Study on the Performance of Carbon Dioxide Trans-Critical Refrigeration Cycle with Vortex Tube Expansion

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Keywords: Carbon dioxide; Trans-critical cycle; Vortex tube; COP

Abstract. In this paper, one carbon dioxide trans-critical refrigeration cycle with vortex tube expansion is proposed to replace traditional expansion valve in order to reduce the irreversible loss of the refrigeration system and improve its COP. The thermodynamic model of carbon dioxide trans-critical refrigeration system is built to simulate the effect of the vortex tube performance on the COP. The results show that the gas cooler outlet temperature and the discharge pressure have significant effects on the COP improvement of the vortex tube system. COP increases with the increase of the gas cooler outlet temperature and the decrease of the discharge pressure. When the gas cooler outlet temperature is 45°C and evaporation temperature is 0°C, the COP increases 28% compared to traditional system with expansion valve. In addition, increasing the isentropic efficiency and the cold mass fraction of the vortex tube can also increase the COP by about 10%.

Introduction

In recent years, the ever-increasing menace of ozone layer depletion and global warming has caused a great deal of attention from insiders to environmental protection and energy-saving refrigerants. The natural refrigerant 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. At a given saturation temperature and pressure, the surface tension, the liquid viscosity, and ratio of liquid to vapor density of CO_2 are the lowest. It also 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°C) and high critical pressure (7.38MPa), a trans-critical 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 trans-critical 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 trans-critical refrigeration cycle were performed by He[4]. Use of vortex tube as an expansion device in CO_2 trans-critical cycle seems to be one of the promising cycle modifications to improve the system performance[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 [6-15]. The influence of the inlet temperature, pressure, the isentropic efficiency and cold mass fraction on the COP improvement of CO_2 trans-critical cycle by using vortex tube

instead of expansion valve is analyzed in this paper, which provides a theoretical basis for the design of the vortex tube and the optimization on the performance of CO_2 trans-critical refrigeration cycle.

CO₂ Trans-critical Refrigeration Cycle with Vortex Tube Expansion

CO₂ Trans-critical Basic Refrigeration Cycle. In the CO_2 trans-critical basic refrigeration cycle, a traditional expansion valve is taken usually, as shown in Fig.1. High pressure and high temperature 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, CO_2 returns to the compressor.
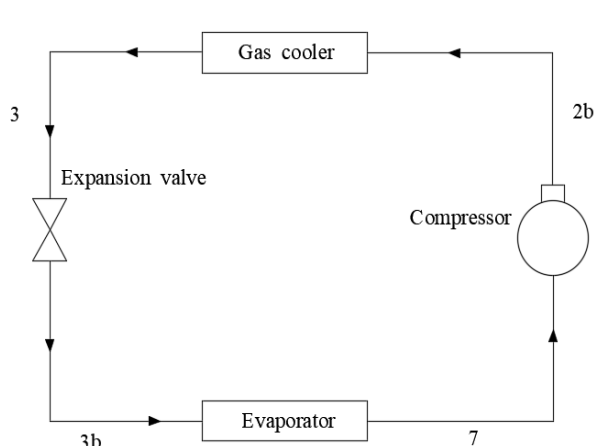

Fig.1. Schematic of CO_2 trans-critical basic refrigeration cycle
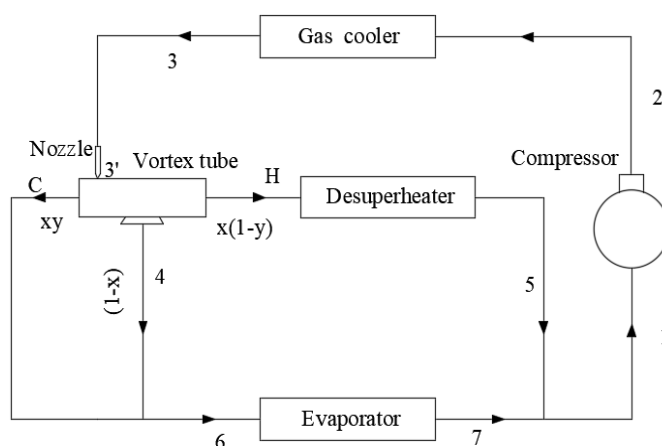
Fig.2. Schematic of CO_2 trans-critical refrigeration cycle with vortex tube expansion

CO₂ Trans-critical Refrigeration Cycle with Vortex Tube Expansion. The traditional vortex tube mainly contains the following parts: an inlet nozzle, a vortex chamber, a cold-end orifice, a hot end control valve and a tube. It converts a gas flow initially homogeneous in temperature, into two separate flows of differing temperatures. 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. The gas is expanded from discharge pressure to evaporation pressure and divided into three fractions: saturated liquid (state 4), which is collected in a ring inside the vortex tube; saturated vapor (state C) and superheated gas (state H), which are created because of the Ranque–Hilsch effect. The saturated liquid and vapor are mixed again (state 6) and going through the evaporator to give useful cooling effect. The superheated gas is cooled in the desuperheater to state 5 and mixed with the gas coming from the evaporator (state 7) before entering the compressor (state 1).

Two types of CO_2 trans-critical refrigeration cycles are presented on the p-h diagram, and each state point corresponds to the state point on Figs.1 and Figs.2. The cycle of the basic system is 7-2b-3-3b-7. It can be seen from Fig.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 trans-critical refrigeration cycle system. 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.
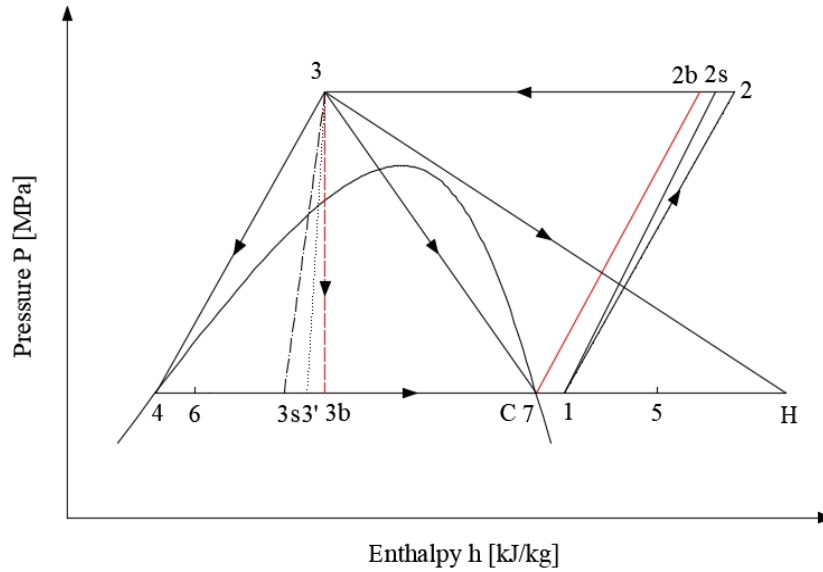
Fig.3. p-h diagram of CO_2 trans-critical refrigeration cycle of the two systems

Thermodynamic Analysis and Calculation of Two Systems

Thermodynamic Calculations of CO_2 Trans-critical Basic Refrigeration Cycle. According to Fig.3, the COP_T formula for CO_2 trans-critical basic refrigeration cycle system is as follows,

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

The enthalpy at the vortex tube nozzle exit for given nozzle efficiency η_e can be calculated by,

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

Then the vapor quality is found $x = x(p_{ev}, h_{3'})$ by using properties relation.

According to the definition of the cold mass fraction y, it can be calculated as the following,

$$y = \frac{m_c}{m_h + m_c} \tag{3}$$

Assuming that all the liquid (1-x) is separated out from state 4, the sum of mass flow rate of saturated vapor at state C and superheated gas at state H, that is,

$$m_h + m_c = x \tag{4}$$

Solving equations (3) and (4) can get the following results: mass fraction of saturated vapor is separated as cold fluid $m_c = xy$, the rest absorbs all the kinetic energies and separated as hot fluid is $m_c = x(1 - y)$. Properties at state point H is given by,

$$h_H[x(1 - y)] + xyh_C + (1 - x)h_4 = h_{3'} \tag{5}$$

State point 6 is the mixing state point of state point 4 and state point C, the inlet enthalpy of evaporator h_6 can be found by,

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

State point 1 is the mixing state of state point 5 and state point 7, the inlet enthalpy of the compressor h_1 is calculated by,

$$h_1 = x(1 - y)h_5 + (1 - x + xy)h_7 \tag{7}$$

As known compressor efficiency η_c and compressor discharge pressure p_3, the outlet enthalpy of the compressor h₂ is found by,

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

(8)

Based on the above calculations, the COP$_v$ of the vortex tube system is obtained by,

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

(9)

The improvement of COP is presented by △COP as the following,

$$\Delta COP = \frac{COP_V}{COP_T}$$

(10)

Calculation Results and Analysis

Assuming that the evaporation temperature is 0°C, the compressor isentropic efficiency 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 T₃, the isentropic efficiency of the vortex tube nozzle η_e, the cold mass fraction y and the discharge pressure P₃ on the system COP are analyzed. The variables setting is shown in Table 1.

Table 1 Thermodynamic cycle change parameter settings

Variables	Gas cooler outlet temperature T₃ [°C]	Vortex tube isentropic efficiency η_e	Cold mass fraction y	Discharge pressure P₃ [MPa]
Reference value	40	0.8	0.5	9
Range	25-45	0-1	0.3-0.8	8-12

Fig.4 - Fig.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 expansion system compared with the valve expansion system are also displayed.
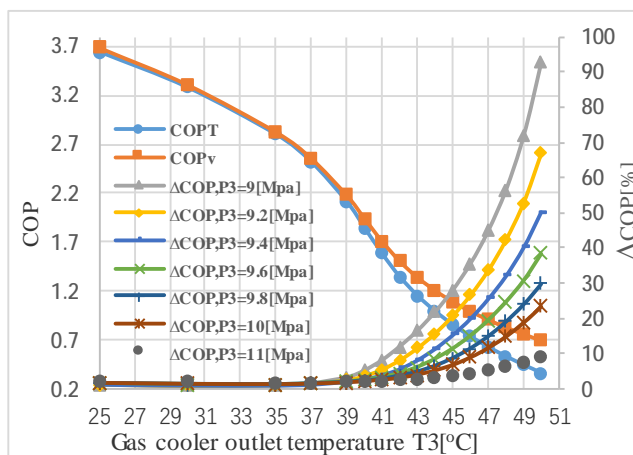

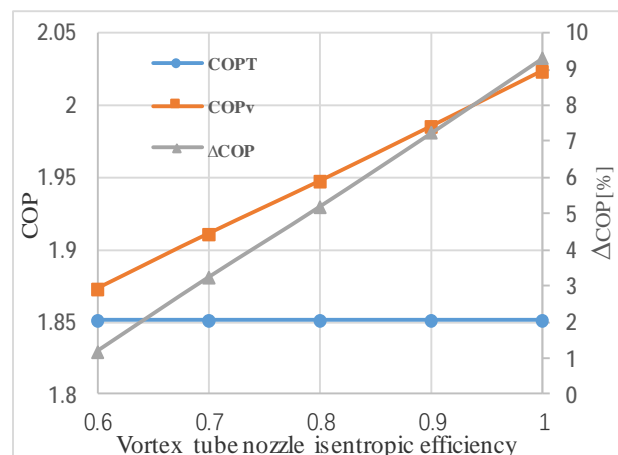Fig.4. Variation of COP with gas cooler outlet temperature

Fig.5. Variation of COP with vortex tube nozzle isentropic efficiency

It can be seen from Fig.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, when 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, the use of vortex tube is more effective in terms of higher COP improvement compared to basic cycle. The reason is that when the gas cooler outlet temperature is higher, the quality of the working fluid at the outlet of the vortex tube is higher which causes more efficient vortex tube energy separation, benefiting the COP improvement.

Fig. 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.

Fig.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 improvement △COP increases due to the increase in heat rejection through the desuperheater.
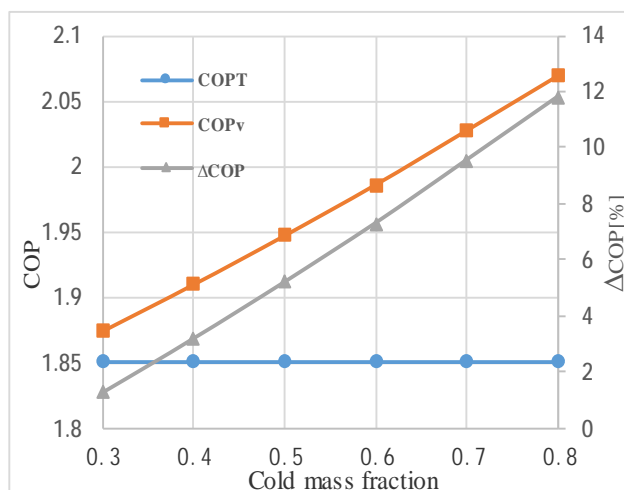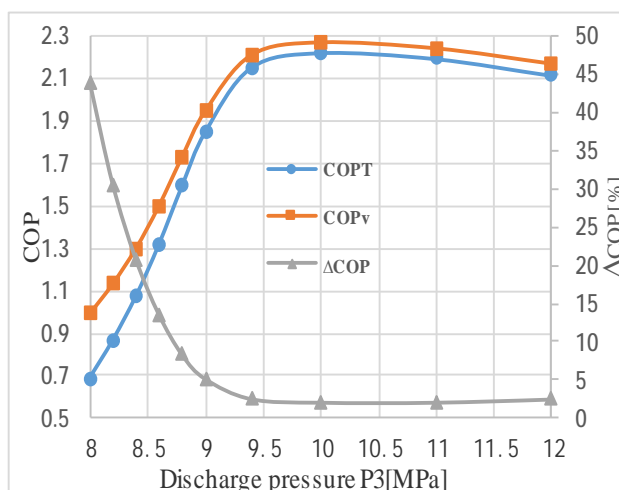

Fig.6. Variation of COP with cold mass fraction

Fig.7. Variation of COP with discharge pressure

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

Conclusion

This paper proposes the use of vortex tube expansion instead of valve expansion to reduce irreversible losses and improve system performance COP in a CO_2 trans-critical refrigeration cycle. The influence of various parameters on the COP is analyzed, the following conclusions are drawn.

(1) The higher the isentropic efficiency and the cold mass fraction of the vortex tube nozzle, the higher the COP improvement of the system, the increment can be up to 10%.

(2) The effect of gas cooler outlet temperature and discharge pressure are more significant compared to the isentropic efficiency and the cold mass fraction of the vortex tube nozzle. The application of vortex tube is highly recommended when a higher outlet temperature and a lower discharge pressure is required. At T_3=45°C, P_3=9MPa, the COP increased by 28%.

Acknowledgements

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