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Research on carbon dioxide transcritical refrigeration cycle with vortex tube

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Yefeng Liu, Ying Sun, Dongliang Wang

University of Shanghai for Science &Technology, School of Energy and Power Engineering, Shanghai, 200093，China

Email: yfliu209@163.com (Yefeng Liu)

Abstract. A kind of carbon dioxide transcritical refrigeration cycle with vortex tube is proposed to replace traditional expansion valve to reduce the irreversible loss of the refrigeration system and improve its COP. The mathematic model of carbon dioxide transcritical refrigeration system is built to simulate the effect of the vortex tube performance on the COP. The results show that outlet temperature of gas cooler, the discharge pressure, cold mass fraction and isentropic efficiency of the vortex tube have significant effects on the COP improvement of the proposed system. COP increases with the increase of cold mass fraction, isentropic efficiency of the vortex tube and the gas cooler outlet temperature. However, COP decreases with the increase of the discharge pressure. When the cold mass fraction is 0.8, the COP improvement increases by 12%.

1. Introduction

The natural refrigerant carbon dioxide (CO_2), which is a probable replacement for traditional refrigerants such as HCFCs (R22) or HFCs (R134a and R404A), is environmentally friendly (ODP=0, GWP=1), non-flammable and non-toxic. CO_2 has excellent heat transfer coefficients and compatibility with material of refrigeration system. As an environmentally friendly refrigerant that replaces traditional refrigerants, CO_2 has been popularized in heat pump water heaters, automotive air conditioners, and heating application[1]. However, CO_2 has a relatively high operating pressure due to its low critical temperature (31.1℃) and high critical pressure (7.38MPa), a transcritical cycle system is usually adopted. When the traditional expansion valve is used for isenthalpic expansion, the irreversible loss caused by the expansion valve is as high as 40% which is higher than that of the compressor [2]. Therefore, the COP of CO_2 transcritical refrigeration system is usually low, reducing cycle throttling loss becomes an important way to improve the COP of CO_2 system.

The vortex tube is a mechanical device operating as a refrigerating machine without any moving parts, by using the Ranque-Hilsch effect to separate a compressed gas stream into a low total temperature region and a high one. In 2000, Li[3] first proposed the use of vortex tubes instead of throttle valves to reduce throttling losses. A certain thermodynamic analysis and calculation under the same working conditions to compare the irreversible loss of the throttle valve, the vortex tube and the turbine expander as the expansion device of CO_2 transcritical refrigeration cycle were performed by He[4]. Using vortex tube as an expansion device is one of the promising cycle modifications to improve COP of CO_2 transcritical cycle [5].

Currently, most of the researches on vortex tube are generally about compressed air, there are few studies on refrigerants such as CO_2 existing in gas-liquid phase transition, and they are not comprehensive enough[6-13]. The influence of the inlet temperature, pressure, the isentropic efficiency and cold mass fraction on the COP improvement of CO_2 transcritical refrigeration system by vortex tube is analyzed in this paper, which provides a theoretical basis for the application of the vortex tube and the optimization on the performance of CO_2 transcritical refrigeration cycle.

2. CO_2 Transcritical refrigeration Cycle with vortex tube

2.1 CO_2 Transcritical Basic Refrigeration Cycle

In the CO_2 transcritical basic refrigeration cycle, a traditional expansion valve is taken usually, as shown in Fig.1. CO_2 gas is discharged from the compressor (state 2b), after no phase-change heat release process (state 3) via a gas cooler, CO_2 enters the expansion valve and is expanded from the supercritical state to the two-phase state (state 3b), then it enters the evaporator to absorb heat. Finally, CO2 returns to the compressor.

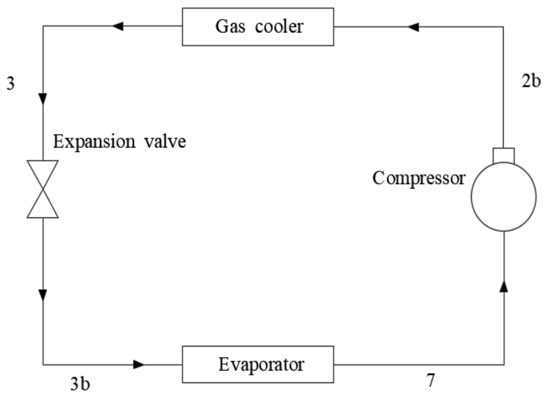

Figure 1. Schematic of CO_2 transcritical basic refrigeration cycle

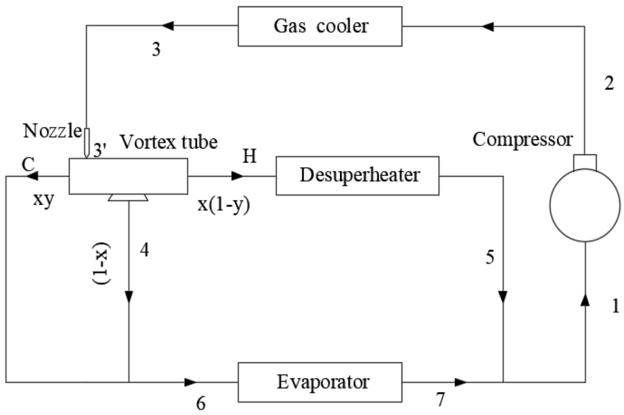

Figure 2. Schematic of CO_2 transcritical refrigeration cycle with vortex tube

2.2 CO_2 Transcritical refrigeration cycle with vortex tube

The traditional vortex tube mainly contains an inlet nozzle, a vortex chamber, a cold-end orifice, a hot end control valve and a tube. Due to the expansion of high pressure CO_2 through the vortex tube nozzle into the two-phase (gas-liquid) region, the vortex tube designed in this paper has added a saturated liquid outlet compared to the traditional vortex tube, as shown in Fig.2. Firstly, the CO_2 gas coming from the gas cooler is expanded to evaporation pressure and divided into saturated liquid (state 4), saturated vapor (state C) and superheated gas (state H). Then, the saturated liquid and vapor are mixed again (state 6) before the evaporator. The superheated gas is cooled in the desuperheater to state 5. Finally, the cooled gas through the desuperheater is mixed with the gas coming from the evaporator (state 7).

Two types of CO_2 transcritical refrigeration cycles are presented on the p-h diagram in Figure 3, and each state point is corresponded to the state point on Figure 1 and Figure 2. The cycle of the basic system is 7-2b-3-3b-7. It can be seen from Figure 3 that the enthalpy of the outlet state point 3'of the vortex tube nozzle after expansion is lower than that of the throttle outlet state point 3b, which is closer to the isentropic state point 3s. This is due to the higher isentropic effect of the vortex tube nozzle, which reduces the irreversible loss of the CO_2 transcritical refrigeration cycle. At the same time, the superheated gas separated by the vortex tube is cooled by the desuperheater to further reduce the compressor inlet temperature and improve the system performance COP.

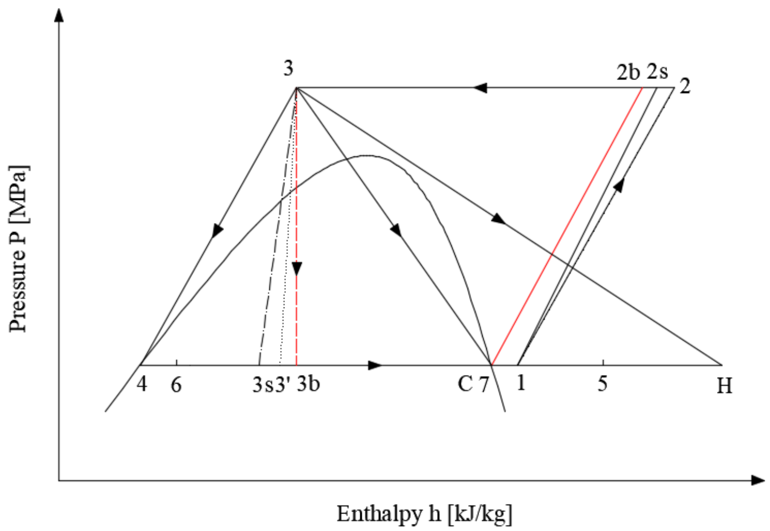

Figure 3. P-h diagram of CO_2 transcritical refrigeration cycle of the two systems

3. Thermodynamic analysis and performance simulation

3.1 CO_2 transcritical refrigeration basic cycle

According to Figure 3, the calculation of COP_b for the basic cycle with throttle valve is the following,

$$COP_T=(h_7-h_{3b})/(h_{2b}-h_7) \tag{1}$$

The vortex tube nozzle outlet enthalpy h_3'at given vortex tube isentropic efficiency η_n is calculated by,

$$h_3'= h_3-\eta_n (h_3-h_{3s}) \tag{2}$$

The quality is found $x = x$ (Pe, h_3') by using thermodynamic property code in REFPROP [14].

According to the definition of the cold mass fraction y, it is calculated by,

$$y = m_c /(m_h+m_c) \tag{3}$$

Assuming that the refrigerant mass flow rate entering the vortex tube is 1 kg/s, and the liquid $(1-x)$ kg is separated from the vortex tube, then the total mass flow rate from cold end at state C and from hot end at state H is x, that is,

$$m_H+m_C = x \tag{4}$$

Equation (3) and equation (4) are solved, then the fraction of saturated vapor is separated at cold end: $m_C = xy$, and the rest separated as superheated gas at hot end: $m_H = x(1-y)$, as shown in Fig.2. Based on the first law of thermodynamics, the enthalpy of the superheated gas h_H is calculated by,

$$h_H = (h_3'-(1-x)h_4-xyh_c)/(x(1-y)) \tag{5}$$

State 6 is the mixing point of state 4 and state C, so h_6 is calculated by,

$$h_6 = ((1-x)h_4+xyh_7)/(1-x+xy) \tag{6}$$

State 5 can be calculated by using the effectiveness of desuperheater ε,

$$t_5 = t_h{\cdot}\varepsilon(t_H-t_{wi}) \tag{7}$$

State 1 is the mixing point of state 5 and state 7, so enthalpy at state 1 is found by,

$$h_1 = x(1-y)h_5+(1-x+xy)h_7 \tag{8}$$

For given compressor efficiencyη_c and compressor discharge pressure P, enthalpy at state 2 can be found by,

$$h_2 = h_1+\eta_c(h_{2s}-h_1) \tag{9}$$

Based on the above calculation, the $COPv$ of the proposed system is obtained by,

$$COP_v = (1-x+xy)(h_7-h_6)/(h_2-h_1) \tag{10}$$

The code "$\triangle COP$" is utilized to describe COP improvement of the proposed system over the basic system, and it is obtained by,

$$\triangle COP = (COP_v-COP_T)*100\%/COP_T \tag{11}$$

4. Results and Discussion

Assuming that the evaporation temperature is 0°C, the isentropic efficiency of compressor is 0.8, the outlet temperature of the desuperheater is the same as the gas cooler outlet temperature. The influences of the gas cooler outlet temperature T_3, the isentropic efficiency of the vortex tube, the cold mass fraction y and the discharge pressure P_3 on the system *COP* are analyzed. The variables setting is shown in Table 1**.**

Table 1. Thermodynamic cycle variables setting

	Gas cooler outlet temperature T_3 (℃)	Vortex tube isentropic efficiency η_n	Cold mass fraction y	Discharge pressure P_3 (MPa)
Reference value	40	0.8	0.5	9
Variation range	24～45	0～1.0	0.3～0.8	8～12

Figure 4 – Figure 7 show the effects of the above variables on *COPs* of the two systems. In addition, the effects on *COP* improvement of the vortex tube system compared with the expansion valve system are also displayed.

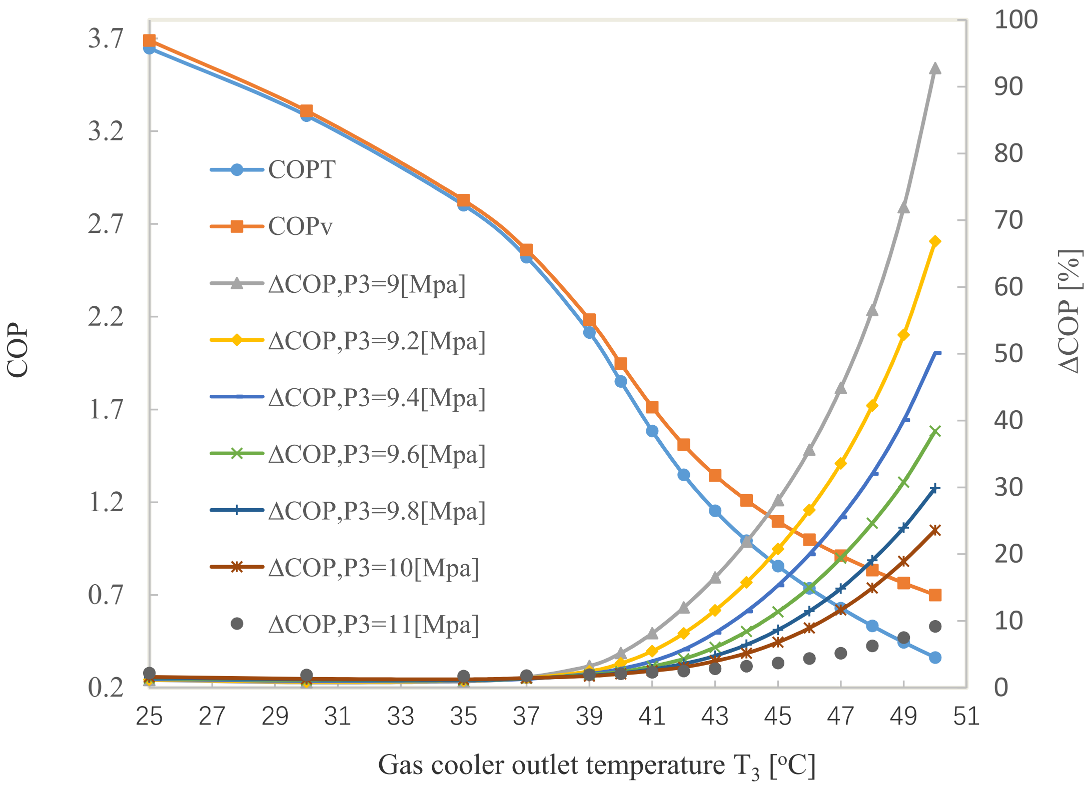

Figure 4. Variation of *COP* with gas cooler outlet temperature

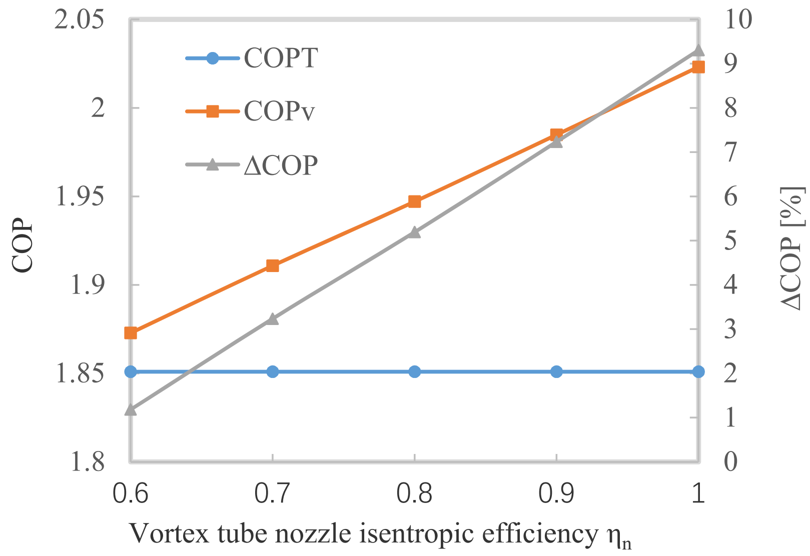

Figure 5. Variation of *COP* with vortex tube isentropic efficiency

It can be seen from Figure 4 that as the gas cooler outlet temperature increases, the COP of the both of the systems decreases. However, the △COP increases with the increase of gas cooler outlet temperatures. Especially when the gas cooler outlet temperature is higher than 40°C, the increases is significantly high. For example, if the discharge pressure is 9 MPa, and the temperature is 45°C, the COP increases by 28%. When the temperature is 47°C, the COP increases by 45%. Therefore, for the higher gas cooler outlet temperature and lower discharge pressure, using vortex tube is very effective way to improve COP. The reason is that when the gas cooler outlet temperature is higher, the vapor quality of the CO2 at the outlet of the vortex tube is higher, which causes more efficient vortex tube energy separation to benefit the COP improvement.

Figure 5 shows that when the isentropic efficiency of the vortex tube nozzle increases, the system COP improvement can increase up to 9.3%. The reason is that the increasing of isentropic efficiency can significantly reduce the irreversible loss of the expansion process and improve the COP.

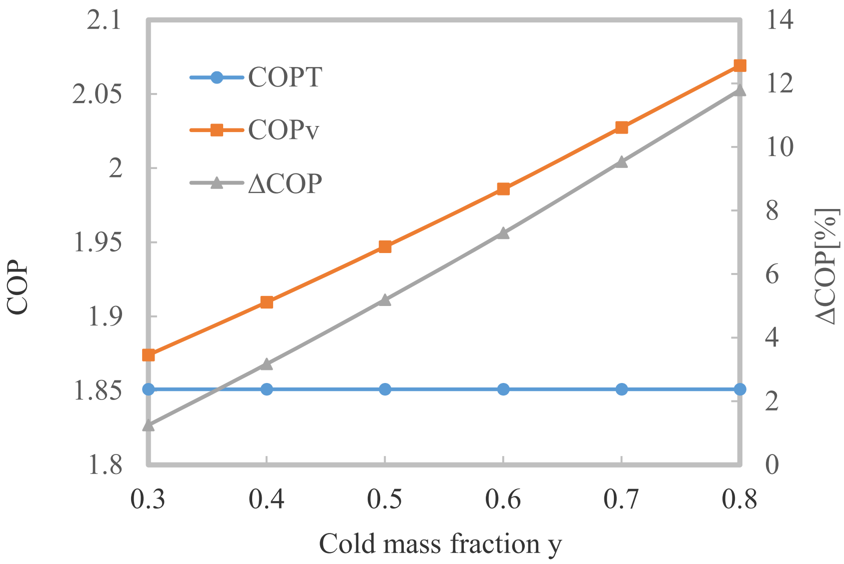

Figure 6. Variation of *COP* with cold mass fraction

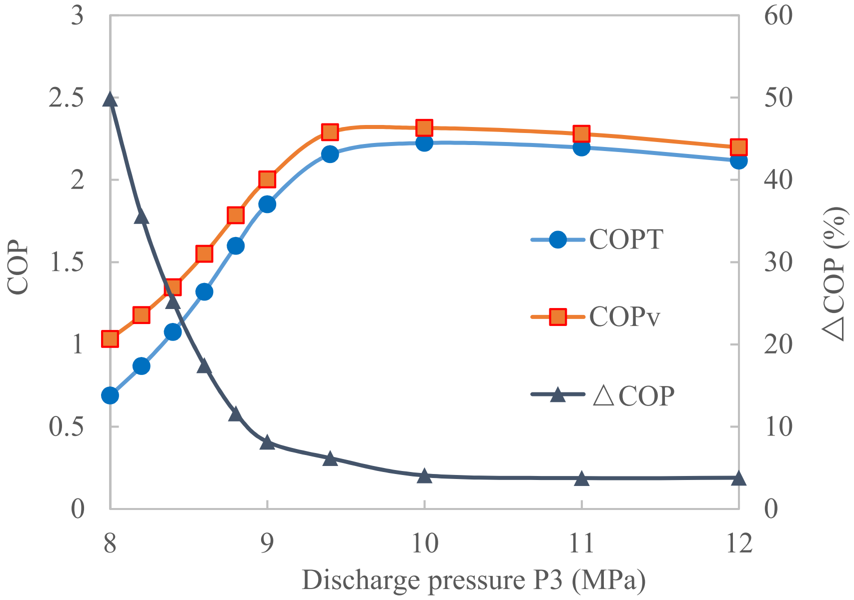

Figure 7. Variation of *COP* with discharge pressure

Figure 6 shows the COP of the vortex tube system gradually increases with the increase of cold mass fraction y, and when y = 0.8, the COP improvement increases by 12%. The △COP increases due to the heat rejection increasing through the desuperheater.

It can be found from Figure 7 that the COP increases with the increase of discharge pressure, but the △COP decreases. For example, when the discharge pressure is 8.4 MPa, the △COP increases by 21%. When the discharge pressure is 9.0 MPa, the △COP increases by 5%. Especially when the gas cooler outlet pressure is lower than 10 MPa, the increases is significantly higher.

5. Conclusion

One new kind of CO_2 transcritical refrigeration cycle is proposed in order to improve *COP*. The operating performance and characteristics of the proposed system are simulated, and the effects of main influence factors on *COP* improvement are analysed. The following conclusions are drawn:

- The higher the isentropic efficiency of vortex tube, the higher the COP improvement is. The COP improvement can be up to 9.3%.
- The COP of proposed system gradually increases with the increase of cold mass fraction, and when cold mass fraction is 0.8, the COP improvement increases by 12%.
- The effect of vortex tube inlet temperature and discharge pressure are more obvious compared to the isentropic efficiency and the cold mass fraction of the vortex tube nozzle. The application of vortex tube in CO_2 transcritical refrigeration cycle is highly recommended when a higher outlet temperature and a lower discharge pressure is required. At T3 = 45°C, P3= 9 MPa, the COP increased by 28%.

Acknowledgements

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