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Energetic and Life Cycle Economic Assessment of Air Source CO_2 Heat Pump for Space Heating Integrated with DMS Utilizing Zeotropic Mixture

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ABSTRACT

Using transcritical CO_2 air source heat pump for space heating is a clean and environmentally friendly solution. Based on the concept of the Lorenz cycle and CO_2 heat pump system with traditional dedicated mechanical subcooling (DMS), CO_2 system with DMS employing zeotropic mixture as working fluid for subcooling is proposed. Its life cycle economic performance is assessed. The results indicate that the energy performance of zeotropic working fluid with high-temperature glide is higher than that of the pure working fluid. The capital cost and fuel cost are obviously reduced by using high-temperature glide zeotropic mixtures for subcooling. The levelized total annual cost is saved by 5.70, and 19.12% compared with that using pure refrigerant, and baseline CO_2 system. The mixture with high-temperature glide is recommended.

Keywords: transcritical CO_2 heat pump, dedicated mechanical subcooling, zeotropic mixture, space heating, life cycle economic performance, energetic performance

NONMENCLATURE

Abbreviations	
ASHP	Air source heat pump
BASE	Baseline CO_2 system
CO_2	Carbon dioxide
COP	Coefficient of performance
DMS	Dedicated mechanical subcooling
GWP	Global warming potential
HP	Heat pump
OMC	Operation and maintenance cost
Symbols	
η	Efficiency

1. INTRODUCTION

China has become the largest energy consumption and CO_2 emission country with the promotion of urbanization [1], However, Space heating in China heavily relies on the coal-burning restricted by its energy structure. To control air pollution, it is necessary to employ advanced methods for space heating.

Air source heat pump (ASHP) is regarded as an environmental protection and energy saving solution to replace the coal-fired boiler for space heating, However, the widely used ASHP units are mostly charged with hydrofluorocarbons (HFCs), such as R410A and R134a, which show very high global warming potential (GWP) and lead to a strong greenhouse effect [2]. They are all listed as the substances to be phased down due to the Kigali Amendment to the Montreal Protocol. The Paris Agreement [3] also regulates to restrict the emission of greenhouse gases to limit the increase in average global temperature. Therefore, to mitigate the greenhouse effect, a new alternative working fluid for ASHP is needed.

Among the alternative working fluids, CO_2 is considered as a promising refrigerant because of its excellent thermal and transport performance and low impact on the environment. CO_2 subcooling has become one of the promising methods to improve the performance of the CO_2 system [4]. The discharge pressure of the CO_2 system can be reduced and the heating/cooling capacity can also be increased effectively by introducing this solution.

For the applications of refrigeration, Dai et al. [5] carried out an energy improvement study of a transcritical CO_2 refrigeration system with DMS. For the applications of space heating, the method of DMS is also utilized. Yang et al. [6] found that the COP of the combined system is 22% higher than that of the baseline CO_2 system.

It can be found from the above summary that the dedicated mechanical subcooling CO_2 system is a feasible solution for room heating. The performance of the refrigeration system can be improved by considering the characteristic of temperature glide for the zeotropic working fluid. an energetic and economic model is developed in this study considering the influence of heat transfer deterioration by using the zeotropic mixture. Then the influence of component selection and the ratio of the zeotropic mixture on the energetic performance and the whole life cycle economic performance of the DMS CO_2 heat pump system is systematically discussed. It can provide a theoretical reference to improve the energy efficiency and the practical operation of the air source CO_2 heat pump system for space heating.

2. SYSTEM DESCRIPTION

2.1 Dedicated mechanical subcooling CO_2 heat pump system

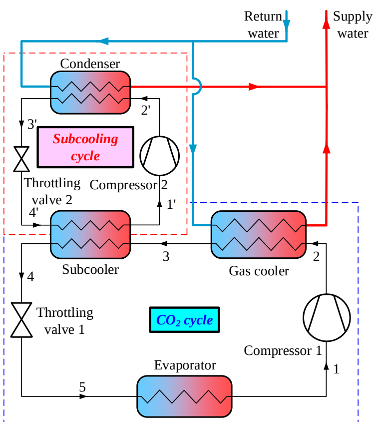

Fig. 1 Schematic of the transcritical CO_2 heat pump system with dedicated mechanical subcooling

Fig. 1 shows the diagram of the air source transcritical CO_2 heat pump system integrated with dedicated mechanical subcooling, which consists of two subsystems: a CO_2 subsystem (1-2-3-4-5-1) and a mechanical subcooling subsystem (1'-2'-3'-4'-1'). For the CO_2 subsystem, the CO_2 working fluid flows through the compressor, gas cooler, subcooler, throttling valve, and evaporator, respectively. The subcooler is also the evaporator of the subcooling subsystem. The return water is split into two branches, one flows into the condenser and the other goes into the gas cooler. The water is heated and then converges to provide heating for the users.

The T-s diagram of the transcritical DMS CO_2 heat pump system with the zeotropic mixture is shown in Fig. 2. The standard operating condition is set as the ambient temperature of -12°C [7], and the supply and return water temperature of the heat pump system is 40°C, and 65°C, respectively according to the national standard [8]. When the working fluid of the subcooling subsystem is zeotropic mixture, the temperature changes as the working fluid evaporates or condenses. Therefore, the temperature matching of the mixture with the water and CO_2 is alleviated. According to the theory of the Lorentz cycle, the energy efficiency of the system can be improved.

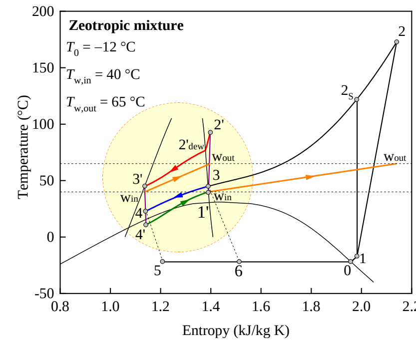

Fig. 2 T-s diagram of transcritical CO_2 heat pump system with dedicated mechanical subcooling for zeotropic mixture

2.2 Working fluid selection for subcooling subsystem

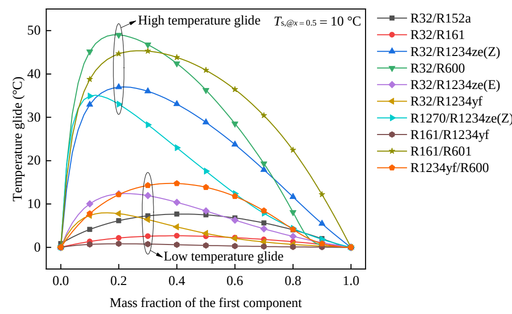

Fig. 3 Temperature glide of zeotropic mixture variation with the mass fraction

The energy efficiency of the CO_2 heat pump system can be improved by using zeotropic mixture as working fluid. Nevertheless, the selection of components and determining the composition ratio of the mixture are important for the system's performance. Ten binary mixtures are selected as the working fluids for the subcooling subsystem and the corresponding temperature glides are shown in Fig. 3. For the ten temperature curves, R1270/R1234ze(Z), R161/R601, R32/R1234ze(Z), and R32/R600 show relatively high-temperature glide, and the maximum value are all above 30°C. Thus, they are named as high-temperature glide mixture in this study. In contrast, the other six zeotropic mixtures show much lower temperature glide, which is classified as low-temperature glide mixture.

3. METHODOLOGY

3.1 Energetic model

The thermodynamic model is established based on the following assumptions:

(1) The transcritical CO_2 heat pump subsystem and the subcooling subsystem both operate under steady working conditions;

(2) Heat loss of the equipment and pressure drop of fluid flowing through the pipeline are ignored;

(3) The pinch point temperature difference in the gas cooler, the condenser, and the subcooler is set as 5°C;

(4) The ambient temperature is 10°C higher than the evaporation temperature.

The energy and exergy performance of the system is analyzed based on each state point of the working fluids and the heat transfer fluids, which are determined by the method of iterative calculation due to the restriction of pinch temperature difference. The properties of the fluids are determined by REFPROP 10.0 [9].

<u>3.1.1 Transcritical CO_2 heat pump system</u>

For the traditional transcritical CO_2 heat pump system, which is short for the BASE in this study, the energy performance can be determined by:

Compressor:

$$W_{CO_2} = m_{CO_2} \times (h_2 - h_1) \tag{1}$$

$$\eta_{g,CO_2} = (h_{2s} - h_1)/(h_2 - h_1) \tag{2}$$

Gas cooler:

$$Q_{h,CO_2} = m_{CO_2} \times (h_2 - h_3) \tag{3}$$

Throttling valve:

$$h_5 = h_4 \tag{4}$$

Evaporator:

$$Q_{c,CO_2} = m_{CO_2} \times (h_1 - h_6) \tag{5}$$

$$COP_{BASE} = Q_{h,CO_2} / W_{CO_2} \tag{6}$$

<u>3.1.2 Transcritical CO_2 heat pump system with DMS</u>

The equations for the subcooling subsystem are expressed as follows:

Subcooler:

$$T_4 = T_3 - \Delta T_{SC} \tag{7}$$

$$m_{Sub} = m_{CO_2} \times (h_3 - h_4)/(h_{1'} - h_{4'}) \tag{8}$$

where ΔT_{SC} is the subcooling degree of CO_2 fluid flowing through the subcooler; m_{Sub} is the refrigerant mass flow rate of the subcooling subsystem.

Compressor:

$$W_{Sub} = m_{Sub} \times (h_{2'} - h_{1'}) \tag{9}$$

$$\eta_{g,Sub} = (h_{2s'} - h_{1'})/(h_{2'} - h_{1'}) \tag{10}$$

Condenser:

$$Q_{h,Sub} = m_{Sub} \times (h_{2'} - h_{3'}) \tag{11}$$

$$m_{w,Sub} = m_{Sub} \times (h_{2'} - h_{3'})/(h_{w,out} - h_{w,in}) \tag{12}$$

Throttling valve:

$$h_{4'} = h_{3'} \tag{13}$$

$$COP_{Sub} = Q_{h,Sub} / W_{Comp,Sub} \tag{14}$$

For the whole system:

$$Q_{h,Tot} = Q_{h,CO_2} + Q_{h,Sub} \tag{15}$$

$$W_{Tot} = W_{CO_2} + W_{Sub} \tag{16}$$

$$COP_{DMS} = Q_{h,Tot} / W_{Tot} \tag{17}$$

3.2 Economic model

A residential building with a heating area of 100 m^2 located in Beijing is selected to evaluate the life cycle economic performance of DMS CO_2 heat pump system with the zeotropic mixture for subcooling, and the economic performance indicator of annual total revenue requirement (TRR) is employed.

<u>3.2.1 Total capital investment</u>

The finite volume method [10] is adopted to determine the heat transfer area.

For traditional transcritical CO_2 heat pump systems:

$$TCI = C_{Evap} + C_{Comp,CO2} + C_{GC} + C_{Add} \tag{18}$$

For mechanical subcooling transcritical CO_2 heat pump system:

$$\begin{aligned} TCI = {} & C_{Evap} + C_{Comp,CO2} + C_{GC} + C_{Subcooler} + C_{Comp,Sub} \\ & + C_{Cond} + C_{TV,CO2} + C_{TV,Sub} + C_{Fan} + C_{Add} \end{aligned} \tag{19}$$

<u>3.2.2 Fuel cost</u>

The bin-method [11] is adopted to calculate the energy consumption for CO_2 heat pump system.

<u>3.2.3 Total revenue requirement</u>

The annual total revenue requirement refers to the compensation for expenses incurred in operating the system through revenue in a certain year to ensure the good operation of the economy [12].

4. ENERGETIC AND ECONOMICAL ASSESSMENT

4.1 Energy analysis

The COP of the DMS CO_2 heat pump system by using R32/R152a (50/50) as refrigerant for the subcooling subsystem is shown in Fig. 4. The working condition is set as the ambient temperature of -12°C, and the supply and return water temperature of 65/40°C. As can be seen, with the increase of discharge pressure and subcooling degree, the COP increases sharply first and then decreases gradually. When the discharge pressure is 10.007 MPa and the subcooling degree is 27.68°C, the maximum COP is obtained with the value of 2.1307, and the corresponding pressure and temperature are the optimum discharge pressure and the optimum subcooling degree.

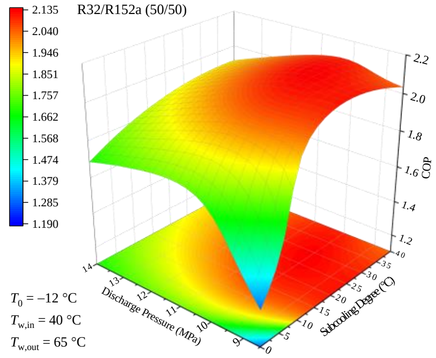

Fig. 4 Variation of system COP with discharge pressure and subcooling degree.

Ten binary mixtures shown in Fig. 3 are used to analyze the energy efficiency performance, and the corresponding COP is shown in Fig. 5. The results indicate that the COP for all the mixtures generally increases first and then decreases, and the highest value is achieved. Additionally, the COP of working fluid with high-temperature glide is significantly higher than small temperature glide for all these mixtures discussed. In contrast, for the mixtures with low-temperature glide, the improvement in COP is not notable, and R32/R152a shows the highest COP of 2.1319, whereas it is 11.22% lower than that of R32/R1234ze(Z) (30/70).

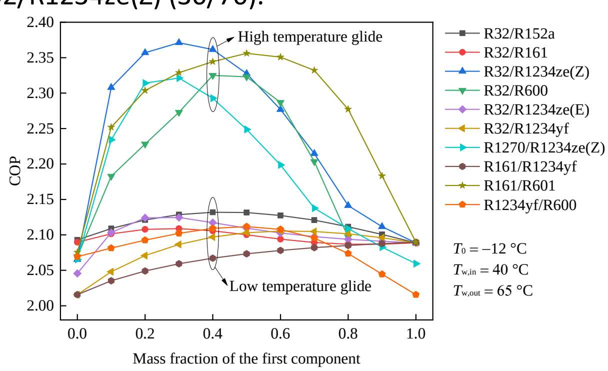

Fig. 5 COP of binary zeotropic mixture variation with the mass fraction of the first component

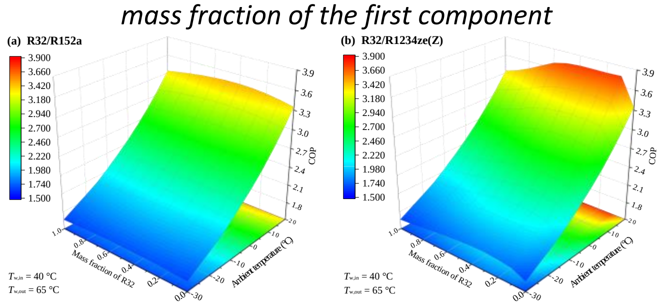

Fig. 6 Variation of system COP with ambient temperature and component. (a) R32/R152a. (b) R32/R1234ze(Z).

The COP of the system with X_{R32} of 0~1 and ambient temperature of -30~20°C is shown in Fig. 6. It indicates that the mixture with large temperature glide, for instance, R32/R1234ze(Z) at 0.2 ⩽ X_{R32} ⩽ 0.6, the enhancement of COP remains at a high level through the ambient temperature varies in a wide range of -30~20°C.

4.2 Exergy analysis

In order to further explain the influence of mass fraction of mixture on the performance of the system from the perspective of exergy, The exergy efficiency of BASE CO_2 system and DMS CO_2 HP system by using R32/R152a and R32/R1234ze(Z) for subcooling is shown in Fig. 7. As can be noted that the exergy efficiency adopting R32/R1234ze(Z) is much higher than that of R32/R152a. It can be explained as follows: The temperature glide of R32/R1234ze(Z) is relatively high, and a good temperature matching with the cooling process of supercritical CO_2 fluid and the heating process of water is obtained. It can be concluded that a higher exergy efficiency of DMS CO_2 system can be achieved when high-temperature glide mixture is utilized for subcooling.

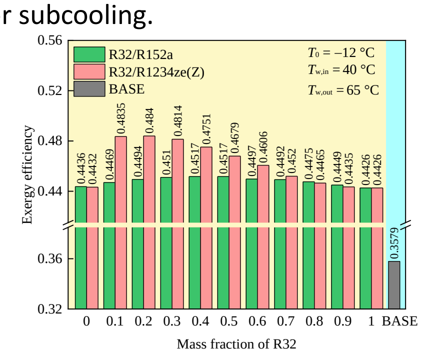

Fig. 7 Exergy efficiency corresponding to R32/R1234ze(Z) and R32/R152a at different mass fractions

4.3 Economic analysis

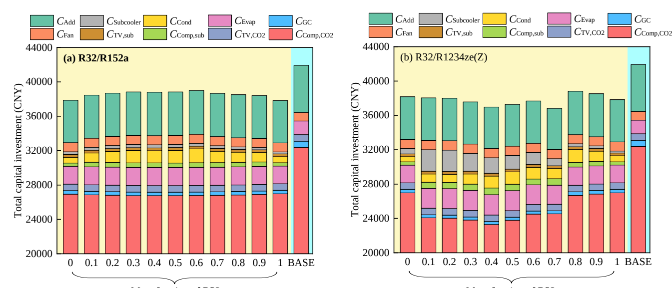

Fig. 8 Total capital investment for different components of the CO_2 system. (a) R32/R152a. (b) R32/R1234ze(Z).

A CO_2 heat pump system with different zeotropic mixtures for subcooling is designed, and then the total capital investment is also assessed taking a residential building with an area of 100 m^2 in Beijing as an example. Fig. 8 shows the cost of the CO_2 heat pump system for each component when the subcooling system is charged with R32/R152a and R32/R1234ze(Z), respectively. It can be clearly observed that the cost of the DMS CO_2 heat pump system is lower than that of the BASE system, which is reduced by 7.46~10.78% and 7.98~13.83% for R32/R152a and R32/R1234ze(Z), respectively. This is mainly due to the low COP of the BASE system, which leads to the designed compressor capacity is much larger.

Due to the analysis, many advantages can be obtained by employing high-temperature glide working fluid as the refrigerant of the subcooling subsystem, it is recommended to use high-temperature glide mixture as the refrigerant for the subcooling subsystem of the DMS CO_2 system.

5. CONCLUSIONS

Transcritical CO_2 air source heat pump system with dedicated mechanical subcooling (DMS) charged with zeotropic working fluid is proposed. The energetic and the life cycle economic performance model of the system is developed, considering the heat transfer deterioration effect due to mass transfer resistance of zeotropic mixture. Finally, the following conclusions are obtained:

(1) A maximum coefficient of performance (COP) is achieved at the optimum discharge pressure and subcooling degree for the DMS CO_2 heat pump using zeotropic working fluid for subcooling subsystem.

(2) The zeotropic mixture with high-temperature glide for subcooling subsystem can improve the energy efficiency of CO_2 heat pump system more significantly than that with low-temperature glide.

(3) DMS CO_2 heat pump system shows a better life cycle economic performance when large temperature glide zeotropic mixture is utilized compared with that using pure working fluid.

(4) Zeotropic working fluids with large temperature glide are recommended to be used as refrigerants for subcooling subsystem, such as R32/R1234ze(Z) (30/70), R161/R601 (50/50), R1270/R1234ze(Z) (30/70), and R32/R600 (40/60).

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