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Thermo-economic Assessment of an Organic Rankine Cycle System for Repowering Application in a Landfill Biogas Power Plant

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Abstract:

Marca Ambiental is a Brazilian company that collects and treats solid waste from Grande Vitória, state of Espírito Santo, since 1997. In order to recovery this waste energy, mainly the landfill biogas, the company installed a steam power plant, in 2008, generating 1 MW of electricity. A study showed that greater efficiency and electric power, by recovering the landfill biogas energy, would be possible using internal combustion engines (ICE). This first repowering has already occurred using three internal combustion engine gensets, generating more than 3 MW of electricity. This work carried out a thermo-economic analyse for this plant new repowering, using organic Rankine cycle (ORC) systems. The energy and exergy balances showed that, although more than 40% of the biogas energy is converted into electricity, more than 54% is lost through exhaust gases (27.2%) and cooling water (27.4%), which represent, respectively, 14.13% and 4.12% of the biogas total exergy. The simulation and economic evaluation of this repowering, using ORC systems, shows technical and economic feasibility, respectively, for 400 kW and 300 kW of additional electricity generation.

Keywords:

Energy Efficiency, Repowering, Exergy Balance, Energy Recovery.

1. Introduction

According to [1], in 2021 the Brazilian electric energy demand was 497 TWh, an increase of 4.6% in relation to the year of 2020. In the same period, the participation of thermoelectric power plants in electricity generation was 23.5%. Nowadays, thermoelectric generation is mostly using fossil fuels. The use of landfill biogas from Urban Solid Waste (USW) landfills may be one of the alternatives to improve this scenario. One of the technologies for thermoelectric generation is Internal Combustion Engines (ICE). According to [2], the waste heat energy from these engine gensets, due to the cooling system (of the engine block, lubricating oil and intercooler) and exhaust gases, is generally rejected to the atmosphere. This waste heat, from exhaust gas and cooling water can be effectively recovered, for the system efficiency improvement and repowering.

Repowering is defined by [3] as an important alternative to achieve improvements in systems and thermoelectric generation. Among the improvements, there are: reduction of the specific fuel consumption and/or costs, reduction of the emissions and least cost option for increasing generation capacity. The repowering methodology presented by [3] is summarized with the following steps: determine the generation system goals; identify the generation system information and restrictions; identify the candidates repowering technologies; Evaluate and select the most feasible technology. In consideration of repowering thermal power plants through the usage of waste heat, a study was conducted by [4] in order to survey waste heat recovery technologies for power plants equipped with internal combustion engines (ICE) aiming at increasing the produced net power and overall efficiency. Among the alternatives, the following were chosen as interesting based on commercial availability and low impact on engine operation:

- Combined cycle with Conventional Rankine Cycle (CRC).
- Combined cycle with Organic Rankine Cycle (ORC).
- Combined cycle with Kalina Cycle (KA).
- Absorption cycle for intake air cooling.

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The results of a study by [4] showed that the Organic Rankine Cycle, using the exhaust gases heat from an ICE) presented the best performance for repowering purposes, considering the maximum power produced, achieving 5.3% in additional power produced. According to [5] ORC involves the same components of a conventional Rankine cycle: evaporator, expander, condenser and pump. One study [6] affirmed ORC is considered a simple technology because most of the heat addition and rejection happens during the phase change of the working fluid in the evaporator and condenser, respectively. The fundamental difference between CRC and ORC is the working fluid. As seen in [6], in ORC the fluid is an organic compound characterized by a lower boiling temperature than water, allowing power generation using heat sources with lower temperatures than in the conventional Rankine cycle. [8] affirms that, consequently, for low temperature waste heat recovery, ORC is a more suitable technology. One study [4] claims that organic fluids are high molecular mass fluids, compared to water, which allows compact equipment and low pipe diameter, for higher mass flow and higher isentropic efficiency of the expander.

In [5], parameter optimizations were performed, considering evaporation and condensation temperature, evaporator pinch-point and different types of organic fluids, as decision variables, in two configurations of ORC systems (simple and regenerative) to recover waste heat from an internal combustion engine. Considering the lowest specific costs for the power generated, as objective function, the best working fluid were selected, Toluene for waste heat recovery from engine exhaust gases and R141b for engine cooling water recovery. In [5], the maximum increase in efficiency and additional power generated was about 7%.

A study [6] was conducted to compare the efficiency of different working fluids in the heat recovery of exhaust gases from an internal combustion engine operating with biogas. The study examined pure organic fluids as well as their mixtures. The findings revealed that the Organic Rankine Cycle (ORC) system operating with toluene demonstrated the highest net electrical energy production, achieving an efficiency of 19.9%. In a related study [7], the use of toluene and cyclohexane in recovering waste heat from the flue gas of a reheat furnace was compared. The results indicated that under similar conditions, the ORC system with toluene exhibited a lower gross electricity production of 30 kW, but a higher energy efficiency of 17.08%. It is worth noting that neither of these studies explored the heat recovery potential from engine cooling water nor conducted an economic feasibility analysis of implementing the ORC technology.

With this in mind, the present work aims to recover both waste heat from the exhaust gases and cooling water of gensets in a Brazilian company involved in the collection and treatment of municipal waste. This company operates a thermal power plant based on three internal combustion engine gensets for landfill biogas energy recovery, generating over 3 MW of electricity. This paper outlines the procedures and discusses the results of a thermo-economic assessment of organic Rankine cycle systems for repowering applications in this landfill biogas power plant. The main contribution and novelty of this work lie in conducting an economic analysis of this plant under Brazilian conditions, as well as utilizing the engine cooling water as an input for a secondary ORC system.

To perform the energy and exergy balance analysis of the gensets, Engineering Equation Solver (EES) software was employed. IPSEpro was used for simulation purposes, and Excel was utilized for the economic evaluation of the ORC systems. Importantly, this work proposes advancements in the utilization of residual heat from an engine by providing a detailed explanation of the economic analysis using Brazilian economic conditions. Furthermore, the utilization of engine cooling water as a heat source is investigated in a real plant setting, with actual operating conditions that have not been previously explored.

2. Case Study Description

Marca Ambiental is an urban solid waste (MSW) recovery company, which has been collecting and treating municipal waste from Grande Vitória, in the state of Espírito Santo, Brazil, since 1997. According to [8], the urban solid waste is deposited in the landfill and then covered with layers of soil from the site itself, isolating it from the environment. Chambers are then formed, in which microbial activity, mainly of anaerobic bacteria that, through their metabolism, transform organic matter into combustible products, such as methane gas and released leachate. The slurry is captured through pipes and drained into treatment tanks.

In 2011, the company installed a steam cycle thermal power plant, generating 1 MW of electric power. Later, as shown in Figure 1, the original system was replaced by three ICE gensets (Genset 1, 2 and 3), generating more than 3 MW of electric power. Figure 1 shows the pipelines for capturing the biogas and directs it to the three engine-generator sets (EG's). There is a forecast for the future installation of two more generator groups. Nowadays, the thermoelectric plant has 3 ICE gensets modules, Jenbacher J416 GS models, 4-stroke Otto engines with mixture compression and exhaust gas turbocharging.

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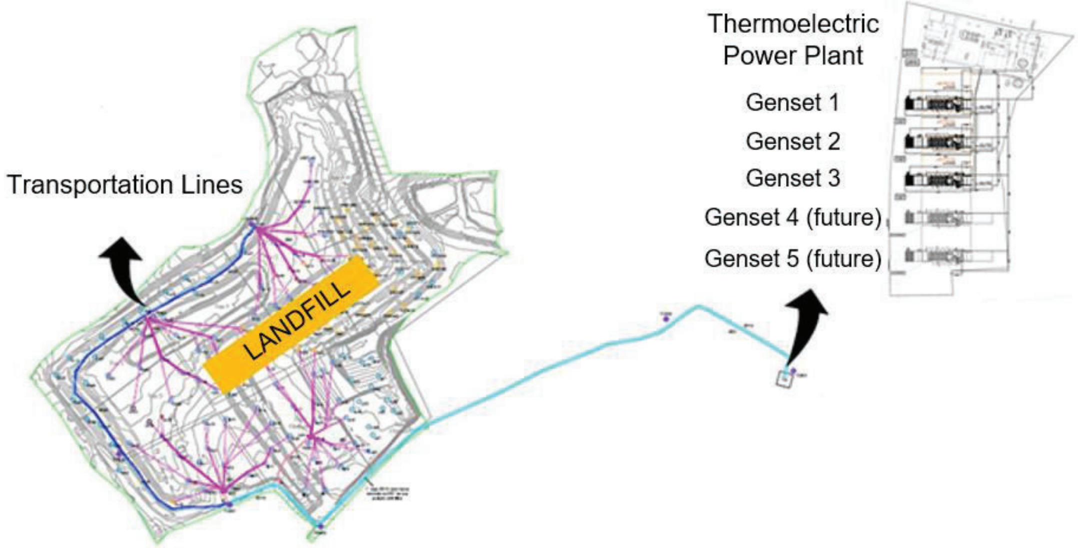

Figure 1 - Marca Ambiental's Biogas Capture and Thermoelectric System.

The average chemical characterization of the biogas generated at the Marca Ambiental landfill is shown in Table 1, as well the air composition, considering the average onsite conditions (25°C, 1 atm and 60% RH).

Table 1 - Biogas Characterization and Environmental Conditions.

Parameters			Value	Unit
Biogas Composition	CH_4		47.57	%
	CO_2		47.34	
	N_2		4.02	
	O_2		1.07	
	Heating Value	Lower (LHV)	12,753	kJ/kg
		Higher (HHV)	14,153	
Onsite Average Conditions	Pressure		1.013	bar
	Temperature		25	°C
	Relative Humidity		60	%

2.1. Gas Exhaust System

Each genset releases exhaust gases through the stack at a temperature of approximately 457 °C and a mass flow rate of 1.54 kg/s. This is the main source of heat with the potential to be recovered. It is important to stress the fact that there is a limit temperature for cooling the exhaust gases in their recovery process, due to the possible presence of sulfur in the fuel composition, which is considered 180ºC, in this work. The molar composition of the exhaust gases, shown in Table 2, was previously calculated with the aid of Engineering Equation Solver (EES) software using a complete combustion model with excess of wet air.

Table 2 - Molar Composition of the Exhaust Gases.

CO_2 (%)	H_2O(%)	N_2(%)	O_2(%)
12.85	14.49	67.57	5.09

2.2. Engine Cooling System

The cooling system of the generating units is done by means of demineralized water in two closed circuits, high temperature (HT) and low temperature (LT), with very low makeup during operation. The heat removed from the two aftercoolers, the lube oil and jacket water must be dissipated in a cooling system in order to allow the closing of the circuit. In the case of the thermoelectric plant, the cooling method adopted, depending on environmental conditions and water availability, is the use of a bank of radiators, with each generator set having two radiators. A thermal scheme of the cooling system, with water, biogas and exhaust gas flow values, can be seen in the engine flowchart, collected from the company documents, in Figure 2.

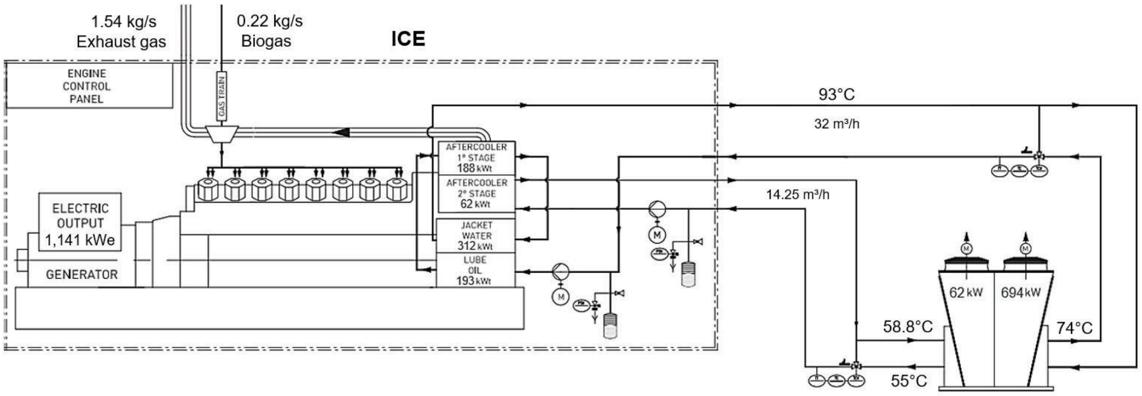

Figure 2 - Engine Flowsheet.

3. Engine Energy and Exergy Balance Methodology

The biogas energy rate is expressed by Eq. (1), as function of lower heating value (*LHV*) and mass flow ($\dot{m}_b$). The exergy rate of the biogas is calculated by Eq. (2), where $\dot{n}_b$ is the molar flow rate of the fuel, e_i^{ch} is the standard specific chemical exergy of each element, $\bar{R}$ the universal gas constant on a molar basis and T_0 is the reference temperature (dead state) in Kelvin scale.

$$\dot{Q}_b = \dot{m}_b \, . LHV \quad (1)$$

$$\dot{Ex}_b = \dot{n}_b \left(\sum_{i=1}^{j} y_i e_i^{ch} + \bar{R} T_0 \sum_{i=1}^{j} y_i \ln y_i \right) \quad (2)$$

The recoverable heat of each engine cooling water circuit is expressed by Eq. (3), where h_O and h_i are the specific enthalpy of the outgoing and incoming water stream, respectively. The sum of the two circuit recoverable heats, due to the two cooling water circuit, is the total recoverable cooling water heat ($\dot{Q}_{cw}$).

$$\dot{Q}_i = \dot{m}_i \, . (h_o - h_i) \quad (3)$$

The exergy of each engine cooling water flow is expressed by Eq. (4). The sum of the two circuit recoverable exergy is the total exergy loss, due two both cooling water circuit ($\dot{Ex}_{cw}$).

$$\dot{Ex}_i = \dot{m}_i \, . (\Delta h_i - T_0 \Delta s_i) \quad (4)$$

The energy contained in the exhaust gases ($\dot{Q}_{eg}$) is calculated using Eq. (5) Where $\dot{n}_{eg}$ is the molar flow rate of the exhaust gases, y_i is the molar fraction of each component in the exhaust gases, and $\Delta\bar{h}_i$ the variation of the specific enthalpy between the standard state and the state of interest.

$$\dot{Q}_{eg} = \dot{n}_{eg} \, . \sum y_i \, . \, \Delta\bar{h}_i \quad (5)$$

To calculate the exergy of the exhaust gas, its chemical and physical parts are calculated ($\dot{Ex}_g^{ch} + \dot{Ex}_g^{phi}$) represented by Eqs. (6) and (7), respectively, where y_i^e represents the molar fraction of the component in the air.

$$\dot{Ex}_g^{ch} = \dot{n}_b \, \bar{R} T_0 \sum_{i=1}^{j} y_i \ln \frac{y_i}{y_i^e} \quad (6)$$

$$\dot{Ex}_g^{phi} = \dot{n}_b \, \sum_{i=1}^{j} y_i (\Delta h_i - T_0 \Delta s_i) \quad (7)$$

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$$\dot{E}x_g = \dot{E}x_g^{ch} + \dot{E}x_g^{phi} \quad (8)$$

Since the mechanical power is pure exergy, then the exergy of this stream is represented by $\dot{E}x_M$. Finally, the overall energy and exergy balance are represented by Eqs. (9) and (10) respectively.

$$\dot{Q}_b = \dot{W}_M + \dot{Q}_{cw} + \dot{Q}_{eg} + \dot{Q}_l \quad (9)$$

$$\dot{E}x_b = \dot{E}x_M + \dot{E}x_{cw} + \dot{E}x_{eg} + \dot{E}_d \quad (10)$$

In Equations 9 and 10, $\dot{Q}_l$ is other heat losses from the internal system and $\dot{E}_d$ is the destroyed exergy involving engine irreversibility and other losses.

4. Recoverable Waste Heat using Reversible Bottoming Cycles

The heat available to be used by each heat source in an engine if the minimum recoverable temperature is reached is shown in Table 3, considering that the heat sink temperature ranges from 27 °C to 32 °C.

Table 3 - Amount of heat available at each source to be recovered.

Range Temperature of the Recoverable Heat Source	Total Heat (kW)
Exhaust Gases (457 °C – 180 °C)	511
Cooling Water HT (93 °C – 74 °C)	694
Cooling Water LT (58.8 °C – 55 °C)	62

In an optimist scenario, where the heat addition line of the bottoming cycle is very close to the hot source line and the heat rejection line is close to the heat sink, one has the maximum technical potential presented in Table 4, considering that the recoverable would be converted into net power using a total reversible cycle.

Table 4 - Maximum Technological Potential of Repowering.

Range Temperature of the Recoverable Heat Source	Net Power (kW)
Exhaust Gases (457 °C – 180 °C)	250
Cooling Water HT (93 °C – 74 °C)	105
Cooling Water LT (58.8 °C – 55 °C)	5

It can be seen that the LT cooling circuit has a small flow rate and temperature variation compared to the remaining heat sources, so this water flow has a very small heat amount. Therefore, only the HT circuit heat source is chosen to be model for repowering purposes using the ORC with cooling water hot source.

4.1. Proposed Heat Recovery Thermal Scheme

The configuration of the ORCs coupled to the ICEs is proposed by joining the flows of the 3 gensets units and proposed to model one ORC for each of the available recoverable heat sources, as shown in Figure 3.

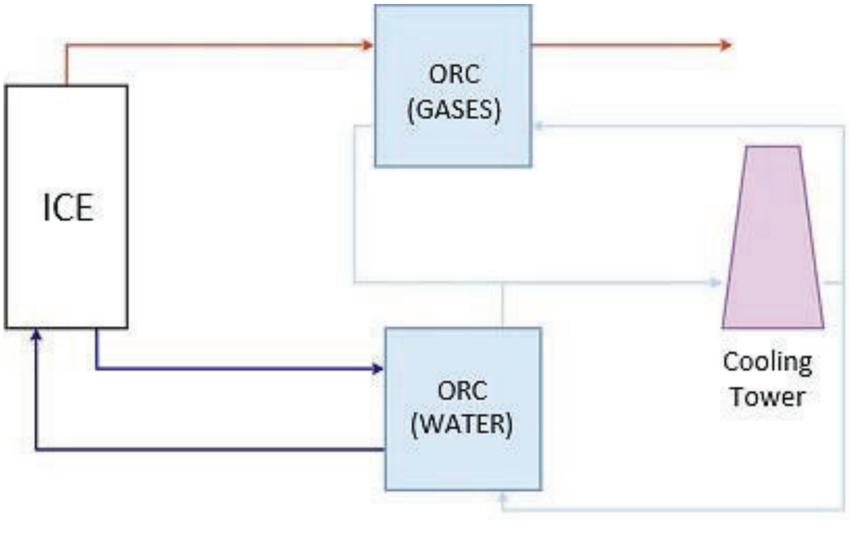

Figure 3 - Configuration of the ORCs to be coupled at the Marca Ambiental Thermal Power Plant.

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5. Thermo-Economic of the ORCs

From the optimization data of [5], which caried out a similar case study, the implementation of the simple ORC was defined. This same study tested several working fluids for the different heat sources for recovery and reached a conclusion that in terms of power production with the lowest specific cost, the best fluids were Toluene and R141b, for the cycle recovering exhaust gases heat source and for the cycle recovering cooling water heat source, respectively. Based on that work, other parameters were taken into account to start modeling the ORCs, such as evaporation and condensation temperature and isentropic efficiencies of turbines and pumps, for each modeled ORC. These parameters are presented in Table 5.

Table 5 - ORC parameters.

Parameter	Fluid	
	Toluene	R141b
T_{evap} (°C)	242.60	62.63
T_{cond} (°C)	57.45	38.35
η_b (%)	71.54	71.10
η_t (%)	81.52	86.27

With the set of equations of the mass and energy balance of the Organic Rankine Cycle it was possible to obtain all the thermodynamic properties (pressure, temperature, enthalpy and entropy) of each state of the cycle, which was obtained as described, according to the configuration represented in Figure 4.

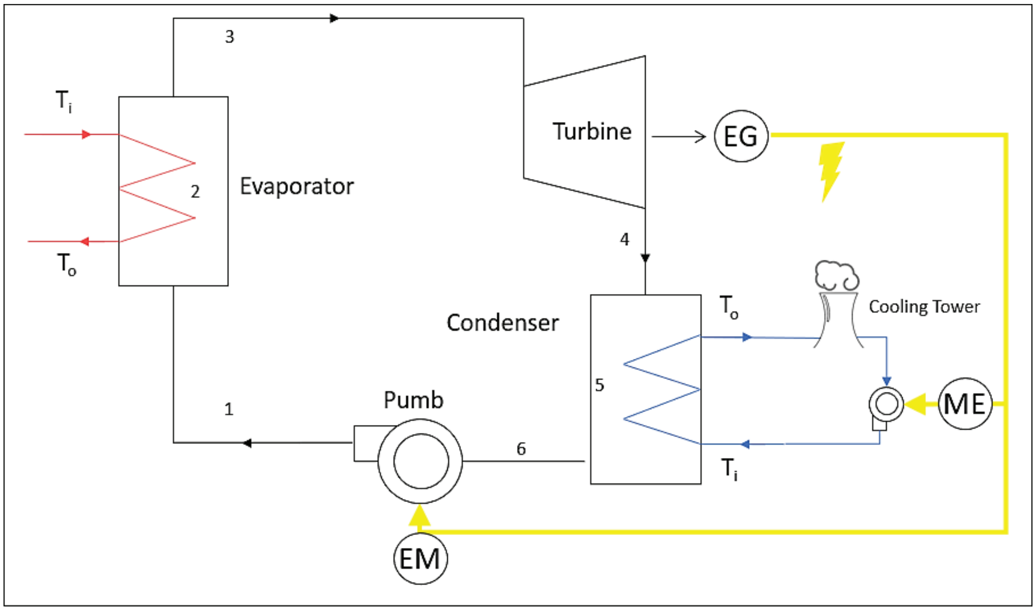

Figure 4 – ORC System Flowsheet.

The successful completion of a thermal design project requires estimation of the major costs involved in the project. Therefore, good cost estimation is a key factor in successfully completing a design project, as shown in [11]. There are many types of capital cost estimations and various methods often provide different results. The economic evaluation in this work is performed according to the module costing technique (MCT), extensively used for preliminary cost estimates of plants by [12]. This technique relates all direct and indirect costs to the purchased equipment cost evaluated for base conditions (C_{PE}) at ambient pressure, and carbon steel construction expressed by Eq. (11), where K_i are constants depending on the equipment type and A is the capacity or size parameter.

$$log_{10} C_{PE} = K_1 + K_2\, log_{10}(A) + K_3[log_{10}(A)]^2 \qquad (11)$$

Deviations from these base conditions are handled by multiplying pressure (F_P) and material (F_M) factors. The pressure factor is given by Eq. (12), where P is the pressure and c_i are constants depending on equipment type.

$$log_{10}\, F_P = C_1 + C_2\, log_{10}(P) + C_3[log_{10}(P)]^2 \tag{12}$$

The additional direct and indirect costs are considered through the bare module factor (F_{BM}) in the module costing technique. The bare module cost is the sum of all direct and indirect cost and can be calculated by Eq. (13).

$$C_{BM} = C_{PE} \cdot F_{BM} \quad [US\$] \tag{13}$$

The values of the bare module cost factors are given for different types of equipment. For heat exchangers and pumps the expression of the bare module cost factor is given by Eq. (14), where B_i are constants depending of the heat exchanger or pump type.

$$F_{BM} = B_1 + B_2 \cdot F_P \cdot F_M \tag{14}$$

For other components the F_{BM} is directly given as a multiplier that accounts for equipment type, operating pressure and construction material. The coefficients for Eq. (11) to (14) are obtain from [12] for carbon steel turbines, pumps and shell and tube heat exchangers. The parameters to estimate the cost of the colling tower was not considered in [11]. Thus, for this equipment, the methodology by [13], was employed in this work, as shown in Equation (15).

$$C_P = C_{ref,i} \cdot \left(\frac{X}{X_{ref}}\right)^m \tag{15}$$

For modifications and expansions of existing thermal systems, there are also other costs that need to be accounted for, like taxes and contingencies costs. According [14] when there are no other recommendations, these costs are 3% and 15% of the bare cost module respectively. Adding these remaining costs, the total module cost is calculated by Eq. (16), where *n* represents the number of the project equipment.

$$C_{TM} = 1.18 \,.\, \textstyle\sum_{i=1}^{n} C_{BM,i}\ [US\$] \tag{16}$$

All the data available in [12] are referenced in 2001. According to [14] the calculated cost updated is made through an appropriated cost index. The cost index is an inflation indicator used to correct the cost of equipment items, material, labor, and supplies to the date of the estimation. For thermal design projects the Chemical Engineering Cost Index (CEPCI) is recommended for total plants, or groups of components. It's adopted the CEPCI of 2022 ($CEPCI_{2022} = 821.3$). Thus, the correct total cost is given by Eq. (17), where the $CEPCI_{2001}$ is 397.

$$C_{Total} = (C_{TM})\ \frac{CEPCI_{2022}}{CEPCI_{2001}}\ [US\$] \tag{17}$$

6. Results and Discussions

The thermodynamic evaluations of the repowering alternative were carried out on energy and exergy basis aiming at providing more support to the analysis, since exergy efficiency, exergy losses and irreversibilities allow a better understanding of the improvement opportunities in the waste heat recovery for power production, once that power is pure exergy. Exergy allows the evaluation of waste heat according to its maximum potential for power conversion, which is not possible using only an energy evaluation, as it overestimates the available heat potential for recovery and repowering. The economic evaluation allows the estimation of the specific capital cost for the repowering technology.

6.1. Recoverable Waste Heat

Based on the balances performed on the engine, the equations were implemented in the EES software. Thus, the results were found for a Jenbacher J416 GS engine of the EG of the Marca Ambiental Thermal Power Plant (TPP). In Figure 5 are the energy and exergy balances of an engine. The available heat energy is 54.60% of the energy provided by the biogas. However, the exergetic heat potential is 18.25% only.

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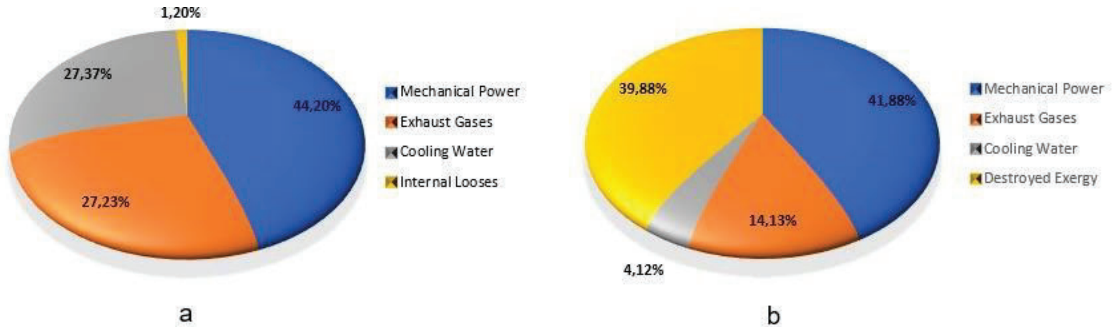

Figure 5 - UTE Marca Ambiental EG balances: a) energy balance, b) exergy balance

6.2. ORC Utilizing the Exhaust Gases Heat

Given the thermodynamic properties of toluene calculated in the EES it is possible to plot the T x s plot of the cycle, as seen in Figure 6. the values of the properties for each point are shown in the Table 6. The cycle has a net electric power production of 300.87 kW. In Table 7 there are the capacities and costs of each equipment.

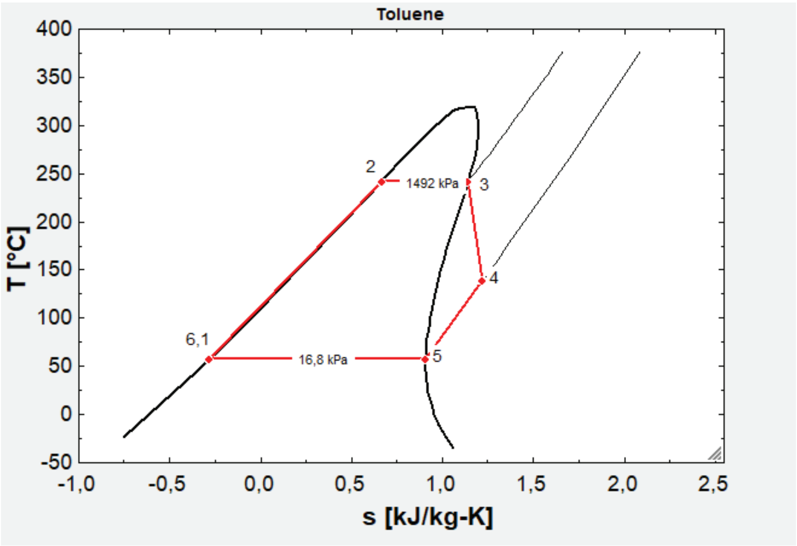

Figure 6 - Toluene T x s Diagram.

Table 6 - Thermodynamic Properties of Toluene at the ORC State for 2.38 kg/s of Mass Flow.

State	Temperature (°C)	Pressure (kPa)	Enthalpy (kJ/kg)	Entropy (kJ/kg.K)
1	58.2	1492.4	-98.8	-0.281
2	242.6	1492.4	301.7	0.667
3	242.6	1492.4	545.7	1.140
4	139.0	16.8	407.9	1.218
5	57.5	16.8	292.8	0.908
6	57.5	16.8	-101.3	-0.283

Table 7 - Capacity and Cost of Each Equipment.

Equipment	Capacity [unit]	C_{BM} [US$]
Evaporator	46.85 [m²]	149,190.03
Condenser	99.52 [m²]	203,466.40
Turbine	328.03 [kW]	543,019.03
ORC Pump	5.90 [kW]	13,469.85
Condenser Pump	2.06 [kW]	10,681.43
Cooling Tower	3.48 [m³/min]	29,867.1

Inserting the fees and updating the values to 2022 the total cost of the project is US$ 2,318,339.01.

Starting from the total cost of the project, we can apply some methods of economic feasibility analysis, such as: payback, NPV (Net Present Value) and IRR (Internal Rate of Return). For this we need the interest rate per year on the investment, the value of the electric energy and the life time of the project. The interest rate of the investment which was adopted is 12% and the life time of the equipment adopted was 20 years. Unitary Variable Cost (UVC), that is, the cost of selling electricity to the grid was US$ 0.130/kWh. The obtained payback was 15.36 years, showing that the initial investment will be paid, even if it is in a time close to the useful life of the equipment of 20 years. The IRR is was approximately 13.38%, greater than 12%.

6.3. ORC Utilizing Cooling Water

Faced with the thermodynamic properties of toluene calculated in the EES it is possible to plot the T x s plot of the cycle, as seen in Figure 7. the values of the properties for each point are shown in the Table 8.

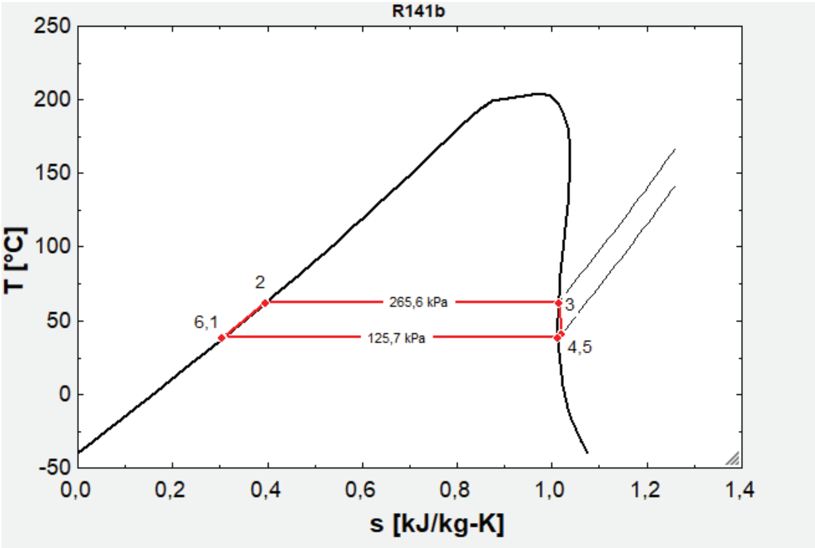

Figure 7 - R141b T x s Diagram.

Table 8 - Thermodynamic Properties of R141b at the ORC State for 8.35 kg/s of Mass Flow.

State	Temperature (°C)	Pressure (kPa)	Enthalpy (kJ/kg)	Entropy (kJ/kg.K)
1	38.50	265.59	83.27	0.306
2	62.63	265.59	112.86	0.397
3	62.63	265.59	320.09	1.014
4	41.58	125.74	306.10	1.021
5	38.35	125.74	303.52	1.013
6	38.35	125.74	83.09	0.305

The cycle has a net electric power production of 100.42kW. In Table 9 re defined the capacities and costs of each equipment.

Table 9 - Capacity and Cost of Each Equipment.

Equipment	Capacity [unit]	*CBM* [US$]
Evaporator	300.71 [m²]	106,597.48
Condenser	891.53 [m²]	143,471.47
Turbine	116.74 [kW]	284,450.35
ORC Pump	1.47 [kW]	10,231.25
Condenser Pump	3.07 [kW]	11,462.07
Cooling Tower	5.34 [m³/min]	45,830.55

Inserting the fees and updating the values to 2022 the total cost of the project is US$ 1,469,673.82.

In this case, the ORC system is not able to return the money invested within the useful life of the equipment of 20 year. The find the IRR, which was approximately 4.06%, comparing to the MARR of 12 %, it can be stated that the IRR rate is extremely lower than the MARR, show the unfeasibility of the project.

7. Conclusions

This paper aimed to perform the thermodynamic modeling of ORC systems, and then analyze the additional power generated, repowering performance, in order to carry out an economic feasibility analysis of the implementation of the ORC systems, using some economic indexes: payback, Net Present Value (NPV), Internal Rate of Return (IRR). Two separately ORC system was simulated for this repowering purpose.

Firstly, energy and exergy balance were performed in order to evaluate the thermodynamic repowering potential, obtaining more than 43% of repowering potential. However, it is important to notice the limitations of the recoverable waste heat sources, which make the repowering potential drop down to approximately 32%. Additionally, it was observed that the LT colling water circuit has a very small repowering potential, representing approximately 5% of the HT cooling water circuit or 1% of the exhaust gas repowering potential.

Bearing this in mind, an ORC system was modeled for the exhaust gases from the 3 engines, operating with toluene as working fluid, generating 300.87 kW, which represents almost 8.8% of repowering. Another ORC was modeled for the HT colling water circuits heat of the 3 engines, operating with R141b as working fluid, generating 100.42 kW of additional electric power, representing 2.9% of repowering. The net efficiencies obtained for each cycle were 20.99% for exhaust gases ORC system and 5.83% for the cooling water ORC.

The economic analyses indexes obtained demonstrated the economic feasibility for the exhaust gases heat ORC system only, with total investment cost of US$ 2,318,339.01 and payback of approximately 15.5 years.

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Nomenclature

Symbols

A	capacity or size parameter, kW, m² or m³/min
B	bare module factor coefficient
C	pressure factor coefficient
E	exergy rate, kW
h	enthalpy, kJ/kg
K	equipment type coefficient
$\dot{m}$	mass flow rate, kg/s
P	pressure, bar
$\dot{Q}$	energy rate, kW
s	entropy, kJ/(kg.C)
T	temperature, °C
$\dot{W}$	power, kW
y	mole fraction
η	efficiency

Subscripts and superscripts

BM	bare module
cond	condensation
b	biogas
cw	cooling water

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evap	evaporation	i	in
eg	exhaust gases	o	out
ex	exergetic	wf	working fluid
ge	generator efficiency		

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