Validation of Displacement Ventilation System in a Residential Room

Lei Xu¹, Weijun Gao² and Toshio Ojima³

¹ Graduate Student, Department of Architecture, Waseda University, Japan
² Associate Professor, Department of Environment Space Design, Kitakyushu University, Japan
³ Professor, Department of Architecture, School of Science and Engineering, Waseda University, Japan

Abstract

The conventional air conditioning system for residential houses aims to a uniform room temperature, even those places near the ceiling is cooled during the summer. But the displacement ventilation system supplies cold air directly to the occupant space. In order to verify the effectiveness of this system for residential houses, it has been introduced in the experiment house in Kitakyushu. According to the field experiment and CFD simulation, the thermal environment and ventilation effectiveness has been analyzed. The stratification in temperature and CO₂ concentration has been verified, and comfortable environment has been achieved in the occupant space. And the energy consumption for cooling by this system is about 20% lower than the conventional system.

Keywords: displacement ventilation system; field experiment; simulation; thermal comfort; ventilation effectiveness

1. Introduction

At present, with the information technology (IT) development, the popularized small office & home office (SOHO) applications lead to the increase of cooling load in houses, which attracts more attention on energy saving for air conditioning and indoor air quality again.

The conventional air conditioning system for residential houses aims to a uniform room temperature, even those places near the ceiling is cooled during the summer. But the displacement ventilation system supplies cold air, which is only several degrees lower than the setting temperature in rooms, directly to the occupant space from the air inlet near the floor. From the point of view in energy conservation, this system shows great promise. Besides, the pollutants from human will be discharged by the buoyancy effects, and will not be circulated in the occupant space.

In order to verify the effectiveness of the displacement ventilation system for residential house, it has been adopted in the experiment house in Kitakyushu. According to the field experiment in the summer¹ and CFD simulation, the temperature distribution, ventilation effectiveness, and energy consumption for cooling by the displacement ventilation system have been studied.

2. The Displacement Ventilation System in the Experiment House

2.1 Details of the Proposed System

The air conditioning system in the experiment house is shown in Fig. 1. The occupant space is under 1.8m above the floor in each room. The cold air at about 18°C, which comes from the thermal storage air-handling unit (AHU) in the machine room, is mixed with the induced room air, and then the mixed air of about 22°C is supplied from the low part of the fan unit. The center of the air outlet is 0.3m above the floor, and the supply air goes directly to the occupant space at a quite low speed. Some of the air at 1.8m in the room is induced into the fan unit, and the other goes back to the AHU as return air, while the exhaust air is discharged from the ceiling. In addition, the louvers inside the outlet can also swing.

The detail constructions of this house are shown in Table 1. A double skin space is to the south of the SOHO room (1F) and the living room (2F), and its outer glazing consists of 8mm clear float with interior light blinds, while the interior glazing consists of 6mm clear float.

Contact Author: Lei Xu, Graduate Student, Dept. of Architecture, Waseda University, Room 08-05, No. 55S Building, 3-4-1, Okubo, Shinjuku-ku, Tokyo, 169-8555 Japan
Tel: +81-3-5286-3132 Fax: +81-3-5291-6855
e-mail: lei@ojima.arch.waseda.ac.jp
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Fig.1. Air Conditioning System in the Experiment House
3.2. Feature of the Proposed System
As using the displacement system, the smaller temperature difference between supply air and room air will leads to a larger airflow rate. In the proposed system, some of the room air is induced into the fan unit, then it is mixed with the cold air from the AHU before it is flown into the room. And the air inlet of return air is set at the height of 1.8m over the floor, which is cooler than the air near the ceiling.

3. Field Experiment in the Summer
The field experiment of displacement ventilation was carried out in the SOHO of the experiment house since July 29 to August 3 in 2001. The layout of the Soho is shown in Fig. 2. The blinds in the double skin were completely shut down during experiment.

3.1. The Setting Conditions for Experiment
In order to keep the supply air temperature at 22°C, the air temperature from AHU is set at 18 ± 1°C. The geometrical, thermal and flow boundary conditions of the experiment are detailed in Table 2.

3.2. The Test Points of Experiment
3.2.1. Air Temperature Distribution
The test points of experiment are detailed in Table 2. The air temperature from AHU is set at 18 ± 1°C. The air temperatures under and above the ceiling are also measured.

3.2.2. Air Velocity Distribution
Air velocity at the height of 1.1m is measured, and there are 8 points (cord1~cord6, cord10, cord12) in the horizontal surface.

3.2.3. CO₂ Concentration
The CO₂ concentration in the occupant space has also been measured at the height of 0.1m, 0.6m, 1.1m and 1.7m², all together there are 48 points to be tested. The air temperatures under and above the ceiling are also measured.

4. CFD Simulation
According to the data from the field experiment at 12:00 on August 2, the initial conditions for simulation have been set as shown in Table 3. The simulation is carried out by using the CFD soft Airpak2.0.

Table 1. Detail Construction and Thermal Resistance

| Details       | Main Construction                                      | R (m²·K/W) |
|---------------|--------------------------------------------------------|------------|
| Roof          | 8mm Glass+140mm Insulation+20mm Recyle Pet Board       | 3.26       |
| Exterior Wall | 8mm Glass+90mm Insulation+20mm Recyle Pet Board        | 2.19       |
| Interior Wall | 70mm Insulation + 10mm Recyle Pet Board                | 1.48       |
| Floor (2F)    | 17mm Plastic +50mm Insulation+50mm Air + 4mm Rubber+12mm Recyle Pet Board | 1.8       |
| Floor (1F)    | 17mm Plastic +100mm Insulation +50mm Air + 4mm Rubber+12mm Recyle Pet Board | 2.7       |
| Window        |北:3mm glass+6mm air+3mm glass                          | 0.22       |
|              | South (double skin outside): 8mm glass                  | 0.17       |
|              | South (double skin inside): 6mm glass                   | 0.15       |

Table 2. The Conditions of the Field Experiment

| Item             | Specification | Center Height | Flow rate |
|------------------|---------------|---------------|-----------|
| Room volume      | 40m³          |               |           |
| Supply air inlet | 0.6×0.3m²     | 0.3 m         | 0.65m/s   |
| Induced air inlet| 0.6×0.16m²    | 1.77 m        | 0.5m/s    |
| Return outlet    | 0.55×0.1m²    | 1.8 m         | 1 m/s     |
| Exhaust outlet   | 0.18×0.1m²    | 2.4 m         |           |
| Equipment and human heat release | | | |
| Light            | 150W          | Human         | 110 W     |
| Computer         | 400W          |               |           |

Note: There were two people in the SOHO during experiment.

T-type thermal couples, together with a data collector are used to record the air temperature. In order to test the air temperature distribution in the occupant space, there are 12 cords in the occupant space (Fig. 2), and 4 points in the vertical direction with the height of 0.1m, 0.6m, 1.1m and 1.7m², all together there are 48 points to be tested. The air temperatures under and above the ceiling are also measured.

Table 3. The Conditions of Simulation

| Item             | Specification | Center Height | Flow rate |
|------------------|---------------|---------------|-----------|
| Temperature      |               |               |           |
| East wall        | 29.6°C        | Ceiling       | 30.6°C    |
| West wall        | 28.3°C        | Floor         | 28°C      |
| North wall       | 28.5°C        | Supply air at inlet | 22.2°C |
| South glass      | 30.8°C        |               |           |
| Relative humidity| 60%           |                |           |
| Supply air inlet |               |                |           |
| Opening conditioning |           |                |           |
| Air inlet        | Vx=0.65, Vy=0.65sin(t/40), Vz=0.15 (m/s) | | |
| Return air outlet| Vx=0.5m/s     | Exhaust air outlet | Ambient pressure |

Note: t is time of operation (s), as the louver swings at a cycle of 40s.

The standard k-epsilon model has been adapted with the turbulent intensity of 35% at the air inlet, while the turbulent length scale is supposed to be 10% of the length of the shortest side of the air inlet.

The distributions of air temperature, velocity, PMV and age of air have been analyzed.

5. Results of Experiment and Simulation
5.1. Climate Conditions
The outside air temperature and the sky radiation on the horizontal surface are given in Fig. 3, which shows that the temperature peak reached to 35.8°C during 12:00 to 14:00.

5.2. Temperatures of Interior Surfaces of the Soho

Temperatures of different surfaces in the room are detailed in the Fig. 4, and the peak temperatures have been reached before the air conditioning system is started. The temperature on the surface of the east wall reaches to 34°C, while the others reach to 30~31°C at 9:30. After the air conditioning system is started, those temperatures go down slowly, and they reach to the balance temperature at about 28°C. On the other hand, the temperature of the south glass reaches its peak 30.8°C during 12:00 to 14:00 because of the strong solar radiation from the south, and it goes down after 15:00.

5.3. Thermal Environment

5.3.1. Temperature Distribution

(1) Stratification in Temperature

Fig. 5 shows temperatures at points of cord 5 (X=2.7m, Y=1.8m). Before air conditioning system is on, the temperature of each point was about 30~31°C and the vertical temperature difference is very small. After air conditioning is started, air temperature falls down. And the temperature distribution becomes stable an hour later. The average temperature from 0.1m to 0.6m is about 25.5°C, 1.1m about 26.5°C and 1.7m about 28°C during 11:00 to 17:00. The air near the ceiling is at about 30°C. Testing points on other cords experienced the same variation as those on cord 5.

The room air temperature goes up slowly after 17:00, because the ice stored in thermal storage AHU has been used up step by step.

Fig. 6 shows the temperatures at different heights at 12:00 by simulation, and the temperature profile at Y=1.8m agrees with that of the experiment, especially under the height of 1.2m (Fig. 7). For example, at the average temperature at 0.1m is 25.2°C, 0.6m 25.9°C, and 1.1m 26.5°C, but above 1.6m, the result of simulation is about 0.5°C higher than the experiment. The reason could be considered that the fixed temperatures of interior surfaces are adopted as the boundary conditions, while the radiation between interior surfaces is neglected in the CFD simulation.

(2) Temperature Gradient

The temperature gradient in the room is shown as
At 9:00 or 21:00, as air conditioning is off, there is almost no temperature gradient, and the temperature difference is within 1°C from floor to ceiling. After the air conditioning is started, the temperature difference becomes greater, and about an hour later, i.e. 11:00, the temperature gradient becomes stable. Temperature difference from 0.1m to 1.1m is within 2°C, while the temperature difference is 3°C near the air inlet (within 1m diameter from the air inlet).

The comparison of the temperature gradient between experiment and simulation is shown in Fig. 9, and the simulation has a quite good accuracy.

(3) Air Humidity

The humidity and temperature of air are measured at the height of 1.3m on the fan unit. From 11:30 to 12:30, the average temperature is 24.5°C, and the average humidity 65%, thus to calculate the absolute humidity to be 0.0126kg/kg.

Take absolute humidity as constant, according to temperature distribution, the humidity distribution can be shown as Fig.10. Humidity in the occupant space is in the range from 50% ~ 60%.

(4) Velocity Distribution

The flow rate of supply air is about 425m³/h, with the average velocity of 0.65m/s at the air inlet, and the turbulence intensity in the room can be defined as equation 1:\[ T_u = \frac{1}{\bar{\nu}} \sqrt{\frac{1}{n-1} \sum (\nu_i - \bar{\nu})^2} \times 100\% \] (1)

Where,

\( \bar{\nu} \): the average velocity;
\( \nu_i \): velocity of point \( i \);
\( n \): point number

Fig. 11 shows air velocity of points on cord1, cord 2, cord 4 and cord 5 at the height of 1.1m. Air velocity is below 0.25m/s and turbulence intensity \( T_u \) is 20%.

Air velocity near ceiling is below 0.2m/s, and it means airflow there is not directly affected by air inlet.
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5.3.2. PMV Distribution

PMV of measured points can be calculated based on measured data of temperature, humidity, air velocity, human activity and clothing.

Here suppose the metabolic rate is 1.2Met (70W/m²)

and the clothing value is 0.5clo, Fig. 12 illustrates the PMV distribution at the section of Y=1.8m of SOHO room at 12:00.

5.4. Air Exchange Efficiency

5.4.1. CO₂ Concentration

After the temperature distribution becomes stable, concentration of CO₂ is measured vertically at section of Y=0.45m (Fig. 13). The concentration of CO₂ is relatively low near the inlet, and rather high near the ceiling.

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5.4.2. Mean Age of Air

Fig. 14 shows the distribution of local mean age of air $t_a$ at the section of Y=1.8m by simulation analysis. The average age of air $t_a$ in the whole room is about 272s. The nominal time constant $t_{nom}$ (=room volume/supply airflow rate) is 339s, thus the average air exchange efficiency is 1.25.

5.5. Energy Consumption of Air Conditioning

\[ Q_{sci} = \sum \alpha_i (T_{ci} - \overline{T}_{air}) + C_p Q_p + C_i Q_i + C_{eq} Q_{eq} \]  

(2)

According to the heat balance for the room air, the sensible air conditioning load can be calculated by equation 2.

Where,

- $\alpha_i$: the convection coefficient of interior surface $i$ and air, 5W/m²;  
- $T_{ci}$: temperature of interior surface $i$, (°C );  
- $\overline{T}_{air}$: the average temperature measured near the surface $i$, (°C );  
- $Q_p$: sensible heat release from human (W);  
- $Q_l$: heat release from light (W);  
- $Q_{eq}$: the convection ratio in $Q_p$, 60%;  
- $C_p$: the convection ratio in $Q_p$, 50%;  
- $C_{eq}$: the convection ratio in $Q_{eq}$, 80%.

According to measured temperatures and the heat release in Table 2, the average sensible heat load for air conditioning in summer is about 70 W/m² (Fig. 15).

6. Assessment on Displacement Air Conditioning System in Residential House

6.1. Thermal Comfort

Temperature gradient in the occupant space from the ankle (0.1m high) to the face (for a sitting man is about 1.1m high), is within 2°C/m. According to the standard of ISO7730, temperature gradient is recommended within 3°C/m. From the guidance of SCANVAC state, temperature gradient of 2°C/m will cause 3% dissatisfaction degree.
The temperature gradient near the air inlet is about 3°C/m, which means 8% dissatisfaction, and the occupant will feel a little cool there. In the occupant space, it can be considered the temperature gradient caused by displacement ventilation has little effect on thermal comfort.

Moreover, according to the PMV distribution (Fig.12), PMV is below 0.5 under the height of 1.1m, which means a comfortable environment.

From 1.1m to 1.7m, PMV becomes greater and corresponds to 20% dissatisfaction degree. But to a sitting person there is few effect. Above 1.7m the average PMV is 1.5, which means a little hot, but it has no effect on occupants.

According to distribution of temperature, humidity, temperature gradient and PMV in occupant space, it can be concluded that comfortable indoor environment is realized.

6.2. Air Exchange Efficiency

Fresh air is induced directly to occupant space in displacement ventilation system. Distribution of CO₂ concentration shows the CO₂ concentration below 1.8m is 30% lower than that above 1.8m. In the occupant space, CO₂ concentration is about 600ppm, while the standard of building environment of Japan recommends to be smaller than 1000ppm.

As the air is supplied directly to the occupant space, the average air exchange efficiency is 1.25 according to the simulation result. While it is usually 1 for the completely mixed air conditioning system. According to the results of simulation, air exchange rate of displacement ventilation system is over 1. It shows air exchange efficiency of displacement ventilation system is better.

Displacement air conditioning system is considered to be a good air change method.

6.3. Energy Consumption of Air Conditioning System

In displacement ventilation system only occupant space (below 1.8m) is conditioned, the temperature of air near the ceiling is rather high, and the heat from the ceiling and light will be discharged with the exhaust air, will have little effect on indoor environment.

According to measured results, sensible heat load in summer is about 70W/m². In conventional air conditioning system, when the indoor air temperature is supposed as 27 or 26.5°C uniformly, the sensible heat load increases about 22% and 38% respectively (Fig.16). This means 20% ~ 40% energy will be saved by displacement ventilation system.

7. Conclusions

In this research field measurement and simulation of the displacement ventilation are conducted in a residential house. Results are as follows:

(1) It is confirmed that the air temperature is stratified vertically in the occupant space. The temperature is about 26~28°C under 1.8m, and it becomes 2~3°C higher above 1.8m. Humidity is about 50%~60%, while PMV is within 0.5 under 1.1m.

(2) Concentration of CO₂ in the occupant space is about 600ppm, and it is about 800ppm near ceiling. The air exchange efficiency of displacement ventilation is good because of exhaustion from ceiling.

(3) Because heat transfer of ceiling and heat release of ceiling lights have little effect on air conditioning load, the cooling load (sensible heat) is small. According to the experiment, the cooling load is about 70W/m², 20% ~ 40% lower than conventional systems.

In this paper the efficiency of displacement ventilation system in a residential room is confirmed, and further study will go on to make improvement in the displacement ventilation system, thus to popularize this system in residential houses.

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