EVALUATION OF A KALINA CYCLE ON A BRAZILIAN FPSO TO RECOVERY WASTE HEAT AND A COMPARASION WITH AN ORC

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Abstract. *Recovery waste heat in any process is very important to reach the sustainable development. Through the First and the Second Law of Thermodynamics, it is possible to research the performance of waste heat recovery thermodynamic cycles. This research studies the Kalina cycle (KCS-34) with that point on a Brazilian FPSO - Floating Productions Storage Offloading. The one transforms thermal energy in shaft power. This cycle usually works with low and average heat sources because works with a mixture of ammonia-water that has low vaporization temperature. The concentration of the mixture is variable in different parts of the system to decrease the thermodynamic irreversibility and increase the global thermal efficiency of the FPSO. This work will apply the waste heat of the fourth stage s CO_2 Compression Unit as the heat source to rate the potential of the Kalina cycle to recovery heat. Lastly, it will be compared with an ORC ó Organic Rankine Cycle that used the same source to compare the results. The commercial software Aspen HYSYS v8.6® is used, as a tool, to simulate the cycle and the activity coefficient model used is the non-random two-liquid (NRTL). The maximum power output gain together with the minimum area was obtained with the higher ammonia mass fraction, lower pinch point, higher terminal temperature differential and 4400kPa of the working pressure, but the cycle not showed a big production of shaft power. However, it can contribute to increase the efficiency of the compression process.*

Keywords: Kalina Cycle, Waste heat, FPSO.

1. INTRODUCTION

The economic growth is strongly connected to productivity and technology. Therefore, there is the necessity of a technological development, mainly related with the energy conversion to respond the demand of the secondary sector and impulse the economic growth. It is necessary to balance the development with sustainability, it means, to demand the unification between economy and ecology. It is important to reflect and act in a different way to get another ways to produce energy.

This structure has helped the growth, focusing on the increasing of the energetic efficiency of the industrial process throughout the development of the equipment and reduction of the energy (waste heat) throw in the environment without any use searching optimization in the use of fossil fuel. This increasing can play an important role in the generation of electric energy in a way to answer new patterns in the environmental rules. It is in this context that this work is done, emphasizing in a study of a thermodynamic cycle to recover the heat in a Brazilian FPSO unit.

In offshore units due to the distance and an isolation from the coast, the mechanic and electric energy is generally produced in the own unit through the burning of fossil fuel available to the activation of gas turbines and internal combustion engines. In general, the thermic efficiency of the productive process and generation of energy in these units are low. In industry, the waste heat can represent around 60 to 80% of the total energy refrained in the primary sources of used energy (Sotomonte, 2015).

The increase of the efficiency in offshore units through the use of the waste heat source in its productive process to generate energy can play an important role in the rentability of this industrial sector, in the reduction of the fossil fuel consume and the emission of environmental pollutant. Although the use of technologies of waste heat recuperation to generate energy is common in industrial installation onshore, these technologies are still not the reality in offshore ones, particular FPSO, due to restrictions associated with the cost, occupied space and building material.

The case study in this work is a Brazilian FPSO defined in Gotelip (2015). In that study, he assessed the availability of low-grade waste heat from different processes in a FPSO. The study of energy flow in this platform indicated the processes with the greatest potential for energy recovery. The CO_2 compression Unit at the FPSO has great potential for waste heat recovery and the fourth stage of the compression will be analyzed.

2. KALINA CYCLE⊘S DESCRIPTION

Created by Alexander Kalina in 1983, the Kalina cycle uses a mixture to generate shaft power. It is a system applied to change thermal energy in mechanical energy and usually works with low heat sources. The cycle uses a working fluid with at least two components, usually ammonia and water. The components⊘ concentration are variables in different parts of the system with the objective to decrease the thermodynamics irreversibility and to get the increase of the thermal efficiency. (Dippipo, 2007; Rodríguez, 2014)

The first central energy working with Kalina cycle was in USA in 1992, it was a trial central to recover the heat of the exhaust gas of gas turbine. Nowadays there are two plants in Japan that use this cycle, both to recover waste heat, and there are two others in Europe to produce energy from geothermal energy (Figueiras, 2011). These cases are show at the Figure 1.

Year of start	Place	Heat source	Electrical energy output
1992-1996	Canoga park ó USA (trial central)	Exhaust of gas turbine	3 MW
1999	Fukuoka ó Japan	Waste heat from incineration plant	5 MW
1999	Kashima steel works ó Japan	Waste heat of production (water 98ºC)	3,1 MW
2000	Husavik - Iceland	Geothermal brine at 124ºC	2 MW
2007	Germany	Geothermal energy	3,4 MW

Figure 1. Adapted from Figueiras (2011) - Kalina cycles applications all over the world.

There are some configurations of the Kalina cycle systems (KCS). The KCS 34, similar that is used in Husavik, is the most suitable for temperatures of the heat sources until 140ºC. If the heat source has temperature between 120ºC and 140ºC it is impossible to vaporize all the ammonia and water in the mixture, to have sure that just steam will enter on turbine it is necessary to use a separator. The KCS 11 is the most applicable for geothermal temperatures from 121ºC until to 204ºC. The system KCS 1-2 is useful for temperatures of the heat source between 200ºC and 400ºC, in this case that is not necessary the use of a separator before the turbine because all the mixture can be vapored (Zhang, 2012).

2.1 Ammonia-water mixture

The ammonia-water is a zeotropic mixture and its characteristics differ from the pure water or the pure ammonia. There are three main differences of the mixture and pure water or pure ammonia. The first is the variable boiling and condensation temperatures keeping the pressure, whereas pure ammonia and pure water have a constant boiling and condensation temperatures. The second one is the physical properties of the mixture can be altered, since keeping the thermophysical properties, through of change the concentrations of ammonia. The last one is these kind of mixtures have thermophysical properties that help to increase or decrease the temperature without change its content energy (Mlack and Mirolli, 2002).

The ammonia is the most volatile component of the mixture because its boiling temperature is lower than water, it means the ammonia will boil first. When the mixture is cooled, the water will condense first.

The phase diagram in the Figure 2 shows temperature versus the concentration of the ammonia solution at 550 kPa. It express what is necessary to know about the vaporizing or cooling process. For example, when the mixture begins to vaporize at the boiling point temperature, given by 3, the molar percent of ammonia is 70% and 30% of water. As the mixture continues in boiling, the temperature increases and point 4 are achieved, where in the concentrations of the remaining liquid and of the vapor formed are given by points 6 (ammonia-poor fluid) and 5 (ammonia-rich fluid). When the point 7 is reached, the mixture is saturated vapor at a temperature above the dew point and the vapor concentration is the same as the concentration of the liquid at the beginning of the evaporation process.

The percentages vapor and saturated liquid can be determined by the lever rule, so the steam fraction (*fr*) of at 70 % of ammonia mass fraction is given by Eq. (1).

$$f_{r_{vapor}} = \frac{f_{r_4} - f_{r_6}}{f_{r_5} - f_{r_6}} = \frac{0.7 - A}{B - A} \tag{1}$$
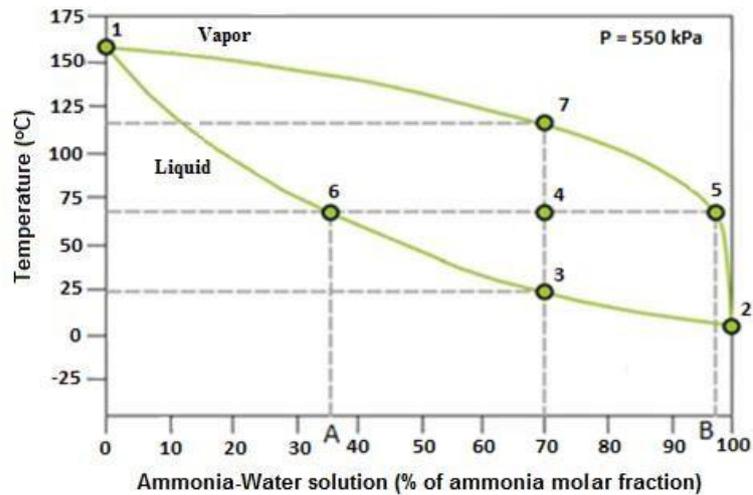
Figure 2 – Phase diagram to obtain the equilibrium for ammonia-water solution keeping the pressure.

The vapor fraction consisting mainly of ammonia can be directly seen in Figure 2 and has the value B. Regarding the liquid phase, the interpretation is similar and shown in Eq. (2).

$$
\begin{cases}
f_{r_{liquid}} = \dfrac{f_{r_5} - f_{r_4}}{f_{r_5} - f_{r_6}} = \dfrac{B - 0.7}{B - A} \quad \text{or} \\[2mm]
f_{r_{liquid}} = 1 - f_{r_{vapor}} \\
f_{r_{liquid}}^{NH_3} = A \\
f_{r_{liquid}}^{H_2O} = 1 - A
\end{cases}
\tag{2}
$$

If the ammonia–water mixture is heated, the ammonia to be more volatile tends to vaporize first than pure water. In the mixture, the ammonia concentrations of the remaining liquid decreases and the saturation temperature increases, providing a better use of the heat source than the constant temperature evaporation of a pure fluid (Rodríguez, 2014).
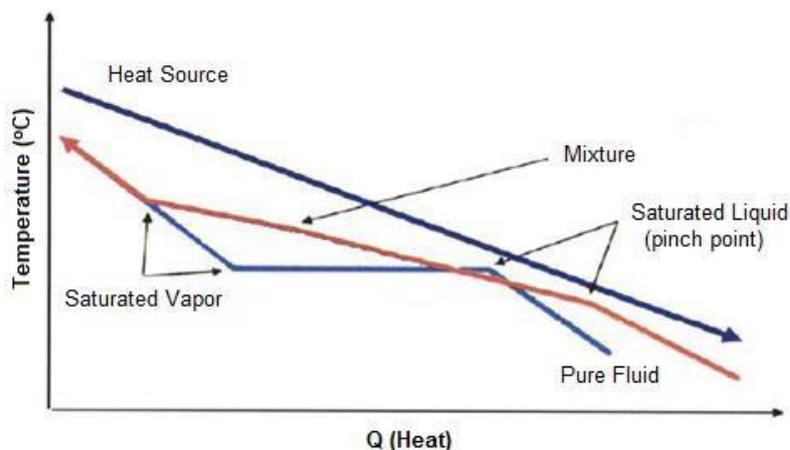

Figure 3 – Adapted from Rodríguez (2014) - Heat transfer diagram of a pure fluid and a zeotropic mixture.

The main concern for engineering application of the Kalina cycle focuses on the environmental and safety features of ammonia–water mixture. Ammonia has a smell and annoyance features, however they are very important to safety, because they are self-alarming. Fire and explosion hazards due to ammonia are very low. Ammonia will not support combustion after the ignition source is withdrawn. Finally, ammonia is gaseous at atmospheric pressure. It is much lighter than air (Zhang, 2012; Napoleão, 2003).

3. THERMODYNAMIC MODELING

An activity model, non-random two liquid (NRTL), was used to calculate the equilibrium of the phases liquid-vapor, liquid-liquid and vapor-liquid-liquid in Aspen HYSYS V8.6. It uses the statistic mechanic and the theory of liquid cell to represent the structure of liquid phase (Renon and Praunitz 1968).

A KCS-34 is adopted in this work and its flow diagram is showed in Figure 4. The heat source of the cycle is an exhaust gases from the fourth stageϕs CO_2 Compression Unit, which is paint of red, the composition and the physical properties of the gas are in the Table 1. The flow of mixture ammonia-water is painted of blue. The heated mixture (4) is separated in a separator in: ammonia-poor liquid (5) and saturated ammonia-rich vapor (6). The flow (5) is cooled (HT Recuperator) and depressurized in a throttle. The flow (6) expanded in a turbine to produce shaft power (W_t), and the turbine exhaust (7) is diluted with (17). After this, these ones are cooled (LT Recuperator) and condensed (Condenser) by cooling water (11 and 12). The saturated liquid mixture (10) is compressed to the working pressure (Pump) and heated in a low-temperature recuperator, high-temperature recuperator and finally the boiler (Evaporators).

Table 1 - Gas Composition (Gotelip, 2015).

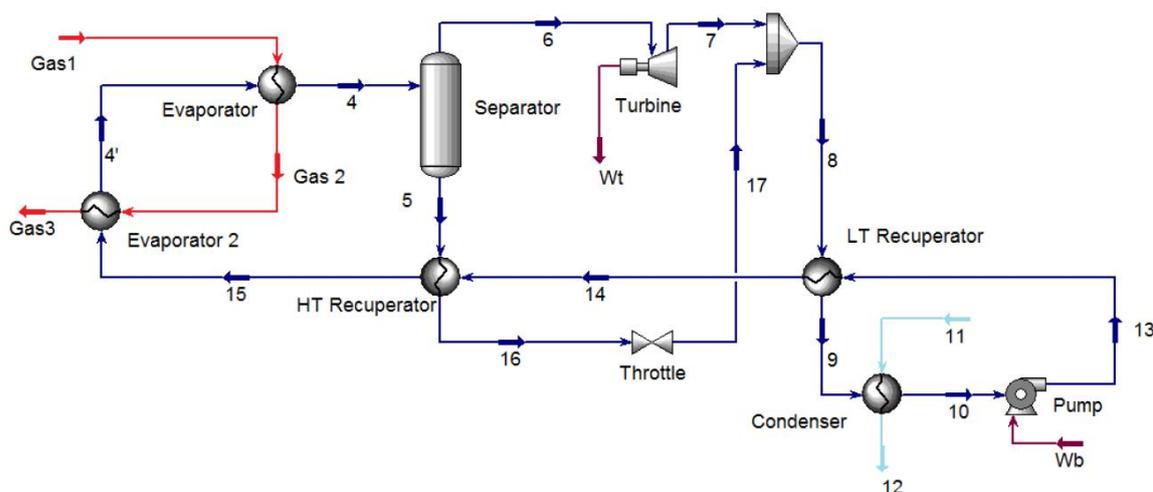| Component	Carbon dioxide	Methane	Ethane	Others
Composition[%]	70.8	27.5	1.2	0.5
Temperature [ºC]	135			
Pressure [kPa]	25110			
Mass flow [kg/s]	55.54			

Figure 4: KCS-34 on Aspen HYSYS V8.6.

If the heat source has low-temperature (Lakew and Bolland, 2010; Rodríguez and Lora, 2013), the design of the heat exchangers are very important. In this case, the amount of heat absorbed by the working fluid is critical to a good use of the heat source, and so there are three parameters to study: pinch point temperature (*PP*), terminal temperature differential (TTD) and working pressure.

The pinch point temperature is the minimum temperature difference between fluids (see Fig. 3) and is an important parameter generally fixed by most of researches (Zhang, 2012; Lakew and Bolland, 2010). A lower *PP*, requires a higher heat exchange area, which also increases the cost of the system. On the other hand, improve the heat exchange in the evaporator, which increases the output power. Several authors agree with the importance of the pinch temperature as a design parameter, essentially, for low temperature applications (Guo et al, 2011), generally assumed between 1 ó 6 K. In this paper is considered a variation of *PP* from 2 to 10 ^{o}C.

Terminal temperature differential (TTD) is the difference between the temperature of the hot source and the inlet temperature of the working fluid in the turbine. For this analysis is assumed a minimum TTD of 10ºC and a maximum of 30ºC. As the TTD increases for a given pressure, it reduces the quality of the vapor and an ammonia-rich vapor with higher concentration of ammonia in the mixture is obtained at the evaporator outlet, enabling work with higher mass flow rates in the system. However, this also increases the ammonia-poor flow that does not pass through the turbine to produce work, maximize the size of system components and induce more pump power consumption (Rodríguez, 2014).

The distillation process in KCS can happen, for an ammonia-water mixture, at different working pressures for the same boiling temperature. Therefore, it could obtain different mass flow of the working fluid and different concentrations of the mixture. It is very important to select the right pressure, if it is higher than the optimum the vapor in the evaporator outlet will have a higher quality with higher ammonia concentrations in the mixture, but the impact of the mass flow causes a decrease of power production. The same result can be achieve if lower values of working pressure are assumed due to a lower quality of ammonia steam is produced allowing it to increase the ammonia-rich flow rate, but the enthalpy drops in the turbine is reduced (Rodríguez, 2014; Zhang, 2012).

The analysis of the design points in the evaporator are made from a complete combinatorial analysis in order to study the different combinations that can be formulated in order to assess the possible combinations among the different design points and thereby determine which variables provide the best thermal performance of the system. This analysis is crucial to evaluate the influence of each parameter on the performance of the thermal system.

In order to simplify the model, pressure drops in the different components of the system, except turbine, are neglected. The number of variables the molar concentration of ammonia will be fixed in 85%. For each component are applied First and Second Law to determinate the produced or consumed work, the added or rejected heat, produced irreversibility of each component. In this work is took into account also the following considerations:

1. Ammonia molar concentration 85%;
2. Pinch point temperature variation between 2 ó 10 ºC;
3. Values of Terminal temperature differential: 10ºC, 20 ºC and 30 ºC;
4. Vapor quality in turbine exhaust up to 90 %;
5. The system operates in steady state, kinetic and potential energy changes are neglected;
6. Motor-pump efficiency and turbine-generator are considered 80 %;

Mass and energy balance for each component of the cycle can be calculated using Eq. (3-4).

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{3}$$

$$Q - W = \sum \dot{m}_{out} h_{out} - \sum \dot{m}_{in} h_{in} \tag{4}$$

The parameter area (*A*) is extremely important

The area calculation (*A*) occupied by the system for an FPSO is often more important than power generation because of the limited space available on the platform. The heat exchange area is calculated from the logarithmic temperature differential method (LTDM) according to Eq. (5) (Incropera, 2013).

$$Q = \bar{U}A \left[\frac{(T_{h_1} - T_{c_2}) - (T_{h_2} - T_{c_1})}{ln \left[\frac{T_{h_1} - T_{c_2}}{T_{h_2} - T_{c_1}} \right]} \right] \tag{5}$$

Considering the overall heat transfer coefficient ( ) constant for the entire trajectory of the fluid in the heat exchanger and there are no partial phase changes in the system. Required values are taken from Dorj (2005) and Mills (1995) and shown in Table 3.

Table 2 ó Required values for different processes of the ammonia-water mixture.

Mixture	Overall heat transfer coefficient ( ) [$W/m^2 K$]
Ammonia-water (recuperator)	1000
Ammonia-water (condensation)	1100
Ammonia-water (evaporation)	900

4. RESULTS

Analyzing just the net power output, it is possible to identify for the different studied design points, the higher net power output is achieved (712 kW). This power is reached with the lowest pinch point value (2ºC) and terminal temperature differential (30ºC), a working pressure around of 4400 kPa and the heat exchanger area is 1257 m².

On the other hand, the point that results the minimum heat exchanger area is the one that produces 209 kW, and is reached with PP of 10ºC, TTD 10ºC, a working pressure around 1750 kPa and 399 m² of heat exchanger area.

Finally, there is a point that generates a good quantity of energy with a smaller area. This rate, between power and area, could be call of specified net power output. Considering the lowest PP and the lowest TTD, the net power output produced is 513 kW, a working pressure is around 3750 kPa and the area is 464 m².

The heat exchange area increases with the increasing of working pressure, but the net power output increases until a maximum point and after decreases. All of the points after the maximum net power output give big areas with power which was just reached before.

The thermal efficiency by the First Law, the amount of the energy that get into the system that is real converted in work, changed a little with different PP, so the black curve represents the thermal efficiency for every TTD. As the working pressure increases the thermal efficiency increases too.
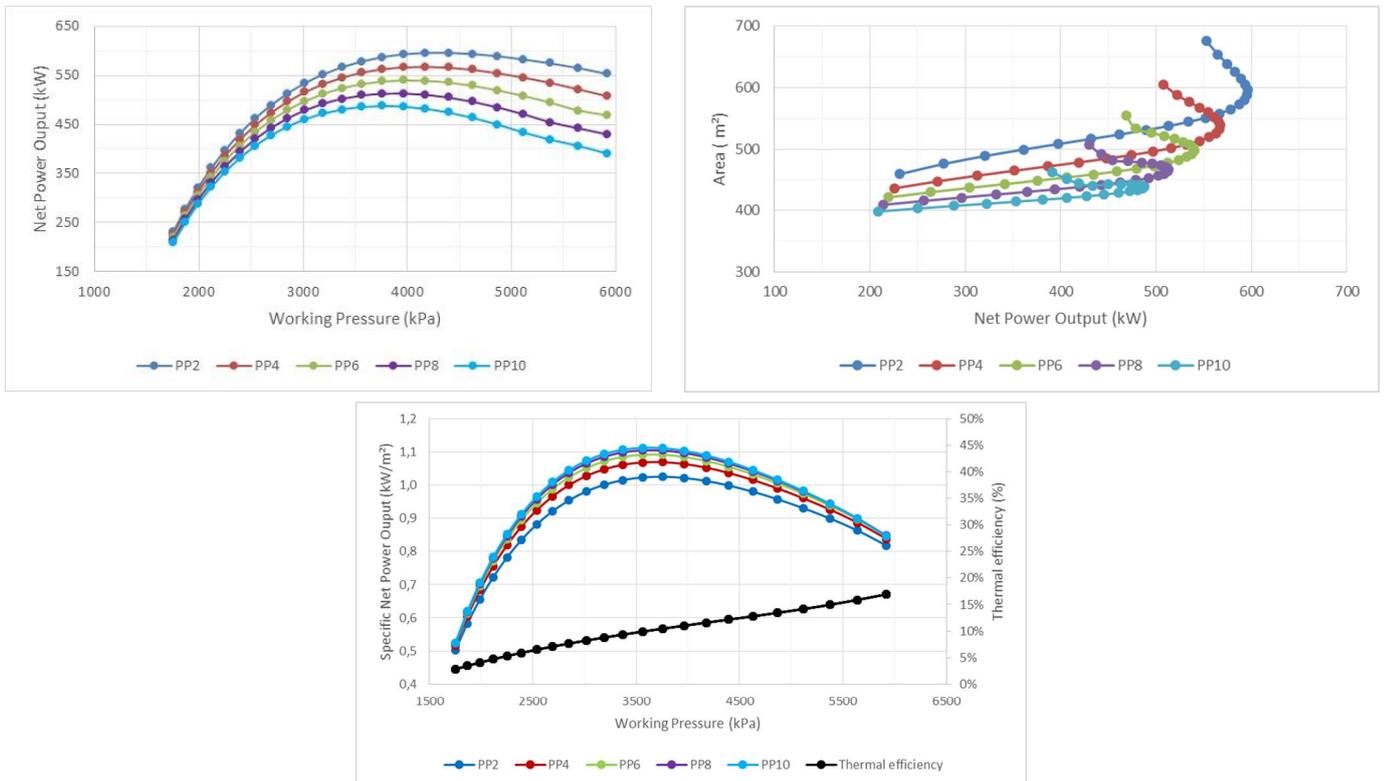

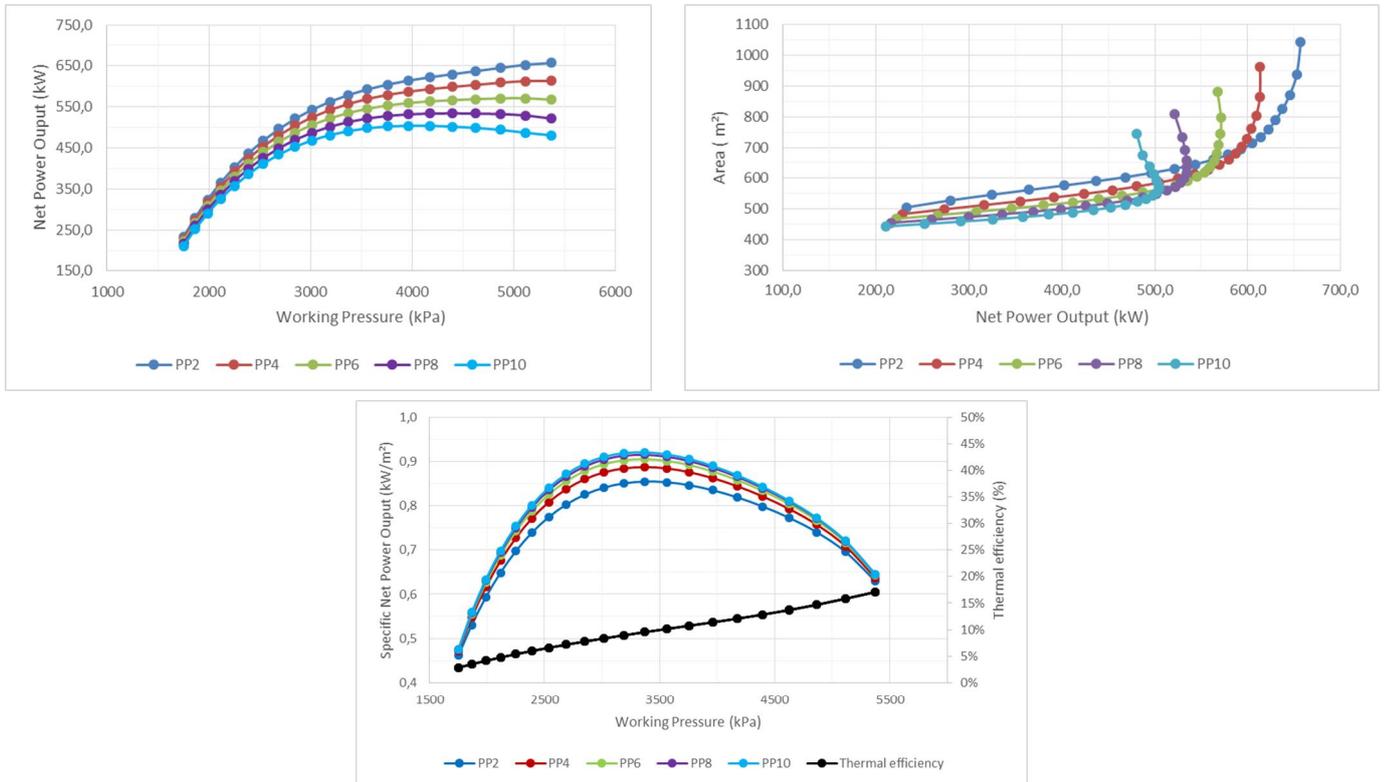

Figure 5: Parameters evaluated for TTD of 10ºC.

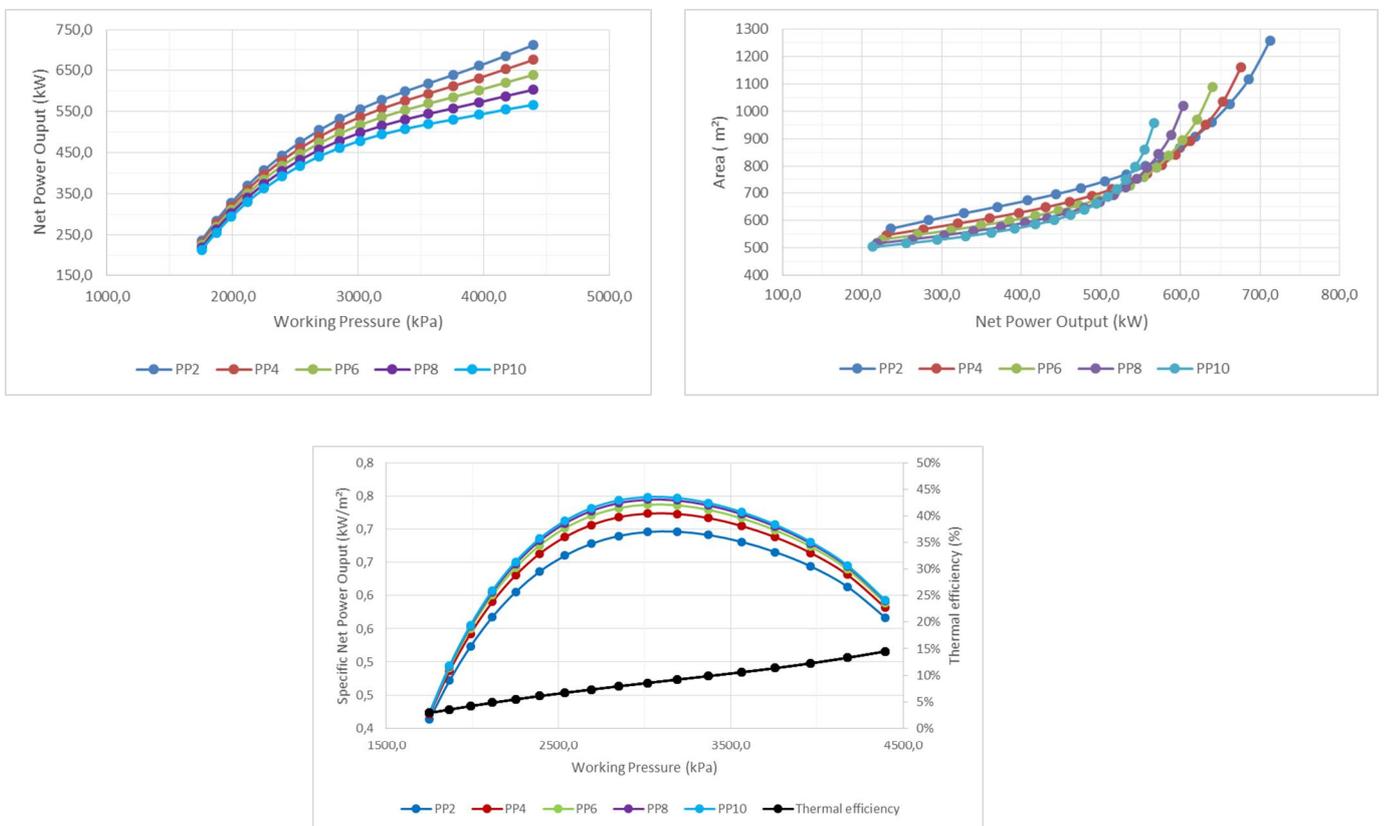Figure 6: Parameters evaluated for TTD of 20ºC.

Figure 7: Parameters evaluated for TTD of 30ºC.

5. CONCLUSIONS

In this work, the performance of the Kalina cycle was examined by using the First Law and Second Law, with an emphasis in low-temperature waste heat recovery in fourth stage of compression in the CO_2 Compression Unit on a Brazilian FPSO. Starting from power generation, the evaluation of the heat exchanger area and thermal efficiency to a fixed ammonia-water composition and design points it is possible to conclude:

1. A small pinch point results in greater heat absorbed by the system and higher net power output, but also require larger system dimension.
2. The increase of TTD generates a greater net power output, but the effect is bigger in the calculus of heat exchange area. Then, lower values of TTD improve the best net power output of the cycle.
3. The heat exchange area for the net power output in the system for different design conditions is an important aspect to be considered in system analysis for use in the FPSO.
4. Last but not least the economic analysis is extremely significant to design this project because it is necessary to choose the point that produces the higher net power output considering the available area, and not necessarily that point is the maximum specific net power output.

Gotelip (2015) studied the area occupied by an ORC and its net power output for different heat sources of the same Brazilian FPSO studied at this work. The results that he obtained to the fourth stageøs Compression CO_2 Unit are shown at the Figure 8. These results reveal that ORC can produce much more net power output than Kalina for the same heat source studied. However, the Kalina cycle needs less area, at least considering just the heat exchange areas. To calculate the total area of the Kalina cycle it has taken in consideration all the equipments.
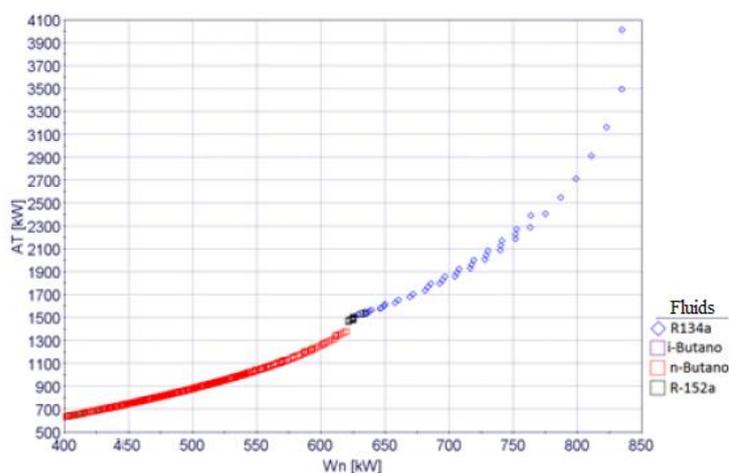

Figure 8: Area and net power output for the forth stageøs Compression CO_2 Unit (Gotelip, 2015).

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