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THEORETICAL ANALYSIS ON A CO_2 HEAT PUMP DRIVEN BY WASTE HEAT

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ABSTRACT

Heat pumping is a useful technology in supplying high temperature heat for industrial purpose or district heating by using low temperature heat simultaneously. The absorption heat pump is a rather complex system while the vapor-compression heat pump requires power for compressing the working fluid. In addition, high amounts of waste heat, even at high temperature is available in industrial processes. This article analyzed a novel heat-driven high temperature heat pump with CO_2 as working fluid and revealed its performance by a theoretical approach. It is driven by cooling a heat source at 300°C, which may be industrial excess heat. The results showed that the heating capacity supplied by the cycle increases with the increase of the cooled pressure while the temperature shows an opposite trend. The maximum coefficient of performance is 1.54 in the considered conditions.

Keywords: high temperature heat pump, CO_2, low temperature waste heat, district heating

NONMENCLATURE

Abbreviations	
COP	coefficient of performance
Symbols	
t	temperature (°C)
h	enthalpy (kJ/kg)
c	specific heat (J/(kg·°C))
r	ratio
$\dot{Q}$	heat capacity (kW)
$\dot{m}$	mass flow rate (kg/s)
E	exergy (kW)
Subscript	
p	isobaric
1, 1a, 1b, 2, 3, 4, 5, 6	state points of the cycle
Superscript	
$'$	inlet
$''$	outlet

1. INTRODUCTION

Heat-driven heat pumping is a useful technology for raising the availability of low-temperature heat or even the surroundings, and also increase its capacity by transferring heat from a high-temperature excess heat source.

The most widely used heat pump is so called vapor compression heat pump which is driven by power. Fatouh et al. [1] carried out an experimental investigation on a vapor compression heat pump with the aim to evaluate the performance of a system for space cooling and hot water supplying simultaneously. It was found that the actual COP under the cooling mode is between 1.9 and 3.1 and that under the heating mode is from 2.9 to 3.3. Ommen et al. [2] studied the technical and economic working domains for industrial heat pumps and gave advices about the refrigerant in different conditions. Tong et al. [3] used the genetic algorithm to optimize a vapor compression heat pump system. The absorption heat pump is another choice for pumping heat from low temperature heat source to a higher level. In addition, it is driven by heat rather than power. Li et

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al. [4] proposed a solar cogeneration system coupled with an absorption heat pump and analyzed its performance. Roeder et al. [5] carried out a simulation study on the ammonia-water mixture desorption for the absorption heat pump. Yan et al. [6] carried out an experimental investigation on a novel triple heat sources-driven absorption heat pump system.

On many occasions, high temperature heating is needed, but it is limited by the maximum operating temperature of the compressor in the vapor-compression heat pump. The mixture refrigerant can be used to achieve this target as well as reducing the discharging temperature of the compressor [7]. This article studied on a novel high temperature heat pump with CO_2 as working fluid and revealed its performance by a theoretical analysis. The main innovation is that a liquid pressurizing process is used to replace a near-critical compression which is necessary in common ejector heat pump or ejector refrigeration.

2. METHODOLOGY

2.1 The cycle and modelling

As shown in Fig. 1, the main components of a heat-driven CO_2 high temperature heat pump include a heater, an ejector, a cooler, a throttle valve, an evaporator, a vapor-liquid separator and a pump.

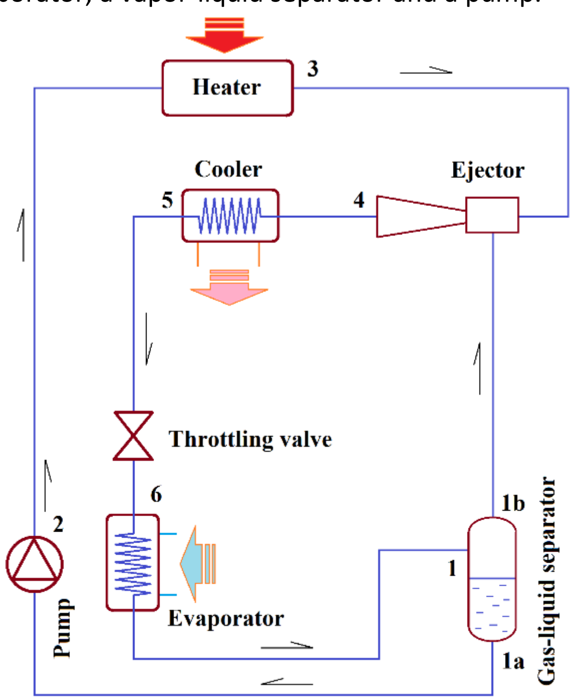

Fig 1 Flow chart of the heat-driven CO_2 high temperature heat pump

The thermodynamic process of the whole cycle is indicated in a temperature entropy diagram, Fig. 2. In the supercritical heater, CO_2 absorbs heat from the driving heat source with the process from state 2 to state 3. The ejector is a core part of the system. It is used to raise the pressure of the saturated vapor from the separator using the primary high-pressure hot CO_2 from the heater. Then, the mixed CO_2 comes to the cooler and is cooled to state point 5. Meanwhile the heating function can be realized. The low-temperature supercritical CO_2, usually in near-critical condition, expands in the throttle valve, becoming a two-phase flow. In the evaporator, some liquid CO_2 evaporates to realize the cooling function. Finally, the separator separates vapor from the two-phase flow. The vapor is sucked away by the ejector while the liquid is pressurized by the pump.

The whole cycle consists of two sub-cycles, namely, the power sub-cycle (1-1a-2-3-4-5-6-1) and the refrigeration sub-cycle (1-1b-4-5-6-1). The power sub-cycle is a driving cycle where the driving heat converts into mechanical energy to compress the saturated CO_2 from the separator. The refrigeration sub-cycle generates a low-temperature condition to condense CO_2 and absorb heat from an excess heat source or the environment. In the expansion process, some shaft power can be recovered if the throttle valve is replaced by an expander.

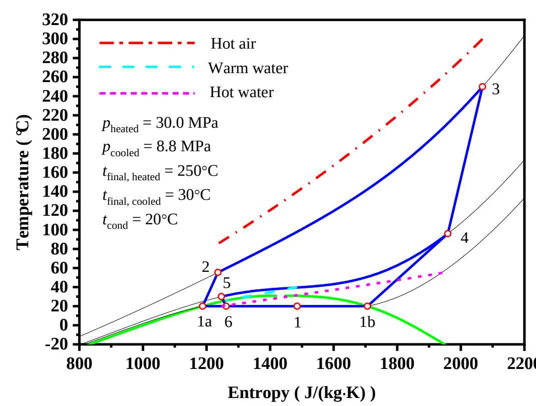

Fig 2 Thermodynamic process of the heat-driven CO_2 high temperature heat pump

However, the expander may operate in the two-phase region, so it is worth noting that though the liquid density is almost the same as that of the vapor in near-critical conditions, the liquid drops are still harmful to an impeller. In view of this, a non-impeller expander may be a better choice for safer operation, such as a piston expander, a screw expander and so on.

The theoretical model was implemented in Matlab. The computational flow is indicated in Fig. 3.

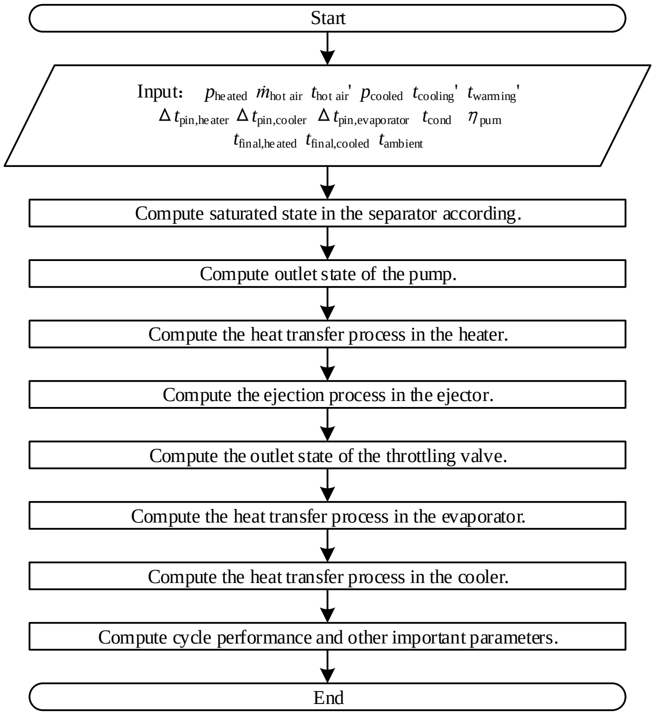

Fig 3 Flow chart of the calculation process

2.2 Calculation process

The thermophysical properties of CO_2 in all states are obtained from REFPROP 9.0 [8] with inputting some state parameters.

This cycle is driven by a heat source, hot air and the heat capacity can be expressed with equation (1). The equation (1a) is used to calculate the heat capacity according to the parameters of the CO_2. Equation (1b) shows the relationship between the heat capacity and the parameters of the hot air. Both of these equations are used in the theoretical analysis.

$$\dot{Q}_{\text{driving}} = \dot{m}_{\text{CO2,normal}} \cdot (h_3 - h_2) \tag{1a}$$

$$\dot{Q}_{\text{driving}} = \dot{m}_{\text{hotair}} \cdot c_{\text{p,air}} \cdot (t'_{\text{hotair}} - t''_{\text{hotair}}) \tag{1b}$$

The heating capacity supplied and the cooling capacity in the evaporator can be found with equation (2-3). The entrainment of the ejection can be calculated with equation (4). Both coefficient of performance and exergy efficiency are used in the analysis and the definition can be found in the equation (5-6).

$$\dot{Q}_{\text{heating}} = \dot{m}_{\text{CO2, total}} \cdot (h_4 - h_4) \tag{2a}$$

$$\dot{Q}_{\text{heating}} = \dot{m}_{\text{hot water}} \cdot c_{\text{p, water}} \cdot (t''_{\text{hot water}} - t'_{\text{hot water}}) \tag{2b}$$

$$\dot{Q}_{\text{cooling}} = \dot{m}_{\text{CO2, total}} \cdot (h_1 - h_6) \tag{3a}$$

$$\dot{Q}_{\text{cooling}} = \dot{m}_{\text{warm water}} \cdot c_{\text{p, water}} \cdot (t'_{\text{hot water}} - t''_{\text{hot water}}) \tag{3b}$$

$$r_{\text{entrainment}} = \frac{\dot{m}_{\text{refrigeration}}}{\dot{m}_{\text{power}}} \tag{4}$$

$$\text{COP} = \frac{\dot{Q}_{\text{heating}}}{\dot{Q}_{\text{driving}} + \dot{P}_{\text{pump}}} \tag{5}$$

$$\eta_{\text{exergy}} = \frac{\Delta E_{\text{hot water}}}{|\Delta E_{\text{hotair}}| + |\Delta E_{\text{warm water}}| + \dot{P}_{\text{pump}}} \tag{6}$$

As shown in equation (6), the exergy efficiency is defined as the ratio of the obtained exergy in the cooler and the consumption of the exergy in the supercritical heater, the evaporator and the pump. In the considered conditions, the exergy of both hot air and warm water decreases, so its absolute value is used in the equation.

2.3 Data source

In the theoretical analysis, the following parameters should be specified. Hot air is selected as the driving heat source and its mass flow rate is specified as 1 kg/s. Its temperature is up to 300°C. The supercritical heated pressure is 30.0 MPa under which the CO_2 is heated in the heater. The final heated temperature is 250°C. The isentropic efficiency of the pump is specified as 0.65. The pinch point temperature difference in the cooler is specified as 6°C. The same pinch point temperature difference is used in the evaporator as that in the cooler. Because of the poor heat transfer of hot air, the pinch point temperature difference in the supercritical heater is specified as 30°C. The initial temperature of the hot water supplied is 20°C. The temperature of the waste water which supplies low-temperature heat to the cycle is 40°C. The condensing temperature is 20°C.

The final cooled temperature is 40°C, while the cooled pressure ranges from 7.6 MPa to 12.0 MPa. It is worth noting that the cooled pressure refers to the pressure under which the CO_2 is cooled by the cooling water and the final cooled temperature is the temperature at the exit of the cooler.

The ambient temperature and pressure are specified as 25°C and 0.1 MPa, respectively, which is used as dead state in the exergy analysis.

3. RESULTS AND DISCUSSION

In this section, some parameters and the cycle performance are discussed, such as the mass flow rates, ejector entrainment, heat capacities, coefficient of performance and exergy efficiency.

As shown in Fig. 4, the mass flow rate of CO_2 in the power sub-cycle keeps constant as 0.63 kg/s and that in the refrigeration sub-cycle decreases with raising the cooled pressure. In the analysis, the mass flow rate and the initial temperature of the hot air are kept constant. In addition, the heater operates under a constant pinch point temperature difference. Therefore, the CO_2 in the power sub-cycle flows through the heater with a constant mass flow rate and absorbs a constant heat rate from the driving source. The ejector entrainment is determined by the primary state, the secondary state and the back pressure which is also known as cooled pressure here. With constant values for both the primary state and the secondary state, the entrainment decreases with increasing the cooled pressure. That is also to say that the mass flow rate of CO_2 ejected in the refrigeration sub-cycle decreases with the decrease of the entrainment. Therefore, the total mass flow rate of CO_2 shows a decreasing trend.

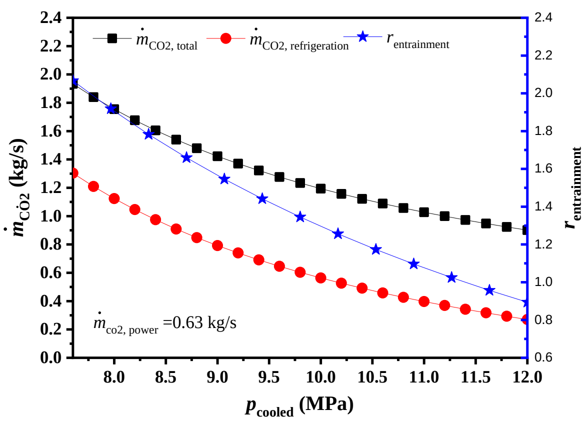

Fig 4 Mass flow rate of CO_2 and ejection entrainment with the cooled pressure

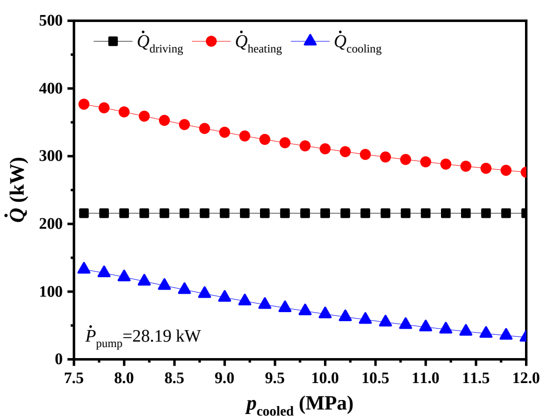

Fig 5 Variation of the heat capacities with the cooled pressure

As shown in Fig. 5, heat capacity in the cooler and the evaporator decrease with raising the cooled pressure. A constant driving heat rate, 215.7 kW, is also indicated as explained above. A constant power of 28.19 kW is consumed by the pump to drive the CO_2 of the power sub-cycle. Both the mass flow rate of CO_2 in the evaporator and the evaporating process impact on the cooling heat capacity. With the increase of the cooled pressure, the vapor quality at the evaporator exit decreases more than that at the entrance, which decreases the heating capacity in the evaporator. In addition, the total mass flow rate of CO_2 in the evaporator also shows a decrease in the series conditions. Therefore, the cooling heat capacity decreases with the increase of the cooled pressure. The heating capacity supplied by the whole cycle comes from the driving heat, the cooling heat capacity and the power consumption by the pump, so it shows the same trend with the cooling heat capacity.

As shown in Fig. 6, the temperature of the hot water increases with increasing the cooled pressure. That is caused by the variation of the isobaric line in the cooling process. It changes into a straighter line, which raises the outlet temperature of the cooling water under the same pinch point temperature difference. It is also indicated in the figure that the mass flow rate of the supplied hot water decreases with increasing the cooled pressure, which is caused by the decrease of the heat capacity in the cooler and the increase of the temperature of the hot water. High temperature and mass flow rate of the hot water should be the purpose of this cycle, but there is a remarkable negative relation between these two parameters. Therefore, the condition should be specified according to the actual need. The maximum temperature of the hot water reaches up to 106°C in the cycle.

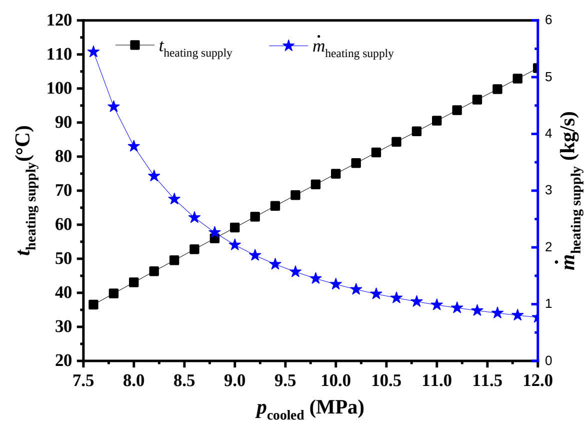

Fig 6 Variation of the temperature and mass flow rate of the supplied hot water with the cooled pressure

As shown in Fig. 7, the coefficient of performance decreases with the increase of the cooled pressure and the exergy efficiency shows an uptrend in the considered conditions. The consumptions of the whole cycle, the driving heat capacity and the power consumption of the pump, are kept constant in the considered conditions. The heating capacity supplied decreases with raising the cooled pressure. Therefore, higher coefficient of performance can be obtained under lower cooled pressure. However, on the other hand, the exergy efficiency shows a completely opposite trend. The reason is that though the heating capacity supplied decreases while the specific exergy increases with raising the cooled pressure. The maximum coefficient of performance is 1.54 and the maximum exergy efficiency is 28.7% in the considered conditions.

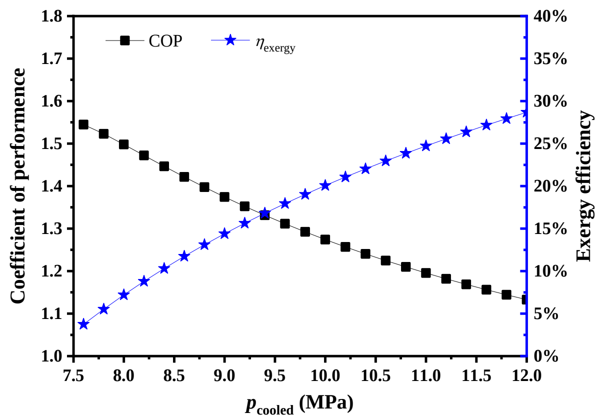

Fig 7 Variation of the coefficient of performance and the exergy efficiency with the cooled pressure

4. CONCLUSIONS

This article studied a novel high temperature heat pump with CO_2 as working fluid and revealed its performance by theoretical analysis. In the considered conditions, the entrainment decreases with increasing the cooled pressure. The heat capacity in the cooler and the evaporator decrease with raising the cooled pressure while the driving heat rate and the power consumed by the pump are constant as 215.7 kW and 28.19 kW, respectively. The maximum coefficient of performance obtained in the analysis is 1.54 and the maximum exergy efficiency is 28.7%. In addition, the maximum temperature of the hot water reaches up to 106°C in the cycle.

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