EVALUATION OF A COMPACT SYSTEM OF TRIGENERATION THROUGH ENERGY AND EXERGY ANALYSIS

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Abstract. First and second laws of thermodynamics are employed to analyze the quantity and quality of energy in a small compact trigeneration system. This combined cycle is composed of a little reciprocating internal combustion engine model GM, 1.0 CORSA, using liquefied petroleum gas as fuel, two heat exchangers and an absorption chiller utilizing water–ammonia as working fluid mixture. The mass and energy balance equations of the engine and subsystems are reviewed in detail. Exergy of each involved stream is calculated and the exergetic balance of each subsystem is presented, as well as the global system, identifying where and why losses and irreversibilities occurs. Efficiencies based on the second law of thermodynamics are calculated for each subsystem and compared. Special attention is given to identification and quantification of second law efficiencies and the irreversibilities of all subsystems. The determination of the irreversibilities in each subsystem is particularly important since they are not identified in traditional first law analysis. Furthermore, this study revealed that the combustion was the most important contributor to the system inefficiency representing 36.0% of the total exergy input and 73% of the total exergy destruction. The exergetic efficiency of the trieneration system is determined to be 51.19%.

Keywords: trieneration, energetic, exergetic, internal combustion engine, efficiencies.

1. INTRODUCTION

Cogeneration or Combined Heating and Power (CHP) system may be defined as the simultaneous production of electrical or mechanical energy and useful thermal or cooling energy from a single energy source, such as oil, coal, natural or liquefied gas, biomass, or solar (Silveira e*t al.,* 2012). The main objective of the CHP is that most of the energy contained in the fuel is used, instead of only a small part. Thus, a more economical method is obtained compared to the systems where electricity and heat are separately produced. The efficiency of energy production can be increased from current levels that vary from 35% to 55% in the conventional power plants to over 80% in the CHP systems (DOE, 2000). Trigeneration or Combined Cooling Heating and Power (CCHP), is a new trend of cogeneration systems, where in addition to producing power and heat, as traditionally, cold water is also produced.

The standard approach used for these studies is usually based on the first law of thermodynamics, analyzing all forms of energy equal and neglecting the quality of the energy. In this way, the overall performance of a system can be calculated using mixed indexes, which, in spite of being easy to calculate, have the drawback of mixing energy streams that are different in quality and characteristics. The second law of thermodynamics deals with the quality of the different forms of energy and studying the causes, which led to its degradation. Kotas (1985), defines power quality as its ability to cause changes. More specifically, the second law is interested in energy degradation over a process, the entropy generation, and missed opportunities to produce work.

Due to the connection between the first and second law of thermodynamics, the exergetic analysis heads to a more suitable study of the CHP systems, as it allows each energy stream quality to be determined, and simultaneously, the type, location and magnitude of irreversibilities to be individuated (Torres and Gallo, 1998). Additionally, this kind of analysis can be precise to comprehend the way to reduce irreversibilities, and improve energetic performance with the help of the operating parameters optimization (Balli and Hepbasli, 2013; Ramos da Costa, *et al.,* 2012). Exergetic analysis has indeed been used by several authors as an instrument of new energy plants thermal purpose and has shown to be a very powerful tool of thermodynamic systems optimization.

The use of LPG as alternative fuel to gasoline is common practice in spark ignition engines. While the main driving force to the use of LPG remains the low cost, usually less expensive than gasoline, for the end user, and its favorable pollutant emissions, it can produce significantly lower amounts of some harmful emissions, in particular the greenhouse gas carbon dioxide. These reasons will probably increase interest in LPG as an IC engine fuel. Although when LPG fuel is used with new generation conversion systems in spark ignition engines, a little reduction in power output of engine occurs. The reason for this reduction is the decrease in the volumetric efficiency of the engine as the result of using LPG, which expands 230–267 times while passing to the gas phase from liquid phase. When using LPG in gas state in spark ignition engines, the volumetric efficiency is more explicitly less than those when using gasoline (Karamangil, 2007).

Based on our search in the open literature, in this study, the first and second Laws of thermodynamics are employed to analyze the quantity and quality of energy in a four-cylinder, with MPFI – Delphi system injection spark engine using liquefied petroleum gas as fuel. The experimental data were collected using steady-state tests which enable accurate measurements of air, fuel and cooling and hot water flow rates, engine load, and all the relevant temperatures.

2. EXPERIMENTAL PROCEDURE

The compact trigeneration system, see Figure 1, used in this work is a "stand alone" system, not connected to the grid. This is especially useful in countries like Brazil, where there is not the possibility to connect the whole territory to the national grid, due to its vastness. This system is located in the Energetic Optimization Systems Laboratory (LOSE) in State University of Sao Paulo and it is constituted by a little internal combustion engine model GM, 1.0 CORSA, see Table 1. Manufacture date "1998", four strokes, with MPFI-Delphi injection system, associated with an electrical generator and two heat exchangers, one heat exchanger makes the function of radiator, is coupled at the engine system cooling, from which the water thermal energy is used for hot water production. The exhaust gases are directed into a second heat exchanger in which their temperature is reduced to a suitable level to run the absorption chiller with a capacity of 5TR, mark ROBUR. The engine study has a "Rodogás" fuel feeding system (pressure reducer) which allows the operation both on GLP or GNV and even on gasoline. Rodogás is an instrument, model TE-01, with capacity for up to 43 m^3/h of fuel, this system serves for engines up to 120 HP. A generator is coupled to the engine through a pulley and belt which receives the mechanical energy produced by the combustion engine. The electric generator (alternator three phase), works at a frequency of 60 kHz, has 4 poles, power of 10 kW-12.5 kW with a current at 32.8A and 220 V, rated values. Before going to the second heat exchanger the gases pass through a catalyst in order to reduce pollution emission. The heat expelled in the reducing temperature process is used for hot water production, in addition to the produced in the engine cooling system (jacket water system). The absorption system works with a water–ammonia mixture and it produces cold water, which can be used in human consumption water machines or fan coils. The electrical power from the generator is dissipated in the resistive load banks. Pressure, temperature and flow gauges are installed in different points of the system, to enable the necessary calculations. The data captured by acquisition system, see Table 2, are sent to a computer through an AQDADOS system, where electrical signals are converted into temperature, pressure and flow measurements, calculations of entropy and enthalpy of each point were performed with the Engineering Equation Solver (EES) software.

Figure 1. Compact small trigeneration system

Table 1. The technical specifications of the GM, 1.0 CORSA engine

Engine Manufacturer	Opel
Coolant Type	Water
Valves per Cylinder	4
Engine Displacement	998 cc
Bore	73,4 mm
Stroke	78,6 mm
Compression Ratio	10,5 : 1
Fuel System Type	Motronic MPFi
Aspiration Type	Normal
Maximum Engine Power	59 HP (44 kW)
Engine Torque at Max. Engine Power rpm	75 N.m (5600 d/d)
stoichiometric air-fuel ratio	15:1

Figure 2 shows the basic layout of the small trigenerator system. In this figure are indicating the flows required for operation of the system, and the CHP applications considered for this case. The engine run with LPG as fuel with a consumption of 3,798 kg/h, generating an output shaft of 12,8 kW at 1800 rpm and drives a electric generator with nominal performance of 95%. The LHV of the fuel is 46 473 kJ/kg (BEN, 2014). The engine works with a real air-fuel ratio 12,1:1. The cylinder cooling system is pressurized at 140 kPa and moves a flow of water of 1980 kg/h. The outgoing temperature is 90°C and the return temperature is 84°C. In Table1 are reported the experimental data for each subsystems. This hot water pass although the heat exchanger one, type water/water (HE1), with an efficiency of 85%. The mass flow rate of the exhaust gas is 49.55 kg/h and the gas emerge from the engine at a temperature of 541°C, entering in the second heat exchanger, type gas/water (EH2), with efficiency of 80%, circulating on the side of the tubes and cooled to 307°C. Then, with the gases temperature levels required are brought to the generator of the absorption machine initiating the process of refrigeration and producing a cool water at 7°C. The absorption machine functions on ammonia/water, has a refrigeration capacity of 17.4 kW (5 TR), consumes the equivalent of 2.55 kg/h of LPG, if it is used with direct burning, and has an electrical consumption (Wp) of 1.275 kW. Hydraulic pumps drive the water heated in the heat exchangers while the water used in the absorption machine is fed by gravity. Electrical energy produced in the generator is dissipated in a resistive load bank submerged in a tank with 500 liters of water and can also operate lamps of 100 W. The Bank consists of 27 resistance of 1kW each, through to avoid the super heating in the resistance, water passes through a cooling cycle through a radiator and air, a pump is responsible to drive back the water to the tank.
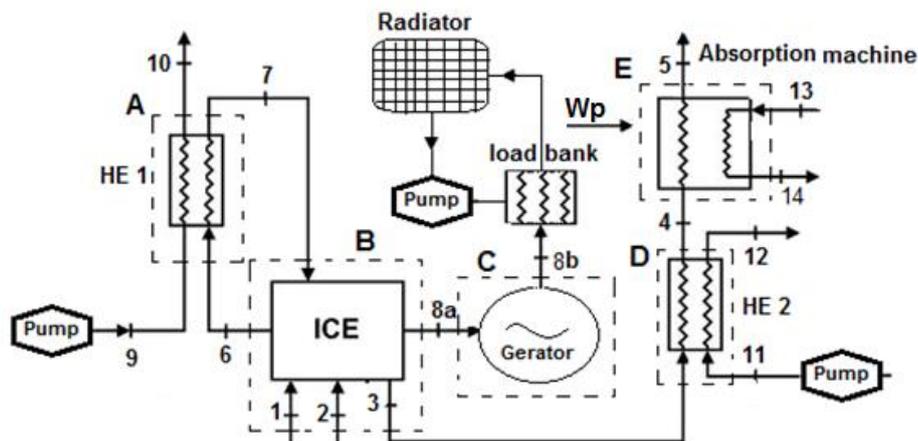

Figure 2. Schematic layout of the experimental installation

Table 2. Experimental data

point	flow	$\dot{m}$ (kg/h)	T (°C)	h (kJ/kg)	s (kJ/kg K)
1	LPG	3.798	25	-	-
2	Air	45.955	25	-	-
3	Exhaust gas	49.554	541.1	883.09	7.954
4	Exhaust gas	49.554	307	647.17	7.572
5	Exhaust gas	49.554	149	423.19	7.232
6	Water	1980	90.3	378.2	1.196
7	Water	1980	84	351.8	1.123
9	Water	265	25	104.8	0.3669
10	Water	265	65	272.1	0.8935
11	Water	66.4	25	104.8	0.3669
12	Water	66.4	65	272.1	0.8935
13	Water	87.9	25	104.8	0.3669
14	Water	87.19	7	29.51	0.1063

2.1 Mathematical modeling

Before the exergy analysis, mass and energy balances on the small scale cogeneration system are required to determine the mass flow rate and energy transfer rate the control surface.

Most transient-flow processes can be modeled as a uniform flow process. With the purpose of simplifying the first law calculations of the test engine, the following assumptions were made:(Y.A. Çengel, 2008; Wark, 199; Sayin & Hosoz, 2007; Sekmen Perihan and Yılbaşı Zeki, 2011).

- The engine powered cogeneration system operates in a steady-state condition.
- The ideal gas principles are applied to air and exhaust gases.
- The combustion reaction in gas engine is complete.
- The kinetic and potential energy changes are negligible.
- The temperature and pressure of dead (environmental) state are taken as the actual ambient conditions.
- Because the state of water in the exhaust is generally vapor in internal combustion engines, the lower heating value (LHV) of the fuel is used.

Energetic model

The total mass flow rate entering in the control volume was assumed to equal the total mass flow rate leaving the control volume. Then we can write:

$$\sum_{in} \dot{m}_{in} = \sum_{out} \dot{m}_{out} \tag{1}$$

Where $\dot{m}_{in}$ and $\dot{m}_{out}$ represent inlet and outlet mass flow rate, respectively

The First Law allows an assessment of the heat transfer rate that is lost by the engine to the environment, based on the assumptions listed above, can be written as fallows:

$$\dot{Q} + \sum \dot{m}_{in} h_{in} = \sum \dot{m}_{out} h_{out} + \dot{W} \tag{2}$$

Then fuel energy rate to the control volume is given by

$$Q_{fuel} = \dot{m} \cdot LHV_{fuel} \tag{3}$$

The equation of complete combustion depends on the chemical composition of the type of fuel. For LPG with 50% of methane and 50% of n-butane and amount of air equal 1.3. the equation is:

$$0.5C_3H_8 + 0.5C_4H_{10} + \alpha(O_2 + 3.76N_2) \rightarrow 3.5CO_2 + 4.5H_2O + 1.15O_2 + 25.94N_2 \tag{4}$$
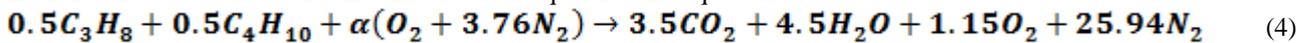
The exhaust loss was calculated using equation (5)

$$Q_{ex} = \sum_{i=1}^{n} n_i \bar{C}_{pi}(T_{ex} - T_0) \tag{5}$$

The specific heat of the exhaust was calculated by an empirical equation (Robert, H.P and Don, 1984)

$$\bar{C}_p = R\left(A + BT_{am} + \frac{C}{3}(4T_{am}^2 - T_1 T_2) + \frac{D}{T_1 T_2}\right) \tag{6}$$

Where $T_{am} = (T_1+T_2)/2$ is the arithmetic mean temperature and A, B, C and D are heat capacities constants for the concerned gases, $\bar{C}_p$ is the average specific heat over temperature change, and R is the universal gases constant R=8.31 J/mol.K.

The transferred or absorbed heat in the heats exchangers and in the absorption machine is calculated by:

$$Q_i = \dot{m}_i \cdot C_{pi} \cdot (T_{in\ i} - T_{out\ i}) \tag{7}$$

The engine block heat loss is calculated using equation (8)

$$Q_{lossICE} = Q_{fuel} - W_u - Q_{ex} - Q_{cw} \tag{8}$$

Where $Q_{lossICE}$ is the heat rejection to the oil plus convection and radiation heat transfer from the engine's external surfaces.

The total heat loss in the compact system is calculated by equation

$$Q_{losssys} = Q_{fuel} - W_{eff} - Q_{HE1} - Q_{HE2} - Q_{AM} \tag{9}$$

Thermal efficiency of the control volume (energy percentage), is usually determined as the ratio of the power output (net work) to the fuel energy input and determined by

$$\eta_{th} = \frac{W_u}{Q_{fuel}} \tag{10}$$

Exergetic model

When evaluating a variation of exergy or exergy flow between two states where the chemical composition of the substance is the same, them the chemical contribution cancels, leaving only the difference of thermomechanical contributions (Moran and Shapiro, 2000). The specific exergy then can be calculated by:

$$e_i = h_i - h_0 - T_0(s_i - s_0) \tag{11}$$

Where h_i and s_i are flow enthalpy and flow entropy per unit mass at the relevant temperature and pressure, respectively, while h_0 and s_0 stands for the corresponding values of these properties when the fluid comes to equilibrium with the reference environment.

The exhaust gas can be assumed as a mixture of ideal gases. Then the exergy is calculated as: (Torres, 2001)

$$e_{exi} = C_{pexi}\left[T_{exi} - T_0 - T_0\ ln\left(\frac{T_{exi}}{T_0}\right)\right] \tag{12}$$

The total exergy for a fuel is exactly equal to the chemical exergy multiplied by the coefficient that is the ratio of the chemical exergy and the lower calorific value of the fuel (Kotas, 1985; Szargut J, Morris D. R, 1988) and can be calculated by:

$$E_{fuel} = \dot{m}_{fuel} \cdot LHV_{fuel} \cdot \phi \tag{13}$$

where: ϕ is 1.0 - 1.06 for a LPG

The net exergy transfer by heat at temperature T, is given by

$$E_{heat} = \sum \left(1 - \frac{T_0}{T}\right) Q \tag{14}$$

Where T is the temperature at which heat transfer takes place. We considers T=313.15 K (temperature of the system boundary where heat is transferred to the environment)(Ramos da Costa et al., 2012)

The exergy produced work is defined by:

$$E_{work} = W \tag{15}$$

The rate of exergy destroyed in the control volume due to irreversibilities is calculated but the version of the Gouy-Stodola Theorem (Szargut, 1980).

$$E_{idest} = \sum E_{iin} - \sum E_{iout} \tag{16}$$

At steady state regime the exergy enters the control volume is equals the exergy exits plus the exergy is destroyed in the system.

The efficiency of the second law of thermodynamics is intended to serve as an approximation measure of a reversible process, is determined with the aid of an exergy balance.

$$\xi = \frac{exergy\ desired}{exergy\ used} = 1 - \frac{exergy\ destroyed}{exergy\ used}$$
(172)

For each sub-system, second law efficiency ξ, also known as rational efficiency (KOTAS, 1985), and ψ, also called degree of thermodynamic perfection (Szargut J, Morris D. R, 1988), have been determined according to the equations:

$$\xi_i = \frac{\sum_i E_{desired}}{\sum_i E_{used}} \tag{18}$$

$$\psi_i = \frac{\sum_i E_{out}}{\sum_i E_{in}} \tag{19}$$

3. RESULTS AND DISCUSSION

Figure 3 shows the energy balance for the internal combustion engine. The thermal efficiency for the ICE is 26%, this value is according to the others value reports for some researchers. Braga S.L. and Braga C.V.M. 2006, report a thermal efficiency of 22% for a diesel internal combustion engine working with loads smaller than 50kW. The total fuel energy input in this research was 49.03 kW.
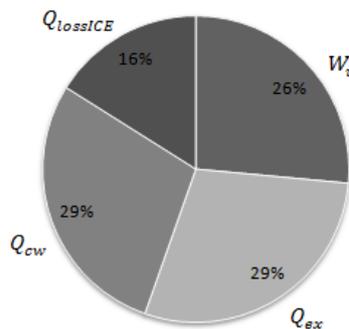

Figure 3. Energy balance in the ICE

In Figure 4 is shown the energy balance of the trigeneration system, the heat loss was 20.7 kW. The thermal efficiency of the system was calculated by equation (10) considering the payload is the water heating load in the heats exchangers and the load in the absorption machine for water cooled, this value was 35%.
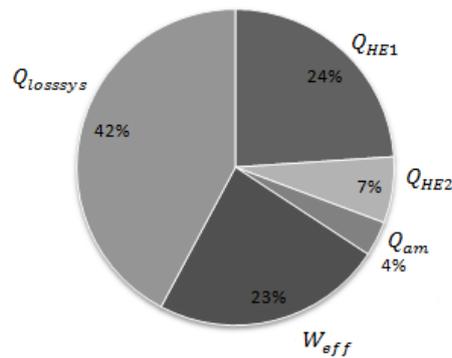
Figure 4. Energy balance of the cogeneration system

Efficiency achieved in the generation of electricity was 23%, finally the overall efficiency (η_{global}) of the plant reached the 57.8%. We can get a better vision of the energy flows proposed system in the Fig. 4, which represented the Sankey diagram. This diagram represents the quantitative distribution of all existing energetic flows in the system.
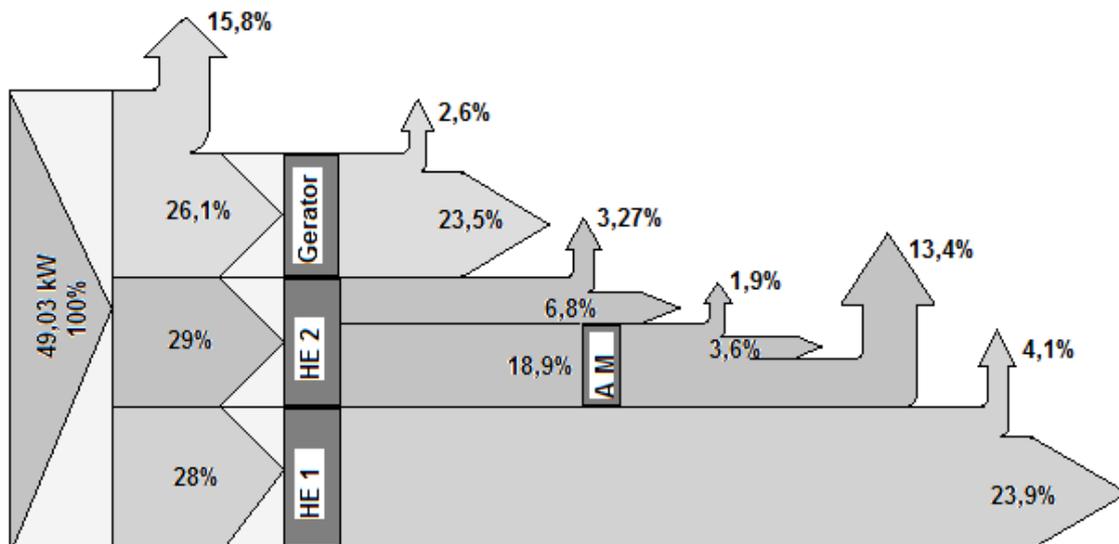

Figure 3. Sankey`s Diagram.

The destruction exergy, the rational efficiency e and the degree of perfection, have been calculated for each sub-system and are reported in Tab. 3. As expected, rational efficiency shows values lower than those do inherent to the *degree of thermodynamic perfection* for each sub-system analyzed.

Table 3. Irreversibility, rational efficiencies and degree of thermodynamic perfection.

Sub-system	ΣE_{in} (kW)	ΣE_{out} (kW)	I (kW)	ξ	ψ
ICE	49.03	30.959	17.693	0.397	0.639
HE1	14.44	12.626	1.823	0.285	0.874
HE2	3.719	2.161	2.079	0.090	0.441
AM	1.435	0.379	1.056	0.052	0.264
Generator	12.8	11.5	1.280	0.900	-

From table 5 it is possible to observe that the ICE shows a rational efficiency of 39.7%; this value is penalised largely because it a significant fraction of the fuel exergy is destroyed by irreversible processes in the engine, such as combustion, heat transfer, friction, etc. See Figure , which is, as is well known, the principal process that causes exergy destruction in a power generation plant (Sue & Chuang, 2004; Verkhivker & Kosoy, 2001). Although very important, combustion irreversibility is called intrinsic, since it is inherent to the combustion process. However this value is very close to those reported by Abusoglu & Kanoglu, 2009 and Sala *et al.*, 2006.

The exergetic efficiencies of the heat exchanger one (HE1), heat exchanger two (HE2) and absorption machine (AM) are calculated as 28.5%, 9.0% and 5.2%, respectively. This values show that the HE2 and the absorption machine are being the less efficient. Exergy destructions in theses heat exchanges units on the ground are mainly due to the high average temperature difference between the two fluid streams without mixing. It can be concluded that the maximum entropy generation occurred in the ICE instead of in water jacket and exhaust gas heat exchangers.
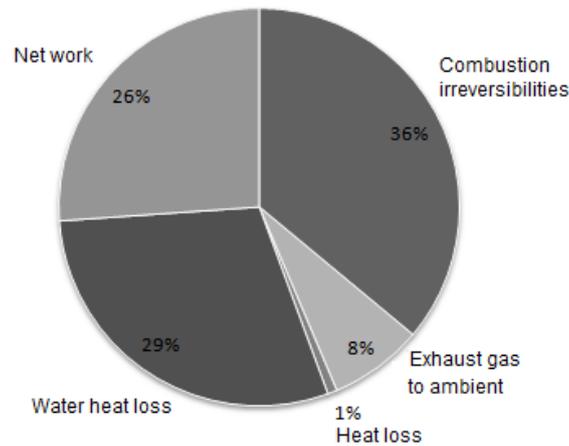

Figure 4. Exergy distribution in the ICE

Figure 4 shows a generation of entropy produced in the water cool system, it is three times more than that produced in the exhaust gas, this occur for the difference from the exhaust gas temperature, and cooling water temperature. A cooling water temperature nearly has no fluctuation. Under the whole operating conditions, the maximum temperature and the minimum temperature only vary 6°C. Hence, cooling water temperature can be almost regarded as a constant. The main reason is that the specific heat capacity and mass flow rate of cooling water are much greater than those of exhaust gas. The difference of temperature between exhaust gas and cooling water determines the difference of their energy characteristics especially the exergy characteristics. These results show that the heat exergy loss represents a small part of the total exergy (1%) , this value is consistent with other referenced works (Ramos da Costa *et al.,* 2012; Ameri, Kiaahmadi, & Khanaki, 2012).

In figure 5, is shown the Grassman Diagram where the exergetic efficiency of the trigeneration system is determined to be 51.19%. The exergy destruction in the engines of the cogeneration system accounts about 36.0% of the total exergy input and 73% of the total exergy destruction in the cogeneration system. The 23.49% of exergy entering the compact system it is converted to electrical power.
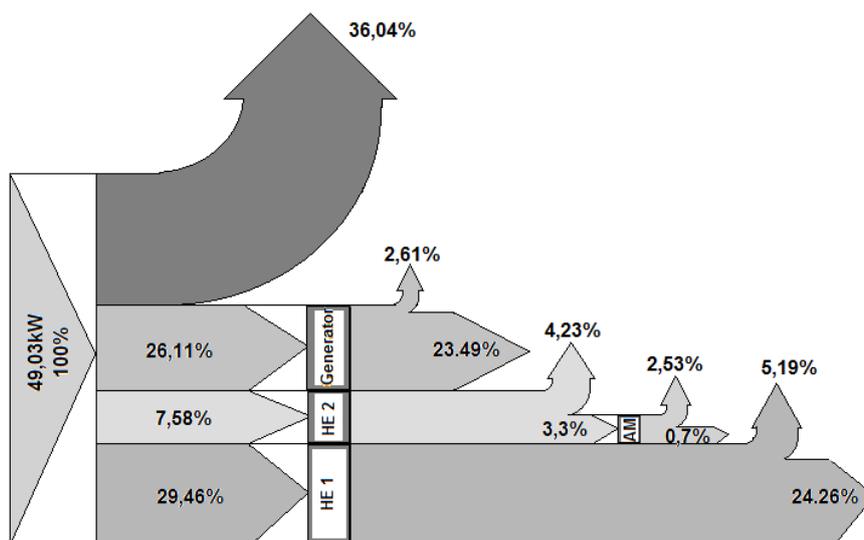

Figure 5. Grassman Diagram

4. CONCLUSIONS

If only power production is considered, it can be seen that first law efficiency is 23.5% and rational efficiency yield very similar values. On the other hand, considering the combined production of hot water, cool water and electrical power, substantially different values emerge (57.8% versus 51.19%).

On the other hand compact trigeneration system applied in the tertiary sector in Brazil is a good technology choice. In this country there are many opportunities to use a system like this, as the vastness of the territory prevents the total connection to the national grid, and there are still many isolated areas.

In addition to these results, this study reveals that a combined energy and exergy analysis provides a much better and more realistic answer.

5. ACKNOWLEDGEMENTS

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