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INVESTIGATION OF SUMMER COOLING IN THE WARM-AIR HEATING RESEARCH RESIDENCE

CONDUCTED BY

THE ENGINEERING EXPERIMENT STATION
UNIVERSITY OF ILLINOIS

IN COOPERATION WITH

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, AND NATIONAL WARM-AIR HEATING AND AIR CONDITIONING ASSOCIATION

BY

ALONZO P. KRATZ
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AND
SEICHI KONZO

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The Engineering Experiment Station,
University of Illinois,
Urbana, Illinois
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INVESTIGATION OF SUMMER COOLING IN THE
WARM-AIR HEATING RESEARCH RESIDENCE

I. INTRODUCTION

1. Preliminary Statement.—This bulletin is a report of the results of an investigation conducted under the terms of cooperative agreements between the National Warm-Air Heating and Air Conditioning Association, the American Society of Heating and Ventilating Engineers, and the University of Illinois, providing for investigations of warm-air furnaces and furnace systems, and investigations of general problems in heating, ventilating and air conditioning. The agreement with the National Warm-Air Heating and Air Conditioning Association was formally approved in August, 1918, and that with the American Society of Heating and Ventilating Engineers in April, 1931.

The cooperating associations have been represented by Advisory Committees, the personnel of which changes somewhat from year to year. During the period of this investigation the National Warm-Air Heating and Air Conditioning Association has been represented by a Research Advisory Committee consisting of:

F. G. Sedgwick, Chairman, Waterman-Waterbury Company, Minneapolis, Minnesota.
R. G. Gulick, May-Fieberger Company, Newark, Ohio.
C. M. Lyman, Honorary Member, Utica, New York.

The American Society of Heating and Ventilating Engineers has been represented by a Technical Advisory Committee on Refrigeration in Relation to Air Treatment consisting of:

A. P. Kratz, Chairman, University of Illinois, Urbana, Illinois
E. A. Brandt, National Association of Ice Industries, Chicago, Illinois
G. B. Bright, Consulting Engineer, Detroit, Michigan
John Everett, Jr., Air and Refrigeration Corporation, New York City
Elliot D. Harrington, General Electric Company, Bloomfield, New Jersey
S. R. Lewis, Consulting Engineer, Chicago, Illinois
H. J. Macintire, University of Illinois, Urbana, Illinois
E. D. Milener, American Gas Association, New York City
K. W. Miller, Utilities Research Commission, Chicago, Illinois
It is the function of these committees to propose such problems for investigation as are of the greatest interest to manufacturers and engineers. 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 cooperating associations provide funds for defraying a major part of the expense of this research work. The studies in summer cooling were conducted in the Research Residence in Urbana, Illinois. This Residence was built, furnished, and completely equipped specifically for research work in warm-air heating, by the National Warm-Air Heating and Air Conditioning Association in December 1924.

2. Objects and Scope of Investigation.—Since a study of summer cooling under actual service conditions must necessarily be confined to the summer season alone, the amount of work that can be completed during any one season is limited by the length and severity of the season. This bulletin contains the results obtained during the three summer seasons of 1932, 1933 and 1934, and the objects and scope of the investigation were somewhat modified by the experience gained as the investigation progressed. The original objects of the investigation consisted of:

(1) The determination of the cooling load and its hourly variation when cooling the Research Residence as a whole, under both night and day conditions.

(2) The determination of the effectiveness of awnings as a means of reducing the cooling load of the Residence as a whole.

(3) The allocation to the various rooms of the heat entering the
Residence and the determination of the hourly variation in the cooling load of the individual rooms.

During the first season it was found that it was not feasible to isolate the different rooms in such a way that heat losses to, or heat gains from, the adjacent rooms could be prevented. It was also found that the disturbing effect of heat lag in the structure could not be eliminated. Hence, studies of the heat loads on the separate rooms were discontinued.

The investigation for the summer of 1933 was undertaken with the object of determining:

1. To what extent the circulation through the Residence of air taken from the outdoors at night could be used to supplement artificial cooling during the day.

2. To what extent the circulation of air taken from the outdoors at night could be used to eliminate the necessity for any additional artificial cooling during the day.

3. The relative merits of natural ventilation, of a fan in the forced-air heating system, and of a fan in the attic as means for circulating the air taken in from outdoors at night.

The investigation for the summer of 1934 was undertaken with the object of determining to what extent a 2½-ton mechanical refrigerating unit could be used to produce satisfactory cooling in the Research Residence, both when supplemented by the circulation of outdoor air through the house at night with the second-story windows open and when not so supplemented.

3. 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. It was initiated under the direction of President A. C. Willard who was at that time Professor of Heating and Ventilation and Head of the Department of Mechanical Engineering, and was under the general administrative direction of the late Dean M. S. Ketchum, who was then Director of the Engineering Experiment Station. The investigation was continued under the general administrative direction of Dean M. L. Enger, Director of the Engineering Experiment Station, and of Prof. O. A. Leutwiler, Head of the Department of Mechanical Engineering.

Acknowledgment is made to E. L. Broderick, now Research Assistant, W. S. Harris, formerly Special Research Assistant and A. F. Hubbard, formerly Research Graduate Assistant for their services in conducting the tests and in the reduction of the test data.
Acknowledgment is also made to the various organizations and companies who contributed funds through the American Society of Heating and Ventilating Engineers and the National Warm-Air Heating and Air Conditioning Association, and to the manufacturers who coöperated by furnishing instruments and equipment used in the investigation.

II. Description of Plants and Apparatus

4. Research Residence.—The Research Residence, shown in Fig. 1 equipped with awnings for operation in summer, faces to the south, and is of standard frame construction, with the exception that the studding is 2 in. x 6 in. instead of the 2 in. x 4 in. more commonly used. The roof consists of copper shingles laid on felt over wood sheathing. The wall section consists of weather boarding, ship-lap siding, building paper, 6 in. studding, wood lath and plaster with rough sand finish. The interior plaster walls and ceilings are painted. The coefficient of
INVESTIGATION OF SUMMER COOLING IN RESIDENCE

FIG. 2. FIRST-STORY PLAN OF RESEARCH RESIDENCE

FIG. 3. SECOND-STORY PLAN OF RESEARCH RESIDENCE
heat transmission for this wall section is 0.25* B.t.u. per sq. ft. per hr. per deg. F., at a wind velocity of 15 mi. per hr. The walls are not insulated, and no weather stripping is used on the windows and doors. The total heated space in winter, including the sun-room, is approximately 17,540 cu. ft. and the heat loss at an indoor-outdoor temperature difference of 70 deg. F. is approximately 137,570 B.t.u. per hr. The Residence has a total of 50 windows and two outside doors, and the ratio of total exposed window and door area to the net wall area is 22.4 per cent. The room arrangement and exposures are shown in the floor plans in Figs. 2, 3 and 4.

For the purpose of this investigation the sun-room was isolated from the rest of the house by closing the doors leading into the dining room. The entire third story was regarded as an attic, and during the daytime was isolated from the rest of the house by closing the door at the head of the stairs. With the exception of the space above the northwest bedroom and the small spaces adjacent to the dormer windows, the third story had hardwood floors laid on pine sub-flooring. The small spaces adjacent to the dormer windows were not floored, and in the space above the northwest bedroom one inch of insulating blanket was nailed to the upper edges of the floor joists. Hence, prac-
tically all second-floor ceilings were at least equivalent to lath and plaster with flooring above it. The total space cooled consisted of three rooms and a breakfast nook on the first story, and three bedrooms and a bathroom on the second story, together with the two interconnecting halls, making in all approximately 14,170 cu. ft. The total calculated heat gain for an outdoor-indoor temperature difference of 11 deg. F. was 37,500* B.t.u. per hr. No cooking was done in the kitchen, but the heat transmitted through the doors from the sun-room, which was not ventilated by opening the windows, tended to offset any reduction in cooling load arising from the fact that no cooking was done. For all tests, with the exception of one series made during the first summer, the Residence was equipped with a total of 19 awnings which were placed at all east, south, and west windows. The awnings were of the hood type with rope pull-up, consisting of a top and two wings with an 11 in. valence all around. They were made from 10 oz. duck, and had gray and green stripes. Unless otherwise specified, the state of the Residence was the same for the work done during the three summers.

5. Forced-Air System.—For the purpose of this investigation the forced-air system used in the winter for heating the Residence was utilized as the distributing system to deliver the cooled air to the rooms and to return the warmed air through the cooling coil to the suction side of the fan. This system has been described in detail in Engineering Experiment Station Bulletin No. 266, and the basement plan, showing the duct system as modified for summer cooling, is given in Fig. 5. Under winter conditions the fan drew the cool air from the rooms through one main and two auxiliary return ducts and delivered it into the furnace casing. From the furnace casing the warmed air passed into the distributing duct system and was delivered into the different rooms through registers. For summer cooling the system was modified by blocking the two auxiliary return ducts and the delivery ducts leading to sun-room and third story. It was also necessary to readjust the dampers in the distributing ducts in order to obtain a proper balance of the cooling required on the two stories. During winter operation the fan delivered 1675 cu. ft. of air per min. Under summer conditions the fan drew warm air from the rooms and delivered cooled air through the distributing duct system and registers. Owing to the increased resistance of the modified system, the capacity

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Fig. 5. Basement Plan Showing Duct System Used for Distributing Cool Air
of the fan was somewhat reduced. With one exception the registers on the delivery side of the system were of the baseboard type. In the east bedroom a wall type register located 7 ft. above the floor was used on some of the tests.

6. Plant for Cooling with Ice.—For the first series of tests a plant using ice as the cooling medium was selected in preference to one using mechanical refrigeration, because it was considered that the cooling load on the Residence could be conveniently and accurately determined from the weight of ice melted, while the determination of the refrigeration load for the mechanical plant offered complications in testing that might be reflected in the accuracy of the results. At this time, no consideration was given to any possible relative merits in the two types of plants from commercial or other standpoints.
The arrangement and details of the cooling plant are shown in Fig. 6. A cooling coil, through which ice water was circulated, was located in a by-pass in the return air duct, and the air to be cooled was passed over the outside surface of the coil and back into the return air duct before going to the fan. The sections of the cooling coil were connected in series, and the flow of water was in the opposite direction to that of the air, thus forming a counter-flow arrangement. Two sections of coil were used, each section consisting of three rows of finned tubing set in headers. The coil contained a total of 308 linear feet of $\frac{3}{4}$-in. tubing having helical fins, spaced 6 fins per inch. The over-all diameter of the tubes, including the fins was 1 1/2 inches. The inside dimensions of the casing were 27 7/8 in. by 49 1/8 in. Since the two sections of the coil were used in tandem, the gross or face area presented to the flow of air was 9.5 sq. ft., and the net free area was 4.2 sq. ft., giving a ratio of free area to gross area of 44.3 per cent. The total cooling surface was 473 sq. ft. A drip pan at the bottom of the casing for the coil served to collect the condensation resulting from the dehumidification of the air, and from the drip pan the condensation was drained into a bucket placed on platform scales.

The chilled water was pumped from the bottom of the ice tank, shown in Fig. 6, and passed through the cooling coil. A calibrated water meter and a thin plate orifice were included in the water line. The orifice was used as an indicator to control the flow of water through the coil. After passing through the coil, the water was returned to spray heads in the top of the ice tank, and sprayed over the ice. The latter was placed on racks, and both the circulating water and the meltage from the ice accumulated in the bottom of the tank. The water space was connected to a weir box, shown as section B-B in Fig. 6, containing a drain pipe which faced upward. This drain pipe had a sharp edge that served as a weir and maintained the water at a constant level just below the ice racks. All of the water that drained out of the weir box, therefore, represented the ice meltage that occurred during the time that the drain pipe was open. By allowing this water to collect in a pan on scales, it was possible to determine the ice meltage for any period of time desired.

All of the air used for cooling was taken from the rooms and recirculated, and the fan delivered approximately 1475 cu. ft. of air per min. computed at a density of 0.0749 lb. per cu. ft. The plant was operated to maintain a constant dry-bulb temperature in the rooms of the Residence, and no attempt was made to control the relative humidity so as to obtain any specified percentage. The control of the dry-
INVESTIGATION OF SUMMER COOLING IN RESIDENCE

bulp temperature was obtained by means of a modulating by-pass damper operated from a thermostat located in the dining room. This damper, shown in Fig. 6, operated so as to cause part of the return air to pass through the cooling coil. The cooled air was then mixed with the balance of the return air on the suction side of the fan, and the mixture was delivered through the furnace casing into the distributing duct system. The amount of air thus by-passed and cooled was at all times just sufficient so that, when it was mixed with the balance of the return air, the resulting temperature of the mixture delivered through the registers was that required to maintain the desired dry-bulb temperature in the rooms. This method of control was selected in preference to any method involving intermittent operation of the fan or water pump because under conditions of continuous operation it was possible to obtain accurate determinations of the amounts of air circulated and ice melted. Under conditions of intermittent operation there is always some doubt concerning the proper weighting of the averages and the interpretation of the results. Since no question of re-heating the air to obtain control of both temperature and humidity was involved, the heat absorbed by the cooling coil was the same as the heat given up by the air, and also the same as the total heat gain of the building. Hence, the same final temperature of the air would be required to maintain a given dry-bulb temperature, and it made no difference whether all of the air was passed through the coil and brought to this temperature, or whether part of the air was cooled to a lower temperature and then mixed with the balance in such proportions as to give the required final temperature. There is no reason to believe that the method of control would have any effect on the amount of cooling required under given outdoor conditions, provided the same degree of refinement in the regulation could be obtained in each case.

7. Plants for Cooling with Air from Outdoors at Night.—These tests involved taking air from the outdoors at night and circulating it through the Research Residence. This was accomplished either by opening the windows and relying on natural ventilation unaided by the use of a fan, by opening the windows and making use of the fan in the forced-air system, or by opening the windows and making use of a special fan installed in the attic. The first arrangement will hereafter be designated as natural ventilation, the second as the basement fan, and the third as the attic fan.

The arrangement of the forced-air heating plant making use of the basement fan is shown in Fig. 5. When used for circulating air from
the outdoors at night, all of the return air ducts were blocked so that the fan could not take any air from the rooms. For a few tests the outdoor air was taken into the fan through a connection from the nearest window. This connection was found to restrict the flow of air, however, and for later tests it was removed, and the fan was allowed to take air directly from the basement through an opening in the duct on the suction side of the fan. When this was done the basement door was opened to permit the entrance of air from outdoors. The fan delivered from 1740 to 2120 cu. ft. of air per min. computed at a density of 0.0749 lb. per cu. ft.

The arrangement of the attic fan is shown in Fig. 7. A 24-in. fan was installed in the doorway at the head of the stairs leading from the second to the third story. Thus the air was delivered into the attic space at a point approximately centrally located with respect to the horizontal plane of the third story. It was allowed to escape through windows on all sides. The use of a box and hinged damper on the suction side of the fan, as shown in Fig. 7, permitted outdoor air to be drawn into the open first- and second-story windows at night, or to be drawn through a duct from a third-story window in order to provide positive ventilation for the third story during the day. This arrangement did not interfere with the full capacity of the fan for circulating
the air at night, and the somewhat reduced capacity was more than sufficient for ventilating the attic during the day. The fan delivered approximately 4000 cu. ft. of air per minute.

8. Plant for Cooling with Mechanical Refrigeration.—For the purpose of these tests the forced-air heating system was modified as shown in Fig. 5 and discussed in Section 5. This plant differed from the one used for cooling with ice in that provision was made for taking in approximately one air change per hour of outdoor air to be used for the purpose of ventilation during the operating periods of the plant. The arrangement of the cooling plant is shown in Fig. 8. The condensing unit consisted of a double-pipe condenser and a four-cylinder compressor driven by a 3 h.p. motor. This unit was self-contained. Cooling was accomplished by direct expansion of Freon in an evaporator unit placed in a by-pass in the central, or main, air return duct. The evaporator unit or cooling coil consisted of 32 rows of finned copper tubing placed 4 rows deep in the direction of air flow and 8 rows high measured along the vertical axis of the air duct. The tubes were 3/8 in. outside diameter and had 6 fins per inch. The overall diameter including fins was 1 1/2 in. The evaporator, as installed in the duct, was 17 in. high and 30 in. wide. The gross or face area presented to the flow of air was 3.54 sq. ft. and the net or free area was 2.18 sq. ft. giving a ratio of free area to gross area of 61.5 per cent. The coil contained a total of 80 lineal feet of tubing and had a total cooling surface of 122 sq. ft. The nominal rating of the condenser and evaporator units as
used was 21,700 B.t.u. per hour with refrigerant temperature of 32 deg. F., water temperature of 80 deg. F., and ambient air temperature of 100 deg. F., or 29,500 B.t.u. per hour with refrigerant temperature of 32 deg. F., and water temperature of 60 deg. F., with an air velocity of approximately 350 f.p.m. across the 3.54 sq. ft. of face area.

The operation of the refrigerating machine was controlled by means of thermostatic expansion valves having the thermostatic bulbs clamped to the tubes of the evaporator, and a water control valve so designed that the degree of opening for the flow of the condenser water was controlled by the condenser pressure. The latter also operated to shut off the supply of water to the condenser when the machine was not running. The operation of the cooling plant as a whole was controlled by means of a room thermostat placed in the hall on the second story, and this served to start and stop the refrigerating machine in accordance with the cooling load required to maintain constant room temperature.

In order to provide for both heating in the winter and cooling in the summer, the evaporator, or cooling coil, was installed in a by-pass in the central return duct, as shown in Fig. 8. For the summer work, the duct was blocked with tightly-fitting dampers at B and C, and all of the air delivered to the fan in the forced-air system passed through the cooling coil when the air in the house was being recirculated. For the purpose of providing outdoor air for cooling during the night, a slide damper, E, was placed in the by-pass on the down-stream side of the cooling coil, and a door, F, was placed in the recirculating duct just ahead of the fan. When outdoor air was required, the basement door and the door in the recirculating duct were opened, and the slide damper was closed. The fan in the forced-air system delivered approximately 1260 cu. ft. of air per minute when recirculating the air in the house and 2180 cu. ft. per minute when using outdoor air at night. Outdoor air, for the purpose of ventilation during the periods when the cooling plant was operating, was provided by means of the duct shown as “Detail A” in Fig. 8. This duct contained a venturi section for the measurement of the volume of air delivered, and it was necessary to use a small fan in order to deliver the equivalent of approximately one air change per hour. Wet- and dry-bulb temperatures of the air and inlet and outlet temperatures of the condenser water were measured by means of thermocouples or thermometers placed at points indicated in Fig. 8.

9. Room Cooling Units.—A few tests were run making use of room cooling units to cool the living room alone and also the entire first
story of the Residence. The two types of commercial room cooling units used are shown in Fig. 9, and their location in the living room is shown in Fig. 2. The first story had a total net volume of 7300 cu. ft. The living room, with a north, east, and south exposure was 13 ft. 6 in. by 20 ft. It had a ceiling height of 8 ft. 11 in., and a total net volume of 2410 cu. ft. This room contained 6 windows, having a total exposed glass area of 90 sq. ft.

Unit A was a portable insulated ice chest, consisting of two compartments, one located above the other. The upper compartment was for ice storage, with a maximum capacity of 300 lb. of ice when fully loaded, while the lower compartment formed a tank for holding the melted ice and water collected from the dehumidification of the air. The ice compartment was provided with special metal surfaces, in contact with the ice on one side and exposed to the air on the other; the side exposed to the air having extended fins which materially increased the heat transfer from air to ice. The warm air from the room, entering through a grilled opening in the front of the unit, Fig. 9, passed under and up the back side of the ice compartment in direct contact with the ice, and with the cold, finned, metal surfaces, into the suction inlet of the twin-rotor centrifugal fans. The latter ejected the cooled air into the room from the top of the cooling unit. The water resulting from the melting of the ice, together with any condensation of moisture resulting from the dehumidification of the warm
room air as it came into contact with the ice and cooling surfaces, was drained into the tank in the lower compartment.

For test purposes, this unit was mounted on a portable platform scale, which was sensitive to 0.01 lb., and both the ice meltage and dehumidification rates were determined by direct weighing.

Unit B, Fig. 9, consisted of an insulated ice storage tank of 500 lb. capacity, located in the basement, and a cooling unit placed in the living room on the first floor, Fig. 2. The cooling unit consisted of an attractively finished cabinet, which enclosed the extended surface cooling coil, dehumidification drip-pan, and fans. The cold water was pumped from the storage tank through the cooling coil, over which the warm room air was circulated by means of a twin-rotor centrifugal fan which could be operated at three different running speeds. For the tests under discussion the lowest fan speed was used. The moisture condensed out of the air drained from the cooling coil into a shallow drip pan beneath the coil from which it was collected and weighed.

It should be noted that neither unit A nor unit B made any provision for introducing air from outside of the house, but merely recirculated the air in the room.

10. *Instruments and Apparatus.*—The Research Residence is equipped with an extensive system of copper-constantan thermocouples for the purpose of obtaining the temperature of the air at different points in the forced-air system, surface temperatures at different points in the structure, and any other incidental temperatures for which thermocouples are particularly adapted. The thermocouple leads are all brought to a constant temperature box from which copper leads are run to a central switch system. A single cold junction immersed in melting ice is used, and all thermocouples are calibrated in place. The electromotive force is read by means of a precision potentiometer used in connection with a sensitive suspension type of galvanometer equipped with a mirror to reflect a beam of light.

In each room a standard located in the central axis of the room bears three mercury thermometers, one placed four inches above the floor, one at the 5-ft. level, and one four inches below the ceiling. In addition to the thermocouples and thermometers, nine recording dry-bulb thermometers were used for the studies in summer cooling. Other instruments used included wet- and dry-bulb mercury thermometers, an aspirating psychrometer, indicating and integrating wattmeters, platform scales, and water meters.
III. TEST PROCEDURE

11. General.—During each successive summer, irrespective of the type of plant used for cooling, continuous records were made, by means of temperature recorders, of the following air temperatures: outdoor, living room, dining room, kitchen, first- and third-story halls, east bedroom, southwest bedroom, and northwest bedroom. A continuous record was also made of the outdoor air taken in for ventilation in case such auxiliary ventilation was employed. Wet- and dry-bulb temperatures of the air entering and leaving the cooling coil were observed at regular intervals by means of wet- and dry-bulb mercury thermometers. Other incidental air temperatures were also observed periodically. Relative humidities both indoors and outdoors were observed by means of an aspirating psychrometer. The outdoor dry-bulb reading on this psychrometer served as a basis for correcting the outdoor temperature read from the recorder chart. Surface temperatures of the outside surface of the roof, the outside and inside surfaces of the north, south, and west walls, and of the under surfaces of the ceilings in the northwest bedroom, the east bedroom, and the living room were obtained periodically by means of thermocouples. A few observations were also made of the temperature of the upper surfaces of tables in the dining room and east bedroom. These data were plotted on a continuous chart with time as a base. This chart furnished a complete history of the variations in indoor and outdoor conditions, both during the periods of tests and the periods preceding and following tests, and was used as a basis for the analysis of the results.

The volumes of air delivered by the fans and passing through the cooling coil were computed from velocity traverses made by means of Pitot tubes. The density of the air varied from day to day and was not the same in different parts of the system. Any statement of air volume without a statement of the accompanying density is incomplete. Hence, in order to avoid the necessity for stating the density each time that an air volume is mentioned, all air volumes in this bulletin have been consistently stated in terms of the equivalent air volume at a temperature of 70 deg. F. and an absolute pressure of 29.92 ins. of mercury; i.e. at a standard density of 0.0749 lb. per cu. ft. Air velocity, however, is stated as the existing velocity at the density actually prevailing. A statement of air quantities in terms of weight of air is more satisfactory than one in terms of air volume. However, this method has not been generally adopted in practice, and all air quantities in this bulletin have been given in terms of air volumes, with
**Table 1**

**Desirable Indoor Air Conditions in Summer Corresponding to Outdoor Temperatures**

Applicable to exposures of less than 3 hours

<table>
<thead>
<tr>
<th>Outdoor Temperature deg. F.</th>
<th>Indoor Air Conditions with Dew-Point Constant at 57 deg. F.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Dry-Bulb deg. F.</td>
</tr>
<tr>
<td>95</td>
<td>80.0</td>
</tr>
<tr>
<td>90</td>
<td>78.0</td>
</tr>
<tr>
<td>85</td>
<td>76.5</td>
</tr>
<tr>
<td>80</td>
<td>75.0</td>
</tr>
<tr>
<td>75</td>
<td>73.5</td>
</tr>
<tr>
<td>70</td>
<td>72.0</td>
</tr>
</tbody>
</table>

*Reproduced from the American Society of Heating and Ventilating Engineers Guide for 1934, Table 2, page 33.

the addition of air weights in cases where such air weights enter into the computation of heat quantities.

12. *Cooling with Ice.*—The total period over which observations were made extended from June 1 to October 1, 1932, and during all weather warm enough to make artificial cooling desirable all windows, including those on the third story, remained closed. For a few of the preliminary tests on days requiring cooling, an attempt was made to maintain the relation between outdoor temperature and desirable indoor temperature given in Table 1. It at once became evident, however, that after the outdoor temperature had reached a peak value and started to decline, it was necessary progressively to reduce the indoor temperature in order to maintain the schedule. This procedure imposed a load on the plant that was regarded as outside of the practical limits of operation. Furthermore, if the conditions indoors were conducive to comfort during the peak outdoor temperature, there was nothing to indicate that the maintenance of these conditions resulted in discomfort after the peak had been passed. This matter is more fully discussed in Section 43.

The following method of operation was therefore adopted as a standard: The cooling plant was started when the effective temperature* indoors rose to 75 deg. F., irrespective of the outdoor temperature. This occurred when the indoor dry-bulb temperature reached a value between 78 and 81 deg. F., depending on the relative humidity. After starting the plant, the dry-bulb temperature was maintained approximately constant through the action of the thermostatically

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*See Appendix A for an explanation of effective temperature.
controlled by-pass damper. When the cooling load decreased until
the by-pass damper remained closed for some time, thus passing no
air through the cooling coil, both the fan and water pump were
stopped, providing that the outdoor temperature was appreciably
lower than that indoors. This usually occurred at night, but some-
times occurred in the daytime resulting from sudden change in weather
conditions. The plant was not started again until the indoor effective
temperature rose to 75 deg. F. If the outdoor temperature at night
did not drop appreciably below 80 deg. F., the plant ran continuously
day and night. Owing to the reduction in relative humidity after the
plant had been in operation for about 1½ hours, the resulting effective
temperature corresponding to the operating dry-bulb temperature of
from 78 deg. F. to 81 deg. F. ranged from 72 deg. F. to 74 deg. F.

A calibration test on the plant, with the by-pass damper opened
to pass the maximum amount of air through the cooling coil, indicated
that if the temperature of the air at the registers was not allowed
to fall below 60 deg. F., 396 gallons of water per hour had to be cir-
culated through the coil. Since the plant was controlled with the by-
pass damper and it was not possible, within the limits of the time
available, to determine the optimum amount of water to be circulated
under various conditions, all tests were run with 396 gallons of water
circulated per hour at a pressure of 23 lb. per square inch. The water
entered the coil at about 35 deg. F. The amount of water circulated
was measured by means of a calibrated water meter and the flow was
controlled by maintaining a constant pressure drop across a thin plate
orifice. The quantity of air circulated was obtained from traverses
made with a Pitot tube at the fan outlet.

The dampers in the ducts were set to balance the plant with all
of the room doors closed. When the plant was operated with the room
doors open, the balance was not disturbed.

13. Cooling with Air from Outdoors at Night.—The first study of
cooling with air taken from outdoors at night extended over the period
from May 23 to September 23, 1933. During the progress of this
study it became evident that the practice of allowing first-story win-
dows to remain open all night might be subject to criticism from the
standpoint of the householder. Studies were therefore undertaken in
which only the second-story windows were opened at night, and these
studies were continued over the period from June 1 to June 20, 1934.
Twenty-four hours of operation constituted a test, and for all tests,
with the exception of those in Series 12-33, the attic was ventilated by
natural circulation through the windows during the day. This was accomplished by opening all of the windows in the attic except the one opposite the door at the top of the stairs. At night, the natural circulation in the attic was employed for all series except those involving the attic fan. In the latter case the fan delivered into the attic the air drawn from the first and second stories, and the circulation was outward through the attic windows. For the tests on which the first-story windows remained closed, eleven windows on the second story were opened by raising the lower sash to the full extent. Two windows which were opposite air registers and one window at the second-story stair landing remained closed. During all of these tests, the general observations discussed in Section 11 were made, and the Residence was operated strictly on the schedules described in connection with the following discussion of the different test series:

Series 1-33. Restricted Natural Ventilation
Eight windows, four on the first story and four on the second, were opened halfway at 9 P.M. and closed at 6 A.M. on the following morning. The attic door was kept closed.

Series 2-33. Maximum Natural Ventilation
All windows on both stories were opened wide from the bottom, and the attic door was opened at 6 P.M. and closed at 6 A.M. on the following morning.

Series 3-33. Restricted Basement Fan Circulation
Eight windows, four on the first story and four on the second, were opened halfway and the basement fan was started, circulating outdoor air, at 9 P.M. The windows were closed and the fan stopped at 6 A.M. on the following morning (comparable with Series 1-33). The fan delivery was 1740 c.f.m., or 7.4 air changes per hour.

Series 4-33. Basement Air Recirculation
Basement air was recirculated for a short period in the early evening hours with the house closed. This was followed by circulation of outdoor air by the basement fan with all the house windows opened wide (1 window in each of 3 second-story rooms was opened from the top), and the attic door was opened. The windows were closed and the fan stopped at 6 A.M. on the following morning.

Series 5-33. Maximum Basement Fan Circulation
All windows were opened wide (1 window in each of 3 second-story rooms was opened from the top), the attic door was opened, and the
basement fan started circulating outdoor air at 6 P.M. The windows were closed and the fan was stopped at 6 A.M. on the following morning (comparable with Series 2-33). The fan delivery was 2120 c.f.m., or 9.0 air changes per hour.

**Series 7-33. Attic Fan Circulation with Windows Opened Half Way (except Halls)**

The fan was operated from 6 P.M. to 6 A.M. on the following morning (comparable with Series 2-33 and 5-33). The air delivery was 3940 c.f.m., or 16.6 air changes per hour.

**Series 8-33. Attic Fan Circulation with Windows Opened Wide (except Halls)**

Same as Series 7-33, except that all of the windows were opened wide. There was no difference in air delivery.

**Series 10-33. Attic Fan Circulation with Only Second-Story Windows Opened Wide**

The fan was operated from 6 P.M. to 6 A.M. on the following morning. The air delivery was 3940 c.f.m., or approximately 33.2 air changes per hour on second story.

**Series 11-33. Natural Ventilation with Only Second-Story Windows Opened Wide**

Same as Series 10-33, except with natural ventilation in effect. The attic door was open.

**Series 12-33. Attic Fan Circulation in Day Time Through Attic Space Only**

Night air circulation was the same as in Series 8-33 with attic fan circulation.

**Series 5-34. Basement Fan Circulation with Only Second-Story Windows Opened Wide**

The second-story windows and the attic and basement doors were closed at 6 A.M., and the fan was stopped. The house remained closed during the day, with no attempt made to control the indoor temperature. The second-story windows and the attic and basement doors were opened at 6 P.M., and the fan was started. During the night the fan took air from outdoors and delivered it into the rooms through the duct system. The fan delivery was 2180 c.f.m., or 9.2 air changes per hour.

**Series 6-34. Basement Fan Circulation with Only Second-Story Windows Opened Wide**

Same as Series 5-34, except that the windows were closed and the fan was stopped at 7 A.M. instead of 6 A.M.
For the purpose of comparing the fan circulation with natural ventilation, these series may be divided into three related groups. The first group consists of Series 1-33 and 3-33; the second of Series 2-33, 5-33, 7-33, and 8-33; and the third of Series 10-33, 11-33, 5-34, and 6-34.

14. Cooling with Air from Outdoors at Night Supplemented by Cooling with Ice during Day.—In two series of tests cooling with outdoor air at night was supplemented either by using a restricted amount of ice in the cooling plant during the day, or by using a room-cooling unit located on the first story. The test procedure was essentially the same as that discussed in Section 13, and the Residence was operated on the following schedules:

**Series 6-33. Day Cooling with Ice Plant**

The ice plant was operated in the day time until the outdoor temperature had dropped below the indoor temperature, when cooling with air from outdoors at night was started, making use of natural ventilation, the basement fan, or the attic fan, as in Series 2-33, 5-33, or 8-33.

**Series 9-33. Room Cooling Unit on First Story**

The room-cooling unit was operated on the first story in the day time. Cooling with air from outdoors at night with the attic fan was the same as in Series 7-33 and 8-33 except for the starting time.

When cooling with outdoor air at night was supplemented by cooling with ice during the day, as in Series 6-33, the total ice meltage was restricted to a maximum of 700 lb. per day. Since the period of maximum demand usually did not extend over a period of more than approximately 5 hours, the rate of meltage was adjusted so that if necessary the total 700 lb. could be used in 5 hours. After closing the windows and stopping the fan in the morning the indoor temperature was allowed to rise until it reached 81 deg. F., and then the cooling plant was started. The plant was allowed to operate at a constant rate until the entire 700 lb. of ice was melted, unless the indoor temperature fell below 79 deg. F., or unless the outdoor temperature dropped more than 3 deg. F. below the inside temperature. In the case first mentioned, the cooling plant was stopped until the inside temperature again rose to 81 deg. F. In the second case the cooling plant was stopped for the night, the windows were opened, and the fan started. Whenever the plant was stopped before the time that the windows were opened, the pump was stopped, thus discontinuing the circulation of water through the cooling coil, but the fan was allowed
to run in order to maintain recirculation of the air in the house. The 3-deg. F. difference between indoor and outdoor temperatures was allowed in order to provide for the difference in relative humidity indoors and outdoors. The latter difference was such that it required about 3 deg. F. lower dry-bulb temperature outdoors in order to have the same effective temperature both indoors and outdoors. If the windows were opened before these two effective temperatures were equalized it resulted in an undesirable increase in effective temperature indoors.

15. Cooling with Mechanical Refrigeration.—This study extended over the period from July 20 to August 10, 1934, and observations of general data were made as discussed in Section 11. All windows on the first and second stories remained closed both during the day and at night, but the windows in the attic, with the exception of one opposite the door at the top of the stairs, were allowed to remain open.

During this series of tests outdoor air amounting to approximately one air change per hour was introduced, as shown in Fig. 8, for the purpose of ventilation. The fan delivered approximately 5.3 air changes per hour through the cooling coil, of which 4.3 air changes per hour were recirculated air and one change per hour was taken from outdoors. The Residence was operated on the following schedule:

Series 4-34. Artificial Cooling During the Day Not Supplemented by Circulation of Outdoor Air at Night

The second-story windows were not opened during the 24 hours. The fan was run continuously during the 24 hours, delivering through the cooling unit both the recirculated air and the one air change per hour admitted for ventilation. The cooling plant was allowed to operate with thermostatic on and off control maintaining 81 deg. F. on the second story, which corresponded to an average of 80 deg. F. for the house.

During the periods over which the plant was in actual operation, observations were made of the weight of water circulated through the condenser and water jackets on the compressor, the temperature of the water entering and leaving the condenser and jackets, the temperature of the refrigerant entering and leaving the evaporator and at the thermostatic control bulb on the evaporator, the head and suction pressures, and the electrical inputs to the compressor and fan motors. The weight of condenser water was obtained by means of a calibrated water meter and the air quantities were obtained from traverses made with Pitot tubes at section D in the central return duct and at the
venturi section in the ventilating air duct, as shown in Fig. 8. The total duration of the running time was recorded by means of an electrical clock connected into the circuit for the compressor motor so that the clock operated while the compressor was running and stopped when the compressor stopped.

16. Cooling with Mechanical Refrigeration During Day Supplemented by Cooling with Air from Outdoors at Night.—This study extended over the periods from June 20 to July 20, 1934 and from August 10, to September 16, 1934. The windows on the first story remained closed both during the day and at night. The windows in the attic, with the exception of the one opposite the door at the top of the stairs, remained open. For the purpose of cooling at night, eleven windows on the second story were opened by raising the lower sash to the full extent. Two windows which were opposite air registers and one window at the second-story stair landing remained closed. Outdoor air for ventilation was introduced in a manner similar to that discussed in Section 8, except that on this series of tests there was one period of the day, extending from the time that the windows were closed until the temperature in the rooms rose to 81 deg. F. and the cooling plant was first started, during which no outdoor air was introduced for ventilation. The general observations made were similar to those discussed in Section 11, and the observations made on the cooling plant itself were similar to those discussed in Section 15. The Residence was operated on the following schedule:

Series 3-34. Artificial Cooling During the Day Supplemented by Circulation of Outdoor Air at Night

The second-story windows and the attic and basement doors were closed at 7 A.M., and the basement fan, which had been delivering outdoor air through the system, was stopped. When the temperature of the indoor air on the second story rose to 81 deg. F., the cooling plant was started, with the fan delivering through the cooling unit both the recirculated air and the outdoor air admitted for ventilation. The former was equivalent to 4.3 recirculations of the air in the house, and the latter was equivalent to one air change per hour, making a total of 5.3 air changes per hour delivered by the fan. The fan was run continuously through both on and off periods of the refrigerating unit. The cooling plant was allowed to operate with thermostatic on and off control, maintaining 81 deg. F. on the second story until the effective temperature outdoors became equal to the effective temperature on the second story indoors, as discussed in Section 14. The refrigerating
unit was then stopped, the second-story windows and attic door were opened, the dampers were set and the basement door was opened, so that the fan delivered outdoor air through the duct system, the fan continuing to run until 7 A.M. The fan delivery was 2180 c.f.m., or 9.2 air changes per hour.

17. Cooling with Room Cooling Units.—During these tests all of the windows in the Residence on the east, south, and west sides, which were exposed to the sun at some time in the day, were shaded with awnings, and the average occupancy was one person per room. A few preliminary tests were run to determine a practical operating schedule for the maintenance of indoor temperatures. The first method of operation involved maintaining the indoor dry-bulb temperature at a value of 10 deg. F. less than the outdoor dry-bulb temperature. The second method involved maintaining the schedule of desirable indoor temperatures given in Table 1, Section 12. In the case of the third method the indoor effective temperature was allowed to rise to 75 deg. F. before the cooling unit was started. This usually occurred when the indoor dry-bulb temperature reached a value between 78 and 82 deg. F., depending upon the prevailing relative humidity. The cooling unit was then started and the dry-bulb temperature was maintained constant during the remaining period of the test. As indicated by the discussion of test results in Section 43, this third method proved to be the most practical and was therefore adopted for all tests.

As stated in Section 9, cooling unit A was mounted on a portable platform scale which was sensitive to 0.01 lb., and both the ice meltage and the dehumidification rates were determined by direct weighing. At the start of each test the cooling unit was charged with ice, thoroughly pre-cooled, the water drained out, and the gross weight of the unit and ice was observed. It was then operated to maintain the required indoor dry-bulb temperature as determined by the third method previously discussed, and at the end of each hour the ice meltage and dehumidification were drained directly into a separate container and weighed, as shown in Fig. 9. At the same time, after the water had been drained out, the gross weight of the unit and ice was observed, and the net ice meltage was obtained from the difference in these hourly readings. Since the water drained from the unit each hour consisted of both the ice meltage and the moisture condensed out of the air, the net amount of moisture condensed was determined from the difference between the total weight of water collected and the net weight of ice melted. By this method of procedure both the
hourly rates of ice meltage and dehumidification were very satisfactorily obtained without handling the ice or the unit.

The majority of the tests with this unit were made with the unit located in the living room of the Residence, Fig. 2, but several tests were also made with it located in the dining room or in the hall. When located in the living room or in the dining room, the doors between that room and the hall were closed, confining the cooling entirely to the one room. When located in the hall, the doors to the living room and dining room were opened and the entire first story was cooled.

In the case of cooling unit B the amount of heat absorbed from the air could be determined either by weighing the meltage from the ice or by using the weight of water circulated through the coil and the temperature rise of the water. It was found, however, that although the temperatures of the entering and leaving water responded very quickly to the operation of the fan (Fig. 32), there was sufficient lag in these temperatures after the fan was started and stopped to make it difficult to accurately determine their average values. Since the total temperature rise was of such small magnitude that a slight variation in either temperature resulted in a comparatively large variation in the temperature rise, and hence in the computed heat quantities, this method of measuring the heat absorption was replaced by the more direct method of weighing the ice melted during a test.

Previous to the start of a test the water in the tank and system was pre-cooled. The weight of the initial charge of ice, and subsequent charges during a test were then accurately observed. At the end of a test the amount of ice remaining in the tank was deducted from the total amount charged, giving the net amount used in cooling and dehumidifying. The moisture condensing on the cooling coil was drained into a container and weighed directly.

All of the tests with unit B were conducted with the cooling section of the unit located near the north wall of the living room, on the first floor, as shown on the floor plan in Fig. 2. During some of the tests the door from the living room to the interconnecting hallway was opened, and the unit allowed to cool the entire first story, consisting of the living room, the hallway, the dining room, the breakfast nook and the kitchen, having a total net volume of 7300 cu. ft.

Unit B was operated on the same temperature schedule as that used for unit A. It was operated on the on and off principle, and was controlled by means of a room thermostat connected into the electric circuit to the fan.
IV. RESULTS OF TESTS ON COOLING WITH ICE

18. General Results.—In general, the results obtained from the plant used to cool the Research Residence by means of ice indicated that the ducts of a central forced-air heating system could be successfully adapted as a distributing system for cooled air without material alteration. In order to balance the plant when cooled air was circulated it was necessary to readjust the dampers so that more air was delivered to the second story than to the first. When the plant was balanced, however, the rooms were uniformly cooled with no indication of the existence of a pool of cold air near the floor, and as the sun changed position, the balance was not materially disturbed when the plant was controlled with a single thermostat placed in the dining room. An average temperature difference of approximately 3.5 deg. F. was observed between the floor and the ceiling, and 1.8 deg. F. between the floor and the breathing level, with the cooler air near the floor. The air was delivered from the registers at temperatures varying from 60 to 70 deg. F., depending on the outdoor temperature, and at average velocities varying from 50 to 450 f.p.m. measured with an anemometer placed one inch from the register face. No objectionable drafts were observed in the rooms, except in one case in which the air was delivered from a baseboard register at a velocity of approximately 450 f.p.m. Even in this case, conditions were easily corrected by employing a baffle in front of the register face in order to direct the air flow toward the ceiling. The best distribution of cooled air in the room was obtained with the wall register placed 7 ft. above the floor. In this case, the velocity of the air leaving the register was approximately 350 f.p.m. The gain in temperature of the air, from the fan inlet to the register faces did not exceed 3 deg. F., and condensation did not appear at any time on the furnace casing or basement duct system. No insulation was used on these surfaces. The basement temperature remained practically constant at 70 deg. F.

The cooling season at Urbana, Illinois, during the summer of 1932, extended from June 1 to October 1, with a total of 62 days on which the maximum temperature reached 85 deg. F. or more. This represented a total of 1471 degree-hours* above 85 deg. F. A total of 43.3 tons of ice was used for the season. This total was determined by weighing, and represents the actual ice used for the season as influenced by the actual conditions under which the different tests were

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*A degree-hour above 85 deg. F. is a temperature one degree in excess of 85 deg. F. maintained for one hour. For any given time the number of degree-hours above 85 deg. F. is the product of the time and the difference between 85 deg. F. and the mean temperature for the period.
<table>
<thead>
<tr>
<th>Date</th>
<th>Test No.</th>
<th>Start of Test</th>
<th>End of Test</th>
<th>Length of Test hours</th>
<th>Outdoor Temperature</th>
<th>Average Indoor Temperature</th>
<th>Average Temperature Difference In-Out</th>
<th>Average Humidity</th>
</tr>
</thead>
<tbody>
<tr>
<td>7-7</td>
<td>P-4</td>
<td>10:45 A.M.</td>
<td>6:45 P.M.</td>
<td>8.0</td>
<td>Maximum 88.0</td>
<td>Minimum 73.5</td>
<td>Average 83.8</td>
<td>Average 6.1</td>
</tr>
<tr>
<td>7-9</td>
<td>P-5</td>
<td>1:15 P.M.</td>
<td>11:15 P.M.</td>
<td>10.0</td>
<td>Maximum 87.0</td>
<td>Minimum 62.5</td>
<td>Average 71.2</td>
<td>Average 5.7</td>
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<tr>
<td>7-10</td>
<td>P-6</td>
<td>1:25 P.M.</td>
<td>5:25 P.M.</td>
<td>4.0</td>
<td>Maximum 90.5</td>
<td>Minimum 70.5</td>
<td>Average 80.3</td>
<td>Average 9.1</td>
</tr>
<tr>
<td>7-11</td>
<td>P-7</td>
<td>2:00 P.M.</td>
<td>6:00 P.M.</td>
<td>6.0</td>
<td>Maximum 85.0</td>
<td>Minimum 68.5</td>
<td>Average 76.2</td>
<td>Average 3.3</td>
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<tr>
<td>7-12</td>
<td>P-8</td>
<td>2:00 P.M.</td>
<td>9:30 P.M.</td>
<td>7.5</td>
<td>Maximum 98.5</td>
<td>Minimum 59.5</td>
<td>Average 84.4</td>
<td>Average 9.8</td>
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<td>7-13</td>
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<td>11:00 P.M.</td>
<td>13.5</td>
<td>Maximum 90.0</td>
<td>Minimum 70.0</td>
<td>Average 81.0</td>
<td>Average 12.6</td>
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<tr>
<td>7-14</td>
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<td>1:55 A.M.</td>
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<td>Minimum 73.0</td>
<td>Average 89.4</td>
<td>Average 8.5</td>
</tr>
<tr>
<td>7-15</td>
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<td>10:00 A.M.</td>
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<td>24.0</td>
<td>Maximum 103.0</td>
<td>Minimum 78.0</td>
<td>Average 88.8</td>
<td>Average 8.0</td>
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<td>7-16</td>
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<td>24.0</td>
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<td>Minimum 77.5</td>
<td>Average 83.2</td>
<td>Average 3.2</td>
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<td>Average 5.1</td>
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<td>24.0</td>
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<td>Maximum 87.0</td>
<td>Minimum 62.0</td>
<td>Average 81.8</td>
<td>Average 6.2</td>
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<tr>
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<td>Maximum 95.0</td>
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<td>Maximum 93.0</td>
<td>Minimum 71.5</td>
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<td>Average 1.9</td>
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</table>

Basement loss approximately 6600 B.t.u. per hr., or equal to an ice meltage of 45.8 lb. per hr. Total correction = (No. of hrs.) \times (45.8).

Test P-16 on 7-22-32 and Test F-23 on 8-17-32 omitted due to unfavorable change in weather conditions.

Test P-17, P-18, and P-19 on 7-26, 7-28, and 8-5-1932, respectively, omitted on account of incomplete data.
<table>
<thead>
<tr>
<th>Date</th>
<th>Test No.</th>
<th>Ice Meltage</th>
<th>Corrected for Basement Loss lb.</th>
<th>Dehumidification Total lb.</th>
<th>Hourly Heat Loads</th>
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</thead>
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<tr>
<td></td>
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<td>Equivalent Total lb.</td>
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<td>Total</td>
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<td>B.t.u. per Hour</td>
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<td></td>
<td>Total</td>
</tr>
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<td>B.t.u. per deg. per hr.</td>
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</table>

**Table 2.**—Continued

**Data and Results for Tests with Ice**

**Part a—Concluded**

<table>
<thead>
<tr>
<th>Date</th>
<th>Test No.</th>
<th>Ice Meltage</th>
<th>Corrected for Basement Loss lb.</th>
<th>Dehumidification Total lb.</th>
<th>Hourly Heat Loads</th>
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</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Equivalent Total lb.</td>
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<td>Total</td>
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<tr>
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<td>B.t.u. per Hour</td>
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<tr>
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<td></td>
<td>Total</td>
</tr>
<tr>
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<td></td>
<td></td>
<td></td>
<td>B.t.u. per deg. per hr.</td>
</tr>
</tbody>
</table>

Basement loss approximately 6000 B.t.u. per hr., or equal to an ice meltage of 45.8 lb. per hr. Total correction = (No. of hrs.) X (45.8).

Test P-16 on 7-22-32 and Test P-23 on 8-17-32 omitted due to unfavorable change in weather conditions.

Test P-17, P-18, and P-19 on 7-26, 7-28, and 8-5-1932, respectively, omitted on account of incomplete data.
### Table 2.—Concluded

**DATA AND RESULTS FOR TESTS WITH ICE**

(Part b)

<table>
<thead>
<tr>
<th>Test No.</th>
<th>Above 85 deg. F.</th>
<th>Above 90 deg. F.</th>
<th>Heat Absorbed During Overall Test Period, B.t.u.</th>
<th>Ratio of Moisture Load to Total Load</th>
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<tr>
<td></td>
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<td>Load Due to Moisture</td>
<td>Sensible</td>
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<td>P- 4</td>
<td>6.5</td>
<td>0</td>
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<tr>
<td>P- 7</td>
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<td>91 508</td>
<td>374 620</td>
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</table>
run. The maximum hourly rate of ice meltage observed was 220 lb., which occurred on Test P-29, when the Residence was not equipped with awnings, and for which the maximum outdoor temperature attained was 95 deg. F. The plant had been in operation for the previous 24 hours. The maximum meltage for any 24-hour period was 4243 lb. This occurred on Test P-11, when the Residence was equipped with awnings, and for which the maximum outdoor temperature attained was 103 deg. F. In this case, however, the plant had not been in operation for the 24 hours previous to the test.

Since the total volumes of air and water circulated were not varied for the different tests, the electrical input to the fan and pump motors remained practically constant. The rate of power consumption for the fan motor was 0.265 kw., and for the pump motor was 1.25 kw. The power required by the pump was influenced by the pressure of 23 lb. per square inch required to operate the spray heads in the ice tank. The use of these spray heads was found necessary in order to break the water up into a sufficiently fine spray to prevent the formation of craters in the ice. These craters filled with water and interfered with the accuracy in the determination of the ice meltage. The gain in accuracy resulting from the elimination of the craters, however, was obtained by a sacrifice in the operating efficiency of the plant as a whole, due to the pressure required by the spray heads. In a service plant, where the accurate determination of the ice meltage is not necessary, the spray heads could be dispensed with and the water showered over the ice in larger streams, thus reducing the power required for the pump.

The cooling load in the basement consisted of the heat gain of the coil, furnace casing and duct system, and the electric load imposed by the fan and pump motors and the lights. Separate determinations of this load indicated that it remained practically constant at 45.8 lb. of ice per hour, or 6600 B.t.u. per hour. A stand-by loss amounting to 100 lb. of ice per 24 hrs. occurred during the periods over which the plant was not in operation, but over which the ice tank remained charged with ice.

The principal results for all tests are given in Table 2. It has not been possible to use all of these for the curves and analyses. The lengths of tests were necessarily determined by the character of the weather. Furthermore, the total ice meltage in the case of short tests was largely influenced by the history of the Residence previous to the start of the test. On some of the shorter tests, the starting load formed an excessive proportion of the total ice meltage for the test.
Hence, many inconsistencies may be observed in comparisons based on the averages and totals for individual tests. It was therefore necessary, for the purpose of analysis, to group the tests made under comparable conditions, and in some cases the analysis has been based on the maximum load extending over a 4-hour period, including the maximum outdoor temperature. Some comparisons have been based on the load corrected for basement losses, since this represented the part of the load influenced by the particular factor under consideration. Furthermore, this basement load would vary for different plants and, by especial attention to features of design favoring higher operation efficiency rather than adaptability to accurate measurement of test data, could be very materially reduced.

In the test plant, the basement load was augmented by the necessity for the use of spray heads in the ice tank and by the use of more basement lights than would be demanded by a service plant. Also, during the tests, an effective temperature (see Appendix A) of approximately 73 deg. F. was maintained in the Residence. This condition was perfectly comfortable with the occupants clothed in light summer clothing. No member of the staff experienced any ill effect either from shock due to sudden changes in environment or from contracting a cold. However, the indications were that somewhat less cooling would be satisfactory for residence service. Just how much less is debatable at present. Hence, it is possible that by varying the design and operation from that used for the test plant, a reduction in ice meltage could be made. The figures presented, however, may be considered as representative, with a reasonable margin of safety, of the cooling requirements for a two-story structure similar in construction to the Research Residence.

19. Graphic Logs of Tests.—The operating characteristics of the cooling plant and a comparison of the actual and calculated cooling loads for the house can best be illustrated by the results obtained on typical days. For this purpose a complete graphic log covering a period of two 24-hour tests with the house equipped with awnings is shown in Fig. 10, and a similar graphic log for two 24-hour tests with the awnings removed is shown in Fig. 11. Figure 10 contains all of the data from the corresponding portion of the continuous chart previously mentioned, with the addition of effective temperatures, ice meltage, and calculated cooling load. In Fig. 11, the data on roof, attic, and ceiling temperatures have been omitted.

From Fig. 10 it is interesting to note that the temperature of the copper shingles on the roof attained a value as high as 169 deg. F.
At times values as high as 175 deg. F. were observed. The temperature of the air in the attic space rose to 112 deg. F. The time shown between the attainment of maximum temperature for the roof surface and that of maximum air temperature in the attic does not represent the time lag of the attic space, however. The data for the roof were taken from the south slope, while the attic space was in a north wing, and the west slope of the roof above it did not receive the sun until afternoon. The time at which the maximum was attained in the attic space corresponded approximately with that for the maximum outdoor temperature, thus indicating comparatively no lag. The temperature of the surface of the ceiling below the attic reached a maximum about
3 hours later than that of the attic itself. In this case, the ceiling was insulated with one inch of insulating quilt, and the surface temperature never rose more than approximately 3 deg. F. above the temperature of the air in the room below it. An uninsulated ceiling below an attic space becomes in effect a large panel radiator under summer conditions, thus adding to the discomfort of the occupants of the room, independently of the temperature of the air itself. The use of effective insulation in a ventilated attic reduces the temperature of the surface of the ceiling and thus directly promotes comfort. The fact that it also effects a desirable reduction in the required cooling load is worthy of favorable consideration, but this may be of secondary importance compared with the gain effected in comfort.

20. Daily Variation in Outdoor Temperatures.—It may be noted from the outdoor temperature curves shown in Figs. 10 and 11 that the difference in temperature between the maximum in the daytime and the minimum during the preceding night ranged from 19 to 28 deg. F., and that for a period of six hours, or more, the outdoor temperature was at least 20 deg. F. below the maximum attained during the same
day. An analysis of all of the daily temperatures during the months from June 1 to October 1 proved that, with very few exceptions, a difference of at least 20 deg. F. was exhibited. The exceptions occurred when the maximum was 85 deg. F. or less. The average difference when the maximum was from 75 to 85 deg. F. was 19 deg. F. For maximums between 85 and 95 deg. F., the average difference was 23 deg. F., and for maximums above 95 deg. F. it was 25 deg. F. This indicated that for considerable periods at night the outdoor air constituted a reservoir from which air could be drawn in order to cool the residence at night, and reduce the cooling load required during the following day. For a large portion of the time, enough cooling could probably be effected in this way to make it unnecessary to start the cooling plant the next day, particularly in the case of an insulated structure. It should be emphasized, however, that in order that this method might be effective, a fan large enough to circulate sufficient outdoor air to cool the walls and contents of the structure, and not merely to cool the air inside of the building, would be required.

21. Relative Humidities and Dehumidification.—Figure 10 illustrates the effect of the indoor relative humidity on the starting load on the plant. The plant was not operated on the day previous to the start of Test P-11. The indoor relative humidity rose gradually until it reached a value of 57 per cent at 10 A.M., July 15; at this time the outdoor temperature was 92 deg. F., and the indoor temperature 81 deg. F., corresponding to an indoor effective temperature of 76 deg. F. (see Appendix A.) At this time the cooling plant was started for Test P-11. At the end of an hour’s operation the relative humidity had dropped to about 46.5 per cent, and at the end of four hours had become stabilized at about 45 per cent. Meanwhile the indoor effective temperature dropped from 76 deg. F. to 73 deg. F., while the indoor dry-bulb temperature remained comparatively constant at about 81 deg. F. The dehumidification, appearing as condensation on the coils, decreased from about 7 lb. per hour to 5.5 lb. per hour. Data on other tests showed decreases from as high as 9 lb. per hour to as low as 4.5 lb. per hour within two hours after the start of the test. The average dehumidification load ranged from approximately 3000 to 8000 B.t.u. per hour, and formed a considerable part of the total load. It averaged about 4000 B.t.u. per hour, and constituted approximately one-fifth of the total load. A number of calculations, based on the condensation weighed after the indoor relative humidity had become stabilized, and at outdoor temperatures near the peak, gave results for
the number of air changes ranging from 0.7 to 1.0 air change per hour. These results were subject to changes in moisture content of the walls, floors, and furniture, but the consistency of the weighed amounts, and the fact that long periods of uniform indoor relative humidities were maintained, indicate that three-quarters of an air change per hour represents the amount of air infiltration with a fair degree of accuracy.

22. Comparison of Actual and Calculated Hourly Cooling Loads.—
The variation in the actual cooling load, as determined by ice meltage from hour to hour, may be observed from Figs. 10 and 11. From these curves it may be noted that while the outdoor air temperature reached a maximum at about 4 p.m., the actual load reached a maximum from two to four hours later. This was particularly true if the plant had been operated during the 24 hours previous to the start of a test. The effect of the sun on the load may be observed by comparing the shape of the load curves in Figs. 10 and 11. In the case of Test P-12, shown in Fig. 10, for which the Residence was equipped with awnings, the load increased gradually to a peak occurring four hours later than the time for the maximum temperature for the day. In the case of Tests P-28 and P-29, shown in Fig. 11, the load rose suddenly, reaching one peak at about noon, then declined somewhat, and increased to reach a second peak at 5:30 p.m. The first peak was probably caused by the sun, which showed its maximum effect at 10 a.m., and the second peak by a combination of the sun, which reached a secondary maximum at 2 p.m., and the heat lag of the structure. The total heat transmitted through the windows, based on the glass area and exposure of the windows, and on the curves given by F. C. Houghten,* and others, was as follows: 6 A.M., 1,145; 8 A.M., 1,380; 10 A.M., 1,750; 12 P.M., 1,780; 2 P.M., 1,455; 4 P.M., 1,170; 6 P.M., 970 B.t.u. per hour. The detailed method for calculating these items is given in Appendix B, and the curves used are shown in Figs. 38 and 39. The heat transmitted through the windows was 97 per cent of the values read from Fig. 39. A limited number of observations on the solar intensity, a sample of which is shown in Fig. 12, indicated that the solar intensity at Urbana, Illinois, which is in practically the same latitude as Pittsburgh, Pennsylvania, or approximately 40 deg. N., agreed reasonably well with the results obtained at Pittsburgh by the authors previously cited. Their curves were therefore accepted as applying at Urbana. The observations of solar intensity were made with

an Abbot silver disk pyrheliometer,* a description of which is given by J. H. Walker† and others. A few observations made through the windows of the Residence indicated that a single thickness of glass intercepted 17 per cent of the total radiation from the sun. This value agreed closely with that obtained by Walker† and others.

A comparison of the actual cooling load, based on measured ice meltage, and the calculated cooling load reduced to terms of ice melt-

*Loaned by the Smithsonian Institution, Washington, D. C.
age, may be made from Figs. 10 and 11. This comparison should be based on the cooling load corrected for loss in the basement, since the calculated load applies only to the portion of the Residence above the basement. The calculated load at any given time consisted of the sum of the sun load for both walls and windows, the dehumidification load, the load due to occupancy, the load due to sensible heat from air infiltration, the load resulting from electrical input to lights and fan motor, and the heat transmission load. The latter was based on the actual outdoor-indoor temperature difference shown by the curves at the time under consideration and the heat transmission coefficients applying to the walls, floors, and ceilings of the Residence. The dehumidification load, and the load due to sensible heat gain of the air inleakage resulting from infiltration were based on the existing indoor and outdoor temperatures and relative humidities, and on the assumption of three-quarters of an air change per hour for the 14 170 cu. ft. of space cooled. Occupancy was assumed to be four people, or 1600 B.t.u. per hour. The methods employed in the calculations are given in detail in Appendix B.

From the curves in Figs. 10 and 11, it is evident that the heat lag of the structure operates in such a way as to make it almost futile to attempt to calculate the cooling load for the building at any specified time. The actual load never attained a value as high as the calculated maximum load corresponding to the maximum outdoor temperature. Furthermore, the true load at the time that the maximum load actually occurred was much greater than the calculated load corresponding to this time. The curves all showed one point in common, however. That is, the calculated load at some time between 5 P.M. and 7 P.M. agreed with the actual load at this time, and corresponded to a value only slightly less than the actual maximum load. It is therefore possible that the maximum load for the purpose of design could be based on the sun effect and the probable outdoor temperature at 6 P.M. for any given locality. Whether or not this is a feasible method of calculation, and whether it would apply to structures differing in character from the Research Residence is admittedly somewhat problematical.

23. Effect of Outdoor Temperature on Cooling Load.—An attempt to correlate the cooling load with the average outdoor-indoor temperature differences shown in Table 2 resulted in many inconsistencies, due to the fact that this average temperature difference was influenced by the character of the daily temperature curve, and the relation between
the maximum outdoor temperature and the balance of the outdoor temperatures. Hence it was possible to obtain approximately the same average temperature with radically different maximum temperatures. Also the total or average cooling load was dependent on the heat lag and previous history of the structure, and on the portion of the day over which it was possible to run the test. Therefore, a number of tests were selected in which the start of the test occurred a sufficient length of time previous to that for attainment of maximum outdoor temperature to permit conditions to become stabilized before the maximum outdoor temperature was reached. For this group of tests, the cooling load per degree difference in temperature was based on the average outdoor-indoor temperature difference for the four hours included between two hours before and two hours after the time for the maximum, or peak temperature, and the average rate of ice meltage for the same four-hour period. The results are shown in Fig. 13.

From Fig. 13 it is evident that the cooling load per degree difference in temperature, and therefore per degree-hour, was not constant, but increased as the outdoor temperature increased. This was true both for the sensible and total cooling loads, the latter including the heat absorbed by dehumidification, and was also true whether or not the house was equipped with awnings. Since the actual cooling load per degree-hour is not constant, any method of calculating the total seasonal cooling load based on the B.t.u. per degree-hour and the total number of degree-hours in the season is incorrect unless some ac-
count is taken of the increase in cooling load per degree difference in temperature as the actual outdoor temperature increases.

24. Effect of Awnings.—The reduction in total cooling load effected by the use of awnings on all east, south, and west exposures may be obtained from the curves shown in Fig. 13. These curves overlap over a range of outdoor temperatures of from 93 deg. F. to 97 deg. F., and it may be observed that over this range the reduction in both total and sensible cooling loads was approximately 32 per cent. This comparison has been made between the cooling loads corrected for basement loss, since the only portion of the load affected by the awnings was that above the basement, and the basement load would vary with different individual plants.

It should be noted that the tests on the Residence without awnings, P-21 to P-30, inclusive, in Table 2, were conducted during the month of August, while those with awnings were conducted during the month of July. A calculation of the total amount of heat received by radiation from the sun on all of the windows in the Residence (shown in Appendix B, Fig. 39) indicates that the possible heat from this source was somewhat greater during August than during July. Of this available heat approximately 97 per cent would be transmitted through windows unprotected by awnings, and 28 per cent through windows protected by awnings. Hence, slight differences would be occasioned by whether the tests with awnings were conducted in July or in August. However, the sun load obtained without awnings during August was probably somewhat greater than it would have been if these tests had been run simultaneously with the tests with awnings in July. The comparison that has been made is equivalent to assuming that both series of tests were run during August. If it had been possible to compare two such series of tests made in July, the sun load with awnings would have been practically the same, but the sun load without awnings would have been somewhat less than the one actually obtained in August. To this extent, therefore, the comparison as made has tended to favor the awnings. It is also possible that the results may be similarly influenced by the variations in the sun effect on the walls from month to month, but it is not possible to separate this for the purpose of analysis. Hence, it is evident that the actual percentage of saving effected by awnings is dependent on what parts of the season are selected for comparison, but it is probable that the 32 per cent based on the comparison as made is representative of the average reduction of load resulting from the use of awnings. This figure is
Referring to Figs. 10 and 11, it may also be noted that the awnings changed the character of the daily cooling load curve, and reduced the extent of the period of maximum cooling load from a duration of approximately 6 hours to one of 2 hours.

25. Estimated Seasonal Loads.—During each series of tests, both with and without awnings, data were obtained over a sufficiently wide range of weather conditions to establish the relation existing between the cooling load, or heat absorbed from the Residence, and the severity of the weather expressed as the degree-hours above 85 deg. F. per day. Curves indicating this relation for both sensible and total heat are shown in Fig. 14. From these curves it may be observed that the heat
absorbed can best be represented as a linear function of the degree-hours per day. The curves were based on the number of degree-hours above 85 deg. F. because experience gained in operation proved that it was not necessary to start the plant on days for which the maximum outdoor temperature did not exceed 85 deg. F. As soon as the plant was started, however, a material ice meltage would be obtained, and for this reason the curves do not indicate zero ice meltage corresponding with zero degree-hours above 85 deg. F.

The difference between the curves for total heat and those for sensible heat in the lower portion of Fig. 14 represents the latent heat load, or the heat absorbed in dehumidification. Without awnings the latent heat load varied from 20 to 21 per cent of the total load, and with awnings it varied from 22 to 26 per cent of the total. These figures are in substantial agreement with the average values given in Section 21.

The season of 1932 consisted of 62 days on which the maximum temperature attained or exceeded 85 deg. F., giving a total of 1471 degree-hours above 85 deg. F. The distribution of these days and degree-hours is more fully discussed in Section 49.

Owing to the fact that two separate series of tests, with and without awnings, were run during the same season, direct experimental results on the total seasonal cooling load for either one of these conditions alone are not available. These seasonal loads, however, can be estimated from the curves in Fig. 14. Since the greater numbers of degree-hours above 85 deg. F. per day correspond with the higher maximum outdoor temperatures, and the ice meltages determining the curves were obtained from the actual weights for the corresponding weather, the curves reflect the increase in cooling load with higher outdoor temperatures which was shown in Fig. 13 and discussed in Section 23.

In order to estimate the seasonal cooling load, the number of degree-hours above 85 deg. F. was tabulated for each day for the months of May, June, July, August and September, together with the corresponding ice meltages read from the curves for awnings and for no awnings shown in Fig. 14. This tabulation showed that there were no days in May for which the degree-hours above 85 deg. F. were above zero. It also showed that there was one day on which the maximum temperature was exactly 85 deg. F., resulting in zero degree-hours, and that there were 10 days during the season on which the degree-hours were not greater than 2 and which were preceded by days on which the degree-hours were zero. Experience had proved that the
indoor temperature would not rise to 80 deg. F. on these days, and that
the plant would not be started. These days were therefore disregarded
and a total of 51 days was used in computing the seasonal meltage.
The summation of all of the values read from the curve for no awnings
indicated that if the plant had been operated all season without awn-
ings on the windows, the total ice meltage would have been 59.7 tons.
The summation of all of the values read from the curve for awnings
showed that if awnings had been used for the whole season, the total
ice meltage would have been 40.3 tons. Thus the use of awnings would
have effected a seasonal saving of approximately 32.5 per cent.

Under the actual operating conditions, the series of tests with the
Residence equipped with awnings extended from June 1 to August 15,
while the series of tests with the Residence not equipped with awnings
extended from August 15 to September 15. A summation of the tab-
ulated values corresponding to these two parts of the season, and read
from the respective curves for no awnings and for awnings, gave an
estimated total ice meltage of 44.1 tons for the season under the actual
conditions of operation. This agrees very closely with the total of 43.3
tons of ice actually used during the season, and confirms the validity
of this method for estimating the total cooling load for a season from
the results obtained over a wide range of weather conditions during a
part of a season.

By plotting a similar set of curves showing the relation between the
hours of fan operation and the degree-hours above 85 deg. F., and
following the same procedure as that outlined for estimating the sea-
sonal ice meltage, it was estimated that the fan would have run a
total of 787 hours for the season with the Residence not equipped with
awnings. If the Residence had been equipped with awnings during the
whole season the fan would have run 563 hours. For the season as run
with two series of tests, with and without awnings, it would have
operated 616 hours. Since both the fan and the pump were running
only when the plant was in operation, the hours of running for the
pump would have been the same as those for the fan.

26. Cost of Operation.—Owing to wide variations in seasonal de-
mands and in local utility rates, a discussion of seasonal costs, particu-
larly from the comparative standpoint, is not conclusive unless
unavoidable differences in existing conditions are fully recognized.
With due appreciation of these limitations, however, operating costs
are undoubtedly of some interest. The following operating costs are
based on local rates of $4.00 a ton for ice and $0.031 a kw-hr. for
electricity. The fan in the forced-air system used electricity at an average rate of 0.265 kw. If it had not been necessary to use high pressure spray heads to overcome experimental difficulties, as explained in Section 18, a reasonable electrical input to the circulating pump would have been 0.5 kw.

Based on these rates, the seasonal cost for operation under the actual running conditions, with the Residence equipped with awnings for a part of the season only, was $176.40 for ice, $5.07 for the fan, and $9.55 for the pump, or a total cost of $191.02. If the Residence had been operated all season without awnings, the operating cost would have been $238.80 for ice, $6.47 for the fan, and $12.20 for the pump, or a total operating cost of $257.47. If the Residence had been operated all season equipped with awnings, the operating cost would have been $161.20 for ice, $4.63 for the fan, and $8.72 for the pump, or a total operating cost of $174.55. Thus it may be noted that a reduction in total cost of approximately 32 per cent would have been effected by the use of awnings.

During the periods over which the plant operated at maximum capacity, 220 lb., or 0.11 tons, of ice were melted per hour. This is equivalent to 31 700 B.t.u. per hr. or 2.64 tons of refrigeration. The cost per hour for operating the plant at maximum capacity was, therefore, 44 cents for ice, 1.55 cents for the pump, and 0.82 cents for the fan. This represents a total cost of 46.37 cents per hour, or a unit cost of 17.6 cents per ton of refrigeration per hour.

27. Conclusions.—From the results of the tests in which ice was used for cooling the Residence, the following conclusions may be drawn, subject to the limitations of the conditions under which the tests were run:

(1) The ducts and registers of a central forced-air heating system can be successfully adapted as a distributing system for cooled air in the summertime without material alterations.

(2) There is no tendency for a pool of cold air to collect near the floor if the temperature of the air entering the room through the registers is not lower than 60 deg. F.

(3) Baseboard registers can be used without objectionable drafts resulting if the velocity of the air at the register face is below 300 ft. per min. although wall registers located 7 ft. above the floor are preferable to baseboard registers with perforated grilles.

(4) A two-story building of the type of the Research Residence may require the equivalent of two tons of ice in 24 hours, including
the basement load, on days when the maximum outdoor temperature reaches 100 deg. F. if an effective temperature of approximately 73 deg. F. is maintained indoors.

(5) An effective temperature somewhat higher than 73 deg. F. may be satisfactory for residence service, and in this case more conservative results for the cooling load can be obtained.

(6) The use of awnings at all windows on east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.

(7) The cooling load per degree difference in temperature is not constant, but increases as the outdoor temperature increases.

(8) The heat lag of the building complicates the estimation of the cooling load under any specified conditions, and makes such estimates based on the usual methods of computation of doubtful value.

V. RESULTS OF TESTS ON COOLING WITH AIR FROM OUTDOORS AT NIGHT

28. Preliminary Statement.—The discussion in Chapter IV, in which it was shown that 43.3 tons of ice were used in cooling the Research Residence during the summer of 1932, indicated that complete artificial cooling of this class of structure by such cooling agents as cold water or ice, or by mechanical or chemical methods, may be a very expensive process unless some modification is made in the conventional or usual methods of operation or in the structure itself. The most obvious modifications in the structure consist of the use of insulation in the walls and ceilings, and the use of awnings at the sun-exposed windows. The use of awnings was investigated during the summer of 1932, (see Section 24) and the use of insulation was considered outside of the immediate scope of this investigation. Thus attention was directed toward some modification in the operating schedule.

For the investigation during the summer of 1932 the windows and outside doors were kept closed as much as possible, and the ice plant was started whenever the effective temperature indoors reached a value of 75 deg. F. During the periods of operation, both day and night, the effective temperature was maintained at approximately 73 deg. F., corresponding to a dry-bulb temperature of from 78 to 80 deg. F., and a relative humidity of about 45 per cent. However, a study of the temperature cycles discussed in Section 20 indicated that the
minimum outdoor temperature at night was always approximately 20 deg. F. lower than the maximum attained during the day, and that during a period extending from 9 P.M. to 6 A.M. the outdoor temperature was from 7.5 to 10 deg. F. lower than that indoors. During this period the temperature of the inside surfaces of the walls was usually less than that of the inside air, so that all of the cooling of these surfaces had to take place by conduction through the wall to the outside air. Hence it seemed evident that if sufficient air at a temperature lower than that of the inside wall surface could be introduced, these surfaces could cool from the inside, and, if the process was continued, the whole structure could be cooled, and the next day started with the entire structure filled with cool air, with the inside wall surfaces and contents at a temperature lower than would otherwise exist, and with the heat-absorbing capacity of the material in the structure increased as a result of the lower wall surface temperatures. The investigation for the summer of 1933 was, therefore, undertaken with the object of determining to what extent the circulation of air taken from outdoors at night could be used to supplement ice cooling during the day, thus reducing the amount of ice required, and to what extent it could be used to eliminate the necessity for any additional cooling during the day.

29. Influence of Methods of Operation on Conditions at Night.—A study of the results of the investigation of cooling with outdoor air at night involves the recognition of three separate natural subdivisions or objectives. These are (1) the study of the conditions at night, between the times of opening and of closing the windows, as influenced by the weather and by the different methods of operation, (2) the study of the conditions in the house the next day resulting from the weather and from the conditions existing at 6 A.M. when the windows were closed, and (3) the prediction of the effectiveness of a given method of operation, or of the cooling required, from the weather reports of a particular summer season.

Table 3 shows the volume of air delivered per minute and the number of air changes per hour produced by the different fan arrangements used, together with the power required to operate the fans. Figure 15 shows characteristic curves for outdoor temperatures, indoor temperatures and indoor relative humidities for the least favorable and most favorable methods of operation for two similar days. In the first case, shown by the upper curves in Fig. 15, eight windows were partly opened at 9 P.M. and no fan was used. During the early morning hours of June 18 the outdoor temperature reached a minimum of
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65.0 deg. F., while the indoor air temperature did not fall below 75.5 deg. F., or a difference of 10.5 deg. F. was shown. During these hours the indoor air temperature remained equal to, or slightly above, the temperature of the inside surface of the exposed walls, thus permitting no transfer of heat from the walls except by conduction to the outdoor air. The temperature of the air in the house rose to 84.5 deg. F. on June 18 following this cycle of operation at night.

In the second case, shown by the lower curves in Fig. 15, the lower sash of all of the windows, which were double hung, was opened to the fullest extent at 6 P.M. and the attic fan was used. The minimum outdoor temperature in the early morning hours of August 12 was 64.0 deg. F., nearly the same as in the first case, but the indoor air temperature dropped to 68.0 deg. F. as compared with 75.5 deg. F. for the first case. That is, the difference between indoor and outdoor
temperatures was 4.0 deg. F. as compared with 10.5 deg. F. During the whole night the temperature of the indoor air was distinctly below that of the inside surface of the exposed walls, thus permitting heat to be transferred from the walls to the air that was being removed from the house. The temperature of the air in the house next day rose to only 79.5 deg. F. before the windows were opened as compared to 84.5 deg. F. for the previous case.

In all cases, the temperature of the surface of the first-story ceilings and of the furniture remained practically the same as that of the indoor air of the room involved. The temperature of the surface of the second-story ceilings was usually from one to three degrees higher than that of the air in the room (see Fig. 21).

From Fig. 15 it may be observed that conditions in the house at 6 A.M., the time at which the windows were closed, were directly dependent on the weather and the method of operation at night between the times of opening and closing the windows. Since during the daytime the house was always closed and no fan was operated, conditions in the house the following day were determined by the conditions existing at 6 A.M. and by the weather during the day. Inasmuch as these conditions at 6 A.M. were dependent on the method of night operation, the conditions next day were a reflection of the method of operation. But since the conditions at 6 A.M. could be considered as having been established by any independent means, and the resulting conditions during the day would be the same, independent of the means required to establish those at 6 A.M., for the purposes of analysis of the different methods of night operation the daytime conditions may be regarded as not directly dependent on these methods, and the attention may be concentrated on the part of the cycle included between the times of opening and closing the windows.

The effectiveness of any given method of night operation is directly indicated by the ratio of the rate of decrease in the indoor temperature to the rate of decrease in the outdoor temperature, and the relative effectiveness of the different methods can be compared by comparing these ratios. However, the rates of decrease are also proportional to the drops in temperature from the time that the indoor and outdoor air attain the same temperature, as represented by the crossing point of the two air-temperature curves shown in Fig. 15, to the times at which each of these two air temperatures becomes a minimum. These temperature drops are also shown diagrammatically as A and B in the inserts in Figs. 17, 18 and 19. Since these temperature drops were useful in subsequent calculations, they were used as the basis of com-
comparison, instead of the ratios of the rates of temperature decrease. These temperature drops are further useful in that, with a known outdoor minimum temperature, and a known or assumed temperature at the crossing point, they serve to establish the minimum indoor temperature that would be attained. That this minimum indoor temperature was also the same as the indoor temperature existing at the time that the windows were closed at 6 A.M. is shown by Fig. 16.

A comparison of the temperature drops, $A$ and $B$, from the crossing point of the indoor and outdoor temperature curves to the minimum outdoor and indoor temperatures for the different methods of night operation is shown in Fig. 17. In comparing different methods of night cooling, temperature drop curves having greater slopes indicate greater effectiveness of cooling, since the maximum possible effectiveness would be attained when the drop $B$ became equal to the drop $A$, or when the indoor temperature was reduced to the same value as the outdoor temperature. Thus the maximum effectiveness would be represented
by a 45 degree line, or one having a slope of 1.0 when the two temperature drops are plotted to the same scale, and any effectiveness less than the maximum would be represented by a line of correspondingly less slope. The time of the crossing point was not always coincident with the time at which the windows were opened, and the temperature at the crossing point was somewhat influenced by the history between these two times. Hence, in order to obtain the effect of the fan alone, the comparison must be made between two series for which the windows were opened at the same time, and which differed only in that the fan was used for one and not for the other. Other comparisons are valid, however, in that the temperature drop curves reflect the whole history from the time of opening the windows, and this whole history is inherent in the particular series under consideration. A complete description of the different series of tests is given in Sections 13 and 14.

From Fig. 17, curve No. 1, it is evident that Series 1-33 in which eight windows were partly opened at 9 P.M. was the least favorable,
since it gave the least drop in indoor temperature for a given drop in outdoor temperature. The house was stuffy and uncomfortable all night, and the temperature was far too high for comfort the next day. A companion series, Series 3-33, curve No. 2, run with the same schedule of window opening and with the basement fan delivering 1740 cu. ft. of air per minute taken from outdoors, resulted in marked improvement, but did not entirely correct conditions.

From the experience with Series 1-33, it became evident that some advantage was to be gained by opening all of the windows, and also opening them earlier, even if the outdoor temperature was somewhat higher than that indoors. Series 2-33 was therefore run without a fan, but with all of the windows in the house opened to the fullest extent possible at 6 p.m., and in addition the attic door opened in order to obtain the advantage of the full chimney action of the heated air in the house. As shown by Fig. 17, curve No. 4, this resulted in very marked improvement over the results from Series 1-33, curve No. 1, and in considerable improvement over those from Series 3-33, shown in curve No. 2. Conditions in the house were quite comfortable at night, and the indoor temperatures the next day were not unreasonable.

It is probable that the Research Residence was better adapted to natural ventilation than the average type of residence would be. The attic was equivalent to a full third story, and the dormer windows extended nearly the full height of the story. A 2.5 ft. by 6.5 ft. door at the head of the stairs provided ample area for the passage of air into the attic. Hence conditions were particularly favorable for obtaining and utilizing a relatively large chimney effect. In houses with only a small trap door in the second-story ceiling and small attic windows, this chimney effect might be greatly reduced.

Series 5-33 was run on the same schedule of window opening as Series 2-33, but with the basement fan delivering 2120 cu. ft. of air per minute taken from outdoors. The results from this arrangement, shown in curve No. 3, were not quite as favorable as those obtained with Series 2-33, curve No. 4, for which no fan was used. This can not be explained on the basis that wind movement aided the natural ventilation in Series 2-33. The different points on the curves represented days covering the whole range of wind movement at night and sun effect by day. During most of the season the nights were comparatively calm, and the wind velocity varied from 2 to 7 miles per hour. Except for an occasional storm the days had a high percentage of sunshine. Hence, the most probable explanation appears to be that more air was circulated through the house as a whole by full natural
ventilation than by the basement fan. It is also possible that in the latter case some of the air short-circuited out of the windows, although no positive evidence could be obtained on this point owing to the low air velocity at the windows.

In Series 4-33 an attempt was made to take advantage of the reservoir of cooler air existing in the basement by recirculating this basement air through the house for about one hour before opening the windows. This series is comparable with Series 5-33 and the two points representing these tests are shown in Fig. 17. This method of operation represented a slight gain over the results from Series 5-33, curve No. 3, but the temperature of the basement air soon rose to practically the same as that upstairs, and the amount of gain cannot be regarded as sufficient to offset the complication in the operating routine. Furthermore, this method of operation appeared to accentuate odors. With all of the different test series, there was a tendency for odors to become slightly noticeable in the afternoon, particularly on the second story, but not to the extent of becoming objectionable.

The most favorable results, shown in Fig. 17, curve No. 5, were obtained from Series 7-33 and 8-33 in which the attic fan, drawing 3940 cu. ft. of air per minute into the first and second stories, was used. No difference was observed between series 7-33 for which the lower sash were raised half way, and Series 8-33 for which they were raised all of the way. From Fig. 17 it is apparent that there was not a great amount of difference between the three most favorable methods of operation, and that, even with the 16.6 air changes per hour produced by the attic fan, the indoor temperature was not reduced to the same value as that of the outdoor air. The latter condition would be represented by the line designated as the theoretical maximum drop. The results of a separate study of the effect of the attic fan in ventilating the attic during the day, and of the heat removal from the attic during the day and from the house during the night are discussed in Section 31.

The results of the tests shown in Fig. 17 indicate that considerable benefit may be obtained from the use of an attic fan drawing a generous amount of air into the house through the open windows at night. In a house similar to the Research Residence, having two full stories and an attic having dormer windows and a large attic door, practically as much benefit may be obtained without any fan by opening all of the windows and the door leading into the attic. In case it is not desirable to open a large proportion of the windows at night considerable benefit may be obtained from the use of either a fan in the attic or one
## TABLE 3
### FAN DATA

<table>
<thead>
<tr>
<th>Test Series No.</th>
<th>Designation of Fan</th>
<th>Windows</th>
<th>Time of Operation</th>
<th>Air Volume c.f.m.*</th>
<th>Air Changes per Hour</th>
<th>Kilowatt hours per 12 Hours</th>
<th>Speed r.p.m.</th>
</tr>
</thead>
<tbody>
<tr>
<td>3-33</td>
<td>Basement</td>
<td>Eight windows on two stories open</td>
<td>9 P.M. to 6 A.M., Night Air</td>
<td>1740</td>
<td>7.4</td>
<td>4.50</td>
<td>567</td>
</tr>
<tr>
<td>4-33, 5-33, and 6-33</td>
<td>Basement</td>
<td>All windows open</td>
<td>6 P.M. to 6 A.M., Night Air</td>
<td>2130</td>
<td>9.0</td>
<td>5.40</td>
<td>567</td>
</tr>
<tr>
<td>4-33 and 6-33</td>
<td>Basement</td>
<td>Windows closed; recirculation</td>
<td>Daytime</td>
<td>1468</td>
<td>6.2</td>
<td>4.20</td>
<td>567</td>
</tr>
<tr>
<td>7-33, 8-33, and 9-33</td>
<td>Attic</td>
<td>All windows open</td>
<td>6 P.M. to 6 A.M., Night Air</td>
<td>3940</td>
<td>16.6</td>
<td>3.85</td>
<td>845</td>
</tr>
<tr>
<td>10-33</td>
<td>Attic</td>
<td>Second-story windows open only</td>
<td>6 P.M. to 6 A.M., Night Air</td>
<td>3940</td>
<td>33.2†</td>
<td>3.85</td>
<td>845</td>
</tr>
<tr>
<td>5-34</td>
<td>Basement</td>
<td>Second-story windows open only</td>
<td>6 P.M. to 6 A.M., Night Air</td>
<td>2180</td>
<td>9.2</td>
<td>4.92</td>
<td>568</td>
</tr>
<tr>
<td>6-34</td>
<td>Basement</td>
<td>Second-story windows open only</td>
<td>6 P.M. to 7 A.M., Night Air</td>
<td>2180</td>
<td>18.4†</td>
<td>4.92</td>
<td>568</td>
</tr>
<tr>
<td>5-34</td>
<td>Basement</td>
<td>Second-story windows open only</td>
<td>6 P.M. to 6 A.M., Night Air</td>
<td>2180</td>
<td>18.4†</td>
<td>4.92</td>
<td>568</td>
</tr>
<tr>
<td>6-34</td>
<td>Basement</td>
<td>Second-story windows open only</td>
<td>6 P.M. to 7 A.M., Night Air</td>
<td>2180</td>
<td>18.4†</td>
<td>4.92</td>
<td>568</td>
</tr>
</tbody>
</table>

*Based on density of air of 0.0749 lb. per cu. ft.
†First-story windows closed and only second story regarded as space affected.
installed in connection with a forced-air heating system and drawing air from the outdoors. The power required, as shown by Table 3, ranged from 3.85 to 5.40 kw-hr. for 12 hours of operation at night.

Since it is undesirable in the average city residence to leave the first-story windows open all night, two series of tests on which the open windows were confined to the second story only were run during the summer of 1933. These tests included Series 10-33 with the attic fan, and Series 11-33 with natural ventilation. Furthermore, since it also seemed more practical from the standpoint of the householder to employ the fan in the forced-air heating system, if one were available, rather than to install additional fans to circulate a greater volume of air, three more series of tests for which the windows were opened on the second story alone were run with the basement fan during the summer of 1934. These tests included Series 3-34, 5-34, and 6-34. In the case of the attic fan 3940 cu. ft. of air per minute were drawn in through the second-story windows giving 33.2 air changes per hour based on the cubic contents of the second story. The basement fan delivered 2180 cu. ft. per minute or 18.4 air changes per hour based on the cubic contents of the second story. The resulting temperature drops for both first and second stories, obtained from the five series of tests, are shown in Fig. 18. Comparing curve No. 7 with curve No. 2, curve No. 5 with curve No. 4, and curve No. 3 with curve No. 1 in Fig. 18, it is evident that in every case, when the windows were opened on the second story only, more cooling was effected on the second story than on the first.

A comparison of curve No. 3 in Fig. 18 with curve No. 4 in Fig. 17 indicates that, with natural ventilation, cooling on the second story alone with the windows open on the second story only was not as effective as cooling the house as a whole with all of the windows open. In the latter case the whole height of the house was effective in producing the motive head, and, as a result, a large quantity of air was drawn in through the open windows. When the first-story windows were closed, the motive head was reduced by the height of the first story, and the amount of air drawn through the house was correspondingly reduced. Apparently, under these conditions, relatively less air was also drawn through the second story only than was the case when all of the windows on both stories were opened, and the cooling on the second story only was less effective than it was for the house as a whole with all of the windows opened. As a result, curve No. 1 in Fig. 18 indicates that very little cooling was effected on the first story, and
that the temperature drop was small as compared with the average temperature drop for the house as a whole when natural ventilation was used on both stories, as shown by curve No. 4 in Fig. 17.

Curves Nos. 3 and 7 in Fig. 18 show that when the attic fan was used in connection with the second story alone, there was more marked improvement on this story over natural ventilation than there was in the case of the house as a whole when the attic fan was used in connection with both stories. This is indicated by the fact that the distance between curves Nos. 3 and 7 is greater than that between the corresponding curves, Nos. 4 and 5 in Fig. 17, and also by the fact that curve No. 7 in Fig. 18 lies above curve No. 6, which has been transferred to Fig. 18 from curve No. 5, Fig. 17. Considering the effect on the first story only, curve No. 2 in Fig. 18 shows that, while the use of the attic fan with only the second-story windows open resulted in some improvement, the temperature drop was still not comparable with that occurring when either the fan or natural ventilation was used in connection with both stories.
Comparison of curve No. 5 in Fig. 18 with curve No. 3 in Fig. 17 indicates that the cooling effect on the second story resulting from the use of the basement fan with only the second-story windows open, was practically the same as the cooling effect on the house as a whole resulting from the use of the basement fan with all windows on both stories open. Furthermore, comparison of curve No. 4 in Fig. 18 with curves Nos. 1 and 2 shows that the basement fan with the windows on only the second story opened accomplished more cooling on the first story than was effected by either natural ventilation or the attic fan under the same conditions; and comparison with curve No. 3 shows that it accomplished more cooling on the first story than was effected on the second story alone with natural ventilation. This relatively high effectiveness in cooling the first story with the basement fan probably arose from the fact that, even though the first-story windows were closed, part of the cool air taken from outdoors was delivered through the first-story registers, and hence had some cooling effect on the first story before it could pass up through the hall ways and escape from the second-story and attic windows. In the cases of the attic fan and natural ventilation, practically no outdoor air could enter the first story with the windows closed, and hence no cooling effect could be obtained except that normally occurring by heat transfer through the walls caused by the temperature difference between indoors and outdoors.

When cooling with outdoor air at night is used to supplement artificial cooling during the day, the effectiveness of night cooling in the house as a whole is of more significance than the effectiveness for either the first or the second story alone. The temperature drops based on the average temperature of the first and second stories when the attic fan and the basement fan were used, with the windows opened on the second story only, are shown in Fig. 19. From Fig. 19, curves Nos. 2 and 4, it may be observed that in the case of the attic fan, drawing approximately 33.2 air changes per hour through the second story alone, the effectiveness in cooling the house as a whole, based on the average indoor temperature for both stories, was much less than that in cooling the second story alone based on the average indoor temperature for the second story. The effectiveness in cooling the house as a whole was slightly greater for the basement fan when circulating approximately 9.2 air changes per hour, with both first- and second-story windows open, as shown in curve No. 3, than it was for the attic fan when drawing in 33.2 air changes per hour through the second-story windows alone, as shown in curve No. 2. The effective-
ness of cooling with both of these methods was somewhat greater than that of cooling the house as a whole with the basement fan circulating 9.2 air changes per hour with the second-story windows only opened, as shown in curve No. 1. The location of the points representing the test data indicates that the effectiveness of cooling with air from outdoors at night was not greatly influenced by the time of closing or opening the windows. It apparently made no difference whether they were closed at 6 A.M., as for Series 5-34, or at 7 A.M., as for Series 6-34; or whether they were opened at 6 P.M. or at the time that the effective temperature became the same indoors and outdoors, as for Series 3-34.

It has been shown that the slopes of the curves in Figs. 17, 18 and 19 afford a measure of the relative effectiveness of the different methods employed for cooling with outdoor air at night. Furthermore, the relative effectiveness of the different methods may be considered as dependent on the amount of air circulated and not inherent in the particular method used for circulating the air. Under these conditions, sufficient data were available from the studies of 1933 and 1934 to establish the range of the number of air changes that would be required to prove effective for night cooling in a house of the type of the Research Residence.
The slopes of all of the curves obtained when the outdoor air was positively circulated by means of some type of fan are shown in Fig. 20, plotted against the corresponding amounts of air circulated expressed as a number of air changes per hour. Since a slope of 1.0 represents the theoretical maximum cooling effect that could be obtained, the ordinates of Fig. 20, if multiplied by 100, also represent a percentage of maximum cooling effect attained with different numbers of air changes per hour. Two curves are shown in Fig. 20. The upper curve represents conditions in a given space in which all of the windows are opened and the effectiveness of cooling is based on the average temperature in the space. Such conditions would be satisfied by a one-story house with all of the windows opened and the temperature based on the single story, by a two-story house with all of the windows opened and the temperature based on the average for the house, by the second story of a two-story house with all of the windows on the second story only opened, and the temperature based on the average for the second story alone, or by the first story of a two-story house with all of the windows on the first story only opened, and the temperature based on the average for the first story alone. The fact that all of the points representing the data fell on a smooth curve, even though the amounts of window opening were not the same, indicates...
that the effectiveness of cooling for a given number of air changes per hour was more or less independent of the number and location of the windows opened, provided that the distribution of the openings was such as to give good distribution of the air. Furthermore, the fact that the point representing the cooling effect with 33.2 air changes per hour, obtained from consideration of the second story only, is on the curve, indicates that this type of curve is applicable to any space when the number of air changes per hour, the amount of window opening, and the indoor air temperature are considered in relation to the given space alone.

The lower curve in Fig. 20 represents conditions in a two-story house in which the windows on the second story only are opened, and the effectiveness of cooling is based on the house as a whole or on the average indoor temperature for both stories. It is evident that the effectiveness of cooling the house as a whole is less when the windows on the first story are not opened than it is when the windows on both stories are opened.

From Fig. 20 it may be observed that with the smaller numbers of air changes per hour the curvature of the curve decreases very rapidly as the number of air changes increases, but that the rate of decrease in the curvature becomes much less for the larger numbers of air changes. A slope of 1.0 would represent the maximum effectiveness of cooling, inasmuch as it would represent conditions under which the indoor temperature would be reduced to the same value as that outdoors. Within the limits of the observed data, the slope curve does not become asymptotic to a line representing a slope of 1.0, indicating that it would require air quantities far in excess of 33.2 air changes per hour in order to reduce the indoor temperature to the same value as that outdoors. It is therefore evident that cooling with outdoor air at night does not become reasonably effective in lowering the temperature in the house until the number of air changes per hour reaches a value of approximately 9, and that above a value of 30 air changes per hour the gain resulting from increasing the amount of air circulated is very small. At some point the gain obtained by increasing the number of air changes per hour will probably be offset by the increased cost of electrical current required to operate the fan.

In discussing the effectiveness of cooling with air from outdoors at night only the effect in cooling the air and the structure at night and its influence on the cooling load next day has been considered. During the period over which the fan was in operation, however, some lowering of the effective temperature, and hence some increase in the com-
### Table 4

**RESULTS OF OBSERVATIONS WITH KATA THERMOMETER**

<table>
<thead>
<tr>
<th>Series No.</th>
<th>Date 1933</th>
<th>Time of Test</th>
<th>Operating Notes</th>
<th>Air Temperature (Average) deg. F.</th>
<th>Air Velocity, f.p.m.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>First Story</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Dry-Bulb  Wet-Bulb</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>5 6</td>
<td></td>
</tr>
<tr>
<td>Still air</td>
<td>6-24</td>
<td>2-4 P.M.</td>
<td>House closed up; no fan</td>
<td>79.2 69.9</td>
<td>82.0 68.5</td>
</tr>
<tr>
<td>2-33</td>
<td>8-22</td>
<td>7-9 P.M.</td>
<td>Natural ventilation. All windows open wide</td>
<td>78.5 69.4</td>
<td>83.8 64.3</td>
</tr>
<tr>
<td>4-33</td>
<td>6-24</td>
<td>4-5 P.M.</td>
<td>Recirculation of house air; house closed</td>
<td>79.8 69.4</td>
<td>81.8 69.1</td>
</tr>
<tr>
<td>5-33</td>
<td>6-26</td>
<td>7-9 P.M.</td>
<td>Basement fan circulation. All windows open wide</td>
<td>81.2 73.3</td>
<td>85.2 73.7</td>
</tr>
<tr>
<td>7-33</td>
<td>7-17</td>
<td>7-9 P.M.</td>
<td>Attic fan; all windows open halfway</td>
<td>74.0 61.3</td>
<td>74.2 61.0</td>
</tr>
<tr>
<td>8-33</td>
<td>7-30</td>
<td>7-9 P.M.</td>
<td>Attic fan; all windows open wide</td>
<td>83.1 71.5</td>
<td>83.9 71.7</td>
</tr>
<tr>
<td>10-33</td>
<td>9-2</td>
<td>8-9 P.M.</td>
<td>Attic fan; second-story windows open only</td>
<td>72.5 67.9</td>
<td>71.7 65.9</td>
</tr>
<tr>
<td>11-33</td>
<td>9-18</td>
<td>7-8 P.M.</td>
<td>Natural ventilation; second-story windows open only</td>
<td>79.0 63.6</td>
<td>74.0 63.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Second Story</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Dry-Bulb  Wet-Bulb</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>5 6</td>
<td></td>
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<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>8 9</td>
<td></td>
</tr>
<tr>
<td>Dry-Kata</td>
<td>4.3</td>
<td>4.6</td>
<td>6.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.7</td>
<td>3.2</td>
<td>3.7</td>
<td>8.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>19.4</td>
<td>3.4</td>
<td>3.7</td>
<td>19.9</td>
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<tr>
<td>16.7</td>
<td>5.2</td>
<td>6.0</td>
<td>10.9</td>
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<td>12.2</td>
<td>25.8</td>
<td>25.8</td>
<td>4.2</td>
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<td>19.2</td>
<td>30.7</td>
<td>30.7</td>
<td>6.9</td>
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<tr>
<td>4.9</td>
<td>17.5</td>
<td>17.5</td>
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</tr>
<tr>
<td>3.9</td>
<td>2.9</td>
<td>2.9</td>
<td></td>
<td></td>
<td></td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>Heat Loss, H, B.t.u. per sq. ft. per hr.</th>
<th>Dry-Kata</th>
<th>Wet-Kata</th>
</tr>
</thead>
<tbody>
<tr>
<td>Series No.</td>
<td>12</td>
<td>13</td>
</tr>
<tr>
<td>Still air</td>
<td>35.5</td>
<td>36.1</td>
</tr>
<tr>
<td>2-33</td>
<td>42.6</td>
<td>49.6</td>
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<tr>
<td>4-33</td>
<td>40.9</td>
<td>43.5</td>
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<tr>
<td>5-33</td>
<td>39.3</td>
<td>43.4</td>
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<tr>
<td>7-33</td>
<td>48.8</td>
<td>56.3</td>
</tr>
<tr>
<td>8-33</td>
<td>29.8</td>
<td>39.4</td>
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<td>10-33</td>
<td>48.9</td>
<td>59.8</td>
</tr>
<tr>
<td>11-33</td>
<td>46.5</td>
<td>42.2</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Weather and Wind Conditions</th>
<th>18</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clear; very hot, max. temp. = 96 deg. F. West wind—13 m.p.h.</td>
<td></td>
</tr>
<tr>
<td>Cloudless after; cooling slowly, temp. = 74 deg. F.; practically calm, wind NE-4.</td>
<td></td>
</tr>
<tr>
<td>Same as (a)</td>
<td>138</td>
</tr>
<tr>
<td>Partly cloudy; wind SW-2</td>
<td>135</td>
</tr>
<tr>
<td>Almost cloudless; warm wind, mild</td>
<td>168</td>
</tr>
<tr>
<td>Almost cloudless; calm, wind SW-3</td>
<td>137</td>
</tr>
<tr>
<td>Very humid; temp. = 70 deg. F.; calm</td>
<td>174</td>
</tr>
<tr>
<td>Cloudless; outdoor temp. = 71 deg. F.; calm</td>
<td></td>
</tr>
</tbody>
</table>

Columns 5 to 8 inclusive, for dry-bulb and wet-bulb temperatures, are the average of 4 readings in each case.
Columns 9, 12, and 15 are the average of Kitchen, Dining Room, and Living Room.
Columns 10, 13, and 16 are the values for first-story hall only.
Columns 11, 14, and 17 are the average of E. Bedroom, S.W. Bedroom, N.W. Bedroom, and Bathroom.
In order to determine whether there was any difference in cooling effect produced by the air movement resulting from the different methods of night operation Kata thermometer readings were made at the centers of the rooms. The results are shown in Table 4. The criterion for comfort as stated by Dr. Leonard Hill is that the cooling effect for the dry Kata should exceed 40 and for the wet Kata 135 B.t.u. per sq. ft. per hr. Comparing the results from Series 2-33 and 7-33 it appears that the use of the attic fan with the windows halfway open increased the air movement over natural ventilation from a velocity of 3.2 to 12.2 ft. per minute on the first story and from 2.8 to 4.2 ft. per minute on the second story. Since the wet- and dry-bulb temperatures were comparable for these two cases the Kata cooling effects are also comparable. The dry Kata cooling effect was increased from 42.6 to 48.8 B.t.u. per sq. ft. per hr. on the first story and from 37.6 to 44.9 on the second, and the wet Kata cooling effect was increased from 167 to 178 on the first story and from 158 to 168 on the second. All of these cooling effects were greater than the 40 and 135 B.t.u. per sq. ft. per hr. required for comfort.

Comparing the results from Series 7-33 and 8-33 it may be noted that, when the attic fan was used, the velocities at the center of the room were increased somewhat when the windows were opened wide instead of only halfway. Owing to the fact that the dry-bulb temperatures were not the same, the Kata cooling effects are not comparable.
Since the first-story hall acted as a duct to convey the air from the first story, the velocity in the hall was considerably higher than that in the first- and second-story rooms.

Comparing the averages for the first and second stories in Series 5-33 with those in Series 7-33 and 8-33 it may be noted that there was some tendency for the basement fan to cause greater disturbance of the air in the rooms than there was for the attic fan. The average velocities determined by the Kata thermometer, excluding the lower hall which acted as a duct when the attic fan was used, were higher for the basement fan than for the attic fan. In the former case the air was delivered into the room as a jet from the register while in the latter it was drawn more uniformly into the room through the windows. Hence more disturbance would result in the former case than in the latter.

Comparing Series 10-33 and 11-33 it is evident that with the windows opened on the second story only, the air movement in the second-story rooms was increased from 2.9 ft. per minute when no fan was used to 17.5 ft. per minute when the attic fan was used. Also this velocity was much greater than the 4.2 and 6.9 ft. per minute obtained on the second story when the attic fan was used and all windows on both stories opened. A study of the velocities at the open windows in the latter case proved that the air was being drawn into the first-story windows at an average velocity of 57.9 ft. per minute and into the second-story windows at an average velocity of 46.4 ft. per minute. This indicated that when the attic fan was used in connection with both stories there was no tendency for the air to short-circuit through the second-story windows at the expense of the first story, but, on the contrary, the suction resulting from the chimney action of the house was greater on the first story than on the second, and, adding to the effect of the suction produced by the fan, caused a greater flow of air into the first-story windows than into the second.

A study was made to determine whether any advantage resulted from recirculating the air in the house by means of the basement fan while the house was closed during the daytime. These results are shown in Table 4, for still air and for Series 4-33, under date of June 24. When no recirculation occurred, the air velocities at the center of the rooms were 4.3 and 3.0 ft. per minute on the first and second stories, respectively. The corresponding dry and wet Kata cooling effects were 35.5 and 28.7, and 133 and 118 B.t.u. per sq. ft. per hr. These cooling effects were below the threshold for comfort. Immediately after making these observations the basement fan was started in
order to recirculate the air. The velocity at the center of the rooms was increased to 19.4 and 6.6 ft. per minute on the first and second stories, respectively, and the corresponding dry and wet Kata cooling effects were increased to 40.9 and 32.6, and 161 and 138 B.t.u. per sq. ft. per hr. The latter are very close to, or just above the threshold for comfort. Hence, while the effect was not very marked, some advantage resulted from recirculating approximately 1468 cu. ft. of air per minute, and, in border-line cases, this procedure might convert an uncomfortable atmosphere into one that is just comfortable.

31. Heat Removal Effected by Attic Fan.—A few tests, designated as Series 12-33, were run with the attic ventilated during the day by means of the attic fan instead of by natural ventilation through the windows. The temperature of the air in the attic, together with the temperature of the surface of the ceiling and the air in the rooms of the second story for a characteristic day, are shown in Fig. 21. The temperatures of the air entering the fan and leaving the attic windows are also shown in this figure. These temperatures, taken in connection with the volume of air delivered by the fan, afforded a means of cal-
calculating the heat absorbed either from the attic or second story alone, or from both.

Considering the attic alone, during the day the air from outdoors entering the fan through the duct shown in Fig. 7 was forced out through the windows, and the temperature rise was representative of the heat absorbed from the attic. A comparison of the power required by the fan motor for day and night operation indicated that no appreciable reduction in fan capacity was caused by the duct arrangement used during the day. Hence, the full fan capacity of 3940 cu. ft. per minute was used in the calculations for curve A shown in Fig. 21. This curve gives the heat absorbed from the attic, and shows a maximum rate of 15,600 B.t.u. per hr., or the equivalent of 1.3 tons of refrigeration. However, a comparison of the results from two similar days, one with the fan in operation and one without, showed no appreciable lowering of the temperature of the air in the attic resulting from the use of the fan, and no appreciable difference in either the temperature of the surface of the ceiling or of the air on the second story for the two cases. The temperature of the air in the attic was measured at two points, approximately 4 ft. above the floor, and it is probable that the temperature of the air in the peak of the roof and near the attic ceilings, when no fan was used, was much higher than that of the air nearer the floor where the measurements were made. This air formed a stationary pool of hot air above the level of the windows that served to reduce the heat transmission through the roof and to insulate the air nearer the floor. When the fan was used, this pool was swept out, and the movement of the air over the upper surfaces increased the heat transmission. Thus, while a large quantity of heat was removed and carried out by the air going through the windows, the greater part of this represented heat that did not penetrate the lower stratum of air when the fan was not used. If this were the case, the measured temperature of the air near the floor would remain about the same whether the fan was or was not used, and the heat transmitted through the floor would remain practically constant for the two cases, resulting in no change in the temperature of the surface of the ceiling, or of the air on the second story. Thus, while the fan apparently removed a large quantity of heat from the attic during the day, no appreciable improvement in conditions on the second story was effected as compared with conditions existing when the attic was ventilated by means of open windows alone. More refined methods of testing might have indicated a slight difference, but if the difference was sufficiently small to demand more refined testing methods, the cost of operating the
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Attic fan during the day could not be justified under the conditions existing at the Research Residence. It is possible that in a poorly-ventilated attic, with no windows, or with small windows in positions not adapted to cross ventilation, the use of an attic fan during the day could be entirely justified.

The rate at which heat was removed from the first and second stories by the operation of the attic fan at night is shown by the curve $H$ in Fig. 21. In this case, the air was drawn into the windows from outdoors, and all of the air passing through the house was delivered by the fan. Hence, the difference in temperature between the air entering the fan and that outdoors was representative of the heat absorbed from the first and second stories. It may be noted that this heat amounted to approximately 12 000 B.t.u. per hr. or the equivalent of one ton of refrigeration, over a period of about 3 hours. Since approximately the same amount of heat loss would occur by conduction to the outdoors through the exposed walls, irrespective of whether outdoor air was or was not circulated, the amount of heat indicated by curve $H$ represents in some measure the gain resulting from the circulation of air from outdoors at night. Owing to the presence of cross currents when full natural ventilation without the fan was used, it was not possible to measure the actual heat absorbed in this case. However, the evidence presented in Fig. 17 indicates that very nearly as much heat was removed by the full open window ventilation in Series 2-33 as was removed by the attic fan in Series 7-33 and 8-33. The portion of the curve $A$ corresponding to curve $H$ in Fig. 21 represents the additional heat removed at night from the attic alone. The whole of this amount cannot be regarded as a gain for the next day, however, because, due to the high heat transmission of the roof and to the fact that the windows in the attic were not closed during the day, the gain could not be conserved, as it was for the first and second stories on which the windows were closed at 6 A.M.

32. Effect of Methods of Night Operation on Conditions the Following Day.—Following a given method of night operation, the maximum temperature attained the next day after the windows have been closed at 6 A.M. depends upon the indoor temperature attained at 6 A.M., and the subsequent rise in outdoor temperature. As indicated by Fig. 16 this indoor temperature at 6 A.M. was the same as the minimum temperature attained indoors. Hence, it was determined by the night history, or the temperature drop shown in Fig. 17 corresponding to the particular method of night operation under consideration. The
FIG. 22. TEMPERATURE RISE OF INDOOR AIR DURING DAY

"C" is the difference between the minimum indoor and the minimum outdoor temperatures.
difference between the minimum outdoor temperature and the indoor temperature at 6 A.M. may be represented by \( C \), shown in the diagram in Fig. 22. The observed data from all of the tests not involving artificial cooling have been classified under four arbitrary values of the temperature difference, \( C \), and the results have been plotted in Fig. 22, giving the relation between the rise in indoor temperature, \( E \), and that in outdoor temperature, \( D \), for the different values of \( C \).

If the temperature corresponding to the crossing point of the indoor and outdoor temperature curves in the early evening hours is known for a given day, and the subsequent minimum outdoor temperature is obtained from the weather reports, both the indoor temperature drop and the difference, \( C \), can be evaluated from the temperature drop curve in Fig. 17, corresponding to the particular method of night operation. The indoor temperature at 6 A.M. can then be found by subtracting the indoor drop from the temperature at the crossing point. By the aid of these data and the rise in outdoor temperature, \( D \), in Fig. 22, the rise in indoor temperature and the maximum indoor temperature attained in the house next day can be calculated. If some relation could be shown between the temperature at the crossing point and the minimum outdoor temperature following it, or the drop from the crossing point to the minimum outdoor temperature, then, for a given method of night operation, the probable maximum temperature attained in the house next day could be predicted for various combinations of outdoor minimums and succeeding outdoor maximums obtained from weather reports.

The relation between the temperature at the crossing point, \( G \), and the minimum outdoor temperature, \( K \), obtained from the observed data is shown in Fig. 23. Lines representing equal temperature drops from the temperature at the crossing point to the minimum outdoor temperature have been added. From the spread of points in Fig. 23, it may be observed that a number of different crossing point temperatures may exist for any given minimum outdoor temperature, but the points corresponding to observed data all lie in a band that can be included between two lines representing upper and lower limits. Under these circumstances, for a given series of tests in which the temperature of the crossing point is known from the test data, the solution for the maximum indoor temperature for the next day is obvious and simple. The difficulty in applying the curves to the prediction of the maximum house temperature in general, when no test data are available but when the method of operation is given and the minimum and maximum outdoor temperatures are known from the weather reports,
lies in the fact that, while all of the other factors are known or determinable from the curves, the temperature at the crossing point is a casual factor depending on the previous history of the house and the weather. It may be noted from Fig. 23, however, that, over the whole range of observed weather conditions, the temperature drops from the crossing point to the minimum outdoor temperature, with few exceptions, were above 10 deg. F., and did not exceed 18 deg. F. Hence, it is possible to establish upper limits for the maximum indoor tempera-
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FIG. 24. CALCULATED MAXIMUM INDOOR AIR TEMPERATURES
ture, above which this temperature would not rise with given combinations of maximum and minimum outdoor temperatures.

Assuming 18 deg. F. and 15 deg. F. drops, from Fig. 23, as characteristic of the most severe and of average weather, respectively, the curves in Fig. 24 have been drawn representing maximum probable indoor temperatures resulting from various combinations of minimum and maximum outdoor temperatures occurring in connection with the most favorable and the least favorable methods of night operation. These may be used to predict the probable maximum indoor temperature from the weather reports of a given summer season.

Experience at the Research Residence seemed to indicate that approximately 82 deg. F. dry-bulb was a critical temperature from the standpoint of comfort. Without artificial cooling, and with the usual indoor relative humidity of from 50 to 60 per cent, conditions were reasonably comfortable if the indoor temperature did not exceed 81 deg. F. This was equivalent to an effective temperature of between 74 and 75.5 deg. F. Hence 81 deg. F. dry-bulb was accepted as the upper limit for comfort without cooling, and for the cooling tests the cooling plant was started when the indoor temperature reached 81 deg. F. on the second story. Under these conditions, owing to the reduction in indoor relative humidity when the cooling plant was in operation, the effective temperature remained somewhat below the 75 deg. F. indicated as the upper limit on the comfort chart.

Figure 25 shows the various combinations of maximum and minimum outdoor temperature occurring at Urbana, Illinois, during the summers from 1927 to 1933. Practically all of the plotted data could be included between two lines representing upper and lower limits. From these curves it may be observed that with a maximum outdoor temperature of 100 deg. F. the preceding minimum would not have exceeded 78 deg. F. Judging from the upper limit line shown in Fig. 23, with this minimum outdoor temperature, the greatest drop from the crossing point to the minimum would be of the order of 13 deg. F., and certainly would not exceed 15 deg. F. Hence, from the upper curves in Fig. 24, for a day on which the maximum outdoor temperature was 100 deg. F., with 78 deg. F. minimum outdoor temperature the night preceding and a 15 deg. F. drop in temperature from the crossing point to the minimum outdoor, a maximum indoor temperature not to exceed 92 deg. F. would be expected if the attic fan had been operated during the night. On the other hand if the house was operated on natural ventilation with partly open windows, as in Series 1-33, an indoor temperature of 94.5 deg. F. might be obtained. During one season comparatively
Few 100 deg. F. days would occur. There might be a considerable number of 95 deg. F. days, however. In this case, Fig. 25 indicates a possibility of a minimum outdoor temperature of 76 deg. F. and Fig. 23 of a 15 deg. F. temperature drop. These conditions would result in a maximum indoor temperature of 88.5 deg. F. succeeding a night of operation with the attic fan, and of 92 deg. F. for natural ventilation with partly open windows. Both of these indoor maximum temperatures are considerably above the 81 deg. F. representing the limit for comfort.

It is of some interest to consider the number of days that the maximum indoor temperature might have exceeded 81 deg. F. if the
attic fan had been operated at night over the whole period of a season. It may be observed from Fig. 23 that, at a minimum outdoor temperature of 68 deg. F., the upper limit line falls midway between temperature drops of 18 deg. F. and 15 deg. F. Hence a maximum temperature drop of 18 deg. F. may be regarded as representative of all minimum outdoor temperatures below 68 deg. F., and one of 15 deg. F. as representative of all minimum outdoor temperatures above 68 deg. F. Assuming this to be true, and obtaining from Fig. 24 the maximum indoor temperature for the actual maximum and minimum outdoor temperatures shown by the weather reports for each day in the summers of 1932 and 1933, it was found that there would have been 37 days in 1932 and 47 days in 1933 on which the maximum indoor temperature might have exceeded 81 deg. F. This, of course, represents the upper limit, and it is possible, since Fig. 23 shows that about $\frac{1}{2}$ of the total number of observed points were below the 15-deg. F. drop line, that actual operation might have resulted in approximately 25 and 32 days, respectively.

The 15 deg. F. and 18 deg. F. temperature drop lines representing an indoor maximum temperature of 81 deg. F. have been transferred from the upper curves in Fig. 24 and shown in Fig. 25. Since, as shown in Fig. 23, the 15-deg. F. drop line is the most probable upper limit for outdoor minimum temperatures above 68 deg. F. and the 18-deg. F. drop line for outdoor minimum temperatures below 68 deg. F., the line $MN$ in Fig. 25 represents a limit line above and to the right of which any combinations of outdoor maximum and minimum temperatures may certainly be expected to result in indoor maximum temperatures exceeding 81 deg. F., and below and to the left of which any combinations may be expected to result in maximum indoor temperatures not exceeding 81 deg. F. From the points of intersection, $M$ and $N$, it is evident that the maximum indoor temperature will always exceed 81 deg. F. if the outdoor maximum reaches 94 deg. F., and may do so if the outdoor maximum rises above 84 deg. F. Hence it is probable that night operation with a fan, even under the most favorable conditions, cannot be depended upon to result in comfort over the whole of a summer season in a climate similar to that of Urbana, Illinois, unless supplemented part of the time by some form of artificial cooling. It can be used, however, to alleviate conditions even in the most severe weather, and in this respect may prove satisfactory to a considerable number of householders to whom the cost of a cooling plant might be prohibitive.
33. Cooling with Air from Outdoors at Night Supplemented by Cooling with Ice During Day.—Since the tests on cooling with air from outdoors at night proved that for part of the season the use of this method alone could not be depended upon to prevent the temperature in the house from rising above 81 deg. F. on the following day, some tests were conducted in which the circulation of outdoor air at night was supplemented by the use of a limited amount of ice during the day. For this purpose the amount of ice used was restricted to a maximum of 700 lb. during the day, and the procedure is discussed in Section 14. The three most favorable methods of night operation, involving the basement and attic fans and natural ventilation, were used with the object of determining the total tonnage of ice that would be necessary for the season if the daily consumption were restricted to a maximum of 700 lb.

Figure 26 shows temperature and indoor relative humidity curves for somewhat similar days and for two characteristic conditions in the
operation of the cooling plant. The test of June 30 illustrates a case in which it was necessary to increase the rate of ice meltage, and for which the 700 lb. allotment was all melted before the outdoor temperature dropped 3 deg. F. below the indoor temperature, thus permitting the windows to be opened. In the case of all of the cooling tests the windows were not opened at any stated time, but the time was determined by the rate of decrease in the outdoor temperature, as discussed in Sections 12 and 14. Only a few days similar to June 30 were encountered for which it would have been necessary to increase the meltage to somewhat more than 700 lb. in order to maintain the indoor temperature below 81 deg. F. In these cases the sacrifice in comfort was slight because the indoor relative humidity did not increase rapidly after the ice meltage was stopped, and the period before the windows were opened was comparatively short. The remaining tests shown in Fig. 26 illustrate cases in which the 700 lb. of ice were more than sufficient to maintain 81 deg. F. until the outdoor temperature was 3 deg. F. below that indoors.

The four tests shown in Fig. 26 prove that the amount of ice used was dependent both on the maximum temperature attained during the day and on the history of the preceding night. The tests of September 8 and 9 indicate that as a hot wave progressed and the maximum and minimum outdoor temperatures increased from day to day, the ice required increased because the cooling plant either had to be started earlier, or had to be operated longer before the outdoor temperature dropped 3 deg. F. below the indoor. The latter case is illustrated by the test of September 9 for which the outdoor temperature did not decrease as rapidly as it did on September 8. No cooling was required on September 7, although the maximum outdoor temperature was 91 deg. F., which was comparable with July 10, and very little cooling was required on September 8 when the maximum outdoor temperature rose to 93 deg. F. July 10 was preceded by a hot day during the night of which the minimum outdoor temperature was 69 deg. F., as compared with 67.5 deg. F. attained in the early morning hours of September 8. Hence the minimum indoor temperature was reduced to only 75 deg. F. to start the day of July 10 as compared with 72.5 deg. F. to start the day of September 8. Again on September 9, the minimum indoor temperature was reduced to 72.5 deg. F. in the early morning hours, and September 9, for which the maximum outdoor temperature reached 96 deg. F., required about the same amount of ice as July 10, for which the maximum was 90 deg. F.
A few tests, Series 9-33, with a room cooling unit in the living room proved that, with outdoor temperatures as high as 93 deg. F., the temperature in the living room, lower hall, and dining room could be maintained at from 77 deg. F. to 80 deg. F. The temperature in the kitchen rose to 82 deg. F. but conditions were not particularly uncomfortable because the relative humidity maintained in the kitchen was the same as that in the other first-story rooms. The relative humidity was reduced from a value of about 70 per cent to one of about 55 per cent. The unit produced no noticeable effect on the conditions on the second story.

In the cases discussed in Section 32, where no cooling was employed, the temperature corresponding to the crossing point of the indoor and outdoor temperature curves was a more or less casual factor, and the prediction of the maximum indoor temperature was confined to certain limiting values. In the case where the cooling plant is to be used consistently throughout the season, however, and the plant is started whenever the indoor temperature rises to 81 deg. F., the temperature at the crossing point could usually be maintained at approximately 80 deg. F. Hence, for a given method of operation at night, the curves in Figs. 17 and 22 used in connection with the minimum and maximum outdoor temperatures obtained from the weather reports for a given season afford a comparatively certain means of predicting the number of days on which the indoor temperature would reach 81 deg. F. and necessitate starting the cooling plant. If the ice meltage is limited to 700 lb. a day, and it is assumed that the 700 lb. are always melted, then the product of 700 and the number of days requiring the use of the cooling plant represents the maximum limit of the amount of ice required per season. It is true that a few days were observed for which somewhat more than 700 lb. of ice would have been required to maintain the indoor temperature below 81 deg. F., but these days were more than offset by the ones not requiring the entire 700 lb. Hence, the amount of ice obtained by the method outlined would be the maximum.

On the assumption that the attic fan is used at night and the crossing point temperature is limited to 80 deg. F., if an outdoor minimum temperature of 68 deg. F. is observed the indoor temperature drop of 8.1 deg. F. may be obtained from Fig. 17. This represents an indoor minimum, or temperature at 6 A.M., of 71.9 deg. F. The temperature difference, $C$, shown in Fig. 22, would then be 3.9 deg. F. If the outdoor temperature attained a maximum of 95 deg. F. the next day, the
### Table 5
**Estimated Ice Requirements for Summers of 1932 and 1933**

<table>
<thead>
<tr>
<th>Series No.</th>
<th>Method of Operation at Night</th>
<th>Summer of 1932</th>
<th></th>
<th></th>
<th></th>
<th>Summer of 1933</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Estimated Number of Days Requiring Cooling†</td>
<td>Ice Requirements*</td>
<td></td>
<td></td>
<td>Estimated Number of Days Requiring Cooling†</td>
<td>Ice Requirements*</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1-33</td>
<td>Natural ventilation; 8 windows partly open at 9 P.M.</td>
<td>21</td>
<td>24</td>
<td>18</td>
<td>2</td>
<td>65</td>
<td>23</td>
<td>25</td>
<td>24</td>
</tr>
<tr>
<td>3-33</td>
<td>Basement fan; 8 windows partly open at 9 P.M.</td>
<td>11</td>
<td>18</td>
<td>11</td>
<td>0</td>
<td>40</td>
<td>14</td>
<td>21</td>
<td>18</td>
</tr>
<tr>
<td>2-33</td>
<td>Natural ventilation; all windows open at 6 P.M.</td>
<td>9</td>
<td>17</td>
<td>8</td>
<td>0</td>
<td>34</td>
<td>12</td>
<td>20</td>
<td>15</td>
</tr>
<tr>
<td>7-33 and 8-33</td>
<td>Attic fan; all windows open at 6 P.M.</td>
<td>7</td>
<td>15</td>
<td>7</td>
<td>0</td>
<td>29</td>
<td>10</td>
<td>19</td>
<td>14</td>
</tr>
</tbody>
</table>

*Based on an ice consumption of 700 lb. per day.
†Based on indoor temperature exceeding 81 deg. F. at some time during the day.
outdoor rise would be 27.0 deg. F., and the indoor rise corresponding to a value of 3.9 deg. F. for \( C \) would be 12.0 deg. F. Hence, the indoor temperature would rise to 83.9 deg. F. if no cooling were employed, indicating that the cooling plant would have to be started. By following this procedure for all of the recorded outdoor maximum and minimum temperatures for the summer, the number of days requiring cooling and the maximum amount of ice required for the season may be obtained. This procedure has been followed for four different methods of night operation for the seasons of 1932 and 1933, and the summary is given in Table 5. Neither season would have required any cooling in the month of May.

During the summer of 1932, which included 1471 degree-hours above 85 deg. F., 43.3 tons of ice were required under the system of operation followed for that summer. It is of considerable interest to note that if advantage had been taken of the possibility of circulating outdoor air at night and an attic fan had been used under conditions corresponding to those for Series 7-33 and 8-33, in which the total daily ice meltage was restricted to a maximum of 700 lb., the probable ice requirement would have been only 10 tons. With natural ventilation with all windows open the requirement would have been 12 tons, and with the basement fan with 8 windows partly open it would have been 14 tons. Even with the least favorable method of night operation the requirement would have been only 23 tons. The summer of 1933 included 2310 degree-hours above 85 deg. F., and, as indicated in Table 5, the ice requirements would have been somewhat greater than those for the summer of 1932. However, the table presents concrete evidence that even the least favorable method of circulating the cool air from outdoors at night, when used to supplement some form of artificial cooling during the day, has merit as a means of reducing the cost of summer cooling to an amount that may not be prohibitive for the average householder.

34. Conclusions.—The following conclusions may be drawn as applying to the Research Residence and the conditions under which the tests were conducted:

(1) The circulation of air from the outdoors at night, when used as a supplement to artificial cooling during the day, has considerable merit in reducing the seasonal cooling load that would otherwise be required.
(2) The circulation of air from the outdoors at night may make the use of artificial cooling unnecessary for a considerable portion of the summer season.

(3) If the best means of circulating air from the outdoors at night had been used at the Research Residence during the summer of 1932, and the amount of ice melted during any one day had been restricted to a maximum of 700 lb., the ice meltage could probably have been reduced from the 43 tons actually used to an amount of the order of 10 tons.

(4) The practice of partly opening a few windows at night is not very effective as a means of circulating air from the outdoors.

(5) The use of a fan in a forced-air heating system to circulate from 6 to 9 air changes per hour from the outdoors at night is much more effective than opening a few windows even when the same amount of window opening is retained for the two cases.

(6) The use of a fan in a forced-air heating system to circulate 9 air changes per hour is more effective if all of the windows and the attic door or hatchway are opened than if only a few windows are opened.

(7) The most effective method of circulating air from the outdoors at night is to open all of the windows and to use a fan drawing the equivalent of approximately 17 air changes per hour into the windows of the first and second stories and discharging the air into the attic to escape from the attic windows.

(8) In the case of a two-story house similar to the Research Residence having an ample attic with dormer windows and large attic door, opening all of the windows and the attic door is nearly as effective for circulating air from the outdoors at night as the use of an attic fan producing approximately 17 air changes per hour.

(9) The use of an attic fan or a fan in the forced-air heating system to circulate outdoor air at night is not as effective in cooling the Residence as a whole when the second-story windows only are opened as it is when windows on both stories are opened.

(10) Satisfactory cooling with outdoor air at night probably cannot be accomplished by employing less than 9 air changes per hour. The gain arising from the use of more than 30 air changes per hour is very small.

(11) There is some advantage in opening the windows at 6 P.M. rather than at 9 P.M., even if the outdoor temperature is slightly higher than the indoor temperature at 6 P.M.
### Table 6

**Typical Operating Data and Results with Mechanical Refrigeration at 2:00 P.M. on June 27, 1934**

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Outdoor air</td>
<td>dry-bulb 97.1 deg. F., wet-bulb 73.6 deg. F., relative humidity 33 per cent 86.4</td>
</tr>
<tr>
<td>2. Indoor air, avg.</td>
<td>first story, 79.4 deg. F., second story, 80.4 deg. F.</td>
</tr>
<tr>
<td>3. Indoor air, house</td>
<td>dry-bulb 79.9 deg. F., wet-bulb 65.2 deg. F., relative humidity 45.5 per cent 70.0</td>
</tr>
<tr>
<td>4. Ventilating air</td>
<td>dry-bulb 94.3 deg. F., wet-bulb 76.5 deg. F., relative humidity 44.5 per cent 105.6</td>
</tr>
<tr>
<td>5. Mixed air enters</td>
<td>dry-bulb 81.5 deg. F., wet-bulb 67.2 deg. F., relative humidity 47.5 per cent 76.0</td>
</tr>
<tr>
<td>6. Mixed air leaves</td>
<td>dry-bulb 65.6 deg. F., wet-bulb 59.9 deg. F., relative humidity 72.0 per cent 68.1</td>
</tr>
<tr>
<td>7. Air temp. drop</td>
<td>15.9</td>
</tr>
<tr>
<td>8. Temperature of</td>
<td>70.0</td>
</tr>
<tr>
<td>9. Air temp. rise in</td>
<td>4.4</td>
</tr>
<tr>
<td>10. Basement air temp.</td>
<td>71.3</td>
</tr>
<tr>
<td>11. Quantity of</td>
<td>1200 c.f.m. or 5660 lb. of dry air per hour 0.0749</td>
</tr>
<tr>
<td>12. Number of air</td>
<td>5.3</td>
</tr>
<tr>
<td>13. cooling coil</td>
<td>face area 3.54 sq. ft., free area 2.18 sq. ft. 365</td>
</tr>
<tr>
<td>14. Ventilating air</td>
<td>598</td>
</tr>
<tr>
<td></td>
<td>226 c.f.m. or 1015 lb. dry air per hour 0.0749</td>
</tr>
<tr>
<td></td>
<td>7.11</td>
</tr>
<tr>
<td></td>
<td>29,065 B.t.u. per hour</td>
</tr>
<tr>
<td></td>
<td>7466 B.t.u. per hour; 25.7 per cent of total heat absorbed 21,599 B.t.u. per hour; 74.3 per cent of total heat absorbed 21,599 B.t.u. per hour; 74.3 per cent of total heat absorbed 0.767</td>
</tr>
<tr>
<td></td>
<td>0.767</td>
</tr>
<tr>
<td></td>
<td>800.3 lb., or 96.1 gal.</td>
</tr>
<tr>
<td>17. Water temp.</td>
<td>32,632 B.t.u. per hour</td>
</tr>
<tr>
<td></td>
<td>9980 B.t.u. per hour</td>
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<td>18. Water temp.</td>
<td>inlet, 42.4, control bulb, 57.1, outlet, 74.0</td>
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<td>19. Quantity of</td>
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<td>20. Heat contained by</td>
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<tr>
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<td>size, 3 h.p., measured power rate, 3.18 kw.</td>
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<td>21. Heat contained by</td>
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<td>size, 3 h.p., measured power rate, 0.330 kw.</td>
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35. Preliminary Statement.—The tests conducted with mechanical refrigeration differed somewhat from those conducted with ice as the cooling medium in that in the latter case no outdoor air was introduced in excess of that resulting from normal infiltration around windows and doors, while in the case of mechanical refrigeration the equivalent of one air change per hour from outdoors was introduced by means of a fan delivering air into the suction side of the system. Furthermore, two series of tests were run, one in which artificial cooling during the day was not supplemented by the circulation of air from outdoors at night, and one in which the second-story windows and the attic door were opened at night and outdoor air was circulated by means of the basement fan. The details for these methods of operation are discussed in Sections 15 and 16.

36. Mechanical Refrigeration Not Supplemented by Cooling with Air from Outdoors at Night.—The operating characteristics of the cooling plant and a comparison of the actual and calculated cooling loads for the house can best be illustrated by the results obtained on a typical day. For this purpose, a test made on June 27, 1934, was selected, and the results are shown in Fig. 27 and Table 6. On this day the outdoor temperature was 97.1 deg. F. at 2 P.M., and reached a maximum of 99.0 deg. F. at 3:45 P.M. At that time the house was being operated on Series 4-34, and the windows had remained closed during the night preceding the test. The fan in the forced-air system had continued to operate during the off period of the compressor, shown from 1:30 A.M. to 6:30 A.M. in Fig. 27, and ventilating air from the outdoors was drawn in, resulting in a rise in the relative humidity indoors. At 6:30 A.M. the compressor was started through the action of the thermostat and operated intermittently until 11:20 A.M. The compressor operated continuously at full load capacity from 11:20 A.M. to 11:15 P.M., at which time intermittent operation was resumed. During the long off period from 1:30 A.M. to 6:30 A.M., the cooling coil evidently warmed up and the temperature became practically the same as the air temperature. The intermittent periods of operation were not sufficiently long to establish equilibrium, and full capacity of the machine was not absorbed from the air until the compressor started to operate continuously.

During the hours from midnight to 5 A.M. the indoor temperature remained constant owing to the effect of the heat capacity of the
structure, while the calculated load curve showed a decreasing cooling load. From 6 A.M. until 10 A.M. the actual average load approximated the calculated load. After 10 A.M., however, the actual load was less than the calculated load until approximately 6:10 P.M., at which time the two became equal. The calculated load attained a maximum of 38 240 B.t.u. per hour, while the actual load never rose above 30 500 B.t.u. per hour. The effect of heat lag of the structure is most strikingly shown in the period from 6:10 P.M. to 11:15 P.M. At the latter time the calculated load had decreased to 13 000 B.t.u. per hour, while 27 500 B.t.u. per hour were still being absorbed from the air. During the day the temperature of the air indoors rose from 79 deg. F. to 81
deg. F. From 6:10 P.M. to 11:15 P.M. the indoor temperature was reduced from 81 deg. F. to 79 deg. F. This was taken into account in the calculated load curve, but is not sufficient to explain the wide discrepancy between the actual and the calculated loads. From 11:15 P.M. to 3 A.M. the calculated load continued to decrease somewhat more rapidly than the actual load.

The calculated cooling load was based on the first and second stories only. It included the heat transmission through the walls, floors, ceilings, and glass, sun effect on walls and glass, sensible heat brought in by the air used for ventilation, heat brought in by moisture in the air used for ventilation, heat from four occupants, heat from lights, and heat supplied by the electrical input to the fan, and was based on the hourly observed indoor and outdoor wet- and dry-bulb temperatures. The heat equivalent of the electrical input to the compressor motor was not included as it was regarded as all being dissipated in the basement. An attempt was made to determine the pressure built up in the first and second stories by the operation of the ventilating fan. While this did not result in a quantitative determination of the actual pressure, it did indicate that the pressure was slightly greater than atmospheric. Hence it was considered that infiltration through windows could be neglected and all infiltration could be credited to the air brought in for ventilation, amounting to approximately one air change per hour. The details for calculating the hourly cooling load are given in Appendix B.

For the purpose of comparison, the design load calculated by the method outlined in the A.S.H.V.E. Guide, 1934, has been shown as a single point in Fig. 27. Certain modifications and assumptions were necessary in applying this method. The heat transmitted through the windows from the sun was obtained by using a value taken at 2 P.M. from the intensity curves given in the Guide, and by multiplying this result by 0.28* to allow for the shading effect of the awnings. In order to obtain the sun effect on the walls a value of 25 deg. F. was added to the difference in temperature between outdoor and indoor air as recommended. The south and west walls only were regarded as exposed to the sun, and 50 per cent of the net area of these walls was regarded as shaded by awnings and shutters. The design temperatures of 91 deg. F. dry-bulb and 75 deg. F. wet-bulb were selected from the table in the Guide, and the average of 80 deg. F. was used for the indoor temperature. The temperature in the attic was assumed to be

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INVESTIGATION OF SUMMER COOLING IN RESIDENCE

15 deg. F. higher than the outdoor temperature. It may be observed that this calculated design load was 37 500 B.t.u. per hour as compared with the actual load of approximately 30 000 B.t.u. per hour. The latter proved sufficient to cool the house satisfactorily.

The indoor relative humidity was approximately 61 per cent until 9 A.M., as indicated in Fig. 27. During the early period of intermittent operation of the compressor the relative humidity decreased until the compressor started to operate continuously; at that time it assumed a value of approximately 45 per cent, and remained practically constant at this value until the compressor again started to operate intermittently.

Characteristic results obtained during the continuous operating periods of the compressor are shown in Table 6. The air conditions at different locations are given in the first ten items of the table. The dry-bulb temperature on the second story was only 1.0 deg. F. higher than that on the first, thus representing a very satisfactory balance of the cooling on the two stories. The ventilating air was taken from a large sunken window areaway on the north side of the Residence. This areaway was at the junction of the main body of the house and the north wing forming the kitchen, and hence was protected from the sun on both the south and the west. The temperature of the ventilating air was therefore somewhat lower and the relative humidity somewhat higher than the corresponding values for the outdoor air. The temperature of the cooled air leaving the registers was never lower than 68.0 deg. F., and no difficulty was experienced from drafts in the rooms. The rise in temperature of the air passing through the furnace casing and ducts was 4.4 deg. F. The quantity of air circulated was 1260 cu. ft. per min., which amounted to 5.3 air changes delivered by the fan per hour or 522 cu. ft. per min. per ton of refrigeration absorbed from the air. With this amount of air, the velocity through the face area of the coil was 368 ft. per min. and the temperature drop was 15.9 deg. F. This corresponded closely with the 350 ft. per min. and the 15 deg. F. drop used for the design. The free area velocity of 598 ft. per min. did not prove to be sufficient to carry any condensed moisture away from the surfaces of the cooling coil.

The useful refrigerating effect, or heat absorbed by the cooling coil was 29 065 B.t.u. per hour, which compared favorably with the nominal rating of 29 500 B.t.u. per hour, although the temperature of the refrigerant as shown by item 25 was considerably above the 32 deg. F. on which the rating was based. Item 25 indicates that considerable superheating occurred in a large portion of the evaporator.
or cooling coil. The ratio of the useful refrigerating effect to the heat absorbed by the water passing through the condenser was 0.95. This high ratio may be explained by the fact that considerable heat was lost from the condenser coil to the air in the basement, which would otherwise have been absorbed by the water and thus would have reduced the value of the ratio. However, this ratio, obtained for long operating periods of the compressor on each test, proved valuable in checking the calculated refrigerating effect for the periods of intermittent operation, during which the validity of using average temperatures for the air and average weights for the water condensed on the coil is somewhat uncertain. Such calculations based on averages for widely fluctuating conditions resulting from intermittent operation are always subject to some uncertainty.

As shown in item 16 the useful refrigerating effect, or heat absorbed by the cooling coil, was obtained by computing the heat absorbed by the air, which was separated into the sensible heat given up by the dry air and the heat given up by the change in the moisture content of the air in passing through the cooling coil. In connection with the latter heat quantity, some difficulty was experienced in obtaining accurate readings of the wet-bulb temperatures as observed by means of recording thermocouples. The wicks used on these recording thermocouples became fouled in a short time, thus affecting the accuracy of the observed wet-bulb temperature. However, a sufficient number of accurate readings was obtained to prove that no great amount of variation occurred in the weight of air and in the entering and leaving wet- and dry-bulb temperatures for the air passing through the cooling coil during periods of compressor operation. The weighed amounts of water condensed on the coil were therefore accepted as being accurate, and the heat absorbed was obtained by multiplying this weight by a constant value representing the heat given up by the change in moisture content of the air per pound of water vapor condensed. This latter included the latent heat and superheat in the water vapor condensed, and the change in the superheat in the water vapor remaining in the air after passing the cooling coil, and amounted to 1050 B.t.u. per pound of vapor condensed. The weight of water vapor condensed varied from 12.5 to 3.4 pounds per hour based on the periods of continuous operation for the compressor, or from 7.2 to 1.2 pounds per hour based on the total length of the test period over which artificial cooling was required.

Item 21, the heat absorbed by the water passing through the condenser, the compressor heads, and the piping in the basement, $H_w$,
represents all of the heat that was removed from the house with the exception of that which might have been lost through the basement walls. In order to separate the heat gains introduced by the presence of the refrigerating machine in the basement, such as electrical and mechanical losses in the compressor motor, mechanical losses in the compressor and drive, and heat losses from the condenser and compressor heads, from the net heat gains which were independent of the presence of the machine, the following heat balance on the house as a whole may be made:

\[ H_\omega = H + H_e, \text{ or } H = H_\omega - H_e. \]

In which \( H_\omega \) = total heat removed by the cooling water from the point of entrance to the point of drainage.

\( H \) = net heat gain of house, including heat transmission through walls, ceilings, and glass, infiltration, ventilating air, heat from lights and fan motor, and heat from occupants, but excluding heat loss through the basement walls.

\( H_e \) = heat equivalent of electrical input to the compressor motor.

The net heat gain, \( H \), is shown as item 23 in Table 6. Item 16, the heat absorbed by the cooling coil, represents the useful refrigerating effect, and since ultimately all of the cooling in the house had to be accomplished by means of the air passing through the coil, it also represents the total amount of refrigeration that had to be expended to cool the house under the conditions of operation with the machine in the basement. If this heat is represented by \( H_r \), then the ratio \( H/H_r \) becomes an index of the additional cooling load imposed by the presence of the refrigerating machine in the basement, since this ratio would be 1.0 if it were possible to operate with no heat loss from the machine or condenser. In some respects this ratio is analogous to the overall house efficiency which may be obtained for winter heating. The value of 0.767, given as item 24, is relatively low as compared with overall house efficiencies obtained for winter heating. Since \( H = H_\omega - H_e \) is the net heat gain for the house, it may be observed that for given outdoor weather conditions any increase in the heat loss from the machine or condenser must be compensated for by readjustments in \( H_\omega \) and \( H_e \). These readjustments must be such that the difference, \( H \), remains constant. Any such increase in losses, however, will be directly reflected in an increase in \( H_r \). Hence the ratio \( H/H_r \) will decrease rapidly as the losses from the machine or condenser are
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<th>Indoor Air Conditions</th>
<th>Temperature Difference</th>
<th>Night Air Cooling</th>
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| Tests Nos. 1, 25, 33, and 34 were omitted on account of incomplete data and unfavorable changes in the weather. |
|1| 3-34 is a Series No. 1-34 test. |
|2| 3-34 is a Series No. 3-34 test. |
|3| Circulating fan was run continuously during the overall test period. |
increased. Insofar as operating costs are concerned, it would therefore seem to be advantageous to insulate the condenser or to install the machine outside of the house.

The general results for all of the tests with mechanical refrigeration are shown in Table 7. It may be observed from column 33 that beginning with Test No. 17-34, a decrease in the ratio $H/H_r$ occurred up to, and including, Test No. 24-34. This was caused by a gradual loss of the charge of refrigerant occurring over this period. As a result, the compressor did not operate as efficiently, and the heat losses from the machine and condenser became a larger portion of the useful refrigerating effect. The refrigerant had no odor and the gradual loss of the charge was obscured by other factors. A small decrease in the drop in temperature was observed, but it was erroneously attributed to an increase in outdoor relative humidity which occurred at about this time. Furthermore, the electrical input to the compressor motor which was observed by means of an integrating wattmeter did not decrease sufficiently to attract attention. The indoor temperature, however, was maintained lower than 81 deg. F. until Test No. 22-34. Hence, only the three tests, Nos. 22-34 to 24-34, inclusive, have been regarded as of doubtful value, and these are useful in determining the effect on comfort of maintaining an indoor temperature higher than 81 deg. F., which is discussed later. The machine was recharged after Test No. 24-34 and on Test No. 26-34, during which the maximum outdoor temperature rose to 101.4 deg. F., the ratio $H/H_r$ rose to a normal value, and the indoor temperature was maintained at 80.3 deg. F., indicating that with a normal charge of refrigerant no difficulty would have been encountered in maintaining an indoor temperature of 80 deg. F. on the three preceding days for which the maximum outdoor temperature was from 100.0 deg. F. to 102.8 deg. F. This experience, however, in which the loss of the charge was obscured by other factors even with the machine under constant observation, indicates that, from the standpoint of the householder, it would be advantageous to introduce some odor, or other positive means of indicating a leak before the loss of the whole charge has occurred.

The total cooling load for the day, expressed as the total B.t.u. absorbed from the air during the overall period of compressor operation, is shown in Fig. 28 plotted against the degree-hours above 85 deg. F. per day. The base temperature of 85 deg. F. was selected because it was found that usually no cooling was required unless the outdoor temperature was approximately 85 deg. F. or above. Curves for both total heat and sensible heat are shown, and the difference
Fig. 28. Heat Absorbed by Cooling Coil, and Hours of Compressor Operation Per Day
between these curves represents the heat due to the change in the moisture content of the air. This moisture load varied from approximately 25 per cent to 30 per cent of the total load. One point, at 60 degree-hours, deviates widely from the total heat curve, but does not deviate very widely from the sensible heat curve. On this particular test, No. 29-34, the outdoor relative humidity was 83.5 per cent, the highest recorded during the summer, and the load due to moisture was excessive.

37. Mechanical Refrigeration Supplemented by Cooling with Air from Outdoors at Night.—A comparison between the heat absorbed by the cooling coil in Series 4-34, for which no cooling with outdoor air at night was employed, and that in Series 3-34, for which supplementary night cooling was used, may be obtained from Fig. 28. The data for Tests Nos. 22-34 to 24-34 inclusive have been omitted from these curves because the average indoor temperature was higher than 81 deg. F. and these tests were not regarded as being comparable with the others. The results from the two series of tests form two fairly well-defined curves. More deviations occurred in the points representing the data from Series 3-34 because the amount of artificial cooling required during the day was directly influenced by the outdoor conditions on the preceding night. The curve represents median conditions, and if the night was exceptionally cool less artificial cooling was required the next day, while if the night was exceptionally warm the reverse was true. In the case of Series 4-34, for which the windows were not opened at night, the variations in the outdoor conditions at night were not so directly reflected in the amount of cooling required, and hence less deviation occurred between the median curve and the points representing individual tests.

From the curves in Fig. 28 it is apparent that considerable saving in the amount of mechanical refrigeration required for cooling may be effected by supplementary cooling with outdoor air at night, even when the latter is confined to that which may be obtained by opening only the second-story windows at night and employing 9.2 air changes per hour as furnished by the fan in the forced-air heating system. This is particularly true for milder weather. For Series 3-34 the windows were opened when the outdoor effective temperature became the same as that indoors. As the nights became warmer, this opening time occurred later, and when the degree-hours per day reached a value of approximately 110 no windows were opened and Series 3-34 merged into Series 4-34. On days for which the maximum outdoor
temperature reached 80 to 85 deg. F. the windows were opened at
about 7 P.M. and no cooling was required the next day. When the
maximum outdoor temperature was from 85 to 95 deg. F. the windows
were opened some time between 8 and 10 P.M., and artificial cooling
was required the next day. When the maximum outdoor temperature
was above 95 deg. F. and the indoor and outdoor effective tempera-
tures did not equalize until later than midnight, the windows were
not opened, and artificial cooling was employed during the whole of
the 24 hours.

The curves in the lower portion of Fig. 28 were computed from the
total heat curves in the upper portion, and represent the total time
that the compressor would be required to run delivering a constant
mean refrigerating load of 29 000 B.t.u. per hour. The actual run-
ning times obtained from the observed data for the individual tests
have been plotted as points on the curves. A study of these points
shows that, with the exception of four points near the upper end of the
curves, they fell in about the same positions relative to the curves as
the points fell relative to the total and sensible heat curves in the
upper portion of Fig. 28. Of the exceptions, the three upper points
represent data from Tests Nos. 22-34 to 24-34 inclusive, which were
excluded from the heat curves because an indoor temperature below
81 deg. F. was not maintained. The fourth point represents Test No.
21-34, which was included in the upper curves because the indoor
temperature was maintained at 80.7 deg. F. From column 33 in
Table 7 it may be noted that for these four tests exceptionally low
values of the ratio $H/H_r$ were obtained. A study of the points there-
fore indicates that when the refrigerant leak started with Test
No. 17-34, a gradual loss in efficiency occurred as evidenced by the
reduction in the ratio of $H/H_r$, but the loss of part of the charge did
not affect the output of the machine until Test No. 21-34 was reached.
At this point the output was decreased and although it was still pos-
sible to maintain an average indoor temperature below 81 deg. F. it
was accomplished by an increase in the running time of the compres-
sor. For the points representing Tests Nos. 22-34 to 24-34 inclusive,
it was not possible to maintain a temperature below 81 deg. F. even
with the compressor operating continuously for 24 hours. Hence these
three tests have been excluded in any consideration of the data. The
point at 60 degree-hours, representing Test No. 29-34, is of interest as
indicating the increase in running time that was made necessary by the
moisture load imposed by an outdoor relative humidity of 83.5 per
cent. It may also be observed that extrapolation of the curves to the
24-hour ordinate indicates that the compressor would not be required to operate continuously over the 24-hour period until the number of degree-hours per day above 85 deg. F. equaled, or exceeded, approximately 150. Since this condition practically never occurs at Urbana, Illinois, the data indicate that a refrigerating machine of 2\(\frac{1}{2}\) tons capacity is ample to maintain an indoor temperature not in excess of 81 deg. F. in a two-story building similar in construction to that of the Research Residence located where the climate is similar to that at Urbana.

38. Comfort Conditions.—On all of the tests in which the average indoor temperature for the test was not above 81 deg. F. the maximum indoor temperature did not rise above 81.5 deg. F., and then only for a short time. With indoor relative humidities between 45 and 55 per cent this was equivalent to effective temperatures between 74 and 75.5 deg. F. During the daytime these conditions were found to be comfortable both by members of the staff who occupied the Residence more or less constantly, and by visitors who were present for short periods only. The maximum discomfort occurred with the Series 3-34 tests during the hour just preceding the start of the compressor. After the windows were closed in the morning, the indoor dry-bulb temperature did not rise sufficiently to start the compressor until the late morning or early afternoon hours. During this time that the house was closed and the plant was not operating, the indoor relative humidity increased, and for a short time the house felt stuffy. Odors were also somewhat noticeable at this time. When the plant was started, both the slight discomfort and the odors disappeared. During the Series 4-34 tests, for which the windows were not opened, the plant operated either continuously or at short intermittent periods over the whole 24 hours, and no discomfort or odors were noticeable. The admission of outdoor air for ventilation amounting to approximately one air change per hour was effective in overcoming any odors. At night, a temperature of 80 deg. F. was not entirely comfortable for a person lying down. The bed prevented the circulation of air over part of the body, and the remaining surface was apparently not sufficient for effective cooling. It is therefore possible that sleeping quarters should be cooled to a somewhat lower temperature than living quarters.

On the three tests, Nos. 22-34 to 24-34 inclusive, for which the average indoor temperature was above 81 deg. F., the maximum temperatures attained during the day were 84.5, 85.2, and 86.0 deg. F.
During these times the house was distinctly uncomfortable to the occupants. On coming in from outdoors an observer would be satisfied for a short time owing to the contrast, particularly if the outdoor temperature was approximately 100 deg. F. This effect was not lasting, however, and indoor temperatures as high as 85 deg. F. do not seem desirable unless relative humidities lower than those obtained at the Research Residence are readily attainable.

### 39. Actual and Estimated Seasonal Cooling Loads.

Table 8 gives a summary of total quantities obtained for the season of 1934. The total of 2658 degree-hours above 85 deg. F., item 3, indicates that this season was severe as compared with those of 1933 and 1932, for which the degree-hours above 85 deg. F. were 2310 and 1471, respectively. This was caused by the comparatively large number of days on which the outdoor temperature rose above 90 deg. F. The hours above 85 deg. F. for the seasons of 1934, 1933, and 1932 were 482, 493, and 329 respectively, while those above 90 deg. F. were 224, 208, and 122.

It is of interest to note that the total heat absorbed during the season with the two methods of operating the Residence from June 20 to September 1 was 8 354 881 B.t.u., which was equivalent to an ice meltage of 29.0 tons. It is not usual to reject the water to the drain at a temperature of 32 deg. F. in the case of a plant using ice as a cooling medium. On the assumption that the water is rejected at approximately 40 deg. F. and that advantage is taken of this additional cooling effect, an equivalent ice meltage of 27.5 tons could be obtained,

<table>
<thead>
<tr>
<th>Table 8</th>
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<tbody>
<tr>
<td><strong>SUMMARY OF RESULTS OF TESTS WITH MECHANICAL REFRIGERATION, SEASON OF 1934</strong></td>
</tr>
<tr>
<td>1. Total hours above 85 deg. F. for season of 1934</td>
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<td>2. Total hours above 90 deg. F. for season of 1934</td>
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<td>3. Total degree-hours above 85 deg. F. for season of 1934</td>
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<td>4. Total degree-hours above 90 deg. F. for season of 1934</td>
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<td>5. Number of tests with mechanical refrigeration</td>
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<td>6. Total running time for fan during night cooling, hours</td>
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<td>7. Average rate of power input to fan during night cooling, watts</td>
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<td>8. Total power input to fan during night cooling, kw-hr</td>
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<td>9. Total running time for fan during test period, hours</td>
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<td>10. Average rate of power input to fan during test period, watts</td>
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<td>11. Total power input to fan during test period, kw-hr</td>
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<td>12. Total running time for fan including night cooling, hours</td>
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<td>13. Total power input to fan including night cooling, kw-hr</td>
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<td>14. Total running time for compressor, hours</td>
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<td>15. Average rate of power input to compressor, watts</td>
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<td>16. Total power input to compressor, kw-hr</td>
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<td>17. Total power input to compressor and fan, including night cooling, kw-hr</td>
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<td>18. Total quantity of cooling water, gallons</td>
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<td>19. Total heat absorbed by cooling coil during season, B.t.u.</td>
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<td>20. Equivalent ice meltage during season, tons</td>
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<tr>
<td>21. Corrected ice meltage during season assuming water to the drain at 40 deg. F., tons</td>
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as shown by item 21. This equivalent tonnage would have been somewhat increased if the plant had been in operation about May 15, but it is probable that the 27.5 tons may be regarded as representative of the requirements under the existing conditions of operation for a season similar to that of 1934. This figure is not comparable, however, with the 43.3 tons of ice actually used during the summer of 1932. For the 1932 season the windows remained closed during the whole period from June 1 to October 1, and no advantage was taken of the possible supplementary cooling with outdoor air at night. Furthermore, for one-half of the season no awnings were used on the windows. For the work in 1934, full advantage was taken of cooling with outdoor air at night for a large part of the season and the Residence was equipped with awnings during the whole season.

Since the period over which the plant was in operation did not comprise the entire season, and since during the operating period two separate series of tests were run, one with supplementary cooling with air from outdoors at night and one without such supplementary cooling, direct experimental results on the total seasonal cooling load for either one of these conditions alone are not available. Furthermore, since the season for 1932 included 1471 degree-hours above 85 deg. F., while that for 1934 included 2658, direct comparison of the seasonal loads obtained experimentally under the same operating conditions could not be made, even if such results were available. All of these seasonal loads can be estimated, however, with sufficient accuracy to permit comparisons by making use of the curves in Figs. 14 and 28, and the method discussed in Section 25. Tabulation of the daily loads from the curves in Fig. 28 corresponding to the degree-hours per day for each day in the season of 1934 indicated that if the plant had been operated all of the season using mechanical refrigeration alone, not supplemented by cooling with air from outdoors at night, it would have operated 61 days and the total seasonal load would have been 15,587,000 B.t.u. This would correspond to an equivalent ice meltage of 51.2 tons, on the assumption that the water was rejected at 40 deg. F. and that the difference between 40 deg. F. and 32 deg. F. was effective for useful cooling. If the plant had been operated all of the season using mechanical refrigeration supplemented by cooling with air from outdoors at night, it would have operated 42 days and the total seasonal load would have been 10,508,500 B.t.u., corresponding to an equivalent ice meltage of 34.5 tons. This represents a reduction in cooling load of 32.6 per cent resulting from the use of cooling with air from outdoors at night to supplement artificial cooling during the day.
It should be emphasized at this point, however, that this 32.6 per cent does not represent the reduction in the cost of operation, because, although the total number of hours that the fan was running was practically the same in the two cases, when cooling with air from outdoors at night more air was circulated, and hence the power input to the fan was greater than it was when the fan was used for the same number of hours to draw the air through the cooling coil.

Under the actual conditions of operation the plant was run from June 20 to July 21 with supplementary cooling with air from outdoors at night, and from July 21 to September 1 without supplementary cooling. If it had been operated from May 1 to July 21 with supplementary cooling and from July 21 to September 1 without supplementary cooling, it would have operated 47 days requiring cooling and the total seasonal load would have been 11,797,500 B.t.u., corresponding to an equivalent ice meltage of 38.7 tons. This is not comparable with the 27.5 tons shown by item 21 in Table 8 obtained from the experimental data for the part of the season from June 20 to September 1, because the former applies to the whole season and the latter to only a part of the season.

During the summer of 1932 data were obtained for the Research Residence when cooled with a plant employing ice as the cooling medium. It is of interest to compare the seasonal equivalent ice meltage that would have been obtained with this plant with that which would have been obtained with the mechanical refrigeration plant when operated during the same season and under similar operating conditions. Data were obtained for both plants when operated a part of a season without supplementary cooling with air from outdoors at night and when the house was equipped with awnings. The season of 1934 was selected as the common season, and the daily ice meltages from the curve for the house equipped with awnings from Fig. 14 corresponding to the degree-hours for each day in the season of 1934 were tabulated. This tabulation indicated that if the plant using ice as a cooling medium had been operated without supplementary cooling with air from outdoors at night and with the house equipped with awnings the total seasonal ice meltage would have been 58.8 tons. Under the same conditions of operation, it has been shown in a preceding part of this section that the plant employing mechanical refrigeration gave an equivalent ice meltage of 51.2 tons. Hence it is evident that the actual tonnage required for a given building is not greatly affected by the character of the plant. If supplementary cooling with air from outdoors at night were used in connection with the
plant employing ice as the cooling medium and for the season of 1934, the cooling load would have been reduced 32.6 per cent. Hence the consumption of ice would have been 36.9 tons as compared with 34.5 tons of equivalent ice meltage for the plant using mechanical refrigeration.

The value of 58.8 tons obtained by using the load distribution curve of Fig. 14 in connection with the daily distribution of degree-hours illustrates the fallacy of calculating the probable total cooling load for a second season by using the load for the first season and the ratio of the total seasonal degree-hours for the two seasons. The probable tonnage for the plant employing ice operated during the season of 1932 comprising 1471 degree-hours above 85 deg. F., and with the Residence equipped with awnings during the whole season, was shown in Section 25 to be 40.3 tons. By using the ratio of the seasonal degree-hours, a seasonal tonnage of \( \frac{2658 \times 40.3}{1471} = 72.8 \) would have been obtained for the season of 1934. This is considerably greater than the 58.8 tons obtained by the use of the distribution curves, and the difference is caused by the fact that the cooling load per degree-hour increases with the outdoor temperature. Hence, owing to the preponderance of milder weather the weighted mean cooling load per degree-hour is less than the arithmetical mean obtained by dividing the total tonnage by the total degree-hours for the season.

40. Cost of Operation.—The limitations applying to cost data have been mentioned in Section 26. With due appreciation of these limitations, however, operating costs are of some interest. The following operating costs are based on local rates of $4.00 a ton for ice, $0.031 a kw-hr. for electricity, and $0.33 per 1000 gallons for water. During periods of operation of the plant, when the fan was drawing air through the cooling coil, the average rate of consumption of electricity was 0.314 kw. During the periods when the fan was drawing air from outdoors at night the average rate of power input was 0.410 kw., and during the running periods of the compressor the average rate of power input was 2.836 kw.

Based on these rates and the total quantities shown in Table 8, the actual operating cost for the season as actually run, consisting of mechanical refrigeration supplemented by cooling with air from outdoors at night from June 20 to July 21, and mechanical refrigeration not supplemented by cooling with outdoor air at night from July 21
to September 1 was $43.75 for electricity, and $8.50 for water; or a total cost of $52.25. If the plant had been ready to operate on May 1, and had been operated from May 1 to July 21 with mechanical refrigeration supplemented by cooling with air from outdoors at night, and from July 21 to September 1 with mechanical refrigeration not supplemented by cooling with air from outdoors at night, an analysis of the data based on a detailed study of the daily performance and the daily degree-hours indicated that the compressor would have run for 407 hours, the fan would have run 732 hours drawing air through the cooling coil and 682 hours drawing air from outdoors at night, and 39 120 gallons of water would have been used. Under these conditions the operating cost would have been $51.60 for electricity and $12.91 for water, or a total of $64.51.

If the plant had been operated during the whole season from May 1 to September 1 using mechanical refrigeration not supplemented by cooling with air from outdoors at night the compressor would have run 538 hours, the fan drawing air through the cooling coil 24 hours a day would have run 1464 hours, and 51 650 gallons of water would have been used. Under these conditions the operating cost would have been $61.60 for electricity and $17.05 for water, or a total of $78.65. If the plant had been operated the whole season from May 1 to September 1 using mechanical refrigeration supplemented by cooling with air from outdoors at night the compressor would have run 362 hours, the fan drawing air through the cooling coil during part of the day would have run 510 hours, and drawing air from outdoors averaging 11 hours at night would have run 946 hours. The total amount of water used would have been 34 830 gallons. Under these conditions the operating cost would have been $48.87 for electricity and $11.50 for water, or a total of $60.37. Thus the reduction in operating cost effected by the use of cooling with air from outdoors at night to supplement artificial cooling during the day was 23.4 per cent as compared with the reduction in cooling load of 32.6 per cent, which was mentioned in Section 39.

During the periods of continuous operation the condensing unit developed a capacity of 29 000 B.t.u. per hour, or 2.41 tons of refrigeration. The electrical input to the compressor motor was 2.836 kw., and that to the fan was 0.814 kw. The condenser water required was 96.1 gallons per hour. The cost per hour for operating the plant under continuous load was 8.79 cents for the compressor, 0.97 cents for the fan, and 3.17 cents for water. The total cost per hour was, therefore, 12.93 cents, and the cost per ton of refrigeration delivered was 5.38
cents. As compared with this, the cost per ton of refrigeration delivered by the plant using ice as a cooling medium, discussed in Section 26, was 17.6 cents.

A direct comparison of the seasonal costs of operation for the plant using ice and the one using mechanical refrigeration is not possible owing to the fact that each plant was operated under two different conditions for parts of the season and that the number of degree-hours above 85 deg. F. was different for the two seasons. By using Fig. 14 and a similar curve for the hours of fan operation in connection with the daily degree-hours per day above 85 deg. F. for the season of 1934, however, it is possible to obtain the total ice meltage and hours of operation for the fan and pump which would have resulted if the plant using ice had been operated during the entire season of 1934. From this study it was determined that if the plant using ice had been operated during the season of 1934 without supplementary use of cooling with air from outdoors at night and with the Residence equipped with awnings the total ice meltage would have been 58.8 tons and the pump and fan would have run for 830 hours. With an electrical input of 0.265 kw. to the fan and 0.50 kw. to the pump motor the total seasonal cost of operation would have been $254.89. Under the same conditions, the seasonal cost for operating the plant using mechanical refrigeration has previously been estimated as $78.65. If supplementary cooling with air from outdoors at night were employed these seasonal costs would be reduced 23.4 per cent and the cost of operation for the plant using ice would be $191.00 as compared with $60.37 for operating the plant using mechanical refrigeration.

41. Conclusions.—The following conclusions may be drawn as applying to the Research Residence and the conditions under which the tests were conducted:

(1) An indoor temperature of approximately 80 deg. F. with relative humidity below 55 per cent results in satisfactory comfort conditions in the living quarters of a residence. For complete comfort in sleeping quarters a somewhat lower temperature is desirable.

(2) The introduction of approximately one air change per hour of outdoor air for the purpose of ventilation is sufficient to prevent objectionable odors.

(3) A mechanical refrigeration unit capable of producing 2½ tons of refrigeration is sufficient to maintain conditions of comfort on two stories of a residence similar to the Research Residence when the outdoor temperature does not exceed 103 deg. F. and an amount of out-
VII. RESULTS OF TESTS ON ROOM COOLING UNITS

42. Preliminary Statement.—This investigation was undertaken to determine a practical method for the control and operation of room cooling units, and to obtain performance data on such units.

43. Operating Temperatures.—One of the most important and at the same time probably one of the most debatable questions involved in space cooling is the one concerning the proper indoor temperature to be maintained, together with its relationship to the outdoor temperature. Sufficient data were obtained from preliminary tests on room cooling unit A, described in Section 9, to permit the construction of the diagrams shown in Fig. 29. These diagrams represent idealized schedules of outdoor and indoor temperatures and the approximate length of operating periods required by cooling unit A to maintain the indoor temperatures shown. Figure 29 shows three possible schedules of indoor temperatures, or methods of operating a room cooling unit. The schedule for the first method, designated as Case 1, is based on the assumption that the indoor temperature should at all times be maintained at a value 10 deg. F. less than the outdoor temperature. This condition is represented by the broken line, and the approximate indoor air temperature actually maintained by the operation of the room cooling unit is shown by the full line in the diagram. It may be noted that with this schedule of operation it was necessary to start the cooling unit a considerable length of time before the maximum outdoor temperature was reached. At the starting time the indoor temperature conditions are indicated by point b, and during the period indicated between the points b and c, the outdoor air temperature increased more rapidly than the indoor air temperature. It should also be noted that after the outdoor air temperature had reached a peak value and started to decline, it was necessary to progressively reduce the indoor air temperature in order to attempt to maintain the constant differential as specified by the schedule. This reduction in indoor air temperature, accompanied by a reduction in the temperature of the contents of the room, such as furniture, inside partition walls,
etc., in the late evening hours when the cooling load was just reaching a peak due to the lag in heat transmission through the outside walls, imposed such an exceedingly heavy load on the unit as to make it impossible to maintain the required indoor temperature, and thus to make this schedule of operation impractical. A unit or plant of sufficient size to maintain this schedule in the late afternoon or evening would be prohibitive in first cost and uneconomical to operate. Furthermore, if the indoor conditions were conducive to comfort during the peak outdoor temperature, there was no indication or reason to believe that the maintenance of these conditions resulted in discomfort after the peak had passed.

With the second method of operation, designated as Case 2 in Fig. 29, an attempt was made to maintain the schedule of desirable
indoor air temperatures given in Table 1, Section 12. This schedule provides for a variable differential between the indoor and outdoor temperatures, the differential increasing or decreasing progressively as the outdoor temperature increases or decreases. The difficulties encountered with this method of operation were similar to those discussed under Case 1, but somewhat less pronounced, because the differences between indoor and outdoor temperatures were not as great for this case as those required for Case 1. In addition, both methods required careful manual control and adjustment, which would be entirely impractical and unsatisfactory in a domestic installation.

With the third and more simple method of operation, which was finally adopted as a standard and is designated as Case 3 in Fig. 29, the cooling unit was started when the effective temperature reached some predetermined value, as for example 75 deg. F., irrespective of the outdoor temperature. This usually occurred when the indoor dry-bulb temperature reached a value between 78 and 81 deg. F., depending upon the relative humidity, although in some of the tests slightly lower dry-bulb temperatures were maintained. The indoor dry-bulb temperature was maintained practically constant by operating the unit intermittently through the action of a simple two-point thermostat placed in the electric fan circuit. This method of operation was very satisfactory, and the indoor temperature conditions were maintained by operating the unit until the outdoor effective temperature dropped to approximately the same value as the indoor effective temperature, at which time the unit was stopped and the windows were opened.

The choice of a particular dry-bulb temperature to be maintained on a given day for any domestic air cooling unit installation is governed by a number of factors, including that of economy of operation. Obviously, for economical reasons, it is advisable to maintain the highest indoor air temperature conducive to comfort. For climatic conditions similar to those of Urbana, Illinois, experience has shown that an indoor dry-bulb temperature of approximately 80 deg. F., with relative humidities between 40 and 60 per cent is very satisfactory, whereas for cooler climates it is probable that a slightly lower air temperature might be necessary. Regardless of the particular indoor air temperature to be maintained, there seems to be little need for complex schedules of operation such as those illustrated by Cases 1 and 2.

44. Performance of Cooling Unit A.—A typical performance record for unit A when cooling the living room only is shown in Fig. 30. In this test the unit was initially charged with 206.0 lb. of ice and the
dry bulb temperature of the air in the living room was maintained at a constant value of approximately 75.5 deg. F. The test was terminated at the end of 13.5 hours when the 206.0 lb. of ice were all melted, and it was merely a coincidence that at this time the effective temperature outdoors was approximately the same as the effective temperature in the living room. It is significant to note, however, that at this time the unit could have been stopped and cooling continued by the introduction of night air even if the ice supply had not been depleted.

The average dry-bulb and wet-bulb temperatures of the air entering and leaving the cooling unit during the periods of fan operation were as follows:

<table>
<thead>
<tr>
<th></th>
<th>Entering Air</th>
<th>Leaving Air</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dry bulb temperature</td>
<td>75.0 deg. F.</td>
<td>63.8 deg. F.</td>
</tr>
<tr>
<td>Wet bulb temperature</td>
<td>64.3 deg. F.</td>
<td>58.0 deg. F.</td>
</tr>
<tr>
<td>Dewpoint temperature</td>
<td>58.6 deg. F.</td>
<td>54.4 deg. F.</td>
</tr>
<tr>
<td>Relative humidity</td>
<td>56.0 per cent</td>
<td>71.0 per cent</td>
</tr>
<tr>
<td>Moisture content of the air</td>
<td>73.6 grains per lb. of dry air</td>
<td>63.0 grains per lb. of dry air</td>
</tr>
</tbody>
</table>

It should be noted that the air was cooled 11.2 deg. F. in passing through the unit, and that the moisture content was reduced 10.6 grains per pound of dry air.

The dry bulb temperature of the air in the living room was maintained at a level approximately 2 deg. F. lower than that in the uncooled hall due to the operation of the unit. It should also be noted...
that a comparable cooling effect was obtained by the reduction in the relative humidity. From Fig. 30 it may be observed that at 10:30 A.M., when the unit was started, the relative humidity of the air in the room was 68 per cent; at noon it had been lowered to 55 per cent, and between that time and 8:00 P.M. it ranged from 51 to 55 per cent. In cooling effect this reduction of approximately 15 points in relative humidity is equivalent to a reduction of 1.5 deg. F. in effective temperature, or to a 2.0 deg. F. reduction in dry-bulb temperature.

The total weight of moisture condensed from the air during the period when 206.0 lb. of ice were melted was 7.1 lb. This change in the moisture content of the air was equivalent to a heat absorption of approximately 7474 B.t.u. The total heat absorbed by the ice melting process was composed of the latent heat of fusion of the ice, plus the sensible heat absorbed by the water resulting from the meltage, and was equal to

\[
(206.0 \times 144) + 206.0 (57-32) = 34814 \text{ B.t.u.}
\]

144 = latent heat of fusion of ice, in B.t.u.
57 = temperature of water going to the drain, in deg. F.

Of the total amount of heat absorbed by the ice melting process, during the entire period of the test, approximately 21.5 per cent was given up by the change in the moisture content of the air, consisting of the total heat given up by the vapor condensed and the change in superheat of the vapor remaining in the air. As shown by the upper curve in Fig. 31, the amount of dehumidification accomplished with
this type of cooling unit was dependent on the percentage of the time that the fan in the unit was operating. For example, the ratio of the heat absorbed in the dehumidification process to the total heat absorbed varied from a value of 37.8 per cent when the fan in the unit was operating continuously, to a value of approximately 16.0 per cent when the fan was not running, and during which time the small amount of ice meltage was that due to standby heat leakage only.

The average rate of ice meltage, as indicated by the lower curve in Fig. 30, was approximately 15.3 lb. per hour for a period of 13.5 hours. The maximum meltage rate observed with the fan operating continuously was 43.8 lb. per hour with a room temperature of about 75 deg. F.

The values of heat absorption may be reduced to an equivalent ice melting capacity expressed as tons of refrigeration by dividing the heat absorption rate in B.t.u. per hour by 12000. Therefore, for the test shown in Fig. 30 the total heat absorption of 34814 B.t.u. for the 13.5 hour period was equivalent to 0.215 tons of refrigeration. The summary of results from six tests is presented in Fig. 31, and it may be noted from the lower curve that the ice melting capacity expressed in tons of refrigeration varied from a value of 0.51 tons with continuous fan operation to a value of 0.06 tons when the fan in the unit was not running.
45. Performance of Cooling Unit B.—The performance of cooling unit B was similar to that of unit A. However, it was possible to obtain a wider variation in cooling rates with unit B by making adjustments in the rate of circulation of the cooling water, by changing the speed of the fan, or by allowing the fan to run a larger proportion of the time. In the tests under consideration the rate of circulation of the cooling water and the fan speed remained constant. The amount of cooling depended entirely upon the length of time that the fan in the unit was operated, and Fig. 32 shows the sensitivity and positive- ness of this method of control as reflected in the temperature rise and fall in the cooling water, corresponding to the operation of the fan. Approximately one minute after the fan started or stopped, the temperature of the water leaving the cooling coil had practically reached equilibrium. For the test shown, the temperatures of the water entering and leaving the coil were approximately 34 deg. F. and 40 deg. F., respectively, giving a temperature rise of about 6 deg. F. through the coil.

Figure 33 shows a typical performance curve with the unit located in the living room, as shown in Fig. 2, and cooling the entire first story consisting of the living room, hall, dining room, breakfast nook and kitchen; or a total volume of 7300 cu. ft. It may be noted that the outdoor temperature reached a maximum value of 93.7 deg. F. at
about 3:30 P.M. and that the temperature in the southwest bedroom on
the uncooled second story attained a maximum value of 87.5 deg. F. at
about 7:30 P.M. During the test period of 11.67 hours it was possible to
maintain average breathing level temperatures of 79.3 deg. F. in the
living room, 81.5 deg. F. in the hall, 82.6 deg. F. in the dining room,
and 82.1 deg. F. in the kitchen, or an average of 81.4 deg. F. for the
entire first story. Although the cooling unit and control thermostat
were unfavorably located with respect to the center of the first story,
the air temperatures in any one room did not deviate more than 2.1
deg. F. from the mean temperature for the entire first story. The
minimum room temperature of 79.3 deg. F. occurred in the living room
where the cooling unit was located, while the maximum room tempera-
ture of 82.6 deg. F. occurred in the dining room. Unquestionably, this
variation in air temperature between rooms would have been reduced
by a more central location of the thermostat and unit. It is also prob-
able that, from the standpoint of temperature equalization in several
rooms, the best method of operation for a room cooling unit of this type
is that which will allow the unit to operate for the greatest portion of
the time, at a rate of cool air delivery just sufficient to balance the
cooling load. With such a unit equipped with a multiple-speed fan this
method of operation can be most closely approximated by running the
fan at the lowest available speed which will handle the load; whereas,
in a single-speed fan unit the adjustment could be made either by em-
ploying adjustable louvers to control the amount of air delivered, or
by using a control valve to vary the rate of water being circulated
through the cooling coil.

The total ice meltage for the test shown in Fig. 33 was 500 lb., and
the total amount of heat absorbed was 82,224 B.t.u., which included
the net rise in the sensible heat of the water in the ice tank.

The average rate of heat absorption, reduced to an equivalent ice
melting capacity, and expressed as tons of refrigeration was 0.59 tons.
In general, the tests with both units A and B on the first story indi-
cated that a cooling unit with a capacity of approximately 0.25 tons
of refrigeration per room would be required to satisfy the maximum
cooling requirements of rooms similar to those in which these tests
were made, and in which no outdoor air other than that which occurs
by natural infiltration was introduced for the purpose of ventilation.
It should be noted that the windows of these rooms which were exposed
to the sun were equipped with awnings, and that the occupancy load
consisted of only one person per room. There was practically no
power load. The estimate of 0.25 tons of refrigeration per room is
INVESTIGATION OF SUMMER COOLING IN RESIDENCE

based on normal running conditions, since starting conditions impose an extra load of considerable magnitude. It is very probable that a larger tonnage capacity per room would be necessary for second-story rooms since the cooling load for the second story is considerably greater than that for the first story and the cooled air would tend to drift downwards to the first story.

The total amount of moisture condensed out of the air during the entire test lasting 11.67 hours was 21.22 lb., and the ratio of the heat absorbed by the change in the moisture content of the air to the total heat absorbed was 27.2 per cent. The great magnitude of the dehumidification load is emphasized by the lower curve in Fig. 33, showing the rate of dehumidification, or the weight of water condensed out of the air in pounds per hour. It may be observed that the initial rate at the beginning of the test was approximately 5.0 lb. per hour, while the average rate for the entire test period was only 1.82 lb. per hour. As shown by the center group of curves in Fig. 33, this drying process substantially reduced the relative humidity of the air in all the rooms on the first story. The average initial relative humidity of approximately 68 per cent was reduced to a value of 58 per cent at the end of one hour's operation, and at the end of five hours' operation it had attained a value of about 50 per cent. Since the dry-bulb temperature remained constant, the reduction in relative humidity also represented a reduction in moisture content, or the pounds of moisture present per pound of air. The fact that during the operation of the cooling unit the moisture content of the air on the second story remained constant indicated that the relative humidities in the rooms on the second story were also somewhat reduced by the dehumidification accompanying the cooling on the first story, since it was observed on other days that when the cooling unit was not operated a gradual increase in the moisture content of the air on the second story always occurred during the day.

46. General Remarks.—Although these tests indicate that a room cooling unit can be successfully adapted to cool from one to three rooms on the same floor of a residence, it is probable that the most economical method of cooling would be to operate such units in conjunction with a fan unit for the purpose of introducing cool air from outdoors at night into the rooms or house. It would be feasible for instance, to locate the cooling unit on the first story of a two-story house and operate it during the afternoon and evening, while the second-story rooms would be cooled at night by introducing outdoor air through the open windows.
The advantageous features of portability and lower initial cost of a unit similar to unit A are somewhat offset by the inconvenience of having the entire unit in the living quarters. It would probably be preferable to locate the unit in the basement and attach a short duct system from the outlet of the unit to a register on the first story, thereby avoiding the necessity for servicing the unit in the living portion of the house.

Adequate insulation of the ice storage compartment in all types of units is very essential for the minimization of standby losses. The cooling load during the summer was extremely variable with the weather conditions fluctuating rapidly, and oftentimes it was found that after the ice storage tank was filled, very little cooling was necessary due to sudden temperature changes outdoors. Under these conditions standby losses become expensive if any appreciable ice meltage takes place between operations. With units similar to type B it is essential that all pipes carrying relatively cold refrigerants be thoroughly insulated.

The matter of noise of mechanical parts and of air motion is extremely important, and great care should be taken in the design and installation of any unit to reduce these factors to a minimum.

47. Conclusions.—The following conclusions are applicable to the conditions under which these tests were made:

(1) Room cooling units should be operated and controlled to maintain a constant dry-bulb temperature in the room, rather than to maintain a complex schedule of indoor temperatures varying with the outdoor temperature.

(2) Room cooling units similar to units A and B can be successfully adapted to cool from one to three medium-sized rooms on the same floor of a residence.

(3) A room cooling unit of 300 lb. ice capacity and having a melting rate of approximately 20 lb. of ice per hour would be adequate to satisfy the normal cooling requirements of a medium-sized first-story room of a residence for most summer days, if no outdoor air other than that which enters by natural infiltration is introduced for ventilation. However, if the occupancy of the room is to be more than two persons it is essential that additional capacity be supplied for the necessary ventilation required to reduce undesirable odors.

(4) The initial ice melting rate, or the melting rate during the first hour after the unit was started, was nearly twice as great as the normal melting rate.
INVESTIGATION OF SUMMER COOLING IN RESIDENCE

(5) Of the total amount of heat absorbed by the room cooling units, approximately from 16 to 38 per cent was absorbed in the dehumidification process.

(6) In addition to the lowering of the dry-bulb temperature the relative humidity of the air in the rooms was, in most cases, reduced about 15 points after three or more hours of continuous operation of the units.

(7) Although not centrally located on the first story of the residence, the cooling units operated in the living room effectively cooled and dehumidified the air in the adjoining rooms, and there was some indication that the reduction in relative humidity extended to the second story.

(8) With room cooling unit B located in the living room and all interconnecting doors on the first story open, the maximum deviation of the breathing level temperature in any room on the first story from the mean temperature of the entire first story was 2.1 deg. F.

VIII. ESTIMATION OF SEASONAL DEMANDS

48. Preliminary Statement.—The actual daily cooling load imposed on a plant used to cool a given structure is dependent on the outdoor temperature, and hence is a function of the number of degree-hours above some selected base occurring on that day. The number of degree-hours occurring on a given day may therefore be regarded as a measure of the potential demand on the plant for that day, and the total number of degree-hours for the season may be regarded as a measure of the seasonal demand. Hence, for the purpose of estimating the probable cost of operation, some estimate must be made of the probable seasonal demand for the coming season. Such an estimate must be based on a knowledge of the seasonal demands of past seasons in the given locality.

The number of degree-hours above any given base temperature is the product of the difference between the base temperature and the mean temperature over the period in hours during which the outdoor temperature is equal to or exceeds the base temperature, and the total length of the period over which this occurs. This procedure involves a knowledge of the hourly temperatures occurring for each day of the season. In many localities such data are not available, since many local weather stations make a practice of observing only the daily maximum and minimum temperatures, and those occurring at 7:00
## Table 9
### Summary of Total Days, Hours, and Degree-Hours Above Base Temperatures of 85, 90, and 95 deg. F. for Summer of 1932
#### At Urbana, Illinois

<table>
<thead>
<tr>
<th>Month</th>
<th>Total Days</th>
<th>Total Hours</th>
<th>Total Degree-Hours</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>85 deg. F. and Above</td>
<td>90 deg. F. and Above</td>
<td>95 deg. F. and Above</td>
</tr>
<tr>
<td>June</td>
<td>18</td>
<td>5</td>
<td>0</td>
</tr>
<tr>
<td>July</td>
<td>22</td>
<td>14</td>
<td>8</td>
</tr>
<tr>
<td>August</td>
<td>18</td>
<td>6</td>
<td>0</td>
</tr>
<tr>
<td>September</td>
<td>4</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Total</td>
<td>62</td>
<td>25</td>
<td>8</td>
</tr>
</tbody>
</table>
A.M., 2:00 P.M., and 7:00 P.M. In most localities, however, records of the daily maximum temperatures are available, and if the degree-hours could be correlated with these daily maximum temperatures with a reasonable degree of accuracy it would afford a means of estimating the seasonal demands for the season under consideration. The following discussion suggests a method for such a correlation.

49. Correlation of Degree-Hours with Maximum Outdoor Temperatures.—Table 9 gives a summary of the actual number of days, hours, and degree-hours above bases of 85, 90 and 95 deg. F. for the summer of 1932 at Urbana, Illinois, based on hourly data observed at the Research Residence. From this table it is evident that the choice of a base temperature is an important item in determining the number of degree-hours in a cooling season. If it is assumed that no cooling will be required until the outdoor temperature reaches 90 deg. F. then the summer of 1932 would have had a total of 409 degree-hours requiring cooling. If, however, a base of 85 deg. F. is assumed, then that summer would have had a total of 1471 degree-hours, or 3.6 times the amount for a 90 deg. F. base.

The uniformity of the curves of outdoor temperature plotted against the time of day, as in Figs. 10 and 11, suggested the possibility that there might be some correlation between the maximum daily temperatures and the number of hours and degree-hours above any given base temperature. The daily outdoor temperatures, in general, followed a cycle similar to the one shown as an inset in Fig. 34. Therefore, if a given base temperature was selected, the length of the chord between the two points at which this base temperature line intersected the curve of outdoor temperatures represented the number of hours for the given day that temperatures at, or above, the base temperature prevailed. Also, the area between the base line and the temperature curve above the base line, shown cross-hatched in Fig. 34, represented the number of degree-hours above the given base. This procedure was followed for each of the 62 days during the summer of 1932 for which the maximum temperature reached 85 deg. F. or above, using three separate base lines, at 85, 90, and 95 deg. F. The resulting number of hours and degree-hours for each day were then plotted against the maximum outdoor temperature for that day, as shown in the two groups of curves in Fig. 34. It was then possible to draw three curves, for base temperatures of 85, 90, and 95 deg. F., representing the average of the points with a fair degree of accuracy. The spread of the points accordingly represents the deviations of the actual
Fig. 34. Curves Showing Relation Between Maximum Outdoor Air Temperature and Number of Degree-Hours per Day Above Stated Base Temperature
cycles for the individual days from the ideal average cycle represented by the curves. As an example, if a base temperature of 85 deg. F. is selected, and the maximum temperature of 92 deg. F. was attained on a given day, then, from the upper group of curves in Fig. 34, a total of 7.2 hours with temperatures above 85 deg. F. could be expected for that day, and from the lower group of curves, a total of 30 degree-hours would be probable. There is no reason to believe that the character of the daily temperature cycles used in constructing these curves is peculiar to Urbana, Illinois, alone, and therefore it is probable that the curves in Fig. 34 are applicable to a wide range of localities. Hence, it should be possible to estimate the probable number of hours and degree-hours above a selected base for any season in a given locality, by counting the total number of days occurring at various maximum temperatures from the U. S. Weather Bureau reports for the locality, and multiplying the number of days thus obtained for each maximum temperature by the number of hours or degree-hours obtained from the curves in Fig. 34 corresponding to the various maximum temperatures selected. The summation of the hours or degree-hours thus obtained represents the total for the season.

An estimation of the number of degree-hours above 85 deg. F. for each of the summers for the years from 1923 to 1934 at Urbana, Illinois, has been made in this manner, and is shown in Table 10. In compiling this table, the various ranges of outdoor temperatures have been grouped as shown in column 1. The mean temperatures for the various ranges are given in column 2, and the number of degree-hours per day, as read from the curves in Fig. 34 corresponding to the mean temperatures in column 2, are given in column 3. In the two columns for each year shown, the first column is the total number of days included in the months of May, June, July, August, and September, having maximum temperatures falling between the limits shown in column 1, and the second column is the product of the number of days and the corresponding number of degree-hours per day from column 3. The first three columns in this table may be considered as applicable to any locality, while the balance of the table applies to Urbana only. The ranges could be made smaller than those shown in column 1, but greater accuracy at this point is hardly warranted when it is considered that the final result is only a reasonably close approximation at best.

That it is a reasonably close approximation is indicated by the fact that the number of degree-hours for the summer of 1932, calculated from the summation of the actual degree-hours for the individual
Table 10

Number of Days of Stated Maximum Temperature and Estimated Number of Degree-Hours Above 85 deg. F. for Season from May 1 to October 1 at Urbana, Illinois

<table>
<thead>
<tr>
<th>Range of Maximum Outdoor Temperature deg. F.</th>
<th>Mean Range deg. F.</th>
<th>Number of Degree-Hours Above 85 deg. F. per Day (from Fig. 34 at Mean of Range)</th>
<th>1934</th>
<th>1933</th>
<th>1932</th>
<th>1931</th>
<th>1930</th>
<th>1929</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Number</td>
<td>Total</td>
<td>Number</td>
<td>Total</td>
<td>Number</td>
<td>Total</td>
<td>Number</td>
<td>Total</td>
</tr>
<tr>
<td></td>
<td>of Days</td>
<td>of Degree-Hours</td>
<td>of Days</td>
<td>of Degree-Hours</td>
<td>of Days</td>
<td>of Degree-Hours</td>
<td>of Days</td>
<td>of Degree-Hours</td>
</tr>
<tr>
<td>85-87</td>
<td>86</td>
<td>1.0</td>
<td>14</td>
<td>14</td>
<td>20</td>
<td>20</td>
<td>23</td>
<td>23</td>
</tr>
<tr>
<td>88-90</td>
<td>89</td>
<td>10.5</td>
<td>12</td>
<td>126</td>
<td>17</td>
<td>179</td>
<td>16</td>
<td>168</td>
</tr>
<tr>
<td>91-93</td>
<td>92</td>
<td>30.5</td>
<td>14</td>
<td>427</td>
<td>18</td>
<td>549</td>
<td>11</td>
<td>336</td>
</tr>
<tr>
<td>94-96</td>
<td>95</td>
<td>56.0</td>
<td>10</td>
<td>360</td>
<td>12</td>
<td>672</td>
<td>4</td>
<td>224</td>
</tr>
<tr>
<td>97-99</td>
<td>98</td>
<td>86.0</td>
<td>7</td>
<td>602</td>
<td>7</td>
<td>602</td>
<td>6</td>
<td>516</td>
</tr>
<tr>
<td>100-102</td>
<td>101</td>
<td>119.0</td>
<td>6</td>
<td>714</td>
<td>0</td>
<td>0</td>
<td>1</td>
<td>119</td>
</tr>
<tr>
<td>103-105</td>
<td>104</td>
<td>162.0</td>
<td>2</td>
<td>304</td>
<td>0</td>
<td>0</td>
<td>1</td>
<td>102</td>
</tr>
<tr>
<td>Total</td>
<td>65</td>
<td>2747</td>
<td>74</td>
<td>2022</td>
<td>62</td>
<td>1538</td>
<td>70</td>
<td>2053</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Number of Days</th>
<th>Total Degree-Hours</th>
</tr>
</thead>
<tbody>
<tr>
<td>1929</td>
<td>17</td>
</tr>
<tr>
<td>1930</td>
<td>15</td>
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<tr>
<td>1931</td>
<td>13</td>
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<td>1932</td>
<td>13</td>
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<tr>
<td>1933</td>
<td>14</td>
</tr>
<tr>
<td>1934</td>
<td>16</td>
</tr>
<tr>
<td>1935</td>
<td>16</td>
</tr>
</tbody>
</table>

122
<table>
<thead>
<tr>
<th>Year</th>
<th>Mean Range of Temperature deg. F.</th>
<th>Number of Days</th>
<th>Total Degree-Hours of Days</th>
</tr>
</thead>
<tbody>
<tr>
<td>1923</td>
<td>55-67</td>
<td>14</td>
<td>120</td>
</tr>
<tr>
<td>1924</td>
<td>60-69</td>
<td>12</td>
<td>120</td>
</tr>
<tr>
<td>1925</td>
<td>64-70</td>
<td>12</td>
<td>120</td>
</tr>
<tr>
<td>1926</td>
<td>64-70</td>
<td>12</td>
<td>120</td>
</tr>
<tr>
<td>1927</td>
<td>64-70</td>
<td>12</td>
<td>120</td>
</tr>
<tr>
<td>1928</td>
<td>64-70</td>
<td>12</td>
<td>120</td>
</tr>
<tr>
<td>1929</td>
<td>64-70</td>
<td>12</td>
<td>120</td>
</tr>
</tbody>
</table>

Total: 776

INVESTIGATION OF SUMMER COOLING IN RESIDENCE

FROM MAY 1 TO OCTOBER 1 AT URBANA, ILLINOIS
days, was 1471, while the number calculated from the average curves and shown in Table 10 was 1538, or a deviation of 4.2 per cent from the actual. A similar approximation for the number of hours above 85 deg. F. gave 318 as compared with the actual number of 329, or a difference of 3 per cent. While the curves in Fig. 34 were derived from the data for one season only, their applicability to different seasons can be shown by comparing the results for the years of 1933 and 1934 from Table 10 with the data obtained from the continuous records made at the Research Residence for those seasons. For the summer of 1934 a total of 2658 degree-hours was obtained from the daily charts made by the recording thermometer, as compared with 2747 shown in Table 10 as calculated from the curves. This represents a deviation of 3.5 per cent. For the summer of 1933 the actual number of degree-hours was 2310 as compared with 2022 shown in Table 10, representing a deviation of 12 per cent. The larger deviation for 1933 is probably caused by the fact that this season contained no days with a maximum outdoor temperature as high as 100 deg. F., while both the seasons of 1932 and 1934, and the data from which the curves in Fig. 34 were derived contained a number of such days.

50. Variability of Seasonal Demands.—From Table 10 it is evident that the cooling season is extremely variable, even in the same locality. During the twelve years, it varied from a maximum of 2747 degree-
hours to a minimum of 327 degree-hours. The latter represents only about one-eighth of the maximum. Furthermore, during this twelve-year period there were four years in which the number of degree-hours was less than one-fourth of the maximum, six years in which it was less than one-third, and eight years in which it was less than two-thirds of the maximum. That the cycle represented by the twelve years selected presents nothing unusual in the way of variation is shown by Fig. 35, in which the averages of all of the maximum temperatures for each day in the months of June, July, August, and September are plotted against the years, over a period of 47 years. It may be noted also from Fig. 35 that the shape of the degree-hour curve, plotted from the data in Table 10, closely approximates the shape of the average maximum temperature curve. Hence, the variations shown by the twelve-year cycle selected may be regarded as typical of the variations that might occur in any twelve-year cycle during the whole period of 47 years, and both the curves and the table serve to emphasize the futility of any attempt to calculate the probable cooling load for a given season from a mean value of the number of degree-hours based on the average for a cycle of seasons.

51. Moisture Content of Outdoor Air.—The number of degree-hours above some given base, while offering a reasonably accurate index, is not a complete measure of the load actually imposed on a cooling plant. Part of this load consists of the moisture load, or as it is usually designated, the latent heat load. This may be separated into the load resulting from moisture carried in by infiltration of outdoor air or by air introduced for ventilation, and the load resulting from moisture given up by people or by other forms of evaporation within the structure itself. The latter is independent of outdoor temperature and must be estimated from what is known of the probable occupancy and the character of any processes, such as cooking, resulting in evaporation. The former is also to a certain extent independent of the outdoor dry-bulb temperature, and therefore of the number of degree-hours, and must be estimated from the probable moisture content of the outdoor air.

Figure 36 shows a plot of the moisture content of the outdoor air for Urbana, Illinois, corresponding to different outdoor dry-bulb temperatures. This plot includes all of the summers from 1927 to 1935, and is based on observations made at 2:00 P.M. It may be noted that for any given dry-bulb temperature less than 95 deg. F., the moisture content of the outdoor air may vary between rather wide limits. For every dry-bulb temperature, however, there are definite upper and
lower limits within which the value for the moisture content will be found. Hence, the chart may be useful in estimating the limiting latent heat load which might be imposed on a given cooling plant.

52. Conclusions.—The following conclusions may be drawn from the data presented:

(1) The daily degree-hours can be correlated with the daily maximum outdoor temperatures with sufficient accuracy to permit the estimation of the seasonal demands for one season from the correlation obtained from the weather reports for a different season.

(2) The seasonal cooling requirements are extremely variable from year to year, and the ratio of the number of degree-hours exhibited by any two seasons occurring within a twelve-year period may be as high as 8.5 to 1. Hence, an average value for the degree-hours per season is comparatively meaningless.
APPENDIX A

EFFECTIVE TEMPERATURES

Under normal conditions, in an environment in which the temperature of the surrounding walls does not differ materially from that of the air, comfort is produced largely through the influence of air temperature, air motion, and relative humidity. The latter can be represented by wet-bulb and dry-bulb temperatures, observed on two thermometers, one of which has the bulb encased in a cloth sheath moistened with distilled water. When air is blown over these thermometers, or they are moved through the air, the dry-bulb thermometer reads a temperature representative of that of the air, and the wet-bulb thermometer reads a lower temperature depending on the relative humidity of the air. These thermometers must always be read with air moving over the bulbs. A chart showing the relations between the dry-bulb temperature, the wet-bulb temperature, and the relative humidity of the air is given in Fig. 37.

Effective temperature lines, or lines representing a composite temperature serving as an index to express the combined thermal effect of the dry-bulb temperature, wet-bulb temperature, and relative humidity existing in a given environment, are shown in Fig. 37.* These lines were established at the Research Laboratory of the American Society of Heating and Ventilating Engineers, and represent the various combinations of wet-bulb and dry-bulb temperatures giving the same feeling of warmth or coldness to an individual seated in still air, or air having a movement of less than 25 ft. per min. That is, all combinations of wet-bulb and dry-bulb temperatures located on a given effective temperature line on the chart are accompanied by the same feeling of warmth or coldness. As an arbitrary scale of effective temperatures, a scale was selected such that it coincided with the wet-bulb temperatures shown along the saturation, or 100-per-cent relative humidity, line on the chart. Hence, effective temperature may also be defined as the temperature of saturated still air which will give the same feeling of warmth or coldness as the combination of wet-bulb and dry-bulb temperatures actually existing in the environment under consideration.

Effective temperature is a reliable index of the feeling of warmth or coldness, but it is not in itself a comprehensive index of comfort.

That is, while it is true that at a given effective temperature which is comfortable at a relative humidity of 50 per cent, an equal feeling of warmth may exist over the whole range of relative humidities from 0 to 100 per cent, it is not necessarily true that an equal degree of comfort exists over the whole range. High relative humidities produce sensations of dampness and stickiness, while low relative humidities produce a sensation of excessive dryness; neither of which may be regarded as comfortable. However, the work on winter comfort con-
ducted at the Research Laboratory of the American Society of Heating and Ventilating Engineers* and later studies on summer comfort carried on at the Harvard School of Public Health† proved that, within reasonable limits, the effective temperature is a fair index of comfort. The experimental data were confined to the range of relative humidities between 30 and 70 per cent. Hence the comfort zones, shown as the cross-hatched portions of the chart in Fig. 37, were limited to a range in relative humidity of from 30 to 70 per cent. During the course of this investigation the sense impressions of a large number of subjects were obtained when they were exposed to a wide range of different combinations of dry-bulb temperature and relative humidity. From the sense impressions of these different subjects it was established that, when normally clothed for indoor activities during the winter, 97 per cent of them felt comfortably warm when the effective temperature was 66 deg. F. This effective temperature was therefore regarded as the optimum for comfort in the winter. It was also determined that 50 per cent of the subjects felt comfortable at effective temperatures of 71 and 63 deg. F., and these effective temperatures were regarded as the upper and lower practical limits of the winter comfort zone. In the same manner, an effective temperature of 71 deg. F. was determined as the optimum for people normally clothed during the summer, and the upper and lower limits of the summer comfort zone were established as 75 and 66 deg. F. effective temperature. Both of these zones are shown as cross-hatched portions in Fig. 37, and any combination of dry-bulb temperature and relative humidity that lies within these zones in the winter or in the summer, respectively, may be regarded as conducive to a reasonable degree of comfort.

APPENDIX B

METHOD USED FOR CALCULATION OF COOLING LOAD

1. General Statement.—The total cooling load imposed on a cooling plant consists of the sum of all heat gains by the structure resulting from indoor-outdoor temperature differences and sun effects on walls and windows, and of all other heat quantities introduced by air in-leakage and heat given up by the occupants and by mechanical or electrical processes. For convenience in calculation, these heat quantities may be itemized as follows:

   Item (1); Heat Transmission Through Walls Not Exposed to Sun
   Item (2); Heat Transmission Through Walls Exposed to Sun
   Item (3); Heat Transmission Through Windows and Doors Not Exposed to Sun
   Item (4); Heat Transmission Through Windows Exposed to Sun
   Item (5); Heat Transmission Through Ceilings
   Item (6); Heat Transmission Through Floors
   Item (7); Heat Brought in by Infiltration or Ventilation Air
   Item (8); Heat Given off by Occupants
   Item (9); Heat Given off by Electric Lights
   Item (10); Heat Given off by Electric Motors

In computing the cooling load for the Research Residence, the values of coefficients of heat transmission and enthalpies of mixtures of water vapor and air as given in the American Society of Heating and Ventilating Engineers' Guide for 1934 have been used. Wherever other sources of information were consulted references to such sources have been given. The following computations all apply to conditions existing at the Research Residence at 2:00 P.M. on June 27, 1934.

2. Item (1); Heat Transmission Through Walls Not Exposed to Sun.—The heat transmission through walls not exposed to the sun and through partition walls was computed from the equation

\[ H = AU (t_o - t_i) \]

(1)

in which \( H \) = heat transmitted in B.t.u. per hr.
\( A \) = wall area, in sq. ft.
\( U \) = coefficient of heat transmission, in B.t.u. per sq. ft. per hr. per deg. F. temperature difference from air to air.
\( t_o \) = outdoor temperature, in deg. F.
\( t_i \) = indoor temperature, in deg. F.
In the Research Residence three types of walls were involved, as follows:

(a) Frame wall consisting of weatherboard, building paper, sheathing, 6-in. studding, wood lath and plaster. The total area of this type of wall was 1767 sq. ft. However, since the portion of this wall which was exposed to the sun at any particular time is separately treated, only the portion not exposed to the sun was used in computing the heat gain by Equation (1). At the time considered this was 1304 sq. ft. The value of $U$ used was 0.25, (Guide 1934, p. 85, Table 10, Wall No. 41 A.)

(b) The chimney was considered as equivalent to a wall section consisting of 12-in. brick with wood lath and plaster finish. The total area not exposed to the sun at the time considered was 117 sq. ft., and the value of $U$ used was 0.24, (Guide 1934, p. 83, Table 8, Wall No. 2 C.)

(c) Partition wall between sun room and dining room, consisting of 6-in. studding with wood lath and plaster on both sides. The total area of this wall was 100 sq. ft., and the value of $U$ used was 0.34, (Guide 1934, p. 86, Table 11, Wall No. 53 B.) The temperature in the sun-room was assumed to be the same as that outdoors.

Sample calculation at 2 p.m., June 27, 1934:

$$H_{1a} = 1304 \times 0.25 (97.1 - 79.9) = 5607 \text{ B.t.u. per hr.}$$

$$H_{1b} = 117 \times 0.24 (97.1 - 79.9) = 483 \text{ B.t.u. per hr.}$$

$$H_{1c} = 100 \times 0.34 (97.1 - 79.9) = 585 \text{ B.t.u. per hr.}$$

Total $H$ for walls = 6675 B.t.u. per hr.

3. Item (2); Heat Transmission Through Walls Exposed to Sun.—The intensity of the sun is generally expressed as the B.t.u. received per hour on one sq. ft. of surface normal to the sun. The intensity received normal to a surface which is in itself not normal to the sun’s rays is dependent on the angle of incidence between the sun’s rays and the surface. For such surfaces the angle of incidence is a function of the orientation of the surface, the month, the day of the month, and the time of day. Curves giving the relation between the time of day and the solar intensity normal to the sun, normal to a horizontal surface, and normal to south, east, and west walls for the months of July, August, and September have been developed at the Research Laboratory* of the American Society of Heating and Ventilating Engineers. These curves are for Pittsburgh, Pennsylvania, but are equally ap-
Applicable to Urbana, Illinois, which is in practically the same latitude.

A set of these curves for July 1 is shown in Fig. 38.

Of the sun’s intensity, \( I \), normal to a given wall, which may be read from the curves in Fig. 38, a portion \( eI \) is absorbed by the surface and a portion, \( (1 - e) \ I \), is reflected from the surface. The portion \( eI \) raises the surface temperature, and a part of it is lost by radiation from the surface and by convection to the outdoor air. The remaining portion is transmitted through the wall. In the case of such sun-exposed walls Equation (1) for heat transfer from air to air on the two sides of the wall is not applicable. The following equations,* however, may be written as applying to this case:

\[
e I = H_1 + H_2, \quad \text{or} \quad H_2 = eI - H_1
\]

\[
H_1 = f_o(t_x - t_o)
\]

\[
H_2 = C(t_x - t_y)
\]

\[
H_2 = f_i(t_y - t_i)
\]

In which \( I \) = solar intensity normal to the surface, in B.t.u. per sq. ft. per hr.

\( e \) = coefficient of absorption of solar radiation.

\( H_1 \) = heat lost from outside surface by radiation and convection, in B.t.u. per sq. ft. per hr.

---

$H_2 =$ heat transmitted through wall, in B.t.u. per sq. ft. per hr.

$f_o =$ outside surface coefficient, in B.t.u. per sq. ft. per hr. per deg. difference in temperature between the surface and the air.

$f_i =$ inside surface coefficient, in B.t.u. per sq. ft. per hr. per deg. difference in temperature between the surface and the air.

$C =$ conductance of wall, in B.t.u. per sq. ft. per hr. per deg. difference in temperature from surface to surface.

$t_o =$ outdoor air temperature, in deg. F.

$t_i =$ indoor air temperature, in deg. F.

$t_s =$ outdoor surface temperature, in deg. F.

$t_v =$ indoor surface temperature, in deg. F.

The general solution of Equations (2) to (4) gives

$$H_2 = \frac{e I + f_o (t_o - t_i)}{1 + f_o \left( \frac{1}{C} + \frac{1}{f_i} \right)} \quad (6)$$

By introducing the equation for the overall coefficient of heat transfer, $U$, or the heat transferred per sq. ft. per hr. per deg. difference in temperature from air to air

$$U = \frac{1}{\frac{1}{f_i} + \frac{1}{f_o} + \frac{1}{C}} \quad (7)$$

Equation (6) may be reduced to the following form:

$$H_2 = \frac{U}{f_o} \left[ e I + f_o (t_o - t_i) \right] \quad (8)$$

For the total area, $A$, of wall involved,

$$H_s = H_2 \times A = \frac{AU}{f_o} \left[ e I + f_o (t_o - t_i) \right] \quad (9)$$

The values of the coefficients used in Equation (9) were $U = 0.25$, (Guide 1934, p. 85, Table 10, Wall No. 41A), and $f_o = 6.00$, (Guide 1934, p. 82, Table 7). Values of $e$ for clean white paint ranging from 0.20 to 0.30 were given by different authorities, and a value of 0.40 was used as being representative of weathered white paint stained with soot and dirt. It was determined by observations made at the
Research Residence that when the sun was shining on a wall only about 50 per cent of the wall area was actually exposed to the sun’s rays. The balance was shaded by shutters and other obstructions. For the south and west walls the total areas of the wall surfaces were 530 and 396 sq. ft. respectively. The actual sun-exposed areas were therefore 265 and 198 sq. ft. At 2 p.m. on June 27, the values of sun’s intensity on the south and west walls, read from Fig. 38, were 63 and 144 B.t.u. per sq. ft. per hr. respectively.

Sample calculation for 2 p.m., June 27, 1934:

South wall,

\[
H_s = \frac{265 \times 0.25}{6.00} \left[ 0.4 \times 63 + 6.00(97.1 - 79.9) \right] = 1420 \text{ B.t.u. per hr.}
\]

West wall,

\[
H_s = \frac{198 \times 0.25}{6.00} \left[ 0.4 \times 144 + 6.00(97.1 - 79.9) \right] = 1330 \text{ B.t.u. per hr.}
\]

Total heat transfer through walls exposed to sun = 2750 B.t.u. per hr.

4. Item (3); Heat Transmission Through Windows and Doors Not Exposed to Sun.—The heat transmission through windows and doors was calculated from Equation (1), Section 2. For windows the value of \( U \) used was 1.13, (Guide 1934, p. 92, Table 8). This value of \( U \) applies to the window as a whole including sash and any separating strips between panes. All windows were shaded by awnings, and the indirect sun effect was obtained by using the glass area alone of the windows on the sides exposed to the sun, as explained in Section 5. Hence the area used in Equation (1) was obtained by subtracting the total area of the glass alone on the sun-exposed sides from the total area of the openings for all windows. That is, it included the total area of openings on the sides of the house not exposed to the sun and the wooden sash and dividing strips on the sides exposed to the sun. At 2:00 p.m. this area was 243.7 sq. ft.

The front door was considered separately. Its area was 24.5 sq. ft. and a value for \( U \) of 0.52 was used, (Guide 1934, p. 92, Table 18).

Sample calculation for 2:00 p.m., June 27, 1934:

Windows, \( H = 243.7 \times 1.13 (97.1 - 79.9) = 4735 \text{ B.t.u. per hr.} \)

Door, \( H = 24.5 \times 0.52 (97.1 - 79.9) = 219 \text{ B.t.u. per hr.} \)

Total = 4954 B.t.u. per hr.

5. Item (4); Heat Transmission Through Windows Exposed to Sun.—The studies of F. C. Houghten* and others indicated that the total

heat gain through a glass area shaded by an awning was 28 per cent of the total heat gain for the same glass area unshaded by an awning and exposed to the sun. Since the latter amounted to approximately 97 per cent of the sun's intensity normal to the window surface, it is sufficiently accurate for all practical purposes to regard the heat gain through a glass area shaded by an awning on the sunny side of a building as equal to 28 per cent of the sun's intensity normal to that area. At 2:00 P.M. on June 27 the sun's intensity as read from Fig. 38, for south and west windows was 63 and 144 B.t.u. per sq. ft. per hr., respectively. The heat transmitted by the windows shaded by awnings was therefore obtained by using 28 per cent of these values and multiplying by the respective glass areas for south and west windows.

The total radiation that would have been received by unshaded windows in the Research Residence for each hour of the day for days included in the months of July, August, and September is shown by the curves in Fig. 39. At any given hour the amount of heat actually transmitted by the windows, which were all shaded by awnings, would be 28 per cent of the value read from these curves.
Sample calculation for 2:00 P.M., June 27, 1934:
South windows, $H_s = 85.6 \times 63 \times 0.28 = 1510$ B.t.u. per hr.
West windows, $H_w = 39.9 \times 144 \times 0.28 = 1610$ B.t.u. per hr.
Total = 3120 B.t.u. per hr.

6. Item (5); Heat Transmission Through Ceilings.—The heat transmission through ceilings was calculated from Equation (1), Section 2. The temperature in the attic space at any given time was obtained from test data. The attic space contained a floored section having an area of 395 sq. ft., for which a coefficient of 0.24 was used, (Guide 1934, p. 87, Table 10); an unfloored section having an area of 304 sq. ft., for which a coefficient of 0.62 was used, (Guide 1934, p. 87, Table 10); and a section of 182 sq. ft. which had $\frac{3}{4}$ in. of blanket insulation nailed to the top of the joists. The calculated value of the coefficient for the latter was 0.19.

Sample calculation for 2:00 P.M., June 27, 1934:
Floored ceiling, $H = 394 \times 0.24(104.7 - 79.9) = 2352$ B.t.u. per hr.
Unfloored ceiling, $H = 304 \times 0.62(104.7 - 79.9) = 4675$ B.t.u. per hr.
Insulated ceiling, $H = 182 \times 0.19(104.7 - 79.9) = 858$ B.t.u. per hr.
Total = 7885 B.t.u. per hr.

7. Item (6); Heat Transmission Through Floor.—The heat transmission from the first story to the basement through the floor was calculated from Equation (1), Section 2. The total area of the floor was 850 sq. ft., and the coefficient used was 0.34, (Guide 1934, p. 87, Table 13, Floor No. 1D). Since the temperature in the basement was less than that on the first story the heat flow was from the first story to the basement. Hence the heat gain in this case was negative.

Sample calculation for 2:00 P.M., June 27, 1934:
$H = 850 \times 0.34(77.3 - 79.4) = -607$ B.t.u. per hr.

8. Item (7); Heat Brought in by Infiltration or Ventilation Air.—Since approximately one air change per hour of air for ventilation was delivered into the house, it was considered that the amount resulting from infiltration was negligible. The heat gain was therefore calculated from the weight of air delivered and the difference in enthalpies for indoor and outdoor air. These enthalpies were determined by the respective wet-bulb temperatures from Table 2, pp. 6-9, Guide 1934. For a wet-bulb temperature of 76.5 deg. F. outdoors the enthalpy was 39.20 B.t.u. per lb., and for a wet-bulb temperature of 65.2 deg. F. indoors it was 29.80 B.t.u. per lb. The weight of air delivered by the ventilating fan, as determined by Pitot tube readings was 1015 lb. per hr.
Sample calculation for 2:00 P.M., June 27, 1934:

\[ H = 1015 (39.20 - 29.80) = 9540 \text{ B.t.u. per hr.} \]

9. Item (8); Heat Given Off by Occupants.—The work of F. C. Houghten* and others indicates that the heat loss from a sedentary individual is approximately 400 B.t.u. per hr. The average occupancy at the Research Residence was equivalent to approximately 4 people. Hence the heat given off by occupants was \( 4 \times 400 = 1600 \) B.t.u. per hr.

10. Item (9); Heat Given Off by Electric Lights.—At 2:00 P.M. on June 27, 1934 no lights were in use.

11. Item (10); Heat Given Off by Electric Motors.—It was considered that approximately 80 per cent of the heat equivalent of the electrical input to the fan motor was absorbed by the air delivered through the system. This electrical input was 340 watts. Hence the heat given off by the motor was,

\[ 340 \times 0.80 \times 3.415 = 929 \text{ B.t.u. per hr.} \]

12. Summary.—The summation of all of the heat gains given in items (1) to (10) inclusive, taking into account the negative value for item (6), gave a total of 36,846 B.t.u. per hour for the calculated cooling load at 2:00 P.M., June 27, 1934.

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