Thermoeconomic analysis of a solar thermal ejector cooling unit

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Abstract:

The fact that the peak of cooling loads in dwelling application happens typically in periods of high solar radiation is favoring systems designed to produce cooling from the solar resource. The ejector cooling cycle is particularly attractive coupled to a solar thermal captation device, as it is compact and it has a simple conceptual layout. On the other hand, some critical points arise in the extreme fluid dynamics performance conditions – notably, a high Mach Number in the ejector. The search for an optimized performance must include the selection of the working fluid, and to a large extent the system performance depends on both the performance of the thermal (solar collector) and on the cooling (ejector) systems, which are coupled through the entrainment ratio.

A complete thermo-fluid dynamics analysis using real fluid conditions is presented, considering both fluid dynamics and thermodynamics constraints and/or preferences of the designer (for example, limitation of ejector Mach number); the system/cycle simulation allows to calculate the energy performance (COP). The study is completed by the calculation of exergy flows in each point of the system, and by the calculation of the thermoeconomic performance.

The results confirm the possibility of achieving a COP in the range 0,28-0,4 from a solar thermal supply at 85-95°C, a temperature range achievable with non-concentrating solar collectors. The largest part of irreversibilities, as well as of the system cost, can be attributed to the solar field.

Keywords:

Ejector Cooling Cycle, Solar Energy, Exergy, Thermoeconomics.

1. Introduction

The use of solar energy for supplying cooling power for dwelling units is very attractive, because the availability of the renewable solar resource is abundant in the summer period, when the cooling load is also relevant. On the other hand, the solar collector array can be used for heating in winter and production of hot water during the whole year, thereby reducing the equipment cost, which is shared between heating in winter and cooling in summer.

The ejector cooling cycle is particularly attractive as it is compact and it has a simple conceptual layout. Several possible circuit schemes are possible, some including regeneration and hybrid compressor/ejector solutions [1]. Having in mind the development of a small unit suitable for the replacement of unit coolers (5 kW design cooling load at the evaporator), it was decided to adopt the simple scheme represented in Figure 1. The working cycle is represented in Figure 2, referred to the case of a common refrigerant (R134a). It is important to distinguish the primary circuit (7-1-9-6-11, represented in blue) and the secondary (entrained-flow) loop (4-5-8-11, represented in red). The two flows m_P and m_{EV} are joined at point 11 (ejector mixing section), after which the mixed flow rate enters a diffuser (11-3) where the fluid is compressed to the condenser pressure level. The section of the circuit 11-3-4 (in green in Figure 2) is operated with the flow rate $m_P + m_{EV}$. The condenser pressure level is fixed by the ambient temperature (heat rejection to the environment); the cycle operates between the upper pressure (heat exchanger/storage) and the lower pressure which is determined by the allowable preset cooling temperature.

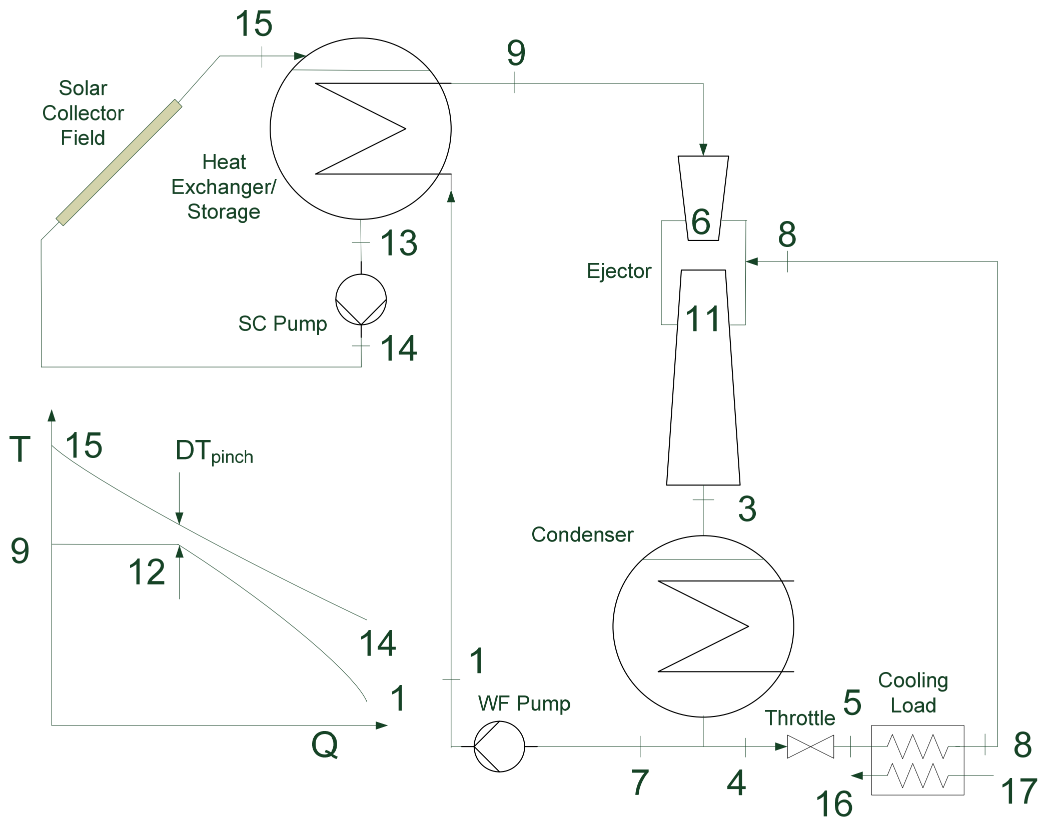

Fig. 1. Schematic of solar-driven ejector cooling system; Left, pinch diagram of the heat exchanger

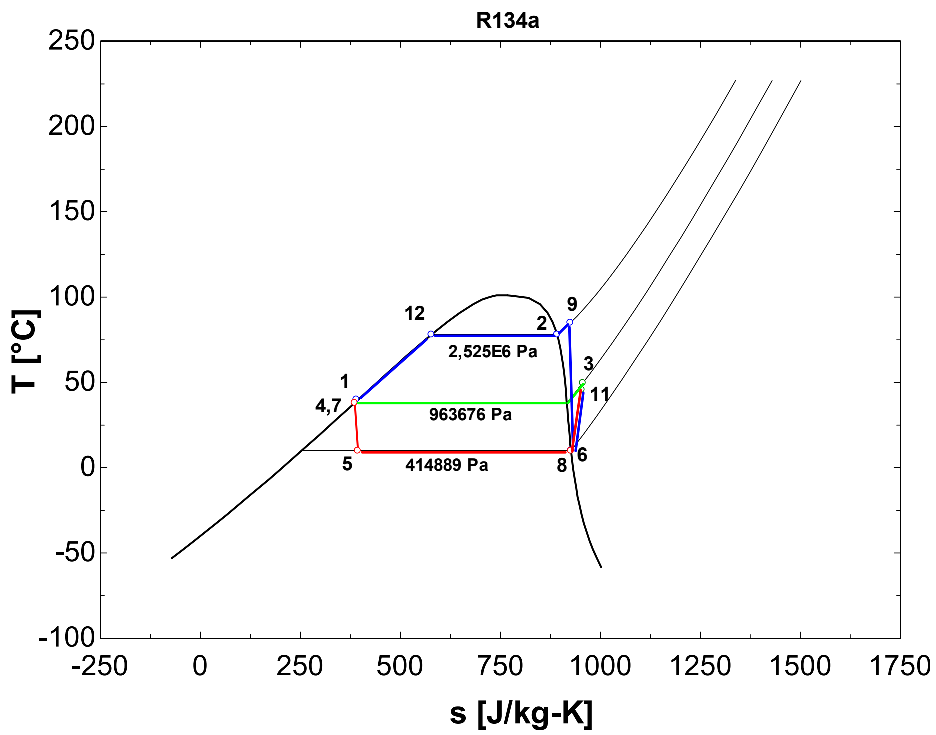

Fig. 2. Ejector cooling cycle (typical)

2. Ejector model

The one-dimensional model of the ejector is adapted from existing literature [2, 3]. The following equations assume $p_6 = p_8$ at the mixing plane, and apply the conservation of mass, momentum and energy:

$$\dot{m}_P + \dot{m}_{EV} = \dot{m}_C \quad (1)$$

$$(p_{11} - p_6)A_{11} + \rho_{11} A_{11} v_{11}^2 = \rho_8 A_8 v_8^2 + \rho_6 A_6 v_6^2 \quad (2)$$

$$\dot{m}_C\ h_{o,11} = \dot{m}_P\ h_{o,6} + \dot{m}_{EV}\ h_{o,8} \quad (3)$$

This set of equations, together with the geometric conditions $A_{11} = A_8 + A_6$, allows to determine the complete conditions at the fully mixed section (11) once the ratio MR between the entrained and primary flow rates is specified. Accepted guidelines for this last suggest that in order to achieve a reasonable ejector efficiency values of MR in the range between 0,3 and 0,5 should be adopted. It is important to underline that, with respect to the previous models [2, 3], no use of compressible flow relations used for perfect gases is done in the present case; it is sufficient to use a real fluid property evaluator [4].

3. System thermodynamic model

The present study is first aimed at a general thermodynamic design of the system, starting from the evaporator and ending with the solar collector field. A feature of the model is that real fluid properties are used throughout, and that thermodynamic and fluid dynamics variables are calculated together so that a feasible design can result.

The main input data are:

Temperature at vapor generator, T_{VG}

Superheating at vapor generator, DT_{SH}

Pressure at condenser, T_C

Pressure at evaporator, T_{EV}

Ejector entrainment ratio, $MR = m_{EV}/m_P$

The evaporator heat rate determines the necessary working fluid flow rate:

$$Q_{EV} = \dot{m}_{EV}(h_8 - h_5) \quad (4)$$

Where $h_5 = h_4$ (throttle) and h_8 is calculated at saturated vapor conditions at p_{EV}.

With the primary flow rate fixed by the assumed MR, and using Eq. 1, it is possible to calculate the vapor generator heat rate, which must be satisfied by the solar collector field:

$$Q_{HE} = \dot{m}_P(h_9 - h_1) = \dot{m}_{SC}(h_{15} - h_{13}) \quad (5)$$

The solar collector is operated with a suitable heat transfer fluid (slightly pressurized water or appropriate water/glycol solution), fixing the delivery temperature T_{15} and the pinch point temperature difference DT_{pp} at evaporator inlet. With these conditions, it is possible to determine the flow rate $\dot{m}_{SC}$, T_{13} and consequently h_{13}. The traditional Hottel-Woertz-Bliss equation [5] was used for the evaluation of the solar collector performance:

$$\eta_{SC} = \frac{\dot{Q}_{HE}}{\dot{Q}_S} = \eta_o - \frac{a_1(T_{av} - T_a)}{I} - \frac{a_2(T_{av} - T_a)^2}{I} \tag{6}$$

With $T_{av} = (T_{15} + T_{13})/2$ and I equal to the global effective radiation (including collector incidence angle modifier). The heat rate output for the solar collector is integrally transferred with negligible losses to the heat exchanger/storage as Q_{HE}. The solar input is $Q_S = A\, I$, which allows to calculate the collector field surface under the design conditions using the same Eq. 6.

The condenser heat rate is given by:

$$Q_C = \dot{m}_C(h_3 - h_4) \tag{7}$$

The primary circuit pump power is:

$$W_P = \dot{m}_P(h_1 - h_7) \tag{8}$$

For the ejector cooling cycle, a COP can be calculated as:

$$COP = \frac{Q_{EV}}{Q_{HE} + W_P} \tag{9}$$

The diffuser (process 3-11) is treated assuming the conservation of energy:

$$h_{o,3} = h_{o,11} \tag{10}$$

A loss coefficient ξ_D is used to evaluate the total pressure loss (momentum equation):

$$p_{o,3} = p_{o,11} - 0{,}5\, \xi_D \rho_{11} v_{11}^2 \tag{11}$$

The value of ξ_D can be calculated using suitable correlations; in the present case, a simple conical diffuser was considered, with correlations taken from accepted literature sources [6, 7]. The value of ξ_D depends mainly on the ratio LDR, and on the value of Reynolds and Mach numbers.

Finally, knowing the thermodynamic conditions of all the circuit points, it is possible to evaluate the entropy and consequently the exergy, using the definitions [8, 9]:

$$e_i = \left(h_{o,i} - h_a\right) - T_a\left(s_{o,i} - s_a\right) \tag{12}$$

$$E = \dot{m}\, e \tag{13}$$

Dealing with a complete thermo-fluid dynamics problem, with locally large contributions of kinetic exergy, reference in eq. 11 is done total enthalpies and entropies, so that exergy includes also kinetic terms.

4. Exergy analysis

The exergy analysis is finalized at identifying the system exergy destructions and losses, following the fundamental methodology described in [9]. The fundamental assumptions are:

- Negligible effect of pressure losses in piping and solar collector
- The specific exergy of solar radiation is assumed equal to the global solar radiation I.
- The reference temperature T_a was set to 30°C, as the system performance is evaluated during hot climate conditions

The following exergy destructions can be defined for the system sketched in Figure 1, at the aggregation level object of the present study:

$$E_{D,HE} = (E_1 + E_{15} - E_9 - E_{13}) \tag{14}$$

$$E_{D,N} = (E_9 - E_6) \tag{15}$$

$$E_{D,M} = (E_6 + E_8 - E_{11}) \tag{16}$$

$$E_{D,D} = (E_{11} - E_3) \tag{17}$$

$$E_{D,TV} = (E_4 - E_5) \tag{18}$$

$$E_{D,P} = (E_7 + W_P - E_1) \tag{19}$$

$$E_{D,PSC} = (E_{13} + W_{PSC} - E_{14}) \tag{20}$$

$$E_{D,EV} = (E_5 + E_{17} - E_8 - E_{16}) \tag{21}$$

$$E_{D,SCHT} = (E_{14} + Q_{HE} - E_{15}) \tag{22}$$

Moreover, the system includes two exergy losses (that is, complete dissipation of exergy to the environment):

$$E_{L,C} = (E_3 - E_4 - E_7) \tag{23}$$

$$E_{L,SC} = (Q_S - Q_{HE}) \tag{24}$$

This last (the solar collector exergy loss) is assumed to take place at the sun equivalent radiation temperature level, so that the heat exergy is equal to the heat itself (Carnot factor = 1) [10, 11, 12]. This is a relatively crude model, which could be improved referring to more advanced model for the exergy of solar radiation [13, 14]; however, as the present analysis is dealing with the low-temperature end of the process of conversion (into cold) of solar energy, the assumption of the simplified model is considered adequate.

The system exergy output is given by(E_{16} - E_{17}) while the inlet exergy is under the above simplifying assumptions $E_{in} = Q_S + W_P + W_{PSC}$, so that the exergetic efficiency can be calculated directly as:

$$\eta_{x,D} = \frac{(E_{16} - E_{17})}{E_{in}} \tag{25}$$

Or indirectly as:

$$\eta_{x,ID} = 1 - \sum_k \frac{E_{DL,k}}{E_{in}} = 1 - \sum_k E_{DLR,k} \tag{26}$$

Where $E_{DL,k}$ represents the exergy destruction or loss in the k-th system component, and $E_{DL,k}$ its non-dimensional value. Similarly, an exergetic efficiency $\eta_{x,CC}$ can be defined with reference to the ejector cooling cycle alone, making reference to $E_{in,CC} = E_9 - E_1 + W_P$. The direct/indirect formulations were used to check the correctness of the exergy balance.

5. Thermoeconomic analysis

The thermoeconomic analysis is run following the direct approach described in [9]. All cost rates Z include capital cost, operation and maintenance, and were levelized to initial time values. Starting from the solar collector field outlet:

$$c_{15} E_{15} = c_{14} E_{14} + Z_{SC} \tag{27}$$

In this equation, the cost of the solar radiation is taken as zero, assuming that we are dealing with a free natural resource.

While at collector field and pump inlet:

$$c_{14} E_{14} = c_{13} E_{13} + c_w W_{PSC} + Z_{PSC} \tag{28}$$

Passing to the coolant fluid, at heat exchanger outlet:

$$c_9 E_9 = c_{15} \left(E_{15} - E_{13}\right) + c_1 E_1 + Z_{HE} \tag{29}$$

The ejector is a compact and expensive device; it is impossible to divide its cost into its three sections (nozzle, mixing chamber, diffuser), so that it was assumed that its component cost was to be divided into three equal parts; under this assumption:

$$c_6 E_6 = c_9 E_9 + \frac{1}{3} Z_{EJ} \tag{30}$$

$$c_{11} E_{11} = c_6 E_6 + c_8 E_8 + \frac{1}{3} Z_{EJ} \tag{31}$$

$$c_3 E_3 = c_{11} E_{11} + \frac{1}{3} Z_{EJ} \tag{32}$$

After the condenser, the flow is divided into two streams, so that:

$$c_4 \left(E_7 + E_4\right) = c_3 E_3 + Z_C \tag{33}$$

and $c_7 = c_4$.

It is now possible to close the left branch loop (Figure 1):

$$c_1 E_1 = c_7 E_7 + c_w W_P + Z_P \qquad (34)$$

Which is linked as input to Eq. 29. On the right branch, proceeding after point 4 we have the throttle valve:

$$c_5 E_5 = c_4 E_4 + Z_{TV} \qquad (35)$$

At the evaporator, the cost of the cold stream is constant so that $c_8 = c_5$, which provides the input to Eq. 31 in order to close the right branch. It is finally possible to evaluate the cost of the cooled hot stream, which is the product to be distributed to the dwelling $c_{cold} = c_{16}$, and is the objective of the thermoeconomic analysis:

$$c_{16} E_{16} = c_5 \left(E_5 - E_8\right) + c_{17} E_{17} + Z_{EV} \qquad (36)$$

In Eq. 36, it is assumed $c_{17} = 0$ as m_{17} is the no-cost return flow of the cooling fluid which is distributed in the dwelling (moreover, the exergy of this flow is very low being close to ambient temperature).

The whole system (including all balances of mass, energy, exergy and thermoeconomic variables) includes over 400 equations, which are solved by the standard dispersed matrix equation solver included in EES [15].

6. Results of the simulation

6.1 –Thermo-fluid dynamics design issues – selection of proper working fluid

Some critical points can arise in ejector cooling systems because of the extreme fluid dynamics performance conditions – notably, a high Mach number in the ejector. In order to limit this last to reasonable (but supersonic) values, it is possible to adjust the value of MR: however, this has direct consequences over the primary flow rate. This last, together with the fluid properties (notably, the temperature at end of compression, point 1, and the amount of heat to be provided for heating the working fluid from point 1 to saturated liquid conditions, corresponding to the heat exchanger pinch point) determines the temperature at solar collector inlet T_{14}. This temperature, for a fixed value of the solar collector field outlet temperature T_{15}, determines the thermal performance of the solar collector through Eq. 6. This means that there is a link between the fluid dynamics performance of the ejector and the thermodynamic performance of the solar collector loop.

Three working fluids were considered as potential candidates for the ejector cooling circuit: the base case R134a, R113 and R236fa. The thermodynamic properties were provided by the EES environment simulation, with a real fluid model used in every point (including the supersonic ejector and mixing chamber). The assumed operating conditions are summarized in Table 1:

Table 1. Summary of assumed operating conditions and parameters

Variable	*Description*	*Value*
T_{15}	Solar collector outlet temperature	95 °C
T_2	Working fluid evaporation temperature	78 °C
T_4	Condenser temperature	38 °C
T_5	Evaporator inlet temperature	10 °C
T_{16}	Cooling fluid production temperature	15 °C
$T_a = T_{17}$	Ambient temperature	30 °C
DT_{pp}	Heat exchanger pinch temperature difference	10 °C
η_P	Working fluid pump efficiency	0,50
η_{PSC}	Solar collector field pump efficiency	0,70
LDR	Diffuser L/D ratio	12
I	Design average global radiation	700 W/m^2
Q_{EV}	Evaporator cooling load	5 kW

The collector field is assembled using FDS ICARO units [16], each having a net absorber area of 1,01 m^2, η_o = 0,679, a_1 = 0,1696 (°Cm^2)/W, a_2 = 0,099 (°C^{2}m^2)/W.

For each working fluid, a careful research of the optimal sizing was performed, with the objective of maximizing the COP. Table 2 reports the resulting fundamental calculated values forthe performance/sizing indicators.

Table 2. Design calculated performance(upper)and sizing (lower) indicators

Variable	R134a	R113	R236fa
COP	0,29	0,343	0,363
Ma_6	1,85	2,22	1,96
η_{SC}	0,484	0,480	0,483
MR	0,35	0,46	0,5
n_{SC}	51	43	40
d_5, mm	5	18	7
T_{13}, °C	84,4	86	84,6
m_{ev}, kg/s	0,033	0,038	0,042
m_{SC}, kg/s	0,095	0,084	0,084

6.2–Exergy destructions and losses

Referring to the design conditions for the three reference fluids reported in Table 2, the exergy balance was calculated. The exergy destructions and losses, as well as the exergetic efficiency, are reported in Table 3.

Table 3. Design calculated exergy balance(absolute and non-dimensional)

Variable	R134a		R113		R236fa	
	W		W		W	
ED_{HE}	754	0,021	432	0,0142	605	0,021
ED_N	37	0,0010	29	0,00095	26	0,00092
ED_M	1054	0,029	1090	0,0359	751	0,0264
ED_D	88	0,0024	45	0,0015	64	0,0022
ED_{TV}	76	0,0021	47	0,0016	70	0,0025
ED_P	124	0,0034	21	0,00069	40	0,0014
ED_{PSC}	30	0,00082	29	0,00096	24	0,00084
ED_{EV}	113	0,0031	108	0,0035	27	0,00095
ED_{SCHT}	14700	0,4027	12210	0,4018	11490	0,4036
EL_C	754	0,021	535	0,018	629	0,022
EL_{SC}	18660	0,5108	15700	0,5170	14600	0,5133
E_{in} kW	36,5		30,4		28,6	
E_{out} kW	0,130		0,130		0,130	
η_{xCC}	0,0547		0,0648		0,0748	
η_x	0,00356		0,00426		0,00457	

6.3–Thermo-economic analysis – Cost of cooling

Referring to the sizing indicators resumed in the lower part of Table 2, an estimate of capital costs (including maintenance) was performed recovering data from literature [17, 18] or directly from manufacturers (in the case of the solar collector). For the ejector, which is a non-standard component, a fixed cost of 1000€ was assumed in all cases. The costs of the evaporator, Heat Exchanger/Vapor Generator, and throttle valve are the same for the three working fluids here considered. For electricity (pump work), the retail price of 0,22 €/kWh was considered. An operation time of the cooling service of 1250 h/year was assumed, for a lifetime of 20 years.

Table4. Levelizedcomponent cost data (€)

Component	R134a	R113	R236fa
Pump	200	90	140
Solar collector*	500	500	500
Condenser	6210	5450	4800
Evaporator	1340	1340	1340
Throttle Valve (T-controlled)	70	70	70
Pump (Solar field)	100	100	90
Overall system cost	35420	30550	28440

*unit cost, number of collectors n_{SC} from Table 2

The results in terms of buildup of the cost of unit exergy across the system components are resumed in Table 5.

Table 5. Calculated stream costs (€/MJ)

Cost of the stream	R134a	R113	R236fa
c_1	1,23	1,20	1,35
c_3	0,81	0,75	0,70
c_4	1,28	1,19	1,41
c_5	1,68	1,43	2,04
c_6	0,82	0,27	0,62
c_7	1,28	1,19	1,41
c_8	1,68	1,43	2,04
c_9	0,81	0,27	0,61
c_{11}	0,79	0,71	0,68
c_{13}	0,10	0,10	0,10
c_{14}	0,10	0,10	0,10
c_{15}	0,10	0,10	0,10
c_{16}	3,27	2,73	2,58

c_{16} is the cost per unit exergy of cold produced at the evaporator. As the total exergy output rate at the evaporator is 0,130 kW, for a cold energy production of 5 kW, the cost of the cold produced per unit energy results for the best case (R236fa) equal to 0,067 €/MJ. This was compared to the cost of cold production from a conventional compression cooling system using electricity (COP = 4; overall cost 2000€), which is about 0,015 €/MJ. On the other hand, the traditional system consumes about 1500 kWh/year, while the ejection system, running on renewable energy, uses electricity only for the operation of pumps (225 kWh/year). A more accurate evaluation should consider year-round off-design operation with variable radiation, which is beyond the purpose of the present analysis. Profitability is largely hindered by the much higher capital cost, with respect to the traditional solution: in practice, a high investment is paid back with an annual saving of about 300 €/year of electricity, which at present makes this system unprofitable in terms of payback time.

7. Analysis of the results - Conclusions

The results presented in Section 6 can be interpreted as follows.

From the thermo-fluid-dynamics point of view, design conditions were identified and a preliminary sizing of the system was performed. Looking at the data in Table 2, R236fa appears to be the best working fluid. In fact, the use of R236fa allows to achieve a better COP in the ejector cycle; the ejector is operated with a MR value well within the recommended range; the ejector Mach number (Ma_6 = 1,94) is high but this is common in all ejector cooling cycles, and the Mach value can be placed in the lower range with respect to the design choice of several researchers. This value of Ma should be effective in producing the oblique shock wave pattern which is largely responsible of a good entrainment performance [3, 19]. The size of piping is reasonable (d_7 = 7 mm at the evaporator), and the lower number of solar collectors needed (n_{SC} = 40; a result of the better heat capacity matching in the heat exchanger) promises an appreciable reduction of the capital cost, which is largely determined by the solar field size.

The exergy analysis (Table 3) confirms the better performance of R236fa. As the objective is to provide a product of E_{16} = 0,13 kW (for all working fluids), the performance is well measured by the lowest value of the input exergy E_{in} = 28,6 kW, opposed to 30,4 kW for R113 and 36,5 kW for R134a. As most of the input exergy is sun radiation, this result is traduced effectively by the lower number of collectors required. The values of the exergy efficiency are low, but it should be reminded that this systems is mainly using a renewable resource, and without any concentration. Even considering the small numbers, R236fa is superior for this application from the point of view of ejector cycle efficiency (nearly 7,5%; this is reflected by homogeneous and low values of exergy destructions in all components of the ejector cooling cycle); however, the main result is the notable decrease of the exergy destruction and exergy loss in the solar collector field, which are the largest negative contributions to the exergy balance; again, this is a direct consequence of being able to satisfy the required cooling load with a reduced solar collector field.

From the thermoeconomics point of view, the proposed system cannot be considered profitable in terms of payback time under the simplified hypotheses of this preliminary study. The performance achieved by the best case (R236fa) leads to a cost of cold which is about 4,4 times that produced by a traditional compression 5 kW cooling unit, running on grid electricity. If this same unit were run with locally produced PV electricity, the ejection cooling system would be profitable at present conditions. Moreover, the cost of the collectors - which represents on the whole about 80% of the capital cost - should be accounted only for the cooling mode operation (the same collectors can be used with simple switch arrangements and adjustments to provide solar heating during winter). The Solar-Driven Ejector Cycle produces a COP (0,3 – 0,36) which is competitive to that of a solar-driven absorption cycle using non-concentrating solar collectors.

A more detailed study should consider off-design operation with variable radiation along the year, including a low-temperature heat storage system (possibly a phase-change storage) to minimize the expected large penalty due to off-design operation.

Nomenclature

A	area, m^2
a_1, a_2	solar collector performance coefficients
c	cost of exergy, €/J
COP	Coefficient of Performance
D	diameter, m
e	specific exergy, J/kg
E	exergy, W
ED	Exergy destruction, W
EDR	Exergy destruction, non-dimensional
EL	Exergy loss, W
ELR	Exergy loss, non-dimensional
h	enthalpy, J/kg
I	solar global radiation incident on the collector surface, W/m^2
L	diffuser length, m
LDR	Length/Diameter ratio of the conical diffuser
Ma	Mach Number
MR	ejector mass flow ratio, $MR = \dot{m}_{EV}/\dot{m}_P$
$\dot{m}$	mass flow rate, kg/s

p	pressure, Pa
Q	heat rate, kW
s	entropy, J/(kgK)
T	temperature, °C
v	velocity, m/s
W	power, W
Z	component cost rate(capital + operation and maintenance), €/s

Greek symbols

η	efficiency
ρ	density, kg/m^3
ξ_D	diffuser loss coefficient

Subscripts and superscripts

a	Ambient
av	Average
C	Condenser
D	Diffuser
EJ	Ejector
EV	Evaporator
HE	Heat Exchanger (Vapor Generator)
HT	Heat Transfer
in	Input
M	Mixing
N	Nozzle
o	Total
out	Outlet
p	Primary
P	Pump
PSC	Pump, Solar Collector loop
S	Sun
SC	Solar Collector
TV	Throttle Valve
x	Exergy
x,CC	Exergy, cooling cycle only
x,D	Exergy, direct
x,ID	Exergy, indirect

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