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USING ORGANIC RANKINE CYCLE FOR ELECTRICITY GENERATION FROM WASTE HEAT

Valmir de Oliveira Junior

Universidade do Estado do Rio de Janeiro - UERJ
valmir.deoliveira@gmail.com

Manoel Antonio da Fonseca Costa filho

Universidade do Estado do Rio de Janeiro - UERJ
manoelantonio.fonsecacosta@gmail.com

Abstract. *The Organic Rankine Cycle (ORC) operates similarly to a steam Rankine cycle with the same components, replacing the water by organic fluids with high molecular mass which have lower boiling points. This enables thermal recovery from milder temperature sources, being suitable for electricity generation from solar, geothermal, biomass and residual heat sources. The waste heat in exhaust gas from internal combustion engines corresponds to approximately 55% of the total heat released by the fuel. This manuscript aims at carrying out a thermodynamic modeling of an ORC designed to waste heat recovery from flue gas from the diesel engine Caterpillar 3516C. In this first approach, all assumptions from ideal Rankine Cycle were taken in account. The thermodynamic model for each component of the cycle is based on the mass and energy balances and was done using the software Engineering Equation Solver (EES) from the F-Chart Software. Only n-Pentane, Benzene and Toluene have proved suitable for operating at the mass flow between 0.4 kg/s and 0.5 kg/s using the regenerative cycle and the condensation, intermediary and evaporation pressures fixed at 0.1 MPa, 1.4 MPa and 2.8 MPa, respectively. Since cycle's efficiencies overcoming 19% were achieved, the ORC can be a practical device when applied for waste heat recovery from high size diesel engines.*

Keywords: *Organic Rankine Cycle, waste heat, energy efficiency. . .*

1. INTRODUCTION

In recent years, the accelerated usage of fossil fuels has caused serious environmental problems, highlighting the global warming. These problems have gradually intensified from the eighteenth century onwards with the beginning of the industrial revolution.

The use of fossil fuels as the main source of energy has contributed to a progressive increase in the concentration of carbon dioxide in the atmosphere and is now considered the biggest factor causing global warming.

In its first report in 2007, the Intergovernmental Panel on Climate Change (IPCC) made the following statement: "It is likely that most of the increase in global mean temperatures since the mid-20th century is due to anthropogenic concentrations of greenhouse effect gases. According to the different scenarios, the average temperature of the Earth should grow between 2 to 4°C before the end of the 21st century, with catastrophic consequences, such as sea level rise, higher occurrence of drought, consequently desertification, higher frequency of natural disasters, negative impact on biodiversity, etc. (Quoilin, 2011)

According to Quadrelli and Peterson (2007), 84% of greenhouse gas emissions are attributable to the energy sector, mainly in the form of carbon dioxide. These emissions are mainly due to industrialized countries, where the top ten emitters generate two-thirds of the world's total. It may be feared that the real development of the less industrialized countries will be accompanied by a large amount of CO₂ emissions.

Most world's energy demand is supplied by burning fossil fuels which are a sources of non-renewable energy. As they become scarce, the world is likely to depend more on electricity for the most diverse situations of the day and will require the search for new sources of primary energy that are easily converted into electricity. (DiPippo (2007))

New energy conversion technologies are needed to utilize adequate energy resources for clean energy generation without causing environmental damage. Low-grade heat sources are considered as candidates for new energy sources. Solar heat, waste heat and geothermal energy are typical examples of these sources, with temperatures available between 60 and 200°C. The use of such sources is an alternative for electricity generating systems. (Yamamoto *et al.* (2001))

The Rankine Cycle is a well-known process of converting thermal energy into useful work, by transferring heat from

a high temperature source to a low temperature source. Most global electricity generation come from Rankine Cycle using water as working fluid. The Organic Rankine Cycle (ORC) operates similarly to a steam Rankine cycle with the same components, replacing the water by organic fluids with high molecular mass which have lower boiling points. This enables thermal recovery from milder temperature sources, being suitable for electricity generation from solar, geothermal, biomass and residual heat sources.

The most critical elements in the development of practical systems with the Rankine Organic cycle are the choice of the most appropriate working fluid to absorb heat and convert it to energy and the cycle's own configuration. (Lu *et al.* (2016)) The working fluid thermophysical properties differ from water in several aspects, which has significant implications for the design of the organic cycle. (Quoilin (2011))

The waste heat in exhaust gas from internal combustion engines corresponds to approximately 55% of the total heat released by the fuel. Advances in internal combustion engines have focused to improve energy efficiency and have progressively reached very few percent reduction in the flue gas waste heat. Cogeneration systems are the most popular strategy for energy recovery in this case, but they request availability of heat consumers close to the facility site.

This manuscript aims at carrying out a thermodynamic modeling of an ORC designed to waste heat recovery from flue gas from a diesel engine.

2. WORKING FLUIDS CANDIDATES

The choice of working fluid, which alternately vaporizes and condenses when it absorbs and releases thermal energy, is the main part for the sizing of the cycle. Most often, it is necessary to make a comparison between a set of possible candidate fluids in terms of thermodynamic performance and based on the final purpose of the cycle.

According to Liu *et al.* (2011), the fluid needs to have specific characteristics depending on the application, for instance: appropriate saturation temperature, low critical pressure and temperature, reduced specific volume, low viscosity and surface tension, high thermal conductivity, non-corrosive and toxic, compatibility with turbine material, where appropriate in accordance with national and global environmental laws.

The software Engineering Equation Solver (EES) from the F-Chart Software (www.fchart.com) was utilized to elaborate the thermodynamic modeling, since it contains a library with various mathematical functions and the requested thermodynamic properties of all working fluids tested are readily available.

Figure 1 shows in the T-s diagram the saturation curves of water and of a few typical organic fluids in ORC applications. Two main differences can be stated:

- The slope of saturated vapor curve (right curve of the dome) is negative for water, while the curve is much more vertical for organic fluids. As a consequence, the limitation of the vapor quality at the end of the expansion process disappears in an ORC cycle, so there is no need to superheat the vapor before the turbine inlet;
- The enthalpy difference between saturated liquid and saturated vapor, i.e. the vaporization heat, is much smaller for organic fluids. Therefore, for the same useful power output, the organic working fluid mass flow rate must be much higher than of water, leading to a higher pump consumption.

Organic fluids usually remain superheated at the end of the expansion, therefore, there is no need for superheating in ORC cycles, contrary to steam cycle. The absence of condensation also reduces the risk of corrosion on the turbine blade, and increases its lifetime. (Bundela *et al.* (2010))

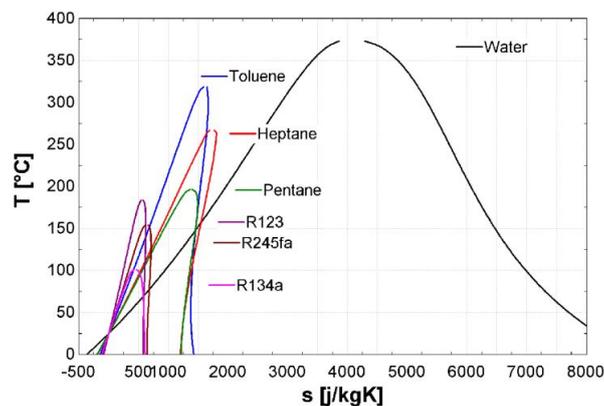


Figure 1. T-s diagram of a few typical organic fluids and of water (Quoilin (2011))

3. MATERIALS AND METHODS

The current investigation was made on the marine propulsion diesel engine Caterpillar 3516C which characteristics are presented in Tab. 1.

Table 1. Diesel Engine Technical Specifications.

Parameter	Specification
Engine type	V-16, 4-Stroke-Cycle-Diesel
Sales Model	3516C
Engine Heat to Atmosphere (kW)	149
Engine Speed (rpm)	1800
Engine Power (BKW)	2525
Exhaust Gas Flow Rate, \dot{m}_{exh} (m ³ /min)	53.9
Exhaust Stack Gas Temperature, T_{exh} (°C)	501.8

Table 2 presents a pre-selection of the working fluids to be tested. It was made in accordance with Quoilin (2011) that summarizes the scientific literature in the field of working fluid selection for ORC systems. After comparing several studies, those authors took in account three characteristics: the target application and the desired ranges of the condensing and evaporating temperatures.

Table 2. Working Fluid Pre-Selection.

Fluids		
R113	R123	R236ea
Benzene	n-Pentane	R11
Toluene	R245fa	R141b

The thermodynamic model for each component of the cycle is based on the mass and energy balances. All thermodynamic property ranges were limited to practical design constraints imposed to each equipment, such as obeying the minimum pinch points in the heat exchanger designs.

In this first approach, all assumptions from ideal Rankine Cycle were taken in account. According to Çengel and Boles (2015) and using the thermodynamic laws, all four components associated with the Rankine cycle (the pump, boiler, turbine and condenser) are steady-flow devices, Fig. 2, and thus all four processes that make up the Rankine cycle can be analyzed as steady-flow processes. The kinetic and potential energy changes of the working fluid are usually small relative to the work and heat transfer terms and are therefore usually neglected. Then the steady-flow energy equation per unit mass of working fluid reduces to:

$$(q_{in} - q_{out}) + (w_{pump,in} - w_{turb,out}) = (h_{end} - h_{initial}) \quad (1)$$

Where: q, w and h represent, respectively, heat, work and enthalpy in SI units.

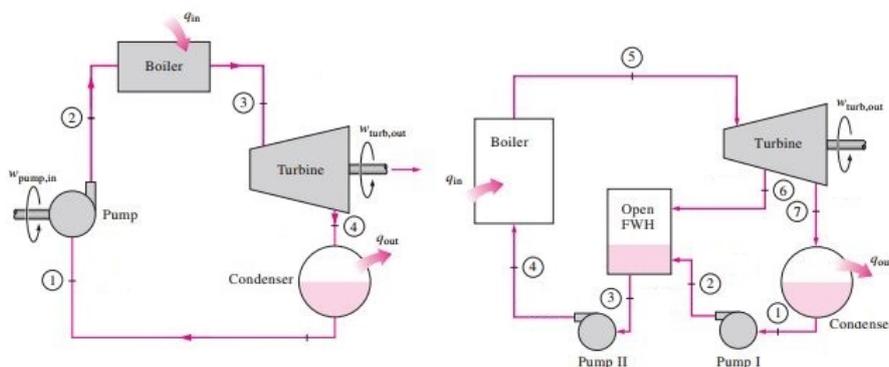


Figure 2. The ideal Simple and Regenerative Organic Rankine Cycle, respectively, (Çengel and Boles (2015)) - Adapted

In this study, due to the high temperature of the exhaust gases, it was necessary to perform the modeling of two types of ORC (simple and regenerative, Fig. 2), for all the fluids suggested by Quoilin (2011) and shown in Tab. 2, varying the organic fluid mass flow, \dot{m} in the case simple cycle and \dot{m}_5 in the regenerative.

The calculation of the pump and turbine power, respectively $w_{pump,in}$ and $w_{turb,out}$, as well as the efficiency of the cycle. The boiler and the condenser do not involve any work, and the pump and the turbine are assumed to be isentropic. According to Çengel and Boles (2015), the value of the isentropic efficiency of a turbine and a pump ranges from 70 to 90 percent.

Then the conservation of energy applied to each device can be expressed as follows:

For ORC simple,

$$w_{pump,in} = \dot{m}(h_2 - h_1) \quad (2)$$

$$w_{turb,out} = \dot{m}(h_3 - h_4) \quad (3)$$

$$\eta_{ORC} = (w_{turb,out} - w_{pump,in})/q_{in} \quad (4)$$

For ORC Regenerative,

$$q_{in} = h_5 - h_4 \quad (5)$$

$$q_{out} = (1 - y)(h_7 - h_1) \quad (6)$$

$$w_{turb,out} = ((h_5 - h_6) + (1 - y)(h_6 - h_7))\dot{m}_5 \quad (7)$$

$$w_{pump I,in} = v_1(P_2 - P_1)\dot{m}_7 \quad (8)$$

$$w_{pump II,in} = v_3(P_4 - P_3)\dot{m}_5 \quad (9)$$

$$w_{pump,in} = (1 - y)w_{pump I,in} + w_{pump II, in} \quad (10)$$

Where y (fraction of steam extracted) can be calculated, through Eq.(11) and Eq.(12)

$$y = \dot{m}_6/\dot{m}_5 \quad (11)$$

$$\dot{m}_5 = \dot{m}_6 + \dot{m}_7 \quad (12)$$

The evaporator effectiveness (ε) is an important design parameter that needs to be selected carefully to prevent water condensation in the evaporator tubing.

The pinch point temperature difference (PPTD) corresponds here to the minimum difference between the exhaust gas temperature and the temperature organic fluid evaporating temperature. El-Wakil (1984) cite that too small a PPTD results in low overall temperature difference and, hence, lower irreversibilities, but in a large and costly steam generator; too large PPTD results in a small, inexpensive steam generator but large overall PPTD and irreversibilities and, hence, reduction in plant efficiency. The most economical PPTD is obtained by optimization that takes into account both fixed charges (based on capital costs) and operating costs (based on efficiency and, hence, fuel costs).

Figure 3 shows temperature-heat exchanger path length for counterflow (steam generators). The minimum approach point between the two lines is the PPTD, represented by b-B, must be defined.

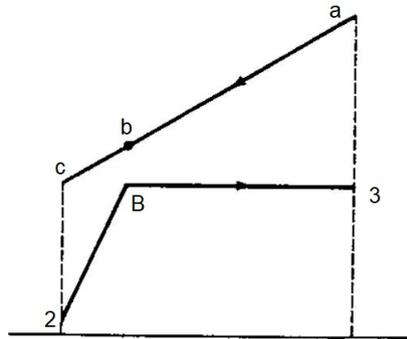


Figure 3. Temperature-heat exchanger path length diagram for the boiler (El-Wakil (1984)) - Adapted

According to Srinivasan *et al.* (2010), the pinch point temperature (b) can be estimated as:

$$T_{pinch} = [(T_a - T_c) / \Delta H_{23}](\Delta H_{2B} - \Delta H_{23}) + T_a \quad (13)$$

The PPTD can be calculated as:

$$PPDT = T_{pinch} - T_B \quad (14)$$

Some considerations about the system modeled:

- The isentropic efficiencies of the pump and turbine are $\eta_p=80\%$ and $\eta_T=82\%$, respectively;
- The effectiveness (ε) of the boiler is 0.7;
- All thermodynamic properties come from the EES database;
- A prefixed minimum PPTD of 10K must be considered;
- The ambient condition is specified by 298K and 1 atm;
- Diesel engine exhaust gas is assumed to be air;
- The characteristic gas constant of the exhaust was assumed to be approximated by the characteristic gas constant of air;
- The condensation, intermediary and evaporation pressures are fixed at 0.1 MPa, 1.4 MPa and 2.8 MPa, respectively.

4. ORGANIC RANKINE CYCLE ANALISYS

To model the ORC, heat and mass balances are done for each component, in both single and regenerative ideal Rankine cycles. Exhaust gas temperature pressure and flow rate from Tab 1 have been used as inputs for the bottoming ORC model to obtain a realistic idea of the possible benefits of Wasted Heat Recovery.

The specific gas constant of the exhaust of the air (R), and the specific heat of the exhaust gases at constant pressure (c_p) was calculated using:

$$c_p = \gamma R / \gamma - 1 \quad (15)$$

Where specific heat ratio is calculated for air at average temperature of 501.8°C (T_{exh})

The maximum energy that can be extracted from the exhaust can be determined as:

For ORC simple

$$\dot{Q}_{max} = \dot{m}_{exh} c_p (T_{exh} - T_2) \quad (16)$$

For ORC Regenerative

$$\dot{Q}_{max} = \dot{m}_{exh} c_p (T_{exh} - T_4) \quad (17)$$

The heat that can be recovered in the boiler as well as the temperature of the exhaust leaving the boiler (T_{amb}) can be estimated using the heat exchanger effectiveness as follows:

$$\dot{Q}_{heat} = \varepsilon \dot{Q}_{max} = \dot{m}_{exh} c_p (T_{exh} - T_{amb}) \quad (18)$$

The thermal efficiency of the Organic Rankine Cycle is determined from

$$\eta = w_{net} / q_{in} \quad (19)$$

$$w_{net} = w_{turb,out} - w_{pump,in} \quad (20)$$

5. RESULTS

Some fluids were discarded after modeling the whole systems, since they did not present the expected result in relation to cycle efficiency, condensation and evaporation temperatures, and restriction of 10K as PPTD in the boiler. The fluids that have presented the best performances were chosen as suitable work fluids: Benzene, Toluene and n-Pentane.

Figure 4 shows the cycle's efficiency as function of the organic fluid mass flow for the n-Pentane, Benzene and Toluene, both for simple cycle and for regenerative cycle.

It is possible to note that small for mass flow rates in both cycles, the Benzene presents a good yield, consequently, producing good power. However, as the flow increases, the supremacy of the regenerative cycle over the simple cycle is clear for all fluids. Among the three fluids, benzene was the one that presented the best performance, as it can be observed in Fig. 4 and in Tab. 3.

6. CONCLUSIONS

In this paper, the exhaust gas waste heat recovery potential from the diesel engine Caterpillar 3516C using a bottoming Organic Rankine Cycle was examined.

Furthermore, it was considered, the isentropic efficiency of cycle's equipment beyond the basic design criteria for the evaporator were established using a Pinch Point analysis.

It can be concluded, that maintaining the organic fluid mass flow between 0.4 and 0.5 kg/s, using the regenerative cycle, the ORC can be a practical device when applied for waste heat recovery from high size diesel engines.

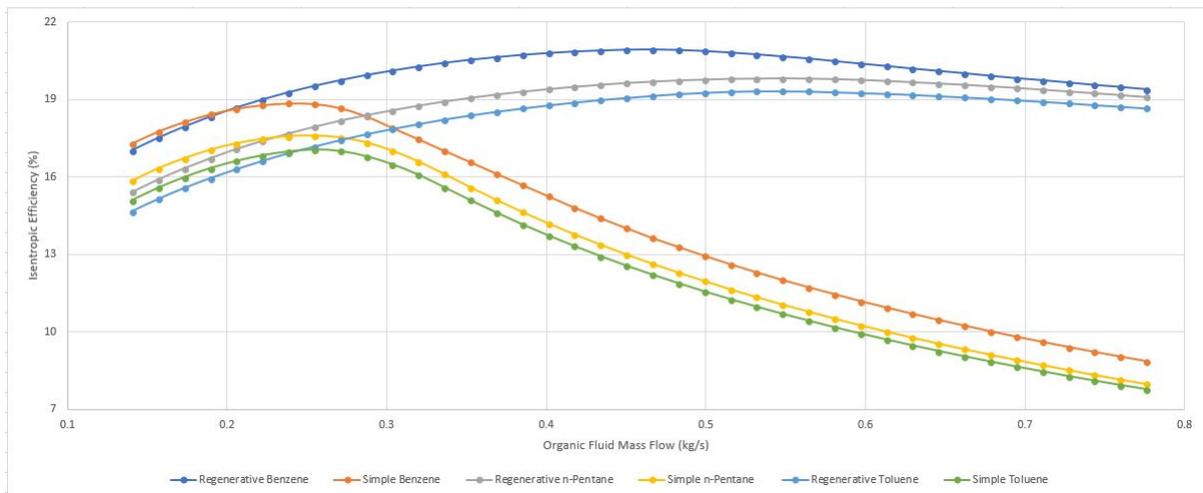


Figure 4. Effect of organic fluid mass flow on cycle efficiency

Table 3. Effect of organic fluid mass flow on the net power produced

Organic fluid mass flow [kg/s]	Regenerative Benzene [kW]	Simple Benzene [kW]	Regenerative n-Pentane [kW]	Simple n-Pentane [kW]	Regenerative Toluene [kW]	Simple Toluene [kW]
0.1894	27.31	27.46	24.92	25.38	23.77	24.33
0.2546	29.08	28.07	26.73	26.21	25.6	25.41
0.3197	30.2	26.04	27.95	24.74	26.9	23.97
0.3849	30.87	23.39	28.76	21.82	27.81	21.1
0.4501	31.17	20.89	29.26	19.38	28.42	18.72
0.5153	31.01	18.81	29.5	17.36	28.77	16.78
0.5805	30.52	17.07	29.49	14.24	28.52	13.79
0.6456	29.95	15.59	29.24	14.24	28.52	13.79
0.7108	29.41	14.31	28.88	12.99	28.19	12.6
0.776	28.89	13.2	28.45	11.9	27.8	11.57

7. ACKNOWLEDGMENTS

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