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RESULTS OF COOLING RESEARCH IN THE I=B=R RESEARCH HOME

By
WARREN S. HARRIS

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Warren S. Harris

FOREWORD

Summary

Research on chilled water residential cooling systems was started at the University of Illinois in the summer of 1953 under the terms of the co-operative research agreement between the Institute of Boiler and Radiator Manufacturers and the University Engineering Experiment Station. The investigation has as its long range objective the determination of the most practical means of providing summer comfort in homes having steam or hot water heating systems. To date the tests have been more or less of an exploratory nature aimed at revealing the general operating characteristics of possible systems and their potentialities as far as development into efficient and economically feasible residential cooling systems are concerned.

During the summers of 1953 and 1954 the I=B=R Research Home was cooled using combination heating and cooling units in each room. The same piping system was used to circulate hot water from the boiler to the units for winter operation and chilled water from the water-cooled water chiller to these same units during summer operation. The room units used during the first summer were of larger capacity than those used the second summer, but both were the same as far as principal of operation was concerned.

The cooling equipment used in the Research Home during the summer of 1955 was completely separate from the heating system. Two chilled water, fan-coil units were used, one serving each story of the house. These units were connected to an air-cooled water chiller by a simple piping system independent of that for the heating system.

EQUIPMENT

I=B=R Research Home

The Research Home (Fig. 1) was a two-story building, typical of the small well-built American home. The construction was brick veneer on wood frame, and all of the outside walls and the second story ceiling were insulated with mineral wool batts 3-5/8 in. thick. A vapor barrier was placed between the studs and the plaster base to retard the passage of water vapor from the rooms into the insulation during the winter months. The calculated coefficient of heat transmission, U, for the wall section was 0.074 Btu per sq ft (hr)(F). All windows and outside doors were weatherstripped. The windows were double-hung wood sash with the exception of wood casement windows in the kitchen.

The floor plans of the Research Home are shown in Fig. 2, and a summary of the estimated maximum cooling loads is given in Table 1. An outdoor temperature of 96 F and an indoor temperature of 75 F were selected as design conditions. Sensible heat gains through the walls and roof were esti-

mated using the equivalent temperature differential method developed by Stewart and described in the "Cooling Load" chapter of the 1953 ASHAE Guide. The heat gains through the windows were also estimated using the method recommended in the Guide. Infiltration loads were estimated by using the air change method and assuming the same number of air changes for summer operation as is recommended in the ASHAE Guide for winter heat loss calculations.

In estimating the design cooling load no allowances were made for such internal loads as lights and occupancy; however, during the tests the house was occupied by an average of four people during the day and two at night. There was a normal usage of lights, but there was no cooking in the house during the testing season.

Cooling Equipment - 1953 and 1954

The cooling systems used in 1953 and 1954 were designed for use in conjunction with a hot water heating system. Schematic diagrams of the systems, consisting of a water chiller and room units resembling convectors through which either chilled or heated water could be circulated, are shown in Fig. 3a and 3b. In these systems the same piping system was used for both winter and summer operation. The water circulation rates for both summer and winter were considered when sizing the piping system. It was found that the piping system sized for winter heating was adequate for summer cooling as well. All pipe carrying chilled water was insulated with a vapor proof insulation to prevent sweating.

Cross sections through the combination heating-cooling rooms units are shown in Fig. 4. All of the Type A units (used in 1953) with the exception of the one in the dining room were of the same size. The next larger unit was used in the dining room.

The Type B units (used during the summer of 1954) were small in physical dimensions and cooling capacity. These units measured approximately 14 in. x 18 in. x 5 in. One unit was located in each major room of the house.

A second style of room unit (Unit C) was located in each of the first story rooms during the summer of 1954. These units were very similar to Unit B except for the direction of air flow. They were used in only a few special tests as reported in the discussion of results.

All room units were selected from those commercially available at the time. The actual cooling capacity of each system tested is given in the discussion of the test results. The locations of the different units as installed in the Research Home are shown in Fig. 2a.

The water chiller was rated at two tons capacity. The compressor was directly connected to a 2 hp, 220 volt, single phase motor. Both the motor and the condenser were water-cooled. The cooling water was not re-used.

Two 22-in. fans, each rated at 3100 cfm (free delivery) were installed in the attic of the Research Home and used during a part of the summer of 1954. Both fans could be used to draw air through the house and exhaust it through the attic at night, while one of the fans was so mounted that it could be used to draw outdoor air through the attic during the day so as to reduce the build-up in attic temperature during the heat of the day. The arrangement of the attic fans is shown in Fig. 5.

Cooling Equipment - 1955

The cooling equipment used in the Research Home during the summer of 1955 was completely separate from the heating system. Two chilled water, fan-coil units were used, one serving each story of the house. These units were connected to an air-cooled water chiller located in the garage by a simple two-pipe system independent of that for the heating system.

It was possible to locate the fan-coil units so that it was not necessary to use sheet metal duct work. The units were enclosed by either a drop ceiling effect or a fiberboard box, and these enclosures served as the air distributing system. High sidewall registers were cut into the sides of the enclosures to supply cooled air to each room and to serve as return grilles. All registers were located on inside walls near the ceiling. The general arrangement of test equipment used during the summer of 1955 is shown in Figs. 2b and 3c.

Controls

The fans on the combination heating-cooling units and the fan-coils were manually controlled and were operated continuously except for special tests in which the windows were open at night, in which case the room unit fans were operated only during the period that the windows were closed.

The operation of the circulating pump and the compressor motor were controlled by a room thermostat located on an inside wall of the living room 30 in. above the floor. The location of the thermostat is indicated in Fig. 2. As the air temperature in the living room increased above the temperature setting of the thermostat, the thermostat started both the circulating pump and the compressor, and both continued to operate until the air temperature in the living room dropped below the temperature setting of the thermostat. The operating differential of the thermostat was approximately 2 F. The chiller was protected by a limit control which would stop the compressor motor at any time the water temperature in the chiller dropped below approximately 39 F.

Testing Apparatus in Research Home

Approximately 100 copper-constantan thermocouples made of No. 22 B and S gage wire were permanently installed in the walls and ceilings in order to measure temperatures at important points in the structure under various operating conditions. At each of eight locations, one on each exposure of each story, nine thermocouples were installed to provide the data for complete temperature gradients through the walls. Another group of approximately 50 thermocouples provided for the measurement of air temperatures at various levels in the center of each room, in the attic, and in the basement. A third group made it possible to study the performance of the component parts of the cooling system. Provision was made for measuring the temperature of the water entering and leaving each room unit as well as the temperature of the water entering and leaving the chiller.

A central switchboard was located in the basement, and all thermocouples were connected to selector switches on this board. In this way the

electromotive force of each thermocouple could be read quickly on a precision potentiometer used in connection with a highly sensitive galvanometer. A 10-point recording potentiometer used in connection with an auxiliary switchboard made it possible to obtain either instantaneous readings or continuous printed records of the emf produced by the thermocouples in any selected group.

Provision for measuring the rate of flow of water through the chiller used during 1953 and 1954 was made by installing an elbow meter(1) in the return main. It was used in connection with a sensitive differential pressure recorder. This type of meter introduces no additional resistance to the flow of water. The pressure drop through each room unit and connecting piping used in 1953 was measured by a mercury manometer. Prior to the testing season, the relationship between the pressure loss and the rate of water flow was determined for each of these room units so that the pressure losses could be used to measure the respective flows during a test. In a similar manner the pressure drop through the fan-coil units was used to determine water flow rates through the system in 1955. There was no method of measuring the water flow through the individual room units used during the summer of 1954.

Recording thermometers were used to make continuous records of the temperatures of the air in each of the six rooms. The moisture content of the air was measured by means of four humidity indicators, one recording hygrometer, and one wet- and dry-bulb recorder, which were checked periodically with an aspirated psychrometer. The electrical inputs to circulator and compressor motors were measured by means of integrating watt-hour meters having scale divisions of 10 watt-hours. Self-starting electric clocks were wired into the compressor and circulator motor circuits in such a way as to indicate the total time of operation.

Methods and Observations

In all tests the thermostat was set to maintain an average indoor temperature of 75 F. Except for some special tests, each test was 24 hours in length. In addition to those conditions continuously recorded by recording instruments, it was customary to read all room-air temperatures at levels of 3 in., 30 in., and 60 in. above the floor and 3 in. below the ceiling, the relative humidity in each room, and the quantity of water removed from the room air by each room unit two to four times during each test. The power consumption of the compressor and circulating pump motors was recorded for each test period. The temperature of the water entering and leaving the chiller and each of the room units as well as the respective flow rates were also taken during at least one cycle of operation for each of the test arrangements. Procedures used during special tests are described along with the discussion of the results of these tests.

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(1) "The Use of an Elbow in a Pipe Line for Determining the Rate of Flow in the Pipe", Univ. of Ill. Eng. Exp. Sta. Bull. No. 289, 1936.

WEATHER

Cooling and Heating Seasons

The American Society of Heating and Air Conditioning Engineers publishes a chart commonly known as the Comfort Chart. This chart shows the relationship between dry-bulb temperature, relative humidity, and the feeling of warmth. It was determined experimentally using a large number of subjects. Better than 95 percent of the subjects reported maximum summer comfort at a condition represented by a dry-bulb temperature of 75 F and a relative humidity of 60 percent. Approximately 70 percent of the subjects reported comfort at a dry-bulb temperature of 80 F and a relative humidity of 50 percent, while 50 percent of the subjects reported comfort at a dry-bulb temperature of 80 F and a relative humidity of 60 percent. From this it would appear that the majority of persons will be comfortable during the summer if the dry-bulb temperature in the home does not exceed 80 F and the humidity is 60 percent or less.

During the summers of 1941 and 1942 a series of tests was made in the I=B=R Research Home to determine the relationship between maximum indoor and maximum outdoor temperature when using only natural ventilation through open windows to cool the house at night. It was found that if the windows were opened at 10:00 p.m. and remained open until about 7:00 a.m. and were kept closed the remainder of the time that the maximum indoor temperature was approximately 10 F lower than the maximum outdoor temperature. On this basis the only days during which mechanical refrigeration would be required to maintain comfortable dry-bulb temperatures would be those for which the maximum outdoor temperature was 90 F or more.

Figure 6 is a plot of the daily maximum and minimum temperatures at Urbana, Illinois, for the years 1954 and 1955. The figure shows that as the maximum outdoor temperature increases the difference between the maximum and minimum temperature also increases. For example, when the maximum outdoor temperature is 20 F the average minimum temperature is about 3 F. On the other hand, when the maximum temperature is 90 F the minimum is about 64 F. Even though the maximum outdoor temperature may be 100 F or more, the minimum temperature in Urbana seldom exceeds 75 F.

A similar analysis shows this same trend for most midwestern cities. For days having a maximum outdoor temperature of 100 F, the average minimum temperature in Austin, Texas, is about 75 F, while for Minneapolis, Minnesota, it is about 77 F, and in Bismark, North Dakota, it is about 70 F. The difference between the maximum and minimum daily temperatures in coastal areas is much less. In New York City the average minimum daily temperature is 82 F when the maximum is 100 F. These observations suggest the possibility of cooling by night time ventilation in certain areas to secure at least part of the total cooling effect required and thus reduce the operating cost.

Table 2 is a summary of a 53-year record of temperatures at Urbana. It shows the relative lengths of the summer cooling season as compared to the winter heating season. Somewhat similar data for several other cities are presented in Table 3.

In the northern part of the United States there may be 10 to 30 days per year in which the maximum temperature exceeds 90 F, while there may be 200 to 250 days per year in which the maximum outdoor temperature is 65 F or less. On the other hand, these figures are almost reversed in certain sections of the south. The data in these tables point out the fact that there is a vast difference in the relative needs of heating and cooling in the different parts of the country, and, therefore, different solutions to the problem of providing year around comfort may be required.

COOLING LOADS

Comparison of Measured and Calculated Cooling Loads

Table 4 shows a comparison of calculated and observed cooling loads for a day when the maximum outdoor temperature was 100 F. Calculating the cooling loads by the procedure outlined in Chapter 13 of the 1953 ASHAE *Guide* indicated that the maximum load should occur about 1:00 p.m., DST. The observed maximum load did not occur until about 1:30 p.m. and continued until about 4:00 p.m. The measured maximum sensible cooling load for the house was only 12,288 Btuh as compared to a calculated sensible load of 14,763 Btuh.

It should be noted that the calculated heat gains for rooms having southern exposures were high as compared to the actual measured cooling load, while for the rooms having northern exposures the calculated loads were lower than those actually observed. It is true that when operating with all room doors open there is air movement from room to room, and hence there is some transfer of load from one room to another. The measured loads as reported in Table 4 were obtained after the water flow to each of the room units had been adjusted to give the best possible balance of room-air temperatures. There was only 3 F difference in temperature between the warmest and the coolest rooms in the house, and furthermore the rooms with south exposures were the warmer rooms. Thus, it is probable that any transfer of load from one room to another was from the rooms on the south side to those on the north side. This would make the differences between measured and calculated loads even greater than that indicated by Table 4.

The magnitude of the differences between measured and calculated heat gains were sufficiently large that they could not be attributed to errors in the estimation of wall and ceiling gains alone. In fact, the data seem to indicate that most of the discrepancy between calculated and measured loads must be in the estimated heat gains through the glass areas. When estimating cooling loads it is common practice to make the assumption that all radiant energy transmitted through glass is immediately available to heat the air in the rooms. Actually this energy is not transformed to heat until it strikes some solid object. The object is first warmed, and then the room air is warmed by convection. Since appreciable time is required for these processes to take place, there is a finite time lag between the time the radiant energy is transmitted through the windows and the time it actually warms the room air. Furthermore, the convective heat transfer rate between the objects warmed by radiation and the air in the room is not necessarily as high as the rate at which solar radiation is received by objects in the room. Ignoring these facts would tend to make the maximum estimated instantaneous load occur earlier and have a larger magnitude than

the actual instantaneous loads on the cooling equipment. Methods of estimating solar heat gains through glass areas which determine the instantaneous transmittance of solar energy through the glass do not necessarily reflect the rate at which the solar energy eventually warms the air in the room. The latter affects the load on the cooling equipment.

EFFECTS OF EQUIPMENT DESIGN ON OPERATION

Effects of Number of Units in Use and Equipment Capacity on Cooling Performance

The principal operating characteristics of the different systems and combinations tested during the three summers of 1953, 1954 and 1955 are summarized in Table 5. Each test in this table represents a day having an outdoor temperature approximating design conditions.

Test I, Table 5, represents conditions obtained with the original cooling equipment operated in the Research Home during the summer of 1953. All six Type A room units were in operation. The average air temperature at the 30-in. level was 76.3 F. The average temperature of the warmest room was about 4.5 F higher than that of the coolest room, and the change in temperature in each room ranged from a low of 2.0 F in the living room to a high of 4.5 F in the kitchen. The average temperature of the first story rooms was about 3 F lower than for the second story.

The total cooling capacity of this system was high as compared to the actual cooling loads, and as a result the system only operated about 7.5 hours per day. This was made up of a number of short cycles averaging about 40 minutes in length. The average temperature of the water supplied to the units during these short cycles was about 54 F. Due to the short period of operation and the relatively high water temperature only 8.4 lb of water was removed from the room air during the test. As a result, the relative humidity in the house was almost 70 percent, too high for comfort.

The same equipment was used during Test II as during Test I except in Test II all room units except those in the living room and northeast bedroom were turned off. Under the test conditions the total cooling capacity of the system during Test II was about 10,500 Btuh. The unit in the northeast bedroom did not have sufficient capacity to cool the entire second story. As a result, second story temperatures were 2 or 3 F higher than normal, and the average indoor temperature for the test was 78.9 F as compared to approximately 76 F for Test I. The temperature difference between rooms was about the same as for Test I, but the temperature variation in each room was less, ranging from 0.5 F in the southwest bedroom to 2.5 F in the northeast bedroom. Had the unit in the northeast bedroom had sufficient capacity to reduce the average air temperature in this room 1.5 F lower than it did in Test II there is every reason to believe that a corresponding reduction in temperature would have occurred in the other bedrooms. This being true, two units of proper capacity could have cooled all of the Research Home with a maximum difference in air temperature from room to room of about 2 F.

In Test II the system operated 10.4 hours, and the average water temperature was 42.4 F. The somewhat longer operating period, coupled with the lower water temperature, made it possible to remove 24.4 lb of water from

the room air. This resulted in an average indoor relative humidity of just under 60 percent. In general, conditions in the house were more uniform and more comfortable during Test II even though the dry-bulb temperature was slightly higher than in Test I.

The system used in Test III, using six Type B units, also had a very low cooling capacity. The cooling capacities of the room units used in the bedrooms were too low to prevent a slight increase in room-air temperature during the heat of the day as is indicated by the higher than normal maximum air temperatures in these rooms. On the other hand, once the system started operation it operated almost continuously until evening, resulting in very gradual change in room-air temperatures. Persons in the house at the times of the tests felt that, everything considered, comfort was at as high a level in Test II as in any of the tests to date.

Tests V and VI represent conditions obtained with the fan-coil units in the summer of 1955. The combined cooling capacity of the two fan-coil units operating at low fan speed (Test V) was about 16,700 Btuh, again above the actual cooling load, and again the operating time was low, and the relative humidity was too high for comfort.

In order to reduce the cooling capacity and thereby obtain longer operating periods, a rheostat was placed in the fan circuit of the first story fan-coil unit, and the fan speed reduced so that the air flow was about 60 percent of normal (Test VI). Under these conditions of operation the total cooling capacity of the system was about 14,400 Btuh. In Test V both the room-air temperature and relative humidity were at satisfactory levels.

Water temperatures were not recorded during Tests V and VI, but representative temperatures of both air and water entering and leaving the fan-coil units are shown in Tables 6 and 7.

Reserve Capacity

During the summer of 1954, while using a cooling system with a total cooling capacity of only 10,800 Btuh, the hottest weather on record at Urbana was experienced. House and outdoor conditions for this period are shown in Fig. 7. The minimum outdoor temperature for the period was 73 F, while the maximum was 107 F. During the last 24 hours of the period the minimum outdoor temperature was 76 F, and the temperature exceeded 100 F for a period of 6.5 hr. At 7:00 a.m. July 12th the thermostat set the cooling system in operation, and the system operated continuously for a period of more than 72 hours. Even though the daily maximum temperatures during this period were all well above the design temperature, the total overrun in room-air temperature was only about 4 F. On the whole, indoor temperatures and humidities were very uniform, and within the comfort range throughout the entire period indicating that even though the system had a total capacity well under the estimated design load, it was capable of maintaining satisfactory indoor conditions with outdoor temperatures as much as 15 F above the design conditions.

Effect of Air Discharge Location

Two types of small room units were installed in each of the first story rooms. The one designated as Type B was used during most of the tests. Air was discharged from the top of this unit and directed at an angle toward the ceiling. The air from the other unit, designated as Unit C, was discharged at the bottom along the floor. Actual rates of air circulation of Units B and C were essentially the same.

Table 8 shows a comparison of living room air temperatures obtained with Units A, B and C. The maximum-minimum air temperature differences obtained at each level with Units A and B were much the same. At the floor level, the difference obtained with Unit C was higher than for either Unit A or B. Unit C also produced the greatest temperature difference between the 3-in. and 30-in. levels. These observations indicate the desirability of projecting the cooled air upward so as to prevent stratification of cool air along the floor.

Night Air Cooling - Open Windows Only

The curves in Fig. 8 show the indoor and outdoor conditions for two 24-hr test periods during the summer of 1953. These two test periods were almost exact duplicates as far as outdoor temperature and humidity were concerned. During one of the test periods the windows were closed all of the time and the cooling system was controlled by the thermostat. During the other all windows were opened from 10:00 p.m. to 7:00 a.m. The cooling unit was not operated during the time the windows were open.

While the outdoor temperature was 95 F from noon to about 5:00 p.m., it was 75 F or less from 9:00 p.m. to 8:30 a.m., reaching a low of approximately 67 F between 5:00 and 7:00 a.m. With a maximum temperature of 95 F minimum temperatures of 65 to 70 F are typical of Urbana weather (see Fig. 6).

Figure 8 shows that during the night, both the indoor dry-bulb temperature and humidity ratio were lower for the test period in which the windows were opened than for the test made with closed windows. During the rest of the day the indoor conditions were essentially the same for both tests.

From the curves at the bottom of Fig. 8, it can be seen that the measured loads were well under the calculated load during the heat of the day. For the test made with closed windows, the measured load exceeded the calculated load after about 8:00 p.m. Because of heat storage in the house and furnishings there was some cooling load all night long during the test with closed windows even though the outdoor temperature was well below the desired indoor temperature. On the other hand, there was no need of cooling in the house from 10:00 p.m. until 1:30 the following afternoon when operating with open windows at night.

Opening the windows at night had little or no effect on the maximum cooling load, and hence on the capacity of cooling equipment required; it merely shortened the length of time the cooling system had to operate in order to maintain comfortable indoor temperatures. In the section of this

report in which costs are discussed, it is shown that by making use of night air cooling, operating costs were reduced by almost 50 percent.

Night Air Cooling with Attic Fan

Tests were made during the summer of 1954 to determine the effectiveness of an attic fan in reducing the load on the cooling system and in producing comfortable bedroom air temperatures at night be drawing cool night air through the house. The results of the tests using the attic fan were compared with results obtained the preceeding summer using open windows at night but without the aid of mechanical ventilation. In all tests discussed in this section, windows were open from 10:00 p.m. to 7:00 a.m. During tests in which attic fans were used, the fans were in operation all the time that the windows were open.

When the outdoor temperature was 71 F or less at 10:00 p.m. and with no attic fan operation, it was found that the second story room-air temperatures throughout the night were 4 to 5 F warmer than the first story air temperatures. With the attic fan in operation there was no change in first story air temperatures, but the second story temperatures were reduced until they were essentially the same as those on the first story. Upon starting the attic fan, the indoor air temperature dropped to within 2 F of the outdoor air temperature within one hour and maintained this difference throughout the remainder of the night. With open windows alone, 6 to 8 hours were required to accomplish this same reduction in indoor air temperature in the first story rooms while the minimum air temperature in the second story rooms was always about 6 F warmer than the outdoor temperature.

If the outdoor temperature was above 71 F to 75 F at 10:00 p.m. neither open windows nor the use of an attic fan were effective in reducing room-air temperature. The air motion produced by the attic fan did produce some "cooling effect" even though there was little or no change in the dry-bulb temperature of the air. The use of an attic fan at night did not reduce the cooling load during the following day any more than did the open windows by themselves since the house temperatures at 7:00 a.m. were about the same with both methods of operation.

Attic Ventilation

In Test IV of Table 5 an attic fan was used to draw outdoor air through the attic all day and thus reduce the build-up of attic temperature during the heat of the day. This reduced the load in the bedrooms enough that the capacity of the Type B room units was sufficient to prevent the overrun in room temperature noted in Test III. The maximum air temperature in the bedroom in Test IV was about 78 F as compared to 80 F in Test III. The daily operating cost was increased 11 cents, the cost of operating the attic fan.

Ventilation Air

To demonstrate the effect of ventilation air on the cooling load a few tests were made in which the cooling system was operated with closed windows at all times, and air was exhausted by means of an exhaust fan in the east

window of the kitchen and operated continuously during the test period. No special provision was made for the entrance of outdoor air into the house. The measured free air delivery of the exhaust fan was 430 cfm.

While too few tests were made to accurately determine the effects of ventilation, it appeared that the ventilation air increased the maximum load by approximately 30 percent and increased the total daily load by about 8 to 10 percent. At no time did those working or sleeping in the house notice that the operation of the exhaust fan appreciably increased the quality or "freshness" of the air in the house. By consensus there seemed to be no great need of mechanical ventilation, although it is probable that in homes some ventilation might be required during periods when food is being cooked or at times of unusual occupancy or heavy smoking.

WINTER OPERATION

Winter Heating

While this is a paper on the performance of residential cooling, winter heating cannot be ignored completely as some of the systems tested were designed to perform both the summer and winter operations. Figure 9 is a graphical representation of air temperature variations obtained in the living room with (a) baseboard, (b) Type A combination room units, and (c) Type B combination room units. Unit fans were operated continuously for winter operation. The combination room units caused a much wider fluctuation in room-air temperature than did the baseboard heating system. The air temperature changes and the air motion produced by the combination room units were noticeable. Ways to correct this condition must be found before these units can be considered wholly satisfactory.

On the other hand, the chilled water, fan-coil cooling system used in the I=B=R Research Home produced satisfactory room-air temperature and humidity during the hot weather and did so with no compromise as far as winter heating performance was concerned.

COSTS

Operating Costs

The operating costs are determined by the quantity of cooling water used and the power consumption of the compressor, pump and fan motors. Representative operating costs for all systems and methods of operation tried in the Research Home are given in Table 9. In each case the maximum outdoor temperature was approximately 95 F. Tests I through VI in Table 9 are for the same tests as the corresponding tests in Table 5.

Tests I and II show that reducing the number of Type A room units in operation from six to two reduced the daily operating cost from $1.14 to $0.82. The operating cost when using six Type B units, Test III, was $1.33. The power used by the fans in the Type B units was less than that used by the fans in the Type A units, but the power consumption of the pump and compressor motors was increased, probably due to an increase in the total daily cooling load resulting from the greater amount of water removed from the room air by the Type B units.

Comparing Tests III and IV it is found that operating the attic fan to ventilate the attic during the heat of the day increased the cost of operation to $1.42, an increase of about 10 cents per day. Practically all of the increase was represented by the power consumption of the attic fans.

Operating cost data are not available for the tests in which a kitchen exhaust fan was used for ventilation purposes, but it is known that the increase in load due to ventilation was about 10 percent, and it is reasonable to expect a corresponding increase in the cost of operating the compressor. The cost of operating the ventilating fan was about 3 cents per day. On this basis, the cost of operation when using two Type A units and ventilating by the use of a kitchen fan should be approximately $0.90 per day.

Comparing Tests VII and VIII it is observed that opening the windows from 10:00 p.m. to 7:00 a.m. reduced the cost of operation from about $0.87 to approximately $0.45 per day.

All the foregoing cost figures are only approximate as the shortness of the testing seasons permitted the running of only a few tests for each operating condition, and this was not sufficient to average out such effects as day-to-day variations in wind, sun intensity, and outdoor humidity.

During the summer of 1955 sufficient data were taken to establish the relationship between cost of operating the fan-coil cooling system and the outdoor temperature. This relationship is shown in Fig. 10.

At an average outdoor temperature of 83 F (maximum outdoor temperature about 93 F) the cost of operation was about $0.95 per day. For the same outdoor temperature the cost of operating the system with a water-cooled condensing unit used in 1954 was about $1.33. Comparing Tests III and VI, Table 9, it may be observed that the air-cooled condensing unit used a little more power than did the water-cooled unit; the power consumed by the pump and the unit fans of the system used in 1955 (air-cooled condenser) was less than that consumed by the pump and fans used in 1954 (water-cooled condenser).

Had the air-cooled condenser been used on last year's system, the total power consumption would have been increased by about 5.3 kw hr and the water consumption reduced to zero. On that basis the operation cost of the system used in 1954 would have been about $1.10 per day at an average outdoor temperature of 83 F instead of $1.33.

Since the cost of operation is dependent on the unit cost of the utilities used as well as the quantity consumed, this comparison of operating costs may not apply in localities having power and water rates which differ from those in Urbana, Illinois.

Installation Cost

One might expect the installation cost of a system such as was used during the summer of 1955 to be high as compared to systems in which much of the equipment is used for both cooling in summer and heating in winter. Table 10 is a breakdown of installation costs for a one-pipe, baseboard heating system, a heating -cooling system using small combination room units in each room, and a heating-cooling system consisting of a baseboard heating system, fan-coil units for cooling and separate piping for the

heating and the cooling systems. These cost estimates are for systems installed in the I=B=R Research Home. It was assumed that an air-cooled water chiller was used on both cooling systems to place the cost estimates on a comparable basis. There was no appreciable difference in the installation cost of the systems used for the two methods of heating and cooling the Research Home. The total increase in the cost of the heating and cooling system as compared to a one-pipe baseboard system appeared to be about $1400. Most of this increase occurred in three items: the chiller, the room units, and labor. It is apparent that if total installed cost is to be reduced, an effort must be made to reduce the cost of these three items.

SUMMARY AND CONCLUSIONS

Tests made in the I=B=R Research Home indicate that:

1. With a maximum outdoor temperature of 100 F, the measured maximum sensible cooling load was about 17 percent less than the maximum sensible load determined by conventional methods of estimating cooling loads.

2. Most of the differences between the estimated and observed maximum cooling loads could be attributed to the method of estimating heat gains through glass areas.

3. In estimating cooling loads it is assumed that all radiant energy transmitted through glass is immediately available to heat the air in the room. Actually it first heats objects in the room, and they in turn heat the room air by convection. This results in a time lag and a reduced rate of heat release to the air.

4. As long as room doors are not closed, a satisfactory cooling job could be done in the entire Research Home by operating cooling units in only two or three rooms.

5. With the type of equipment used in these tests, the installation of room units having a cooling capacity in excess of the maximum required decreases the quality of performance from the standpoint of comfort and frequently increases the cost of operation.

6. For satisfactory results the capacity of the water chiller must be sufficient to quickly reduce the temperature of the water in the system to the minimum value.

7. A cooling system having a total cooling capacity of only 10,800 Btuh successfully maintained indoor temperatures in the Research Home below 80 F with an outdoor temperature in excess of 105 F even though the calculated heat gain of the Research Home is 17,962 Btuh with an outdoor temperature of 95 F.

8. Chilled air from the room units should be projected upward toward the ceiling to prevent stratification and the resulting "pool" of cool air at the floor with little cooling at the higher levels.

9. Opening windows at night had no effect on the maximum cooling load, but except for a very few days in Urbana, this practice results in equal or better comfort conditions in the house than obtained by operating with the windows closed at all times.

10. Opening windows at night reduced the cost of operation by as much as 50 percent.

11. Ventilating the attic during the day did not reduce the load sufficiently to affect the operating cost, but it did reduce the maximum second story air temperatures by 3 to 4 F.

12. Opening windows at night and using an attic fan to draw outdoor air through the house had the same effect on operating costs as did opening windows at night with no use of an attic fan.

13. Using an attic fan at night reduced the room-air temperatures to within about 2 F of the outdoor temperature within one hour. About six hours were required by natural ventilation through open windows. Air movement produced by the fan created some additional "cooling effect".

14. Operating a kitchen exhaust fan having a free air delivery of 430 cfm increased the maximum cooling load by about 25 percent and increased the daily cost of operation by about 10 percent.

15. From the standpoint of comfort there appeared to be no need of mechanical ventilation for the test conditions. Some such ventilation might be required during periods when foods are being cooked or at times of unusual occupancy or heavy smoking.

16. The room units designed for both summer and winter operation maintained satisfactory room-air temperature and humidity during the summer, but winter performance was not as satisfactory as the performance of a hot water baseboard heating system.

17. The summer performance of the fan-coil system was comparable to that of the combination room units. The fan-coil cooling system did not affect the performance of the heating system used.

TABLE 1

ESTIMATED COOLING LOADS, I=B=R RESEARCH HOME

Room	Design Sensible Cooling Load, Btuh
Kitchen	1,942
Dining Room	5,240
Living Room	1,138
Northeast Bedroom	1,207
Northwest Bedroom	1,383
Southwest Bedroom	2,562
Total Sensible Load	13,472
Latent Load Allowance	4,490
Total Estimated Design Load	17,962

Estimated design loads based on a maximum outdoor temperature of 95 F and indoor temperature of 75 F.

Design loads estimated by the equivalent temperature method given in the 1953 ASHAE *Guide*.

TABLE 2

OUTDOOR TEMPERATURES - URBANA, ILLINOIS
(U. S. Weather Bureau Records from 1901 through 1954)

	Average Number of Days		
	Maximum Outdoor Temperature 90 F and Above	Minimum Outdoor Temperature 32 F and Below	Degree Days
First Quarter	0	69	2952
Second Quarter	6	6	637
Third Quarter	20	0	92
Fourth Quarter	0	38	2125
Total for Year	26	113	5806
Record High for Year	56		
Record Low for Year	4		

TABLE 3

WEATHER DATA

City	Average Number of Days per Year with Maximum Temperature			
	90 F and above	100 F and above	65 F and below	30 F and below
Bismark, North Dakota	28.2	5.6	223.2	90.6
Newark, New Jersey	10.2	0.4	207.6	22.8
St. Paul, Minnesota	17.0	1.6	227.2	87.6
San Antonio, Texas	107.0	5.0	67.4	0.2

TABLE 4

COMPARISON OF CALCULATED AND OBSERVED COOLING LOADS

Maximum Outdoor Dry-Bulb Temperature = 100 F
Average Indoor Dry-Bulb Temperature = 75 F
Six Type A Room Units in Operation
September 1, 1953

	Calc. Loads at 1:00 PM (Max. Total Load) Btuh						Observed Max. Load (1:30 to 4:00 PM, CST)			Calc. Sens. Load Minus Observed Total Load	Observed Room-Air Temp. at 30" Level	
	Walls	Ceil.	Glass	Infil.	Total Walls and Ceil.	Sens. Total	Sens.	Latent	Total		Max.	Min.
Kitchen	241	-	1,735	131	241	2,107	1,718	0	1,718	389	77	74
Din Rm	173	-	4,925	604	173	5,702	2.360	27	2,387	3,315	77	73
Liv Rm	276	14	990	277	290	1,557	2,134	171	2,305	-748	76	74
Total 1st Story	690	14	7,650	1,012	704	9,366	6,212	198	6,410	2,956		
N.E. Bed	245	159	908	140	404	1,452	1,741	74	1,815	-363	75	72
N.W. Bed	131	222	890	140	353	1,383	2,193	56	2,249	-866	74	71
S.W. Bed	240	220	1,962	140	460	2,562	2,142	59	2,201	361	76	73
Total 2nd Story	616	601	3,760	420	1,217	5,397	6,076	189	6,265	-868		
House Total	1,306	615	11,410	1,432	1,921	14,763	12,288	387	12,675	2,088		

TABLE 5

COOLING SYSTEM PERFORMANCE

TEST I, July 28, 1953, Six Type "A" Combination Units

Maximum Outdoor Temperature........................92.5 F
Average Outdoor Temperature........................83.1 F
Cooling Capacity of System........................21,900 Btuh
Average Indoor Temperature........................76.3 F
Average Indoor Relative Humidity........................69.9 %
Total Water Removed from Air........................8.4 lb
Operating Time........................7.52 hr
Average Temperature of Chilled Water Supplied to Units......54.3 F

Room	Kitch	Din Rm	Liv Rm	SW Bed	NW Bed	NE Bed	Max. Diff. Between Rooms
Unit in Operation	Yes	Yes	Yes	Yes	Yes	Yes	
Fan Speed	High	High	Low	High	High	High	
Max. Air Temp., 30" Level, F	75.5	77.0	76.5	79.5	79.5	79.5	4.0
Min. Air Temp., 30" Level, F	71.0	73.5	74.5	76.5	76.0	76.0	5.5
Max. - Min. Air Temp., F	4.5	3.5	2.0	3.0	3.5	3.5	

TEST II, August 3, 1953, Type "A" Combination Units

Maximum Outdoor Temperature........................95.5 F
Average Outdoor Temperature........................84.3 F
Cooling Capacity of System........................10,500 Btuh
Average Indoor Temperature........................78.9 F
Average Indoor Relative Humidity........................59.5 %
Total Water Removed from Air........................24.4 lb
Operating Time........................10.37 hr
Average Temperature of Chilled Water Supplied to Units......42.4 F

Room	Kitch	Din Rm	Liv Rm	SW Bed	NW Bed	NE Bed	Max. Diff. Between Rooms
Unit in Operation	No	No	Yes	No	No	Yes	
Fan Speed	-	-	High	-	-	Low	
Max. Air Temp., 30" Level, F	78.5	78.5	77.0	81.0	81.0	79.0	4.0
Min. Air Temp., 30" Level, F	77.5	77.5	75.0	80.5	80.0	76.5	5.5
Max. - Min. Air Temp., F	1.0	1.0	2.0	0.5	1.0	2.5	

TEST III, June 20, 1954, Type "B" Combination Units

Maximum Outdoor Temperature........................94.0 F
Average Outdoor Temperature........................83.1 F
Cooling Capacity of System........................10,800 Btuh
Average Indoor Temperature........................76.1 F
Average Indoor Relative Humidity........................52.2 %
Total Water Removed from Air........................21.2 lb
Operating Time........................17.5 hr
Average Temperature of Chilled Water Supplied to Units......40.0 F

Room	Kitch	Din Rm	Liv Rm	SW Bed	NW Bed	NE Bed	Max. Diff. Between Rooms
Unit in Operation	Yes	Yes	Yes	Yes	Yes	Yes	
Fan Speed	#6	#6	#6	#6	#6	#6	
Max. Air Temp., 30" Level, F	76.0	77.0	76.5	79.0	81.0	79.0	5.0
Min. Air Temp., 30" Level, F	74.0	74.5	74.0	76.0	76.0	75.5	2.0
Max. - Min. Air Temp., F	2.0	2.5	2.5	3.0	5.0	3.5	

TABLE 5 (continued)

TEST IV, June 29, 1954, Type "B" Combination Units, Mechanical Ventilation of Attic During Day

Maximum Outdoor Temperature	96.0 F
Average Indoor Temperature	75.8 F
Average Indoor Relative Humidity	51.6 %
Total Water Removed from Air	26.2 lb

Room	Kitch	Din Rm	Liv Rm	SW Bed	NW Bed	NE Bed	Max. Diff. Between Rooms
Unit in Operation	Yes	Yes	Yes	Yes	Yes	Yes	
Max. Air Temp., 30" Level, F	76.5	77.5	76.5	78.0	78.0	77.5	1.5
Min. Air Temp., 30" Level, F	74.0	75.5	74.0	75.0	76.0	75.0	2.0
Max. - Min. Air Temp., F	2.5	2.0	1.5	3.0	2.0	1.5	

TEST V, July 4, 1955, Two Fan-Coil Units; CFM: First Story = 280; Second Story = 230

Maximum Outdoor Temperature	95.0 F
Average Outdoor Temperature	80.9 F
Cooling Capacity of System	16,700 Btuh
Average Indoor Temperature	75.7 F
Average Indoor Relative Humidity	66.0 %
Total Water Removed from Air	6.5 lb
Operating Time	9.3 hr

Room	Kitch	Din Rm	Liv Rm	SW Bed	NW Bed	NE Bed	Max. Diff. Between Rooms
Max. Air Temp., 30" Level, F	76.5	76.5	76.5	75.5	-	78.5	3.0
Min. Air Temp., 30" Level, F	73.5	75.5	75.0	74.0	-	76.0	2.5
Max. - Min. Air Temp., F	3.0	1.0	1.5	1.5	-	2.5	

TEST VI, July 28, 1955, Two Fan-Coil Units; CFM: First Story = 155; Second Story = 230

Maximum Outdoor Temperature	95.0 F
Average Outdoor Temperature	83.7 F
Cooling Capacity of System	14,400 Btuh
Average Indoor Temperature	75.7 F
Average Indoor Relative Humidity	57.2 %
Total Water Removed from Air	40.1 lb
Operating Time	17.1 hr

Room	Kitch	Din Rm	Liv Rm	SW Bed	NW Bed	NE Bed	Max. Diff. Between Rooms
Max. Air Temp., 30" Level, F	76.5	77.0	76.5	78.5	78.0	76.5	1.5
Min. Air Temp., 30" Level, F	73.5	74.5	74.0	75.0	75.0	75.0	1.5
Max. - Min. Air Temp., F	3.0	2.5	2.5	3.5	3.0	1.5	

TABLE 6

TYPICAL AIR TEMPERATURES, SUMMER COOLING

		Series A-55	Series B-55
First Story Fan-Coil, In	F	72.5	75.2
First Story Fan-Coil, Out	F	58.0	59.2
Temperature Drop	F	14.5	16.0
Second Story Fan-Coil, In	F	73.7	78.0
Second Story Fan-Coil, Out	F	59.2	63.5
Temperature Drop	F	14.5	14.5

TABLE 7

TYPICAL WATER TEMPERATURES
SUMMER COOLING 1955, FAN-COIL SYSTEM

		Series A-55	Series B-55
Chiller, In	F	46.3	49.2
Chiller, Out	F	37.4	40.6
Temperature Drop	F	8.9	8.6
First Story Fan-Coil, In	F	37.6	41.6
First Story Fan-Coil, Out	F	45.2	48.6
Temperature Rise	F	7.6	7.0
Second Story Fan-Coil, In	F	37.6	41.4
Second Story Fan-Coil, Out	F	45.4	49.3
Temperature Rise	F	7.8	7.9

TABLE 8

LIVING ROOM TEMPERATURE GRADIENTS PRODUCED
BY DIFFERENT COOLING UNITS

Unit Type	Air Discharge	CFM	Level	Temperatures Max.	Min.	Diff.
A*	Top of Unit	167	Ceil.	77.0	75.0	2.0
			30"	76.5	74.0	2.5
			3"	76.5	73.5	3.0
B**	Top of Unit	50	Ceil.	78.0	77.0	1.0
			30"	76.5	74.5	2.0
			3"	77.0	74.0	3.0
C**	Bottom of Unit	54	Ceil.	77.5	75.0	2.5
			30"	76.0	73.5	2.5
			3"	75.0	70.0	5.0

Maximum Outdoor Temperatures = 93.8 ± 0.8
Minimum Outdoor Temperatures = 72.0 ± 2.2
* Used during summer of 1953.
** Used during summer of 1954.

TABLE 9

OPERATING COST OF COOLING SYSTEMS IN THE RESEARCH HOME

Test	I		II		III		IV		V		VI		VII		VIII	
No. of Units Operating	6 Type "A"		2 Type "A"		6 Type "B"		6 Type "B"		2 Fan-Coil*		2 Fan-Coil**		3 Type "A"		3 Type "A"	
Night Air Cooling	No		No		No		No		No		No		Yes		No	
Ventilation Air	No		No		No		No		No		No		Yes		Yes	
Attic Ventilation during Day	No		No		No		Yes		No		No		No		No	
Maximum Outdoor DB	92.5		95.5		94.0		96.0		94.0		91.0		96.0		96.0	
Minimum Outdoor DB	71.5		77.0		73.5		69.0		75.0		75.0		67.0		72.0	
Average Outdoor DB	83.1		84.3		83.1		78.9		83.1		82.8		81.0		82.3	
	KWH	Cost	KWH	Cost	KWH	Cost	KWH	Cost	KWH	Cost	KWH	Cost	KWH	Cost	KWH	Cost
Compressor & Condenser	16.3	$0.407	14.4	$0.360	22.8	$0.571	23.7	$0.593	23.9	$0.598	28.1	$0.707	7.3	$0.182	14.8	$0.370
Pump	1.9	0.048	3.2	0.080	5.3	0.133	5.4	0.135	1.2	0.030	2.0	0.050	1.0	0.025	1.9	0.046
Unit Fans	15.3	0.381	3.9	0.099	9.1	0.227	9.1	0.227	6.0	0.150	4.6	0.116	3.1	0.078	5.3	0.137
Attic Fans	-	-	-	-	-	-	3.2	0.081	-	-	-	-	-	-	-	-
Kitchen Ventilating Fan	-	-	-	-	-	-	-	-	-	-	-	-	0.6	0.015	1.1	0.027
Total Power	33.5	0.836	21.5	0.539	37.2	0.931	41.4	1.036	31.1	0.778	34.7	0.873	12.0	0.300	23.1	0.577
	Cu ft		Cu ft		Cu ft		Cu ft		Cu ft		Cu ft		Cu ft		Cu ft	
Cooling Water	83.0	0.303	77.3	0.284	102.8	0.395	105.4	0.385	-	-	-	-	39.1	0.146	79.5	0.292
Total Operating Cost		$1.139		$0.823		$1.326		$1.421		$0.778		$0.873		$0.446		$0.867
Power Cost in Percent of Total		73.2		66.5		71.0		73.0		100		100		67.3		66.5

* CFM First Story Fan-Coil = 280; Second Story Fan-Coil = 230
** CFM First Story Fan-Coil = 155; Second Story Fan-Coil = 230

TABLE 10

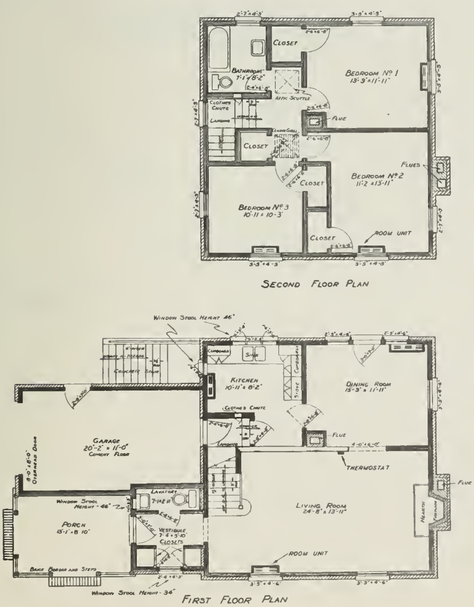
INSTALLATION COSTS

	One-Pipe Baseboard (Heating Only)	Heating and Cooling: Combination Room Units 1954	Heating and Cooling: Fan-Coil Units and Baseboard System 1955
Pipe and Fittings	$104.96	$ 140.77	$ 148.19
Pipe Covering	0.00	62.94	37.45
Labor (piping)	190.17	346.33	320.37
Boiler and Burner	255.50	255.50	255.50
Chiller (air-cooled)	0.00	700.00[a]	700.00[a]
Room Units	196.58	562.42	496.58[b]
Pump and Controls	63.82	127.64	127.64
Electrical Outlets (material and labor)	0.00	45.00[c]	20.00[c]
Fan-Coil Enclosures (material and labor)	0.00	0.00	139.59
Total	$811.03	$2,240.60	$2,245.32

(a) Average of trade prices ranging from $650 to $760.
(b) Average of trade prices for fan-coil units ranging from $102 to $191.
(c) Does not include cost of 220 volt power supply to chiller.

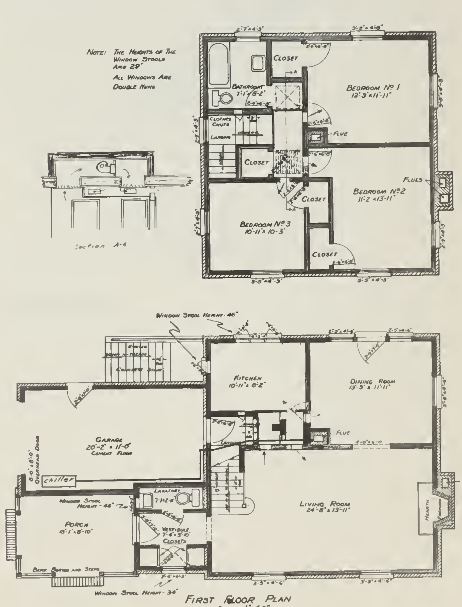
FIGURE 1. I=B=R. RESEARCH HOME.

FIGURE 2A.



"LOCATION OF ROOM UNITS"

FIGURE 2B.



"LOCATION OF FAN COIL UNITS"

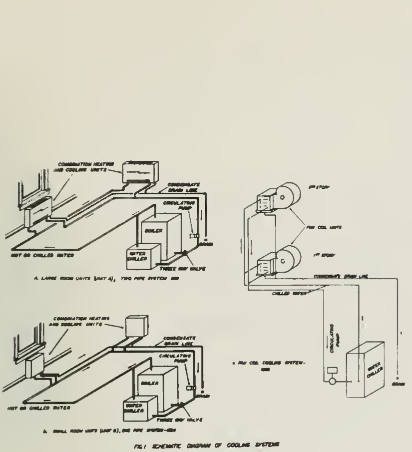


FIGURE 3.

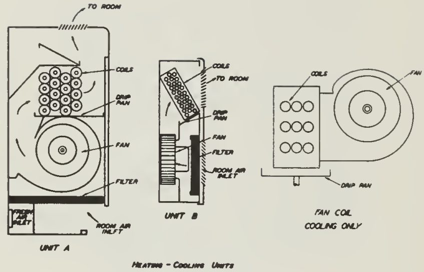


FIGURE 4.

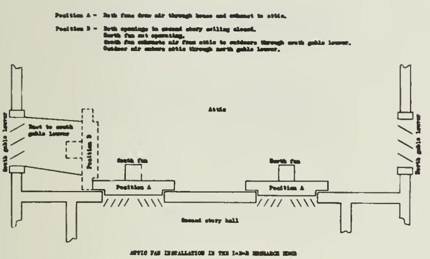


FIGURE 5.

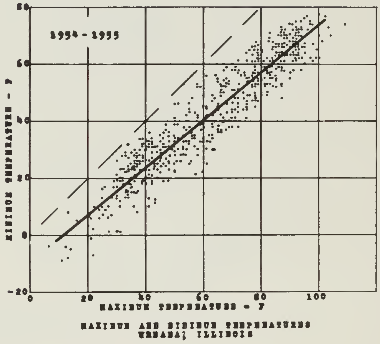
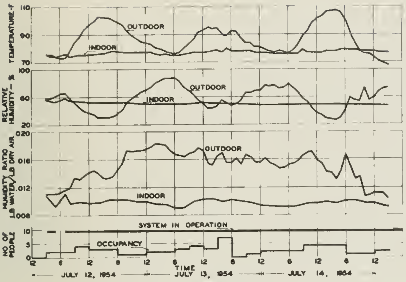
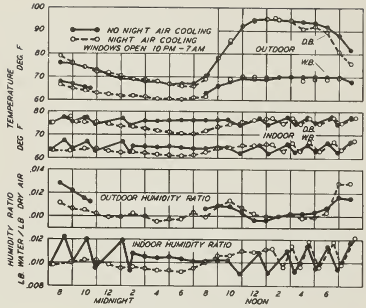


FIGURE 6.



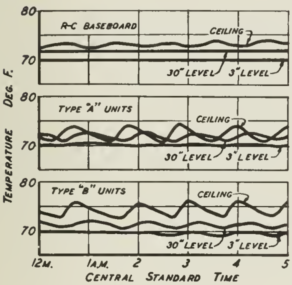
OPERATING CONDITIONS — SUMMER COOLING
SIX "B" UNITS IN I B R RESEARCH HOME

FIGURE 7.



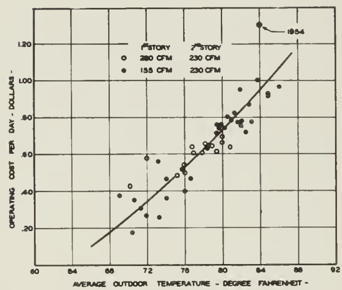
TEMPERATURE AND HUMIDITY
WITH AND WITHOUT NIGHT AIR COOLING

FIGURE 8.



ROOM TEMPERATURES, LIVING ROOM
AVERAGE OUTDOOR TEMP. APPROX. 31F.

FIGURE 9.



DAILY OPERATING COSTS
1955

FIGURE 10.