Performance evaluation of a solar ejector-vapour compression cycle for cooling application

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Abstract. This study deals with the performance of the ejector-vapour compression cycle assisted by solar. The effect of operating conditions on the combined cycle performance is examined. Also, a comparison of the system performance with environment friendly refrigerants (R134a, R600, R123, R141b, R142b, R152a, R290, and R245fa) is made. This performance is calculated using an empirical correlation. Thermodynamic properties of functioning fluids are obtained with package REFPROP 8. Using the typical meteorological year file containing the weather data of the city of Tunis, the system performance is computed for three collector types. The theoretical results show that the R290 offers the highest coefficient of performance, COP=3.75, for generator temperature T_B = 78°C, condenser temperature T_c = 30°C and the intercooler temperature T_e = 15°C.

Keywords: Solar energy, Ejector, Compression system, Performance, Combined refrigeration cycle

1. Introduction

An ejector system is relatively simple to construct, to operate and to control. This makes it appears like an attractive solution in the countries having good sunshine. Several research groups have investigated the performance of a simple stage ejector cooling cycle.

Zeren et al. [1] considered the commercial potential of the solar-driven ejector cycle with R12 as the refrigerant. They concluded that the efficiency of such a system depends mainly on the availability of a free solar heat source. Huang et al. [2] have examined the solar ejector cooling system using R141b as the working fluid and have determined the COP~ 0.22 at practical operating conditions for air conditioning. Nahdi et al. [3] studied a solar ejector system for the air-conditioning application, with several environment-friendly working fluids. The result shows that the coefficient of performance of 0.28 can be achieved on an average when the evaporating temperature is at 10 °C.

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The COP of the simple stage ejector cooling cycle is still relatively low as a compared to that of vapour compression and absorption systems. The aim of this work is to focus the study on a combined ejector-vapour compression cooling cycle assisted by solar energy that appears as a remarkable alternative solution to improve the COP of the whole system and very little work has been done in this area [4-6]. Sun [7] conducted the mathematical simulation of an environmental friendly solar system. Steam and R134a were used as the refrigerant in an ejector and a compression sub-cycles, respectively. The simulated results showed that the COP of the system could be improved up to 50% compared to the conventional vapour compression system. The system consisted of a solar collector, hot water storage tank, a conventional compression and an ejector sub-cycles with heat exchanger as an interface between them. The COP drops dramatically and even to zero when it is under atrocious weather. In order to prevent the failure of solar system, an auxiliary burner, driven by gas or oil, may be added to ensure that the pressure and temperature conditions required by the ejector are guaranteed. The water carries heat from the collector and releases it in the storage tank. Then, the hot water is pumped from the storage tank to the generator which vaporizes the refrigerant.

In case the heat delivered from the storage tank is not enough to drive the cycle, additional heat is produced in the auxiliary burner and delivered to the R290. At the same time, the vapour leaving the intercooler enters into the ejector and generates a mixed stream that is discharged into the condenser.
This saturated liquid is divided into two streams; one goes into the pump where is pumped back to the generator and the other goes to the expansion valve. It is then expanded to the intercooler where it is evaporated by the heat rejected from the vapour mechanical compression cycle.
At the mechanical subsystem, the compressed R290 vapour coming from the compressor is condensed in the intercooler, then exposed to pressure reduction in the throttling valve and then enters the evaporator where it is evaporated to produce the necessary cooling effect. The vapour is finally compressed to a higher pressure by the compressor and then enters to the intercooler, thus completing the combined cycle. A solar energy assisted combined ejector-vapour compression system is schematically shown in Fig.1.

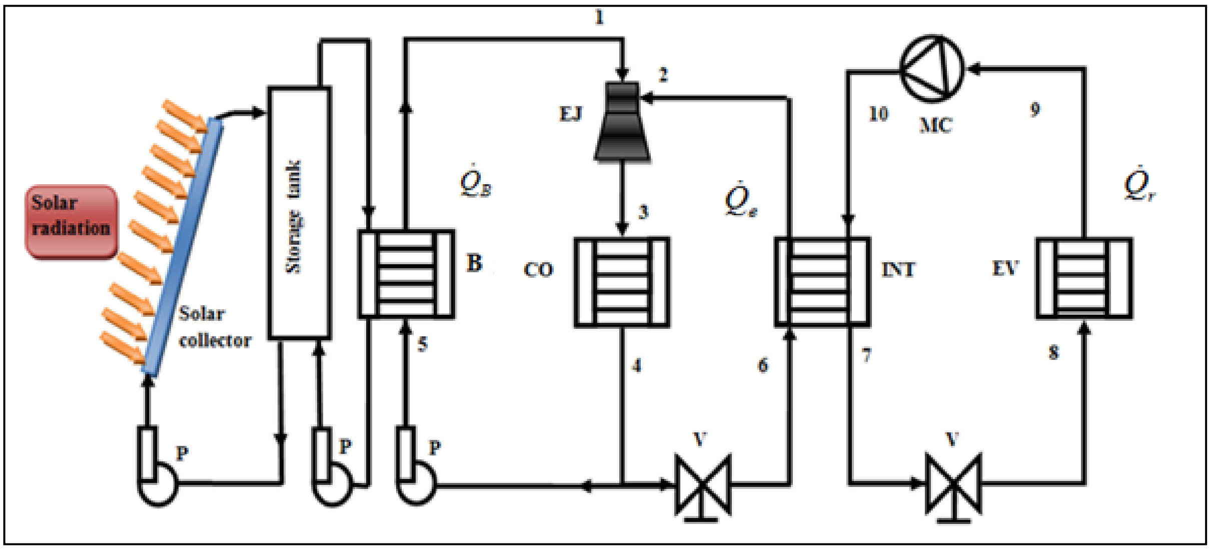

Figure 3. Schematic of a solar ejector-vapour compression system

2. Solar collector performance

The efficiency of a solar system depends on the collector type, solar radiation intensity and the system operating conditions. The solar collector efficiency η_c is defined as the ratio of the gain of the useful heat over any time period to the solar radiation over the same period. It was defined by Duffie and Beckmain [8] as:

$$\eta_c = F_R(\tau\alpha) - F_R U_L \frac{T_i - T_a}{I} \quad (1)$$

Where $F_R(\tau\alpha)$ and $F_R U_L$ are the average heat removal factor and the heat loss coefficient (W $m^{-2}K^{-1}$), respectively. They change depending on the collector type.

T_a : The ambient temperature.

I : The total solar radiation on a flat surface per unit area.

T_i : The collector's inlet temperature.

In simulation models found in the literature T_i is considered to be 10°C higher than the generator temperature [2].

Pridasawas and Lundqvist [9] proposed three solar collectors for the heat supply of the solar ejector system to investigate the effect on the system's performance and comparison.

Type 1 is a single glazed flat plate solar collector:

$$\eta_c = 0.8 - 5\frac{T_i - T_a}{I} \quad (2)$$

Type 2 is a double glazed flat plate solar collector:

$$\eta_c = 0.8 - 3.5\frac{T_i - T_a}{I} \quad (3)$$

Type 3 is an evacuated-tube solar collector:

$$\eta_c = 0.8 - 1.5\frac{T_i - T_a}{I} \quad (4)$$

3. Modeling of the ejector :

The ejector is the heart of the ejector refrigeration sub-cycle. This component replaces the compressor in the conventional compression refrigeration cycle. The ejector consists of four parts: a nozzle section for primary flow, a section chamber for the secondary flow, a mixing section for the primary and secondary flows, a throat section and the diffuser section for the mixed fluid to recompress to the previous pressure Fig.2.

The refrigerant vapour enters into a primary nozzle where the energy of pressure is converted into velocity energy in a suitable experimental condition; the driving pressure ratio corresponds to the supersonic flow of the jet in the divergent part of the primary nozzle. The high speed jet creates the entertainment force for entrainment of secondary fluid then occurs and the velocity energy of the mixture is converted into pressure in the secondary tube. The mixture at higher pressure returns to the condenser.

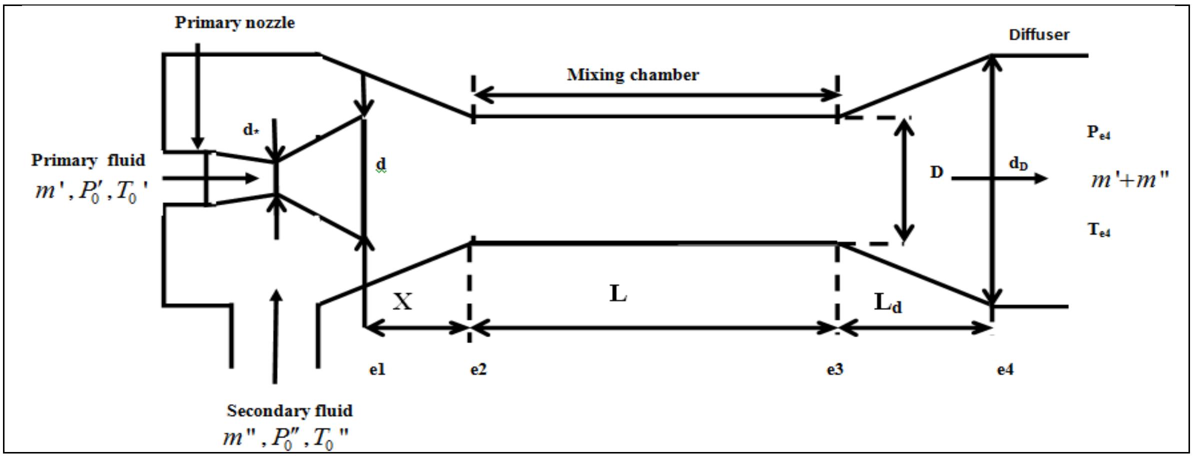

Figure 3. Schematic view of an ejector

Nahdi et al. [10] Holton and Schiltz [11], Work and Haedrich [12] shown that a great number of parameters that determine the performance of the ejector are related to the nature of the fluid, the thermodynamical properties and the ejector geometry.

Lu [13] showed that the basic thermodynamical parameters are the entrainment ratio (U) (the flow rate of the entrained vapour divided by the flow rate of the motive vapour). The expansion ratio (Γ), it is defined as the ratio of the motive vapour pressure to the entrained vapour pressure. The compression ratio (r) gives the pressure ratio of the compressed vapour to the entrained vapour. The driving pressure ratio (ξ) is defined as the ratio of the motive vapour pressure to the back pressure, and the main geometrical parameter ratio Φ (ratio of the areas of the mixing chamber to the primary nozzle throat).

On the basis of the previous experimental investigation, an empirical correlation was derived for the performance prediction of ejectors in optimal regime and independent of the fluid nature [14].

$$U = U_{opt} = 3.32\left[\frac{1}{r} - \frac{1.21}{r\xi}\right]^{2.12} \qquad (5)$$

4. Performance analysis of a combined ejector-vapor compression cycle

In the configuration shown in Fig. 1, the solar heat that drives the refrigeration system is determined by the operation temperature T_i. The combined cycle consists of two sub-cycles. The heat exchanger which is an evaporator of the ejector sub-cycle is used to increase the COP of the overall system. Its working principle is depicted in [4-15].

Let's define the COP_s of the combined cycle, the ejector cycle and the vapour compression cycle as:

$$COP = \frac{\dot{Q}_r}{\dot{Q}_B + \dot{W}_{comp}} \qquad (6)$$

$$COP_{ej} = \frac{\dot{Q}_e}{\dot{Q}_B} \qquad (7)$$

$$COP_m = \frac{\dot{Q}_r}{\dot{W}_{comp}} \qquad (8)$$

The overall thermal efficiency of the ejector sub-cycle is defined as the ratio of the refrigeration $\dot{Q}_e$ to the solar radiation on the collectors [3].

$$COP_s = \frac{\dot{Q}_e}{\dot{Q}_s} = \frac{\dot{Q}_e . \dot{Q}_B}{\dot{Q}_B . \dot{Q}_s} = \eta_c COP_{ej} \qquad (9)$$

However, it should be noticed that the efficiency of the solar collector efficiency η_c defined as $\dot{Q}_B / \dot{Q}_s$ will depend on the heat transfer process which takes place with phase-change in the generator heat exchanger and therefore will also depend on the outlet refrigerant temperature, T_B.

5. Results and discussion

In the present work, the computer program is developed and simulated with FORTRAN. The refrigerant properties are evaluated by using REFPROP V8.0 [16]. This study used 8 $\leq T_e \leq$26 °C and 30$\leq T_c \leq$42°C. The isentropic efficiencies of the pump and the compressor are assumed to be 0.85 and 0.6 respectively.

5.1. *Effect of fluid nature on the COP*

Some properties of the selected refrigerants are given in table 1. Comparative calculations have been made for R134a, R600, R123, R141b, R142b, R152a, R290, and R245fa.

Table 1: some properties of the selected working refrigerants

Name	NBP	ODP relative to R11	GWP relative to CO_2
R134a	-26.1	0	1600
R600	-0.6	0	<10
R123	27.81	1.0	77
R141b	32.05	0.1	630
R142b	-9	0.043	2300
R152a	-24.3	0	140
R290	-42.2	0	3
R245fa	14.9	0	820

Fig.3 represents the COP of the solar refrigeration system for various refrigerants. It can be seen that the performance varies from refrigerant to refrigerant. The refrigerant R290 gives the highest COP (COP=3.75) for the operating temperatures (T_B = 78°C, T_e = 15°C, T_c=30°C).

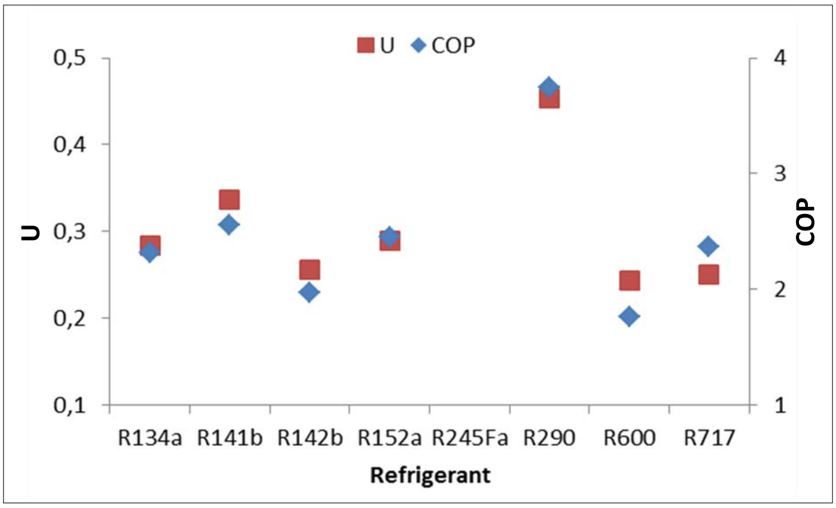

Figure 3. The entrainment ratio U and the COP of combined ejector cycle for various refrigerants

5.2. *Effect of T_e*

The variations of the performance of ejector cycle COP_{ej} and the combined cycle performance COP with T_e are represented in Fig.5.

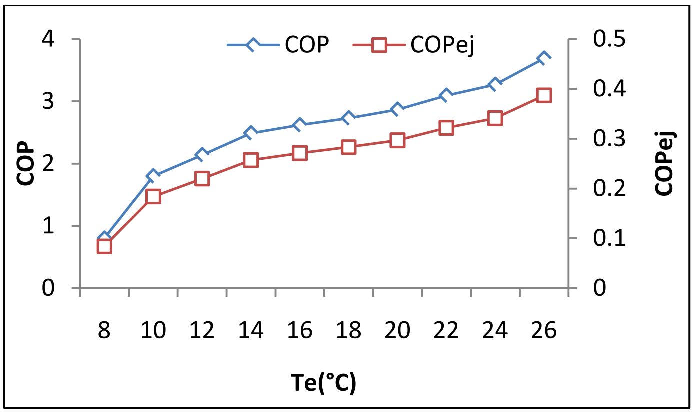

Figure 5. Coefficient of performance of the combined system against T_e

The ejector entrainment ratio, COP_{ej} and COP increase with T_e. Increasing T_e leads to an increase in the secondary flow on the one hand and to a reduction in the latent heat of evaporation on the otherhand. The increasing ratio in the secondary flow is higher and therefore, the COP and the COP_{ej} increases with an increasing T_e for fixed boiler and condenser temperature.

5.3. *Effect of T_c*

The condensing temperature, T_c, mainly depends on the environmental temperature. The following results were obtained by varying the condensing temperature from 30 °C to 42 °C with T_e=10 °C and

T_B=78 °C. The effect of the condenser temperature on the COP is examined. As indicated in Fig.6, the increase in the condenser temperature causes the COP to decrease. Since the same amount of vapor cannot be compressed to a higher condenser pressure, COP will decrease.

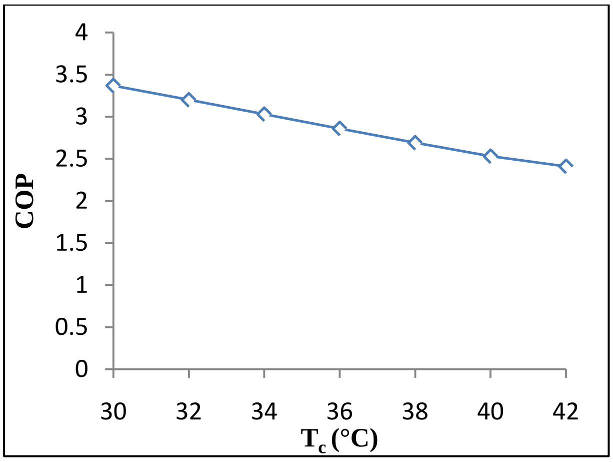

Figure 6. Effect of T_c on the COP

5.4. *Effect of I:*

The meteorological data were supplied by the Tunisian Institute of Meteorology for the city of Tunis [17]. These data considered of total solar radiations on horizontal surface for a representative day. The incidence angle of the collector is taken as 30°. For all months, the hourly solar radiation intensity incident upon the collector surface was calculated from the meteorological data [17] and presented in Fig. 7. Maximum solar radiation takes place as I=1030 W m^{-2} at 12.00 in August.

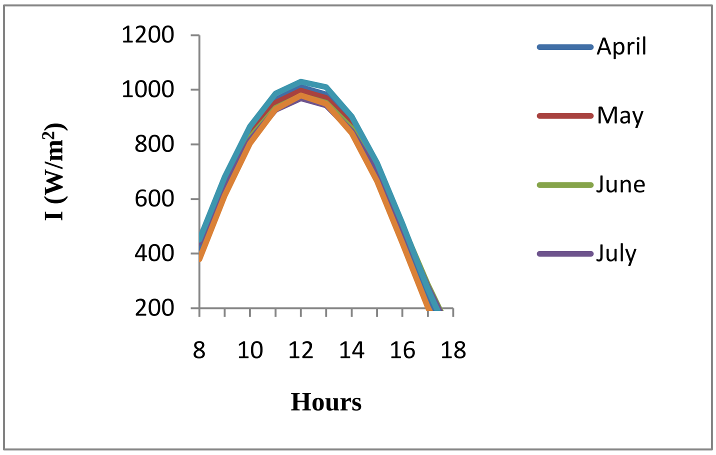

Figure 7: (color online) Comparison of the hourly solar radiation

The monthly average of the efficiency of the solar collector is given in Fig.8. The highest monthly efficiency of the flat plate collector η_c is determined as 0.7 in August.

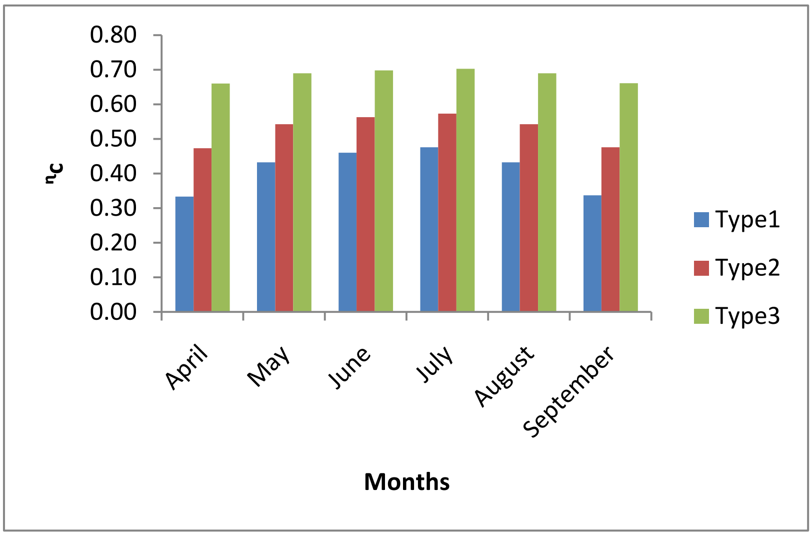

Figure 8: Monthly average efficiency of three flat plate collectors

The hourly variations of the collector efficiency are shown in Fig. 9. As seen, the collector efficiency depends on the collector type. It increase and reaches maximum value at 12:00 with the rise of the solar radiation. The evacuated-tube solar collector provides higher efficiency.

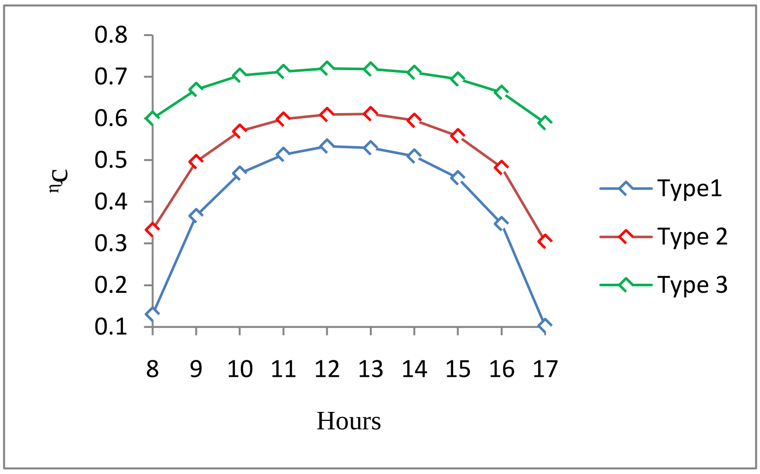

Figure 9: Hourly variations of efficiency of three flat plate collectors

Figures 10 and 11 show the hourly and monthly performances of the solar combined ejector system when the system operates with different types of solar collectors. The evacuated-tube solar collector gives higher COP_s values than a single glazed flat plate solar collector and a double glazed flat plate solar collector for the operating temperatures T_B=78 °C, T_e=10 °C and T_c=35 °C. The highest COP_s is determined to be 1.9 for collector type1, 2.3 for collector type 2 and 2.8 for collector type 3.

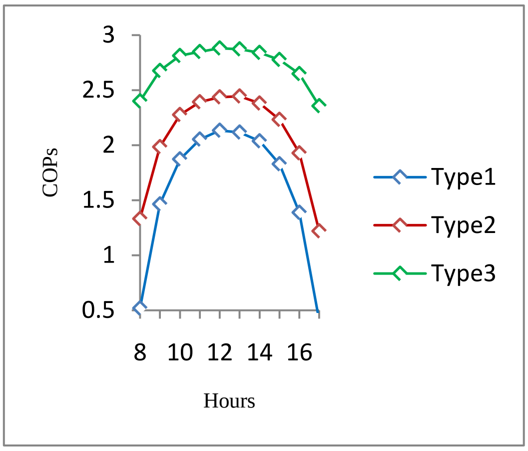

Figure 10. Hourly variation of the COP_s for three flat plate collectors

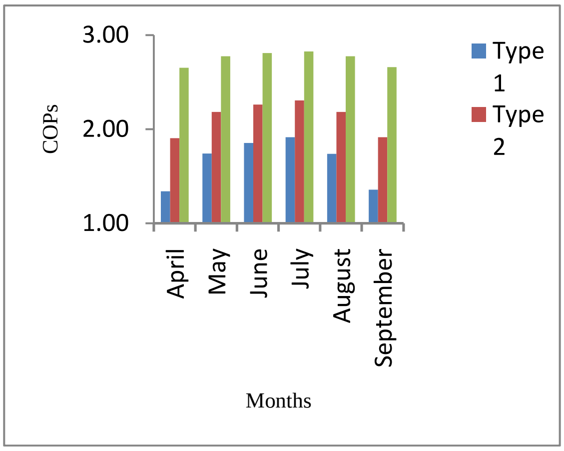

Figure 11. Monthly average performance of the solar combined ejector system for the three flat plate collector types

6. Conclusion:

The main objective of this work was to analyse a combined ejector-vapour compression system assisted by solar energy. The system analysis was based on the empirical correlation and friendly working fluid.

The authors believe that the analysed system is well posed to offer a good answer to the increased demand for refrigeration and air-conditioning. The COP of the combined ejector cycle is mainly influenced by the type and the efficiency of the solar collector. For the considered refrigerants, it has been observed for a given operating conditions that the best performances are obtained with R290. Even though, is necessary to investigate the performance of the system under different conditions from the ones considered in the present study like the use different design parameters.

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