Chapter 4
Key Components of the THIC System: Outdoor Air Handling Methods

Abstract In the THIC system, outdoor air is usually handled to a state dry enough to remove indoor moisture for humidity control during cooling season. In this chapter, requirements for outdoor air handling devices within different climate regions are discussed, and the basic air handling devices for heat recovery, dehumidification, and humidification are introduced. Evaporative cooling method, which is feasible for outdoor air handling process in the dry region, the corresponding devices, as well as the analyzing method are introduced. As to the humid region in summer, dehumidification is required to handle the outdoor air to a state dry enough. Condensation dehumidification method and solid desiccant dehumidification method are introduced in this chapter.

4.1 Basic Outdoor Air Handling Devices

4.1.1 Requirements for Outdoor Air Handling Devices in Different Climate Regions

The requirements for outdoor air handling process in the THIC system have been examined in Sect. 2.4.2 of Chap. 2. The requirements for outdoor air handling devices can be determined according to the required parameters of the supplied air. Moreover, the outdoor air handling devices should satisfy the handling requirements for different outdoor conditions. The outdoor climate parameters for summer and winter in China have been analyzed in Sect. 2.4.3 in Chap. 2; the country can be divided into two regions (Zone I and Zone II) according to the different outdoor air handling requirements, as shown in Fig. 4.1.

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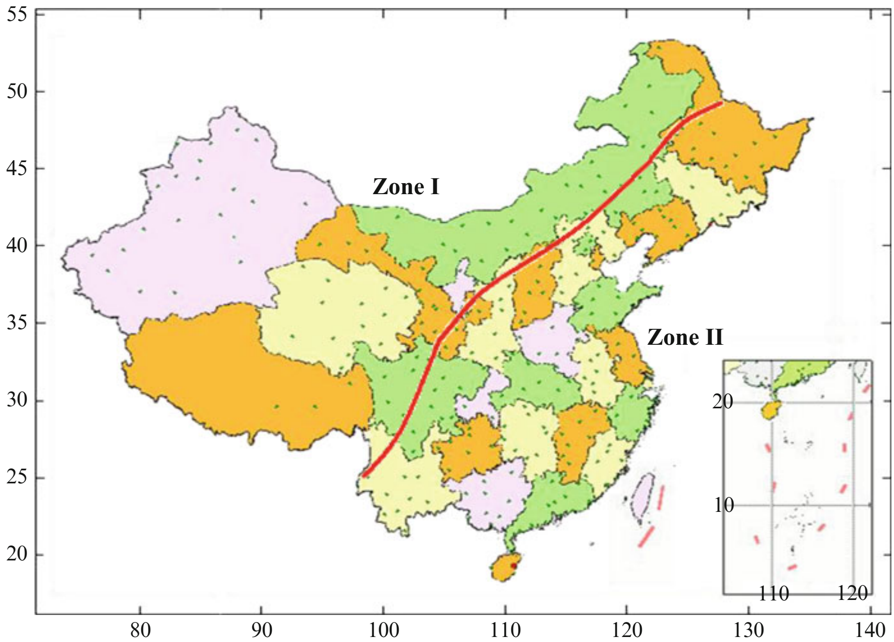


Fig. 4.1 Main climate regions of China

4.1.1.1 Dry Region in the West (Zone I)

In the northwest of China, the outdoor air humidity ratio is low enough in summer that only sensible cooling is required to handle the outdoor air before it is supplied to the indoor environment. The typical annual outdoor air parameters of Urumchi are listed in Fig. 4.2, and it can be seen that the outdoor air humidity ratio is low throughout the year. Because of the dry outdoor air humidity ratio, the evaporative cooling method can be adopted to cool the outdoor air in summer. Section 4.2 will introduce the different evaporative cooling solutions and devices in detail. In this region, the outdoor air temperature and humidity ratio are both quite low in winter, so there are both heating and humidification requirements in the outdoor air handling process.

4.1.1.2 Humid Region in the East (Zone II)

In the THIC system, dry air is supplied to the indoor environment to extract the indoor moisture load. The outdoor climate is hot and humid during summer in the southeast of China. The typical annual outdoor climate parameters of Beijing and Shanghai are shown in Figs. 4.3 and 4.4, respectively. The mean outdoor humidity ratios in July and August approach 17 g/kg for Beijing and 19 g/kg for Shanghai. Thus, the outdoor air has to be dehumidified before being supplied to the indoor environment.

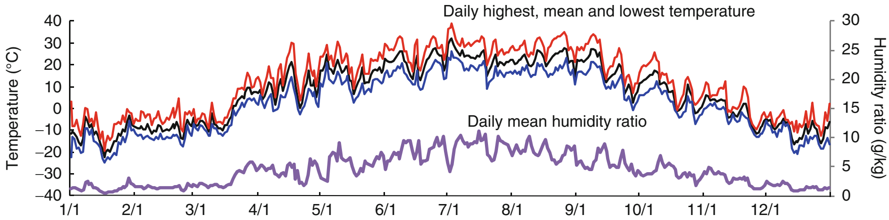


Fig. 4.2 Annual outdoor climate of Urumchi

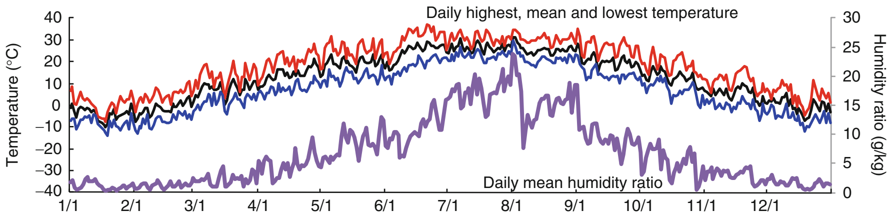


Fig. 4.3 Annual outdoor climate of Beijing

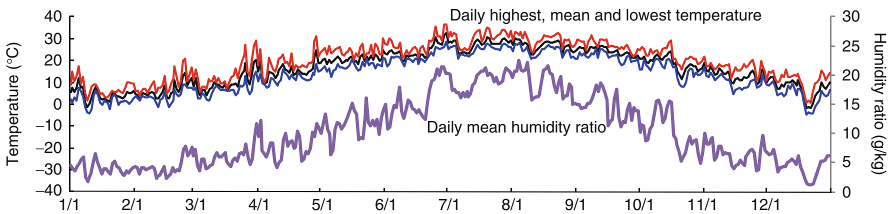


Fig. 4.4 Annual outdoor climate of Shanghai

According to the analysis of the requirements for outdoor air handling processes from Sect. 2.4.2 in Chap. 2, as well as the performance requirements for outdoor air handling devices in different climate regions of China discussed above, various kinds of outdoor air handling solutions and devices can satisfy these requirements. The following subsection will introduce the basic handling devices in the outdoor air handling process.

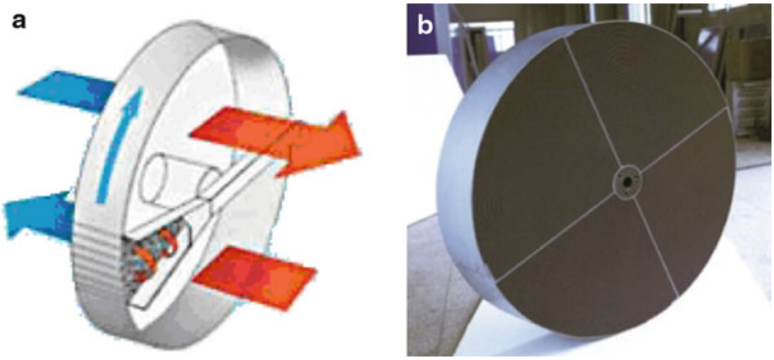


Fig. 4.5 Desiccant wheel used for heat recovery: (**a**) schematic diagram and (**b**) rotary wheel

4.1.2 Heat Recovery Devices

Heat recovery devices are very useful for reducing the energy consumption of the outdoor air handling process, especially in hot and humid climates. Heat recovery devices can be classified into two types: sensible heat recovery devices and enthalpy recovery devices. Both sensible heat and latent heat from the indoor exhaust air can be recovered by enthalpy recovery devices, which demonstrate greater performance than sensible heat recovery devices. There are many different configurations of heat recovery devices, as described in detail in various handbooks (Lu 2008, China Institute of Building Standard Design & Research 2006). In this section, only enthalpy recovery devices using rotary wheels and those using liquid desiccant are examined.

4.1.2.1 Enthalpy Recovery Device Using a Rotary Wheel

The enthalpy recovery device with a rotary wheel has a honeycomb cylinder structure made of paper coated with desiccant material, as shown in Fig. 4.5. Rotation speed is usually around 8–10 rpm (480–600 r/h) for this kind of heat recovery device. When the water vapor pressure and temperature of the solid desiccant on the wheel are lower than that of the outdoor air, the moisture transfer direction and the heat transfer direction are both from the outdoor air to the desiccant, which makes air dehumidification possible. With the rotation of the wheel, the fully adsorbed desiccant rotates to the indoor exhaust air zone and is dried and cooled by the exhaust air. In this process, exhaust air and outdoor air flow through the top and bottom sections of the wheel, respectively, with a counterflow configuration to achieve enthalpy recovery.

4.1.2.2 Enthalpy Recovery Device Using Liquid Desiccant

The heat recovery device using liquid desiccant is also an option for recovering energy from the indoor exhaust air. Packed-bed towers are common handling

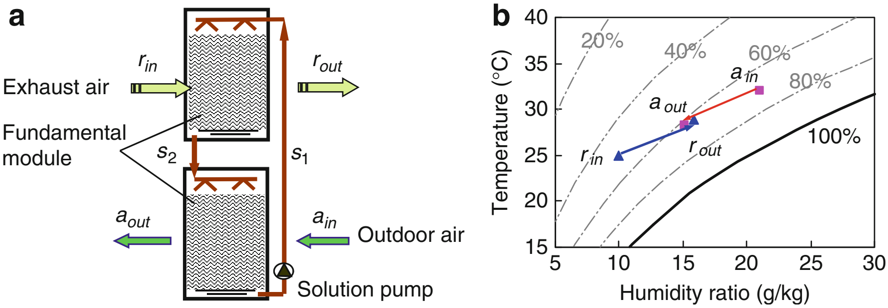


Fig. 4.6 Single-stage enthalpy recovery device using liquid desiccant: (**a**) operating schematic and (**b**) air handling process in psychrometric chart

devices in liquid desiccant systems, and the packing in the tower can increase the contact area between the solution and the air, thereby enhancing the effectiveness of the heat and moisture transfer process. Figure 4.6 shows a typical cross-flow single-stage enthalpy recovery device using liquid desiccant, where the indoor exhaust air is used to preprocess the outdoor air. The device consists of two packed towers and a circulating solution pump: the top module is the indoor exhaust air handling module, and the bottom module is for handling the outdoor air. The outdoor air and the exhaust air are represented by a and r, respectively, and the solution state is represented by s, as shown in Fig. 4.6.

For the operating process in summer, the solution pump transports the liquid from the bottom of the solution tank in the bottom module to the top of the top module, and then the solution is sprayed from the top to wet the packing. The coupled heat and mass transfer process proceeds between the indoor exhaust air and the solution. The exhaust air is heated and humidified by the solution and then flows out of the module. The cooled and concentrated solution mixes together and flows out of the top module. The solution is then sprayed from the top of the bottom module and is evenly distributed to the bottom packing. The outdoor air flows into the bottom module and is cooled and dehumidified, as the solution temperature or vapor pressure in the packing is lower than the air. Then the solution is diluted and flows to the bottom of the bottom module, completing the cycle. In this enthalpy recovery device, the outdoor air is cooled and dehumidified, and the indoor exhaust air is heated and humidified with the help of the solution cycle, achieving energy recovery. The operation in winter is similar to that in summer, with only the direction for energy transfer, which indicates that the outdoor air is heated and humidified, being reversed.

The multistage device is composed of a series of the single-stage devices shown in Fig. 4.6. Figure 4.7 illustrates the operating schematic of a three-stage enthalpy recovery device using liquid desiccant, with the outdoor air and the exhaust air flowing through the modules in reverse order. The temperature and concentration of the solution sprayed in each stage are determined by the parameters of the outdoor air and the exhaust air. The air handling process is shown in Fig. 4.7b.

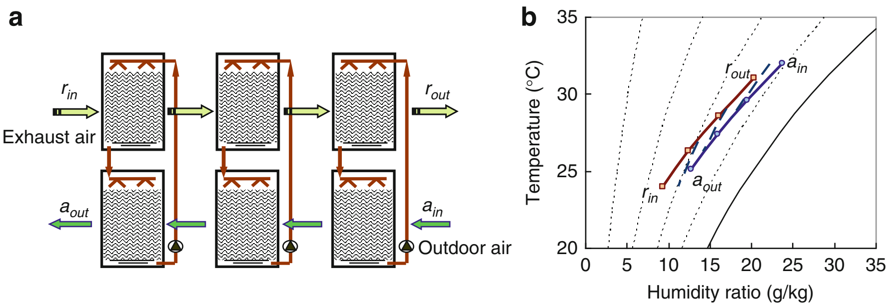


Fig. 4.7 Three-stage enthalpy recovery device using liquid desiccant: (**a**) operating schematic and (**b**) air handling process in psychrometric chart

4.1.3 Dehumidification Devices

4.1.3.1 Condensation Dehumidification Methods

Condensation dehumidification uses chilled water or refrigerant with a temperature low enough to cool the humid air; as the air temperature drops below its dew point temperature, the moisture condenses and the air is dehumidified. The operating schematic of the condensation dehumidification method is demonstrated in Fig. 4.8.

The relatively low-temperature cooling medium flows into the cooling coil, and then the temperature of the humid air flowing through the coil decreases. When the saturation condition is achieved, the humid air is dehumidified, and the moisture continues to be condensed as the temperature continues to decrease. Figure 4.9 shows the reachable handling region (nearly a triangular region) of the outlet air state using the condensation dehumidification method. The temperature and humidity ratio of the processed outlet air are both lower than those of the inlet, and the state of the processed outlet air usually approaches the saturation state.

4.1.3.2 Dehumidification Methods Using Liquid Desiccant

The operating principle of the typical dehumidification-regeneration cycle using liquid desiccant is shown in Fig. 4.10. The left side of this figure is the air dehumidification process, and the right side is the desiccant regeneration process. In the air dehumidification process, moisture transfers from the gas phase (air) to the liquid phase (solution) because the partial pressure of water vapor in the air is greater than that in the solution. With the help of the mass transfer process, the humidity ratio of the moist air is reduced, i.e., the air is dehumidified, and the solution is diluted due to moisture absorption. Thus, the partial pressure of the water vapor in the solution is gradually increased, so that the water vapor pressure difference between the solution and the air is gradually reduced. As a result, the

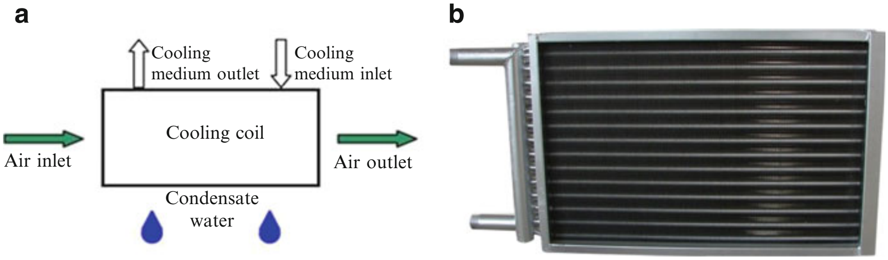
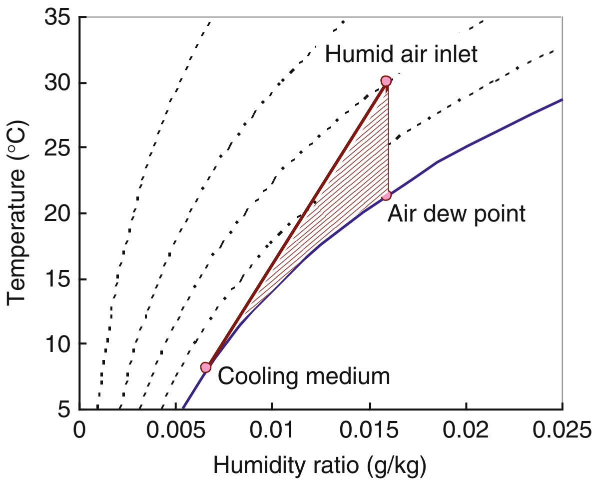


Fig. 4.8 Operating schematic of the condensation dehumidification method: (**a**) operating principle and (**b**) picture of the cooling coil

Fig. 4.9 Reachable air outlet region of the condensation dehumidification method



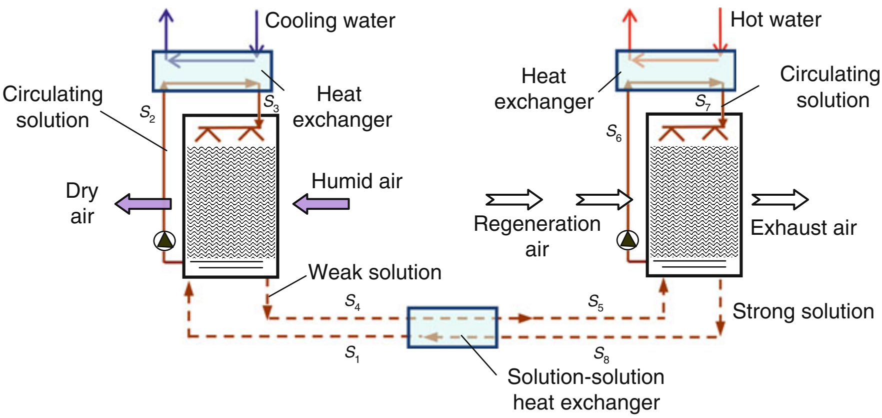


Fig. 4.10 Principle of the typical liquid desiccant dehumidification-regeneration cycle

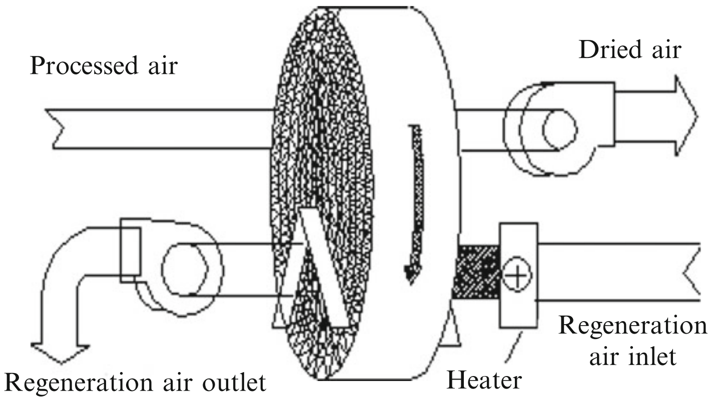


Fig. 4.11 Operating principle of the desiccant wheel used for dehumidification

dehumidification ability of the solution is weakened, and regeneration of the diluted solution is required. As shown in the right part of Fig. 4.10, hot water provides the heat required for the desiccant regeneration process. A solution-to-solution heat exchanger is usually adopted in this air handling system to precool the solution flowing into the dehumidifier and to preheat the solution flowing into the regenerator. Thus, the heat recovery process between solutions of different temperatures is realized, and the energy performance is improved.

4.1.3.3 Dehumidification Methods Using Solid Desiccant

There are two types of dehumidification methods that use solid desiccant: those with rotary wheels and those with fixed desiccant beds. Desiccant wheels are widely used due to their ability to achieve continuous dehumidification and regeneration. The operating principle of a desiccant wheel is shown in Fig. 4.11. Dehumidification wheels are similar to the enthalpy recovery desiccant wheels introduced in Sect. 4.1.2, which have a honeycombed channel coated with desiccant material. Heat and mass transfer processes proceed simultaneously in the channels. In desiccant wheels used for air dehumidification, most of the surface area is used for dehumidification, and the remainder is used for regeneration. An optimal rotation speed is around 0.2–0.5 rpm (12–30 r/h). The differences between enthalpy recovery wheels and dehumidification wheels have been examined in previous studies (Zhang and Niu 2002).

The fixed desiccant bed is another kind of dehumidification device that utilizes solid desiccant. In contrast to the rotation mode of desiccant wheels, desiccant beds realize the shifting between dehumidification and regeneration modes by shifting the air flow direction. Figure 4.12 illustrates the working principle of a fixed descant bed (Suzuki and Oya 1983; Zhang 2005). During the first half of the cycle, the left desiccant bed works in regeneration mode, while the right bed works in dehumidification mode. Humid air enters the right bed, and cooling water enters the right

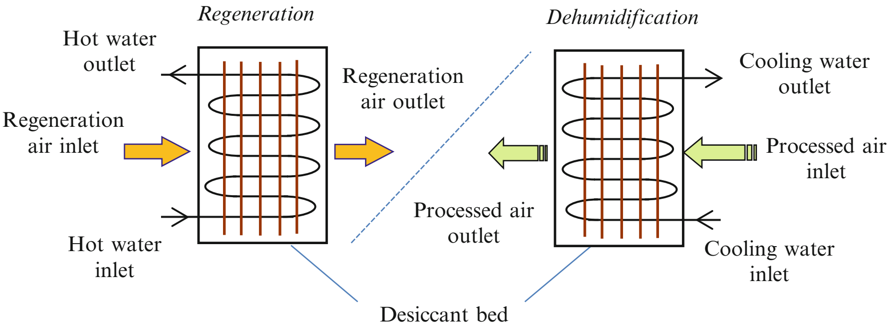


Fig. 4.12 Operating principle of the fixed desiccant bed (the first half of the cycle)

bed to take away the adsorption heat. Regeneration air enters the left bed, and hot water enters the left bed to provide desorption heat. During the second half of the cycle, the left bed works in dehumidification mode (with cooling water entering it), and the right bed works in regeneration mode. During the dehumidification period, the supplied air humidity ratio changes periodically due to the shifting of fan valves and water valves, and this instability of the supplied air parameters limits the use of fixed desiccant beds to some extent.

4.1.4 Humidification Devices

There are multiple methods for air humidification in air-conditioning systems. Common air humidification solutions adopted in air-conditioning systems (The 10th Design and Research Institute of Ministry of Electronics Industry 1995) include wet film evaporative humidification, dry steam humidification, electrode-type humidification, and ultrasonic humidification. The operating schematic of the wet film evaporative humidification method is shown in Fig. 4.13. The water distributed from the sprinkler is sprayed on the wet film. The dry air flows through the wet film, and the sprayed water turns into steam due to evaporation. The moisture then flows into the air, and the dry air is humidified. The humidification process of this type of wet film evaporative humidification method is similar to evaporative cooling. In the operating period during winter, the inlet dry air is usually preheated before flowing through the wet film to ensure the required humidification capacity.

The operating schematic of the dry steam humidification process is shown in Fig. 4.14. The saturated steam flows into the evaporation chamber through the bend. A baffle plate ensures the smooth flow of the steam flowing into the evaporation chamber. The saturated steam flowing out of the evaporation chamber then flows into the drying chamber and extracts all the water to ensure that the dry saturated steam flows out. A control valve is adopted to regulate the flow rate of the dry saturated steam flowing into the effuser. Then the dry saturated steam erupts from the effuser and diffuses uniformly along the width of the air duct, realizing the humidification of the dry air.

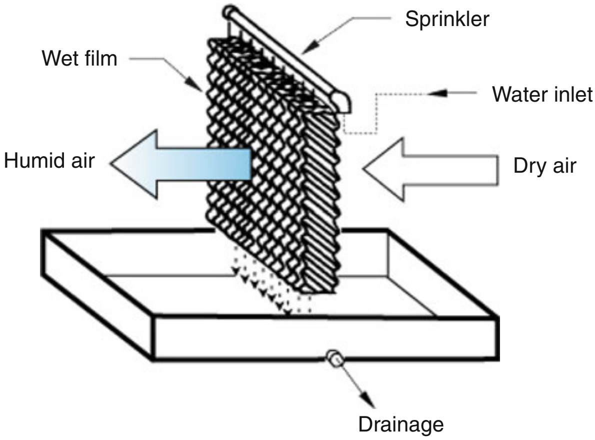


Fig. 4.13 Operating schematic of the wet film humidifier

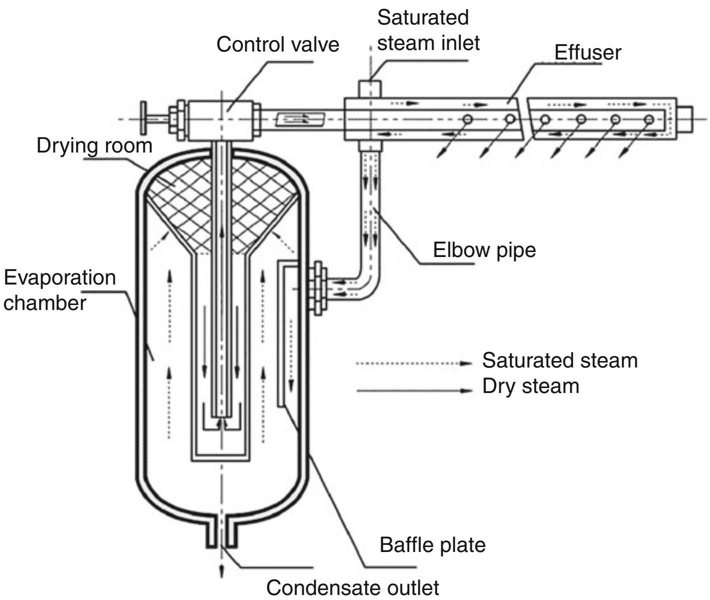


Fig. 4.14 Operating schematic of the dry steam humidifier

In the electrode-type humidification method, an electrode is immersed in the water, and the water is treated as the electrical resistance. The electrical voltage heats the water, and steam is produced, which is then distributed to the area where humidification is required.

However, the application conditions for the various humidification methods are different. If there is a steam source, then the dry steam humidification process should be given priority; this dehumidification method is often utilized in the operating rooms of hospitals. If no steam source can be adopted directly, the wet film evaporative humidification method or the electrode-type humidification method could be utilized.

4.2 Outdoor Air Handling Process in the Dry Region

For most parts of northwestern China, the humidity ratio of the outdoor air is seldom higher than 12 g/kg during summer. Thus, the dry outdoor air could be supplied to the indoor environment to remove indoor moisture. Therefore, the major task of handling the outdoor air is to cool it to a state with a suitable temperature. According to the outdoor air parameters, the air handling devices using direct or indirect evaporative cooling methods could be applied to handle the outdoor air to a state with a temperature of about 18–21 °C and a humidity ratio of about 8–10 g/kg. When the temperature of the handled outdoor air is a bit lower than the indoor temperature, the supplied outdoor air could also extract part of the sensible load, as well as the indoor moisture load. However, the supply air parameters for evaporative cooling rely on the local dry-bulb and wet-bulb temperatures. In designing an evaporative cooling outdoor air handling processor, the stage number and the handling process must both be determined to ensure that the outdoor air can effectively extract the indoor moisture load.

4.2.1 Outdoor Air Handling Process Using Evaporative Cooling

Evaporative cooling devices include direct evaporative cooling devices, indirect evaporative cooling devices, and devices combining direct and indirect evaporative cooling. The operating schematics and corresponding air handling processes for direct and indirect evaporative cooling devices are shown in Figs. 4.15 and 4.16, respectively. For the direct evaporative cooling process, the lowest temperature of the outlet air is equal to the wet-bulb temperature of the inlet air in theory. For the indirect evaporative cooling process, the lowest temperature of the outlet air is equal to the dew point temperature of the inlet air in theory (Huang 2010). In the indirect evaporative cooling process, only the temperature of the handled air decreases, while the humidity ratio stays constant, achieving a cooling process with a constant humidity ratio. The indirect evaporative cooling device shown in Fig. 4.16 is composed of a direct evaporative cooling device and a sensible heat exchanger. The indirect evaporative cooling device could also be constructed by combining a sensible heat exchange component inside a direct evaporative cooling device, which would approach the functionality of an internally cooled indirect evaporative cooling device.

A multistage evaporative cooling process can be constructed based on the direct and indirect evaporative cooling devices introduced above. Figure 4.17 shows a

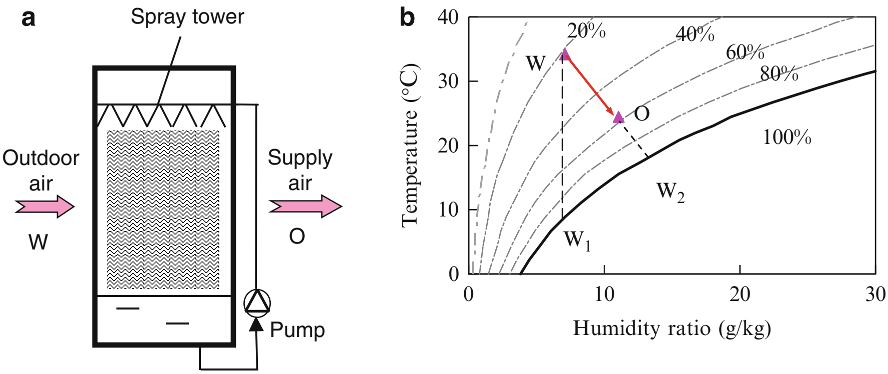


Fig. 4.15 Direct evaporative cooling device: (**a**) schematic diagram and (**b**) air handling process in psychrometric chart

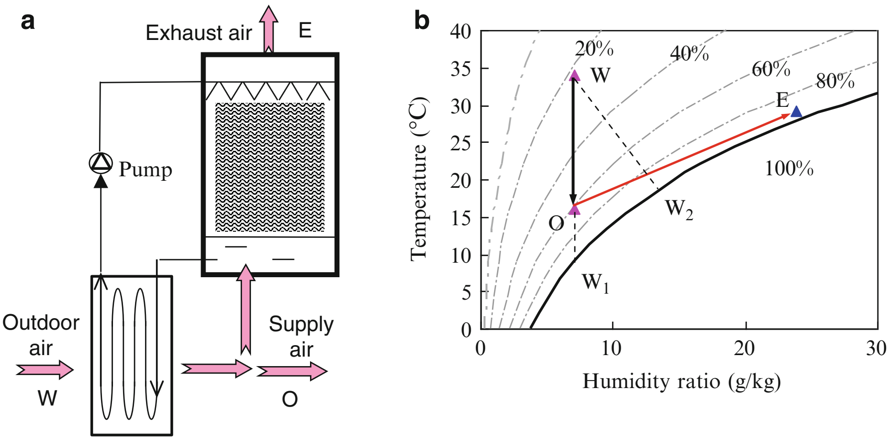


Fig. 4.16 Indirect evaporative cooling device: (**a**) schematic diagram and (**b**) air handling process in psychrometric chart

two-stage evaporative cooling device that includes two indirect evaporative cooling components (Xie and Jiang 2010). Figure 4.18 shows a two-stage air handling process that combines the direct and indirect evaporative cooling methods. In the indirect evaporative cooling process, the limiting temperature of the inlet air to be cooled is determined by the source of the secondary air (i.e., the air participating in the direct evaporative cooling process). If the secondary air is a part of the outlet primary air, as shown in Fig. 4.17, the limiting temperature of the outlet air for this indirect evaporative cooling process is the dew point of the inlet air. In the air handling process shown in Fig. 4.18, the outdoor air is first cooled at a constant humidity ratio by the indirect evaporative cooling method and then cooled with humidification in the direct evaporative cooling module. In practice, the limiting

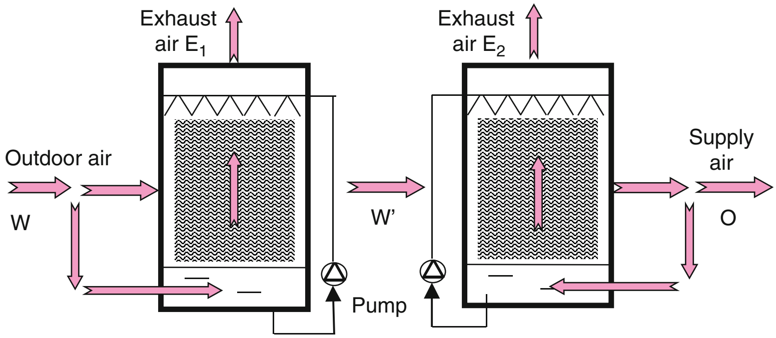


Fig. 4.17 A two-stage indirect evaporative cooling device

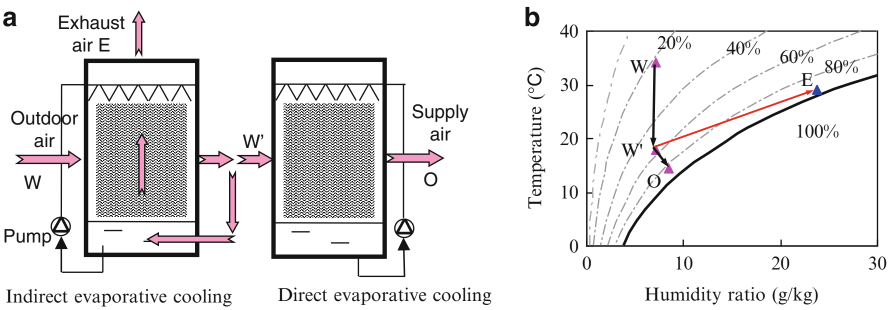


Fig. 4.18 Air handling process combining indirect and direct evaporative cooling: (**a**) operating principle and (**b**) air handling process in psychrometric chart

temperature of the air to be cooled is lower than the wet-bulb temperature of the outdoor air but higher than the dew point of the outdoor air.

In the handling processes using direct evaporative cooling, indirect evaporative cooling, or a combination of direct and indirect types, the outlet air parameters can be expressed by Eqs. (4.1) and (4.2), where W is the outdoor air state, W_1 is the relative dew point state, W_2 is the relative wet-bulb state, and O is the supplied air state after evaporative cooling (shown in Figs. 4.15, 4.16, 4.17 and 4.18, respectively). The temperature and humidity ratio of the supplied air are expressed as follows (Xie and Jiang 2010):

$$t_O = t_W - \eta_1 \cdot (t_W - t_{W1}) - \frac{r}{c_{p,m}}(d_O - d_W) \tag{4.1}$$

$$d_O = d_W + \eta_2 \cdot (1 - \eta_1) \cdot (d_{W2} - d_W) \tag{4.2}$$

where r is the latent heat of water vaporization, $c_{p,m}$ is the specific heat capacity of the humid air, η_1 is the device efficiency of the indirect evaporative cooling module (taking the outdoor dew point as the limiting temperature), and η_2 is the device efficiency of the direct evaporative cooling module humidifying the air. The definitions of $\eta_1$and η_2 can be expressed as follows:

$$\eta_1 = \frac{t_W - t_1}{t_W - t_{W1}} \tag{4.3}$$

$$\eta_2 = \frac{d_W - d_O}{d_W - d_2} \tag{4.4}$$

where t_1 is the dry-bulb temperature of the outlet air of the indirect evaporative cooling module and d_2 is the humidity ratio of the saturated air, with a temperature equal to the wet-bulb temperature of the inlet air in the direct evaporative cooling module. Thus, the values of the relative coefficients in Eqs. (4.1) and (4.2) can be determined for the three kinds of evaporative cooling devices:

- Direct evaporative cooling method: $\eta_1 = 0$ and $0 < \eta_2 \leq 1$
- Indirect evaporative cooling method: $\eta_2 = 0$ and $0 < \eta_1 < 1$
- Combination of direct and indirect evaporative cooling methods: $0 < \eta_1 < 1$ and $0 < \eta_2 < 1$

For internally cooled or externally cooled indirect evaporative cooling devices, dew point efficiency η_1 expressed by Eq. (4.3) is determined by the ratio between the primary air and the secondary air, the flow rate ratio between the air and the water in the direct evaporative cooling module, and the input heat transfer ability of the indirect evaporative cooling module. For handling processes combining direct and indirect evaporative cooling methods, the higher the efficiency (η_1) of the indirect evaporative cooling module, the lower the humidification efficiency of the air supplied by the device, as indicated in Eq. (4.2). The reason is that, as the efficiency increases for the indirect evaporative cooling module, the outlet air approaches the saturation line, leading to a decrease of the driving force for cooling and humidification in the direct evaporative cooling module.

Of all the current outdoor air handling devices that utilize evaporative cooling methods, the structure of the direct evaporative cooling type is the simplest, and its air humidification efficiency (η_2) can be about 90 % or even higher in some cases. For the indirect evaporative cooling type, the air cooling efficiency (η_1) based on the outdoor dew point temperature is usually 40–70 %, mostly depending on the structures and input parameters. Table 4.1 lists the outlet air parameters of different evaporative cooling devices in typical cities of Xinjiang province, with η_1 calculated as 60 % and η_2 calculated as 90 % to simplify the analysis.

Table 4.1 Outlet air states using evaporative cooling

Area	Atmospheric pressure (kPa)	Outdoor design parameters in summer				Outlet air of DEC device (one stage)		Outlet air of IEC device (one stage)		Outlet air of IDEC device (two stages)	
		t_{db} (°C)	t_{wb} (°C)	t_{dew} (°C)	ω (g/kg)	t (°C)	ω (g/kg)	t (°C)	ω (g/kg)	t (°C)	ω (g/kg)
Altay	93.4	30.6	18.7	12.6	9.9	26.3	14.2	19.8	9.9	18.1	11.6
Karamay	96.9	34.9	19.1	9.4	8.2	29.2	13.9	19.6	8.2	17.3	10.5
Gulja	94.2	32.2	21.4	15.7	12.9	28.2	16.9	22.3	12.9	20.7	14.5
Urumchi	91.8	34.1	18.5	7.5	8.5	28.5	14.1	18.1	8.5	15.9	10.8
Turpan	101.3	40.7	23.8	12.3	11.8	34.5	18.0	23.7	11.8	21.2	14.3
Kumul	93.1	35.8	20.2	11.3	9.9	30.1	15.6	21.1	9.9	18.8	12.2
Kashgar	87.2	33.7	19.9	13.4	11.4	28.6	16.5	21.5	11.4	19.5	13.4
Khotan	86.2	34.3	20.4	13.6	12.2	29.3	17.2	21.9	12.2	19.9	14.2

Notes: t temperature, ω humidity ratio, t_{db}, dry-bulb temperature, t_{wb} wet-bulb temperature, t_{dew}, *dew* point temperature, *DEC* direct evaporative cooling, *IEC* indirect evaporative cooling, *IDEC* combination of direct and indirect evaporative cooling

4.2.2 *Outdoor Air Humidification in Winter*

4.2.2.1 Operating Schematic of the Handling Device

In the dry region, the outdoor air temperature and humidity ratio are both relatively low in winter, indicating that the outdoor air has to be heated and humidified before being supplied to the indoor environment. Figure 4.19a, b illustrates the typical outdoor air handling device and the handling process, respectively, where the cold and dry outdoor air first flows through the heater to be preheated and then flows into the packed tower to be humidified to reach a satisfactory state for the indoor environment.

4.2.2.2 Discussion of the Position of the Heater

For these common air humidification processes, the water or the air is usually heated in the humidification process to achieve a better humidification result. The heater could be used to heat the inlet water or the inlet air. For air humidification processes that use a heater to heat the inlet air, there are some cases in practice when the humidification effect is not satisfactory or is lower than the requirement. What could account for these insufficient humidification results? And does the difference in inlet heating targets have any affect?

Figure 4.20 illustrates two humidification cases in an attempt to answer the above questions. A heater is adopted to heat the inlet air in scenario A and the inlet water in scenario B. Figure 4.21 illustrates the air handling processes that correspond to the two devices described above. The conditions of the inlet air (point a_1)

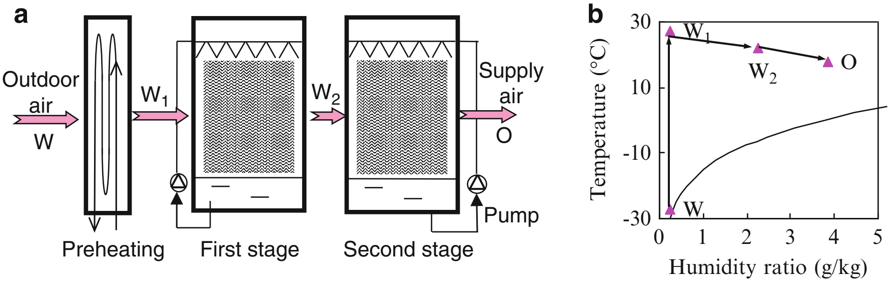


Fig. 4.19 Outdoor air humidification process in the dry region during winter: (**a**) operating principle and (**b**) air handling process in psychrometric chart

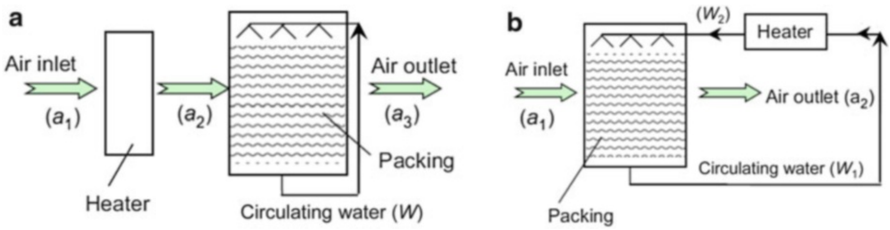


Fig. 4.20 Operating schematics of two air humidification processes: (**a**) heating the inlet air and (**b**) heating the inlet water.

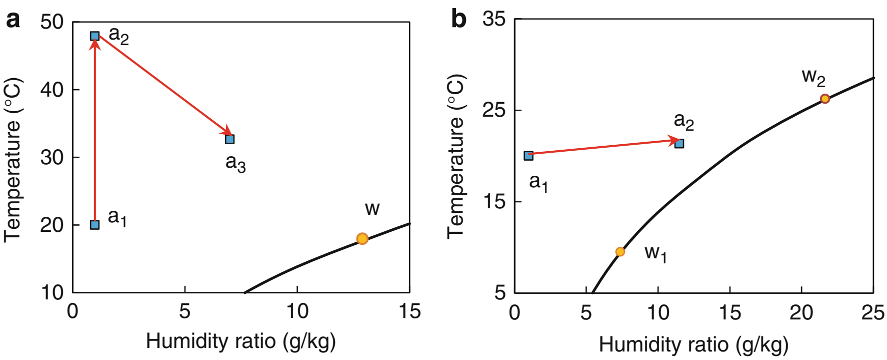


Fig. 4.21 Air handling processes of two different humidification processes in psychrometric chart: (**a**) heating the inlet air and (**b**) heating the outlet air

are 1 kg/s, 20 °C, and 1 g/kg, and the flow rate of the circulating water (point w) is 0.4 kg/s. The heater capacity is 28 kW, and the NTU_m value of the packed-bed humidifier is 1. The humidification performances of the two modes are shown in Fig. 4.21 and Table 4.2.

Table 4.2 Calculated results for different air humidification processes

	Inlet parameters of the packed module			Outlet parameters of the packed module			
Heating object	$t_{a,in}$ (°C)	$\omega_{a,in}$ (g/kg)	$t_{w,in}$ (°C)	$t_{a,out}$ (°C)	$\omega_{a,out}$ (g/kg)	$t_{w,out}$ (°C)	Humidification effect (g/kg)
Air	47.9	1.0	18.0	32.6	7.0	18.0	6.0
Water	20.0	1.0	26.2	21.4	11.5	9.5	10.5

Notes: *t* temperature, *ω* humidity ratio; subscripts: *a* air, *w* water, *in* inlet, *out* outlet

As indicated by the results listed in Table 4.2, when the heater is adopted to heat the inlet air in scenario A, the actual humidification effect of the outlet air is 6.0 g/kg; when the heater is adopted to heat the inlet water in scenario B, the actual humidification effect of the outlet air is as high as 10.5 g/kg. In other words, when inlet water is the heating target, 75 % more humidification is achieved compared to the scenario when inlet air is the heating target. Therefore, for the packed air-water device for humidification shown in Fig. 4.20, the inlet water should be heated, rather than air, in order to gain better humidification performance. Thus, by adjusting the heating target, the humidification effect can be improved significantly in the air-water packed device.

4.3 Outdoor Air Handling Process in the Humid Region

In this region, the outdoor air temperature and humidity ratio are both relatively high in summer, where the outdoor air has to be cooled and dehumidified before being supplied to the indoor environment. Therefore, maximizing the efficiency of the outdoor air handling process is the key issue. In this section, air dehumidification methods that utilize condensation dehumidification and solid desiccant are investigated. Dehumidification methods using liquid desiccant will be discussed in Chap. 5.

4.3.1 Condensation Dehumidification Method

Condensation dehumidification is widely utilized in conventional air-conditioning systems, and it can also be adopted in THIC systems. In conventional air-conditioning systems, chilled water with a temperature of around 7 °C is usually adopted to handle the outdoor air, with the temperature and humidity ratio of the outdoor air both decreasing. Taking a system utilizing an FCU with an outdoor air handling device as an example, the outdoor air is usually handled to a state with a similar humidity ratio as the indoor state before being supplied to the indoor environment. The indoor moisture load (e.g., the moisture generated from occupants) is then extracted by the

FCU operating in the wet condition. In the THIC system, where dry air is responsible for removing the indoor moisture, the outdoor air has to be handled to a state lower than the indoor humidity ratio and dry enough to extract the indoor moisture load. Compared with the conventional system, the required humidity ratio of the supplied outdoor air in the THIC system is lower, leading to a higher performance requirement for outdoor air handling processors.

As the indoor design parameters are 26 °C and 60 % (with a corresponding humidity ratio of 12.6 g/kg), if only the moisture generated from occupants is taken into account, then the humidity ratio of the handled outdoor air in the conventional system is about 12.6 g/kg. For the THIC system, the required humidity ratio of the handled outdoor air is about 9.6 g/kg, as calculated by Eq. (2.20), if the outdoor air flow rate is 30 m^3/h per person. The above parameters of the handled outdoor air in conventional and THIC systems are listed in Table 4.3. Taking the outdoor climate design parameters of Beijing (dry-bulb temperature of 33.2 °C, wet-bulb temperature of 26.4 °C, with a corresponding humidity ratio of 19.1 g/kg) as an example, the required parameters of cooling coils and chilled water flow rate can be determined. The inlet chilled water temperature is 7 °C, and the outdoor air flow rate is 12,000 m^3/h. The required rows of cooling coils and the water flow velocity are listed in Table 4.4, which shows that a 4-row cooling coil is sufficient for the conventional system. Due to its lower required humidity ratio, the THIC system requires more rows of cooling coils than the conventional system, which means a 6-row cooling coil is sufficient.

When the condensation dehumidification method is adopted for the outdoor air handling process, there is a significant temperature difference between the inlet outdoor air and the cooling source. Thus, this low-temperature cooling source leads to an obvious temperature mismatch for the total heat transfer process, resulting in heat transfer loss. To reduce this kind of heat transfer loss caused by the significant temperature difference between the inlet fluids, using an appropriate high-temperature cooling source to precool the outdoor air is a feasible solution, which results in a cascade process and improved energy performance. On the other hand, the handled outdoor air usually approaches the saturated state, and its temperature is too low to be supplied directly to the indoor environment, even though the humidity ratio is satisfactory for humidity control. Thus, reheating is needed to a certain extent. To avoid energy dissipation caused by the reheating process, the indoor exhaust air or the outdoor air itself could be used to reheat the handled air (MOHURD, China Institute of Building Standard Design & Research 2009). The following subsection focuses on common precooling methods and reheating solutions.

4.3.1.1 Precooling the Outdoor Air with Heat Recovery from the Indoor Exhaust Air

If the gas tightness of a building's windows and that of the building envelope are satisfactory, some indoor air should be extracted to maintain the air balance when the outdoor air is supplied to the conditioned space. By setting an appropriate indoor exhaust air system and venting the indoor exhaust air in an organized way,

Table 4.3 Supply air parameters after condensation dehumidification

Type	Dry-bulb temperature (°C)	Relative humidity (%)	Humidity ratio (g/kg)	Wet-bulb temperature (°C)	Specific enthalpy (kJ/kg)
Conventional system	18.5	95	12.6	17.9	50.6
THIC system	14.2	95	9.6	13.8	38.6

Table 4.4 Selected cooling coils for condensation dehumidification

Type	γ	Cooling coil type	Rows	Air velocity (m/s)	Water velocity (m/s)	η_h
Conventional system	2.14	JW20-4	4	1.78	1.01	0.56
THIC system	2.29	JW20-4	6	1.78	1.39	0.72

Notes: γ ratio of the total cooling capacity to the sensible cooling capacity, η_h heat exchange efficiency

the heat recovery device can be implemented between the indoor exhaust air and the outdoor air, where the energy can be recovered. Figure 4.22 illustrates the outdoor air handling process using condensation dehumidification with the enthalpy recovery module. It can be seen that the enthalpy of the outdoor air decreases after the enthalpy recovery process (from state W to W_1). Thus, the enthalpy difference required in the dehumidification process decreases, helping to reduce the energy consumption of the outdoor air handling process.

4.3.1.2 Precooling the Outdoor Air with High-Temperature Chilled Water

To improve the operating performance of the outdoor air handling process, the high-temperature chilled water (about 16–18 °C) of the THIC system could be adopted to precool the air. The lower-temperature chilled water could be used to dehumidify the air further, as shown in Fig. 4.23. The high-temperature chilled water could be directly obtained from natural cooling sources such as underground water and could also be available from the high-temperature water chiller. With the help of the precooling process, the outdoor air could be cooled from the hot and humid state to the saturated state (or approaching the saturated state). The major task of the precooling process is to cool the air but not to dehumidify it, i.e., from state W to state W_1, as shown in Fig. 4.23b. Low-temperature chilled water is then adopted to dehumidify the air from state W_1 to state O, satisfying the humidity ratio requirement of the supplied air. Moreover, using high-temperature chilled water for precooling takes full advantage of the energy efficiency of the high-temperature cooling source.

Based on the variances of the outdoor air parameters and the required supply air parameters, the air handling processor can meet the requirement of the supplied air

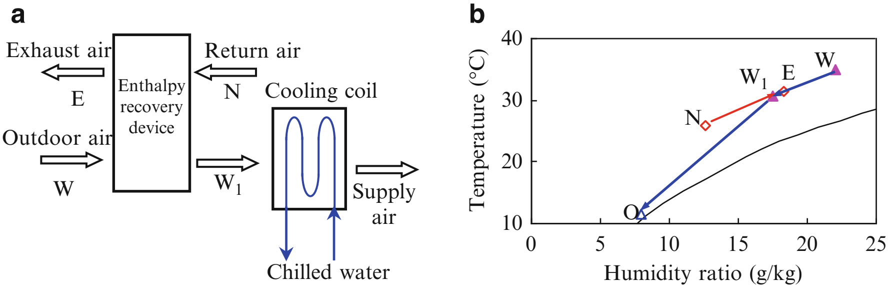


Fig. 4.22 Condensation dehumidification process with the enthalpy recovery module: (**a**) operating principle and (**b**) air handling process in psychrometric chart

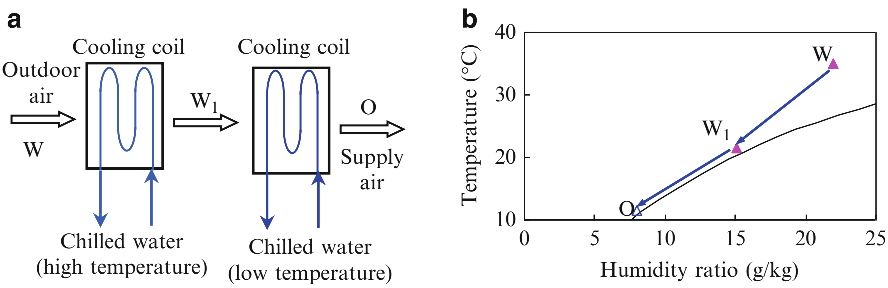


Fig. 4.23 Condensation dehumidification process using high-temperature chilled water for precooling: (**a**) operating principle and (**b**) air handling process in psychrometric chart

by regulating the flow rate of the low-temperature chilled water and ensuring the use of high-temperature chilled water as much as possible. Taking the required humidity ratio of 8 g/kg (the relative temperature is 11.5 °C with a relative humidity of 95 %) as an example, Table 4.5 lists the enthalpy differences using the high-temperature chilled water to precool the air as a function of the outdoor climate, where the high-temperature chilled water is 16 °C. As indicated by the results, the enthalpy difference of the precooling process is significantly higher as the outdoor temperature and humidity ratio are higher. The enthalpy difference of the precooling process accounts for about 50 % of the total enthalpy difference during the outdoor air handling process. Thus, precooling using high-temperature chilled water could undertake the load of handling the outdoor air effectively and is a feasible approach for improving the efficiency of the outdoor air handling process.

For the outdoor air handling process using high-temperature chilled water to precool the air in Fig. 4.23, low-temperature chilled water is adopted for further dehumidification, leading to two plumbing systems for chilled water of different temperatures. To make the air handling processor more flexible, some improvements are proposed for the condensation dehumidification outdoor air handling processes utilizing high-temperature chilled water for precooling. Figure 4.24 illustrates an

Table 4.5 Outdoor air enthalpy differences under typical conditions

Outdoor air parameters					
Temperature (°C)	Humidity ratio (g/kg)	Enthalpy (kJ/kg)	Δh_1 (kJ/kg)	Δh_2 (kJ/kg)	$\Delta h_1/(\Delta h_1 + \Delta h_2)$
35	22	91.6	32.7	27.2	54.7 %
30	22	86.4	29.1	25.6	53.2 %
35	16	76.2	22.0	22.5	49.4 %
30	16	71.0	18.3	21.0	46.7 %

Notes: Δh_1 is the enthalpy difference during precooling, Δh_2 is the enthalpy difference during dehumidification

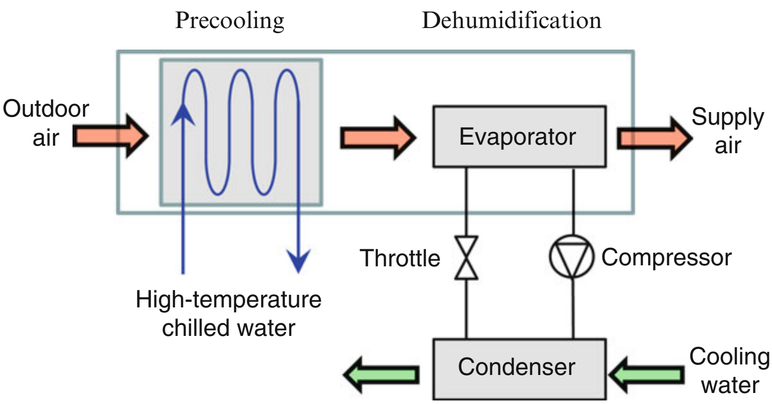


Fig. 4.24 Condensation dehumidification outdoor air handling process with a separate heat pump

improved outdoor air handling process using condensation dehumidification. In the handling process, the outdoor air is first precooled by the high-temperature chilled water (16–18 °C) from the cooling source. The air is dehumidified further by the evaporator of the separate heat pump cycle to meet the humidity ratio requirement. The condenser of the heat pump could be an air-cooled type utilizing indoor exhaust air or a water-cooled type utilizing cooling water. For the outdoor air handling process shown in Fig. 4.24, the refrigerant inside the evaporator evaporates directly, and the air is dehumidified by the heat transfer process between the refrigerant and the moist air. As the separately installed heat pump is responsible for dehumidification, only a single plumbing system for the high-temperature chilled water is required, resulting in a much simpler arrangement of the processors.

For the aforementioned outdoor air handling processes using condensation dehumidification, a common problem is that the supply air temperature is usually too low to be supplied to the indoor environment directly. As the humidity ratio is about 8–10 g/kg, the corresponding air temperature is about 11.5–14.8 °C. If air with such a low temperature is supplied to the conditioned space, the occupants could experience thermal discomfort. Thus, an air diffuser with good inductivity or

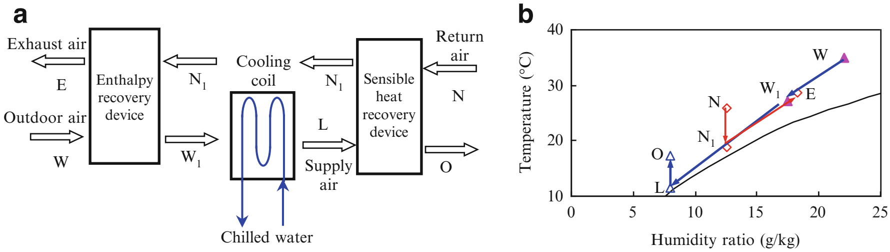


Fig. 4.25 Condensation dehumidification process using return air for precooling and reheating: (**a**) operating schematic and (**b**) air handling process in psychrometric chart

diffusivity is required, and the air distribution should be checked carefully. However, there are some cases when the indoor temperature is too low due to the low supplied air temperature. This is because the building sensible load is related to the outdoor condition, while the indoor moisture load is mostly related to the variance of moisture sources (including the number of occupants). When the variance of occupant number does not fluctuate significantly, indicating that the indoor moisture load is steady, the required humidity ratio for indoor humidity control is therefore steady. If the condensation dehumidification method is adopted, the required supplied air state is then fixed. As a result, during the partial sensible cooling load, the supplied outdoor air temperature can sometimes become too low because of lower outdoor air temperatures or insufficient solar radiation, inevitably resulting in a decrease of the indoor temperature.

If the dehumidified air is directly supplied to the indoor environment, it may lead to overcooling in the partial load. Therefore, the dehumidified air using condensation dehumidification should be reheated to reach an appropriate temperature before being supplied to the indoor environment. Common reheating methods include electrical reheating and steam reheating. However, these methods lead to additional energy dissipation and should be avoided in practice, except for certain special requirements. Instead, reheating the supplied air after dehumidification using the indoor exhaust air or the outdoor air is a more feasible solution, since it achieves the reheating effect while reducing unnecessary energy consumption.

4.3.1.3 Reheating the Handled Air with Indoor Exhaust Air

Figure 4.25 shows the operating schematic of a method for reheating the supply air. The sensible heat recovery process is conducted between the dehumidified outdoor air (L) and the indoor return air (N), realizing the reheating of the outdoor air (from L to O). The return air is cooled to state N_1 by the dehumidified air and then flows through the enthalpy recovery device. The outdoor air is first precooled by the indoor exhaust air before being dehumidified, from state W to state W_1. Thus, in this process, the indoor exhaust air is first used to reheat the dehumidified air. It then absorbs more heat from the enthalpy recovery module and precools the outdoor air.

Table 4.6 Performance of the condensation dehumidification process using exhaust air for precooling and reheating

State W (outdoor air)			State W_1			State O	State N_1	State E		
t (°C)	ω (g/kg)	h (kJ/kg)	t (°C)	ω (g/kg)	h (kJ/kg)	t (°C)	t (°C)	t (°C)	ω (g/kg)	h (kJ/kg)
35.0	22.0	91.6	27.3	17.5	72.0	17.3	18.7	28.6	18.3	75.3
30.0	22.0	86.4	24.6	17.5	69.3	17.3	18.7	25.5	18.3	72.2
35.0	16.0	76.2	27.2	14.4	64.0	17.3	18.7	28.5	14.7	66.1
30.0	16.0	71.0	24.6	14.4	61.3	17.3	18.7	25.5	14.7	63.0

Notes: t temperature, ω humidity ratio, h enthalpy

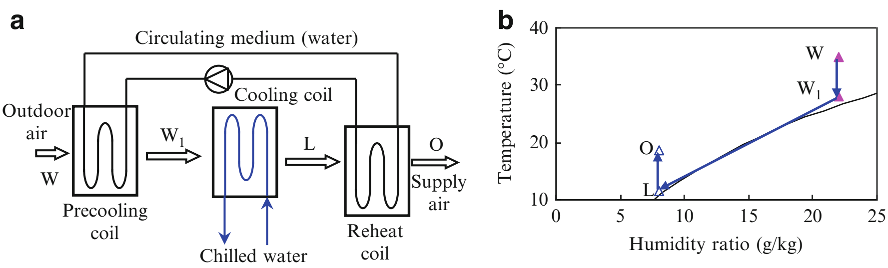


Fig. 4.26 Outdoor air handling process using intermediate for precooling and reheating: (**a**) operating schematic and (**b**) air handling process in psychrometric chart

Table 4.6 lists the operating performances of the sensible heat exchange module and the total heat exchange module of the outdoor air handling device shown in Fig. 4.25 that utilizes condensation dehumidification. The indoor design parameters are 26 °C with a relative humidity of 60 % (the corresponding humidity ratio is 12.6 g/kg). The flow rate ratio between the indoor exhaust air and the outdoor air is 0.8, and the dehumidified air state (L) is 11.5 °C with a relative humidity of 95 % (the corresponding humidity ratio is 8 g/kg). The supply air temperature could be around 17.3 °C using the indoor exhaust air to reheat the dehumidified air, which is suitable for the indoor environment. Based on these indoor parameters, it can be said that this reheating method is superior in relatively stable operating conditions.

4.3.1.4 Reheating the Handled Air with Outdoor Air

As the fluid intermediate can act as the medium for the energy exchange between the outdoor air and the dehumidified air, it could be adopted for precooling and reheating, as shown in Fig. 4.26. In contrast to the condensation dehumidification process shown in Fig. 4.25, this kind of precooling and reheating process uses the intermediate, which circulates in a closed loop. The intermediate's major task is to cool or reheat the air in the handling process undertaking the sensible load.

Table 4.7 Temperatures of the outdoor air handling process using intermediate for precooling and reheating (°C)

State *W* (outdoor air)	State W_1	State *L*	State *O* (supply air)	Intermediate
35.0	27.9	11.5	18.5	23.2
32.0	25.8	11.5	17.6	21.7
29.0	23.7	11.5	16.7	20.2
26.0	21.6	11.5	15.8	18.7

Table 4.7 lists the precooling and reheating process in typical conditions using Freon as the intermediate, where the dehumidified air state (*L*) is 11.5 °C with a relative humidity of 95 % (the corresponding humidity ratio is 8 g/kg). This table shows that the effect of the precooling and reheating process using the circulating fluid is affected by the variance of outdoor air temperature; the supply air temperature can fluctuate after reheating, and the reheating effect cannot be guaranteed. In some outdoor air handling devices using condensation dehumidification, sensible heat exchangers are installed between the outdoor air inlet and the dehumidified air and use the outdoor air itself to reheat the supplied air. However, since the reheating effect is also influenced by the outdoor air temperature, this is not regarded as a solution that can satisfy a steady reheating requirement.

Thus, for this kind of reheating process that uses the intermediate or the outdoor air itself, the influence of the variance of outdoor air parameters on the reheating effect should be taken into account, especially when the outdoor air temperature is low (e.g., during plum rain season in Shanghai), as the supplied air temperature after reheating cannot be guaranteed.

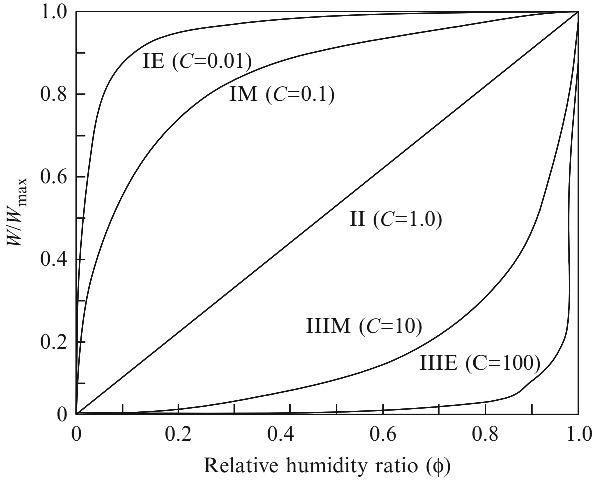
4.3.2 Solid Desiccant Dehumidification Method

There are two kinds of solid desiccant dehumidifiers: the rotary wheel type (continuous dehumidification process) and the fixed bed type (noncontinuous dehumidification process). In this section, a desiccant wheel dehumidifier and the DESICA fixed bed system (developed by Daikin Industries, Ltd.) are introduced.

4.3.2.1 Properties of Commonly Used Solid Desiccant Materials

Solid desiccant material can be used to adsorb water vapor molecules in the air to achieve dehumidification, and the water vapor pressure difference between the air and the desiccant surface is the driving force for this moisture transfer process. As humid air flows through the surface of the solid desiccant material, water vapor is adsorbed and the air is dehumidified. Air states approach the isenthalpic line, with the humidity ratio decreasing and the temperature increasing. Thus, the air temperature after dehumidification by solid desiccant is usually too high to be supplied

Fig. 4.27 Adsorption isotherms of ideal adsorbents



directly to the indoor environment. To realize a lower supplied air temperature, the dried air usually has to be cooled down by associated cooling sources. Furthermore, as the moist air is dehumidified, the water content of the desiccant material increases, and the equilibrium humidity ratio on the surface also increases. Desiccant material loses its adsorption ability when an equilibrium state is reached, at which point it has to be regenerated.

When the desiccant material reaches the equilibrium state, the mass of adsorbed water per unit mass of desiccant material, which is called the absorbed content, is an important index for evaluating the performance of the desiccant material. The water content at the equilibrium state can be described by Zhang et al. (2006):

$$\frac{W}{W_{\max}} = \frac{\phi}{C + (1 - C)\phi} \tag{4.5}$$

where W is water content, kg (water)/kg (desiccant material); $W_{\max}$ is the maximum water content if the moist air is of 100 % relative humidity, kg (water)/kg (desiccant material); C is a shape factor of the desiccant material; and ϕ is the relative humidity of the moist air.

According to different equilibrium water contents under variant relative humidity, adsorption isotherms of the solid desiccant can be obtained. Adsorption isotherms can be divided into different types according to various isotherm characteristics. Type I (including IE and IM), type II, and type III (including IIIE and IIIM) adsorption isotherms have corresponding shape factors of greater than 1, equal to 1, and lower than 1, respectively, as shown in Fig. 4.27. For dehumidification processes using desiccant wheels, desiccant materials with adsorption isotherms of types I and II are usually preferred.

Solid desiccant materials commonly used for dehumidification include silica gel, activated aluminum oxide, molecular sieves, calcium chloride, and zeolite.

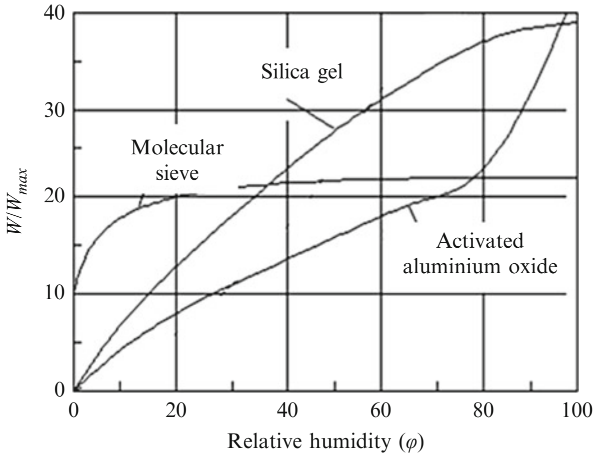


Fig. 4.28 Adsorption isotherms of commonly used adsorbents (Zhang 2005)

Figure 4.28 shows the adsorption isotherms of these desiccant materials, where the molecular sieve is type I, silica gel is type II, and activated aluminum oxide is type III.

Silica gel is a kind of nontoxic, odorless semitransparent crystalline, with a porosity greater than 70 % and W_{max} higher than 30 % (Suzuki and Oya 1983; Jia 2006). Some kinds of silica gel are macroporous, while others are microporous; some change color from blue to pink when adsorbing water because of the addition of cobalt chloride, while others do not. Macroporous silica gels can be saturated easily, while microporous silica gels can adsorb water vapor for a longer period of time. Silica gel is commonly used in air-conditioning systems. Hot air with a temperature between 100 and 150 °C is usually used for the regeneration of silica gel to remove the adsorbed water from the desiccant.

Activated aluminum oxide is highly attracted to water, and under certain circumstances, it can dry air to a dew point lower than −70 °C. Its regeneration temperature is much lower than that of a molecular sieve.

Calcium chloride is a salty, white porous crystal with strong adsorption ability and can deliquesce easily after adsorbing water. Calcium chloride solutions have strong corrosion effects on metal, making the substance inconvenient to use. Nevertheless, because it is cheap and can be easily regenerated, it is a commonly used desiccant material.

Zeolite is an aluminosilicate compound with a strong adsorption ability that is related to the ratio between Si and Al; a smaller ratio helps to increase the adsorption ability (Suzuki and Oya 1983). Synthetic zeolite (a molecular sieve) is of balanced microporosity, and it can be used to produce dehumidification products with different porosity characteristics.

When the solid desiccant and humid air reach equilibrium, their water vapor pressures are identical. In other words, the equivalent humidity ratio of the desiccant is the same as the humidity ratio of the air. The state of the desiccant can be represented in a psychrometric chart according to the equilibrium of the air state with the desiccant. Figure 4.29 depicts the states of commonly used desiccants in a

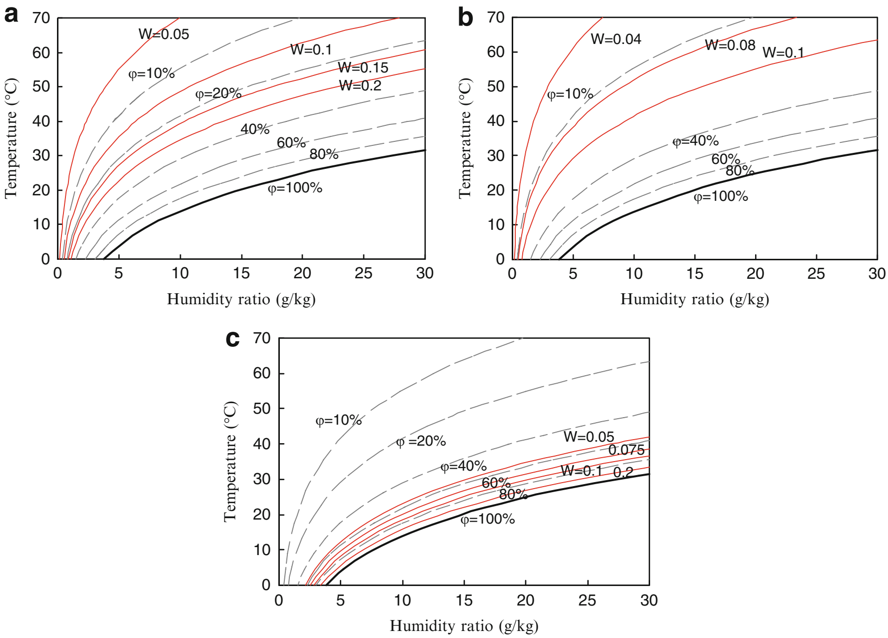


Fig. 4.29 States of commonly used desiccants in psychrometric chart: (**a**) RD-type silica gel, (**b**) molecular sieve, and (**c**) activated aluminum oxide

psychrometric chart. For example, the states of RD-type silica gel (Pesaran and Mills 1987) are shown in Fig. 4.29a, with *W* representing water content (kg water/kg desiccant). As indicated by this figure, the iso-water content lines of the silica gel coincide with the iso-relative humidity lines of the humid air.

It can be observed that, for the iso-water content line, lowering the temperature helps to lower the vapor pressure and strengthen its adsorption ability. In the process of dehumidification, the driving force for mass transfer is the vapor pressure difference between the desiccant and the air. Reducing the desiccant's temperature and water content helps to increase the mass transfer driving force. The regeneration process is similar to the dehumidification process but with an opposite mass transfer direction.

4.3.2.2 Dehumidification Process Using Desiccant Wheel

Figure 4.30 shows a dehumidification system using a desiccant wheel developed by the Swedish company Munters (2004), which uses silica gel as the desiccant material. As the processed air flow rate is 790 m^3/h, to achieve a dehumidification capacity of 3.7 kg/h, the regeneration air temperature needs to be around 130 °C with a flow rate of 220 m^3/h; the power consumption of the heater is 3.9 kW. Defining dehumidification efficiency as the latent heat difference of the processed air divided by the heater's power,

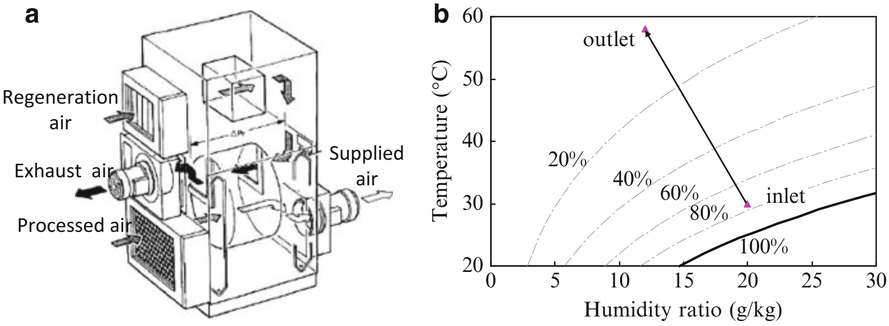


Fig. 4.30 Desiccant wheel developed by Munters, series SX: (**a**) schematic diagram and (**b**) air handling process in psychrometric chart

the dehumidification efficiency of this wheel is about 0.7 ((3.7/3,600 × 2,500)/3.9 = 0.7). Figure 4.30b shows the air handling process if the processed air inlet state is 30 °C, 20 g/kg, and that of the supplied air is 58 °C, 12 g/kg, with a temperature increase of 28 °C.

Because the processed air is handled approaching the isenthalpic line, the supplied air temperature is usually very high, so a cooling system should be implemented to cool down the air. Heat recovery from the indoor exhaust air is a common way to cool the processed air. However, the cooling ability is limited since the supplied air temperature cannot be significantly lower than the indoor air temperature. Thus, other cooling sources such as high-temperature cooling water are preferable options to cool the processed air. Figure 4.31 illustrates a rotary wheel dehumidification system, which uses external cooling coils to precool the processed air and cool down the supplied air.

There are many air handling processes that use desiccant wheels, such as the ventilation cycle desiccant dehumidification process, the Dunkle cycle, and the recirculation cycle desiccant dehumidification process (La et al. 2010). Figure 4.32 shows the air handling process of the ventilation cycle. Seen from the psychrometric chart, there are four approximate isenthalpic processes, including the desiccant dehumidification process (f–g), the desiccant regeneration process (d–e), and the direct cooling processes (a–b and h–i).

As analyzed above, in the dehumidification process using the desiccant wheel, the processed air is handled approaching the isenthalpic line, which requires auxiliary heat recovery and direct cooling components. If the task is to dehumidify the processed air to a state with a humidity ratio lower than that of the indoor air, a regeneration temperature higher than 100 °C is usually required. To reduce the required regeneration temperature, it is advised to split one wheel into two (Ge et al. 2008), as depicted in Fig. 4.33.

Figure 4.34 compares the minimum regeneration temperature of the single-stage desiccant wheel system to that of the two-stage desiccant wheel system. The processed air (W) is 33.2 °C, 19.1 g/kg (Beijing design condition); the required supplied air humidity ratio is 8 g/kg; the dehumidification process is assumed to be along the

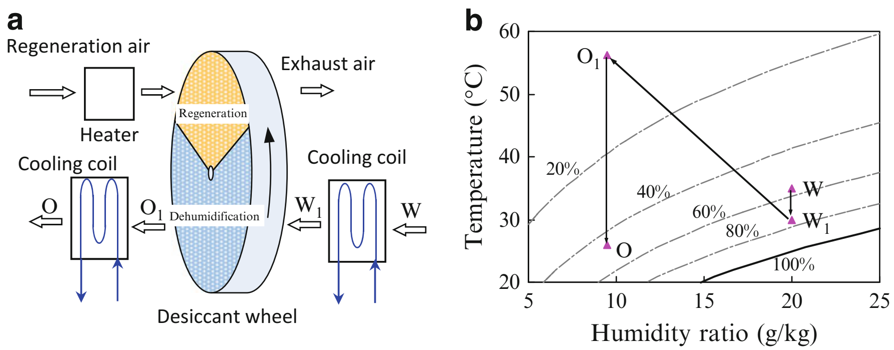


Fig. 4.31 Desiccant wheel with cooling devices: (**a**) schematic diagram and (**b**) air handling process in psychrometric chart

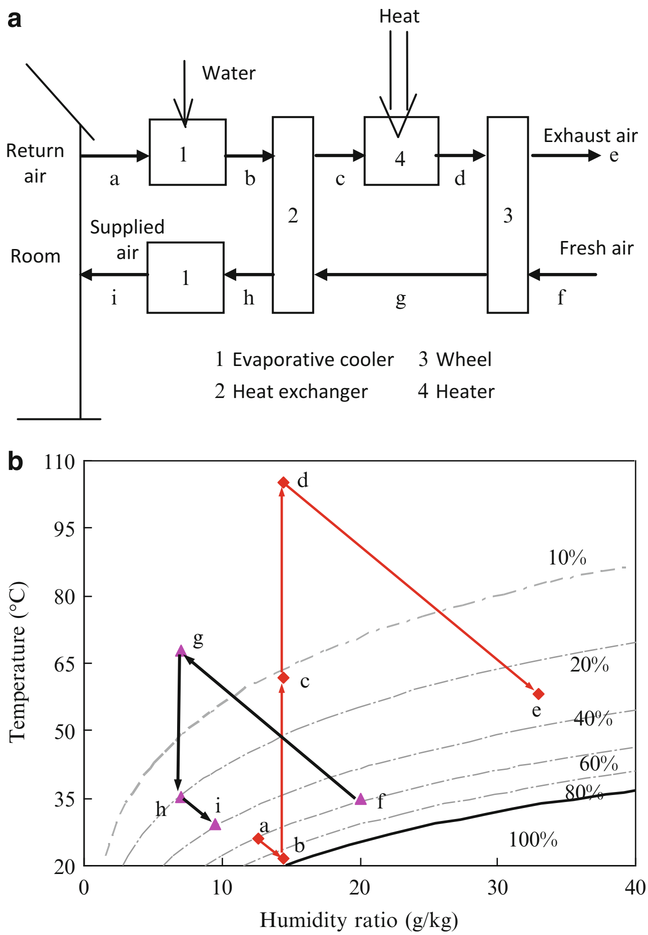


Fig. 4.32 Ventilation cycle solid desiccant dehumidification process: (**a**) schematic diagram and (**b**) air handling process in psychrometric chart

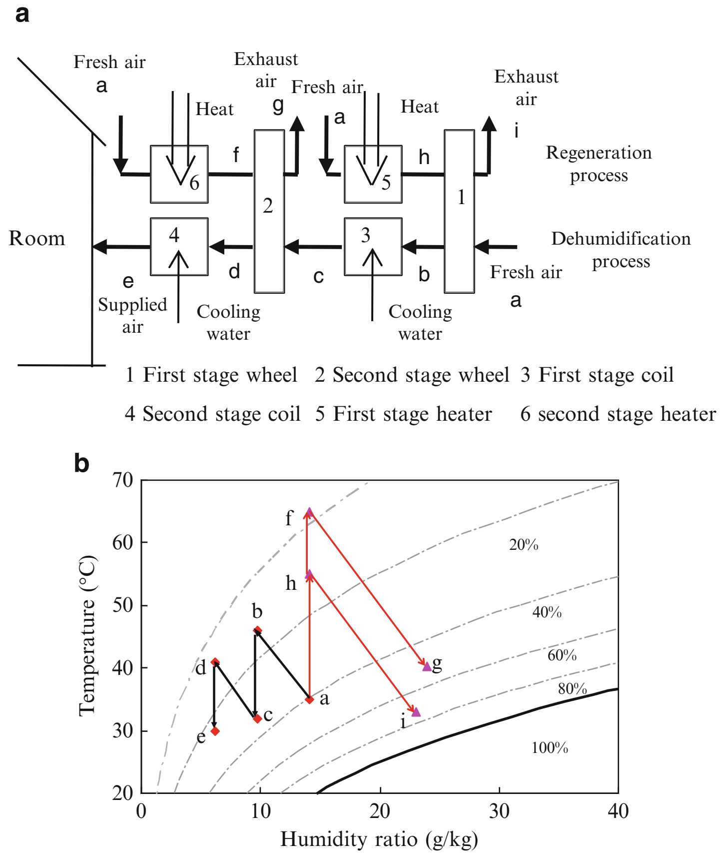


Fig. 4.33 Two-stage desiccant wheel system (La et al. 2011): (**a**) schematic diagram and (**b**) air handling process in psychrometric chart

isenthalpic line; and the outdoor air is used for regeneration. In the single-stage desiccant wheel system, the processed air is handled from point *W* to point *O* (61 °C, 8 g/kg, relative humidity of 6 %). The minimum regeneration temperature is the temperature at intersection point *M* obtained by the iso-relative humidity lines *O* and *W*. The minimum regeneration temperature is obtained under the following assumptions: (1) the temperature and humidity ratio differences between the air and the desiccant material are ignored; (2) the variance of the moisture content for the hygroscopic material inside the desiccant wheel is ignored, and a constant adsorbing capacity is maintained in the process, i.e., the state of the solid desiccant remains at the iso-relative humidity line corresponding to *O*; (3) the driving force of the mass transfer between the desiccant and the regeneration air required in the regeneration process is ignored, as an identical water vapor pressure for the desiccant and the regeneration air is sufficient to realize the desiccant regeneration process; and (4) the driving force of the heat transfer between the desiccant and the regeneration air required in the regeneration process is ignored,

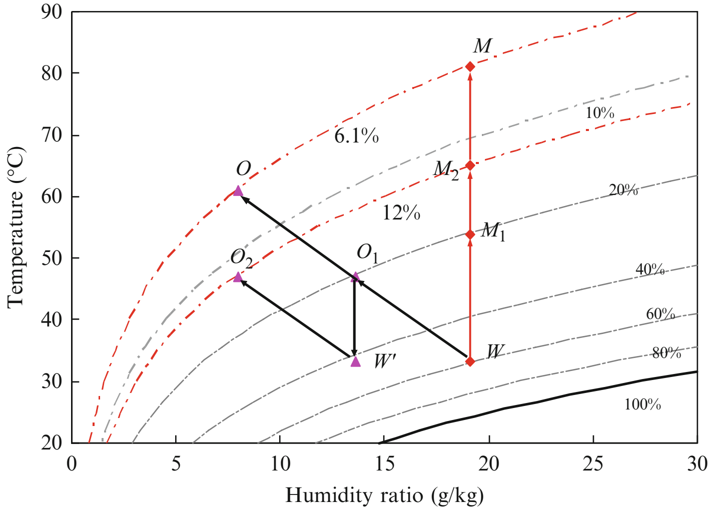


Fig. 4.34 Minimum regeneration temperatures of a single-stage desiccant wheel and a two-stage desiccant wheel

as the solid desiccant is heated to a state with the same temperature as the regeneration air (point M). Under these assumptions, the temperature at point M is the theoretical minimum regeneration temperature, which is significantly lower than the actual required regeneration temperature.

The same method can be applied to the two-stage desiccant wheel system to analyze its minimum regeneration temperature. Assuming that each wheel is responsible for half of the dehumidification task, the processed air state after the first wheel is O_1 (47 °C, 13.6 g/kg, relative humidity of 20 %); it is cooled to W' and is then handled by the second desiccant wheel to O_2 (47 °C, 8 g/kg, relative humidity of 12 %). The minimum regeneration temperature of the first wheel is 54 °C (relative humidity of 20 %), and the minimum regeneration temperature of the second is 65 °C (relative humidity of 12 %).

Table 4.8 compares these two systems, and it can be seen that when the wheel is divided into two wheels, the regeneration temperature can be reduced considerably. Under the same operating condition, the minimum regeneration temperature of the single-stage desiccant wheel system is 81 °C, while those of the two-stage desiccant wheel system are 65 and 54 °C. This can be explained by the fact that, in the two-stage desiccant wheel system, the desiccant maintains high relative humidity, which means a lower temperature can meet the regeneration requirement under the same humidity ratio.

4.3.2.3 Fixed Desiccant Bed

Daikin Industries, Ltd. developed an outdoor air handling processor called DESICA (Xi 2010), which combines solid desiccant materials with a vapor compression

Table 4.8 Minimum regeneration temperatures of a single-stage desiccant wheel system and a two-stage desiccant wheel system

	Air handling process	Typical parameters	Relative humidity of air leaving wheels	Minimum regeneration temperature, °C
Single stage	$W \rightarrow O$	W: 33.2 °C, 19.1 g/kg O: 61.0 °C, 8.0 g/kg M: 81.1 °C, 19.1 g/kg	6.1 %	81.1
Two stage	First stage $W \rightarrow O_1$ Second stage $W' \rightarrow O_2$	W: 33.2 °C, 19.1 g/kg W': 33.2 °C, 13.6 g/kg O_1: 47.0 °C, 13.6 g/kg O_2: 47.1 °C, 8.0 g/kg M_1: 53.8 °C, 19.1 g/kg M_2: 65.1 °C, 19.1 g/kg	First stage 20.3 % Second stage 12.0 %	First stage 53.8 Second stage 65.1

refrigeration system, as shown in Fig. 4.35. In contrast to the desiccant wheel, in which the air handling process approaches the isenthalpic line, the desiccant material is coated on the surface of both the evaporator and condenser in DESICA. As a result, the cooling and heating capacity can be transferred effectively to the desiccant material, and the air handling process is directly cooled (regeneration air is heated) during the dehumidification (regeneration) process.

- *Dehumidification process*: Outdoor air makes contact with the desiccant coated on the evaporator. Because the equilibrium humidity ratio on the surface of the desiccant is lower than that of the processed air, water vapor is transferred from the air to the desiccant. Adsorption heat of the dehumidification process, as well as sensible heat of the processed air, is taken away by the evaporator. The processed air is dried and cooled at the same time.
- *Regeneration process*: Indoor exhaust air makes contact with the desiccant coated on the condenser. As the equilibrium humidity ratio on the surface of the desiccant is higher than that of the regeneration air, water vapor is transferred from the desiccant to the air. The condenser provides desorption heat and heats the air. Desiccant regeneration is realized with a relatively low temperature compared to the desiccant wheel.

Air flow direction and refrigerant flow direction have to be changed at the same time to realize the shift between dehumidification and regeneration modes, as shown in Fig. 4.36. In the dehumidification process, when the desiccant fully adsorbs the water and has to be regenerated, the four-way valve of the refrigeration cycle changes direction to shift the functions of the two heat exchangers (i.e., the

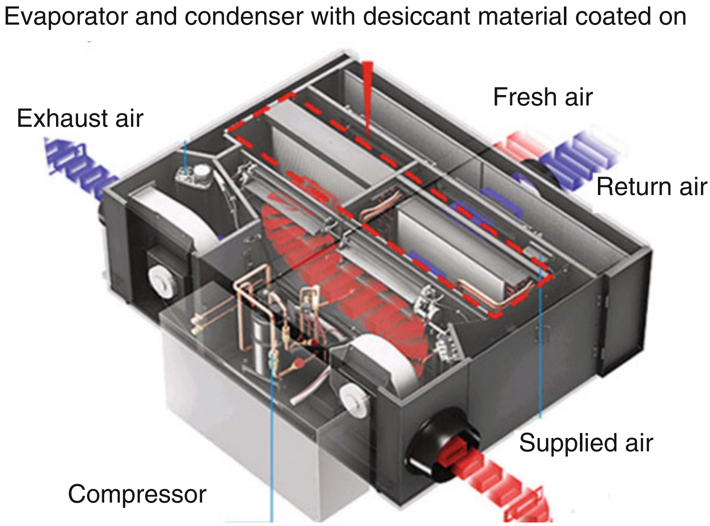


Fig. 4.35 Schematic diagram of DESICA

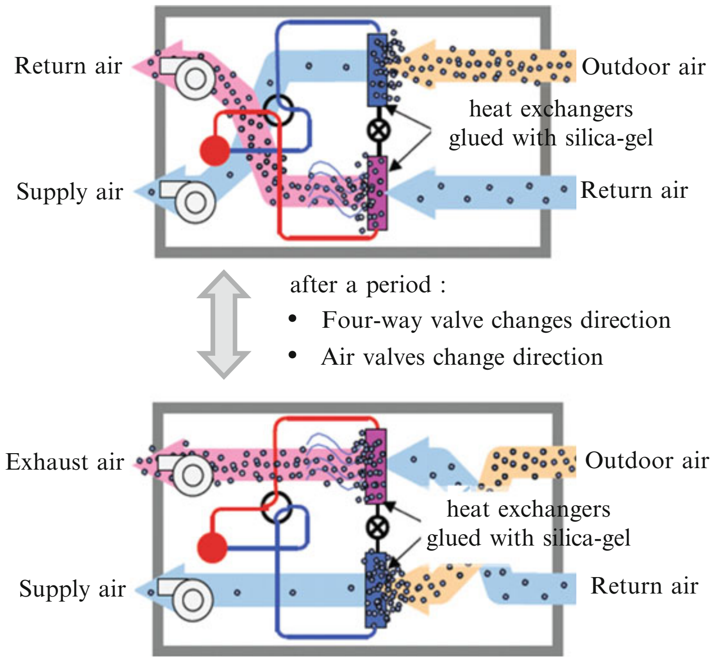


Fig. 4.36 Working principle of DESICA

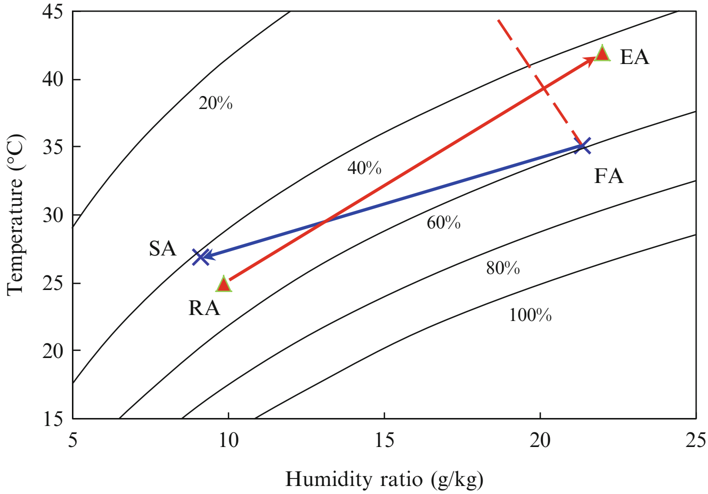


Fig. 4.37 Typical air handling process of DESICA

Table 4.9 Air states of DESICA

Point	Temperature (°C)	Humidity ratio (g/kg)	Relative humidity (%)	Enthalpy (kJ/kg)
FA	35.0	21.4	59.9	90.1
RA	25.0	9.9	49.9	50.2
SA	26.8	9.1	41.5	50.3
EA	41.9	22.0	42.4	98.9
SA_0	65.8	9.1	5.7	90.1

evaporator and condenser). At the same time, fan valves have to change the direction to send the indoor exhaust air to the new condenser, where the desiccant material is to be regenerated. If the regeneration process is complete, a four-way valve and fan valves change the direction back to continue the dehumidification process. Every 3–5 min, the shift occurs to realize the change between dehumidification and regeneration. However, the change of refrigerant direction leads to heating-cooling offset loss.

Figure 4.37 and Table 4.9 present the typical air handling process of DESICA. It can be seen that FA (relative humidity is around 60 %) reaches SA (relative humidity is around 40 %) after the processed air is dehumidified. The air handling process approaches the iso-relative humidity line, as opposed to the rotary wheel dehumidification process, which approaches the isenthalpic line. The indoor exhaust air changes from RA to EA to achieve regeneration, which also approaches the iso-relative humidity line. The exhaust air temperature is around 40 °C, which is much lower than the regeneration temperature in rotary wheel dehumidification. If the rotary wheel is used to handle FA, the state of the processed air changes along the dashed line in Fig. 4.37. The air parameters required to reach the same supplied humidity ratio, represented as SA_0, are listed in Table 4.9 (not shown in Fig. 4.37). Moreover, the processed outlet air temperature is higher than 65 °C, which is much higher than the 27 °C of DESICA.

Thus, this kind of solid desiccant outdoor air handling processor effectively combines desiccant material with an evaporator and condenser to realize internally cooled dehumidification and internally heated regeneration. It can be seen that the operating performance of DESICA is greatly improved compared to the desiccant wheel system.

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