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Design methodology for radial turbo expanders in mobile organic Rankine cycle applications ☆

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HIGHLIGHTS

- We describe a detailed design methodology for ORC radial turbo expanders.
- Toluene is selected as the working fluid for diesel engine waste energy recovery.
- A first turbine of 15.5 kW is designed but yields too small inlet blade heights.
- A second turbine for minimum power generates 34.1 kW with 51.5% efficiency.
- A third turbine for maximum efficiency produces 45.6 kW at 56.1% efficiency.

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ABSTRACT

Future vehicles for clean transport will require new powertrain technologies to further reduce CO_2 emissions. Mobile organic Rankine cycle systems target the recovery of waste heat in internal combustion engines, with the exhaust system identified as a prime source. This article presents a design methodology and working fluid selection for radial turbo expanders in a heavy-duty off-road diesel engine application. Siloxanes and Toluene are explored as the candidate working fluids, with the latter identified as the preferred option, before describing three radial turbine designs in detail. A small 15.5 kW turbine design leads to impractical blade geometry, but a medium 34.1 kW turbine, designed for minimum power, is predicted to achieve an isentropic efficiency of 51.5% at a rotational speed of 91.7 k min^{-1}. A similar 45.6 kW turbine designed for maximum efficiency yields 56.1% at 71.5 k min^{-1}. This emphasizes the main design trade-off – efficiency decreases and rotational speed increases as the power requirement falls – but shows reasonable radial turbine efficiencies and thus practical turbo expanders for mobile organic Rankine cycle applications are realizable, even considering the compromised flow geometry and high speeds imposed at such small scales.

1. Introduction

The global drive towards reducing CO_2 emissions from all forms of transport will require vehicle manufacturers to develop new technology to improve powertrain system efficiency. Despite the growth of the hybrid and electric passenger vehicle segments, internal combustion (IC) engines still power the vast majority of vehicle fleets. In the case of heavy-duty on-highway trucks, and off-road machines in particular, being neither candidates for full electrification nor especially well-suited to hybridization, the ability to reduce CO_2 emissions will depend on improvements in conventional powertrains because the IC engine will continue to be the prime mover for decades to come [1].

1.1. The organic Rankine cycle for waste heat recovery

Considering that ~22–35% [2] of the energy contained in the fuel is rejected to the exhaust, it is clear that waste heat recovery (WHR) technologies represent one of the best routes to achieving the required IC engine system efficiency improvements [3]. Although a significant part of the exhaust enthalpy will be extracted by turbocharging, there still exists an opportunity to recover some of the remaining lower grade heat energy. It is in this category, so-called *bottoming cycles*, where the organic Rankine cycle (ORC) is being investigated.

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Nomenclature

Symbols

A	area (m^2))
B	blockage coefficient (–)
b	blade height (mm)
C	absolute velocity ($m\ s^{-1}$)
D	diameter (mm)
h	enthalpy ($J\ kg^{-1}$)
i	incidence angle (radians)
M	Mach number (–)
$\dot{m}$	mass flow rate ($kg\ s^{-1}$)
N	rotational speed
N_s	specific speed (–)
r	radius (mm)
Re	Reynolds number (–)
s	blade span (mm)
SP	size parameter (m)
t	blade thickness (mm)
U	blade speed ($m\ s^{-1}$)
$\dot{V}$	volume flow rate ($m^3\ s^{-1}$)
V_R	volume flow rate ratio (–)
W	relative velocity ($m\ s^{-1}$)
$\dot{W}$	power (kW)
Z_b	blade number (–)

Greek letters

α	absolute flow angle (radians)
β	relative flow angle (radians)
ϵ	clearance (mm)
η	efficiency (–)
Λ	degree of reaction (–)
Φ	flow coefficient (–)
ϕ	velocity coefficient
Ψ	blade loading coefficient (–)
ρ	density ($kg\ m^{-3}$)
ζ	loss coefficient (–)

Subscripts

0	stagnation value
1	stage inlet station
2	stator outlet or rotor inlet station
3	stage outlet station
act	actual value
av	average
ex	exit
f	effective
geom	geometrical
h	hub
in	inlet station
is	isentropic value
min	minimum value
m	meridional component
N	nozzle
opt	optimum
out	outlet station
R	rotor
r	radial
s	shroud
s	static value
t	total value
win	windage
w	relative component

Superscripts

n	exponent in Table 11

Abbreviations

1D	one-dimensional
3D	three-dimensional
CFD	computational fluid dynamics
D4	octamethylcyclotetrasiloxane
DOE	Department of Energy
IC	internal combustion
MD2M	decamethyltetrasiloxane
MD3M	dodecamethylpentasiloxane
MDM	octamethyltrisiloxane
MM	hexamethyldisiloxane
ORC	organic Rankine cycle
WHR	waste heat recovery

ORC has been applied to WHR in IC engines before [4–7], and even as a direct replacement for a bus engine [8], but has yet to be commercialized, with the additional cost, complexity, and packaging of a mobile ORC system so far proving prohibitive [9]. However, with the aforementioned drive towards lowering CO_2 emissions from heavy-duty vehicles, the recent US DOE-backed *SuperTruck* program developed a mobile ORC system for an American on-highway truck, demonstrating better than 50% brake thermal efficiency [10]. Even so, no such system is known to have been successfully applied to an heavy-duty *off-road* machine.

1.2. Expanders for mobile ORC systems

Selection of the appropriate expander can be made by considering the availability and temperature of the heat source, and the turbine power that can be generated. For the present application these parameters can be determined from knowledge of the range of exhaust gas mass flow rates and temperatures experienced. A comparison of radial, screw, and scroll expanders [11] suggests that for a high temperature WHR application with an expected regenerated power in the order of 15 kW, a radial machine can be considered. Other expander types have also been assessed over the years [12–15].

2. Working fluid selection

2.1. Working fluid selection procedure

Selection of a suitable working fluid is crucial for ORC systems [11,16–18], and influences overall system performance and expander efficiency. The procedure used herein is shown in Fig. 1. Key factors in the selection process include: matching the working fluid critical temperature and that of the heat source (thereby avoiding prohibitive volume ratios across the expander [14,11]); matching the working fluid evaporating temperature and that of the heat source (to reduce irreversibilities [19]); and ensuring good chemical and thermal stability at temperatures well above operating conditions [20].

The preliminary screening step considers whether operation is at close to critical conditions. In the current work, the candidate fluids are down-selected to Toluene and Siloxanes, which have critical temperatures suitable for a ~300 °C heat source.

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