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Energy performance of independent air dehumidification systems with energy recovery measures

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Abstract

Independent air dehumidification provides an attractive alternative to traditional coupled air dehumidification with reduced energy use, better humidity control and indoor air quality. According to this concept, latent load is treated by an independent system and the sensible load is treated by chilled-ceiling panels. In this work, four independent air dehumidification systems with energy recovery strategies are proposed. They are as follows: system 1, mechanical dehumidification with heat pump; system 2, mechanical dehumidification with sensible heat exchanger; system 3, mechanical dehumidification with membrane-based total heat exchanger; and system 4: a heat pump incorporating an active desiccant wheel and evaporative cooler. They are compared with a mechanical dehumidification system with no heat recovery. Hour-by-hour energy analysis is performed on the systems proposed. The results show that the system of mechanical dehumidification with membrane total heat recovery (system 3) consumes the least primary energy. However, since, the systems employ energy recovery measures, the energy savings for the four systems are in the same order, around 4.40×10^6 kJ per person.

Keywords: Energy; Heat recovery; Air dehumidification; Air conditioning

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Nomenclature

$\dot{m}_a$	mass flow rate of air (kg/s)
Δp	total pressure rise (Pa)
c_p	Specific heat of the air (kJ kg^{-1} K^{-1})
q	heat (kW)
T	temperature (K)
V_a	air (water) volumetric flow rate (m^3/h)

Greek letters

η	efficiency
ε	effectiveness
ϕ	relative humidity

Subscripts

Eva	evaporator
Sen	sensible
Wb	wet bulb

1. Introduction

People's concern of indoor air quality has greatly deepened since, the outbreak of the SARS epidemic (severe acute respiratory syndrome) that devastated South China and some other parts of the world in the Spring of 2003. SARS was controlled at last, but many challenges will remain for HVAC engineers: What can people do to prevent the pollutions of the indoor air by fatal pollutants such as SARS virus? The mechanism of the spread of SARS virus is complicated and still remains a mystery. However, it is believed that increased fresh air ventilation is helpful in decreasing the possibility of being infected by the disease.

Increased ventilation rates usually lead to high energy consumption in air-conditioning. This is because the relative humidity in a occupied building must be controlled to within 40–60% for health and comfort. Ventilation air is the major source of moisture load in air conditioning. As shown in Fig. 1 for a moisture load estimation of a medium size retail store [1], ventilation air constitutes about 68% of the total moisture load in most commercial buildings. As a consequence, treatment of the latent load from the ventilation air is a difficult and imminent task for HVAC engineers, especially in hot and humid climates like South China.

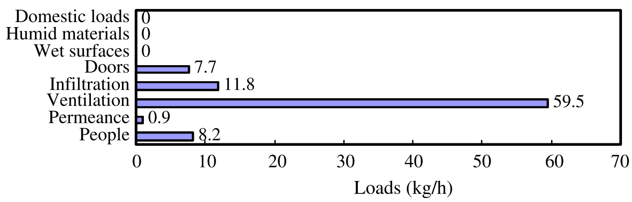

Fig. 1. Sources of moisture loads in a medium size retail store.

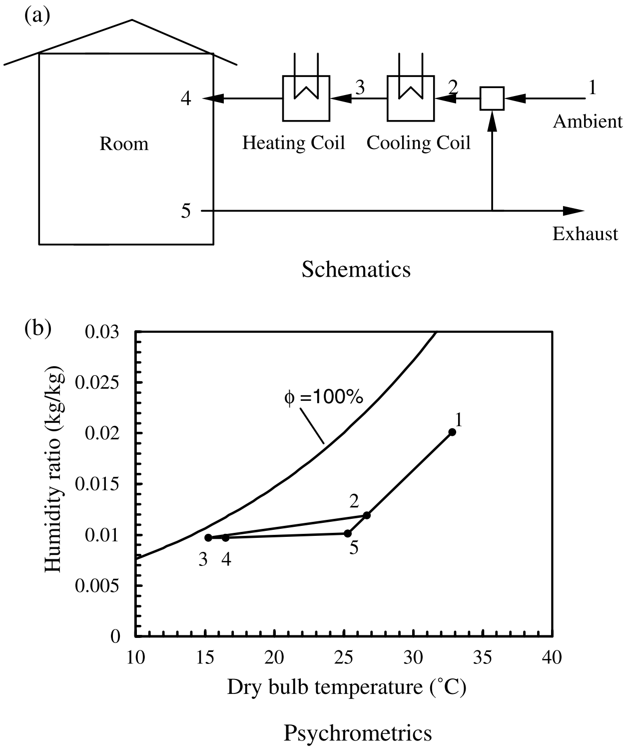

Fig. 2. A traditional constant volume all-air system, (a) schematic and (b) psychrometric chart.

There are various techniques for air dehumidification [2]. Traditionally, latent load and sensible load are treated in a coupled way. A schematic of the method and the processes in a psychrometric chart are shown in Fig. 2. Ambient air is mixed with return air first. Then the mixed air (state 2) flows through an AHU (air handling units) where it is cooled and dehumidified. It should be mentioned that to remove moisture, all the air must be cooled to below its dew point at state 3, (around 15 °C). Air at this temperature could not be directly supplied to room, to prevent cold-drafts and over-cooling that would cause discomfort to occupants. As a result, supply air is subsequently reheated by a heating coil to state 4 before distributed into the conditioned space. Because air is not only for ventilation, but also a heat transfer medium, and a large quantity of air is needed to extract the sensible load, energy requirements are very high. Another problem with this technique is that in transition seasons, when it is at part load conditions, humidity control will be lost [3].

There is an increasing trend to separate the treatment of sensible and latent load by using an independent humidity control system. According to this concept, the latent load of a room is treated by an independent humidity control system, while the sensible load is treated with some other alternative cooling techniques such as chilled-ceiling panels, or phase-change materials [4–6]. Since, the circulating air is dramatically reduced, energy consumption can be reduced substantially. Another benefit is that chilled water or suction temperatures can be raised, resulting in increased equipment efficiency and decreased operating costs. It is estimated that with a new system of chilled-ceiling panels combined with independent humidity control, 30% of energy could be saved in comparison to a traditional coupled system [7–9]. Nevertheless, due to the hot and humid climates in south China, energy for moisture control with an independent humidity control system still accounts for 25% of the total energy for air conditioning. To further reduce the energy consumption in the treatment of fresh air, energy recovery measures must be combined to an independent humidity control system.

In this study, four independent humidity control systems with heat recovery measures are proposed. These systems are: system 1, mechanical dehumidification with a heat pump; system 2, mechanical dehumidification with a sensible heat exchanger; system 3, mechanical dehumidification with a membrane-based total heat exchanger; and system 4: Desiccant wheel driven by a heat pump. Through hour-by-hour analysis, the annual primary energy consumption for the proposed systems proposed is discussed.

In the investigation, an office building with five occupants in a 20 m^2 room is considered. The set points for indoor air are as follows: temperature, 25 °C; relative humidity, 50%. Fresh air is supplied at 37.5 m^3/h, 20 °C in temperature and 7 g/kg in humidity ratio. The air dehumidification systems are operated when the offices is opened, i.e. from 9:00 to 18:00. At nights, the systems are shut down, to save energy. The air flow rates are selected according to ASHRAE Standard 62-1999 [10], which is determined by several factors such as room area, building occupancy pattern, and building types. The sensible load of the room is treated by chilled-ceiling panels. The total ventilation load includes two fractions: moisture load from fresh air and moisture load from human activities. The fresh air sensible and latent load is a variable relating to weather conditions, while the load from human activities is assumed to be kept at 50 g/h.

2. System descriptions

Four independent air dehumidification systems with heat recovery measures are proposed. The operations of the four systems are controlled to satisfy the load and weather conditions. The air flow rates are fixed. For mechanical dehumidification, both the evaporating and condensing temperatures are controlled according to the load and outside temperatures. For desiccant wheels, the regeneration temperature is adjusted to fit the load.

2.1. System 1: mechanical dehumidification with heat pump

A schematic and the corresponding psychrometric charts are shown in Fig. 3. Fresh air (point A) first flows through a cooling coil where it is dehumidified below the dew point (point B). Then the air flows through a heating coil where it is heated to the set points of supply air. The system comprises a heat pump: the cooling coil acts as the evaporator, and the heating coil acts as a condenser. The heat that should be rejected is larger than that that is required to heat the supply air, so an additional condenser is necessary to reject the surplus heat of the heat pump. Exhaust air from indoor space is pumped through this condenser to enhance the performance of the heat pump system.

2.2. System 2: mechanical dehumidification with sensible heat exchange

This concept uses a sensible heat exchanger to recover the heat of the supply air itself after it flows through the dehumidification cooling coil. As shown in Fig. 4, fresh air at point A flows through the

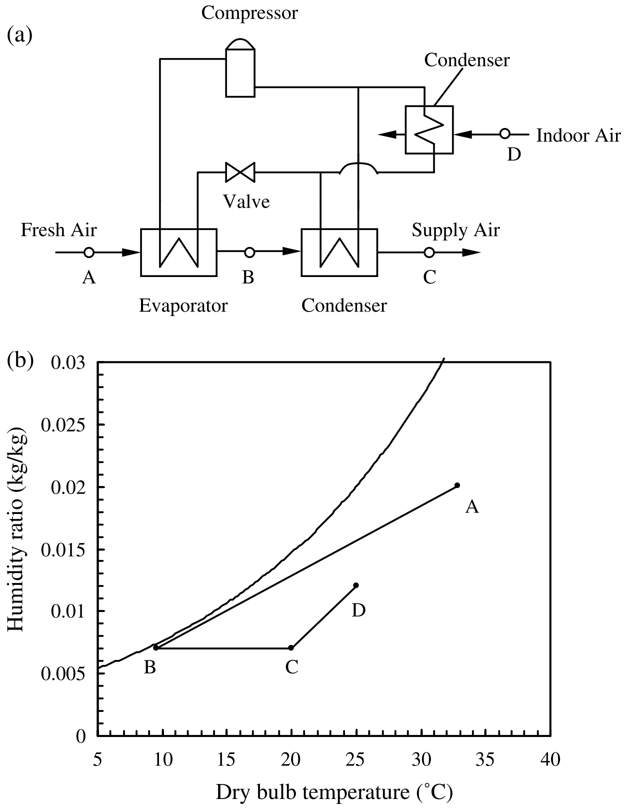

Fig. 3. Schematics (a) and psychrometrics (b) of system 1: mechanical dehumidification with heat pump.

sensible heat exchanger, where it is cooled down (in some cases, it also loses some water content due to condensation) to point B. Subsequently, it is dehumidified by a cooling coil and returns to the sensible exchanger at point C. After heated up, it is supplied to the room. The cooling coil can be an evaporator of a small refrigeration system. Then the incoming fresh air is dehumidified and subsequently heated by a heat pump system that is similar to system 1.

Without any use of the exhaust air, this design uses an air-to-air heat exchanger to pre-cool and reheat the outdoor air that is dried with a mechanical dehumidifier. Heat pipes, coil run-around loops and plate type heat exchangers are used for this purpose.

2.3. System 3: mechanical dehumidification with a membrane-based total heat exchange

In this system, a membrane based total heat exchanger is used before the fresh air is pumped to a heat pump for air dehumidification. The total heat exchanger has a membrane core where the incoming fresh air exchanges moisture and temperature simultaneously with the exhaust air. In this manner, the total heat or enthalpy from the exhaust is recovered. The schematic and the processes are shown in Fig. 5 for this system. This system is also relatively simple, since, the membrane system has no moving parts, and is compact.

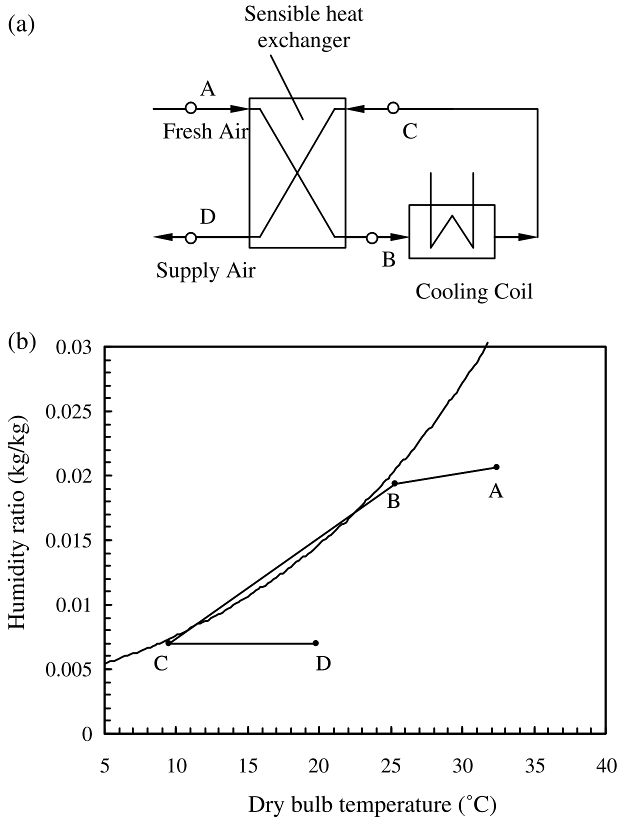

Fig. 4. Schematics (a) and psychrometrics (b) of system 2: Mechanical dehumidification with sensible heat exchanger.

2.4. System 4: mechanical dehumidification with a desiccant wheel

This two-stage equipment uses the condenser heat from a mechanical dehumidifier to re-active a desiccant wheel. First, fresh air is pre-cooled and partially dried by the mechanical dehumidifier. Then the air is dried more deeply and also reheated by the desiccant wheel. This arrangement uses both technologies at favorable points of performance. At higher inlet humidities, the mechanical refrigeration system can operate at a higher coil temperature and suction pressure, thereby saving energy. Dehumidified air from a desiccant wheel is very hot. Therefore, before it is supplied to rooms, it should be cooled down to set points first. To recover the energy from exhaust air, an evaporative cooler is used to cool down the exhaust air, which is then used to cool the supply air from desiccant wheel. Under extremely humid ambient conditions, energy required for re-activation exceeds available heat from the heat pump. In such cases, an auxiliary electric heater is used to accomplish the regeneration of the desiccant wheel. This system is relatively complex due to the rotating character of the wheel and the reactivation of the desiccants. If low grade waste heat is available, the desiccant wheel system becomes superior to other systems in energy efficiency (Fig. 6).

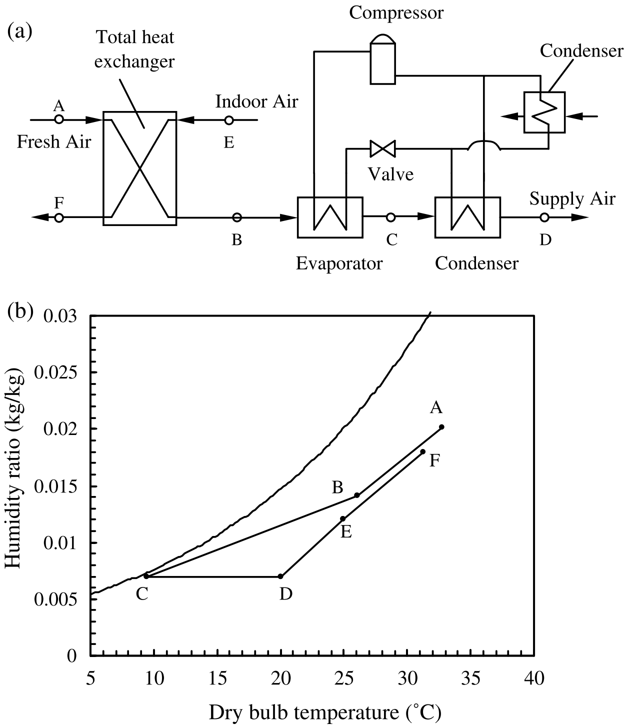

Fig. 5. Schematics (a) and psychrometrics (b) of system 3: mechanical dehumidification with membrane-based total heat exchanger.

The system with no heat recovery is shown in Fig. 7. It is the base system studied.

3. Mathematical modeling

3.1. Component modeling

At this point, energy performance of the four building dehumidification system previously described will be calculated by thermodynamic calculations. The modeling methods will be introduced taken system 4 as an example.

3.1.1. The cooling coil

Cooling and dehumidification of the incoming fresh air are performed in the evaporator. The temperature effectiveness in an evaporator is governed by the effectiveness relationship:

$$\varepsilon_{\mathrm{Eva}} = \frac{[T_{\mathrm{A}} - T_{\mathrm{B}}]}{[T_{\mathrm{A}} - T_{\mathrm{Eva}}]} \tag{1}$$

where T is temperature (K), and subscripts A, B, and Eva denotes points A, B and evaporator, respectively. A temperature effectiveness of 0.90 is assumed for the evaporator. The outlet humidity of

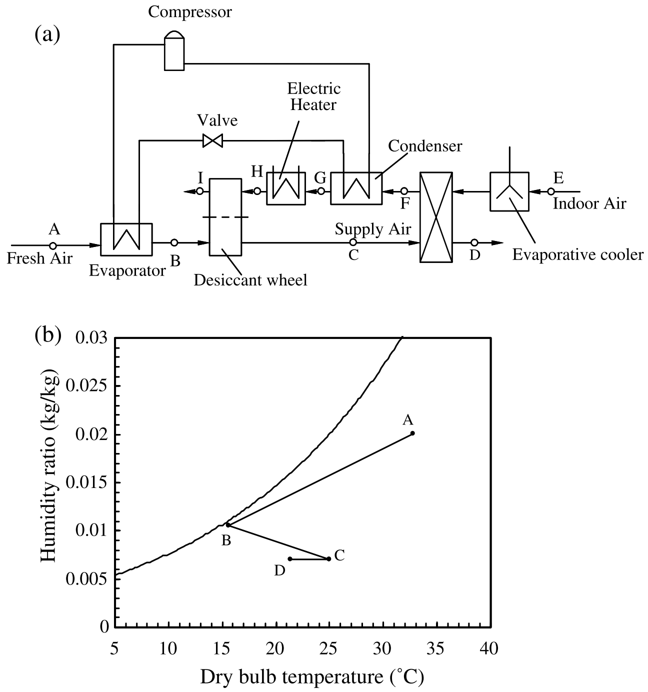

Fig. 6. Schematics (a) and psychrometrics (b) of system 4: mechanical dehumidification + desiccant wheel.

air at point B is governed by

$$\phi_B = 0.95 \tag{2}$$

3.1.2. Desiccant wheel

The purpose of the rotary dehumidifier is to dehumidify the supply air stream as it passes from state B to C. In doing so, the temperature of the supply air is raised. The humidity ratio of the return air is increased and the temperature decreased as the desiccant is regenerated.

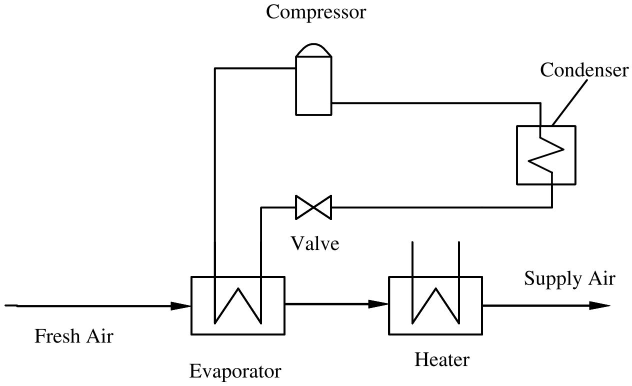

Fig. 7. The base system, air dehumidification with no recovery.

The models for the desiccant wheel can be classified into two categories: finite difference models, and correlation models. The former models are described in detail in Ref. [11], the latter one is given in Ref. [12]. The finite difference models are detailed, but complex and difficult to find stable solutions. The correlations, which are used in this study, are simple, yet sufficient for an energy analysis. The correlations are summarized as follows:

$$\eta_{f1} = \frac{f_{1so} - f_{1si}}{f_{1ei} - f_{1si}} \tag{3}$$

$$\eta_{f2} = \frac{f_{2so} - f_{2si}}{f_{2ei} - f_{2si}} \tag{4}$$

$$f_{1j} = -2865 T_j^{-1.49} + 4.344 w_j^{0.8624} \tag{5}$$

$$f_{2j} = T_j^{1.49}/6360 - 1.127 w_j^{0.07969} \tag{6}$$

where subscripts ‘s’ and ‘e’ mean ‘supply’ and ‘exhaust’, respectively, and ‘i’ and ‘o’ mean ‘inlet’ and ‘outlet’, respectively. Effectiveness η_{f1} and η_{f2} could be pre-determined by the finite difference equations, considering wheel structure and dimensions.

The desiccant dehumidifier configurations are: wheel weight, 2.5 kg; duct wall thickness, 0.2 mm; duct geometry, sinusoidal; material, silica gel; effective material fraction, 0.7; wheel length, 0.2 m, rotary speed, 30 RPH. The coefficients are: $\eta_{f1} = 0.29$; $\eta_{f2} = 0.85$.

3.1.3. Sensible heat exchanger

The purpose of the sensible heat exchanger is to transfer the heat of sorption in the dehumidified supply air stream to the exhaust air stream. The effective method can be applied to a compact heat exchanger of this type in the conventional manner. By assuming equal heat capacities for the two air flows, temperature at point D is calculated by:

$$T_D = T_C - \varepsilon_{Sen}(T_C - T_{E1}) \tag{7}$$

where ε_{Sen} is the effectiveness for the sensible heat exchanger. Based on information on the construction details of the exchanger, effectiveness values can be calculated from several sources. A constant effectiveness of 0.90 is selected for this study. In other words, a latent effectiveness of 0.72 is feasible from our experiences with membrane-based energy recovery ventilators [13].

3.1.4. The evaporative cooler

In the return air side and before the sensible heat exchanger, a rigid media air cooler is used to cool the air stream. This particular type of evaporative cooler consists of rigid, corrugated materials, which form the wetted surface. Moist air flows through the corrugations. Water enters the top of evaporative cooler and flows by gravity through the wetted surfaces. Commercially available evaporative coolers are rated according to their saturation effectiveness, ε_C, defined as

$$\varepsilon_C = (T_E - T_{E1})/(T_E - T_{Ewb})(100\%) \tag{8}$$

where T_E is the entering air dry bulb temperature, T_{Ewb} is the entering air wet bulb temperature, and T_{E1} is the exiting air dry bulb temperature.

Therefore, exit air state at point E_1 is

$$T_{E1} = T_E - \varepsilon_C\left(T_E - T_{Ewb}\right) \tag{9}$$

Process $E-E_1$ is a constant enthalpy process. Moisture content at point E_1 can be calculated from psychrometrics. Saturation effectiveness ε_C in the range of 0.7–0.9 is attainable.

3.1.5. The heating coil

Heating of the air is performed in the condenser and the electric heater. Its performance in a condenser is described by the effectiveness relationship:

$$\varepsilon_{Con} = \frac{[T_G - T_F]}{[T_F - T_{Con}]} \tag{10}$$

where T is temperature (K), and subscripts G, F, and Con denotes points G, F and condenser, respectively. An effectiveness of 0.85 is assumed for the condenser. In the electric heater, temperature rise is determined by the electric power as

$$q_{Ele1} = \dot{m}_a c_P\left(T_H - T_G\right)/\eta_{heater} \tag{11}$$

where q_{Ele1} is electricity consumed by the heater (kW), $\dot{m}_a$ is the mass flow rate of air (kg/s), c_p is the specific heat of the air (kJ kg^{-1} K^{-1}), η_{heater} is the efficiency of the heater.

It should be noted that a detailed modeling of the evaporator and the condenser are rather complicated. Usually, they are divided into regions associated to the phase of the refrigerant. Each region constitutes a separate heat exchanger. In the case of the condenser, the superheated vapor, the condensation and subcooled liquid regions are considered, whereas for the evaporator it is divided into the evaporating and superheated vapor regions. For each region, the refrigerant side and air side convective heat transfer coefficients need be calculated from the established correlations for single-phase and two-phase flow. Such methodology may be too complex for an hour-by-hour calculation. Therefore, a relative simpler, but valid method, effectiveness-NTU method, is used.

Usually, a 10 °C log mean temperature difference between the condensing refrigerant and the air flowing through it is required. Consequently, the condensing temperature is varied according to outside weather conditions.

3.1.6. Heat pump

The cooling coil acts as an evaporator of a heat pump, and the heating coil acts as a condenser for the heat pump. The efficiencies vary with evaporating and condensing temperatures. The heat pump efficiency is defined as

$$\varepsilon_{HP} = \frac{q_{Con}}{q_{Ele2}} \tag{12}$$

where q_{Con} is the heat rejected at the condenser side (kW), and q_{Ele2} is electricity consumed by the compressor (kW). The above equation is used to calculate the electric energy to drive the heat pump, from the condensing energy required and heat pump efficiencies. Depending on the operating and condensing temperatures, the heat pump efficiencies are in the range of 3–5 [14].

3.1.7. The membrane-based total heat exchanger

In case 3 for the system with membrane enthalpy exchanger, besides the component mentioned above, membrane systems should be calculated. Two effectiveness: sensible effectiveness (ε_S) and latent effectiveness (ε_L) are defined. Air state at point B in Fig. 5 is calculated by

$$T_B = T_A - \varepsilon_S (T_A - T_E) \tag{13}$$

$$\omega_B = \omega_A - \varepsilon_L (\omega_A - \omega_E) \tag{14}$$

A sensible effectiveness of 0.8 is easily obtained with a membrane system, and the latent effectiveness is about 90% of the sensible effectiveness [13].

The definition of sensible and latent effectiveness is as follows:

$$\varepsilon_S = \frac{T_A - T_B}{T_A - T_E} \tag{15}$$

$$\varepsilon_L = \frac{\omega_A - \omega_B}{\omega_A - \omega_E} \tag{16}$$

Fan and pump energy is an important factor in the annual energy consumption of an HVAC system. Fan (pump) performance can be characterized by its efficiency, which itself is dependent on operational air-flow rate. Mostly, rated volumetric flow rate, pressure rise and efficiency are available from the manufacturer. Then rated power can be calculated as

$$\text{Fan(pump)power} = V_a \Delta p / (3600 \eta_f)(W) \tag{17}$$

where V_a is air (water) volumetric flow rate (m^3/h), Δp is total pressure rise (Pa), η_f is fan (pump) efficiency.

Effectiveness of the main components are related to design and operating conditions. When the operating conditions fluctuate near design conditions, the effectiveness changes only in a small range. To simplify analysis, constant effectiveness for various components is assumed [7].

The simulations are conducted on an hour-by-hour basis. The operating hours are from 9:00 to 18:00. Fan efficiency is selected as 0.6. For the convenience of comparison, energy consumed in the form of electricity is converted to primary energy by a factor of 3.3.

The dehumidified supply air temperature is set to and fixed at 20 °C. The indoor is 25 °C. So the dehumidified air has no sensible load. Rather, it will extract a small fraction of the sensible load from the building.

Table 1
Annual primary energy requirements (kJ/person) for each person under various dehumidification strategies

	Cooling	Heating	Electricity	Condensing	Auxiliary	Fan	Total
No recovery	5.5×10^6	2.1×10^6	3.7×10^6	6.6×10^6	9.1×10^4	2.6×10^5	6.2×10^6
System 1	5.5×10^6	1.6×10^6	3.7×10^6	6.7×10^6	3.8×10^4	3.9×10^5	4.1×10^6
System 2	4.2×10^6	1.6×10^6	3.8×10^6	5.4×10^6	3.3×10^4	5.9×10^5	4.4×10^6
System 3	4.4×10^6	5.8×10^5	3.0×10^6	5.4×10^6	4.5×10^4	5.9×10^5	3.6×10^6
System 4	3.3×10^6	2.3×10^6	2.3×10^6	4.1×10^6	9.7×10^5	8.8×10^5	4.1×10^6

4. Results and discussions

4.1. Annual primary energy requirements

The systems are used to treat ventilation fresh air. To make comparisons, the annual energy requirements of the four proposed systems and a dehumidification system with no heat recovery measures as shown in Fig. 7 are computed and listed in Table 1. As can be seen, the total energy of the systems with heat recovery saved 29–42%, depending on the systems involved. In the table, the values in the column ‘cooling’ refers the energy required in the cooling coil for air dehumidification; ‘Heating’ refers to the energy need to heat the dehumidified air to the set points; ‘Electricity’ refers to the converted primary energy of electricity used by compressors; ‘Condensing’ refers to the energy rejected by the condenser of the heat pump; ‘Auxiliary’ means the converted primary energy used in electric heating, for example, the electric heater in system 4. The column ‘Fan’ refers to the converted primary energy used to circulate the air. All the energy values are calculated on a per-person basis.

In the analysis, the systems are used to treat the latent load solely, the sensible load of the room is around 50 W/m^2, which will be extracted by chilled-ceiling panels.

The comparisons of the different systems are plotted in Fig. 8. It is shown that among the 5 systems, system 3 consumes the least energy, and system 2 consumes the most. Generally speaking, energy savings for the four systems are in the same order. This is due to the reason that all the 4 systems newly proposed take into account the energy recovery measures of the exhaust air.

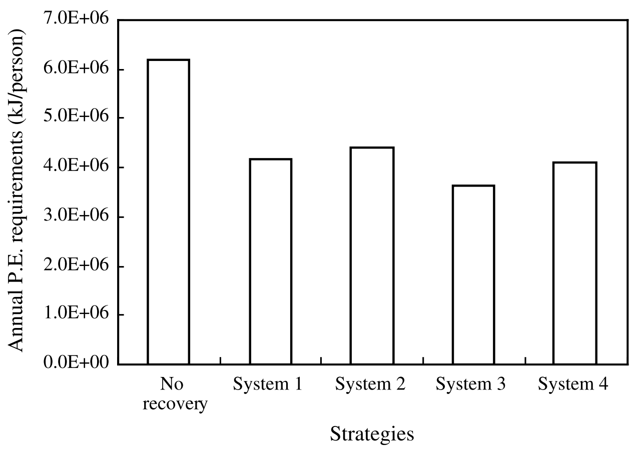

Fig. 8. Annual primary energy consumptions by air dehumidification for each person with four systems proposed.

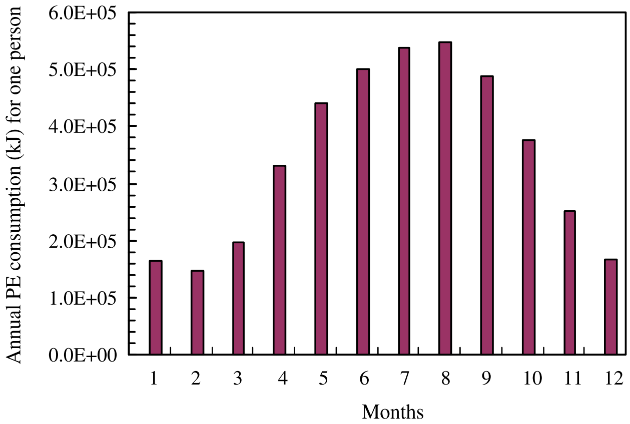

Fig. 9. Primary energy consumptions by dehumidification in each month for system 1.

Of the systems studied, three systems recover, more or less, the energy from exhaust air. They are the same use as an economizer. Outside air is fresh air that needs to be dehumidified and treated. In system 1, the indoor air is used to cool the condenser. In system 3, indoor air is used to cool and dehumidify the fresh air in a total heat exchanger. In system 4, indoor air is used to cool down the dehumidified air. system 3 is the best.

4.2. Seasonal energy requirements

Fig. 9 plots the primary energy requirements by dehumidification system 1 for each person from January to December. In hot and humid regions like Guangzhou, air dehumidification is required most of

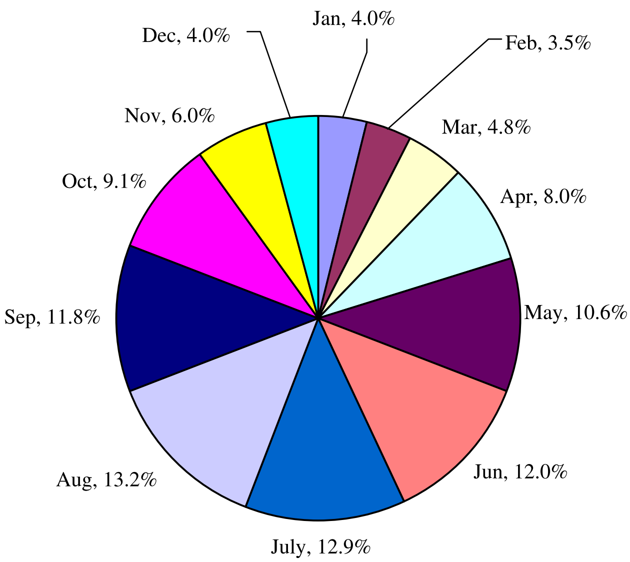

Fig. 10. Percentages of primary energy consumed in each month in a year, system 1.

the year. The energy values vary from 1.5×10^5 kJ to 5.3×10^5 kJ per month per person. In January and February, when it is the dry season, energy requirements are the least energy. In August, it is the most humid season, required energy is the largest.

The percentages of the energy required in each month to the annual total energy are shown in Fig. 10. Five months from May to September account for 60.5% of the yearly energy load. Other 7 months account for the rest 39.5%. South China has a long hot and humid summer. The energy for air dehumidification is substantially large. To save energy while ensuring a healthy built environment with enough fresh air ventilation, an independent air dehumidification system with energy recovery is necessary.

5. Conclusions

Concerns on indoor air quality have prompted the research of novel air dehumidification techniques. This study proposed 4 systems in which mechanical dehumidification is combined with energy recovery measures like a heat pump, membrane enthalpy recovery, sensible heat exchanger and desiccant wheel. An hour-by-hour simulation reveals that the independent air dehumidification with heat recovery could save 29–42% of primary energy, depending on the system involved. Of the systems proposed, the mechanical dehumidification with a sensible heat exchanger consumes the largest energy. In contrast, the mechanical dehumidification with a membrane enthalpy exchanger consumes the least. Because all the four systems use recovery measures, their energy consumptions are in the same order. The annual total primary energy used for independent air dehumidification is around 4.40×10^6 kJ per person.

Acknowledgements

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