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Adaptive Full-range Decoupled Ventilation Strategy and Air-conditioning Systems for Cleanrooms and Buildings Requiring Strict Humidity Control and Their Performance Evaluation

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Abstract: The air-conditioning systems in buildings and spaces, such as cleanrooms, requiring strict space humidity control are usually energy intensive, where significant energy wastes often occur due to improper system design and control. Dedicated outdoor air ventilation strategy, as the most recommended solution today, offers good energy performance by fully decoupling the cooling and dehumidification process. But its energy saving potential is restricted to a range of working conditions, due to the excessive outdoor air intake, it cannot provide energy-efficient operation when the internal latent load and ambient enthalpy are high. This paper therefore proposes a novel "adaptive full-range decoupled ventilation strategy", which offers optimized energy-efficient operation by incorporating the advantages of different ventilation strategies and adopting an adaptive economizer. This study also addresses the design of the air-conditioning systems for the implementation of the proposed ventilation strategy. The energy performance and economic analysis of the air-conditioning systems adopting the proposed ventilation strategy are investigated and compared with most updated strategies available. Results show that the proposed ventilation strategy can offer superior energy performance over the full range of internal load and weather conditions while the initial cost is even lower than that for the most recommended ventilation strategy today.

Keywords: dedicated ventilation, adaptive full-range decoupled ventilation, strict humidity control, energy conservation, air-conditioning system.

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Nomenclature

ADV	Adaptive full-range decoupled ventilation
AHU	Air-handling unit
IC	Interactive control with economizer
DV	Dedicated outdoor air ventilation
FL	Following latent load
FS	Following sensible load
LL	Lower-limit humidity control
MAU	Make-up air-handling unit
PAU	Primary air-handling unit
PD	Partially decoupled control
SHR	Sensible heat ratio
SSCL	Space sensible cooling load (W/m^2)
C_t	Cumulative total cost (USD/m^2)
C_{in}	Initial cost (USD/m^2)
C_e	Electricity cost (USD/m^2)
C_{ma}	Maintenance cost (USD/m^2)
COP_c	Overall coefficient of performance of cooling system
COP_{he}	Overall coefficient of performance of heating system
COP_{hu}	Overall coefficient of performance of humidification system
h	Enthalpy of air (kJ/kg)
h_{fg}	Latent heat of vaporization (kJ/kg)
Δp	Total pressure rise (kPa)
Q_{sen}	Internal sensible load (W/m^2)
Q_{lat}	Internal latent load (W/m^2)
$Q_{cc,MAU}$	Cooling coil cooling load of make-up air-handling unit (kW)
$Q_{cc,AHU}$	Cooling coil cooling load of supply air-handling unit (kW)
$Q_{he,AHU}$	Heater heating load of supply air-handling unit (kW)
$Q_{hu,AHU}$	Humidification load of supply air-handling unit (kW)
t	Temperature (°C)
w	Humidity ratio (kg/kg)
$w_{MAU,min}$	MAU minimum outlet humidity ratio (kg/kg)
W_{sf}	Supply fan power (kW)
W_{mf}	Make-up supply fan power (kW)

Greek letters

α	Outdoor air ratio
ε	Motor installation factor
η_f	Fan efficiency

Subscripts

c	Cooling system
cc	Cooling coil
e	Electricity
f	Fan
he	Heating system
hu	Humidification system
in	Initial
lat	Latent
ma	Maintenance
min	Minimum
mf	Make-up fan
sen	Sensible
sf	Supply fan
t	Total

1 Introduction

Strict space humidity control is needed in many applications such as hospitals, manufacturing facilities, pharmaceutical cleanrooms, art galleries, libraries, special hotel spaces and other commercial facilities [1]. In the subtropical climate, the humid outdoor air together with the ventilation requirement increase the dehumidification load and hence the energy consumption of air-conditioning systems. Conventional air-conditioning systems with the common purpose of thermal comfort rely on temperature-based control strategies to remove moisture, and thus allow the relative humidity to fluctuate [2, 3]. In the process of meeting the sensible cooling loads of buildings, the air is often treated below its dew-point temperature, coincidentally providing dehumidification in parallel with the indoor sensible heat removal. Therefore, the mismatch between building latent load and air-conditioning system dehumidification capacity, which also depends on the working conditions, can result in unsatisfactory indoor humidity control and cause degraded occupant comfort and productivity, and even damage from mold and condensation [4]. However, in some commercial and manufacturing facilities, most known as cleanrooms, both temperature and humidity need to be directly and strictly controlled. Adding humidity as a control objective would raise the control difficulties since cooling and dehumidification processes are highly coupled, leading to two highly coupled control loops [5-7].

In a particular working condition, the match between space sensible heat ratio (SHR) and SHR of a specific air-conditioning system depends significantly on the control strategy used. The most common control method is to employ cooling (i.e. sub-cooling) and reheating processes for eliminating the coupling between temperature and humidity control loops, while the counteraction of these processes may waste a large amount of energy [8]. Decoupled control strategies seem to be promising solutions for addressing this problem. Dedicated outdoor air systems (DOAS) for space cooling, which are regarded as fully decoupled control strategies, have been widely investigated and successfully implemented worldwide [9-11]. A typical DOAS consists of two independent systems: a dehumidification system dedicated to condition outdoor air (OA) that handles all indoor redundant latent load and part of the sensible load and a terminal system to remove the remaining sensible heat. As such, no extra overcooling and reheating for air treatment is needed to achieve satisfactory humidity control. Chilled ceiling combined with a desiccant dehumidification system [12, 13], which is one common type of DOAS systems, can provide a healthy and comfortable indoor environment with high energy performance. However, there are some limitations for its application. First, the chilled ceiling is suitable for environment control for human comfort and not suitable for environment control for industrial processes [14]. Secondly, the volatile problem of some desiccants is another concern for the environment for the industrial process as well as for human comfort [15]. Thirdly, the desiccant system and the dual-temperature chiller may significantly increase the initial cost and system complexity [16]. Therefore, the all-air systems (i.e., the combined use of make-up air-handling units (MAUs) (or primary air-handling units (PAUs)) and air-handling units (AHUs)) still widely existed in the real applications [17-20]. The MAU/PAU is used as a dehumidification component, while the AHU is regarded as a terminal component. Nevertheless, to remove the moisture produced by the occupants or manufacturing processes in such systems, the outdoor airflow rate sometimes exceeds the minimum outdoor airflow rate required for maintaining the acceptable indoor air quality or space positive pressure in a space. A large cooling capacity of the MAU is usually required to meet its high outdoor cooling demand, and outdoor air treatment can represent 30% to 65% of total air-conditioning electricity consumption [21]. To overcome the drawbacks of fully decoupled control strategy, the coauthors of this paper proposed a “partially decoupled control strategy” [22], which decouples the cooling and dehumidification process of air conditioning with the minimum set-point of the required outdoor airflow, and applied the method in an existing air-conditioning system of a building which has similar system configuration to the conventional design. The results show that around

69.6% of electricity and 87.8% of town gas consumptions can be reduced respectively in the test period. However, this method is applicable to the system serving spaces with relatively low internal latent load at most of the operational time due to the fact that the simultaneous cooling and reheating process would occur when the internal latent load is high.

Design and selection of an appropriate HVAC and dehumidification system for strict humidity and temperature controls require that the system meet both sensible and latent loads not only in the design condition but also in different possible working conditions. The air-conditioning systems should be properly designed to satisfy the needs of different working conditions and the control strategies, otherwise some problems [23, 24] such as over/under-sizing of air-conditioning components and energy waste would occur. For example, Pacific Gas and Electric Company [14] investigated a 1,200 m^2 cleanroom facility and found that the actual cooling load of the make-up air in operation is only about one-fourth of its design value (160 tons versus 630 tons), which subsequently increased the unnecessary capital costs.

This study therefore proposes a novel "adaptive full-range decoupled ventilation strategy" by systematically addressing its energy performance under different internal load and outdoor weather conditions, the economics of its implementation as well as the proper design of the associated air-conditioning systems. The proposed novel "adaptive full-range decoupled ventilation strategy" incorporates the advantages of "dedicated outdoor air ventilation strategy", "partially decoupled control strategy" and the adaptive economizer control, and therefore offers superior energy efficiency over the full range of internal load and weather conditions. The cleanrooms in a pharmaceutical factory, which have strict indoor environment control requirement located in Hong Kong, a humid sub-tropical city, is selected to test and validate the proposed ventilation strategy. The energy performance, energy saving potential, air-conditioning component design and economic analysis of the systems adopting the proposed ventilation strategy are investigated and compared with the systems adopting other most updated ventilation strategies available today.

2 Existing and proposed ventilation strategies and their comparison

The basic operation mechanisms of the proposed ventilation strategy and three existing ventilation strategies, which are most updated or recommended today, are presented and compared in this session. The advantages and limitations of each strategy are highlighted. To describe these ventilation strategies, a typical air-conditioning system configuration, i.e. a blow-through type MAU and a draw-through AHU served for a space, is selected as shown in Fig. 1. The MAU consists of a cooling coil, an axial fan and filters for conditioning make-up air. The AHU contains a cooling coil, a heater, an axial fan, a humidifier and filters for conditioning the supply air. A chiller plant supplies the chilled water to both MAU and AHU cooling coils. The summer condition in Hong Kong is considered in Section 2.1 to illustrate the air handling processes of the system.

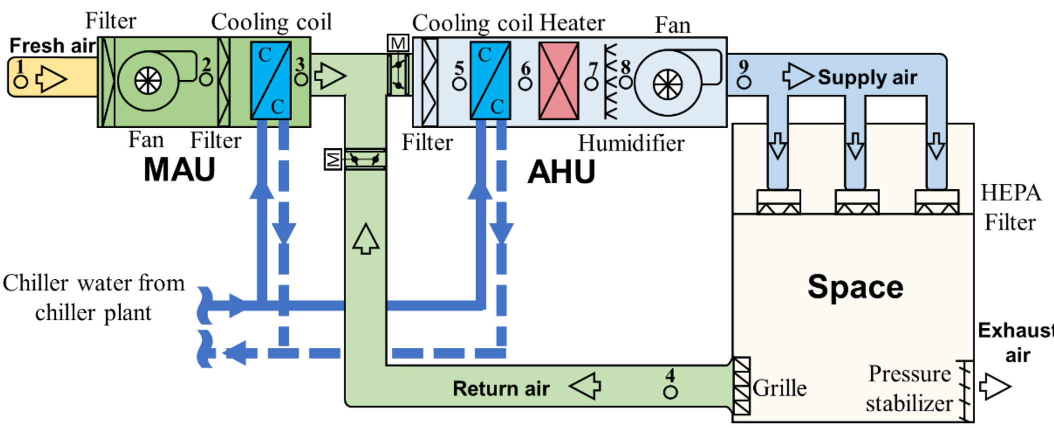

Fig. 1 System configuration of a typical air-conditioning system

2.1 Existing ventilation strategies

Interactive control with economizer (IC)

Fig. 2(A) shows the air-handling process on the psychrometric chart adopting the interactive control strategy [22], one of the most updated ventilation strategies for spaces requiring strict humidity control. The minimum required outdoor airflow is first introduced by the MAU with some temperature rise due to heat generation from the

MAU fan motor (1→2). The MAU cools the outdoor air down to its apparatus dew point (2→3), which is near the indoor air enthalpy. The cooled outdoor air is then mixed with the recirculation air in the AHU (3→5, 4→5) before being handled by the AHU cooling coil (5→6) for space dehumidification and then reheated by the heater (6→7) if necessary, further gains heat from the AHU fan motor (7→9) and eventually reaches the supply air temperature set-point.

However, during transient or winter seasons when the outdoor air is dry or with low enthalpy, an enthalpy-based economizer is activated according to the enthalpy differential between State 2 and State 4. If the enthalpy of State 2 is higher than that of State 4, the minimum outdoor airflow would be set. Otherwise, the outdoor airflow would be adjusted according to the AHU cooling coil outlet temperature, and the MAU cooling coil value would be closed. The outdoor air ratio of the IC strategy is expressed as Eq. (1). Where, α_{IE} is the outdoor air ratio of adopting the IC strategy. α_{min} is the minimum outdoor air ratio.

$$\begin{cases} \alpha_{IE} = min(max\left(\frac{t_4-t_6}{t_4-t_2}, \alpha_{min}\right), 1) & h_2 < h_4 \\ \alpha_{IE} = \alpha_{min} & h_2 \geq h_4 \end{cases} \quad (1)$$

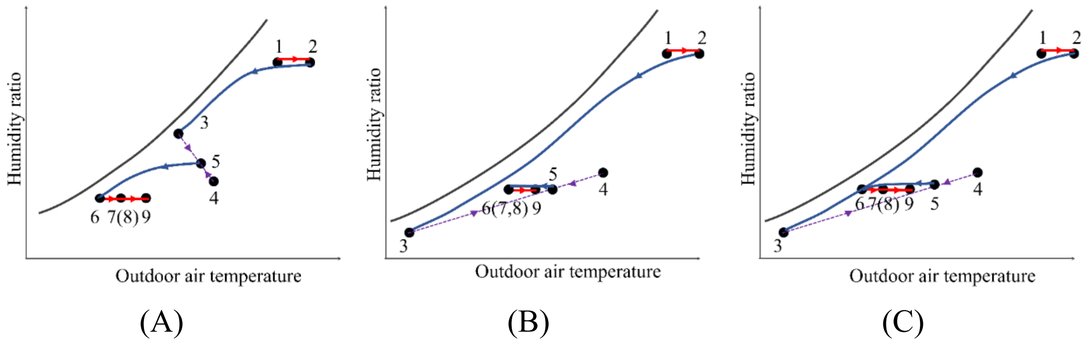

Fig. 2 Psychrometric process of different ventilation strategies for hot and humid weather conditions

Dedicated outdoor air ventilation (DV)

Dedicated outdoor air ventilation strategy, also named “fully decoupled control strategy”, is a ventilation strategy recommended currently for its promising energy performance. For the DV strategy, the MAU cools and dehumidifies the outdoor airflow to handle all internal latent load and part of the internal sensible load and the AHU removes the residual internal sensible heat. The outdoor air ratio of dedicated outdoor air ventilation strategy is determined according to the internal latent load, which is allowed to be higher than the minimum requirements under high internal latent load conditions, as shown in Eq. (2).

$$\alpha_{DV} = min(max\left(\frac{Q_{lat}}{h_{fg}(w_4 - \min(w_2, w_{MAU,min})}, \alpha_{min}\right), 1) \tag{2}$$

where, α_{DV} is the outdoor air ratio of adopting the DV strategy. Q_{lat} is the internal latent load (W/m^2). $w_{MAU,min}$ is the minimum outlet humidity ratio determined by MAU lower-limit outlet temperature (kg/kg). h_{fg} (2501kJ/kg) is the latent heat of vaporization.

The air-handling process of the DV strategy is shown in Fig. 2 (B) [2]. The indoor relative humidity and dry-bulb temperature are controlled by the MAU and AHU respectively. Different from the IC strategy, the outdoor air is cooled below the dew-point of the indoor air (2→3) and the MAU outlet air temperature set-point can be adjusted/reset until its lower limit. The MAU can induce higher outdoor airflow than its minimum requirement for the purpose of space dehumidification if the internal latent load is high. In this case, no reheating is needed.

Partially decoupled control (PD)

The outdoor air ratio of adopting the partially decoupled control strategy keeps the outdoor airflow rate at its lower limit, which is expressed as Eq. (3). Where, α_{PD} is the outdoor air ratio of adopting the PD strategy.

$$\alpha_{PD} = \alpha_{min} \tag{3}$$

The mechanisms and air-handling processes of the PD strategy are demonstrated in [22], as elaborated in Fig.2 (B-C). Fig. 2 (B) illustrates the air-handling process of the PD strategy when the internal latent load is relatively low. In this circumstance, since the outdoor air is dried effectively by the MAU which can handle all outdoor and indoor latent loads, there is no need to subcool and reheat the supply air in the AHU (State 6 and State 7 overlapping). When the internal latent load increases, the air-handling process is shown in Fig. 2(C), where the AHU only undertakes part of the latent load restricted by the use of fixed outdoor airflow set-point. To handle the rest of the latent load, the supply air needs to be cooled by the AHU cooling coil (5→6). The air is then reheated by the heater (6→7) (unless the indoor sensible load is high enough), which is similar to the air-handling process of the IC strategy.

2.2 Proposed adaptive full-range decoupled ventilation strategy

As described and compared above, all three most updated existing ventilation strategies have different limitations in applications. A novel "adaptive full-range decoupled ventilation strategy" (ADV) is therefore developed to overcome these limitations and minimize the energy consumption by compromising properly "inducing more outdoor air" and "sub-cooling and reheating process with minimum outdoor airflow" under high internal latent load conditions. Meanwhile, a new "adaptive economizer" is incorporated into the proposed strategy, which considers the needs of both temperature and humidity controls comprehensively during the cool and cold seasons. The adaptive economizer adopts three economizer control modes, named "following sensible load" (FS), "following latent load" (FL) and "lower-limit humidity control" (LL) as shown in Fig. 3.

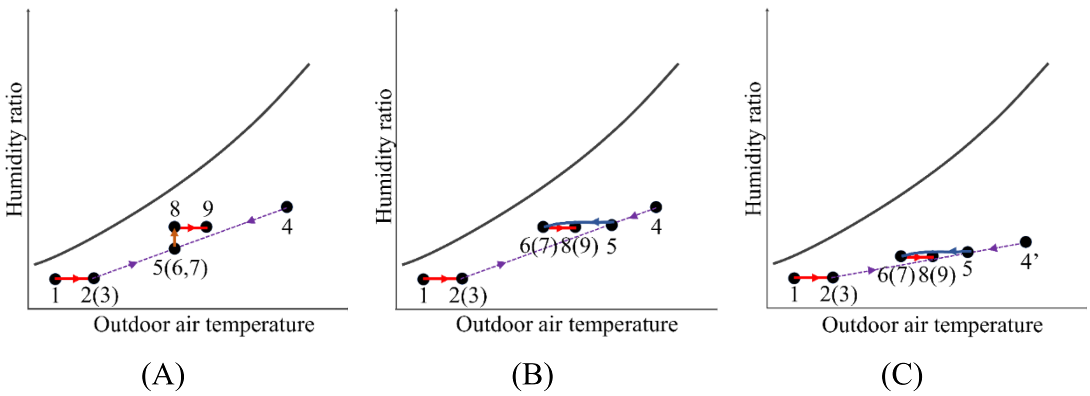

Fig. 3 The psychrometric process of economizer for cool/cold and dry weather

conditions

(A): Free cooling, (B): Free dehumidification, (C): Lower-limit humidity control

Fig. 3(A) shows the air-handling process on the psychrometric chart adopting the FS mode. In this mode, the outdoor airflow is adjusted according to the internal sensible load and the humidification might be needed if the internal sensible load is high (5→8). Fig. 3(B) shows the air-handling process on the psychrometric chart adopting the FL mode. In this mode, the latent heat is removed by adjusting outdoor airflow rate, keeping indoor relative humidity at the upper limit, while the remaining sensible cooling load is handled by AHU cooling coil (5→6). Fig. 3(C) shows the air-handling process on the psychrometric chart adopting the LL mode. Compared with the FL mode, the indoor relative humidity is always controlled at the lower limit (State 4'), so more outdoor air can be induced for removing the internal sensible heat.

The outdoor air ratios of different economizer modes are listed in Table 1. Where, α_{FS} is the outdoor air ratio of adopting the FS mode. α_{FL} is the outdoor air ratio of adopting the FL mode. α_{FL} is the outdoor air ratio of adopting the LL mode. w_4 *and* $w_{4'}$ are upper and lower limit settings of indoor humidity respectively (kg/kg). Q_{sen} is the internal sensible load (W/m^2). W_{sf} is the supply fan power (W). ε is the motor installation factor, which is used to indicate the motor location.

Table 1 Outdoor air ratios of different control modes of adaptive economizer

Control mode	Outdoor air ratio	
Following sensible load	$\alpha_{FS} = min\left(max\left(\frac{Q_{sen}+\varepsilon W_{sf}}{c_p(t_4-t_2)}, \alpha_{min}\right), 1\right)$	(4)
Following latent load	$\alpha_{FL} = min\left(max\left(\frac{Q_{lat}}{(w_4-w_1)}, \alpha_{min}\right), 1\right)$	(5)
Lower-limit humidity control	$\alpha_{LL} = min\left(max\left(\frac{Q_{lat}}{(w_{4'}-w_1)}, \alpha_{min}\right), 1\right)$	(6)

In operation, the proposed ventilation strategy identifies the economic operation mode under the dynamic weather and internal load conditions. Under the weather conditions that are hot and humid, if “inducing more outdoor air” is identified to be the most economical, the system will operate as Fig. 2(B), allowing the outdoor airflow to exceed minimum outdoor airflow rate required for maintaining an acceptable indoor air quality or space positive pressure in a space. Otherwise, the air-handling will follow the process shown in Fig. 2(C) using minimum outdoor airflow rate. In the weather conditions that are cool/cold and dry, the proposed ventilation strategy will choose the economizer operation modes which has better energy performance.

The process and steps of selecting the optimal mode using the ADV strategy are illustrated in Fig. 4. At each time step, the design indoor condition, outdoor condition, internal sensible and latent load, etc., are measured directly or computed based on measurements, and the required supply air state including the temperature and humidity can be then determined. By considering the outdoor condition (air temperature and humidity) and required supply air state, the required outdoor air ratios of different operation modes (i.e., PD, DV and economizer modes) can be obtained using Eqs. (2-6). The feasibilities of utilizing particular operation modes are therefore assessed by verifying whether the required outdoor air ratio is within the required range (i.e. lower than 1). The electrical loads of the feasible operation modes thus can be calculated according to their corresponding working principles using Eqs. (7-8, 12-17). The ADV strategy will eventually select the operation mode with the minimum electrical load.

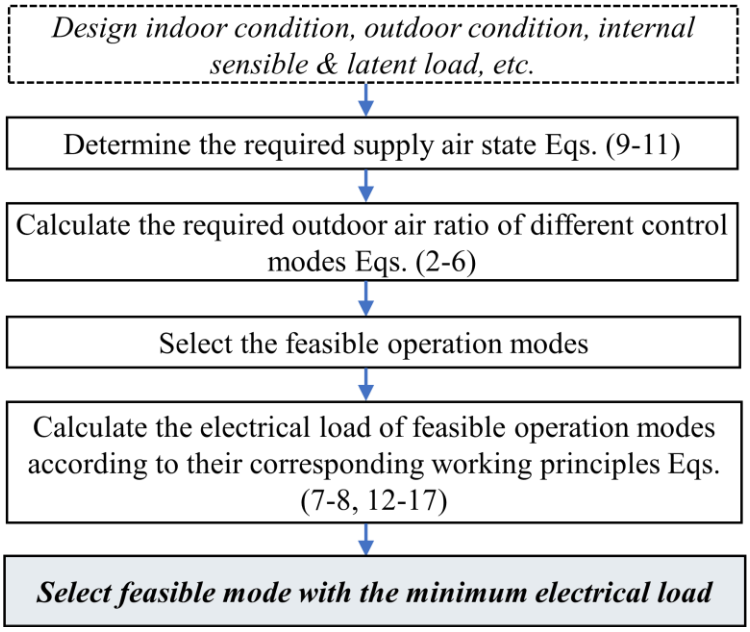

Fig. 4 Process and steps of selecting optimal operation mode by the ADV strategy

2.3 Comparison of existing ventilation strategies and proposed strategy

The induced outdoor air volume will significantly affect the energy consumption of air-conditioning systems when the outdoor enthalpy is high. The amount of outdoor air needed for a space depends on what type of the ventilation strategy is employed. Table 2 summarizes the capital cost, application and limitations of the existing ventilation strategies. The IC strategy has relatively good energy performance under cool/cold weather conditions since it makes some use of "free cooling and dehumidification" capacity of the outdoor air by adopting an enthalpy-based economizer. The DV strategy has a higher capital cost due to its requirement for the larger MAU cooling capacity, and its energy-saving potential is restricted, due to the excessive outdoor air intake, when the internal latent load and ambient enthalpy are high. Therefore, this ventilation strategy is applicable for the buildings in moderate or cold climate zones. The PD strategy is regarded as a method that can substitute the IC strategy during high enthalpy weather conditions because it is energy-efficient in part load conditions due to its decoupled control. It is worth noting that, when the internal latent load is high under hot and humid weather conditions, although the IC and PD strategies must consort to the overcooling and reheating process, the overall energy consumptions of these two strategies are possibly less than that of the DV strategy due to the energy savings on the outdoor air treatment.

Different from the IC strategy, the DV and PD strategies do not consider the economizer control because it may increase the difficulties for the MAU to control indoor relative humidity. Therefore, in some cool or cold weather conditions, the uses of "free cooling" and "free dehumidification" capacities of outdoor air are limited in these two strategies.

Table 2 Comparison of existing and proposed ventilation strategies

Strategy		Applicable situations	Limitations	Capital cost
Existing strategies	Interactive control with economizer	Buildings in moderate or cold climate zones	The counteraction between cooling and reheating for humidity and temperature controls	Low
	Dedicated outdoor air ventilation	Buildings in moderate or cold climate zones OR buildings with low internal latent load	The energy efficiency is low in the conditions when outdoor air enthalpy is high	High
	Partially decoupled control	Buildings with low internal latent load OR in hot climate zones	The counteraction between cooling and reheating for humidity and temperature controls occurs when the internal latent load is high	Low
Proposed	Adaptive full-range decoupled ventilation	Buildings over the full range of internal load and weather conditions	The more complicated control logic is needed	Medium

As shown in Table 2, the proposed ventilation strategy incorporates the advantages of the "partially decoupled control strategy", the "dedicated outdoor air ventilation strategy" and an adaptive economizer, and offers superior energy-efficiency over the full range of internal load and weather conditions. The capital cost of the proposed strategy is medium mainly due to the required capacities of some air-conditioning components are in-between that of three existing strategies, which will be elaborated further in Section 6.1.

3 Test system set-up for energy performance assessment

The air-conditioning system used in the energy performance assessment has the typical configuration as shown in Fig. 1. In the tests to assess the energy performance, the capacities of the subsystems/components are assumed to be as high as needed by the ventilation strategies.

3.1 Test building and weather/load conditions

A pharmaceutical factory building located in Tai Po district of Hong Kong is selected for the comparison study. It has five floors and the total cleanroom area is about 3620 m^2. All the production areas are designed as Class ISO 8 cleanrooms based on [25]. The configuration of a typical cleanroom air-conditioning system, i.e. the system serving part of the cleanrooms at 2nd floor, is shown in Fig. 1. A typical air-conditioning system of the cleanrooms consists of the axial fans, chilled water-cooling coils, an electric heater, an electric steam humidifier and other accessories.

For the cleanrooms concerned, the minimum total supply and outdoor airflow rates are designed as 20 air change rates per hour (ACH) ($2.88m^3/s$) and 2 ACH ($0.29m^3/s$) respectively to meet the requirements of indoor cleanness and pressurization [25-27]. The cleanroom air-conditioning systems adopt usually constant air volume (CAV) systems in practice. It is due to the fact that, as the high supply air flowrate (as the result of high ACH required) is required, even the minimum supply airflow can handle the peak internal load. It is worth noticing that for the cleanroom with the ISO class 3-7 (higher cleanliness), an even higher supply ACH is required. As Cleanroom Energy Efficiency Baselines conducted by PG&E [28], from Class 7 to Class 3, the minimum requirement of supply ACH varies from 50 to 800. For the higher cleanliness class cleanrooms, the system type and airflow pattern may be different (i.e. utilizing fan filter units (FFU) for unidirectional airflow pattern), but the way to handle the indoor latent load is similar.

The selected cleanrooms operate 24/7 for a total of 8760 hours. The design loads and schedules of lighting, occupancy and equipment are estimated according to the standards, field survey and machinery lists. The air-conditioning system of the cleanrooms operates continuously throughout a year. The design room conditions, equipment configuration and control requirements are summarized in Table 3.

Table 3 Design room conditions, equipment configuration, and control requirements

Description	Parameter	Value
Envelope details	Wall ($W/m^2 \cdot K$)	1.5
	Roof ($W/m^2 \cdot K$)	0.8
	Window ($W/m^2 \cdot K$)	2.7
	Window to wall ratio (WWR)	0.2
Indoor design conditions	Temperature (°C)	20±3
	RH (%)	55±10
	Volume (m^3) (length*width*height)	518 (12.5*14.7*2.8)
Outdoor and supply airflow rate	Outdoor air changes per hour	≥2
	Supply air changes per hour	≥20
Installed fans specification	MAU fan pressure (Pa)	1600
	AHU fan pressure (Pa)	1350
	Fan efficiency (%)	60
Internal loads (Sensible & latent heat)	Lighting (W/m^2)	13.9 + 0
	Occupants (W/m^2)	22 + 37
	Equipment (W/m^2)	45 + 15
Internal loads schedule	Lighting	00:00-24:00
	Occupants	05:00-21:00
	Equipment	00:00-24:00

The hourly weather conditions of Hong Kong in 1995, which is the typical meteorological year of Hong Kong [29], is used for calculating hourly space cooling load and the energy consumptions of adopting different ventilation strategies. The climate of Hong Kong is humid subtropical. The relative humidity in Hong Kong is rather high (80% on daily average), while the variations occur predominantly between 70% and 90% [30].

Assuming that the indoor temperature of the cleanrooms is controlled at its upper limit (23℃) with relative humidity varied at the desired range, the annual space cooling load profile is calculated in TRNSYS 18 [31] as shown in Fig. 5. It can be found that the cooling is required almost all the year due to the subtropical climate. The peak space total cooling load and sensible cooling loads are 145.77 W/m^2 and 92.46 W/m^2 respectively, which appears at 7:00 pm on July 22.

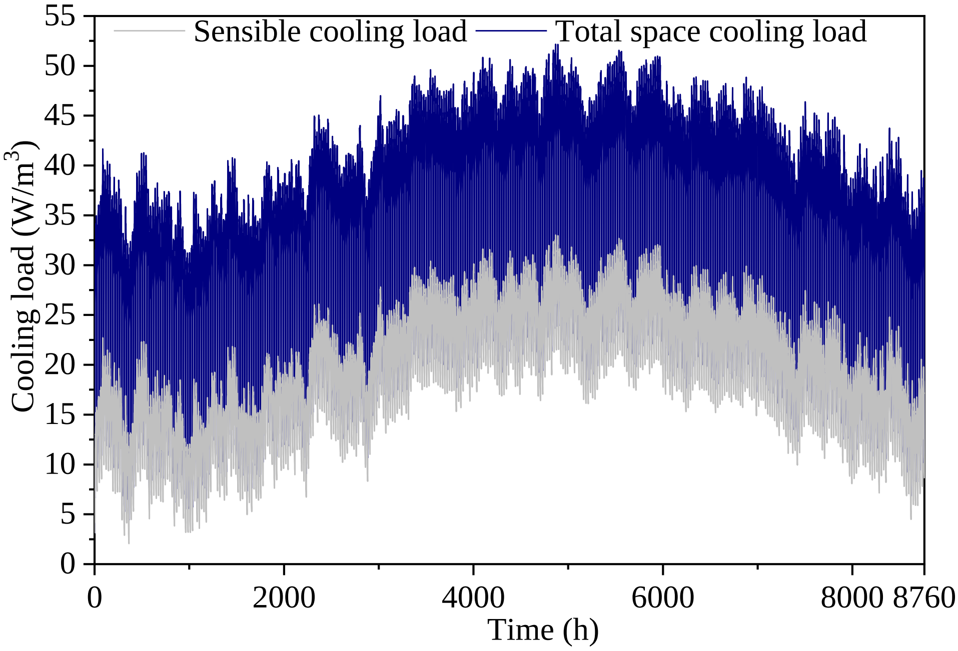

Fig. 5 Annual hourly space cooling load of the cleanrooms

3.2 Air-conditioning subsystem energy models

The total electrical load of the air-side components is calculated using Eq. (7), which includes the electrical load of the MAU/AHU cooling coils, the AHU heater and humidifier, the make-up air fan and the supply air fan.

$$E_{tot} = \frac{Q_{cc,MAU}+Q_{cc,AHU}}{COP_c} + \frac{Q_{he,AHU}}{COP_{he}} + \frac{Q_{hu,AHU}}{COP_{hu}} + W_{mf} + W_{sf} \tag{7}$$

where, COP_c is the overall coefficient of performance of the cooling system. COP_{he} is the overall coefficient of performance of the heating system. COP_{hu} is the overall coefficient of performance of the humidification system. $Q_{cc,MAU}$ is the cooling coil cooling load of the make-up air-handling unit (kW). $Q_{cc,AHU}$ is the cooling coil cooling load of the supply air-handling unit (kW). $Q_{he,AHU}$ is the heater heating load of the supply air-handling unit (kW). $Q_{hu,AHU}$ is the humidification load of the supply air-handling unit (kW). W_{mf} is the fan power of the make-up air-handling unit (kW). W_{sf} is the fan power of the supply air-handling unit (kW).

The energy models of the subsystems are described as follows.

<u>*Fan model*</u>

The fan powers of MAU/AHU fans are characterized by their volumetric flow rate, pressure rise and efficiency, as shown in Eq. (8). Where, W_f, is total fan power (kW). V is air volumetric flow rate (m^3/s). Δp is total pressure rise (kPa). η_f is fan efficiency.

$$W_f = \frac{V\Delta p}{\eta_f} \tag{8}$$

<u>*System energy balance model*</u>

The thermodynamic states of the system are determined by heat balance Eqs. (9)-(16) with three main assumptions: 1) The pressure drops through the ducts are constant and the air heat loss through the duct is neglected. 2) The minimum outlet temperatures of both MAU and AHU are set as 13°C, a setting typically used in practice [32], which are the lower limit of AHU and MAU outlet temperatures in design calculation when dehumidification requirements are concerned. 3) The saturated relative humidity is set at 95% when processed air reaches the apparatus dew point. 4) Perfect air mixing inside all ducts and the thermal space. 5) The ducts are well sealed without air leakage. 6) The air states at the outlets of MAU and AHU vary simultaneously with the instantaneous sensible and latent cooling demands. 7) The overall coefficient of performance of the cooling system (COP_c), heating system (electric heater) (COP_h) and humidification system (electric humidifier) (COP_{hu}) are assumed to be 2.5, 1.0 and 1.0 respectively as constants.

After determining the air states for each ventilation strategy (see also Fig. 2), the cooling/heating loads for the cooling coils/heater can be estimated using Eqs. (14)-(16). Where, m_s is the supply air mass flow rate (kg/s). h is the enthalpy of air (kJ/kg). The fan motor installation factor ε is used to indicate the motor location. It equals 1.0 if the fan motors are installed inside the MAU/AHU, and it equals the motor efficiency if motors are installed outside the MAU/AHU. The fan motor installation factor η equals 1 in the case study.

$$Q_{sen} = m_s c_p (t_4 - t_9) \tag{9}$$

$$Q_{lat} = Q_{sen}\left(\frac{1\text{-}SHR}{SHR}\right) \tag{10}$$

$$Q_{lat} = m_s h_{fg} (w_4 - w_9) \tag{11}$$

$$w_5 = \alpha w_3 + (1 - \alpha) w_4 \tag{12}$$

$$t_8 = t_4 - \frac{1}{m_s c_p}(Q_{sen} + \varepsilon W_{sf}) \tag{13}$$

$$Q_{cc,AHU} = (m_s - \alpha m_s) h_4 + \alpha m_s h_3 - m_s h_6 \tag{14}$$

$$Q_{he,AHU} = m_s (h_7 - h_6) - \varepsilon W_{sf} \tag{15}$$

$$Q_{cc,MAU} = \alpha m_s (h_1 - h_3) + \varepsilon W_{mf} \tag{16}$$

During the transient seasons and winter, when outside air is dry, the humidifier may be activated to avoid too low space humidity (i.e. lower than its lower-limits). The humidification load of the humidifier can be shown in Eq. (17).

$$Q_{hu,AHU} = m_s h_{fg} (w_8 - w_7) \tag{17}$$

4 Energy performance of proposed strategy and comparison with existing strategies

The energy performance of the proposed ventilation strategy is assessed and compared with the existing ventilation strategies under different internal load and weather conditions. Section 4.1 presents the energy performance of the proposed strategy and its comparison with the existing strategies under various internal load conditions and the Hong Kong design weather condition without adopting the economizer. Section 4.2 presents the energy performance of the proposed strategy and its comparison with the existing strategies under different weather conditions and a typical internal load condition when the adaptive economizer is adopted. Section 4.3 presents the energy performance of the proposed strategy and its comparison with the existing strategies in six selected cases (conditions).

4.1 Energy performance under different internal load conditions

The system performance maps for the IC, DV, PD and ADV strategies are presented in Fig. 6 (A)-(D), which represents the situation at Hong Kong design weather condition when no economizer control is adopted. As recommended by ASHRAE fundamentals 14.6 [27], the dew-point temperature corresponding to 1% annual cumulative frequency of occurrence (ASHRAE 1% DP-MCDB) is used as the design weather condition (i.e., 29.1 °C, 84.4%) when space dehumidification is the duty of the outdoor air ventilation system (i.e. make-up air-handling units). The total electrical load (Eq.7) of each strategy is calculated under 6,147,072 working points/conditions (3072×2001), with the interval of 0.05 W/m^2 and 5×10^{-4} in terms of space sensible cooling load (SSCL) and sensible heat ratio (SHR), respectively. Where, the total electrical load is the sum of the electrical loads of the cooling coils, the heater, the humidifier (if needed) and fans as calculated by Eqs. (7-17). It is worth noting that although the selected weather condition is hot and humid without involving the economizer, the induced outdoor air flowrates of different ventilation strategies are still different and the outdoor air flowrates of some ventilation strategies may exceed the lower limits of their set-points. This is due to different MAU dehumidification requirements of different strategies. For the IC and PD strategies, the outdoor air flowrate is set at the lower limit as a constant value (i.e., 2 ACH or outdoor air ratio 0.1 in the case study). For the DV and ADV strategies, the outdoor air flowrate is adjusted according to the internal latent load, allowing the amount of outdoor air higher than the lower limit to remove moisture under relatively high internal latent load conditions, as shown in Fig. 7(A) and (B), respectively.

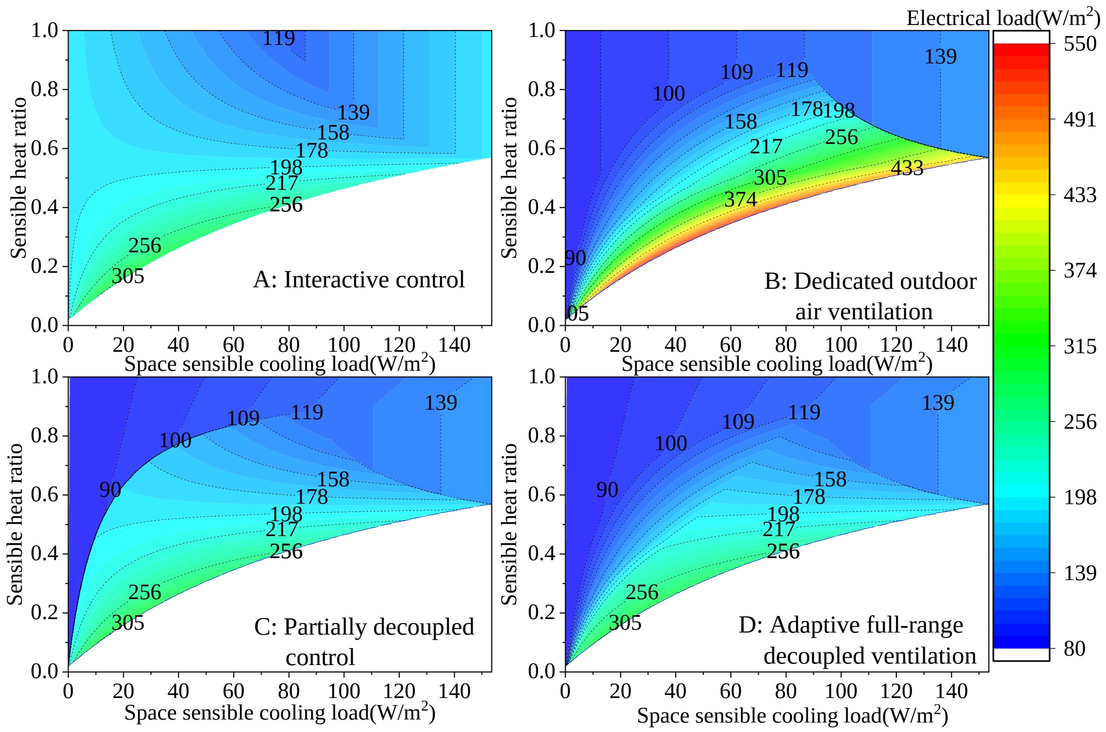

Fig. 6 Space air-conditioning energy maps of four ventilation strategies under Hong Kong design weather condition

(A: Interactive control; B: Dedicated outdoor air ventilation; C: Partially decoupled control; D: Adaptive full-range decoupled ventilation)

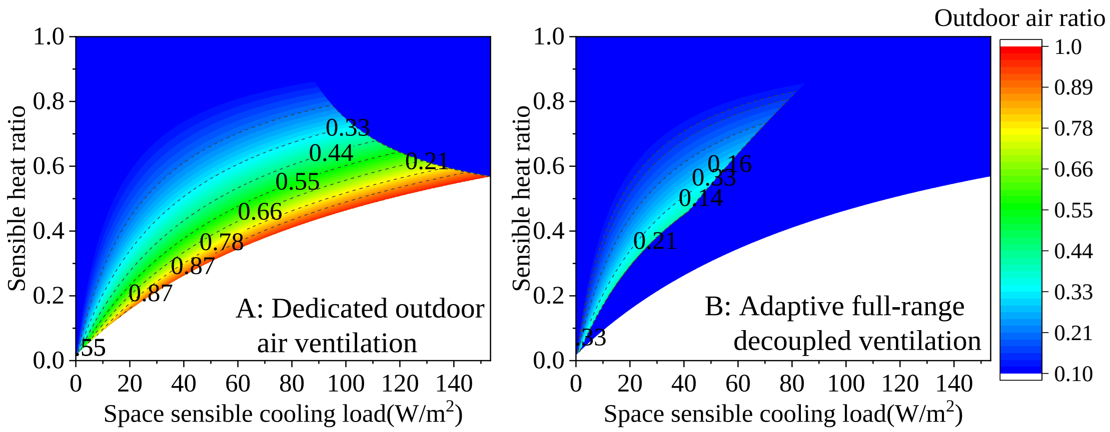

Fig. 7 Outdoor air ratio of two ventilation strategies under Hong Kong design weather condition

(A: Dedicated outdoor air ventilation; B: Adaptive full-range decoupled ventilation)

By comparing the energy performance of the existing and proposed strategies under different internal load regions, the preferred ventilation strategies (i.e. most energy-efficient strategy) in different regions are shown in Fig. 8. Where, blue dot lines are the contour lines of energy saving ratio. The energy saving ratio of the proposed ADV strategy ranges up to 59.2% compared with that of the least energy-efficient strategy. In Region 1, the proposed ADV strategy as well as the PD and DV strategies have the superior energy performance compared with the IC strategy, where the space has comparatively high sensible heat ratio. In Region 2, the proposed ADV strategy and the DV strategy have the superior energy performance compared with the other two strategies, where the space has medium sensible heat ratio. In Region 3, the proposed ADV strategy as well as the IC and PD strategies are the superior options, where the space has low sensible heat ratio.

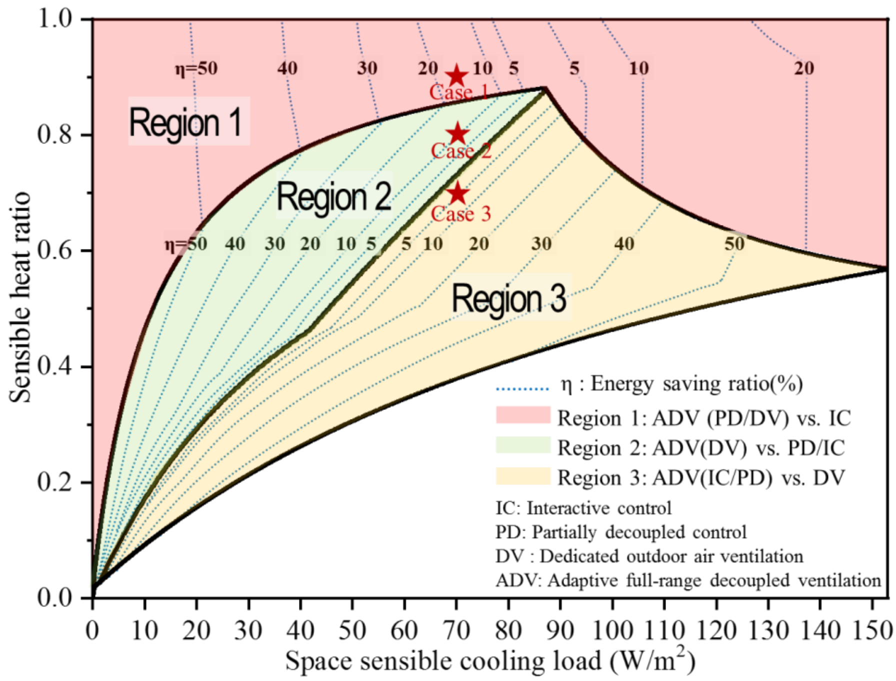

Fig. 8 Preferred ventilation modes/strategies and their energy saving ratio in different internal load regions under Hong Kong design weather condition

It can be summarized that although the energy performance of some existing ventilation strategies is as good as that of the proposed strategy in some of the three regions, the proposed strategy is always the superior strategy in all regions or working conditions concerned. Table 4 shows the preferred ventilation strategies in different regions of working conditions according to the energy performance comparison results. It can be seen that the proposed ventilation strategy offers superior energy-efficiency over the full range of internal loads under the Hong Kong design weather condition. It can be also observed that the proposed strategy has adopted the advantages of best ventilation strategies, which are different in different internal load regions.

Table 4 Superior ventilation strategies in different internal load regions under Hong Kong design weather condition

Strategy	Region 1	Region 2	Region 3
ADV	**√**	**√**	**√**
IC			√
PD	√		√
DV		√	

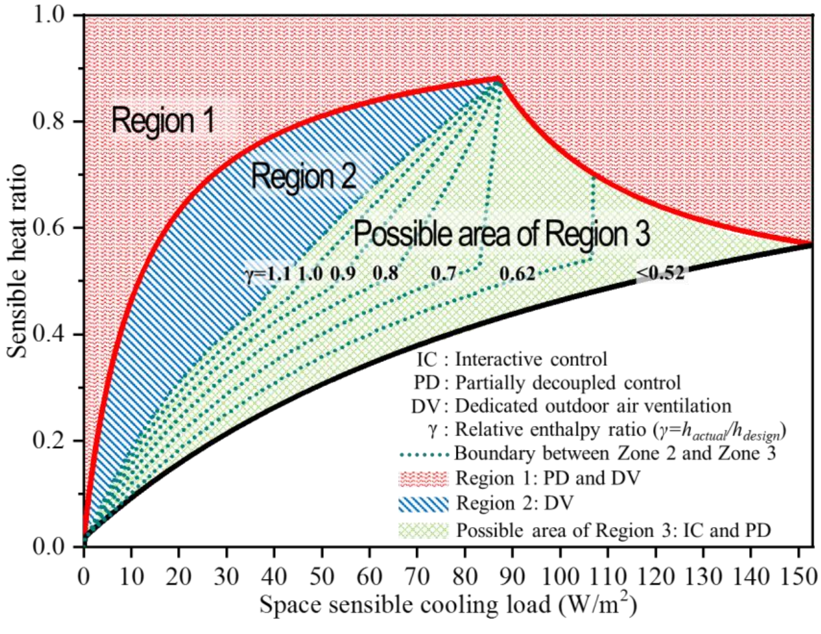

Fig. 9 Boundary changes of preferred ventilation modes/strategies of the ADV strategy in different internal load regions under different outdoor weather conditions

The preferred ventilation modes of ADV strategy in different internal load regions under Hong Kong design weather condition is presented in Fig. 8. However, it is also worth noticing that the boundary for selecting superior ventilation mode is affected significantly by the outdoor weather condition. As mentioned in Section 2.2, the proposed ADV strategy consists of three operation modes and it operates at its most economical mode at a particular state. To illustrate how most economical mode is affected by different indoor and outdoor weather conditions, a further analysis is therefore conducted. Fig. 9 presents the boundary of preferred ventilation strategies in different internal load regions, which moves when the outdoor weather condition varies. This map is made under the condition when the absolute humidity of the outdoor air is no less than that of the indoor air or the economizer is not applicable. Similar to Fig. 8, in Region 1, the PD and DV modes/strategies are the preferred modes, i.e. better than the IC mode/strategy. In Region 2, the DV mode/strategy is the preferred mode, i.e. better than the IC and PD modes/strategies. In Region 3, the IC and PD modes/strategies are the preferred options. The key difference between Fig. 9 and Fig. 8 is that, in Fig. 9, the boundary between Region 2 and Region 3 moves when outdoor enthalpy changes. When the enthalpy of outdoor air reduces, this boundary line moves down and the area of Region 2 increases. Where, relative enthalpy ratio (γ) of the outdoor air is defined as the ratio of its actual enthalpy (h_{actual}) to that of the air at the design weather condition (h_{design}) (29.1 °C, 84.4%), as shown in Eq. (18). When the relative enthalpy ratio of the outdoor air is 0.62, which is equal to the enthalpy of the indoor design condition (23 °C, 65%), the area where the DV mode/strategy is the preferred mode covers most of the possible working conditions as indicated by 88.4% of the total operation area in the figure. It is worth noticing that, when the relative enthalpy ratio of the outdoor air decreases to 0.52 or below (equivalent to that at (19.2°C, 70%) or (17.3 °C, 85%)), the DV mode/strategy is the preferred mode for all possible working conditions if no economizer is adopted and the Region 3 disappears practically.

$$\gamma = \frac{h_{actual}}{h_{design}} \tag{18}$$

4.2 Energy performance under different weather conditions

Fig. 10 (A)-(D) present the system performance maps for the IC, DV, PD strategies and the proposed ADV strategy at a given typical internal load condition (Qs=90W/m^2, SHR=0.8). The total electrical load (Eq.7) of each strategy is calculated under 1,806,301 working points/conditions (6001×301), with the interval of 0.01°C and 1×10^{-4} in terms of outdoor air temperature and humidity ratio, respectively.

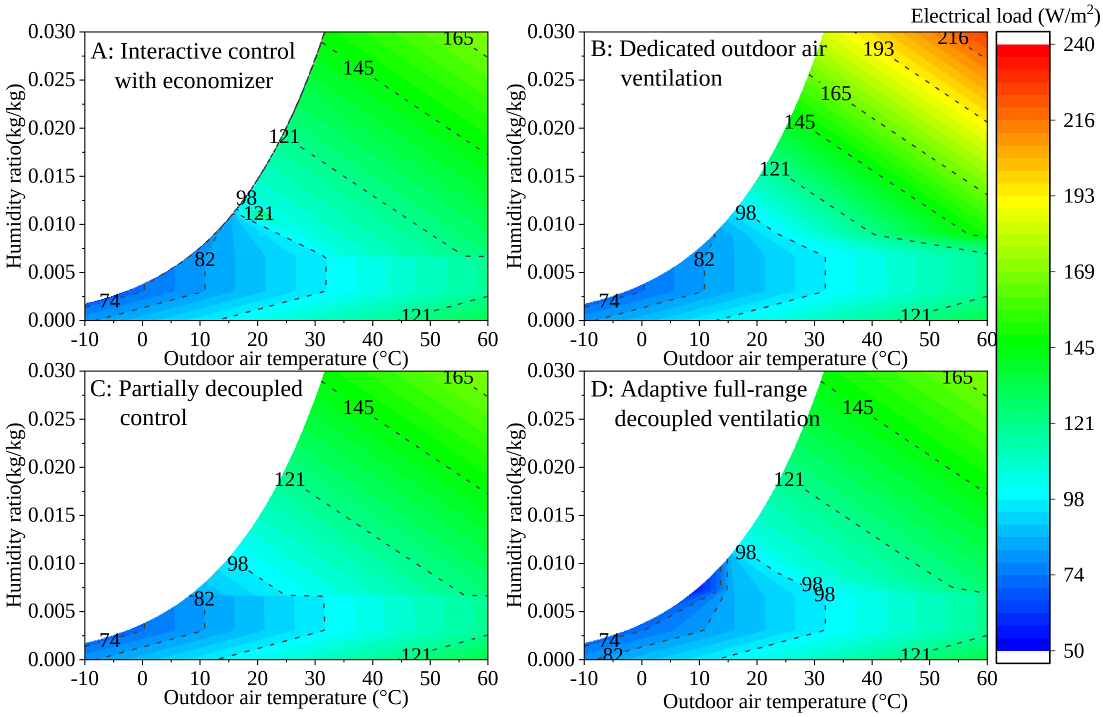

Fig. 10 Space air-conditioning energy maps of four ventilation strategies under a typical internal load condition (Qs=90W/m^2/SHR=0.8)

(A: Interactive control with economizer; B: Dedicated outdoor air ventilation; C: Partially decoupled control; D: Adaptive full-range decoupled ventilation)

An adaptive economizer, which involves three economizer operation modes, marked as "following sensible load" (FS), "following latent load" (FL) and "lower-limit humidity control" (LL), is investigated and adopted by the proposed ADV strategy. Fig. 11(A) shows the applicable weather condition regions for activating the adaptive economizer. The adaptive economizer is not applicable in Regions 1-3 since the enthalpy of outdoor air is comparatively high (i.e. Region 1 and 2) or the minimum outdoor airflow is enough for "free dehumidification" (i.e. Region 3). By contrast, the adaptive economizer is beneficial and more energy-efficient in Regions 4-6 compared with that of the ventilation strategy without the economizer. The preferred economizer modes in different weather condition regions are shown in Fig. 11(B), which is actually the enlarged figure of Regions 4-6 in Fig. 11(A). The energy saving ratio of the proposed ADV strategy adopting the adaptive economizer ranges up to 27.4% compared with that of the ADV strategy without adopting an economizer. In Region 4, the proposed ADV strategy adopting the FL mode has the superior energy performance, where outdoor airflow rate is adjusted to control the indoor humidity at its upper limit. In Region 5, the proposed ADV strategy adopting the FS mode has the superior energy performance, where the outdoor airflow rate is adjusted to control the indoor temperature while the indoor humidity varies in an allowable range. In Region 6, the proposed ADV strategy adopting the LL mode is the superior option, where the outdoor airflow rate is adjusted to control the indoor humidity to its lower limit.

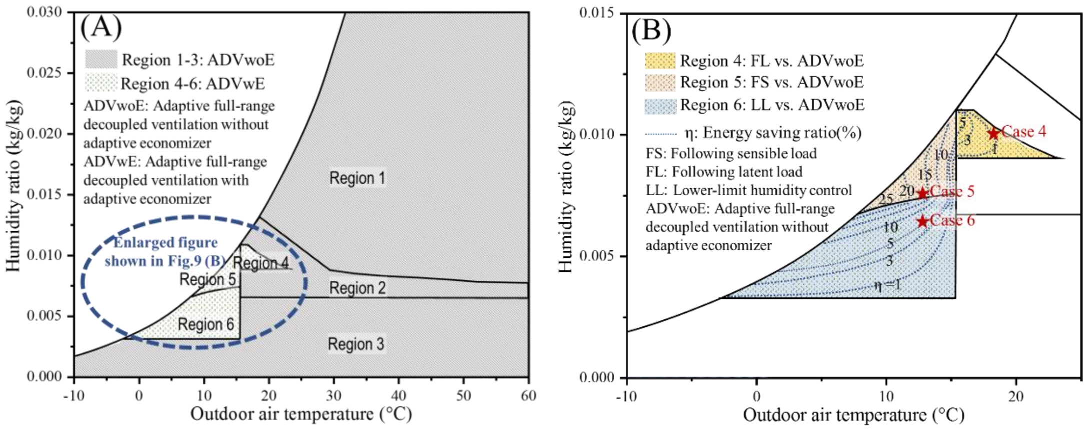

Fig. 11 Applicable weather condition regions for adaptive economizer and preferred economizer modes at a typical internal load condition(Qs=90W/m^2/SHR=0.8)

Table 5 shows the preferred ventilation strategies in different regions of weather conditions according to the energy performance comparison results. It can be seen that the proposed ventilation strategy offers superior energy-efficiency over the full range of weather conditions under a typical internal load condition (Qs=90W/m^2/SHR=0.8). In Region 1, the proposed ADV strategy as well as the IC and PD strategies are the most energy-efficient compared with the DV strategy, where the outdoor air has high enthalpy. In Region 2, the proposed ADV strategy and the DV strategy have superior energy performance compared with the other two strategies, where the outdoor air enthalpy is not high. In Region 3, all the ventilation strategies have the same energy performance, where the minimum outdoor airflow is enough for “free dehumidification” since the outdoor air is extremely dry. In Region 4, the proposed ADV strategy and the IC strategy offer superior performance compared with the other two strategies due to the adoption of the economizer. In Regions 5-6, the proposed ADV strategy is the superior option since it adopts the adaptive economizer and makes full use of the “free cooling and dehumidification” capacity of the outdoor air.

Table 5 Superior ventilation strategies in different weather conditions and the typical internal load condition

Strategy	Region 1	Region 2	Region 3	Region 4	Region 5	Region 6
ADV	**√**	**√**	**√**	**√**	**√**	**√**
IC	√		√	√		
PD	√		√			
DV		√	√			

4.3 Limitations of existing strategies and benefits of proposed strategy

To summarize the limitations of the existing ventilation strategies and the benefits of the proposed strategy, six typical and representative cases are selected to assess and compare the energy performance of the proposed and the existing most-updated ventilation strategies, as shown in Table 6. It is worth noting that, in this table, the electrical loads of the MAU/AHU cooling coils, AHU heater and AHU humidifier are their cooling/heating loads (Eqs.14-17) divided by the cooling/heating COPs respectively (i.e., 2.5/1.0 in this case study). The powers of MAU/AHU fans are calculated using Eq. (8).

The energy performance of different ventilation strategies under different internal load conditions is assessed in Case 1-3 (marked in Fig. 8), while their performance under low/medium/high internal latent load conditions is compared. Similarly, the energy performance of different ventilation strategies under different outdoor conditions is assessed in Case 4-6 (marked in Fig. 11(B)), where the space sensible cooling load (SSCL) and sensible heat ratio (SHR) in three cases are set the same.

Table 6 Electrical loads of studied cases under different internal load and weather conditions

Case	Internal load SSCL (W/m^2)	Internal load SHR	Weather condition Temperature (°C)	Weather condition RH (%)	Ventilation strategy	Outdoor air ratio	Electrical load/ power (W/m^2) MAU cooling	MAU fan	AHU cooling	AHU heating	AHU humidifier	AHU fan	Total
1	70	0.9	29.1	84.4	IC	0.1	25.90	4.16	52.04	14.70	0.00	35.03	131.84
					ADV/PD/DV	0.1	35.61	4.16	35.57	0.00	0.00	35.03	**110.37**
2	70	0.8			IC	0.1	25.90	4.16	58.18	20.11	0.00	35.03	143.38
					PD	0.1	38.44	4.16	45.64	20.11	0.00	35.03	143.38
					ADV/DV	0.146	56.04	6.05	31.01	0.00	0.00	35.03	**128.13**
3	70	0.7			DV	0.25	96.11	10.38	23.16	0.00	0.00	35.03	164.67
					PD	0.1	38.44	4.16	53.60	27.24	0.00	35.03	158.48
					ADV(FL)/IC	0.1	25.90	4.16	66.14	27.24	0.00	35.03	**158.48**
4	90	0.8	18	78	FS	1	0.00	41.46	28.89	0.00	0.00	35.03	105.37
					PD	0.1	7.74	4.16	48.82	2.97	0.00	35.03	98.72
					DV	0.187	14.51	7.78	35.87	0.00	0.00	35.03	93.19
					ADV(FL)/IC	0.272	0.00	14.49	42.64	0.00	0.00	35.03	**92.15**
5			13	80	PD	0.1	1.66	4.16	45.95	2.97	0.00	35.03	89.77
					LL	0.937	0.00	38.86	0.00	12.81	0.00	35.03	86.70
					IC/FL	0.118	0.00	4.86	43.05	0.00	0.00	35.03	82.94
					DV	0.118	1.95	4.86	41.10	0.00	0.00	35.03	82.94
					ADV(FS)	0.85	0.00	35.24	0.00	0.00	0.00	35.03	**70.27**
6			13	70	FS	0.85	0.00	35.24	0.00	0.00	31.57	35.03	101.84
					IC/FL	0.1	0.00	4.16	44.11	0.00	0.00	35.03	83.30
					PD/DV	0.1	1.66	4.16	42.44	0.00	0.00	35.03	83.30
					ADV(LL)	0.353	0.00	14.65	29.21	0.00	0.00	35.03	**78.89**

Case 1 represents the condition when indoor has a relatively low latent load. Where all the latent gains can be removed by the MAU with the minimum outdoor airflow. The electrical load of the IC strategy is the largest, which is 16.3% more than that of the proposed ADV strategy as well as the PD and DV strategies, due to the sub-cooling and reheating process for removing the latent gains. Case 2 represents the condition when indoor has a medium latent load. Where only part of the latent gains can be removed by the MAU with the minimum outdoor airflow using the PD strategy and therefore the reheating occurs. The electrical loads of the IC strategy and the PD strategy are the same, which are 10.6% more than that of the proposed ADV strategy and the DV strategy. Case 3 represents the condition when indoor has a high latent load. The DV strategy has higher electrical load compared with the proposed ADV strategy as well as the IC and PD strategies since the amount of the AHU electrical load decrease overtakes that of increased the MAU electrical load due to excessive high-enthalpy outdoor airflow. Case 4 represents the condition when the weather is cool and dry. The FS strategy has the highest electrical load, which is 12.6% more than that of the most energy-efficient strategy (the proposed ADV strategy adopting the FL mode or the IC strategy), due to the excessive outdoor air intake. In this case, the electrical loads of the PD and DV strategies are also 6.1% and 1.1% more than that of the proposed ADV strategy respectively, since these two strategies cannot make full use of the “free dehumidification” capacity of the outdoor air. Case 5 represents the condition when the weather is cold and dry. The energy electrical loads of the PD and LL strategies are 21.7% and 20.0% more than that of the proposed ADV strategy (adopting the FS mode) respectively, due to the activation of heating processes. In addition, the electrical loads of the DV, IC and FL strategies are 15.3% more than that of the proposed ADV strategy (adopting the FS mode), since these three strategies cannot make full use of the “free cooling” capacity of the outdoor air. Case 6 represents the condition when the weather is cold and extremely dry. The FS strategy has a highest electrical load, which is 22.5% more than that of the proposed ADV strategy (adopting the LL mode) since the humidification needs to be activated due to the excessive dry outdoor air intake. In this case, the electrical loads of the PD and DV strategies are also 5.3% more than that of the proposed ADV strategy (adopting the LL mode), since these two strategies cannot make full use of the "free cooling and dehumidification" capacity of the outdoor air.

By comparing the energy performance of six typical working conditions (Case 1-6), it can be concluded that each of the existing ventilation strategies has its limitations under certain internal load or weather conditions. The proposed ADV strategy, having incorporated the advantages of different ventilation strategies and an adaptive economizer, could offer the superior energy performance in all cases by setting optimal outdoor airflow and activating the most energy-efficient operation modes of the adaptive economizer.

4.4 Comparison of annual energy consumptions

Using the hourly sensible and latent cooling loads calculated by TRNSYS 18 as mentioned in Section 3.2, the MATLAB 2017 is then used to calculate the hourly energy consumption when using different ventilation strategies according to Eqs. (7-17) and their working principles as shown in Fig. 2 and Fig. 3. For the purpose of energy saving, when the economizer modes are not adopted (i.e. adopting IC, PD and DV modes), the indoor temperature and humidity are set at their higher limits (i.e. 23°C, 65%), while at the economizer modes, the indoor temperature and humidity vary within the acceptable ranges (i.e. 20±3°C, 55±10%) according to the actual outdoor and load conditions. Through the hourly energy simulation over a typical year in Hong Kong climate, the energy consumptions using different ventilation strategies for the air-conditioning of the cleanrooms from January to December are estimated as shown in Fig. 12. During the summer months (May-September) when outdoor weather is extremely hot and humid, the energy consumption of the DV strategy is the highest due to the introduction of the excessive high-enthalpy outdoor air, while the consumption of the PD strategy is much lower and the consumption of the proposed ADV strategy is even lower (the lowest). However, during the winter and transition seasons (October-April), the energy consumption of the PD strategy is the highest, followed by the IC strategy, while the consumptions of the proposed ADV strategy and the DV strategy are almost the same and lowest. The reason is that, during the transition seasons and winter, the intake of higher outdoor airflow can provide more “free cooling and dehumidification” and thus reduces the cooling energy consumption although it may increase the MAU fan energy consumption.

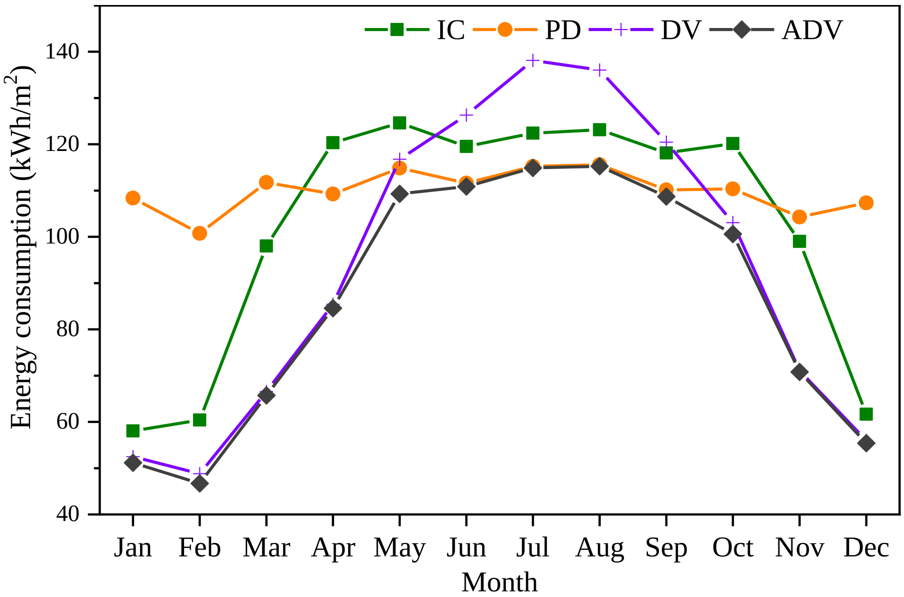

Fig. 12 Comparison of monthly energy consumptions of proposed and existing ventilation strategies

The annual energy consumptions of different air-conditioning components using the proposed and existing ventilation strategies are compared in Table 7. The overall annual energy consumption of the proposed ADV strategy is 21.64 %, 15.63% and 7.77% less than that of the PD, IC and DV strategies respectively. For the PD strategy, the heating and annual energy consumptions are the highest among the four ventilation strategies since this strategy uses the least “free cooling and dehumidification” capacity of the outdoor air. For the IC strategy, although the MAU energy consumption is the lowest, the AHU energy consumption is the highest compared with the other three ventilation strategies. For the DV strategy, although the AHU energy consumption is the lowest, the MAU energy consumption is the highest compared with the other three ventilation strategies. Due to the high internal latent load, the energy consumption of the humidifier is the same for all four strategies since it only opens in extremely dry weather conditions.

Table 7 Comparison of annual energy consumptions of proposed and existing ventilation strategies

Energy consumption (kWh/m^2)	ADV	IC	PD	DV
MAU cooling	289.92	84.40	166.01	573.44
AHU cooling	257.12	503.64	489.39	121.17
AHU heating	86.86	255.06	320.93	2.25
AHU Humidifier	0.32	0.32	0.32	0.32
MAU fan	93.19	75.49	36.34	117.29
AHU fan	306.60	306.60	306.60	306.60
Total	1034.00	1225.51	1319.58	1121.06

The differences between the four ventilation strategies in terms of energy performance are mainly due to the volume of outdoor airflow induced for space cooling and dehumidification. Fig. 13 presents the outdoor airflow settings of the four strategies. The proposed ADV strategy always induces the optimal outdoor airflow for the air-conditioning systems. For the PD strategy, the outdoor airflow is always set as the lower limit (2 ACH in this case) while the outdoor airflow varies for the other three strategies. Using the IC strategy, the outdoor airflow is kept at the lower-limit during the summer period and it may rise to 20 ACH (outdoor/total supply air ratio equals 1) during the winter and transition seasons due to the utilization of the enthalpy-based economizer. It is worth noting that, in most time of the hot summer, the outdoor airflow of the proposed ADV strategy maintains at the lower limit while it is rather high when using the DV strategy. For maintaining the indoor relative humidity under high internal latent load conditions in the hot summer period, the DV strategy requires the intake of excessive high-enthalpy outdoor airflow for space dehumidification and this explains why the energy consumption of this strategy is highest in the period.

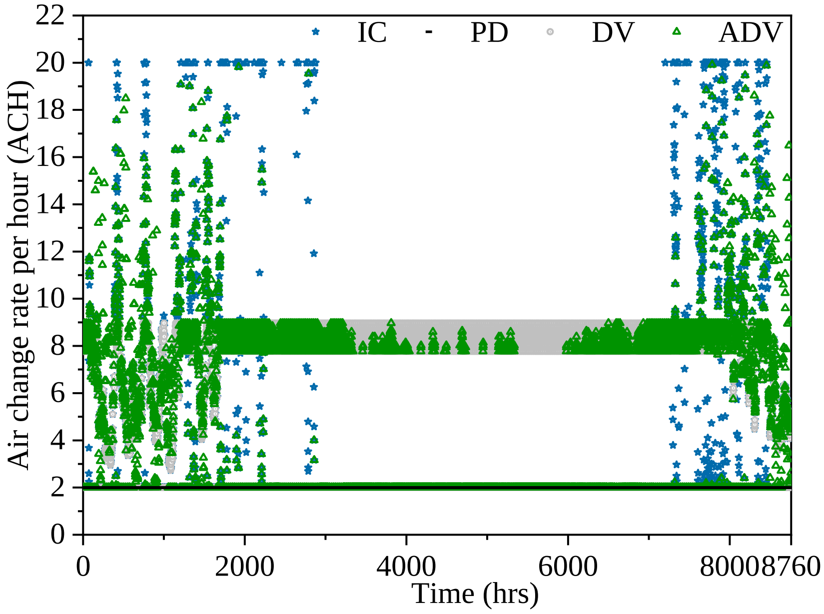

Fig. 13 Outdoor airflow rate settings for four ventilation strategies

To illustrate how the ADV strategy incorporates the advantages of using different ventilation strategies and the adaptive economizer, the operation hours (frequencies) of different operation modes in the studied year is listed in Table 8.

Table 8 Annual operation time of different operation modes using ADV strategy

Operation mode	DV	PD	PD/DV	FS	FL	LL
	non-economizer mode			economizer mode		
Operation time (h)	2,750	2,527	1,958	649	726	150
(%)	31.39	28.85	22.35	7.41	8.29	1.71

In the studied year, the non-economizer and economizer modes account for 7235 h (82.59%) and 1525 h (17.41%) respectively and the DV mode accounts for the longest hour. The PD/DV modes represent that there is no difference between the PD mode and DV mode since the indoor moisture load can be handled at the lower limit (i.e. 2 ACH) of outdoor air flowrate. It is worth noticing that the IC mode is not used in the case studied because the energy performance of IC mode in the studied year is the same or worse than that of other operation modes.

5 Air-conditioning system design and comparative economic analysis

The implementation of the ventilation strategies requires the air-conditioning systems to be designed properly. This section, therefore, investigates the design of the air-conditioning subsystems/components and economic analysis of the proposed ventilation strategy compared with the most updated existing ventilation strategies.

5.1 Air-conditioning component design

Due to the very strict requirements on cleanroom temperature and relative humidity controls, it is assumed that cooling and heating capacities of their air-conditioning components must meet the environment control requirement of the cleanrooms throughout a year. Therefore, the hourly maximum cooling and heating demands are used for determining the cooling coil and electric heater design capacities. The maximum outdoor and total supply airflow rates are used to determine the MAU and AHU fan design capacities (powers) respectively. The maximum humidification load is used to determine the design capacity of the humidifier.

The required design (maximum) capacities of the air-conditioning components for implementing the proposed and the existing ventilation strategies are listed in Table 9. For the DV strategy, since the largest amount of outdoor air is needed for space dehumidification even during hot and humid periods, the required MAU cooling coil design capacity is the largest, which can be as large as 6 times of that for the IC strategy. However, it requests the smallest AHU cooling coil design capacity. For the IC strategy, due to the use of the enthalpy-based economizer, the MAU needs the largest design fan (power) capacity to induce 100% outdoor airflow. For the PD strategy, the heater of the largest design capacity is needed since the heater may need to be switched on due to insufficient outdoor airflow for space dehumidification even in cool and dry seasons. For the proposed ADV strategy, the required design capacities of the MAU/AHU cooling coils, heater and MAU fan are in-between that of three existing strategies. It is worth noting that since the weather in Hong Kong is humid throughout the year, the required design capacity of the humidifier is the same for all the four strategies. The AHU fan design capacities for all the four strategies are also the same as it is determined by the minimum total supply airflow.

Table 9 Air-conditioning component design capacities for implementing different ventilation strategies

Ventilation strategy	Required capacity (kW)					
	MAU cooling coil	AHU cooling coil	AHU heater	AHU Humidifier	MAU fan	AHU fan
ADV	61.39	31.88	10.09	2.01	7.64	6.48
IC	15.36	37.68	14.83	2.01	7.67	6.48
PD	21.16	31.88	16.16	2.01	0.77	6.48
DV	93.39	22.20	2.92	2.01	3.44	6.48

The required capacities of different ventilation strategies listed in Table 9 can be also used as a check reference when retrofitting the existing air-conditioning subsystems. Concerning the system design options in this case study, the full implementation of the ADV strategy requires the following modifications.

For the system designed for the IC strategy, a larger cooling capacity of the MAU (i.e. 4 times that of the existing capacity) is needed while other components can remain unchanged. For the system designed for the PD strategy, a larger cooling capacity of the MAU (i.e. 3 times that of the existing capacity) and a larger volume variable-speed MAU fan are needed while other components can remain unchanged. For the system designed for the DV strategy, all the components should be selected with larger capacities except the cooling coil of the MAU.

5.2 Economic analysis and comparisons

The economic performance of the air-conditioning systems when adopting different ventilation strategies is assessed using the cumulative total cost C_t, shown in Eq. (19).

$$C_t = \sum_{i=0}^{n} C_{in,i} + C_{e,i} + C_{ma,i} \tag{19}$$

where, C_t refers to the sum of the initial cost C_{in}, the operation cost (electricity cost) C_e and the maintenance cost C_{ma} accumulated over the increased length of service n. The cumulative total cost at the start of services (i.e. year $i=0$) represents the initial cost. C_{in}

is obtained using the sizes of the components. C_e is calculated from the electricity consumption and local electricity price (USD/kWh). C_{ma} is calculated based on the maintenance cost database (USD/m^2).

In the initial cost calculation, the air-side air-conditioning components considered and the associated costs are as follows: cooling coil (369.1 USD/kW), electric heater (93.75 USD/kW), electric steam humidifier (183.75 USD/kg/h) and axial fan (684.81 USD/kW). Where, the cost estimates are based on RSMeans Mechanical Cost Data [33]. The initial cost of the economizer is estimated as 8 USD/m^2 of the floor area considering the installation of additional dampers, sensors and actuators [34]. The electricity cost is calculated according to the local electricity price in Hong Kong, which takes the average price of 0.141 USD/kWh [35]. It is assumed that the maintenance cost (or repair cost) for these four strategies is the same and set as 3.66 USD/m^2 per year according to the mean value of ASHRAE Owning and Operating Costa Database [36].

Fig. 14 shows the cumulative total costs of different ventilation strategies. The initial cost of the DV strategy is highest, followed by the proposed ADV strategy and the IC strategy, while it is the lowest for the PD strategy. The proposed ADV strategy requires the lowest annual operation cost since it offers the best energy performance over the full range of internal load and weather conditions. Compared with the PD and IC strategies, although a slightly higher initial cost is required, the cumulative total cost of the proposed ADV strategy is less than that of the PD and IC strategies after 2.7 years and 2.9 years respectively. The cumulative total cost of the proposed ADV strategy is always less than that of the DV strategy.

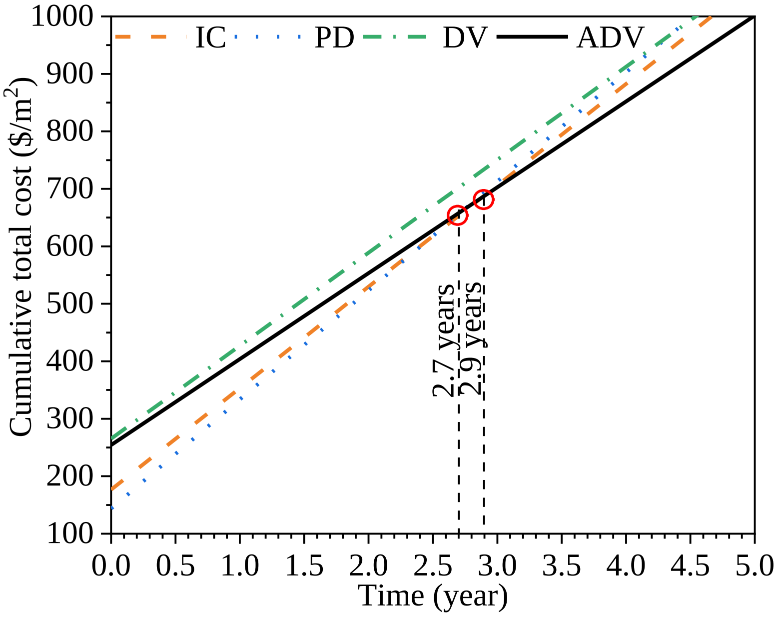

Fig. 14 Cumulative total costs for the four ventilation strategies

6 Conclusion and discussion

This paper proposed an “adaptive full-range decoupled ventilation strategy”. The proper system design for its practical implementation is studied and compared with the other three existing strategies. The proposed ventilation strategy is implemented and tested on the simulation test platform constructed based on the cleanroom air-conditioning systems of a pharmaceutical manufacturing building located in Hong Kong. The system energy performance and economic analysis of the air-conditioning systems adopting the proposed ventilation strategy are also investigated and compared with the cases when the three existing strategies are used.

Based on the results of the test and investigation, some detailed conclusions can be drawn as follows:

- Compared with the existing ventilation strategies, the proposed strategy has superior energy performance over the full range of internal load and weather conditions. The overall annual energy consumption of the proposed ADV strategy is 21.64 %, 15.63% and 7.77% less than that of the partially decoupled control strategy, the interactive control strategy and the dedicated outdoor air ventilation strategy, respectively.

- The proposed new strategy requires appropriate system design since the required design capacities of the air-conditioning components/subsystems are different from that of the existing ventilation strategies. For the proposed strategy, the design capacities of the MAU/AHU cooling coils, heater and MAU fan are in-between that of three existing strategies.

- The cumulative total cost of the proposed strategy is always lower than that of the dedicated outdoor air ventilation strategy as the initial cost of the proposed strategy is also lower than that of the dedicated outdoor air ventilation strategy. Compared with the partially decoupled control strategy and the interactive control strategy, although its initial cost is slightly higher, the cumulative total cost of the proposed strategy will be lower than that of these two strategies after 2.7 and 2.9 years respectively since its saving of the operation cost can then compensate the slightly higher initial costs.

It is worth noting that the proposed ventilation strategy can be also implemented in retrofitting the operation/control of existing air-conditioning systems without additional equipment or change of the system configuration. If the capacities of the air-conditioning subsystems can meet the needs in operation, the strategy can be fully implemented and work over the full range of the operation conditions. The strategy may not work in some working condition if the capacity of the certain subsystem is not sufficient and in this case, the strategy can be partially implemented only. For the buildings with a lower ACH requirement (i.e. lower than 20 ACH), such as some exhibition rooms, animal labs, etc., the supply air temperature can be set lower compared with the space in above case. The dehumidification capacity for the AHU would be higher and the control for the indoor environment would be easier than the case of this study. For such applications, the ADV strategy can be also implemented, probably with less impact.

It is also worth noticing that, For the system configuration with one MAU serving for multiple AHUs, the system design process would be similar but the design of the MAU would be different. The diversity of cooling loads of zones served by different AHUs in practical operation should be considered to avoid oversizing the MAU.

In this study, the performance of the "adaptive full-range decoupled ventilation strategy", as well as the other strategies for comparison, is evaluated under the condition that ideal controls are adopted to allow the systems to operate according to the working principles of the ventilation strategies. The online control strategy is needed for ensuring the actual operation of the ventilation systems to follow the operation principle of the proposed ventilation in actual applications according to actual load and weather conditions as well as the subsystem capacities.

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