Independent Temperature and Humidity Control Air-Conditioning Systems

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Abstract

Air-conditioning systems play an important role in maintaining the indoor built environment. Coupled heat and mass handling is usually applied for the current state-of-the-art air-conditioning systems. With the advance of society, conventional air-conditioning methods have been challenged by the demand for a more comfortable indoor environment and a higher system energy efficiency. Continuing to improve the energy efficiency and reducing the energy consumption of air-conditioning systems in order to provide a suitable and comfortable environment are foundations to the development of new strategies for the indoor built environment. Taking these requirements into account, the THIC (temperature and

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humidity independent control) air-conditioning system is generally considered to be a possible and effective solution. The innovative THIC air-conditioning systems are introduced in this chapter. Theoretical analysis, key components specially developed, and design methodology of the THIC systems are emphasized. Terminal devices of the temperature control subsystem consist of a heat exchanger and a heat transfer fluid, which could be a high-temperature chilled water or refrigerant. Different terminal devices for handling the indoor sensible load, including radiant terminal devices and dry FCUs, could be utilized in the THIC systems. Different air dehumidification methods could be proposed as well as different high-temperature cooling sources. On the basis of key components, a THIC air-conditioning system could be designed and proposed in terms of indoor requirements for temperature and humidity ratio. It is believed that practical and convenient reference content could be provided for researchers, designers, and engineers.

Keywords

THIC (temperature and humidity independent control) · Air-conditioning system · Terminal device · Air handling process · High-temperature cooling source · Energy performance · Building energy saving

Operating Principle of the THIC Air-Conditioning System

The basic tasks of air-conditioning systems are to extract indoor sensible heat, moisture, CO_2, etc., effectively [1, 2]. For these tasks, different solutions are proposed as follows:

- The extraction of sensible heat can be achieved by adopting various approaches; it is not limited to the direct contact method. Radiant heat transfer is applicable for heat extraction, as well as convective heat transfer.
- The extraction of indoor moisture, CO_2, or other gases has to be achieved by supplying air with a lower humidity ratio or concentration. This is a mass transfer process between the indoor air and the supply air.

The required outdoor air flow rate and variation tendency of the task of indoor moisture extraction are consistent with those of the task of indoor CO_2 extraction. The indoor temperature can therefore be regulated by an independent system, and the required temperature of the cooling source increases compared to that in the conventional system.

Temperature and humidity independent control (THIC) air-conditioning system is proposed as an effective solution [3]. Figure 1 illustrates the operating principle of the THIC air-conditioning system, with an outdoor air handling subsystem and a relatively high-temperature cooling source subsystem that can separately regulate the indoor temperature and humidity, respectively. As indoor temperature and humidity are regulated by independent subsystems, the THIC system can satisfy the variance of the indoor

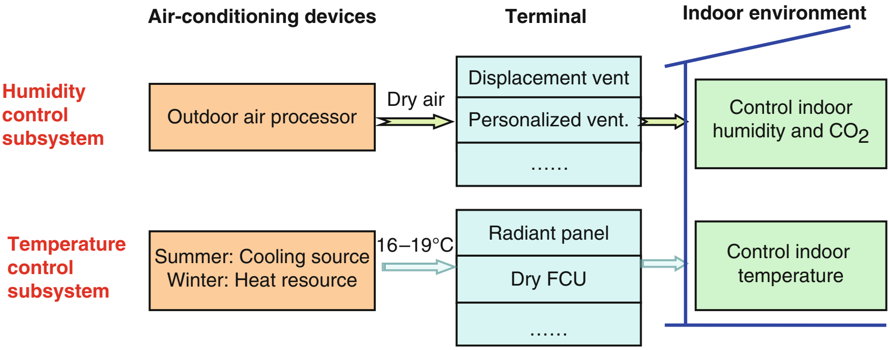

Fig. 1 Operating principle of the THIC air-conditioning system

Table 1 Basic characteristics of the THIC air-conditioning system

	Problem in conventional systems	THIC air-conditioning system
1	Loss in coupled temperature and humidity handling process	Avoids this kind of loss; regulates indoor temperature and humidity separately
2	Offset in cooling and heating, dehumidification and humidification	Avoids offset
3	Insufficiently accommodates variance of heat moisture ratio	Separate subsystems are responsible for temperature and humidity; accurate regulation
4	Indoor terminals	The same terminal is adopted both for cooling and heating; radiant terminals are a feasible solution
5	High energy consumption of transportation	Supplying air is only for moisture extraction; water or refrigerant is recommended as the temperature control medium
6	Influence on indoor air quality	Indoor sensible terminals operate under dry conditions without condensation

heat moisture ratio. Therefore, the THIC system can avoid the imbalance of indoor parameters, and it addresses the conventional system's inability to meet temperature and humidity requirements simultaneously. Table 1 shows a detailed comparison of THIC systems and conventional systems.

In the humidity control subsystem, the handled outdoor air assumes the two main tasks of the air-conditioning system, i.e., supplying enough outdoor air to meet the requirements of indoor air quality and supplying sufficiently dry outdoor air to remove the total moisture load in the building to control the indoor humidity [4]. The humidity control subsystem consists of the outdoor air handling processors and the indoor air supply terminals. As the task can only be completed by supplying air, outdoor air is usually chosen as the medium; however, the required air flow rate will be much lower than in the CAV or VAV systems, where air is used for extracting the entire load.

As for the indoor sensible load, another subsystem with a relatively higher-temperature cooling source can be utilized to control the indoor temperature. To reduce the energy consumption of transportation systems, the minimum air flow rate is recommended because water pumps consume only 10–20% of the energy consumed by fans when transporting the same cooling or heating capacity. Therefore, either water or refrigerant, but not air, is the recommended fluid in the temperature control subsystem. The cooling source temperature needed for temperature control could be increased to about 16–18°C, which is much higher than that of conventional systems. This makes it more convenient to adopt a natural cooling source with the appropriate temperature; if this is not permitted, the mechanical refrigeration method can satisfy the requirement. The energy efficiency of the mechanical cooling source will be improved considerably due to the increased evaporating temperature. Both radiant terminals and common FCUs are feasible for extracting the sensible load. Moreover, there will be no risk of condensation, as a high-temperature cooling source is adopted and the indoor humidity is regulated by the other subsystem.

Figure 2a illustrates the operating schematic of a THIC system. Taking an outdoor air processor and a dry FCU as examples, the air handling processes in this THIC system are illustrated in Fig. 2b. The outdoor air is dehumidified to a sufficiently dry state, and then supplied to the conditioned space to extract the indoor moisture, which is quite different from the conventional system. For the outdoor air handling processor, the high-temperature cooling source, and the indoor terminals, there are different requirements in the THIC system compared to conventional systems.

Indoor Terminals

Terminal devices of the temperature control subsystem include a heat exchanger and high-temperature chilled water or refrigerant that is transported to the end of the heat exchanger and then transfers heat with the indoor air or surfaces through convection or radiation. In the humidity control subsystem, handled outdoor air with a lower humidity ratio is supplied into the indoor space to extract the indoor moisture load. Terminal devices of the temperature control subsystem and humidity control subsystem complement each other, completing the task of constructing the indoor thermal built environment together. Here performance of the radiant terminal as an example is introduced [5–7].

Performances of Different Types of Radiant Panels

Concrete Radiant Floors

The concept of the concrete core follows that of the heating floor, which fixes plastic or stainless steel pipes with concrete reinforcing bars into poured concrete. This structure is widely used in Sweden and has also been used in the Beijing Tiptop International Apartments (Fig. 3). This kind of radiant panel has a mature structure technology and a low cost. On one hand, the large thermal mass of the concrete floor slab allows for adequate thermal storage. On the other hand, the large thermal inertia of the floor heating/cooling system means that the start-up time is long and the dynamic response is slow, leading to difficulties in both regulation and control.

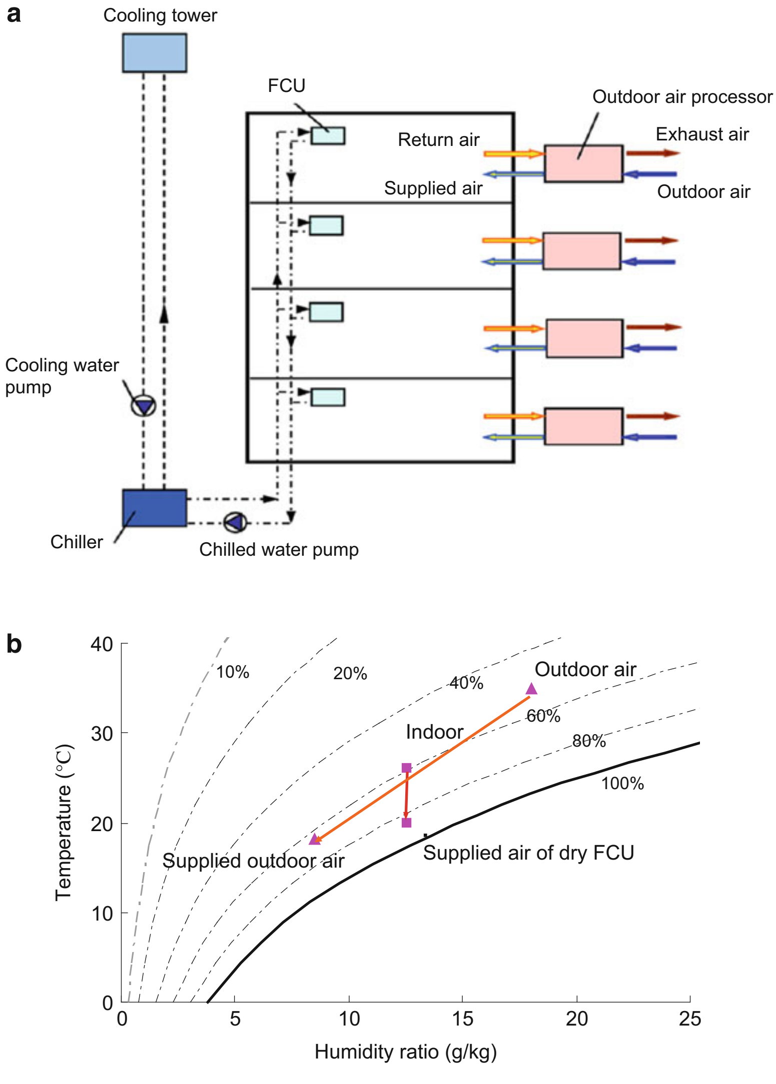

Fig. 2 Typical air handling process of the THIC air-conditioning system: (**a**) schematic diagram of the system and (**b**) air handling process in psychrometric chart

Table 2 lists the thermal resistance and time constant of the concrete radiant floor with typical structures. Thermal resistance of the granite is around 0.1 ($m^2 \cdot °C$)/W, while that of the plastic is around 0.15–0.2 ($m^2 \cdot °C$)/W. The time constant of the radiant panel is around 3–4 h. From the calculated results of the thermal inertia for radiant panels, it can be seen that the time constant of this kind of radiant panel is large: it takes a long period of time to reach a stable cooling/heating effect. Compared to FCUs, concrete radiant floors have greater heat inertia if used for

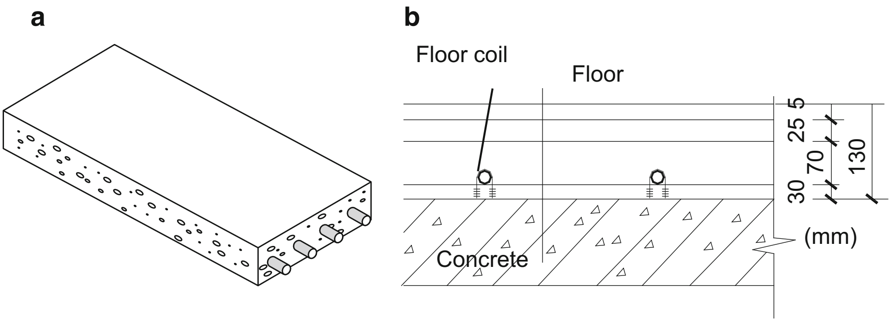

Fig. 3 Typical structure and form of concrete radiant panels: (**a**) schematic diagram and (**b**) concrete radiant floor

Table 2 Thermal resistances and time constants of typical concrete radiant floors

	Configuration (from bottom to top)	Diameter of water pipe, mm	Space between adjacent pipes, mm	R^*, ($m^2 \cdot °C$/W)	Time constant, *h*	
					Floor cooling	Floor heating
Structure I	Pea gravel concrete (70 mm), cement mortar (25 mm), granite (25 mm)	20	150	0.098	3.8	3.2
Structure II	Pea gravel concrete (70 mm), cement mortar (25 mm), granite (25 mm)	20	200	0.116	4.2	3.4
Structure III	Pea gravel concrete (70 mm), cement mortar (25 mm), granite (25 mm)	20	250	0.138	4.6	3.7
Structure VI	Pea gravel concrete (70 mm), cement mortar (25 mm), plastic (3 mm)	25	200	0.160	3.7	2.9
Structure VII	Pea gravel concrete (70 mm), cement mortar (25 mm), plastic (5 mm)	25	200	0.200	4.1	3.1

Notes: The thermal resistance of the radiant panels in the table is the same as R'(R = R' + R"), because R" is much smaller compared to R' (referring to Table 3). The thermal resistance for each material is as follows: pea gravel concrete is 1.84 W/(m·°C); cement mortar is 0.93 W/(m·°C); granite is 3.93 W/(m·°C); PERT plastic pipe is 0.4 W/(m·°C); and plastic floor is 0.05 W/(m·°C)

heating/cooling. For FCUs, heat exchange between the air and the water is realized through forced convection of fans. For radiant panel systems, heat is transferred from the cooling/heating fluid to the panel surface through heat conduction. The thermal inertia problem of radiant floors is significant because of the thickness of the

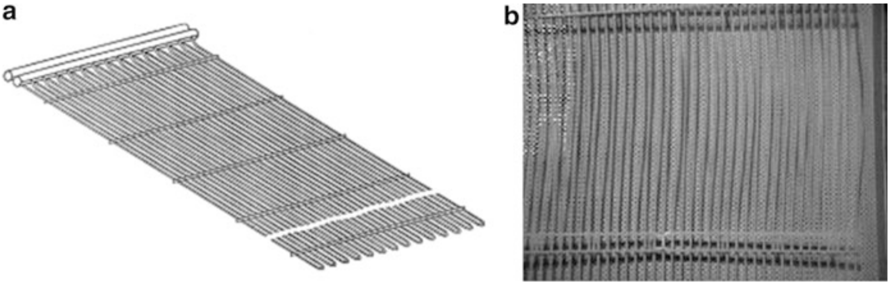

Fig. 4 Capillary radiant panel: (**a**) structure diagram and (**b**) sample

material. If applied in airports and railway stations, where 24-h operation of air-conditioning systems is required, the effects of thermal storage are not obvious. However, these problems should be taken into consideration when applied in buildings with part-time operation.

Capillary Radiant Panels

Capillary radiant panels are made of densely covered plastic pipes with a small diameter (2–3 mm) and a small interval between neighboring pipes (10–20 mm). The pipes connect the distributor pipe on one side and the collector pipe on the other side to form a grid structure, as shown in Fig. 4. Water flow velocity is slow in the pipes; it is around 0.05–0.2 m/s, which is similar to that of blood flow in the capillaries of the human body. This structure can be combined with metal panels or concrete slabs and is thus widely used in practice in retrofitting projects.

When the capillary radiant panel is combined with a metal plate, good contact should be guaranteed in order to achieve a better heat transfer effect. This leads to a high heat transfer resistance, and its cooling capacity is much lower compared to structures without metal plates.

Table 3 lists the thermal resistances of plastered capillary radiant panels with typical structures, where the capillary diameter is 3.35 mm. The thermal resistance and time constant of the capillary radiant panels, which are around 0.02–0.06 ($m^2 \cdot °C$)/W and 5–15 min, respectively, are obviously lower than those of the concrete radiant floor.

Flat Metal Radiant Ceilings

The sandwich structure of the flat metal radiant ceiling mainly consists of metal, such as copper, aluminum, or steel. From the cross-sectional diagram, it can be seen that the pipe is in the middle, while the insulation material and the cover plate are on the top and the backing strap is on the bottom, as shown in Fig. 5a, b. As this type of structure includes decorative features, it is the most widely used structure. The indoor appearance after setup can be seen in Fig. 5c.

Because of the metal components in this structure, the mass is usually large, leading to a higher cost. Moreover, the fin efficiency is usually low due to the influence of the thickness of the panel.

Table 3 Thermal resistance and time constant of plastered capillary radiant panels

	Packing layer material	Packing layer thickness, mm	Space between adjacent pipes, mm	R', $(m^2 \cdot °C)/W$	R'', $(m^2 \cdot °C)/W$	R, $(m^2 \cdot °C)/W$	Time constant, min		
							Ceiling cooling	Vertical heating or cooling	Ceiling heating
Structure I	Parget ($\lambda = 0.45$ W/(m·°C))	20	15	0.046	0.004–0.007	0.050–0.053	11	12	13
Structure II	Parget ($\lambda = 0.87$ W/(m·°C))	20	15	0.025	0.004–0.007	0.029–0.032	7	8	8
Structure III	Parget ($\lambda = 0.45$ W/(m·°C))	20	30	0.063	0.008–0.013	0.071–0.076	13	15	16
Structure IV	Parget ($\lambda = 0.45$ W/(m·°C))	10	15	0.024	0.004–0.007	0.028–0.031	4	4	4
Structure V	Cement mortar ($\lambda = 1.5$ W/(m·°C))	20	15	0.015	0.004–0.007	0.019–0.022	8	9	9

Notes: R'' is related to water velocity inside the pipes, which is taken as 0.05–0.2 m/s; the higher the velocity, the larger the R'' value

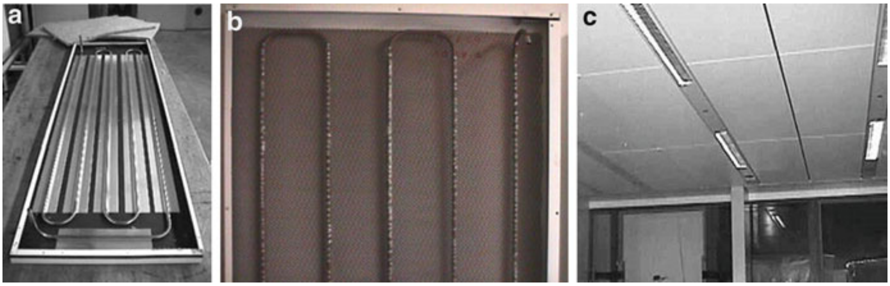

Fig. 5 Flat metal radiant ceiling panel: (**a**) sample; (**b**) top view; and (**c**) indoor appearance after setup

Table 4 Thermal resistance and time constant of metal (aluminum) panels with typical structures

	Thickness of metal δ, mm	Space between adjacent pipes L, mm	R', (m^2·°C/W)	R", (m^2·°C/W)	R, (m^2·°C/W)	Time constant, min	
						Ceiling cooling	Ceiling heating
Structure I	0.5	80	0.009	0.001–0.002	0.010–0.012	0.2	0.2
Structure II	0.5	100	0.014	0.001–0.002	0.015–0.016	0.2	0.2
Structure III	1.0	100	0.007	0.001–0.002	0.008–0.009	0.3	0.3
Structure IV	1.0	140	0.013	0.001–0.003	0.014–0.016	0.5	0.5
Structure V	1.5	140	0.009	0.001–0.004	0.010–0.013	0.5	0.5

Notes: R" is related to water velocity inside the pipes, which is taken as 0.4–0.8 m/s; the higher the velocity, the larger the value of R"

Thermal resistances of radiant panels with different thicknesses and intervals between neighboring pipes can be calculated. The results indicate that, if the ratio of temperature difference between the air and the radiant panel at the end portion to that at the base portion is within 0.8–1.0 (e.g., air temperature is 26 °C, panel surface temperature at the base portion is 20 °C, and panel surface temperature at the end portion is lower than 21.2 °C), the thermal resistance of the metal radiant panel is less than 0.015 (m^2·°C)/W. Table 4 lists the thermal resistance and time constant of metal panels with typical structures. The time constants of the metal panels are quite small, i.e., less than 1 min.

Performance Summary of Different Radiant Panels

Table 5 gives the performance comparison of the five types of radiant panels described in this section. For the flat metal radiant ceiling and the suspended metal radiant ceiling, the thermal resistance along the direction of plate thickness δ is relatively small; the main thermal resistance is that along the direction of interval L between neighboring pipes. Thus, it is easy for the surface temperature distribution to be nonuniform, and the lowest surface temperature of the metal radiant panel approaches the supply chilled water temperature. For the concrete radiant floor and the capillary

Table 5 Performance comparison between different radiant panels

	$R = R' + R''$		Time constant	Δt between the surface and the supply water
	R', ($m^2 \cdot °C$)/W	R'', ($m^2 \cdot °C$)/W		
Concrete radiant floor	–0.1 (granite)	<0.005	3–4 h	Δt is relatively larger
Capillary radiant ceiling with plastering	0.02–0.06	–0.005	5–15 min	Large
Flat metal radiant ceiling	<0.02	<0.005	<1 min	Approaching the supply water temperature

radiant ceiling with plastering, the thermal resistance is mainly along the direction of the plate thickness, which helps to make the surface temperature distribution more uniform. However, the temperature difference between the surface temperature (or the lowest surface temperature) and the supply water temperature is relatively larger compared to the metal type.

Performance in Summer

Concrete Radiant Floors

Table 6 lists the operating performances of concrete radiant floors for cooling. Due to a relatively large thermal resistance of this kind of radiant panel, the surface temperature distribution is almost uniform if there is no partially shaded area. Thus, in this table, only the variances of the surface mean temperature and the cooling capacity of the radiant panels are listed. For the same indoor environment, if the thermal resistance of the radiant floor is larger, the required mean temperature of the supply and return chilled water will be lower for the same surface temperature of the radiant panel. For example, if $T_a = AUST = 26$ °C and the shortwave radiation $q_{sr} = 0$, in order to achieve a surface temperature of 21 °C for the radiant floor, the required mean temperature of the supply and return chilled water ($\overline{T}_w$) is about 18 °C for structure II, with a thermal resistance of 0.116 ($m^2 \cdot °C$)/W. For structure VII with a thermal resistance of 0.2 ($m^2 \cdot °C$)/W, the required $\overline{T}_w$ is as low as 16 °C.

The cooling capacity of the radiant floor per unit area is significantly influenced by the operating environment (i.e., the environmental parameters including air temperature, surrounding wall temperatures, solar radiation, etc.) and the temperatures of the supply and return water. Taking structure I of the concrete radiant floor as an example, with a structure of 70-mm-thick pea gravel concrete + 25-mm-thick cement mortar + 25-mm-thick granite, an external diameter of 20 mm for the supply and return water pipe, a distance of 150 mm between neighboring pipes, and $\overline{T}_w$=16 °C:

- If the indoor air temperature and the surrounding wall temperatures are all 26 °C, the floor surface mean temperature is 19.5 °C, and the radiant floor heat flux (q) is

Table 6 Operating performances of concrete radiant floors with different structures for cooling

	R, (m^2·°C)/W	$\overline{T}_w$, °C	$T_a = 26$ °C, $AUST = 26$ °C, $q_{sr} = 0$		$T_a = 26$ °C, $AUST = 28$ °C, $q_{sr} = 0$		$T_a = 26$ °C, $AUST = 26$ °C, $q_{sr} = 50$ W/m^2	
			T_s, °C	q, W/m^2	T_s, C	q, W/m^2	T_s, °C	q, W/m^2
Structure I	0.098	16	19.5	35.4	20.1	42.2	22.7	68.1
		18	20.8	28.3	21.4	35.1	24.0	61.0
		20	22.1	21.2	22.7	28.0	25.3	53.9
Structure II	0.116	16	19.9	33.3	20.6	39.7	23.4	64.0
		18	21.1	26.6	21.8	33.0	24.6	57.3
		20	22.3	20.0	23.1	26.4	25.9	50.7
Structure III	0.138	16	20.3	31.0	21.1	37.0	24.2	59.6
		18	21.4	24.8	22.2	30.8	25.4	53.4
		20	22.6	18.6	23.4	24.6	26.5	47.2
Structure VI	0.160	16	20.6	29.0	21.5	34.6	24.9	55.8
		18	21.7	23.2	22.6	28.8	26.0	50.0
		20	22.8	17.4	23.7	23.0	27.1	44.2
Structure VII	0.200	16	21.2	26.0	22.2	31.0	26.0	50.0
		18	22.2	20.8	23.2	25.8	27.0	44.8
		20	23.1	15.6	24.1	20.6	27.9	39.6

35.4 W/m^2 when there is no shortwave radiation (i.e., solar radiation) directly reaching the surface.

- If the surrounding wall temperatures are 28 °C and other conditions are the same as above, the mean surface temperature for the radiant floor increases to 20.1 °C, and the heat flux is then about 42.2 W/m2, representing an increase of about 20%.
- If the indoor air temperature and the surrounding wall temperatures are all 26 °C and there is solar radiation of 50 W/m2 directly reaching the floor surface, the mean surface temperature for the radiant floor goes up to 22.7 °C. The radiant floor heat flux increases to 68.1 W/m2, which is almost twice as large as the heat flux with no solar radiation.

Capillary Radiant Ceilings with Plastering

Table 7 lists the operating performances of the capillary radiant ceilings with plastering for cooling. It can be seen that similar conclusions can be drawn for this type of radiant panel as for the concrete radiant floors described above. For the same indoor environment, the required mean temperature of the supply and return chilled water will be lower if the thermal resistance of the radiant floor is larger for the same average surface temperature of the radiant panel. In addition, the cooling capacity of the capillary radiant ceiling per unit area is significantly influenced by the operating environment (i.e., air temperature, surrounding wall temperature, solar radiation, etc.) and the temperature of the supply and return water.

Table 7 Operating performances of capillary radiant ceilings with plastering

	R, (m^2·°C)/W	$\overline{T}_w$,°C	$T_a = 26$ °C, $AUST = 26$ °C, $q_{sr} = 0$		$T_a = 26$ °C, $AUST = 28$ °C, $q_{sr} = 0$	
			T_s, °C	q, W/m^2	T_s, °C	q, W/m^2
Structure I	0.051	16	19.2	62.0	19.5	68.8
		18	20.5	48.6	20.8	55.6
		20	21.8	35.6	22.2	42.6
Structure II	0.030	16	18.2	72.3	18.4	80.2
		18	19.7	56.5	19.9	64.5
		20	21.2	41.2	21.5	49.3
Structure III	0.073	16	19.9	54.0	20.4	60.0
		18	21.1	42.5	21.5	48.6
		20	22.3	31.2	22.7	37.4
Structure IV	0.029	16	18.1	72.9	18.3	80.8
		18	19.7	57.0	19.9	65.0
		20	21.2	41.5	21.4	49.7
Structure V	0.020	16	17.6	78.6	17.7	87.2
		18	19.2	61.3	19.4	70.0
		20	20.9	44.6	21.1	53.3

Comparing the cooling capacities of the capillary radiant ceilings with plastering listed in Table 7 and the concrete radiant floors listed in Table 6, it can be seen that the cooling capacities of the capillary radiant ceilings with plastering are higher than those of the concrete radiant floors (with the same mean temperature of the supply and return water and the same indoor environment). This performance discrepancy is mainly due to two key factors: first, the convective heat transfer coefficient for the radiant cooling ceiling is significantly higher than that of the radiant cooling floor; second, the thermal resistances of the capillary radiant ceilings are considerably lower than those of the concrete radiant floors.

Metal Radiant Ceilings

Table 8 shows the performance comparison between the flat metal radiant panels and the suspended metal radiant panels for cooling, where the angle of slant for the installation of the suspended metal type is 45°, indicating that its area for heat convection is about 2.8 times larger than that of the flat metal radiant panel with the same projected area. The cooling capacities listed in this table are calculated with the same supply water temperature for the radiant cooling panels; the lowest temperature is about 18 °C for all surfaces. The cooling capacity of the metal radiant panel increases with the decrease of the temperature difference between the supply and return water (in the condition when the lowest surface temperature remains the same). Taking structure I of the flat metal radiant roof as an example, the cooling capacities of the radiant panels are 44.4 W/m^2 and 52.5 W/m^2 when the supply/return water temperatures are 18 °C/23 °C and 18 °C/21 °C, respectively. Therefore, the increase of the cooling capacity with a lower water temperature difference is about 18%. This is

Table 8 Cooling performance comparison of flat and suspended metal radiant ceilings ($T_a = AUST = 26$ °C, $q_{sr} = 0$)

	Inlet and outlet temperature of chilled water, °C	Flat metal radiant panel		Suspended metal radiant panel	
		T_s, °C	q, W/m^2	T_s, °C	q, W/m^2
Structure I	18, 21	20.0	52.5	20.3	86.1
	18, 23	20.9	44.4	21.1	72.9
Structure II	18, 21	20.2	50.5	20.6	81.0
	18, 23	21.1	42.7	21.4	68.6
Structure III	18, 21	19.9	53.3	20.1	88.3
	18, 23	20.8	45.1	21.0	74.7
Structure IV	18, 21	20.2	50.6	20.6	81.3
	18, 23	21.1	42.8	21.4	68.8
Structure V	18, 21	20.0	52.4	20.3	85.9
	18, 23	20.9	44.3	21.2	72.7

why the operating mode of "higher water flow rate and lower temperature difference" is usually chosen for radiant panels in the condition of no condensation.

Compared with the flat metal radiant panel, the suspended metal radiant panel effectively increases the area for heat convection with the same projected area; thus, its cooling capacity increases significantly. The results shown in Table 8 indicate that the increasing percentage of the cooling capacity is about 60–65%. The cooling capacity for the suspended metal radiant roof per unit of projected area is 68–75 W/m^2 if the supply and return water temperatures are 18 °C/23 °C and increases to 81–88 W/m^2 if the water temperatures are 18 °C/21 °C. The cooling capacity of the suspended metal radiant ceiling is sufficient for the cooling requirement in most commercial buildings.

Outdoor Air Handling Process in the THIC System

In the humid 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.

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

Table 9 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 10 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

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, 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 9. 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 10, which shows that a four-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 six-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,

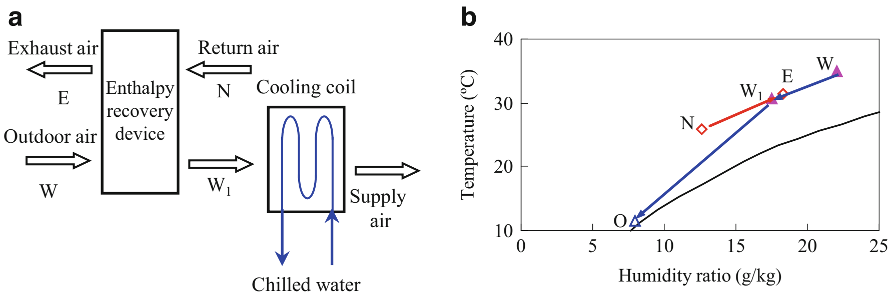

Fig. 6 Condensation dehumidification process with the enthalpy recovery module: (**a**) operating principle and (**b**) air handling process in psychrometric chart

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. The following subsection focuses on common precooling methods and reheating solutions.

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, the heat recovery device can be implemented between the indoor exhaust air and the outdoor air, where the energy can be recovered. Figure 6 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.

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. 7. The high-temperature chilled water could be directly

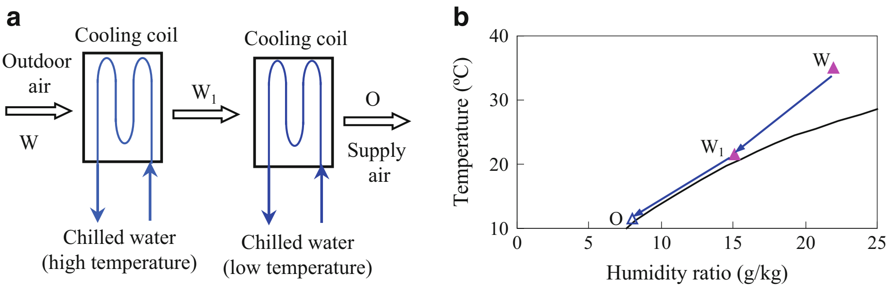

Fig. 7 Condensation dehumidification process using high-temperature chilled water for pre-cooling: (**a**) operating principle and (**b**) air handling process in psychrometric chart

Table 11 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

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. 7b. 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 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 11 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

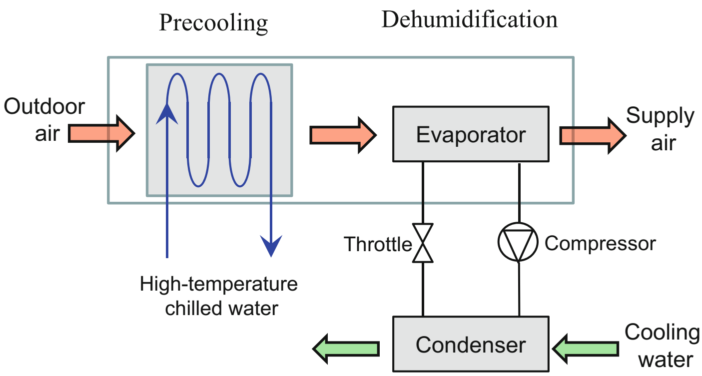

Fig. 8 Condensation dehumidification outdoor air handling process with a separate heat pump

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. 7, 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 8 illustrates an 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. 8, 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 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, steam reheating, etc. 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.

Performance of Liquid Desiccant Outdoor Air Handling Processors

For liquid desiccant outdoor air handling processors operating in summer, the outdoor air is dehumidified in the dehumidifier, and then the diluted solution is regenerated in the regenerator [8]. According to whether there is indoor exhaust air that can be adopted as regeneration air, a distinction can be made between two different kinds of outdoor air handling processors that use liquid desiccant. If there is sufficient indoor exhaust air, the processor with enthalpy recovery from the indoor exhaust air and the process of utilizing the indoor exhaust air for desiccant regeneration could be adopted for outdoor air dehumidification. Alternatively, if there is not sufficient indoor exhaust air to be utilized directly, outdoor air could be adopted as the regeneration air, and a process using high-temperature chilled water to precool the outdoor air could be constructed to improve the performance of the outdoor air handling process in the THIC system. The following subsection examines the performance of these two kinds of liquid desiccant outdoor air handling processors, in which exhaust heat from the condenser is utilized to heat the desiccant coming into the regenerator, so the air handling processes are close to the iso-relative humidity line (rather than the isenthalpic line).

Figure 9 demonstrates the operating principle of a two-stage heat pump-driven liquid desiccant outdoor air processor operating in summer, which is composed of a two-stage enthalpy recovery device (spray modules numbered I and II), a two-stage dehumidifier/regenerator, and a vapor compression refrigeration cycle. In this figure, the straight lines and dashed lines stand for liquid desiccant and refrigerant, respectively. The top channel is for the indoor exhaust air, and the bottom channel is for the outdoor air. The outdoor air first enters the two-stage enthalpy recovery device and then flows into the evaporator-cooled two-stage dehumidification modules (numbered III and IV) before being supplied into occupied spaces. In the same way, the

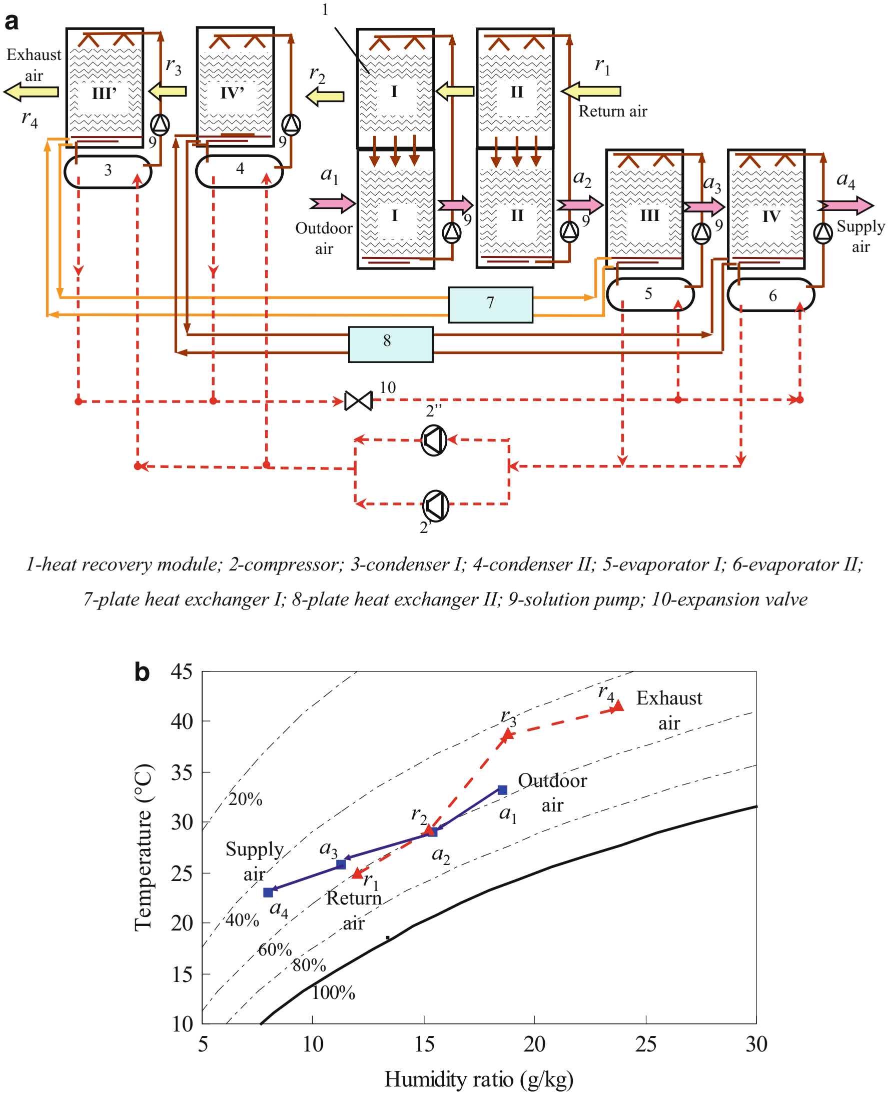

Fig. 9 Summer operation principle of the two-stage liquid desiccant outdoor air processor with enthalpy recovery: (**a**) operating schematic and (**b**) air handling process shown in psychrometric chart

indoor exhaust air first enters the enthalpy recovery device and then flows into the condenser-heated modules (numbered IV' and III') before being exhausted to the outdoor environment. The evaporator of the heat pump is adopted to further cool and dehumidify the outdoor air coming out of the enthalpy recovery device to the desired supplied temperature and humidity ratio.

The liquid desiccant in this outdoor air processor is divided into two parts. One part is stored in spray modules I and II (in Fig. 9) for the purpose of enthalpy recovery, the equivalent state of which is decided by the outdoor air and the indoor exhaust air simultaneously. The other part is stored in spray modules III, III', IV, and IV' to exchange heat with the evaporator and condenser, respectively. Taking the solution circulating between spray modules III and III' as an example to illustrate the operating principle in summer, the solution is first cooled by the evaporator (labeled 5 in Fig. 9a) and then exchanges heat and mass with the outdoor air in spray module III, where the solution is diluted and heated. The solution heated by the condenser (labeled 3 in Fig. 9a) enters spray module III' to be regenerated. The diluted solution and the regenerated solution are connected by solution pipes, and a plate heat exchanger (labeled 7 in Fig. 9a) is adopted for the internal heat recovery of the solution. The principle of spray modules IV and IV' is similar to that of modules III and III'. It is obvious that the enthalpy recovery device can efficiently reduce the energy consumption of the outdoor air handling processor. The cooling capacity and condensation heat of the heat pump are both effectively utilized in this outdoor air processor. Two parallel compressors are utilized in the heat pump cycle to operate efficiently under the partial load condition, so that the processor can have higher energy efficiency and control accuracy at partial load.

Tested Performance and Analysis at Full Load (Typical Hot and Humid Condition)

Because the heat pump system in the outdoor air processor utilizes two compressors working in parallel, and the solution in each stage exchanges heat with an individual evaporator (condenser), the dehumidification requirement at partial load can be achieved by adjusting the on-off time of these two compressors. As the required cooling temperature of the solution is about 15–20 °C, the evaporating temperature can be increased to about 10 °C. Meanwhile, the condensing temperature is only around 45 °C since the required heating temperature of the solution is about 40 °C. The test results of the outdoor air processor shown in Fig. 9 under the summer design condition are listed in Tables 12 and 13. The coefficient of performance COP_{air} is defined as

$$COP_{\mathrm{air}} = \frac{\text{Cooling capacity of the outdoor air}}{\text{Power consumption of compressor and solution pumps}} \tag{1}$$

As shown in Tables 12 and 13, *COP* of the heat pump (COP_{hp}) in the outdoor air processor is 4.0, while *COP* of the processor (COP_{air}) is 5.0. It is worth noting that the superheating temperature is 8.3 °C, which is higher than the designed 5 °C, and the subcooling temperature is 4.9 °C, which is higher than the designed 3 °C. The high superheating temperature is due to the slightly smaller expansion valve. The superheating temperature can be lowered by regulating the electronic expansion valve, and then COP_{hp} can be improved further. Moreover, the refrigerant at the expansion valve outlet is 15.0 °C, which is 8 °C higher than the evaporating

Table 12 Test results of the outdoor air processor under the summer design condition

Parameter	Unit	Value	Parameter	Unit	Value
Outdoor air flow rate	m^3/h	4058	Exhaust air flow rate	m^3/h	4021
Outdoor air dry-bulb temperature	°C	36.0	Return air dry-bulb temperature	°C	26.0
Outdoor air humidity ratio	g/kg	25.8	Return air humidity ratio	g/kg	12.6
Supply air dry-bulb temperature	°C	17.3	Exhaust air dry-bulb temperature	°C	39.1
Supply air humidity ratio	g/kg	9.1	Exhaust air humidity ratio	g/kg	38.6
Power consumption of the compressors	kW	14.6	Power consumption of the solution pump	kW	1.92
Outdoor air cooling capacity	kW	82.7	Exhaust air heating capacity	kW	101.9
Outdoor air dehumidification rate	kg/h	80.1	Cooling capacity of the heat pump	kW	59.0
COP_{air}	W/W	5.0	COP_{hp} of the heat pump	W/W	4.0

Table 13 Test results of the heat pump system

Temperature at the evaporating side			Temperature at the condensing side		
Evaporating temperature	°C	7.0	Condensing temperature	°C	45.0
Suction temperature of compressor I	°C	9.6	Exhaust temperature of compressor I	°C	67.2
Suction temperature of compressor II	°C	11.6	Exhaust temperature of compressor II	°C	66.8
Temperature at evaporator outlet	°C	15.3	Temperature at condenser outlet	°C	40.1
Temperature at expansion valve outlet	°C	15.0	Temperature at expansion valve inlet	°C	39.1

temperature. Similarly, the refrigerant at the evaporator outlet is 15.3 °C, while the suction temperatures of the two compressors are only 9.6 °C and 11.6 °C. However, the refrigerant temperature difference between the condenser outlet and the expansion valve inlet is less than 1 °C. These uncommon phenomena can probably be attributed to the unreasonable refrigerant piping layout, which results in a large pressure drop along the refrigerant pipes. Therefore, special attention should be paid to the layout of the refrigerant piping when designing the processors. The number of elbows and tees, as well as the length of copper pipe, should be minimized to reduce the pressure drop of the refrigerant along the way.

Tested Performance and Analysis at Partial Load (Summer)

Based on the test results under the design condition, performance of the processor at partial load was then tested, and the main results are listed in Table 14. Since only one compressor is in operation at partial load, while the heat transfer areas of the evaporator and that of the condenser are identical, the heat transfer between the refrigerant and the solution became more sufficient, and the heat transfer temperature differences decrease accordingly. As a result, the evaporating temperature rises, and

Table 14 Test results of the outdoor air processor at partial load under the summer condition

Parameter	Unit	Value	Parameter	Unit	Value
Outdoor air flow rate	m^3/h	4058	Exhaust air flow rate	m^3/h	4021
Outdoor air dry-bulb temperature	°C	30.0	Return air dry-bulb temperature	°C	26.0
Outdoor air humidity ratio	g/kg	17.4	Return air humidity ratio	g/kg	12.7
Supply air dry-bulb temperature	°C	17.3	Exhaust air dry-bulb temperature	°C	39.1
Supply air humidity ratio	g/kg	9.6	Exhaust air humidity ratio	g/kg	26.6
Power consumption of the compressors	kW	5.8	Power consumption of the solution pump	kW	1.43
Outdoor air cooling capacity	kW	42.7	Cooling capacity of the heat pump	kW	33.2
COP_{air}	W/W	5.9	COP_{hp} of the heat pump	W/W	5.7
Evaporating temperature	°C	11.0	Condensing temperature	°C	37.6
Superheating temperature	°C	5.6	Subcooling temperature	°C	2.7

the condensing temperature drops. As indicated by the test results listed in Table 14, the evaporating temperature increases to 11 °C, while the condensing temperature decreases by 7.4 °C compared to the full-load condition to 37.6 °C. Thus, COP_{hp} and COP_{air} increase to 5.7 and 5.9, respectively. As the outdoor air handling processor was running at partial load for more than 70% of the time, the comprehensive energy efficiency of this liquid desiccant processor could be as high as 5.5. In addition, the expansion valve can meet the refrigerant flow rate requirement at partial load, so both the superheating temperature and the subcooling temperature will be very close to the design values.

Based on these test results, it can be seen that the heat pump-driven liquid desiccant outdoor air processor can effectively meet the dehumidification requirement in summer with high comprehensive energy efficiency. The humidity ratio of the dehumidified outdoor air can meet the demand for humidity control when the supply air temperature is higher than 17 °C, which could be supplied directly into the conditioned room without further reheating or cooling. Moreover, the enthalpy recovery modules are adopted in this process to recover energy from the return air. In summary, the outdoor air processor can meet the needs of running at full load and partial load with comprehensive energy efficiency up to 5.5.

High-Temperature Cooling Sources

In THIC systems, the required temperature of the high-temperature cooling source in summer is significantly higher than that of conventional systems. As there is no dehumidification requirement, the chilled water temperature could be increased from about 5–7 °C in conventional systems to about 16–18 °C in THIC systems. This offers the possibility to utilize natural cooling sources, e.g., deep phreatic water, ground heat exchangers, direct or indirect evaporative cooling methods in certain dry regions, etc. If no natural cooling sources can be adopted directly for temperature control in the THIC system, a vapor compression refrigeration system can be utilized instead. Owing to the increased evaporating temperature of the vapor compression refrigeration cycle, the operating compression ratio of chillers in THIC systems is significantly different from that in conventional systems. Thus, new requirements are proposed for system design and device development of the vapor compression refrigeration cycle for THIC systems. In this chapter, common types of high-temperature cooling sources are examined.

In the temperature control subsystem, since the required temperature of the cooling source in the THIC system is about 16–18 °C (compared to 7 °C in conventional systems), natural cooling sources can be utilized directly as high-temperature cooling sources. If no natural cooling sources are available, mechanical chillers are needed to satisfy the requirement for the high-temperature chilled water or refrigerant to buildings. The cooling sources that produce chilled water or refrigerant with a temperature of 16–18 °C in THIC systems are referred to as high-temperature chillers in the following analysis, in order to distinguish them from the chillers adopted in conventional systems. There is a significant increase in the evaporating temperature of high-temperature chillers in the THIC system compared to conventional chillers, and theoretically, the *COP* of these high-temperature chillers is much higher than that of conventional chillers.

Key Issues for the Development of High-Temperature Water Chillers

Because of the increase of evaporating pressure, the compression ratio of high-temperature water chillers is much lower than that of conventional low-temperature water chillers, which results in new requirements for compressors and other components. Analyses of piston, scroll, screw, and centrifugal compressors for high-temperature chillers are summarized below.

Fixed Volume Ratio Compressors (Scroll, Screw, etc.)

For scroll, screw, and other fixed volume ratio compressors, overcompression or insufficient compression processes exist when the external compression ratio is not equal to the internal compression ratio. For such compressors, increasing the evaporating temperature inevitably leads to an increase in overcompression loss and a reduction of compressor efficiency. If these kinds of compressors are used under high-temperature conditions, the actual *COP* values will be far lower than the theoretical values.

"Self-Adaptive" Compressors (Piston, Centrifugal, etc.)

The suction and discharge pressures of the piston compressor are equal to the evaporating and condensing pressures, respectively. As a consequence, there is no overcompression loss. The centrifugal compressor is another type of "self-adaptive" compressor since there is no internal compression process in the compressor. Among these "self-adaptive" compressors, as the internal efficiency of piston compressors is relatively lower, centrifugal compressors are more suitable for high-temperature water chillers. To achieve efficient operation with a small compression ratio, the centrifugal compressor should adjust the inlet guide vanes and the rotation speed.

Since the refrigerant pressure difference between the inlet and the outlet of the throttle device is remarkably reduced, the throttle device of the conventional water chiller requires some regulations to satisfy the small compression ratio condition in the high-temperature water chiller.

According to the above analyses, the basic design principles of high-temperature water chillers compared to conventional water chillers can be summarized as follows:

- *Compressor*: a smaller compression ratio (centrifugal and piston compressors or specially designed compressors with fixed volume ratios) and a higher rated motor power
- *Throttle device*: greater capacity and superior regulation performance under a lower compression ratio and a lower working pressure difference
- *Evaporator and condenser*: a greater capacity and higher heat transfer efficiency (by improving the heat transfer coefficient or area to strengthen the heat transfer ability)
- *Oil return system*: working properly under the small compression ratio condition

Development Case: Centrifugal Refrigerant Cycle (GREE)

Optimal Design of the Chiller

The high-temperature centrifugal water chiller developed by GREE, shown in 6, is adopted as an example in this subsection to illustrate the development process of the high-temperature centrifugal chiller [9–11]. With an increase of the evaporating temperature in the high-temperature water chiller, some significant changes occur in the centrifugal compressor. Table 15 lists the compressor design parameters of a 4000 kW centrifugal chiller when producing 7 °C and 18 °C chilled water, using R134a as the refrigerant. Compared to the 7 °C condition, the compression ratio of the 18 °C condition decreases to 72%, while the inlet volume flow rate decreases to 67%. If a conventional 7 °C water chiller operates at a high evaporating temperature, the working condition will deviate significantly from the design condition because the system flow rate, the compression ratio, and the gas suction state all change considerably. Since the conventional 7 °C water chiller operating at a high evaporating temperature reduces system reliability without significantly improving performance, a high-temperature centrifugal water chiller must be specially designed.

Table 15 Design parameters of the compressor under different working conditions

Condition	Suction temperature (°C)	Suction pressure (kPa)	Compression ratio	Volume flow rate (m^3/s)
7 °C outlet water	5.0	345	2.7	1.5
18 °C outlet water	16.5	485	1.95	1.0

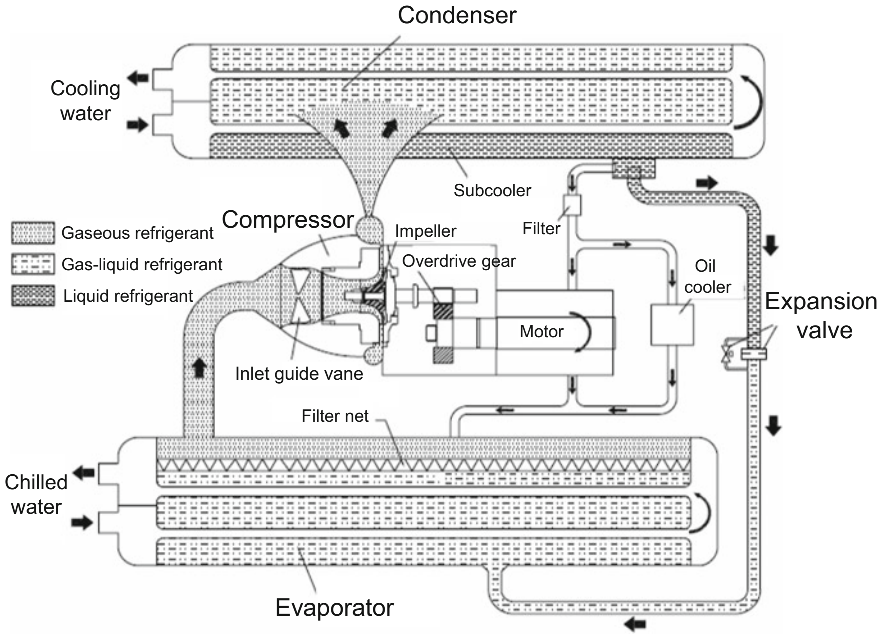

Fig. 10 Refrigeration cycle of a centrifugal high-temperature water chiller developed by GREE

The refrigerating cycle of a high-temperature centrifugal water chiller developed by GREE is shown in Fig. 10, the core technology of which is the high-temperature centrifugal compressor. The main features of the water chiller include the following: (1) The compressor effectively combines a variable geometry diffuser and adjustable inlet guide vanes to ensure the efficient and stable operation of the compressor at partial load; (2) a special motor and a lubricating oil cooling mode ensure that the motor and the lubricating oil are fully cooled and that the machine operates reliably under the small compression ratio condition; (3) the oil return process is designed to be isolated from the influence of the gas suction flow rate and the pressure difference, which ensures the reliability of the oil return under the condition of high-evaporation temperature and low compression ratio; (4) when the chiller operates in surge mode, the control system can make efficient adjustments to keep the chiller away from the surge zone and avoid the occurrence of the surge; (5) a subcooler is introduced into the condenser to improve the subcooling temperature by 3–5 °C, helping the chiller run more efficiently; and (6) a double gearbox design is adopted to reduce the gear noise; the noise of the chiller is only 80–86 db.

Tested Performance

It can be seen that the performance of the high-temperature centrifugal water chiller is improved by optimizing the structure and the working parameters. The performance of a GREE high-temperature centrifugal chiller with a rated cooling capacity of 4000 kW was tested when producing 16 °C and 18 °C high-temperature chilled

Table 16 Test results under the 16 °C chilled water condition

Item	Parameter	Unit	Load ratio			
			100%	75%	50%	25%
Test conditions	Chilled water outlet temperature	°C	16	16	16	16
	Chilled water flow rate	m^3/h	688	688	688	688
	Cooling water inlet temperature	°C	30	26	23	19
	Cooling water flow rate	m^3/h	860	860	860	860
Test results	Cooling capacity	kW	3826	2953	2034	1123
	COP	W/W	8.58	10.10	9.52	6.88
	IPLV	W/W	9.47			

Table 17 Test results under the 18 °C chilled water condition

Item	Parameter	Unit	Load ratio			
			100%	75%	50%	25%
Test conditions	Chilled water outlet temperature	°C	18	18	18	18
	Chilled water flow rate	m^3/h	688	688	688	688
	Cooling water inlet temperature	°C	30	26	23	19
	Cooling water flow rate	m^3/h	860	860	860	860
Test results	Cooling capacity	kW	3966	3046	1978	1322
	COP	W/W	9.18	10.10	9.80	8.41
	IPLV	W/W	9.77			

water. According to the test results listed in Tables 16 and 17, the *COP* of the chiller at 100% full capacity for producing 18 °C chilled water is 9.18, while the *COP* at full capacity for producing 16 °C chilled water is 8.58.

Performance Comparison

If the conventional 7 °C centrifugal water chiller is used directly to produce high-temperature chilled water, the performance is significantly different compared to the performance-optimized high-temperature water chiller. In order to clarify the performance discrepancy between the two kinds of chillers, a conventional GREE 7 °C chiller with a rated cooling capacity of 4000 kW was tested when producing 10 °C, 16 °C, and 18 °C chilled water. The results are shown in Table 18.

As shown in Table 18, if the conventional water chiller is used directly to produce high-temperature chilled water, its energy performance is significantly lower than that of the specially designed high-temperature chiller. The *COP* values for the conventional chiller producing 16 °C and 18 °C chilled water are 6.80 and 7.05, respectively. Thanks to the improvements and optimizations mentioned previously, the performance of the newly designed centrifugal high-temperature water chiller is significantly improved. Compared to the conventional chiller producing high-temperature chilled water, the energy performance of the modified high-temperature chiller producing 16 °C and 18 °C chilled water improved by 26% and 30%, respectively.

Table 18 Test results for a conventional chiller with different chilled water temperatures

Item	Parameter	Unit	Standard condition	10 °C chilled water	16 °C chilled water	18 °C chilled water
Test conditions	Chilled water outlet temperature	°C	7	10	16	18
	Chilled water flow rate	m^3/h	688	688	688	688
	Cooling water inlet temperature	°C	30	30	30	30
	Cooling water flow rate	m^3/h	860	860	860	860
Test results	Cooling capacity	kW	3963	4065	4088	4095
	Power consumption	kW	683	601.2	601.2	580.8
	COP	W/W	5.78	6.25	6.80	7.05
	η_{ad}	/	0.81	0.79	0.70	0.65
	Compression ratio	/	2.68	2.37	2.00	1.95

Design and Operation of THIC Systems

Overview of System Design

There are two subsystems in THIC system: humidity control subsystem (outdoor air handling system) is used to satisfy indoor requirements for outdoor air and removing indoor moisture load and temperature control subsystem (indoor sensible terminals) for removing indoor sensible load to control indoor temperature. Based on the fact that indoor moisture load and outdoor air flow rate required could be regarded to be proportional to occupants in most buildings, requirements for indoor humidity control and outdoor air can be both satisfied by adjusting the humidity ratio of supplied outdoor air to a certain value and introducing the handled outdoor air to indoor space with a flow rate proportional to the number of occupants. At the same time, there is another system used to regulate the cooling capacity of sensible terminals to effectively control indoor temperature. With the help of the THIC idea for thermal built environment, indoor requirements for outdoor air, humidity control, and temperature control can be satisfied at the same time.

If the required indoor temperature is 25 °C, theoretically speaking, any cooling source with a temperature lower than 25 °C can be used for the air-conditioning system in summer. In this way, natural cooling sources as well as high-temperature cooling sources with high efficiency could be used in THIC systems. However, in conventional system, under most circumstances, cooling source with a much lower temperature is required, like 7 °C chilled water, because of the need to control humidity and temperature at the same time. If condensation dehumidification method is adopted to dehumidify outdoor air and indoor air, the temperature of the cooling source must be low enough. However in summer, it is very difficult to find cooling sources with a temperature of around 7 °C naturally existing or cheaply produced. Efficiency of chillers used to produce chilled water with a temperature of around 7 °C is much lower than that of chillers used to produce chilled water with a temperature of around 15–20 °C. Besides, in actual summer conditions, 60–80% of the cooling capacity of air-conditioning system is used for sensible load to satisfy indoor temperature demand, and the application of THIC system can greatly reduce the energy consumption for indoor temperature control.

In conventional system, indoor temperature and humidity are regulated at the same time. As air exchange is the only solution to realize humidity regulation, introducing cold air to occupants' space is the way commonly used for indoor thermal built environment. However, if the task is only to remove indoor sensible load, it is not necessary to be limited by air circulation any more. Instead, water circulation can be used to directly remove sensible heat to achieve temperature regulation, such as radiant panel for cooling in summer and heating in winter. Because heat capacity of water is much higher than that of air, the profits are not only using the same unit both in summer and winter but also a great reduction of fan power consumption for ventilation and air circulation. Using water circulation, cooling/heating capacity can be transported directly into indoor environment, and the link of forced heat convection between water and air can be omitted, which reduces fan power consumption and space occupied by air ducts.

Based on the analysis above, adopting THIC system can help to greatly reduce energy consumption for air conditioning from the aspects of cooling sources and distribution system and also makes it possible to effectively use natural cooling sources. It is an energy saving approach, which can also improve indoor thermal environment, and is considered as one of the most promising directions of air-conditioning system in future. There are different types of THIC air-conditioning systems, regarding to different climates, geographical conditions, building functions and load characteristics, etc. Referring to Fig. 11, it usually consists of outdoor air handling processors and supplied terminals for outdoor air used to control indoor humidity and provide outdoor air, cooling sources used to produce high-temperature chilled water around 15–19 °C, and the corresponding indoor sensible terminals used to control indoor temperature. The main devices are briefly introduced as follows.

Outdoor Air Handling Processor

Outdoor air handling processor is used to provide outdoor air with the task of maintaining indoor air quality and extracting indoor moisture load. The air flow rate is designed to satisfy occupants' health requirement, and the indoor humidity requirement can be satisfied with a supplied air humidity ratio lower than the indoor condition (usually around 9 g/kg). During the most humid period, the outdoor air system is running with the lowest air flow rate. If outdoor air humidity ratio is around 10–11 g/kg, the air can be sent directly indoor with a larger flow rate, and the energy consumption of handling outdoor air can be saved. If the actual number of indoor occupants varies a lot, the indoor terminal with variable air volume should be adopted, regulating air flow rate according to the number of occupants, indoor humidity ratio, or CO_2 concentration. In this way, outdoor air flow rate can be reduced if the number of indoor occupants becomes less, resulting in less outdoor air processing power consumption as well as fan power consumption. In order to keep the balance of

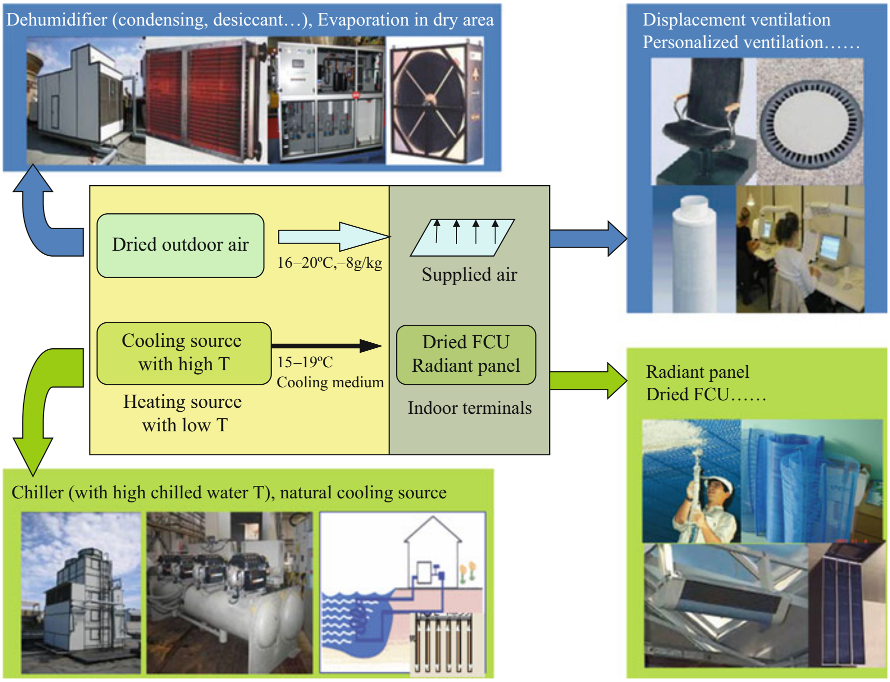

Fig. 11 Key components of THIC air-conditioning systems

indoor air pressure, indoor exhaust air with a certain flow rate should be removed in most cases. In summer and winter, exhaust air state is much closer to the required supply air state than that of outdoor air, and efficient heat recovery from indoor exhaust air is necessary to save energy consumed for outdoor air handling process.

As for the outdoor air in THIC system, the supplied air humidity ratio is around 8–10 g/kg. This can be realized by conventional condensation dehumidification system; however, this also brings the problem of a too low supplied air temperature. At present, outdoor air handling processor using liquid desiccant is recommended to be used to regulate supplied air humidity ratio by dehumidification in summer or humidification in winter and achieve energy recovery from exhaust air at the same time. Combined with enthalpy recovery device, condensing dehumidifier can realize outdoor air dehumidification in summer; however, another humidifier is required if applied in winter.

In the northwestern China, outdoor air humidity ratio is rarely higher than 12 g/kg in summer. Thus the outdoor air can be supplied indoor after being cooled down to realize removing extra moisture load without the need of dehumidification.

Production of High-Temperature Cooling Sources

Cooling source's temperature in THIC system is around 15–20 °C, instead of 7 °C in conventional air-conditioning system. The energy saving potential of THIC system can be fully realized if high-temperature cooling sources are taken advantage of.

In the eastern region of China (north of Yangtze River), annual mean temperature is usually lower than 18 °C. And underground water temperature is around 15–20 °C. If allowed by geographical condition, cooling capacity from underground water can be used by drifting wells and recharging. The operating costs are only power consumption of water pumps. As to the small-scale buildings, pipes, with water circulating in, can be buried underground as heat exchanger to get cold water with a temperature of around 20 °C, and there is no need of getting water from and recharging water into underground. If there are sufficient spaces to bury pipes and annual energy balance could be solved, this is an efficient method to obtain high-temperature cooling sources.

By the seashore, lakeside, and riverside, if temperatures of the seawater, lake water, and river water are lower than 18 °C, water can be used as the high-temperature cooling sources through heat exchangers. However, pump power consumption used for transporting water may exceed the power consumption of chillers, and then the advantage for energy saving disappears.

Chillers can be adopted when there are no natural cooling sources or it is difficult to be used. To produce high-temperature chilled water, energy efficiency of such chillers could be higher than conventional chillers. At present, centrifugal chiller and screw chiller specially developed for THIC system have been marketed in China, which can be used to produce high-temperature chilled water. To produce 16 °C chilled water, *COP* of the centrifugal chillers can exceed 8.5, and screw chillers can exceed 7.0, which are much higher than chillers used to produce 7 °C chilled water. Besides, refrigerant could be used as the high-temperature cooling source directly, with no need of secondary water loop. For example, VRF using refrigerant as the intermediate medium is a feasible solution for indoor temperature control of THIC system in small-scale buildings. Compared with conventional VRF for temperature and humidity control, this kind of VRF with an increased evaporating temperature could be more energy efficient, with a *COP* as high as 6.0 as analyzed in section "Operating and Regulating Strategy of THIC System."

Actually, in the western part of China, climate in summer is hot but dry, with the dew point temperature lower than 15 °C. Under this condition, indirect evaporating cooler can be used to produce high-temperature chilled water, which could be about 2–3 °C higher than dew point temperature of outdoor air. Without the need of vapor compression refrigeration cycle, this shows up obvious energy saving effects. At present there are already indirect evaporative cooling products widely used in large-scale commercial buildings of Xinjiang, Ningxia, Inner Mongolia, and so on, achieving comfortable indoor environment and energy saving effects.

Terminals Used to Regulate Indoor Temperature

High-temperature chilled water is needed to take away sensible heat from indoor to regulate indoor temperature, and corresponding terminals are needed. The terminals are used to deal with sensible heat only, and chilled water temperature is higher than indoor dew point temperature, so there is no condensation phenomenon. Radiant cooling panel and dry FCU are commonly used.

In the last two decades, floor radiant heating has been promoted in the northern of China with fine heating effects. In the same way, chilled water with a temperature of around 15–20 °C can be sent into the floor coils to deal with 30–50 W/m^2 cooling load (if there is solar radiation, the cooling capacity will be increased). With the help of outdoor air handling processor, which undertakes indoor moisture load, under most circumstances, air-conditioning requirements of office buildings can be satisfied. After importing from European countries and domestically developed, capillary radiant terminal has been put forward to realize indoor temperature control in China, which is installed on the ceilings or vertical walls to realize cooling or heating with cold or hot water circulating inside. With radiant cooling method, as panel surface temperature is lower than the surrounding air dew point temperature, there will be condensation risk on the panel surface. So, outdoor air systems are important to realize indoor air humidity control and ensure the normal operation of sensible terminals. Because the density of dry air is higher than that of humid air, humid air stays in the top of occupants' space. If displacement ventilation is used to supply dry air from below, humidity ratio in the lower space will be lower than that in the upper. Thus the possibility of condensation is higher for ceiling radiant cooling than that of floor radiant cooling. "On-off" control method is regarded as the best way to regulate indoor temperature for radiant cooling/heating, which has been used in many projects with floor heating and shows fine indoor temperature regulating performance.

According to conventional method using forced air circulation for air-conditioning system, air is used as the medium, which exchanges heat with high-temperature chilled water and removes indoor sensible load. The corresponding terminals include dry FCU and passive chilled beam. The former uses fans to realize forced air circulation and heat transfer, and the later depends on the natural convection caused by the sinking of cold air around the beam. The water temperature in the air-water heat exchanger is higher than indoor dew point temperature, and there is no condensation phenomenon. The condensation pipe or plate is no longer needed, the problem of condensation water leakage can be avoided, and environment pollution problem caused by mold breeding on the humid surface can be solved. However, dry FCU works at dry conditions; therefore, temperature difference between water and air is significantly reduced, requiring a larger heat transfer area for a same cooling capacity.

In recent years, there have been a number of large-scale office buildings and hospitals applying the THIC systems, located in the climate regions from Southern China to Northern China and from eastern coastal cities to Xinjiang and Inner Mongolia. After operating for a few years, the new THIC air-conditioning method shows great superiorities in indoor environment control and energy conservation.

Load Calculation of THIC System

Loads in the buildings include indoor load (sensible cooling load through envelop, solar radiant, occupant, equipment, lighting, etc., and moisture load from people) and outdoor air load (caused by the temperature and humidity ratio differences between outdoor air and indoor states). The load calculation methods are the same with those of conventional air-conditioning system. In this chapter, indoor load characteristic is introduced, and load splitting is discussed in different THIC systems.

Appointment of Indoor Sensible Load

In THIC system, the entire indoor moisture load is removed by the humidity control subsystem. If supplied air temperature of humidity control subsystem is different from indoor air temperature, humidity control subsystem removes or brings in a part of sensible load from (to) the indoor environment. According to the outdoor air handling methods, supplied air temperatures of different methods are different. Thus, the sensible heat taken away (brought in) by the humidity control subsystem will be also different.

Indoor sensible load influenced by the supplied outdoor air, Q_{HS}, is calculated by Eq. (2), where G is the outdoor air flow rate, m^3/s; c_p is specific heat capacity of air, kJ/(kg·°C); ρ is air density, kg/m^3; and t_N and t_S are indoor temperature and supplied outdoor air temperature, respectively, °C:

$$Q_{HS} = c_p \rho G(t_N - t_S) \tag{2}$$

- If condensation dehumidification and liquid desiccant dehumidification methods are used to handle outdoor air, supplied air temperature t_S is usually lower than indoor air temperature t_N, and humidity control subsystem undertakes part of indoor sensible load.
- If rotary wheel is used to handle outdoor air, supplied air temperature is usually higher than indoor temperature. Temperature control subsystem has to undertake this part of sensible load caused by the temperature difference between supplied outdoor air and indoor air, as well as entire indoor sensible load.

Figure 12 is an example if the supplied outdoor air temperature is lower than the indoor air temperature, indicating the apportionment of indoor sensible heat in THIC system. As the supplied outdoor air temperature t_S is lower than the indoor designed temperature t_N, supplied outdoor air undertakes part of indoor sensible heat, and

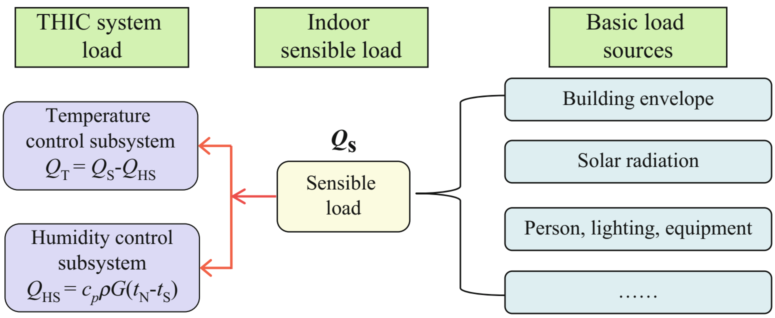

Fig. 12 Indoor sensible load appointment of the THIC system ($t_S < t_N$)

temperature control subsystem undertakes the rest of indoor sensible heat, Q_T, as shown in equation below, where Q_S is the indoor sensible heat load (sensible heat load of outdoor air is not included here), kW:

$$Q_T = Q_S - Q_{HS} \tag{3}$$

Load of Major Devices

Load of major devices in temperature control subsystem and humidity control subsystem of THIC system is analyzed in the following. In THIC system, outdoor air handling processor is the main device in humidity control subsystem, and high-temperature cooling source (usually high-temperature water chiller), distribution system, and sensible terminal devices are the main devices in temperature control subsystem. According to the plans of THIC air-conditioning systems in design stage, devices to be used for temperature and humidity control subsystems can be determined. And according to the characteristics of different devices and load calculation method mentioned above, the load of key devices can be calculated.

Load of Outdoor Air Handling Processor

The main device in humidity control subsystem is outdoor air handling processor, with the major task to produce dry and clean outdoor air and supply it into the room to control indoor humidity. Outdoor air changes from outdoor state to supply air state in outdoor air handling processor, so the load of the processor is calculated by the enthalpy variance of the outdoor air.

If there is no need for precooling or cooling using high-temperature cooling source in humidity control subsystem, load of outdoor air handling processor Q_{OAP} is calculated as Eq. (4), where Q_{OAP} is the product of outdoor air mass flow rate and enthalpy difference between outdoor air and indoor air. The outdoor air handling processor is responsible for all of indoor moisture load $Q_{in,\,m}$ and outdoor air load Q_{OA} and some of indoor sensible load Q_{HS}:

$$Q_{OAP} = Q_{in,m} + Q_{HS} + Q_{OA} \tag{4}$$

If precooling or cooling is required for outdoor air, the precooling load Q_{pre} needs to be subtracted in calculating Q_{OAP} by Eq. (5):

$$Q_{OAP} = Q_{in,m} + Q_{HS} + Q_{OA} - Q_{pre} \tag{5}$$

Load of High-Temperature Cooling Source

Major devices for temperature control subsystem include high-temperature cooling source, distribution system, and sensible terminal devices. The main task of high-temperature cooling source is for extracting the sensible load, while sometimes sensible cooling is also needed in the outdoor air handling process. Load of high-temperature cooling source is affected by different air handling methods.

If there is no need for precooling or cooling in humidity control subsystem, load of high-temperature water chiller Q_{HTC} equals to the indoor load extracted by the temperature control subsystem Q_T:

$$Q_{\mathrm{HTC}} = Q_{\mathrm{T}} \tag{6}$$

If precooling or cooling is required for outdoor air, precooling load Q_{pre} should be added to the load of high-temperature water chiller Q_{HTC}, as shown in Eq. (7):

$$Q_{\mathrm{HTC}} = Q_{\mathrm{T}} + Q_{\mathrm{pre}} \tag{7}$$

Case Study

Taking an office building in Beijing as an example, Indoor sensible peak load Q_{S}, indoor moisture peak load, outdoor air peak load, and outdoor air flow rate per unit area are 62.5 W/m^2, 6.7 W/m^2, 43.3 W/m^2, and 3.4m^3/h, respectively. THIC air-conditioning system is adopted in this building. Outdoor air (humidity control subsystem) is responsible for extracting the entire indoor moisture load, and the design temperature of the supplied outdoor air is 20 °C.

The calculation and analysis process is as follows.

Calculating the Humidity Ratio of Supplied Outdoor Air

In THIC system, humidity ratio of the supplied outdoor air is lower than indoor design humidity ratio, and the humidity ratio difference is the driving force to remove indoor moisture load. The humidity ratio of the supplied outdoor air required is as follows:

$$\omega_s = \omega_{a,in} - \frac{W}{\rho G} = 8.5 \;\; \mathrm{g/kg} \tag{8}$$

Therefore, temperature and humidity ratio of the supply air for outdoor air handling processor are 20 °C and 8.5 g/kg, respectively.

Appointment of Indoor Sensible Load

In this case, for the supply outdoor air temperature is lower than indoor design temperature, the supply air could also undertake part of indoor sensible load, which is calculated as follows:

$$Q_{\mathrm{HS}} = c_p \rho G \left(t_{\mathrm{a,in}} - t_{\mathrm{sa}}\right) = 1005 \times 1.2 \times 3.4/3600 \times (26 - 20) = 6.8\mathrm{W/m}^2 \tag{9}$$

Therefore, indoor sensible load undertaken by indoor sensible terminals is as follows:

$$Q_{\mathrm{T}} = Q_{\mathrm{S}} - Q_{\mathrm{HS}} = 62.5 - 6.8 = 55.7 \ \mathrm{W/m2} \tag{10}$$

Loads of Outdoor Air Handling Processor and High-Temperature Cooling Source

Load appointment of outdoor air handling processor and high-temperature cooling source is closely related to the devices. Here the analysis of two typical conditions is given:

- If the outdoor air handling processor is an independent device (e.g., built-in compressor), high-temperature cooling source just needs to meet the demand of indoor sensible terminals. Loads of the outdoor air handling processor and the high-temperature cooling source are calculated by Eqs. (4) and (6), respectively. The load of outdoor air handling processor is $Q_{in,\ m} + Q_{HS} + Q_{OA} = 6.7 + 6.8 + 43.3 = 56.8$ W/m^2. The load of high-temperature cooling source is $Q_T = 55.7$ W/m^2. Percentages of load undertaken by outdoor air handling processor and high-temperature cooling source are about 50% and 50%.
- If precooling by high-temperature chilled water is needed in outdoor air handling process, the high-temperature cooling source has to meet the demands of indoor sensible terminals and outdoor air precooling process at the same time. Loads of outdoor air handling processor and high-temperature cooling source are closely related to the condition of outdoor air after precooling. Then loads of outdoor air handling processor and high-temperature cooling source are calculated by Eqs. (5) and (7), respectively. Taking the precooling of outdoor air by 16 °C high-temperature chilled water as an example, as the precooling load is 34.1 W/m2, loads of outdoor air handling processor and high-temperature cooling source are 56.8–34.1 = 22.7 W/m2 and 55.7 + 34.1 = 89.8 W/m2, respectively. Percentages of load undertaken by outdoor air handling processor and high-temperature cooling source are 20% and 80%.

Annual Cooling Consumption Analysis

According to Appendix C, annual cooling loads of indoor sensible load, indoor moisture load, and outdoor air are 61.6, 3.6, and 6.8 kWh/m^2, respectively. As the supplied outdoor air temperature is 20 °C, during the whole cooling season, indoor sensible cooling load undertaken by supplied outdoor air is 6.0 kWh/m^2 and that of sensible terminals is 61.6–6.0 = 55.6 kWh/m^2.

As the outdoor air handling processor is the independent type, annual cooling load is 6.8 + 3.6 + 6.0 = 16.4 kWh/m2. And the cooling load of the high-temperature cooling source is 55.6 kWh/m2. The percentages of cooling loads are then 23% and 77%, respectively.

If precooling of outdoor air is needed, under above parameters for precooling, the cooling loads of outdoor air handling processor and high-temperature cooling source are 8.5 kWh/m2 and 63.5 kWh/m2 (including precooling outdoor air), respectively.

Operating and Regulating Strategy of THIC System

The common points for operating strategies between THIC system and conventional system are regulating methods of the chillers, chilled water pumps, cooling water pumps, and cooling towers and supplying the handled outdoor air from the processor to indoor space. The present section only illustrates the differences between the operation strategies of two systems [12–14].

Based on the concept of THIC system, a new control mode is proposed for indoor thermal environment. In this control mode, priority is given to the passive approaches that natural cooling sources and waste heat are recommended to be adopted to maintain a comfortable indoor environment. In transition season, “free-cooling” by outdoor air could be used to take away indoor heat and moisture, decreasing the operation time of the active air-conditioning system. One thing to note is to check whether the air flow rate of outdoor air ventilation system is sufficient to meet the dehumidification requirements. If natural ventilation is insufficient, active humidity control system is required to meet the dehumidification requirements. The operation modes of natural ventilation are as follows:

- If outdoor temperature and humidity ratio are both lower than these of indoor condition, directly use natural ventilation to extract indoor moisture and sensible load.
- If outdoor temperature is higher than indoor temperature, but outdoor humidity ratio is lower than indoor, natural ventilation could be used to extract indoor moisture load, but sensible terminals are required to extract indoor sensible load and control indoor temperature.
- If outdoor humidity ratio is higher than indoor, stop natural ventilation. In this condition, the passive way could not be adopted for extracting indoor moisture and sensible load any more.

There are temperature control and humidity control subsystems in THIC system, regulating indoor temperature and humidity separately. Thus, the logic of the operation strategy for THIC system is easier than that of conventional system, since there is no longer coupled temperature and humidity control. As the outdoor temperature is lower than indoor temperature but outdoor humidity ratio is higher, outdoor air dehumidification processor is required to meet the demands for outdoor air and dehumidification. In summer, condensation phenomenon should be strictly avoided. For the buildings where air-conditioning systems do not need to be in operation 24 h continuously, the devices of THIC system should be turned on in a different order compared with those of conventional air-conditioning system.

Taking a THIC system with the high-temperature water chiller, separate outdoor air handling processor (such as outdoor air handling processor using liquid desiccant or driven by separate heat pumps) and dry FCU as an example, the suggested operating order for the THIC system is as follows:

- Switch on the outdoor air handling processor in advance, and the exact time should be determined according to the realities of situation.
- Get the dew point information with the help of indoor temperature and humidity sensors. If indoor dew point temperature is lower than the supply chilled water temperature which is set about 16–18 °C, switch on the FCU and terminal water valve. Then the high-temperature water chiller could be in operation.
- The order to switch on the high-temperature water chiller is as follows successively: cooling pumps →cooling towers →chilled water pumps →chillers.

- After normal operation of the THIC system, motor damper of outdoor air branch should be automatically regulated based on the monitoring data of temperature and humidity sensor, and the water valve of FCU should be automatically regulated and switched based on the monitoring data of temperature sensor for regulating water temperature.
- The order to switch off the THIC system is as follows: high-temperature water chiller →chilled water pumps →cooling water pumps →cooling towers →dry FCU → outdoor air handling processor.

The basic concept for controlling and regulating of the entire THIC system is as follows:

- Outdoor air processor: regulating the outdoor air handling processor based on the difference between the measured value of humidity ratio (which could be calculated by the dry-bulb temperature and relative humidity) and the set point. One solution is to adopt a variable air flow rate and a fixed humidity ratio of the supply air. Another solution is to adopt a variable humidity ratio and a fixed outdoor air flow rate.
- Indoor sensible terminals (dry FCUs and radiant terminals): regulating the devices based on the difference between measured indoor temperature and the set value. The air flow rate and the water valve of the dry FCUs could both be regulated. For radiant terminals, variable duty ratio with fixed water flow rate, variable supply water temperature with mixed water pumps, and variable water flow rate could be used to regulate indoor temperature.

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