Energy 88 (2015) 897–906

Contents lists available at ScienceDirect

Energy

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

Optimal Heat Source Temperature for thermodynamic optimization of sub-critical Organic Rankine Cycles

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ARTICLE INFO

Article history:
Received 13 December 2014
Received in revised form
11 June 2015
Accepted 12 July 2015
Available online 1 August 2015

Keywords:
Optimal Heat Source Temperature
Organic Rankine Cycle
Heat Transfer Fluid
Working fluid
Pinch point
Exergetic efficiency

ABSTRACT

Based on a sub-critical ORC (Organic Rankine Cycle) process, this study introduces the term OHST (Optimal Heat Source Temperature) with consideration of a suitable thermal match between heat source and working fluid. A theoretical formula is developed for predicting the OHST, which shows that OHST only depends on evaporation pressure and pinch point in the preheater and evaporator. A comparative study between the predicted OHSTs and those obtained from cycle simulations is performed, showing that the proposed formula is reliable, provided that HTF (Heat Transfer Fluid) is homogeneous and has good consistency in terms of heat capacity for different temperatures. To demonstrate the application of the proposed OHST-theory for thermodynamic optimization of ORC systems, a case study is presented based on a simple ORC coupled with thermal water at 140 °C. Consequently, using R227ea leads to the highest system efficiency of 10.38%, due to a better thermal match in the preheater and evaporator. In order to increase the exploitation of the thermal potential from the heat source, a dual-fluid-ORC is proposed, where R245fa and R227ea are considered for the high and low temperature ORC processes, respectively. Finally, this combination leads to the highest system efficiency of 11.07%.

1. Introduction

Throughout the last decades, due to increasing concerns over the unecological utilization of fossil fuels, power generation from low-to medium temperature heat sources has attracted growing interests. Heat at low-to medium temperature levels accounts for 50% or more of the total heat generation worldwide [1] and it exists in various forms, ranging from waste heat in industrial sector, concentrated solar thermal energy to geothermal energy. A number of technologies exist to convert these low-to medium heats into electricity. Among them, the ORC (Organic Rankine Cycle) has been widely used due to some advantages, such as simplicity and relatively high efficiency compared to a conventional Clausius–Rankine-Cycle.

The working mediums in an ORC application differ from one other in many aspects, such as thermophysical properties, environmental impacts, safety, and compatibility with component materials [2,3]. Therefore, an efficient way to optimize the ORC system is the selection of a working fluid which has preferable thermophysical properties and matches best with the available heat source [4–6]. However, for a typical sub-critical ORC process, a suitable selection of working fluids cannot avoid isothermal evaporation which causes a large fraction of irreversibilities in the heat transfer process [7,8]. Several methods are proposed, which aim to reduce such irreversibilities, e.g. using triangular ORC [9] or a supercritical fluid [10–13] to bypass the isothermal evaporation, or using a zeotropic mixture [14–17] to obtain a non-isothermal evaporation. Although these methods are effective under certain working conditions, complexity of the ORC system increases, leading to higher manufacturing costs.

To reduce the irreversibilities caused by the isothermal evaporation, an approach is proposed in this paper, where a high evaporation temperature is assumed for a sub-critical ORC in order to reduce the heat amount that is transferred during evaporation. The drawback of this approach is, however, obvious: the pinch point, located in most cases at evaporator inlet, can greatly limit the mass flow rate of working fluid, which constrains further heat source

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http://dx.doi.org/10.1016/j.energy.2015.07.040
0360-5442/

Nomenclature

c_p	specific heat capacity, kg/kg K
e	specific exergy, kJ/kg
h	specific enthalpy, kJ/kg
m	mass flow rate, kg/s
p	power, kW or pressure, bar
$\dot{Q}$	heat flow, kW or %
s	specific entropy, kJ/kg K
t	temperature, °C
ALT	Atmospheric Lifetime
GWP	Global Warming Potential
ODP	Ozone Depletion Potential
OEP	Optimal Evaporation Pressure, bar
OHST	Optimal Heat Source Temperature, °C
HTF	Heat Transfer Fluid
ORC	Organic Rankine Cycle
RSD	Relative Standard Deviation, %

Greek letters

η	efficiency, %
ρ	density, kg/m^3

Subscripts

0	ambient state
1,2…i	working states
cond	condensation
evp	evaporation
ex	exergy
HE	heat exchanger
hs	heat source
in	inlet
is	isentropic
out	outlet
p	pump
pre	preheater
sys	system
t	turbine
th	thermal
wf	working fluid

utilization. To overcome this drawback, it is a prerequisite for the proposed approach to find a suitable working fluid, for which the pinch point is not located at the evaporator inlet but at the preheater inlet. As a result, whole heat transfer process can be optimized by increasing evaporation temperature, leading to a better thermal match and hence a higher system efficiency [18].

Although the proposed approach is promising, few authors have paid attention to it because of difficulties in predicting pinch point position. By means of heuristic approaches, it was found that pinch point position depends not only on the working fluid but also on the heat source temperature for which the ORC is designed. The results of some parametric studies [14,18,19] showed that for a pure working fluid at a constant sub-critical pressure, a certain heat source temperature can be identified, for which the pinch point switches its position from the evaporator inlet to the preheater inlet. Moreover, such a heat source temperature indicates a maximization of exergetic efficiency in the heat transfer process. Therefore, this temperature is defined as an OHST (Optimal Heat Source Temperature) for a specific fluid. However, no prior works are available so far that accurately predict the above-described OHST. Therefore, the main task of this study is to develop a theoretical method to predict the OHST and to utilize it for further design and optimization of an ORC process.

This paper is structured as follows: a description regarding a simple ORC in Section 2, which is followed by Section 3, where efficiencies used for performance evaluation are described. In Section 4, a conventional method of determining the OHST by cycle simulations is reviewed, along with a theoretical model for the OHST calculation. Subsequently, a comparative study is performed by comparing the predicted OHSTs against the obtained ones from cycle simulations. In Section 5, two ORC systems are presented and discussed in order to demonstrate the proposed OHST model for optimization of more complex ORC applications. Finally, conclusions are given in Section 6, along with suggestions for future work.

2. Description of ORC process

Fig. 1 shows the simplest configuration and corresponding T–s-diagram of a sub-critical ORC. Fundamental components include: pump, preheater, evaporator, turbine and condenser. The working fluid at a saturated liquid state (state 1) is pressurized in a feed-pump to a designed pressure (state 2). Then it is fed to a preheater where the sub-cooled working fluid is heated until becoming a saturated liquid (state 3). Afterwards, the working fluid is evaporated in an evaporator to a slightly superheated vapour (state 4) which is fed to an expansion machine (turbine). The vapour expands in the turbine isentropically, which rotates a shaft and generates electricity in the generator. Finally, a condenser rejects heat from the super-heated vapour (state 5) until it becomes a saturated-liquid again.

3. Thermodynamic analysis

This section describes efficiencies in terms of the first and second laws of thermodynamics, which are used for further performance evaluations. Thermodynamic analyses are based on the following assumptions:

- No pressure drop and heat loss inside and between components,
- Slight super-heating at the evaporator outlet is neglected.

3.1. Global system efficiency

Global system efficiency is used as the main criterion for further performance evaluations due to the advantage that it combines both thermal efficiency (η_{th}) and heat transfer efficiency (η_{HE}) of an ORC system [10]. The global system efficiency is expressed by:

$$\eta_{sys} = \eta_{th} \cdot \eta_{HE} \tag{1}$$

where η_{th} and η_{HE} are equal to:

$$\eta_{th} = \frac{P_{net}}{\dot{Q}_{HE}} = \frac{\dot{m}_{wf} \cdot [(h_4 - h_5) - (h_2 - h_1)]}{\dot{Q}_{HE}} \tag{2}$$

$$\eta_{HE} = \frac{\dot{Q}_{HE}}{\dot{m}_{hs} \cdot (h_6 - h_0)} \tag{3}$$