Kobayashi, N. and Chen, Q. 2003. "Floor-supply displacement ventilation in a small office," *Indoor and Built Environment*, 12(4), 281-292.

Floor-Supply Displacement Ventilation in a Small Office

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Key Words
Displacement ventilation · Computational fluid dynamics · Experimental measurements · Floor supply · Indoor air quality · Thermal comfort

Abstract
This paper studied the performance of a floor-supply displacement ventilation system using computational-fluid-dynamics (CFD). The experiment was carried out in a full-scale environmental chamber with a floor-supply system to obtain reliable flow information for the validation of a CFD program. The validated program was used further to evaluate the performance of the floor-supply displacement ventilation system with different air change rates, diffuser numbers, furniture arrangement, and cooling loads. The evaluation criteria are thermal comfort level and indoor air quality.

Nomenclature

A_{object}	surface area of a heated object
C	contaminant concentration in the room air
C_e	contaminant concentration in the exhaust air
C_s	contaminant concentration in the supply air
G	turbulent production
G_B	turbulent production due to buoyancy
g_i	gravity in i-direction
k	turbulent kinetic energy
L	the thermal load on the body

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p	air pressure
$Q_{convection}$	convective part of the heat flow from a heated object
$Q_{radiation}$	radiative part of the heat flow from a heated object
Q_{totak}	total heat flow from a heated object
S_C	concentration source
S_T	heat source
S_ϕ	source term
T	room air temperature
T_{air}	air temperature around a heated object
T_o	reference air temperature
T_{object}	surface temperature of a heated object
Tu	turbulence intensity
T_{walls}	surface temperature of the surrounding walls
t	time
u	air velocity [m/s]
x_j	coordinate in j direction (j = 1, 2, 3)
β	air expansion coefficient
Γ_ϕ	diffusion coefficient
ε_{object}	surface emissivity of the heated objects
ε_{wall}	surface emissivity of the surrounding walls
η	ventilation effectiveness
μ	laminar viscosity
μ_t	turbulent viscosity
ρ	air density
σ	Stefan-Boltzmann constant
τ	the mean age of air
ϕ	dependent variables

Introduction

Research on the indoor environment has received more attention recently because of the increase in reports of symptoms and other health complaints related to the indoor environment. The majority of health problems reported in buildings, which are nonspecific complaints sometimes called the “sick building syndrome,” cannot be attributed to specific exposures. Available evidence suggests that multiple factors, including indoor air quality, are involved.

Indoor air quality is related to HVAC systems. Most of the HVAC systems used in the U.S. buildings are mixing systems. The design assumes that fresh air delivered by the HVAC systems would completely mix with the indoor pollutants to reduce their concentration level to be acceptable. In fact, complete mixing is difficult to achieve. The concentration level in some parts of an indoor space often exceeds the permitted level. In addition, the complete mixing would cause cross-infection between occupants.

To improve indoor air quality, displacement ventilation seems to be a good alternative. The system provides fresh air directly to the occupied zone. Therefore, compared to conventional mixing ventilation, the advantage of displace ventilation is that generally it can create better indoor quality, such as lower contaminant concentration, in the occupied zone. The most common system is a horizontal discharge system from a low sidewall, as reviewed by Yuan et al. [1]. However, the airflow from the displacement ventilation system is not one-dimensional in the occupied zone; there are slow recirculations in the lower part of the room [2]. These recirculations can cause cross-infection between occupants. A floor-supply displacement ventilation system could eliminate the recirculations. A large-scale measurement confirmed that the floor-supply system could provide very good indoor air quality [3]. In the past few years, the system has been widely studied in different parts of the world [4-9].

Sodec and Craig and other researchers [9-11] reported that the under-floor air supply system generated an S-shaped vertical temperature pattern. The air temperature in the lower part of a room is rather uniform. These studies found that the air temperature gradient in an indoor space with the floor-supply displacement ventilation system depends on

- measuring point position (horizontal temperature distribution)
- cooling load
- ventilation rate
- diffuser shape

However, this system may have potential problems when it is applied to indoor spaces with high cooling loads. With a high cooling load, the room air temperature gradient becomes very high as well and would not be acceptable in terms of thermal comfort [11]. However, if the air supply is increased to reduce the temperature gradient, the supply air velocity would be higher and the risk of cold drafts would be higher. This is because the cold air is directly supplied to the occupied zone. In addition, the energy consumption by the HVAC system would be higher due to the high ventilation rate. Nevertheless, the floor-supply ventilation system is still superior to the sidewall displacement system because it can also be used directly for heating.

Despite many studies that are available for the floor-supply displacement ventilation systems, their performance has not been thoroughly understood. The objective of this research is to systematically study the impact of ventilation rates, supply air temperature, diffuser location, exhaust location, occupant location, and furniture and partition arrangement, on the thermal comfort level and indoor air quality. The research used a Computational Fluid Dynamics (CFD) program. Detailed experimental data obtained in a full-scale environmental chamber with a floor-displacement ventilation system were used to validate the CFD program. The results from this investigation can then be used for the designing of the optimal system.

Research Approach

In general, two main research methods are available to study the indoor environment: experimental investigation and computer simulation. Experimental measurements give the most realistic information concerning indoor air quality and thermal comfort - such as the distributions of air velocity, temperature, relative humidity, and contaminant concentrations. To obtain quality results, the experiment requires an environmental chamber with a controlled indoor and outdoor environment, as well as well-equipped sensors and data acquisition systems to measure air velocity, temperature, relative humidity, and contaminant concentrations. Furthermore, the chamber should also have flexibility in changing from one spatial configuration to another. In order to obtain detailed and meaningful data measurements, a complete measurement takes time. Therefore, the experimental measurements are expensive and time consuming.

On the other hand, the indoor air quality and thermal comfort can also be determined computationally by solving a set of conservation equations describing the flow, energy, and contaminants in the indoor environment. Due to the limitations of the experimental approach and the increase in the performance and affordability of high-speed computers, the numerical solution of these conservation equations provides a practical option for computing the airflow and pollutant distributions in buildings. This method is known as the Computational Fluid Dynamics (CFD).

The CFD is a powerful tool for solving indoor environmental problems, such as airflow pattern and the distributions of air velocity, temperature, relative humidity, turbulent intensity, and contaminant concentrations. The CFD program that was used [12] solves the following governing equations for the conservation of mass, momentum, heat, species concentrations, and turbulence quantities:

$$\frac{\partial}{\partial t}(\rho\phi) + \frac{\partial}{\partial x_j}(\rho u_j \phi) = \frac{\partial}{\partial x_j}(\Gamma_\phi \frac{\partial \phi}{\partial x_j}) + S_\phi \qquad (1)$$

Table 1 provides detailed information concerning the different terms in Equation (1).

Because of the limited computer power and capacity available at present, turbulence models are used in CFD in order to solve indoor airflows on a PC. The present study used the re-normalization group (RNG) k-ε model [13]. The governing equations are highly non-linear and self-coupled. They are solved by discretizing the equations by using the finite volume method with the SIMPLE algorithm [14].

However, the use of turbulence models leads to uncertainties in the computed results, because the models do not have universal validity for all kinds of indoor spaces. Therefore, it is essential to validate a CFD program by using appropriate experimental data.

Many experimental data are available in the literature, but very few of them can be used for validation. This is because experimental data for CFD validation must contain detailed information concerning the flow and thermal boundary conditions, as well as the flow and thermal parameters measured in the space. In addition, the flow and heat transfer features of the airflow measured must be similar to those to be studied [15]. For example, the experimental data used for the present study should come from a displacement ventilation system. Therefore, we have conducted experimental measurements of the floor-supply displacement ventilation system to obtain the data to validate the CFD program. Since the experimental data are used only for the CFD validation, only a few measurements are need for typical cases. Then the validated CFD program can be used to study different problems in a room with floor-supply displacement ventilation. This effort would save time and costs compared with the research uses only experimental measurements.

CFD Validation and Experimental Data

This validation is to demonstrate the ability of the CFD program to accurately predict an indoor environment with displacement ventilation by using reliable data. It shows the combined ability of the CFD program and the user in predicting indoor environments in the real world, according to the validation procedure developed by ASHRAE [16]. The validation strategy is to first identify suitable experimental data, then to make sure that all the important physical phenomena in the problem of interest are correctly modeled, and finally, to quantify the error and uncertainty in the CFD simulation.

Experimental Facility and Measurements

The present investigation used an environmental chamber to simulate a two-person office, as shown in Fig. 1. The chamber is a well-insulated box (R = 5 K m^2/W) with a movable wall that divides the box into a test chamber and a climate chamber. The test chamber can simulate an indoor environment and the climate chamber can simulate different weather conditions. The net dimension of the entire box is 9.00 m long, 3.65 m wide, and 2.27 m high. The current partition makes the test chamber 5.16 m long. The movable wall has a window as wide as the room. Because of the limitation of the experimental facility, the ceiling height is only 2.27m, which is much lower than that of an ordinary room. However, the ceiling height does not have an impact if the results are used for validating the CFD results. With a slightly lower than normal ceiling height, the Reynolds and Grashof numbers are close to those in an ordinary room. The flow and thermal characters in the experiment are the same as those in the ordinary room.

Fig. 2 shows the office configuration used in the experiment. Table 2 lists the internal heated objects. An SF_6 tracer gas source with a rate of 348 ml/min was introduced at the top of occupant 2. This study used two round shape floor diffusers, as shown in Fig. 3, which are widely used for the under-floor air-conditioning system in the U.S. The diffuser diameter is 200 mm and its height is 150 mm.

Measurements were conducted under steady state conditions. Air velocity, air temperature, and the SF_6 concentration were measured in nine different positions, as shown in

Fig. 4. Each position had six anemometers, six sampling tubes for SF_6, and two thermocouples that were mounted from the floor to the ceiling. Omni-directional hot-sphere anemometers were used for measuring air velocity and temperature in the room. The anemometers can measure a velocity between 0.05 to 5 m/s with a repeatability of 0.01 m/s. The measuring errors for air temperature by the anemometers are 0.4 K, including the error by the data acquisition systems. The error for measuring temperature by the thermocouples is also 0.4 K. A multi-gas monitor and analyzer system was used to measure SF_6 concentration. The minimum measurement range of the system is 0.01 ppm and the repeatability is 1% of measured value.

The air change rates used in the measurements were 4 and 8 air changes per hour (ACH). According to the survey conducted by Yuan et al. [17] in Greater Boston area, the cooling load in those buildings can be as high as 140 W/m^2. Therefore, unlike the European system of displacement ventilation that uses 100% fresh air, return air or heat exchanger needs to be used in U.S. buildings with displacement ventilation in order to remove the high cooling load. Since the temperature difference between the supply and return air is rather fixed, it is rather common that the ventilation rate could be between 4 to 8 ACH in those buildings. With an air change rate of 4 ACH, the temperature gradient in the lower part of the room was normally much larger than that in the upper part, because most heat sources, such as the PCs and occupants, were located in the lower part of the room. When the ventilation rate is increased to 8 ACH, the temperature gradient in the lower part of the room can be greatly reduced. This is because the higher air velocity supplied from the floor diffusers turned the stratified conditions into a mixed condition. A very similar trend can be found for the SF_6 concentration. The SF_6 concentration at 8 ACH in the room was almost uniform, compared with a stratified concentration at 4 ACH. Since the present investigation was interested in stratified flow, the case with 4 ACH was used for CFD validation.

Validation of the CFD Program

Correct simulation of airflow in a room by CFD depends on proper specifications of the boundary conditions. The boundary conditions include the heat transfer from the heated objects and building enclosure, the mass flow from the inlet and outlet, as well as information about the tracer gas source. The experiment measured the surface temperatures of the building enclosure (ceiling, floor, window, and walls). For the walls, the important parameter of the boundary conditions of the walls is the h_c, the convective heat transfer coefficient for each wall. The coefficient is defined based on the temperature difference between the wall surface and room air. However, the heat transfer coefficient, h_{CFD}, which is an input for the CFD program, uses the temperature difference between the wall surface and the first cell from the wall. Therefore, it is necessary to adjust h_{CFD} to ensure the heat flux to the wall calculated by CFD is the same as that estimated from the measured wall temperatures.

On the other hand, the experiment did not measure surface temperatures of the heated objects, such as the two occupants, PCs, and overhead lighting, but the total heat generated from those objects. Therefore, the heat fluxes are specified as the boundary conditions for the objects. However, the total heat consists of convection and radiation. The radiative heat is already considered in the wall temperatures measured. The CFD program uses the following method to estimate the convective heat:

$$q_{convection} = h_c (T_{object} - T_{air}) \tag{2}$$

$$q_{radiation} = \varepsilon_{object} \varepsilon_{wall} \sigma (T_{object}^4 - T_{walls}^4) A_{object} \tag{3}$$

$$Q_{total} = q_{convection} + q_{radiation} \tag{4}$$

By assuming h_c and T_{air} , T_{object}, $q_{convection}$ and $q_{radiation}$ can be calculated. Then compare the T_{air} assumed with that calculated, the h_c can be adjusted to reach a converged solution of the convective heat. These procedures may need no more than five iterations.

The round floor diffusers used in the experiment discharges a swirl jet (Fig. 3). This study simulated the round floor diffuser through nine small rectangular cells, as shown in Fig. 4. Each cell has a different airflow direction, except for the center one, which was completely blocked. The airflow directions were obtained through smoke visualization. Since it is difficult to find an exact direction, this investigation used different angles that best fitted the measured room air distribution as the "correct angles". Even though the opening area and the total airflow rate from the diffuser are known, the diffuser is complex. It is difficult to find the accurate velocities from the diffuser openings by calculation. Hence, directly measured velocities at the diffuser by an anemometer were used in the simulation.

The exhaust was simulated with zero pressure and zero gradient for all the flow parameters. Considering that the amount of total mass was very small, and that it would not affect the air distribution much, a zero-momentum was assumed for the tracer-gas source.

Figures 6, 7, and 8 present, respectively, the measured and computed air temperature, air velocity, and SF_6 concentration in the test chamber with 4 ACH. It is clear that the floor-supply system created a stratified flow. Due to the thermal plumes, the temperature in the lower part of the chamber was lower than that in the upper part. The computed temperature profiles are in excellent agreement with the measured data. The velocity is less than 0.2(m/s) except around the diffusers. The agreements between the computed velocity and measured data are acceptable in both cases, although the velocity is low in the most part of the room. However, the agreements between the computed tracer-gas concentration and measured data are not as good as the temperature at some positions. Since the tracer-gas is a point source and the concentration is very sensitive to the position, it is difficult to get good agreement between the computed tracer-gas concentration and measured data at every point. Nevertheless, the accuracy of the computed concentration is acceptable for indoor environmental design.

The above-comparison concludes that the CFD simulation can compute the room air distribution with acceptable accuracy. Hence, it was further used to evaluate the performance of the floor-supply displacement ventilation system in indoor spaces.

Performance Evaluation of the Floor-Supply Displacement Ventilation System

Evaluation Criteria

The performance of a HVAC system should be evaluated based on thermal comfort and indoor air quality. The current study used the following parameters to evaluate the thermal comfort in a room with the floor-supply displacement ventilation:

- Temperature gradient
- Percentage dissatisfied people due to draft (PD)
- Percentage predicted dissatisfied people (PPD)

The CFD program calculates the air temperature in an indoor space. It is easy to determine the temperature gradient. According to ASHRAE Standard 55-1992, the temperature difference between the foot level (0.1m) and the head level (1.7m) should be less than 3 °C. The CFD program also calculates the air velocity and turbulence distributions. It is easy to calculate the PD and PPD. The PD was determined from (Fanger et al. 1989):

$$PD = (34 - T)(u - 0.05)^{0.62}(3.14 + 0.37u\,Tu) \quad [\%] \qquad (5)$$

and $Tu = 100\,(2k)^{0.5}/u$. For PD>100%, PD=100%. The PPD was calculated via (ISO 1993):

$$PPD = 100 - 95\exp(-0.03353PMV^4 - 0.2179PMV^2) \quad [\%] \qquad (6)$$

where the predicted mean vote in the equation, PMV, is determined by:

$$PMV = [0.303\exp(-0.036M) + 0.028]\,L \qquad (7)$$

The following parameters were used to evaluate the indoor air quality:

- Contaminant concentration distributions
- Mean age of air (MAA)
- Ventilation effectiveness

The CFD program can directly calculate the contaminant concentrations. In this study, CO_2 was used as an indicator for indoor air quality. The CO_2 concentration in the outdoor air was assumed to be 400 ppm, and the air-handling unit used 100% fresh air. The indoor CO_2 sources were from the occupants.

The mean age of air, τ, was defined as the average time for all air particles travel from an air supply diffuser to a specific point [17]. The mean age of air was calculated by a transport equation:

$$\frac{\partial}{\partial t}(\rho\tau) + \frac{\partial}{\partial x_j}(\rho u_j \tau) = \frac{\partial}{\partial x_j}(\Gamma_\tau \frac{\partial \tau}{\partial x_j}) + \rho \tag{8}$$

with τ =0 at the supply diffuser and $\frac{\partial \tau}{\partial x_j}$ =0 at the exhaust and walls.

The ventilation effectiveness is defined as:

$$\eta = \frac{C_e - C_s}{C - C_s} \tag{9}$$

In a condition of completely mixed ventilation, the ventilation effectiveness is 1.

Performance Evaluation

This study investigated numerically the impact of the following parameters on the thermal comfort and indoor air quality in a room with the floor-supply displacement ventilation systems:

- Supply airflow rate (air change rate)
- Supply air temperature
- Number of diffusers
- Partition arrangement
- Distance between the occupants
- Furniture arrangement
- Exhaust location and number
- Diffuser location
- Cooling load

The reference case is the one used for the validation, and Table 2 lists its detailed boundary conditions. The cooling load in the reference case was moderate (approximately $40 W/m^2$). The present investigation used the same room for the performance evaluation of the displacement ventilation system. Seven parameters were varied to study their impact on thermal comfort and indoor air quality, as shown in Table 3. Two values were used for each parameter, and only one parameter was varied from the reference case for all the other cases studied. The variation of the parameters in all the cases are marked as bold. Since the cooling load in buildings becomes higher and higher due to the massive use of electronics, this investigation studied two high cooling loads of recent typical high-tech buildings, 80 and 100 W/m^2. In the numerical simulations shown in this section, the room ceiling height was adjusted to 2.7 m. The thermal and geometric boundary conditions in these high-cooling-load cases were the same as those in the reference case, except for those listed in Table 4. The supply air temperatures were adjusted to maintain the room air temperature at 25 °C in the center of the occupied zone.

All the computed results are summarized in Table 5. Case 1 shows that a higher air change rate is better for reducing the temperature gradient and for indoor air quality. However, the higher air change rate may result in draft because of the higher supply air velocity. Although this investigation did not study energy consumption of the air handling system, it is obvious that a higher air change rate would require a large fan and a big air-handling unit. A higher air change rate will therefore consume more energy.

The number of diffusers and the supply air velocity are related to each other, since the total airflow rate is determined from the cooling/heating load and the supply air temperature. If more diffusers are used, the airflow rate per diffuser and supply air velocity will become smaller. Case 2 shows that neither a lower supply air velocity nor a smaller number of diffusers can obtain a better indoor environment than the reference case. With more diffusers (lower supply air velocity), the vertical air temperature gradient is large, due to the poor entrainment of the diffusers. With only one diffuser, the mean age of air is old, because of the poor air distribution in the room. To obtain a better indoor environment, it is necessary to find a good balance between the supply air velocity and the number of diffusers in the room.

Case 3 studied the effect of partitions on the thermal comfort and indoor air quality with the use of the floor-supply system. In Case 3, the floor diffusers were installed inside the partition area, where the heat and contaminant sources were located. With such an arrangement, the room environment was better in terms of the mean age of air than that of one without partitions. If the floor diffusers and contaminant sources are not in the same partitioned space, the air quality might be worse than that of a space without partitions.

With the present configuration of the space, heated objects, and furniture, the locations of the occupant(s) do not affect the indoor environment much, as shown in the results for Case 4.

Furniture normally does not affect the indoor environment of the room unless it disturbs the airflow from the floor diffusers, as shown in Case 5. Case 5-2 had a table placed above the floor diffusers so that the PPD was slightly higher than that of Case 5-1.

Generally, an exhaust location does not have an impact on the room airflow pattern. This is true of all the cases studied here. However, Case 6 shows that exhaust locations had an influence on the indoor air quality of the room. In Case 6-2, two exhausts were almost directly above the contaminant sources (occupants). The contaminant can then be removed fast, and the indoor air quality was better.

Case 7 shows that the distance between the two diffusers did not affect the indoor environment much, as long as the distance is not less than about one meter (so to avoid interference). The same applies to the distance between a diffuser and a solid object close to it.

With a high cooling load (Cases 8 and 9), the large temperature gradient in the room is a major concern. In order to reduce the gradient to be less than 3 °C, a higher air change rate is needed. For example, a 100W/m^2 cooling load needs 12 ACH, and 10 ACH is not acceptable. On the other hand, a large air-change-rate creates high PD distributions, due to a high supply air velocity. It is important to ensure that there is no draft condition in the room. There is a trade-off between the temperature gradient and the draft risk.

Conclusions

A CFD program with the RNG k-ε model of turbulence was used to predict the airflow, air temperature, and tracer gas concentration distributions in a small office. The CFD program was validated by comparing computed results to the experimental data.

By using the validated CFD program, the performance of the floor-supply displacement ventilation system with vortex diffusers was evaluated. Table 6 displays the impact of the parameters studied on the thermal comfort and indoor air quality in the room with the displacement ventilation system. If the cooling load is high, the air change rate should be carefully selected to avoid a large temperature gradient or a high PD distribution.

Acknowledgements

This article was supported by research grant number 1R01 OH04076-01A1 from the Centers for Disease Control and Prevention (CDC). Its contents are solely the responsibility of the authors and do not necessarily represent the official views of the CDC.

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Table 1. Values of ϕ , Γ_ϕ , and S_ϕ in Equation 1.

ϕ	Γ_ϕ	S_ϕ
1	0	0
u_j	$\mu + \mu_t$	$-\partial p/\partial x_i - \rho g_i \beta (T - T_0)$
k	$(\mu + \mu_t)/\sigma_k$	$G - \rho\varepsilon + G_B$
ε	$(\mu + \mu_t)/\sigma_\varepsilon$	$(C_{\varepsilon 1} G - C_{\varepsilon 2}\rho\varepsilon + C_{\varepsilon 3} G_B)\varepsilon/k + R$
T	$\mu/\sigma_1 + \mu_t/\sigma_t$	S_T
C	$(\mu + \mu_t)/\sigma_c$	S_C

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon},\ G = \mu_t \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right),\ G_B = -g_i \beta \frac{\mu_t}{Pr_t}\frac{\partial T}{\partial x_i},\ R = \frac{C_\mu \eta^3 (1 - \eta/\eta_0)}{1 + \beta\eta^3}\frac{\varepsilon^2}{k}$$

$$\eta = S\frac{k}{\varepsilon},\ S = (2 S_{ij} S_{ij})^{1/2},\ S_{ij} = \frac{1}{2}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right),\ C_\mu = 0.0845,\ C_{\varepsilon 1} = 1.42,\ C_{\varepsilon 2} = 1.68,$$

$C_{\varepsilon 3} = 1.0$, $\sigma_k = 0.7194$, $\sigma_\varepsilon = 0.7194$, $\sigma_1 = 0.71$, $\sigma_t = 0.9$, and $\sigma_c = 1.0$.

Table 2. The size and location of the heated objects in the test chamber.

Objects	size (m)			location (m)			Capacity (W)
	Length	Width	height	x	y	z	
Room	5.16	3.65	2.27	0	0	0	
Window		3.35	1.16	5.16	0.15	0.94	*
Diffuser1	0.2	0.2		1.82	1.82	0	
Diffuser2	0.2	0.2		3.84	1.82	0	
Exhaust	0.45	0.45		2.37	1.61	2.27	
Occupant 1	0.4	0.35	1.1	1.98	0.85	0	75
Occupant 2	0.4	0.35	1.1	3.13	2.45	0	75
PC 1	0.4	0.4	0.4	1.98	0.1	0.75	108.5
PC 2	0.4	0.4	0.4	3.13	3.15	0.75	173.4
Desk 1	2.23	0.75	0.75	0.35	0	0.75	
Desk 2	2.23	0.75	0.75	2.93	2.9	0.74	
Box 1	0.95	0.58	1.24	4.21	0	0	
Ceiling lamp 1	0.2	1.2	0.15	1.03	0.16	2.18	68
Ceiling lamp 2	0.2	1.2	0.15	3.61	0.16	2.18	68
Ceiling lamp 3	0.2	1.2	0.15	1.03	2.29	2.18	68
Ceiling lamp 4	0.2	1.2	0.15	3.61	2.29	2.18	68
*Temperature in the climate chamber : 34 (°C)							

Table 3. Thermal and flow boundary conditions for the parameter studies.

	ACH	Supply Air Tempera-ture	Number of Diffusers	Parti-tion	Occupants Location	Furniture	Exhaust Location	Diffuser Location
Case R	**4**	**19**	**2**	**0**	**Far (3m)**	**1 box**	**center**	**regular**
Case-1-1	**2**	**16**	2	0	Far (3m)	1 box	center	regular
Case-1-2	**8**	**22**	2	0	Far (3m)	1 box	center	regular
Case-2-1	4	19	**1**	0	Far (3m)	1 box	center	regular
Case-2-2	4	19	**4**	0	Far (3m)	1 box	center	regular
Case-3-1	4	19	2	**2**	Far (3m)	1 box	center	regular
Case-3-2	4	19	2	**4**	Far (3m)	1 box	center	regular
Case-4-1	4	19	2	0	**Close (1m)**	1 box	center	regular
Case-4-2	4	19	2	0	**Moderate (2m)**	1 box	center	regular
Case-5-1	4	19	2	0	Far (3m)	**6 boxes**	center	regular
Case-5-2	4	19	2	0	Far (3m)	**6boxes +1table**	center	regular
Case-6-1	4	19	2	0	Far (3m)	1 box	**near window**	regular
Case-6-2	4	19	2	0	Far (3m)	1 box	**two exhausts**	regular
Case-7-1	4	19	2	0	Far (3m)	1 box	center	**near**
Case-7-2	4	19	2	0	Far (3m)	1 box	center	**far**

Table 4. Thermal and flow boundary conditions for the high cooling load cases.

	Cooling Load (w/m2)	ACH	Supply Air Tempera-ture	Number of Diffusers	Number of Occupants	Number of PCs	Number of Printers
Case-8-1	**80**	**8**	**17**	4	4	**5**	**1**
Case-8-2	**80**	**10**	**18**	4	4	**5**	**1**
Case-8-3	**80**	**12**	**20**	4	4	**5**	**1**
Case-8-4	**80**	**16**	**22**	4	4	**5**	**1**
Case-9-1	**100**	**8**	**16**	4	4	**6**	**2**
Case-9-2	**100**	**10**	**17**	4	4	**6**	**2**
Case-9-3	**100**	**12**	**19**	4	4	**6**	**2**
Case-9-4	**100**	**16**	**21**	4	4	**6**	**2**

Table 5. The computed results for all the cases studied.

TEM - averaged room air temperature, DT1 – air temperature difference between 1.7 m to the floor and 0.1 m to the floor, DT2 - air temperature difference between 1.1 m to the floor and 0.1 m to the floor, PD – percentage dissatisfied due to draft, PPD – predicted percentage dissatisfied, VE – ventilation effectiveness, C – CO_2 concentration, MAA – mean age of air

	TEM	DT1	DT2	PD	PPD	VE	C	MAA
	(^oC)	(1.7m-0.1m)	(1.1m-0.1m)	(%)	(%)	(-)	(ppm)	(s)
Case R	26.8	3.1	2.2	0.1	9.0	1.1	589	811
Case 1-1	27.7	3.3	2.7	0.1	10.7	0.6	773	1619
Case 1-2	25.7	0.8	0.3	4.8	6.6	2.7	487	342
Case 2-1	26.8	2.9	2.0	0.3	9.4	1.0	607	866
Case 2-2	27.3	3.5	2.8	0.1	10.0	1.1	593	852
Case 3-1	26.8	3.1	2.2	0.2	9.4	1.1	590	774
Case 3-2	26.6	3.0	2.0	0.4	8.4	1.2	589	700
Case 4-1	27.1	3.1	2.3	0.1	9.7	1.1	587	875
Case 4-2	26.9	3.1	2.2	0.1	9.1	1.2	588	842
Case 5-1	26.9	3.2	2.3	0.2	9.2	1.2	581	740
Case 5-2	27.1	3.4	2.7	0.2	9.8	1.2	582	746
Case 6-1	26.8	3.0	2.2	0.2	8.5	1.0	605	876
Case 6-2	26.8	3.1	2.2	0.1	8.8	1.2	579	775
Case 7-1	26.9	3.0	2.2	0.2	9.4	1.1	590	804
Case 7-2	26.8	3.1	2.2	0.2	8.7	1.1	589	801
Case 8-1	25.5	5.1	3.0	0.3	9.0	1.3	563	249
Case 8-2	25.6	3.2	1.2	4.1	6.7	1.3	526	198
Case 8-3	25.2	1.4	0.3	8.3	6.4	1.4	498	170
Case 8-4	25.6	0.7	0.2	9.2	6.6	1.4	480	136
Case 9-1	26.0	6.0	3.8	0.5	9.9	1.4	545	262
Case 9-2	25.8	4.3	2.1	2.9	7.0	1.3	531	203
Case 9-3	25.2	2.1	0.6	8.6	6.5	1.4	500	169
Case 9-4	25.2	1.1	0.2	10.3	6.4	1.5	475	128

Table 6. Summary of the impact of the thermal/flow conditions on the thermal comfort and indoor air quality in the room with the floor-supply displacement ventilation system.

Impact on comfort and IAQ	*Parameters studied*
Large	Air change rate Supply air velocity Supply air temperature Number of diffusers
Medium	Partitions Exhaust location
Small	Diffuser distance Occupant location Furniture arrangement

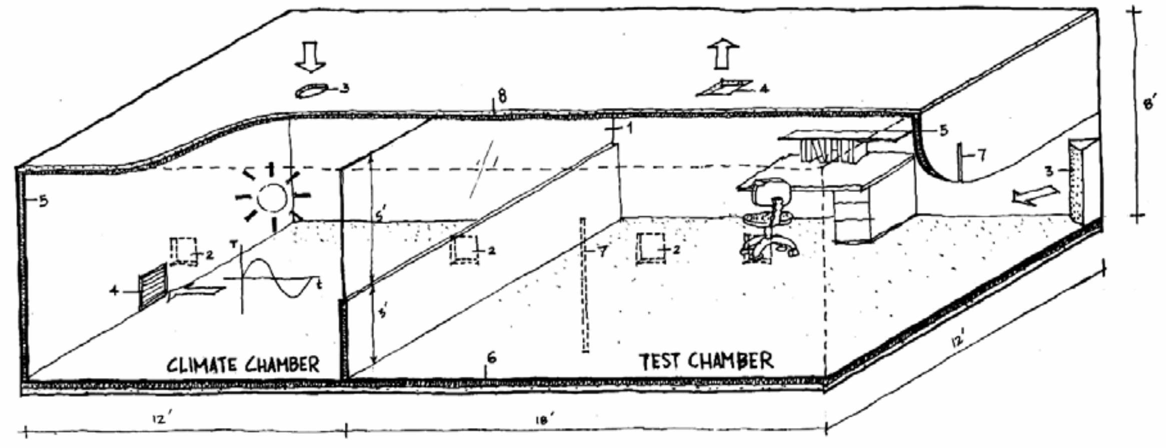

Fig. 1. The schematic of the environmental chamber.

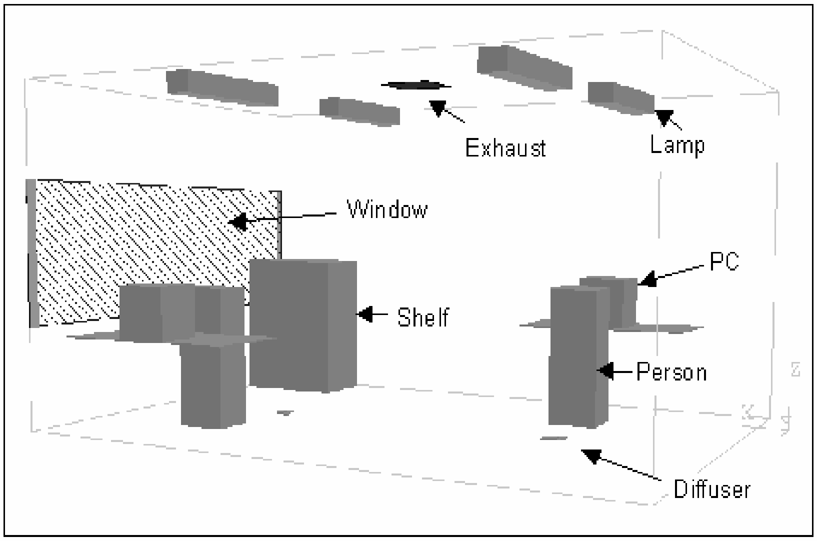

Fig. 2. Space configuration used in the experiment.

Fig. 3. A round swirl floor diffuser.

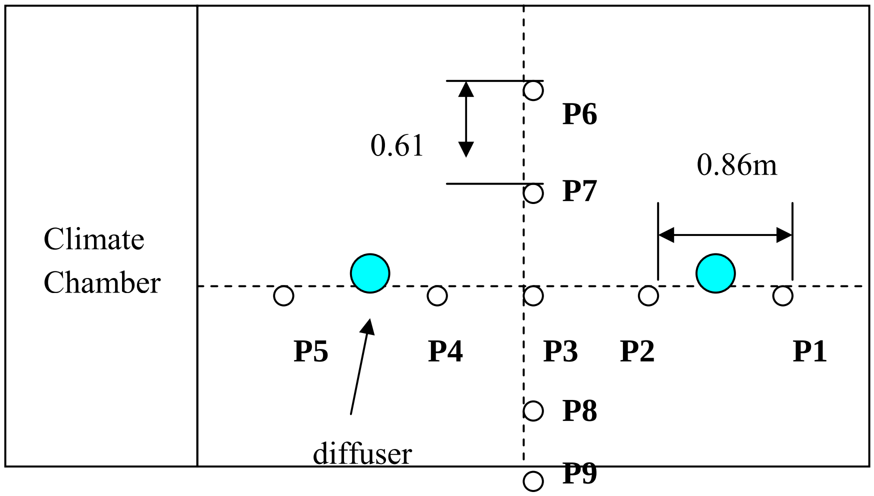

Fig. 4 . The location of the nine measuring positions in the room floor plan.

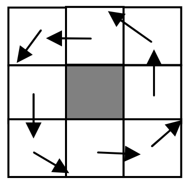

Fig. 5. Floor diffuser simulation.

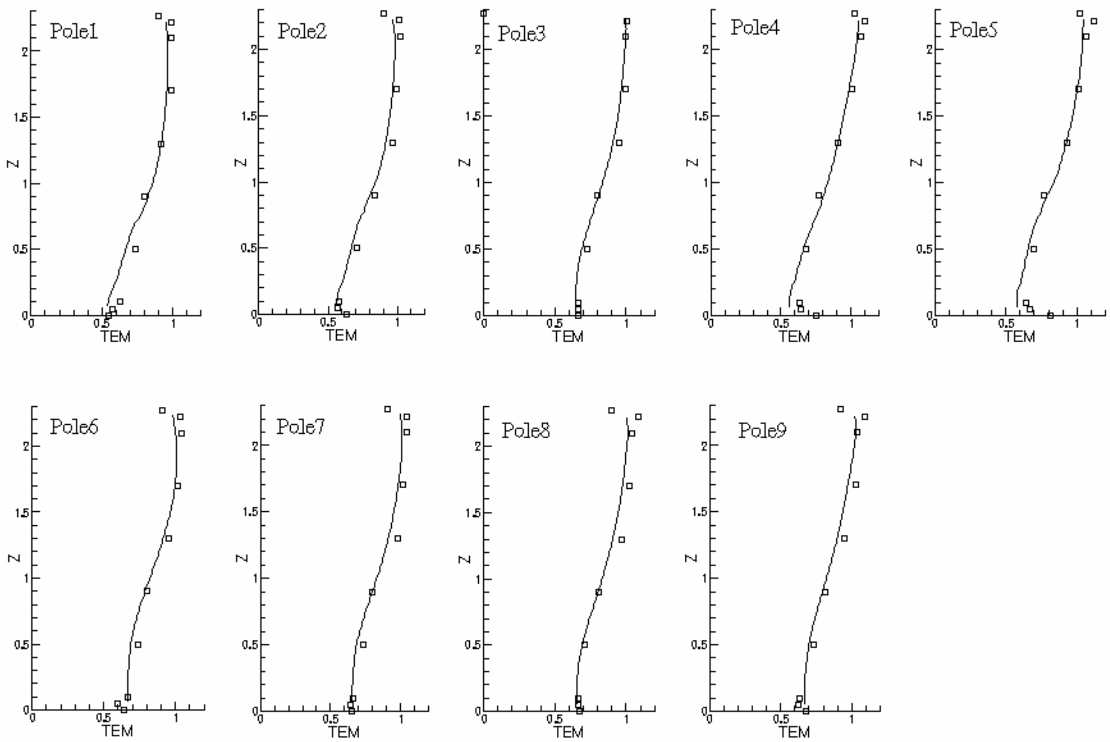

Fig.6. Comparison of the vertical temperature profiles.

TEM = (T - T_{supply})/($T_{exhaust}$ - T_{supply}), Z = height from the floor. Symbols: experiment, Lines: computation

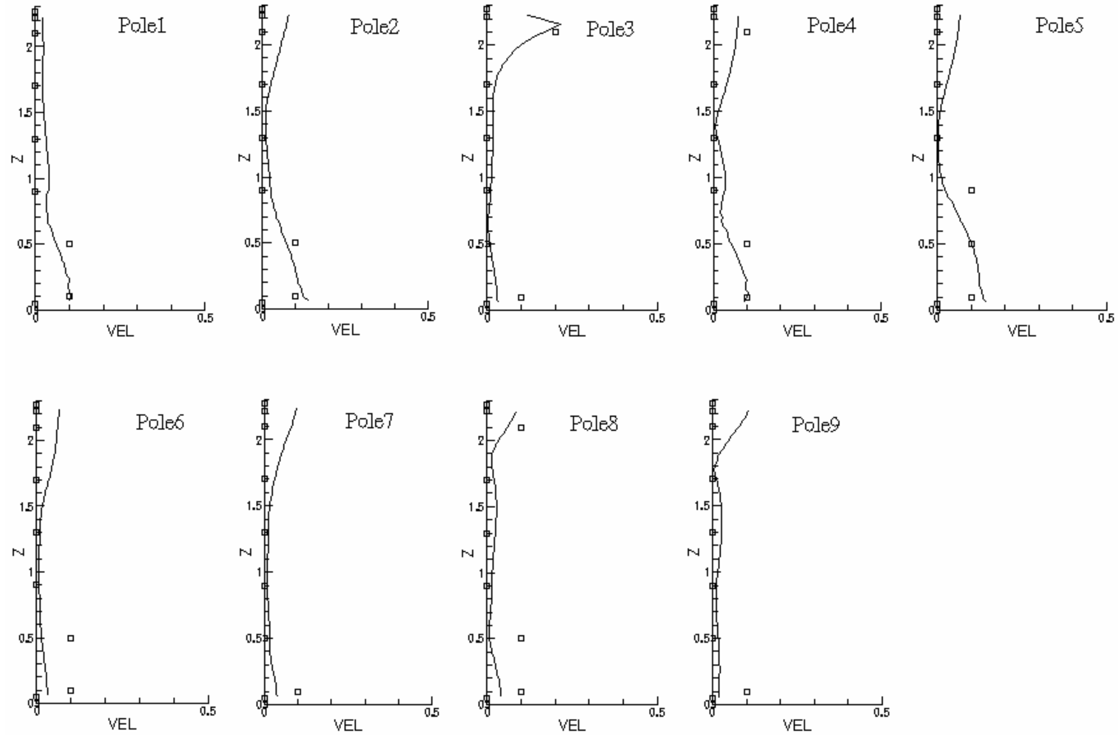

Fig.7. Comparison of the vertical velocity profiles.
VEL = velocity (m/s), Z = height from the floor. Symbols: measurement, Lines: computation.

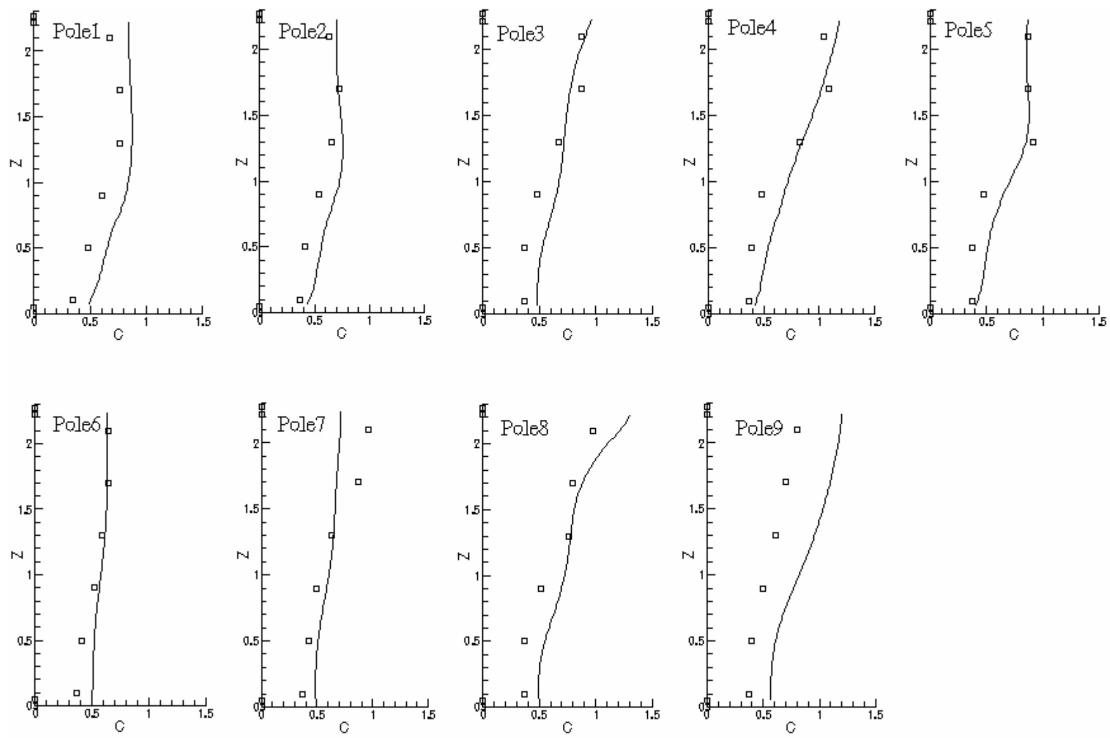

Fig. 8. Comparison of the SF_6 profiles.
$c = (C - C_{inlet})/(C_{exhaust} - C_{supply})$, Z = height from the floor. Symbols: measurement, Lines: computation