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A new method for preventing air-source heat pumps and refrigerators from frosting

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Abstract

In this study, we proposed a new method to prevent air-source heat pumps and refrigerators from frosting, in which a desiccant-coated heat exchanger (DCHE) is used to dehumidify the air before it enters the evaporator, consequently retarding frosting on its surface. Theoretical and experimental studies were conducted for the proposed frost-free heat pump and frost-free household refrigerator-freezer. Simulation results show that the proposed systems have high energy efficiency, while experimental results verify that frost-free operation could be realized and the DCHE could be regenerated under the condensation temperature of the proposed systems.

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Keywords: heat pump, refrigerator, frost, desiccant, dehumidification;

1. Introduction

Air-source heat pumps (ASHPs) are a promising technology, which is widely applied as an economic form of heating. However, one of the largest problems encountered is evaporator frosting and the subsequent need to defrost at a low ambient temperature and high relative humidity. Frost normally accumulates when the air temperature is -7 to 5.5ºC and the relative humidity exceeds 60%; a frequent atmospheric combination [1]. In fact, frosting commonly occurs when the air temperature is lower than -7ºC. The frosting-defrosting process causes significant problems, such as reduced energy efficiency and heating shutdown, which underlines the need to prevent or delay the frosting process when designing an air-source heat pump. Generally, defrosting can be achieved by supplying heat to the heat exchanger by various methods: electric resistance heater defrosting, warm-air defrosting [1], reverse cycle defrosting cycle [2-3] and hot-gas bypass cycle defrosting [4-7]. Defrosting using an electric resistance heater or warm air is not commonly employed in heat pump operations because heating from electrical resistive heaters is expensive. Reverse cycle defrosting is accomplished by reversing the normal heating mode. As for the hot-gas bypass defrosting method, the superheated refrigerant from the compressor flows directly to the evaporator; bypassing the condenser and expansive device and this approach is considered one of the most effective means of defrosting. However these defrosting methods result in heating shutdown and decrease system efficiency. Another disadvantage is that some melted water remains on the heat-exchanger surfaces. As the defrost cycle ends and the heat pump switches to heating mode, this water freezes to form a high-density frost, which is slow to melt during subsequent defrosts [1].

Furthermore, frost formation is a common phenomenon observed in refrigerators, including household refrigerator-freezers, refrigerated display cabinets, refrigerated warehouses and so on. The accumulation of frost on the evaporator surface of refrigerators increases thermal resistance and the air side pressure drops, resulting in higher electrical power input to the compressor and decreased refrigerating capacity. Modern refrigerators

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generally have an automatic mechanism to remove frost using an electrical resistance heater before their performance declines significantly. Although some energy-saving efforts have been made in the defrosting process of refrigerators, the process still requires a significant amount of energy. The efficiency of a defrost heater, defined as the ratio of total energy input to the energy required to melt the frost, was measured at 15 ~ 30% [8-9] in a household refrigerator-freezer, while the electrical power consumption of the refrigerator-freezer was found to increase by about 18% due to the automatic defrosting [8][10]. Bahman et al. reported that, depending on store size, humidity in store and the cabinet types, the energy associated with defrosting refrigerated display cabinets may exceed 15% of the total refrigeration electrical energy in a supermarket [11]. Furthermore, during defrosting, the compressor and fan remain off and part of the heat provided by the electrical heater is transferred to the refrigerated compartments. It was observed that the air temperature of the cabinet during the defrosting process rose from -1.57 to 6.8ºC for the refrigerator compartment (R. Compartment) and -23.1 to -11.8ºC for the freezer compartment (F. Compartment) [12] in a household refrigerator-freezer. This temperature fluctuation affects food quality and the compressor also has to run for longer to compensate for this extra thermal load after the defrosting process. The defrosted water runs through a tube and is collected in a tray, which is often located close to the compressor or condenser and finally evaporates from the tray due to the heat of the compressor or condenser. This system works satisfactorily in countries where the ambient air has low relative humidity and the amount of defrosted water is also low. However, under hot and humid ambient conditions, more defrosted water is generated, but the evaporation speed is slower, which may lead to water overflowing from the tray [13].

In our study, we focused on developing frost-free air-source heat pump and refrigeration systems, in which a desiccant is applied to remove moisture from the air entering the evaporators, thereby preventing frost formation. Theoretical and experimental studies of the proposed frost-free heat pump and refrigeration systems were conducted, the results of which are reported in this paper.

2. Proposed systems

2.1. Frost-free air-source heat pump (ASHP) system

In our study, we proposed a frost-free air-source heat pump (ASHP) for water heating applications. A schematic diagram of the frost-free ASHP water heater system is shown in Fig. 1, with the process illustrated on a pressure (*P*)-enthalpy (*h*) diagram of refrigerant in Fig. 2. This system comprises a desiccant-coated heat exchanger (1), an air heat exchanger (2), a compressor (3), a water heat exchanger (4), a water storage tank (5), a pump (6), two expansion valves (7, 8), a fan (9) and three valves (10, 11, 12). Note that the proposed frost-free ASHP can be applied to air-conditioning, in which the water heat exchanger (4) is replaced by an air heat exchanger and the water storage tank (5) is not necessary. Compared to a conventional ASHP water heater system, this system is equipped with a desiccant-coated heat exchanger that can dehumidify the outside air before it enters the air heat exchanger, consequently retarding frosting on its surface. However, the moisture absorption capacity of the desiccant will decrease and eventually expire over time, hence the need to desorb the desiccant. With respect to moisture moving in the desiccant, the operation process of the ASHP water heater system can be classified into two modes respectively: adsorption and desorption (AD and DE) modes.

While the system works in AD mode, the refrigerant is vaporized in two evaporators (1, 2) at two different temperature levels by controlling the two expansion valves (7, 8). Hot water is produced at the water heat exchanger (4). Valves (11, 12) are opened and valve (10) is closed. Outside air (OA) is blown in by a fan (9) across the desiccant-coated heat exchanger (1) and the air heat exchanger (2), then exhausted at state EA. The process air follows the two processes. OA→AA: OA is dehumidified at (1) and leaves as dry air (state AA). The adsorption heat is used to vaporize one part of the refrigerant. AA→EA: AA is then passed through (2), in which refrigerant is completely vaporized by obtaining sensible heat from AA before the air is finally exhausted at state EA. Because the dew point of the air (AA) entering the air heat exchanger is lower than the evaporation temperature of the refrigerant, frost-free operation could be realized in the system. When the moisture content of the desiccant becomes relatively high, so that the vapor partial pressure of the desiccant exceeds that of the OA, the desiccant should be regenerated, whereupon the system will be operated in desorption mode.

While in DE mode, the expansion valve (7) is opened and another expansion valve (8) throttled. Simultaneously, valve (10) is opened and valves (11, 12) closed to form an air cycle between (1) and (2). The refrigerant exiting compressor (3) initially removes part of the heat into the water at (4), then heats the return air (RA) to regenerate the desiccant at (1). Consequently, RA is heated and humidified to state DA (RA→DA). The

hot and humid air (DA) is then passed through (2), in which the refrigerant is vaporized and the DA is cooled and dehumidified to state RA (DA→RA). Therefore, the total heat load added to DA in the desiccant regeneration process, including sensible and latent heat, could be completely recycled in (2). This is one of the key characteristics of the proposed system, for which high system performance is thus expected. It should be noted that the evaporation temperature of the refrigerant in (2) must exceed 0°C while in DE mode. A single period is ended when this moisture adsorbed by the desiccant in AD mode is completely discharged in DE mode.

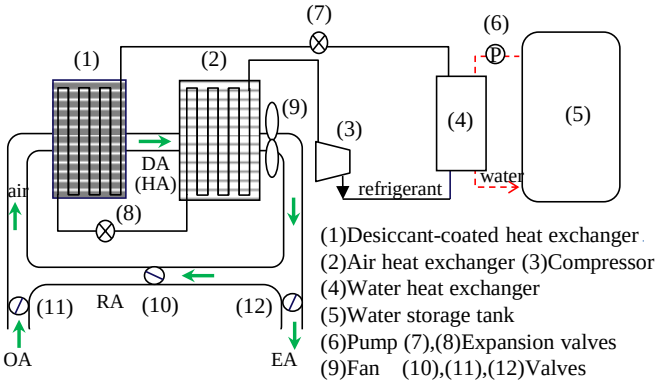


Fig. 1 Schematic diagram of the frost-free ASHP water heater system

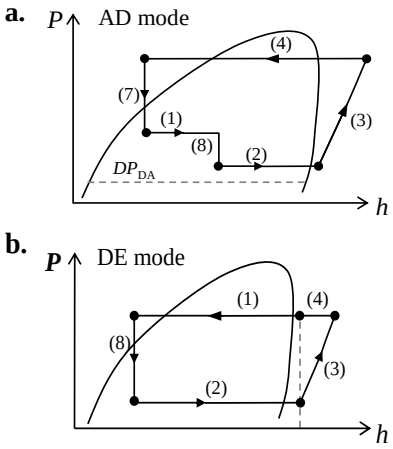


Fig. 2 P-h diagram of refrigerant

The coefficient of performance (COP) of this system, defined as the ratio of heat output to water (Q_w) to the electrical consumption of the compressor (W) over a single period, can be expressed as functions of $COP_{HP,ad}$, $COP_{HP,de}$, SHF_{ad} and SHF_{de} by Eq. (1)

$$\frac{1}{COP-1}=\frac{1}{COP_{HP,ad}-1}+\frac{1-SHF_{ad}}{1-SHF_{de}}\cdot\frac{1}{COP_{HP,de}-1} \quad (1)$$

Where, $COP_{HP,ad}=\frac{Q_{w,ad}}{W_{ad}}$; $COP_{HP,de}=\frac{Q_{w,de}+Q_{a,heat}}{W_{de}}$; $SHF_{ad}=\frac{Q_{a,from}^{sen}}{Q_{a,from}^{sen}+\Delta M_{vap}\gamma}$; $SHF_{de}=\frac{Q_{a,heat}^{sen}}{Q_{a,heat}^{sen}+\Delta M_{vap}\gamma}$,

$COP_{HP,ad}$ and $COP_{HP,de}$ are coefficients of performance of the heat pump unit in AD and DE modes, separately; SHF_{ad} and SHF_{de} are the sensible heat factors (SHF) of air in AD and DE modes, respectively. $Q_{a,from}$ is the heat obtained from outside air, [kJ]; $Q_{a,\,heat}$ is the heating load of air required to regenerate the desiccant, [kJ]; ΔM_{vap} is the total moisture transfer amount during the AD or DE process of a single period, [kg] and r is the vaporization latent heat of water, [kJ kg^{-1}]. The subscripts of "ad" and "de" express AD and DE modes. The superscript of "sen" means the sensible heat load of air. From Eq. (1), we can conclude that the COP of this system rises as the performance of the heat pump unit (COP_{HP}) improves. SHF also has a significant effect on COP: increasing SHF_{ad} in AD mode or decreasing SHF_{de} in DE mode results in increasing COP.

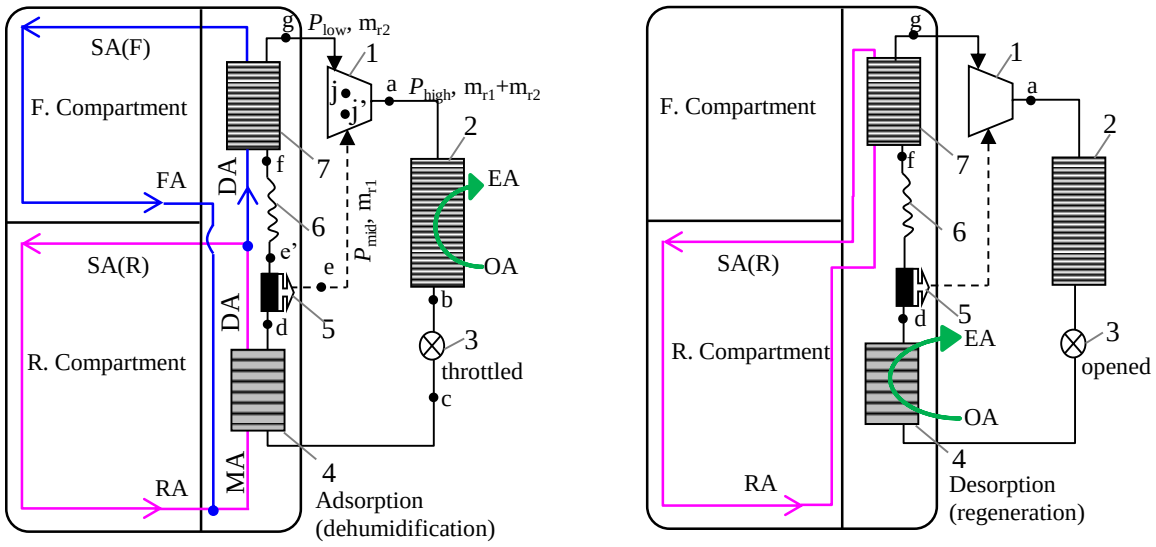
2.2. Frost-free refrigerator

The method of using a desiccant to prevent the evaporator from frosting can also be applied to refrigeration systems, such as household refrigerator-freezers, refrigerated display cabinets, refrigerated warehouses and so on. In our study, we proposed a frost-free household refrigerator-freezer (HR-F) and a frost-free refrigerated display cabinet (RDC). Here we introduce the frost-free HR-F system and the frost-free RDC can be referred to in our previous paper [14].

A schematic diagram of the frost-free HR-F is shown in Fig. 3, with the process illustrated on a pressure (P)-enthalpy (h) diagram of refrigerant in Fig. 4. The system comprises a compressor (1), an air-cooled heat exchanger (2), an expansion valve (3), a desiccant-coated heat exchanger (4), a gas-liquid separator (5), a capillary (6) and an air cooling heat exchanger (7). With respect to the available cooling load temperature and the moisture removing characteristic of the desiccant, the operation process of the system can be classified into two modes: Freezing-Refrigeration-ADsorption mode (F-R-AD mode) and Refrigeration-DEsorption (R-DE mode). One of the characteristics is the ability to regenerate the desiccant using the condensation heat of the refrigerant that would otherwise be simply ejected into the ambient air in a conventional refrigeration system. Accordingly, this system could not only retard evaporator frosting, but also achieve high energy efficiency by regenerating the desiccant with the system's exhaust heat.

While the system is in the F-R-AD mode, the expansion valve (3) is throttled, so the DCHE (4) and the air cooling heat exchanger (7) work as evaporators at two different evaporation pressures, P_{mid} and P_{low}, respectively and the air-cooled heat exchanger (2) works as a condenser. The refrigerant exiting the compressor (1) dissipates the condensation heat into the outside air (OA) at (2). After the refrigerant leaves (2), it is expanded to an intermediate pressure (P_{mid}) by traversing (3). The air (MA) mixed by the freezer air (FA) and refrigerator air (RA) is dehumidified at (4) due to the adsorption process of the desiccant. The dry air (DA) exiting (4) is split into two air streams: a portion of which (SA(R)) is supplied to the refrigerator compartment (R. Compartment) and the rest to (7) where it is further cooled and supplied to the freezer compartment (F. Compartment). The adsorption heat of the desiccant of (4) is used to vaporize one part of the refrigerant. The refrigerant from (4) flows into the gas-liquid separator (5), where the refrigerant liquid and vapor are separated. The saturated refrigerant liquid at point "e' " flows into (7) after it has been throttled to the required evaporator pressure (P_{low}) by traversing the capillary (6). The saturated refrigerant vapor at point "e" is then fed to the compressor (1) and compressed via a condensation pressure (P_{high}). At (7), the refrigerant is vaporized and the DA is cooled and leaves at the state of SA(F), which is supplied to the F. Compartment. The refrigerant from (7) at point "g" is fed back to the compressor (1) where two-stage compression is conducted: P_{low} to P_{mid} for the refrigerant, m_{r2}; and P_{mid} to P_{high} for the total refrigerant, $m_{r1}+m_{r2}$. When the water content of the desiccant becomes relatively high so that the vapor pressure of the desiccant exceeds that of the DA, the desiccant should be regenerated, whereupon the system will be operated in desorption mode.

During R-DE mode, the expansion valve (3) is opened, so the DCHE (4) works as a condenser and the air cooling heat exchanger (7) as an evaporator. The refrigerant exiting the compressor (1) traverses (2) and (3), where the temperature and pressure remain constant and flows into (4). At (4), the desiccant is heated by the condensation heat of the refrigerant and moisture moves from the desiccant to the outside air (OA). Consequently, the desiccant is regenerated and the outside air (OA) is heated, humidified and exhausted at state EA. After leaving (4) the condensed liquid refrigerant traverses the gas-liquid separator (5) and capillary (6) and flows into (7). It is noted that no refrigerant vapor can be separated at (5) during R-DE mode. The refrigerant evaporates at (7) and the refrigerator air (RA) from the R. Compartment is cooled and returned to the R. Compartment. Because the evaporation temperature of the refrigerant of (7) is controlled to keep it higher than the dew point of RA, frost-free operation is also feasible in R-DE mode.



(a) F-R-AD mode

(b) R-DE mode

1- Compressor,
2- Air-cooled heat exchanger,
3- Expansion valve,
4- DCHE,
5- Gas-liquid separator,
6- Capillary,
7- Air cooling heat exchanger

F. Compartment- Freezer compartment,
R. Compartment- Refrigerator compartment,
FA- Freezer air, RA-Refrigerator air,
MA- Mixed air, DA-Dry air,
OA- Outside air, EA-Exhausted air,
SA(F)- Supplying air to F. compartment,
SA(R)- Supplying air to R. compartment

Fig. 3 Schematic diagram of the frost-free household refrigerator-freezer

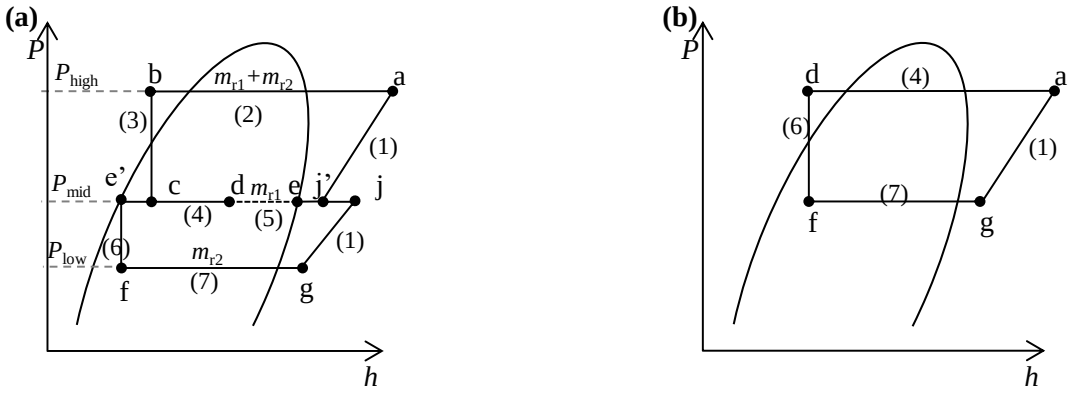


Fig. 4 P-h diagram of refrigerant

The coefficient of performance of this system (COP), defined as the ratio of total cooling load of R. and F. compartments (Q_c) to the electrical consumption of the compressor (W) in a single period, can be expressed as,

$$COP = COP_{ad} \cdot \gamma \tag{2}$$

Focusing on Eq. (2), we found that the COP of this system comprises two parts: one is COP_{ad}, the coefficient of the performance of the refrigeration cycle during F-R-AD mode, in which the freezing and refrigeration loads are supplied under frost-free operation conditions because of the adsorption process of the desiccant; another is coefficient γ, which refers to the influence of the desiccant desorption process. We found that γ is within the range 0.9 ~ 1 in most cases [15].

3. Theoretical study of the proposed systems

A detailed description of the mathematical models of the two proposed systems could be found in [14] and [16]. The following section introduces the main simulation results:

3.1. Simulation results of the frost-free air –source heat pump (ASHP)

The main parameters of the analytical calculation are shown in Table 1. Calculations were carried out for a frosting atmospheric air conditions: air temperature (T_{oa}) of -7 - 5ºC, air relative humidity (RH_{oa}) of 60 - 80%. Inlet and outlet water temperatures ($T_{w,in}$ and $T_{w,out}$) are 5 and 65ºC, separately. Hot water heating load (Q_w) is 4.5kW. The adiabatic efficiency of the compressor (η_{com}) is assumed to be 0.75. Thermal effectiveness (η_T) and humidity effectiveness (η_x), defined by Eqs. (3) and (4), are used to calculate the heat and mass (moisture) transferring loads of the DCHE. The operation time of AD and DE modes (τ_{ad} and τ_{de}) are 30 and 10 minutes. CO_2 is used as a working fluid in this heat pump system. The thermophysical properties of CO_2 were calculated using REFPROP7.0 (NIST) [17]. The evaporation temperature of the heat pump is determined to be 7ºC lower than T_{oa}. The minimum temperature difference method [19] is adopted to determine the discharge pressure of the compressor (P_{high}) in the current work.

$$\eta_T = \left| \frac{T_{a,in} - T_{a,out}}{T_{a,in} - T_e} \right| \tag{3}$$

$$\eta_X = \left| \frac{X_{a,in} - X_{a,out}}{X_{a,in} - X_e} \right| \tag{4}$$

where, T_e represents the ideal outlet air temperature [ºC], that equals the evaporation or condensation temperature of the refrigerant in the heat pump, respectively. X_e, the air humidity ratio in equilibrium with the desiccant [g (kg dry air)$^{-1}$], is a function of the water content (C) and temperature (T_d) of the desiccant. $T_{a,in}$ and $T_{a,out}$ are the inlet and outlet air temperatures of the DCHE, [ºC] and $X_{a,in}$ and $X_{a,out}$ are the inlet and outlet air humidity ratios of the DCHE, [g (kg dry air)$^{-1}$].

Table 1. Parameters of the analytical calculation

Parameters	T_{oa}	RH_{oa}	$T_{w,in}$	$T_{w,out}$	Q_w	η_{com}	η_t	η_x	$\Delta\tau_{ad}$	$\Delta\tau_{de}$
Values	-7~5ºC	60~80%	5ºC	65ºC	4.5kW	0.75	0.8	0.8	30min	10min

Figure 5 shows the calculation results of four factors influencing the system COP: SHF_{ad} of 0.73, SHF_{de} of 0.28, COP_{ad} of 3.78 and COP_{de} of 6.5, under air conditions of T_{oa}=2ºC and RH_{oa}=80%. Substituting the four factors to Eq. (1), the system COP is calculated as 3.37. Figure 6 shows the influences of temperature (T_{oa}) and relative humidity (RH_{oa}) of the atmospheric air on the system COP. It is found that RH_{oa} has a significant effect on COP: the higher the RH_{oa}, the lower the COP. COP rises with increasing T_{oa} due to the evaporation temperature of the heat pump rising with increasing T_{oa}. Furthermore, Fig. 6 shows a comparison of COP between the frost-free ASHP water heater system proposed in this work and a hot–gas defrosting ASHP water heater system. The COP of the hot-gas defrosting ASHP water heater system is calculated by assuming that the performance of the heat pump system will be reduced by 20% [18] compared to a base ASHP water heater system [19], which operates at the same atmospheric air temperature, but low relative humidity (i.e. non-frosting conditions). Results show that the COP of the proposed frost-free ASHP water heater system is higher than that of the hot-gas defrosting ASHP water heater system for any frosting temperature and humidity condition. In particular, the COP of the frost-free ASHP water heater system is increased by 25 - 30% under low-temperature (-7 ℃) or low relative humidity (60%) conditions compared to the hot-gas defrosting ASHP water heater system.

3.2. Simulation results of the frost-free household refrigerator-freezer

Simulations were carried out under the operating conditions shown in Table 2. The air flow rates of the refrigerator (R.) and freezer (F.) compartments (m_{ra} and m_{fa}) are 0.1 and 0.3 m³/min. The air temperature and relative humidity in R. and F. compartments are (4ºC, 50%) and (-18ºC, 50%), separately. The adiabatic efficiency of the compressor (η_{com}) is assumed to be 0.7, while the thermal effectiveness (η_T) and humidity effectiveness (η_x) of the DCHE are 0.8. The operation times of the AD and DE modes (τ_{ad} and τ_{de}) are 7.5 and 0.5 hours, individually. Isobutane (R600a) is used as a working fluid in this system. The thermophysical properties of R600a were calculated using REFPROP7.0 [17].

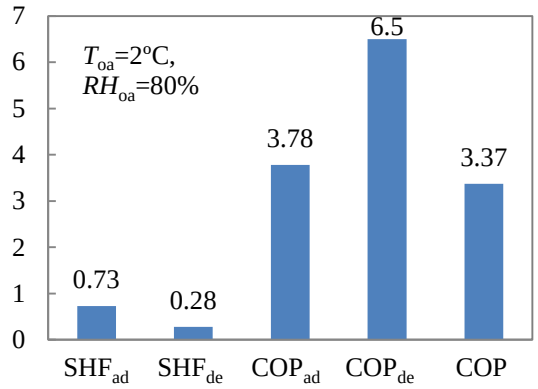


Fig. 5 Calculation results under the conditions of T_{oa}=2ºC and RH_{oa}=80%

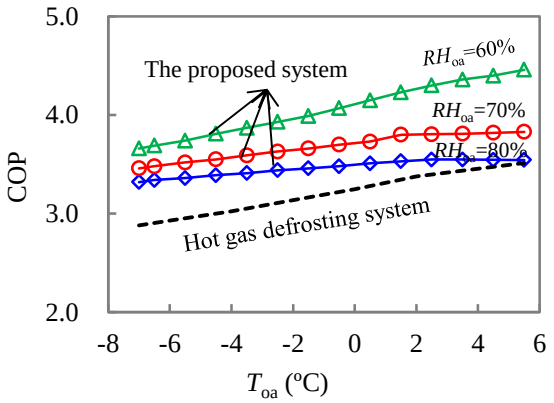


Fig. 6 Effect of T_{oa} and RH_{oa} on system COP

Table 2 Parameters of the analytical calculation

Parameters	m_{RA} m³/min	T_{RA} [ºC]	RH_{RA} [%]	m_{FA} [m³/min]	T_{FA} [ºC]	RH_{FA} [-]	T_{OA} [ºC]	RH_{OA} [-]	η_{com} [-]	η_x [-]	η_t [-]	τ_{ad} [h]	τ_{de} [h]
Values	0.1	4	50	0.3	-18	0.5	26	0.5	0.7	0.8	0.8	7.5	0.5

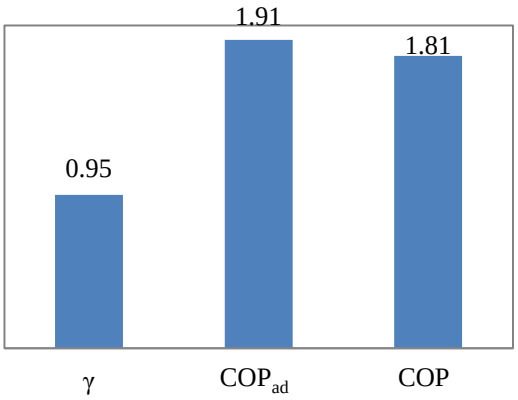


Fig. 7 Calculation results of γ, COP_{ad} and COP

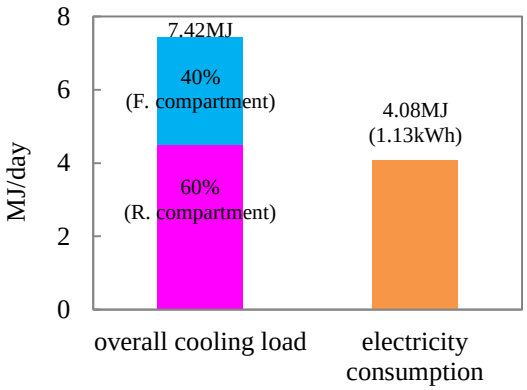


Fig. 8 Overall cooling loads of R. and F. compartments and the electric energy consumption for one day

Figure 7 shows the calculation results of the two factors those influence the system COP: γ of 0.95 and COP_{ad} of 1.91. Substituting the two factors to Eq. (2), the system COP is calculated as 1.81.The overall cooling load handled by the refrigeration system and electric power consumed by the compressor for one day are calculated and the results are shown in Fig. 8. Among the overall cooling load of 7.42MJ/day, the cooling load of R. compartment has a ratio of 60%. The electric power consumption of the compressor for one day is 1.13 kWh.

4. Experimental study of DCHEs

The results of the theoretical analysis show that the two proposed systems can prevent evaporators from frosting and have high energy efficiency. In the following, we want to confirm experimentally whether a frost-free state could be realized and whether the DCHE could be effectively regenerated under the condensation temperature of heat pumps and refrigerators.

4.1. Experimental apparatus

A schematic diagram and photo of the experimental apparatus are shown in Fig. 9, comprising an air loop, two brine loops: cooling brine and hot brine loops, a test section, an air rectification box and a convergence chamber located separately upstream and downstream of the test section and some measurement instruments. DCHE, meanwhile, is installed inside the test section. Air flows through the DCHE and contacts the desiccant to exchange moisture with it, while cooling or hot brine inside the tubes is used to remove the adsorption heat during the AD process, or heat the desiccant to regenerate it during the DE process. In this study, two DCHEs, one coated by AQSOA (a kind of desiccant belonging to the zeolite family, developed by Mitsubishi Chemical Corporation) and another coated by polymer sorbent (developed by Japan Exlan), were assessed experimentally. The physical characteristics of the two DCHEs are shown in Table 3.

During adsorption experiments (AD Exp.), valves (V1 and V2) were opened and valves (V3 and V4) were closed, so that the cooling brine was passed through the DCHE. The process air, conditioned to the required temperature and humidity using a constant temperature and humidity air supplier, traversed the DCHE and returned into the air supplied. The process air was dehumidified due to the DCHE adsorption process, whereupon the adsorption heat was dissipated into the cooling brine. The air leaving the DCHE had a lower dew point than that of the air at the DCHE inlet. When the humidity ratios of the air at the inlet and outlet of the DCHE became equal, AD Exp. was finished. Subsequently, valves (V1 and V2) were closed, while others (V3 and V4) were opened, so that the hot brine was passed through the DCHE. Desorption experiment (DE Exp.) was started, during which the desiccant released its moisture into the air. As a result, the desiccant is regenerated and the air leaving the DCHE has a humidity ratio that exceeds that of the inlet air. When the humidity ratios of the air at the inlet and outlet of the DCHE became equal, DE Exp. was finished. AD and DE experiments were continuously conducted with at three cycles per test and the experimental data on the second or third cycle were used in the following data reduction.

Table 3 Physical characteristics of the DCHEs

Parameters	DCHE (AQSOA)	DCHE (polymer sorbent)
Type of heat exchanger (HE)	Aluminum fin and copper tube HE	All aluminum, plate tube HE with corrugated fins
Height of the DCHE-H (m)	0.242	0.2
Length of the DCHE-L (m)	0.3	0.5
Width of the DCHE-W (m)	0.0508	0.016
Fin pitch-P_f (m)	1.8×10^{-3}	1.2×10^{-3}
Mass of dry desiccant-$M_{d,dry}$ (kg)	0.756	0.0878

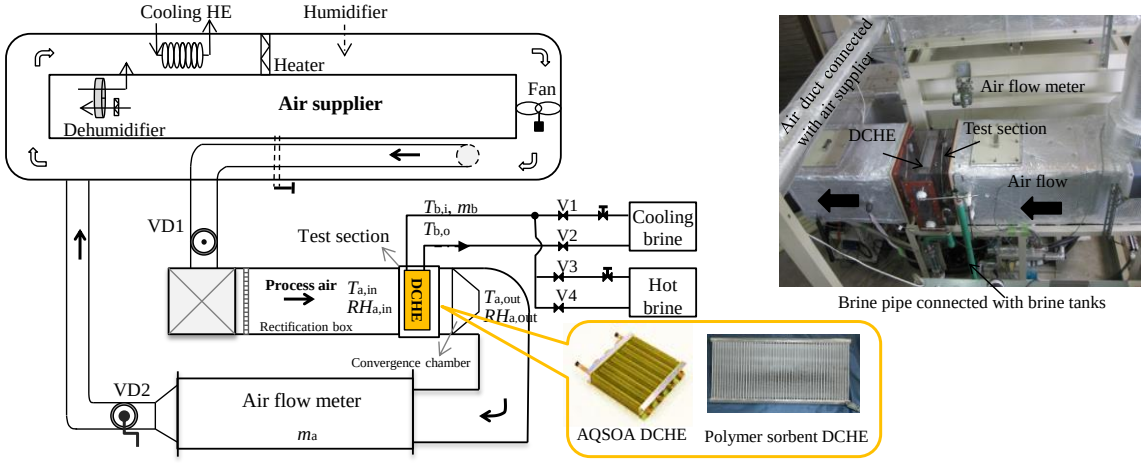


Fig. 9 Experimental apparatus

4.2. Experimental apparatus

· *Whether can frost-free be realized?*

Two experimental results corresponding to the two DCHEs, under the typical frosting air conditions of air-source heat pumps in Japan: T_{oa}=2ºC and RH_{oa}=80% and the air temperature and humidity of freezer compartment of household refrigerator-freezers: T_{oa}=-18ºC and RH_{oa}=60%, were shown in Figs. (10) and (11), separately.

The evaluation criteria of the frost-free operation of the proposed heat pump and refrigerator are given as,

$$DP_{a,out} < T_{eva} \tag{5}$$

where, $DP_{a,out}$ is the dew point of the DCHE outlet air and T_{eva} is the evaporation temperature of the heat pump or the refrigerator. Here, we assumed that T_{eva} is 7ºC and 10ºC lower than the DCHE inlet air temperature of the heat pump and refrigerator, individually.

From Fig. 10, the experimental result of the AQSOA-DCHE under heat pump operation condition, we found the evaluation criteria were satisfied and the operation time satisfying the evaluation criteria was about 20 minutes. The experimental result of the polymer sorbent-DCHE under freezer operation condition, is shown in Fig. 11. It is verified that the polymer sorbent is capable of adsorbing moisture from the air of -18ºC and the operation time satisfying the evaluation criteria of the frost-free freezer is about 30 minutes.

· *Whether can the desiccant be regenerated under the condensation temperature?*

The AD experimental results shown in Figs. 10 and 11 were obtained after the DCHEs were regenerated by passing 55ºC (the same temperature levels as the condensation temperature of the heat pump) and 43ºC (the

same temperature levels as the condensation temperature of the refrigerator) hot brines through the DCHEs,

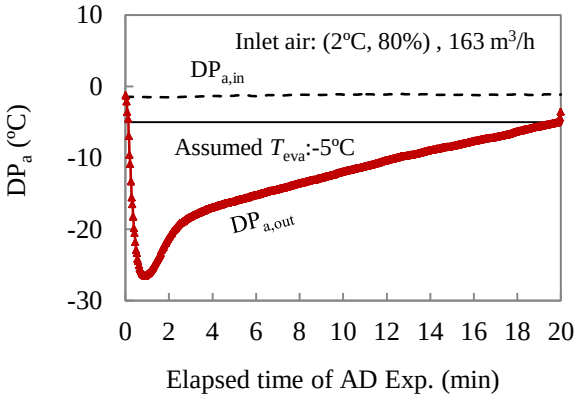


Fig. 10 Experimental result of AD Exp. of AQSOA-DCHE for frost-free heat pump application

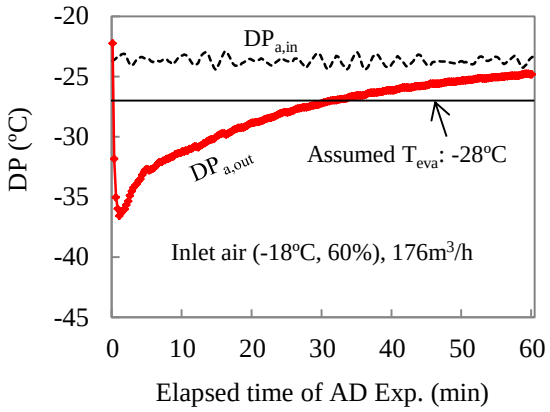


Fig. 11 Experimental results of AD Exp. of polymer sorbent DCHE for frost-free refrigerator application

separately. Therefore, it is verified that the two kinds of desiccants: AQSOA and polymer sorbent, could be regenerated under the condensation temperature and are suitable for the two proposed frost-free systems.

5. Conclusions

This paper introduced a proposed frost-free air-source heat pump and frost-free household refrigerator-freezer. Theoretical study shows that the two proposed systems have high energy efficiency. Furthermore, it was also experimentally verified that the frost-free operation of the proposed systems could be realized and DCHE regenerated under the condensation temperatures of heat pumps and refrigerators.

Nomenclature

AD – adsorption
ASHP – air-source heat pump
COP – coefficient of performance
DCHE – desiccant-coated heat exchanger
DE – desorption
HR-F – household refrigerator-freezer
SHF – sensible heat factor

References

[1] Ameen F. R., Coney J. E., Sheppard C. G. W. Experimental study of warm-air defrosting of heat-pump evaporators. Rev. Int. Froid. 1993; 16 (1): pp. 13-18.
[2] Ding Y. J., Ma G. Y., Q. H., Jiang Y. Experiment investigation of reverse cycle defrosting methods on air source heat pump with TXV as the throttle regulator. Int. J. Refrig. 2004; 27: pp. 671-678.
[3] O'Neal D. L., Peterson K. T., Anand N. K. Effect of short-tube orifice size on the performance of an air source heat pump during the reverse-cycle defrost. Int. J. Refrig. 1991; 14 (1): pp. 52-57.
[4] Byun J. S., Lee J., Jeon C. D. Frost retardation of an air-source heat pump by the hot gas bypass method. Int. J. Refrig. 2008; 31: pp. 328-334.
[5] Tso C. P., Wong Y. W., Jolly P. G., Ng S. M. A comparison of hot-gas by pass and suction modulation method for partial load control in refrigerated shipping containers. Int. J. Refrig. 2001; 24 (6): pp. 544-553.
[6] Hoffenbecker N., Klein S. A., Reindl D. T. Hot gas defrost mode development and validation, Int. J. Refrig.2005; 28 (4): pp. 605-615.
[7] Yaqub M., Zubair S. M., Khan J. R. Performance evaluation of hot-gas by-pass capacity control schemes for refrigeration and air–conditioning system. Energy 2000; 25: pp. 543-561.
[8] Pradeep, B., David, F., Ryan, F.. Thermal analysis of the defrost cycle in a domestic freezer. Int. J. Refrig. 2010; 33: pp. 589-599.

[9] Niederer, D. H. Frosting and defrosting effects on coil heat transfer, ASHRAE Transactions 1986; 82(1): pp. 467-473.
[10] Knabben, F. T., Hermes, C. J. L., Melo, C. 2011. Numerical and experimental investigation of the frosting and defrosting processes in no-frost evaporators. XI Ibero-American Congress of Refrigeration and Air conditioning – CIAR, Mexico.
[11] Bahman, A., Rossario, L., Rahman, M. M. Analysis of energy savings in a supermarket refrigeration/HVAC system. Applied Energy 2012; 98: pp. 11-21.
[12] Zakrzewskib,. Lokietek T., Hrycyk E. 2011. Defrosting of air cooler in refrigerating appliances and heat pump. The 23rd IIR International Congress of Refrigeration, Prague Czech Republic.
[13] Xie, G., Bansal, P. K. Analysis of defrosted water evaporation from three water trays in refrigerators. Appl. Therm. Eng. 2000; 20: pp. 651-669.
[14] Zhang L., Fujinawa T., Saikawa M. Theoretical study on a frost-free refrigerated display cabinet. Int. J. Refrig. (Accepted), http://dx.doi.org/10.1016/j.ijrefrig.2016.09.027
[15] Zhang L., Fujinawa T., Saikawa M. Theoretical study on a frost-free household refrigerator –freezer. Int. J. Refrig. 2016; 62: pp. 60-71.
[16] Zhang L., Fujinawa T., Saikawa M. A new method for preventing air-source heat pump water heater from frosting. Int. J. Refrig. 2012; 35: pp. 1327-1334.
[17] NIST reference fluid thermodynamic and transport properties database: version 7. 2007.
[18] Byun J. S., Lee J., Jeon C. D. Frost retardation of an air-source heat pump by the hot gas bypass method. Int. J. Refrig. 2008; 31: pp. 328-334.
[19] Neksa P., Rekstad H., Zakeri G. R., Schiefloe P. A. CO_2 –heat pump water heater: characteristics, system design and experimental results. Int. J. Refrig. 1998; 21 (3): pp. 172-179.