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Energy and Exergy Analyses of Different Transcritical CO₂ Refrigeration Cycles

Rabah GOMRI, Nadim KAROUNE, Nabil KHELLAF

University Mentouri Constantine 1, Department of Genie Climatique, 25000, Constantine, Algeria
rabahgomri@yahoo.fr

Abstract: Carbon dioxide has received increasing attention owing to its zero ODP and negligible GWP. Furthermore, carbon dioxide also has desirable thermodynamic properties, such as large specific heat, low viscosity, and large heat conductivity. Carbon dioxide CO_2 has low critical pressure and temperature which are 7.36 MPa and 31.1°C, respectively. The low critical temperature causes the heat rejection process to occur above the critical point and heat absorption process to happen below the critical point. However, due to the high throttling loss, the energy efficiency of the basic transcritical CO_2 cycle is lower than that of the conventional low pressure refrigeration cycle. In the present study three different kinds of transcritical carbon dioxide cycles that are the transcritical cycle with expansion valve (called also the conventional transcritical cycle), the transcritical cycle with expander and the transcritical cycle with ejector are analyzed. The effect of operating parameters on the maximum performance and exergy efficiency of the three cycles is investigated. Results reveal that replacing the expansion valve by an expander or an ejector does not only improve the maximum COP and the exergy efficiency but also reduces the optimal heat rejection pressure.

Keywords: Transcritical, CO₂ Refrigeration Cycle, Expansion, COP, Exergy.

Farklı Transkritik CO₂ Soğutma Çevrimlerinin Enerji ve Ekserji Analizleri

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Öz: Karbon dioksit sıfır ODP ve ihmal edilebilir GWP'ye bağlı olarak artan ilgiye sahip olmuştur. Ayrıca, karbon dioksit ayrıca büyük özgül ısı, düşük viskozite ve büyük ısı iletkenliği gibi arzu edilen termodinamik özelliklere sahiptir. Karbondioksit CO_2, sırasıyla 7.36 MPa ve 31.1 °C olan düşük kritik basınç ve sıcaklığa sahiptir. Düşük kritik sıcaklık, kritik noktanın üzerinde gerçekleşmesi için kritik reddetme işleminin ve ısı emme işleminin gerçekleşmesine neden olur. Bununla birlikte, yüksek bastırma kaybına bağlı olarak, temel transkritik CO_2 döngüsünün enerji verimliliği, geleneksel düşük basınçlı soğutma döngüsünden daha düşüktür. Bu çalışmada, genleşme valfi (geleneksel transkritik çevrim olarak da adlandırılır) ile transkritik döngü olan üç farklı transkritik karbon dioksit döngüsü, genleştirici ile transkritik döngü ve ejektör ile transkritik döngü analiz edilmiştir. Çalışma parametrelerinin üç çevrimin maksimum performansı ve ekserji verimliliği üzerindeki etkisi araştırılmıştır. Sonuçlar, genleşme vanasının bir genişletici veya bir ejektör ile değiştirilmesinin sadece maksimum COP'yi ve eksi verimi arttırmayı değil, aynı zamanda optimal ısı reddetme basıncını da düşürdüğünü ortaya koymaktadır.

Anahtar kelimeler: Transkritik, CO₂ Soğutma Çevrimi, Genleşme, COP, Ekserji

1. Introduction

Because of the ozone depletion caused by chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) followed by the global warming impact of hydrofluorocarbons (HFCs), the use of natural refrigerants has attracted worldwide attention. As a natural refrigerant, carbon dioxide has an ODP of zero and negligible GWP therefore this working fluid has become an important alternative [1]. Many researchers reported that the use of carbon dioxide as a refrigerant gives the impression to be the most promising alternative [2-7]. The low critical temperature of 31.1 °C of carbon dioxide and the surrounding environment temperature, which is typically higher than carbon dioxide critical temperature, were the main reasons that have led to the development of transcritical carbon dioxide cycle in which the process of evaporation remains subcritical and the process of heat rejection is not any more condensation through condenser, but supercritical vapor cooling with another kind of heat exchangers which is called gas cooler. Lorentzen and Pettersen [8] through a pioneering work have proved that the problem of low critical temperature of carbon dioxide can be effectively overcome by rejection the heat in the supercritical region. Many researchers [3-7] reviewed the recent development for transcritical CO_2 cycle technology in refrigeration, heat pump, automobile air-conditioning, and residential air-conditioning systems applications in details. The transcritical cycle gives low performance compared to the traditional vapor compression cycle (subcritical cycle).

One of the improvements to the vapor compression cycle is the use of expander instead of expansion valve. Based on the first and second laws of thermodynamics, Yang et al. [8] performed a comparative study for the transcritical carbon dioxide refrigeration cycles with an expansion valve and with an expander. It is found that the COP and exergy efficiency of the expander cycle are on average 33% and 30% higher than those of the expansion valve cycle respectively. Zhang et al [9] presented the state of the art for recent literature concerning various two-phase expanders in transcritical CO_2 refrigeration cycles and reviewed the integration concepts for the expander in transcritical CO_2 refrigeration cycle. In a recent comparative study, Shariatzadeh et al [10] found that the use of expander has a privilege over that of expansion valve from energy point of view.

Employing an ejector as expansion device is also one of the alternative ways of improving transcritical cycle performance. Nowadays, ejector has been received more attention due to its noticeable benefits, such as no moving parts, low cost and low maintenance requirements. As a result, a lot of theoretical and experimental studies about ejector transcritical CO_2 refrigeration cycles have been performed. The research results prove that use of ejector as an expansion device in transcritical CO_2 cycle is considered a promising cycle modification to increase the system performance [11-17]

Based on the published literatures mentioned previously, it can be see that the two dominant modifications of the single stage leading to enhance the performance of the CO_2 transcritical cycle were the use of expander or ejector as an expansion device instead of expansion valve.

The aim of this paper is to carry out a comparative theoretical study among three different transcritical carbon dioxide cycles that are the conventional transcritical cycle (called also the base cycle, which is the transcritical cycle with expansion valve), the transcritical cycle with expander and the transcritical cycle with ejector.

2.   Description of Transcritical Carbon Dioxide Configurations Materials

A typical transcritical carbon dioxide refrigeration cycle consists of a compressor, a gas cooler, an evaporator and an expansion device, which is an expansion valve. The schematic of the transcritical cycle with expansion valve is shown in Figure 1.

The transcritical cycle operating principle is as follow: From suction line at state (1), the working fluid enters the compressor where it is elevated to high-pressure and high-temperature corresponding state (2). The high-pressure vapor leaving the compressor flows through the gas cooler rejecting heat to the environment, afterward the cooled supercritical CO_2 refrigerant at state (3) flows into the expansion valve, where it is throttled to low-pressure and low-temperature state (4). Finally after absorbing heat from the cold space in the evaporator, the refrigerant evaporates and re-enters the compressor at state (1).
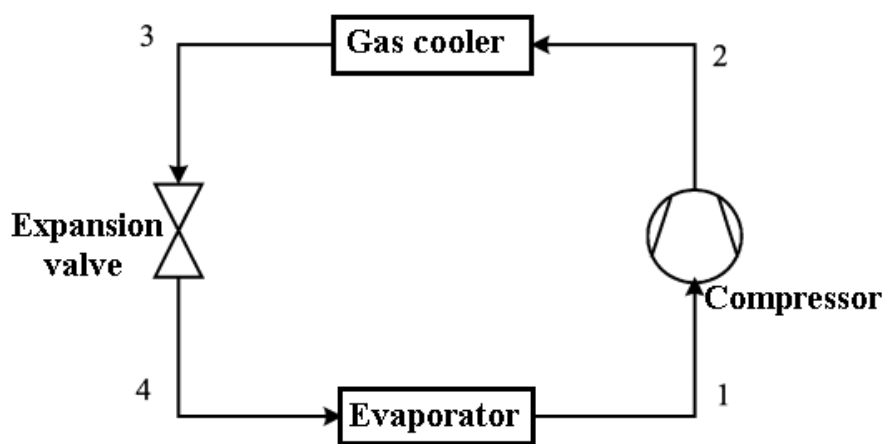

Figure1. Schematic of the transcritical cycle with expansion valve

The schematic of the transcritical cycle with expander is presented in Figure 2. In the transcritical refrigeration cycle with expander, the vapor from evaporator is compressed to supercritical pressure (1-2), then rejected heat to the surrounding environment via gas cooler and flows through expander to be expanded from supercritical pressure to subcritical.
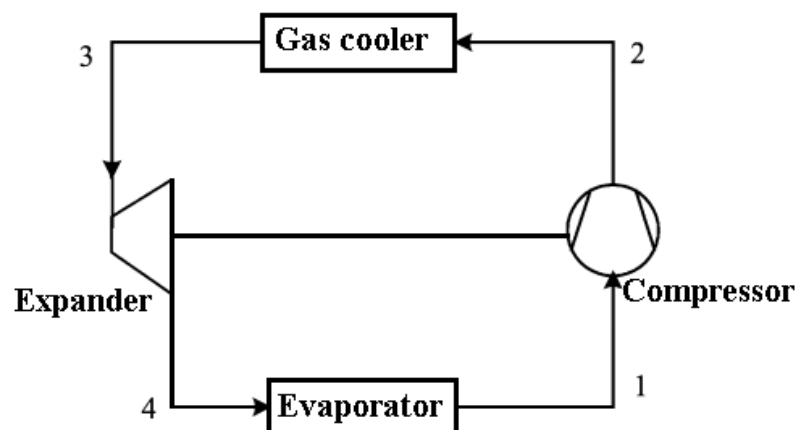

Figure 2. Schematic of the cycle with expander

The transcritical carbon dioxide cycle with ejector consists of a compressor, a gas cooler, an ejector, a separator, an expansion valve and an evaporator.

The schematic diagram of the transcritical system with ejector is shown in Figure 3. The ejector includes a motive nozzle, a suction chamber, a mixing section and a divergent diffuser.

3. Thermodynamic Analysis

In order to investigate these cycles the following assumptions are made:
-   There are no pressure losses in pipes and heat exchangers.
-   The flow across the expansion valve is isenthalpic.
-   The ejector entrainment ratio, U (m_8/m_3) is defined as the ratio of the ejector suction mass flow rate to the motive mass flow rate.
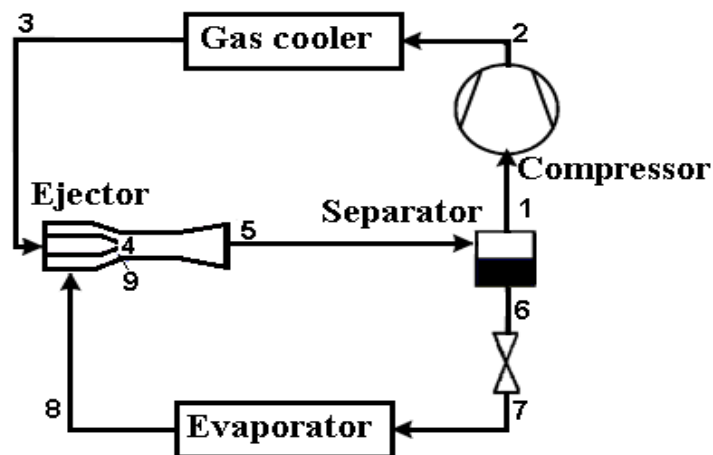

Figure 3. Schematic of the transcritical cycle with ejector

Energy analysis

The cycle performance is evaluated by its coefficient of performance COP, which is defined as the ratio of the cooling capacity (q_e) to the power consumption of compressors (W):

$$COP = \frac{q_e}{W} \tag{1}$$

Where the power consumption of the compressor "W" is calculated as:

For the conventional cycle and the cycle with ejector

$$W = W_c \tag{2}$$

For transcritical cycle with expander

$$W = W_c - W_{exp} \tag{3}$$

Where:
W_{exp} is the work generated by the expander

Exergy analysis

The refrigeration cycle includes various irreversible processes. Exergy or availability of a system at given state represents its maximum work potential. Therefore, the exergy loss provides a very important criterion to evaluate the thermodynamic performance of a system [12].

Exergy analysis is usually aimed to determine the maximum performance of the system and identify the locations of exergy destruction and to show the direction for potential improvements [18].

The total exergy destruction rate of the cycle is the sum of the exergy destruction rate in each component:

$$I_t = \sum I_{components}$$ (4)

The exergy efficiency for the refrigeration cycle can be defined as the ratio of the minimum work requirement to the actual work input [18].

$$\eta_{ex} = \frac{W_{rev}}{W} = \frac{W_{rev}}{W_{rev} + W_l} = 1 - \frac{W_l}{W}$$ (5)

Where W_{rev} and W are the work inputs to a reversible and an actual refrigeration cycle, respectively, and W_l is the lost work or the total exergy loss.

An exergy analysis has been performed for each individual component of the three cycles.

The three cycles are simulated under the following operating conditions:
- The isotropic efficiency of the compressor and the expander is assumed to be 75% and 65 %, respectively.
- The expansion efficiency of the motive and suction streams is taken to be 70%
- The efficiency of the ejector diffuser is equal to 80%
- The evaporating temperature:-20-10°C
- The outlet temperature of the gas cooler: 35 to 50°C
- The environment temperature: $T_0 = 30°C$

4. Results and Discussion

Effect of heat rejection pressure

Figure 4 shows the variation of performance coefficient (COP) and the exergy efficiency for the three cycles versus heat rejection pressure at the evaporating and gas cooler outlet temperatures of 0°C and 40°C, respectively.

It is clear that whatever the kind of the transcritical cycle considred, it exists an optimum pressure so that it is recommended that the transcritical CO_2 cycles should operate at their optimal heat rejection pressure in order to keep their maximum COP.

Further, it can be noticed that the transcritical cycle with expander outperforms the other transcritical cycles in terms of the COP and exergy efficiency. It gives at the optimum heat rejection pressure a values of 2.77 and 0.25 for the COP and the exergy efficiency respectively. Followed by the transcritical cycles with ejector and with expansion valve that provides at the optimum pressure a values of 2.50 and 0.22 of COP and 2.10 and 0.19 of exergy efficiency respectively.

In addition, the transcritical cycle with expander shows the lowest optimum pressure (97 bar) whereas the optimum pressure of the ejector cycle and expansion valve cycle is 99 and 102 bar respectively.

In Figure 6, the exergy efficiency of the three presented cycles is plotted as a function of gas cooler outlet temperature at optimum pressure and a given evaporating temperature (0°C).

Figure 6 shows clearly a downward trend of exergy efficiency with the rise of the gas cooler outlet temperature for all the considered cycles. It should be also noted that, for the studied range of the gas cooler outlet temperature. The cycle with expander has the highest exergy efficiency compared to the other cycles whereas the transcritical cycle with ejector outperforms the transcritical cycle with expansion valve in term of exergy efficiency.

Effect of gas cooler outlet temperature

The maximum COP and the corresponding optimum pressure of the three cycles at different outlet temperature of the gas cooler under the same evaporating temperature of 0°C are shown in Figure 5. The COP drops rapidly with the increase of the gas cooler outlet temperature. This is because, with the increase of the outlet temperature of the gas cooler, the cooling capacity remains almost the same but the power consumption increases.

It is can be seen from Figure 5 that, the transcritical cycle with expander presents the best performance along the whole range of gas cooler outlet temperature followed by the transcitical cycle with ejector and the transcritical cycle with expansion valve that comes at the last order. The cycle with expander provides an average improvement of 35 % whereas the cycle with ejector gives an average improvement of 21 % compared to the transcritical cycle with expansion valve.
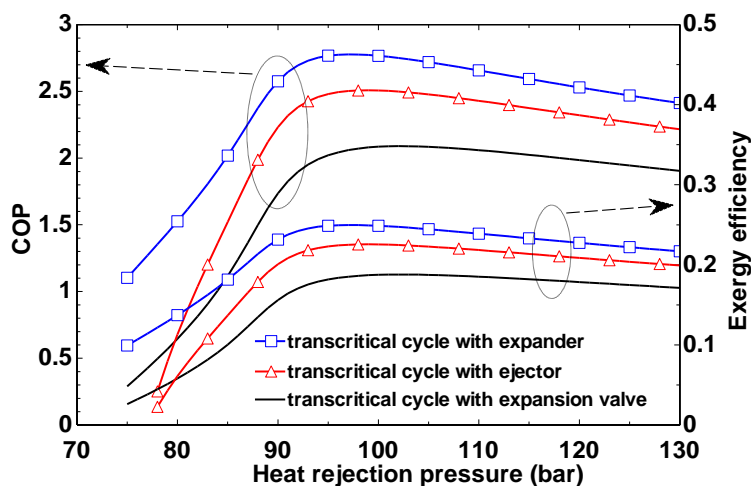

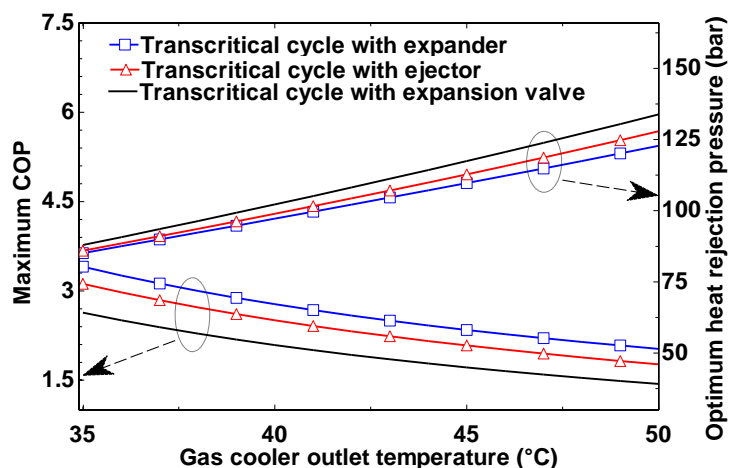Figure 4. Effect of heat rejection pressure on COP and exergy efficiency

Figure 5. Relationship between maximum COP and corresponding optimum heat
rejection pressure with gas cooler outlet temperature

Comparing the presented cycles in terms of optimum pressure, the descending order for the three cycles is the transcritical cycle with expansion valve, the transcritical cycle with ejector and the transcritical cycle with expander that gives on average optimum pressure of 110.1, 106.2 and 103.6 bar respectively

The exergy efficiency value of the cycle with expander is on average 35 % greater than that of the transcritical cycle with expansion valve while the cycle with ejector is on average of 21 % compared to the same cycle.

Effect of evaporating temperature

The effect of evaporating temperature on the maximum COP and optimum heat rejection pressure of the investigated cycles under the same gas cooler outlet temperature (40°C) is presented in Figure 7.

As the evaporating temperature increases, the maximum COP rises for all the considered configurations. This is due to the fact that when the evaporating temperature increases, the cooling capacity remains the same whereas the power consumption decreases and consequently the COP decreases. Additionally, when varying evaporating temperature it can be remarked that the maximum COP of transcritical cycles behaves similarly to the COP of conventional subcritical compression refrigeration cycles.
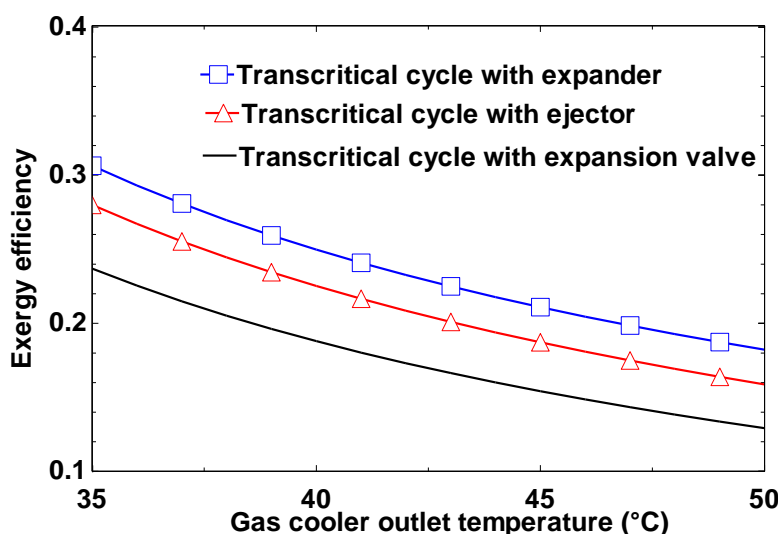
Figure 6. Exergy efficiency as a function of gas cooler outlet temperature

Compared to the transcritical cycle with expansion valve, the maximum COP value of the transcritical cycle with expander shows an average of 34 % performance improvement while the transcritical cycle with ejector gives an average of 21 %.

Figure 8 shows the exergy efficiency behavior of the studied cycles versus evaporating temperature variation at optimum discharge pressure and a constant gas cooler temperature (40°C). It is clear that, as the evaporating temperature ascends, the exergy efficiency of the different studied cycles falls gradually. From the same Figure 8, it is clear that the transcritical cycle with expander shows the highest exergy efficiency among the considered cycles.

From the Figure 8, the optimum pressure lines show a slight downward trend with growing along of the evaporating temperature. Further, the evaporating temperature has relatively a minor influence as compared with the gas cooler outlet temperature discussed in the previous subsection. Further, Compared to the transcritical cycle with expansion valve, it is found that employing an expander instead of an expansion valve enhances the exergy efficiency on average of 34%, whereas substituting the expansion valve by ejector improves the exergy efficiency on average by 21%.
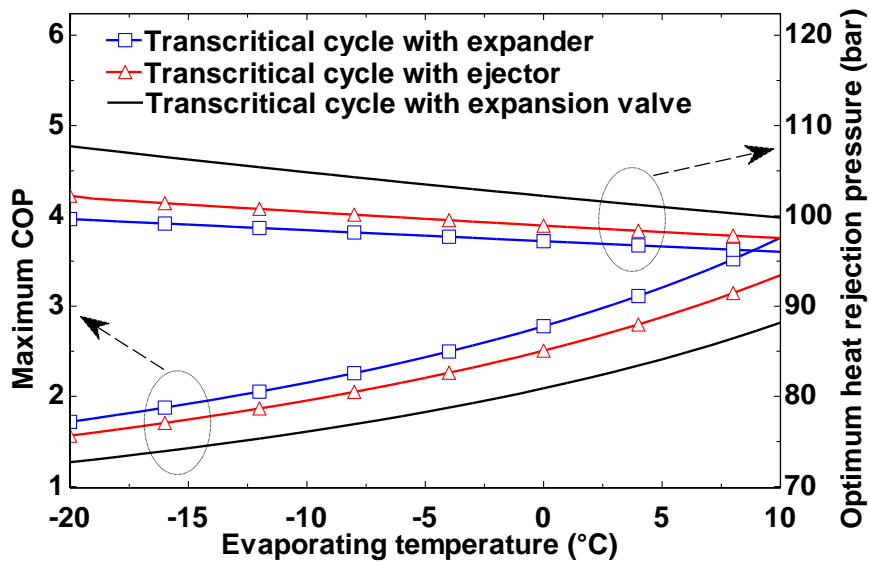

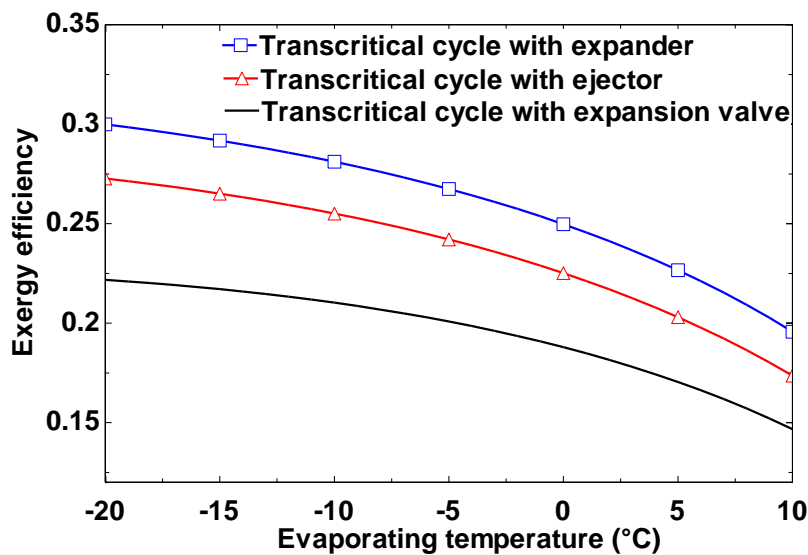Figure 7. Maximum COP and optimum heat rejection pressure at different evaporating temperatures

Figure 8. Exergy efficiency versus evaporating temperature

5. Conclusions

In this paper a detailed comparative study of three transcritical CO_2 cycles is presented, these cycles are transcritical cycle with expansion valve, transcritical cycle with ejector and transcritical cycle with expander. The expander cycle improves both the maximum COP and exergy efficiency by an average of 35%, whereas the ejector cycle also enhances the maximum COP as well as the exergy efficiency by an average of 21%. Further, replacing the expansion valve by an expander or an

ejector does not only improve the maximum COP and the exregy efficiency but also reduces the optimal heat rejection pressure.

References

[1]   Bolaji, B.O., Huan, Z., "Ozone depletion and global warming: Case for the use of natural refrigerant – a review", Renewable and Sustainable Energy Reviews, 2013, 18:49-54.

[2]   Lorentzen, G., "The use of natural refrigerant: a complete solution to the CFC/HCFC predicament", International Journal of Refrigeration, 1994, 18:190-197.

[3]   Kim, M.H., Pettersen, J., Bullard, C.W., "Fundamental process and system design issues in CO_2 vapor compression systems", Vol. 30, No.2, Progress in Energy and Combustion Science, 119-174:2004.

[4]   Austin, B.T., Sumathy, K., "Transcritical carbon dioxide heat pump systems: A review", Renewable and Sustainable Energy Reviews, 2011, 15:4013-4029.

[5]   Bansal, P. , "A review Status of CO_2 as a low temperature refrigerant: Fundamen tals and R&D opportunities", Applied Thermal Engineering, 2012, 41:18-29.

[6]   Ma, Y., Liu, Z., Tian, H.A., "Review of transcritical carbon dioxide heat pump and refrigeration cycles", Energy, 2013, 55:156-172.

[7]   Zhang, J.F., Qin, Y., Wang, C.C, "Review on CO_2 heat pump water heater for residential use in Japan", Renewable and Sustainable Energy Reviews, 2015, 50: 1383-1391.

[8]   Yang, J.L., Ma, Y.T., Li, M.X, Guan, h. Q. ,"Exergy analysis of transcritical carbon dioxide refrigeration cycle with an expander", Energy, 2005, 30:1162-1175.

[9]   Zhang, Z., Ma, Y. , Li, M. , Li, Z., "Recent advances of energy recovery expanders in the transcritical CO_2 refrigeration cycle", HVAC&R Research, 2013, 19:376-84.

[10]   Shariatzadeh, O. J., Abolhassani, S., Rahmani, M., Nejad, M. Z.,"Comparison of transcritical CO_2 refrigeration cycle with expander and throttling valve including/excluding internal heat exchanger: Exergy and energy points of view", Applied Thermal Engineering, 2016, 93:779–787.

[11]   Li, D., Groll, E. A., "Transcritical CO_2 refrigeration cycle with ejector-expansion device", International Journal of Refrigeration, 2005, 28:766–773.

[12]   Deng, J. Q., Jiang, P.Q., Lu, T,. Lu, W.,"Particular characteristics of transcritical CO_2 refrigeration cycle with an ejector", Applied Thermal Engineering, 2007, 27:381-388.

[13]   Elbel, S.,"Historical and present developments of ejector refrigeration systems with emphasis on transcritical carbon dioxide air-conditioning applications", International Journal of Refrigeration, 2001, 34:1545-1561.

[14]   Sumeru, K., Nasution, H., Ani, F. N., "A review on two-phase ejector as an expansion device in vapor compression refrigeration cycle", Renewable and Sustainable Energy Reviews, 2012, 16:4927-4937.

[15]   Sarkar, J., "Ejector enhanced vapor compression refrigeration and heat pump systems-A review", Renewable and Sustainable Energy Reviews, 2012, 16:6647-6659.

[16]   Xiangjie, X., Omer, S., Worall, M , Riffat, S., "Recent developments in ejector refrigeration technologies", Renewable and Sustainable Energy Reviews, 2013, 19: 629-651.

[17]   Besagni, G., Mereu, R., Inzoli, F.,"Ejector refrigeration: A comprehensive review", Renewable and Sustainable Energy Reviews, 2016, 53:373-40.

[18]   Robinson, D. M., Groll E. A., "Efficiencies of transcritical CO_2 cycles with and without an expansion turbine", International Journal of Refrigeration, 1998, 21:577-589.