# ENERGY AND EXERGY ANALYSIS OF AN EJECTOR REFRIGERATION SYSTEM

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*In this study we conducted an energy and exergy analysis of an ejector refrigeration system with different working fluids. The system operates on waste heat provided by the exhausted gas of an internal combustion engine.Four working fluids are studied: water, methanol, ammonia and R134a. Best performances were achieved for water. The influence of the heating temperature ($t_G$), evaporation ($t_{Ev}$) and condensation ($t_{Cd}$) was studied. An optimum value was reached for $t_G$=140° C, $t_{Cd}$=30° C, $t_{Ev}$=5°C resulting COP=0.48 and $\eta_{ex}$=0.085. The numerical simulations were carried in Engineering Equation Solver (EES ).*

**Keywords:** ejector, refrigeration, exergy, water, performance

## 1. Introduction

Nowadays in order to assure comfort during summer, mechanical compression air conditioning systems are widely used. These systems have the advantage of being easy to use, and having a simple adjustment of temperature. The weakness of these systems is due to increasing electricity price. An alternative is represented by the three-therm systems, namely those with absorbtion or ejection. These systems have two major advantages: energy is the heat source that can be provided by renewable sources: solar, geothermal or waste heat and as well as increased reliability, due to the fact that there are no moving parts, except the solution pump. Due to high initial cost for the absorbtion systems, the ejection systems are more advantageous [1].

This study focuses on the optimization of an ejection refrigerating machine, both in terms of working fluid and of the system parameters. Different researchers [2-8], performed theoretical and experimental studies over the jet compression systems.

In 1999, Sun [2] realized a theoretical study comparing the COP of an ejector system using working fluids such as R718, R123, R134a, R11, R12, R113, R21, R142b, R152a, R318 and R500. The results show that steam jet

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systems have very low COP values. The system using R152a as an working fluid has better performance.

Rogdakis and Alexis [3], theoretically studied a jet system using ammonia as refrigerant. The authors developed a computational model and studied the effect of the variation of different temperatures in the components of the system such as: generator, condenser and evaporator over the COP and the ejector performance.

Comparative studies were also developed by Cizung et al. (2001) [4], Selvaraju and Mani (2004) [5]. They concluded that the system performance depends mainly on the ejector geometry, compression ratio and on the type of refrigerant.

Riffat and Omer [6] present the results of an analysis and an experimetal investigation of a system with methanol. They obtained experimental values of COP between 0.2 and 0.4 at operating conditions achievable using low-grade heat such as solar energy and waste heat [6].

Sankaral and Mani [7] in 2007 published the results of an experiment performed over an ejector system with ammonia, and concluded that entrainment ratio and COP increase with increase in ejector area ratio and expansion ratio and with the decrease of the compression ratio.

Ziapour and Abbasy [8] theoretically investigated an ejector system with water according to first and second law.The second law efficiency of the heat pipe/ejector refrigeration cycle increases with increasing in evaporator temperature and decreasing in condenser temperature.

In this paper, we have concentrated our study on the ejector refrigerating that will operate on waste heat provided by the exhausted gas of an internal combustion engine. We developed a computational model in Engineering Equation Solver (EES) according to first and second law. In the first part of the work, we have conducted a comparative study of the performances of the system with different refrigerants: water, methanol, ammonia and R134a. The results show that the system using water in given conditions has better perfomance and also is a feasible solution, considering the boiling pressure level.

The objective of the present work was to study in terms of energy and exergy the performance of the system and it's main components, for different temperatures. Simulations were made for variations in the temperature of the generator, the condenser, the evaporator and an optimum was established.

## 2. Description of the system

In this paper we studied different operating regimes of a jet compression system which should ensure a cooling load of 45.6 kW. The main parameters of the system, the temperatures of: evaporation, condensation, cooled and cooling water are presented in Table 1.

*Table 1*

**Basic parameters of the refrigerating machine**

| | |
|---|---|
| $Q_{Ev}$ [kW] | 45,6 |
| $t_{Ev}$ [°C] | 5 |
| $t_{Cd}$ [°C] | 30 |
| $t_{Evi}$ [°C] | 12 |
| $t_{Eve}$ [°C] | 7 |
| $t_{Cdi}$ [°C] | 25 |
| $t_{Cde}$ [°C] | 29 |

Considering the parameters from Table 1, a computer simulation was done according to energy and exergy analysis. The temperature was varied at the steam generator. The simulation was performed for the next agents: R718 (water), methanol, R717 (ammonia) and R134a.

In an ejection system, the mechanical compressor is replaced by the ejector. The ejector is the key component in this combined cycle. The ejector consists of several parts: a nozzle section for a primary flow and a suction chamber for the secondary flow, a mixing chamber where the primary flow and secondary flow mix, a throat section in which the mixed fluid undergoes a transverse shock and a pressure rise [9]. The last section is the diffuser in which the mixed fluid recompresses to the back pressure.The schematic of an ejector refrigeration system is presented in Fig. 1.

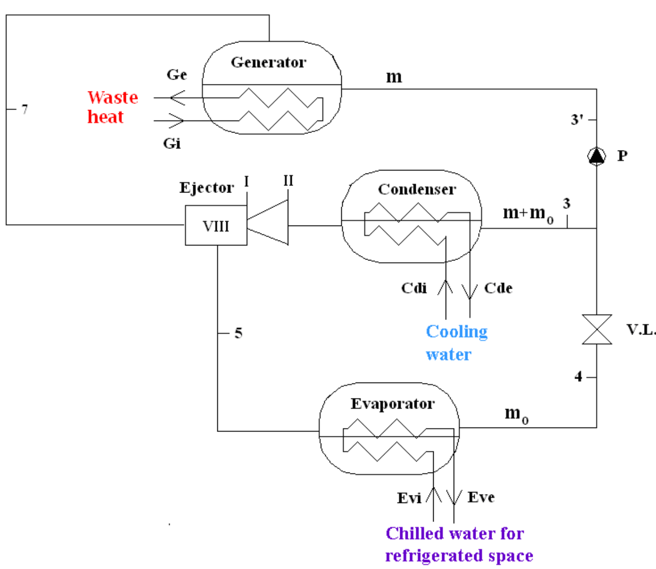


Fig. 1 Scheme of an ejector refrigeration system

## 3. Analysis

### 3.1. Energy analysis

An energy analysis was perfomed using the EES software. EES or Engineering Equation Solver, gives the numerical solution of a set of equations. In order to simulate the ejector system, the mathematical model of the machine was build. The model of the ejector chiller is constituted of a set of energy, mass and momentum equations. In order to establish the model, the following assumptions are made: the flow inside the ejector is in steady state, the pressure drops due to friction in evaporator and on the pipes are neglected, the mixing process in the mixing section of ejector occurs at constant pressure and complies with the conservation of energy, the ejector is adiabatic.

The equations that describe the processes in the system and in the ejector are according to the model presented by other authors [10, 11]. The equations that describe the processes in the ejector, the heat capacities, the mass and energy balance for the various components of the ejection system are presented below.

- velocity at the exit of the primary nozzle of the ejector

$$w_{VIII} = \varphi_1 \sqrt{2(h_7 - h_{VIII'})} \quad (1)$$

where $\varphi_1$=0,95 with it's recommended values between 0,92 – 0,96 [11], is a coefficient of velocity reduction due to gas friction.

- velocity at the exit of the mixing chamber of the ejector

$$w_I = \varphi_2 \frac{w_{VIII}}{1+u_r} \quad (2)$$

where $\varphi_2$=0,975 is the coefficient of velocity reduction due to friction

- enthalpy increase in the mixing chamber

$$\Delta h_{MC} = \frac{w_{VIII}^2}{2(1+u_r)}\left(1-\frac{\varphi_2^2}{1+u_r}\right) \quad (3)$$

In order to determine the real state at the exit of the mixing chamber, state I, we apply the energy balance equation for this part of the ejector:

$$\dot{m} h_{VIII} + \dot{m}_0 h_5 + (\dot{m}+\dot{m}_0)\Delta h_{MC} = (\dot{m}+\dot{m}_0)h_I \quad (4)$$

$$h_I = \frac{h_{VIII} + u_r h_5 + (1+u_r)\Delta h_{MC}}{1+u_r} \quad (5)$$

- enthalpy at the exit of the difuser

$$h_{II} = h_I + \frac{h_{II'} - h_I}{\varphi_4^2} \quad (6)$$

where $\varphi_3$=0,90 is a coefficient of velocity reduction due to friction with it's recommended values between 0,88 – 0,92.

After the determination of the real state, it is necessary to recalculate the ejection factor and specific steam consumption according to the relations 7 and 8.

$$u_{rec} = \varphi_1 \varphi_2 \varphi_3 \sqrt{\frac{h_7 - h_8}{h_{II'} - h_I}} - 1 \tag{7}$$

$$a_{rec} = \frac{1}{u_{rec}} \tag{8}$$

With these recalculated values further on are determined: flows, thermal loads and COP.

- determination of the specific heat capacity of the evaporator

$$q_{Ev} = h_5 - h_4 \tag{9}$$

- secondary flow or refrigerant flow

$$\dot{m}_0 = \frac{\dot{Q}_{Ev}}{q_{Ev}} \tag{10}$$

- primary flow or entraining steam

$$\dot{m} = a_{rec}\, \dot{m}_0 \tag{11}$$

- the heat load of the components

$$\dot{Q}_{Cd} = (\dot{m}_0 + \dot{m})(h_{II} - h_3) \tag{12}$$

$$\dot{Q}_G = (\dot{m}_0 + \dot{m})(h_7 - h_{3'}) \tag{13}$$

- performance of the cycle

$$COP = \frac{\dot{Q}_{Ev}}{\dot{Q}_G + \dot{W}_P} \tag{14}$$

In our study, it will be assumed that the pumping work, $\dot{W}_P$, is neglected.

### 3.2. Exergy analysis

Exergy analysis is used to estimate the exergy destructions and thus to point out the malfunctions occurring in every component of the refrigerating system [12]. An exergetic analysis was performed in EES software. The reference status was considered at $p_0$=1 atm and $t_0 = t_{Cdi}$=25ºC. In order to study every component according to second law, the following elements were associated: a Fuel (the exergetic resources supplied or the exergetical potential at the beginning of the process), a Product (what offers the component exergetically) and the Irreversibilities, meaning the exergy consumed [13].

A mathematical model for the study on the entropy generation for a refrigeration machine was developed by Petrescu et. al in 2012, considering the direct method[14]. Also an exergy analysis for choosing the optimal temperature difference in a recuperative heat exchanger of a cryogenic system was performed by Dobrovicescu et al. [15].

The corresponding expressions for the Fuel, Product and Irreversibility of the system components are presented below according to Dobrovicescu, [13,15].

First of all we have calculated the reference state to which we relate.

$$A_0 = h_0 - T_0 s_0 \tag{15}$$

where $A_0$ is the water exergy at reference state "0", respectively $t_0=t_{Cdi}=25^oC$ and $p_0=101{,}325kPa$.

The specific exergy of every state point is calculated as follows:

$$ex = h - T_0 s - A \tag{16}$$

Evaporator:

$$P_{Ev} = \dot{E}x_{Eve} - \dot{E}x_{Evi} \tag{17}$$

$$Cb_{Ev} = \dot{E}x_5 - \dot{E}x_4 = \dot{m}_0(ex_5 - ex_4) \tag{18}$$

$$\eta_{exEv} = \frac{P_{Ev}}{Cb_{Ev}} \tag{19}$$

$$I_{Ev} = Cb_{Ev} - P_{Ev} \tag{20}$$

$$Ir_{Ev} = \frac{I_{Ev}}{I_{tot}} 100 \tag{21}$$

The expressions for the exergy efficiency (η), exergy destruction (I) and percentage of exergy destruction (Ir), relative to total destruction, are similar to all components.

Condenser:

$$P_{Cd} = \dot{E}x_3 + \dot{E}x_{Q_{Cd}} = (\dot{m} + \dot{m}_0)ex_3 + \dot{E}x_{Q_{Cd}} \tag{22}$$

$$\dot{E}x_{Q_{Cd}} = \dot{Q}_{Cd}(1 - \frac{T_0}{T_0}) = 0 \tag{23}$$

$$Cb_{Cd} = \dot{E}x_{II} = (\dot{m} + \dot{m}_0)ex_{II} \tag{24}$$

Steam Generator:

$$P_G = \dot{E}x_7 = \dot{m}ex_7 \tag{25}$$

$$Cb_G = \dot{E}x_{3'} + \dot{E}x_{QG} = \dot{m}ex_{3'} + \dot{E}x_{QG} \tag{26}$$

$$\dot{E}x_{QG} = \dot{Q}_G(1 - \frac{T_0}{T_G}) \tag{27}$$

Ejector:

$$P_{Ej} = \dot{E}x_{II} = (\dot{m} + \dot{m}_0)ex_{II} \tag{28}$$

$$Cb_{Ej} = \dot{E}x_5 + \dot{E}x_7 = \dot{m}_0\, ex_5 + \dot{m}\, ex_7 \tag{29}$$

The overall exergetic efficiency of the refrigerating machine is:

$$\eta_{exMEJ} = \frac{P_{Ev}}{\dot{E}x_{Q_G}} \tag{30}$$

## 4. Results and discussions

4.1. Numerical results for different working fluids

The ejector refrigerating machine will operate on waste heat provided by the exhausted gas of an internal combustion engine of 3319 $cm^3$. We studied the performance of the system, working with different fluids considering various generator temperatures, while $t_{Cd}$=30°C and $t_{Ev}$=5°C The working fluids are: water (R718), methanol, ammonia (R717) and R134a. The results are presented in table 2.

*Table 2*

**Variation of the energy and exergy performance with the generator temperature**

| | Water | | Methanol | | Ammonia | | R134a | |
|---|---|---|---|---|---|---|---|---|
| $t_G$ | $\eta_{ex}$ | COP | $\eta_{ex}$ | COP | $\eta_{ex}$ | COP | $\eta_{ex}$ | COP |
| °C | - | - | - | - | - | - | - | - |
| 70 | 0.0597 | 0.1442 | 0.4572 | 0.1435 | 0.0173 | 0.1309 | 0.0146 | 0.0968 |
| 80 | 0.7088 | 0.2033 | 0.2592 | 0.2025 | 0.0233 | 0.1877 | 0.0198 | 0.1402 |
| 90 | 0.0772 | 0.2546 | 0.2076 | 0.2538 | 0.0278 | 0.2376 | 0.0232 | 0.1762 |
| 100 | 0.0809 | 0.2996 | 0.1823 | 0.2988 | 0.0312 | 0.2820 | 0.0240 | 0.2037 |
| 110 | 0.0830 | 0.3392 | 0.1660 | 0.3381 | 0.0336 | 0.3223 | | |
| 120 | 0.0841 | 0.3744 | 0.1558 | 0.3746 | 0.0352 | 0.3600 | | |
| 130 | 0.0846 | 0.4058 | 0.1478 | 0.4070 | 0.0361 | 0.3982 | | |
| 140 | 0.0850 | 0.4341 | 0.1410 | 0.4357 | | | | |
| 160 | 0.0841 | 0.4829 | 0.1327 | 0.4896 | | | | |
| 180 | 0.0832 | 0.5242 | 0.1255 | 0.5325 | | | | |
| 200 | 0.0822 | 0.5601 | 0.1202 | 0.5694 | | | | |
| 220 | 0.0814 | 0.5928 | 0.1178 | 0.6044 | | | | |
| 240 | 0.0808 | 0.6236 | 0.1194 | 0.6391 | | | | |

The system that operates with water and methanol has the best COP, while for exergy efficiency methanol shows better characteristics. With increasing boiling temperature, the exergy performance for methanol decreases, being closer to the water performance. On the other hand, as it can be seen in Table 3, the boiling pressures for methanol, ammonia and R134a have important values with the increasing in $t_G$ , which involves the use of pipes with high nominal pressures and risk of explosion. For example considering a generator temperature of 100° C, the vapor saturation pressure is 1 bar for water, 62 bar for ammonia and almost 40 bar for R134A. If we consider methanol, the pressures are lower, but even so, for temperatures over 160° C, pressures also increases to high values.

These elements, combined with the fact that the heat source are the exhaust gases from an internal combustion engine, which provides high temperatures, recommend using water as the most suited fluid for given conditions.

*Table 3*

**Boiling pressure variation with $t_G$**

| $t_G$ [°C] | 70 | 80 | 90 | 100 | 110 | 120 | 130 | 140 | 160 | 180 | 200 | 220 | 240 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| $p_G$ Water [bar] | 0.31 | 0.47 | 0.70 | 1.01 | 1.4 | 1.9 | 2.7 | 3.6 | 6.2 | 10.0 | 15.5 | 23.2 | 33.5 |
| $p_G$ Methanol [bar] | 1.2 | 1.8 | 2.5 | 3.5 | 4.7 | 6.3 | 8.3 | 10.7 | 17.3 | 26.7 | 39.7 | 57.2 | 80.7 |
| $p_G$ Ammonia [bar] | 33.1 | 41.4 | 51.2 | 62.6 | 75.8 | 91.2 | 108 | | | | | | |
| $p_G$ R134a [bar] | 21.2 | 26.3 | 32.5 | 39.7 | | | | | | | | | |

4.2. Model validation

In order to validate the mathematical model, several theoretical and experimental studies of ejection refrigerating machines were considered [5-8].

In Figure 2, simulation results obtained in this work are compared to those of a simulation of a system that has water as refrigerant [8]. The parameters of simulation were $t_{Cd}$=15° C and $t_{Ev}$=5° C.

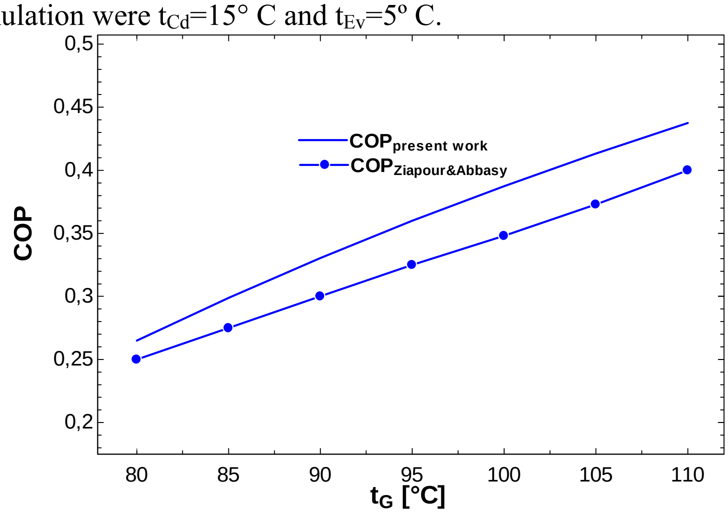


Fig. 2 – COP comparation between present work and (Ziapour and Abbasy,2010)

These authors [10], studied the system according to the second principle of thermodynamics. For $t_G$ =100° C resulted the values presented in Table 4.

*Table 4*

**Percentage of exergy destruction/component for $t_G$=100ºC present work vs. the study of Ziapour and Abbasy, (2010)**

| Ircomponent[%] | Condenser | Ejector | Evaporator | Generator |
|---|---|---|---|---|
| Present work | 11 | 81 | 2,5 | 5,5 |
| Ziapour and Abbasy, 2010 | 14 | 79 | 2 | 5 |

Riffat and Omer in 2001[6] conducted a theoretical study over a ejector system working on methanol. For $t_G$=180ºC, $t_{Cd}$=28ºC and $t_{Ev}$=-2ºC the COP was 0,4, while for the present work simulation resulted a COP of 0.42.

Selvaraju and Mani in 2004 [5] developed an experimental study of ejector refrigerating machine using different refrigerants. For R134a considering $t_G$ = 80° C, $t_{Cd}$ = 25° C and $t_{Ev}$ = 5 ° C they obtained a COP of 0.26. Applying the simulation program designed in this paper, we obtain a value of 0.247.

Sankaral and Mani in 2007 [7], in an experimental study, for and ejector system working on ammonia, for $t_G$ = 72° C, $t_{Cd}$ = 27° C and $t_{\backslash Ev}$ = 15° C obtained a COP of 0,29. For the same parameters, in the present work simulation was achieved a COP of 0.35.

The simulation for the verification of the accuracy of the model was made considering the same: type of the working fluid, $t_{G,}$, $t_{Cd}$ and $t_{Ev}$ , data presented by the authors in their papers. We didn't possess the informations regarding to the other coefficients, dimmensions and parameters that defines the ejector in order to simulate with 100% accuracy the systems presented in the previous studies. Taking into account the above considerations, the results presented and the relatively errors, lying between 5-10%, comparatively with previous studies [5-7, 10],all these elements confirm the validity of the model.

### 4.3. Parametric study of the ejector refrigeration system with water as working agent

Next we proceeded to study the behavior of the ejection system in which the refrigerant is water. We started by varying the temperature of the generator, while the other parameters presented in table 1, remain constant.

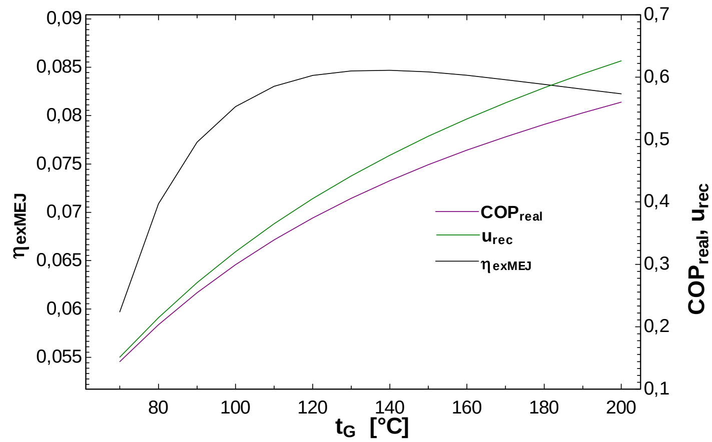


Fig. 3 – Variation of COP, ejection factor and exergy efficiency with $t_G$

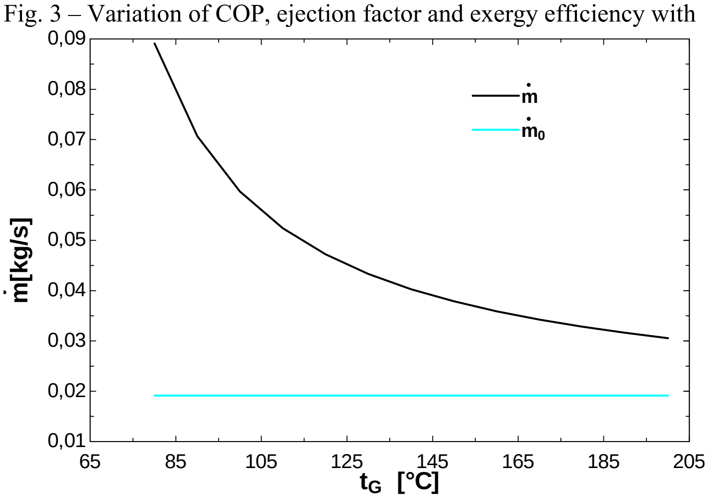


Fig. 4 – Variation of primary and secondary flow with $t_G$

In Fig. 3 one can observe that the exergy performance of the system reaches a peak at about 140° C. In terms of the first principle, the energy performance (COP) and the ejection factor ($u_{rec}$), which by definition is the ratio between the flows of cold vapours and of the entraining steam [10], increases with the increasing in the boiling temperature. This is explained by the increased speed of the steam leaving the convergent-divergent nozzle. This fact leads to a decrease in the pressure in the mixing chamber. Knowing that the speed is inversely proportional to the pressure, this involves an increase in the gap between the evaporating pressure, or the pressure of the cold vapour and steam, which increases the driving capability of the steam. But considering the fact that the cooling load, $Q_{Ev}$, remains constant and so does the secondary flow, $\dot{m}_0$, (Figure 4), in order to obtain the same effect of refrigeration, a smaller amount of steam is required. So an increase of the $u_{rec}$ means a decrease in the flow of steam, $\dot{m}$, required to produce the same amount of cold, so the capacity to produce refrigeration effect is increased, respectively the COP.

Ejection factor increases with the increase of $\frac{p_G}{p_{Ev}}$ and it is inversely proportional with $\frac{p_{Cd}}{p_{Ev}}$ ratio, that represents the compresion factor.

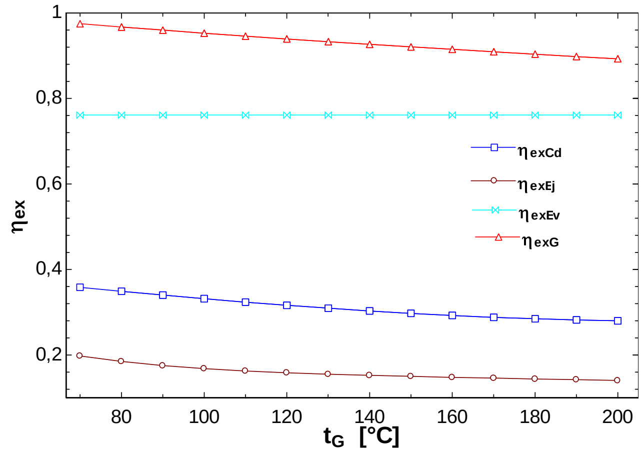

Fig. 5 – Variation of exergy efficiency of the components with $t_G$

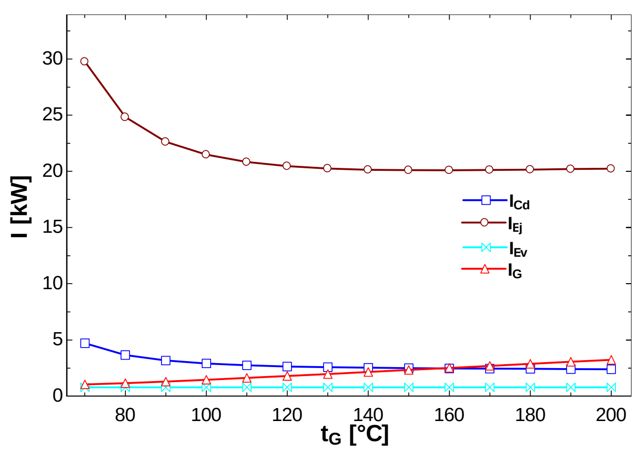

Fig. 6 – Variation of the exergy destruction by components with $t_G$

As it can be seen in Figures 5 and 6, the ejector is a key component in the whole system. It has the lowest exergy efficiency ranging between 0.3 and 0.15, Figure 5, and the highest exergy destruction, Figure 6. It results that any increase or decrease in functional performances of this component leads to major influences on the functioning of the entire system.It can be observed the correlation between variation of $\eta_{exMEJ,}$ which reaches a maximum at $t_G$ = 140 ° C, and the decrease of the destruction in the ejector, at the same temperature, Figure 6. Subsequent drop of exergy efficiency over 140° C is justified by the increasing of destruction in the generator, while at the ejector it remains almost constant.

The rate of exergy destruction variation shown in Figure 7, reconfirms that the destruction of the ejector has a share of over 80% in the system.

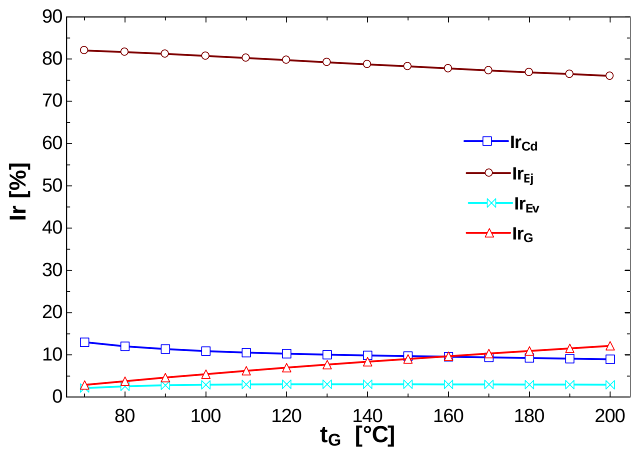


Fig. 7 – Variation of the rate of the exergy distruction in the main components

Fig. 8 presents the influence of the condensing temperature over the energy and exergy performance. The study is performed for a constant evaporator temperature of 5° C and for a $t_G$ of 100°, 120°, 140° and 160°C. The best energy performance is achieved for $t_{Cd}$=30° C and $t_G$=160°C. If the criteria is the second law, the best performance is achieved for $t_{Cd}$=30° C and $t_G$=140°C.

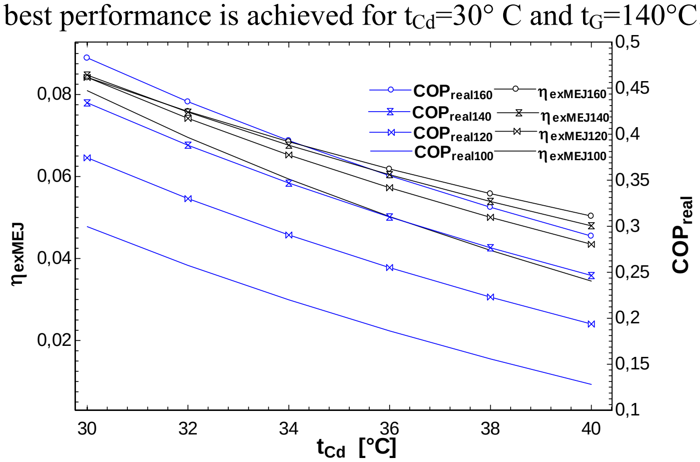


Fig. 8 – The effect of $t_{Cd}$ on COP and $\eta_{exMEJ}$

The ejector has two roles:

- to compress the total flow pumped into the system from $p_{Ev}$ to $p_{Cd}$. An increase in $p_{cd}$ leads to a higher input work, which involves a decrease in the COP and the exergy efficiency of the system, Fig. 7.

- to extract $\dot{m}_0$ flow, that comes from the evaporator at $p_0$ pressure. An increase in $t_{Ev}$, consequently $p_{Ev}$, leads to a decrease in driving power required to the ejector, which has a positive influence on system performance, Figure 9.

An increase of the evaporation temperature decreases the total irreversibility mainly due to the reduction of the required heat to the generator and the rejected heat from the condenser.

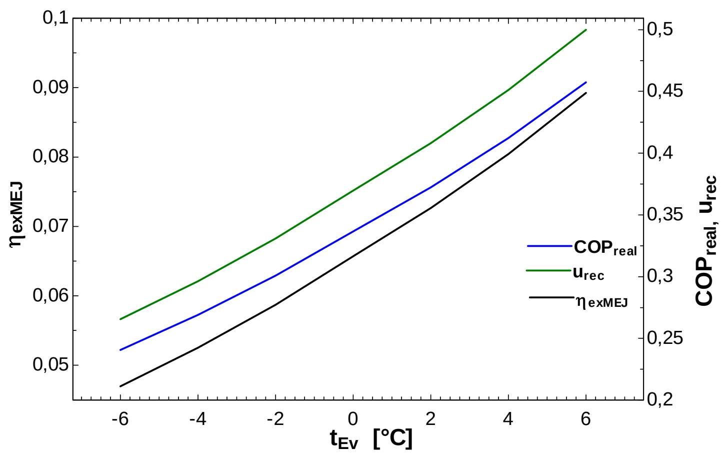


Fig. 9 – The influence of $t_{Ev}$ on COP and on exergy efficiency for $t_G$=140°C and $t_{Cd}$=30°C

## 4. Conclusions

A comparison according to first and second law, has been made on different working fluids for the same operating parameters. It can be concluded that the performances of all the selected working fluids are improved with higher generator temperatures. Considering the boiling pressure, the performances and that the fuel is waste heat, we considered water as the most suited fluid for the given conditions. The variation in the performance of the water ejector refrigeration system was evaluated based on a computer simulation in EES. The operational parameters $t_G$ , $t_{Cd}$ and $t_{Ev}$ have been varied.

The conclusions of the computer simulation are:

- the COP of the ejector refrigeration system increases with increasing the generator and evaporator temperature and decreasing condenser temperature.

- an increased performance involves a greater ejection factor, respectively a lower specific steam consumption.
- the exergy efficiency curve follows the same variation with the COP, except the variation with the $t_G$, when it reaches a maximum at 140° C. For higher $t_G$ , the exergy efficiency slowly decreases.
- the second-law efficiency of the ejector refrigeration system increases with increasing evaporator temperature and decreasing condenser temperature.
- the ejector is a key component in the system; about 80% of the exergy is destroyed here.

***Nomenclature***

| | |
|---|---|
| $a$ | specific steam consmuption, [kg primary steam/ kg cold vapours ] |
| $A_0$ | exergy of the reference state |
| *COP* | coefficient of performance |
| *Cb* | fuel (resource), [kW] |
| *ex* | specific exergy, [$kJkg^{-1}$] |
| $\dot{E}x$ | exergy flow rate, [kW] |
| $h$ | specific enthalpy, [$kJkg^{-1}$] |
| $I$ | exergy destruction, [kW] |
| *Ir* | rate of exergy destruction, [%] |
| $\dot{m}$ | mass flow rate primary steam, [$kg\ s^{-1}$] |
| $\dot{m}_0$ | mass flow rate cold vapours, [$kg\ s^{-1}$] |
| $p$ | pressure, [bar] |
| $P$ | product, [kW] |
| $q$ | heat capacity, [$kJkg^{-1}$] |
| $\dot{Q}$ | heat flow rate, [kW] |
| $s$ | specific entropy, [$kJkg^{-1}K^{-1}$] |
| $t$ | temperature, [°C] |
| $T$ | temperature, [K] |
| $u$ | ejection factor, [kg cold vapours / kg primary steam] |
| w | velocity, [$ms^{-1}$] |
| W | work [W] |

***Greek letters***

| | |
|---|---|
| $\varphi$ | coefficient of velocity reduction |
| $\Delta t$ | temperature difference, [°C] |
| $\eta_{ex}$ | exergetic efficiency |

***Subscripts***

| | |
|---|---|
| Cd | condenser |
| Cde | cooling water exit from condenser |
| Cdi | cooling water inlet to condenser |
| Ej | ejector |

| | |
|---|---|
| Ev | evaporator |
| Eve | evaporator exit of the cooled water |
| Evi | evaporator inlet of the cooled water |
| G | generator |
| Ge | heat exit to engine |
| Gi | heat inlet from engine |
| MEJ | refrigerating ejector machine |
| P | pump |
| r | real state |
| rec | recalculated |
| V.L. | throttling valve |
| 0 | environmental state |
| 1-5,I,II,VIII | state points |

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