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Recuperated power cycle analysis model: Investigation and optimisation of low-to-moderate resource temperature Organic Rankine Cycles

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ABSTRACT

A numerical model for recuperated power cycles for renewable power applications is described in the present paper. The original code was written in Python and results for a wide range of working fluids and operating point conditions are presented. Here, the model is applied to subcritical and transcritical Rankine cycles. It comprises a brute-force search algorithm that covers a wide parametric study combining working fluid, resource and cooling temperatures as well as high-side pressures in order to ascertain the best working fluid for a given resource temperature and operating point. The present study determined the fluids that maximise the specific energy production from a hot stream for a range of low-to-medium temperature (100–250 °C) resources. This study shows that for the following resource temperatures: 100 °C, 120 °C, 150 °C, 180 °C and 210 °C, R125, R143a, RC318, R236ea and R152a were found to maximise specific energy production, respectively. In general, the inclusion of a recuperator within the power cycle results in greater specific energy production for a given operating temperature. However, it was found that for all fluids there was a threshold pressure above which the inclusion of a recuperator did not enhance system performance. This may have design and economic ramifications when designing next-generation transcritical and supercritical power cycles.

1. Introduction

The use of steam-based Rankine cycle power systems has been the main-stay of electricity production [1] for decades. However, steam-based systems are unable to achieve high efficiencies when converting low-grade heat (such as that obtained from geothermal, waste-heat, small-scale solar) into electricity [2]. *ORCs* (Organic Rankine Cycles), which make use of an organic working fluid, such as *HFCs* (hydrocarbons and hydrofluorocarbons), are an attractive solution to harness the energy of lower temperature resources (100 °C–250 °C). The use of such working fluids may be considered advantageous as they are able to provide higher cycle efficiencies than steam-based cycles when considering low temperatures [1].

The thermodynamic analysis, including comparison of working fluid performance and optimisation of *ORCs,* has received increased attention in recent years and has been relatively well-covered in the literature [1–10]. The work presented in Ref. [9] showed that turbine inlet pressure and temperature together with pinch and approach temperature difference play an important role in determining the net power output as well as the surface area of the heat exchangers employed in the cycle. A comprehensive review of works regarding *ORC* optimisation may also be found in Ref. [10]. In this latter work, resource (geothermal) temperatures between 120 °C and 180 °C were investigated. This corresponds to a sub-set of the resource temperature range investigated in the present paper (100 °C–250 °C), for a large number of working fluids.

These studies are complemented by techno-economic analyses presented by other authors (e.g., [11,12]) regarding the utilisation of *ORC* systems. A review of works involving the techno-economic optimisation of *ORCs* may be found in Ref. [13].

Nevertheless, results for various working fluids and for a wide range of temperatures and pressures while considering the inclusion of a recuperator in the system has been less well reported. The

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inclusion of a recuperator can adjust the overall rank of performance of a fluid relative to another. Here we present a full set of algorithms that can be used to determine the operating conditions to make best-use of any temperature resource as well as the results for a range of fluids at operating temperatures between 100 °C and 250 °C and pressures up to 6.1 MPa. These results serve as a starting point for further system design investigations, such as turbine design.

2. Model definition

This model and was originally reported as part of the body of work presented in Ref. [14]. The details relevant to this study are discussed below.

A simplified flow diagram of the cycle, including the key state points, components, features, and simplifications assumed, is presented in Fig. 1. Overall, the *rf* (resource fluid) transfers heat to the *wf* (working fluid) in an external *HSHE* (hot-side heat exchanger) whilst a *cf* (cooling fluid) condenses the working fluid in another external *CSHE* (cold-side heat exchanger). A recuperator is placed between the outlets of both the turbine and the pump to extract energy from the turbine exit stream.

The subcritical and transcritical cycle configurations covered by the model are presented in Fig. 2. This figure depicts both temperature vs. entropy (*T–s*, left of the figure) and the temperature vs. heat exchanged (*T-Heat Exchanged*, right of the figure) diagrams for subcritical and transcritical cycles. The numbered state points from Fig. 2 match with those denoted in the flow diagram seen in Fig. 1. The calculation of these points is further described in the following sections.

2.1. Calculation process

The model runs by performing parametric sweeps of the following variables: fluid type, high-side operating pressure, $P_{TURBINE\ IN\ (wf)}$ and hot-side operating temperature, $T_{HSHE\ IN\ (rf)}$. The working fluid mass flow rate, isentropic efficiencies of the pump and turbine and conversion efficiency of the generator were held constant throughout the simulations. The applicable parametric ranges and fixed-values used for key variables are specified in Table 1 below.

In the present model, the pressure and heat losses occurring between each stage are considered negligible [15]. The same simplification is assumed for the heat exchanger stages in the cycle: hot-side (*HSHE*) and cold-side (*CSHE*) heat exchangers and recuperator. For these latter stages, a single passage counter-flow heat exchanger configuration is assumed and a conservative overall maximum heat exchanger approach temperature, ΔT_{HE} of 10 °C was considered.

For the present study, a range of resource fluid temperatures between 100 °C and 250 °C were tested using a step size of 10 °C. Similarly, a range of high-side operating pressures between a minimum and maximum of 300 kPa and 6100 kPa were considered. For the fluids considered, pressures above ~6000 kPa were found to yield only minimal performance gains, thus setting the constraint for the maximum pressure considered.

The model starts by calculating the thermodynamic state points (Section 2.1.1). These results are analysed against predefined feasibility constraints (further discussed in section 2.1.2) that must be simultaneously satisfied. This defines the thermodynamic state points and allows the calculation of the mass flow rate and temperature of the resource fluid at the outlet of the hot-side heat exchanger. This ultimately leads to the calculation of the thermal and electrical efficiencies of the thermodynamic cycle.

Fig. 3 illustrates the calculation process for the model. This figure is complemented by Fig. 4 and Fig. 5, which provide detailed algorithms of the recuperator and heat exchanger models, respectively.

2.1.1. Calculation of state points

This section covers the calculation of the working fluid thermodynamic state points. These calculations follow the state points denoted in the flow and temperature–entropy (*T–s*) diagrams in Figs. 1 and 2. The determination of the thermodynamic and transport properties of the working fluid at any point is based on providing two known values to a thermodynamic and transport property database: *NIST REFPROP* [16] and obtaining the necessary properties via a custom Python-based library. This is illustrated in Table 2 where the necessary properties are calculated as a (*NIST REFPROP*) function of other two known variables: unknown variables/properties = *f* (*variable 1, variable 2*). It should be noted however, that any pressure and heat losses in between as well as within the components were considered negligible [15].

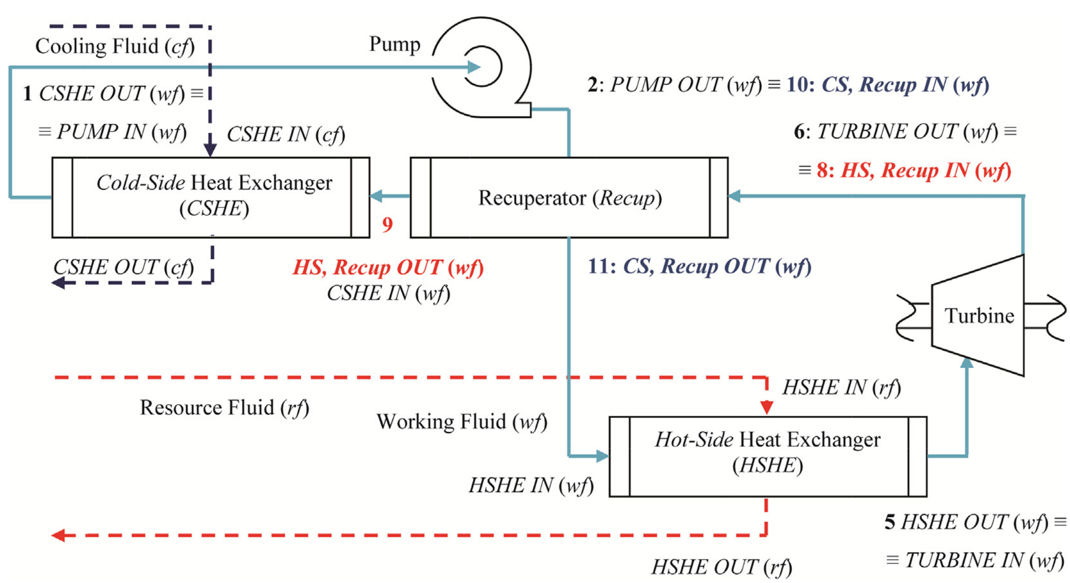

Fig. 1. Flow diagram, state points and nomenclature for the cycle analysis program [14].