Year-round Energy Saving Potential for a Stratum Ventilated Subtropical Office

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Abstract

Stratum ventilation has been proposed to cope for elevated indoor temperature recommended by governments in East Asia. TRNSYS is used for computation of the space cooling load and system energy consumption. A typical Hong Kong office is investigated. Compared with mixing ventilation and displacement ventilation, stratum ventilation derives its energy saving potential largely from the following two factors: a reduced ventilation load and increased coefficients of performance (COP) for chillers. The year-round energy saving is found to be substantial at 25% and 44% when compared with displacement ventilation and mixing ventilation respectively.

Keywords: stratum ventilation, displacement ventilation, mixing ventilation, annual, energy saving, TRNSYS simulation

Introduction

Minimizing the energy consumption by air conditioning systems would help to reduce CO_2 emission. Proactive actions in this regard have been taken by several governments in East Asia [1,2,3,4,5]. In order to cope with the recommended elevated room temperatures, stratum ventilation, a new ventilation method, was proposed by Lin et al. [6] with the following characteristics:

1. reversed temperature gradient in the occupied zone;
2. higher air speed at the head-chest level for the same air supply volume;
3. higher supply air temperature; and
4. higher evaporating temperature of the associated chiller, thus higher coefficient of performance (COP) of the chiller.

Tian et al. investigated the diffusion of CO_2 released by the manikin and thermal comfort under stratum ventilation experimentally [7]. The results demonstrated the flow pattern formed by stratum ventilation was able to provide good IAQ in the breathing zone.

System Description

Mixing ventilation (MV), displacement ventilation (DV) and stratum ventilation (SV) may be used to cool a building zone. Figure 1 shows the schematic diagram for the three systems. For MV, air is supplied to the building zone at high velocity at ceiling level which mixes fully with the room air before being extracted from the building zone (usually at ceiling level). The air temperature and pollutant concentration is assumed to be homogeneous within the building zone. For DV, air is supplied at/near the floor level at low velocity which fills the entire floor area and then displaces upward for extraction at ceiling level. The temperature (and usually also the pollutant concentration) increases with the height. For SV, air is supplied at an intermediate level close to the breathing zone/head level which keeps the temperature and pollutant concentration to a low level within the breathing zone. Consequently, a positive temperature and pollutant concentration gradient is established above the breathing zone while a negative gradient is developed below the breathing zone [8].

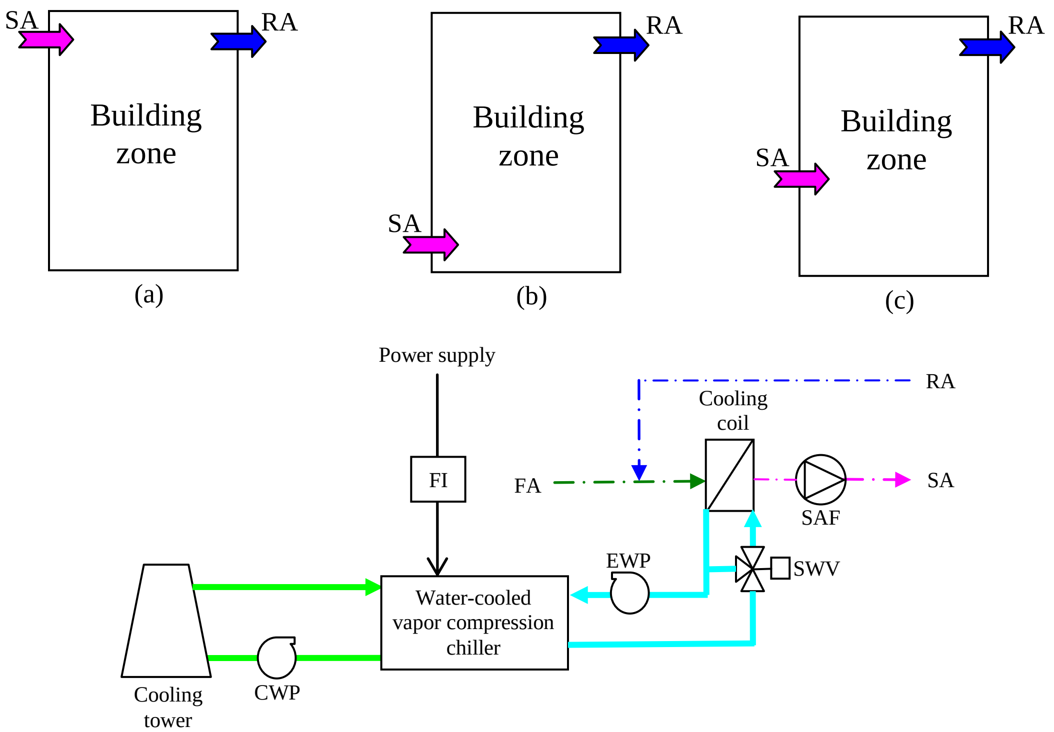

Figure 1 Schematic diagrams of (a) mixing ventilation, (b) displacement ventilation and (c) stratum ventilation systems equipped with an air-handling unit and a variable-speed water-cooled vapor compression chiller

An air-handling unit (AHU), consisting of a cooling coil and a supply air fan (SAF), and a water-cooled vapor compression chiller are used to provide the cooled supply air (SA) to the building zone. The chilled water entering the cooling coil is controlled by a three-way supply chilled water valve (SWV) based on the reference zone air temperature in the cases of DV and SV, or the return air (RA) temperature in the case of MV. The AHU controller generates a linear signal from 0 to 1 corresponding to nil to full flow to the coil when the zone temperature varies between the temperatures within

$\pm 1^{o}C$ of the set point. To avoid instability when simulating the cooling coil, the valve will open at least 30% when the output from the AHU controller is greater than zero. The operation of the chiller, the condenser water pump (CWP) and the cooling tower is governed by a return chilled water thermostat with a dead band of $3^{o}C$. The cooling tower is additionally controlled by a return condenser water thermostat triggered between 15 and $20^{o}C$. The fresh air (FA) amount is based on $0.01m^3/s$ per occupant. The AHU and the chilled water pump (EWP) will always be running within the daily occupancy schedule. A typical Hong Kong office is investigated. The zone loadings are calculated based on the varying ambient conditions and the pre-defined internal heat sources. Hence, the annual performance of the system can be studied.

Model Formulations

Lee presented a model for a variable-speed ground-source heat pump which is also applicable to a variable-speed chiller [9]. A parametric model as used by Jin and Spitler was adopted for the compressor where the various parameter values including the swept volume, volumetric coefficient, polytropic compression coefficient and the mechanical efficiency were determined using a parameter estimation technique based on performance data from the manufacturer (www.emersonclimate.com) [10]. The performance of the condenser and evaporator coils were calculated by dividing each coil into a maximum of three portions corresponding to the different states of the refrigerant in the coils. By employing different temperature effectiveness formulations for the coil portions, the performance of the entire coil was then solved by an iterative method. A thermostatic expansion valve was used which maintained a constant degree of superheat at the compressor suction. The solution for the complete chiller cycle was then determined based on the energy and refrigerant mass balances for the entire cycle as proposed by Domanski and Didion [11].

For DV, Skistad proposed a rule-of-thumb which stated that the room temperature sufficiently distant from the supply air terminal at 0.1m above the floor is approximately the mean between the supply and return/exhaust air temperatures. Moreover, the room temperature increased linearly from the floor to the ceiling. The reference zone temperature was defined at 1.1m above the floor [12]. Hence,

$$T_{fl} = \frac{T_s + T_e}{2} \quad , \qquad T_r = T_{fl} + T_{grad} \tag{1}$$

where

$$T_{grad} = \frac{T_e - T_{fl}}{H_{room} - 0.1} \tag{2}$$

and

$$Q_{sen} = mc(T_e - T_s) \tag{3}$$

Equations (1) to (3) can be solved if the supply air conditions and the building load are known.

For SV, the correlation of the reference zone temperature to the supply and exhaust air temperatures are represented by a dimensionless temperature defined as

$$\theta_r = \frac{T_r - T_s}{T_e - T_s} \tag{4}$$

based on experimental results [13]. Equation (3) also applies for SV. The same form of mathematical expression was also used to present experimental results both for temperature gradient and for humidity ratio gradient [14]. In estimating the reference zone humidity for both DV and SV, the humidity ratio is deemed to vary in the same fashion as the temperature along the room height.

System Specification

Figure 2 shows the configuration of the typical office used in this study. The office is divided into two zones, namely the individual room and the open concourse with two AHU's used. The west external wall of the individual room zone, with a window area ratio of 0.5, is 8.4 m long. All other partitions are assumed to be thermally symmetrical, meaning that there is no heat transfer across those boundaries.

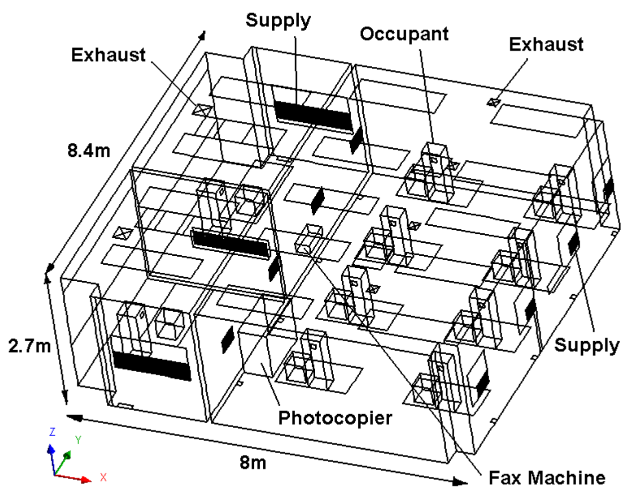

Figure 2 Configuration of typical office

Design loads are summarized in Table 1. The equipment load is assumed to be 30% radiative and 70% convective.

Table 1 Summarized design loads

Type	Room	Concourse
Occupant	2	8
Lighting ($W \cdot m^{-2}$)	19	19
Computer (W per occupant)	150	150
Equipment (W)	0	1850

Figures 3 and 4 show the daily operating schedules. Combined with the design loads stated in Table 1, the peak cooling load can be calculated.

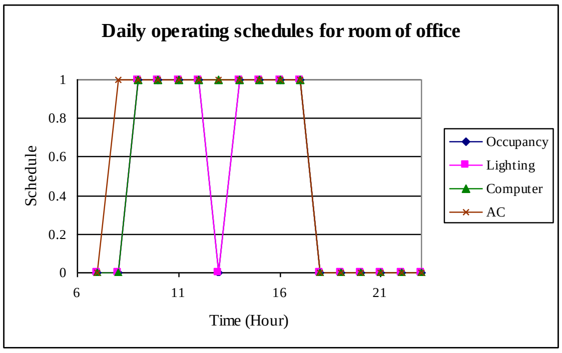

Figure 3 Daily operating schedules for individual office rooms

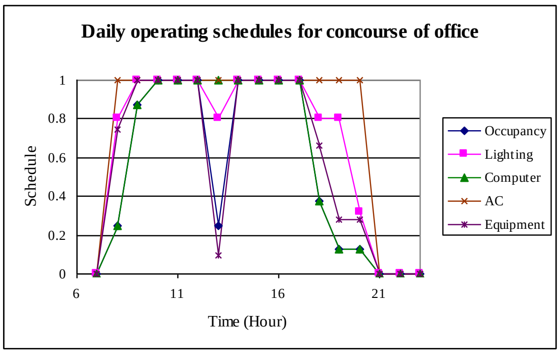

Figure 4 Daily operating schedules for office concourse

The thermal neutral temperatures under the three ventilation methods are different. The design temperatures for rooms ventilated with these methods should therefore not be identical. As a starting point, the design methods of DV and SV consider sensible loads only - no design relative humidity is defined. The corresponding settings are 24°C/54% for MV, 25°C for DV and 26.5°C for SV. The design outdoor condition is 33.5°C/68%. Table 2 summarizes the design cooling loads and required supply air conditions for the various cases. To allow better comparison of energy consumption, the design air flow rate of the MV case is also applied to the DV and SV cases. The system load includes the ventilation load resulting from the fresh air intake. The zone loads of the SV systems are the lowest followed by those of the DV systems, though the differences are less than 4% when compared with those of the MV systems. This is because the higher design indoor temperatures adopted, the less heat transmits through the external wall. The systems loads drop much more than the zone loads with the adoption of DV and SV systems. This is because the exhaust air or RA temperatures are substantially higher for the DV and SV systems which results in a large reduction in the ventilation loads. With a smaller chiller capacity, the corresponding requirements for the water pumps, the cooling tower and the cooling coil can also be reduced which means that both carbon foot print and cost in the construction stage can be lowered. This is also true in the operation stage.

Table 2 Summarized design requirements for the air-conditioning systems

Case	**Total zone load (kW)**	**System load (kW)**	**Supply airflow rate ($m^3 \cdot s^{-1}$)**	**Supply air temperature (^oC)**
Office rooms, MV	3.265	4.167	0.259	13.97
Office rooms, DV	3.201	3.834	0.259	18.17
Office rooms, SV	3.137	3.510	0.259	21.69
Office concourse, MV	4.342	7.943	0.315	13.62
Office concourse, DV	4.324	6.854	0.315	17.81
Office concourse, SV	4.306	5.732	0.315	21.53

With the system requirements fixed, the various parameters for the chiller are to be selected. As the supply air temperatures for the different systems vary significantly, the design chilled water supply temperature should also be different. The higher design supply air temperatures of the DV and SV systems allow higher chilled water temperatures to be adopted. This improves the coefficient of performance (*COP*) of the chiller and offers significant energy saving. To provide the same reasonable difference between the air temperature and chilled water temperature at the cooling coil, the corresponding design chilled water return temperatures are 13^{o}C for MV systems, 18^{o}C for DV systems and 21^{o}C for SV systems. The design condenser water return temperature is 30^{o}C in all cases. To set a comparable baseline in selecting the chiller parameters, *COP* of the chillers are assumed to achieve a rated of 3 when the chilled water return temperature is 13^{o}C in all cases. The design *COP* can then be determined based on the design chilled water return temperature (only applicable for the DV and SV systems). The condenser and chilled water flow rates are selected so that the temperature changes across the chiller are around 5^{o}C. Table 3indicates the various parameter values selected for the air-conditioning systems under different ventilation methods. Only single chiller is used for each case. The supply air fan head is taken as 750 Pa with a fan efficiency of 70% in all cases. The cooling tower fan head is taken as 200 Pa with a fan efficiency of 65%. All water pump efficiencies are assumed to be 60%.

Table 3 Parameter values for the chiller system

	MV	DV	SV
Chiller			
Condenser water mass flowrate ($kg{\cdot}s^{-1}$)	0.77	0.68	0.59
Chilled water mass flowrate ($kg{\cdot}s^{-1}$)	0.58	0.51	0.44
Swept volume of compressor (litre)	0.651	0.576	0.465
Volumetric coefficient of compressor	-0.225	-0.225	-0.239
Compressor polytropic compression coefficient	2.218	1.897	2.131
Electro-mechanical efficiency of compressor (%)	98.41	95.26	93.11
Degree of superheat at compressor suction (^oC)	11.1	11.1	11.1
Volume of refrigerant in condenser coil (litre)	3.00	3.00	2.50
Volume of refrigerant in evaporator coil (litre)	3.00	3.00	2.50
Volume of refrigerant in liquid line (litre)	0.20	0.15	0.15
Overall heat transfer value of condenser coil ($kW{\cdot}K^{-1}$)	1.18	1.00	1.00
Overall heat transfer value of evaporator coil ($kW{\cdot}K^{-1}$)	0.88	0.75	0.70
Refrigerant charge (kg)	1.38	1.00	1.38
Design *COP*	3.00	3.24	3.42
Auxiliary equipment			
Cooling tower air volume flowrate ($m^3{\cdot}s^{-1}$)	0.611	0.556	0.472
Condenser water pump head (kPa)	150	138	130
Chilled water pump head (kPa)	130	125	118

Methodology of Analysis

The TRNSYS simulation software is used for the analysis. New TRNSYS component is developed for the variable-speed chiller. System simulations are made for one year based on the typical weather data of Hong Kong [15] using a simulation time step of 3 minutes. The total primary energy consumption and system performance for the various cases with and without the part-load control are compared. An energy efficiency of 33% is assumed for the electric power plant in relation to the primary energy input.

Results and Discussions

Table 4 summarizes the year-round total energy consumption for the cases investigated. The percentage primary energy saving is at least 25% with the adoption of a DV system and 44% with the use of a SV system when compared with a MV system. A SV system offers at least 21% energy saving when compared with a DV system. The saving is substantial even taking account additional dehumidification of the fresh air might be needed.

Table 4 Year-round total primary consumption

MV (kWh)	35,248
DV (kWh)	26,221
SV (kWh)	19,609
Saving from MV to DV	25.61%
Saving from DV to SV	25.22%
Saving from MV to SV	44.37%

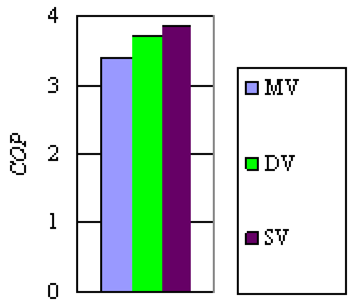

Figure 5 Year round mean COP for the different cases

Figure 5 shows the year-round-averaged *COP* for the different cases investigated. Figures 6 and 7 indicate the variation of the monthly-averaged zone conditions for the different ventilation methods. The temperature change only mildly throughout the year under the temperature control, but the relative humidity varies widely which reaches a maximum in the peak-load season. With the adoption of a DV or SV system, the relative humidity increases significantly due to the higher supply air temperatures used. With cooling-only dehumidification, the higher supply air temperature, the more moisture is introduced into the building zone. A high relative humidity impairs the indoor air quality besides the thermal comfort. On the other hand, the higher the zone temperature, the higher moisture content in the air for a specific relative humidity. This reduces the dehumidification load for SV.

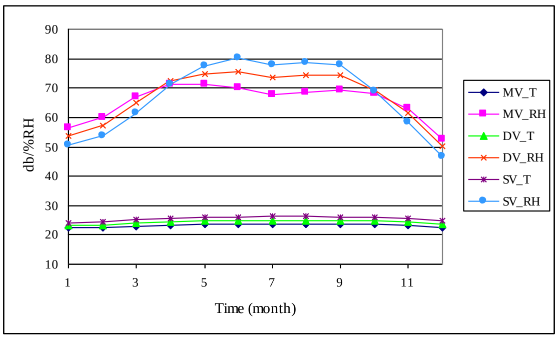

Figure 6 Monthly-averaged zone conditions for office rooms

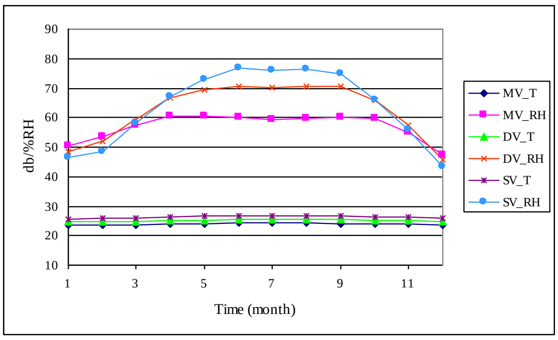

Figure 7 Monthly-averaged zone conditions for office concourse

Conclusion

The energy simulation software TRNSYS is used for computation of the space cooling load and system energy consumption. Year-round energy consumptions of three systems of stratum ventilation, displacement ventilation and mixing ventilation are compared for a typical office configuration in Hong Kong. The energy saving potential of stratum ventilation is derived mainly from two factors: the reduced ventilation load and increased COP of chillers. Because of the higher supply air temperature of stratum ventilation the energy consumed for the treatment of fresh air is much reduced. The increased COP reduces chiller energy in stratum ventilation. Through the year-round energy simulation of the typical office in Hong Kong, stratum ventilation has shown to be able to provide substantial energy saving when compared with conventional ventilation methods. The figure is 25% and 44% when compared with displacement ventilation and mixing ventilation respectively.

Nomenclature

c	Specific heat capacity of air ($kJ \cdot kg^{-1} \cdot K^{-1}$)
COP	Coefficient of performance
H_{room}	Zone height (m)
m	Supply air mass flow rate ($kg \cdot s^{-1}$)
Q_{sen}	Sensible load of building zone (kW)
T	Temperature (^{o}C)
T_{grad}	Temperature gradient along the zone height ($^{o}C \cdot m^{-1}$)
θ_r	Dimensionless temperature coefficient as defined in Equation (4).

Subscript

e	Exhaust air
fl	Zone air at 0.1m above floor
r	Reference zone point
s	Supply air

Abbreviation

AHU	Air-handling unit
CWP	Condenser water pump
DV	Displacement ventilation
EWP	Chilled water pump
FA	Fresh air
FI	Frequency inverter
MV	Mixing ventilation
RA	Return air
SA	Supply air
SAF	Supply air fan
SWV	Supply chilled water valve
SV	Stratum ventilation

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