ISSN 2176-5480

22nd International Congress of Mechanical Engineering (COBEM 2013)
November 3-7, 2013, Ribeirão Preto, SP, Brazil

EXERGY ANALYSIS OF CASCADE REFRIGERATION SYSTEM FOR LOW TEMPERATURES USING ECOLOGICAL FLUIDS

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Abstract. *Cascade refrigeration systems work with two or more cycles in serial and can obtain internal temperatures below -60°C, necessary to several activities in medicine and scientific research. The main objective of this article is to present a thermodynamic analysis of cascade system refrigeration with the use of natural refrigerant fluids for low temperatures. These fluids have ecological character and are an alternative to hydrochlorofluorocarbons (HCFCs) and to hydrofluorocarbons (HFCs). The energy and exergy analysis were done from a thermodynamic model based on the law of conservation of mass, on the first and second laws of Thermodynamics and were developed a simulator to assess the technical practicability of this system.*

Keywords: *exergy, cascade refrigeration, ecological fluids*

1. INTRODUCTION

Typically cascade refrigeration systems are employed in applications with temperatures below -60 ° C, which it is not possible to remove heat from the lowest temperature level and reject it to the atmosphere directly (Cleto, 2008). They are composed of independent cycles each one with its own refrigerant, but with a heat exchanger in common. Its main advantage in relation to refrigeration cycles multistage is that: cycles do not contain the same refrigerant fluid, so same refrigerant doesn't need to operate at higher and lower pressure levels.

Usually, cascade systems operate using synthetic refrigerants such as HCFCs and HFCs, which are currently being replaced by natural or ecological fluids, attending to an environmental cause. The hydrochlorofluorocarbons (HCFCs) destroy the ozone in the atmosphere, which protects the planet from ultraviolet radiation. This phenomenon occurs because the rise to the stratosphere, the gases undergo the action of UV rays, releasing free radicals that react with the ozone molecules, forming an oxygen molecule (O_2) and a molecule of chlorine oxide (ClO), causing destruction of O_3. The ClO is short-lived and quickly reacts with a free oxygen atom, releasing free radical which returns to destroy another O3 molecule. The hydrofluorocarbons (HCFs) do not contain chlorine and therefore do not deplete the ozone. However, they have, as well as HCFCs, a halogen (fluorine) in its composition, contributing to the greenhouse effect.

The cascade refrigeration systems using natural refrigerants therefore present themselves as an ecologically relevant alternative.

The theoretical analysis of thermal systems and processes have historically been based on the application of the laws of conservation of mass, momentum and energy. The energy performance of refrigerants systems is usually evaluated based on the first law of Thermodynamics. However, compared to energy analysis, exergy analysis can better and accurately show the location of inefficiencies. The exergy method is a relatively new technique in which the basis of evaluation of thermodynamic losses follows the second law rather than the first law of Thermodynamics. The results from exergy analysis can be used to assess and optimize the performance of refrigerant systems (Ahamed *et al*, 2011).

The main objective of this work is to perform an assessment of the energy and exergy point view of a cascade refrigeration system for low temperatures using natural fluids. Thus, it is possible determine the efficiencies of the processes based on the first and second laws of thermodynamics and the thermodynamic properties of the working

ISSN 2176-5480

fluids. Therefore, simulations were made on the platform Engineering Equation Solver (EES), considering their operating conditions and external influences as a way to assess the technical feasibility of implementation.

2. BIBLIOGRAPHIC REVIEW

2.1 Cascade refrigeration system

Cascade refrigeration systems consist of at least two refrigeration cycles that work independently. The two refrigeration cycles are connected by a cascade heat exchanger where heat is released in the condenser low-temperature circuit and it is absorbed from the evaporator high-temperature circuit (Alhamid *et al*, 2010).

Thus, the desired refrigeration effect occurs in the evaporator low-temperature, and rejection of heat from the cycle as a whole takes place in the high temperature condenser.

In systems in which the same refrigerant passes through the stages of high and low temperature, extreme values of pressure and specific volume can cause problems. In fact, when the evaporation temperature is too low, the specific volume of refrigerant vapor in the compressor suction is high, which implies a high volumetric capacity compressor.

Regarding pressure, an important aspect of the cascade system is that the refrigerant in two or more stages can be selected so as to present reasonable pressures in the evaporator and the condenser in two or more temperature intervals.

In a double cascade system (Fig. 1), a refrigerant to be selected for the cycle A must have such a relationship between the saturation pressure and temperature that permits refrigeration at a relatively low temperature without excessively low pressure in the evaporator. The refrigerant cycle B must have characteristics of saturation that allows condensation to the desired temperature in the absence of excessively high pressures in the condenser.

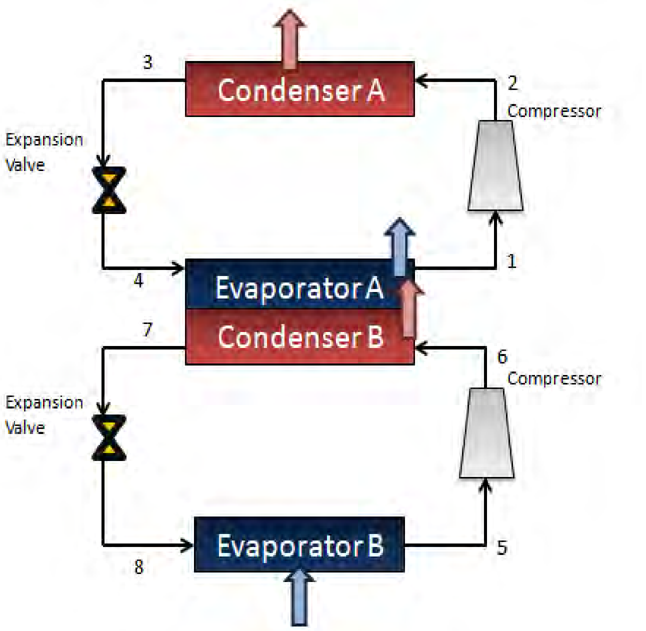

Figure1. Cascade refrigeration system

2.2 Exergy

Exergy is the maximum theoretical work that can be obtained from a global system, as it comes into equilibrium with the environment (reaches the dead state). Thus, if the pressure and temperature are equal to room temperature, this is the dead state, with no longer able to generate work (Moran and Shapiro, 2009).

Thus, exergy is a property that quantifies the availability of a system, which is usually largely destroyed because of the irreversible nature of thermodynamic processes. However, the exergy not only can be destroyed by irreversibilities, but can also be transferred to others systems. The exergy transferred from one system to its surroundings and which is not used usually represents a loss. We can obtain a better utilization of energy resources by reducing the exergy destruction within a system and / or reducing losses (Kotas, 1995).

The balance of exergy for a control volume is presented by (Szargut *et al*, 1988):

$$\frac{dE_{vc}}{dt} = \sum_j \left(1 - \frac{T_0}{T_j}\right)\dot{Q}_j - \left(\dot{W}_{vc} - p_0 \frac{dV_{vc}}{dt}\right) + \sum_e \dot{m}_e e_{fe} - \sum_s \dot{m}_s e_{fs} - \dot{E}_d \qquad (1)$$

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where $\frac{dE_{vc}}{dt}$ is exergy flow rate of a control volume, the terms $(1-\frac{T_0}{T_j})\dot{Q}_j$ represent the transfer of exergy flow accompanying the heat transfer rate $\dot{Q}_j$, which occurs at border points at which the instantaneous temperature is T_j. $\dot{W}_{vc}$ represent the energy transfer by work flow rate for control volume and the transfer of exergy flow rate is given by $\dot{W}_{vc} - p_0\frac{dV_{vc}}{dt}$, where $\frac{dV_{vc}}{dt}$ is the volume flow rate. The terms $\sum_e \dot{m}_e e_{fe} - \sum_s \dot{m}_s e_{fs}$ represent the exergy transfer with mass entering and leaving the control volume, respectively. $\dot{E}_d$ is the exergy destruction flow rate due to irreversibilities.

In steady state, $\frac{dE_{vc}}{dt} = \frac{dV_{vc}}{dt} = 0$, thus obtaining exergy balance for the steady state in terms of rate,

$$0 = \sum_j(1-\frac{T_0}{T_j})\dot{Q}_j - \dot{W}_{vc} + \sum_e \dot{m}_e e_{fe} - \sum_s \dot{m}_s e_{fs} - \dot{E}_d \tag{2}$$

where e_{fe} denotes the exergy per unit mass that enters and traverses the e_{fs} denotes the exergy per unit mass passing through the exit s. These terms, known as exergy specific flow, are expressed by

$$e_f = h - h_0 - T_0(s - s_0) + \frac{V^2}{2} + gz \tag{3}$$

where h and s represent specific enthalpy and entropy, respectively, at the inlet or outlet considered; h_0 and s_0 represent the respective values of these properties when evaluated in T_0, P_0, temperature and pressure in dead state.

2.3 Refrigerant fluids

Since the establishment of the Montreal Protocol, the refrigeration industry has sought substitutes for CFCs and HCFCs refrigerants.

The use of hydrocarbons (HCs) as refrigerants in some applications of refrigeration and air conditioning has been a valid alternative. Compared with CFCs, hydrochorofluocarbons (HCFCs) and hydrofluorocarbons (HFCs), hydrocarbons refrigerants offer zero Ozone Depletion Potential (ODP) and extremely low Global Warming Potential (GWP) and, in regard to their performance, they offer in general: high efficiency, reduces charge sizes, good miscibility with mineral oils (synthetic lubricants are not required), lower compressor discharge temperatures, and slightly better heat transfer within heat exchangers (Corberán *et al*, 2008).

From the hydrocarbons, isobutane (R600a) is the most frequently used refrigerant. In Europe it is used in household refrigerators, with a market share of more than 95% in many countries. Propane (R290) and propylene (R1270) are used by many heat pump manufactures, and have also been used in air conditioners and in commercial refrigeration systems (Palm, 2008). Ethylene (R-1150) is one of the most important raw materials in the petrochemical industry and is used in the synthesis of a series of products such as: ethylene oxide, ethylene glycol, ethyl alcohol, polyethylene, polystyrene. It is used, mixed with nitrogen, to accelerate the ripening other fruits.

The cascade systems currently still use refrigerant fluids with high GWP, such as R-508B and R-404a. These fluids are HFCs, are not inflammable and have low toxicity. In cascade systems, R-404A is refrigerant fluid from high temperature cycle and R-508B, refrigerant fluid from low temperature cycle.

The global warming potential (GWP) of hydrocarbons is 3, while for R-508B it is 10350 and for R-404A it is 3260 (Hansen and Haukas, 2000).

Thus, hydrocarbons can be substitutes also relevant for these systems, being technically feasible and can operate over a wide range of evaporation temperature (until the lower limit of -170 ° C). However, practical applications are restricted by security codes and national regulation because they are flammable fluids. Thus, appropriate safety measures should be used during handling, fabrication, maintenance and servicing.

3. METHODOLOGY

3.1 Thermodynamic modeling

In the modeling of cascade refrigeration system (Fig. 1), some assumptions were adopted to perform exergy analysis:

- All systems are operating at steady state;
- Negligible pressure and heat losses/gains in the pipe networks or system components;
- Variation of kinetic and potential energies in all equipments are neglected;
- Isenthalpic expansion in expansion valve.

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T-S diagram of the refrigeration cycle is shown in Fig. 2. In the diagram, corresponding to high temperature cycle (Fig. 1), 1-2 is isentropic compression in the compressor. The other states 2-3, 3-4 and 4-1 show condensation, throttling in the expansion valve and evaporation in evaporator respectively. The same process occurs for the low temperature cycle.

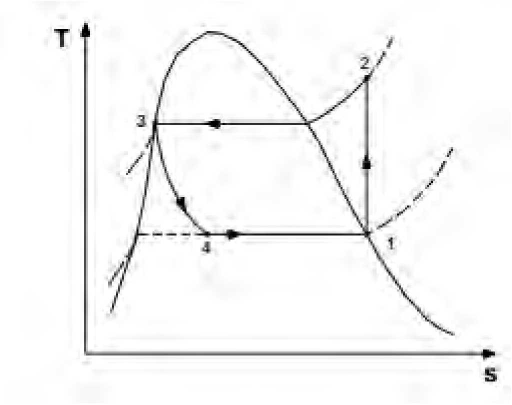

Figure 2. T-S diagram of vapor compression refrigeration system

We applied the balances of mass, energy and exergy for the control volumes of the cycles of high (Cycle A, Fig. 1) and low temperature (Cycle B, Fig. 1): condenser, expansion valve, compressor, and evaporator. This is the balance equations for each component of the system:

For compressor from high temperature-circuit, Eq. (4) to (6):

$$\dot{m}_2 = \dot{m}_1 \tag{4}$$
$$\dot{W}_H = \dot{m}_1(h_{2s} - h_1) \tag{5}$$
$$\dot{E}_{d\ comp\ H} = \dot{W}_H + \dot{m}_1(ex_2 - ex_1) \tag{6}$$

For condenser from high temperature-circuit, Eq. (7) to (9):

$$\dot{m}_3 = \dot{m}_2 \tag{7}$$
$$\dot{Q}_C = \dot{m}_3\,(h_3 - h_2) \tag{8}$$
$$\dot{E}_{d\ cond} = \dot{m}_3(ex_3 - ex_2) \tag{9}$$

For expansion valve from high temperature circuit, Eq. (10) to (12):

$$\dot{m}_4 = \dot{m}_3 \tag{10}$$
$$h_4 = h_3 \tag{11}$$
$$\dot{E}_{d\ exp\ H} = \dot{m}_4(ex_4 - ex_3) \tag{12}$$

For cascade condenser, Eq. (13) to (15)

$$\dot{m}_7 = \dot{m}_6 \tag{13}$$
$$\dot{m}_4 = \dot{m}_1 \tag{14}$$
$$\dot{Q}_{cas} = \dot{m}_4(h_1 - h_4) = \dot{m}_7(h_7 - h_6) \tag{15}$$
$$\dot{E}_{d\ cas} = \dot{m}_4(ex_4 - ex_1) - \dot{m}_7(ex_7 - ex_6) \tag{16}$$

For compressor from low temperature-circuit, Eq. (17) to (19):

$$\dot{m}_5 = \dot{m}_6 \tag{17}$$
$$\dot{W}_L = \dot{m}_5(h_{6s} - h_5) \tag{18}$$
$$\dot{E}_{d\ comp\ L} = \dot{W}_L + \dot{m}_5(ex_6 - ex_5) \tag{19}$$

For evaporator from low temperature-circuit, Eq. (20) to (23):

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$$\dot{m}_8 = \dot{m}_5 \tag{20}$$
$$\dot{Q}_E = \dot{m}_8\,(h_5 - h_8) \tag{21}$$
$$\dot{E}_{evap\,L} = -\left(1 - \left(\frac{T_0}{T_5}\right)\right) * \dot{Q}_E \tag{22}$$
$$\dot{E}_{d\,evap} = \dot{m}_8(ex_8 - ex_5) \tag{23}$$

For expansion valve from low temperature circuit, Eq. (24) to (26):

$$\dot{m}_7 = \dot{m}_8 \tag{24}$$
$$h_7 = h_8 \tag{25}$$
$$\dot{E}_{d\,\exp L} = \dot{m}_8(ex_8 - ex_7) \tag{26}$$

For all the system, Eq. (27) to (29)

$$COP_{cas} = \frac{\dot{Q}_E}{W_H + W_L} \tag{27}$$

$$\dot{E}_d = \dot{E}_{d\,comp\,\mathrm{L}} + \dot{E}_{d\,cas} + \dot{E}_{d\,evap} + \dot{E}_{d\,exp\,H} + \dot{E}_{d\,\exp L} + \dot{E}_{d\,cond} + \dot{E}_{d\,comp\,H} \tag{28}$$
$$\varepsilon = \frac{\dot{E}_{evap\,L}}{W_H + W_L} \tag{29}$$

where $\dot{m}$ is mass flow rate, $\dot{W}$ is work, h is specific enthalpy, $h_s$is specific enthalpy calculated at suction entropy, ex is exergy specific, $\dot{E}_d$ is exergy destruction, $\dot{Q}$ is heat transfer rate, COP is coefficient of performance and ε is exergetic efficiency.

3.2 Fluid Selection

The refrigerant fluids selection for thermodynamic modeling was performed by analysis of their physical properties, observing critical temperatures and pressures. In cascade refrigeration system, the evaporation temperature of the refrigerant cycle low temperature is -92°C and the condensation temperature was established as -30°C like a medium value, so that the difference of temperature and pressure was not so high, and it was not necessary a high compression ratio. Therefore, the average evaporation temperature of the refrigerant cycle high temperature is -30°C and condensation temperature, 35°C (environment temperature).

Thus, we sought fluids to work at those temperatures with pressures above atmospheric pressure so as to avoid pressure with negative values. Analyzing the values of pressure at the temperatures established (Tab. 1 and 2), it was discovered that ethylene is the refrigerant which best suits the conditions of low temperature cycle, and propylene and propane are best suited the conditions high-temperature cycle. However, the propane is a fluid refrigerant whose use is more common.

Therefore, we selected propane and ethylene as refrigerant fluids for cascade refrigeration system.

Table 1 – Physical properties of low temperature cycle refrigerant fluids

	CO_2	Ethylene	Methane	Ethane	Ethylene
Working temperature range	-50 °C a 30°C	-165°C a 5°C	-185°C a -85°C	-175°C a 30°C	-140°C a 90°C
Pressure at -92°C (MPa)	-	0,1928	3,6399	0,0939[1]	0,0060[1]
Pressure at -30°C (MPa)	1,4278	1,9366	-	4,6535	0,2122

(1) pressure values below atmospheric pressure
ASHRAE (2000)

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Table 2 – Physical properties of high temperature cycle refrigerant fluids

	Propylene	Propane	Isobutane	Butane
Working temperature range	-140°C a 90°C	-150°C a 95°C	-100°C a 130°C	-100°C a 150°C
Pressure at -30°C (MPa)	0,2122	0,1676	0,0461[1]	0,0282[1]
Pressure at 35°C (MPa)	1,5080	1,2471	0,4772	0,3388

(1) pressure values below atmospheric pressure
ASHRAE (2000)

3.3 Simulation

The simulation was done in the computational program EES (Engineering Equation Solver), where an estimated value of the thermal load to be removed by the cascade system such as 800 W, a value corresponding to a commercial freezer for ultra-low temperatures.
All refrigerant thermophysical properties were obtained from the EES, for several state points as shown in Fig. 1 and Fig. 2 and are directly calculated for the system analysis program.

The input parameters for the simulation of thermodynamic model of the system are:

-T_0: 35°C
-P_0: 101 kPa
-Evaporation temperature of low cycle: -92 °C
-Evaporation temperature of high cycle: -35°C
-Condensation temperature of the high temperature cycle: 41°C
-Condensation temperature of the low temperature cycle: -25°C

4. RESULTS

When performing the simulation with fluids propane-ethylene and propylene-ethylene, there were obtained the results that can be seen in Tab. 3.

Table 3 - Output values of the simulation in EES

	Propane-Ethylene	Propylene-Etylene
W_H (kW)	0,6807	0,6613
W_L (kW)	0,489	0,489
$\dot{Q}_C$ (kW)	-2,067	-2,025
$\dot{Q}_{cas}$ (kW)	-1,36	-1,36
$\dot{Q}_E$ (kW)	0,8484	0,8484
COP_{cas}	0,7253	0,7375
$\dot{E}_{d\ comp\ L}$ (kW)	1,361	1,323
$\dot{E}_{d\ comp\ H}$ (kW)	0,978	0,978
$\dot{E}_{d\ cond}$ (kW)	0,04292	0,04791
$\dot{E}_{d\ evap}$ (kW)	0,00001009	0,00001009
$\dot{E}_{d\ exp\ H}$ (kW)	0,2304	0,2129
$\dot{E}_{d\ \exp L}$ (kW)	0,1613	0,1613
$\dot{E}_{d\ cas}$ (kW)	0,1403	0,1083
$\dot{E}_d$ (kW)	2,607	2,913

ISSN 2176-5480

In examining this table, it can be seen that the results are similar for the two fluids. Thus, propane and ethylene were selected as default pair of fluids, due to number of compressors using propane which are available in the market, more than the number of compressors which use propylene.

For the purpose of benchmarking with the systems currently used and the possibilities using hydrocarbons as refrigerants, we performed a simulation for the same values of system capacity using the following pairs:

- R404A x R508B
- Propane x Ethylene
- Propylene x Ethane
- Isobutane x Ethane

In Fig. 3, it is possible to check the comparison between the alternative selected of hydrocarbons (propane and ethylene) and HFC refrigerants (R404A X R508B). Hydrocarbons have a better relationship between the refrigeration capacity and the work consumed by compressors. This improved efficiency is increased with the need for greater refrigeration power.

In Fig. 4, we can evaluate other options of pairs of hydrocarbons used in cascade. In this graphic, we see that the difference in power required for small refrigeration capacity is small. From approximately 800W capacity required, the pair of fluids propane / ethylene begins to show a better relationship between the values of refrigeration capacity and the work consumed by compressors.

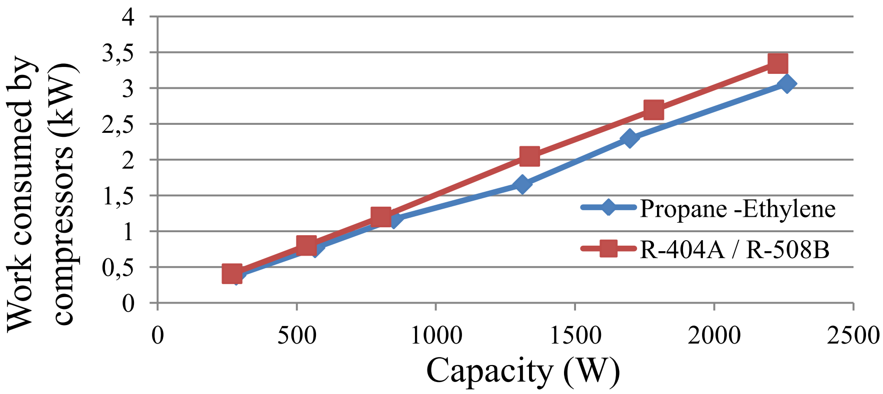

Figure 3 - Relationship between the work consumed by the compressor and the capacity of the system to the refrigerants propane/ethylene and R404A/R508B

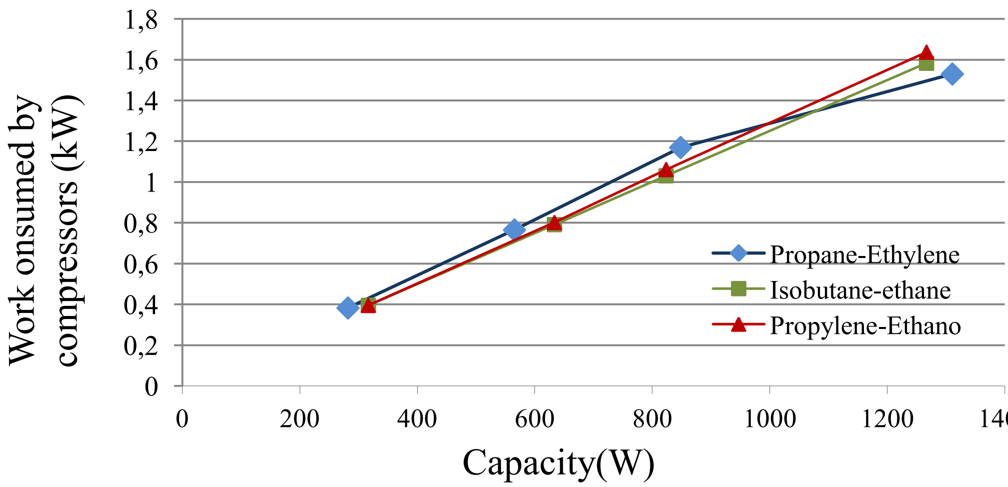

Figure 4 - Relationship between the work consumed by the compressor cycles and the capacity of the system to the ethylene/propane, isobutane/ethane and propylene/ethane

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In Fig. 5, the pair of fluid presents the best coefficient of performance is isobutane/ethane, while R404A/R508B has the worst coefficient of performance among the fluids.

In Fig. 6, we can see that the pairs of refrigerants exhibit behavior similar to that shown in the previous figure, where the pair which features improved exergy efficiency is the isobutane / ethane and what is the worst R-404A/R-508B.

In Fig. 7, it is found that components have a higher irreversibility are compressors from high and low temperature, as the evaporator provides a very low irreversibility. The pair of refrigerants that have a lower irreversibility in their components are isobutane/ethane, followed by propylene/ethane.

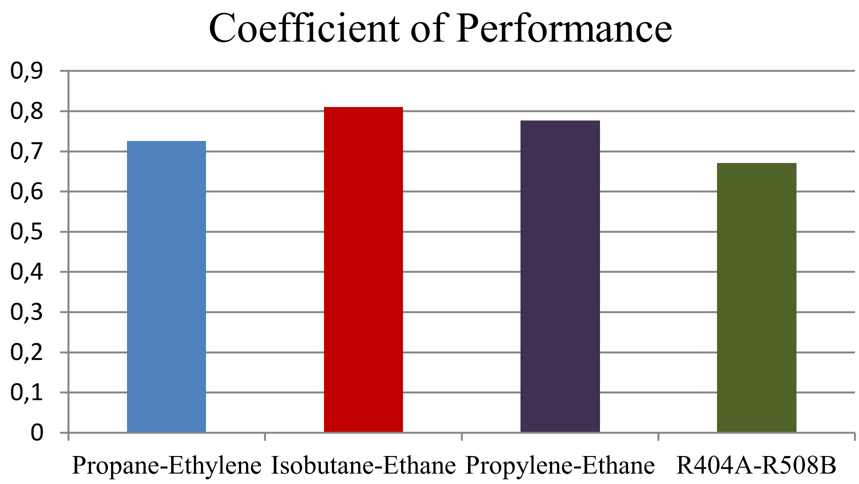

Figure 5 - Relation between the COP of the system and the refrigerant pair used

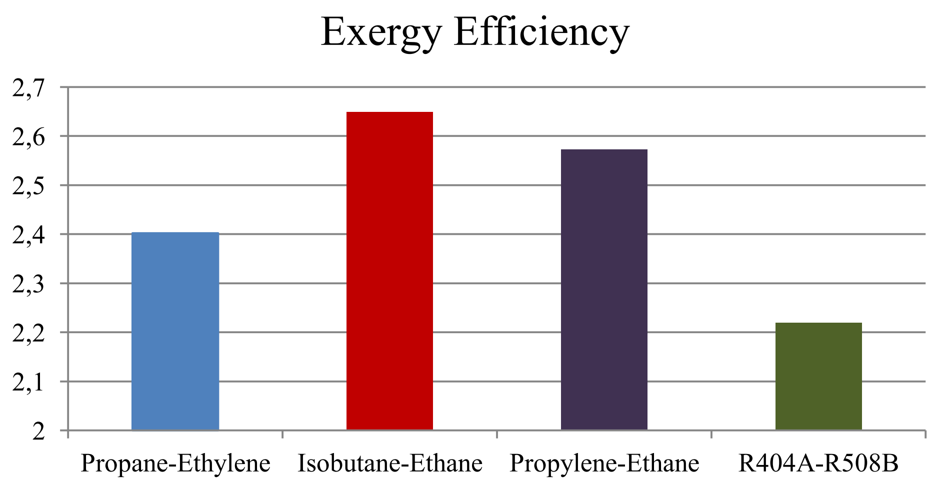

Figure 6 - Relation between the Exergy Efficiency of the system and the refrigerant pair used

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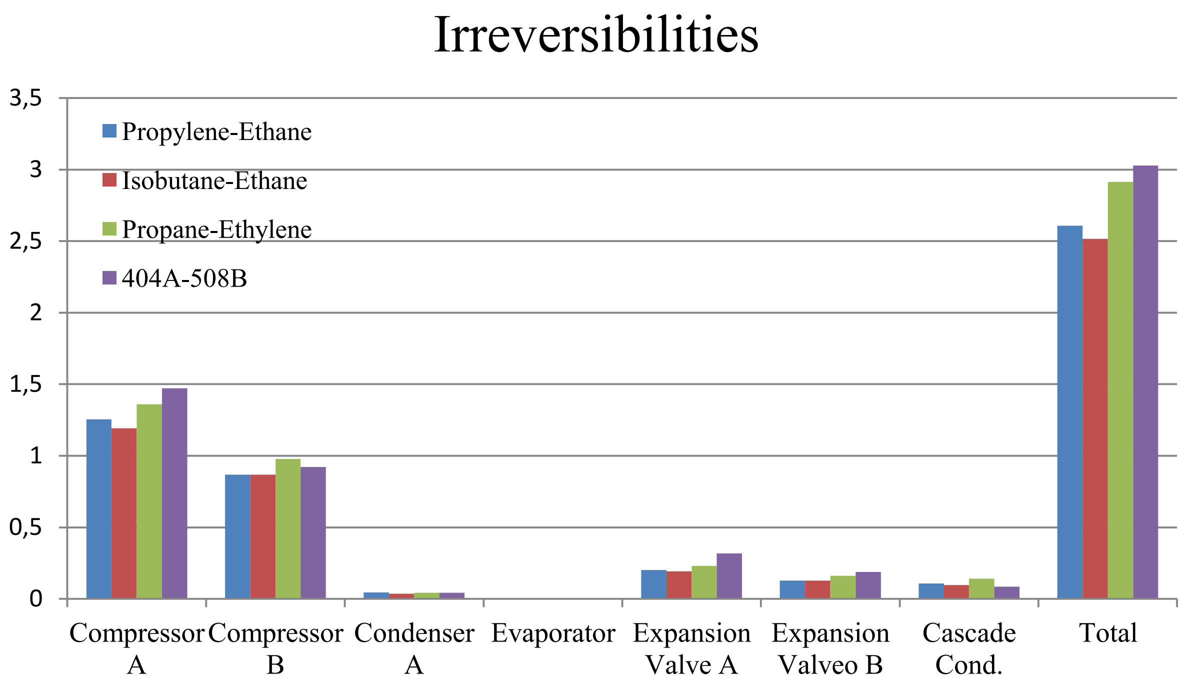

Figure 7 - Relationship between the irreversibility of each component and pair of refrigerant used

5. CONCLUSION

After thermodynamic modeling, simulation and choice of fluids, it could be observed the results and concluded that the cascade refrigeration system using hydrocarbons is feasible from the point of view of energy and exergetic.

Through this study it was possible to quantify the heat that is absorbed or rejected in each heat exchanger and work consumed by each compressor system. With the exergy analysis, it is possible to quantify the exergy destroyed in each component unit. Thus, one could identify the points where there is greater irreversibility, such as compressors from high and low temperature and verify the need to evaluate these components to reduce these losses.

In relation to the pair of fluids presently used in cascade systems (R-404A/R-508B), all hydrocarbons have better performance, both energy and exergy point of view. Among the hydrocarbons, it was found that the pair which included the fluid ethane as refrigerant from low temperature cycle showed a better performance and lower irreversibility components. However, this fluid presents suction pressure value below atmospheric pressure when working at extremely low temperatures. Thus, for safety, the pair propane / ethylene shows up as the most suitable for the application.

The next step is to carry out work on the dimensioning of the components of the values obtained through the simulation, followed by the assembly of a prototype refrigeration system using refrigerants selected.

Thus, the study reveals that the refrigerants propane and ethylene have perform well in a cascade system for low temperatures, showing up as a good alternative to HFCs currently used fluids.

6. ACKNOWLEDGEMENTS

We thank Instituto Federal de Educação, Ciência e Tecnologia da Bahia (IFBA) and Conselho Nacional de Desenvolvimento Científico e Tecnológico (CNPq) for supporting the development of research.

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