Since January 2020 Elsevier has created a COVID-19 resource centre with free information in English and Mandarin on the novel coronavirus COVID-19. The COVID-19 resource centre is hosted on Elsevier Connect, the company's public news and information website.

Elsevier hereby grants permission to make all its COVID-19-related research that is available on the COVID-19 resource centre - including this research content - immediately available in PubMed Central and other publicly funded repositories, such as the WHO COVID database with rights for unrestricted research re-use and analyses in any form or by any means with acknowledgement of the original source. These permissions are granted for free by Elsevier for as long as the COVID-19 resource centre remains active.
Energy consumption analysis on a dedicated outdoor air system with rotary desiccant wheel

Weiwei Liu\textsuperscript{a}, Zhiwei Lian\textsuperscript{a,}*, Reinhard Radermacher\textsuperscript{b}, Ye Yao\textsuperscript{a}

\textsuperscript{a}Institute of Refrigeration and Cryogenics, Shanghai Jiao Tong University, 800 Road Dongchuan, Shanghai 200240, China
\textsuperscript{b}Department of Mechanical Engineering, University of Maryland, College Park, MD 20742-3035, USA

Received 18 October 2005

Abstract

A dedicated outdoor air system (DOAS) with rotary desiccant wheel is the combination of a desiccant dehumidification system and a vapor compression refrigeration system. An energy consumption model of this hybrid DOAS is established for its analysis. Coefficient of performance, \textit{COP}, is appropriately defined for evaluation on performance of the hybrid DOAS. The results indicate that, compared with a conventional DOAS, energy savings are possible for the suggested DOAS, when solar energy or natural gas is used for regeneration. Ventilation air flow rate, temperature or humidity of outdoor air, as well as regeneration-to-process air ratio, influence the energy consumption and the \textit{COP} of the hybrid DOAS, greatly.

\textcopyright{} 2006 Elsevier Ltd. All rights reserved.

Keywords: Dedicated outdoor air system; Energy consumption; Rotary desiccant wheel; Hybrid

1. Introduction

The severe acute respiratory syndrome (SARS) has made people attach importance to the possibility of disease transmission by air. So, as the most important air-handling unit, air conditioning systems become the focus of attention. Aiming at the challenges by the SARS calamity for air conditioning systems, it is required that the air conditioning systems should not only maintain the thermal and humidity environment for comfort and the indoor air quality for health, but also reduce humidified surfaces and the possibility of internal air circulation between different zones [1].

A dedicated outdoor air system (DOAS) is a good choice to meet these new requirements. In this system, outdoor air, the ventilation makeup air, is separately conditioned without first being mixed with indoor air, with the entire humidity load and part of sensible load handled in the process (other part of sensible load is handled by the indoor air conditioning unit) [2,3]. Because indoor air does not recirculate among different zones, the possibility of disease transmission by it is greatly reduced.

A conventional DOAS uses cooling coils to accomplish dehumidification. On the one hand, higher the humidity load, lower is the dew-point temperature required. But, the evaporator temperature of the chiller must not go below 0 \textdegree{}C, the freezing point, which means that the conventional DOAS cannot be applied when the required dew-point temperature is lower than 0 \textdegree{}C. Therefore, solving the problem of the limitation on the dew-point temperature is the crux to employ DOAS widely. On the other hand, there is condensation water during the cooling dehumidification process, which is a breeding ground for mildew.

Desiccant technology is a promising alternative for dehumidification. Compared with cooling dehumidification, it has lots of advantages, three of which are that no limitation on the dew-point temperature, no condensation water, and having greater energy saving potential. So, a DOAS using desiccant dehumidification will be an ideal system without the problems existing in the conventional DOAS. In this paper, a DOAS with rotary desiccant wheel is suggested.
A solid desiccant-based hybrid air-conditioning system (HACS) with return air used can save energy [4,5]. A DOAS with rotary desiccant wheel (a hybrid DOAS) suggested in this paper is a hybrid air-conditioning system without return air. It is uncertain whether energy consumption for this hybrid DOAS is larger or smaller compared with the conventional one. So analysis on energy consumption of the hybrid DOAS is one of the main purposes in the present study. Some factors, such as temperature and humidity of outdoor air and regeneration-to-process air ratio, can affect energy consumption and performance of HACS [5–7]. Accordingly,
effects of these factors on the hybrid DOAS are also studied here.

2. DOAS with rotary desiccant wheel

In a conventional DOAS (as shown in Fig. 1), outdoor air is first dehumidified by cooling below its dew-point temperature (the entire humidity load and part of sensible load are handled in the process), and then reheated to a proper temperature (when required) via a heater, before entering the room. The indoor sensible cooling unit handles other part of the sensible load. The corresponding psychrometric process is indicated in Fig. 3(a). As shown in Fig. 3(a), delivered into the room, the ventilation air absorbs excess heat and moisture in the room, with both its temperature and humidity increasing to the same with indoor air (from point “O” to point “N”).

As depicted in Fig. 2, the suggested DOAS with rotary desiccant wheel is the combination of a desiccant dehumidification system (DDS) and a vapor compression refrigeration system (chillers). It is the same in the two DOAS that outdoor air must remove the whole humidity load and part of the sensible load, and the indoor sensible cooling unit handles the other part of the sensible load. But the means of dehumidification is different. In the hybrid DOAS, outdoor air, the process air, is dehumidified when passing through the desiccant wheel. In this process, its temperature increases, which is the result of the dehumidification effect and the heat transfer from the regeneration airstream via the desiccant wheel. As the process air flows through the heat recovery wheel, the heating effect is partially offset by transferring heat from the process air to the cooler incoming regeneration air. The process air is further cooled to the required temperature in the chiller 1, and then enters the room. The whole psychrometric process is shown in Fig. 3(b), and it can be seen that there is truly no limitation on the dew-point temperature in the suggested hybrid DOAS.

In the DDS, the regeneration air is heated to a sufficiently high temperature to bring out the moisture extracted by the sorbent in the desiccant wheel. Before entering the heater, the regeneration air is preheated by the heat recovery wheel, which leads to less energy consumption. The regeneration air can be outdoor or indoor air or a mixture of both. In order to avoid the pollution of the process air by the regeneration air, because of the leakage between the process area and the regeneration area, outdoor air is a better choice.

In the two DOAS, heat recovery of the exhaust air from the room is not considered here.
3. Computational modeling

The energy consumption of the hybrid DOAS is composed of that resulting from dehumidification and that for chillers (energy consumption for fans/pumps is not considered here). Based on the energy consumption model, three coefficients of performance \((COP)\) are separately defined for evaluation on performance of the chiller, the DDS and the whole hybrid DOAS.

3.1. Mass flow rate of ventilation air

Since the ventilation air assumes the entire humidity load, its mass flow rate is found from

\[
G = \frac{HL}{W_N - W_o}. \tag{1}
\]

In DOAS, the ventilation air is of 100% outdoor air.

3.2. Energy consumption for DDS

DDS is shown in Fig. 4. Energy consumption of DDS is mostly that heating up the regeneration air to the regeneration temperature. The heat can be calculated as

\[
Q_r = m_r c_r (t_r - t_2). \tag{2}
\]

There are several ways to heat the regeneration airstream, such as electricity, natural gas, solar energy or waste heat. So the energy consumption of DDS varies with the different methods. For energy comparison, all the energy inputs in these methods are normalized to electrical power consumption in this paper.

If electricity is directly used to heat the regeneration airstream, the electrical power input should be equal to the heat

\[
E_h = Q_r. \tag{3}
\]

If natural gas is used, the currency of natural gas can be expressed in the electrical power with a conversion factor, which is defined as incorporating the heating value per unit cost of natural gas and electrical energy

\[
C_{ge} = \frac{C_e}{C_g}. \tag{4}
\]

So

\[
E_h = C_{ge} Q_r. \tag{5}
\]

For solar energy, since it is renewable, it can be considered that almost no electrical power needs to be consumed

\[
E_h = 0. \tag{6}
\]

The regeneration temperature and the outlet states of the process or regeneration air are determined by the heat and mass transfer process in the desiccant wheel. These parameters can be obtained by solving the governing equations or by use of the performance curves supplied by wheels manufacturers.

When the process or regeneration air passes the heat recovery wheel, its humidity ratio keeps unchanged, but its
temperature changes, which can be obtained as
\[ t_{p3} = t_{p2} - \eta \min(m_p c_p, m_r c_r) (t_{p2} - t_{i1}) / m_p c_p, \]
(7)
\[ t_{i2} = t_{i1} + \eta \min(m_r c_r, m_p c_p) (t_{p2} - t_{i1}) / m_r c_r. \]
(8)
Here \( m_p = G \).

The efficiency of heat recovery wheels is about 0.7 [3,8].

3.3. Energy consumption for chillers

The power needed for chillers can be calculated as
\[ E_c = Q_c / \text{COP}_c. \]
(9)
The chillers in the hybrid DOAS only handle the sensible load, so \( Q_c \) can be expressed as follows.

For chiller 1
\[ Q_c = G c_p (t_{p3} - t_o). \]
(10)
For chiller 2
\[ Q_c = Q - Q_p, \]
where \( Q \) is the total indoor sensible load and
\[ Q_p = G c_p (t_N - t_o). \]
(12)

3.4. Definitions for coefficient of performance

To evaluate the performance of an air-conditioning system, coefficient of performance, COP, is appropriately defined as
\[ \text{COP} = \frac{\text{The cooling capacity of the system}}{\text{The energy used by the system}}. \]

Based on this definition, the COP of the chiller, the DDS and the whole hybrid DOAS can be separately given as follows.

3.4.1. COP of the chiller

The chiller adopts a vapor compression refrigeration system, as shown in Fig. 5(a).

There are pressure drops in both the condenser and the evaporator. The compression is not an isentropic process.

Considering the complexity of the actual refrigeration cycle, some assumptions have to be made before calculation:

(a) mass flow rate of refrigerant in each component is the same;
(b) the efficiency of the compressor is regarded as a constant;
(c) the condensation of refrigerant in the condenser is an isobaric heat rejection process and the evaporation in the evaporator is an isobaric heat absorption process;
(d) the degrees of subcooling and superheating are maintained at fixed values;
(e) the refrigerant throttling process through the expansion valve is adiabatic.

According to Fig. 5(b), COP of the chiller, \( \text{COP}_c \), is
\[ \text{COP}_c = \eta_{\text{com}} \frac{h_1 - h_4}{h_2 - h_1}. \]
(13)

Considering the actual existence of irreversible losses, including the non-isentropic process in the compression of compressor, a factor, “\( a \)”, is introduced to reflect the situation, Eq. (13) can be expressed as
\[ \text{COP}_c = a \eta_{\text{com}} \frac{h_1 - h_4}{h_2 - h_1} = \eta_{\text{c,com}} \frac{h_1 - h_4}{h_2 - h_1}, \]
(14)
where \( \eta_{\text{c,com}} \) is the equivalent compressor efficiency.

Usually the efficiency of adiabatic compression is about 0.85, and 0.9, 0.8–0.9 for motor and mechanical efficiency, respectively [9].

3.4.2. COP of the DDS

An appropriate definition for the COP of the DDS, \( \text{COP}_l \), is
\[ \text{COP}_l = Q_l / E_h, \]
(15)
where \( Q_l \), the equivalent latent cooling load, is calculated as
\[ Q_l = G (h_{pl} - h_{p1}). \]
(16)

Point \( P_l \) in Fig. 3(b) represents a hypothetical thermodynamic state corresponding to the temperature of the
process inlet air and the humidity ratio of the process exit air [7].

3.4.3. COP of the DOAS

$COP_s$ of the DOAS, $COP_s$, is defined as

$$COP_s = \frac{Q_s}{E_s}. \quad (17)$$

For the conventional DOAS, the total cooling capacity, $Q_s$, is the cooling load handled by the two chillers. And for the rotary desiccant wheel based DOAS, $Q_s$ is the sum of the sensible load handled by the two chillers and the latent cooling load handled by the DDS.

For the conventional DOAS, the energy expended to achieve the desired cooling capacity $Q_s$ is

$$E_s = E_c. \quad (18)$$

For the rotary desiccant wheel based DOAS

$$E_s = E_c + E_h. \quad (19)$$

4. Results and analysis

In this calculation, performance of the desiccant wheel was obtained via the software of DW [10]. And the software of engineering equation solver (EES) is used to calculate the thermal physical properties of refrigerant R22 and $COP$ of the chillers [11].

4.1. Desiccant wheel used in the hybrid DOAS

The desiccant wheel is an important component in the hybrid DOAS. Here, the outlet states of the process and regeneration air were obtained by using the performance curves supplied by a desiccant wheels manufacturer (the software of DW), which was also adopted in the simulations of Refs. [12,13]. The structure parameters of the desiccant wheel are listed in Table 1 and its performance is shown in Table 2.

4.2. Comparisons between the two DOAS by energy consumption and $COP$

The comparisons were done for two mass flow rates of ventilation air, 0.833 and 6.667 kg/s, respectively, under the outdoor conditions of 35 °C air temperature and 18 g/kg d.a. moisture content. For the indoor conditions, the air temperature was set at 26 °C and moisture content at 11.6 g/kg d.a. The indoor sensible load was 200 kW and the humidity load 0.004 kg/s. The temperature difference between indoor air and ventilation air was taken as 8 °C. Table 3 shows humidity ratio of ventilation air (state o in Fig. 3(b)), dew temperature for the conventional DOAS and regeneration temperature for the DDS. Evaporator temperatures of the chillers in the two DOAS are separately presented in Table 4.

In the DDS, regeneration mass flow rate here is equal to that of the process air ($R/P = 1.0$). Energy consumption and $COP$ were discussed when electricity, natural gas, and solar energy were separately used to regenerate the air. According to the heating value per unit cost of natural gas and electricity in Shanghai in China, the conversion factor $C_{ge} = 0.34$.

Energy consumption of the two DOAS is shown in Fig. 6. Energy consumption of chiller1 in the hybrid DOAS is always smaller than that in the conventional DOAS, because chiller1 in the hybrid DOAS only handles part of the sensible load, while chiller1 in the conventional DOAS handles the entire humidity load and part of the sensible load, which are larger, and $COP$ of chiller1 in the hybrid DOAS can be higher than that in the conventional DOAS. In the two DOAS, chiller2 handles the same sensible load and has the same $COP$, so energy consumptions of chiller2 are equal. Compared with the conventional DOAS, energy

| Table 1      | Structure parameters of the desiccant wheel |
|--------------|-------------------------------------------|
| Desiccant media | Wound silica gel |
| Wheel diameter (m) | 1.525 |
| Wheel depth (m) | 0.20 |
| Regeneration portion (%) | 25 |
| Wheel speed (rph) | 24 |

| Table 2      | Performance data of the desiccant wheel |
|--------------|-----------------------------------------|
| Process air mass flow rate (kg/s) | Regeneration/ process air volume ratio | Heater temperature (°C) | Humidity ratio decrease of process air (g/kg d.a.) | Temperature rise of process air (°C) |
| 4           | 1.0     | 90       | 7.3     | 48.4       |
|            |         | 110      | 7.7     | 54.5       |
|            |         | 130      | 8.0     | 60.2       |

| Table 3      | Humidity rate, dew temperature, regeneration temperature |
|--------------|----------------------------------------------------------|
| Mass flow rate (kg/s) | Humidity ratio at state o (g/kg d.a.) | Dew temperature (°C) | Regeneration temperature (°C) |
| 0.833       | 6.8         | 10       | 110       |
| 6.667       | 11.0        | 17.2     | 72.2       |

| Table 4      | Evaporator temperatures of the chillers (°C) |
|--------------|---------------------------------------------|
| $G = 0.833$ kg/s | Conventional DOAS | Hybrid DOAS |
| Chiller1       | 2     | 10    | 10 |
| Chiller2       | 12    | 12    | 12 |

| $G = 6.667$ kg/s | Conventional DOAS | Hybrid DOAS |
| Chiller1       | 2     | 10    | 10 |
| Chiller2       | 12    | 12    | 12 |
consumption of the heater in the hybrid DOAS is higher when electricity or natural gas is used.

If solar energy is used for heating the regeneration air, it can be simply considered that no energy consumed for the heater, so the total energy consumption for the hybrid DOAS is lowest. But, if electricity is used, the total energy consumption for the hybrid DOAS is highest. These results agree with the findings based on the experiments in Ref. [6]. When natural gas is used, the result will vary with mass flow rates of ventilation air. For example, when the mass flow rates of ventilation air takes 0.833 kg/s, the total energy consumption for the hybrid DOAS is 6.6% less than that of the conventional DOAS, while it takes a larger value, 6.667 kg/s, 22.5% more energy is consumed for the hybrid DOAS.

Fig. 7 depicts $COP$ of the two DOAS. In the hybrid DOAS, $COP$ of chiller1 and chiller2 will not change, no matter what kind of energy is used for regeneration, but $COP$ of the whole air conditioning system (ACS) is not the same that it is highest if solar energy is used, while lowest with electricity used.

Compared with the conventional DOAS, when the mass flow rate of ventilation air is 0.833 kg/s, higher evaporator temperature of chiller1 leads to higher $COP$ of chiller1 in the hybrid DOAS, and $COP$ of the whole system is lower if electricity is used for regeneration, while that is slightly higher if natural gas is used. However, results are different when the mass flow rate is larger (6.667 kg/s): $COP$ of chiller1 in the two DOASs is equal, but $COP$ of the whole hybrid DOAS is lower unless solar energy is used. The two DOASs have the same $COP$ of chiller2 because of the same evaporator temperatures.

4.3. Effects of mass flow rate of ventilation air on energy consumption and $COP$

Suppose electricity is used for regeneration, variation of energy consumption and $COP$ are illustrated in Figs. 8 and 9, respectively, when mass flow rate of ventilation air changes from 0.65 to 2.25 kg/s. Table 5 lists the corresponding moisture removal rate and the humidity rate of ventilation air at state o in Fig. 3(b).
According to Fig. 8, it is certain that the total energy consumption rises as ventilation air flow rate increases, because of more ventilation air flow rate, more humidity load and sensible load. In the total energy consumption, based on the same reason, energy consumption for chiller1 and heater also goes up. However, energy consumption for chiller2 reduces, the reason for which is that since the total indoor sensible load is invariable, increase in the part of this sensible load handled by chiller1 leads to reduction in the part handled by chiller2 that, in turn, reduces the energy consumption for chiller2.

As long as evaporator temperatures of the chillers keep unchanged, their COP certainly does not change. And it appears that ventilation air flow rate only has slight effect on COP of the DDS according to Fig. 9. But, it can be seen from Fig. 9 that COP of the whole system unexpectedly reduces when ventilation air flow rate increases. It can be explained as follows.

The COP of the whole system lies on the COP of the components (the DDS and the chillers) and their ratio of energy consumption to the total. Though the COP of the components changes little, the percentage of energy consumption for the chiller2, COP of which is highest, decreases, which reflects in the reduction in the COP of the whole system.

### 4.4. Effects of humidity and temperature of outdoor air on energy consumption and COP

First, temperature of outdoor air will affect the heat conducting through the envelope of the building, which can be calculated as

\[
Q = KF(t_W - t_N).
\]

It presumes that 40% of the total indoor sensible load, 80 kW, is due to the heat conducting through the envelope under 35°C outdoor air temperature. So \( KF = \frac{80}{(35 - 26)} = 8.9 \text{ kW/°C} \). When temperature of outdoor air increases from 28 to 36 °C, change of the corresponding total indoor sensible load is shown in Table 6.

Mass flow rate of the ventilation air (outdoor air) is 1.429 kg/s, and \( R/P \) is 0.5.

As shown in Fig. 10, the effects of outdoor air humidity and temperature on energy consumption of the components are different. When temperature or humidity of outdoor air rises, chiller1 has to handle more sensible heat due to the dehumidification, so its energy consumption increases too. As to chiller2, humidity of outdoor air exerts no effect on the indoor sensible load handled by it, as well as its energy consumption. However, higher outdoor air temperature leads to greater indoor sensible load, as inferred from Table 6. So, chiller2 is required to handle more sensible load, with energy consumption increasing.
According to Fig. 10(c), energy consumption for DDS goes up greatly as outdoor air humidity rises, while temperature of outdoor air only has slight effect on that, which is determined by the change of regeneration temperature of the DDS. Consequently, based on the energy consumptions of the components, the total energy consumption increases with the increasing temperature or humidity of outdoor air.

From Fig. 10(d), the total energy consumption increases 78.2%, when outdoor conditions change from 28 °C air temperature and 13 g/kg moisture content to 36 °C and 17 g/kg.

Fig. 11 shows the effects of humidity and temperature of outdoor air on COP. It can be seen that the effects on COP of the DDS (COP₁) is tiny, and COP₁ changes between 0.62 and 0.70. As to COP of the whole system (COPₛ), it is enhanced as temperature of outdoor air goes up, but when humidity of outdoor air rises, the result is reversed. In the calculation, the suggested hybrid DOAS has the highest COPₛ, 3.044, in the outdoor conditions of 36 °C air temperature and 13 g/kg moisture content, while the lowest

| Temperature of outdoor air (°C) | 28  | 30  | 32  | 34  | 36  |
|---------------------------------|-----|-----|-----|-----|-----|
| Total indoor sensible load (kW) | 137.8 | 155.6 | 173.4 | 191.2 | 209.0 |

Table 6
Corresponding total indoor sensible load

![a](image1)

![b](image2)

![c](image3)

![d](image4)

Fig. 10. Effects of humidity and temperature of outdoor air on energy consumption.
COP, 2.367, in that conditions of 28 °C and 17 g/kg. The reason to explain the effects on COP of the whole system can be consulted that in Section 4.2.

When outdoor conditions change, regeneration temperature must be adjusted to keep steady indoor conditions. The change of regeneration temperature is provided in Fig. 12.

Higher humidity ratio of outdoor air, more moisture needs to be removed, which leads to increase in the regeneration temperature. However, it is strange that this temperature also rises as outdoor air temperature goes up, though the humidity ratio remains constant. This result can be explained as follows, based on the heat and mass transfer process in the desiccant wheel.

Suppose humidity ratio of outdoor air does not change, that is to say the moisture removal capacity of the desiccant wheel is invariable. In the process area, increase in outdoor air temperature induces increase in temperature of the sorbent, as a result of which less water content in the sorbent is required to keep humidity difference between the sorbent surface and the process air (outdoor air) unchanged. Similarly, in the regeneration area, when the water content in the sorbent becomes less, the temperature of the sorbent must increase to keep the humidity difference constant. That means the higher regeneration temperature is needed.

4.5. Effects of regeneration to process flow ratio on energy consumption for dehumidification

It is known that the desiccant dehumidification introduces an additional sensible load, which results in the rising of the temperature of process air. Considering this, energy consumption for dehumidification is the summation of that for DDS and that for handling the additional heat load.

The inlet conditions of both process air and regeneration air are assumed to be 35 °C air temperature and 18 g/kg moisture content, and the outlet conditions of process air are kept at 35 °C and 10 g/kg. The process air flow rate still takes 1.429 kg/s.

Regeneration temperature varies with regeneration to process flow ratio, which is shown in Fig. 13. The larger the regen-to-process flow ratio, the higher is the regeneration temperature required. However, when the ratio is bigger than a value, change of the regeneration temperature is slight.

In the present study, increasing regen-to-process flow ratio from 0.5 to 1.0, leads to a 40.5 °C reduction in the regeneration temperature, while that only reduces 2.8 °C, as the ratio increases from 1.0 to 2.0.
It can be seen that almost the same conclusions about the additional sensible load and the energy consumption for handling it are attained. When regen-to-process flow ratio goes up from 0.5 to 1.0, the additional sensible load decreases 60.8%, with a 61.7% reduction in the corresponding energy consumption for handling the load. However, when the ratio is larger than 1.0, there is no obvious change in the load and the corresponding energy consumption.

According to Fig. 13, the energy consumption for the DDS has a trend that it reduces at first, and then increases, as the ratio rises. Based on Eq. (2), this can be explained that the regeneration temperature decreases a lot, although the mass flow ratio of the regeneration air increases, when the regen-to-process flow ratio increases from 0.5 to 1.0. With the ratio increasing from 1.0 to 2.0, the regeneration air flow ratio increases much while the regeneration temperature changes slightly, which results in the increase of the energy consumption.

The total energy consumption for dehumidification almost has the same trend as that for the DDS. And it has a minimum when the regen-to-process flow ratio takes 1.0. The results are useful in improving performance of the DDS.

5. Conclusions

A DOAS with rotary desiccant wheel is suggested to meet the new requirements for air-conditioning systems. Based on the calculation and analysis for its energy consumption and COP, following conclusions can be obtained:

(1) Compared with the conventional DOAS, energy savings are possible for the suggested hybrid DOAS, when solar energy or natural gas is used for regeneration. As solar energy is used, the energy consumption for the suggested DOAS is lowest. (However, depending on the application, the initial cost of the solar equipment may neglect the saving achieved in the cooling and dehumidification process.) Natural gas is also a better choice for regeneration. But, if electricity is used to heat the regeneration airstream, the suggested DOAS will have the largest energy consumption and poorer COP.

(2) More ventilation air flow rate, more energy consumption and lower COP for the suggested DOAS. Therefore, a smaller ventilation air flow rate is appropriate.

(3) Temperature and humidity of outdoor air have great effects on energy consumption and COP for the suggested DOAS. The energy consumption increases with the rising of the temperature or humidity of outdoor air, and the COP goes up when the outdoor air temperature rises or the humidity decreases. Considering these effects, in order to keep steady indoor conditions, it is important to adjust the chillers and the DDS, respectively, in response to the change of outdoor conditions. Consequently, an optimal control module for that is required.

(4) There is an optimum regeneration to process flow ratio leading to the lowest energy consumption for
dehumidification. Accordingly, it is significant to determine this ratio.

References

[1] Jiang Y. Consideration of air conditioning system configuration after the SARS calamity. HV&AC 2003;33(HV&AC and SARS Supplement):4–7.
[2] John D, Roth KW, Brodrick J. Dedicated outdoor air systems. ASHRAE J 2003;45(3):58–9.
[3] Mumma SA. Designing dedicated outdoor air systems. ASHRAE J 2001;43(5):28–31.
[4] Henderson HI, Walburger AC, Sand JR. The impact of desiccant dehumidification on classroom humidity levels. ASHRAE Trans 2002;108(Part 2):HI-02-2-2.
[5] Dhar PL, Singh SK. Studies on solid desiccant based hybrid air-conditioning systems. Appl Therm Eng 2001;21:119–34.

[6] Jalalzadeh-Azar AA. Consideration of transient response and energy cost in performance evaluation of a desiccant dehumidification system. ASHRAE Trans 2000;106(Part 2):4386.
[7] Jalalzadeh-Azar AA, Glenn Steele W, Hodge BK. Performance characteristics of a commercially available gas-fired desiccant system. ASHRAE Trans 2000;106(Part 1):4326.
[8] Mazzei P, Minichiello F, Palma D. Desiccant HVAC systems for commercial buildings. Appl Therm Eng 2002;22:545–60.
[9] Yan Q. Refrigeration technology used in air conditioning system. 2nd ed. Beijing: China Mechanical Industry Publisher; 1996.
[10] Novelaire Technologies. Desiccant wheel simulation program—version 3.2.1. See also: (http://www.novelaire.com).
[11] Klein S, Alvarado F. Engineering equation solver. F-chart software, Middleton, WI, 2000.
[12] Zadpoor AA, Golshan AH. Performance improvement of a gas turbine cycle by using a desiccant-based evaporative cooling system. Energy 2006;31:2652–64.
[13] Camargo JT, Ebinuma CD, Silveira JL. Thermoeconomic analysis of an evaporative desiccant air conditioning system. Appl Therm Eng 2003;23:1537–49.