WORLD BIOENERGY 2012
Conference & Exhibition on Biomass for Energy
29 - 31 MAY 2012, JÖNKÖPING - SWEDEN

# ELECTRICITY GENERATION FROM BIOMASS: ORGANIC RANKINE CYCLE VERSUS STEAM CYCLE

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ABSTRACT: To generate electricity from biomass combustion heat, geothermal wells, recovered waste heat from internal combustion engines, gas turbines, or industrial processes, both the steam cycle and the organic Rankine cycle (ORC) are widely used. Both technologies are well established and can be found in comparable industrial applications. In this paper, we present a thermodynamic analysis and a comparative study of the cycle efficiency for a simplified steam cycle versus an ORC. We examine the application area of several working fluids based on their physical properties, considering some of the most commonly used organic fluids (R245fa, toluene, pentane, cyclopentane, and Solkatherm) and two silicon oils (MM and MDM). From computer simulations, we gain insight into the effect of several process parameters, such as the turbine inlet and condenser temperatures, the turbine isentropic efficiency, the vapor quality and pressure, and of the addition of a regenerator. We demonstrate that the thermal efficiency is primarily determined by the temperature level of the heat source and by the condenser conditions; and that the temperature profile of the heat source is the principal restricting factor for the evaporation temperature and pressure levels. Finally, we discuss some general and economic considerations relevant to the choice between a steam cycle and an ORC.
Keywords: organic rankine cycle (ORC), efficiency, electricity generation.

## 1 INTRODUCTION

The past years have seen considerable growth in the generation of power from industrial waste heat. Due to the rising energy prices, even the recovery of low grade waste heat is becoming increasingly profitable. A frequently applied technology is the transformation of waste heat into electricity by means of a conventional steam turbine. This method utilizes the waste heat to produce steam which is then expanded over the turbine to generate electricity.

However, the obtainable electric efficiency of this power cycle is limited for the often low temperature levels of waste heat sources, which put a constraint on the related maximum superheating temperature and evaporation pressure of the generated steam. An alternative solution, based on the same technology, is the organic Rankine cycle (ORC). An ORC installation uses the same components as a conventional steam power plant, i.e., a heat exchanger, an evaporator, an expander and a condenser, to generate electric power, but its working fluid is an organic medium instead of water and steam. These organic fluids have some favorable properties compared to water and steam [1–4]. For instance, most of these fluids can be characterized as 'dry' fluids, implying that, at least theoretically, superheating of the vapor is not necessary. These fluids can be used at a much lower evaporation temperature – and pressure – than water in a conventional steam cycle, while still yielding competitive electric efficiencies or, at low temperatures, even showing superior performance.

Biomass combustion heat occurs at higher temperature levels. Also in this regime, the ORC can be a viable alternative for the steam turbine, certainly when the simplicity of the operation is taken into account.

Today, standard ORC modules are commercially available in the power range from a few kW up to 10 MW. The technology has been proven and successfully applied for several decades in geothermal plants and in solar and biomass fired combined heat and power (CHP) plants. Waste heat is abundantly available in industrial processes, often at low temperature levels and on small to moderate thermal power scales. Several studies of the working fluid [1–14] and of the optimization, control and economic aspects of ORCs [15–20] have appeared in the literature. The objective of this paper is to evaluate and compare the performance of an organic Rankine cycle with a classic steam cycle for small and low temperature heat sources.

**Table I:** Thermophysical properties of the fluids in this study.

| Fluid | MW [kg/mol] | $T_{crit}$ [°C] | $P_{crit}$ [bar] | BP [°C] | $E_{evap}$ [kJ/kg] |
|---|---|---|---|---|---|
| Water | 0.018 | 373.95 | 220.64 | 100.0 | 2257.5 |
| Toluene | 0.092 | 318.65 | 41.06 | 110.7 | 365.0 |
| R245fa | 0.134 | 154.05 | 36.40 | 14.8 | 195.6 |
| n-pentane | 0.072 | 196.55 | 33.68 | 36.2 | 361.8 |
| cyclopentane | 0.070 | 238.55 | 45.10 | 49.4 | 391.7 |
| Solkatherm | 0.185 | 177.55 | 28.49 | 35.5 | 138.1 |
| OMTS | 0.237 | 290.98 | 14.15 | 152.7 | 153.0 |
| HMDS | 0.162 | 245.51 | 19.51 | 100.4 | 195.8 |

## 2 ORGANIC WORKING FLUIDS

To evaluate the characteristics of several organic fluids, we used the simulation software packages Fluidprop and Cycle Tempo [21], developed at Technical University of Delft. We considered the following commonly used organic fluids: R245fa, toluene, pentane, cyclopentane, and Solkatherm, and the silicon oils MM and MDM. In Table I, we reproduce some thermophysical properties for these fluids and for water.

From Table I it follows that the critical pressure and, consequently, also the operating pressure at the inlet of the turbine in a (subcritical) ORC system are much lower than in the case of a classical steam cycle. Although steam turbines exist that operate in a low pressure steam regime, the thermal efficiency of a steam cycle also decreases with lower turbine pressure.

All of the above organic fluids are dry fluids. Dry fluids are characterized by a positive slope of the saturated vapor curve in the temperature-entropy ($T$-$s$) diagram. Water, on the other hand, is a 'wet' fluid, with a negative slope. Dry fluids do not require superheating and saturated vapor can thus be supplied to an ORC expander. After expansion, the working fluid remains in the superheated vapor region. By comparison, in a steam cycle, the steam is usually superheated to avoid moisture formation in the final turbine stages, which would otherwise affect the performance and durability of the steam turbine.

As a general rule, a fluid with a higher boiling point has a lower condensation pressure at ambient temperature, and, after expansion, a lower density and a higher specific volume. For water and steam in particular, large diameters of the final turbine stages and a voluminous condenser are expected. Organic fluids have densities that are an order of ten times higher than the density of water/steam and therefore require smaller turbine diameters. However, the evaporation heat of organic fluids is also about ten times smaller, which corresponds to higher mass flows in the ORC, and hence much bigger feed pumps.

In short, the thermophysical properties of the working fluid have an effect on the design and complexity of the heat exchangers, the turbine and the condenser, and need to be taken into account during the economic analysis and comparison.

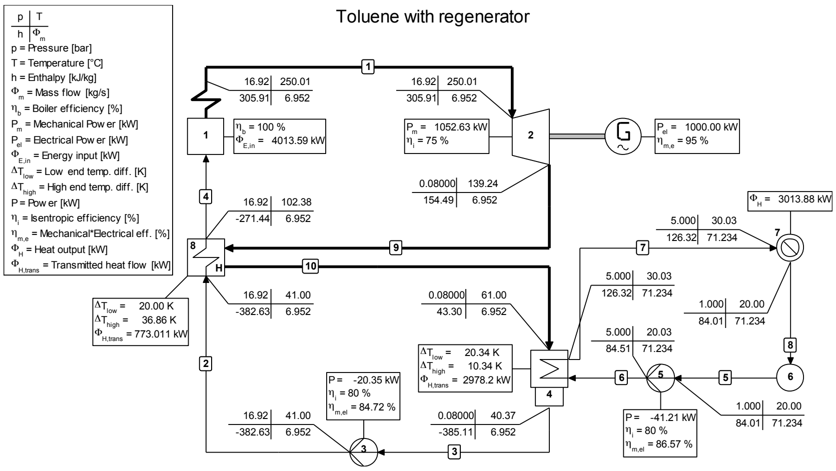


**Figure 1:** Cycle diagram of a toluene based ORC, equipped with a regenerator.

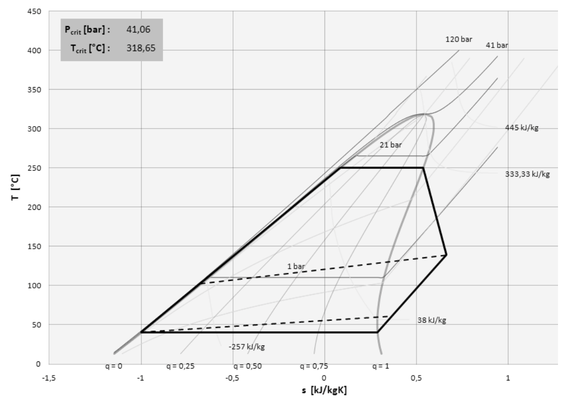


**Figure 2:** $T$-$s$ diagram of toluene and the ORC cycle steps corresponding to the specifications in Figure 1.

3 ORC VERSUS STEAM CYCLE

3.1 Organic Rankine cycle

In Figure 1, we reproduce a cycle diagram made with Cycle Tempo [21] of an ORC with toluene as the working fluid and with a regenerator. The corresponding thermodynamic cycle is represented by the $T$-$s$ diagram in Figure 2. A regenerator is often used to reach a higher cycle efficiency. After expansion, the organic fluid remains considerably superheated above the condenser temperature. This sensible heat can be exploited to preheat the organic liquid in a heat exchanger after the condenser stage. The higher the evaporation temperature, the higher the effect of a regenerator on the cycle efficiency. In Figure 3, this influence of a regenerator on the cycle efficiency is made apparent for the specific case of an ORC based on the silicon oil MM as a working fluid, assuming the parameters listed in Table II.

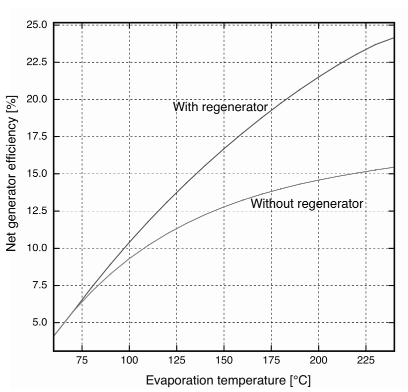


**Figure 3:** Net generator efficiency as a function of the evaporation temperature for an MM (silicon oil) based ORC, with and without regenerator. Parameters as in Table II.

**Table II:** Cycle parameters used in the analysis of ORCs and the steam cycle.

| Parameter | Unit | Value |
|---|---|---|
| Condenser temperature | [°C] | 40 |
| Isentropic efficiency turbine | [%] | 75 |
| Isentropic efficiency pump | [%] | 80 |
| Electromechanical efficiency pump | [%] | 90 |
| Electromechanical efficiency generator | [%] | 90 |
| Superheating temperature ORC | [°C] | 5 |
| Regenerator pinch | [°C] | 15 |
| Inlet turbine ORC | | Saturated |
| Inlet turbine steam | | Superheated |
| Vapor quality steam at outlet turbine | [%] | 90 |

3.2 Simplified steam cycle

Whereas saturated vapor can be applied in ORCs, a classic steam cycle commonly works with superheated steam. Steam turbines that can manage saturated steam typically have very poor isentropic efficiencies.

The inlet and outlet conditions of a steam turbine are correlated via the isentropic efficiency of the turbine. As a consequence, to achieve a prescribed vapor quality at the turbine outlet, the correspondence between the evaporation pressure and the minimum superheating temperature must be taken into account.

3.3 Assumptions

The assumptions mentioned in the presentation above of the ORC and the steam cycle, apply to the remainder of this paper. The performance of the cycles is evaluated assuming stationary conditions of all the components, with parameters mentioned in Table II. In addition, we assume that mass and energy are conserved in each cycle component, and that no pressure and energy losses occur. To enable the comparison between cycles based on wet and dry fluids, the optimal cycle determined by the predefined set of temperature levels of the heat source and the condenser is considered for each case. In this part of the study, we assume that the heat source is at a constant temperature level that also defines the turbine's inlet temperature. Hence, only cycles with identical turbine inlet temperatures and condenser temperatures are compared. Later in this paper, we extend our analysis by considering a predefined temperature profile of the heat source, together with an optimized turbine inlet pressure to maximally exploit the available heat.

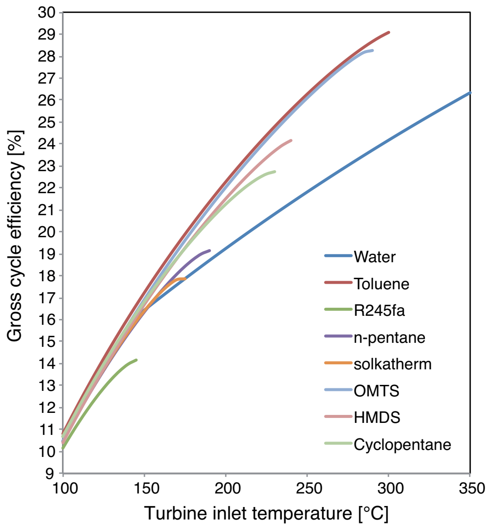


**Figure 4:** Cycle efficiency as a function of the turbine inlet temperature for all considered fluids.

3.4 Results

In Figure 4, the achieved cycle efficiency is plotted as a function of the turbine inlet temperature for all considered fluids. We note that for inlet temperatures below circa 130 °C it is impossible for the considered steam cycle to reach the preset turbine outlet conditions. Based on these results, we can state that: (i) ORCs display a superior performance compared to the simplified steam cycle, assuming equal temperatures at the turbine inlet; (ii) of the working fluids considered, the best ORC performance is achieved by toluene; and (iii) the application area of ORCs based on conventional working fluids, without superheating, is confined to temperatures below 300 °C.

3.5 Additional remarks and considerations

We conclude this part of the study with some additional remarks and considerations. In the field, there is a variety of expanders (e.g., turbine, screw expander) found in ORCs. Although isentropic efficiencies of 85% to 90% are attainable for turbines with a dedicated design, in practice, the isentropic efficiencies of small scale steam turbines, for low pressure applications, and with limited superheating temperature, are found to be lower than 75%. The efficiencies of commercially available ORCs may also fall short of our predictions, depending on the validity of our assumptions, such as for the pressures and temperatures at the inlet and outlet of the turbine and for the isentropic efficiency.

## 4 INFLUENCE OF THE TEMPERATURE PROFILE OF THE HEAT SOURCE

In reality, the temperature of a waste heat source does not remain at a constant level, but has a given temperature profile. This profile defines the thermal power Pth available between the inlet and outlet temperatures, and is dependent on the mass flow and medium type of the heat source. The closer the heating curves of the cycle (preheating, evaporation, and superheating) fit this temperature profile, the more efficiently the waste heat will be transformed by the ORC or steam cycle. In this part of the paper, we report our simulations for an arbitrary temperature profile of the waste heat source. In Table III, we list the parameters assumed for this part of the study.

**Table III:** Parameters used in the analysis of ORCs and the steam cycle.

| **Waste heat source** | | **Simplified steam cycle** | |
|---|---|---|---|
| $T$-profile | 350-120 °C | $T_{cond}$ | 40 °C |
| $P_{th}$ | 3000 kW$_{th}$ | $\eta_i$ turbine | 70-80% |
| pinch | 20 °C | vapor quality $q$ | 93% |
| **ORC** | | **Components** | |
| medium | HMDS | $\eta_i$ pump | 80% |
| $\Delta T_{sup}$ | 10 °C | $\eta_{m,e}$ pump | 90% |
| $T_{cond}$ | 40 °C | $\eta_{m,e}$ generator | 90% |
| $\eta_i$ turbine | 70-80% | | |

Design calculations of a heat exchanger, used in the recovery of industrial waste heat, lie beyond the scope of this study. Nevertheless, the effectiveness of the heat exchanger is taken into account by defining a 'pinch line' with an offset of 20 °C with respect to the temperature profile of the waste heat source. The attainable superheating temperature for the simplified steam cycle is then a function of the evaporation pressure $p_{evap}$, the vapor quality $q$, the condenser temperature $T_{cond}$ , and the isentropic efficiency $\eta_i$ of turbine, and is bounded by this pinch line.

In Table IV, we present our results for the gross ($P_{gen,bto}$) and net generator power ($P_{gen,nto}$) and the cycle efficiency η. The net generator power is given by $P_{gen,nto} = P_{gen,bto} - P_{pump}$, with $P_{pump}$ the pump power. Depending on pevap and the superheating temperature T su p, only a fraction Pth,reco of the thermal energy of the heat source can be recovered. In Figure 5, the corresponding heating profiles for the cases of Table IV are reproduced. This figure makes visible that the ORC pinch point is determined by the temperature after the regenerator. For the steam cycle, the selected evaporation pressure and superheating temperature are the constraining variables. Since the evaporation heat $E_{evap}$ for organic fluids is much smaller than for water, a higher evaporation temperature can be selected and therefore less thermal energy is required at higher temperature levels in an ORC. The result is a higher cycle efficiency and a 10 to 15% increase in electric power generation for the ORCs presented in this case study.

Also included in Table IV are our results for the optimization with respect to the net generator power of an ORC ($O_{opt}$) and a steam cycle ($S_{opt}$), for isentropic turbine efficiencies of 70% and 80%, and under the conditions of Table III.

**Table IV:** Gross and net cycle efficiencies, $\eta_{cycle,bto}$ and $\eta_{cycle,nto}$, and gross and net generator power, $P_{gen,bto}$ and $P_{gen,nto}$, of MM based ORCs ($O_1$, $O_2$, $O_3$, $O_4$) and steam cycles ($S_1$, $S_2$, $S_3$, $S_4$, $S_5$) with a temperature profile and parameters as indicated in Table III.

| Case | $\eta_i$ turb [%] | peva p [bar] | $T_{su\,p}$ [°C] | $\eta_{cycle,bto}$ [%] | $\eta_{cycle,nto}$ [%] | $P_{gen,bto}$ [$kW_e$] | $P_{gen,nto}$ [$kW_e$] | $P_{th,reco}$ [$kW_{th}$] |
|---|---|---|---|---|---|---|---|---|
| $O_1$ | 70 | 14.0 | 234 | 20.4 | 19.7 | 506 | 488 | 2479 |
| $O_2$ | 70 | 17.6 | 248 | 21.3 | 20.4 | 509 | 487 | 2388 |
| $O_{opt}$ | 70 | 14.9 | 239 | 20.7 | 19.8 | 513 | $\boxed{492}$ | 2478 |
| $S_1$ | 70 | 6.0 | 219 | 16.1 | 16.0 | 440 | 439 | 2737 |
| $S_2$ | 70 | 12.0 | 272 | 18.5 | 18.5 | 442 | 441 | 2386 |
| $S_3$ | 70 | 18.0 | 305 | 19.9 | 19.9 | 426 | 424 | 2134 |
| $S_{opt}$ | 70 | 7.9 | 320 | 17.6 | 17.6 | 454 | $\boxed{453}$ | 2572 |
| $O_3$ | 80 | 14.0 | 234 | 22.6 | 21.9 | 574 | 556 | 2540 |
| $O_4$ | 80 | 17.6 | 248 | 23.6 | 22.7 | 578 | 556 | 2452 |
| $O_{opt}$ | 80 | 16.3 | 244 | 23.3 | 22.4 | 583 | $\boxed{561}$ | 2505 |
| $S_4$ | 80 | 6.0 | 267 | 18.7 | 18.7 | 509 | 508 | 2715 |
| $S_5$ | 80 | 12.0 | 330 | 21.6 | 21.5 | 509 | 508 | 2357 |
| $S_{opt}$ | 80 | 7.9 | 320 | 20.2 | 20.1 | 519 | $\boxed{518}$ | 2571 |

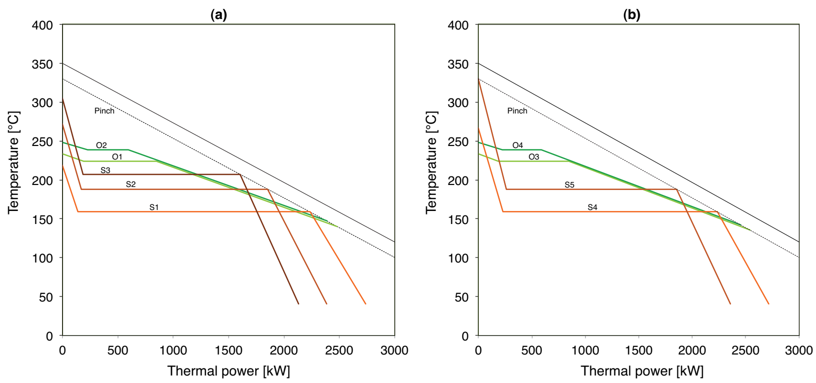


**Figure 5:** Heating profiles for the cases of Table IV. In (a) the isentropic turbine efficiency is 70%, in (b) it is 80%.

## 5 SELECTION ARGUMENTS

Based on our research, other studies, extensive experience and shared knowledge with constructors, suppliers and operators of both steam cycle and ORC based power plants, we list some general arguments that we think should be considered when faced with the choice between a steam cycle and an ORC. These factors should then be included in an investment, maintenance and exploitation plan.

The following are arguments in favor of an ORC implementation.

• Most organic fluids used in ORC installations are dry fluids that do not require superheating. Important factors in the total installation cost are the design and the dimensions of the heat exchangers (i.e., preheater, evaporator, and superheater) for the waste heat recovery.

• The isentropic efficiency of a turbine varies with its power range and its design. In general, ORC expanders with a dedicated design have a higher efficiency than small scale steam turbines in the same power range.

• There is no need for meticulous process water treatment and control, nor for a deaerator.

• The installation is less complex, which is desirable when starting from a green field or when there is no steam network with appropriate facilities already present on the site.

• Maintenance costs are very low and the availability is

high.
• The operation is very simple, usually only involving start and stop buttons.
• The behavior and efficiency under partial load is good.
• The system pressure is much lower and the applicable safety legislation is less stringent.
• A qualified operator is not required.
• Electrical outputs of less than 1 kWe are available. Small scale steam turbines (e.g. 10 kWe)
exist, but steam turbines only become profitable at higher power outputs (above 1 MWe).

The next arguments support the choice for a steam cycle.
• Water as a working fluid is cheap and abundant, while ORC fluids can be very expensive or their use restricted by environmental arguments. Large on-site steam networks, which require high amounts of working fluid (steam), are feasible.
• The flexibility of the power / heat ratio – important for biomass fired CHPs – can be higher due to the possibility to add steam extraction points on the turbine or by using a back pressure steam turbine.
• Direct heating and evaporation is possible in (waste) heat recovery heat exchangers, therefore there is no need for an intermediate (thermal oil) circuit.
• Some standard ORCs are designed to work with an intermediate thermal oil circuit to transport the waste heat to the ORC preheater and evaporator. This technique requires less ORC fluid, but tends to make the installation more complex and expensive, and results in an additional temperature drop. Furthermore, some fire accidents with thermal oil circuits are known.

## 6 CONCLUSIONS

The main conclusions that we draw from this study are the following. First, ORCs can function in combination with low temperature heat sources, characterized by low to moderate evaporation pressure, and still achieve better performances than steam cycles. Next, ORCs require larger feed pumps, because of their inherent higher mass flow, which in turn affects the net electric power. And finally, the heating curves of ORCs can be better matched to the temperature profiles of waste heat sources, resulting in higher cycle efficiencies and higher thermal power recovery ratios.

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