Improvements of ADPI and ventilation effectiveness of a classroom by a dedicated outdoor air system

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ABSTRACT

School indoor air quality has become of concern recently in Korea. In this paper, it is intended to investigate the ventilation performance and thermal comfort characteristics of a classroom, when an outdoor air system is installed in addition to a ceiling-mounted heat pump system. Experiments were conducted in a full-scale model classroom to collect experimental data to validate numerical schemes. Three-dimensional temperature distributions were measured with thermocouples distributed throughout the space, and ventilation effectiveness was measured using a tracer gas technique. CFD calculations were performed to investigate the effects of various system parameters on air diffusion performance index and air change efficiency. This paper focuses mainly on the effect of the discharge angle of the air jets from a four-way-cassette heat pump mounted on a ceiling by the interaction with the fresh air supplied from the outdoor air system.

INTRODUCTION

Conventional classrooms in Korea used to be equipped only with heating facilities, such as steam radiators, gas heaters, etc. In recent years, floor-standing air-conditioner or ceiling-mounted heat pumps are installed in existing and new classrooms to provide cooling as well as heating capabilities. In order to save electric energy, however, buildings are made more air-tight, and window openings become less frequent. School indoor air quality is therefore of concern, since most school kids spend a lot of their time in highly populated classrooms. Many students complain of drowsiness and asthenia due to lack of fresh outdoor air. It is reported that carbon dioxide concentration can be well over the indoor concentration guideline of 1000 ppm (Kim, 1995). According to the Act of Indoor Air Qualities for public spaces (Ministry of Environment, 2003) recently passed in Korea, it is required to install proper ventilation systems so as to supply minimum outdoor air.

As far as heating and cooling are concerned, temperature and humidity are the major

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parameters determining indoor thermal comfort conditions (ASHRAE, 1992). When a dedicated outdoor air system is installed in addition to a heating/cooling system, ventilation performance for distributing fresh outdoor air is important as well. In this study, it is intended to investigate air change efficiency and air diffusion performance index depending on various airflow patterns created by mutual interactions of the discharged air from the heat pump system and the supplied air by the outdoor air system.

MODEL ROOM AND EXPERIMENTS

The model room has the dimensions of a typical school classroom. It is constructed in an outdoor environmental chamber, which is 17.4 m × 11.1 m × 7.2 m in internal dimensions. The environmental chamber provides outdoor air conditions. The temperature can be controlled from –20 to 60°C (± 0.5°C), and relative humidity from 5 to 90% (± 5%).

A schematic drawing of the model room is shown in Figure 1. The floor area is 88.1 m^2, and the height is 2.7 m. A four-way cassette heat pump is located nearly at the centre of the ceiling, and four supply diffusers and four exhaust outlets are installed at ceiling corners.

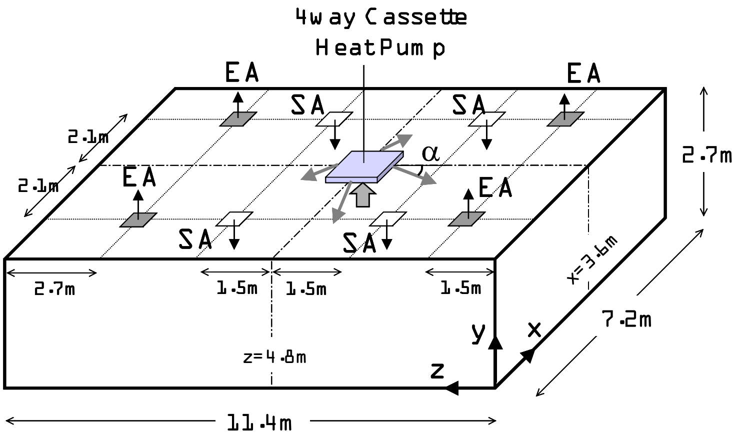


Figure 1 Schematic drawing of the model room.

The heat pump discharges heated or cooled air into the room through four-way slots to take care of the heating or cooling loads. Room air is sucked into the bottom face of the heat pump and re-circulated through the internal coils before being discharged. The re-circulating airflow rate is 0.483 m^3/s (1740 CMH).

Fresh outdoor air is supplied through square diffusers of 305 × 305 mm, and room air is exhausted to the outside of the building. The temperature difference from the exhaust air is recovered by a plate-type air-to-air heat exchanger. The ventilation airflow rate is 0.194 m^3/s (700 CMH), and is equally distributed to four diffusers. It corresponds to the air change rate of 3.16 ACH. The experimental reference conditions are shown in Table 1.

Temperature distributions were measured using 990 T-type thermocouples evenly distributed in the three-dimensional space. A steady state was reached approximately 4 h after an experiment started, and data were taken afterwards for 1 h. Air change efficiency was measured using a tracer gas technique (Sandberg, 1983). Sulfur hexafluoride was injected as a tracer and mixed with room air using portable electric fans. Step-down concentration decays were measured continuously in the exhaust duct using a multi-channel gas analyser.

Table 1 Experimental reference conditions

	$T_{outdoor}$ (°C)	T_{indoor} (°C)	Four-way cassette heat pump			Outdoor air system		
			Flow rate (CMH)	Discharge angle	$T_{discharge}$ (°C)	Flow rate (CMH)	Supply angle	T_{supply} (°C)
Cooling	33.0	26.0	1740	20-50	12.0	700	50	31.8
Heating	7.0	20.0	1740	20-50	38.0	700	50	13.0

NUMERICAL ANALYSIS

A commercial CFD package was used to simulate the airflow and temperature distributions in the room. The numbers of grids are 78 × 50 × 105. Details of governing equations and boundary conditions are not listed here. From the velocity and temperature data, the effective draft temperature (EDT) is calculated at every grid point using the equation below:

$$T_{\mathrm{eff}} = (T - T_{\mathrm{ave}}) - 8.0(V - 0.15) \qquad (1)$$

where T is a local temperature and T_{ave} is the zone average temperature both in degrees Celsius. The local air velocity, V, is in m/s. The air diffusion performance index (ADPI) is defined as the percentage of the comfortable area, excluding hot, cold or drafty areas. The comfortable area is the region where EDT is between –1.7 and +1.1°C, and the air velocity is less than 0.35 m/s (Kirkpatrick, 1996).

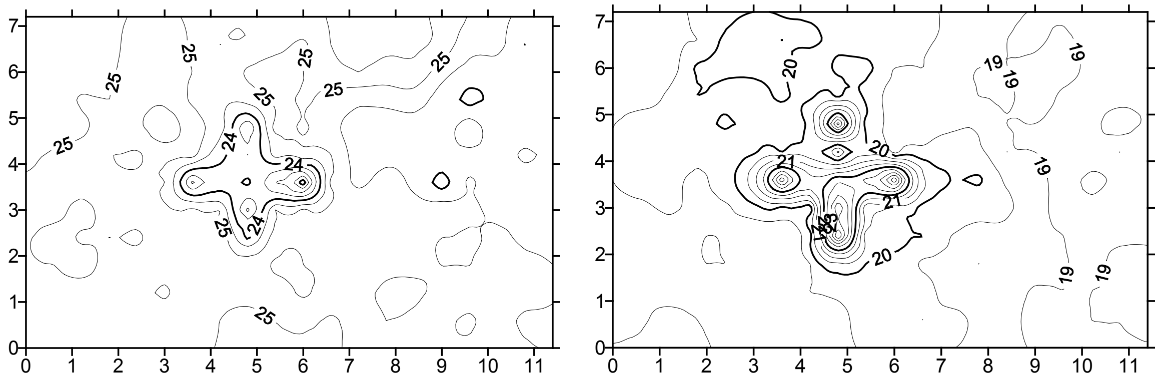
The ventilation performance of a room is evaluated by the air change efficiency (ACE). It is defined as the nominal ventilation time relative to the average age of air in the room.

$$\varepsilon = \frac{\tau_{\mathrm{n}}}{\langle \tau_{\mathrm{p}} \rangle} \qquad (2)$$

where τ_n is the nominal ventilation time, defined as the time required to flush the room air with the ventilation airflow rate, τ_p is the local mean age at an internal point and $\langle\ \rangle$ represents the average value over the volume it is averaged. The local mean age distributions are obtained from the steady-state concentration calculations when uniform contaminant sources are present in the entire domain (Han, 1992).

RESULTS AND DISCUSSIONS

Figure 2 shows the temperature distributions experimentally measured in a horizontal plane. Isothermal contours created by the discharged four-way jets can be seen below the heat pump for both cases. Except the regions right below the heat pump, the temperatures are maintained nearly at the setting temperatures. Figure 3 shows the temperature variations along the horizontal centerlines below the heat pump for the cooling condition. The numerical results superimposed are in good agreement with the experimental data. A slight stratification can be observed in the vertical direction.



(a) Cooling condition (b) Heating condition

Figure 2 Isothermal contours measured in the horizontal plane at the height of $y = 1.6$ m.

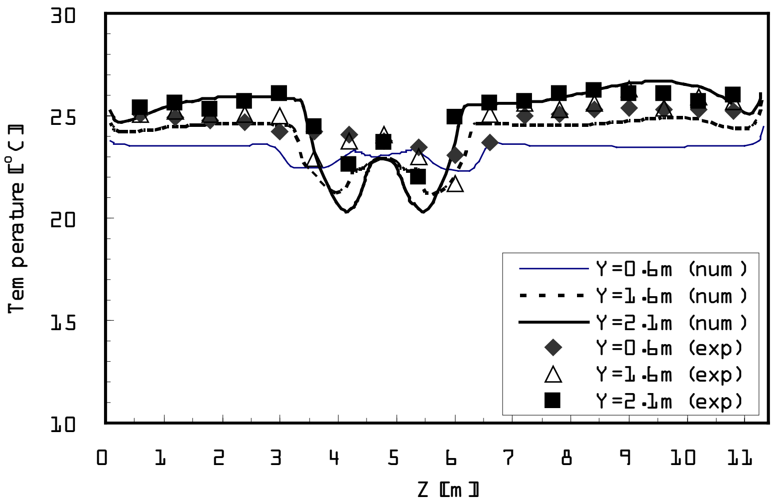
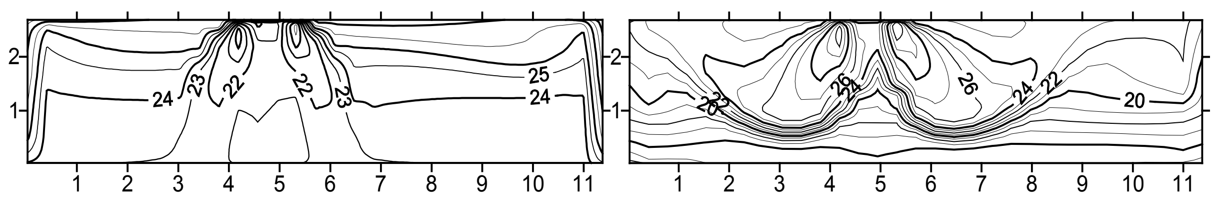


Figure 3 Comparison of experimental and numerical temperature distributions along the horizontal lines in the z-direction at $x = 3.6$ m.

The behaviour of the discharged jets can be clearly observed in vertical planes. Figure 4(a) shows the cold jets reach the floor vertically, and move towards the side walls. In Figure 4(b), however, the hot air jets discharged with an angle deflect upward because of the buoyancy effect. For both cases, the discharge angle is 50° from the ceiling.



(a) Cooling condition (b) Heating condition

Figure 4 Temperature distributions calculated in the vertical cross-sectional plane of $x = 3.6$ m.

Numerical simulations have been conducted for various discharge angles. When the angle is small, the discharged air moves almost horizontally along the ceiling and makes a large circulation after hitting the vertical wall for each direction. The circulation helps mixing the entire room air effectively. The air diffusion performance index calculated in an occupant zone below 1.8 m is shown in Figure 5 as a function of the discharge angle. As the angle is decreased from the experimental condition below 50°, the ADPI decreases slightly for cooling conditions, but increases for heating conditions. There is a large increase in ADPI as the angle is changed from 50° to 40° for heating conditions. This indicates the horizontal jets can provide more uniform and comfortable temperature distributions in an occupant region.

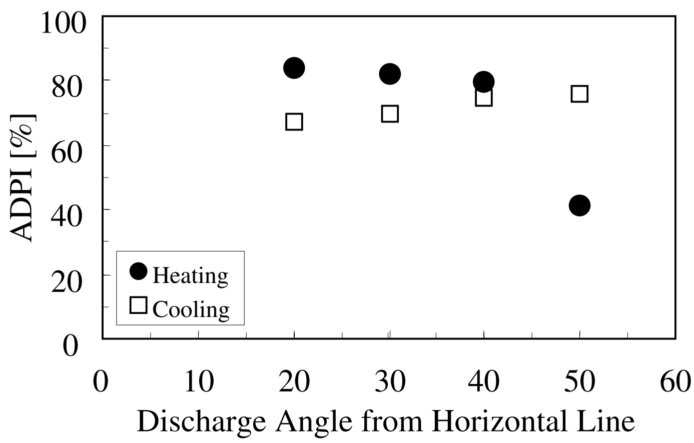


Figure 5 Air diffusion performance index as a function of the discharge angle of heat pump.

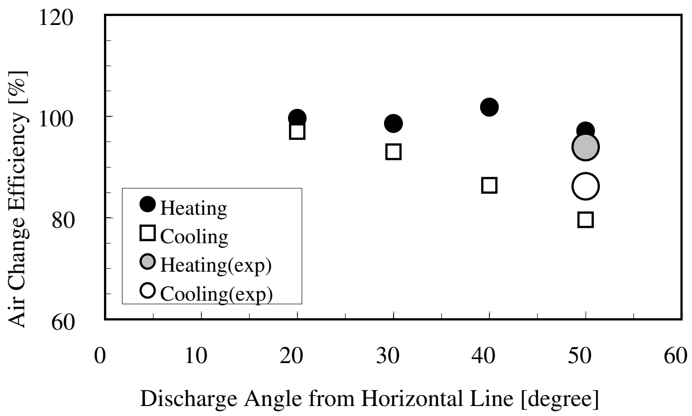


Figure 6 Air change efficiency as a function of the discharge angle of heat pump.

The introduction of outdoor air through supply diffusers makes the room airflow patterns more complicated. For summer conditions, warm supply air stays near the ceiling before being mixed with the cold air from the heat pump. Some of the supply air is entrained into the jets from the heat pump and moves along with them. Some moves slowly toward the center of the room along the ceiling, and is sucked into the return grille of the heat pump. For winter conditions, the supply air from diffusers is relatively cold, and pours downward into the room. The fresh air has enough time to be mixed with the room air before being exhausted compared to summer conditions, and hence shows better air change efficiency.

Figure 6 shows the air change efficiency of the room for the discharge angle between 20° and 50°. The ACE is greater for heating conditions than cooling conditions for large discharge angles, which is in agreement with experimental results. The ACE measured by the tracer gas experiment is 0.86 for the summer condition, and 0.94 for the winter condition at the angle of 50°. For small discharge angles, the ACE becomes nearly 100% for both heating and cooling conditions. As was discussed previously, it is due to the vigorous air circulations created by the horizontal discharge jet from the heat pump regardless of heating or cooling.

CONCLUSIONS

The ventilation and thermal environmental characteristics of a school classroom are investigated when a dedicated outdoor air system is installed in addition to an existing four-way cassette heat pump. The discharge airflow by the heat pump helps mixing the outdoor air coming through supply diffusers with indoor air effectively. Room airflow patterns are totally different depending on the discharge angle and the air temperature from the heat pump. Buoyancy force creates deflected upward airflow patterns for heating conditions especially when the discharge angle is large. For discharge angles less than 30°, the horizontal jets create large air circulation regardless of heating or cooling.

The air diffusion performance index increases slightly as the discharge angle increases for cooling conditions, but decreases for heating conditions. The air change efficiency is nearly constant for heating conditions, but decreases for cooling conditions as the discharge angle increases. A discharge angle of approximately 30° from the ceiling is found to be optimal for the present configurations, which is applicable for both winter and summer weather conditions. The effect of the locations of the supply and exhaust outlets on ADPI and ACE need be further investigated.

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