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Energy Efficient Design and Control of Cleanroom Environment Control Systems in Subtropical Regions – A Comparative Analysis and On-site Validation

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Abstract: Compared with spaces air-conditioned for thermal comfort, cleanrooms often have special requirements on dry bulb temperature, relative humidity and particle concentrations. It is a challenging task to achieve those requirements with minimum energy consumption, especially when different parameters interfere with each other. A significant amount of energy would be wasted if the system is not properly designed and controlled. This paper firstly provides an overview and a discussion on the essentials for design and control of cleanroom air-conditioning systems. The existing systems and controls are categorized into three typical options and their performances are then analyzed based on different weather and load conditions. For new design, the "fully decoupled option" is the preferred option for humid sub-tropical regions. The analysis results are applied in a retrofit project for a pharmaceutical factory located in Hong Kong, a humid sub-tropical city, which employed the "interactive option". This system is proposed to operate as a "partially decoupled option" in this project since such retrofit requires no modification on the existing hardware. The retrofitted system option has been on-site tested in mild weather condition, which provided 69.6% and 87.8% reductions of cooling and heating consumptions respectively. More comprehensive comparison tests are also conducted on a dynamic platform built on Matlab/Simulink.

Key words: cleanroom; air-conditioning; energy saving; dehumidification; humidity control.

1 Introduction

Cleanrooms can be 30-50 times more energy intensive than the average US commercial buildings due to high ventilation rates required for maintaining low particle concentrations [1]. Cleanroom environment control systems or air-conditioning systems consume about 30-65% of the total energy use in a high-tech fabrication plant [2]. High energy consumptions often represent high operation costs: “A Class 10 environment typically costs about US$2000 per square foot to build and US$1 million a year to operate.” [3][4]. It is essential to reduce energy use in cleanrooms for two main reasons. First, the cleanroom area has been growing fast, which increased from 4.2 million m^2 in 1993 to the estimated 15.5 million m^2 in 2015 in the US [5], and it increases even faster in South China. Second, the energy consumptions and their energy saving potentials are very high compared with many other air-conditioning systems.

Design and control of cleanroom environment control or air-conditioning systems are both essential for energy efficiency, which are closely interrelated concerning both the environmental control performance and the energy performance. Only when an air-conditioning system is properly designed, appropriate control can be implemented to achieve the desired environment control with high energy efficiency. In addition, control engineers should be involved in the design process so that all elements are considered [6].

Many researchers have addressed the design of cleanroom air-conditioning systems. Hansz [7] provided five steps to collect required information for designing cleanrooms, i.e. establishing goals, analyzing facts, examining concepts, establishing needs, and stating problems. Yang et al [4] analyzed the essential elements of cleanrooms design that significantly affect the construction costs. Tschudi et al [8] provided strategies for designing and controlling of air change rates in cleanrooms, i.e. demand-controlled filtration (DCF) based on real-time monitoring of particle concentrations. Lin et al [9] developed a fan dry coil unit return system for improving the energy efficiency of cleanrooms. Hu and Tsao [10] compared the energy efficiency of five different cleanroom air-conditioning systems made up of different combinations of recirculation air unit (RCU), make-up air unit (MAU), fan coil unit (FCU), dry cooling coil (DCC), fan-filter unit (FFU), etc. The results indicated that the system with combined MAU and FFU provided the highest energy efficiency. They also proposed a make-up air system for energy conservation [11]. Kircher et al [1] compared energy efficiency methods through modeling and simulation on four systems, including a heat recovery system, solar preheating for dehumidification system, lighting control, and demand-controlled filtration. Some other studies addressed energy recovery from exhaust air using technologies like heat pipes or regenerative-desiccant wheel [12][13][14][15][16].

Though many studies for cleanroom design appear in literature, few studies have investigated the control of cleanroom air-conditioning systems. A well designed but not properly controlled cleanroom air-conditioning system may still consume a large amount of energy. Some studies discussed the control of cleanroom pressure. Wang et al [17] provided an operation strategy to control pressure gradient in a multi-zone cleanroom. Their experimental investigation showed that the strategy achieved an energy saving of about 24.5%. Brink et al [18] also proposed an improved pressure control in cleanrooms with a focus on pressure deviations during the entry of cleanrooms.

However, very few published research works have addressed the control associated with design for energy efficiency in cleanrooms. Because of the special requirements, those approaches for thermal comfort air-conditioning systems, such as reducing outdoor air flow[19] and some complex control methods[20][21], may not be applicable for cleanroom applications. Only a few publications addressed local controls that aim at controlling process variables to follow their set-points in cleanrooms. For instance, Tan et al [22] provided an automatic tuning approach for variable structure control of temperature in cleanrooms.

Since a majority of the cleanroom air-conditioning systems use cooling process for dehumidification, counteraction between heating and cooling for humidity and temperature control may waste a large amount of energy. Alternative approaches should be considered to enhance the energy efficiency of cleanroom air-conditioning systems. First, the system design should consider the climate conditions. Secondly, the control of the system in the year-round operation conditions should be properly considered at the design stage and the control design should consider the system energy use seriously.

This paper therefore provides an overview and discussion of the key issues of design and control of cleanroom air-conditioning systems. Three typical systems are then described and comparatively analyzed in different weather and load conditions. The analysis results are applied in a retrofit project for a pharmaceutical factory located in Hong Kong, a humid sub-tropical city, which originally employed an air-conditioning system of “interactive option”. This system is retrofitted to operate as a system of “partially decoupled option” in this project since such retrofit is the most cost effective as it requires no modification of the existing hardware.

Though different cleanrooms may have different requirements, they have the same problem of high operation cost due to similar reasons. The results of comparative analysis of the systems of three categories as well as the case study presented in this paper provide very useful guidance for the design and control of environment control systems or air-conditioning systems in both retrofit and new projects.

The rest of this paper is organized as follows: Section 2 presents an overview and discussion on the key issues of design and control of cleanroom air-conditioning systems. Section 3 describes the configurations and control methods of three typical systems for cleanrooms. Section 4 shows the comparative analysis on these three systems. Section 5 introduces a retrofit project, including a description of the actual building air-conditioning, its retrofit on-site test plan, and the dynamic simulation platform for comprehensive tests and validation; Section 6 presents the on-site and simulation test results and analysis. On-site implementation results are also presented. Section 7 draws the conclusions.

2 Key issues of design and control of cleanroom air-conditioning systems

Compared with common purpose of air-conditioned spaces for thermal comfort, cleanrooms often have special requirements on dry bulb temperature (DBT), relative humidity (RH) and particle concentrations. A high supply air flow rate is often required in cleanrooms for removing airborne particle pollutants. Besides the large fan power consumption, the high supply air flow rate may also cause great power consumption for dehumidification and temperature control of cleanrooms. The conventional design and control methods directly cool down the supply air to its dew point for dehumidification and then reheat it to achieve the desired temperature in cleanrooms. Because of the high supply air flow rate, the required cooling and reheating energy for dehumidification would be extremely high even when the humidity load is low.

To solve the problem of high energy consumption caused by high supply air flow rate, some key issues should be addressed properly at design stage. First, it is essential to design the supply air ducts with low air flow resistance, so that the supply air fan power can be reduced. Second, conventional air handling processes have to be replaced by alternative approaches for controlling dry bulb temperature, relative humidity and particle concentration. Two common approaches can be summarized to address the problem of high energy consumption and the relative concepts are described as follows.

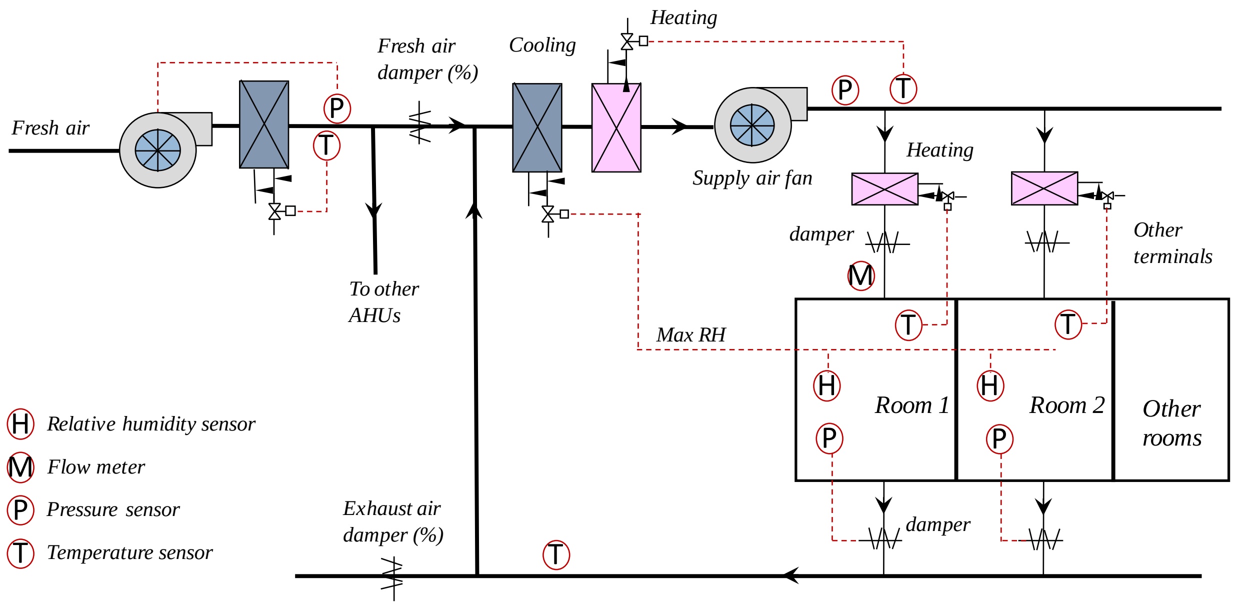
Approach 1: Dehumidify the outdoor air in primary air handling units (PAUs) to decouple humidity and temperature controls while the dried outdoor air is used to dehumidify air in the indoor spaces. This system is particularly suitable for cleanrooms of relatively low dehumidification load.

Approach 2: Decouple temperature and humidity controls using two parallel cooling coils besides heating. Supply air flow is separated into two streams. One stream goes through the wet coil for humidity control. Its flow rate is optimized so that the wet coil consumes minimum cooling energy for dehumidification. The other stream goes through the dry cooling coil for temperature control only. Moreover, the total flow rate of the two streams equals the required total supply air flow rate for the control of particle concentrations.

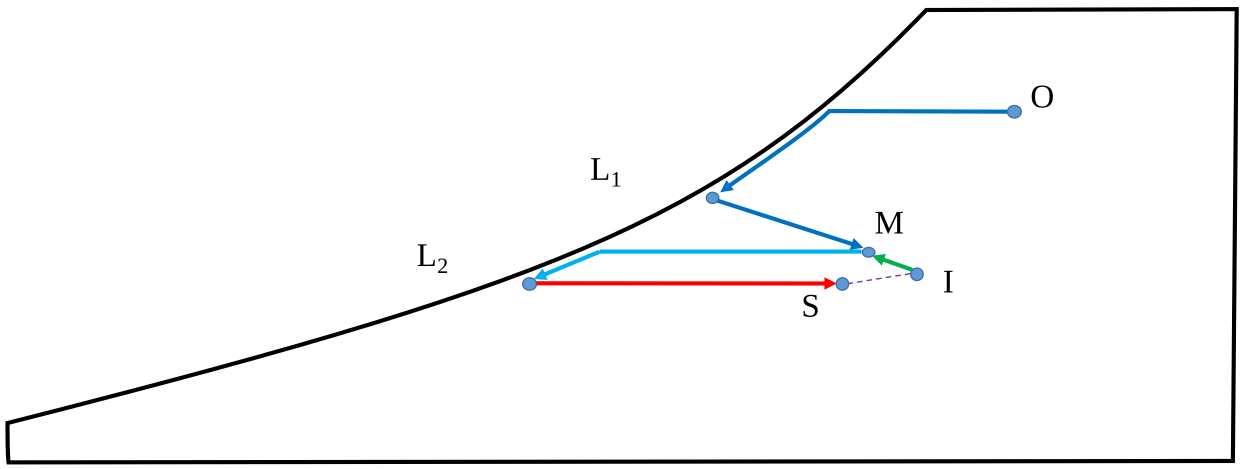
3 Three typical air-conditioning system options

Based on a survey of the existing systems and controls appearing in literature and real practice, cleanroom environment control systems or air-conditioning systems are categorized into three typical options according to the degree of decoupling, the control of dry bulb temperature, and relative humidity as well as particle concentrations.

3.1 Option A – Interactive Option



(a) System configuration of Option A



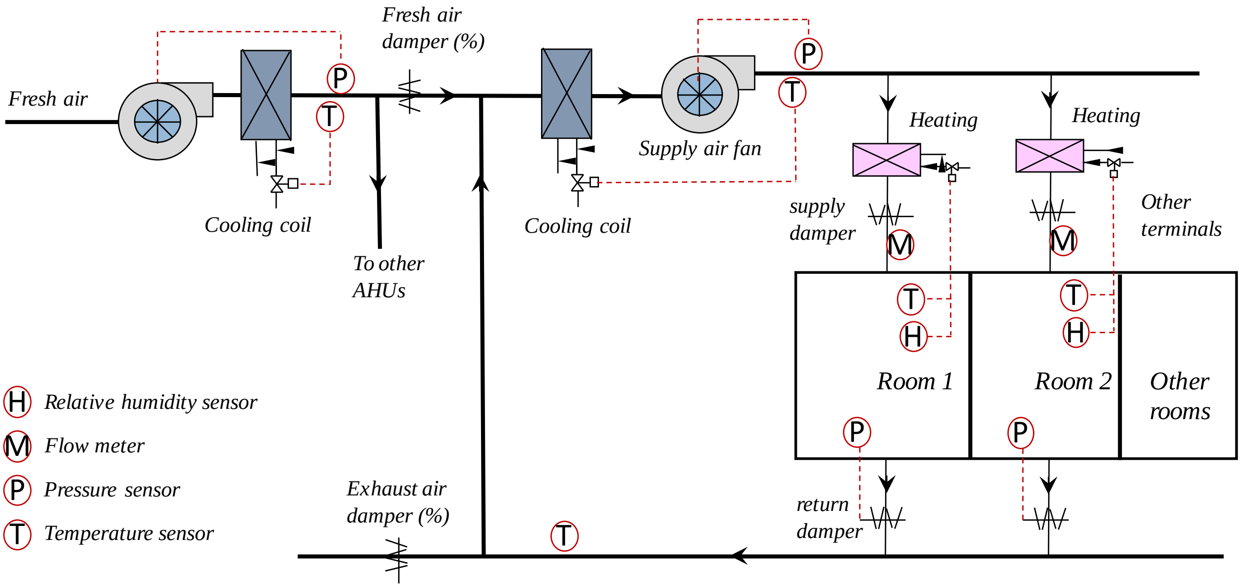
(b) Air handling process control in PAU and AHU of Option A

Fig. 1 System configuration and air handling process control of Option A - Interactive Option

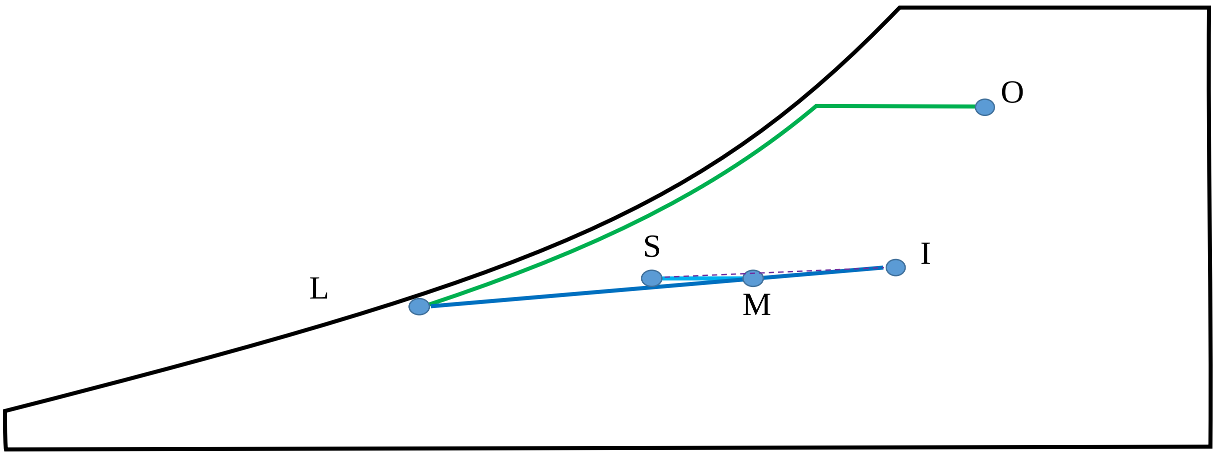
Fig. 1(a) shows the configuration of Interactive Option, and Fig. 1(b) demonstrates the air handling process control on the psychrometric chart. The system consists of a PAU and several AHUs. Each AHU serves several cleanrooms. The PAU cools the outdoor air down to a certain temperature, e.g. 15°C, at its mechanic dew point (O→L_1). The cooled outdoor air is then mixed with the recirculation air in an AHU with a certain outdoor/recirculation air flow ratio, e.g. 15/85, (L_1→M, I→M). The mixed air is further handled by the AHU cooling coil for dehumidification and possibly cooling as well (M→L_2) and heated by AHU heating coil if needed to reach the supply air temperature set-point while providing necessary dehumidification (L_2→S). The supply air is further heated if needed by the cleanroom reheating terminals to maintain the individual space temperatures at their set-points.

Fig. 1(a) also demonstrates the control mechanism of AHUs and cleanrooms. PAU fan speed and opening of the cooling coil valve are modulated by PID controllers to maintain the outlet static pressure and temperature at their corresponding set-points. The maximum relative humidity among all associated cleanrooms is controlled by a PID controller below a preset upper limit (threshold) by modulating the AHU cooling coil valve. The AHU heating coil valve is modulated to control the AHU supply air temperature at its set-point when the supply air is overcooled for the purpose of dehumidification or heating is needed in the cleanrooms in cold seasons. The valves of the terminal reheating coils in individual cleanrooms are modulated by their PID controllers to control cleanroom dry bulb temperatures at their set-points. AHU fan speed is modulated to maintain the air static pressure set-point in the supply duct at its set-point, in order to provide sufficient air flow to all associated cleanrooms. The supply air dampers of cleanrooms are used to achieve proper air flow balance among cleanrooms. The required supply air flow rates in individual cleanrooms are maintained by modulating the corresponding supply air dampers based on the measured supply air flow rates either at commissioning stage or in the online control process. The PID controllers control the static pressures of individual cleanrooms at their set-points by modulating corresponding return air dampers.

3.2 Option B – Partially Decoupled Option



(a) System configuration of Option B



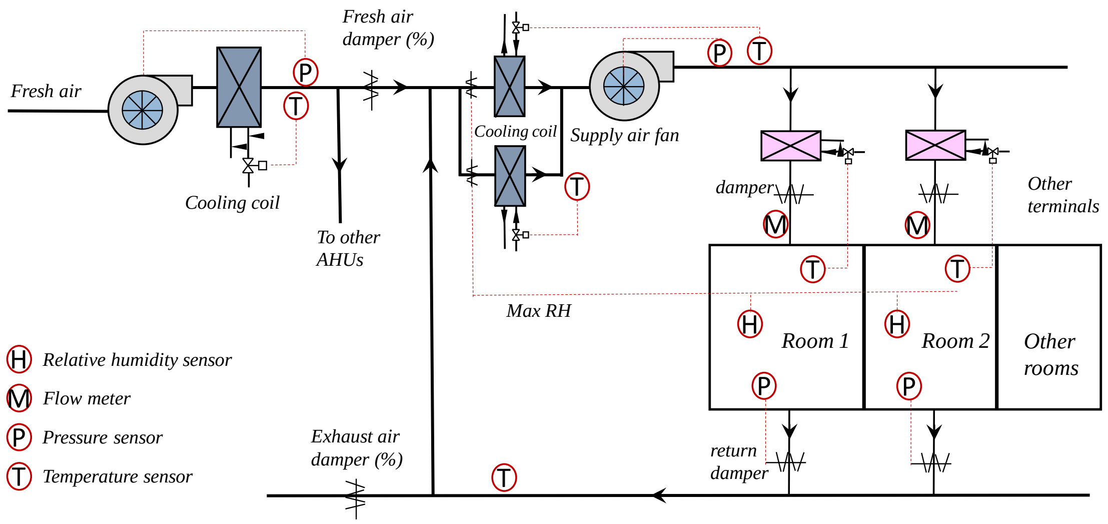
(b) Air handling process control in PAU and AHU of Option B

Fig. 2 System configuration and air handling process control of Option B - Partially Decoupled Option

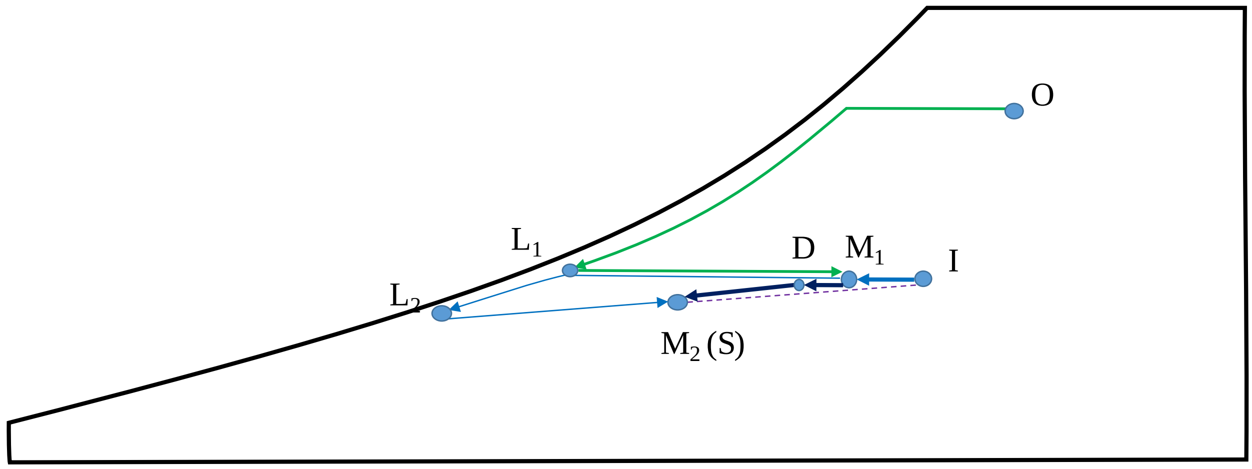
The system configuration of Option B is quite similar to Option A as shown in Fig. 2(a). The only difference is that Option B does not require a heating coil in AHUs. However, the air handling process control of Option B is quite different from that in Option A, as demonstrated on the psychrometric chart in Fig. 2(b). Only the dry outdoor air is used for the dehumidification of humid air in cleanrooms. In case the humidity ratio of the outdoor air is higher than that of the indoor air, the outdoor air is cooled and dried in PAU, i.e. the O→L curve in Fig. 2(b). The PAU outlet air temperature set-point needs to be set rather low, so that the air in the cleanrooms can be dehumidified effectively by the outdoor air from the PAU only. In case the outdoor air humidity ratio is low enough, the air in cleanrooms can be dehumidified by directly supplying the dry outdoor air to the AHUs without any cooling in the PAU. The outdoor air flow rate is determined according to the need of indoor pollutant control or relative humidity control in cleanrooms, depending on which one is critical.

The control of Option B is also demonstrated in Fig. 2(a). The PAU is controlled using the same logic as that in Option A, but the PAU outlet air temperature is reset online according to humidity ratios of the outdoor air and indoor air. The AHU supply air fan speed is modulated to maintain the static pressure in supply air duct at its set-point. Its cooling coil valve is modulated to control the AHU supply air temperature at its set-point, which is reset automatically to maintain cleanroom dry bulb temperature below the preset upper limit with minimum heating energy consumption in the terminals. The terminal heating coils in individual cleanrooms are controlled to ensure the cleanroom dry bulb temperature above the preset lower limits and cleanroom relative humidity below their upper limits. The control mechanism of supply air flow and static pressure in the cleanrooms are the same as that of Option A.

3.3 Option C – Fully Decoupled Option



(a) System configuration of Option C



(b) Air handling process control in PAU and AHU of Option C

Fig. 3 System configuration and air handling process control of Option C - Fully Decoupled Option

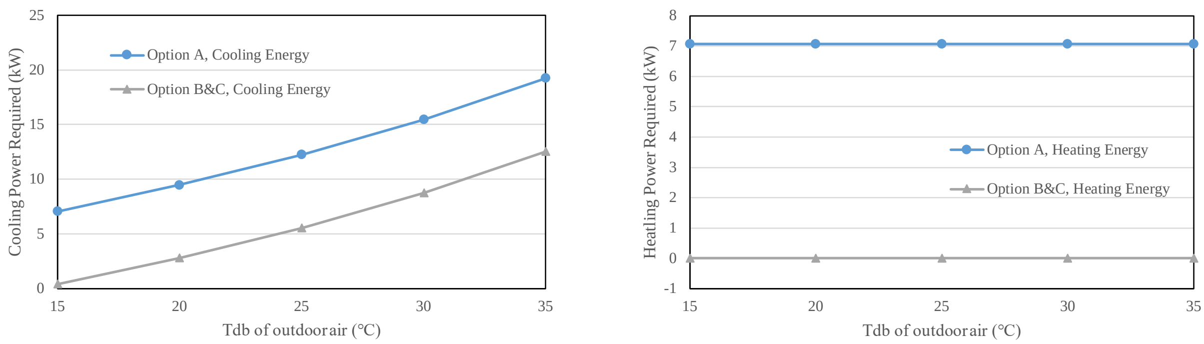
Fig. 3 shows the system configuration and air handling process of Option C. This configuration differs from Option A and Option B mainly in the AHU, because the AHU employs two parallel cooling coils. The air handling process is, therefore, more complicated than the other two Options, as demonstrated in Fig. 3(b). The PAU cools down the outdoor air to certain low temperature to share parts of cooling and dehumidification loads of the outdoor air and cleanrooms ($O \rightarrow L_1$). The outdoor air is mixed with the recirculation air ($L_1 \rightarrow M_1$, $I_1 \rightarrow M_1$) and is then divided into two streams, each goes through one cooling coil and mixed again as the supply air to terminals. One of the AHU cooling coil is used as wet coil for dehumidification ($M_1 \rightarrow L_2$), the other one is used for temperature control ($M_1 \rightarrow D$). The two air streams are mixed as ($L_2 \rightarrow M_2$, $D \rightarrow M_2$).

The control of Option C is demonstrated in Fig. 3(a). Compare with Option A and Option B, the main difference of this option is the cleanroom humidity control. The maximum relative humidity of the associated cleanrooms is used to control the air flow ratio of the two air streams in AHU. The outlet air temperature of the wet cooling coil is controlled at a low set-point for dehumidification, and the other coil is controlled to maintain a reasonable supply air temperature based on the need of sensible cooling to avoid terminal reheating if not necessary. The controls of terminal reheating, supply air dampers, return air dampers are the same as that of Option B.

4 Comparative analysis on the performance of typical air-conditioning systems

The three typical system options are analyzed based on the same working conditions: a cleanroom with an area of 64.2 m^2, and a volume of 162.5 m^3. The required air change per hour (ACH) is 20, and the minimum outdoor air flow is 150 L/s. The internal sensible heat ratio is 0.95 and fixed. The performance of the three system options is compared in different weather and load conditions. The calculation analysis was conducted based on moist air property and the air handling process described in section 3, by using R language [23], RStudio [24] and the CoolProp library [25].

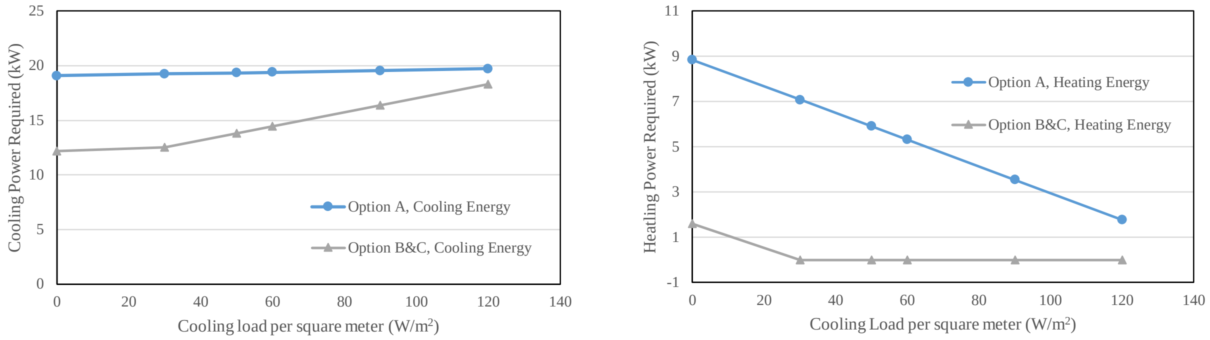
Fig. 4 shows the required cooling and heating energy in the three system options when the cooling load is 30 W/m^2 and the outdoor air dry bulb temperature (T_{db}) varies from 15 °C to 35 °C. Option A requires more cooling and heating energy than the other two options. Option B and C show the same performance since they require the same amount of cooling and heating energy in these working conditions. By comparing the three systems, it is clear that counteraction between heating and cooling exists in Option A.



(a) Cooling energy consumption (b) Heating energy consumption

Fig. 4 Energy consumption comparison in different weather conditions at fixed cooling load

Fig. 5 shows the cooling and heating energy when the internal cooling load changes and the outdoor air dry bulb temperature is fixed at 35 °C. As the internal cooling load increases, cooling energy required by Option A remains at a high level with slight increase, while the cooling energy required by the other two options increases obviously. Heating energy required by Option A decreases with the increase in cooling load. Option B and C require very limited heating energy in very low cooling load working conditions, and do not require any heating when cooling load is a bit higher. It can be seen that the difference between the energy consumptions of Option A and the other two options reduces as cooling load increases. In other words, Option B and C have apparent advantage over Option A in part load conditions.



(a) Cooling energy consumption (b) Heating energy consumption

Fig. 5 Energy consumption comparison in different load conditions at fixed outdoor temperature

Theoretically, Option C reaches the boundary of energy efficiency, because it fully decouples temperature and humidity control in the system. It is therefore recommended for new systems. Although Option B have the same performance with Option C in the analyzed condition, its dehumidification capacity is limited to the amount of fresh air flow.

Since system configuration of Option B is nearly the same as that of Option A, it is therefore proposed for retrofit projects where system configuration of Option A is originally designed, as the case of the retrofit project reported in this paper. Hereafter, the control method of Option A is named as "reference method" and the control method of Option B is named as "proposed method".

5 Retrofit project and validation test arrangements

5.1 Building air-conditioning system and cleanroom requirements

The pharmaceutical factory building is located in Hong Kong, a humid sub-tropical city. It has a height of around 23 m and a gross floor area of about 9,000 m^2. The total floor area of cleanrooms is about 2,700 m^2, and the rest is for general use such as staff offices. All production areas are Class ISO 8 cleanrooms based on ISO 14644-1 cleanroom standards [26], while the laboratory spaces are Class ISO 7 cleanrooms. The cleanrooms have strict requirements on dry bulb temperature, relative humidity, air change per hour, and static pressure. The dry bulb temperature of all cleanrooms should be controlled between 16 °C and 24 °C, and the relative humidity should be within 35% – 65% or 35% – 70%, depending on the function of different cleanrooms. The air change per hour is required to be not less than 20 for cleaning airborne particles in the space. Different cleanrooms may also have different requirements on positive static pressure (i.e. 0 Pa, 15 Pa, 30 Pa, 45 Pa, 60 Pa or 75 Pa) depending on their functions. This is to ensure the air, and hence particulate contaminant, does not pass from adjacent areas into the cleanrooms of higher grade [27].

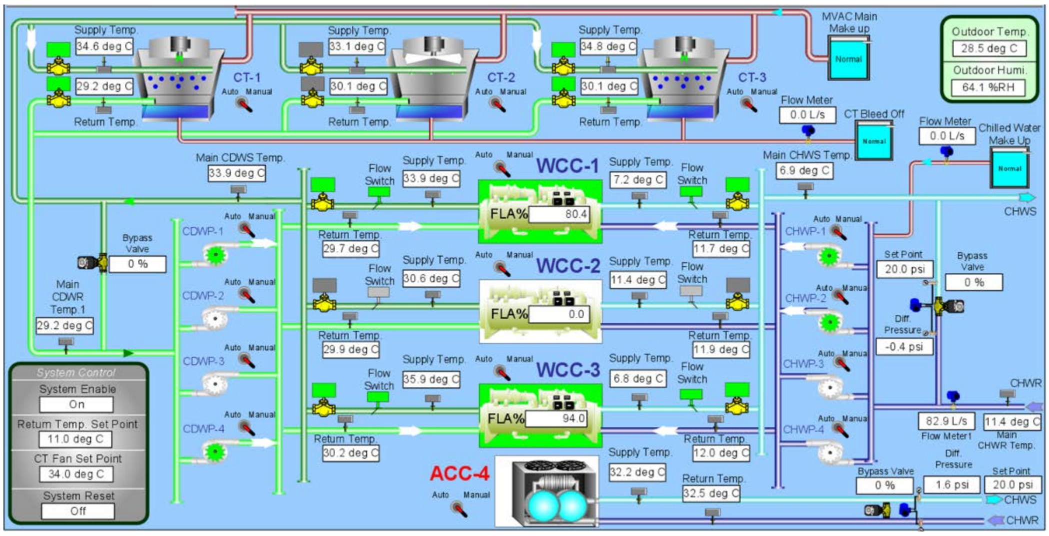


Fig. 6 System configuration of the chiller plant in the building

Fig. 6 shows the configuration of the chiller plant for the building. The entire system consists of three water-cooled screw chillers for duty and an air-cooled chiller for backup. The chilled water delivery system is a primary pump only system installed with variable speed drivers, i.e. a primary variable flow system. “Common head” is applied to cooling towers, cooling pumps, chillers and primary chilled water pumps, so that all equipment backups each other within the groups.

On the air side, two types of systems are used, i.e. PAU+AHU systems for cleanrooms and PAU+FCU systems for spaces of general use.

5.2 On-site test

On-site tests were conducted to evaluate the actual performance of the proposed control method in comparison with the reference method. Prior to full implementation, an on-site trial test was conducted to evaluate the performance of the proposed method when switched from the reference method. After that, the proposed method was further fully implemented in the air-conditioning system of one zone in the building, and the test results were compared with that of the air-conditioning system of another zone of the same configuration using the reference control method.

5.3 System dynamic simulation platform

A dynamic system simulation platform was also built to evaluate the performance of the proposed methods and the interaction among different control loops comprehensively. The simulation platform was built using Matlab/Simulink and based on a single AHU that serves six cleanrooms. Details on these cleanrooms and the control settings are provided in Table 1.

Table 1 Parameters and control settings of cleanrooms served by an AHU

Room No.	Room function	Volume (m^3)	Air change per hour	Pressure (Pa)	Dry bulb temperature (°C)	Relative humidity (%)
216	Raw material Sampling entrance	38.15	≥20	30	16-24	35-65
217	Shoe changing	10.11	/	0	16-24	35-65
218	Gowning	10.68	≥20	15	16-24	35-65
219	Hand disinfection	9.58	≥20	30	16-24	35-65
220	Raw material sampling area	59.44	≥20	45	16-24	35-65
221	Raw material sampling & exit	34.56	≥20	30	16-24	35-65

The dynamic simulation platform was built based on a few simplifications and assumptions as follows. (1) Supply and return air flow rates to individual cleanrooms are proportional to the opening of the corresponding dampers. (2) Hot water and chilled water flow rates in the coils are proportional to the openings of the corresponding valves. (3) The cleanrooms are strictly airtight; (4) The outdoor air from PAU is well mixed with recirculation air. (5) The air in each cleanroom is well mixed.

The platform mainly consists of a cleanroom model, an AHU model, and mass and energy balance models. Those models are integrated in the platform to simulate the dynamic air handling process and the thermodynamics of the air in cleanrooms (including humidity, dry bulb temperature and static pressure).

Cleanroom model

Derived from the ideal-gas equation of state [28], Eq. (1) is used to estimate cleanroom pressure based on air mass and temperature in cleanrooms. During the estimation, the air in cleanrooms is assumed as ideal gas, and molar mass of the air is assumed to be same as dry air.

$$P = \frac{R^*}{M}\rho T \tag{1}$$

where, P is cleanroom absolute pressure, Pa. R^* is the ideal gas constant, 8.31446 J/(K • mol). M is the molar mass of dry air, 0.029 kg/mol. ρ is the density of humid air, kg/m^3. T is the air temperature, K.

AHU model

The AHU heating and cooling coils are simulated using the model developed by Lebrun et al. [29][30]. The outlet air and chilled water temperature from coil can be calculated by Eqs. (2-3) based on heat balances of both sides.

$$t_{a,out} = t_{a,in} - \frac{SHR(t_{a,in} - t_c)}{R_1 C_a} \tag{2}$$

$$t_{w,out} = t_{w,in} + \frac{t_c - t_{w,in}}{R_2 C_w} \tag{3}$$

where, C is capacity flow rate. t is temperature. *SHR* is sensible heat ratio. Subscript "*w*", "*a*", "*c*", "*in*", and "*out*" represent water, air, coil, inlet, and outlet, respectively. R_1 and R_2 are heat transfer resistances in water side and air side, respectively.

The classical number of transfer unit (NTU) and effectiveness (ε) method is used for heat transfer calculation. Two different methods are applied for dry and wet regions on the air side. In the dry region, Eqs. (4-6) are used to calculate the overall heat transfer resistance (R). The function used in Eq.(5) is actually the heat transfer effectiveness relations, which is detailed in [31].

$$NTU = \frac{UA}{C_{\min}} = \frac{A}{C_{\min}\left(R_a + R_m + R_w\right)} \tag{4}$$

$$\varepsilon = f\left(N_{row}, \frac{C_{\min}}{C_{\max}}, NTU\right) \tag{5}$$

$$R = \frac{T_{a,in} - T_{w,in}}{Q} = \frac{1}{\varepsilon C_{\max}} \tag{6}$$

where, A is the total heat transfer surface area. R_a, R_m, R_w are heat transfer resistances of air side convection, coil metal conduction, water side convection. N_{row} is the number of tube rows, which is used to determine flow types. If $N_{row} > 2$, it is considered to be counter flow, otherwise, it is cross flow.

In the wet region, a fictitious air flow is assumed. Eqs. (7-8) are used to calculate the air capacity flow rate and air convention coefficient of the fictitious air flow. The same method used in dry region is then used to calculate the overall heat transfer resistance (R), as shown in Eqs. (9-13).

$$C_{af} = m_a c_s \tag{7}$$

$$h_{a,wt} = h_a \frac{c_s}{c_{pi}} \tag{8}$$

$$NTU_f = \frac{UA}{C_{\min,f}} = \frac{A}{C_{\min,f}\left(R_{a,wt} + R_m + R_w\right)} \tag{9}$$

$$\varepsilon_f = f\left(N_{row}, \frac{C_{\min,f}}{C_{\max,f}}, NTU_f\right) \tag{10}$$

$$Q_{wt} = \varepsilon_f C_{\min,f}\left(T_{aw,in} - T_{w,in}\right) \tag{11}$$

$$\varepsilon_{wt} = \frac{Q_{wt}}{\left(T_{a,in} - T_{w,in}\right)C_{\min,f}} \tag{12}$$

$$R = \frac{T_{a,in} - T_{w,in}}{Q_{wt}} = \frac{1}{\varepsilon_f C_{\min,f}} \tag{13}$$

where, c_s is specific heat of saturation moisture air at the mean temperature of inlet air wet bulb temperature and $t_{w,in}$. C_{af} is air capacity flow rate. h is air convection coefficient. c_{pi} is the specific heat of moisture air. Subscript "*wt*" and "*f*" represent wet region and fictitious air flow respectively. R is the overall heat transfer resistance.

With the overall heat transfer resistance computed, the R_1 and R_2 in Eqs. (2-3) can be calculated using Eqs. (14-15).

$$R_1 = R\frac{R_a + R_m/2}{R_a + R_m + R_w} \tag{14}$$

$$R_2 = R\frac{R_w + R_m/2}{R_a + R_m + R_w} \tag{15}$$

6 Results and discussion of simulation tests and on-site tests

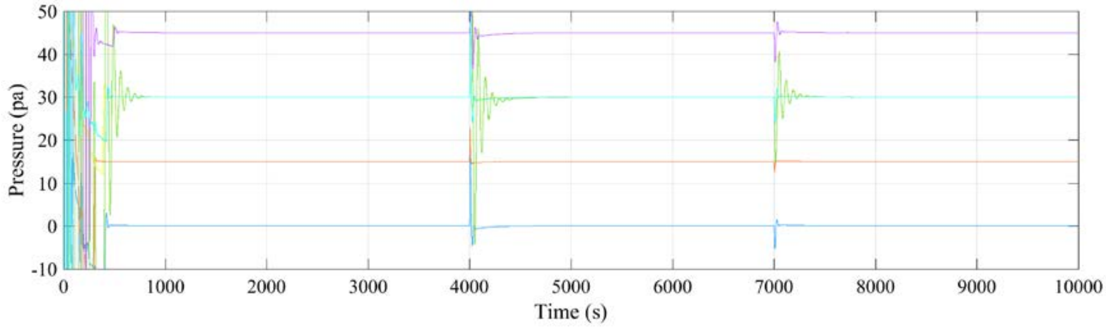
6.1 Simulation test results

The simulation validation tests were conducted on the simulation platform prior to on-site tests. During the tests, the cleanroom static pressure, dry bulb temperature and relative humidity were initialized to be 0 Pa, 28 °C and 90% respectively. Step changes were introduced to evaluate the system dynamics and process control performance. The internal heat gain and internal moisture gain in all cleanrooms were constant at the beginning (0.02 kW/ m^2, and 0.00005 kg/s/ m^2). The internal heat gain was suddenly increased to and then remained at 0.05 kW/m^2 at time 4000 s. The internal moisture gain was increased by 20% and then remained at 0.00006 kg/s/m^2 at time 7000 s.

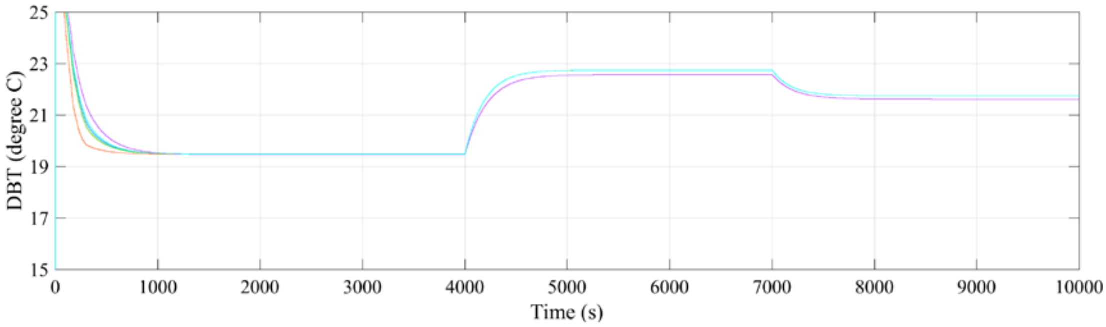
The test results are presented in Fig. 7(a-c). At starting stage, the cleanroom pressures fluctuated dramatically due to two reasons. Firstly, the initial cleanroom pressure was too far from the set-points. Secondly, it was affected by the other two parameters (dry bulb temperature and relative humidity) being tuned. At time 4000 s and 7000 s, when there was a step increase in internal heat gain and internal moisture gain respectively, the pressures in all cleanrooms could be well adjusted to the original set-points after a few fluctuations.

Air dry bulb temperatures in cleanrooms increased from 19.5 °C to about 22.5 °C when there was a step increase in internal heat gain at 4000 s. They decreased to about 21.5 °C when internal moisture gain had a step increase at 7000 s. It is noticeable that different cleanrooms may have different temperatures, but the temperatures were still maintained within the required range.

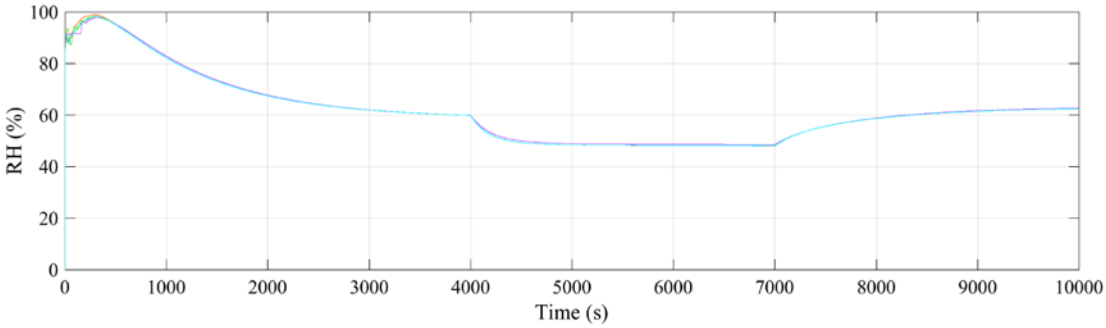
Air relative humidity in the cleanrooms increased a little bit at the starting stage because of the decrease of dry bulb temperature. It then gradually decreased to 60% at 4000 s. When there was a step increase in internal heat gain, the relative humidity decreased due to the increase of dry bulb temperature. When internal moisture gain increased at 7000 s, relative humidity in cleanrooms also increased, but they remained below the preset upper limit. However, it is noticeable that relative humidity in cleanrooms may not be maintained within the preset range if the internal moisture gain increased too much. The proposed method may therefore not be applicable for cleanrooms where the internal moisture loads are too high.



(a) Static pressure in cleanrooms



(b) Dry bulb temperature in cleanrooms

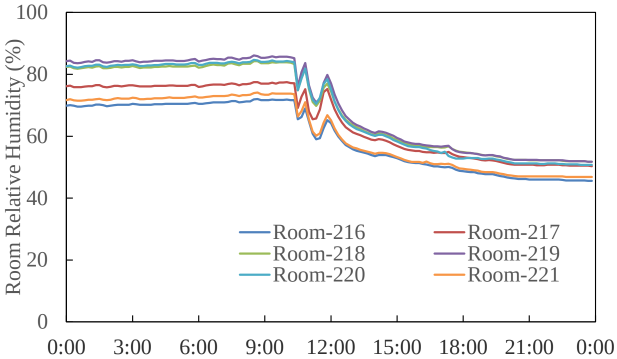


(c) Relative humidity in cleanrooms

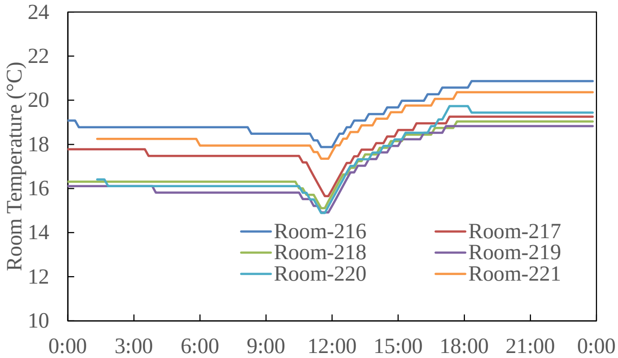
Fig. 7 Controlled indoor variables against internal load changes in simulation tests

6.2 On-site test results

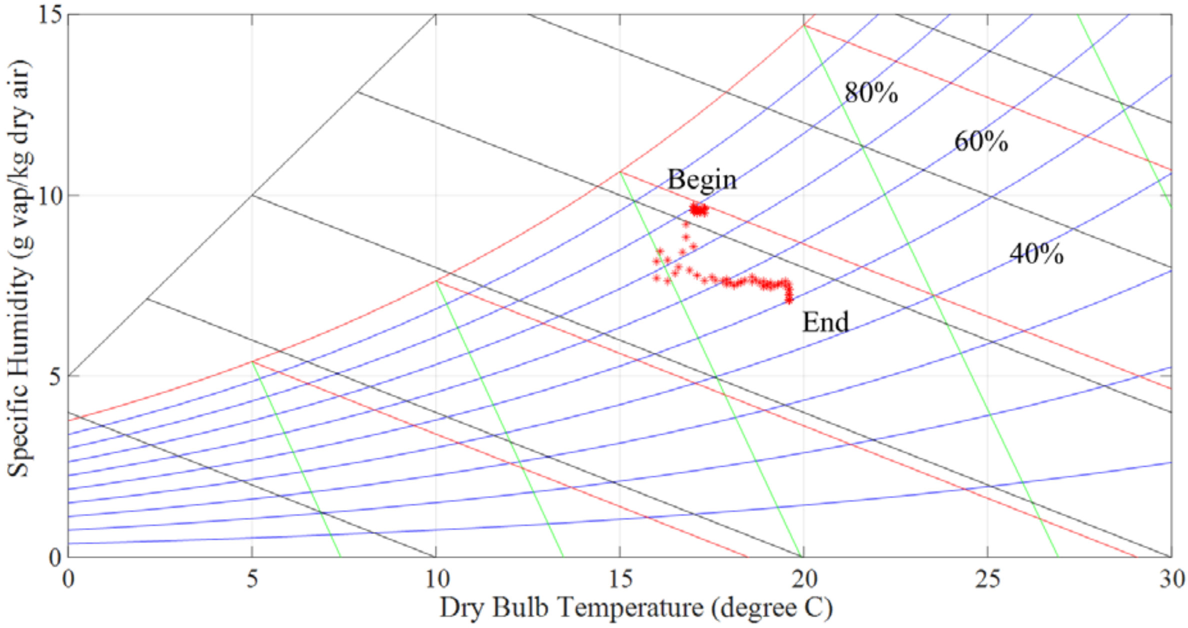
Results of both on-site trail test and implementation of proposed control method are presented in this section. Prior to full implementation, an on-site trial test was conducted to evaluate the proposed method. The heating in the entire system of a zone concerned was turned off during the test day. At the start of the test day, the reference control method was used and it is switched to the proposed control method at around 11:00AM. Fig. 8 shows the temperature and relative humidity of the cleanrooms served by the AHU concerned. When the AHU was controlled by the reference method, the indoor relative humidity cannot be controlled within their preset ranges, as there was no heating supply. When switched to proposed control methods, the relative humidity decreased and the dry bulb temperature increased. Fig. 8(c) shows the process in the psychrometric chart. After about two hours, the indoor relative humidity could be eventually controlled within the preset limit (60%) without the use of heating while the indoor temperatures were controlled at a higher level within the preset range. It can be also observed that it took a relatively long time for the relative humidity to be reduced below the preset upper limit. That indicates that the dehumidification capacity of using the outdoor air only is limited and might not be suitable in the cases cleanrooms have high internal dehumidification loads. But this option is a proper option in the cases the internal dehumidification loads are not very high and there is no sudden large increase in dehumidification load.



(a) Cleanroom relative humidity



(b) Cleanroom dry bulb temperature

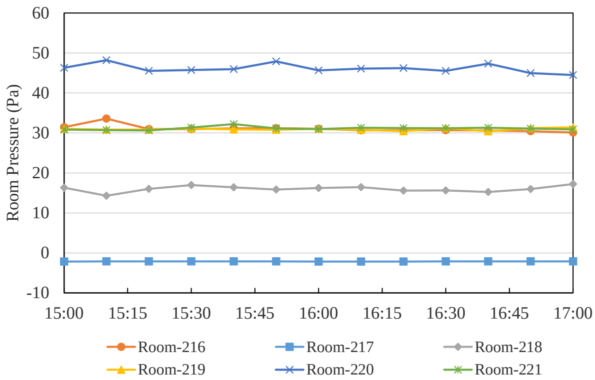


(c) Average cleanroom conditions on psychrometric chart during the on-site test

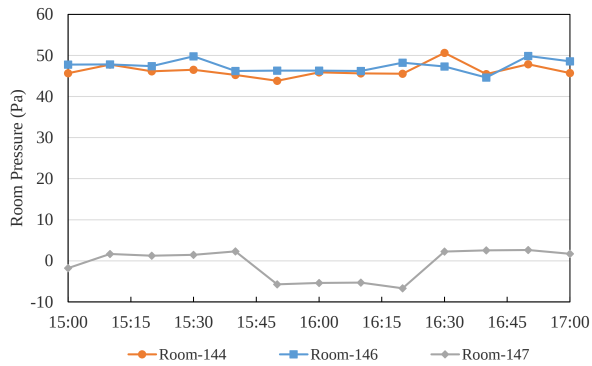
Fig. 8 On-site test of switching from the reference method to the proposed method

The proposed method was further fully implemented in the air-conditioning system of one zone (System Z-I) in the building before the full implementation, and the test results were compared with that of the air-conditioning system of another zone (System Z-II) of the same configuration using the reference control method. The System Z-I consists of 1 PAU, 5 AHUs and 17 cleanrooms. The System Z-II consists of 1 PAU, 6 AHUs and 18 cleanrooms. During the test day, the average outdoor air temperature and relative humidity were 21.8 °C and 75% respectively. Operation data of two AHUs (each from the two systems) on the same test day were extracted from the building management system and they are presented in Fig. 9.

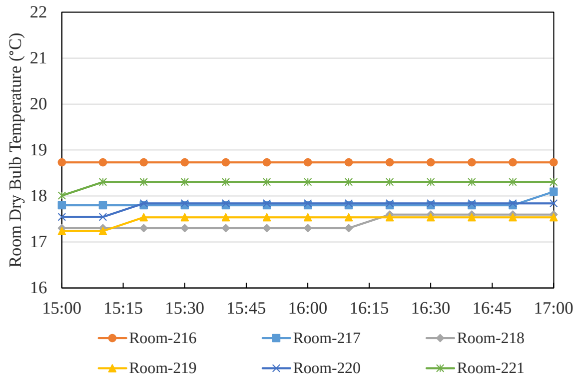
As shown in Fig. 9(a-b), both control methods can control the cleanroom pressure well. However, comparing Fig. 9(c) with Fig. 9(d), it can be observed that the cleanroom temperature controlled by proposed method was higher than that controlled by the reference method whereas the cleanroom humidity controlled by the proposed method was lower than that controlled by the reference method (Fig. 9(e-f)). Fig. 9(g-j) shows the valve openings of the cleanroom reheating coils and AHU heating/cooling coils. It can be seen that the proposed method fully closed the heating/reheating coil and cooling coil valves and did not use any heating and cooling. That indicates cooling was purely provided by the PAU. In the case of using the reference control method, those valves were open and caused significant energy for heating/reheating and cooling. One of the terminals even fully used its reheating capacity since its reheating valve was fully open. The AHU cooling valve was always fully open during the test day, and its AHU heating valve was fluctuating between 0% and 40%. Operating data of PAU, Fig. 9(k-l), show that the proposed method maintained a lower PAU outlet air temperature and larger cooling valve opening than that using the reference method.



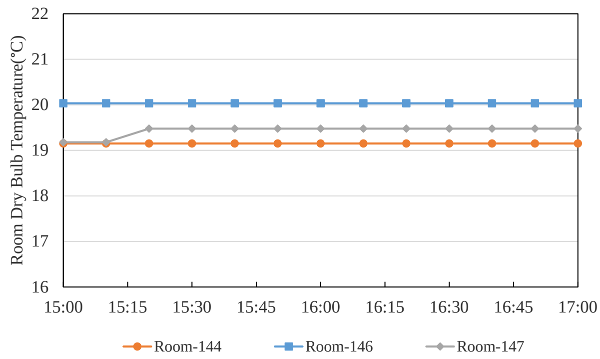
(a) Static pressure in cleanrooms



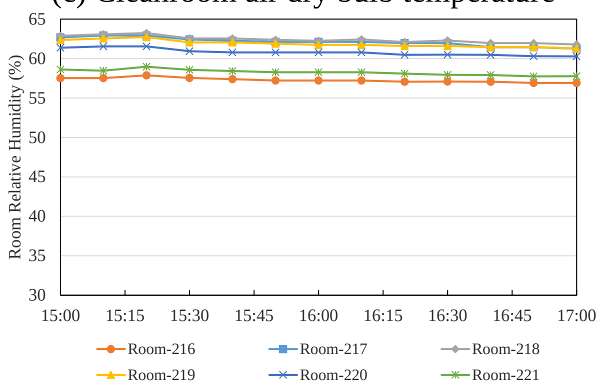
(b) Static pressure in cleanrooms



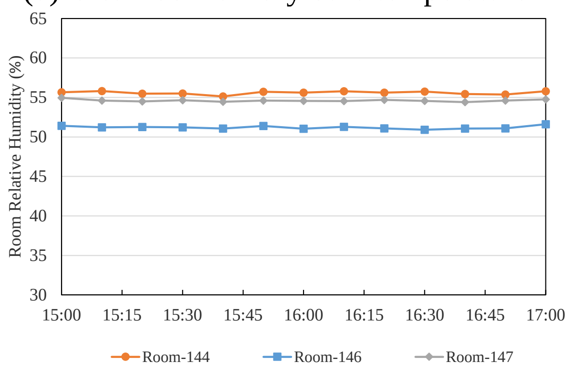
(c) Cleanroom air dry bulb temperature



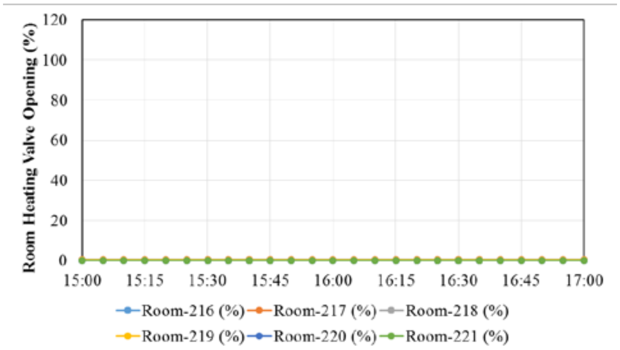
(d) Cleanroom air dry bulb temperature



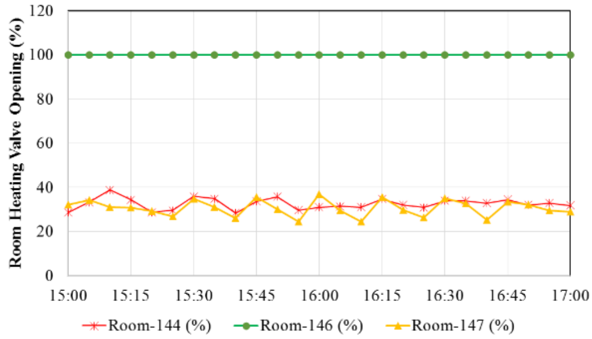
(e) Relative humidity of air in cleanrooms



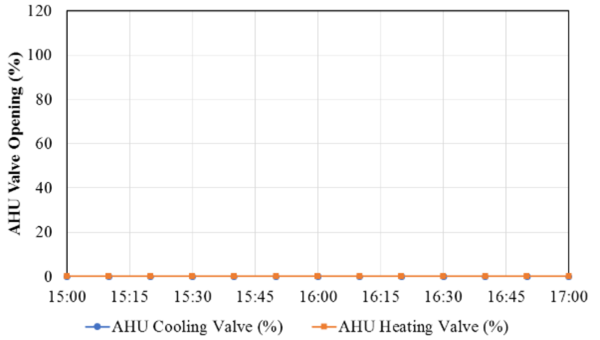
(f) Relative humidity of air in cleanrooms



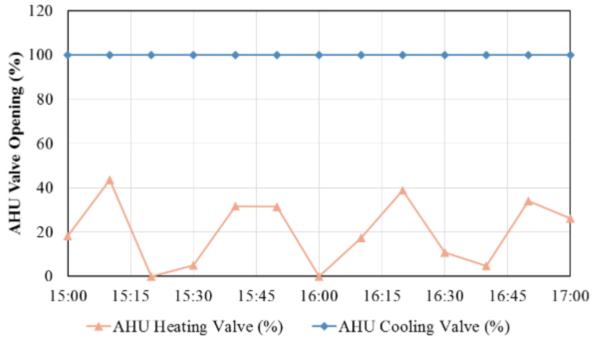
(g) Valve opening of terminal reheating coils



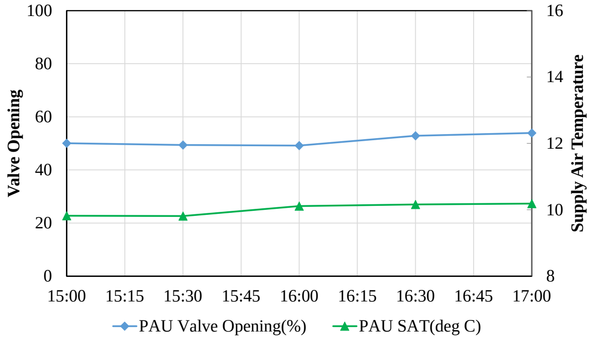
(h) Valve opening of terminal reheating coils



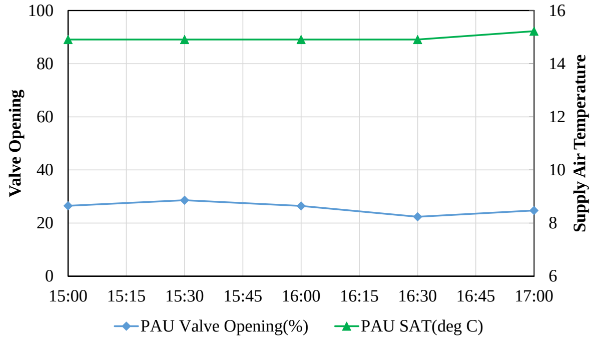
(i) Valve opening of AHU heating/cooling coils



(j) Valve opening of AHU heating/cooling coils



(k) PAU cooling valve opening and outlet air temperature



(l) PAU cooling valve opening and outlet air temperature

Fig. 9 Performance comparison between proposed method (left) and reference method (right)

6.3 Energy saving

To estimate the energy saving brought by the proposed method over the reference method, the energy saving of System Z-I is calculated by comparing the cases when the operation parameters are the actually controlled values and when they were the controlled values in System Z-II using reference control method in the same test day. The operation parameters controlled by two methods are shown in Table 2:

Table 2 Typical operation parameters controlled by two different methods in the test day

Control method	Reference method	Proposed method
Outdoor /recirculation air flow ratio	15:85	
Outdoor air conditions	dry bulb temperature is 21.8°C, relative humidity is 75.2%	
Average air dry bulb temperature in cleanrooms	20°C	18°C
Average air relative humidity in cleanrooms	53%	60%
PAU outlet temperature	15°C	10°C
AHU cooling coil outlet air temperature	10.1°C	16.8°C

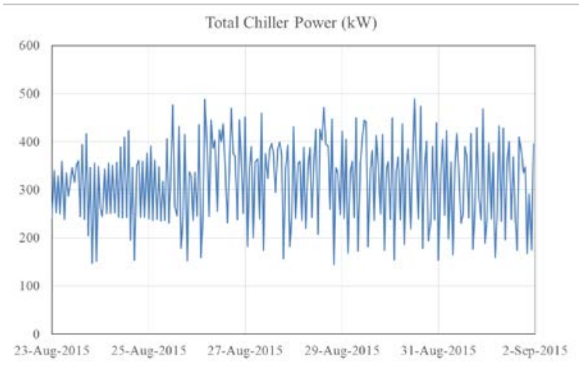
Table 3 shows the cooling and heating energy consumptions when using the two control methods. It can be observed that the proposed method saved 48.4 kW (69.6%) for cooling and 50.9 kW (87.8%) for heating/reheating. The calculated savings in cooling and heating are very close confirming the reliability of calculation and measurements. The difference is due to the effect of site measurement deviations. It can be concluded that the counteraction between cooling and heating was huge, i.e. high as 48.4-50.9 kW and about 77.9% of heating and cooling energy, when using the reference method in the test day.

Table 3 Comparison between energy consumptions of air-conditioning system in a zone using two different control methods

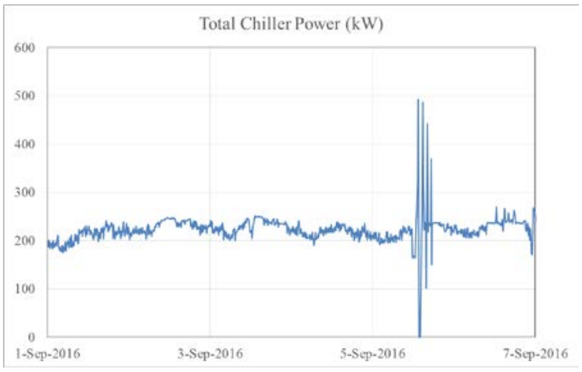
	Reference method (kW)	**Proposed method** (kW)	**Saving** (kW)
Cooling (PAU+AHU)	9.7 + 59.8	21.1 + 0	48.4 (69.6%)
Heating (AHU+terminals)	58.0	7.1	50.9 (87.8%)

It is worth noticing that the proposed control method has been fully implemented in all the zones to retrofit the reference control method originally used in the building except three very small zones due to the limitations of system designs. The total annual cost saving of electricity consumption for cooling and gas consumption for heating is about 4.6 million HKD. Fig. 10 provides a comparison of the energy consumption in the building in similar weather conditions (DBT: 27°C-28.9°C, RH: 78%-88%) before and after the implementation. The average reduction was 94 kW in chillers power consumption. And the estimated saving in chiller plant was 129 kW by assuming: (a) the COP

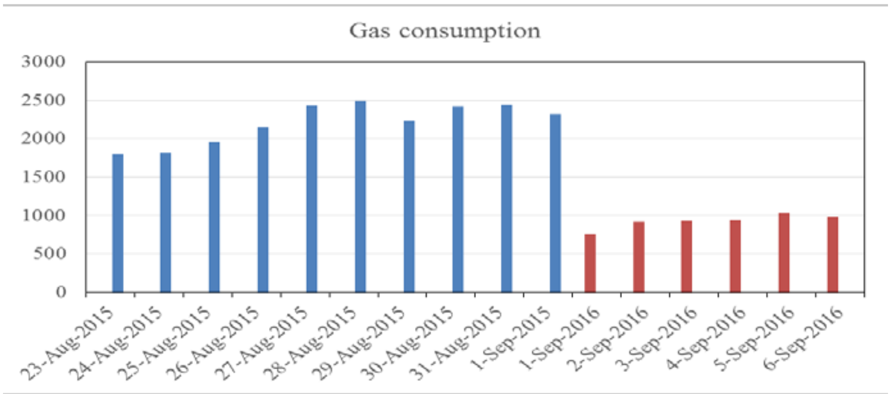
(Coefficient Of Performance) of chiller plant and the chillers are 4.0 and 5.5 respectively, and (b) the savings in cooling towers and pumps are proportional to that in chillers. The average daily reduction was 1276 units (1 unit = 48 MJ) in gas consumption. Because of different weather conditions, the relative saving is different from the values shown in Table 3.



(a) Chiller power (before**)**



(b) Chiller power (after)



(c) Gas consumption (before and after)

Fig. 10 Chiller power consumption and gas consumption before and after the implementation

6.4 Discussion on test results

Two main phenomena were observed from the on-site test results by replacing the reference method with the proposed method. (1) Dry bulb temperatures of cleanrooms are controlled to be closer to that of outdoor air, but still within the required range allowing less cooling or heating energy; (2) Cooling/heating coils in AHUs and cleanroom heating terminals are used much less, but PAU cooling is used much more. These two phenomena accordingly could be explained as follows:

(1) Using the proposed method, dry bulb temperature and relative humidity of cleanrooms are not strictly controlled at certain fixed points as that in the reference method. Instead, the two parameters

are allowed to vary within their required range according to ambient conditions. For instance, cleanroom temperature will be low if the ambient temperature is low.

(2) Using the proposed method, The PAU cools down the outdoor air to certain lower temperature to share parts of cooling and dehumidification loads of the outdoor air and cleanrooms. As a result, the AHU does not need to use extra cooling for dehumidification and provide extra reheating to achieve cleanroom temperature requirement.

Energy saving is therefore achieved by allowing cleanroom conditions to vary within the required range according to ambient conditions and particularly eliminating the counteraction between cooling and heating. Although the proposed control method requires more cooling energy to operate the PAU than the reference method, the overall energy consumption is reduced tremendously.

In real application, apart from the detailed control methods provided in section 3, attention should be paid to the control of individual cleanroom conditions. Actuators only perform the tuning when the cleanroom conditions are about to reach their predefined boundaries. In this way, the cleanroom conditions are allowed to drift within the required range.

7 Conclusions

This study investigated few essential issues for the design and control of cleanroom air-conditioning systems. The existing systems and controls are categorized into three typical options and their performance is analyzed based on different weather and load conditions.

The "partially decoupled option" is selected for retrofitting the existing air-conditioning system of a building which has similar system configuration with the conventional design and a significant energy saving is achieved (4.6 million HKD per year). The on-site validation tests on the performance of systems using "partially decoupled option" for retrofitting the existing systems using "coupled option" were conducted as well as being tested on a dynamic platform.

The simulation test results show that using “partially decoupled option”, the air humidity, temperature and pressure in cleanrooms can be controlled properly and simultaneously. On-site test results were compared with the “interactive option” that was the original design of the air-conditioning system in the building. Although PAU consumed more energy than that in the original “interactive option”, the “partially decoupled option” successfully eliminated the counteraction between cooling and heating. Analysis on the energy saving shows that the “partially decoupled option” reduced the electricity and gas consumptions for cooling and heating by 69.6% and 87.8%, respectively.

Since adopting “partially decoupled option” requires no hardware modification when retrofitting the existing system of “interactive option”, it is particularly suitable for retrofit projects of similar situations. Considering dehumidification capacity, systems of “partially decoupled option” is applicable for cleanrooms with relatively low internal moisture gains, or cleanroom air-conditioning systems with relatively large PAU cooling coils. For those cleanrooms with high internal moisture gain, “fully decoupled option” might be more suitable. For the design of new air-conditioning systems, it is very important to note that systems of “interactive option” often have very high energy consumption in practical applications when the dehumidification loads are high compared with the sensible cooling loads such as in humid subtropical regions. This problem of high energy consumption is also due to the fact that the control of such systems is very sensitive to load conditions and easy to fall into the control mode, which has serious and simultaneous cooling and heating.

Acknowledgement

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Highlights

1. Three typical air-conditioning system options are comparatively analyzed

2. The “partially decoupled option” is proposed for retrofit projects

3. Simulation and on-site test results show the proposed method performed properly

4. The annual cost saving is about 4.6 million HKD in the reported retrofit project