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Thermodynamic Analysis of a Multi-Ejector, CO_2, Air-To-Water Heat Pump System

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Abstract

Nowadays, air conditioning systems for residential and office buildings, contribute largely to the energy consumptions and to the direct and indirect emissions of greenhouse gases. Carbon dioxide (CO_2) could be an interesting option to replace traditional HFCs in space heating applications, due to its environmentally friendly characteristics: zero ODP and extremely low GWP, but, in order to spread its use, improvements in performances are needed. In fact, CO_2 requires transcritical cycles with high expansion losses. The use of an ejector can reduce these losses and improve the performances up to 30% (depending on the performances of the ejector itself and on the operating conditions). In the a/c applications, characterized by variable operating conditions, multi-ejector systems could be used, where some ejectors work in parallel, in different combination, varying the operating conditions. Currently, a project of DTE-PCU-SPCT Department of ENEA and Industrial Engineering Department of Federico II University of Naples, is in progress, in order to evaluate experimentally the effect of several ejectors geometries on the global performance of a CO_2 heat pump working with a transcritical cycle. As a part of this project, a complete heat pump system for production of hot water for sanitary use and for space heating is tested to investigate the effect of the ejector size on the balancing of the global performance of the whole system.

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Keywords: carbon dioxide, ejector, transcritical cycle, heat pump, space heating

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1. Introduction

The latest F-Gas Regulation [1] (and the new proposed amendment to the Montreal Protocol) has imposed more and more strict limits on the consumption of HFC refrigerants, attaining a limit in 2030 equal to 21% of the current equivalent CO_2 emissions. Starting from 2025, the GWP limit will be 750 in the residential air conditioning sector. As a consequence, the most common refrigerants used, such as R410A and R134a, characterized by high value of GWP (2088 and 1430 respectively), will be banned. Nowadays, the possible alternatives are the fluids with low GWP: the fluids based on R32, the new synthetic fluids, such as HFO fluids, and the natural fluids. The first two groups give good performance, but their use is limited by their slight flammability (ASHRAE A2L safety classification). Among the natural fluids, it is possible taking into account the propane and CO_2. Several studies as those of Granryd [2] and Palm [3] show that the propane has similar or better performance compared to the traditional HFC refrigerants. However, the limit of the propane is represented by its high flammability (ASHRAE A3 safety classification). The use of CO_2 as refrigerant in the sanitary hot water production is well known, as shown by Neksa et al. [4]. In particular, in case of water heating starting from low temperature, CO_2 shows high performance. Vice versa, the performance is not as good when CO_2 is used in air conditioning systems, as shown by Calabrese et al. [5]. Lower performances are due to the throttle losses in the expansion process. In particular, higher water inlet temperature at gas cooler leads to higher exergetic throttling losses. The use of an ejector system as substitute of a common expansion valve can improve the performance of the overall system. Minetto et al. [6], through a numerical simulation, pointed out that only the use of an ejector system can lead to a carbon dioxide heat pump performance comparable to that of R410A. Elbel et al. [7] and Lucas et al. [8] confirm that the use of the ejector improves the COP and exergy efficiency up to 17% in transcritical refrigeration cycle. However, in air conditioning systems, where the boundary conditions are variable, the use of an ejector as an expansion device does not lead necessarily to better performance. In this case there is the need to have several ejectors, conveniently sized, working in parallel, able to adjust, for each operative condition, the appropriate ejector, improving the system performance, as exposed by Banasiak et al. [9]. The multi-ejector system is widely used in transcritical CO_2 refrigeration cycles, with COP improvements up to 7% as confirmed by experimental study of Haida et al. [10]. However, multi-ejector systems are barely diffused in air conditioning applications. Moreover, Lawrence and Elbel [11] underline that the performance of an ejector cycle is strictly related to the performance of the ejector. In the present work, a multi-ejector heat pump system was tested using CO_2 as refrigerant. A thermodynamic and exergetic analysis, with varying the ejector system configuration characterized by different geometries, was run to investigate the effect of the ejector size on the balancing of the whole system and consequently on its global performances.

2. Experimental setup

2.1. Test facility

The tests were run with the experimental facility "Calorimetro Enea" at ENEA (Casaccia) research center. The climatic chamber allows to test air-to-water reversible heat pump, with thermal capacity until 50 kW, according to UNI EN 14511/2011 [12]. It is possible to measure and control the air conditions in terms of temperature, relative humidity and the air speed in the climatic chamber under 1 m/s. Therefore, the stability of the test conditions and the measure's accuracy within the limits are ensured. It is possible to control the air temperature in a wide range (-15°C/+35°C) and the relative humidity between 10 to 95%. Furthermore, it is possible to set the water mass flow rate at the gas cooler. The prototype is an air-to-water reversible carbon dioxide heat pump with heating capacity of 30 kW at nominal conditions (water inlet temperature of 40°C, water outlet temperature of 60°C and air temperature of 7°C). The multi-ejector CO_2 system consists of an alternative semi-hermetic compressor (CP) driven by an inverter, a plate heat exchanger (GC), a finned coil (EVAP), a plate internal heat exchanger (IHE), an electronic valve (EEV) and a multi-ejector expansion pack (EJEC), including four different ejector geometries, with throat diameters from 0.7 mm to 2.0 mm (Fig. 1). The heat pump control system can activate each ejector independently, depending on boundary conditions, with 15 different configurations. K-type and J-type thermocouples and pressure sensors are allocated at the inlet and outlet of each component. The water volumetric flow rate is measured by an electromagnetic transmitter. The electrical power is measured by a wattmeter. All measurement instruments are characterized by high accuracy, according to UNI EN 14511 [13]. Table 1 summarizes the instrument specifications and their uncertainty, as indicated by the manufactures.

2.2. Experimental procedure

All tests were run according to UNI EN 14511, part 3 [12]. During the tests, it was possible to set the outlet evaporator and inlet compressor superheating. Furthermore, it was possible to set manually the chosen ejectors configuration. The trends of the thermodynamic measured variables were monitored via a software designed specifically. The main thermodynamic parameters were recorded and processed using Matlab software [14]. The instabilities generated by the control system regulation are limited in according to UNI EN 14511, part 4 [13].

2.3. Data reduction

The following quantities have been calculated from the measured parameters. The heating capacity $\dot{Q}_{GC}$ (kW) of the ejector CO_2 system and the primary mass flow rate was evaluated as follows:

$$\dot{Q}_{GC} = \rho_W \dot{V}_W c_{p,W} (T_{W,\mathrm{OUT}} - T_{W,IN}) \tag{1}$$

$$\dot{m}_{PF} = \frac{\dot{Q}_{GC}}{h_{IN,GC} - h_{\mathrm{OUT},GC}} \tag{2}$$

where $\dot{V}_W$ is the water volumetric flow rate in m^3/s; ρ_W, $c_{p,\mathrm{W}}$, $T_{W,OUT}$ and $T_{W,IN}$ are the density (kg/m^3), the specific heat at constant pressure (kJ/(kgK)), the gas cooler outlet and inlet temperatures (°C) of water respectively; $h_{IN,GC}$ and $h_{OUT,GC}$ are the CO_2 gas cooler inlet and outlet enthalpy in kJ/kg, calculated via Refprop 9.1 [15].

The coefficient of performance was evaluated by Eq. (3):

$$COP = \frac{\dot{Q}_{GC}}{\dot{L}_{HP}} \tag{3}$$

where $\dot{L}_{HP}$ is the electrical power (kW), including the power required by the compressor and auxiliary components. The ejector entrainment ratio was evaluated by Eq. (4), where $\dot{m}_{SF}$ (kg/s) is the secondary mass flow rate, evaluated by an energy balance on the evaporator:

$$\mu = \frac{\dot{m}_{SF}}{\dot{m}_{PF}} \tag{4}$$

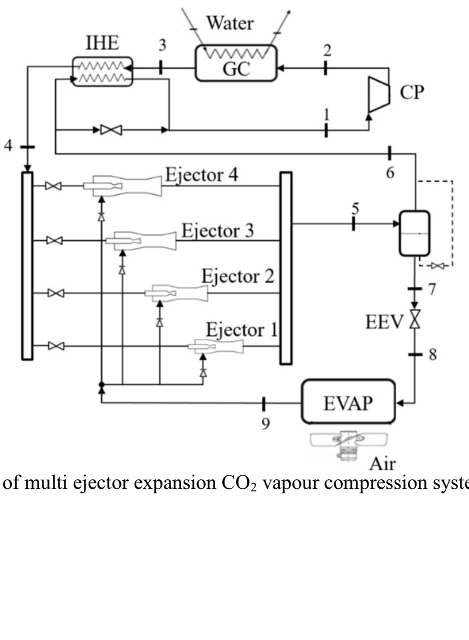

Fig. 1. Layout of multi ejector expansion CO_2 vapour compression system test setup.

Table 1. Measurement instrument and calibrated uncertainties.

Measurement	Range/Unit	Uncertainty
Temperature (K-type)	0/150 °C	± 0.4 % of reading
Temperature (J-type)	-40/80 °C	± 0.4 % of reading
Pressure	0/60 - 0/100 - 0/160 bar	0.08%
Water volumetric flow rate	0/200 l/min	0.02% of reading
Electrical power	0/25 kW	Precision class 0.5

3. Experimental results

3.1. Variation of performance with ejector geometry

For fixed boundary conditions, it has been analyzed the influence of the ejector's internal geometry on the main thermodynamic and performance parameters of the heat pump. During the test, the combination of active ejectors has been varied. In Fig. 2 are shown the *T-s* and *p-h* diagrams: it has been taken into account a test characterized by a water temperature at the gas cooler inlet of 40 °C, a water temperature at the gas cooler outlet of 60 °C and an air temperature of 7 °C. The inverter has been set to 50 Hz. The test has been performed varying gradually the active ejectors combination and therefore, the overall cross section available to the fluid. Multi-ejector system affects the thermodynamic cycle: it occurs an approach of the pressure levels that characterizes the cycle as the passage section increases. Thus, the compressor runs under lower compression ratio, β, as shown in Fig. 3. Furthermore, the ejector allows a pressure recovery compared to the evaporation pressure due to the higher mass flow rate; in opposition to this effect, the system has a lesser ability to entrain the secondary flow rate (lower entrainment ratio), as can be seen in Fig. 3. This result is consistent with the experimental results shown in [8]: lower values of the entrainment ratio correspond to higher pressure lifts and vice versa. With increasing ejector cross section, the heating capacity reaches a maximum due to two opposing effects: the increase of the primary mass flow rate and the decrease of the gas cooler enthalpy variations, because of the decrease of the gas cooler pressure and the outlet compressor temperature. In terms of overall performance, there is an overall ejectors cross section which determines an increase of the COP up to 10% compared to the worst case. However, the ejector efficiency, in according to the method proposed by Elbel et al. [7], reaches a maximum in an ejectors' configuration different from the previous one (Fig. 4.). The results are in line with the results shown by Liu et al. [16], which have experimentally investigated the influence of the ejector's internal geometry on the CO_2 heat pump performances. Their experimental results show that the heating capacity and the overall system COP reach a maximum in correspondence of the motive nozzle throat diameter of 2 mm, in the operating condition investigated.

3.2. Exergetic analysis

In order to analyze the influence of the individual processes on the heat pump performances, an exergetic analysis has been carried-out. The climatic chamber temperature has been chosen as the reference temperature. The following equations express the thermodynamic losses, π, referred to a unit primary mass flow rate of refrigerant.

$$\pi_{CP} = T_a(s_2 - s_1) \tag{5}$$

$$\pi_{GC} = (\mathrm{h}_2 - h_3) - T_a(s_2 - s_3) - (\mathrm{h}_2 - h_3)\tau_{GC} \tag{6}$$

$$\pi_{EV} = \mu\left[T_a(s_9 - s_8) - (\mathrm{h}_9 - h_8) + (\mathrm{h}_9 - h_8)\left|\tau_{EV}\right|\right] \tag{7}$$

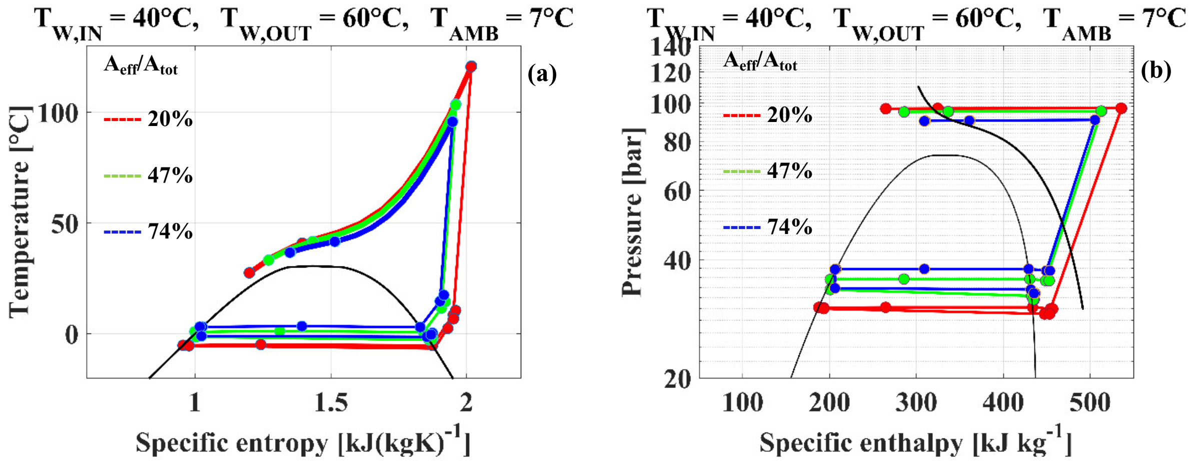

Fig. 2. (a) T-s diagram and (b) p-h diagram, variation of the thermodynamic cycle varying the overall ejectors cross section for fixed boundary conditions, represented on the top of the figure.

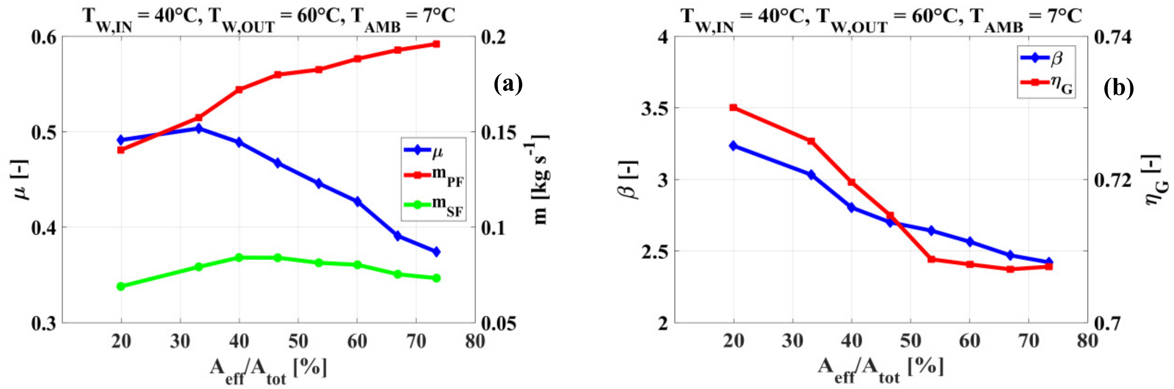

Fig. 3. Performance variations as a function of the overall ejectors cross section. 3.a) entrainment ratio (left) and mass flow rates (right); 3.b) pressure ratio (left) and compressor global efficiency (right).

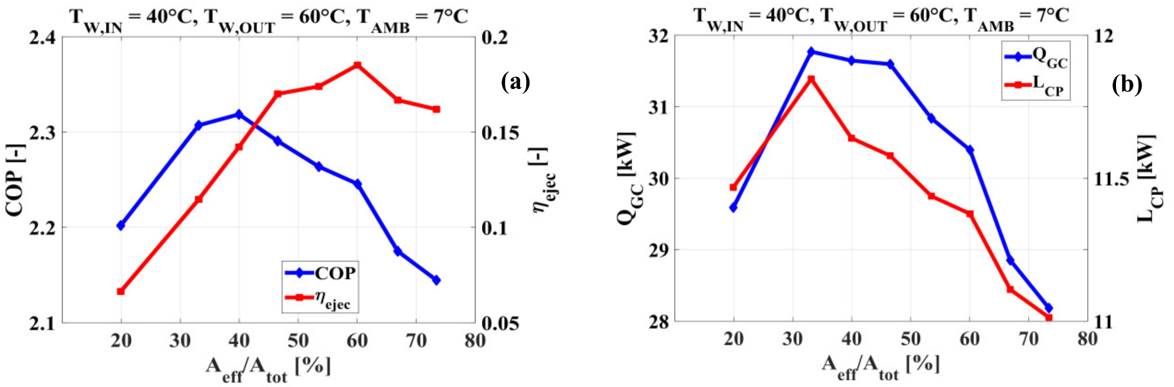

Fig. 4. Performance variation as a function of the overall ejectors cross section. 4.a) COP (left) and ejector efficiency (right); 4.b) heating capacity (left) and compression work (right).

$$\pi_{IHE} = (h_3 - h_4) - T_a(s_3 - s_4) - (h_1 - h_6) + T_a(s_1 - s_6) \quad (8)$$

$$\pi_{EJEC} = (h_4 - h_5) - T_a(s_4 - s_5) + \mu[(h_9 - h_5) - T_a(s_9 - s_5)] \quad (9)$$

$$Q_0 = h_2 - h_3 \quad (10)$$

$$E_{Q0} = |Q_0|\left(1 - \frac{T_a}{T_{set,w}}\right) \quad (11)$$

where:

$$\tau_{GC} = 1 - \frac{T_a}{T_{set,w}} \quad (12a), \qquad \tau_{EV} = 1 - \frac{T_a}{T_{set,a}} \quad (12b)$$

In the above equation, h and s are respectively, specific enthalpy, in kJ/kg, and specific entropy, in kJ/(kgK), of the refrigerant evaluated through the experimental measurements using Refprop 9.1 [15], T is the temperature. $T_{set,a}$ and $T_{set,W}$ are the equivalent temperatures of the ambient and the water, respectively. Q_0 is the desired output of the cycle; E_{Q0} is the exergy associated to Q_0, equal to the minimum compression work required in an ideal cycle. The actual compression work required (eq. 13) can be calculated as the minimum compression work plus the sum of the individual process losses. All the terms in Eq. (13) can be represented as an area in the *T-s* diagram, as shown in Fig. 5.

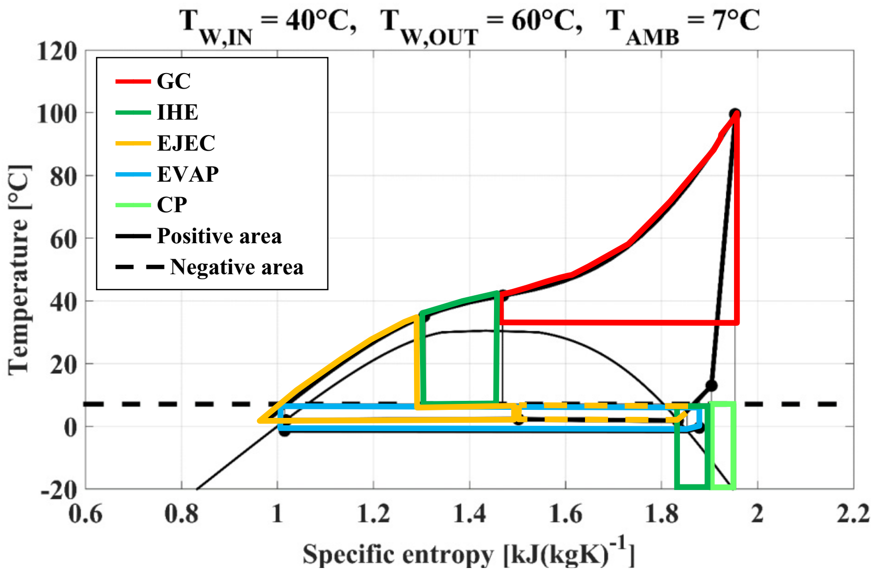

Fig. 5. Exergy losses π in the T-s diagram for transcritical CO_2 multi-ejector cycle.

$$|W_c| = E_{Q0} + \pi_{CP} + \pi_{GC} + \pi_{EV} + \pi_{EJEC} + \pi_{IHE} \quad (13)$$

Normalizing the individual losses to the required work $|W_c|$, it is possible to evaluate the efficiency defects δ (eq. 14), where *i* represents the main heat pump components.

Therefore, it is possible to evaluate the exergetic efficiency with the equation (15):

$$\delta_i = \pi_i / |W_c| \quad (14)$$

$$\eta_{ex} = 1 - \delta_{CP} - \delta_{GC} - \delta_{EV} - \delta_{EJEC} - \delta_{IHE} = 1 - \sum \delta \quad (15)$$

The efficiency defects estimate the influence of each single process on the degradation efficiency of the thermodynamic cycle. In order to have a comparison with standard technology currently on the market, it has been performed a comparison between the experimental multi-ejector cycle with an expansion valve cycle and an internal heat exchanger cycle, at the same heating capacity, evaporating pressure and compressor efficiency. The comparison results are shown in Fig. 6: in this case it has been chosen the multi ejector configuration that maximizes the COP. The same analysis has been performed in the multi-ejector configuration that maximizes the ejector efficiency. The results shown in Fig. 7, are slightly better than the previous case. In both cases, the use of the ejector system as expansion device reduces the throttling losses to 46% but the other efficiency defects are equally distributed throughout the whole system. Furthermore, in terms of overall exergetic efficiency, the improvement is not so high. In the best case, the improvement is equal to 9% compared to a basic cycle. This is due to the difficulty to optimize the ejector's performances when COP reaches the maximum value. Overall, it has been evaluated an ejector efficiency up to 18%, lower than the values published in scientific literature. Higher ejector's performances, resulting from an optimization sizing, can lead to better overall performances compared to those of a conventional cycle.

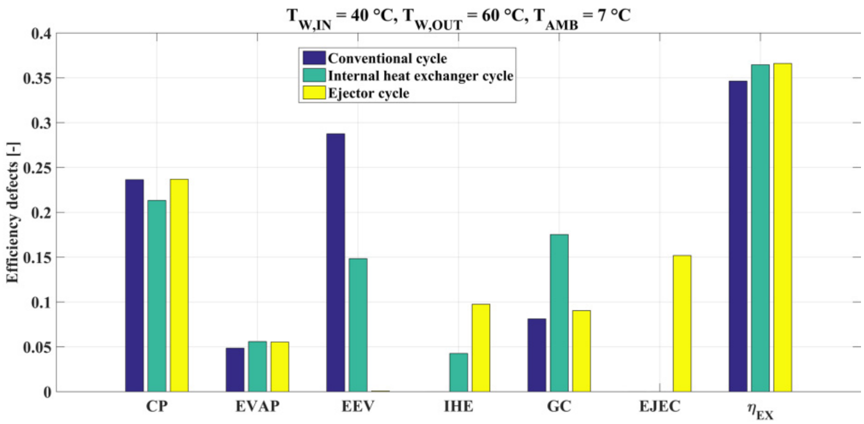

Fig. 6. Exergy losses comparison between ejector cycle, expansion valve cycle and internal heat exchanger, in the maximum COP configuration.

Conclusion

Carbon dioxide is an interesting option to replace traditional HFCs in vapor compression systems, due to its environmentally friendly characteristics: zero ODP and extremely low GWP. In order to optimize the use of the multi-ejector system with natural refrigerant CO_2, experimental tests on a complete heat pump system with multi-ejector pack and internal heat exchanger for production of hot water for sanitary use and for space heating have been carried-out. In the investigated conditions, this experimental analysis shows that it is possible to find an optimal multi-ejector configuration that leads a maximum in terms of COP, heating capacity and ejector efficiency. The comparison of the present multi-ejector system performance, at the optimal configuration, to a basic and a regenerated cycle with a throttling valve (imposing the same evaporating pressure and efficiency of the compressor) has been performed. Although the present multi-ejector system is designed for refrigeration systems (which require higher pressure difference compared to applications for air-conditioning systems), its exergetic efficiency is similar or slightly better compared to a cycle with the regenerator. The increment of performance due to the ejector is reduced since the actual balancing point of the system led to an operating condition where the efficiency of the ejector is marginal with respect to the nominal efficiency. Therefore, further studies on ejector systems specifically designed for air-conditioning applications are necessary.

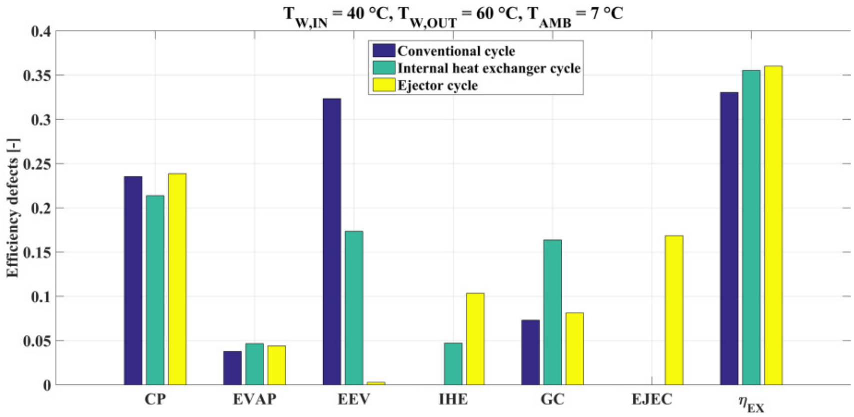

Fig. 7. Exergy losses comparison between ejector cycle, expansion valve cycle and internal heat exchanger, in the maximum ejector efficiency configuration.

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