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Energy Saving Characteristics of Hybrid Displacement Ventilation System with Vertical Supply Duct

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Abstract **Original Research Article**

The study on saving energy is being conducted widely in the ventilation system of industrial building with high-ceiling.

The mixing ventilation system (MVS) is less effective for the high-ceiling buildings from the view of energy saving. The displacement ventilation system (DVS) has a shortcoming when process equipment with high thermal load is utilized in industrial building.

In this paper, the characters of indoor airflow and temperature distribution in hybrid displacement ventilation system (HDVS) where the vertical supply duct has been installed on the middle height of the building have been analyzed by computational fluid dynamics (CFD). And the specific influence of the velocity of air inflow and the length of the supply duct on the air temperature has been analyzed, and the existence of the effective length of vertical supply duct in this system has been discovered from the view of energy saving.

The accuracy of simulated results has been validated comparing with the experimental data. The operation of this system is more cost effective as the temperature of the air in the working zone has decreased 3.2-5.4℃ compared to the MVS.

Keywords: ventilation, energy saving, industrial conditioning, HDVS, vertical supply duct, effective.

1. Introduction

A lot of energy is consumed for the buildings around the world, and therefore the study on energy saving in this field is being conducted widely [1]. Mixing ventilation system (MVS) is one of the mechanical ventilation systems that makes the entire indoor air mixed completely using the jet flow made from blast air. Several Mathematical modeling and experimental researches have been conducted to improve the characteristics of airflow and temperature distribution of MVS [2, 3]. Since the indoor airflow is ensured using mainly the mechanical ventilation force while the aim is to make the air quality of the entire space uniform in MVS, the temperature, humidity and concentration of the air at every indoor place is the same and therefore the whole environment is uniform. The use of MVS for the air conditioning for the only small volume of the floor occupied by the people is not cost effective as the entire volume is unnecessarily cooled [4]. Though the introduction of MVS to the subject with heavy cooling load was difficult, its experimental and designing methods and the mathematical analyzing method have been suggested. In particular, MVS is unfavorable for the high-ceiling buildings and the process equipment with high thermal load from the view of energy saving [4].

The displacement ventilation system (DVS) is one of the mechanical ventilation systems which blow the air with lower temperature than indoor air from the lower part of the room and cause upward moving of indoor air by buoyant force to exhaust it at the ceiling. DVS makes flow of indoor air mainly by the buoyant force and MVS is using mechanical ventilation force, it is the basic difference[5, 6]. As DVS is aimed at meeting only the requirements of the workplace the thermal and concentration stratification are formed between lower and upper part[7, 8]. DVS makes quality of the air better and exausts the polluted air effectively and therefore the saving efficiency of energy is remarkable[9]. The height of thermodynamic stratification has to be controlled to be higher than the height of the workplace the people in the designing of DVS, but it should not be too high[10]. If not, there is increment in the amount of air blast while waste is caused. So the determination of the height of thermodynamic stratification is one of the vital problems in the application of DVS. In DVS, ventilation volume for keeping the temperature below the required level is too much when the cooling load is heavy. For DVS, the experiments, experiences and engineering design methods have already been presented and the mathematical analysis methods have been studied. However, there is little data about DVS for high-ceiling rooms.

Heating energy consumptions were compared when impinging jet ventilation (IJV) system and MVS, DVS are used in ventilation of large-height spaces[11].

Retrofit hybrid displacement ventilation system (RHDVS) with the duct getting down from the ceiling at the middle level of the room instead of the duct fixed on the wall has been used[4].

According to the references, the character of indoor air flow has been simulated under the complicated turbulence flow and the convection heat transfer using continuity, momentum and energy equation together with the turbulent kinetic energy and dissipation transport equations for buoyant flow by CFD [5, 12]. But there are a few data of simulation and experiment for the characteristic of temperature change of work zone air according to the length of supply duct in the hybrid displacement ventilation system.

The change of the temperature at the workplace where the people occupy when there is heat source in MVS and HDVS with the vertical supply duct is reviewed using CFD and its effectiveness is presented in this paper. And the existence of the effective length of vertical supply duct in HDVS with the supply duct is verified from the view of energy saving. Then it is compared with MVS in the industrial condition technologically and economically and the application character is analyzed. The results are useful for ventilation system design and thermal comfort study in high-ceiling building.

2. Methodology

In this paper, the study on the characteristics of indoor air flow and temperature distribution depending on the length of the supply duct rather than the air temperature and air flow change phenomena depending on the supply and exhaust position and the output of the heat source has been focused.

The allocation of supply and exhaust is to be constrained as the working zone of the people is defined under the condition that the structure of the building is available and the heat source is distributed in the room. However, suitable temperature and velocity can be provided by controlling the air velocity and effusion characteristics at exhaust, according to the characteristics of the heat source and space arrangement *i.e.* HDVS is effective in improving energy-effectiveness of the ventilation in the high-ceiling rooms. So it is very important to ensure accuracy in the analysis of the characteristics of the air-flow distribution and temperature change in the working zone considering the low temperature radiation between the heat source and the walls of buildings and to evaluate the effectiveness of supply duct accurately as well in the designing the system. Being functional enough for the air and fluid flow analysis in the process of indoor ventilation, CFD can be used without limit for the simulation of the heat flow depending on the change in the length of supply duct, velocity at drain and heat source.

2.1 Basis of the modeling method

The indoor geometrical size of the object is 9m in width (B) and $6m$ in the height (H).

The heat source is rectangular in its shape with the height h of 1.5m and a width b of 3m and it is symmetrically located centering with x=6m in the z direction. The position of the air supply is as $(x, y, z) = (1.5, 6, 0)$, and the position of the exhaust is as $(x, y, z) = (7.5, 6, 4)$. The position of the center of heat source is as $x = 6m$ (Fig 1.).

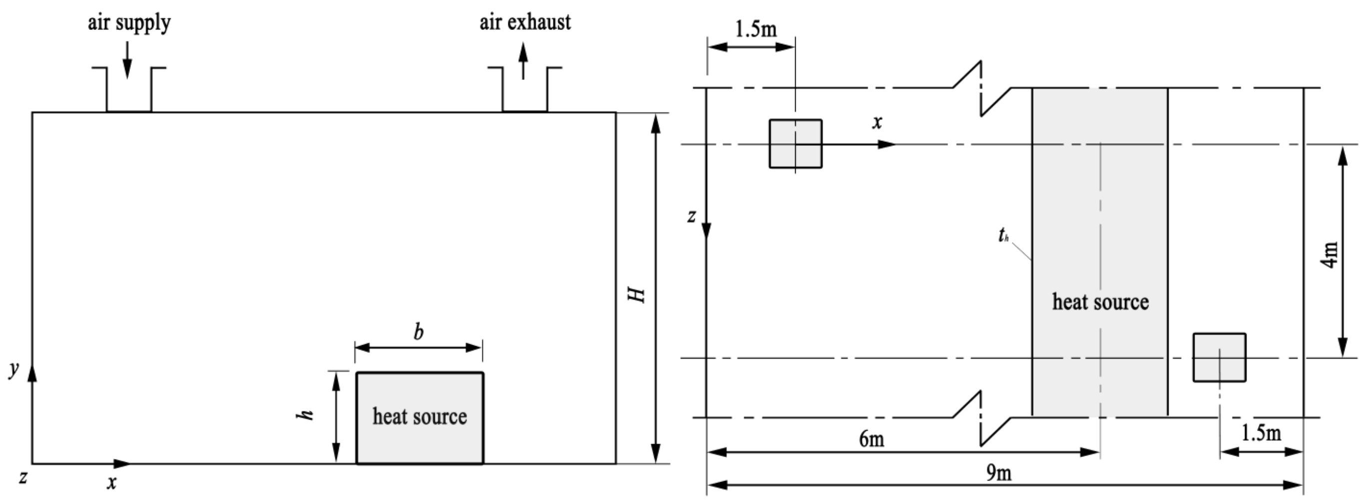

Fig. 1. Schematics of simulated room and the determination of coordinate
a) side view *b*) plan view

For the provision of some positive pressure in the room, a little much amount of air inflow is provided. The inflowing air is supplied at constant temperature through the process of mixing, filtering, preheating and cooling of air recycled in the air conditioning device and outdoor air. The supply and exhaust duct are fixed and their sections are rectangular shape (400×400mm).

The characteristics of air flow distribution and temperature change between the centers of supply duct and exhaust duct in the direction of length of heat source.

2.2 Mathematical modeling

For the mathematical modeling of the indoor ventilation process where there's heat source following hypothesis has been set out. The flow of medium is turbulence flow and T_h, the temperature of heat source is constant. And no loss of air through walls, floor and ceiling is available. To make the review brief only the air and heat flow between the centers of supply and exhaust duct in the direction of axis z are considered and the one for the entire region is considered as its symmetrical issue. The air temperature in the working zone is expressed as T_{av}. The air temperature in the working zone means the average temperature at the level of a height of 2m from the floor as the height occupied by the people is within a height of 2m. The density change is considered as the flow of air in the room is non-isothermal flow of incompressible fluid. T_h, the temperature of heat source ranges 50-80°C and T_c, the temperature of cooling air in supply duct is set out as 20°C. The time averaged continuity, momentum and energy equations are written as

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0 \qquad (1)$$

$$\frac{\partial(\rho uu)}{\partial x}+\frac{\partial(\rho uv)}{\partial y}+\frac{\partial(\rho uw)}{\partial z}= \\ -\frac{\partial P}{\partial x}+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial u}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial u}{\partial z}\right)+S_u \qquad (2)$$

$$\frac{\partial(\rho vu)}{\partial x}+\frac{\partial(\rho vv)}{\partial y}+\frac{\partial(\rho vw)}{\partial z}= \\ -\frac{\partial P}{\partial y}+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial v}{\partial z}\right)+S_v \qquad (3)$$

$$\frac{\partial(\rho wu)}{\partial x}+\frac{\partial(\rho wv)}{\partial y}+\frac{\partial(\rho ww)}{\partial z}= \\ -\frac{\partial P}{\partial z}+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial w}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial z}\right)+S_w \qquad (4)$$

$$\left.\begin{aligned} S_u &= \frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial x}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial x}\right) \\ S_v &= \frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial y}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial y}\right)+g\beta\Delta T \\ S_w &= \frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial z}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial z}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial z}\right) \end{aligned}\right\} \qquad (5)$$

where $\Delta T = T - T_c$ is difference between temperatures of indoor air and cooling air(°C).

$$\mu_{eff}=\mu+\mu_t$$

$$\mu_t=\rho C_\mu\left(\frac{k^2}{\varepsilon}\right)$$

$$\frac{\partial(C_P\rho uT)}{\partial x}+\frac{\partial(C_P\rho vT)}{\partial y}+\frac{\partial(C_P\rho wT)}{\partial z}= \\ \frac{\partial}{\partial x}\left[(\lambda+\lambda_t)\frac{\partial T}{\partial x}\right]+\frac{\partial}{\partial y}\left[(\lambda+\lambda_t)\frac{\partial T}{\partial y}\right]+\frac{\partial}{\partial z}\left[(\lambda+\lambda_t)\frac{\partial T}{\partial z}\right] \qquad (6)$$

$$\lambda_t=C_P(\mu_t/\sigma_T)$$

Medium flow is turbulence, so $k-\varepsilon$ equation of standard is used.

$$\frac{\partial(\rho uk)}{\partial x}+\frac{\partial(\rho vk)}{\partial y}+\frac{\partial(\rho wk)}{\partial z}=\frac{\partial}{\partial x}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x}\right]+ \\ \frac{\partial}{\partial y}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial y}\right]+\frac{\partial}{\partial z}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial z}\right]+\mu_t S-\rho^2\sigma_\mu\frac{k^2}{\mu_t} \qquad (7)$$

$$\frac{\partial(\rho u\varepsilon)}{\partial x}+\frac{\partial(\rho v\varepsilon)}{\partial y}+\frac{\partial(\rho w\varepsilon)}{\partial z}=\frac{\partial}{\partial x}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x}\right]+$$
$$\frac{\partial}{\partial y}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial y}\right]+\frac{\partial}{\partial z}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial z}\right]+C_1\rho\sigma_\mu kS-C_2\rho\frac{\varepsilon^2}{k^2} \tag{8}$$
$$S=2\left[\left(\frac{\partial u}{\partial x}\right)^2+\left(\frac{\partial v}{\partial y}\right)^2+\left(\frac{\partial w}{\partial z}\right)^2\right]+\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2+\left(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y}\right)^2+\left(\frac{\partial w}{\partial x}+\frac{\partial u}{\partial z}\right)^2$$

The empirical constants σ_k, σ_μ, σ_ε, σ_T, C_1, C_2 and C_μ are $\sigma_k=1.0$, $\sigma_\mu=1.0$, $\sigma_\varepsilon=1.4$, $\sigma_T=0.9$, $C_1=1.44$, $C_2=1.92$ and $C_\mu=0.09$, $P=101\ 325$ Pa.

Initial and boundary conditions are set out. Indoor air temperature at the beginning, *i.e.* τ =0, the first moment of air inflow is set as follow.

$$T(x,y,z,0)=T_0 \tag{9}$$

The boundary condition of air inlet is set as follow.

$$\begin{cases} v=v_0 \\ k=1.5I^2 \\ \varepsilon=\dfrac{k^{\frac{3}{2}}C_P^{\frac{3}{4}}}{l_t}v \\ T=T_0 \end{cases} \tag{10}$$

where I is turbulent intensity, $I=0.16\times\mathrm{Re}^{-\frac{1}{8}}$; l is distance of turbulent flow.

The boundary condition of air outlet is set as follow.

$$\begin{cases} \dfrac{\partial T}{\partial n}=0 \\ \dfrac{\partial k}{\partial n}=0 \\ \dfrac{\partial\varepsilon}{\partial n}=0 \end{cases} \tag{11}$$

where n is length vertical to isothermal-surface.

The temperatures of the wall, ceiling and the floor can be considered as constant (T_{wi}=20°C), since there's almost no change in indoor temperature of wall. Here, suffix i=1-3 represents the wall, ceiling and floor. Effective radiation of wall is as following.

$$q_{eff}=(1-\varepsilon_{wi})q_{in}+\varepsilon_{wi}\sigma(T_{wi}^4-T^4) \tag{12}$$

Intensity of radiation arriving at wall(q_{in}) is as following.

$$q_{in}=\int_{\vec{s}\cdot\vec{n}>0} I_{in}\vec{s}\cdot\vec{n}d\Omega$$

where T_{wi}, ε_{wi} is temperature and blackness of *i*-th wall, Ω is solid angle, $\vec{s}$ is ray vector, $\vec{n}$ is normal line. I_{in} is intensity of entering radiation,

$$I_{in} = \frac{\varepsilon_{in-s}\sigma(T_{ws}^4 - T^4)}{\pi}$$

where T_{ws}, ε_{in-s} is temperature and blackness of radiated wall.

A network with 500 thousands of hexahedron elements was prepared to perform simulation analysis using CFD. The air-blast has been assumed as velocity inlet, air-exhaust as pressure outlet and both sides of surface as symmetry condition while the wall and heat source have been set as wall boundary in the course of preparing the network. The floor surface has been considered as wall surface.

3. Numerical experiments and comparative analysis

The indoor air temperature distribution with the condition of H = 6m, $v_0 = 4$ m/s, $h = 1.5$ m, $T_h = 60$ °C, $b = 3$ m based on the above mathematical modeling is shown in Fig.2. where v_0 is the velocity of supply air in supply duct. As shown in Fig. 2, the temperature of the working zone in front of heat source is higher than 27°C.

Pilot measurement has been performed in the room conditioned with height H =6m, height of heat source of 1.5m, width of 3m, v_0=4m/s to verify the correctness of the model. The average temperature of the front side of the heat source is 60°C. As shown in Fig.3 three measuring objects have been installed at 1m (in the direction of z axis) intervals on 2m high level for the measurement to be conducted.

Considering that temperature of back side and upside of heat source didn't affect the change of temperature of working zone, its measurement hasn't been performed. Three times of measurement has been performed until the temperature of front side of heat source has been confirmed to get to the constant value and not to change any more. The result of the measured temperature is shown in Fig 4.

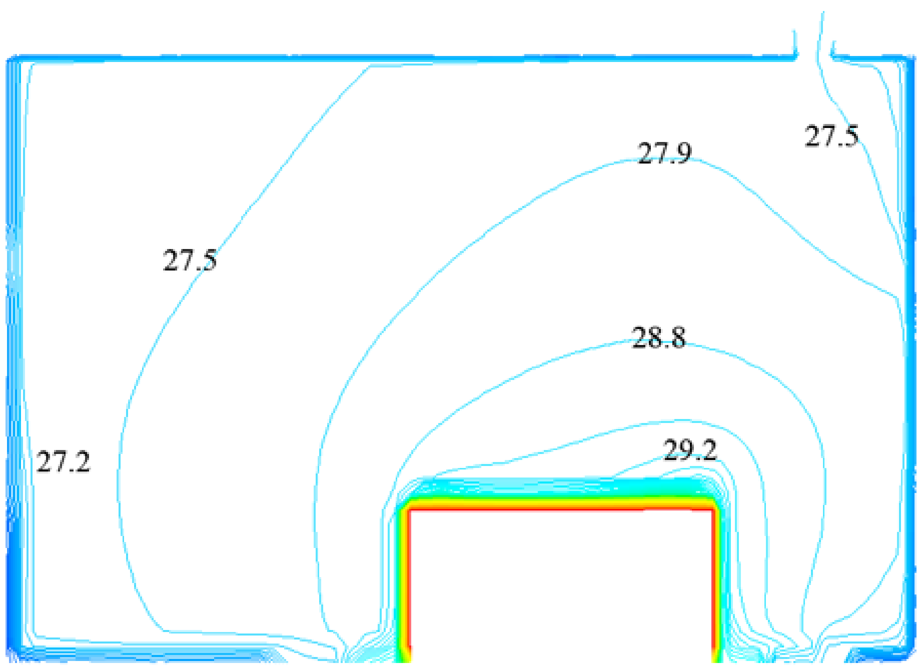

Fig. 2. Isothermal line at the position of $x = 4$ m

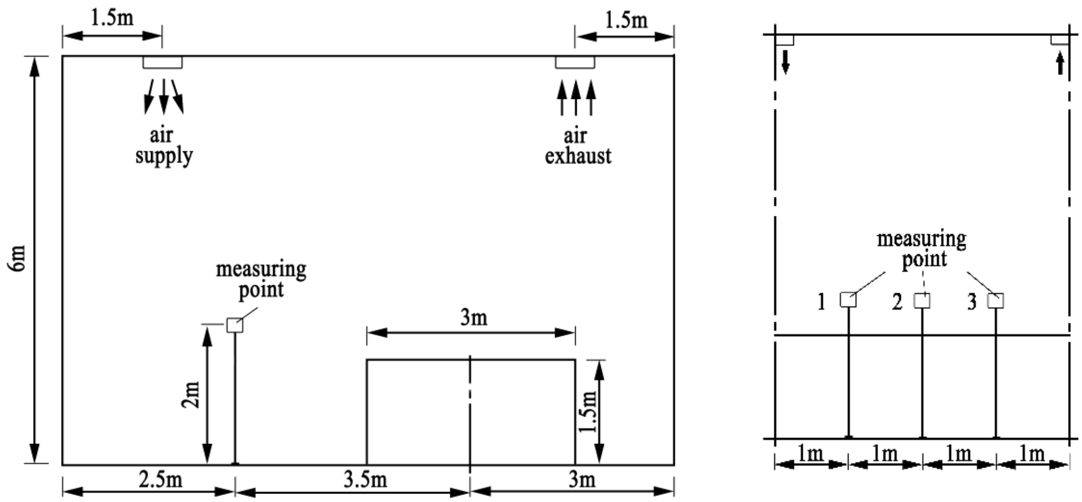

Fig. 3. Schematics of measurement point.
a) side view *b*) front view

As shown in Fig 4, the temperature of indoor air rises a little getting to the exhaust from supply position of inflow air. The relative error of the measured between the calculated values ranges within 1.2 percent. This difference is due to the assumption that temperature of the front surface of the heat source is constant and there is a small difference in the air flow between the model and the actual state. The Fig. 5 is shown the linear chart of the change of T_{av} / T_s depending on v_0 and T_h in the room where its height is 6 m. The standard temperature of working zone T_s has been set as 24°C.

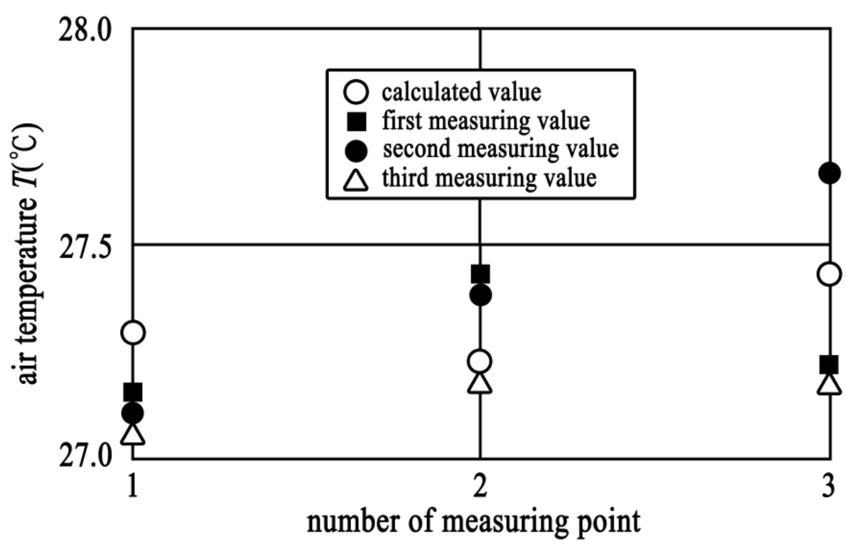

Fig. 4. Comparison of analyzed value with measured one.

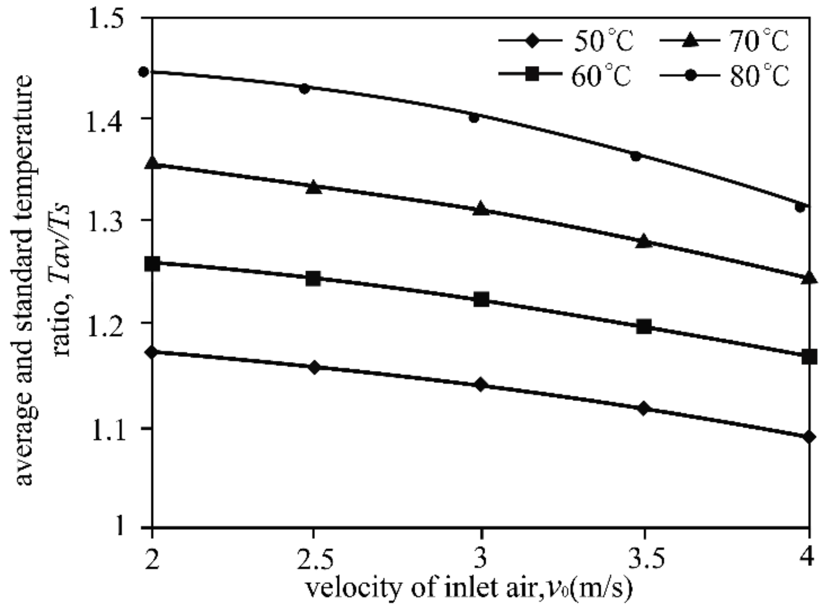

Fig. 5. Change of T_{av} / T_s depending on v_0 (T_c=20°C, T_h=50-80°C)

As shown in Fig 5, T_{av} gets decreased gradually as the velocity of air inflow or the amount of air inflow gets increased and $T_{av} / T_s < 0$ when the velocity of inlet air reaches 4m/s. And T_{av} / T_s is higher as the temperature on the surface of heat source increases while its value is lower as v_0 increases. This is because indoor air flow is affected by the equipment installed with a certain degree of temperature and height in the room. And it is explained that it happens with the influence between flow of supply air towards working zone and air flow caused by buoyancy generated by the temperature of heat source.

4. Simulation of the HDVS with vertical supply duct

The simulation of the HDVS described above in this paper was performed by mathematical modeling. Fig. 6 shows calculation model.

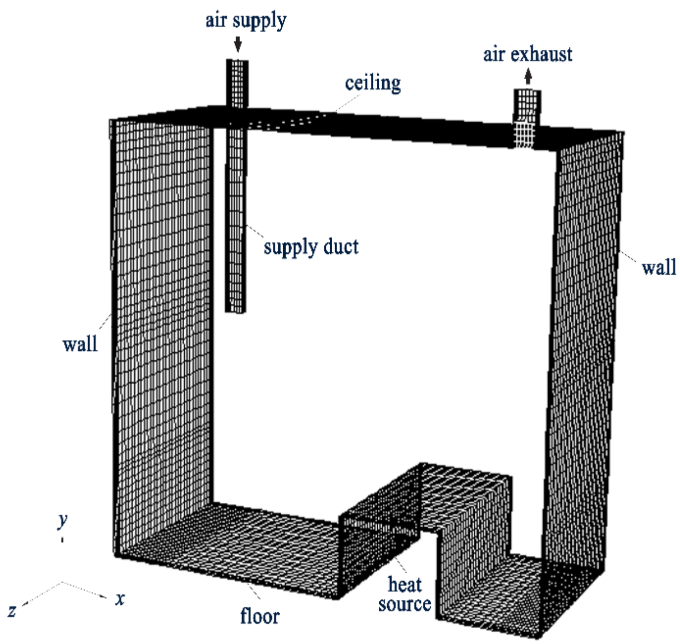

Fig. 6. Simulated room of HDVS with supply duct.

Fig. 7 shows the change of T_{av} in the case of H =6m, h =1.75m.

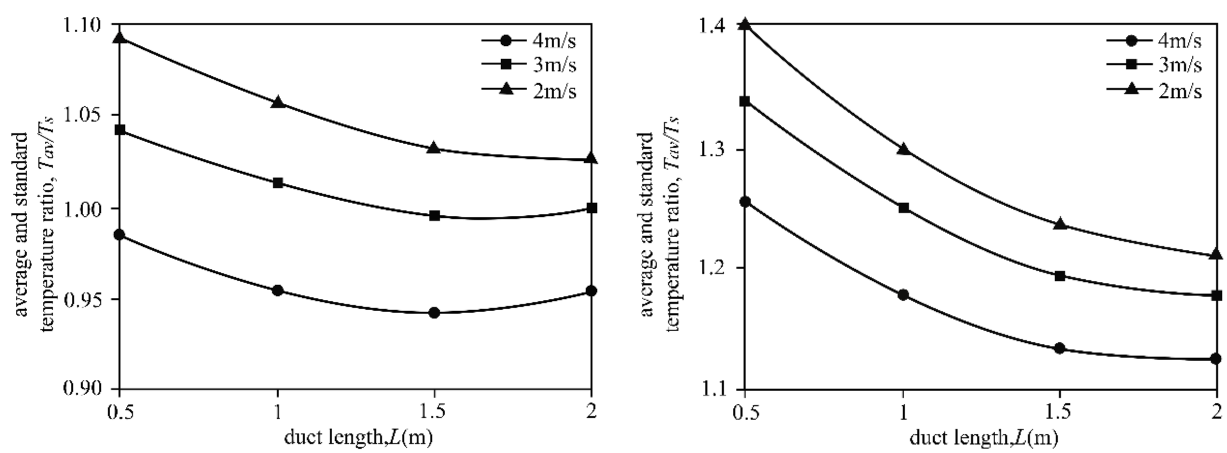

Fig. 7. Change of T_{av} depending on L ($H = 6$ m, $h = 1.75$ m)

a) T_h =50℃ *b*) T_h =80℃

As shown in Fig 7, $T_{av} \le T_s$ is ensured when v_0 =3-4m/s by controlling L in the condition of T_h = 50℃. Here, L means the length of supply duct which is placed between the ceiling and the supply position. Finally, energy consumed for ventilation can be saved a lot by installing the supply duct. As shown in Fig. 7, T_s can be ensured without increasing the ventilation volume if the supply duct is available.

It is shown in Fig 7*a* that the curve of temperature change has concave shape in the condition of v_0=2~4 m/s and thus it has the extreme point. L, the length of supply duct with the extreme point gets long as v_0 gets small and T_h is high under the condition that H is given. When T_h is higher than 60°C, the temperature of the working zone is not changed any more from L, a certain threshold even though there's increase in the length of supply duct while the characteristic is presented that there' s no extreme point formed even though the length of supply duct is increased up to 2 m. As shown in the Fig 7b, no extreme point is formed even though the length of supply duct is increaed at v_0= 2~4m/s and T_h =80°C, and T_{av} can't reached T_s. Therefore, in this case the temperature of supply air has to be lower accordingly .

Fig.8 shows the change of T_{av} / T_s depending on L under the condition of $H = 8$ m, $B = 9$ m, h=1.75m, b =2m and T_h =50°C. Fig. 8 shows that T_{av} , temperature of working zone gets lower gradually as L, the length of supply duct and v_0, the velocity of supply air are increased.

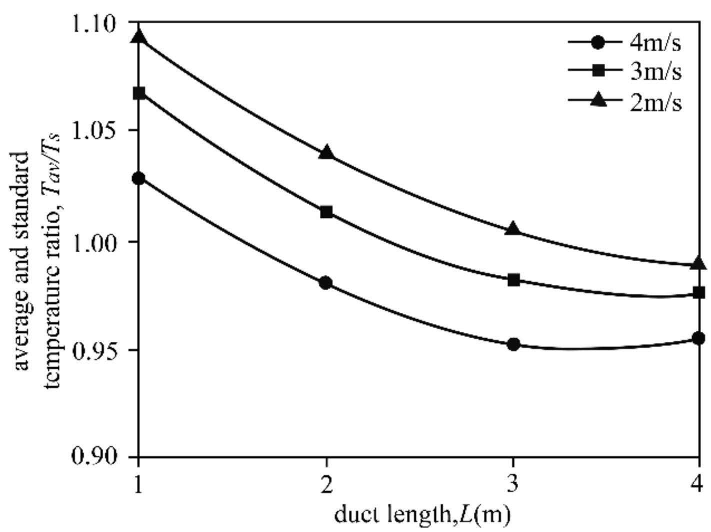

Fig. 8. Change of T_{av} depending on L (T_h=50°C).

As shown in Fig. 8, $T_{av} \le T_s$ can be secured with the condition of v_0 =2~4m/s when T_h is 50°C adjusting L. It means T_s can be secured with no increasing the ventilation volume when the supply duct is used. It is shown that the curve of temperature change has concave shape in the condition of v_0=2-4 m/s and thus it has the extreme point. L, the length of supply duct where the extreme point is formed gets long as v_0 gets decreased. The extreme point is not formed and T_s can't be secured as a result under the condition of v_0<3m/s even though the length of supply duct is increased, and therefore, the temperature of air inflow(T_c) should be lowered accordingly in this case. As shown in Figures, T_{av}, temperature of working zone gets lower gradually as L, the length of supply duct and v_0, the velocity of supply air are increased.

Fig. 9 and Fig.10 present the isothermal line and equal velocity line at the section of exhaust under the condition of H=8m, B=9m, h=1.75m, b=2m, T_c =20°C, L=4m and T_h=60°C, v_0=4m/s.

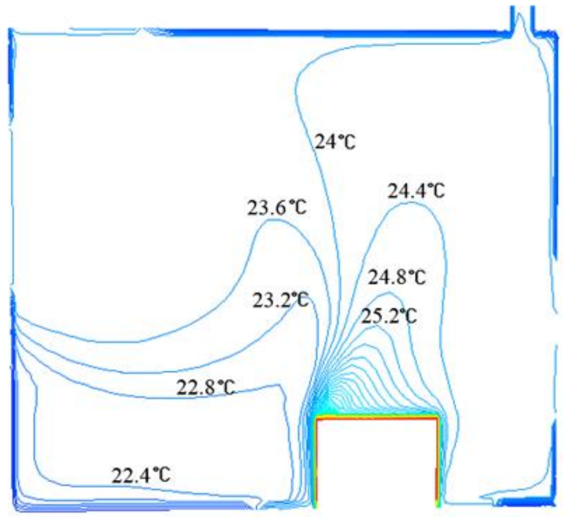

Fig. 9. . Isothermal line at z =4m.

5. System design and discussion

The case study of introduction of model suggested in this paper to the working building has been presented herewith. The height of new building is 8m, compared to old height of 6m, to establish new production process with new equipments. Since the building structure has already been determined and the equipment of heat source has already been placed, the position of the supply and exhaust can't be changed, whereas the temperature of surface of the heat source is given.

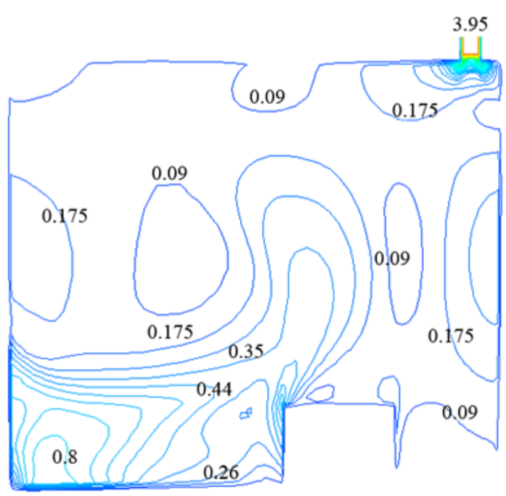

Fig. 10. Equal velocity line at z =4m

It has been determined to opt the hybrid displacement ventilation system and the designing considered the ventilation volume and the length of supply duct contributed to the minimized energy consumption has been conducted.

5.1 The character of ventilation object and the system designing based on that.

The object of ventilation is a workshop with a height of 8m, width of 9m and the length of 92m. 12 supply ports and 11 exhaust ports have been installed.

The supply and exhaust ducts are fixed and their sections are of rectangular shape (400×400mm). It is impossible to lower the building height since there's an elevator structured with a height of 7 m in the front and middle of the workplace. Heat source has got conditions such as h=1.75m and $b=2$m and it is placed symmetrically with x=6m as a center in the direction of axis z. The heat source is placed in the direction of axis z as it is the linking process with other equipment while the surface temperature is different in compliance with the intervals ranging 50 ~ 80°C. The focal position of the heat source is as x =6m. The position of supply duct is as $(x,y,z)=(1.5,8,0)$ and the position of the exhaust duct is as $(x,y,z)=(7.5,8,4)$ on the ceiling. The temperature of supply air in supply duct(T_c) is 20°C.

5.2 The effectiveness analysis of supply duct

As described above, T_{av} is rather increased than decreased when the supply duct is lengthened longer than critical value under the given condition, namely, L_{eff}, the effective length of supply duct exists. Thus, the effective length of supply duct L_{eff} exists.

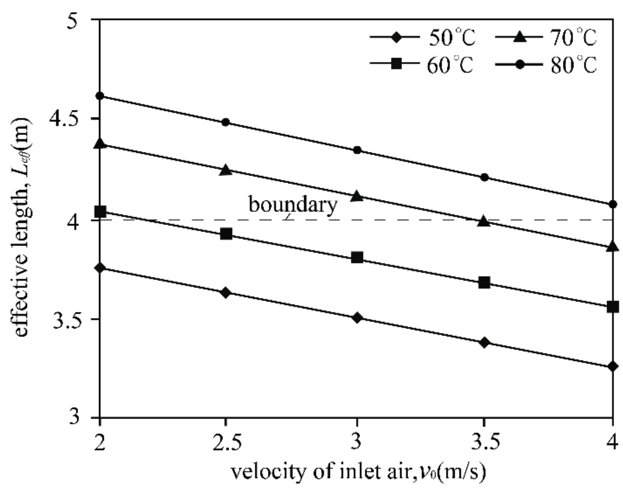

Fig. 11. Change of L_{eff} depending on v_0 and T_h

Fig. 11 shows the change of L_{eff} depending on v_0, T_h at H=8m. As shown in Fig. 11, L_{eff}, the length of air supply duct gets longer as T_h gets long and v_0 gets bigger *i.e,* the effective length of the duct decreases linearly as v_0 gets increased and the change of L_{eff} compared to the change range of T_h is almost the same. As shown in Fig.11, L_{eff} exceeds length limit under the condition of $T_h > 70$ °C. The allowable length was set out as 4m because the height of building was 8m and it didn't cause any inconvenience to the people's activities. The distance from the bottom to the exit of supply duct should be 4m at least.

Fig. 12 shows ΔT_{av}, the difference in air temperature in the working zone when the duct was installed in as L=0 and L=L_{eff} under the condition of v_0=4m/s, T_c=20°C.

As shown in Fig. 12, T_{av} the temperature of working zone can be lowered 3.2°C, 3.9°C, 4.6°C and 5.4°C respectively under the condition of T_h =50°C, T_h=60°C, T_h=70°C and T_h=80°C in the case of $L=L_{eff}$ compared to the case of L=0. It proves use of the vertical supply duct is very effective. The higher T_h, the higher the ΔT_{av} in average.

The analysis has been conducted for the simulated results compared to the measured results at measuring points shown in Fig. 3 (Fig. 13).

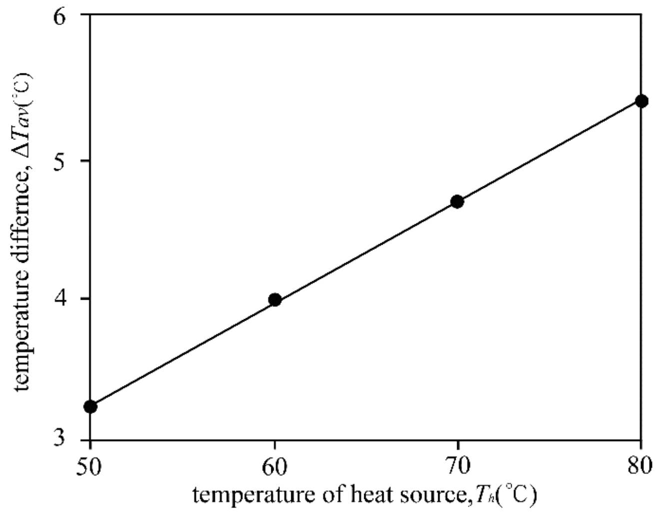

Fig. 12. Change of ΔT_{av} depending on T_h

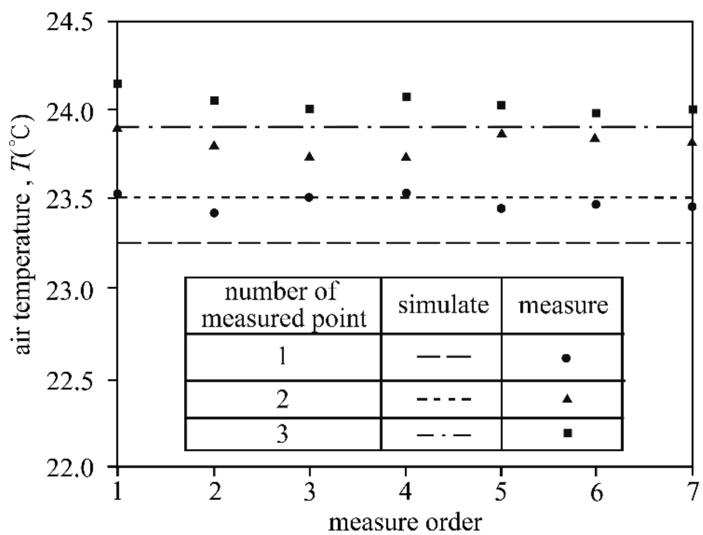

Fig. 13. Comparison between simulated and measured values

The error between them is 1.8 percent and measured value is a little higher that all the simulated values. It is supposed that the simulated values are not considered the low-temperature radiation between heat source and measuring element, but measured values are affected by it.

In practical application, the length of supply duct has been set as 4m from an aesthetic view. The velocity of supply air has been set out as 2 m/s, 2.2 m/s, 3.2 m/s and 4 m/s respectively in the volume with surface temperature of the process equipment as 50℃, 60℃, 70℃ and 80℃ respectively (Fig. 11. reference).

6. Conclusions

In this paper the energy saving character of HDVS which has vertical supply duct installed and with heat source has been analyzed compared to MVS for ventilating high-ceiling buildings. The modeling and calculation have been done using CFD and based on that the change of T_{av}, the average temperature of the air in the working zone depending on the temperature of the heat source and the velocity of air inflow has been studied. Under conditions of provision of equal temperature and velocity of air inflow T_{av}, the temperature of the air in the working zone depends on the length of vertical supply duct.

And the temperature of the air in the working zone increases when the vertical supply duct is longer than the limited value. In this model the character of heat source has been presented in the surface temperature instead of released amount of heat and the air temperature of working zone has been calculated considering low temperature radiation. The result is significant to verify that effective length of vertical supply duct exists from view of energy saving in HDVS equipped with the vertical supply duct. This enables the power consumption of the air conditioning to reduce to the maximum by increasing cooling air temperature and therefore, it is the reasonable proposal for substantial energy saving. This system is presented to allow 3.2~5.4℃ of temperature of air supply to be decreased or amount of air supply to be reduced, accordingly, compared to the mixing ventilation system. This system contributes to the energy consumption of 35kw compared to the mixing ventilation system, which leads to the annual energy saving of about 80,000kWh.

In the future, for energy saving in the ventilation system, we will study about the effective length of supply duct according to changes of heat source characteristic and height of building.

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Nomenclature

B width of workshop (m)

b width of heat source (m)

C_1, C_2, C_μ turbulent constants

C_p specific heat capacity at constant pressure ($Jkg^{-1}K^{-1}$)

H height of workshop(m)

h height of heat source(m)

I turbulent intensity

I_{in} intensity of entering radiation (Wm^{-2})

k kinetic energy of turbulence (m^2s^{-2})

L length of the supply duct from the ceiling to the supply air sector (m)

L_{eff} effective length of supply duct (m)

l distance of turbulent flow (m)

P air pressure (Pa)

q_{in} Intensity of radiation arriving at wall (Wm^{-2})

S_u , S_v , S_w term of momentum conservation

T air temperature (℃)

T_{av} average temperature of indoor air (℃)

T_c temperature of cooling air in supply duct (℃)

T_h heat source temperature (℃)

T_s standard temperature of working zone (℃)

T_{wi} temperatures of *i*-th wall (℃)

T_{ws} temperature of radiated wall(℃).

v_0 air velocity in supply duct (ms^{-1})

u, v, w air velocity in direction of *x, y, z* axial (ms^{-1})

x, y, z Cartesian coordinates (m)

Greek symbols

β coefficient of cubic expansion(K^{-1})

ε viscous dissipation in turbulent flow

ε_{wi} blackness of *i*-th wall

ε_{in-s} blackness of radiated wall.

λ heat conductivity of general air ($Wm^{-1}K^{-1}$)

λ_t heat conductivity of turbulence air($Wm^{-1}K^{-1}$)

μ_{eff} effective molecular viscosity of air(Pa s)

μ dynamic viscosity (Pa s)

μ_t turbulent eddy viscosity (Pa s).

ρ density of air($kg\ m^{-3}$).

σ Stefan-Boltzmann constant ($Wm^{-2}K^{-4}$)

σ_k, σ_μ, σ_ε, σ_T empirical constants

τ time(s)

Subscripts

av average
eff effective
h heat source
s standard
t turbulent
wi *i*-th wall
ws radiated wall

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Highlights

·we present design using vertical supply duct in hybrid displacement ventilation system (HDVS).

··we establish simulation evaluation to optimize the design of ventilation system by CFD.

··For saving energy, it is authenticated to exist the effective length of vertical supply duct in HDVS.

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To
Jianlei Niu, Editors-in-Chief
"Energy and Buildings"

Dear Jianlei Niu,
I am submitting a manuscript titled "Energy Saving of hybrid displacement ventilation system with vertical supply duct in the industrial conditioning" by Paek Myong-Chol, Jon Chol-Jin to the "Energy and Buildings". The contents of this manuscript have not been copyrighted or published previously.
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Best Regards.
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