Optimization and performance experiments of a $\text{MnCl}_2/\text{CaCl}_2\text{-NH}_3$ two-stage solid sorption freezing system for a refrigerated truck

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Abstract: Based on the earlier-established $\text{MnCl}_2/\text{CaCl}_2\text{-NH}_3$ two-stage solid sorption freezing system for a refrigerated truck, a series of optimization designs are conducted in this prototype system. For sorption beds consisting of many unit tubes, the arrangement mode is changed to the staggered arrangement. Off-the-shelf heat exchangers from refrigeration industries are chosen as the evaporator and the condenser, and an expansion valve is also used. The total mass of the optimized system is reduced to approximately 150 kg. Firstly, different refrigerating temperatures ranging from -25°C to 0°C are investigated, and experimental results show the optimized system can easily satisfy requirement even for transporting frozen goods. The earlier-established system can only satisfy requirement for transporting fresh goods. Simultaneously, the cycle time can be reduced to 45 min. Through the optimization, both the refrigerating capacity and the total mass of the system can satisfy requirement of this refrigerated truck.

Keywords: optimization; solid sorption freezing cycle; two-stage; refrigerated trucks; exhaust gas

Nomenclature

A	heat exchange area, m^2
C	specific heat, $\text{kJ kg}^{-1} \text{K}^{-1}$
h	convective heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$
K	total heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$

L	latent heat of evaporation, kJ kg^{-1}
l	length of the unit tube, m
M	total mass, kg
m	mass flow rate, kg s^{-1}
P	pressure, MPa
Pr	Prandtl number
Q	heat transfer rate, W
R	fouling resistance, $\text{m}^2 \text{K W}^{-1}$
r	radius, m
Re	Reynolds number
S	spacing between two unit tubes
T	temperature, K
t	time, s
ΔH	sorption reaction heat, kJ kg^{-1}
ΔT	logarithmic mean temperature, T
Δx	cycle sorption quantity, kg kg^{-1}
λ	thermal conductivity, $\text{W m}^{-1} \text{K}^{-1}$
ε	correction factor

Subscripts

a	sorption equilibrium
ave	average
c	condensation
chilled air	chilled air that flows through the evaporator
cyc	cycle
des	desorption
e	evaporation

exh	exhaust
ext	external
g	desorption equilibrium
heat	sorption heat or desorption heat
in	inlet
inn	inner
ins	instantaneous
met	metallic materials
out	outlet
re	resorption
ref	refrigerant
sor	sorbent
tube	unit tube

Abbreviations

AV	ammonia valve
CFD	computational fluid dynamics
COP	coefficient of performance
CV	air valve
DT	temperature change
EAV	expansion ammonia valve
EV	exhaust valve
ENG-TSA	expanded natural graphite treated with sulphuric acid
HTS	high-temperature salt
min	minute

MTS	middle-temperature salt
PM	particulate matter
SCP	specific cooling power per kilogram sorbent
RC	refrigerating capacity

1. Introduction

In recent decades, people's living standards have been improved greatly, and the demand for fresh and frozen food is increasing rapidly, which contributes to the rapid development of refrigerated trucks. Refrigerated trucks are utilized to carry fresh, frozen and perishable goods, and refrigerators are essential for refrigerated trucks.

At present the predominant technology is mechanical vapor compression refrigeration driven by a diesel engine, which certainly will increase greenhouse gas and particulate matter (PM) emissions. A typical refrigerator produces approximately 50 tons of CO₂ annually with a fuel consumption of 0.47 L h⁻¹ per 1 kW refrigerating capacity, operating at an ambient temperature of 30°C and a refrigerating temperature of -20°C, and the CO₂ emission will increase more than 30% after six years and more than 50% after nine years (Chatzidakis and Chatzidakis, 2004). For internal combustion engines, only 30-40% of the total heat generated from fuel combustion is converted to useful mechanical work, so the remaining energy is expelled directly to the environment through the engine cooling system and the exhaust gas, which account for 30% and 30%-40% of the total heat, respectively (Jadhao and Thombare, 2013).

For the engine cooling water, the temperature at outlet under mapping characteristics can reach up to 90°C, and for the exhaust gas immediately leaving the engine, the temperature can reach 450°C-600°C (Mohamed et al., 2015). Considering that the engine waste heat has a relatively high temperature and a large amount of heat, the solid sorption refrigeration cycle that is driven by the low-grade heat source can

be used to recover the waste heat and provide refrigerating capacity (Wang and Oliveira, 2006; Pang et al., 2013; Goyal et al., 2016).

Much work about adsorption air-conditioning systems powered by waste heat from internal combustion engines has been done (Ali and Chakraborty, 2015; Amir et al., 2015; Jiangzhou et al., 2005; Jribi et al., 2013, 2014; Lu et al., 2003; Ramji et al., 2014; Wang et al., 2006; Wang et al., 2007; Wang et al., 2008). Just as mentioned above, the waste heat is from the engine cooling water and the exhaust gas.

For the exhaust gas, some typical examples are as follows: Lu et al. (2003) and Jiangzhou et al. (2005) established a locomotive driver-cabin adsorption air-conditioner using zeolite-water as the working pair. Ramji et al. (2014) conducted the parametric investigation of a heat-driven adsorber for the air-conditioning system employing the activated carbon-methanol working pair, and the computational fluid dynamics (CFD) was utilized. Amir et al. (2015) analyzed the impact of the fin spacing on temperature distribution in the adsorption cooling system for vehicle air-conditioning applications. Ali and Chakraborty (2015) presented a two stage indirect exhaust heat recovery system of an automotive engine employing an effective lumped parameter model to simulate the dynamic behavior of an adsorption chiller for the car cabin, and the CaCl_2 -in-silica gel-water was the best candidate.

For the engine cooling water, some typical examples are as follows: Wang et al. (2007) built a split heat pipe type compound solid sorption ice making test unit for fishing boats, and the compound sorbent of CaCl_2 /activated carbon-ammonia was the working pair. Jribi et al. (2013, 2014) established a transient mathematical model of a 4-bed adsorption chiller using Maxsorb III as the adsorbent, and the refrigerants of CO_2 and R1234ze (E) were analyzed.

The researches mentioned above mostly focus on the adsorption air-conditioning system for which the refrigerating temperature is above 5°C , so they are not suitable for refrigerated trucks. It is because

the required refrigerating temperature of refrigerated trucks generally ranges from -18°C to 0°C . Based on the previous researches (Jiang et al., 2015; Wang et al., 2013), to make the sorption refrigeration cycle suitable for refrigerated trucks, Gao et al. (2016a, 2016b) proposed and established a $\text{MnCl}_2/\text{CaCl}_2\text{-NH}_3$ two-stage solid sorption freezing system driven directly by the exhaust gas, and composite sorbents of MnCl_2 and CaCl_2 developed by the expanded natural graphite treated with sulphuric acid (ENG-TSA) were utilized. The advantage of the two-stage sorption refrigeration cycle is that it can reduce the required heat source temperature (Alam et al., 2004; Khan et al., 2007), which make it suitable for the critical working conditions such as the low exhaust temperature at the low-medium load of the truck engine. Due to the employment of the $\text{MnCl}_2/\text{CaCl}_2\text{-NH}_3$ two-stage solid sorption freezing cycle, experimental results showed that the refrigerating capacity obtained could satisfy requirement for transporting fresh goods even under the low engine load condition in summer, but the system has some problems as follows:

(1) The system can reach the required refrigerating capacity only for transporting fresh goods at the refrigerating temperature of 0°C and -5°C . However, it is difficult to satisfy the required refrigerating capacity when the refrigerating temperature is lower than -10°C .

(2) The cycle time of the system is up to 240 min, and the resorption time is approximately 120 min. Because no refrigerating capacity is output during the resorption process, a reasonable resorption time is quite essential for improving the storage effect of the goods.

(3) The total mass of the system is relatively heavy, almost 260 kg.

To solve the above problems, a series of optimization designs are conducted in this paper and the optimized system is also established. Simultaneously, corresponding experiments have been carried out.

2. System description

The refrigerator on the truck is utilized to maintain the internal temperature of the cabin constant, so that the goods don't deteriorate during transportation. The required refrigerating temperature of refrigerated trucks generally ranges from -18°C to 0°C . In this paper, a refrigerated truck with a rated power of 80 kW is analyzed, and the cabin size is $4035 \times 1935 \times 1800$ mm. The rated load capacity of this refrigerated truck is 1500 kg, and the required refrigerating capacity is shown in Table 1 (Gao et al., 2016a, 2016b).

Table 1 Required refrigerating capacity under different refrigerating temperatures

Refrigerating temperature/ $^{\circ}\text{C}$	0	-5	-10	-15	-18
Required refrigerating capacity/kW	0.97	1.13	1.29	1.45	1.55

The schematic diagram of the $\text{MnCl}_2/\text{CaCl}_2\text{-NH}_3$ two-stage solid sorption freezing system is shown in Fig. 1a, which consists of a high-temperature salt sorption bed (HTS bed), a middle-temperature salt sorption bed (MTS bed), an evaporator, a condenser, an expansion valve (EAV), three ammonia valves (AV), two exhaust valves (EV), two air valves (CV) and a liquid storage tank. Ammonia is chosen as the refrigerant. Composite sorbents of MnCl_2 and CaCl_2 developed by the matrix of ENG-TSA are filled in the HTS bed and MTS bed, respectively.

(a)

Fig. 1 $\text{MnCl}_2/\text{CaCl}_2\text{-NH}_3$ two-stage solid sorption freezing system: (a) schematic diagram; (b) $\ln P$ - T diagram

The detailed working principles of the optimized system are as follows:

(1) Sorption process of the HTS bed and desorption process of the MTS bed. For this process, air valve CV1, exhaust valve EV2, expansion valve EAV, ammonia valve AV1 and AV2 are opened. The HTS bed is cooled by ambient air, and the MTS bed is heated by the engine exhaust gas. The working pressure of the HTS bed decreases gradually from P_{re} to P_e , and the working pressure of the MTS bed increases from P_{re} to P_c , as shown in Fig. 1b. The ammonia is desorbed from the MTS bed and condensed into liquid in the condenser. The liquid ammonia flows into the liquid storage tank, and then passes through the expansion valve (EAV). Consequently, the pressure of the liquid ammonia changes from P_c to P_e , and correspondingly, the temperature of the liquid ammonia decreases quickly from T_c to T_e . The low-temperature and low-pressure liquid ammonia flows into the evaporator and evaporates. The evaporation process of the ammonia inside the evaporator produces the refrigeration effect, and the chilled air transfers the refrigerating capacity into the cabin. Finally, the ammonia is sorbed by the HTS bed. Compared with the $\text{CaCl}_2\text{-NH}_3$ solid sorption cycle, for the evaporating temperature of T_e , the sorption temperature of the two-stage solid sorption cycle increases from the T_{a1}' to T_a (shown in Fig. 1b), which can adapt to the high ambient temperature at summer.

(2) Resorption process between the HTS bed and the MTS bed. For this process, air valve CV2, exhaust valve EV1 and ammonia valve AV3 are open. The HTS bed is heated by the engine exhaust gas, and the MTS bed is cooled by the ambient air. The ammonia is desorbed from the HTS bed and flows to the MTS bed. The MTS bed sorbs the ammonia and the reaction heat is released to the ambient air. Compared with the $\text{MnCl}_2\text{-NH}_3$ solid sorption cycle, for the condensing temperature of T_c , the desorption temperature of the two-stage solid sorption cycle decreases from T_{g1}' to T_{g1} (shown in Fig. 1b), which can adapt to the low heat source temperature, i.e., the low exhaust temperature at the low-medium load of the engine.

The two-stage solid sorption freezing cycle is the combination of the conventional sorption process and the resorption process, and the refrigerating capacity is output only at the sorption process of the HTS bed.

To solve problems encountered in the earlier-established system, some optimization designs are conducted, which are as follows:

For the sorption beds, they consist of many unit tubes, which are similar to the two-stage solid sorption freezing system previously established by Gao et al. (2016a, 2016b). In the earlier-established system, the cycle time is up to 240 min, which leads to the low average refrigerating capacity. The long cycle time is mainly due to the poor heat exchange between the air and unit tubes in the sorption bed. Through the numerical calculation by Fluent, the path lines of air in the earlier-established HTS bed are obtained, as shown in Fig. 2a. From Fig. 2a, it can be found that the effective heat transfer area of each unit tube is quite limited. To enhance heat exchange, the arrangement mode of unit tubes is changed from the previous aligned arrangement to the staggered arrangement. Similarly, through the numerical calculation by Fluent, the path lines of air in the optimized HTS bed are obtained, as shown in Fig. 2b.

From Fig. 2b, it can be found that the effective heat transfer area of each unit tube is enlarged significantly, and it is because each unit tube can be almost totally surrounded by air.

Fig. 2 Path lines of the air in the HTS bed: (a) the earlier-established HTS bed; (b) the optimized HTS bed

Composite sorbent is filled in the unit tube. The mass ratio between the metal chloride and the ENG-TSA is 5:1, and the packing density of the composite sorbent is approximately 500 kg m^{-3} . The length and the diameter of the unit tube are 460 mm and 42 mm, respectively, and the mass of the composite sorbent filled in each unit tube is about 240 g.

To predict the performance of the optimized sorption beds, corresponding thermodynamic and heat transfer models needs to be established. Considering that the sorption beds are composed of many unit tubes, a good method is to establish the thermodynamic and heat transfer models of one unit tube.

As mentioned above, the sorption bed is heated by the engine exhaust gas in the desorption process and cooled by the ambient air in the sorption process. The heat input for each unit tube in the desorption process is mainly composed of three parts, i.e., desorption heat of the composite sorbent, the sensible heat of the composite sorbent and the metallic materials of the unit tube. The average heat input of each unit tube in the desorption process $Q_{\text{ave, tube, heat}}$ is:

$$Q_{\text{ave, tube, heat}} = KA_{\text{tube}} \Delta T = \frac{1}{t_{\text{des}}} (M_{\text{tube, sor}} \Delta x_{\text{sor}} \Delta H_{\text{sor}} + C_{\text{sor}} M_{\text{tube, sor}} \Delta T_{\text{sensible}} + M_{\text{tube, met}} C_{\text{met}} \cdot \Delta T_{\text{met}}) \quad (1)$$

where $M_{\text{tube, sor}}$ is the mass of the composite sorbent filled in the unit tube, and $M_{\text{tube, met}}$ is the metallic mass

of the unit tube. Δx_{sor} and ΔH_{sor} are the cycle sorption quantity and reaction heat, respectively. Δx_{sor} is:

$$\Delta x_{\text{sor}} = \frac{\Delta M_{\text{tube,ref}}}{M_{\text{tube,sor}}} \quad (2)$$

where $\Delta M_{\text{tube,ref}}$ is the refrigerant mass that can be sorbed by the composite sorbent in the unit tube.

When the exhaust gas flows outside the tube, the total thermal resistance is composed of four resistances in series, namely, the convection between the exhaust gas and the external surface of the unit tube, a fouling resistance, the conduction across the metal tube, and the conduction across the composite sorbent. The equation is:

$$\frac{1}{K} = \frac{1}{h_{\text{exh}}} + R + \frac{A_{\text{tube}} \ln(r_{\text{tube}}/r_{\text{sor,ext}})}{2\pi\lambda_{\text{tube}}l} + \frac{A_{\text{tube}} \ln(r_{\text{sor}}/(r_{\text{sor,ext}} - r_{\text{sor,inn}}))}{2\pi\lambda_{\text{sor}}l} \quad (3)$$

where the λ_{sor} is the thermal conductivity of the composite sorbent, which is $18.7 \text{ W m}^{-2} \text{ K}^{-1}$ with the packing density of 500 kg m^{-3} (Jiang et al., 2014).

The convection heat transfer coefficient of the exhaust gas (Zhukauskas, 1986) is:

$$h_{\text{exh}} = \varepsilon \times 0.35 \times \left(\frac{S_1}{S_2} \right)^{0.2} \text{Re}_{\text{exh}}^{0.6} \text{Pr}_{\text{exh}}^{0.36} (\text{Pr}_{\text{exh}}/\text{Pr}_{\text{tube}})^{0.25} \times \lambda_{\text{exh}} / (2 \times r_{\text{tube}}), \frac{S_1}{S_2} \leq 2 \quad (4)$$

The average refrigerating capacity obtained from one unit tube:

$$Q_{\text{ave,tube,ref}} = \frac{\Delta M_{\text{tube,ref}} \times L}{t_{\text{cyc}}} \quad (5)$$

Based on the above mathematic models of the unit tube, the thermodynamic and heat transfer calculation of the unit tube is conducted by MATLAB. For the HTS bed, the number of unit tubes is reduced from 120 to 66, and for the MTS bed, the number of unit tubes is reduced from 80 to 55. Consequently, the total mass of the sorption beds is reduced dramatically.

Considering that the pressure drop of the staggered arrangement of unit tubes in the sorption bed is higher than that of the aligned arrangement, the spacing S_1 between two unit tubes is a very important

parameter for the pressure drop. To accurately determine the pressure drop between the air inlet and outlet of the sorption bed, a test rig is established, as shown in Fig. 3a. The wooden club is used as the unit tube, and the arrangement mode is shown in Fig. 3b. The air flow of the fan is adjusted by a voltage controller. The volumetric flow of the air is measured by the Pitot tube anemograph (ZC1000-1F), and its accuracy is 0.5%R. And the 0.5%R means that the measure error of the Pitot tube anemograph is the 0.5% of the reading. The pressure difference is measured by the micro difference pressure sensor (Setra 261C), and its accuracy is $\pm 0.4\%R$.

Different spacing of 52 mm and 57 mm is chosen, and test results are show in Fig. 4. Both the volumetric flow and the pressure drop increase with the rising voltage. Under the same voltage, the volumetric flow at the spacing of 52 mm is lower than that at the spacing of 57 mm, but the pressure drop at the spacing of 52 mm is higher than that at the spacing of 57 mm. When the voltage is 225 V, the volumetric flow at the spacing of 52 mm and 57 mm is $1514 \text{ m}^3 \text{ h}^{-1}$ and $1765 \text{ m}^3 \text{ h}^{-1}$, and the corresponding pressure drop is 559 Pa and 363 Pa. Considering that no fan is installed at the sorption bed in the system installed on the truck, the small pressure drop is beneficial for the system. As a result, the spacing of 57 mm is chosen. Consequently, the size of the HTS bed and MTS bed is $799 \times 520 \times 341 \text{ mm}$ and $699 \times 520 \times 341 \text{ mm}$. The optimized HTS bed is shown in Fig. 5a.

Fig. 3 Test rig for measuring pressure drop between the air inlet and outlet of the sorption bed: (a)

experimental system; (b) arrangement mode of unit tubes

Fig. 4 Test results vs. voltage: (a) volumetric flow; (b) pressure drop between the air inlet and outlet of the sorption bed

For the evaporator, a plate-fin type heat exchanger is utilized, and the base tube is made of aluminum. For the condenser, a parallel-flow heat exchanger with the characteristics of high efficiency and compactness is used. Both the evaporator and the condenser are off-the-shelf components of refrigeration industries, so they are bought from the company of SONGZ Automobile Air Conditioning CO., LTD. at a reasonable price, which decrease the cost of the system greatly.

Additionally, the expansion valve shown in Fig. 5b is firstly employed, and the number of liquid storage tanks is reduced from 2 to 1.

The installation of the sorption freezing system on the truck is shown in Fig. 5c. Two sorption beds are mounted on two sides of the truck beam, and the available space on the two sides of the truck beam is $1665 \times 625 \times 515$ mm and $1060 \times 625 \times 515$ mm, respectively. The condenser, the liquid storage tank and the evaporator are installed at the upper side of the cabin, which is similar to the mechanical vapor compression refrigeration system.

Fig. 5 $\text{MnCl}_2/\text{CaCl}_2\text{-NH}_3$ two-stage solid sorption freezing system: (a) HTS bed (MnCl_2 bed); (b) expansion valve; (c) installation of the system on the truck

3. Performance calculation

The average heat input $Q_{\text{ave, heat}}$ of the sorption bed in a cycle:

$$Q_{\text{ave, heat}} = \frac{\int_0^{t_{\text{cyc}}} C_{\text{exh}} \cdot m_{\text{exh}} \cdot (T_{\text{exh, in}} - T_{\text{exh, out}}) dt}{t_{\text{cyc}}} \quad (6)$$

The instantaneous refrigerating capacity $Q_{\text{ins, ref}}$:

$$Q_{\text{ins, ref}} = C_{\text{chilled, air}} \cdot m_{\text{chilled, air}} \cdot (T_{\text{chilled air, in}} - T_{\text{chilled air, out}}) \quad (7)$$

where the $T_{\text{chilled air, in}}$ and $T_{\text{chilled air, out}}$ are the chilled air inlet and outlet temperatures of the evaporator, and the latent heat term is not considered.

The average refrigerating capacity $Q_{\text{ave, ref}}$ in a cycle:

$$Q_{\text{ave,ref}} = \frac{\int_0^{t_{\text{cyc}}} C_{\text{chilled,air}} \cdot m_{\text{chilled,air}} \cdot (T_{\text{chilledair,in}} - T_{\text{chilledair,out}}) dt}{t_{\text{cyc}}} \quad (8)$$

The coefficient of performance:

$$\text{COP} = \frac{Q_{\text{ave,ref}}}{Q_{\text{ave,heat}}} \quad (9)$$

where the power consumed by the evaporator fan and the condenser fan is not considered.

The specific cooling power per kilogram of sorbent:

$$\text{SCP} = \frac{Q_{\text{ave,ref}}}{M_{\text{tot,sor}}} \quad (10)$$

4. Experimental results and discussion

The $\text{MnCl}_2/\text{CaCl}_2\text{-NH}_3$ two-stage solid sorption freezing experimental system after optimization is established, as shown in Fig. 6. For the heating process of sorption beds, the hot air heated by the electric heater, whose maximum power is 21 kW, is utilized to simulate the engine exhaust gas. The hot air temperature of the electric heater can be adjusted from the ambient temperature to 250°C with an accuracy of $\pm 3^\circ\text{C}$. Two centrifugal fans are utilized to cool two sorption beds separately to simulate the cooling process of the sorption beds when the truck is running. To consume the refrigerating capacity, the electric heater controlled by the temperature controller is installed in the cabin. Temperature sensors (PT100) and pressure transmitters (YSZK-31) are mounted in the experimental system; their accuracies are $\pm 0.1^\circ\text{C}$ and 0.3%R, respectively. The flow rate of the air is measured by the Pitot tube anemograph (ZC1000-1F). In addition, a high-pressure sight glass is installed at the liquid storage tank. A data acquisition unit of the Agilent 34970A is utilized for the collection of the data.

Fig. 6 $\text{MnCl}_2/\text{CaCl}_2\text{-NH}_3$ two-stage solid sorption freezing experimental system

To determine the performance of the optimized system, different refrigerating temperatures of 0°C , -5°C , -10°C , -15°C , -20°C and -25°C , different resorption time of 20 min, 30 min and 40 min, and different hot air temperatures of 210°C and 230°C are investigated. Meanwhile the performance of the optimized system is compared with that of the earlier-established system.

The air inlet and outlet temperatures of the sorption beds under a typical operational condition are shown in Fig. 7a. The hot air temperature and the ambient temperature are 210°C and 15°C . For the HTS bed, from 0 min to 30 min the resorption process between the HTS bed and the MTS bed occurs, and from 30 min to 65 min the sorption process of the HTS bed occurs. As mentioned in Section 2, the sorption process of the HTS bed and the desorption process of the MTS bed occur simultaneously. But for the MTS bed, after 56 min the liquid level in the liquid storage tank is almost constant according to the sight glass, which means that there is no ammonia desorbed from the MTS bed to the condenser and the desorption process of the MTS bed is completed. After that the MTS bed is precooled by ambient air right now, which is beneficial for the following resorption process.

The working pressure of the sorption beds is shown in Fig. 7b. For the HTS bed, because the

resorption happens between MnCl_2 bed and CaCl_2 bed, the working pressure is at a relatively low value ranging from 0.083 MPa to 0.479 MPa. Based on the equilibrium lines of chlorides reacting with ammonia (Neveu and Castaing, 1993), the relatively low working pressure corresponds to the relatively low equilibrium temperature; i.e., such a process effectively decreases the required heat source temperature. Meanwhile, for the MTS bed, because desorption occurs between the sorption bed and the condenser, the working pressure is much higher and ranges from 0.110 MPa to 0.973 MPa.

Fig. 7 Performance of sorption beds: (a) air inlet and outlet temperatures of the HTS and MTS bed; (b) working pressure

4.1 Performance influenced by different refrigerating temperatures

Just as mentioned before, the required refrigerating temperature of refrigerated trucks generally ranges from -18°C to 0°C , so different refrigerating temperatures are investigated. The hot air temperature and the resorption time are controlled at approximately 210°C and 30 min, respectively.

The instantaneous refrigerating capacity for different refrigerating temperatures of 0°C , -5°C , -10°C , -15°C , -20°C and -25°C is shown in Fig. 8. Taking the refrigerating temperature of -10°C as an example (Fig. 8c), when the HTS bed begins to sorb the ammonia refrigerant at 35 min, the chilled air outlet temperature of the evaporator drops quickly from -2.1°C to -10°C in one min, and then, the electric heater installed in the cabin begins to heat the chilled air. Such a short time is due to that the expansion valve and the plate-fin type heat exchanger are utilized in the sorption freezing system. After

that, the chilled air inlet temperature increases gradually. Based on eq. 7, the instantaneous refrigerating capacity is calculated, and the result is shown in Fig. 8c. The refrigerating capacity is output only at the sorption process of the HTS bed.

Fig. 8 Instantaneous refrigerating capacity under different refrigerating temperatures of: (a) 0°C; (b) -5°C; (c) -10°C; (d) -15°C; (e) -20°C; (f) -25°C

Based on eq. 8, the average refrigerating capacity in a cycle for different refrigerating temperatures is calculated and is shown in Fig. 9a. The average refrigerating capacity decreases

gradually with the decreasing refrigerating temperature, which is due to the critical mass transfer performance under the lower evaporating pressure of the ammonia refrigerant, i.e., the lower refrigerating temperature. The average refrigerating capacity for the refrigerating temperatures of 0°C, -5°C, -10°C, -15°C, -20°C and -25°C are 2.46 kW, 2.32 kW, 2.06 kW, 1.94 kW, 1.69 kW and 1.63 kW. Compared the required refrigerating capacity shown in Table 1, the average refrigerating capacity obtained experimentally is enough to meet the requirement. Additionally, the COP and the SCP for different refrigerating temperatures are shown in Fig. 9b. Both the maximum COP and the maximum SCP are obtained at the refrigerating temperature of 0°C, and they are 0.288 and 186.4 W kg⁻¹, respectively.

Fig. 9 Performance under different refrigerating temperatures: (a) average refrigerating capacity; (b) COP and SCP

4.2 Performance influenced by different resorption time

Considering that no refrigerating capacity is output during the resorption process between the HTS bed and the MTS bed, a reasonable resorption time is quite essential for improving the storage effect of the goods. Consequently, different resorption time of 20 min, 30 min and 40 min is investigated, which corresponds to different cycle time of 45 min, 65 min and 80 min shown in Fig. 10a. Experimental results show that the refrigerating capacity obtained at the resorption time of 20 min and 40 min can meet the requirement. For the resorption time of 20 min, the refrigerating capacity at the refrigerating temperatures of -5°C, -10°C and -15°C is 2.15 kW, 1.9 kW and 1.87 kW. Additionally, the refrigerating

capacity, the COP and the SCP are compared. From Fig. 10b, 10c and 10d, it can be found that the maximum refrigerating capacity, the maximum COP and the maximum SCP are obtained at the resorption time of 30 min, so the optimum resorption time at the hot air temperature of 210°C is 30 min. Although no refrigerating capacity is output during the resorption process, the internal temperature of the cabin during the complete cycle will change little.

According to the regulation from the Carrier Transicold distributor, the refrigerating temperature of -10°C is for the frozen meat products. Taking the example for transporting the frozen meat products, the operational condition of -15°C refrigerating temperature and 30 min resorption time is chosen. Under such a condition, the average refrigerating capacity in a cycle is 1.94 kW, which is higher than the required refrigerating capacity of 1.45 kW shown in Table 1. For the refrigerated truck analysed in this paper, the load capacity is approximately 1500 kg. For the frozen meat products the specific thermal capacity is about 3.2 kJ kg⁻¹ K⁻¹. As shown in Fig. 8d, the average refrigerating capacity in the sorption process of the HTS bed is 3.84 kW, so the frozen meat products store the extra refrigerating capacity in this process, and the time for this process is about 31 min. The temperature change of the frozen meat products is $DT = (3.84 \text{ kW} - 1.46 \text{ kW}) \times (31 \times 60 \text{ s}) / (1500 \text{ kg} \times 3.2 \text{ kJ kg}^{-1} \text{ K}^{-1}) = 0.9 \text{ K}$, so the temperature of the frozen meat products decreases gradually from -10°C to -10.9°C. During the resorption process no refrigerating capacity is output, and the temperature of the frozen meat products increases from -10.9°C to -10°C, which is acceptable for storage process.

(a)

(b)

Fig. 10 Performance under different resorption time: (a) cycle time; (b) average refrigerating capacity; (c) COP; (d) SCP

4.3 Performance influenced by different hot air temperatures

Hot air temperature of 230°C is employed to drive the two-stage solid sorption freezing system, and the resorption time is controlled at 30 min. Corresponding experimental results are compared with those obtained at the hot air temperature of 210°C, which are shown in Fig. 11. From Fig. 11a, 11b, and 11c, it can be found that the refrigerating capacity, the COP and the SCP are almost same, but the effective refrigerating time shown in Fig. 11d increases from 26 min to 30 min. The effective refrigerating time refers in particular to the time that the chilled air outlet temperature is at the setting temperature, such as -10°C or -15°C. The increasing effective refrigerating time is related to the increasing driving temperature potential, which is the difference between the heat source temperature and the equilibrium desorption temperature. When the difference is larger, the heat transfer performance is better, and much more ammonia is desorbed from the HTS bed for the same resorption time.

Fig. 11 Performance under different hot air temperature: (a) average refrigerating capacity; (b) COP; (c) SCP; (d) effective refrigerating time

To show clearly the refrigerating capacity (RC) obtained under different operational conditions, the RC is summarized in Table 2.

Table 2 Refrigerating capacity (RC) obtained under different operational conditions

Refrigerating temperature/°C	Required RC/kW	RC/kW			
		Hot air temperature (°C) / Resorption time (min)			
		210/20	210/30	210/40	230/30
0	0.97		2.46		
-5	1.13	2.15	2.32	2	2.28
-10	1.29	1.9	2.06	1.87	2.11
-15	1.45	1.87	1.94	1.68	1.98
-20	1.62		1.69		
-25	1.78		1.63		

4.4 Performance comparison of two solid sorption freezing systems

The performance of the experimental system after optimization is compared with that of the earlier-established experimental system (Gao et al., 2016a, 2016b). The detailed comparison of two systems is shown as follows:

(1) The earlier-established system can reach the required refrigerating capacity only at the refrigerating temperature of 0°C and -5°C. However for the optimized system the refrigerating capacity obtained experimentally can meet the requirement even at the refrigerating temperature of -18°C, which

is needed for transporting frozen goods. The experimental results show that even at the refrigerating temperature of -25°C , and the corresponding refrigerating capacity is still up to 1.63 kW, as shown in Fig. 9a. For the refrigerating temperature of -10°C the refrigerating capacity of 2.06 kW obtained in the optimized system is almost one time higher than that of 1.09 kW obtained in the earlier-established system.

(2) The cycle time of the earlier-established system is up to 240 min, and the resorption time is approximately 120 min. However for the system after optimization, the cycle time is approximately 65 min, and the resorption time is 30 min. A reasonable resorption time is quite essential for improving the storage effect of the goods, because no refrigerating capacity is output during the resorption process.

(3) For the earlier-established system, the time that the chilled air outlet temperature of the evaporator drops to -15°C is approximately 15 min. For the optimized system, the time needed is only 1.5 min, as shown in Fig. 8d. Such a short time is due to that the expansion valve and the plate-fin type heat exchanger are utilized in the system.

(4) For the sorption beds, the total number of unit tubes is reduced from 200 to 121. Additionally, both the evaporator and the condenser are off-the-shelf components of refrigeration industries. Consequently, the total mass of the system is reduced from 260 kg to 150 kg.

Finally, the comparison of the COP of the sorption freezing system established in this paper to the ammonia-water absorption refrigeration system is conducted. Generally, there are two kinds of ammonia-water absorption chillers, i.e., small capacity chillers and large capacity chillers. Experimental performance of some ammonia-water absorption chillers is shown in Table 3. Small capacity chillers usually refer to the ammonia-water diffusion absorption chillers, and pressure compensated auxiliary gas (helium or hydrogen) is used. Because there is no moving part in diffusion absorption chillers, they can operate with the absolute noiselessness, which have been commercially developed for the hotel mini-bars and the hospital rooms, but both the refrigerating capacity and the COP are relatively low. The refrigerating capacity is usually below 1 kW, and the COP is about 0.12. Consequently, the diffusion absorption chillers are difficult to meet the requirement of refrigerated trucks. For large capacity chillers

such as a single-stage ammonia-water absorption chiller the refrigerating capacity is above 5 kW, and the COP is above 0.3. And the technology of the spray falling film evaporation is utilized in the system. When trucks vibrate continually at driving conditions, the vibration certainly will deteriorate the performance, which is due to that the liquid that is sprayed cannot exchange heat with the tube very well. Meanwhile, the solution pump is an essential part of the system, which makes the system more complex and also consumes some electricity. Additionally, in consideration of the flow and cost of solution pump, an appropriate solution pump for refrigerated trucks is difficult to be found in the market. As a result, the large capacity ammonia-water chillers are also not suitable for refrigerated trucks at present.

Table 3 Experimental performance of some ammonia-water absorption chillers

Author	Mixture	Heat source/ Temperature(°C)	Evaporation temperature/°C	COP	Refrigerating capacity/kW
Bazzo et al.(2013)	Ammonia-water	Steam/160.9	-9.5	0.35	7
Said et al.(2016)	Ammonia-water	Hot water/129	-7	0.3	4.8
ACF60 LB (2016)	Ammonia-water	Natural gas/>500	-10	0.48	13.3
Jakob et al.(2008)	Ammonia-water				
	-helium	Electricity/129	6	0.12	0.7
Mazouz et al.(2014)	Ammonia-water				
	-hydrogen	Electricity/185	-14	0.12	0.005
Rêgo et al.(2014)	Ammonia-water	Engine exhaust			
	-hydrogen	gas/200	-7	-	-

Note: “-” indicates that the data isn’t shown in the paper.

5 Conclusion

Based on the earlier-established $\text{MnCl}_2/\text{CaCl}_2\text{-NH}_3$ two-stage solid sorption freezing system for a refrigerated truck with the rated power of 80 kW, a series of optimization designs are proposed in this paper, and the optimized system is built. For the sorption beds consisting of many unit tubes, the

arrangement mode is changed from the previous aligned arrangement to the staggered arrangement for enhancing heat exchange. For the evaporator and the condenser, both of them are off-the-shelf components of refrigeration industries. Simultaneously, an expansion valve is utilized in the system. To determine the performance of the optimized system, a series of experiments are carried out. Conclusions are as follows:

(1) Different refrigerating temperatures ranging from -25°C to 0°C are investigated at the hot air temperature of 210°C and the resorption time of 30 min, and the refrigerating capacity obtained is enough to meet the requirement of this refrigerated truck. Even at the refrigerating temperature of -25°C , the refrigerating capacity is approximately 1.63 kW. The maximum COP and the maximum SCP are 0.288 and 186.4 W kg^{-1} , respectively.

(2) Different resorption time of 20 min, 30 min and 40 min is investigated, and experimental results show that the refrigerating capacity obtained at the above resorption time can meet the requirement. For the resorption time of 20 min, the refrigerating capacity at the refrigerating temperatures of -5°C , -10°C and -15°C is 2.15 kW, 1.9 kW and 1.87 kW, respectively.

(3) Experimental results show that the higher hot air temperature can prolong the effective refrigerating time. It is because much more ammonia is desorbed from the HTS bed at the same resorption time.

(4) The earlier-established system can reach the required refrigerating capacity only at the refrigerating temperature of 0°C and -5°C . For the optimized system, the refrigerating capacity can meet the requirement even at the refrigerating temperature of -18°C , which is for transporting frozen goods.

Through a series of optimization designs, not only the refrigerating capacity but also the total mass of the system can satisfy requirement of this refrigerated truck. Considering that the optimized system is

prototype level at present, in the future the system will be installed in the truck to conduct the test.

Although the initial investment of the sorption system is larger than that of the mechanical vapor compression refrigeration system, the use-cost of the sorption refrigeration system is less than that of the mechanical vapor compression refrigeration system. The high initial investment is mainly due to that the two sorption beds are manufactured by us and their cost is approximately six times more than the cost of the compressor. If the sorption beds can be manufactured in large batch, their cost can be reduced greatly. It is predicted that compared with the use-cost of the mechanical vapor compression refrigeration system, the initial investment for the sorption refrigeration system will pay back in less than a year.

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