EXPERIMENTAL STUDY ON FLOOR-SUPPLY DISPLACEMENT VENTILATION SYSTEM

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ABSTRACT

Thermal performance of the floor-supply displacement ventilation system was evaluated in a large climatic chamber designed to simulate a single span of an office building. Detailed measurements were conducted to determine the indoor environment and skin temperature of a thermal manikin. Temperature gradient in the room could be kept smaller compared to conventional wall-supply unit displacement ventilation system,[(1)(2)] owing to the floor cooling effect of the floor-supply system. Altering the supply air volume, the amount of heat load, and the position of heat sources showed a great influence on vertical temperature difference. Special care would be required to maintain the thermal stratification in a perimeter zone with cold or warm windows.

INTRODUCTION

Energy conservation is one of the main topics concerned in modern society. In the process of improvements in the field of air-conditioning, which occupies a large part of energy consumption in buildings, displacement ventilation system was developed as one of the most promising systems to realize high ventilation efficiency and energy conservation.[(3)(4)(5)] However, the vertical temperature difference and draft remain a critical issue in Japan, because higher cooling load might be expected by hot and humid summer.[(1)(2)] To cope with this problem, the floor-supply displacement ventilation system with breathable carpet was devised and tested in a full-scale climatic chamber. The basic idea behind this system is shown in Figure 1. The purpose of this study is to identify the indoor thermal environment of the floor-supply displacement ventilation system in a controlled chamber simulating a typical office space where numerous heat-generating equipment such as personal computers and copying machines are scattered throughout the room under various conditions.

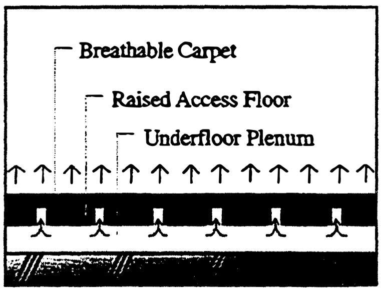

Figure 1 Section of raised access floor

METHODS

Experimental Chamber

All experiments were performed in a large climatic chamber designed to resemble a single span of an office building as shown in Figure 2, during the summer season in 1996. The floor is completely covered with the breathable carpet over a raised access floor, producing a 120 mm high subfloor plenum. The underfloor space was used as a pressurized plenum chamber of the supply air fed from a supply duct connected to one side. Air is extracted through the outlets in the lighting fixtures into the ceiling plenum space used as a return chamber. This double-chambered structure is also equipped with an artificial sunlight and regulatory compartment outside the window to reproduce summer and winter conditions in the perimeter zone.

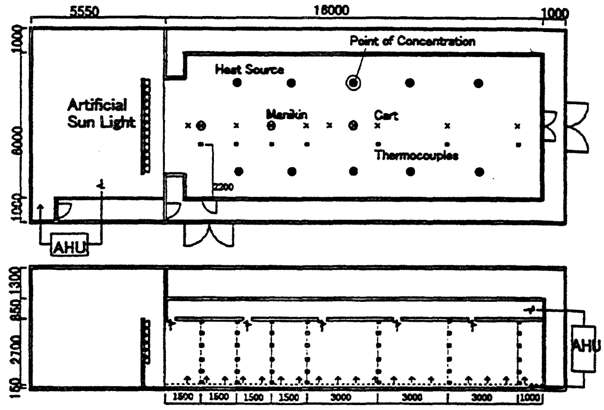

Figure 2 Plan and section of experimental chamber with measurement points (unit in mm)

Measurement Methods

Three types of measurements were conducted under steady-state conditions, utilizing an instrumental cart devised to measure 7 thermal parameters (8 locations), thermocouples (7 locations), and a thermal manikin (3 locations). Measured items are shown in Tables 1 and 2. For the cart measurements, locations were changed for conditions with scattered heat sources (1.0/1.5/3.0/4.5/6.0/9.0/12.0/15.0 m from the window) and clustered heat sources (1.0/1.5/3.0/4.5/6.0/7.0/8.0/9.0 m from the window). The thermal manikin was set on an office chair facing the window. See Figure 2 for measurement locations.

Table 1 Measured items and points of the instrumental cart

Item	Instrument	Height
Air Temperature	Thermocouple	0.1,0.6,1.1,1.6m
Globe Temperature	Small Globe Thermometer	0.1,0.6,1.1,1.6m
Air Velocity	Indoor Climate Analyzer	0.1, ,1.1,1.6m
Solar Radiation	Solar Meter	1.1m
Humidity	Relative Humidity Sensor	0.6m
Equivalent Temperature	Comfort Meter	0.6m
Radiant Temperature	Indoor Climate Analyzer	1.1m /6 sides

Table 2 Measured points of temperature distribution

Vertical Distribution	9 heights(0,0.1,0.6,1.1,1.7,2.2,2.7,3.0)×7points
Under Floor	42points
Wall Surface	7points
Window Surface	15points
Room Air Outlet	9points
Outer Chamber	6points
Air-Conditioning Unit (return/supply air,etc.)	7points

Experimental Conditions

Light bulbs covered with aluminum cylinders were used as heat sources (100W×10, 20), and were either scattered in 2 lines or clustered together in one location, 8.0 m apart from the window, as shown in Figure 2. Supply air volume was kept constant during each experiment. Supply air temperature was controlled to maintain 25℃ room air temperature at a representative point near the entrance, 1.1 m above floor level. Experiments were conducted on 12 conditions, altering the supply air volume (1890 m^3/h, 1350 m^3/h, 810 m^3/h), the amount of heat load (26W/m^2, 36W/m^2), the height of heat sources (high, middle, low), and the arrangement of heat sources (scattered, clustered). In addition, 2 cases of typical summer and winter conditions were reproduced to investigate the thermal environment near the window. See Table 3 for details.

Table 3 Experimental conditions

Legend	Heat Load	Height of Heat Source	Arrangement of Heat Source	Perimeter Load	Supply Air Volume	Room Air Set Point Temperature
26HS-7	26W/m^2	H (2.0m)	Scattered		1890m^3/h (7h^{-1})	25°C
26MS-7	26W/m^2	M (1.0m)	Scattered	—	1890m^3/h	25°C
26LS-7	26W/m^2	L (0.2m)	Scattered		1890m^3/h	25°C
26HC-7	26W/m^2	H	Clustered		1890m^3/h	25°C
26MC-7	26W/m^2	M	Clustered	—	1890m^3/h	25°C
26LC-7	26W/m^2	L	Clustered		1890m^3/h	25°C
26MS-5	26W/m^2	M	Scattered	—	1350m^3/h (5h^{-1})	25°C
26MS-3	26W/m^2	M	Scattered		810m^3/h (3h^{-1})	25°C
36HS-7	36W/m^2	H	Scattered		1890m^3/h	25°C
36MS-7	36W/m^2	M	Scattered	—	1890m^3/h	25°C
36LS-7	36W/m^2	L	Scattered		1890m^3/h	25°C
36MC-7	36W/m^2	M	Clustered	—	1890m^3/h	25°C
SMR	26W/m^2	M	Scattered	Artificial Sunlight	1890m^3/h	25°C
WTR	26W/m^2	M	Scattered	Cold Window	1890m^3/h	25°C

*h^{-1}=room air changes per hour

RESULTS

Heat Balance

Figure 3 illustrates the fact that exhausted heat calculated from the temperature difference of supply and exhausted air agreed well with the given heat load. Also, the heat load was reduced down to 50-60% before the supply air entered the occupied space. This means that the supply air penetration through the floor panels caused the whole floor to cool down, which in turn gave rise to a considerable heat transmission between the occupied space and the underfloor plenum.

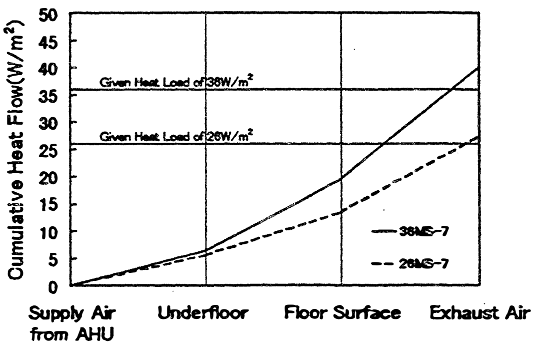

Figure 3 Heat balance of the experiments

Temperature Distribution

Figure 4 presents the vertical air temperature profiles for 1890 m^3/h supply air volume in the case of 36 W/m^2 of scattered heat load with the height of heat sources altered. All of the measured values at 7 locations are plotted in the figure, and the profiles are almost identical for each condition. Note that the temperature gradient changes at the height of the heat sources. This illustrates the fact that the lower the height of the heat sources, the greater the vertical air temperature difference in the occupied zone. This effect became more significant when the heat load was increased. However, the vertical temperature difference between 0.1m and 1.1m was 2.1℃ at the severest condition of "36LS-7", while that of others were kept below 1.5℃. This will be hard to be accomplished with the wall-supply unit displacement ventilation system under the same conditions.[1][2] The floor cooling effect of this floor-supply system as mentioned earlier is found quite important.

The vertical temperature profiles with different supply air volumes are shown in Figure 5. The vertical temperature difference increased when the supply air volume was reduced. The system could not meet the heat discharge requirements at 810 m^3/h, causing a temperature rise throughout the chamber.

The local vertical air temperature profiles were no longer identical when the heat sources were clustered together at one location, 8m from the window. As shown in Figure 6, the air temperature arose from 0.3 to 0.5℃ near the heat sources, but the temperature gradient was maintained to be nearly constant throughout the room. Also, the vertical temperature difference was kept below 2℃ even at the location nearest to the heat sources. Supposedly, these results owe much to the strong tendency of air to spread horizontally.

See Figure 7. When artificial sunlight was brought in to reproduce a typical summer condition of August, the thermal stratification was destroyed throughout the room, even though the blinds were lowered. In a winter condition where the temperature of the exterior compartment outside the window was controlled at 0℃, the effect of cold draft was observed in the perimeter zone, but the thermal stratification at 0.6 m above the floor level was kept undisturbed.

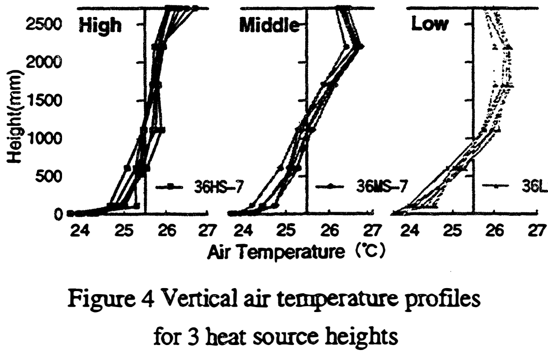

Figure 4 Vertical air temperature profiles for 3 heat source heights

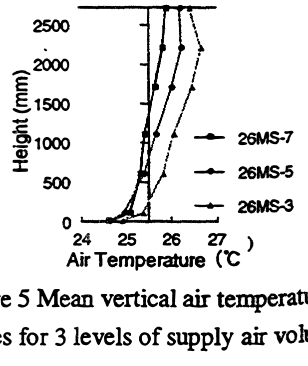

Figure 5 Mean vertical air temperature profiles for 3 levels of supply air volume

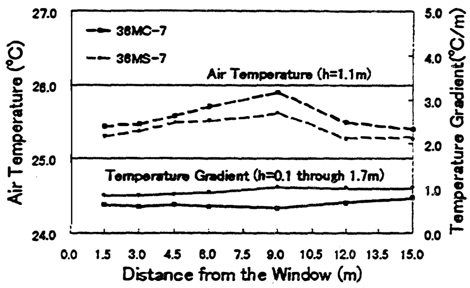

Figure 6 Air temperature and temperature gradient for clustered heat sources

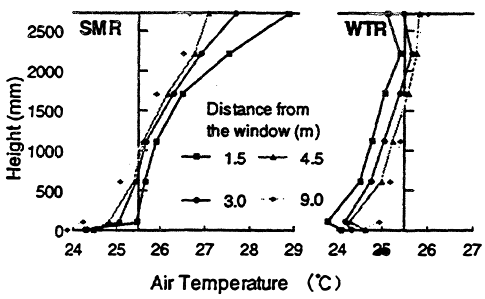

Figure 7 Vertical air temperature distribution near the window for summer and winter conditions

Air Velocity, PMV, and Skin Temperature of Thermal Manikin

Air velocities for all the conditions except "WTR" were kept under 0.1 m/s, well below the standard of ISO-7730[6] and ASHRAE 55-92[7]. For "WTR" condition, the cold draft caused an increase of air velocity below the window up to 0.15 m/s. All the PMV values based on 1.1 met and 0.6 clo fell between 0 and 0.5 except for "SMR" and "WTR". Careful measurements were conducted to investigate occupancy satisfaction. In the present experiment, skin temperature controlled thermal manikin was used for this purpose. A large number of data were collected for further analyses. A strong influence of radiation and ambient air temperature on the skin temperature of a thermal manikin can be confirmed in Figure 8.

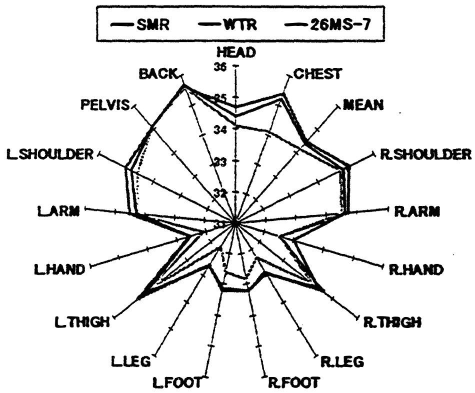

Figure 8 Skin temperature of thermal manikin, 1.5m from the window (℃)

DISCUSSION

The following conclusions and recommendations were drawn from the current study.

1. Heat transmission between the occupied space and underfloor plenum due to the floor cooling effect of the floor-supply displacement ventilation system kept the temperature gradient substantially low in the occupied zone compared to the wall-supply unit system.
2. The lower the height of the heat sources, the greater the vertical air temperature difference in the occupied zone. An increase of heat load or a decrease of supply air volume also

caused a greater vertical air temperature difference in the occupied zone.

3. A slight air temperature rise was observed near the clustered heat sources, but the temperature gradient was kept nearly constant throughout the room.
4. The thermal stratification near the window was destroyed under the summer condition. An effect of cold draft was observed under the winter condition, but the thermal stratification above 0.6m was kept undisturbed.
5. A strong influence of radiation and ambient air temperature on the skin temperature of a thermal manikin was observed to be significant.

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