Tecnologia/Technology

ORGANIC RANKINE CYCLE ASSOCIATED WITH AN ABSORPTION CHILLER FOR BIOMASS APPLICATIONS

C. A. R. Sotomonte,

S. Ribeiro,

E. Oliveira,

E. E. S. Lora,

and O. J. Venturini

Universidade Federal de Itajubá

Departamento de Engenharia Mecânica

Bairro Pinheirinho

CP. 37500-903, Itajubá, Mina Gerais, Brasil

c.rodriguez32@unfei.edu.br

ABSTRACT

The main goal of this work is to develop a calculation process, based on the second law of thermodynamic, for evaluating the potential of a small Organic Rankine Cycle (ORC) cogeneration (electrical and cooling power) plant using biomass as fuel. Two different configurations and several organic working fluids are presented and assessed. A mathematical model has been developed to find thermodynamically suitable fluids for ORC in biomass cogeneration plants. The main results show that, the family of alkylbenzenes has higher exergetic efficiencies found to be around 16% almost 5 percent higher efficiency than the cycles that use siloxanes as working fluids; in spite of their lower cooling power capacity. The results demonstrate that the cycle efficiency is more dependent on the thermodynamic properties of the working fluids than on the system configuration

Keywords: Cogeneration, Exergetic Analysis, Biomass, Organic Rankine Cycle

NOMENCLATURE

C   cooling power, kW
P   Pressure, kPa
T   average fluid temperature, K
W_n net electric power, kW
m   working fluid mass flow, kmol/s
m_B biomass consumption, kg/s
H   enthalpy, kJ/kmol
S   entropy, kJ/kmol K
Q   heat flow, kJ/s
E   exergy. kW
I   irreversibility, kW

Greek symbols

η   energy efficiency
ε   exergy efficiency

Subscripts

b   boiler
t   turbine
p   pump
con   condenser
is   isentropic
evp   evaporator
g   electric generator
F   fuel
P   product

INTRODUCTION

During the last years, Brazil has been studying the utilization of biomass as fuels for distributed generation systems aiming the diversification the country's energetic matrix and also to supply electricity to small communities situated in isolated regions. Currently, several prime movers such as: Micro-turbines, Internal Combustion Engines and Stirling Engines are being proposed to generate electric energy for small scale applications. However, the utilization of biomass with these technologies requires a previous stage of the biomass conversion into a liquid or gaseous fuel via a chemical, thermochemical or biochemical process. Although feasible, most of these conversion technologies still do not have competitive cost, for this reason, the most used technology for energy conversion from biomass is the direct combustion.

As a result, electricity generation technologies, such as steam turbines, although having low efficiencies, would have a better performance due to their high level of development. For the improvement of these thermal systems, especially when they are related to efficiency gain, the reduction of losses in conversion process is mandatory. In this way, power and cooling cogeneration systems can be interesting; particularly for tropical regions such as the Brazilian Amazonia.

The application of combustion based biomass to electricity conversion technologies for small capacities is complicated by the fact that small conventional axial steam turbines have very low efficiencies, due to the small blades height and the necessity of partial flow implementation in the flow section. There are two solutions for this problem: the utilization of another fluid as the working one, an organic fluid (Organic Rankine Cycle- ORC), the utilization of another prime mover instead of the conventional axial turbines (radial turbines, screw or scroll expanders, or steam piston engines), or both

those modifications simultaneously. The last case corresponds the EFGT technology – the externally fired gas turbine operating using air heated in a specially designed heat exchanger using high temperature combustion gases. From all these alternative options only the ORC systems are just commercial, however its costs is still high.

Rentizilas et al. (2009), compared the technologies and the costs of the direct use of the biomass in an ORC plant and its gasification. They showed that gasification demands higher investments (about 75% more) and operation and maintenance costs (about 200% more). Moreover, according to Duvia et al. (2007); Quoilin and Lemort (2009), comparing to competing technologies, the main advantages of an ORC power plant are: high cycle efficiency; very high turbine efficiency; low mechanical stress of the turbine, due to low tangential velocity; the low turbine RPM allows the direct coupling to the electrical generator, without gear box; there is no erosion on the turbine blades, due to moisture absence in the exhaust vapor flow; long equipment life time; there is no need of water treatment systems. Besides the advantages presented above, the success of the biomass ORC technology can be explained by its modular characteristics, which makes this technology particularly suitable for distributed generation to smaller scale power where the conventional Rankine steam power cycles are not cost-effective in this power range.

These systems are associated with a minimization of environmental impacts, especially when the goal is the reduction of CO_2 emissions, as contemplated in the Kyoto protocol, since, biomass fuels are considered to emit almost zero net CO_2.

Considering these scenarios, this paper presents a comparative study of a thermal cogeneration system using steam turbines with several organic working fluids, based on exergetic analysis; in order to, clarify which is the best cycle configuration, evaluate the exergy destruction and the exergetic efficiency in each component of the system.

BIOMASS COGENERATION SYSTEM DESCRIPTION

The cycles proposed in this paper include two different configurations of the ORC associated an absorption chiller, one with and the other without a regenerator. The absorption unit is a NH_3-H_2O single effect, which is installed to increase the efficiency of the thermodynamic cycle using part of the energy contained in the exhaust vapor flow in the turbine, which is generally rejected to the environment through the condenser. These cycles were modeled to produce 300 kW of net electric power (W_n) and until 210 kW of cooling power (C), the average consumption of an 200 – 300 people community that has a sawmill and a refrigeration unit to sell ice and to keep fruit pulp and fish.

The cycle considered (Fig. 1) is a simple ORC associated with an absorption chiller, consisting of a pump, evaporator, turbine and condenser. The principle of energy generation by an ORC process is similar to the conventional Rankine cycle process. The main difference is that ORC process uses an organic working fluid, instead of water, with favorable thermodynamic properties. Another difference is that the boiler does not heat the working fluid directly, but rather a thermal oil, which exchanges heat with the organic fluid in the evaporator. Consequently, when the efficiency of the system is calculated, the boiler efficiency ($_b$) of the thermal oil boiler has to be taken into consideration, the maximum thermal oil temperature is limited to about 630 K (Dresher and Bruggemann, 2007).
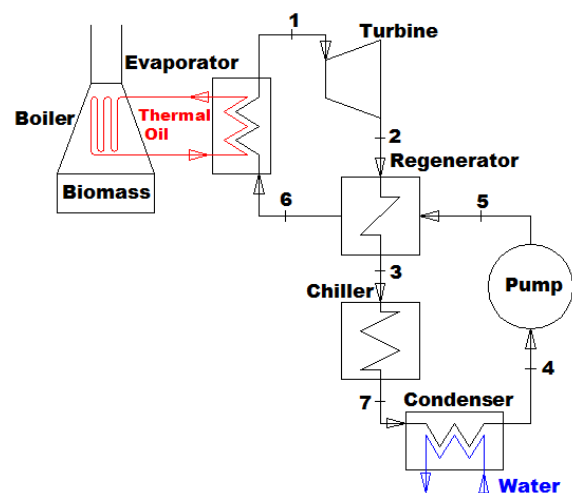

Figure 1. Organic Rankine cycle cogeneration.

The pressurized organic working fluid, saturated vapor, expands in the turbine to produce useful work; the vapor exhaust, after passing through the chiller, is cooled in the condenser and passes through the pump, returning to the evaporator in the desired pressure. It is still possible to use a heat regenerator, which uses the flow in the turbine outlet to preheat the flow that enters at evaporator, in order to increase the cycle efficiency.

Pump ($_p$) and turbine ($_t$) efficiency was assumed as 80% and electric generator efficiency ($_g$) 98%. The minimum pressure in the condenser was set to 10 kPa, pressure accepted for a commercial condenser. If required, the condenser pressure could be raised to make the vapor temperature equal 393.15 K, the lowest heat source temperature accepted for a single effect NH_3-H_2O absorption unit.

Minimum approach temperature of 10 ^{o}C in the evaporator and 15 ^{o}C in the condenser is assumed. To reduce expenses with safety measures and material, the maximal pressure of the cycle (P_l) was fixed in 2MPa (Dresher and Brüggemann, 2007). However, due to some organic working fluids has critical pressure (Pc) below 2MPa, this pressure been limited

at 70% of the critical pressure for this fluids. The solution of the energy and mass balances determines the mass flow rate of the working fluid (m), biomass consumption (m_B), efficiencies and temperature at each point.

In the thermodynamic analysis the following general assumptions are made: 1) The kinetic and potential energy are neglected; 2) The reference state temperature (T_0) and pressure (P_0) are 298.15 K and 101.325 kPa; 3) The temperature and pressure of fuel and air inlets are 298.15 K and 101.325 kPa; 4) Steady state operation; 5) Load condition 100%. The main operating parameters are summarized in Table 1.

For the purpose of evaluate the most suitable organic working fluid in order to increase the efficiency of biomass ORC power plant detailed analysis for typical fluids has been carried out. Seven organic working fluids; four siloxanes the most widely used fluids currently in the ORC plants and three alkylbenzenes that showed higher cycle efficiency according to Drescher and Brüggemann, (2007) were selected.

Besides the thermodynamics parameters, environmental and safety characteristics are going to be taken into account. The safety and environmental parameters analyzed in the organic fluid selection are the global warming potential (GWP), ozone depletion potential (ODP) and the ASHRAE 34 toxicity and flammability classification (Table 2).

Table 1**.** Operating conditions of the power plant.

Maximum Turbine Inlet Vapor Pressure (kPa)	2000
Thermal Oil Boiler Efficiency (%)	80
Turbine Isentropic Efficiency (%)	80
Electric Generator Efficiency (%)	98
Regenerator Effectiveness	0.5
Minimum Chiller Heat Source Temperature (K)	393.15
COP	0.5
Pump Isentropic Efficiency (%)	80
Biomass Lower Heating Value (kJ/kg)*	13800

*Wasted wood (moisture content 20%)

MATHEMATICAL MODEL

Thermodynamic Properties

The study of the ORC requires the thermodynamic properties of the organic working fluid. These properties can be obtained by the Standard Cubic Equations of State (*EoS*), using the critical point properties (T_C and P_C) and the acentric factor (w) as fluid parameters. Peng-Robinson (*PR*) and Peng-Robinson modified by Stryjek and Vera (*PRSV*) are documented examples of the use of these EoS in the analysis of organics fluids in thermal systems. According to (Dresher and Bruggemann, 2007); Angelino and Colonna (1998), the PR and

PRSV EoS provides, aside from the critical point, relative discrepancies less than ±3% for calculated gas and liquid heat capacity and below ±1.5% for cycle efficiency (Dresher e Bruggemann, 2007); accurate estimates of the saturation pressures and can be used together with modern mixing rules to estimate the properties of mixtures with different fluids (Angelino and Colonna, 1998).

$$Z = \frac{V}{V-b} - \frac{a \cdot V}{R \cdot T \cdot (V^2 + 2 \cdot b \cdot \mathbf{V} - b^2)} \qquad (1)$$

$$b = 0,077796 \cdot \frac{R \cdot T_c}{P_c} \qquad (2)$$

$$a = 0,457235 \cdot \frac{R^2 \cdot T_c^2}{P_c} \cdot \alpha \qquad (3)$$

$$\alpha = \left[1 + k_0 \cdot (1 - \sqrt{T_R}) + k_1 \cdot (1 - T_R) \cdot (0,7 - T_R) \right]^2 \quad (4)$$

These models have limitations, namely: (a) incorrect functional form at high subcritical reduced temperatures (TR) and pressures (PR) and in the superheated region close to the critical point ($0.7 \leq P_R \leq 1$ and $0.95 \leq T_R \leq 1.01$), (b) low accuracy of saturated and subcooled liquid densities (Colonna *et al.* 2006). In this paper the PRSV EoS (Eq. 1) proposed by Stryjek and Vera (1986), was used to evaluate the thermodynamic properties of the different working fluids analyzed. Where (k_1) is obtained from experimental data or from fitting values to an accurate Eq. (1). Values of k_1 parameter (Tab. 2) for the selected fluids are reported in Colonna et al. (2008) and Proust and Vera (1989).

$$\frac{H^{ig} - H}{R \cdot T} = \int_V^\infty \left[T \cdot \left(\frac{\partial Z}{\partial T} \right)_V \right] \frac{dV}{V} + 1 - Z \qquad (5)$$

$$\frac{S^{ig} - S}{R} = \int_V^\infty \left[T \cdot \left(\frac{\partial Z}{\partial T} \right)_V \right] \frac{dV}{V} - \int_V^\infty [1 - Z] \frac{dV}{V} - \ln Z \quad (6)$$

All the thermodynamic properties of interest of any substance can be calculated from volumetric data and thermal measurements (Poling et al. 2004). Maxwell's relations and the P-v-T relation of Eq. (1) provide the necessary information for calculation of the properties enthalpy (H), Eq. (5) and entropy (S) Eq. (6) used in this study. Where (H^{ig}) and (S^{ig}) are the respective enthalpy and entropy at the hypothetical ideal gas reference state.

A description of the mathematical development Eqs. (5-6) are reported in Poling *et al.* 2004 and Smith, (2005).

Table 2. Toxicity and flammability classification.

Fluid	T_b (K)	T_c (K)	P_c (kPa)	w	k_1	ODP	GWP[a]	ASHRAE 34
Toluene (C_7H_8)	384	591.8	4106	0.26323	0.03849	0	-	B3
Ethylbenzene (C_8H_{10})	409	617.2	3606	0.3027	0.03994	0	-	B3
Propylbenzene (C_9H_{12})	432	638.32	3200	0.34513	0.02715	0	-	B2
Octamethyltrisiloxane (MDM)	425	564.09	1415	0.5297	0.12634	0	-	A2
Decamethyltetrasiloxane (MD_2M)	467	599.4	1227	0.668	0.03079	0	-	A2
Dodecamethylpentasiloxane (MD_3M)	503	628.36	945	0.7218	0.16431	0	-	A2
Dodecamethylcyclohexasiloxane (D_6)	518	645.78	961	0.7361	0.09627	0	-	A2

(a) Apparently these fluids do not have a direct influence on global warming. However, they may have an indirect effect to trigger the formation of photochemical oxidants in the atmosphere.

Energy Analysis

The equations presented in this paper represent the mass and energy balances, for each subsystem that compose the power cogeneration cycle, taking into consideration the assumptions already presented.

For the turbine (1-2):

$$\eta_{is,t} = \frac{h_1 - h_2}{h_1 - h_{2,s}} \qquad (7)$$

$$W_t = \dot{m} \cdot (h_1 - h_2) \qquad (8)$$

For the regenerator (2-3) (5-6):

$$h_2 - h_3 = h_6 - h_5 \qquad (9)$$

$$T_3 = T_2 - \varepsilon_r \cdot (T_2 - T_5) \qquad (10)$$

Where ε_r is effectiveness for the regenerator:

For the chiller (3-7):

$$C = \dot{m} \cdot (h_3 - h_7) \cdot COP \qquad (11)$$

For the condenser (7-4)

$$Q_{cond} = \dot{m} \cdot (h_3 - h_4) \qquad (12)$$

For the pump (4-5):

$$\eta_{is,b} = \frac{v_4 \cdot (P_5 - P_4)}{h_5 - h_4} \qquad (13)$$

$$W_b = \dot{m} \cdot (h_5 - h_4) \qquad (14)$$

For the evaporator (6-1):

$$Q_{evp} = \dot{m} \cdot (h_1 - h_6) \qquad (15)$$

For the boiler:

$$Q_T = \frac{Q_{evp}}{\eta_b} \qquad (16)$$

$$Q_T = \dot{m}_B \cdot LHV \qquad (17)$$

Net power output:

$$W_n = (W_t \cdot \eta_g) - W_b \qquad (18)$$

Global energy efficiency

$$\eta_c = \frac{W_n + C}{Q_T} \qquad (19)$$

Eqs. (7-18) are the mathematical model for each component in the cycle. The energy efficiency is quantified as the ratio of useful energy output and total input energy in the system (Eq. 19), while electric efficiency don´t takes into account the cooling power generation.

Exergetic Analysis

To evaluate the efficiency of a cogeneration system that produces cold and electric power; the First Law of Thermodynamics, states that, this is quantified as the ratio of useful energy output and total input energy in the system and is calculated by Eq. (19). Where W_n is the net electrical power generated in the cycle, C is the cooling power of the chiller and Q_T is the total energy supplied to the thermodynamic cycle. However, this analysis is not enough to identify energy losses and efficiencies of these systems. Since; this principle only takes into account the quantity of energy, but not its quality, and the result is an overvaluation of the thermal component of the system.

To overcome this problem, exergy (E) can be used as a quality and quantity measure of energy which involves the first and second thermodynamic laws, so an exergetic analysis is useful to identify and

quantify the thermodynamic efficiencies of a combined cool and power system. Thus, exergy efficiency (ε) of the cogeneration system being defined as follows:

$$\varepsilon = \frac{\dot{W}_n + \dot{E}_{chiller}}{\dot{E}_T} \qquad (21)$$

Where $\dot{E}_{chiller}$ is the exergy variation of the working fluid (NH_3/H_2O) in the chiller evaporator and $\dot{E}_T$ the total exergy supplied to the thermal system.

According to Tsatsaronis (1993), resources (F) are flows that act as exergy sources, necessary for the manufacturing of a certain product (P). Consequently, the exergy efficiency of equipment (ε_n) is given by:

$$\varepsilon_n = \frac{\dot{E}_P}{\dot{E}_F} \qquad (22)$$

Using the concept of resources (F) and products (P), if the system operates in steady state and neglecting the heat losses in equipment, the exergy balance for calculation of irreversibilities (I) in the equipment can be written as follows:

$$\dot{I} = \dot{E}_F - \dot{E}_P \qquad (23)$$

The parameter presented by exergy analysis provides a clear criterion for evaluating the performance of each thermal system and its components. A good description of the concepts used to evaluate the exergy flows are reported in Kotas (1985); Szargut *et al.* (1988) and Zaleta *et al.* (2007).

RESULTS AND DISCUSSION

The intention of the first part (Fig. 2-5) is a comparison of global energy efficiency, electric efficiency, biomass consumption and cooling power production at different evaporator pressure while keeping the turbine inlet temperature at saturated conditions and the condensation pressure at 10 kPa for all the working fluids. According to the information already presented the absorption cooling systems have an optimal operation temperature. Hence, the use of alkylbenzenes like working fluids does not allow the use of a heat regenerator operating under the conditions described previously; due to the lower temperature in the fluid flow at the outlet of the turbine.

Figure 2 demonstrates that for all the fluids used, with and without heat regenerator, the system electric efficiency increases with the increment of the turbine inlet pressure. Lower inlet turbine pressure increases both the evaporator heat flow and the working fluid mass flow which leads to a decrease in

the system efficiency for a fixed electrical net power (300 kW).

Figure 2 also shows that cycles working with alkylbenzenes, even without heat regenerator, have the best electrical performance among the organic fluids. Toluene shows highest efficiency among the alkylbenzenes while D_6 shows the worst performance of the siloxanes. These results clearly demonstrate that the cycle efficiency is more dependent on the thermodynamic properties of fluids than on the system configuration. Hence, the use of regenerative ORC is not justified for all fluids from the thermal efficiency point of view; however, others parameters have to be considered in this analysis, such as biomass consumption, cooling power generation and global energy efficiency.
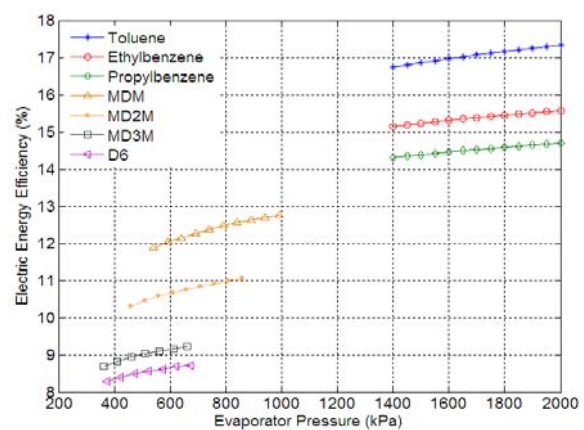

Figure 2. Variation of the electric efficiency with the turbine inlet pressure.

Biomass consumption (Figure 3) is consistent for all working fluids. Since, this parameter is a direct measure of the heat flow required by the evaporator. Thus, for a fixed electric power output, the cycles with lower biomass consumption have higher energy efficiency. This is due to the decrease in the working fluid mass flow as a result of the increase in the turbine inlet pressure.
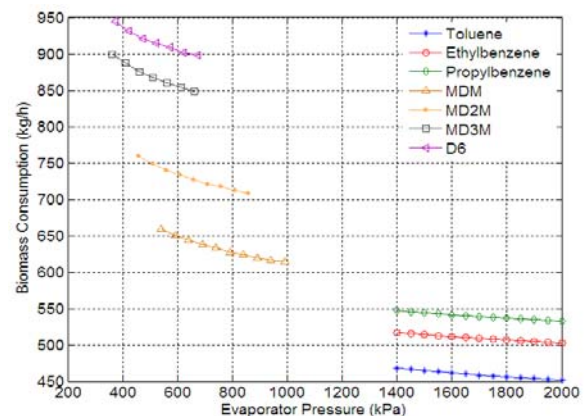

Figure 3. Variation of the biomass consumption with the turbine inlet pressure.

The absorption unit was modeled to produce up to 210 kW (60 TR) of cooling power (*C*). Figure4 shows that, Toluene, Ethylbenzene and MDM did not reach the maximum cooling power production. This is due to the lower temperature of the MDM working fluid before entering the chiller when compared to the other siloxanes and both the smaller mass flows and the lower temperature for Toluene and Ethylbenzene working fluids when compared to the others alkylbenzenes analyzed (Tab. 3).
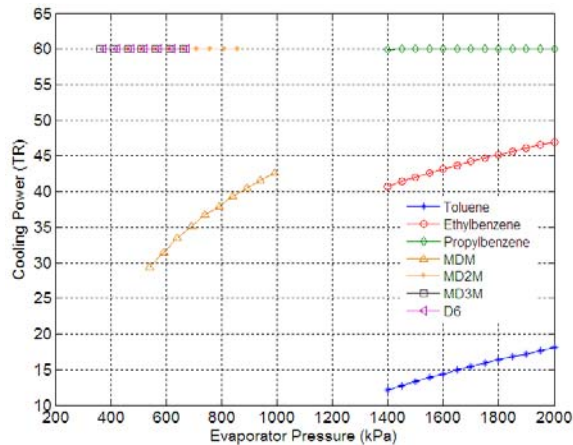

Figure 4. Variation of the cooling power production with the turbine inlet pressure.

Figure5 presents the global energy efficiency for each analyzed fluid. The results show that the cycles working with alkylbenzenes have the best performance among the evaluated organic working fluids. Propylbenzene shows the highest global efficiency among the alkylbenzenes while D6 shows the worst performance among the siloxanes. These results show the increase in the efficiency of the cycle caused by the recovery of part of the energy contained in the exhaust vapor flow in the turbine. Fluids with higher density and higher boiling point, operating with the same parameters in terms of evaporation and condensation pressure, have both higher evaporation temperatures and working fluid mass flow which allow to generate more thermal energy in the chiller increasing the global thermal efficiency.

Table 3 Summarizes the main results of simulations of the analyzed cycles at the maximum evaporator pressure.

The thermodynamic analysis presented in this paper introduces the exergetic efficiency and exergy destruction as evaluation parameters of the real performance. In that way, the irreversibility (*I*) and exergetic efficiency (*ε*) in the thermal system and each subsystem for the working fluids at maximum operating pressure are determined.
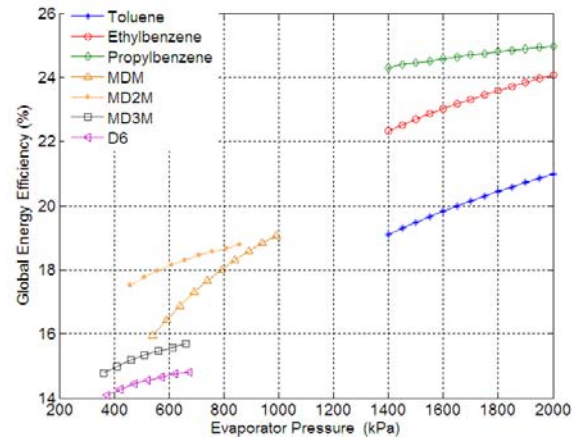

Figure 5. Variation of the global energy efficiency with the turbine inlet pressure.

Figure 6 shows that, for cycles working with alkylbenzenes, 15,8%, 14,9% and 14,4% for Toluene, Ethylbenzene, Propylbenzene respectively, of the resources that enter in system are transform into electricity and cold. While cycles that use the siloxanes have less exergetic efficiency, this is, 12,1%, 10,8%, 9,0% and 8,5% for MDM, MD$_2$M, MD$_3$M and D$_6$ respectively. According to the results presented in Fig. 6 Propylbenzene shows the highest global thermal efficiency. However, a combined first and second law analysis shows that the use of Toluene as a working fluid increases the system exergetic efficiency showing the importance of energy quality of the different products in the thermal system performance. Figure 7 can also be used to analyze the influence of the fluid boiling point temperature and density on the system thermal efficiency. Toluene has both the highest boiling point and the highest density among the selected fluids, while the fluid with the worst exergetic efficiency, Dodecamethylcyclohexasiloxane (D6), has both the lowest boiling point temperature and the lowest density. Similar tendency is observed for the remaining working fluids selected in this paper. This difference in exergetic efficiencies is due to the bigger exergy destruction (Figure 7). In cycles that use alkylbenzenes as working fluid the irreversibilities are among 30% - 55% lower when compared to cycles that use siloxanes.

The exergy analysis of each component is based on the operating conditions shown in Table. 3. The boiler and the condenser are the main responsible for the irreversibility rise in the cycle (Figure8). This is attributable to high biomass consumption in the boiler and higher quantity of heat rejected in the condenser. It is observed, in Fig. 8 , that these devices together are responsible for more than 90% of irreversibilities.

Table 3. Results of the cogeneration cycle.

| PARAMETERS	C_7H_8	C_8H_{10}	C_9H_{12}	MDM	MD_2M	MD_3M	D_6
Net Electrical Power (kW)	300	300	300	300	300	300	300
Cooling Power (kW)	63	164	210	149	210	210	210
Biomass Consumption (kg/h)	451	503	533	614	708	848	898
Working fluid mass flow (kg/h)	6734	6842	8048	18733	23488	29379	32675
Evaporation Temperature (K)	535	570	600	541	577	605	622
Condensation Temperature (K)	318	340	359	355	392	421	433
Auxiliary Equipment Consumption (kW)	5.5	6.0	6.7	8.3	8.4	8.9	8.8
Electric Efficiency (%)	17.3	15.5	14.6	12.7	11.0	9.2	8.7
Global Energy Efficiency (%)	20.9	24.0	24.9	19.0	18.7	15.6	14.8
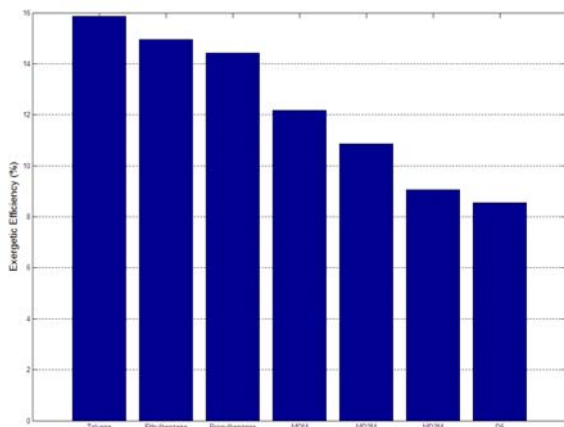
Figure 6. Exergy efficiency of the organic working fluids.

In the boiler (Fig. 8), the exergy destruction is caused mainly by the combustion process and the low rates of heat transfer between the combustion gases and the organic working fluid. Results show that working fluids with higher normal boiling point (T_b), even with lower average temperature difference with the combustion gases, have higher irreversibility in this equipment. The cycles with higher biomass consumption have higher exergy destruction rate during the biomass combustion. It is important to remark, that, in the exergy analysis in this paper, the boiler joins the evaporator to form a single subsystem.

In the condenser, the working fluids with higher density have higher condensing temperature. This is among 45 °C for Toluene with the lowest condenser irreversibility until 165 °C for D_6 with the highest condenser irreversibility at the same condensing pressure (10 kPa). In other words, fluids with higher densities, such as siloxanes, have higher average temperature difference with the cooling water. This may be explored, if there is a commercial or industrial district in a close distance to the cogeneration plant.

The rest of exergetic losses are concentrated in the vapor expansion system and cooling production. In the turbine the ratio of exergy destruction is around 3% for all alkylbenzenes and MDM while for the remaining working fluids is around 1,5%. Results show that cycles which use alkylbenzenes as working fluid have higher exergy destruction even with lower mass flow rates. Therefore those fluids have a higher specific entropy generation during the vapor expansion. Depending on which fluid is used (Alkylbenzenes) this difference can be up to 2-6 times higher than a siloxanes specific entropy generation in the turbine. In the chiller, Toluene, MDM and Ethylbenzene which are fluids with less capacity to produce cooling power, have lower exergy destruction respectively if compared to the other fluids. Those fluids have lower operating temperatures, so for that reason they have lower specific entropy generation reducing the irreversibilities in this equipment.
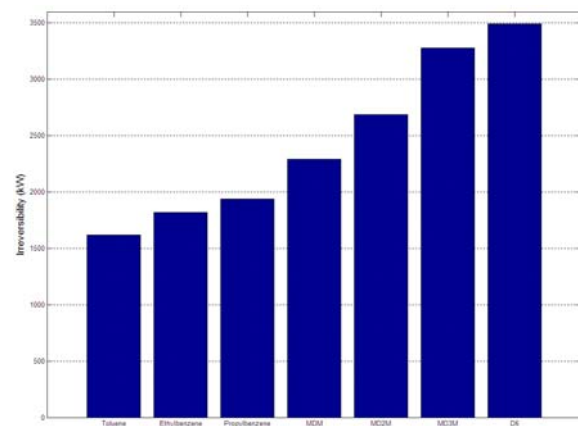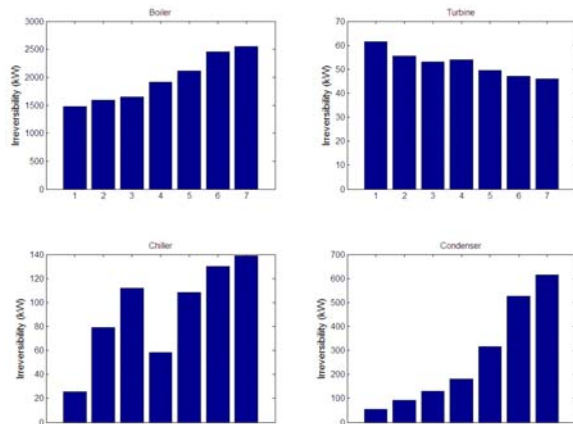

Figure 7. Exergy destruction of the organic working fluids.

1) Toluene, 2) Ethylbenzene, 3) Propylbenzene, 4) MDM, 5) MD_2M, 6) MD_3M, e 7) D_6

Figure 8. Exergy destruction by component.

CONCLUSIONS

The aim of the exergy analysis for a cogeneration power plant is usually to identify and quantify the sites of exergy destruction. Once used, it makes the irreversibility determination easier, identifying the components that cause more exergy destruction in the plant; allowing for the evaluation of the efficiency in each equipment. This information may be used to improve the performance of the system in order to have an effective use of the energy resources and to decide about improvement measures.

In general, the values of irreversibilities and exergy efficiencies depend on the working fluid that is being used. The results show that alkylbenzenes, even without heat regenerator, have the highest cycle exergy efficiency and the lowest exergy destruction. Although, those fluids show up more efficiency, they have a higher toxicity and flammability levels. For that reason, it is important to research thoroughly the use of those fluids in this type of energy systems.

ACKNOWLEDGEMENTS

REFERENCES

Angelino, G. and Colonna, P., 1998, Multicomponent working fluids for organic Rankine cycle (ORCs), Energy, Vol. 23, pp. 449-463.

Colonna, P., Nannana, N. R. and Guardone A., 2008, Multiparameter equations of state for siloxanes: $[(CH_3)_3\text{-Si-O}_{1/2}]_2\text{-}[O\text{-Si-}(CH_3)_2]_{i=1,\dots,3}$, and $[O\text{-Si-}(CH_3)_2]_6$, Fluid Phase Equilibria, Vol. 263, pp. 115–130.

Colonna, P., Nannana, N.R. and Guardone A., Lemmon, E.W., 2006, Multiparameter equations of state for selected siloxanes, Fluid Phase Equilibria, Vol. 244, pp. 193–211.

Drescher, U. and Brüggemann, D., 2007, Fluid selection for the Organic Rankine Cycle (ORC) in biomass power and heat plants, Applied Thermal Engineering, Vol. 27, pp. 223 – 228.

Duvia, A., Bini, R., Spanring, H. and Portenkirchner, K., 2007, Application of ORC units in the MDF and particleboard sector - general considerations and overview of the experiences of the first ORC in this industry installed at MDF Hallein, in: *15th European Biomass Congress and Exhibition*, Berlin, Germany.

Kotas, T. J., 1985, *The Exergy Method of Thermal Plant Analysis*, Departament of Mechanical Enginnering, Queen Mary College, University of London Butterwonths.

Poling, B. E., Prausnitz, J. M. and O'connell, J. P., 2004, *The properties of gases and liquids*, McGraw-Hill, fifth edition. New York, United States, 768 p.

Proust, P. and Vera, J. H., 1989, PRSV: The Stryjek-Vera Modification of the Peng-Robinson Equation of State. Parameters for Other Pure Compounds of Industrial Interest, The Canadian Journal of Chemical Engineering, Vol. 67, pp. 170-173.

Quoilin, S. and Lemort, V., 2009, Technological and Economical Survey of Organic Rankine Cycle Systems, in: *5th European Conference: Economics and Management of Energy in Industry*, Algarve, Portugal.

Rentizelas, A., Karellas, S. and Kakaras, E., Tatsiopoulos, I., 2009, Comparative techno-economic analysis of ORC and gasification for bioenergy applications, Energy Conversion and Management, Vol, 50, pp. 674–681.

Smith, R., 2005, *Chemical process design and Integration*, Ed Wiley, second edition. University of Manchester, England, 712 p.

Stryjek, R. and Vera, J. H., 1986, PRSV: An Improved Peng- Robinson Equation of State for Pure Compounds and Mixtures, The Canadian Journal of Chemical Engineering, Vol. 64, pp. 323-333.

Szargut J., Morris D. R. and Steward F.R., 1988, *Exergy Analysis of Thermal Chemical, and Metallurgical Processes*, Taylor & Francis Inc.

Tsatsaronis, G., 1993, Thermoeconomic Analysis and Optimization of Energy Systems, Progress in Energy and Combustion Systems, Vol 19, pp. 227-257.

Zaleta, A. A., Correas, U. L., Kubiak, S.J. and Sierra, E.F.Z., 2007, Concept on Thermoeconomic Evaluation of Steam Turbines, Applied Thermal Engineering, Vol. 27, pp. 457–466.