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An adaptive full-range decoupled ventilation strategy for buildings with spaces requiring strict humidity control and its applications in different climatic conditions

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Abstract: The buildings with spaces, requiring strict temperature and humidity controls, such as cleanrooms, are usually very energy-intensive due to improper system design and operation. To address this challenge, a novel "adaptive full-range decoupled ventilation strategy" (ADV strategy) is proposed, which incorporates the advantages of different operation modes and offers energy-efficient operation. In this paper, the energy and economic performance as well as the recommended operation modes of the ADV strategy are investigated under different climatic conditions. The energy performance of a typical pharmaceutical cleanroom air-conditioning system is evaluated by simulation tests in nine cities in five typical climate zones. The test results show that adopting the ADV strategy can achieve 6.8 to 40.8% energy savings compared with that of a most commonly used existing ventilation strategy. Dedicated outdoor air ventilation (DV) is recommended as the main mode of the ADV strategy in severe cold/cold/moderate climate zones while the following sensible load (FS) is recommended as the main economizer mode in hot/temperate climate zones. For existing system retrofits and new system designs, the payback periods are less than 4 years and 2 years respectively in most climates when the ADV strategy is fully implemented.

Keywords: ventilation strategy; cleanroom; air-conditioning; energy conservation; design; retrofitting.

Nomenclature

ADV	Adaptive full-range decoupled ventilation
AHU	Air-handling unit
C	Cost (CNY)
CAP	Capacity (kW)
c_p	Air specific heat ratio (kJ/(m^3·°C)
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
DV	Dedicated outdoor air ventilation
E	Electrical load (kW)
FL	Following latent load
FS	Following sensible load
h	Enthalpy of air (kJ/kg)
h_{fg}	Latent heat of vaporization (kJ/kg)
IC	Interactive control
Inc	Investment cost (CNY)
LL	Lower-limit humidity control
m	Air mass flowrate (kg/s)
MAU	Make-up air-handling unit
PD	Partially decoupled control
Δp	Total pressure rise (kPa)
Q_{sen}	Space sensible cooling load (kW)
Q_{tot}	Space total cooling load (kW)
$Q_{cc,MAU}$	Cooling coil cooling load of make-up air-handling unit (kW)
$Q_{cc,AHU}$	Cooling coil cooling load of supply air-handling unit (kW)
$Q_{he,AHU}$	Heating load of supply air-handling unit (kW)
$Q_{hu,AHU}$	Humidification load of supply air-handling unit (kW)
t	Temperature (°C)
V	Air volumetric flow rate (m^3/s)
w	Humidity ratio (kg/kg)
W_{sf}	Supply fan power (kW)
W_{mf}	Make-up fan power (kW)

Greek letters

η_f	Fan efficiency
£	Price

Subscripts

c	Cooling system
cc	Cooling coil
e	Electricity
f	Fan
he	Heating system
hu	Humidification system
mf	Make-up fan
sen	Sensible
sf	Supply fan
tot	Total

1 Introduction

Buildings with spaces requiring strict temperature and humidity controls, such as semiconductor fab, biological lab, pharmaceutical cleanroom, data center, museum and hospital (hereafter denoted as “cleanrooms” for brevity), have been growing very fast in terms of the total floor area and energy consumption. The cleanrooms can be 30 to 50 times more energy-intensive than typical commercial buildings (Mathew, 2008), or 10 to 100 times more energy-intensive than office buildings (Mills et al., 2008; Tschudi & Xu, 2001). However, there was no significant improvement in terms of energy efficiency in cleanroom buildings over the past 17 years due to the insufficient concerns on energy conservation (Kircher, Shi, Patil, & Zhang, 2010; Mills, 1996). In addition, current engineering practices focus more on the improvement of cleanroom environment controls, while the energy conservation issues are not sufficiently considered (Dixon, 2016). The industries, which may need cleanroom production environments, usually involve highly skilled and knowledge-intensive manufacturing processes, and the products are of high profit. The efforts on cleanroom air-conditioning systems in the past are mainly on the means to provide a satisfactory indoor environment in order to maintain the reliable operation of the machines/equipment. Therefore, the energy conservation issue in such applications has not gained sufficient attention. According to a retrofitting work by Shan and Wang (2017), the energy saving of retrofitting an existing cleanroom air-conditioning system is up to 42% by optimizing the control strategies without any hardware modification.

Generally, the energy consumption of air-conditioning in cleanrooms accounts for 40-50% of the total energy consumption, while make-up air handling units (MAUs) consume nearly half of the power load of the air-conditioning systems (Hu & Chuah, 2003). The introduction of proper outdoor airflow is the key issue to reduce the energy use of cleanroom air-conditioning systems. In hot climatic conditions, three existing ventilation strategies, namely "interactive control" (IC) (Jouhara & Ezzuddin, 2013; Kakkar, 2017; Yau, 2010), "dedicated outdoor air ventilation" (DV) (Zhang & Zhang, 2014)" and "partially decoupled control strategy" (PD) (Shan & Wang, 2017), have their own advantages and limitations. IC strategy, which adopts overcooling the reheating processes, is the most commonly used method to remove the indoor moisture. The air is overcooled to lower than its dewpoint and then reheated (if needed) to the required supply air state while the counteraction processes would waste a large amount of energy. DV strategy, which decouples the temperature and humidity control loops, can significantly improve the energy efficiency of air-conditioning systems. The MAU handles all indoor latent load and part of indoor sensible load, while the residual indoor sensible load is handled by the supply air-handling unit (AHU). However, when the indoor latent load is high, the excessive outdoor airflow (e.g., higher than the minimum outdoor airflow for indoor pressurization) is required to be introduced, which indeed needs a large MAU size. PD strategy always induces the minimum required outdoor airflow for decoupling the temperature and humidity controls. However, the overcooling and reheating processes would occur when the indoor latent load is high, due to the insufficient dehumidification capacity of the MAU.

In cool or cold climatic conditions, adopting an economizer system can be a superior option to reduce cooling energy. An economizer system can save energy by strategically introducing outdoor air into occupied spaces based on the comparison of the outdoor and return air states (i.e., temperature, enthalpy, or humidity differences). Many studies have investigated the energy saving potentials of the economizer in commercial and office buildings. Fasiuddin and Budaiwi (2011) found adopting an enthalpy economizer in a shopping mall can achieve energy saving by 3% in Saudi Arabia. Yao and Wang (2010) evaluated potential energy conservation brought by different air-side economizers in office buildings, and the results indicate that energy savings by adopting economizer are about 10-20% under hot climate zones and 5-10% under cold climate zones. Son and Lee (2016) obtained around 10% energy savings by using a differential enthalpy control method in a three-story office building in Korea. However, current economizer operation strategies are generally developed for buildings requiring thermal comfort control only, where the main purpose of utilizing the economizer is for energy conservation. For the buildings with spaces requiring strict temperature and humidity controls, the existing economizer operations are required to be improved, to meet the strict control requirements. For instance, during very dry climatic conditions, the amount of outdoor air should be properly induced to ensure the indoor air humidity is not lower than its lower limit. Although several studies were carried out to demonstrate the potential energy benefits of utilizing economizer in data centers (J. Cho, Lim, & Kim, 2012; K. Cho, Chang, Jung, & Yoon, 2017; Ham, Park, & Jeong, 2015; Lee & Chen, 2013; Park & Seo, 2018), the economizer operation for other types of buildings, especially for the buildings/spaces with the high internal latent load, are still not fully investigated.

To overcome the drawbacks of existing operation strategies, the authors of this paper proposed a novel “adaptive full-range decoupled ventilation strategy” (ADV strategy) (Zhuang, Wang, & Shan, 2019). In hot climatic conditions, the proposed ventilation strategy incorporates the advantages of the PD and DV strategies by compromising properly “inducing more outdoor air” and “sub-cooling and reheating process with minimum outdoor airflow”. In cool or cold climatic conditions, the proposed ventilation strategy adopts three economizer control modes, named “following sensible load” (FS), following latent load” (FL) and “lower-limit humidity control” (LL). Therefore, the proposed ventilation strategy can offer superior energy performance over the full range of internal load and weather conditions. More importantly, the requirements of indoor temperature and humidity can also be attained using this ventilation strategy by proper selection of operation mode. However, the successful and full applications of the ADV strategy in practical applications are still facing the following questions: i) actual energy performance in different climates; ii) the new design and the needs of modifications in retrofitting existing systems and their cost-benefits; iii) proper selection of the optimal alternatives from various operation modes in different climatic conditions.

The objectives of this study therefore include: i) to evaluate energy and economic performance of implementing the ADV strategy in different climate zones, ii) to investigate the air-conditioning system design and the needs of existing system retrofitting, and iii) to identify the most suitable operation modes of the proposed ADV strategy when implemented in different climatic conditions. The application potentials of the proposed ADV strategy are evaluated in a typical pharmaceutical cleanroom air-conditioning system in nine Chinese cities in five main climatic conditions by simulation tests.

2 System concerned and mechanisms of existing and proposed ventilation strategies

2.1 System configuration of a typical cleanroom air-conditioning system

Fig. 1 shows the required air-conditioning components, i.e., a blow-through type MAU and a draw-through AHU, serve for cleanrooms and spaces with strict temperature and humidity controls. This configuration is the most commonly seen in cleanroom air conditioning systems (i.e., ISO8 cleanrooms). The MAU consists of a cooling coil, a fan and filters for conditioning the outdoor air. The AHU contains a cooling coil, a heater, a fan, a humidifier and filters for conditioning the supply air. The chilled water to both MAU and AHU cooling coils is supplied by a chiller plant. Generally, cleanrooms require positive static pressure to avoid the infiltration of pollutants from the outdoor air or adjacent area. The pressure stabilizer (i.e., adjustable automatic relief damper) controls the difference between the total supply and return airflow rates in order to maintain a positive pressure in the cleanroom. The amount of the exhaust air, which needs to be discharged, highly depends on the amount of induced outdoor air (i.e., make-up air) and leakage air. It is worth noting that, in this study, only the major components (e.g., fans, cooling coils in the MAU/AHU, heater and humidifier in the AHU), which would significantly influence the temperature and humidity control performance of ventilation strategies, are involved for the system design and performance evaluation. For the cleanrooms in cold regions, the MAU may also have heaters to avoid freezing problems, and minor preheat energy is needed likely with a proper preheat control scheme. This configuration, therefore, may have a minor difference compared with the systems concerned in this study.

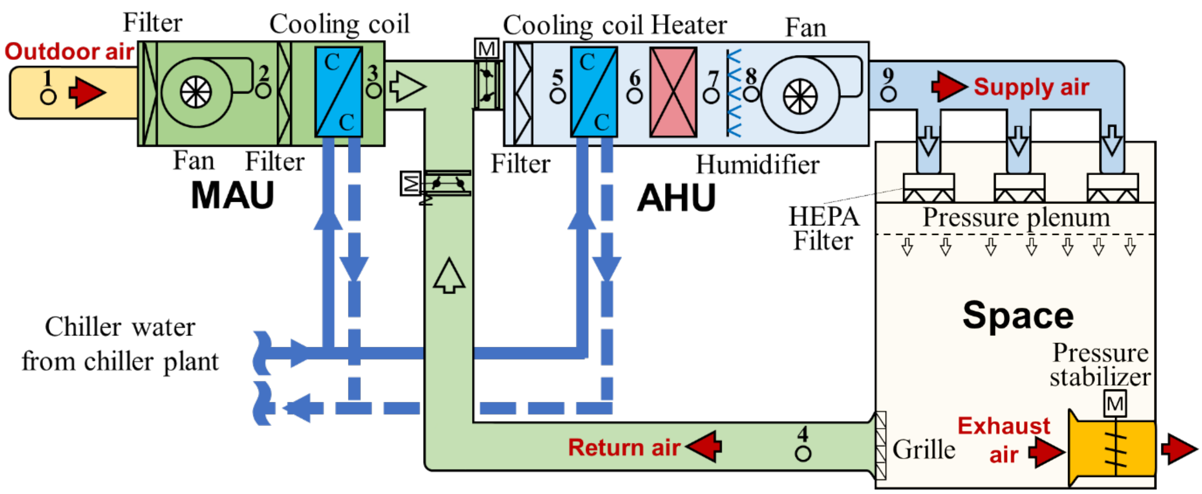

Fig. 1. System configuration of a typical cleanroom air-conditioning system (Zhuang et al., 2019)

2.2 Existing ventilation strategy (IC strategy) and its limitations

The cleanroom air-conditioning system is originally designed using interactive control and an enthalpy-based economizer (IC strategy). In hot weather conditions (Fig. 2A), the minimum required outdoor airflow is introduced by the MAU with a certain temperature rise due to heat generation from the MAU fan motor (1→2). The MAU cools the outdoor air down to a near the indoor air enthalpy

(2→3), at its apparatus dew point. The cooled outdoor air is then mixed with the recirculation air in an AHU (3→5, 4→5) before being further handled by the AHU cooling coil for dehumidification (5→6) and heated by the AHU heater (6→7, if needed) to reach the required AHU supply air state. In cool/cold weather conditions, an enthalpy-based economizer is activated to induce outdoor air for cooling/dehumidification (e.g., 100% outdoor air) according to the enthalpy differential of outdoor and indoor air, as shown in Fig. 2B. The amount of outdoor air to be introduced is determined based on the required supply air state (State 9). It can be seen that for the IC strategy, most dehumidification load is handled by AHU consorting to the overcooling and reheating processes, which is an energy-intensive way for humidity control.

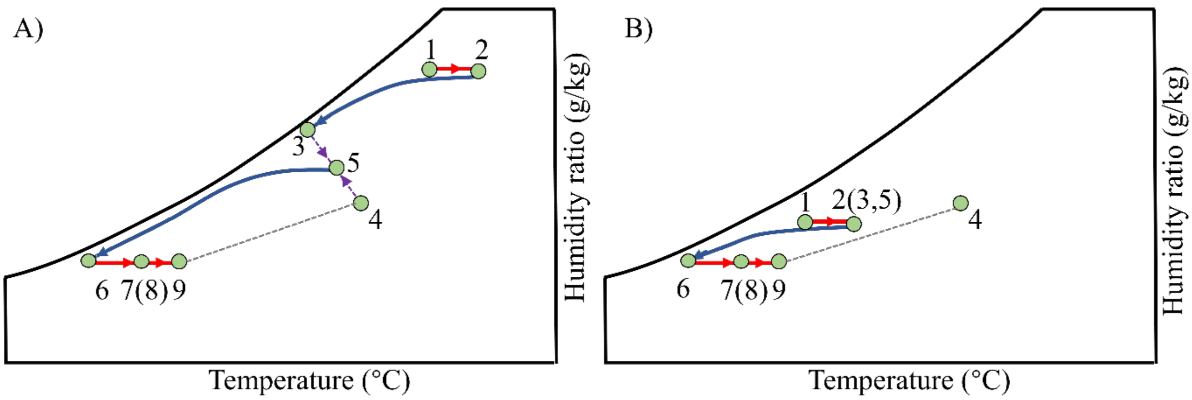

Fig. 2. Air handling processes of IC strategy in A) hot weather conditions B) cool/cold weather conditions

2.3 Proposed ventilation strategy (ADV strategy) and its benefits

To full use of the cooling/dehumidification capacity of outdoor air and avoid unnecessary reheating processes, an ADV strategy is proposed (Zhuang et al., 2019), which contains non-economizer operation modes (i.e., PD and DV modes) and economizer operation modes (i.e., FL, FS and FL modes) for different application situations. The mechanisms and descriptions of non-economizer and economizer operation modes of the ADV strategy are highlighted in Table 1. Due to the combination of the advantages of different operation modes, the ADV strategy can offer superior energy performance over the full range of internal load and weather conditions by avoiding sub-cooling and reheating as far as beneficial via the best use of MAU and economizer for cooling and dehumidification.

Table 1 Mechanisms and descriptions of operation modes of ADV strategy (Zhuang et al., 2019)

Operation mode		Mechanism	Description	Limitations/challenges
Non-economizer operation mode	Dedicated outdoor air ventilation (DV)	MAU handles all latent heat and part of space sensible heat while the AHUs remove the rest of space sensible heat	Outdoor air is treated below the indoor air dewpoint Outdoor airflow is adjusted according to the internal latent load	High ventilation energy demand
	Partially decoupled control (PD)	MAU handles all latent heat and part of space sensible heat while the AHUs remove the rest of space sensible heat under low internal load conditions	Outdoor air is treated below the indoor air dewpoint Outdoor airflow is always set at the minimum	Simultaneous cooling and reheating under high internal latent load
Economizer operation mode	Following sensible load (FS)	Control indoor temperature by properly setting the outdoor airflow	Outdoor airflow is adjusted according to the internal sensible load	Humidification might be needed if the internal sensible load is high
	Following latent load (FL)	Control indoor humidity at upper limit by properly setting the outdoor airflow	Outdoor airflow is adjusted to control the indoor relative humidity at upper limit	Can't make full use of the "free cooling" capacity of outdoor air sometimes
	Lower-limit humidity control (LL)	Control indoor humidity at lower limit by properly setting the outdoor airflow	Outdoor airflow is adjusted to control the indoor relative humidity at lower limit	More MAU fan energy consumption

Selecting the best operation mode of ADV strategy involves four steps. In the first step, the required air state can be determined based on the internal load and design indoor conditions. In the second step, the feasibility of operation modes (economizer and non-economizer modes) is assessed based on the weather and load conditions as well as the working principles (i.e., air-handling processes) of the corresponding operation modes. In the third step, the total electrical load of the available operation modes is calculated according to the energy balances (shown in Section 4.2). Eventually, the operation mode with the minimum electrical load would be selected.

3 Climatic zones and selection of test locations

Due to the large land area across a wide range of latitudes and complexities of topography, the climates are of large diversities in China (He, Yang, & Ye, 2014). In terms of the thermal design of buildings, a major climate classification is developed to distinct climatic features (MOHURD, 1993). The five major climate zones of this classification are as follows: severe cold, cold, hot summer and cold winter, temperate as well as hot summer and warm winter. Fig. 3 shows the overall layout of the five major climate zones. Due to varying topology and elevation, the five climate zones are further divided into two or three subregions although they are not shown in the figure (MOHURD, 2015). Nine test locations, including Harbin (severe cold, 45.8°N and 126.8°E), Urumqi (severe cold, 43.8°N and 87.6°E), Beijing (cold, 39.9°N and 116.3°E), Lhasa (cold, 29.7°N and 91.0°E), Shanghai (hot summer and cold winter, 31.2°N and 121.4°E), Chongqing (hot summer and cold winter, 29.9°N and 108.6°E), Kunming (temperate, 25.0°N and 102.7°E), Nanning (Hot summer and warm winter, 22.8°N and 108.4°E) and Hong Kong (Hot summer and warm winter, 22.3°N and 114.2°E), within each of the five climate zones are selected for the analysis.

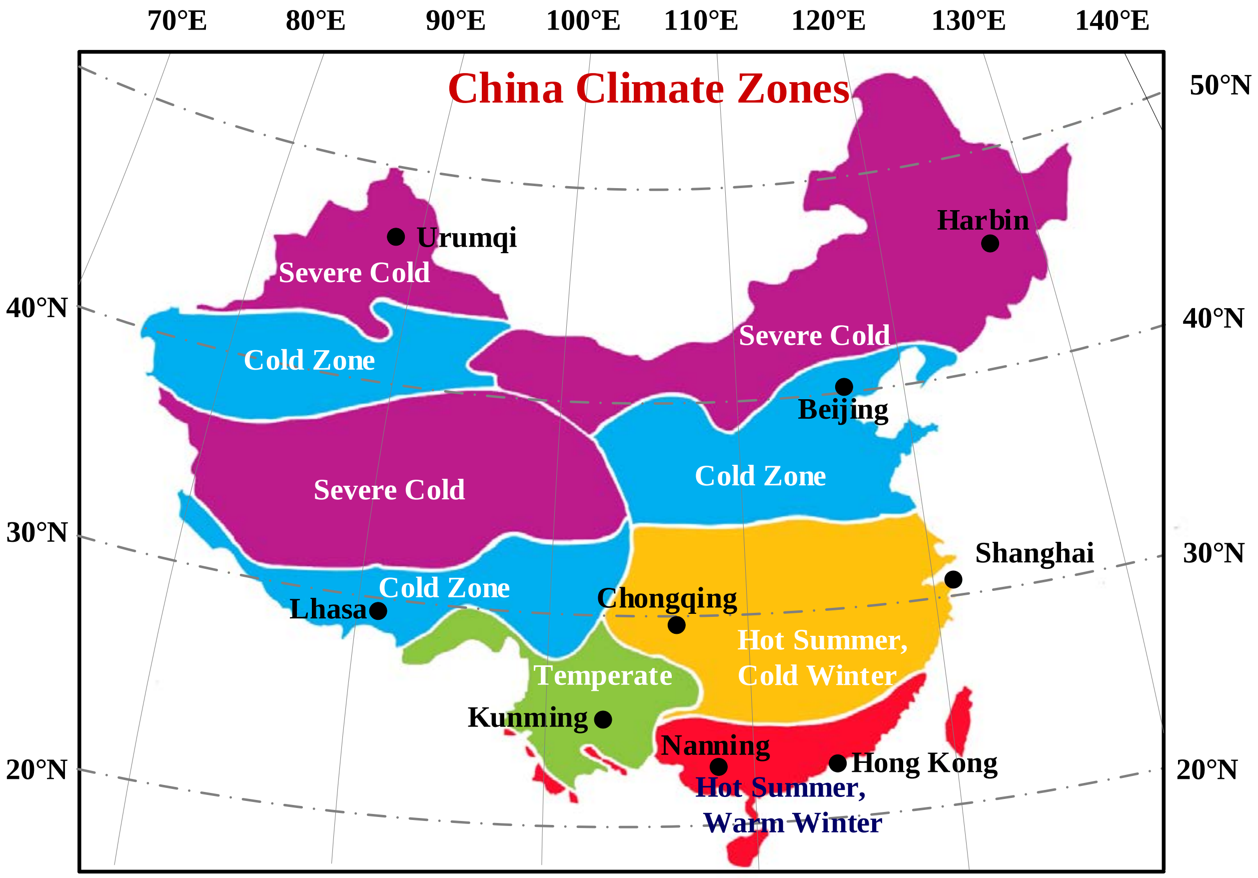

Fig. 3. Geographical distribution of the five major climates and the nine cities (MOHURD, 1993).

4 Test system set-up for energy performance assessment

4.1 Building description, weather and load conditions

4.1.1 Building description

A typical pharmaceutical building is selected for comparative energy studies and the cleanroom spaces are designed as Class ISO 8 cleanrooms based on the ISO standard (2015). The pharmaceutical building model is assigned envelope parameters, ventilation rates, internal loads, operating schedules, as well as indoor temperature and relative humidity setpoints compliant with prescriptive requirements or recommended design values in Chinese building energy efficiency standards (Qingqin & Miao, 2015). The detailed characteristics of the simulated building are shown in Table 2.

The air-conditioning subsystem concerned (i.e., serving the space with a floor area of 369.7 m^2) has the system configuration as shown in Fig. 1, containing the axial fans, chilled water-cooling coils, an electric heater, an electric steam humidifier and other accessories. It is worth noting that due to the positive pressure control requirement on cleanrooms, no infiltration is considered in the load calculation.

Table 2 Characteristics of the simulated building

Description	Parameter	Value
Envelope details (MOHURD, 2015)	Roof thermal transmittance ($W/m^2 \cdot K$)	0.28 (Harbin), 0.35 (Urumqi), 0.45 (Beijing & Lhasa), 0.5 (Shanghai & Chongqing), 0.8 (Kunming, Nanning & Hong Kong)
	Wall thermal transmittance ($W/m^2 \cdot K$)	0.38 (Harbin), 0.43 (Urumqi), 0.5 (Beijing & Lhasa), 0.8 (Shanghai & Chongqing), 1.5 (Kunming, Nanning & Hong Kong)
	Window thermal transmittance ($W/m^2 \cdot K$)	2.7 (Harbin), 2.9 (Urumqi), 3.0 (Beijing & Lhasa), 3.5 (Shanghai & Chongqing), 5.2 (Kunming, Nanning & Hong Kong)
	Window to wall ratio (WWR)	0.2
Indoor design conditions	Temperature (°C)	20 ± 3
	RH (%)	55 ± 10
	Concerned space volume (m^3) (length × width × height)	1035 (23.7×15.6×2.8)
Outdoor and supply airflow	Outdoor air changes per hour (ACH)	2
	Supply air changes per hour (ACH)	20
Installed fans specification	MAU fan pressure (Pa)	1100
	AHU fan pressure (Pa)	850
	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)	57 + 19
Internal loads schedule	Lighting Occupants Equipment	See Fig. 4, referring to the defaults given in BEAM Plus (BEAM, 2009)

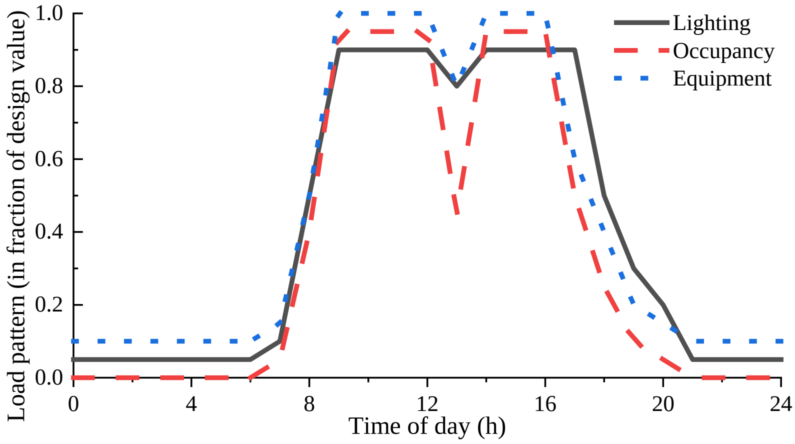

Fig. 4. Schedules for lighting, occupancy and equipment load in the simulated building

4.1.2 Weather conditions

Typical Meteorological Year of different locations (Marion & Urban, 1995), which contains typical hourly weather data obtained from the Meteonorm database (Remund & Kunz, 1997), are used for the whole-year building energy analysis. Fig. 5 presents the hourly TMY2 weather data on the psychrometric chart and the distribution of outdoor conditions in different regions for 1 year in Hong Kong. By selecting the indoor humidity at the lower-limit (23 °C, 45%) and upper-limit (23 °C, 65%) as reference indoor conditions (point O_1 and O_2), the psychrometric chart can be divided into four regions based on differences between the outdoor and indoor air states. The air-conditioning systems should operate using non-economizer modes (i.e., the economizer is not activated) in regions 1 and 2, while the systems are of high probability to operate using economizer modes in regions 3 and 4. Region 1 represents the humid weather conditions. The outdoor air humidity is higher than the upper-limit humidity of indoor air, so the MAU needs to dehumidify the outdoor air for removing the moisture. Region 2 represents the hot-dry weather conditions. It is possible to remove all the indoor moisture by the introduction of the proper outdoor airflow under low internal latent load conditions, while the cooling is still required to remove the sensible heat. Region 3 represents the cool-dry weather conditions. Through the introduction of cool outdoor air by the economizer cycle, the indoor environment is likely to be controlled at the allowable range even without additional cooling/dehumidification. Region 4 represents the cold-dry weather conditions. Due to low temperature and humidity of outdoor air, heating and humidification are possibly needed to maintain the required indoor temperature and relative humidity.

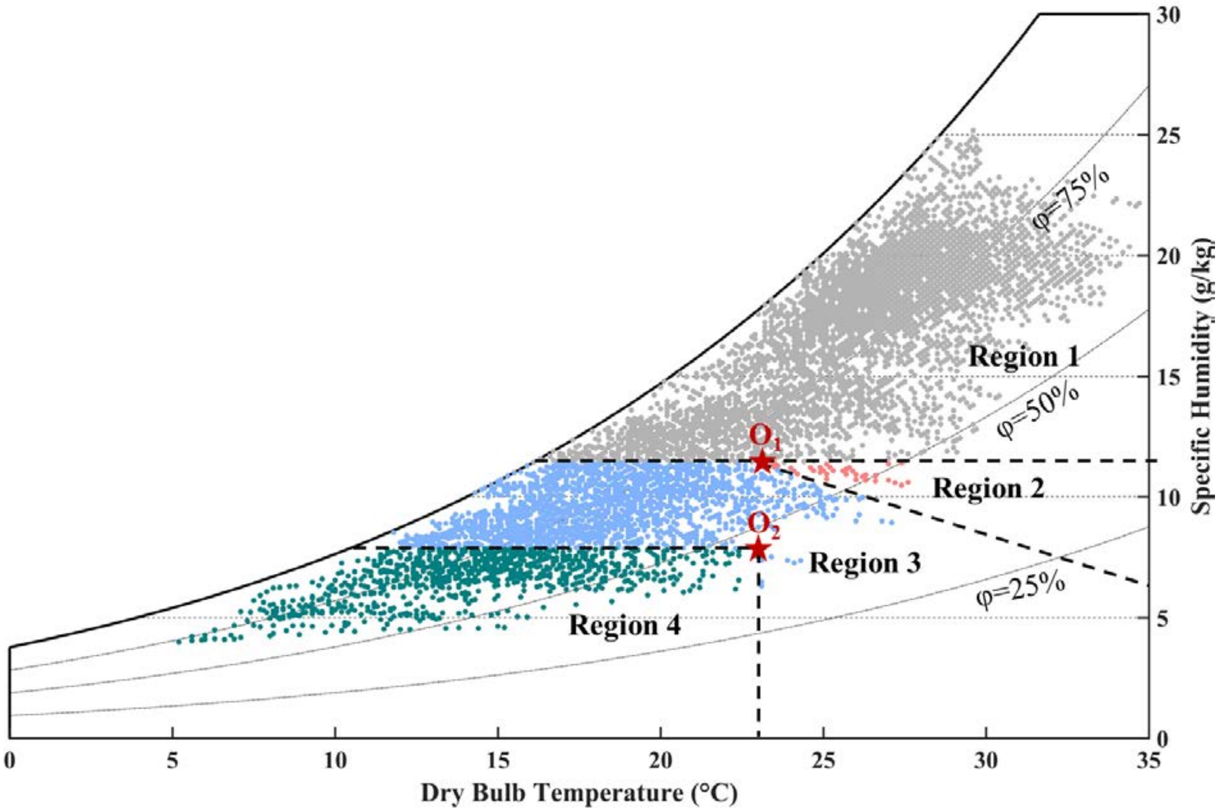

Fig. 5. Hourly outdoor conditions for 1 year in Hong Kong

Fig. 6 presents the fraction of outdoor conditions that fall into each psychrometric region for nine selected cities. Nanning and Hong Kong, located in hot summer and warm winter climate zone, are hot and humid with over 60% of outdoor conditions fall into region 1. Harbin, Urumqi, Beijing and Lhasa, located in the cold/severe cold zones, have more than 60% of time in the cold-dry region (i.e., region 4). Shanghai and Chongqing, located in hot summer and cold winter climate zone, have more than 30% of time in both region 1 and region 4, with distinct dry and wet climates. Kunming, located in temperate climate zone, has more than 30% of time in cool-dry weather conditions (region 3), significantly higher than that of the other cities. It can also be seen that, in moderate, cold and severe cold climate zones, the outdoor conditions have a significant fraction falling into regions 3 and 4. These are the regions in which the economizer modes should be well-selected and applied. For the cities in hot climatic conditions, the non-economizer modes operate for most time through a year.

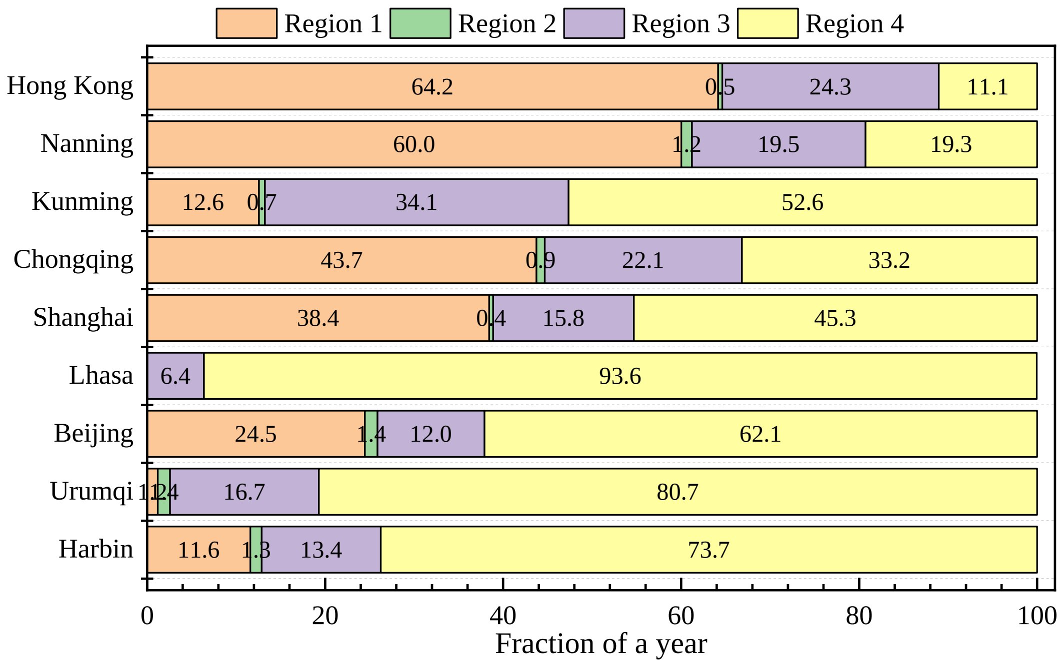

Fig. 6. Fraction of a year for different psychrometric chart regions in different cities.

4.1.3 Load conditions

Assuming that the indoor temperature and relative humidity of the cleanrooms are controlled at its upper limit (23 °C, 65%), the hourly space cooling loads of cleanrooms in nine cities are calculated in TRNSYS 18 (TRNSYS, 2017) as shown in Fig. 7. Where, the cooling loads result from heat transfer processes through the building envelope (external elements) and internal sources while the outdoor air load is not taken into account. This is due to the outdoor air load is determined by the ventilation strategy adopted and handled by the MAU before entering the spaces. The mean sensible cooling loads vary between 23.6 W/m^2 (Kunming) and 36.4 W/m^2 (Hong Kong), and the mean total cooling loads vary between 44.3 W/m^2 (Kunming) and 57.4 W/m^2 (Hong Kong). Although the selected cities locate in different climate zones, the space cooling load distributions have no significant difference when the outdoor air load is not taken into account.

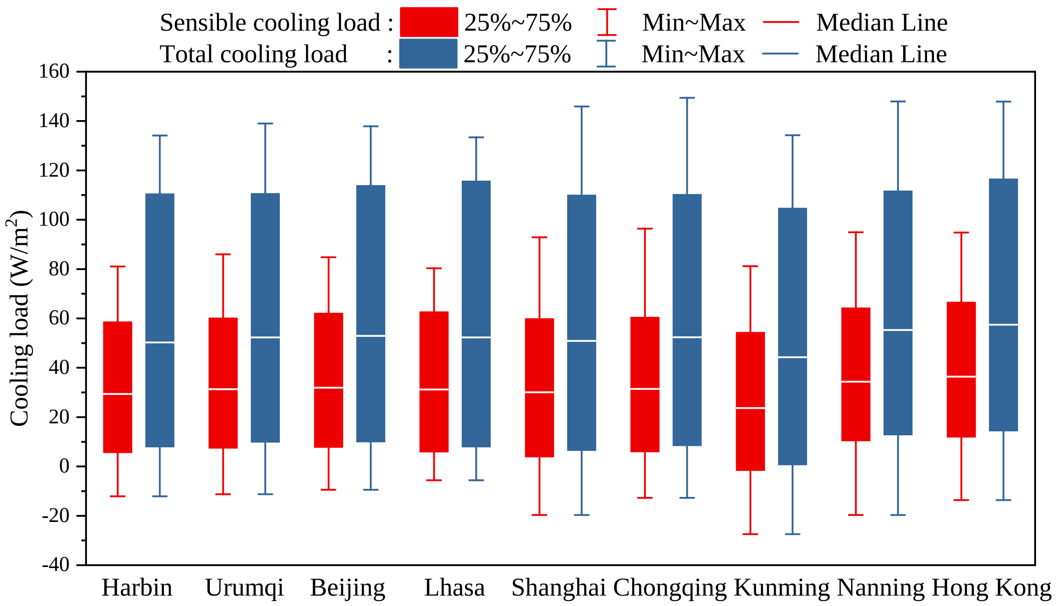

Fig. 7. Space cooling load distributions of cleanrooms in nine cities

4.2 Air-conditioning subsystem energy models

The total electrical load of the cleanroom air-conditioning system can be calculated using Eq.1, which includes the electrical loads of the MAU/AHU cooling coil, AHU heater, AHU humidifier, make-up air fan and supply air fan. Here, Q_{cc} is cooling load of the cooling coil (kW). Q_{he} is heating load of the heater (kW). Q_{hu} is humidification load of the humidifier (kW). COP_c, COP_{he} and COP_{hu} are the overall coefficient of performance of the cooling system, heating system and humidification system, respectively. $W_{f,MAU}$ and $W_{f,AHU}$ are fan powers of the MAU and AHU (kW), respectively. In this paper, for the central cooling system, the overall coefficient of performance of the central cooling system (COP_c) including the pumps and the chiller is assumed to be constant as 2.5. COPs of the heating system (COP_{he}) and humidification system (COP_{hu}) are assumed to be constant as 1.0 due to the heating/humidification energy is provided by the electric heaters/humidifiers.

$$E_{tot} = \frac{Q_{cc,MAU+}Q_{cc,AHU}}{COP_c} + \frac{Q_{he,AHU}}{COP_{he}} + \frac{Q_{hu,AHU}}{COP_{hu}} + W_{f,MAU} + W_{f,AHU} \quad (1)$$

Several simplified models are used to evaluate the electrical loads of these components shown as follows.

<u>*Fan model:*</u> The fan power of MAU/AHU fans is characterized by their volumetric flow rate, pressure rise and efficiency, as shown in Eq. 2. Here W_f is the total fan power (kW). V is the air volumetric flow rate (m^3/s). Δp is the total pressure rise (kPa). η_f is fan efficiency.

$$W_f = \frac{V\Delta p}{\eta_f} \quad (2)$$

<u>*System energy balance model:*</u> The cooling loads of the air-conditioning components are determined based on the energy and mass balances, to process the air to the required supply air state (i.e., State 9 in Fig. 1). In each time step, the supply air temperature and humidity are first obtained as Eqs. 3-4. The air flowrates and inlet/outlet air states (States 2-8) are then determined according to the air-handling processes of the adopted ventilation strategy as presented by Zhuang et al. (2019). The equivalent electrical loads of cooling coils (E_{cc}), heaters (E_{he}) and humidifier (E_{hu}) are calculated according to their inlet and outlet air states and the cooling/heating/humidification system overall COPs using Eqs. 5-8. Here t and w are the dry-bulb air temperature (*°C*) and absolute humidity (*g/kg*), respectively. Q_{sen} and Q_{tot} are the space sensible and total cooling load (excluding the outdoor air load, kW), respectively. m_{fh} and m_s are the outdoor air mass flowrate (*kg/s*) and AHU supply air mass flowrate (*kg/s*), respectively. h is the air enthalpy (*kJ/kg*). c_p is the air specific heat ratio ($kJ/(m^3 \cdot °C)$). h_{fg} (2501 *kJ/kg*) is the latent heat of vaporization.

$$t_9 = t_9 - \frac{Q_{sen}}{m_s C_p} \quad (3)$$

$$w_9 = w_4 - \frac{Q_{tot} - Q_{sen}}{m_s h_{fg}} \quad (4)$$

$$E_{cc,MAU} = \frac{Q_{cc,MAU}}{COP_c} = \frac{m_{fh}(h_3 - h_2)}{COP_c} \tag{5}$$

$$E_{cc,AHU} = \frac{Q_{cc,AHU}}{COP_c} = \frac{m_s(h_6 - h_5)}{COP_c} \tag{6}$$

$$E_{he,AHU} = \frac{Q_{he,AHU}}{COP_{he}} = \frac{m_s(h_7 - h_6)}{COP_{he}} \tag{7}$$

$$E_{hu,AHU} = \frac{Q_{hu,AHU}}{COP_{hu}} = \frac{m_s h_{fg}(w_8 - w_7)}{COP_{hu}} \tag{8}$$

5 Test results and analysis

5.1 Energy performance and recommended operation modes in different climate zones

The energy performance of the ADV strategy is assessed and compared with the existing IC strategy in the selected nine cities. Fig. 8 shows the annual energy consumption of different components adopting the IC strategy and ADV strategy. It can be seen that the annual energy consumption of the air-conditioning systems adopting the ADV strategy can be 6.8-40.8% less than that of the IC strategy. Compared with adopting the IC strategy, although more MAU cooling is required, a large amount of AHU cooling and heating energy consumption is reduced. This indicates that the overcooling and reheating processes are significantly reduced by adopting the proposed ADV strategy. In addition, for the cities in hot/temperate regions (i.e., Shanghai, Chongqing, Nanning, Hong Kong and Kunming), the energy savings (over 34%) are more significant than that in cold regions (i.e., Harbin, Urumqi, Beijing and Lhasa). It is worth noticing that applying the ADV strategy in Urumqi can only achieve 6.8% energy saving, which is least among all the cities. The reason is that Urumqi is cool/cold and dry throughout a year, and the dry outdoor air can be directly induced for removing the indoor latent heat to avoid overcooling and reheating processes, resulting in little difference between using the ADV strategy and IC strategy in most operation time.

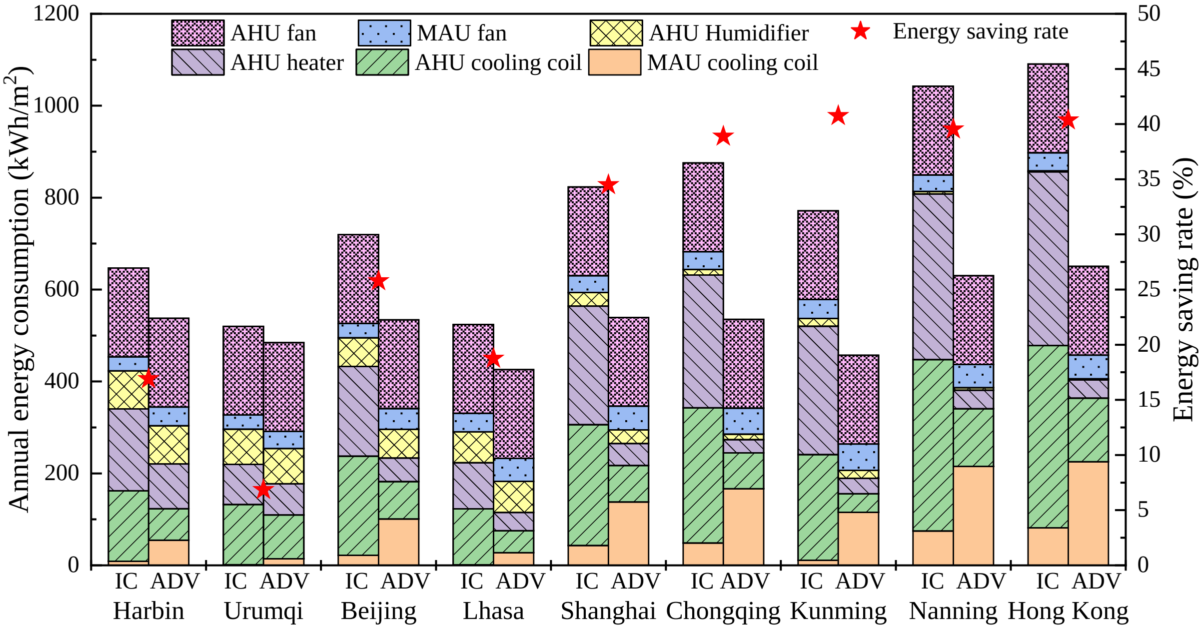

Fig. 8 Annual energy consumption of adopting existing IC strategy and proposed ADV strategy in 9 cities

The running frequencies (%) of different control modes of ADV strategy are presented as in Fig. 9, to illustrate how the significant energy savings are achieved. The non-economizer modes account for more than 50% running time through a year in all selected cities. Here the PD/DV mode represents that there is no difference between the PD mode and DV mode, which indicates all moisture heat can be removed at the minimum required outdoor airflow (i.e., 2 ACH in this case). The PD/DV mode accounts for the largest proportion among all operation modes (more than 42%). It is worth noticing that overcooling and reheating processes (i.e., represented by PD mode in this figure) to remove the internal latent load are seldom used in cold and temperate regions. However, for the cities in hot climate regions, such as Hong Kong, the overcooling and reheating processes are still adopted due to the high internal latent load and outdoor air enthalpy. The FL mode has the lowest running frequency in all cities. The running frequency of economizer mode is highest in Lhasa (45.6%), due to the high cooling/dehumidification potentials under plateau climatic conditions.

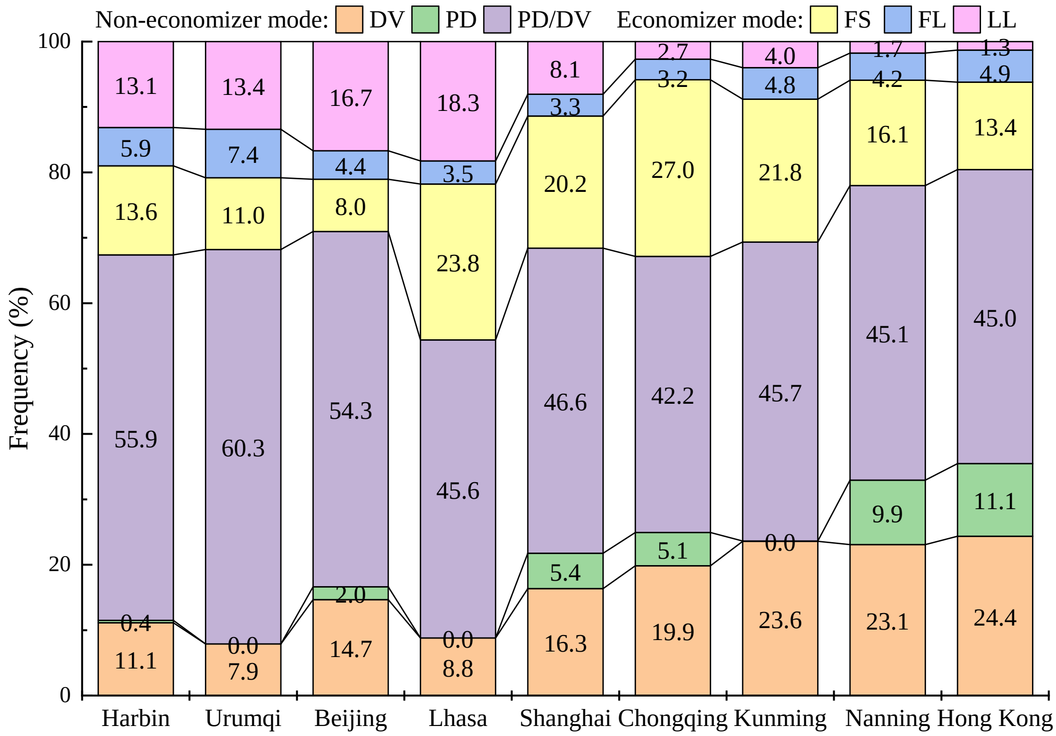

Fig. 9 The running frequencies (%) of different control modes of ADV strategy

To simplify the operations adopting the ADV strategy in real applications, it is recommended to only use DV mode in severe cold/cold and moderate regions due to the low running frequency of using PD mode (less than 2%). In addition, when the economizer is activated, the FL and FS modes are the main economizer modes in severe cold and cold regions (Harbin, Urumqi, Beijing and Lhasa), while the FS mode is the main economizer mode in hot or temperate regions (Shanghai, Chongqing, Nanning, Hong Kong and Kunming).

Both the operation frequencies of economizer mode and energy saving ratios when adopting the ADV strategy have a high correlation with outdoor air humidity of locations as shown in Fig. 10. Fig. 10A presents the relations between the annual mean absolute humidity of outdoor air and frequencies of economizer mode in different locations. It can be seen that economizer modes are more frequently used for the cities in dry climatic conditions. Fig. 10B presents that the overall energy saving ratios are higher in humid climates than that in dry climates when adopting the ADV strategy. This indicates climate conditions, especially the outdoor air humidity, significantly influence the selection of best operation modes, and thus the energy performance of the systems.

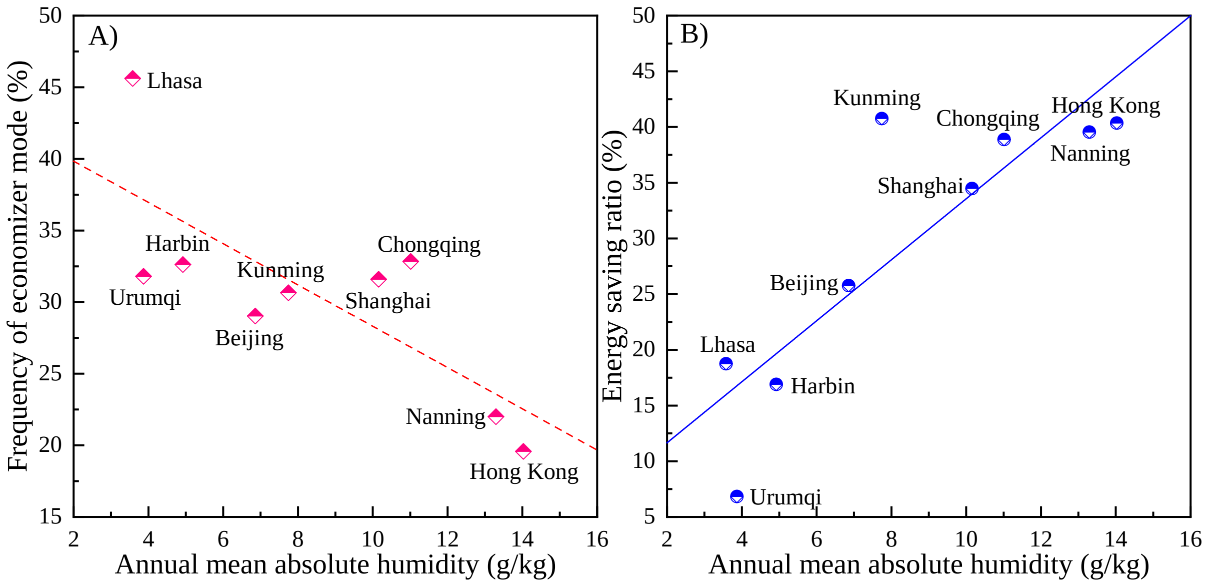

Fig. 10 Annual mean absolute humidity of outdoor air vs. A) frequencies of economizer mode B) energy saving ratios in different locations

5.2 Impacts on air-conditioning component system design and economic analysis

Due to the strict requirements on temperature and humidity controls in cleanrooms, it is assumed that the component capacities should meet the maximum cooling/heating demands of a year. Therefore, the required capacities of the cooling coils, heater and humidifier are determined by the hourly maximum cooling, heating and humidification demands, respectively. The required powers of the MAU and AHU fans are determined by the maximum outdoor and supply airflow, respectively. The required capacities of air-conditioning component for implementing the IC and ADV strategies are shown in Fig. 11. In general, the implementation of the ADV strategy requires larger capacities of MAU cooling coil, which can be 3.1 (Chongqing) to 9.4 times (Urumqi) as larger as that for IC strategy. In contrast, the required capacities of other components (i.e., fans, AHU cooling coil, humidifier and heater) are not larger or equal to that for IC strategy. This result also offers a good reference for the existing system (i.e., initially adopting IC strategy) retrofit and new system design. For the full implementation of ADV strategy in an existing system, only the size of MAU cooling coil is required to be enlarged while the other components can be unchanged. For the full implementation of the ADV strategy in a new system, compared with the design for the IC strategy, the required capacity of MAU cooling coil is larger while the required capacities of AHU cooling coil and heater are smaller. It is worth noticing that, when adopting the IC strategy, the required fan power (9.9 kW) in Urumqi is lower than the others. When the enthalpy economizer is activated under dry weather conditions adopting IC strategy, the amount of outdoor air is induced to control indoor relative humidity at its higher limit (i.e., 65%). Due to the dry climatic conditions in Urumqi (Fig. 10), it is found that the maximum 94% (proportion to total supply airflow) outdoor airflow is required to be induced in the test year. For other cities, which is not as dry as Urumqi, the maximum 100% outdoor air is required to be induced according to the mechanism of the economizer adopting IC strategy. For a new design of the case in Urumqi, the required MAU fan power (10.5 kW) adopting ADV strategy is larger than that of adopting IC strategy (9.9 kW). However, in the retrofitting case (originally designed for adopting IC strategy), the modification on MAU fan is not required due to that the fan capacity can almost meet the outdoor air ventilation requirements when the economizer is activated.

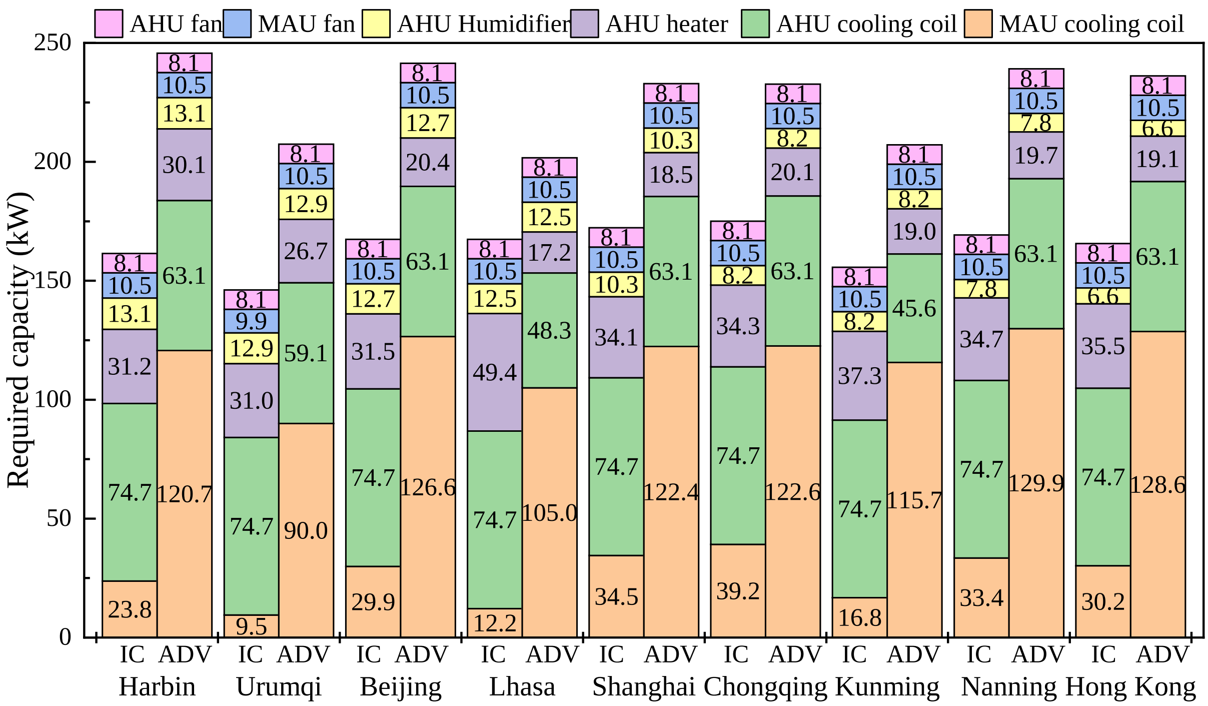

Fig. 11 Air-conditioning component required capacities for implementing the IC and ADV strategies

The payback period (PBP) indicates the number of years needed for the payback of the surplus capital cost for using the ADV strategy to replace IC strategy. This value is the ratio of capital cost difference and operation cost difference as shown in Eq. 9, where ΔC_C and ΔC_O are the increase of system capital cost and the annual operation (electricity) cost saving, respectively. For new system designs, the total capital cost is the sum of component investment costs. For existing system retrofits, the total capital cost includes the component investment costs and refurbishment costs. The componment investment cost includes the cost of AHU/MAU cooling coils (IC_{CC}), AHU/MAU axial fans (IC_{fan}), AHU electric heater (IC_{he}) and AHU electric humidifier (IC_{hu}). The cost estimates, which are the functions of the corresponding component capacities, are based on RSMeans Mechanical Cost Data (Mossman, 2008) as shown in Appendix A. Here £$_e$ is the local electricity price (RMB/kWh), as shown in Appendix B. It is worth noting that, the component initial cost considering the inflation is used to fit the cost equations. In this study, the component initial cost (in 2019) is used for cost analysis, which is obtained by adding the inflation on the cost data in 2008. It is also recommended to use actual manufacturers' quotations, if reliable, or newly updated cost data for cost analysis. For the retrofitting case, the refurbishment cost of a component (including additional installation cost) is assumed as 30% of its incremental initial cost (ΔIC)(delete) considering the possible modification and maintenance of accessories and packages (Hang et al., 2014).

$$PBP = \begin{cases} \frac{\Delta C_C}{\Delta C_O} = \frac{(IC_{cc}+IC_{he}+IC_{hu}+IC_{fan})_{ADV}-(IC_{cc}+IC_{he}+IC_{hu}+IC_{fan})_{IC}}{£_e \times (E_{tot,IC}-E_{tot,ADV})}, & \text{New system designs} \\ \frac{\Delta C_C}{\Delta C_O} = \frac{1.3[(IC_{cc}+IC_{he}+IC_{hu}+IC_{fan})_{ADV}-(IC_{cc}+IC_{he}+IC_{hu}+IC_{fan})_{IC}]}{£_e \times (E_{tot,IC}-E_{tot,ADV})}, & \text{Existing system retrofits} \end{cases} \quad (9)$$

Fig. 12 shows the payback periods for existing system retrofit and new system design adopting the ADV strategy. Due to the high energy saving potentials adopting ADV strategy, when retrofitting an existing system, the system payback periods in all cities except for Urumqi are all less than 4 years. Due to the required smaller sizes of the AHU, for new system design, the system payback periods in all cities except for Urumqi are all less than 2 years. In addition, the cities in hot climate zones (Shanghai, Chongqing, Kunming, Nanning and Hong Kong) have shorter payback periods than that of the cities in cold/severe cold climate zones (Harbin, Beijing and Lhasa). Urumqi requires the longest payback periods compared with other cities, which are 13.6 years for the existing system retrofit and 8.3 years for the new system design. The reason is that Urumqi is dry throughout a year, and only slight energy saving (6.8%) can be achieved by adopting the ADV strategy.

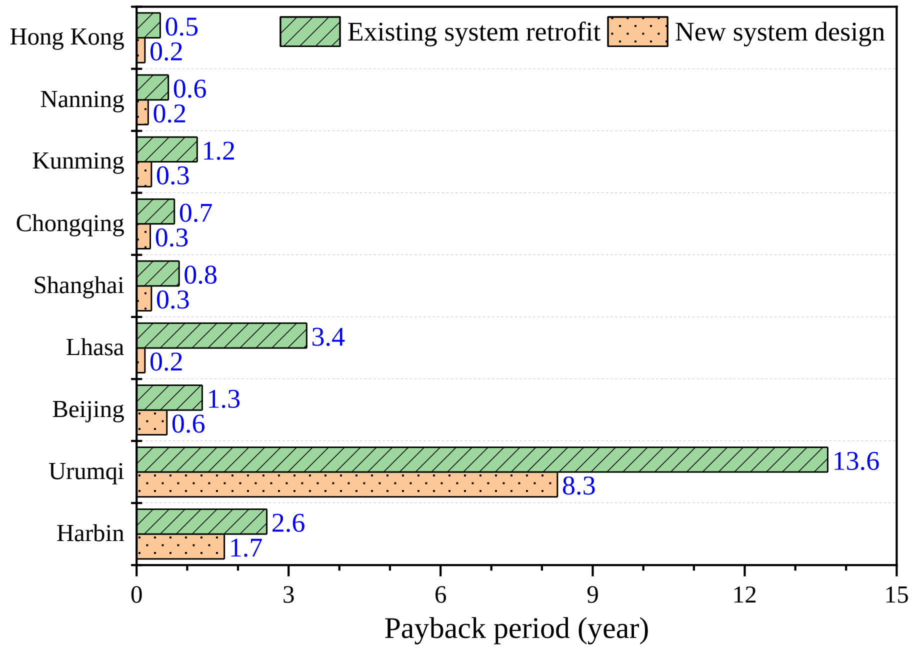

Fig. 12 Payback periods for existing system retrofits and new system designs

The energy and economic performance of the proposed strategy is compared with desiccant cooling strategies, which are the most-updated strategies for indoor temperature and humidity controls. Table 3 lists the energy performance and payback periods of desiccant cooling systems compared with conventional vapor compression air-conditioning systems in the existing studies. Compared with the conventional vapor compression air-conditioning systems, although the energy performances of the desiccant cooling systems are superior, both the initial cost and system complexity of them are increased significantly. Generally, the payback periods of desiccant cooling systems are 2-15 years, with the energy savings of 6-78%, depending on the selection of hybrid systems and climatic conditions.

It is worth noticing that its acceptance for cleanroom applications might still be an issue to be confirmed. It should be also noted that, in these cases, the payback periods presented are that for new system designs. In cases of retrofitting the conventional vapor compression air-conditioning systems, the payback periods of desiccant cooling strategies will be much higher, due to the requirements of replacing the existing equipment. The proposed ADV strategy offers a much better solution for conventional system retrofits. A significant energy saving can be achieved by adopting the proposed ADV strategy while only minor hardware (i.e. sizing) modification is required.

Table 3 Overview of desiccant cooling systems for temperature and humidity controls.

Author (Year)	City/ Country	System	Energy performance	Payback period (years)
Zhang and Niu (2003)	Hong Kong	Chilled ceiling combined with desiccant cooling	40% primary energy saving	15
Mazzei et al (2005)	Rome	Desiccant wheel hybrid system	23-38% operating cost savings	2-3
Gasparella et al. (2005)	Bolzano	Ground source heat pumps with chemical dehumidification	30% primary energy saving	7.8
Hirunlabh et al. (2007)	Thailand	Solid air-conditioning system	24% electricity saving	4
Khalid et al. (2009)	Karachi	Solar assisted pre-cooled hybrid desiccant cooling system	6% thermal energy saving	14
Ge et al. (2010)	Berlin and Shanghai	Solar driven two-stage rotary desiccant cooling system	78% (Berlin) and 69% (Shanghai) electricity saving	4.7 (Berlin) / 7.2 (Shanghai)
Li et al. (2010)	Hong Kong	Solar-assisted liquid desiccant cooling system	Maximum 59.6% electricity saving	7
Qi et al. (2015)	Singapore	Solar-assisted liquid desiccant cooling system	40% electricity saving	6

6 Conclusions

This study evaluates the energy and economic performance, investigates the required air-conditioning system design and identifies the most suitable operation modes of the proposed “adaptive full-range decoupled ventilation strategy” (ADV strategy) for buildings with spaces requiring strict temperature and humidity controls in different climatic conditions. Based on the results of the tests and investigation, some detailed conclusions can be drawn as follows:

- The ADV strategy offers significant and promising energy savings in different climate zones.

The annual energy consumption of the air-conditioning systems can be reduced by 6.8-40.8% when adopting the proposed ADV strategy, compared with the most commonly used existing interactive control (IC) strategy. It has higher energy saving in humid climates compared with that in dry climates.

- When the economizer is not activated, dedicated outdoor air ventilation (DV) mode is highly recommended as the main operation mode of the ADV strategy in severe cold/cold and moderate regions. When the economizer is activated, the following sensible load (FS) and lower-limit humidity control (LL) modes are the recommended operation modes in cold/severe cold climate zones while the following sensible load (FS) mode is the recommended operation mode in hot/temperate climate zones.

- For the full implementation of ADV strategy in the existing systems (i.e. initially adopting the IC strategy), only the size of MAU cooling coil needs to be enlarged while the other components can keep unchanged. For new system designs, the required capacities of AHU cooling coil and heater are even smaller compared with that designed for the IC strategy.

- For existing system retrofits and new system designs, the payback periods are less than 4 years and 2 years respectively in most climates when the ADV strategy is fully implemented.

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Appendix A The investment of components

The capital costs of air-side components are the functions of the corresponding component capacities in this study. The capital cost (RMB) of the axial fan, cooling coil, electric heater and electric humidifier can be estimated by Eqs. A.1-4 based on RSMeans Mechanical Cost Data (Mossman, 2008). The RSMeans Mechanical Cost Data was built based on continuously monitoring the available costs from the manufacturers in the construction industry. Considering the inflation, the component initial cost (IC_l) used in this study is calculated as Eq.A.5. Here the *NC* is the initial cost provided by RSMeans Data, including the installation cost and capital cost. *i* is the annual inflation rate set as 4% (Daud & Ismail, 2006/2012), and *n* is the number of time periods (years) past, selected as 11 years in this study.

$$IC_{fan} = 2019.4flow + 9563.8 \qquad flow \in [0.5, 6] \tag{A.1}$$

$$IC_{cc} = -9.05CAP_{cc}^2 + 1122.6CAP_{cc} + 11878 \qquad CAP_{cc} \in [2, 55] \tag{A.2}$$

$$IC_{he} = 1502.9CAP_{he} + 1435.1 \qquad CAP_{he} \in [0.5,20] \tag{A.3}$$

$$IC_{hu} = 508.52CAP_{hu} + 22801 \qquad CAP_{hu} \in [5,90] \tag{A.4}$$

$$IC = NC(1 + i)^n \tag{A.5}$$

Here CAP_{cc} (kW), CAP_{he} (kW), CAP_{hu} (kW) and *flow* (m^3/s) are the capacities of the cooling coil, electric heater, electric humidifier and design air flowrate, respectively. It is worth noting that when the design capacities of the cooling coil, electric heater and electric humidifier are larger than the upper limit of the ranges (e.g., 55 kW for cooling coil), several same components would be selected. For example, if the design cooling coil capacity is 100 kW, the initial cost of the cooling coil is the sum of the initial costs of two cooling coils, each with the capacity of 50 kW.

Appendix B Electricity prices in each selected city

Table B.1 lists the electricity prices (RMB/kWh) for non-residential buildings in the selected cities, where electricity prices of all cities except for Hong Kong are obtained from National Development and Reform Commission (NDRC, 2018) and the electricity price of Hong Kong is obtained from China Light & Power Company (CLP Hong Kong, 2018).

Table B.1 Electricity prices (RMB/kWh) for non-residential buildings in each simulated city.

City	Electricity price (RMB/kWh)
Harbin	0.7499
Urumqi	0.4814
Beijing	0.8203
Lhasa	0.72
Shanghai	0.79
Chongqing	0.7132
Kunming	0.6075
Nanning	0.7365
Hong Kong	0.9588

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