Energy and Exergy analysis of Carbon Dioxide Transcritical Booster System

Analyse énergétique et exergétique d'un système à dioxyde de carbone d'amplification transcritique

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ABSTRACT. In this paper, energy and exergy analysis of CO2–booster two-stage chiller combining transcritical cycle in cascade with a subcritical are carried out. The examined system produce refrigeration at two temperatures levels and hot water at 73°C.The necessary thermodynamic parameters for analysis are calculated using the software Engineering Equation Solver. A parametric study is conducted to investigate the effect of MT-evaporating and heat rejection temperatures on the coefficient of performance, the total lost work and the exergy lost in different components in the CO2 cycle. The largest lost was recorded in the MT compressor; followed by the high-pressure expansion valve, the Gas cooler, the intermediate pressure valve, LTV and MTV expansion valves, the compressor LT and IP Receiver .

RÉSUMÉ. Dans cet article, l'analyse énergétique et exergétique d'un refroidisseur à deux étages CO2-amplificateur combinant un cycle transcritique en cascade avec un cycle sous-critique est réalisée. Le système examiné produit de la réfrigération à deux niveaux de température et de l'eau chaude à 73°C. Les paramètres thermodynamiques nécessaires à l'analyse sont calculés à l'aide du logiciel Engineering Equation Solver. Une étude paramétrique est menée pour étudier l'effet des températures d'évaporation de la magnétoscopie et de rejet de chaleur sur le coefficient de performance, le travail total perdu et l'exergie perdue dans les différents composants du cycle du CO2. La perte la plus importante a été enregistrée dans le compresseur MT; suivi du détendeur haute pression, du refroidisseur de gaz, de la soupape de pression intermédiaire, des détendeurs LTV et MTV, du compresseur LT et du récepteur IP.

KEYWORDS. Energy, Exergy, CO2–booster, Coefficient of performance, Parametric study.

MOTS-CLÉS. Énergie, Exergie, CO2-Amplificateur, Coefficient de performance, Etude paramétrique.

1. Introduction

Active research is being carried out nowadays to improve the coefficient of performance (COP) of vapor compression refrigeration systems. Carbon dioxide (CO2, R744) is increasingly becoming the working fluids for refrigeration systems of choice to replace environmentally harmful CFCs and HCFCs[1, 2]. R744 is ecofriendly with and. Further advantages of carbon dioxide are high latent heat of evaporation, non-flammability and non toxicity. CO2 refrigeration units are now used in commercial supermarkets all around the world. Research is performed to investigate the CO2 refrigeration systems in order to increase their performance for the reason that it is generally low when compared to other conventional refrigerants mainly because of the low-critical point of the CO2 (30.98°C). Many researchers have considered the optimization of the high pressure in the trans-critical mode in order to enhance the performance of CO2 refrigeration systems (Kauf [5] and Liao et al. [6]). An internal heat exchanger used for sub-cooling after the gas cooler as well as cascade systems have been considered [7-9]. Later, CO2 two-stage chiller combining transcritical cycle in cascade with a subcritical (conventional) have been examined and COP enhancement up to 25% have been found [10].Chesi et al.[11] examined the use of an auxiliary compressor for the transcritical cycle. They found up to 30% increases in the COP compared to the simple one stage compression unit. Despite the great interest in CO2 booster systems, their performances is predominantly assessed via energy based method However, the most appropriate measures aimed at enhanced the performance of any energy system can be brought to light by applying an exergy analysis. The exergy analysis describes the losses in the whole system and its components [12].

2. Analysis

2.1. *Description of the cycle*

Figure 1 shows schematically the CO2 booster refrigeration configuration considered. In this system, the refrigerant exiting the gas cooler is expanded with a high-pressure expansion valve (6-7) into a liquid receiver at an intermediate pressure level(Pint). The saturated liquidrefrigerant (10) from the flash gas separator is divided in two streams (11 and 14) and expanded in expansion valves MTV and LTV to provide cooling effect at two temperature levels in MT-evaporator and LT-evaporator, respectively. The gas flow (4) in the suction line of the MT compressor is composed of streams exiting three different unit components: Bypass flash gas out of the liquid receiver (8-9), discharge gas from the LT-stage compressor (2) and superheated gas from the MT-evaporator (13).The compressed gas from MT-compressor are mixed before entering the gas cooler for heat rejection to an external fluid(water at 25°C).

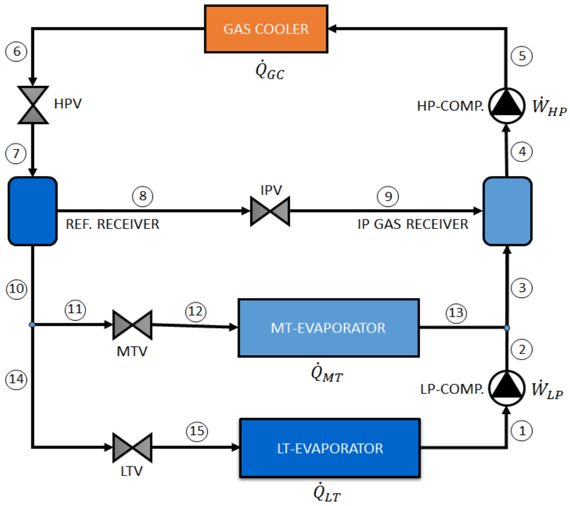

Figure 1. *Schematic diagram of the CO2 booster refrigeration system.*

2.2. *CO2 refrigerator model equations*

The formulation of the examined refrigeration system is given with all the proper details. Some assumptions for thermodynamic calculations are presented as below:

– Analysis under steady state conditions.

– Isenthalpic throttling.

– Adiabatic but non-isentropic compressions.

– Isobaric evaporation and gas cooling.

– Saturated vapor at inlet of compressors LT and MT.

– Saturated vapor exiting intermediate pressure receiver.

– the kinetic and potential energies are not taken into account

– The mass flow rate of refrigerant circulating in the first cycle is 1 kg/s.

– The reference temperature and pressure are 25°C and 1 bar

– Water temperature entering gas cooler: 25°C.

– Water temperature leaving gas cooler: 74°C.

– the heat loss to the environment from the cycle component are ignored.

In this section, the first and the second laws of thermodynamics are combined to derive useful equation for computing the total lost work of the refrigeration system.

Energy balances :

$$\sum\left(\dot{m}H\right)_{in}-\sum\left(\dot{m}H\right)_{out}+\sum\left(\dot{W}\right)_{in}-\sum\left(\dot{W}\right)_{out}+\sum\left(\dot{Q}\right)_{in}-\sum\left(\dot{Q}\right)_{out}=0 \quad [1]$$

Entropy balances:

$$\Delta\left(\dot{m}S\right)_{folowingstreams}-\frac{\dot{Q}_0}{T_0}-\sum\left(\frac{\dot{Q}_i}{T_i}\right)=\Delta\dot{S}_{irr} \quad [2]$$

Availability balance:

$$\sum\left(\dot{m}B\right)_{in}-\sum\left(\dot{m}B\right)_{out}+\sum\left(\dot{W}\right)_{in}-\sum\left(\dot{W}\right)_{out}+\sum\left[Q\left(1-\frac{T_0}{T}\right)\right]_{in}-\sum\left[Q\left(1-\frac{T_0}{T}\right)\right]_{out}=L\dot{W} \quad [3]$$

Where $B=H-T_0S$ is the availability function

The thermodynamic efficiency of this refrigeration unit is given by:

$$\eta_{ex}=\frac{Q_{LT}\left(1-\frac{T_0}{T_{LT}}\right)+Q_{gc}\left(1-\frac{T_0}{T_{win}}\right)Q_{MT}\left(1-\frac{T_0}{T_{MT}}\right)}{\dot{W}_{LT}+\dot{W}_{MT}} \quad [4]$$

Energy and exergy of each component of the CO2 booster are calculated in the following model equations:

Valv HPV :

$$h_6=h_7;I_{HPV}=\dot{m}_5\left(\left(h_6-T_0\times s_6\right)\right)-\left(h_7-T_0\times s_7\right) \quad [5]$$

Valv IPV :

$$h_9=h_8;I_{IPV}=\dot{m}_8\left(\left(h_8-T_0\times s_8\right)\right)-\left(h_9-T_0\times s_9\right) \quad [6]$$

Valv MTV

$$h_{12}=h_{10};I_{MTV}=\dot{m}_{11}\left(\left(h_{11}-T_0\times s_{11}\right)\right)-\left(h_{12}-T_0\times s_{12}\right) \quad [7]$$

Valv LTV:

$$h_{15}=h_{10} \quad [8]$$

$$I_{LTV}=\dot{m}_1\left(\left(h_{10}-T_0\times s_{10}\right)\right)-\left(h_{15}-T_0\times s_{15}\right) \quad [9]$$

Ref .Reciever:

$$\dot{m}_8 = \dot{m}_5 \times x_7 \quad [10]$$

$$\dot{m}_{10} = \dot{m}_5 - \dot{m}_8 \quad [11]$$

MTEvaporator

$$Q_{MT} = \dot{m}_{11} \times (h_{13} - h_{10}) \quad [12]$$

$$Ev_{MT} = \dot{m}_{11} \times ((h_{12} - T_0 s_{12}) - \times (h_{12} - T_0 s_{12})) + Q_{MT}\left(1 - \left(\frac{T_0}{T_{13} + 273{,}15}\right)\right) \quad [13]$$

LT Evaporator

$$\dot{m}_1 = \dot{m}_{10} - \dot{m}_{11} \quad [14]$$

$$Q_{LT} = \dot{m}_1 \times (h_1 - h_{10}) \quad [15]$$

$$Q_{LT} = \beta \times Q_{MT} \quad [16]$$

$$Ev_{LT} = \dot{m}_1 \times ((h_{15} - T_0 s_{15}) - \times (h_1 - T_0 s_1)) + Q_{LT}\left(1 - \left(\frac{T_0}{T_1 + 273{,}15}\right)\right) \quad [17]$$

LT Compressor

$$\dot{W}_{LT-Comp} = \dot{m}_1 \times (h_2 - h_1) \quad [18]$$

$$I_{LPC} = \dot{m}_1((h_1 - T_0 \times s_1)) - (h_2 - T_0 \times s_2) + \dot{W}_{LT-Comp} \quad [19]$$

HT Compressor:

$$\dot{W}_{HT-Comp} = \dot{m}_5 \times (h_5 - h_4) \quad [20]$$

$$I_{HPC} = \dot{m}_5((h_4 - T_0 \times s_4)) - (h_5 - T_0 \times s_5) + \dot{W}_{HT-Comp} \quad [21]$$

Gas Cooler:

$$\dot{Q}_{GC} = \dot{m}_5 \times (h_6 - h_5) \quad [22]$$

$$I_{GC} = \dot{m}_5((h_5 - T_0 \times s_5)) - (h_6 - T_0 \times s_6) + \dot{m}_w((h_{wi} - T_0 \times s_{wi}) - (h_{wo} - T_0 \times s_{wo})) \quad [23]$$

IP Receiver:

$$\dot{m}_5 \times h_4 = \dot{m}_8 \times h_8 + \dot{m}_{10} \times h_3 \quad [24]$$

$$I_{IPR} = -\dot{m}_5 \times (h_4 - T_0 \times s_4) + \dot{m}_8(h_8 - T_0 \times s_8) + \dot{m}_{10}(h_3 - T_0 \times s_3) \quad [25]$$

$$I_{Mix} = -\dot{m}_{10} \times (h_3 - T_0 \times s_3) + \dot{m}_{11} (h_{13} - T_0 \times s_{13}) + \dot{m}_1 (h_2 - T_0 \times s_2) \quad [26]$$

The total lost work of the system is the sum of availability in different equipments of the system and it is calculated as follows:

$$I_{tot} = I_{GC} + I_{Mix} + I_{HPC} + I_{LPC} + I_{IPR} + I_{MTV} + I_{LTV} + I_{IPV} + I_{HPV} \quad [27]$$

2.3. RESULTS AND DISCUSSION

Exergy destruction and relative irreversibility of the system and its components are presented in Table 1.

Component	Lost work(kW)	Percentage of total lost(%)
Gas cooler	8,005	19,59
Valv MTV	2,84	6,95
Valv LTV	3,58	8,76
LT Compressor	2,66	6,51
MT Compressor	8,33	20,38
IP Receiver	0,14	0,34
Mix	0,345	0,84
IPValv	6,63	16,22
Valv HPV	8,33	20,38

Table 1. *Exergy destruction and relative irreversibility of the system and its components.*

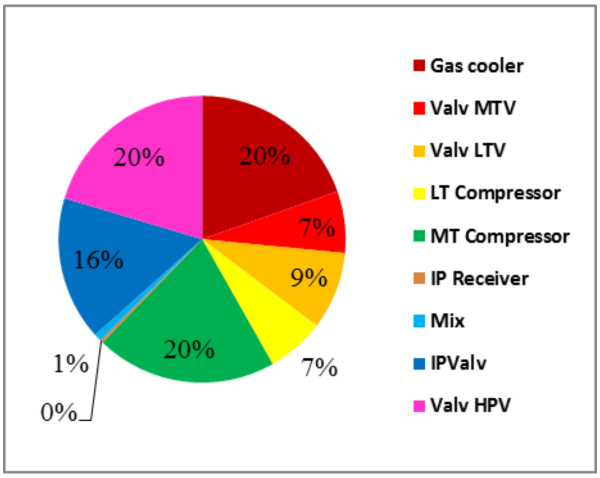

Figure 2. *Relative irreversibility of different equipment of the CO2 booster refrigeration.*

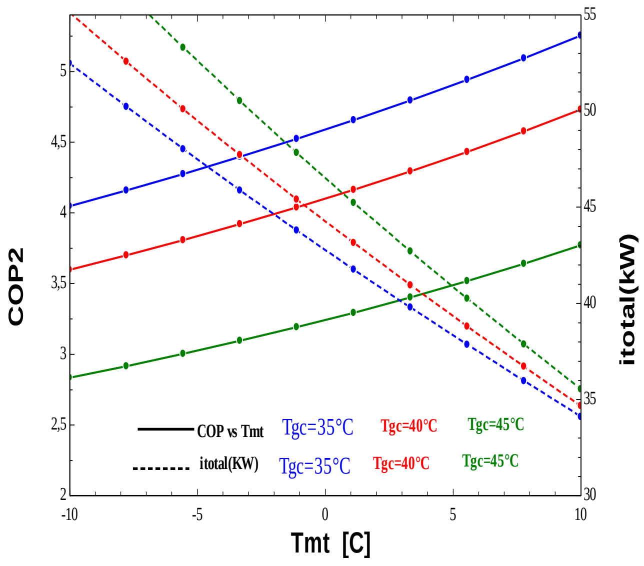

Figure 3. *COP of the CO2 cycle (continous lines) and the total lost work(dotted lines) for different MTevaporating and gas cooler outlet temperature.*

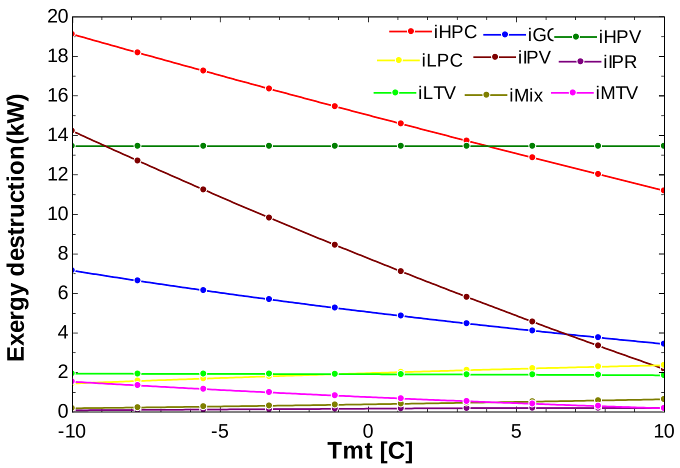

Figure 4. *Variation of irreversibility of different components with the MT evaporator temperature.*

The exergy destruction of the main components is presented in Figure 2. The largest lost work was recorded in the MT compressor, the HPV valve followed by the gas cooler, the IPV, LTV and MTV expansion valves and the compressor LT. The contribution of the MT compressor is 8.33 kW (20.38%), 8.33 kW (20.38%) for the HPV valve, 8kW(19.57%) for the gas cooler, 6.63kW(16.63%) for the IP valve, 3.58kW(8.76%) for the LT valve, 2.84kW(7%) for the MT valve, 2.67kW(6.52%) for the LT compressor and 0.76% for the rest of components. The total exergy destruction is 40.90 kW.

Figure 3 exhibit the coefficient of performance of CO2 cycle and the total lost work. The total irreversibility is more intense for lower MT evaporator temperature and in high gas cooler outlet temperature (heat rejection temperature). It is remarkable that the performance of the R744 booster increase with the increases of the MT evaporator temperature and decreases with the increase of the gas cooler outlet temperature. For instance, when Tgc=35°C, the COP of refrigeration unit is ranged from 2.04 to 2.74, while for Tgc=40°C it is ranged from 1.78 to 2.28.

Lost work values that belong to each CO2 Booster system component have been found depending on the evaporating temperature change in the MT stage and it is given in Figure 4.

3. Conclusion

The detailed energy and exergy analysis of the CO_2 booster system with flash gas bypass have been carried out in this study. It's observed that the performance of the CO_2 cycle increases when MT evaporating temperature increases and COP decreases when the heat rejection temperature increases. Irreversibility values have reached the highest values in the MT Compressor.

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