Title
Testing and optimizing the performance of a floor-based task conditioning system

Authors
Bauman, Fred
Arens, Edward A
Tanabe, S.
et al.

Publication Date
1995

Peer reviewed

Testing and Optimizing the Performance of a Floor-Based Task Conditioning System

F. Bauman, E. Arens,
S. Tanabe, H. Zhang, and A. Baharlo

CEDR-R06-94

Energy and Buildings 22 (1995) 173-186

Center for Environmental Design Research
390 Wurster Hall
University of California
Berkeley, California 94720

Submitted to *Journal of Energy Efficiency*

TESTING AND OPTIMIZING THE PERFORMANCE OF A FLOOR-BASED TASK CONDITIONING SYSTEM

F.S. Bauman, P.E.   E.A. Arens, Ph.D.   S. Tanabe, Ph.D.*   H. Zhang   A. Baharlo

Center for Environmental Design Research
University of California
Berkeley, California
CEDR-R06-94

March 2, 1993

ABSTRACT

During recent years an increasing amount of attention has been paid to air distribution systems that individually condition the immediate environments of office workers within their workstations. As with task lighting systems, the controls for these "task conditioning" systems are partially or entirely decentralized and under the control of the occupants. Among the primary types of task conditioning systems (floor-, desktop-, and partition-based), floor-based designs are the most common, having been widely developed and used in South Africa and Europe, and are now gaining acceptance in the United States.

This paper reports the results of recently completed laboratory measurements investigating the thermal performance of a floor-based task conditioning system. The experiments were performed in a controlled environment chamber configured to resemble a modern office space with modular workstation furniture and partitions. Tests were conducted to study the effects of supply volume, supply temperature, supply direction, and heat load levels in the space. In addition to detailed temperature and velocity measurements, a new skin-temperature-controlled thermal manikin was used to evaluate the non-uniform thermal environments produced by the floor-based system, in both sitting and standing positions. Under the low to intermediate supply volumes investigated in this study, the overall performance of the floor-based system resembled that of displacement ventilation systems. Primarily by controlling supply volume the floor-based system can be operated to maintain acceptable thermal comfort in the occupied zone of the building, while at the same time taking advantage of the temperature stratification inherent in displacement flow to achieve savings in space conditioning energy use.

* Department of Human Environmental Engineering, Ochanomizu University, Tokyo, Japan

INTRODUCTION

During recent years an increasing amount of attention has been paid to air distribution systems that individually condition the immediate environments of office workers within their workstations. As with task lighting systems, the controls for these systems are partially or entirely decentralized and under the control of the occupants. Typically, the occupant has control over the speed and direction, and in some cases the temperature, of the incoming air supply. The systems have been variously called "task conditioning," "localized thermal distribution," and "personalized air conditioning" systems. These task conditioning systems provide supply air and (in some cases) radiant heating directly into the workstation, either through a raised access floor system, or in conjunction with the workstation furniture and partitions.

The primary types of task conditioning systems at this time can be categorized as floor-based, desktop-based, and partition-based. Floor-based systems are the most widely used; the earliest such systems were extensively developed and used in South Africa and Europe [Sodec 1984, Spoormaker 1990, Sodec and Craig 1991], and are now gaining acceptance in the United States [Shute 1992]. In the floor-based system that is the subject of this paper, air is drawn from an underfloor plenum by local variable speed fans and delivered to the space through floor-level supply grills. The other most common floor-based design does not use local fans, but rather relies on the central air handler to force the supply air through the sub-floor plenum and into the space through floor-level grills.

Task conditioning systems have the potential to affect many of the ways modern offices perform, as described briefly below.

1. The thermal comfort of the occupants is perhaps the area of greatest potential improvement in that individual differences or preferences can be accommodated. In fact, a recently completed survey (60 respondents) of industry perspective on task conditioning systems found that the provision of individual control was considered to be the most attractive feature of these systems [Bauman et al. 1992]. However, the comfort is obtained by imposing environments that are sometimes thermally asymmetrical, with air movement and radiation directed on some parts of the body and not on others. In order to optimize such system designs, new understanding is needed of the thermophysiological effects of such asymmetrical thermal environments.

2. Ventilation efficiency and air quality also have the potential to be improved over conventional uniformly-mixed systems, in that the fresh supply air can be delivered at breathing level and near the occupant.
3. Energy use can be raised or lowered depending on system design and operation [Heinemeier et al. 1991, Seem and Braun 1992, Bauman et al. 1992]. In this regard, the most important aspect of installing and operating an energy efficient task conditioning system is the level of sophistication with which it is integrated with the design and operation of the building's central HVAC system. The influence of occupancy sensors can be particularly strong; by shutting off individual workstations when they are unoccupied, substantial savings can be realized. It is also possible that energy savings can be obtained by conditioning only the smaller volume of the occupied workstations or by allowing some amount of properly controlled thermal stratification in the space. Offsetting these advantages are inefficiencies associated with the small electric motors powering the fans.
4. Occupant satisfaction and productivity can also be increased as a result of improved thermal comfort and control over the environment. The financial implications of such improvements have the potential to be very large. However, proving and quantifying such effects is a difficult undertaking.

Should there be substantial improvement in any of the above areas, the task conditioning technology is likely to grow prominently.

This paper reports the results of recently completed laboratory measurements investigating the thermal performance of a floor-based task conditioning system and focusing on how to optimize its performance in terms of both thermal comfort and energy efficiency. The floor-based system and the background for the present study are described below.

PRESENT INVESTIGATION

The floor-based task conditioning system consists of individual floor supply modules designed to be incorporated into a raised access floor system. Each module, measuring 2 ft by 2 ft (0.6 m by 0.6 m), can be located at any position simply by exchanging it with a solid floor panel of equal size. Figure 1 shows a cutaway diagram of the floor supply module. A fan/motor assembly draws air from the sub-floor plenum and supplies it to the room through four 5-in. (127 mm) diameter discharge grills. The grills are molded of durable, fire-resistant polycarbonate. Individual vanes are inclined at 40° from vertical and each grill can be rotated 360°, allowing office workers to control the direction of air supplied from the module. Under normal operating conditions, air is supplied by a conventional air-handling unit (AHU) to a sub-floor plenum maintained at very nearly the same pressure as the conditioned space. By using a rotary speed control knob, building occupants can control the quantity of air delivered by the floor supply module to the space over the range of 90 to 190 cfm (40 to 90 L/s), when the fan is turned on, or they can turn the fan off. Over this same range of airflows the maximum outlet velocity measured at the grill varies between 6 and 19 fps (2 and 6 m/s). Supply outlet temperatures are typically maintained above 63°F to 64°F (17°C to 18°C) to avoid overcooling nearby occupants. Air is normally returned from the space near ceiling level, producing an overall upward movement of air in the room.

Previous laboratory experiments by the authors and colleagues investigated the thermal performance [Bauman et al. 1991, Arens et al. 1991] and ventilation performance [Fisk et al. 1991] of this same floor-based task conditioning system. Most of the tests were done under cooling applications, the dominant space-conditioning requirement in the interior zones of office buildings. One of the major findings was that under low flow conditions the overall ventilation performance of the room resembled that of displacement ventilation systems. In a displacement ventilation system, cool air is supplied at very low velocity through large-area supply devices near floor level, and is returned at ceiling level. The upward movement of air in the space is driven by natural convection from the heat sources, resulting in vertical temperature stratification. This stratification, if properly controlled, can lead to reductions in the space cooling load by as much as 15% compared to conventional well-mixed ventilation systems [Shute 1992]. Displacement ventilation is characterized by the creation of two distinct horizontal zones with different airflow patterns. (1) The lower zone contains the cooler and fresher supply air which moves upward in a piston-like flow pattern. (2) The upper zone is relatively well-mixed and contains warmer air and greater concentrations of space contaminants. Under steady-state conditions, the two zones are separated by a horizontal plane, called the "stationary front" or "transition plane," whose height in the room depends on the relationship between the strength of heat sources in the space and the amount of supply air introduced at floor level by the ventilation system. Higher heat loads tend to lower the transition plane while increased supply volumes tend to raise it. Ideally, operating conditions for an optimized displacement ventilation system would allow the transition plane to be maintained at a height above the occupant's breathing level, allowing acceptable occupant comfort to be maintained (by limiting stratification), and improving air exchange efficiency and ventilation efficiency in the occupied zone and reducing space cooling energy use compared to conventional ceiling-based "mixing" ventilation systems.

Displacement ventilation systems have been widely and successfully used in Nordic countries, particularly in industrial applications. Unfortunately, when applied to office configurations with high heat load densities (> 20-30 W/m^2) and reduced ceiling heights compared to industrial buildings, displacement systems can not satisfy the cooling demand without imposing excessive thermal stratification in the space and overly cool conditions in the occupied zone. In response to these comfort limitations recent research has suggested that higher air exchange rates should be used [Sandberg and Blomqvist 1989]. Svensson 1989 has suggested that higher outlet velocities (although still relatively low) should also be used, producing greater mixing within the occupied zone and reducing the risk of comfort problems. By permitting greater mixing to occur, air exchange and ventilation efficiency rates will not be as high as those obtained with true displacement flow. However, if the mixing is confined to the occupied zone (up to heights of 2 m), some amount of improvement in the efficiency of air exchange and ventilation can still be realized.

Floor-based task conditioning systems, such as the one investigated in the current study, differ from displacement ventilation systems primarily in the manner in which air is delivered to the space: (1) higher air supply volumes are used, allowing higher cooling loads to be met, (2) air is supplied at higher velocity through smaller-sized supply outlets, and (3) local air supply conditions are generally under the control of the occupants, allowing comfort conditions to be optimized. Because of the more powerful air supply conditions with task conditioning systems, however, it is also possible to disrupt the stable upward flow pattern in the space. For example, previous laboratory experiments on this same floor-based system [Bauman et al. 1991, Fisk et al. 1991] demonstrated that when the floor supply module was operated at its maximum air supply volume, the cool supply jets were able to reach the ceiling, thereby minimizing stratification and producing close to uniform ventilation conditions. While this operating strategy of providing more complete mixing parallels that of conventional overhead air distribution systems, it reduces the potential improvements in energy and ventilation performance described above. A preferred operating strategy for task conditioning systems would limit the immediate impact of local air supply to room air conditions within the occupied zone of the space.

The laboratory experiments described in the current paper were carried out to study the thermal performance of the floor-based task conditioning system under low flow conditions. The primary objectives of these tests were to investigate how closely the low-flow performance resembles that of displacement ventilation, and to analyze how well the task conditioning system can be controlled to optimize the height of the stationary front. To evaluate the non-uniform thermal environments produced by the floor-based system, a new skin-temperature-controlled thermal manikin [Tanabe et al. in press] was used in both sitting and standing positions. Related measurements of ventilation and pollutant removal efficiencies for this same floor-based system under similar operating conditions were also performed using the same test facility, and are described elsewhere [Faulkner et al. 1992].

EXPERIMENTAL METHODS

Controlled Environment Chamber

The floor-based system was tested in a controlled environment chamber (CEC) measuring 18 ft by 18 ft by 8 ft, 4 in. (5.5 m by 5.5 m by 2.5 m) located in the Building Science Laboratory, Department of Architecture, University of California, Berkeley. The CEC is designed to resemble a modern office space while still allowing a high degree of control over the test chamber's thermal environment [Bauman and Arens 1988]. To study the performance of the floor-based system in an office environment, a modular workstation configuration shown in Figure 2 was installed in the CEC. Solid partitions (65 in. [1.65 m] tall) were set up to produce two small 60 in. by 75 in. (1.5 m by 1.9 m) workstations and one double-sized 120 in. by 75 in. (3.0 m by 1.9 m) workstation. The arrangement of the furniture, including desks, side tables, and overhead storage bins, is also shown in the figure. A floor supply module was installed as indicated in each of the three workstations During all tests, conditioned air was delivered to the room using sub-floor plenum supply to the floor supply modules. Air was returned from the room at the ceiling level through a single ducted perforated return grill.

The CEC air distribution system also allows a separately conditioned airflow to be provided within the plenum wall construction of the two exterior chamber walls and between the inner and outer window panes in the area called the annular space. During the tests reported here, airflow through the annular space maintained the temperature of the interior window pane at approximately the average indoor air temperature. By eliminating the external loads to the chamber, test conditions resembled those found in interior office spaces.

Heat loads were provided to simulate typical office load distributions and densities. Personal computers, containing small internal cooling fans, and monitors (≈90 W [310 Btu/h] total) were placed on each of the three desktops. Each workstation had a 75 W (256 Btu/h) task light above the desk. A second 75 W light bulb was located at the 1.1 m level near the edge of the desk to simulate the sensible heat load from a typical office worker. The experimenter and computer-based data acquisition system also added approximately 260 W (885 Btu/h) to the total room load during these tests. During some tests a higher heat load was produced by placing a 200 W (680 Btu/h) electric radiant heater on the floor under one or more of the desks to represent larger computer processing units.

Instrumentation

Detailed air velocity and temperature measurements within the test room were accomplished by using a lightweight sensor rig fabricated of aluminum tubing that allowed a vertical array of sensors to be positioned at desired measurement heights and moved around the room to map out a grid of selected measurement locations. At each location in the room, air velocity and temperature were measured at six heights: 4 in. (0.1 m); 2 ft (0.6 m); 3 ft, 7 in. (1.1 m); 5 ft, 7 in. (1.7 m); 6 ft, 7 in. (2.0 m); and 7 ft, 9 in. (2.35 m). The 0.1-m, 0.6-m, and 1.1-m levels correspond to recommended measurement heights for seated subjects, and the 0.1-m, 1.1-m, and 1.7-m levels correspond to heights recommended for standing subjects, as specified by ASHRAE (1992). Velocities were measured with spherical-element omnidirectional anemometers having a range of 0-700 fpm (0-3.5 m/s), and temperatures were measured with shielded thermistor temperature probes. All sensors were calibrated prior to testing. The measurement error of the anemometers was estimated to be ±4 fpm (±0.02 m/s) over the range 0 to 80 fpm (0 to 0.4 m/s), and ±8 fpm (±0.04 m/s) at higher velocities. The measurement error of the temperature sensors was ±0.2°F (±0.1°C). Temperature and velocity sensors were sampled 50 times over a 90-second measurement period.

Supply and return air volumes were monitored with a high-precision flow measurement setup consisting of a series of small-diameter straight-length pipes with pitot tubes mounted to measure the fully developed flow. The volume of supply air entering the test room through a floor supply module was checked and balanced with a flow hood. Additional temperature measurements in the ducts and room were monitored and recorded through the personal-computer-based direct digital control system for the CEC. Other details of the measurement equipment, sensor calibration, and data acquisition system are described by Bauman et al. (1991).

Thermal Manikin

Thermal manikins, originally developed to measure thermal insulation of clothing, are heated dummies that simulate the heat transfer between humans and their thermal environment. Commonly used thermal comfort indices such as Predicted Mean Temperature (PMV) [Fanger

1970] and Standard Effective Temperature (SET*) [Gagge et al. 1973] are based on the heat balance of the whole body, and as a result, may be less applicable to the evaluation of non-uniform thermal conditions. A realistically-shaped thermal manikin, however, is a useful tool for the direct measurement of heat exchange, and has been used in the current study to investigate the nature of the locally asymmetrical thermal environments produced by the floor-based task conditioning system. Previous human subject research on the effects of radiant asymmetry has shown that within limits, if overall thermal neutrality is maintained, the subjects experience no significant discomfort attributable to the asymmetry [McNall and Biddison 1970]. This indicates that the human body does a good job of averaging out thermal non-uniformity's. Earlier, Wyon and Sandberg (1990) have used a manikin to predict discomfort due to displacement ventilation.

Tanabe et al. (in press) describes the thermal manikin used in the current study, and also reviews the use of manikins in previous comfort research. Briefly, the outer surface of the manikin is heated by wound nickel wire (0.3 mm diameter) that covers the entire surface area at a maximum spacing of 2 mm. The manikin is segmented into sixteen individually-controlled body parts, so that under steady-state conditions, the local heat loss can be derived from the electricity consumption of each part. Table 1 lists the sixteen body parts and their respective surface areas. Exact details of how the surface temperature of the manikin is controlled to account for evaporative heat loss are described by Tanabe et al. (in press). A computer-based data acquisition system records the manikin's skin temperature and heat loss for each body part at one-minute intervals. Each data record represents the mean of 60 individual measurements; an average of five records (300 observations) was used for data analysis.

Test Procedures

As described earlier, this series of experiments investigated the performance of the floor-based system under low flow conditions. Air supply volumes below the minimum (non-zero) rate possible with a factory-shipped floor supply module (90 cfm per module) were obtained by partially obstructing the inlet on the underside of each module in the sub-floor plenum. All tests were performed under steady-state conditions in the test chamber controlled to represent an interior zone of an office building. To conduct these experiments, the room was preheated overnight and in the early morning using the electrical heat sources in the room until the expected average room temperature was reached. After completing the warm-up, the mechanical system was turned on, the floor-based system supply air volume, temperature, and direction were adjusted to their selected setpoints, heat sources were set at desired levels, and conditions in the room were allowed to further stabilize. During the tests, typical control of the supply air temperature entering the room through the floor supply grills was to within ±1°F (±0.5°C) of the desired setpoint.

During each test, thermal conditions were measured with the previously described sensor rig at 15 locations in the room, focusing on points in front of the desk in each workstation and also including a few points outside of the workstations (see Figure 3). The measurement locations were centered over one of the access floor panels, producing 2-ft (0.6-m) intervals between points in front of the desks. After completing the temperature and velocity measurements, the thermal manikin was positioned in a chair at the work location in workstation #3 (refer to Figure 2), turned on, and allowed to reach equilibrium with the environment (a minimum of one hour when first turned on and at least 30 minutes for each subsequent test). An open-mesh type chair was used to expose the whole body surface to the environment. The female manikin (Anne) was dressed in typical office attire (panties, bra, long sleeve cotton shirt, cotton pants, and shoes), which was measured to represent a thermal insulation value of 0.55 clo for sitting position and 0.73 clo for standing position. The higher effective insulating value in standing position results from a thicker air layer within the clothing and the slightly higher temperatures at head level for the standing manikin. After measuring the manikin in sitting position, it was repositioned in standing posture at the same location, allowed to stabilize again with the environment, and measured before ending the test. Pairs of tests that differed only by the direction of the supply grills were often performed consecutively on the same day.

Table 2 presents a complete list of the average conditions maintained during each of the 23 tests of the floor-based system. Three floor supply modules were operated during each test at approximately equal supply volume (balanced with a flow hood). The tests focused on variations in supply volume and heat load level because these variables strongly affect displacement flow characteristics in the room. Supply temperature was also varied, but only in combination with other parameters to maintain the average room air temperature around 77°F (25°C), in the middle of the ASHRAE-prescribed comfort zone [ASHRAE 1992]. The tests investigated the following ranges of test parameters: (1) supply volume per unit from 34 cfm to 98 cfm (16 L/s to 47 L/s); (2) supply air temperature from 61°F to 71°F (16.0°C to 21.6°C): (3) heat load per workstation from 256 Btu/h to 1430 Btu/h (75 W to 420 W); (4) average heat load density from 8 Btu/h·ft^2 to 16 Btu/h·ft^2 (25 W/m^2 to 50 W/m^2); (5) average room temperature from 75°F to 81°F (23.7°C to 27.3°C); and (6) supply grills directed toward the desk in the workstation, or supply grills directed inward toward the center for the floor module.

RESULTS

Due to the large amount of experimental data, a limited number of tests have been selected from Table 2 for presentation and discussion. The emphasis of the data presented here is on the local thermal conditions within each workstation. For brevity, average conditions at a given height in a workstation are defined as the velocity or temperature calculated by averaging the measured values from the measurement locations in front of the desk. Referring to Figure 3, average conditions in workstation #1 (WS#1) are based on the four points, 8-11; workstation #2 (WS#2) is based on the four points, 12-15; and workstation #3 (WS#3) is based on the four points, 1-4.

Effect of Supply Volume

Figures 4a and 4b present average velocity and temperature results for tests 13, 21, and 23, comparing three different supply volumes (48, 79, and 98 cfm per unit) at the same medium heat load density (30 W/m^2). Data are shown for WS#1, WS#3, and point 7 (see Figure 3), representing a location in the room outside of the three workstations. The supply grills were turned toward the work location in front of the desk in all three workstations. Due to an instability in the anemometer located at the 2.0 m height, velocity measurements are not reported at that height. To aid the comparison of the temperature profiles, the data are presented in terms of a non-dimensional temperature calculated as follows: measured temperature (T) minus supply temperature (T_{sup}) divided by the difference between return (T_{ret}) and supply temperatures.

The velocity results demonstrate the influence of the supply air jets in WS#1 and WS#3. In WS#1, average velocities at the 0.6 m height are greater than 0.25 m/s in all three tests, exceeding the acceptable summer comfort limit specified in the previous version of ASHRAE Standard 55 (1981). The recently released revised version of the standard (ASHRAE 1992) now allows higher velocities when under individual control of the occupant, as is the case with the floor supply system. The elevated air velocities in WS#1 also demonstrate the negative effects of the small-sized workstation in which one of the measurement locations is directly over the floor supply module. A more appropriate installation of the floor-based system in an office would use larger workstations so that the floor supply modules could be installed further away from the work location, allowing the occupant to easily avoid the supply air jets if desired. The larger WS#3 provides a more realistic arrangement in this regard and the velocity results are easier to interpret as discussed below.

At the minimum air flow rate (test 21) the jets do not reach the 1.1 m level, as all of the extremely low velocities (≤ 0.06 m/s [12 fpm]) at this height and above represent essentially still-air conditions. At the middle supply rate (test 23) the jets reach the 1.1 m height, representing the head level of a seated person. The largest supply volume tested (test 13) shows evidence of reaching the 1.7 m level. These results have an important effect on the establishment of the "transition plane" separating the lower and upper zones in the room, as discussed further below. Outside of the workstations at point 7, all velocities are very low as this location is beyond the local impact of the floor supply jets.

The temperature results indicate that the lowest measured air temperatures are always warmer than the floor supply air temperature due to mixing of the supply jets and room air. The maximum measured air temperatures at or near the ceiling level are essentially equal to the return temperature. Within WS#1 and WS#3 the supply jets increase the mixing at lower heights, and eliminate or actually reverse the stratification that would normally be present in the lower zone with true displacement ventilation. Temperature profiles at the 1.7 m level and above are quite consistent for all three tests, and indicate that a warm and fairly well mixed upper zone is maintained in the room, corresponding to the expected displacement system flow pattern.

The location of the transition plane is most clearly demonstrated to be between the 0.6 m and 1.1 m heights for the results of test 21, having the lowest supply volume. The cooler lower zone is separated by a significant increase in temperature up to the upper zone, which begins at the 1.1 m height. Although the exact height of the transition plane can not be determined due to the limited number of measurement heights, this result is in good agreement with the tracer gas measurements of Faulkner et al. (1992). The low flow conditions of test 21 would not provide acceptable comfort because the stratification produced in both WS#1 and WS#3 exceeds the comfort limits for a vertical temperature difference of 3°C (5°F) between the 0.1 m (4 in.) and 1.7 m (5 ft, 7 in.) levels as specified by ASHRAE (1992). In fact, the 3°C temperature difference is exceeded between the 0.1 m and 1.1 m heights due to the low elevation of the transition plane.

In test 13 with the highest supply volume (although still representing the manufacturer's minimum supply rate), the supply jets clearly reach the 1.1 m level rather directly in WS#3, as the minimum temperature is recorded at this level. Due to the higher supply volume of this test, overall stratification in the room is significantly reduced in comparison to test 21.

Outside of the workstations at point 7, average temperature profiles show stratification of a similar magnitude to those obtained in WS#1 and WS#3, indicating that despite the existence of the office partitions, the cool supply air in the lower zone is pooling around on the floor to cover the whole office. The vertical temperature difference between the 0.1 m and 1.7 m heights is 2.8°C for test 21, 1.9°C for test 23, and 1.0°C for test 13, demonstrating how the supply volume can be used to control the level of stratification in the entire space.

Effect of Grill Direction

Figures 5a and 5b present average velocities and temperatures for tests 14 and 15, both performed with a high heat load density (50 W/m^2) and a supply volume close to 68 cfm per unit. In test 14 the supply grills were turned inward, producing a vertical air jet above the floor supply module, and in test 15, the grills were directed toward the work location in front of the desk. Results are shown for WS#1 and WS#3, comparing the thermal conditions in the small- and large-sized workstations.

In WS#1, due to the close proximity of the floor supply module to the desk, relatively high velocities are obtained, particularly at the 0.6 m height. When the grills are directed inward, the vertical supply jet reaches the 1.1 m height. However, when the grills are turned toward the desk, the jet reaches only to the 0.6 m height, as all measured velocities at the 1.1 m height and above are less than 0.05 m/s (10 fpm). In WS#3, where the floor supply module is further away from the desk, the supply jet is able to reach the head level of a seated person (1.1 m) in front of the desk when the grills are directed toward the work location. When the grills are turned inward, except near the floor, all velocities are quite low because the floor supply module is outside of the measurement area in front of the desk. Previous experimental results have shown that when the grills are in this position, the region of elevated velocities is confined to the narrow space directly above the floor module [Bauman et al. 1991]. The higher velocities near the floor are caused by the entrainment effect of the emerging supply jet.

The temperature results are consistent with the velocity measurements discussed above. When the grills are turned toward the desk, the supply jets produce the coolest temperature at the 0.6 m height in WS#1 and at the 1.1 m height in WS#3. When the grills are turned inward, there is no reversal of the temperature stratification. The position of the floor module near the desk in WS#1 produces noticeably cooler temperatures in the lower zone compared to those measured in WS#3. As a result, the vertical temperature difference between the 0.1 m and 1.7 m heights exceeds the 3.0°C comfort limit in WS#1 during both tests. In comparison, this same temperature difference was less than 2.0°C in WS#3 and was close to 2.5°C outside of the workstations at point 7, for both tests.

Archimedes Number

As discussed previously, the relative strengths of buoyant forces (heat load levels) and inertial forces (supply volume) have an important influence on the displacement flow characteristics in the room. The Archimedes number (Ar), a non-dimensional number that represents the ratio of buoyant forces in the room to inertial forces in the supply air jets, is defined here as $Ar = g \cdot d_0 \cdot \Delta T_0 / (v_0^2 \cdot T_r)$, where g is the acceleration due to gravity (m/s^2), d_0 is the outlet diameter (m), v_0 is the outlet velocity (m/s), ΔT_0 is the temperature difference at jet entry (K), and T_r is the room temperature (K). Faulkner et al. (1992) used the Archimedes number to analyze their tracer gas measurements of ventilation performance of this same floor-based system. They found that the slope of the average age of air versus height in the room could be correlated quite well with Ar and the ratio of supply volume to internal heat load. This result indicated a displacement flow pattern with lower ages of air and improved ventilation efficiency in the lower zone.

To investigate the nature of the thermal stratification produced by the floor-based system, we plotted vertical temperature difference versus Archimedes number. The results are shown for tests with the supply grills turned inward in Figure 6a for the difference between temperatures at the 1.7 m and 0.1 m heights ($\Delta T_{1.7}$), and in Figure 6b for the difference between the 1.1 m and 0.1 m heights ($\Delta T_{1.1}$). Results for tests with the supply grills directed toward the desk are shown in Figures 7a and 7b. In the calculation of Ar, ΔT_0 represents the temperature difference between the room air and the average (of three supply modules) supply air temperature, and v_0 is the average outlet velocity. v_0 was not measured for every test, but was estimated to be proportional to supply volume. Results are shown for WS#3, representing a realistic workstation configuration for the floor supply module, and for point 7, a location outside of the workstations.

At low Ar over the range tested (higher supply volume, lower heat load), the force of inertia of the supply jets is stronger and reduces the level of stratification to less than 1°C (2°F) in WS#3. Even outside of the workstations, $\Delta T_{1.7}$ is only slightly larger than 1°C. This degree of stratification could be reduced to even lower levels if the supply volume was increased above the relatively low rates of these tests. When the supply grills are turned toward the desk in WS#3, the supply jets reach the 1.1 m height in front of the desk, causing negative values for $\Delta T_{1.1}$ (Figure 7b). On the other hand, when the grills are turned inward there is increased mixing of the supply and room air, producing small, but positive temperature stratification.

At the higher Ar values (lower supply volume, higher heat load), buoyancy forces have become more important as the supply jets do not reach as high in the space. Greater stratification is produced in the room with values for both $\Delta T_{1.7}$ and $\Delta T_{1.1}$ exceeding the 3°C (5°F) comfort limit in WS#3 when the supply grills are turned toward the desk. These temperature differences also approach or exceed 3°C outside of the workstations at point 7. The height of the transition plane separating the lower and upper zones in displacement flow is clearly below the 1.1 m height for these conditions.

At intermediate Ar values over the range studied ($\approx 9 \times 10^{-3}$), the results indicate that the supply jets are reaching the 1.1 m level (negative $\Delta T_{1.1}$ in WS#3 when the supply grills are turned toward the desk). The transition plane appears to be above this height as values for $\Delta T_{1.7}$ are close to 2.5°C at point 7 outside of the workstations. If Ar is maintained at less than about 0.01, this may be the critical range for optimized operation of the floor-based system by providing thermal control for a seated office worker (supply jets reach the head level) while maximizing (within acceptable limits) stratification in the overall space.

Thermal Manikin Results

To further evaluate thermal comfort conditions produced by the floor-based task conditioning system in WS#3, selected results from the thermal manikin measurements are presented below.

Figure 8 presents the heat loss from each part of the clothed thermal manikin for the conditions of test 21. Results are shown for both sitting and standing positions to compare the effect of the large thermal stratification ($\Delta T_{1.1} > 3.0$°C) that occurred during this test at a very low flow rate. Data for the left foot are taken to be equal to that of the right foot due to an unresolved shift in calibration for that body part during the tests. No other measurements were affected. The higher temperatures encountered by the standing manikin produce a noticeable reduction in heat loss from most of the body. The mean heat loss was 49 W/m^2 for standing position compared to 58 W/m^2 when seated, representing a 15% decrease. The highest rates of heat loss are observed at the left foot and left hand because these parts are exposed to the cool supply air and have little clothing insulation.

Tanabe et al. (in press) describe how the measured heat loss by the manikin is used to calculate a manikin-based equivalent temperature (t_{eq}), defined as the temperature of a uniform enclosure in which a thermal manikin with realistic skin surface temperatures would lose heat at the same rate as it would in the actual environment. This equivalent temperature can then be inserted into the

ISO (1984) computer program for calculating PMV, based on Fanger's PMV model [Fanger 1970]. To complete the calculation, the following values are used: air velocity is assumed to be still air, relative humidity is 50%, the manikin's clothing insulation value is 0.55 clo, and the activity level is 1.1 met (representing typical office work). The PMV for the seated manikin is -0.8 compared to -0.5 for the standing manikin, demonstrating that a seated office worker would experience a significant increase in whole-body PMV of 0.3 simply by standing up.

Figure 9 presents the predicted values for equivalent temperature for each part of the manikin for the conditions of test 19. The results are divided into left and right sides to demonstrate the thermal asymmetry produced during this test, using a combination of one of the highest supply volumes tested with the highest heat load level. Referring to Figure 2, cool air from the floor module was directed toward the left side of the manikin, and the extra radiant heat source in WS#3 was positioned to the right of the manikin. The mean whole-body t_{eq} is 24.4°C (76°F), producing a PMV value for this test of -0.2 (calculated as described above), indicating that conditions are well within the acceptable comfort range. However, individual parts experience substantially different thermal sensations depending on their exposure to the sources of cooling and heating. The entire left side is cooler than the corresponding right-side parts, clearly demonstrating the thermal asymmetry that could not be detected with a single temperature sensor. The coolest sensations occur at the left hand and head, as these parts are exposed directly to the supply air jets and have little clothing insulation. The warmest sensations occur on the right shoulder, arm, and hand, as these parts are closest to the radiant heater. The overall difference in t_{eq} between left and right sides is 1°C (1.8°F), estimated to represent a whole-body PMV value of 0.3.

Tanabe et al. (in press) provide additional PMV results based on the thermal manikin for the floor-based system. The manikin is seen to provide a useful means of evaluating thermal comfort based on an equivalent rate of whole-body heat exchange with the environment, particularly when asymmetric conditions exist.

CONCLUSIONS

Laboratory measurements were made to investigate the performance of an occupant-controlled floor-based task conditioning system. The experiments were carried out in a controlled environment chamber configured to resemble a modern office space with modular partitioned workstations. Detailed tests were conducted to study the effects of supply volume, supply temperature, supply direction, and heat load levels in the space. In addition to detailed temperature and velocity measurements, a new skin-temperature-controlled thermal manikin was used to evaluate the non-uniform thermal environments produced by the floor-based system, in both sitting and standing positions. The major findings are summarized below:

1. Under the relatively low supply volumes investigated in this study, the overall performance of the floor-based system resembled that of displacement ventilation systems. Under these conditions, air temperatures increased with height in the space (thermal stratification), complementing previous related tracer gas experiments by Faulkner et al. (1992) on this same floor-based system that showed that the age of air also increased with height (improved ventilation efficiency in the occupied zone).
2. The higher supply velocities associated with this task conditioning system increase the mixing with room air and disrupt the typical piston-like upward flow in the lower zone of a true displacement system, thereby decreasing (and sometimes reversing) the level of stratification in this region.
3. The degree of overall stratification in the space is strongly dependent on the relative strengths of the floor supply jets and internal heat load (Archimedes number). Correspondingly, the height of the "transition plane" separating the characteristic lower and upper zones in the displacement flow pattern can also be controlled for a given heat load level primarily by adjusting the floor supply volume. In fact, it has been shown previously that at maximum supply volumes, the floor supply jets can reach the ceiling, thereby minimizing stratification and producing close to uniform thermal and ventilation conditions in the space [Bauman et al. 1991].
4. In optimizing the performance of the floor-based system, the height of the transition plane in the space is a critical performance criteria. It should be possible to maintain the transition plane above the occupant's breathing level and below the ceiling level, allowing acceptable occupant comfort to be maintained (by limiting stratification), and improving air exchange efficiency and ventilation efficiency in the occupied zone, while using the warmer upper zone to reduce space cooling energy use compared to conventional ceiling-based "mixing" ventilation systems.
5. The floor supply module should be installed at a reasonable distance (1-1.5 m [3-5 ft]) from the work location in front of the desk. If the supply module is too close, an office worker may experience unacceptably persistent cool and drafty conditions near the floor.
6. Manikin-based measurements indicated that thermally asymmetric conditions (stratification and locally cooling air jets) are produced by the floor-based system under certain operating conditions. If these non-uniform thermal environments are adjusted to provide the necessary

rate of whole-body heat exchange (by controlling the floor supply module), acceptable thermal comfort can be maintained.

FUTURE WORK

Future work on floor-based task conditioning systems should address the following issues:

1. Field monitoring projects of operational systems are needed to test the optimization criteria suggested by the present study. Data can be collected to demonstrate occupant response, thermal comfort, indoor air quality, and energy use implications under a wider range of environmental conditions.
2. The existing ASHRAE comfort standard [ASHRAE 1992] specifies limits to vertical (5°C [9°F]) and horizontal (10°C [18°F]) radiant asymmetry for acceptable comfort. Additional subjective and thermo-physiological data based on human subject testing are needed to describe the maximum limits for acceptable thermally non-uniform conditions associated with the local cooling by air jets produced by the floor-based system. Experimental work of this nature is now in progress in our controlled environment chamber.
3. Improved integration of the design and control of task conditioning systems with the building's central HVAC system is needed. Overall building energy performance is closely related to the sophistication with which this integration occurs.
4. There is significant interest in pursuing the investigation of worker productivity issues related to task conditioning systems.

ACKNOWLEDGMENTS

The research reported here was funded by the California Institute for Energy Efficiency (CIEE), a research unit of the University of California. Publication of research results does not imply CIEE endorsement of or agreement with these findings, nor that of any CIEE sponsor. We would like to acknowledge the support and technical assistance of Gene Brager of Dynamic Building Systems and John Zeren, both formerly with Tate Access Floors, Jessup, MD. This research was also aided significantly by generous furniture and partition donations from Steelcase, Inc., Grand Rapids, MI. The authors also appreciate the administrative support provided by Nora Watanabe of the Center for Environmental Design Research, University of California, Berkeley.

REFERENCES

Arens, E., F. Bauman, L. Johnston, and H. Zhang. 1991. "Testing of localized ventilation systems in a new controlled environment chamber." *Indoor Air*, No. 3, pp. 263-281.

ASHRAE. 1981. *ANSI/ASHRAE Standard 55-1981*, "Thermal environmental conditions for human occupancy." Atlanta: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

ASHRAE. 1992. *ANSI/ASHRAE Standard 55-1992*, "Thermal environmental conditions for human occupancy." Atlanta: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

Bauman, F.S., and E.A. Arens. 1988. "The development of a controlled environment chamber for the physical and subjective assessment of human comfort in office buildings." In: *A New Frontier: Environments for Innovation, Proceedings: International Symposium on Advanced Comfort Systems for the Work Environment*, W. Kroner, ed. Troy, NY: Center for Architectural Research.

Bauman, F.S., L.P. Johnston, H. Zhang, and E.A. Arens. 1991. "Performance testing of a floor-based, occupant-controlled office ventilation system." *ASHRAE Transactions*, Vol. 97, Pt. 1.

Bauman, F., G. Brager, E. Arens, A. Baughman, H. Zhang, D. Faulkner, W. Fisk, and D. Sullivan. 1992. "Localized Thermal Distribution for Office Buildings: Final Report - Phase II." Center for Environmental Design Research, University of California, Berkeley, December.

Fanger, P.O. 1970. Thermal Comfort. Copenhagen: Danish Technical Press.

Faulkner, D., W.J. Fisk, and D.P. Sullivan. 1992. "Indoor air flow and pollutant removal in a room with floor-based task ventilation: results of additional experiments." In Bauman, F., G. Brager, E. Arens, A. Baughman, H. Zhang, D. Faulkner, W. Fisk, and D. Sullivan. 1992. "Localized Thermal Distribution for Office Buildings: Final Report - Phase II." Center for Environmental Design Research, University of California, Berkeley, December.

Fisk, W.J., D. Faulkner, D. Pih, P.J. McNeel, F.S. Bauman, and E.A. Arens. 1991. "Indoor air flow and pollutant removal in a room with task ventilation." *Indoor Air*, No. 3, pp. 247-262.

Gagge, A.P., Y. Nishi, and R.R. Gonzales. 1973 "Standard effective temperature - a single temperature index of temperature sensation and thermal discomfort." Proceeding of the CIB Commission W45 (Human Requirements), Symposium, Thermal Comfort and Moderate Heat Stress. BRS, pp. 229-250.

Heinemeier, K.E., G.S. Brager, C.C. Benton, F.S. Bauman, and E. A. Arens. 1991. "Task/Ambient Conditioning Systems in Open-Plan Offices: Assessment of a New Technology." Center for Environmental Design Research, University of California, Berkeley, September.

ISO. 1984. *International Standard 7730*, "Moderate thermal environments -- determination of the PMV and PPD indices and specification of the conditions for thermal comfort." Geneva: International Standards Organization.

McNall, P.E., and R.E. Biddison. 1970. "Thermal and comfort sensations of sedentary persons exposed to asymmetric radiant fields." *ASHRAE Transactions*, Vol. 76, Pt. 1.

Sandberg, M., and C. Blomqvist. 1989. "Displacement ventilation systems in office rooms." *ASHRAE Transactions*, Vol. 95, Part 2.

Seem, J.E., and J.E. Braun. 1992. "The impact of personal environmental control on building energy use." *ASHRAE Transactions*, Vol. 90, Pt. 1.

Shute, R.W. 1992. "Integrating access floor plenums for HVAC air distribution." *ASHRAE Journal*, Vol. 34, No. 10, October.

Sodec, F. 1984. *Air Distribution Systems Report No. 3554A*. Aachen, West Germany: Krantz GmbH & Co., Sept. 19.

Sodec, F., and R. Craig. 1991. *Underfloor Air Supply System: Guidelines for the Mechanical Engineer*. Report No. 3787 A. Aachen, West Germany: Krantz GmbH & Co., January.

Spoormaker, H.J. 1990. "Low-pressure underfloor HVAC system." *ASHRAE Transactions*, Vol. 96, Part 2.

Svensson, A.G.L. 1989. "Nordic experiences of displacement ventilation systems." *ASHRAE Transactions*, Vol. 95, Part 2.

Tanabe, S., E.A. Arens, F.S. Bauman, H. Zhang, and T.L. Madsen. In press. "Evaluation of thermal environments with a skin-surface-temperature-controlled thermal manikin." Submitted to *ASHRAE Transactions*.

Wyon, D.P., and M. Sandberg. 1990. "Thermal manikin prediction of discomfort due to displacement ventilation." *ASHRAE Transactions*, Vol. 96, Pt. 1.

LIST OF TABLES AND FIGURES

Table 1 Name and surface area of each body part

	Name of part	Area(m^2)
1	Left Foot	0.0446
2	Right Foot	0.0437
3	Left Leg	0.0892
4	Right Leg	0.0879
5	Left Thigh	0.1630
6	Right Thigh	0.1670
7	Crotch	0.1740
8	Head	0.1100
9	Left Hand	0.0397
10	Right Hand	0.0394
11	Left Arm	0.0490
12	Right Arm	0.0500
13	Left Shoulder	0.0736
14	Right Shoulder	0.0778
15	Chest	0.1380
16	Back	0.1270
Total		1.4739

Table 2. TEST CONDITIONS

TEST NO.	TOTAL SUPPLY VOLUME (cfm)	SUPPLY VOLUME PER UNIT (cfm)	SUPPLY AIR TEMP. (°C)			HEAT LOAD (W)				ROOM TEMP.* (°C)	RETURN AIR TEMP. (°C)	RETURN AIR VOLUME (cfm)	GRILL DIRECTION
			WS#1	WS#2	WS#3	WS#1	WS#2	WS#3	**TOTAL**				
1	101	34	17.0	17.0	17.0	200	75	220	**750**	25.4	26.3	42	Toward
2	152	51	17.6	17.2	16.9	200	75	220	**750**	25.1	24.8	78	Toward
3	206	69	16.9	17.2	17.0	200	140	220	**1110**	25.2	25.8	133	Inward
4	205	68	16.8	17.1	17.1	200	140	220	**1110**	25.3	25.8	132	Toward
5	199	66	18.0	18.0	18.0	200	220	220	**1090**	23.7	24.9	195	Toward
6	267	89	21.1	20.8	21.1	200	220	220	**1090**	25.7	26.0	224	Inward
7	269	90	21.0	20.7	21.0	200	220	220	**1090**	25.7	26.1	223	Toward
8	265	88	19.3	18.6	19.3	400	420	420	**1490**	25.4	26.2	222	Inward
9	265	88	19.3	18.5	19.3	400	420	420	**1490**	25.8	26.2	222	Toward
10	208	69	19.2	18.3	19.1	400	420	420	**1490**	24.8	26.5	196	Inward
11	208	69	18.6	17.3	18.4	400	420	420	**1490**	25.5	26.9	196	Toward
12	295	98	21.4	21.1	21.4	200	220	220	**890**	26.5	27.0	227	Inward
13	294	98	21.5	21.2	21.6	200	220	220	**890**	27.3	27.3	228	Toward
14	207	69	17.6	16.1	17.3	400	420	420	**1490**	24.8	26.2	198	Inward
15	200	67	17.7	16.0	17.3	400	420	420	**1490**	24.7	26.3	196	Toward
16	210	70	17.5	16.2	17.3	400	420	420	**1490**	24.5	26.0	201	Inward
17	205	68	17.6	16.2	17.2	400	420	420	**1490**	24.6	26.1	194	Toward
18	276	92	18.3	17.6	18.4	400	420	420	**1490**	24.8	25.4	223	Inward
19	268	89	18.3	17.5	18.3	400	420	420	**1490**	25.0	25.8	224	Toward
20	139	46	18.1	17.8	17.8	200	220	220	**890**	24.3	26.2	130	Inward
21	144	48	17.0	18.0	16.7	200	220	220	**890**	24.4	26.3	130	Toward
22	235	78	20.9	20.3	21.1	200	220	220	**890**	25.0	26.1	144	Inward
23	237	79	20.9	20.3	21.1	200	220	220	**890**	25.6	26.6	145	Toward

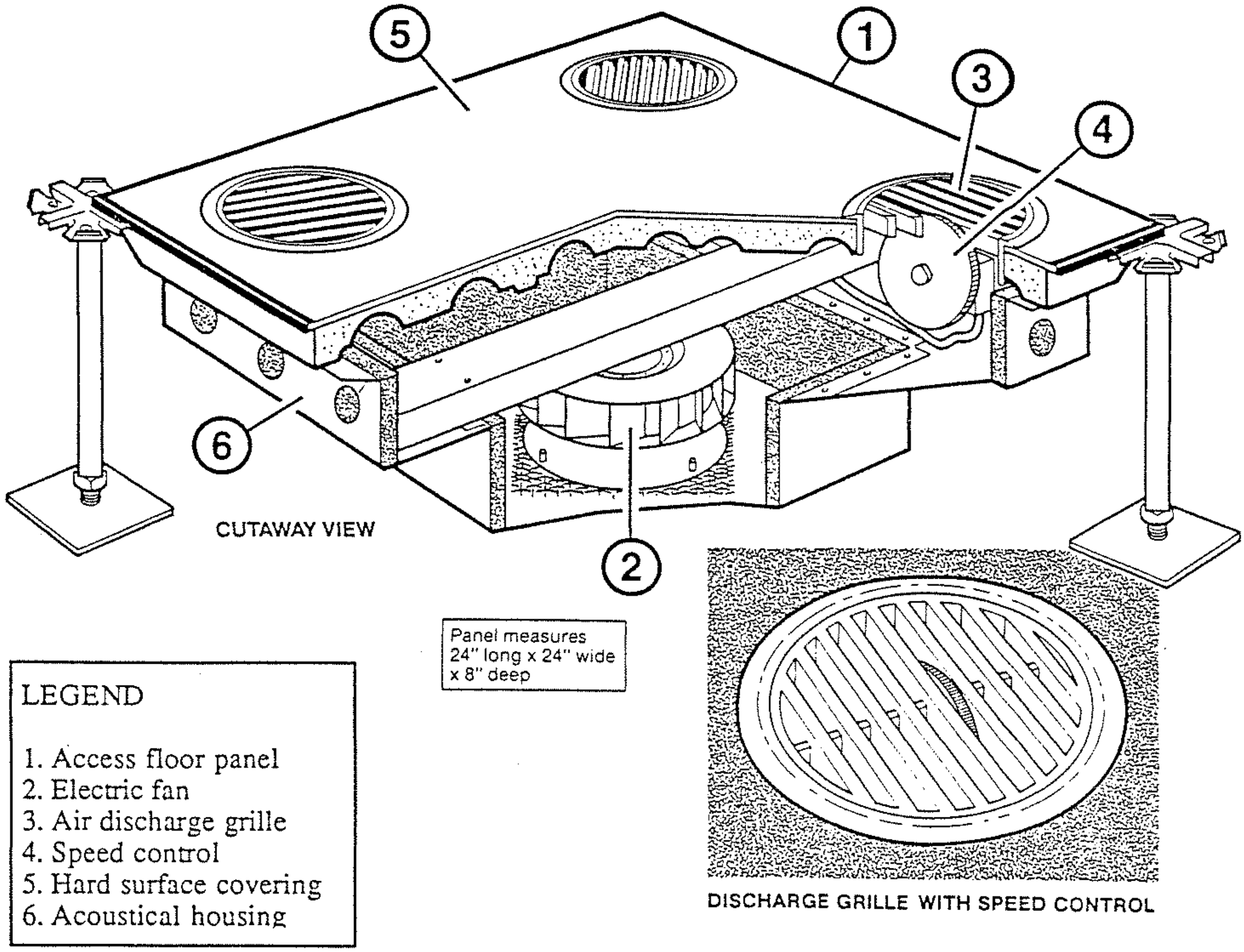
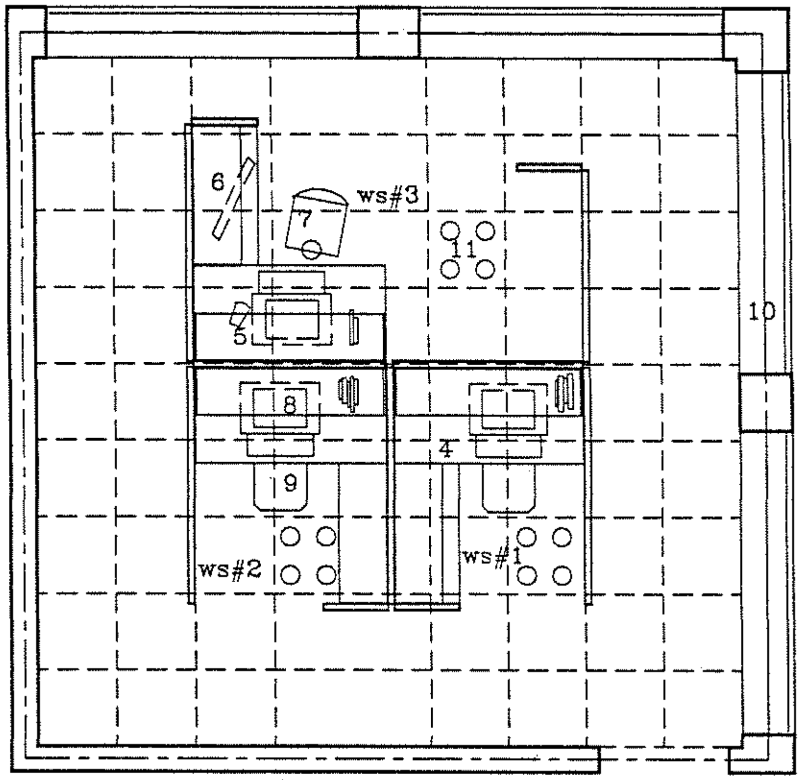


Figure 1. Floor Supply Module



ws#1 – workstation #1	7 – manikin
ws#2 – workstation #2	8 – computer
ws#3 – workstation #3	9 – chair
4 – desk	10 – annular space
5 – task light	11 – floor supply module
6 – radiation heat panel	

Figure 2. Chamber Plan

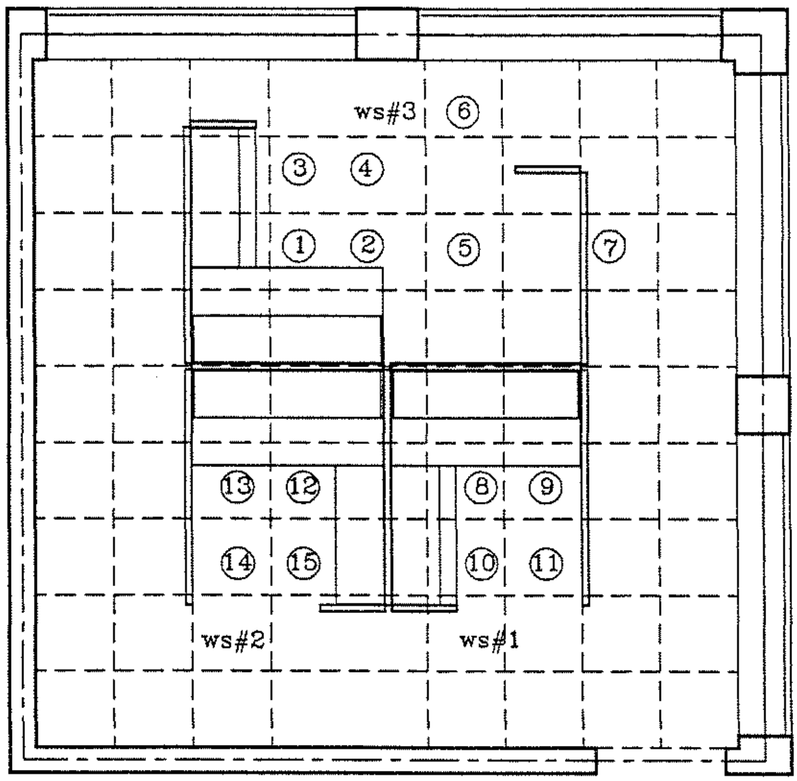


Figure 3. Measurement Locations

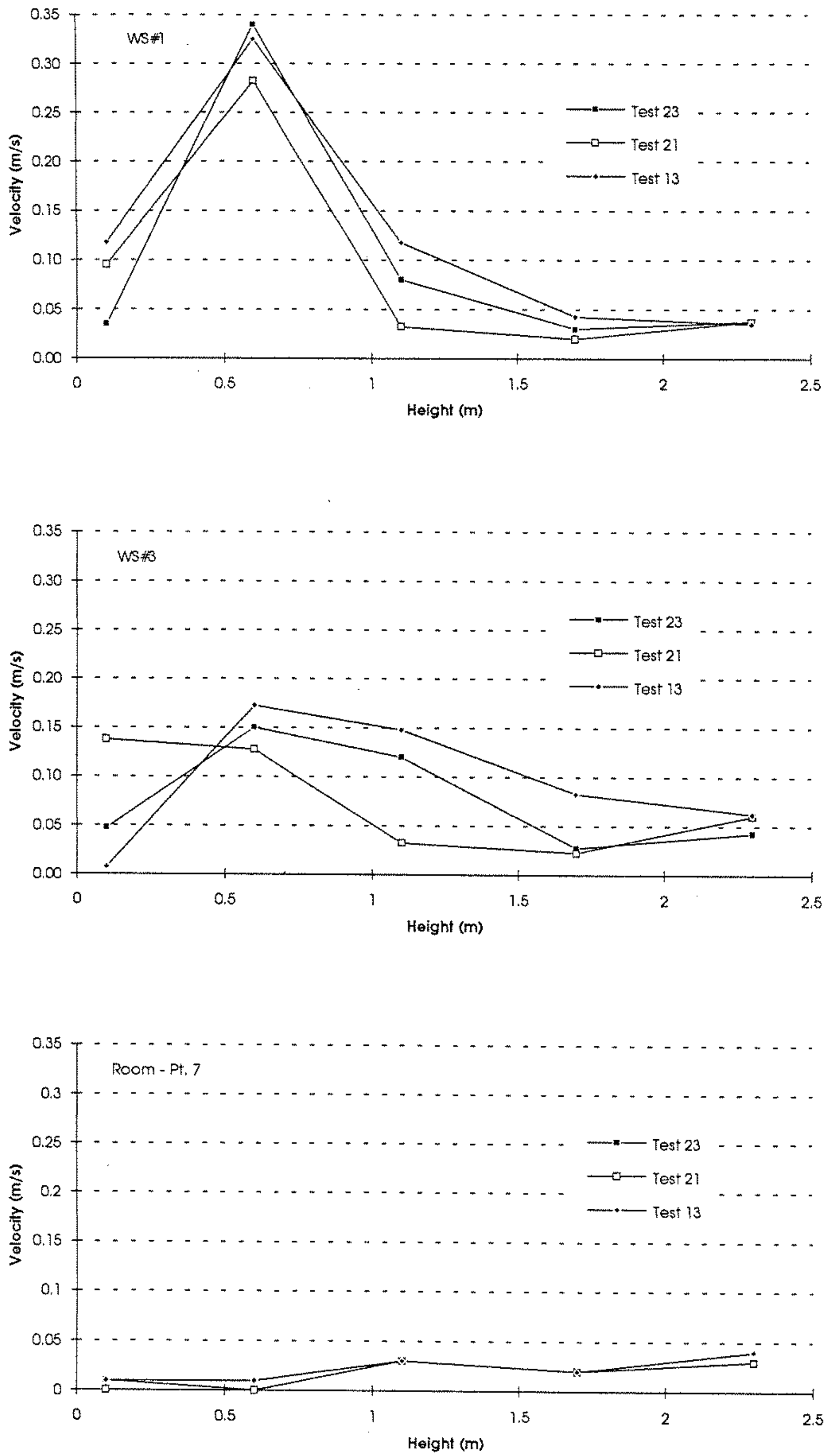


Figure 4a. Effect of supply volume: average velocities

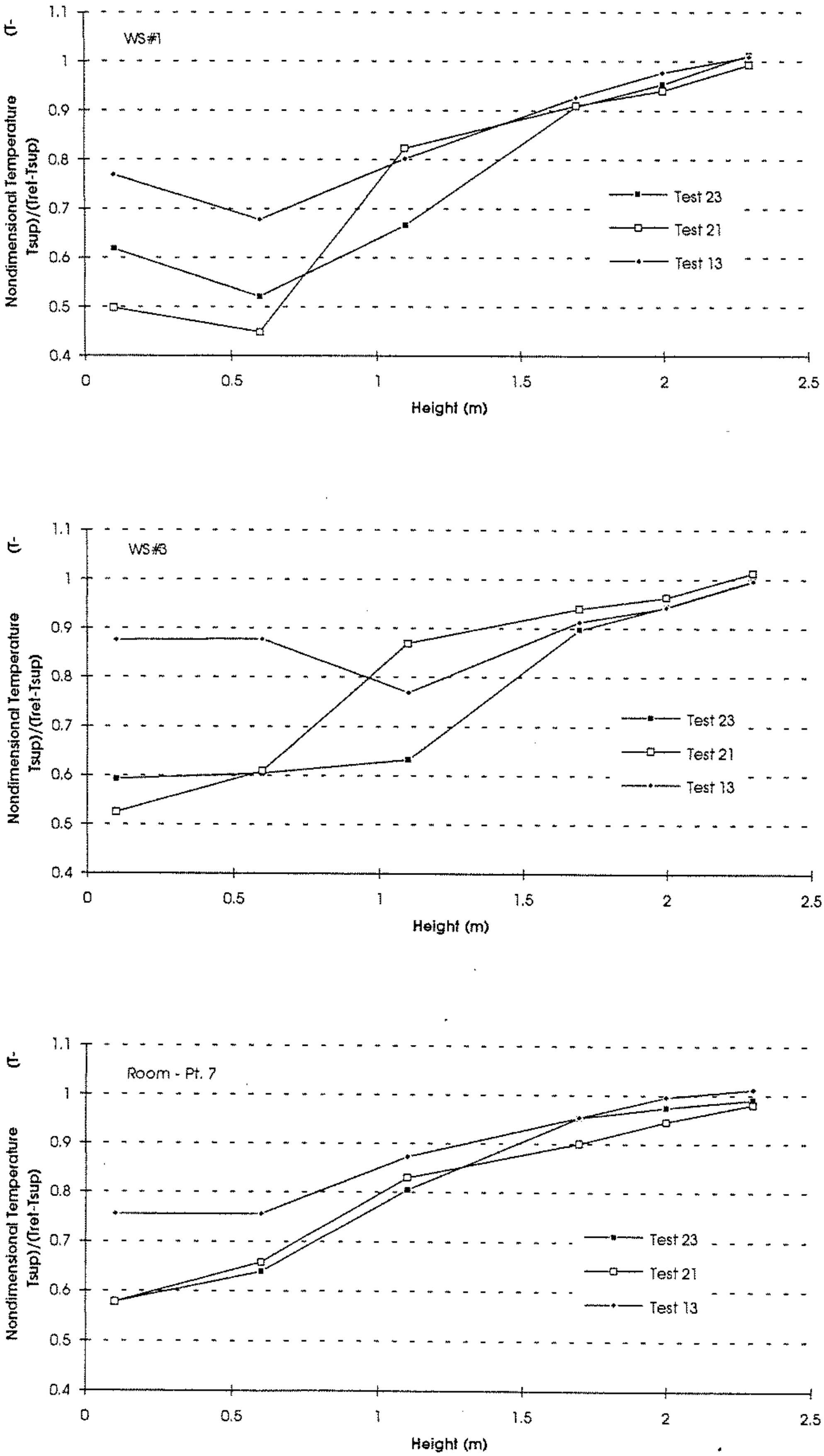


Figure 4b. Effect of supply volume: average temperatures

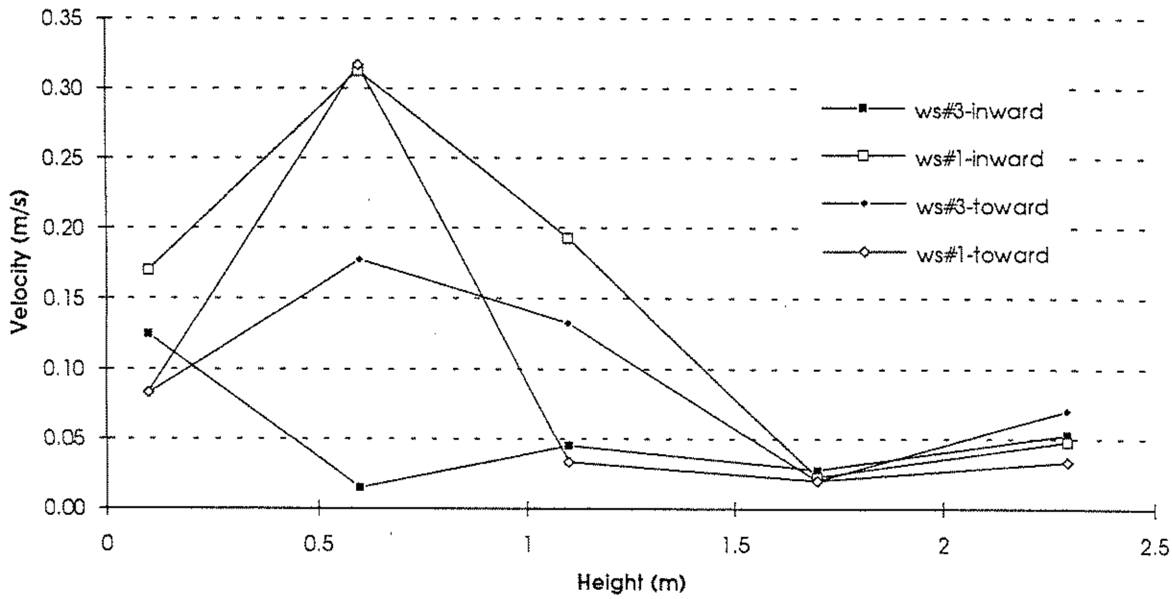


Figure 5a. Effect of grill direction: average velocities

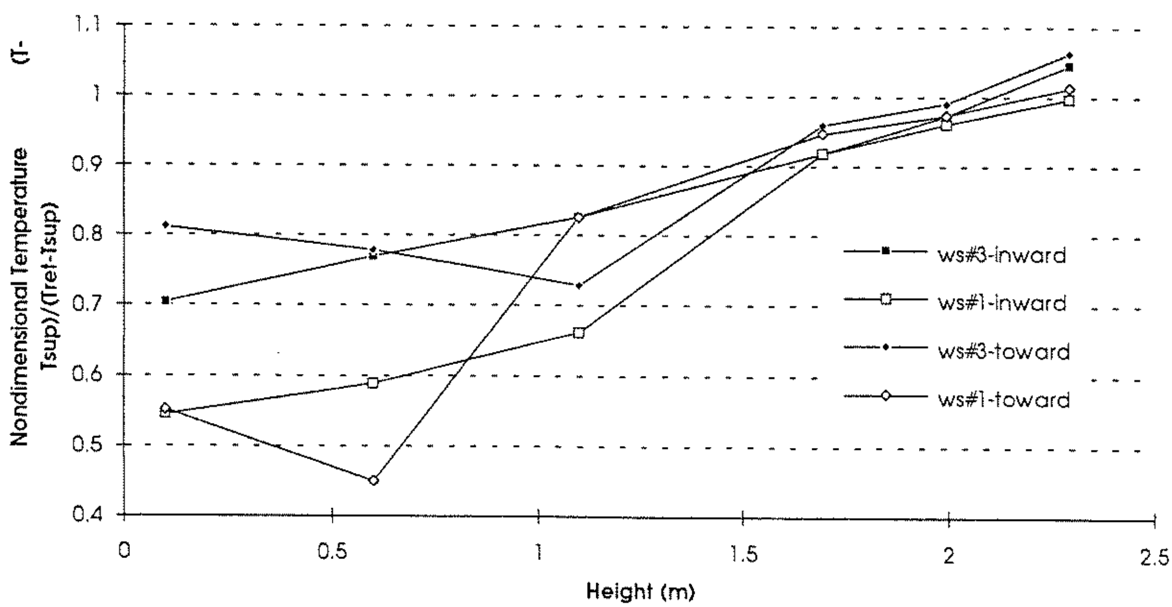


Figure 5b. Effect of grill direction: average temperatures

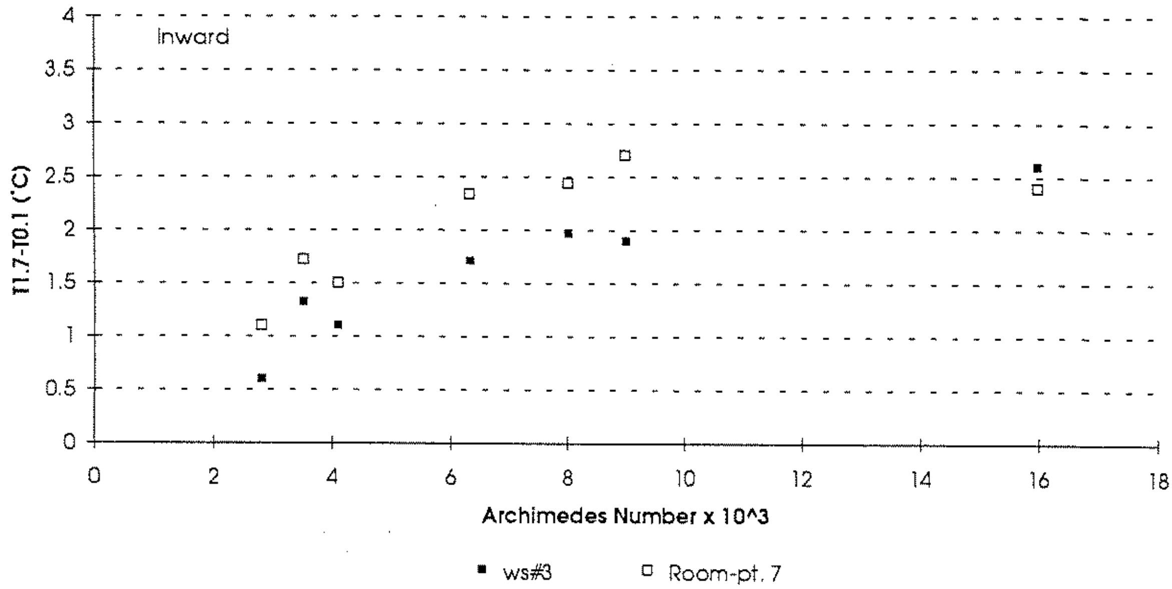


Figure 6a. Archimedes number versus temperature difference between 1.7 m and 0.1 m heights: inward grill direction

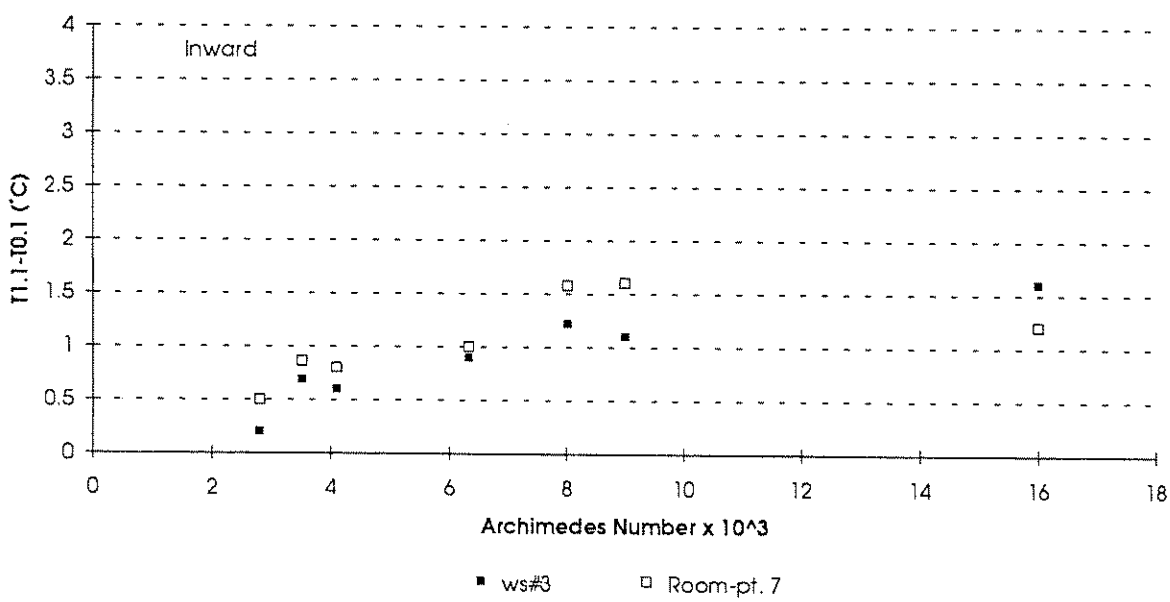


Figure 6b. Archimedes number versus temperature difference between 1.1 m and 0.1 m heights: inward grill direction

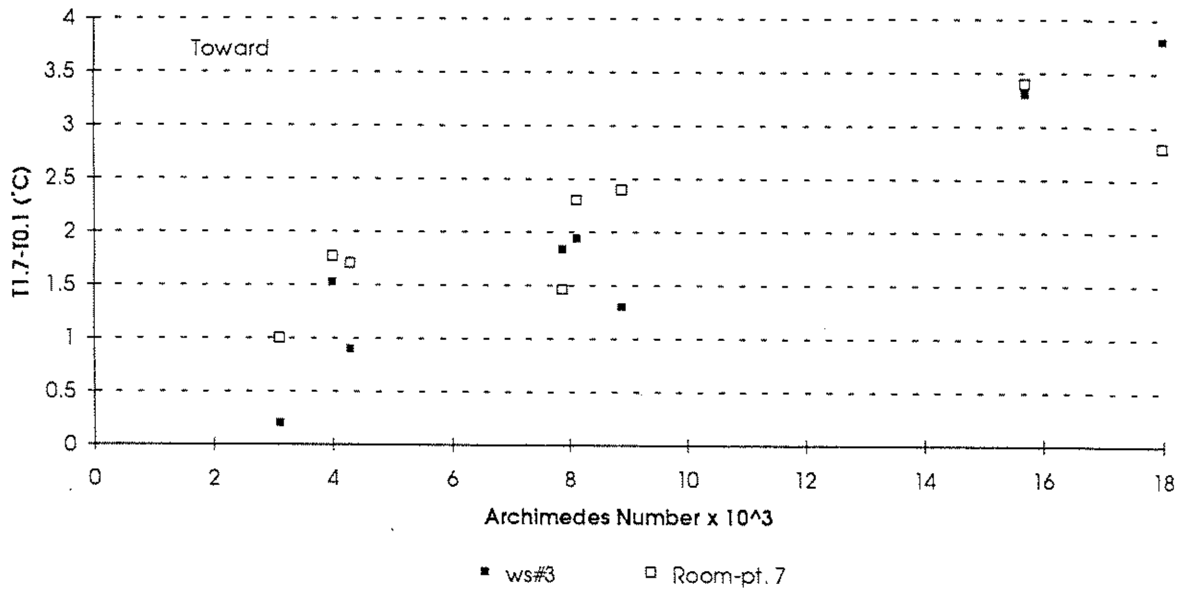


Figure 7a. Archimedes number versus temperature difference between 1.7 m and 0.1 m heights: toward grill direction

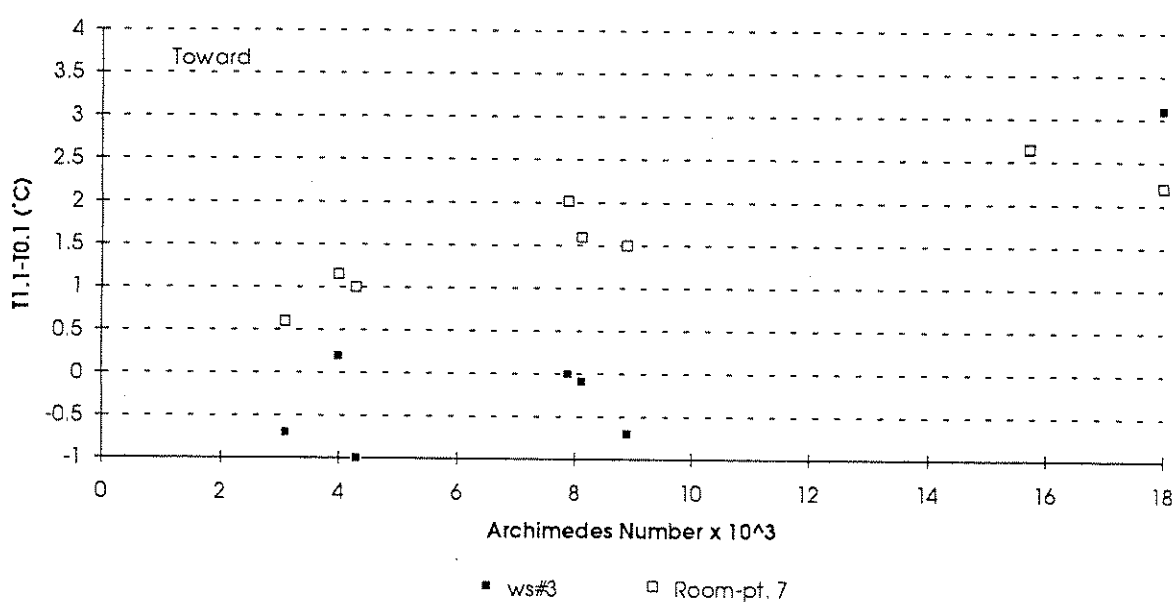


Figure 7b. Archimedes number versus temperature difference betrween 1.1 m and 0.1 m heights: toward grill direction

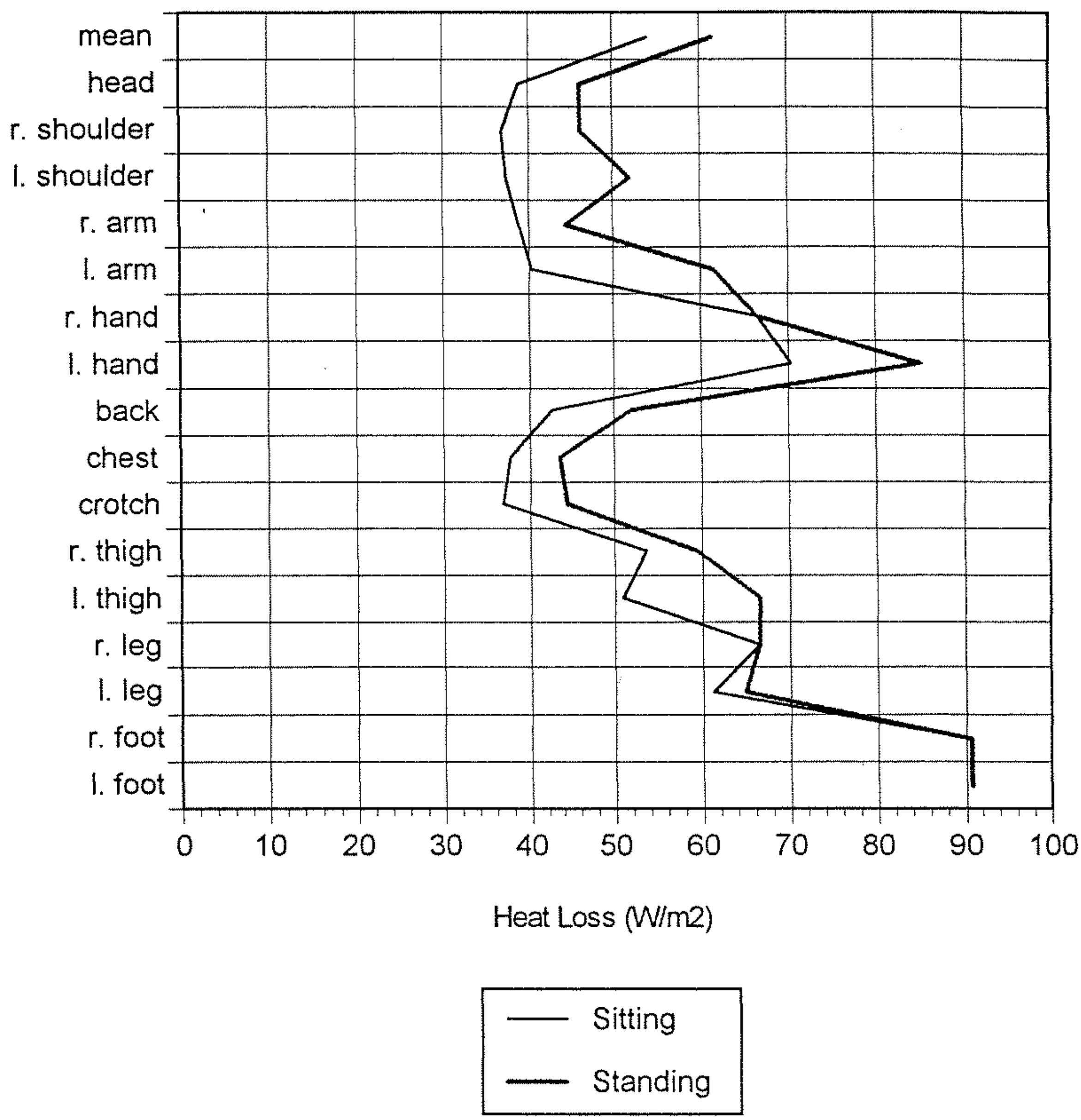


Figure 8. Heat loss for sitting versus standing manikin: test 21

Figure9. Manikin-based equivalent temperature for left versus right sides: test 19