Optimization of A Solar Ejector Refrigeration System in the Region of Adrar, Algeria

Zoheir Derghout[1,2,3,*] and Mohammed Salah Rouabah[1]

[1]Department of Climatic Engineering, Brothers Mentouri Constantine 1 University, Constantine, Algeria
[2]Department of Mechanical Engineering, Kasdi Merbah-Ouargla University, Ouargla, Algeria
[3]Energy and Environment Laboratory, Salah Boubnider Constantine 3 University, Constantine, Algeria

([*]Corresponding author's e-mail: z.derghout@umc.edu.dz)

Abstract

This study aims to simulate a solar ejector refrigeration system in the region of Adrar (27.82 N, 0.18 W). It is the first investigation of such refrigeration system in this region, the importance of this study is to benefit from the free intense solar energy for cooling and air conditioning and reduce using electrical energy produced by thermal power station which is harmful to the environment and causing warming global. It is a theoretical study using real gas equation (REFPROP database). The system is inspected for various evaporator temperatures from 0 to 30 °C and for various condenser temperatures from 25 to 50 °C. For each situation, the system is optimized by choosing the best ejector parameters and the ideal generator temperature. Different refrigerants are used as R11, R134a and R717 (ammonia). The last one is observed to be the best fluid in all the working conditions. More explicitly, the highest COPs is 0.28 when the system produces refrigeration at 30 °C and the boiler temperature is 90 °C. The COPs values are acceptable and very interesting, and it's higher than other cities in the world like Athens, this study can be generalized in other hot regions. The examination is performed with a code written in Fortran.

Keywords: Ejector refrigeration systems, Solar energy, Performance, Simulation, Optimization

Nomenclature:

A: Surface (m^2)
COP: Coefficient of performance
COPs: Whole coefficient of performance
D: Diameter (m)
h: Enthalpy (Kj/Kg)
M: Mach number
η_{SC}: Solar collector yield
I: Solar radiation (W/m^2)
Lx: Elementary length (m)
$\dot{m}_1$: Primary mass flow rate (Kg/s)
$\dot{m}_2$: Secondary mass flow rate (Kg/s)
P: Pression (KPa)
Q: Capacity (kW)
T_e: Evaporator temperature (°C)
T_c: Condenser evaporator temperature (°C)
T_b: Boiler temperature (°C)

Subscripts
c: Col
ODP: Ozone depletion potential
GWP: Global warming potential
Sc: Solar collector
U: Optimal training ratio
W: Work

Greek symbols
$\mathcal{E}$: Motor ratio

Introduction

Air conditioning and refrigeration systems are among the most energy demanding equipment in the building sector [1]. On another side, cooling systems are with the most energy-consuming systems either in domestic applications or in the industrial sector. It will often require cheaper, economic and ecological systems, especially in hot and arid zones such as Southern Algeria. According to global estimates, the refrigeration system amount to 15 % of the world's electricity consumption [2]. In Algeria, most of this energy is produced by thermal power station [3] which emit greenhouse gases and use fossil fuels. These, have not only a negative effect on the health of the human being, caused by the general pollution they incur, but also, they seriously damage the whole environment quality (including global warming, destruction of the ozone layer and others) due to the several toxic gas emissions.

To reduce the impact of these problems, it is important to pay an interest to new techniques using clean and economical energy like solar or geothermal energy and exploiting waste heat as energy sources for the operation of different devices. The compressor in the mechanical vapor refrigeration system is the main responsible for the largest proportion of the electrical energy consumption [4]. In such case, the solution is to replace it with a thermal compressor composed of 2 parts, ejector and boiler.

The ejector is an immobile device intended to suck and mix gases to compress the steam from the evaporator to reach the condensation pressure. Ejector cooling technology is attractive compared to conventional mechanical compression system because it is simple, less expensive and its electrical energy consumption is negligible compared to the thermal energy generated by the boiler.

Many studies of a solar ejector refrigeration system realized in the past, Al-Khalidy [5] in 1997 made an experimental study of ejector refrigeration cycle using R113 powered by solar energy. The solar collector used in the study is a solar collector with synthetic oil, the maximum efficiency of the solar collector is 20 %. The system used for both mode, refrigeration and heating. One year after Huang *et al* [6] proposed a solar ejector cycle model using refrigerant R141b. The results revealed a coefficient of performance equal to 0.5 by using the single-stage experimental cycle. Boiler, evaporation and condensation temperatures are respectively 90, 8 and 28 °C. For solar irradiation equal to 700 W/m^2, the temperatures of the generator and the evaporator can be reached 95 and 8 °C, respectively. The COPs of the whole system is 0.22. In 2005, Alexis and Karayiannis [7] conducted a study on a solar ejector machine for refrigeration applications. The refrigerant used is HFC R134a. Based on climatic data taken from Athens (Greece) statistics from 1952 to 1962. The system operating with the solar collector at intermediate temperature is simulated for the 5 months from May to September. The results show values of the coefficient of performance COP between 0.035 and 0.199 for the following machine working conditions: The temperature of the generator between 82 and 92 °C, the temperature of the condenser between 32 and 40 °C and the evaporator temperature between −10 and 0 °C, in refrigeration mode, the coefficient of the whole system COPs varying from 0.014 to 0.101 for the same working conditions, the radiation between 536 and 838 W/m^2 in July. In 2008, Nehdi *et al* [8] made a study on an air conditioning system by ejector using solar energy, tested several fluids friendly with the environment. The calculation was done by empirical correlations, the thermodynamic characteristics of refrigerants are obtained from the RERPROP database. According to this study, the R717 fluid gives the best COP values for a boiler temperature of 90 °C, a condenser temperature of 35 °C and an evaporation temperature of 15 °C. The COP is about 0.408. Using the meteorological conditions of Tunis city, 3 types of solar collectors are tested for 6 months; from April to September, from 8 AM to 5 PM, the coefficient of performance for the complete system cycle have been calculated, and the system of solar collection are varied from 0.21 to 0.28, and the exergetic efficiency between 0.14 and 0.19 for the same working conditions and solar radiation in July (351 - 875 W/m^2). In the same year, Guo and Chen [9] modeled a solar-powered ejector refrigeration system for an office air conditioning application in a building in Shanghai, China. A dynamic model was proposed to investigate the performance of the refrigeration cycle operated with the refrigerant 134a. The local weather database was used to provide the outdoor temperature and solar irradiation between 9 AM and 5 PM, the average COP, the solar fraction are 0.48 and 0.82 respectively at the time when the conditions are: 85 °C for the boiler, 8 °C for the evaporator, the condenser temperature varied with the outdoor temperature, this system can save up to 80 % of electrical energy compared to the conventional mechanical compression system used for the same local, so this system is a good alternative to conserve energy at administrative buildings level . In 2009, Varga *et al* [10] study the minimum conditions to have an acceptable efficiency and calculate the right design parameters to produce 5 kW of

cold for the entire installed system. They used the 1D ejector theory with constant pressure mixing between the primary and the secondary fluids at the ejector mixing chamber, the used refrigerant is the water. The results obtained show that the optimum temperatures in order to have values of the acceptable coefficient of performance, the temperature of the boiler must greater than 90 °C, the evaporator temperature must be greater than 10 °C, and finally the condenser temperature does not exceed 35 °C. Meyer *et al* [11] did a study on the solar ejection refrigeration cycle, but this time the authors focused much more on geometric parameters and their influences on the coefficient of performance. The boiler works over a temperature range between 85 and 140 °C, the temperature of the evaporator between 5 and 10 °C, the temperature of the condenser between 15 and 35 °C, they found that the increase in diameter of the throat of the primary nozzle causes a decrease in the boiler temperature. The choice of collector type according to available solar irradiation. Salimpour *et al* [12] did a study about thermodynamic and exergo-economic analysis of a solar ejector refrigeration system at a load of 5 kW for the 4 proposed scenarios. The low cost one worth (0.1891 $/h) and possesses the highest exergetic efficiency, they found that the evaporator assembly has the highest investment cost of all configurations

The present research relates to a study of a thermal machine using the principle of the ejector. It is also part of a project work tackling improved systems to produce cold from solar energy. It pays a particular interest to the influence of the 3 sources temperatures and the fluid nature used in this system, together with the influence of solar radiation on the coefficient of performance of the overall system.

Materials and methods

In this section, the real gas equation is used to define the thermo-physical characteristics of the fluids chosen, this calculation is carried out in the FORTRAN software using an internal subroutine to calculate the thermo-physics properties of fluids either for the design or the simulation mode, which allows us to avoid the use of external software to save time and facilitate the export of results in order to plot the curves and interpret them.
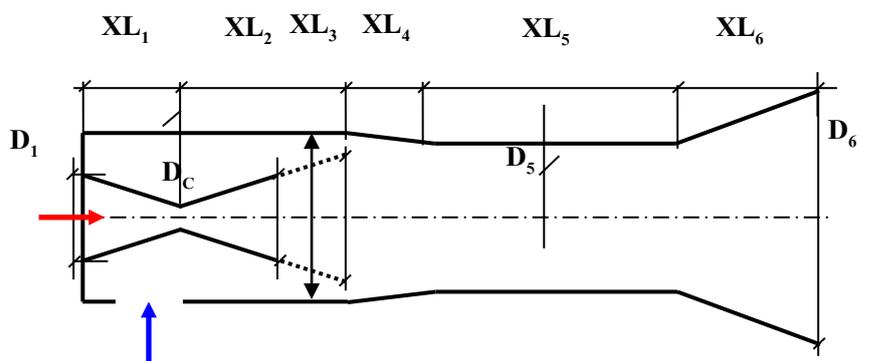

Figure 1 Ejector schematics [13].

This **Figure 1** represents the different parts of the ejector, starting with the primary nozzle, the secondary nozzle, the mixing chamber, the diffuser and also the different symbols of the geometry parameters like the diameter, also the thermo-physical parameters specially the pressure and temperature at several essential points in the ejector, these points are represented in the figure, this is called the constant pressure model, because the mixing of the primary and secondary fluid takes place at constant pressure in the mixing chamber.

Calculation procedure

In this mode, the thermo-physical parameters of the working fluid at the primary nozzle inlet and the secondary nozzle are known, so it only remains to size the ejector, so that the primary fluid sucks the secondary fluid and mixes it up. Homogeneous, and of course, having the highest possible ejector outlet pressure to remain the condenser pressure.

At the inlet of the primary nozzle, we have the stop or initial parameters P_1, T_1 and the mass flow rate m_1, knowing at least 2 parameters, we can calculate the other thermo-physical characteristics of the chosen fluids, also we have the Mach number value at the pass which is equal to unity. To consider friction and energy loss, according to the literature, we take an isentropic efficiency value equal to 0.9 [13]. Using the energy conservation equation:

$$\dot{m}_1(h_1 + \tfrac{V1}{2}) = \dot{m}_1(h_x + \tfrac{Vx}{2}) = \text{constant} \tag{1}$$

The calculation is carried out point by point up to the throat of the primary nozzle.

Mixing chamber

To have a homogenous mixture fluid (primary and secondary) with the same pressure, the calculation of the primary nozzle and secondary converge is needed, by iteration until pressures they are equal each other.

The conservation of momentum equation:

$$\text{Qmm} = \text{Qmp} + \text{Qms} \tag{2}$$

$$\dot{m}_1 V_5 + P_5 A_5 + \dot{m}_2 V_4 + P_4 A_4 \quad \text{with} \quad P_4 = P_5$$

$$Qm = \dot{m}_m V_m + P_m A_m \rightarrow \text{Vm}$$

Energy conservation:

$$E_m = E_p + E_s = \dot{m}_1(h_5 + \tfrac{V^2{}_5}{2}) + \dot{m}_2(h_4 + \tfrac{V^2{}_4}{2}) \quad \rightarrow \quad h_m = \dot{m}_1(h_m + \tfrac{V^2{}_m}{2}) \tag{3}$$

The same calculation procedure as the converging primary nozzle, up to the outlet of the secondary nozzle, respecting the model of the chosen ejector (**Figure 1** of the constant pressure ejector).
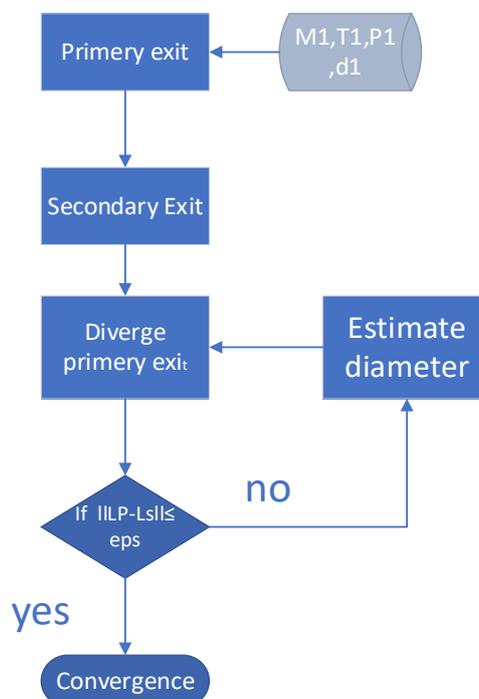

Figure 2 Organigram to calculate geometry of ejector and thermo-physics characteristic of refrigerant.
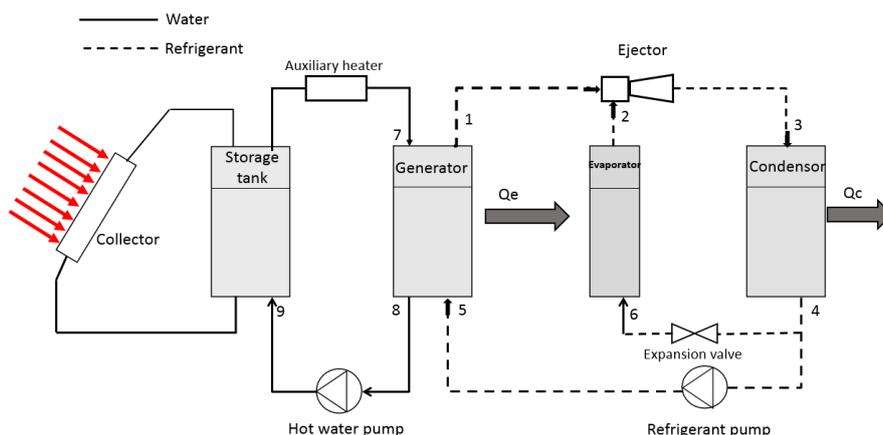
Figure 3 Diagram of solar ejector refrigeration cycle with storage [14].
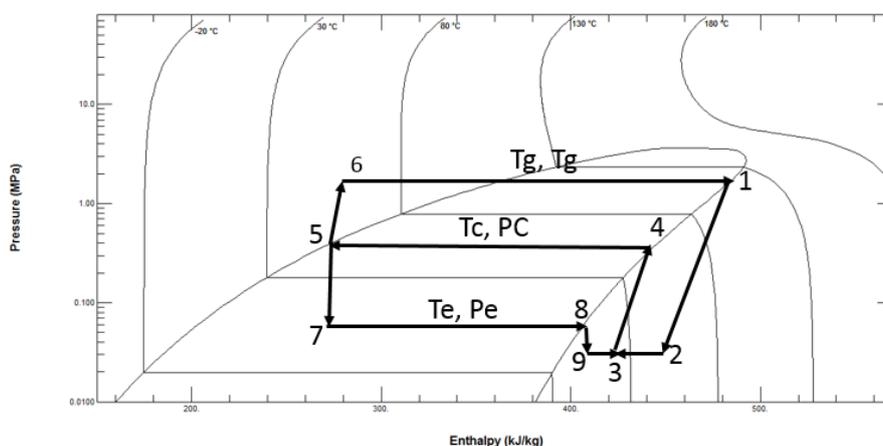

Figure 5 Ejector thermodynamic cycle representation [14].

For an optimal operation, and that is for all refrigerants, the following relation makes it possible to determine the optimal training ratio [15]:

$$U_{opt} = 3.32 \left[\frac{1}{r} \left(1 - \frac{1.21}{\varepsilon} \right) \right]^{2.12} \tag{4}$$

Or r is the compression ratio P_C/P_E of the generator part and ε is the motor ratio P_B/P_C of the ejector. For the pressure calculation, the REFPROP database are used.

Calculation of COP$_S$
The whole coefficient of performance is given by:

$$COPs = COP * \eta_{SC} \tag{5}$$

The ordinary coefficient of performance COP of an ejector refrigeration cycle given by the following equation:

$$COP = \frac{\text{Energy absorbed by the evaporator}}{\text{Energy generated by the boiler+Wpump}} = \frac{mp\Delta he}{ms\Delta hb} = U \frac{\Delta he}{\Delta hb} \tag{6}$$

The electrical energy W consumption by the pump is negligeable,
U is the entrainment ratio which is the ration between the primary and secondary fluids,
Energy absorbed by the evaporator: $mp\Delta he = mp(Hes - Hee)$,

Energy absorbed by the boiler: $ms\Delta he = ms(hbs - hbe)$.

The solar collector yield is given by the following expression:

$$\eta_{SC} = 0.80 - C_i \frac{Tec - Ta}{I_G} \text{ [7]} \tag{7}$$

With $C_i = 2$ W/m^{2}K (type of collector) [16], it's a coefficient given by the manufacturer of solar collector, it's preferable to choose a under vacuum solar thermal collector type, for maximum profit from the solar radiation to fueled the boiler. Tec and Ta are the input temperatures in the collector and the ambient respectively, or the solar collector intake temperature (Tec) has been assumed 10 °C higher than the boiler temperature, Tec = Tg + 10 [7]. This is a result of thermal loses between collector outlet and the boiler inlet, also the thermal energy losses from the storage equipment. Ambient temperature (Ta) and total solar radiation (GI) are estimated for weather conditions in Adrar using solar atlas of Algeria.

1) The average ambient temperature for the day (16-August) is Ta = 45 °C (Meteonorm database)

2) I_G: is the incident total solar radiation received on the surface of the solar collector, where the average solar radiation for a day (16 August in Adrar) $I_G = 825$ W/m^2 [17]. The solar collector efficiency (η_{SC}) and the overall performance coefficient of the overall system (refrigeration machine and the solar collector) was calculated, the values obtained are shown in the graphs below.

$$COP_S = COP * \eta_{SC}$$

Results and discussion

This section summarizes and discusses the main findings of the work, like the variation of the ambient temperatures, the solar radiation in Adrar, and finally the performance coefficient of overall system for different refrigerants in different conditions (condenser temperature) and for different applications (evaporator temperature), and according to the temperature of energy source (boiler), this last of course is related with the solar radiation in Adrar in different season.

Solar energy source

Adrar situated in the south-west of Algeria and his coordinates are: Between latitude (20 °13' - 31 ° 40'N) and longitude (5 ° 46'W - 2 ° 15'E), is endowed with high intensity of global solar radiation greater than 2,200 kwh/m^2/year [18], the sunshine duration is between 8 and 14 h for all season (meteonorm database), using meteonorm software version 7, the below graphs are drawing.
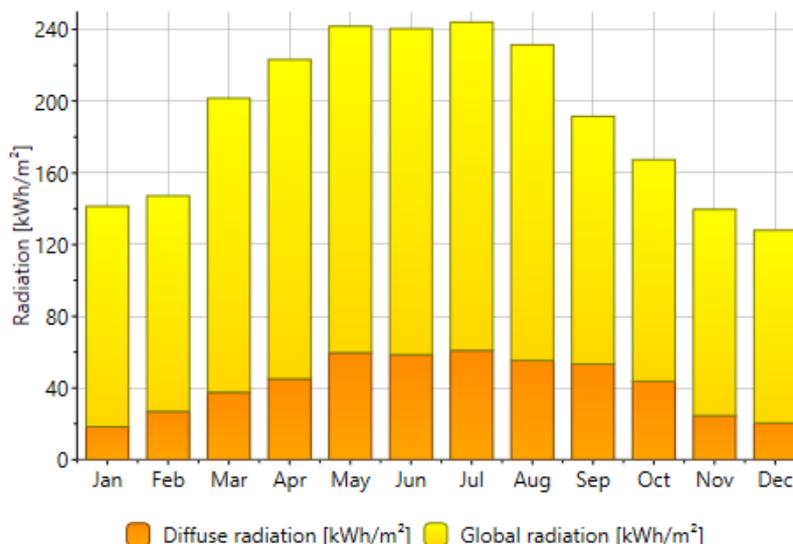

Figure 6 Monthly solar radiation in Adrar.

In the **Figure 6**, the monthly solar radiation in Adrar is very interesting to use solar collector as a worm source for a thermal refrigeration system, especially in summer season for air conditioning system or for refrigeration, the values of monthly global solar radiation are greater than 120 kwh/m^2 from Mars to September.
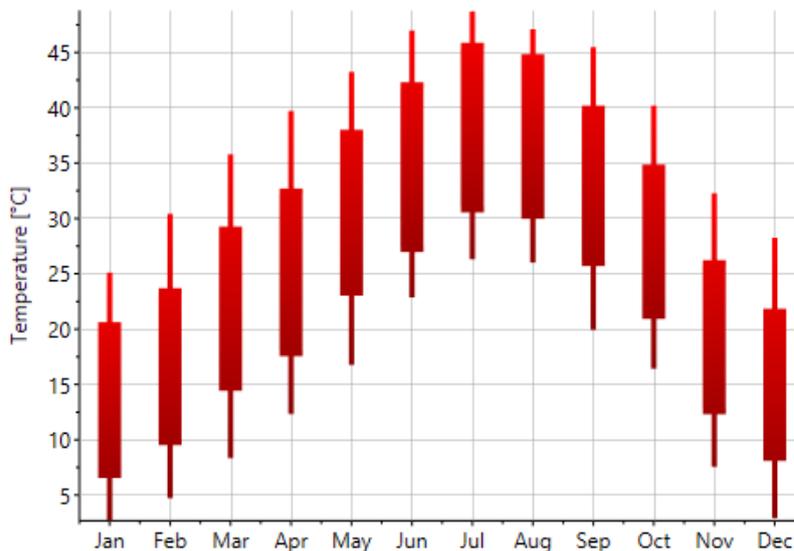

Figure 7 Yearly temperature variation in Adrar.

The temperatures in Adrar are very high specially in summer, in July for example, the temperature is higher than 45 °C, for this reason, an economic and efficiency refrigeration system is needed.
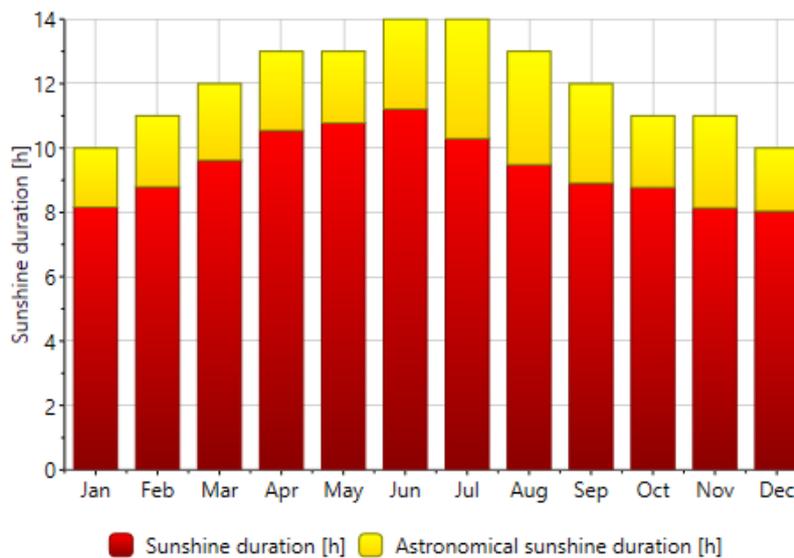

Figure 8 Yearly sunshine duration in Adrar.

The sunshine duration in Adrar is long, in all seasons from January to December, this is very helpful to receive huge quantities of solar radiation and use it to powered a boiler.

Using TRNSYS [19] dynamic simulation software version 17 and meteonorm database for weather, by choosing 2 type of collector, plat one and parabolic, below is the summer useful energy gain variation for both solar panels.
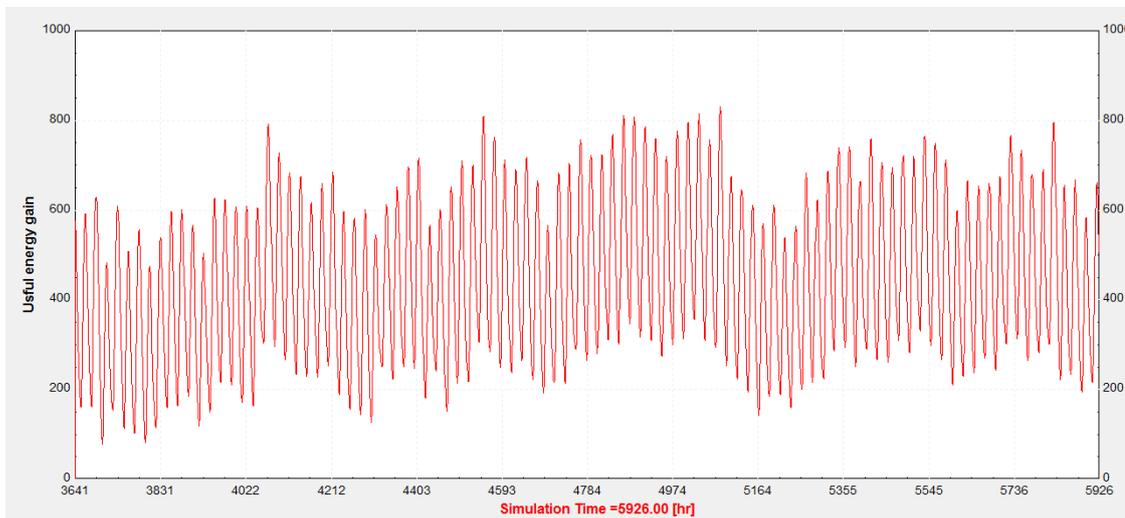

Figure 9 Summer useful energy gain by plat solar panel in Adrar.
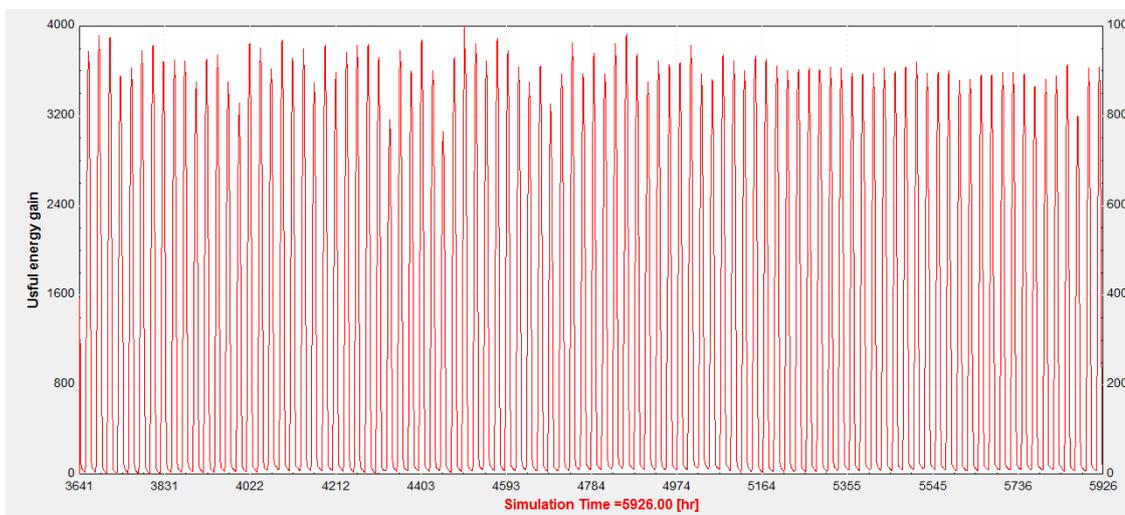

Figure 10 Summer useful energy gain by parabolic solar panel in Adrar.

The comparison between the 2 graphs shown that useful energy gain for the parabolic panel is far greater than the plat one, for this reason is preferable to using the parabolic solar panel if we have the sufficient place for this type of collector.

The used refrigerants

Below is a table for thermo-physics characteristics for the 3 fluids used, from the REFPROP database.

Table 1 Refrigerants characteristics.

	R11	R134a	R717
ODP	1	0.000015	0
GWP	4660	1360	0
Cp (at 25 °C) (KJ/Kg.K)	L 1.07 G 1.67	L 1.11 G 1.71	L 1.4 G 5.31
ρ (Kg/m³)	5	33.33	6.66
Tmax (°C)	198.05	101.06	132.35

Refrigerant R717 (ammonia) is the best fluid from an environmental point of view, which has no influence on the ozone layer and global warming too. Freon R11 has the highest critical temperature, so the possibility of using it with a parabolic solar concentrator. Also, the refrigerant R134a has the highest density, therefore the advantage of being used in micro-exchangers. The heat capacity of the 3 fluids are converging from one to another. R717 is toxic at certain concentrations; therefore, it should be used only in industrial installations where safety precautions are controlled.
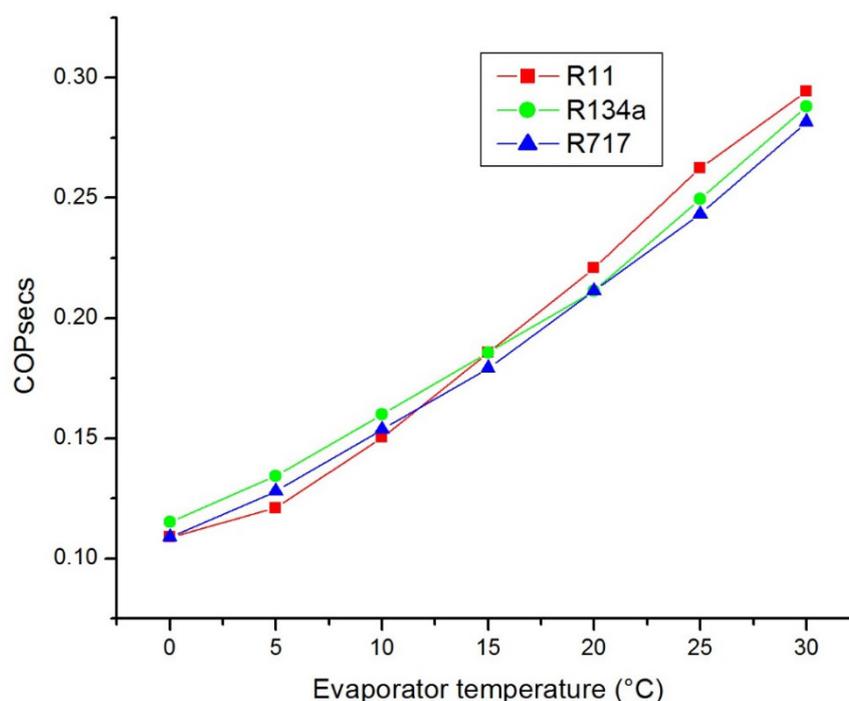
Figure 12 Variation of COP_S as a function of the evaporator temperature for R134a, R11 and R717.

From the diagram in **Figure 12**, which indicates the performance coefficient variation of the overall system (COP_S) as a function of the evaporator temperature T_E, it's noted that the increase of the coefficient of performance COP_S is proportional to the increase of evaporator temperature T_E, and the solar collector yield it remains constant to the variation of evaporator temperature T_E, under 17 °C (refrigeration mode), the fluid R134a gives the best values of performance, but above this temperature (Air conditioning mode), the CFC R11 is the best among the 3 fluids tested.
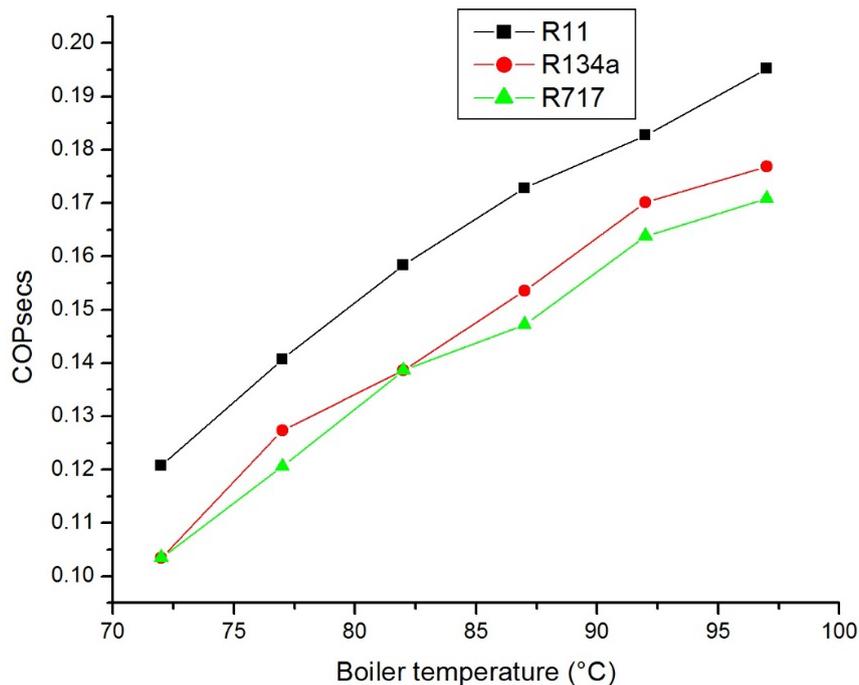
Figure 13 COP_S Variation according to the boiler temperature for R134a, R11 and R717.

From the diagram in **Figure 13** which indicates the COP_S variation according to the boiler temperature T_B, it's noted that the COP_S is increased with the increase of the boiler T_B for the 3 refrigerants.
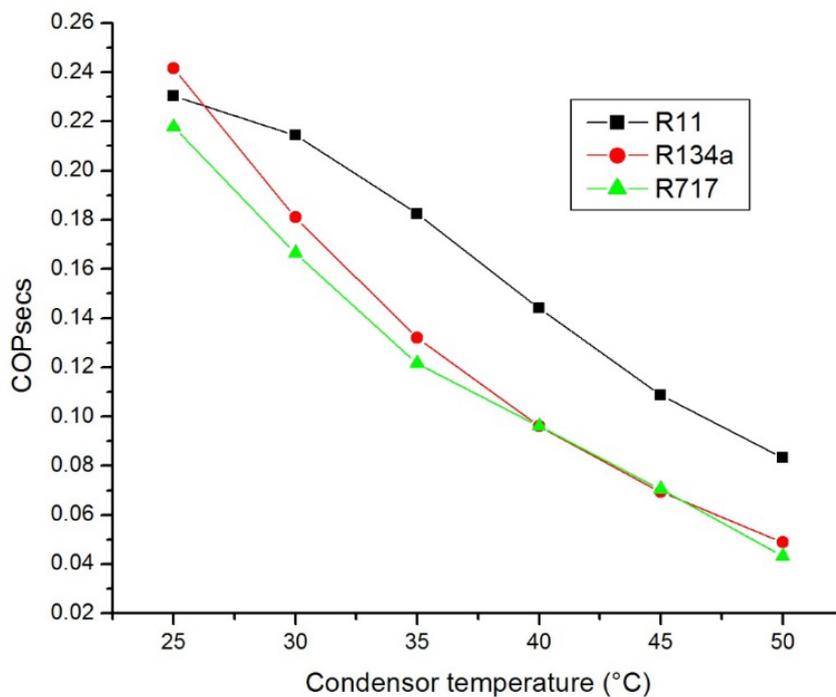

Figure 14 COP_S variation as a function of the condenser temperature for R11, 134a and R717.

Illustrating, shows an increase of Tc that causes a decrease in the value of COPs. In this study a solar collector vacuum tube is chosen, despite the collector cost is very high, but they give a better value of solar fraction.

The results revealed that in optimal operating conditions, the performance coefficient of the overall system COP_S automatically varies as the COP according to the evaporator temperature, the condenser temperature and the boiler temperature.

Conclusions

The most relevant conclusions can be summarized as follows: The overall performance coefficient (COP_S) of the solar ejector refrigeration for a three chosen refrigerant (R11, R134a and R717) as working fluids are investigated. The boiler temperature (T_B) varying from 70 to 90 °C. For R134a, the condenser temperature (T_C) wide-ranged from 15 to 45 °C which represent the outdoor ambient temperature and evaporator temperature (T_E) wide-ranged from 7 to 25 °C to ensure refrigeration or air conditioning of a given interior. In each case, we set the temperatures of the two sources constants and modify the temperature of the third source. From this work, it is found that the performance coefficient of the overall system (ejector cycle with solar subsystem) is lower than the refrigeration system coefficient of performance under the same conditions compared than simple ejector refrigeration system (without a solar collector), because the COP_S is attached to η_{SC}, for this reason it is better to use high performance solar collector (for example, vacuum tube sensor). The CFC R11 gives the best value of performance but is harmful for environment, causes global warming and depletion of the ozone layer, it's used just for comparison because it is an old efficiency refrigerant. The HFC R134a also gives a better performance coefficient than the natural fluid R717, and it's a real alternative as a transition fluid for a multiple application. For this reason, it's recommended to use the HFC refrigerant R134a for a solar ejector refrigeration system, especially for a domestic application. The natural refrigerant R717 (ammonia) is preferable to using it for industrial applications because it's toxic at certain concentrate and of its high critic temperature. Finally, the use of solar ejector refrigeration systems in hot and arid climate like Adrar in Algeria is investigated, and the values of the coefficient of performance are acceptable wherever the summer is very hot where the outdoor temperatures are very high specially in July and August, wherever the temperatures are superior to 44 °C. The value of the COPs is better than other studies in other cities like the study of Bellos *et al* in Athens [10], because of the high values of solar radiation in Adrar (among the highest solar radiation in the world).

The use of solar energy in cooling systems is well suited to hot regions, whose needs coincide with the maximum hours of sunshine. In most cases, there is a lag that requires storage to meet the needs of the non-sunning period: Storage tank associated with a solar collector for refrigeration needs during the nighttime or cloudy days specially for refrigeration mode in winter (Adrar is rarely cloudy in summer). The performance of the solar ejector refrigeration system in Adrar is investigated and it's the first investigation of this very hot region (the temperature reaches 50 °C in summer), the values of COPs are very interesting for air condition and refrigeration to implicate and use of this system in Adrar and it can be generalized this study in other very hot regions in Algeria or around the world.

References

[1] R Tang, S Wang, K Shan and H Cheung. Optimal control strategy of central air-conditioning systems of buildings at morning start period for enhanced energy efficiency and peak demand limiting. *Energy* 2018; **151**, 771-81.

[2] D Coulomb. Refrigeration Main Challenges. International Institute of Refrigeration, Available at https://energy-learning.com/index.php/opinion/78-what-s-new/108-refrigeration-main-challenges, accessed June 2020.

[3] B Belabes, A Youcefi, O Guerri, M Djamai and A Kaabeche. Evaluation of wind energy potential and estimation of cost using wind energy turbines for electricity generation in north of Algeria. *Renew. Sustain. Energ. Rev.* 2015; **51**, 1245-55.

[4] TG Hovgaard, LFS Larsen, MJ Skovrup and JB Jørgensen. Power consumption in refrigeration systems - modeling for optimization. *In*: Proceedings of the 2011 International Symposium on Advanced Control of Industrial Processes, Hangzhou, China. 2011, p. 234-9.

[5] N Al-khalidy. Experimental investigation of solar concentrating collectors in a refrigerant ejector refrigeration machine. *Int. J. Energ. Res.* 1997; **21**, 1123-31.

[6] BJ Huang, JM Chang, VA Petrenko and KB Zhuk. A solar ejector cooling system using refrigerant R141b. *Sol. Energ.* 1998; **64**, 223-6.

[7]   GK Alexis and EK Karayiannis. A solar ejector cooling system using refrigerant R134a in the Athens area. *Renew. Energ.* 2005; **30**, 1457-69.

[8]   E Nehdi, L Kairouani and M Elakhdar. A solar ejector air-conditioning system using environment-friendly working fluids. *Int. J. Energ. Res.* 2008; **32**, 1194-201.

[9]   J Guo and HG Shen. Modeling solar-driven ejector refrigeration system offering air conditioning for office buildings. *Energ. Build.* 2009; **41**, 175-81.

[10] S Varga, AC Oliveira and B Diaconu. Analysis of a solar-assisted ejector cooling system for air conditionin. *Int. J. Low-Carbon Tech.* 2009; **4**, 2-8.

[11] AJ Meyer, TM Harms and RT Dobson. Steam jet ejector cooling powered by waste or solar heat. *Renew. Energ.* 2009; **34**, 297-306.

[12] MR Salimpour, A Ahmadzadeh and AT Al-Sammarraie. Comparative investigation on the exergoeconomic analysis of solar-driven ejector refrigeration systems. *Int. J. Refrigerat.* 2019; **99**, 80-93.

[13] M Ouzzane and Z Aidoun. Model development and numerical procedure for detailed ejector analysis and design. *Appl. Therm. Eng.* 2003; **23**, 2337-51.

[14] VO Petrenko and OS Volovyk. Theoretical study and design of a low-grade heat-driven pilot ejector refrigeration machine operating with butane and isobutane and intended for cooling of gas transported in a gas-main pipeline. *Int. J. Refrigerat.* 2011; **34**, 1699-706.

[15] R Dorantes, 1992. Performances theoriques et experimentales d'une machine frigorifique tritherme a ejecto-compression: Influence de la nature du fluide de travail: Analyses energetique et exergetique. Ph. D. Dissertation. The Institut National des Sciences Appliquées de Lyon, Villeurbanne, France.

[16] BJ Huang, VA Petrenko, IYA Samofatov and NA Shchetinina. Collector selection for solar ejector cooling system. *Solar Energ.* 2001, **71**, 269-74.

[17] M Capderou. *Atlas Solaire de l'Algérie, modèles théoriques et expérimentaux.* Office des Publications Universitaires, Ben Aknoun, Algeria. 1987.

[18] NIBK Bouchouicha and M Bellaoui. Cartography of Global solar radiation over Adrar. *In*: Proceedings of the International Conference of the Green Energy Environ. 2010, p. 1-6.

[19] A Fiksel, JW Thornton, SA Klein and WA Beckman. Developments to the TRNSYS simulation program. *J. Sol. Energ. Eng.* 1995; **117**, 123-7.