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FLOOR-SUPPLY DISPLACEMENT VENTILATION FOR WORKSHOPS

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ABSTRACT

This paper reported the investigation of the performance of floor-supply displacement ventilation with swirl diffusers or perforated panels under a high cooling load (nearly 90W/m^2). The experiment was carried out in a full-scale environmental chamber to obtain reliable data on the floor-supply displacement ventilation for the validation of a computational-fluid-dynamics (CFD) program. Numerical simulations using CFD program were to evaluate the performance of the system for a large workshop. The impacts of several parameters, such as the air change rate, number of diffusers, diffuser location, occupant location, furniture arrangement, partition location, and arrangement of exhausts, on the indoor environment were investigated based on the thermal comfort level and indoor air quality. This study ranked the impacts of these parameters on indoor environment.

KEYWORDS

Floor-supply displacement ventilation, swirl diffusers, perforated panels, computational-fluid-dynamics, thermal comfort, indoor air quality

INTRODUCTION

Most of the heating, ventilating and air-conditioning (HVAC) systems used in U.S. buildings are mixing ventilation. The system design assumes that fresh air delivered from the HVAC systems will completely mix with the indoors pollutants to reduce the concentration level of the pollutants to an acceptable level. However, a complete mixing is difficult to achieve and as a result, the concentration level in some parts of an indoor space may exceed the permitted level [1]. In addition, the complete mixing could enhance cross contamination between occupants due to the re-circulation inside the room.

On the other hand, displacement ventilation provides fresh air directly to the occupied zone. Heated objects, such as the occupants and equipment, will bring the contaminants to the upper zone through the thermal plumes generated by the heat. Return exhausts in displacement ventilation are located at or close to the ceiling through which the warm air with higher contaminant concentrations is removed. The most common configuration for displacement ventilation supplies air horizontally from a diffuser from a low side-wall. Unfortunately, the airflow in such displacement ventilation is not one-dimensional in the occupied zone. Yuan et al. [2], Zhao et al. [3] and Mattsson [4] showed experimentally and numerically cross recirculations in the lower part of the room (Figures 1a and 1b).

These recirculations present the risk of cross contamination between the occupants, thus it is better to use ventilation systems without the risk of cross contamination.

Floor-supply displacement ventilation could be a solution to avoid the recirculation problem. Floor-supply system uses perforated panels or swirl diffusers [5,6] to supply air from the floor and extract air from the ceiling. As reviewed by Akimoto [7], the Japanese have used the perforated floor-supply system in office buildings and acceptable thermal comfort was achieved. The floor-supply displacement ventilation system is superior to the side-wall supply system because it can also be used directly for heating. The system seems superior to other ventilation systems especially for workshops where the risk of cross contamination between the workers can be high. Much research on this ventilation system involves its application in offices [8-13], classrooms [4] or theaters [14, 15] in which the occupants are at a regular activity level (55-70W/m^2 of body surface area). However, more performance data and studies are needed to investigate under workshop configuration in which the activity level of workers is higher (105-140W/m^2 of body surface area) and the heat load density could be higher than the office configuration.

Thus, this investigation is to examine the performance of floor-supply displacement ventilation systems under high cooling load. More specifically, the investigation is to:

1. Compare the performance of the floor-supply ventilation system with swirl diffusers to that with perforated panels;
2. Study indoor environment under a high cooling load situation (nearly 90W/m^2) in a full-scale environmental chamber;
3. Optimize the ventilation system to minimize the risk of cross contamination with suitable air supply and exhaust locations; and
4. Perform a parametric study with different workshop arrangements on thermal comfort and indoor air quality with a computational-fluid-dynamics (CFD) program.

RESEARCH METHODS

CFD was used in this study to investigate thermal comfort and indoor air quality for different workshop settings. Since the CFD used approximations, it was necessary to validate the CFD program before it was used as a tool of study [16]. The validation was done by comparing the CFD results with experimental data obtained in an environmental chamber with a floor-supply displacement ventilation system. The experimental measurements were carried out to obtain the flow characteristics in the chamber with swirl diffusers and perforated panels. With the CFD and experimental results, it is possible to evaluate thermal comfort and indoor air quality through appropriate indices for a room with floor-supply displacement ventilation.

Measurement procedure

Our environmental chamber consisted of a box covered by well-insulated walls (with thermal resistance - 5.45 Km^2/W) as shown in Figure 2. It had a movable wall, which divides the chamber into two parts. One was the test chamber, in which the measurements were conducted and the other was the climate chamber, which simulated different outdoor conditions. The movable wall had a double-glazing window and an

insulated wall underneath the window. The dimensions of the test chamber were 4.91 m long, 2.44 m high, and 4.31 m wide. The perforated floor panels were 0.6 m square, constructed of a 0.025 m thick high density particle board core, structurally bonded to electro-galvanized steel top and bottom sheets. The cavity under the raised floor is pressurized by the supply air and used as a supply air plenum feeding the floor diffusers or perforated panels. Each chamber has an individual HVAC system so that different conditions can be made for the test chamber and the climate chamber. For the cases studied, the climate chamber was used to simulate summer conditions at 32^{o}C. The test chamber was arranged as a workshop and we assumed the occupants were doing machine work in the light electrical industry. According to ASHRAE 55-2004 [17], the heat rate per body surface area is 105 W/ m^2. Since the surface area of the manikin is 1.92 m^2, the total heat released from each manikin should be 201.6W. However, for the simplicity, 200W was used because light bulbs with standard rating were used to generate heat in the manikin. The chamber contained four occupants with a heat rate of 200 W (sensible heat) per person, four equipment boxes each with 100 W and six ceiling lamps of 64 W per lamp. The total cooling load is very high (nearly 90W/m^2) and the break-down of the load was shown in Table 1. Workshop-1 is the case with the swirl diffusers and Workshop-2 is the case with the perforated panels.
Figure 3 shows the floor plan used in the experiments. The internal heat sources and the tracer gas source used in the experimental cases are listed in Table 2. The measurements for air velocity, air temperature, and Sulfur Hexafluoride (SF_6) concentrations were conducted on nine removable poles installed in the chambers. SF_6 is a tracer-gas used to simulate a contaminant. Each pole had seven hot-sphere anemometer probes for air temperature and velocity measurements and six sampling tubes for SF_6 concentration measurements at different heights. Fifty-eight thermocouples were used to measure the temperatures of the walls, ceiling, window, and floor, as well as the supply and exhaust air. The accuracy of the temperature reading on the pole was ±0.3 K. The anemometers were omni-directional and had the measurement range from 0.05 m/s to 5 m/s. The accuracy of the velocity reading was 0.02m/s or ±1% of readings. The error for measuring temperature by the thermocouples was ±0.4 K. A multi-gas monitor and analyzer system was used to measure SF_6 concentration. The minimum measurement range of the system was 0.01 ppm and the repeatability was 1% of the measured value.

CFD models

This investigation used a commercial CFD program [18] that solves the governing equations for the conversation of mass, momentum, heat, contaminant concentrations, and turbulence quantities. The CFD program used the Re-Normalization Group (RNG) k-ε model [19] because it is slightly more accurate than the standard k-ε model for indoor airflow simulations [20]. Since the indoor heat source is a main force for the buoyancy flows, the Boussinesq approximation, which relates density change with temperature difference, is also employed. Since the RNG k-ε model is valid for high Reynolds number turbulent flow, generalized (non-equilibrium) wall functions are needed for the near wall region where Reynolds number is low. The simulations use the finite volume differencing scheme and SIMPLE algorithm [21]. Structured Cartesian meshes are used to discretize the cases studied since the problems studied were in the rectangular configurations. The total number of grids was 69x78x43 for Workshop-1 and it was

56x67x43 for Workshop-2. Finer grids were used for more critical areas such as diffusers and the region near to the wall and heated objects. After generating a numerical grid on which the discrete algebraic equations were solved and specifying a set of problem-dependent boundary conditions, the calculation was iterated until prescribed convergence criteria were met. The convergence criteria were set such that the respective sum of the absolute residuals of mass continuity, three components of momentum, contaminant concentration, kinetic energy of turbulence, and dissipation rate of turbulence energy at each cell was less than 10^{-3} of the total inflow of the corresponding parameters and that of temperature was less than 10^{-2} of the total energy inflow. The computation for each case took about 48 hours on a personal computer with a processor running at a speed of 2.6 GHz.

Boundary conditions

Correct simulation of airflow in a room by CFD depends on proper specification of the boundary conditions by the user. The surface temperatures on the ceiling, floor, side walls, window, occupants and equipment were measured in the experiment and were used as the boundary conditions for the walls and heated objects in CFD simulations. When there was temperature difference between surfaces, radiant heat warms up the colder surfaces; these surfaces then became secondary convective heat sources to the air. Therefore, the measurement for surface temperature at steady-state already takes into account the affect of heat from radiation to the air. For surface temperature on the walls, temperature stratification was found as the air temperature was also stratified and, therefore, in the CFD model, wall surfaces were sub-divided into three levels to give a more realistic simulation. Round swirl diffusers and perforated panels were simulated by CFD using the momentum method [22] because it is almost impossible to calculate directly the complicated geometry of the diffusers with many tiny holes. The momentum method required information of turbulence intensity, flow directions and velocity magnitudes around a diffuser that was obtained through measurements. Figure 4 shows how flow direction was obtained by smoke visualization for a swirl floor diffuser and a perforated panel, respectively. The air velocity from the perforated panels was upwards and uniform while the air velocity from the swirl diffuser was a 30 degree bend from vertical axis and with strong turbulence. In the CFD simulations, the perforated panel was represented as a 0.6 m x 0.6 m square opening with a fixed mass inflow and velocity; and the swirl diffuser was simulated as rectangular shape divided into nine small cells (Figure 5). Each cell has a different airflow direction.

The exhaust was simulated with zero pressure and zero gradient conditions for all the flow parameters. Considering that the amount of total mass of SF_6 was very small, and that it would not affect the air distribution much, a zero-momentum was assumed for the tracer-gas source.

Indices for thermal comfort and indoor air quality

Since the main objective for a ventilation system is to create an appropriate thermal environment and indoor air quality for building occupants, this investigation evaluated the performance of the floor-supply displacement ventilation system by using different indices. The following parameters were used for thermal comfort:

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

The CFD program calculates the three-dimensional airflow pattern and air velocity, air temperature, and turbulence distributions. With the distributions of air velocity, temperature and turbulence, it is easy to calculate the point average of PD and PPD. Fanger et al. [23] developed a model to calculate the PD as:

$PD = (34 – T)(u – 0.05)^{0.62}(3.14 +0.37u\ Tu)$ [%] (1)
and $Tu=100(2k)^{0.5} /u$ [%] for PD>100%, PD=100%
Where T = air temperature [°C]
u = air velocity [m/s]
Tu = turbulence intensity [-].
k = turbulent kinetic energy. [J/kg]

The PPD was calculated according to ISO 1993 [24]:
$PPD = 100 –95\exp(-0.03353PMV^4 – 0.2179PMV^2)$ [%] (2)
The predicted mean vote, PMV, in the equation is determined by:
$PMV = [0.303\exp(-0.036M) + 0.028] L$ (3)
Where L = the thermal load on the body. [W]
M = activity level (metabolic rate) [met]

A well designed ventilation system is not only providing thermal comfort to the occupants inside the space, but it should also allow occupants to stay in a clean and healthy space. The following parameters were used to evaluate the indoor air quality:

- Contaminant concentration distributions
- Mean age of air
- Ventilation effectiveness

In this study, CO_2 is used as an indicator for contaminant concentration. The CO_2 concentration at the supply air is assumed to be 400 ppm in the CFD study. The indoor CO_2 sources are from the occupants. According to the ASHRAE Standard 62-2001 [25], the CO_2 concentration should be less than 1000 ppm in the room. Since the CO_2 concentration in the atmosphere fluctuated, SF_6 was used as a tracer gas to simulate indoor contaminants when we validated the CFD program. SF_6 has a zero background concentration in the atmosphere.

The mean age of air, τ, is defined as the average time for all air particles travel from the supply diffuser to that point. It can be derived from the measured transient history of the tracer-gas concentration [26]. The mean age of air is governed by a transport equation:

$$\frac{\partial}{\partial t}(\rho\tau)+\frac{\partial}{\partial x_j}(\rho u_j\tau)=\frac{\partial}{\partial x_j}\left(\Gamma_\tau\frac{\partial\tau}{\partial x_j}\right)+\rho \qquad (5.1.5)$$

ρ = density of air [kg/m^3]

(4)

τ = mean age of air [sec]
Γ_τ = effective diffusion coefficient [-]

with the following as the boundary conditions:

τ =0 at the supply diffuser

$\frac{\partial \tau}{\partial x_j}$ =0 at the exhaust and walls

The ventilation effectiveness [27] is defined as:

$$\eta = \frac{c_e - c_s}{c - c_s} \tag{5}$$

where η = ventilation effectiveness [-]
c_e = contaminant concentration in the exhaust air [ppm]
c_s = contaminant concentration in the supply air [ppm]
c = contaminant concentration in the room air [ppm]

In the condition of complete mixing ventilation, the ventilation effectiveness is 1.

RESULTS

CFD result validation

The following figures show the vertical profiles of air velocity (Figures 6a and 6b), air temperature (Figures 7a and 7b), and SF6 concentrations (Figures 8a and 8b) at two typical locations in the chamber with the two different air supply diffusers. The corresponding CFD results are also plotted in the figures as comparison. The results presented were located at Poles 4, 5 and 9 as shown in Figure 3. Although measurements were taken in many more more locations, these three poles were selected because these locations can represent the flow characteristics in a room with an underfloor ventilation system. Pole 4 was in between two occupants; while Pole 5 was right below the exhaust of the room and Pole 9 was close to the supply diffusers or panels. The flow pattern at the three locations was complicated and representative.

Figures 6a and 6b show the air velocity profiles for Workshop-1 and Whokshop-2, respectively. The velocities in the room were generally low for both cases (less than 0.2 m/s except around the diffusers and exhaust). The agreement between the computed velocity and measured data are acceptable, although discrepancies were found in some locations. Since the CFD model is not perfect and the omni-directional anemometers are not free from errors, it is difficult to identify the exact reason for the discrepancies.

Shown in Figures 7a and 7b, the temperature gradient in the lower part of the room was much larger than that in the upper part, because most heat sources, such as equipment boxes and occupants, were located in the lower part of the chamber. Comparing the vertical temperature distribution of Workshop-1 with that for Workshop-2, the one with swirl diffusers induced more turbulence flow near the diffuser region and the air temperature, therefore, was less stratified than that with perforated panels. Smaller air temperature stratification is preferred for thermal comfort. For Workshop-2, this thermal comfort requirement was not satisfied at Pole 9 according to ASHRAE Standard 55-2004

[17] that restricts the vertical temperature difference between the head and the feet of a person to no more than 3 K. The air temperature close to that diffuser at the feet level was too cold. Remediation of this situation was not carried out because the supply air temperature and velocity were not individually adjustable at the perforated panels, nevertheless, the computed temperatures agree with the measured data within 1.5 °C difference.

Figures 8a and 8b show tracer gas distributions simulated by SF_6. The agreement between the computed tracer-gas concentration and measured data is not as good as that for the temperature and velocity. Measured data and the simulation results did not agree well for poles 5 and 9 which were located at the strong turbulence regions and at a higher air velocity. 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. Our experience shows that the steady state of the tracer-gas is very difficult to achieve. We experienced the disturbance at the source and the SF_6 dosing system might create a jump unexpectedly. Therefore, the result of the SF_6 measurement might have a high uncertainty. Nevertheless, the general trend of the computed concentration seems to still reflect the trend of the measured one.

From the flow patterns from the smoke visualization and the CFD simulation, both swirl diffusers and perforated panels effectively reduced the cross contamination by creating a more unidirectional flow. The slow recirculation at the occupant zone was eliminated as illustrated in Figures 9a and 9b.

Workshops with the Floor-Supply Displacement Ventilation System

The validated CFD program was used to evaluate the performance of floor-supply displacement ventilation in a workshop. The impacts of different parameters on thermal comfort and air quality were investigated. A large workshop that is 13.1 m long, 10.2 m wide, and 4.5 m high was simulated to represent a light electrical assembly workstation or a sewing workshop. The workers were seated and worked in front of an assembly machine or a sewing machine as shown in Figure 10. The dimensions of this workshop and the loading density of occupants, equipment/machine and overhead lighting were developed with the reference of the survey of Yuan et al. [28]. There were 28 occupants each generating a heat rate of 200 W due to metabolism, 28 machines each with 100W and 18 ceiling lights each with 62 W. Fig. 10 shows the configuration of the reference workshop. The following parameters were studied with this workshop:

- Exhaust location and number
- Supply airflow rate and supply air temperature
- Partition arrangement
- Number of diffusers
- Diffuser location
- Distance between the workers
- Furniture arrangement

Seven different cases were made from the reference workshop (Case R), and each case has two variations. Table 3 shows the different parameters of each case were marked as bold letters. By studying these parameters with the degree of impacts on the thermal comfort and indoor air quality, the significance of each of these parameters could be compared. The best arrangement of a workshop with floor-supply displacement ventilation system could also be identified. Table 4 summarizes the results of the parametric studies on the horizontal plane at 1.1 m above the floor. The height at 1.1m was selected because it was the height of a seated person and representing the breathing height of the workers in the workshops.

Exhaust location and number

When exhaust location and number were changed in Case 1, the vertical temperature distributions were almost the same. Unlike the study by Mattsson [4], the impact of the exhaust location on air quality is not significant in the large workshops. This is because the contaminant sources were quite evenly distributed in the room. Therefore, changing the exhaust location will not affect the ventilation effectiveness significantly, especially in the region below the occupant height (1.1m).

Supply airflow rate and supply air temperature

Since the indoor heat load was not changed in Case 2, the total heat removed by this system was the same. The supply air temperature is usually adjusted according to the cooling load and the air change rate. Comparing to Case R, Case 2a had a lower air change rate (4 ACH) with a reduced temperature of supply air (14.5°C) while the Case 2b had a higher air change rate (6 ACH) with an increased temperature of supply air (18.3°C). From the results, the higher the air change rate, the smaller the air temperature stratification. Even at the reduced flow rate in Case 2a, the temperature difference between head and foot level was still at an acceptable level for thermal comfort. From the Case 2a, a smaller air change rate reduced the PD significantly at the height 1.1m (from 7.3% in Case R to 4.7% in Case 2a). It was because the air velocity of each diffuser was changed when the total air change rate varied. However, the impact on PPD was not as significant as on PD. This is because PPD is based on heat balance by considering the temperature of the room and the heat lost of an occupant.

On the other hand, the higher air change rate, the younger the age of air and the better the air quality. The age of air was the youngest for Case 2b (467 s at 1.1m) and that for Case 2a was the oldest (516 s at 1.1m) while that for Case R was in between (486 s at 1.1m). Clearly, there is trade-off between the energy used and indoor air quality. A higher air change rate implies more energy consumed by the fan.

Partition arrangement

In a workshop, partition may be used to separate the room into different zones for different types of workers. The use of partition might be a problem because they can be obstacles for air movement. Therefore, the parameter changed in Case 3 is the number of partitions. The influence of the partitions was not significant in these workshop cases studied. This is because the partition was arranged in parallel to the airflow direction in the workshop so the partitions did not block the flow. Therefore, the conclusion is only applicable to the cases studied and should not be generalized.

Number of diffusers

The number of diffuser was considered to be an interesting parameter, because more diffusers allow a more uniform distribution in the horizontal plane of the workshop. Since the total air change rate was fixed at 5 ACH and the total heat load was also unchanged, Cases 4 changed the individual flow characteristics of the diffusers when the number of diffusers in the room changed. The air velocity at each diffusers was 2.25 m/s in Case R, 3.01 m/s in Case 4a, and 1.5m/s in Case 4b. For Case 4a where the number of diffusers was fewer than that of the reference case, the air velocity at the diffusers was higher, which induced a higher turbulence and mixing effect. The temperature stratification was reduced. Comparatively, Case 4b where the air velocity at the diffusers was reduced had less mixing effect and more stratification. For Case 4b, the ΔT_2 increased by 83% compared with that of Case 4a and by 57% with Case R. The change of ΔT_1 was more significant (60% and 100% higher than that in Case R and Case 4a). Obviously, the diffusers in Case 4b did not have the minimum air velocity in order to keep the throw height and to avoid large temperature gradient.

For both Case 4a and Case 4b, the age of air was older than that of Case R. For Case 4a, the age of air at 1.1m above the floor was 656 s while that for case R was 486 s. The age of air in Case 4a was still older since the diffusers were further away from the occupants and it took a longer time to reach the occupant. Comparatively, too many diffusers (Case 4b) would not improve air quality either (with age of air 705 s) because the diffusers were not operating at the suggested flow rate. The supply air formed a stagnant layer at the lower level and there was weak momentum force to bring the air up to the breathing zone.

Diffuser location

Case 5 was designed to study the impact of diffuser location on indoor environment. The total number of diffusers was kept the same. Only the location and distance between the diffusers were changed. In the reference case, the diffusers were arranged along the three walkways between tables. Eight diffusers were located in the center walkways while four diffusers on each remaining walkway. The distance between each diffuser was 1.6 m. For Case 5a and Case 5b, two more diffusers were put on each side of the walkways and only four diffusers were remained in the middle. For Case 5a, the diffusers distance kept the same as case R (1.6 m). For Case 5b, a wider distance was allowed (3.2m). The average temperature and contaminant concentration did not vary much. The height of the stratification layer was very close among the three cases. Therefore, the distance of the diffusers would not affect the global temperature and contaminant distribution, however would affect local values. Special attention should be paid to the distance between occupants and diffusers, since the PD and PPD were high and exceed the thermal comfort limit within the radius distance of 0.5 m from a diffuser. In other words, an occupant should be seated at the distance of 0.5 m from the diffusers to avoid local draft.

Distance between the workers

Different occupant seating arrangements from the reference case were also used. Instead of arranging the equipment and the workers in four rows seated along the two long tables, the occupants and the equipments were arranged sitting around the walls of the room and the center table in the middle as shown in Figure 11. The new arrangement of the occupants did not change the temperature and contaminant stratifications. The vertical distributions of the parameters were similar for the three cases.

Furniture arrangement

In order to study the influence of furniture on the thermal comfort and air quality in the workshop with floor-supply displacement ventilation, eight file cabinets were added in Case 7a and then two more benches were added in the Case 7b. The temperature distribution of Case R is similar to that of Case 7a because those file cabinets on the two sides were not heat sources and they did not obstruct the airflow in the room. However, in Case 7b, the benches obstructed the flow from the diffusers so the PD was increased from 7.3% in Case R to 16.2% at 1.1 m above the floor.

The average age of air and the CO_2 concentration for all the cases did not vary much (Within ±10% of the R-case which is about 600 ppm) as the sources of the CO_2 were at the head level of the occupants and the influence of the furniture arrangement below this height would not be significant. Therefore, as soon as benches and tables do not obstruct the airflow from diffusers, such furniture has minimal impact on the thermal comfort and air quality in the room.

CONCLUSIONS

The results from this study show that a workshop with floor-supply displacement ventilation can improve indoor air quality because the contaminant concentration in the breathing zone is lower than that of mixing system. The ventilation effectiveness is always greater than 1 for all the cases studied. Since a more unidirectional flow was created, the slow recirculation at the occupant zone was eliminated for the floor-supply ventilation with both swirl diffusers and perforated panels, and the risk of cross contamination can be effectively reduced. However, the indoor spaces with floor-supply displacement ventilation could have a higher risk of discomfort, because of high temperature stratification between the ankle and head levels when compared to traditional mixing ventilation. The system with the swirl diffusers can provide a better comfort level than that with the perforated panels due to the mixing by the diffusers. The draft risk can be high in an area within 0.5 m around the swirl diffuser.

This research used a validated CFD program as a tool to further study the impacts of several parameters on indoor environment in the workshop with floor-supply displacement ventilation, such as exhaust location and number, supply airflow rate and supply air temperature, partition arrangement, number of diffusers, diffuser location, distance between the workers, and furniture arrangement. The parameters with significant impact on thermal comfort and air quality in the workshop are air change rate, supply air velocity, supply air temperature, and number of diffusers; with moderate impact are partition location, and exhaust location; and with little impact are diffuser location, occupant location, and furniture arrangement.

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NOMENCLATURE

c = contaminant concentration in the room air [ppm]
c_e = contaminant concentration in the exhaust air [ppm]
c_s = contaminant concentration in the supply air [ppm]
ΔT_1 = air temperature difference between 1.7m and 0.1m [^oC]
ΔT_2 = air temperature difference between 1.1m and 0.1m [^oC]
k = turbulent kinetic energy. [J/kg]
L = the thermal load on the body. [W]
M = activity level (metabolic rate) [met]
PD = percentage dissatisfied due to draft [%]
PPD = predicted percentage dissatisfied [%]
T = air temperature [^oC]
TEM = averaged room air temperature [^oC]
Tu = turbulence intensity [-]
u = air velocity [m/s]
VE = ventilation effectiveness [-]
ρ = density of air [kg/m^3]
τ = mean age of air [sec]
Γ_τ = effective diffusion coefficient [-]

η = ventilation effectiveness [-]

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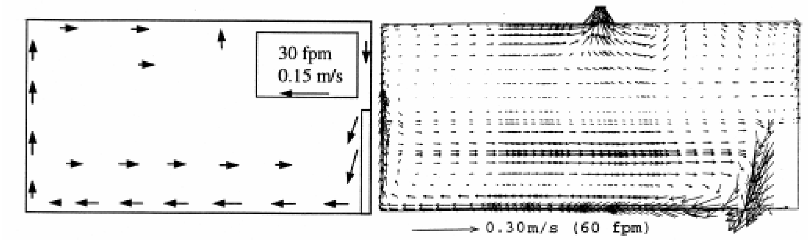


Fig. 1a Airflow pattern observed by using smoke visualization and computed by the CFD program (side view of the room). The length of the arrow is proportional to the velocity level. (source: Yuan et al. 1999 [2])

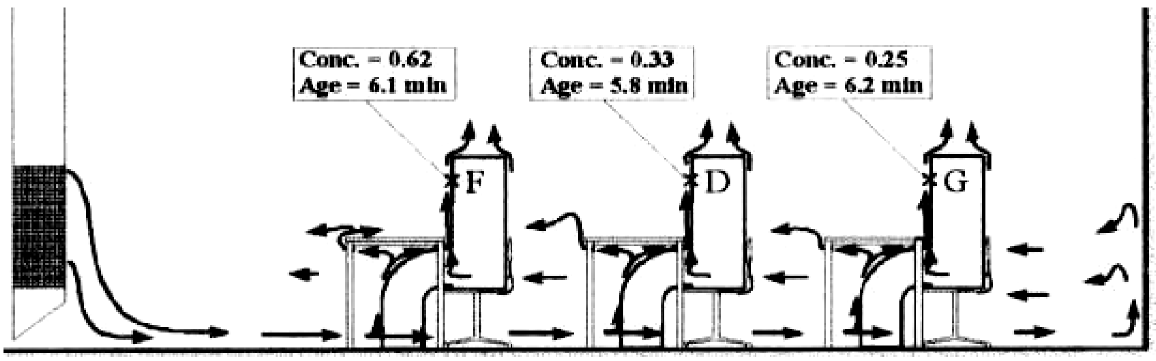


Fig. 1b Measured airflow pattern in a room with side-wall-supply displacement ventilation. (source: Mattsson. 1998 [4])

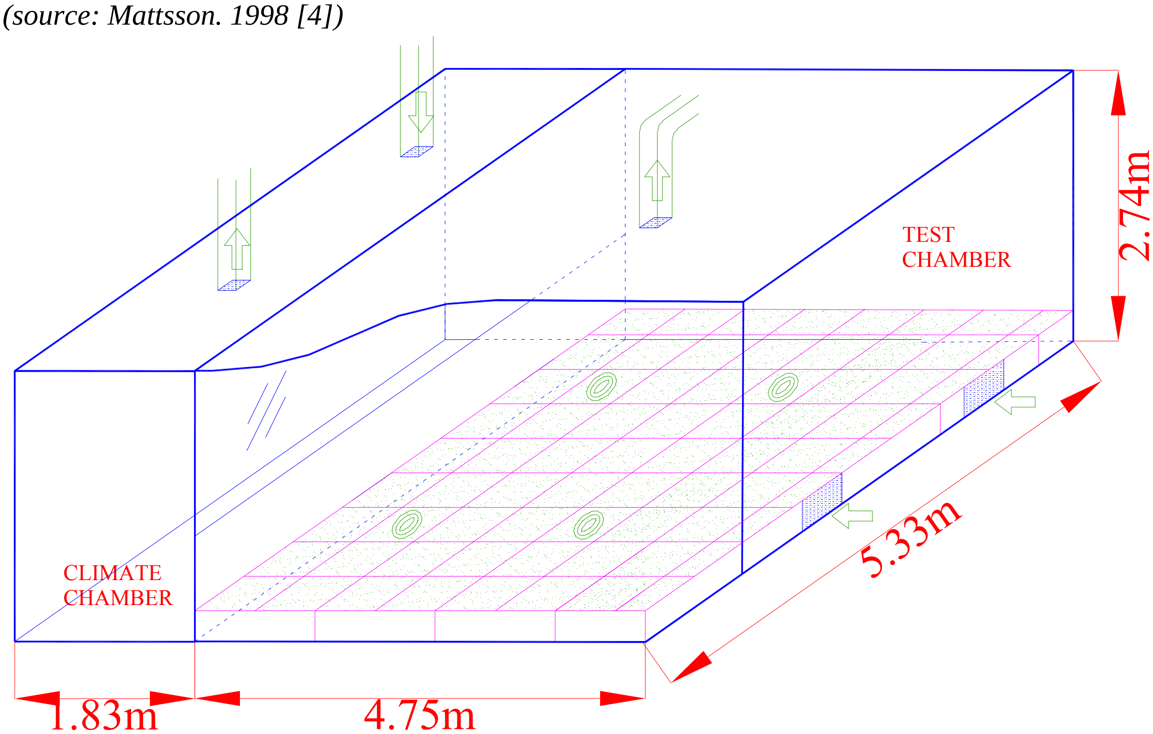


Fig. 2. Environmental chamber (marked with external dimensions)

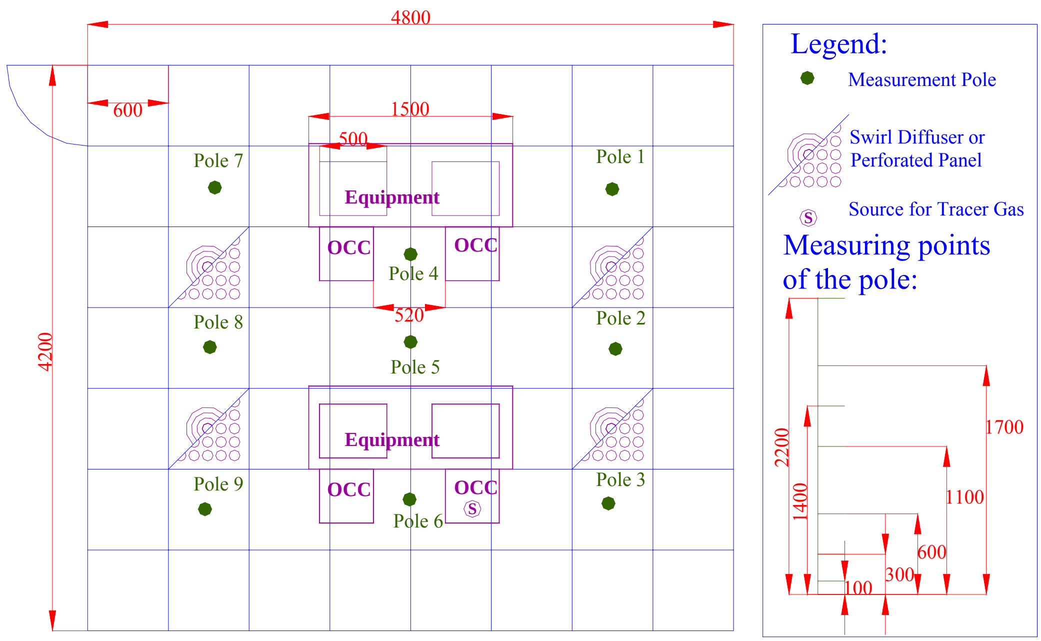


Fig. 3. Floor plane of workshop-1 and 2

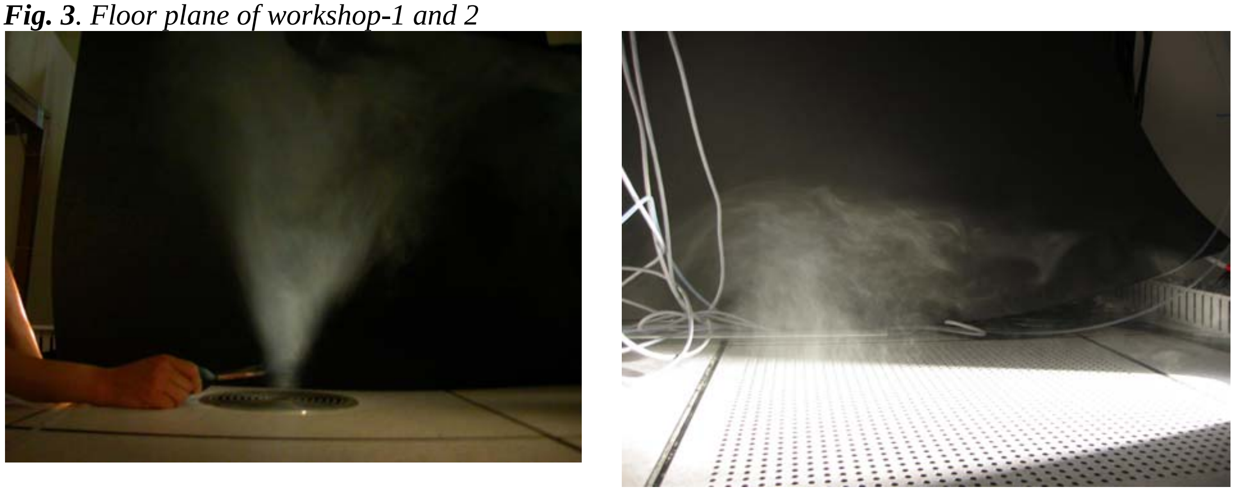


Fig. 4. *Smoke visualization of the airflow pattern from the floor diffuser (left) and from the perforated panel (right).*

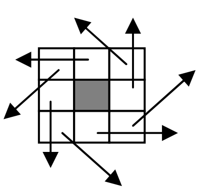


Fig. 5*. Floor diffuser simulation*

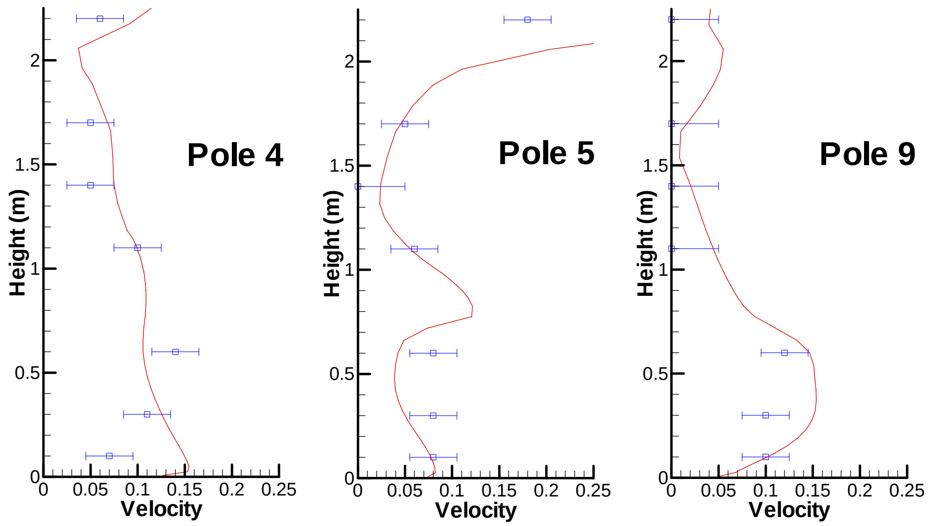


Fig. 6a. Comparison of the vertical velocity profiles of Workshop-1with swirl diffusers. units: (m/s), Symbols: measurement, Lines: computation

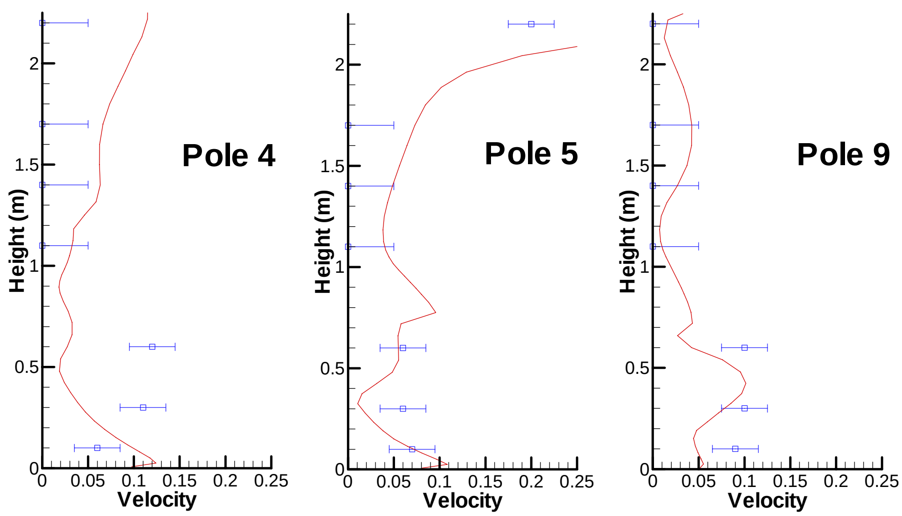


Fig. 6b. Comparison of the vertical velocity profiles of Workshop-2 with perforated panels. units: (m/s), Symbols: measurement, Lines: computation

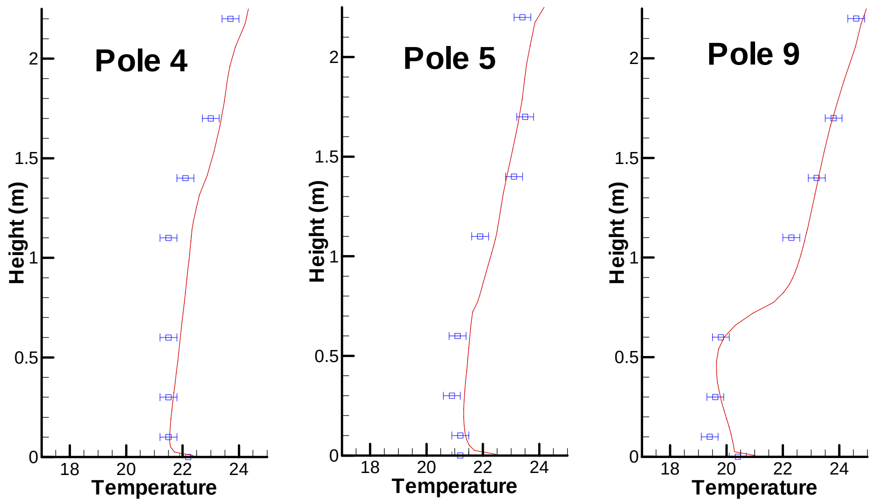


Fig. 7a. Comparison of the vertical temperature profiles of Workshop-1with swirl diffusers. units: (^{O}C), Symbols: measurement, Lines: computation

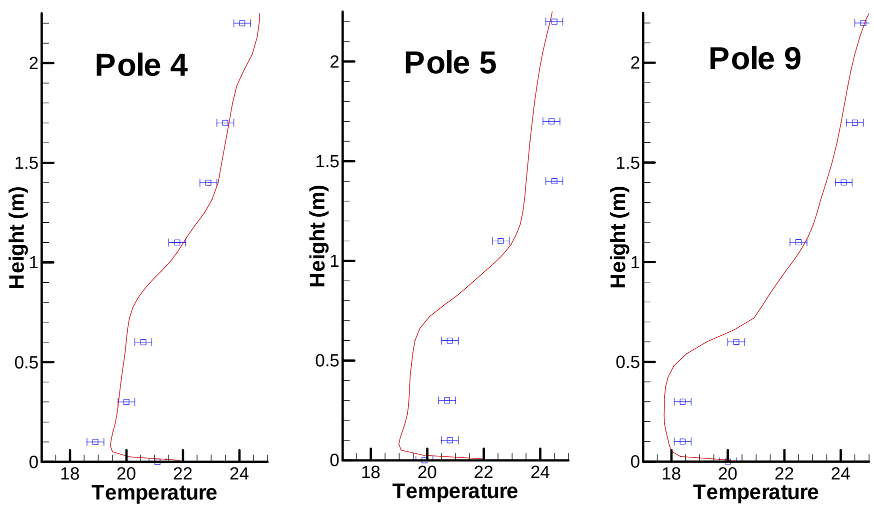


Fig. 7b. Comparison of the vertical temperature profiles of Workshop-2 with perforated panels. units: (^{O}C), Symbols: measurement, Lines: computation

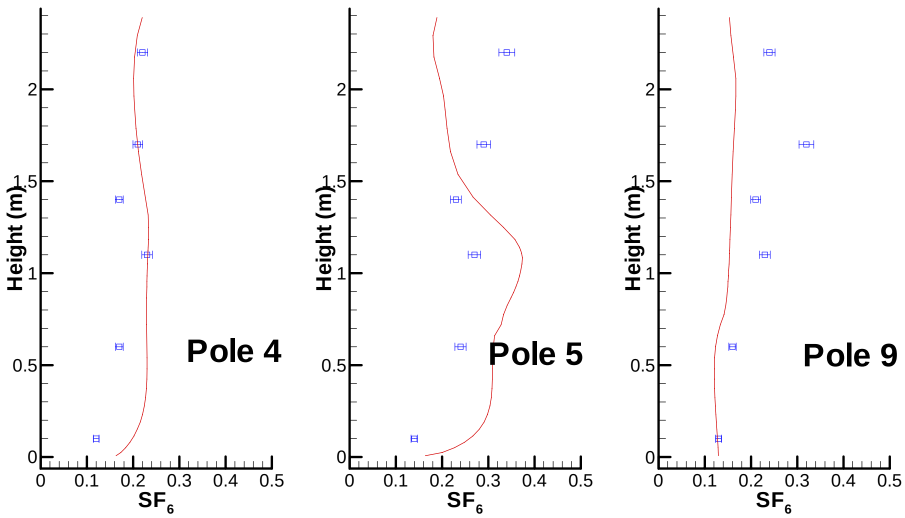


Fig. 8a. Comparison of the vertical SF_6 concentration profiles of Workshop-1 with swirl diffusers. units: (ppm), Symbols: measurement, Lines: computation

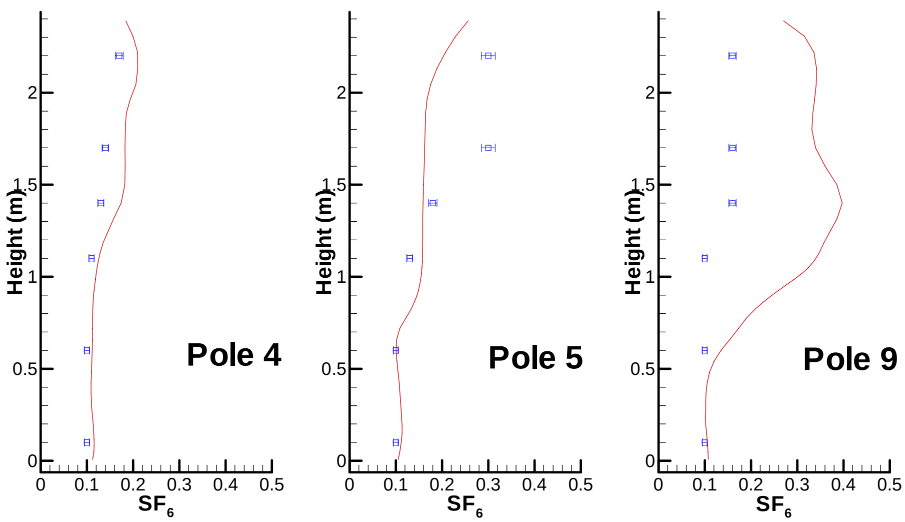


Fig. 8b. Comparison of the vertical SF_6 concentration profiles of Workshop-2 with perforated panels.
units: (ppm), Symbols: measurement, Lines: computation

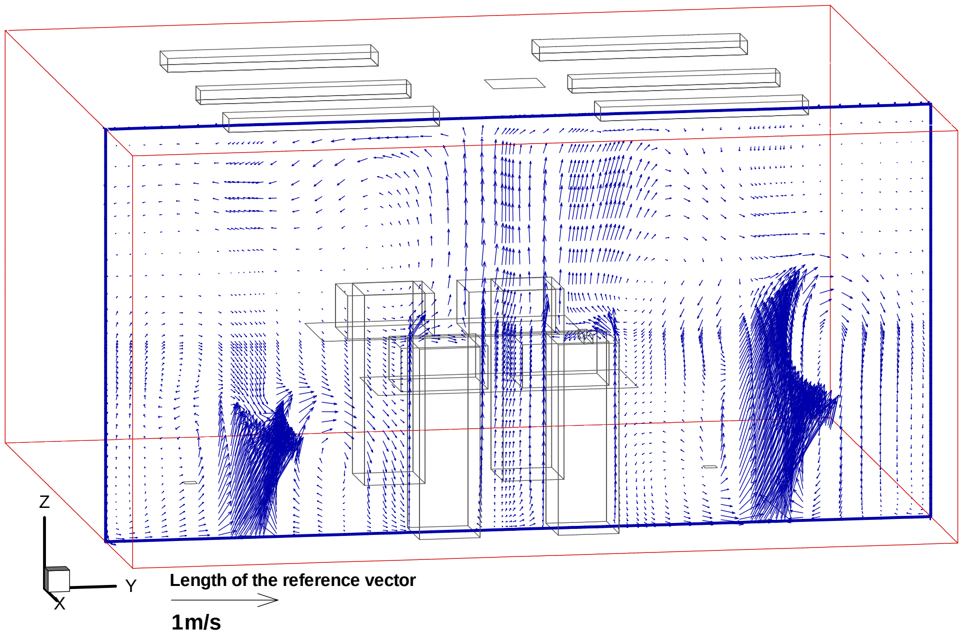


Fig. 9a. Flow pattern of Workshop-1 with swirl diffusers that has no recirculation in the lower part of the room.

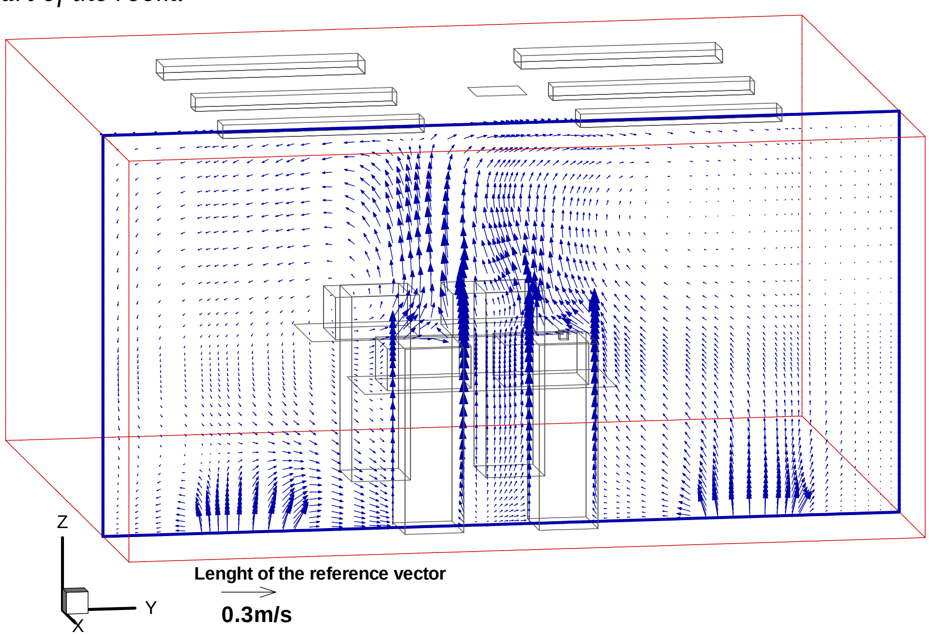


Fig. 9b. Flow pattern of Workshop-2 with perforated panels that has no recirculation in the lower part of the room.

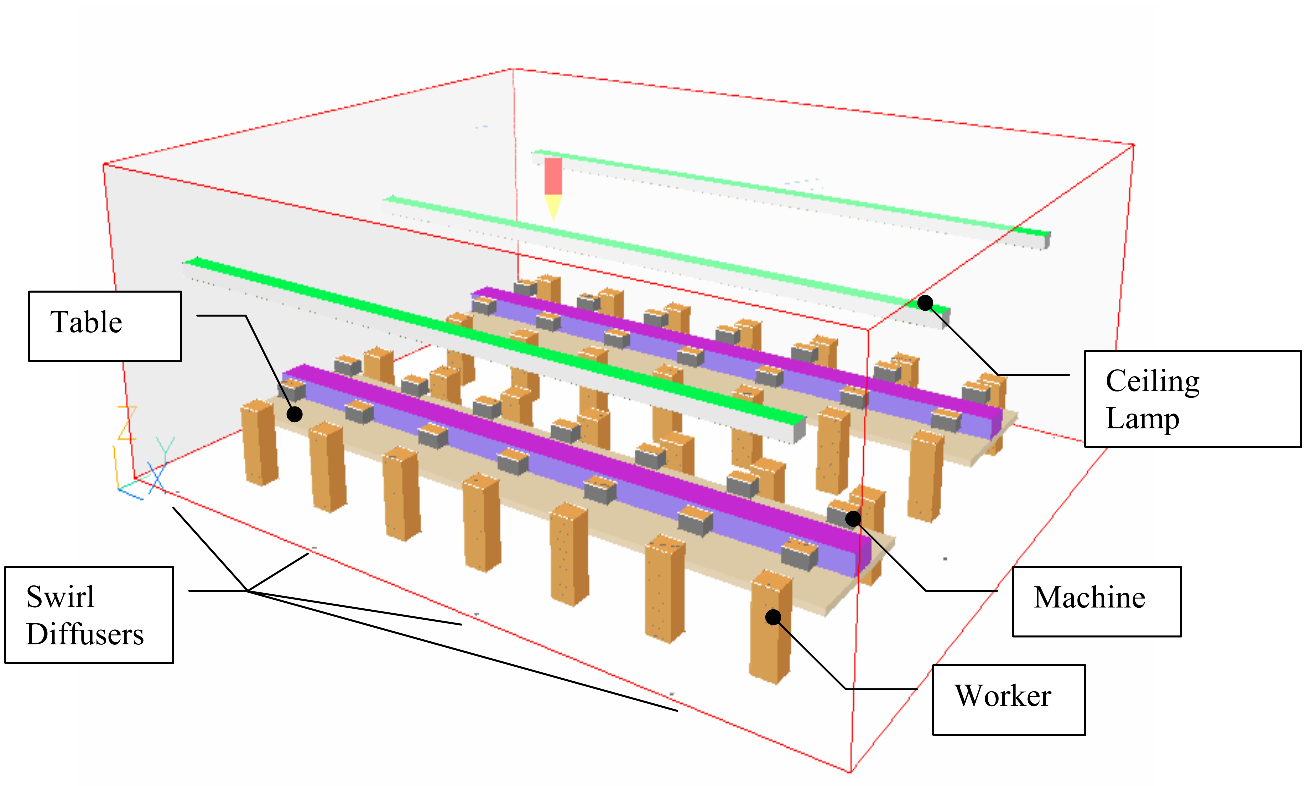
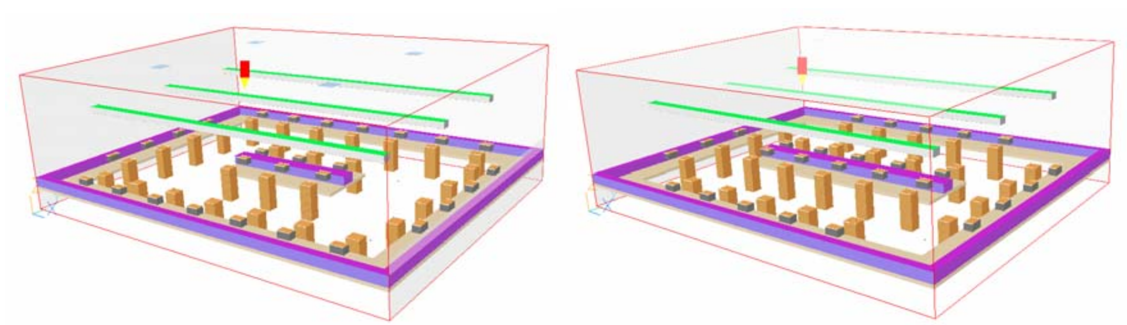


Fig. 10. Configuration of the reference case (Case R)



Case 6a *Case 6b*

Fig. 11. Different arrangements for occupants in Case 6

Table 1. Break-down of the cooling load in the room for the experiment

Cooling Loads in Test Chamber	Heat (W)	
$Q_{envelope}$	0	(or negligible) because of no significant temperature difference between chamber and surrounding environment)
$Q_{parti\text{-}wall}$	19.42	=11.7*5.85/3.53 (#1)
$Q_{parti\text{-}glass}$	171.7	=5.7*7.35/0.25 (#2)
Q_{equip}	1584	=(100W+200W)*4+64W*6
Q_{total} (W)	1,775.12	
Q_{total} (W/m^2)	88.05	
Temperature difference(K) (from measurement)	Area (m^2)	Resistance (K*m^2/W)
Partition-wall (#1)		
11.7	5.85	3.53
Partition-glass (#2)		
5.7	7.35	0.25

Table 2. Internal heat source and the tracer gas source

Heat Sources				
Item	size (m)			Heat (W)
	length	width	height	
Occupant (for workshop cases)	0.4	0.4	1.1	200
Equipment	0.5	0.4	0.25	100
Ceiling lamp	1.2	0.15	0.08	64
Temperature in the climate chamber : 32 (C)				
Tracer gas (SF_6) Source				
Item	Height (m)		SF_6 (ml/min)	
Occupant	1.1		500	

Table 3. *Summary of the parameters studied for the workshop*

Case	ACH	Supply air temp. (oC)	Number of diffusers	Partition	Exhaust Location[1]	Diffuser location (# of diffusers per row)	Occupant arrangement	Furniture
Case R	**5**	**16.8**	**16**	**0**	**2 sides (on E and W)**	**regular (4,8,4)**	**4 rows**	**2 big tables**
Case 1a	5	16.8	16	0	**2 sides (on N and S)**	regular (4,8,4)	4 rows	2 big tables
Case 1b	5	16.8	16	0	**4 sides**	regular (4,8,4)	4 rows	2 big tables
Case 2a	**4**	**14.5**	16	0	2 sides (on E and W)	regular (4,8,4)	4 rows	2 big tables
Case 2b	**6**	**18.3**	16	0	2 sides (on E and W)	regular (4,8,4)	4 rows	2 big tables
Case 3a	5	16.8	16	**2**	2 sides (on E and W)	regular (4,8,4)	4 rows	2 big tables
Case 3b	5	16.8	16	**4**	2 sides (on E and W)	regular (4,8,4)	4 rows	2 big tables
Case 4a	5	16.8	**12**	0	2 sides (on E and W)	regular (4,8,4)	4 rows	2 big tables
Case 4b	5	16.8	**24**	0	2 sides (on E and W)	regular (4,8,4)	4 rows	2 big tables
Case 5a	5	16.8	16	0	2 sides (on E and W)	**regular (6,4,6)**	4 rows	2 big tables
Case 5b	5	16.8	16	0	2 sides (on E and W)	**far (6,4,6)**	4 rows	2 big tables
Case 6a	5	16.8	16	0	4 sides	regular (4,8,4)	**Arrangement A[2]**	2 big tables
Case 6b	5	16.8	16	0	4 sides	regular (4,8,4)	**Arrangement B[2]**	2 big tables
Case 7a	5	16.8	16	0	2 sides (on E and W)	regular (4,8,4)	4 rows	**Type A[3]**
Case 7b	5	16.8	16	0	2 sides (on E and W)	regular (4,8,4)	4 rows	**Type B[3]**

[1]Note A: “E and W” means “East and west sides” & “N and S” means “North and South sides”
[2]Arrangement A: 22 on the surrounding and 6 in middle, and Arrangement B : 18 on the surrounding and 8 in middle
[3]Type A: add 8 boxes, and Type B: add 8 boxes + 2 side benches

Table 4. The computed results for all the cases studied on horizontal plane at 1.1 m above the floor.

Case	**TEM**	**ΔT_1**	**ΔT_2**	**PD**	**PPD**	**AGE**	**CO_2**	**VE**
	(^oC)	(K)	(K)	(%)	(%)	(sec)	(ppm)	(-)
Case R	23.7	1.5	0.7	7.3	6.1	486	597	1.116
Case 1a	23.7	1.2	0.6	7.1	6.2	504	615	1.163
Case 1b	23.7	1.3	0.6	7.8	6.1	511	609	1.055
Case 2a	23.5	2.1	1	4.7	6	516	642	1.033
Case 2b	24.1	0.9	0.5	8.8	6.4	467	591	1.111
Case 3a	23.3	1.1	0.4	7.4	6.2	486	606	1.067
Case 3b	23.2	1	0.3	7.6	6.3	471	605	1.073
Case 4a	24.1	1.2	0.6	8.9	6.4	656	630	1.152
Case 4b	24.1	2.4	1.1	4.3	5.8	705	597	1.116
Case 5a	23.9	1.3	0.6	6.5	6	490	600	1.1
Case 5b	23.8	1	0.4	5.6	6	512	572	1.279
Case 6a	22.7	1	0.3	9.8	6.7	426	581	1.215
Case 6b	22.6	1	0.2	9.3	6.7	438	586	1.183
Case 7a	23.5	1.4	0.6	7.2	6.2	511	620	1.022
Case 7b	22.5	1	0	16.2	8.5	356	611	1.043

Keys: TEM- averaged air temperature on plane at 1.1m, ΔT_1 – air temperature difference between the head and ankle level of a standing person, ΔT_2 – air temperature difference between the head and ankle level of a sitting person, PD – percentage dissatisfied due to draft, PPD – predicted percentage dissatisfied, VE – ventilation effectiveness, AGE – mean age of air