Air-Source Heat Pump Systems

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Contents

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R. Wang, X. Zhai (eds.), *Handbook of Energy Systems in Green Buildings*,
https://doi.org/10.1007/978-3-662-49120-1_2

Abstract

Applying heat pumps to space heating for residential buildings in cold regions will reduce the combustion of gas, oil, and other fossil fuels and the emissions of greenhouse gases. An air-source heat pump (ASHP), which uses the easily available air as heat source, is more easily deployed and applied than other types of heat pumps, such as geothermal heat pumps. In terms of an ASHP, however, it is hard to effectively maintain a high capacity all the time not only because of a variety of instantaneous loads and demands affecting efficiency curves but also due to the unstable outdoor air temperature and humidity during summer and winter seasons. These uncertainties will increase the difficulty to control, rate, and select ASHP units. Moreover, when an ASHP runs at low ambient temperatures, several problems restrict its applications, deteriorated heat output, high discharge temperature, and declining coefficient of performance (COP), due to an increase of the pressure ratio. The high discharge temperature may even result in an abnormal shutdown of the system.

This chapter principally probes into discussions regarding ASHP systems, including the topics of system components, system performance ratings, defrosting methods, system design selections, unit(s) energy regulations, as well as installation considerations and technical measures. Additionally, this chapter also covers the most updated concepts to promote the performance of ASHP, including the subcooling cycle, the multistage cycle, the throttling losses recovery cycle, the multifunction cycle, the saturation cycle, and the frost-free system.

Keywords

Air-source heat pump (ASHP) · Improved cycle · Subcooling cycle · Multistage cycle · Throttling losses recovery cycle · Multifunction cycle · Frost-free cycle

Introduction to Air-Source Heat Pump Systems

Introduction

An ASHP has been widely used in outdoor air as a heat sink/source, usually serving as the first choice for space heating or domestic hot water in residential buildings, owing to its characteristics of relatively easy deployment and less investment compared to other types of heat pumps, such as the sewage source heat pumps and the ground source heat pumps.

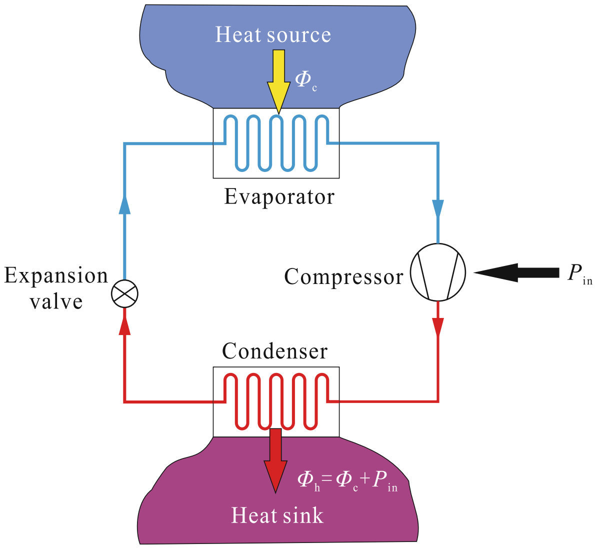


Fig. 1 Basic cycle of an ASHP

Four basic components, i.e., the compressor, the condenser, the evaporator, and the expansion devices, are used in an ASHP to absorb heat from the outdoor air (the heating source) and restore it into a heat sink, as shown in Fig. 1.

An ASHP distinguishes itself from other types of heat pumps in its exceptional heat exchanger at the side of the heat source. Generally, the heat exchanger of an ASHP is composed by a finned-tube coil and a fan: the outdoor air flows over the coil in such a forced manner as to pass heat to the refrigerant inside the coil with an acceptable difference in heat transfer temperatures. An ASHP might be equipped with valves (four-way valve, etc.) to allow for reverse-cycle (cooling) operation. The compressor, the heart of the system, can be driven either by an electric motor or by a gas engine. Gas engine-driven heat pump (GHP), i.e., a heat pump with its compressor driven by a gas engine, is beyond the scope of this chapter. This chapter will mainly focus on the discussion of electrically operated ASHPs unless otherwise stated.

The arrangement of the heat exchanger at the side of the heat sink includes two types. The first one is an air-refrigerant heat exchanger (the indoor heat exchanger) and the corresponding system, shown in Fig. 2, is called an air-to-air type ASHP. In terms of this type of ASHP, the air obtains heat from the indoor heat exchanger under heating mode and subsequently is blown into the room with the assistance of a built-in fan. The merits of this type of ASHP rest with its relatively simple installation, and the demerits lie in its strong blowing feelings, which make people feel uncomfortable under heating modes.

Another type is a water-refrigerant type heat exchanger (the indoor heat exchanger), and the corresponding system is called an air-to-water type ASHP, as shown in Fig. 3. With regard to this type of ASHP, the water or aqueous solution absorbs heat from the indoor heat exchanger under heating mode and is later

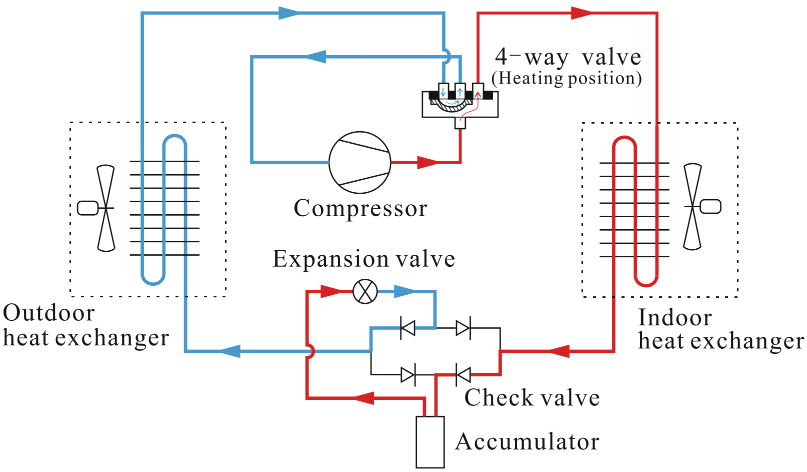


Fig. 2 Air-to-air type air-source heat pump

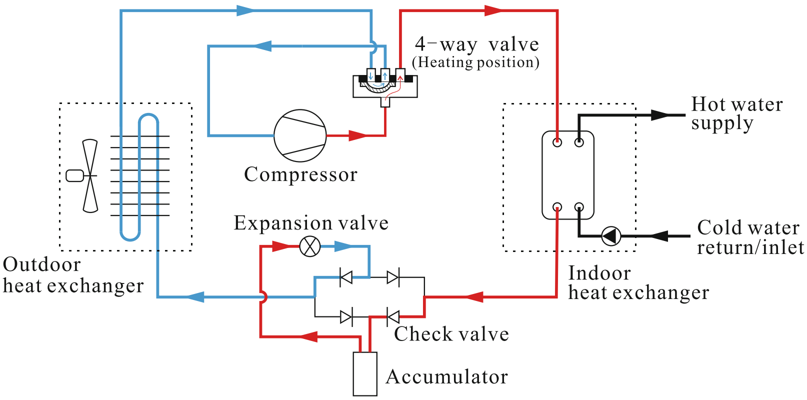


Fig. 3 Air-to-water type air-source heat pump

transferred to indoor terminals, such as fan coils, low-temperature hot water floor radiant heating systems, radiators, etc., by means of a water pump. The selection of the terminals is flexible. In winter when only heating demands exist, a low-temperature hot water floor radiant heating system is more favored since it caters for better comfort.

Outdoor air is endowed with a low heat capacity compared with liquids and solids because of its low density. As a result, the volume flow rate of the air in an air-to-refrigerant heat exchanger is far larger than that of the water in a water-to-refrigerant heat exchanger. The former one results in relatively large size of an ASHP unit and unavoidable noise caused by the fan within an air-to-refrigerant heat exchanger. In general, the design difference between the evaporating temperature and inlet air temperature is about 10 °C, and an air flow of around $0.1 m^3 s^{-1}$ is used for every kilowatt of heat extracted [1].

However, it is hard to effectively maintain a high capacity of an ASHP, mainly due to the unstable outdoor air temperatures hour by hour. The outdoor air temperature will exert a significant impact on an ASHP's COP and capacity and even result in an abnormal shutdown of the system, which are discussed as follows:

1. An ASHP uses finned-tube coil as its evaporator, and when the surface temperature of the coil is below both the dew point of the outdoor air and 0 °C, frost formation will emerge on the surfaces of the outdoor heat exchanger. Heavy frost thickness will deteriorate the heat exchange efficiency between the outdoor air and the refrigerant inside the evaporator, resulting in an abnormally low evaporating temperature. Under such circumstances, the ASHP has to stop to defrost, giving rise to discontinuous heating outputs. Hence, people in the air-conditioned room may probably feel uncomfortable. Therefore, the defrosting of an ASHP accounts for an essential task in humid climates.
2. When the outdoor air temperature drops, the evaporating temperature of the ASHP decreases and the compressor suction refrigerant specific volume increases, which together result in a decrease in the theoretical mass flow rate and an increase in the specific power of the compressor. Moreover, the pressure ratio of the compressor becomes larger when the evaporating temperature drops, resulting in the deterioration in both volumetric efficiency and electric efficiency. All these factors combine to result in the decrease in heat capacity and COP of an ASHP.
3. Refrigerants with small molar heat capacities may undergo a high temperature after compression of the compressor when the pressure ratio is great. Too high discharge temperature often leads to the failure of the lubricating oil in the compressor and even leads to the motor fault. This serves as a major reason accounting for why an ASHP cannot operate at very low ambient temperatures.
4. The heating capacity and heating COP of a single-speed ASHP will decrease with a drop in outdoor air temperatures, and the latter will decrease with an increase in the hot water supply temperatures or the indoor supply air temperatures.

Overview

The efforts to “pump” heat in a thermodynamic cycle from a lower-temperature to a higher-temperature level had its origins in the 1850s, and since then more and more attention has been paid to refrigeration [2]. The first book dealing with the heat pump was published in 1910, and from that time on the heat pump technology has experienced a rapid development. It appears that Haldane was the first engineer in Britain (and, possibly, internationally) to construct, monitor, and record the performance of a heat pump system for space heating [3] . Haldane's experimental heat pump, installed in his Scottish home in 1927, was a typical air-to-water type air-source heat pump in Britain, providing hot water for space heating and domestic water with ammonia as the refrigerant. Another champion of heat pump heating was Guarini, possibly the pioneer to propose a reversible heat pump for space heating and cooling, proved that economy of the heat pump was comparable to that of coals or gases [2]. It was pointed out that over 90% of the world population live in the regions where air-source heat pumps are suitable for heating and cooling [4]. In the 1950s, the USA embarked on mass production of ASHP units, and in the 1980s, Japan started the mass production of kinds of air-source type air conditioners. In China, the market for ASHP units develops rapidly since the 1990s, and China has become the largest production center worldwide. Air-to-air type ASHPs are widely installed in Southern Europe because of their ability to supply air conditioning in summer and the interest in them has increased dramatically in the Nordic countries, while the air-to-water ASHPs have been the most popular ones in Central Europe [5].

System Components

Compressor

The compressor constitutes the "heart" of an ASHP system, which pumps refrigerant around the circuit and produces the required substantial increase in the pressure of the refrigerant. The discharged refrigerant from the compressor, in high pressure and temperature, enters the condenser, to release heat into the heat sink.

The compressors can be divided into two main categories, i.e., the displacement compressors and dynamic compressors [6]. Normally, the displacement compressors are deployed in ASHP, whereas the dynamic compressors are generally used as substitutes of positive displacement compressors for very large capacities and are available with above 300 kW cooling capacity. The displacement compressors take advantage of the shaft work to increase the refrigerant pressure through reducing its volume in the chamber, including reciprocating, rotary, scroll, and crew compressors. The cooling capacity ranges of these compressors are listed in Table 1 according to Ref. [7].

According to Ref. [8], the electrical efficiency of a scroll compressors basically varies from 0.5 to 0.75, with a focus on section 0.6–0.7; the efficiency of a piston compressors is principally between 0.4 and 0.7, with a focus on section 0.5–0.65; the efficiency of a screw compressors is on the whole between 0.5 and 0.75, with a focus on section 0.65–0.75; and the efficiency of compressors reaches its best when its compression ratio is between 2 and 4.

Heat Exchangers

Heat exchangers refer to devices through which heat is transferred from one fluid to another without the two being mixed. In an ASHP, there exist two types of heat exchanger: the air-to-refrigerant and the refrigerant-to-water heat exchanger.

Table 1 Cooling capacity ranges of the displacement compressors

Compressor type	Cooling capacity range
Reciprocating compressor	100 W–200 kW
Rotary compressor	100 W–10 kW
Scroll compressor	5 kW–70 kW
Crew compressor	150 kW–1500 kW

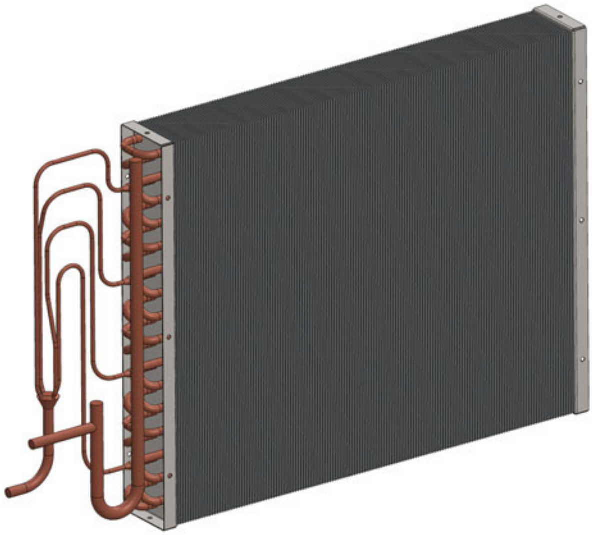


Fig. 4 Finned-tube heat exchanger

Air-to-Refrigerant Heat Exchanger

The heat transferred between air and refrigerant are usually implemented with a finned-tube heat exchanger, as shown in Fig. 4. When the refrigerant evaporates or condenses in the tubes, the heat transfer coefficient at the refrigerant side is far larger than that at the air side. Hence, fins are always equipped outside the tubes to facilitate heat transfer between the tube and the air. There are four commonly used fines with different surface shapes, plain, wavy, slit, and louvered, as shown in Fig. 5. The heat transfer performances of slit and louvered fines are higher than that of plain and wavy fines, whereas the pressure losses at the air side of the former are higher as well. The typical heat transfer coefficient of a finned-tube heat exchanger is 30–40 $\mathrm{Wm^{-2}\ k^{-1}}$ (with air-side heat exchange area as the calculation base) [9]. The heat transfer and friction correlations of the air side for fin-and-tube heat exchangers can be found in Refs [10–15].

Refrigerant-to-Water Heat Exchanger

In ASHP, the refrigerant-to-water heat exchanger is used as the condenser to transfer heat from the refrigerant to the water. Commonly used refrigerant-to-water heat exchangers in an ASHP are shell-and-tube heat exchangers, double-pipe heat exchangers, and vertical brazed plate heat exchangers.

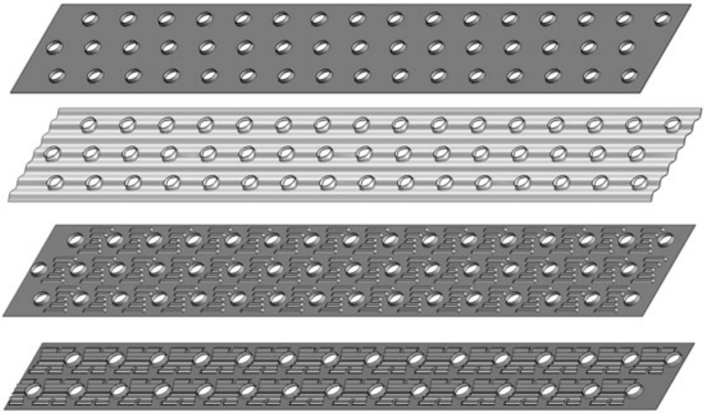
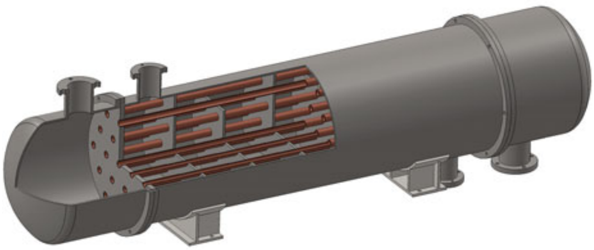


Fig. 5 Fines with different surface shapes

Fig. 6 The structure of a shell-and-tube heat exchanger



A shell-and-tube heat exchanger is built of round tubes mounted in large cylindrical shells with the tube axis parallel to that of the shell [16]. The shell is welded by a steel plate or is cut by a seamless steel pipe, with two pieces of plate welded at both ends of it. A schematic diagram is shown in Fig. 6.

A double-pipe heat exchanger consists of two seamless steel pipes or copper tubes with different diameters. In general, the outer tube is a seamless steel tube with large diameter, while the inner tube is a copper tube with smaller diameters and smooth or low-finned surface, as shown in Fig. 7. This type of heat exchanger is easy to manufacture, whereas the pressure losses in the refrigerant side are always high. As a result, it is generally used as the condenser rather than the evaporator.

A vertical brazed plate heat exchanger consists of a series of thin plates with corrugations separating the fluids, as shown in Fig. 8. The plate is generally made of stainless steel or titanium alloy material in various shapes, such as herringbone plate, corrugated plate, etc., and the design pressure is generally not more than 4.0 MPa. In the heat exchanger, the pattern shape of adjacent plates is always on the opposite direction so that the ridge line intersects each other with contacts. The contacts are brazed to form isolated mesh channels, and the brazing is always done with copper. Ports are placed at each of the four corners of the plates, two inlets and two outlets. The vertical brazed plate heat exchanger is endowed with the merits of compact structure and small volumes. Its

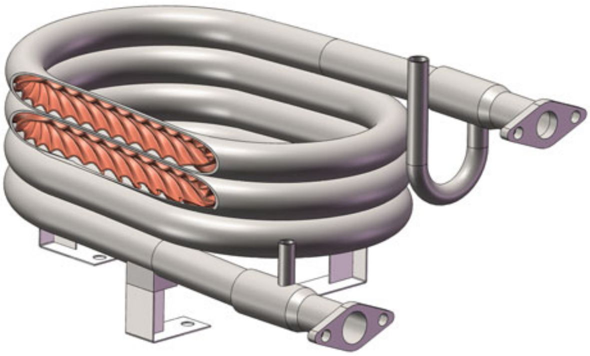


Fig. 7 The structure of a double-pipe heat exchanger

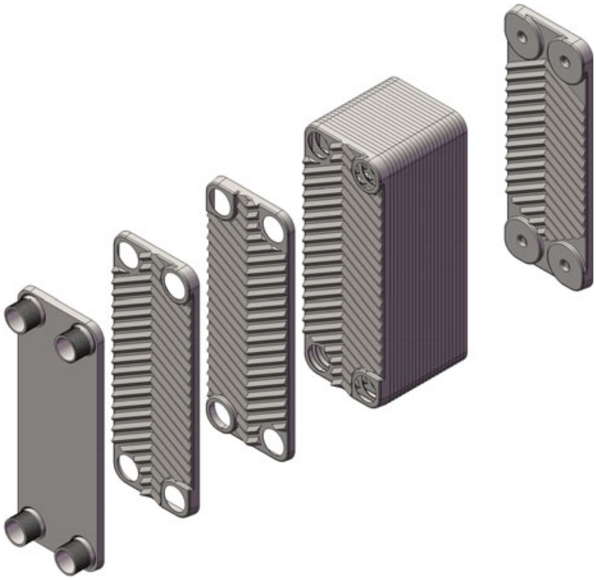


Fig. 8 The structure of a vertical brazed plate heat exchanger

weight is only about 25% of the shell-and-tube heat exchanger with the same heat transfer area. Because the flow passages are quite small, strong eddying brings about high heat transfer coefficients, which may be 1.1–1.7 times of the shell-and-tube heat exchanger.

The approximate scope of the heat transfer coefficients of the three types of refrigerant-to-water heat exchanger [9] is summarized in Table 2.

Expansion/Throttling Structures

A throttling device adjusts the refrigerant mass flow rate to match the displacement of the compressor within a heat pump cycle. Commonly used throttling devices include capillary tubes, thermostatic expansion valves, and electronic expansion valves.

Table 2 Approximate scope of the heat transfer coefficients of the three types of refrigerant-to-water heat exchanger [9]

Heat exchanger type	Heat transfer coefficient ($Wm^{-2}\ k^{-1}$)	Corresponding conditions
Shell-and-tube heat exchanger	800–1200 (R22, R134a, R404A) (taking the tube outside surface area as the heat transfer area)	Temperature rise of the cooling water: 4–6 °C Mean temperature difference: 7–9 °C Water velocity: 1.5–2.5 ms^{-1} Low ribbed copper tube Low ribbed copper tube and ratio of the surface area of the finned side to that of the smooth side ≥ 3.5
Double-pipe heat exchanger	800–1200 (R22, R134a, R404A) (taking the outside surface area of the inner pipe as the heat transfer area)	Mean temperature difference: 8–11 °C Water velocity: 1.0–2.0 ms^{-1} Low ribbed copper tube and ratio of the surface area of the finned side to that of the smooth side ≥ 3.5
Vertical brazed plate heat exchanger	2300–2500 (R22, R134a, R404A)	Stainless steel brazed plate

Capillary Tube

A capillary tube is generally made of a copper pipe with an inner diameter of 0.7–2.5 mm, and the length of the capillary tube varies between 0.6 and 6 m according to its own design conditions. The structure of a capillary tube is simple, whereas its internal flow mechanism is very complicated. When the capillary size is constant, the mass flow rate depends on the density, the pressure, the degree of subcooling, and the viscosity of inlet refrigerant [17]. Such a tube allows a greater mass flow rate of liquid than of vapor. As pressure falls along the tube, evaporation starts to occur at some point, with vapor bubbles forming in the refrigerant flow. If the flow rate decreases, due to a change in conditions in the rest of the circuit, the pressure drop along the capillary is reduced, and accordingly the bubbles of vapor will not form so early, and as a consequence the flow through the capillary will increase to restore the circuit balance [1]. However, the regulation ability of capillary tube is weak, and several parallel capillary tubes are always used in the condition that the operating condition of the unit varied greatly.

Thermostatic Expansion

TEV refers to the thermostatic expansion valve. It consists of an orifice whose opening areas can be adjusted by a spring-loaded needle or plunger, a diaphragm connected to the plunger, and a sensing bulb connected via a capillary tube to the other side of the diaphragm. The bulb is located at the exit of the evaporator to sense the temperature of the vapor leaving the evaporator. The plunger side of the diaphragm is exposed to the liquid-line refrigerant pressure at the entrance to the evaporator (sometimes can be treated as the evaporating pressure). The combination of spring pressure, bulb pressure, and line pressure is designed to control the refrigerant flow to maintain a fixed vapor superheat at the sensing bulb. An increase in superheat results in an increased bulb pressure, which depresses the diaphragm, opens the valve, and increases the refrigerant.

In contrast, a decrease in superheat has a reverse effect. In order to eliminate the effluence of the pressure drop inside the evaporator, an external equalizer, connected via a capillary tube to the exit of the evaporator, is used rather than the internal equalizer.

Electronic Expansion

An electronic expansion valve is often referred to as an EEV. Application of an EEV requires a valve, controller, and control sensors. The control sensors may include a set of pressure and temperature sensors. The valve is driven by the electric motor, such as the step motor. The EEVs may be controlled by either digital or analog electronic circuits. Electronic control gives additional flexibility over TEV to consider control schemes that would otherwise be impossible, including stopped fully closed or full flow when required [17]. The increasingly popular EEV, offering a refrigerant flow control (i.e., endowed with a pretty wide load range), is more flexible, self-adapting, intelligent, and non-refrigerant-specific.

Auxiliary Components

An axial-flow fan is always equipped in the outdoor fin-and-tube heat exchanger, while a cross-flow fan is generally equipped in the indoor unit for an air-to-air type ASHP. Equipment intended for use with field-installed duct systems may be supplied with matched fans by the manufacturer.

With regard to an air-to-water ASHP, some manufacturers provide built-in water pumps and electrical resistance along with their products. In the absence of a water pump or electrical resistance inside the ASHP unit, the engineers should select proper pumps and electrical resistance (or other types of supplemental heat source) for the space-heating project.

Components Arrangement

The components of an ASHP can be arranged into different combinations, which would result in different classifications of the ASHP equipment [18, 19]. Figure 9 demonstrates some common arrangements.

The basic components of an ASHP, i.e., the compressor, the heat exchangers, the expansion valve, and the auxiliary components, such as the water pump, can be equipped in one package, with pipes or ducts connected to the indoor system. For an ASHP, the outdoor unit will always contain an air-to-air type heat exchanger, which in most cases is a fin-and-tube heat exchanger. The outdoor unit may only be equipped with a fin-and-tube heat exchanger, and the rest components are placed inside the indoor unit, as shown in Fig. 9c. However, the most common arrangement

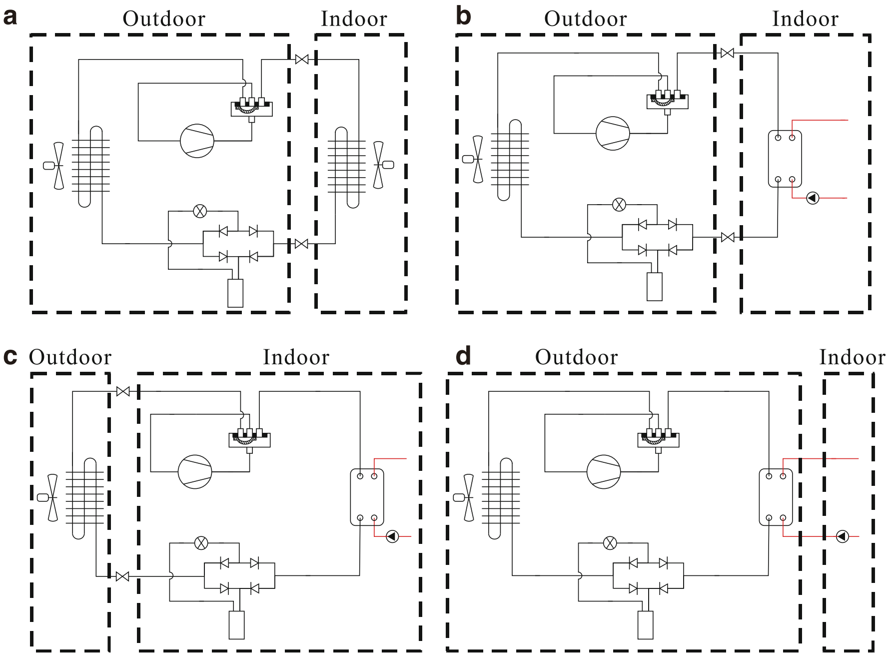


Fig. 9 Schematic of some typical components arrangement

is to place the heat exchanger (refrigerant-to-air type or refrigerant-to-water type) and the corresponding fans or pumps inside the indoor unit and leave the others outside, as shown in Fig. 9a, b. The noise will be a trouble when the equipment is installed in a residential house when placing the compressor(s) inside the indoor unit, and necessary damping measures should be taken. The installment of the equipment of an air-to-water type with all components equipped inside outdoor unit should be carefully designed to avoid bursting in cold or severe cold areas.

The AHRI Standard 210/240 [18] renders a classification of factory-made air-source unitary heat pumps covered within its scope, which sheds light on different arrangements of the components of an ASHP. It is noted that all equipment under such standard are of air-to-air type with their cooling capacities (at AHRI Standard Rating Conditionings) below 19 kW. Refer to AHRI Standard 340/360 [19] if interested in classification of air-source unitary heat pumps with cooling capacities above 19 kW.

System Performance

Efficiencies

Quite a few criteria could be used to describe heat pump efficiency. In all of these criteria, higher number suggests higher efficiency of the system. Some commonly used criteria are discussed as follows.

Heating Coefficient of Performance (COP_H)

COP_H refers to the ratio of the heat pump's heating capacity in watts to the electrical power input values in watts at any given set of rating conditions, the measurement of which is based on laboratory tests, as given below:

$$COP_H = \Phi_H / P_{in} \tag{1}$$

where Φ_H refers to the heating capacity, W, and P_{in} refers to the electrical energy input, W.

An ASHP generally has a COP_H of 2–4, which means it delivers two to four times more energy than they consume. As has been discussed above, the COP_H of an ASHP decreases as the outdoor air temperature drops. The rating of COP_H for a given unit is usually carried out at nominal conditions. The rating conditions vary according to the type and application of the ASHP, which are described in the corresponding rating criteria [18–20]. When comparing COP_H, make sure ratings are based on the same wording conditions.

While comparing COP_H is helpful, it fails to provide enough information of an ASHP. First, frost formation on the outdoor coils is actually possible when an ASHP is operating in cold climates and the unit has to defrost when necessary. To defrost the coils, the unit reverses its cycle and moves heat from the house to the outdoor coil to melt the frost. This significantly reduces the average COP_H. Moreover, the heating capacity of an ASHP shrinks significantly as the outdoor air temperature drops. For most of the heating projects, the installed units cannot provide enough heat during the coldest days of the winter, so that the equipment of backup heating systems are always necessary. This backup is often electric resistance heat with a maximum COP_H of 1.0 only, leading to the decrease of the overall system efficiency.

Energy Efficiency Ratio (EER)

EER represents the ratio of the cooling capacity in Btu/h to the power input values in watts at any given set of rating conditions expressed in Btu/W·h, the measurement of which is also based on laboratory tests, as given below:

$$EER = \Phi_C / P_{in} \tag{2}$$

where Φ_C is the cooling capacity, Btu/h, and P_{in} is the electrical energy input, W.

EER is used for evaluating a heat pump's efficiency under cooling mode. The same rating system is used for air conditioners, making it easy to compare different units.

Heat Recovery Coefficient of Performance (COP_{HR})

COP_{HR} refers to the ratio of the net heat recovery capacity plus the net refrigerating capacity to the total input power input at any given set of rating conditions [20]. COP_{HR} applies to units that are operating in a manner that uses either all or only a portion of heat generated during chiller operation to heat the occupied space, while the remaining heat, if any, is rejected to the outdoor ambient. The calculation of COP_{HR} takes into account the beneficial cooling capacity as well as the heat recovery capacity.

Simultaneous Heating and Cooling Coefficient of Performance ($COPS_{HC}$)

$COPS_{HC}$ indicates the ratio of the net heating capacity plus the net refrigerating capacity to the total input power input at any given set of rating conditions [20]. COP_{SHC} applies to units that are operating in a manner that uses both the net heating and refrigerating capacities generated during operation. COP_{SHC} takes into account the beneficial capacity as well as the heating capacity.

Integrated Energy Efficiency Ratio (IEER)

IEER refers to a single metric for the annualized performance of the mechanical cooling system expressed in Btu/W·h, with the purpose to allow for comparison of mechanical cooling systems at a common industry metric set of conditions. It is based on a volume-weighted average of 3 building types and 17 climate zones in the USA and includes 4 rating points at 100%, 75%, 50%, and 25% load at condenser conditions seen during these load points [19]. It is calculated using tested derived data under the following equation:

$$\mathrm{IEER} = (0.020 \cdot A) + (0.617 \cdot B) + (0.238 \cdot C) + (0.125 \cdot D) \tag{3}$$

where A, B, C, and D are EERs at 100%, 75%, 50%, and 25% load rating points, respectively.

IEER refers to a comparative metric representing the integrated full-load and part-load annualized performance of the mechanical cooling of the air conditioning unit over a range of operating conditions. However, it should not be used to predict the annual energy consumption of a specific building in a given climate area. To more accurately estimate energy consumption of a specific building, an energy analysis using an hour-by-hour analysis program should be performed for the intended building using the local weather data, and added features, if any, like energy recovery, heat reclaim, etc., should be considered.

Heating Season Performance Factor (HSPF)

Since an ASHP's performance significantly depends on the hour-by-hour varied outdoor air temperature, a more realistic measurement is calculated on a seasonal basis. These measurements are referred to as the heating season performance factor (HSPF) for the heating cycle and the seasonal energy efficiency ratio (SEER) for the cooling cycle.

HSPF is defined as the total space heating required during the space-heating season, expressed in Btu, divided by the total electrical energy consumed by the heat pump system during the same season, expressed in watt-hours. HSPF accounts for the reductions in efficiency caused by defrosting, temperature fluctuations, supplemental heat, fans, and on-off cycling. The equation is as follows:

$$\mathrm{HSPF} = \frac{\sum_{\mathrm{j}}^{\mathrm{J}} \frac{n_{\mathrm{j}}}{N} \cdot BL(T_{\mathrm{j}})}{\sum_{\mathrm{j}}^{\mathrm{J}} \frac{e_{\mathrm{h}}(T_{\mathrm{j}})}{N} + \sum_{\mathrm{j}}^{\mathrm{J}} \frac{RH(T_{\mathrm{j}})}{N}} \cdot F_{\mathrm{del}} \tag{4}$$

where $BL(T_{\mathrm{j}})$ is the building space-heating load corresponding to an outdoor temperature of T_{j}, Btu/h; $e_{\mathrm{h}}(T_{\mathrm{j}})$/N is the electrical energy consumed by the heat pump during periods of the space-heating season when the outdoor temperature fell with the range represented by bin temperature T_{j} to the total number of hours in the heating season (N), W; $RH(T_{\mathrm{j}})/N$ refers to the ratio of the supplemental heat energy used for space heating when the outdoor temperature fell with the range represented by bin temperature T_{j} to the total number of hours in the heating season, W; T_{j} is the outdoor bin temperature, °F; n/N is the fractional bin hours for the heating season; j is the bin number, dimensionless; J is the total number of temperature bins for the specified climatic region, dimensionless; and F_{def} is the demand defrost credit, dimensionless. Refer to [18] for detailed information on the calculation of HSPF for a specified ASHP unit. HSPF can be deemed as the "average COP" for the entire heating system. To estimate the average COP, multiply HSPF by 0.293 since 1 Btu = 3.412 Wh.

Seasonal Energy Efficiency Ratio (SEER)

SEER means the total heat removed from the conditioned space during the annual cooling season, expressed in Btu, divided by the total electrical energy consumed by the heat pump during the same season, expressed in watt-hours, which is expressed as

$$\mathrm{SEER} = \frac{\sum_{\mathrm{j}=1}^{8} \frac{q_{\mathrm{c}}(T_{\mathrm{j}})}{N}}{\sum_{\mathrm{j}=1}^{8} \frac{e_{\mathrm{c}}(T_{\mathrm{j}})}{N}} \tag{5}$$

where $q_{\mathrm{c}}(T_{\mathrm{j}})/N$ represents the ratio of the total space cooling provided during periods of the space cooling season when the outdoor temperature fell within the range represented by bin temperature T_{j} to the total number of hours in the cooling season (N), Btu/h; $e_{\mathrm{c}}(T_{\mathrm{j}})/N$ is the electrical energy consumed by the test unit during periods of the space cooling season when the outdoor temperature fell within the range represented by bin temperature T_{j} to the total number of hours in the cooling season, W; T_{j} is the outdoor bin temperature, °F; and j is the bin number and ranges from 1 to 8 for cooling season calculations. Refer to [18] for detailed information on the calculation of SEER for specified ASHP units.

Off-Design Performance

The performance changing with the outdoor air temperature constitutes a critical characteristic for an ASHP, which is distinct from other types of heat pump and determines both the design and the engineering application of an ASHP unit. The off-design performance, to a certain extent, determines a unit's efficiency, such as HSPF or SEER, in operation. Hence it is important to lay emphasis on the off-design performance of an ASHP unit in addition to its nominal performance. Figure 10 illustrates the heating performances of an air-to-water type ASHP with a vapor

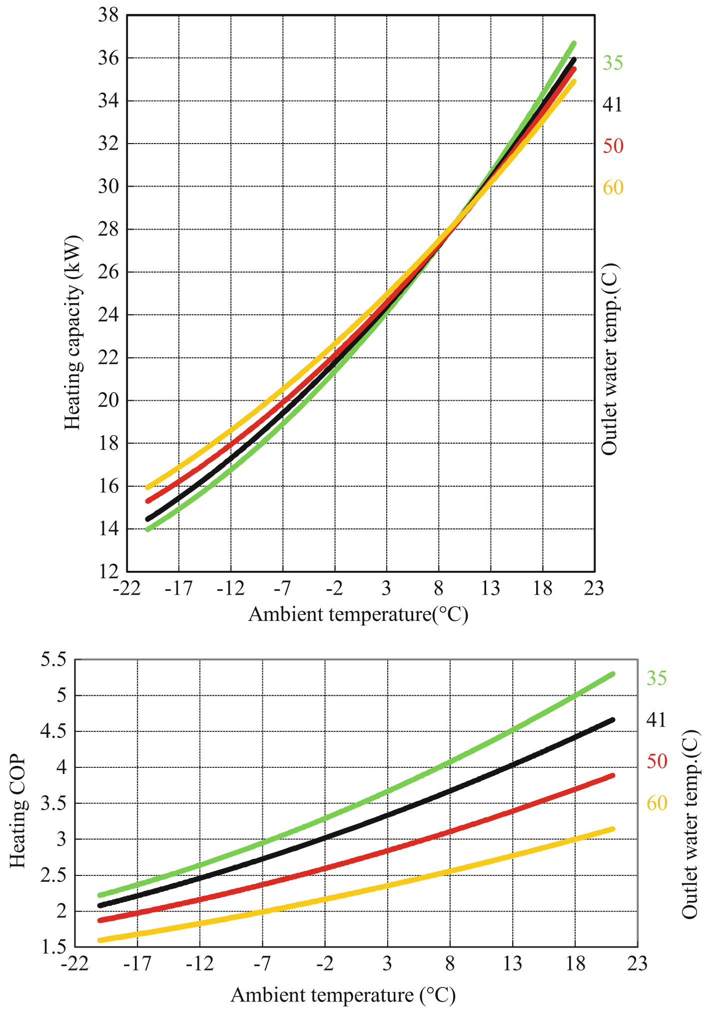


Fig. 10 Heating performance characteristics of an air-to-water type ASHP with vapor injection technology

injection scroll compressor (the vapor injection technology will be discussed later in this chapter). It is found that the heating capacity of the tested unit dropped remarkably as the outdoor air temperature decreases, and the lowest heating capacity (about 14 kW@−20 °C/35 °C, A/W) is less than half of the highest heating capacity (about 37 kW@20 °C/35 °C, A/W). Similarly, the heating COP also drops with the decrease of the outdoor air temperature. Moreover, it decreases with an increase of the outlet water temperature at the same outdoor air temperature. Hence, it may be technically feasible to install an ASHP unit in areas where the ambient outdoor temperature may drop to −20 °C, for example, whereas it may not be economically feasible. Moreover, it may be advisable to design a space-heating project with lower water supplying temperature, for example, 40 °C rather than 60 °C, to obtain a higher HSPF and hence relatively lower operating costs. Moreover, it may be worth pondering over at which ambient temperature the capacity of the ASHP unit should be evaluated and selected in order to achieve a better economy.

Defrosting Method

Frost formation represents the most detrimental problem that happens on the finned-tube evaporator in an ASHP used in cold or severe cold regions. When the surface temperature of the evaporator is below both water freezing temperature and air dew point temperature, the water vapor in the air will condensate on the surface in the form of solid deposit or frost. The frost will accumulate on the surface of the fin-and-tube heat exchanger. The outdoor air temperature and relative humidity (RH) serve as two major influential factors. It is found that the frost accumulated significantly when the air temperature falls between −7 °C and 5.5 °C with RH larger than 60% [21]. The frost may increase the air-side heat transfer coefficient at the beginning. However, when it accumulates severely, the micro-gap between the fins is reduced, giving rise to a much more severe drop of air-side pressure and an increase in the power consumption of the fan at the same time. The air flow rate will be reduced significantly, followed by the appreciable degradation of the total heat transfer coefficient. The evaporating temperature, as a result, may drop to a very low level and sometimes lead to system shutdown eventually. Hence, effective defrosting is necessary for an ASHP equipment when freezing takes place.

In terms of defrosting in actual system operation, there are four types of methods, namely, the on-off defrosting, the electrical heating defrosting, the reverse-cycle defrosting, and the hot gas bypass defrosting.

On-Off Defrosting

The principal merits of this type of defrosting method lie in its simplicity and economy. The application of such method will lead to the shutdown of the system, although the fans of the outdoor unit may still keep running until the frost accumulated on the heat exchanger melts completely. Because the melting point of the frost is always above 0 °C, this type of defrosting method is usually adopted combined with other types of methods: when the outdoor ambient temperature is above some point, for example, 5 °C, the on-off method is utilized; otherwise, the other method, for example, the reverse-cycle defrosting method, is adopted.

Electrical Heating Defrosting

The electrical heating defrosting works through using electric resistive heater embedded in the outdoor heat exchangers. When it needs to defrost, the system will be turned off, and the electric resistive will be turned on until the frost melts completely. The installment and control of the electric resistive are so easy that many manufacturers choose to use it despite of its relative high electricity energy consumptions for defrosting.

Reverse Cycle Defrosting

This type of defrosting method is the most used method currently. When a defrosting period begins, the flow direction of the discharged refrigerant runs contrary to the normal operation via the on-off of the four-way valve. The outdoor heat exchanger then becomes the condenser, and the hot gas from the compressor melts the frost. The fans of the outdoor must be turned off to speed up the defrosting period. In terms of an air-to-air type ASHP, the fan of the indoor unit is always turned off to avoid uncomfortable blowing feeling. However, with regard to an air-to-water type ASHP, the water pump must be carefully kept running during the whole defrosting period to avoid freezing damages of the refrigerant-to-water heat exchanger.

In terms of this type of defrosting method, a four-way valve will be necessary even if the system is designed for heating only. Moreover, when the defrosting task ends, it will take several minutes to restore to its normal working state. In this period, the system especially calls for careful control if an EEV serves as the throttling structure since the pressure of the system is opposite at the beginning.

Hot Gas Bypass Defrosting

This type of defrosting method implements a bypass valve, which is a normally closed on-off solenoid valve, between the compressor discharge line and the inlet of the outdoor heat exchanger, as shown in Fig. 11. The start of the defrosting suggests the shutdown of both the fan of the outdoor heat exchanger and the fan/pump of the indoor heat exchanger. In the meantime, the bypass valve is opened, and then the hot refrigerant gas from the compressor flows into the outdoor heat exchanger, in which the refrigerant gas condenses and releases heat to melt the frost outside the heat exchanger. When the defrosting period ends, the bypass valve is closed, and the system restores to its heating supply mode again.

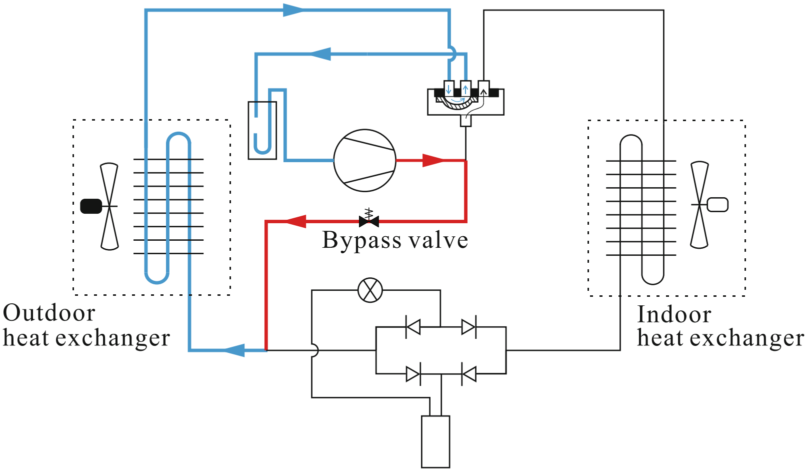


Fig. 11 Hot gas bypass defrosting cycle

Compared with the reverse-cycle defrosting method, the hot gas bypass defrosting method is easier to control, and the switch between the defrosting mode and the normal working mode is smoother. However, an additional solenoid valve and corresponding pipes are needed, and a suction accumulator is always installed before the compressor to avoid the risk of liquid slugging, directly complicating the design of the system. Moreover, because the heat comes only from the compressor during the hot gas bypass defrosting process, the defrosting period is always longer than that of the reverse-cycle defrosting method.

Application

Design Selection

Inconsistency of Heating Demand and the Heating Capacity

The heating demand refers to the heat to be delivered to a room to maintain the specified temperature at any moment. In heating seasons, the difference of indoor and outdoor temperatures is always large so that an approximately linear relationship is assumed to exist between the heat demands and the temperature differences. This approximation can also be found under the degree-day methods and bin methods [22]. According to this approximation, the heating demand can be calculated under the following equation:

$$\Phi_{\mathrm{i}} = K_{\mathrm{tot}}(t_{\mathrm{bal}} - t_{\mathrm{o,i}}) \quad (6)$$

where t_{bal} is defined as that value of the outdoor temperature at which, for the specified value of the interior temperature, the total heat loss is equal to the heat gain from sun, occupants, lights, and so forth; K_{tot} is the total heat loss coefficient of the building in W/K.

According to Eq. 6, the heating demand increases as the outdoor air temperature decreases. On the other hand, the heating capacity and efficiency of an ASHP shrink significantly with outdoor temperature, as shown in Fig. 10. Figure 12 sheds light on performance characteristics of a single-speed, air-source heat pump, along with heating loads for a typical building. The shrink of heat pump heating capacity accompanies the decline of ambient temperatures. This characteristic runs contrary to the trend of the building load. As a result, the inconsistency of the heating demand and the heating capacity emerges as an important problem for the design of an ASHP heating project.

The outdoor temperature at which a heat pump's heating capacity equals the building heating load is called the heating balance point ($t_{bal,H}$). It is the lowest temperature at which the heat pump alone can accommodate the heating load [23]. When outdoor temperature is below the balance point, supplemental heat (usually electric resistance) must be added to make up the difference, as shown by the shaded area.

In terms of an ASHP space-heating project, the building heating load is calculated using standard practices in selecting the proper size of the ASHP. The heating balance point may be lowered by improving the structure's thermal performance or by choosing a heat pump larger than the heating load requires, as shown in Fig. 12.

If the heating balance point is selected too low, for example, as low as the local extreme minimum temperature, no supplemental heat is needed to satisfy the heating demand even on the extremely cold days. In this case, however, there will be more redundant heat when the outdoor air temperature is above the heating balance point temperature and the costs of the units will be much higher. Moreover, excessive oversizing of the capacity causes excessive cycling, probably resulting in uncomfortable temperature feelings.

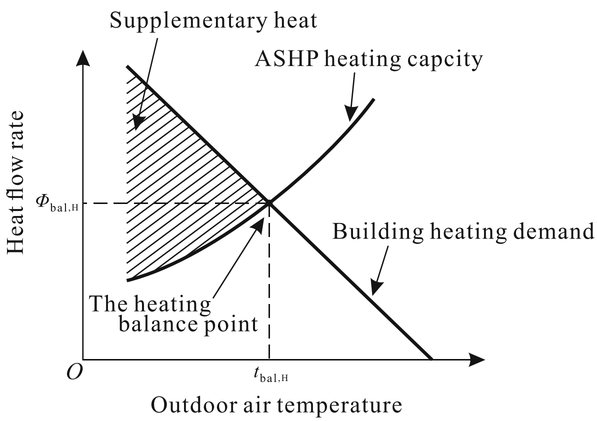


Fig. 12 Characteristics of an ASHP space-heating system

On the contrary, if the heating balance point is excessively high, the size of the units will be small, and more supplemental heat (usually electric resistance) will be needed during colder days, leading to higher operation costs or lower energy utilization rates.

Hence, it is necessary to choose a reasonable heating balance point temperature to ensure that the capacity of the selected unit is not too large and meanwhile, in most of the time, that the heating system operates at a higher efficiency. This will save not only the initial investment but also the operating costs.

The Optimal Heating Balance Point

There exist different two temperatures to determine the optimal heating balance point: the heating optimal energy balance temperature and the heating optimal economic balance temperature. The corresponding two optimal heating balance points are described below.

Heating Optimal Energy Balance Point (OENT)

OENT refers to the heating balance temperature at which the heating season performance factor (HSPF, defined by Eq. 4) of the selected ASHP unit is maximum. For a certain area where the building is located, the HSPF is only a function of OENT when the characteristics of the building envelope and the unit are specified, and the indoor design temperature is given according to the design code. OENT will be obtained when HSPF is maximized [24].

Heating Optimal Economic Balance Point (OECT)

OECT is the heating balance temperature at which the ASHP heating system is designed and meets the minimum cost requirements of the investment and the operation. The annual costs are usually used to evaluate the economy of an ASHP heating system, which is calculated under the following equation [25]:

$$A_a = C_t \frac{i(1 + i^n)}{(1 + i^n) - 1} + C_r \tag{7}$$

where C_t represents the total initial investment cost; C_r refers to annual operating costs, including costs of the energy, maintenance, administration, etc.; i is basic discount rate (BDR); and n is the service life period and can be set as 15 years.

A_a is only a function of OECT when BDR, the service life period, the price of the unit, and the energy are specified. OECT will be obtained when A_a is minimized.

Due to the influence of the energy pricing policies, the obtained ONET and ONCT are always not consistent. Generally, people pay more attention to the economy of an ASHP system than the energy saving, leading to a more common use of OECT.

Auxiliary Heat Source

Since the heating balance point temperature is always higher than the local extreme minimum temperature, auxiliary heat sources are always needed to meet the heating demands during colder days when the ASHP units alone cannot handle the heating load.

Electric resistance is usually chosen as the auxiliary heat source, for its low initial investment costs, highly flexible modulation capacities, stable and reliable operations, low failure rates, small sizes, and easy installment. In terms of an air-to-air type ASHP unit, the electric resistance is usually installed at the indoor supply side; while with regard to an air-to-water type ASHP unit, the electric resistance is usually installed at the outlet side of the hot water.

In addition, steam can also be selected as the auxiliary heat source if available. For areas where there exist great differences of the electric power costs between on-peak and off-peak, electrical energy storage technologies can be used as an effective method for peak load shifting in power grid and to meet the lacking heating load. When the electric power is short and expensive, an oil burning or gas-fired boiler may be selected as an alternative.

Economic analysis is necessary when a variety of auxiliary heat sources are available. The method to determine the heating optimal economic balance point (OECT) can be used to seek an optimal auxiliary heat source by minimizing the annual costs.

The auxiliary heat source is always placed at the heating supply side of the ASHP unit, due to the fact that an ASHP works more efficiently at relatively low condensing temperatures.

Energy Regulation

When the selected ASHP units run at a temperature higher than the heating balance point, the heating capacity of the units is larger than the building heating load, which changes with the fluctuation of the outdoor air temperature. Hence, the units have to adjust the heating capacities to adapt the change of the heating loads. Using variable-speed, multispeed, or multiple compressors and variable-speed fans can improve matching of both heating and cooling loads over an extended range. This equipment can reduce cycling losses and improve comfort levels.

There are several ways for energy regulation in a space-heating project. One way is to choose three to five units rather than single unit in the project. The units are arranged in parallel so that the numbers of the units in operation can be selected to adjust the total heating output. When the heating load decreases to a certain extent, the control system commands one of the remaining running units to stop. Generally, the control system will unload the unit with the longest operation period, and load the unit with the shortest operation time, to keep the balance of the operation time among the units.

Another way is to choose units equipped with a variable-speed compressor. The variable-speed compressor changes its displacement by varying its driving frequency so that its heating capacity is changed. Because the frequency of the compressor is continuously adjustable, the capacity modulation will be more precise, and fluctuations of the room temperature will be smaller. Some commercialized products are designed by the combination of a variable-speed compressor and a constant-speed compressor to reduce the manufacturing cost while keep the modulation capacities.

In terms of the project using air-to-air type ASHP, units with variable-speed compressors should be preferred for the sake of thermal comfort and energy saving. In contrast, as for the project using air-to-water type ASHP, especially when the low-temperature hot water floor radiant heating system is designed as the indoor terminal, parallel units with constant-speed compressors may be preferred. In the air-to-water system, the thermal inertia is prominent due to the water storage in pipes of the system, and on-off of the units will not be frequent and hence will not exert a significant impact on thermal comfort. Due to the fact that a unit with a variable-speed compressor is much expensive than that with a constant-speed compressor, the initial investment costs will be saved.

Selection Steps

1. Determine the heating balance point, $t_{\mathrm{bal,H}}$. This can be determined with the above optimal heating balance point determination method or simply by choosing the outdoor temperature for heating design in design handbooks.
2. Calculate the heating load (Φ_{h}) of the targeted building at outdoor temperature of $t_{\mathrm{bal,H}}$.
3. Find out the heating capacity of the unit at outdoor temperature of $t_{\mathrm{bal,H}}$, $\Phi_{\mathrm{h,0}}$, by using its heating capacity character curves provided by the manufacturer.
4. The number of the units is calculated under the following equation:

$$n = \Phi_{\mathrm{h}} / (k \cdot \Phi_{\mathrm{h},0}) \tag{8}$$

   where k is the loss coefficient of the heating capacity due to frost. The value of k depends on the climatic characteristic where the units is placed, ranging from 0.7 to 1.0. For high humidity areas, a smaller value of k should be chosen. The number of the units should not too much, and 3–5 is appropriate.
5. Calculate the supplementary heat according to Fig. 12.

Installation

The noise reduction and heat exchange with air are the two main concerns for installment of the ASHP units. Some considerations and technical measures are given below [26–28].

- In general, install products containing compressors on solid, level surfaces.
- Avoid mounting products containing compressors (such as remote units) on or touching the foundation of a house or building.
- Avoid mounting products containing compressors near buildings with higher noise control requirements.
- A separate pad that does not touch the foundation is recommended to reduce any noise and vibration transmission through the slab.
- Leave enough spaces around every outdoor air-to-refrigerant heat exchanger. Do not box in outdoor units with fences, walls, overhangs, or bushes. Otherwise, the

air-moving capability of the unit will be lowered, and hence the efficiency will be reduced.

Outdoor units are generally mounted on the roof, balcony, and ground. The outdoor units must be designed to satisfy the requirements of no interference and no air return. When the outdoor unit is installed on the roof, the heat exchange effect is good, while the maintenance and management are not favorable. However, the arrangement of multiple units should avoid air suction interfering with each other. When the outdoor machine is arranged on the balcony, the backflow phenomenon should be avoided. Especially when there are several outdoor units vertically arranged at each layer of the balcony, the discharge air of the units may be suctioned by the units installed above them. Measures should be taken to prevent this from happening.

- As to a split-system remote unit, choose an installation site that is close to the indoor part of the system to minimize pressure drop in the connecting refrigerant tubing. Also, a smaller refrigerant charge and shorter refrigerant lines are used when this distance is short.
- For VRF units, locate the refrigerant pipes' headers so that the length of refrigerant pipes is minimized.
- The prevailing wind direction should not cause the unit to malfunction during periods of high wind velocity.
- The electrical supply panel should be as close to the outdoor unit as possible to reduce the costs of running the electric wire.
- Snow cover should be considered over the outdoor unit to keep out snow, and the foundation should be heightened to raise the unit above the anticipated snow depth where annual snow fall is heavy.
- The defrost water may freeze and accumulate under the outdoor unit under cold climate. If the unit is installed on the ground, the foundation should be high enough so that the accumulated ice will not reach the bottom of unit and block up the drain hole. If the unit is mounted on the wall, make sure the defrost water is drained to somewhere so that no ice hangs on the unit; otherwise the hanged ice will be a threat to the safety of the pedestrians passing by.
- Contact the unitary equipment manufacturer or consult installation instructions for further information on installation procedures. Remember that not all heat pumps have the same installation requirements.

Improved Cycles/Systems

Introduction

A major characteristic for the vapor compression heat pump system is that its efficiency decreases when the difference between the condensing and evaporating temperature increases. In addition, the use of throttling structures instead of expander also led to reduced efficiency because of the loss of expansion work. Importantly, for real cycles using refrigerant with lower molar heat capacities, the discharge temperature of the compressor will be high, especially when the evaporating temperature is too low or the condensing temperature too high. Too high discharge temperature often gives rise to the failure of the lubricating oil in the compressor and even leads to the motor fault, constituting a major reason accounting for why an ASHP cannot operate at very low ambient temperatures. Moreover, when the outdoor air temperature drops or the defrosting cycle starts, the deteriorated heat capacity and COP of an ASHP will emerge as a major problem, or barrier, in terms of its application in space heating. Some improved cycles or concepts are proposed to improve system efficiency or increase the functionality.

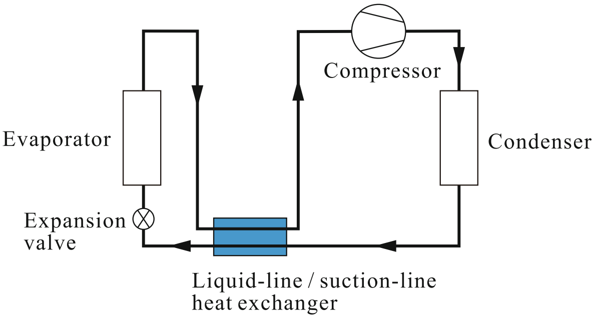
Subcooling Cycle

Typically, the subcooled liquid is designed to serve as the refrigerant entering the expansion device, so as to maintain a stable refrigerant flow rate in the expansion device. The increase of the degree of subcooling will contribute to the refrigeration effect and potentially promote the coefficient of performance (COP).

A common approach to increase the degree of subcooling is to use a liquid-line/suction-line heat exchanger (SSLL-HX), as shown in Fig. 13. The SSLL-HX is located between the condenser outlet and expansion valve inlet and also between the evaporator outlet and compressor inlet. Cold refrigerants from the suction line cool down the refrigerants at the condenser outlet, decrease the enthalpy of the refrigerants entering the evaporator, and hence enlarge the evaporator capacity. On the other hand, the rise of the compressor suction refrigerant temperature gives rise to the deterioration of the compressor suction refrigerant temperature as well as the compressor efficiency and the promotion of the theoretical specific work of the compressor. It was found that the benefits of the SSLL-HX depend on both the operating conditions and the properties of the refrigerant employed [29]. Klein et al. [30] identified a new dimensionless group to correlate performance impacts attributable to SSLL-HX and concluded that the SSLL-HX was detrimental to system performance in systems using R-22, R-32, and R-717, whereas it was useful for systems using R-507A, R-134a, R-12, R-404A, R-290, R-407C, R-600, and R-410A when the SSLL-HX have a minimal pressure loss on the low-pressure side.

The subcooling can also be accomplished by adding a mechanical loop in/to a conventional vapor compression cycle. The subcooling system can either be an integrated system [31], as shown in Fig. 14, or a dedicated system [32], as shown in Fig. 15. In an integrated system, there is only one condenser and a single refrigerant, whereas in a dedicated system, there are two separate condensers and different refrigerants. Figure 16 illustrates a subcooling method for vapor compression refrigeration cycle based on expansion power recovery [33]. In this method, the expander output power in the main refrigeration cycle is employed to drive a compressor of the auxiliary subcooling cycle, and the refrigerant at the outlet of condenser is subcooled by the subcooler, significantly promoting the performance of the system. It is reported that this system achieves much higher COP than the conventional VCC system, conventional mechanical subcooling system, and

Fig. 13 Schematic of the cycle with liquid-line/suction-line heat exchanger (Reprinted from [29], Copyright(1994), with permission from Elsevier)



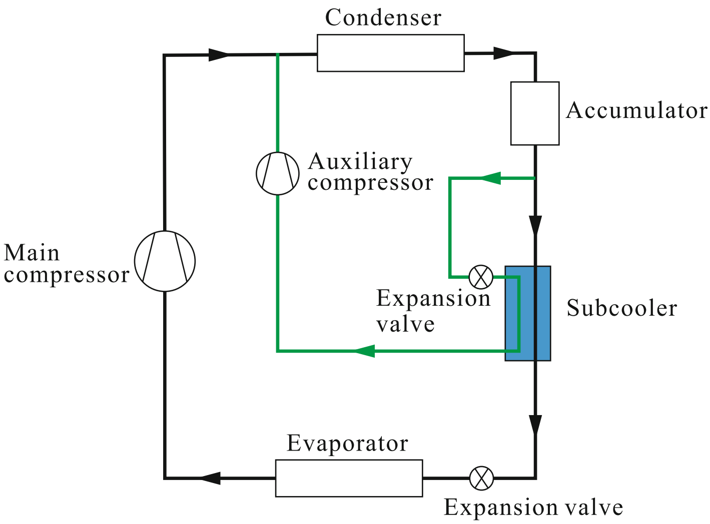


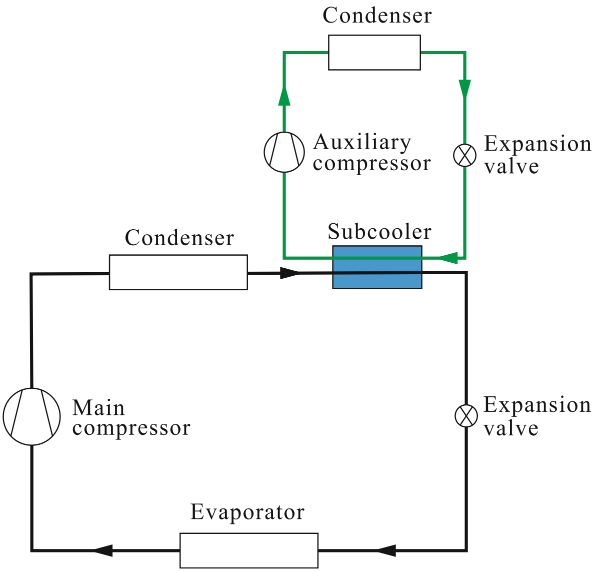
Fig. 14 Schematic of integrated mechanical subcooling (Reprinted from [31], Copyright(2000), with permission from Elsevier)

conventional expansion power recovery system, with maximum COP increments 67.76%, 19.27%, and 17.73%, respectively, when R744 works as the refrigerant in the main refrigeration cycle [33].

Multistage Cycle

The increase in the temperature differences between the condenser and the evaporator would give rise to the deterioration of the performance and reliability of the heat pump, which is the case where an ASHP runs at low ambient temperatures, for example, below −10 °C, for space heating. Several concepts have been proposed in previous literatures to improve the performance of ASHP at low ambient temperature [34–37], and one of the most effective concepts lies in the adoption of one of the

Fig. 15 Schematic of dedicated mechanical subcooling (Reprinted from [32], Copyright(2012), with permission from Elsevier)



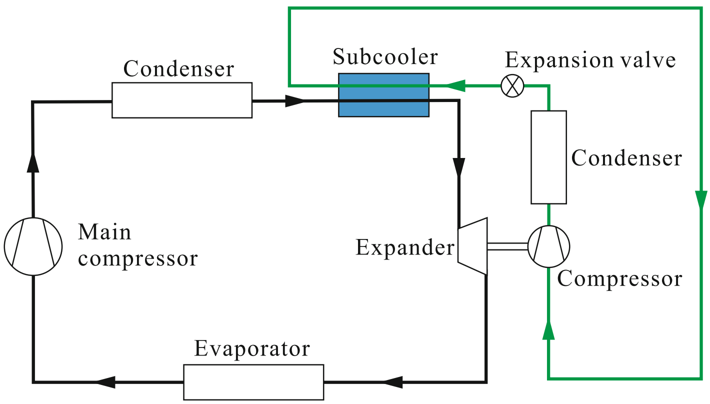
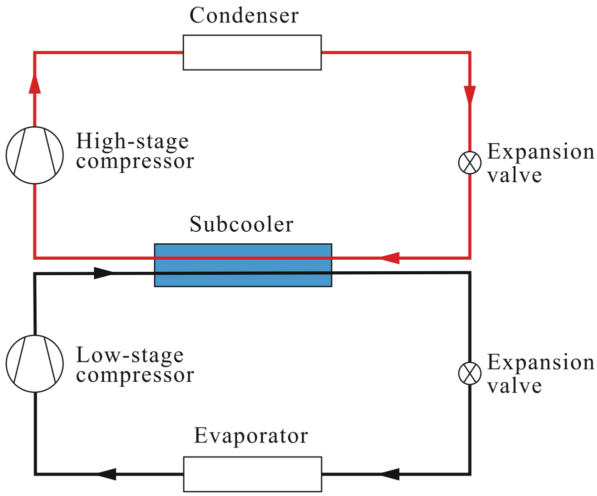


Fig. 16 Schematic of mechanical subcooling cycle based on expansion power recovery (Reprinted from [33], Copyright(2014), with permission from Elsevier)

two main types of multistage system: (a) staged compression or the (b) cascade system.

Unlike the staged compression system, the cascade system contains several separate refrigerant loops that are connected in series with the internal heat exchangers, as shown in Fig. 17. It is advantageous to use a cascade system when the temperature differences between the condenser and the evaporator is so large that it is difficult to find a refrigerant with suitable properties at both the condensing and evaporating temperature [38].

Fig. 17 Schematic of a cascade system with two circuits



In staged compression, the same refrigerant is used throughout the system and compressed in stages, which may be performed in one compressor or in physically distinct compressors. Interstage configurations, such as the flash intercooler, the flash tank, and the subcooler, are always implemented in staged compression, which will further promote the performance of the VCC system.

Experimental studies were carried out to evaluate the performance of a two-stage staged compression system under different operating conditions. For example, Tian and Liang [39] proposed and examined a two-stage compression variable frequency ASHP with a subcooler as the interstage configuration and experimental results demonstrated that the heating COP is higher than 2.0, the discharge temperature of the high-stage compressor was below 120 °C when the condensing temperature was 50 °C, and the evaporating temperature was −25 °C. Bertsch and Groll [34] designed, constructed, and tested a two-stage heat pump with subcooler for water and heating, which was able to operate at ambient temperatures between −30 °C and 10 °C with supply water temperatures of up to 50 °C. The system could run in either single-stage or in two-stage operating mode, and at the same ambient temperature, two-stage mode operation approximately doubles the heating capacity compared to single-stage mode operation. It is reported that a heating COP of 2.1 was observed at an ambient temperature of −30 °C. The discharge temperatures of the compressors in two-stage mode stayed below 105 °C at all times. Jin et al. [35] designed and tested a two-stage heat pump with flash tanks. Experimental results demonstrated that the system was capable of providing 50 °C heating hot water when outside temperature was −20 °C, with a heating capacity of 4.71 kW and COP of 1.76, and the discharge temperature of the high-stage compressor was below 100 °C.

There are several interstage configurations that can be implemented in a two-stage compression system, as presented in Fig. 18. All these cycles presented are corresponding to direct two-stage compression plants commercially used. Every cycle has its own merits and demerits. Take cycle (g) and (h), for example, one

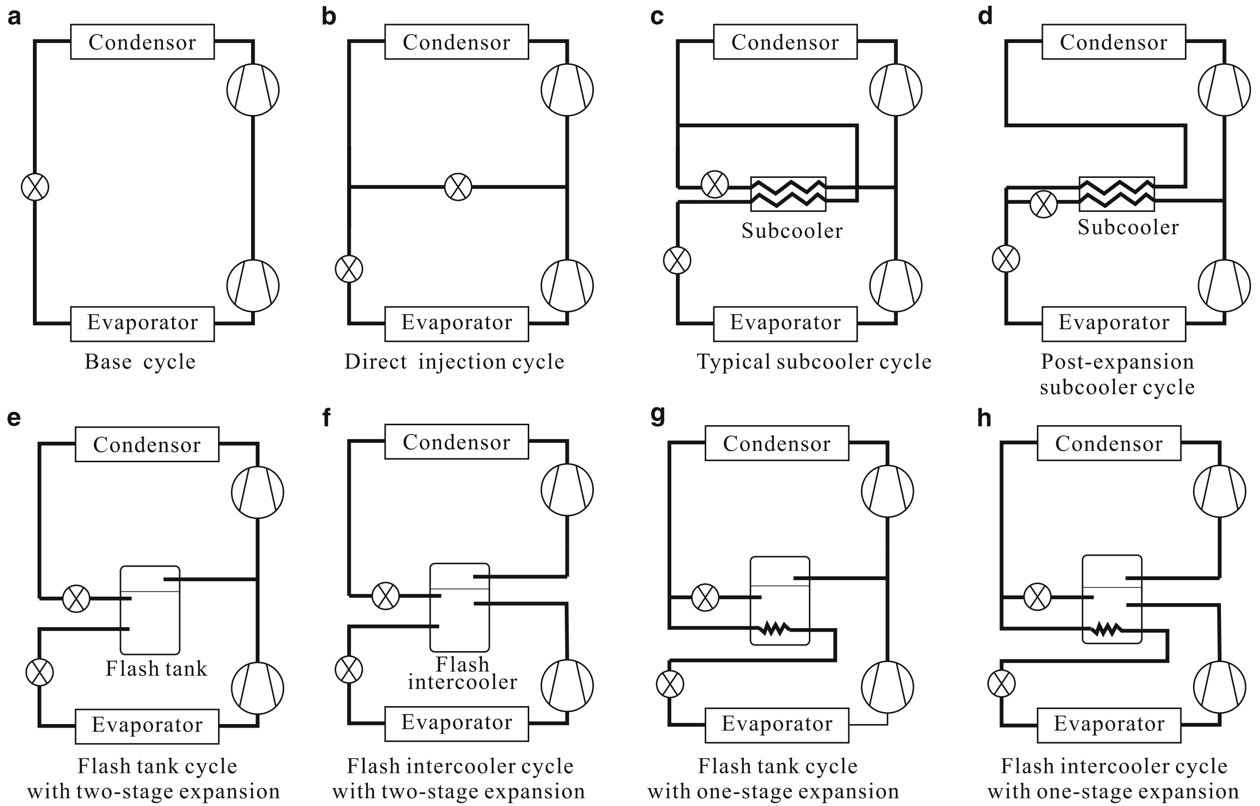


Fig. 18 Kinds of interstage configurations and the corresponding two-stage compression cycles (Reprinted from [40], Copyright(2015), with permission from Elsevier)

salient characteristic of such cycles lies in the refrigerant after interstage configuration is still at a high pressure. If the liquid must flow through a long line before it reaches the expansion valve, there is less possibility that the pressure drop in the line will flash the liquid to vapor and thus restricting the flow through the expansion valve. It is a great advantage when the evaporator is placed far away from the unit. Cycle (f) has a saturated suction state for the high-stage compressor; therefore, it is more suitable for the refrigerants with relatively high discharge temperatures (a consequence of the low specific capacity), such as ammonia, R32, etc. Cycle (d) is different with cycle (e) in the branching points of the injection stream. As a consequence, the performance will be different when the same subcooler is employed. Moreover, for cycle (d), injection stream starts after the main stream flows out from the subcooler. Subcooling provides 100% refrigerant liquid to enter the injection expansion device, preventing vapor bubbles from impeding the flow of refrigerant through the expansion valve; as a result, the injection stream is easier to show consistency in terms of its mass flow rate control for cycle (d).

Single compressors such as compound crew compressor and compound reciprocating compressor that can achieve two-stage compression have been used in refrigeration for years [41]. Schematic diagram of the system to achieve two-stage compression by using a single compressor is shown in Fig. 19. In terms of the compound crew compressors, the refrigerant liquid or vapor at an intermediate pressure is injected into the rotors and compressed along the compression process. As for compound reciprocating compressors, several cylinders perform the low-stage compression and the remaining cylinders accomplish the high-stage compression.

Recently, the twin rotary and scroll compressors are adopted in air-source heat pump systems, with an aim to solve the heating problems in cold climates [42, 43]. Schematic diagram of a twin rotary compressor is illustrated in Fig. 20. The refrigerant from the evaporator is firstly sucked and compressed by cylinder 1, then mixed with the injection refrigerant, and ultimately is sucked and compressed

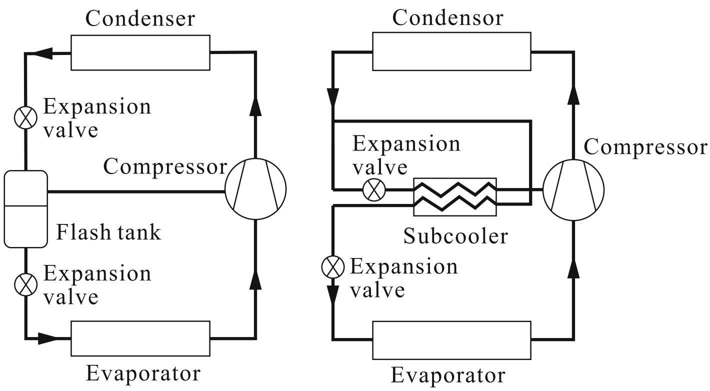


Fig. 19 Schematic diagram of the system to achieve two-stage compression by using a single compressor

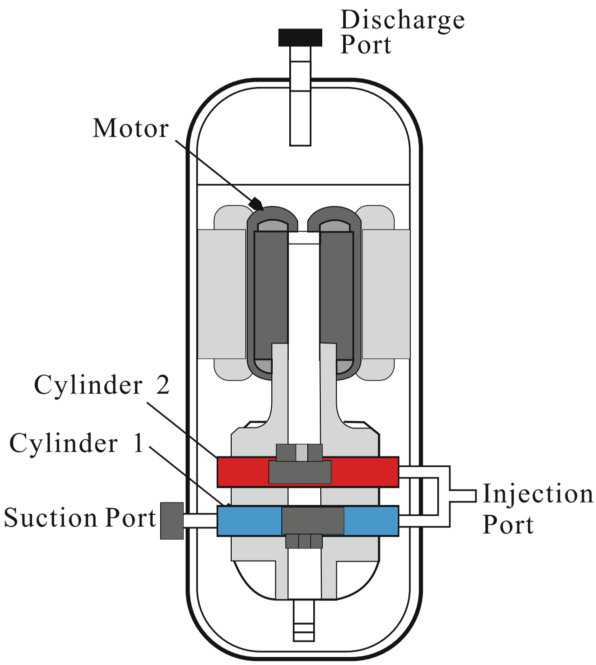


Fig. 20 Schematic diagram of a twin rotary compressor

by cylinder 2. The two cylinders are driven by a single electric motor. The vapor-injected scroll compressor has an injection port at its intermediate stage, allowing the refrigerant with intermediate pressures to enter the sealed compression chamber, as shown in Fig. 21. Wang et al. [44] conducted an experimental investigation on an 11 kW R410A heat pump system with a vapor-injected scroll compressor; a cooling capacity gain of around 14% with 4% COP improvement at the ambient temperature of 46.1 °C and about 30% heating capacity improvement with 20% COP gain at the ambient temperature of −17.8 °C were found as compared to the conventional system with the identical compressor displacement volumes.

Throttling Losses Recovery Cycle

The throttling process of a typical VCC is considered as an isenthalpic process leading to thermodynamic losses compared with an isentropic process. To recover the loss in an isenthalpic expansion process, it is better to utilize the devices capable of achieving isentropic condition or approximate isentropic condition in the throttling process, such as the expander or ejector, instead of the ordinary expansion/throttling structures described above.

The expander cycle is endowed with a great potential to improve the VCC performance, which attracts attention in heating and cooling systems, especially in CO_2 systems. Huff et al. [45] probed into the performance potential of CO_2 systems with expander and concluded that an ideal expander can promote the energy

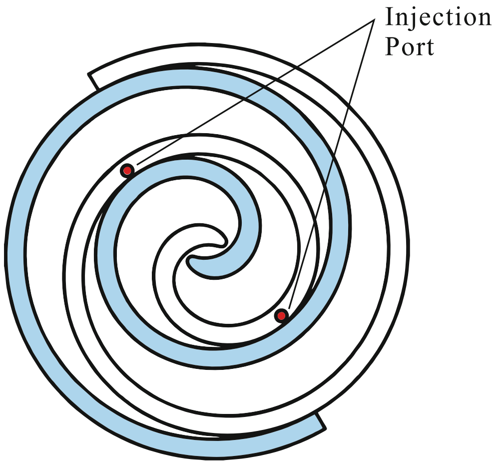


Fig. 21 Injection port of the vapor-injected scroll compressor

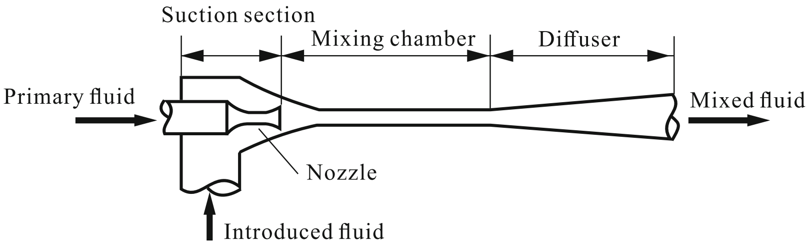


Fig. 22 Schematic of an ejector with cylindrical mixing chamber

efficiency of a transcritical CO_2 system under typical residential A/C operating conditions by 41% to 81%. Subiantoro and Ooi [46] performed an economic analysis of the installation of expanders on to existing VCC cooling systems. It was found that when the expander efficiency is 50%, the payback periods of most of the conventional systems are below 3 years in high-temperature countries with high electricity tariffs and are above 5 years in other countries; expanders are especially attractive for the transcritical CO_2 and the R404A systems.

There are several options for the expander cycle to extract the work from the expander [47]: (1) The shaft of the expander is combined with that of the compressor. (2) The generated electricity from a generator connected to the expander can be used for the compressor. (3) The multiple expanders can be utilized to recover the expansion loss more efficiently. The limitation of the expander cycle lies in its relatively low efficiency.

An ejector shows another way in work recovery. The ejector principally consists of a nozzle, mixing chamber, and diffuser, as shown in Fig. 22. The main fluid with high-pressure level passes through the nozzle, where its velocity increases to a very high level, maybe reaching the local sound velocity. If a diffuser is connected to the nozzle, the fluid velocity may reach supersonic velocity in the diffuser. The static

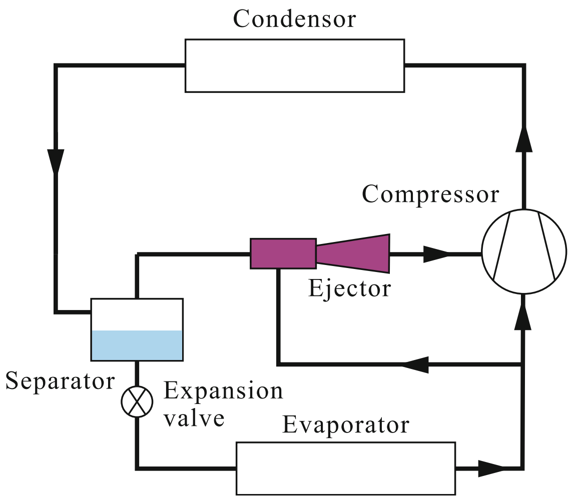


Fig. 23 Two-stage compression system coupled with ejector (Reprinted from [49], Copyright(2011), with permission from Elsevier)

pressure around the exit of the nozzle becomes lower than the introduced fluid. As a result, the introduced fluid is sucked and then fully mixed with the main fluid in the mixing chamber. The mixed fluid slows down in the diffuser and recovers its static pressure.

Reference [48] introduces two types of ejector cycle. The first cycle is a standard ejector cycle in which the ejector is treated as another type of "expander." The ejector transforms expansion losses into kinetic energy, sucks refrigerant from the evaporator, and then restores to an increased pressure, which should be a part of the compressor's work. The compressor continues to compress the refrigerant with higher pressure at the exit of the ejector so as to reduce the compression work. The other one is a dual evaporator ejector cycle, in which the exit of the ejector is connected with a high-temperature evaporator. The merits of this system lie in the fact that different temperature levels of heat sources can be used in practice. Experimental results suggested that maximum COP improvements of 12% with R1234yf and 8% with R134a can be reached by using this system when compared to the normal two expansion valves cycle [48].

The ejector can also be coupled with two-stage system to further improve the system performance. Figure 23 illustrates the ejector system with ejector before flash tank. Xu and Ma [49] explored this system and reported that the exergetic efficiency can be improved about 3–5%. Another type of system with ejector before the flash tank was analyzed and experimentally tested by Yu [50] with similar conclusions.

The ejector efficiency plays a crucial role in improving the performance of the system. When designing an ejector, its geometry strictly hinges on the design conditions. However, the operating condition may vary from time to time for an ASHP, and at off-design conditions, the efficiency of the ejector may drop significantly. Therefore, it is of great significance to design an ejector with wide range of operation, such as an ejector with adjustable geometry parameters.

Multifunction Cycle

Multifunction systems are always developed for improving the energy performance of the traditional heat pump technologies when special demands are required.

Figure 24 shows a two-stage compression ASHP with a subcooler as the interstage configuration, which can be operated in either single- or two-stage mode [51]. Moreover, the two compressors in the system are characterized by variable frequency. The system and pressure-enthalpy (ln(p)-h) diagram for this two-stage cycle are presented as well. For this system, the solenoid valve is switched on for the two-stage mode and off for the single-stage mode. In the single mode, the second EEV is closed, and either of the two compressors can be selected as the operation compressor. The four-way valve is opened only in cooling or defrosting operation. ON-OFF table for this system in shown in Table 3. This system shows the high modulation capacity of the two-stage compression technology and is suitable for space heating and cooling in cold climates.

Figure 25 illustrates the system diagram of an ASHP with heat recovery. The system is composed of a compressor, two four-way valves, two water-to-refrigerant heat exchangers, an air-to-refrigerant heat exchanger, three EEVs, three check valves, one high-pressure accumulator, and one low-pressure accumulator. By switching ON-OFF the four-way valves and the EEVs, any two of the total three heat exchangers can be selected as evaporators or condensers, and the other one serves as the condenser or evaporator. For example, the air-to-refrigerant heat exchanger (EX1) serves as an outdoor unit, one of the two remaining heat exchangers (EX2) supplies cooling water for air conditioning usage, and the other one (EX3) connects to a hot water tank. For the air conditioning project, EX3 serves

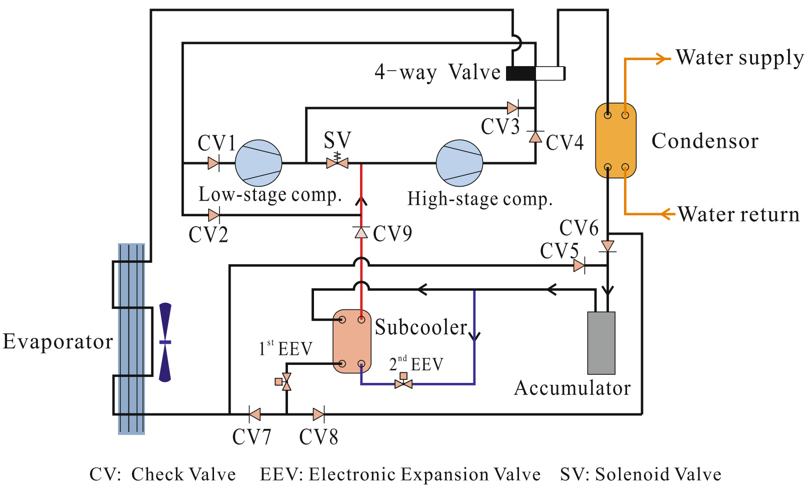


Fig. 24 System diagram of the two-stage air-source heat pump designed for space heating, which can be operated in both single and two-stage mode (Reprinted from [51], Copyright(2016), with permission from Elsevier)

Table 3 ON-OFF table for the heat pump unit under different operation modes

			Four-way valve	*High-stage comp.*	*Low-stage comp.*	*Solenoid valve*	*2nd EEV*
Heating	*Single-stage*	*High-stage comp.*	○	●	○	○	○
		Low-stage comp.	○	○	●	○	○
	Two-stage		○	●	●	●	●/○
Cooling	*Single-stage*	*High-stage com.*	●	●	○	○	○
		Low-stage com.	●	○	●	○	○
	Two-stage		●	●	●	●	●/○

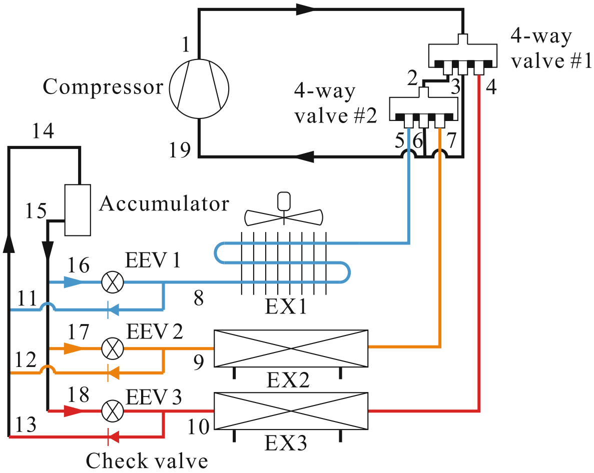


Fig. 25 System diagram of an ASHP with heat recovery

as the condenser to produce hot water when hot water demand exists (the flow direction of the refrigerant is 1-4-10-13-14-15-17-9-7-6-19, with four-way valve #1 ON, four-way valve #2 ON, and EEV1 OFF), and EX1 switches to serve as the condenser when the temperature of the hot water in the tank reaches the set value (the flow direction of the refrigerant is 1-2-5-8-11-14-15-17-9-7-6-19, with four-way valve #1 OFF, four-way valve #2 OFF, and EEV3 OFF). In the seasons when no cooling but only hot water demand exists, EX1 serves as the evaporator, and EX3 serves as the condenser, which is a typical air-to-water type ASHP system (the corresponding flow direction of the refrigerant is 1-4-10-13-14-15-16-8-5-6-19, with four-way valve #1 ON, four-way valve #2 OFF, and EEV2 OFF). Another example of its application is to build a dual-source heat pump system, a combination of an ASHP and a geothermal heat pump. In this case, EX1 is used as the heat exchanger of the air source, and EX2 is used as the heat exchanger of the geothermal source. EX3 supplies hot water for space heating or domestic use. The control system judges

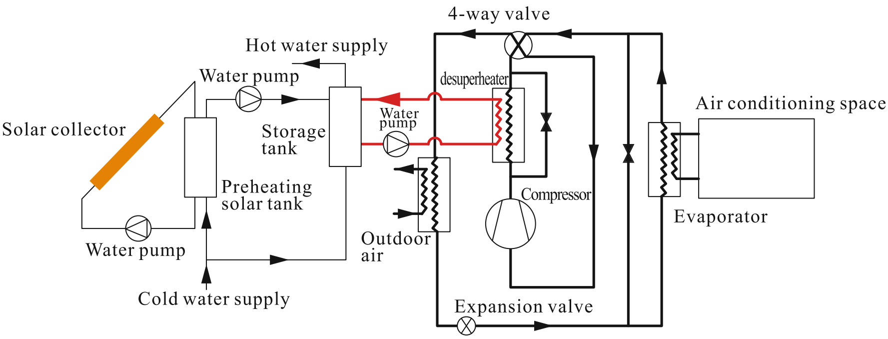


Fig. 26 A multifunctional solar-assisted air-source heat pump system (Reprinted from [52], Copyright(2009), with permission from The Hong Kong Polytechnic University)

and chooses the better source and makes it as the evaporator to obtain better performances.

Figure 26 sheds light on a multifunctional solar-assisted air-source heat pump system [52]. This system is endowed with two cycling loops, i.e., the solar collector loop and the multifunctional ASHP circuit, which are coupled together with a storage hot water tank. In summer, when the solar radiation is not strong enough, the superheated refrigerant vapor in the desuperheater sequentially heats the hot water; on sunny days, the bypass valve 1 is opened so that the required hot water is only obtained through the solar collector loop. In mild seasons, when air conditioning is not necessary and the solar radiation is not strong enough, the valve 2 is switched on so that the ASHP unit becomes a typical ASHP water heater.

Saturation Cycle

Lee et al. first proposed the saturation cycle in 2013 [53]. Before that, the performances of the scroll compressor with multiple vapor injection port and the cycle with continuous refrigerant injection have been investigated by Mathison et al. [54, 55]. A saturation cycle may be achieved by multistage two-phase refrigerant injection cycle. Figure 27 illustrates the schematic diagram of four-stage vapor compression cycle with two-phase refrigeration injection. Figure 28 shows the saturation cycle in the ln(p)-h diagram. In the saturation cycle, the compression process follows the saturated vapor line. Although this process is not as good as the isothermal compression process, it is a practical ideal compression process since the compressor handles only vapor phase for reliability reasons [53]. Moreover, the expansion process follows the saturated liquid line in the saturation cycle, which is superior to the isentropic expansion and can be achieved by an ideal expander expansion process or an ejector expansion process [53].

When the saturation cycle is applied, the COP and its improvement of the R123 cycle are higher than any other refrigerants studied (including R134a, R22, R410A,

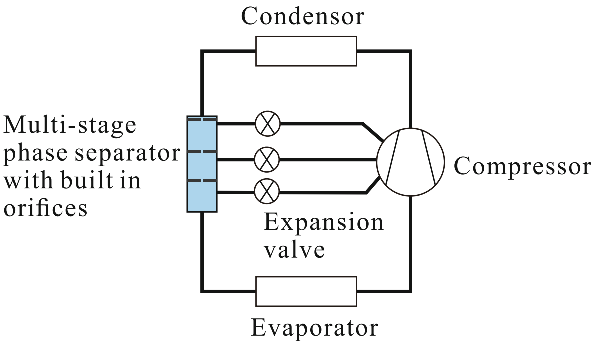


Fig. 27 Schematic diagram of four-stage vapor compression cycle with two-phase refrigeration injection (Reprinted from [53], Copyright(2013), with permission from Elsevier)

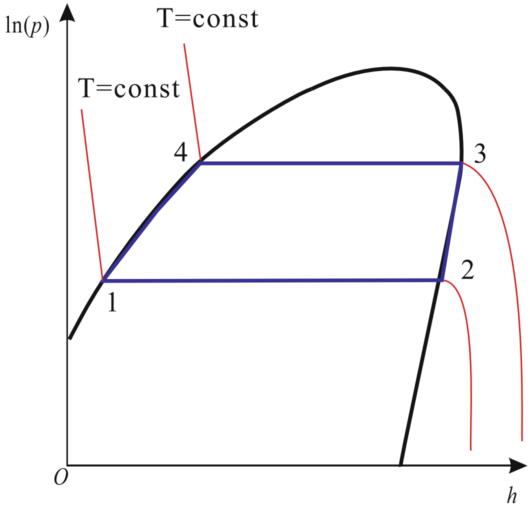


Fig. 28 Saturation cycle in the ln(p)-h diagram (Reprinted from [53], Copyright(2013), with permission from Elsevier)

R717, R290, R60a, R1234yf, R1234ze, R41, R152a, R32/R152a(95/5), R32, R32/R134a(95/5)), and the improvements of the cooling and heating mode are over 32.5 and 25.9%, respectively [53].

Frost-Free System

The problem of frosting in an outdoor unit constitutes a major obstacle in promoting and developing the ASHP system in a cold region. In order to overcome this shortcoming of the ASHPs, Wang et al. [56] developed a novel frost-free air-source heat pump water heater system coupled with an extra heat exchanger coated by a solid desiccant (EHECSD) with an energy storage device. Figure 29 illustrates the frost-free system, which consists mainly of a rotary compressor (1), overload protection devices (2, 14), a four-way valve (3), a condenser which wraps around outer of the water tank (4), solenoid valves (5, 13, 15, 18), an energy storage device

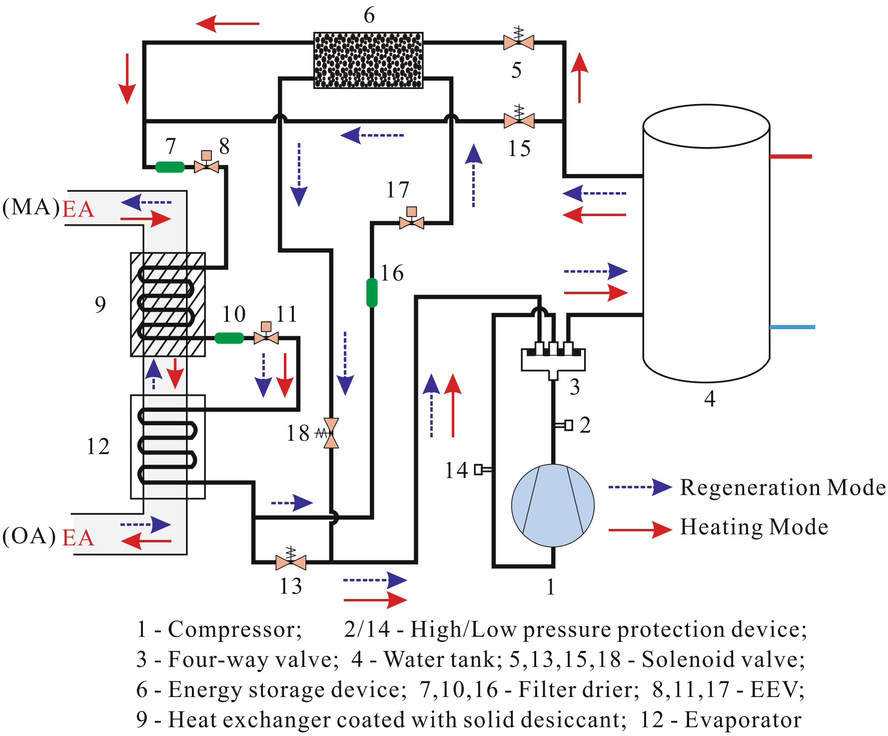


Fig. 29 Schematic diagram of a frost-free ASHP water heater combined with energy storage and dehumidification (Reprinted from [56], Copyright(2015), with permission from Elsevier)

(ESD) (6), filter driers (7, 10, 16), electronic expansion valves (EEVs) (8, 11, 17), an EHECSD (9), and an outdoor heat exchanger (12). In the heating mode, the EHECSD absorbs the water vapor of the outdoor air for dehumidification so that the defrosting problem will not happen as long as the dew point temperature of the dehumidified dry air is lower than the evaporating temperature. The system is switched to the regeneration mode when the vapor partial pressure of the solid desiccant is higher than that of the outdoor air. In the regeneration mode, the discharged refrigerant from the compressor flows into the condenser first and then the EHECSD where the water vapor is removed by the preheated outdoor air. The ESD acts as an evaporator in the regeneration mode. The system kept the evaporator frost-free for 32, 34, and 36 min during heating mode at the ambient temperatures of −3 °C, 0 °C, and 3 °C and 85% RH. Compared with the reverse-cycle defrosting (RCD), COP of the frost-free ASHPWH are 17.9% and 3.4% higher at the ambient temperature of −3 °C and 3 °C, respectively. It was concluded that this system can realize continuous heating and excellent performance at a low ambient temperature.

Li et al. [57] proposed a novel frost-free air-source heat pump (FFASHP) system, which can realize heating while does not need defrost in winter. The schematic

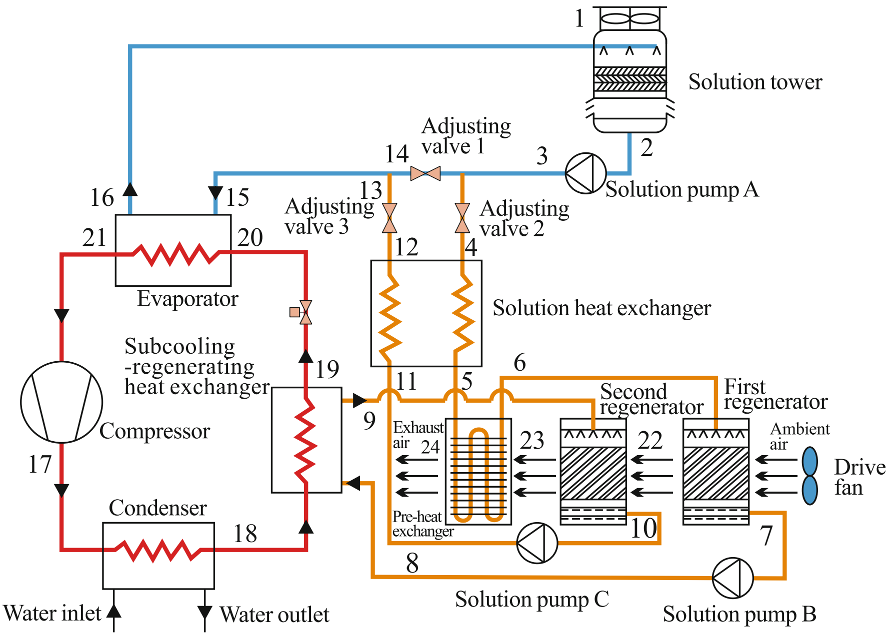


Fig. 30 Schematic diagram of the frost-free ASHP system (Reprinted from [57], Copyright(2011), with permission from Elsevier)

diagram of the FFASHP system is shown in Fig. 30. The system is composed of three subsystems: a compression refrigeration subsystem, a solution endothermic subsystem, and a solution regeneration subsystem. The compression refrigeration subsystem consists of the compressor, the condenser, the subcooling-regenerating heat exchanger, the expansion valve, and the evaporator. The compression refrigeration subsystem absorbs heat from the solution and releases heat to the hot water. The subcooling-regenerating heat exchanger provides heat for the regeneration of the solution. The solution endothermic subsystem, coupled with the compression refrigeration subsystem by the evaporator, is composed of the solution tower, the solution pump A, and the adjusting valve 1. The solution endothermic subsystem absorbs heat from the outdoor air. The solution regeneration subsystem, coupled with the compression refrigeration subsystem by subcooling-regenerating heat exchanger, is composed of the adjusting valve 2, the solution heat exchanger, the preheat exchanger, the first regenerator, the second regenerator, and the solution pumps B and C. The solution regeneration subsystem increases the concentration of the solution. Based on some analysis results, Li et al. [57] concluded that even the ambient air temperature reaches −20 °C, the system still operates at an acceptable performance, and the FFASHP system can be operated continuously and needs not to run in defrosting mode periodically, thus broadening the application of ASHP.

Summary

This chapter introduces the basic characteristics and the other issues worthy of attention in the design and application of an ASHP system, as well as some improved cycles to promote the system performance.

An ASHP, which uses the widely available air as heat source, is more easily deployed and applied than other types of heat pumps, such as geothermal heat pumps. Nevertheless, the performance of the ASHP changes hour by hour all year round because of the fluctuations in both temperature and humidity of the atmosphere air. Moreover, the unavoidable frosting phenomenon and the corresponding defrosting measure will result in deteriorated performance of an ASHP when compared with its normal operations. As a result, firstly, it is a little more complicated to rate an ASHP than other types of heat pump, and for a complete evaluation of an ASHP, the essay takes into account the combinations of some efficiencies, such as the heating coefficient of performance (COP_H), the simultaneous heating and cooling coefficient of performance ($COPS_{HC}$), or the heating season performance factor (HSPF). Secondly, the inconsistency of the heating demand of a building and the heating capacity of the ASHP emerge as an important problem for the design of an ASHP heating project, calling for the determination of the optimal heating balance point. Corresponding to the targets of minimizing energy consumption and maximizing economy, the heating optimal economic balance point (OECT) and heating optimal economic balance point (OECT) are used to choose the optimal heating balance point.

The installation of an ASHP is not as complicated as other types of heat pump. Air flow and noise control for both indoor and outdoor units serve as the two principal aspects calling for attention.

In the case that an ASHP is applied in cold areas where the outdoor temperatures are extremely low, the evaporating temperature decreases, and the temperature difference between the heat source and sink increases, leading to the deteriorated heating capacity, declining coefficient of performance, and even abnormal shut down due to the high discharge temperature. The last part of this chapter explores diverse approaches regarding how to promote the thermodynamic performance of an ASHP as well as to diversify functions of the system, including the subcooling cycle, the multistage cycle, the throttling loss recovery cycle, the multifunction cycle, the saturation cycle, and the frost-free system. Some of these approaches have been used in commercial products for years, whereas some are only in the experimental stage.

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