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Thermal Comfort Conditions in the NBS/DoE Direct Gain Passive Solar Test Facility

U.S. DEPARTMENT OF COMMERCE
National Bureau of Standards
National Engineering Laboratory
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U.S. DEPARTMENT OF COMMERCE, Malcolm Baldrige, ***Secretary***
NATIONAL BUREAU OF STANDARDS, Ernest Ambler, ***Director***

FOREWORD

This report is one of a series documenting NBS research and analysis efforts to support the Department of Energy Passive and Hybrid Solar Energy Program. The work reported here was sponsored by the Passive and Hybrid Solar Energy Division, Office of Solar Heat Technologies, U.S. Department of Energy, Washington, DC, under Interagency Agreement No. EA-77-A-01-6010.

ABSTRACT

The thermal comfort conditions in a direct gain cell of the NBS/DoE passive solar test facility were analyzed in accordance with the criteria specified in the recently revised ASHRAE Comfort Standard 55-1981, using test data collected during the month of October 1981 and the month of January 1982. It was found that the daytime operative temperature (as measured by the black globe temperature sensors) in an area near the large south glazing exceeded the upper boundary of the ASHRAE comfort envelope by a large amount on a clear day during both the thermal transition month of October and the cold winter month of January. The generally accepted method of computing the mean radiant temperature based only on the interior surface temperatures was found to produce large errors. The reflected solar radiation from the interior surfaces and the snow covered ground was believed to play a significant role on the measured black globe temperature and should be included in the computation of the mean radiant temperature for a space with large glazed areas.

Key words: ASHRAE Comfort Standard 55-1981; black globe temperature; comfort envelope; direct gain room; operative temperature; passive solar test facility; solar radiation; thermal comfort.

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1. INTRODUCTION

Over the past several years, the national concern with the increasing cost of conventional energy has stimulated the utilization of solar energy for the heating and cooling of buildings. Specifically, the design of residential buildings with passive solar heating features has received increasing attention from both the U.S. Department of Energy (DoE) and the general public because of the potential for large energy savings at lower cost than for many active solar heating systems. In a passive system, solar energy which impinges on the building is collected and stored by the architectural elements of the building such as heavy-mass walls, floor, or roof, and is distributed by natural means such as conduction, convection, and radiation without the use of solar collectors and conventional mechanical equipment. However, the acceptance and growth in the number of passive solar heated buildings will depend to a large extent on occupant comfort in such buildings. The American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE) Standard 55-1981, "Thermal Environmental Conditions for Human Occupancy," [1], which defines the condition for thermal comfort in a building, has recently been revised to include adjustments for extending the comfort zone due to the effect of air movement. The revisions also set limits on the amounts of temperature drift within the occupied zones, and on the non-uniformity of vertical temperature distribution, radiant asymmetry and floor temperature. All these have a direct bearing on the determination of the conditions of thermal comfort existing in passive solar buildings.

This report describes an investigation of thermal comfort condition in a direct gain type room which is one of the popular generic type configurations in passive solar heated buildings. In the direct gain configuration, the sun's rays are allowed to enter the building's living space directly through a large expanse of south facing glass arrangement (windows, clerestory, or sliding glass door). The surfaces and masses of the floor, walls, or ceilings, which are in direct contact with the building's interior living space, are used as the thermal collecting, transfer, and storage media for the entering solar energy. This direct and larger than normal amount of solar radiation into the living space causes a higher than usual surface temperature on the storage walls and floor, and a larger radiant energy flux passing through the space due to reflections and surface emissions. Both the higher surface temperatures and the larger radiant flux may cause thermal discomfort for the occupants and are two of the criteria established in the ASHRAE Standard 55-1981 for the evaluation of thermal comfort.

The building selected for the comfort analysis in this report is the direct gain cell of the NBS/DoE passive solar research test facility [2]. The test facility is one of the DoE Class A passive research facilities constructed in support of the DoE Program Area Plan [3, 4]. The facility has been in operation since October 1981, and several preliminary tests, each of from two to three weeks duration, have been conducted over the fall and winter seasons. The direct gain cell was extensively instrumented for thermal performance data collection during those tests. The collected data from two of the preliminary tests, one in the thermal transition month of October 1981, and one in the prime heating month of January 1982 were analyzed and discussed on the basis of the criteria contained in the ASHRAE Standard 55-1981. A brief description of the ASHRAE Standard, the direct gain room in the test facility, the test configurations and indoor physical environment settings, and the evaluation of the thermal comfort conditions, are presented in the following sections.

2. ASHRAE STANDARD 55-1981

The standard specifying the thermal environmental conditions in a building for the comfort of healthy people is ASHRAE Standard 55-1981, "Thermal Environmental Conditions for Human Occupancy" [1], revised in 1981. It states in part that at the center of the room and 0.6 m (2 ft) from the center of each exposed wall, the following conditions should be met at all times:

1. The operative temperature, T_o, measured within the occupied zone shall be on the boundary or within the "comfort envelope". The operative temperature is defined as the uniform temperature of a radiantly black enclosure in which an occupant would exchange the same amount of heat by radiation plus convection as in the actual nonuniform environment. The comfort envelope is defined as a quadrangle with the following corner coordinates when plotted on the psychrometric chart:

   Winter: T_o = 19.5-23°C (67.1-73.4°F) at 16.7°C (62°F) dew-point temperature (DP), and T_o = 20.2-24.6°C (68.4-76.4°F) at 1.7°C (35°F) DP.

   Summer: T_o = 22.6-26°C (72.7-78.8°F) at 16.7°C DP and T_o = 23.3-27.2°C (74-80.9°F) at 1.7°C DP

2. The humidity as described in terms of dew-point temperature shall not be less than 1.7°C (35°F) or greater than 16.7°C (62°F).

3. The average air movement in the occupied zone shall not exceed 0.15 m/s (30 fpm) in winter and 0.25 m/s (50 fpm) in summer. However, the comfort zone can be extended above 26°C (79°F) if the average air movement is increased 0.275 m/s for each °C (30 fpm for each °F) of increased temperature to a maximum temperature of 28°C (82.5°F) and 0.8 m/s (160 fpm).

4. When the mean radiant temperature (defined as the uniform surface temperature of a radiant black enclosure in which an occupant would exchange the same amount of radiant heat as in the actual nonuniform space) differs from the air temperature in the occupied zone, the air temperature shall be adjusted to keep the operative temperature within the appropriate comfort zone. For low air movement, T_o is approximately the average of the air temperature and the mean radiant temperature.

5. Monotonic, steady, non-cyclical temperature changes (drifts) shall not extend beyond the comfort zone by more than 0.6°C (1°F) and for longer than one hour.

6. The radiant temperature asymmetry for a plane element 0.6 m (2 ft) above the floor, shall be less than 5°C (9°F) in the vertical direction and less than 10°C (18°F) in the horizontal direction.

7. The surface temperature of the floor for people wearing appropriate indoor footwear shall be between 18°C (65°F) and 29°C (84°F).

The above criteria are specified for persons clothed in typical summer or winter clothing, at light, mainly sedentary, activity. The insulating value of clothing is defined in units of Clo, where 1 Clo = 0.155 $m^2 \cdot K/W$ (0.88 $ft^2 \cdot h \cdot F/Btu$). An example of typical winter clothing is given in ASHRAE 55-1981 as consisting of heavy slacks, long sleeved shirt and sweater, with an overall insulating value of 0.9 Clo, and for summer clothing, light slacks and short sleeve shirt (0.5 Clo). In addition to the above criteria, ASHRAE 55-1981 also states the conditions for temperature cycling, nonuniformity in vertical air temperature, for sedentary but nontypical clothing, and for active persons. Details of these are contained in Reference [1].

For the present analysis, the occupants were assumed to be in typical winter or summer clothing, and at sedentary or light activity level. Since the tests were conducted during the low humidity and colder months of October and January, and no humidifier was installed in the cell, the comfort limits of the comfort envelope under the low dew-point temperature (1.7°C) setting will be applied in the analysis in this report. Namely, at a dew-point temperature of 1.7°C (35°F),

Winter (0.9 Clo): T_o = 20.2 - 24.6°C (68.4 - 76.4°F)

Summer (0.5 Clo): T_o = 23.3 - 27.2°C (74-80.9°F).

3. DESCRIPTION OF THE TEST FACILITY

The NBS/DoE passive test [2] facility is located at Gaithersburg, Md. (latitude 39°N, longitude 77.3°W) less than one mile from the main NBS campus, on an open field with no shading from the surroundings. The building is a one-story, slab-on-grade, rectangular, frame structure with the long axis running from east to west. Figure 1 shows the south and east elevations of the building. The building is divided into four re-configurable test cells of equal areas. A floor plan of the test cells is shown in figure 2. Each cell has a floor area of 30.1 m^2 (342 ft^2) and contains a different south facing passive solar feature. South facing vertical clerestory windows are common to all the cells. The roof is divided into two sections; a 4.5 m (15 ft) south section of flat built-up construction and a 4 m (13 ft) north section of asphalt shingle covered, sloped roof to accomodate the clerestory windows. The ceiling height under the flat roof is 2.46 m (8.08 ft). It varies from 4.64 m (15.25 ft) at the south to 2.46 m (8.08 ft) at the north under the sloped roof section. The total volume of each of the cells is 88.4 m^3 (3122 ft^3). Both sections of the roof are insulated with 240 mm (9.5 in) of fiberglass batt insulation. The slab-on-grade floor consists of a 100 mm (4 in) concrete slab over 100 mm of gravel on compacted soil. The floor is insulated with 50 mm (2 in) extruded polystyrene insulation board along the perimeter of the slab down to the footings. The north outside wall is of singly-studded (2 x 4's) frame construction covered with insulating sheathing and cedar sidings and insulated with 90 mm (3.5 in) fiberglass batt insulation. The east and west outside walls are of doubly-studded (two rows of 2 x 4's separated by air space) frame construction covered with insulating sheathing and cedar sidings and insulated with two layers of 90 mm (3.5 in) fiberglass batt insulation. The double-studding and extra insulation are used to reduce the heat transfer through the end walls. The adjacent cells are separated by heavily insulated partition walls (38 mm (1.50 in) extruded polystyrene plus 90 mm fiberglass batt insulation). A heavy metal door insulated with 38 mm (1.50 in) of styrofoam serves as passage way between the cells. Besides the special solar features described later, the glazed areas common to all four cells consist of a double-pane, wood casement window with an unobstructed glass area of 0.67 m^2 (7.2 ft^2) located on the corner of the north wall of each cell, and a clerestory window located on the upper level of the south wall. The clerestory window consists of a three-section, double-pane, wood awning window which spans the entire width of the cell with an unobstructed glass area of 3.65 m^2 (39.25 ft^2). Both the north window and the clerestory window are provided with an operable interior insulating shutter with a thermal resistance value of 1.23 $m^2 \cdot K/W$ (R=7).

A different passive solar feature is incorporated into each of the cells. The direct gain cell (cell 4 in figure 2) studied in this report contains a double pane wood sliding glass door covering the entire width of the lower level south facing wall. The sliding glass door consists of two fixed sections and one sliding section. All three sections are of equal areas. The total unobstructed glass area is 6.27 m^2 (67.53 ft^2).

Besides the thermal storage provided by the concrete floor, this cell contains an additional thermal storage in the form of a 2.08 m wide by 2.46 m high (6.83 ft by 8.08 ft) concrete block wall located inside the north wall. The block wall is made of 200 mm by 400 mm by 200 mm (8 in by 16 in by 8 in) concrete masonry unit (CMU) with cores filled with mortar. The surface of the block is unpainted. A detailed description of this cell and the other three cells is given in [2].

4. INSTRUMENTATION

Extensive instrumentation was installed in 1981 in the direct gain cell as an initial step in instrumenting the entire test facility. The measured data included the following;

- £ Solar radiation: total horizontal, total south facing vertical inside and outside of the sliding glass door, normal beam radiation, and infrared sky radiation.
- £ Weather: wind speed and direction, barometric pressure, ambient air temperature.
- £ Relative humidity: inside and outside.
- £ Temperature: globe temperature, inside air temperatures, temperatures on all inside surfaces including the floor and ceiling, ground temperatures, and foundations wall temperatures.
- £ Heat fluxes: at the ceiling, east wall, north storage wall, and floor.
- £ Air infiltration.
- £ Energy consumption: lights and equipment, auxiliary heating, and cooling.

Figures 3 and 4 show the locations of the relevant air and surface temperature thermocouples, and black globe sensors. A detailed description of all the sensors and their locations is given in [2].

With the exception of the air infiltration data, which is a one-time measurement, and the energy consumption data which are measured separately, all other data are taken by an on-site data acquisition system and recorded onto magnetic tape at ten-minute intervals. The ten-minute interval data are numerically integrated and averaged into an hourly data set.

5. TEST CONDITIONS

Several preliminary thermal performance tests of the direct gain cell, each for a duration of from two to three weeks, have been conducted over the fall and winter seasons since October 1981. The collected data from two of the tests, one in the thermal transition month of October 1981 and one in the prime heating month of January 1981, were used for the thermal comfort analysis presented in this report. The October test was a floating temperature test where the room air temperature was unrestricted (free floating) with no auxiliary cooling or heating. The January test was a fixed temperature test where the minimum room air temperature was fixed by the heating thermostat setting at 18.3°C (65°F). The air temperature was free to float upward without restriction since no auxiliary cooling was provided.

During both of these tests, the south facing clerestory window shutter and the north window shutter were closed. No interior drapery or shading device was provided for the south facing sliding glass door which was kept closed during the tests. The room air temperature was kept mixed by the action of two small destratifying fans which were enclosed in two ducts and placed near the center of the room. The ducts extended from the floor to the sloped ceiling level and were open on both ends. One fan takes the room air from the floor level and discharges it at the sloped ceiling level near the clerestory window; the other fan takes the room air from the sloped ceiling level and discharges it at the floor level. Destratification was also promoted by the blower in the fan-coil unit below the north window which was turned on and run continuously throughout the two tests.

6. TEST RESULTS AND DISCUSSIONS

Relevant test data from the two test periods were analyzed and the thermal comfort conditions in the direct gain cell were evaluated on the basis of the ASHRAE Standard 55-1981 as discussed below. The relevant data for thermal comfort analysis were the solar radiation, the outside air temperature and the room air temperature variations, the black globe temperature variations at three locations in the room, the floor surface temperature variations at four locations, and the surface temperatures of the interior surfaces.

6.1 SOLAR RADIATION DATA

The solar radiation data on a horizontal surface for an 18-day (432 hour) period during each of the two tests are shown in figures 5 and 6 for the October test and the January test, respectively. The test periods contain a mix of clear, partly clear, and cloudy days. It is worth mentioning that several snow falls occurred during the January test and the ground around the test facility was covered with snow most of the time. This caused a large increase in the solar reflectivity of the ground and increased significantly the amount of solar radiation through the south glass as will be discussed later in this report.

6.2 ROOM AIR TEMPERATURE AND THERMAL COMFORT CONDITION

Figures 7 and 8 show the ambient temperature and room air temperature variations in the direct gain cell for each of the 18-day (432 hour) test periods in October 1981 and January 1982, respectively. The horizontal line on figures 7 and 8 are the upper bound (for summer time clothing) and lower bound (for winter time clothing) of the ASHRAE 55-1981 comfort limits described in section 2 of this report. It is assumed that the occupants will adjust their clothing when necessary. It is seen in figure 7 that without thermostat control (no auxiliary heating or cooling) during the October test, the room air temperature fluctuated daily between the two limits. This fluctuation is due to the effects of the solar radiation and higher ambient temperature at daytime, and the absence of solar radiation and lower ambient temperature at night. The predominant effect of solar radiation on the daytime room air temperature is evident in figure 8 during the January test. The room air temperature is seen to float upward from the heating thermostat setting of 18.3°C (65°F) whenever the solar radiation was high, and actually reached the ASHRAE 55-1981 upper limit on three days during the 18-day test period. Because the room air temperature was well mixed and controlled at the 18.3°C setting during the January test, the effect of low ambient and glass surface temperatures on the thermal comfort conditions during cloudy and snowing days and at nighttime was not evident in the room air temperature measurement. However, their effect will be shown in the globe temperature measurement discussed in the following section.

6.3 BLACK GLOBE TEMPERATURE AND THERMAL COMFORT CONDITION

The black globe temperature is measured as the equilibrium temperature at the center of a 150 mm (6 in) diameter thin wall-sphere with a nonreflective surface having an emissitivity of 0.95. At low air movement condition and in an environment where infrared radiation predominates, it is approximately a measure of the operative temperature and therefore the thermal comfort conditions in an occupied zone [1]. Figures 9 and 10 show the black globe temperature variations in the direct gain cell at three locations for each of the 18-day test periods in October 1981 and January 1982, respectively. Again shown in the figures are the lower and upper operative temperature limits for comfort in an occupied zone as specified in ASHRAE 55-1981. The upper limit is for a person in typical summer clothing at light activity level and the lower limit is for a person in typical indoor winter clothing at light activity level. It is assumed that the occupants will adjust their clothing when necessary. The three black globes sensors (shown in figure 3) were located along the south to north center line of the room, 1.52 m (5 ft) above the floor (in the shade all the time), and at a distance of 1.52 m (5 ft), 4.72 m (15.5 ft), and 7 m (23 ft) from the south glazing, respectively. It is seen from figures 9 and 10 that the black globe temperature 1.52 m from the south glazing (directly above the sunlit floor and close to the south glazing) frequently exceeded the upper ASHRAE comfort limit during the daytime hours by from 1°C (1.8°F) to as much as 7.5°C (13.5°F) and for a period of more than six hours whenever the days are clear (see figures 5 and 6). The daytime ambient temperature played only a minor role as can be seen by comparing the results of the mild October days with the very cold January days. Both the magnitude and the length of time where the black globe temperatures were over the limit greatly exceeded the comfort criteria for the temperature drift specified in reference [1] and described previously in item 5 of section 2 in this report (not greater than 0.6°C over the the limit and not longer than one hour), indicating that overheating and thermally uncomfortable conditions existed in the occupied space at least 1.52 m (5 ft) away from the south glazing. This condition occurred even when the air temperature in the space remained within a comfortable range most of the time. This uncomfortably hot condition was much less severe further away from the south glazing as can be seen from data on the other two black globe temperature sensors. Figures 11 and 12 show a comparison of the room air temperature with the black globe temperature at the middle of the test cell for the two test periods. It is seen that the black globe temperature and the air temperature became much closer to each other, as is generally the case in a normal living space other than a direct gain space.

In figures 9 and 10, it is noted that the lower operative temperature limit, as well as the criteria for temperature drift of the ASHRAE Comfort standard, was exceeded frequently, especially during the January test when the weather was cold. This is caused by the radiation heat loss to the large cold glass surface and the cold interior wall surfaces during non-solar hours and cloudy days. However, this cold condition is correctable by employing drapery or movable nighttime insulation over the glazing to reduce the radiation heat loss and by increasing the setting of the heating thermostat. On the other hand, the hot condition in an area as much as 1.52 m to 3 m (5 ft to 10 ft) away from the large south glazing will be harder to correct since any covering of the glazing during the cold January month will defeat the purpose of the direct gain design. This is not true in October where movable covering in the daytime when needed, would be very helpful in reducing the overheating condition. Figures 13 and 14 show a replot of the data on figures 9 and 10, but for a selected period of only four days for easier tracking and identification of the hourly variation of the black globe temperatures. The four days in October (19,20,21, 22), figure 13, covered a partly sunny day followed by two clear days and a partly sunny day. The four days in January (17,18,19,20), figure 14, covered two clear days with a partly sunny and a cloudy day in between.

6.4 COMPARISON OF COMPUTED OPERATIVE TEMPERATURE AND BLACK GLOBE TEMPERATURE

In an effort to identify the source of the thermally uncomfortable condition in the region near the south glazing, a computation was made to estimate the operative temperature at the south black globe temperature sensor location, using the method described in the ASHRAE Standard 55-1981 [1]. By assuming an average air movement of less than 0.15 m/s (30 fpm), the operative temperature at a location can be approximated as the arithmatic average of the air temperature and the mean radiant temperature at that location in the living space. The mean radiant temperature, T_{mrt}, can be computed from the inside surface temperatures and the associated angle factors as given by ASHRAE 55-1981. In equation form, the mean radiant temperature is written as [1]

$$T_{mrt}^4 = F_{p-1}\, T_1^4 + F_{p-2}\, T_2^4 + \cdots + F_{p-n}\, T_n^4 \qquad (1)$$

where T_n = absolute temperature of surface n,

F_{p-n} = angle factor from person at location p to surface n,

n = total number of surfaces.

For the present study, the mean radiant temperature and the operative temperature for a clear day in October (10-21-81) and a clear day in January (1-17-82) were computed, using the average temperature data on the interior surfaces at areas near the south glazing. Since the inside surface temperature of the glazing was not measured, that temperature was estimated by a simple quasi-steady state (mass of glass ignored) energy balance, using the measured solar radiation data on the south vertical surface, the wind speed, and the inside and outside air temperatures data as input, and an estimated glass (per layer) solar transmittance of 0.73 and solar reflectance of 0.07. Details of the energy balance on the glass are given in the appendix of this report. The results of this investigation are shown in figures 15 to 20.

Figures 15 and 16 show the measured interior surface temperatures, the interior room and ambient air temperatures, the estimated inside glass surface temperature, and the measured black globe temperature, used in the computation of the operative temperature for the two selected days. It is noted that most of the interior surfaces and air temperatures were less than the measured black globe temperature during the daytime hours. This fact is shown in figures 17 and 18. Figures 17 and 18 show the comparison of the computed and measured (the black globe temperature) operative temperature at the location 1.52 m (5 ft) from the south glazing and 1.52 m above the floor. The agreement is very good at the non-solar hours, but becomes very poor during the high solar radiation hour with the measured value above the computed value by as much as 3°C (5.3°F) in October and 8°C (14.4°F) in January. Since the interior surface temperatures and air temperature were all below the measured black globe temperature, it appears that this difference was caused mainly by the short wave solar radiant fluxes passing through the interior space where the black globe sensor was located. Parts of those solar fluxes are reflected by the portions of the floor and interior surfaces near the glazing that receive the solar beam radiation, and another parts come from the ground reflected solar radiation that passes through the glazing into the interior space. These radiant fluxes are not considered in the ASHRAE equation (equation (1)) for the mean radiant temperature because their effects are considered small for a normal sized window area, however, for the room with a large direct gain type glazing, and highly reflective surfaces in the solar spectrum, equation (1) should be modified to include terms for the interior and exterior (ground) reflected solar fluxes.

At this point it should be noted that the black globe sensor measures the highest possible mean radiant temperature due to its high absorptivity in both the infrared and the solar spectrum. For surfaces with a lower solar absorptivity such as people in light colored summer clothing, the effect of the reflected solar fluxes will be less than those measured by the black globe sensors reported here.

In figures 17 and 18, it is noted that the disagreement is much greater in January than in October. This can be explained by the larger sunlit area on the interior surfaces caused by the lower altitude angle of the sun, and by the solar radiation data shown in figures 19 and 20. From figure 19, the ratio of the insolation on the south vertical surface to that on the horizontal surface at noon time is 1.24, and the ground reflectivity is 0.2, while from figure 20, they are 2.03 and 0.55, respectively. The large ground reflectivity in January was caused by ground snow cover, which, in addition to giving a larger reflected solar radiation, would also cause some specular reflection (especially when glazed over) of the direct beam radiation through the large glass into the normally shaded interior space high above the floor level. This specular reflected beam radiation would have a larger effect on the mean radiant temperature than the diffuse reflected radiation since its intensity is not diminished by the distance between the south glazing and a point in the interior space. This is part of the reason that the black globe temperature in the middle of the room was higher than the air temperature in a January day shown in figure 12.

6.5 FLOOR SURFACE TEMPERATURE

Figures 21 and 22 show the surface temperature of the concrete floor at four locations along the south to north centerline of the direct gain cell during the October and January tests, respectively. During the month of October, approximately 25 percent of the floor area received direct solar radiation at solar noon. Specifically, the portion of the floor measured within 2.2 m (7.2 ft) from the south glazing in October received direct solar radiation at solar noon. The effect is shown in figure 21 where the line labeled $L = 1.02$ m shows a much larger increase during the day than the rest of the locations. The temperatures remain essentially uniform for the portions of the floor which do not see direct sunlight during the day. The nonuniformity in floor temperature is even more pronounced during the month of January due to the lower altitude angle of the sun in the sky; during this month, the portion of the floor up to 3.4 m (11.2 ft) from the south glazing is sunlit at solar noon. This effect is shown in figure 22 where the locations on the floor at 2.2 m and 3.1 m from the south glazing show a large difference in temperature from the rest of the floor area. It is noted that the location closest to the south glazing (at L = 1.02 m) also recorded the lowest temperature at nighttime and during cloudy days. This is caused, in addition to the normal perimeter slab losses, by the radiant heat loss from the floor to the cold glazing surface during those time periods. Movable nighttime type insulation or drapery should reduce these heat losses and provide a more comfortable floor temperature during those non-solar hours.

In figures 21 and 22, the ASHRAE 55-1981 limits on floor surface temperature (described in section 2 of this report) are also shown. It is seen that the upper limit was frequently exceeded in the sunlit floor area during the October test, and the lower limit was exceeded in the same area at nighttime and cloudy days during the January test. Movable window covering in the daytime in the thermal transition month of October and in the nighttime in the cold January month would reduce or eliminate these problems.

6.6 EFFECTS OF NONTYPICAL CONDITIONS

1. Air movement: In the present study, natural circulation was assumed to exist within the living space where the air movement in the occupied zone was less than 0.15 m/s (30 fpm). ASHRAE Std. 55-1981 specifies that the upper boundary of the comfort zone can be extended to above 26°C (79°F) if the average air movement is increased 0.275 m/s for each 1°C of increased temperature (30 fpm for each 1°F) to a maximum temperature of 28°C (82.5°F) and air movement of 0.8 m/s (160 fpm). Therefore, if some mechanical means were used to increase the air movement 0.8 m/s during high operative temperature periods, the occupied zone would stay within the extended upper comfort boundary for a longer period of time. However, as is evident from figures 9 and 10, overheating and uncomfortable condition would still exist near the south glazing area.

2. Nontypical clothing: ASHRAE Std. 55-1981 specifies that for each 0.1 decrease or increase in Clo value, the comfort zone boundary operative temperature can be increased or decreased respectively by 0.6°C. With a minimal clothing of 0.05 Clo, the upper boundary can be extended to 29°C (84°F). Therefore, if the occupants would adjust (reduce) the clothing to below the typical summer clothing's value of 0.5 Clo during the high operative temperature period of the day, the length of the time period when the comfort criteria was exceeded would be reduced or sometimes eliminated.

3. Active persons: The comfort zone boundary values stated in section 2 are for a sedentary or slightly active person. If the activity level of the occupants are increased, the standard requires a decrease in the upper boundary value. For example, for a person doing domestic work (medium activity) and wearing clothes with a Clo value of 0.35, the reduction in

the upper boundary temperature would be by 3.2°C (5.8°F). This would cause the operative temperature at the center of the room to exceed the allowable boundary value of 24°C (75.2°F) frequently in both October and January as can be seen in figures 9 and 10.

7. CONCLUSIONS

Based on the study of the preliminary test data collected during the October 1981 and January 1982 tests on the direct gain cell of the NBS/DoE passive test facility, the following conclusions can be drawn:

1. The occupied zone near the south glazing of a direct gain type space with large south glazing area will exceed the upper comfort limit specified in the ASHRAE thermal comfort standard 55-1981 during daytime hours in a clear day during both the thermal transition month of October and the prime heating month of January. However, mechanical cooling and movable window covering should reduce or eliminate this condition during October and presumably also during the summer months.

2. The same region described above would also fall below the lower comfort limit at nighttime and during cloudy or snowing days in the winter months, caused by nighttime infrared radiation heat loss to the large cold south glass surface. Removable insulation or window covering may help in alleviating this condition.

3. Ground and interior surface reflected solar radiation fluxes near the south glazing area contribute significantly to the uncomfortably warm condition near the large south glass area common to a direct gain space. Snow covered ground in winter, which may reflect specularly, plays a significant role in increasing the operative temperature of the occupied space.

4. For a direct gain type building with large glass area, the equation described in the ASHRAE Standard 55-1981 for the estimation of the mean radiant temperature in an occupied space may need revision to include terms for the reflected solar radiation fluxes passing through the space near the large south glass even when the space is shaded by the overhang roof from the direct solar radiation.

5. Nighttime infrared radiation heat loss from the floor to the large and cold south glazing caused the floor surface temperature to drop below the ASHRAE specified limit on floor temperature during the cold winter season. Removable window covering should again be able to alleviating this condition.

It should be noted that the results of this analysis are limited by the conditions under which the tests were conducted. In order to obtain and provide clean data to be used for the verification of energy analysis computer programs, those preliminary tests were run without any window covering at any time during the test. There is no doubt that window covering, even the everyday type, will change the indoor environment to a more comfortable level during some parts of these tests as discussed above. Additional testing with the proper and realistic application of window covering as part of the test procedure is therefore recommended for a better evaluation and understanding of the thermal comfort conditions in a direct gain space. Finally, the role of the reflected solar radiations from both the interior surfaces and the ground, especially if the ground is covered with snow, on the room mean radiant temperature computation should be studied in more detail and quantified.

8. ACKNOWLEDGMENTS

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4. Hamilton, B.L. and Scofield, P.M., "National Passive/Hybrid Performance Evaluation Program for System Development," Proceedings of the 5th National Passive Solar Conference, Amherst, Ma., October 1980.

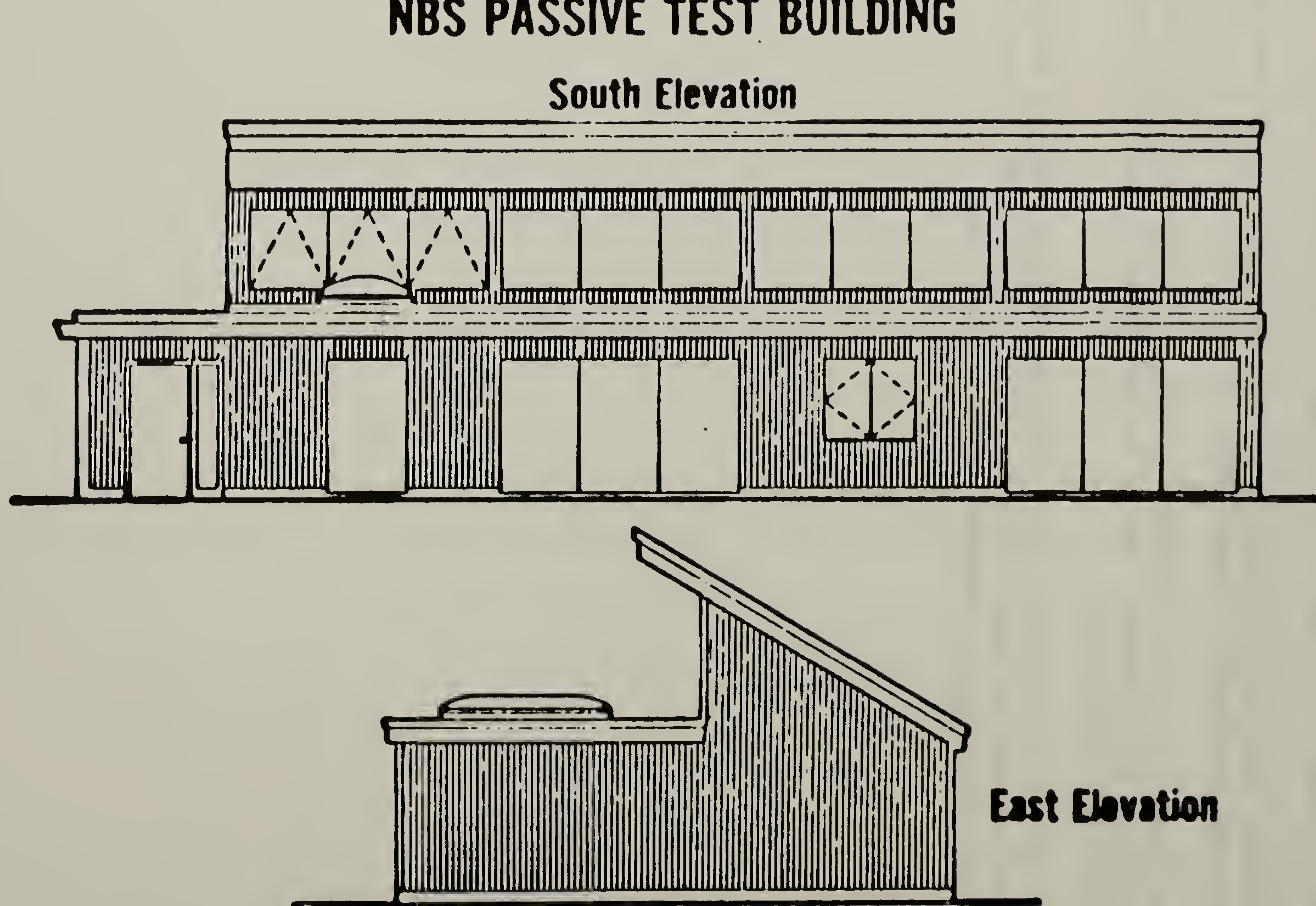


Figure 1. The NBS passive solar test building

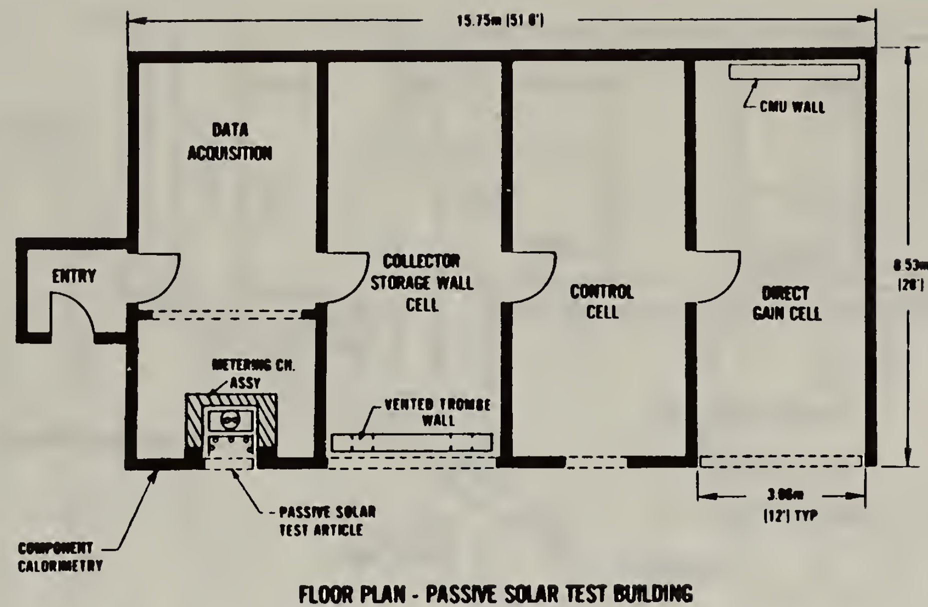
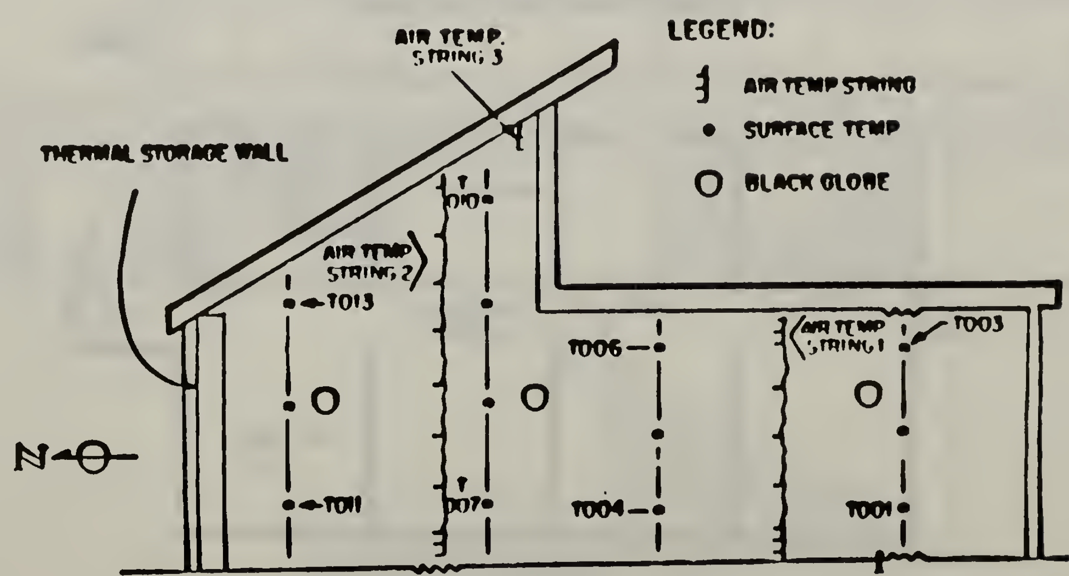


Figure 2. Floor plan of the NBS passive solar test building



VIEW TOWARD EAST WALL (SAME FOR WEST WALL)

Figure 3. Sensor location - room and globe temperature along the center line, and surface temperature on east and west walls

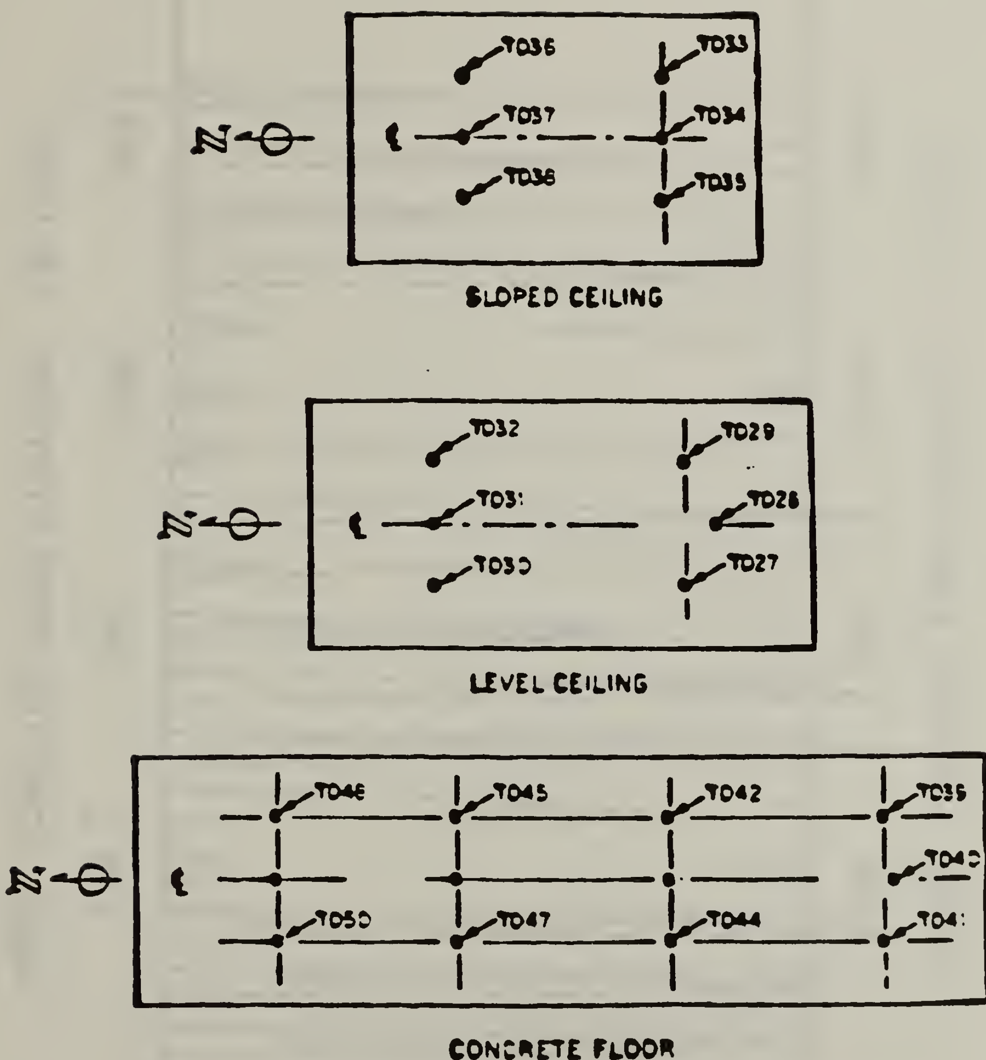


Figure 4. Sensor location - ceiling and floor surface temperature

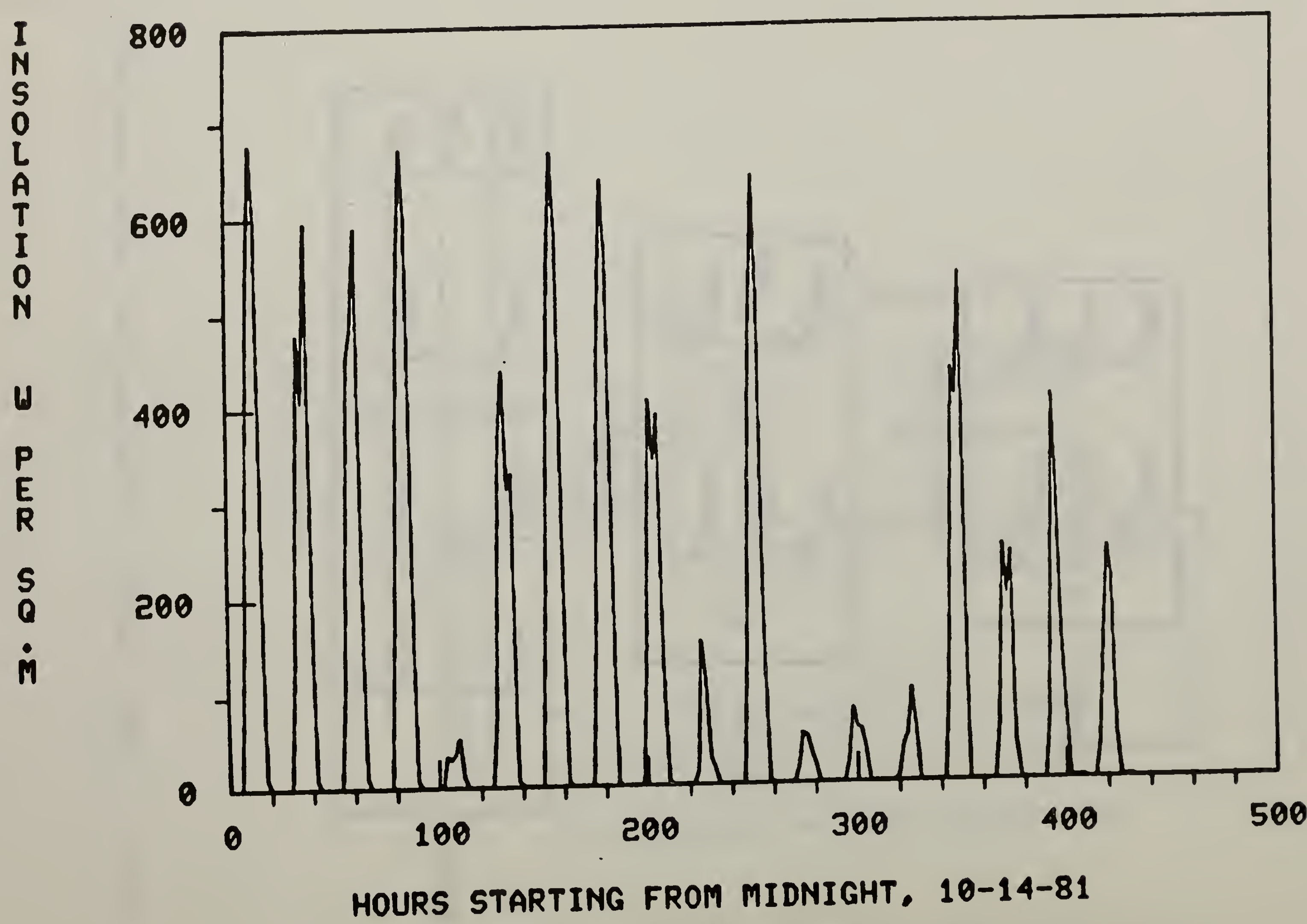


Figure 5. Solar radiation on a horizontal surface during the October 1981 test

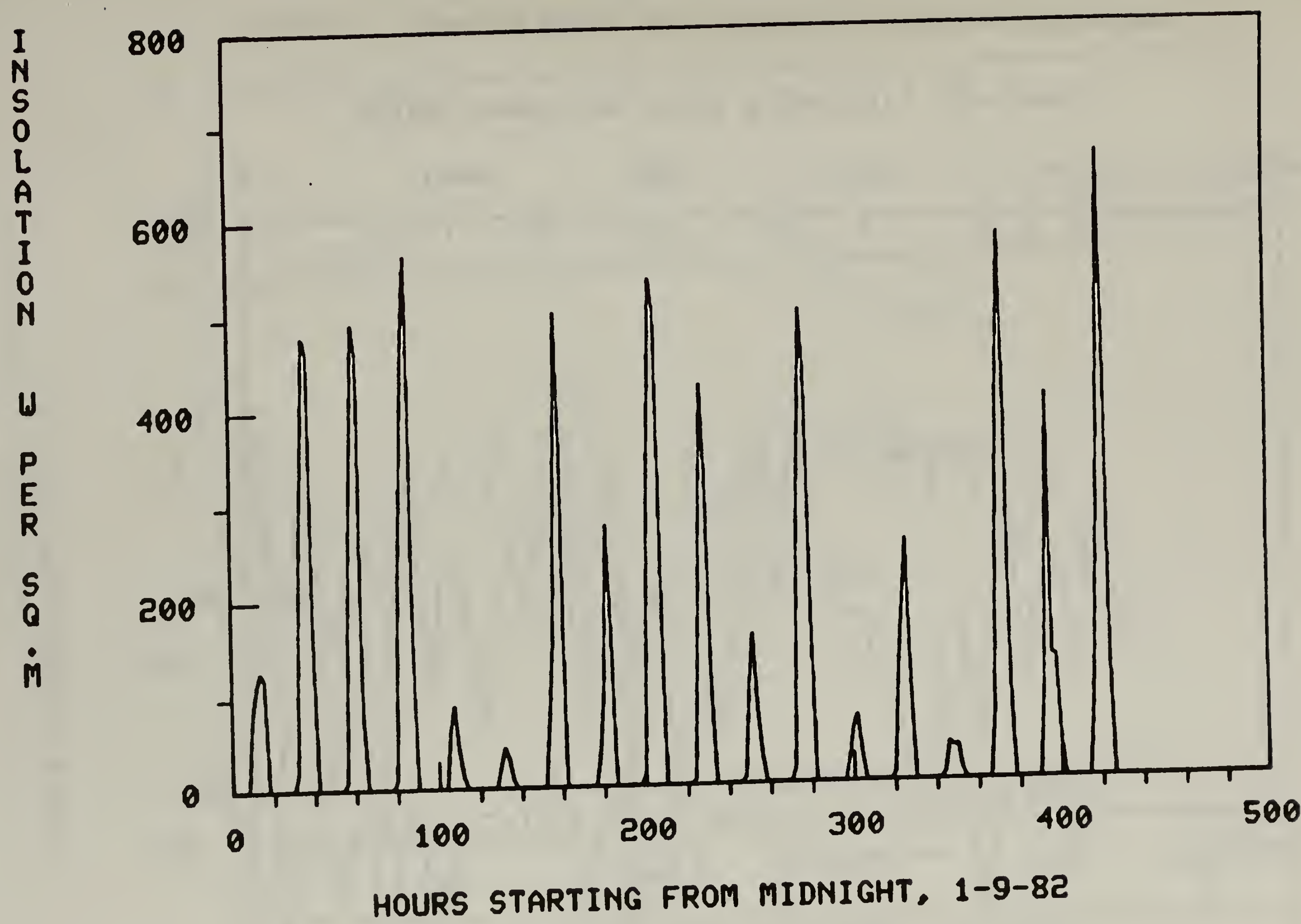


Figure 6. Solar radiation on a horizontal surface during the January 1982 test

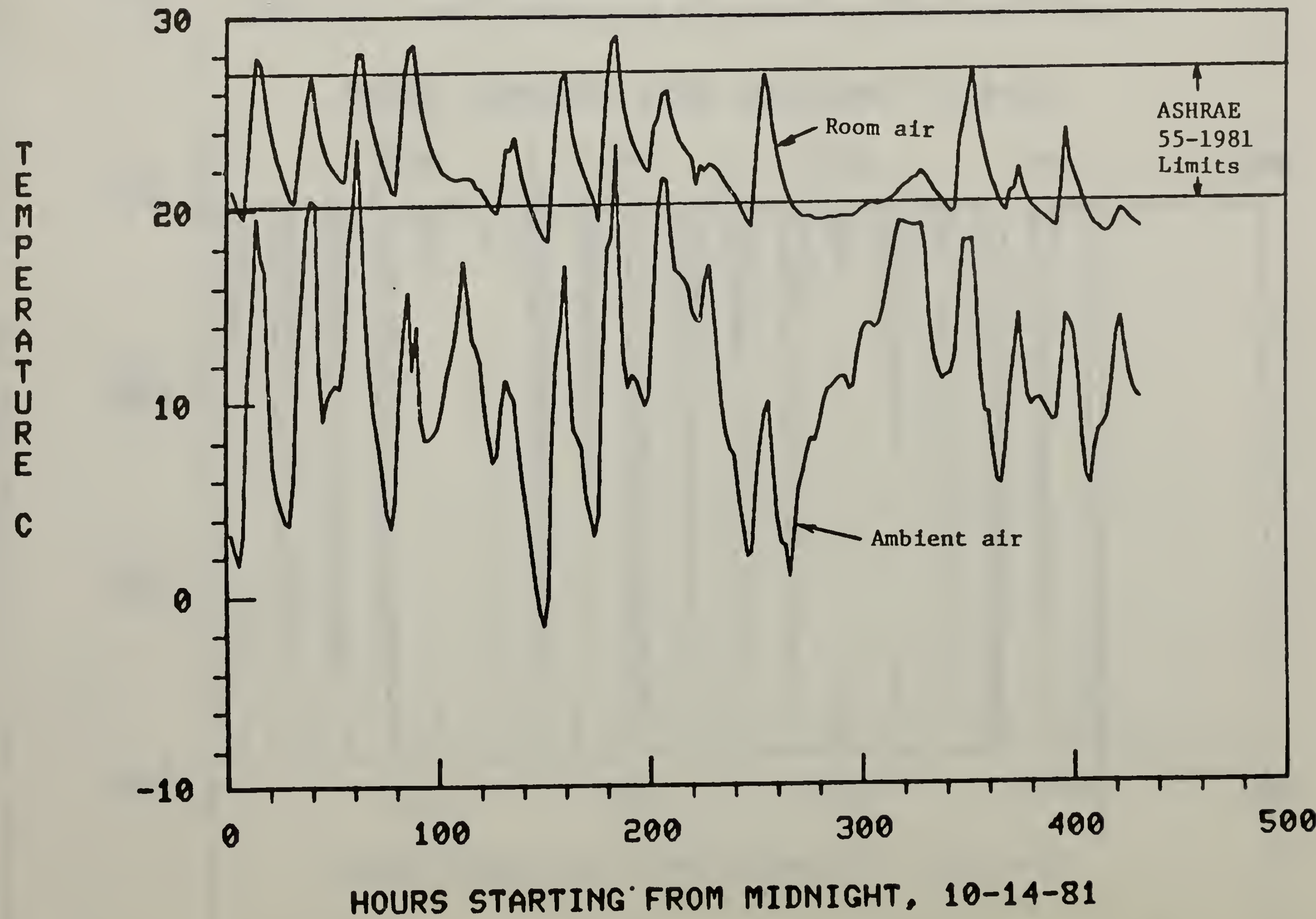


Figure 7. Room and ambient air temperature during the October test

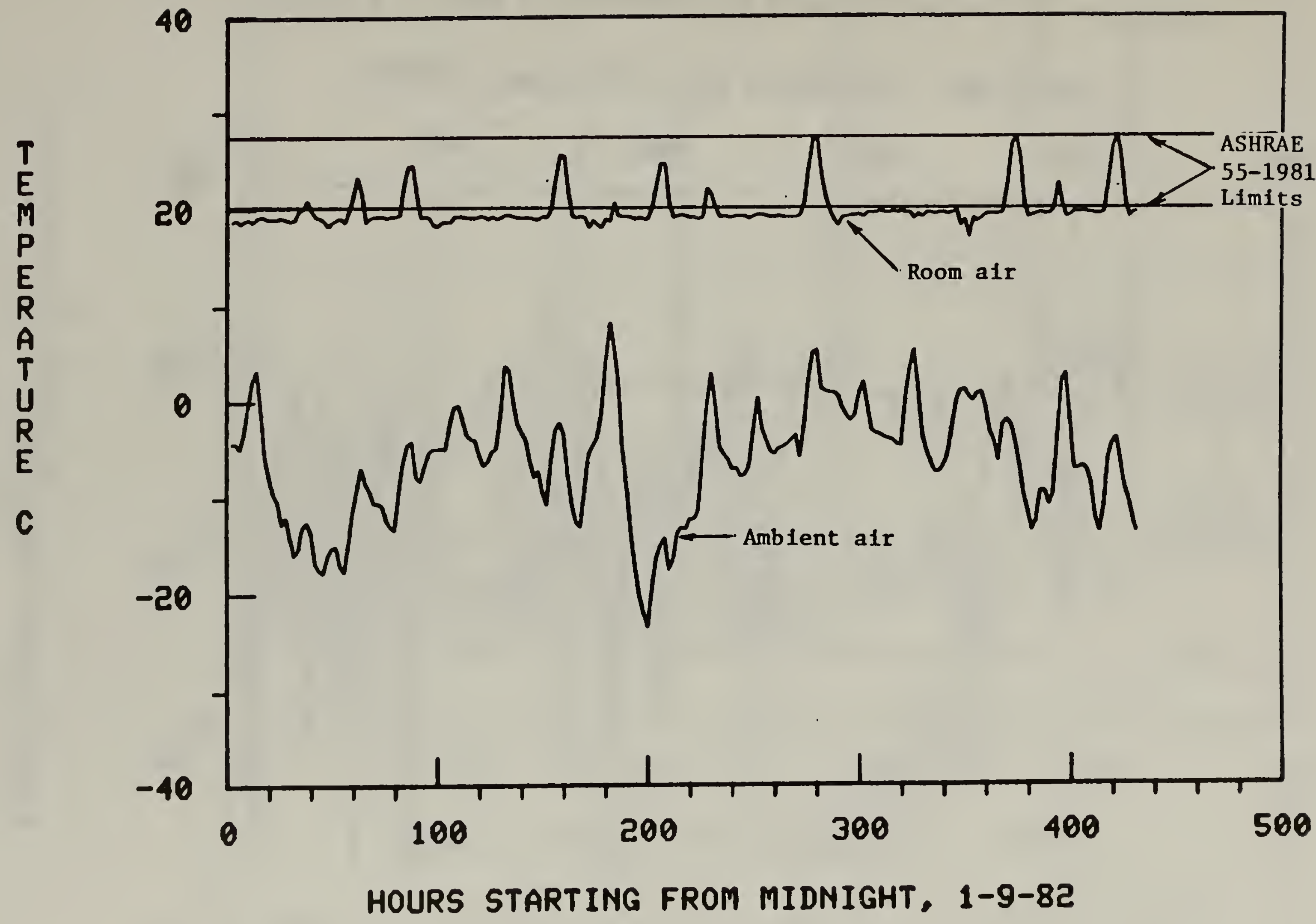


Figure 8. Room and ambient air temperature during the January test

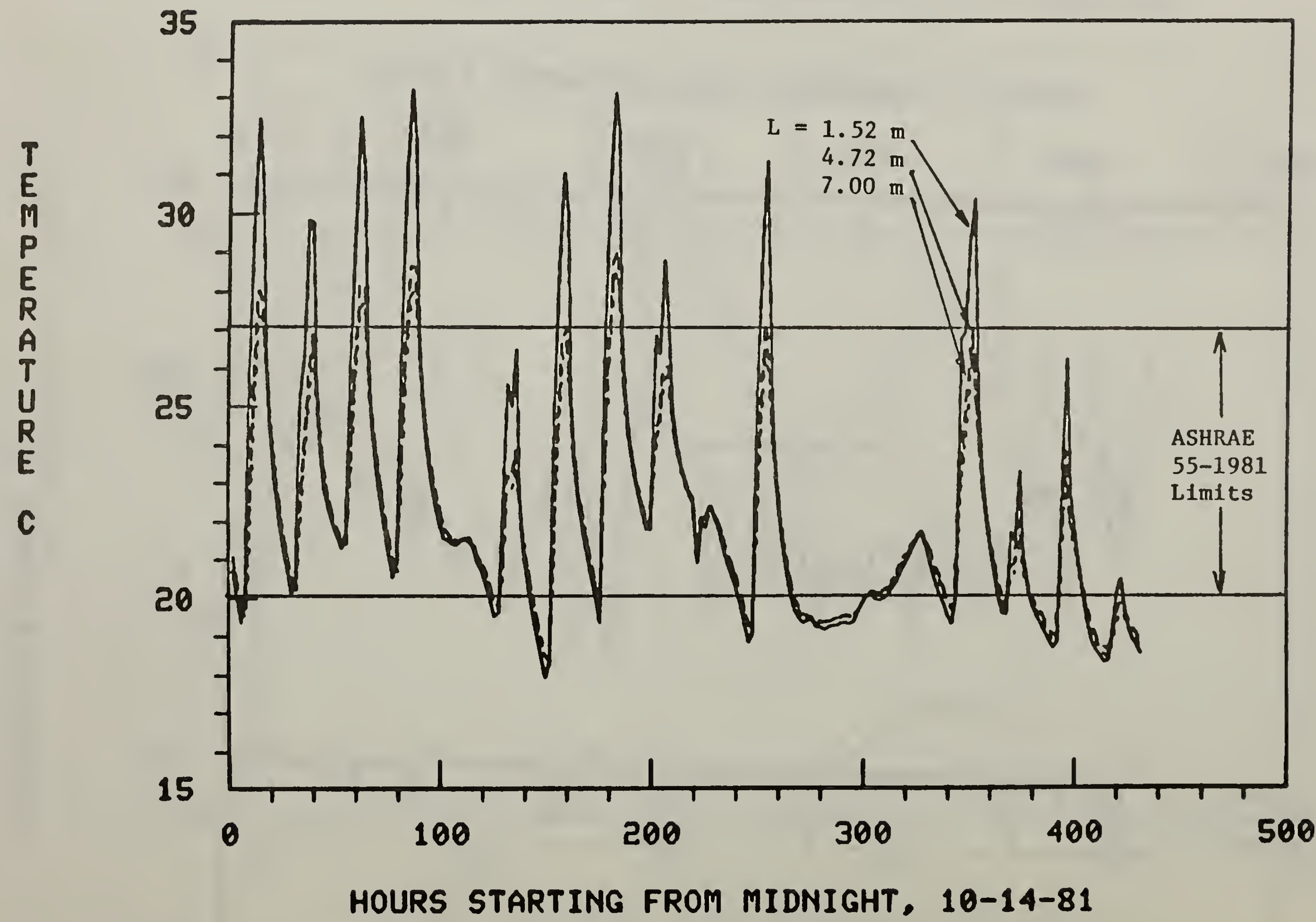


Figure 9. Black globe temperature along the center line at distance L from south glazing and 1.52 m above the floor during the October test

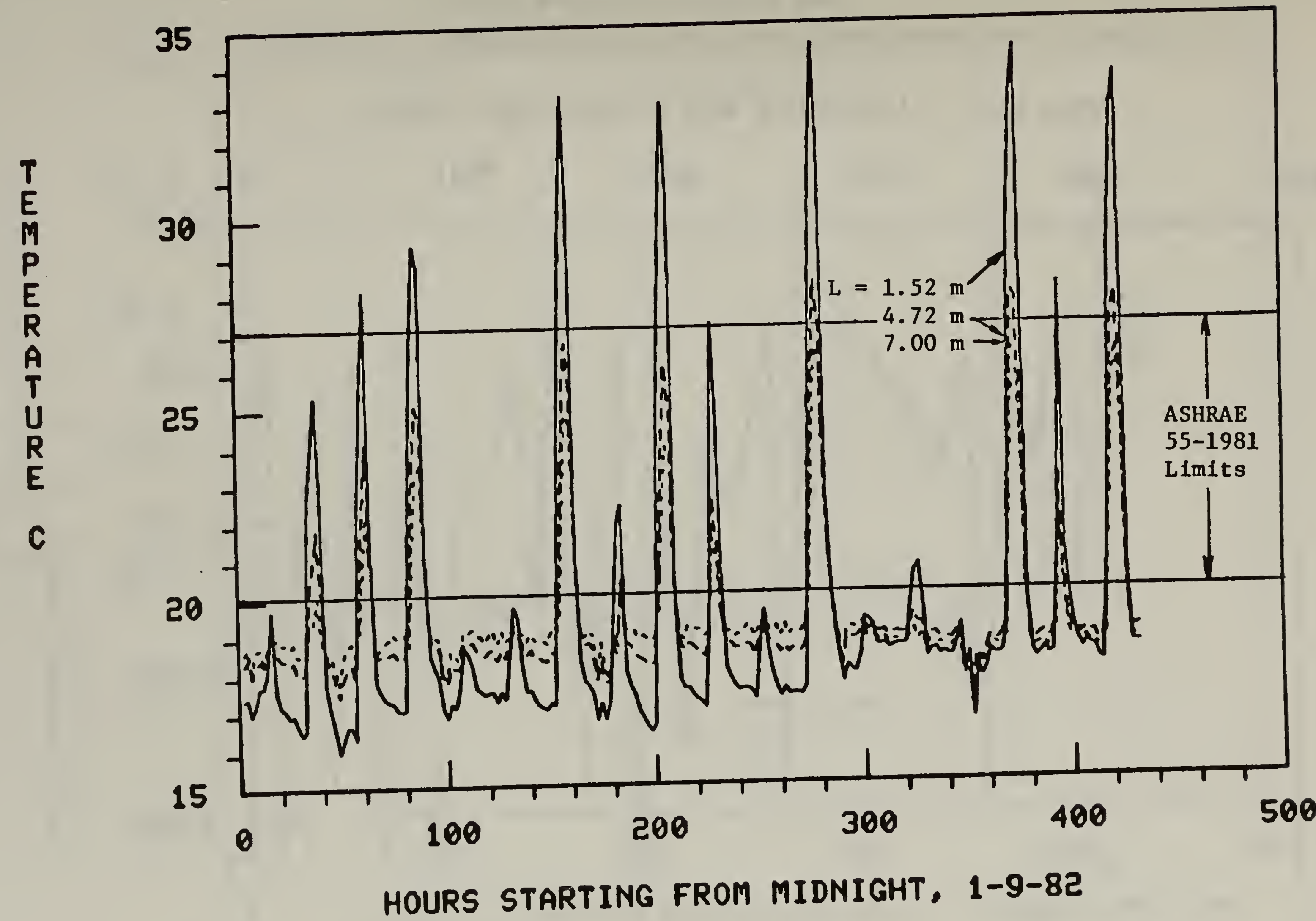


Figure 10. Black globe temperature along the center line at distance L from south glazing and 1.52 m above the floor during the January test

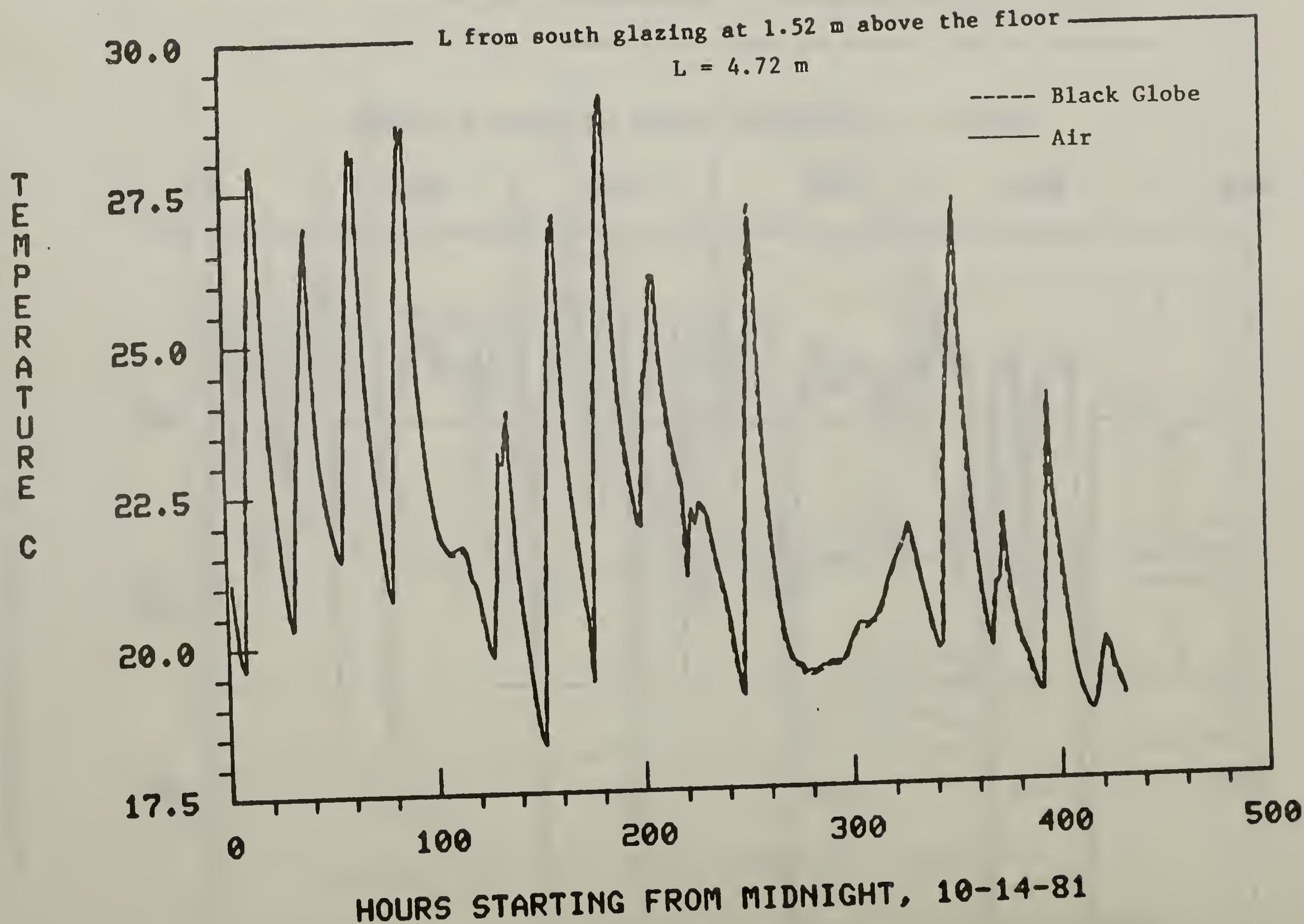


Figure 11. Comparison of air and black globe temperature at center of room during the October test

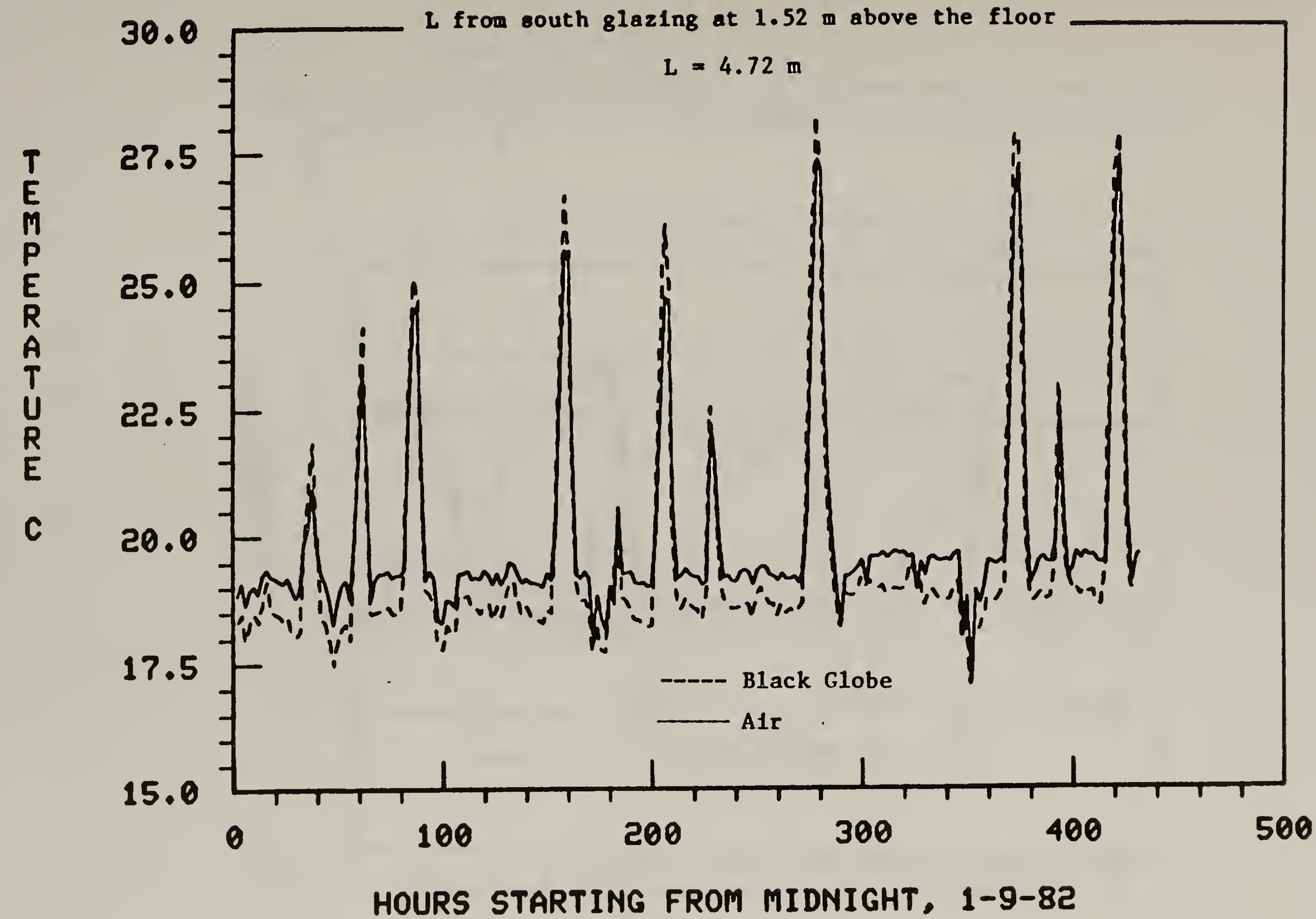


Figure 12. Comparison of air and black globe temperature at center of room during the January test

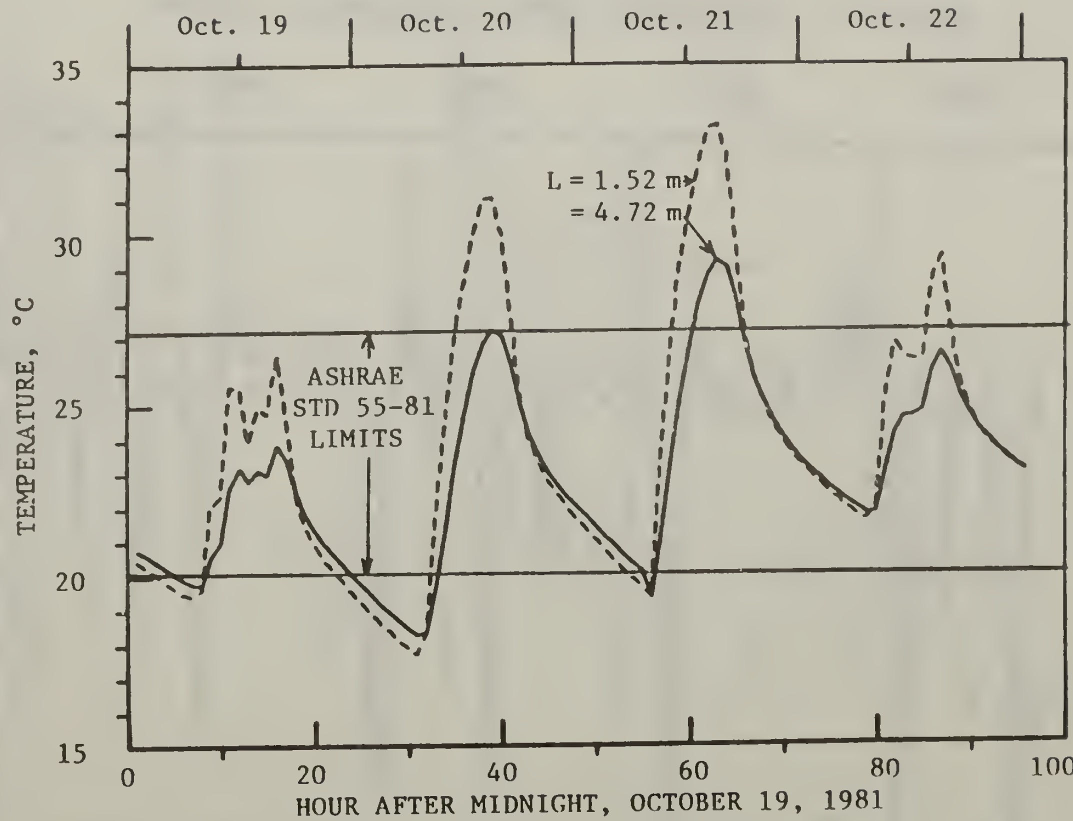


Figure 13. Black globe temperature along the center line at distance L from south glazing at 1.52 m above the floor during October 19-22, 1981

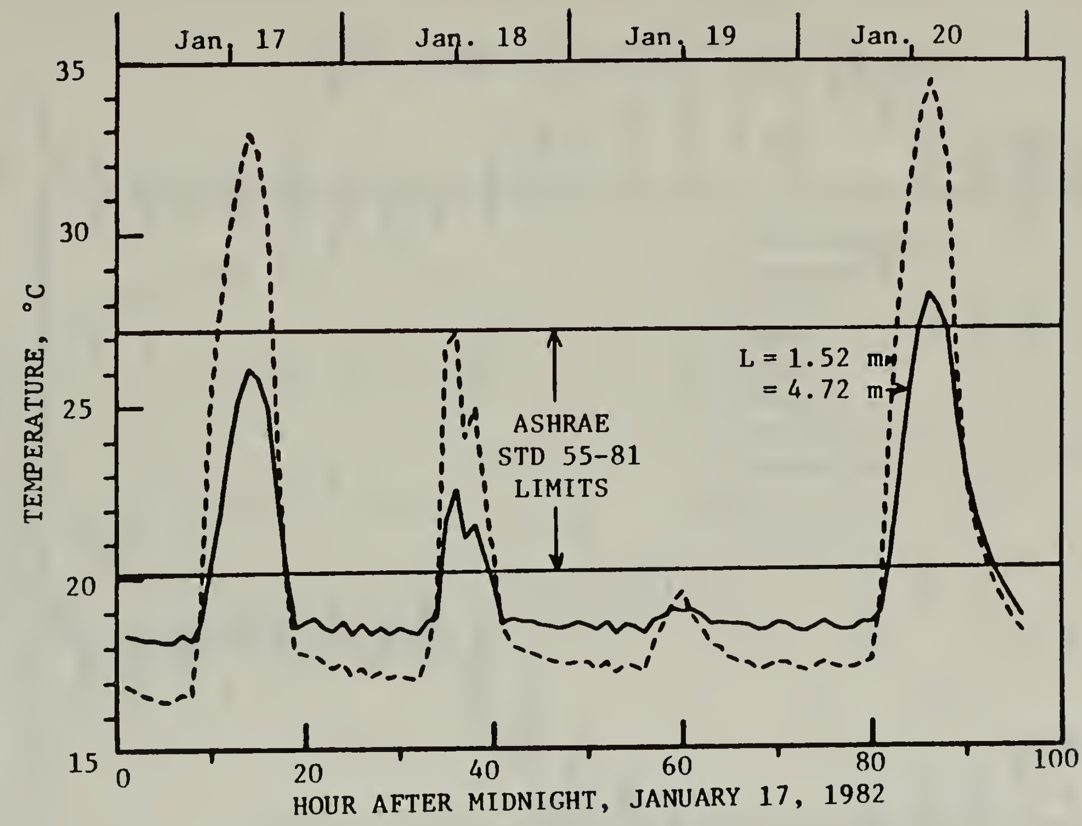


Figure 14. Black globe temperature along the center line at distance L from south glazing at 1.52 m above the floor during January 17-20, 1982

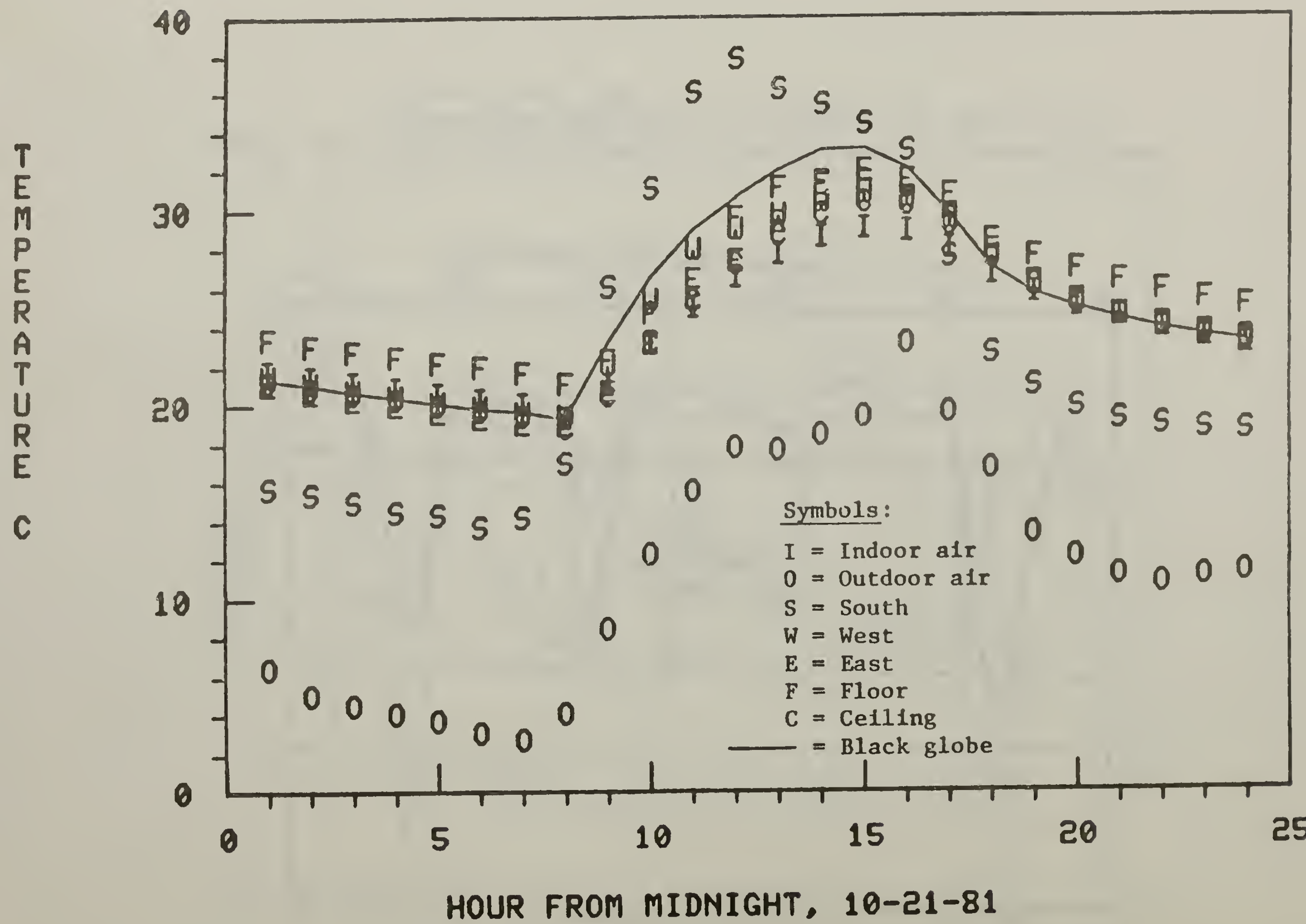


Figure 15. Average hourly surface temperatures near the south glazing during October 21, 1981

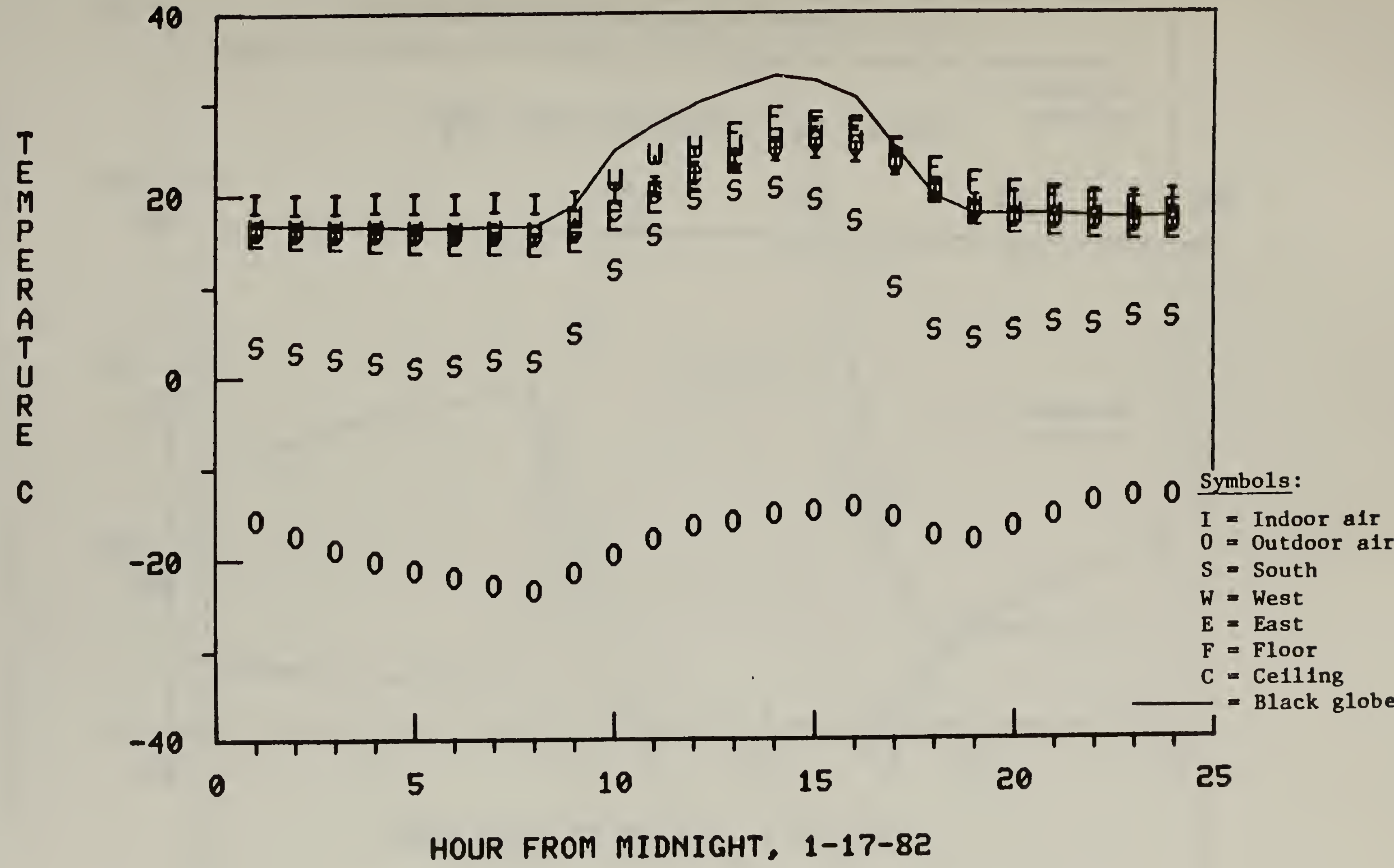


Figure 16. Average hourly surface temperatures near the south glazing during January 17, 1982

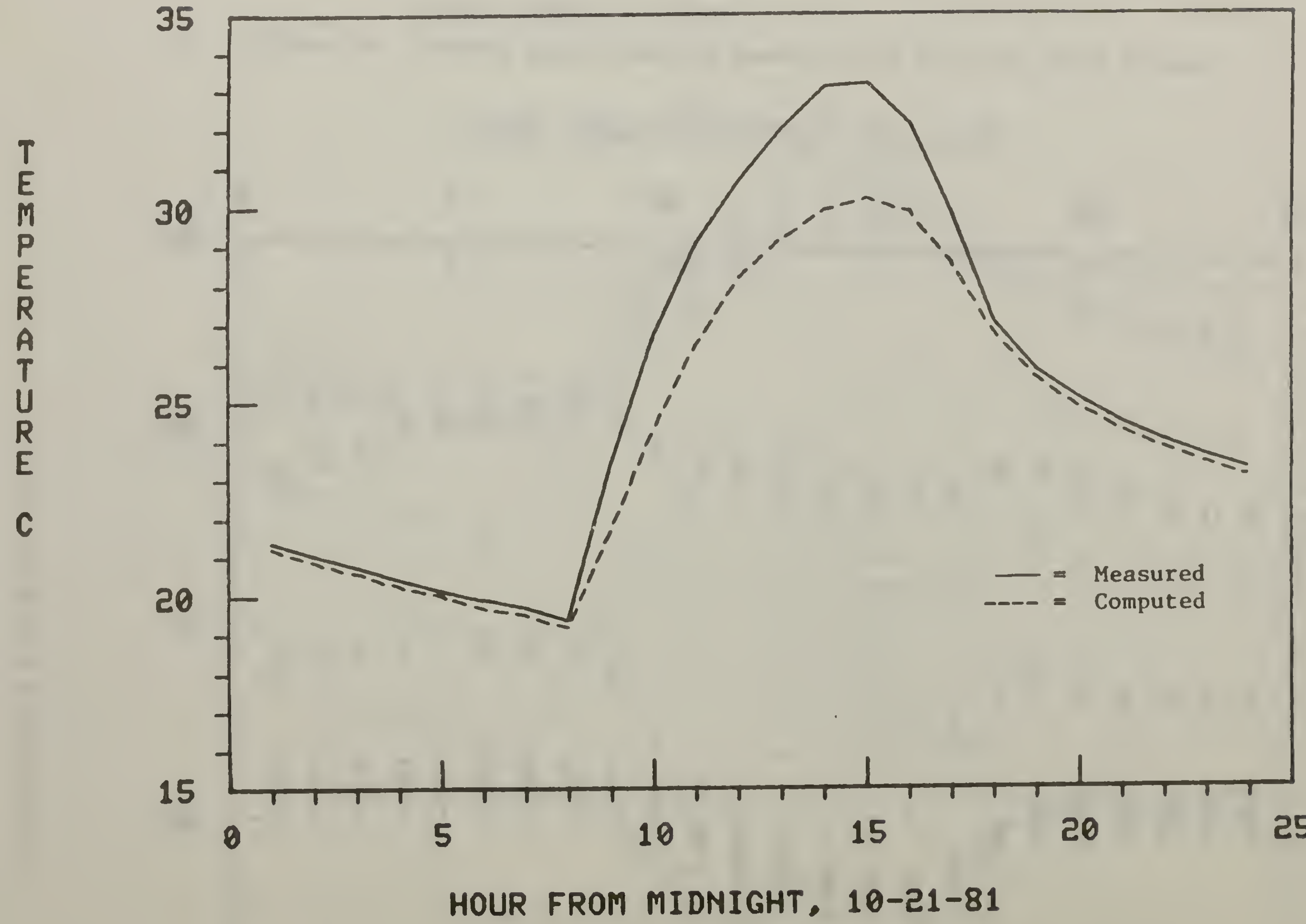


Figure 17. Comparison of measured and estimated operative temperature near the south glazing during October 21, 1981

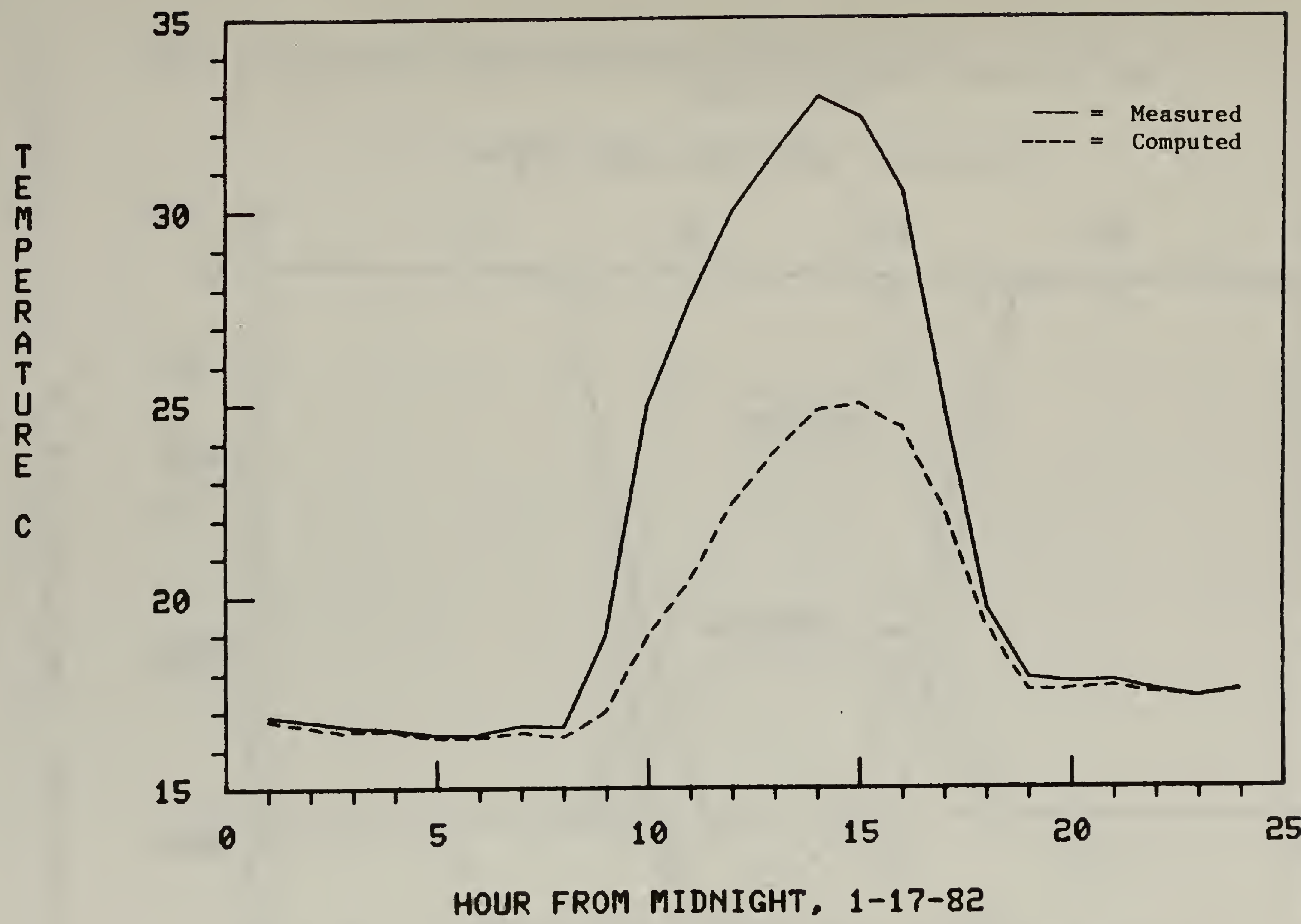


Figure 18. Comparison of measured and estimated operative temperature near the south glazing during January 17, 1982

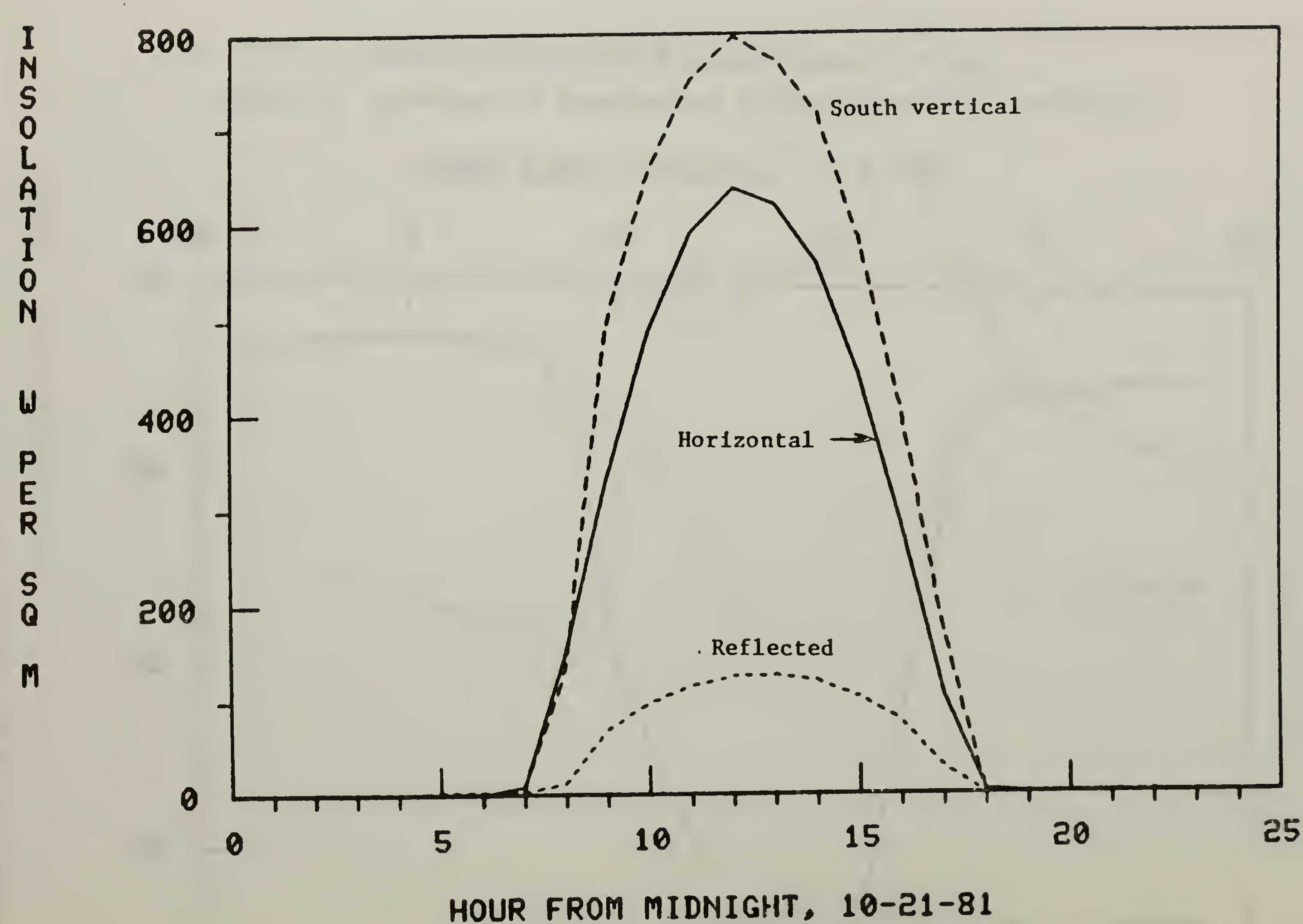


Figure 19. Hourly solar radiation data during October 21, 1981

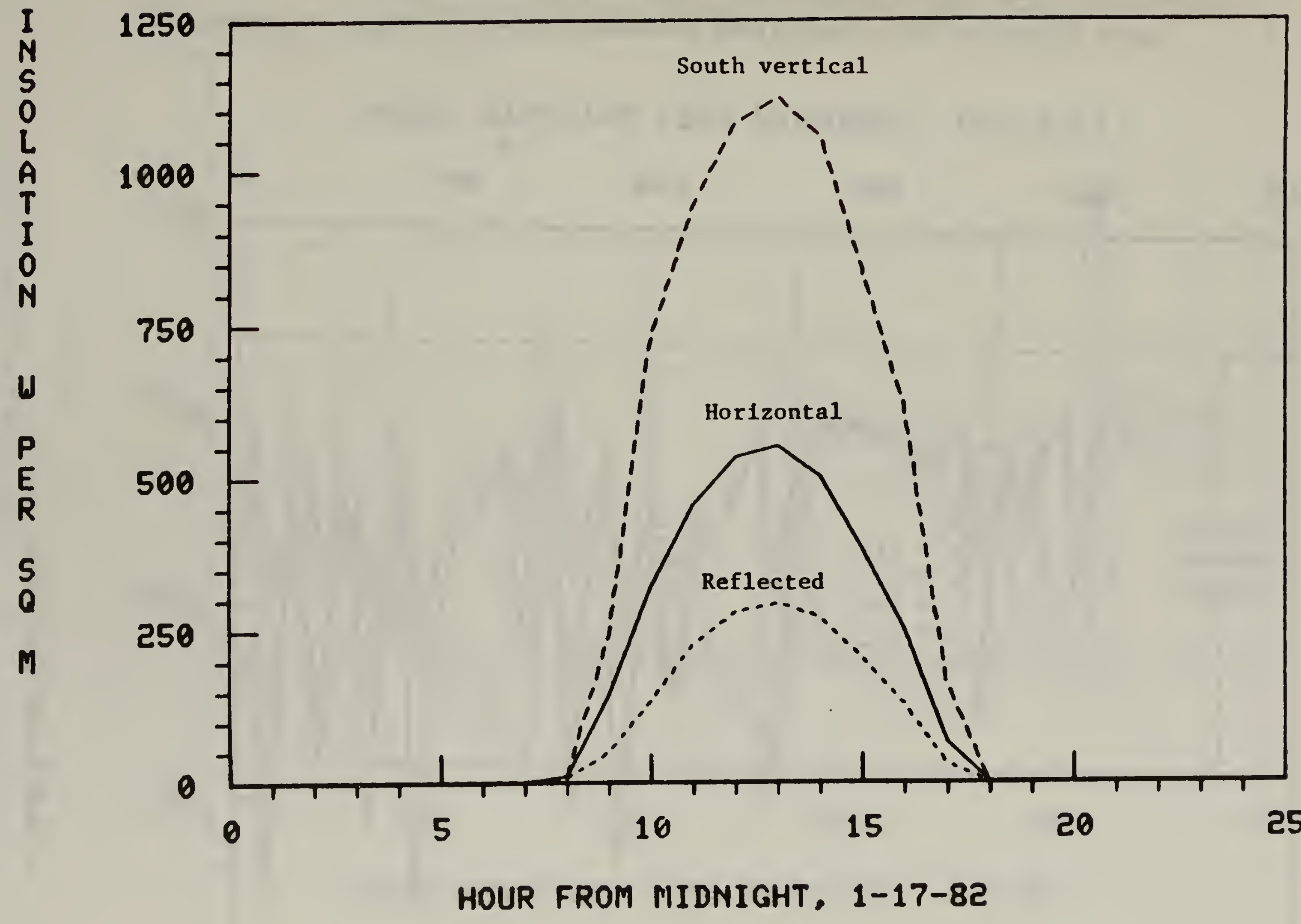


Figure 20. Hourly solar radiation data during January 17, 1982

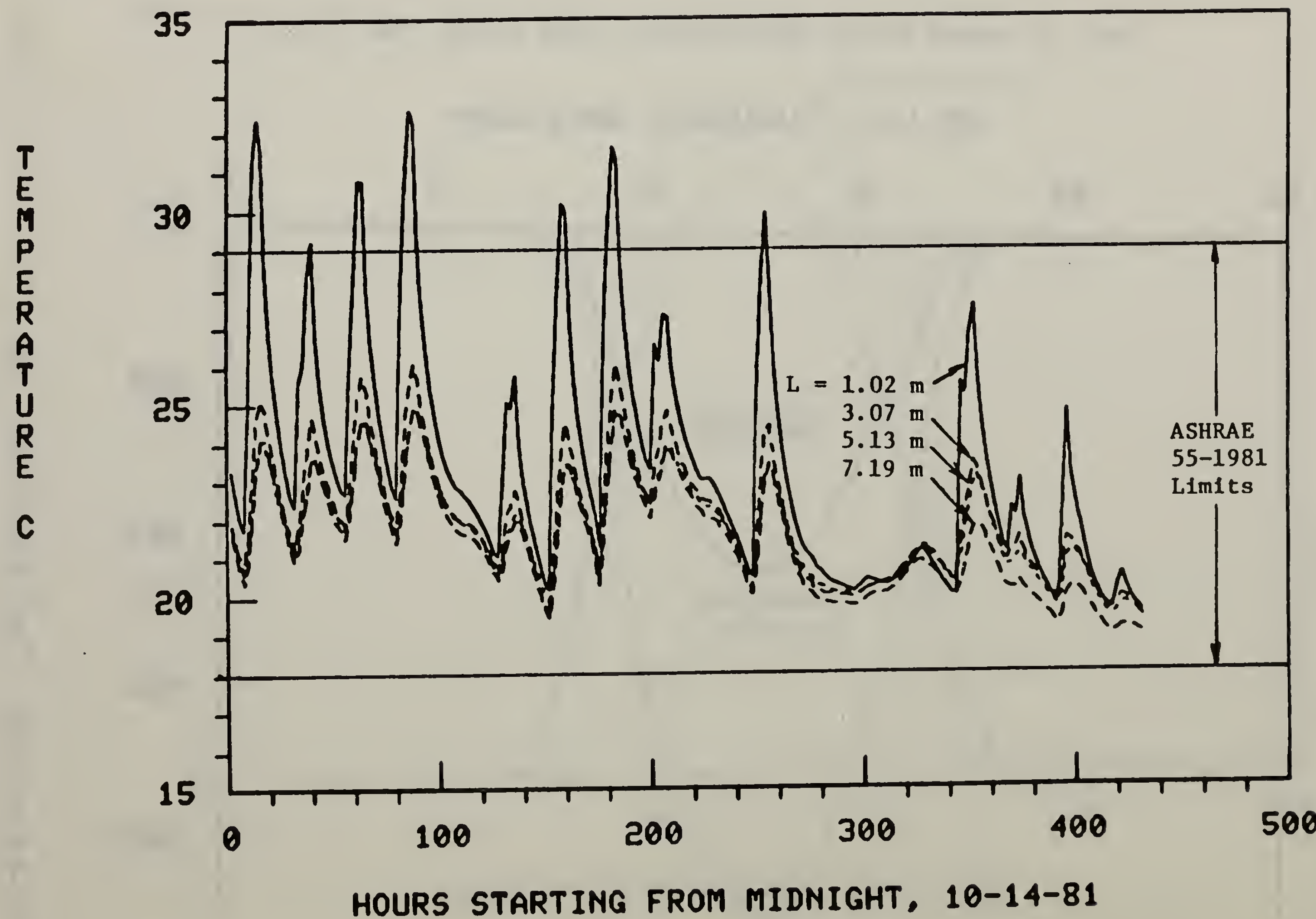


Figure 21. Floor surface temperature variations along the center line at a distance L from the south glazing during the October test

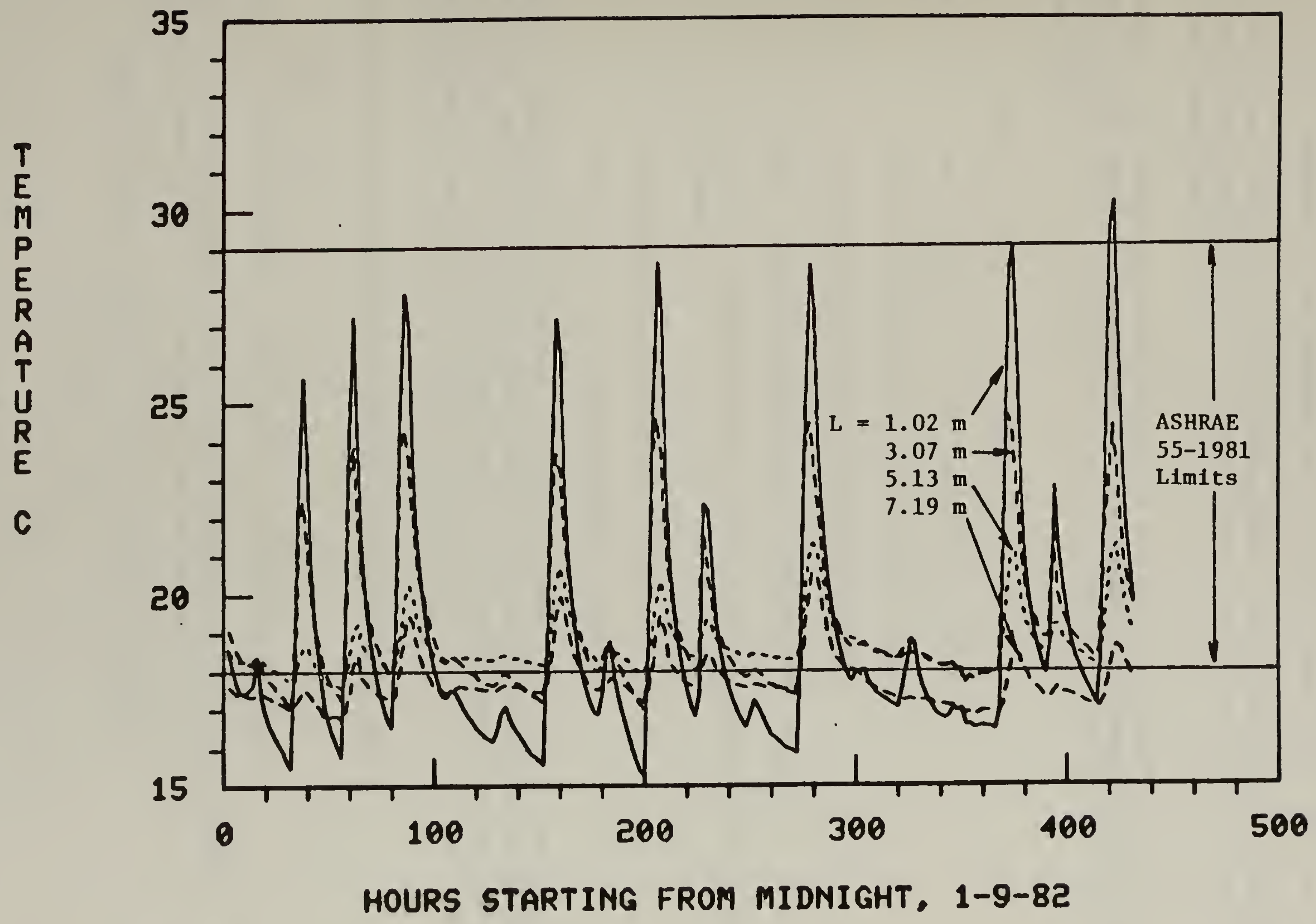


Figure 22. Floor surface temperature variations along the center line at a distance L from the south glazing during the January test

APPENDIX - ESTIMATION OF INNER SURFACE TEMPERATURE OF A DOUBLE-PANE GLASS WINDOW

For the present analysis, both layers of the double-pane glass window are assumed to be thermally mass-less so that quasi-steady state exists at any time of the day. Namely, it is assumed that for each glass layer, the heat gain by the absorption of solar radiation and convection is equal to the heat loss by convection and radiation to the surroundings. In addition, it is assumed that no temperature gradient exists in each glass layer, that is, the layer is at a uniform temperature.

Assuming that the solar radiation on the vertical glass surface H, the wind speed W_s, the ambient temperature T_o, the inside room temperature T_i, and the solar properties of the glass are known, an energy balance on the outside glass layer gives,

$$\alpha(H + \tau\rho H) = h_o(T_1 - T_o) - u_s(T_2 - T_1) \quad \text{(A-1)}$$

where α = solar absorptivity of the glass layer

τ = solar transmissivity of the glass layer

ρ = solar reflectivity of the glass layer

h_o = heat transfer coefficient of the outside surface

u_s = thermal conductance of the air space between the two layers of glass

T_1 = temperature of the outside glass layer

T_2 = temperature of the inside glass layer

The left hand terms in equation (1) are the portions of the incoming solar radiation and the reflected solar radiation from the inside layer surface that are absorbed by the outside glass layer. Any re-reflection is considered small and ignored. The right hand terms are heat loss to the ambient air and heat gain from the inside layer through the air space.

Similarly, an energy balance on the inside glass layer gives

$$\alpha(\tau H) = h_i(T_2 - T_i) + u_s(T_2 - T_1) \quad \text{(A-2)}$$

where h_i = heat transfer coefficient of the inside surface.

Equations (A-1) and (A-2) are the two equations for the two unknown T_1 and T_2. Combining equations (A-1) and (A-2) and solve for T_2 gives

$$T_2 = \frac{h_o u_s T_o + h_o h_i T_i + h_i u_s T_i + \alpha H((1 + \tau\rho)u_s + \tau(h_o + u_s))}{(h_o h_i + u_s h_o + u_s h_i)} \quad \text{(A-3)}$$

Values of h_i and u_s are given in the ASHRAE Handbook of Fundamentals. For this report, the values of h_i and u_s are

$$h_i = 8.29 \text{ W/m}^2 \cdot \text{K}$$

$$u_s = 7.84 \text{ W/m}^2 \cdot \text{K}$$

h_o is a function of the wind speed and is also given by ASHRAE, for smooth glass surface, as

$$h_o = -0.0355 \cdot W_s^2 + 3.327 \cdot W_s + 8.233 \quad \text{W/m}^2 \cdot \text{K}$$

where W_s = wind speed in m/s.

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☐ Document describes a computer program; SF-185, FIPS Software Summary, is attached.

11. ABSTRACT (A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here)

The thermal comfort conditions in a direct gain cell of the NBS/DoE passive solar test facility were analyzed in accordance with the criteria specified in the recently revised ASHRAE Comfort Standard 55-1981, using test data collected during the month of October 1981 and the month of January 1982. It was found that the daytime operative temperature (as measured by the black globe temperature sensors) in an area near the large south glazing exceeded the upper boundary of the ASHRAE comfort envelope by a large amount in a clear day during both the thermal transition month of October and the cold winter month of January. The generally accepted method of computing the mean radiant temperature based only on the interior surface temperatures was found to produce large errors. The reflected solar radiation from the interior surfaces and the snow covered ground was believed to play a significant role on the measured black globe temperature and should be included in the computation of the mean radiant temperature for a space with large glazed areas.

12. KEY WORDS (Six to twelve entries; alphabetical order; capitalize only proper names; and separate key words by semicolons)

ASHRAE Comfort Standard 55-1981; black globe temperature; comfort envelope; direct gain room; operative temperature; passive solar test facility; solar radiation; thermal comfort

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