Study the Thermal Effect of Ceiling-mounted Light Fixtures for a Displacement Ventilation System

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ABSTRACT

This paper investigates the thermal effect of ceiling-mounted light fixtures for a displacement ventilation system. The influence of fluorescent lamps on a ceiling is examined by computational fluid dynamics and experimental methods. The test room is a 5.3m ×5.44m ×2.5m height office. Two supply air diffusers and two exhaust grilles are dimensioned by 0.9m x 0.2m each. A heat source is installed at the center of the room. Supply air velocity is treated as a parameter from 0.1 to 0.5 m/s. The finite volume method and low-Reynolds number k-ε turbulence model are employed for CFD analysis. Thermocouples are installed vertically above floor at the center of the room for the experiments. Temperature data are collected automatically by a data logger every minute.

Dimensionless vertical temperature profiles are compared for CFD and experimental results for each supply air velocity. Thermal stratification in the test room is investigated at 8 points of the room by CFD analysis.

As a result, dimensionless vertical temperature profiles indicate the similarity for different supply air velocities. Ceiling-mounted light fixtures stabilize thermal stratification as well as heating by ceiling panels according to CFD results. Thermal comfort in the test room is assessed according to ISO7730-1994 recommendations for the test room.

1. INTRODUCTION

Displacement ventilation is widely applied to various types of buildings as presented by Skistad (1994) and Skistad et al (2002). Previous papers presented by Hashimoto (2004) and Hashimoto (2005) investigated thermal comfort in the test room numerically. This study compares the experimental and numerical results for the test room with ceiling-mounted light fixtures. The previous papers exclude lighting load. This study purposes to validate the influence of lighting load from ceiling-mounted lighting fixtures.

2. METHODLOGY

2.1 Experiment

The conditions of the experiments are indicated in Table 1. Experiments are carried out to measure the vertical and horizontal temperature distributions in a small room which dimensions 5.3m by 5.44m by 2.5m height equipped with displacement ventilation. The internal loads are 40W×24 of ceiling-mounted light fixtures and two 1.25kW heaters. The average supply air velocities are 0.1 to 0.4m/s and the supply air temperature is 20 ° C. Fig.1 shows the picture of the test room. The wall of the test room is made of vinyl sheets inside the laboratory and kept of penetrating heat flow from outside.

The layout of the test room is illustrated as Fig.2, referred to the previous papers. Two supply air grilles and two exhaust air grilles are dimensioned for 0.2m by 1.0m each. Two supply air grilles are installed quite near above floor. Two exhaust air grilles are installed at the opposite wall near the ceiling. Two 1.25kW heaters are installed in a steel mesh box of 0.9m cubic at the center of the test room to simulate the internal cooling loads. Thermocouples are installed in the test room to measure the vertical temperature distribution. The data are collected by a data logger automatically every minute, connected to the sensors.

2.2 CFD

A commercial CFD code (The STREAM for Windows) is used to predict flow and temperature field for a displacement ventilation system. The governing equations are the incompressible Navier-Stokes equations and the continuity equation in Cartesian coordinates. The Boussinesq hypothesis is used for the buoyant force term. The steady-state calculations are carried out for all the cases.

The standard k- ε turbulence model is widely used to analyze a turbulent flow field. However, this model is assumed to apply for a fully turbulent flow and indicated inappropriate for a buoyant flow. This paper applies the non-linear low Reynolds number k- ε turbulence model with a damping function to reduce the turbulent viscosity near a wall (the AKN Model, Abe et al.(1994)). The used damping function is described as follows;

$$\nu_t = C_\mu f_\mu \frac{k^2}{\varepsilon} \tag{1}$$

$$f_\mu = \left[1 - \exp\left(-\frac{y^*}{14}\right)\right]^2 \left[1 + \frac{5}{R_t^{3/4}} \exp\left\{-\left(\frac{R_t}{200}\right)^2\right\}\right] \tag{2}$$

where

ν_t=turbulent viscosity

C_μ= turbulence model coefficient

f_μ= damping function for the AKN model

R_t= turbulent Reynolds number

y^*= wall normal coordinate

This turbulence model is evaluated numerically stable.

The conditions for numerical calculations are shown in Table 2. The test room has a dimension of 5.3m by 5.44m by 2.5m height to compare with the experimental results. Air outlets are located at the bottom of the wall and air inlets are at the top of the opposite wall. The number of grid cells is 68×112×60 =456,960 for the test room. Supply air velocity is varied for a variety of total heat source capacity of 1.25kW and 2.5kW. Lighting load is given as 3 rows of 320W heaters installed on the ceiling. Each case is examined without lighting load and

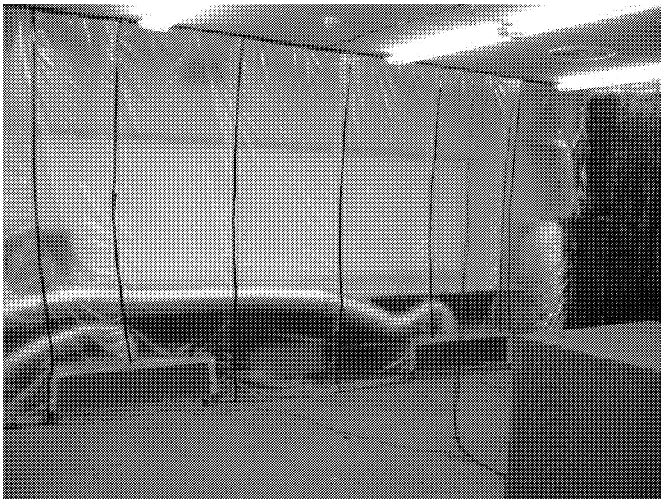


Fig.1 The Test Room

Table 1 Conditions for Experiments

Items	Conditions
Dimension of the test room	5.3m×5.44m×2.5m Height ($72.1\ m^3$)
Supply air velocity	0.1, 0.2, 0.3, 0.4 m/s (2.0, 4.0, 6.0, 8.0 ACH)
Supply air temperature	20°C
Horizontal points	3 points shown in Fig.2
Height above floor of vertical points	FL+0, 0.1, 0.6, 1.1, 1.5, 2.0, 2.4, 2.5m (H=2.5m)
Internal loads	Heat source 1.25kW / 2.5kW Lighting fixtures 40W×24

Table 2 Conditions for CFD

Items	Conditions
Turbulence model	non-linear low Reynolds number k- ε model (the AKN model)
Dimension of the test room	5.3m×5.44m×2.5m Height ($72.1\ m^3$)
Supply air velocity	0.1, 0.2, 0.3, 0.4, 0.5 m/s
Ceiling height	H=2.5m
Horizontal points	8 points shown in Fig.2 (except No.2)
Height above floor of vertical points	FL+0, 0.1, 0.6, 1.1, 1.5, 2.0, 2.4, 2.5m
Heat capacity	Heat source 1.25kW / 2.5kW Lighting fixtures 320W×3

Table 3 Variety of Internal Loads

Case Name	Heat Source	Lighting Load
CASE1	1.25kW	No
CASE2	2.5kW	No
CASE1L	1.25kW	Lighting Load
CASE2L	2.5kW	Lighting Load

with lighting load. Table 3 presents the CFD cases according to the variety of internal loads. Thus, 20 cases of numerical simulations are totally carried out for the test room with a displacement ventilation system. The case names of CASE1L and CASE2L are commonly used for the experimental cases.

3. RESULTS AND DISCUSSIONS

3.1 Dimensionless Vertical Temperature Profiles

Dimensionless temperature θ is defined as follows;

$$\theta = \frac{T - T_s}{T_e - T_s} \tag{3}$$

where T is a local temperature of each measurement point, Ts is supply air temperature and Te is exhaust air temperature.
Dimensionless height is defined below;

$$h^* = \frac{h}{H} \tag{4}$$

where h is a local height and H is the ceiling height (H=2.5m).
Dimensionless vertical temperature profiles are compared for CFD and experimental results for each supply air velocity for the test room of 2.5m height. Fig.3 shows the dimensionless vertical temperatures for the experimental results for the supply air velocities of 0.1 to 0.4m/s at the point of No.1.
The dimensionless vertical temperatures from the experimental results are observed nearly similar for the variety of supply air temperatures for CASE1L and CASE2L. The results show that displacement ventilation is suitable for a VAV system (a Variable Air Volume system) to save fan power for the variations of the internal cooling load as the previous papers investigate.
The dimensionless vertical temperature gradients are considered almost linear from the floor to the ceiling. This means that the stratification inside the test room is quite stable.

3.2 Influence of Ceiling-mounted Light Fixtures

Fig.4 to Fig.7 present the dimensionless vertical temperature profiles for the cases without lighting load and with lighting load in case of 1.25kW and 2.5kW heater load. It is known that heated ceiling panels stabilize the thermal stratification for displacement ventilation. As heated ceiling panels, ceiling-mounted lighting fixtures are considered to stabilize vertical temperature profiles.

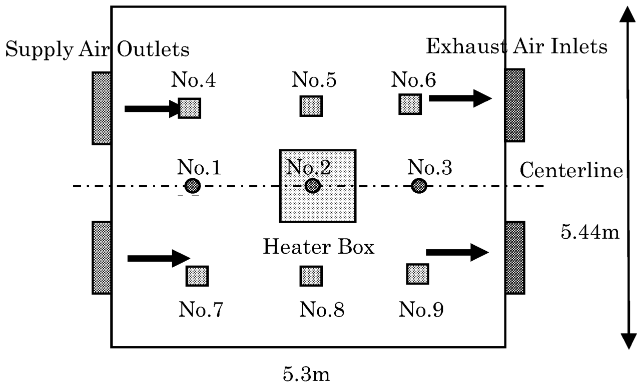
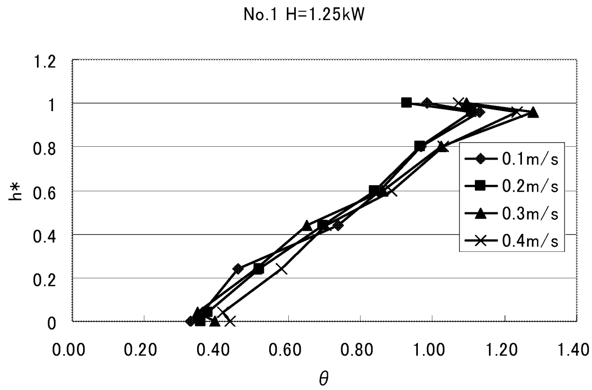
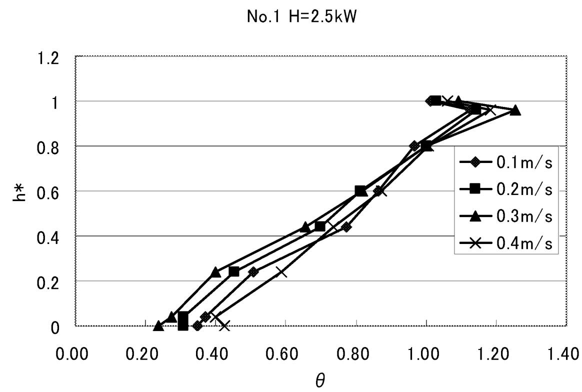


Fig.2 Layout of the Test Room

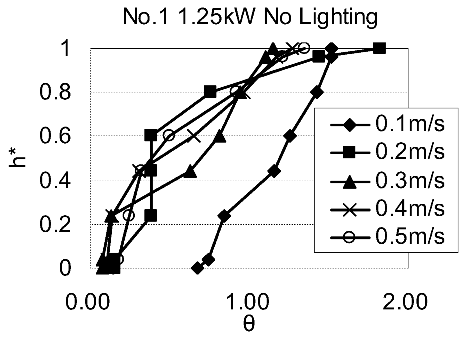


(a) CASE1L

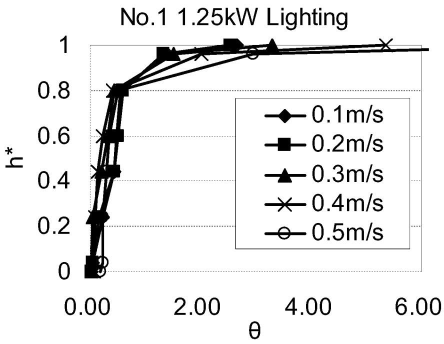


(b) CASE2L

Fig.3 Dimensionless Vertical Temperature Profiles for the Experimental Results (at the point of No.1)

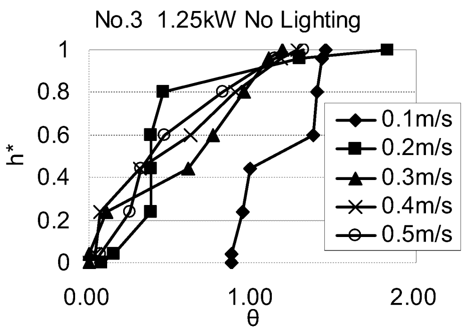


(a) CASE1

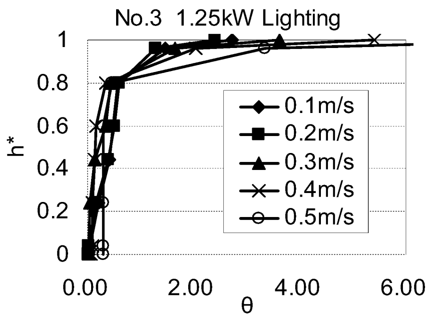


(b) CASE1L

Fig.4 Dimensionless Vertical Temperature Profiles for the CFD Results (1.25kW No.1)

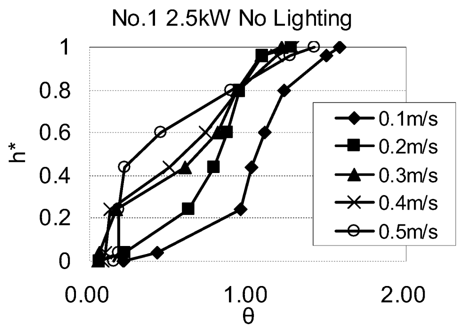


(a) CASE!

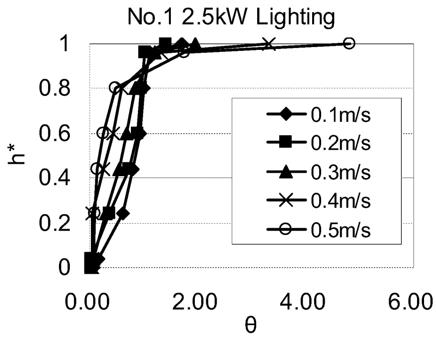


(b) CASE1L

Fig.5 Dimensionless Vertical Temperature Profiles for the CFD Results (1.25kW No.3)

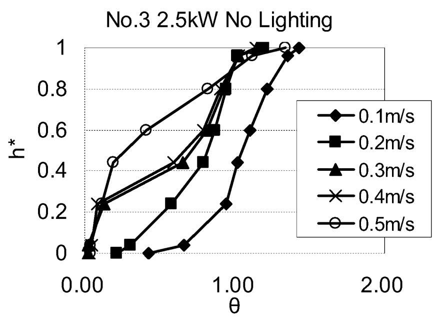


(a) CASE2

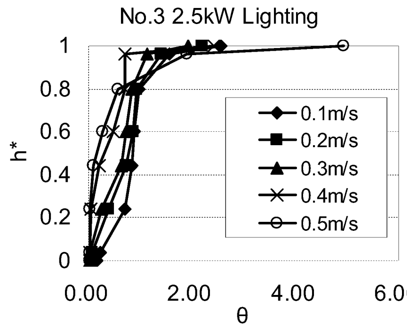


(b) CASE2L

Fig.6 Dimensionless Vertical Temperature Profiles for the CFD Results (2.5kW No.1)



(a) CASE2



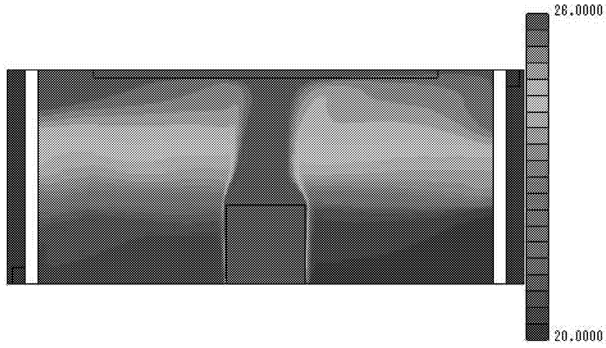
(b) CASE2L

Fig.7 Dimensionless Vertical Temperature Profiles for the CFD Results (2.5kW No.3)

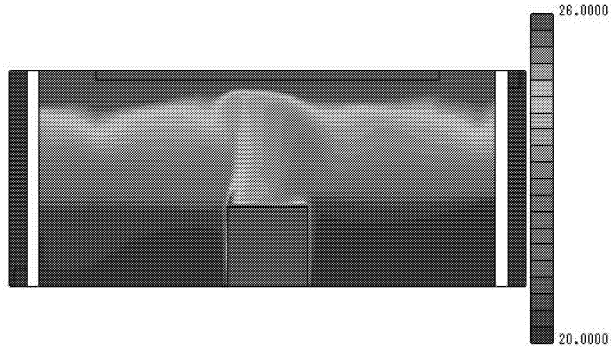
In CASE1 and CASE2, the dimensionless vertical temperature profiles are quite different for the variety of supply air velocity. However, in CASE1L and CASE2L, the vertical temperature profiles are observed nearly similar for the variety of supply air velocity in the occupied zone. Vertical temperature gradients are becoming greater close to the ceiling with lighting load. The reason is considered that the given lighting load for the numerical calculations is overestimated compared to the experimental conditions. No.1 is located at upward of the heat source and No.3 is at leeward. Nevertheless, the dimensionless similarity of vertical temperature profiles as the vertical temperature profiles of No.1 and No.3 are considered close. This means that the heat source does not influence horizontally for a displacement ventilation system.
Generally, the dimensionless vertical temperature gradients in the occupied zone are observed larger in case of 2.5kW heater load than in case of 1.25kW.
Roughly speaking, the dimensionless vertical temperature gradients in the occupied zone are not so different between the experimental and numerical results in case of giving the lighting load except near the ceiling and floor. The dimensionless temperatures on the ceiling by the CFD results are extremely larger and those on the floor are lower compared with the experimental results. The reasons which caused the differences between the experimental and numerical results near the ceiling and floor are considered that the influences of radiation from the ceiling-mounted lightings are neglected and the floor and ceiling are regarded as adiabatic surfaces for the CFD.
Fig.8 presents typical vertical temperature contours at the centerline of the test room for CASE1 and CASE1L. Vertical temperature contours in the occupied zone are likely to be more favorable for CASE1L than CASE1. Plume from the heat source comes straight upward and does not spread horizontally. The influence of the heat source is not observed horizontally from the vertical temperature contours for both cases.

3.3 Evaluation f Thermal Comfort

Table 4 shows the temperature differences



(a) CASE1, v=0.3m/s



(b) CASE1L, v=0.3m/s

Fig.8 Typical Vertical Temperature Contours

between 0.1m and 1.1m above floor with 1.25kW and 2.5kW heat source at 8 points shown in Fig.2 except No.2 which is the location of the heat source. ISO7730 recommends the vertical temperature difference should be less than 3 ° C for thermal comfort. For CASE1 and CASE1L, ISO7730 recommendation is satisfied when the supply air velocity is 0.3 to 0.5m/s. However, for CASE2 and CASE2L, supply air velocity to meet the recommendation is different. The temperature differences are observed smaller for the supply air velocity of 0.4m/s in CASE2L than in CASE2. The lighting load seems to influence on the vertical temperature gradients.

4. CONCLUSIONS

This study investigates vertical temperature profiles for a displacement ventilation system by experimental and numerical methods for a variety of supply air velocity and internal loads.
- As the experimental result, dimensionless

vertical temperature profiles are observed nearly similar for a variety of supply air velocity.
- From the CFD simulations, differences of the vertical temperature contours are indicated due to the presence of ceiling-mounted lighting fixtures. By giving lighting load on the ceiling, dimensionless vertical temperature profiles become nearly similar.
- As the CFD results, the temperature differences between 0.1m and 1.1m above floor are presented smaller in CASE1L and CASE2L than in CASE1 and CASE2 for higher supply air velocity.

A displacement ventilation system is recommended to satisfy thermal comfort for occupants and energy saving of fan power for an office space.

Future study concerns the influences on a ceiling height for a displacement ventilation system.

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Table 4 Temperature Differences between 0.1m and 1.1m above Floor (CFD) [°C]

(a) CASE1

	No.1	No.3	No.4	No.5	No.6	No.7	No.8	No.9
0.1m/s	6.9	7.1	7.5	7.2	7.2	7.3	7.0	7.4
0.2m/s	5.6	4.6	5.3	4.7	4.4	5.6	5.0	4.7
0.3m/s	2.6	2.9	3.0	2.9	3.0	2.8	2.9	2.9
0.4m/s	0.1	0.2	0.5	0.5	0.4	0.5	0.5	0.4
0.5m/s	0.1	0.1	0.5	0.5	0.4	0.5	0.5	0.4

(b) CASE1L

	No.1	No.3	No.4	No.5	No.6	No.7	No.8	No.9
0.1m/s	8.6	5.5	6.2	6.1	5.6	5.9	5.4	5.5
0.2m/s	8.2	8.5	9.9	8.4	8.7	9.0	8.2	8.3
0.3m/s	1.6	1.5	2.0	1.7	2.1	1.9	1.9	1.4
0.4m/s	0.5	0.7	1.1	1.0	1.0	1.2	1.0	0.7
0.5m/s	0.1	0	0.5	0.4	0.4	0.5	0.4	0.4

(c) CASE2

	No.1	No.3	No.4	No.5	No.6	No.7	No.8	No.9
0.1m/s	7.6	4.6	7.8	6.4	0.7	5.8	6.2	5.4
0.2m/s	6.7	5.7	7.1	5.9	6.1	7.8	6.5	6.6
0.3m/s	5.1	6.0	5.8	5.9	5.8	6.4	6.2	5.5
0.4m/s	2.8	3.8	3.7	3.6	3.4	3.8	3.5	3.7
0.5m/s	0.1	0.4	0.8	0.7	0.6	0.8	0.7	0.6

(d) CASE2L

	No.1	No.3	No.4	No.5	No.6	No.7	No.8	No.9
0.1m/s	8.4	8.1	9.4	8.2	7.6	8.8	8.2	8.1
0.2m/s	12.3	11.8	12.7	13.6	13.2	12.5	11.5	12.3
0.3m/s	9.2	11.1	12.2	9.7	8.7	11.3	10.7	8.9
0.4m/s	1.7	1.6	2.8	2.2	2.2	2.3	2.5	1.9
0.5m/s	0.3	0.3	0.8	0.7	0.5	0.8	0.6	0.5