

NUMERICAL STUDY ON AIRFLOW IN AN OFFICE ROOM WITH A DISPLACEMENT VENTILATION SYSTEM

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ABSTRACT

This paper investigates airflow in an office room with a displacement ventilation system parametrically using three-dimensional CFD. A numerical study is executed for a typical office room with a dimension of 6m by 6m by 3m height according to a variety of supply air velocities, supply air temperatures and heat source conditions. Internal cooling load is given as 28W/m² to 111W/m². In CASE1, a heat source is installed at the center of the room. In CASE2, four heat sources are located separately in the modeled room. The finite volume method and the low Reynolds number k- ϵ turbulence model are employed to solve the governing equations numerically. As a result, the temperature field in the modeled room for a displacement ventilation system is concluded according to a variety of the above parameters. The stratification profiles in the room depend especially on the supply air velocity. This means VAV systems are applicable to control room air temperature for a displacement ventilation system.

INTRODUCTION

Displacement ventilation has been applied mainly in Scandinavian countries and spreading worldwide. It causes slow upward airflow in a room with bottom wall outlets. Supply air is introduced to the room at a low velocity and at a temperature slightly below the desired room temperature. It exhausts contaminants generated in the room occupied effectively together with heat plumes. The goal of this paper is to validate that a displacement ventilation system achieves not only thermal comfort but energy conservation.

This paper investigates temperature field in a typical office room parametrically for a displacement ventilation system using CFD. Computational fluid dynamics (CFD) technique is widely used to predict airflow in ventilated rooms. Especially for parametric studies, CFD is a useful scheme to save cost and time, compared to experimental methods. Recent computer technology encourages the utilization of CFD due to higher calculating performances. CFD requires iterating calculations and post processing techniques. This study is dedicated to a three dimensional numerical analysis of a displacement ventilation system. Parametric simulations are carried out for a typical office room with a variety of supply air velocities, supply air temperatures and heat source

conditions. As a result, displacement ventilation is concluded more effective than to employ traditional mixing ventilation for an office room.

NUMERICAL METHODS

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. Previous paper employs the standard k- ϵ turbulence model with a standard log law wall function, but this paper applies the low Reynolds number k- ϵ turbulence model with a damping function to reduce the turbulent viscosity near a wall (the AKN Model). The used damping function is described as follows;

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

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

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 1. The modeled room has a dimension of 6m by 6m by 3m height. Air outlets are located at the bottom of the wall and air inlets are at the top of the opposite wall. Supply air velocity and temperature are varied for a variety of total heat source capacity. Internal cooling load is given as 28W/m² to 111W/m². Two cases are prepared for the location of internal heat sources. In CASE1, a heat source is installed at the center of the room. The geometry of the modeled room is shown in Fig.1 for CASE1. In CASE2, four heat sources are located separately in the modeled room. A heat source is a cube with a dimension of

1m by 1 m by 1m. Heat generation is assumed uniform. The number of grid cells is 50x58x32 =92,800 for CASE1 and 60x72x30=129,600 for CASE2 as used in the previous paper. Thus, 90 numerical simulations are made for the modeled room with a displacement ventilation system.

RESULTS AND DISCUSSION

Fig.2 shows the plan of the modeled room. A heat source is installed at the center of the room for CASE1 and 4 heat sources are installed separately in the room for CASE2 as mentioned previously. The numerical results are shown in Fig.3 to Fig.6 for temperature fields at the centerline of the modeled room as typically encountered in displacement ventilation systems. The temperature stratifications are formed generally in the whole space for all cases. This indicates that the driving force of displacement ventilation is buoyancy generated by heat sources. The Archimedes number is the ratio of buoyancy and inertia force expressed by the next formula.

$$Ar = \frac{\Delta\rho g L}{\rho v^2} \quad (3)$$

where

$\Delta\rho$ = density difference between the colder and the warmer air [kg/m^3]

g = gravitational acceleration = 9.8m/s^2

L = a characteristic length [m]

ρ = air density [kg/m^3]

v = air velocity [m/s]

In these cases, Ar is estimated roughly between 0.3 and 30.

Vertical Temperature Profiles

Fig.3 and Fig.4 show typical examples of isothermal vertical sections (x - z sections) at the centerline in CASE1 when a heat source of 1000W and 2000W is given relatively. As the supply air velocity increases, vertical temperature difference decreases. Temperature stratifications are formed clearly except above heat sources. Although heat plumes break temperature stratifications above heat sources, they give no influences to the adjacent zones.

Fig.7 shows vertical temperature profiles at 3 points of the centerline for a variety of supply air velocities. At low supply air velocities, vertical temperature differences in occupied zones are large. As supply air velocities increases, smaller vertical temperature distributions are formed. ISO7730-1994 presents moderate thermal environments and recommends that vertical temperature difference between 0.1m and 1.1m above floor shall be less than 3°C for thermal comfort. For a heat source of 1000W, supply air velocity of 0.2m/s satisfies ISO7730 recommendation. And for 2000W, supply air velocity of 0.3m/s gives the satisfaction. However, temperature stratification is observed more clearly in case of lower air velocity. The temperature stratifications are formed generally

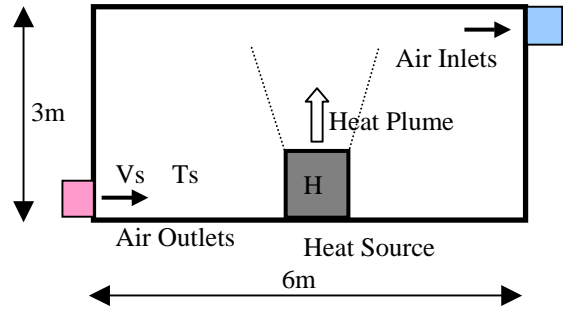


Fig. 0. Geometry of the Modeled Room in CASE1

Table 1. The Conditions for Numerical Calculations

Supply Air Velocity V_s [m/s]	Supply Air Temperature T_s [$^\circ\text{C}$]	Total Heat Source Capacity H [kW]
0.1 (3.3ACH)	18	1
0.2 (6.7ACH)	19	2
0.3 (10ACH)	20	4
0.4 (13.3ACH)		
0.5 (16.7ACH)		

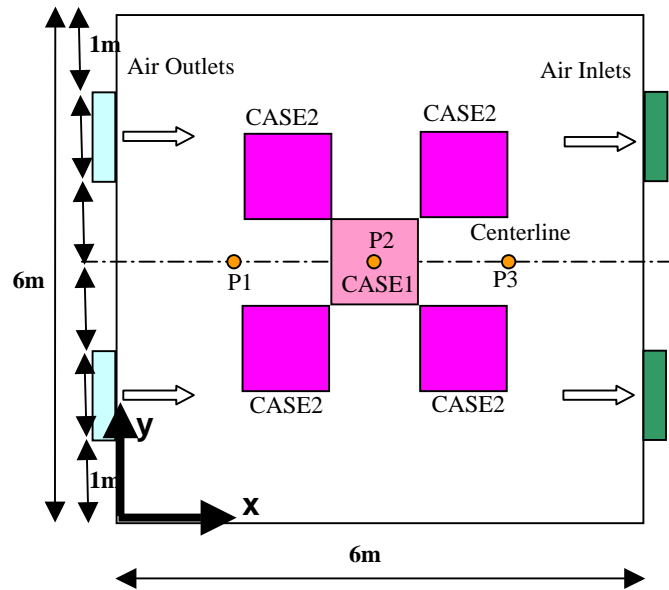
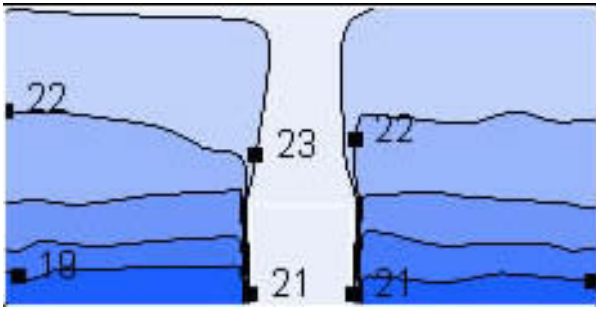
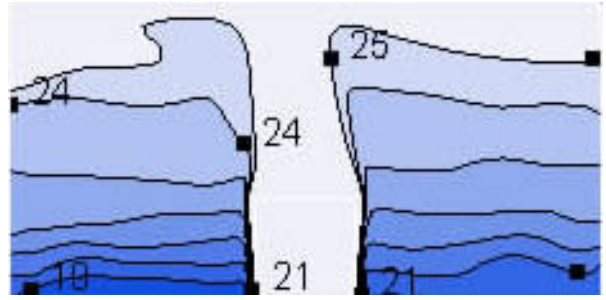


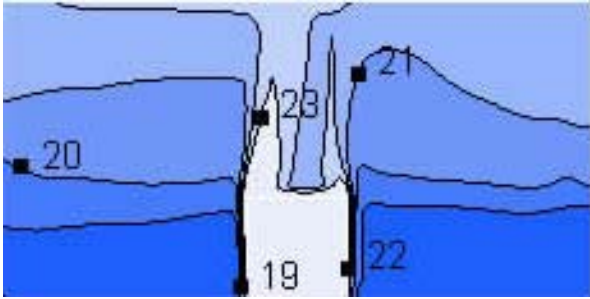
Fig.2 Plan of the Modeled Room in the whole space for all cases. In CASE2, heat plumes affect each other and tend to perturb temperature stratifications at the heat source sections. Fig.5 and Fig.6 show typical examples of isothermal vertical sections in CASE2. For a variety of supply air velocity, temperature field changes in Fig.8. Vertical temperature distributions depend largely on supply air velocity as shown in Fig.8. As supply air velocity increases, it is observed clearly that vertical temperature difference decreases. Fig.8 shows how the vertical air temperature difference can be decreased due to the increase of supply air velocity for displacement ventilation. Thus, thermal comfort is maintained by the regulation of supply air velocity.



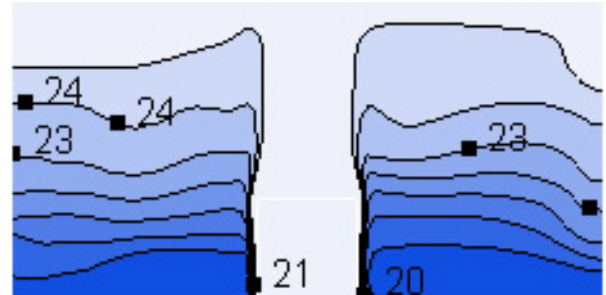
(a) $V_s=0.1\text{m/s}$, $T_s=18^\circ\text{C}$, $H=1000\text{W}$



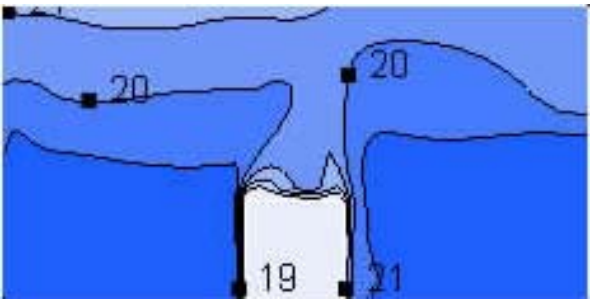
(a) $V_s=0.1\text{m/s}$, $T_s=18^\circ\text{C}$, $H=2000\text{W}$



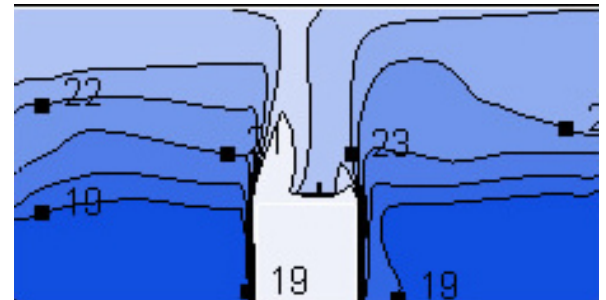
(b) $V_s=0.2\text{m/s}$, $T_s=18^\circ\text{C}$, $H=1000\text{W}$



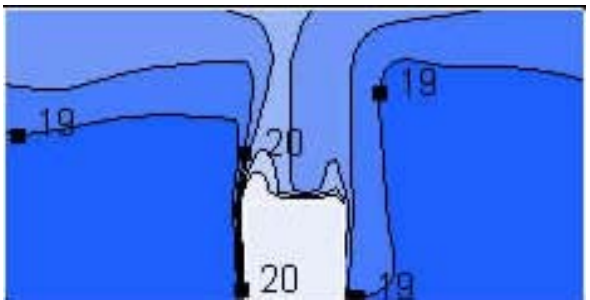
(b) $V_s=0.2\text{m/s}$, $T_s=18^\circ\text{C}$, $H=2000\text{W}$



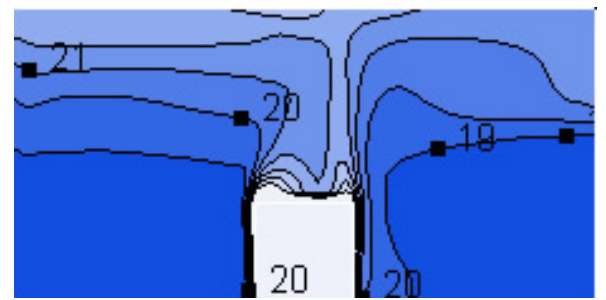
(c) $V_s=0.3\text{m/s}$, $T_s=18^\circ\text{C}$, $H=1000\text{W}$



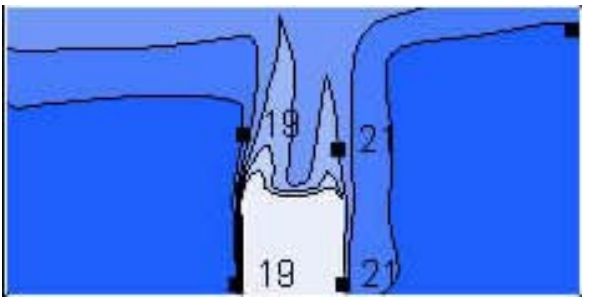
(c) $V_s=0.3\text{m/s}$, $T_s=18^\circ\text{C}$, $H=2000\text{W}$



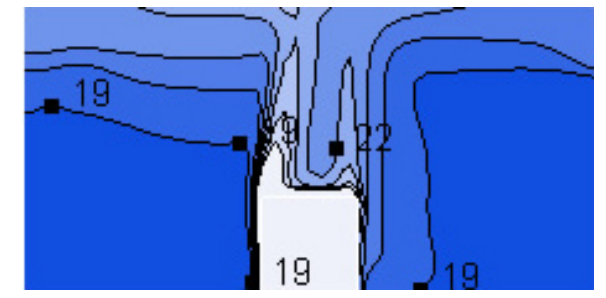
(d) $V_s=0.4\text{m/s}$, $T_s=18^\circ\text{C}$, $H=1000\text{W}$



(d) $V_s=0.4\text{m/s}$, $T_s=18^\circ\text{C}$, $H=2000\text{W}$



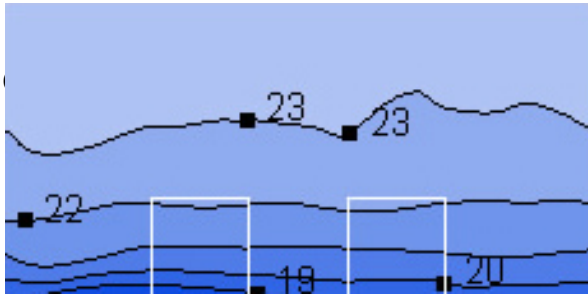
(e) $V_s=0.5\text{m/s}$, $T_s=18^\circ\text{C}$, $H=1000\text{W}$



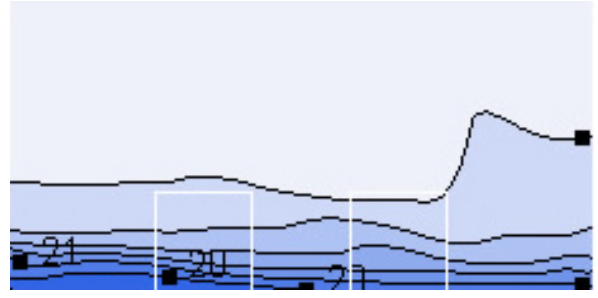
(e) $V_s=0.5\text{m/s}$, $T_s=18^\circ\text{C}$, $H=2000\text{W}$

*Fig.3 Isothermal Vertical Sections
at the Centerline in CASE1
(H=1000W)*

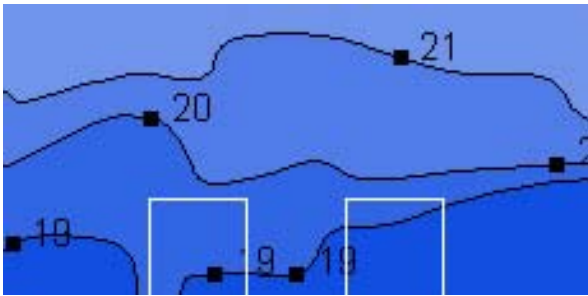
*Fig.4 Isothermal Vertical Sections
at the Centerline in CASE1
(H=2000W)*



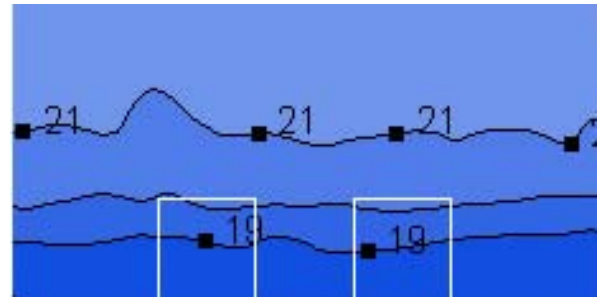
(a) $V_s=0.1\text{m/s}$, $T_s=18^\circ\text{C}$, $H=1000\text{W}$



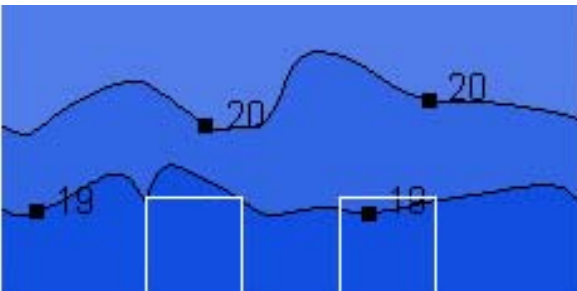
(a) $V_s=0.1\text{m/s}$, $T_s=18^\circ\text{C}$, $H=2000\text{W}$



(b) $V_s=0.2\text{m/s}$, $T_s=18^\circ\text{C}$, $H=1000\text{W}$



(b) $V_s=0.2\text{m/s}$, $T_s=18^\circ\text{C}$, $H=2000\text{W}$



(c) $V_s=0.3\text{m/s}$, $T_s=18^\circ\text{C}$, $H=1000\text{W}$



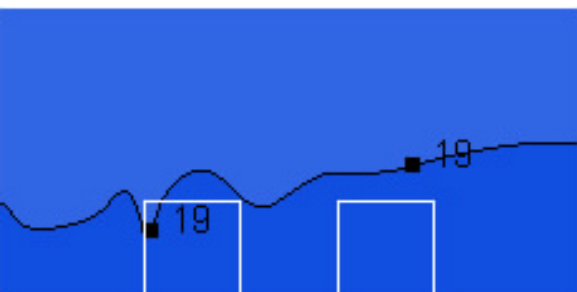
(c) $V_s=0.3\text{m/s}$, $T_s=18^\circ\text{C}$, $H=2000\text{W}$



(d) $V_s=0.4\text{m/s}$, $T_s=18^\circ\text{C}$, $H=1000\text{W}$



(d) $V_s=0.4\text{m/s}$, $T_s=18^\circ\text{C}$, $H=2000\text{W}$



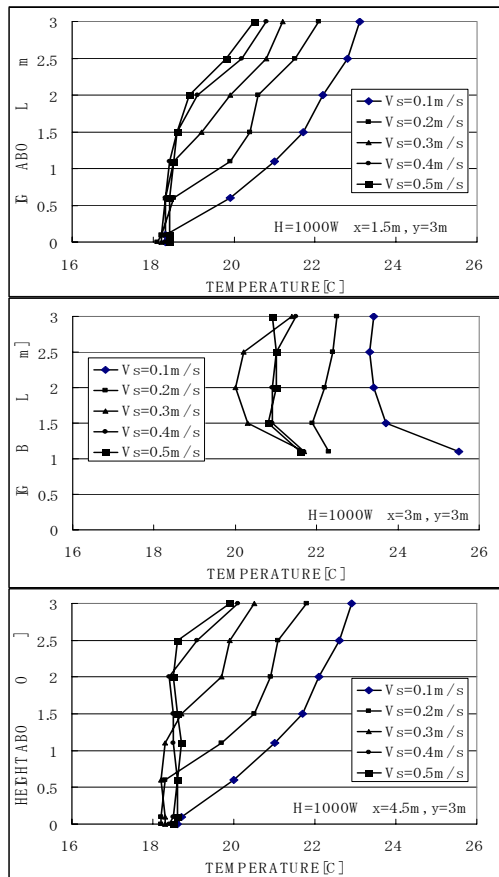
(e) $V_s=0.5\text{m/s}$, $T_s=18^\circ\text{C}$, $H=1000\text{W}$



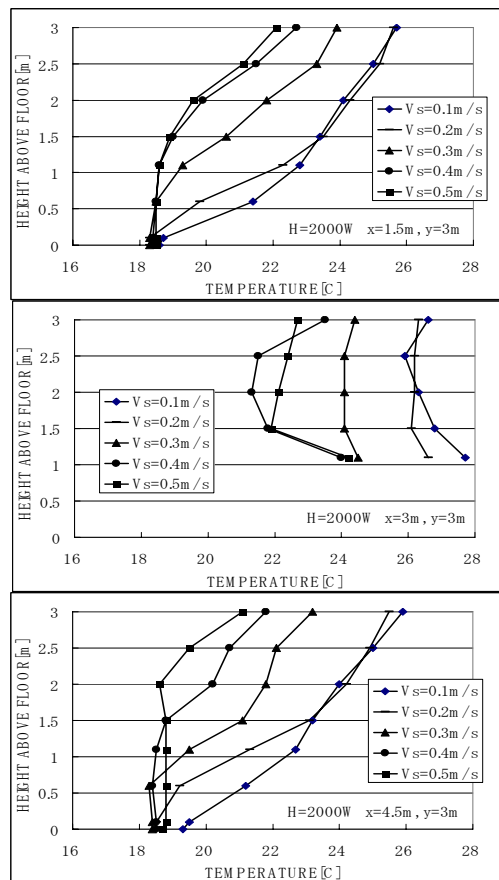
(e) $V_s=0.5\text{m/s}$, $T_s=18^\circ\text{C}$, $H=2000\text{W}$

*Fig.5 Isothermal Vertical Sections
at the Centerline in CASE2
($H=1000\text{W}$)*

*Fig.6 Isothermal Vertical Sections
at the Centerline in CASE2
($H=2000\text{W}$)*

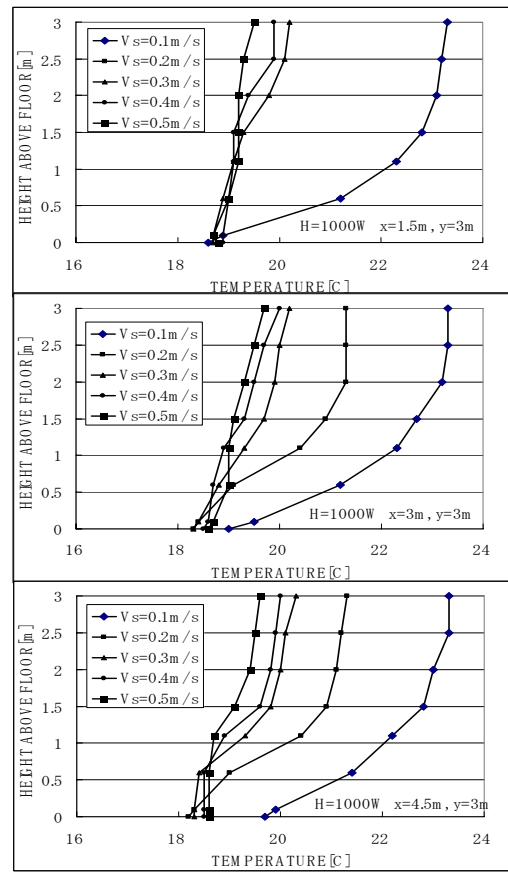


(a) $T_s=18^\circ\text{C}, H=1000\text{W}$

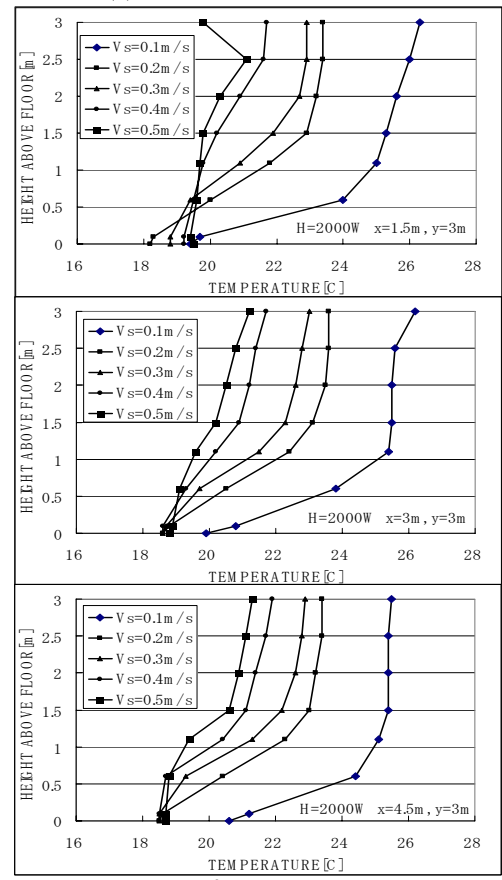


(b) $T_s=18^\circ\text{C}, H=2000\text{W}$

Fig.7 Vertical temperature Profiles at the Center Line of The Room in CASE1



(a) $T_s=18^\circ\text{C}, H=1000\text{W}$



(b) $T_s=18^\circ\text{C}, H=2000\text{W}$

Fig.8 Vertical Temperature Profiles at the Center Line of the Room in CASE2

Thermal Comfort

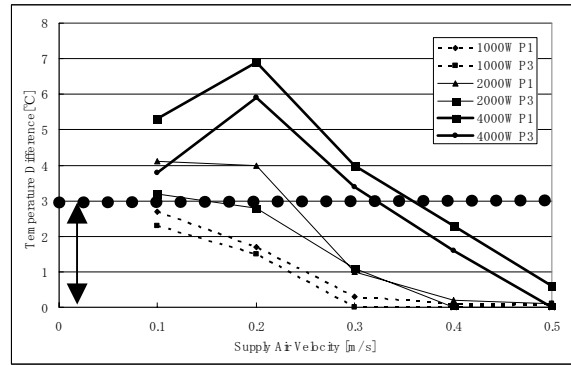
Fig.9 shows the temperature differences between 0.1m and 1.1m above floor for all the cases. In CASE1, supply air velocity of 0.1m/s is enough to satisfy ISO7730 recommendation for a heat source of 1000W, but 0.3m/s is needed for 2000W and 0.4m/s is required for 4000W. In CASE2, supply air velocity of 0.2m/s is enough to satisfy ISO7730 recommendation for 1000W heat sources, but 0.3m/s is needed for 2000W and more than 0.5m/s is required for 4000W. However, this means that the increase of supply air velocity realizes thermal comfort for displacement ventilation, because thermal stratification is formed in all the cases as presented previously. Therefore, it is possible to adapt a VAV system for displacement ventilation to save fan energy without reducing thermal comfort of the occupants.

Influence of Supply Air Temperature

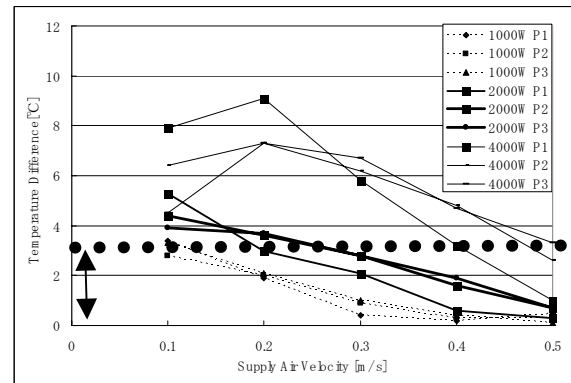
The typical vertical temperature profiles at different supply air temperatures at P1(x=1.5m, y=3m) are shown in Fig.10. The common conditions are given as the supply air velocity of 0.5m/s and total heat source capacity of 2000W for CASE1 and CASE2. The vertical temperature profiles shift approximately in parallel according to the change of supply air temperature. The similar tendency is observed at the other points. This means that room air temperature at any height is controlled according to the regulation of supply air temperature when the setpoint room air temperature changes. Therefore, the room air temperature may be successfully controlled when constant air volume system is employed for a displacement ventilation system.

Discussion about the Turbulence Models

Fig.11 shows typical examples in comparison with two different turbulence models. In CASE1 and CASE2 at any point, vertical temperature profiles are generally estimated lower by the low Reynolds number $k-\epsilon$ model than by the standard $k-\epsilon$ model. Similar tendency can be observed in any cases. The differences between the results achieved by two turbulence models are not estimated large. All the rough discussions can be concluded similar by both results. If discussions are carried out for thermal comfort inside rooms, there are no significant difference between the results by two turbulent models. Therefore, it is allowable to employ the standard $k-\epsilon$ model for these cases shown in the previous paper.

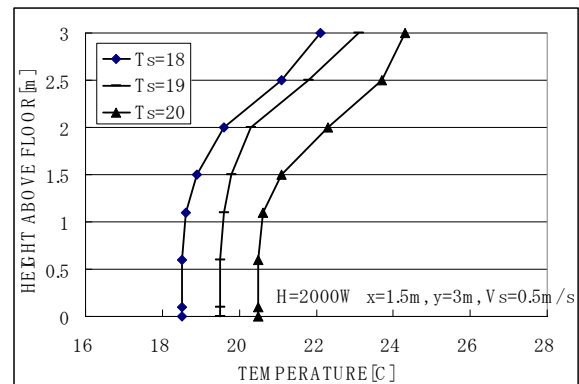


(a) CASE1

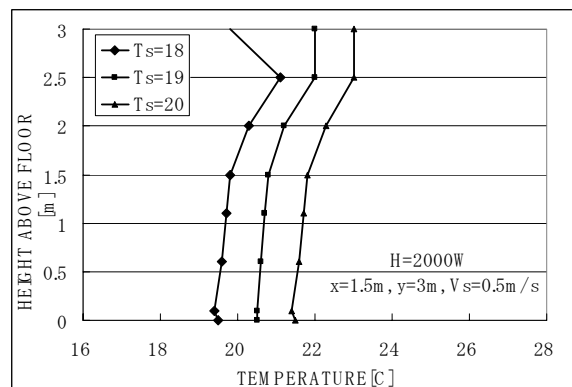


(b) CASE2

Fig.9 Temperature Differences between 0.1m and 1.1m above Floor at $T_s=18^\circ\text{C}$



(a) CASE1



(b) CASE2

Fig.10 Vertical Temperature Profiles at Different Supply Air Temperatures $V_s=0.5\text{m/s}$, $H=2000\text{W}$

CONCLUSION

The present paper studies on temperature field in a modeled office room for a displacement ventilation system using three-dimensional CFD. The low Reynolds number $k-\epsilon$ turbulence model is employed for the turbulent viscosity. As a result, thermal comfort in a typical office room is maintained by regulating supply air velocity as well as temperature. This results in the possibility of a VAV system for displacement ventilation. Thus, a displacement ventilation system is expected to achieve not only thermal comfort in a served room but energy conservation of fan power.

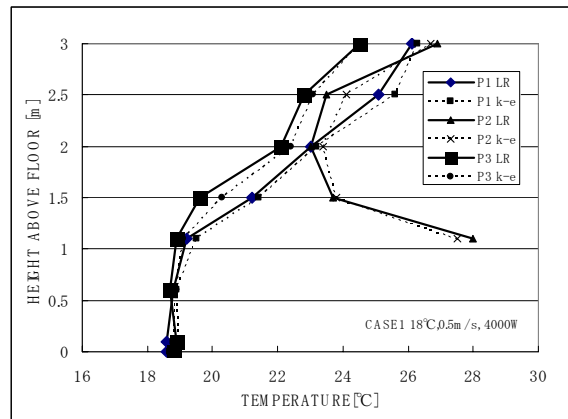
This study is dedicated to three-dimensional CFD. Future work should be followed by experimental results to validate the numerical discussions.

ACKNOWLEDGMENT

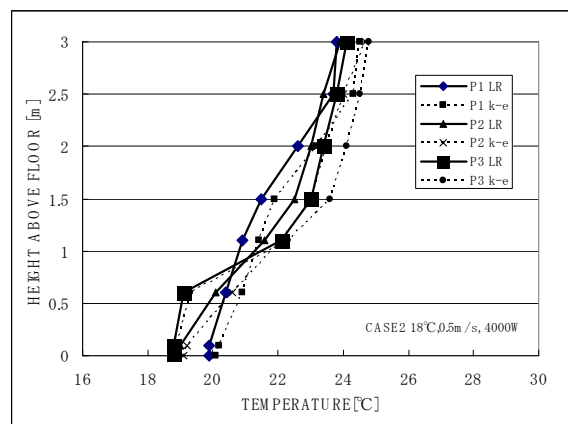
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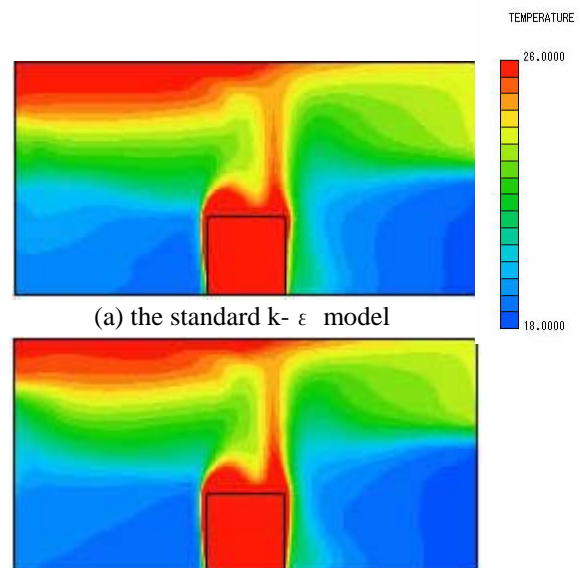


(a) CASE1



(b) CASE2

Fig.11 Comparison of Vertical Temperature Profiles between the results of the standard $k-\epsilon$ and the low Re number $k-\epsilon$ model



(b) the low Re number $k-\epsilon$ model

Fig.12 Typical Examples of Vertical Temperature Contours at $T_s=18^\circ\text{C}$, $V_s=0.5\text{m/s}$, $H=4000\text{W}$ in CASE1

