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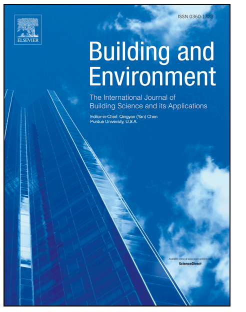

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Optimization on Fresh Outdoor Air Ratio of Air Conditioning System with Stratum Ventilation for Both Targeted Indoor Air Quality and Maximal Energy Saving

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Abstract

Stratum ventilation can energy efficiently provide good inhaled indoor air quality with a proper operation (e.g., fresh outdoor air ratio). However, the non-uniform CO_2 distribution in a stratum-ventilated room challenges the provision of targeted indoor air quality. This study proposes an optimization on the fresh outdoor air ratio of stratum ventilation for both the targeted indoor air quality and maximal energy saving. A model of CO_2 concentration in the breathing zone is developed by coupling CO_2 removal efficiency in the breathing zone and mass conservation laws. With the developed model, the ventilation parameters corresponding to different fresh outdoor air ratios are quantified to achieve the targeted indoor air quality (i.e., targeted CO_2 concentration in the breathing zone). Using the fresh outdoor air ratios and corresponding ventilation parameters as inputs, energy performance evaluations of the air conditioning system are conducted by building energy simulations. The fresh outdoor air ratio with the minimal energy consumption is determined as the optimal one. Experiments show that the mean absolute error of the developed model of CO_2 concentration in the breathing zone is 1.9%. The effectiveness of the proposed optimization is demonstrated using TRNSYS that the energy consumption of the air conditioning system with stratum ventilation is reduced by 6.4% while achieving the targeted indoor air quality. The proposed optimization is also promising for other ventilation modes for targeted indoor air quality and improved energy efficiency.

Keywords: Fresh outdoor air ratio; Targeted indoor air quality; Energy saving; CO_2 removal efficiency; Stratum ventilation

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Nomenclature

a, b, c	constant coefficients	Q_c	rated chiller capacity (kW)
ACH	air changes per hour	Q_{cl}	room cooling load (kW)
c_p	specific heat capacity of air (1.004 kJ/(kg•^{0}C))	Q_{cl}'	chiller cooling load (kW)
C_e	exit CO_2 concentration (ppm)	Q_f	proposed fresh airflow rate (ACH)
C_o	outdoor CO_2 concentration (400 ppm)	$Q_{f,c}$	conventional fresh airflow rate (ACH)
C_r	CO_2 concentration in breathing zone (ppm)	r_1	capacity ratio
C_s	supply CO_2 concentration (ppm)	r_2	COP ratio
COP	coefficient of performance (5)	T_e	exit air temperature (^{0}C)
EC	total energy consumption (kJ/hr)	T_{icw}	inlet cooling water temperature (^{0}C)
EC_{chi}	chiller energy consumption (kJ/hr)	T_{ocw}	outlet chilled water temperature (^{0}C)
EC_{fan}	fan energy consumption (kJ/hr)	T_r	room air temperature (^{0}C)
EC_{pump}	pump energy consumption (kJ/hr)	T_s	supply air temperature (^{0}C)
$FFLP$	fraction of full load power	V	room volume (m^3)
G	CO_2 generation rate (m^3/s)	η_{fan}	fan efficiency (70%)
k	number of room occupants	η_{pump}	pump efficiency (60%)
j	jth case	ρ_{air}	air density (1.2 kg/m^3)
$\dot{m}_{wat}$	water flow rate (kg/s)	ρ_{CO_2}	CO_2 gas density (kg/m^3)
MAE	mean absolute error (%)	ρ_{wat}	water density (1000 kg/m^3)
n	number of cases	Δh	head of water flow (m)

N	supply airflow rate (ACH)	ΔP_{air}	total pressure (Pa)
PLR	part load ratio	γ	fresh outdoor air ratio (%)
PMV	Predicted Mean Vote	ε	CO_2 removal efficiency in breathing zone

1. Introduction

Modern people spend most of their time (80%-90%) in indoor environments [1, 2]. Indoor air quality significantly affects occupants' health and productivity [3, 4]. For quality indoor air, the fresh outdoor air is conditioned and supplied into the indoor environment by the air conditioning system, so that the indoor air pollutants can be diluted. However, conditioning the fresh outdoor air consumes more energy when its enthalpy is higher than that of indoor air, particularly in subtropical/tropical and humid regions [5-7]. Besides indoor air quality, improving the energy efficiency of the air conditioning system is also a common concern, since the air conditioning system accounts for the significant part (around 50%) of the building energy consumption [8, 9]. Taking into consideration both indoor air quality and energy efficiency of the air conditioning system, a part of the indoor air is recirculated in practice. The recirculated indoor air and fresh outdoor air constitute the air supplied into the indoor environment. The airflow rate of the fresh outdoor air is required to satisfy indoor air quality, and the total air supply, which determines the air movement, to the indoor environment is required to meet the requirement of thermal comfort [10-12]. The ratio of the fresh outdoor airflow rate to the total supply airflow rate is defined as the fresh outdoor air ratio [13, 14]. Intuitively, increasing the fresh outdoor air ratio might improve the indoor air quality but deteriorate the system energy efficiency. Thus, the fresh outdoor air ratio needs to be optimized to provide satisfactory indoor air quality and minimize the energy consumption of the air conditioning system simultaneously.

The air conditioning system with stratum ventilation can energy efficiently create a health and thermally comfortable indoor environment [15-18]. It horizontally supplies conditioned air into the breathing zone, thereby resulting in lower concentration of air pollutants in the breathing zone than that of mixing ventilation [19, 20]. The air distribution pattern of stratum ventilation is a “sandwich”, with the lowest air temperature and the highest air velocity at the head level [21, 22]. The head is the most sensitive part of the body regarding thermal comfort, so that stratum ventilation can efficiently cool the body and provide thermal comfort [23]. Stratum ventilation has been experimentally verified to provide a uniformly comfortable thermal environment for a group of occupants [17]. Air movement is preferred at thermally neutral and slightly warm thermal environments [22], and is recommended to be not greater than 0.8 m/s [24]. With satisfactory indoor air quality and thermal comfort, stratum ventilation can reduce the annual energy consumption of the air conditioning system by at least 37.7% as compared with mixing ventilation [25]. Stratum ventilation is designed for small-to-medium sized rooms [24], and has been experimentally confirmed to be able to provide a uniform thermal environment for occupants in multiple rows [17, 26, 27].

To achieve high performance in practice, stratum ventilation is required to be carefully operated [16, 22]. However, few existing studies on the operation optimization of stratum ventilation are available [25, 28]. Zhang et al. [12] proposed an operation optimization of the supply air temperature and supply airflow rate for stratum ventilation to achieve the targeted thermal condition and save energy. For this operation optimization, the fresh outdoor air ratio is determined to introduce a constant amount of the fresh outdoor air according to the minimal requirement of ASHRAE Standard 62 [25, 29]. ASHRAE Standard 62 stipulates the minimal fresh outdoor airflow rate of 10 L/s per person to account for both human bio-effluents and contaminants related to building materials and furnishings [11, 25 30]. However, there are no justifications for the stipulated minimal amount of the fresh outdoor air whether it is adequate for indoor air quality [31]. An appropriate operation with a variable amount of the fresh outdoor air can generally provide more energy efficient indoor air quality compared with that of the constant fresh outdoor air. The appropriate operation with a variable amount of the fresh outdoor air has been reported to save energy up to 60% by the studies on different air conditioning systems under different outdoor weather conditions

[31, 32]. Thus, the operation optimization on the fresh outdoor air ratio of stratum ventilation is needed for energy efficiency improvement while providing satisfactory indoor air quality.

Quantifying the fresh outdoor air ratio based on demand-controlled ventilation is getting more and more attentions from researchers and policymakers for both indoor air quality and energy saving [31]. Demand-controlled ventilation adjusts the fresh outdoor air ratio as a response to the indoor pollutant load [33, 34]. The indoor pollutant load is indicated by the concentration of a pre-defined surrogate of indoor air quality, e.g., CO_2, humidity and radon [35, 36]. The demand-controlled ventilation methods essentially control the fresh outdoor airflow rate to achieve the targeted indoor CO_2 concentration when CO_2 is used as the surrogate of indoor air quality [37, 38]. CO_2 is a widely used surrogate of indoor air quality [32, 39]. Although indoor CO_2 concentration can be up to 10,000 ppm without severe health damage on occupants, a high indoor CO_2 concentration indicates that the fresh airflow rate is inadequate to dilute indoor pollutants [39, 40]. For acceptable indoor air quality, indoor CO_2 concentration can be below 1000 ppm or 650 ppm above that of outdoor air [40, 41].

The accurate measurement/prediction of indoor CO_2 concentration is the core of the demand-controlled ventilation [34, 42]. Based on the methods of obtaining the indoor CO_2 concentration, demand-controlled ventilation is classified into sensor-based and model-based types [31, 43]. However, the non-uniform distribution of indoor CO_2 under stratum ventilation challenges both types of the demand-controlled ventilation [44]. Experiments showed that under stratum ventilation, the ratio of CO_2 concentration difference between the exit air and supply air to that between air in breathing zone and supply air could reach 1.76, indicating a substantial non-uniformity of indoor CO_2 distribution [45]. The CO_2 concentration in the breathing zone is most critical to the inhaled air quality. However, the existing model-based demand-controlled ventilation is developed for an indoor environment with a uniform indoor CO_2 distribution, and cannot accurately predict the non-uniform indoor CO_2 distribution under stratum ventilation [46]. As a result, implementing the existing model-based demand-controlled ventilation for stratum ventilation would fail to maintain the targeted CO_2 concentration in the breathing zone thereby deteriorating the inhaled air quality [47]. For the sensor-based demand-controlled ventilation, the non-uniform distribution of indoor CO_2 requires multiple sensors to be installed in the breathing zone. The multiple-sensors in breathing zone firstly increase the initial cost and operation complexity, and secondly obstruct the space use of the occupants [48]. Practically, the non-uniform distribution of the indoor CO_2 concentration is commonly encountered under different ventilation modes, e.g., mixing ventilation, displacement ventilation and task/ambient air conditioning system [49-52]. Since the existing demand-controlled ventilation methods ignore the non-uniformity of indoor CO_2 concentration distribution, the field tests in school and office buildings have reported that the existing demand-controlled ventilation methods failed to provide the targeted indoor air quality [47]. Moreover, the existing demand-controlled ventilation methods harvest the energy saving mainly from the variation of the number of occupants [31, 53]. When the number of occupants decreases, the fresh outdoor airflow rate can be decreased to save energy [53, 54]. However, when the number of occupants is fixed, by properly determining the supply air parameters, the indoor air distribution has potential to be improved to more energy-efficiently provide indoor air quality [45, 47, 51]. Since the existing demand-controlled ventilation methods ignore the effects of supply air parameters on the indoor air distribution, they are unable to capture the associated energy saving.

This study proposes an optimization of the fresh outdoor air ratio for stratum ventilation to minimize the energy consumption of the air conditioning system and to achieve the targeted indoor air quality (i.e., targeted CO_2 concentration in the breathing zone) simultaneously. The proposed optimization belongs to the model-based demand-controlled ventilation category. CO_2 is used as the surrogate of indoor air quality. A model of CO_2 concentration in breathing zone is developed and experimentally validated. The proposed optimization will be elaborated in Section 2, and its effectiveness will be demonstrated in Section 3. By accounting for the non-uniform distribution of indoor CO_2 concentration, the proposed method mainly has two advantages over the exiting demand-controlled ventilation methods. Firstly, the proposed method can control the CO_2 concentration in the breathing zone at the targeted value using the developed model, while the existing demand-controlled ventilation methods would fail to fulfill the targeted CO_2 concentration in the breathing zone. Secondly, the proposed method can properly determine the supply air parameters to improve indoor air distribution to more energy-efficiently provide indoor air quality, while the existing demand-controlled ventilation methods cannot. The proposed method can be conveniently implemented in practice. This is because the developed model mainly requires the inputs of the supply airflow rate and fresh outdoor air ratio which can be obtained from the building management system. Thus, the proposed method contributes to the proper operation of stratum ventilation to achieve both the targeted indoor air quality and maximal energy saving. Moreover, the proposed method is also promising for other ventilation modes with non-uniformly distributed indoor CO_2 concentration as discussed in Section 4.

2. Methodology

2.1 Overview of proposed optimization on fresh outdoor air ratio

As shown in Figure 1, the proposed optimization on the fresh outdoor air ratio of stratum ventilation mainly includes two issues: (1) modeling CO_2 concentration in breathing zone and (2) energy performance evaluations of the air conditioning system with different fresh outdoor air ratios. For the first issue, following the law of mass conservation, under steady states, the mass sum of the CO_2 entering the room in the supply air and the CO_2 generated indoors is equal to the mass of CO_2 removed by the exit air (Equation 1); and the CO_2 concentration in the supply air equals the CO_2 concentration in the mixture of the recirculated air and fresh outdoor air (Equation 2) (Figure 2) [29, 43]. CO_2 removal efficiency in breathing zone is used to correlate CO_2 concentrations in the breathing zone, supply air and exit air (Equation 3). Therefore, CO_2 concentration in breathing zone is quantified by the fresh outdoor air ratio, outdoor CO_2 concentration, supply airflow rate, indoor CO_2 generation rate and CO_2 removal efficiency in breathing zone (Equation 4) (i.e., f_1 in Figure 1). Then the CO_2 removal efficiency in breathing zone is correlated to the supply airflow rate based on experiments (i.e., f_2 in Figure 1). The CO_2 removal efficiency in the breathing zone might be affected by the supply airflow rate, supply air temperature and exit air temperature [45].

However, the preliminary analyses show that only the supply airflow rate is statistically significant to the CO_2 removal efficiency in the breathing zone (Section 3.1). The correlation between the CO_2 removal efficiency in the breathing zone and the supply airflow rate will be validated by the experiments (Section 3.1). In this study, the breathing zone refers to the zone between the heights of 0.9 m and 1.3 m above the floor for seated occupants [45]. However, when other special zones are concerned, the proposed method is also applicable by replacing the CO_2 concentration in the breathing zone (Equations 3 and 4) with the CO_2 concentration of the specifically concerned zone (e.g., the occupied zone as discussed in Section 4).

$$\frac{\rho_{\mathrm{CO_2}} N V C_{\mathrm{s}}}{3600} + \frac{\rho_{\mathrm{CO_2}} G}{10^6} - \frac{\rho_{\mathrm{CO_2}} N V C_{\mathrm{e}}}{3600} = 0 \tag{1}$$

$$C_{\mathrm{s}} = (1-\gamma) C_{\mathrm{e}} + \gamma C_{\mathrm{o}} \tag{2}$$

$$\varepsilon = \frac{C_{\mathrm{e}} - C_{\mathrm{s}}}{C_{\mathrm{r}} - C_{\mathrm{s}}} \tag{3}$$

$$C_{\mathrm{r}} = C_{\mathrm{o}} + \frac{3600G}{NV\varepsilon} + \frac{3600G(1-\gamma)}{NV\gamma} \tag{4}$$

where C_{e}, C_{o}, C_{r} and C_{s} are CO_2 concentrations in exit air, outdoor air, breathing zone and supply air respectively (ppm); G is the indoor CO_2 generation rate which is determined by the number of occupants [50] (m^3/s); N is the supply airflow rate (ACH); V is the volume of indoor environment (m^3); $\rho_{\mathrm{CO_2}}$ is the CO_2 gas density (kg/m^3) and is regarded as constant for the indoor environment; γ is the fresh outdoor air ratio (%); ε is CO_2 removal efficiency in the breathing zone.

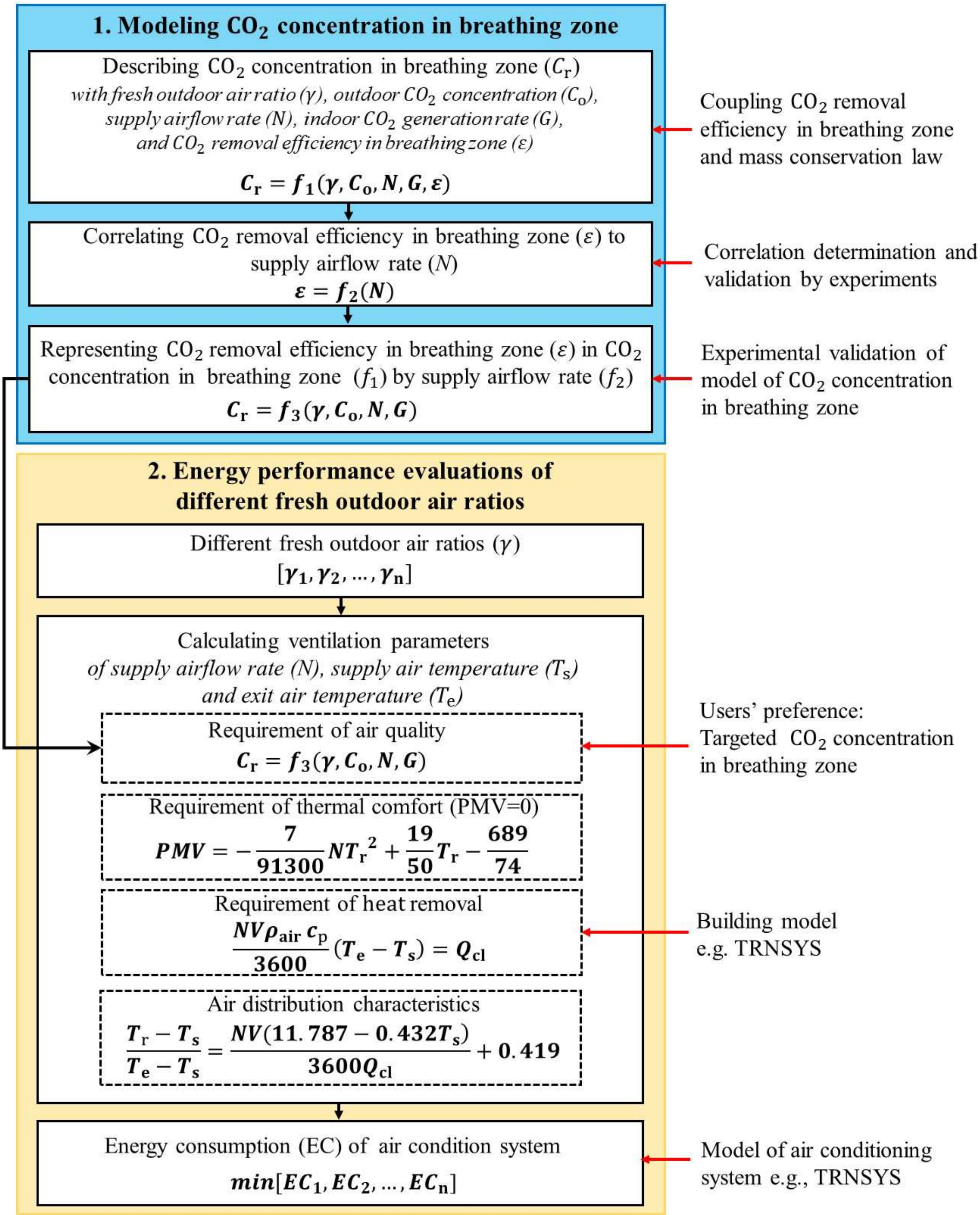


Fig.1. Flowchart of proposed optimization on fresh outdoor air ratio for stratum ventilation.

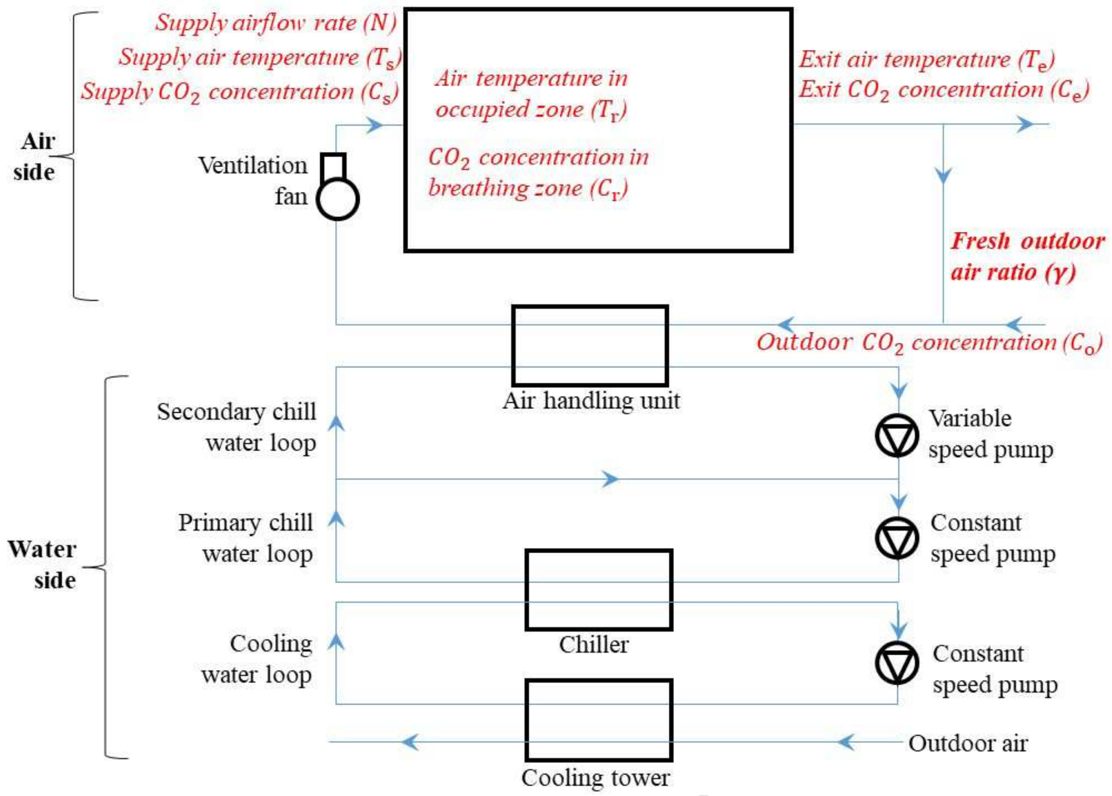


Fig.2. Schematic diagram of air side and water side of air conditioning system.

Based on the validated correlation (i.e., f_2 in Figure 1), the CO_2 removal efficiency in the breathing zone can be replaced by the supply airflow rate for quantifying the CO_2 concentration in the breathing zone (i.e., f_1 in Figure 1). As a result, the CO_2 concentration in the breathing zone is modeled as a function of the fresh outdoor air ratio, outdoor CO_2 concentration, supply airflow rate, indoor CO_2 generation rate (i.e., f_3 in Figure 1). The outdoor CO_2 concentration can be monitored by the building management system, or assumed to be constant since its variation is relatively small (e.g., the outdoor CO_2 concentration in Hong Kong can be assumed to be 360 ppm) [39, 55]. The supply airflow rate can also be monitored by the building management system [28]. The indoor CO_2 generation rate can be calculated from the indoor occupancy [56, 57]. Thus, the CO_2 concentration in the breathing zone can be determined by the fresh outdoor air ratio. The model of CO_2 concentration in breathing zone is directly derived from the conservation of mass law and the experimentally validated correlation of the CO_2 removal efficiency in the breathing zone, thus the model of CO_2 concentration in the breathing zone is reliable. The accuracy of the model of CO_2 concentration in the breathing zone will be demonstrated by comparing its predicted CO_2 concentrations in the breathing zone with those from the experiments (Section 3.2).

For the second issue, with the given outdoor CO_2 concentrations and indoor CO_2 generation rate, one supply airflow rate is determined for each fresh outdoor air ratio to achieve the targeted CO_2 concentration in the breathing zone according to the model of CO_2 concentration in the breathing zone (i.e., f_3 in Figure 1). With the determined supply airflow rate, the air temperature in the occupied zone is calculated to meet the requirement of thermal comfort according to the modified PMV model for stratum ventilation (Equation 5) [12, 58, 59]. The other two ventilation parameters (i.e., supply air temperature and exit air temperature) can be solved by the requirement of heat removal of the indoor environment (Equation 6) and air distribution characteristics of stratum ventilation (Equation 7) [60]. The heat required to be removed from the indoor environment (i.e., room cooling load) can be simulated by building models in TRNSYS or by other existing room cooling load prediction models [61, 62]. It is noted that since the indoor air temperature distribution of stratum ventilation is non-uniform [26, 27, 63], TRNSYS [64] is recommended to be integrated with the multi-node model [65] to accurately predict the room cooling load. The targeted CO_2 concentration in the breathing zone can be determined in compliance with standards of indoor air quality or according to the preferences of the users. Thus, for each fresh outdoor air ratio, the corresponding ventilation parameters of the supply airflow rate, supply air temperature and exit air temperature are obtained for stratum ventilation, which satisfies the targeted indoor air quality. The fresh outdoor air ratios and corresponding ventilation parameters are used as inputs for the energy performance evaluation of the air conditioning system by building energy simulation tools (e.g., TRNSYS [64]). The fresh outdoor air ratio with the minimal energy consumption of the air conditioning system is identified as the optimum. Therefore, the proposed optimization can identify the optimal fresh outdoor air ratio and the corresponding ventilation parameters to maximize the energy saving of the air conditioning system and achieve targeted indoor air quality simultaneously. It is noted that the proposed method can also satisfy different thermal comfort preferences by replacing the PMV value of zero in

Figure 1 with the preferred value. Since the main contributions of this study are to optimize the fresh outdoor air ratio for the targeted CO_2 concentration in the breathing zone and the maximal energy efficiency, the merit of the proposed method of providing the preferred PMV is not emphasized.

$$PMV = -\frac{7}{91300} N{T_\mathrm{r}}^2 + \frac{19}{50} T_\mathrm{r} - \frac{689}{74} \tag{5}$$

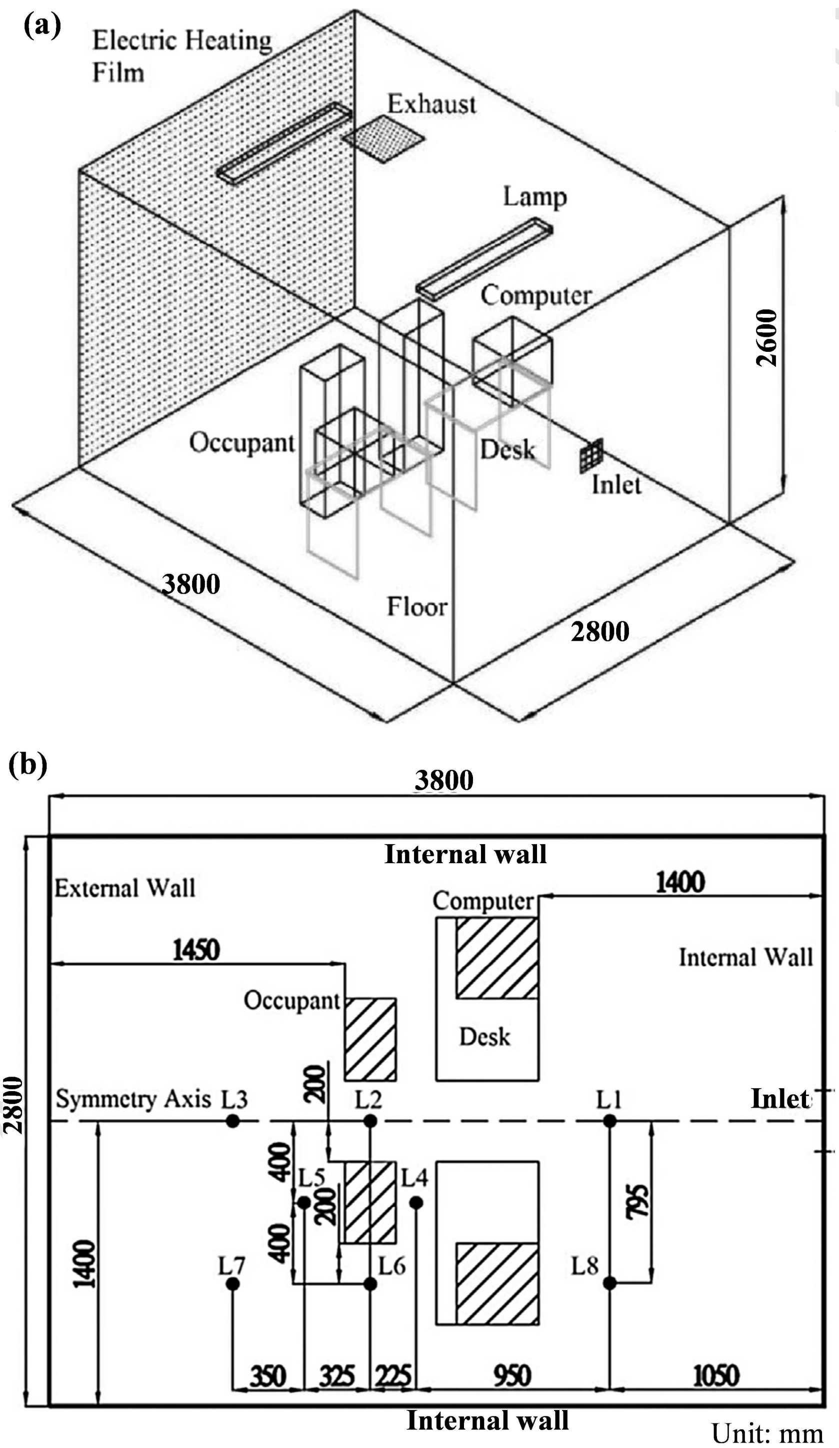
$$Q_\mathrm{cl} = \frac{NV\rho_{air}c_\mathrm{p}}{3600}(T_\mathrm{e} - T_\mathrm{s}) \tag{6}$$

$$\frac{T_\mathrm{r} - T_\mathrm{s}}{T_\mathrm{e} - T_\mathrm{s}} = \frac{NV(11.787 - 0.432T_\mathrm{s})}{3600Q_\mathrm{cl}} + 0.419 \tag{7}$$

where c_p is the specific heat capacity of air ((kJ/(kg·^{0}C)); PMV is Predicted Mean Vote; T_e, T_r and T_s are the temperatures of the exit air, air in the occupied zone and supply air (^{0}C); Q_cl is the room cooling load (kW).

2.2 Experiments for modeling CO_2 concentration in breathing zone

A typical office cooled with stratum ventilation at Xi'an Jiaotong University, Xi'an, is shown in Figure 3. The cool air is supplied from the middle level of one side wall and exhausted from the ceiling. The inlet terminal is a 200 mm × 195 mm grille, and the exit terminal is a 600 mm × 600 mm perforated panel. The office is designed for two occupants, with the dimensions of 3800 mm (length) × 2800 mm (width) × 2600 mm (height) as the experiments described by Huan et al. [45]. This environmental chamber is surrounded by an air-conditioned room. The heat generation by the electric heating film embedded in one side wall (regarded as the exterior wall) is used to simulate the effects of outdoor weather condition. The other walls are insulated and assumed to be the interior enclosure without heat transmission from the ambiance. The internal heat sources include two occupants (65 W each), two computers (80 W each) and two lamps (70 W each). The two seated occupants are simulated by manikins, with the dimensions of 400 mm (length) × 250 mm (width) × 1200 m (height). Each manikin is heated by bulbs with a power of 65 W. At the 1.1 m height of the manikin, CO_2 is released to simulate the exhalation, with a flow rate of 320 ml/min and a temperature of 36^{0}C. Eight sampling lines for CO_2 concentration are placed on one side of the symmetric axis of the office (Figure 3), with three sampling points for each sampling line in breathing zone for sedentary occupants (i.e., at the heights of 0.9 m, 1.1 m and 1.3 m). The TES-1370 non-dispersive infrared radiation system is used to measure CO_2 concentration, with a measuring accuracy of $\pm 3\%$ for the measuring range of 0 ppm to 6000 ppm.



Note: L1-L8 are the sampling lines of CO_2 concentration for seated occupants, with *three height levels of each sample line, i.e., 0.9 m, 1.1 m and 1.3 m.*

Fig.3. (a) Setup of environmental chamber and (b) sampling lines for CO_2 concentration in breathing zone.

Thirteen experimental cases are designed to cover a wide range of the supply airflow rate from 5 ACH to 17 ACH and the supply air temperature from 17 ^{0}C to 21 ^{0}C for a broad range of the room cooling load from 0.43 kW to 1.49 kW. The room cooling load is adjusted by the electric heating film simulating the heat gain from the ambiance. For all experimental cases, the air temperature in the occupied zone is controlled within 25.7 ^{0}C to 26.3 ^{0}C for thermal comfort. The experimental cases are divided into Series 1 (Case 1-6) and Series 2 (Cases 7-13). Series 1 is used for the correlation identification of the CO_2 removal efficiency in the breathing zone by ANOVA (analysis of variance) and multiple regression (Section 2.1). Series 2 is not involved in the correlation identification but used for validating the developed correlation. Both the experiments for model development and validation are randomly determined. However, the experiments for model validation (Experiments 7-13) are designed to cover a broader range of the supply airflow rate than that of the experiments for model development (Experiments 1-6) to test the robustness of the developed model of CO_2 concentration in the breathing zone (see Section 3.1). More detailed information on the experiments can be found in Huan et al. (2016) [45].

Table 1. Experimental case designs and resulted CO_2 removal efficiency in the breathing zone.

Cases		Q_{cl} (kW)	N (ACH)	T_s (°C)	T_r (°C)	ε
Series 1	1	0.43	5.7	21	25.8	1.70
	2	0.76	7.2	17	25.7	1.62
	3	0.76	9.2	19	26.0	1.40
	4	1.24	9.5	17	25.8	1.51
	5	0.99	11	19	26.0	1.34
	6	1.24	15	19	26.1	1.32
Series 2	7	0.54	5	17	25.7	***1.76***
	8	0.54	6.5	19	26.3	1.68
	9	0.99	8.5	17	26.3	1.52
	10	0.54	9	21	25.7	1.51
	11	1.49	10.2	17	25.7	1.48
	12	0.76	11.5	21	25.8	1.35
	13	0.99	17	21	25.9	***1.29***

Note: N is the supply airflow rate; Q_{cl} is the room cooling load; T_r and T_s are the room air temperature and supply air temperature respectively; Series 1 is used for the correlation identification of CO_2 removal efficiency in the breathing zone (ε), and Series 2 is not involved in the correlation identification but used for the correlation validation.

2.3 Energy performance evaluation of air conditioning system

TRNSYS [64], one of the most popular building simulation tools, is used for the energy performance evaluation of the air conditioning system (Figure 2). The energy consumption of the air conditioning system is mainly counted by the water-cooled chiller (including the chiller and cooling water loop), pumps of the primary and secondary chilled water loops and ventilation fans [66, 67]. Energy performance models of these components in TRNSYS are standardized and have already been validated by the developers (e.g., Type 666 for the chiller, Type 742 for the constant speed pump in the primary chilled water loop, Type 741 for the variable speed pump in the secondary chilled water loop, Type 744 for the ventilation fan), which are the same as those in previous studies [12, 67-72]. The energy consumption of the chiller is calculated from Equations 8-12, and the energy consumptions of the pumps and fans are estimated using Equations 13 and 14 respectively [71-73]. The key parameters of the air conditioning system in TRNSYS are summarized in Table 2, more detailed descriptions of the energy consumption evaluation of the air conditioning system are available in studies [67-72].

$$EC_{chi} = 3600FFLP \frac{Q_c \, r_1}{COP \, r_2} \tag{8}$$

$$FFLP = f_4(PLR) \tag{9}$$

$$PLR = \frac{Q_{cl}'}{Q_c \, r_1} \tag{10}$$

$$r_1 = f_5(T_{ocw}, T_{icw}) \tag{11}$$

$$r_2 = f_6(T_{ocw}, T_{icw}) \tag{12}$$

where COP is the rated chiller COP, which is set to 5; EC_{chi} is the chiller energy consumption (kJ/hr); $FFLP$ is the fraction of full load power which is related to the part load ratio (i.e., f_4 provided by TRNSYS) [64, 74]; PLR is the part load ratio; Q_c is the rated chiller capacity (kW) [68]; and Q_{cl}' is the chiller cooling load [66]; r_1 is the capacity ratio, which is calculated from the outlet chilled water temperature (T_{ocw}) and inlet cooling water temperature (T_{icw}) (i.e., f_5 provided by TRNSYS) [64]; r_2 is the COP ratio, which is calculated from the outlet chilled water temperature and inlet cooling water temperature (i.e., f_6 provided by TRNSYS) [64].

$$EC_{pump} = \frac{3600\dot{m}_{wat} \, \Delta h}{\eta_{pump} \, \rho_{wat}} \tag{13}$$

$$EC_{fan} = \frac{NV \, \Delta P_{air}}{\eta_{fan}} \tag{14}$$

where EC_{fan} is the fan energy consumption (kJ/hr); EC_{pump} is the pump energy consumption (kJ/hr); $\dot{m}_{wat}$ is the water flow rate (kg/s) [71]; η_{fan} is fan efficiency, which is set to 70%; η_{pump} is pump efficiency, which is set to 60%; ρ_{wat} is water density ($\mathrm{kg/m^3}$); Δh is the head of the water flow (m) [75]; ΔP_{air} is the total pressure (Pa) [71].

Table 2. Key parameters of the air conditioning system [68].

Component	Parameter	Value	Unit
Chiller (Type 666)	Rated COP	5	-
	Set point temperature of chill water	7	℃
	Cooling water temperature	32	℃
Variable speed Pump (Type 741)	Overall pump efficiency	60	%
	Motor efficiency	90	%
Constant speed Pump (Type 742)	Overall pump efficiency	60	%
	Motor efficiency	90	%
	Pressure drop	100	kPa
Ventilation fan (Type 744)	Overall fan efficiency	70	%
	Motor efficiency	90	%

3. Results

3.1 Correlation between CO_2 removal efficiency in breathing zone and supply airflow rate

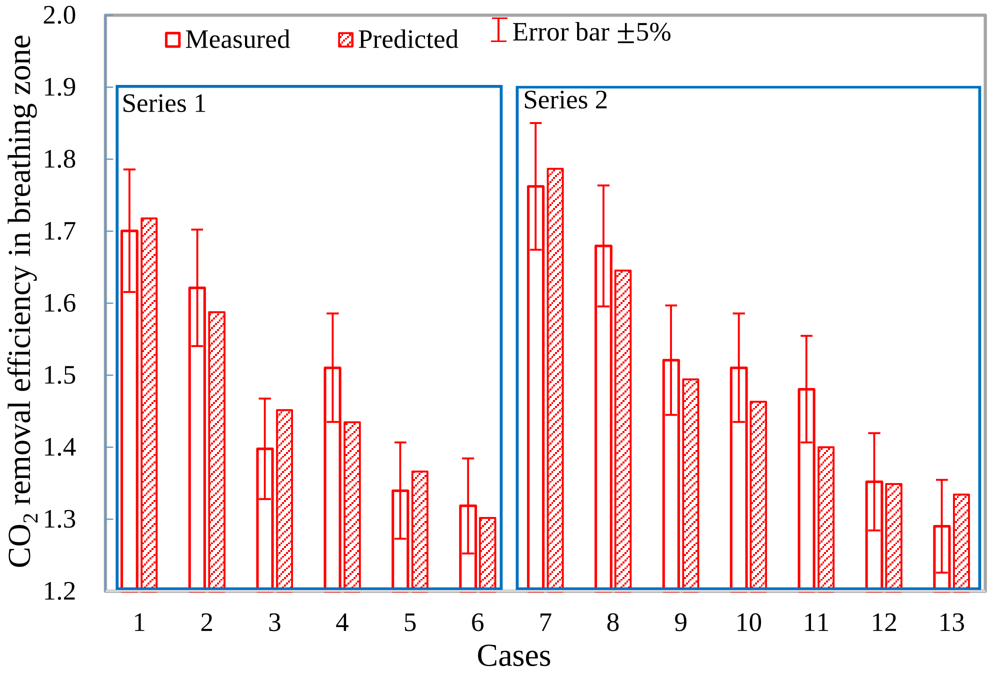
Table 1 and Figure 4 show that the CO_2 removal efficiency in the breathing zone of Series 1 and Series 2 from the experiments ranges from 1.29 to 1.76, indicating that the CO_2 distribution of stratum ventilation is significantly non-uniform. The maximal CO2 removal efficiency is achieved at the supply airflow rate of 5 ACH (Experiment 7), and the minimal CO2 removal efficiency is achieved at the supply airflow rate of 17 ACH (Experiment 13). Based on the results of Series 1, the CO_2 removal efficiency in the breathing zone is correlated to the supply airflow rate as shown in Equation 15, with a coefficient of determination R^2 of 0.91. Software Design Expert [76] is used to obtain the regression model of Equation 15. The CO_2 removal efficiency in the breathing zone of Series 1 predicted by the correlation agrees reasonably with the experimental data, with a maximal absolute error no greater than 5% and a mean absolute error of 2.5% (Equation 16 [77]). The correlation is further validated that the maximal absolute error for Series 2 is no greater than 5% except Case 11 (5.4%) and the mean absolute error is 2.5%. Thus, the correlation between CO_2 removal efficiency in the breathing zone and supply airflow rate is accurate.

$$\varepsilon = 0.0054N^2 - 0.1565N + 2.4344, \qquad R^2 = 0.91 \quad (15)$$

$$MAE = \frac{\sum_{j=1}^{n}\left|\frac{mea_j - pre_j}{mea_j}\right|}{n} \quad (16)$$

where MAE is the mean absolute error; mea_j and pre_j are the measurement and prediction respectively; j is for the j^{th} case; n is the number of cases.

Moreover, the validations imply that the correlation is general, which can be extended to conditions beyond those used for the correlation identification. The correlation is identified under conditions with a supply airflow rate from 5.7 ACH to 11 ACH (i.e., Series 1 in Table 1), but can be satisfactorily applied to conditions with a supply airflow rate extended to between 5 ACH and 17 ACH (i.e., Series 2 in Table 1). The generality of the developed correlation will be further discussed in Section 4 by extending to other ventilation modes.



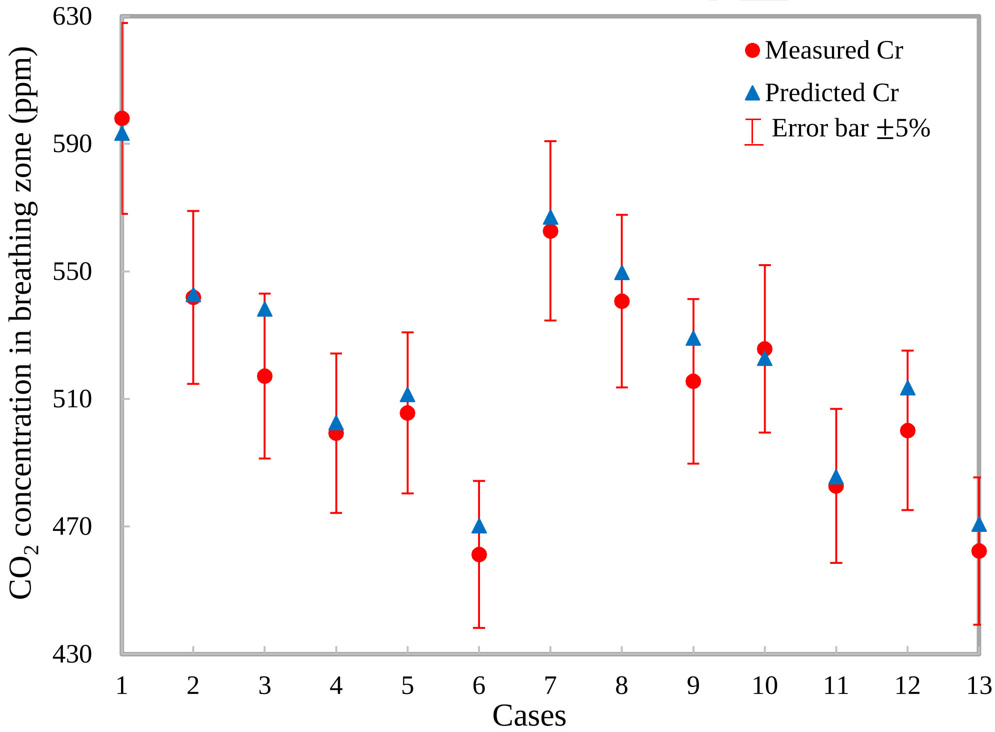
Note: Error bar of ±5% indicates that the relative difference between the predicted and measured CO_2 removal efficiencies in the breathing zone is within ±5%.

Fig.4. Comparisons between predicted and measured CO_2 removal efficiencies in breathing zone.

3.2 Model of CO_2 concentration in breathing zone

By replacing the CO_2 removal efficiency in the breathing zone (ε) in Equation 4 with the validated correlation in Section 3.1 (i.e., Equation 15), the model of CO_2 concentration in the breathing zone is finally expressed as Equation 17. Since Equation 4 is the equation of the CO_2 mass conservation (Section 2.1) and Equation 15 has been experimentally validated, Equation 17 (i.e., the model of CO_2 concentration in the breathing zone) should be regarded as accurate. This is further validated by the experiments. As shown in Figure 5, for Cases 1-13, the predicted CO_2 concentration in the breathing zone by the model (Equation 17) is fairly close to that of the experiments (Section 2.2), with the maximal error less than 5% and the mean absolute error of 1.9%.

$$C_r = C_o + \frac{3600G}{V(0.0054N^3 - 0.1565N^2 + 2.4344N\ \ )} + \frac{3600G(1-\gamma)}{NV\gamma} \qquad (17)$$



Note: Error bar of ±5% indicates the relative difference between the predicted and measured CO_2 concentrations in the breathing zone is within ±5%.

Fig.5. Comparisons between predicted and measured CO_2 concentrations in breathing zone.

3.3 Energy consumptions of different fresh outdoor air ratios

The room studied is the same as the environmental chamber in Figure 3 (Section 2.2). The cooling load is 0.96 kW (i.e., 90 W/m^2) [45]. The outdoor air temperature and relative humidity are 33.5 ^{0}C and 68% respectively [78]. The outdoor CO_2 concentration is 400 ppm [39]. For each fresh outdoor air ratio, the ventilation parameters including the supply airflow rate, fresh outdoor airflow rate and temperatures of supply and exit air are quantified to achieve the targeted CO_2 concentration of 1000 ppm in the breathing zone (Section 2.1) (Table 3).

Table 3. Ventilation parameters corresponding to different fresh outdoor air ratios for targeted indoor air quality quantified by the proposed optimization.

γ (%)	N (ACH)	Q_f (ACH)	T_s (^{0}C)	T_e (^{0}C)
10.0	22.2	2.2	25.4	30.1
12.5	17.9	2.2	23.3	29.1
15.0	14.9	2.2	21.7	28.7
17.5	12.7	2.2	20.3	28.5
20.0	10.9	2.2	19.1	28.6
22.5	9.6	2.2	17.9	28.7
25.0	8.5	2.1	16.7	28.9
27.5	7.6	2.1	15.6	29.2
30.0	6.8	2.0	14.4	29.6
32.5	6.2	2.0	13.2	30.0

Note: N is the supply airflow rate; Q_f is the fresh outdoor airflow rate; γ is the fresh outdoor air ratio; T_e and T_s are the temperatures of exit air and supply air respectively.

Figure 6 shows that the proposed optimization can maintain the targeted CO_2 concentration of 1000 ppm in the breathing zone, while the conventional method fails to meet the targeted CO_2 concentration in the breathing zone. The conventional method refers to that the fresh outdoor airflow rate is 10 L/s for each occupant which takes into consideration CO_2 removal efficiency in the breathing zone (Equation 18) [24]. When the fresh outdoor air ratio increases from 10% to 17.5%, the CO_2 concentration in the breathing zone with the conventional method decreases from 1123 ppm to 1073 ppm. When the fresh outdoor air ratio continuously increases to 32.5%, the CO_2 concentration in the breathing zone with the conventional method increases to 1243 ppm. The corresponding supply airflow rate decreases from 18.7 ACH to 4.3 ACH, which is calculated from the fresh outdoor airflow rate and fresh outdoor air ratio. When the fresh outdoor air ratio increases to be larger than 32.5%, the supply airflow rate of the conventional method decreases to be less than 4 ACH which might deteriorate the ventilation efficiency and even fail to satisfy the airflow characteristics of stratum ventilation due to the too small supply momentum [45]. Thus, the fresh outdoor air ratio larger than 32.5% is not considered. The variations of the CO_2 concentration in the breathing zone with the conventional method are mainly caused by the variations of the CO_2 removal efficiency in the breathing zone, which firstly decreases and then increases with the variations of the supply airflow rate (Equation 15). The CO_2 concentration in the breathing zone with the conventional method is also calculated by the model validated for the CO_2 concentration in the breathing zone (Equation 17), while the supply airflow rate is different from the proposed optimization. The proposed optimization increases the supply airflow rate thereby the fresh outdoor airflow rate as compared with the conventional method (Figure 6), so that the CO_2 concentration in the breathing zone with the proposed optimization can be maintained at 1000 ppm (Section 2.1).

$$Q_{f,c} = \frac{36k}{\varepsilon V} \tag{18}$$

where $Q_{f,c}$ is the fresh outdoor airflow rate with the conventional method (ACH); k is the number of occupants.

Although the CO_2 concentration in the breathing zone with the conventional method could be different from that shown in Figure 6 when the outdoor CO_2 concentration and indoor occupancy change, the variations of the CO_2 concentration in the breathing zone indicate that the conventional method fails to maintain the CO_2 concentration constant at the targeted level in the breathing zone. The main reason for the conventional method failing to maintain the targeted CO_2 concentration in the breathing zone is that the conventional method assigns a fixed airflow rate of fresh outdoor air for one occupant (i.e. 10 L/s) which has not been justified [31].

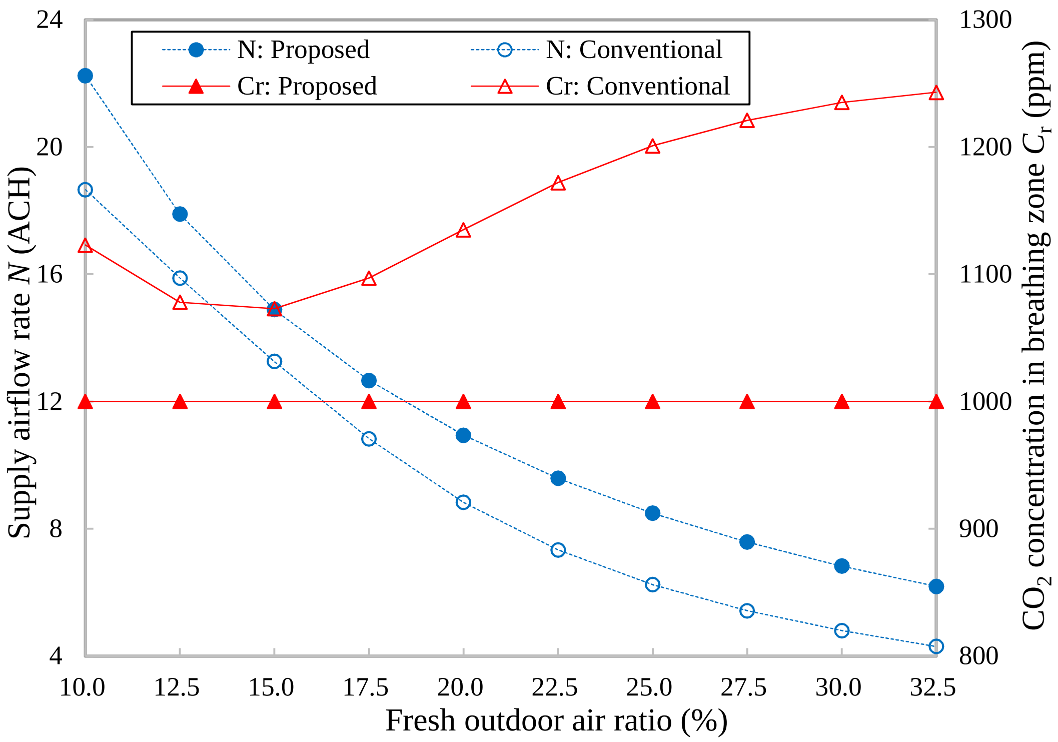


Fig.6. Comparisons between proposed and conventional methods: Variations of supply airflow rate and CO_2 concentration in breathing zone with fresh outdoor air ratios.

Using the ventilation parameters listed in Table 3 as the inputs, Figure 7 shows the variations of the energy consumptions of the air conditioning system and its components with the fresh outdoor air ratios from the proposed optimization. Since the conventional method fails to provide the targeted indoor air quality, one of the primary tasks of the air conditioning system, it makes no sense to evaluate the energy performance of the air conditioning system with the conventional method [2, 31]. With the proposed method, when the fresh outdoor air ratio increases from 10% to 32.5%, the energy consumption of the ventilation fan decreases from 890 kJ per hour to 170 kJ per hour. This is because the supply airflow rate decreases with the increasing fresh outdoor air ratio (Figure 6). However, the energy consumptions of the chiller and pumps increase from 1082 kJ per hour to 1807 kJ per hour. This is mainly caused by the largely reduced supply air temperature (Table 3) thereby a lower coefficient of performance (COP) of the chiller [64, 79, 80]. Although Table 3 also shows the fresh outdoor airflow rate slightly decreases from 2.2 ACH to 2.0 ACH which helps to decrease the chiller cooling load thereby reducing the energy consumption, its effects are overwhelmed by those of the largely decreased supply air temperature. As a trade-off between the variations of the energy consumption of the ventilation fan and those of the chiller and pumps, the total energy consumption of the air conditioning system firstly decreases and then increases with the minimal value achieved at the optimal fresh outdoor air ratio of 15%. Compared with the worst case (i.e., the fresh outdoor air ratio of 32.5%), the energy saving by the optimal fresh outdoor air ratio is 6.4%. Adopting the worst case as the benchmark is because no commonly recommended fresh outdoor air ratios for stratum ventilation are available.

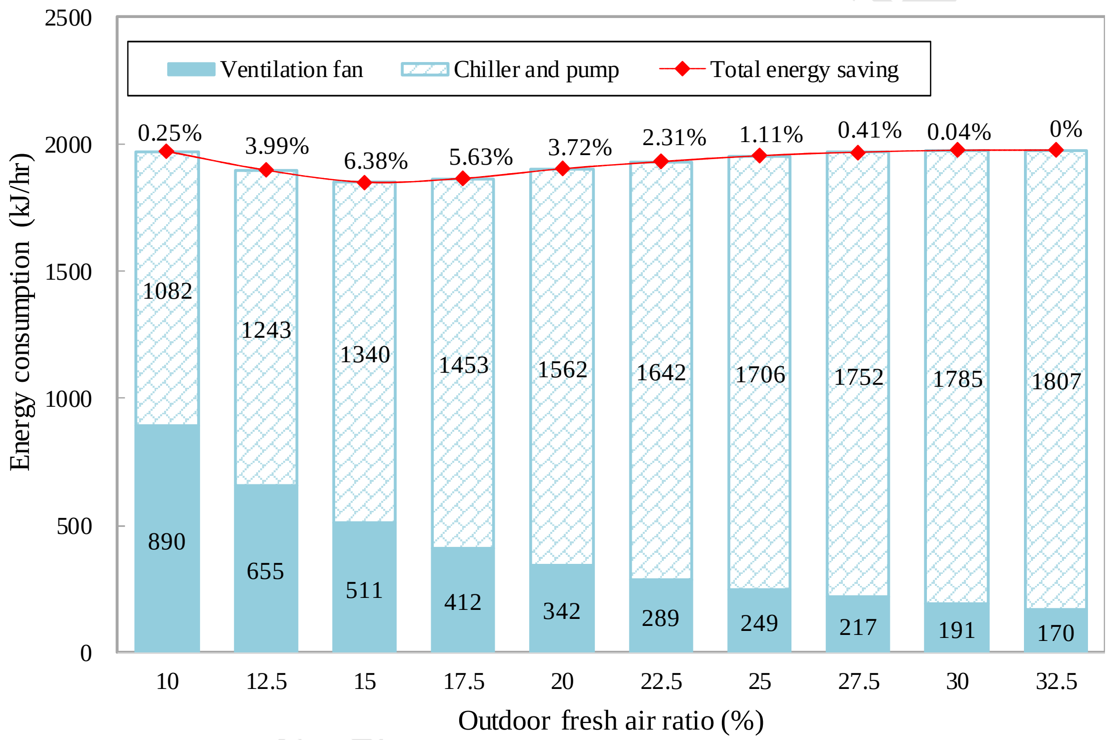


Fig.7. Variations of energy consumptions of air conditioning system and its components with fresh outdoor air ratios.

When the outdoor weather condition varies, the room cooling load also varies and the corresponding ventilation parameters for the targeted indoor air quality change (Section 2.1). With changed input ventilation parameters, the energy consumption of the air conditioning system would also change. The variations of the outdoor weather condition also affect the ventilation cooling load thereby impacting the energy performance of the air conditioning system [81]. Thus, the optimal fresh outdoor air ratio and its corresponding ventilation parameters need to be updated according to the outdoor weather condition [82]. It is noted that TRNSYS is just one of the feasible methods for the energy consumption evaluation of the air conditioning system. If the meteorological data and the performance data of the air conditioning system are available, similar to Study [83], the energy performance models of the air conditioning system can be coded to calculate the energy consumption by hand.

4. Discussion

The above case studies demonstrate the effectiveness of the proposed optimization method on the fresh outdoor air ratio for stratum ventilation to achieve both the targeted air quality and maximal energy saving. For the optimization method, three elements are specific for stratum ventilation, i.e., the model of the CO_2 concentration in the breathing zone (Equation 17), the PMV model (Equation 5) and the air distribution characteristics (Equation 7). As long as these three elements are available for other ventilation modes, the proposed optimization can also be applied. Both the PMV model and air distribution characteristics for other ventilation modes (e.g., mixing ventilation and displacement ventilation) can be derived according to the methods reported in the previous studies [12, 16]. Thus, when the model of the CO_2 concentration in the breathing zone for other ventilation modes is available, the proposed optimization can be extended to those ventilation modes. It is noted that when the design of stratum ventilation substantially changes (e.g., different terminal layouts), Equations 5, 7 and 15 are also recommended to be revised according to the methods proposed by Study [12], Study [16], and this study (Section 2.1) respectively.

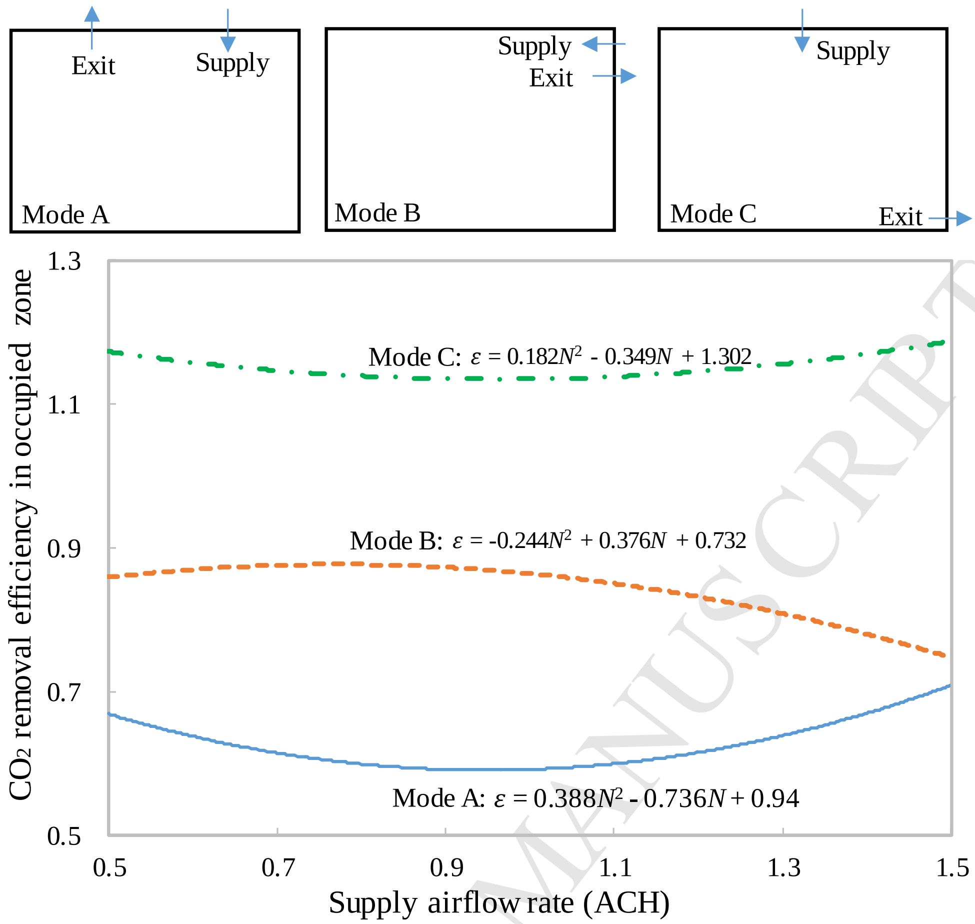
Krajčík et al. [51] experimentally determined that the CO_2 removal efficiency in the occupied zone of the three different ventilation modes (Modes A, B and C in Figure 8) can also be correlated to the supply airflow rate in a way similar to Equation 15. They focused on the CO_2 concentration in the occupied zone while this study focuses on the CO_2 concentration in the breathing zone. The occupied zone is broader than the breathing zone [45, 51]. Controlling CO_2 concentration in the occupied zone at a targeted level is a stricter requirement for indoor air quality. However, since the breathing zone is the critical zone for inhaled air quality, controlling CO_2 concentration in the breathing zone at a targeted level is adequate for indoor air quality [45]. It seems to be general for the CO_2 removal efficiency of the breathing zone/occupied zone to be correlated to the supply airflow rate as presented in

Equation 19. Besides the experiments of this study and the study by Krajčík et al. [51], the experiments from Olesen et al. (2011) [84] and Jurelionis et al. (2015) [85] also confirmed the generality of the correlation between CO_2 removal efficiency in the breathing zone/occupied zone and supply airflow rate presented in Equation 19. The constant coefficients in Equation 19 are different from ventilation modes to ventilation modes, and can be determined by experiments/CFD simulations using ANOVA (analysis of variance) and multiple regression. As a result, the model of the CO_2 concentration in the breathing zone/occupied zone can be obtained as shown in Equation 20 by replacing the CO_2 removal efficiency in the breathing zone/occupied zone in Equation 4 with Equation 19. Thus, the proposed optimization is also promising for other ventilation modes for the targeted indoor air quality and maximal energy saving.

$$\varepsilon = aN^2 + bN + c \tag{19}$$

$$C_{\mathrm{r}} = C_{\mathrm{o}} + \frac{3600G}{V(aN^3 + bN^2 + cN)} + \frac{3600G(1-\gamma)}{NV\gamma} \tag{20}$$

where a, b and c are the three constant coefficients in the correlation between the CO_2 removal efficiency in the breathing zone/occupied zone and supply airflow rate.



Note: The results are reported by Krajčík et al. (2012) [51]; for Mode A, both the supply and exit air terminals are on the ceiling; for Mode B, both the supply and exit air terminals are on the upper part of the same wall with the supply air terminal above the exit air terminal; for Mode C, the supply air terminal is on the ceiling and the exit air terminal is on the wall at floor level.

Fig.8. Variations of CO_2 removal efficiency in occupied zone with supply airflow rates for three ventilation modes.

It is noted that the proposed method is oriented for the supervisory control. The supervisory control has been widely implemented for the operation optimization of the air conditioning system [86, 87]. It updates the optimal settings of the operation parameters with a certain frequency, usually one time per hour [57, 86]. The steady-state assumption of the proposed method (Section 2.1) is acceptable for a time interval of one hour [73, 88]. The local control, i.e., the dynamic control of CO_2 concentration with a smaller time interval (e.g., one minute) considering the non-uniform distribution of indoor CO_2 concentration, needs to be further developed in the future.

5. Conclusions

This study proposes an optimization method on the fresh outdoor air ratio of stratum ventilation for both targeted indoor air quality and maximal energy saving. The indoor air quality is indicated by the CO_2 concentration in the breathing zone. A model of the CO_2 concentration in the breathing zone is developed by coupling the CO_2 removal efficiency in the breathing zone and the mass conservation law. Compared with the experiments, the mean absolute error of the model of the CO_2 concentration in the breathing zone is 1.9%. Based on the model of the CO_2 concentration in the breathing zone, for each fresh outdoor air ratio, the corresponding ventilation parameters can be quantified to maintain the CO_2 concentration in the breathing zone at a targeted level. As a comparison, the conventional method fails to achieve the targeted CO_2 concentration in the breathing zone. Using the fresh outdoor air ratios and quantified ventilation parameters as inputs, evaluations of the energy performance of the air conditioning system are conducted by building energy simulations with TRNSYS. The fresh outdoor air ratio with the minimal system energy consumption is identified as the optimum. Case studies show that the proposed optimization reduces the system energy consumption by 6.4% while achieving the targeted indoor air quality. The proposed optimization method can contribute to the improved performances of the air conditioning system with stratum ventilation, and is also promising for the air conditioning system with other ventilation modes.

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Highlights

- Optimization method is proposed for fresh outdoor air ratio of stratum ventilation.
- CO_2 concentration in breathing zone is modeled and experimentally validated.
- Targeted CO_2 concentration in breathing zone is achieved.
- Energy consumption of air conditioning system is minimized.
- Proposed optimization is promising for other ventilation modes.