Ejector refrigeration: A comprehensive review

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The increasing need for thermal comfort has led to a rapid increase in the use of cooling systems and, consequently, electricity demand for air-conditioning systems in buildings. Heat-driven ejector refrig-eration systems appear to be a promising alternative to the traditional compressor-based refrigeration technologies for energy consumption reduction. This paper presents a comprehensive literature review on ejector refrigeration systems and working fluids. It deeply analyzes ejector technology and behavior, refrigerant properties and their influence over ejector performance and all of the ejector refrigeration technologies, with a focus on past, present and future trends. The review is structured in four parts. In the first part, ejector technology is described. In the second part, a detailed description of the refrigerant properties and their influence over ejector performance is presented. In the third part, a review focused on the main jet refrigeration cycles is proposed, and the ejector refrigeration systems are reported and categorized. Finally, an overview over all ejector technologies, the relationship among the working fluids and the ejector performance, with a focus on past, present and future trends, is presented.

Keywords:
Ejector
Refrigeration system
Working fluid
Cycle configuration
Technology comparison

1. Introduction

The increasing demand for thermal comfort has led to a rapid increase in cooling system use and, consequently, electricity demand due to air-conditioning in buildings [1]. Deployment of thermal energy refrigeration, using low-grade heat or solar energy, would provide a significant reduction of energy consumption [2–6]. Among the various technologies for thermal refrigeration, heat-driven ejector refrigeration systems (ERSs) seem a promising alternative to the traditional compressor-based technologies owing to their reliability, limited maintenance needs and low initial and operational costs. Moreover, ERSs may help in the reduction of greenhouse effect emissions through both saving in primary energy and avoidance of environmental harmful refrig-erants [7,8]. Nevertheless, ejector refrigeration has not been able to penetrate the market due to its low performance coefficient and severe degradation in performance when not operating under idealized design conditions [9].

In the existing literature, different reviews on ejector technologies have been presented [10–23]. All of the previous reviews are focused on a particular aspect or aspects of ejector refrigeration, whereas the goal of the present review is to propose a comprehensive view of all ejector refrigeration technologies and the impact of working fluids on their performance. This review has four main parts that each have sub-sections. In the first part, ejector technologies are described. In the second part, a detailed description of refrigerant properties and their influence over ejector performance is presented. In the third part, a review focused on the main jet refrigeration cycles is proposed and analyzed. This section is divided into eight subsections and covers all of the main refrigeration technologies presented in the literature (Fig. 1): the concepts and main aspects of each study have been described in detail and linked to other studies. Finally, an overview is presented covering all of the ejector technologies, the relationships between working fluids and ejector performance, with a focus on past, present and future trends.

2. Ejectors technology

2.1. Technology

An ejector is a simple component: a primary flow enters into a primary nozzle accelerating and expanding entraining a secondary flow entering from a suction chamber. The flows mix and a diffuser compresses the stream (Fig. 2).

2.2. Ejector classification

An ejector can be classified by (i) the nozzle position, (ii) nozzle design and (iii) the number of phases, as outlined in Table 1. In the following paragraphs, these classifications will be detailed.

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Published Journal Article available at: http://dx.doi.org/10.1016/j.rser.2015.08.059

Nomenclature

Acronyms

CAM	constant-area mixing ejector
CC	cooling capacity
CFD	computational fluid dynamics
COP	coefficient of performance
CPM	constant-pressure mixing ejector
CRMC	constant rate of momentum-change ejector
NXP	nozzle exit position
SERS	single ejector refrigeration system
ERS	ejector refrigeration system
SoERS	solar-powered ejector refrigeration system
BERS	bi-ejector refrigeration system
EAbRS	combined ejector–absorption refrigeration system
EAdRS	combined ejector–adsorption refrigeration system
CERS	vapor compression–ejector refrigeration system
EERS	ejector expansion refrigeration system
MERS	multi-components ejector refrigeration system
TERS	transcritical ejector refrigeration system

Greek letters

η	efficiency
ϕ	ejector area ratio (A_m/A_t)
ω	entrainment ratio (m_s/m_p)

Parameters

h	specific enthalpy [kJ/kg]
m	mass flow rate [kg/s]
p	static pressure [Pa]
Q_e	evaporation heat energy(cooling effect) [J]
L	mechanical work [j]
R_c	compression ratio (p_c/p_e) [dimensionless]
R_d	expansion ratio (p_g/p_c) [dimensionless]
T	temperature [°C]

Subscripts

c	condenser or mixed flow
ejector	parameter referred to the ejector
mec	machanical efficiency
overall	overall efficiency
pump	mechanicl pump
e	evaporator or secondary flow
g	generator or primary flow
in	inlet
m	mixing chamber
out	outlet
p	pressure or primary flow
s	secondary flow
t	throat

2.3. Nozzle position

Two common ejector nozzle configurations are the constant-pressure mixing ejector (CPM), in which the nozzle exit is in the suction chamber and the constant-area mixing ejector (CAM), in which the nozzle exit is placed in the constant-area section. The mixing process occurs in the suction chamber for CPM ejectors and in the constant area section for CAM ejectors.

CPM ejectors are widely used because of their ability to operate against larger backpressures. Accordingly, CPM ejectors generally perform better than CAM ejectors despite CAM ejectors being able to provide higher mass flow rates [24]. Eames [25] proposed a

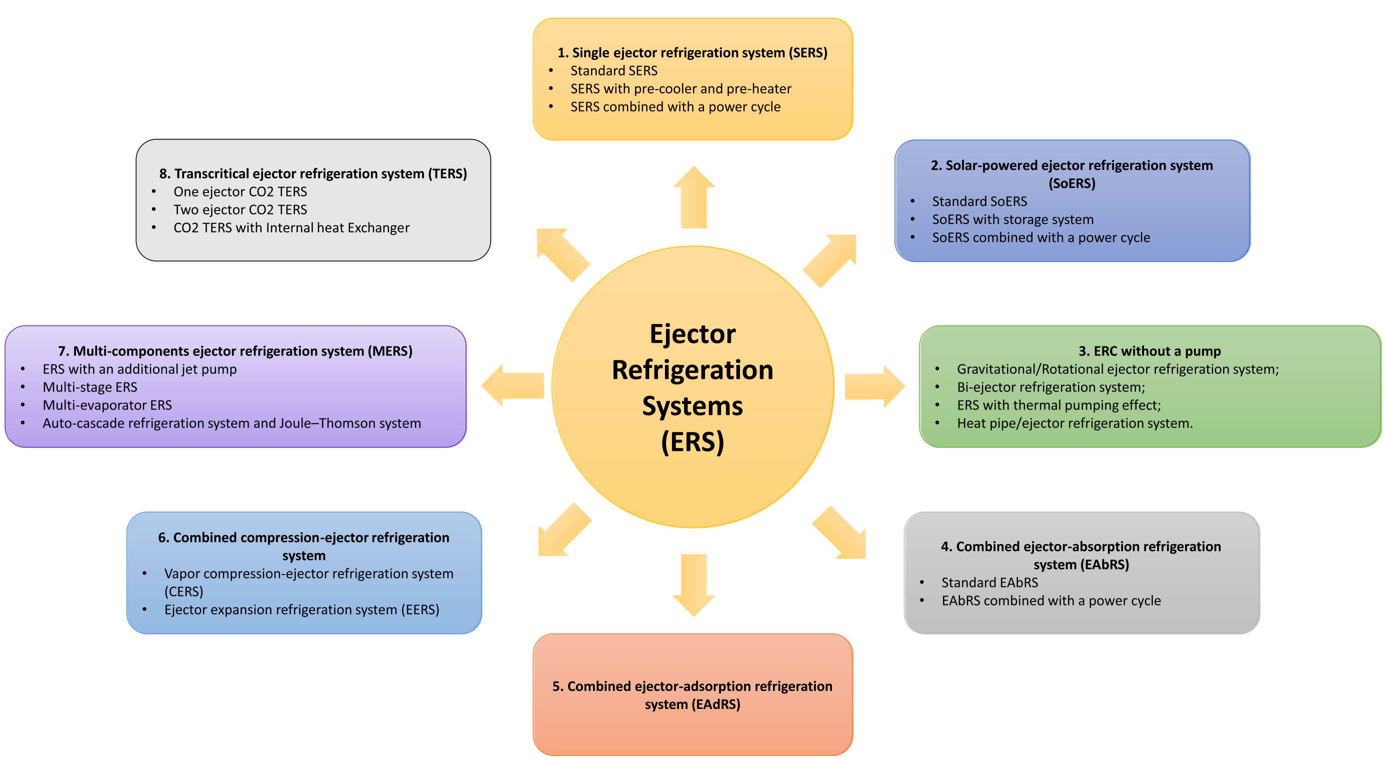

Fig. 1. Overview of ejector refrigeration systems.

constant rate of momentum-change (CRMC) ejector, which seeks to combine the best aspects of CPM and CAM ejectors. The CRMC configuration uses a variable area section rather than a constant area section, which provides an optimum flow passage area to reduce the thermodynamic shock thus increasing ejector performance. The method assumes a constant rate of change of momentum within the duct.

2.4. Nozzle design

Nozzle geometry affects ejector operation. Specifically, the nozzle shape can be convergent, i.e., the ejector works in a subsonic regime and it can reach, at most, a sonic condition at the suction exit, or it can be convergent–divergent, thus flow through the ejector may reach supersonic velocities. The choice between the two types of ejectors depends largely on the specifics of the end application.

Subsonic ejectors are not designed to produce a significant fluid compression, but they must provide little pressure loss. In the energy industry, they can be employed in industrial plants for exhaust gases [26], proton exchange membrane fuel cell (PEMFC) systems [27–33], chemical looping combustion (CLC) power plants [34,35] and transcritical CO_2 ejector refrigeration systems (TERS) [16,36]. Supersonic ejectors are used when there is a need to generate a high pressure difference: in the supersonic regime, the primary flow can entrain a high quantity of suction fluid because of the lower-pressure at the nozzle exit and high momentum transfer. Main energy applications are fuel cell recirculation sys-tems [37], i.e., molten carbonate fuel cells [38,39] and solid oxide fuel cells [40,41], ejector metal topping power plants [42,43], ejector organic Rankine cycles [44] and ejector refrigeration sys-tems (ERS), which are the topic of this review.

The actual operating conditions will depend, however, on the backpressure value and the fixed primary and secondary flow conditions. In the following, the operating conditions of subsonic and supersonic ejectors are described, and details of their fluid dynamics are outlined.

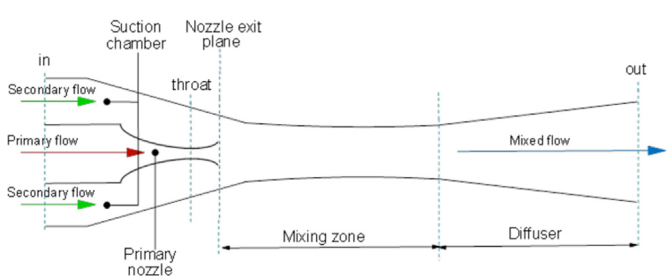

Fig. 2. Ejector layout.

2.4.1. Subsonic ejector

The subsonic ejector can work in three different modes, as shown in Fig. 3. In the critical mode, the primary flow is choked and the secondary mass flow rate is constant. The subcritical mode, the primary flow is not choked and there is a high depen-dence of the secondary mass flow rate on the backpressure value is present. In the malfunction mode (back-flow) the secondary flow is reversed causing ejector malfunction.

2.4.2. Supersonic ejector

The supersonic ejector can work in three different modes, as shown in Fig. 4. In the critical mode (double-choking), the entrainment ratio is constant because of the choking of the primary and secondary flows. In the subcritical mode (single-choking), the primary flow is choked and a linear entrained ratio change with backpressure is present. In the malfunction mode (back-flow), the secondary flow is reversed causing ejector malfunction.

An important phenomenon related to secondary flow is the choking phenomenon that, in critical mode, limits the maximum flow rate through the ejector and thus cooling capacity (*CC*) and the coefficient of performance (*COP*) remain constants (refer to the next section for the detailed definition of these parameters). More precisely, primary fluid expanded waves, due to under-expansion, create a converging duct where there is no mixing. The entrained flow feels the cross-section constriction, reaches sonic speed and chokes in a certain position that varies with the operating conditions [45]. Thus, the secondary mass flow is not dependent on the downstream pressure and can be raised with the upstream pressure only. In contrast, during the subcritical mode, ejectors are influenced by the backpressure: upon increasing the backpressure, a shock wave moves into the mixing chamber interacting with the mixing and, increasing the backpressure further, the primary flow reverses back in the suction chamber. It is very complicated to describe in detail the flow characteristics because a series of oblique or normal shock waves occur and interact with shear layers. These complex fluid dynamics influence the performance of ejectors. Of particular importance is the dissipative effect of the shock trains as it produces a compression and a shift from supersonic to subsonic conditions. There is considerable research concerning experimental [46–65] and numerical [66–81] studies of the flow phenomena inside an ejector. Even further detailed knowledge and modeling of these phenomena should allow for better component design.

Table 1
Ejector classification.

Parameters	Condition			Classification	Remarks
Nozzle position	Inside suction chamber			CPM ejector	Better performance if compared with CAM ejector
	Inside constant-area section			CAM ejector	–
Nozzle design	Convergent			Subsonic ejector	–
	Convergent-divergent			Supersonic ejector	–
Number of phases	*Primary flow*	*Secondary flow*	*Exit flow*		
	Vapor	Vapor	Vapor	Vapor jet ejector	Possible two-phase flow Possible shock waves
	Liquid	Liquid	Liquid	Liquid jet ejector	No shock waves, single-phase flow only
	Vapor	Liquid	Liquid	Condensing ejector	Two-phase flow with primary flow condensation Strong shock waves
	Liquid	Vapor	Two-phase	Two-phase ejector	Two-phase flow Shock waves possible

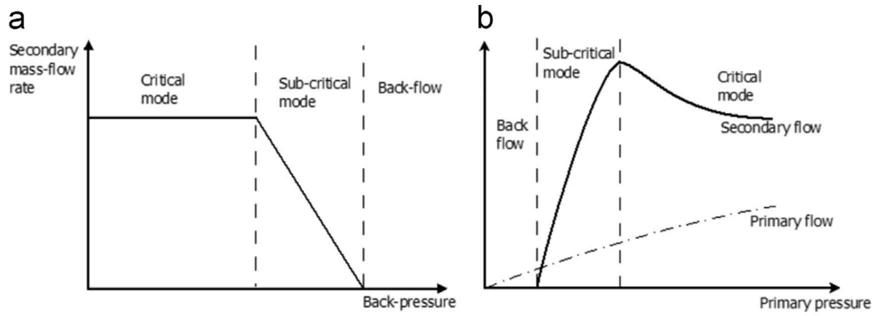

Fig. 3. Subsonic ejector operational mode (a) fixed primary pressure and (b) fixed backpressure.

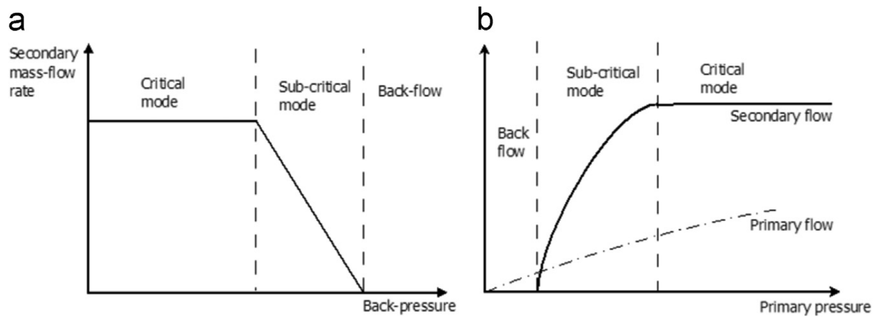

Fig. 4. Supersonic ejector operational mode (a) fixed primary pressure, and (b) fixed backpressure.

2.4.3. Number of phases

Depending on the primary and secondary flow conditions (Table 1), the flow inside the ejector can be either single phase (gas-gas or liquid-liquid) or two-phase. A two-phase ejector may be classified by the nature of the two-phase flow: (i) a condensing ejector (the primary flow condensates in the ejector) and (ii) a two-phase ejector (where the flow at the outlet is two-phase). The single phase ejectors are widely studied in the literature and the previous section references refer to them. The understanding and modeling of two-phase ejectors, however, is still limited.

The complete physics of fluid flow in a condensing ejector is very complex [82–84], making modeling very difficult [85–88]. The condensing ejector combines a subcooled liquid stream and a vapor stream, whereby a liquid stream is formed via condensation, which has a stagnation pressure potentially higher than the inlet pressure. The phase change phenomenon is governed by both two phase heat transfer and the mixing, favored by the high relative velocity and the large temperature difference between the vapor and liquid streams. Vapor condenses onto the liquid stream and the momentum of the liquid increases accordingly. The rapid condensation process causes shock waves resulting in a completely liquid state downstream of the shock [85,89,90]. In configurations where condensation is present, the steam is often assumed to be a perfect gas, a rather strong simplification that can result in significant errors. A more correct description of the steam is obtained by considering metastable behavior. This is related to the short time available for expansion in a supersonic nozzle preventing establishment of thermodynamic equilibrium resulting in frequent occurrence of metastable states [91]. Moreover, droplet nucleation and the subsequent development of condensation result in an energy transfer that cannot be accurately simulated when assuming the steam to be a perfect gas. Therefore, recent computational fluid dynamics, CFD, simulations of steam ejector performance have incorporated droplet nucleation and condensation using the homogeneous model [92–94]. For the ejector shape, a re-design of the nozzle is required to account for the nucleation downstream of the throat to provide a sufficient distance for avoiding the presence of flow oscillations across the sonic section [91].

When the fluid exiting the ejector is two-phase, both a liquid state and a vapor state exists in which either [95] (i) the primary fluid is a liquid that entrains a gas or (ii) the primary fluid is high pressure steam that entrains a liquid secondary flow. The detailed modeling of such a hydrodynamic process is also very difficult; one possible way is to apply an Eulerian two-fluid approach [95]. When using an Eulerian two-fluid approach, a proper solution for the two-phase flow depends on the correct modeling of interphase forces and turbulence models. These closure models must describe complex phase interactions. Although this topic has been widely discussed for other types of two-phase flows, e.g., bubble columns, the closures for ejector two-phase flow are not yet clear. The closures may involve drag and lateral forces, i.e., the lift force, the wall and the turbulent dispersion force. Another possible solution method could be the tracking interface method, but at present, there are not clear guidelines for this framework.

2.5. Performance parameters

Several parameters are used to describe the performance of ejectors in refrigeration cycles, as provided below.

- The entrainment ratio, ω, is the ratio between the secondary flow mass flow rate, m_s, and the primary flow mass flow rate, m_p:

$$\omega = \frac{m_s}{m_p} \tag{1}$$

- The compression ratio, R_c, is the static pressure at the exit of the diffuser, p_c, divided by the static pressure of the secondary flow, p_e:

$$R_C = \frac{p_c}{p_e} \tag{2}$$

  The entrainment ratio evaluates the refrigeration cycle efficiency, and the pressure lift ratio is a measure of the operative range of the cycle.
- The coefficient of performance, *COP*, is the ratio between evaporation heat energy, Q_e (cooling effect), and the total incoming

Table 2
Refrigerant classification and safety characteristics.

Group		Safety group [96] (toxicity/flammability)	Working fluid
Halocarbon compounds	CFC	A1	R11, R12, R113, R114
	HCFC	A1–B1	R21, R22, R123, R141b, R142b, R500, R502
	HFC	A1–A2	R134a, R152a, R236fa, R245fa
	HFO	A2L	R1234yf
Hydrocarbon compounds	HC	A3	R290, R600, R600a
Other refrigerants		B1	CH_3OH
		B2L	R717
		A1	R718b, R744

energy into the cycle ($Q_g + L_p$).

$$COP = \frac{Q_e}{Q_g + L_p} \quad (3)$$

- The cooling capacity, CC, is given by

$$CC = m_e(h_{e,out} - h_{e,in}) \quad (4)$$

- Concerning the ejector itself, there are many ways to define the ejector efficiency, $\eta_{ejector}$. The efficiency used by ASHRAE is defined as the ratio between the actual recovered compression energy and the available theoretical energy in the motive stream [14]:

$$\eta_{ejector} = \frac{(m_g + m_e)(h_{c,in} - h_{e,out})}{m_g(h_{g,out} - h_{e,out})} \quad (5)$$

3. Ejector refrigeration: working fluids

In this section, we will present and discuss the main working fluids used in the ERS. The selection of the appropriate refrigerant is of fundamental importance in the design of an ERS. In the past, the main principle for selection was the maximization of the performance; more recently, several factors (safety, cost, etc.) are considered, and the final choice depends on the compromise between the performance and the environmental impact. The working fluids can be classified based on the chemical compounds and can be classified into three main groups [10] (Table 2): (i) the halocarbon group (i.e., chlorofluorocarbons (CFCs), hydrochlorofluorocarbons (HCFCs), hydrofluorocarbons (HFCs) and hydrofluoroolefin (HFO) and the hydrocarbon group (HC)), (ii) organic compounds consisting of hydrogen and carbon (i.e. R290, R600, R600a) and (iii) other refrigerants, i.e., water R718b, ammonia R717 and carbon dioxide R744.

3.1. Criteria for working fluid selection

Generally speaking, a suitable refrigerant for a refrigeration system should be able to guarantee high performance for the required operating conditions. Accordingly, working fluid thermo-physical properties must be taken into account. Thermo-physical properties should satisfy some constraints: they should have a large latent heat of vaporization and a large generator temperature range for limiting the circulation rate per unit of CC and the fluid should have a high critical temperature to compensate large variations in generator temperatures. The fluid pressure should not be too high in the generator for the design of the pressure vessel and for limiting the pump energy consumption. Moreover, the viscosity, the thermal conductivity and the other properties that influence the heat transfer should be favorable. A high molecular mass is desirable to increase ω and $\eta_{ejector}$ [37]; however, this requires smaller ejectors (for the same output), introducing design difficulties and performance issues related to small-scale components. Low environmental impact, as defined by the global warming potential, GWP, and the ozone depletion potential, ODP, is also an important factor for consideration. The fluid should also be non-explosive, non-toxic, non-corrosive, chemically stable, cheap and available on the market. Finally, the dry or wet working fluids must be considered on the basis of the differential entropy equation for an ideal gas:

$$dS = c_p\frac{dT}{T} - R\frac{dp}{p} \quad (6)$$

An increase in temperature or a decrease in pressure will raise the fluid entropy. Depending on which effect prevails between temperature and pressure, the saturated vapor line in the *T–s* diagram can have either a negative slope or positive slope. In a simple molecular compound, the pressure effect is typically dominant, whereas in a complex molecular compound, due to its high molar heat capacity, the thermal effect typically has a greater influence. According to the saturated vapor line slope in the *T–s* space, a working fluid can be defined as follows: (i) wet vapor, if the saturated vapor line forms a negative slope (low molecular complexity); (ii) isentropic vapor, if the saturated vapor line is approximately vertical; and (iii) dry vapor, if the saturated vapor line forms a positive slope (high molecular complexity).

In a dry or isentropic vapor, phase change is typically not present in the primary nozzle expansion. This is in contrast to a wet vapor where drops can appear near nozzle outlet. These drops may block the effective area with the presence of unsteady flow and, by consequence, lead to unstable system operation [91]. A possible solution can be to superheat the fluid before passing into the nozzle even if it decreases the ejector efficiency [10,97,98]. However, it is noted that even for the isentropic and dry fluids, isentropic expansion can occur in the two-phase zone. If the saturation temperature is close to the critical value, the expansion may lead to the same problems found using wet fluids. As a result, for some dry and isentropic fluids, it is best to avoid temperatures approaching the critical value for ejector refrigeration systems. It should be noted that in actual application, fluid dynamic losses will actually reduce this problem because the state at the nozzle exit is much closer to the vapor saturation line.

3.2. Working fluids in ejector refrigeration

The versatility of the ejector technology has allowed testing different working fluids (Table 3). Using water (R718b) as a working fluid provides many advantages [99–124]: it has a high heat of vaporization, is inexpensive and has minimal environmental impact; however, the cooling cycle temperature is limited to above 0 °C, limiting the obtainable COP to less than 0.5 [125]. Moreover, due to the large specific volume of the water, large diameter pipes are required for minimizing the pressure loss [126]. Therefore, water is often employed in experimental devices but is rarely used in real refrigeration systems. The halocarbon compounds can be used for providing cooling temperature below 0 °C and exploit low-grade thermal energy at approximately 60 °C producing an accep-table COP (0.4–0.6) [98,99,106,118,127–189]. For example, the low-pressure refrigerant R113 has a high molecular mass and is able to produce a high mass ratio (0.5–0.6), a good ejector efficiency (0.5–0.55) and a high compressibility factor (0.9–0.995) [135]. However, several high performance halocarbon refrigerants are not environmentally friendly, having ODP or a high GWP. After the Montreal Protocol, some refrigerants have been banned, which has

Table 3
Working fluids for ejector refrigeration systems.

Ref.	Wet/Dry vapor	Molecular mass [kg/kmol]	Boiling point [°C]	Latent heat at 10 °C [kJ/kg]	GWP (100 yr)	ODP	Employment in ERS Ref.
R11	Wet	137.4	23.7	186.2	4750	1	[99,106,118,127–130,135]
R12	Wet	120.9	−29.8	147.8	10,900	1	[99,106,118,130,131,135]
R22	Wet	86.5	−40.8	196.8	1790	0.05	[129–133]
R113	Dry	187.4	47.6	155.9	6130	0.85	[99,118,127,130,134–136]
R114	Dry	170.9	3.8	133.7	9180	0.58	[129,130,135,137,138]
R123	Dry	152.9	27.9	177.5	77	0.01	[99,106,118,129,139–142,144,145,179]
R134a	Wet	102.0	−26.1	190.9	1370	0	[98,99,106,118,129,133,139, 144,146–162,181,182,185–187, 234,235]
R141b	Dry	116.9	32.1	233.1	717	0.12	[129,144,149,157,161,163–170,178,183,184]
R142b	Dry	100.5	−9.2	212.0	2220	0.06	[99,118,129,149,161,171–175]
R152a	Wet	66.1	−24.0	295.8	133	0	[98,99,118,129,133,139,144,146, 149,156,157]
R245fa	Dry	134.1	15.1	199.0	1050	0	[98,149,170,176,177,180,188,189]
RC318	Dry	200.0	−6.0	110.7	10,300	0	[99,118,129]
R290	Wet	44.1	−42.1	360.3	20	0	[98,144, 146, 149, 156, 157, 190–192]
R500	Wet	99.3	−33.6	–	8100	0.61	[99,118,130]
R502	Wet	111.6	−45.3	–	4600	0.31	[106,130]
R600	Dry	58.1	−0.5	376.1	20	0	[98,146,149,156,191,193,194]
R600a	Dry	58.1	−11.8	344.6	20	0	[98,144,156,157,171,191,192,195–198,200]
CH_3OH		32.0	64.7	1194.5	–		– [109, 206–209]
R717	Dry	17.0	−33.3	1226.1	0	0	[106,130,135,139,144,146,149,157,192,202–205]
R718b	Wet	18.0	100	2477.2	0	0	[99–124,236]
R744	Wet	44.0	−78.5	197.7	1	0	[205,210–227]

led to the adoption of considerably different working fluids. For example, HFCs have significant benefits regarding safety, stability and low toxicity and are appropriate for large-scale applications. Even more promising for the future are HFOs. They can offer balance among performance, environmental impact, safety and durability. However, they belong to A2L safety group; thus, they will require changes to equipment safety standards. In addition to the new halocarbon compounds, the low environment impact of HCs make them possible alternatives [98,144,146,149,156,157,171,190–200]. Unfortunately, HC refrigerants are highly flammable, which may limit their usage [201]. These concerns can be relieved with additional research on new mixtures of HCs and HFCs [8]. Another working fluid that has been studied is ammonia (R717) [106,130,139,144,146,149,157,192,202–205] for its low cost, high performance and more favorable thermodynamic properties, and it does not create environmental problems. However, it likely will remain restricted to industrial applications, as it is unsuitable for domestic use due to its toxicity [13]. Another interesting option is methanol thanks to its appropriate thermo-physical properties, low environmental impact and low cost, and it is able to provide a cooling effect at evaporation temperatures below the freezing point of water [109,206–209]. On the other hand, methanol is toxic and highly flammable; therefore, important preventive measures should be taken. Recently, many studies have focused on the carbon dioxide (R744) refrigerant: CO_2 is a natural substance, is non-flammable and has negligible GWP and zero ODP [205,210–227]. In particular, the refrigeration cycles using carbon dioxide are transcritical (the critical temperature of CO_2 is approximately 30.85 °C). In the recent years, the regulations are becoming stricter in terms of environmental protection. The EU Regulation 517/2014 will phase out and limit the use of refrigerants with high GWP values such as R134a, R404a and R410a in the next future. Therefore, environmentally friendly halocarbons, hydrocarbons, natural refrigerants (R717, R744) and HFC/HFO mixtures will be increasingly adopted [228]. Due to the limitations in existing working fluids, there is increasing research about potential substitutes (i.e. R1234yf [229] as potential sostitute for R134a [20,230–232]) and refrigerant blends, e.g., Hernandez et al. [233] studied blends of 410A and 507. The results indicated that for a certain range of generator temperatures, the refrigerant blend has higher performance if compared with either of the individual refrigerants.

3.3. *Screening of working fluids in ejector refrigeration*

The goal of this section is to provide an overview of studies concerning the screening of the working fluids, without focusing on cycle performance. For a detailed analysis, the reader should refer to the next sections where these studies are discussed and compared. Herein, only the studies comparing at least three or four refrigerants are listed. The details of these studies can be found in the referred sections.

3.3.1. *Single Ejector Refrigeration Cycle (Section 4.1)*

Dorantes and Lallemand [129] (R11, R22, R114, R123, R133a, R134a, Rl41b, R142b, R152a, RC318 and non-azeotropic mixtures) reported R123 (*COP*=0.20), R141b (*COP*=0.21) and RC318 (*COP*=0.20) to have the best performance. Sun [99] (H_2O, R11, R12, R113, R21, R123, R142b, R134a, R152a, RC318 and R500) obtained the best results with R152a (*COP* of 0.09–0.50) and R500 (*COP*= 0.09–0.47), whereas the steam jet systems had low performance (*COP*=0–0.35). Cizungu et al. [139] (R123, R134a, R152a and R717) reported R134a and R152a to be appropriate for heat sources at 70–80 °C and R717 is appropriate for temperatures higher than 90 °C, with R134a the working fluid with the highest *COP* (0.1–0.45). Similar results were shown by Selvaraju and Mani [146] (R134a, R152a, R290, R600 and R717): R134a had the highest *COP* (0.12–0.40) and critical ω (0.20–0.45). Hernandez [156], reported (in order) R152a, R134a, R600a and R600 in terms of *COP*, ω, efficiency and the least ϕ. Kas-perski and Gil [191] compared nine heavy hydrocarbons (R290, R600, R600a, R601, R601a, R602, R602a, R603 and R604) and calculated the optimal temperature ranges of vapor generation for each fluid; each hydrocarbon had its own maximum ω at its unique optimal tem-perature. The highest *COP*, equal to 0.32 was achieved for R600a at the temperature of 102 °C and a *COP* equal to 0.28 for R601 at 165 °C. The same authors [237] compared refrigerants (organic and non-flammable) for a high temperature heat source (acetone, benzene, cyclopentane, cyclohexane, toluene, R236ea, R236fa, R245ca, R245fa, R365mfc and RC318): no single refrigerant could ensure high performance across the entire operating range. Among the non-flam-mable refrigerants, R236fa was the most beneficial, providing a maximum *COP* equal to 0.23. The use of organic solvents may be applied for high T_g values, and, among the different working fluids, cyclopentane had the highest values of both ω and *COP* across the entire operating range. Each substance has its own maximum ω and *COP* at its unique optimal temperature. The use of non-flammable synthetic refrigerants provides higher *COP* values in the low primary vapor temperature range. R236fa was the most beneficial among the non-flammable synthetic refrigerants tested. The use of organic solvents can be justified only for high values of motive steam temperature. Among the solvents, the highest values of ω and *COP* throughout the range of motive temperature were noted for cyclopentane. Toluene was found to be an unattractive refrigerant from the ejector cooling point of view. Chen et al. [98] (R134a, R152a, R290, R430A, R600, R245fa, R600a, R1234ze and R436B) found R600 to be viable option for an ejector refrigeration system considering system performance and environmental aspects; flammability was left for further analysis. Shestopalov et al. [189] (R123, R141b, R142b, R236fa, R245ca, R245fa, R600 and R600) considered low-pressure refrigerants and R600, R600a and R245fa had the best performance combinations. In particular, the authors suggested R245fa for the thermodynamic properties and the non-corrosive, non-toxic, and non-flammable characteristics.

3.3.2. Solar-powered ejector refrigeration systems (Section 4.2)

Al-Kahlidy [135] (R11, R12, R113, R114, R717) selected R113 for its high molecular weight and large compressibility factor. Zhang and Mohamed [199] (R1234yf, R1234ze, R290, R600, R600a, R601, R744, R134a) suggested R601 for a combined power and ejector cooling cycle with a high critical temperature (196.7 °C) for wide operating temperature range applications in the hot climates. Tashtoush et al.[238] (R717, R134a, R600, R600a, R141b, R152a, R290 and R123) reported a better COP for R717, R290, R152a and R134a

3.3.3. Ejector refrigeration systems without pump (Section 4.3)

Shen et al. [106] (R11, R12, R22, R134a, R123, R502, R717 and H_2O) reported a high *COP* equal to 0.26 using R717 in a bi-ejector refrigeration system.

3.3.4. Combined ejector–absorption refrigeration systems (Section 4.4)

Jaya et al. [152] (DMAC-R32, DMAC-R124 and DMAC-R134a) reported on R124-DMAC and R134a-DMAC having found a *COP* of approximately 1.0 at low generator and evaporator temperatures (T_g of 100–110 °C, T_e of 5 °C) and found R32-DMAC to have high circulation ratios and high generator pressures.

3.3.5. Combined compression–ejector refrigeration systems (Section 4.6.1)

Sun [118] evaluated a combined CERS (R11, R142b, R12, R134a, R21, R152a, R113, R123, RC318, H_2O and R500); the system had a significant increase in performance using dual refrigerants: R718 for the ejector cycle and R21 for the vapor compression cycle.

3.3.6. Combined compression–ejector refrigeration systems (Section 4.6.2)

Kornhauser [130] analyzed an EERS (R11 R12 R22 R113 R114 R500 R502 R717). R502 had the highest *COP* compared with the other refrigerants ($COP=5.67$); R717 also had notably high per-formance ($COP=5.33$). For these refrigerants, the potential increase in *COP* with the ejector expansion cycle is much greater. Nehdi et al. [161] (R134a R141b R142b R404A) reported the best *COP* improvement (+22%) was obtained with R141b. Sarkar [192](R290 R600a R717) provided maximum performance improve-ment for R600a, whereas minimum performance improvement was achieved for ammonia.

3.3.7. Multi-components ejector refrigeration system (Section 4.7)

Elakdhar et al. [144] (R123, R124, R134a, R141b, R152a, R290, R717 and R600a) and Kairouani et al. (2009) [157] (R290, R600a, R134a, R152a, R717 and R141b) reported R141b to give the best performance.

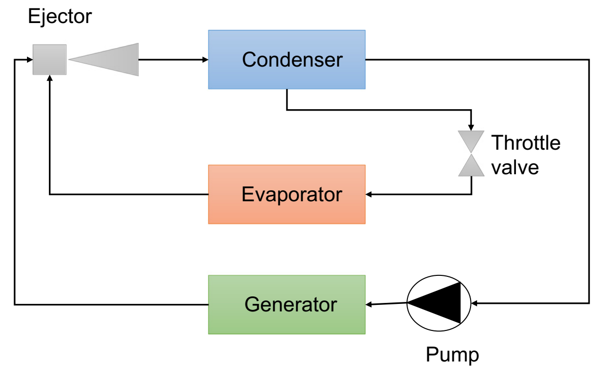

Fig. 5. Standard ejector refrigeration system.

4. Ejector refrigeration: technologies

4.1. Single ejector refrigeration system (SERS)

Single ejector refrigeration systems (SERSs) may be divided into three sub-categories: (i) standard SERS, (ii) SERS with a pre-cooler and a pre-heater and (iii) SERS combined with a power cycle. In the following, for each section, we present a comprehensive collection of all existing literature regarding these systems.

4.1.1. Standard SERS

The standard cycle is structured as detailed in Fig. 5. The generator supplies low-grade heat energy for working fluid vaporization. Upon reaching saturation conditions, the flow at high pressure (primary flow) is sent to the nozzle entraining the secondary flow from the evaporator, i.e., vapor at low pressure. Mixing of the two streams is obtained, and the resulting mixed flow leaves the ejector being dispatched to the condenser, where condensation takes place with a heat flux rejected to the environment. The liquid then splits: one part expands isenthalpically through the valve and is fed into the evaporator, producing the desired cooling effect; the other part is pulled back into the generator by pumps. Thus, the generator is used to produce high-pressure vapor to drive the ejector. The tasks of the ejector are vapor “entrainment” and recompression before exiting the evaporator and being discharged into the condenser. Main features of a standard SERS are [13,69]: (a) the setting of generator and evaporator operating conditions, i.e., the ejector working at critical conditions and providing constant *COP* and *CC* (when exceeding the critical pressure, secondary flow is reduced and thus ω and *COP* decrease significantly). (b) Increasing the generator pressure will decrease ω but enhance the critical condenser pressure for a fixed evaporator pressure. This is related to the increase of the primary mass flow and the consequent growth of the expansion angle causing a reduction of the annular effective area; thus, less secondary flow is entrained. However, jet core momentum and mixed flow increase and the shock wave position moves downstream and such that the critical pressure grows reducing the *CC* and *COP*. (c) Once the generator conditions are fixed, an increase in pressure in the evaporator determines the increase of ω and the critical pressure at the condenser. This is due to the reduction of the under-expanded wave angle: a larger effective area is obtained resulting in an increase in the secondary flow. The jet core momentum is reduced, but the total momentum related to the mixed flow is higher due to the large secondary pressure. The shock position is pushed further downstream and the ejector can thus work against a higher backpressure. Thus, increases of *CC* and *COP* result.

This section is divided into two parts. The first focuses mainly on working fluid impact, and the second focuses on ejector geo-metry and operating conditions.

4.1.1.1. Working fluids influence. There has been significant attention given toward the selection of an appropriate working fluid for ejector refrigeration since the earliest studies. Dorantes and Lallemand [129] proposed to use non-azeotropic mixtures [239,240] and investigated a SERS applied to air conditioning systems using classical refrigerants (R11, R22, R114); pure and cleaner refrigerants, such as R123, R133a, R134a, Rl41b, R142b, R152a and RC318; and non-azeotropic mixtures. From their results, it is possible to deduce that with variable heat sink and source temperatures (T_e= 10–20 °C and T_g=90–130 °C), *COP* and ω are mainly dependent on the working fluid and the mixture com-position R123 (*COP*=0.20), R141b (*COP*=0.21), and RC318 (*COP*=0.20) show the best performance. A comparison of the performance of various working fluids was also obtained by Sun [99] based on a thermodynamic model. Among the eleven fluids tested (water, several halocarbon compounds, an organic fluid and an azeotrope R500), the best results were obtained with R152a (*COP*=0.09–0.50) and R500 (*COP*=0.09–0.47), and the steam jet systems had low performance (*COP*=0–0.35). The *COP* variation range for several working fluids is similar to the ω range. Cizungu et al. [139] compared R123, R134a, R152a and R717. The data obtained by the authors suggested a strong dependence of *COP* and ω on ejector geometry and compression ratio at different values of T_g. Furthermore, it was observed that the working fluids R134a and R152a are appropriate for heat sources at 70–80 °C and R717 is appropriate for temperatures higher than 90 °C; R134a had the highest *COP* of 0.1–0.45. Similar results were shown by Sel-varaju and Mani [146], who compared ERS performance using R134a, R152a, R290, R600 and R717. Even in this study, R134a provided the highest *COP* (0.12–0.40) and critical ω (0.20–0.45). More recent studies have focused on the screening of working fluids. Roman and Her-nandez [156], using a validated 1-D model with low ecological impact refrigerants, found that the R290 shows better performance. The working fluid permits the highest system *COP*, ω and efficiency and the least ϕ. Ranking by performance, R152a, R134a, R600a and R600 were also investigated.

Recently, Kasperski and Gil [191] presented a theoretical analysis based on a 1D model developed by Huang et al. (1999) [241]. Nine heavier hydrocarbons were tested and the optimal temperature range of vapor generation for each fluid was calculated: each hydrocarbon has its own maximum ω at its unique optimal temperature. Moreover, the optimal vapor generation temperature and maximum values of ω increase according to the hydrocarbon heaviness; peak values of *COP*, however, do not follow the same trend. The highest *COP*, equal to 0.32, was achieved for R600a at a temperature of 102 °C and a *COP* of 0.28 was obtained for R601 at 165 °C. R603 and R604 can be ignored. Chen et al. [170] studied the ejector operating characteristics, investigating possible general interactions and relationships of the external para-meters (T_g=75–125 °C, T_e=0–16 °C, T_c=27–43 °C and primary and secondary flow superheating ΔT=0–10 °C) and the internal para-meters (efficiencies of ejector components 0.7–0.98). The ejector per-formance is influenced by all internal, external and geometric parameters, as characterized by *COP*, ω and ejector internal entropy production. In particular, *COP* and ω increase with increasing T_g and T_e, but decrease with increasing T_c. Although a higher T_g increases *COP*, an excessively high T_g may decrease the ideal efficiency. Thus, an optimal T_g is observed for the maximum ideal efficiency (the optimal T_g is 100 °C for R141b, 95 °C for R245fa and 110 °C for R600a), whereas a higher T_e and a lower T_c reduce the irreversibility into the ejector. Moreover, the system *COP* and the ejector behavior are influenced by component efficiencies and the type of refrigerant used; R141b pro-vided the largest *COP*. Finally, an influence of the primary or secondary flow superheat is observed on ejector and system performance when wet working fluids are used, regardless of whether this is an evident advantage for R141b, R245fa and R600a. In an investigation by Chen et al. [98], wet fluids (R134a, R152a, R290 and R430A) and dry fluids (R245fa, R600, R600a and R1234ze) and an isentropic fluid (R436B) were analyzed in a numerical model to compare their performance capabilities and applicability in an ejector refrigeration system. To avoid droplet formation inside the ejector when working with wet fluids, the primary flow should be superheated before the ejector nozzle inlet. In some cases, superheating may also be desirable for dry fluids and isentropic fluids. The authors also proposed a numerical approach for determining the minimum superheat before the ejector nozzle inlet, which is not known a priori. For a wet fluid, the ideal amount of superheat is the minimum amount that eliminates droplet formation, i.e., when the flow exiting the ejector nozzle ends is at saturation. This optimal superheat relies on both the generator saturation temperature and the nozzle efficiency; over-superheating of the primary flow has a limited effect on ω and no effect on *COP*. However, excessive superheat leads to a decrease in ideal efficiency. Using the same methodology for dry and isentropic fluids, the need for superheat can be avoided as long as fluids are not operating at the high temperatures adjacent to their critical values. Accordingly, R600 appears to be a viable option for ejector refrigeration systems con-sidering system performance and environmental aspects; flamm-ability has not yet been addressed. Gil and Kaspergi [237] tested dif-ferent working fluids (acetone, benzene, cyclopentane, cyclohexane, toluene, R236ea, R236fa, R245ca, R245fa, R365mfc and RC318) for high temperature heat sources (T_g=70–200 °C, T_e= 10 °C, T_c=40 °C.). They found no one working fluid could accommodate the entire operating range, and each working fluid had its own maximum ω and *COP* at a certain optimal T_g. For the low T_g range, R236ea, R236fa and RC318, performed better than the other working fluids considered. A max-imum *COP* of 0.23 was found for R236fa (T_g=95 °C). For T_g values from 105 °C to 125 °C, the highest *COP* values were obtained for R236ea (*COP*=0.21). Above a T_g of 125 °C, the best fluid was found to be R123. The use of organic solvents may be applied for T_g > 120 °C. A value of *COP* above 0.35 was observed only for cyclopentane (T_g > 190 °C). The worst results were obtained for toluene: a *COP* lower than 0.2 was found across the entire operating range.

Some studies have focused on methanol. Riffat and Omer [206] studied an SERC by an experimental campaign and a CFD analysis. The results indicated that an ERS fed by methanol is able to pro-vide a cooling effect for temperature values lower than the water's freezing point (T_e= −2–14 °C), achievable using low-grade heat (T_g =80–100 °C), such as waste heat or solar energy. A study by Alexis and Katsanis [207] investigated ejector performance in a refrig-eration system using methanol and a thermal source with a medium temperature and a superheated temperature equal to 150 °C. Three independent variables can be considered for an ejector system: (i) the generator, (ii) the evaporator and (iii) the condenser conditions with the maximum *COP* linear function of generator (T_g=117.7–132.5 °C), cubic function of condenser (T_c=42–50 °C) and evaporator (T_e= −10–5 °C) temperatures:

$$COP_{max}= \sum_{i=0}^{1} B_i T_g^i \tag{7}$$

$$B_0= \sum_{i=0}^{3} T_e^i \sum_{j=0}^{3} \alpha_{ij} T_c^j \tag{8}$$

$$B_1= \sum_{i=0}^{2} T_e^i \sum_{j=0}^{3} \beta_{ij} T_c^j \tag{9}$$

One of the first exergy analyses of ERSs was presented by Alexis [101]. The results demonstrated that improving the ejector quality affects the system efficiency more than improving other components. This is explained by ejector exergy loss that is equal to 54%of the total irreversibility loss. The other exergy losses are due to the condenser (26.9%), the generator (10.8%), the evaporator (7.4%) and the expansion valve (1%). At design conditions, the second law efficiency is approximately 17%.

4.1.1.2. Geometry and operating conditions influence. In addition to studies focused on working fluids, an increasing number of studies have focused on the dependence of system performance on ejector geometry and operating conditions. In this section, a selection of these studies is presented.

The experimental and theoretical analysis presented by Sun [120] highlighted the limits of the use of fixed-geometry ejector in refrigeration cycles for low *COP* (approximately 0.2–0.3) and the difficulty in obtaining high performance under several operating conditions. From this study, the necessity of variable ejector geometry used in refrigeration cycles is evident, as variable geometry would increase performance across variable operating conditions and maintaining improved constant cooling system capacity. Such characteristics would allow ejector-refrigeration systems to obtain better performance with respect to conventional ejector systems making them comparable with conventional refrigeration and air-conditioning systems.

Concerning the nozzle shape and position, Aphornratana and Eames [119] found an apparent link between primary nozzle position and ejector performance based on *COP*, CC and critical condenser pressure for a refrigerator with a jet. *CC* and *COP* increase when retracting the nozzle into the mixing chamber. According to the authors, a specific nozzle position was necessary for each ejector and was not possible to find a unique optimum nozzle position for all operating conditions. Chunnanond and Aphornra-tana [100] analyzed static pressure trends through the ejector with variable operating temperatures (T_g=120–140 °C, T_e=5–15 °C and T_c=22–36 °C), and varied superheated level of the primary flow (heat input of 0–100 W) along with different geometry and posi-tions of the nozzle $NXP = -10$–20 mm (ϕ can be changed by the spindle position). This work found that a primary flow decrease and a secondary flow increase, i.e., a decrease in the boiler pressure, increased the *COP* (0.25–0.48) and *CC*. Consequently, a decrease of the mixed stream momentum was observed, leading to a reduction in the critical condenser pressure ($p_{c,cr}$=40–65 mbar). Furthermore, an increase in evaporator pressure (sacrificing the desired cooling temperature) increased the critical condenser pressure ($p_{c,cr}$= 48–55 mbar). This also led to the increase in the total mass flow and consequently increased *COP* and *CC* (*COP*=0.28–0.48). The cycle performance was not influenced by the superheating level of the motive fluid before entering the nozzle. Finally, when retracting the nozzle out of the mixing chamber, *COP* and *CC* increased and the critical condenser pressure was reduced ($p_{c,cr}$=41–47 mbar). Another experimental analysis was presented by Eames et al. [176]. They described and evaluated the design of a jet-pump refrigerator. Performance maps were used to evaluate the use of R245fa and the effect of the operational parameters. They found that ω and *COP* strongly depend on the nozzle geometry and position. The values varied up to 40% by changing the nozzle exit position by 10 mm (from −10 to 0 mm). The importance of nozzle exit position (NXP) and shape were also investigated by other authors by CFD and experimental techniques [69,71,242–244]. They found significant effect of the nozzle position on ejector performance. The influence of the nozzle parameters was also investigated by Hu et al. [245], that studie an adjustable two-phase ejector by experimentas and numerical simulations. They investigated the influence of throat diameter and NXP finding the optimum geometrical paremeters. A large amount of studies is focusing on the role of nozzle shape for improving the performances. Some examples may be the rotor-vane/pressure-exchange ejector [246], the petal nozzle [247], the lobel nozzle [248] and circle, cross-shaped, square, rectangular and elliptical nozzles [249]. Another work is the experimental investigation of Rao and Jagadeesh, testing Tip Ring Supersonic Nozzle and Elliptic Sharp Tipped Shallow nozzles [250] of the research of Zhu and Jiang on a bypass ejector [251]. Sharifi [252] investigated, by using CFD, the influence of the nozzle profile at constant area ratio. The resulting ejector was manufactured and tested, showing good agreement with the predicted performance.

Concerning the area ratio influence, Selvaraju and Mani (2006) [147] studied 6 different geometric configurations of the ejectors switching evaporator, generator and T_c. For a given ejector configuration and fixing T_e and T_c, an optimum temperature of the primary flow can be defined permitting to maximize ω and *COP*. They obtained some correlations via regression analysis to calcu-late *COP* and ω at critical conditions. *COP* can be evaluated by the following relation:

$$COP = -0.27238R_d - 0.37332R_c + 0.202621\phi + 0.968945 \quad (10)$$

where R_d is the expansion ratio (p_g/p_c), R_c is the compression ratio (p_c/p_e) and ϕ is the ejector area ratio (A_m/A_t). When increasing ϕ (at fixed primary flow conditions), ω increased but the pressure recovery decreased. According to Varga et al. [242], with increasing ϕ, the critical back-pressure decreases and ω increases; therefore, depending on operating conditions, an optimal value should exist. Cizungu et al. [203] analyzed a two-phase ejector using ammonia. From the modeling of the ejector a quasi-linear relation between the expansion rate and ϕ was found. Furthermore, the optimal primary nozzle diameter was found to decrease increasing the boiler temperature. The influence of ϕ (ϕ=4, 5.76 and 8.16), R_c (R_c=1.6/2.25) and R_d (R_d=2.1/2.6) on ejector performance (*COP*=0.12/0.30) was studied by Sankarlal and Mani (2007) [202]. They showed that by increasing the ejector ϕ and the R_d or decreasing the R_c, the *COP* and ω of the system increase. Further-more, performance of the ejector refrigeration system was found to be independent to the nozzle and mixing chamber diameters. Finally, *COP* decreased with R_c and increased with R_d. Yapici et al.[145], using R123, theoretically and experimentally determined the optimum for T_g and the maximum for *COP* as a function of ϕ at given evaporator and condenser conditions. *COP* decreases faster when the T_g decreases from the optimal temperature for a given ϕ. Yapici [140], analyzing ejectors with a movable primary nozzle, also observed an improvement of the ejector performance if it is care-fully designed and realized. The analysis indicated that the opti-mum position of the nozzle to obtain better performance is 5 mm outwards from the mixing chamber and for a T_g higher than 97 °C, *CC* remained constant but *COP* decreased. Chen et al. [253] applied a lumped parameter model for investigating the ejector optimum performance as and the optimum area ratio. It resulted that T_c and a greater influence than T_g on the ejector performance parameters (ω and ϕ) and suggested the use of variable area ejectors. Del Valle et al. [186] tested a R134a ejector focusing on the role of three mixing chamber for enhancing of the pressure recovery. The shape of the mixing chamber was found to have a large influence over the ejector performance, but further investigations (i.e., by CFD analy-sis) are needed for giving an insight view of the local phenomena. Finally, concerning the operating conditions (on-design and off-design), among the different studies, we propose the one by Aidoun and Ouzzane [97], where they conducted a simulation of an ejector-based system via a thermodynamic model considering different ejector operation characteristics. The fluid mixing conditions, related to the mixing chamber geometry, the fluid type and the inlet and outlet conditions, can lead the ejector to work in off-design conditions with a decrease in performance.

Moreover, in off-design conditions the increase of the internal superheat generation, due to inefficient mixing and normal shock waves, becomes relevant. The authors concluded that to prevent internal condensation, an inlet superheat of approximately 5 °C is necessary. A larger superheat limits the condenser efficiency. A numerical analysis conducted by Boumaraf and Lallemand [171] evaluated performance and operating cycle characteristics of the ERS using R142b and R600a. Results found by the authors suggest that for an ejector operating at critical mode, for a given geometry and T_e, *COP* decreases if the T_g exceeds the design point (T_g=120–135 °C). Therefore, designing the ejector at the highest possible temperature is preferred, guaranteeing a better performance at a lower source temperature. Furthermore, if an ERS designed for working with R142b and R600a at a defined temperature operates with the fluid R142b, the system *COP* increases by approximately 70%. Shestopalov et al. [188,189] studied (numerically and experimentally) the on-design and off-design operating conditions of an ERC. At first, a lumped parameter model for on-design and off-design operation is developed and a screening of working fluids is performed, suggesting R145fa. Then, an experimental setup was built and results were used for validating the model. Furthermore, NXP and the shape of the mixing chamber of system performance were investigated. The problem of the optimum operating condition has been addressed by Sadaghi et al. [254] proposing an energy, exergy and exergoeconomic analysis and optimizing the refrigeration system by means of an algorithm. On the other hand, ejector behavior can also be predicted by means of maps: Zegenhagen and Ziegler [181] experimentally investigated a R134a cooling system to develop three dimensional maps of the ejector operating conditions.

Finally, Ruangtrakoon and Aphornratana [123] designed, by means of CFD, and built a prototype of an SERC (CC=3 KW, *COP*=0.45). This work is an example of a successful coupling of the COF approach as a support for the system design.

4.1.2. SERS with pre-cooler and pre-heater

In some studies a regenerator (also called pre-heater) and a pre-cooler are added to the SERC to increase the system efficiency [15]. A SERS with pre-cooler and pre-heater is presented in Fig. 6. The liquid refrigerant returning to the generator is pre-heated by the regenerator using the hot refrigerant arriving from the ejector exhaust. The liquid refrigerant is cooled by pre-cooler using the cold vapor refrigerant leaving the evaporator before reaching the evaporator. The refrigerant arriving from the condenser is heated and cooled before passing through the boiler and evaporator reducing the heat entering the generator and the cooling load to the evaporator of the system.

Huang and Jiang [134] used R113 as the working fluid in their experimental study. A performance map was constructed to show the ejector characteristics from which the design analysis of the ERS was carried out. They experimentally demonstrated that the sec-ondary flow choking phenomena play a very important role in ejector performance. In this early study, operation was at critical conditions, at which the ejector system should work, was identified and discussed. Sun and Eames [141] presented a numerical model for an ERS based on a thermodynamic model. If regenerators are intro-duced into the cycle, the heat input and cooling load are reduced and *COP* can be improved by approximately 20%. An additional two heat exchangers are needed leading to additional costs and system com-plications. Introducing a regenerator can significantly increase the system *COP*, but adding a pre-cooler does not.

Therefore, we may conclude that the introduction of a pre-cooler and a pre-heater in these refrigeration systems seems to be a poor techno-economical choice for general application. On the other hand, for specific applications, e.g., automobile air conditioning as in references [136,255], these technologies could be attractive.

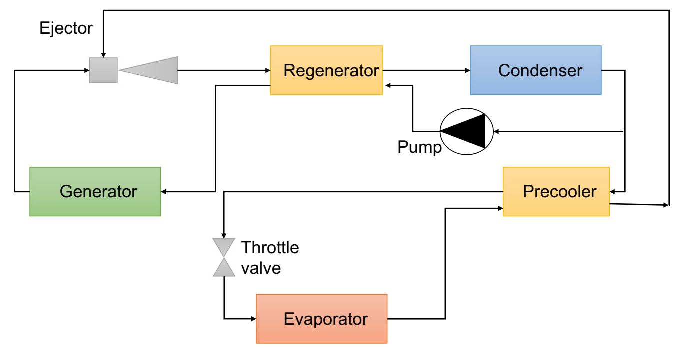

Fig. 6. SERS with pre-cooler and pre-heater.

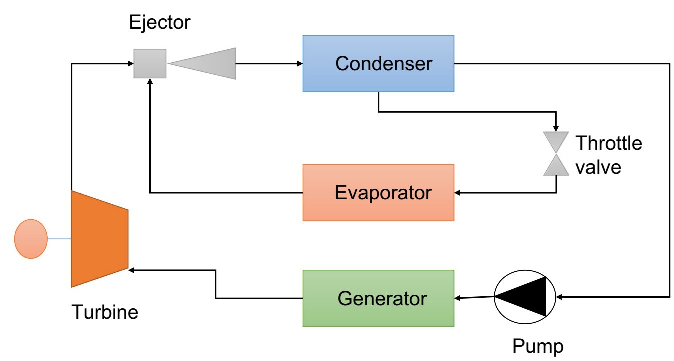

Fig. 7. Combined SERC and power system.

4.1.3. SERS combined with a power cycle

Cogeneration and tri-generation provide multiple useful outputs from one system. These systems are widely studied and applied presenting technological challenges at small scales. Different studies have tried to investigate power production ERC coupled systems.

4.1.3.1. Organic ranking – ERC systems. Zhang and Weng [180] investigated a combined Rankine cycle and a R245fa ERS for low temperature heat sources. In this configuration (Fig. 7) the primary flow of the ejector is the turbine outlet flow. They found a thermal efficiency of 34.1%, a first law efficiency of 18.7% and an exergy efficiency of 56.8% (T_g=122 °C, T_c=25 °C, T_e=7 °C). The influence of T_g was reported to have a significant impact on the cycle, i.e., from 60 to 140 °C, ω increased from 0.15 to 0.35 and the first law efficiency from 0.15 to 0.35 Wang et al. [256,257] investigated a combined Rankine cycle and ERS using ammonia–water mixtures and R123a. The authors studied the influence of the operating conditions and have performed an exergy analysis finding that the exergy destruction in ejector is not negligible. The authors also proposed another configuration of the cycle [114], combing absorption tech-nology (for the discussion of the absorption technology, refer to Section 4.4). Habibzadeh et al. [258] studied a coupled Rankine cycle to an ERS with different working fluids (R123, R141b, R245fa, R600a, R601a): R141b had the lowest optimum pressure and R601a had the highest thermal efficiency and the lowest exergy destruc-tion. Alexis [259] proposed a coupled 2 MW Rankine cycle to an ERS, as an alternative solution to absorption technologies.

4.1.3.2. Gas turbine – ERC systems. Different from the systems dis-cussed in Section 4.1.3.1, some studies have investigated hybrid gas turbines systems. Invernizzi and Iora [260] studied a coupled 30 kWe micro-gas turbine with an ERC using water, ammonia and R134a. A maximum *COP* of approximately 0.3 was found. This high performance is due to the high condensation temperature of the cycle, i.e., 40 °C for the wet cooling tower and 50 °C for the surface

Table 4
Operating conditions and performance of state-of-the-art of SERS and ERS: (T) theoretical study and (E) experimental study.

Ref.	Working fluid	Generator temperature [°C]	Evaporator temperature [°C]	Condenser temperature [°C]	*COP* [–]	*CC* [kW]
[190] T	R290	85	−15	30	0.12	–
[136] T	R113	76	27	67	0.24	3.5
[127] T	R11 R113	70–90	0–5	30–35	0.08–0.65 0.10–0.60	–
[128] T	R11	80–104	−1–20	30–55	0.15–0.42	–
[129] T	R11 R22 R114 R123 R133a R134a R141b R142b R152a RC318	90–130	10–20	25	0.10–0.25	–
[99] T	H_2O R11 R12 R113 R21 R123 R142b R134a R152a RC318 R500	80–90	−5–5	25–35	0.02–0.50	–
[206] E	CH_3OH	80–100	−2–14	16–28	0.20–0.40	0.5
[139] T	R123 R134a R152a R717	60–90	−5–14	25–40	0.05–0.45	–
[146] T	R134a R152a R290 R600a NH_3	60–90	5	24–36	0.05–0.40	–
[207] T	CH_3OH	118–132.5	−10–5	42–50	0.14–0.47	–
[120] E	H_2O	95–130	5–15	25–45	0.05–0.75	5
[119] E	H_2O	120–140	2.5–16	22–32	0.10–0.40	2
[100] E	H_2O	120–140	5–15	22–36	0.28–0.48	3
[101] T	H_2O	165	4–8	44–50	0.40–0.60	100
[147] E	R134a	65–90	2–13	26–38	0.03–0.16	0.5
[202] E	R717	62–72	5–15	30–36	0.12–0.29	2
[176] E	R245fa	100–120	8–15	30–40	0.25–0.70	4
[145] E	R123	80–105	9–15	32–37	0.22–0.50	–
[140] E	R123	83–103	0–14	29–38	0.12–0.39	2
[171] T	R142b R600a	120–130	10	20–35	0.11–0.13 0.06–0.08	10
[156] T	R290 R123 R600 R600a R134a R152a	70–100	5–15	25–35	0.30–0.85	1
[191] T	R290 R600 R600a R601 R601a R602 R602a R603 R604	70–200	10	40	0.05–0.32	–
[170] T	R141b R245fa R600a	75–125	0–16	27–43	0.35–0.42	–
[98] T	R134a R152a R290 R430A R600 R245fa R600a R1234ze R436B	75–125	0–16	27–43	0.05–0.50	5
[134] E	R113	65–80	7–12	28–45	0.16–0.24	1.6
[141] T	R123	80–90	5–10	30	0.19–0.29	–
[123] T E	H_2O	−110–130	10	30	0.3–0.47	3
[189] T	R123 R141b R142b R236fa R245ca R245fa R600 R600	85	12	32	0.4–0.7	–
[188] E	R245fa	90–100	8	29–38	0.27–0.689	12
[237] T	Acetone Benzene Cyclopentane Cyclohexane Toluene R236ea R236fa R245ca R245fa R365mfc RC318	70–200	10	40	0.05–0.6	
[180] T[a]	R245fa	60–140	7	25	0.15 to 0.35[b]	–
[260] T[a]	Water, ammonia and R134a	100–150	5	20–50	0.3–1	–

The values provided in the table represent an indicative range of the conditions considered in each study analyzed.

[a] Combined ERC – power cycle.

[b] Combined cycle first low efficiency.

heat exchanger. Appling other cooling techniques, such as water cooling in the condenser, *COP* could increase to approximately 1 (cooling the exhaust gasses from 150 °C to 100 °C, with T_c=20 °C and T_e=5 °C). Ameri et al. [261] studied a coupled 300 KWe micro-gas turbine with ERC for cogeneration and tri-generation systems, showing that this system can reduce the fuel of about 23–33%, depending on the time of the year, if compared with single plants for heating, cooling and electricity. While considering tri-generation systems, Godefroy et al. [262,263] studied tri-generation systems based on a gas engine unit and an ERC (electric power 5.5 kW_e). The authors have also shown that with accurate design and analysis, these systems can reach overall efficiencies of 50–70%.

4.1.3.3. Other configurations. Other configuration may concern the applications of ejectors to district heating systems. Sun et al. [264] have studied district heating system based on the coupled heat and power production. This system was based on ejector heat exchangers and absorption heat pumps

4.1.4. Summary

ERC have been widely studied and an intensive research in ongoing in order to improve the system performance. Indeed, ejector is the critical component of these systems: for example, Chen et al. [265] using conventional and advanced exergy analysys remarked that the system performance can be largely enhanced through improvements of the ejector. All of the previously mentioned studies are summarized in Table 4. In this table particular attention is given to the working fluids, operating conditions and performances. SERS performances strongly depend on working fluid and for each refrigerant there are appropriate operating conditions. Theoretical and experimental stu-dies show the advantages of using R134a [139,146], R152a [99], R141b [129], R142b [171] and finally R600a [191] to obtain high *COP*, working under the typical operating conditions of the ejectors. It was observed that the working fluids R134a, R152a are appropriate for heat sources at 70–80 °C and R717 is appropriate for temperatures higher than 90 °C, with R134a the working fluid with the highest *COP*=0.1/0.45. Tests over different working fluids (acetone, benzene, cyclopentane, cyclohexane, toluene, R236ea, R236fa, R245ca, R245fa, R365mfc and RC318) for high temperature heat source (T_g=70–200 °C, T_e=10°C, T_c=40 °C.) show that no one is able to cover all the operating range, and each working fluid has its own maximum ω and *COP* at a certain optimum T_g [237]. However, working fluids with limited environ-mental impact and good performance are needed and using hydro-carbon refrigerants can be a viable technical and environmental option when ensuring requisite care surrounding their flammability is taken, e.g., by developing safety procedures to use them [11]. In addition, it is very important the effect of some geometric parameters, like nozzle position and ϕ. Experimental and theoretical studies highlighted the limits of the use of fixed-geometry ejector in refrigeration cycles for low *COP* (approx. 0.2/0.3) and the difficulty in obtaining high perfor-mance under several operating conditions [102]. Concerning the nozzle shape and position an evident link between primary nozzle position and ejector performance (*COP*, CC and critical condenser pressure) in the case of a refrigerator with jet was found [101]. The importance of Nozzle Exit Position (NXP) and shape was also inves-tigated by means of CFD and experimental techniques finding the great influence of the nozzle position as ejector design parameter [60,62,208–210].

Even if a single ERS has a large range of applications, its maximum R_c, equal to 4, limits its use to air-conditioning devices [11]. Future studies are needed for improving its performance and allow a wider use of ejector for waste heat upgrade in large plants [266] and in medium/large scale refrigeration applications [224,267]. Some studies focused on the use of regenerator (also called pre-heater) and pre-cooler added to the SERC to increase the system efficiency [14]. From these studies we may conclude that the introduction of the pre-cooler and the pre-heat in the refrigeration systems seems to be a bad technical-economical choice. It could be taken into account only in particular applications, i.e. air con-ditioning in automotive field [118,212]. Some results about the use of ERS combined with a power cycle are also reported for Organic Rankine–ERC and Gas turbine–ERC coupled systems. Future study should focus on the dynamic modeling of the whole ejector based system. For example, Xue et al. [268] proposed the dynamic mod-eling of some components (i.e., heat exchangers) and the static modeling of the other components (i.e., the ejector).

4.2. Solar-powered ejector refrigeration system (SoERS)

The solar-powered ejector refrigeration system (SoERS) configuration is similar to the SERS one. In the SoERS, the thermal source is the solar thermal energy provided by a solar collector and transferred by using an intermediate working fluid to an heat exchanger. The intermediate fluid between the solar collector and the heat exchanger should have the following properties: (i) high boiling point, (ii) low viscosity and (iii) good heat transfer properties. Generally speaking, above the 100 °C oil transforming and below 100 °C water (with a corrosion inhibitor) can be used [13]. In order to evaluate SoERS performance, another efficiency defi-nition is introduced. The overall efficiency of the SoERS can be expressed as [15]:

$$COP_{overall}=\eta_{solar}COP_{ejector} \tag{11}$$

where η_{solar} is the solar collector efficiency and COP_{ejetor} is the ejector sub-cycle *COP*. Therefore, not only should the refrigeration cycle be optimized but also the solar part of the system. η_{solar} depends on the collector characteristics, the operating conditions and the radiation intensity. The collector type limits the temperature of the cycle; for further details on collector technology, the reader may refer, for example, to Charalambous et al. [269]. Although a high η_{solar} may significantly increase $COP_{overall}$, economic constraints must be considered [15]. With the proliferation of renewable energy technology, the SoERS has been widely studied, and we may divide the studies into three sub-categories:(i) standard SoERS, (ii) SoERS with a storage system and (iii) SoERS combined with a power cycle.

4.2.1. Standard SoERS

Al-Kahlidy [135] performed a theoretical screening of working fluids (R11, R12, R113, R114 and R717), proposing different refrigerant selection criteria. R113 was then chosen for the experimental setup because it has a high molecular weight and has the greater compressibility factor. For this configuration, $COP_{ejector}$ reached 0.42 (T_g = 100 °C, Te = 18 °C, Tc = 50 °C). Another comparison of SoERS using eight working fluids, was performed by Nehdi et al.[149]. The comparative study revealed that R717 provided the highest performance ($COP_{overall}$ =0.21-0.28), with an exergy effi-ciency between 0.14 and 0.19. Similar performances have been obtained by Huang et al. [163] with an R141b SoERS: the $COP_{ejector}$ obtained exceeded 0.5 and the $COP_{overall}$ was 0.22. Smierciew et al.[195,196] experimentally investigated an SoERS driven by low temperature solar heat (< 75 °C). This case is of particular interest since, in this range, the ejector cycles can be considered compe-titive with absorption refrigeration systems. In fact, 80 °C can be considered as the minimum value at which the absorption cycle can still operate, whereas there is no physical limitation for operation of the ejector systems at lower temperatures. The results confirmed that the ejector cycle operating with R600a may be used for air conditioning, powered by a low temperature heat source, either for individual or commercial households.

SoERS should be evaluated with a reference to a certain geographical area in a certain period of the year. Alexis and Karayiannis [148] evaluated the performance of an SoERS using R134a in the Athens area in summer months. η_{solar} was between 0.319 to 0.507 and the $COP_{overall}$ was between 0.011 and 0.101. The $COP_{ejector}$ was found to be an exponential function of T_g, T_c and T_e. Ersoy et al. [142] studied an SoERS using R123 in the Turkish area in August. The η_{solar} of an evacuated tube solar collector varied depending on the ambient condition and the solar radiation. Therefore, to operate with continuity, an auxiliary heat source should be employed. The maximum $COP_{overall}$ and *CC* were 0.197 and 178.26 W/m^2, respectively (T_g=85 °C, T_c=30 °C, T_e=12 °C, at 12:00). Tashtoush et al. [185], after a preliminary study on the ejector cooling cycle [238], performed dynamic hourly simulation of 7 kW of SoERC in a Jordan location. The influence of cycle parameters (ie., storage tank size, collector type, collector area and flow rate) were studied and optimized. The evacuated tube collector berformed better than the flat plate type The resulting cycle, under peak solar radiation, has $COP_{overall}$=0.32–0.47, $COP_{ejector}$=0.52–0.547 and, the efficiency of the solar collector was between 0.52 and 0.92.

Concerning the influence and the role of the collectors, Huang et al. [270] compared the performance of a SoERS using three different collectors. Small differences in solar collector efficiency can yield a proportionally larger difference in overall *COP*. Prida-sawas and Lundqvist [193] carried out an exergy analysis and optimization of the system. The largest losses are located in the solar collector and in the ejector, equal to 51% and 16% of the overall system losses, respectively. The optimum T_g is approxi-mately 80–100 °C, depending on T_e (a low temperature collector can be used). The overall thermal energy efficiency at T_g=90 °C is approximately 11%.

Variations in solar irradiation intensity are a critical issue in SoERSs that do not allow a steady T_g. If a fixed ejector geometry is used, the refrigeration cycle would not consistently provide the designed *COP*. At low ambient temperatures, the cycle is limited by choking and, and at high ambient temperatures, the ejector requires more power than can be supplied by the collector. A larger throat can accommodate a larger solar collector and a wider range of T_g, but the component may be overdesigned (especially for off-design conditions) and increases cost. In contrast, a smaller throat limits the range of T_g. For all of the aforementioned reasons, a variable area ejector is attractive. For example, a spindle can be used for maintaining a particular value for ϕ that ensures optimal performance. Ma et al. [102] controlled the primary flow using a spindle: moving the spindle toward the nozzle, the CC and the primary flow decreased. The authors reported that an optimal ω and *COP* exists and are related to the optimal ϕ. The maximum CC was found at a T_g=92.8 °C and the maximum ω and *COP* were found at T_g of 90 °C. Finally, the system performance (*CC*, ω and

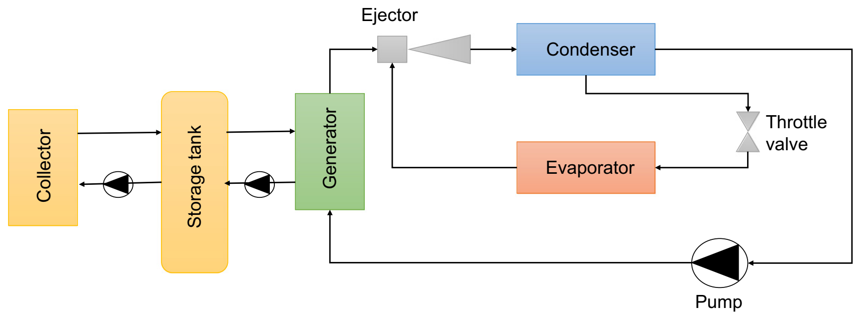

Fig. 8. Solar-driven ejector refrigeration system with hot storage tank.

COP) increase significantly with T_e, whether, the critical back-pressure increases slowly with an increase of the T_e. Another method for dealing with the transient phenomena is a variable throat ejector. A variable throat ejector was studied by Yen et al. [103] using CFD simulations using R145fa, T_c values between 35–40 °C and T_g between 90 and 110 °C. Pereira et al. [200] experimentally studied R600a ejector with variable geometry: if compared to a fixed ejector, COP would increase by 80%. The reader may also refer to the experimental and numerical studies by Varga et al. [271,272] on the topic. Dennis et al. [177] studied a SoERC using R245fa and proposed an algorithm to design a vari-able geometry nozzle diameter. This algorithm takes into account the behavior of the solar collector and the vapor generator was modeled with a fixed collector area of 16 m^2 for T_g values between 90 and 110 °C and T_e between 4 and 14 °C. A correlation was provided between the optimal nozzle throat diameter and the ambient and operating conditions.

4.2.2. SoERS with storage system

The major technical problem of SoERS is the strongly reliance of the system on environmental conditions [13]. To mitigate these negative aspects, one solution is to introduce an integrated thermal storage system for dealing with the problem of intermittent energy supply and continuous cooling demand. The storage system should have a minimum temperature variation to ensure nearly constant operating conditions and high cooling performance [273]. This solution is receiving growing attention [13]. In SoERSs, two energy storages can be applied: hot storage, (located at the solar collector side of the system) and cold storage (located at the evaporator side of the system). A cold storage can be supported by phase changing materials, ice storage or cold water [274]. Fig. 8 represents the case of hot storage tank. Therefore, the major components of the systems are: solar collector, a hot/cold storage, an ejector sub-cycle and, eventually, an auxiliary heat supply for ensuring the on-design operating conditions.

4.2.2.1. Hot storage system. Dorantes et al. [172] simulated the dynamic thermal behavior of a R142b SoERS. The obtained $COP_{overall}$ was as high as 0.34 (T_g=105 °C, T_c=30 °C, T_e=−10 °C), and the annual average efficiency was 0.21. A comparison between two periods of the year was also presented, and the average values, over the year, for the system and collector efficiency were 0.11 and 0.52, respectively. The authors compared their results with an intermittent single effect absorption system and the *COP* of the ejector cycle was similar, whereas the cycle configuration is simpler. Vidal et al. [164] conducted an hourly simulation of an SoERC with a hot water storage and an auxiliary heat source. A parametric study was conducted for selecting the optimum system size, which was found to feature a collector area of 80 m^2 with a solar fraction of 42% and a thermal capacity of 10.5 kW. The sto-rage tank size has a large influence on the auxiliary heat and a slight influence on the heat gain of the system.

Pridasawas and Lundqvist [197] studied an SoERC with R600a, selecting Bangkok as simulation location, having an average yearly $COP_{ejector}$ of 0.48. A comparison between three solar collectors is also presented: the installation cost of the flat plate collector is lower, but this system it is not economically favorable due to the auxiliary heat required. Using an evacuated tube with a collector area of approximately 50 m^2 and a hot storage tank volume of 2 m^3 for a solar fraction of 75% the *CC* was 2.5–3.5 kW. Varga et al.[104] studied an SoERC with H_2O, selecting the Mediterranean as location. For obtaining a *COP* of approximately 0.6, the T_g should not be below 90 °C, requiring a collector output temperature of approximately 100 °C (evacuated tube collectors) If the T_e is less than 10 °C, then *COP* will be less than 0.1, confirming that water may not be suitable for low temperature applications. For high values of T_c (> 35 °C) and a T_e of approximately 10 °C, the required solar collector area is greater than 50 m^2. The authors also noted that auxiliary heating is required even for 800 W/m^2 solar radiation. Guo and Shen [150] investigated office building air conditioning in Shanghai. Employing a vacuum tube collector of 15 m^2, during business hours, the average *COP* and solar fraction was 0.48. Compared with conventional compressor technologies, the solar-powered ERS can save more than 75% of electric energy. Golchoobian et al. [178] performed a dynamic simulation of a R141 system with a hot water storage tank for an office application in Tehran. As expected, the results demonstrated that a dynamic analysis provides more accurate results than a steady state analysis. The highest exergy destruction occurs in the collector and next the ejector. It is also interesting that in the first and the last hours of the days, second law efficiencies are lower. *COP* had a value around 0.1 in the first hours of the day, reached 0.7 in the middle of the day and dropped to 0.1 in the last hours of sunlight.

4.2.2.2. Cold storage system. Diaconu et al. [275] simulated an SoERS with and without cold storage located in a Algeria. Only the system with the cold storage was able to provide satisfying internal comfort conditions. The same authors [273] continued his work presenting a quantitative energy analysis on an office building For the best configuration tested, the maximum value of the cooling load was 6.6 kW and the $COP_{ejector}$ was 0.61 and the $COP_{overall}$ was 0.3. Dennis et al. [165] investigated a variable geo-metry ejector with cold storage. Without energy storage, both fixed and variable ejector systems had solar fractions up to 4% and 17%, respectively; with cold storage a variable geometry ejector was able to increase solar fractions to 8–13% greater than that for a fixed geometry ejector. Eames et al. [121] experimentally studied an ejector refrigeration cycle with a jet spray thermal ice storage system. The low T_e of this system ensures a low overall *COP*=0.162. The authors argued that this system is suitable for off-design operating conditions. Recently, Chen et al. [276] have studied (experimentally) a cold storage proving that its integration with ejector system would help keeping a more stable COP.

4.2.3. SoERS combined with a power cycle

The ejector refrigeration community is continually looking for new plant configurations for improving the performance of SoERSs. Recently, a solar-powered combined Rankine and ejector refrigeration cycle was proposed (as discussed in Section 4.1.3). In these systems, when cooling is not needed, the cycle is applied for power generation only.

Gupta et al. [122] studied a combined cycle by thermodynamic analysis (turbine inlet pressure 0.9–1.3 MPa, the $T_e = -11$ to -3 °C, $T_c = 24$–30 °C, extraction ratio 0.2–0.8 and direct normal radiation per unit area 0.8–0.9 kW/m^2). In the proposed cycle, the solar energy is exploited by means of the concentrating solar tower [277]. The results revealed that, approximately 14.81% of the inlet energy is available as useful energy output: 10.62% is the net power and 4.19% is the refrigeration output. Approximately 88.1%of the input (solar heat) exergy is destroyed due to irreversibility; the remainder, 11.36% of exergy, is associated with the net power output and 0.54% exergy is associated with the refrigeration output. The same research group [278] investigated a solar-driven triple-effect cycle. This cycle integrated three cycles: ejector, absorption, and cascaded refrigeration and has a first law efficiency equal to 11.5%. The second law efficiencies is, on the other hand, the 2% due to the losses, especially in the central receiver and, then, in the heliostat field. Another triple effect cycle powered by the solar source was proposed by Khaliq [279] using CO_2 in the refrigeration cycle. The first and second law efficiencies ranged between 33.77% and 36.06% and 2.78–2.9%, respectely, Zhang and Mohamed [199] proposed a similar plant configuration where the steam extraction to supply the ejector is downstream of the turbine. A latent heat storage unit between the combined cycle and the solar receiver is introduced for dealing with the transient phenomena and the change of conditions at night. Different refrigerants (R1234yf, R1234ze, R290, R600, R600a, R601, R744 and R134a) were evaluated and compared. R601 was found to have great potential in the proposed framework (combined power and ejector cooling cycle in hot climates) due to its high critical temperature (196.7 °C). This value accommodates a wide operating range above the ambient temperature of 40 °C. Finally, a thermodynamic analysis of the combined system has been presented and thermal and exergy efficiencies 15.06% and 19.43%, respectively, were found at $T_e = 12$ °C and $T_g = 148.83$ °C. Finally, when considering the optimization of multi effect cycles powered by solar energy, the reader may refer to the study of Wang et al. [280].

4.2.4. Summary

SoERSs are attractive systems due to their simplicity, use of solar energy and incorporation of the well-known SERS technology (refer to Section 4.1). However, there are some drawbacks that limit the system performance including the solar collector technology and the discontinuous nature of the solar energy.

The solar collector efficiency depends on the technology and further advancement will improve the performance of the whole system. Concerning the discontinuous nature of the solar energy, the performance of the system should be evaluated for one for more year(s) taking into account real ambient conditions of the selected location. SoERS should be evaluated with a reference to a certain geographical area in a certain period of the year, e.g. the performance of a SoERS using R134a in the Athens area in summer months has been evaluated [148]. Efficiency η_{solar} was between 0.319 and 0.507 and $COP_{overall}$ was among 0.011 and 0.101. $COP_{ejector}$ was found to be an exponential function of T_g, T_c and T_e.

The solar collector efficiency depends on the technology and advancement in the research.. Concerning the discontinuous nature of the solar energy, the performance of the system should be evaluated for over one or more years taking into account the real ambient conditions of the selected location. Also, prototypes should be built and tested for investigating the bahaviour of the system under variable operating conditions. The interested reader may reader to the tests performed by Huang et al. [281] for an example of this approach and for useful information.

Furthermore, the models typically employed need to be improved to account for not only the off-design operating conditions but also transient phenomena. Such work has been initially proposed by Pollerberg et al. [282] and later applied by a few authors, e.g., Golchoobian et al. [178]. A possible solution for dealing with the transient phenomena is the thermal storage; however, the storage tanks need to be carefully designed and the economical evaluation of the system should be clarified via prototypes. In SoERS two energy storages can be applied: the hot storage, (located at the solar collector side of the system) and the cold storage (located at the evaporator side of the system). A cold storage can be supported by phase changing materials, ice storage or cold water [274]. Another method for dealing with the transient phenomena is the variable throat ejector. e.g. an ejector with a movable nozzle or a movable spindle, can widen the range of operating conditions. The variable throat ejector was also analyzed by Yen et al. [103] using CFD simulations using R145fa for T_c among 35–40 °C and T_g among (90–110 °C). Dennis et al. [177] studied a SoERC using R245fa and proposed an algorithm to design a variable geometry nozzle diameter.

In recent years, coupled Rankine and SoERC systems have been proposed, and they can be energy-efficient, reliable and flexible in operation [199]. However, efforts are needed to optimize these cycles and for developing models able to consider transient phenomena in every component of the cycle. Table 5 provides a general overview about solar-driven ERS performance and operating conditions. Another proposal, different from the previous ones and not reported above, is the coupled photovoltaic-heat pump systems for water heating [283]. This system was propsed for and industry application. The system may suffer of control issues (i.e., difficulty of maintaining the vacuum required by the low evaporation temperature) and further studies are required.

In a SoERC, the *COP* of the ejector sub-cycle ranges between 0.1 and 0.5, whereas the T_g and the overall *COP* are also dependent on the collector used. In Table 6 the characteristics of the solar collector used in existing literature and, where required, the type of storage system are reported. The information contained in this table can help elucidate the influence of the efficiency of the solar system on the overall system. The collector efficiency also varied between 0.1 and 0.65, depending on the technology, the ambient conditions and the operating conditions.

4.3. Ejector refrigeration system without pump

The pump does not determine a high growth in cost or electricity consumption (i.e., in Ref. [193] the required pump power consumption is approximately 0.18% of the energy received from the solar collector). However, the pump requires more main-tenance than other parts because it is the only moving part in the system. Hence, to replace the pump, several solutions have been found:

- Gravitational/rotational ejector refrigeration system;
- Bi-ejector refrigeration system;
- ERS with thermal pumping effect;
- Heat pipe/ejector refrigeration system.

In this way, the ejector refrigeration systems acquire additional benefits, such as the potential for a very long lifetime with low maintenance, high reliability and no moving parts [105].

Table 5
Operating conditions and performance of state-of-the-art of SoERS: (T) theoretical study and (E) experimental study.

Ref.	Working fluid	Generator temperature [°C]	Evaporator temperature [°C]	Condenser temperature [°C]	$COP_{ejector}$ [–]	CC [kW]
[135] E	R11, R12, R113, R114, R717, H_2O	60–100	10–18	40–50	0.42 (max)	0.21
[163] T	R141b	80–120	−6–8	30–36	0.20–0.50	10.5
[193] T	R600	85–125	5–15	37	0.20–0.40	5
[148] T	R134a	82–92	−10–0	32–40	0.035–0.20	–
[142] T	R123	85	12	30	0.20	3.7
[149] T	R134a R141b R142b R152a R245fa R290 R600 R717	90	15	35	0.30–0.41	–
[102] E	H_2O	84–96	6–13	21–38	0.17–0.32	5
[195] E	R600a	50–64	4–7	22–32	0.15–0.20	2
[103] T	R245fa	90–110	12–20	35–40	0.2–0.55	10.5
[172] T	R142b	105	−10	30	0.34	2
[164] T	R141b	80	8	32	0.39	10.5
[197] T	R600a	70–120	5–15	$T_{amb}+5$	0.35–0.48	3.5
[104] T	H_2O	90–110	5–15	30–40	0.10–0.55	5
[150] T	R134a	85	8	$T_{amb}+\Delta T$	0.30–0.53	6
[165] T	R141b	80–110	2–14	20–40	1.5 (max)	3.5
[121] E	H_2O	110–135	2.5–10	21–30	0.5 (max)	–
[122] T	H_2O	150	−11 to −3	24–30	$\eta_I=0.148$[a]	–
[199] T	R1234yf, R1234ze, R290, R600, R600a, R601, R744	150	12	*50*	$\eta_I=0.151$[a]	–
[200] E	R600a	83	9	21–29	0.2–0.58	–
[238] T	R717 R134a R600 R600a R141b R152a R290 R123	80–100	8–12	28–40	0.59–0.67	
[185] T	R134a	26 bar	8	30	0.52–0.547	7
[178] T	R141	85	35	8	0.1–0.7[b]	5

The values provided in the table represent an indicative range of the conditions considered in each study analyzed.

[a] Solar-powered combined Rankine and ejector refrigeration cycle.
[b] Dynamic simulation.

Table 6
Characteristics of the solar collector used and the of storage system (where required).

Ref.	Solar collector and storage system	Solar radiation intensity [kW/m²]	Efficiency [%]	Area [m²]
[135]	Parabolic trough concentrator	0.762–0.874	20	15
[163]	Double-glazed selective surface flat-plate solar collector	0.7	50	68
[193]	Double-glazed selective surface flat-plate solar collector	0.7	48	–
[148]	Evacuated-tube solar collector	0.536–0.838	31.9–50.7	–
[142]	Evacuated-tube solar collector	0.200–0.896	28–36	19.7–21.5
[149]	Single-glazed selective surface flat-plate solar collector	0.351–0.875	40	–
	Double-glazed selective surface flat-plate solar collector		50	
	Evacuated-tube solar collector		65	
[102]	Evacuated-tube solar collector	–	–	–
[172]	Evacuated-tube solar collector+hot liquid storage tank	0.311	52	18
[164]	Single-glazed selective surface flat-plate solar collector+hot liquid storage tank	–	–	80
[197]	Evacuated-tube solar collector+hot liquid storage tank	–	47	50
[104]	Evacuated-tube solar collector+hot liquid storage tank	0.8	–	50
[150]	Evacuated-tube solar collector+hot liquid storage tank	0.2–0.9	–	15
[165]	Evacuated-tube solar collector+cold storage system	–	–	12–22
[122]	Heliostat for solar tower CSP	0.8–0.9	75	3000
[185]	Evacuated-tube solar collector	0.2–1.1	0.52–0.92	60–70
[178]	Evacuated-tube solar collector	0.1–0.9	10–65	–

The values provided in the table represent an indicative range of the conditions considered in each study analyzed.

4.3.1. Gravitational/rotational ejector refrigeration system

The layout of a gravitational ejector refrigeration system refrigeration cycle is presented in Fig. 9. Kasperski [107] proposed a gravitational ejector. In this configuration, the heat exchangers are placed on different vertical positions, equalizing the pressure differences between them. The steam generator has the highest pressure, and the evaporator has the lowest pressure. There are also complex mechanisms of self-regulation of the generator, evaporator and condenser. A major drawback of the system is the requirement of height differences (depending on the working fluid and on the temperature differences) and the length of pipes (which causes high friction and heat losses). At T_g=80 °C, Tc=35 °C and Te=15 °C, the *COP* is 0.16. The same author [108] developed the concept of the gravitational ejector into a rotating ejector, which is able to decrease the size of the gravitational refrigerator and the amount of working fluid (at, for example, approximately 1000 rpm). The performance is similar to those of the gravitational ejector [107]: *COP*=0.16 (T_g=90 °C, T_c=35 °C, T_e=15 °C). Nguyen et al. [105] investigated a solar ERS based on the natural convection: gravity ensures the liquid recirculation from the condenser to the boiler (height of the system was above 7.5 m). The system was proposed for air-conditioning use with used water as the refrigerant. This system also provides heating in the winter season and was evaluated and installed in an office building in England. The prototype system had a nominal *CC*=7 kW and operated with a *COP* of up to 0.3. The investment payback period was 33 years, and the economic performance was analyzed for future market viability. In addition to the economic aspects, this

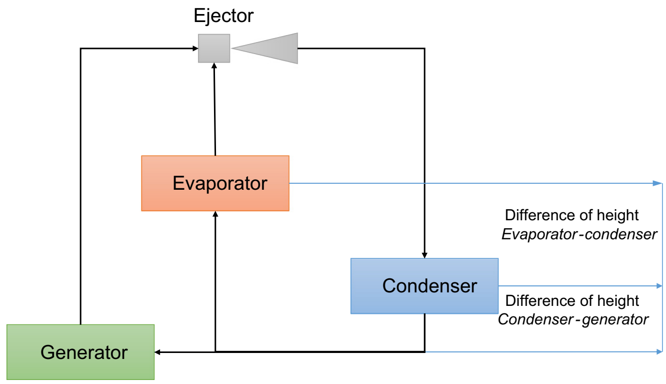

Fig. 9. Gravitational ERS.

system has other critical factors, in particular, the large thermal inertia, which affects the start-up and shut-down performance. Moreover, the use of an additional burner is required during off-design operation for additional heating and to avoid thermal transients.

4.3.2. *Bi-ejector refrigeration system*

In the bi-ejector refrigeration system (BERS), a second ejector, which replaces the pump, carries the liquid condensate to the generator. Therefore, the ejector is a vapor/liquid ejector. The layout of a BERS is presented in Fig. 10. During ideal operation, this system does not consume electricity, which makes it attractive. Shen et al. [106] numerically studied this configuration, and the numerical results showed that the cycle *COP* is mainly influenced by ω for all the tested refrigerants (R11, R12, R22, R134a, R123, R502, R717 and H_2O). The highest *COP* was 0.26 using R717. However, Wang and Shen [179] investigated a solar BERS using R123. They showed that increasing generation temperature ω of the two ejectors results in different behaviors: one increases and the other decreases. Therefore, the overall thermal efficiency of the cycle has an optimum value equal to 0.13 (T_g = 105 °C, T_c = 35 °C, T_e = 10 °C). With increasing T_c, the ω of the two ejectors and the system efficiency decrease. Yuan et al. (2014) investigated a bi-ejector absorption power cycle with two ejectors for an ocean thermal energy conversion. Ammonia–water is used as the working fluid, and the ejectors are driven by vapor and solution from the sub-generator. The results show that the absorption temperature is increased by 2.0–6.5 °C by using the bi-ejector ejector cycle if compared with a single ejector cycle. The proposed cycle is investigated by the first law and the second law: this cycle can reach to 3.10% and 39.92%, respectively (49.80% of exergy loss occurs in the generators and reheater, followed by the 36.12% of exergy loss in the ejectors).

4.3.3. *ERS with thermal pumping effect*

ERS with thermal pumping effect may be multi-function generator (MFG) or workless-generator-feeding (WGF). Huang et al. [166] proposed a multi-function generator (MFG): the system includes two generators constituted by a boiler and an evacuation chamber. The boiler heats the liquid, and the evacuation chamber provides a cooling effect. The system is composed of many elements, which leads to a consumption of thermal energy. The experimental results reported *COP*=0.22 (T_g=90 °C, T_c=32.4 °C, T_e =8.2 °C), without considering the extra heat required for the MFG operation. Taking into account the required extra heat, the total *COP* is observed to decrease to 0.19. To replace R141b, Wang et al. [143] designed the ejector system to work with R365mfc. In particular, the authors showed that R365mfc can replace R141b while maintaining the performance of the system. At T_g=90 °C, $COP_{ejector}$ =0.182–.371, the total *COP* = 0.137 to 0.298, and *CC* = 0.56 kW to 1.20 kW for T_e = 6.7 to 21.3 °C. Srisastra et al. [183,184] presented a workless-generator-feeding (WGF), using R141b, system working without a pump. This system is based on filling phase and feeding, controlled by a system of valves. Another thermal pumping system, activated by solar energy, was presented by Dai et al. [151], reaching a *COP* = 0.13.

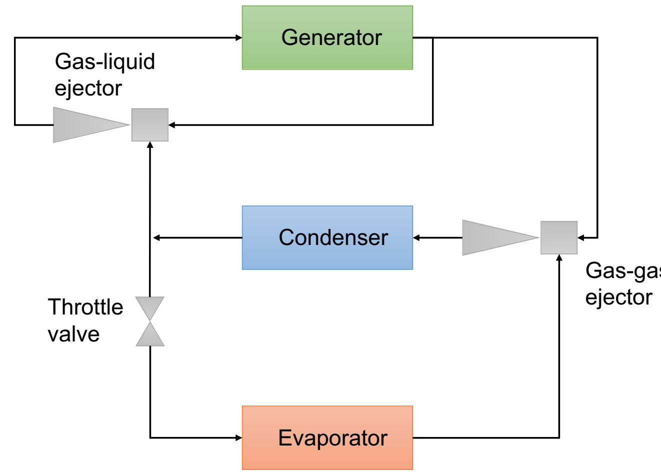

Fig. 10. Bi-ejector refrigeration system without pump proposed by Wang and Shen (2009).

4.3.4. *Heat pipe/ejector refrigeration system.*

An interesting technology is the coupling between the ejector and the heat pipe. The coupling of the heat pipe and the ejector technology is interesting because it results in a system that is both compact and with high performance. This system is composed of a heat pipe, an ejector, an evaporator and an expansion valve; the working principles will not be described here because they are the same as those of other ejector refrigeration systems. A description can be found in the work of Smirnov and Kosov [284]. Riffat and Holt [109] performed computer modeling of the system using ethanol, methanol and water. The *COP* of methanol was higher than that of the other fluids, approximately 0.7. In general, $COP \approx 0.5$ is achievable using low-grade heat operating conditions. A heat pipe/ejector system for air-conditioning and building cooling was proposed by Ziapour and Abbasy [110] using energy and exergy analysis. The simulation results indicate that *COP* = 0.30 (T_e = 10 °C, T_c = 30 °C, and T_g = 100 °C) and the maximum *CC* could be obtained for heat pipes with large diameters. Finally, another system, with a vertical arrangement of the ejector, was proposed by Ling [285].

4.3.5. *Summary*

Ejector refrigeration systems without the use of a pump are very interesting due to the prospects of energy saving. The performances of the plant configurations that do not involve the use of a mechanical pump are summarized in Table 7. All the proposed systems are interesting, but the performances are low and there is a lack in experimental large scale works and modeling techniques. Only the gravitational and the ERS with thermal pumping effect have been experimentally studied. Solar ERS based on the natural convection have an investment payback period of 33 years and present criticalness, in particular the large thermal inertia, which affects the start-up and shut-down performance. Moreover, the use of an additional burner is required during off-design operation for additional heating and avoid thermal transient. Among the different alternatives, the gravitational/rotational cycle is interesting and can be used in different applications (i.e

Table 7
Operating conditions and performance of state-of-the-art of ERSs without the use of a mechanical pump: (T) theoretical study and (E) experimental study.

Ref.	Technology	Working fluid	Generator temperature [°C]	Evaporator temperature [°C]	Condenser temperature [°C]	*COP* [–]	*CC* [kW]
[105] E	Gravitational ejector refrigeration system	H_2O	90	10	35	0.30	7
[151] E	Thermal pumping system	R134a	75–80	10–18	31–36	0.08/0.13	1.5
[106] T	Bi-ejector refrigeration system	R11 R12 R22 R134a R123 R502 R717 H_2O	75–100	3–15	28–40	0.04–0.26	–
[179] T	Bi-ejector refrigeration system	R123	80–95	7–15	30–39	0.15–0.30	–
[143] E	Multi-function generator	R365mfc	90	6.7/21.3	40	0.182/0.371	0.56–1.20
[107] T	Gravitational ejector refrigeration system	H_2O	80	15	35	0.16	0.12
[108] T	Rotational ejector refrigeration system	H_2O	90	15	35	0.16	0.08
[166] E	Multi-function generator	R141b	90	8	32	0.22	0.8
[109] T	Heat pipe/ejector refrigeration system	H_2O CH_3OH C_2H_5OH	80–100	5–10	24–32	0.40–0.70	–
[110] T	Heat pipe/ejector refrigeration system	H_2O	90–100	10–15	30–32	0.30–0.50	1–5.5

The values provided in the table represent an indicative range of the conditions considered in each study analyzed.

air-conditioning, food storage, internal cooling of rotors and so on), but there are some drawbacks to be addressed, such as the difficulties in the experimental studies (also because of the difficulties, due to damaged measuring sensors and disturbance in the electric signals by the sliding contacts). However, it should not escape notice that the roto-gravitational system needs a rotating cylinder driven by electricity. Therefore, this system replace the pump, but still need electricity. The most promising system appears to be the integrated heat pipe/ejector system: the expected *COP* is similar to the one of absorption systems, but in heat pipe/ejector system is cheaper, with low maintenance, compact and without moving parts [109]. Unfortunately, experimental investigations are not available

4.4. *Combined ejector–absorption refrigeration system (EAbRS)*

The main components of an absorption refrigeration system are the pump, the generator and the absorber. A detailed description of an absorption cycle will be not presented here because it has been well detailed elsewhere [3,13]. In an absorption system, almost any type of heat source can be utilized. This system is, however, more complex and has a lower *COP* compared to con-ventional vapor compression systems. Adding an ejector (thus developing the "Combined ejector–absorption refrigeration sys-tem", EAbRS) can improve the system efficiency by, for example, increasing the refrigerant flow from the evaporator. Moreover, the EAbRS is quite simple, has low investment cost and the resulting systems have generally high *COP* [13].

EAbRS may be divided into two sub-categories: (i) standard EAbRS, (ii) EAbRS SERS combined with a power cycle. In the following, for each section, we present a comprehensive collection of all existing literature regarding these systems.

4.4.1. *Standard EAbRS*

One of the first studies of the EAbRS was proposed by Chen [132], who studied an EAbRS in which the ejector outflow is sent to the absorber (Fig. 11). The system is highly dependent on the ejector geometry, and the optimum ϕ yields a maximum $COP=0.85$, while the performance of a conventional cycle is $COP=0.68$ under the same conditions ($T_g=120$ °C, $T_c=40$ °C, and $T_e=5$ °C). By reducing the condenser temperature to $T_c=30$ °C, *COP* reaches the maximum value of $COP=1.5$. Sozen and Ozalp [112] proposed a solar-driven (Turkey region) EAbRS; using the ejector at the absorber inlet, the *COP* improved by approximately 20%, reaching 0.6–0.8. The influence of the ejector geometry over the cycle performance was studied by Vareda et al. [286]. The authors reported that the activation temperature decreased if compared with a conventional single-effect absorption cycle and COP increased for medium temperatures. An analysis of the performance of this configuration was also proposed by Sozen et al. [287,288] using different nnumerical methods. A comparison of this configuration and single/stage was proposed by Jelinek et al. [289] and Garousi Farshi et al. [290] showing an increase of performance (first and second law) and lower activation temperatures. Performance enhancement can be achieved placing the ejector between the generator and the condenser, as proposed by Sun et al. [111] (Fig. 12). The authors found that the EAbRS using a high generator temperature ($T_g=220$ °C) can have high *COP* ($COP=2.4$). This value is approximately twice that of a conventional single-effect absorption machine. However, the required generator temperatures cannot be easily reached using low-grade energy sources. This system has better performance is compared to the previous one (Fig. 11), as confirmed by experimental and numerical investigations (i.e., COP increase form 0.274–0.382 to 1.099–1.355, under the same temperature range of the generator and evaporator) [111,291,292].

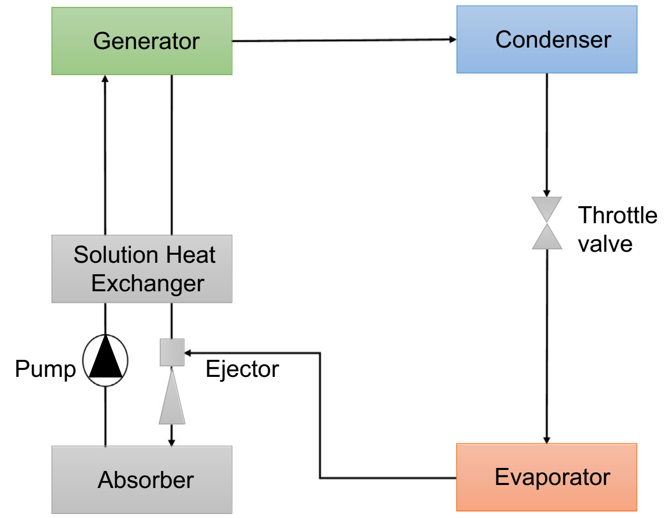

Fig. 11. Combined ejector–absorption refrigeration system (EAbRS): ejector outflow is sent to the absorber.

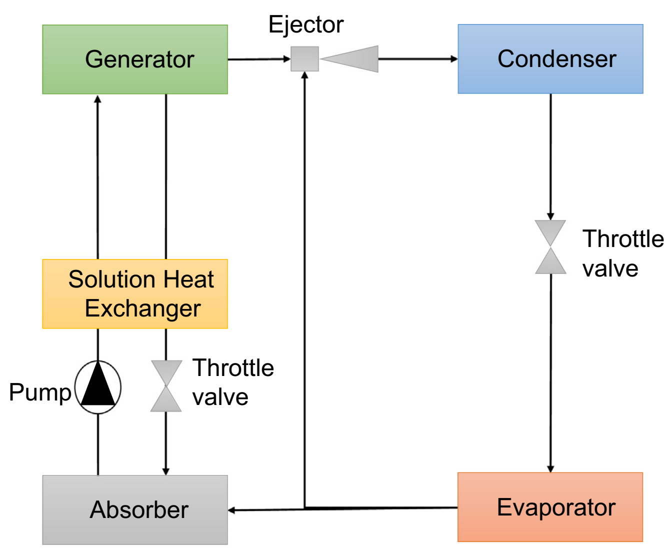

Fig. 12. Combined ejector–absorption refrigeration system (EAbRS). ejector outflow is sent to the condenser.

Some other configurations and comparative studies have been proposed in the literature. Hong et al. [113] proposed a modified EAbRS: this cycle functions as a double-effect cycle for high heat source temperature and as a single-effect cycle for lower temperature. As a consequence, *COP* is 30% higher than a conventional single-effect cycle. Sirwan et al. [204] proposed using a flash tank between the condenser and the evaporator to improve both ω and the cooling effect. In particular, the study is focused on the case of the use of solar energy, where the performance is limited by the solar collector (heat source) and the range of the high ambient temperature *COP* of the modified cycle (0.49–0.86) is higher compared to that of a combined absorption–ejector cooling cycle (0.42–0.75) and of the basic absorption cycle (0.18–0.575). Jelinek, and Borde [293] studied a single- and double-stage cycle with different working fluids (fluorocarbon refrigerants and organic absorbents). A system with a concentrator has been proposed by Eames and Wu [294,295] by numerical and experimental investigations ($COP=1.03$ in the experimental investigation). Vereda et al. [296] studied a single-effect absorption refrigeration cycle coupled with a triple purpose ejector (i.e., pressure booster, adiabatic absorber and controlled solution expansion valve): this configuration was found to have improved CC and lower activation temperature. Abed et al. [297] propose internal heat recovery for enhancing the system performance (ie., the COP was reported to increase by the 12.2%). Jiang et al. [208] compared, via a thermo-economic analysis, three EAbRSs and a double-effect absorption cycle. The former system has a value of *COP* of up to 0.9–1.0 ($T_g=160$ °C), which is slightly lower than that of the commercial double-effect absorption refrigeration system. A comparative study of the working fluids was performed by Jaya et al. [152], considering R124-DMAC, R134a-DMAC and R32-DMAC. The use of R124-DMAC and R134a-DMAC provided $COP \approx 1.0$ at low temperatures of the generator ($T_g=100$ to 110 °C) and evaporator ($T_e=5$ °C). R32-DMAC has some drawbacks: high circulation ratios and high generator pressures.

4.4.2. *EAbRS combined with a power cycle*

Also EAbRS can be coupled with power cycle. Wang et al. [114] presented a combined EAbRS with a Rankine cycle; this system could produce both power ($P=612.12$ kW) and refrigeration ($CC=245.97$ kW) outputs. The various performance metrics of the cycle (i.e., refrigeration output, net power output, and exergy efficiency) are highly influenced by the operating conditions (i.e., generator, condenser and evaporator temperature, turbine inlet and outlet pressure, and solution ammonia concentration). Khaliq et al. [298] investigated a coupled power and EAbRS: the coupled systems provide approximately 22.7% of the input exergy and 19.7% of the input energy available as the useful output. Finally, Kumar [299] investigated an EAbRS using an R-152a ejector on cycle and a LiBr-H_2O absorption cycle integrated with a renewable energy power generator. The useful exergy and energy output are approximately 7.12% and 19.3%, respectively. Khaliq [300] investigated a multieffect cycle based on an ORC, an ejector–absorption cycle and ejector expansion Joule–Thomson (EJT) cycle. The first and secon law effciiencies were 22.5% and 8.6% respectely. The criogenic cicles are detaile din Section 4.7.4. Yang et al. [301] studied a a coupled power and EAbRS using zeotropic mixture. The authors have studied the second law efficiency as function of the mixture used as working fluid: the maximum efficiency was 7.83%,

4.4.3. *Summary*

Summarizing the above studies, the coupling of the absorption cycles and the ejector component combines the advantages of two systems, and the resulting systems exhibit high values of *COP* (0.4–2.4). However, the *COP* of the system strongly depends on the ejector performance [113] and, therefore, detailed models for the off-design of the component should be developed along with an optimization of the ejector geometry [132]. When considering hot climates, in which the condenser has a lower efficiency, the solution proposed by Sirwan et al. [204] may enable the system to perform well. A summary of the EAbRS studies is presented in Table 8.

4.5. *Combined ejector–adsorption refrigeration system (EAdRS)*

It is well known from the literature that the absorption and the adsorption processes differ from each other. The former is a surface phenomenon, and the latter is a volumetric phenomenon [3]. In an adsorption system, the main component is a porous surface, which is able to provide a large surface and a high adsorptive capacity. The detailed analysis of the adsorption process is, of course, far beyond the scope of this paper; however, for the sake of clarity, some explanations will be provided. The adsorption process can be divided in different phases. Initially, the surface is free of molecules. Subsequently, a vapor molecule approaches the surface and, via an interaction, the molecule is adsorbed onto the surface. The molecule then releases energy because of the exothermic adsorption [2, 3]. In an adsorption cycle, there are both adsorption and desorption processes. In a real system operation, at

Table 8
Operating conditions and performance of state-of-the-art of EAbRS: (T) theoretical study and (E) experimental study.

Ref.	Working fluid	Generator temperature [°C]	Evaporator temperature [°C]	Condenser temperature [°C]	*COP* [–]	*CC* [kW]
[132] T	DME-R22	120–180	5	30–50	0.5–1.5	–
[111] T	LiBr–H_2O	–180–240	5–15	22–40	0.7–2.4	–
[208] T	LiBr–$ZnCl_2$–CH_3OH	170	7	42	0.9–1.0	30
[112] T	NH_3–H_2O	50–130	−5–5	25–40	0.6–0.8	–
[152] T	DMAC-R32 DMAC-R124 DMAC-R134a	70–140	−5–15	20–34	0.4–1.2	–
[113] T	LiBr–H_2O	120–150	5	40	0.8–1.2	–
[114] T	NH_3–H_2O	62	−5	31	–	858 ($CC+P_{el}$)
[204] T	NH_3–H_2O	65–120	–14–14	20–50	0.4–0.85	–

The values provided in the table represent an indicative range of the conditions considered in each study analyzed.

Table 9
Operating conditions and performance of state-of-the-art of EAdRS: (T) theoretical study and (E) experimental study.

Ref.	Working fluid	Generator temperature [°C]	Evaporator temperature [°C]	Condenser temperature [°C]	*COP* [–]	*CC* [MJ/kg]
[115] T	13 × -H_2O	120	10	40	0.4	–
[116] T	13 × -H_2O	150–200	5	30	0.33	0.15–0.34

The values provided in the table represent an indicative range of the conditions considered in each study analyzed.

least two beds are necessary to ensure the continuity of the process. Li et al. [115] studied an EAdRS (zeolite 13X-water system); the authors focused on the problem of the intermittence of adsorption refrigeration, taking into account the processes occurring during daytime and nighttime. The authors demonstrated that $COP_{ejector}$ increases with increasing the temperature or decreasing the pressure of the adsorbent. Zhang et al. [116] analyzed a solar-powered EAdRS coupled to an heating hybrid system; when the high temperature in the adsorbed can be used for heating water, the value of *COP* was 0.33, corresponding to an improvement of 10% compared with a system without ejector. A prototype of this system was also designed.

Generally speaking, taking into account the theory of the adsorption process, the following should be considered: reducing the pressure or increasing the temperature of the adsorbent can increase COP_{ejector}. Finally, we may state that the main problem of this cycle is the intermittent effect over *COP* and CC. Table 9 summarizes the results of the above studies. Despite this system could be interesting, there is a very limited amount of research and no experimental data is available at this moment. Future studies should clarify the performance of the system under a wider range of operating conditions and perform a better comparison of this system and the other technologies.

4.6. Combined compression–ejector refrigeration system

According to the function performed by the ejector, there are two types of combined compression–ejector refrigeration systems. In the first type, the ejector still has the goal of increasing the working fluid pressure into the cycle. In the second type, a two-phase ejector acts as an expansion device to improve the performance of vapor compression refrigeration systems. Two sub-categories will be presented in the next sections: (i) vapor compression-ejector refrigeration system (CERS) and (ii) ejector expansion refrigeration system (EERS). However, a brief explanation is required to clarify some aspects concerning the approach followed in this paragraph. In 1990, Sokolov and Hershgal [137] first proposed the CERS in various plant configurations, for ejector-compression refrigeration systems. Among these technologies, the more interesting type is the combined ejector-compressor refrigeration cycle, consisting of a standard ejector and a vapor compression refrigeration system in the cascade configuration. The second sub-category is the ejector expansion refrigeration system. In this plant configuration, in which the ejector assumes a new role, the compressor cannot be replaced. Therefore, the EERS will be presented inside this section.

4.6.1. Vapor compression–ejector refrigeration system (CERS)

In a CERS, the *COP* is still defined as the cooling effect and the total incoming energy in the cycle ratio, which, in this case, also includes the electric work consumed by the compressor or the booster. However, a different definition of the *COP* in the CERS is necessary to represent the real economics [137] with a more direct economic implication, for which COP_{mec} is defined as:

$$COP_{mec}=\frac{Q_e}{L_c}=\frac{Q_e}{L_{pump}+L_{compressor}} \tag{12}$$

In this way, the ERS increases its range of application and increases its efficiency with a reduced electrical requirement for the mechanical compression refrigeration system.

Sokolov and Hershgal [137] suggested two basically different approaches to improve the *COP* of the ejector refrigeration system. These approaches are based on the dependency of the ejector performance on the secondary flow pressure, and if all other cycle parameters are constant, an increment of the secondary flow pressure can cause an increase in either condenser pressure or ω. In the remainder of this section, the main studies concerning CERS are detailed to analyze the evolution from the initial configura-tions to the most recent proposed configurations.

The first configuration proposed is the booster assisted ejector cycle: similar to conventional ERS, but with a pressure booster compressing the secondary flow before entering in the ejector (e.g., Dorantes et al. [172], Fig. 13). The value of *COP* is improved ($COP=0.767$, more than double the *COP* of the SERS), but the coupling of the booster and ejector in series may cause control issues.

The second configuration proposed is a coupled ejector-compressor refrigeration cycle. The bottoming cycle is a conventional ERS or a booster ERS, while the topping cycle is a vapor compression cycle moved by a compressor. In this configuration, the heat (and eventually the mass) is transferred between the two cycles in an inter-cooler, which replaces the evaporator of the ejector cycle. This arrangement can reduce the variability of the working conditions and guarantee more stable operating conditions.

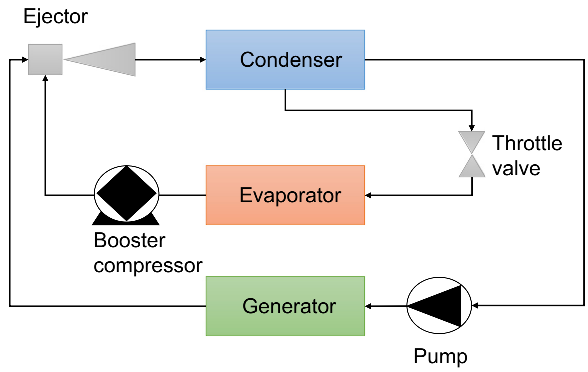

Fig. 13. ERS with a booster compressor.

Moreover, considering a single refrigerant, the intercooler may combine both heat and mass transfer, thereby providing inter-bal-ancing effects of the thermodynamic state in each of the cycles. Otherwise, the intercooler is only a heat exchanger, permitting the use of two refrigerants and the selection of the most appropriate refrigerant for each subsystem.

In 1993, Sokolov and Hershgal [138] developed a single-refrigerant compression enhanced refrigeration system, in which the inter-cooler allows for both heat and mass transfer. They demonstrated that this system could operate using solar energy, but to enhance the system availability, the use of storage is recommended in this case. In particular, the authors suggested the use of a cold storage tank because the hot storage approach is wasteful due to the low-thermal system efficiency. This system configuration has been widely studied. Indeed, the same system was studied by Arbel and Sokolov [173] but using R142b as the working fluid. According to the authors, a combined CERS with moderate condensing temperatures producing air-conditioning, hot water, and solar space-heating could be a very feasible and economical system. Hernandez et al. [174] tested R142b and R134a on the same systems, driven by solar energy and considering the ice production application: the system using R134a at a moderate T_c (approx. 30 °C) exhibited the best performance, while the use of a higher T_c with R142b provided better performance.

Sun [117] proposed a solar-driven combined CERS for air-conditioning and refrigeration purposes. The refrigerant in the ejector sub-cycle is water when the refrigerant in the vapor compression sub-cycle is R134a. The combined cycle shows a potential increase of the system *COP* (50% over the conventional cycles) and a decrease of the electrical energy requirements (to half of the conventional cycles). Sun [118] evaluated a combined CERS for refrigeration and an air-conditioning operating with single or dual refrigerants. To identify suitable dual refrigerants, azeotrope R500, CFCs (R11, R12, R113), HCFCs (R21, R123, R142b), HFCs (R134a, R152a), organic compound RC318, and water (R718) are used in combined systems. Numerical results demonstrated an improvement of performance and achievement of *COP* ($COP=0.8$) values similar to the single-effect absorption system ones ($COP=0.6$–0.8). Considering the cost of the waste heat used for supplying the system as being negligible, the *COP* can be higher. The performance can be further increased if dual refrigerants are used, with the optimum pair composed of R718 for the ejector cycle and R21 for the vapor compression cycle. Another CERS powered by the solar source was presented by Vidal and Colle [168], who performed a study with hourly simulation and thermo-economical optimization of a solar CERS with a thermal storage tank. R141b and R134a were used as the working fluids for the ejector and compressor cycle, respectively. The final optimized system of 10.5-kW cooling capacity has a flat plate collector of area of 105 m^2 and an inter-cooler temperature of 19 °C, resulting in a system solar fraction of 82% and a value of *COP* equal to 0.89.

A combined CERS moved by waste heat and with a pre-cooler in the bottom cycle was built and tested by Huang et al. [167]. The working fluids used are R22 in the topping cycle and R141b in the ejector cycle. The *COP* can be improved by 24%, with potential for further improvement because the prototype does not operate at optimal conditions.

Worall et al. [209] designed a hybrid jet-pump compression system with carbon dioxide for transport refrigeration; a hybrid system was simulated, and its performance was determined for different operating conditions and optimized using entropy generation minimization. The jet-pump circuit working fluid of methanol was used to recover heat from the discharge gases and vehicle exhaust and to sub-cool the CO_2 transcritical sub-system. Sub-cooling improved the refrigeration effect, reducing the gas cooler outlet temperature below the critical point and thus improving heat transfer. The temperature of exhaust gases from the engines varies from 300 °C to 500 °C, and consequently, the available heat is variable, depending on the cooling capacity and hence the engine power output.

Zhu and Jiang [133] proposed CERS using different working fluids. The simulation results demonstrated that *COP* increased by 5.5% with R152a and 8.8% with R22 when compared with the basic system. The value of *COP* of the hybrid system increases with T_e and decreases with T_c, as in the basic vapor compression refrigeration system.

Mansour et al. [153] compared a conventional vapor-compression refrigeration system, a boosted assisted ERS and a combined CERS at fixed evaporation, condensation and boiling temperatures. Considering nominal conditions of cooling capacity equal to 5 kW, the boosted ERS and the cascade CERS show interesting performance: the compression ratio substantially decreased with work decreasing early by 24% and 35%, respectively. Consequently, performance is improved by 21% and 40%over the reference for the same capacity.

Šarevski et al. [124] studied a double stage R718 CERS: the first stage was provided by a centrifugal compressor and the second stage was provided by two-phase ejector. The proposed system has $COP_{mec}=5.4$–8.3 ($T_e=10$ °C, $T_c=35$ °C), depending on the ejector component efficiencies.

Also, for CERS systems, cogenerative systems have been proposed. For example, Petrenko et al. [194] proposed a micro-tri-generation system composed of a cogeneration system and a cascade refrigeration cycle (the coupling of a CO_2 compression refrigerating system, and a R600 ejector cooling system). The *CC* was 10 kW and the $COP=1.4$ when the system is operating under the design conditions.

Applying a CERS, instead of a SERC, improve the performance of the refrigeration cycle ($COP=0.2$–1.52, depending on the systems). Future studies may concern the economical evaluation of the CERNS technology in comparison with SERC. Also, an exergy analysis using the same framework, may evaluate the advantages of CERS. However, as CERS requires electricity as input, the evaluation of these systems should be performed taking into account the energy system of the country analyzed. For example, Italy has higher electricity cost if compared to other countries, or developing countries have lack of energy access. Table 10 summarizes and compares the above-mentioned studies.

4.6.2. Ejector expansion refrigeration system (EERS)

The performance of a compression refrigeration cycle can be improved using an ejector as the expansion device (EERS) instead of the expansion valve (isenthalpic process). An ejector may reduce both expansion irreversibility and the compression work (raising the suction pressure), thus leading to a *COP* improvement. Both expansion valve losses and compressor superheat losses have

Table 10
Operating conditions and performance of state-of-the-art of CERS: (T) theoretical study and (E) experimental study.

Ref.	Working fluid	Generator temperature [°C]	Evaporator temperature [°C]	Condenser temperature [°C]	*COP* [–]	COP_{mec} [–]	*CC* [kW]
[137] E	R114	86	−8	30	0.77	8.1	2.9
[138] T	R114	76	4	50	0.85	5	3.5
[117] T	H_2O–R134a	110–140	5–15	35–45	0.3–0.4	5–7	5
[118] T	R11 R142b R12 R134a R21 R152a R113 R123 RC318 H_2O R500	70–100	5	35–45	0.5–0.8	–	–
[167] E	R141b–R22	68	−5–5	35–55	0.5–0.8	1.9–2.6	3.9
[173] T	R142b	100	4	50	0.32 − 1.52	5 − 20	3.5
[174] T	R142b R134a	80-115	−10	30–40	0.1–0.5	–	1
[168] T	R141b–R134a	80	8	32–34	0.8–0.9	–	10.5
[209] T	CH_3OH–R744	90–140	−15	35	0.8–1.3	1.3–3	3
[194] T	R600–R744	80–140	−40 to 0	28–40	0.4–0.9	2.5	10
[153] T	R134a	90	0	40	–	4.49 5.21	5
[133] T	R134a R152a R22	90	−10–10	45–55	0.6–0.7	2.2–2.4	7–12
[124] T	R718	–	35	10	–	5.4–8.3	–

The values provided in the table represent an indicative range of the conditions considered in each study analyzed.

important effects on the cycle *COP*. With the ejector expansion cycle, the expansion valve losses are reduced. Thus, potential refrigerants, which are unacceptable due to large expansion valve losses in a standard vapor-compression cycle, may be much more attractive when used in an ejector expansion cycle [12]. The ejectors used are two phase ejectors, which introduces modeling difficulties and challenges in the manufacturing of the system. Kornhauser and Menegay [302] patented a solution for increasing the velocity of the motive nozzle flow based on the bubble breakup at the nozzle entrance. Another study of the two phase flow in a nozzle is the report of Nakagawa and Takeuchi [303], who studied the divergent length of the nozzle. Longer nozzles would allow the two-phase flow to reach equilibrium, thus increasing the performance. The authors also investigated the throat diameter, showing that with an increase of the throat dia-meter, the CC, *COP* and ω all increase.

The first proposal of this configuration dates back to 1931, with the patent of Gay [304]. However, Kornhauser [130] first analyzed the EERS using different working fluids (R11, R12, R22, R113, R114, R500, R502 and R717). To compare the performance of the EERS with the standard vapor-compression cycle, simulations of the two cycles were conducted for the same values of T_e, T_c, compressor efficiencies, and heat loads. The improvement in *COP* with the ejector expansion sys-tem varies from refrigerant to refrigerant because the sources of loss in the standard vapor-compression cycle vary (+12 to 30%). For some refrigerants, such as R717 (*COP*=5.33), a large part of the loss is due to heat transfer from the superheated vapor: the potential increase in *COP* by reducing the loss in the expansion process is limited. For other refrigerants, such as R502 (*COP*=5.67), little discharge of superheat occurs and almost all the loss is in the expansion process. For these refrigerants, the potential increase in *COP* with the ejector expansion cycle is much greater and, in fact, R502 had the highest *COP* improvement compared to the other refrigerants. The *COP* improve-ment decreases when T_e increases. Also Nehdi et al. [161] compared different working fluids and focused particularly on synthetic refrig-erants (R134 a, R141b, R142b and R404A); the best *COP* improvement (+22%) was obtained with R141b. The authors also studied the dependence of the optimum ejector parameter for the operating temperatures and studied the influence of ϕ on and T_e.ForagivenT_e, the *COP* of the standard cycle decreases much more than the *COP* of theEERSwhenT_c increases, and vice versa. Sarkar [192] compared natural refrigerants (R290, R600a, R717) and observed that the use of R600aandammoniaguarantee themaximumandminimum performance increase, respectively. Furthermore, the dependence on the ejector parameters was studied: the optimum ϕ increases with T_e and decreases with T_c, whereas the *COP* improvement compared to the basic expansion cycle increases with the increase in T_c and decreases when T_e increases.

Concerning, the effect of the heat source and the heat sink temperature on the EERS performance, we highlight two studies. Disawas and Wongwises [160] investigated a R134a EERS and found that the primary mass flow rate was strongly dependent on the heat sink temperature and not dependent on the heat source tempera-ture, due to the choking phenomena in the nozzle. As result, the CC and *COP* increase with the increase of the heat source temperature and decrease with the increase of the heat sink temperature. Chaiwongsa and Wongwises, used R-134a and reported (i) the primary mass and the secondary mass flow rate slightly increase as the heat source temperature increases and (ii) the CC varies inversely with the heat sink temperature. The authors also tested three nozzle outlet diameters, showing the great influence of the geometrical parameters on the cycle performance.

It is widely accepted that this cycle configuration is interesting and enhances the system performance. Bilir and Ersoy [159, 305] studied the performance improvement of EERS over the standard cycle using the R134a refrigerant: the *COP* was found to increase by 10.1–22.34%, and the reduction in exergy destruction was found to be up 58.7%. The *COP* improvement increases with T_c. and the optimum ϕ increases with the decrease in ejector component efficiencies. Dokandari et al. [205] evaluated the ejector impact on the performance of the cascade cycle that uses CO_2 and NH_3 as refrigerants. The maximum *COP* and the second law efficiency are approximately 7% and 5% higher than those of the conventional cycle. Ersoy and Bilir Sag [187] tested a R124a EERS and, depending on the operating condition, the COP was 6.2–14.5% higher than that of the conventional system. Bilir Sag et al. [182] (experimental study using R134a) reported an increase of *COP* by 7.34–12.87%and an increase of the exergy efficiency of 6.6–11.24% compared to a conventional system. An EERS provide performance enhance-ment due to two effects: the liquid-fed evaporator and work recovery. Unal and Yilmax [306] reported an increase in the COP of the 15%. Pottker and Hrnjak [307] experimentally investigated and quantified these two contributions:. The system was first com-pared to a system with liquid-fed evaporator at matching CC: system performance improved from 1.9% to 8.4% due to the work recovery. When compared to a conventional expansion valve

Table 11
Operating conditions and performance of state-of-the-art of EERS: (T) theoretical study and (E) experimental study.

Ref.	Working fluid	Evaporator temperature [°C]	Condenser temperature [°C]	COP_{mec} [–]	CC [kW]
[130] T	R11 R12 R22 R113 R114 R500 R502 R717	−15	30	5.3–5.7	–
[160] E	R134a	8–16	27–37	4.5–6	3
[161] T	R134a R141b R142b R404A	−15	30	4–4.7	–
[192] T	R290 R600a R717	−15 to −5	35–55	6.1–6.2	–
[162] E	R134a	8–16	27–38.5	2.5–6	3
[158] E	R134a	8–16	27–38.5	3–6	3
[159] T	R134a	−25–5	35–50	3–5.5	–
[205] T	R744-R717	−55 to −45	30–40	2.5–6.5	–
[182] E	R134	40	55	2.62-3.53	–
[187] E	R134	10	55	2.1–2.4	
[230] T	R134-R1234yf	−5–0	20–90	0.5–9.5	–
[232] T	R134-R1234yf	−10–10	30–55	3–7	

The values provided in the table represent an indicative range of the conditions considered in each study analyzed.

system at the same CC, the EERS improved COP from 8.2% to 14.8% due to simultaneous benefits of the two combined effects. The reader may also refer to the study of Wang et al [308] focused on the comparison of different ejector–expansion vapor-compression cycles by using a mathematical model. The authors also proposed a novel configuration with better performance, where ejector was placed between the evaporator and the separator. Other configurations may concern an additional flash tank [309] (COP increased by the 6 and 10%) or a mechanical subcoooler [310] (COP increased by 7 and 9.5%).

Due to regulations concerning the refrigerants, alternatives for R134a should be selected and a possible candidate is R1234fa. Some studies have compared the performance of both refrriger-atns showing that R1234yf is a valuable candidate [230–232]. Boumaraf et al. [230] reported an improvement in COP higher than 17% (T_c=40 °C) for both R134a and R1234yf. R1234yf was found to have higher COP, especially at high T_c. Li et al [232] reported that EERS with R1234yf EERC has better performance than that of the standard cycle, especially at high T_c and low T_e condensing tem-perature and lower evaporation temperature. Lawrence et al. [231] compared EERC with conventional systems and reported a COP improvements of up to 6% with R1234yf and 5% with R134a. However, further studies are needed for better investigating the role of R1234fa under a larger range of operating conditions.

Despite the advantage on the performance, however, some disadvantages should be considered in this configuration, i.e., high refrigerant flow rate, insulation of the piping and installation cost. Table 11 summarizes and compares the above-mentioned studies.

4.7. Multi-components ejector refrigeration system (MERS)

Multi-components ejectors can be used for maintaining the highest possible performance at varying working conditions (i.e., lower T_g). The main multi-components ERS analyzed over the years by researchers are the ERS with an additional jet pump, the Multi-stage ERS and the Multi-evaporator ERS.

4.7.1. ERS with an additional jet pump

The layout of an ERS with an additional ejector is presented in Fig. 14. Yu et al. [154] proposed the addition of a second ejector in series to the main one: the jet-pump (liquid jet ejector) receives the mixing flow of the first ejector as the secondary flow and the liquid condensate as the primary flow. As a result, the ejector backpressure can be reduced, increasing ω (ω=0.6, at maximum value) and COP (COP=0.3). The results of the simulations indicated that COP can increase by 45.9% and 57.1% with R134a, and R152a, respectively, compared with a conventional cycle. Yu and Li [169] suggested another system with a similar configuration using R141b but in the regenerative configuration for preheating the working fluids. The exhaust flow of the ejector is divided: (i) the first part is discharged at the condenser pressure, and (ii) the second part at higher pressure, is redirected to the jet pump. The cycle increases the COP by 9.3–17.8% compared to a conventional cycle. The same research group proposed some other solutions [175]: a mechanical sub-cooling ejector refrigeration cycle with R142b improved the COP up to 10% compared with a conventional cycle. However, despite the increase of performance, difficulties exist in the system control [11]. Cardemil and Colle [311] studied a cascade system composed by two ejector refrigeration systems using H_2O and CO_2, respectively, obtaining a COP=0.2. The condenser and the evaporator in the H_2O system are the boiler and the condenser for the CO2 system. He et al. [236] investigated a two stage ERC and investigated the performance of each ejector. The two-stage system has better performance than the single-stage one fot T_g=150 °C, T_c=54 °C. For lower condensing tem-perature, a single stage cycle is competitive. As a conclusion, for different operating conditions, different operational models should be considered for a two stage system.

Another possible configuration is the two stage ejector proposed by Grazzini et al. [312,313]: the ejector is composed by two sub-ejectors: the first sub-ejectos has no diffuser and its oulet is the second ejector inlet. This system is able to increase the pres-sure lift by the 12.7%, when compared to a SERC (the working fluid was water). The layout of the system is proposed in Fig. 15: the vapor coming from the generator is spitted in two streams and is the primary fluid of the first sub-ejector and the secondary fluid of the second sub-ejector. Recently, Kong et al. [64,314] presented a numerical investigation of the local phenomena in a two-stage ejector system. A dual ejector configuration was also proposed by Zhu et al. [315] using R410A. COP was increased by 4.60–34.03% over conventional system. However, further studies are needed for an improved design of the double ejector systems (i.e., the ejector design as function of the operating conditions, ejector component efficiencies, etc.).

4.7.2. Multi-stage ERS

Multi-stage ejector refrigeration systems are another type of multi-component ERSs, in which some ejectors are placed in parallel before the condenser (Fig. 16). Sokolov and Hershgal [137] proposed the following arrangement: each ejector operates in a different operative range of condenser pressure. Multi-stage ejectors attempt to solve the main problem afflicting the ERS, namely, the difficulty to maintain the system operating in the on-design mode, even after a change in the operating conditions. This challenge is especially true for the solar-driven ejectors, whose performances are highly dependent upon environmental conditions, i.e., the level of solar radiation.

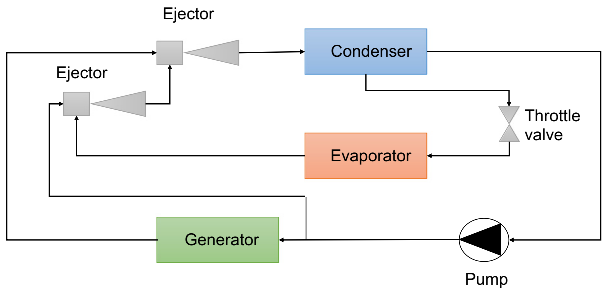

Fig. 14. ERS with an additional ejector by Yu et al. (2006).

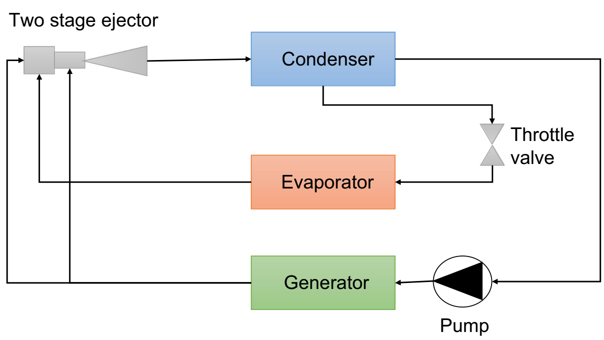

Fig. 15. Two-stage ejector ERS by Grazzini et al (1998).

4.7.3. Multi-evaporator ERS

Elakdhar et al. [144] proposed a two-evaporator system that operates at different pressure levels as a solution for domestic refrigeration. In the proposed configuration, the ejectors combine the streams coming out from the two evaporators into a single mixed stream at intermediate pressure. For this system, light refrigerants (R123, R124, R134a, R141b, R152a, R290, R717 and R600a) were studied, and R141b was found to provide the best performances. The cycle improved the *COP* by 32% compared with a conventional cycle. Note that the system makes use of a compressor: it requires less mechanical work but does not eliminate the compressor; as a result, the electricity consumption is not negligible. Kairouani et al. (2009) [157] suggested a solution similar to the previous one, but with three evaporators and two ejectors (Fig. 17). Also, in this case, the ejectors are placed at the evaporator outlets and, as a consequence, the compressor specific work decreases, thereby improving the *COP*. The authors investi-gated R290, R600a, R134a, R152a, R717 and R141b and, as in the previous work of Elakdhar et al. [144], R141b provides the best performance, increasing the *COP* by 15% compared with a con-ventional cycle.. A similar study (both numerical and experi-mental) was performed by Li et al. [234, 235] using R134a as a refrigerant. The system is highly dependent upon the cooling load: the authors concluded that the primary and the secondary flow rate cannot change more than $\pm 5\%$ and 10%, respectively, from the on-design operating conditions to maintain the evaporating temperature within the range of ± 2 °C. Liu et al. (2010) [198] presented different circulatory systems in the hybrid two-eva-porator cycle: (i) series hybrid, (ii) parallel hybrid and (iii) hybrid cross-regenerative thermal system. For the first two systems, the power consumption reduction compared to a system without ejector is negligible. With the third method, the power con-sumption decreased to 0.655 kWh/day while maintaining the on-design operating condition. Thus, the power consumption decreased by 7.75% compared to the original prototype. Recently, Minetto et al. [316] performed an experimental investigation focused on parallel evaporator feeding. This experimental

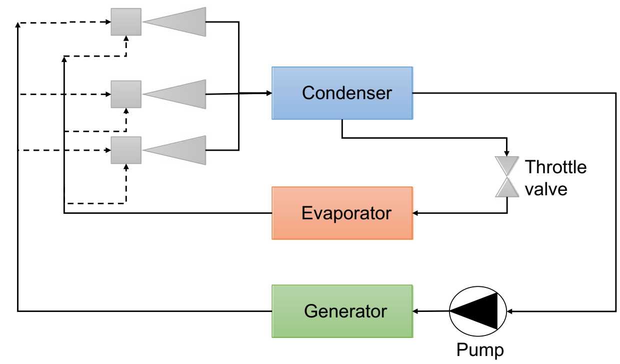

Fig. 16. Multi-stage ERC.

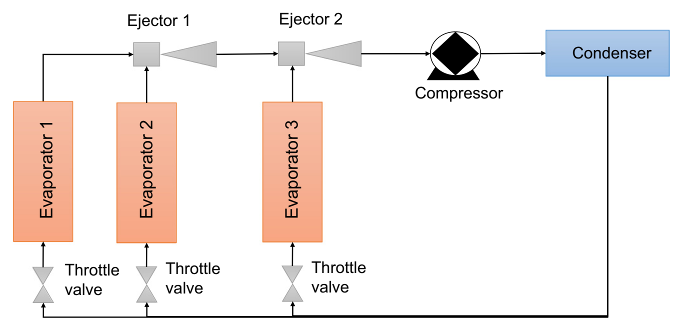

Fig. 17. Multi-evaporator ERC.

investigation may suggest methods for the scale up of these plants on an industrial scale.

4.7.4. Auto-cascade refrigeration system and Joule–Thomson system

Auto-cascade and Joule–Thompson systems can be classified as cryogenic ERS. the autocascade system uses one compressor to achieve the lower refrigerating temperature (i.e., -40 °C and -20 °C). In these systems, an ejector is introduced for recovering the expansion process kinetic energy (reducing the throttling loss). The ejector is, in other words, used for increasing the suction pressure of the compressor. Yu et al. [155] studied this system (Fig. 18) using R23/R134a. The application of the ejector increased the *COP* by 19.1% and decreased the compressor pressure ratio compared to a conventional autocascade cycle. In this paper, an auto-cascade ejector refrigeration cycle (ACERC) was proposed to obtain a lower refrigeration temperature based on the conven-tional ejector refrigeration and auto-cascade refrigeration princi-ple. Tan et al. [317] studied an autocascade refrigeration systems using R32/236fa (zeotropic refrigerant mixture). Using this work-ing fluid, the numerical results showed that this cycle can reach the lowest refrigeration temperature of -30 °C. A Joule–Thomp-son ERC has been proposed by Yu et al. [318] (Fig. 19), improving by 41.5% the performance of the systems, compared to a system without ejector. Cryogenic ejector refrigeration cycle (in the Joule–Thompson implementation), have also been included in multi-effect cycle [300] (Section 4.4.2).

4.7.5. Summary

All the different MERS solutions ensure a performance improvement, compared to conventional ejector refrigeration systems. However, the impact of the complexity of the equipment and its management must be considered. In the future, detailed models of the complete systems should be developed taking into account both on-design and off-design operating conditions and the economical evaluation of the cycle. A large amount of research

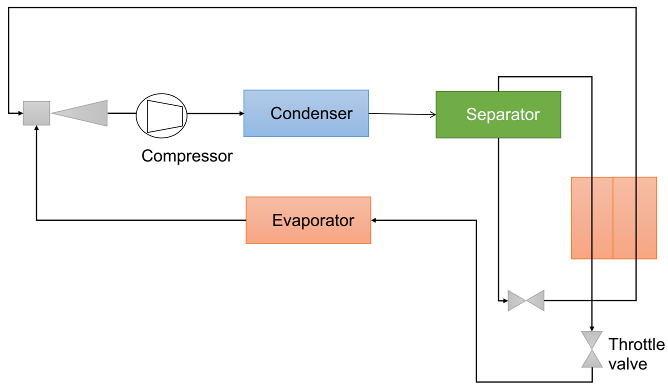

Fig. 18. Cryogenic ERS: autocascade system.

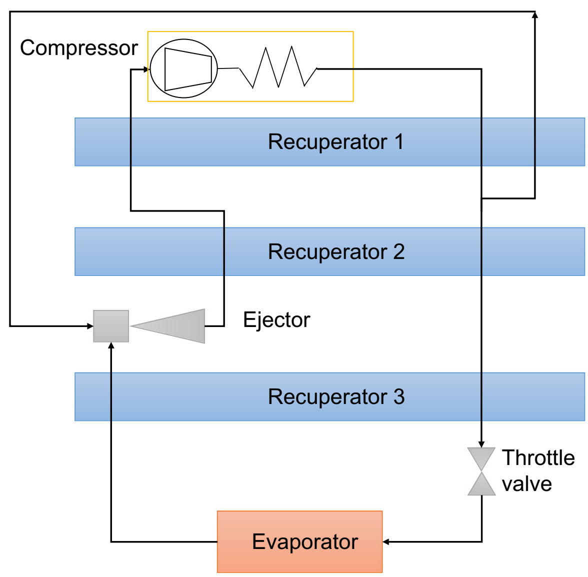

Fig. 19. Cryogenic ERS: Joule–Thompson system.

(theoretical and experimental) should be considered to better evaluate the performance of these systems.

ERS with additionals jet pumps shows an improvement of the performance (if compared to a SERC) till the 57%. However, a critical issue, in these systems, is the off design performance of the ejectors. Future studies should apply off-design models and study the performance of these systems. In particular it should be investigated how the change in operating condition of one ejector influence the others. Double-stage ejector systems have also been proposed, but a better investigation of the ejector design, ejector modeling and ejector component efficiencies as function of the operating conditions and working fluids is needed. Moreover, all these studies are theoretical investigations and no experimental data are available. Multi-stage ERS has been found to have an appreciable *COP* (between 1.2 and 2.2), however there is a very limited amount of research and further studies should be performed for this system. More studies have focused on Multi-evaporator ERS and autocascade systems; in particular autocascade refrigeration seems a promising technologies for reaching low cooling temperature (-40 °C). Despite the cryogenic refrigeration systems are interesting and further numerical and experimental investigations are necessary to verify the COP_{mec} improvements. In particular, the models for cryogenic refrigeration systems should be improved and particular care should be taken to equations of state and ejector component efficiencies. Table 12 summarizes and compares the above-mentioned studies.

4.8. *Transcritical ejector refrigeration system (TERS)*

Differently from other ejector refrigeration systems, that operate in the subcritical region, the transcritical ejector refrigeration system (TERS) involves a refrigerant operating over the critical conditions. In TERS systems, the generation process occurs at supercritical pressure, and the density of the primary working fluid decreases until the vapor state is achieved. The supercritical vapor expands through the ejector nozzle and entrains the flow from the evaporator. To maintain the required performance, the operation of the transcritical process requires control of the high-side pressure. In these cycles, both the pump discharge pressure and the generator outlet temperature are operation parameters. Furthermore, the ejector could involve two-phase flows, depending on the operating conditions (primary flow pressure and T_g). A more detailed analysis of these system can be found in Yu et al. [319]. Yu et al. [319] compared the above-described cycle with a subcritical cycle using R143a. The first cycle showed considerable advantages; in fact, it presented a maximum value of $COP=0.75$, while the subcritical cycle exhibited a $COP=0.45$. The authors indicated the problem of controlling the high pressure. Finally, the higher working pressure resulted in a more compact system.

Different from the previous study, the most common TERSs are operated with the carbon dioxide (R744). We may divide the studies as follows: (i) one ejector CO_2 TERS, (ii) two ejector CO_2 TERS and (iii) CO_2 TERS with an internal heat exchanger.

4.8.1. *One ejector CO_2 TERS*

One of the first CO_2 TERS studies was published by Liu et al. [320]. Their thermodynamic analysis was based on the work of Kornhauser [130]. Compared to a traditional vapor-compression cycle, in this configuration, an ejector replaces the throttling valve (for the same reasons detailed elsewhere in the paper). Through the ejector, the compressor suction pressure increases compared to a standard cycle, resulting in higher efficiency of the systems (less compression work). However, this layout creates some difficulties regarding control of the operating conditions due to the close link among the quality of the ejector outlet stream and ω [12]. Therefore, Li and Groll [210] proposed feeding some of the vapor in the separator back to the evaporator through a throttle valve (Fig. 20), increasing *COP* by approximately 18% compared with the basic transcritical cycle. Deng et al. [211] presented a thermodynamic analysis of a CO_2 TERS cycle. The improvement of the *COP* achieved is $+22\%$ compared to a standard cycle. The sum of the throttling and ejector exergy losses of the TERS is lower than the one of a standard vapor compression cycle, and the exergy loss in the compressor is lowered. The results also indicated that ω influenced significantly the refrigeration effect. An experimental investigation on a similar system was performed by Elbel and Hrnjak [321]. The *COP* and *CC* were found to increase by up to 7% and 8% compared to a conventional expansion valve system. Fangtian and Yitai [216] compared a CO_2 TERS with an ejector and with a throttling valve: the ejector cycle increased the *COP* by more 30% and reduced the exergy loss by more than 25%. The results showed that *COP* (1–3) is greatly affected by the operating condi-tions. Ahammed et al. [215], experimentally studied CO_2 TERS sys-tems, demonstrating that, at lower heat sink temperatures, the performance is slightly better towards low gas cooler pressure; however, the CC significantly decreases. They also showed that at higher ambient temperature, a high gas cooler pressure leads to an improvement in the performance. In addition, a comprehensive exergy analysis was implemented, and the resulting second law efficiencies obtained were 6.6% and 7.52% for conventional and ejector based systems, respectively. Bai et al. [222] studied a CO_2 TERS cycle with a sub-cooler (ESCVI). The proposed cycle was found to have better performance than the conventional vapor injection cycle, with an increase of *COP* up to 7.7%. The gas cooler and ejector

Table 12
Operating conditions and performance of state-of-the-art of MERS: (T) theoretical study and (E) experimental study.

Ref.	Configuration	Working fluid	Generator temperature [°C]	Evaporator temperature [°C]	Condenser temperature [°C]	*COP* [–]	*CC* [kW]
[154] **T**	ERS with an additional jet pump	R134a R152a	80–100 80–98	5	35	0.20–0.30	1
[169] **T**	ERS with an additional jet pump	R141b	80–160	10	35–45	0.20–0.40	1
[175] **T**	ERS with an additional jet pump	R142b	80–120	5	35	0.30	1
[311] T	ERS with an additional jet pump	H_2O CO_2	80–95	−7 to 3	25	0.20	–
[215] E	ERS with an additional jet pump	R718	130–150	6–30	45–54	0.05–1	–
[144] **T**	Multi-evaporator ERS	R123 R124 R141b R134a R152a R290 R600a R717	–	−5–10 −40 to −20	28–44	1.20–2.20	0.5+0.5
[157] **T**	Multi-evaporator ERS	R290, R600a, R717, R134a, R152a, and R141b	–	−28 −18 5	45	0.5–4	–
[155] **T**	Cryogenic ERS	Mix R23/R134a	0–25	−35 to −20	40	0.6–0.9	–
[317] **T**	Cryogenic ERS	Mix R32/236fa	73–93	−25–14	18-28	0.04	–

The values provided in the table represent an indicative range of the conditions considered in each study analyzed.

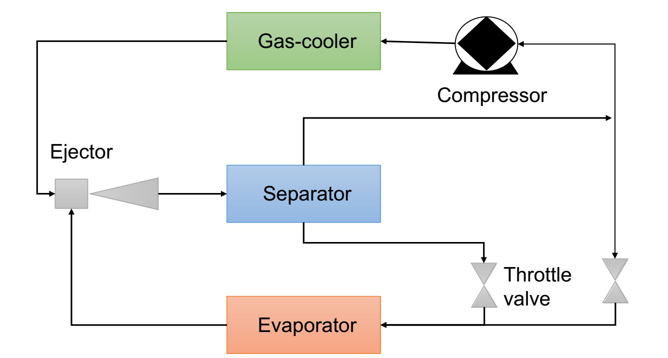

Fig. 20. Transcritical ejector expansion refrigeration system proposed by Li and Groll (2005).

both exhibit low exergy efficiency (57.9% and 69.7%, respectively). The results also revealed the great influence of the ejector compo-nent efficiencies on the performance.

4.8.2. *Two ejector CO_2 TERS*

Using a parametric analysis, Yari and Mahmoudi studied and optimized CO_2 cascade refrigeration cycles with a TERS top cycle and a bottom cycle (sub-critical CO_2 cycle). Energy and exergy analysis suggest that the proposed cycles exhibit a $COP=2.5–2.9$ with a discharge temperature lower than that of the conventional cycles. Cen et al. [219] introduced a two ejectors cycle to recover more expansion loss (Fig. 21). The value of *COP* ranged between 2.75 and 7. The authors indicated that such high values can be difficult to achieve in practice, as the high values are due to the calculation assumptions. In particular, the ejector component efficiencies were assumed to be constants, and the results highly depend on these values. Indeed, Liu et al. (2012) [221] experi-mentally investigated ejector component efficiencies in a CO_2 TERS. The ejector efficiencies were found to depend upon the geometry and operating conditions. Xing et al. [223] studied a transcritical CO_2 heat pump cycle with two ejectors. The ejector are placed at low and high pressure lines of the cycle. The pro-posed cycle increases the COP of 10.4% is compared with a con-ventional cycle. The authors have also studied the influence of an Internal Heat exchanger (please refer to the next paragraph), showing a further increase of the performance of 10.5–30.6%. Also the influence of ejector component efficiencies were studied showing a large influence over the results. Bai et al. [225] studied a

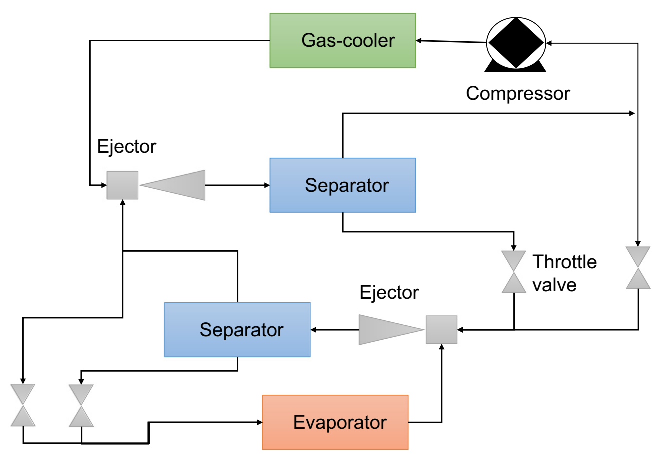

Fig. 21. Two-ejector transcritical ejector expansion refrigeration.

double evaporator system with two ejectors. The first and second law efficiency improved by the 37.61% and 31.9% if compared to a single ejector system ($T_{gascooler,exit}$ =35–50 °C, $T_{e,high}$ = −5–5 °C, $T_{e,low}$= −35 to −15 °C).

4.8.3. *CO_2 TERS with internal heat exchanger*

Some studies focused on the influence of an internal heat exchanger (IHE). Yari and Sirousazar [212] studied a CO_2 TERS with an IHE and an intercooler. Compared to conventional ejector–expansion TERS, the *COP* increased by 55.5%, and the second law efficiency was 26%. Furthermore, Yari [213] also proposed corre-lations to predict the design parameters for the following ranges: $T_{gas\ cooler\ outler}$ from 35 to 55 °C and T_e from −30 to 0 °C. Nakagawa et al. [214] experimentally investigated the role of the mixing length for different systems (conventional expansion systems or with ejector and with and without IHE). The mixing length is a critical parameter for ω and the pressure recovery; for all the operating conditions tested, the authors concluded that the mix-ing length of 15 mm yielded the highest ejector efficiency and the *COP*. A longer mixing length leads to a minor variation in the pressure recovery but a significant decreases ω. Moreover, the use of internal heat exchanger enhanced the system performance, increasing the *COP* by up to 26%. However, the improper mixing length lowered the *COP* by 10%. Manjili and Yavari (2012) [220] studied a multi-intercooling CO_2 TERS, comparing it to a standard

Table 13
Operating conditions and performance of state-of-the-art of TERS: (T) theoretical study and (E) experimental study.

Ref.	Working fluid	Primary flow conditions [°C]/[MPa]	Secondary flow temperature [°C]	Outlet mixing flow temperature [°C]	*COP* [–]	*CC* [kW]
[319] T	R143a	60–100 6–10	10	30–40	0.3–0.75	1
[210] T	R744	36–48 8–12	5	15	+7–18%	–
[211] T	R744	36–40 8–12	0–10	4–20	1.5–3.5	–
[212] T	R744	40–50 8–12	−20–10	13	1–4	–
[213] T	R744	35–55 7.5–12	−30–5	–	1–3.5	–
[216] T	R744	40–45 8–9	−5–17	–	2.5–2.9	–
[214] E	R744	41–44 9–10.5	2–8	–	1–2	1–2.5
[219] T	R744	40–43 9–11.5	5	40	2.75–7	–
[217] T	R744	36–40 8–12	5	40	1.5–3.5	–
[220] T	R744	36–54 8–12.5	−15–5	–	2.2–2.8	–
[215] T	R744	30–45 8–12	0–10	35	2–3.6	3.5
[218] T	R744	36–40 8–12	5	40	1.5–3.25	–
[222] T	R744	35–50 8.5–12	−25–5	25	2.5–4	–
[223] T	R744	36–40 8–11.5	−30–0	–	3.12–4.25	

The values provided in the table represent an indicative range of the conditions considered in each study analyzed.

ejector refrigeration and to an heat exchanger ejector refrigeration cycle (IIE). The proposed configuration has the maximum *COP* (2.2–2.8) and the IEC has the minimum *COP* (1.4–2.2). The max-imum *COP* of the multi-intercooling cycle is 15.3% and 19.6%higher than those of a conventional cycle and the IEC, respectively. Finally, the exergy destruction of the compressors and in the gas cooler decrease by 60.89 and 51.61%, respectely, comparing to a conventional ejector refrigeration cycle. The influence of the IIE on CO_2 TERS was also studied by Zhang et al. [217] using a thermodynamic model. The addition of IHE increases ω (+20–30%) and decreases pressure recovery (approximately −30%) for the same gas cooler pressures. However, the *COP* is not always improved: this depends on the isentropic efficiency of the ejector. The *COP* is increased for lower ejector isentropic efficiencies or higher $T_{gas\ cooler\ outler}$. Zhang et al. [218] also investigated the influence of the suction nozzle pressure drop. This parameter has little impact on ω, but an optimum value for the pressure recovery and *COP* exists: optimizing the geometrical parameter, the *COP* increases by 45.1% and the exergy loss reduces by 43.0% compared to the basic cycle. The optimum value is influenced by the ejector component efficiencies, but it is independent of the gas cooler outlet temperature and the evaporating temperature. Also Xing et al. [223] studied the influence of an Internal Heat exchanger reporting an increase of the performance of 10.5–30.6%. Other configurations have been proposed by Goodarzi et al. [226, 227] (i.e., extracting a saturated vapor from separator and feeding to the intercooler or using a multi intercool system): both these studies reported an increase of the system performance. In particular, the system with vapor extraction increase the *COP* by the 26.87% in compared with a conventional cycle. Beside TERS systems, the interested reads may refer to Butrymowicz, et al. [322] for a discussion on internal heat exchanger in ejector systems.

4.8.4. *Summary*

Significant *COP* improvement (+7 ± 18%) has been observed if compared with conventional cycles and the CO_2 is a natural, nontoxic and non-flammable refrigerant. However, despite the interesting technology and the increasing number of studies, some experiments are still needed and the technical and economic feasibility of this choice on a large scale plant must be evaluated. Furthermore, the role of the ejector in the modeling of these cycles is still not clear and deserves more attention. In particular, an increasing number of studies is focusing the attention on ejector efficiencies in CO_2 TERS. These efficiencies works critically in the evaluation of the system performances. For example, Cen et al. reported a *COP*=7 because of the efficiency value. Further research (experimental and numerical) should be performed concerning the ejector component efficiencies for both on-design and off-design operating condition as function of the geometry. Table 13 summarizes and compare the above-mentioned studies.

5. Ejector refrigeration systems: comparison

In the previous paragraphs, we have examined different ejector refrigeration technologies; in this section, we have collected all the data (from the previous sections), organized by technology, to provide summary charts able to compare the different performances of the technologies in terms of historical evolution, T_g and working fluids. The goal of this section is, therefore, to present a comprehensive view of the studies of the ejector technology and research and to provide a useful tool for the selection of the appropriate technology and working fluids. The charts presented in this section shown the main results and the maximum performances reported in the original references.

5.1. *Historical evolution*

Fig. 22 shows the historical evolution of the *COP* for the different ejector technologies (expect for the combined refrigeration and power production systems). The development of new technological solutions resulted in an increase of the system performance.

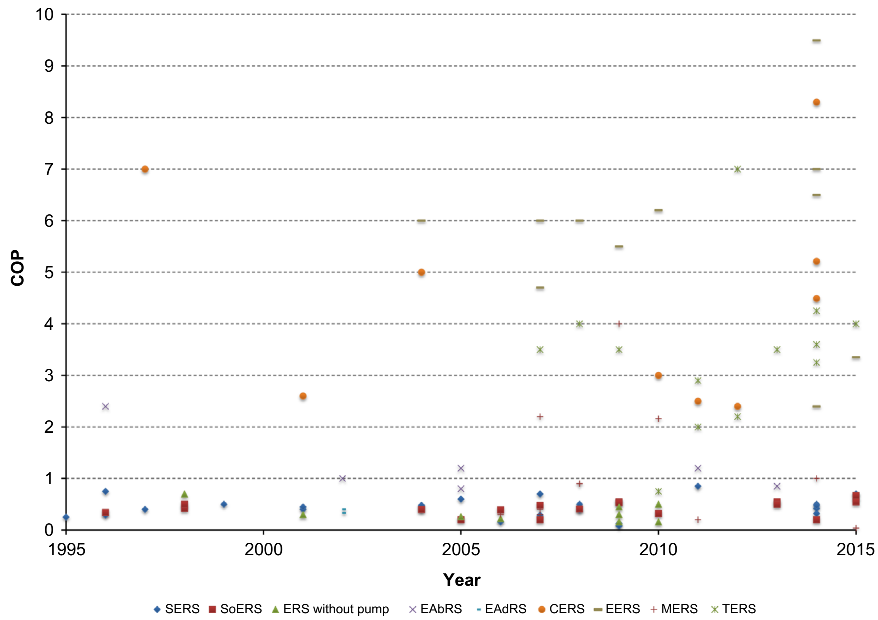

Fig. 22. Performance trend of ERS technologies over the years.

The SERS exhibited a growth in the performance in the last 20 years, passing from $COP=0.12$ in 1995 to the value of $COP=0.75$ achieved in more recent years. A similar trend is shown for the SoERS: starting from a coefficient of performance equal to 0.34 obtained in 1996, managed to stabilize to a value of approximately $COP=0.6$. COP increase also for the ERS without pump, but it is still lower with respect to the other systems; however, the research for these systems is still limited. The increasing trend of the *COP* may not be always so clear because other variables are also involved in the ERS operation. Particularly interesting, the growth of the *COP* obtained with the combined systems (i.e., EERS and CERS) is not lower than the one obtainable with the other refrigeration systems, such as absorption or vapor compression systems. The coupling of the absorption cycles and the ejector component combines the advantages of two systems, and the resulting systems exhibit high values of *COP* (0.4–2.4) if compared to SERC systems. The coupling of adsorption cycles and the ejector component is promising, but the research is very limited. The MERSs, presented in the last decade, ensure a performance improvement, compared to conventional ejector refrigeration systems. The first EERS was proposed in 1990, and its coefficient of performance was equal to 5. Since then, the *COP* has continued to grow, and fourteen years later, it has reached the value of 6.5–7.5.

This evolution was made possible due to the great efforts of researchers to develop and improve the ejector refrigeration systems. In light of this evolution, it is reasonable to expect, for the future, a further improvement of the ERS performances, as well as the development of new plant configurations.

5.2. Generator temperature

Fig. 23 illustrates the relationship between *COP* and T_g. An increase in the value of T_g determines an increase in the performance. However, the operating conditions are determined by the availability of the energy source and, for each application, there is a more suitable technology. Among the different technologies, the EERS and the TERS, have a high coefficient of performance and are also able to work with low T_g (<60 °C). The SERS, SoERS and CERS operate with intermediate temperatures, in the range of 60 °C to 140 °C. Particularly interesting are the CERS, able to have higher Cop if compared to the other technologies in the intermediate temperature range. The ERS without a pump operate in a narrow range of generator temperature between 80 and 110 °C. The EAbRS requires, instead, a high value of T_g greater than 120 °C. In addition, the graph shows that, such as expected, the coefficient of performance increases with the value of T_g for each technology. Depending on the heat source available, this chart may provide a useful tool for the selection of the appropriate technology.

5.3. Working fluids

The effect of the working fluid is shown in Figs. 24 and 25. The figures represent the historical trend of the working fluid used in the ejector refrigeration systems and the former relates each technology with its working fluid. The information in these figures should be coupled with the discussion in Section 3.3 concerning the screening of the working fluids for ejector refrigeration system. Hydrocarbon and halocarbon compounds with low ODP and GWP were widely considered as valuable working fluids. Generally speaking, the halocarbon compound providing the best performance is R134a (HFC compound), which is able to provide high performances with all types of ERS technologies, in particular, with the EERS (the value of *COP* is approximately 6). The hydrocarbon compounds are sufficiently versatile, but appear to provide the best results when used in simple systems. As the most econom-ically and environmental friendly refrigerant, water has been tes-ted as a refrigerant for ERS, and carbon dioxide has recently attracted increasing interest. In particular, by using transcritical cycles, the carbon dioxide can provide good performance ($COP=3$–6). Even if ammonia and the methanol have good properties as refrigerants, they do not adapt well with the best-performing systems (in particular, EERS and TERS). In the future it is expected a further evolution of the working fluids used in ejector refrig-eration system due to the recent regulations. For example, The EU Regulation 517/2014 will phase out and limit the use of refrigerants with high GWP values such as R134a, R404a and R410a. Therefore, it is expected that environmentally friendly halocarbons, hydrocarbons, natural refrigerants (R717, R744) and HFC/HFO mixtures will be increasingly adopted [228]. Further research should be considered for potential substitutes: for example R1234yf [229] can be a valuable for R134a and has already been investigated for ejector expansion refrigeration system [20,

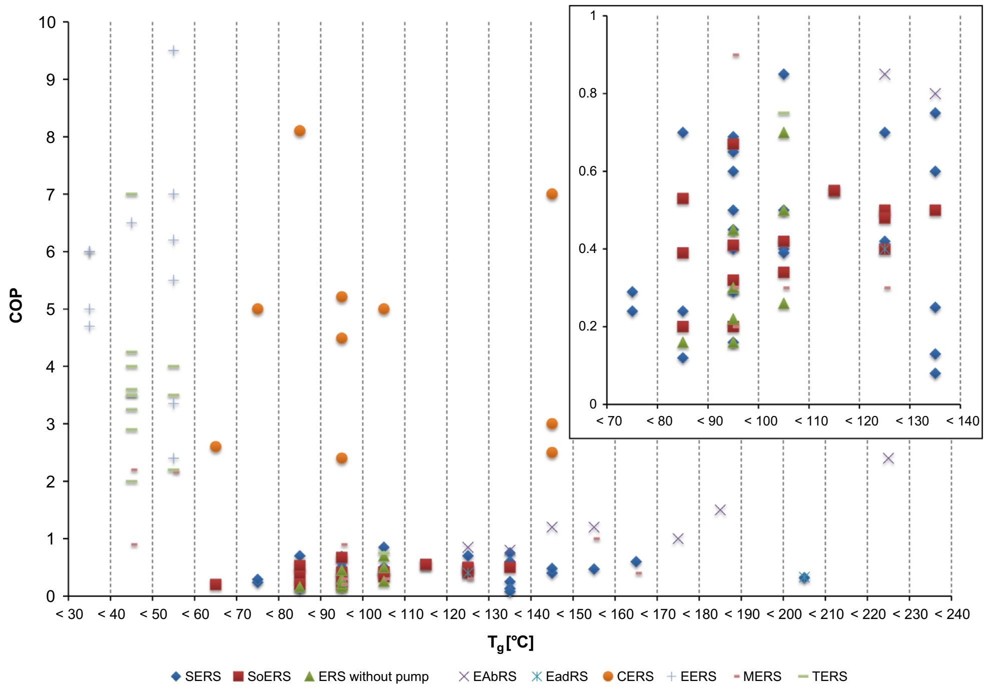

Fig. 23. Performance trend of ERS technologies as a function of the generator temperature.

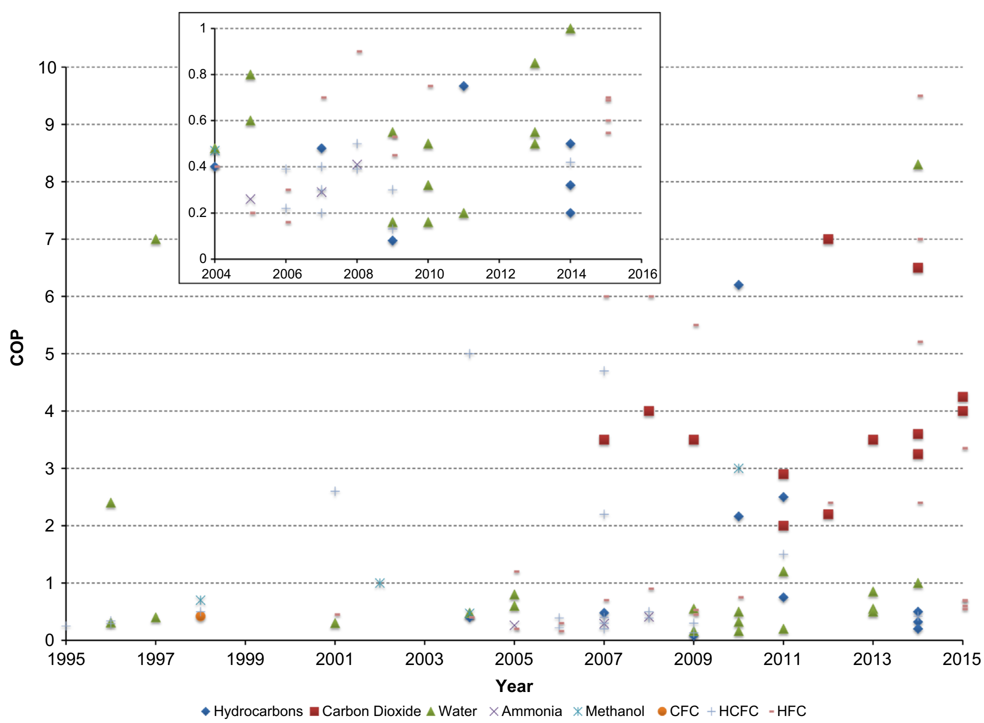

Fig. 24. Performance trend of ERS over the years for the different working fluids.

230–232] and other refrigeration systems [323–326]. Future stu-dies should also consider refrigerant blends [233].

6. Conclusions

ERS is a promising technology for producing a cooling effect by using low-grade energy sources with different working fluids. In this paper, ejector technology, refrigerant properties and their influence over the ejector performance, the main jet refrigeration cycles, and all of the types of ejector technologies (Fig. 1) were analyzed in depth, with a focus on past, present and future trends. Ejector allows the use of many refrigerants and many studies have tested the influence of the fluid on the refrigeration cycle. A recent driver on the study and selection of the working fluid is the EU Regulation 517/2014 that is going to phase out and limit the use of refrigerants with high GWP value, like the most used R134a, R404a and R410a. Therefore, environmental friendly halocarbons, hydrocarbons, natural refrigerants (R717, R744) and HFC/HFO mixtures will be increasingly employed for their low ODP and GWP values. As the most economically and environmental friendly

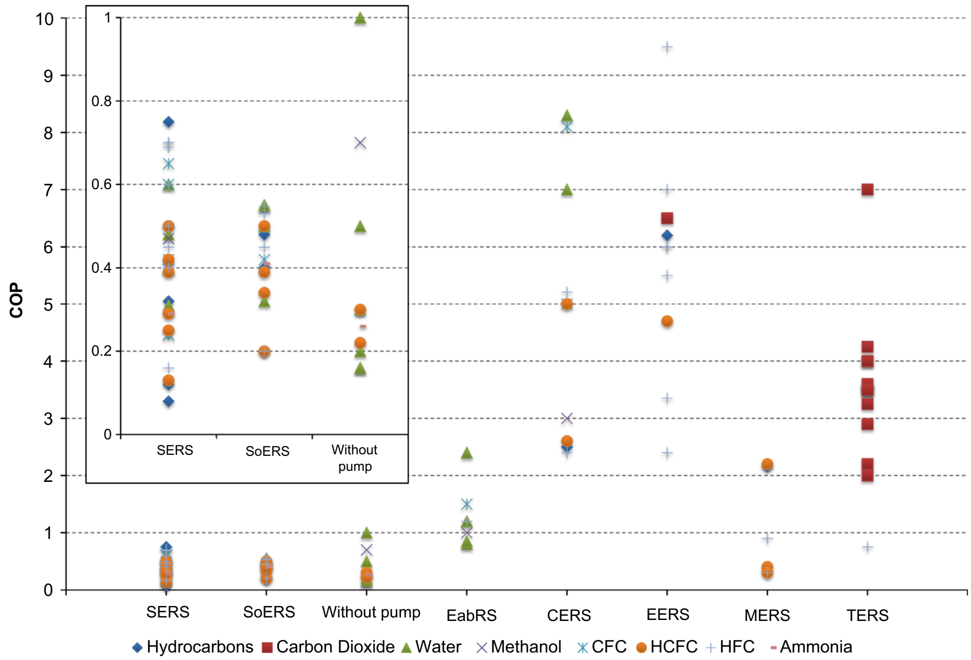

Fig. 25. Performance trend of ERS technologies organized by working fluid.

refrigerant, water, has been tested for ERS application and, recently, carbon dioxide has attracted a growing interest too. Further studies should also consider other working fluids, such mixture and blend of refrigerants. Furthermore, most of the studies concerning the screening of working fluids have considered subcritical cycle only: future studies should take into account critical and subcritical cycles too. A complete review of the working fluids is reported in Section 3 and related subsections.

Different configurations for ejector refrigeration have been investigated. SERSs are simple refrigeration systems with a low coefficient of performance and many studies have focused on the enhancement of system performance: possible solutions are the use of different refrigerants, storage systems and the reduction of the mechanical work. Some evolutions of this technology have been presented based on alternative energy source, pumping system, ejector purpose to improve the system performance or reduce costs. Solar energy can drive the system (i.e, for air-conditioning system), however, the system performance highly depends on ambient conditions, the use of energy storage is proposed for solving the problem. However, dynamic simulations are required for the design and study of these refrigeration systems. ERS without pump have been proposed, but further research, modeling studies and experi-mental investigations are needed for clarifying theirs performance and off-design behavior. The use of combined systems (ejector–absorption, ejector–adsorption or ejector-compression) allows extending the jet compressor application range and hybrid cycles allow the use of different working fluids for each subsystem. The transcritical ERS cycles have attracted a growing attention because they could provide higher potential in utilizing low-grade heat. Using ejector as an expansion device (EERS) improves *COP* in vapor compression refrigeration cycles, but, for better exploiting this advantage, more studies on the two-phase ejector local phenomena are required. Particularly interesting are the combined power and ejector refrigeration systems able to provide electricity and refrig-eration effect simultaneously.

In the Section 5 of the paper, we have collected the data, organized by technology, to provide summary charts able to compare the different performances of the technologies in terms of historical evolution, T_g and working fluids. A comprehensive view of the ejector technology and research is provided. The chart presented may help in the selection of the appropriate technology and working fluids, as reported in Figs. 22–25.

When considering the above-mentioned and other ejector technologies reported in this review, the performance are compared in terms of efficiencies. While the first law efficiencies are straightforward, for the second law efficiencies there are some issues. Indeed, exergy analyses have been widely applied without using a common basis making difficult to compare the exergy efficiencies. A common basis when considering the second law analysis should be applied (i.e. the same reference temperature, for example 298 K). Beside the efficiency evaluation, economical evaluations should be performed. In future research this should be considered and, when performing economic analysis, different scenarios should be always investigated and compared for every system.

Finally, for all the ejector technologies some main considerations should be taken in account: (a) further studies concerning on-design and off-design operating conditions are needed using both experimental and numerical studies; (b) non-steady-state models should be developed for considering the dynamic behavior of the system (i.e., the start-up phase) and, for the solar based system, dynamic simulations should be considered for taking into account the discontinuous nature of the solar energy; (c) when applying lumped parameter models for studying ejector perfor-mance, the ejector component efficiency used for investigating the ejector performance should be verified by means of numerical or experimental studies. If this would not be possible, a sensitivity analysis should be always performed; (d) the use of studies with constant ejector component efficiencies is questionable and variable formulation should be proposed; (e) lot of studies has been proposed for single phase ejector, but these data and models can not be used for studying two phase ejectors because a large number of differences exist. Furthermore, most of the studies concerning two-phase ejectors are numerical and mainly based on one-dimensional homogeneous equilibrium model with few experimental data available. A more advanced analysis of these cycles could be performed by using variable ejector efficiencies and multi-dimensional non-homogeneous flow.

In conclusion, ejector refrigeration systems are a promising technology that can be applied for different applications and operating conditions. Their market spread can be supported by providing accurate off-design ejector modeling techniques, reliable two phase ejector models and large scale experimental investigations in a large set of operating conditions.

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