# TECHNO-THERMODYNAMIC MULTI-OBJECTIVE OPTIMIZATION OF ORGANIC RANKINE CYCLES FOR WASTE HEAT RECOVERY

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## ABSTRACT

Organic Rankine cycles (ORC) are considered a mature technology in waste heat recovery applications. Yet, many of the commercially available implementations have comparable design characteristics. They operate under subcritical regime and only few working fluids are considered. In this work, three ORC configurations are analyzed on thermodynamic and technical grounds. The cycles under consideration are the subcritical cycle (SCORC), the transcritical cycle (TCORC) and the partial evaporation cycle (PEORC). Technical design constraints on the expander generally limit the cycles which can be considered feasible. Yet, their impact on the optimization process is not clear. Therefore, in this work, the technical performance criteria are optimized in conjunction with maximization of the net power output. Thus the investigation results in a multi-objective optimization strategy. The proposed strategy is performed on two representative but distinctive waste heat recovery cases. The results of the investigation are particularly useful for manufacturers of ORCs. The thermodynamic performance of the cycles is compared under equal boundary conditions, expander criteria are taken into account, actual cases are used and a ranking is created.

## INTRODUCTION

The organic Rankine cycle is a mature and cost effective technology to convert low capacity/low temperature heat to electricity. Typical benefits attained to the ORC are: autonomous operation, favorable operating pressures and low maintenance costs [1]. Different heat sources are used as input to the ORC. Waste heat applications roughly consist of up to 20% [2] of the ORC market, preceded by geothermal and biomass installations. Yet, the share of waste heat recovery applications can be expected to increase.

First, in view of increasing energy demand and environmental concerns it becomes essential to use our natural resources more efficiently. This is directly reflected in the 202020 targets [3] which aims for a 20% improvement in the EU's energy efficiency. Recovery of heat from industrial process is evidently an effective measure to make better use of our resources. Second, there are still large amounts of waste heat which are unused. In Canada 70% [4] of the input energy from the eight largest manufacturing sectors is discarded to the atmosphere. For the U.S., estimates are between 20% and 50% [5] of industrial energy that is lost as waste heat. In Europe alone, 140 TWh/j [6] of waste heat is available.

## NOMENCLATURE

| | | |
|---|---|---|
| $\dot{E}$ | [kW] | Exergy flow rate |
| $\dot{e}$ | [kW/kg] | Specific exergy |
| $F$ | [-] | Dimensionless ORC parameter |
| $h$ | [kJ/kg] | Specific enthalpy |
| $\dot{m}$ | [kg/s] | Mass flow rate |
| $N$ | [-] | Number of segments |
| $p$ | [Pa] | Pressure |
| $PP$ | [°C] | Minimum pinch point temperature difference |
| $\dot{Q}$ | [kW] | Heat flow rate |
| T | [°C] | Temperature |
| $v$ | [m³/kg] | Specific volume |
| $VC$ | [m³/MJ] | Volume coefficient |
| $\dot{W}$ | [kW] | Power |
| Special characters | | |
| $\eta$ | [-] | efficiency |
| Subscripts | | |
| $0$ | | Dead state |
| $I$ | | Thermal efficiency |
| $II$ | | Second law effciency |
| $avg$ | | Average |
| $c$ | | Condenser |
| $cf$ | | Cooling loop condenser |
| $crit$ | | Critical |
| $e$ | | Evaporator |
| $hf$ | | Heat carrier |
| $isen$ | | Isentropic |
| $net$ | | Nett |
| $wf$ | | Working fluid |
| Abbreviations | | |
| SCORC | | Subcritical ORC |
| TCORC | | Transcritical ORC |
| PEORC | | Partial evaporation ORC |

An additional drive for adoption of ORC technology is increasing their performance. The current commercially available ORCs are typically of the subcritical type. However, performance gains over the subcritical ORC are reported for, amongst others, multi-pressure cycles [7-10], triangular cycles [11-13], cycles with zeotropic working fluids [14-16] and transcritical cycles [11, 17-20]. These last three cycles have an identical basic component layout in common, consisting of an expander, evaporator (or vapour generator for the transcritical cycle), condenser and pump.

The expander is a key component of the system. Technical limitations to the expander design limit the choice of working fluids and cycle architectures. The thermodynamic performance criteria associated to the expander design are the size factor (SF) and volume ratio (VR) for the turbine and the volume coefficient (VC) for volumetric machines. A limited region of feasible values can be associated to these performance criteria [21-24]. The finale design follows from a techno-economic optimization but falls within the technical limitations. Therefore a preliminary selection of cycles and working fluids can be made.

As several performance criteria need to be optimized simultaneously a multi-objective optimization is preferred. In literature, this type of optimization is already applied on ORCs [25, 26]. However, to the authors' knowledge the expander performance criteria are in multi-objective optimizations never considered as objective criteria.

Therefore, in the presented paper, a methodology for the pre-selection of working fluids and cycle architectures is proposed based on a multi-objective optimization. Three cycle types, namely the subcritical ORC (SCORC), partial evaporation ORC (PEORC) and transcritical ORC (TCORC) are investigated. In total 67 possible working fluids are considered. At the hand of two representative waste heat recovery cases the capability of the optimization strategy is highlighted. Furthermore, the impact of expander criteria on ORC performance is investigated. General guidelines and recommendations are made.

## ORC CYCLES

Three cycles architectures are investigated: the subcritical ORC (SCORC), the partial evaporation ORC (PEORC) and the transcritical ORC (TCORC). The cycle layout is identical for the three architectures and shown in Figure 1. The T-s diagram which introduces the nomenclature used is given in Figure 2.

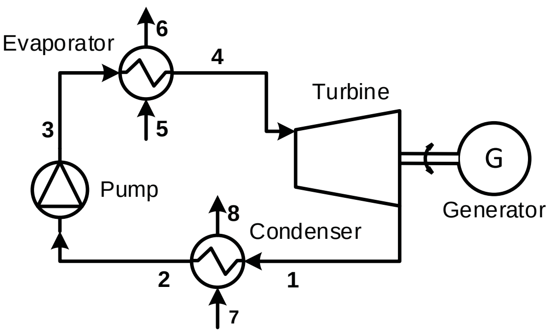


**Figure 1:** Common cycle architecture component layout

## CASE DEFINITION

A classification of four main waste heat recovery groups is compiled in Table 1. Low temperature cooling loops are found in incinerator installations [27], aluminum factories [28] and combined heat and power installations [28]. High temperature cooling loops consist of intermediate thermal oil loops or pressurized water circuits. This type of waste heat can again be found in incinerator installations [29] but also in chemical or steel industry [30]. Low temperature flue gas is found in drying processes, annealing furnaces or exhaust gas from internal combustion engines [1]. High temperature flue gas is commonly found in the steel industry, cement industry or exhaust gasses from gas turbines [1].

**Table 1** Four main waste heat recovery groups.

| Group | Description | Loop | T range [°C] |
|---|---|---|---|
| I | Low temperature cooling | Closed | 80-100 |
| II | High temperature cooling | Closed | 100-180 |
| III | Low temperature flue gas | Open | 180-250 |
| IV | High temperature flue gas | Open | 250-350 |

The closed loop configurations have a fixed heat input to the ORC, the goal is to increase the thermal efficiency, while for the open loop the goal is to increase the net power output [31]. Furthermore, an upper cooling limit can be assigned to the open loops. This is done in order to avoid condensation of flue gasses. This again results in a fixed heat input to the ORC.

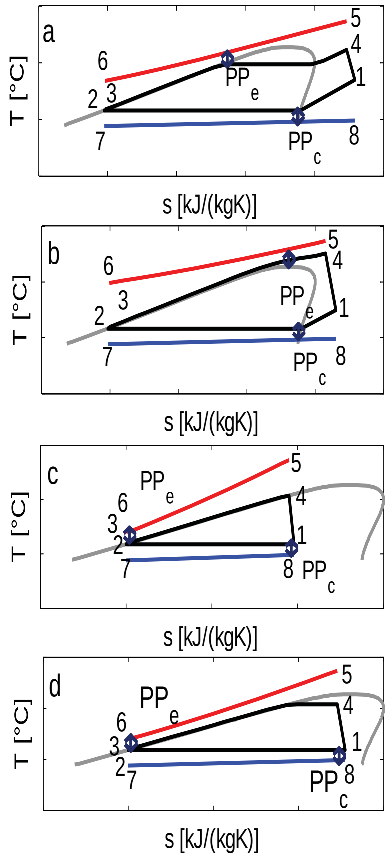


**Figure 2:** T-s diagram (a) SCORC, (b) TCORC, (c) TLC, (d) PEORC.

A representative case for each group is given in Table 2. The proposed cases and classification are based on data gathered in the ORCNext project [6]. In this work both Case 3 and Case 4 will be investigated with volumetric machines as expander.

**Table 2** Definition of waste heat recovery cases.

| Case | Group | Description | $T_{avg}$ [°C] | $\dot{Q}$ [MWt] | $\dot{m}$ [kg/s] |
|---|---|---|---|---|---|
| 1 | I | Water cooling loop from incinerator | 90 | 1.5 | 10 |
| 2 | II | Pressurized water cooling loop from incinerator | 180 | 15 | 70 |
| 3 | III | Flue gas from drying process | 240 | - | 36 |
| 4 | IV | Flue gas from electric arc furnace | 305 | - | 43 |

## MODEL AND ASSUMPTIONS

The parameters characterizing the cycle are shown in Table 3. The cycles are modelled under the assumption of steady state operation. Heat losses to the environment and pressure drops in the heat exchangers are, considered negligible. A discretization approach is implemented for modelling the heat exchangers. The evaporators are segmented into $N$ parts. As such, changing fluid properties are taken into account. This is particularly essential for the TCORC vapour generator. Details about the modelling approach are found in previous work by the authors [15, 32].

**Table 3** Thermodynamic cycle parameters.

| Parameter | Description | Value |
|---|---|---|
| $\eta_{pump}$ | Isentropic efficiency pump [%] | 70 |
| $\eta_{turbine}$ | Isentropic efficiency turbine [%] | 80 |
| $PP_e$ | Pinch point temperature difference evaporator [°C] | 5 |
| $PP_c$ | Pinch point temperature difference condenser [°C] | 5 |
| $T_5$ | Heat carrier inlet temperature [°C] | Case dependent |
| $T_2$ | Cooling loop inlet temperature [°C] | 20 |
| $\Delta T_{cf}$ | Cooling loop temperature rise [°C] | 10 |
| $\dot{m}_{hf}$ | Mass flow rate heat carrier [kg/s] | Case dependent |

The total heat to the cycle follows from:

$$\dot{Q}_{evap} = \sum_{i=1}^{N} (h_{wf,e,(x+1)} - h_{wf,e,x})\dot{m}_{wf} \quad (1)$$

The turbine and pump are modelled by their isentropic efficiency:

$$\dot{W}_{pump} = \frac{h_{pump,out}^{isen} - h_{pump,in}}{\eta_{pump}} \dot{m}_{wf} \quad (2)$$

$$\dot{W}_{pump} = \frac{h_{pump,out}^{isen} - h_{pump,in}}{\eta_{pump}} \dot{m}_{wf} \quad (3)$$

The net power output is given as:

$$\dot{W}_{net} = \dot{W}_{turbine} - \dot{W}_{pump} \quad (4)$$

## PERFORMANCE EVALUATION CRITERIA

The thermodynamic performance criteria used in this work are briefly introduced. First the thermal efficiency is defined as:

$$\eta_I = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \quad (5)$$

The second laws efficiency is given as:

$$\eta_{II} = \frac{\dot{W}_{net}}{\dot{E}_{hf,in}} \quad (6)$$

The exergy flow $\dot{E}$ is obtained by multiplying the specific exergy with the mass flow rate:

$$\dot{E} = \dot{m}e \quad (7)$$

The specific exergy e for a steady state stream, assuming potential and kinetic contributions are negligible, is defined as:

$$e = h - h_o - T_0(s - s_0) \quad (8)$$

The dead state ( $p_0, T_0$ ) is defined as the inlet temperature of the condenser cooling loop.

The volume coefficient is given as:

$$VC = \frac{v_{\exp,out}}{h_{in,\exp} - h_{out,\exp}} \quad (9)$$

The VC value directly correlates with the size of the expander. General design ranges can be associated to the expander evaluation criteria. This provides a rough means for pre-selection of working fluids and cycle architectures. In refrigeration and heat pump applications the VC ratio is typically between 0.25 and 0.6 m³/MJ [21].

## OPTIMIZATION STRATEGY

The proposed model has two degrees of freedom left. Depending on the cycle architecture these are typically defined as:

- The superheating and evaporation pressure, for the SCORC.
- The vapour quality and evaporation pressure, for the SCORC
- The turbine inlet temperature and supercritical pressure (TCORC).

In this work however, two dimensionless parameters $F_p$ and $F_s$ are introduced. Depending on their value they correspond to one of the three cycles under investigation. The benefit is that there is only one uniquely defined search space under which the three cycles under investigation are simulated.

Both dimensionless parameters have a range [0,1] and their definition is given below:

$$F_p = \frac{p_{wf,e} - p_{\min}}{p_{\max} - p_{\min}} \quad (10)$$

$$p_{\max} = 1.3 p_{wf,crit} \qquad T_5 - PP_e > T_{wf,crit} \quad (11)$$

$$p_{wf,sat}(T = T_5 - PP_e) \qquad T_5 - PP_e < T_{wf,crit} \quad (12)$$

$$p_{\min} = p_{wf,sat}(T = T_8 + PP_c) \quad (13)$$

and

$$F_s = \frac{s_4 - s_{\min}}{s_{\max} - s_{\min}} \tag{14}$$

$$s_{\min} = s_{wf,sat,liq}(p = p_{wf,e}) \qquad p_{wf,e} < p_{wf,crit} \tag{15}$$

$$s_{\min} = s_{wf,crit} \qquad p_{wf,e} > p_{wf,crit} \tag{16}$$

$$s_{\max} = s_{wf}(p = p_{wf,e}, T = T_5 - PP_e) \tag{17}$$

For the cycles under consideration with volumetric expanders both the second law efficiency and the VC are minimized. The VC is constraint to a range [0.1, 3.5]. The optimization problem is formulated as:

$$\begin{aligned} &\text{minimum } [VC(x), \eta_{II}(x)] \\ &x = (F_s, _p) \\ &s.t \qquad 0 < F_s < 1 \\ &\qquad\quad\; 0 < F_p < 1 \end{aligned} \tag{18}$$

A genetic algorithm is used to perform the multi-objective optimization. The implementation is based on the NSGA-II algorithm [33]. The Parameter settings of the genetic algorithm are provided in Table 4. A population size of 10000 with 100 generations gave an acceptable calculation time of approximately 4 hours using an Intel E5-2736 v2 processor. Yet, in other works, with a comparable number of optimization variables, population sizes between 40 [34] and 150 [35] are employed.

**Table 4** Parameter settings of the genetic algorithm.

| Parameter | Value |
|---|---|
| Generations | 100 |
| Population size | 10000 |
| Crossover rate | 0.8 |
| Migration rate | 0.2 |
| Mutation type | Gaussian (shrink = 1, scale = 1) |
| Pareto fraction | 0.35 |

## RESULTS AND TRENDS

The Pareto plots of Case 3 and Case 4 are respectively visualized in Figure 3 and Figure 4. It is directly clear that for these cases there are working fluids available which attain both good second law efficiency and feasible VC values to design a volumetric expander. The working fluids following from the optimization are ammonia, propyne, R142b, isobutene, trans-2-butene, R1233ZDE, R123, isopentane and n-pentane for Case 3 and ammonia, sulferdioxide, R21, R11, R141b and cyclopentane for Case 4. For case 4, finding an environmentally friendly working fluid (Ozone Depletion Potential < 0 and Global Warming Potential < 150) proves to be difficult as only cyclopentane and ammonia qualify.

Furthermore, for increased second law efficiency there is a clear tradeoff between $\eta_{II}$ and VC. For each working fluid a small change in VC results in large differences in $\eta_{II}$. Optimizing the operating parameters allows to significantly increase $\eta_{II}$ but the impact on the VC is small. Thus a good working fluid selection is dominant over optimizing the operating parameters.

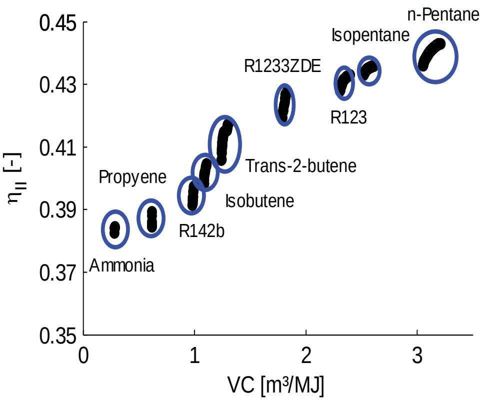


**Figure 3: Case 3, Pareto front of VC and $\eta_{II}$**

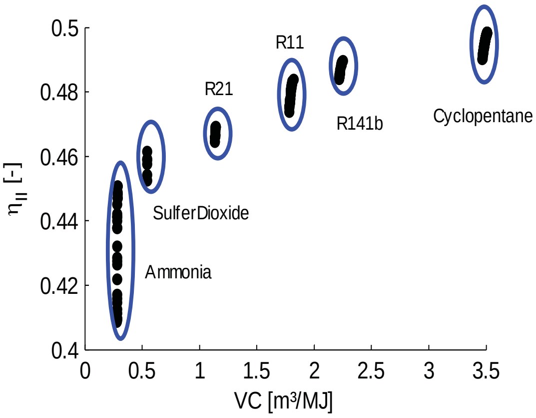


**Figure 4: Case 4, Pareto front of VC and $\eta_{II}$**

**Table 5** Critical temperature, critical pressure and temperature at normal boiling point for the resulting working fluids.

| Working fluid | $T_{crit}$ [K] | $p_{crit}$ [MPa] | $T_{boiling}$ [1] [K] |
|---|---|---|---|
| n-pentane | 467.7 | 3.37 | 309.2 |
| isopentane | 460.0 | 3.38 | 300.9 |
| R123 | 456.8 | 3.67 | 300.9 |
| R1233zde | 438.7 | 3.57 | 291.4 |
| trans-2-butene | 428.6 | 4.03 | 274.0 |
| isobutene | 418.1 | 4.00 | 266.1 |
| R142b | 410.2 | 4.05 | 264.0 |
| propyne | 402.4 | 5.63 | 248 |
| ammonia | 405.4 | 11.33 | 239.8 |
| cyclopentane | 511.7 | 4.57 | 322.4 |
| R141b | 477.5 | 4.21 | 305.2 |
| R11 | 471.1 | 4.41 | 296.8 |
| R21 | 451.5 | 5.18 | 282.0 |
| sulfordioxide | 430.6 | 7.88 | 263.1 |

[1] at 101.325 kPa

Next the cycle types are investigated. For Case 3 all cycles are of the TCORC type. This is explained by the good match of working fluid supercritical temperature and heat carrier inlet temperature. As such, high second law efficiencies are attained.

In contrast, for Case 4 the optimal cycle type is always a SCORC. For these high temperatures, going to a supercritical state is less beneficial in terms of second law efficiency. Therefore the dominant factor in the optimization is the reduction of the VC. Furthermore from Table 5 follows that a decrease in normal boiling point temperature results in both lower VC and second law efficiency.

Due to the clear link between VC and second law efficiency, many working fluids can already be discarded and a ranking can be made. However the final operating parameters and sizing of the cycle should follow from thermo-economic considerations. In the thermo-economic optimization also the specific heat transfer equipment should be designed. The pre-selection made in this work allows focusing on the most promising operating regimes and cycle architectures.

## CONCLUSION

In this work a novel multi-objective optimization strategy is presented for ORCs. Taking into account expander constraints a pre-selection strategy of working fluids is proposed. Furthermore, a classification scheme into four main waste heat recovery groups is introduced. From these, two representative cases are detailed.

For Case 3 (medium temperature heat carrier at 240 °C) the optimal working fluids are ammonia, propyne, R142b, isobutene, trans-2butene, R1233ZDE, R123, isopentane and n-Pentane, while for Case 4 (high temperature heat carrier at 305 °C) the optimal working fluids are ammonia, sulferdioxide, R21, R11, R141b and cyclopentane. A clear Pareto front is found in function of the objective criteria second law efficiency and volume coefficient. For increasing volume coefficient an increase in second law efficiency is observed.

In future work, all four cases will be examined. As such the majority of possible waste heat recovery applications are covered. Also turbine technology will be added. Subsequently, the optimal working fluids and cycles resulting from the pre-selection strategy will be investigated based on thermo-economic criteria. A framework for this was already proposed by the authors in previous works [32, 36].

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