Energy 58 (2013) 97–106

Contents lists available at SciVerse ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

An Organic Rankine Cycle off-design model for the search of the optimal control strategy

Giovanni Manente [a,*], Andrea Toffolo [b], Andrea Lazzaretto [a], Marco Paci [c]

[a] University of Padova, Department of Industrial Engineering, via Venezia 1, 35131 Padova, Italy
[b] Luleå University of Technology, Department of Engineering Sciences and Mathematics, Energy Science Division, 97187 Luleå, Sweden
[c] ENEL Engineering and Innovation, via Andrea Pisano 120, 56126 Pisa, Italy

ARTICLE INFO

Article history:
Received 1 August 2012
Received in revised form
6 December 2012
Accepted 14 December 2012
Available online 26 January 2013

Keywords:
Organic Rankine Cycle
Off-design
Dynamic model
Control strategy

ABSTRACT

Power generation from low enthalpy geothermal resources using Organic Rankine Cycle systems is markedly influenced by the temperature level of the heat source and heat sink. During plant operation the actual temperature of the geofluid may be different from the value assumed in the design phase. In addition, the seasonal and daily variations of the ambient temperature greatly affect the power output especially when a dry condensation system is used. This paper presents a detailed off-design model of an Organic Rankine Cycle that includes the performance curves of the main plant components. Two capacitive components in the model have the key function of damping the temporary disequilibrium of mass and energy inside the system. Isobutane and R134a are considered as working fluids, mainly operating in subcritical and supercritical cycles, respectively. The off-design model is used to find the optimal operating parameters that maximize the electricity production in response to changes of the ambient temperatures between 0 and 30 °C and geofluid temperatures between 130 and 180 °C. This optimal operation strategy can be conveniently applied both to already existing plants and to the choice of new design plant configurations.

1. Introduction

Organic Rankine Cycles are a viable option for high efficiency/low cost exploitation of low temperature geothermal flows for electricity production. Some studies in the literature searched for parameters which are most suitable to correctly analyze the overall system performance. Among them, the works [1–3] emphasized the importance of using a heat recovery efficiency in addition to thermal efficiency to correctly quantify the system capability of using the available energy/exergy content of the geothermal source. Other studies [4–6] focused mainly on the best choice of the cycle operating fluids both in terms of efficiency and costs. The common approach in these studies consists in building system models and in comparing the obtained results for a given inlet temperature of the sensible heat source. Only few works, among these [7,8], present instead a comparison between calculated and experimental values.

A different approach was used by the authors in Ref. [9] to search for the optimal and sub-optimal design conditions of Organic Rankine Cycle systems, which consists in keeping the design of the "heat transfer section" within the system (which appears as a black-box including hot and cold thermal streams, see Fig. 1) independent of the design of the rest of the system itself (which is called basic plant configuration). This is done using the so called HEATSEP method [10] which considers the temperatures at the boundaries of the two parts of the systems as decision variables in the design optimization procedure. So, the design of the heat exchanger network can be performed only after these optimal temperature values are calculated.

In Ref. [11] the optimal thermodynamic solutions found in Ref. [9] for the working fluids isobutane and R134a were evaluated from the economic point of view. For various temperature values of the geothermal source in the range 130–180 °C, the heat exchanger network, previously left undefined inside the black box, was obtained (Fig. 2). It was composed by a preheater-vaporizer or a supercritical evaporator, an air cooled condenser and, only in some cases, a recuperator. The recuperator was "introduced" in the system structure by the optimization procedure when the thermal energy required to heat the operating fluid between pump outlet

* Corresponding author. Tel.: +39 (0)498277478; fax: +39 (0)498276785.
E-mail addresses: giovanni.manente@unipd.it (G. Manente), andrea.toffolo@ltu.se (A. Toffolo), andrea.lazzaretto@unipd.it (A. Lazzaretto), marco.paci@enel.it (M. Paci).

0360-5442/$ – see front matter © 2013 Elsevier Ltd. All rights reserved.
http://dx.doi.org/10.1016/j.energy.2012.12.035

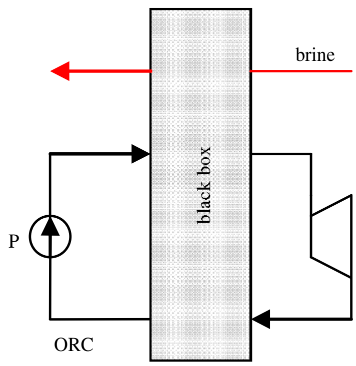

Fig. 1. Basic plant configuration of the single pressure level ORC according to the HEATSEP method [10].

and turbine inlet was greater than the thermal energy made available by the geothermal source.

Although ORC systems are rather easy to operate, particular care must be taken in controlling and monitoring the system during transient conditions when the load demand or the quality or flow of the low temperature heat source changes. These conditions can be profitably predicted by off-design models. In Ref. [7] one of the present authors built an off-design stationary model of an existing 30 MW ORC plant in Aspen® environment using real characteristic curves. The model was then used to calculate the values of the operating parameters which maximize the power output from the available geothermal resource, and to change the actual operating criterion accordingly. Dynamic off-design models of ORC plants were recently proposed in Refs. [12,13]. In Ref. [12] the authors focused on the correct representation of evaporator and condenser dynamic behavior. The model was validated against experimental data available from a pilot 100 kW ORC system using R245fa as working fluid. In Ref. [13] a dynamic model was built for a small scale Organic Rankine Cycle including a volumetric expander. The model was used to find the optimal control strategy to recover energy from a variable flow rate and temperature waste heat source using R245fa as working fluid. In particular, the authors developed dynamic models for the evaporator and the "receiver" (an accumulator) at the condenser exhaust whereas they used a steady state model for the expander.

In this paper a new off-design dynamic model of the ORC system is presented. Design conditions are assumed to be those obtained by the thermodynamic optimization of ORCs presented in Ref. [9]. Performance data and curves of the main components (e.g. feed pump, turbine, heat exchangers) were obtained starting from datasheets provided by the manufacturers of a recently built geothermal binary cycle power plant, and adjusted according to the

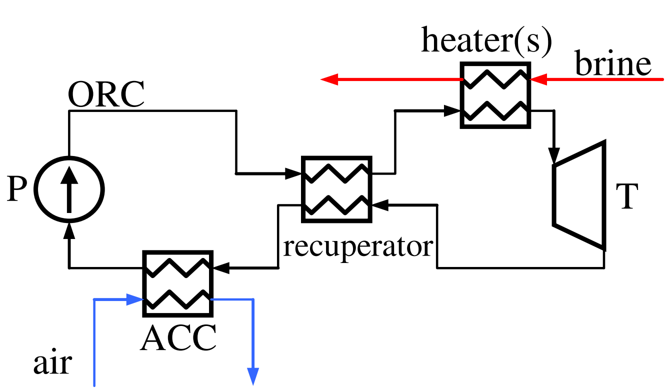

Fig. 2. Configuration of the single pressure level ORC which includes the heat exchanger network.

assumptions used here (e.g. brine flow rate, brine temperature etc.). So, the model is representative of a real plant behavior. When the ambient temperature and/or the geofluid temperature depart from the design values new operating parameters and thermodynamic cycles can be found by the model, which depend on the components performance curves and the plant control criterion. On this basis, an optimal control strategy is suggested to search for the values of the operating parameters which maximize the power generated at any predictable off-design conditions.

2. The off-design model

The off-design model built in the Simulink® environment to simulate the system is composed by the sequence of components shown in Fig. 3. Each of these components is described by mass and energy balances and one or more performance curves set in the design phase.

The mass rate balance, in general, is given by:

$$\frac{dm}{dt} = \sum_i \dot{m}_i - \sum_e \dot{m}_e \tag{1}$$

where dm/dt is the time rate of change of mass within the control volume and $\dot{m}_i$ and $\dot{m}_e$ are the instantaneous mass flow rates at the inlet and exit, respectively.

The energy rate balance (neglecting the kinetic and potential energy terms) is:

$$\frac{dU}{dt} = \dot{Q} - \dot{W} + \sum_i \dot{m}_i h_i - \sum_e \dot{m}_e h_e \tag{2}$$

where dU/dt is the time rate of change of the internal energy contained within the control volume, the terms $\dot{Q}$ and $\dot{W}$ account for the net rate of energy transfer by heat and work over the boundary of the control volume, $\dot{m}_i h_i$ and $\dot{m}_e h_e$ account for the enthalpy rates of the entering and exiting streams.

At steady state there is no net change in the total mass and total energy within the control volume and the mass and energy rate balance equations respectively reduce to:

$$\sum_i \dot{m}_i = \sum_e \dot{m}_e \tag{3}$$

$$0 = \dot{Q} - \dot{W} + \sum_i \dot{m}_i h_i - \sum_e \dot{m}_e h_e \tag{4}$$

In this study the transient behavior of the ORC system was modeled by adding two capacities in the plant layout that were analyzed using the unsteady form of the mass and energy rate balances. The remaining plant components were modeled using the steady state form of the mass and energy rate balances.

In addition to mass and energy balances the behavior of each component in the model is described by its performance curves which depend on rotational speeds (pump), blade settings (turbine) and flow conditions (heat exchangers).

The application of the dimensional analysis [14] to the pump states that the head coefficient ψ and the pump efficiency η_p are unique functions of flow coefficient ϕ (ignoring Reynolds number effects):

$$\psi, \eta_p = f(\phi) \tag{5}$$

where $\psi = gh/\omega^2 D^2$ and $\phi = Q/\omega D^3$.

From Eq. (5) the dimensional head – volumetric flow rate characteristics ($h = f(Q, N)$) are obtained.