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Analysis of Three Perimeter Heating Systems by Air-Diffusion Methods

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ABSTRACT

The end result of a good HVAC system is a suitable thermal environment for the occupants of a conditioned space. ASHRAE Standard 55-1981 establishes measurable limits for the level of drafts, temperature gradients, and radiant temperature asymmetry necessary to assure an acceptable thermal environment. Tests have been conducted to evaluate a space against requirements based on this standard.

A test chamber was built to investigate the thermal environment in a perimeter office under typical winter conditions. Sufficient measurements were made to compile detailed draft and temperature data throughout the room. With graphical analysis of the data, performance can be shown to respond to selected input variables. Tests of the type performed could be used by a building designer to make selective choices in the perimeter heating design.

Additional tests were conducted to evaluate radiance effects. These tests compare alternate methods of determining heat loads and MRT values in conjunction with the requirements of Standard 55-1981.

INTRODUCTION

By using recently developed testing procedures and equipment, in conjunction with the new ASHRAE standard on thermal acceptibility, three different methods of heating a perimeter office space were evaluated. 1-3 These tests were conducted at a constant heating load using (1) all air, (2) ceiling radiant heat, and (3) below-window radiant heat to offset a cold glass wall. Multiple-point airspeed and temperature profiles were developed and analyzed by the use of a draft equation employing a "thermally optimum velocity." This technique allows for computer analysis of the data over the expected range of space operating temperatures and yields graphs relating certain thermal environmental variables for varying input conditions. These graphs give a visual indication of the limiting factors with each design variable for each type of heating method. 1

Additional research was conducted to compare alternate methods of determining mean radiant temperatures, of comparing operative temperature sensors to established measurement methods, and of investigating alternate methods of determing heat transfer across a surface. (The operative temperature is defined by ASHRAE Standard 55-1981 as the average, weighted by respective heat-transfer coefficients, of the air and mean radiant temperatures. 3

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TEST DESCRIPTION

A 15 ft (4.5 m) x 15 ft (4.5 m) test room was built inside a larger temperature-controlled space. One wall of this test room is an insulated, movable cold wall simulating the perimeter wall of an office building. This wall contains 6 ft (2 m) of single-pane vertical glass which extends from 2 ft (0.6 m) above the floor to within 1 ft (0.3 m) of the ceiling. (This wall was described in an earlier paper by the author.) 4

Three commercially available modular linear diffusers were placed in the ceiling in a line parallel to the cold wall and at a distance of 3.5 ft (1 m) from the wall. The performance parameters for this diffuser are listed in Appx A. Three 2 x 4 ft (0.6 x 1.2 m) radiant panels composed of a resistive film over 0.5 in. (13 mm) thick fiberglass boards in a metal frame, were installed in one of two locations: either below the glass or in the ceiling between the glass and the air diffuser (Fig. 1). Three flat-lensed light fixtures are also located in the ceiling. The remainder of the ceiling is composed of 1.5 in. (38 mm) glass-cloth-faced fiberglass ceiling panels. The concrete floor is carpeted.

Conditioned air is supplied through flexible ducts to each ceiling air diffuser. The air temperature is controlled by heating slightly cooled air with either a proportionally controlled or operative-sensor-controlled electric resistance heater. 5 Air volumes delivered to the space are determined by measurement of discharge velocity and the data in Appx A. Power to the electric resistance radiant panels is controlled by either the operative temperature controller or with a variac on the 220 volt AC line.

EQUIPMENT DESCRIPTION

Several electronic devices were used during these tests to evaluate new instruments against established test instruments and to validate load and radiance equations:

1. Six constant-temperature, multidirectional, temperature-compensated anemometer transducers are positioned on a movable tree at intervals of 1 ft (0.3 m) from the floor. An additional probe is located 0.5 ft (0.2 m) above the floor. A radiant shielded thermistor temperature probe is located at each anemometer position. A 2 in. (51 mm) diameter thin-walled globe, painted a flat gray color with a thermistor located at its center, is located 3 ft (0.9 m) above the floor, also on the tree. These feed information into a programmable data-acquisition device, which converts the signals to appropriate engineering units against individual calibration constants and curves. It also averages rapidly scanned data at selected time intervals. The equipment and calibration are described in detail in an earlier paper by the author. 2
2. A "Comfy Test" meter, designed for direct measurement of combined factors affecting the thermal environment and human comfort, is used as a baseline data instrument. Several papers have been published comparing the response of this device to human response. 6-7
3. An operative temperature sensor/controller 5 is used both to quantify the environmental conditions at a point and to control the heating elements in several tests.
4. Three commercially available heat flowmeters, consisting of thin metal plates surrounding a thermopile embedded in a plastic material, about 1/4 in. (6 mm) thick, are used as one means of determining the load imposed on the test space. These are placed in contact with the cold glass, the radiant panels, the light fixtures, etc., for comparison to other means of determining load factors. These are calibrated in thermal test devices meeting the requirements of ASTM C 518 for meter factor determination. The meter factors are placed into the data-acquisition device so that readouts are in engineering units.
5. A commercially available "heat flow" gun was investigated as an alternate load determinant. This device does not have an electronic output and therefore is recorded manually at intervals and compared to the averaged scanner output.

TESTS CONDUCTED

A variety of heating tests were conducted to determine the design limitations in providing an acceptable level of temperature variation in the test room. One major concern when heating a space within a building is to control stratification. The tests conducted employed different strategies to control the stratification to within limits considered acceptable by ASHRAE Standard 55-1981 and to yield acceptable values for the Air Diffusion Performance Index, or ADPI. For the "Comfy Test" meter and for ASHRAE Standard 55-1981 calculations, some assumptions were made about the occupants of the test zone. Their activity was assumed to be sedentary, at 0.8 to 1.2 Met; their clothing was assumed to be 1 Clo; there was no sunlight in the zone; and the relative humidity was set at 50% RH. 3 The tests are divided into three groups;

1. All air tests; no heat provided by the panels

2. Radiant panels in ceiling; heat provided by panels and/or by air

3. Radiant panels in cold wall; heat provided by panels and/or by air

The tests were conducted by measuring point temperatures and airspeeds in a single plane through the room along a line representative of the air diffusion in the space(Diagram 2). For the first series of tests, used to quantify loads and measurement techniques, no furniture was present in the test room. For the actual series of performance tests, a desk was placed in the zone to obtain more realistic draft conditions (Fig. 3). Numerous readings of all surface temperatures as well as other instruments were obtained during each test. Each test was conducted under steady-state conditions, typically requiring two hours to stabilize between runs. Each required about 45 minutes to conduct. From the data, a modified ADPI, called ADPI* is calculated at 69 degrees F (19 degrees C). 1 (The Air Diffusion Performance Index, ADPI, is a single-number rating for the level of drafts and temperature variations in a zone (9). This quantity is determined by the percentage of points meeting criteria for acceptable calculated draft temperatures. The criteria were developed around comfort data of the 1960s and assume an ideal point airspeed of 30 fpm (0.15 m/s). The ADPI* equation utilized for this analysis, however, uses a variable ideal airspeed based on the temperature at each point (see test results). Selected tests are also depicted by ADPI* versus mean air temperature graphs. A sample data sheet is included, as well as several of these graphs, in the appendixes.

Sixty separate temperature/velocity profiles were determined using the "Proposed Method of Test of Air Distribution Performance." 1 These were conducted over a range of heating demand loads. The results are summarized according to the mode of heating. For all tests, the air inside the cold wall cavity was maintained at about 20 degrees F (-6.6 degrees C). This resulted in a load of about -365 Btuh/ft (-325 W/m) of wall (loads reported are per horizontal linear dimension). The MRT at each test position was calculated from the globe temperature measured at the point 3 ft (0.9 m) above the floor. The globe temperature was combined with the airspeed and temperature at that point, per ASHRAE Standard 55-1981 and the test procedures, 1 and the point MRT was calculated. The operative temperature and optimum velocity were then calculated for that test position according to the test procedures. 1

First Series of Tests

The first series of tests investigated calibration factors previously determined for the heat flowmeters. By comparing measured temperatures and the amount of heat supplied to the space, the output of the heat flow meters was analyzed. The initial series was also used to perform several additional tests:

1. Heat Flow Gun - The output of the heat flow gun was compared to the heat flowmeter values for the various surfaces in the room.

2. Globe Sensor Time Response - The globe sensor, mounted on the tree, was moved from a stable condition at one point to another close to the glass. The output was monitored over time and recorded.

3. MRT Calculation - ASHRAE Standard 55-1981 describes two methods of determining the value of mean radiant temperature at a point. One method is to correct a globe thermometer reading and combine that with the air temperature at a point to determine MRT. The second involves detailed angle factor calculations and measured surface temperatures throughout the room. Both were employed on data acquired during the initial series of tests.

4. Experimental Operative Temperature Sensor - The output of an experimental operative temperature sensor 5 was compared to values calculated from globe and air temperature data according to the equations in the referenced test procedure, which were derived from ASHRAE Standard 55-1981.

Additional Tests

Fifty tests were conducted for the three types of heating systems operating under steady-state conditions. Additional heat was provided by the diffusers in some panel heating tests. A slight mismatch of the test room and the larger room temperatures results from the difficulties of determining the average room temperature when significant stratification is occurring (see "discussion" section). Heat supply from the air diffusers is calculated from air volume and temperature data. This is compared with the other load factors and any mismatch is reported as the "apparent floor heat load".

1. All-Air Tests - Twenty-two tests were conducted at steady-state conditions over a range of -280 to -420 Btuh/ft (-269 to -405 W/m) of wall heat demand load. A calculated apparent floor heat load of from 0 to 17 Btuh/sqft (0 to 53 W/sqm) resulted with the remainder of the heat supplied by the three ceiling air diffusers. The diffusers supply air in two directions, toward and away from the glass wall.

2. Below-Glass Panels - Eleven tests were conducted with panels located below the glass (Fig. 1). These tests were conducted over a range of -350 to -425 Btuh/ft (-366 to -409 W/m) of wall heat demand load. A calculated apparent floor load of from 0 to 17 Btuh/sqft (0 to 53 W/sqm) resulted, with the remainder provided by the radiant panels below the glass and by air from the ceiling diffusers.

3. Ceiling-Mounted Panels - Eight tests were conducted with ceiling-mounted radiant panels (Fig. 1). These tests were conducted at a load of about -350 Btuh/ft (-336 W/m) of wall heat demand load and no apparent floor load. The ceiling panels and air diffusers supply all the required heat.

TEST RESULTS AND DISCUSSION

Preliminary tests

The first ten tests were conducted to evaluate six different instrumentation factors (for use in quantifying later tests). In each test, the test room was allowed to come to equilibrium to insure that steady-state measurements were being made. Data were then collected and analyzed for several heat source conditions:

1. Heat Flowmeter Output - Meter factors were determined from thermal test equipment (ASTM C 518). Comparison with factors calculated from room data shows a fair agreement. Room data calculations include "outside air" temperatures, film coefficients calculated from measured airspeeds at both surfaces of the glass, ASHRAE Handbook 9 thermal transmittance values, and test room air temperatures in the vicinity of the glass. Slight temperature differences between the test room and the outer rooms were unavoidable. The effect of this was considered as an

interior load upon the test zone. This effect is utilized during the environmental tests to allow for various rates of heat supply by the active heating components with a fairly constant cold wall load. (This effect is valid only for small differences between the temperature of the test and outer rooms.)

2. Heat Flow Gun - The heat flow gun is used by first calibrating the unit against a thermally "neutral" reference surface, then pointing it at the surface to be measured. This device reads in units of Btuh/sqft. It was found that good correlation exists between the gun and the heat flowmeter only when a reference surface is chosen that is at the same temperature as the air surrounding the surface to be measured. When significant stratification is present in the test room, this requires a careful choice of reference location. When properly referenced, the gun offers a reasonably accurate instantaneous heat flow measurement on any surface in the room.

3. Globe Sensor Time Constant - The transient globe response is illustrated in Fig. 6. While slow to respond, the time constant is a composite of many component factors and may not be exactly exponential. During these tests, the measured globe and calculated mean radiant temperatures varied only by about 2 degrees F (1 degree C) throughout the test room and differed little from the air temperature at each point. Data taken at three-minute intervals at each test position showed the same globe temperature as temperatures averaged over ten minutes at each position. To get a significant change, it is necessary to place the globe very close to the glass.

4. MRT Analysis - Both MRT and radiant asymmetry were calculated for several runs:

--MRT values were computed by angle factor analysis for runs 4, 5, 6, 7, and 9 and compared to those determined by globe thermometer.

Run Number	4	5	6	7	9
Calculated MRT, degrees F	68.4	68.7	68.1	68.5	70.4
Measured MRT, degrees F	68.4	68.1	68.2	68.6	69.6
Deviation, degrees F	+0.03	+0.57	-0.07	-0.14	+0.76
Average Deviation	--+0.29 Degrees F (0.62%)---				

These values indicate that the globe data is an accurate and repeatable measure of MRT, if sufficient time is allowed for equilibrium, at least with tests of this magnitude of radiant effects.

--Asymmetric radiation in the vertical direction and horizontally from the glass wall toward the rear of the room were calculated from the "Occupant Position" (Fig. 2) according to ASHRAE Standard 55-1981. These calculations involve a detailed analysis that includes the location of each light, panel, or window.

Run Number	4	5	6	7	9
Delta Pr, Horizontal	6.83	7.84	9.59	8.32	3.07
Delta Pr, Vertical	-3.77	-2.36	-0.83	-2.10	-4.15

(ASHRAE Standard 55-1981 places an upper limit of acceptability for vertical asymmetry at 9 degrees F, and for horizontal asymmetry at 18 degrees F.)

The heat source for the above tests was as follows;

Run 4, All Air
Run 5, Air Ceiling Panel
Run 6 & 7, Ceiling Panel
Run 9, Window Panel

5. Operative Temperature Sensor Controller - This experimental sensor performs as expected. 5 The data from the sensor tend to read 1.3 degrees F (0.7 K) higher than that from the thermistors. This is probably due to a basic calibration difference as it is apparent for operative, globe, and air temperature data. The unit adequately controls both air temperature from the diffusers and panel temperatures in both operative and air temperature mode. Selecting a suitable location for the sensor, however, is the major difficulty in applying the device to an actual environment. In many tests a nonuniform environment exists; while it may be ideal at the sensor location, it may be very unsuitable at other locations.

6. Comfy Test Meter - This device correlates well with both the globe/air-temperature-calculated operative temperature and the operative temperature sensor-predicted comfort levels. For all tests, the test room temperature is adjusted to read minimum discomfort (+/-3 PPD) (Percent Persons Dissatisfied) on the "Comfy Test" meter. Again, however, the location of the sensor (which was adjacent to the operative sensor) is critical when the environment is nonuniform.

<u>Environmental Tests</u>

The results of the 50 environmental tests are summarized in Appx B, grouped by the major source of heat to the test room. A number of graphs were prepared from the data acquired during these tests. These graphs were prepared from computer analysis of the temperature/velocity profiles measured during each run. The data from a run is manipulated by raising or lowering all the temperatures in the zone in incremental amounts and recomputing the ADPI*. As proposed, 1 ADPI* differs from the traditional ADPI calculation described in the <u>ASHRAE Handbook 9</u> by the inclusion of a variable airspeed value in the draft temperature equation. The "optimum thermal velocity," which is based on the airspeed limits of <u>Standard 55-1981</u> at fixed levels of clothing, activity, and humidity, is dependent on the operative temperature determined at that point. The higher the temperature at a point, the higher the set airspeed. The percentage of points meeting acceptable draft temperature limits (-3 to +2 degrees F) is therefore temperature sensitive. This method of data reduction should be valid over the expected range of space operating temperatures, assuming that the magnitude of the drafts will remain fairly constant for a given room-discharge delta T. The reported loads are still the same when the temperatures are adjusted, which simulates that a change has occurred in the outside air temperature of the same magnitude as the change in mean space temperature. (While not a problem with this study, this would pose some problems if a design outside air temperature were a part of the analysis.) The resulting data are as ADPI* versus mean temperature. The graphs allow a comparative analysis of the effects of design variables and limits of acceptable thermostat settings.

1. All-Air Tests - Tests with all the required heat supplied from the ceiling air-diffusion system indicate that the control of stratification is of prime importance. Selecting the correct levels of air volume and discharge temperature can control stratification to within the limits considered acceptable by <u>ASHRAE Standard 55-1981.</u> While all-air heating systems have the advantage of flexibility in office arrangements, definite performance limits are apparent. In order to achieve ADPI levels greater than 80%, this particular diffuser has an apparent maximum allowable discharge to room delta T of 10 degrees F (5.6k). If the perimeter wall meets the design requirements of <u>ASHRAE Standard 90-1981,</u> the maximum heat loss through a typical wall will seldom exceed -250 Btuh/ft (-240 W/m) of wall. When combined with the typical internal heat gain from occupants, lights, etc., which are typically on the order of 3 to 11 Btuh/sqft (11 to 33 W/sqm), the demand on the heating system will seldom require more then 8 degrees F delta T (4.4k) at 20-25 cfm/ft of wall (30 to 39 l/s.m). The effect of night setback will have to be evaluated to determine start-up heat loads, which may be higher.

2. Ceiling Radiant Panels - The use of ceiling radiant panels requires a significant quantity of conditioned air to prevent stratification in excess of the ASHRAE allowable 5 degrees F (3 K). A complaint typically expressed by building occupants exposed to ceiling radiant panels is a sensation of excessive radiant heat on the tops of their heads. This has not been found with these tests and can indeed be disproved by plane radiant asymmetry calculations. What probably occurs is excessive air temperature stratification, aggravated by slight radiant asymmetry. By supplying sufficient quantities of conditioned air, the tendency to stratify can be overcome. Unfortunately, these required air supply levels are far in excess of minimum ventilation levels. Supplying cool air can be used to limit the level of stratification at lower volumes but will result in a certain degree of energy waste.

3. Panels Below The Window - No stratification occurred during any of the tests with panels located below the window. Placing the radiant panels below the window would seem to be the best choice, except for interference with furniture along the glass. Panels in this location must have higher abuse-resistance and lower maximum temperature limits than ceiling panels. Also, the loss of heat through the back side of the panels must be included in any analysis of energy use.

Five graphs of ADPI* versus temp. have been included as figures 4 to 8. These illustrate the effect of different combinations of air volume and temperature with differing designs. Graphs of this type can be used to establish operation limits for a HVAC design.

<u>CONCLUSIONS</u>

1. The ADPI* computations and analysis, employing the "Thermally Optimum Velocity," 1 can be used to evaluate design limitations of perimeter heating systems over a range of space temperatures and operating conditions. Comparison of the detailed draft data obtained during these tests to the acceptance limits of <u>ASHRAE Standard 55-1981</u> shows there is good correlation between the ADPI* concept and the limits of thermal acceptability of the Standard. ADPI* values in excess of 80% are required to insure that most points meet the ASHRAE limits (at fixed levels of clothing, activity, and humidity). While the test procedure employed requires significantly more measurement points than the ASHRAE Standard, it should yield more repeatable and representative data of the test area.

2. The instrumentation utilized for these tests, including the 2 in. (51 mm) globe sensor and multidirectional anemometer, have been shown to give repeatable and accurate results. The time constant of the globe sensor, while longer than the scan time at each position, has not posed any problems with the systems evaluated here. Systems with greater radiant asymmetry will require a longer time of scan at each position. All the anemometer probes remained within calibration limits (+/-5fpm (+/-0.025m/s)) for the duration of testing.

3. An operative sensor can be effectively used to control space heating elements and maintain a satisfactory thermal environment at the sensor location. Spatial uniformity throughout the zone must be maintained by proper design of the entire system.

4. Analysis of globe and calculated Mean Radiant Temperatures throughout the space indicates that there is little drop in MRT near the glass with any of the three systems evaluated. A lower MRT had been expected near the glass with the all-air system. What was found was that the stratification of warm air at the ceiling elevates the temperature of the ceiling boards, causing them to radiate heat into the zone and offsetting the effect of the cold glass. The operative temperature calculated for each test position (at the 3 ft (0.9 m) test point) was never more than 0.2 degrees F (0.1 K) different from the uncorrected globe temperature reading (see attached sample data sheet). This suggests that, at least for tests of this magnitude of radiant effects, a 2 in. (51 mm) globe thermometer may be used directly as a measure

of operative temperature. High airspeeds or direct sunlight may increase this difference. (The data also suggest that a good measure of the comfort level may be obtained without any measurement of the radiant temperatures in the zone, at least when no sunlight is present.)

5. The question of energy use must still be answered. Measurements of air temperature and airspeeds at the glass and heat demands at varying airflows show there is little relationship between airflow rates and heat loss through the glass at the levels of this study. A thin air film moving downward at over 200 fpm (1.0 m/s) was measured for all three conditions, regardless of air volume or temperature. Using smoke traces, this film appears to be less than 0.5 in. (13 mm) thick and is apparently tied to the glass surface temperature. Measurements of radiative effects with panels are difficult to perform due to problems in quantifying heat transmission differences between transparent glass and opaque heat meters. Actual energy use tests must be performed to quantify the energy parameters more accurately. Within the limits of the measurement systems employed for these tests, differences of energy use have been determined to be minimal.

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TABLE 1
Modular Air Diffuser Air Delivery Performance

Volume, CFM	Slot Velocity, FPM	Static Pressure, in. H2O	Throw, Ft (to 50 fpm)
50	550	0.020	4.0
75	875	0.050	6.5
100	1100	0.080	9.0
125	1375	0.130	12.5
150	1675	0.195	15.5
175	1925	0.270	18.5
200	2200	0.350	22.5

TABLE 2

Data Summary, Panels Pelow Window

Run Number	CFM/ sq.ft	Delta T (Room Discharge), Deg. F	Cold Wall Load, BTUH/ft	Panel Heat, BTUH/ft	Apparent Floor Heat Load, W/sq.ft	ADPI* (at 69 deg. F) %
28	1.35	+3.2	-425	288	1.3	96
30	1.35	+2.0	-350	376	-1.3	98
45	1.2	+6.4	-395	99	+3.4	92
46	1.2	+6.2	-396	119	+3.0	92
44	1.2	+5.5	-392	57	+4.4	90
47	1.2	+5.1	-391	300	-0.15	96
48	1.2	+3.8	-376	396	-1.8	96
32	1.1	-0.3	-375	428	-0.9	100
33	0.61	+0.9	-375	424	-1.3	100
49	0.42	-1.9	-340	389	-0.7	100
34	0.36	+0.2	-375	404	-0.56	100

Data Summary, No Panels

Run Number	CFM/ sq.ft.	Delta T (Room To Diffuser), Deg. F	Cold Wall Load, BTUH/ft	Apparent Floor Heat Load, W/sq.ft	ADPI* (at 69 deg. F) %	Comments
25	1.5	+15.0	-425	1.2	73	
39	1.5	+13.3	-426	2.0	80	
38	1.5	+10.0	-415	3.4	86	
37	1.5	+8.5	-410	4.0	88	
59	1.5	+8.5	-337	2.6	88	
60	1.5	+5.5	-350	4.2	78	Floor Heater (600 W)
61	1.5	+4.9	-353	4.6	79	Floor Heater (600 W)
24	1.3	+19.0	-425	0.5	71	
21	1.15	+22.2	-425	0.2	71	
20	1.15	+22.0	-425	0.2	70	
58	1.10	+11.9	-345	2.7	69	
36	1.04	+19.2	-425	2.0	71	
19	1.0	+24.9	-425	0.4	69	
18	1.0	+25.7	-380	0.0	67	
17	1.0	+26.5	-425	0.0	67	
40	1.0	+16.7	-413	2.8	68	
11	1.0	+14.6	-280	0.9	61	
41	1.0	+13.3	-396	3.5	71	
42	1.0	+12.0	-396	3.9	65	
43	1.0	+9.7	-408	4.9	82	
14	0.75	+19.8	-280	0.8	72	
15	0.54	+21.6	-280	1.8	72	

Data Summary, Ceiling Panels

Run Number	CFM/ sq.ft	Delta T (Room Discharge), Deg. F	Cold Wall Load, BTUH/ft	Panel Heat, BTUH/ft	Apparent Floor Heat Load, W/sq.ft	ADPI* (at 69 deg. F) %
54	1.60	+2.3	-364	452	-2.8	91
55	1.5	-0.7	-348	416	-1.3	89
53	1.15	+1.4	-367	410	-1.3	84
56	1.15	-0.4	-351	389	-0.7	93
52	0.75	+3.0	-371	383	-0.8	69
57	0.60	+0.6	-350	+357	0.0	73
50	0.41	0.0	370	376	0.0	69
51	0.0	0.0	-322	358	-0.7	65

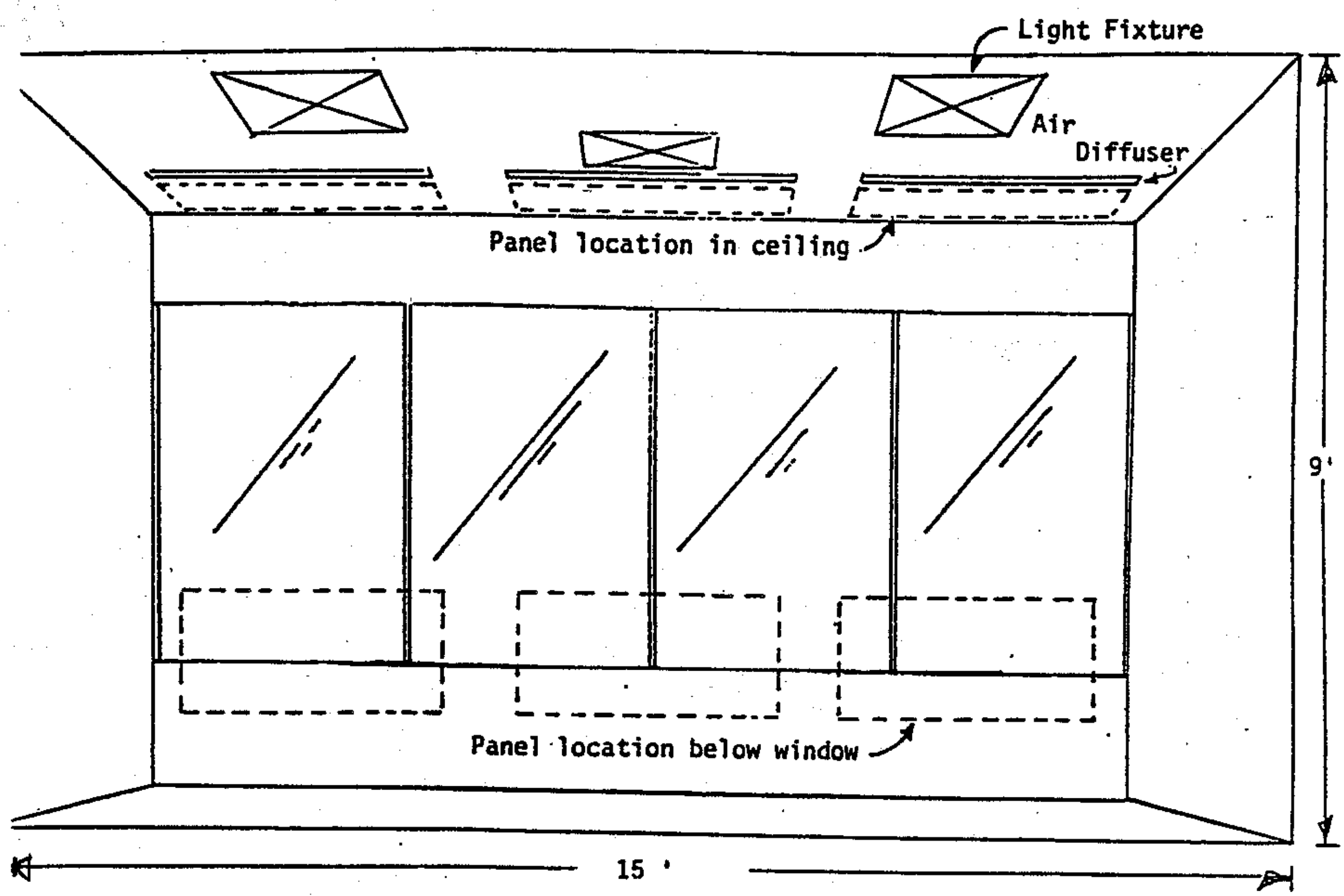

Figure 1. Room details

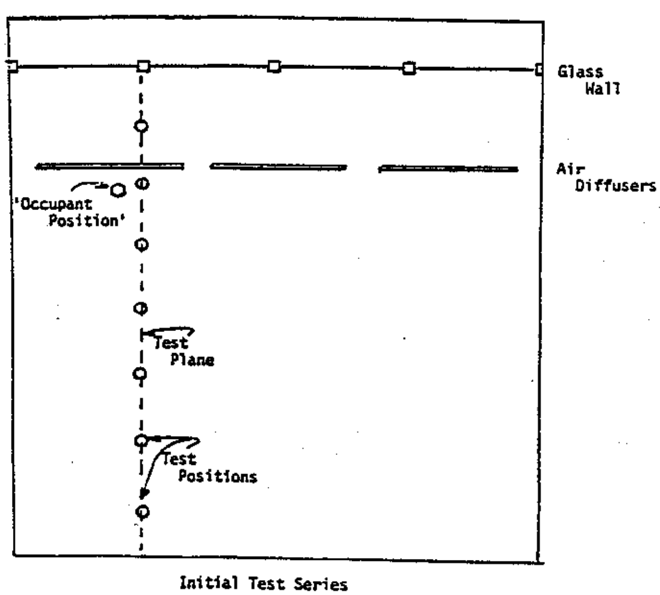

Figure 2. Test planes, initial tests

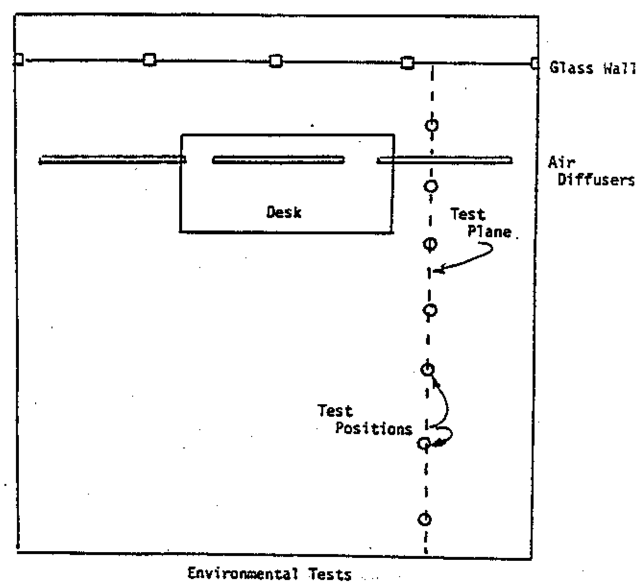

Figure 3. Test planes, environmental tests

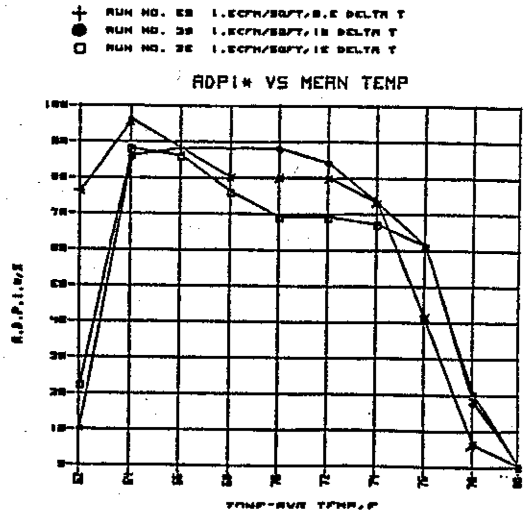

Figure 4. All-air system, discharge to room Delta T at the highest level of airflow tested, 1.5 cfm/sq. ft or 23 cfm/ft of wall, indicating 10 degrees. Delta T is probably the maximum allowable temperature difference.

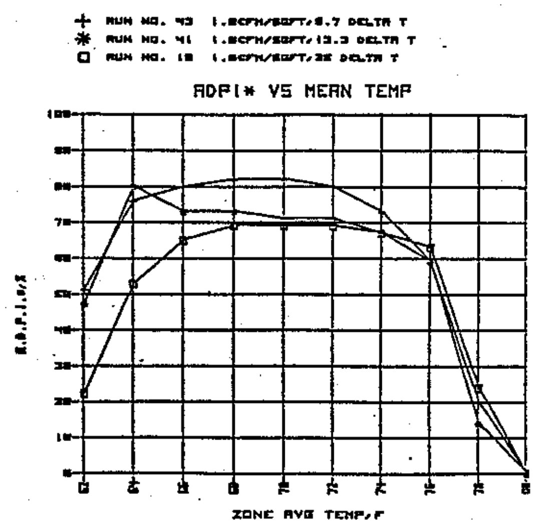

Figure 5. All-air system, is similar to Fig. 4, but at 1.0 cfm/sq. ft or 15 cfm/ft of wall. This indicates that this is probably the lowest acceptable airflow rate.

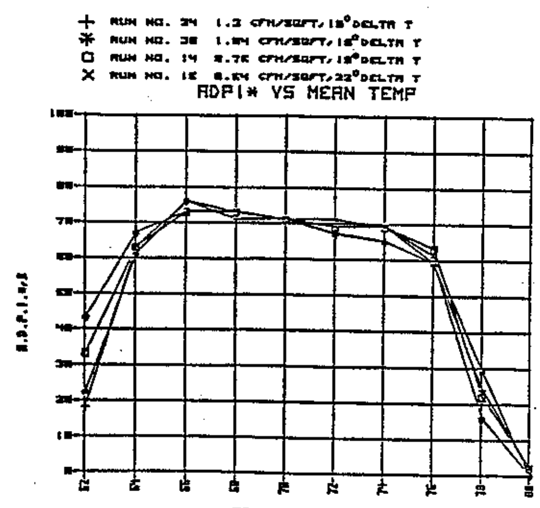

Figure 6. All-air system, illustrates the fact that at higher Delta Ts, the airflow rate does not seem to be a dependent variable.

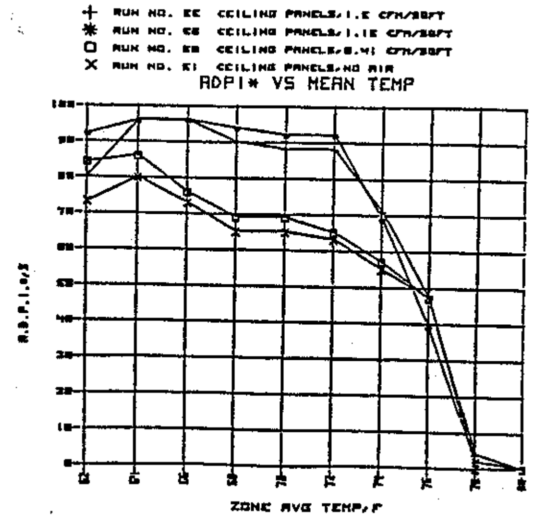

Figure 7. With ceiling panels, high airflow is required to prevent stratification with the lower limit at about 1.0 cfm/sq. ft. Utilizing colder air allows lower airflow, but increases the heating requirement of the panels.

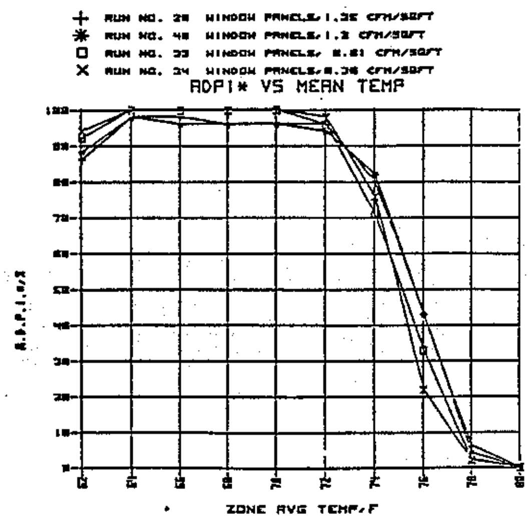

Figure 8. With panels located below the window, the air flow rate is not a variable, and excellent spacial uniformity is achieved for all tests.

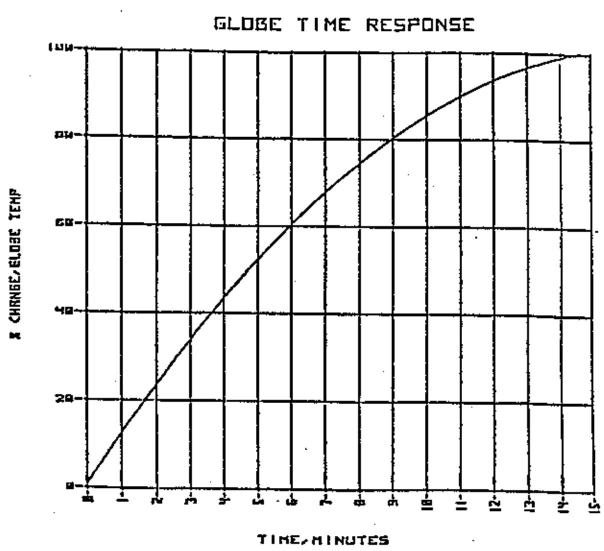

Figure 9. Globe time response.