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Energy-efficient and -economic technologies for air conditioning with vapor compression refrigeration: a comprehensive review

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Abstract
Vapor Compression Refrigeration Systems (VCRS) are widely used to provide cooling or freezing for domestic/office buildings, supermarkets, data centres, etc., which expend 15% of globally electricity and contribute to ~10% of greenhouse gas emissions globally. It is reported that cooling demand is expected to grow tenfold by 2050. Therefore, it is critical to improve the efficiency of the VCRS. In this paper, a comprehensive review of advanced and hot technologies is conducted for the VCRS. These technologies include radiative cooling, cold energy storage, defrosting and frost-free, temperature and humidity independent control (THIC), ground source heat pump (GSHP), refrigerant subcooling, and condensing heat recovery. Radiative cooling could produce a cold source ~8 ^{o}C lower than the surroundings, which reduces the electricity consumption of the VCRS by ~21%; cold energy storage is used to shift the peak cooling load, and as a result, the electricity consumption and operation cost of the VCRS could be reduced by ~12% and ~32%, respectively; frosting is a big issue of the VCRS especially for freezing applications, and more than 60% of electricity consumption for defrosting could be saved with the advanced defrosting and frost-free technologies; THIC deals with the building sensible load and latent load separately, which not only increases the COP of the VCRS by ~35%, but also improves the building thermal comfort; GSHP uses the ground as a low-temperature cooling source for condensing the refrigerant in the VCRS in summer, which decreases the condensing temperature by ~5 ^{o}C and correspondingly increases the COP of the VCRS by ~14%; refrigerant subcooling and condensing heat recovery can increase the refrigerating capacity and achieve multi-functions of the VCRS, respectively. The review is summarized in terms of the technology classification, basic ideas, advantages/disadvantages, current research status and efforts to be made in the future.

***Keywords*:** Radiative cooling; Energy storage; Defrosting and frost-free; Air conditioning; Heat recovery; Heat pump

Contents

Nomenclature

AC	Air conditioning
COP	Coefficient of performance
ED	Electrodialysis
GSHP	Ground source heat pump
Le	Lewis
MPCMs	Microencapsulated phase change materials
NTU	Number of transfer unit
PCMs	Phase change materials
TES	Thermal energy storage
THIC	Temperature and humidity independent control
VCRS	Vapor compression refrigeration system

1. Introduction

The refrigeration system plays an indispensable role in many areas, such as residential or commercial buildings, industry, cold chains, etc. It provides thermal comfort for buildings, keeps food or medicine at desired temperatures, and is essential for some industrial processes as well, such as air liquefaction. With the increase of population and development of urbanization, the demand for cooling increases significantly, which is reported to grow tenfold by 2050 [1]. Hence, the increase of energy consumption of refrigeration systems is expected. However, according to the annual energy outlook 2017 (with projections to 2050) [2], it is found that energy consumption of refrigeration and cooling, in both residential and commercial sectors, remains relatively flat or decline slightly from 2016 to 2040, as shown in Fig. 1, despite the growth in the number of households and the amount of commercial floor space. This is mainly due to the improved refrigeration efficiencies by the use of advanced technologies.

Fig. 1 Residential and commercial sectors delivered energy consumption [2]

Up to now, Vapor Compression Refrigeration System (VCRS) is the most popular and widely used refrigeration system, which has a market share of 80%. Therefore, it is critical to improve the efficiency of the VCRS for decreasing the energy consumption of refrigeration. As a matter of fact, relevant technologies have been investigated since early last century and some of them have been widely applied in the VCRS. These technologies include radiative cooling, cold energy storage, defrosting and frost-free, temperature and humidity independent control (THIC), ground source heat pump (GSHP), refrigerant subcooling, and condensing heat recovery.

The radiative cooling technology is based on the mechanism that a body at typical ambient temperatures can radiate heat into the colder outer space at 273 K, which cools down the body’s temperature and achieves cooling effect. Vall and Castell [3] made a review on radiative cooling consisting of backgrounds, theoretical approaches, numerical simulations and prototypes, but they did not discuss the potential of the radiative cooling for improving the VCRS performance.

The basic idea of the cold energy storage technology is to generate cold energy at off-peak times, store it with energy storage media, and then release it at peak times. It can not only save energy by storing excess cold energy of the VCRS, but also reduce the operation cost due to the cheap off-peak electricity. Moreno et al. [4] conducted a review of thermal energy storage of heat pumps for building cooling and heating, using phase change materials (PCMs) as the energy storage media.

Frost formation is inevitable in the VCRS when surface temperature of heat exchangers is below 0 ^{o}C. On the one hand, frosting significantly decreases the operating efficiency of the VCRS due to the increased thermal resistance. On the other hand, it consumes large amounts of energy for defrosting. Therefore, energy-efficient defrosting or frost-free technologies are required. Wang et al. [5] presented a review of anti-frosting technologies in refrigeration and air conditioning fields, with an emphasis on the surface treatment technology.

The THIC deals with the sensible load and latent load of air conditioning individually: the latent load is processed by liquid or solid desiccant dehumidification; the sensible load could be processed by the VCRS with a much higher evaporating temperature at ~15 ^{o}C (traditionally it is ~5 ^{o}C), which significantly improves the performance of the VCRS. Giampieri et al. [6] reviewed the working principle of liquid desiccant systems, focusing on the thermodynamic properties of the desiccant solutions to identify which thermodynamic property influences the liquid desiccant process. Sultan et al. [7] made a review on the solid desiccant air conditioning system and comparisons were made with the conventional vapor compression air conditioning system.

Ground source heat pump (GSHP) for cooling relies on the fact that the ground has a lower temperature than the air in summer. Therefore, the ground could work as a low-temperature cooling source for condensing the refrigerant in the VCRS and hence increase the system performance. Lucia et al. [8] made a review on the GSHP systems with first and second law thermodynamic approaches for modeling. However, they only covered the horizontal and vertical ground heat exchangers without the pile foundation ground heat exchanger.

In the VCRS, subcooling the refrigerant at the inlet of the expansion device can not only increase the refrigerating capacity but also reduce throttling losses in the expansion device. Qureshi and Zubair [9] conducted a review of using refrigerant subcooling to save energy and increase performance of the VCRS, while it was only focused on the mechanical subcooling method.

There is a large amount of condensing heat in the VCRS, which is ~1.2 times of the refrigerating capacity. What's more, the condensing heat could be up to 80 ^{o}C in some cases, such as the CO_2 system. Therefore, it is necessary to recover the condensing heat for other applications, which could enrich the functions of the VCRS. Some researchers proposed to use the condensing heat for hot water or liquid desiccant regeneration. However, no literature review is available till now.

Based on the literature survey and to the best of the authors' knowledge, no recent review has been conducted on the VCRS for summarizing, describing, and comparing these technologies in depth till now. The latest review on the VCRS was made by Park et al. [10] in 2015, but it was mainly focused on the configuration of the VCRS, such as expansion loss recovery and multi-stage compression, without integration with other technologies. Considering that there is a high demand on cooling in the near future, it is quite necessary to summarize the most advanced technologies that can be applied in the VCRS and provide a clear vision to achieve energy-efficient and -economic cooling. In the paper, a comprehensive review of advanced technologies is conducted for the VCRS. These technologies have high potential to integrate with the current VCRS, as shown in Fig. 2. For each technology, it is summarized in terms of the technology sub-categories, merits and drawbacks of each technology and current research status. More importantly, it also identified possible development space and necessary efforts to be made in the future.

Fig. 3 Advanced technologies for improving the performance of vapor compression refrigeration systems.

2. Radiative cooling technology

Radiative cooling is a passive cooling strategy based on the mechanism that the atmosphere of the Earth acts as a transparent window for electromagnetic waves of 8-13 μm, which are just the wavelengths of peak thermal radiation of a body at the typical ambient temperature [11]. Through the window, a body is cooled through radiating the heat into the colder outer space which is regarded as an infinite-size reservoir of 273 K. For example, the morning dew and frost are the direct consequence of the above effect.

The radiative cooling is attractive as it can improve the energy efficiency of refrigeration systems by only consuming little energy with circulation pumps, and has great potential to work with the VCRS as a supplemental cooling source and hence reduce the energy consumption of the VCRS.

2.1 Nighttime radiative cooling

It has been extensively investigated of nighttime radiative cooling systems since 1918 [12], with both organic and inorganic materials of promising infrared-emissivity. The idealized spectral emissivity (absorptivity) of the selective materials for night cooling is presented in Fig. 4 [13].

Fig. 5 Idealized spectral emissivity (absorptivity) of the selective materials from ultraviolet to mid-infrared [13]

In recent years, Tevar et al. [14] described the design and establishment of three different night cooling systems with low cost, easy to install panels, which reduced the cooling load significantly. Experimental results showed night cooling power was 63.7 W/m^2 for an organic panel, 42.2 W/m^2 for a metallic plate with white coating, and 7.7 W/m^2 for a metallic plate with black coating. Hu et al. [15] proposed a new photovoltaic-photothermic-nocturnal radiative cooling system, which collected heat energy at daytime by photovoltaic and photothermic conversions and generate cooling effect at nighttime with radiative cooling. The temperature difference between surroundings and the collecting plate varied between 8 and 13 ^oC at nighttime radiative cooling.

Considering that space cooling is more crucial than heating for buildings in hot regions, Zhao et al. [**Error! Bookmark not defined.**] proposed a building integrated photovoltaic-radiative cooling system that can generate electricity via photovoltaic conversion during daytime and generate cooling energy via radiative cooling during nighttime. The schematic of the system was shown in Fig. 6. The total electricity production and cooling energy gain of the system were 96.96% higher than those of the building integrated photovoltaic system.

Fig. 7 Schematic of the building integrated photovoltaic-radiative cooling system [Error! Bookmark not defined.]

Nighttime radiative cooling provides cold energy at night, while most of time, the cooling demand is required during office hours in the daytime. Therefore, it is necessary to store the cold energy at night and then discharge it during the day, where water is normally used as the cold storage medium [16]. To increase the energy storage density, Zhang and Niu [17] investigated the use of microencapsulated phase change materials (MPCMs) with water as the carrier fluid for a nighttime radiative cooling system. It was concluded that MPCMs slurry appeared to be a good energy storage medium, and energy saving potential can reach up to 77% for low-rise buildings.

2.2 Daytime radiative cooling

Significant cooling under direct sunlight is only achieved recently, because solar irradiation during the day results in heating of the radiative cooler which counteracts the radiative cooling effect. To realize daytime cooling, the radiative material should be a broadband mirror of solar light and a strong thermal emitter in the atmospheric transparency window as well [11]. Recent advancement in nano-photonics and meta-materials makes daytime radiative cooling possible.

In 2013, Rephaeli et al. [11] presented a metal-dielectric photonic structure, as shown in Fig. 8. This structure behaved as a broadband mirror of solar light and emitted strongly in the atmospheric transparency window, resulting in a net cooling power up to 100 W/m^2 at ambient temperatures. Further in 2014, Raman et al. [18] introduced an integrated photonic radiative cooler composed of several layers of HfO_2 and SiO_2, reflecting 97% of incident sunlight and also emitting effectively in the atmospheric transparency window. Experimental results demonstrated that, exposed to direct sunlight over 850 W/m^2, the temperature of the photonic radiative cooler was cooled to 4.9 ^{o}C lower than the ambient air temperature and had a cooling power of 40.1 W/m^2 at the ambient air temperature. Recently in 2017, Goldstein et al. [1] proposed to use a radiative cooling panel as a condenser of the VCRS to decrease the condensing temperature of refrigerants, as shown in Fig. 9. Simulation results showed that electricity consumption in cooling was reduced by 21% for a two-story office building in a region of hot and dry climate.

Fig. 10 The metal-dielectric photonic structure for daytime radiative cooling [11]

Fig. 11 A radiative-cooled vapor compression refrigeration system [1]

The nanophotonic approach needs strict and nanometer-precision fabrication, so it is difficult to be applied to the large area of the residential and commercial buildings. Hence in 2017, Zhai et al. [19] integrated resonant polar dielectric microspheres in a polymeric matrix randomly, generating a metamaterial which was wholly transparent to the solar spectrum and had infrared emissivity larger than 0.93 across the atmospheric window. As shown in Fig. 12, the metamaterial consisted of a visibly transparent polymer which encapsulated distributed silicon dioxide (SiO_2) microspheres randomly and backed with silver coating. Under direct sunshine, it showed a radiative cooling power of 93 W/m^2 at noon-time.

Fig. 13 A schematic of the polymer-based hybrid metamaterial with randomly distributed SiO_2 microspheres for large-scale radiative cooling [19]

In addition, harnessing radiative cooling at a low energy flux as small as 100 W/m^2 or less, is challenging. This is because, in a conventional radiative cooling system, the collection and storage of cold energy requires an electric pump to drive the heat transfer fluid, which accounts for ~20% of the total cold gain. To reduce the self-energy consumption, Zhao et al. [20] proposed a single-phase thermosiphon, using the buoyancy force to drive heat transfer fluid. It did not consume electricity so increase the net gain of the radiative cooling system. Meanwhile, Zhang et al. [21] had proposed a pipe network with low-pump-power requirement for cold energy collection generated by the radiative cooling.

2.3 Summary

Radiative cooling is a free cooling technology taking advantage of the Earth's transparent window. Nighttime radiative cooling has been widely investigated since 1918. However, the materials in the early stage can only provide cooling at night. The reason is that the radiative materials also absorb the solar light in the daytime which counteracts the cooling effect. To make it use in daytime, a good method is to reserve the cold energy at night while release it in the daytime.

Daytime radiative cooling was realized only recently in 2013 with the invention and application of nano-photonic materials, which act as a broadband mirror of solar light and emit large amounts of energy simultaneously. Then in 2017, meta-materials were developed with high-throughput, economical roll-to-roll manufacturing for large scale applications. It shows a large opportunity to couple daytime radiative cooling to some energy systems to improve their energy efficiency. In addition, it is found that the reflected sunlight in the daytime is still unused, and methods to use the reflected sunlight will be a good research direction.

3. Cold energy storage technology

The demand of cold energy varies significantly during peak and off-peak periods, leading to the inefficiency of refrigeration systems. In addition, the refrigeration system is usually over-designed to cope with the load fluctuation, while in most cases the system operates in partial-load.

It is a good idea to generate cold energy in the midnight or off-peak periods, and release it in the daytime or peak periods, as shown in Fig. 14. On the one hand, the refrigeration system is able to handle the cooling load fluctuation more flexibly and the system capacity could be significantly reduced. On the other hand, it could mitigate the peak electricity usage and the operation cost of the refrigeration system is therefore reduced since the off-peak electricity is quite cheap. However, there is a temporal mismatch between cold generation and usage, which could be eliminated by the cold energy storage systems. For refrigeration, three types of cold storage media are commonly used: chilled water, ice and phase change materials (PCMs) [22].

Fig. 15 Schematic diagram of the cold energy storage

3.1 Chilled water storage

In chilled water storage systems, a constant source of water (usually a water tank) is utilized to store the energy which can be provided by a central or off-site chilled water plant. At the early stage, the research concentrated in the investigation of the design and performance of the chilled water storage tank. In 2003, Fahl´en and Karlsson [23] predicted the gains with employing a buffer tank to match the capacity and load better. Then they [24] adopted parallel design with a storage tank in the by-pass line, making the flows in both of the heat pump and heating system keep at their optimum. Sebzali and Rubini [25] investigated the influence of chilled water storage on the performance of air cooled chillers in Kuwait. The results demonstrated that chilled water storage could reduce the peak electrical load of a chiller by up to 100% and decrease the nominal chiller size by up to 33% with the operating strategy. Corberan et al. [26] installed a tank at the exit of the internal circulation pump to produce thermal inertia of the system. They developed a mathematical model to describe the performance of the above ground-source heat pump, which was used to provide heat and cool to an institutional building located in a mediterranean climate. Cervera-Vázquez et al. [27] sized the volume of the buffer tank by employing the minimum ON time as the criterion for adequate compressor oil return. Recently in 2016, Cervera-Vázquez et al. [28] carried out a more comprehensive study of the sizing of the buffer tank in a chilled water air conditioning system. They found that three factors were affected by the tank volume, i.e, ON cycle time, OFF cycle time and the number of starts of the chiller compressor. The design guideline was proposed to decide the minimum volume of buffer tanks.

3.2 Ice storage

Ice storage is another popular cold storage method. Ice is generated during off-peak hours and melts to provide cold energy during peak hours. To make ice, the evaporating temperature of an ice storage refrigeration system is reduced by 8-10 °C compared with a conventional refrigeration system, which results in an efficiency reduction by 30-40% [29]. In Table 1, it summarized the research on ice storage air conditioning system.

Table 1 Research on ice storage air conditioning systems

Firstly, the design and optimization of ice storage control systems has always been the hot topic, as experience states that poor design and operation could lead to disappointing results [30,31,32]. Early in 1992, Braun [33] had conducted daily simulation of an ice storage system and compared its performance under chiller-priority, storage-priority and optimal control. Later in 2012, Hajiah and Krarti [34,35] demonstrated a simulation environment to predict the benefits of using building thermal capacitance and ice storage systems simultaneously, and developed corresponding optimal controls to minimize energy charges, demand charges, or combined energy and demand charges. The results showed that the savings of energy cost, demand cost and total cost could be up to 11.2%, 13.2% and 10.8%, respectively. More recently, Candanedo et al. [36] formulated a model-based predictive control algorithm for the cooling plant of a small commercial building. With the control algorithm, the operation cost was saved by 5-20% with the modified storage-priority strategy and 20-30% with the chiller-priority strategy. Powell et al. [37] presented a novel technique to handle the dynamic chiller loading with thermal energy storage in a district cooling system. Two objective functions were solved to minimize total energy consumption and total cost over a 24h time period. Recently, Lin et al. [38] optimized the ice storage air conditioning (AC) system with a hybrid algorithm (Ant-Based Radial Basis Function Network). The simulation results indicated that with the algorithm, the ice storage AC system provided greater energy efficiency in dispatching chillers, and hence reduced the electricity cost.

Secondly, a lot of research had been done to investigate the performance of the ice storage system with experiment or simulation. Chaichana et al. [39] reported that the ice storage system could reduce 55% of electricity consumption compared with the conventional system. Sebzali and Rubini [40] selected a building in Kuwait to study the effects of ice storage on the size of chiller and storage unit, decrease of peak electrical demand and chiller energy consumption, with the ESP-r building energy simulation program. Fang et al. [41] experimentally studied the performance of ice storage AC systems with separate helical heat pipes. Fang and Liu [42] compared the exergy performance between the ice storage AC systems with heat pipe and ice-on-coil. The simulation showed the exergy efficiency of the former system was 9.55% higher than that of the latter one. Yau and Lee [43] employed TRNSYS to analyze the feasibility of incorporating an ice slurry-cooling coil for AC systems. It was found the modification could improve the energy management and dehumidification performance of the system. Sehar et al. [44] used a quick assessment tool of demand response to simulate the ice storage systems for shifting the peak cooling demand in large and medium-sized office buildings. Zhang et al. [45] analyzed the performance of an ice storage AC system in Xi'an Xianyang International airport. It was found that ice storage could decrease the operating costs, and the supplied chilled water temperature could be as low as 3 ^{o}C. As shown in Fig. 16, Sanaye and Hekmatian [46] analyzed an ice storage AC system at full and partial operating modes. The electricity consumption was decreased by 11.83% and 10.23% for full and partial operating modes respectively. Correspondingly, the electricity cost was reduced by 32.65% and 13.45% due to the switch of the peak load. Honsef and Yari [47] employed ice storage in residential AC systems to balance the energy consumption in Iran. They conducted a case study in Iran and found, with an ice storage AC system, the cooling charge was approximately 18.6% lower than that of the conventional system. Gasia et al. [48] carried out an experimental study to evaluate the influence of dynamic melting in a shell-and-tube heat exchanger adopting water as the phase change material. From experiments, it was concluded that dynamic melting was effective to enhance heat transfer during the melting process of PCMs.

Finally, there are few literatures about the cost effectiveness of ice storage systems, considering the capital investment and electricity cost savings. To verify the economic feasibility under different tariff structure, Chan et al. [49] evaluated the performance of district cooling plants adopting ice storage. The payback period was estimated to be ~22 years if a 60% ice storage system with chiller priority was operated. Then, Habeebullah [50] investigated the economic feasibility of retrofitting the old air conditioning plant with an ice storage system for the Grand Holy Mosque in Makkah, Saudi Arabia. The investment could be paid back with 10 years, with base tariff strategy. The saving for the first 10 years was 163 $/d, and the net daily saving later would be 363$/d. Campoccia et al. [51] discussed the influences of ice storage systems on the power daily profile of residential buildings, and examined the economic repercussions. Sanaye and Shirazi [52] performed thermo-economic optimization and environmental penalties of an ice storage system. The results showed that the electricity consumption and CO_2 emission of the ice storage system were 9% and 9.8% lower than a conventional system, respectively. Furthermore, the capital cost could be paid back in 3.43 years.

Fig. 17 The schematic diagram of the ice storage system for full operating mode: (a) discharging cycle; (b) charging cycle [46]

3.3 PCMs storage

Energy storage with phase change materials (PCMs) has attracted more and more attention in recent years as a result of the advantages, such as large energy storage density, energy storage and release at relatively constant temperatures, compactness and low weight per unit storage capacity [53]. In Fig. 18, it shows the families of PCMs [54]. Since the 1800s, PCMs have been used for various energy storage, but they have started to be used as cold storage media much later from around 1990s [55]. For cold energy storage, the commonly used PCMs are organics (especially paraffin), salt hydrates, and salt solutions.

Fig. 19 Families of phase change heat storage materials [54]

The most popular PCMs are organics, especially the paraffin, which are commonly used for cold storage above 0 ºC. He et al. [56,57] demonstrated the potential of using n-tetradedcane and n-hexadecane and their binary mixtures as PCMs for cool storage. Wu et al. [58] presented a model to study the dynamic performance of a packed bed adopting n-tetradecane as PCMs. Numerical simulations were conducted to investigate the influences of some factors, such as inlet temperature and flow rate of energy transfer medium, porosity of the packed bed, and size of the capsules, on the thermal performances of the thermal energy storage system. Stritih and Butala [59] conducted an experiment using paraffin to store cold energy at the night-time and to cool the air at the daytime in summer. Parameshwaran et al. [60] investigated a novel system which was a combination of variable air volume based chilled water air conditioning system and thermal energy storage system. The PCMs showed good characteristics of charging and discharging, resulting in saving energy used for cooling and ventilation. Then Parameshwaran and Kalaiselvam [61,62] applied the nanocomposite PCMs as the media of thermal energy storage. The new PCMs improved the thermal conductivity by 7.3-58.4% and its time for complete freezing was decreased by 15%. Barzin et al. [63] investigated combining night ventilation with PCMs-impregnated gypsum boards for cooling purposes experimentally. If night ventilation was adopted to charge PCMs, a saving of 73% was achieved for weekly electricity. Rahdar et al. [64] compared the exergetic, economic and environmental performance of ice and PCMs thermal energy storage for air-conditioning systems in the office building. The main outcomes are shown in Table 2. Al-Abidi et al. [65] reviewed previous works on the PCMs used in air conditioning systems and the existed methods to enhance their heat transfer performance. De Falco et al. [66] presented an innovative PCMs device for air conditioning, as shown in Fig. 20. They added a storage tank to a traditional chiller-fan coil system to shift the electricity peak loads. The data from experiment and simulation with a 5 kWh prototype demonstrated that the system could save energy significantly as a result of chiller efficiency improvement.

Table 2 Results of multi-objective optimization for the end points of the ice storage and PCMs storage AC systems [64]

Fig. 21 PCMs cold storage tank layout (a) and 3D rendering (b) [66]

Salt hydrates are usually used for cold storage above 0 ºC. In 2001, Vakilaltojjar and Saman

[67] proposed a PCMs system for air conditioning applications. The PCMs consisted of calcium chloride hexahydrate and potassium fluoride tetrahydrate. They developed a model for calculation, and the effect of design parameters was investigated. Tyagi et al. [68] experimentally studied a thermal management system using calcium chloride hexahydrate for cool energy storage. It was found that the room temperature could be maintained for long time with the thermal management system even there was an active heating load. Chen et al. [69] presented an experimental study on the features of cold storage ejector cooling system with a cylindrical PCMs tank. The PCMs they used were composed of inorganic salts, nucleating agents and thickening agents. The results showed that hybrid of the PCMs cold storage and ejector could achieve a more stable COP. Moreno et al. [70] experimentally tested a heat pump integrated with thermal energy storage tanks operated in summer. Compared with the water tank, the PCMs tank was capable of supplying 14.5% additional cold and maintaining the indoor temperature at a comfort level about 20.65% longer. However, the tank charge time was much longer as well, i.e., 4.55 times of the water tank.

Salt solutions are used for cold storage below 0 ^{o}C, and commonly used in refrigerated truck, domestic and commercial freezers. Azzouz et al. [71] presented experimental results of the performance of a household refrigerator with PCMs. It was found that the storage capacity with a eutectic aqueous solution was a litter smaller than with water, but it has the ability to maintain the air in the refrigerated cell at proper temperature values. To maintain refrigerated trucks at the desired thermal conditions, Liu et al. [72] proposed an innovative refrigeration system with PCMs. The PCMs were made of an inorganic salt-water solution. The system consumed less energy and was more environmentally friendly. Oro´ et al. [73] improved the thermal performance of commercial freezers by employing PCMs under the condition of door openings and electrical power failure. Later, Oro´ et al. [74] provided a review of the potential energy savings and CO_2 mitigation with PCMs for cold storage and transportation systems in Spanish and European. Perier-Muzet et al. [75] studied a large solar thermoacoustic refrigerator which was able to achieve temperatures of the industrial refrigeration. To guarantee abundant cooling capacity to deal with refrigeration loads of the production fluctuations, the refrigerator was integrated with latent cold storage. Kozak [76] explored both experimentally and theoretically the thermal performance of transported insulated cold storage packages that can keep products at a low temperature (-33 ^{o}C). Liu et al. [77] presented a novel air-cooled household refrigerator with PCMs, with which the energy consumption was reduced by 18.6% and the compressor ON-time ratio was reduced by 13.6%.

3.4 Summary

Table 3 summarized the advantages and disadvantages of chilled water, ice and PCMs energy storage, among which PCMs attract the most attention and chilled water is the least popular mainly due to its low energy storage density.

Table 3 Comparison of chilled water, ice and PCMs storage

Ice energy storage has a large latent heat capacity of phase change (334 kJ/kg) and low cost. However, it suffers from a melting point of 0 ^{o}C along with a supercooling degree of 4-6 ^{o}C, resulting in an increase in energy consumption for lowering the evaporation temperature of refrigerating systems.

PCMs energy storage uses phase change materials, such as organics (especially paraffin), salt solutions and salt hydrates, and could match well with the evaporation temperature of refrigeration systems. Salt solutions and salt hydrates have a large energy storage density and lower cost, while their disadvantages include supercooling, phase segregation and corrosion to metals. Organics possess the advantages of suitable phase change temperature, high latent heat and good stability, making them promising for use as cold storage media, but it has a higher cost and is only for cold storage above 0 ^{o}C.

Both ice and PCMs have a low thermal conductivity, which results in a slow phase change process since heat is transferred mainly by conduction for static ice or PCMs. Currently, there are different methods to improve the heat transfer rate, among which PCMs/ice slurries may be most potential for energy storage and heat transportation in rapid cooling applications. It is because that they are fluid so can flow during phase transition. Therefore, the convective heat transfer will happen instead of conduction to enhance the heat transfer during melting and solidification processes.

4. Defrosting and frost-free technologies

Frost formation on surfaces of heat exchangers is a big issue of vapor compression refrigeration systems in different applications, such as refrigerated trucks, fridges, air-source heat pumps, etc. Frost forms when three necessary conditions are satisfied: (1) moist air is in contact with the heat exchanger; (2) the temperature of the air should be decreased below its dew point temperature; (3) the temperature of the heat exchanger surface is below 0 ^{o}C.

Frost depositing and accumulating will be a layer of thermal resistance between the wall and air, hence degrading system performance. To solve the above problems, two technologies are usually used: the defrosting technology and frost-free technology. For the defrosting technology, the basic idea is to retard the growth of the frost or melt the formed frost, but frost formation is unavoidable; while, for the frost-free technology, the frost is preventing from forming on the heat exchanger surface.

4.1 Defrosting technology

Currently, there are two main defrosting methods: passive defrosting and active defrosting. The passive defrosting uses surface coating to delay the frost formation without additional power consumption, while the active defrosting requires additional power input, and includes the methods of reverse cycle and ultrasonic vibration.

4.1.1 Surface coating

Hydrophobic or super hydrophobic surface coating attract much attention because of its better performance of defrosting. The reason is that the hydrophobic surface has a larger contact angle and lower surface energy, so the frost structure becomes comparatively fragile.

The defrosting performances of different surfaces, namely, bare, hydrophilic and hydrophobic surfaces, had been compared. Early in 1983, Seki et al. [78] studied the hydrophilic and hydrophobic surfaces and found that frost formation on the hydrophilic surface was quicker than that on the hydrophobic surface. In 2015, Wang et al. [79,80] reported that the melting time was 128 s for hydrophilic, 147 s for bare and 107 s for super hydrophobic heat exchanger. Besides, as shown in Fig. 22, it was easier for condensate droplets to roll and depart on the super hydrophobic surfaces than on the hydrophilic and bare surfaces.

Fig. 23 Molten water retention on three types of fins [80]

In recent two years, more research focuses on the defrosting mechanism of hydrophobic surfaces. Chu et al. [81] developed a theoretical model to analyze the energy change during defrosting and to predict the meltwater shrink angle. Kim et al. [82] investigated the methods of fabrication and frosting features of hydrophobic surfaces and discussed potential directions to improve the hydrophobic surface technology. In 2017, Sommers et al. [83] proposed a semi-empirical relational expression to predicate the frost density on vertical hydrophilic and hydrophobic surfaces. Chu [84] studied the droplet freezing and frost melting phenomena of an ultra slippery super hydrophobic surface.

But there are some different voices. Okoroafor and Newborough [85] found that the frost was reduced by 10-30% when they used a polymeric material based hydrophilic surface for more than 2 h. Therefore, some special hydrophilic surfaces might also have a positive influence on defrosting. In 2011, Kim and Lee [86] reported that the required defrosting time was almost the same for various surface treatments, including hydrophilic, bare, and hydrophobic surfaces.

Therefore, it deserves further study to uncover the underlying mechanism of the enhancement of defrosting with the surface coating.

4.1.2 Reverse cycle defrosting

Reverse cycle defrosting is a widely used active method of defrosting. It is realized by reversing the heat exchanger from an evaporator mode to a condenser mode with valve regulating. In general, the defrosting period is expected to be as short as possible. The reverse cycle defrosting might cause low-pressure cutoff or wet compression, which may result in system shut-down and potential compressor damages. Besides, the melted water stays at the heat exchanger surfaces which must be removed to prevent it from frosting again.

To solve the above problems, a lot of efforts have been done. Ding et al. [87] proposed to by-pass the thermal expansive valve in the mode of defrosting. The results showed that there was almost no improvement in shortening the defrosting time, but the system could return to a normal working mode smoothly without activating the low-pressure switch. Hu et al. [88] developed a new defrosting method based on reverse cycle for air-source heat pumps with thermal energy storage. As shown in Fig. 24, a heat exchanger of phase change material was added to store the excess heating capacity in part loads, which was then used as an evaporator in defrosting. The corresponding defrosting time was shortened by 38%, and the risk of compressor shutting down was minimized. Qu et al. [89] introduced an electronic expansion valve to a 6.5 kW residential air-source heat pump. Experimental results revealed that when the valve was adjusted by some degree of superheat during defrosting, a better defrosting performance and less heat wastage would be achieved. Later, Qu et al. [90] employed a reverse cycle defrosting method on the basis of thermal energy storage (TES) for cascade air-source heat pumps. Using TES shortened the defrosting time by 71.4-80.5%, and reduced the energy consumption by 65.1-85.2%. Moreover, the stored thermal energy could be a heating source for indoor space as well. Song et al. [91] varied the heat provision for each refrigerant circuit to mitigate uneven defrosting of air-source heat pumps. This measure could decrease the defrosting energy consumption to 94.6% and reduce the defrosting time by 7 s. Liu et al. [92] proposed a thermal storage defrosting system for household refrigerators. In refrigerating modes, the condensation heat of the refrigerant was stored in a thermal storage heat exchanger, which worked as an evaporator in defrosting modes.

The defrosting speed was 50% less than the original model, and electric consumption for defrosting was reduced by around 71%.

Fig. 25 Schematics of a novel reverse-cycle defrosting method based on thermal energy storage [88]

4.1.3 Ultrasonic vibration

Ultrasonic vibration is another kind of active methods of defrosting. At the beginning, the vibration of low frequencies ranging from 100-200 Hz was studied [93,94]. But it was found that the low-frequency mechanical vibration had little positive effects on defrosting. Then in 2003, Adachi et al. [95] studied the frost accumulation at the surface of an aluminum alloy plate which vibrated at about 37 kHz with an environment of 100% RH and 2 ^{o}C. It was found that ultrasonic vibrations of 3.4 mm amplitude suppressed frost accumulation by approximately 60%. Li et al. [96] experimentally conducted a study of frost formation at a cold flat surface with 20 kHz ultrasound. The experimental results showed that the ultrasound had a strong capacity of suppressing both of the initial frost nucleation and frost growth, as shown in Fig. 26. Wang et al. [97] pointed out that the ultrasonic frost restraint mainly resulted from the mechanical vibrations of high frequency ultrasonic. Tan et al. [98] put forward a new defrosting method on the basis of the ultrasonic resonance. It was reported that the ultrasonic defrosting was mainly attributed to the resonance effect, and the optimal working mode was vibrating 1 min with an intermittency of 4 min. Later, Tan et al. [99] demonstrated that with the ultrasonic defrosting, the defrosting energy consumption was reduced by 3.14-5.46%, and system COP was increased by 6.51-15.33%.

To sum up, all the previous research demonstrated that the ultrasonic vibration has a positive influence on defrosting. However, the ultrasonic defrosting system also has disadvantages. For example, it consumes larger power, radiates larger electromagnetic waves, and has the potential of secondary frosting. Therefore, further study in this field is still needed before the actual application of ultrasonic vibration in defrosting.

Fig. 27 Side-view of frost formation comparison on a cold flat surface [96]

4.2 Frost-free technology

Even though the above defrosting technology could suppress frost formation, there is still frost existing on the heat exchanger surface, which reduces the thermal performance. Hence, the frost-free technology emerges, aiming to totally avoid frost formation. Currently, the frost-free technology includes the method of reversibly used cooling tower and desiccant dehumidification.

4.2.1 Reversibly used cooling tower

As mentioned above, one necessity of frost formation is that moist air should contact with the heat exchanger. If the moisture air does not directly contact the heat exchanger, there should be no frost formation on the surface of the heat exchanger. Hence, a reversibly used cooling tower with anti-freezing solution was proposed, which was mainly used for air-source heat pumps. The outdoor heat exchanger of air-source heat pumps extracts heat from the anti-freezing solution like glycol which will then extract heat from outdoor air in the reversibly used cooling tower. In the above process, the outdoor heat exchanger is not directly in contact with the outdoor air, and hence frost formation is fully avoided.

Li et al. [100] established a theoretical model to analyze the performance of a frost-free air-source heat pump. Results indicated that the new system was more efficient than the conventional one in winter, with COP improvements by 80-100%. Besides, it was not necessary for the novel system to run in the defrosting mode periodically, which improved the thermal comport of indoor space. Wen et al. [101] experimentally analyzed the effect of inlet parameters on heat transfer coefficients of a reversibly used cooling tower. Jiang et al. [102] selected glycerol as a spray solution, and experimentally investigated the influence of glycerol solutions of different concentration and mass flow rate on a frost-free air-source heat pump. Liang et al. [103] analyzed similarities and differences between the anti-freezing solution regeneration in a reversibly used cooling tower and liquid desiccant solution regeneration in a regenerator. Cui et al. [104] proposed a reversibly used cooling tower with upward spraying. In the tower, the aqueous solution was sprayed upward to decrease the drag resistance and improve the efficiency, as shown in Fig. 28.

Fig. 29 Frost-free air source heat pump with a reversibly used cooling tower [104]

4.2.2 Desiccant dehumidification

As mentioned above, another necessity of frost formation is that moist air should be cooled below the dew point temperature. When the air dew point temperature is lower than the evaporator surface temperature, there will be no condensed water on the evaporator surface and hence no frost formation. As well known, decreasing the air humidity with solid or liquid desiccants could reduce its dew point temperature. Therefore, the idea of desiccant dehumidification is that the moist air is dehumidified to achieve a lower dew point temperature before flowing through the evaporator.

Wang and Liu [105] developed an air-source heat pump with air dehumidified by a solid adsorbent before it entered the evaporator. The air humidity decreased and the air temperature increased as a result of the absorption latent heat. In this way, the frost was avoided. Zhang et al. [106] proposed a frost-free air-source heat pump water heater system with an extra heat exchanger coated by solid desiccants. In the heating mode, outdoor air first flowed through the extra heat exchanger and was dehumidified by solid desiccants, before it entered the evaporator. In the regeneration mode, condensation heat of the refrigerant was collected to heat the solid desiccant and the evaporator was heated by the air leaving the extra heat exchanger. Results showed that system COP was within the range of 3.3-3.8, which was improved by 5-30%. However, in the regeneration mode, it will cause frosting on the evaporator when the refrigerant temperature was below 0 ^{o}C. So Wang et al. [107] put forward to add an energy storage device in the above system. As present in Fig. 30, the device stored partial condensation heat of the refrigerant in the heating mode, and worked as an evaporator in the regeneration mode. It could avoid frosting on the evaporator in the heating mode and supply heating in the regeneration mode. Further in 2017, Wang et al. [108] numerically studied the system performance with R134a and R407C as the refrigerant. In addition to the air-source heat pumps, the frost-free technology with solid desiccants was also used for the refrigerated space. Zhang et al. [109] developed a frost-free refrigerated display cabinet. The frosting could be suppressed and the system efficiency was high as the solid desiccant was regenerated with the exhaust heat.

Fig. 31 Schematic diagram of a novel frost-free air-source heat pump water heater system based on solid desiccant dehumidification [107]

Besides the solid desiccant, liquid desiccant is also used to dehumidify the air to avoid frost formation. In 2010, Zhang et al. [110, 111] present a novel frost-free air source heat pump combined with a liquid desiccant system, as shown in Fig. 32. The humidity of air flowing through the evaporator was decreased by liquid desiccant so as to retard the frosting. Recently, Su and Zhang [112] dehumidified the air with a liquid-to-air membrane dehumidifier in an air-source heat pump system. Results demonstrated that system COP was improved by 64.3% compared to the conventional reverse-cycle defrosting system. And then, in 2017, Su et al. [113] applied a compression-assisted regeneration cycle to assist the liquid regeneration in the air-source heat pump system.

Fig. 33 Schematic diagram of a hybrid frost-free air conditioning system with liquid desiccant dehumidification [111]

4.3 Summary

The defrosting and frost-free technologies are essential for vapor compression refrigeration systems. Generally, the defrosting technologies could be divided into the methods of reverse cycle, surface coating and ultrasonic vibration, and the frost-free technologies include the methods of reversibly used cooling tower and desiccant dehumidification. A summary of the technologies is shown in Table 4.

Table 4 Summary of the defrosting and frost-free technologies

5. Temperature and humidity independent control (THIC) technology

In conventional air-conditioning systems, the indoor sensible load and latent load are dealt with simultaneously. The moist air temperature is firstly reduced below the dew-point temperature by the VCRS to dehumidify the moist air, and it is reheated to meet the inhabitants' comfort requirements before entering the conditioned space. The above process results in significant energy waste. Therefore, it is proposed to deal with the indoor sensible load and latent load separately [114], i.e. temperature and humidity independent control (THIC). The operating principle is presented in Fig. 34.

In the THIC system, air is dehumidified by liquid/solid desiccants and the sensible load could be handled by the VCRS at a much higher evaporating temperature, improving the system COP and save a large amount of energy. After dehumidification, the liquid/solid desiccant, in order to be used again, must be regenerated, such as by waste energy, renewable energy or electric fields.

Fig. 35 Operating principle of the temperature and humidity independent control air-conditioning system [114]

5.1 Solid desiccant air conditioning system

As shown in Fig. 36 [115], solid desiccants are mounted on a wheel which slowly rotates between the incoming air stream and reactivation air stream. Desiccant materials could be silica gel, molecular sieves, activated alumina, etc. In general, three quarters of rotation time is spent on absorbing moisture from the incoming air, and the remaining time desorbing moisture into hot reactivation air. The mass transfer between the moist air and solid desiccant is driven by the vapor pressure differences. To desorb the water from solid desiccant, the reactivation air must be heated by external heat sources. As for heat sources, the solid desiccant air conditioning system could be divided into the solar energy driven solid desiccant system and heat pump driven solid desiccant system.

Fig. 37 Rotary wheel filled with solid desiccants [115]

5.1.1 Solar energy driven solid desiccant system

Solar energy is one of the most popular heat sources for solid desiccant regeneration. As shown in Fig. 38. López et al. [116] proposed to integrate a solar desiccant air handling unit to the conventional cooling coil, resulting in 35% reduction of energy consumption. In 2014, Ge et al. [117] presented a detailed review on rotary desiccant wheel cooling systems powered by solar energy. Recently, Wang et al. [118] validated the performance of a novel solid desiccant cooling system of self-cooling, whose average moisture removal rate and system COP were about 17% and 6% higher than that of the system without self-cooling. Abbassi et al. [119] analyzed and compared the transient performance of solar desiccant cooling systems with different configurations.

Fig. 39 Solar energy driven solid desiccant system [116]

5.1.2 Heat pump driven solid desiccant system

Recovering the condensation heat of a condenser for regeneration is another popular method. Tu et al. [120] investigated the influences of the irreversible processes of the desiccant wheels and cooling sources on the performance of desiccant wheel dehumidification and cooling systems. Nie et al. [121] theoretically and experimentally studied a heat pump assisted solid desiccant cooling system, as shown in Fig. 40. Considering that the amount of condensing heat was always more than that of the required regeneration heat, another condenser was designed to take away the surplus condensing heat. In 2017, Tu et al. [122] proposed a heat pump fresh air handling unit of multi-stage with solid desiccant plates. Influences of the total thickness of desiccant plates, the total stage number and the switching period on the system performances were analyzed.

Fig. 41 Heat pump driven solid desiccant system [121]

5.2 Liquid desiccant air conditioning system

The air can also be dehumidified by liquid desiccants with high concentrations as a result of vapor pressure differences between the air and liquid desiccant. Liquid desiccants could be LiCl solution, LiBr solution, $CaCl_2$ solution, etc. Unlike the solid desiccant dehumidification, the liquid desiccant dehumidification does not have moving parts except the liquid pumps. In addition, the liquid desiccant has the property of sterilization.

After air dehumidification, the solution concentration decreases due to absorbing moisture from the process air. To reuse the solution, it must be regenerated with external driven sources. Currently, there are three main types of liquid desiccant systems in terms of driven sources: heat-pump driven, solar energy driven liquid desiccant systems and electric field driven liquid desiccant systems.

5.2.1 Heat pump driven liquid desiccant system

It is quite popular to integrate a heat pump to a liquid desiccant dehumidification/regeneration system. The condensing heat of the refrigerant is used for desiccant regeneration, and the refrigeration effect of the evaporator is used for air dehumidification and cooling. Zhang et al. [123] tried to optimize the performance of a heat pump driven liquid desiccant dehumidification system. As shown in Fig. 42, Chen et al. [124] developed a new THIC liquid desiccant air-conditioning system. To make the condensation heat satisfy the demand of desiccant solution regeneration, the air was dehumidified with liquid desiccant of low concentration and low temperature. Then they [125] conducted an experimental study of the above system. The classical NTU-Le model was modified to simulate the air dehumidification. The system transient performance was analyzed under a typical summer condition, and the system average COP was 4.0. Recently, Xie et al. [126] focused on a counter-flow heat pump driven liquid desiccant system. A multi-stage heat pump was adopted to enhance the match between the solution and refrigerant. Cai et al. [127] theoretically and experimentally studied a small-scale heat pump type air conditioner integrated with a liquid desiccant cycle. The results showed that while the evaporating temperature increased to 15 ^{o}C, the energy consumption was decreased by 22.64%, and the COP was 35.3% greater than the conventional air conditioner.

Fig. 43 Heat pump driven liquid desiccant system with low solution concentration [124]

5.2.2 Solar energy driven liquid desiccant system

Liquid desiccant can also be regenerated by heat sources of low-grade, among which solar energy is strongly recommended for the desiccant regeneration due to its sustainability. As shown in Fig. 44 [128], solar energy finds a wide application in liquid desiccant regeneration.

Fig. 45 Typical liquid desiccant regeneration driven by solar energy [128]

Cheng and Zhang [128] had extensively reviewed the research on solar energy regeneration methods for the liquid desiccant before 2013. Mohaisen and Ma [129] developed and simulated a novel solar driven liquid desiccant air-conditioning system. The simulation results showed that an average daily thermal COP of 0.5-0.55 could be achieved with the new system in Sydney. Coca-Ortegón et al. [130] studied the hybrid system of a liquid desiccant system and a conventional vapor compression chiller, as presented in Fig. 46. Dynamic simulations were conducted to investigate the seasonal performance for obtaining a suitable control strategy. In 2017, Das et al. [131] experimentally investigated a small capacity liquid desiccant air conditioning system applied to the small office. The major energy consumption for the regeneration process was from solar energy. The COP of the system varied from 0.25 and 0.8 depending on the operating conditions.

Fig. 47 Hybrid liquid desiccant system with solar energy [130]

5.2.3 Electric field driven liquid desiccant system

Besides thermal energy, liquid desiccant could also be regenerated by a direct electric field based on the electrodialysis (ED) theory. Unlike the thermal energy driven regeneration method, the electric driven method is not affected by air parameters. As shown in Fig. 48 [132], in an ED regenerator, anions will move across the anion-exchange membrane while cations will move across the cation-exchange membrane under a direct electrical field. In this way, the concentration of liquid desiccant in regenerate cells is increased, and the strong liquid desiccant can be used for air dehumidification.

Fig. 49 An electrodialysis regenerator for a liquid desiccant air conditioning system [132]

Cheng et al. [133] experimentally investigated an ED regenerator under different solution flow rate. Results showed that the largest mass concentration difference of the concentrated desiccant at the inlet and outlet of the ED regenerator was 0.03% after 30 mins of running. Guo et al. [134] studied the influences of the circulation flow rate, supplied current density and solution initial concentration on the ED regenerator. Cheng et al. [132] examined the influence of the concentration difference between the dilute and concentrated desiccant on the performance of an ED regenerator, and it showed that increasing the concentration difference was not favorable to the ED regenerator performance. Cheng and Xu [135] also developed an advanced multi-function liquid desiccant regeneration system. Results showed that the ideal COP of the above system could be above 7.

5.3 Summary

The basic idea of the THIC is to handle the sensible and latent loads separately, which saves much energy. However, this system is more complicated than the traditional air conditioning system due to the integration of dehumidification and regeneration devices, resulting in large-scale sizes and difficult operation. To promote its application, there are several efforts to be made: firstly, the performance of existing devices can be further improved, especially the regenerator; secondly, there is still a big issue in the control strategies of daily and annual operation; finally, small-scale THIC systems should be developed for residential users. When liquid desiccant is used for air dehumidification, corrosion is a serious issue which remains to be solved in the future research.

6. Ground source heat pump (GSHP) technology

Ground source heat pump (GSHP) is increasingly popular due to its potential to reduce primary energy consumption and greenhouse gas emissions. The GSHP releases heat from the building into the ground (soil) in summer and absorbs heat stored in the ground into the building in winter. Esen et al. [136] showed that the GSHP was more economical than the air source heat pump. This relies on the fact that the ground has a relatively constant temperature, cooler than the air in summer and warmer than the air in winter. In summer, the ground acts as a low-temperature cooling source, which lowers the condensing temperature of the refrigerant and hence improves the COP of the VCRS. Based on the types of ground heat exchangers in the GSHP, it could be classified as horizontal, vertical, and pile foundation ground heat exchangers.

6.1 Horizontal ground heat exchanger

Fig. 50 shows the schematic diagram of the GSHP with horizontal ground heat exchangers in the summer cooling mode. The horizontal ground heat exchangers are composed of parallel pipes (19-38 mm in diameter and 121.9-182.9 m in length) and horizontal boreholes (0.91-1.83 m in depth). In general, it is much cheaper to install the horizontal ground heat exchanger, but it will occupy much space. In addition, as it is located in a shallow soil, it is subject to a big temperature fluctuation mainly affected by the ambient conditions, such as air temperature, humidity, and wind speed.

Esen et al. [137] made experiments and economic analyses of a horizontal GSHP in Turkey. They reported that the GSHP was more economical than the traditional heating systems driven by electric resistance, fuel oil, liquid petrol gas, coal and oil, while it was not a profitable substitute of a natural gas furnace. Later, they [138] studied the performance of a solar-assisted slinky GSHP with simulation and experiment. It showed that the horizontal slinky-type ground heat exchanger was better than the vertical one.

Considering that the soil temperature strongly affected the performance of the horizontal heat exchanger, Kayaci and Demir [139] investigated the effects of burial depth, distances between pipes and surface effects on soil temperatures. Results showed that higher distances between pipes more than 2 m had no significant effects on soil temperatures and larger burial depth led to lower soil temperatures. Later, they [140] obtained the soil temperature profile to predict the performance of a horizontal heat exchanger by considering realistic boundary and operating conditions. Gan [141] considered not only heat transfer but also moisture transfer in soil, and developed a coupled heat and moisture transfer model to generate moisture and temperature profiles in soil for accurately predicting the seasonal thermal performance of a horizontal ground heat exchanger. Results showed that the maximum difference between models with and without moisture transfer for the prediction of heat transfer through a heat exchanger was 24% in clay sand soils.

Fig. 51 Ground-source heat pump with horizontal ground heat exchangers for cooling.

6.2 Vertical ground heat exchanger

Fig. 52 shows the schematic diagram of the GSHP with vertical ground heat exchangers (vertical GSHP) in the summer cooling mode. The vertical ground heat exchangers are composed of boreholes (30-120 m in depth and 76-127 mm in diameter) backfilled with a material that prevents contamination of ground water, and U-shaped pipes (19-38 mm in diameter) through which the heat transfer fluid flows. One of the difficulties for designing the vertical GCHP lies in the appropriate depth of boreholes.

Extensive studies have been made on the vertical GSHP. Shang et al. [142] conducted an intermittent experiment of the vertical GSHP, and found that the insufficient soil recovery time could result in the rapid decline of system performance. Hence, the optimum intermittent time was suggested. Zhang et al. [143] tested the performance of the vertical GSHP in the hot summer and cold winder area in China. It was found that a high cooling load contributed to a high COP. Zhai et al. [144] designed and installed a minitype vertical GSHP in Shanghai, China. The test results showed that the ground temperature was ~6 ^{o}C lower than the air temperature in summer and COP of the GSHP was 14.3% higher than that of the traditional VCRS. Lu et al. [145] made economic analyses on the feasibility of using vertical GSHP in Melbourne, Australia. It was found that the vertical GSHP provided considerably more savings than other alternative air conditioning systems for a design life of 40 years.

Considering that the borehole drilling cost is significant, measures have been taken to decrease the borehole depth. Zhou et al. [146] compared single- and double-U type vertical ground heat exchangers, and concluded that the double-U type had a higher heat transfer flux per unit borehole depth than the single-U type. Saeidi et al. [147] proposed to use a spiral tube with aluminum fins for enhancing heat transfer and decreasing the borehole depth. 3D numerical modeling showed that spiral type tubes were the best choice. Narei et al. [148] used Al_2O_3/water nanofluid instead of water as heat transfer fluid to reduce the borehole depth by 1.3% for a vertical ground heat exchanger.

Fig. 53 Ground-source heat pump with vertical ground heat exchangers for cooling.

6.3 Pile foundation ground heat exchanger

It is costly and challenging to lay heat exchangers underground, which makes using the pile foundation as a ground heat exchanger more attractive. The pile foundation ground heat exchanger emerged at the end of last century [149]. It is a built-in heat exchanger which makes full use of pile foundation inner space, and can obtain more heat exchange amount per unit length than the traditional ground heat exchanger. Fadejev et al. [150] reported pile foundation applications and their potential as a renewable energy solution; properly sized GSHP with pile foundation ground heat exchangers (pile foundation GSHP) could achieve a seasonal COP higher than 4.5. The schematic diagram of the pile foundation GSHP is shown in Fig. 54 for the summer cooling mode.

The configuration of the pile foundation ground heat exchanger can be classified as U-type and spiral-type. Zarrella et al. [151] investigated a thermal response test on a pile foundation ground heat exchanger (20 m in depth) with a double U-tube circuit to determine the ground's thermal conductivity. Park et al. [152] carried out a series of in-situ thermal performance tests on a large-diameter (1.5 m) pile foundation ground heat exchanger by controlling intermittent cooling and heating loads, which can provide sufficient heat exchange compared with horizontal and vertical types. Cui and Zhu [153] studied the year-round performance of the pile foundation GSHP with a pile depth of 10 m based on a 3D transient heat transfer model. The maximum cooling COP was 4.73 in the first year operation period, which decreased by ~8% in the tenth year.

Some researchers found that the U-type pile foundation ground heat exchanger had a lower thermal performance than the conventional vertical ground heat exchanger. This is because the pile foundation is usually 20-50 m deep while the vertical borehole could be as deep as 200 m [154]. Considering that the pile diameter is obviously larger than that of vertical boreholes, it is feasible to adopt spiral-type pipes rather than U-type pipes to increase the heat transfer area and enhance the heat exchange capacity [155].

When the pipes are configured in the form of spiral coils, three models are usually used for simulation: cylindrical [156], ring-coil [157] and spiral-coil [158] heat source models. The cylindrical heat source model treats the spiral coils as a cylindrical surface without considering the pitches of spiral coils, and the medium inside the cylinder is assumed to be the same as the outside. The ring-coil heat source model is an improved model compared with the cylindrical heat source model, which considers the spiral coils as a series of discrete ring heat sources without any connection between adjacent coils. The spiral-coil heat source model is more accurate and realistic, where a real spiral-coil replaces the ring-coil. Zhou et al. [159] investigated the temperature responses of the cylindrical and ring-coil heat source models and established analytical solutions using the Green's function method. Lei et al. [160] proposed a decomposition algorithm to evaluate the heat conduction in a particular direction, considering that the pile had thermal properties different from the soil. Yang et al. [161] performed an experimental study on the effect of working parameters on the pile foundation GSHP, including the inlet fluid temperature, intermittent operation modes, spiral pitch and pile material.

Fig. 55 Ground-source heat pump with pile foundation ground heat exchangers for cooling.

6.4 Summary

Ground source heat pump (GSHP) is a great alternative of traditional air conditioning systems as it could make good use of the geothermal energy. The biggest obstacles of its promotion consist of the high drilling cost and necessary space for installation.

Many attempts have been made to solve the above issues, mainly focused on the optimization of ground heat exchangers (horizontal, vertical and pile foundation types). The horizontal ground heat exchanger is much cheaper since it is installed in the shallow soil (1-2 m deep), but it will occupy much space and is subject to a big temperature fluctuation. The vertical ground heat exchanger occupies less space and achieves more stable temperatures and a higher COP. But the drilling cost is much higher for a borehole at the depth of ~100 m. What’s more, it is difficult to design a suitable borehole depth, which requires more research. The pile foundation ground heat exchanger is more promising, which can effectively reduce the drilling cost and save the space as well. However, the current research is mainly based on simulation, more experimental work is needed to validate its applicability.

In addition, the future research should also evaluate the effect of GSHP on the earth’s environment (soil), which is not considered by all the current research.

7. Refrigerant subcooling technology

It is often assumed the condition of the refrigerant at the inlet of the expansion device in the VCRS is saturated liquid. However, cooling the refrigerant liquid below saturation could enhance the refrigerating effect so as to increase the COP. What’s more, liquid subcooling reduces throttling losses in terms of the second law of thermodynamics. Through investigation, refrigerant subcooling is generally achieved by liquid-suction/internal heat exchanger, mechanical subcooling, ice storage subcooling and liquid desiccant dehumidification subcooling.

7.1 Liquid-suction/internal heat exchanger

In the traditional VCRS, a liquid-suction/internal heat exchanger is usually added between the outlets of the condenser and evaporator, as shown in Fig. 56 [162]. In this way, the liquid refrigerant can be subcooled to avoid the formation of flash gas at the inlets of expansion valves, and all residual liquid can evaporate before entering the compressor so as to improve the system performance.

Fig. 57 Schematic of the vapor compression refrigeration system with a liquid-suction/internal heat exchanger [162]

Early before the century, thermodynamic performance of liquid-suction heat exchangers had been investigated [163]. In 2000, Klein et al. [164] analyzed the effect of pressure drops of heat exchanger on performance of the system with different refrigerants. Then, in 2001, through experimental study, Boewe et al. [165] reported that the internal heat exchanger could enhance the cycle efficiency of a prototype transcritical mobile VCRS using R-744 with 25%. A simulation model was then used to optimize the design of the internal heat exchanger. Masrullo et al. [166] made a simple chart to identify the effectiveness of installing liquid-suction heat exchangers. Hermes [162] found that the introduction of internal heat exchanger would increase or decrease the COP of the VCRS, depending on several factors. Later he [167] reported the refrigerant charge could be reduced by using the liquid-suction heat exchanger. Navarro-Esbrí et al. [168-169] found the presence of the internal heat exchanger had a positive effect on the system COP when R1234yf, R1234ze(E) and R450A were used instead of R134a. Pottker and Hrnjak [170] experimentally studied the effect of condenser subcooling on the performance of VCRS, under the same operating conditions.

Some researchers chose CO_2 as refrigerant and presented detailed analyses of the internal heat exchanger effect on the system performance. In 2005, Chen and Gu [171] developed a relationship of optimum high pressure for the CO_2 trans-critical VCRS with internal heat exchangers. Zhang et al. [172] conducted a detailed analysis of the impact of the internal heat exchanger on CO_2 inverse cycles. Torrella et al. [173] experimentally proved that both the efficiency and cooling capacity of the VCRS would be increased by the use of the internal heat exchanger. And the improvement depended on the evaporating temperature, gas-cooler refrigerant outlet temperature and gas-cooler pressure. Besides, internal heat exchanger did not have too much effect on the compressor power consumption. Llopis et al. [174] presented an assessment on a CO_2 subcritical VCRS with internal heat exchangers, which was found to improve the system energy performance.

7.2 Mechanical subcooling

It is well-known that the liquid-suction/internal heat exchanger achieves subcooling by transferring the refrigerant heat from the condenser outlet to the compressor inlet. The larger superheated degree at the compressor inlet results in lower refrigerant flow rate, which will reduce the system performance. Therefore, a liquid-suction/internal heat exchanger is not applicable for all VCRS.

In order to eliminate the drawbacks of the liquid-suction/internal heat exchanger, some researchers proposed a mechanical subcooling method to enhance the system COP and decrease the superheated degree as well. As shown in Fig. 58 [175], in the mechanical subcooling method, a small VCRS is coupled to the main VCRS at the exit of the condenser to achieve the subcooling of the main VCRS. The analysis had been carried out for the mechanical subcooling system in terms of first-law (energy-balance) and second-law (irreversibility) of thermodynamics. It demonstrates that the system coefficient of performance (COP) would be improved with a decrease in the temperature difference between the condenser and evaporator, and COP could be significantly improved by reducing the irreversibility of the expansion process.

Khan and Zubair [176] developed thermodynamic models of an integrated mechanical subcooling system to simulate the actual performance of the subcooling system. Compared with the simple system without subcooling, the performance of the overall system was improved and the improvements depended on the refrigerant saturation temperature of the subcooler. Yang and Zhang [177] presented a model-based comprehensive analysis on the subcooler design and discussed the optimal subcooling control. It revealed the criteria of selecting proper subcooling to size the subcooler at lower cost. Qureshi and Zubair [178] investigated the impact of fouling on performance of the VCRS with integrated mechanical subcooling for various applications. It was also found the performance degradation due to fouling of the evaporator always had a larger effect on cooling load capacity, while the performance degradation of the condenser always had an overall larger effect on the sub-cooler compressor power requirement as well as the COP of the system. Qureshi and Zubair [179] also studied the effect of refrigerant combinations on performance of the VCRS with dedicated mechanical sub-cooling. For scratch designs, R134a used in both cycles produced the best results in terms of COP, COP gain and relative compressor sizing. In retrofit cases, it seemed that dedicated mechanical sub-cooling may be more suited to cycles using R134a as the main cycle refrigerant. Qureshi and Zubair [180] gave thermoeconomic considerations to heat exchanger inventory allocation in the VCRS with mechanical subcooling. It was found that no minima exist for any of the cost functions, and the cost optimization of the integrated mechanical subcooling system was qualitatively the same as the dedicated subcooling system. Qureshi et al. [181] made an experimental investigation into the effects of employing a dedicated mechanical subcooling cycle with the VCRS. In the experiment, R22 was employed as the refrigerant in the main cycle and R12 in the dedicated subcooling cycle. The experimental outcomes indicated that the second-law efficiency of the cycle increased by an average of 21%. Xing et al. [182] proposed a novel VCRS with mechanical subcooling using an ejector. The results demonstrated that the capacity was improved by 11.7% with R404A and 7.2% with R290, and the COP improvements were achieved by 9.5% with R404A and 7.0% with R290. Llopis [183] theoretically analyzed the possibilities of enhancing the energy performance of CO_2 transcritical refrigeration systems using a dedicated mechanical subcooling cycle. It was observed that the cycle combination allowed increasing the COP up to a maximum of 20% and the cooling capacity up to a maximum of 28.8%. Llopis [184] did the experiment to show the energy improvements of the use of a mechanical subcooling cycle in combination with a CO_2 transcritical refrigeration plant. The maximum increments of capacity and COP were 55.7% and 30.3%, respectively.

Fig. 59 Schematic of a vapor compression refrigeration cycle with dedicated mechanical sub-cooling [175]

7.3 Ice storage subcooling

As the mechanical subcooling refrigeration system requires one more compressor for subcooling, it consumes more electricity so as to influence the COP of the system. Besides, it has higher initial cost. Considering that, some researchers utilized cold storage units with energy storage materials as subcoolers [185]. The extra cold energy would be stored if the cooling load is lower than the cooling capacity of the system, while some cold energy would be released to subcool the condenser outlet refrigerant if the cooling load is larger than the cooling capacity.

As shown in Fig. 31, Huang et al. [185] investigated a cold storage air conditioning system with a thermal battery as a subcooler experimentally. To transform water into ice, two expansion devices had to be installed. One was at the air side and the other was at the ice storage side. The expansion device of ice storage side generated larger pressure drop and therefore reduced the system COP. To eliminate the disadvantages discussed above, Hsiao et al. [186] introduced another cold storage unit of one expansion device. Hsiao et al. [187] reported that the heat pump

COP with an ice storage subcooler was 12% and 15% greater than that without the ice storage tank under the charge mode and the discharge mode, respectively.

Fig. 31 Schematic of subcooled ice storage air-conditioning system [185]

7.4 Liquid desiccant dehumidification subcooling

Liquid desiccant system could use low-grade heat, including solar energy, geothermal energy and waste heat, to dehumidify the air. In 2014, She et al. [188] proposed a liquid desiccant dehumidification subcooling method combined with the vapor compression refrigeration system. As shown in Fig. 602, it includes a vapor compression refrigeration cycle, liquid desiccant cycle and an evaporative cooling cycle. The condensing heat (40-60 ^{o}C) is used to drive the liquid desiccant cycle to generate dry air in the dehumidifier; then the dry air absorbs water vapor from spray water in the indirect evaporative cooler to generate cooling water of low-temperature. The cooling water is used to subcool the refrigerant at the outlet of the condenser. Thermodynamic studies present that the maximum COP of the hybrid system was 18.8% higher than that of the conventional system. System optimizations were also conducted, including the optimum solution concentration [189], effective use of condensing heat [190] and air flow patterns [191].

Fig. 612 Schematic diagram of the liquid desiccant dehumidification subcooling method for vapor compression refrigeration systems [188]

7.5 Summary

Through investigation, it is found, among the four different subcooling technologies, the one with liquid-suction/internal heat exchangers has the longest history and therefore the most literature. Whether the installation of liquid-suction/internal heat exchangers will increase the system COP depends on lots of factors, such as the refrigerants, the working pressure, the heat exchanger effectiveness, etc. The biggest disadvantage of the technology is that it increases the superheated degree of the refrigerant at the compressor inlet so as to decrease the refrigerant flow rate. In the above context, the mechanical subcooling technology is proposed. It eliminates the drawbacks of the liquid-suction/internal heat exchanger by coupling a small refrigeration cycle to the main refrigeration cycle at the outlet of the condenser. However, the additional compressor means higher initial cost and more electricity consumption.

The ice storage subcooling technology is invented to store extra cold energy when the cooling load is lower and release the cold energy to subcool the refrigerant while the cooling load is larger. Currently, there is no much literature on this technology. The reason may be due to the complexity of this technology, such as frequent switches between cold storage and subcooling. Future work regarding this technology includes the optimum design of the cold storage unit, flexible control strategy, and various energy storage materials besides water.

The liquid desiccant dehumidification subcooling technology is proposed recently by using low-grade heat (40-60 ^{o}C) to produce cooling water to subcool the refrigerant. It broadens the temperature range of low-grade heat use for refrigeration, which is generally above 60 ^{o}C. However, the hybrid system size is much larger than the traditional vapor-compression refrigeration system, which is a disadvantage. In addition, the additional pumps and fans also reduce system performance. Current research is far from enough. Further work on this technology includes economic assessment, experimental optimization of system configuration, and integration with external waste heat.

8. Condensing heat recovery technology

Air conditioning system generates a lot of condensing heat, which is around 1.25 times of the refrigeration effect. It is meaningful to recover all or part of the condensing heat for possible use. Usually, the condensing heat could be recovered to provide domestic hot water directly or indirectly, and to achieve liquid desiccant regeneration.

8.1 Domestic hot water

In summer, most buildings should provide both of air cooling and water heating. It is conventionally achieved by two independent systems. The air conditioning system provides air cooling and the hot water supplying system offers hot water, respectively. The condensing heat is discharged into the surroundings directly. Meanwhile daily hot water are needed in many buildings, which are generally gained with an individual boiler driven by electric or fuel gas. It is known that the cooling water outlet temperature is between 35 and 40 ^{o}C for air conditioning systems. And the temperature could satisfy the requirement for washing or bathing in summer. Thus it is feasible to collect the condensing heat to meet the demand for daily hot water so as to save energy. Depending on if there is storage of thermal energy, the condensing heat could be recovered directly or indirectly to provide domestic hot water.

8.1.1 Direct domestic hot water

As shown in Fig. 33 [192], in a direct domestic hot water system, the heat recovery systems directly heat water to users. Early in 1965, Healy and Wetherington [193] initially showed that the condensing heat could potentially generate hot water. Since then, many researches had done the investigation in the field. Their experimental and numerical results showed that not only the domestic hot water could be provided with recovering about 20% of the total discharged condensing heat but also the COP of the system increased.

Fig. 33 Schematic of a vapor-compression refrigeration cycle with direct heat recovery [192]

Ji et al. [194, 195] demonstrated that the energy performance of a split-type air-conditioner could be improved by installing a water heater in the outdoor unit to collect the condensing heat as space cooling and water heating can be achieved simultaneously. Willy [196] theoretically and experimentally investigated a combination of an air conditioning system and a tap water heating plant. The system fitted to regions of year-around cooling requirement. Wang et al. [197] studied a split air conditioner integrated with an energy storage unit and a water heater. The storage tank was specially designed to regulate the capacity of the storage coils. It was found that the average cooling capacity and COP of the new developed system was 28.2% and 21.5% greater than the original air conditioner, respectively. Jiang et al. [198] experimentally studied an air conditioner with domestic hot water supply. The results showed that system COP can be improved by about 38.6% compared with the traditional vapor compression system and electric water heater. Abu-Mulaweh [199] demonstrated that the effect of thermosiphon can be used to recover the heat released by an air conditioner. Yi and Lee [200] studied a domestic water-cooled air-conditioner with a heat exchanger to preheat the domestic hot water. It was found that the heat recovery coefficient ranged between 0.542 and 0.654, and the maximum comprehensive COP of the heat recovery water-cooled air-conditioner was 4.92. Chen and Lee [201] conducted questionnaire surveys and site measurements to evaluate the hybrid space cooling and hot water preheating system. Jiang et al. [202] investigated the impact of recovering condensing heat on the dynamic performance of air conditioners. Gong et al. [192] conducted a thermodynamic simulation study for a single stage centrifugal chiller with a heat exchanger between compressor outlet and the condenser to supply hot water in a hotel located in South China.

8.1.2 Indirect domestic hot water

The mismatch exists between air conditioning and hot water usage, as the hot water is most required in the evening, while the maximum cooling load occurs around noon. To solve the problem, thermal energy storage is proposed. As phase change materials (PCMs) can store large amount of energy, it is a good way to store condensing heat through PCMs. As shown in Fig. 34 [203], unlike the direct domestic hot water system, some heat accumulators are installed to store the condensing heat for regulating the demand and supply of hot water.

Gu et al. [203] studied a heat recovery system with PCMs to recover the condensing heat for producing hot water of low temperature. They also investigated the thermal properties of paraffin wax and the mixtures of paraffin wax. Using paraffin wax as PCMs, Zhang et al. [204] conducted a charging and discharging experiment to compare air conditioning with and without the thermal storage container. Recently, in 2016, Xia et al. [205] designed a cold storage for condensing heat recovery using PCMs. They developed some novel materials which had larger latent heat, higher thermal conductivity, and a more appropriate phase change temperature.

Fig. 34 Schematic of air conditioning systems with indirect condensing heat recovery [203]

8.2 Liquid desiccant regeneration

In a liquid desiccant air conditioning system, weak desiccant regeneration could be achieved by the heat sources of low-grade. Therefore, it was proposed to utilize the condensing heat to regenerate the liquid desiccants. Depending on the heated objects, there are four ways to use condensing heat for regenerating the desiccant.

In the first type, the condensing heat is put into the regenerator directly, where the solution and regeneration air are heated at the same time. Yadav [206] conducted studies of a hybrid solar space-conditioning system composed of conventional vapor compression and liquid desiccant units. As shown in Fig. 35, Yamaguchi et al. [207] developed a hybrid system whose main feature was the dehumidifier and regenerator were integrated with the evaporator and condenser respectively.

Fig. 35 Schematic diagram of hybrid liquid desiccant air-conditioning system [207]

In the second type, one part of condensing heat is to heat the liquid desiccant, and the other part is to preheat the regeneration air [188, 189, 208]. Lazzarin and Castellotti [208] investigated a self-regenerating liquid desiccant cooling system, where the liquid desiccant was heated in a primary condenser and the regeneration air heated in a secondary heat pump condenser. Recently, She et al. [188] put forward a hybrid liquid desiccant refrigeration system, where the heat of the condensing process were collected for heating the weak solution and regeneration air. Later, optimum solution concentration to match different condensing temperatures (45-55 ^{o}C) was studied [189].

In the third type, all the condensing heat is used to heat the regeneration air, and then the heated air is conveyed to the regenerator for desiccant regeneration. Dai et al [209] developed a hybrid air conditioning system which was composed of a desiccant dehumidification unit, evaporative cooling unit and a vapor compression air conditioning unit. An air-cooled condenser was used to remove the condensing heat and the air after the condenser was used in the regenerator for desiccant regeneration. A special designed regenerator, using solar energy or waste heat, was backup in case the condensing heat was not enough for regeneration. The experimental results demonstrated that cooling capacity and COP of the new hybrid system could be much greater than those of the single vapor compression system. Mohan et al. [210] carried out a theoretical study of a novel hybrid liquid desiccant vapor compression air conditioning system. Later in 2015, Mohan et al. [211] fabricated and tested the above system. In this system, as presented in Fig. 36, the regenerator was placed after the condenser and the ambient air was heated by the condensing heat (40-50 ^{o}C) before entering the regenerator.

Fig. 36 Block diagram of the proposed system [211]

In the final type, all of the condensing heat is used for heating and regenerating the solution. Zhu et al. [212] presented a liquid desiccant heat pump air-conditioning system. The liquid desiccant was heated by the condensing heat before entering the regeneration unit. A performance investigation was performed in summer in an ecological building of Shanghai Research Institute of Building Sciences. Zhao et al. [213] implemented liquid desiccant fresh air handling units driven by heat pumps (condensing temperature around 51 ^{o}C) in an office building in Shenzhen of China. Experimental results demonstrated that the system could keep a comfortable indoor environment and the COP of the entire system reached 4.0. She et al. [190] discussed different use of condensing heat to drive the liquid desiccant regeneration in a hybrid refrigeration system. Results showed that all the condensing heat to heat the solution was the best for system performance. Later in 2016, optimizations of air flow patterns in the evaporative cooling and dehumidification process were conducted to achieve better system performance with variable ambient air conditions [191]. Recently Su and Zhang [214] proposed a hybrid compression-absorption refrigeration coupled with a liquid desiccant air-conditioning system, where condensing heat (44-77 ^{o}C) was used to regenerate the liquid desiccant solution. All the condensing heat can be recovered to regenerate the diluted liquid desiccant solution. Meanwhile the performance of the absorption refrigeration system was improved as the evaporation temperature in the absorption refrigeration system increased.

8.3 Summary

Air conditioning system generates a lot of condensing heat, direct discharge of which into the environment not only wastes energy but also harms the environment. Therefore, it is recommended to recover it for potential uses. It finds the initial application in producing domestic hot water directly for residential buildings. Through integrating a water heater with the condenser of an air-conditioner, space cooling and water heating can be achieved simultaneously with system energy improvement. However, mismatch exists between air conditioning and hot water demand. To solve the problem, thermal energy storage, i.e. indirect domestic system, was proposed. On the other hand, with the development of the material science, PCMs are verified to be good materials for condensing heat storage. More research should be conducted on the indirect system with PCMs as the energy storage media in the future research.

In the liquid desiccant system, weak liquid desiccant can be regenerated at a relatively low temperature. Hence, it is a good method to utilize the condensing heat to regenerate the liquid desiccants. The hybrid of conventional vapor-compression refrigeration and liquid desiccant cycles, not only recovers the waste heat effectively but also improves the system COP. Depending on the heated objects, there are four methods to use the condensing heat for liquid desiccant regeneration: (a) regeneration solution and air are heated simultaneously by the condensing heat; (b) all the condensing heat is chosen to heat regeneration air; (c) part of the condensing heat is used to heat regeneration air, while the remaining is for regeneration solution; (d) all the condensing heat is allocated to regeneration solution.

9. Conclusions

To meet the challenges of rapidly increasing cooling demand, it is crucial to improve the efficiency of vapor compression refrigeration systems (VCRS) and hence reduce the energy consumption caused by the rapid increase of cooling demand. In this paper, seven energy-efficient and -economic technologies for the VCRS are reviewed, including the radiative cooling, cold energy storage, defrosting and frost-free, temperature and humidity independent control (THIC), ground source heat pump (GSHP), refrigerant subcooling, and condensing heat recovery. Main findings for each technology could be summarized as follows,

- Radiative cooling is a free cooling technology taking advantage of the Earth’s transparent window. Combined with the radiative cooling, the electricity consumption of the VCRS could be decreased by ~21%. Nighttime radiative cooling can only provide cooling at night due to the limitation of general materials. However, cooling demand usually occurs in the daytime, thus cold energy storage could be integrated with the nighttime radiative cooling in the future research. Daytime radiative cooling is recently emerged with the invention of nano-photonic materials and meta-materials. The future work includes the design of new devices/systems with the integration of daytime radiative cooling, reducing its self-energy consumption by pumps, and collecting the reflected sunlight for hot water supply.
- Cold energy storage stores cold energy at off-peak times and releases it at peak times. With cold energy storage, the electricity consumption and operation cost of the VCRS could be reduced by ~12% and ~32%, respectively. Phase change materials (PCMs) are the most promising among energy storage media, due to their advantages such as large energy storage density and stable operating temperatures. However, PCMs have a low thermal conductivity, which leads to a slow phase change process. Future work includes enhancing the thermal conductivity without phase segregation and designing heat exchangers with high heat transfer rates. In addition, for the salt-based PCMs, there is a large degree of subcooling and a corrosion issue, which remain to be solved in the future research.
- With the defrosting and frost-free technologies, electricity consumption for defrosting in the VCRS could be saved more than 60%. Surface coating is the most promising defrosting method. However, the coating strength should be further improved with simple preparation methods for a long life-cycle. What’s more, the defrosting mechanism is controversial and remains to be further clarified in the future. The frost-free technologies include the reversibly used cooling tower and desiccant dehumidification. The main obstacle of the two technologies for widely application is the large system size, which results from the poor heat and mass transfer between the air and solution in the packing tower. Therefore, the future

work should include the enhancement of heat and mass transfer in the packing tower to decrease the system size.

- The THIC technology dehumidifies the process air with liquid/solid desiccants and then cools it down by the VCRS, which makes the VCRS work at a much higher evaporating temperature and thus significantly improves the COP of the VCRS by ~35%. One of the main disadvantages is that, after air dehumidification, the liquid/solid desiccants, must be regenerated with external heat sources. Therefore, the future work needs to be focused on reducing the heat source temperature through enhancing the heat and mass transfer between air and desiccants. In addition, the THIC system is more complicated than the traditional air conditioning system. There are several efforts to be made, including improving the performance of the components, minimizing its size and developing mature control strategies. What's more, for the liquid desiccant, there is a serious issue of corrosion, which is urgently required to be solved in the future research.
- The GSHP uses the ground as a low temperature cooling source in summer, which decreases the condensing temperature of the refrigerant and thus increases the COP of the VCRS by ~14%. The main challenges for the GSHP include the drilling cost and occupied space. The pile foundation ground heat exchanger is more promising compared with the horizontal and vertical types, which can reduce the drilling cost and save the space. The current research is mainly based on simulation, and more experimental work needs to be done for fully validation. In addition, the impact of the GSHP on the ecology should also be investigated in the future research.
- Among the refrigerant subcooling technologies, the ice storage subcooling method is promising which stores excess cold energy of the VCRS at part loads, and then releases it to subcool the refrigerant when the VCRS is at full loads. However, this system usually works at two evaporating temperatures and is more complicated to control. Future work includes the optimum design of the cold storage unit, flexible control strategy, and development of PCMs whose phase change temperatures match well with the evaporating temperatures. The liquid desiccant dehumidification subcooling method is recently proposed. The main drawback is the large system size, and further work includes experimental optimization of the system configuration, performance evaluation of annual operation, and economic assessment.
- The condensing heat of the VCRS could be recovered to provide domestic hot water for buildings, which could improve the system COP by ~38.6%. However, most of the research is focused on direct hot water supply. Considering that the hot water is usually in demand at night while the cooling is required in the daytime, future work should pay more attention to the integration of heat energy storage with the VCRS, including the development of high performance PCMs and heat exchangers. It has been widely studied to use the condensing heat for liquid desiccant regeneration. However, the common problems related to the liquid desiccant include corrosion problems and poor heat and mass transfer, which are required to be solved in the future research.

These findings will give a big picture of the advanced technologies that could be applied in the VCRS to achieve energy-efficient and economic cooling. The readers can also get a clear vision for the future development of each technology. This paper gives necessary guidance to the industry for developing new refrigeration devices.

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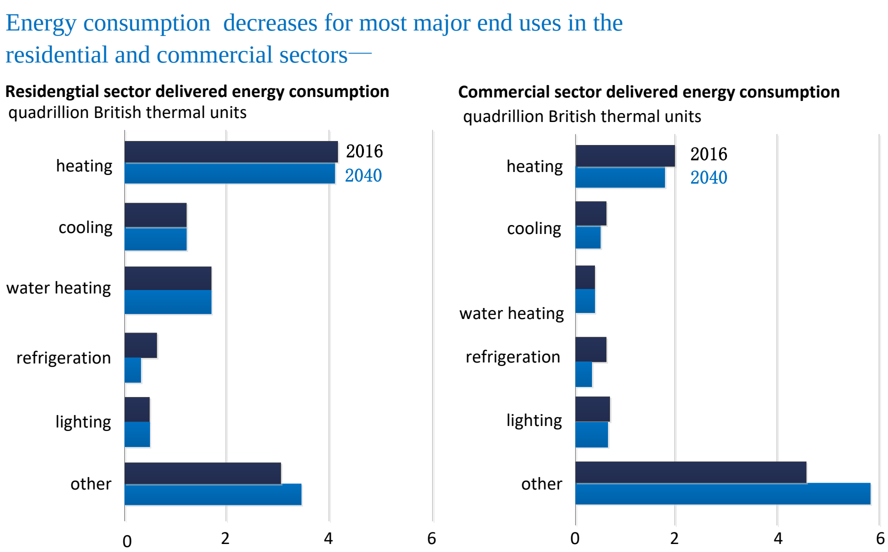


Fig. 1 Residential and commercial sectors delivered energy consumption [2]

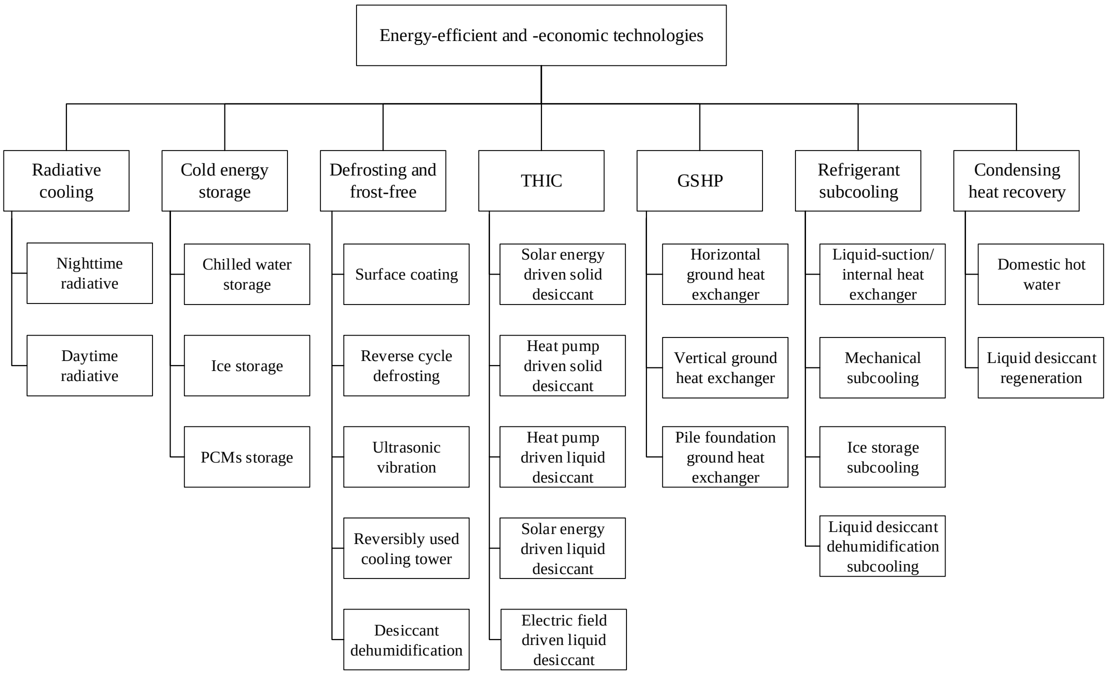


Fig. 2 Advanced technologies for improving the performance of vapor compression refrigeration systems.

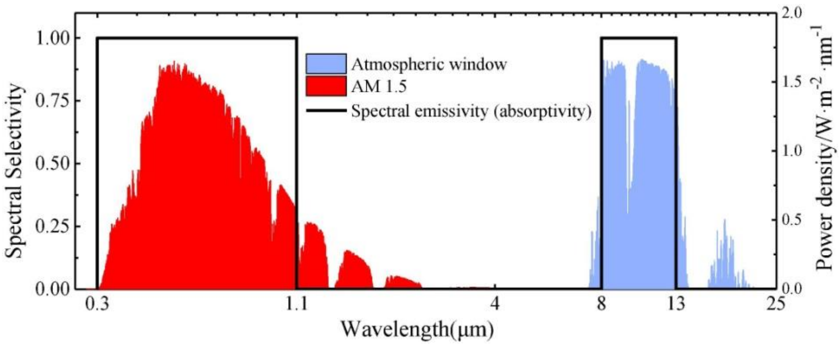


Fig. 3 Idealized spectral emissivity (absorptivity) of the selective materials from ultraviolet to mid-infrared [13]

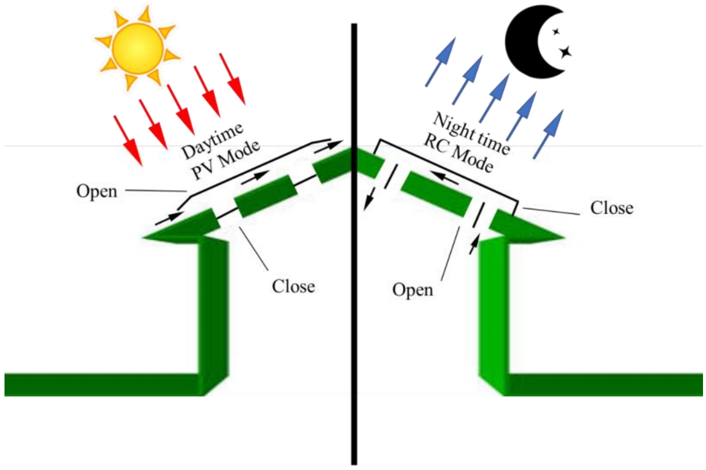


Fig. 4 Schematic of the building integrated photovoltaic-radiative cooling system [13]

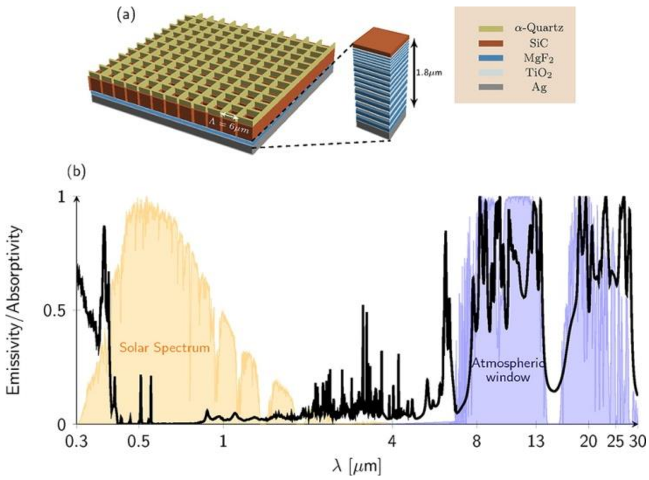


Fig. 5 The metal-dielectric photonic structure for daytime radiative cooling [11]

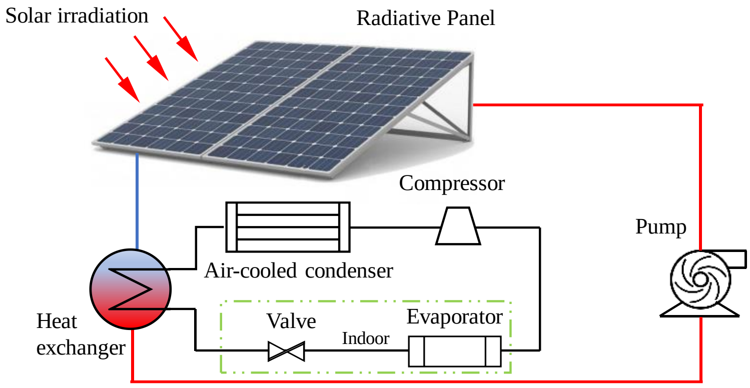


Fig. 6 A radiative-cooled vapor compression refrigeration system [1]

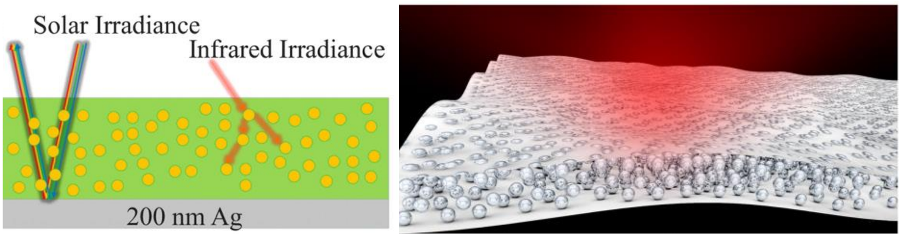


Fig. 7 A schematic of the polymer-based hybrid metamaterial with randomly distributed SiO_2 microspheres for large-scale radiative cooling [19]

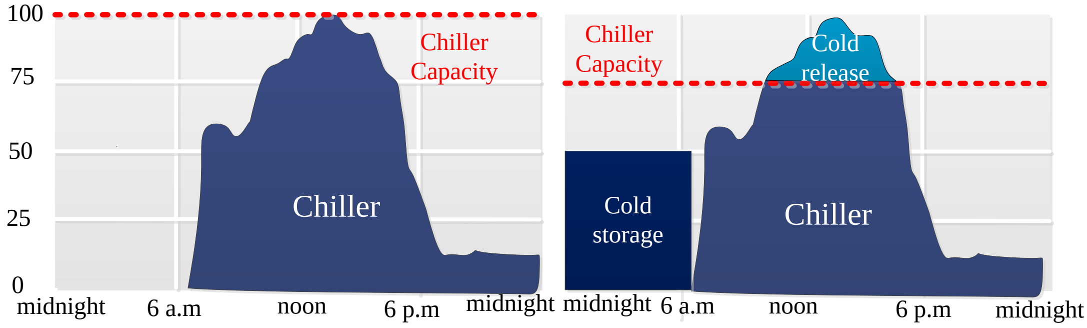


Fig. 8 Schematic diagram of the cold energy storage

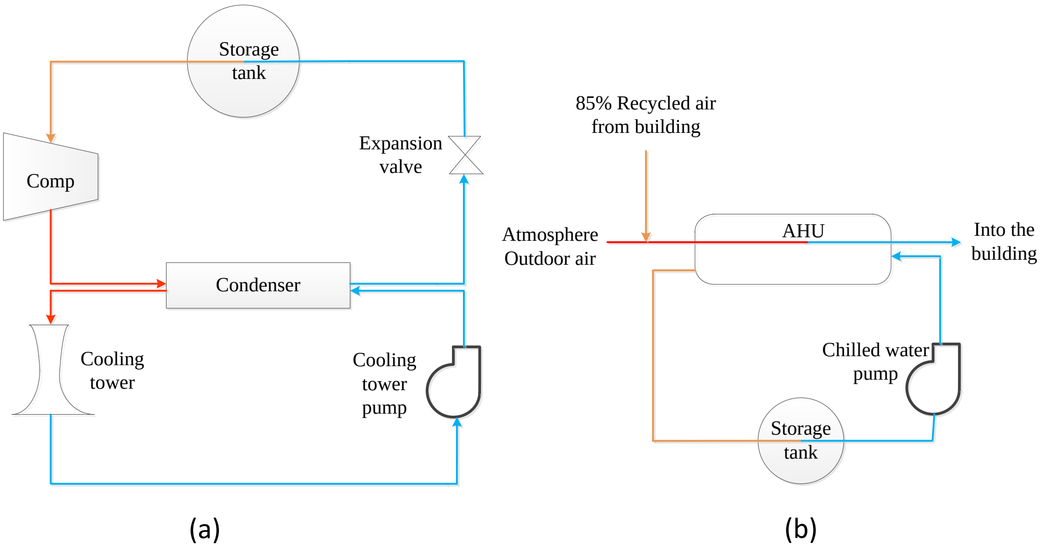


Fig. 9 The schematic diagram of the ice storage system for full operating mode: (a) discharging cycle; (b) charging cycle [46]

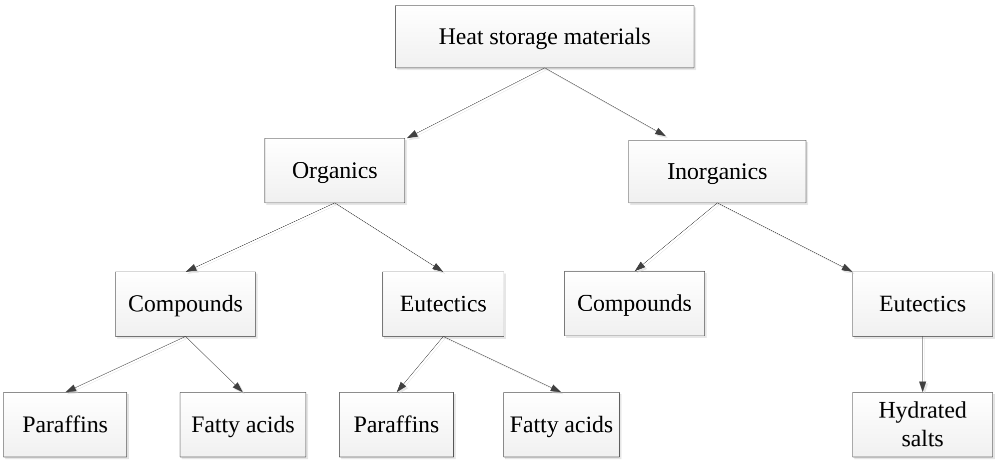
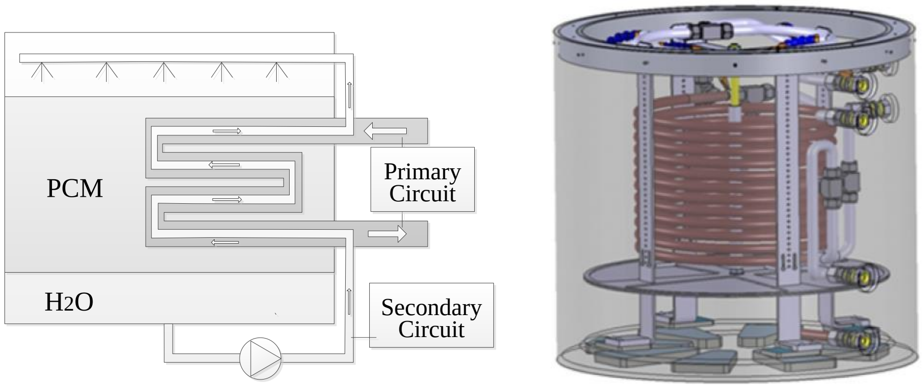


Fig. 10 Families of phase change heat storage materials [54]



(a) (b)

Fig. 11 PCMs cold storage tank layout (a) and 3D rendering (b) [66]

(a) Hydrophilic fin (b) Bare fin (c) Super hydrophobic fin

Fig. 12 Molten water retention on three types of fins [80]

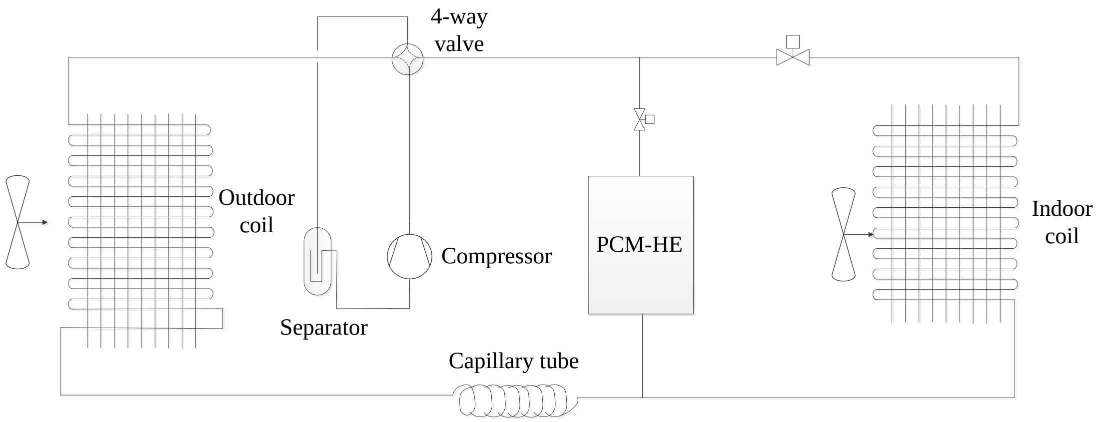
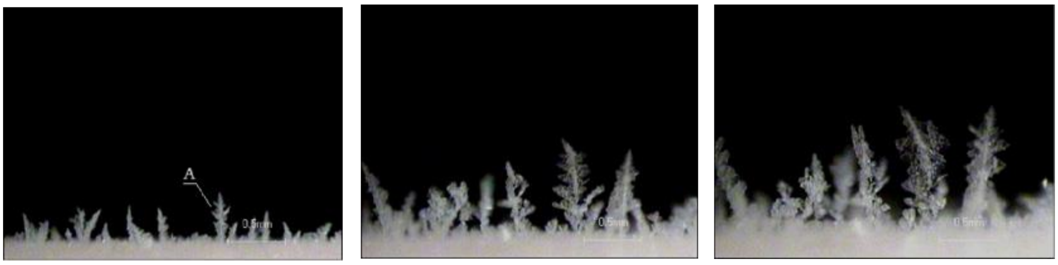
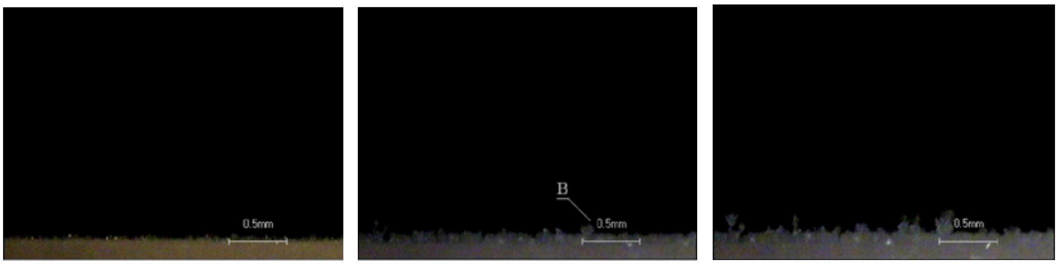


Fig. 13 Schematics of a novel reverse-cycle defrosting method based on thermal energy storage [88]



270 s 420 s 600 s

(a) without ultrasound



270 s 420 s 600 s

(b) with ultrasound

Fig. 24 Side-view of frost formation comparison on a cold flat surface [96]

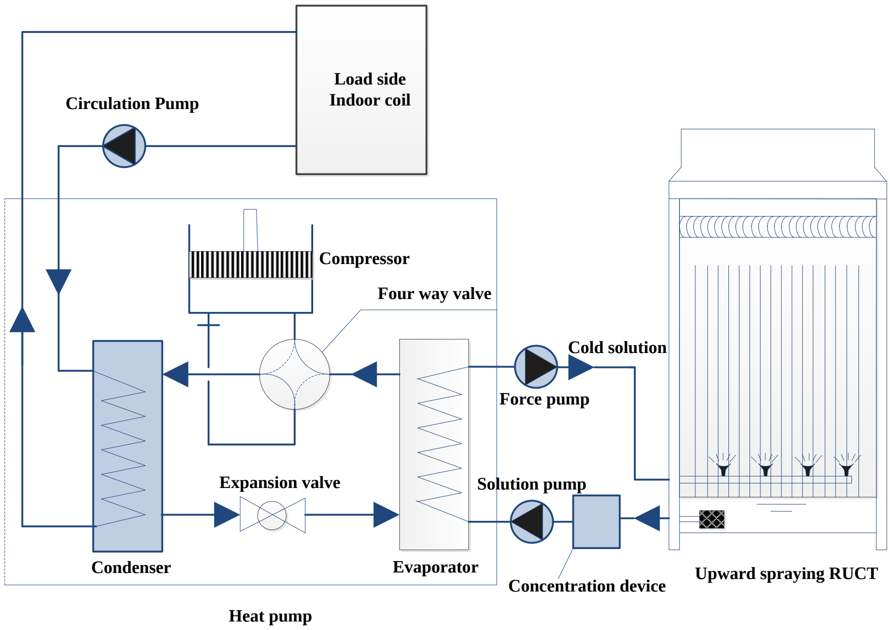


Fig. 15 Frost-free air source heat pump with a reversibly used cooling tower [104]

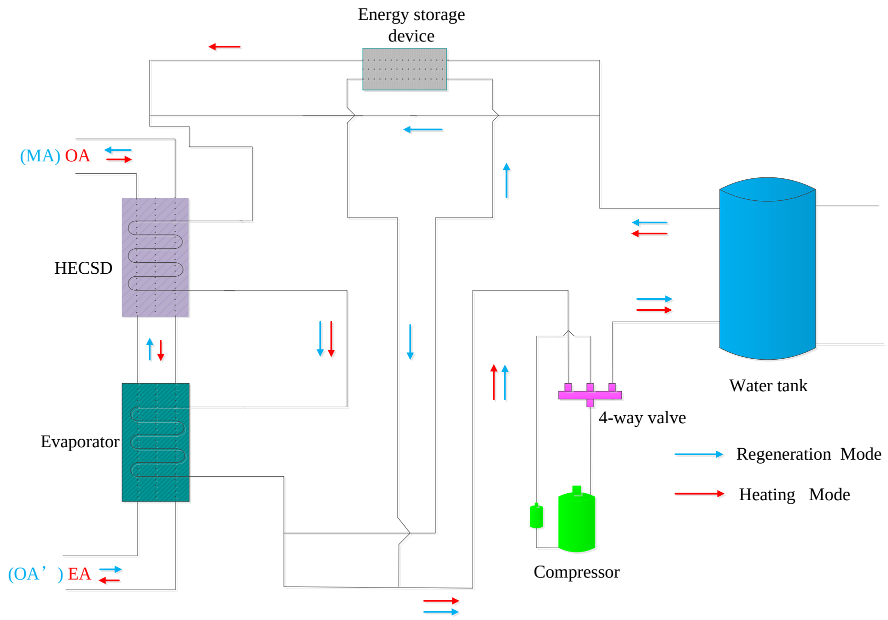
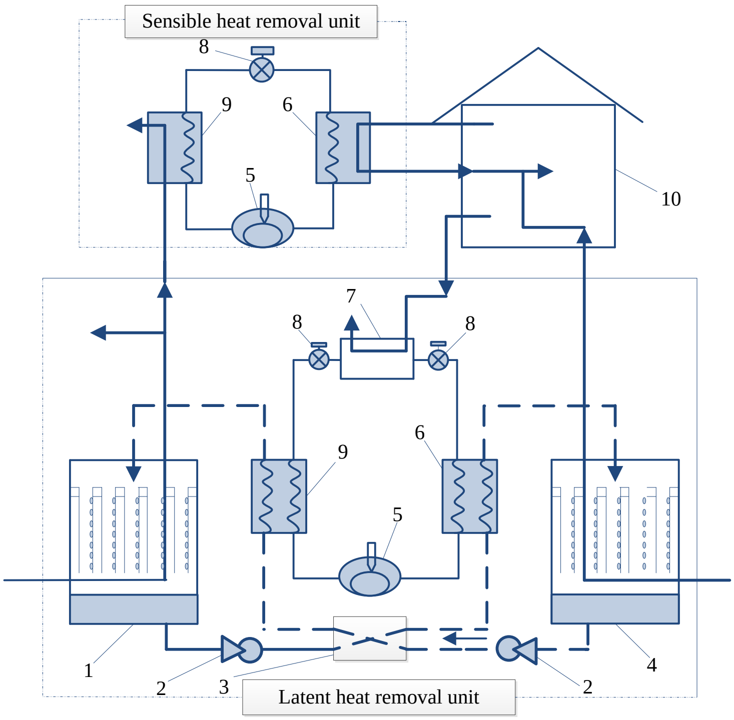


Fig. 16 Schematic diagram of a novel frost-free air-source heat pump water heater system based on solid desiccant dehumidification [107]



1-Dehumidifer or Regenerator, 2-Pump, 3-Solution heat exchanger, 4-Regenerator or dehumidifier, 5-Compressor, 6-Condenser or Evaporator, 7-Auxiliary heat exchanger, 8-Valve, 9-Evaporator or Condenser, 10-Air-conditioning room

Air — LiCl solution — Refrigerant

Fig. 17 Schematic diagram of a hybrid frost-free air conditioning system with liquid desiccant dehumidification [111]

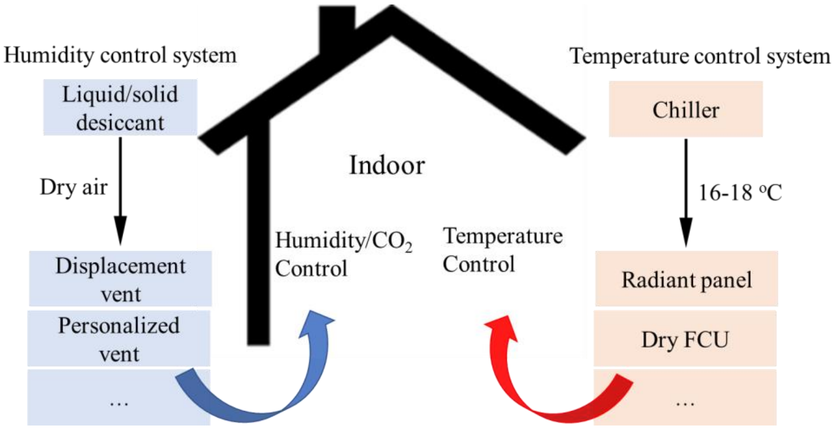


Fig. 18 Operating principle of the temperature and humidity independent control air-conditioning system [114]

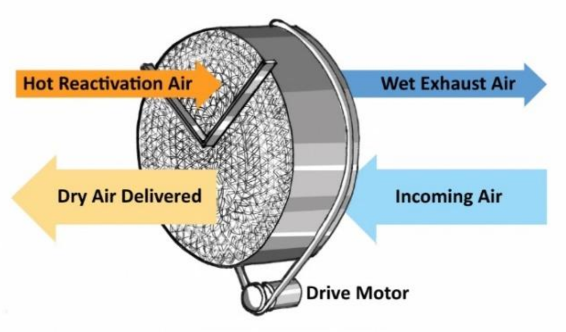


Fig. 19 Rotary wheel filled with solid desiccants [115]

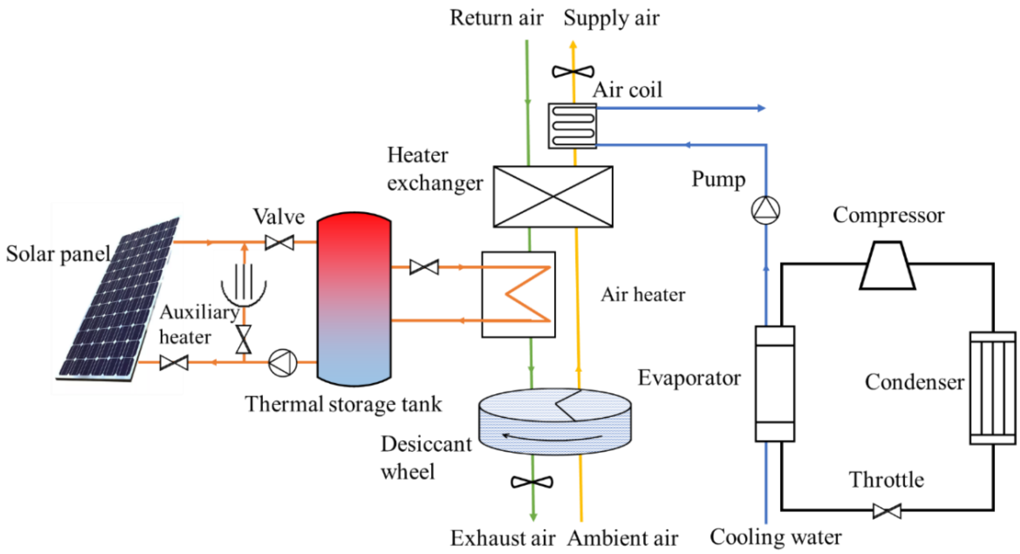


Fig. 3 Solar energy driven solid desiccant system [**Error! Bookmark not defined.**]

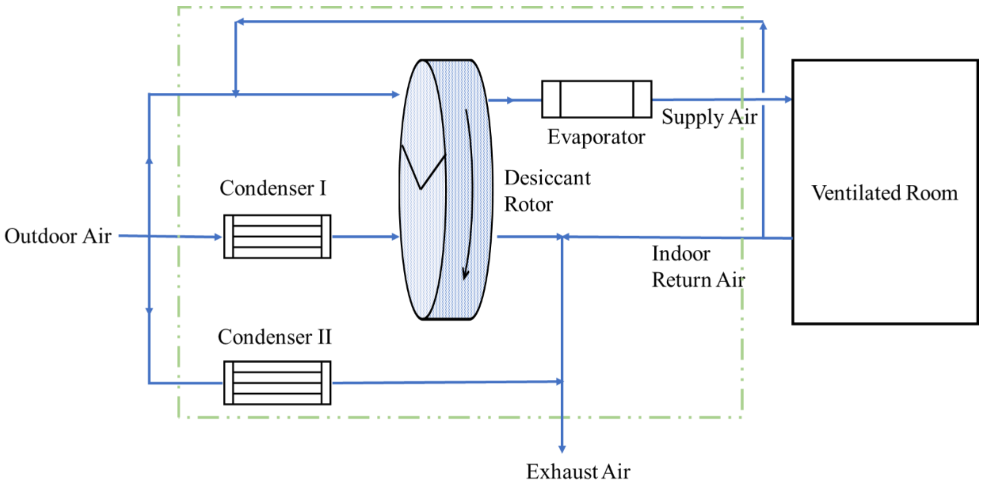
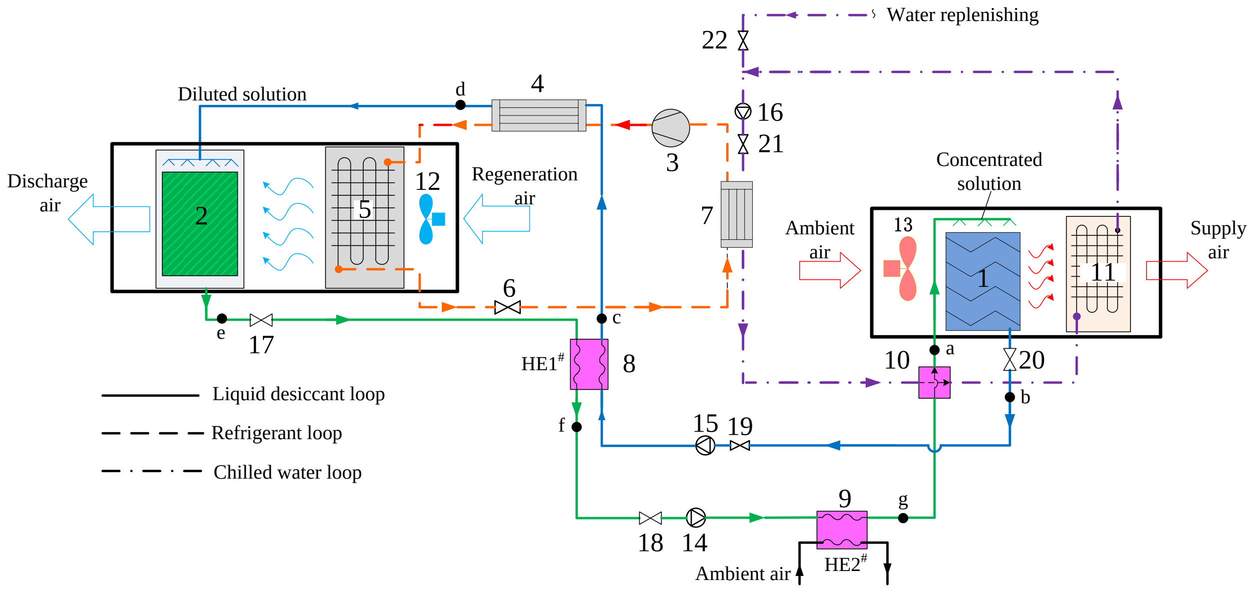


Fig. 4 Heat pump driven solid desiccant system [**Error! Bookmark not defined.**]



1-Dehumidifier; 2-Regenerator; 3-Compressor; 4-Solution-cooled condenser; 5-Air-cooled condenser; 6-Expansion valve; 7-Evaporator; 10-Solution cooler; 11-Air cooler

Fig. 5 Heat pump driven liquid desiccant system with low solution concentration [**Error! Bookmark not defined.**]

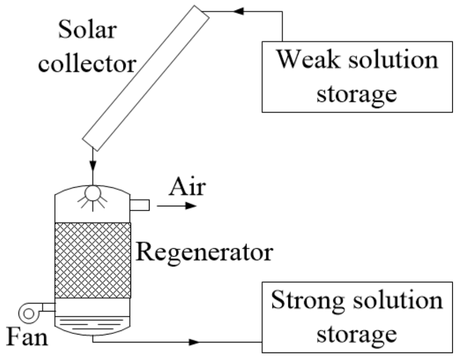


Fig. 6 Typical liquid desiccant regeneration driven by solar energy [**Error! Bookmark not defined.**]

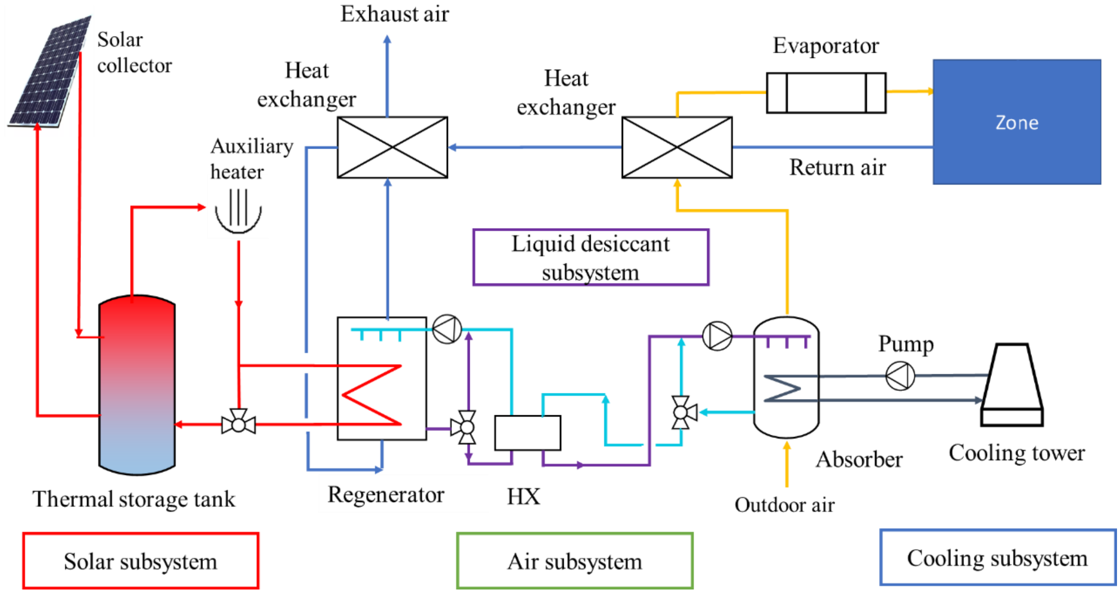
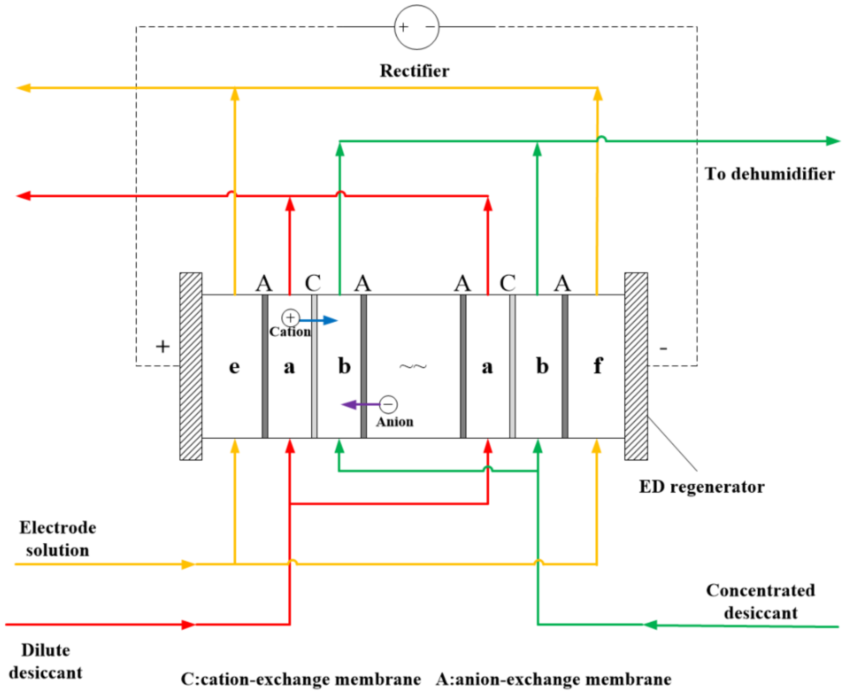


Fig. 7 Hybrid liquid desiccant system with solar energy [**Error! Bookmark not defined.**]



a. dilute cell; b. regenerate cell; e. anode cell; f. cathode cell; A. anion-exchange membrane; C. cation-exchange membrane

Fig. 8 An electrodialysis regenerator for a liquid desiccant air conditioning system [**Error! Bookmark not defined.**]

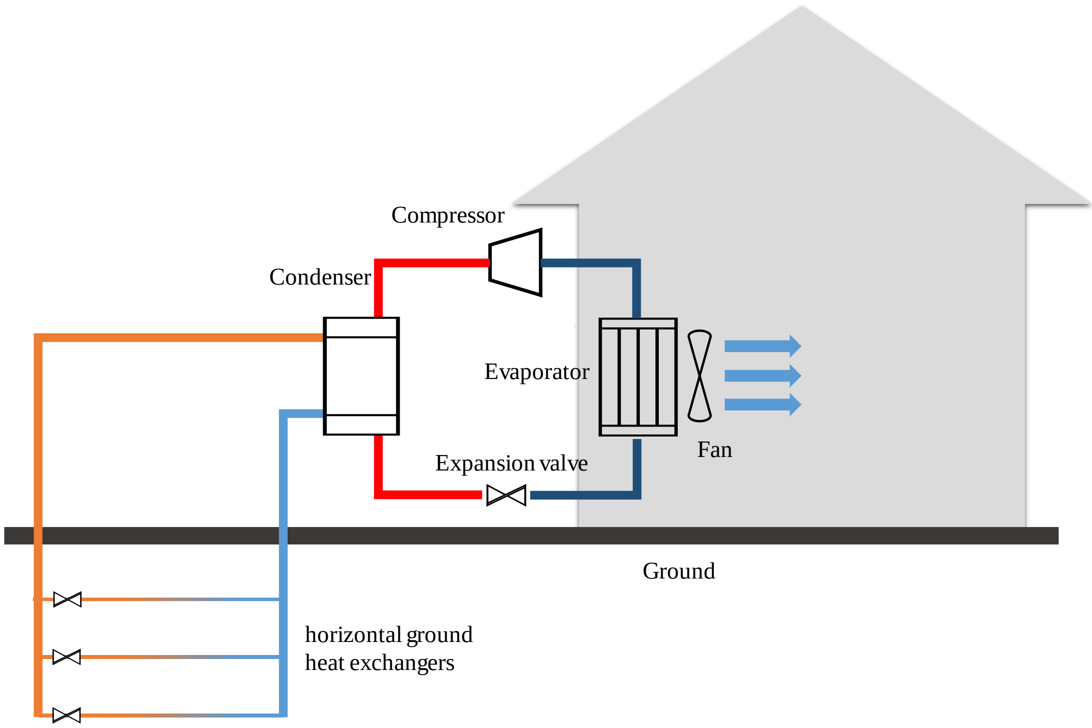


Fig. 9 Ground-source heat pump with horizontal ground heat exchangers for cooling.

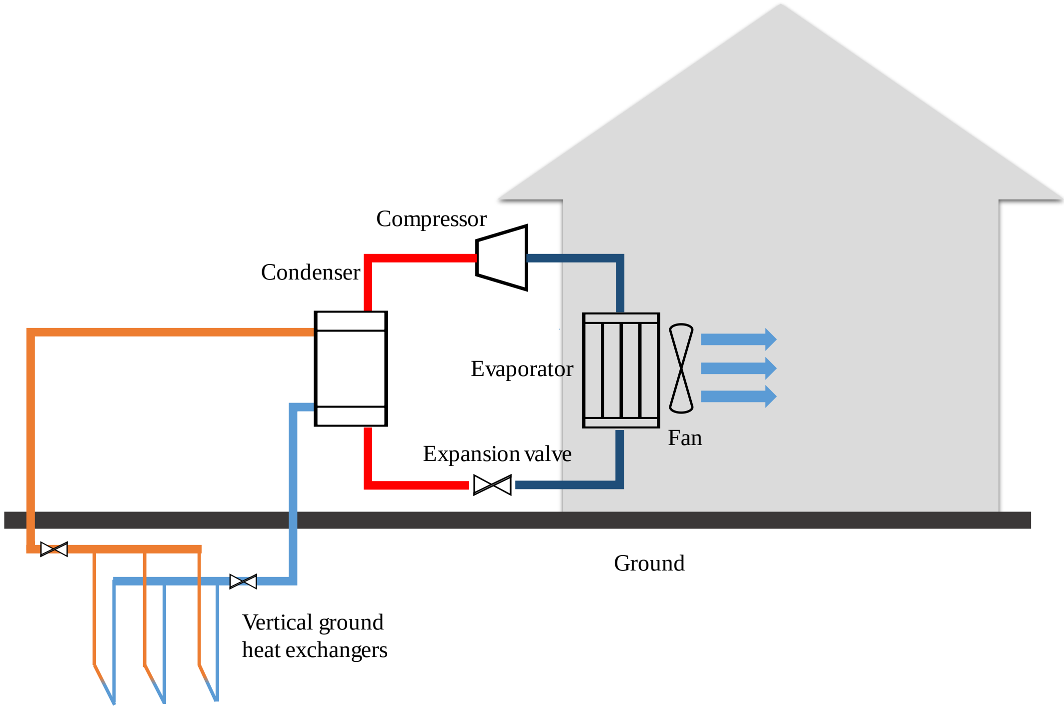


Fig. 10 Ground-source heat pump with vertical ground heat exchangers for cooling.

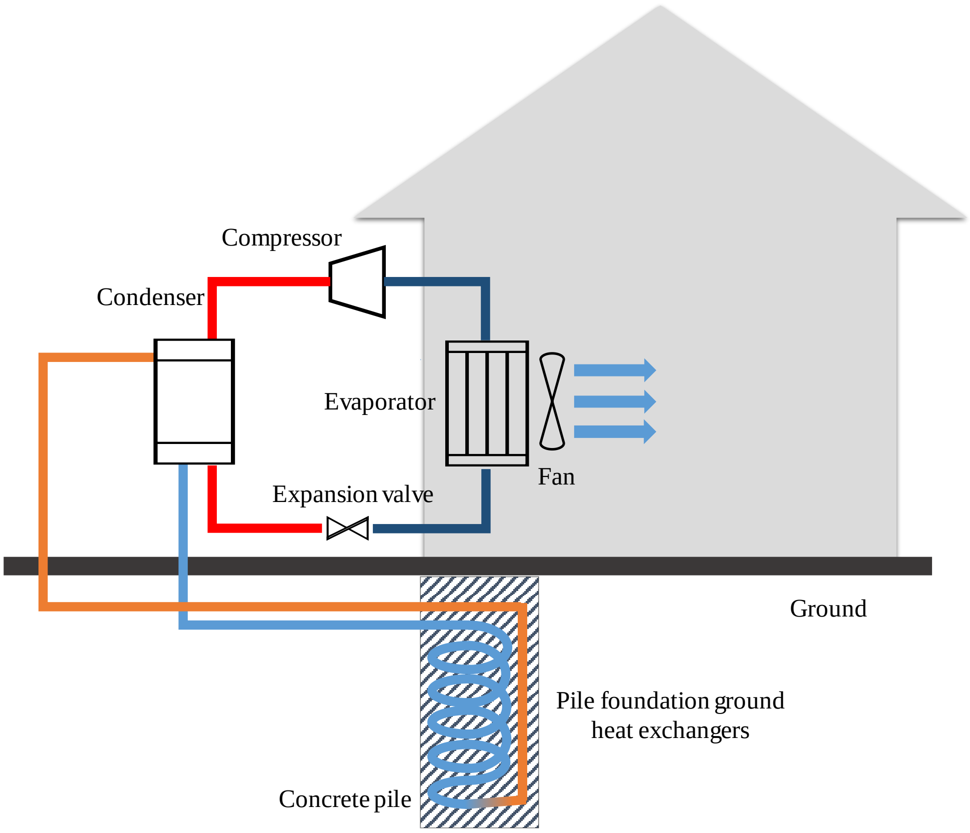


Fig. 11 Ground-source heat pump with pile foundation ground heat exchangers for cooling.

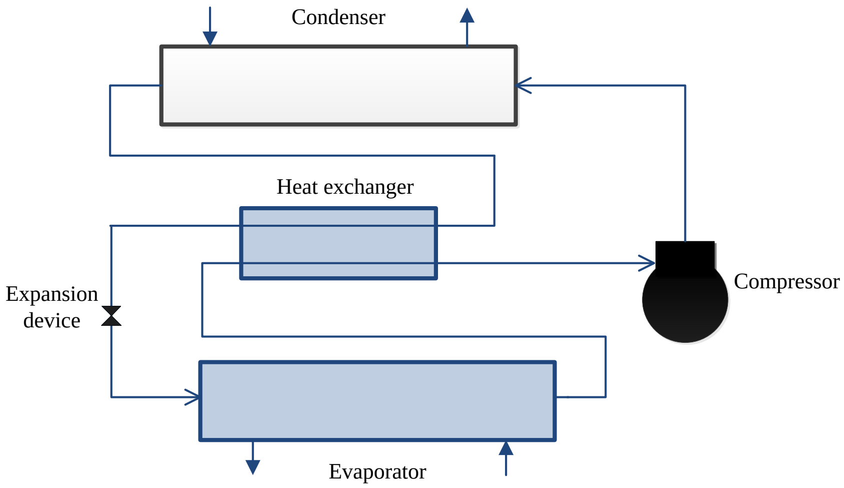


Fig. 12 Schematic of the vapor compression refrigeration system with a liquid-suction/internal heat exchanger [**Error! Bookmark not defined.**]

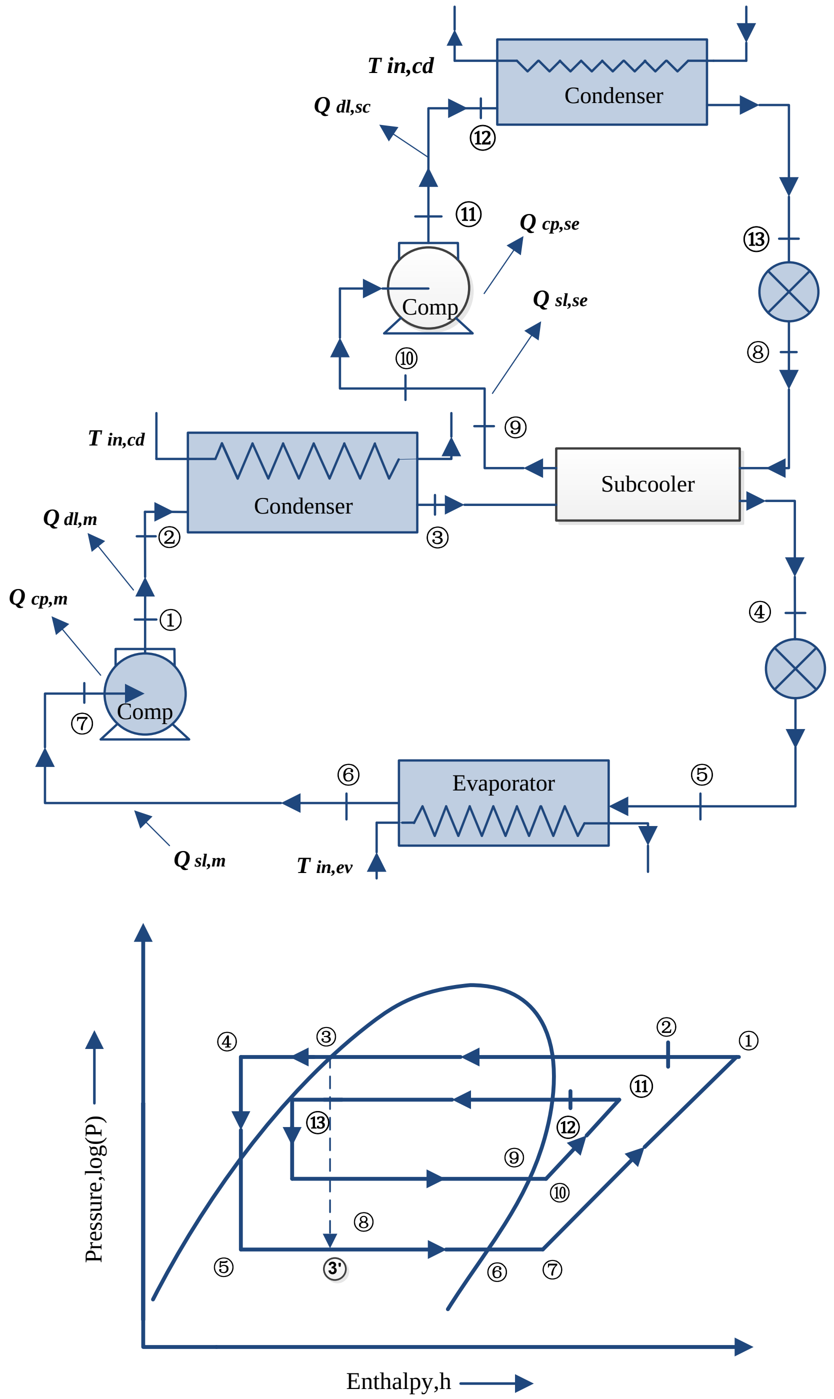


Fig. 13 Schematic of a vapor compression refrigeration cycle with dedicated mechanical sub-cooling [**Error! Bookmark not defined.**]

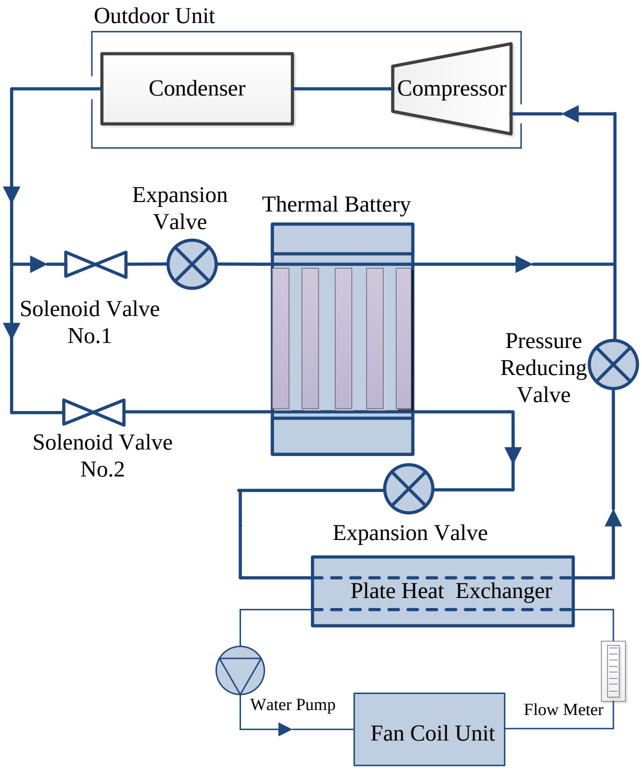


Fig. 31 Schematic of subcooled ice storage air-conditioning system [**Error! Bookmark not defined.**]

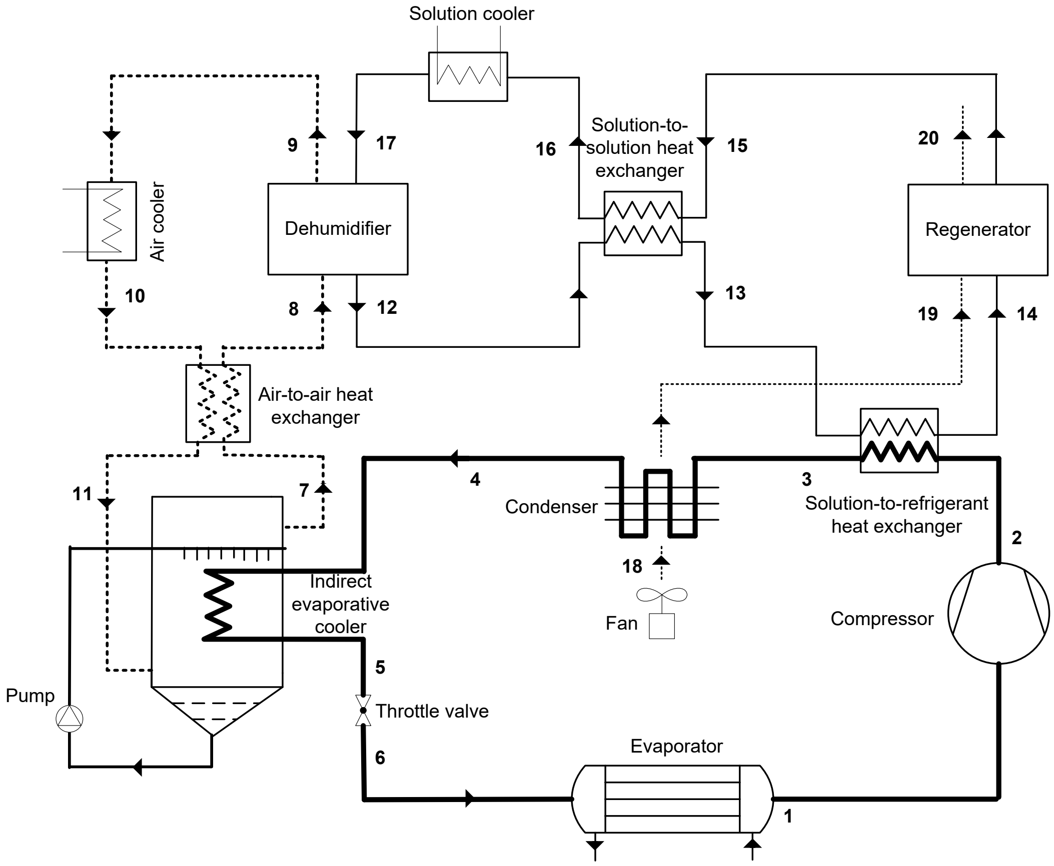


Fig. 142 Schematic diagram of the liquid desiccant dehumidification subcooling method for vapor compression refrigeration systems [**Error! Bookmark not defined.**]

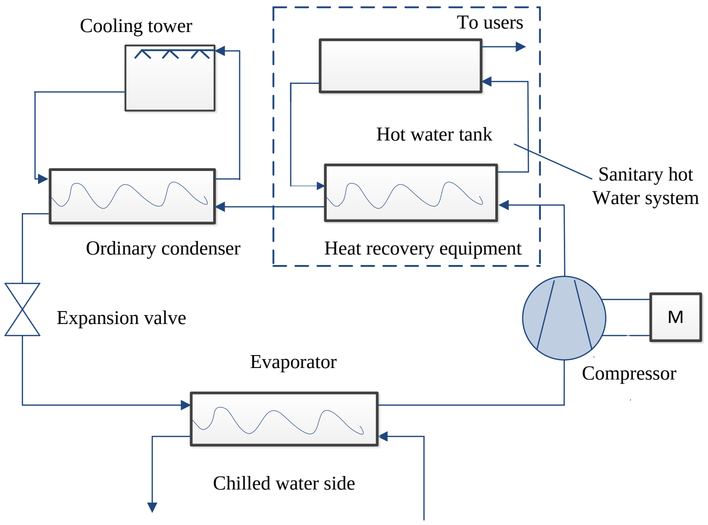
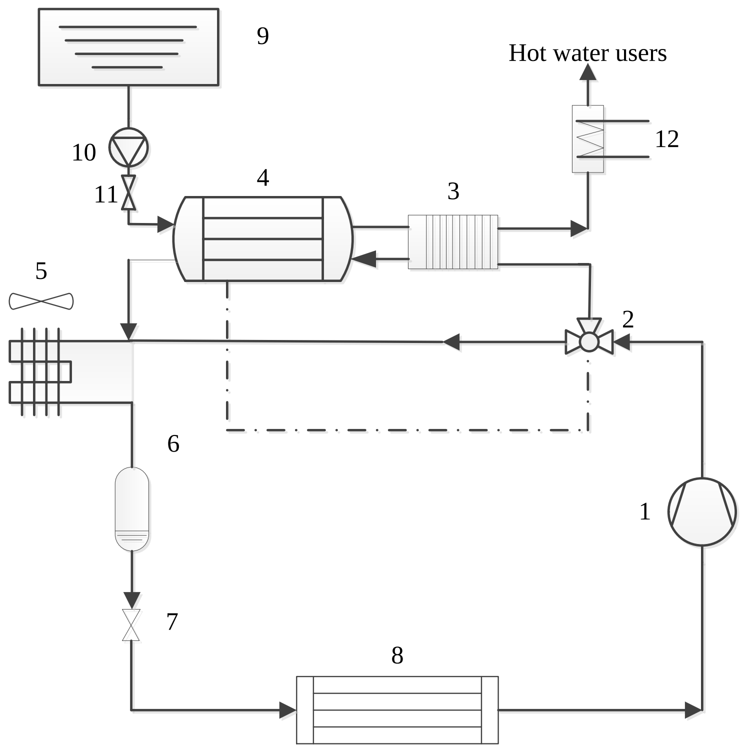


Fig. 33 Schematic of a vapor-compression refrigeration cycle with direct heat recovery [**Error! Bookmark not defined.**]



1-Compressor, 2-Three-way valve, 3-Higher temperature accumulator (accumulator 1), 4-Lower temperature accumulator (accumulator 2), 5-Cooling tower, 6-Liquid storage tower, 7-Valve, 8-Evaporator, 9-Tap water tank, 10-Water pump, 11-Tap water valve

Fig. 34 Schematic of air conditioning systems with indirect condensing heat recovery [**Error! Bookmark not defined.**]

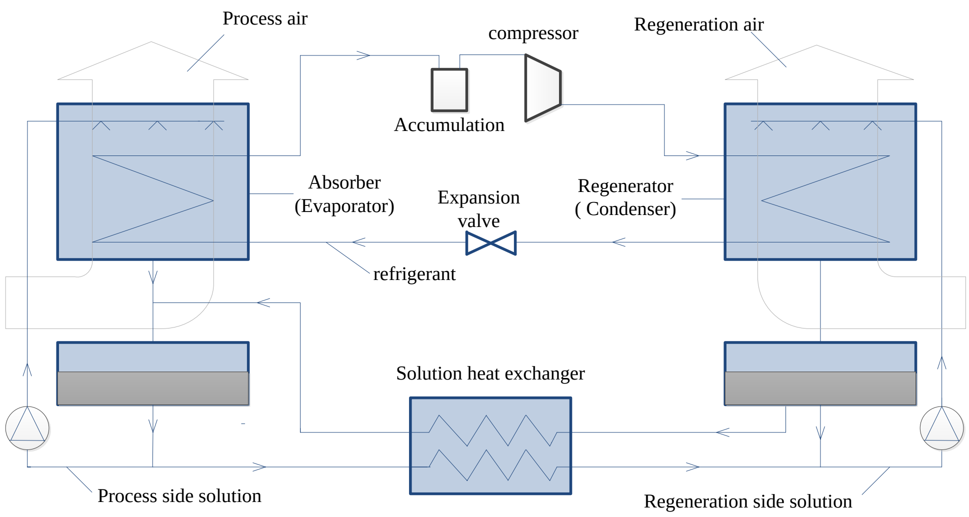


Fig. 35 Schematic diagram of hybrid liquid desiccant air-conditioning system [**Error! Bookmark not defined.**]

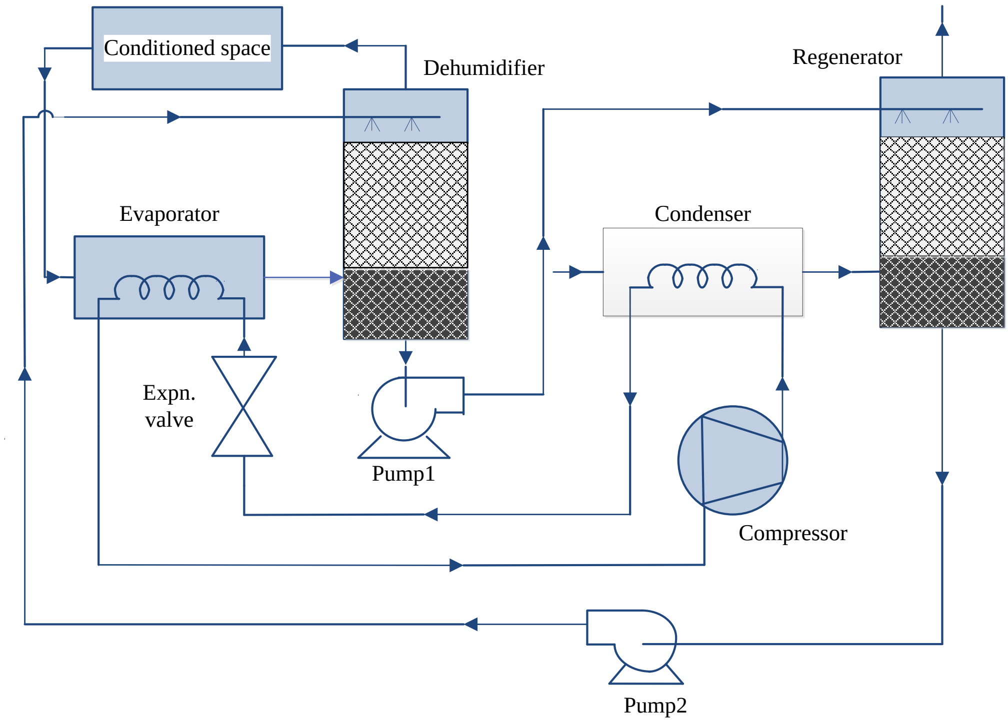


Fig. 36 Block diagram of the proposed system [**Error! Bookmark not defined.**]

Table 1 Research on ice storage air conditioning systems

Research emphasis	Nature of work	Main outcomes	References
Optimal control	Simulation	With optimal control strategies, energy savings could be as high as 9.4%; electricity cost savings up to 30%	[30-38]
Thermodynamic performance	30% Experiment 70% Simulation	Energy consumption decreases by 11.83%; electricity cost saves up to 55%	[39-48]
Economic effectiveness	Simulation	Payback period varies from 3.43 to 22 years based on different tariff structures	[49-52]

Table 2 Results of multi-objective optimization for the end points of the ice storage and PCM storage AC systems [64]

Objects	Ice storage AC system	PCM storage AC system
Maximum exergy efficiency (%)	48.36	55.57
Total cost rate at maximum exergy efficiency (USD s^{-1})	0.13	0.25
Minimum exergy efficiency (%)	21.97	30.45
Total cost rate at minimum exergy efficiency(USD s^{-1})	0.07	0.13
Annual investment and operating cost of entire system($)	482,823	806,591
Annual CO_2 emission amount (kg)	3,015,225	2,670,306

Table 3 Comparison of chilled water, ice and PCMs storage

Type	Advantage	Disadvantage	Applications	Cold storage medium
Chilled water storage (4-12 ^{o}C)	1) Simple system structure 2) Low investment 3) Low level technical demand	1) Low energy storage density 2) Occupy large place	Air conditioning	Water
Ice storage	1) High energy storage density 2) Narrow melting temperature 3) low investment 4) Compactness	1) Low compressor COP	Air conditioning	Ice/Water
PCMs storage	1) High energy storage density 2) Narrow melting temperature 3) Compactness 4) Good equipment compatibility	1) Low thermal conductivity 2) Corrosion, subcooling and phase segregation 3) Stability loss 4) High investment	Air conditioning; Commercial refrigerator and freezer; Refrigerated truck	Paraffin wax [55-59, 63, 64, 66] Dimethyl adipate [60-62] $CaCl_2 \cdot 10H_2O$ and $KF \cdot 4H_2O$ [67] $CaCl_2 \cdot 6H_2O$ [68] Commercial salt hydrate S15 [69] and S10 [70] New water-salt solution [72] Commercial $NaNO_3$ solution (Climsel-18) [73] Commercial water-salt solution SN26 [75] 18% NaCl solution [77]

Table 4 Summary of the defrosting and frost-free technologies

	Technology	Basic idea	Remarks
Defrosting	The reverse cycle	Evaporator is converted into a condenser by valve regulating	It is the most widely used defrosting method. Integrated with thermal energy storage, defrosting time is significantly decreased and the risk of low-pressure cutoff could be avoided
	Surface coating	Hydrophobic surface has a larger contact angle and lower surface energy, so the frost structure becomes fragile	It is one of the most promising methods. The defrosting mechanism is not fully clear.
	Ultrasonic vibration	Ultrasound has a complex mechanical or acoustical effect on small ice crystals.	It is not widely studied, and the disadvantages remain to be overcome, such as larger power consumption and electromagnetic radiation.
Frost-free	Reversibly used cooling tower	Anti-freezing solution is used as the circulation medium between the evaporator and air to avoid their direct contact	It is mainly used for air-source heat pumps and is almost fully developed.
	Desiccant dehumidification	Air is dehumidified to reduce its dew point temperature before flowing through an evaporator	It attracts much attention recently and is a promising frost-free technology both for refrigerated space and air-source heat pumps. The disadvantage is the occupation of large space.