Arrangement of Ceiling Fan for Airflow Control in Rooms by Coupled Simulation of Convection and Radiation

Yoshihisa Momoi¹, Kazunobu Sagara¹, Toshio Yamanaka¹ and Hisashi Kotani¹

¹Osaka University, Japan

Corresponding email: momoi@arch.eng.osaka-u.ac.jp

ABSTRACT

In large space, such as atria, it is possible to enumerate a problem that the high ceiling tends to generate a vertical temperature gradient due to the effect of buoyancy. This study examines the way of utilizing a ceiling fan for airflow control in a large air-conditioned room. In order to discuss the effect of the distance between the ceiling fan and the air supply openings on the air velocity distribution, temperature and thermal comfort inside the room, CFD simulation was conducted. Configuration factors were obtained by Monte Carlo method and radiation heat transfer calculation was conducted with Gebhart's absorption factor method. Convective heat was calculated using convective heat transfer coefficient. In the CFD simulation, two hypothetical boxes are assumed above and below the ceiling fan and measured values of air velocity and turbulent parameters were given as boundary conditions, in order to model the airflow produced by the ceiling fan. This study reveals that the standard deviation of PMV and the removal heat of FCU have the relation of ‘trade-off’. It was found out that actual installed location of the ceiling fan was appropriate.

INTRODUCTION

Airflows from horizontal supply openings often used a large air-conditioned room are affected by buoyancy, and quickly decline during cooling. Therefore, cold air may reach the occupants in the room, causing the discomfort. Furthermore, as there may be areas that the supply air does not reach, the temperature gradient within the room can easily become uneven. In particular, during heating, warm air remains near the ceiling, and this is also unhelpful from an energy conservation perspective. In this study, the prevention of cold drafts and the dissolution of temperature gradients in winter are proposed by converting supply airflows into upward flows, through the use of ceiling fans.

Previous studies [1] have empirically examined the effects of ceiling fans by carrying out measurements of indoor airflows and temperature gradients in summertime, in existing buildings with large spaces. They have examined also the numerical accuracy of Computational Fluid Dynamics (CFD) simulation in the large space, by comparing the measurements with CFD simulations based on the measurement data of wall temperatures, supply air velocities and temperatures. The measurement values of average air velocities and turbulence parameters near the ceiling fan were entered into CFD simulation as a ceiling fan airflow model [2]. In the previous study [3], it has been shown that when the distance between the ceiling fan and ceiling is above 40 cm, decreases in air volumes can be disregarded, and ceiling fan airflow models based on measurements taken in free space can be applied without problem.
When carrying out air-conditioning design for the large room with ceiling fans, if the effects of ceiling fan airflows on supply airflows are too great during cooling, temperature stratification will be lost due to airflows circulating vertically in the room. This may be considered unhelpful from an energy conservation perspective. Consequently, in this paper the effects of ceiling fan location on room airflow distribution and temperature gradient were examined, by conducting CFD simulation with ceiling fan location (distance from supply opening and height above supply opening) as a parameter, while fixing the boundary conditions of supply air velocities and temperatures. The airflow control effects of ceiling fans on energy efficiency and thermal comfort were also investigated through the use of Predicted Mean Vote (PMV).

**METHODS**

As the installation of ducts is difficult in the rehabilitation room of the welfare facility (Fig. 1), which has a high wooden ceiling, there is horizontal supply air-conditioning by fan coil units (FCU) inside the northern wall from a height of 2.65m above the floor. For many elderly people, particular consideration must be given to thermal environment in planning. Therefore, airflow control is performed by a ceiling fan (Matsushita Electric Industrial, F-M131H-W: Fig. 2) installed 3.1m above the floor. The position of the ceiling fan is designed to be above the path line of supply airflow, as predicted by equations of jet flow, where the angle of blow-off is set at 20° upward (Fig. 3).

Simulation was carried out in the calculation area (W: 13.5 m × D: 4.05 m × H: 2.8 m) assuming the calculation boundary between one ceiling fan and another to be a symmetrical plane with no inflow or outflow of air and heat (free-slip and insulation). The spatial division of calculation was 10 cm, and all conditions assumed the same number of spatial divisions (Fig. 4). After ceiling fan positions were changed from their actual installation positions to horizontal directions in three conditions (B1, B2, B4), and vertical directions in three conditions (A3, C3, D3), CFD simulation was carried out (Fig. 5). Calculation was also performed in the condition where the ceiling fan was turned off (S). The supply air velocity
was assumed to be 3.93 m/s, as it was when measured. In order to keep room temperature at its target value (25°C-27°C) when the ceiling fan was turned off, FCU supply air temperature was set at 24°C. The external temperature with exceedance probability of 5% (33.2°C) and the sol-air temperature SAT (65.7°C) in the summer determined by the typical meteorological year of expanded AMeDAS weather data were given as boundary conditions. The convection heat transfer rate of each surface was set at 4.0W/m²K. Configuration factors were calculated using the Monte Carlo method, while radiation heat transfer was calculated using the Gebhart’s absorption factor method. Ceiling fan airflows were assumed to be only in upward flows, so as to reduce drafts. The standard k-ε model was used as a turbulence model. All calculation conditions are shown in table 1.

For data of blow-off flow and suction flow of a ceiling fan, a hypothetical box with 140 cm in length and 10 cm in height was set 20 cm above and below the ceiling fan, as shown in Fig. 6. The horizontal planes of each box were divided into 10 cm sections, and on the mesh center of each plane the same data was given. The input data were given as: air velocity components U, V, W, turbulent kinetic energy k and energy dissipation rate ε [4, 5]. Among these, the air velocity components U and V, parallel to the boundary surface, were given as the momentum of airflow. For details of ceiling fan airflow models, please refer to the previous paper [2].
RESULTS AND DISCUSSIONS

Average Air Velocity Vectors and Average Air Velocity Distributions

The left and center diagrams in Fig. 7 show the calculation results of averaged air velocity vectors and air velocity distributions. In cases like B2, where supply airflows flow in the side of the model, the airflow data given to the ceiling fan airflow model is affected by the influence of supply airflows, and may diverge from airflow data measured in the free space. However, the volume of supply airflow is only 1/8 of the volume of the blow-off flow from the ceiling fan, and the airflow from the ceiling fan may become dominant near the ceiling fan. Therefore, the following discussions will take into account the fact that there remain problems with analytical accuracy in situations where supply outlets are near ceiling fans.

When the ceiling fan was turned off, supply airflows declined, and air velocities above 0.3 m/s flew towards the occupancy area, potentially causing drafts. By contrast, in B1 and B2, where ceiling fans were set at a height lower than the supply openings, it was seen that declines in supply airflows were controlled and diffused, and that direct incursion of supply airflows towards occupancy areas was prevented. In B3 and B4, as ceiling fans were removed further from the height of the supply openings, the effects of ceiling fan airflows on air velocity distribution decreased, and results approached those found under the condition where the ceiling fan was turned off. In A3, even though incursion of supply airflows into the occupancy area was observed, air velocity was only 0.2 m/s, which is insufficient to cause drafts. The reason the effects of the ceiling fan were smaller in A3 than B3 is that supply airflows are affected by the suction of the ceiling fan before they diffuse and dissipate, and so they maintain their kinetic momentum and blow through, without being sucked in by the ceiling fan. In C3 and D3, the air velocity of the occupancy area was almost the same as when the fan was turned off (S). The effects of the ceiling fan on supply airflows were not observed, and high velocity airflows reached the occupancy area.

Room Temperature gradient

The diagram on the right in Fig. 7 shows the calculation result of room temperature gradient. As declines of supply airflows are being controlled in B1 and B2, where supply openings is at a height lower than the ceiling fan, the temperature gradient in the occupancy area has a roughly uniform temperature of 0.5°C lower. However, because cold air blown off can easily mix with warm air in the upper parts of rooms in these conditions, the temperatures in lower
parts of the room are 0.5°C higher than when the ceiling fan is turned off. In B3 and B4, as ceiling fans moved further above the height of the supply openings, the effect of ceiling fan airflows on temperature gradient decreased, and approached the results found when the fan was turned off (S). In A3, supply airflows are affected by the suction of the ceiling fan before they diffuse and dissipate, and lower-temperature air mixed with high-temperature air in the rooms upper parts, thereby causing the temperature in the occupancy area to be 1°C above the temperature found when the fan was turned off (S). In C3 and D3, where effects of the ceiling fan on supply airflows were not observed, supply airflows declined, and the temperature gradient was almost the same as when the fan was turned off (S).

Figure 7. CFD simulation results with location of ceiling fan changed. Left) airflow vectors, Center) air velocity distribution, Right) air temperature distribution
Relations between Averaged PMV, Standard Deviation of PMV and Removal Heat of FCU

PMV was calculated from indoor air velocity, air temperature, and wall temperature obtained by CFD simulation, assuming a metabolic rate of 1met (58.2W/m²), clothing thermal insulation of 0.5clo and humidity of 60%.

Fig. 8 shows calculation results of PMV in a cross section 1.1m above floor level. On this occasion, with supply air temperature set as a boundary condition of CDF simulation, room temperature was set so as to reach target room temperature (25°C-27°C) when the fan was turned off. However, due to the effects of radiation on ceiling surfaces, the PMV was calculated slightly higher. In the condition where the fan was turned off (S), in the position of 3.5m from the supply opening the PMV declined and indicated a negative value. In conditions where the ceiling fan was near the supply opening (B1, B2, B3, B4 and A3) PMV standard deviations were smaller than those found in S, and regional cold draft decreased. On the other
hand, in conditions where the ceiling fan was far from the supply openings (C3, D3) PMV standard deviations were roughly the identical to those found in S, and no reductive effects of ceiling fans on cold drafts were observed.

Fig. 9 and Fig. 10 show the relation between cooling energy required to treat heat flowing into the room (the removal heat of FCU), and averages and standard deviations of PMV. Due to the mixture of indoor air caused by the ceiling fan, the temperature stratification was disturbed and the temperature near the ceiling decreased. The removal heat of FCU was larger in all conditions than the condition when the ceiling fan was turned off, resulting in inefficient air-conditioning. In A3, where the ceiling fan was near the supply openings, much of the cold air was carried to the upper part of the room by the ceiling fan, resulting in a large removal heat of FCU. In situations where the ceiling fan was near the supply openings (A3, B1, B2), although removal heat of FCU were large, standard deviations of PMV were small. On the other hand, in the situation where the fan was turned off (S), and in C3 and D3, where the ceiling fan was far from the supply openings, although removal heat of FCU were small, standard deviations tended to be large.

![Figure 9. Relationship between average of PMV and removal heat of FCU.](image1)

![Figure 10. Relationship between standard deviation of PMV and removal heat of FCU.](image2)

**CONCLUSIONS**

CDF simulation was performed with ceiling fan position as a parameter, and calculations of PMV and the removal heat of FCU were calculated. From this it could be seen that there is a trade-off relationship between reduction of drafts from ceiling fans and reductions in the removal heat of FCU. In addition, actual installation positions of ceiling fan were appropriate from the perspectives of the energy conservation and the thermal comfort. This paper conducted CFD simulation with the ceiling fan position changed. However, it is conceivable that rotational speed of the ceiling fans may also heavily affect airflow control. Thus, in future it will be necessary to consider rotational speed as well as ceiling fan locations when planning air-conditioning with ceiling fans.

**REFERENCES**


