

Analysis of a Solar Assisted Ejector Cooling Cycle for Air Conditioning in a Mediterranean Climate

Szabolcs Varga¹, Armando C. Oliveira^{1*} and Bogdan Diaconu^{1,2}

¹Faculty of Engineering, University of Porto
Dept. Mechanical Eng. & Industrial Management
Rua Roberto Frias, 4200-465 Porto, Portugal

²University "Constantin Brancusi" of Tg-Jiu
Str. Republicii nr.1, 210152 - Tg-Jiu, Romania

* acoliv@fe.up.pt

Abstract: Using low grade thermal energy instead of electricity to operate a refrigerator can have important environmental benefits, especially when it is powered by a renewable energy source (e.g. solar thermal). Ejector refrigeration is one of the most promising technologies because of its relative simplicity and low initial cost. In this work, a theoretical study has been carried out to asses system and refrigeration efficiencies of a solar assisted ejector cycle using water as the operating fluid.

The model was based on a 1D ejector approach, including both the refrigeration and solar collector cycles. In the development of the mathematical model, several assumptions were made e.g. steady state, negligible thermal losses during transport, constant ejector efficiencies. Ejector performance was evaluated for different operating conditions. The results indicated that in order to achieve an acceptable COP, generator temperatures should not fall below 90°C. Evaporator temperatures below 10°C and condenser temperatures over 35°C resulted in a significant drop of system efficiency, and therefore these conditions can be identified as minimal (reference) design values. The required solar collector area to provide 5 kW of cooling power was calculated for different operating conditions. Ejector dimensions were also calculated using the constant pressure mixing ejector theory.

Keywords: ejectors, solar thermal, modelling, operating conditions

Nomenclature

A area [m²]
COP coefficient of performance
d diameter
F_R(τ_α) collector optical efficiency
F_RU_L collector loss coefficient [W/m²K]
F_{sol} solar fraction
h specific enthalpy [J/kg]

I solar radiation [W/m²]
m mass flow rate [kg/s]
Q power, [W]
T remperature [K]
v velocity [m/s]
W pumping power [W]
η efficiency
λ entrainment ratio

Subscripts

a	auxiliary
coll	collector
entr	entrainment
ex	exit
g	generator
in	inlet
is	isentropic
nozz	nozzle
sys	system
∞	ambient

1. INTRODUCTION

The urban style of living in southern and eastern Mediterranean countries has changed significantly over recent decades and this has been accompanied by a massive increase in the use of electrical energy. Air conditioning equipment is one of the main energy consumers in this region. Solar driven ejector cooling seems can be an attractive option, due to its capacity to use low temperature heat supply [1] such as flat plate or vacuum tube solar collectors [2].

A number of studies can be found in the literature concerning solar ejector cooling. A theoretical work on collector selection can be found in [2], for different operating conditions using R141b as working fluid. It

collectors economically more attractive in spite their lower efficiency. This conclusion was re-evaluated later by [3]. R600a was used as working fluid in this theoretical work. Based on yearly dynamic simulations, it was concluded that vacuum tube solar collectors are more attractive than flat plate type due to their significantly lower need for auxiliary heating. Water as a working fluid was used in [4]. An attractive COP of 0.3 was obtained in a test location in the UK. No details concerning the solar collector area were given. Water was also used as refrigerant in the modelling study presented by [5]. Very low generator temperatures (50 °C) and cooling loads (~400 W) were considered. A review of ejector applications (including solar driven) for refrigeration purposes can be found in [6].

A schematic representation of a solar driven ejector cooling system, consisting of a collector circuit and an ejector cooling cycle, is shown in Fig.1. The major components of the solar collector circuit are the collector array, a storage tank and an auxiliary heater (Q_a) which is an additional heat source to provide constant inlet temperature (T_s) for the generator. The refrigerant is vaporised on the ejector cycle side of the generator, at relatively high pressure. This high pressure vapour is accelerated to supersonic velocity through

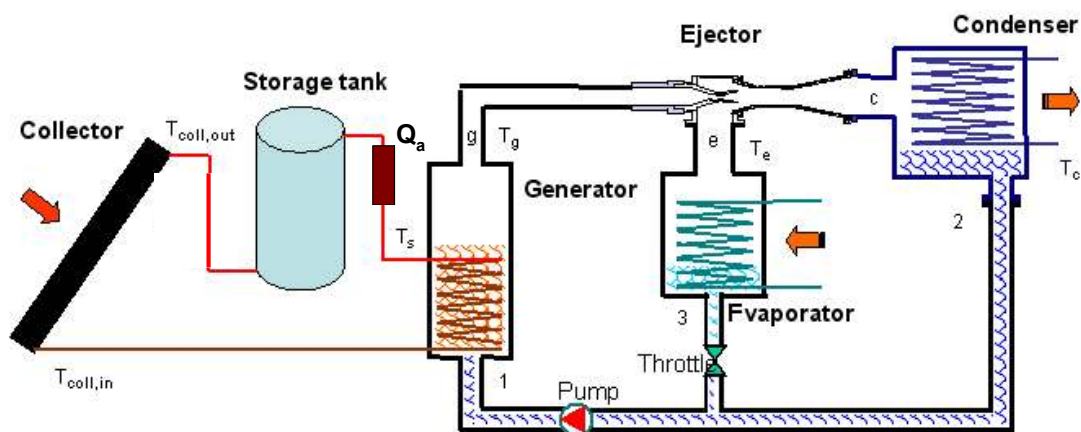


Figure 1: Schematic representation of the solar assisted ejector cooling system

was concluded that the high price of vacuum tube collectors makes the use of flat plate

the primary nozzle of the ejector (primary flow) creating low pressure at the exit

section. This low pressure entrains the secondary flow coming from the evaporator where the cooling effect is obtained. In the converging section of the suction chamber, a shear layer between the motive and secondary fluids develops and the secondary fluid gets to be accelerated to sonic velocity (mixing). After the mixing process is completed, a final shock occurs, resulting in a subsonic downstream condition. The pressure is then further increased in the subsonic diffuser to the conditions present in the condenser. The cycle is closed by returning the condensate to the generator, using a pump, and to the evaporator through an expansion valve. Detailed discussion of the ejector operation can be found e.g. in [7].

In the present study, the performance characteristics of the solar driven ejector cycle is analysed theoretically, in order to identify potential design conditions for an actual application in Mediterranean countries. This is a first step towards the development of the system described in [8].

2. MATHEMATICAL DESCRIPTION

In the mathematical model of the solar assisted ejector cooler only steady state conditions were considered. It was assumed that the thermal losses during the fluid transport were negligible. The model was a simple 1D approach based on energy, mass, and momentum conservation equations. Water was selected as working fluid for the ejector cycle as well as in the solar collectors, since it represents no risk to the environment and compared to other refrigerants water is easily available.

2.1 Solar collector circuit

The performance of the solar collector array can be assessed by the solar collector efficiency (η_{coll}) given by:

$$\eta_{coll} = F_R(\tau\alpha) - F_R U_L \left(\frac{T_{coll,in} - T_\infty}{I} \right) \quad (1)$$

Typical values for the optical efficiency ($F_R(\tau\alpha)$) and the loss coefficient ($F_R U_L$) can

be found in [9]. From Eq.1 the useful heat can be obtained as:

$$Q_{coll} = \eta_{coll} A_{coll} I \quad (2)$$

In this work the thermal inertia caused by the storage tank was not modelled. The required thermal input from the auxiliary heater was simply determined from:

$$Q_a = Q_g - Q_{coll} \quad (3)$$

It was assumed that T_s was 10°C higher than the generator temperature T_g .

2.2 Ejector cooling cycle

A typical ejector is shown in Fig.2. Several simplifying assumptions were made in the theoretical analysis of the ejector cycle. It was considered that the ejector was adiabatic. The fluid was saturated vapour at point e and g (see Fig.2) and saturated liquid at point 2. Ejector inlet and outlet velocities were neglected. Irreversibilities in the primary nozzle, mixing chamber and diffuser were accounted for by applying efficiency constants. More detailed discussion of the constant pressure mixing ejector theory can be found in [7] and [10].

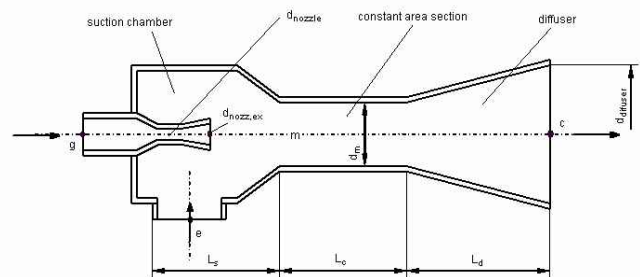


Figure 2: Typical ejector.

The energy balance over the ejector can be written as:

$$\dot{m}_g h_g + \dot{m}_e h_e = (\dot{m}_g + \dot{m}_e) h_c \quad (4)$$

In the primary nozzle, the motive fluid undergoes an adiabatic expansion resulting in a high velocity and low pressure at the exit. The nozzle exit velocity and enthalpy can be calculated from:

$$v_{nozz,ex} = \sqrt{2(h_g - h_{nozz,ex})} \quad (5)$$

$$h_{nozz,ex} = h_g - \eta_{nozz} (h_g - h_{nozz,ex,is}) \quad (6)$$

Mixing of the primary and secondary flow occurs at constant pressure by the

motive fluid partially transmitting its kinetic energy to the secondary flow. Applying an entrainment efficiency constant (η_{entr}), the energy balance of the mixing process can be written as [11]:

$$\eta_{entr} \dot{m}_g (h_g - h_{nozz,ex}) = (\dot{m}_g + \dot{m}_e)(h_c - h_m) \quad (7)$$

The ejector exit enthalpy can be obtained assuming a negligible exit velocity by:

$$h_c = h_m + (h_{c,is} - h_m) / \eta_{dif} \quad (8)$$

The condenser output, pumping and generator power were calculated as:

$$Q_c = (\dot{m}_g + \dot{m}_e)(h_c - h_2) \quad (9)$$

$$Q_g = \dot{m}_g (h_g - h_1) \quad (10)$$

$$W_{pump} = \dot{m}_g (p_g - p_c) / \rho \quad (11)$$

The required secondary mass flow rate to provide Q_e cooling capacity in the evaporator is given by:

$$\dot{m}_e = Q_g / (h_e - h_3) \quad (12)$$

In the present work, Eqs 1-12 were solved using EES program package (F-Chart Software, Madison). Physical properties of water were determined using the physical property functions of EES [12]. Nozzle, entrainment and diffuser efficiencies were taken as 0.85, 0.7 and 0.7 respectively.

2.3 Performance of the ejector cycle

The performance of a refrigeration cycle is generally expressed through the coefficient of performance, which is the output cooling power for a unit energy input:

$$COP = Q_e / (Q_g + W) \quad (13)$$

Thus the system efficiency can be estimated by [2]:

$$\eta_{sys} = \eta_{coll} \times COP \quad (14)$$

Ejector performance can also be measured by the entrainment ratio (λ) defined as:

$$\lambda = \dot{m}_e / \dot{m}_g \quad (15)$$

For a given cooling power, the required evaporator flow rate is approximately constant. The higher the entrainment ratio is, the lower the flow rate on the primary

nozzle side and consequently the lower the required generator power.

Since the main source for the ejector cooling system is solar energy, the overall effectiveness can be quantified through the ratio of the useful solar power to the power necessary in the generator to run the system:

$$F_{sol} = Q_{coll} / Q_g \quad (16)$$

3. Results and discussion

COP and system efficiency as a function of condenser and generator temperatures are shown in Fig.3. It is clear from the figure that the performance of the ejector cooling cycle decreases quickly with increasing condenser temperature. Considering climate conditions in Mediterranean countries, it is likely that condenser temperatures would be 35°C or higher. Calculated COP for 90°C generator temperature was 0.13. This can be only increased by increasing the generator temperature, e.g. for $T_g = 110^\circ\text{C}$ the calculated COP was 0.31. Within the temperature range considered in this work, system efficiency increased with generator temperature. It is worth noting however that T_g cannot be increased beyond a given temperature without having a negative impact on system efficiency. Although COP increases with generator temperature, the collector efficiency decreases, which can result in an actual decrease in η_{sys} for very high values of T_g [2].

Cooling cycle and system efficiencies increase with evaporator temperature, as shown in Fig.4. As can be seen, for a generator temperature of 110°C, the evaporator temperature should be about 7°C or higher to achieve a COP of at least 0.2. For lower generator temperatures, e.g. 90°C, the required evaporator temperature should be as high as 13°C to obtain the same COP. In a typical air conditioning unit, where the evaporator temperature is about 10°C, the generator should operate at least at 100°C.

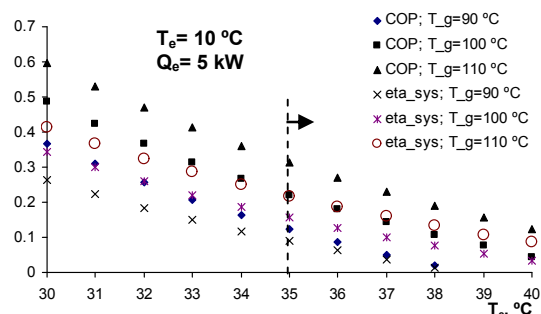


Figure 3: COP and system efficiency for different condenser and generator temperatures

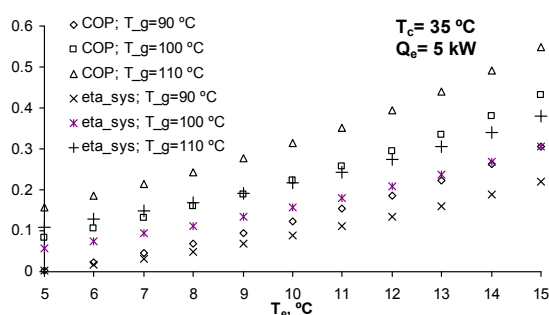


Figure 4: The effect of evaporator temperature on COP and system efficiency for different generator temperatures.

The required collector area to provide the necessary generator input (with $F_{sol}=1$) at a constant incident radiation of 800 W/m^2 was calculated. The cooling capacity of the ejector cycle was set to 5 kW . It was assumed that collectors were of evacuated tube type. Fig.5 shows the collector area as a function of condenser and generator temperatures. It can be seen that the required solar collector area increases exponentially with condenser temperature. A very large collector area would increase significantly the initial cost of such a system. Considering a 50 m^2 collector field, the maximum considered as economically feasible, it is clear from the figure that the generator temperature should be at least 100°C for a condenser temperature of 35°C .

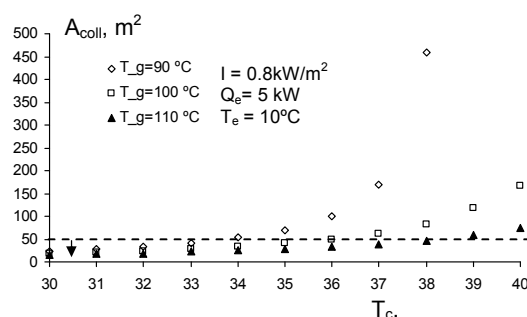


Figure 5: Required solar collector area as a function of condenser temperature for different generator conditions.

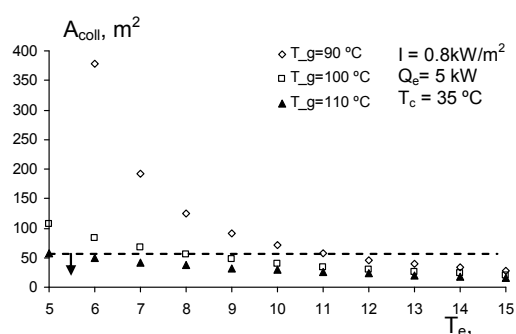


Figure 6: Solar collector area as a function of evaporator temperature for different generator conditions.

Fig.6 shows the effect of evaporator temperature on the required collector area. The collector area seems to be more sensitive on evaporator conditions for low generator temperatures (90°C). As the generator temperature increases, A_{coll} becomes less sensitive on evaporator conditions. In general, the required collector area decreases with evaporator temperature due to the increase in COP and the corresponding decrease in the generator input to provide a constant cooling capacity. It can also be seen from the figure that for a collector field of 50 m^2 , the generator can be operated without auxiliary power input (except for the pump) for an evaporator temperature as low as 5°C if a T_g of 110°C can be reached. For lower T_g the minimum evaporator temperature should be higher, e.g. 11°C for $T_g=90^\circ\text{C}$.

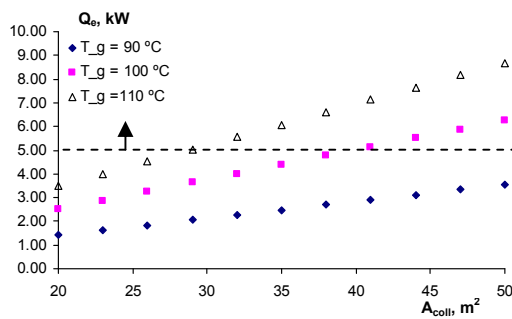


Figure 7: Cooling capacity vs. solar collector area for different generator conditions ($T_e = 10^\circ\text{C}$, $T_c = 35^\circ\text{C}$, $I = 800 \text{ W/m}^2$).

The estimated cooling capacity of the ejector cooling cycle as a function of A_{coll} is shown in Fig.7. It can be seen that the cooling capacity of the system increases almost linearly with collector area. For a generator temperature of 90°C , a 50 m^2 evacuated tube solar collector array can only provide a cooling capacity of $\sim 3.5 \text{ kW}$ without auxiliary power input assuming 800 W/m^2 solar radiation. The same power output can be expected for a collector area of about half if the collectors supply the generator at 110°C . For 5 kW cooling power, A_{coll} should be about 40 m^2 and 29 m^2 for T_g of 100 and 110°C , respectively.

Ejector dimensions for different operating conditions and 5 kW cooling capacity were estimated based on an ejector model similar to [10]. The results are summarised in Table I. Operating conditions and flow rates were given as input parameters, the major dimensions were outputs as shown in the table. The diameter of the primary nozzle depended only on the

mass flow rate of the fluid through the generator. Since the secondary flow rate is practically constant for a given cooling capacity, the higher the entrainment ratio, the smaller the primary nozzle is. The exit diameter of the primary nozzle depended on flow rate, T_g and downstream conditions. This is indicated by the diameter ratio ($d_{\text{ratio}} = d_{\text{nozz,ex}}/d_{\text{noz}}$). Increasing back pressure and decreasing T_g resulted in a decrease in d_{ratio} . The mixing section diameter (d_m) mostly depended on entrainment ratio. Higher λ resulted in smaller d_m . Improving the entrainment ratio can decrease the ejector size. It can also be seen from Table I that very low generator (90°C) and evaporator (5°C) temperatures resulted in very poor performance and consequently impractical ejector size, considering water as working fluid.

4. Conclusions

Theoretical analysis of a solar assisted ejector cooling system based on a simplified 1D approach has been carried out. The system performance and required collector area to operate a small scale air-conditioning device for use in Mediterranean countries was studied using water as working fluid.

The results indicated that both COP and system efficiencies increased with generator temperature, within the range considered in this work. In order to obtain acceptable values, generator temperature should not be below 90°C . This would require a collector output temperature of about 100°C . Therefore, evacuated tube collectors are

Table I: Ejector dimensions for different operating conditions

$T_g, ^\circ\text{C}$	$T_e, ^\circ\text{C}$	$T_c, ^\circ\text{C}$	mass flow, kg/s		λ	Diameter, m		d_{ratio}	Diameter, m		
			generator	evaporator		d_{noz}	$d_{\text{noz,ex}}$		secondary inlet	d_m	diffuser
110	10	35	0.0063	0.00211	0.34	0.006	0.020	6.8	0.162-0.186	0.043	0.103-0.143
100	10	35	0.00893	0.00211	0.24	0.009	0.025	5.2	0.172-0.198	0.045	0.109-0.152
90	10	35	0.0161	0.00211	0.13	0.014	0.035	3.7	0.198-0.227	0.052	0.126-0.175
100	10	40	0.0444	0.00213	0.05	0.020	0.055	3.5	0.259-0.298	0.068	0.165-0.230
110	10	40	0.0161	0.00213	0.13	0.010	0.031	5.0	0.189-0.217	0.050	0.120-0.167
90	5	35	0.915	0.00212	0.00	0.106	0.297	2.9	1.179-1.353	0.311	0.749-1.044

better suited for operating an ejector cooling system. Evaporator temperatures below 10°C also resulted in very poor system performance ($COP < 0.1$). Ejector cooling systems using water as working fluid are not suitable at very low evaporator temperatures.

For high values of condenser temperature ($>35^{\circ}\text{C}$) and low values of evaporator temperature (10°C) the required solar collector area is large ($>50\text{m}^2$). Auxiliary heating is required even for relatively high solar radiation (800 W/m^2), as is the case in Mediterranean countries.

The presented analysis was a simplified approach. It is believed that using a more detailed approach, e.g. CFD, ejector geometry can be optimised to improve COP by increasing the entrainment ratio.

ACKNOWLEDGEMENTS

The work was developed within the framework of the Mediterranean-Aircond Project, which was funded by the Commission of the European Union (DG Research), through the Energy research programme (FP6): contract INCO-CT2006-032227. The authors would like to acknowledge other project partners are also.

REFERENCES

- [1]Vidal, H, Colle, S & Perreira, G S. 2006. Modelling and hourly simulation of a solar ejector cooling system. *Applied thermal Engineering*. Vol 26. pp663-672.
- [2]Huang, B J, Petrenko, V A, Samofatov, I YA & Shchetinina, N A. 2001. Collector selection for solar ejector cooling system. *Solar Energy*. Vol 71. No 4. pp269-274.
- [3]Pridasawas, W & Lundqvist, P. 2007. A year-round dynamic simulation of a solar-driven ejector refrigeration system with iso-butane as a refrigerant. *International Journal of Refrigeration*. Vol 30. pp840-850.
- [4]Nguyen, V M, Riffat, S B & Doherty, P S. 2001. Developmen of a solar-powered passive ejector cooling system. *Applied thermal Engineering*. Vol 21. pp157-168.
- [5]Khattab, N M & Barakat, M H. 2002. Modeling the design and performance characteristics of solar steam-jet cooling for comfort air conditioning. *Solar Energy*. Vol. 74. pp257-267.
- [6]Chunnanond, K & Aphornratana, S. 2004. Ejectors: application in refrigeration technology. *Renewable and Sustainable Energy Reviews*. Vol. 8. pp129-155.
- [7]Eames, I W, Aphornratana, S & Haider, H. 1995. A theoretical and experimental study of a small-scale steam jet refrigerator. *International Journal of Refrigeration*. Vol 18. No 6. pp378-386.
- [8]Oliveira, A C et al. 2007. Presentation of a Novel Solar Air Conditioning System Using a Hybrid Ejector/Exchanger Unit. *Proceedings SET2007, Chile*.
- [9]Pridasawas, W. 2006. Solar driven refrigeration systems with focus on the ejector cycle. Ph.D Thesis. Royal Institute of Technology. Stockholm, Sweden.
- [10]Huang, B J, Chang, J M, Wang, C P & Petrenko, V A. 1999. A 1-D analysis of ejector performance. *International Journal of Refrigeration*. Vol 22. pp354-364.
- [11]Tyagi, K P & Murty, K N. 1985. Ejector-compression system for cooling: utilising low grade waste heat. *Heat Recovery Systems*. Vol 5. pp545-550.
- [12]ESS user manual, Madison, USA: F-Chart Software.