International Journal of Engineering Sciences & Research Technology

(A Peer Reviewed Online Journal)
Impact Factor: 5.164

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INTERNATIONAL JOURNAL OF ENGINEERING SCIENCES & RESEARCH TECHNOLOGY

COMPARATIVE THERMODYNAMIC PERFORMANCE ANALYSIS OF A CASCADE SYSTEM USING DIFFERENT REFRIGERANT COUPLES

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DOI: 10.5281/zenodo.3653477

ABSTRACT

In present study the comparison of thermodynamic analysis of cascade refrigeration system has been done with refrigerant pairs such as CO2-HFE7000, CO2-R134a, CO2-R152a, CO2-R32, CO2-R1234yf, CO2-NH3, CO2-Propane and CO2-Propylene. In these systems, performance of two stage cascade compression system using above different refrigerant couples, have been studied and the effect of condenser temperature & evaporator temperature, has been done. Thermodynamic analysis is carried out by developing computational model in Engineering Equation solver (EES).

KEYWORDS: Cascade refrigeration system, low temperature circuit (LTC), high temperature circuit (HTC), coefficient of performance (COP), global warming potential (GWP), ozone depletion potential (ODP)

1. INTRODUCTION

Many industrial applications require low temperature refrigeration such as quick freezing biomedical preservations, manufacturing of dry ice, liquefaction of petroleum vapors, pharmaceutical reactions etc. where evaporating temperature requires between -40ºC to- 80ºC. Condensing temperature is governed by temperature of cooling tower water which is about 35 ºC. Thus, system has to work for wide range of temperature. Single stage vapor compression system is not feasible for such application and its performance decreases below -35 ºC. Multistage or compound systems can be useful but no refrigerants available to work efficiently for high temperature lift. Also, it will be difficult to balance the oil level in compressor because of large difference in suction pressures of low stage and higher stage compressors. Cascade refrigeration system has two different stages which permits appropriate selection refrigerants to maximise system performance. Synthetic refrigerants prominently used in till now due to their excellent thermodynamic properties but owing to higher ODP (Ozone Depletion Potential), GWP (Global worming Potential) they are contributor to ozone depletion and global warming. Cascade refrigeration system is the combination of two single stage vapor compression system together, condenser of LTC and evaporator of HTC is cascaded and forms the heat exchanger where evaporator cascade absorbs the heat from the condenser cascade which further leads to better refrigeration effect.

Amongst the natural refrigerants, Lorentzen and Petterson [1] suggested the use of carbon dioxide (CO2) and seems to be the most promising one especially as the natural refrigerant [1-6]. The key advantages of CO_2 include the fact that is not explosive, non-toxic, easily available, environmental friendly and has excellent thermo-physical properties. On the other hand, researches in Norway in 1993 showed that the refrigerant leakages coming from the commercial sector were 30% of the annual total [7]. In this research, the use of a cascade system using CO2 in the low temperature stage and NH_3 in the high temperature stage turned out to be an excellent alternative for cooling applications at very low temperatures [8-10]. Researches from Eggen and Aflekt [11], Pearson and Cable [12] and Van Riessen [13] show practical examples of the use of a cascade system of CO2/NH3 for cooling in supermarkets. Eggen and Aflekt [11] developed research based on a prototype of a cooling system built in Norway. Pearson and Cable [12] showed data from a cooling system used in a Scottish supermarket line, (ASDA), and Van Riessen [13] carried out technical energy and economic research of a cooling system used in a Dutch supermarket. In the same way, different researches about the performance of different cooling systems involving CO2 have been carried out together with its reuse as a refrigerant fluid. Lorentzen and Petterson [1] evaluated the possibility of the use of a heat exchanger in a CO2 transcritical system. Hwang et al. [6] showed experimental results and simulation research including expansors and double stage cycles. Groll et al. [14] carried out a numerical analysis of a double stage cycle changing the

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compression ratio of each compression stage. It is well-known that cascade refrigeration system (CRS) is usually adopted to meet the low-temperature cooling requirement in many commercial and industrial applications where single-stage or multistage systems are insufficient. There are two cycles in a cascade refrigeration system: the high-temperature cycle (HTC) is used to absorbed the energy released by the low-temperature cycle (LTC) during the condensation process [1]. In this way, CRS can satisfy the low-temperature cooling requirement range from -30°C to -55°C [2]. Regarding energy shortage problems, much attention has been devoted to the optimization of CRS performance. One of the research topics is the selection of refrigerant couples [3]. A suitable refrigerant couple is able to provide a large temperature lift while improving system performance [2]. The HTC of a CRS can normally be charged as an intermediate-temperature refrigerant with a normal boiling point ranging from 0 °C to -60°C, such as R22 [4], R404A [5], R290, NH_3(R717), propylene (R1270), R12, R134a, and R410a, whereas the normal boiling points of low-temperature refrigerants such as R23, carbon dioxide (R744) and N2O are usually lower than -70°C. However, there is no definite temperature boundary between,

2. SYSTEM MODELLING

To aid in analysis of engineering problem it is necessary to realize the Physical model in a mathematical model. To do this, we first write state point equations of thermodynamic properties and then develop a polynomial for thermodynamic properties with the help of software or, directly taken from the reference. Therefore this chapter involves the description of physical model, mass, and energy balance, assumptions, state point equations and thermodynamic properties. To show the superiority of cascade system for low temperature application or to justify the utility of cascade system for low temperature cooling (below -40° c), it becomes necessary to analyse them separately .Thus this chapter deals with the mathematical modelling of two sections. The schematic diagram of a typical CRS and the corresponding p–h plot are shown in Fig. 1 a, b, respectively. In CRS, two single-stage vapour compression refrigeration cycles are connected with each other in series through a cascade heat exchanger. This cascade heat exchanger serves as condenser for the low-temperature circuit (LTC) and as evaporator for the high-temperature circuit (HTC). In the cascade heat exchanger, the LTC refrigerant rejects the heat which is absorbed by HTC refrigerant. In the evaporator, Qeva amount of heat is absorbed by the LTC refrigerant at evaporator temperature of Teva and gets evaporated. The vapour refrigerant then enters the LTC compressor (state 1), where WLTC amount of work is supplied to raise its temperature and pressure (state 2). It is then sent to the cascade heat exchanger where LTC refrigerant rejects Qcas amount of heat at LTC condenser temperature, TLC which is absorbed by the HTC refrigerant at HTC evaporator temperature, THE. This causes condensation (state 3) of the LTC refrigerant and evaporation (state 5) of the HTC refrigerant. The condensed LTC refrigerant then enters the LTC throttle device and expanded to the evaporator pressure (state 4) and enters the evaporator. The vapourized HTC refrigerant, coming out from the cascade heat exchanger, enters the HTC compressor. WHTC amount of work is consumed by the HTC compressor to compress the refrigerant to the condenser pressure, and this compressed and superheated refrigerant enters the condenser at state 6. The refrigerant vapour is first desuper heated and then condensed to saturated liquid (state 7) at condenser temperature of Tcond by rejecting Qcond amount of heat. The condensed HTC refrigerant is then passed through the HTC throttle device and expands to the HTC evaporator pressure at state 8. The evaporator temperature, condenser temperature, LTC condenser temperature and the temperature difference in the cascade heat exchanger are the four most important parameters which have great influence on the performance of system.

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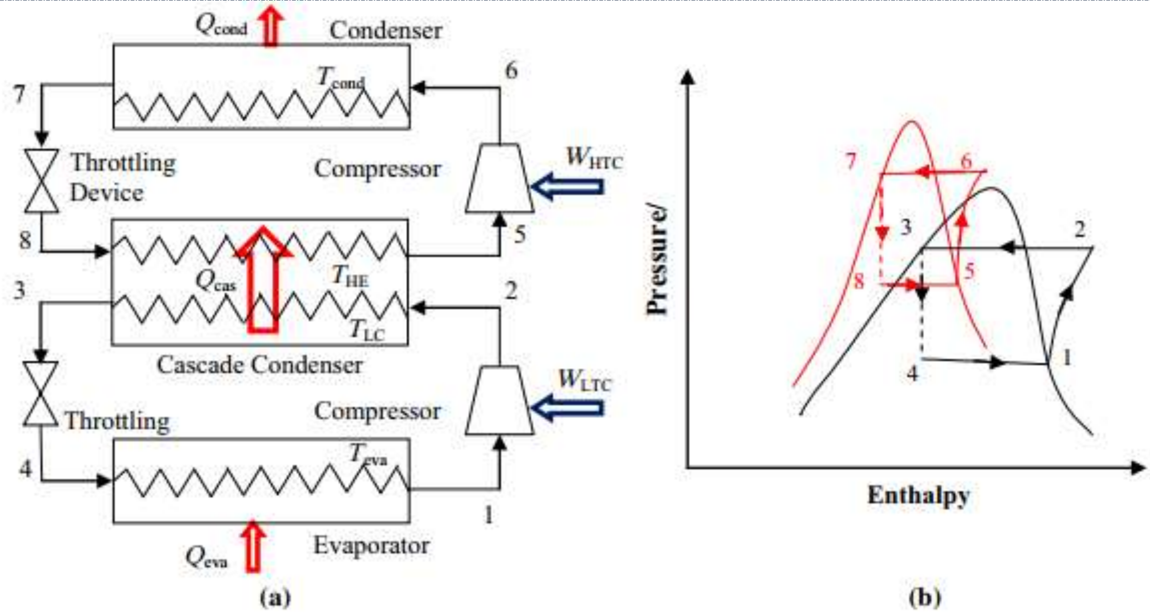

Fig. 1: Schematic diagram of the cascade refrigeration system

2.1 Assumptions

1. All components are assumed to be a steady state and steady flow processes. The changes in the potential and the kinetic energy of the components are negligible.
2. The low circuit compressor is isentropic
3. All throttling devices are isenthalpic.
4. Refrigerants at the cascaded heat exchanger outlet, condenser outlet and evaporator outlet are saturated.
5. Negligible pressure and heat losses or gains in the pipe networks or system components.
6. The dead state is T_a=25°C and P_a=1 atm.

Table 1: Input parameter values assumed in the simulation models (Yilmaz & Selbaş, 2019)

Input parameters	Value
Q_{eva} (KW)	10
ΔT (°C)	5
T_{con} (°C)	50
$T_{cas,eva}$ (°C)	0
T_{eva} (°C)	-30
$T_{cas,cond}$ (°C)	10
η_h	0.8
η_l	0.8
T_a (°C)	25

2.2 Mass Balance and Energy Balance

In table 2 specific equations for each system's component are summarized. The system's COP has been calculated by the following equation

$$COP = Q_L / W_{Total}$$

Table 2: Energy and Mass Balance for R22-R404a Cascade System

Component	**Mass**	**Energy**
HTC-Compressor	$m_7=m_8$	$W_{comp} = m_1.(h_8-h_7)/\eta_l$
LTC-Compressor	$m_6=m_5$	$W_{comp}= m_2.(h_3-h_2) /\eta_h$
HTC- Exp. Device	$m_5=m_6$	$h_5=h_6$
LTC- Exp. Device	$m_4=m_1$	$h_4=h_1$
Evaporator	$m_6=m_7$	$Q_{evap} = m_1(h_7-h_6)$
Condenser	$m_4=m_3$	$Q_{cond} = m_5(h_4-h_3)$
Cascade heat exchanger	$m_1=m_2, m_5=m_8$	$m_1.(h_1-h_2)=m_2.(h_5-h_8)$

3. THERMODYNAMIC ANALYSIS

The thermodynamic analysis of CO_2-HFE7000, CO_2-R134a, CO_2-R152a, CO_2-R32, CO_2-R1234yf, CO_2-365mfc, CO_2-NH3, CO_2-Propane and CO_2-Propylene based cascade refrigeration system performed based on the following assumptions.

1. Compression process is adiabatic with an isentropic efficiency of 0.8 in both HTC and LTC;
2. The expansion process is isenthalpic;
3. Negligible heat interaction in the cascade heat exchanger with surrounding;
4. Negligible changes in kinetic and potential energy;
5. The system is at steady state condition. All processes are steady flow processes.
6. Temperature difference in the cascade heat exchanger is 5°C.

The thermo-physical properties of CO_2-HFE7000, CO_2-R134a, CO_2-R152a, CO_2-R32, CO_2-R1234yf, CO_2-365mfc, CO_2-NH3, CO_2-Propane and CO_2-Propylene were specified in this work were calculating using a software package called engineering equation solver (EES). A major feature of EES is the high accuracy thermodynamic and transport property database that is provided for hundreds of substances in a manner that allows it to be used with the equation solving capability. The cycle is modelled by applying mass balance and energy balance equation for each individual process of the cycle. The equations for the different components of the cascade refrigeration system are given in the previous chapter.

4. MATHEMATICAL FORMULATION

Mass balance

$$\sum_{in} m^{\cdot} = \sum_{out} \dot{m}$$

Energy balance:

$$\dot{Q} - \dot{w} = \sum_{out} \dot{m}h - \sum_{in} \dot{m}h$$

Exergy balance:

$$\dot{X}_{Lost} = \sum_{out} \left(1 - \frac{T_o}{T_i}\right) \dot{Q}_i - \dot{w} + \sum_{in} \dot{m}\varphi - \sum_{out} \dot{m}\, \varphi$$

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$$\dot{m}_l = \frac{Q_e}{h_1 - h_5}$$

The compressor input work of LTC:

$$w_{comp.l} = \frac{\dot{m}_l(h_2 - h_1)}{\eta_l}$$

The heat load of cascade heat exchanger:

$$Q_{cas} = \dot{m}_l \times (h_2 - h_4) = \frac{Q_e(h_2 - h_4)}{h_1 - h_5}$$

The coefficient of performance in LTC:

$$COP_l = \frac{Q_e}{w_{comp.l}} = \frac{(h_1 - h_5)\eta_l}{h_2 - h_1}$$

The mass flow rate of HTC:

$$\dot{m}_h = \frac{Q_{cas}}{h_6 - h_{10}} = \frac{Q_e(h_2 - h_4)}{(h_6 - h_{10})(h_1 - h_4)}$$

$$w_{comp.h} = \frac{\dot{m}_h(h_7 - h_6)}{\eta_h}$$

The coefficient of performance in HTC:

$$COP_h = \frac{Q_{cas}}{w_{comp.h}}$$

The total input work of the both compressors:

$$w_{comp} = w_{comp.l} + w_{comp.h}$$

The heat load of the condenser:

$$Q_{cond.h} = \dot{m}_h(h_7 - h_9)$$

The overall coefficient of performance of CRS:

$$COP = Q_e / w_{comp}$$

The exergy loss in the compression process in LTC:

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$$X_{comp.l} = T_a \dot{m}_l (s_2 - s_1)$$

The exergy loss in the expansion process in LTC:

$$X_{exp.l} = T_a \dot{m}_l (s_5 - s_4)$$

The exergy loss in the evaporation process in LTC:

$$X_{eva.l} = T_a \left[\dot{m}_l (s_1 - s_5) - \frac{Q_e}{T_e + \Delta T} \right]$$

The exergy loss in the compression process in HTC:

$$X_{comp.h} = T_a \dot{m}_h (s_7 - s_6)$$

The exergy loss in the condensation process in HTC:

$$X_{cond.h} = T_a \left[\dot{m}_h (s_9 - s_7) + \frac{Q_{cond}}{T_a} \right]$$

The exergy loss in the expansion process in HTC:

$$X_{exp.h} = T_a \dot{m}_h (s_{10} - s_9)$$

The exergy loss in cascade heat exchanger in the refrigeration system:

$$X_{cas} = T_a [\dot{m}_l (s_4 - s_2) + \dot{m}_h (s_6 - s_{10})]$$

The total exergy loss in the cascade refrigeration system:

$$X_{total} = X_{comp.l} + X_{comp.h} + X_{cond.h} + X_{exp.h} + X_{exp.l} + X_{eva.l} + X_{cas}$$

The exergy efficiency of the system

$$\eta = \frac{w_{comp.l} + w_{comp.h} - X_{total}}{w_{comp.l} + w_{comp.h}}$$

5. RESULTS AND DISCUSSION

In this work, CCS for heating and cooling applications is investigated by thermodynamic performance assessment perspective. The thermodynamic properties of the subcomponents of CCS are calculated by EES software (EES, 2017). The temperature, pressure, enthalpy, entropy, and exergy rate of CCS stages for different refrigerants are illustrated in Tables (3–8). The values given in Tables (3-7) are calculated under the same thermodynamic condition (Tev = −20°C and Tc = 40°C). In Tables (5.1-5.5), the temperature difference at point 6 is due to saturated steam of refrigerants. These tables allow a detailed view of the thermodynamic properties of each point of the CCS for the different refrigerant pair. Because of the constant CO_2 refrigerant is used in the LTC section, the thermodynamic values between 1 and 4 points are the same.

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Table 3: Process data flows in the CO_2-HFE7000 CCS (Tev = −20°C and Tc = 40°C)

State point	T (°C)	P (kPa)	h(kJ/kg)	s (kJ/Kg-°C)
1	-18	1970	-67.34	-0.7804
2	45.36	4502	-25.86	-0.76
3	11.47	4502	-277	-1.637
4	-20	1970	-277	-1.608
5	5	27.77	352	1.55
6	40	124.6	373.2	1.564
7	35	124.6	243.6	1.15
8	3	27.77	243.6	1.158

Table 4: Process data flows in the CO_2-R134a CCS (Tev = −20°C and Tc = 40°C)

State point	T (°C)	P (kPa)	h(kJ/kg)	s (kJ/Kg-°C)
1	-18	1970	-67.34	-0.7804
2	45.36	4502	-25.86	-0.76
3	11.47	4502	-277	-1.637
4	-20	1970	-277	-1.608
5	5	326.2	254	0.9363
6	51.38	1017	283.9	0.9549
7	35	1017	100.8	0.371
8	3	326.6	100.8	0.3818

Table 5: Process data flows in the CO_2-R152a CCS (Tev = −20°C and Tc = 40°C)

State point	T (°C)	P (kPa)	h(kJ/kg)	s (kJ/Kg-°C)
1	-18	1970	-67.34	-0.7804
2	45.36	4502	-25.86	-0.76
3	11.47	4502	-277	-1.637
4	-20	1970	-277	-1.608

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5	5	294	510.9	2.126
6	59.48	910.3	558.2	2.155
7	35	910.3	262.3	1.212
8	3	294	262.3	1.226

Table 6: Process data flows in the CO_2-R32 CCS (Tev = −20°C and Tc = 40°C)

State point	T (°C)	P (kPa)	h(kJ/kg)	s (kJ/Kg-°C)
1	-18	1970	-67.34	-0.7804
2	45.36	4502	-25.86	-0.76
3	11.47	4502	-277	-1.637
4	-20	1970	-277	-1.608
5	5	894.2	518.3	2.153
6	78.01	2478	569.9	2.182
7	35	2478	265.1	1.218
8	3	894.2	265.1	1.235

Table 7: Process data flows in the CO_2-R1234yf CCS (Tev = −20°C and Tc = 40°C)

State point	T (°C)	P (kPa)	h(kJ/kg)	s (kJ/Kg-°C)
1	-18	1970	-67.34	-0.7804
2	45.36	4502	-25.86	-0.76
3	11.47	4502	-277	-1.637
4	-20	1970	-277	-1.608
5	5	349.2	367.1	1.605
6	43.51	1018	391.2	1.62
7	35	1018	247.6	1.162
8	3	349.2	247.6	1.172

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5.1 Performance of Cascade System According to the Studied Working Fluids

To validate the present model, the simulation results have been compared with the available numerical data in the literature using the different refrigerant couples. The results of this study were compared with the simulation data published by Yilmaz & Selbaş (2019). Table 5.1 presents the combined system performances for the same operating conditions used by Yilmaz & Selbaş (2019).

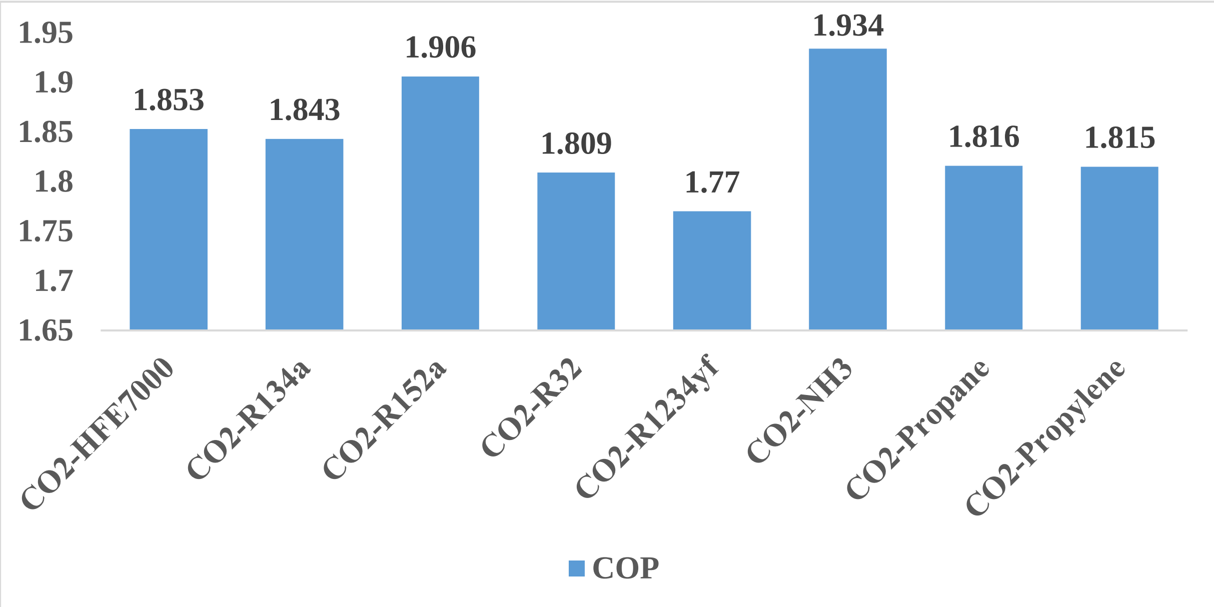

Fig. 2: Variation of COP with different refrigerant couples

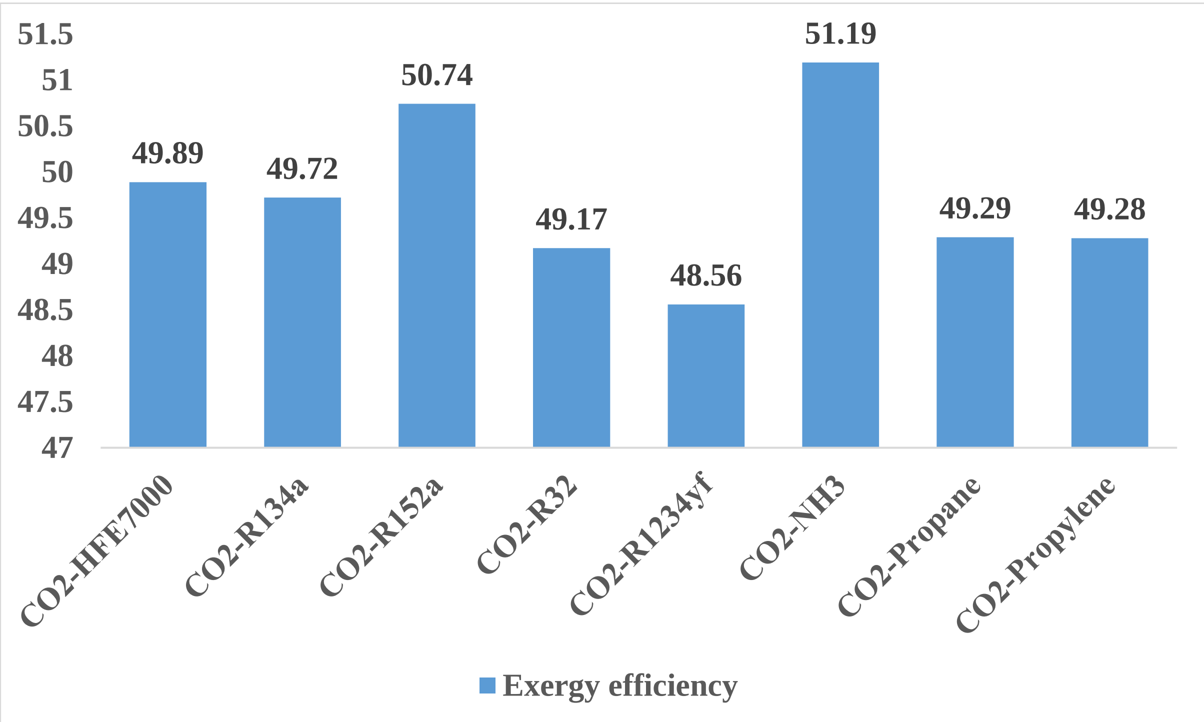

Fig. 3: Variation of exergy efficiency with different refrigerant couples

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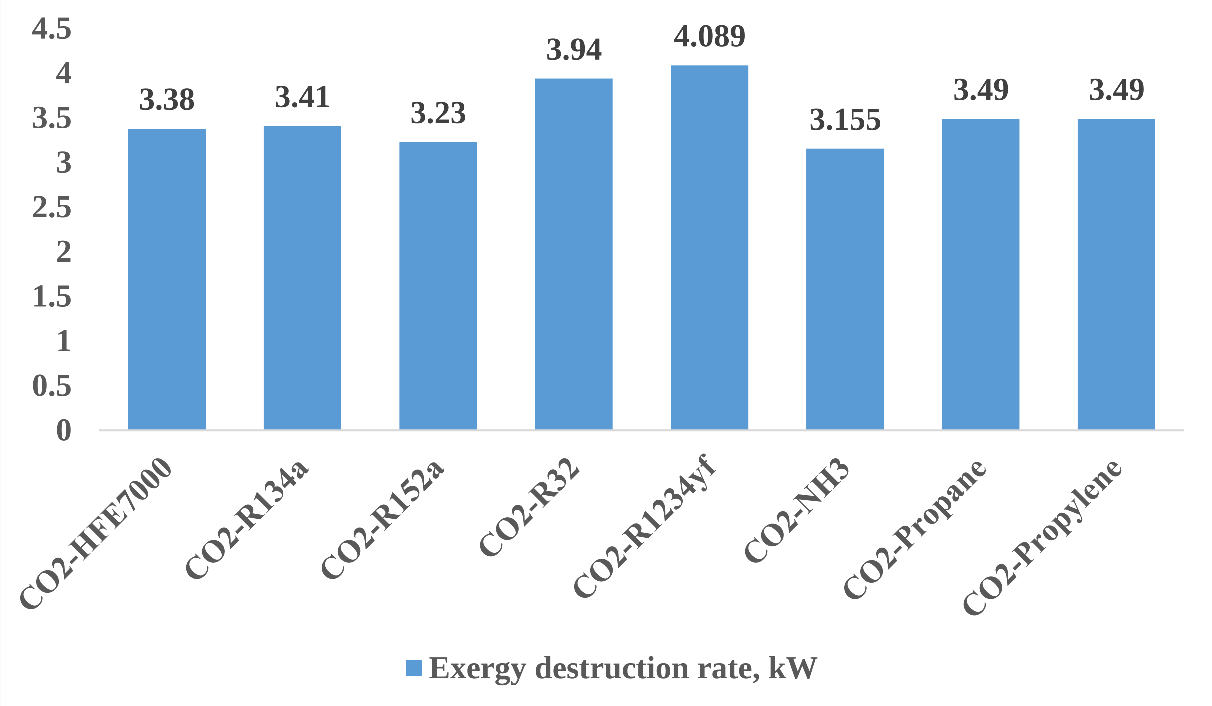

Fig. 4: Variation of exergy destruction rate with different refrigerant couples

According to comparison, the agreement between the two simulation results is good. The results in Table 8 show that among all candidates, CO_2-NH_3 has a highest COP among the all refrigerant couples at evaporator temperature -30°C and condenser temperature of 50°C. With respect to refrigerant couple CO_2-NH_3 exergy efficiency is maximum among the all refrigerant couples. Exergy destruction rate is minimum in case of refrigerant couple CO_2-NH_3 among the all refrigerant couples.

5.2 Effect of Evaporator Temperature on the Performance of Cascade System

In the present work thermodynamic model has been developed in Engineering Equation Solver software and results of the analysis have been given in the following sections.

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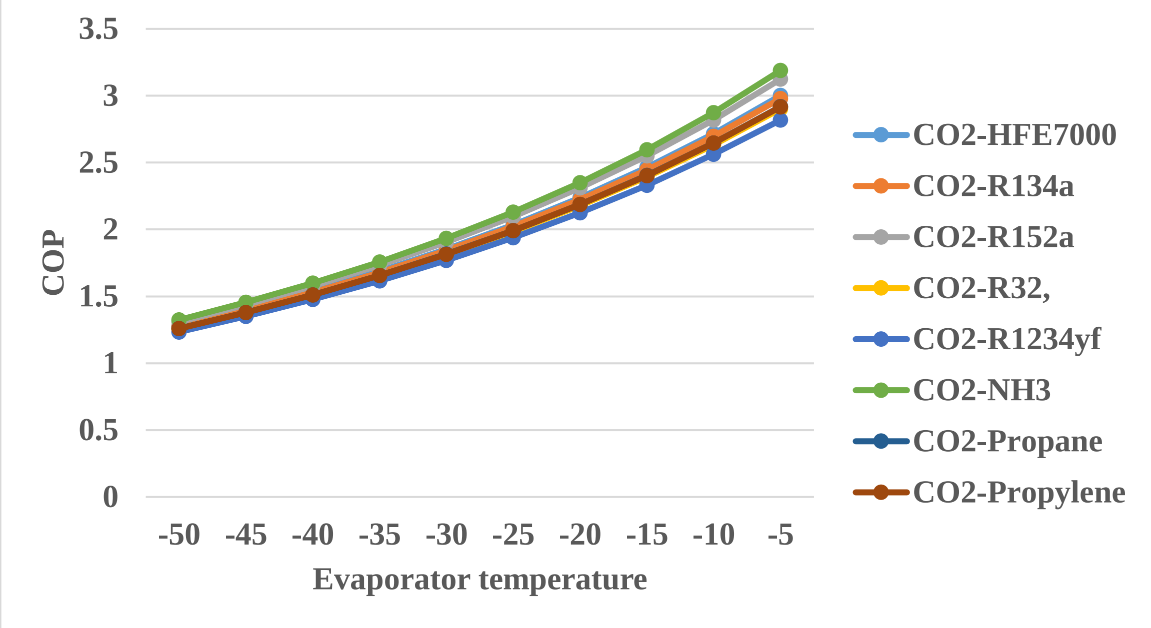

Fig. 5: Variation of COP with different refrigerant couples

Figure 5 explains the effects of varying evaporator temperature on the COP of the CCS, at constant 50°C condenser temperature. In Figure 5.4, the evaporator temperature increasing from −50 to −5°C, the COP of CCS is increasing. Here, the highest COP of CCS is calculated as 3.189 for the CO_2/NH_3 refrigerant couple, at the −5°C evaporator and 50°C condenser temperatures. It is understood from Figure 5, when the evaporator temperature increases, the COP of CCS is increases. In addition, the rise in evaporator temperature has a favorable effect on system efficiency.

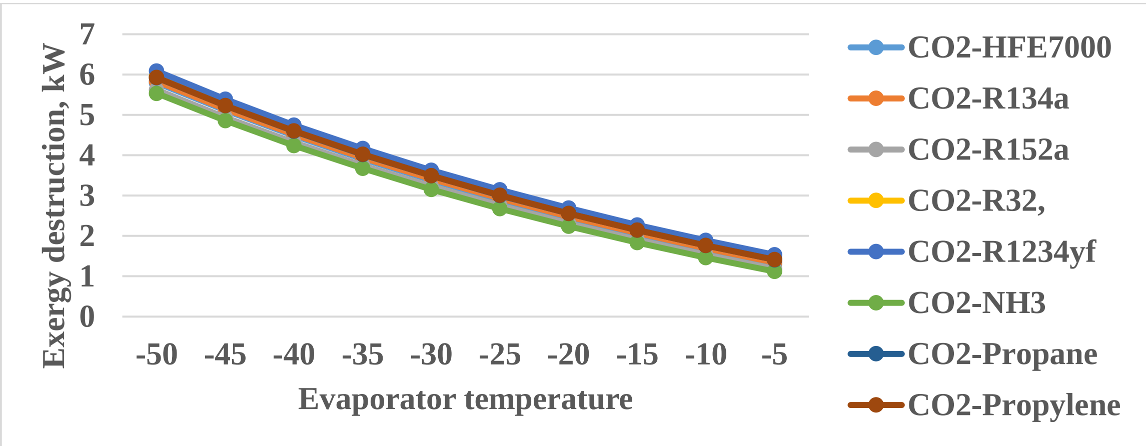

Fig. 6: Variation of exergy destruction rate with different refrigerant couples

Figure 6 explains the effects of varying evaporator temperature on the exergy destruction rate of the CCS, at constant 50°C condenser temperature. In Figure 6, the evaporator temperature increasing from −50 to −5°C, the exergy destruction rate of CCS is decreasing. Here, the highest exergy destruction rate of CCS is calculated as 5.796 kW for the CO_2-HFE7000 refrigerant couple, at the −50°C evaporator and 50°C condenser temperatures. Figure 7 explains the effects of varying evaporator temperature on the exergy efficiency of the CCS, at constant 50°C condenser temperature. In Figure 7, the evaporator temperature increasing from −50 to −5°C, the exergy efficiency of CCS is increasing. Here, the highest exergy efficiency of CCS is calculated as 68.42 % for the CO_2-HFE7000 refrigerant couple, at the −50°C evaporator and 50°C condenser temperatures.

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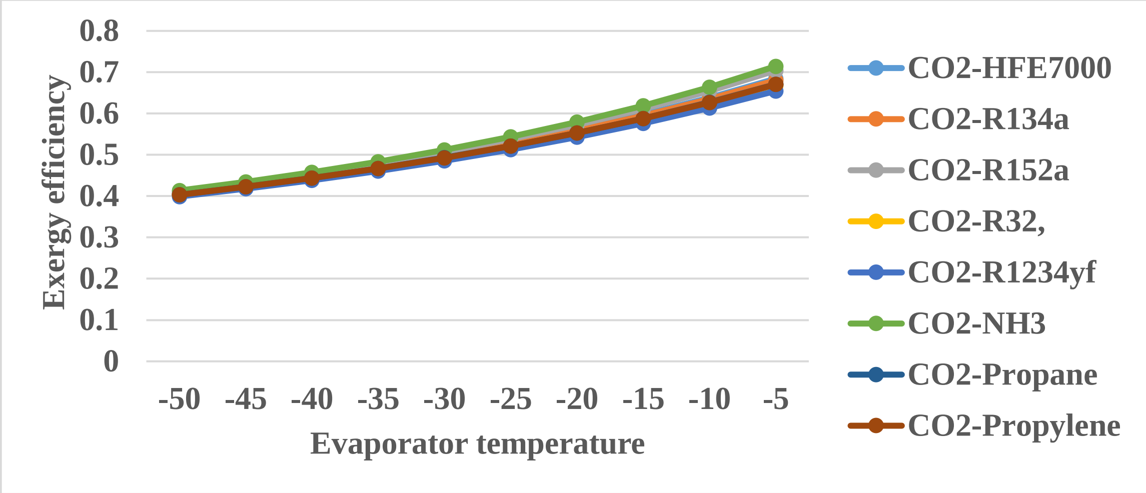

Fig. 7: Variation of exergy efficiency with different refrigerant couples

As the evaporator temperature increases, the refrigeration effect increases marginally and the required compressors work decrease significantly, therefore the performance of the cascade system increases considerably.

5.3 Effect of Condenser Temperature

In the present work thermodynamic model has been developed using Engineering Equation Solver software and results of the analysis have been given in the following sections. Fig. 8 shows the effect of condenser temperature on the COP of cascade system, refrigeration effect and required work in compressors. The results are obtained at fixed -50°C evaporator temperature. As the evaporator and coupling temperatures are fixed, the refrigeration effect will be constant for entire range of condenser temperature. However required work in HTC increases due to increase in pressure ratio in HTC. Hence combined work required increases, therefore the COP of cascade system decreases.

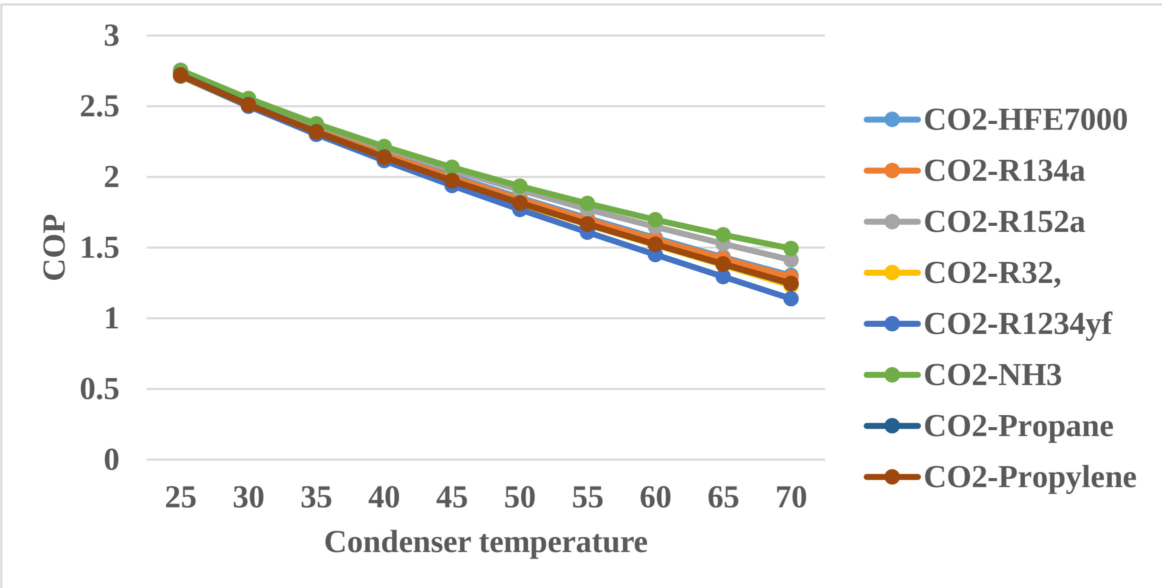

Fig. 8: Variation of COP with different condenser temperature

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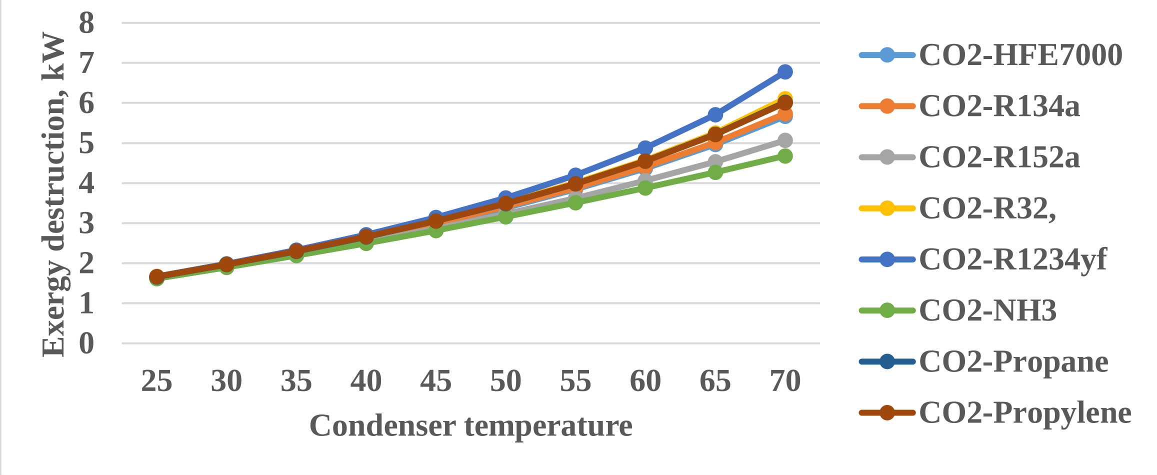

Fig. 9: Variation of exergy destruction rate with different condenser temperature

Figure 9 explains the effects of varying condenser temperature on the exergy destruction rate of the CCS, at constant -50°C evaporator temperature. In Figure 9, the condenser temperature increasing from 25 to 70°C, the exergy destruction rate of CCS is increasing. Here, the highest exergy destruction rate of CCS is calculated as 6.006 kW for the CO_2-HFE7000 refrigerant couple, at the −50°C evaporator and 50°C condenser temperatures. Figure 10 explains the effects of varying condenser temperature on the exergy efficiency of the CCS, at constant -50°C evaporator temperature. In Figure 10, the condenser temperature increasing from 25 to 70°C, the exergy efficiency of CCS is increasing. Here, the highest exergy efficiency of CCS is calculated as 44.08% for the CO_2-NH_3 refrigerant couple, at the −50°C evaporator and 70°C condenser temperature.

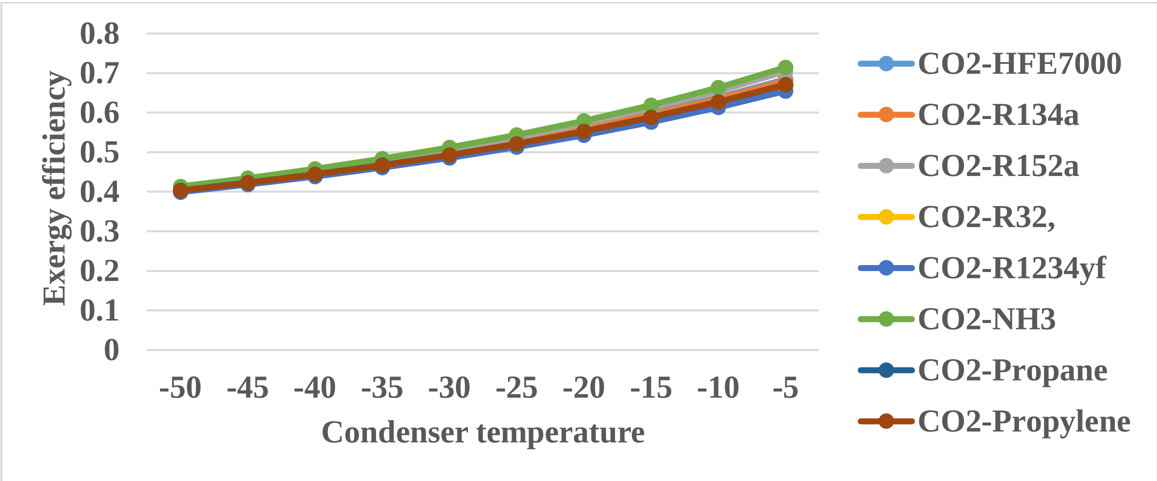

Fig. 10: Variation of exergy efficiency with different refrigerant couples

As the evaporator and coupling temperatures are fixed, the refrigeration effect will be constant for entire range of condenser temperature. However required work in HTC increases due to increase in pressure ratio in HTC. Hence combined work required increases, therefore the COP of cascade system decreases.

6. CONCLUSION

In the proposed work, a comparative thermodynamic performance evaluation of CCS for cooling and heating applications with various refrigerant couples is performed. In this context, the COP, exergy destruction rate, and exergy efficiencies of the overall system are analyzed with the thermodynamic equilibrium equations, such as mass, energy, entropy, and exergy, and also the energy and exergy efficiency equations. In addition, the parametric study is performed to comprehend how evaporator and condenser temperatures influence on the energy and exergy efficiency, and exergy destruction rate of the system for a more efficient process design. Finally, the concluding remarks from the results of this paper can be written as follows:

1. Environment-friendly refrigerants are offered in way of ODP and GWP values.
2. The CO_2 refrigerant is preferred in LTC to reach in lower temperatures.
3. The highest coefficient of performance CCS for cooling application is found as 1.934 for CO_2-NH_3 refrigerant couple, whereas the lowest coefficient of performance CCS is obtained as 1.777 for CO_2-R1234yf refrigerant couple.
4. The exergy efficiencies of CCS for cooling application are calculated as 49.89, 49.72, 50.74, 49.17, 48.56, 51.19, 49.29 and 49.28 for CO_2-HFE7000, CO_2-R134a, CO_2-R152a, CO_2-R32, CO_2-R1234yf, CO_2 – NH_3, CO_2-Propane, CO_2 – Propylene.
5. The largest exergy destruction rate occurs in the heat exchanger for all the CCS components. In addition, the exergy destruction rates of the heat exchanger are close to each other in different refrigerant pairs because it is operated at the same temperature ranges.
6. The expansion valves have the lowest exergy destruction rate among the CCS components. The COP of CCS for cooling and heating applications decreases at a constant evaporator temperature by increasing the condenser temperature.
7. The exergy efficiency and COP of the CCS increased positively with increased evaporator temperature.
8. The CO_2- NH_3 refrigerant pair is the best couple for the proposed CCS among all considered couples according to exergy efficiency.

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