Performance prediction and design optimization of a kW-size reciprocating piston expander working with low – GWP fluids

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Outlines

Context
- Need of low-GWP working fluids -
- Need of kW-size expanders optimization -

Introduction to the work
- Micro-ORC test bench -
- Aim and methodology -

The integrated model
- Expander model -
  Correction of the heat transfer parameters
- Pump model -
  Correction of the slope of the pump characteristic curve

Results and discussion
- Fluids simulation and comparison -
- Built-in volume ratio optimization -

Conclusions
Outlines

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**Context - Need of low-GWP working fluids**

The regulation introduces a phase-down mechanism involving a gradually declining of high GWP fluids, as R134a.

### Montreal protocol EU legislation

<table>
<thead>
<tr>
<th>CFCs</th>
<th>HCFCs</th>
<th>HFCs</th>
<th>HFOs</th>
</tr>
</thead>
<tbody>
<tr>
<td>1900</td>
<td>1990</td>
<td>mid 1990</td>
<td>2005</td>
</tr>
</tbody>
</table>

- ozone depleting and very high GWP
- non ozone depleting but high GWP
- low GWP and no ozone depletion effect

**Refrigerant**

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>GWP</th>
<th>ODP</th>
</tr>
</thead>
<tbody>
<tr>
<td>HFC-134a</td>
<td>1430</td>
<td>0</td>
</tr>
<tr>
<td>HFO-1234yf</td>
<td>4</td>
<td>0</td>
</tr>
<tr>
<td>HFO-1234ze(E)</td>
<td>6</td>
<td>0</td>
</tr>
</tbody>
</table>

**GWP expected reduction VS years**

- R134a
- R245fa

**Suitable for hot source with temperature lower than 150 °C**
Context - Need of kW-size expanders optimization

Isentropic efficiency of the expander at maximum power in comparison with maximum attainable efficiency of the expander

Most of the experiments present a mismatch between:

- the cycle expansion ratio
  (imposed by the boundary conditions, i.e. hot and cold source temperatures)

- and

- expander expansion ratio
  (imposed by the built-in volume ratio)

Thus, isentropic efficiencies drop at maximum power output conditions due to over- and under-expansion losses

To achieve the optimum efficiency, the expander sizing should exactly match the design conditions
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Introduction to the work – Micro-ORC test bench

UNIBO LAB of MICRO-GENERATION

3 kW SIZE ORC SYSTEM for residential application

OPERATING TEMPERATURE: < 100 °C
FLUID: R134A
EXPANDER ARCHITECTURE: 3 RADIAL RECIPROCATING PISTONS - 230 cm³

Ref: Experimental Performance of a Micro-ORC Energy System for Low Grade Heat Recover. Bianchi et Al., ORC 2017
Introduction to the work – Aim and methodology

Previous works

Comprehensive experimental test of the micro-ORC; Calibration and validation of an expander semi-empirical model

Aim

Low-GWP fluids simulation; Expander optimization;

This work

Development of a model for performance prediction of the expander when working with fluids different from R134A

1. Introduction of a semi-empirical model of the gear pump to be integrated with the expander one, with the aim of predicting the expander performance in its real operation into the actual cycle;

2. Update of the models parameters related to thermofluid-dynamic properties of the working fluids, in order to account for the fluid substitution;

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The integrated model

**MODEL OUTPUTS**
- Electric power output ($\dot{W}_{el}$)
- Exhaust temperature ($T_{ex}$)
- Rotational speed ($N_{exp}$)

Imposed boundary condition = MODEL INPUTS

### Expander supply temperature ($T_{su}$)
- Manipulated by: Puffer heaters and hot water circuit
- Exp. data range: 65 – 85 °C

### Expander exhaust pressure ($p_{ex}$)
- Manipulated by: Cold water temperature
- Exp. data range: 5 – 9 bar

### Organic fluid mass flow rate ($\dot{m}$)
- Manipulated by: Pump rotational speed, controlled by pump frequency drive
- Exp. data range: 0.05 – 0.15 kg/s

### Number of activated loads ($n_{loads}$)
- The expander rotational speed is imposed by the equilibrium between the generator torque and the set load resistance

- $T_{hot,in}$ ($T_{su}$)
- $T_{su}$
- $T_{cold,in}$ ($p_{ex}$)
- $T_{ex}$
- $n_{loads}$
- $\dot{m}$
- $f_{PUMP} (\dot{m})$
The integrated model

**Why?**

Evaporation pressure and mass flow rate are independent input variables of the expander model, when the expander behavior is simulated without considering its integration into the ORC circuit, but in the real operation of the system, they are not.

**INPUTs**

- \( n_{loads} \)
- \( f_{pump} \)
- \( T_{H_2O\ hot\ IN} \)
- \( T_{H_2O\ cooling\ IN} \)

**Calculation code implemented on Matlab + CoolProp library**

**HP:**

- Steady-state condition
- Temperature delta at the evaporator;
- Pressure drop between the pump outlet and the expander inlet;
- Fluid at the state of saturated liquid at the exit of the condenser;

**Experimental trend of the pressures at the expander inlet and outlet vs ORC mass flow rate**

**INTEGRATED MODEL**

\[
p_{sat}(T_{H_2O\ cooling\ IN}) = p_{ex}
\]

\[
T_{H_2O\ hot\ IN} - \Delta T = T_{su}
\]

**OUTPUTs**

- \( N_{exp} \)
- \( \dot{W}_{el} \)
- \( T_{ex} \)
Reciprocating piston expander model

SEMI-EMPIRICAL MODEL – LUMPED PARAMETERS APPROACH
Model based on a combination of:
- a limited number of physically meaningful equations
- essential empirical parameters that must be calibrated with exp. data

Ref: Bianchi et Al. Application and comparison of semi-empirical models for performance prediction of a kW-size reciprocating piston expander.
**Reciprocating piston expander model**  
- Correction of the heat transfer parameters (AU)

EQUATIONS

**Heat transfer coeff. definition**

\[ U = \frac{Nu \cdot \lambda}{L} \quad \text{[1]} \]

**Dittus-Boelter correlation**

\[ Nu = 0.023 \cdot Re^{0.8} \cdot Pr^m \quad \text{[2]} \]

\[ \frac{(AU)_{\text{ref, fluid}}}{(AU)_{\text{ref, R134a}}} = \frac{Nu_{\text{fluid}} \cdot \lambda_{\text{fluid}}}{Nu_{R134a} \cdot \lambda_{R134a}} \quad \text{[3]} \]

\[ (AU)_{\text{ref, fluid}} = (AU)_{\text{ref, R134a}} \cdot \left( \frac{\rho_{\text{fluid}}}{\rho_{R134a}} \right)^{0.8} \cdot \left( \frac{c_{\text{pfluid}}}{c_{pR134a}} \right)^m \cdot \left( \frac{\lambda_{\text{fluid}}}{\lambda_{R134a}} \right)^{1-m} \cdot \left( \frac{\mu_{R134a}}{\mu_{\text{fluid}}} \right)^{0.8-m} \quad \text{[4]} \]

Ref: Giuffrida. Modelling the performance of a scroll expander for small organic Rankine cycles when changing the working fluid.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Fluids</th>
</tr>
</thead>
<tbody>
<tr>
<td>(AU)\text{su,ref} [W/K x 10^5]</td>
<td>R134a</td>
</tr>
<tr>
<td>5.65</td>
<td>6.38</td>
</tr>
<tr>
<td>(AU)\text{ex,ref} [W/K x 10^5]</td>
<td>R134a</td>
</tr>
<tr>
<td>9.23</td>
<td>10.19</td>
</tr>
</tbody>
</table>

The **thermodynamic properties of the fluids have been evaluated in the design operating point**: the reference state for the parameter (AU)\text{su,ref} is defined by a pressure of 15 bar and a temperature of 75 °C, while the reference state for (AU)\text{ex,ref} is defined by a pressure of 7 bar and a temperature of 50 °C.
The characteristic curves of the volumetric pump are defined by the trend of the pressure head as function of the volume flow rate for different pump frequencies.

The resistance of the system is influenced by the number of activated resistive loads dissipating the electrical power generated by the expander (i.e. by the resistance torque).
The characteristic curves of the volumetric pump are defined by the trend of the **pressure head as function of the volume flow rate** for different pump frequencies.

The **resistance of the system is influenced by the number of activated resistive loads** dissipating the electrical power generated by the expander (i.e. by the resistance torque).
The actual operating point of the pump is determined by matching the characteristic curve of the pump and the resistance characteristic of the circuit.
Gear pump model
- Correction of the slope of the pump characteristic curve

EQUATIONS

\[ \dot{V} = \dot{V}_{th} - \dot{V}_{leak} \quad [1] \]

- Leakage through internal clearance:
  \[ \dot{V}_{leak} = \frac{b \cdot h^3 \cdot \Delta p}{12 \cdot \mu \cdot l} \quad [2] \]
  \( b \) = meatus width; \( h \) = meatus height; \( l \) = meatus length

- Theoretical vol. flow rate:
  \[ \dot{V}_{th} = V_{cc} \cdot \frac{N_{pump}}{60} = \dot{V} (\Delta p = 0) \quad [3] \]

\[ \Delta p = (c_1 \cdot N_{pump} - \dot{V} \cdot c_2) \cdot \mu \quad [4] \]

is only influenced by the fluid viscosity:
change of the working fluid
variation of the curve slope

Constants depending on the pump geometry

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( c_1 )</td>
<td>5.65 \times 10^2 (-)</td>
</tr>
<tr>
<td>( c_2 )</td>
<td>5.24 \times 10^8 (m^3)</td>
</tr>
<tr>
<td>( V_{cc} )</td>
<td>64.7 (cm^3)</td>
</tr>
</tbody>
</table>

Corrected pump characteristic

Fluids Saturation liquid viscosity at 20 °C [Pa·s] \times 10^4

<table>
<thead>
<tr>
<th>Fluids</th>
<th>HFC - 134a</th>
<th>HFO - 1234yf</th>
<th>HFO - 1234ze(E)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viscosity</td>
<td>2.07</td>
<td>1.54</td>
<td>2.00</td>
</tr>
</tbody>
</table>

The viscosity of the fluid has been evaluated, for all the analyzed fluids, in the reference condition of saturated liquid at 20 °C.
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- Fluids simulation and comparison -
- Built-in volume ratio optimization -
Results and discussion
- Fluids simulation and comparison

Design conditions setting:
• Hot source temperature = 75 °C
• Cooling source temperature = 20 °C
• Activated loads = 5

Parametric study
varying the feed pump frequency between 25 and 45 Hz

<table>
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<tr>
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<th>Pressure ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>R134a</td>
<td>1.6 – 3.2</td>
</tr>
<tr>
<td>R1234yf</td>
<td>1.6 – 3.3</td>
</tr>
<tr>
<td>R1234ze(E)</td>
<td>1.8 – 4</td>
</tr>
</tbody>
</table>

Electric power output VS pressure ratio

Isentropic electric efficiency VS pressure ratio

Why?
Substitutes VS R134a
main contributes of influence
• Higher heat losses
• Higher pump internal leakages
Results and discussion
- Built-in volume ratio (BVR) optimization

Design conditions setting:
- Hot source temperature = 75 °C
- Cooling source temperature = 20 °C
- Activated loads = 5
- Expander shaft speed = 700 rpm

(the elaborated mass flow rate becomes an output of the model in place of the shaft speed)

Parametric study
varying the intake stroke between 0.2 and ~ 1

\[
\alpha = \frac{V_2 - V_1}{V_s} = \frac{1}{r_{v,exp}}
\]

Built-in volume ratio
Parameter of the expander model

Specific work and elaborated mass flow rate VS intake stroke

Electric power output VS intake stroke
The optimization of the BVR could lead to an increase of the electric power output of +40% with respect to the current value.

Why?

optimal BVR VS current BVR:

Reducing the intake stroke significantly decreases under-expansion losses.
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A semi-empirical model of the gear pump has been introduced and integrated with the expander one, with the aim of predicting the expander performance in its real operation into the actual cycle;

The model parameters related to thermofluid-dynamic properties of the working fluids have been updated in order to account for the fluid substitution;

The electric power output decreases by -45% when using R1234yf and by −27% in case of R1234ze(E)

R1234ze(E) seems to be the best candidate to maximize the electric power output, in place of R134a. However the use of low-GWP fluids affects the system performance

The optimization of the BVR could lead to an increase of the electric power output of about +40% with respect to the current value

The optimization of the BVR for the design conditions is fundamental to improve the expander performance
Fluid machines and Energy Systems

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