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NUMERICAL STUDY ON THE INFLUENCE OF A CEILING HEIGHT FOR DISPLACEMENT VENTILATION

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ABSTRACT

This study analyses vertical temperature profiles for displacement ventilation depending on a ceiling height of an small office room. This study purposes to investigate the influence of the ceiling height on thermal comfort of the occupants. Numerical simulations are carried out for a modeled room which dimensions 5.3m by 5.44m by 2.5m, 3m and 3.5m ceiling height. A finite volume method is used for solving the governing equations and low Reynolds number k-ε model is applied as a turbulent model. A heat source is installed at the centre of the modeled room and lighting load is given on the ceiling. The surface temperature of the ceiling-mounted lighting fixtures is given as a boundary condition by measuring surface temperature of fluorescent lamps using a thermograph. Normalised vertical temperature profiles are compared for each ceiling height room. Numerical results are compared to experimental results for 2.5m ceiling height room. As a result, thermal comfort is surveyed for different ceiling height rooms for displacement ventilation.

INTRODUCTION

Displacement ventilation has been widely applied to different types of buildings. Theoretical and experimental studies have reported in many aspects. Skistad (1994) and Skistad et al. (2002) summarise technical information on displacement ventilation. Chen and Glicksman (2003) report a design guideline for displacement ventilation. However, the effects of ceiling heights of the room have rarely been discussed for small offices. Previous papers presented by Hashimoto (2004), Hashimoto (2005) and Hashimoto and Yoneda(2008) investigated thermal comfort in a test room numerically for displacement ventilation. Last paper reports that ceiling-mounted light fixtures stabilise thermal stratification as well as heating by ceiling panels according to CFD results. This study analyses vertical temperature profiles for displacement ventilation depending on a ceiling height of an office room. This study purposes to investigate the influence of the ceiling height on thermal comfort of the occupants. This paper explores the thermal effect of ceiling height for a small office room for a displacement ventilation system by using computational fluid dynamics analysis. It is difficult to examine the influences of a ceiling height by an experimental method. Therefore, the influence of a ceiling height is examined by computational fluid dynamics.

Firstly, results from an experimental and numerical method are compared. The test room for the experiments is a 5.3m×5.44m ×2.5m height room. Two supply air diffusers and two exhaust grilles are dimensioned by 0.9m×0.2m each for displacement ventilation. A localised heat source is installed at the centre of the test room. Supply air velocity is given from 0.1 to 0.4 m/s. Thermocouples are installed vertically above floor at the centre of the room for the experiments. Numerical calculations are carried out to compare to the experimental results. The finite volume method and linear low-Reynolds number k-ε turbulence model are employed for CFD analysis.

Secondly, the influence of a ceiling height of a test room is discussed by using CFD analysis. The common conditions for the numerical calculations are given for a different ceiling height of the room geometry.

Finally, normalised vertical temperature profiles are compared by using CFD results for each supply air velocity and ceiling height. Thermal stratification in the test room is investigated at the centreline of the room by CFD analysis.

As a result, normalised vertical temperature profiles indicate the similarity for different supply air velocities for experimental data. Thermal comfort in the test room is assessed according to ISO7730-1994 recommendations for the test room of a different ceiling height.

METHODOLOGY

Experimental Study

Experiments are performed in a full-scale room set up like a small office room. 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 that 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 regulated to 0.1 to 0.4m/s and the supply air temperature is controlled at 20°C. Fig.1 shows the picture of the test room. The wall of the test room is made of vinyl sheets inside a laboratory and avoids 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 0.9m each. Two supply air grilles are installed quite near above floor. Two exhaust air grilles are installed near the ceiling at the opposite wall. Two 1.25kW heaters are put in a steel mesh box of 0.9m cubic at the centre of the test room to give an internal cooling load for a typical office room. Copper-constantan thermocouples are installed in the test room to measure vertical temperature distributions at 3 horizontal points (d, e and f in Fig.2). Thermocouples are also attached at the surfaces of supply air outlets and exhaust air inlets. Temperature data are collected by a data logger automatically every minute. Vertical temperature profiles are evaluated at steady-state.

CFD Analysis

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 performed 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 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;

$$\nu_t = C_\mu f_\mu \frac{k^2}{\varepsilon} \qquad (1)$$

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

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 listed in Table 2. The whole computational domain of 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

Fig.1 The Test Room

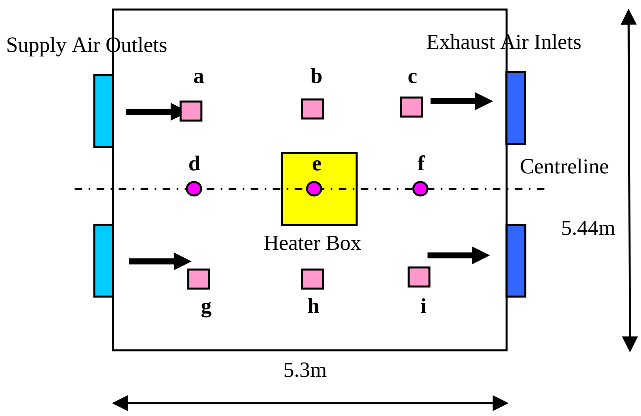


Fig.2 Layout of the Test Room

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 measuring points	3 points shown in Fig.2 (d, e,f)
Vertical measuring points	FL+0, 0.1, 0.6, 1.1, 1.5, 2.0, 2.4, 2.5m (CH=2.5m)
Internal loads	Heat source 1.25kW ($50W/m^2$) / 2.5kW ($100W/m^2$) Lighting fixtures 40W×24

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. The boundary condition for the lighting fixtures is given as the surface temperature of 45° C. It is obtained by measuring the surface temperature of the lighting fixtures on the ceiling with a thermograph. Last paper overestimated heat flow from the lighting fixtures. In addition to the test room described above, test rooms with CH=3.0 and 3.5m are given for the comparison. The grid cells are 68 × 112 × 68 =517,888 and 68×112×77 =586,432, relatively. Radiation is neglected for the CFD analysis.

Table 3 presents the CFD case names according to the variety of ceiling heights, supply air velocities and internal loads. Thus, 3 × 5 × 2=30 cases of numerical simulations are totally performed for the test rooms of different ceiling height with a displacement ventilation system.

RESULTS AND DISCUSSIONS

Normalised Vertical Temperature Profiles from the Experiment

Normalised temperature θ is defined as follows;

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

where T is a local air temperature of each measurement point, T_s is supply air temperature and T_e is exhaust air temperature.

Normalised height is defined below;

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

where h is a local height and H is the ceiling height (CH=2.5m, 3.0m, 3.5m) of the test room.

Normalised 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 normalised vertical temperatures for the experimental results for the supply air velocities of 0.1 to 0.4m/s at the point of d.

The normalised vertical temperatures from the experimental results are observed nearly similar for the variety of supply air temperatures for CASE A1a to A4a and CASE A1b to A4b. 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 present.

The normalised vertical temperature gradients are considered almost linear from the floor to the ceiling. This means that the thermal stratification inside the test room is quite stable.

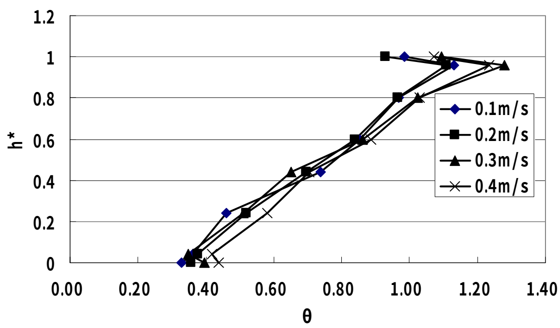
Table 2 Conditions for CFD

Items	Conditions
Turbulence model	linear low Reynolds number k-ε model (the AKN model)
Dimension of the test room	5.3m×5.44m×Ceiling Height
Supply air velocity	0.1, 0.2, 0.3, 0.4, 0.5 m/s
Ceiling height	CH=2.5m, 3.0m, 3.5m
Internal loads	Heat source 1.25kW / 2.5kW Lighting fixtures 320W×3 Surface temperature 45° C

Table 3 Names of Cases

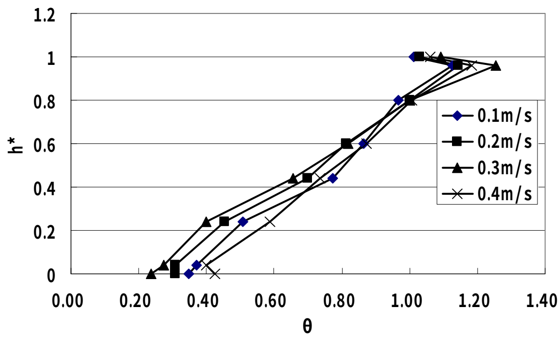
Ceiling Height	Supply Air Velocity	Heat Source Capacity
2.5m=A	0.1m/s=1	1.25kW=a
3.0m=B	0.2m/s=2	2.5kW=b
3.5m=C	0.3m/s=3	
	0.4m/s=4	
	0.5m/s=5	

d



(a) CASE A1-4a (H=1.25kW)

d



(b) CASE A1-4b (H=2.5kW)

Fig.3 Normalised Vertical Temperature Profiles for the Experimental Results (at the point of d)

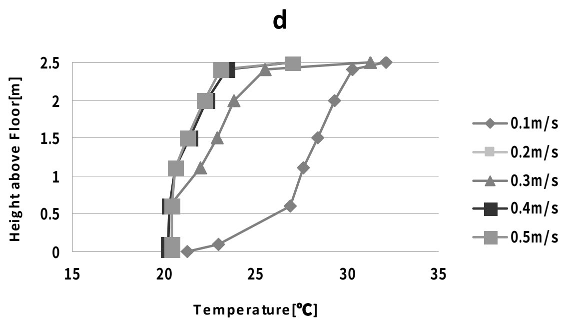
Influence of a Ceiling Height on Vertical Temperature Profiles

Fig.4 to Fig.6 present the vertical temperature profiles at the horizontal point of d for the cases with CH=2.5, 3.0 and 3.5m in case of 1.25kW and 2.5kW heater load with the variety of the supply air velocity from the CFD results. To obtain smaller vertical temperature differences, larger supply air velocity is needed. For 2.5kW internal load, this tendency is observed more clearly.

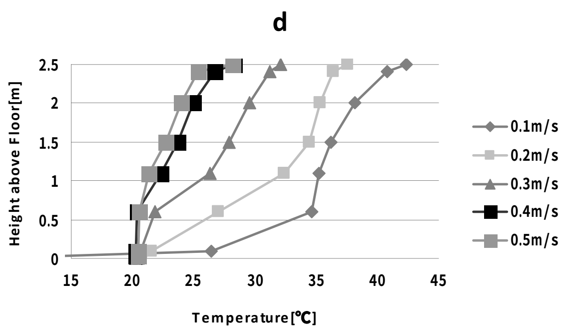
As the supply air velocity increases, vertical temperature differences decreases in the occupied zone, if the whole tendency is roughly estimated. Vertical temperature profiles are more different in case of larger capacity of a heat source, when the supply air velocity changes.

As the ceiling height of the room is higher, vertical temperature differences decrease for the equal heat source capacity and the equal supply air velocity. This means that a room with a higher ceiling height is more suitable for displacement ventilation. The similar tendency is observed at any other horizontal positions.

As stated previously, vertical temperature profiles will be controlled if a VAV system is employed and responded by a measured temperature in an occupied zone. Measured temperature in the occupied zone feeds back to supply air volume and improves the vertical temperature profile.

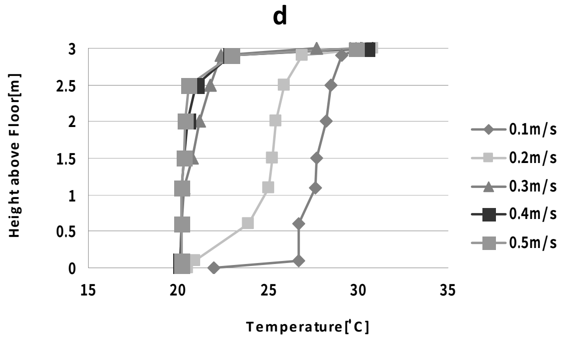


(a) H=1.25kW

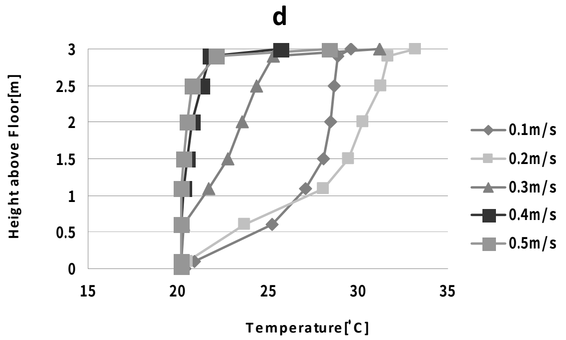


(b) H=2.5kW

Fig.4 Vertical Temperature Profiles for the CFD results (CH=2.5m)

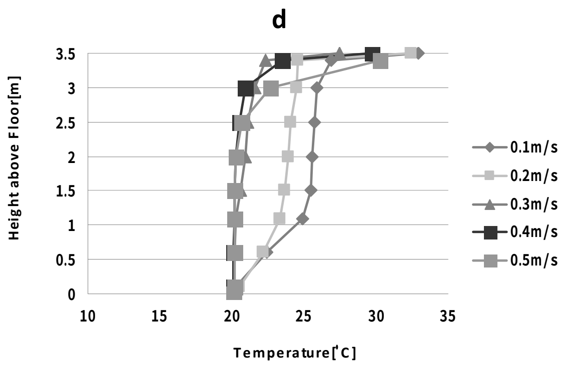


(a) H=1.25kW

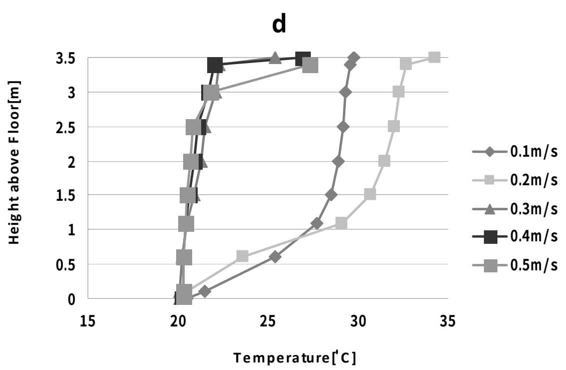


(b) H=2.5kW

Fig.5 Vertical Temperature Profiles for the CFD results (CH=3.0m)



(a) H=1.25kW



(b) H=2.5kW

Fig.6 Vertical Temperature Profiles for the CFD results (CH=3.5m)

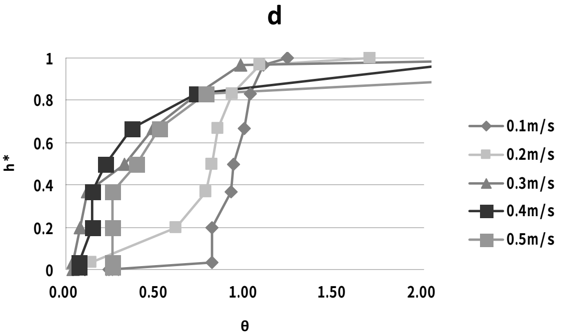
Influence of a Ceiling Height on Normalised Vertical Temperature Profiles

Fig.7 to Fig.9 present the normalised vertical temperature profiles for a variety of suppy air velocity in case of CH=2.5m, 3.0m and 3.5m. Similarity is obtained for each ceiling height when the supply air velocity is higher.

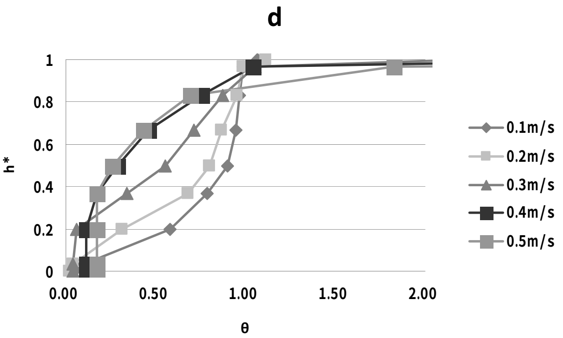
The normalised vertical temperature profiles on the surfaces of the floor and the ceiling are quite different from the experimental results. The reason is considered that the effect of radiation is completely neglected in all cases.

Approximate similarity is generally observed when supply air velocity is 0.4 and 0.5m/s for any cases. Non-linearity appears especially near the ceiling. The reason is considered that the surface temperature of the ceiling is overestimated due to the neglect of radiation.

Although normalised vertical temperature gradients show large in the occupied zone at low supply air velocities, they appear relatively small at high supply air velocities.

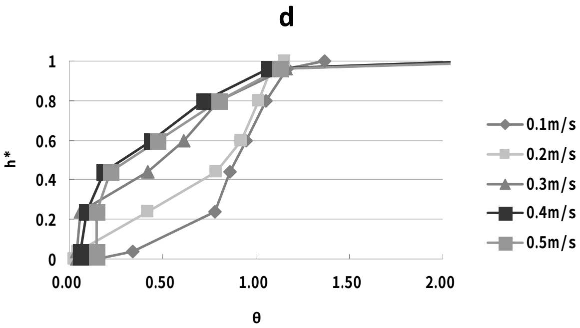


(a) H=1.25kW

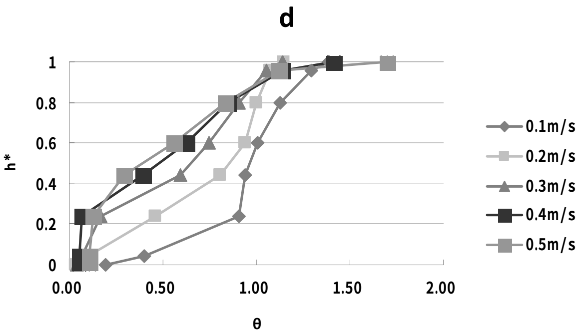


(b) H=2.5kW

Fig.8 Normalised Vertical Temperature Profiles for the CFD results (CH=3.0m)

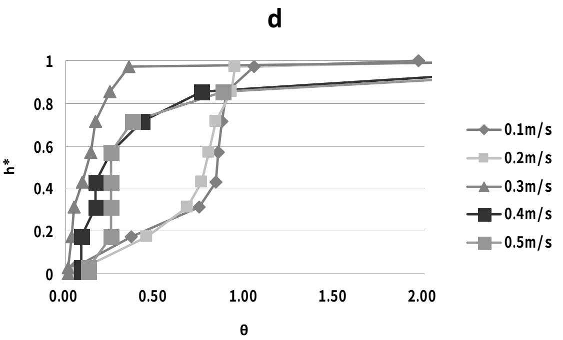


(a) H=1.25kW

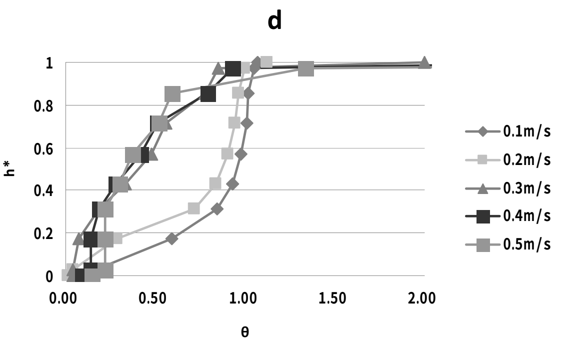


(b) H=2.5kW

Fig.7 Normalised Vertical Temperature Profiles for the CFD results (CH=2.5m)



(a) H=1.25kW



(b) H=2.5kW

Fig.9 Normalised Vertical Temperature Profiles for the CFD results (CH=3.5m)

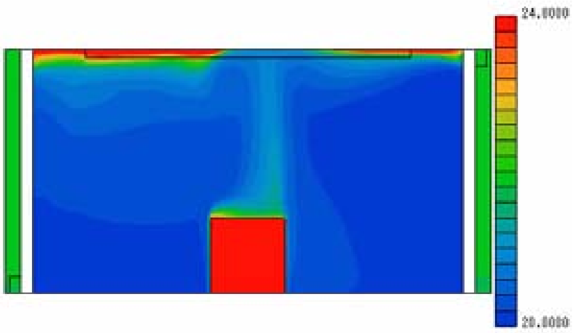
Influence of a Ceiling Height on Vertical Temperature Contours

Fig.10 to Fig.12 show typical vertical temperature contours at the cemtreline of the room from the CFD results at supply air velocity of 0.5m/s in case of CH=2.5m, 3.0m and 3.5m. The colour bars commonly range between 20 to 24 ° C in the all figures.

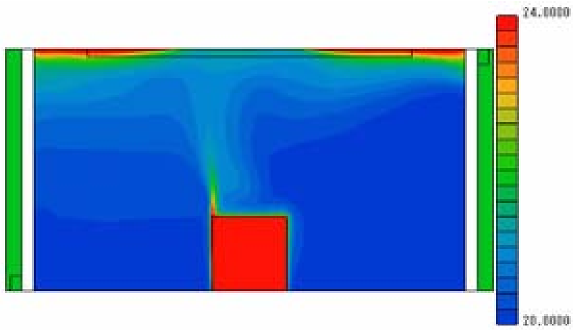
Buoyant plumes develop above the localised heat source. Thermal stratification is clealy observed in any cases. Thermal plumes do not expand to horizontal directions in the occupied zones. Temperature is more clearly stratified in case of 2.5kW heat capacity. An upper zone near a ceiling serves as a buffer space of heat plumes. A heat plume once stays in an upper zone near a ceiling, and is exhausted from exhaust inlets. Therefore, vertical temperature gradients decrease in a higher ceiling height room due to a larger upper space.

When the ceiling height is 2.5m, the thermal influence of a heat source is observed in the occupied zone from temperature contours. However, it is not clearly presented when the ceiling height is 3.0m and 3.5m as described above.

Although the heat plumes come from the upper right side of the heat source in case of H=1.25kW, they rise from the upper left side in case of H=2.5kW. The reason can be explained due to the strength of heat flux.

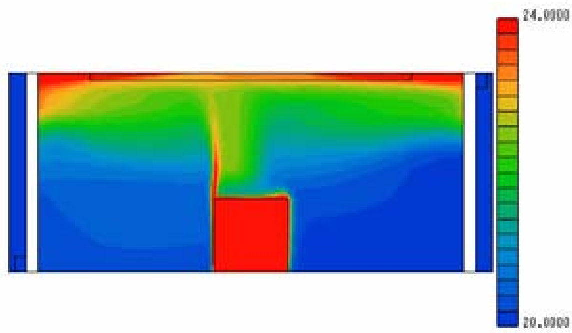


(a) H=1.25kW

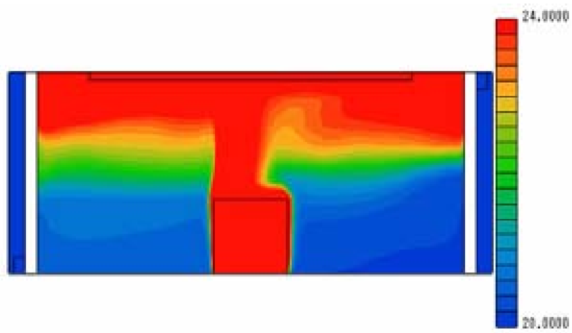


(b) H=2.5kW

Fig.11 Vertical Temperature Contours for the CFD results (CH=3.0m, v=0.5m/s) [° C]

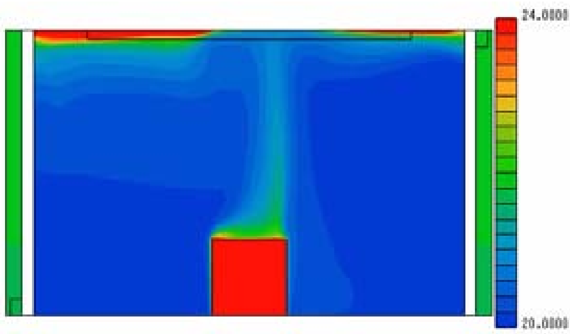


(a) H=1.25kW

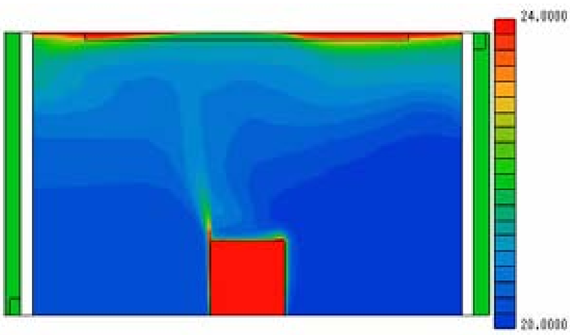


(b) H=2.5kW

Fig.10 Vertical Temperature Contours for the CFD results (CH=2.5m, v=0.5m/s) [° C]



(a) H=1.25kW



(b) H=2.5kW

Fig.12 Vertical Temperature Contours for the CFD results (CH=3.5m, v=0.5m/s) [° C]

Influence of a Ceiling Height on Thermal Comfort

ISO7730-1994 recommends that temperature difference between 0.1m and 1.1m above floor should be within 3 °C for thermal comfort. Temperature differences between 0.1m and 1.1m above floor at 8 points are shown in Table 4, 5 and 6 for different ceiling height of the room with displacement ventilation from the CFD results. Temperature differences decrease as the ceiling height is larger or as the supply air velocity increases. Generally, a higher ceiling height room improves thermal comfort for displacement ventilation. Therefore, a higher ceiling room helps to save energy by decreasing supply air volume at the equal internal load.

Temperature differences are relatively smaller at lower supply air velocity in case of CH=3.0m and H=1.25kW. The reason is estimated that temperature at 0.1m above floor is rather high.

Temperature differences between 0.1m and 1.1m above floor are almost uniform horizontally. This shows that temperature stratification is uniformly created in a room.

Table 7 presents temperature differences between 0.1m and 1.1m above floor from the experimental results. To compare with CFD results, temperature differences are smaller especially at low supply air velocities. The reason is considered that the walls are not completely insulated in the experimental set-up. Temperature differences are larger at the position of f , downstream of the heat source, than at the position of d. The influence of a heat source is observed in the experimental results, although the differences between the temperature of d and f are not seen in the numerical results.

CONCLUSION

This paper has reported how a ceiling height of a small office room influences on thermal stratification for a displacement ventilation system.

Normalised vertical temperature profiles show similarity and linearity for different supply air velocities from the experimental results.

Thermal comfort is preferable for a higher ceiling room for displacement ventilation from the numerical calculations.

A higher ceiling room helps to save energy due to a smaller vertical temperature gradient in a occupied zone at the equal internal load and supply air velocity.

This study also demonstrates how CFD simulations are helpful to examine temperature field for displacement ventilation. CFD analysis can provide parametric research more easily than experimental methods.

Future research will be conducted to apply displacement ventilation to different types of buildings.

Table 4 Temperature differences between 0.1m and 1.1m above floor (CH=2.5m)

(a)H=1.25kW [° C]

	a	b	c	d	f	g	h	i
0.1m/s	4.5	4.0	2.9	4.6	3.6	4.2	3.9	4.2
0.2m/s	8.6	8.6	8.9	9.0	9.2	9.3	8.9	8.3
0.3m/s	1.6	1.9	1.9	1.8	1.5	1.6	2.0	1.7
0.4m/s	0.8	0.7	0.6	0.4	0.6	0.7	0.7	0.5
0.5m/s	0.8	0.7	0.5	0.2	0.2	0.7	0.6	0.5

(b)H=2.5kW [° C]

	a	b	c	d	f	g	h	i
0.1m/s	7.0	6.6	2.8	8.8	6.0	7.5	7.1	7.1
0.2m/s	11	11	11	11	11	11	11	11
0.3m/s	6.4	6.1	5.8	5.6	6.1	6.4	6.7	6.3
0.4m/s	2.3	2.2	2.4	2.0	1.8	2.0	2.4	2.0
0.5m/s	1.8	1.4	1.2	0.9	0.9	1.5	1.5	1.0

Table 5 Temperature differences between 0.1m and 1.1m above floor (CH=3.0m)

(a)H=1.25kW [° C]

	a	b	c	d	f	g	h	i
0.1m/s	0.8	0.6	0.2	0.9	0.9	0.7	0.6	0.8
0.2m/s	1.3	1.3	1.4	1.3	1.3	1.3	1.3	1.1
0.3m/s	0.5	0.4	0.3	0.2	0.7	0.5	0.6	0.7
0.4m/s	0.2	0.2	0.2	0.1	0	0.2	0.2	0.2
0.5m/s	0.2	0.2	0.2	0	0	0.2	0.2	0.2

(b)H=2.5kW [° C]

	a	b	c	d	f	g	h	i
0.1m/s	6.6	6.1	5.3	6.2	5.5	6.0	4.8	5.7
0.2m/s	8.0	7.6	7.7	7.6	7.5	7.6	7.1	7.9
0.3m/s	2.0	2.0	1.8	1.5	1.7	1.8	2.1	2.3
0.4m/s	0.3	0.3	0.3	0.1	0	0.3	0.3	0.3
0.5m/s	0.3	0.3	0.3	0	0	0.2	0.3	0.3

Table 6 Temperature differences between 0.1m and 1.1m above floor (CH=3.5m)

(a)H=1.25kW [° C]

	a	b	c	d	f	g	h	i
0.1m/s	4.9	4.8	4.6	4.6	4.5	4.5	4.3	4.5
0.2m/s	3.0	2.6	2.4	2.8	2.6	3.0	2.5	2.3
0.3m/s	0.3	0.2	0.2	0.2	0.2	0.3	0.3	0.2
0.4m/s	0.2	0.2	0.1	0.1	0	0.2	0.2	0.2
0.5m/s	0.2	0.2	0.2	0.1	0	0.2	0.2	0.2

(b)H=2.5kW [° C]

	a	b	c	d	f	g	h	i
0.1m/s	6.9	6.7	5.9	6.2	6.3	6.7	6.1	6.2
0.2m/s	9.4	8.8	8.9	8.6	8.9	8.6	9.2	8.6
0.3m/s	0.5	0.5	0.4	0.4	0.3	0.4	0.4	0.3
0.4m/s	0.5	0.4	0.4	0.1	0.1	0.4	0.4	0.3
0.5m/s	0.4	0.4	0.3	0	0	0.3	0.4	0.3

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Table 7 Temperature differences between 0.1m and 1.1m above floor by the experiment (CH=2.5m) [° C]

	H=1.25kW		H=2.5kW	
	d	f	d	f
0.1m/s	2.3	2.5	3.5	4.1
0.2m/s	1.8	2.2	2.7	3.2
0.3m/s	1.3	1.5	2.1	2.8
0.4m/s	1.4	1.8	2.5	2.7