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Energy and Exergy Analysis of Transcritical Carbon Dioxide Refrigeration Cycle for Different Working Conditions

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Abstract: The effects of conventional refrigerants on ozone layer and its effect on global warming are known. Refrigerants based on chlorofluorocarbons (CFC) and hydrochlorofluorocarbons (HCFCs) have been used in refrigeration systems for a long time and are still being used. Instead of these refrigerants, the use of natural refrigerants which are much less harmful to the environment, has been given importance. Carbon dioxide is the first that comes to mind among these fluids. In this study, energy and exergy analysis of transcritical carbon dioxide refrigeration cycle examined as theoretically. Energy and exergy analysis were made for different evaporator temperatures and outlet temperatures from gas cooler. Exergy losses of each component of the transcritical carbon dioxide refrigeration cycle were determined. In addition, the first and second law efficiencies of the refrigeration cycle were determined. When the evaporator temperature is -5 ^{o}C and the outlet temperature from the gas cooler is 20 ^{o}C, the highest exergy efficiency value was determined as 41.19 in this study.

Keywords: carbon dioxide, exergy analysis, transcritical cycle, coefficient of performance

Transkritik Karbondioksit Soğutma Çevriminin Farklı Çalışma Şartları için Enerji ve Ekserji Analizi

Öz: Konvansiyonel soğutucu akışkanların ozon tabakası üzerindeki etkileri ve küresel ısınmaya etkisi bilinmektedir. Kloroflorokarbon (CFC) ve hidrokloroflorokarbon (HCFC) bazlı soğutucular, soğutma sistemlerinde uzun süredir kullanılmaktadır. Bu soğutucu akışkanların yerine çevreye çok daha az zararlı olan doğal soğutucu akışkanların kullanımına önem verilmiştir. Bu akışkanlar arasında ilk akla gelen karbondioksittir. Bu çalışmada, transkritik karbondioksit soğutma çevriminin enerji ve ekserji analizi teorik olarak incelenmiştir. Evaporatör ve gaz soğutucu çıkış sıcaklıkları için enerji ve ekserji analizleri yapılmıştır. Transkritik karbondioksit soğutma çevriminin her bir bileşeninin ekserji kayıpları belirlenmiştir. Ayrıca soğutma çevriminin birinci ve ikinci kanun verimleri belirlenmiştir. Bu çalışmada en yüksek ekserji verim değeri, evaporatör sıcaklığı -5 ^{o}C ve gaz soğutucu çıkış sıcaklığı 20 ^{o}C için 41.19 olarak belirlenmiştir.

Anahtar Kelimeler: karbondioksit, ekserji analizi, transkritik çevrim, soğutma performans katsayısı

1. Introduction

The history of refrigeration goes back to very ancient times when stored ice is used, water is evaporated and nutrients are stored. In the 1830s, Perkins invented the first vapor compression cooling system and conventional refrigerants are used in these systems. Today, different types of cooling systems are used. Increased use of refrigeration systems has also increased the use of conventional refrigerants. The global warming and greenhouse effect of conventional refrigerants

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required the use of alternative refrigerants due to international agreements. International agreements such as Montreal, Copenhagen Protocols and the Kyoto Protocol are prohibited with limited use of conventional refrigerants. Today, instead of CFC (Chlorofluorocarbon) and HCFC (Hydrochlorofluorocarbon) refrigerants, alternative refrigerants which are less harmful to the environment are used. Low global warming potential and ozone depletion potential are the most important characteristics desired in alternative refrigerants. Due to these limitations, environment-friendly carbon dioxide refrigerant has been widely used in recent years. Schematic diagram of transcritical CO_2 refrigeration cycle is given Fig. 1. P-h diagram of transcritical carbon dioxide refrigeration cycle is given Fig. 2.

Gomri et al. examined three different types of transcritical CO_2 cycles. Their study was determineted that the effect of working parameters on the peak performance and exergy efficiency of the three cooling cycles. It was found in their study that an expander or an ejector used in place of the expansion valve improves both COP and exergy efficiency [1]. Yang et al. have made comparative performance analysis of transcritical CO_2 refrigeration cycles. They were examined that the effects of evaporator temperature and outlet temperature of gas cooler on COP, exergy losses and the exergy efficiencies. It was found that a large part of the irreversibility occurred in the gas cooler and compressor [2]. Fangtian and Yitai have made comparative work on transcritical CO_2 cooling cycle with ejector. They have found that ejector instead of expansion valve can reduce more 25% exergy loss and increase COP more 30% [3]. Bai et al. have made exergy analysis of ejector expansion transcritical CO_2 cooling system. They have found that 43.4% of total exergy damage of the cooling system could be prevented by the development of system components [4]. Papadaki and Stegou-Sagia have examined that exergy analysis of exergy use of CO_2 in two stage and single stage heat pumps. They compared results of their study with other result of refrigeration cycles and they developed a mathematical model for these cycles [5]. Purjam et al. have made thermodynamic analysis of the transcritical CO_2 with ejector cycle. COP is 1.4 for evaporator temperature 45 ◦C in their study. They determined that this presented cycle can not reach a reasonable performance for deep-freezing applications [6]. Elbarghthi et al. were carried out exergy analysis of the transcritical CO_2 cooling system with ejector. The ejector was delivered maximum exergy efficiency of 23% in cooling system. The results of their work show to achieving better performance when the ejector worked at transcritical conditions [7].

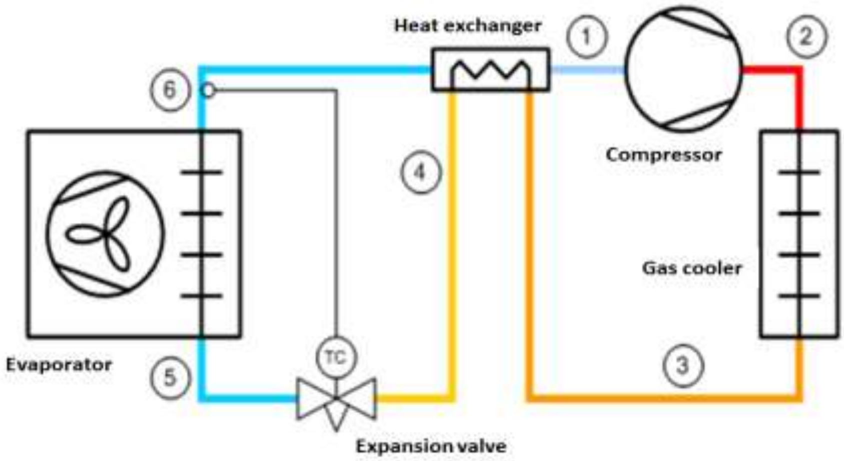

Figure 1. Schematic diagram of transcritical CO_2 refrigeration cycle.

Gullo has investigated exergy analysis of transcritical CO_2 supermarket cooling systems. The result of the study showed that the use of more efficient compressors can greatly reduce the overall irreversibility [8]. Song et al. have made energy and exergy analyses of a transcritical CO_2 air conditioning system for an electric bus. Results showed that the cycle possessed a good refrigeration performance in hot weather and increasing the indoor air flow rate could improve the exergy efficiency of the cycle [9]. Kumar et al. have examined hybrid transcritical CO_2 vapor compression and vapor ejector cooling cycle. The results of their work show that R32 is most suitable for the vapor ejector cooling cycle amongst the five fluids analyzed [10].

In this study as unlike the literature studies, the exergy efficiency, COP values of the refrigeration system and the exergy loss for every component of the transcritical CO_2 refrigeration cycle were determined for the different evaporator temperatures and outlet temperature from the gas cooler. COP, exergy efficiency and exergy losses of the transcritical CO_2 refrigeration cycle have been determined and compared for different working conditions.

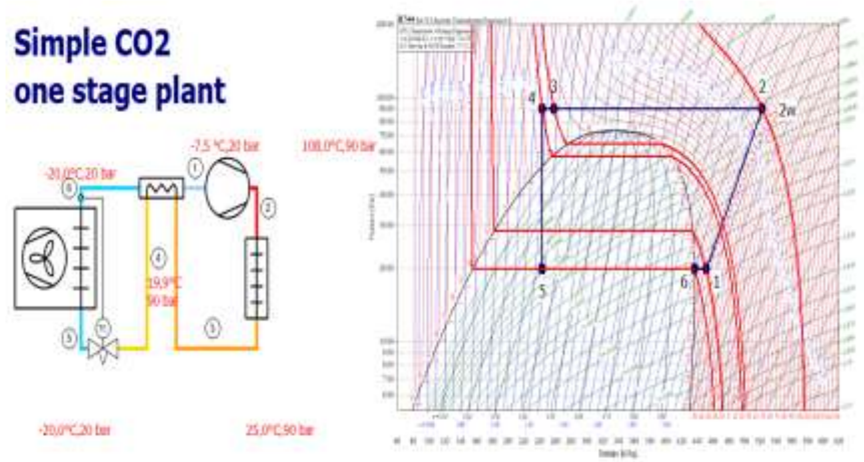

Figure 2. P-h diagram of transcritical CO_2 refrigeration cycle.

2. Thermodynamics Analysis

When the first law of thermodynamics is applied to the transcritical CO_2 cycle, the energy balance is written as follows;

$$\dot{Q}_e + W = \dot{Q}_g \quad (1)$$

where e and g refers to the evaporator and gas cooler, respectively. The COP of the refrigeration system is obtained by dividing the evaporator heat load by the compressor net work.

$$COP = \frac{\dot{Q}_e}{W} \quad (2)$$

According to the principle of conservation of energy, work and heat transfer in each component in a fixed open system can be written as follows;

$$\dot{Q}_e - W = \Delta H \quad (3)$$

The following assumptions were made during exergy analysis of the system.

- All system is working at a steady-state.
- The kinetic energy and potential energies of the system components are negligible.
- The pressure losses through the connecting pipes are negligible.
- Isentropic efficiency of compressor is %90.
- Reference temperature of 298.15 K

Exergy analysis is performed to convert the possibility of obtaining useful work from the energy of a system to the highest level of application. Most basic exergy expression;

$$\psi = (h - h_0) - T_0(s - s_0) \tag{4}$$

where h_0 and s_0 refers to values of enthalpy and entropy at the environmental temperature T_0.

<u>Compressor;</u>

Compressor workload is [11];

$$E_{in} = \dot{W} = \dot{m}(h_2 - h_1) \tag{5}$$

The exergy recovered in the compressor is;

$$\psi_C = \dot{m}(\psi_1 - \psi_2) \tag{6}$$

The exergy loss of the compressor is;

$$\Delta\psi_C = \psi_1 + \dot{W} - \psi_2 = E_{in} - \psi_C = \dot{m}(T_0(s_2 - s_1)) \tag{7}$$

<u>Evaporator;</u>

The exergy afforded of the evaporator is [12];

$$\psi_S = \dot{Q}_E\left(1 - \frac{T_0}{T_r}\right) \tag{8}$$

The exergy recovered in the evaporator is;

$$\psi_R = \dot{m}(\psi_7 - \psi_6) \tag{10}$$

The exergy loss of the evaporator is;

$$\Delta\psi_{ev} = \psi_7 + \dot{Q}_E\left(1 - \frac{T_0}{T_r}\right) - \psi_6 = \dot{m}(h_7 - T_0 s_7) + \dot{Q}_E\left(1 - \frac{T_0}{T_r}\right) - \dot{m}(h_6 - T_0 s_6) \tag{11}$$

Where $\dot{Q}_E\left(1 - \frac{T_0}{T_r}\right)$ is thermal exergy loss rate, $\dot{Q}_E$ is heat load in evaporator, T_r is temperature of the component's boundary. Expansion valve;

$$\Delta\psi_V = \dot{m}(\psi_6 - \psi_5) = \dot{m}(T_0(s_5 - s_6)) \tag{12}$$

<u>Gas cooler;</u>

The exergy afforded of the gas cooler is [13];

$$\Delta\psi_S = \dot{m}(\psi_2 - \psi_4) \tag{13}$$

The exergy recovered in the gas cooler is;

$$\psi_R = \dot{Q}\left(1 - \frac{T_0}{T_{GC}}\right) \tag{14}$$

The exergy loss of the gas cooler is;

$$\Delta\psi_{GC} = \psi_2 + \dot{Q}_{ğ}\left(1 - \frac{T_0}{T_{gc}}\right) - \psi_4 = \dot{m}(h_2 - T_0 s_2) + \dot{Q}_g\left(1 - \frac{T_0}{T_{gc}}\right) - \dot{m}(h_4 - T_0 s_4) \tag{15}$$

In heat transfer processes where temperatures are variable, entropic average temperature is used to facilitate analysis and is expressed as follows;

$$T_{gc} = \frac{h_2 - h_4}{s_2 - s_4} \tag{16}$$

Internal heat exchanger (SGHX);

The exergy afforded of the internal heat exchanger is [14];

$$\Delta\psi_S = \dot{m}(\psi_4 - \psi_5) \tag{17}$$

The exergy recovered in the internal heat exchanger is;

$$\Delta\psi_R = \dot{m}(\psi_1 - \psi_7) \tag{18}$$

The exergy loss of the internal heat exchanger is [15];

$$\Delta\psi_{HEX} = \dot{m}((\psi_4 - \psi_5) - (\psi_1 - \psi_7) = \dot{m}(\big(h_4 - h_5 + T_0(s_5 - s_4)\big) - \big((h_1 - h_7) + T_0(s_7 - s_1)\big)) \tag{19}$$

The analysis of irreversibility in all components of the transcritical refrigeration cycle is completed with the entropy balance method.

$$\sum \delta S_i = \frac{Q_{gc}}{T_{agc}} - \frac{Q_e}{T_{ae}} \tag{20}$$

where δS_i is the irreversible entropy increase in the transcritical refrigeration cycle, T_{agc} and T_{ae} are the average temperatures of the environments of gas cooler and evaporator, respectively, Q_{gc} and Q_e are the heat transfer rate of the gas cooler and evaporator. Irreversibility can be determined for each component [16];

Gas cooler, $\delta S_{gc} = \frac{Q_{gc}}{T_{agc}} - \frac{Q_{gc}}{T_{gc}}$ (21)

Evaporator, $\delta S_e = \frac{Q_e}{T_e} - \frac{Q_e}{T_{ae}}$ (22)

Compressor, $\delta S_c = s_2 - s_1$ (23)

Expansion valve, $\delta S_{ev} = s_6 - s_5$ (24)

Internal heat exchanger, $\delta S_{sghx} = (s_5 + s_1) - (s_4 + s_7)$ (25)

3. Results and Discussion

Each system component in the transcritical CO_2 cycle of exergy loss have been found depending on the outlet temperature from gas cooler change and it is given in Fig. 3. When the evaporator temperatures are kept constant (T_E=-5, -10, -15, -20, -25 and -30 ^oC, respectively) and the outlet temperature from gas cooler are changed in the transcritical CO_2 cycle, the highest exergy loss has occured in the evaporator, gas cooler, expansion valve, compressor and heat exchanger, respectively. It is seen in Fig. 3 that when the outlet temperature from the gas cooler increases, the exergy loss of the gas cooler, heat exchanger and compressor remains approximately the same. However, the exergy loss in the evaporator decreased and the exergy loss of the expansion valve increased. When the evaporator temperature is -30 ^oC and the outlet temperature from the gas cooler is 45 ^{o}C, the highest exergy loss was determined as 50 kJ kg^{-1} in the expansion valve for the operating conditions presented in Fig. 3.

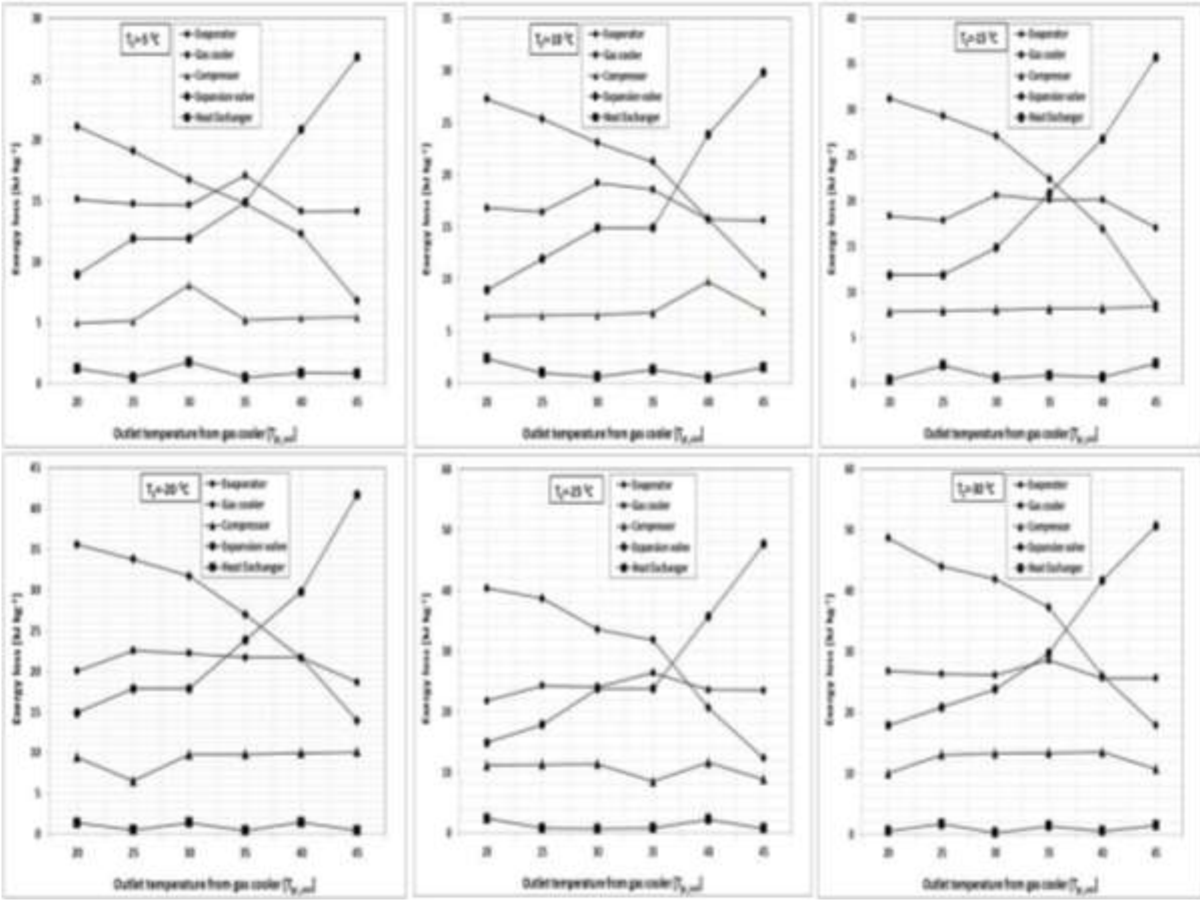

Figure 3. Variation of exergy loss with outlet temperature from gas cooler.

Each system component in the transcritical CO_2 cycle of exergy loss have been found depending on the evaporator temperature change and it is given in Fig. 4. When the outlet temperature from gas cooler are kept constant (T_{GC_out}=20, 25, 30, 35, 40 and 45 ^{o}C, respectively) and the evaporator temperature are changed in the transcritical CO_2 cycle, the highest exergy loss has occured in the evaporator (for T_{GC_out}=20, 25, 30, 35 ^{o}C), and gas cooler (for T_{GC_out}=40 ^{o}C and 45 ^{o}C). It is seen in Fig. 4 that when the evaporator temperature increases, the exergy loss of each system component in the transcritical CO_2 cycle decreases. However, the exergy loss of the heat exchanger remains approximately the same. When the evaporator temperature is -30 ^{o}C and the outlet temperature from the gas cooler is 45 ^{o}C, the highest exergy loss was determined as 50 kJ kg^{-1} in the expansion valve for the operating conditions presented in Fig. 4.

Transcritical CO_2 refrigeration cycle of total exergy loss and COP have been found depending on the outlet temperature from gas cooler change and it is given in Fig. 5. The evaporator temperatures are kept constant (T_E=-5, -10, -15, -20, -25 and -30 ^{o}C, respectively) and the outlet temperature from gas cooler are changed in the transcritical CO_2 refrigeration cycle. It is seen in Fig. 5 that when the outlet temperature from the gas cooler increases, COP values decrease. Moreover, while COP values decreases, exergy loss values increases. Therefore, it can be said that COP values and total exergy loss values of the system are inversely proportional to each other. When the evaporator temperature is -5 ^{o}C and the outlet temperature from the gas cooler is 20 ^{o}C, the highest COP value was determined as 3.8 kJ kg^{-1} in this study.

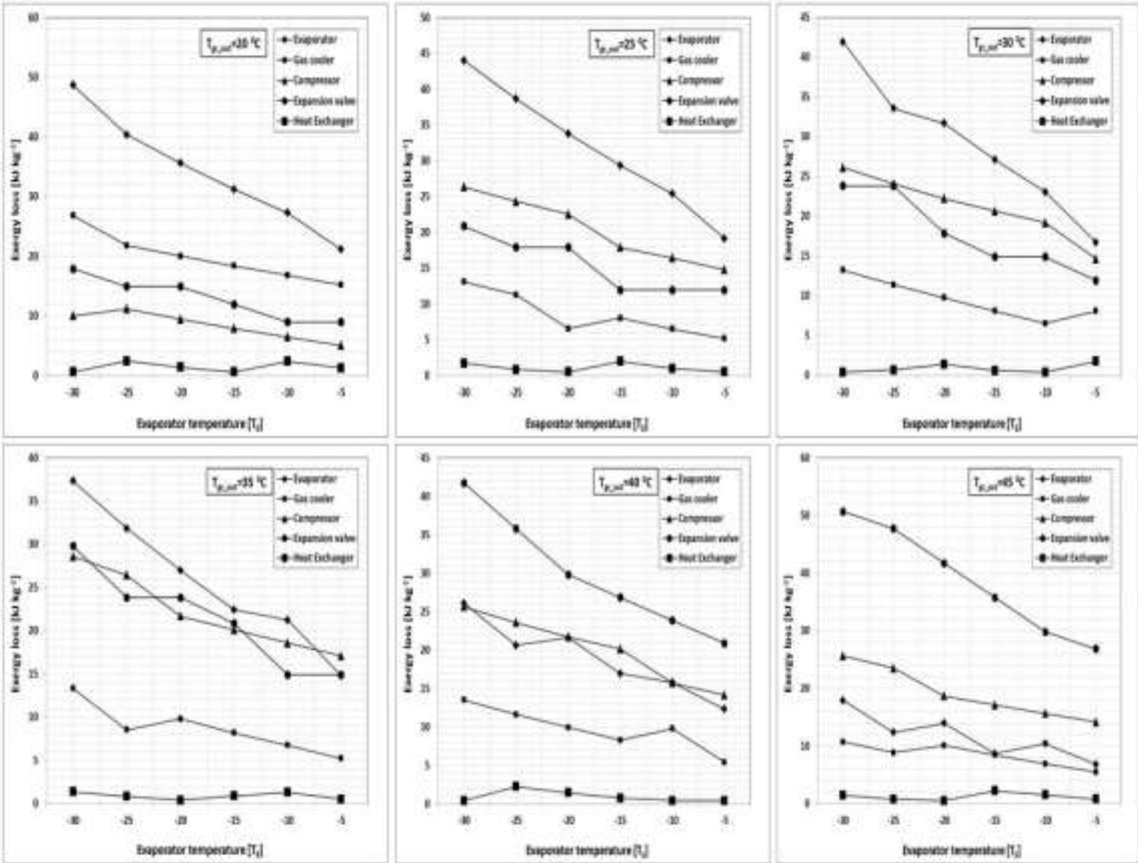

Figure 4. Variation of exergy loss with evaporator temperature.

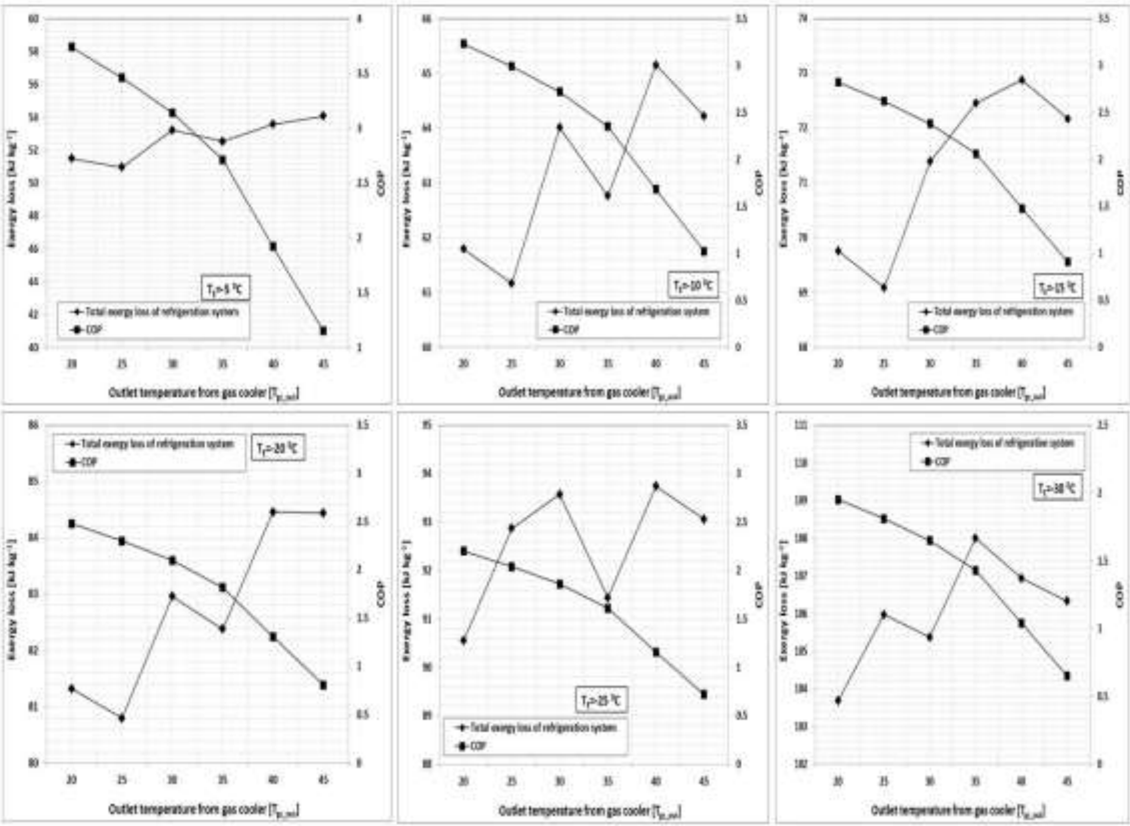

Figure 5. Variation of total exergy loss and COP of system with outlet temperature from gas cooler.

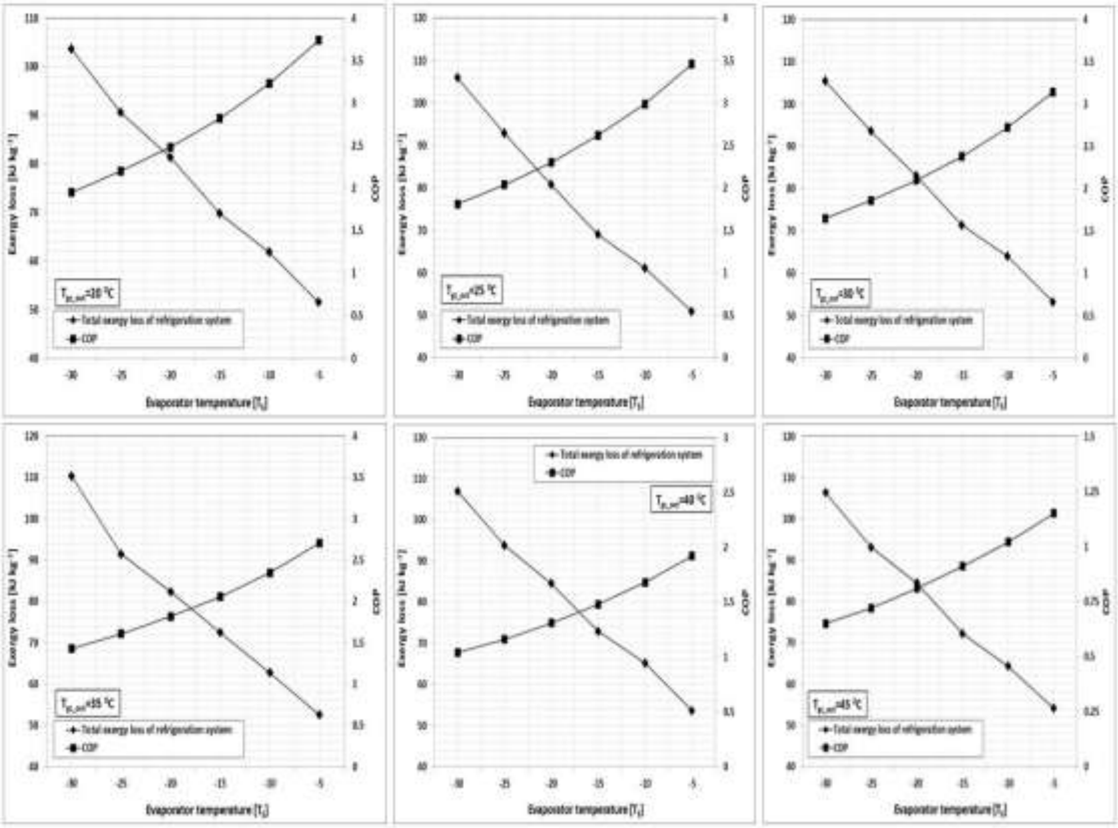

Figure 6. Variation of total exergy loss and COP of system with evaporator temperature.

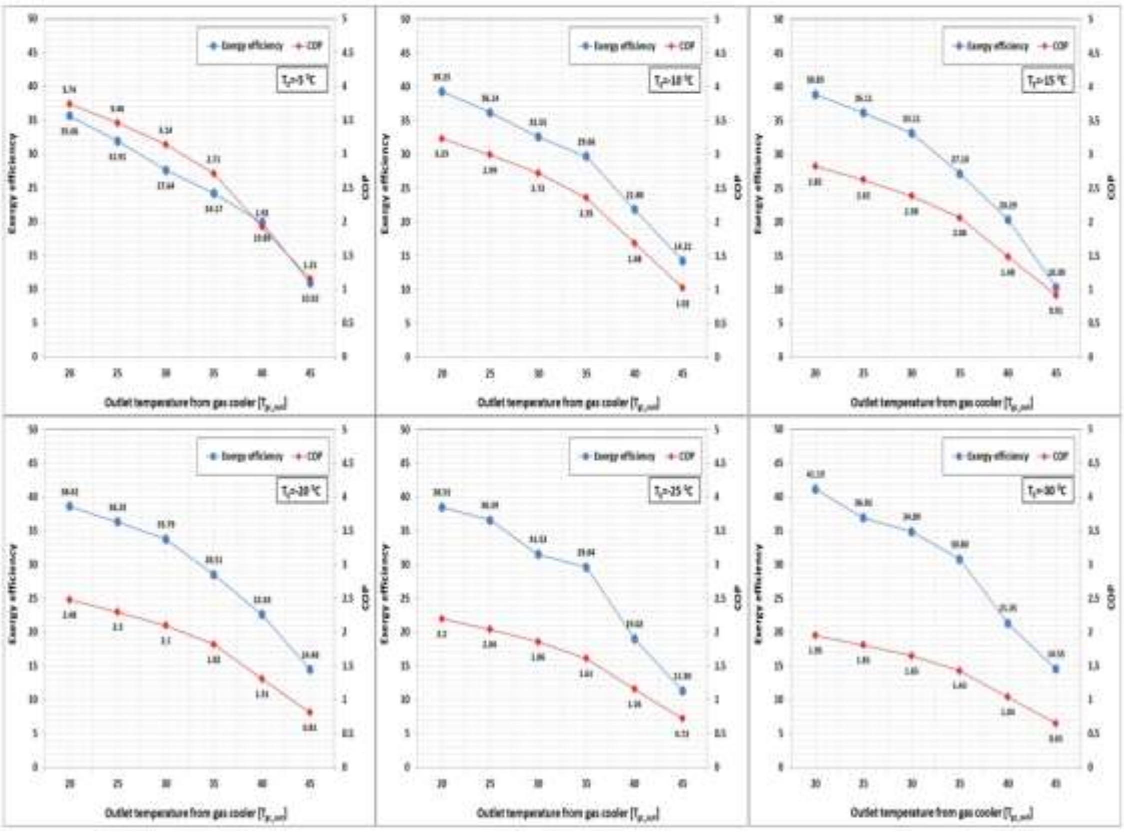

Figure 7. Variation of exergy efficiency and COP with outlet temperature from gas cooler.

Transcritical CO_2 refrigeration cycle of total exergy loss and COP have been found depending on the evaporator temperature change and it is given in Fig. 6. The outlet temperature from gas cooler are kept constant (T_{GC_out}=20, 25, 30, 35, 40 and 45 ^oC, respectively) and the evaporator temperature are changed in the transcritical CO_2 refrigeration cycle. It is seen in Fig. 6 that when the evaporator temperature increases, COP values increase, too. Moreover, while COP values increases, exergy loss values decreases. When the evaporator temperature is -5 ^oC and the outlet temperature from the gas cooler is 20 ^oC, the highest COP value was determined as 3.8 kJ kg^{-1} in this study.

It is given in Fig. 7 that transcritical CO_2 refrigeration cycle of exergy efficiency and COP depending on the outlet temperature from gas cooler change. The outlet temperature from gas cooler are kept constant and the evaporator temperature are changed in the transcritical CO_2 refrigeration cycle. The exergy efficiency of the transcritical CO_2 refrigeration cycle was calculated for these operating conditions. It is seen in Fig. 7 that when the outlet temperature from gas cooler increases, both exergy efficiency and COP values decrease. When the evaporator temperature is -5 ^oC and the outlet temperature from the gas cooler is 20 ^oC, the highest exergy efficiency value was determined as 41.19 in this study. However, the COP value in these working conditions is 1.95. This is a low value for the COP value. It is seen that the most suitable value for both exergy efficiency value and COP value is -10 ^oC evaporator temperature and 20 ^oC outlet temperature from gas cooler when Fig. 7 is examined. At this point, the COP value is 3.23, while the exergy efficiency value is 39.25.

4. Conclusions

In this study, the exergy efficiency and COP values of the transcritical CO_2 refrigeration and the exergy loss for every component of the refrigeration cycle were determined for the different evaporator temperatures and outlet temperature from the gas cooler. In the transcritical CO_2 refrigeration cycle, the evaporator temperature and the outlet temperature from the gas cooler have different effects on COP, exergy losses and exergy efficiency. Therefore, it is very important to determine the optimal evaporator and outlet temperature from the gas cooler for the refrigeration system to operate at high performance. In this study, the highest exergy losses occurred in the evaporator. Therefore, reducing exergy losses in the evaporator is crucial to improving the performance of the transcritical CO_2 cycle. Reducing exergy losses of the each component of the refrigeration system will improve the performance of the cooling system. Thus, efficient use of energy will be ensured.

Authors' contributions

BK designed the structure of article, carried out the theoretical calculations and wrote up the article. Author read and approved the final manuscript.

Competing Interests

The author declare that they have no competing interests.

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