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Energy Procedia 161 (2019) 309–316

Energy Procedia

www.elsevier.com/locate/procedia

2nd International Conference on Sustainable Energy and Resource Use in Food Chains, ICSEF 2018, 17-19 October 2019, Paphos, Cyprus

Experimental investigations on a transcritical CO_2 refrigeration plant and theoretical comparison with an ejector-based one

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Abstract

The paper presents the development of an experimental research facility based on a transcritical single stage vapour compression refrigeration cycle with CO_2 as the working fluid. The experimental setup includes instrumentation and controls which enable tests in a broad range of operating conditions. The measurements presented refer to external temperatures between 21.0°C and 33.5°C; in this latter operating point, energy and exergy analysis allowed the breakdown of irreversibilities. In particular, the gas cooler contributes to the 42.6% of the total exergy losses while the share due to the high-pressure expansion valve is 27.2%. In order to improve the performance of the refrigeration system, a theoretical model was developed including an ejector as the replacement of the expansion valve. The results show that the Coefficient of Performance (COP) is strongly dependent on the entrainment ratio; a value of greater than 0.6 seem to lead to higher COP values, even at low external temperatures.

Keywords: transcritical CO_2 refrigeration; energy saving; ejector;

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10.1016/j.egypro.2019.02.097

Nomenclature

h	enthalpy [J/kg]	comp	compressor	*COP*	Coefficient of Performance
ṁ	mass flow rate [kg/s]	ej	ejector	*EEV*	Electronic Expansion Valve
p	pressure [Pa]	evap	evaporator	*FG*	Flash Gas
s	entropy [J/(kg K)]	ext	external	*ORC*	Organic Rankine Cycle
I	irreversibility [J/kg]	in	inlet	*TFC*	Trilateral Flash Cycle
T	temperature [J/kg]	isent	isentropic		
W	work [J/kg]	out	outlet		
X	quality	ref	reference		
η	efficiency	DC	dry cooler		
μ	entrainment ratio	GC	gas cooler		

1. Introduction

The global extent of refrigeration, air conditioning and heat pump systems involves around 3 billion units and creates jobs for almost 12 million employees [1]. Besides, the impact on electricity consumptions due to the refrigeration sector (including air conditioning) accounts for about 17.2% of the global amount, namely 295 Mtoe [1, 2]. With regards to CO_2 emissions, the indirect ones (due to electricity consumption) are equal to 1680 $MtCO_2$ while the direct ones (due to leaks of fluorocarbon refrigerants) are roughly the 25% of indirect emissions [3]; this leads to a total carbon footprint of 2.1 $GtCO_2$, 6.5% of the global figures [2].

Although the consumption of a typical household refrigerator has dropped by around 65% within 15 years [4], recent statistics also forecast a 70% increase in the cooling needs of the European building sector by 2030 [5]. Therefore, in order to address the growing demand of cold energy and the need for lowering the energy and carbon impacts of household and industrial refrigeration systems, industry and academia have been lately investigating multiple energy saving and recovery approaches. Among them, heat recovery could lead to 4% energy savings while the optimisation of compressor set points and expansion valve might contribute to 15% and 10% energy saving potentials respectively [6]. Indeed, exergy analysis on refrigeration plants showed that the major factor responsible for exergy destruction are compressors and condenser/gas coolers [7, 8]; while the compressor efficiency resides mostly in its technology [9, 10], condenser behaviour is strictly related to the hot source and medium-low temperatures [11]. In this regard, the possibility to recover the thermal power from a low temperature stream (such as the cold source of an industrial refrigeration plant) has been widely studied in literature, in particular making use of thermodynamic cycles, like Organic Rankine Cycles (ORC) [12, 13] or Trilateral Flash Cycles (TFC) [14].

Another promising area of improvement would be the replacement of the throttling valve, which can currently waste up to 20% of the compressor input [15], with a direct expander or an ejector. According to [16, 17], the integration of an expander in a commercial refrigeration system become economically feasible if the expander has an efficiency greater than 50%. On the other hand, theoretical studies claim an increase of 20% in the Coefficient Of Performance (COP) by using ejectors, although only few of the experimental research works could achieve improvements over 10% [18].

In particular, transcritical refrigeration plants using CO_2 (R744) as working fluid suffer from operational and cost limits, which prevent their application in warm locations. Nonetheless, these constrains might be overcome through energy saving solutions [19]. The use of ejector-based layouts seem to have great potential: indeed, keeping the same cooling capacity on the evaporator, the ejector permits to have a lower mass flow rate flowing in the compressor, lower compression ratio and, definitively, lower energy consumption and higher COP [18, 20].

In this paper, the authors propose a preliminary assessment of the energy saving potentials achievable with the introduction of an ejector in the layout of the plant, instead of the high-pressure isenthalpic valve. The novelty with respect to the state of the art lies in the measurement campaign which provided the baseline for the retrofit study; the transcritical CO_2 refrigeration system is in fact an actual industrial unit but with high accuracy instrumentation and customised controls. After the description of the experimental facility, the comparison of the two different technologies is made and some techno-energetic conclusions are eventually drawn.

2. Experimental Setup

The developed transcritical CO_2 refrigeration facility is displayed in Figure 1. Rated power is 18kW thermal and 12kW electric. Starting from the receiver the two-phase refrigerant (mix of liquid and vapour) at point (7) is split into saturated liquid (1) and saturated gas (1b). The latter is bypassed by a flash gas valve, and the former flows into expansion valves where the refrigerant pressure drops to the minimum (2) pressure level. The electric expansion valve (EEV) is driven by hysteresis ON/OFF controller to regulate the temperature of the fridge display cases. Flowing through evaporators, the refrigerant absorbs heat from the cold reservoir while a superheat controller acts on the valve. This is to make sure the refrigerant leaving the evaporators toward compressors is completely vaporised (only in vapour phase). All mass flows at the outlet of evaporator and flash gas valve are collected by the suction manifold at point (4) where the pressure and enthalpy are increased to the highest point (5) by the compressors. Afterwards, the gas phase refrigerant enters the condenser to deliver the absorbed heat from cold reservoirs to the surrounding where its enthalpy decreased significantly from (5) to (6) followed by a small pressure drop. At the outlet of the high-pressure control valve (7) (ICMT), the pressure drops to an intermediate level and the refrigerant, which is now in two-phase, flows into the receiver and the cycle is completed.

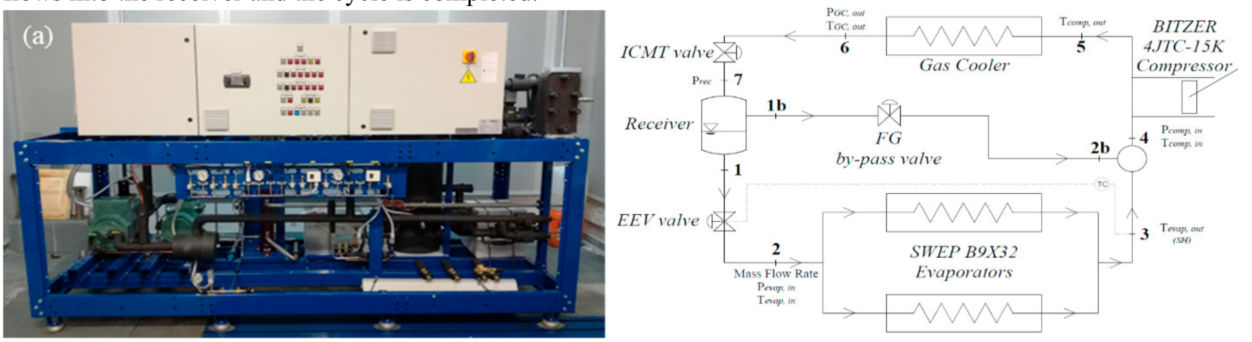

Fig. 1. (a) Picture and (b) scheme of the transcritical CO_2 refrigeration facility at the University of L'Aquila

With reference to Figure 2, from a thermodynamic point of view, the facility operates between two thermal sources: the cold one represents the cooling demand of the plant and it is fed by the evaporation section; the second one is needed to supply the thermal power at high temperature to the environment and it is realised by a cooling of the working fluid (which, in certain thermodynamic conditions, could also condensate) due to external air usually forced by GC fans.

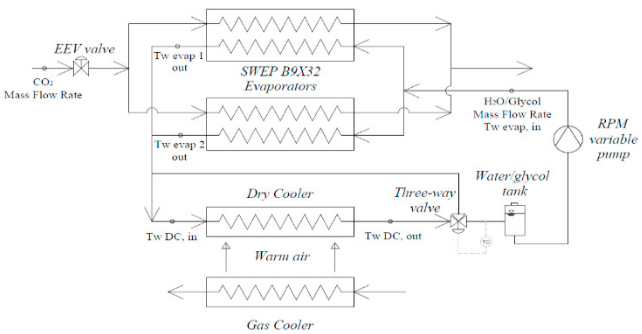

Fig. 2. Ancillary air and water/glycol loops

Table 1. Instrumentation and accuracy.

n°	Instrument	Accuracy
5	PT1000 RTDs	0.1 K
1	K-thermocouple	2.2 K
1	FoxBoro IMT25 Mass Flow (Water/glycol)	0.25%
1	SITRANS FC MASS 6000 IP67 (CO2)	10% max

Due to the transcritical nature of the plant, the hot side operates at very high-pressure levels (up to 120 bar), while the low-pressure side is related to the cold temperature demanded by the user. To simulate the cooling load at the evaporators, a water/glycol (30%) circuit was developed. The temperature in the cold source circuit is controlled by a three-way mechanically valve with thermal probe, which opens a branch towards the heat sink, namely a dry-cooler. This device may use the heat coming from the gas cooler to warm the cold loop and to restore the desired temperature in the evaporators. The dry-cooler is placed above the gas cooler so that its wasted heat is used to balance the energy flows in the water/glycol circuit. The instrumentation to characterize the water/glycol loop is shown in Figure 2. The data acquisition system is based on BOSS/Carel and DAQ/Agilent devices that acquire signals from the transducers listed in Table 1.

3. Analysis of Measured Data

The aim of this experimental activity summarised in Figures 3 and 4 was to analyse the plant at operating conditions which usually lead to low efficiency of a CO_2 refrigeration unit, namely with medium-high external temperatures. The trends reported in Figures 3 and 4 can be justified as follows: with higher external temperature, the plant requires higher operating pressures to realise the transcritical cycle, so that the compressor needs more power with a consequent decrease in the COP. If gas cooler outlet temperature increases, in hotter days, the receiver will have a higher quality (X_7) with a worsening of the COP.

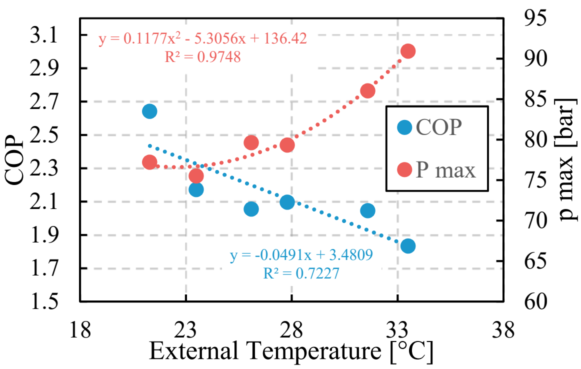

Fig. 3. COP and p max as function of external temperature

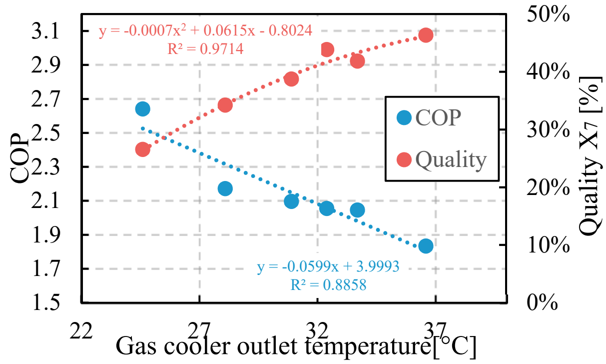

Fig. 4. COP and Quality as function of gas cooler outlet temperature

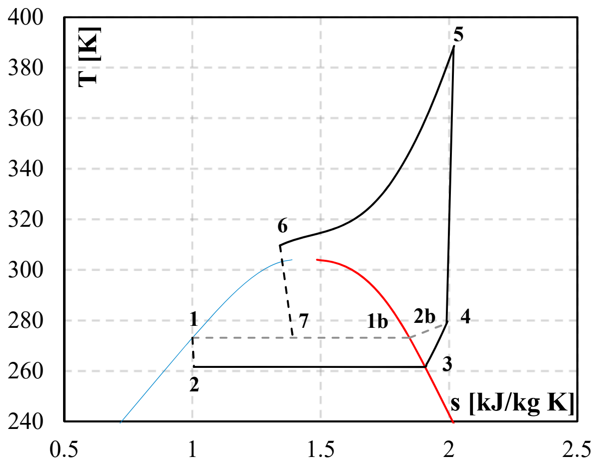

Fig. 5. Transcritical CO_2 cycle at T_{ext} = 33.5 °C

Table 2. Experimental data for the test case shown in Fig. 5

External temperature	33.5	°C
CO_2 Mass Flow Rate	0.10	kg/s
Discharge temperature	115.0	°C
Gas cooler outlet temperature	36.5	°C
Gas cooler pressure	90.9	bar
Suction pressure	25.3	bar
Suction temperature	5.6	°C
Superheat	15.7	°C
Receiver pressure	34.8	bar
Evapourating pressure	25.3	bar
Evapourating temperature	-10.2	°C
Gas cooler thermal power	28.0	kW
CO_2 critical pressure	73.8	bar
CO_2 critical temperature	31.1	°C

Figure 5 displays the Temperature entropy diagram for the test with external temperature at 33.5°C. The numbering is in agreement with Figure 1.b and refers to the measurement data listed in Table 2. In this operating point, the COP was equal to 1.83, calculated according to Eqn. (1) and related to the vapour quality of the CO_2 at point 7. This quantity has been calculated according to Eqn. (2), i.e. the energy balance at the flash gas receiver. The test case of Table 2 has been therefore chosen as the baseline to design the retrofit configuration based on ejector as the replacement of the high pressure expansion valve.

$$COP = (1 - X_7)\frac{(h_4 - h_2)}{(h_5 - h_4)} \quad (1) \qquad \frac{\dot{m}_{evap}}{\dot{m}_{comp}} = \frac{(h_7 - h_{1b})}{(h_1 - h_{1b})} = (1 - X_7) \quad (2)$$

In the reference test case, measured mass flow rates are 0.10 kg/s for CO_2 and 0.78 kg/s for the water/glycol mixture. More specifically, CO_2 mass flow rate is 0.055 kg/s at the evaporator and 0.045 kg/s at the flash gas by-pass circuit. Air mass flow of the gas cooler has been derived from technical data and it is eventually equal to 2.43 kg/s. This information allowed to reconstruct the temperature-heat diagrams for the gas cooler (Fig. 6a) and at the evaporators (Fig. 6b). The resulting thermal power exchanged is 23.30 kW at the gas cooler and 14.27 kW at the evaporators.

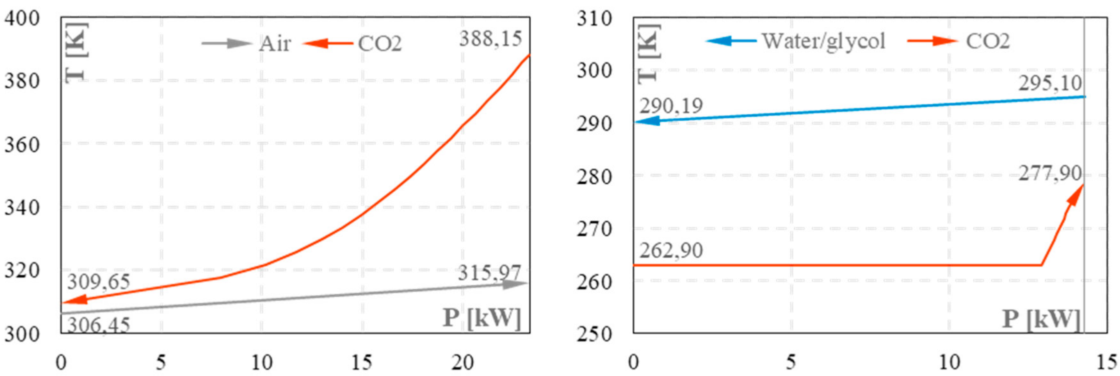

Fig. 6. (a) Gas cooler, CO_2-air heat exchange diagram; (b) evaporators, CO_2-water/glycol heat exchange diagram.

3.1. Exergetic analysis

With reference to the set of equations proposed in [21] and reported in Eqn. (3), an exergetic analysis was carried out for the operating point of Table 2. This allowed to evaluate the exergy performance and the resulting improvement potential in each component as well as for the whole plant.

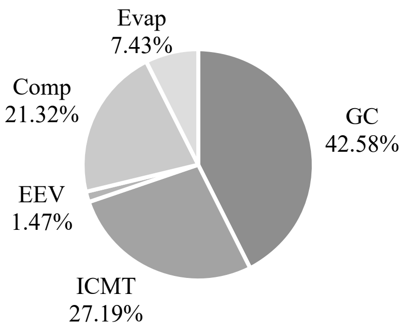

Fig. 7. Share of irreversibilities of each component of the CO_2 refrigeration tested plant

$$\begin{cases} \dot{I}_{comp} = W_{comp} - \dot{m}_{comp}\left[h_5 - h_4 - T_{ref}(s_5 - s_4)\right] \\ \dot{I}_{evap} = T_{ref}\left[\dot{m}_{evap}(s_4 - s_2) + \dot{m}_{water}(s_{water,out} - s_{water,in})\right] \\ \dot{I}_{GC} = T_{ref}\left[\dot{m}_{comp}(s_6 - s_5) + \dot{m}_{air,GC}(s_{air,out} - s_{air,in})\right] \\ \dot{I}_{EEV} = \dot{m}_{evap}\left[h_1 - h_2 - T_{ref}(s_1 - s_2)\right] \\ \dot{I}_{ICMT} = \dot{m}_{comp}\left[h_6 - h_7 - T_{ref}(s_6 - s_7)\right] \\ \dot{I}_{ej} = \dot{m}_{comp}\left[h_{6'} - h_{8'} - T_{ref}(s_{6'} - s_{8'})\right] + \\ +\dot{m}_{evap}\left[h_{3'} - h_{8'} - T_{ref}(s_{3'} - s_{8'})\right] \end{cases} \quad (3)$$

The breakdown of exergy losses is shown in Fig. 7: more than 50% of irreversibilities lie in the two heat exchangers (Gas cooler – GC – and evaporator), due to the mismatch of the temperature profiles of the two heat streams. Almost 21% is related to the compressor efficiency, while, very significantly exergy destruction is realised in the ICMT – high pressure expansion valve; negligible values are obtained for the EEV – low pressure expansion valve. The overall exergy efficiency is 29.40%.

4. Ejector-based CO_2 refrigeration plant

A significant amount of exergy losses of the tested CO_2 refrigeration plant take place between the top and intermediate pressure levels and are related to the isenthalpic expansion realised by the high-pressure expansion valve. This situation can be overcome by using a layout in which the afore mentioned expansion is done through an ejector. In fact, unlike the ICMT valve, the ejector is able to recover the enthalpy drop. Hence, a theoretical analysis has been performed, reconstructing the thermodynamic cycle of a CO_2 refrigeration plant equipped with an ejector, evaluating its performances and comparing it to the experimental values obtained for the baseline plant.

4.1. Mathematical model

The ejector-based CO_2 refrigeration plant has been conceived with the same cooling power of the baseline plant. Considering Figure 8, the new plant has a different layout: compression is realised from intermediate pressure to the highest one, while the mass flow rate that flows through the evaporator is used as secondary stream in the ejector.

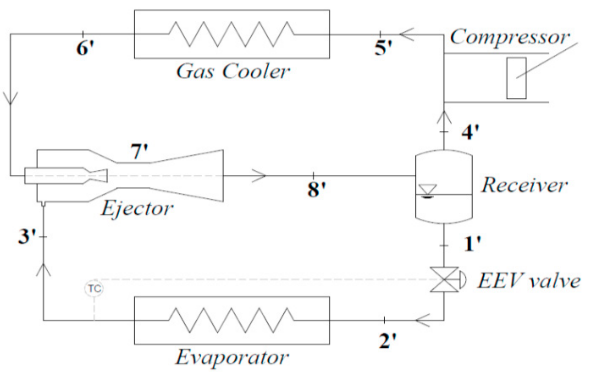

Fig. 8. Ejector-based plant layout

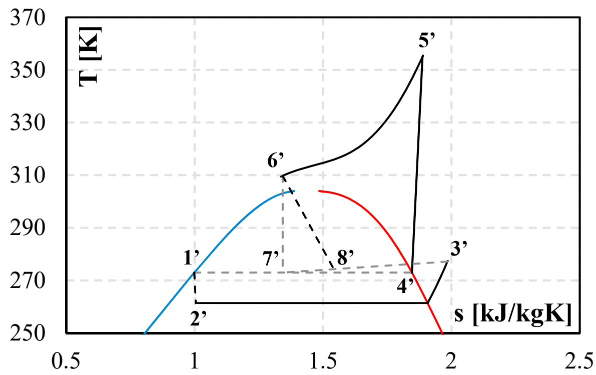

Fig. 9. Ejector-based plant T-s diagram for environmental temperature equal to 33.5°C

In order to compare the new layout with the baseline plant, the three pressure levels have been kept equal to the experimental data of the tested plant, as well as the temperature of the CO_2 exiting the gas cooler (which is strictly related only to the ambient temperature). The model starts evaluating the energy balance on the receiver, Eqn. (4).

$$\dot{m}_{evap} h_{1'} + \dot{m}_{comp} h_{4'} = (\dot{m}_{evap} + \dot{m}_{comp}) h_{8'} \tag{4}$$

In Eqn. (4), state points 1’ and 6’ are known, having fixed the intermediate pressure. Also the evaporator mass flow rate is known, being equal to the one of the experimental tests (i.e. same cooling request). Therefore, the mass flow rate that crosses the compressor is known if the vapour quality of the point 8’ is imposed and used as parameter.

Once the two mass flow rates have been calculated, the entrainment ratio of the ejector can be computed as in Eqn. (5) [22]:

$$\mu = \frac{\dot{m}_{evap}}{\dot{m}_{comp}} \tag{5}$$

$$\eta_{ej} = \mu \frac{(h_{3',isent} - h_{3'})}{h_{7'} - h_{6'}} \tag{6}$$

Hence, the ejector efficiency can be defined as in Eqn. (6), being the ratio between the energy saved by the ejector, i.e. the enthalpy difference between the secondary stream enthalpy and its final value if it is compressed along an isentropic transformation, and the maximum energy available [23]. This energy is the enthalpy difference between the isentropic and isenthalpic expansion from gas cooler exit (high pressure) to intermediate pressure (Eqn. (6)). Remembering that thermodynamic state point 6, in baseline case, is equal to state point 6’ in ejector-based case.

In order to simplify the analysis, the detailed ejector behaviour has been neglected: only the two inlet points and the final exiting one have been considered. This simplification does not affect the final energy balance of the ejector-based plant [24, 25]. Finally, considering a fixed isentropic efficiency of the compressor (equal to the reference experimental case), the compressor work can be calculated.

5. Results

Due to the ejector-based layout, the temperature of the CO_2 at the exit of the compressor is about 15-25 K lower than the baseline case (Fig. 10), producing a lower compression work up to 22 kJ/kg, in the environmental conditions considered. These values do not depend on the entrainment ratio of the ejector, while it affects the vapour quality downstream the ejector and, so, the ejector efficiency (Fig. 11).

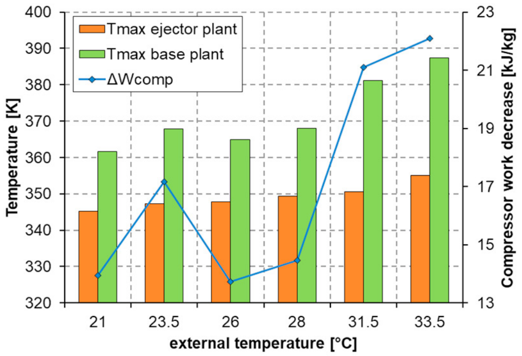

Fig. 10: Tmax of the refrigerating cycles and delta compressor work

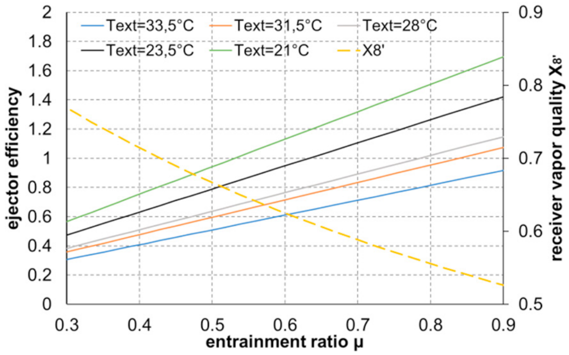

Fig. 11. Vapour quality of the stream at the separator inlet and ejector efficiency as function of entrainment ratio of the ejector

The COP is strongly dependent on the entrainment ratio (in particular on the compressor mass flow rate), but a value of μ>0.6 seem to lead to higher COP values, also for low external temperatures (Fig. 12). The entrainment ratio is the result of the geometry of the ejector and the operation condition of the plant (environmental conditions), so, it can be used as design parameter. Fig. 13 shows the result of the exergetic analysis: ejector accounts only for 12.5% of the whole irreversibilities, leading to an exergetic efficiency of about 34.8%.

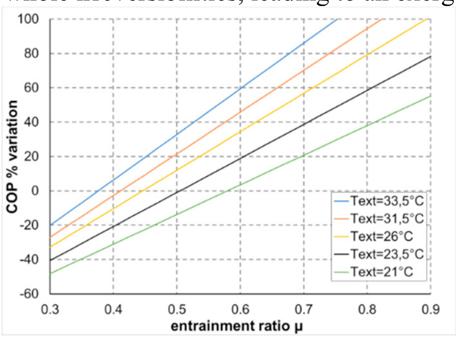

Fig. 12. COP variation as function of entrainment ratio of the ejector

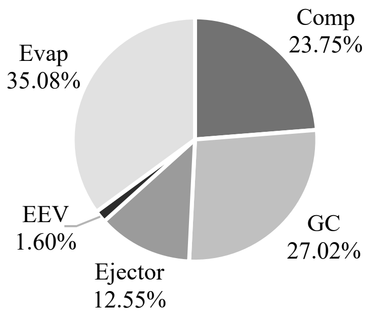

Fig. 13. Share of irreversibilities of each component of the CO_2 ejector-based refrigeration plant

6. Conclusions and Future Work

In this paper the development of a research facility for CO_2 refrigeration has been presented. The system is based on a transcritical, single-stage vapour compression cycle and includes instrumentation and controls to perform a wide range of measurements. In the test campaign herein reported, the focus was on the effects that warm ambient temperatures have on the CO_2 refrigeration system. Based on a test case with at 33.5°C, energy and exergy analysis were carried out and allowed to outline performance improvement strategies. In particular, the theoretical potential achievable through a replacement of the high-pressure expansion valve with an ejector having an entrainment ratio of 0.54, considering vapour quality $X_{8'}$ equal to 0.65, would lead to a COP increase from 1.83 to 2.64 and an exergy efficiency from 29.4% to 34.8%.

Future activities will aim at a broader experimental assessment as well as at a comparison of the ejector technology with alternative ones, such as the usage of direct expanders as a replacement of the high-pressure expansion valve or bottoming thermodynamic TFC and ORC system fed by the waste heat upstream of the gas cooler.

Acknowledgements

The Authors acknowledge the financial support of EPTA S.p.A. and the knowledgeable contribution of prof. Roberto Cipollone (University of L'Aquila) throughout the research activities. Also Mr. Francesco Fedele has been acknowledged for technical support.

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