23[rd] IIR International Congress of Refrigeration, Prague, Czech Republic
Refrigeration for Sustainable Development

ID: 490

# NOVEL CASCADE EJECTOR CYCLE USING NATURAL REFRIGERANTS

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## ABSTRACT

The present work is focused on the thermodynamic analysis of a cascade ejector refrigeration cycle. Two environmentally friendly fluids are use: water for the upper cycle and $CO_2$ in 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 the characteristics of these natural fluids, the traditional one dimensional analysis cannot be applied to ejector's performance prediction. Thus, the ejectors are analyzed using an improved methodology using real gas equations and Wood's approximation for two phase sound speed calculations. Using this methodology simulations are carried out in order to analyze the operation temperatures for the intercoolers, regarding to the operation of a solar-assisted icemaker. The results show that under the base case conditions the COP developed by the systems is 0.2, which is highly dependent on the operational temperature of the intercooler.

## 1. INTRODUCTION

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

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

The present work proposes to explore the performance of a novel cascade ejector cycle, using the same natural refrigerants as in the aforementioned work. Therefore, though conjugating these single stage ejector cycles it is possible to combine the environmental benefits granted by the use of natural refrigerants with the possibility of achieving temperatures below 0°C, using a passive system in the circumstance solar energy is available.

### 1.1. Cycle description

Figure 1 shows the arrangement of the proposed cycle. It consist of two single stage ejector refrigeration cycles coupled by two heat exchangers, called intercoolers, namely A and B. The condenser and the evaporator of the steam cycle play the rule of boiler and 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 $CO_2$ cycle.

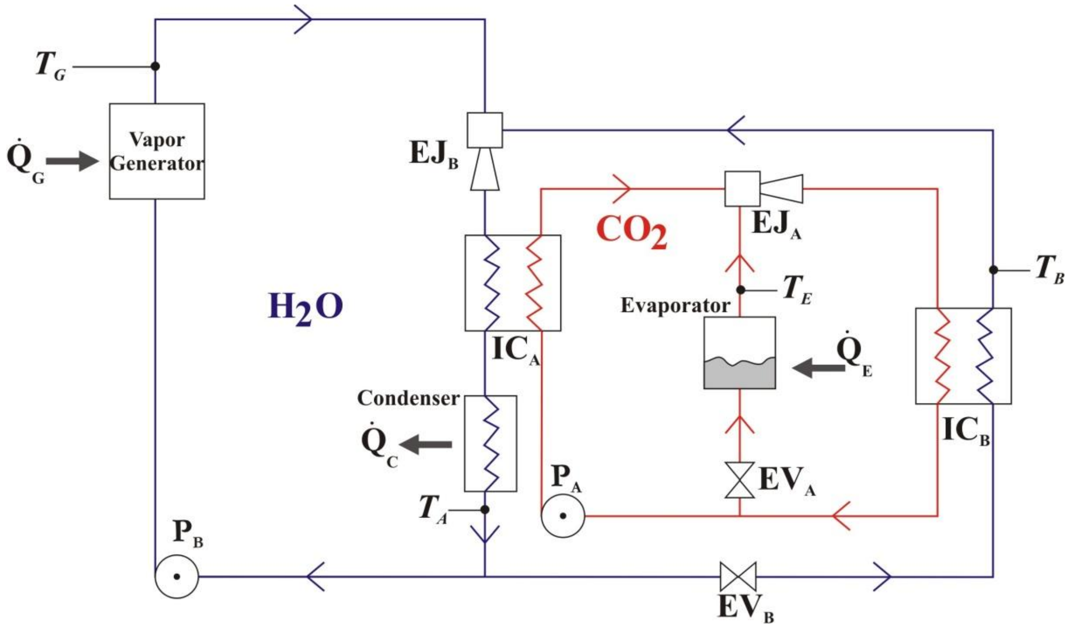



**Figure 1: Schematic of the cascade ejector cycle**

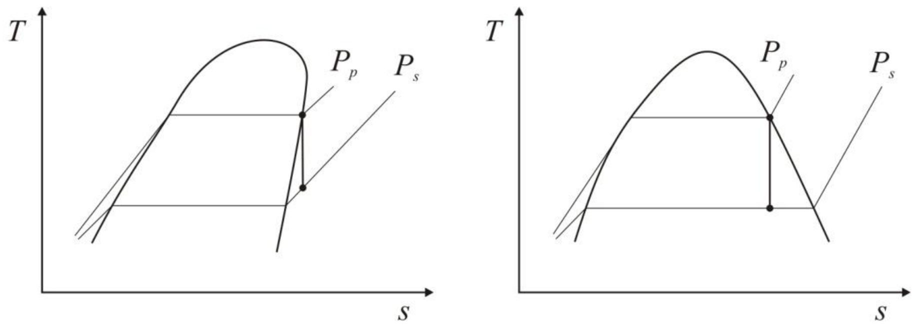
The working principle of the cycle is as follows: solar collector or a waste thermal energy supply heat to a vapor generator, which operates as the heat source for the water ejector cooling cycle. Water evaporates in the vapor generator as temperature $T_G$. The steam flows through the convergent-divergent nozzle of the ejector. As it enters to the mixing section, a low pressure region is caused by the expansion, which induces the secondary steam flow from the intercooler $IC_B$, operated at temperature $T_B$. The primary and secondary steam flows 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, and the intercooler $IC_B$, after passing through an expansion valve ($EV_B$), respectively. 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 release heat at the intercooler $IC_B$ before split again into primary and secondary and flow back to intercooler $IC_A$ and evaporator, respectively.

The proposed cycle allows using a low temperature heat for developing a refrigeration effect at a temperature level below 0°C. Therefore, it represents an interesting cycle, 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 as water and $CO_2$ are considered a workable option due to its stability and availability in the environment (Calm, 2008). Nevertheless, the refrigerants listed before present some disadvantages like the low critic temperature and high operational pressure for $CO_2$ or the lower temperature limit for water.

Although the ejector systems are known from over a century, there is still a clear need to improve modeling the cycle due of difficulties in system's operation. Ejector modeling is commonly based on ideal gas dynamics models. However depending on the characteristics of the working fluids this assumption 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 wet vapor fluid, its saturated vapor line shows a negative slope in the *T-s* diagram. For dry vapor fluids there is no phase change during the expansion process through the primary nozzle. Nevertheless, for wet vapor fluid, small droplets may be formed at nozzle exit, inducing condensation shocks. This can be eliminated by superheating the fluid before entering the nozzle. However, the use of superheated motive steam causes a slight decrease in ejector efficiency (Power, 1993).




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

**Figure 2 : Temperature-Entropy charts of expansion process of refrigerants through the nozzle**

Halocarbons-based refrigerants as HCFC141b and HFC134a are considered dry vapour fluids; instead natural refrigerants as water and $CO_2$ are considered wet vapour fluids.

## 2. EJECTOR ANALYSIS

The heart of the cooling cycle is the ejector; hence information on its design and performance prediction is critically important. Such information can be obtained by 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 managing with real gases and the two phase flow expected in ejectors operated with wet fluids, as the case of water and $CO_2$. Nevertheless, considering the operational pressures for the two fluids studied in this work, the ideal gas assumption is not appropriate. Some authors reported in literature have devoted efforts in modeling ejector's performance, by assuming real gas state equations and two phase flow. However, they used empirical correlations for managing the flow complexity involved (Cizungu et al., 2005; Zhu & Li, 2009). Therefore, here the authors of the present work propose an improved methodology for ejector design and evaluation, based on the model of Sherif et al. (2000). This authors take into account real gas state equations, however assumes to be previously known the isentropic efficiency of the expansion and compression processes that occur in the ejector. 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 here.

Sound velocity calculations in two phase mixtures is a complicate 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. Therefore, the choking phenomenon was analyzed using the Wood's approximation for the two phase sound speed calculations (Wood, 1930), defined as follows:

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

where $c$ is the sound speed for the two phase mixture. $\rho_l$ and $\rho_v$ are the density of the sturated liquid an vapor, respectively, and $\varepsilon$ is the viod fraction. The volumetric density mean ($\rho_m$) and the void fraction are calculated by:

$$\rho_m = \rho_l(1-\varepsilon) + \rho_v\,\varepsilon \tag{2}$$

$$\varepsilon = \frac{x\rho_l}{x\rho_l + (1-x)\rho_v} \tag{3}$$

where $x$ is the quality of the two phase mixture.

The thermodynamic state conditions inside the ejector, as well for the whole cycle, were evaluated using the equations of state available in the Engineering Equation Solver, EES (Klein & Alvarado, 2010).

Using the improved methodology, the main geometry relations for the two ejectors are specified attending the operational conditions for the cascade cycle proposed. It is assumed here that the cycle condenser is water cooled, where water comes from abundant rivers in Brazil. Thus, defining the base case of operating the proposed cycle for an icemaking application of 3 TR (10,55 kW), located in the Amazon River area, the following sink temperatures should be considered: $T_G = 85℃$, $T_A = 25℃$, $T_B = 7℃$ and $T_E = -5°$. According to Huang et al. (1999) two area relations should be taken into account for predicting ejector performance: primary throat area to primary nozzle exit area $(A_t/A_{ne})$, and primary throat area to mixing chamber area $(A_t/A_m)$ . Hence, for the base case the geometry relations estimated are listed in Table 1.

**Table 1 : Ejector's main cross areas 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 entrained ratio $(\omega)$. For the base case specified above and the area ratio of Table 1 the entrained ratio calculated is 0.258 for the $CO_2$ ejector and 0,681 for the steam ejector. Therefore, the process undergoing in the ejectors according to the aforementioned methodology is described in the Mollier's diagrams illustrated in Figures 3 and 4. Although, the outlet flow of the steam ejector is superheated, a two phase flow occurs during the processes that occur in the ejector. For the case of $CO_2$ ejector the outlet mixed stream is a two phase mixture with vapor quality equals to 0.9. Nevertheless, during the expansion in the primary nozzle for both ejectors the vapor quality of the primary fluid is around 0.75.

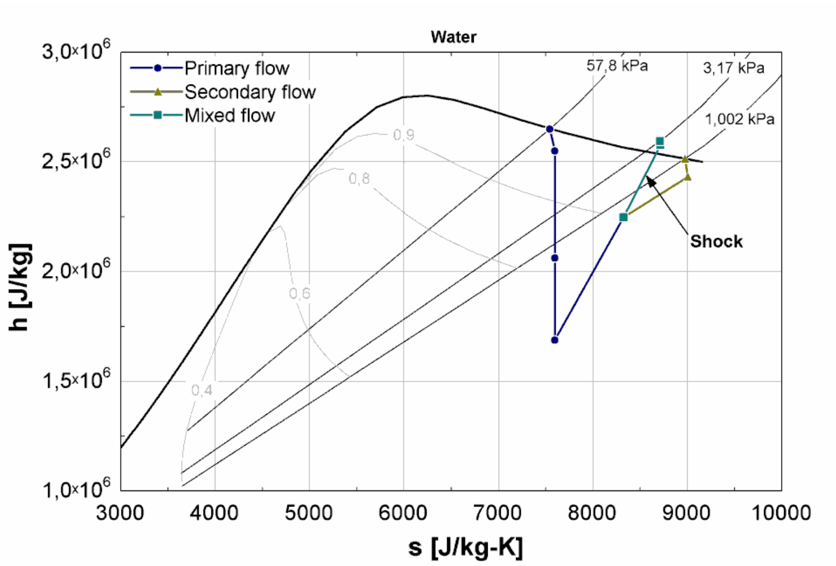



**Figure 3: Mollier's diagram of the steam ejector**

## 3. CYCLE ANALYSIS

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

$$COP_{H2O}(T_G, T_A, T_B) = \frac{\dot{Q}_{IB}}{\dot{Q}_G + \dot{W}_{H2O}} \tag{4}$$

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

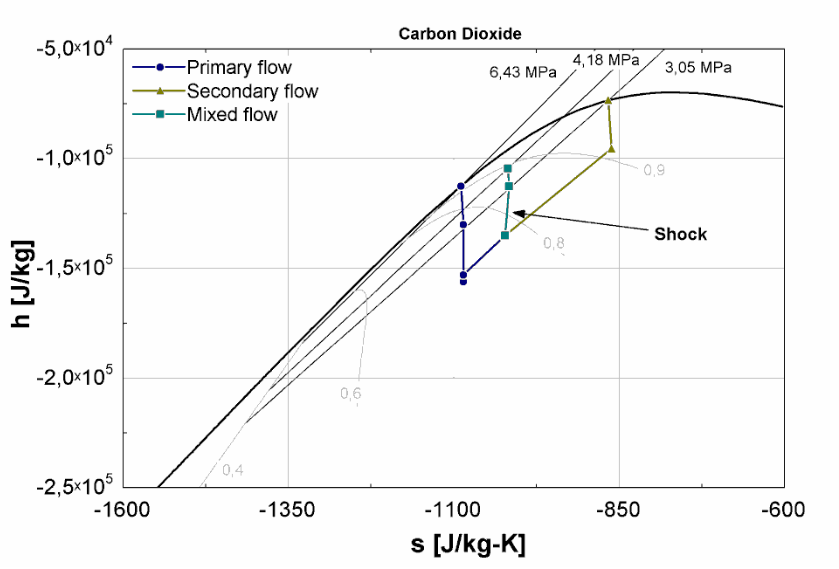



**Figure 4 : Mollier's diagram of the $CO_2$ ejector**

Analogously, for the $CO_2$ cycle:

$$COP_{CO2}(T_A, T_B, T_E) = \frac{\dot{Q}_E}{\dot{Q}_{IA} + \dot{W}_{CO2}} \tag{5}$$

where $\dot{Q}_{IA}$ is the heat transferred at 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.

Defining the COP for the combined cycle:

$$COP(T_G, T_A, T_B, T_E) = \frac{\dot{Q}_E}{\dot{Q}_G + \dot{W}_{CO2} + \dot{W}_{H2O}} \tag{6}$$

The pumping power for the combined cycle:

$$\dot{W}_P = \dot{W}_{CO2} + \dot{W}_{H2O} \tag{7}$$

## 4. RESULTS AND DISCUSION

In a simulation of the cycle, for the base case defined above, a global COP of 0.17 is achieved. In contrast to Carnot cycle, COP for the same sink temperatures is 1.45. Absorption systems based on lithium-bromide cannot achieve temperatures below 0℃ and ammonia based ones require higher temperature heat sources. Thus, though the cycle's COP represents approximately 10% of Carnot performance, it offers a possibility for renewable energy refrigeration, not covered in the range of application of absorption machines and also has the ecological benefits of 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, for a maximum overall COP. All the cases simulated here consider a fixed cooling capacity and optimized ejector's geometry for each condition.

### 4.1. Effect of intercooler $IC_A$ temperature

Figure 5 shows that an optimum temperature for intercooler $IC_A$ operation exists, for temperatures higher than 25℃ the entrained ratio of $CO_2$ cycle decreases. However, for temperatures lower than 25℃ the steam ejector entrained ratio decreases, too. It can be observed that as lower the operation temperature of the evaporator, higher optimum temperature of intercooler $IC_A$. Nevertheless, exists a thermodynamic restriction on this temperature, because it must not be too close of 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 is observed that as higher the intercooler $IC_A$ operation temperature, lower pumping power is required. This is explained because the approximately 90% of the power used for pumping is due the circulation pump of $CO_2$, hence as entrained ratio of $CO_2$ increases when $T_A$ increases, the mass of $CO_2$ pumped decreases.

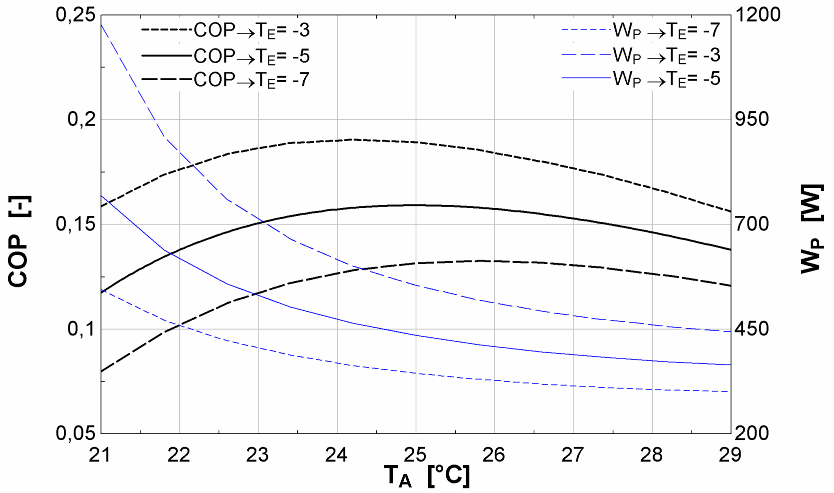



**Figure 5 : Effect of intercooler $IC_A$ temperature on COP**

### 4.2. Effect of intercooler $IC_B$ temperature

Figure 6 shows the effect of intercooler $IC_B$ temperature on overall COP. Similarly to the aforementioned analysis, it is observed the existence of an optimum temperature. However, that optimum appears to be closely to 0℃, which is a thermodynamic limit for the steam ejector cycle. The proximity between the optimum and the water freezing point is even higher when the evaporator temperature decreases.

As the entrained ratio of $CO_2$ decreases as $T_B$ increases, the mass flow of $CO_2$ pumped to the intercooler $IC_A$ increases with the temperature $T_B$. Therefore, the required pumping power of whole cycle increases too.

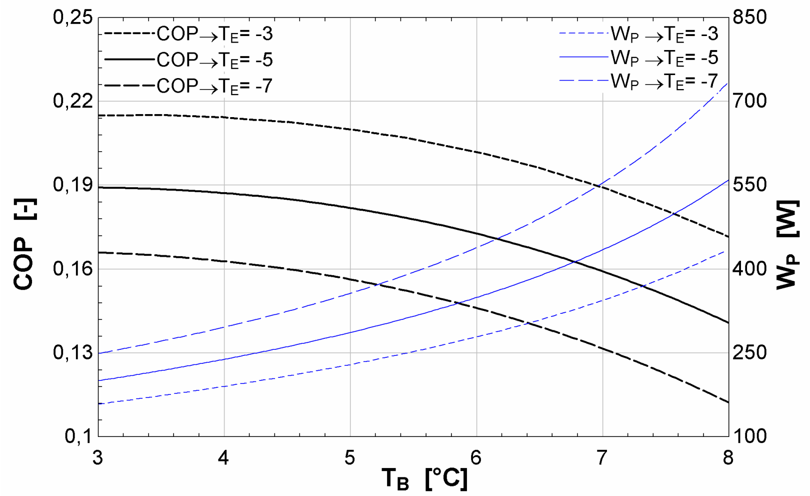



**Figure 6 : Effect of intercooler $IC_B$ temperature on COP**

### 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.

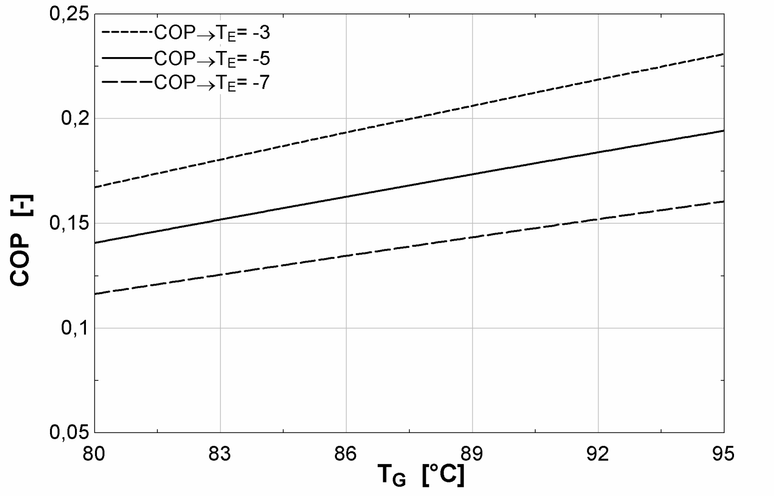



**Figure 7 : Effect of boiler temperature on COP**

## 5. CONCLUSIONS

A cascade ejector cycle using natural refrigerants is proposed. Besides the environmental benefits, this configuration allows to generate a refrigeration effect below water freezing point by considering low temperature heat sources. Although, the cycle is not completely passive, because uses electricity for the circulation pumps, that power may be supplied by 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 cycle's efficiency and heat exchangers size, which certainly affects the overall system cost.

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