24th ABCM International Congress of Mechanical Engineering
December 3-8, 2017, Curitiba, PR, Brazil

COBEM-2017-1121
ENERGETIC AND EXERGETIC ANALYSIS OF SUPERCRITICAL CO_2, ORGANIC RANKINE AND KALINA CYCLE FOR WASTE HEAT RECOVERY IN A FPSO UNIT

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Abstract. *This article aims to study various power generation alternatives to improve energy use in an FPSO along different points of its operation. Waste heat recovery is a promising alternative mainly due to its capacity of energy generation with low and intermediate heat sources, allowing to profit waste heat of industrial processes. The heat source used will be the exhaust flue gases of the gas turbines. Supercritical* CO_2*, Kalina and Organic Rankine (ORC) are analyzed and compared from an energetic and exergetic point of view. Thermoflex® and EES are used to develop thermodynamic models, with the scope of studying simultaneously maximum energetic and exergetic efficiency, along with obtained net power. Simulations are realized in identical conditions, in order to compare the different technologies, its vantages and disadvantages. Two production periods were considered. The year 15, in which there is large oil and gas production and the year 18, in which the oil production is declining and the volume of water brought by petroleum is 50% of total. Obtained net power for the regenerative ORC case is equal to 7.8% of total power demand in the first period. In second period only two gas turbines remain operative and the obtained power was lower due to the inferior energetic availability, this result corresponded to 5.0% total required demand for this period.*

Keywords: *ORC, Supercritical* CO_2*, Kalina, FPSO, Exergetic analysis.*

1. INTRODUCTION

In the last decade more offshore platforms have been installed in Brazil due to the increasing exploitation of oil and gas. FPSO units (Floating Production Storage Offloading) have proven to be cost-effective for deep water exploitation. Electricity generation in the platform is performed by an electricity module which cover the demand in all oil production periods. The electricity module is composed by a group of four gas turbines, one of them remains in stand-by, and the remaining three are in charge of the electricity generation. Fuel gas is produced in the platform, and along with the atmospheric air, serve as inputs for the gas turbines which can be turned on or off depending on the production demand characteristics.

According to Worldbank (2014) the population of the planet in the year 2000 was 6 billion inhabitants. Since then, the population has grown by 1% per year and is forecast to reach 8 billion people by the year 2025, which is related to rising demand for electricity and consumption of fossil fuels. Despite the evolution in industrial processes and the technologies involved to improve the quality of life of people, great impact has been generated to the planet due to emissions of gases harmful to humans and the planet.

Energy efficient technologies have been encouraged due to the increasing concern of greenhouse gases emissions. These innovations allow increasing efficiency and cost-effectiveness of industrial processes, and at the same time establishing environmentally friendly solutions. In this context, using waste heat recovery have been scope of several alternatives around the globe in order to generate electricity without necessarily burning additional fossil fuels. The scope of this article is limited to Organic Rankine Cycle (ORC), addressed in works of Mondejar et al. (2017), Suarez de la Fuente et al. (2017), Song et al. (2015); Kalina cycle, studied in Larsen et al. (2014), Nguyen et al. (2014), Junior et al. (2016) and Supercritical CO_2 cycle, detailed in Ahn et al. (2015), Tahmasebipour et al. (2014).

2. PROPOSED SYSTEMS

The production phases in an oil platform have a variable electric energy demand over the years and therefore, the generation system must be designed in a way that meets all production periods. The hot water system is also an important process in the platform, being required by fuel-gas, separation and dehydration systems. Table 1 shows the number of active turbines and the total demand required in each period, as well as the values of mass flow, temperature, pressure and exergy of a portion of the gas turbine exhaust gases that will not have any heat recovery purpose in the process, ie they will be discharged directly through the chimneys after the heat recovery by the hot water system. These data will be used as input variable in the simulation for later comparison between the cycles for waste heat recovery.

Table 1 – Information on the portion of exhaust gases from the gas turbines directed to the chimney in each period of oil exploration.

Period	Active Turbines	Total Demand [MW]	Mass flow $\dot{m}_g$ [kg/s]	Temp. [°C]	Pressure [kPa]	Physical exergy [kW]	Chemical exergy [kW]
Year 3	3	63.74	35.54	500.0	102.7	7476.2	390.3
Year 12	3	65.00	36.10	500.8	102.7	7822.8	412.3
Year 15	3	62.97	38.15	499.1	102.5	7992.4	415.5
Year 18	2	42.89	16.58	502.8	104.2	3537.7	183.8
Year 21	2	40.70	23.20	499.2	103.6	4881.3	248.9

The composition of exhaust gases is of paramount importance for determining their properties in each thermodynamic state. Table 2 shows the compositions of the exhaust gases for each period of operation of the platform. Note that the parameters are very similar, which would result in very close conclusions. Therefore, only two extremes will be analyzed: the maximum exergy available period (Year 15) and the minimal exergy available period (Year 18).

Table 2 – Average molar composition of the exhaust gases for each period of exploration.

Period	N_2 [%]	O_2 [%]	H_2O [%]	CO_2 [%]	Ar [%]
Year 3	75.21	14.31	6.40	3.17	0.91
Year 12	75.20	14.27	6.43	3.19	0.91
Year 15	75.22	14.34	6.37	3.16	0.91
Year 18	75.20	14.28	6.43	3.19	0.91
Year 21	75.24	14.40	6.33	3.13	0.91

2.1 Organic Rankine cycle (ORC)

The organic Rankine cycle is similar to the conventional Rankine cycle, except that the work fluid is an organic fluid, which has a low boiling point and latent heat of vaporization lower than that of water, which allows its evaporation at lower temperatures making it interesting for heat recovery from relatively low temperature sources. Figure 1 A) shows the simple ORC and Figure 1 B) presents the regenerative ORC adopted in the simulations.

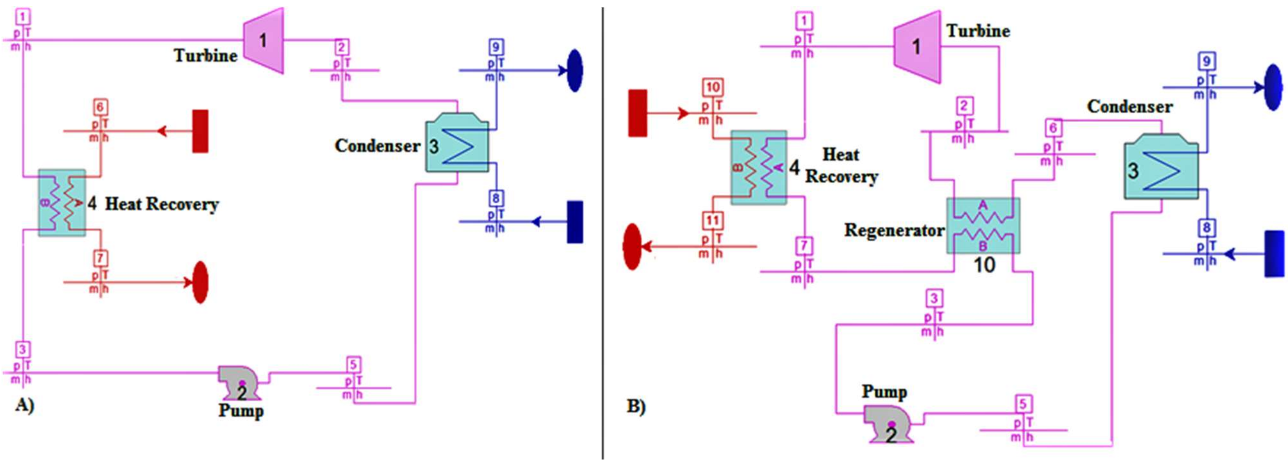

Figure 1. A) Simple ORC cycle B) Regenerative ORC cycle.

Siddiqi & Atakan (2012) investigated hydrocarbons from n-pentane to n-dodecane in comparison to water, benzene and toluene for application in organic Rankine cycle for waste heat recovery of gas turbine. It has been found that if the heat source is at higher temperature, 500°C, n-dodecane and toluene are suitable fluids. Therefore, toluene will be the working fluid used in the study.

2.2 Supercritical CO_2 cycle.

The main advantages of using the supercritical CO_2 cycle are related to the reduction of turbomachinery. As the system operates above the critical point, the minimum pressure is higher when compared to the conventional Rankine and Brayton cycle, causing the fluid density to remain high throughout the power system, resulting in lower volumetric flow, which reduces the size of turbomachinery compared to conventional cycles. On the other hand, the pressure ratio of the supercritical cycle is much lower than the Rankine steam cycle and the outlet temperature in the turbine is relatively high, making it interesting to recover heat from a regenerator, which has great influence of the system efficiency (Ahn et al., 2015). In addition, CO_2 is less corrosive, non-flammable, non-toxic, has low cost and is abundant, characteristics that makes it interesting in this type of application. The configuration of the proposed cycle is shown in Figure 2.

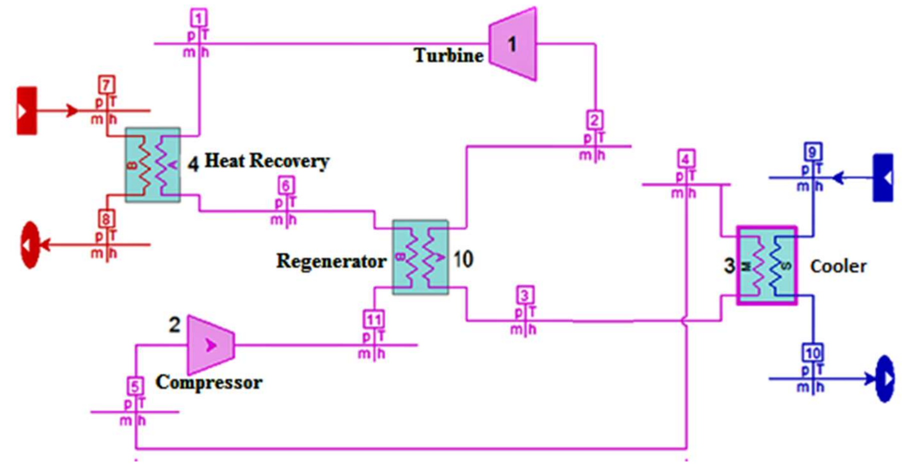

Figure 2. Supercritical CO_2 cycle.

2.3 Kalina cycle.

This cycle is basically a Rankine cycle except that the work fluid is a Water-Ammonia mixture. According to Ganapathy (2002) this cycle has the potential to be 10 to 15% more efficient than the Rankine cycle and conventional materials are used in its construction, making the technology feasible. Furthermore, in the Kalina cycle the heat is added and rejected with the temperature varying, as the water-ammonia mixture changes phase with variable temperature, which makes the temperature of the hot source and that of the work fluid closer, that is, reduces the irreversibility of the heat exchange process. Figure 3 shows the configuration of the proposed Kalina cycle.

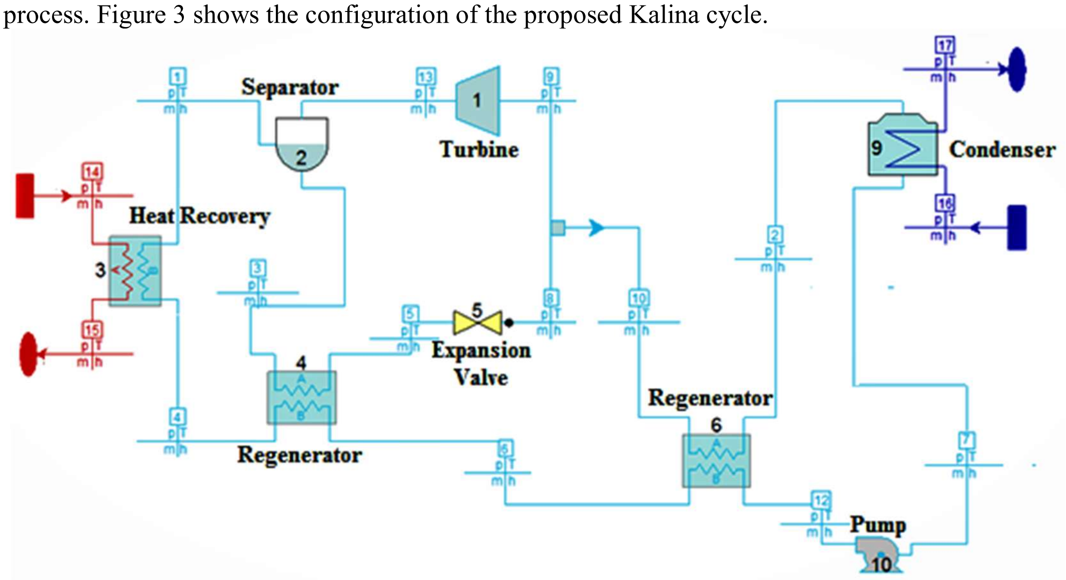

Figure 3. Kalina cycle.

2.4 Considerations and modeling of thermodynamic cycles.

The energy analysis was performed in Thermoflex® software, in which thermodynamic models of each cycle were developed to perform the simulations with various configurations and operating conditions. Each thermophysical state of the cycles are exported to the EES® software (Engineering Equation Solver) to develop the exergetic analysis. To follow a pattern of analysis between the models some considerations are made in the simulations:

- Condenser cooling fluid: Water
- Fluid enters the pump in the state of saturated liquid;
- Pressure drop disregarded in the heat exchange equipment;
- Pump and turbine efficiency: 85%;
- Effectiveness of the heat exchange equipment: 90%;
- Minimum Pinch Point of the heat exchange equipment: 5°C;
- ΔT of the condenser cooling water: 10°C;
- Condensation temperature in all cycles: 40°C.

The ideal gas mixture enthalpy, h_{mix}, is described in Nellis and Klein (2012) as the weighted average of the gas fractions at the temperature of the mix. The calculation of the ideal gas mixture entropy, s_{mix}, is based to partial pressure of each constituent of the blend, which is proportional to its molar fraction for an ideal gas model. The enthalpy and entropy of gas mixture are obtained from Eq. (1) and Eq. (2).

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

$$s_{mix} = \sum y_i s_i - R \sum y_i \ln y_i \tag{2}$$

Principles of mass, species and energy conservation are applied to a steady state control volume in each cycle equipment, neglecting the effects of kinetic and potential energy, as presented in Eq. (3) and Eq. (4).

$$0 = \sum_{vc} \dot{m}_{in} - \sum_{vc} \dot{m}_{out} \tag{3}$$

$$0 = \dot{Q}_{vc} - \dot{W}_{vc} + \sum_{vc} \dot{m}_{in} h_{in} - \sum_{vc} \dot{m}_{out} h_{out} \tag{4}$$

Pump/compressor, $\eta_{C(B)}$, and turbine efficiency, η_T, are modeled as a function of the isentropic and real enthalpies of the equipment, according to Eq. (5) and Eq. (6).

$$\eta_T = \frac{h_{in} - h_{out,r}}{h_{in} - h_{out,s}} \tag{5}$$

$$\eta_C = \eta_B = \frac{h_{out,s} - h_{in}}{h_{out,r} - h_{in}} \tag{6}$$

The total exergy of a system, ex^t, is the sum of the physical exergy, ex^{ph}, and chemical exergy, ex^{ch}, as presented in Eq. (7) in the specific form. In the organic Rankine cycle and supercritical CO_2 cycles there is no change in the composition of the work fluid (wf), that is, the specific physical exergy is sufficient in each thermodynamic state and is given by Eq. (8). In the Kalina cycle the composition changes at each point of the system, because it is a water-ammonia mixture, and therefore besides the physical exergy, it is necessary to account for the chemical exergy as presented in Eq. (9), where z is the mass fraction of ammonia, M is the molar mass of the substance and e_{ch}^{0} is the standard chemical exergy for water and ammonia, and their values are given in Ahrendts (1980).

$$ex^t = ex^{ph} + ex^{ch} \tag{7}$$

$$ex^{ph} = (h - h_0) - T_0(s - s_0) \tag{8}$$

$$ex^{ch} = \left(\frac{z_i}{M_{NH_3}}\right) e^0_{ch,NH_3} + \left(\frac{1 - z_i}{M_{H_2O}}\right) e^0_{ch,H_2O} \tag{9}$$

The exergy balance presented in Eq. (10) is applied to each equipment individually to obtain its exergy destruction, $\dot{E}_d$.

$$0 = \sum_j \left(1 - \frac{T_0}{T_j}\right)\dot{Q}_j - \dot{W}_{vc} + \sum_{cv}(\dot{m}.ex^t)_{in} - \sum_{cv}(\dot{m}.ex^t)_{out} - \dot{E}_d \quad (10)$$

In the presence of irreversibility, exergy is destroyed. Second-law efficiencies measure losses in exergy during a process. The general second-law efficiencies, ε_{II}, is defined as Eq. (11).

$$\varepsilon_{II} = 1 - \frac{\text{Exergy destruction}}{\text{Exergy input in equipment}} = 1 - \frac{\dot{E}_d}{\Delta(\dot{m}.ex^t)_{wf}} \quad (11)$$

3. RESULTS

As already mentioned, only the periods of maximum exergy (year 15) and minimum exergy (year 18) will be presented in the results. The condensation temperature of the cycles is set at 40 °C, which sets the condensing pressure for each cycle. Then, two parameters of the cycles were changed simultaneously: the upper pressure and the mass flow of the cycle but fixing the exhaust gas inlet conditions in order to obtain the point of maximum power and efficiency. The results are shown as a function of the ratio between exhaust gas mass flow, $\dot{m}_g$, and cycle working fluid mass flow, represented by Ψ.

3.1 Year 15 – Maximum Available Exergy.

The maximum available exergetic is in year 15, whose gas temperature in the chimney is 499.1 °C and mass flow rate is 38.15 kg/s. The organic Rankine cycle regenerative (ORC R) achieves maximum net power and efficiency, but not too far from the simple organic Rankine cycle (ORC S), which reaches net power close to the regenerated cycle but with lower thermal efficiency, as shown in Table 3. One disadvantage of ORC is the large water mass flow rate in condenser, $\dot{m}_{cond}$, for cooling of the cycle working fluid, which directly impacts in size of the equipment. This happens because working fluid exits the turbine with the high temperature and enters in condenser still in the superheated state, being necessary its cooling until 40 °C for later change of phase. By recovering waste heat ORC R obtains 7.8% in relation to total power demand in year 15, accompanied by a 13.2% higher thermal efficiency compared to the Kalina cycle.

Table 3 – Summary of the results for the maximum power point. (Year 15)

Cycle	Fluid	Net Power [kW]	Cycle Efficiency [%]	Upper pressure [bar]	Lower pressure [bar]	ψ	$T_{out,HR}$ [°C]	$\dot{m}_{cond}$ [kg/s]
ORC R	Toluene	4920.7	32.5	35	0.08	1.55	137.2	243.7
ORC S	Toluene	4751.7	26.4	35	0.08	1.66	65.3	316.6
sCO_2	CO_2	2877.9	27.7	300	73.89	0.89	252.9	180.1
Kalina	H_2O/NH_3	2945.0	19.3	100	12.95	3.18	134.2	293.9

It was noted from the simulations, that to reduce value of Ψ also causes reduction of the exit temperature in gases of the heat recovery, $T_{out,HR}$, (65.3 °C for the ORC S in the maximum power point), which will cause the condensation of the water vapor present in the gases, which can result in highly corrosive and oxidizing substances, such as carbonic acid, damaging the heat exchanger. The maximum organic fluid temperatures in the ORC S is 318.2 °C and 311.0 °C for the ORC R, well below the autoignition temperature of the Toluene (480 °C according to Ldpi-inc (2017)), which makes it feasible for the conditions studied.

The condenser is the equipment with the lowest exergetic efficiency in all cycles analyzed. In ORC R, the heat recovery and turbine equipment are the main destroyers of the work potential, as shown in Figure 4 a). In ORC S, the heat recovery and condenser are responsible for the great portion of exergy destruction, as shown in Figure 4 (b). The absence of regenerator to take advantage of the output exergy of the turbine to preheat the working fluid at the inlet of the heat recovery causes accentuated exergy destruction in condenser. In sCO_2 cycle the cooler is responsible for much of the cycle exergy destruction along with the turbine, as shown in Figure 5 a). In Kalina cycle, the heat recovery is responsible for the greatest amount of cycle exergy destruction, followed by the condenser, as shown in Figure 5 b). Therefore, it is verified that the heat recovery unit is responsible for the destruction in working potential in the cycles, except for the supercritical cycle in which the condenser is the most irreversible.

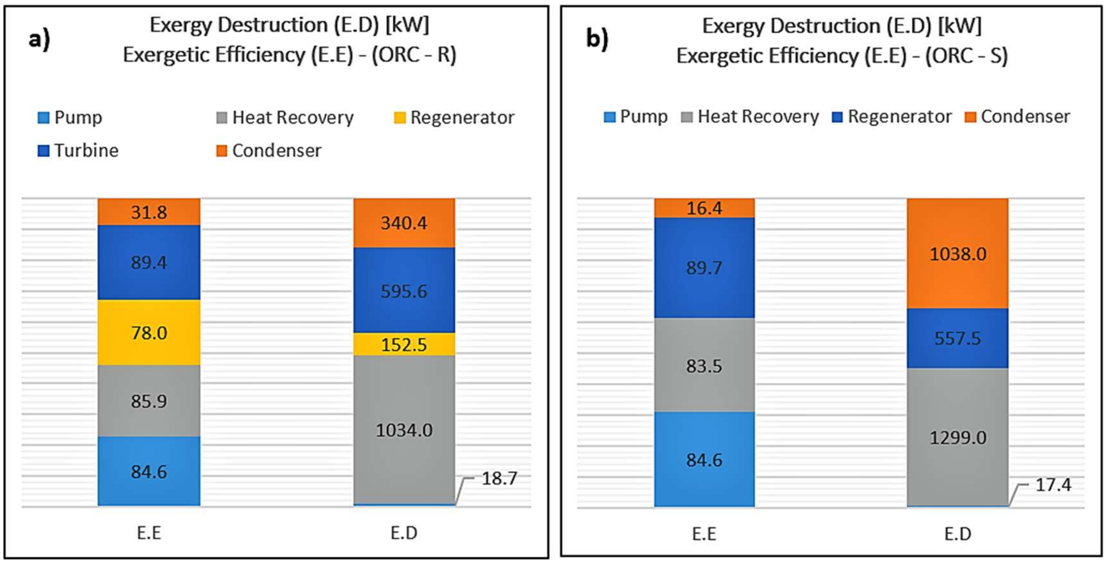

Figure 4. Exergetic efficiency and destroyed exergy in the main equipment in ORC R and ORC S cycle - year 15.

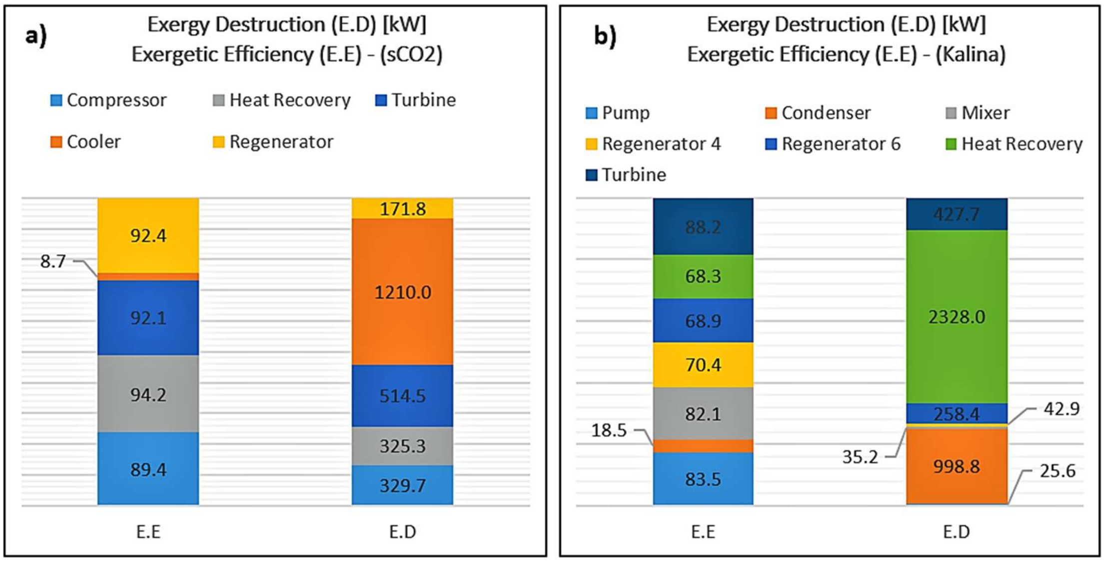

Figure 5. Exergetic efficiency and destroyed exergy in the main equipment in sCO_2 and Kalina cycle - year 15

Figure 6 shows the exergetic efficiency of the cycles and their corresponding exergetic destruction to the maximum power point. This way is possible to be more comprehensive with regard to the maximum useful work possible of each cycle. It can be observed that the Kalina cycle is the one with the lowest exergetic efficiency, that is, it presented less potential of utilization of the available exergy, resulting in the cycle with greater exergy destruction. In addition, Kalina cycle has more equipment than other cycles, which directly affects the space and weight on the FPSO platform. The regenerative organic Rankine cycle is 26.4% more exergetically efficient than the Kalina cycle, therefore, its exergy destruction potential becomes reduced in relation to the other cycles.

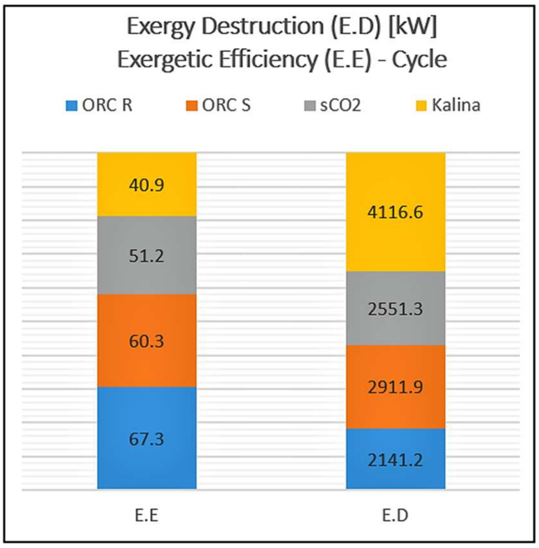

Figure 6. Exergetic efficiency and destroyed exergy in cycles - year 15

3.2 Year 18 – Minimum Available Exergy.

The minimum exergetic availability is in year 18, whose hot source temperature is 502.8°C and the mass flow rate of gases in the chimney is 16.58 kg/s. Table 4 shows the results relative to the maximum net power point obtained in the simulations. In relation to case of maximum available exergy there is a reduction in net power in all cycles due to the reduction in exergetic availability, but it is again verified that the ORC S is problematic due to the reduced temperature of the gases, $T_{out,HR}$, in the outlet of heat recovery. The ORC R presents the highest net power and efficiency, in addition, this system operates at relatively low pressures, which adds another advantage in its use. The maximum temperature of the toluene was 322.6°C in ORC S cycle and 315.1°C in ORC R cycle. The Kalina cycle works with higher pressure than the ORCs and again is not interesting as it provides net power 39.8% lower than the ORC R. The same happens with the supercritical CO_2 cycle, in which the work pressure is higher than all other cycles and results in net power 41.3% lower than the ORC R.

Table 4 – Summary of the results for the maximum power point. (Year 18)

Cycle	Fluid	Net power [kW]	Cycle efficiency [%]	Upper pressure [bar]	Lower pressure [bar]	ψ	$T_{out,HR}$ [°C]	$\dot{m}_{cond}$ [kg/s]
ORC S	Toluene	2080.6	26.4	35	0.08	1.67	65.51	138.8
ORC R	Toluene	2159.6	32.9	35	0.08	1.57	142.1	105.1
sCO_2	CO_2	1267.7	28.7	300	73.89	0.93	263.3	75.1
Kalina	H_2O/NH_3	1300.1	19.5	100	12.95	3.32	135.7	128.4

The condenser again shows the equipment with less exergetic efficiency in all the cycles. The heat recovery in ORC R cycle is responsible for the greatest amount of exergy destruction, as shown in Figure 7 a). In ORC S, the heat recovery and the condenser are responsible for the exergy destruction in cycle, as shown in Figure 7 b). In the supercritical CO_2 cycle the cooler has a high exergy destruction, according to Figure 8 a). For the Kalina cycle the heat recovery is responsible for the exergy destruction, followed by the condenser, according to Figure 8 b).

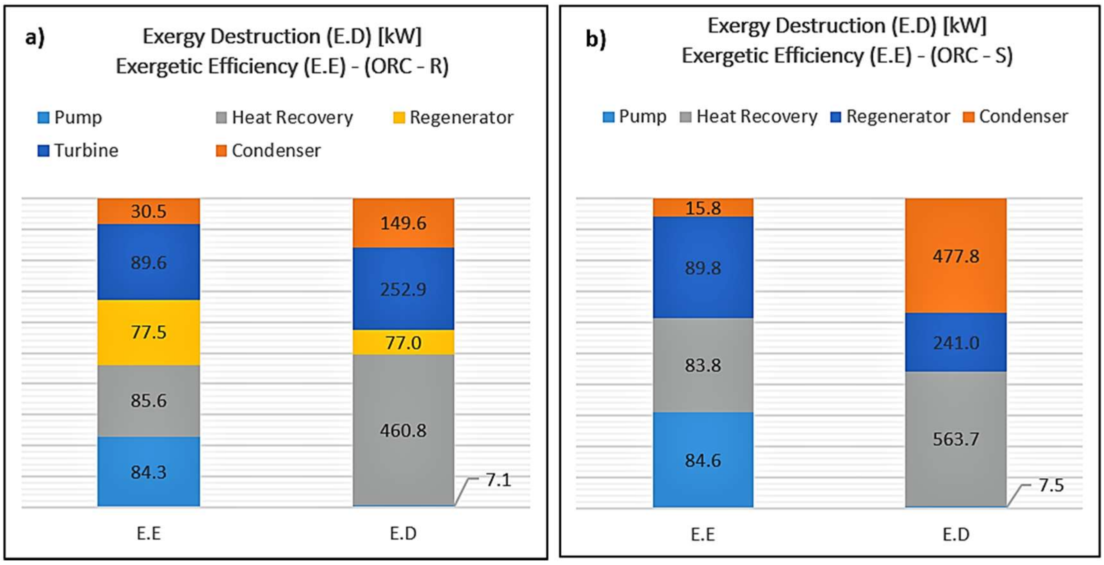

Figure 7. Exergetic efficiency and destroyed exergy in the main equipment in ORC R and ORC S cycle – year 18.

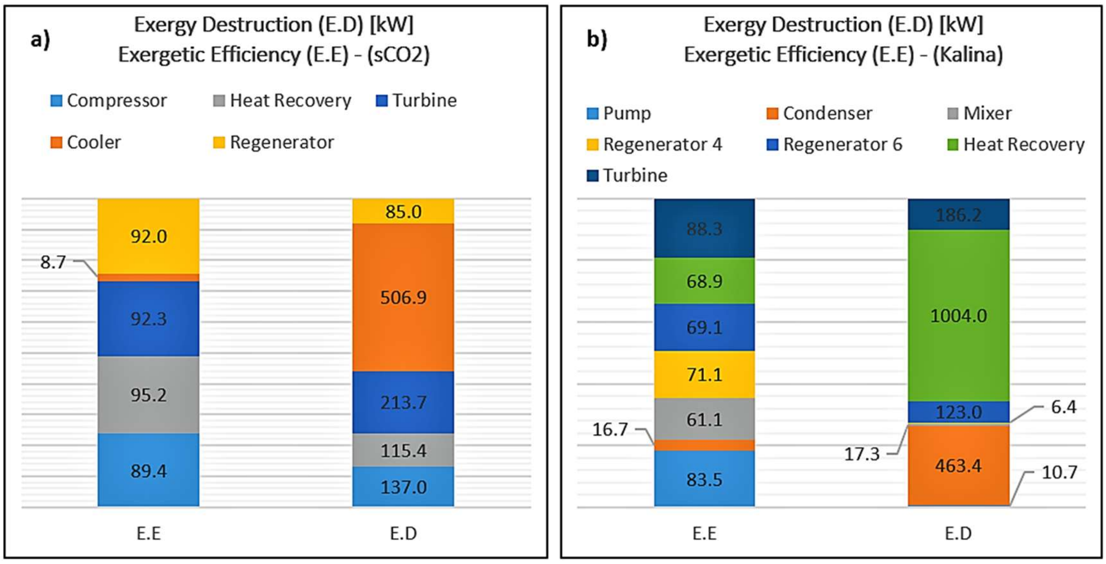

Figure 8. Exergetic efficiency and destroyed exergy in the main equipment in sCO_2 and Kalina cycle – year 18.

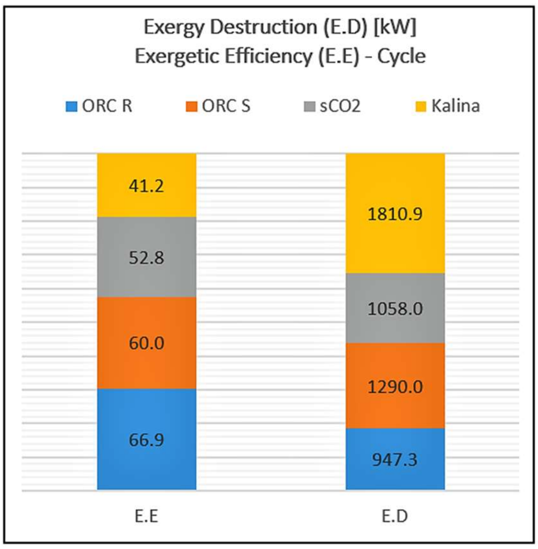

Figure 9. Exergetic efficiency and destroyed exergy in cycles – year 18.

Figure 9 shows an overview of cycles in terms of exergetic efficiency and exergy destruction. It is noted that the Kalina cycle is the one that has lesser exergetic efficiency and is responsible for the greater exergy destruction. The regenerative organic Rankine cycle presents the best results for exergetic efficiency and for exergy destruction. The ORC R is 25.74% more exergetic efficient than the Kalina cycle leading to a reduced exergy destruction.

4. CONCLUSIONS

Faced with the need for energy recovery in FPSO platforms, the possibilities of recovering residual energy from the existing processes in the platform through unconventional thermodynamic cycles were evaluated. Due to weight and space limitations on such platforms, simpler cycles were chosen without large amount of equipment. In addition, the simulations were performed under identical conditions, allowing the comparison of the different technologies, their advantages and disadvantages.

The organic Rankine cycle proved to be superior for energy recovery even in its simplest configuration. When a regenerator is inserted in this cycle, it is possible to obtain higher net power values and increase its thermal efficiency in relation to the other cycles. It is possible to obtain 41.5% more net power from the regenerative ORC in relation to the supercritical CO_2 cycle. The supercritical cycle has a lower net power than the organic cycles but practically equates to the Kalina cycle, however with higher thermal efficiency. However, these cycles operate at relatively high pressures, which bring a drawback from the point of view of equipment design. The Kalina cycle is not superior in any of the analyzed cases. In addition, a relatively high work pressure and a greater quantity of equipment is required, which can impact factors such as cost and physical space for installation in the FPSO platform.

According to exergetic analysis of each cycle equipment, the heat recover and the condenser are responsible for destruction of a great portion of the potential to perform work, requiring a greater attention in these equipment to reduce their irreversibility and increase the exergetic efficiency. In general, the regenerative organic Rankine cycle can better exploit the exergy available in the heat source, obtaining exergetic efficiency around 67%, while the Kalina cycle presented an exergetic efficiency approximately 26% lower, resulting in the cycle with the worst potential to perform work.

Finally, the organic Rankine cycle is an interesting alternative for waste heat recovery in the FPSO oil platform, obtaining net power corresponding to 7.8% of the total power demand in the period of maximum available exergy and 5.0% in relation to the total demand of the period of minimum exergy available.

5. ACKNOWLEDGEMENTS

The authors would like to thank BG E&P Brazil (BG-20) for the financial support to the project "BG20: Energy efficiency and CO_2 emissions reduction for petroleum maritime operations" and the information supply about FPSO. This paper also has the financial cooperation of CNPq Brazilian research agency.

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