Design optimization of small-scale ORC cycles for fluctuating heat source

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Abstract

Organic Rankine cycles (ORC) are efficient technologies for waste heat recovery (WHR) at low to mid temperatures. For the design of ORC power cycles, several thermodynamic parameters should be considered. A challenge related to small scale (<50 kW) ORC cycles is to define the optimal process given frequent variability in a heat source. Many relevant applications require robust ORC systems to perform under varying heat source loads. This is an area where the body of knowledge must be further developed.

In this work, the design of small-scale ORC cycles with varying heat source conditions is addressed by means of system modelling, simulation, and optimization. A framework is presented that consists of multi-scale optimization for the design of small-scale ORC systems considering seasonal and hourly heat source variations. The framework is developed as a flexible tool allowing to include fit-for-purpose models of key elements of the cycle, such as expander and heat exchanger, to suitably simulate off-design performance.

The optimization framework has been tested on a case study representing a woodchips-fired micro-cogeneration unit via ORC. The case study is representative of an existing unit operating at the Czech Technical University (CTU) in Prague. The results indicate that the tool delivers an ORC design that has a 5 % larger accumulated power production with the hourly variation of the heat source during one year than the original ORC solely optimized at the design heat source condition. The optimal ORC system also shows a 33 % smaller nominal capacity and size of heat exchangers than the ORC at the reference design, indicating a potential reduction in the capital cost.

1. Introduction

For low temperature heat sources, organic Rankine cycles demonstrated to be an advantageous technology (Macchi and Astolfi, 2016). When considering ORC design, several thermodynamic parameters should be considered, as well as a several organic working fluids and process configurations. Optimization could be performed both on thermodynamic and techno-economic parameters (Colonna *et al.*, 2015). A main challenge related to small scale (<50kW) ORC cycles is to define the optimal process design given a specific application when there is frequent variability in heat source. Many relevant applications will require robust ORC systems to perform under varying heat source (Petrollese and Cocco, 2019). This is an area where the body of knowledge is less developed.

This paper presents a tool under development for the design optimization of ORC systems accounting for off-design operations. The primary focus will be on small scale distributed energy systems. The objective of this work is to describe the methodology at the basis of the tool and present the first implementation, highlighting the expected benefits by analyzing a case study. In the future, the tool will be further developed along different lines. The off-design models will be refined to include more advanced approaches, potentially validated on experimental data. The framework will also include multi-scale optimization opportunities, where the cycle components design will be optimized together with the process. For instance, the aerodynamic design of the expander will be embedded into the optimization framework. In view of future developments, the tool will strive to enable a high degree of flexibility. The goal is to ensure the possibility to accommodate modules and layers in a simple manner, therefore allowing analyses at the requested level of complexity.

The paper is structured as follows. Section 2 describes the methodology used for the study, including the modelling work and optimization framework. The section concludes with an overview of the case study and boundary conditions for the analysis. Section 3 presents the results obtained from the given case study. Section 4 outlines the main findings.

2. Methodology

2.1. Tool for optimal design of ORC systems for small scale distributed energy systems

This study presents the first version of a tool for multi-scale design optimization of ORC systems. The tool is developed as a flexible platform for simulation and optimization. In this first version, the tool includes a model for simulation of ORCs as well as off-design models for predicting part load performances of key elements of the cycle. An optimization framework integrates the various models allowing to identify a design that results in the optimal performance considering all expected operating conditions. Future versions of the tool will allow for incorporating more advanced off-design models, better control structures and modules for multi-scale optimization. However, those aspects are not part of this study.

The approach used for design optimization is depicted in Figure 1. Given an objective function $\bar{z}$, a design is defined in terms of a set of selected independent variables (see Table 1). The simulation at such design point provides the process thermodynamic values of the cycle at design – in terms of mass flow rates, temperature and pressure levels and also sizes of the heat exchangers. That information is used for the off-design simulations that will be carried out for each operating conditions deemed relevant to describe the operation of the ORC. Each simulation returns the value of the objective function at the specific conditions. The overall objective function is the weighed sum of the specific values obtained from the off-design simulations. For this study, the main objective function was selected to be the accumulated produced electric power over the time horizon of operation. Such objective function is more suitable for systems where power is the main output, while our case study will be based on a combined heat and power (CHP) unit. However, the focus of this first implementation is to prove the methodology. More detailed considerations on relevant objective functions will be made in future studies.

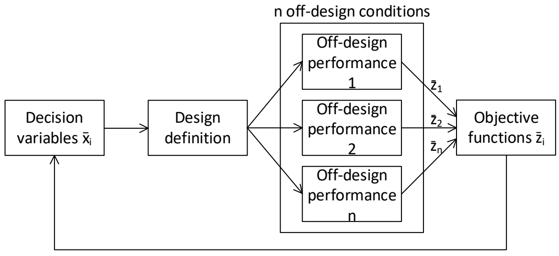

Figure 1. Flowchart of the optimization framework

2.1.1. Rankine Lab code

The Rankine Lab tool[1] (Agromayor and Nord, 2017) has been used as model basis for design optimization. It is an open-source tool in MATLAB that can be used to analyze and optimize Rankine cycles. It utilizes a gradient-based optimizer (SQP), and several cycle configurations are possible to analyze. In this design optimization tool, CoolProp, which is also open-source, is applied for thermodynamic property calculations during the ORC simulation. Various working fluids can be chosen within CoolProp and several parameters can be tuned to perform a simulation. For working fluids that are not supported by CoolProp, RefProp can be linked to the Rankine Lab tool via the CoolProp interface.

The ORC considered is shown in Figure 2 (note that the recuperator is originally included in the Rankine Lab tool, but it is not activated for this analysis). The working fluid selected is MM (hexamethyldisiloxane). For the design optimization, 5 decision variables and 8 inequality constraints are defined as shown in Table 1 and Table 2. The upper and lower bounds of the variables are also reported in Table 1.

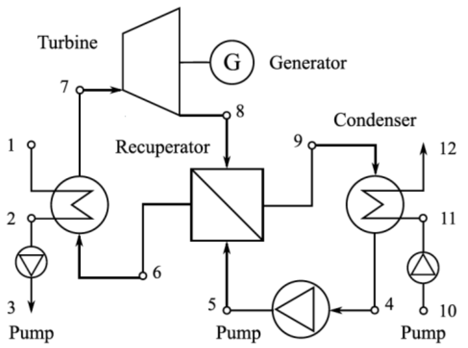

Figure 2. Process flowsheet of the reference ORC.

[1] For the open code and technical documentation, refer to https://github.com/RoberAgro/RankineLab

Table 1. Optimization variables and related bounds.

Independent variables	Lower bound	Upper bound
T_3	$T_{flue,min}$	$T_{flue,supply}$
p_4	$p_{cycle,min}$	$p_{cycle,max}$
h_4	$h_{sat}(T_0)$	$h(T_1, p \to 0)$.
p_7	p_{triple}	$p_{cycle,max}$
h_7	$h_{sat}(T_0)$	$h(1000\ K, p \to 0)$

Table 2. Nonlinear optimization constraints.

Independent variables	Value
ΔT within the evaporator*	$\Delta T_{evap} \geq \Delta T_{evap,min}$
ΔT within the condenser*	$\Delta T_{cond} \geq \Delta T_{cond,min}$
Subcooling at pump inlet	$\Delta T_{sub,min} \leq T_{sat}(p_4) - T_4 \leq \Delta T_{sub,max}$
Superheating at expander inlet	$\Delta T_{sup,min} \leq T_7 - T_{sat}(p_7) \leq \Delta T_{sup,max}$
Vapor quality within expander	$q_{exp} \geq q_{exp,min}$
Expander capacity	$W_{exp} \leq W_{exp,max}$
Expander inlet temperature	$T_7 \leq T_{exp,in,max}$
Expander inlet pressure	$p7 \leq p_{exp,in,max}$
Expander outlet pressure	$T_8 \geq T_{exp,out,min}$

*This constraint is evaluated at each discretization step of the heat exchanger

2.1.2. Code for off-design:

The off-design model was based on the work in (Riboldi and Nord, 2018) and adapted to an ORC. The inputs to the off-design model are the design parameters (see those parameters with subscript *d* in the following equations) obtained by the solution of Rankine Lab at the design tested.
The heat recovery unit (HRU) is modelled through the relation from Incropera et al. (Incropera *et al.*, 2007), where the off-design heat transfer coefficient is calculated as:

$$UA = UA_d \left(\frac{\dot{m}}{\dot{m}_d}\right)^{\gamma} \quad (1)$$

where U is the overall heat transfer coefficient, A is the heat transfer area, ṁ is a mass flow rate and γ is the exponent of the Reynolds number in the heat transfer correlation. γ was set equal to 0.6. Assuming a shell and tube heat exchanger configuration, it is assumed that the heat transfer inside the tubes is the dominating factor (Orlandini *et al.*, 2016). The condenser is more simply modelled as a fixed pressure component. The performance of the expander at off-design conditions is model based on the two following equations:

$$C_S = \frac{\dot{m}\sqrt{T_{in}}}{\sqrt{p_{in}^2 - p_{out}^2}} \quad (2)$$

$$\frac{\eta_T}{\eta_{T,d}} = 2\sqrt{\frac{\Delta h_{T,is,d}}{\Delta h_{T,is}}} - \frac{\Delta h_{T,is,d}}{\Delta h_{T,is}} \quad (3)$$

where C_S is the constant flow coefficient (a constant and determined at design conditions), ṁ is the mass flow rate, T_{in} is the turbine inlet temperature, p_{in} is the turbine inlet pressure and p_{out} is the turbine outlet pressure, η_T is the isentropic efficiency of the turbine at off-design and $\Delta h_{T,is}$ is the isentropic enthalpy difference due to the expansion in the turbine. The former is the Stodola's cone law and determines the mass flow rate of the cycle as a function of inlet pressure, outlet pressure and the fluid density at the turbine inlet. The latter is the relation proposed by Schobeiri (Schobeiri, 2005) to predict the isentropic efficiency at part-load. The efficiency of the generator is calculated as follows (Haglind and Elmegaard, 2009):

$$\eta_{gen} = \frac{load \cdot \eta_{gen,d}}{load \cdot \eta_{gen,d} + (1 - \eta_{gen,d})[(1 - F_{CU}) + F_{CU} load^2]} \quad (4)$$

The efficiency of the pumps at off-design is defined as a function of the volumetric flow rate, according to the relation developed by Veres (Veres, 1994):

$$\frac{\eta_{pump}}{\eta_{pump,d}} = -0.029265\left(\frac{\dot{V}}{\dot{V}_d}\right)^3 - 0.14086\left(\frac{\dot{V}}{\dot{V}_d}\right)^3 + 0.3096\left(\frac{\dot{V}}{\dot{V}_d}\right)^2 + 0.86387 \quad (5)$$

where η_{pump} is the isentropic efficiency of the pump and $\dot{V}$ is the volumetric flow rate.

The pressure drops (Δp) are modelled with a quadratic dependence from the mass flow rate (Lecompte *et al.*, 2013):

$$\Delta p = \Delta p_d \left(\frac{\dot{m}}{\dot{m}_d}\right)^2 \quad (6)$$

where Δp is the pressure drop and $\dot{m}$ is the mass flow rate.

To cope with the different input conditions to the ORC, a simple control scheme was implemented. The pressure was left varying according to a sliding pressure control mode. This is a common control scheme in ORC applications (Imran *et al.*, 2020). The maximum temperature entering the expander is also kept within a maximum threshold while the expander outlet pressure was maintained at the design value.

2.2 Case Study description

The reference case study is biomass-fired micro-cogeneration of heat and power via ORC. The case study is representative of an existing unit operating at the Czech Technical University (CTU) in Prague for supplying heat (design 120 kW_{th}) and electricity (design 6.2 kW_{el}) to the university research center. The working fluid is MM (hexamethyldisiloxane), while the expander is a rotary vane expander. The unit is woodchips-fired.

2.3 Scenario and boundary conditions

Actual operating data has been provided and used as the basis for the analysis. In this study, only the variations of the flue gas flow rate were considered, and the cycle is simulated to provide the maximum power output. This is a simplification as the actual ORC is designed to provide both heat and backup power. However, the goal of this study was to demonstrate the methodology rather than to simulate a real system.

Figure 3 shows the variation of flue gas flow rate (i.e., the heat source for the ORC) over one year. Significant fluctuations can be noticed, resulting in the ORC operating frequently far from its design. Such design was selected as that at 0.078 kg/s of flue gas flow rate and is indicated as 100 % flow rate. Figure 4 shows the distribution of flue gas flow rates throughout one year.

The low load operation will result in the ORC system at the design condition being oversized most of the time. Thus, in this work, the ORC system is optimized to find the optimal system capacity and operating conditions that give the largest power production throughout the year. The capacity of the ORC system is varied by changing the design flue gas flow rate. During this optimization, the excess amount of the flue gas over 115 % of its design value is assumed to be not utilized in the ORC system.

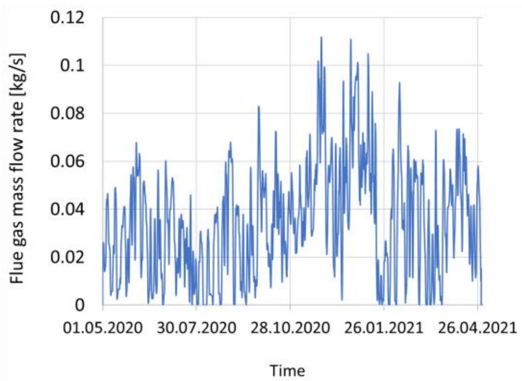

Figure 3. Hourly variation of the heat source (flue gas mass flow rate) over a year period at the Czech Technical University (CTU) campus.

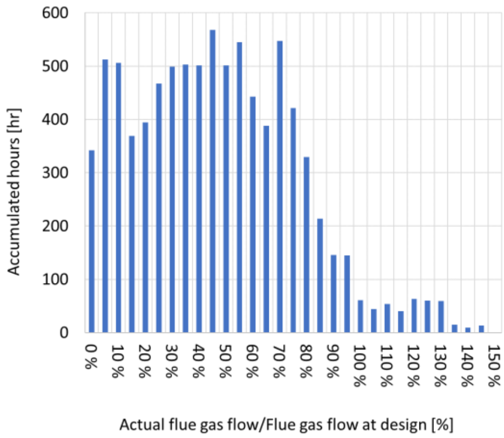

Figure 4. Distribution of the flue gas flow rate at the Czech Technical University (CTU) campus (accumulated hours for one year of operation).

The constraint values in Table 2 can be varied depending on the assumptions made to analyze an ORC system. For example, the practical conditions include limitations to the cycle parameters and operating conditions that are given by the utilization of state-of-the-art equipment.

In this work, however, the ORC is constrained to reflect the actual rig conditions considering the physical limitations of process equipment and working fluids. One example is the cycle is restricted to be operated at sub-critical conditions in order to prevent any mechanical issues. To avoid the thermal degradation of MM, the maximum cycle temperature (expander inlet temperature) is also limited to 190 °C considering an extra temperature margin.

Table 3. Design basis and constraints applied in the MM based ORC.

Design parameters and constraints	**Rig conditions**
$T_{flue,supply}$	1442 °C
$T_{flue,min}$	132 °C
$p_{cycle,max}$	10 bara
$p_{cycle,min}$	0.01 bara
$\Delta T_{evap,min}$	50 °C
$\Delta T_{cond,min}$	2 °C
$\Delta T_{sub,min}$	9.6 °C
$\Delta T_{sup,min}$	10 °C
$q_{exp,min}$	1
$W_{exp,max}$	15.5 kW
$T_{exp,in,max}$	190 °C
$p_{exp,in,max}$	8 bara
$p_{exp,out,min}$	0.2 bara

It is worth noting that the heated cooling water from the condenser is utilized as hot water at the CTU campus. Thus, the cooling water supply (T_{10}) and return temperature (T_{12}) are set to 58 °C and 80 °C to meet the hot water specification. Other design conditions and constraint values are listed in Table 3.

3. Results

Figure 5 presents the performance of two different designs of the MM based ORC system. One is the ORC system optimized for the design flue gas flow rate (referred to as design ORC). Based on the ORC design, the off-design performance is estimated to calculate the accumulated power production per year. The other is the ORC system optimized to maximize the accumulated power production throughout the year while varying the capacity (referred to as optimal ORC). The temperature-enthalpy diagram of the optimal cycle is also presented in Figure 6.

The results indicate that the optimal ORC system has a 5 % larger accumulated power production throughout the year compared to the ORC system at the design. As shown in Figure 5, the design ORC achieves a lower energy efficiency below 80 % load compared to the optimal ORC. Considering the frequent low load operation, such lower efficiency results in smaller accumulated power production as well. Although the ORC at the design outperforms at a larger flue gas flow rate of over 80 %, the fraction of the occurrence is not significant (see Figure 4), thus having a marginal impact on the accumulated power production.

Table 4 introduces the operating conditions of the two different designs. It is worth noting that the optimal ORC has a smaller capacity than the ORC at the design. Based on the flow rate of the working fluid and the expander power output, the capacity of the optimal system is estimated to be around 68 % of the ORC at the design. The lower design capacity of the ORC system allows the cycle to achieve a higher energy efficiency at a lower load where the system is operated most of the time span. The reduced capacity of the ORC will also decrease the capital cost and improve the economic feasibility of the small-scale ORC system, which is one of potential issues regarding the deployment (Tocci *et al.*, 2017).

Although the two different design solutions have a relatively large performance difference, some of the operating conditions at design point are close to identical as indicated in Table 4. The main changes are observed on the size of the heat exchangers (evaporator and condenser). One of the possible reasons will be the limited feasible region that is caused by the severe constraints considering the rig setup, the low limit of the maximum cycle pressure and temperature due to the working fluid characteristics, and the warm supply and return temperatures of the cooling water.

Relaxation of such constraints and design basis will allow this tool to have a wider search space to identify improved operating conditions considering the off-design performance of the ORC system throughout a year. Applying the tool to other case studies and heat sources might also result into a wider search space.

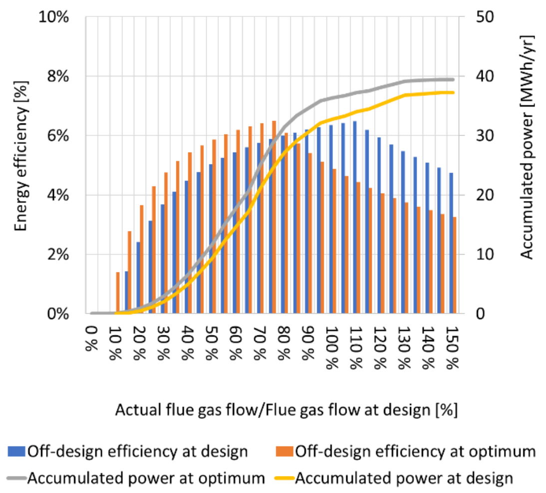

Figure 5. Energy efficiency and the accumulated power output of the MM based ORC system with a hourly variation of the flue gas flow rate for a full year period (Energy efficiency = $W_{net}/\Delta Q_{flue,max}$).

Table 4. Key parameters of the original design and the optimum design accounting for off-design performance (the operating conditions that are similar between the two designs are presented in italic).

Parameter	Unit	Design	Optimum
$W_{accumulated}$	MWh/yr	37.26	39.44
m_{MM}	kg/s	0.37	0.25
m_{CW}	kg/s	1.56	1.04
W_{net}	kW	11.05	7.40
W_{exp}	kW	11.54	7.73
T_{exp_in}	°C	*190*	*190*
P_{exp_in}	bara	*6.42*	*6.42*
T_{exp_out}	°C	*164.51*	*164.51*
P_{exp_out}	bara	*0.42*	*0.42*
UA_{evap}	kW/C	0.39	0.26
$\Delta T_{min,evap}$	°C	69.16	69.16
UA_{cond}	kW/C	17.59	11.79
$\Delta T_{min,cond}$	°C	2.00	2.00

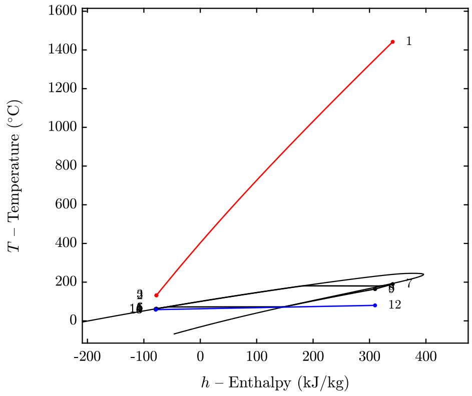

Figure 6. Temperature-enthalpy diagram of the optimal ORC.

4. Summary and Discussions

This work introduces an open-source-based ORC design and optimization tool that can reflect off-design performance. The optimization tool is tested by a case study on an ORC unit operating at the Czech Technical University (CTU) in Prague for supplying heat and electricity to the university research center. Actual heat source variations are considered for the analysis. The results indicate the optimal size of the system is 33 % smaller than the original ORC design, which is optimized without part-load performance estimation. The accumulated power production is also increased with the optimal ORC design by 5 %. Further improvements in the annual accumulated power production with the optimization framework could potentially be achieved with relaxed constraints and design basis, representing practical conditions.

As a next step, the framework will be further tested with different working fluids, heat sources and heat sinks. The tool will be updated with improved process unit and off-design expander models based on experimental data, allowing the framework to be capable of robust and practical multi-scale optimization.

Acknowledgment

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