NOVEL CASCADE EJECTOR CYCLE USING NATURAL REFRIGERANTS

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ABSTRACT

This study focuses on the thermodynamic analysis of a cascade ejector refrigeration cycle. Two environmentally-friendly fluids are use: water for the upper cycle and CO_2 for the lower one. The conjugation of the two ejector cycles is proposed in order to operate the CO_2 sub-cycle with subcritical pressures, thus increasing the coefficient of performance (COP). Due to the characteristics of these natural fluids, the traditional one-dimensional analysis cannot be applied to ejector performance prediction. Thus, the ejectors are analyzed using an improved methodology based on real gas equations and Wood's approximation for two-phase speed of sound calculations. Using this methodology simulations are carried out in order to analyze the effect of the operation temperature of the intercoolers, regarding to the operation of a solar-assisted ice maker. The results show that under the base case conditions the COP of the system is 0.2, and this value is highly dependent on the operational temperature of the intercooler.

1. INTRODUCTION

Refrigeration and air conditioning systems use large amounts of electricity, especially in tropical regions like Brazil. According to PROCEL (2008), these systems represent 40% of the Brazilian domestic electricity demand and 20% of the overall national energy demand. Currently, the refrigeration market is dominated by mechanical vapor compression systems, due to their compactness and efficiency. The growing awareness of global warming has encouraged the scientific community to carry out research on thermal compression systems and hence, in recent decades, a rapid growth of the use of these systems has been observed (Hwnag et al., 2008). Of the thermal compression technologies available, the ejector refrigeration systems have generally been used in niche applications, because of the low COP when compared to vapor compression systems. Nevertheless, given the possibility of using solar or waste energy to supply the motive heat, the challenge in developing applications for thermal compression technologies is to achieve systems which are economically competitive with traditional vapor compression cycles.

In order to improve the ejector cycle performance, Sokolov & Hershgal (1990) proposed a solar-assisted ejector refrigeration cycle using a booster. The advantage of this cycle is a substantial increase in the cycle COP, compared with a single-stage ejector cycle operating at the same sink temperatures. In relation to this model, and considering the potential impact of industrial refrigerants on the earth's atmosphere and on global climate change, the authors reported a cost assessment of the system using natural refrigerants such as water and CO_2 (Colle et al., 2009). This configuration allows the operation of the CO_2 cycle at sub-critical pressures, thereby enhancing the overall performance of the cycle.

In this study the performance of a novel cascade ejector cycle was investigated using the same natural refrigerants as in the aforementioned work. By combining these single-stage ejector cycles it is possible to obtain the environmental benefits provided by the use of natural refrigerants whilst achieving temperatures below 0°C.

1.1. Cycle description

Figure 1 shows the arrangement of the proposed cycle. It consists of two single-stage ejector refrigeration cycles coupled to two heat exchangers, called intercoolers (A and B). The condenser and the evaporator of the steam cycle play the role of the boiler and the condenser of the CO_2 cycle, respectively. The heat source is supplied to the generator of the steam cycle and the refrigeration effect is produced at the evaporator of the CO_2 cycle.

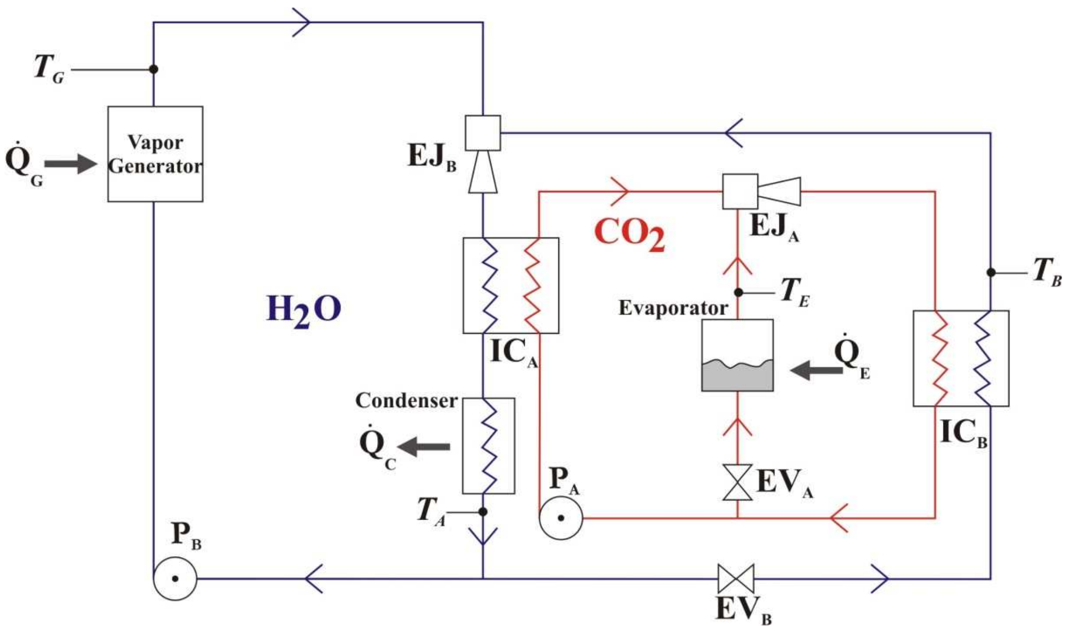


Figure 1: Schematic diagram of the cascade ejector cycle

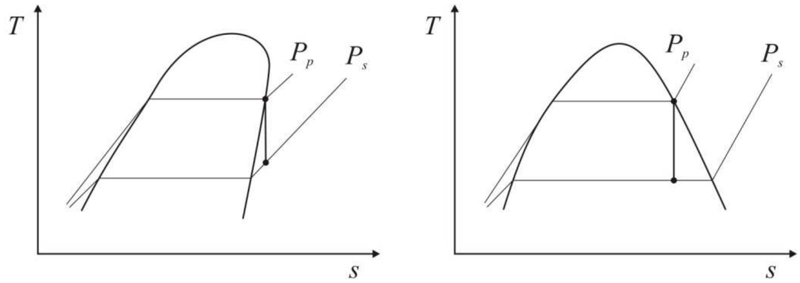
The working principle of the cycle is as follows: solar collectors or a waste thermal energy source supply heat to a vapor generator, which operates as the heat source for the water ejector cooling cycle. Water evaporates in the vapor generator at temperature T_G. The steam flows through the convergent-divergent nozzle of the ejector EJ_B. As it enters the mixing section, a low pressure region develops due to the expansion, which induces the secondary steam flow from the intercooler IC_B, operated at temperature T_B. The primary and secondary steams are mixed in the ejector, and the combined stream flows to the condenser and loses heat at temperature T_A in two heat exchangers, firstly at the intercooler IC_A and the remaining heat is released at the condenser. After the condenser, the flow splits into primary and secondary flows, which are pumped back to the vapor generator (P_B) and the intercooler IC_B, respectively, after passing through an expansion valve (EV_B). The CO_2 ejector cycle works analogous to that described above. The heat source that drives the cycle is supplied by the water ejector at the intercooler IC_A. The vapor generated flows to the ejector, where the secondary flow is induced from the evaporator. The mixed flow releases heat at the intercooler IC_B before it splits again into primary and secondary flows, which flow back to the intercooler IC_A and the evaporator, respectively.

The proposed cycle allows the use of a low-temperature heat to provide a refrigeration effect at a temperature below 0°C. Therefore, it represents a cycle of considerable interest, since it offers a solution not covered by absorption cycles, i.e. low-temperature refrigeration using low-temperature heat.

1.2. Working fluids

Natural refrigerants such as water and CO_2 are considered as workable options due to their stability and availability in the environment (Calm, 2008). Nevertheless, these refrigerants present the disadvantages of low critic temperature and high operational pressure for CO_2 or a lower temperature limit for water.

Although ejector systems have been around for over a century, there is still a clear need to improve the modeling of the cycle due of difficulties associated with the system operation. Ejector modeling is commonly based on ideal gas dynamics models. However, depending on the characteristics of the working fluids these models may be not acceptable. According to Chen et al. (1998) the working fluids for a jet refrigerator can be categorized as wet vapor and dry vapor, as shown in Figure 2. For a wet vapor fluid, the saturated vapor line has a negative slope in the *T*-*s* diagram and for dry vapor fluids there is no phase change during the expansion process when passing through the primary nozzle. Nevertheless, for a wet vapor fluid, small droplets may be formed at the nozzle exit, inducing condensation shocks. This can be eliminated by superheating the fluid before it enters the nozzle. However, the use of superheated motive steam causes a slight decrease in the ejector efficiency (Power, 1993).



a) Dry fluid **b) Wet fluid**

Figure 2 : Temperature-entropy charts for the expansion of refrigerants as they pass through the nozzle

Halocarbon-based refrigerants, such as HCFC141b and HFC134a, are considered as dry vapor fluids, whilst natural refrigerants, such as water and CO_2, are considered as wet vapor fluids.

2. EJECTOR ANALYSIS

The heart of the cooling cycle is the ejector and hence information on its design and performance prediction is critically important. Such information can be obtained using a mathematical model commonly based on the 1-D theory initially proposed by Keenan et al. (1950). The model was based on ideal gas dynamics and the principles of conservation of mass, momentum and energy. This model was later modified by Munday & Bagster (1977) and Huang et al. (1999), who introduced the expansion inefficiencies. However, these theories present some difficulties on dealing with real gases and the two-phase flow expected in ejectors operated with wet fluids, as in the case of water and CO_2. Nevertheless, considering the operational pressures for the two fluids investigated in this study, the ideal gas assumption is not appropriate. Some authors have published attempts to model the ejector performance by assuming real gas state equations and two-phase flow. However, they used empirical correlations to deal with the complexity associated with the two-phase flows (Cizungu et al., 2005; Zhu & Li, 2009). In this study, we propose the use of the methodology developed by Sherif et al. (2000) for ejector design and evaluation. These authors take into account real gas state equations; however, they assume that the isentropic efficiency of the expansion and compression processes that occur in the ejector are previously known. The empirical coefficients for taking into account the losses in the mixing chamber, introduced by Huang et al., (1999) and Eames et al., (1995), are also considered herein. Thus, the isentropic efficiencies are 0.95, 0.95 and 0.9 for the primary nozzle, secondary inlet and diffuser, respectively, whilst the mixing losses coefficient is 0.88.

The fundamental expression for the speed of sound is defined as:

$$c^2 = \left(\frac{\partial P}{\partial \rho}\right)_s \tag{1}$$

where c is the speed of sound, P is the pressure and c is the density. The calculation of the speed of sound in two-phase mixtures is a complex task, since the compressions and rarefactions produced by the sound wave are, in these cases, accompanied by mass transfer from one phase to the other.

Sherif et al. suggested the evaluation of the speed of sound for a two-phase flow through numerical differentiation of Eq. (1). Nevertheless, considering the possibility of metastable conditions occurring in the nozzle exit, the authors propose to analyze the choking phenomenon using Wood's approximation (Wood, 1930) for the speed of sound calculations in two-phase mixtures. This approximation is defined as follows:

$$c_{tp} \approx \left[\rho_m \left(\frac{1-\varepsilon}{\rho_l {c_l}^2} + \frac{\varepsilon}{\rho_v {c_v}^2}\right)\right]^{-1/2} \tag{2}$$

where c_{tp} is the speed of sound passing through the two-phase mixture, ρ_l and ρ_v are the densities of the saturated liquid and vapor, respectively, and ε is the viod fraction.

The analysis is performed with EES (engineering equation solver). This software (Klein & Alvarado, 2011) has the advantage of including fluid properties and ready-to-use optimization tools. It uses the same equation of state as REFPROP-NIST (Lemmon, Mclinden, & Huber, 2002). The predictions obtained were compared with those of REFPROP-NIST and are essentially identical.

Using the methodology of Sherif et al., the main geometry relations for the two ejectors are specified according to the operational conditions of the cascade cycle proposed. It is assumed here that the cycle condenser is water-cooled, where the water comes from the abundant rivers in Brazil. Thus, defining the base case of operating the proposed cycle for an ice making application of 3 TR (10.55 kW), in the Amazon River region, the following sink temperatures should be considered: $T_G = 85°C$, $T_A = 25°C$, $T_B = 7°C$ and $T_E = -5°$. According to Huang et al. (1999), two area ratios should be taken into account for predicting ejector performance: the primary nozzle exit area to primary throat area (A_{ne}/A_t); and the mixing chamber area to primary throat area (A_m/A_t). Hence, for the base case the geometry relations estimated are listed in Table 1.

Table 1 : Main cross-section area ratios of the ejectors for the base case

	CO_2 Ejector	H_2O Ejector
(A_{ne}/A_t)	1.252	3.776
(A_m/A_t)	1.930	35.95

The performance of the ejectors is commonly evaluated in terms of the entrainment ratio (ω). For the base case specified above and the area ratios listed in Table 1, the entrainment ratios calculated were 0.258 for the CO_2 ejector and 0.681 for the steam ejector. Therefore, the process occurring in the ejectors, according to the aforementioned methodology, can be described in the Mollier diagrams shown in Figures 3 and 4. Although, the outlet flow of the steam ejector is superheated, a two-phase flow occurs due to the processes that occur in the ejector. In the case of the CO_2 ejector the outlet mixed stream is a two-phase mixture with a vapor quality is 0.9. Nevertheless, during the expansion in the primary nozzle for both ejectors the vapor quality of the primary fluid is around 0.75.

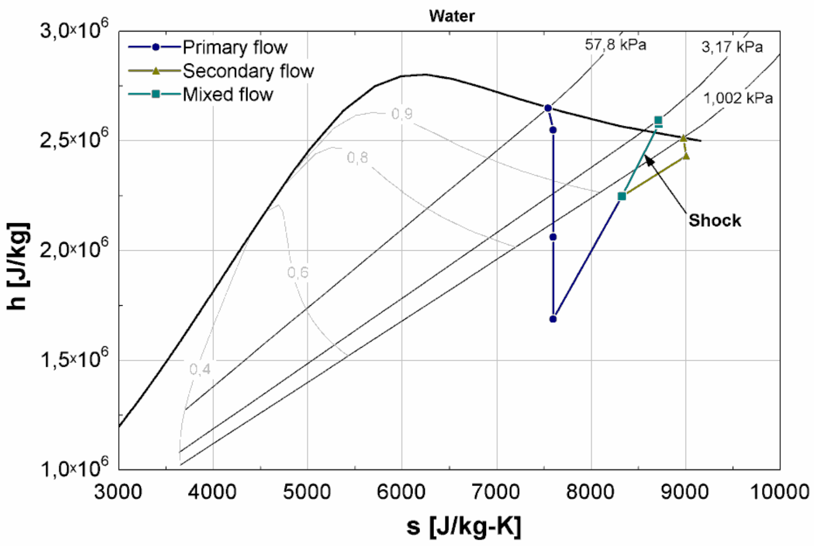


Figure 3: Mollier diagram of the steam ejector

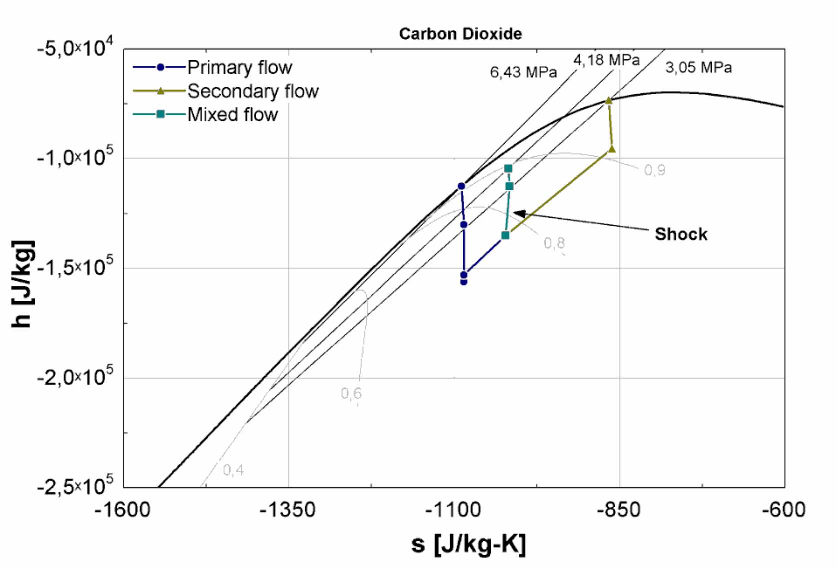


Figure 4 : Mollier diagram of the CO_2 ejector

3. CYCLE ANALYSIS

The COP for a single stage ejector is defined by the ratio between the heat transfer at the heat exchanger which leads to the refrigeration effect (commonly the evaporator) and the heat transfer in the vapor generator. The pumping power is generally neglected because its represent less than 1% of the heat transferred to the boiler. Nevertheless, the present analysis takes the pumping power into account, in order to determine the different characteristics of the two cycles. For instance, the COP of the steam ejector is defined as follows:

$$COP_{H2O}(T_G, T_A, T_B) = \frac{\dot{Q}_{IB}}{\dot{Q}_G + \dot{W}_{H2O}} \tag{3}$$

where $\dot{Q}_{IB}$ is the heat transfer at the intercooler IC_B, $\dot{Q}_G$ is the heat supplied to the vapor generator and $\dot{W}_{H2O}$ is the power required by the water circulation pump.

Analogously, for the CO_2 cycle:

$$COP_{CO2}(T_A, T_B, T_E) = \frac{\dot{Q}_E}{\dot{Q}_{IA} + \dot{W}_{CO2}} \tag{4}$$

where $\dot{Q}_{IA}$ is the heat transferred at the intercooler IC_A, $\dot{Q}_E$ is the refrigeration power of the device and $\dot{W}_{CO2}$ is the power required by the CO_2 circulation pump.

The COP for the combined cycle can be defined as:

$$COP(T_G, T_A, T_B, T_E) = \frac{\dot{Q}_E}{\dot{Q}_G + \dot{W}_{CO2} + \dot{W}_{H2O}} \tag{5}$$

Therefore, the pumping power for the combined cycle is:

$$\dot{W}_P = \dot{W}_{CO2} + \dot{W}_{H2O} \tag{6}$$

4. RESULTS AND DISCUSSION

In a simulation of the cycle, for the base case defined above, a global COP of 0.17 was achieved. In contrast, for the Carnot cycle, the COP for the same sink temperatures is 1.45. Absorption systems based on lithium-bromide cannot achieve temperatures below 0°C and ammonia-based ones require higher-temperature heat sources. Thus, although the cycle COP is approximately 12% of the value for the Carnot performance, it offers an opportunity for refrigeration based on renewable energy, not covered by the range of application of absorption machines, and also has the ecological benefits of using natural refrigerants.

The parametric analysis of the cycle described in this paper shows that the cycle conditions can be optimized, within the range of low-temperature collectors, to achieve the maximum overall COP. All the cases simulated here consider a fixed cooling capacity and an optimized ejector geometry for each condition.

4.1. Effect of intercooler IC_A temperature

Figure 5 shows that there is an optimum temperature for the intercooler IC_A operation since at temperatures higher than 25°C the entrainment ratio of the CO_2 cycle decreases. However, for temperatures lower than 25°C the steam ejector entrainment ratio also decreases. It can be observed that the lower the operation temperature of the evaporator, the higher the optimum temperature of the intercooler IC_A. Nevertheless, there is a thermodynamic restriction on this temperature, because it must not be too close to the critical temperature of CO_2.

Figure 5 also shows the effect of the temperature T_A on the pumping power required by the cycle. It can be observed that the higher the operation temperature of the intercooler IC_A, the lower the pumping power required. This is because approximately 90% of the power used for pumping is due to the CO_2 circulation pump and hence as the entrainment ratio of CO_2 increases when T_A increases, the mass of CO_2 pumped decreases.

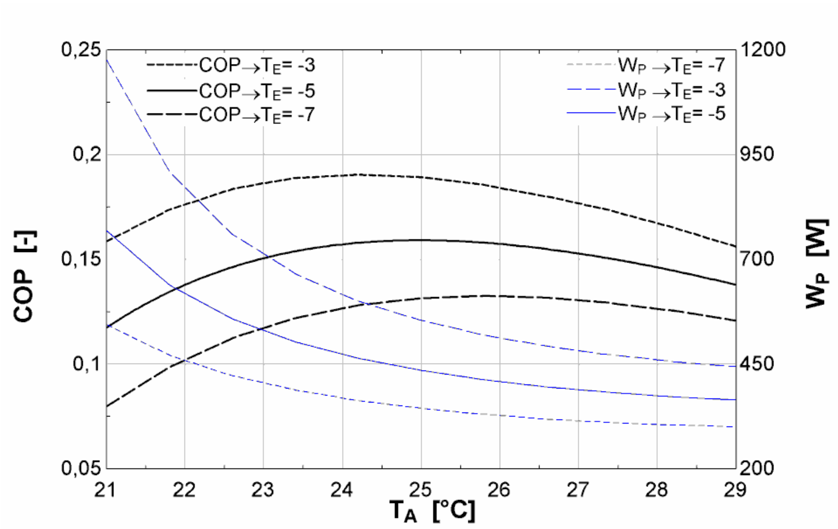


Figure 5 : Effect of intercooler IC_A temperature on COP

4.2. Effect of intercooler IC_B temperature

¡Error! No se encuentra el origen de la referencia. shows the effect of the intercooler IC_B temperature on the overall COP. Similarly, in the aforementioned analysis, the existence of an optimum temperature is observed. However, this optimum appears to be close to 0°C, which is a thermodynamic limit for the steam ejector cycle. The proximity between the optimum and the water freezing point is even greater when the evaporator temperature decreases.

The entrainment ratio of CO_2 decreases as T_B increases, thus the mass flow of the CO_2 pumped to the intercooler IC_A increases with the temperature T_B. Therefore, the pumping power required by the whole cycle also increases.

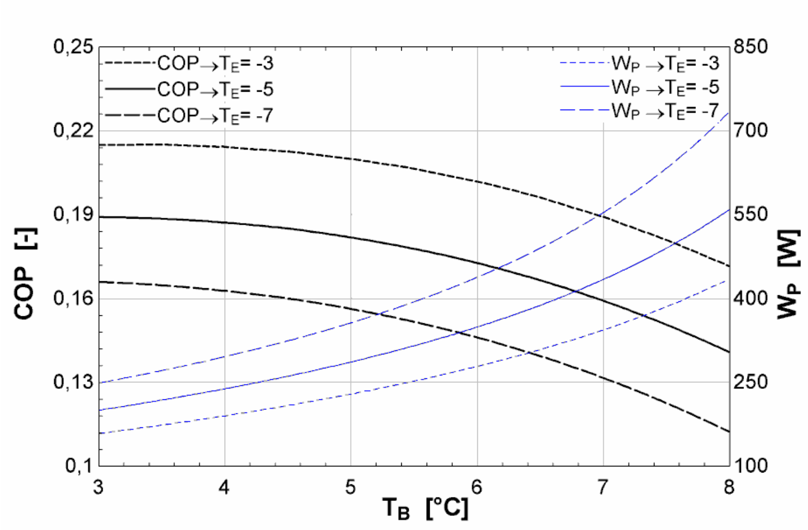


Figure 6 : Effect of intercooler IC_B temperature on COP

4.3. Effect of vapor generator temperature

The effect of the boiler temperature is shown in Figure 7. Since, the temperatures of both intercoolers are fixed, the geometric characteristics of the CO_2 ejector are unmodified during this analysis. Therefore, for each boiler temperature simulated the steam ejector is optimized. Hence, as the boiler temperature increases the COP will also increase, as expected.

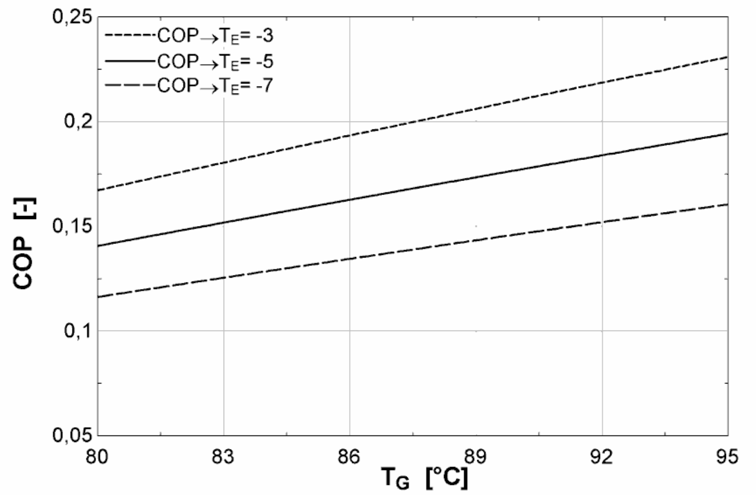


Figure 7 : Effect of boiler temperature on COP

5. CONCLUSIONS

A cascade ejector cycle using natural refrigerants is proposed herein. Besides the environmental benefits, this configuration allows the generation of a refrigeration effect below the freezing point of water by considering low-temperature heat sources. Although the cycle is not completely passive, because it uses electricity for the circulation pumps, the power required may be supplied by solar PV panels (considering that solar collectors also supply the thermal energy).

Regarding the feasibility of the proposed cycle, an economic evaluation must include the trades-off between the cycle efficiency and heat exchanger size, which will certainly affect the overall cost of the system.

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