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Experimental verification of the self-condensing CO_2 transcritical power cycle

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A B S T R A C T

CO_2 is an excellent natural working fluid for both power cycles and refrigeration cycles. However, it limits the actual application of the CO_2 transcritical power cycle that subcritical CO_2 is hard to be condensed by conventional cooling water. Aiming to search solutions for this condensing problem, this work carried out an experimental verification of a novel cycle named self-condensing CO_2 transcritical power cycle and got some operation laws of the system. The results showed that the self-condensing CO_2 transcritical power cycle can operate well with conventional cooling water around 30 °C. In most cases, the operation was steady and could be adjusted easily. The saturated liquid CO_2 even as cold as 5 °C was generated for the pump. It is beneficial for the power sub-cycle to keep inlet pressure and outlet temperature of the throttle valve as high as possible. In the throttling process, the maximum pressure drop and temperature drop were about 3.6 MPa and 25 °C, respectively. A transcritical or near-critical throttling occurred in the experimental investigation and the pump usually experienced a transcritical compression under high pump inlet temperature. In summary, the self-condensing CO_2 transcritical power cycle had been verified and is fully feasible in a practical application.

1. Introduction

The steam Rankine cycle is the most widely used technology for power generation from thermal energy. It can be found in coal power plants, waste heat power generation systems, solar thermal power generation systems and so on. However, it faces two development bottlenecks. Firstly, a large superheat is required to avoid liquid hammer in a turbine of the steam Rankine cycle system, which decreases the average heat absorbing temperature in the boiler or the evaporator for the same heat source. According to Carnot theorem, a high heat absorbing temperature and a low heat releasing temperature contribute to a high thermal efficiency while a decrease of the heat absorbing temperature and an increase of the heat releasing temperature harm to the cycle performance. Therefore, it shows a very low thermal efficiency for low temperature heat source [1]. On the other hand, the operating pressure and temperature of the steam Rankine cycle have to be increased higher and higher in order to achieve higher efficiency. The ultra-supercritical units can reach efficiencies close to 45% while they require more rigorous specifications for equipment materials which need to withstand very high pressure and temperature. The materials for the ultra-supercritical unit are very expensive. The optimization of working fluid is a potential way to solve these two bottlenecks.

ORC (Organic Rankine cycle) is considered to be a very potential cycle for generating power from low temperature heat source [2]. Its working fluid is organic, like HFCs and hydrocarbons. Liu et al. analyzed the performance of a geothermal ORC using hydrocarbons as working fluids [3,4]. Tian et al. [5] and Wang et al. [6] carried out studies on the ORC performance for recovering engine waste heat. Because most of the considered organic fluids decompose at higher temperature, they can only be used for low temperature heat source. Dai et al. [7] contributed to the thermal stability of several hydrocarbons for screening working fluid for the ORC.

CO_2 is another widely concerned working fluid. It is a natural working fluid with environmentally friendly properties. As shown in Table 1, the ODP is 0 and the GWP is 1 [8]. In addition, it is non-toxic, non-flammable and cheap. Because of its thermal stability, CO_2 power cycles can be applied for not only low temperature heat source but also high temperature heat source. Dostal et al. showed a thermal efficiency of 46.07% for a supercritical CO_2 Brayton cycle

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Nomenclature

t	temperature (°C)
p	pressure (MPa)
h	enthalpy (kJ/kg)
$\dot{m}$	mass flow rate of the working fluid (kg/s)
$\dot{P}$	power (kW)
M	molecular weight (g/mol)
abbreviation	
LFL	Lower Flammable Limit
ASHRAE	American Society of Heating, Refrigerating and Air-conditioning Engineers
ODP	Ozone Depletion Potential
GWP	Global Warming Potential
subscript	
b	boiling
c	critical
cond	condense
isen	isentropic
exp	expander
com	compressor
1, 1a, 1b, 2, 3, 4, 5, 6, 7, 8, 9	state points of working fluid
superscript	
‘	inlet
″	outlet

with turbine inlet temperature of only 550 °C [9]. Iverson et al. [10] studied on advantages of the supercritical CO_2 Brayton cycle and claimed that it has significant efficiency especially as solar thermal power plants. Aiming at industrial waste heat, Pan et al. [11] established a CO_2 transcritical power cycle and obtained 1.1 kW of stable power output. Huang et al. [12] carried out an experiment on a small-scale axial turbine expander that is used in CO_2 transcritical power cycle and obtained the maximum power generation of 692 W. Ge et al. [13] built a test rig of a small-scale low-grade power generation system with a transcritical CO_2 power cycle on the top floor of an 80 kW micro-turbine CHP unit. The maximum power generation was about 500 W. Shu et al. [14] showed some configurations selection maps of CO_2-based transcritical Rankine cycle for thermal energy management of engine waste heat. Li et al. [15] integrated the supercritical CO_2 power system with an absorption power generation systems to improve the basic cycle. Du et al. [16] focused on the off-design performance of a CO_2 transcritical power cycle using a radial turbine. Several layout of supercritical CO_2 Brayton power cycles caught much attention. Syblik et al. [17] analyzed the recompression supercritical CO_2 Brayton power cycles in nuclear and fusion energy. Wang et al. [18,19] compared performance of several supercritical CO_2 Brayton cycles with solar thermal energy. Some processes, such as recompression, reheating pre-compression, split expansion, partial-cooling and inter cooling, are used in these cycles. Olumayegun et al. [20] proposed a supercritical CO_2 Brayton cycle system with solvent-based CO_2 capture for coal-fire power generation.

The power cycle using the mixture of CO_2 and organic is also an interesting topic. Pan et al. [21] studied on the properties of a zeotropic mixture (R290/CO_2) and its performance for a transcritical power cycle. Xia et al. [22] compared thermo-economic performance of several CO_2-based mixtures as working fluids. However, most organics are flammable, so the operating safety of the CO_2-based mixtures also attracts researchers' attention. Tian et al. [23] carried out an experimental and theoretical study on the flammability limits of several hydrocarbon/CO_2 mixtures, namely, propance/CO_2, n-butane/CO_2, isobutene/CO_2, n-pentane/CO_2 and isopentane/CO_2. Pan et al. [24] gave the detailed combustion characteristic of n-butane/CO_2 and isobutene/CO_2 at different mixing ratios.

However, because the critical temperature of CO_2 is only 31 °C, the subcritical CO_2 is hardly condensed in the CO_2 transcritical power cycle by conventional cooling water at about 30 °C. This problem limits the actual application of the CO_2 transcritical power cycle. Pan et al. [25] proposed a novel CO_2 power cycle named self-condensing CO_2 transcritical power cycle in which a throttling process is added to couple a power sub-cycle and a refrigeration sub-cycle that produce liquid CO_2 for the pump. In this novel cycle, the transcritical cycle is realized using conventional cooling water, whereas it hadn't been verified by experiments.

In order to verify the self-condensing CO_2 transcritical power cycle, an experimental system was established and several series of verification experiments were carried out. During the experiments, a power sub-cycle and a refrigeration sub-cycle were realized successfully, together with the whole self-condensing CO_2 transcritical power cycle. Considering that the transcritical expansion or throttling is the key process for this novel cycle, states of CO_2 in the throttle valve and the pump are monitored and analyzed. In addition, several temperature levels of liquid CO_2 were achieved by using conventional cooling water.

2. Methodology

2.1. The self-condensing CO_2 transcritical power cycle

As shown in Fig. 1, the cycle consists of two sub-cycles, a power sub-cycle (1-1a-2-3-4-5-6-8-9-1) and a refrigeration sub-cycle (1-1b-7-8-9-1). The former is the main cycle supplying power and the latter is an auxiliary cycle supplying liquid CO_2 for the former. In that cycle, the CO_2 is cooled by conventional cooling water at 30 °C and a pump is used to pressurize the liquid working fluid.

The saturated liquid (1→1a) and vapor (1→1b) of CO_2 are generated by using a vapor-liquid separator, so the total mass flow rate is equal to the mass flow rate of CO_2 in the power sub-cycle plus that in refrigeration sub-cycle, as shown in equation (1). The liquid CO_2 is pressurized by a pump from state point 1a to 2 and the power consumption in this process can be expressed as equation (2). Furthermore, the isentropic efficiency of the pump is calculated by equation (3). Then the CO_2 under supercritical pressure is heated in the recuperator (2 → 3) and the supercritical heater (3 → 4) one by one. The process from state point 4 to 5 refers to the supercritical expansion in the expander or turbine and power output is shown in equation (4). After the first cooling step (5 → 6) in the recuperator, the supercritical CO_2 mixes with the vapor from the compressor

Table 1
Thermal and environmental properties of CO_2.

M g/mol	t_b °C	t_c °C	p_c MPa	LFL %	ASHRAE 34 safety group	Atmospheric life yr	ODP /	GWP 100 yr
44.01	−78.4	31.0	7.38	none	A1	>50	0.00	1

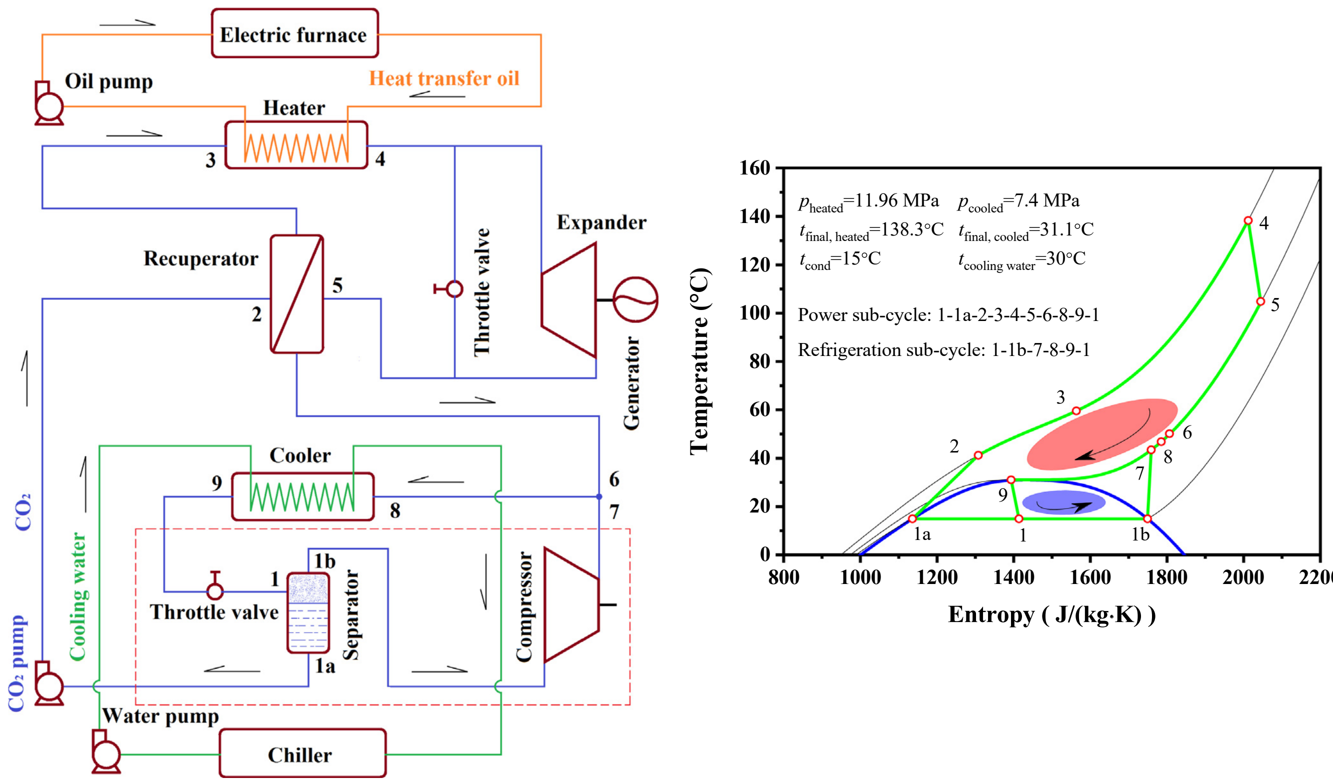

Fig. 1. (a)Flow chart of the self-condensing CO_2 transcritical power cycle.(b)The self-condensing CO_2 transcritical power cycle (Based on data in an experimental condition).

(6 + 7 → 8). Then it is cooled again by the cooling water (8 → 9) under supercritical pressure. An expansion process is used to get a two phase flow at low temperature (9 → 1) and liquid CO_2 can be generated for the power sub-cycle. In the refrigeration sub-cycle, the saturated vapor separates from the two phase flow in the vapor-liquid separator (1→1b) and is compressed by a compressor (1b→7). The power consumption of the compressor can be found in equation (5). Then it mixes into the CO_2 flow in the power sub-cycle (7 + 6 → 8) and the cooling water cools the mixed flow (8 → 9). The two phase flow is generated by using a throttle valve or an expander (9 → 1).

$$\dot{m}_{total} = \dot{m}_{power} + \dot{m}_{refrigeration} \quad (1)$$

$$\dot{P}_{pump} = \dot{m}_{power} \cdot \left(h''_{pump} - h'_{pump}\right) \quad (2)$$

$$\eta_{pump,isen} = \frac{h''_{pump,isen} - h'_{pump}}{h''_{pump} - h'_{pump}} \quad (3)$$

$$\dot{P}_{exp} = \dot{m}_{power} \cdot \left(h'_{exp} - h''_{exp}\right) \quad (4)$$

$$\dot{P}_{com} = \dot{m}_{refrigeration} \cdot \left(h''_{com} - h'_{com}\right) \quad (5)$$

The transcritical expansion is the most important process which coupled the two sub-cycles. Some shaft power can be recovered from it using an expander rather than a throttle valve. However, it is worth noting that two-phase flow is harmful to the impeller, though the density difference of the vapor and the liquid is very low in near critical conditions. It is a better choice to recovery the transcritical expansion power by a non-impeller expander.

2.2. The experimental system

Fig. 2 shows the experimental system for the self-condensing CO_2 transcritical power cycle. A compressor, a throttle valve and a vapor-liquid separator were fixed in it, which is different from a conventional CO_2 transcritical power cycle system. There are three loops in the whole system, namely a heat transfer oil loop, a cooling water loop and the CO_2 power cycle loop. The heat transfer oil loop is used to simulate the industrial waste heat source. An electric furnace supplies initial energy to heat the heat transfer oil that transfers heat from the electric furnace to the supercritical heater. On the experimental system of the basic CO_2 transcritical power cycle, a chiller is used to supply cold water to condense subcritical CO_2. It is hard to do that with conventional water cooling. In the system the chiller rather than a cooling tower is used to reduce the temperature of the cooling water. Then it is easier to get cooling water with lower temperature like 20 °C. The main components of the CO_2 power cycle loop are a heater, an expander, a recuperator, a cooler, a throttle valve, a vapor-liquid separator, a CO_2 pump and a compressor.

Here, a throttle valve is used to realize the transcritical or near critical expansion process, so some expansion shaft is lost. The vapor-liquid separator separates the vapor CO_2 for the compressor and the liquid CO_2 for the pump. In this study, a compressor is used to drive the auxiliary sub-cycle. The detailed information of the components can be found in Table 2.

2.3. Operating indicator

For the self-condensing CO_2 transcritical power cycle, the most important parameters are the temperature of the cooling water and the states in the pump and the throttle valve. Then this investigation revealed the variations of the above parameters, such as the

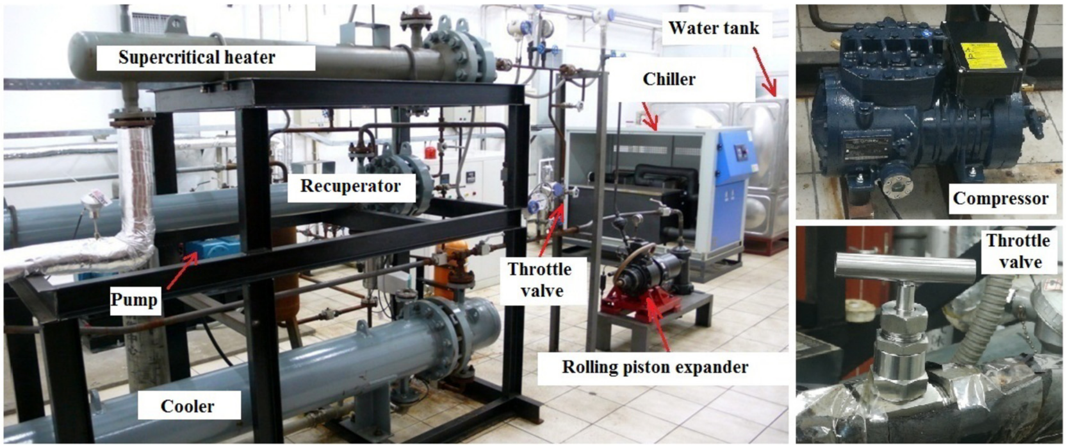

Fig. 2. The experimental system for the self-condensing CO_2 transcritical power cycle.

Table 2
Detailed information of the components.

Item	Information
Supercritial heater	A shell and tube heat exchanger is used as supercritical heater in the CO_2 transcritical power cycle system. CO_2 flows in the tube pass while thermal oil in the shell pass. The design pressure for tube pass is 20 MPa in the tube pass. The heat exchange area is 1.19 m^2.
Recuperator	A shell and tube heat exchanger is used as recuperator in the system. Supercritical pressure CO_2 flows in the tube pass while subcritical pressure CO_2 in the shell pass. The design pressure for tube pass is also 20 MPa in the tube pass. The heat exchange area is 1.59 m^2.
Condenser	A shell and tube heat exchanger is used as condensor in the system. CO_2 flows in the tube pass while cooling water in the shell pass. The design pressure for tube pass is 8.0 MPa in the tube pass. The heat exchange area is 8.09 m^2.
Pump	A plunger pump is used to transport liquid CO_2 in the system. The model number is SJ4-M-630/2.5. The rated flow is 0.4 m^3/h. The rated inlet pressure is 2.0 MPa and the rated outlet pressure is 20 MPa. Its speed can be controlled by a frequency converter.
Compressor	The model is CD180H. The displacement is 1.12 m^3/h under 50 Hz.
Expander	A rolling piston expander is used in the system. The rated power is 2.0 kW and the rated rotational speed is 1000 rpm.
Generator	It is a three-phase permanent magnet generator. The rated rotational speed is 1000 rpm. The rated power and the rated voltage are 2.0 kW and 220 V (AC), respectively. It is fixed in the expander shell in order to achieve good sealing effect.
Chiller	The chiller unit can provide cooling water with temperature in the range from 0 °C to 30 °C. The control accuracy is ±1 °C.
Mass flow meter	Coriolis mass flowmeter is used in the system. The model number is SITRANS F C MASSFLOW MASS2100 DI 15. The measuring range is 0−5600 kg/h and the accuracy is 0.1%. Working pressure is 0−12.8 MPa.
Power meter	Model number of the digital power meter used in the experiment is WT230. Measuring range of voltage is 0−600 V and measuring range of electric current is 0−20 A. Applicable frequency range is 0 and 0.5−100 kHz. The basic accuracy is 0.1%.

inlet and outlet temperature of the cooling water, the temperature and pressure at the entrance and exit of the pump and that of the throttle valve. These parameters can show the operating condition of the novel cycle.

2.4. *Uncertainty of the measurement*

On this experimental system, temperature is measured by the K-type temperature thermocouples whose accuracy is ±0.5% (±1.0 °C). Some pressure transmitters are used to measure the pressure of the CO_2 in the system. Their accuracy is ±0.5% (±0.05 MPa). The data of temperature, pressure and mass flow rate is collected and sent to the computer by a data acquisition and processing system with 12 bit precision. The accuracy of the mass flow meter and the power meter is indicated in Table 2.

3. Results and discussion

3.1. *With cooling water of 20 °C and heated pressure of 11.5 MPa*

In this section, the temperature of the cooling water was specified as 20 °C and the heated pressure was 11.5 MPa. Actually, the temperature of cooling water can be decreased to as low as local wet-bulb temperature by a cooling tower and it isn't constant in different places or seasons. In the most unfavorable condition, e. g. in summer, it is commonly around or higher than 30 °C. Here, the cooling water was already much cooler than the conventional one supplied by a cooling tower, whereas was still a little warmer than that for a basic CO_2 transcritical power cycle system [11].

It is worth noting that the heated pressure refers to the pressure under which the CO_2 is heated in the supercritical heater by the thermal oil.

As shown in Fig. 3, very cool liquid CO_2 was obtained for the pump with the cooling water at 20 °C. The temperature of the cooling water was kept around 20 °C. However, cooling capacity of the chiller in this experimental system is very large and the cooling water tank is a little small, so the temperature is slightly unstable. In the most considered conditions, the CO_2 was cooled to 23 °C at which it began to expand in the throttle valve. During the expansion process near the critical point, the temperature of the CO_2 was reduced and much cold liquid CO_2 was generated. The temperature of the liquid CO_2 before entering the pump was successively adjusted to 10 °C, 5 °C, 15 °C, 20 °C and 23 °C one by one. It kept generally stable with only a slight fluctuation because of the instability of the cooling water. It took very little time and was very easy to transfer the condition from one to another. Therefore, the cycle operated well with the cooling water at 20 °C. However, it should be noted that the CO_2 was cooled under subcritical pressure, as shown in Fig. 3b. Subcritical CO_2 at about 23 °C and 6.3 MPa got into the throttle valve and began to expand. It should be pointed

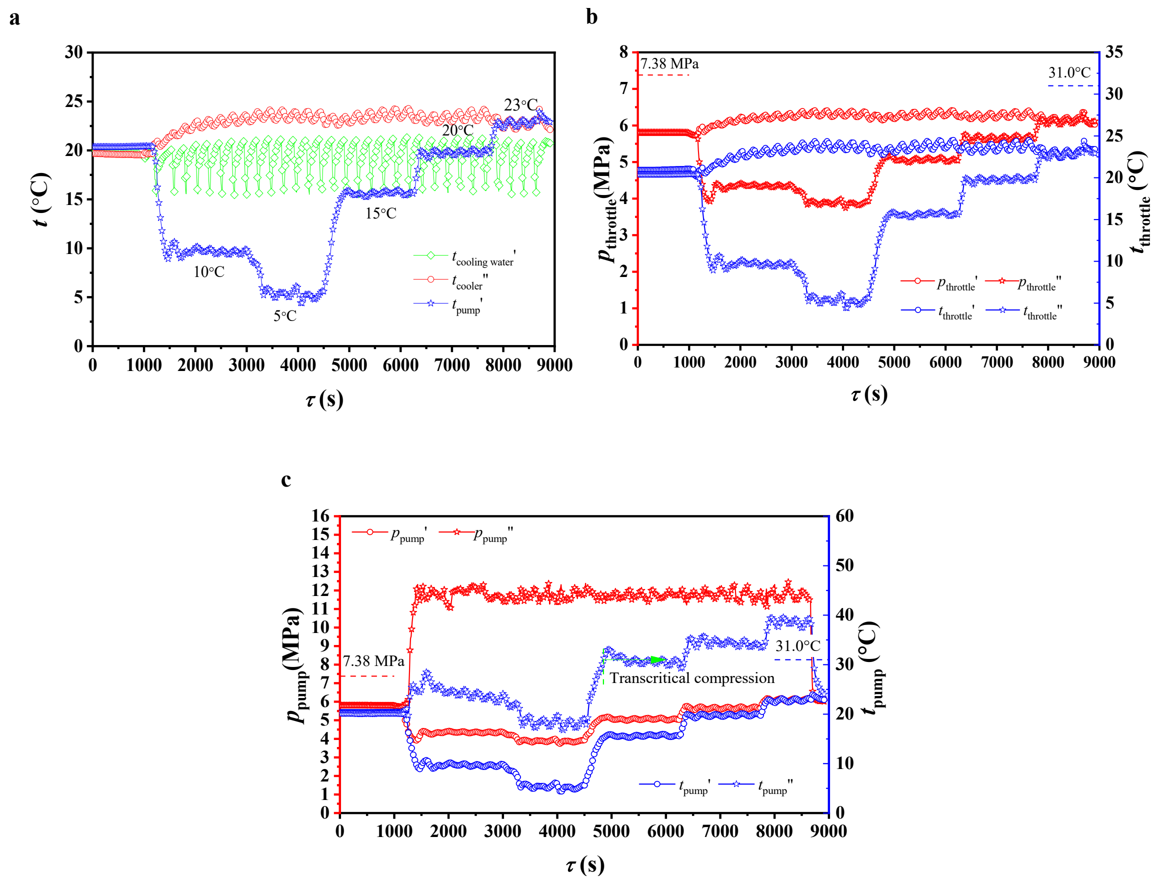

Fig. 3. Variation of the considered parameters. (a. the cooling effect; b. states in the throttle valve; c. states in the pump).

out that the pressure showed in the figures is the absolute pressure of CO_2. According to data from REFPROP [26], the saturated temperature corresponding to 6.3 MPa can be found to be 24 °C, so it was a little supercooled or around saturated at the entrance of the throttle valve. The cooling water was as low as 20 °C, so it was able to condense the subcritical CO_2. But the subcritical expansion here produced colder liquid CO_2 and still contributed to the stability of the pump. In order to realize a transcritical expansion in the throttle valve which exists in the self-condensing CO_2 transcritical power cycle, the temperature of the cooling water and the operating pressure are raised in the following experimental series.

The inlet and outlet parameters in the pump are also shown in Fig. 3c. The outlet pressure was around 11.5 MPa and the inlet pressure varied with the variation of the inlet temperature. The reason is that the CO_2 at the entrance of the pump was at saturated state. For example, the pressure of the liquid CO_2 that got into the pump at about 5 °C was about 4 MPa. Though the outlet pressure almost kept constant, the outlet temperature increased with raising the inlet temperature. The reason is that the outlet state was in the supercooled region or the supercritical region. The temperature and the pressure no longer corresponded one-to-one. It can also be found that the pump experienced a transcritical compression after 5000 s. In other words, the outlet pressure was higher than 7.38 MPa and the outlet temperature was no longer lower than 31 °C. Because the supercritical fluid has much lower viscosity and density than liquid, both the sealing difficulty and power consumption of the pump increases under supercritical condition. For example, with constant pressure of 8 MPa, CO_2's viscosity and density is 7.57×10^{-5} Pa s and 827.7 kg/m^3 at 20 °C while 2.23×10^{-5} Pa s and 277.9 kg/m^3 at 40 °C.

3.2. *With cooling water of 20 °C and heated pressure of 13.5 MPa*

In this section, the heated pressure was around 13.5 MPa and the cooling water at 20 °C was still used. As shown in Fig. 4a, four experimental conditions were carried out and the pump inlet temperature was 20 °C, 15 °C, 10 °C and 5 °C one by one. The CO_2 was already cooled to about 20 °C at the exit of the cooler, so its temperature is around or lower than 20 °C at the exit of the throttle valve. It is also indicated that there was a sharp decrease and increase for the pump inlet temperature during the transition from the 20 °C condition to the 15 °C condition. That was caused by the hysteresis of the human operation and can be avoid by an automatic control system.

As shown in Fig. 4b, the pressure and temperature at the entrance of the throttle valve were still very lower than the critical values. The trends of the outlet pressure and temperature were similar, which indicated that the CO_2 went into the two-phase region after the throttling process. The maximum temperature drop in the throttle valve was about 20 °C and the maximum pressure drop was around 2.5 MPa. Indeed, the auxiliary cycle or so-called the refrigeration sub-cycle generates cooling in the throttling process. The pressure potential energy was lost and the cooling was generated.

As shown in Fig. 4c, the pump outlet pressure was as high as 13.5 MPa and was generally stable in the whole experiment series. However, during the 20 °C condition, the pump outlet temperature gave a steady decline instead of a stable trend. In addition, it went upwards to as high as 50 °C in the pump. The temperature rise in this condition was 30 °C which was much higher than that in the other conditions from 10 °C to 17 °C. The temperature rise indeed

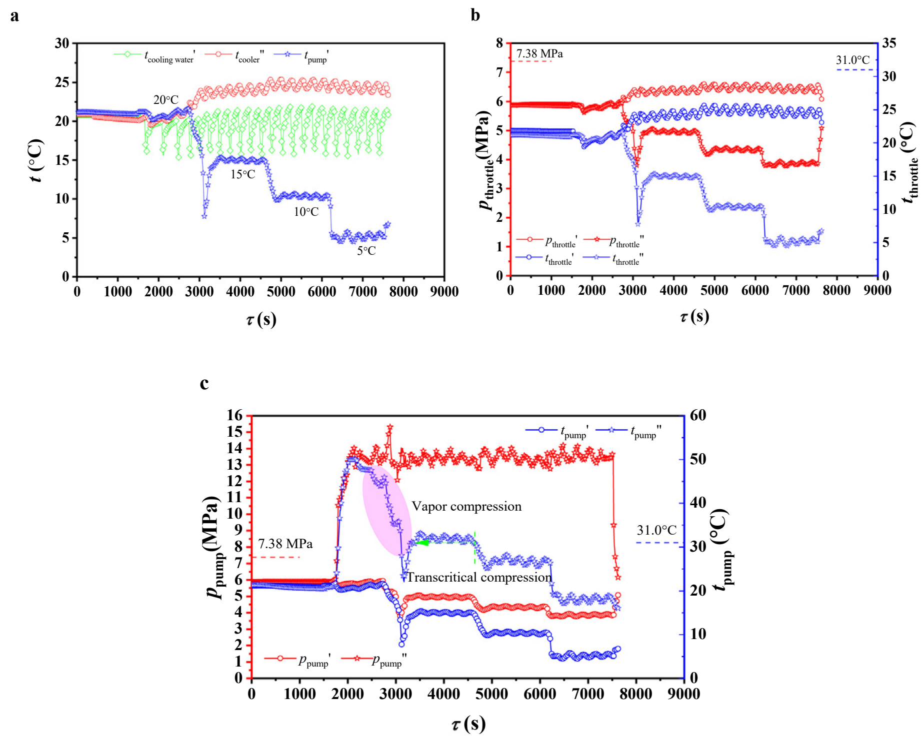

Fig. 4. Variation of the considered parameters. (a. the cooling effect; b. states in the throttle valve; c. states in the pump).

increased with the pressurization process going towards the critical point, but it was found that the CO_2 state at the pump entrance was already in the superheat region. For example, the inlet pressure was 4.98 MPa and the inlet temperature was 17.34 °C at 3000 s, but the saturated temperature under 4.98 MPa is 14.12 °C. There was a superheat degree as high as 3.22 °C at the pump entrance. Though there is a vapor-liquid separator, the CO_2 goes into the superheated region after the throttling process if the vapor quality is high enough at the entrance. Therefore, the vapor compression process in the pump caused the instability of the outlet temperature and the difficulty for controlling the inlet temperature on the pump. In order to check if the vapor compression occurs in other conditions, the condition at 4000 s is selected for the verification. The inlet pressure at that point was 5.01 MPa corresponding a saturated temperature of 14.37 °C. The actual measured inlet temperature was 15.16 °C. The accuracy of the thermocouple used in the system is ±1.0 °C. Consequently, the CO_2 entering the pump was at saturated state at 4000 s. The pressure rise in the pump was up to 10 MPa.

3.3. With cooling water of 30 °C and heated pressure of 11.5 MPa

In this section, the temperature of the cooling water was specified as high as 30 °C, similar to the actual cooling water. The heated pressure was kept as 11.5 MPa. As shown in Fig. 5a, five experimental conditions were finished with different and very stable pump inlet temperature. Using cooling water at 30 °C, liquid CO_2 at temperature as low as 5 °C were generated for the pump which driven the power sub-cycle. It also should be pointed out that colder liquid CO_2 even can be produced, but CO_2 below 0 °C is harmful to the water cooling part of the pump and is unnecessary. It was able to reach up to 25 °C which was higher than that in the above experiments, the reason for which was that the pressure at the entrance of the throttle valve was much higher than that in the above series. As shown in Fig. 5b, the valve inlet pressure was as high as 7.4 MPa and the inlet temperature was able to exceed 31 °C. Therefore, a transcritical or near-critical throttling occurred in the refrigeration sub-cycle. With constant temperature, the higher the pressure is, the lower the specific enthalpy is. The specific enthalpy at the exit is equal to that at the entrance. Therefore, high inlet pressure is helpful for decreasing the outlet specific enthalpy and producing more liquid in the throttling process. The throttle valve inlet temperature should be decreased as far as possible to strengthen the power sub-cycle. In other words, a transcritical throttling or expansion should be used to make the whole cycle adapt with conventional or warmer cooling water. In the considered conditions, the maximum pressure drop and temperature drop were about 3.6 MPa and 25 °C, respectively.

As shown in Fig. 5c, the pump outlet pressure was about 11.5 MPa and many pressurization processes were transcritical compressing. The most parameters were very stable except the pump outlet pressure that showed a slight fluctuation at 5 °C of the pump inlet temperature. As the saturated temperature decreased, the mass flow fraction in the power sub-cycle decreased fast and that in the refrigeration sub-cycle increased correspondingly. It was more difficult to stabilize the pump outlet pressure with much lower mass flow rate.

3.4. With cooling water of 30 °C and heated pressure of 13.0 MPa

One more experiment series was carried out with the cooling water at 30 °C and the heated pressure of 13.0 MPa. The pump inlet temperature varied from 5 °C to 25 °C, as shown in Fig. 6a. The condition at 5 °C showed an instability as a result of the very low mass flow rate of the CO_2 in the power sub-cycle. Therefore, the pump inlet temperature should not be adjusted to too low temperature. With the aim to enhance the power output, the mass flow rate in the power sub-cycle should be raised as far as possible and that in the refrigeration sub-cycle should be reduced. As shown in Fig. 6b, in this experiment series, the inlet state of the throttle value

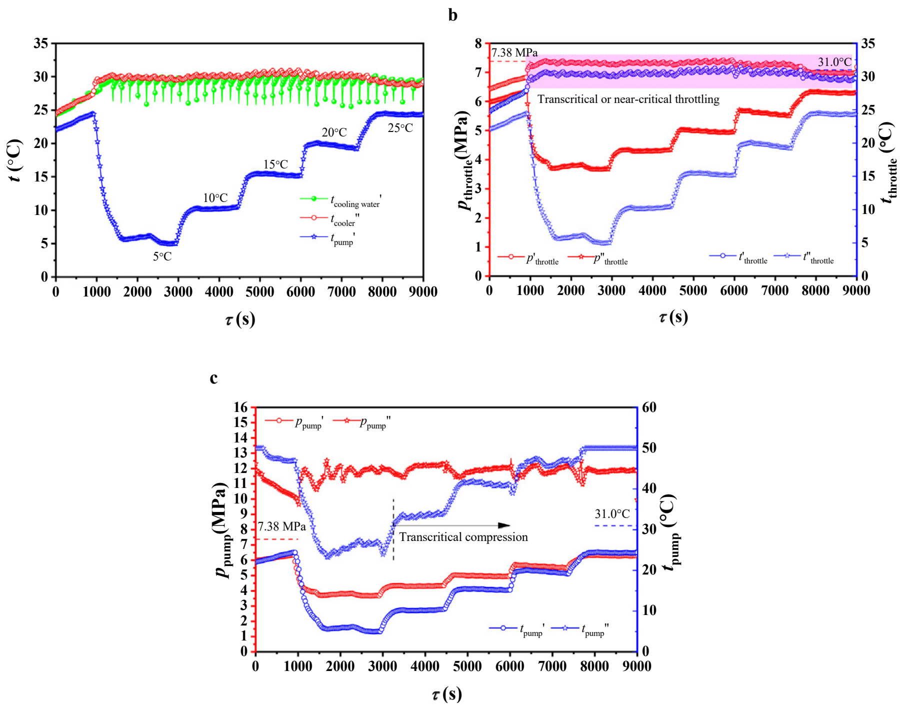

Fig. 5. Variation of the considered parameters. (a. the cooling effect; b. states in the throttle valve; c. states in the pump).

was much closed to the critical point but its values still didn't exceed the critical values. Consequently, the expansion didn't cross the critical point and only occurred in the near-critical region. In the steady conditions, the maximum pressure drop and temperature drop are about 3 MPa and 20 °C, respectively.

As shown in Fig. 6c, a transcritical compression existed in all stable pressurization processes. Fluctuation was the main feature of the state parameters both at the entrance and exit of the pump. It should be explained that there is an error of the constant pump outlet temperature of 50 °C after 4890 s. The actual temperature was slightly beyond the measuring scope of the thermal couple, so the real values at these conditions were lost. It isn't able to reach the limited temperature 50 °C for the CO_2 pump in normal conditions, but the pump inlet temperature was raised very high in this experiment series.

3.5. Discussion on the cycle

The main purpose of using the self-condensing CO_2 transcritical power cycle is to realize the operation of the CO_2 transcritical power cycle with conventional water cooling. Both throttling process and the compression process are monitored and analyzed detailedly as well as showing the condensing effect of the CO_2 in this cycle. The transcritical or near-critical throttling process combined the power sub-cycle and the refrigeration sub-cycle. It is worth noting that the expansion component is replaced by a throttle valve. Therefore, the supercritical CO_2 experienced a throttling and further natural cooling process, indicated from state 4 to 5. This replacement does not influence the verification effect of the whole power cycle. The cycle in a typical experimental condition is already shown in Fig. 1b. In that condition, the heated pressure is 11.96 MPa and the final heated temperature is 138.3 °C. Using 30 °C cooling water, the supercritical CO_2 is cooled to 31.1 °C and then reaches the two phase state at 15 °C after the throttling process. It should be explained that the cooler has a very large area and then has ability to cool CO_2 to 31.1 °C which is almost equal to the inlet temperature of the cooling water. The pump and the compressor have higher actual capacity than the need in the experiment, so their efficiency is lower than that in the rated condition. This phenomenon is common in experimental study. Therefore, the power consumption of the pump and the compressor was not monitored. However, the pressure and temperature of the working fluid at the entrance and exit were measured for each component. Therefore, the state points can obtained according to the corresponding pressure and temperature data. The enthalpy rise and the isentropic efficiency of the pump are 60.8 kJ/kg and 13.5% respectively. Because of the low efficiency of the pump, the entropy increase is very high as well as the temperature rise in the pressurization process. There is an enthalpy rise of 15.5 kJ/kg in the compressor.

In the experiment, the temperature of the cooling water was specified as 20 °C and 30 °C. The cooling water can be cooled to this temperature grade easily by a cooling tower. However, the cooling water at higher temperature like 30 °C or even 40 °C can also be used in this cycle theoretically, but the thermal efficiency decreases correspondingly. The reason is that it is harmful to thermal efficiency to raising the temperature of the heat sink. It is worth noting that higher cooled pressure should be used to generate enough liquid CO_2 for the power sub-cycle under warmer cooling water.

4. Conclusions

CO_2 is an excellent natural working fluid for both power cycle and refrigeration cycle, but the condensing problem of the subcritical CO_2 limits the application of the CO_2 transcritical power cycle. Based on the newly proposed cycle, named self-condensing CO_2 transcritical power cycle, a corresponding experimental system was established for the verification. Four experiment series were carried out with changing the temperature of the cooling water and the heated pressure.

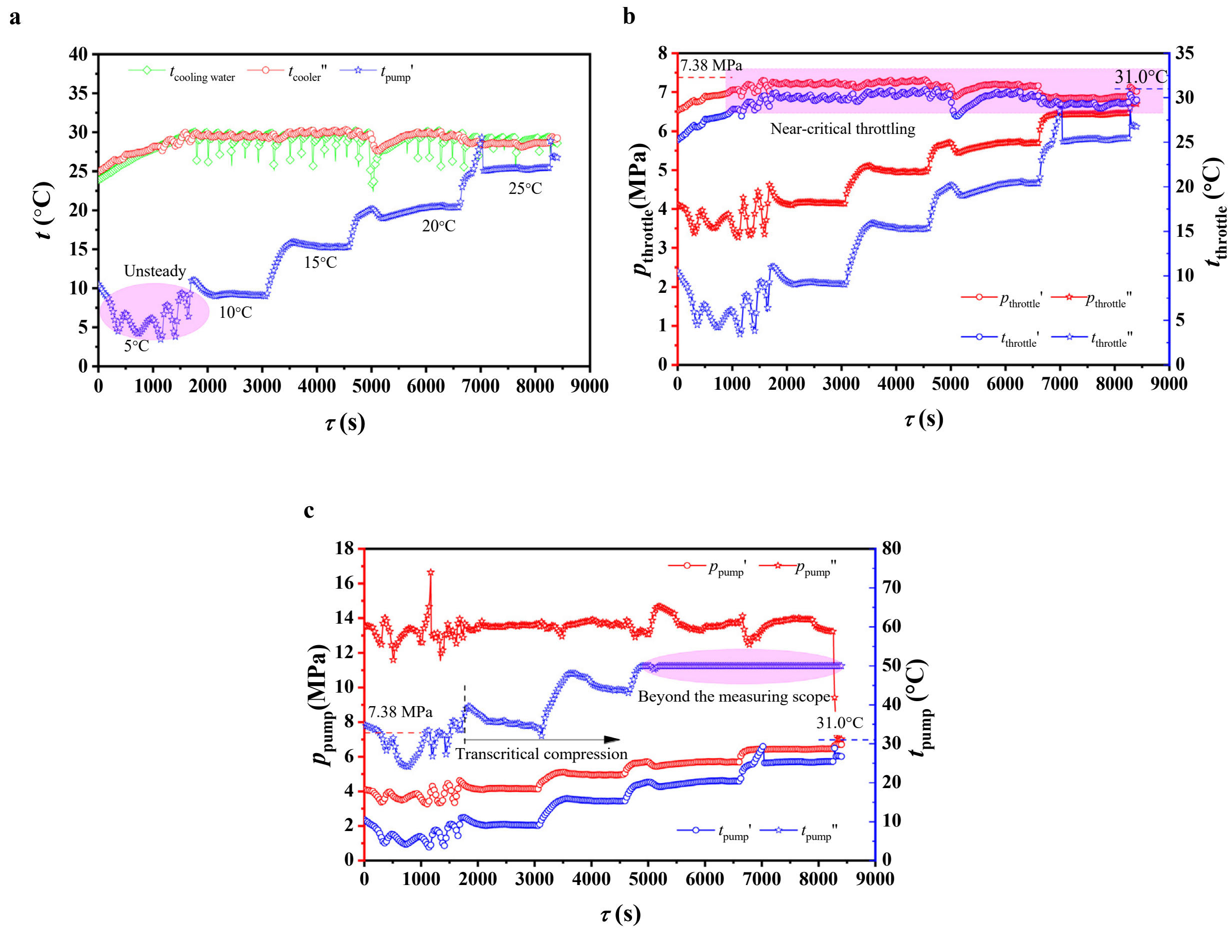

Fig. 6. Variation of the considered parameters. (a. the cooling effect; b. states in the throttle valve; c. states in the pump).

(1) The self-condensing CO_2 transcritical power cycle can operate well with the cooling water as warm as 30 °C. In addition, the coldest saturated liquid CO_2 at 5 °C can be generated for the pump.

(2) Low temperature at the exit of the throttle valve, or called the pump inlet temperature, is harmful to strengthening the power sub-cycle. Therefore, appropriate temperature should be set for liquid CO_2 at the entrance of the pump. The determination principle is to achieve safe operation of the pump and high thermal efficiency.

(3) When conventional cooling water at 30 °C was used, the transcritical or near-critical throttling that is necessary in this novel cycle occurred in the experimental investigation. Under high pump inlet temperature, the pump usually experienced a transcritical compression.

Declaration of competing interest

I wish to confirm that there are no known conflicts of interest associated with this publication and all the organizations that funded the research are mentioned in the manuscript.

I confirm that all the persons who contributed to this paper have been mentioned as authors in the manuscript. There are no other persons who satisfied the criteria for authorship but are not listed.

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