Energy improvements of CO_2 transcritical refrigeration cycles

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ABSTRACT:

A hybrid vapor compression refrigeration (HVCR) system which combines a vapor compression refrigeration (VCR) system and an ejector refrigeration (ER) system was developed. The waste heat energy from the gas cooler in the VCR system is applied as driven source towards ER system.

The thermodynamic investigations on the performance of the HVCR system using CO_2 as the refrigerant are performed with energetic methods, and the comparative analyses with the VCR system are conducted. Comprehensive effects of key operating parameters on performance are also studied. The results indicate that for the same cooling capacity, the coefficient of performance (COP) of HVCR system are around 28% higher than that of conventional VCR system. The performance characteristics of the proposed cycle show its application potential in cooling and air-conditioning.

Keywords: Thermodynamic analysis; Ejector; energy; waste heat

1. Introduction:

Due to the environmental concerns about ozone depletion and global warming, CFC, HCHC and HFC refrigerants are now being regulated. Adopting appropriate natural refrigerant such as CO_2, is one of the advanced methods of enhancing system performance and reducing ozone depletion and global warming. Carbon dioxide as a refrigerant has attracted the interest of researchers because of its unique thermal characteristics, such as low viscosity, excellent heat transfer coefficient, no toxicity and no inflammability. At the same time, CO_2 has zero ODP (Ozone depletion potential), negligible GWP (Global warming potential) and very low cost. However, in common refrigeration conditions, the refrigeration system using CO_2 needs to be operated with a transcritical cycle mode because of the lower critical temperature of CO_2 which results in a lower first and second law efficiency. Large amount of efforts were devoted to enhance the performance of these cycles.

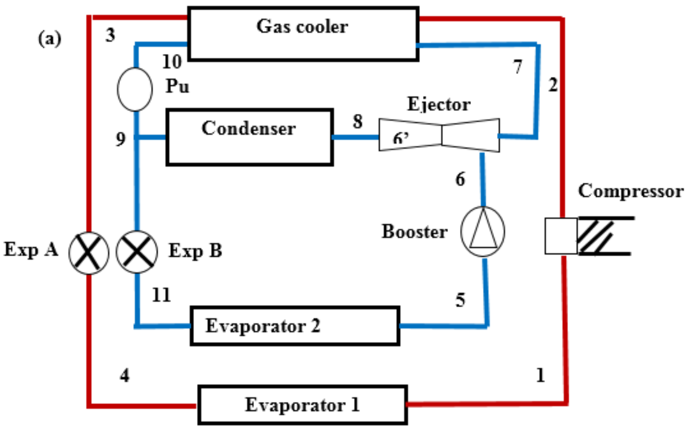
For thermodynamic reasons, the working fluid exiting the compressor (Usually at 100- 200 °C for CO2) should be cooled down to about 35- 55 °C in the gas cooler providing a large amount of energy and exergy, which will be rejected to the environment. This is an ideal thermal energy source to be utilized to increase the refrigeration system performance.

In order to improve the system performance of the CO_2 VCR system and obtain dual-temperature refrigeration function, an ejector cooling cycle driven by low grade heat energy is considered to be combined with the transcritical refrigeration cycle in the present paper. It is hoped that the obtained results could provide some guidelines for refrigeration system designers.

2. Cycle description:

The layout and pressure-enthalpy diagrams of the HVCR system are shown in Figure 1. The proposed system consists of two cycles. It includes the basic refrigeration cycle and a heat driven ejector cooling cycle.

The cycle working principle is described as follows: the superheated gas discharge by the compressor (state 2) enters the gas cooler and leaves at (state 3). In gas cooler, the compressed gas rejects heat to the fluid coming from the condenser. This heat is used to drive the ER system. The gas from state 3 enters a throttling device (Exp A), where the pressure and temperature are reduced and enters evaporator (A) at state 4, where it evaporates and absorbs the heat in evaporator (A). After this, the working fluid enters the compressor as saturated vapor (state 1), and the cycle continues. The ER system is driven thermally by waste heat of the VCR system. The heat transfer between these two cycles is linked through the gas cooler. The high temperature and pressure refrigerant vapor (the primary flow, state 7) enters the ejector through a convergent-divergent nozzle, which accelerates the primary flow from subsonic to supersonic velocity and creates a low pressure region at the nozzle exit. This entrains the low temperature, low-pressure vapor (the secondary flow, state 5) from the evaporator (B) outlet. The secondary flow from the evaporator (B) is first compressed to a relatively high pressure in the booster and then enters the ejector at (state 6). The primary and secondary flows mix in the ejector and then the mixture is discharged to condenser (state 8). After the condensation, the refrigerant at state (9) is divided into two streams; the first one is directed through a pump to gas cooler at state (10) and the second one is directed through an expansion valve (B) (state 11) and into the evaporator (B). The liquid vapor refrigerant mixture evaporates in the evaporator (B) and the ER system thus provides an additional cooling capacity.



(b)

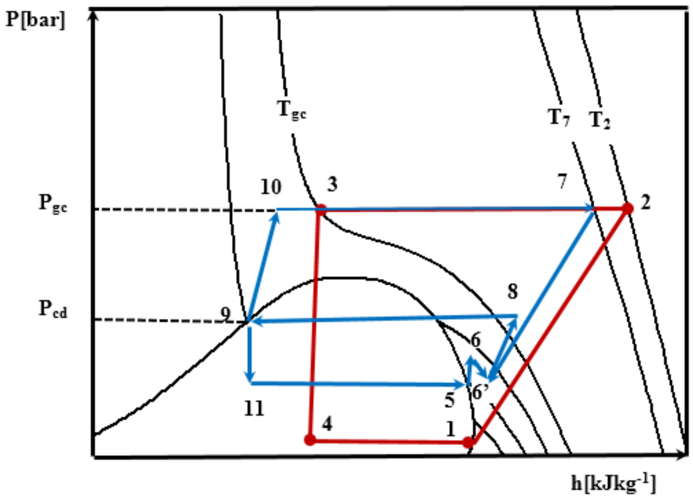


Figure 1. (a) Schematic of a HVCR system. (b) P-h diagram of a HVCR

3. Energetic model :

For the compressor, the input power can be calculated as:

$$W_c = \dot{m}_1 (h_{2,s} - h_1) / \eta_{is} \tag{1}$$

$$\eta_{is} = 0.9343 - 0.04478(\frac{P_2}{P_1}) \tag{2}$$

The input power of the liquid pump is calculated by isentropic efficiency method expressed as,

$$W_{pu} = \dot{m}'(h_{10,s} - h_9) / \eta_{pu} \tag{3}$$

Where η_{pu} represents the isentropic efficiency of the liquid pump and is assumed to be 0.8.

For a real non-isentropic efficiency of the compression process the input power to the booster can be expressed as

$$W_{bo} = \dot{m}''(h_{6,s} - h_5) / \eta_{bo} \tag{4}$$

Where η_{bo} represents the isentropic efficiency of booster and is assumed to be constant.

The total power input the system can be given as:

$$\mathrm{W} = W_c + W_{pu} + W_{bo} \tag{5}$$

For the evaporator A in the VCR system, the cooling capacity is:

$$Q_{e\_A} = \dot{m}_1 (h_1 - h_4) \tag{6}$$

For the evaporator B in the ER system, the cooling capacity is:

$$Q_{e\_\mathrm{B}} = \dot{m}''(h_5 - h_{11}) \tag{7}$$

The total cooling capacity of the HVCR system is:

$$Q_{sys} = Q_{e\_A} + Q_{e\_B} \tag{8}$$

The heat flow rate at the gas cooler is:

$$Q_{gc} = \dot{m}_1(h_2 - h_3) \tag{9}$$

Using the ideal heat exchange condition and from an energy balance principle:

$$\dot{m}_1(h_2 - h_3) = \dot{m}'(h_7 - h_{10}) \tag{10}$$

The heat flow rate at the condenser is:

$$Q_{cd} = \dot{m}_{tot}(h_9 - h_8) \tag{11}$$

Where:

$$\dot{m}_{tot} = \dot{m}' + \dot{m}'' \tag{12}$$

For expansion valves:

$$h_4 = h_3 \tag{13}$$

$$h_{11} = h_9 \tag{14}$$

For ejector:

$$\dot{m}_{tot} h_8 = \dot{m}'' h_6 + \dot{m}' h_7 \tag{15}$$

Entrainment ratio also is an important parameter that describes the ER system performance. This parameter is related to the cooling capacity and directly depends on the refrigeration type and ejector geometry. The entrainment ratio is the ratio between the mass flow rate of the secondary flow and the mass flow rate of the primary flow, given as:

$$U = \frac{\dot{m}''}{\dot{m}'} \tag{16}$$

Where $\dot{m}''$ and $\dot{m}'$ represent the mass flow rates of the secondary and the primary flow, respectively.

The coefficient of performance of the HVCR system, COP is:

$$COP = \frac{Q_{sys}}{W} \tag{17}$$

For comparison, the coefficient of performance of the VCR system, COP_{basic} is:

$$COP_{basic} = \frac{Q_{e\_A}}{W_c} \tag{18}$$

The ER system performance can be evaluated based on the coefficient of performance, COP_{ej},

$$COP_{ej} = \frac{Q_{e\_B}}{W_{pu} + W_{bo} + Q_{gc}} \tag{19}$$

Another criterion is the COP improvement COP_{imp} which could be used to compare the performance of HVCR system with this of the conventional VCR system, is given as follows:

$$COP_{imp}(\%) = \left(\frac{COP - COP_{basic}}{COP_{basic}}\right) * 100 \tag{20}$$

4. RESULTS AND DISCUSSION :

Results are obtained from the developed model. Initially, for specified temperatures (T_{e_A} = -20 °C, T_{e_B} = 0°C, T_{gc} = 45°C) the effect of varying gas cooler pressure on the performance of a CO_2 HVCR system is reported. Optimum values of performance parameters (at different gas cooler pressure for the HVCR system are compared with those of the VCR baseline system. During the entire analysis the cooling capacity of the HVCR system is assumed to be the same as the basic VCR system and both are assumed to be 100 kW.

The effect of varying gas cooler pressure P_{gc} on the COP and the total mechanical power consumption for the HVCR and the VCR systems are shown in Figure 2.

As can be seen (Figure 2), the total mechanical work consumption W, of both cycles drops first and then increases. For this reason, the system COP rises first and then decreases as previously noted. Since the cooling capacity is the same for the two refrigeration systems, the performance improvement appears in terms of lower mechanical power consumption. The main reason for this is that the large temperature in the gas cooler is utilized to drive an ejector cooling cycle. In this case, the HVCR system exhibits a reasonable value of COP.

In order to illustrate the energy saving potential and advantages of excellent cooling performances, the energetic performance comparisons between HVCR and VCR systems using CO_2 are carried out at the specific operating condition and for the same cooling capacity. It can be observed that the HVCR system has more advantages than the VCR system in terms of system COP. Under the given operating condition and at the optimum gas cooler pressure, the HVCR system shows 25 % higher COP and the total mechanical power consumption W_{sys} is reduced by 20 %. Therefore, it could be concluded that the use of the waste heat from the gas cooler in the VCR system to drive the ejector cooling system could significantly improve the system performance. This indicates that applying HVCR system to obtain better performance with lower energy consumption is feasible.

The variation of the entrainment ratio U and the ejector cycle COP with the gas cooler pressure P_{gc} are presented in Figure 3. It is clear from Figure 3 that the entrainment ratio increases with the gas cooler pressure P_{gc} and vary in the range of 0.43-0.57. It is also demonstrated that the COP_{ej} is low in compare to the VCR system COP, the ejector cycle improve the COP because the heat energy utilized in the ejector cycle is waste heat from the gas cooler.

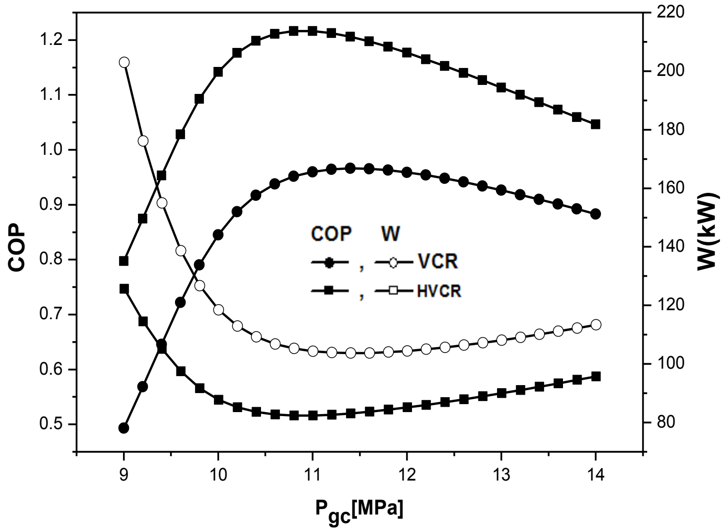


Figure 2. The effect of the gas cooler pressure on the COP end the mechanical power consumption of two systems under specified operation conditions

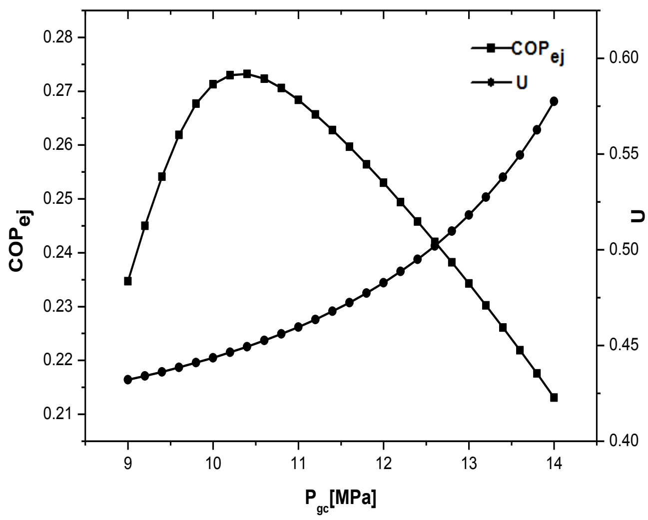


Figure 3. Variation of the ejector COP and the entrainment ratio with the gas cooler pressure

Conclusion:

A proposed hybrid vapor compression refrigeration system (HVCR) is proposed in this paper. The results showed that the HVCR system is proposed to be a candidate system in the cooler applications