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EXERGETIC AND THERMOECONOMIC ANALYSIS OF A COGENERATION PLANT ON A BRAZILIAN PRE-SALT OIL OFFSHORE PLATFORM

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Abstract. *The usage of natural gas in the Brazilian pre-salt oil offshore platforms has turned into a major concern for the industry. This is mainly due to the reinforcement of the environmental legislation and public pressure on the rational use of energy. Firstly, this work will give the present picture of one of those platforms, containing four gas turbines generating power (100 MW installed) and a heat exchanger fed by the exhaust gases, transferring energy to process water. Secondly, it intends to promote alternatives in order to increase the efficiency of the cogeneration plant that is currently very low (around 55 %), mitigate emissions and eventually rationalize the system economically. To do so, the exergetic analysis will be essential because of its capability to point out the useful energy of the streams and the generated irreversibility in each equipment. Furthermore, the thermoeconomic analysis is also very useful to assess the cost of the streams, based on energetic or monetary terms. Among the changes proposed, modifications of the heat exchange system are required for improved performance, especially because of the high remaining exergy of the exhaust gases that is wasted. Additionally, a diagnosis of power partial load conditions is provided.*

Keywords: *Exergy, Thermoeconomics, Cogeneration, Oil Platforms, Natural Gas*

1. INTRODUCTION

An offshore platform, called FPSO (floating, production, storage and off-loading), located in the Brazilian pre-salt field is analyzed here; it consists of a static ship that is able to process at the top, store at the hull and off-load the whole production to transport units. Its cogeneration plant is presented and analyzed thermodynamically and economically through the exergetic and the thermoeconomic analysis; both of these are based on The 2^{nd} Law of Thermodynamics. Looking up at the literature, this kind of analysis can be found on Voldsung *et al.* (2013), at the North Sea platforms, or even at the pre-salt, by de Oliveira Jr and Sanchez (2014). Previous experiences, as the ones cited, have shown that there is a great potential for improvement, specially at the main irreversibility generation sites, such as the petroleum separation, the gas compression and the gas turbine plants, which is the object of this study.

A significant amount of natural gas is recovered alongside oil, and natural gas management has become the subject of many studies because of its thermodynamic and economic potential, as well as its environmental damage potential. Instead of flaring or venting natural gas, which means wasting its physical and chemical potential, interesting alternatives have taken place such as gas lifting the oil, re-injecting into the reservoir, rationally supplying to matching the electrical load of the platform, or even exporting it as liquified natural gas (LNG) or through pipelines.

This work is focused on major objectives: 1) give an overall view of the standard type of generation, locating irreversibility generation and high-cost flows; 2) simulate improvements of thermal and electrical orders, and finally 3) propose technical solutions to be implemented in the future.

2. METHODOLOGY

The gas turbine cogeneration plant is composed of four gas turbines, three of which are active, and each one of them containing a recovery heat exchanger at the bottom of the nozzle to supply process hot water by means of the exhaust gases. The gas turbine is analyzed isolated, which will be referred to as “GT” (gas turbine), and accounting for the heat exchanger at the bottom, referred to as “GT + WHR” (gas turbine + waste heat recovery). The effect of the thermal system into the electrical system is that the turbine will experience a smaller pressure ratio due to the pressure loss at the heat exchanger, which has to be compensated by the turbine in order to reach the atmospheric pressure at the bottom of the system. The great temperature difference between the fluids in the heat exchanger also generates great irreversibility and should be improved.

Partial load conditions are extremely important for this study because of the varied electrical demand of the platform. The eletrical demand depends on a number of factors, such as: production volume, petroleum quality, field age, among others. This methodology also contemplates that sort of concern.

The current system divides the thermal load into three active turbines, which will be discussed first, but it will be suggested to use only two of the turbines for that end, allowing further expansion in one of them, which is later implemented in the study case. That is the main change proposal for the thermal system, which is operational, not structural, that would directly affect the electrical and the whole plant's efficiency.

Modeling the gas turbine was the first step of the analysis. It can be seen from the figure below that air is compressed, some of it bypasses the combustion chamber to serve as a turbine cooler fluid, while some of it burns the fuel; next, the burnt gases expand in the turbine, generating power, and the exhaust gases are finally let out to the atmosphere or serve as a hot working fluid to supply process hot water in the recovery heat exchanger.

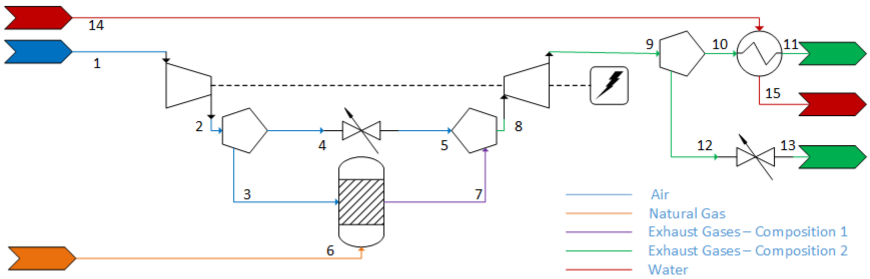

Figure 1. Gas turbine model

There are nine material flows for the GT because there is no heat recovery, and fifteen for the GT + WHR. There are also seven complementary flows in both of these models, corresponding to various types of electrical flows. The following table describes how the organization was established, depending on the number of total flows.

Table 1. Electrical flows

Type	GT flow number	GT + WHR flow number
Compressor shaft power	10	16
Turbine shaft power	11	17
Total shaft power	12	18
Gross power	13	19
Auxiliary power: compressor	14	20
Auxiliary power: turbine	15	21
Net power	16	22

In order to make a comparison between the different disposals cited, it will be presented a case study with particular electrical and thermal demands. It will be possible to verify the parameters' differences more clearly by the comparison of the disposals. For the first model, three GT + WHR at 100 % load are included, while the proposed includes two GT + WHR at 100 % and one GT at 91.66 % load, resulting in the same power output, due to the GT's higher power capacity.

The computational resources for this work are the EES ® (Engineering Equation Solver, F-Chart) software and Thermoflex ® (Thermoflow), that are available at the Faculty of Mechanical Engineering at the university. The thermodynamic states of the flows were simulated in Thermoflex and then programs and sub-routines were written in EES to obtain the results, including thermodynamic properties and equipment performance data.

2.1 Thermodynamic approach

The gas turbine used is the Rolls Royce RB 211; technical specifications, provided by the software Thermoflex (theoretical data), and simulation specifications (model apart from the software) are in Table 2.

Natural gas composition is obtained from the BG Group ® data, and is specified in Table 3. It is important to clear that this composition corresponds to a specific combination of working conditions, and the composition may vary with mode of operation, field composition and time.

Thermodynamic properties' reliability is essential for any work in the field. With that in mind, the substances were brought to a common reference state into EES. Enthalpy and chemical exergy are relative to 25 ^{o}C and 101.3 kPa, while entropy is calculated according to The 3rd Law of Thermodynamics, at 0 K and 101.3 kPa. The following relations for ideal gas mixtures were taken from Klein and Nellis (2012) and Bejan *et al.* (1996), and used in this work.

$$h_{mix} = \sum x_i h_i \tag{1}$$

Table 2. Technical specifications at ISO condition - Rolls Royce RB 211 turbine

Model	Theoretical	Simulated
Shaft power [kW]	25887.0	25730.7
Exhaust temperature [^{o}C]	501.9	501.3
Efficiency$^{(1)}$	36.78 %	36.56 %

$^{(1)}$ relative to the shaft power

Table 3. Natural gas composition

Substance	Molar fraction
N_2	0.60 %
CO_2	3.00 %
C_4H_10	75.66 %
C_2H_6	10.97 %
C_3H_8	6.65 %
C_4H_{10} - n	0.92 %
C_4H_{10} - iso	1.55 %
C_5H_{12}	0.65 %

$$s_{mix} = \sum x_i s_i - R \sum x_i ln(x_i) \tag{2}$$

$$b_{ph,mix} = (h_{mix} - h_{0,mix}) - T_0(s_{mix} - s_{0,mix}) \tag{3}$$

$$b_{ch,mix} = \sum x_i b_{ch,mix} + RT_0 \sum x_i ln(x_i) \tag{4}$$

The chemical exergy values are listed on Szargut *et al.* (1988).

2.2 Thermoeconomic approach

Thermoeconomics is a broad field relating quality and cost of energy, including many methodologies found in the literature. It was primarily discussed by Tribus *et al.* (1966) in the 1960s, and a widespread academic development has been observed since then. In the beggining of the 1990s, the CGAM problem (Valero *et al.*, 1993), united many academic authors in publications comparing their methodology to the same system. However, the exergetic and exergoeconomic costs, introduced by Valero and Lozano (1993) were used, from all of those, as sufficiently reliable and useful tools for this work.

The Exergetic Cost Theory is a rational manner of dealing with the flow costs. Through four propositions, it is possible to allocate costs, in energetic units, to each of the flows according to the system's irreversibility generation involving those flows, which means that the higher the irreversibility generation for the flow production, the higher the exergetic cost; another, but coincident, approach to the exergetic cost is relating it to the amount of exergy destinated to the formation of the flow, with higher costs allocated to the flows with higher expenditure of exergy. An interesting assessment to these results is obtained by the unit exergetic cost, which is $k^* = \frac{B^*}{\dot{B}}$; this way, all the flows are in dimensionless mode.

The Exergoeconomic Theory is based on the same idea, in monetary units, considering capital and operation and management (O&M) investments for every equipment along time. There is also the specific cost, which is $c = \frac{\dot{C}}{\dot{B}}$; it is analogous to the unit exergetic cost.

According to the gas turbine total price, taken from the Thermoflex software, the costs of each of its equipments were assigned based on Gomes (2001).

Table 4. Investment cost for the equipments

Equipment	Investment cost$^{(2)}$ [%]
Compressor	25.00
Combustor	4.09
Turbine	25.00
Recovery Heat Exchanger	37.45
Generator	45.91

$^{(2)}$ relative to the gas turbine total price

The prices are taken to a time dependency considering the capital investment on the equipment (Z), capital recovery factor (CRF), the O&M complimentary cost of investments (ϕ) based on Z and the plant's yearly operational period (N). The cost of capital investment is US$ 11.2 million, divided as in Table 4, CRF is considered to be 18.2 %, ϕ is 1.06 and N is 8000 hours.

$$\dot{Z} = Z\,\frac{CRF\phi}{3600N} \tag{5}$$

Since the costs are not destroyed or lost, the cost balance for the whole system is:

$$\dot{C}_{fuel} + \dot{Z}_{GT} + \dot{Z}_{HE} = \dot{C}_{power} + \dot{C}_{heat} + \dot{C}_{exhaust} \tag{6}$$

An important task in the thermoeconomic approach is to allocate costs between products, in the case of a cogeneration plant. Following the second proposition of The Exergetic Cost Theory straight would lead to assigning the costs of the exhaust gases only to heat, because the exhaust gases are released to the atmosphere right after the recovery heat exchanger, resulting in a poor impact to the electrical system. The most recommended allocation is assigning exhaust gases' cost to heat and power according to their exergy, and that is the method chosen here.

One of the main factors that affect the thermoeconomic results is the natural gas price, which may vary significantly from a low price, when the platform produces it, to a high price, when it has to be imported. A sensitivity analysis over natural gas price is presented in order to measure how relevant that impact is, but the standard value was chosen as US$ 4/GJ, as in Tsatsaronis and Pisa (1994).

2.3 Heat transfer approach

The heat transfer approach is based on the methodology presented by Incropera *et al.* (2012) referring to the effectiveness and overall heat transfer coefficient. The only phenomenon taken into consideration for heat transfer was convection, which is acceptable, as the walls of the pipe are considered thin and the heat loss to the environment is between 0.5 and 1.0 % for every simulated case.

The convection coefficients, for both gases and water, are calculated by the relations of Dittus-Boelter, and the pipe heat exchange system is considered to be a counter-current heating concentric pipe. The following equations were adopted, where the Nusselt number (Nu_D) relates convection and conduction.

$$Nu_D = 0.023\;Re_D^{0.8}\;Pr_D^{0.4} \tag{7}$$

$$NTU = \frac{1}{C_r - 1}\,ln\left(\frac{\epsilon - 1}{\epsilon C_r - 1}\right) \tag{8}$$

The process subcooled water mass flow is 422.4 kg/s, the inlet temperature is 100^{o}C, the inlet pressure is 11.114 bar, the outlet temperature is 130^{o}C and the outlet pressure is 9.944 bar. When there are three heat exchangers active, the mass flow of water in each heat exchanger is 140.8 kg/s, while, with two heat exchangers, it is 211.2 kg/s, since the mass flow is divided equally for each heat exchanger.

The heat exchanger's maximum capacity is 26.8 MW, which will be reached when the thermal load is divided by only two heat exchangers. Also, the pinch temperature was designed to be 10^{o}C, so no matter what is the temperature at the inlet, the outlet temperature of the hot fluid is 10^{o}C plus the inlet temperature of the cold fluid (process water), therefore 110^{o}C. The variable that will control the heat transfer, for that reason, is the mass flow of the hot fluid, since the inlet and outlet temperature and mass flow of the cold fluid is constant.

3. RESULTS AND DISCUSSIONS

Many results have been collected with the methodology that was previously introduced. The main exergetic and thermoeconomic parameters considered for flows are: exergy, unit exergetic cost, monetary cost and specific monetary cost; when it comes to the equipments, other criteria have been used: exergetic efficiency and generated irreversibilities.

3.1 Exergetic analysis

Three of the main parameters listed before are presented ahead for flows and equipments of the GT + WHR. The percentage shown in the legend represents the load case.

It can be noticed that flow 8, corresponding to the fluid entering the turbine has the highest exergy, which makes sense, bearing mind that the main goal of the system is to get this flow expanded for power generation. Also, flow 17, corresponding to the turbine shaft power is the highest among energy flows, because that is the origin of all the other energy flows. The electrical exergetic efficiency stands for the division of exergy of flow 22 (net power) and flow 6 (fuel

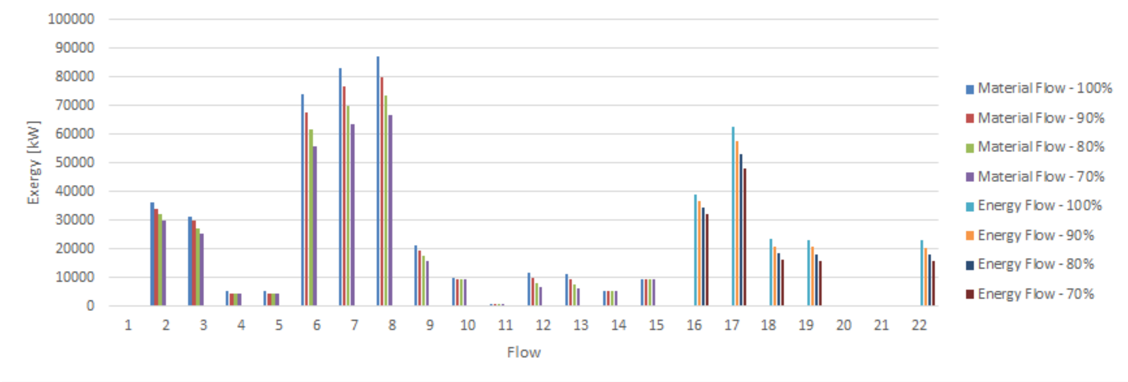

Figure 2. Exergy - GT + WHR

exergy). Also, the difference between exergies 14 and 15 (thermal exergy) is very low. The exergy decays with the load, except for flows 14 and 15, since the thermal load remains constant.

Figure 3 shows that the exergetic efficiency varies little with the load; some of the results must be highlighted, though. The exergetic efficiency of the combustor decays with the load, despite of the fact that the irreversibilities also decay, falling 2.2 % from 100 to 70 % load; that can be explained by the fact that the fuel flow does not decay at the same pace as irreversibility. The exergetic efficiency of the recovery heat exchanger goes in the opposite way, since the irreversibility generation decays with the load, rising 2.0 % from 100 to 70 % load.

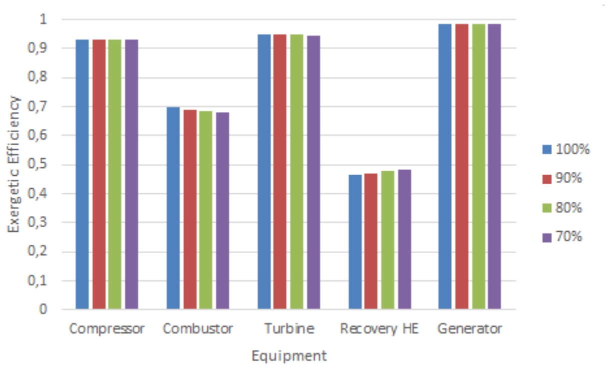

Figure 3. Exergetic efficiency of equipments - GT + WHR

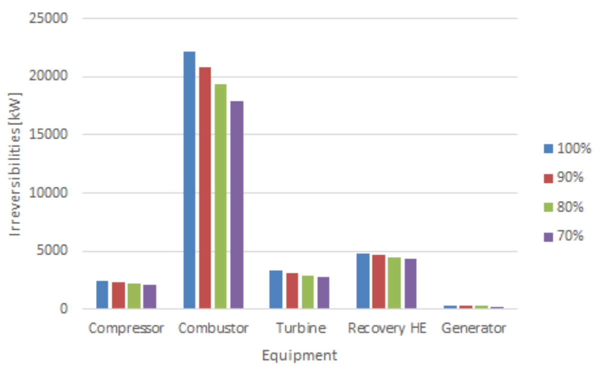

Figure 4. Irreversibilities of equipments - GT + WHR

As it is shown in Figure 5, for a constant thermal demand, the higher the electrical demand, the lower the energetic efficiency and the higher the exergetic efficiency. The reason for that behavior is that, as Figure 6 would tell: $\left|\frac{\partial \eta_T}{\partial Load}\right| > \left|\frac{\partial \eta_e}{\partial Load}\right|$, but $\left|\frac{\partial \epsilon_T}{\partial Load}\right| < \left|\frac{\partial \epsilon_e}{\partial Load}\right|$. That is a real important result, because the higher exergetic efficiency leads to a lower energetic efficiency for the same type of disposal. The reason for that comes from the fact that the water energy is high, but its exergy is low.

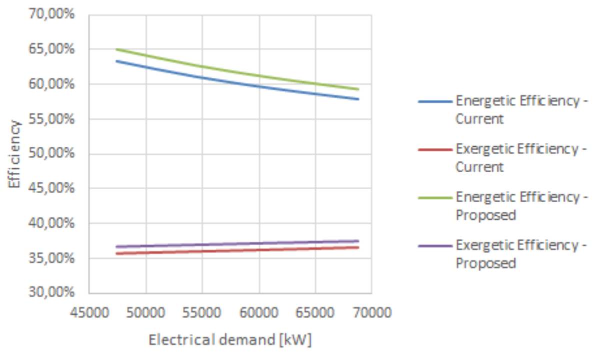

Figure 5. Efficiency relative to electrical load

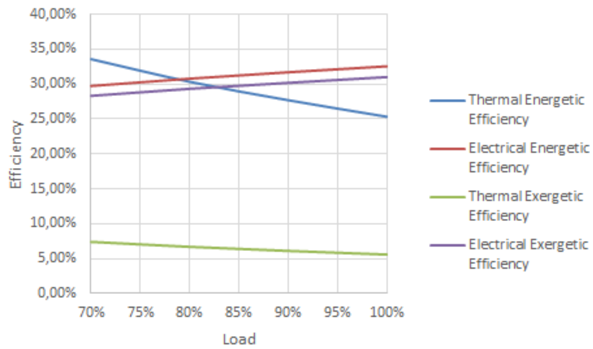

Figure 6. Efficiency relative to turbine load

As it can be seen, the energetic efficiency ranges from 57.91 % (100 % load for the current case) to 65.05 % (70 % load for the proposed case), while the exergetic efficiency is more stable.

3.2 Thermoeconomic analysis

The monetary cost of the flows, for the same case as in the previous section, was chosen to illustrate the thermoeconomic. It can be noticed in Figure 7 that, again, flow 8 has the highest cost, in monetary terms. However, flow 8 is not a final product, so the costs that have to be looked at are flows 11, 13 (exhaust gases), 15 (hot water) and 22 (net power). Until this moment the exhaust gases' cost has not yet been allocated to the products.

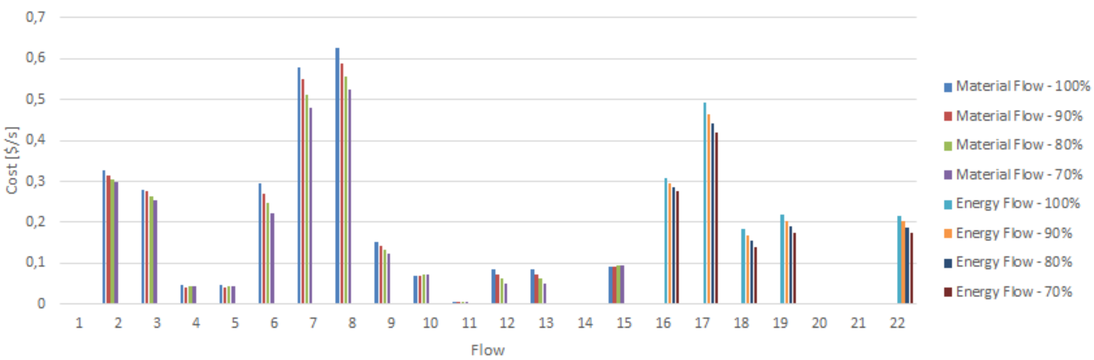

Figure 7. Monetary costs - GT + WHR

By allocating the losses due to the exhaust gases, the results are shown in Table 5.

Raising the costs was expected, specially for the electrical costs, with a much higher exergy. Special attention must be paid to lowering those values.

Table 5. Allocation of products - GT + WHR

Cost	Before allocation	After allocation
Electrical [$/MWh]	34.08	48.08
Thermal [$/GJ]	9.74	13.36

4. STUDY CASE

The study case is based on a specific electrical demand: 68.8 MW. The thermal demand is 53.5 MW. The solution through the current case involves three GT + WHR at 100 % load, while the proposed case involves two GT + WHR at 100 % load and one GT at 91.66 % load, as explained previously. The main results are presented ahead.

As it can be seen, the losses reduction is really affecting the system, as the costs of the losses are immediately allocated to the products, so costs will be higher for that reason in the current case. Also, fuel and irreversibilities are lowered for the proposed case, since the GT is more efficient; that also affects the energetic and exergetic efficiency of the system, and the emissions, which are smoothly improved. The energetic efficiency goes from 57.91 % to 59.37 %, while the exergetic efficiency goes from 36.60 % to 37.49 %.

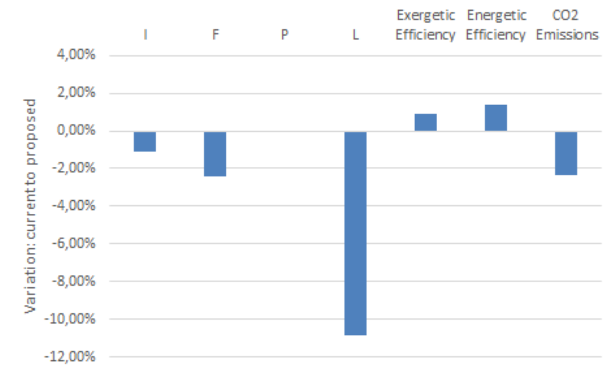

Figure 8. General improvement

Through Figure 9, in a relative basis, it can be seen that when the natural gas cost is increased, the thermal cost gains through the proposed case decays, while the electrical cost goes in the opposite direction. However, when it comes to the total cost, the thermal cost gain reduction prevails, which means that the total cost is less improved as natural gas prices go higher. Even so, in absolute terms, the total cost is continuously reduced as natural gas prices go higher.

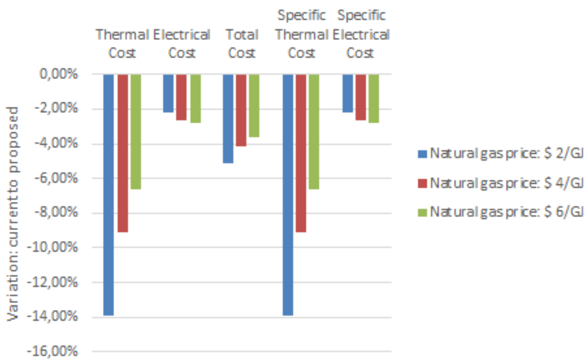

Figure 9. Cost Variation

5. CONCLUSION

The cogeneration plant of a Brazilian off-shore oil platform was analyzed, and exergetically and thermoeconomically characterized successfully through a series of well-known methodologies found in the literature.

There are many more ways to work thermoeconomically with this plant: by dividing exergy between chemical and physical exergies, or narrowing it down a little bit more, by diving physical exergy between thermal and mechanical exergies. This could bring up a more precise result, as for instance, in the compressor case: the main product of the compressor is the mechanical exergy of air, and its thermal exergy is a by-product; from the current methodology there are no differences among them, so their cost is unique. The methodology found in Tsatsaronis and Pisa (1994), would suggest that the average cost of adding thermal exergy would be the same as adding mechanical exergy, but their cost would be totally different.

As it could be seen in the study case, the power and heat generation is still lacking in efficiency. This occurs because of the fact that the thermal demand is considered constant and independent, and for the study case presented too low compared to the electrical demand.

The combustor was the highest irreversibility generation equipment, as it would have been expected, because of combustion. But the recovery heat exchanger has also shown great irreversibility generation, and specially, low exergetic efficiency. The heat exchanger flows (exhaust gases and process water) are very distinct and cannot rationally interact. Since there is no property demand for process water, as it serves as an intermediary fluid for heating all the systems, its average temperature should be fit into irreversibility generation minimization.

The products' cost allocation proved to be really important, and the exergetic criteria is found in the literature to be reliable and the most widespread because of its results' quality.

Natural gas price has shown great importance in the overall economic performance of the system. Imported gas is, therefore, not an attractive alternative for low electrical costs, unless the cost of production is higher than the importing cost, which is usually not the case.

Two interesting alternatives that should be taken into consideration to improve the overall performance are: 1) install a regenerator into the system, in order to pre-heat the air entering the combustion chamber; 2) install a steam power system in a combined cycle to recover the exhaust gases' exergy. Preliminary calculation have shown, for the first option, that the electrical efficiency could be raised from 6 to 8 % through regeneration, even though the turbine pressure ratio had to be lower, for the exhaust gases to be capable of pre-heating the air. For the second alternative, space could be allocated for the steam turbine in the hull of the FPSO, if the project allows it, and efficiency gains would be even higher.

6. ACKNOWLEDGEMENTS

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