A Study on the Evaluation of Air Change Efficiency of Multi-Air-Conditioner Coupled with Ventilation System

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ABSTRACT: Indoor air quality becomes of a concern recently in view of human health. This study investigates the air diffusion performance and the air change efficiency of a classroom, when outdoor air is introduced in two different ways in addition to the heating/cooling operation of a ceiling-mounted heat pump. A CFD analysis has been performed to investigate the effect of the discharge angle of the air jets from the heat pump for both parallel and series types of outdoor air system. It is observed that the series type creates more uniform indoor environment compared to the parallel type in general. It can be concluded the discharge angle should not be larger than 40o for the parallel type, in order not to generate thermal stratification in the room.

Nomenclature

$ADPI$	: Air diffusion performance index [%]
V_x	: Velocity at point x [m/s]
T_x	: Temperature at point x [℃]
T_c	: Room setting temperature [℃]
X	: Horizontal coordinate [m]
Y	: Vertical coordinate [m]
Z	: Longitudinal coordinate [m]

Greek symbols

a	: Discharge angle [deg]
ε	: Air change efficiency [%]
ϕ	: Effective draft temperature [℃]
τ_n	: Nominal time constant [s]
τ_p	: Local mean age [s]

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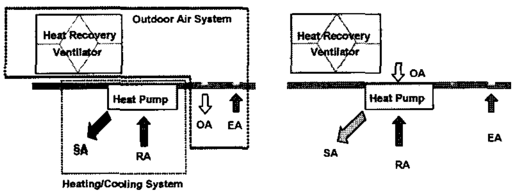
1. Introduction

Recently, ceiling-mounted heat pumps are widely used in classrooms to provide cooling as well as heating capabilities. In order to save electric energy, however, school buildings are made more air-tightened, and window openings become less frequent. According to the Act of Indoor Air Qualities[1] for public spaces legislated in 2003 by Ministry of Environment, 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 major parameters determining indoor thermal comfort conditions.[2] 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 (ACE) and air diffusion performance index (ADPI) depending on various airflow patterns created by mutual interactions between the discharged air from a heat pump system and the supplied air by an outdoor air system. Major parameters used in this study in evaluating ACE and ADPI are the discharge angle of the air jets from the 4-way cassette heat pump and operating conditions of the heat pump coupled with the outdoor air system.

2. Simulations

2.1 Numerical conditions

An outdoor air system applied to a classroom is coupled with a heat pump system in two different ways, i.e. a parallel type and a series (direct-coupled) type as shown in Fig. 1. In parallel type, outdoor air is supplied into the room separately in parallel with the conditioned air by the heat pump. However, in series type, the outdoor air is supplied into the coil of the heat pump and it is mixed with the recirculated air in the heat pump.



(A) Parallel type (B) Series type

Fig. 1 Schematic drawing of the heat pump coupled with an outdoor air system.

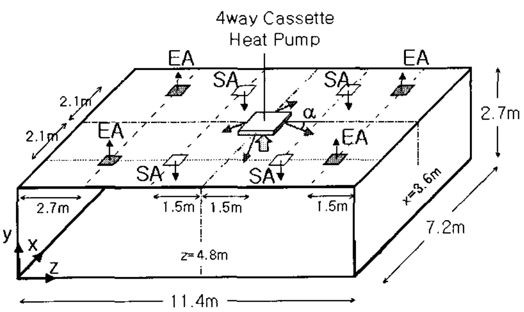


Fig. 2 Schematic drawing of the model room.

The model room has the dimension of a typical school classroom, which is 7.2 × 2.7 × 11.4 m in internal dimensions. A schematic drawing of the model room is shown in Fig. 2. The floor area is 88.1m^2, and the height is 2.7m. A four-way cassette heat pump is located nearly at the center of the ceiling, and four supply diffusers for parallel type and four exhaust outlets are installed at ceiling corners. 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. 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 for parallel type, and through discharge intake of heat pump for series type. The regulation on indoor air quality states that CO_2 concentration

Table 1 Boundary conditions for numerical analysis

Air volume rate of heat pump	29 CMM (1740 CMH)
OA volume rate	700 CMH (3.15 ACH)
Setup temperature	20 ℃ at $\alpha = 50°$
Skin load	11300 W (9748 kcal/h)
OA intake dimension	305 mm × 305 mm
OA discharge angle	50° (only for parallel)
Kinetic energy of inlet	$(0.05 \times V_{in})^2$
Dissipation rate of inlet	$0.163 \times k_{in}/(0.1 \times D)$
Dissipation rate of wall	$(C_\mu^{3/4} k^{3/2})/(\kappa y)$

Table 2 Discharge and outdoor air conditions

	Heat pump discharge		Outdoor air supplied into indoor	
	Temp.	Velocity	Temp.	Velocity
Parallel type	38.7 ℃	5 m/s	13 ℃	0.5 m/s
Series type	31.3 ℃	7 m/s	-	-

should be kept below 1000 ppm on 1-hour basis. There are 35~41 students in each school classroom, and the occupancy density is 0.5 person/m^2 approximately. As it is higher than in office buildings in general and classrooms must be installed with an adequate facilities for mechanical ventilation with heat recovery system for energy savings.

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 numerical boundary conditions are shown in Table 1. Because the exhausted heat is recovered by a plate-type air-to-air heat exchanger, temperature of fresh outdoor air supplied into indoor is higher than outdoor air temperature as shown in Table 2. A major difference between the parallel and the series types, as shown in Table 2, is that discharge velocity of a series type is higher than that of a parallel type. Thereby, discharge temperature of heat pump for a series type is lower than that for a parallel type.

2.2 Numerical method

A commercial CFD code based on the finite differential method has been utilized, which uses SIMPLEST algorithm for pressure equation. A continuity equation and momentum equations are solved simultaneously along with temperature and concentration equations. A standard k-ε model is used to calculate turbulent quantities. The numbers of grids are 78 × 50 × 105. Boundary conditions used are no slip conditions for velocity and adiabatic conditions for heat and mass transfer along the boundaries except at supply/exhaust diffusers. Pressure boundary conditions are applied for the exhausts. To check the convergence, the maximum mass source of all the control volumes, SMAX, is monitored. The convergence criteria of SMAX = 10^{7} is applied. From the velocity and temperature data, the effective draft temperature (EDT) is calculated at every grid point using the equation below.

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

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 place where EDT[3] is between -1.7 ℃ and +1.1 ℃, and the air velocity is less than 0.35 m/s.

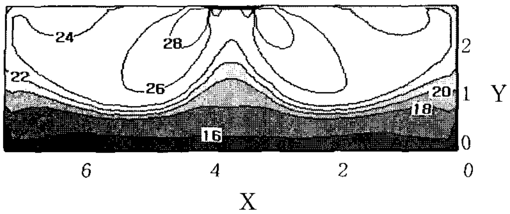
The ventilation performance of a room is evaluated by the air change efficiency (ACE). It is defined as the nominal ventilation time relative to the average age of air in the room.

$$\varepsilon = \frac{\tau_n}{<\tau_p>} \tag{2}$$

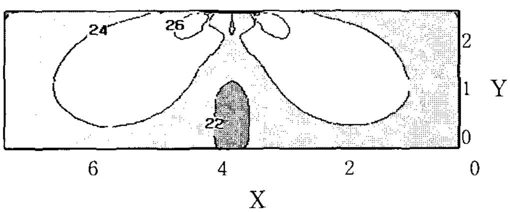
where τ_n is the nominal time constant (1139.7 s), defined as the time required to flush the room air with the ventilation airflow rate. τ_p is the local mean age at an internal point, and < > represents the average value over the volume it is averaged. The local mean age distributions are obtained from the steady-state concentration calculations when uniform contaminant sources are present in the entire domain.[4]

3. Results and discussion

Numerical simulations have been conducted for various discharge angles. Fig. 3 shows temperature distributions calculated in vertical center planes. For both cases, the discharge angle is 50 degree from the ceiling. Isothermal contours created by the discharged four-way jets can be seen below the heat pumps for both types. The behavior of the discharged jets can be clearly observed in the vertical planes. For

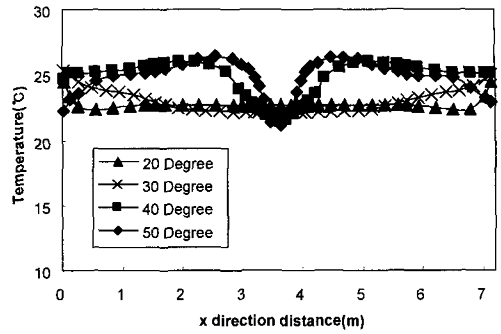


(a) Parallel type

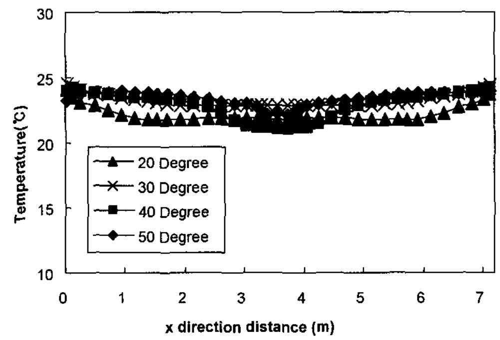


(b) Series type

Fig. 3 Temperature distributions of the vertical cross-sectional plane at Z = 3.6 m (α = 50°).



(a) Parallel type

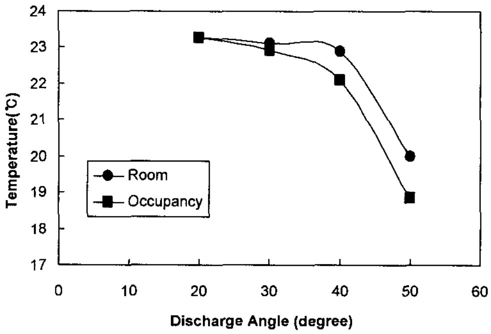


(b) Series type

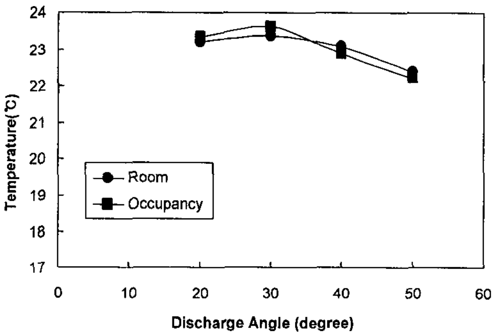
Fig. 4 Temperature distributions along x direction at Y = 1.6 m and Z = 4.8 m.

parallel type, temperature distributions at occupant zone below 1.0 m are maintained lower than the setting temperature. It is shown that the hot air jets discharged with an angle deflect upward because of the buoyancy effect. For series type in Fig. 3(b), however, air jets reach the floor, and move towards the side walls. It is due to a relative lower discharge temperature discharged by a heat pump. The temperature variations obtained along the centerline below the heat pumps (Y = 1.6 m and Z = 4.8 m) are shown in Fig. 4 as a function of discharge angle. There is a large temperature gradient in the regions below heat pump for parallel type as the angle is changing from 40 to 50 because of the strong stratifications in the room. For series type, however, relatively lower discharge temperature can provide more uniform temperature in the region below the heat pump than parallel type. When the angle is small, 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 entire room air and maintaining uniform temperature regardless of the types.

Fig. 5(a) shows the mean temperatures of entire room and occupant region for parallel type. When the angle ranges from 20 to 30, the mean temperatures of room and occupant region are nearly the same by the large circulation as was discussed previously. As the angle is increased up to 50, the mean temperature difference between room and occupant zone increases over 1 ℃, because of the temperature stratifications. In addition, the mean room temperature increases abruptly over 3 ℃ by decreasing the discharge angle from 50 to 40. For series type, however, the mean temperatures of room and occupant region are about the same regardless of the discharge angle as shown in Fig. 5(b). A decrease in room mean temperature as the angle changing from 50 to 40, is not significant because of the weak stratifications in the room by a decreased dis-

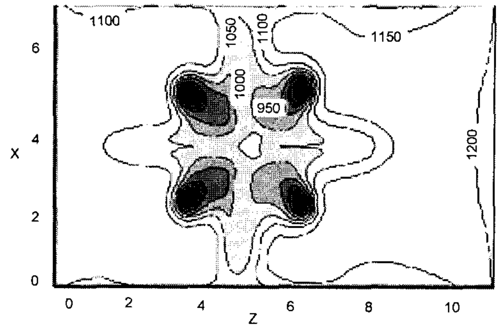


(a) Parallel type

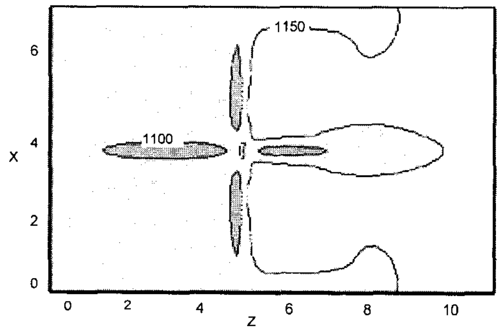


(b) Series type

Fig. 5 Mean temperatures as a function of the discharge angle of the heat pump.



(a) Parallel type



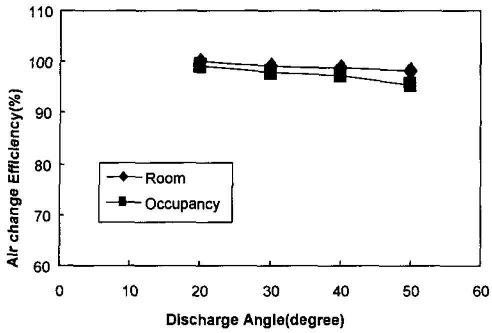
(b) Series type

Fig. 6 Local mean age contours of the horizontal plane at Y = 1.6 m(α = 50°).

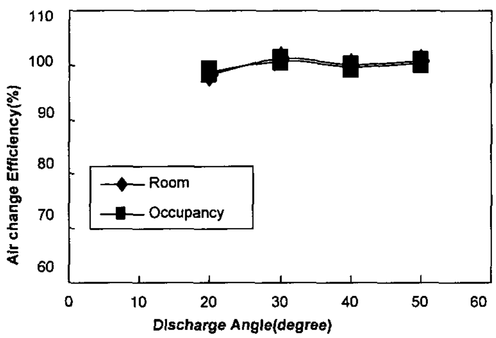
charge temperature of the heat pump.

Fig. 6 shows the local mean age (LMA) distributions calculated in a horizontal plane of breathing zone. For parallel type, because relatively cold air jets are discharged through supply diffusers, they are dropped in the region below supply air diffusers near heat pump. The LMA values are small in the region. Except the regions right below the heat pump for parallel type, the LMA distributions are maintained nearly at the nominal time constant. However, since the fresh outdoor air is supplied by way of the discharge slots of the heat pump for series type, the LMA's are smaller compared to those by parallel type. It is evident that heat pump helps complete mixing between the entire room air and fresh outdoor air.

Fig. 7 shows the air change efficiency (ACE) of the room as the discharge angle ranges from 20 to 50. For series type, the ACE of room and occupant region becomes nearly 100% regardless of the discharge angle. As was discussed previously, it is due to the vigorous air circulations created by the fresh outdoor air discharged by way of the heat pump. For parallel type, however, the ACE's of room and occupant zone are slightly decreasing as the discharge angle increases. The difference of ACE's between room and occupant region is slightly increasing with the discharge angle. It is due to the short recirculation of fresh outdoor air created by hot air jets raised upward because of the buoyancy effect. It is evident that the fresh air of parallel type has not enough time to be mixed with the room air before being exhausted compared to series type, and hence shows better air change efficiency. The air diffusion performance indices calculated in occu-

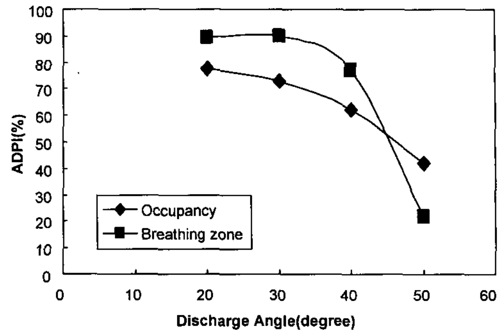


(a) Parallel type

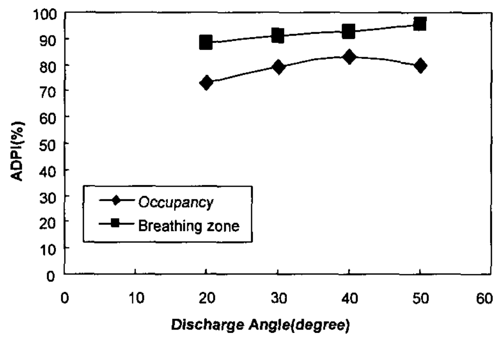


(b) Series type

Fig. 7 Air change efficiency as a function of the discharge angle of the heat pump.



(a) Parallel type



(b) Series type

Fig. 8 ADPI as a function of the discharge angle of the heat pump.

pant region and breathing region below 1.8 m are shown in Fig. 8 as a function of the discharge angle. As the angle is decreased at 50, the ADPI decreases slightly for series type, but increases sharply for parallel type. This indicates the horizontal jets can provide more uniform and comfortable temperature distributions in an occupant and breathing region. The introduction of outdoor air through supply diffusers makes the room airflow patterns more complicated. For series type, as was discussed previously, recirculated room air is mixed well with fresh outdoor air before being discharged by the heat pump. and hence shows better the ADPI.

4. 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 4-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 only for parallel type especially when the discharge angle is large. For discharge angles less than 30, the horizontal jets create large air circulation regardless of the ventilation types. The air diffusion performance index increases slightly as the discharge angle increases for series type, but decreases for parallel type. The air change efficiency is nearly constant for series type, but decreases for parallel type as the discharge angle increases. The discharge angle of approximately 30 degree from the ceiling is found to be optimal for the present configurations, which is applicable for both parallel and series types. The effect of the locations of the supply and exhaust outlets on ADPI and ACE need to be further investigated.

References

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