Energy 87 (2015) 60–76

Contents lists available at ScienceDirect

Energy

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

Methodical thermodynamic analysis and regression models of organic Rankine cycle architectures for waste heat recovery

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ARTICLE INFO

Article history:
Received 28 January 2015
Received in revised form
25 March 2015
Accepted 21 April 2015
Available online 27 May 2015

Keywords:
organic Rankine cycle
Cycle architectures
Working fluids
Optimization
Exergy
Second law efficiency

ABSTRACT

The ORC (organic Rankine cycle) is an established technology for converting low temperature heat to electricity. Knowing that most of the commercially available ORCs are of the subcritical type, there is potential for improvement by implementing new cycle architectures. The cycles under consideration are: the SCORC (subcritical ORC), the TCORC (transcritical ORC) and the PEORC (partial evaporation ORC). Care is taken to develop an optimization strategy considering various boundary conditions. The analysis and comparison is based on an exergy approach. Initially 67 possible working fluids are investigated. In successive stages design constraints are added. First, only environmentally friendly working fluids are retained. Next, the turbine outlet is constrained to a superheated state. Finally, the heat carrier exit temperature is restricted and addition of a recuperator is considered. Regression models with low computational cost are provided to quickly evaluate each design implications. The results indicate that the PEORC clearly outperforms the TCORC by up to 25.6% in second law efficiency, while the TCORC outperforms the SCORC by up to 10.8%. For high waste heat carrier inlet temperatures the performance gain becomes small. Additionally, a high performing environmentally friendly working fluid for the TCORC is missing at low heat carrier temperatures (100 °C).

1. Introduction

The ORC (organic Rankine cycle) is considered a mature technology for low to medium temperature heat to electricity conversion. Besides the capability to operate at low temperatures, the highlights of the ORC include: low maintenance cost, autonomous operation and favourable operating pressures [1]. Commercial ORC installations typically operate with pure working fluids under subcritical conditions [2]. Therefore the heat carrier and working fluid temperature profile have a suboptimal match due to the isothermal evaporation [3]. Advanced cycle architectures have the potential to improve this match by effectively reducing exergy losses associated to finite temperature heat transfer [4]. Performance gains over the subcritical ORC are reported for, amongst others, multi-pressure cycles [5–8], trilateral [3,9–12] and transcritical cycles [3,13–17]. Considering the above, advances in cycle architectures can further push adoption of ORC technologies for low temperature heat conversion. The TLC (trilateral cycle) and the TCORC (transcritical cycle) are further investigated in this work. Furthermore, they share the same component arrangement with the SCORC (subcritical ORC). Only the operating regime is different, necessitating technical modifications. Also a generalization of the TLC, named the PEORC (partial evaporating ORC) [18] is studied. In contrast to a TLC, the working fluid is allowed to partially evaporate.

The working fluid in a TLC, in contrast to the SCORC, does not undergo isothermal boiling but is only heated to a saturated liquid state. An ideal TLC would have a perfect triangular shape with the expansion process stopping in the two-phase region. If the expansion process stops in the dry region, the TLC is sometimes called a quadrilateral cycle [11]. Smith et al. [9] were amongst the first to extensively investigate the TLC. The optimization potential of modified ORC cycles can be appreciated by comparing the Carnot cycle and the ideal TLC. Evaluating the ideal TLC and Carnot cycle with finite heat capacity streams in their optimal upper cycle temperature results in an ideal conversion efficiency of the TLC which is roughly twice that of the Carnot cycle [9,19].

For a TCORC, the working fluid is brought to supercritical pressure and heated to a supercritical state. The heat rejection process is still done by condensing the working fluid. Both the TLC and TCORC approach a triangular shape in a T-s diagram, providing a good match with a finite capacity heat source [3].

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http://dx.doi.org/10.1016/j.energy.2015.04.094
0360-5442/

Nomenclature

$\dot{E}$	exergy flow, kW
e	specific exergy, kJ/(kg)
h	specific enthalpy, kJ/(kg)
$\dot{I}$	irreversibility rate, kW
$\dot{m}$	mass flow rate, kg/s
N	number of segments, –
PP	pinch point temperature difference, °C
$\dot{Q}$	heat transfer rate, kW
R^2	coefficient of determination, –
s	specific entropy, kJ/(kgK)
$\dot{W}$	work, kW
x	vapour quality, –
y_D	exergy destruction ratio, –
y_L	exergy loss ratio, –

Abbreviations

ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
AWF	all working fluids
CHP	combined heat and power
EWF	environmentally friendly working fluids
GWP	global warming potential
NFPA	National Fire Protection Agency
ODP	ozone depletion potential
PEORC	partial evaporation organic Rankine cycle
SCORC	subcritical organic Rankine cycle
SSE	sum of squares due to error
TCORC	transcritical organic Rankine cycle
TLC	trilateral cycle

Greek symbols

η	efficiency, –
ε	effectiveness, –

Subscripts

0	dead state
I	first law
II	second law
cf	cold fluid
crit	critical
evap	evaporator
ext	external
hf	heat carrier fluid
in	inlet
int	internal
isen	isentropic
liq	liquid
out	outlet
rec	recuperator
sat	saturated
wf	working fluid

However, there is no general consensus in scientific literature on the best performing cycle type. Schuster et al. [3] recommends a TCORC to approximate the ideal trilateral cycle. While in a study by Fischer et al. [10], the TLC had an improved net power output over both the SCORC and TCORC. In the latter study the TLC worked with water as working fluid while the TCORC employed R141b, R123, R245ca or R21. The inlet temperature pairs of heat carrier and condenser cooling fluid were (350 °C, 62 °C), (280 °C, 62 °C), (280 °C, 15 °C), (220 °C, 15 °C) and (150 °C, 15 °C). Yamada et al. [20] also investigated the SCORC, TLC and TCORC for the working fluid R1234yf. In their simulations the heat carrier stream is however neglected. The highest thermal efficiency was found for a supercritical cycle, however nothing is mentioned about the point of maximum net power output. Furthermore, several papers study either the TCORC or TLC, see Table 1 for a non exhaustive overview. To the authors' knowledge no paper:

- Compared and analysed the three cycle architectures; SCORC, TCORC and PEORC considering a large set of boundary conditions.
- Includes a systematic analysis of the constraints: restriction to environmentally friendly working fluids, superheated state after the expander, restriction on the heat carrier outlet temperature and the effect of the recuperator.
- Provides regression models for the three ORC architectures under consideration.

Table 1
Articles which investigate either the TLC or TCORC.

Reference	Cycle	Topic
[9,45]	TLC	Analysis and working fluid selection
[46]	TLC	Development of a screw expander
[11]	TLC	Comparison of TLCs
[47]	TLC	Piston engine as expander
[48]	TLC	Analysis of an ammonia-water TLC
[49]	TCORC	Parametric study
[50]	TCORC	Plate heat exchanger design
[51]	TCORC	Performance analysis in near-critical conditions
[52]	TCORC	Working fluid selection
[53]	TCORC	Working fluid selection

Because scattered boundary conditions and assumptions are used in literature, it is hard to make a fair comparison between different working fluids and cycle architectures. Therefore the effort was set up to analyse a large set of working fluids for several waste heat and cooling conditions in a methodological approach and this for the SCORC, PEORC and TCORC.

Another challenge is the availability of high performing ORC working fluids in the near future. From an environmental viewpoint several working fluids are a priori flagged as unsuitable. The ODP (ozone depletion potential) and GWP (global warming potential) are typically employed to assess respectively the impact on the ozone layer and the greenhouse effect. Working fluids with an ODP > 0 are banned by the Montreal protocol [21]. In anticipation of new European F-gas regulations [22] the incentive is launched to ban fluids with a GWP value of >150. By 2015 this rule would apply to domestic freezers and refrigerators and by 2022 in extension to some commercial installations. While the current rules apply for refrigerators and freezers an analogues restriction can be expected for power producing cycles. In addition to environmental constraints, safety is a main concern. The ASHRAE 34 standard provides a classification system which takes into account toxicity and flammability for refrigerants. Working fluids with a classification of A1 are preferred. These are non toxic and non flammable when tested in air at 21 °C and 101 kPa. However, in contrast to ODP and GWP regulations, there is no general consensus on acceptable or permitted working fluids concerning safety. For example, both