International Journal of High-Rise Buildings
March 2019, Vol 8, No 1, 71-82
https://doi.org/10.21022/IJHRB.2019.8.1.71

International Journal of
High-Rise Buildings
www.ctbuh-korea.org/ijhrb/index.php

Energy Saving Potential and Indoor Air Quality Benefits of Multiple Zone Dedicated Outdoor Air System

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Abstract

The purpose of this study is to evaluate the indoor air quality (IAQ) and energy benefits of a dedicated outdoor air system (DOAS) and compare them with a conventional variable air volume (VAV) system. The DOAS is a decoupled system that supplies only outdoor air, while reducing its consumption using an enthalpy wheel. The VAV system supplies air that is mixed outdoor and transferred indoor. The VAV has the issue of unbalanced ventilation in each room in multiple zones because it supplies mixing air. The DOAS does not have this problem because it supplies only outdoor air. That is, the DOAS is a 100% outdoor air system and the VAV is an air conditioning system.

The transient simulations of carbon dioxide concentration and energy consumption were performed using a MATLAB program based on the thermal loads from the model predicted by the TRNSYS 18 program. The results indicated that when the air volume is large, such as in summer, the distribution of air is not appropriate in the VAV system. The DOAS however, supplies the outdoor air stably. Moreover, in terms of annual primary energy consumption, the DOAS consumed approximately 40% less energy than the VAV system.

Keywords: Variable air volume system, Dedicated outdoor air system, Multiple zone, Indoor air quality, Energy consumption

1. Introduction

Over the last decade, dedicated outdoor air systems (DOAS) have attracted a lot of attention thanks to their benefits in energy consumption by decoupling sensible and latent cooling. Furthermore, compared with conventional systems, such as variable air volume (VAV) systems, they have advantages in indoor air quality, because DOAS intakes only outdoor air.

Jeong et al. (2003) demonstrated the energy conservation benefits of a DOAS with parallel sensible cooling by ceiling radiant panels. The annual electrical energy consumption of the DOAS is 42% less than that of the VAV system. Kim et al. (2016) also studied the energy performance of a DOAS and compared it to those of a desiccant-enhanced evaporative air conditioner (DEVap) developed based on a VAV system.

However, the studies addressing the indoor air quality of these systems in multiple zone are scarce, as the established investigations were focused primarily on the energy saving potential. Although there are indoor air benefits in single zone DOAS, it is not clear whether there are indoor air benefits in multiple zone DOAS because the VAV system introduces larger amounts of outdoor air according to the multiple spaces method of the ASHRAE Standard 62.1.

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Consequently, the indoor air quality of VAV system and DOAS with parallel cooling and heating unit, was evaluated by simulating the transient variations of carbon dioxide concentration in the room supplied by each system. Moreover, the energy consumption was calculated using the energy simulation model. The simulations of the transient variations of carbon dioxide concentration and energy consumption were performed using a MATLAB program based on the thermal loads from a model predicted by the TRNSYS 18 program.

2. System Overview

2.1. Dedicated Outdoor Air System

Unlike the VAV system, the DOAS does not mix outdoor air with exhaust air. There are no issues such as cross contamination, which may occur in a VAV systems. The DOAS also consumes less energy than the VAV system because it uses the enthalpy wheel, which can exchange the enthalpy of the outdoor and exhaust air.

The DOAS purpose is dehumidification and the operation of each component is determined by the operation mode. In summer and winter, enthalpy exchange between outdoor and exhaust air, through the enthalpy wheel, has energy benefits, while the enthalpy wheel is not used in the intermediate seasons.

The cooling coil is activated to satisfy the target humidity ratio of the supply air in the DOAS. Part of the sensible load can be treated through a cooling coil operated for

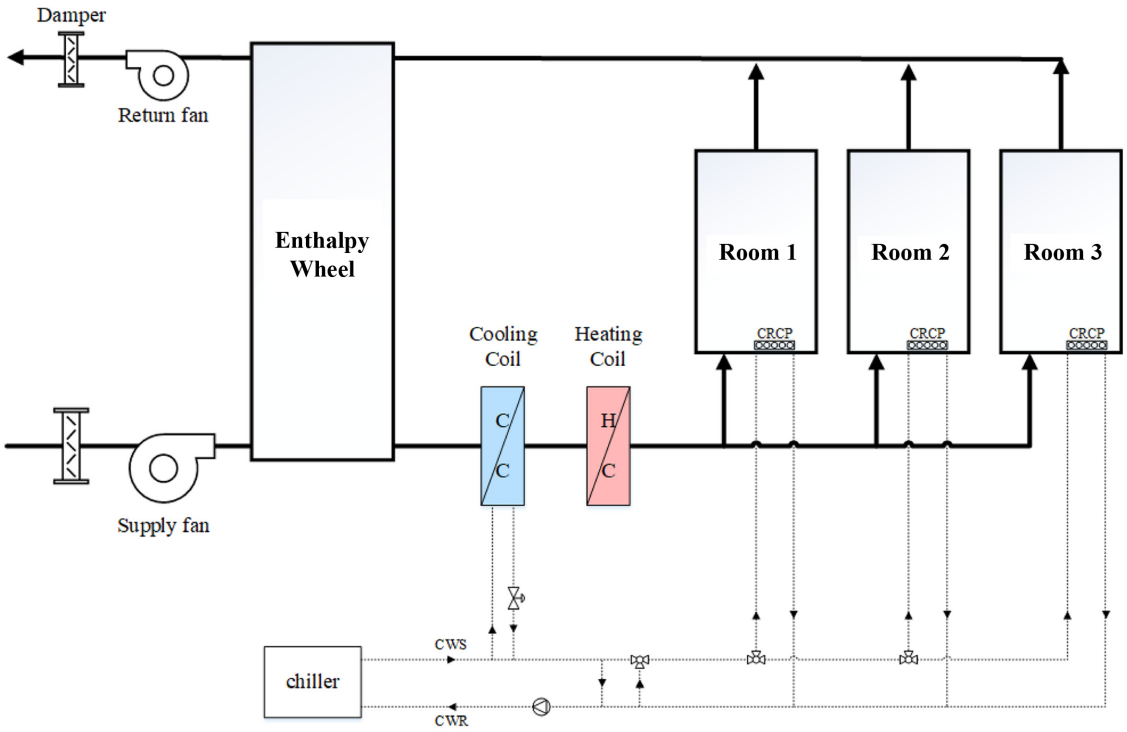


Figure 1. Dedicated outdoor air system schematic.

dehumidification, but it cannot be completely removed. The remaining sensible load is treated through a ceiling radiant cooling panel located in each room.

On the other hand, during the heating mode, the heating coil is operated. The outdoor air is heated up to 20°C through the heating coil, and the remaining sensible load is treated through the heating device located in each room. Normally this function is performed by the sensible wheel, which exchanges the outdoor and exhaust air sensible heat, however, in this study the heating coil is used instead.

In the case of the VAV system, the air flowrate and the operation of the cooling and heating coils are determined by the load. As the temperature and relative humidity of the supply air are fixed, the load generated in the room may not be removed. However, in the case of the DOAS, a cooling coil is used to treat the indoor latent load and thus a part of the sensible load can be treated. In addition, the remaining sensible load is treated through the ceiling radiant cooling panel and the indoor heating device, so that the load can be completely separated and processed.

2.2. Variable Air Volume System

The VAV system controls the room temperature by keeping the supply air (SA) temperature constant and changing the SA volume depending on the load.

The VAV system has the advantage of consuming less energy compared with the outdoor air (OA) system. This is because it takes in the mixed outdoor and return air (RA). However, it has the problem of cross contamination because it uses the contaminated return air. This cross contamination is a serious disadvantage especially for buildings such as hospitals and schools.

In this study, the minimum ventilation rate is calculated based on the ASHRAE Standard considering the maximum occupancy rate and the occupant scheduling, because this system uses demand airflow control. In addition, the SA flowrate is determined using the sensible and latent loads, which are calculated from TRNSYS 18.

During the cooling mode, the cooling and heating coils are operated. Given that the VAV system has a constant supply temperature and a variable air volume, the air temperature, after the cooling coil, is set to 13°C and the relative humidity to 100%. The air supplied directly under these conditions is very cold, therefore a heating coil is used to reheat the air to 15°C. Finally, the supply air is at a dry bulb temperature of 15°C and a relative humidity of about 88%.

During the heating mode, only the heating coil is operated. At this time, the temperature of the air passed through the heating coil is set to 45°C. Given that this process is a sensible heating process, the relative humidity is not considered.

3. Simulation Overview

3.1. Building Model Overview

In this study, the simulation is performed for office buildings. The indoor carbon dioxide concentration in multiple

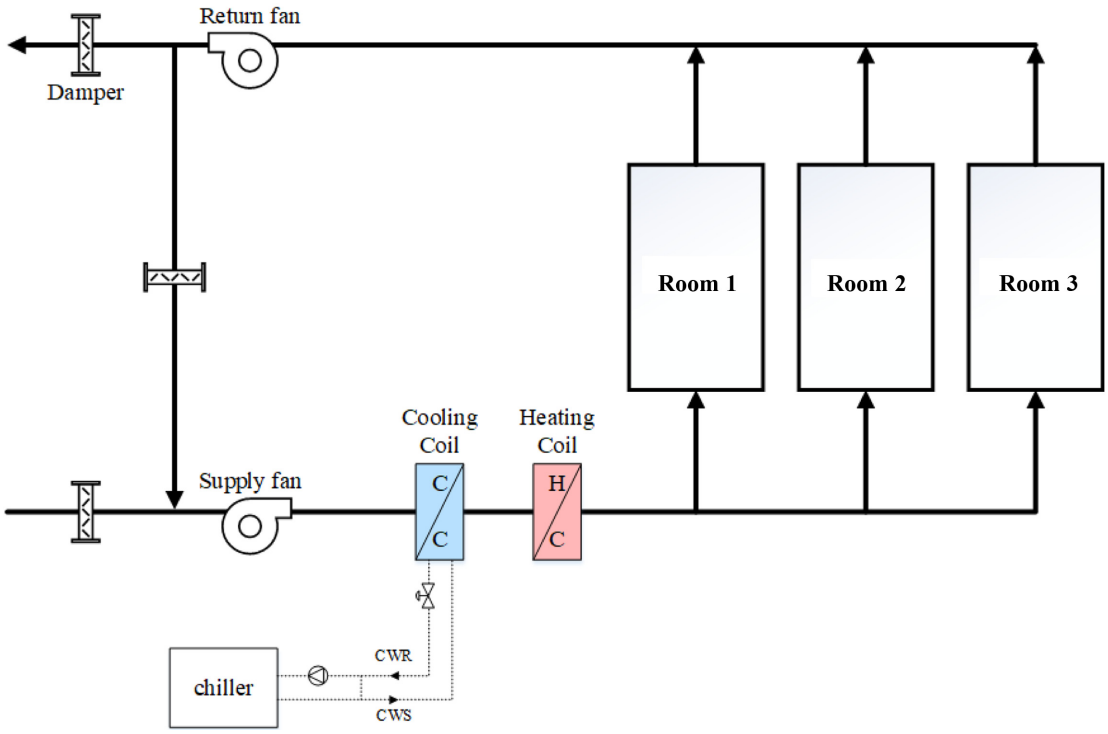


Figure 2. Variable air volume system schematic.

zones and the energy of system operation are the main subjects of this study in all systems.

Sensible and latent loads are calculated from TRNSYS 18 based on the annual outdoor air condition provided by the same program. Occupant and lighting schedules are also considered. Based on the data in (Ferguson, 2013), the minimum ventilation rate, SA flowrate, indoor carbon dioxide concentration, and energy consumption are calculated.

The office building has three rooms. As shown in Table 1, room1, room2, and room3 have the same floor area and a maximum of 5, 10, and 15 occupants, respectively.

Fig. 3 shows the load from TRNSYS 18 of each room, under the conditions outlined in Table 1. The latent load is always positive. The sensible load is almost positive in summer, and negative in winter. Every room has similar dimensions, has and different amounts of load. Room3, which has more occupants, has a larger load than room1.

3.2. Supply Air Volume Flow Rate

In this study, ASHRAE Standard 62.1 was used for the calculation of the minimum ventilation rate (Hedrick et al., 2015.). The minimum ventilation rate (V) is calculated by zone floor area (A_z), zone population (P_z), outdoor airflow rate required per person (R_p), and outdoor airflow rate required per unit area (R_a) (Eq. (1)).

$$V = R_p \times P_z + R_A \times A_z \tag{1}$$

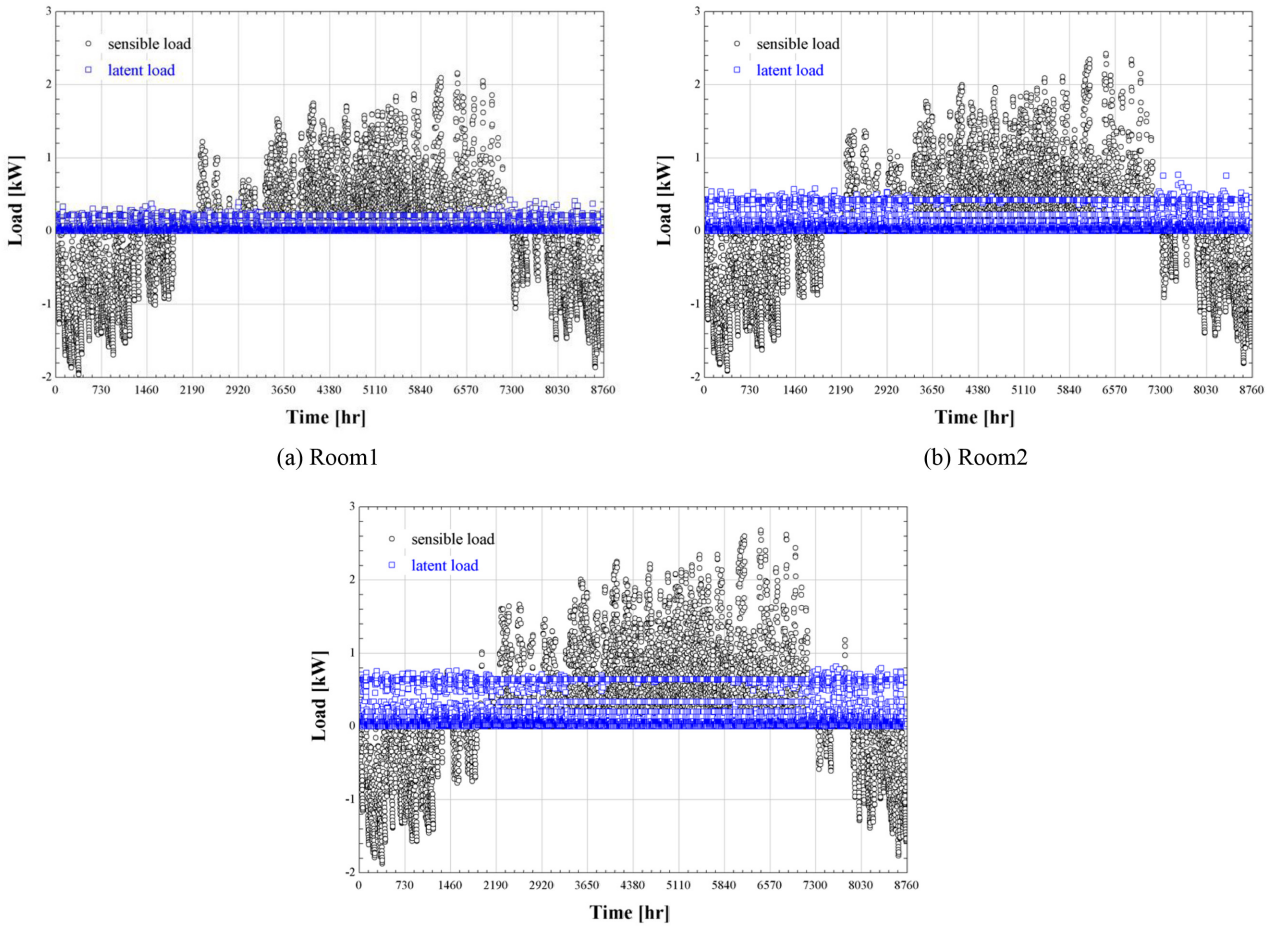
The outdoor airflow rate required per person and the outdoor airflow rate required per unit area are, respectively, 2.5 L/s·person and 0.3 L/s·m^2. The zone air distribution effectiveness (E_z) is considered to calculate the total minimum ventilation rate (V_{ot}). In this case, E_z has a value of one.

$$V_{ot} = V/E_z \tag{2}$$

In the case of multiple zones, the uncorrected ventilation

Table 1. Office building conditions

	Room1	Room2	Room3
Maximum occupants	5	10	15
Floor area	100 m^2		
Height	3 m		
Window area	South side: 10 m^2. North side: 10 m^2		
Exterior wall U-Value	0.252 W/m^2·K		
Roof U-Value	0.129 W/m^2·K		
Floor U-Value	0.297 W/m^2·K		
Window U-Value	1.4 W/m^2·K		



(a) Room1

(b) Room2

(c) Room3

Figure 3. Sensible and latent loads in each room.

flowrate (V_{ou}) is calculated as a sum of the minimum ventilation rate of each zone. The ratio of occupant (D) is considered to calculate the uncorrected ventilation flowrate.

$$V_{ou} = D \times \Sigma_{all\ zones}(R_p \times P_z) + \Sigma_{all\ zones}(R_A \times A_z) \quad (3)$$

In case of the VAV system, unlike the DOAS the system ventilation efficiency (E_v) is considered. E_v is calculated by Z which is the ratio of SA and OA and is called the primary outdoor air fraction or OA ratio. The E_v of the maximum primary outdoor air fraction value of each zone can be found in Table 2.

Table 2. System ventilation efficiency

MAX (Z)	E_v
≤ 0.15	1.0
≤ 0.25	0.9
≤ 0.35	0.8
≤ 0.45	0.7
≤ 0.55	0.6
> 0.55	Use Eqs. (4) and (5)

If the maximum primary outdoor air fraction value is greater than 0.55, use the Eqs. (4) and (5).

$$X = V_{ou} / V_{sa} \quad (4)$$

$$E_v = 1 + X - Z \quad (5)$$

Where, X is system primary outdoor air fraction and V_{sa} is total SA flowrate.

Finally, the total minimum ventilation rate is calculated by Eq. (6).

$$V_{ot} = V_{ou} / E_v \quad (6)$$

The SA flowrate of VAV and DOAS are different. Although the DOAS SA flowrate is the same as the minimum ventilation rate, the VAV SA flowrate considers the RA flowrate. Consequently, each system can have different SA flowrate.

The VAV SA flowrate can be calculated using the following energy balance equation.

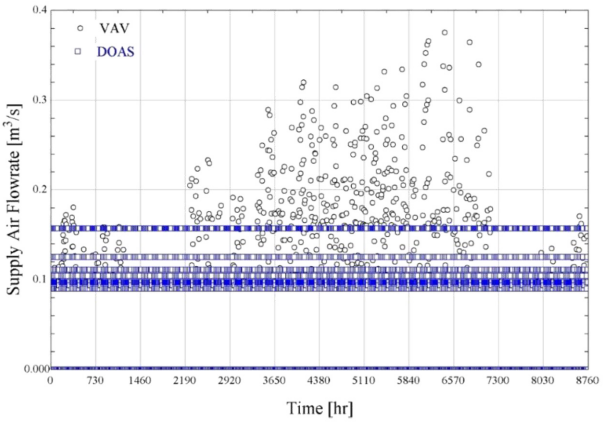


Figure 4. Supply air flowrate.

$$\dot{Q} = V_{sa} \times c_p \times (T_{sa} - T_{ra}) \quad (7)$$

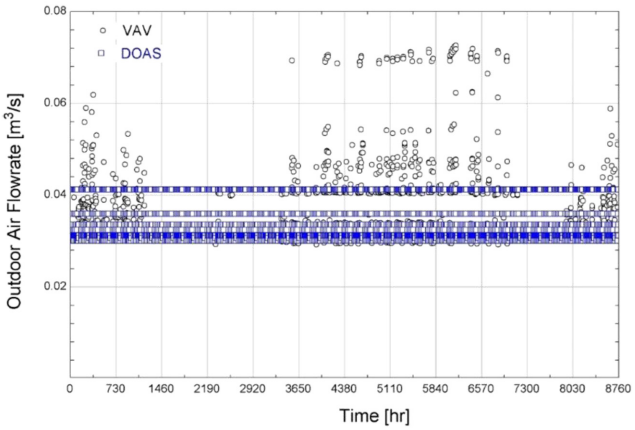
Where, $\dot{Q}$ is load in each room, c_p is specific heat of air, and T_{sa} and T_{ra} are temperature of SA and RA, respectively.

If the calculated SA flowrate is smaller than the minimum ventilation rate, the minimum ventilation rate is considered instead of the calculated SA flowrate. The RA flowrate is the difference between the SA flowrate and the minimum ventilation rate. Therefore, when supplying the minimum ventilation rate, the RA flowrate is zero.

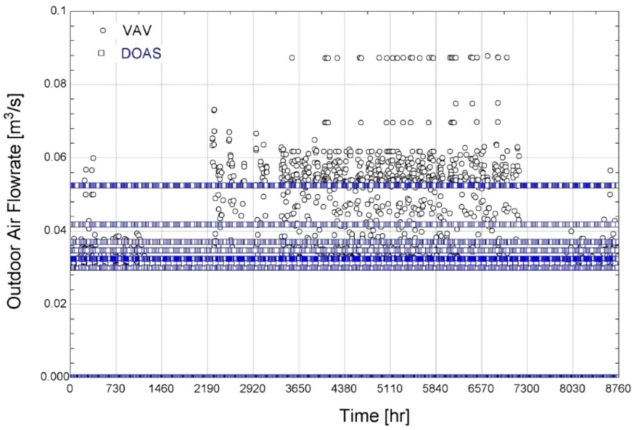
The SA flowrate is shown in Fig. 4.

Furthermore, the OA flowrate for each room is different depending on the system. Because the minimum ventilation rate is equal to the SA flowrate, the OA flowrate of each room is equal to the minimum ventilation rate in each room with the DOAS. However, in the VAV system, mixed air or minimum ventilation rates are supplied through comparing the sum of each room's SA flowrate and the minimum ventilation rate sum. In the process of distributing air to each room, the outdoor air is not properly introduced owing to the supply air having the same OA ratio, which it supplied according to SA flowrate ratio of each room.

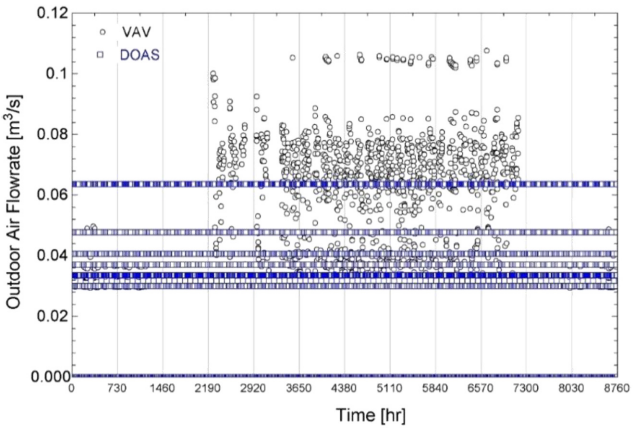
Fig. 5 shows the OA flowrate distributed in the room for each system. Almost the same OA flowrate is supplied during the intermediate and winter seasons, but it is different during summer. In room1, the VAV system supplies less



(a) Room1



(b) Room2



(c) Room3

Figure 5. Outdoor air flowrate of each room.

outdoor air than the DOAS. On the other hand, in room2 and room3, the outdoor intake is less with the DOAS than with the VAV system.

3.3. Governing Equation of Carbon Dioxide

Indoor carbon dioxide mass ($\dot{m}$) is defined as Eq. (8) (M. Meckler, 1996).

$$\dot{m} = \dot{m}_g + \dot{m}_{sa} - \dot{m}_{ra} - \dot{m}_f - \dot{m}_{abs} \tag{8}$$

Where, $\dot{m}_g$, $\dot{m}_{sa}$, $\dot{m}_{ra}$, $\dot{m}_f$ and $\dot{m}_{abs}$ are carbon dioxide mass of generation, supply air, return air, filtration and absorbed, respectively.

The calculated indoor carbon dioxide mass can be converted to ppm (C), which represents the concentration. To convert to concentration, the molecular weight of carbon dioxide and the volume and density of air should be known. The volume and density of the air should also be considered in terms of the temperature.

$$\dot{m} = \frac{C \times molecular\ weight}{air\ volume} \times \dot{V}_a \tag{9}$$

Where, $\dot{V}_a$ is the volume flowrate of air.

Some assumptions are needed to convert the mass into concentration. First, the densities of the outdoor air and exhaust air are assumed to be the same. Then, the mass of carbon dioxide removed by the filtration, soaking, and leaking is assumed to be negligible. Thus, Eq. (8) can be written as follows

$$\dot{m} = \dot{m}_g + \dot{m}_{sa} - \dot{m}_{ra} \tag{10}$$

The rate of oxygen consumption and carbon dioxide production depend on physical activity. In this study, 18,000 mL/h of carbon dioxide emissions were used at the activity level of 1.2 met because office buildings were considered.

The governing equation of carbon dioxide can be expressed in term of concentration, as well as mass. Similar to the mass balance equation, the concentration equation is a function of carbon dioxide concentration produced by people (G) and removed by filtration, changing concentration by ventilation, absorption and emission. This equation is expressed as a function of time (t).

$$V\frac{dC(t)}{dt} = G + C_{oa}\dot{V}_{sa} - C(t)\dot{V}_{abs} - C(t)\dot{V}_f \tag{11}$$

The following equation is obtained by integrating Eq. 11 in terms of time:

$$C(t) = C_0 \exp\left(-\frac{(\dot{V}_{sa} + \dot{V}_{abs} + E\dot{V}_f)t}{V}\right) + \frac{C_{oa}\dot{V}_{sa} + G}{\dot{V}_{sa} + \dot{V}_{abs} + E\dot{V}_f} \times \left\{1 - \exp\left(-\frac{(\dot{V}_{sa} + \dot{V}_{abs} + E\dot{V}_f)t}{V}\right)\right\} \tag{12}$$

Where, C_0 is initial condition of carbon dioxide concentration and E is efficiency of filtration.

Similarly to the mass balance equation, the mass of carbon dioxide removed by filtration, soaking, and leaking are assumed to be negligible. Therefore, Eq. (12) can be written as follows:

$$C(t) = C_{oa} + (C_0 - C_{oa})\exp\left(-\frac{\dot{V}_{sa}t}{V}\right) + \left(\frac{G}{\dot{V}_{sa}}\right)\left[1 - \exp\left(-\frac{\dot{V}_{sa}t}{V}\right)\right] \tag{13}$$

In this study, Eq. (13) is used to compare the indoor carbon dioxide concentration with the VAV system and DOAS.

3.4. Energy Consumption

3.4.1. Cooling and Heating Loads

The cooling and heating loads can be easily obtained for the VAV system as the supply air temperature and the relative humidity do not change. First, during the cooling mode, mixing air, which is calculated from the minimum ventilation rate and the SA flowrate, is cooled through the cooling coil to 13°C and to a relative humidity of 100%. The amount of cooling coil load (Q_c) can be obtained from Eq. (14). The heating load required to reheat the air to 15°C (Q_{rh}) can be obtained from the Eq. (15), and the heating load for reheating is constant during cooling mode.

$$\dot{Q}_c = \dot{m}_a \times \Delta h \tag{14}$$

$$\dot{Q}_{rh} = \dot{m}_a \times c_p \times \Delta T \tag{15}$$

During the heating mode, the heating load required to heat the air to 45°C (Q_h) is calculated from Eq. (16).

$$\dot{Q}_h = \dot{m} \times c_p \times \Delta T \tag{16}$$

In this process, it is assumed that the cooling and heating coils are not activated at cooling mode in winter and heating mode in summer. Therefore, the cooling load in winter and heating load in summer are not considered in this simulation.

The calculation of the load in the DOAS is performed by calculating the condition of the air passing through the enthalpy wheel. This condition can be obtained from the Eqs. (17) and (18) (Jeong and Mumma, 2006). The enthalpy wheel is operated only in summer and winter.

$$\eta = \frac{T_{oa} - T_{ew,o}}{T_{oa} - T_{ra}} \tag{17}$$

$$\eta = \frac{w_{oa} - w_{ew,o}}{w_{oa} - w_{ra}} \tag{18}$$

Where, η is the efficiency of the enthalpy wheel; 0.8. T and w are temperature and humidity ratio.

The humidity ratio of supply air (w_{sa}) during the cooling mode is then calculated.

$$\dot{q}_{lat} = 0.3 \times V_{oa} \times (w_{ra} - w_{sa}) \tag{19}$$

Where, $\dot{q}_{lat}$ is latent load of room.

The cooling coil is operated when the humidity ratio of

the outdoor air is higher than the humidity ratio of the supply air. The cooling coil load ($\dot{q}_{cc}$) is calculated by Eqs. (20)-(22).

$$\dot{q}_{lat,cc}=0.3\times V_{oa}\times(w_{c,i}-w_{c,o}) \tag{20}$$

$$\dot{q}_{sen,cc}=\dot{m}_{a}\times c_{p}\times(T_{c,i}-T_{c,o}) \tag{21}$$

$$\dot{q}_{cc}=\dot{q}_{lat,cc}+\dot{q}_{sen,cc} \tag{22}$$

Given that a certain amount of sensible load is removed by the cooling coil for dehumidification, the removed sensible load ($\dot{q}_{DOAS,SA,sen}$) is calculated using Eq. (23):

$$\dot{q}_{DOAS,SA,sen}=\dot{m}\times c_{p}\times(T_{ra}-T_{sa}) \tag{23}$$

If the removed sensible load is less than the sensible load of the room, the remaining sensible load ($\dot{q}_{sen,parallel}$) should be removed by the ceiling radiant cooling panel. If the cooling coil is not operated, the ceiling radiant cooling panel is operated depending on the sensible load ($\dot{q}_{sen}$) of the room.

$$\dot{q}_{sen,parallel}=\dot{q}_{sen}-\dot{q}_{DOAS,SA,sen} \tag{24}$$

The cooling load ($\dot{q}_{c}$) during the cooling mode can be calculated by Eq. (25).

$$\dot{q}_{c}=\dot{q}_{cc}+\dot{q}_{sen,parallel} \tag{25}$$

During the heating mode, the heating coil load ($\dot{q}_{hc}$) is determined by the air passing through the enthalpy wheel heated to neutral temperature ($T_{hc,o}$).

$$\dot{q}_{hc}=\dot{m}\times c_{p}\times(T_{hc,o}-T_{hc,i}) \tag{26}$$

In this case, a parallel heating unit is needed to remove the sensible load as the heating coil cannot completely remove sensible load. The parallel heating unit load ($\dot{q}_{h,sen}$) is equal to the sensible load of the room, and the total heating load ($\dot{q}_{h}$) is the sum of the heating coil load and the parallel heating unit load.

$$\dot{q}_{h}=\dot{q}_{hc}+\dot{q}_{h,sen} \tag{27}$$

3.4.2. Chiller Energy

The chiller is a device that supplies cold water to the cooling coil and the ceiling radiant cooling panels. Cooling coils cool the air through heat exchange between water and air, thus cold water must be continuously supplied to cool the air. The chiller energy (P_c) can be calculated by Eqs. (28)-(32) (Jeong et al., 2003.).

$$CAPET=f(T_{cw,l},T_{cond,e}) \tag{28}$$

$$EIRFT=f(T_{cw,l},T_{cond,e}) \tag{29}$$

$$EIRFPLR=f(PLR) \tag{30}$$

$$PLR=\frac{\dot{q}_{c}}{\dot{Q}_{ref}\times CAPFT} \tag{31}$$

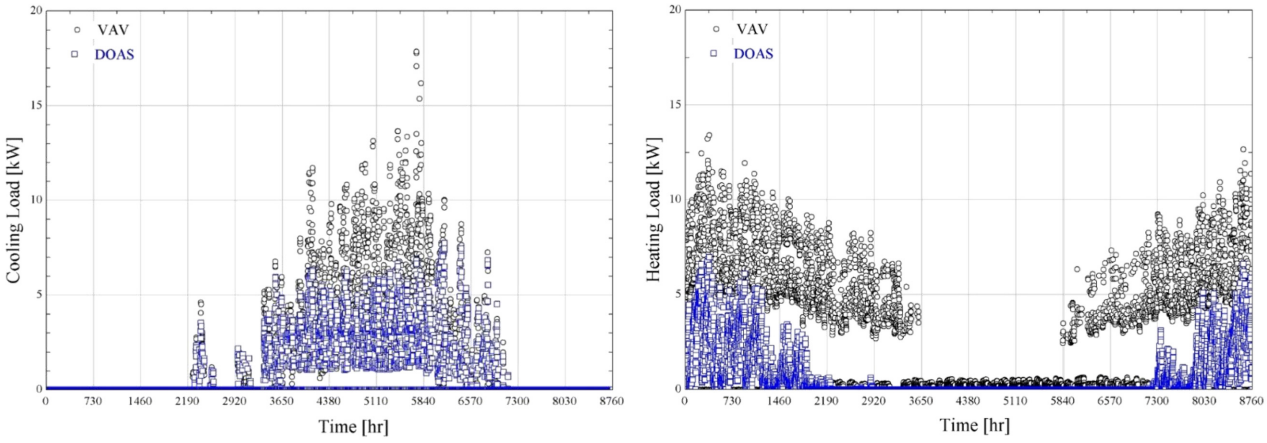
$$P_{c}=P_{ref}\times CAPET\times EIRFT\times EIRFPLR \tag{32}$$

Where, *CAPET* is cooling capacity factor, *EIRFT* is full load efficiency, *EIRFPLR* is partial load efficiency of a chiller, and *PLR* is part load ratio. Q_{ref} is rated cooling capacity and P_{ref} is input power of model.

In this study, different chiller models were used owing to the two systems having a large difference of maximum cooling load. The chiller model used are outlined in Table 3.

3.4.3. Pump Energy

The pump energy (P_{pump}) is needed to supply the water from the chiller to the cooling coil and the ceiling radiant cooling panel. Therefore, the pump energy is expressed as a function of the water flowrate (Lim and Jeong, 2018).



(a) Cooling load (b) Heating coil

Figure 6. Cooling and heating loads.

Table 3. Chiller model of each system

	DOAS	VAV
MAX($\dot{q}_c$) [kW]	7.75	17.87
Model	HLLA 03SI	HLLA 08SI
P_{ref} [kW]	3.85	8.15
Q_{ref} [kW]	9.2	24.2

Table 4. Total head of each system

Total head [m]	DOAS	VAV
Cooling coil	20	20
Ceiling radiant cooling panel	7	-

Table 5. Pressure drop of each system

ΔP [Pa]		DOAS	VAV
SA Fan	Enthalpy wheel	120	-
	Cooling coil	100	100
	Heating coil	50	100
	Duct	10	10
	System balance	-	200
RA Fan	Enthalpy wheel	120	-
	Duct	10	10
	System balance	-	200

$$P_{pump} = \frac{\dot{m}_w \times H \times g}{\eta_{pump}} \tag{33}$$

Where, η_{pump} is pump efficiency; 0.8. H is total head and g is gravity acceleration.

The water mass flowrate ($\dot{m}_w$) of each system is calculated for maximum cooling load, by dividing the load by the specific heat and the temperature difference based on the energy balance equation. The difference between the temperature (ΔT) of the water flowing in and out of the chiller is set to about 5°C. The flowrate was set to the safety factor (*Eff*) of 80%.

$$\dot{m}_w = \frac{MAX(\dot{q}_c)}{c_{p,w} \times \Delta T \times Eff} \tag{34}$$

Where, $c_{p,w}$ is specific heat of water.

Table 4 shows the total head for the pump energy calculation.

3.4.4. Fan Energy

Air travels through the ducts, which requires energy to move. Energy is consumed in the form of fan rotation and the energy consumption (P_{fan}) can be calculated from the supply and exhaust fans (Kim et al., 2016).

$$P_{fan,design} = \frac{V_{design} \times \Delta P}{\eta_{fan}} \tag{35}$$

$$P_{fan} = P_{fan,design} \times (0.0013 + 0.1470PLR + 0.9506PLR^2 - 0.0998PLR^2) \tag{36}$$

The part load ratio (*PLR*) is variable in the VAV system because the SA flowrate changes frequently. However, even if it operates with a part load, it will still consume some energy. Therefore, if the part load ratio is less than 0.25, it will still be assumed to operate at 0.25.

In both systems, the maximum flow speed was set to 3 m/s and the duct diameter was set to the maximum flowrate divided by the flow speed, which were 0.28 m and 0.43 m in the DOAS and the VAV system, respectively. The friction loss is found from the duct diameter. Assuming a duct length of 200 ft, the pressure drops are 10 Pa in both the DOAS and VAV system (McQuiston et al., 2005).

The pressure drop for each component is shown in Table 5. In the DOAS, the pressure drops used in the supply fan of the enthalpy wheel, cooling coil, heating coil, and duct are, respectively, 120 Pa, 100 Pa, 50 Pa, and 10 Pa, and the pressure drops used in the exhaust fan of the enthalpy wheel and the duct are, respectively, 120 Pa and 10 Pa. In VAV System, the pressure drops used in the supply fan of the cooling coil and heating coil are 100 Pa and the pressure drops of the duct and system balance are, respectively, 10 Pa, and 200 Pa, and the pressure drops used in the exhaust fan are 10 Pa on the duct and 200 Pa for the system balance.

3.4.5. Heating Coil and Parallel Heating Energy

When there is a sensible load in the VAV system, the heating coil is activated for heating and reheating. Therefore, the heating energy is only calculated from the heating coil energy for heating and reheating. On the other hand, in the DOAS system, the heating coil consumes the energy for heating from neutral temperature, and the parallel heating unit consumes the energy for treating the remaining sensible load when there is a sensible heating load. If there is a sensible cooling load, the heating coil consumes the energy for reheating similarly to the VAV system. the heating energy was calculated as the sum of the heating coil energy and the parallel heating unit.

3.4.6. Primary Energy

To compare the overall energy performance, the energies calculated for each component were converted into the primary energy by multiplying the local primary energy factor. The local primary energy factors were 2.75 for electricity and 1.1 for natural gas. In this study, the chiller, pump, fan, and parallel heating unit in the DOAS use electricity and the heating coil uses natural gas.

4. Simulation Results

4.1. Indoor Carbon Dioxide Concentration

In this study, indoor carbon dioxide concentration was used as an index to compare the indoor air quality. The indoor carbon dioxide concentration was compared between the rooms for system operation in each season, that is, the intermediate, winter, and summer. Fig. 7-9 show the indoor carbon dioxide concentration for a week of each season.

Fig. 7 shows the indoor carbon dioxide concentration in

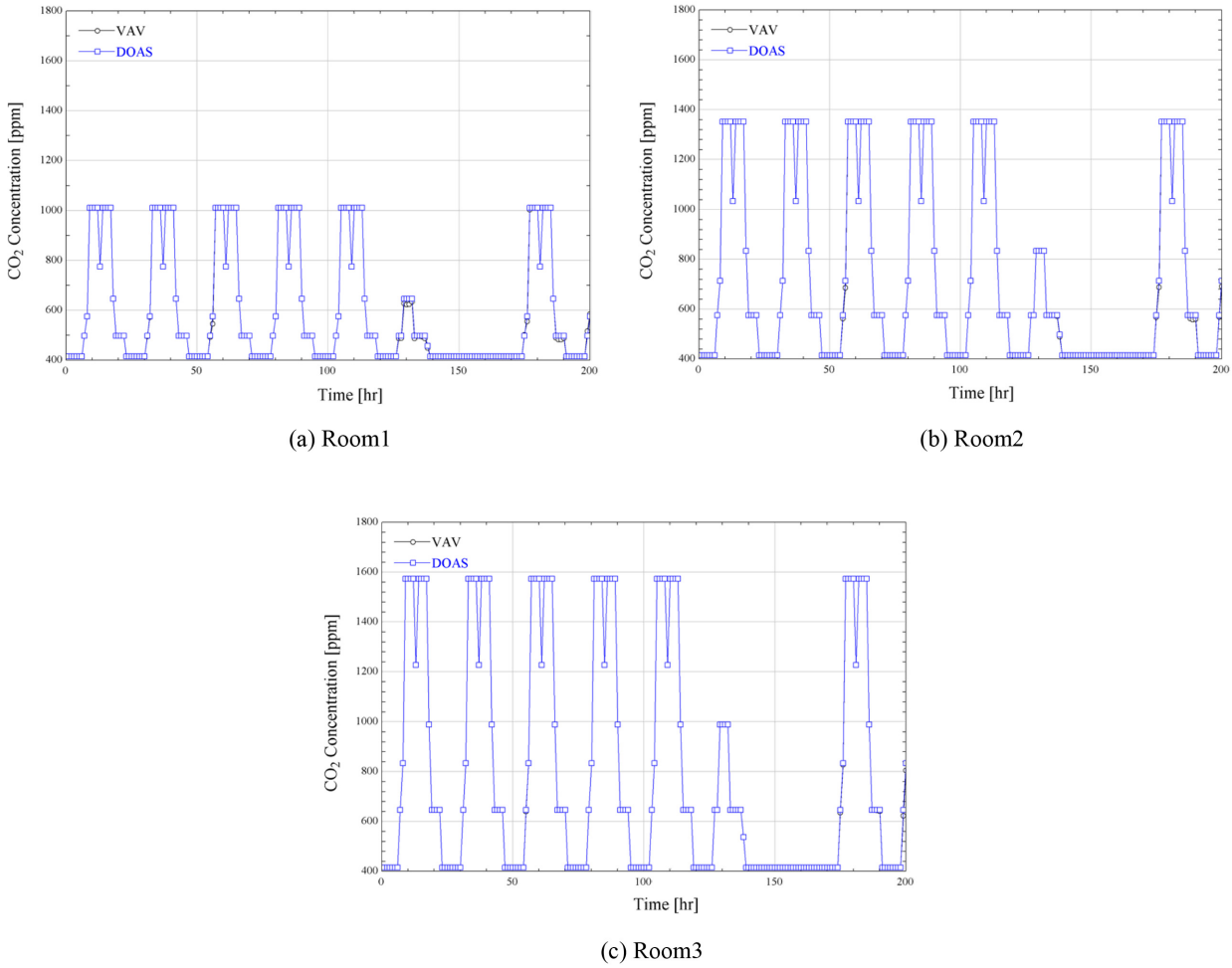


Figure 7. Indoor carbon dioxide concentration in the intermediate season.

the intermediate season. In room1, which has a 100 m^2 floor area and a maximum of 5 occupants, the maximum carbon dioxide concentration is constant at 1000 ppm in both systems. The concentrations are about 1380 ppm in room2, which has the same floor area and a maximum of 10 occupants, and about 1590 ppm in room3, which has the same floor are and a maximum of 15 occupants.

The amount of outdoor air intake into each room is almost the same in the intermediate season. Therefore, the indoor carbon dioxide concentration does not change in the two systems.

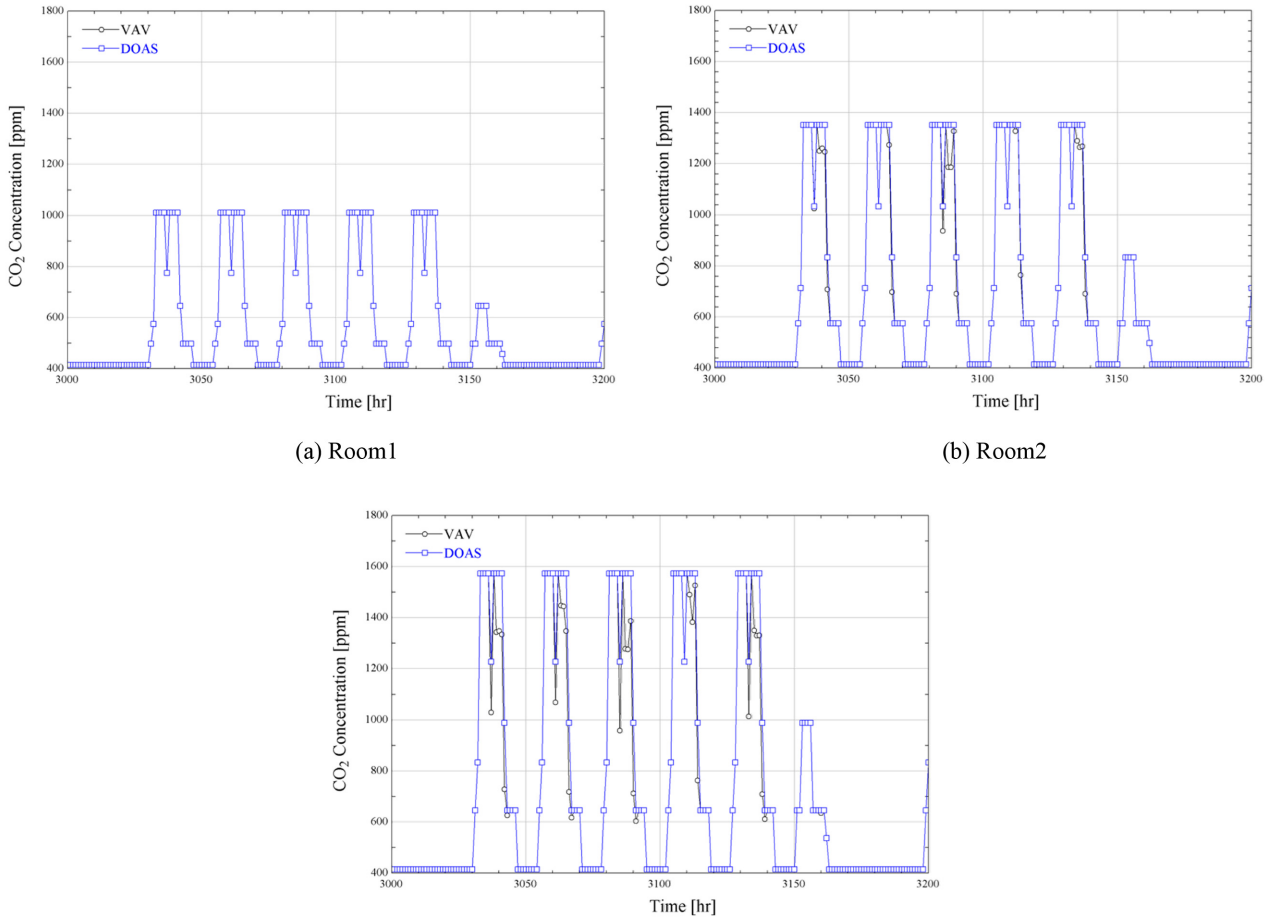
Fig. 8 shows the indoor carbon dioxide concentration in winter. Room1, room2, and room3 have maximum concentrations of, respectively, 1000 ppm, 1380 ppm, and 1590 ppm, which is similar to the intermediate season. Given that the amount of outdoor air intake into each room is almost the same in winter and the intermediate seasons, the indoor carbon dioxide concentration is similar in the two systems.

On the other hand, the results are different for both systems in summer.

The results for summer are the same as in winter and the intermediate season with the DOAS but are different with the VAV system. The SA flowrate in summer is much greater than that in winter and the intermediate seasons. This means that the outdoor air is not supplied to each room as needed. Therefore, the results shown in Fig. 9 are obtained. In room1, the indoor carbon dioxide concentration with the DOAS is about 150 ppm less than that with the VAV system. On the other hand, in room2 and room3, the concentration with the DOAS is maximum 50 ppm higher than with the VAV system. That is, some rooms are supplied much more or less SA flowrate.

4.2. Annual Energy Consumption

The primary energy of the chiller, pump, fan, heating coil, and parallel heating unit is shown in Fig. 10. The DOAS consumes about 35.78% less chiller energy than the VAV and about 74.31% less pump energy than the VAV. Especially, pump energy changes depending on the water flowrate, that is, because the water flowrate in DOAS is much less than in VAV, the former consumes less pump



(a) Room1

(b) Room2

(c) Room3

Figure 8. Indoor carbon dioxide concentration in winter.

energy than the latter. On the other hand, fan energy consumption is almost the same in the two systems. Fan energy changes depending on the air flowrate, but considering the part load ratio, a similar result can be obtained. Although the DOAS has much less air flowrate than VAV, the former consumes about 8.77% less fan energy than the latter due to the part load ratio of air. The heating energy includes the heating coil energy and the parallel heating unit energy. The DOAS uses the heating coil and the parallel heating unit, and the VAV uses the heating coil. The DOAS consumes about 72.81% less heating energy than VAV, even though the VAV uses only the heating coil. Finally, it was found that the DOAS consumes about 40.43% less total primary energy than VAV.

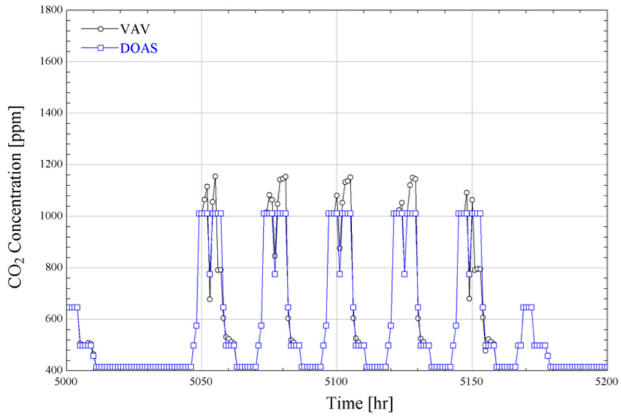
5. Conclusions

In this study, DOAS and VAV systems operating in office buildings are compared in terms of indoor carbon dioxide concentration and energy consumption. For the simulation, the load in each room was estimated by TRNSYS 18. The indoor carbon dioxide concentration was calculated from the governing equation, and the energy consumption was calculated by analyzing each component separately. The simulation results are the following.

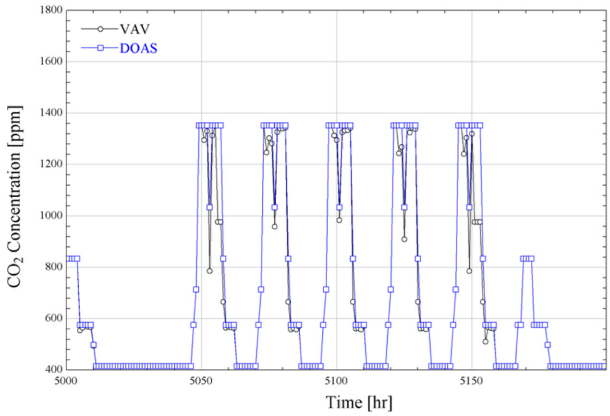
In winter and the intermediate seasons, the indoor carbon dioxide concentration is almost the same in every room for both systems.

In summer, the indoor carbon dioxide concentration depends on the number of indoor occupants. Furthermore, in summer, the DOAS performs similarly to winter and the intermediate seasons. However, when the VAV operates in summer the carbon dioxide concentration is higher in room1, lower in room2 and room3 compared to results with the DOAS. This means that the outdoor air cannot be stably supplied, which is due to the OA ratio of each room not being constant when the SA flowrate is greater in summer.

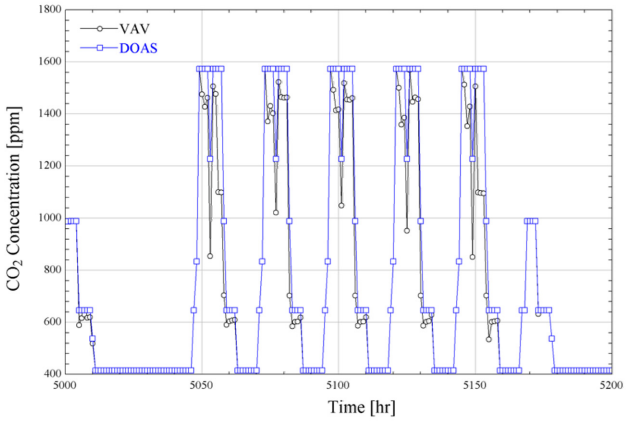
In terms of energy consumption, chiller, pump, fan, and heating energy are less in the DOAS than in the VAV. Every component of the DOAS consumes less energy, and about 40.43% less primary energy than the VAV.



(a) Room1



(b) Room2



(c) Room3

Figure 9. Indoor carbon dioxide concentration in summer.

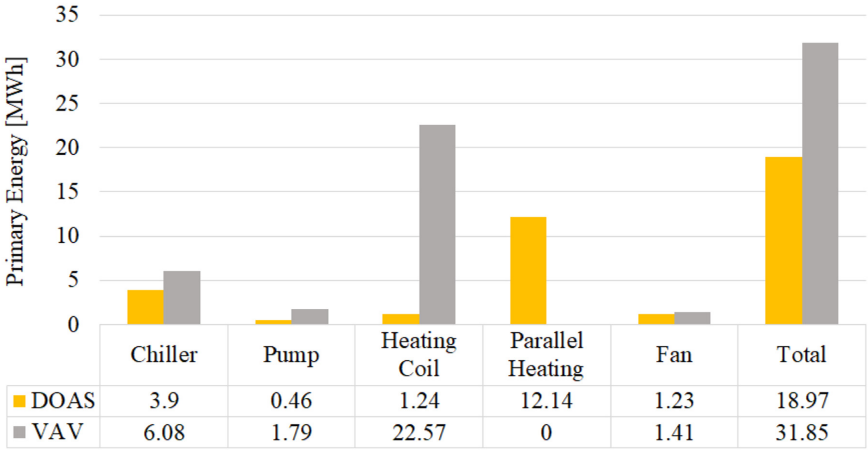


Figure 10. Annual primary energy consumption.

DOAS has many advantages compared to VAV in multiple zones office building because it consumes the less energy while more stably supplying outdoor air.

Finally, this study evaluated the compatibility of the systems with multiple zone office buildings via indoor air quality and energy consumption. As a result, it was concluded that the DOAS is more suitable for multiple zone office buildings.

Acknowledgements

This work was supported by the Korea Agency for Infrastructure Technology Advancement (KAIA) grants (19CTAP-C141826-02).

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