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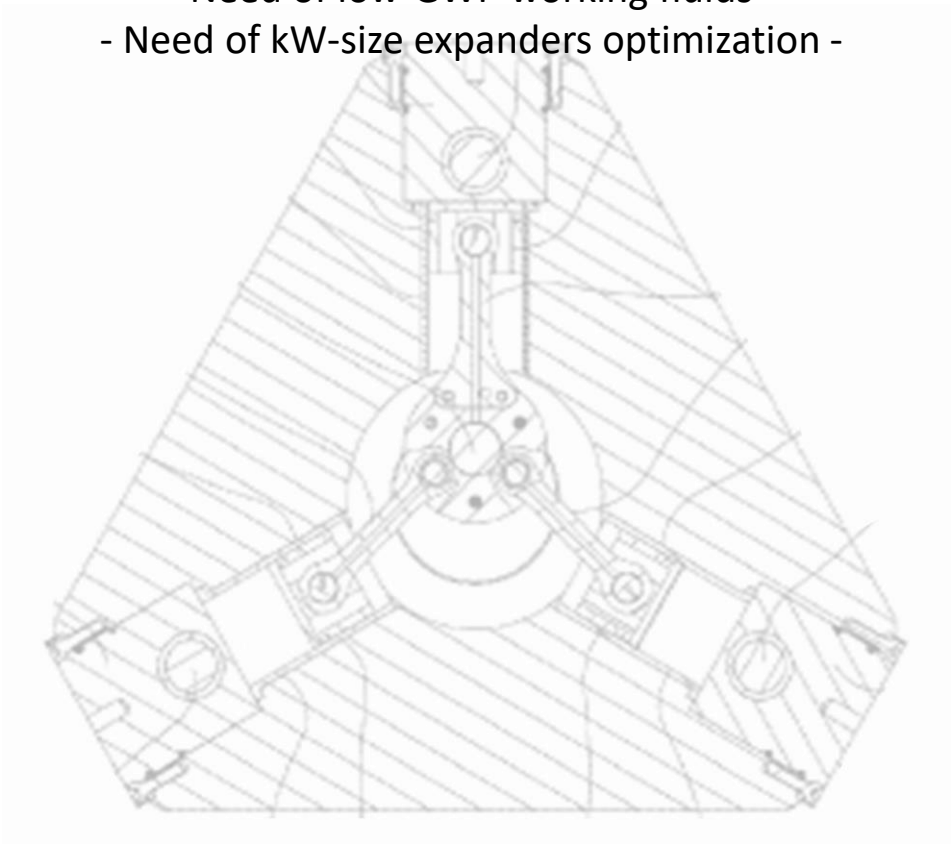
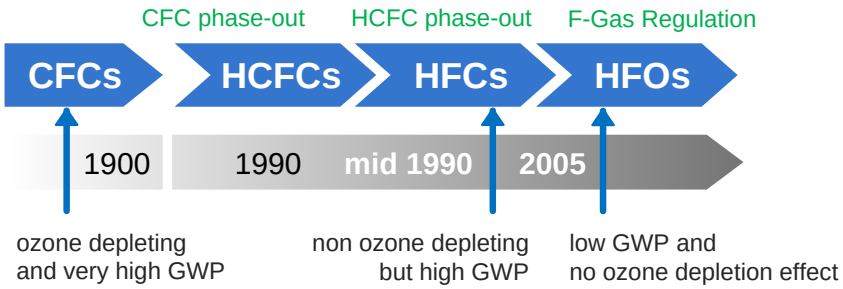
- Need of low-GWP working fluids -
- Need of kW-size expanders optimization -



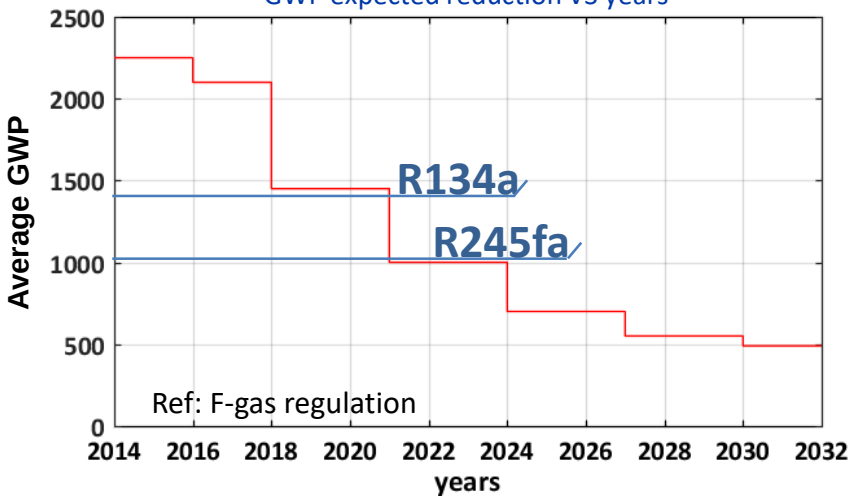
Context - Need of low-GWP working fluids

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

Montreal protocol EU legislation

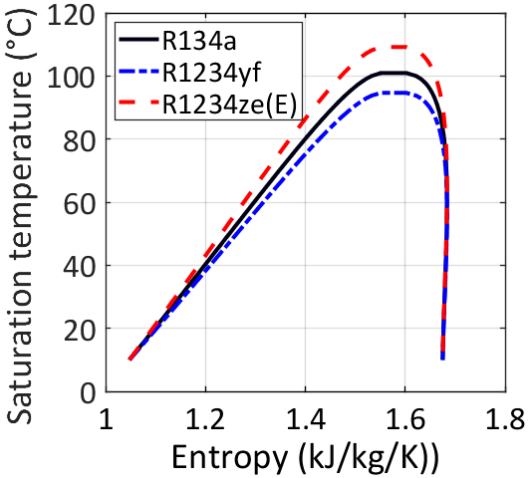


GWP expected reduction VS years



R134a substitutes: HFO-1234yf & HFO-1234ze(E)

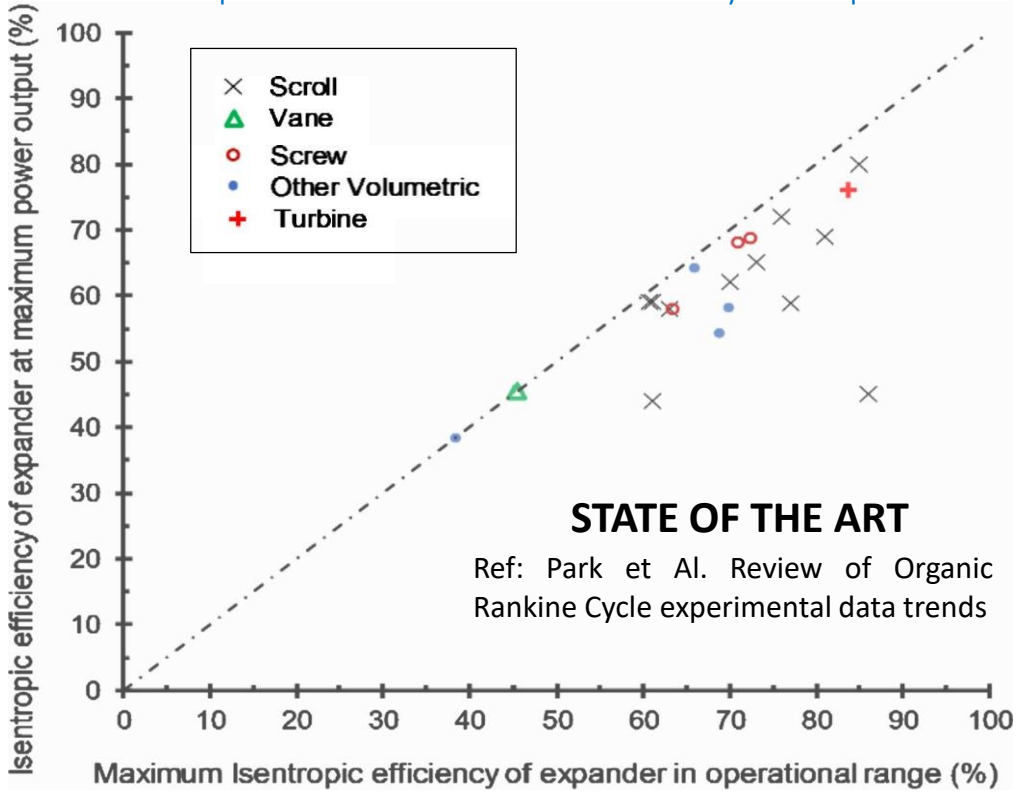
Refrigerant	GWP	ODP
HFC-134a	1430	0
HFO-1234yf	4	0
HFO-1234ze(E)	6	0



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



To achieve the optimum efficiency
the expander sizing should exactly match the design conditions

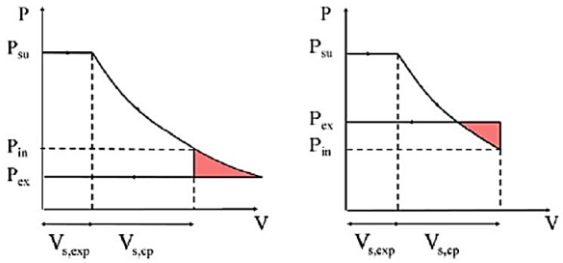
Most of the experiments present a **mismatch** between:

- 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**



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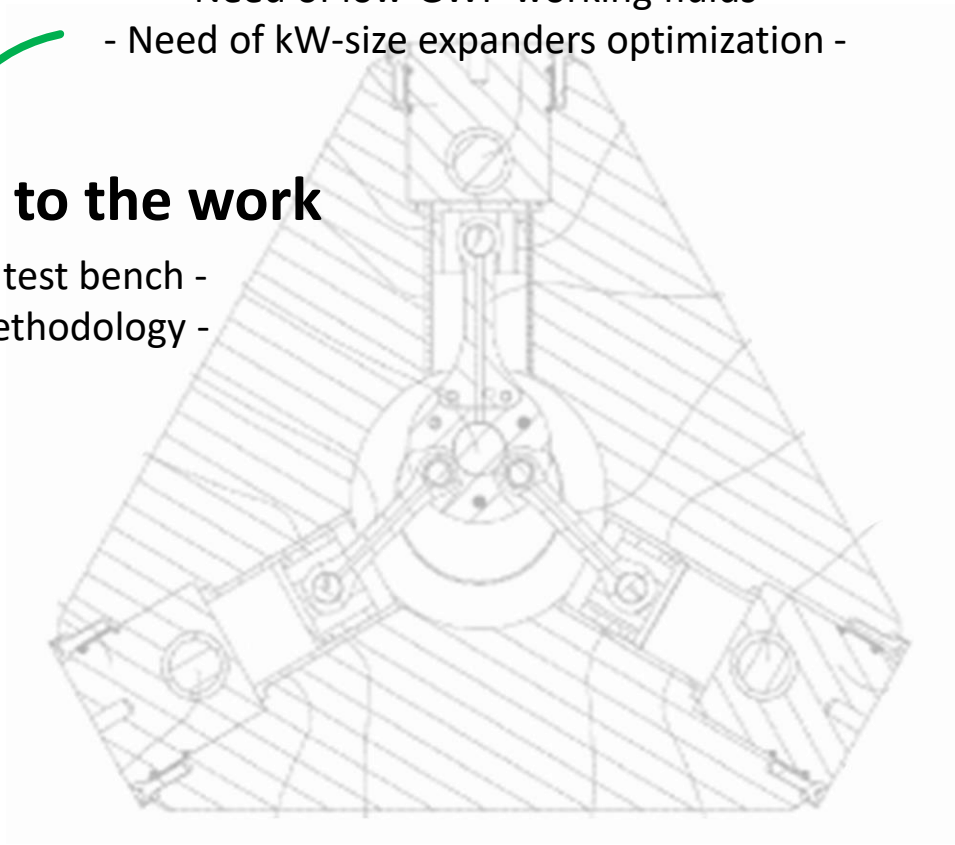
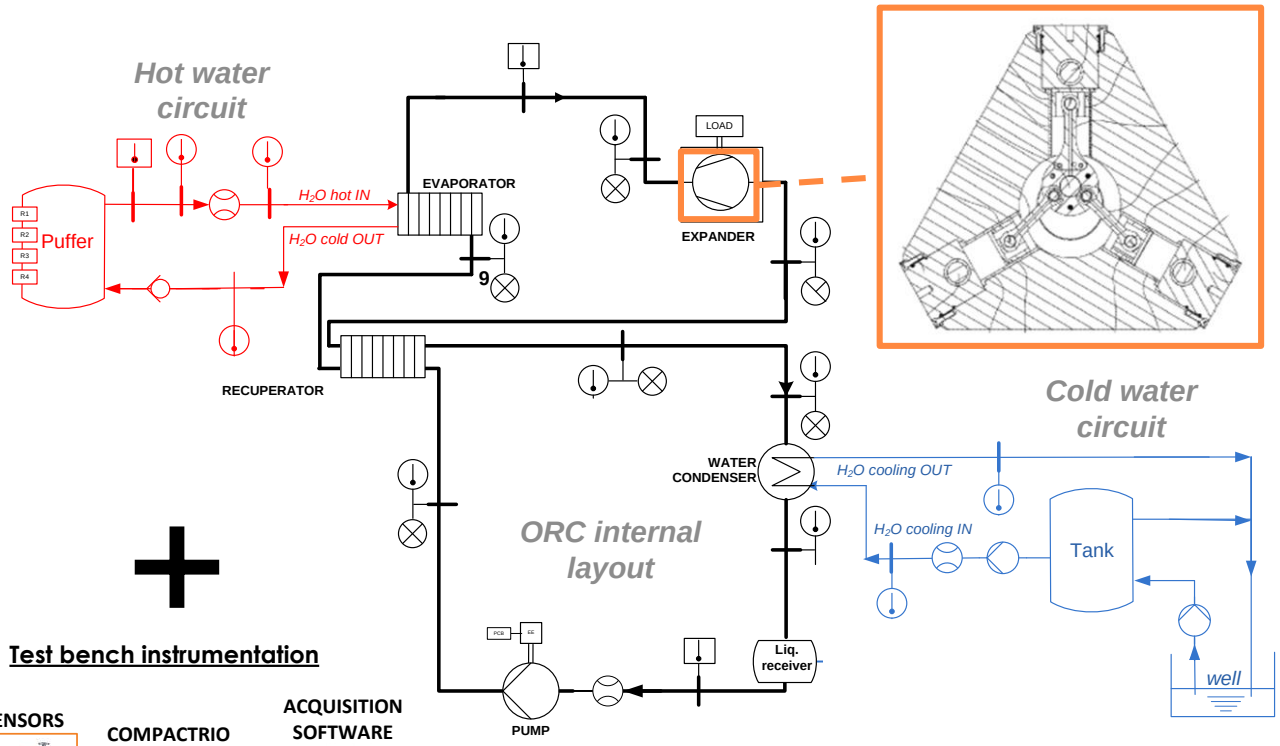


Introduction to the work – Micro-ORC test bench

UNIBO LAB of MICRO-GENERATION

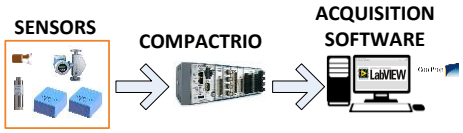
3 kW SIZE ORC SYSTEM for residential application

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



Ref: Experimental Performance of a Micro-ORC Energy System for Low Grade Heat Recover. Bianchi et Al., ORC 2017

Test bench instrumentation



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;
3. **Prediction of the performance** of the expander using low-GWP working fluids and identification of the **optimal built-in volume ratio** in design conditions.

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- Need of kW-size expanders optimization -

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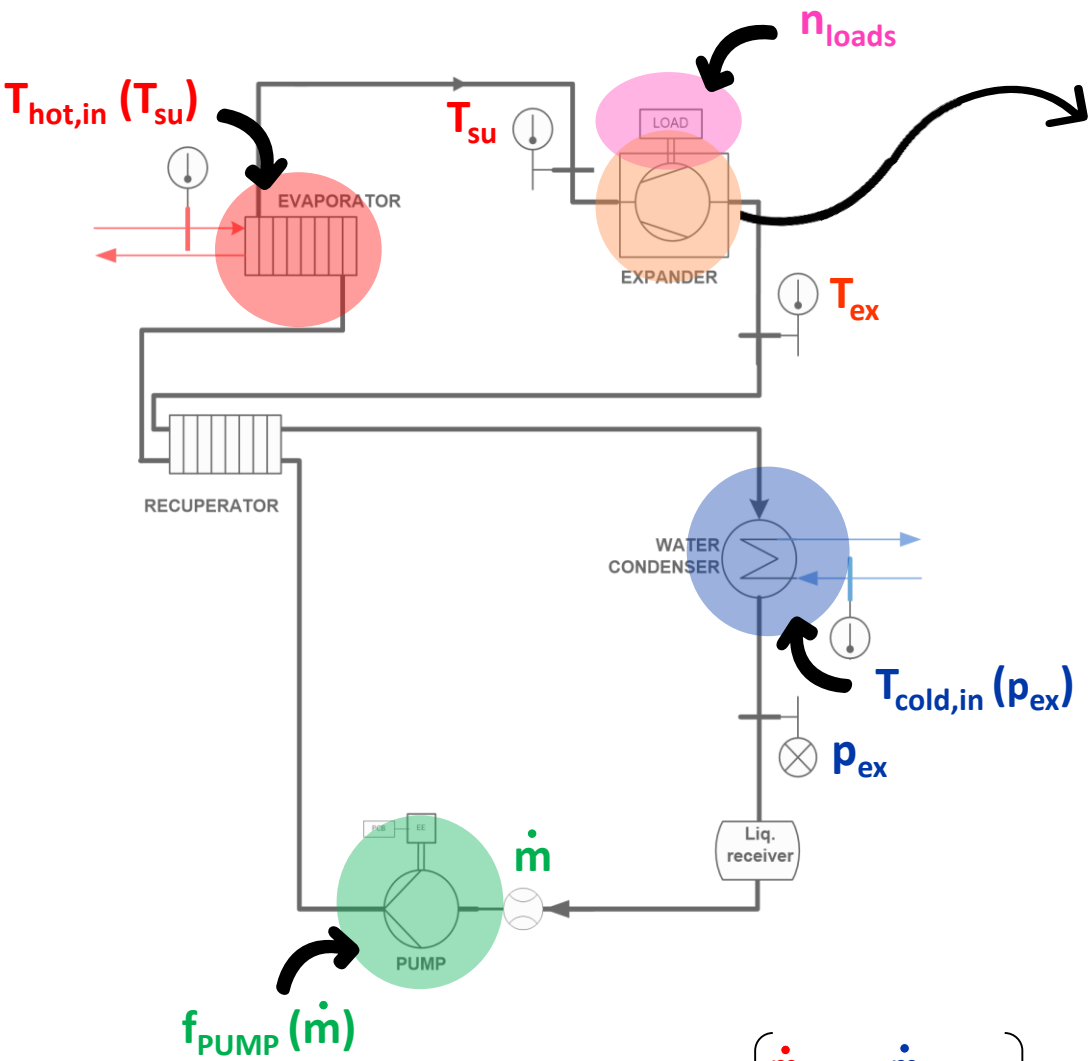
Correction of the heat transfer parameters

- Pump model -
Correction of the slope of the pump characteristic curve

The integrated model



$\dot{m}_{H2Ohot}$ $\dot{m}_{H2Ocold}$
kept ~constant

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

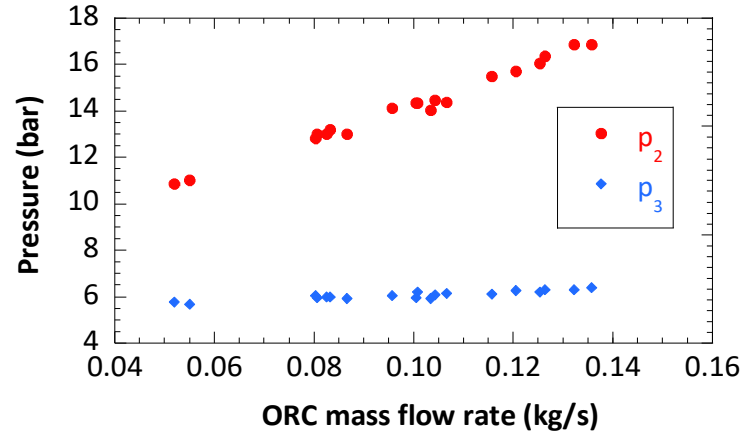
Number of activated loads (n_{loads})
The expander rotational speed is imposed by the equilibrium between the generator torque and the set load resistance

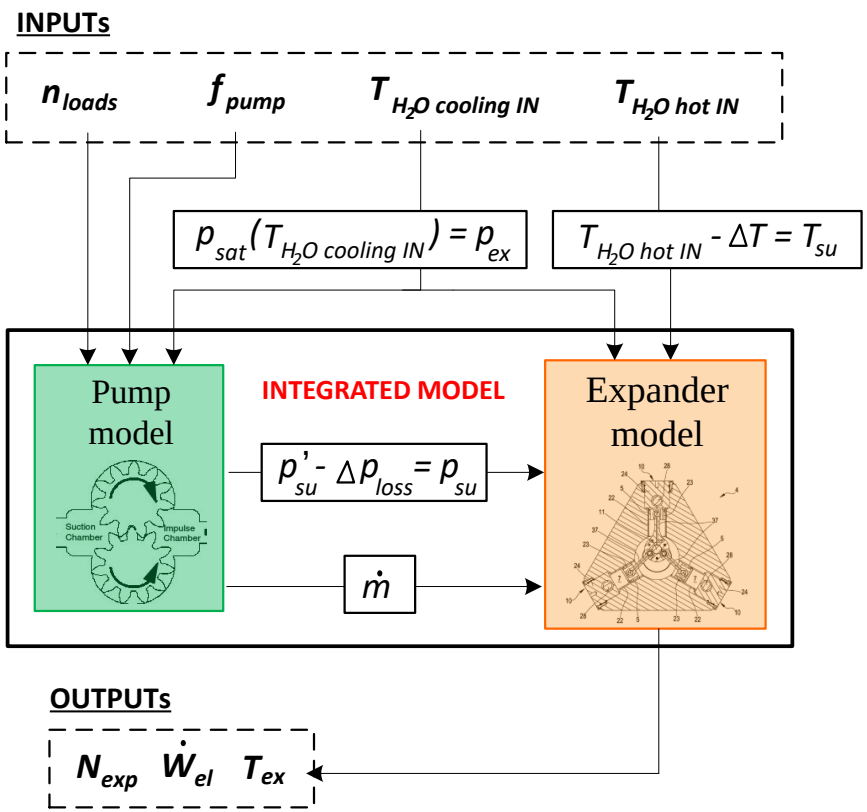
The integrated model

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.

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





Calculation code implemented on Matlab + CoolProp library

HP:

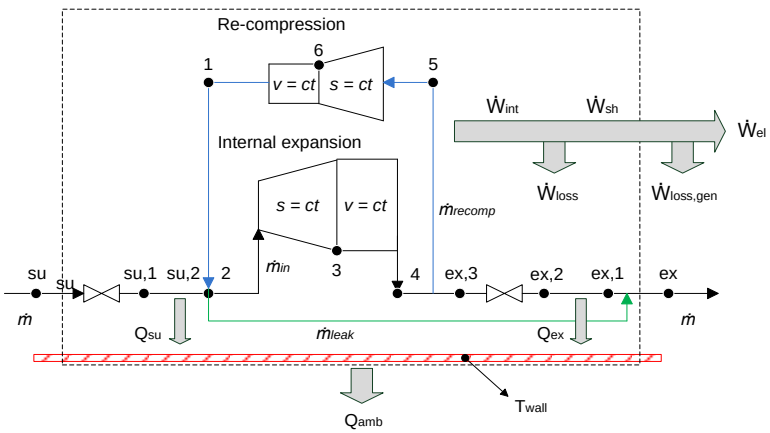
- Steady-state condition
- Temperature delta at the evaporator;
- Pressure drop between the pump outlet and the expander inlet;
- Fluid at the state of saturated liquid at the exit of the condenser;

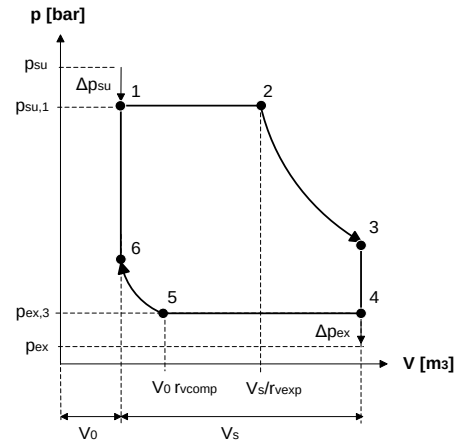
Reciprocating piston expander model

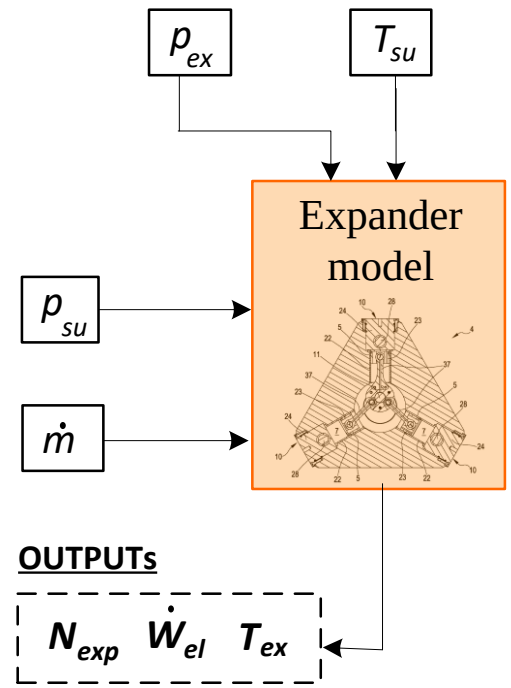
SEMI-EMPIRICAL MODEL – LUMPED PARAMETERS APPROCH

Model based on a combination of:

a limited number of physically **meaningful equations** + essential **empirical parameters** that must be calibrated with exp. data







Model parameters	Calibrated value		
$(AU)_{su,ref}$	Supply heat transfer coefficient	5.65e-05	(W/K)
$(AU)_{ex,ref}$	Exhaust heat transfer coefficient	9.23e-05	(W/K)
$(AU)_{amb}$	Ambient heat transfer coefficient	0.96	(W/K)
$r_{v,exp}$	Built-in volume ratio	1.459	-
$r_{v,comp}$	Re-compression volume ratio	1.25	-
V_0	Clearance volume	2.32e-02	(cm^3)
A_{leak}	Equivalent leakage area	5.51e-06	(m^2)
A_{su}	Supply nozzle equivalent section	1.47e-05	(m^2)
$W_{loss,ref}$	Constant friction losses	0.198	(W)
$W_{loss,N}$	Proportional friction losses	1.07e-05	(W/min)

Ref: Bianchi et Al. Application and comparison of semi-empirical models for performance prediction of a kW-size reciprocating piston expander.
Ref: Glavatskaya et Al. Reciprocating Expander for an Exhaust Heat Recovery Rankine Cycle for a Passenger Car Application.

Reciprocating piston expander model

- Correction of the heat transfer parameters (AU)

EQUATIONS

Heat transfer coeff. definition

$$U = \frac{Nu \cdot \lambda}{L} \quad (1)$$

Dittus-Boelter correlation

$$Nu = 0.023 \cdot \mathrm{Re}^{0.8} \cdot \mathrm{Pr}^{m} \quad (2)$$

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

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

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

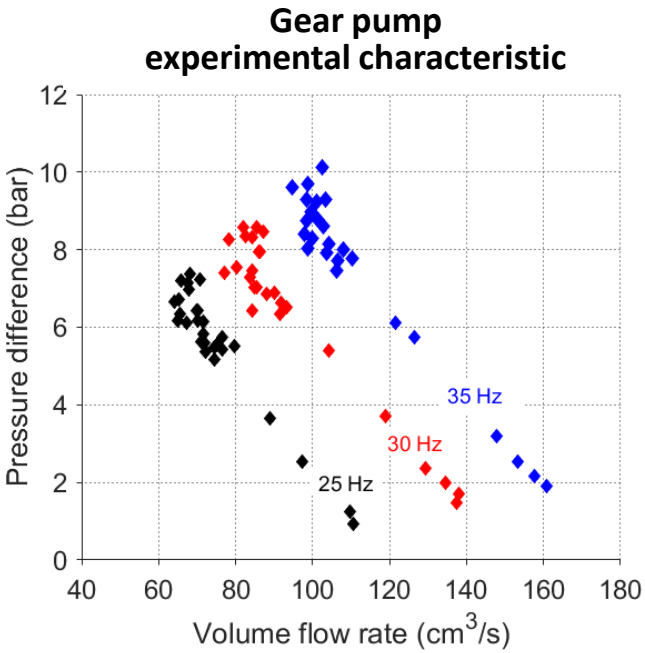
Parameters	Fluids		
	R134a	R1234yf	R1234ze(E)
$(AU)_{su,ref}$ [W/K x 10^5]	5.65	6.38 ↑	6.53 ↑
$(AU)_{ex,ref}$ [W/K x 10^5]	9.23	10.19 ↑	10.13 ↑

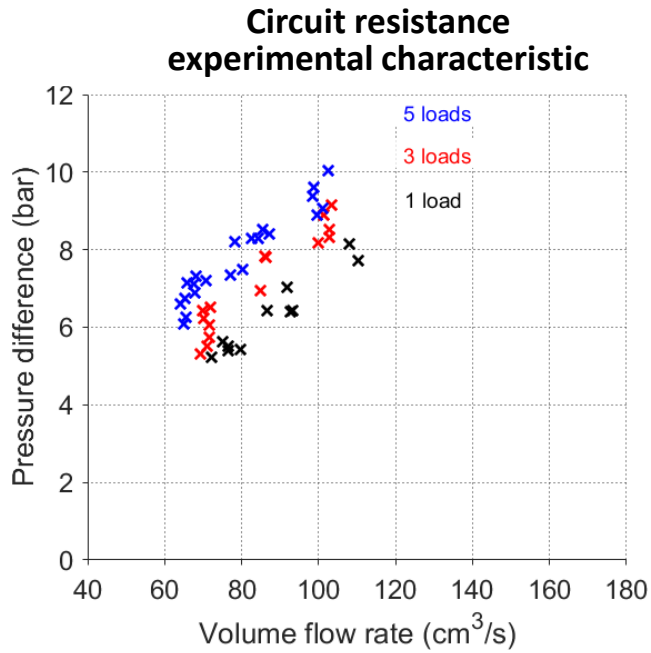
HIGHER HEAT LOSSES

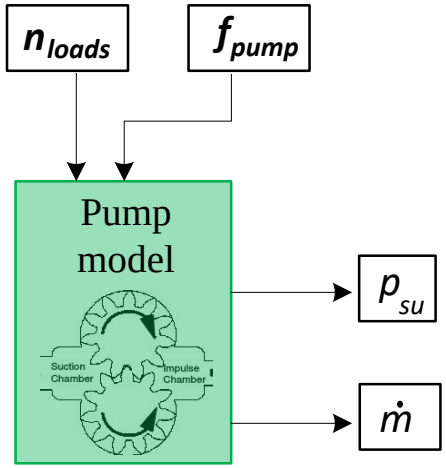
The **thermodynamic properties of the fluids have been evaluated in the design operating point**:
the reference state for the parameter $(AU)_{su,ref}$ is defined by a pressure of 15 bar and a temperature of 75 °C, while the reference state for $(AU)_{ex,ref}$ is defined by a pressure of 7 bar and a temperature of 50 °C.

Gear pump model

EXPERIMENTAL CHARACTERISATION







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.

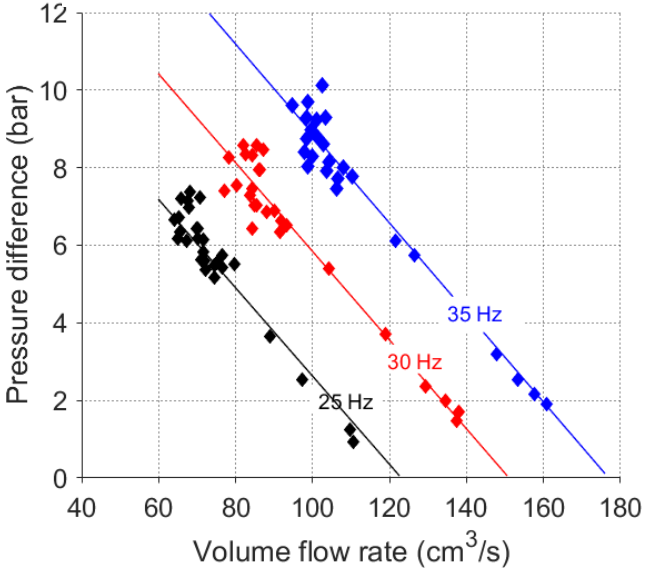
The **resistance of the system is influenced by the number of activated resistive loads** dissipating the electrical power generated by the expander (i.e. by the resistance torque)

Gear pump model

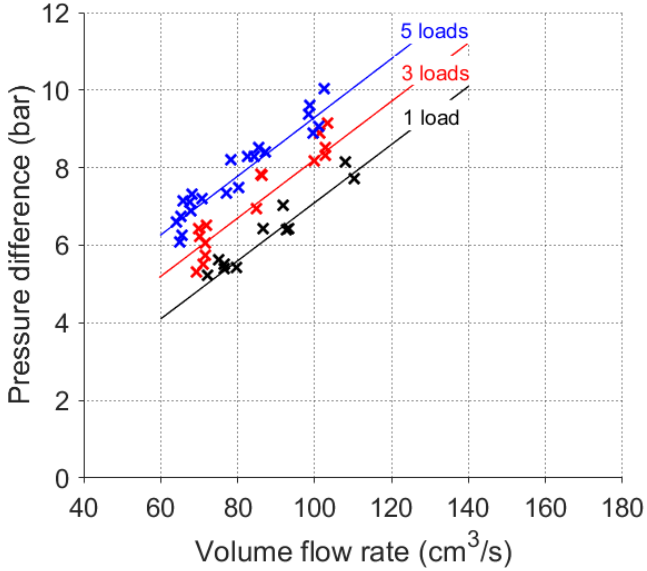
SEMI-EMPIRICAL MODEL –VOLUMETRIC PUMP CHARACTERISTIC

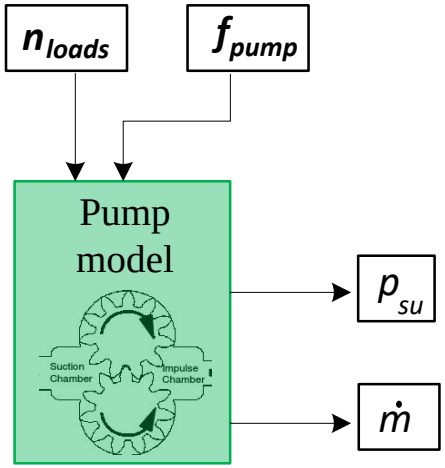
Interpolation of experimental data

Gear pump experimental characteristic



Circuit resistance experimental characteristic





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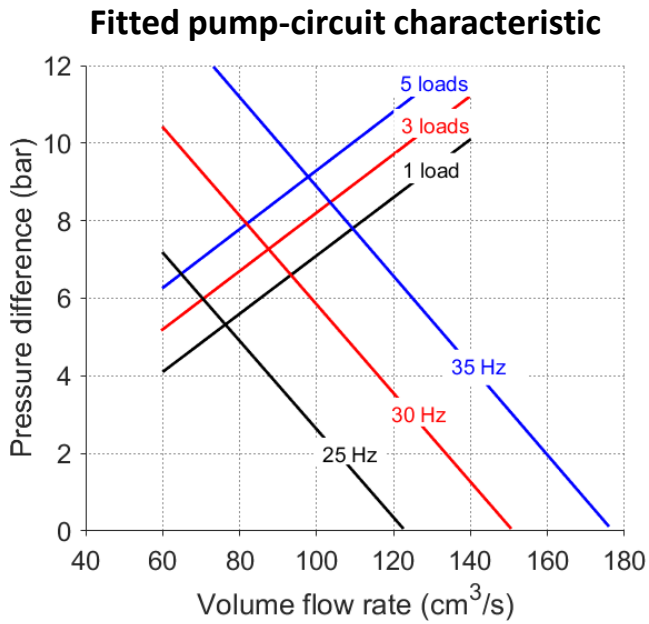
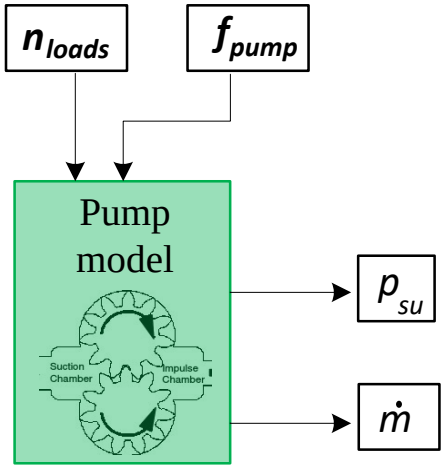




The **actual operating point** of the pump is determined by **matching**

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: **Poiseuilles law** $\dot{V}_{leak} = \frac{b \cdot h^3 \cdot \Delta p}{12 \cdot \mu \cdot l}$ (2)

  b = meatus width; h = meatus height; l = meatus length

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

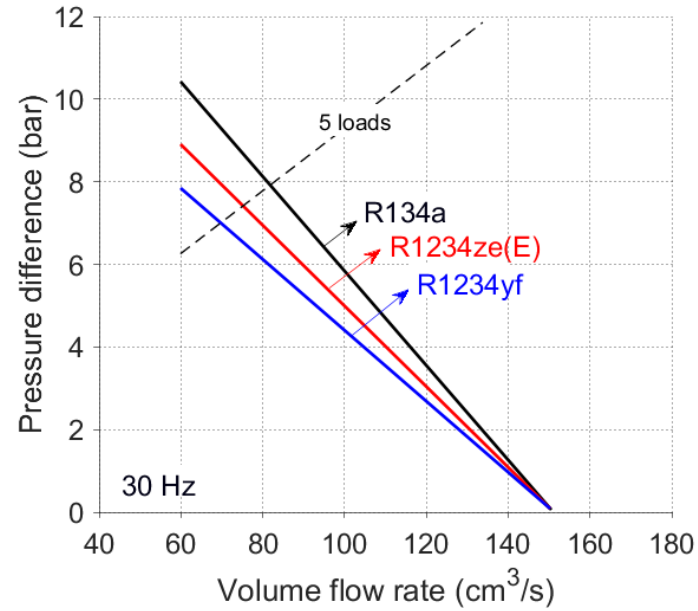
$$\Delta p = (c_1 \cdot N_{pump} - \dot{V} \cdot c_2) \cdot \mu \quad (4)$$

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

Constants depending on the pump geometry

Parameter	Value
c_1	5.65×10^2 (-)
c_2	5.24×10^8 (m^{-3})
V_{cc}	64.7 (cm^3)

Corrected pump characteristic



Fluids Saturation liquid viscosity at 20 °C [Pa·s] x 10^4		
HFC - 134a	**HFO - 1234yf**	**HFO - 1234ze(E)**
2.07	1.54 ↓	2.00 ↓

HIGHER LEAKAGE LOSSES

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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- Built-in volume ratio optimization -

Results and discussion
- Fluids simulation and comparison

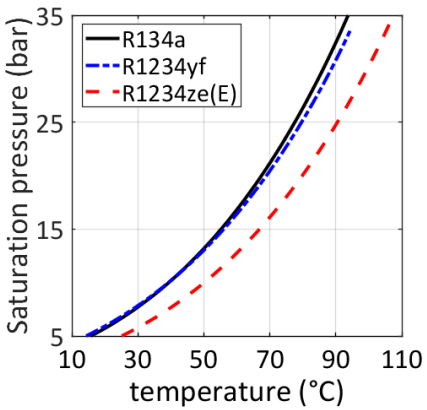
Design conditions setting:

- Hot source temperature = **75 °C**
- Cooling source temperature = **20 °C**
- Activated loads = 5

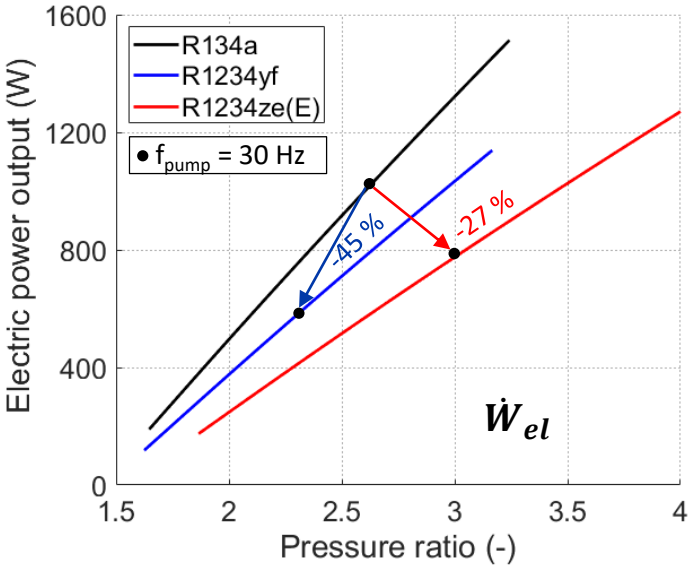
Parametric study

varying the feed pump frequency between 25 and 45 Hz

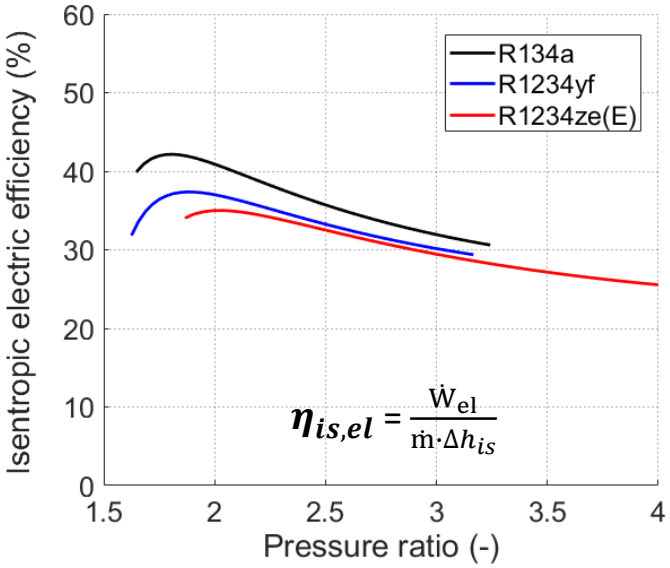
Fluid	Pressure ratio
R134a	1.6 – 3.2
R1234yf	1.6 – 3.3
R1234ze(E)	1.8 - 4



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)

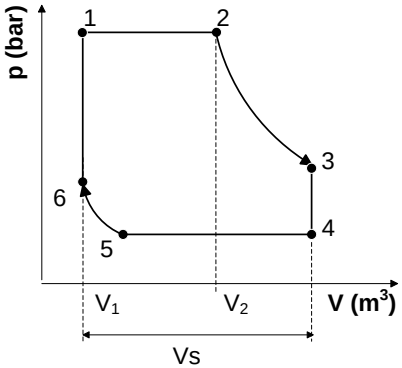
Parametric study

varying the intake stroke between 0.2 and ~ 1

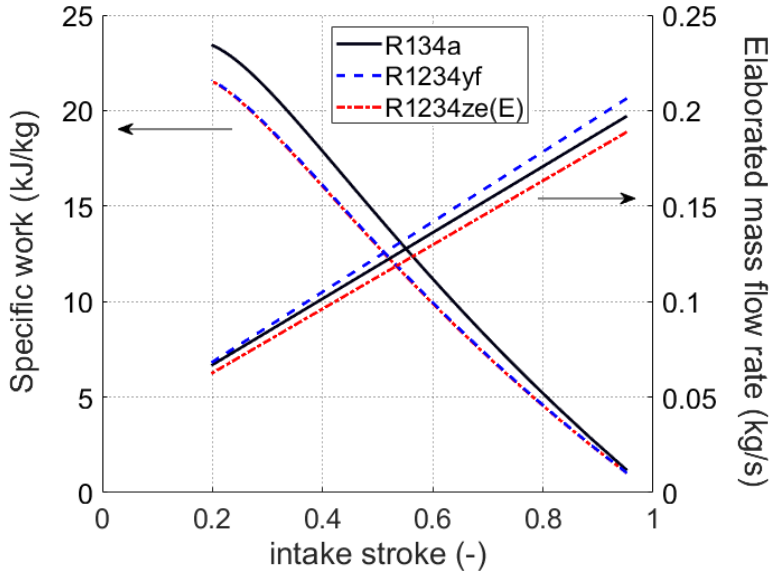
$$\alpha = \frac{V_2 - V_1}{V_s} = \frac{1}{r_{v,exp}}$$

||

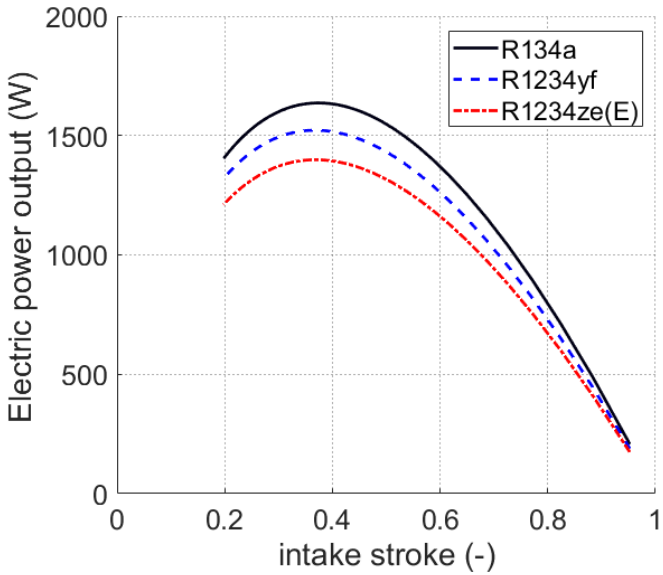
Built-in volume ratio
Parameter of
the expander model



Specific work and elaborated mass flow rate VS intake stroke

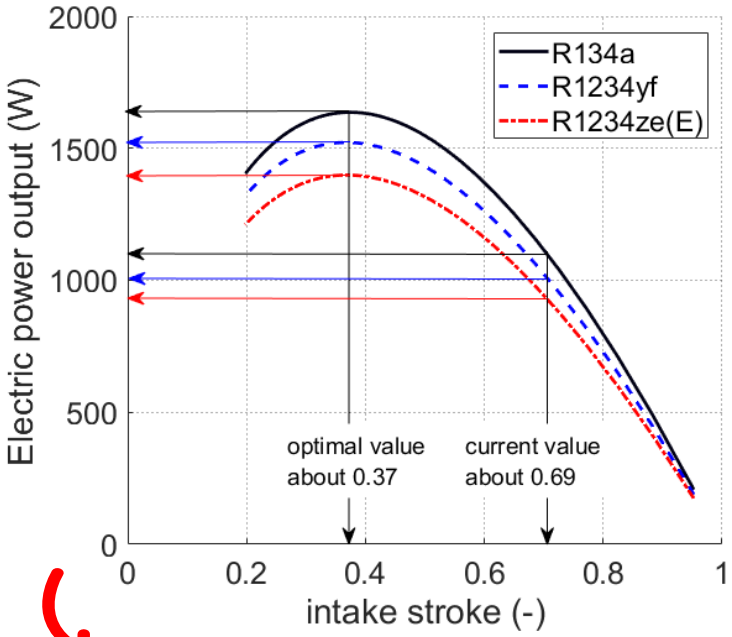


Electric power output VS intake stroke



Results and discussion

Electric power output VS intake stroke



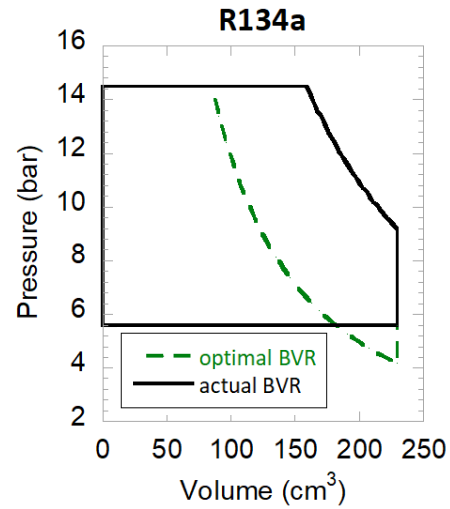
The **optimization of the BVR** could lead to an increase of the electric power output of **+40 %** with respect to the current value

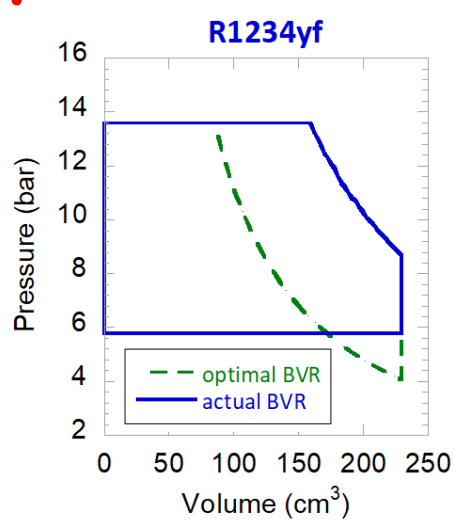
Comparison between indicator diagram obtained with the optimal BVR value and the one obtained with the current BVR value

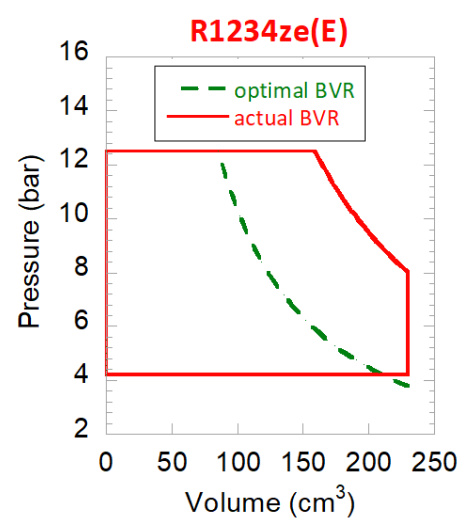
Why?

optimal BVR VS current BVR:

Reducing the intake stroke significantly decreases under-expansion losses







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Conclusions

- **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** decreasesby **-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

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