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Comparison of Underfloor Vs. Overhead Air Distribution Systems in an Office Building

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ABSTRACT

Thermal environments, indoor air quality, and the ventilation effectiveness of an underfloor air distribution system were measured and evaluated in an actual office building with comparison of an overhead system in the same building with the same floor plan. The main findings were as follows.

1. The horizontal air temperature distribution in a room (3.6 ft [1.1 m] above floor level) showed a min-max difference of 2.9ºF (1.6ºC) and a standard deviation of 0.7ºF (0.4ºC) in summer for both systems; that is, the two systems were identical in this respect.

2. The evaluation of thermal nonuniformity showed that legs and feet tended to be cooler in the underfloor system, while the head tended to be cooler in the overhead one. However, in both cases, the results suggested that there was probably no discomfort arising from thermal nonuniformity.

3. The underfloor system indicated lower values in both concentration of mass and total number of airborne particles than the overhead system. However, both systems showed low levels of concentration compared to the acceptable value.

4. Measurements of the local ventilation effectiveness and local air change index showed that the ventilation performance of the underfloor system was better than that of the overhead system.

INTRODUCTION

The underfloor air distribution system was introduced in Japan in office buildings at the beginning of the 1990s, and many practical examples of the technology as an air-conditioning system have been established. Though this system was first introduced in Germany, original research and system development are required since the cooling load is bigger in Germany than in Japan. Many researchers have reported studies[1-15] on the performance of underfloor air distribution systems in Japan. The authors have also published the results of experimental studies on the pressure distribution and air temperature distribution under the floor and its effect on the thermal environment in the room[16] and have also published the results of field measurements.[17]

In experimental studies, underfloor systems can be compared to conventional overhead systems, and clear comparisons can be made. However, those experiments are usually carried out in small unrealistic chambers in the laboratory using heaters instead of an actual heat load such as workers, copy machines, etc.

On the other hand, direct comparison is usually difficult in field measurement, but it is possible to obtain valuable data with an actual heat load while an actual air-conditioning system is in operation. So far no comparative field measurements with conventional overhead systems in an actual building have been reported, although many field studies have been done and have reported fairly good thermal environments created by underfloor systems in actual offices.

The authors had an opportunity to carry out a comparative field measurement of two such systems. This paper describes the results of the comparative field measurements, and evaluates the performance of an underfloor air distribution system based on a direct comparison with an overhead air distribution system.

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DESCRIPTION OF AN OFFICE BUILDING FOR MEASUREMENTS

The structure in the present study is a large-scale office building that was completed in March 1994 in the Minato Mirai district of Yokohama, Japan.

The following is a brief description:

- Floor plate: 66,286 ft^2 (6,158 m^2)
- Total floor area: 1,194,273 ft^2 (110,948 m^2)
- Floor arrangement: 2 basement floors, 34 aboveground floors
- Structure: Steel and steel-reinforced concrete
- Height: 477 ft (145.5 m)
- Main use: Office, retail, exhibit areas

Offices are found on the 6th through 33rd floors. There are 10 floors from the 22nd floor upward that use an underfloor air distribution air-conditioning system. The other floors employ conventional "breeze line" office overhead air-conditioning with ceiling return system. Figure 1 illustrates both air-conditioning systems. In addition, fan coil units (FCUs) are employed in the perimeter area for both underfloor and overhead air-conditioning floors.

DESCRIPTION OF MEASUREMENTS

Air-Conditioning System and Heat Load in Measured Area

The areas measured were the 18th floor for the overhead air-conditioning system and the 23rd floor for the underfloor system. The measured areas on both floors were in the southern part of the west zone, as shown in Figure 2 (floor area, 2,799 ft^2 [260 m^2]; ceiling height, 8.9 ft [2.7 m]). The underfloor air conditioning in this building uses an underfloor pressurization system, consisting of a pressurization chamber under the OA floor. Return air is circulated through a line-type intake in the ceiling and return ducts, which are installed one per span, to circulate the air. The floor outlet employs whirling flow with air volume of 59 ft^3/min (100 m^3/h) at each installation (static pressure difference of 0.03 in. H_2O [7.3Pa] between the below-floor area and the room). The volume of the outlets can be changed manually. There were a total of 56 outlets in each measured area. For climate control in all of these rooms, the perimeter FCUs in both winter and summer were started up at 8:30 a.m.; then, at 9:00 a.m., the interior underfloor and overhead air-conditioning systems were switched on. The measured heat load for each floor is shown in Table 1.

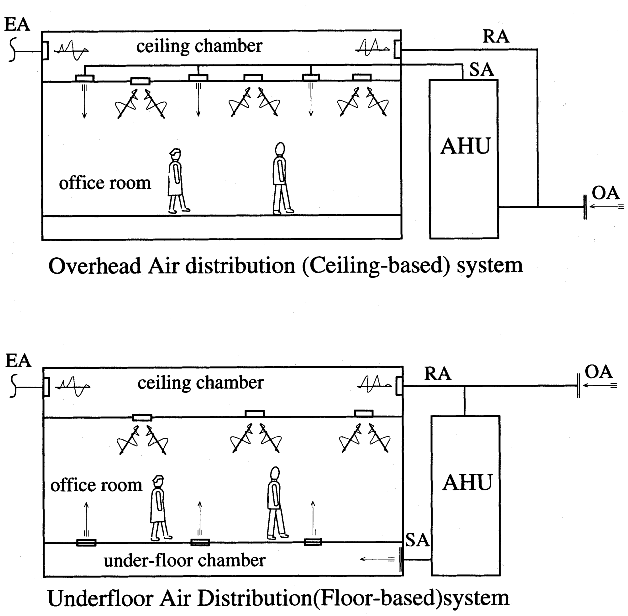


***Figure 1** Air-conditioning system.*

Measurement Items and Methods

Measurements were taken during two seasons—summer (early August 1994) and winter (early February 1995)—at the same locations. Table 2 shows the items, equipment, and methods used for the measurements, while Figure 2 shows the positions of measuring points on the underfloor air-conditioning floor and vertical air temperature measuring points. The same kind of setup was used for the overhead air-conditioning floor.

During the measurements, a fixed volume of air was supplied to the rooms for each type of system, and it was confirmed beforehand that each span was supplied about the same amount of air. Furthermore, when sunlight was shining through the windows, all blinds were drawn shut before measurements were taken.

Taking measurements in an actual building where people are working differs from laboratory experiments in that there are extraneous factors. For example, room temperature and the coming and going of people cannot be kept constant; moreover, severe restrictions are placed on intentional large changes in conditions. However, it is possible to obtain valuable data while an actual air-conditioning system is in opera-

TABLE 1
Heat Load in the Office

		Heat Load in the Office (W/m^2)
Floor-based system floor	human body	5 to 7
	lighting	25
	personal computer et al.	9 to 27
	sum	40 to 58
Ceiling-based system floor	human body	4 to 5
	lighting	25
	personal computer et al.	6 to 18
	sum	35 to 48

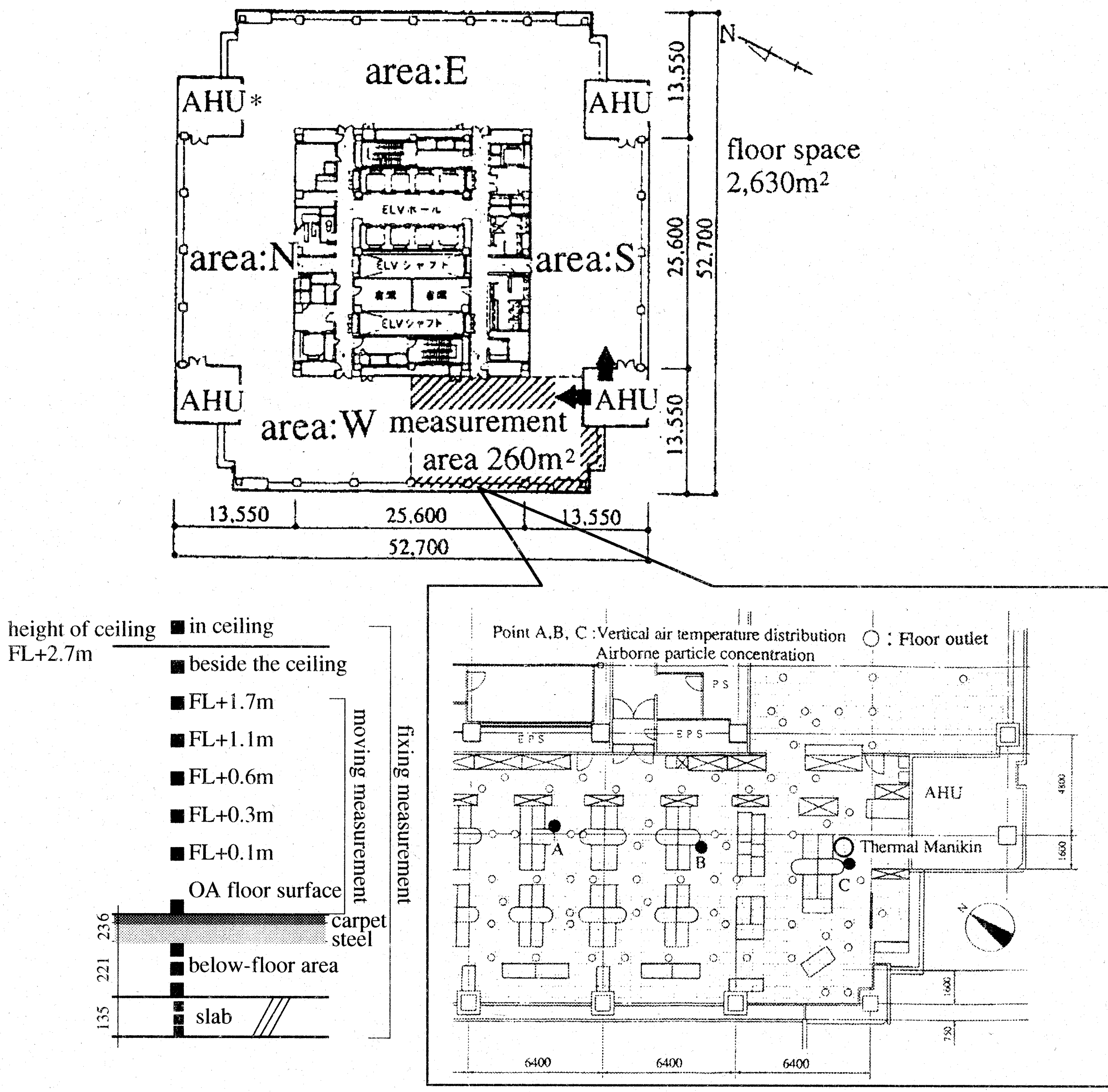


Figure 2 *Floor plan and measured areas with floor outlet layout (plan) on the underfloor air-conditioning floor and vertical temperature measurement points (*AHU is air-handling unit).*

TABLE 2
Items, Equipment, and Methods Used for the Measurements

Measurement Item	Measuring Equipment	Measurement Model
a) Air temperature below the floor	• Copper constantan T-type thermocouple 0.32 ϕ • Hybrid recorder	Placed in 30 measuring points in a ca. 3 m grid under the floor of the measurement zone shown in Figure 2. Continuous measurements taken at 10-minute intervals in summer and winter
b) Thermal environment in a room • vertical and horizontal temperatures • radiation temperature • airflow speed • relative humidity • temperature evaluation at human body parts using a thermal mannequin • survey of office workers	• Copper constantan T-type thermocouple 0.32 ϕ • Indoor climate analyzer • Data collector • Hybrid recorder • Thermal mannequin	1. *Fixed Point Continuous Measurements.* Continuous measurements were taken at fixed vertical measuring points at the points A, B, and C in Figure 2 for one week in summer and winter at 10-minute intervals. 2. *Mobile Measurements.* Measurements were taken at 23 points for three days each in summer and winter, three times a day (10:00, 2:00, 4:00). The indoor climate analyzer was set at FL + 1.1 m to measure air temperature, airflow velocity, relative humidity, and radiation temperature from six directions. In addition, a thermocouple was used to measure the vertical temperature at each of the mobile points. 3. *Thermal Mannequin.* Placed in a seated position in interior sections of the measured zones. When the amount of heat loss stabilized, heat loss was measured at various parts of the human body. 4. *Survey of Office Workers.* A total of six written surveys were distributed three times a day (10:00, 2:00, 4:00) for two days to all workers in the measured zones.
c) Room air quality environment • concentration of airborne particles • survey of office workers	• Digital dust counter • Laser particle counter	A digital dust gauge was set up at a typical point (near Point C in Figure 2) to provide a continuous record during the measurement period. Coinciding with the survey distribution, the laser particle counter was used to measure the number of particles 0.3 μ m or larger at Points A, B, and C. A survey similar to the room heat environment survey was administered.
d) Ventilation capacity (winter only) • local air ventilation efficiency *1 • air purity index*2	• Multi-gas monitor	SF_6 was used as a tracer gas and a polytetrafluorethylene tube was placed at each point to take samples. Age of air used in calculating air ventilation efficiency was measured at each point using the step-up and step-down methods. To measure static concentrations, SF_6 was pumped into roughly the center of a room and measurement commenced when the concentration had stabilized at each point. Calculations were derived from concentration standards for supplied air.

tion. The present measurements were designed to examine an actual spatial environment under actual operations. Measurements were taken in two different seasons (summer and winter) for one week each, and intensive and detailed continuous measurements were taken for three days during each session.

RESULTS AND DISCUSSION

Air Temperature Distribution Under the Floor

An underfloor air distribution system is characterized by the fact that, in summer, the temperature of the cool air supplied to the space under the floor increases as it absorbs heat from the concrete slab and room. Figure 3 shows the air temperature distribution under the floor in summer and winter. At the far side edges of the measurement area, the summertime temperature was 75ºF to 77ºF (24ºC to 25ºC), and the temperature increase rate in the flow direction from the below-floor air outlet was 0.15ºF/ft (0.28ºC/m), which is in close agreement with previous results because of high heat load.[18] In winter, air conditioners were also running under the cooling condition in the interiors of measured floors, and the rate of increase in the X direction was 0.08ºF/ft (0.15ºC/m). The main reason why the increase was less in winter is that the surface temperature of the floor slabs near the center of the rooms was 74.5ºF (23.6ºC) in summer and only 68.0ºF (20.0ºC) in winter; the difference between this temperature and the temperature of the air supplied toward the space under the floor (supplied air temperature in Figure 3) was less in winter. The below-floor temperature should not be increased erratically so that the temperature of the air being supplied into the office area will be consistent. However, if the effect it has on the thermal environment for workers within an office is small, this does not pose a problem. The next section will evaluate the horizontal air temperature distribution at FL + 3.6 ft (FL + 1.1 m).

Thermal Environment

Horizontal Thermal Environment Distribution. Figure 4 shows the horizontal temperature distribution at FL + 3.6 ft (FL + 1.1 m). In summer, for underfloor air-conditioning rooms, the temperature difference between the machine room-side zone (below-floor air outlet side) and the far side of the zone was about 1.8ºF (1.0ºC), while there was a 9.0ºF (5.0ºC) difference with the below-floor air temperature immediately under the same points.This indicates that the temperature distribution in the room was fairly consistent compared to that under the floor. These results are quite close to those of laboratory experiments[16] and other measurements.[18] The average air temperature in the rooms in summer was 75.9ºF (24.4ºC), with a 2.9ºF (1.6ºC) difference between minimum and maximum and a standard deviation of 0.7ºF (0.4ºC). In winter, the average air temperature in the rooms was 73.2ºF (22.9ºC), with a 0.9ºF (0.5ºC) difference between minimum and maximum and a standard deviation of 0.4ºF (0.2ºC). Thus, winter air temperature distributions were within a narrower range.

In Figure 4, we can also see that in summer, the average air temperature was 77.0ºF (25.0ºC), with a 2.9ºF (1.6ºC) difference between minimum and maximum and a standard deviation of 0.7ºF (0.4ºC). In winter, the average air temperature was 74.7ºF (23.7ºC), with a 2.0ºF (1.1ºC) difference between minimum and maximum and a standard deviation of 0.5ºF (0.3ºC). These results tell us that there were no differences between the two air-conditioning systems. The apparent reasons why the horizontal air temperature distributions between the two systems were quite similar involved the flow and mixing of supplied air and the dispersion of heat. These factors are also examined in some of the references.[19, 20] Another reason was that the amounts of air supplied into the room, the heat load within a room, etc., were roughly equal in all measured areas.

There were no great differences in air temperature distribution, but in our evaluation of the thermal environment, we need to take a holistic approach, which includes the distributions of airflow and radiation temperature. Table 3 shows the actual measured values for each floor and season by the measurement item. Here we calculated PMV using the mean radiant temperatures (defined in the footnote of Table 3) from the measured radiation temperatures in six directions shown in Table 4. PMV value differentials were 0.06 in summer and 0.16 in winter—not large by any measure—and both were close to neutral. Table 3 also shows the LPPD proposed by Fanger[21] (less than 6% is the recommended value) to evaluate the distribution of PMV. As we can see, the 6.3% barely exceeds the recommended 6% on the overhead floor in summer, but all other values are at the 5% level; thus, overall, there is not much nonuniformity in either the underfloor or overhead systems, and no large difference is seen on either floor. It should be noted that in underfloor air-conditioning systems, sometimes the coolness of the floor might result in

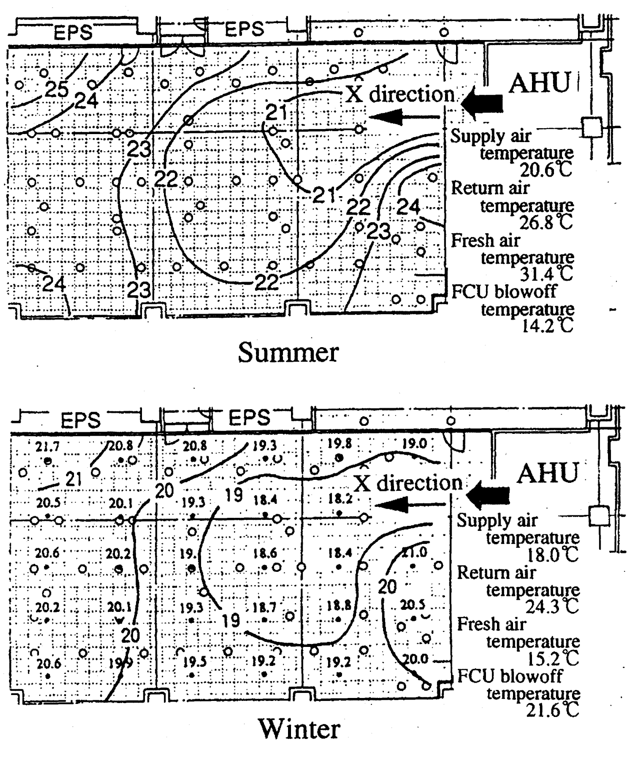


Figure 3 *Air temperature distribution under the floor on a underfloor system floor.*

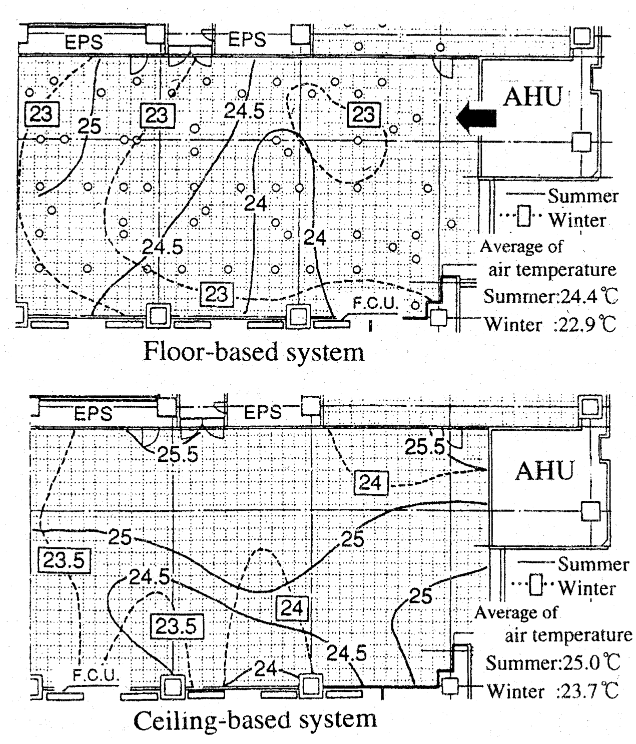


Figure 4 *Horizontal distribution of air temperature at FL + 1.1 m.*

TABLE 3 *
Result of Each Measurement Item

		PMV[*1]	PPD	LPPD	Temperature ºC	Relative Humidity %	Air Velocity m/s	MRTºC[*2]
Summer	Ceiling-based system floor	0.19	7.4	6.3	24.9	65	0.19	26.7
	Floor-based system floor	0.12	5.8	5.6	24.4	60	0.12	26.0
	Difference	–0.06	–1.5	–0.7	–0.5	–5	–0.07	–0.8
Winter	Ceiling-based system floor	0.12	6.4	5.6	23.6	36	0.15	24.7
	Floor-based system floor	–0.04	5.6	5.4	22.4	42	0.12	23.9
	Difference	–0.16	–0.8	–0.2	–1.1	6	–0.03	–0.9

* 1) summer: 0.5 clo, winter: 0.8 clo, 1.2 met
* 2) Here, radiation temperature is defined by the following equation:

$$t_r = \{0.18(t_{pr,u} + t_{pr,d}) + 0.22(t_{pr,r} + t_{pr,l}) + 0.30(t_{pr,f} + t_{pr,b})\}/2(0.18 + 0.22 + 0.30)$$

where *u* is up; *d* is down; *r* is right; *l* is left; *f* is front; and *b* is back.

TABLE 4
Radiant Temperature from Each Direction (ºC)

		Ceiling	Floor	East	North	West	South
Summer	Ceiling-based system floor	27.2	26.2	26.8	26.6	27.0	26.6
	Floor-based system floor	26.9	24.9	25.8	26.1	26.3	25.8
	Difference	–0.3	–1.3	–1.1	–0.5	–0.7	–0.8
Winter	Ceiling-based system floor	25.2	24.2	24.7	24.9	24.9	24.6
	Floor-based system floor	25.0	22.7	23.7	24.0	23.9	23.8
	Difference	–0.2	–1.4	–1.0	–0.9	–1.0	–0.8

discomfort due to cold radiation. Table 4 shows the radiation temperature from each direction; in each case, the radiation temperature on the underfloor air-conditioning floor was lower than that on the overhead air-conditioning floor, and the differences in radiation temperatures from the floor surface were greater than from other directions. However, as we can see in Table 3, air temperature for the underfloor air-conditioning floor was 0.9ºF (0.5ºC) lower in summer and 2.0ºF (1.1ºC) lower in winter than those for the overhead AC floor; taking this into account, we can see that, practically speaking, the differences in radiation temperature from the floor were not so large. Further, in terms of mean radiant temperature, we can see that there is actually very little difference between the two systems. Furthermore, if we consider the radiation temperature that is lower than air temperature as cold radiation, the averaged radiation temperature from the floor surface for the underfloor air-conditioning floor is a little higher than the air temperature, so we cannot really label this as cold radiation. This is probably due to the effect of mutual radiation exchange with the ceiling that has light bulbs on it.

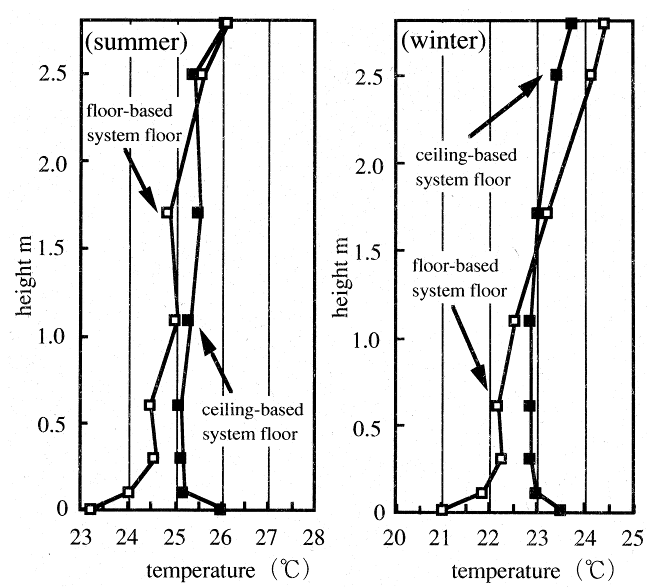


Figure 5 *Vertical air temperature distribution.*

Vertical Temperature Distribution

Figure 5 shows the vertical temperature distribution at Point B in the center of the room. ASHRAE Standard 55-1992 recommends that the difference in temperature between 0.33 ft (0.1 m) and 5.6 ft (1.7 m) above the floor be within 5.4ºF (3.0ºC); ISO7730 recommends the 5.4ºF (3.0ºC) maximum between 0.33 ft (0.1 m) and 3.6 ft (1.1 m). We can see in the figure that these differences in the underfloor system were 1.4ºF (0.8ºC) in summer and about 2.3ºF (1.3ºC) in winter; for the overhead system, they were less than 0.9ºF

(0.5ºC) in both seasons. These results show that in all cases, the temperature difference was well within the recommended limits. The vertical temperature distribution in the space for workers depends on the shape of the floor outlets, the temperature of the air being supplied, the amount that is supplied, and the Archimedes number of the air being supplied. Figure 6 shows the relationship between $\Delta T/Q^2$ obtained from the experiments[16] and measurements,[17] and the difference in vertical air temperature, including the data from the present study. In references 19 and 22, the difference in vertical air temperature is depicted as the relationship with the Ar number. Here, in order to compare the performance of differently shaped outlets, we established the parameter $\Delta T/Q^2$, where ΔT is the difference between room temperature and supplied air temperature, and Q is the volume of air coming from one outlet. Figure 6 shows the relationship between $\Delta T/Q^2$ and the vertical air temperature difference. Since the present study was conducted under controlled conditions, in which the supply air temperature was kept at 64.4ºF (18.0ºC) or more, $\Delta T/Q^2$ tended to concentrate around relatively small values, and a good correlation was not obtained; nevertheless, judging from the plot in the figure, we can assume that the outlet used in the present study (spiral grooves, whirling air flow) appears to be a kind of cross between the outlet with slanted, whirling airflow used in the experiments and the outlet with vertical, whirling flow that is used in another measured building.[18]

Thermal Environment Evaluation Using a Thermal Manikin

A thermal manikin is a device that is used to evaluate thermal nonuniformity in a room. The heat loss at each position of the thermal manikin is often used to calculate the equivalent homogeneous temperature or EHT.[23, 24] EHT is defined as the temperature of a uniform enclosure in which a manikin would lose heat at the same rate as it would in the actual nonuniform environment.[25] Here, the manikin was set up in interior sections of both the underfloor and overhead air-conditioning floors to measure the heat loss. Figure 7 shows the EHT at each position of the manikin and the deviation (DEHT) from the EHT of the entire body in summer (76.8ºF [24.9ºC] for underfloor, 77.9ºF [25.5ºC] for overhead systems). From these results, we can see that the cool area in the underfloor air-conditioning floor was at the thighs and below, while it tended to be at the head in the overhead air-conditioning floor. Excluding the waist for the overhead system, the data obtained were almost in the comfort range shown by DEHT between –3.6ºF (–2ºC) and +3.6ºF (+2ºC), indicating that there was no large thermal nonuniformity in either system. The very large value at the waist for the overhead system may be due to the difference of contact on the chair from that in the experiment in which reference values were obtained.

Thermal Sensation Survey

Table 5 shows the results of a survey on thermal sensation for the entire body and for its various parts, as well as the significant test results.

1. *Thermal sensation at each part of the body:* Figure 8 shows the results of subjective evaluations for thermal sensation at each part of the body. Significant differences were apparent in the head and hands in summer, and thermal sensation in the overhead distribution system tended to be cool. For winter, significant differences were seen at all body positions except for the thighs; in the overhead system, the

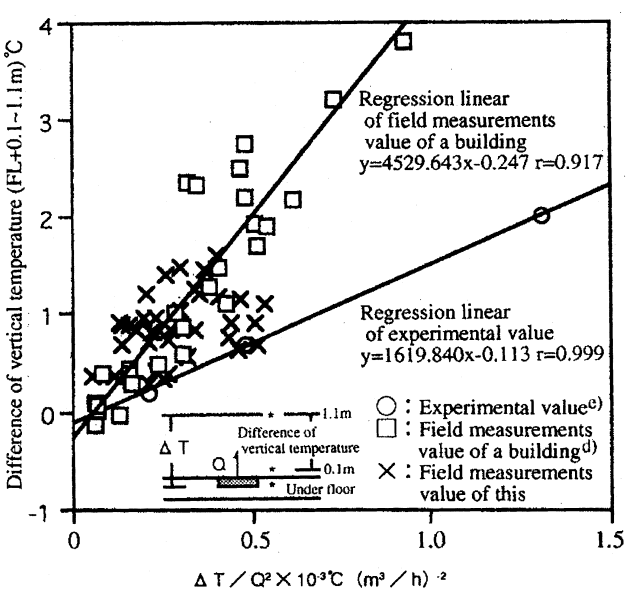


Figure 6 *Relationship between $\Delta T/Q^2$ and difference in vertical temperatures.*

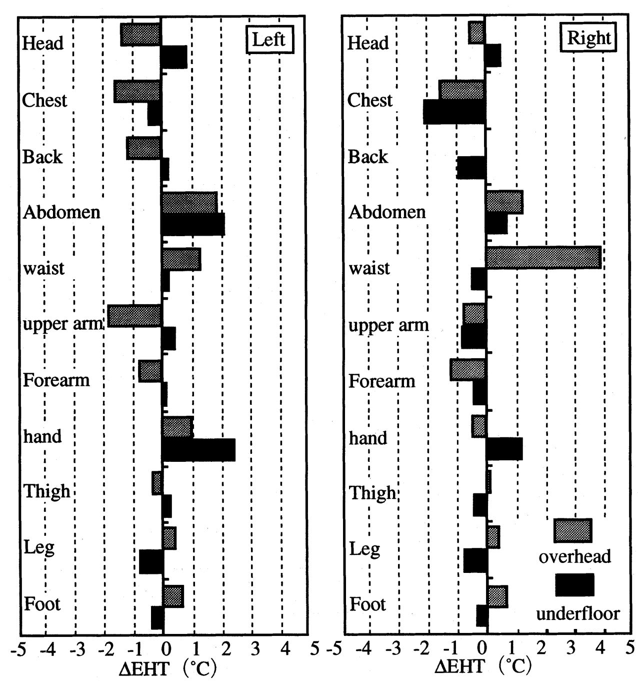
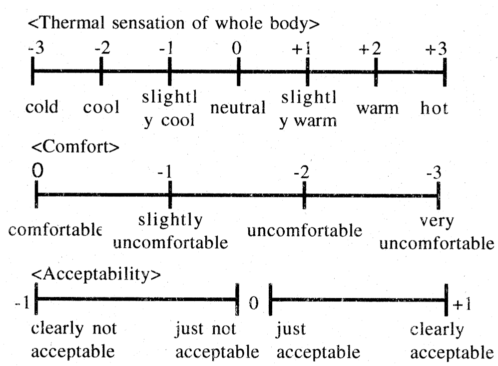


Figure 7 *Distribution of ΔEHT (summer).*

TABLE 5
Result of Significance of Thermal Sensation of Each Part and Whole Body

		Thermal Sensation of Whole Body	Comfort	Acceptable	Thermal Sensation of Each Part					
					Head Face	Shoulder Waist Abdomen	Hand Arm	Thigh	Leg	Foot
Summer	Ceiling-based system	–0.40	–0.44	0.40	–0.22	–0.32	–0.54	–0.27	–0.33	–0.32
	Floor-based system	–0.09	–0.31	0.58	0.21	–0.10	–0.21	–0.10	–0.20	–0.15
	Difference	0.31	0.13	0.18	0.43	0.22	0.33	0.17	0.13	0.17
	Significance	❍	×	×	❍	×	❍	×	×	×
Winter	Ceiling-based system	–0.61	–0.69	0.20	–0.36	–0.48	–0.50	–0.41	–0.40	–0.60
	Floor-based system	–0.35	–0.52	0.14	0.13	–0.18	–0.20	–0.47	–0.67	–0.89
	Difference	0.26	0.17	–0.06	0.48	0.30	0.30	–0.06	–0.26	–0.29
	Significance	Δ	❍	×	❍	❍	❍	×	❍	❍

❍: Significance of 5% Δ: Significance of 10% ×: No significance



Clo value (assumed value) summer: 0.5 clo, winter: 0.8 clo
Met value (assumed value) summer and winter: 1.2 met
Head count of answerer of ceiling-based system floor
Summer male: 30 female: 13 Winter male: 35 female: 16
Head count of answerer of floor-based system floor
Summer male: 32 female: 12 Winter male:49 female:19

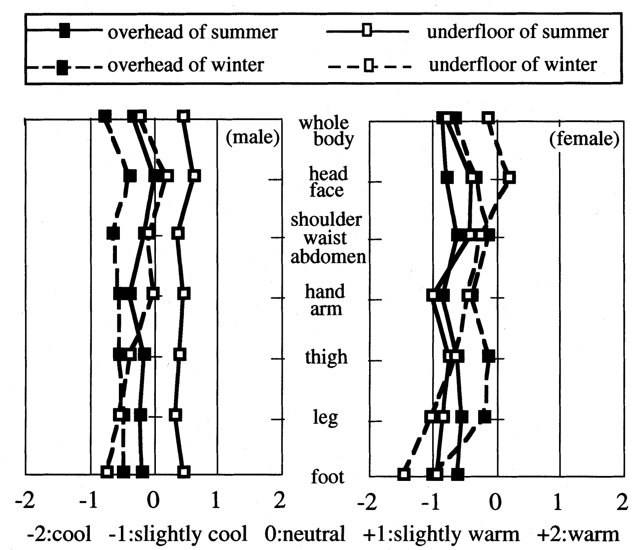


Figure 8 *Thermal sensation of each part of the body.*

upper part of the body tended to feel cool, while the lower body was cooler in the underfloor system.

2. *Acceptability with thermal environment:* Figure 9 shows the results of subjective evaluations for acceptability with the thermal environment for both systems in both seasons. For both systems, about 90% of respondents were pleased with the temperature in summer, while only 75% of respondents expressed satisfaction with the winter thermal environment. Table 5 shows no significant differences in terms of acceptability between the two AC systems.

Based on the chart for the draft risk in ASHRAE Standard 55,[26] the possibility of draft discomfort was examined. According to the draft chart, assuming turbulence intensity is 30%, the velocities for 15% discomfort sensation are about 0.16 m/s at 69.8°F (21°C), 0.165 m/s at 71.6°F (22°C), 0.175 m/s at 73.4°F (23°C), 0.185 m/s at 75.2°F (24°C), and 0.20 m/s at 77.0°F (25°C). In both systems and in both seasons, the average velocities in Table 3 are lower than those 15% discomfort levels at each measured temperature level. In Figure 5, the floor surface temperature at 0 m is pretty low and about 21°C for the underfloor system in winter. However, the measured air velocity (0.12 m/s at 1.1 m level as shown in Table 3) is lower than the criterion (0.16 m/s at 69.8°F [21°C]). Thus, as far as we employ 0.12 m/s as the air velocity at foot level, the discomfort level at foot level would not be very high. As a matter of fact, even though the thermal sensation at the feet is –1.5 for underfloor in winter as shown in Figure 8, the acceptability for the thermal environment is the same as that for the overhead system as shown in Figure 9.

Indoor Air Quality Environment

Some people who work in rooms with an underfloor air-conditioning system are concerned about airborne particles being circulated in the air. However, previous measured data[18,27] and laboratory data[28, 29] have shown that the air quality is somewhat better than that from an overhead system. Airborne particles in the present study were measured with a halogen lamp dust counter and a laser scanning particle counter, and an air quality survey was also taken.

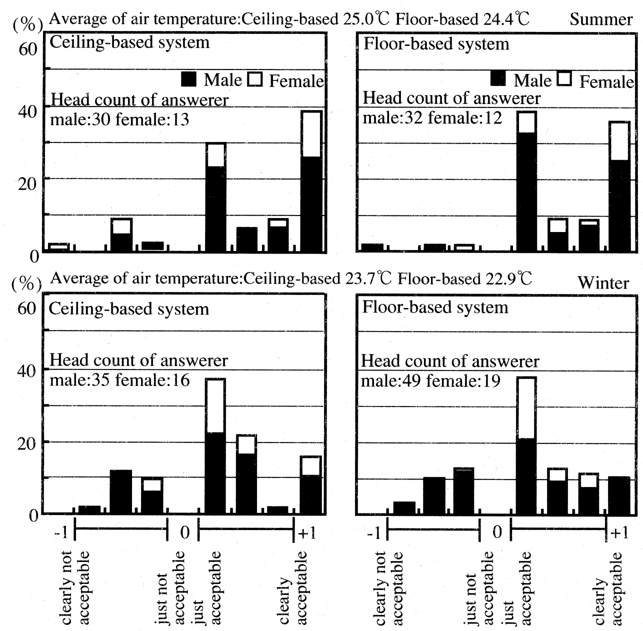


Figure 9 *Results of acceptability for thermal sensation.*

Airborne Particle Concentrations

Figure 10 shows changes in airborne particle concentrations over time. The halogen lamp dust counter showed these concentrations to be from 0.075 to 0.19 × 10-6 lb/ft^3 (0.012 to 0.03 mg/m^3), which is well within the acceptable value of 0.94 × 10-6 lb/ft^3 (0.15 mg/m^3) or less set by the Building Control and Health Law in Japan. The main reason for the low concentrations of airborne particles is that smoking is prohibited in the rooms and is only allowed in designated areas in the NW and SE corners of each floor. The overhead system tended to circulate more dust. The laser scanning particle counter indicated the value for underfloor systems to be 1 million particles per cubic foot (35 million particles per cubic meter) for 1 µft (0.3 mm) or higher, while the values for overhead systems were 3.5 to 4.5 million particles per cubic foot (123 to 158 million particles per cubic meter). The likely reason for the lower particle values for underfloor systems is that in underfloor systems, air coming from the outlets does not mix as much with room air and does not spread contaminants throughout the rooms. That is also clear from the measured local ventilation effectiveness in the subsequent sections, as well as references 28 and 29, in which it was made clear that the underfloor system tends to circulate less contaminants and has a high attenuation of contaminants.

Acceptability with the Air Quality Environment

Table 6 lists the results of significance tests for the air quality for the both air-conditioning systems using the survey results. The table also shows the level of subjective evaluations. Here we can see that in terms of air quality sensation, comfort, and satisfaction, there are no large differences in the sensation level between the systems.

Figure 11 shows the results of subjective evaluations for acceptability of the air quality environment. Sensation votes for an acceptable environment were 90% or more in summer vs. 80% in winter. These results indicate that differences in amounts of airborne particles did not appear in the sensation survey. The reason for this is that in terms of absolute amount, there were few floating particles in both systems, which were

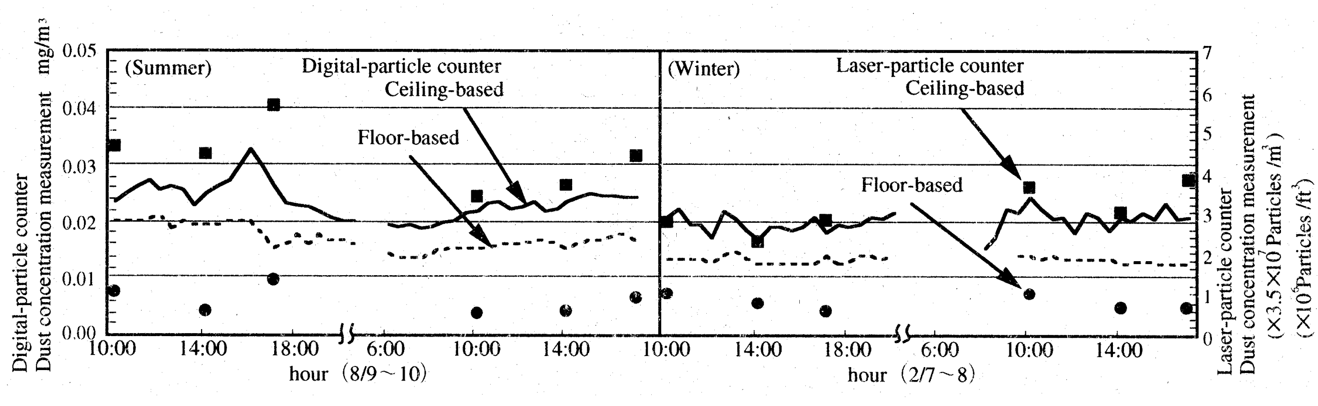
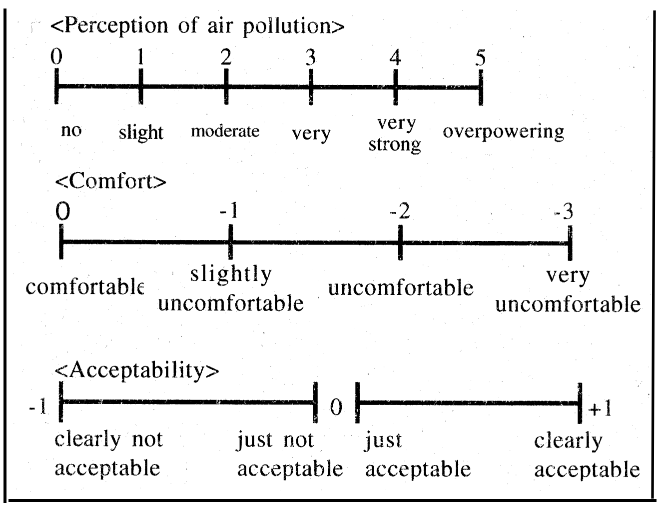


Figure 10 *Changes in airborne particle concentrations over time.*

TABLE 6
Result of Significance for Air Quality

		Perception of air pollution	Comfort	Acceptable
Summer	Ceiling-based system	0.35	–0.30	0.53
	Floor-based system	0.21	–0.15	0.56
	Difference	–0.14	0.15	0.03
	Significance	Δ	×	×
Winter	Ceiling-based system	0.38	–0.32	0.31
	Floor-based system	0.34	–0.35	0.32
	Difference	–0.04	–0.03	0.01
	Significance	×	×	×

Δ: Significance of 10% ×: No significance



generally of a diameter of 33 μft (10 mm) or less, and also those were thus not visually apparent.

Evaluation of Ventilation Performance

In the present study, ventilation performance was evaluated with two indices: local air change index*[1] and local ventilation effectiveness*[2].

Local Air Change Index.

a. *Measurement method:* Tracer gas (SF_6) was pumped at a fixed volume (0.9 L/min) into outdoor air intake ducts of air conditioners for measured zones. Time series data for concentrations at three points (A, B, and C in Figure 2) and in the exhaust, supplied air, and the adjacent space were obtained with a gas monitor during the step-up and step-down measurements. The amount of outdoor air was measured with an anemometer. For step-up measurements, when the increase in concentrations in the exhaust became 8% less than it had been an hour before,[30] the condition was considered to stabilize.

b. *Calculation results:* Nominal ventilation time τ_n was derived from the external air volume and was used to calculate the local air change index ε_p. Table 7 shows the calculated results. The underfloor system shows higher values than the overhead system, indicating it has better ventilating performance.

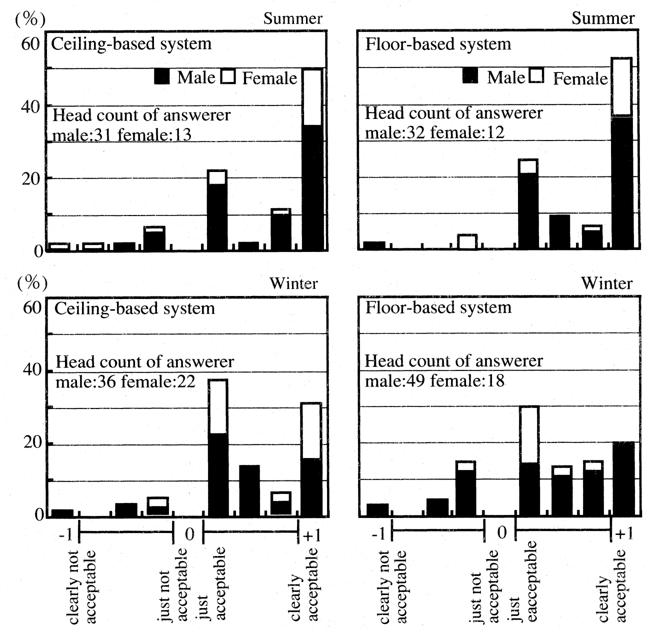


Figure 11 Results of acceptability for air quality.

Local Ventilation Effectiveness.

a. *Measurement method:* SF_6, a tracer gas, was pumped into roughly the center of a room, then, at the time when the concentrations in the exhaust had stabilized, several measurements were taken of gas concentrations at each interior point in Figure 12 (▲ indicates, FL + 3.6 ft [FL + 1.1 m]). Measurements of concentrations in supplied air were also made.

b. *Calculation results:* Figure 12 shows the horizontal distribution of the local ventilation effectiveness. Here we can see that the underfloor air-conditioning system tended not to spread contaminants throughout the room.

CONCLUSIONS

To empirically evaluate an underfloor air distribution system, comparative measurements of room thermal and air quality environments were taken with an overhead system and an underfloor system in the same building with the same floor plan. The main findings were as follows:

TABLE 7
Local Air Change Index for Each Nomial Ventilation Time

	Ceiling-Based System					Floor-Based System				
Measurement time min	355					322				
Measurement point	Exhaust air	Supply air	A	B	C	Exhaust air	Supply air	A	B	C
Final concentration ppm	31.9	34.5	32.4	33.8	34.1	27.8	33.5	33.5	34.1	34.8
Age of air min	98.5	76.0	98.1	86.1	86.0	94.5	75.6	87.0	80.6	78.1
Nominal time constant min	74.9					79.6				
Local air change index	0.76	0.99	0.87	0.87	0.74	0.84	1.05	0.92	0.99	1.02
Room mean air change index	0.83					0.97				

1. Thermal Environment

    a. The horizontal air temperature distribution in a room (FL + 3.6 ft [FL + 1.1 m] above floor level) showed a minimum-maximum difference of 2.9ºF (1.6ºC) in summer for both systems and a standard deviation of 0.7ºF (0.4ºC); that is, the two systems were identical in this respect.

    b. LPPD, the horizontal distribution of temperature sensation that was derived from PMV, exceeded the recommended 6% level (at 6.3%) in summer in the overhead system; but in winter both systems were less than 6%, and there was no significant difference between them.

    c. In terms of vertical air temperature distribution, the underfloor system showed a greater difference (maximum difference of 2.0ºF [1.1ºC] at 3.6 to 5.6 ft [1.1 to 1.7 m] above floor level). Looking at the effect of this vertical difference in terms of thermal sensation for various body parts, significantly cooler sensation was obtained in the legs and feet with the underfloor system than with the overhead system in winter.

    d. ΔEHT showed that legs and feet tended to be cooler with the underfloor system, while the head tended to be cooler with the overhead one. However, in both cases ΔEHT was almost within ±3.6ºF (±2.0ºC), indicating that there was probably no discomfort arising from thermal nonuniformity.

    e. The results of a questionnaire survey on the thermal environment of an entire space showed that there was no significant difference in the two systems in terms of acceptability, with 75% to 90% of respondents expressing their acceptance. However, in winter, the underfloor air-conditioning system was viewed as slightly more comfortable than the overhead one.

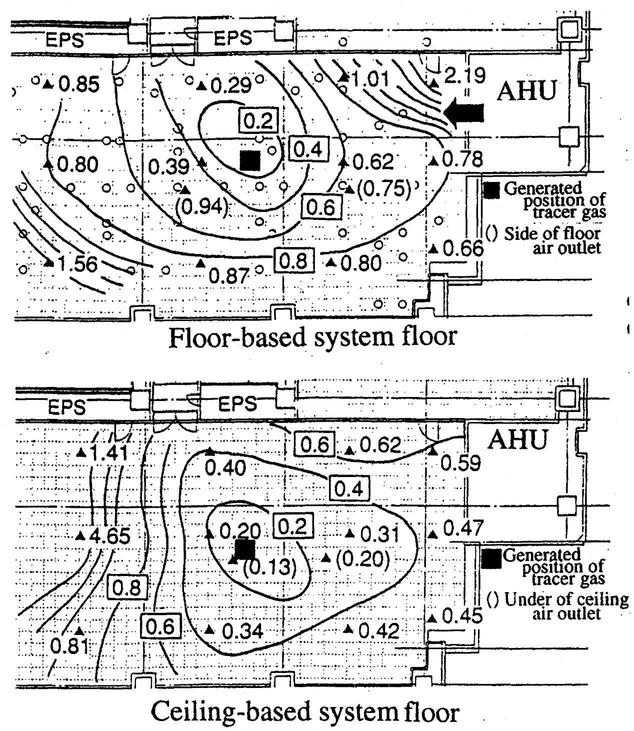


Figure 12 *Horizontal distribution of local ventilation effectiveness.*

2. Air Quality Environment

    a. The underfloor system was lower in both concentration of mass and total number of airborne particles than the overhead system. However, both systems showed low levels of concentration compared to the acceptable value.

b. Measurements of the local air change index and local ventilation effectiveness showed that the ventilation performance of the underfloor system was better than that of the overhead system.

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