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Optimization of organic Rankine cycle power systems considering multistage axial turbine design

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HIGHLIGHTS

- A design methodology for ORC considering multistage axial turbine design is presented.
- A multistage axial turbine model is presented and validated.
- The methodology is applied to a waste heat recovery case on a container ship.
- n-butane and R245fa show the best trade-off between cycle and turbine design.
- The best solutions feature a single-stage with highly supersonic flow conditions.

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ABSTRACT

Organic Rankine cycle power systems represent a viable and efficient solution for the exploitation of medium-to-low temperature heat sources. Despite the large number of commissioned units, there is limited literature on the design and optimization of organic Rankine cycle power systems considering multistage turbine design. This work presents a preliminary design methodology and working fluid selection for organic Rankine cycle units featuring multistage axial turbines. The method is then applied to the case of waste heat recovery from a large marine diesel engine. A multistage axial turbine model is presented and validated with the best available data from literature. The methodology allows the identification of the most suitable working fluid considering the trade-off between cycle and multistage turbine designs. The results of the optimization of cycle and turbine suggest that the fluid n-butane yields the best compromise in terms of cycle net power output, turbine cost and efficiency for the considered case study. When a conservative design approach is adopted, the turbine features a two-stage configuration with supersonic converging nozzles and post-expansion. Conversely, a single-stage turbine featuring a supersonic converging-diverging nozzle and Mach number up to 2 is the resulting ideal choice when a more advanced design approach is implemented.

1. Introduction

Organic Rankine cycle power systems are employed in a range of different fields such as biomass applications, geothermal heat recovery, industrial waste heat recovery and solar applications. Compared to gas and steam turbines, the relatively small enthalpy drop in the expansion is advantageous for the ORC turbine design since it allows for the use of only one or a few turbine stages. For a given external heat source, the preliminary design of the ORC systems is performed by optimization of the thermodynamic cycle, which is typically hampered by technical and practical constraints. The system components such as the expander, heat exchangers and pump can be included in the cycle optimization or can be optimized in a subsequent stage of the design process.

Even though different aspects can be included in the preliminary design of ORC systems, a number of authors [1–4] have highlighted that the expander and working fluid are the two key elements.

Song et al. [5], Yun et al. [6], Yang and Yeh [7,8] and Andreasen et al. [9] performed design and optimization of ORC systems for marine applications but neglected the details of the expander design. Toffolo et al. [10], Maraver et al. [11] and Vivian et al. [12] proposed different methodologies for selecting the optimal working fluid and cycle configuration in ORC systems; however, in all cases the isentropic turbine efficiency was set to a fixed value. Astolfi et al. [13] and Martelli et al. [14] performed detailed thermodynamic and techno-economic analyses

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Nomenclature

C	absolute velocity [m·s^{-1}]; cost [k€]
$Dev.$	deviation [–]
M	Mach number [–]
N	rotational speed [rpm]
P	cycle power output [kW]
PI	performance index [kW·k€$^{-1}$]
PR	pressure ratio [–]
Re	Reynolds number [–]
T	temperature [K]
U	peripheral velocity [m·s^{-1}]
W	relative velocity [m·s^{-1}]
$\dot{Q}_{sc}$	scavenge air heat flow rate [W]
$\dot{m}$	mass flow rate [kg·s^{-1}]
c	axial chord [m]
c_p	molar heat capacity [J·kg^{-1}·K^{-1}]
e	blade rear surface suction curvature [m]
h	blade height [m]; specific enthalpy [J·kg^{-1}]
k_s	blade surface roughness [m]
n	number of stages [–]
o	blade opening [m]
p	pressure [Pa]
r	radius [m]
ss	nozzle to rotor axial clearance [m]
ss_{st}	interstage axial clearance [m]
t	trailing edge thickness [m]
t_{cl}	blade tip clearance [m]
z	number of blades [–]
DR	stage degree of reaction [–]
SP	turbine size parameter [m]
UA	overall heat transfer coefficient times area [kW·K^{-1}]

Abbreviations and acronyms

CFD	computational fluid dynamics
GWP	global warming potential
ODP	ozone depleting potential
ORC	organic Rankine cycle
TURAX	axial turbine simulation tool

Greek letters

α	absolute flow angle [°]
β	relative flow angle [°]
ϵ	residual error [–]
η	turbine isentropic efficiency [–]
ϕ	flow coefficient [–]
ψ	isentropic stage loading coefficient [–]

Subscripts

0	total conditions
1	nozzle inlet
2	rotor inlet
3	rotor outlet
a	axial component
c	critical
el	electrical
gear	gearbox
gen	generator
m	referred to the mean diameter
max	maximum
min	referred to the minimum opening
n	nozzle
net	net power
o	outlet
r	rotor
s	isentropic conditions
st	referred to one stage
t	turbine
tip	referred to the blade tip
tot	referred to the total turbine expansion
ts	total-to-static
tt	total-to-total
turb	turbogenerator
w	referred to the relative coordinate system

on ORC systems including multistage turbine optimization for geothermal and biomass applications, respectively. The aforementioned authors designed the ORC system by means of an assumption of the turbine efficiency, or estimated the turbine performance by interpolation of a statistical correlation developed for single-stage machines. Da Lio et al. [15] tackled the issue by constructing performance maps for single-stage axial turbines including the critical temperature of the working fluid as an additional parameter. Recently, White and Sayma [16] proposed a coupled analysis and optimization process of a small-scale ORC and a single-stage radial turbine with the aim to improve the economy-of-scale of the system. The authors developed performance maps based on a modified similitude theory to predict the performance of the turbine and of the ORC system.

In previous publications the present authors [17,18] developed a methodology for a coupled optimization of an organic Rankine cycle unit and an axial turbine, and demonstrated that it is essential to include the design of the turbine in order to make accurate estimations of the cycle performance. In Refs. [17,18], a single-stage axial turbine model was validated and employed. However, multistage turbine solutions are more suitable for applications requiring large power output and high expansion ratio, since they ensure better efficiency and limit the turbine loading under these conditions [19]. Multistage axial turbine arrangements are used by many ORC manufacturers, as documented in Colonna et al. [20], and a number of exemplary turbines have been documented, see Table 1. However, the literature on their design and optimization procedures is very limited. Some authors [21–25] presented the main criteria used in the development of the turbines in Table 1, however without providing the full details of their design and optimization methods. Other authors [26–28] have focused only on the analysis of the flow and performance of multistage ORC turbines using CFD techniques for a given turbine geometry. In all previous references, i.e. Refs. [21–28], the multistage turbine design was decoupled from the optimization of the ORC system. To the best of our knowledge, the most advanced approach where the multistage turbine performance is included in the ORC system optimization, is that proposed recently by Macchi and Astolfi [19]. They constructed a priori correlations to predict the turbine isentropic efficiency as a function of the size parameter and volume flow rate ratio for a fixed number of stages (up to three). The formula was developed using a mean-line model [34] employing ideal gas assumptions and optimizing the rotational speed. By using the proposed correlations, the thermodynamic cycle can be optimized to assess the impact of the different number of stages on the system performance. The approach by Macchi and Astolfi [19] is very general; however, it does not provide a specific optimal design for the considered application, i.e. the rotational speed is always set to the optimal value, the fluid is considered as an ideal gas, and the number of stages is limited to three. Moreover, the approach follows the same design philosophy for single and multi-stage turbine