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THERMODYNAMICS ANALYSIS OF COGERATION PLANT WHIT INTERNE COMBUSTION ENGINE AND ABSORPTION REFRIGERATION SYSTEM

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Abstract. *In this work will be performed thermodynamic modeling and energy analysis of a cogeneration system that uses an internal combustion engine that operates on the diesel cycle and provides the resultant energy of the exhaust gases to a simple effect absorption refrigeration system using the pair of water-lithium bromide to generate cold. The cogeneration system under study, which can be employed in many industrial and commercial applications, comprises an internal combustion engine, an intermediate heat exchanger (which recovers energy from exhaust gases from the burning fuel) and cooling system simple absorption effect. These equipments are analyzed using the laws of thermodynamics. The system resulting equations of thermodynamics modeling is solved using computational platform EES (Engineering Equation Solver). The results have been evaluated by means of tables generated by ESS software, identifying the best operating conditions of the cogeneration system. The system may be implemented in industrial plants and commercial applications, thus contributing to energy reuse of non-renewable resources and sustainable development.*

Keywords: *Cogeration, Internal Combustion Engine, Refrigeration*

1. INTRODUCTION

In the current global energy scenario, the efficiency of thermal systems has been a major source of discussion on the preservation of natural resources and the reduction of energy costs of industrial processes involved in these systems. With the increasing scarcity of natural resources and environmental problems arising from the power generation, there is a greater pursuit of rational use of energy and a deeper understanding in studies involving cogeneration systems. According to Guimarães (2011), the various applications for refrigeration systems make their essential use. Electricity is the main source for the activation of these systems. However, there are other technologies to generate cold at thermal sources, not just electricity. Some of these heat sources for power generation are exhaust heat from heat engines, solar energy, among others.

In recent years, the large increase in the use of internal combustion engines and continuing increases in energy costs have recommended the emergence of new technologies and energy efficient equipment in the cogeneration systems. Under this new perspective, the small-scale cogeneration comprising an internal combustion engine and an absorption chiller (using as an example the pair water-lithium bromide) has been studied by several national and international researchers (Santos, 2005).

According to Brunetti (2012), the thermal machines are devices that let you transform heat into work. This heat can be obtained from different sources: combustion, electricity, etc. Mechanical work is achieved by a sequence of processes performed in a substance called active fluid, which in this case is formed by the air-fuel mixture in the volume control input and the output of combustion products. Therefore, in internal combustion engines this fluid directly participate combustion.

The combustion process in diesel engines occurs at longer time intervals as compared to engines operating according to the Otto cycle. Because of this long duration, the ideal engine combustion process operating according to the diesel cycle is approached, with a heat supply process at constant pressure. The Internal Combustion Engine operating according to the standard diesel cycle air, or watching the respective theoretical cycle, place four reversible processes. Analyzing P-v diagram, it is observed that the process 1-2 is an isentropic compression process; 2-3 represents the process to provide heat at constant pressure; the process 3-4 is the isentropic expansion; and the process 4-1 heat rejection at constant volume (Shapiro and Moran, 2006).

Therefore, four processes are internally reversible:
1-2 isentropic compression;
2-3 heat supply at constant pressure;
3-4 isentropic expansion;

4-1 heat rejection at constant volume.

The Fig. 1 below shows the P-v and T-s diagrams that standard Diesel cycle the air.

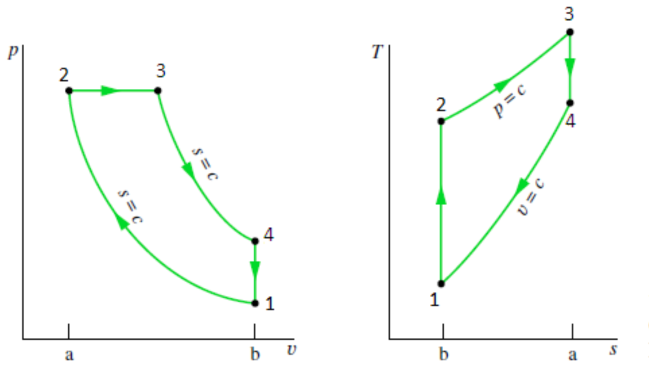

Figure 1. Diagram P-v and T-s standard diesel cycle air.

The refrigeration system for absorption or systems powered heat has presented as technologically and energetically viable solution, gaining market share reserved for large cooling capacities equipment, with economic viability with long-term attractive return. According to Moreira (2004), a major advantage of the absorption cycle is the cost of pumping a liquid instead of steam, a low-pressure region to a high pressure. Because of a liquid pumping, the work is considerably smaller, and there is the use of thermal energy at the expense of electricity use, as in the vapor compression cooling systems. Soon, these absorption refrigeration systems are considered when the cost of thermal energy is low and is planned to remain low relative to the cost of electricity.

Below in Fig. 2, the detailed process of the single effect absorption refrigeration system using the pair of water-lithium bromide.

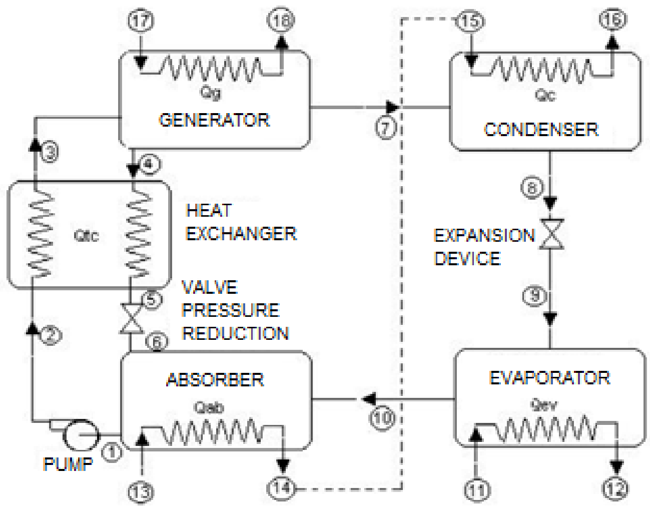

Figure 2. Detailed process of the refrigeration system by simple absorption effect.

The process detailed for each component of the absorption refrigeration system according to the items below.

a) Process in the absorber: the concentrated solution of lithium bromide-water originating from the generator is continuously sprayed onto the absorber tubes in which circulate the cooling water (stream 13-14). The solution absorbs water vapor from the evaporator (stream 10) occurring simultaneously diluting the solution and release heat because it is an exothermic reaction. The cooling water (stream 13-14) dissipates the resulting heat.

b) Process in the regenerator (heat exchanger): the arising dilute solution from the absorber is pumped, increasing its pressure (flow 2). When passing through the regenerator is warmed by the concentrated solution to a temperature higher than the generator returns (flow 4), and there is a preheating of the diluted solution before entering the generator (stream 3). The use of this intermediate heat exchanger (regenerator) improves the cycle performance and requires less amount of heat from the hot source.

c) Process in the generator: the diluted solution (flow 3) from the absorber and heat exchanged in the regenerator is heated by heat exchange with the flow of exhaust gases that circulate inside the tubes, coming from the internal combustion engine operating on the Diesel cycle. In this heat exchange of the water (coolant) evaporating the solution (stream 7), making the solution concentrated again. The more concentrated solution returned to the absorber by gravity, passing again through the heat exchanger (stream 4).

d) Process in the evaporator: the refrigerant from the expansion valve after the throttle (stream 9) is collected inside the evaporator, and exchanges heat with cooling water flowing in the pipes of the same component beams (stream 11-12). In this heat exchange, the refrigerant evaporates due to heat supplied by the water that flows within the evaporator.

e) Process in the condenser: the refrigerant vapor from the generator (stream 7) is cooled by the cooling water inside the condenser tube bundles (stream 15-16). By gravity, the condensed water in the state of saturated liquid (refrigerant) is directed into the evaporator (stream 9) occurring in the path, the throttling pressure to the same pressure through the expansion valve (flow 8).

2. DESCRIPTION OF THE PROBLEM

The purpose of this work is to develop a thermodynamic analysis of a cogeneration system that utilizes a small-scale internal combustion engine operating according to the diesel cycle producing useful mechanical power and as an energy source for generating cold by the refrigeration cycle absorption of simple effect using the pair of water-lithium bromide. A computer program will be used for this purpose. The relevance of this work to those already published, is to compare the data obtained by computer simulation with those identified as standards recommended by the manufacturers of the cycle devices.

3. MODELING

Modeling the cogeneration plant that can be applied in various industrial and commercial applications is presented here. The mathematical model is based on mass conservation equations, the species and the energy to volume control on a permanent basis, with constant properties. Such equations are as follows:

- Conservation of mass to volume control on a permanent basis:

$$\sum_{in} \dot{m_{in}} - \sum_{out} \dot{m_{out}} = 0 \tag{1}$$

- Conservation of the energy for volume control in continuous operation:

$$\dot{Q}_{vc} - \dot{W}_{vc} + \sum_{in} \dot{m_{in}} \left(h_{in} + \frac{v_{in}^2}{2} + gz_{in}\right) - \sum_{out} \dot{m_{out}} \left(h_{out} + \frac{v_{out}^2}{2} + gz_{out}\right) = 0 \tag{2}$$

3.1 Internal Combustion Engine

For the cogeration system was chosen an internal combustion engine that operates on the ideal Diesel cycle and having a theoretical capacity due to its rotation 300 kW. According to the literature it was selected an engine from Doosan brand, P126TI-II model 300 KVA/300 kW (Santos, 2005). The theoretical cycle diesel engine that is composed of four processes: 13-14 isentropic compression process, process 14-15 by addition of heat at constant pressure, process 15-16 and 16-13 isentropic expansion and heat removal process volume constant.

The conditions of point 19 are the conditions of atmospheric air. Therefore, knowing the temperature and pressure point, and assuming that the process (19-20) is adiabatic and reversible (isentropic) can determine the temperature and the pressure point 20 using Eq. (3) and (4), respectively:

$$\frac{T_{20}}{T_{19}} = r_c^{(k-1)} \qquad \frac{P_{20}}{P_{19}} = r_c^k \qquad \text{(3) and (4)}$$

where r is the compression ratio is k and the ratio of specific heats (c_p/c_v).

The combustion process is simulated by supplying heat at constant pressure, represented by process (20-21). An energy balance applied to the motor determines the exhaust temperature at the point 21, as will be shown later. As the process (20-21) occurs at constant pressure, the pressure at the point 21 is equal to the pressure point 20. Given the temperature and pressure values in point 21, you can determine the temperature and pressure at the point 22 seconds the equations below:

$$\frac{T_{21}}{T_{22}} = r_c^{(k-1)} \qquad \frac{P_{22}}{P_{19}} = \frac{T_{22}}{T_{19}} \qquad \text{(5) and (6)}$$

The exhaust temperature (point 21) is determined through an energy balance applied to the combustion engine. The energy balance for reagent systems in continuous operation can be seen admitting the hypothesis that the air-fuel mixture and the combustion products are considered ideal gases. Neglecting the kinetic and potential energy changes, we have,

$$q_{vc} - w_{vc} + \sum_R n_{in} \left(\bar{h}^\circ{}_f + \overline{\Delta h}\right)_{in} - \sum_P n_{out} \left(\bar{h}^\circ{}_f + \overline{\Delta h}\right)_{out} = 0 \tag{7}$$

As Çengel and Boles (2011) considered the calorific value of fuel equal to the absolute value of the enthalpy of combustion given in the above equation, we have the lower PCI is such combustion reaction enthalpy. Thus, the rate of heat released on burning of the diesel is given by:

$$\dot{Q}_{combustion} = P_{engine} + \dot{m}_{fuel} \cdot PCI_{diesel} \tag{8}$$

Whereas diesel liquid enters the combustion chamber 27 °C where it is mixed and burned with 40% excess air.

$$C_{12}H_{26} + 25{,}9 \cdot (O_2 + 3{,}76N_2) \longrightarrow 12CO_2 + 13H_2O + 7{,}4O_2 + 97{,}38N_2 \tag{9}$$

Heywood (1988) references the specific fuel consumption of internal combustion engines with compression ignition. That way you can get the mass flow of the fuel inlet by the following Eq (10):

$$sfc = \frac{\dot{m}_{fuel}}{P_{engine}} \cdot 3{,}6 * 10^6 \tag{10}$$

The air-fuel ratio is determined by the following equation:

$$AC = \overline{AC}\frac{M_{ar}}{M_{combustivel}} \tag{11}$$

where $\overline{AC}$ is the air-fuel ratio on a molar basis $\left(\overline{AC} = \frac{n_{ar}}{n_{combustivel}}\right)$.

The mass flow of exhaust gas (combustion released) in one cycle is calculated through the air-fuel mixture introduced into the engine to mass assuming complete combustion of the mass. Therefore, the conservation of mass, the mass flow of gases can be estimated by the following expression.

$$\dot{m}_{gases} = \left(\frac{\pi}{4}\right) \cdot 0{,}80 \cdot D^2_{cylinder} \cdot L_{piston} \cdot n_{cylinder} \cdot \rho_{mixture} \cdot N \tag{12}$$

where D_{int} is the is the internal diameter of the cylinder, L_{piston} is the offset (course) of the piston, $n_{cylinder}$ is the number of engine cylinders, N is the engine speed and $\rho_{mixture}$ is the density of the mixture is calculated according to the air-fuel ratio as follows:

$$\rho_{mixture} = \frac{AC+1}{\frac{AC}{\rho_{air}} + \frac{1}{\rho_{fuel}}} \tag{13}$$

Thus, the specific heat provided by the fuel for two rotations of the crankshaft is given by:

$$q_{in} = \dot{Q}_{combustion}/\dot{m}_{gases} \tag{14}$$

The combustion process is simulated by supplying heat at constant pressure, represented by process (20-21). The exhaust temperature at the point 21 is determined by the specific heat entering the cycle for the combustion reaction. By Eq. 15 below it can be determined that temperature.

$$q_{in} = c_p \cdot (T_{21} - T_{20}) \tag{15}$$

After certain thermodynamic properties of all parts of the Diesel cycle, it was held the energy balance in the engine. In the final process, heat rejection at constant volume (process 22-19), it was determined that heat is rejected from the cycle by mass as follows:

$$q_{out} = u_{22} - u_{19} \tag{16}$$

For heat out of the system, it was found that 60% comes from combustion gases that will be used in a cogeration process according Pulkrabek. Thus, we have:

$$q_{out;exhaust} = 0{,}6 \cdot q_{out} \tag{17}$$

Therefore, the net work per unit mass is the difference between the energies provided to the diesel cycle and discarded. Thus:

$$w_{net} = \Delta q = q_{in} - q_{out} \tag{18}$$

The thermal efficiency of the Diesel cycle was obtained by the ratio between the net work per unit mass and the specific heat that enters the cycle. Thus, we have:

$$\eta_{t;Diesel} = \frac{w_{net}}{q_{in}} \cdot 100 \tag{19}$$

3.2 Refrigeration System for Absorption (LiBr-H_2O)

The absorption refrigeration system uses a pair of water-lithium bromide as working fluid. This thermodynamic system is subdivided into control volumes and each system component, an energy analysis is performed using Eq. 1 and 2. Some design parameters were established from the available heat generator (heat from exhaust gases released Diesel engine combustion chamber), the concentrations of lithium bromide solution, the condensation temperature and evaporation temperature of the water and the chilled water passing through the evaporator. This analysis is based on the application of the mass conservation equations and energy.

Some simplifying assumptions applied to the volume control were made for the purpose of calculations of absorbed or rejected heats the refrigeration cycle:

01 - The kinetic and potential energy changes are negligible.
02 - The system operates on a permanent basis.
03 - The components are thermally insulated, i.e., no heat exchange with the environment.
04 - The expansion valves of the cycle have isenthalpic flow.
05 - The recirculation pump of the lithium bromide-water solution accomplishes isentropic work.

The Fig. 2 presented above, shows the points of single effect absorption refrigeration system that uses the pair of water-lithium bromide.

Moreira (2004) carried out the modeling applying the first law of thermodynamics with its simplifying assumptions for each component of the absorption refrigeration cycle, resulting in a set of governing equations, which were resolved through ESS platform (Engineering Equation Solver) allowing calculate the heat released or absorbed components. For refrigeration unit for absorption in question, the analysis is made for each control volume highlighting: absorber, vapor generator, condenser, evaporator, expansion valves, solution circulation pump and heat exchanger.

The energy efficiency of the absorption refrigeration cycle is the ratio of energy removed from the chilled water from the evaporator system and energy from the diesel engine combustion gases.

$$\eta_{absorption} = \frac{\dot{m}_{11}(h_{11}-h_{12})}{\dot{m}_{17}(h_{17}-h_{18})} \,.\, 100 \tag{20}$$

The coefficient of performance of the absorption refrigeration cycle (COP) is defined by the ratio desired output (quantity of heat removed or cold generated by the evaporator) and the required input (generator heat supplied from the exhaust gas).

$$COP_{absorption} = \frac{\dot{Q}_{evaporator}}{\dot{Q}_{generator}} \tag{21}$$

3.3 Plant of Cogeration

The cogeneration plant is evaluated in all its aspects through the overall efficiency and the first law of thermodynamics. This energy efficiency is expressed by the ratio of the unit's products (engine power and the evaporator heat transfer rate) and the raw material for the production (combustion heat rate).

$$\eta_{cogeration} = \frac{\dot{Q}_{evaporator}+\dot{W}_{engine}}{\dot{Q}_{combustion}} \,.\, 100 \tag{22}$$

4. RESULTS

The results are presented primarily for the internal combustion engine then analyzes the absorption refrigeration system. Finally, an analysis was made of cogeration plant.

4.1 Internal Combustion Engine

Initially, the input parameters necessary for the computer simulation governing thermodynamic engine model were defined. The input data are listed in Tab. 1 below.

Table 1. Input data for the simulation engine.

Model Engine = Diesel Doosan P126TI-II 300kW (Santos, 2005)
P_{engine} = 300 kW (theoretical power as a function of engine speed)
$n_{cylinder}$ = 6 (number of cylinders)
r_c = 16 (compression ratio)
$D_{cylinder}$ = 0,15 m (cylinder diameter)
L_{piston} = 0,20 m (piston stroke)
N = 1500 rpm (rotational speed of the crankshaft)
ρ_{air} = 1,161 kg/m^3 (density of atmospheric air)
ρ_{fuel} = 853 kg/m^3 (specific fuel weight)
PCI_{diesel} = 45200 kJ/kg (lower calorific value of diesel)
sfc = 85 g/kWh (specific fuel consumption)
C_p = 1,005 kJ/kg.K (specific heat at constant pressure)
C_v = 0,7177 kJ/kg.K (specific heat at constant volume)
k = 1,4 (ratio of specific heats)

From the initial data of ambient temperature, atmospheric pressure, the pressure specific heat constant and volume compression ratio adopted in the design and the specific heat entering the loop, it was possible to determine all the thermodynamic properties of the four points of the theoretical Diesel cycle. The input parameters needed for computer simulation are arranged in the Tab. 2 below.

Table 2. Thermodynamic properties of the points of the Diesel cycle.

Point	P [kPa]	T [C]	T [K]	h [kJ/kg]	s [kJ/kg-K]	u [kJ/kg]	v [m³/kg]
19	100	27	300,2	300,6	5,706	214,4	0,8615
20	4849	636,6	909,7	944,1	5,75	683	0,05384
21	4849	1833	2107	2386	6,753	1781	0,1247
22	231,6	421,9	695,1	708,3	6,327	508,8	0,8615

Then in Tab. 3 those found values that characterize the thermodynamic analysis of the internal combustion engine through the energy balance are shown. These values were obtained by computer code EES.

Table 3. Quantities obtained by the motor energy balance.

$Q_{combustion}$ = 620,2 kW (heat rate released from burning)
m_{fuel} = 0,007083 kg/s (mass flow rate of fuel input)
m_{gases} = 0,5157 kg/s (mass flow rate of exhaust gases for a complete cycle)
AC = 20,99 (air-fuel ratio)
q_{in} = 1203,0 kJ/kg (specific heat supplied to the fuel cycle)
q_{out} = 294,3 kJ/kg (specific heat rejected by the cycle)
$q_{out,exhaust}$ = 176,6 kJ/kg (specific heat rejected from the gases)
w_{net} = 908,2 kJ/kg (specific net work obtained by the cycle)
$\eta_{t,diesel}$ = 75,52% (energy efficiency of the Diesel cycle)

4.2 Refrigeration System for Absorption (LiBr-H_2O)

The starting point for computational thermodynamic modeling of the absorption refrigeration cycle is supplied to the heat generator obtained from the exhaust gases of the internal combustion engine. This input parameter was considered the most important for the simulation. In Tab.4 presents the results of the simulation of simple effect absorption refrigeration system using the pair of water-lithium bromide. The results of all the points shown in Fig. 2 are shown in the table below. The temperature of the exhaust gas reaches the value of 421.9 °C, which is more than sufficient to provide the energy required for the generator of the system above. The values of temperature, pressure, concentration of solution, mass flow rate, enthalpy and entropy related to the 18 points scored by the computer code are represented.

Table 4. Thermodynamic properties of the 18 points of the absorption refrigeration cycle.

Point	State Thermodynamic	P [kPa]	T [ºC]	h [kJ/kg]	s [kJ/kg-K]	$\dot{m}$ [kg/s]	x_i
1	Diluted solution - saturated liquid	0,8726	34,27	82,44	0,2087	0,216	0,549
2	Diluted solution - compressed liquid	6,28	34,27	82,44	0,2087	0,216	0,549
3	Diluted solution - compressed liquid	6,28	70,86	157,7	0,4399	0,216	0,549
4	Concentrated solution - saturated liquid	6,28	90,64	236,1	0,483	0,1853	0,64
5	Concentrated solution - saturated liquid	6,28	54,06	170	0,2913	0,1853	0,64
6	Concentrated solution - mixing biphasic	0,8726	54,06	170	0,2913	0,1853	0,64
7	Coolant (water) - superheated water vapor	6,28	80	2649	8,558	0,03072	-
8	Refrigerant - saturated liquid	6,28	37	155	0,532	0,03072	-
9	Refrigerant - mix biphasic	0,8726	5	155	0,07626	0,03072	-
10	Refrigerant - saturated steam	0,8726	5	2510	9,024	0,03072	-
11	Cold water - compressed liquid	100	12	50,23	0,1804	3,456	-
12	Cold water - compressed liquid	100	7	29,3	0,1063	3,456	-
13	Cooling Water - compressed liquid	100	29,5	123,5	0,4296	7,272	-
14	Cooling Water - compressed liquid	100	32,48	136	0,4706	7,272	-
15	Cooling Water - compressed liquid	100	32,48	136	0,4706	7,272	-
16	Cooling Water - compressed liquid	100	35	146,5	0,5049	7,272	-
17	Combustion products - air at atmospheric pressure	100	421,9	1766	8,61	0,5157	-
18	Combustion products - air at atmospheric pressure	100	321,9	1347	8,291	0,5157	-

In Tab.5 below are shown the found values that characterize the thermodynamic analysis of absorption refrigeration cycle through energy balance. These values were obtained by computer code EES.

Table 5. Quantities obtained from the energy balance in Refrigeration cycle.

$Q_{absorber}$ = 90,79 kW (heat rate obtained in the absorber)
$Q_{condenser}$ = 76,63 kW (heat rate obtained in the condenser)
$Q_{evaporator}$ = 72,34 kW (heat rate obtained in the evaporator)
$Q_{evaporator,TR}$ = 20,57 TR (heat rate obtained in the evaporator in TR's)
$Q_{regenerator}$ = 4,006 kW (heat rate obtained in the intermediate heat exchanger)
$\eta_{t,absorption}$ = 33,51% (energy efficiency of the absorption refrigeration cycle)
$COP_{absorption}$ = 0,7943(coefficient of performance of the refrigeration cycle)

The comparison parameters for the single effect system were removed from Moreira work (2004) in which is made an energetic analysis for the same system under study.

According (Moreira, 2004), thermodynamic modeling of energy analysis allows to vary some relevant parameters such as the condensation temperature of the refrigerant (water vapor) relative to the coefficient of performance of the absorption refrigeration cycle (COP). This situation is shown in Fig.3 below where condensation with increasing temperature there is a significant decrease in the COP of the cycle. This decrease in COP can be explained by the need for increased fuel burning in generator to increase the saturation pressure of the solution and consequently the steam saturation temperature of water.

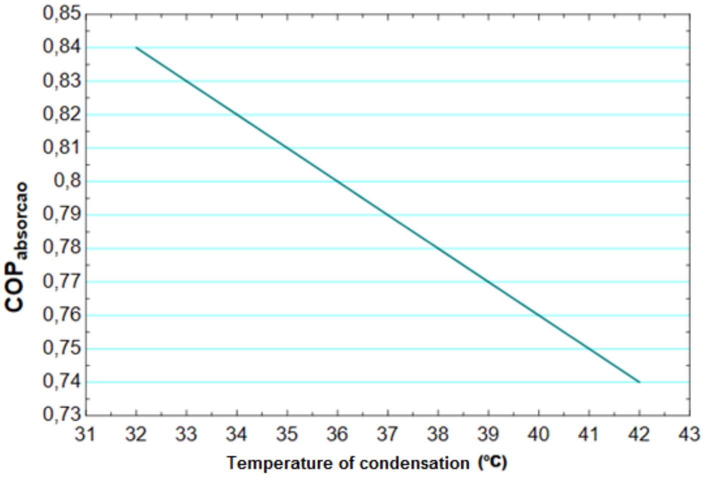

Figure 3. Graphic temperature condensation T_8 versus COP cycle.

The condensation temperature variation for this simulation was performed in the range of 30 to 45 ºC, which is the range suggested by Moreira (2004).

4.3 Plant of Cogeration

An important point of entry into the simulation of the cogeneration system is the engine power according to their rotation. According to the Eq. 23, was obtained through the energy balance for low adopted plant cogeneration system. The value obtained by computational code was EES: $\eta_{cogeration}$ = 60,04% (energy efficiency of plant of cogeration).

5. CONCLUSION

A cogeneration system composed for an internal combustion engine and a cooling system by simple absorption effect (LiBr/H_2O) was simulated in the EES software (Engineering Equation Solver) using diesel as fuel. From the results, it can be concluded that the fuel has met the energy requirements to power the generator of the absorption system, reaching a temperature of 421.9 ºC for diesel at the output of the engine exhaust gases. However, it must be analyzed using thermoeconomics studies is compensated to obtain a greater cooling capacity observed in the refrigeration cycle the evaporator through absorption and hence a better COP at the cost of having a larger amount of burnt fuel.

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