ENERGY AND EXERGY EFFICIENCIES OF DIFFERENT CONFIGURATIONS OF THE EJECTOR-BASED CO_2 REFRIGERATION SYSTEMS

S. TASLIMITALEGHANI, M. SORIN & S. PONCET
Department of Mechanical Engineering, Université de Sherbrooke, Sherbrooke (QC), Canada.

ABSTRACT

Carbon dioxide (CO_2) is an appropriate replacement for conventional refrigerants due to its low global warming effects. However, its application within a traditional refrigeration compression cycle leads to low thermodynamic performance due to the large expansion losses in a throttling process. The application of ejectors allows reducing these losses. Many scenarios of ejector-based cycles have been proposed. Among them four different configurations may be distinguished: an expansion work recovery cycle (EERC), a liquid recirculation cycle (LRC), an increasing compressor discharge pressure cycle (CDPC) and a vapor jet refrigeration cycle (VJRC). This study deals with the comparative analysis of these cycles. In order to study the performance of the cycles, the numerical simulations are developed using EES software. Two performance criteria, energy efficiency (COP) and exergy efficiency are evaluated for each cycle. The highest values of these criteria point to the most thermodynamically efficient cycle. The results show that the EERC has the highest COP and exergy efficiency compared to other cycles. For example, the COP of the EERC is 3.618 and the exergy efficiency is 9.68%. The COP (resp. exergy efficiency) is approximately 23.3% (resp. 23.3%), 24.9% (resp. 25.5%) and 5.6 times (resp. 56.2%) higher than the corresponding energy and exergy efficiencies of LRC, CDPC and VJRC. Moreover, in comparison with a basic throttling valve cycle, the COP and exergy efficiency in EERC are higher up to 23% and 24% correspondingly. The detailed exergy analysis of EERC cycle has pinpointed the equipment where the major exergy losses take place. The largest losses occur in the evaporator (about 33% of the total exergy destruction of the cycle) followed by the compressor (25.5%) and the ejector (24.4%).

Keywords: comparative analysis, COP, ejector, exergy efficiency, refrigeration systems, transcritical CO_2 cycles, two-phase.

1 INTRODUCTION

Carbon dioxide (CO_2) is an appropriate replacement for conventional refrigerants due to its low global warming effects. One of the disadvantage of the cycle is a large exergy loss due to an important pressure reduction during expansion of CO_2 from the supercritical to the sub-critical state in a throttling valve. Among different devices for expansion work recovery, ejector is a favourable equipment, which enables to reduce losses by recovering part of the expansion work in a throttling process and improve the cycle's efficiency.

The first application of two-phase ejector to the transcritical CO_2 cycle was first described by Gay [1]. It was proposed to replace the expansion valve by a two-phase ejector to reduce the losses due to the throttling process.

Kornhauser [2] was the first to develop a one-dimensional and homogeneous model of a two-phase ejector using R12 as a refrigerant in the ejector expansion refrigeration system (EERS).

Li and Groll [3] adapted the Kornhauser's model for an ejector used within a transcritical CO_2 air-conditioning system.

A thermodynamic exergy analysis of transcritical CO_2 ejector refrigeration system was performed by Fangtian and Yitai [4]. They evaluated COP and exergy destruction of the

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system. Their results showed an improvement of 30% in COP and decreasing exergy loss more than 25% compared to the conventional system.

The present study is focused on a thermodynamic comparative analysis of the performance of the different transcritical CO_2 ejector cycles to identify the most efficient one. The COP, exergy efficiency and exergy destructions are calculated and compared for the expansion work recovery cycle (EERC), liquid recirculation cycle (LRC), increasing compressor discharge pressure cycle (CDPC) and vapor jet refrigeration cycle (VJRC). Exergy analysis is employed to determine the amount and locations of irreversibilities within different components of each cycle.

2 EJECTOR APPLICATIONS FOR TRANSCRITICAL CO_2 CYCLES

Different applications of the ejector in CO_2 air-conditioning and refrigeration systems used in this study are as follows:

- Ejector for utilization of low-grade energy (vapor jet ejector systems, VJRC)
- Ejectors for expansion work recovery cycle (standard two-phase ejector, EERC)
- Ejectors for liquid recirculation cycle (LRC)
- Ejector for increasing compressor discharge pressure (CDPC)

2.1 Vapor jet ejector systems (single-phase ejectors)

In the vapor jet cycle, a pump, a generator, and an ejector replace the compressor. A fraction of the liquid from the condenser is pumped to a high pressure and temperature. The fluid absorbs heat at a constant pressure from a low-grade energy source in the generator. The heated flow expands in a primary nozzle to a high velocity and a low pressure. This low pressure entrains the secondary flow from the evaporator into the mixing chamber of the ejector. The irreversible mixing of the two fluids occurs in the mixing chamber depending on the ejector geometry at the constant pressure or at the constant area. Finally, the flow decelerates in the diffuser by converting the remaining kinetic energy into the pressure increase. The vapor exiting the diffuser is condensed at a constant pressure. The liquid at the condenser exit is pumped to the generator. The vapor is sent through the metering valve to the evaporator.

The main advantage of the VJRCs is that they can produce a refrigeration effect by using the low-grade waste heat for heating the primary flow in the generator.

Compared to a conventional system, for the same pressure increase, the work of the liquid pump in the VJRC is less than the compressor work and it does not also require any lubrication [5]. The schematic of a transcritical CO_2 VJRC and corresponding temperature-specific entropy diagram are shown in Fig. 1. It can be seen the flow through the mixing section and the diffuser remains vapor so the ejector works in a single-phase mode.

2.2 Two-phase ejectors for expansion work recovery (EERC)

A two-phase ejector can be used in vapor compression systems for recovery of the expansion work by reducing the throttling losses to improve the performance of the system.

As shown in Fig. 2, the subcritical CO_2 coming from the vapor port of the separator is compressed to high pressure and temperature to the supercritical state. It releases heat in the gas cooler. After the gas cooler exit, the stream enters the primary nozzle of the ejector and expands at the mixing section. The secondary vapor stream pre-accelerates into the mixing section. The

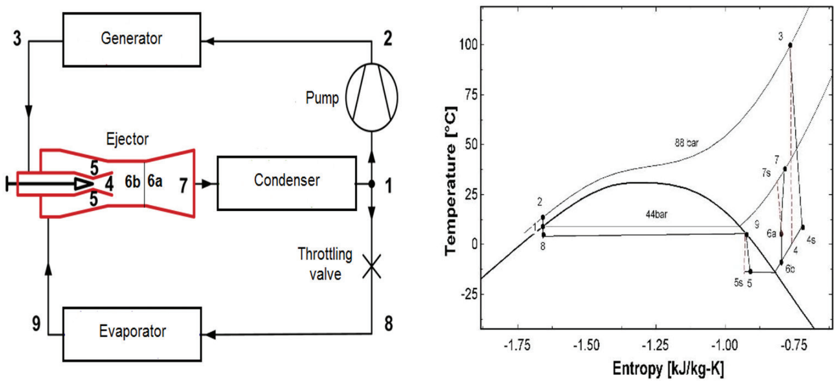

Figure 1: Transcritical CO_2 vapor jet refrigeration cycle and corresponding temperature-specific entropy diagram.

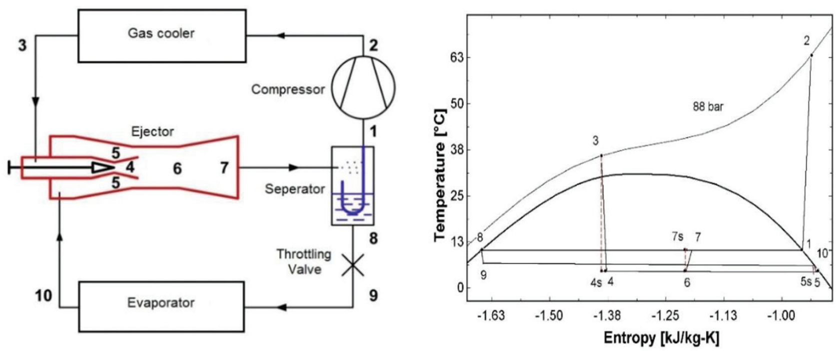

Figure 2: Transcritical CO_2 ejector expansion recovery cycle and the corresponding temperature-specific entropy diagram.

mixture then flows through the diffuser what causes a compression before entering the separator. Vapor portion of the two-phase flow returns to the compressor while the pressure of the liquid portion is reduced through the metering valve before entering the evaporator. The stream absorbs heat in the evaporator before it enters the ejector.

EERC has two main advantages. First, the cooling capacity increases because the isentropic expansion inside the primary nozzle in comparison to an isenthalpic expansion valve of a conventional system has a larger enthalpy difference. Second, the compressor work is decreased due to the increase of the suction pressure of the compressor resulting in COP improvement.

2.3 Two-phase ejectors for liquid recirculation

In this cycle, the ejector is used to recirculate liquid and improve the evaporator performance. It was first patented by Phillips [6] and later by Lorentzen [7]. Figure 3 shows a schematic of the cycle and corresponding T-S diagram for transcritical CO_2. The expansion work

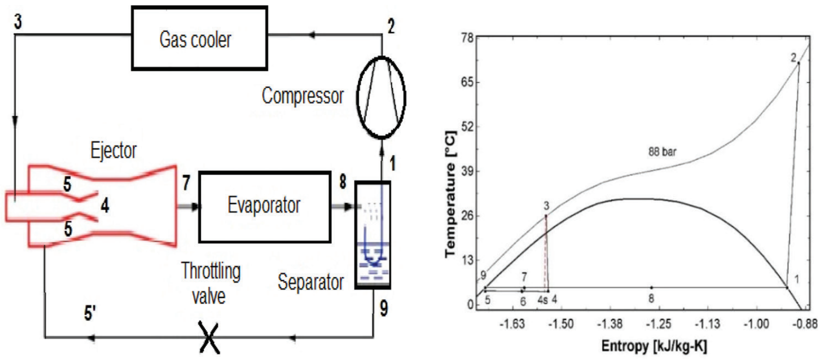

Figure 3: Transcritical CO_2 liquid recirculation cycle and the corresponding temperature-specific entropy diagram.

recovered by the ejector is used to liquid recirculation. The amount of COP improvement by using liquid recirculation is dependent on the working fluid used. There is also an optimum value for every system and operating condition because with increasing the recirculation both the heat transfer coefficient and the pressure drop increase [8].

Lawrence and Elbel [8] studied two applications of the two-phase ejector cycle for CO_2 as a working fluid: first was liquid recirculation cycle that used the ejector to improve the evaporator performance and other was a standard two-phase ejector that used work recovery of the ejector to increase the compressor pressure. The COP improvement of 3% through CO_2 ejector was obtained in the recirculation cycle as it could reach up to 25% in a standard two-phase ejector to directly unload the compressor pressure.

2.4 Two-phase ejector for increasing compressor discharge pressure

This innovative cycle was recently introduced by Bergander [9]. In this cycle, ejector is used as a second-stage compressor. Unlike standard two-phase ejector which increases the suction pressure of the compressor, in this cycle, the ejector is used to increase the compressor discharge pressure. In a subsequent work, Bergander [10] developed a thermodynamic ejector model for R22 and conducted experiments that showed 16% COP improvement. In this cycle, there is a two-phase flow inside the ejector, liquid for the primary flow and vapor for the secondary one. The primary flow enters the ejector after exiting the pump and mixes with the secondary flow that comes from the compressor. The flow at the exit of the ejector enters the gas cooler. The layout of this cycle and corresponding T-S diagram are presented in Fig. 4.

3 EXERGY ANALYSIS OF DIFFERENT EJECTOR CYCLES

The exergy analysis of the ejector cycles introduced in Section 2 is carried out to investigate the exergy destruction of the different components of the system to determine the maximum performance and potential improvements of the cycles.

3.1 Modeling of two-phase flow ejector

Different models of the ejectors exist according to assumptions, governing equations, auxiliary conditions, mixing mechanism and solution methods. Thermodynamic modeling is a simple way to solve the equations in one dimension. It is also easily integrated into a system.

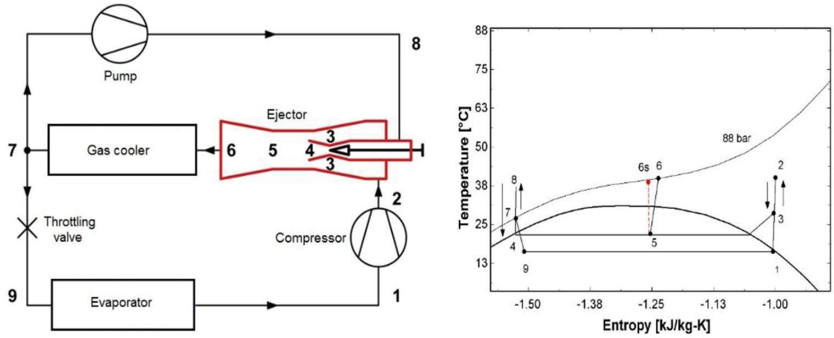

Figure 4: CO_2 transcritical ejector cycle to increase the compressor discharge pressure and corresponding temperature-specific entropy diagram.

Conservation equations of mass, energy and momentum, some gas dynamic equations, state equations, isentropic relations as well as some appropriate assumptions, initial and boundary conditions are used to solve the flow within the ejector. Some assumptions that are usually employed to simplify the problem are as follows: adiabatic walls of the ejector, steady-state flow, isentropic efficiencies for the nozzles and the diffuser, stagnation points of the streams at inlets and outlet of the ejector and mixing coefficient for mixing losses.

Most ejector models presented for CO_2 two-phase flows are based on a homogeneous equilibrium model in which both gas and liquid are in thermodynamic and mechanical equilibrium. It means that both phases have the same pressure, temperature, velocity, turbulence kinetic energy and turbulence dissipation rate [2, 3, 11–16].

3.1.1 Assumptions and calculation procedure

The thermodynamic model of the two-phase ejectors (EERC, LRC, CDPC, section 2.2~2.4) is based on the following assumptions:

1. Flow is one-dimensional, steady state and adiabatic through the ejector.
2. The homogeneous equilibrium is assumed for two-phase flow.
3. The CO_2 thermodynamic and transport properties of the primary and secondary flows are obtained from real fluids properties.
4. Flow losses in the pipes and heat exchangers are negligible.
5. Kinetic energies of the refrigerant are negligible at the ejector inlet and outlet.
6. Friction losses are defined in terms of isentropic efficiencies in the nozzles, diffuser and mixing.
7. Mixing occurs under a constant pressure in the ejector mixing section with the assumption that the fluid momentum is conserved.
8. Pressure loss of the secondary flow is assumed $\Delta P=1bar$ for EERC, LRC, CDPC.
9. Critical-mode operation is applied for VJRC and normal shock takes place at the end of the constant area mixing chamber [17, 18] .
10. The secondary inlet flow is considered as a saturated vapor in EERC, saturated liquid in LRC, superheated vapor in CDPC.
11. The heat sink temperature (or the ambient temperature) is 35°C for EERC, LRC, CDPC and $\left(T_{gen,in}+T_{gen,out}\right)/2+5$°C for VJRC; the heat source temperature is 27°C.

Table 1: Constant parameters used in the simulation.

Pressure of gas cooler	88 bar	Cooling capacity	72 kW
Temperature of gas cooler exit	36°C	Efficiency of Nozzles	0.8
Temperature of evaporator	5°C	Efficiency of diffuser	0.8
Temperature of generator exit (VJRC)	100°C	Efficiency of mixing	0.95
Pressure of generator (VJRC)	88 bar	Efficiency of compressor	0.75
Pressure of evaporator	39.69 bar	Efficiency of pump	0.75

The constant parameters used in the simulations of the cycles are shown in Table 1.

An engineering equation solver (EES) program is used to solve the proposed models in Section 2, which combines non-linear equations with thermo-physical property functions.

The modeling of the ejector expansion recovery cycle is based on one unit of mixing refrigerant mass flow in the mixing sector of the ejector. Therefore, the primary mass flow from the gas cooler is $1/(1+\mathrm{ER})$ and the secondary mass flow from the evaporator is ER/(1 + ER).

The model is solved according to the relationship between the vapor quality of the ejector outlet and the entrainment ratio. The solution converges when eqn (1) is satisfied to maintain a balance between liquid and vapor in the expansion recovery cycle:

$$x_{out,diff} = \frac{1}{1+ER} \tag{1}$$

First the properties at different states of the cycles are calculated. In EERC, according to Fig. 3, the specific enthalpy at gas cooler and evaporator exit (h_3, h_{10}) are defined. The motive flow expands to mixing pressure with a nozzle efficiency η_{pn} defined as:

$$\eta_{pn} = \frac{h_3 - h_4}{h_3 - h_{4s}} \tag{2}$$

where

$$\mathrm{h}_{4s} = \mathrm{f}\left(\mathrm{P}_{evap} - \Delta P, \mathrm{s}_3\right) \tag{3}$$

By applying energy conservation law between state 3 and state 4, the velocity at state 4 is obtained:

$$\frac{1}{2}u_4^2 = h_3 - h_4 \tag{4}$$

The velocity of secondary flow (u_5) is calculated in the same way as that of the primary flow and then the velocity of mixed flow is determined by the momentum equation in mixing chamber according to:

$$u_6\ \mathrm{m}_6 = \mathrm{m}_4 u_4 + \mathrm{m}_5 u_5 \tag{5}$$

The mixing efficiency is defined as [19] :

$$\eta_{mix} = \frac{\frac{1}{2}m_{4'}u_{4'}^2}{\frac{1}{2}m_4 u_4^2} \tag{6}$$

where $u_{4'}$ is the corrected velocity at state 4 which takes into account the mixing loss.

The energy conservation between two inlets and outlet of the ejector is as follows:

$$m_3 h_3 + m_{10} h_{10} = m_7 h_7 \tag{7}$$

The energy conservation between inlet and outlet of the diffuser is described as:

$$\frac{1}{2} u_6^2 + h_6 = h_5 \tag{8}$$

The diffuser efficiency is defined as:

$$\eta_{diff} = \frac{h_{7s} - h_6}{h_7 - h_6} \tag{9}$$

The pressure and quality at the ejector outlet (state 7) are obtained as

$$p_7 = f\left(s_6, h_{7s}\right) \tag{10}$$

$$x_7 = f\left(p_7, h_7\right) \tag{11}$$

This quality satisfies the eqn (1). The cooling capacity of the cycles (Figs. 1–4) can be written:

$$Q_{ev} = m_{evap}\left(h_{out,evap} - h_{in,evap}\right) = \frac{ER}{\left(1+ER\right)}\left(h_{out,evap} - h_{in,evap}\right) \tag{12}$$

The compressor power consumption is

$$W_{comp} = m_{comp}\left(h_{out,comp} - h_{in,comp}\right) = \frac{1}{\left(1+ER\right)}\left(h_{out,comp} - h_{in,comp}\right) \tag{13}$$

The gas cooler capacity is:

$$Q_{gc} = m_{gc}\left(h_{out,gc} - h_{in,gc}\right) = \frac{1}{\left(1+ER\right)}\left(h_{out,gc} - h_{in,gc}\right) \tag{14}$$

The cooling coefficient of performance (COP_c)for EERC and LRC is obtained using:

$$COP_C = \frac{Q_{evap}}{W_{comp}} \tag{15}$$

For the cycle including the pump (CDPC) the COP_c is defined as

$$COP_C = \frac{\mathrm{Q}_{evap}}{\mathrm{W}_{comp} + \mathrm{W}_{pump}} \tag{16}$$

For vapor jet refrigeration system, VJRC, it is expressed as

$$COP_C = \frac{\mathrm{Q}_{evap}}{\mathrm{Q}_{gen} + W_{pump}} \tag{17}$$

3.1.2 Exergy calculations

The exergy in all states is calculated based on the unit mass flow of mixing refrigerant in the ejector:

$$ex_k = m_k \cdot \left[\left(h_k \quad h_0 \right) - T_0 \cdot \left(s_k \quad s_0 \right) \right] \tag{18}$$

where m_k is the mass flow at the cycle state k.

The exergy destructions in the various processes are calculated as follows:

Compressor:

$$ex_{loss,comp} = ex_{in,comp} - ex_{out,comp} + W_{comp} \tag{19}$$

Gas cooler:

$$ex_{loss,gc} = ex_{in,gc} - ex_{out,gc} + Q_{gc} \cdot \left[1 - \left(\frac{T_0}{T_{sink}} \right) \right] \tag{20}$$

Ejector:

$$ex_{loss,ej} = ex_{in,pn} + ex_{in,sn} - ex_{out,diff} \tag{21}$$

Evaporator:

$$ex_{loss,evap} = ex_{in,evap} - ex_{out,evap} + Q_{evap} . \left[1 - \left(\frac{T_0}{T_{source}} \right) \right] \tag{22}$$

Throttling valve:

$$ex_{loss,th} = ex_{in,th} - ex_{out,th} \tag{23}$$

The total exergy destruction is calculated using:

$$ex_{loss,tot} = ex_{comp} + ex_{ej} + ex_{th} + ex_{evap} + ex_{gc} + ex_{gen} \left(VJRC \right) + ex_{cond} \left(VJRC \right) \tag{24}$$

The exergy efficiency of the cycles is evaluated as:

$$\eta_{ex} = 1 - \frac{ex_{loss,tot}}{W_{comp}} \tag{25}$$

For the cycle includes the pump, CDPC, exergy efficiency is as following:

$$\eta_{ex} = 1 - \frac{ex_{loss,tot}}{W_{comp} + W_{pump}} \tag{26}$$

For VJRC (Fig. 1), exergy efficiency and total exergy loss are calculated by:

$$\eta_{ex} = 1 - \frac{ex_{loss,tot}}{W_{pump} + ex_{Q,ge} + ex_{Q,cond}} \tag{27}$$

$$ex_{loss,tot} = ex_{pump} + ex_{gen} + ex_{cond} + ex_{ej} + ex_{th} + ex_{ev} \tag{28}$$

where $ex_{Q,gen}$ and $ex_{Q,cond}$ are exergy transfer by heat in generator and condenser respectively which are defined as:

$$ex_Q = Q . \left[1 - \left(\frac{T_0}{T} \right) \right] \tag{29}$$

4 RESULTS

The comparison of the results for four transcritical CO_2 refrigeration cycles is presented below. Exergy destructions and exergy efficiencies of the cycles are calculated under constant cooling capacity and corresponding parameters listed in Table 1.

As shown in Tables 2 and 3, the EERC has the highest COP and exergy efficiency. For the high-side pressure of 88 bar, the COP for EERC is 23.3%, 24.9% and 5.6 times higher than the COP for LRC, CDPC and VJRC, respectively.

It is also shown that EERC improves the COP by up to 23.1% compared to basic cycle without the ejector, while the COP of LRC and CDPC remains almost constant and for VJRC, the COP is very low. For given operating conditions, the pressure ratio of EERC is also the largest among other cycles.

Table 2: Comparison of the ejector's performance of the cycles.

Device	Ejector performance				
	EERC	LRC	CDPC	VJRC	BC
COP	3.618	2.935	2.896	0.6476	2.938
ER	0.564	0.641	1.558	0.921	-
P_{ratio}	1.15	1.03	1.01	1.11	-

Table 3: Exergy destructions and exergy efficiencies of the cycles (P_{gc} = 88 bar, T_{evap} = 5°C, Q_{evap}=72KW).

	Exergy Loss, Kw									
Device	EERC		LRC		CDPC		VJRC		BC	
	Loss, kW	%	Loss, kW	%	Loss, kW	%	Loss, kW	%	Loss, kW	%
Compressor	4.58	25.5	5.539	24.55	5.475	23.86	–	–	5.533	24.5
Gas cooler	2.817	15.68	3.515	15.55	1.679	7.317	–	–	3.511	15.55
Ejector	4.382	24.38	7.705	34.08	2.082	9.074	12.82	43.91	–	–
Valve	0.338	1.878	–	–	7.696	33.54	0.2198	0.753	7.696	34.07
Evaporator	5.854	32.57	5.847	25.86	5.847	25.48	5.808	19.89	5.847	25.88
Generator	–	–	–	–	–	–	4.931	16.89	–	–
Condenser	–	–	–	–	–	–	4.73	16.2	22.59	–
Pump	–	–	–	–	0.165	0.72	0.686	2.35	–	–
Total	17.97	100	22.61	100	22.94	100	29.2	100	24.51	100
W_{comp}	19.9	–	24.53	–	24.19	–	–	–	–	–
W_{pump}	–	–	–	–	0.668	–	2.558	–	–	–
$ex_{Q,evap}$	-1.919	–	-1.919	–	–	–	-1.919	–	-1.919	–
$ex_{Q,,gen}$	–	–	–	–	–	–	8.8	–	–	–
$ex_{Q,cond}$	–	–	–	–	–	–	19.76	–	–	–
Q_{gen}	–	–	–	–	–	–	108.6	–	–	–
η_{ex}	–	9.683	–	7.856	–	7.718	–	6.197	–	7.831

Table 3 shows the exergy losses in each component and exergy efficiency of all cycles. It can be noticed that EERC has the maximum exergy efficiency. The entrainment ratio and pressure ratio are 0.564 and 1.15 for EERC, respectively. The exergy loss in the evaporation process is the largest one in this system, while for LRC and VJRC, the largest loss occurs in the ejector.

The throttling exergy loss in the basic cycle is 7.69 KJ kg^{-1} that constitutes 34.07% of the total exergy loss. However, it is only 0.34 KJ kg^{-1}, 1.88% of the total exergy loss in EERC and the ejector's exergy loss is also 4.383 KJ kg^{-1}, 24.4% .The sum of these two losses is 26.28% of the total exergy loss of the system which is less than the throttling loss in the conventional cycle. The exergy loss in compressor and gas cooler are also reduced in EERC, and it is almost constant in the evaporator.

The use of liquid recirculation in refrigeration system improves the entrainment ratio compared to EERC; however, COP and exergy efficiency remain constant compared to conventional cycle. Therefore, despite a large amount of work that can be recovered with the CO_2 ejector, there is not the COP improvement for LRC.

The CDPC simulation shows that the ejector integration with this cycle is not efficient. It is due to the fact that the pressure lift is accomplished mainly by the compressor not the ejector. The pressor ratio is obtained 1.01 for this cycle.

The jet refrigeration cycle (single phase ejector, Fig. 1) achieves the lowest COP and the lowest exergy efficiency compared to other cycles. The low COP value around 0.65 is obtained. The high exergy losses of heat transfer in condenser (16.2%) and generator (16.9%) result in low exergy efficiency.

5 CONCLUSION

A comparative study based on the first and second laws of thermodynamics is performed for different transcritical CO_2 refrigeration cycles that use an ejector: EERC, LRC, CDPC and VJRC. The analysis for given conditions led to the following conclusions:

- Transcritical CO_2 refrigeration cycles, EERC has the highest COP and exergy efficiency. For the given operating conditions, it improves the COP and exergy efficiency by up to 23% and 24%, respectively, compared to the basic throttling cycle.
- In EERC, the irreversibility loss of the expansion process is significantly reduced compared to basic throttling valve cycle and as a result the exergy efficiency is increased.
- The COP of EERC is improved by up to 23.3%, 24.9% and 5.6 times compared to the LRC, CDPC and VJRC.
- The exergy loss in the evaporation process is the largest loss in EERC, whereas for LRC and VJRC the ejection process has the largest loss.
- The use of liquid recirculation improves entrainment ratio compared to EERC, however, COP and exergy efficiency decrease.
- CO_2 can gain more benefit from EERC compared to other cycles. CO_2 ejector liquid recirculation cycle and VJRC has a low potential for COP improvement.
- Ejector is not effective in the cycle for increasing compressor discharge pressure because the pressure lift is mainly accomplished by the compressor.

Nomenclature

Symbol/ Abbreviation		**Greek**	
BC	Basic cycle without ejector	η	isentropic efficiency
CDPC	Increasing compressor discharge pressure cycle	ρ	density, kg/m^3
COP	Coefficient of performance	**subscripts**	
EERC	Expansion work recovery cycle	0	dead state in the exergy analysis
ER	Entrainment ratio	C	cooling
ex	Exergy	comp	compressor
Ex_Q	Exergy transferred by heat	cond	condenser
h	Specific enthalpy, kJ/kg	diff	diffuser
LRC	Liquid recirculation cycle	ej	ejector
m	Mass flow rate, kg/s	evap	evaporator
P	Pressure, kPa	gc	gas cooler
P_{ratio}	Pressure ratio (pressure lift)	gen	Generator
Q	Heat rate, kW	in	Inlet
s	Specific entropy, kJ/kg K	mix	Mixing
T	Temperature, K	out	Outlet
u	Velocity, m/s	pn	primary nozzle
VJRC	Vapor jet refrigeration cycle	sn	secondary nozzle
W	Work rate, kW	th	Throttling
x	Quality	tot	Total
ΔP	Pressure drop		
η_{ex}	Exergy efficiency		

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REFERENCES

[1] Gay, N.H., Refrigerating system. Google Patents, 1931.

[2] Kornhauser, A.A., The use of an ejector as a refrigerant expander. *Proceedings of the 1990 USNC/IIR – Purdue Refrigeration Conference*, Purdue University, West Lafayette, IN, US, pp. 10–19, 1990.

[3] Li, D. & Groll, E.A., Transcritical CO_2 refrigeration cycle with ejector-expansion device. *International Journal Refrigeration*, **28**(5), pp. 766–773, 2005. https://doi.org/10.1016/j.ijrefrig.2004.10.008

[4] Fangtian, S. & Yitai, M., Thermodynamic analysis of transcritical CO_2 refrigeration cycle with an ejector. *Applied Thermal Enginnering,* **31**(6), pp. 1184–1189, 2011. https://doi.org/10.1016/j.applthermaleng.2010.12.018

[5] Elbel, S. & Hrnjak, P., Ejector refrigeration: an overview of historical and present developments with an emphasis on air-conditioning applications. *International Refrigeration and Air Conditioning Conference*, pp. 2350-1–2350-8, 2008.

[6] Phillips, H.A., Refrigeration system. Google Patents, 1938.

[7] Lorentzen, G., Throttling, the internal haemorrhage of the refrigeration process. *Proceedings of the Institute of Refrigeration*, **80**, pp. 39–47, 1983.

[8] Lawrence, N. & Elbel, S., Mathematical modeling and thermodynamic investigation of the use of two-phase ejectors for work recovery and liquid recirculation in refrigeration cycles. *International Journal Refrigeration*, **58**, pp. 41–52, 2015. https://doi.org/10.1016/j.ijrefrig.2015.06.004

[9] Bergander, M.J, *New regenerative cycle for vapor compression refrigeration, Final Scientific Report,* DOE Award DE-FG36-04GO14327. Madison, CT, USA, 2005.

[10] Bergander, M., Refrigeration cycle with two-phase condensing ejector, *International Refrigeration and Air Conditioning Conference*, Purdue, USA, pp. R008-1-R008-8, 2006.

[11] Elbel, S.W. & Hrnjak, P.S., Effect of internal heat exchanger on performance of transcritical CO_2 systems with ejector. *International Refrigeration and Air Conditioning Conference*, pp. R166-1–R166-8, 2004.

[12] Li, D., *Investigation of an ejector-expansion device in a transcritical carbon dioxide cycle for military ECU applications* [Ph. D. Dissertation]. Purdue University, West Lafayette, IN, 2006.

[13] Nehdi, E., Kairouani, L. & Bouzaina, M., Performance analysis of the vapour compression cycle using ejector as an expander. *International Journal Energy Research*, **31**(4), pp. 364–375, 2007. https://doi.org/10.1002/er.1260

[14] Sarkar, J., Optimization of ejector-expansion transcritical CO_2 heat pump cycle. *Energy,* **33**(9), pp. 1399–1406, 2008. https://doi.org/10.1016/j.energy.2008.04.007

[15] Ameur, K., Aidoun, Z. & Ouzzane, M., Modeling and numerical approach for the design and operation of two-phase ejectors. *Applied Thermal Engineering,* **109**, pp. 809–818, 2016. https://doi.org/10.1016/j.applthermaleng.2014.11.022

[16] Smolka, J., Bulinski, Z., Fic, F., Nowak, A.J., Banasiak, K. & Hafner, A., A computational model of a transcritical R744 ejector based on a homogeneous real fluid approach. *Applied Mathematical Modelling,* **37**(3), pp. 1208–1224, 2013. https://doi.org/10.1016/j.apm.2012.03.044

[17] Khalil, A., Fatouh, M. & Elgendy, E., Ejector design and theoretical study of R134a ejector refrigeration cycle. *International Journal Refrigeration*, **34**(7), pp. 1684–1698, 2011. https://doi.org/10.1016/j.apm.2012.03.044

[18] Selvaraju, A. & Mani, A., Analysis of an ejector with environment friendly refrigerants. *Applied Thermal Engineering*, **24**(5–6), pp. 827–838, 2004. https://doi.org/10.1016/j.applthermaleng.2003.08.016

[19] He, Y., Deng, J. & Zhang, Z., Thermodynamic study on a new transcritical CO_2 ejector expansion refrigeration system with two-stage evaporation and vapor feedback. *HVACR Research*, **20**(6), pp. 655–664, 2014. https://doi.org/10.1080/10789669.2014.929422