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THERMODYNAMIC ANALYSIS OF A TRIGENERATION SYSTEM INTEGRATED WITH WATER STEAM OR ORGANIC RANKINE CYCLES

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Abstract. *Trigeneration systems combine the production of cooling, heating and power utilities obtained from one primary energy source. This thermal integration is one of the ways to decrease global primary energy consumption and greenhouse gas emission. Conventional trigeneration systems consist of a Brayton cycle, followed by a heat recovery steam generator (HRSG) and finally an absorption chiller. In this work, two different configurations of a trigeneration system were studied, evaluating their thermodynamic performance. These systems differ from each other by the inclusion of a second power cycle, after the absorption chiller, which is a Rankine cycle operating with water (SRC) or an organic fluid (ORC). An ammonia/water single-effect absorption chiller was used. Thermodynamic performance of these two configurations was evaluated considering both energetic and exergetic approaches and a parametric investigation (varying inlet pressure in turbines and pinch point temperature in heat exchanger) was performed. The energetic efficiency of the best performance of this trigeneration integrated with a Rankine cycle was 57.56 % for the ORC, and 55.91 % for the SRC and exergetic efficiencies were 39.41 % and 37.83 %, respectively.*

Keywords: *trigeneration, organic Rankine cycle, energy, exergy, efficiency*

1. INTRODUCTION

Fossil fuel depletion and global warming due to greenhouse gases emissions (GGE) are two of the main concerns of the energy sector at all levels (industrial, commercial and residential). Due to the constant increase of energy demand in many processes and in daily life, it is necessary to evaluate new alternatives towards reducing energy consumption, mainly in the industrial sector, which is responsible for the greatest energy consumption in different forms.

The efficiency of a conventional power plant, which is usually based on single prime movers, is usually less than 40 % and most of the input energy is lost as waste heat (Ahmadi *et al.*, 2012). Therefore, the integration of energy subsystems to provide cooling, heating (and hot water) with a conventional power plant can increase the overall plant efficiency and reduce the emission of environmental pollutants.

An efficient process developed to have a lower consumption of a primary energy source and lower GGE is the combined cooling, heating and power (CCHP) system, known as the trigeneration power plant (Leonzio, 2018). Trigeneration systems can be applied in any enterprise where demands for electricity, heating and cooling are present. The greater availability and wider choice of suitable technologies have made trigeneration an attractive and practical option for a wide range of applications (Memon and Memon, 2017).

Trigeneration systems, with gas turbine, steam turbine or an internal combustion engine for cogeneration and conventional vapor compression refrigeration system (VCRS) or vapor absorption refrigeration system (VARS) for cooling, are commercially available (Jradi and Riffat, 2014). However, trigeneration systems, which use an organic Rankine cycle (ORC), have been investigated as a possible technology to improve system efficiency (Safarian and Aramoun, 2015).

An ORC uses an organic working fluid to generate power, while a conventional Rankine cycle uses water. Differently from water, which is considered a wet fluid, an organic fluid is a dry fluid, which is the one that does not goes under a phase changing during the expansion process in a turbine, due to the negative slope of its vapor saturation curve. Hence, a superheated state is not always needed in the inlet stream of the turbine.

According to Al-Sulaiman *et al.* (2012) and Ahmadi *et al.* (2012) an efficient ORC should operate with a high critical temperature working fluid, thus, the waste heat can be used more efficiently. On the other hand, Liu *et al.* (2004) and Sotomonte (2015) have stated that the thermal efficiency of a thermodynamic system has a weak dependence on the critical temperature of the working fluid, since the maximum value of the heat recovery efficiency occurs at an appropriate vaporization temperature, which is characteristic of each fluid.

Heberle and Bruggemann (2010) have identified that for the same operating conditions, the most suitable working fluid is strongly dependent on the configuration of the thermal system, as well as on the electrical and thermal energy requirements.

Al-Sulaiman *et al.* (2012) have studied a biomass trigeneration system using an ORC, with an efficiency up to 89 %. Pastel *et al.* (2017) presented a thermo-economic analysis of a novel ORC integrated with a cascaded vapor compression–absorption refrigeration system. The energetic efficiency and payback reported were 79 % and 4.9 years, respectively. Sevinchan *et al.* (2019) analysed a biogas multigeneration system used to generate electrical power for at least 300 houses, where the highest energetic and exergetic efficiencies were 72.5 % and 30.44 %, respectively.

In a conventional trigeneration system, power is obtained through a Brayton cycle; heating is provided by a boiler or a heat recovery steam generator (HRSG), while cooling comes from a single-effect absorption chiller. According to Dinçer and Rosen (2007), the absorption chiller has attracted interest because of its advantages over a conventional vapor compression refrigeration systems, including quiet operation, high reliability and long service life. In addition, the use of an absorption cycle increases the overall efficiency of the process.

In this study, two different trigeneration systems, which integrates ORC or SRC after the absorption chiller to increase overall efficiency, is proposed. The objective of this study is to provide a better understanding of the systems proposed, identifying possible improvements, comparing both systems by their energetic and exergetic analysis. Exergy destruction rate in each equipment was computed as well.

2. METHODOLOGY

2.1 System description

The trigeneration system investigated in this study is presented in Fig. 1. It consists of an open-type Brayton cycle, followed by a HRSG (represented as a heat exchanger), a single-effect absorption chiller, and finally a Rankine cycle (operating with water – SRC or an organic fluid – ORC). Natural gas was selected as the primary energy source to be used in the Brayton cycle due its availability, low cost and reduced environmental impact (Ameri *et al.*, 2010). The organic fluid select for the ORC is cyclohexane, which has a high critical temperature, 553.6 K (Aspen Hysys v.10 Databank).

The thermodynamic analysis performed considered an energetic and exergetic approach which are discussed in the following subsection. Process simulator Aspen Hysys v.10, was used for modelling the trigeneration system. The input parameters assumed in the simulations are listed in Tab. 1. Brayton cycle and single-absorption chiller were validated using data from Ameri *et al.* (2010) and Harold *et al.* (2008).

Table 1: Input parameters assumed in the simulation.

Parameter	Value
Brayton	
Fuel mass flow rate (kg/h)	200
Fuel inlet pressure in combustion chamber (kPa)	1000
Fuel inlet temperature in combustion chamber (K)	298
Air inlet temperature in combustion chamber (K)	850
Gas Turbine outlet pressure (kPa)	110
Excess air (%)	200
Compressor and gas turbine isentropic efficiencies (%)	80
Compression ratio	10
Heat loss value from combustion chamber (%)	3
Chilling cycle	
Mass flow rate (kg/h)	4530
Low pressure in the absorber (kPa)	400
High pressure in the generator (kPa)	1700
Light key component (NH_3) in the bottom stream	0.3
Heavy key component (H_2O) in the distillate stream	0.0001
NH_3 mass fraction in the vapor phase at the evaporator	0.975
ORC and SRC	
Turbine inlet pressure base case (kPa)	2000
Turbine and pump isentropic efficiencies (%)	80
Pump inlet temperature base case (K)	343.15
Heat exchanger pinch point temperature base case (K)	10

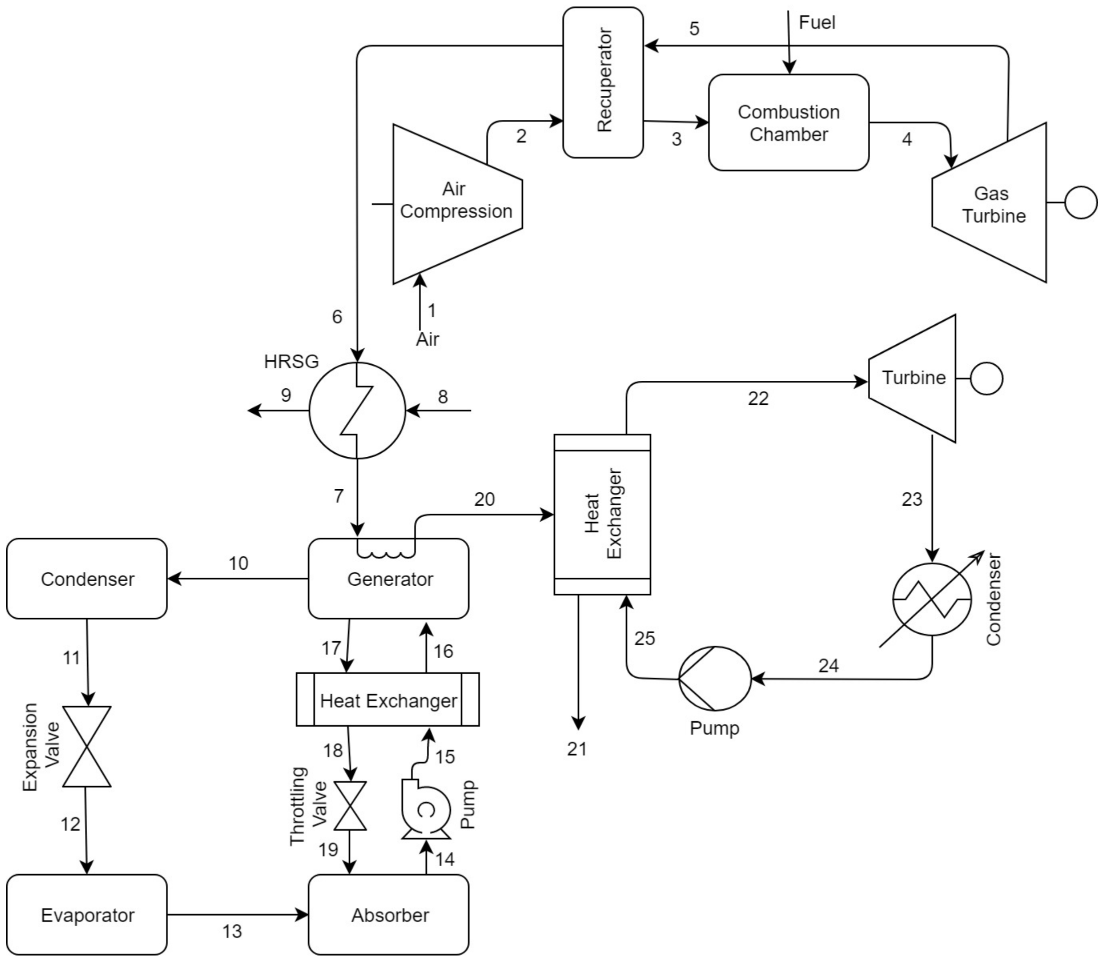


Figure 1: Trigeneration plant configuration.

The simplifications adopted in the simulations are as follows:

- The system and all of its components are operating in steady state;
- Compressor, turbine and pumps are adiabatic;
- Variation in kinetic and potential energies and exergies terms are negligible;
- Pressure drop in pipelines and heat exchangers are neglected;
- Working fluid enters into Rankine turbines as saturated vapor and leaves the condenser as saturated liquid .

2.2 Energetic analysis

An energetic analysis of the system was performed through a parametric analysis considering the influence of some operating variables over the performance of the system. The energy utilization factor (EUF) of the system is defined by Eq. (1). The subscript "tri" indicates trigeneration.

$$EUF_{\mathrm{tri}} = \frac{\mathrm{Energy}_{\mathrm{output}}}{\mathrm{Energy}_{\mathrm{input}}} = \frac{\dot{W}_{\mathrm{net}} + \dot{Q}_{\mathrm{h}} + \dot{Q}_{\mathrm{c}}}{\dot{Q}_{\mathrm{in}}} \tag{1}$$

where $\dot{W}_{\mathrm{net}}$ is the net power produced by the system, $\dot{Q}_h$ is the heat load available in the heat exchanger, $\dot{Q}_c$ is the cooling rate produced by the system and $\dot{Q}_{\mathrm{in}}$ indicates total energy rate input into the system. These variables are defined by Eq. (2) - Eq.(5).

$$\dot{W}_{\mathrm{net}} = \dot{W}_{\mathrm{Brayton}} + \dot{W}_{\mathrm{ORC/SRC}} - \dot{W}_{\mathrm{comp}} - \dot{W}_{\mathrm{pumps}} \tag{2}$$

$$\dot{Q}_{\mathrm{h}} = \dot{m}_6 \cdot (h_6 - h_7) \tag{3}$$

$$\dot{Q}_{\mathrm{c}} = \dot{m}_{12} \cdot (h_{12} - h_{13}) \tag{4}$$

$$\dot{Q}_{\mathrm{in}} = \dot{m}_{\mathrm{ng}} \cdot \mathrm{LHV} \tag{5}$$

where $\dot{m}_{\mathrm{ng}}$ is mass flow rate of the natural gas and LHV is its lower heating value calculated from Aspen Hysys v.10, which is 48.46 MJ/kg.

2.3 Exergetic analysis

Exergy is defined as the maximum theoretical useful work that can be obtained while a system interacts with its neighborhood until an equilibrium is reached between them. The exergy balance is similar to an energy balance but has a fundamental difference that, the former is a statement of the lay of conservation of energy, while the latter may be looked upon as a statement of the law of degradation of energy. Degradation of energy is equivalent of the irretrievable loss of exergy due to all real processes being irreversible (Kotas, 1995).

The specific exergy for each process stream in the system is calculated by the sum of a physical (PH) and a chemical (CH) component as shown in Eq. (6).

$$ex = ex^{\mathrm{PH}} + ex^{\mathrm{CH}} \tag{6}$$

ex^{PH} is defined by Eq.(7).

$$ex^{\mathrm{PH}} = h_j - h_0 - T_0\left(s_j - s_0\right) \tag{7}$$

where j is the process stream number, and "0" stands for a property taken in the reference state (considered to be 1 atm and 298.15 K). Equation (8) determines the specific chemical exergy, ex^{CH}.

$$ex^{\mathrm{CH}} = \left(\sum y_i \overline{ex}_i^{\mathrm{CH}} + \mathrm{R}T_0 \sum x_i \ln(y_i)\right) \tag{8}$$

where ex_j^{CH} is the standard chemical exergy, R is the universal gas constant and y_i is the molar fraction of component i at stream j.

The values of standard chemical exergy of the species related to the analysis of the system considered are listed in Tab 2.

Table 2: Values of standard chemical exergy $\left(\overline{ex}^{\mathrm{CH}}\right)$.

Species	$\overline{ex}^{\mathrm{CH}}$ (kJ/mol)
CH_4	831.65
C_2H_6	1495.84
C_3H_8	2154.00
CO_2	19.87
H_2O	9.5
N_2	0.72
O_2	9.97

Source: Szargut *et al.* (1988).

The total exergy destruction rate indicates the potential work lost due to irreversibilities and may be calculated by Eq. (9).

$$\dot{E}x_d = \sum_j \left(1 - \frac{T_0}{T_j}\right) \cdot \dot{Q}_j - \sum \dot{\mathrm{W}} + \sum_i \dot{m}_i ex_i - \sum_e \dot{m}_e ex_e \tag{9}$$

where,

- $\sum_j \left(1 - \frac{T_0}{T_j}\right) \cdot \dot{Q}_j$ represents the transfers of exergy associated with heat;
- $\sum \dot{\mathrm{W}}$ represents the transfers of exergy associated with power/work;
- $\sum_i \dot{m}_i ex_i - \sum_e \dot{m}_e ex_e$ represent the transfers of exergy associated to the mass flows of the control volume.

Finally, the exergetic efficiency of the system can be defined as the ratio between exergy output of the system and chemical exergy of the fuel, given by Eq. (10).

$$\eta_{ex,tri} = \frac{\dot{W} + \left(1 - \frac{T_0}{T_h}\right) \cdot \dot{Q}_h + \left(\frac{T_0}{T_c} - 1\right) \cdot \dot{Q}_c}{\dot{E}x_{\mathrm{f}}} \tag{10}$$

where $\dot{E}x_{\mathrm{f}}$ is the fuel (natural gas) exergy.

2.4 Parametric analysis

In order to understand how some operating conditions impact the EUF and exergy efficiencies, allowing a comparison between both configurations considered in this work, a parametric study was conducted varying the following parameters of the Rankine cycles:

- Heat exchanger pinch point temperature (T_{PP});
- Turbine inlet pressure (T_{34});
- Pump inlet temperature (P_{22}).

The T_{PP} is responsible for defining the heat flow supplied to the Rankine thermal system. As this parameter decrease, the amount of energy recovered from the heat source increase, so does the cost of the heat exchanger. The base case T_{PP} was 10 K (recommendation from Smith (2005)), and it was varying from 10 K to 30 K.

The T_{34} is responsible for defining the turbine outlet pressure. The SRC outlet pressure of the turbine is limited to 10 kPa to avoid the condensation pressure getting too low and leading to air infiltration (Zang *et al.*, 2016). On the other hand, ORC outlet pressure of the turbine is limited to 20 kPa to avoid ΔT_{min} < 10 K in the condenser. Besides that, the pump inlet temperature must be a temperature able to use tower cooling water. Based on these information, the pump inlet temperature was varying from 232.15 K to 363.15 K. The base case selected was 343.15 K;

The P_{22} is responsible for defining the mass flow rate of the working fluid and the enthalpy turbine inlet. The critical pressure of cyclohexane is 4080 kPa (Refprop 9.1). The base case was 2000 kPa (cyclohexane half critical pressure) and it was varying from 1500 kPa to 2500 kPa.

3. RESULTS AND DISCUSSION

The thermodynamic performance of the two trigeneration configurations was evaluated considering both energetic and exergetic approach. The EUF and exergetic efficiency were examined through changing the parameters presented in Section 2.

Input data of the system were kept constant as shown in Tab. 1. The utilities and power production for the base case are presented in Tab. 3.

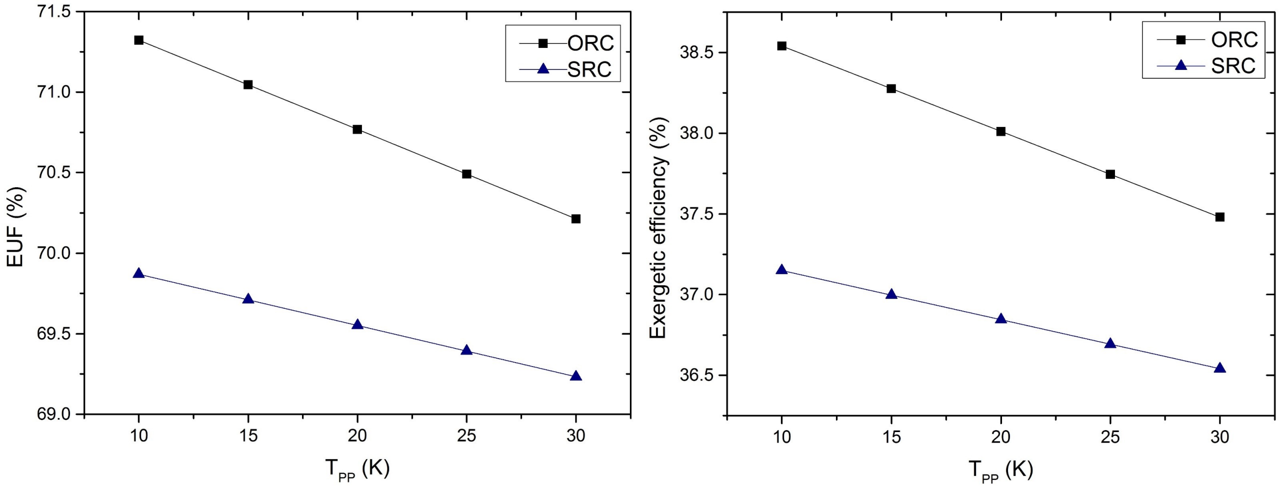
Table 3: Output data of the trigeneration system.

Variable	Value (kW)
Net output Brayton power	890
Heating load	360
Cooling load	200
Net output ORC power	121
Net output SRC power	82

3.1 Influence of the pinch point temperature of the ORC and SRC heat exchanger

The influence of the pinch point temperature (T_{PP}) of the ORC and SRC heat exchanger was examined under the base case turbine inlet pressure, 2000 kPa, and pump inlet temperature, 373.15 K. The minimum pinch point temperature selected was 10 K, following the recommendation from Smith (2005) and it was varyed from 10 K to 30 K, as presented in Fig 2. In this variation range, the output power was varying from 125.5 kW to 94.4 kW for the ORC, and from 82.4 kW to 65.0 kW for the SRC.

It can be observed in Fig. 2 that the EUF and exergetic efficiency decrease as the pinch point temperature increase in both ORC and SRC trigeneration system. The EUF and exergetic efficiency are functions of the electrical power



(a) EUF (%). (b) Exergetic efficiency (%).

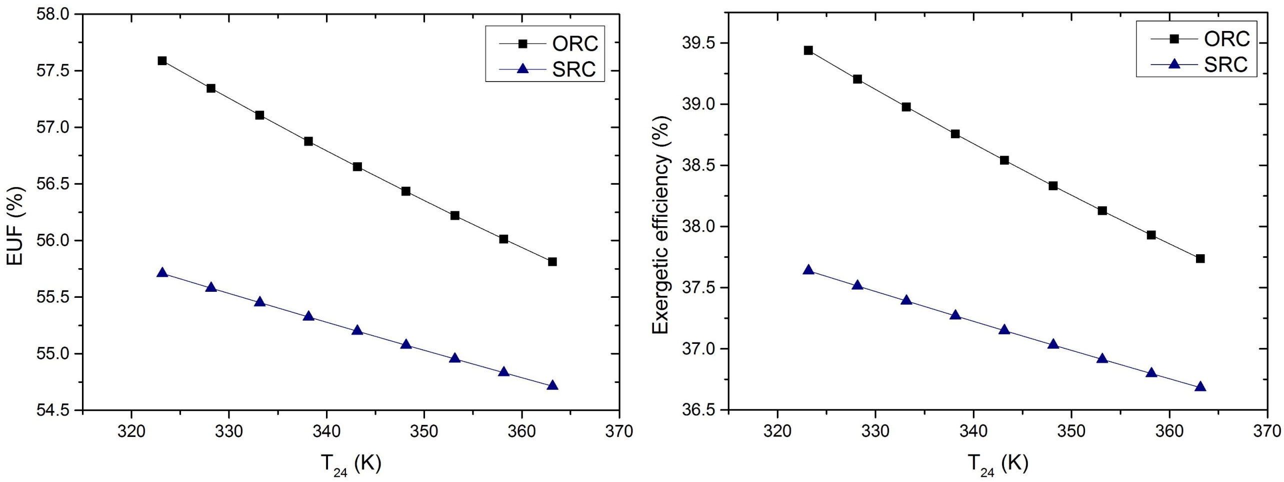
Figure 2: Influence of the ORC and SRC heat exchange pinch point at P_{22} = 2000 kPa and T_{24} = 343.15 K.

produced, and as the pinch point temperature increases, electrical power decreases. The pinch point temperature analysis is important due to the fact that the heat exchanger cost depends on this minimal temperature difference. The ORC trigeneration system shows a better performance than the SRC trigeneration system.

3.2 Influence of the inlet temperature of the ORC and SRC pump

In this study, the inlet pump temperature (T_{24}) was varyed from 232.15 K to 363.15 K under the base case, which represents the ORC and SRC turbine outlet pressure from 36.85 kPa to 131.9 kPa and from 12.34 kPa to 31.29 kPa, respectively. In this variation range, the output power was varying from 146.6 kW to 98.5 kW for the ORC, and from 95.8 kW to 68.8 kW for the SRC.

Figure 3 shows the influence ORC and SRC pump inlet temperature on the EUF and exergetic efficiency. It can be observed that as the temperature increase, both EUF and exergetic efficiency in ORC and SRC configuration decrease. In addition, ORC also had a better performance comparing to SRC.



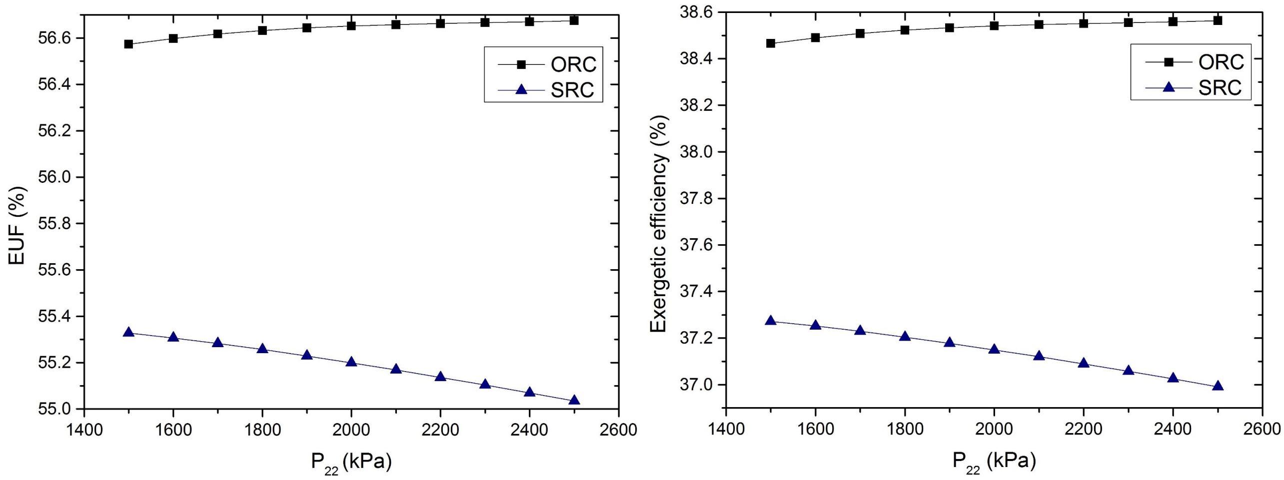
(a) EUF (%). (b) Exergetic efficiency (%).

Figure 3: Influence of the ORC and SRC pump inlet temperature at P_{22} = 2000 kPa and T_{PP} = 10 K.

Besides that, ORC presented smaller expansion ratios and lower enthalpy variation comparing to the SRC, allowing for a favorable turbine design. According to Astolfi *et al.* (2014), this results in a turbine with a high isentropic efficiencies and a reduced number of stages, with a reasonable size, and hence competitive manufacturing costs. On the contrary, SRC had a high expansion ratio, making the turbines of steam cycles larger and more expensive, and even with a multi-stage expansion needed since the single-stage one cannot fulfill the overall volume ratio or the power duty.

3.3 Influence of the inlet pressure of the ORC and SRC turbine

Figure 4 shows the influences of the turbine inlet pressure on the performance of the ORC/SCR trigeneration system. The turbine inlet pressure (P_{22}) was varyed from 1500 kPa to 2500 kPa under the base case. In this variation range, the output power was varying from 119.2 kW to 122.0 kW for the ORC, and from 85.4 kW to 77.5 kW for the SRC.



(a) EUF (%).

(b) Exergetic efficiency (%).

Figure 4: Influence of the ORC and SRC pump inlet temperature at T_{24} = 343.15 K and T_{PP} = 10 K.

Figure 4 shows that the ORC inlet pressure influence on both EUF and exergy efficiency is insignificant. As the inlet pressure increase, the ORC pump power also increases, making the ORC net power almost constant. The results are important since they indicate that the ORC can be operated under lower pressures, resulting in lower costs. On the other hand, regarding the SRC, as the turbine inlet pressure increases, both EUF and exergetic efficiency decrease. This behavior is the opposite of expected, however in order to maintain the heat exchanger pinch point temperature at 10 K, mass flow varies as pressure changes, decreasing the SRC performance, as presented in Tab. 4.

Table 4: Mass flow variation as a function of turbine inlet pressure.

Pressure (kPa)	ORC mass flow (kg/h)	SRC mass flow (kg/h)
1500	4610.7	628.1
1750	4417.1	594.5
2000	4250.5	563.7
2250	4109.6	535.2
2500	3996.3	508.6

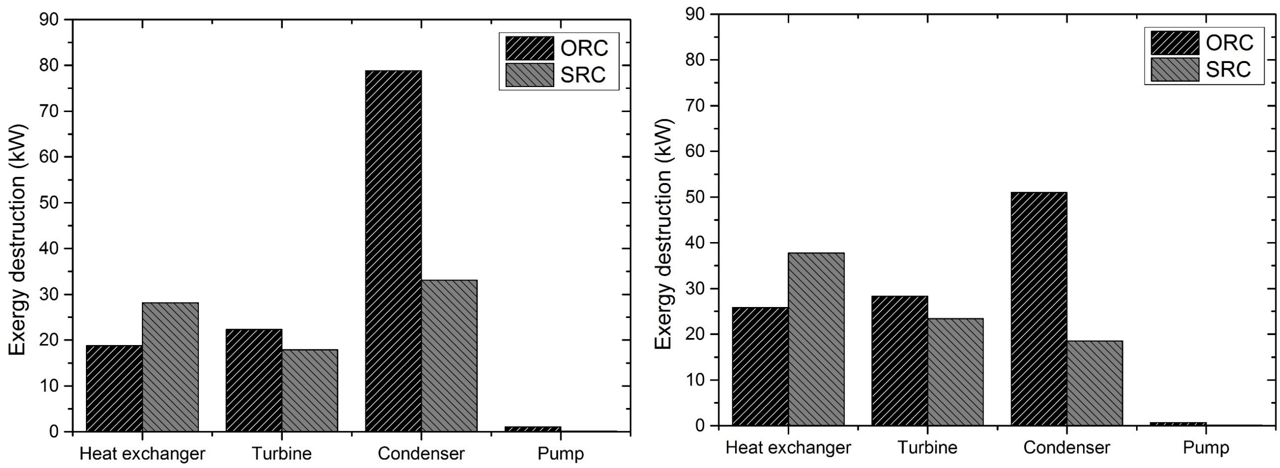
It can be observed in Tab. 4 that the SRC requires a smaller mass flow rate through the power conversion process. According to Tian and Shu (2017), the unconventionally small mass flow rate brings great technological difficulties in designing and manufacturing expansion components. On the other hand, most ORC systems operate with reasonable mass flow rates and allow conventional technologies to be used, facilitating of the development of efficient low power capacity ORC turbo machinery.

3.4 Exergy destruction rate

The exergy destruction analysis for the different components of the system in ORC and SRC at the base case input data and at the best performance is presented in Fig. 5.

It should be noted that for the best performance case the exergy destruction rate is better distributed than the base case, not overloading a specific equipment. Comparing the base case to the best system performance, there was a significant decrease on the ORC condenser exergy destruction, due to the decrease in the condenser pinch point temperature.

This study shows that when designing a similar ORC trigeneration system as in this study, the most important component that needs considerable care in their design and selection is the condenser, probably considering another cold stream fluid.



(a) Base case.

(b) Best performance case.

Figure 5: Exergy destruction rate in the ORC and SRC.

4. CONCLUSION

Energy and exergy analyses of a trigeneration system integrated with Water Steam or Organic Rankine Cycles were conducted. The main conclusions from this study were:

- ORC trigeneration system was able to produce 1000.5 kW power, 360 kW heating, and 200 kW cooling, while SRC trigeneration system was able to produce 959.8 kW power, 360 kW heating, and 200 kW cooling, considering their best operating conditions;
- The best performance of the trigeneration system is obtained at T_{PP} = 10 K, T_{24} = 323.15 K, and P_{22} = 1500 kPa;
- The performance of ORC is insensitive to the turbine inlet pressure variation, so operation is recommended for an ORC under low pressure, reducing costs. On the other hand, as the pressure increases in the SRC, its thermodynamics performance decreases, being recommended to operate the SRC also on low pressure;
- Reasonable ORC mass flow rate and low enthalpy variation in the turbine result in high isentropic efficiencies and reasonable size. The SRC smaller mass flow rate and high expansion ratio make the turbine larger and more expensive;
- The exergy destruction rate in the best performance case was better distributed.

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