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Yingbai Xie
North China Electric Power University

Ganglei Sun
North China Electric Power University

Liyong Lun
North China Electric Power University

Hang Su
North China Electric Power University

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Yingbai Xie[1], Ganglei Sun[2]*, Liyong Lun[3], Hang Su[4]

[1,2,3,4] Department of Power Engineering, North China Electric Power University,
Baoding, Hebei, China
Tel: 86-312-7522882, Fax: 86-312-7522440

[1] E-mail: xieyb@ncepubd.edu.cn

[2] E-mail: sunganglei2003@163.com

[3] E-mail: lunliyong@126.com

[4] E-mail: mona-liassteor@sohu.com

ABSTRACT

Two-stage vapor compression refrigeration cycle is accomplished with an intercooler. Statues of the refrigerant vapor leaving the intercooler will have the effect on the system performance. The thermodynamic analysis of the supercritical two-stage refrigeration cycle using CO_2 as the refrigerant is processed, with a condensing temperature of 35℃ and an evaporating temperature of -5℃. When the intercooler operates at about the geometric mean of the evaporating and condensing pressures, compared to the single stage cycle, the coefficient of performance (COP) of the two-stage is improved. It reaches maximum when the temperature entering into the high pressure compressor is saturated. As the degree of the vapor leaving the intercooler increases, the COP of the two-stage system decreases.

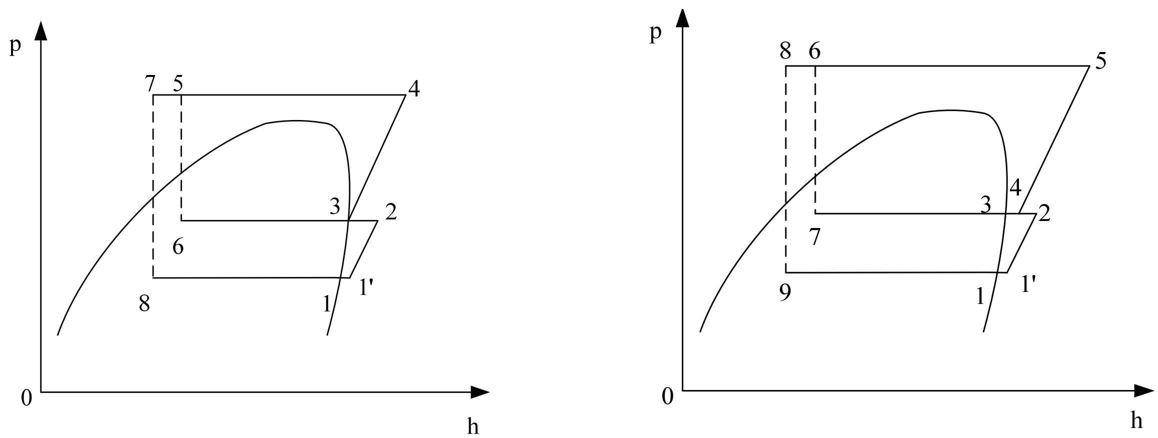
1. INTRODUCTION

According to the negative impact of the ozone layer destruction and the greenhouse effect, natural working fluids are used to substitute the man-made refrigerants (Ding, 2002, Zhang and Wang, 2007, Ma et al, 2002). In addition to water and air, CO_2 is the most environmental benign refrigerant among the commonly used natural refrigerants in the refrigeration cycle. It has many advantages, such as the ODP = 0, GDP = 1; safety usage, non-toxic; good physical and chemical characteristics; large unit volumetric refrigeration capability, etc. CO_2 is becoming the best alternative substitution to CFCs and HCFCs substance in the refrigeration field.

During the cooling mode operation, the system pressure of the cycle adopted the CO_2 as refrigerant is generally above its critical point, for the critical temperature of CO_2 is only 31℃, which lower than that of the typical summer conditions (35 ℃). Lorentzen and Pettersen (1993) have shown in their seminal studies that difficulties connected with the low critical temperature of CO_2 can be successfully overcome by operating the system in the supercritical mode, where single phase heat rejection occurs above the critical temperature in the gas cooler instead of condenser as in conventional systems, and where pressure and temperature can be controlled independently to obtain optimum performance. The gliding temperature in the gas cooler makes the supercritical CO_2 systems more economical for simultaneous cooling and heating applications. At the same time, for supercritical CO_2 refrigeration cycle， the lower critical temperature results in two issues: First, the operation pressure of the system is relatively higher; Secondly, the coefficient of the performance of the cycle is relatively low. To improve the system COP, one scheme is using the two-stage supercritical CO_2 compression refrigeration cycle (Wang et al, 2001, Yang et al, 2005, Guan et al, 2004). In this paper, cyclic model of two-stage supercritical CO_2 compression refrigeration cycle is established, and the thermodynamic principle is applied to analyze the impact of the high pressure compressor suction refrigerant steam conditions.

2. THERMODYNAMIC MODEL

Using two-stage supercritical CO_2 compression refrigeration cycle is propitious to reduce the compressor discharge temperature and improve the system COP (Liu et al, 2002). The P-h Diagrams of the two-stage supercritical CO_2 compression refrigeration cycle with high pressure compressor suction saturated or not are shown in Figure 1.



(a) Complete intercooling (b) Incomplete intercooling

Figure 1: Pressure-Enthalpy diagram for two-stage supercritical CO_2 compression refrigeration cycle

For system shown in Figure 1, mathematical model is presented as below:

ⅰ. For complete intercooling situation, the following formulas for two-stage supercritical CO_2 compression refrigeration cycle are obtained

$$q_{ev} = h_1 - h_8 \tag{1}$$

$$w_{lpc} = h_2 - h_{1'} \tag{2}$$

$$w_{hpc} = h_4 - h_3 \tag{3}$$

$$q_{gc} = h_4 - h_5 \tag{4}$$

ⅱ. For incomplete intercooling situation, the following formulas for two-stage supercritical CO_2 compression refrigeration cycle are obtained

$$q_{ev} = h_1 - h_9 \tag{5}$$

$$w_{lpc} = h_2 - h_{1'} \tag{6}$$

$$w_{hpc} = h_5 - h_4 \tag{7}$$

$$q_{gc} = h_5 - h_6 \tag{8}$$

iii. Energy balance equation for the entire system is

$$q_{ev} + w_{comp} = q_{gc} \tag{9}$$

Where
$$w_{comp} = w_{lpc} + w_{hpc} \tag{10}$$

iv. Coefficient of performance is defined as

$$cop = \frac{Q_0}{\overset{\circ}{m}_{lpc}\, w_{lpc} + \overset{\circ}{m}_{hpc}\, w_{hpc}} \tag{11}$$

v . Pressure of the intercooler is the geometric mean of the evaporating and condensing pressures

$$p_{opt} = \sqrt{p_{ev} p_{gc}} \tag{12}$$

The following assumptions have been made to simplify the analyses:
(1) System operates in steady state.
(2) The isentropic efficiencies for low pressure compressor and high pressure compressor are 0.8, separately.
(3) Heat transfer with the surroundings is negligible.
(4) No pressure drop occurs in evaporator, gas cooler, intercooler and their connecting pipelines.
(5) The saturated evaporator temperature is -5℃, the exit temperature of gas cooler is 35℃.

Based on the above formulas and assumptions, numerical simulation for two-stage supercritical CO_2 compression refrigeration cycle is processed. REFPROP 6.0 is used to calculate the thermodynamic properties of CO_2.

3. RESULTS AND ANALYSIS

The COP of two-stage supercritical CO_2 compression refrigeration cycle is compared with that of the single stage cycle as shown in Figure 2. Heat transfer in the intercooler has a temperature difference of 5℃. Intercooler operates at the geometric mean of the evaporating and condensing pressures bases on equation (16). As Figure 2 shown, either the single stage or the two-stage cycles have the same COP configuration with the pressure variation in gas cooler. But the COP of the two-stage is higher than that of the single stage. Moreover, the COP of complete intercooler is higher than that of the incomplete intercooler situation. For the three system type, with a gas exit temperature at 35℃ and an evaporating temperature at -5℃, the COP reaches maximum when the gas cooler pressure is about 9MPa. After this pressure value, the COP of the cycles decrease when the pressure in gas cooler increase, but the slope is flat before the pressure reaches this pressure.

The impact of the high pressure compressor suction refrigerant steam conditions is shown in Figure 3. When intercooler and the gas cooler operate at their optimum pressure, as the degree of superheated refrigerant CO_2 steam temperature increases, the COP of two-stage system decreases.

Figure 4 shows the work consumed by high pressure compressor to the degree of superheated suction refrigerant steam. As the degree of superheated suction refrigerant steam increases, Power input per unit of refrigerating capacity also increases. Figure 5 shows the heat load of gas cooler with the degree of superheated suction refrigerant steam. It states that the degree of superheated suction refrigerant steam on the heat load of gas cooler has the same effect as the work consumed. The variation of the outlet pressures with the high pressure compressor and the low pressure compressor are shown in Figure 6. At specified condition, the ratio of compensating pressure and evaporating pressure is near same.

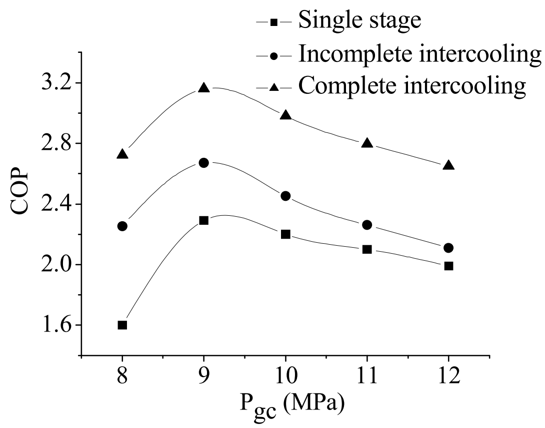


Figure 2: The COP of two-stage system compares to the single-stage system

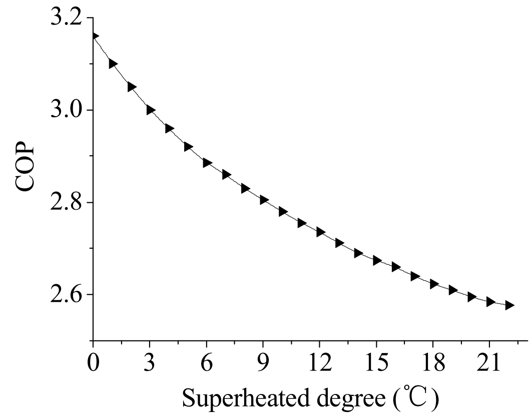


Figure 3: Impact of degree of superheat on the system performance

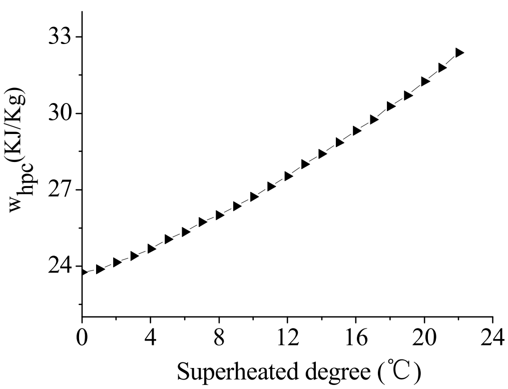


Figure 4: Impact of degree of superheat on power input

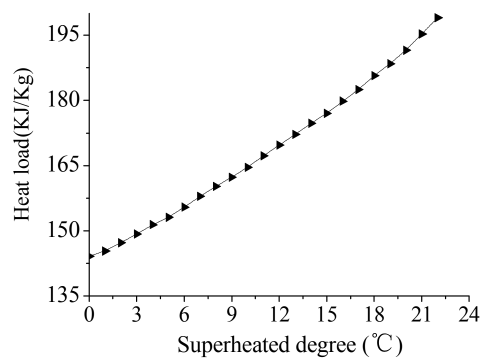


Figure 5: Impact of degree of superheat on the heat load of the gas cooler

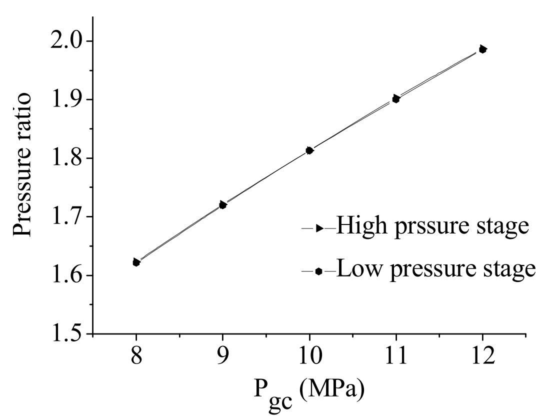


Figure 6: Variation of the outlet pressure with the high and the low pressure compressor

4. CONCLUSIONS

The thermodynamic analysis and performance simulation of a two-stage supercritical CO_2 refrigeration cycle have been processed. Results show that the COP for two-stage supercritical CO_2 refrigeration cycle is higher than that of the single stage cycle. When the refrigerant enters into high pressure compressor is saturated, system has a maximum COP. As the superheated degree of the suction refrigerant steam increases, the COP, work consumed by high pressure compressor and the heat load of gas cooler also increase. As a result of list analyses, it is suggested that the refrigerant enters into the high pressure compressor is saturated or not should be considered in the optimum system design.

NOMENCLATURE

$\mathring{m}$	mass flow rate	(Kg/s)
P	pressure	(MPa)
h	enthalpy	(KJ/Kg)
q	specific heat transfer	(KJ/Kg)
w	specific work	(KJ/Kg)
Q_0	refrigeration quantity	(KW)
COP	coefficient of performance	(–)

Subscripts

1-9	refrigerant state points
1′	refrigerant state point
ev	evaporator
gc	gas cooler
comp	compressor
lpc	low pressure compressor
hpc	high pressure compressor
opt	optimization
p	participants

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