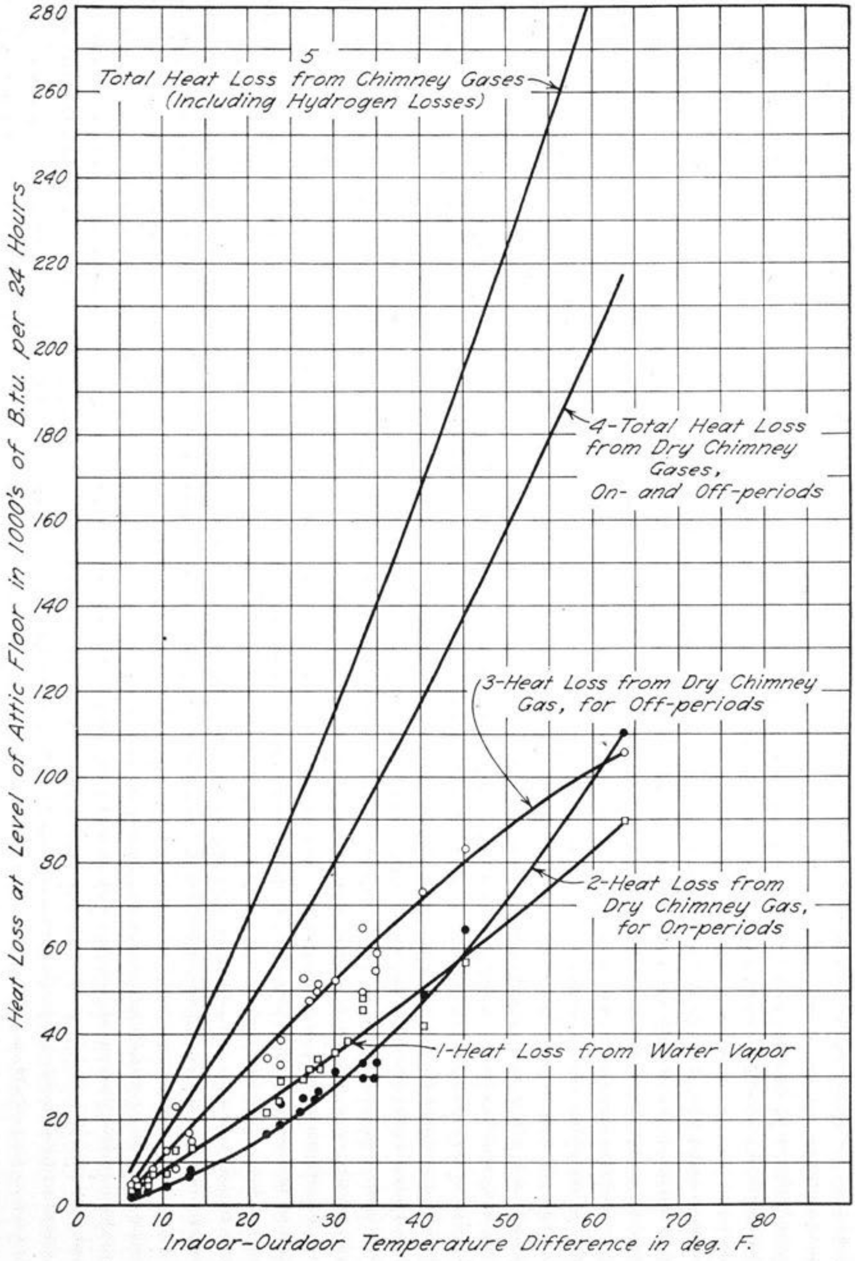

FIG. 30. DAILY HEAT LOSSES FROM CHIMNEY GASES

rate of flow of the chimney gases during the off-periods was determined with an accuracy of ± 5 per cent. From the rates of flow so determined, the velocities of the chimney gases were computed and plotted on logarithmic paper against the corresponding observed drafts as was done for the on-periods. The characteristic flow equation in this case was of the form:

$$V = 1.26\sqrt{\frac{D}{\rho}} \tag{13}$$

in which

V = velocity, ft. per sec.
D = observed draft, in. of water
ρ = density of chimney gases at the level of the attic floor, lb. per cu. ft.

Expressing the velocity in terms of weight of gases flowing in pounds per hour and the density in terms of the temperature of the gases, Equation (13) reduces to:

$$W = 14\,920\sqrt{\frac{D}{t + 460}} \tag{14}$$

in which

W = weight of chimney gases, lb. per hr.
D = observed draft, in. of water
t = temperature of chimney gases at the level of the attic floor, deg. F.

In order to obtain the rate of flow for any given off-period, the average temperature and draft were first obtained by using Equations (10) and (12), respectively, in connection with values read from the recorder charts. The average temperature and draft were then substituted in Equation (14).

6. *Heat Losses From Chimney Gases.*—The daily heat losses in the chimney gases over the whole range of outdoor temperatures as computed from the daily test data and the formulas given in Sections (1) to (5) of the Appendix are shown in Fig. 30. For each individual on-period, the heat loss in water vapor formed by the combustion of hydrogen was obtained by substituting in Equation (2) the proper average values of W_o, H_2, h_c, h_{fi}, and the total time, Φ, as applying to that period. The summation of these losses for all of the on-periods of the burner occurring during the twenty-four hour test period repre-

sented by any given indoor-outdoor temperature difference is shown as curve (1) in Fig. 30. The heat loss in the dry chimney gases for each individual on-period was computed by substituting in Equation (1) the duration of the on-period in hours and the average temperature and average rate of flow obtained from Equations (6) and (9), respectively. The summation of these on-period losses for the dry chimney gases is represented by curve (2). In the same way the heat loss in the chimney gases for each individual off-period was computed by substituting in Equation (1) the duration of the off-period in hours and the average temperature and average rate of flow obtained from Equations (10) and (14), respectively, and the summation of these losses is shown by curve (3).

Curve (4) was obtained by adding curves (2) and (3), and represents the total loss in the dry chimney gases for the day. Curve (5), obtained by adding curves (4) and (1), represents the total daily heat loss due to the chimney gases.

It may be observed that in average winter weather, at an indoor-outdoor temperature difference of 34 deg. F., the losses during the off-periods amount to approximately 9 per cent of the heat input from the oil burned. It is reasonable to assume that about two-thirds of this loss results from air drawn through the burner and the rest from other sources of leakage. This seems to indicate the possibility that approximately 6 per cent of the total heat input might be saved by proper design of the burner to prevent entrance of air during the off-periods.

RECENT PUBLICATIONS OF THE ENGINEERING EXPERIMENT STATION†

Circular No. 39. Papers Presented at the Fifth Short Course in Coal Utilization, Held at the University of Illinois, May 23-25, 1939. 1939. *Fifty cents.*

Reprint No. 15. Stress, Strain, and Structural Damage, by H. F. Moore. 1940. *None available.*

Bulletin No. 318. Investigation of Oil-fired Forced-Air Furnace Systems in the Research Residence, by A. P. Kratz and S. Konzo. 1939. *Ninety cents.*

Bulletin No. 319. Laminar Flow of Sludges in Pipes With Special Reference to Sewage Sludge, by Harold E. Babbitt and David H. Caldwell. 1939. *Sixty-five cents.*

Bulletin No. 320. The Hardenability of Carburizing Steels, by Walter H. Bruckner. 1939. *Seventy cents.*

Bulletin No. 321. Summer Cooling in the Research Residence With a Condensing Unit Operated at Two Capacities, by A. P. Kratz, S. Konzo, M. K. Fahnestock, and E. L. Broderick. 1940. *Seventy cents.*

Circular No. 40. German-English Glossary for Civil Engineering, by A. A. Brielmaier. 1940. *Fifty cents.*

Bulletin No. 322. An Investigation of Rigid Frame Bridges: Part III, Tests of Structural Hinges of Reinforced Concrete, by Ralph W. Kluge. 1940. *Forty cents.*

Circular No. 41. Papers Presented at the Twenty-seventh Annual Conference on Highway Engineering, Held at the University of Illinois, March 6-8, 1940. 1940. *Fifty cents.*

Reprint No. 16. Sixth Progress Report of the Joint Investigation of Fissures in Railroad Rails, by H. F. Moore. 1940. *Fifteen cents.*

Reprint No. 17. Second Progress Report of the Joint Investigation of Continuous Welded Rail, by H. F. Moore, H. R. Thomas, and R. E. Cramer. 1940. *Fifteen cents.*

Reprint No. 18. English Engineering Units and Their Dimensions, by E. W. Comings. 1940. *Fifteen cents.*

Reprint No. 19. Electro-organic Chemical Preparations, Part II, by Sherlock Swann, Jr. 1940. *Thirty cents.*

Reprint No. 20. New Trends in Boiler Feed Water Treatment, by F. G. Straub. 1940. *Fifteen cents.*

Bulletin No. 323. Turbulent Flow of Sludges in Pipes, by H. E. Babbitt and D. H. Caldwell. 1940. *Forty-five cents.*

Bulletin No. 324. The Recovery of Sulphur Dioxide from Dilute Waste Gases by Chemical Regeneration of the Absorbent, by H. F. Johnstone and A. D. Singh. 1940. *One dollar.*

Bulletin No. 325. Photoelectric Sensitization of Alkali Surfaces by Means of Electric Discharges in Water Vapor, by J. T. Tykociner, Jacob Kunz, and L. P. Garner. 1940. *Forty cents.*

Bulletin No. 326. An Analytical and Experimental Study of the Hydraulic Ram, by W. M. Lansford and W. G. Dugan. 1940. *Seventy cents.*

Bulletin No. 327. Fatigue Tests of Welded Joints in Structural Steel Plates, by W. M. Wilson, W. H. Bruckner, J. V. Coombe, and R. A. Wilde. 1941. *One dollar.*

Bulletin No. 328. A Study of the Plate Factors in the Fractional Distillation of the Ethyl Alcohol-Water System, by D. B. Keyes and L. Byman. 1941. *Seventy cents.*

Bulletin No. 329. A Study of the Collapsing Pressure of Thin-Walled Cylinders, by R. G. Sturm. 1941. *Eighty cents.*

Bulletin No. 330. Heat Transfer to Clouds of Falling Particles, by H. F. Johnstone, R. L. Pigford, and J. H. Chapin. 1941. *Sixty-five cents.*

Bulletin No. 331. Tests of Cylindrical Shells, by W. M. Wilson and E. D. Olson. 1941. *One dollar.*

Reprint No. 21. Seventh Progress Report of the Joint Investigation of Fissures in Railroad Rails, by H. F. Moore. 1941. *Fifteen cents.*

Bulletin No. 332. Analyses of Skew Slabs, by Vernon P. Jensen. 1941. *One dollar.*

Bulletin No. 333. The Suitability of Stabilized Soil for Building Construction, by E. L. Hansen. 1941. *Forty-five cents.*

†Copies of the complete list of publications can be obtained without charge by addressing the Engineering Experiment Station, Urbana, Ill.

Circular No. 42. Papers Presented at the Twenty-eighth Annual Conference on Highway Engineering, Held at the University of Illinois, March 5-7, 1941. 1941. *Fifty cents.*

Bulletin No. 334. The Effect of Range of Stress on the Fatigue Strength of Metals, by James O. Smith. 1942. *Fifty-five cents.*

Bulletin No. 335. A Photoelastic Study of Stresses in Gear Tooth Fillets, by Thomas J. Dolan and Edward L. Broghamer. 1942. *Forty-five cents.*

Circular No. 43. Papers Presented at the Sixth Short Course in Coal Utilization, Held at the University of Illinois, May 21-23, 1941. 1942. *Fifty cents.*

Circular No. 44. Combustion Efficiencies as Related to Performance of Domestic Heating Plants, by Alonzo P. Kratz, Seichi Konzo, and Daniel W. Thomson. 1942. *Forty cents.*

Bulletin No. 336. Moments in I-Beam Bridges, by Nathan M. Newmark and Chester P. Siess. 1942. *One dollar.*

Bulletin No. 337. Tests of Riveted and Welded Joints in Low-Alloy Structural Steels, by Wilbur M. Wilson, Walter H. Bruckner, and Thomas H. McCrackin. 1942. *Eighty cents.*

Bulletin No. 338. Influence Charts for Computation of Stresses in Elastic Foundations, by Nathan M. Newmark. 1942. *Thirty-five cents.*

Bulletin No. 339. Properties and Applications of Phase-Shifted Rectified Sine Waves, by J. Tykocinski Tykociner and Louis R. Bloom. 1942. *Sixty cents.*

Bulletin No. 340. Loss of Head in Flow of Fluids Through Various Types of One-and-one-half-inch Valves, by Wallace M. Lansford. 1942. *Forty cents.*

Bulletin No. 341. Effect of Cold Drawing on Mechanical Properties of Welded Steel Tubing, by Winston E. Black. 1942. *Forty cents.*

Circular No. 45. Simplified Procedure for Selecting Capacities of Duct Systems for Gravity Warm-Air Heating Plants, by Alonzo P. Kratz and Seichi Konzo. 1942. *Fifty-five cents.*

Circular No. 46. Hand-Firing of Bituminous Coal in the Home, by Alonzo P. Kratz, Julian R. Fellows, and John C. Miles. 1942. *Twenty-five cents.*

Circular No. 47. Save Fuel for Victory. 1942. *Twenty-five cents.*

Reprint No. 22. Eighth Progress Report of the Joint Investigation of Fissures in Railroad Rails, by Herbert F. Moore. 1942. *Fifteen cents.*

Reprint No. 23. Numerical Procedure for Computing Deflections, Moments, and Buckling Loads, by Nathan M. Newmark. 1942. *None available.*

Bulletin No. 342. Pressure Losses in Registers and Stackheads in Forced Warm-Air Heating, by Alonzo P. Kratz and Seichi Konzo. 1942. *Sixty-five cents.*

Bulletin No. 343. Tests of Composite Timber and Concrete Beams, by Frank E. Richart and Clarence B. Williams, Jr. 1943. *Seventy cents.*

**Bulletin No. 344.* Fatigue Tests of Commercial Butt Welds in Structural Steel Plates, by Wilbur M. Wilson, Walter H. Bruckner, Thomas H. McCrackin, Jr., and Howard C. Beede. 1943. *One dollar.*

**Bulletin No. 345.* Ultimate Strength of Reinforced Concrete Beams as Related to the Plasticity Ratio of Concrete, by Vernon P. Jensen. 1943. *Seventy cents.*

**Bulletin No. 346.* Highway Slab-Bridges With Curbs: Laboratory Tests and Proposed Design Method, by Vernon P. Jensen, Ralph W. Kluge, and Clarence B. Williams, Jr. 1943. *Ninety cents.*

**Bulletin No. 347.* Fracture and Ductility of Lead and Lead Alloys for Cable Sheathing, by Herbert F. Moore and Curtis W. Dollins. 1943. *Eighty cents.*

Reprint No. 24. Ninth Progress Report of the Joint Investigation of Fissures in Railroad Rails, by Norville J. Alleman, Ralph E. Cramer, Russell S. Jensen. 1943. *Fifteen cents.*

Reprint No. 25. First Progress Report of the Investigation of Shelly Spots in Railroad Rails, by Ralph E. Cramer. 1943. *Fifteen cents.*

Reprint No. 26. First Progress Report of the Investigation of Fatigue Failures in Rail Joint Bars, by Norville J. Alleman. 1943. *Fifteen cents.*

**Reprint No. 27.* A Brief History of Lime, Cement, Concrete, and Reinforced Concrete, by Jasper O. Draffin. 1943. *Fifteen cents.*

**Bulletin No. 348.* Fuel Savings Resulting From Closing of Rooms and From Use of a Fireplace, by Seichi Konzo and Warren S. Harris. 1943. *Forty cents.*

**Bulletin No. 349.* Performance of a Hot-Water Heating System in the I=B=R Research Home at the University of Illinois, by Alonzo P. Kratz, Warren S. Harris, Maurice K. Fahnestock, and Ross J. Martin. 1943. *Seventy-five cents.*

*A limited number of copies of bulletins starred are available for free distribution.

This page is intentionally blank.

This page is intentionally blank.

UNIVERSITY OF ILLINOIS

Colleges and Schools at Urbana

College of Liberal Arts and Sciences.—Curriculum in the General Division; general curriculum with majors in the humanities and sciences; specialized curricula in chemistry and chemical engineering; general courses preparatory to the study of journalism, law, medicine, and dentistry.

College of Commerce and Business Administration.—Specialized curricula in accountancy, banking and finance, commerce and law, commercial teaching, economics, industrial administration, management, marketing, and public affairs.

College of Engineering.—Curricula in agricultural, ceramic, chemical, civil, electrical, general, mechanical, metallurgical, mining, and public health engineering.

College of Agriculture.—Curricula in agriculture, dairy technology, floriculture, home economics, and vocational agriculture; pre-professional training in forestry.

College of Education.—Curricula in education, agricultural education, home economics education, and industrial education. The University High School is the practice school of the College of Education.

College of Fine and Applied Arts.—Curricula in architecture, landscape architecture, art, music, and music education.

College of Law.—Professional curricula in law.

School of Journalism.—Editorial, advertising, and publication management curricula.

School of Physical Education.—Curricula in physical education.

Library School.—Curriculum in library science for college graduates.

Graduate School.—Advanced study and research.

Summer Term.—Courses for undergraduate and graduate students.

University Extension Division.—Courses taught by correspondence, extramural courses, science aids service, speech aids service, and visual aids service.

Colleges in Chicago

College of Dentistry.—Professional curriculum in dentistry.

College of Medicine.—Professional curriculum in medicine.

College of Pharmacy.—Professional curriculum in pharmacy.

University Experiment Stations, and Research and Service Organizations at Urbana

Agricultural Experiment Station
Engineering Experiment Station
Extension Service in Agriculture and Home Economics
Bureau of Economic and Business Research
Bureau of Community Planning
Bureau of Educational Research
Bureau of Institutional Research
Personnel Bureau
Radio Station (WILL)
University of Illinois Press

State Scientific Surveys and Other Divisions at Urbana

State Geological Survey
State Natural History Survey
State Water Survey
State Diagnostic Laboratory (for Animal Pathology)
U. S. Regional Soybean Laboratory

For general catalog of the University, special circulars, and other information, address
The Registrar, University of Illinois
Urbana, Illinois