

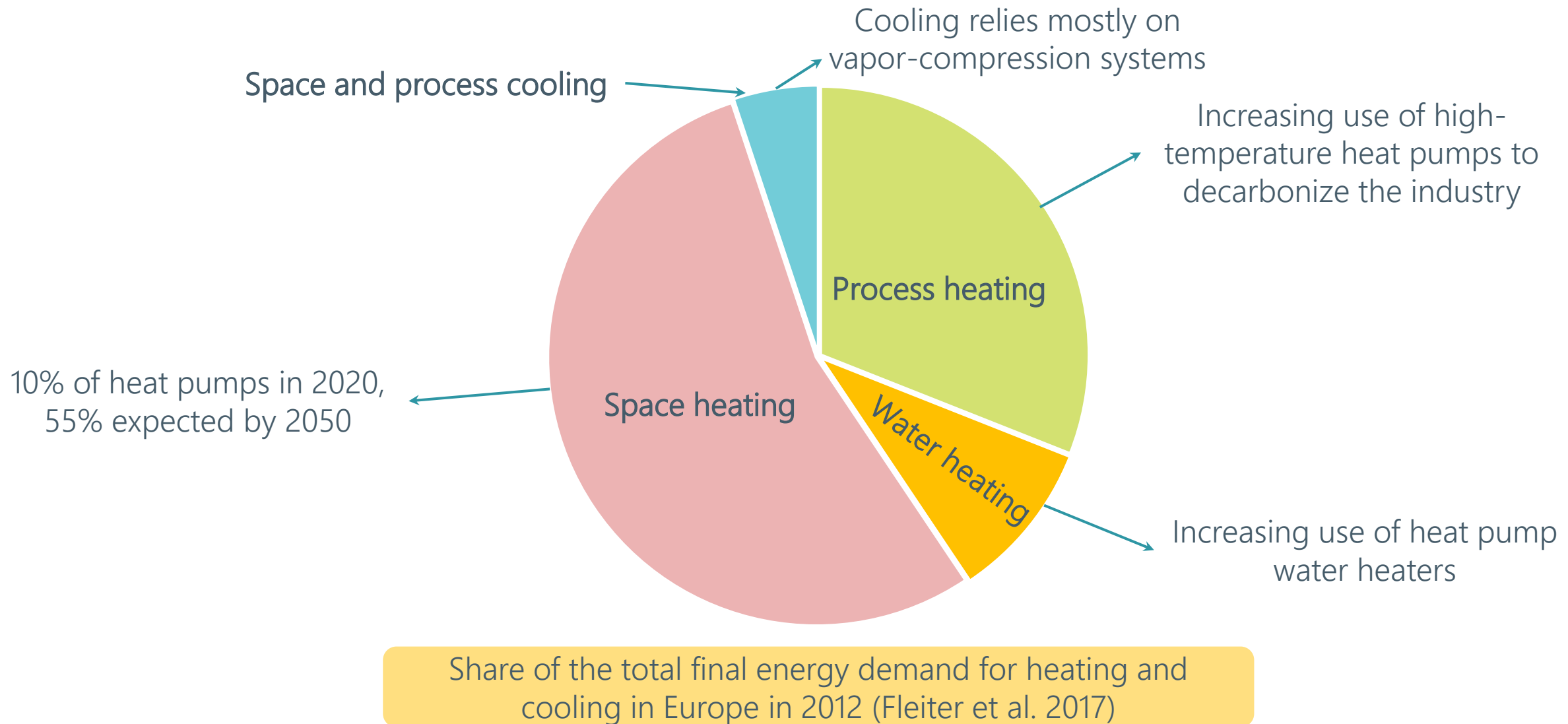
Contribution to the improvement of scroll compressors operating under two-phase conditions

Nicolas Leclercq – PhD public defense

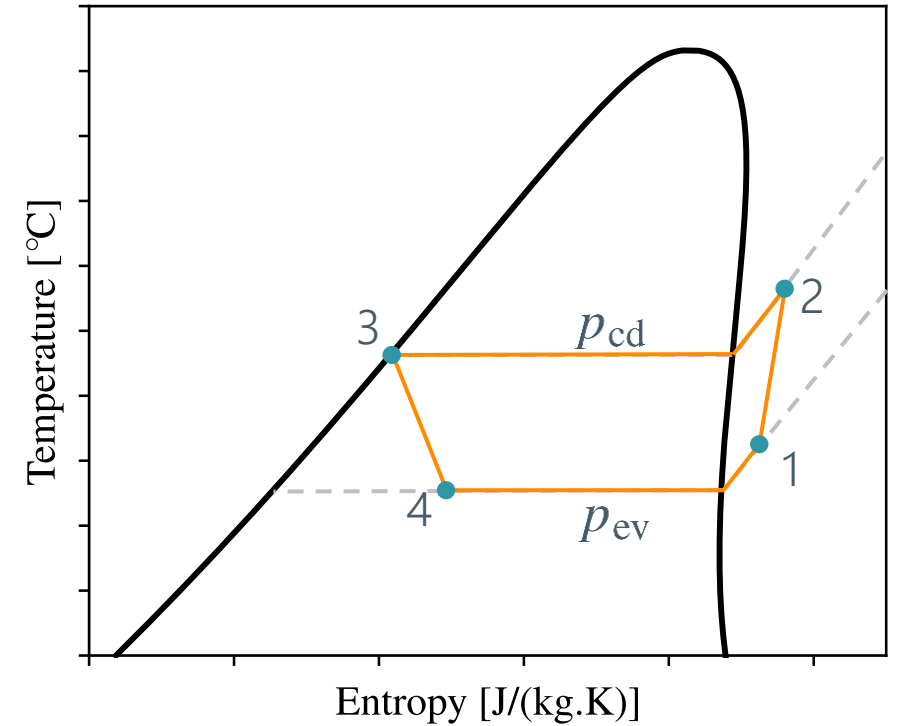
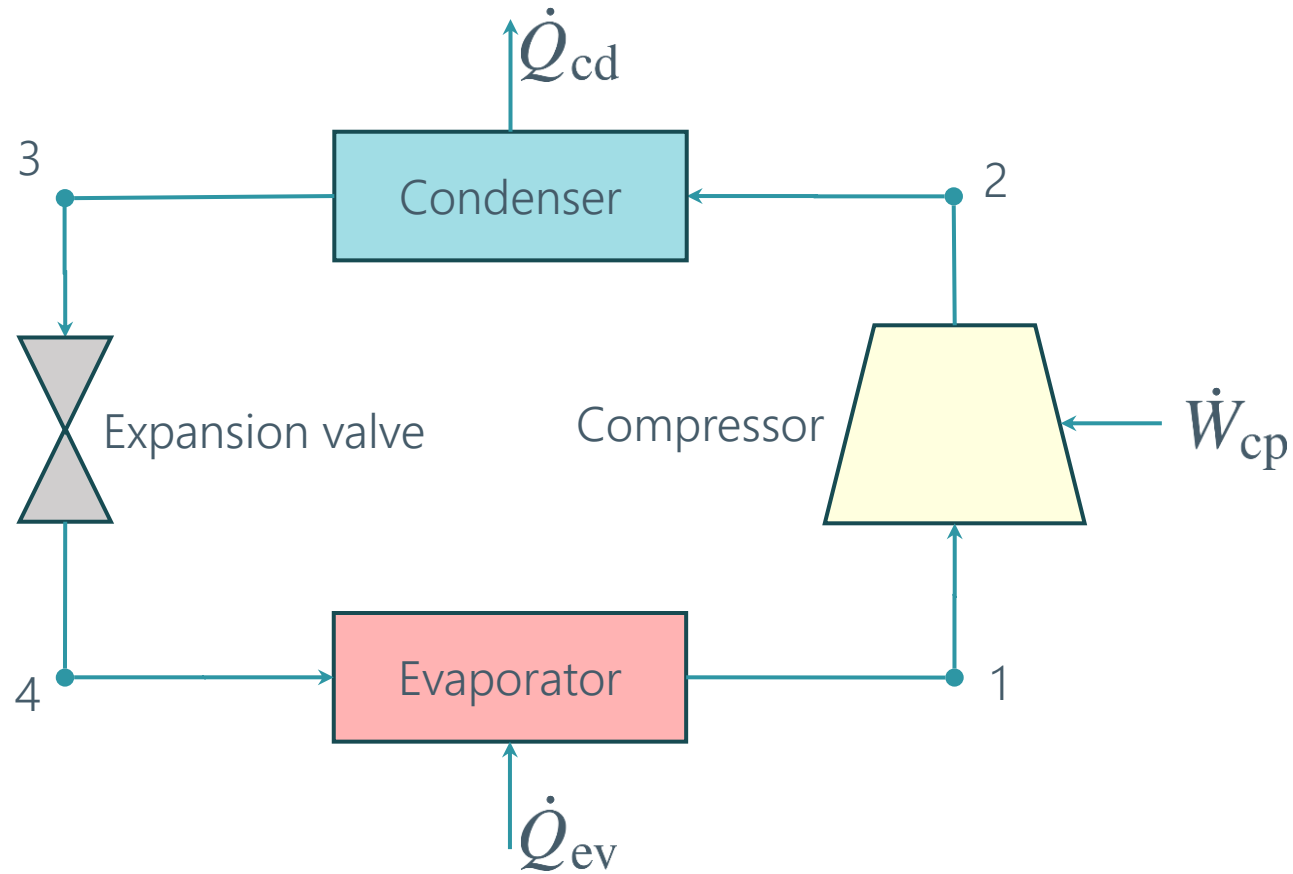
June 16th, 2026



Energy context : increasing use of vapor-compression cycles



Vapor-compression cycle

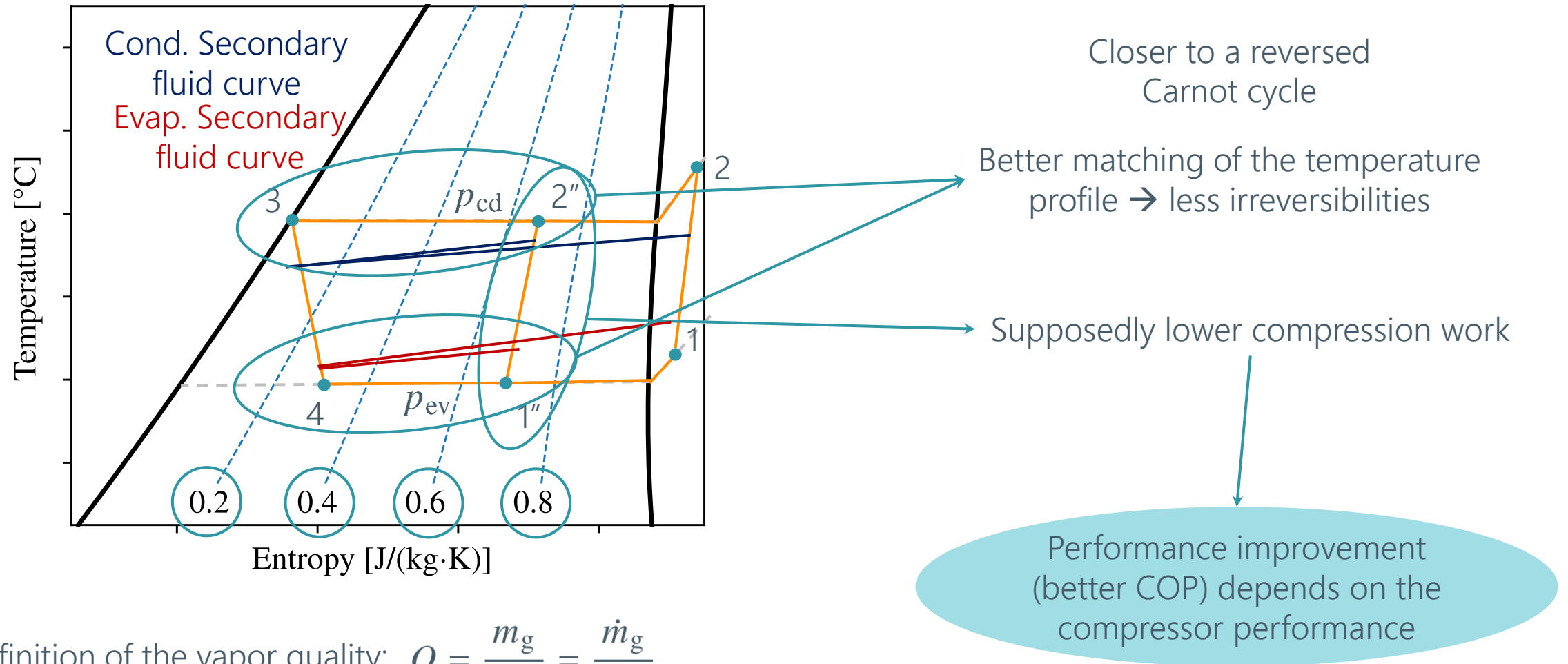


Coefficient of performance

Heat pump $\rightarrow \text{COP} = \frac{\dot{Q}_{cd}}{\dot{W}_{cp}}$

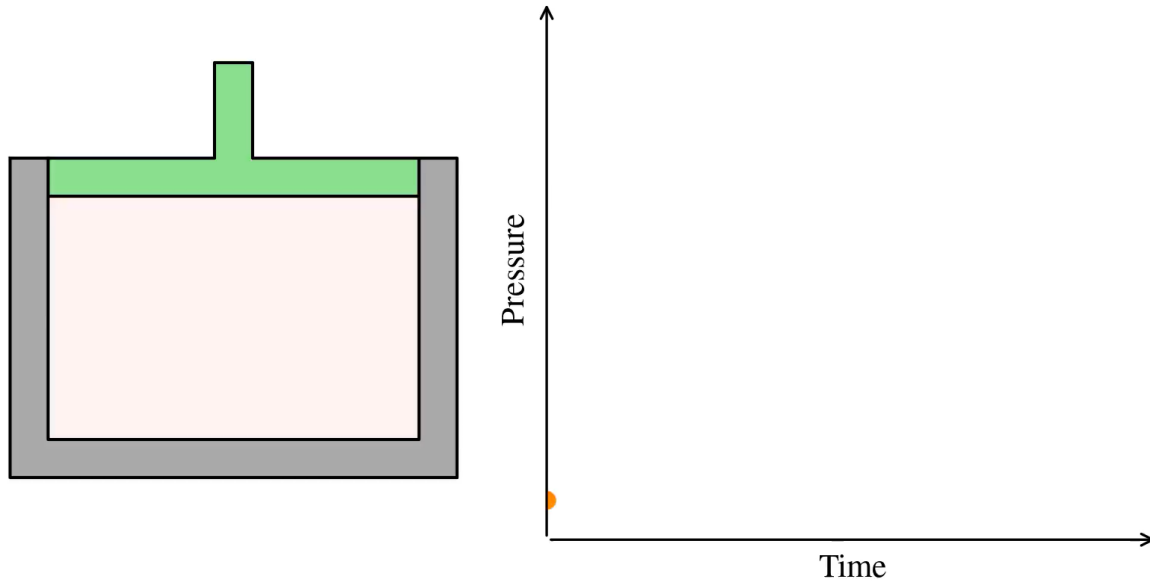
Refrigeration $\rightarrow \text{COP} = \frac{\dot{Q}_{ev}}{\dot{W}_{cp}}$

Use of two-phase compression in vapor-compression cycles

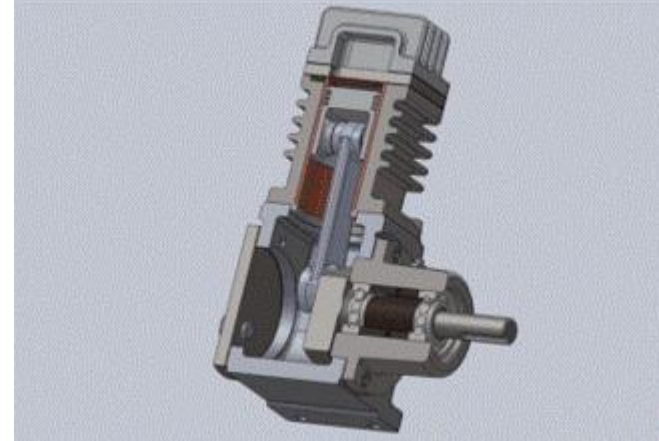


Definition of the vapor quality: $Q = \frac{m_g}{m_{tot}} = \frac{\dot{m}_g}{\dot{m}_{tot}}$

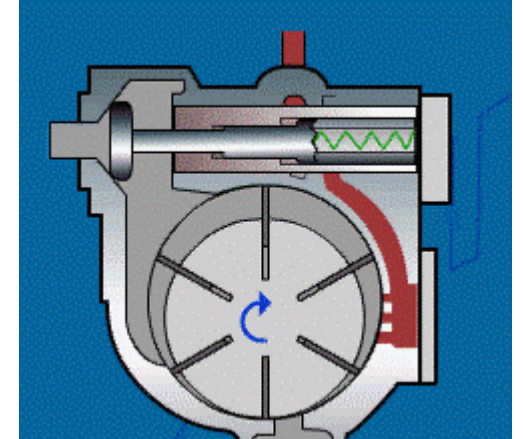
Positive-displacement compressors



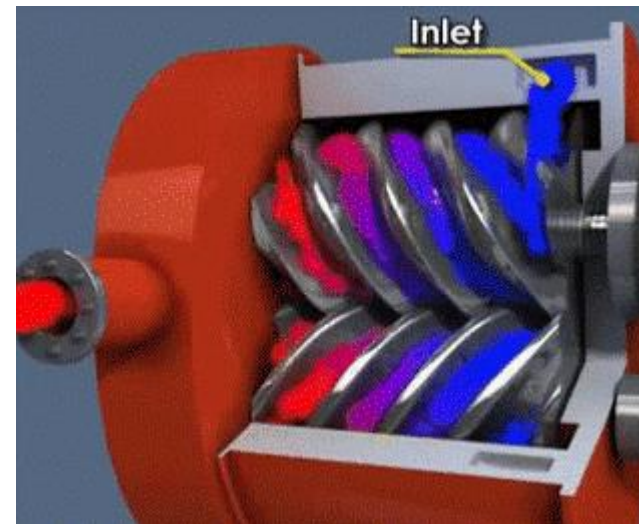
Reciprocating



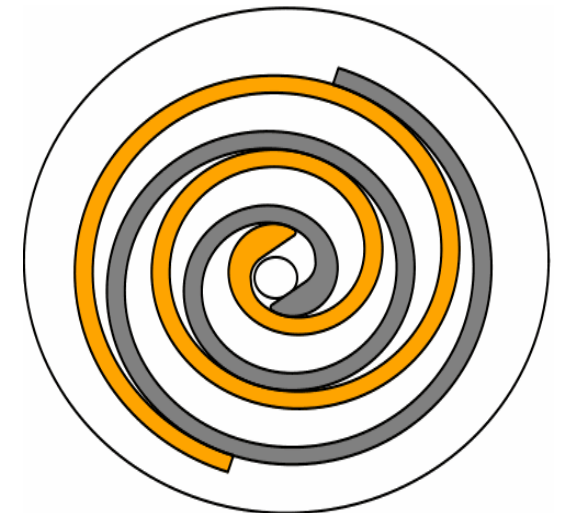
Rotary



Screw



Scroll



Research question

Research question

"How can two-phase compression in scroll compressor be made efficient enough to be of practical interest?"

Thesis title

"Contribution to the improvement of scroll compressors operating under two-phase conditions"

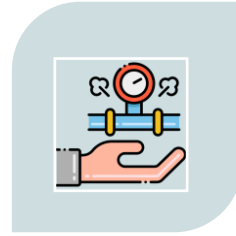
Presentation overview



1. Introduction
and Motivations



2. Oil-Refrigerant
Mixture Modeling



3. Experimental
Investigations



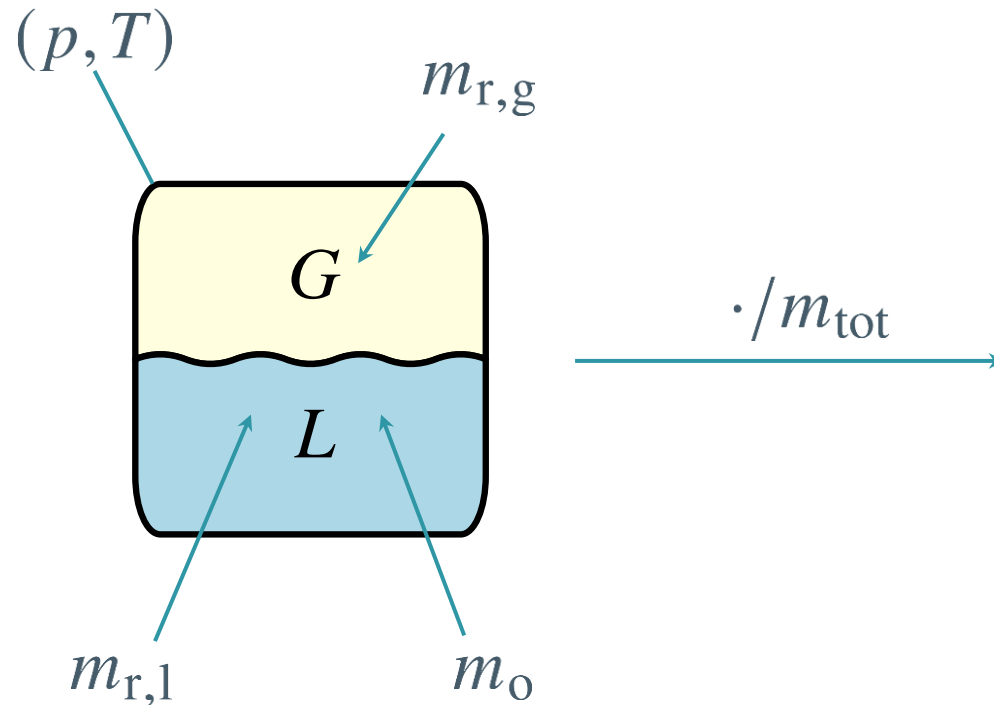
4. Modeling and
Validations



5. Results &
Conclusions

1. Introduction and Motivations
2. Oil-Refrigerant Mixture Modeling
3. Experimental Investigations
4. Modeling and Validation
5. Results and Conclusions

Definition of an oil-refrigerant mixture



	Ref	Oil	
Vapor	$z_{r,g}$	0	Q
Liquid	$z_{r,l}$	$z_{o,l}$	$1 - Q$
	z_r	z_o	1

Key variable defining the solubility:

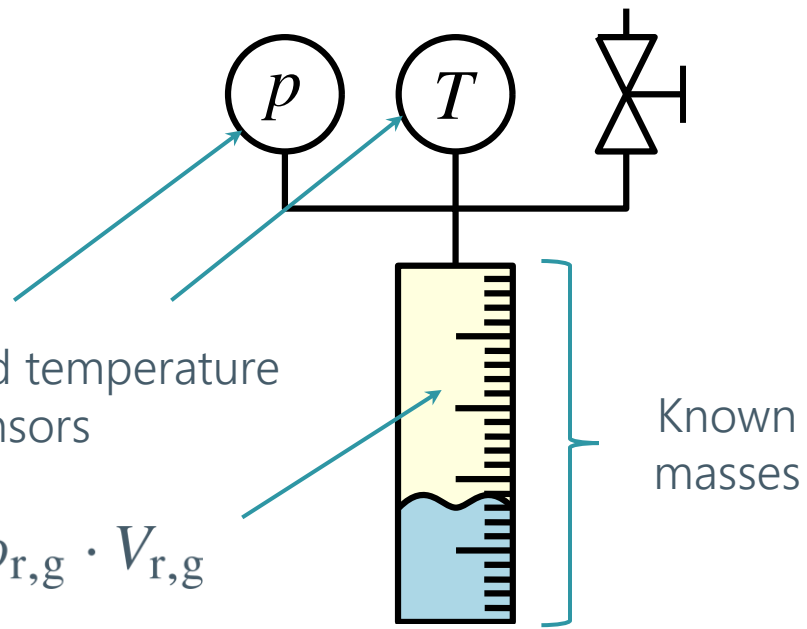
$$x_r = \frac{m_{r,l}}{m_l} = \frac{z_{r,l} \cdot \dot{m}_{tot}}{(1 - Q) \cdot \dot{m}_{tot}} = f(p, T)$$

$$Q = f(z_r, p, T)$$

Experimental liquid composition

Climatic chamber → fix the temperature

Get $x_r = f(p, T)$



Pressure and temperature sensors

Known masses

$$m_{r,g} = \rho_{r,g} \cdot V_{r,g}$$

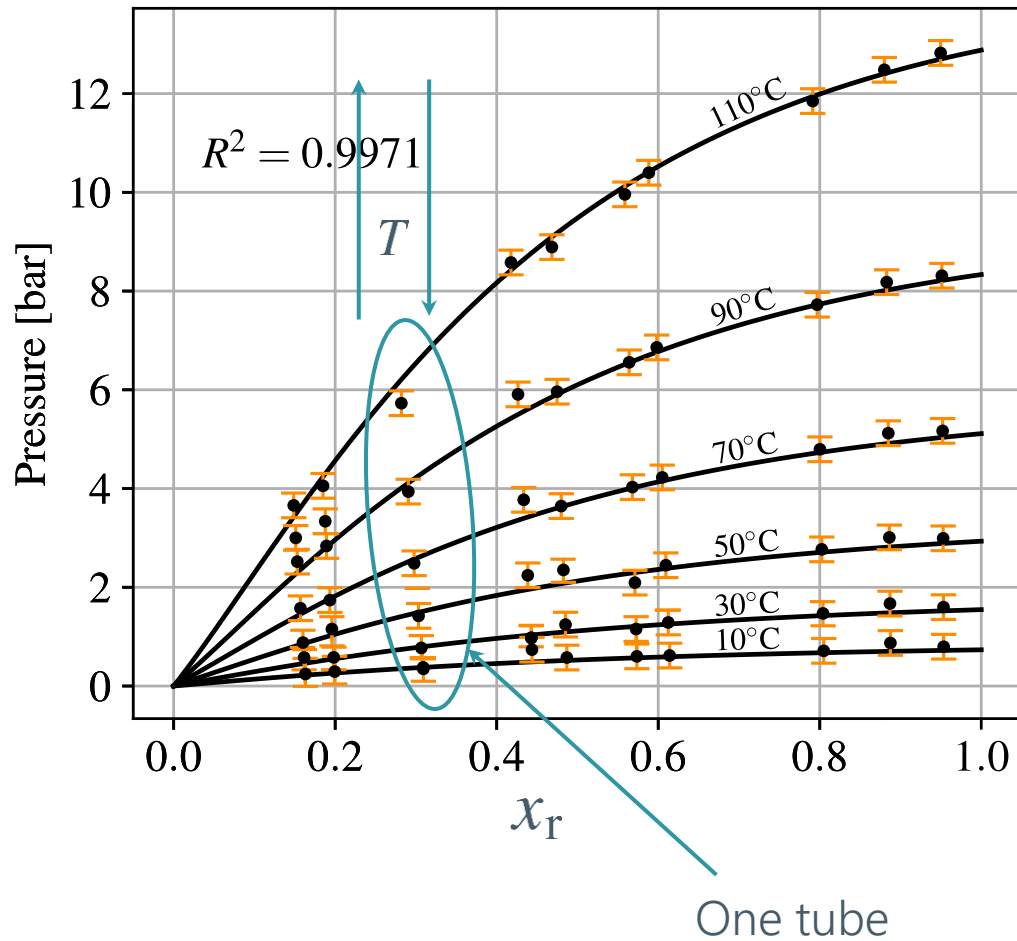
R1233zd(E) – POE oil

$$\left. \begin{array}{l} m_{r,l} = m_r - m_{r,g} \\ m_l = m_{tot} - m_{r,g} \end{array} \right\} x_r = \frac{m_{r,l}}{m_l}$$



Modeling of the solubility

R1233zd(E) – POE oil



Two ways of modeling

- Fugacity equality and use of an equation of state
- Empirical correlations

$$\tilde{y}_i \phi_i^V p = \gamma_i \tilde{x}_i \phi_i^\sigma p_i^\sigma \mathcal{P}_i$$

$$p(x_r, T) = x_r p_r^\sigma + x_r \cdot (1 - x_r) \cdot (a_1 + a_2 T + a_3 T^2 + a_4 + a_5 x_r T + a_6 x_r T^2)$$

Similar techniques can be used for the density, the viscosity and other transport properties

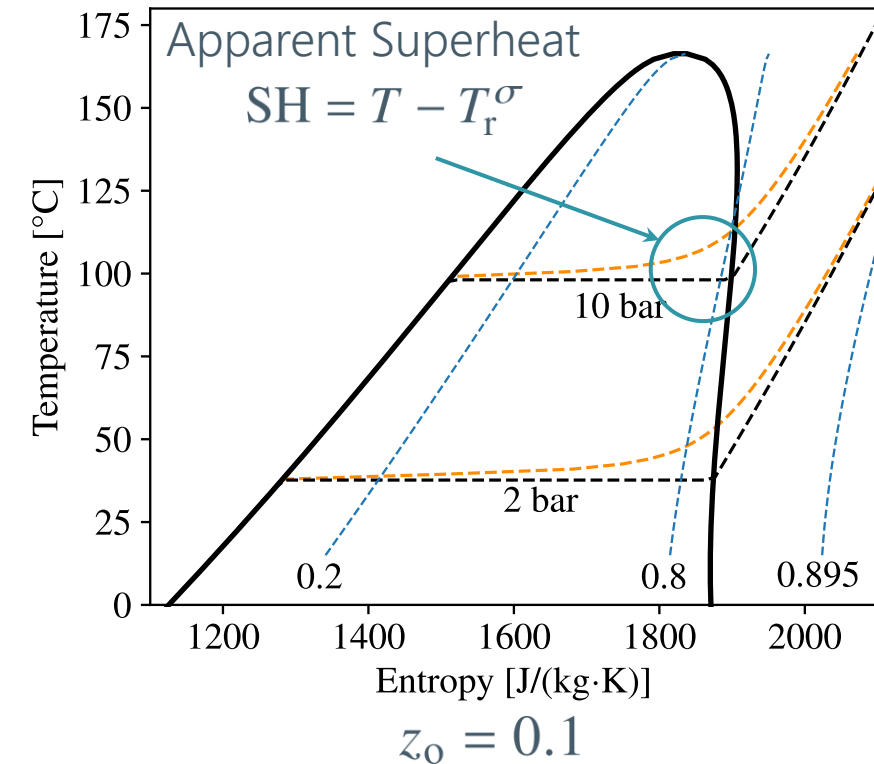
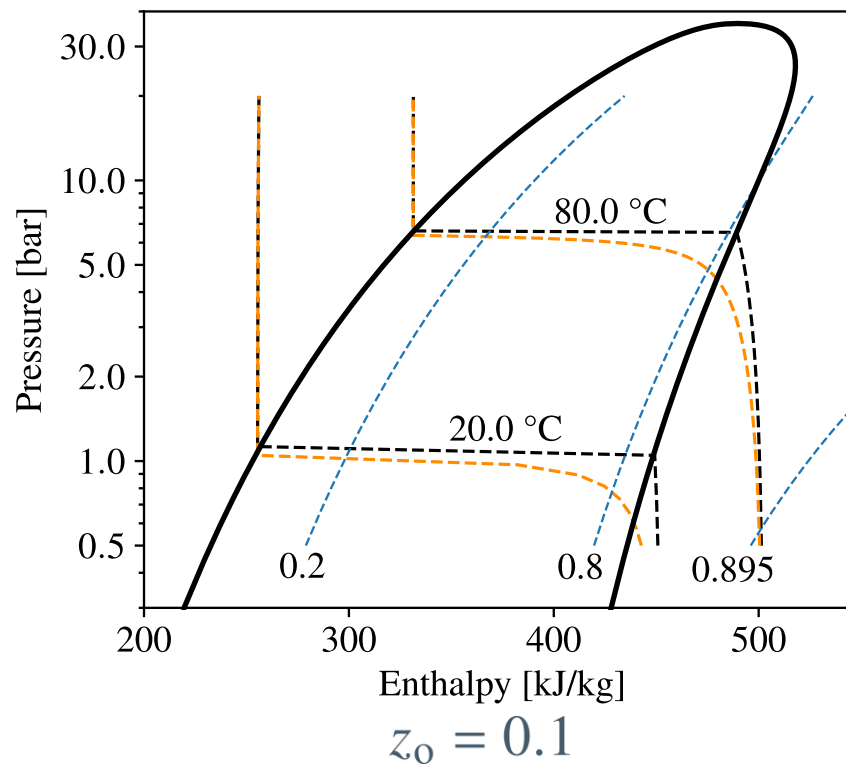
Enthalpy of an oil-refrigerant mixture

Applies also to the entropy and the internal energy

$$h_m(p, T) = \underbrace{Q h_{r,g}(p, T)}_{\text{Vapor contribution}} + \underbrace{z_o h_o(p, T)}_{\text{Oil contribution}} + \underbrace{(1 - Q - z_o) h_{r,l}^\sigma(T)}_{\text{Liquid refrigerant contribution}} + \underbrace{(1 - Q) h^E}_{\text{Excess enthalpy of the solution (Neglected)}}$$

Same reference state

- Pure ref. bell
- - - Mixt. iso-Q
- - - Mixt. iso-p
- - - Pure ref. iso-p



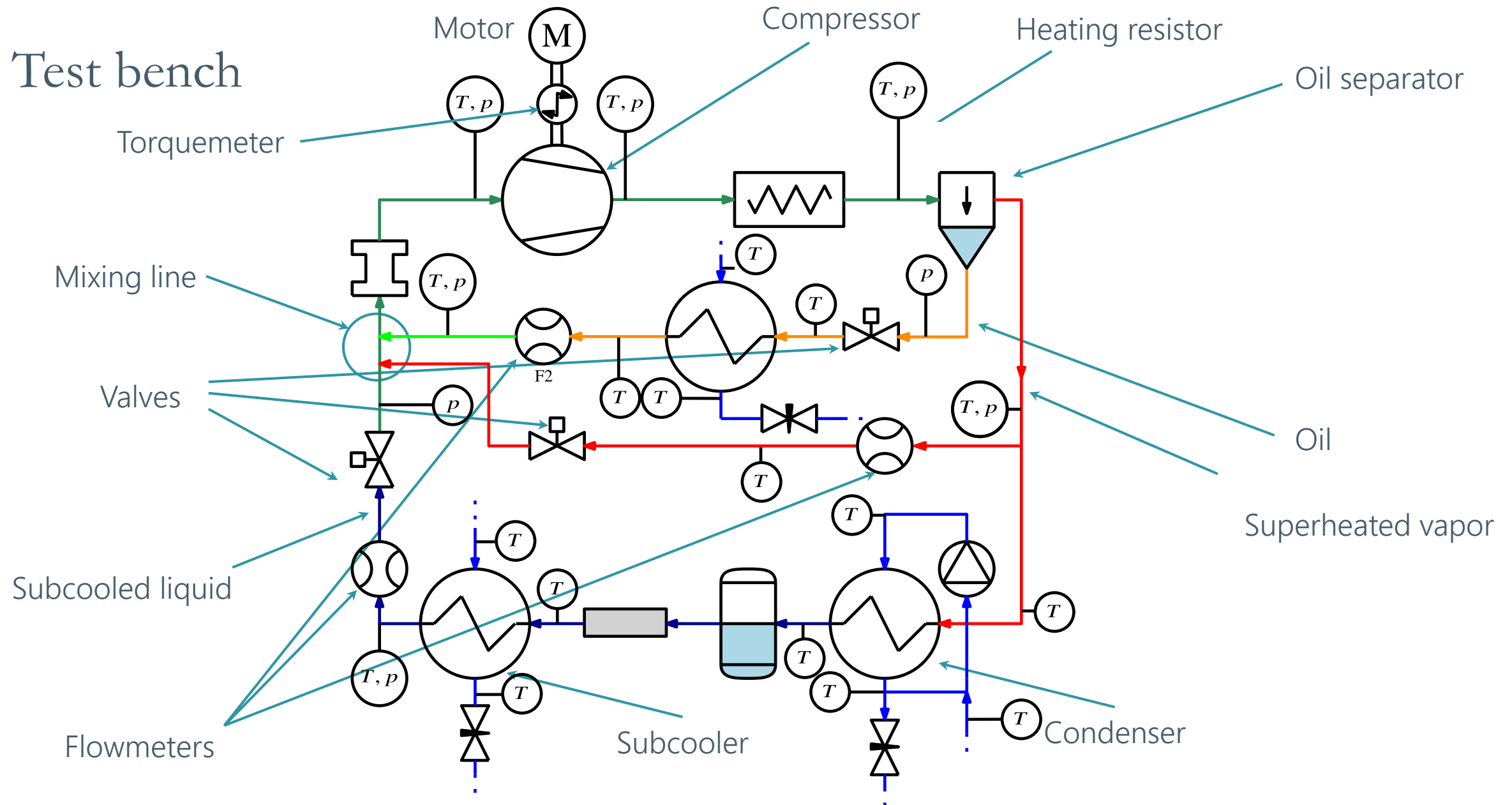
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Test bench

- Testing of compressors with two-phase flows
- Adjustable inlet pressure with heating resistor
- Adjustable pressure ratio and inlet vapor qualities with valves
- Adjustable compressor speed with a variable frequency drive
- Adjustable oil circulation ratio (OCR) with an oil separator and a control valve

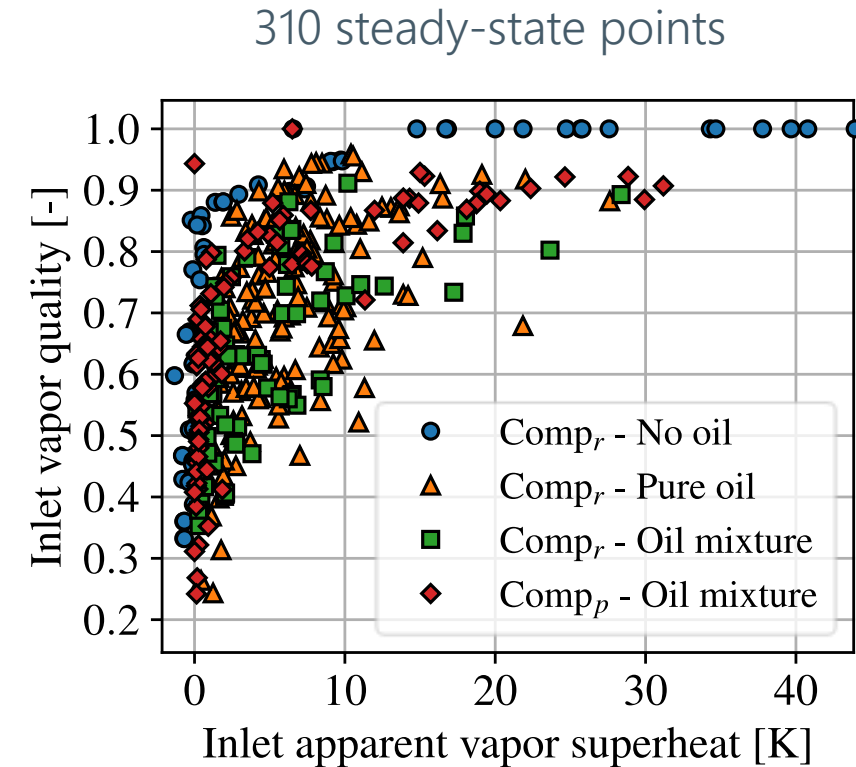
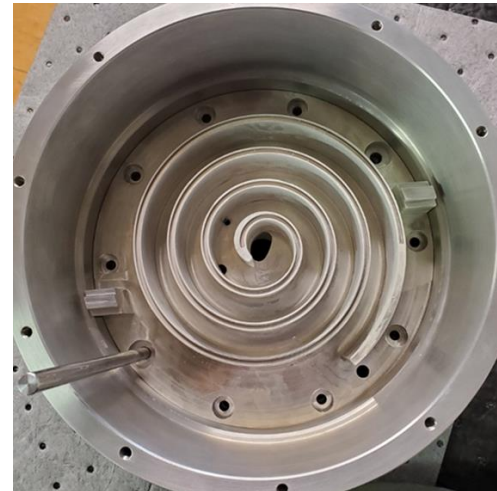


Test bench



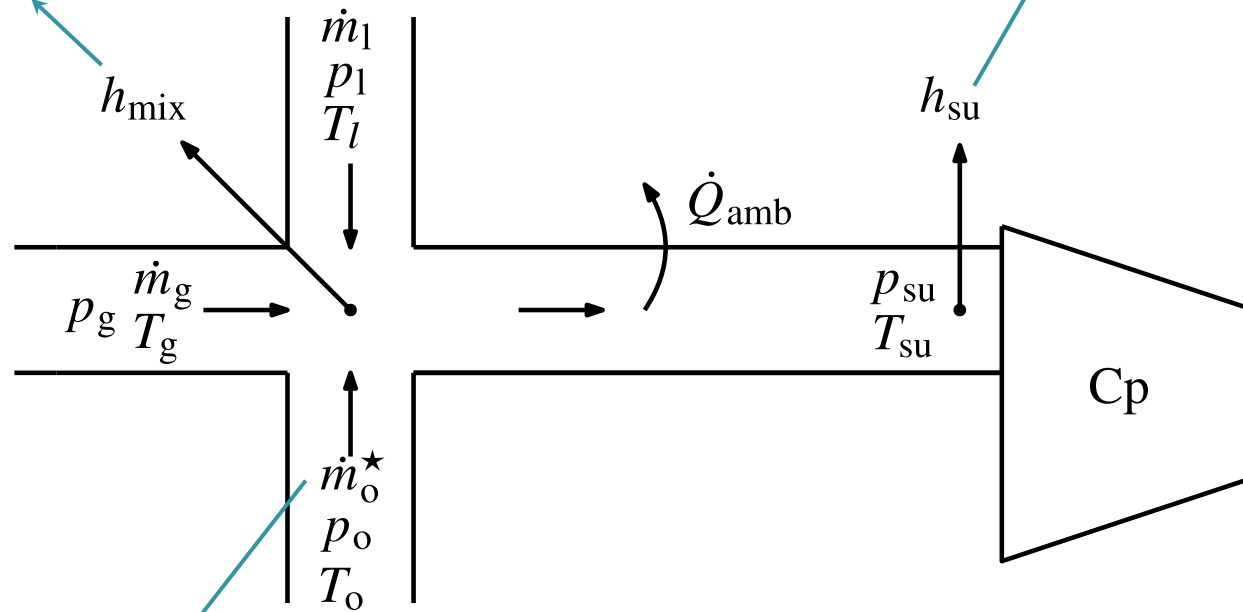
Tested compressors (open-drive scroll compressors)

	Retrofitted compressor	Lab-scale prototype
Displacement volume [cm ³]	86	200
Built-in volume ratio [-]	2.3	3
Orbiting mechanism	Balls	Oldham
Axial compliance	No	No
Radial compliance	No	Yes
Reed valve and IDV	Yes	Yes



Determination of the inlet vapor quality

$$h_{\text{mix}} = \frac{\dot{m}_g}{\dot{m}_{\text{tot}}} h_g(p_g, T_g) + z_o h_o(p_o, T_o) + \frac{\dot{m}_l}{\dot{m}_{\text{tot}}} h_{r,l}^\sigma(T_l) \quad h_{\text{su}} = Q_{\text{su}} h_{r,g}(p_{\text{su}}, T_{\text{su}}) + z_o h_o(p_{\text{su}}, T_{\text{su}}) + (1 - Q - z_o) h_{r,l}^\sigma(T_{\text{su}})$$



Without correction:

$$z_o = \frac{\dot{m}_o^*}{\dot{m}_{\text{tot}}}$$

With correction:

$$z_o = \frac{\dot{m}_o^* (1 - x_{r,o})}{\dot{m}_{\text{tot}}}$$

$f(p, T)$ before separation

M1	M2
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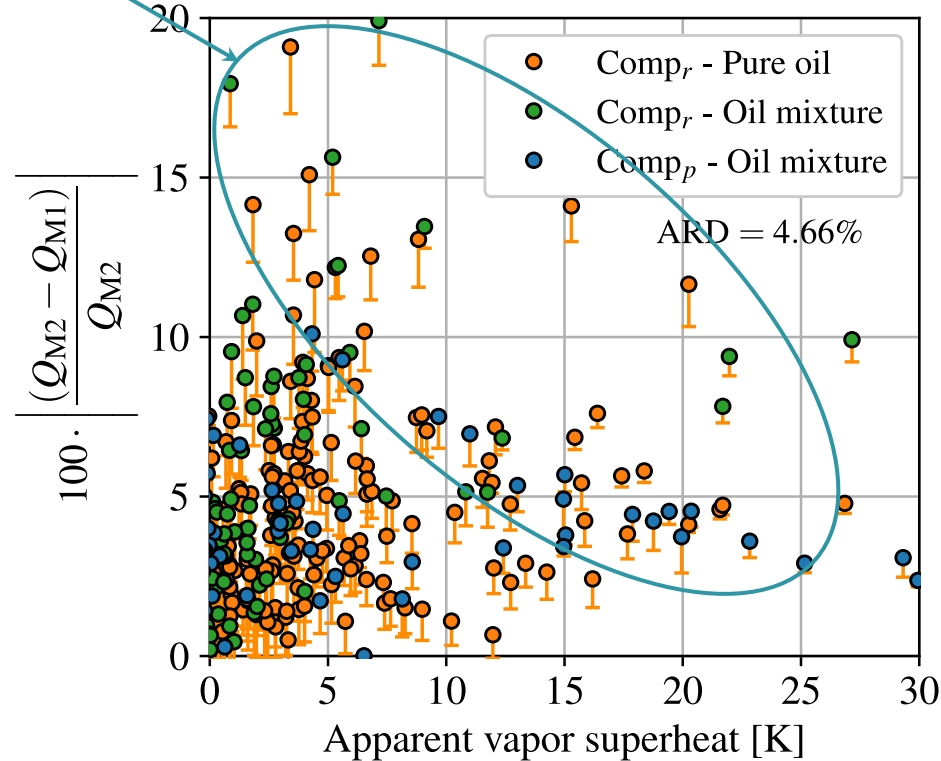
$$h_{\text{mix}} = h_{\text{su}} + \frac{\dot{Q}_{\text{amb}}}{\dot{m}_{\text{tot}}}$$

p_{su} known T_{su} known Q_{su} unknown	p_{su} known T_{su} unknown Q_{su} unknown $Q_{\text{su}} = f(z_o, p_{\text{su}}, T_{\text{su}})$
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Determination of the inlet vapor quality

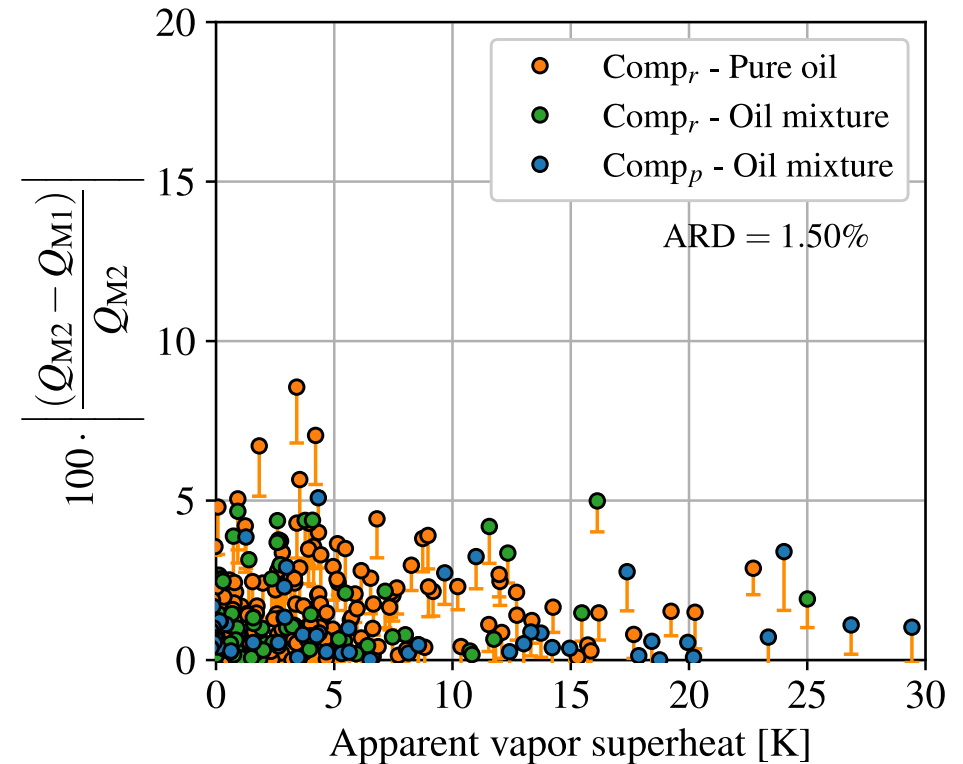
Important deviations

Without correction



Correction of the OCR
important

With correction



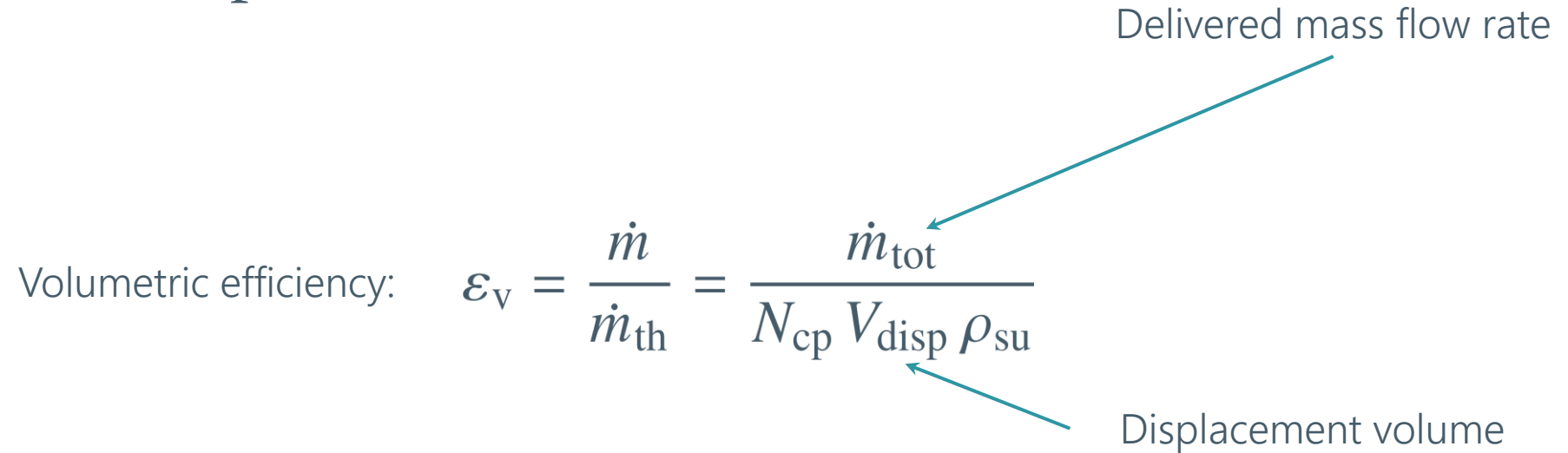
Two valid
methodologies

Definition of the performance indicators

Volumetric efficiency: $\mathcal{E}_v = \frac{\dot{m}}{\dot{m}_{th}} = \frac{\dot{m}_{tot}}{N_{cp} V_{disp} \rho_{su}}$

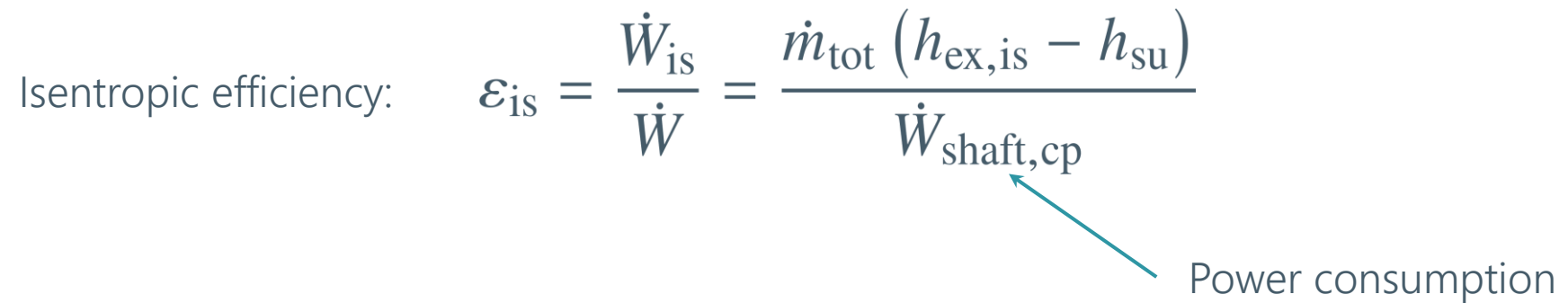
Delivered mass flow rate

Displacement volume

The diagram shows the equation for volumetric efficiency. A teal arrow points from the text 'Delivered mass flow rate' to the symbol $\dot{m}_{tot}$ in the numerator. Another teal arrow points from the text 'Displacement volume' to the term V_{disp} in the denominator.

Isentropic efficiency: $\mathcal{E}_{is} = \frac{\dot{W}_{is}}{\dot{W}} = \frac{\dot{m}_{tot} (h_{ex,is} - h_{su})}{\dot{W}_{shaft,cp}}$

Power consumption

The diagram shows the equation for isentropic efficiency. A teal arrow points from the text 'Power consumption' to the symbol $\dot{W}_{shaft,cp}$ in the denominator.

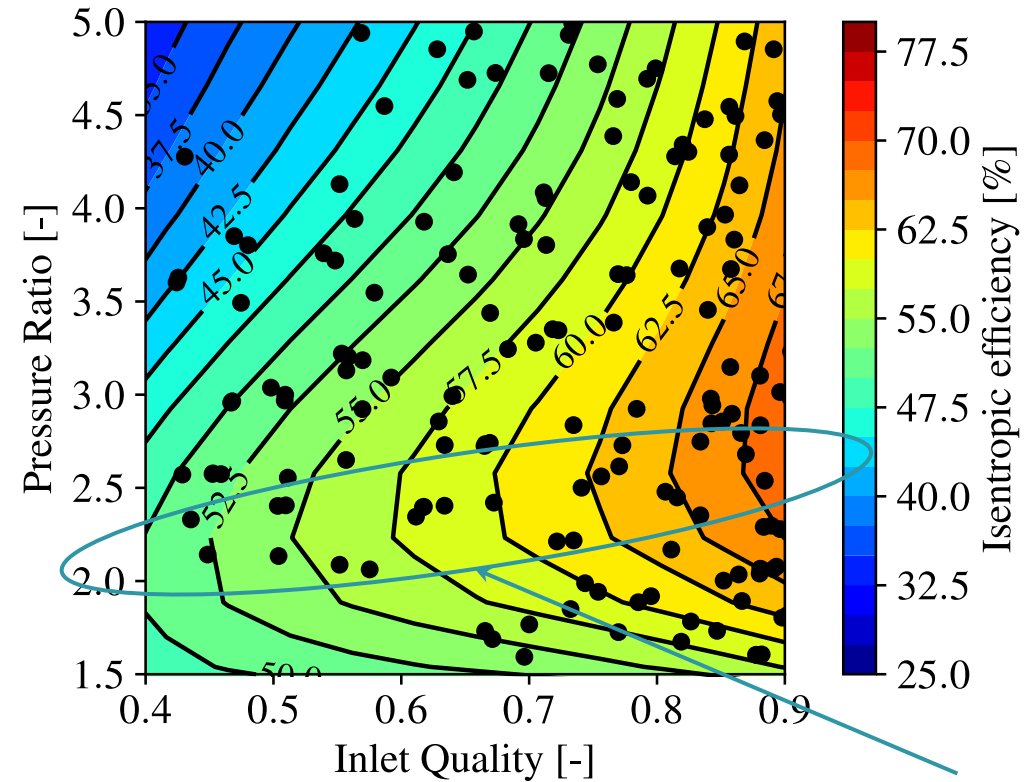
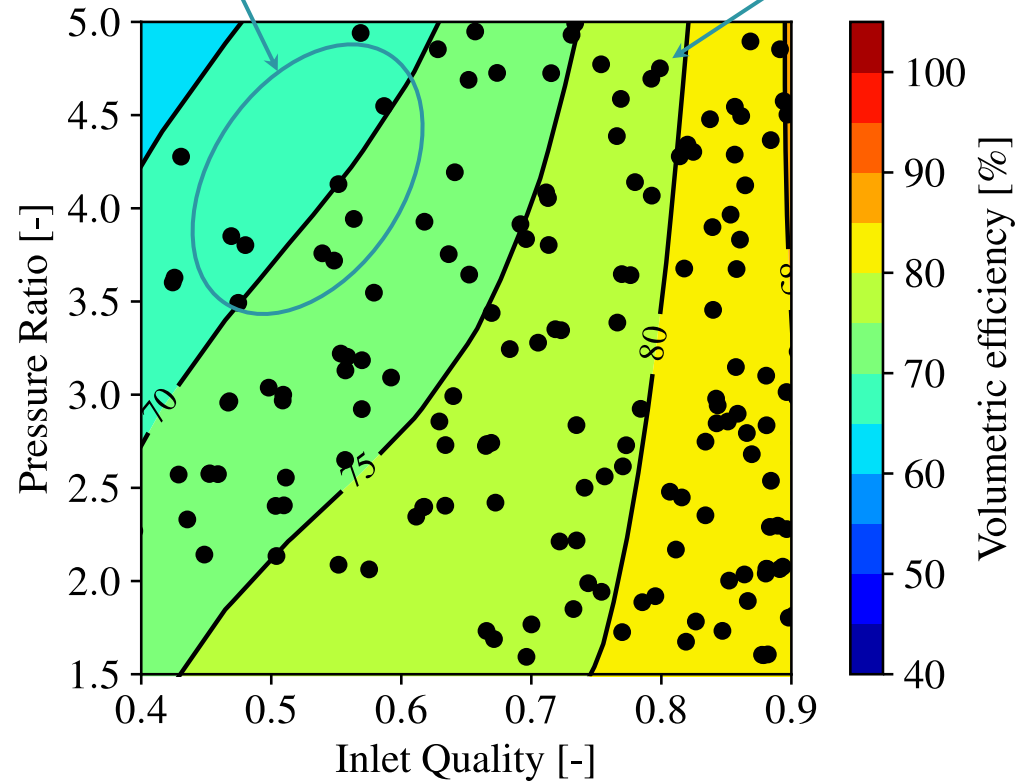
Retrofitted compressor results

Important decrease at high pressure ratios and low vapor qualities

Data points at different:

- Inlet pressures
- Compressor speeds
- OCR

Gaussian process interpolation tool

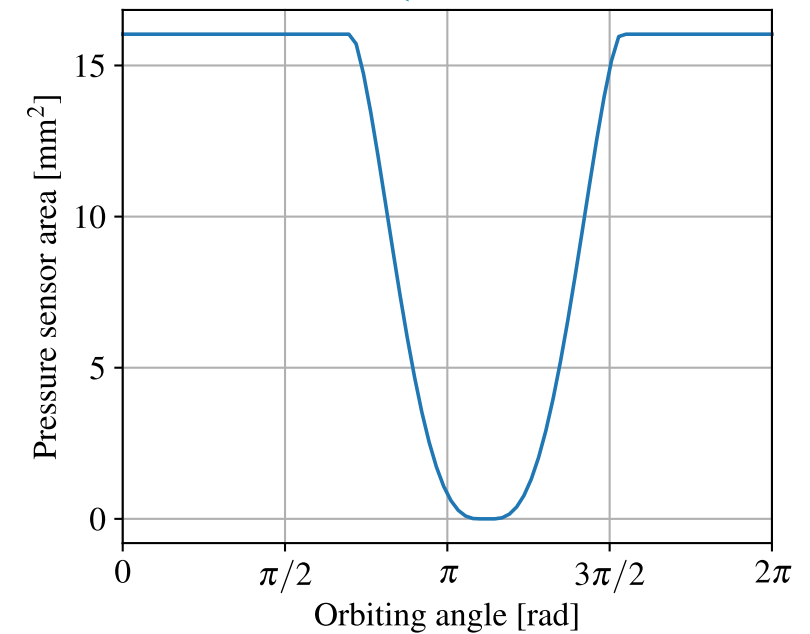
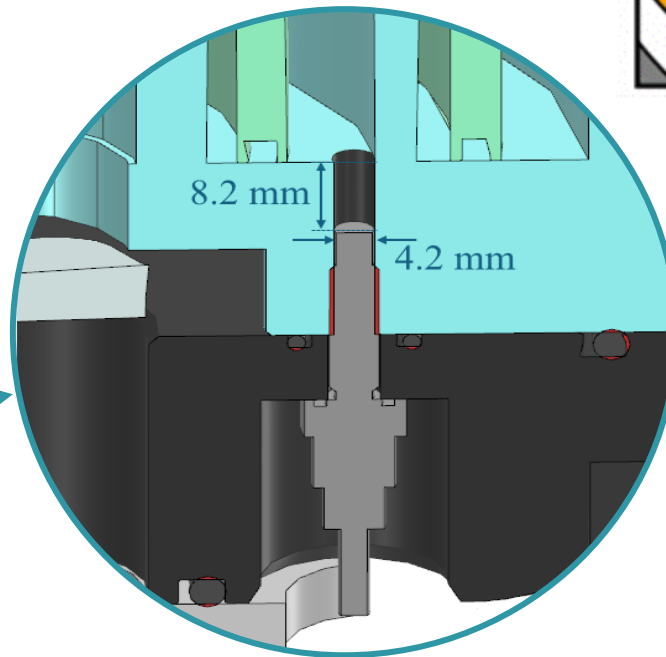
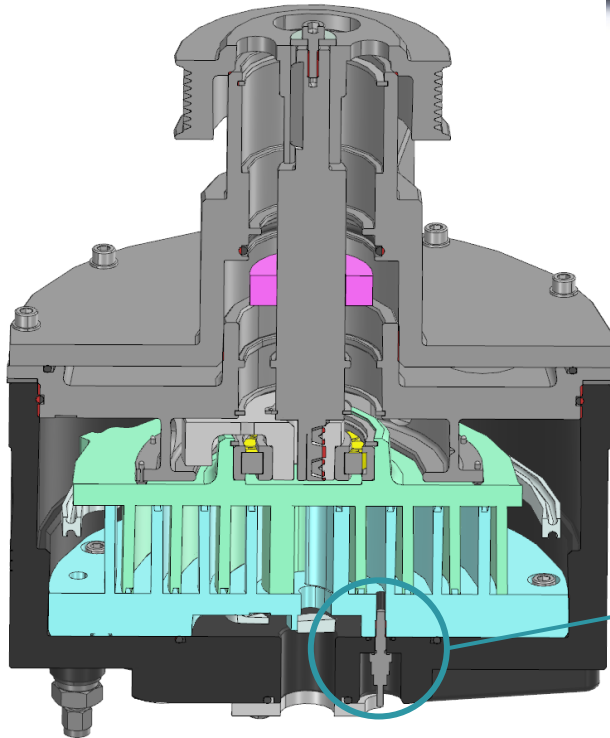
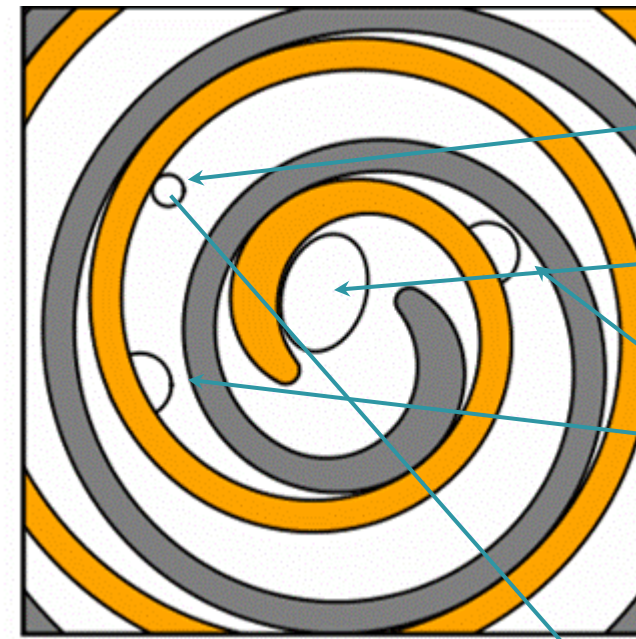


$p_{su} = 1.5$ [bar] $OCR = 5$ [%] $N = 2500$ [RPM]

Ideal pressure ratio shifted towards lower value

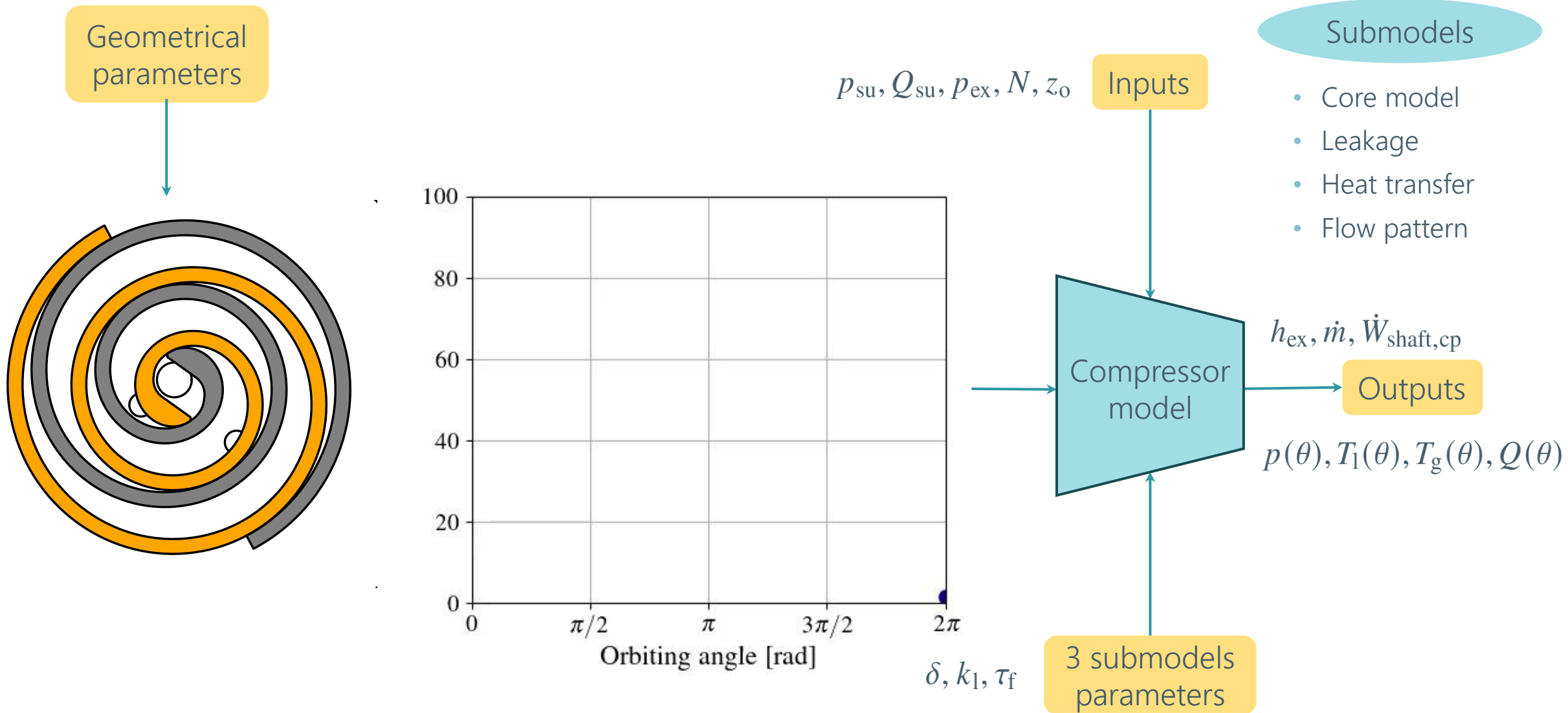
Dynamic pressure sensor

Kulite pressure sensor



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Two-phase scroll compressor deterministic model



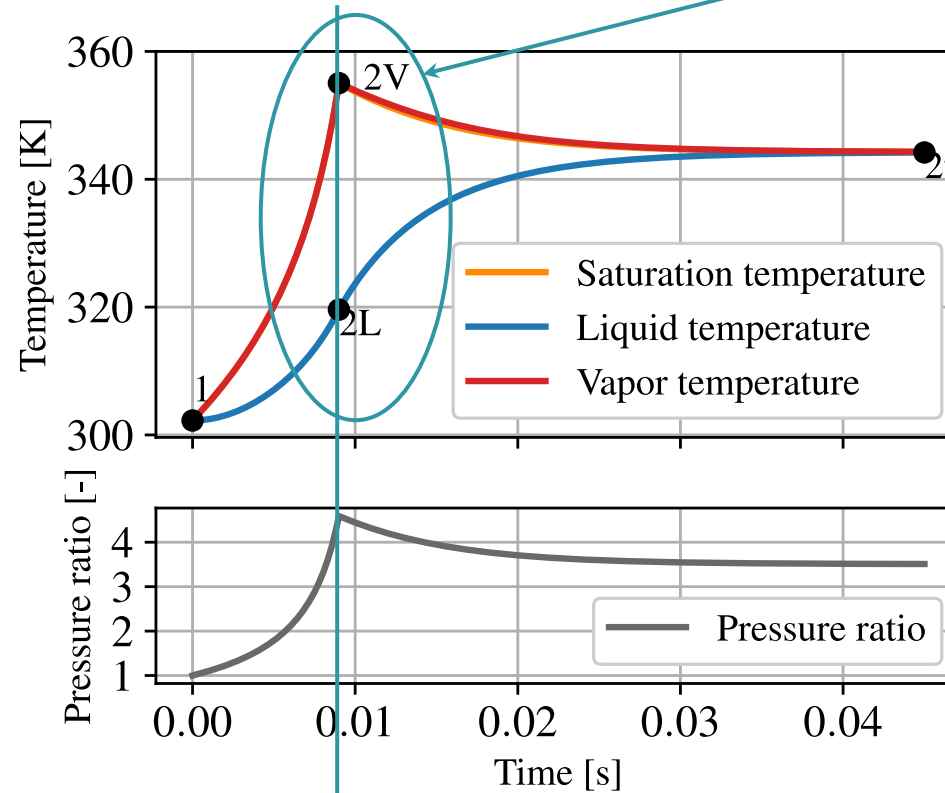
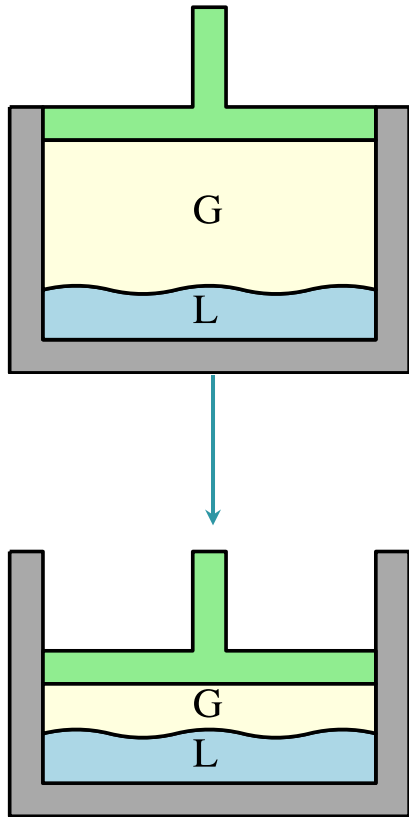
The diagram illustrates a closed system containing three phases: Vapor (G), Liquid (L), and Oil. The system is bounded by a thick black line. The Vapor (G) phase is yellow, Liquid (L) is light blue, and Oil is orange. The Oil phase is enclosed by a dashed black line. The following variables are shown:

- $\Sigma_i \dot{m}_{r,g}^i$: Mass flow rate out of the Vapor phase.
- $\dot{Q}_g$: Heat flow out of the Vapor phase.
- $\frac{dV_{CV}}{d\theta}$: Work flow out of the system.
- T_g : Temperature of the Vapor phase.
- $\dot{m}_{gl}$: Mass flow rate from Vapor to Liquid.
- $\dot{\Phi}_{gl}$: Energy flow from Vapor to Liquid.
- T_l : Temperature of the Liquid phase.
- $\Sigma_i \dot{m}_{r,l}^i$: Mass flow rate out of the Liquid phase.
- $\dot{Q}_l$: Heat flow out of the Liquid phase.
- $\Sigma_i \dot{m}_o^i$: Mass flow rate out of the Oil phase.



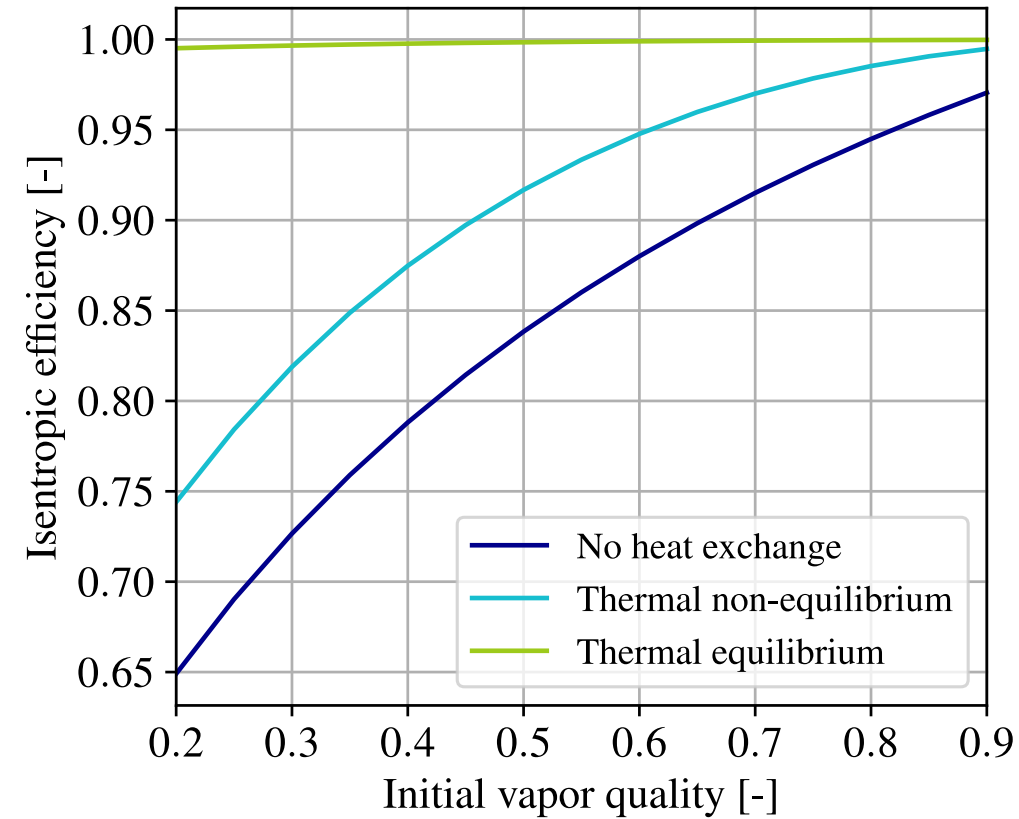
Heat exchanges generate irreversibilities

Core model

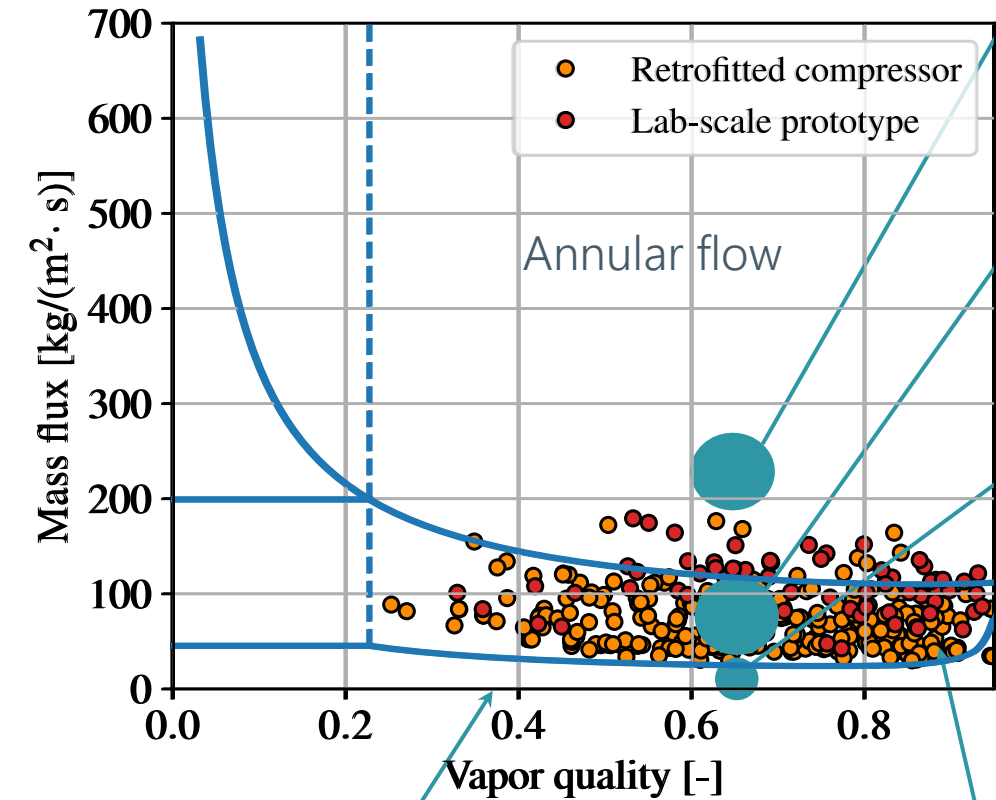


Volume reduction phase

Resting phase

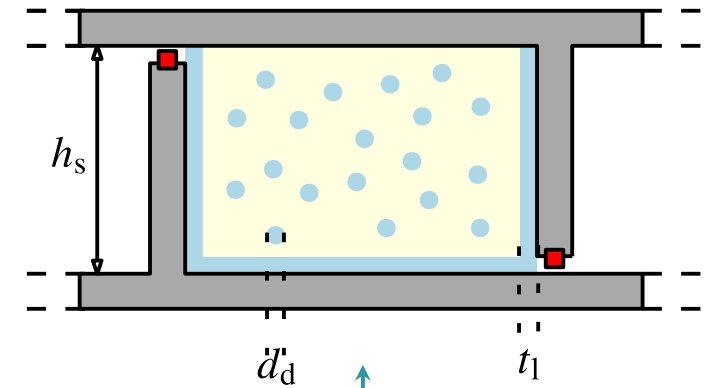


Flow pattern and liquid distribution

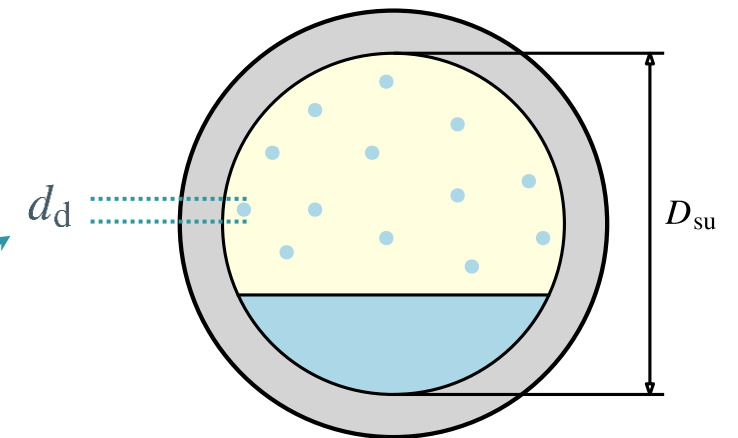


Entrainment ratio:

$$\psi = \frac{\dot{m}_{l,g}}{\dot{m}_{l,tot}}$$

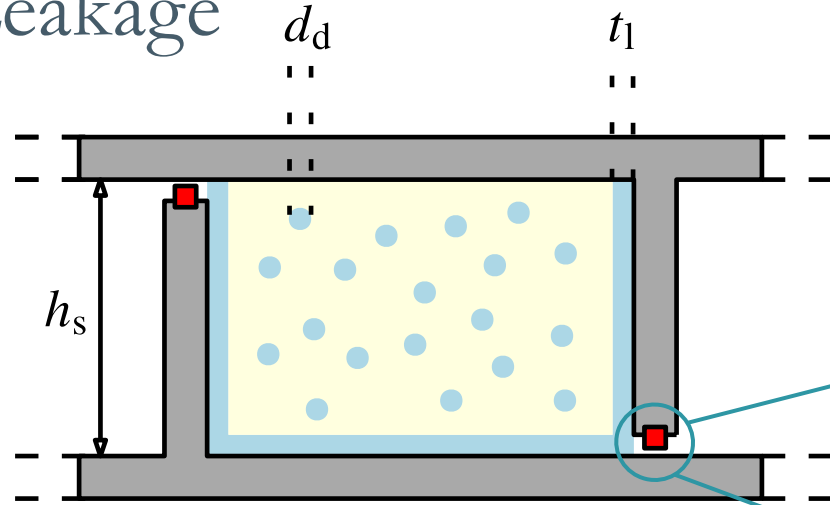


Similar cross-section areas



Supply pipe

Leakage



2 parameters

Height ratio:

$$r_h = \min \left(1, \frac{t_l \cdot k_l}{\delta} \right)$$

Flow area liquid phase:

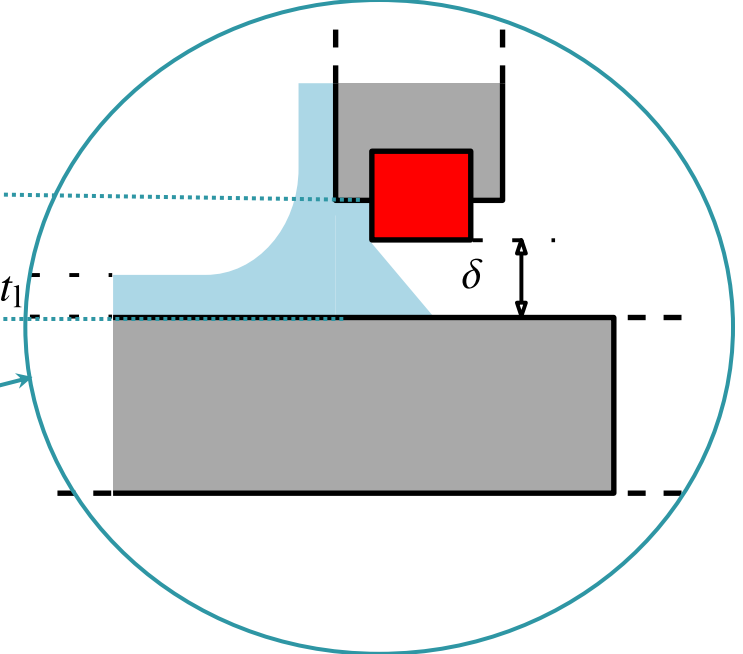
$$A_l = A \cdot r_h$$

Flow area vapor phase:

$$A_g = A(1 - r_h)$$

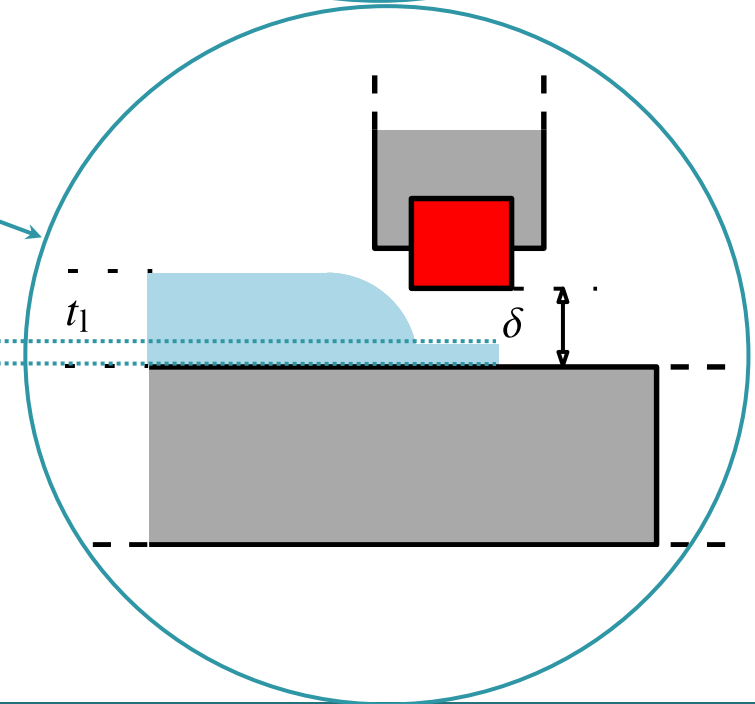
$$t_l \cdot k_l$$

$$r_h = 1$$

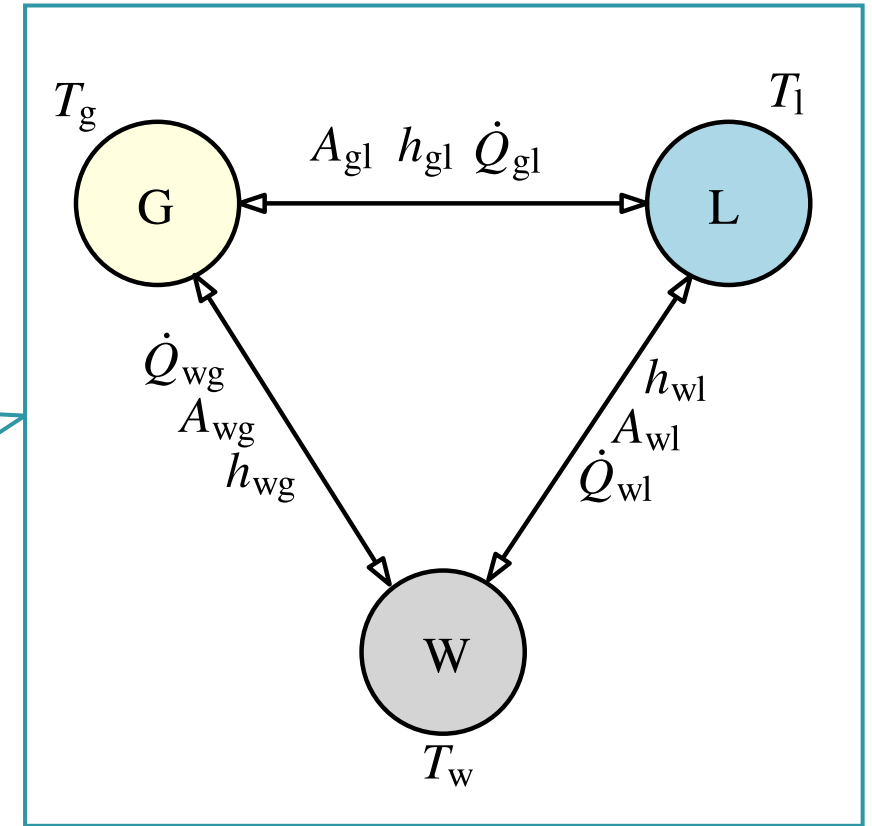
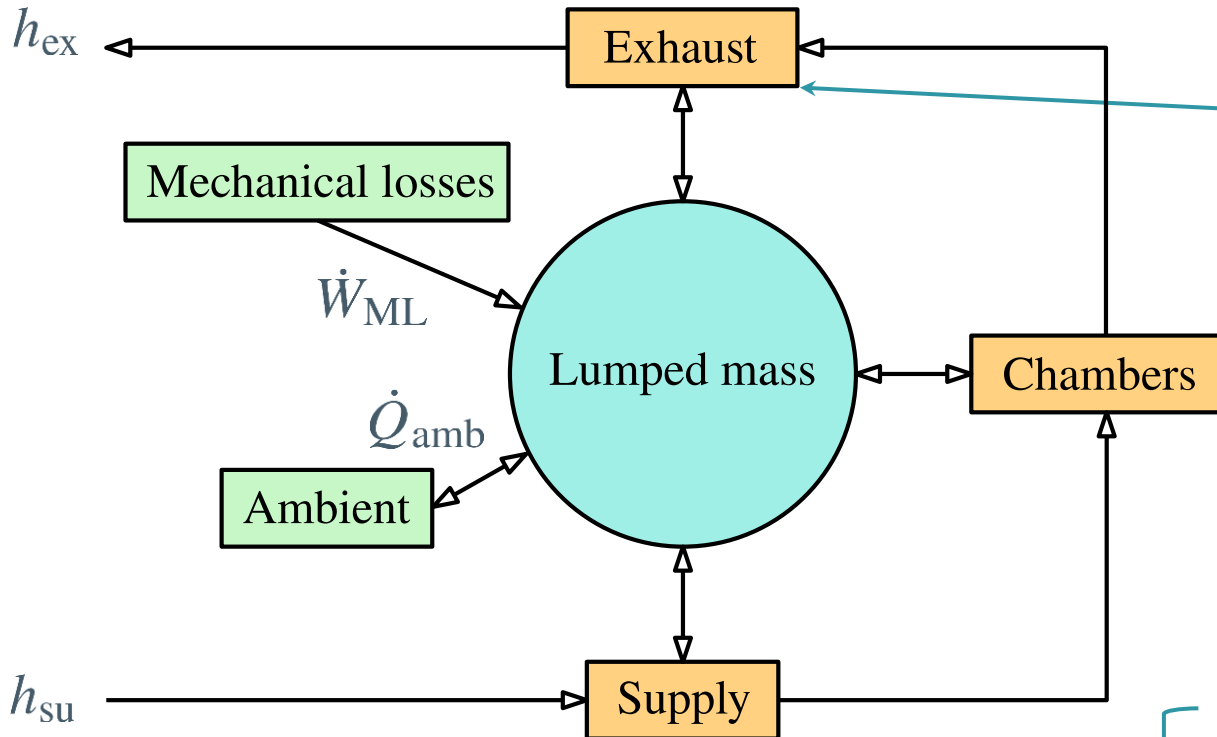


$$t_l \cdot k_l$$

$$r_h \approx 0.3$$



Heat transfer



Heat exchange coefficients defined based on correlations

Results of the model:

- Retrofitted compressor at 1000 RPM: $A_{gl} \cdot h_{gl} \approx 10 \text{ W/K}$
- Prototype at 5000 RPM: $A_{gl} \cdot h_{gl} \approx 10000 \text{ W/K}$

Mechanical power losses

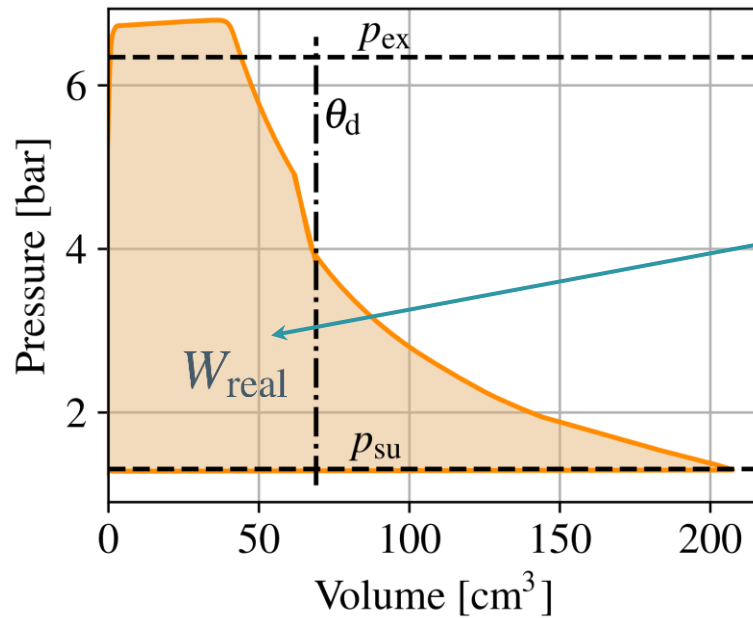
Non-zero mechanical losses

Constant friction torque

Mechanical losses in friction parts:

$$\dot{W}_{\text{ML}} = \dot{W}_{\text{ML},0} + \alpha \cdot \dot{W}_{\text{in}}$$

$$\dot{W}_{\text{ML}} = \tau_f \cdot \omega$$



- Indicated power

$$\dot{W}_{\text{in}} = N \cdot W_{\text{real}}$$

Shaft power: $\dot{W}_{\text{shaft}} = \dot{m}(h_{\text{ex}} - h_{\text{su}}) - \dot{Q}_{\text{amb}}$

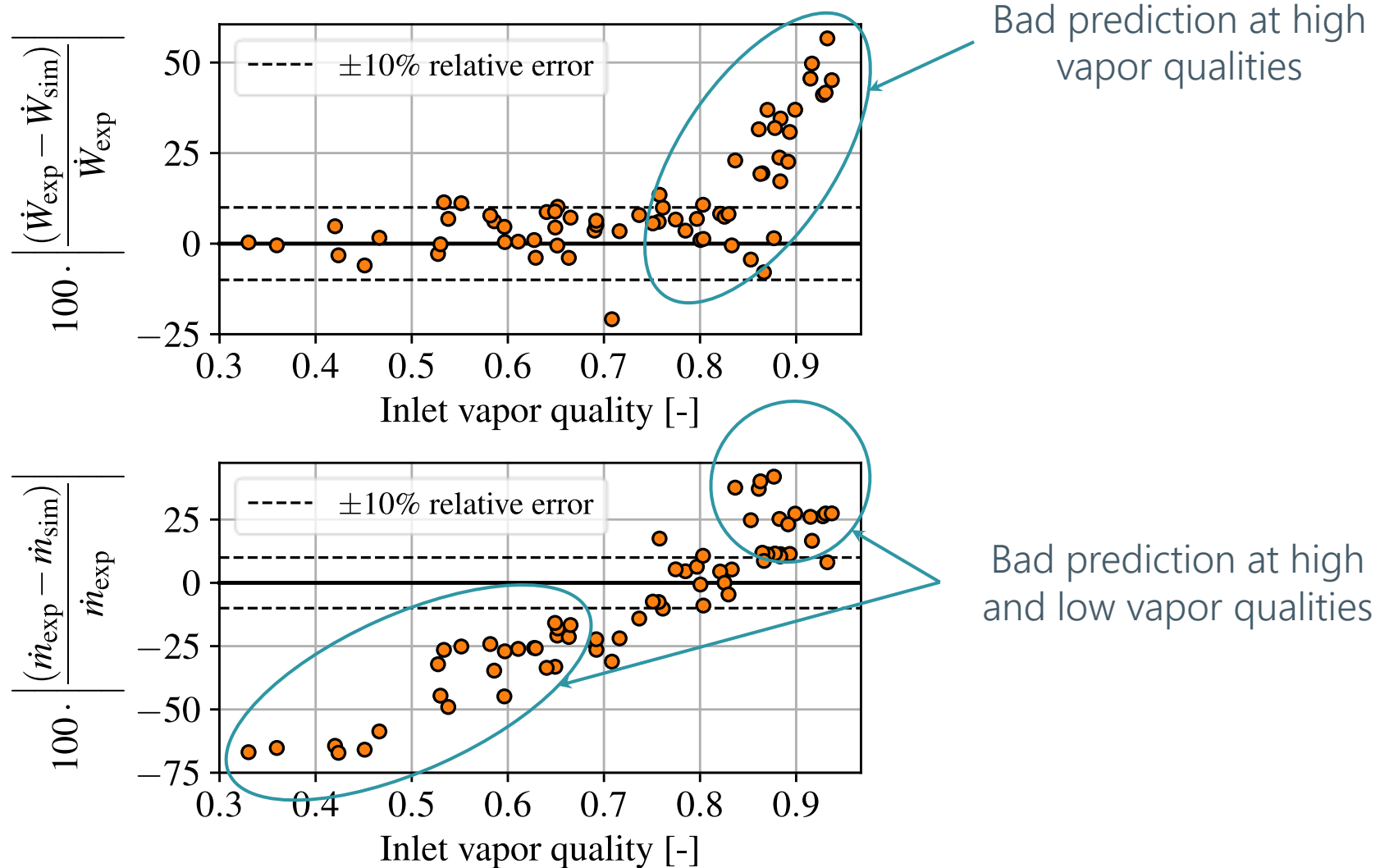
Hydraulic losses
investigated but neglected

Validation – fixed parameters - prototype

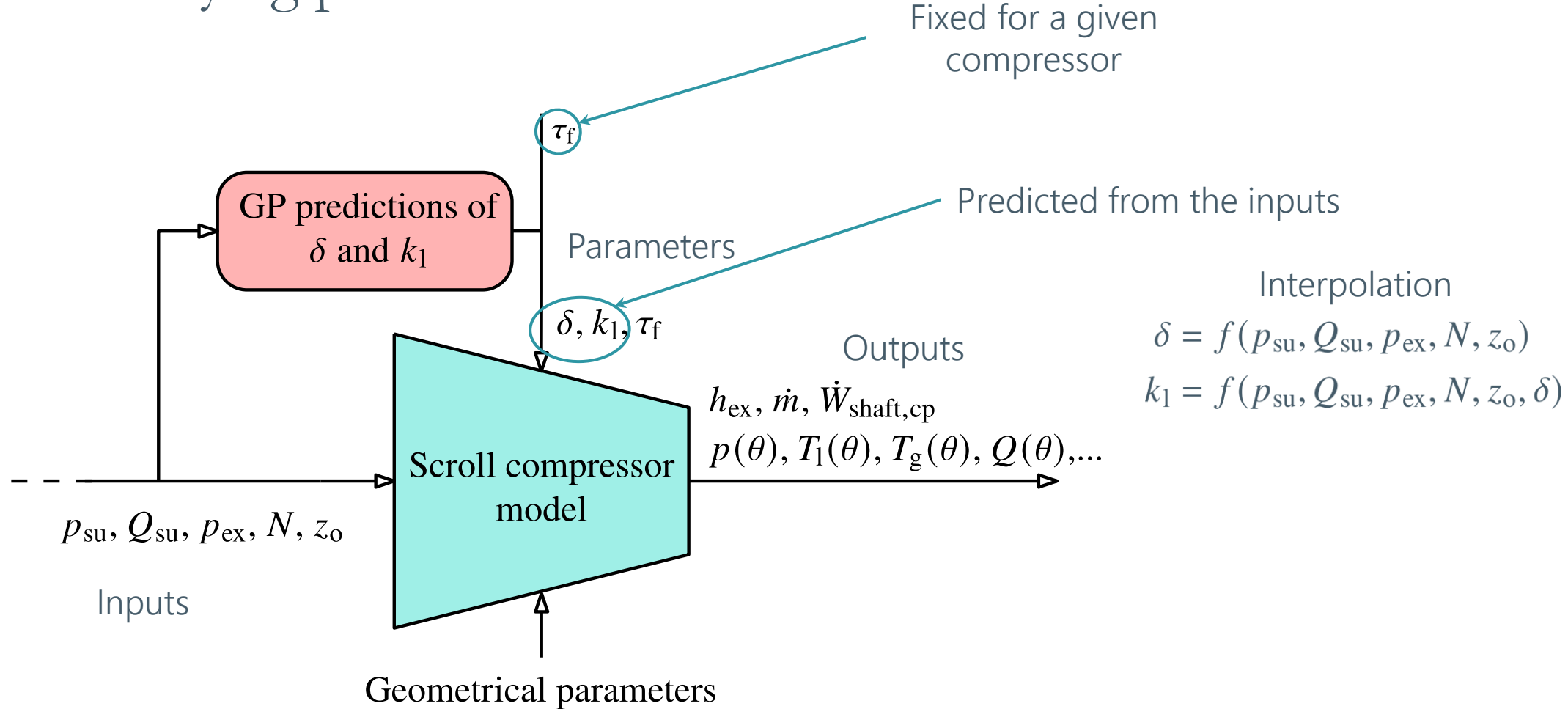
$$\begin{array}{lcl} \delta & \longleftrightarrow & \frac{\dot{m}_{\text{exp}} - \dot{m}_{\text{sim}}}{\dot{m}_{\text{exp}}} \\ k_1 & \longleftrightarrow & \frac{\bar{p}_{\text{exp}} - \bar{p}_{\text{sim}}}{\bar{p}_{\text{exp}}} \\ \tau_f & \longleftrightarrow & \frac{\dot{W}_{\text{exp}} - \dot{W}_{\text{sim}}}{\dot{W}_{\text{exp}}} \end{array}$$

Prototype parameters

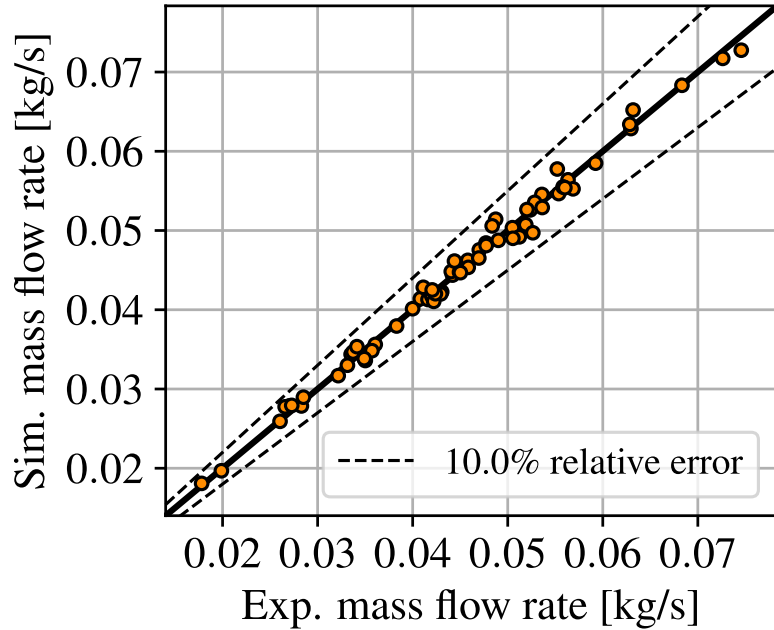
δ [μm]	30.22
k_1 [-]	2.38
τ_f [Nm]	1.6



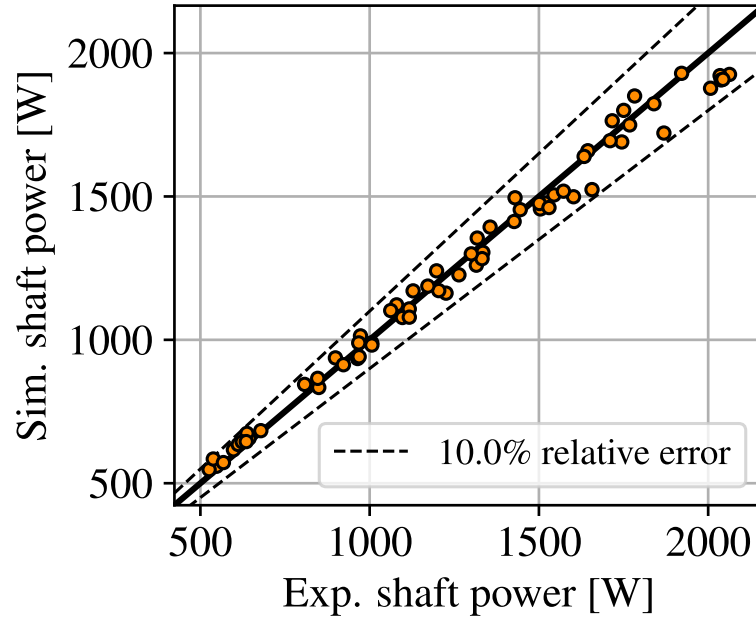
Validation – varying parameters



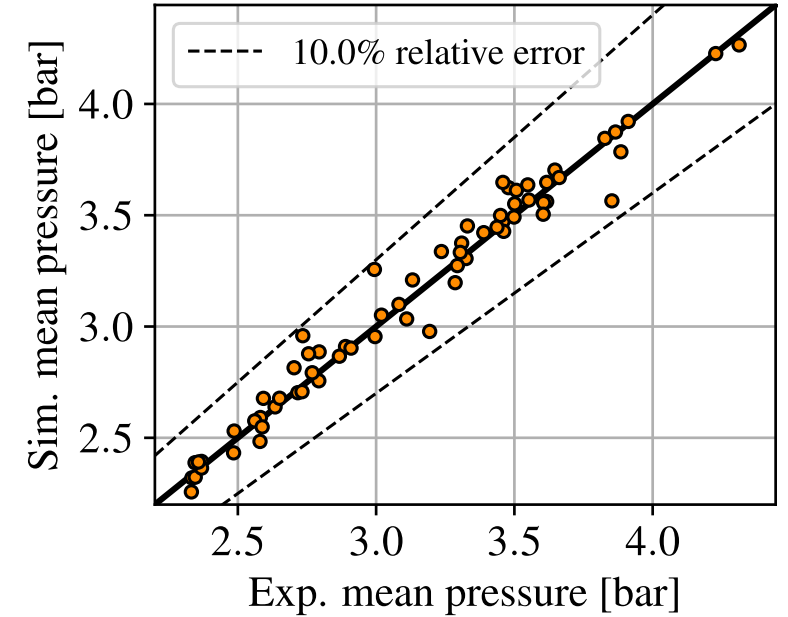
Validation results: prototype



ARD = 2.11%

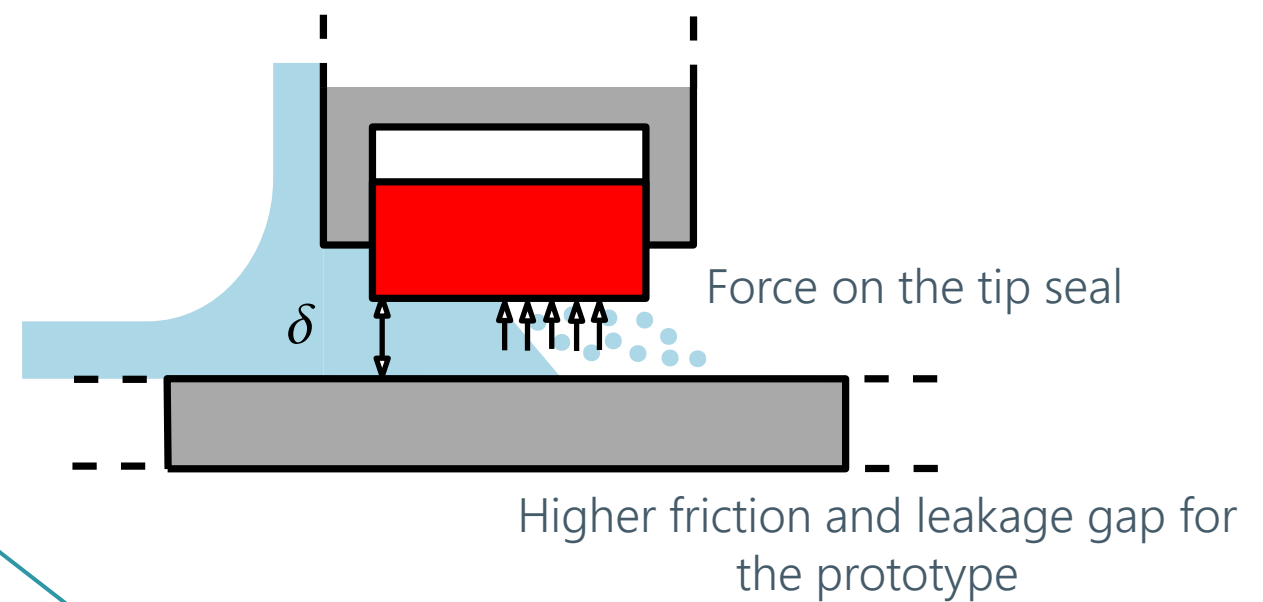


ARD = 3.41%



ARD = 2.51%

Validated models



Same law for both machines

	Retrofitted compressor	Lab-scale prototype
Displacement volume [cm ³]	86	200
Built-in volume ratio [-]	2.3	3
Orbiting mechanism	Balls	Oldham
Friction torque [Nm]	0.42	1.6
Leakage gap δ [μm]	[4-38]	[6-78]
Liquid-thickness correction factor k_1 [-]	[0.1-6.8]	[0.1-7.9]

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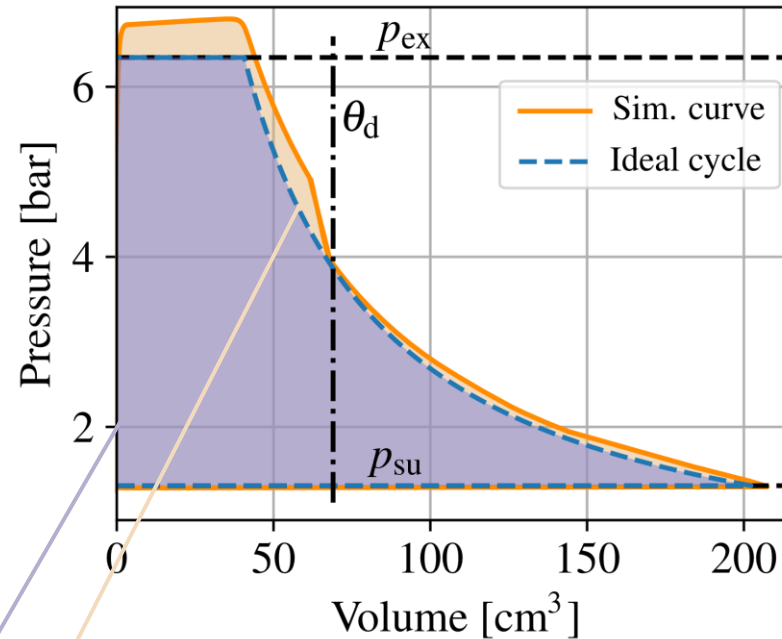
Performance indicators

$$\varepsilon_v = \frac{\dot{m}}{\dot{m}_{th}} = \frac{\dot{m}}{\dot{m}_{in}} \cdot \frac{\dot{m}_{in}}{\dot{m}_{th}} = \varepsilon_{v,lk} \cdot \varepsilon_{v,PT}$$

Mass flow rate without leakage

Leakage-related volumetric efficiency

Heat transfer and pressure drops-related volumetric efficiency



$$\varepsilon_{is} = \frac{\dot{m} (h_{ex,is} - h_{su})}{\dot{W}_{shaft,cp}} = \frac{\dot{m}}{\dot{m}_{th}} \cdot \frac{\dot{W}_{in}}{\dot{W}_{shaft,cp}} \cdot \frac{W_{ideal}}{W_{real}} = \varepsilon_{v,lk} \cdot \varepsilon_{v,PT} \cdot \eta_m \cdot \varepsilon_{in}$$

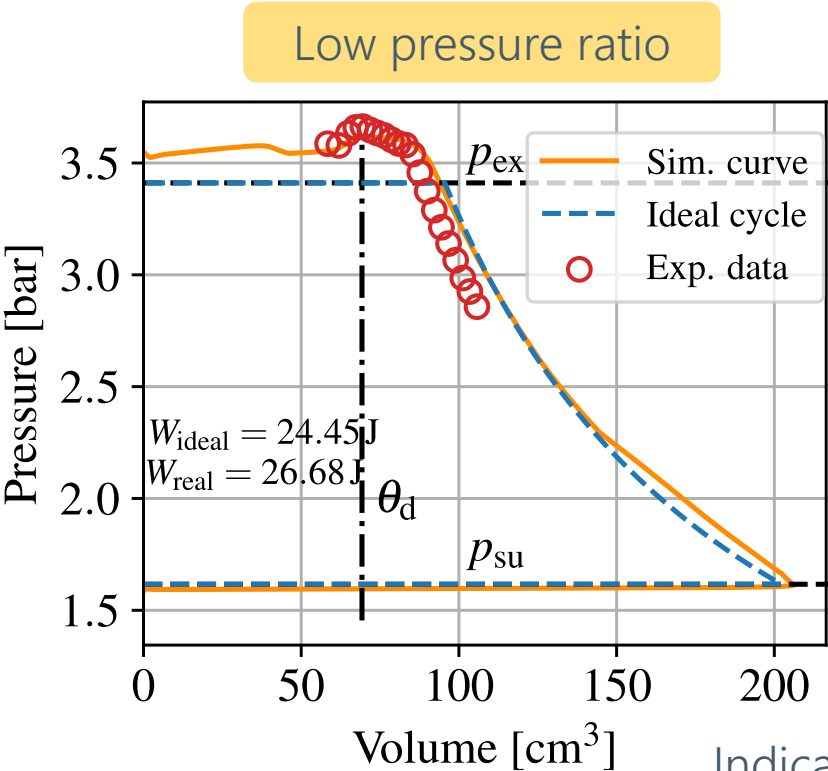
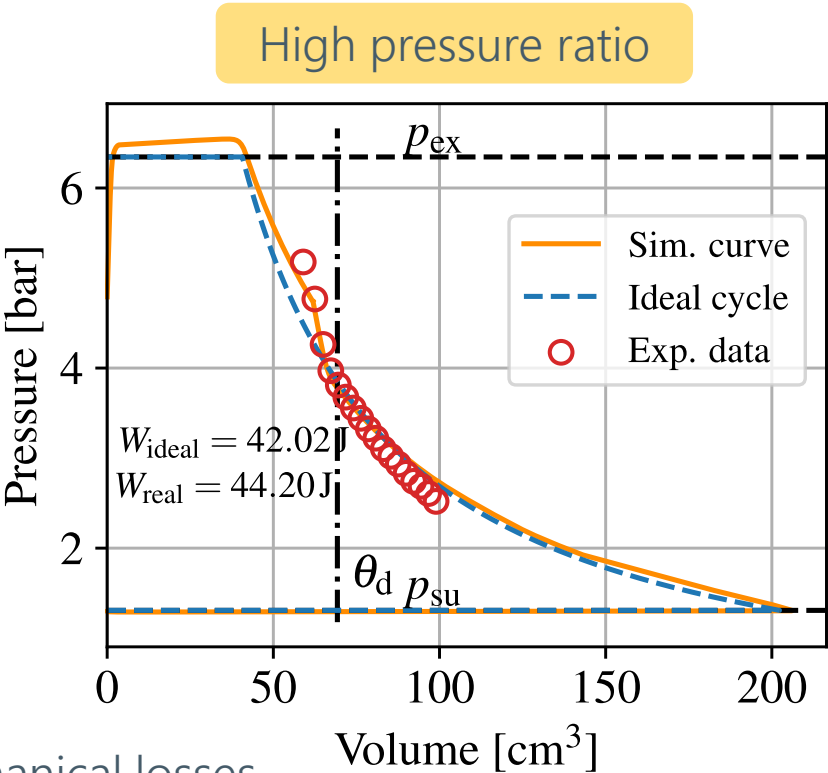
$N \cdot W_{real}$

Indicated power

Mechanical efficiency

Indicated efficiency

Pressure-volume diagrams at high vapor qualities

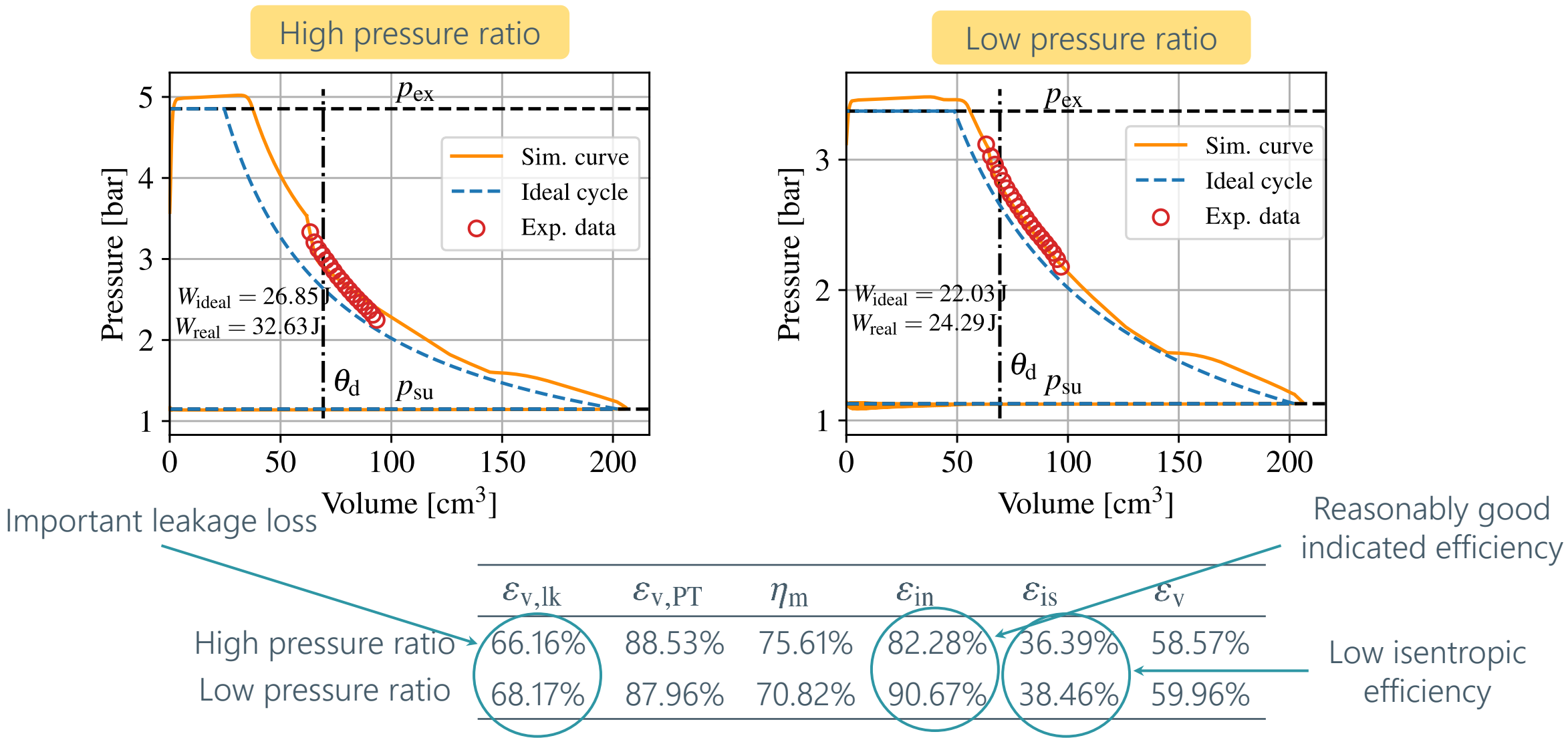


High mechanical losses

Indicated efficiency lower for overcompression

	$\epsilon_{v, \text{lk}}$	$\epsilon_{v, \text{PT}}$	η_m	ϵ_{in}	ϵ_{is}	ϵ_v
High pressure ratio	89.18%	96.66%	79.88%	95.02%	65.34%	86.20%
Low pressure ratio	90.82%	96.13%	76.72%	91.65%	61.31%	87.3%

Pressure-volume diagrams at low vapor qualities



Important leakage loss

Reasonably good indicated efficiency

High pressure ratio
Low pressure ratio

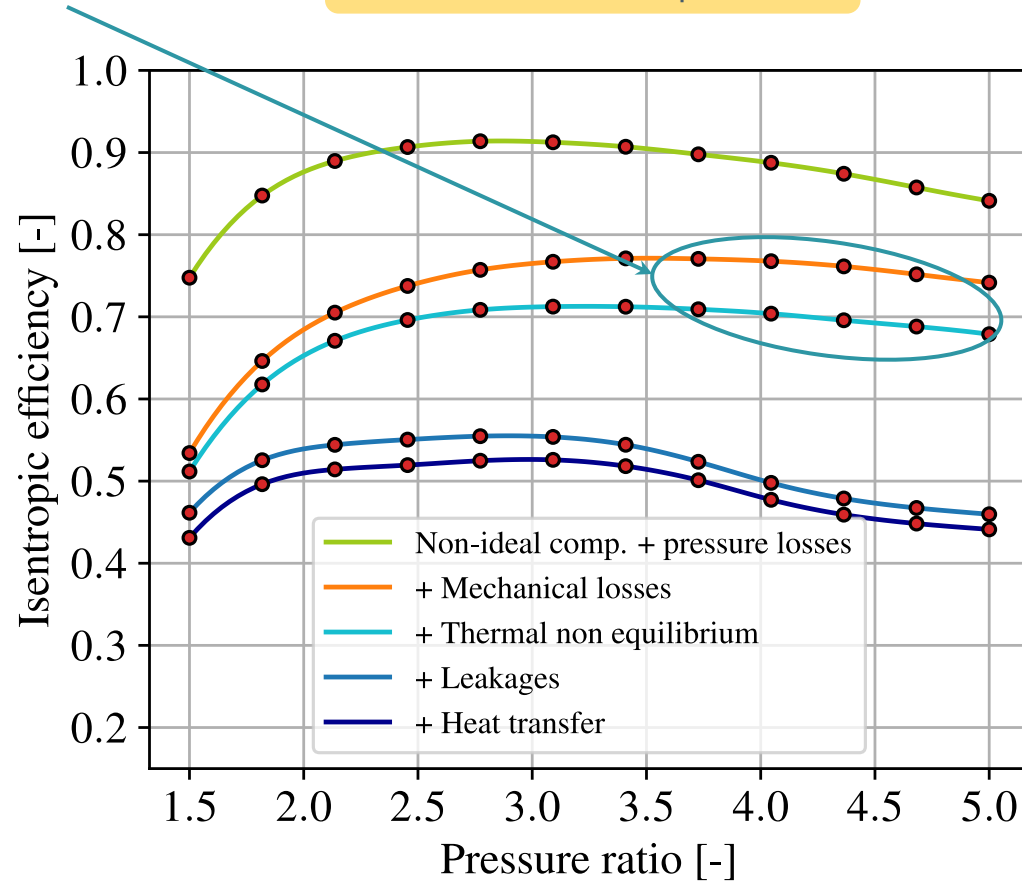
$\epsilon_{v, lk}$	$\epsilon_{v, PT}$	η_m	ϵ_{in}	ϵ_{is}	ϵ_v
66.16%	88.53%	75.61%	82.28%	36.39%	58.57%
68.17%	87.96%	70.82%	90.67%	38.46%	59.96%

Low isentropic efficiency

Sensitivity analysis: isentropic efficiencies

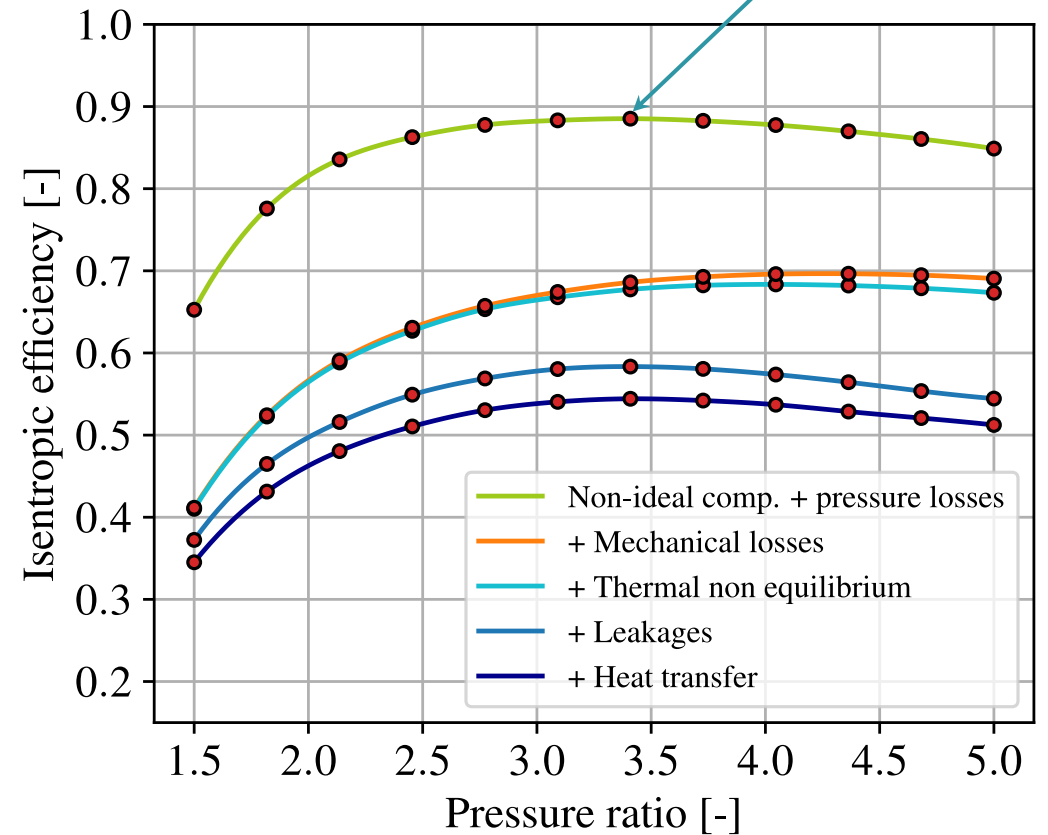
High impact of the thermal non-equilibrium

Retrofitted compressor



High discharge pressure losses

Lab-scale prototype



$$Q_{su} = 0.5 [-]$$

$$N = 4000 [\text{RPM}]$$

$$p_{su} = 1.5 [\text{bar}]$$

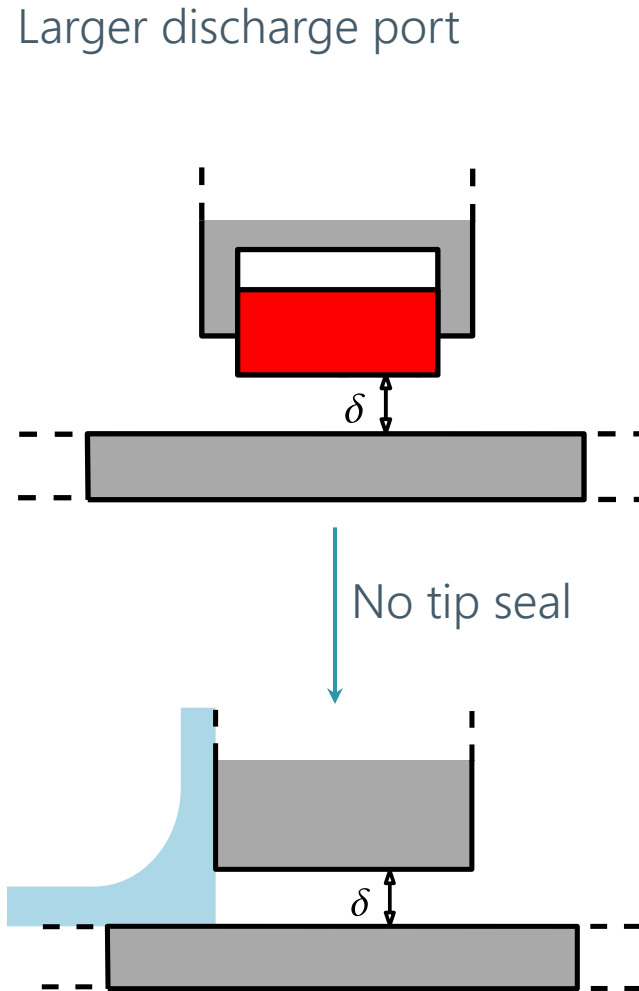
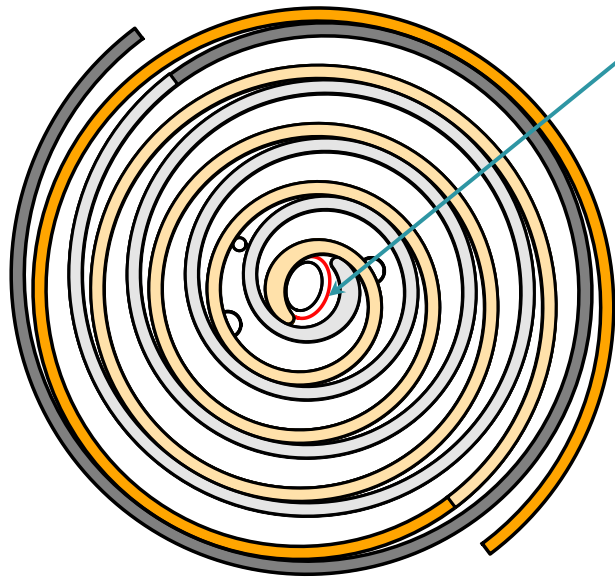
$$OCR = 5 [\%]$$

Conclusion: optimal compressor for two-phase regimes

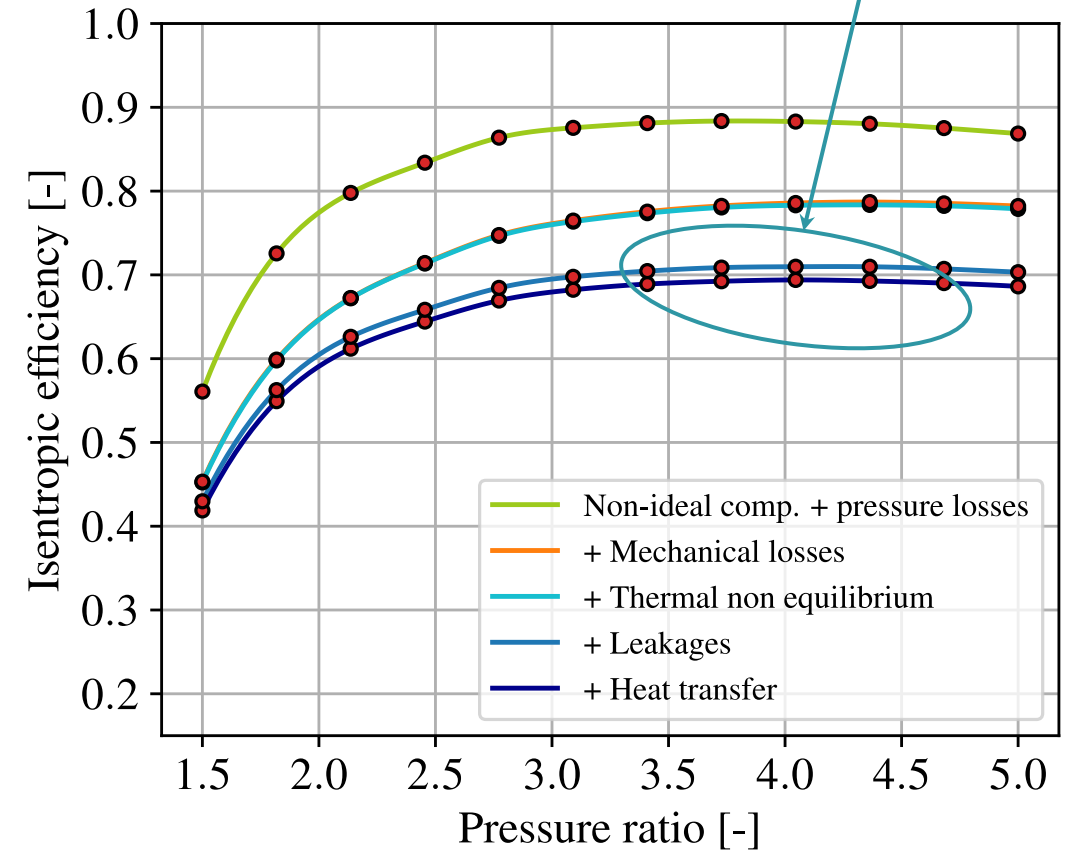
Sources of losses	Solution		Retrofitted compressor	Lab-scale prototype	Optimized design
Undercompression losses	Higher built-in volume ratio	$r_{v,in} [-]$	2.3	3	4
High pressure losses	Larger ports	$A_{dis} [mm^2]$	95	160	260
High leakage	Better sealing mechanism	$\bar{\delta} [\mu m]$	21	42	20
High friction	Enhanced orbiting mechanism	$\tau_f [Nm]$	0.42	1.6	0.8
Thermal non-equilibrium	Larger size	$V_{disp} [cm^3]$	86	200	220

Conclusion: optimal compressor for two-phase regimes

70% of isentropic efficiency achievable



Ball mechanism



$$Q_{su} = 0.5 [-]$$

$$N = 4000 [\text{RPM}]$$

Conclusion: Contributions and Perspectives of the Thesis

Contributions

- Modeling overview of oil-refrigerant mixtures
- Experimental testing of two scroll compressors under two-phase
- Deterministic model for two-phase compression under thermal non-equilibrium and special leakage management
- Validation of the model based on three measured outputs
- Optimized design of scroll compressor for two-phase flows

Perspectives

- Further investigations of thermodynamic cycles with two-phase compression
- Testing and modeling of an hermetic compressor
- Lifetime experimental study, is an oil-free machine possible?
- Manufacturing and testing the compressor proposed design
- Additional pressure sensors for an even better model validation

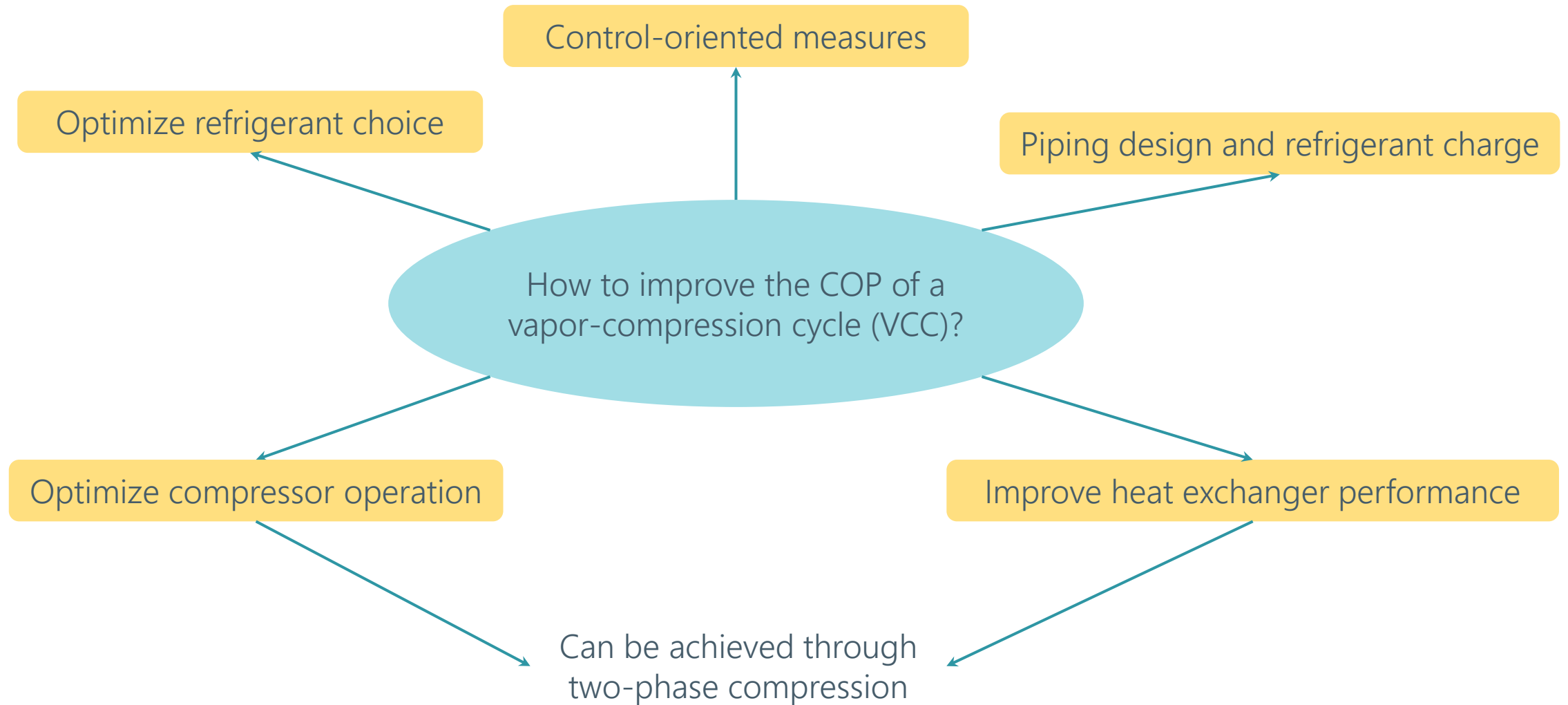


Thank you!
Questions?

Back-up slides

1. Introduction and Motivations

Improvement of vapor-compression cycles



Positive-displacement compressors vs two-phase

Reciprocating

- High risk of liquid compression
- Studied for expansion

Rotary

- Avoided at start-up
- High risk of liquid compression

Screw

- Leakage reduction and cooling effect
- Compressor performance
- Compression-resorption heat pump

Scroll

- Air conditioning at high ambient temperature
- Sealing and cooling effect?
- Two-phase compression in cycles?

Research gap

"Contribution to the improvement of scroll compressor operating under two-phase conditions"

"How can two-phase compression in scroll compressor be made efficient enough to be of practical interest?"

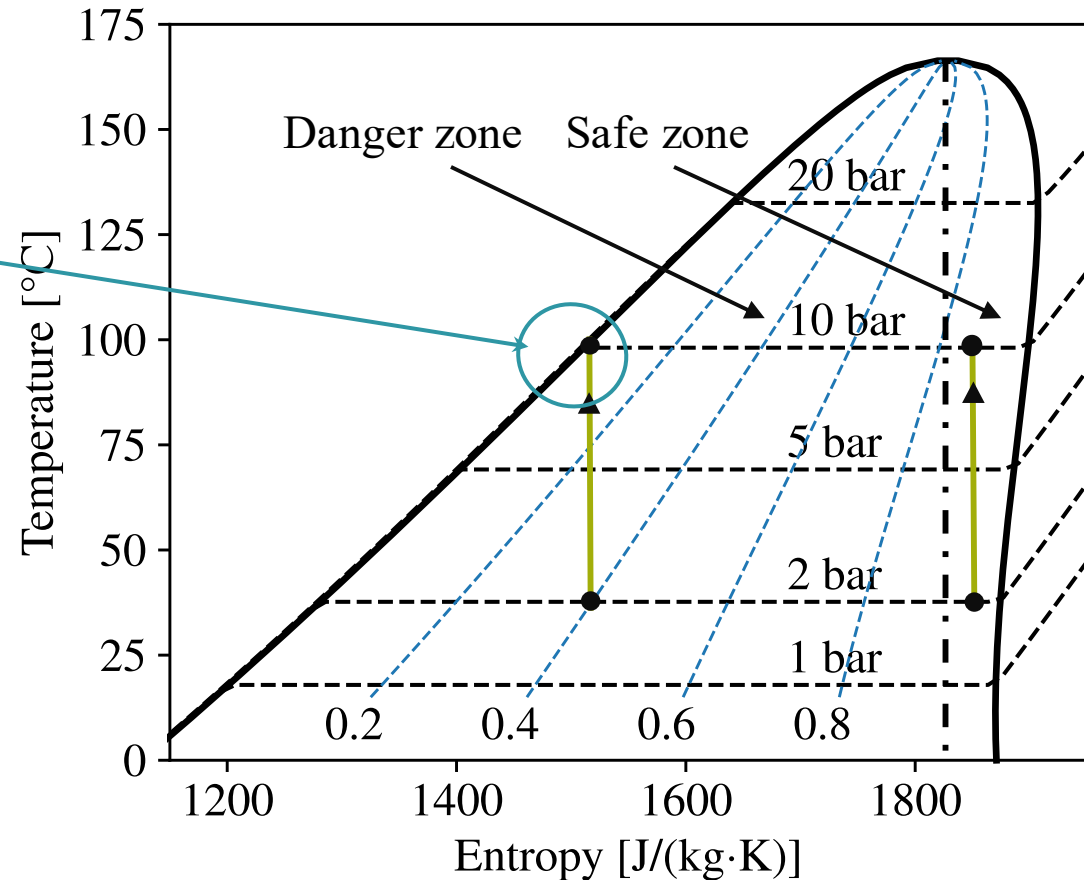
Two-phase in positive-displacement compressors

Why is two-phase usually undesired in compressors?

- Danger for the compressor (liquid slugging phenomenon)
- Operating condition usually not controlled
- Cannot work with all positive-displacement compressors
- Two-phase is difficult to quantify

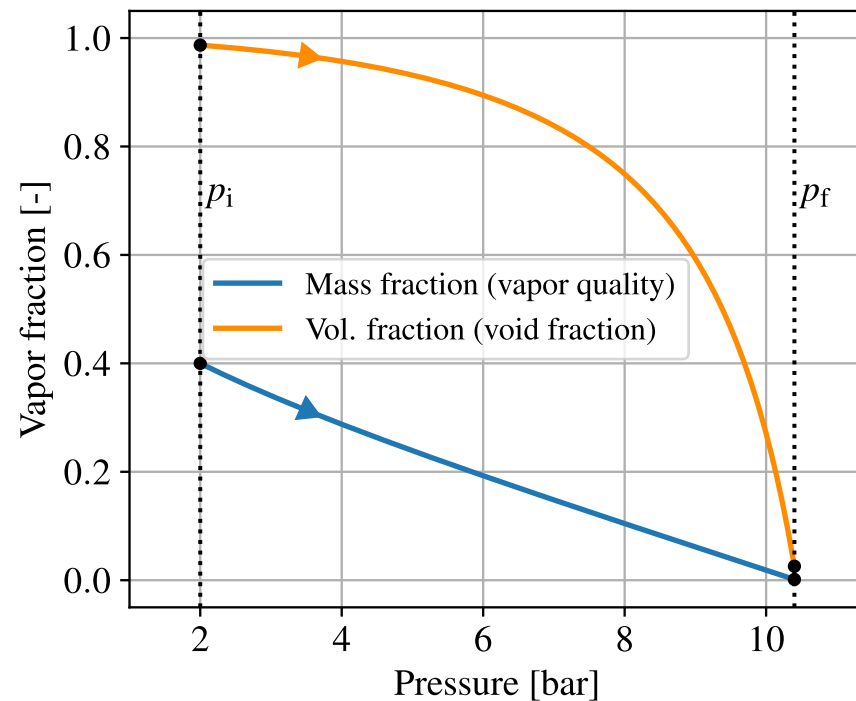
Definition of the vapor quality:

$$Q = \frac{m_g}{m_{\text{tot}}} = \frac{\dot{m}_g}{\dot{m}_{\text{tot}}}$$

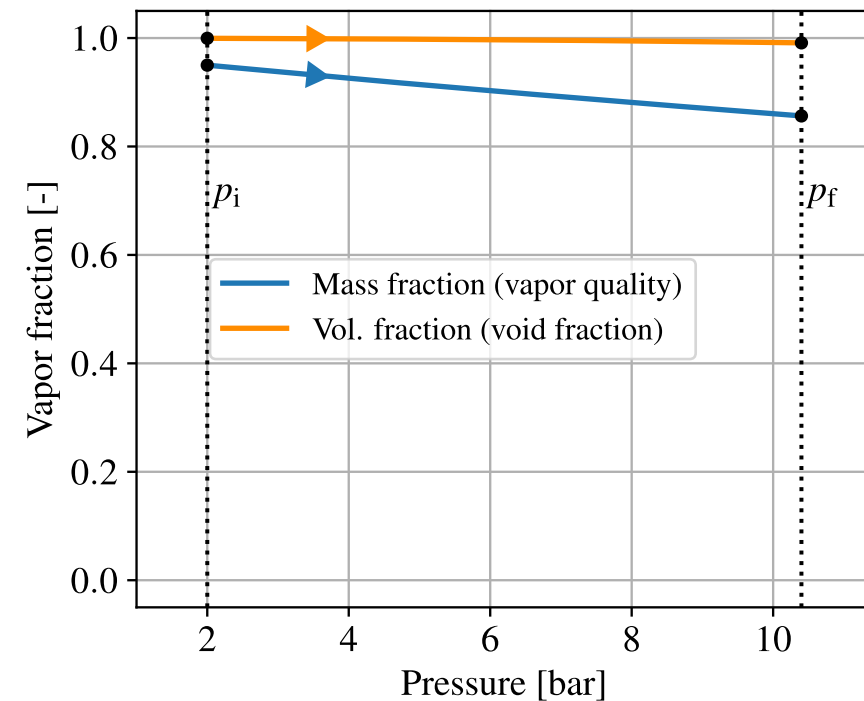


Two-phase in positive-displacement compressors

Definition of the void fraction:
$$\alpha(Q) = \frac{V_g}{V_{\text{tot}}} = \frac{Q v_g^\sigma}{Q v_g^\sigma + (1 - Q) v_l^\sigma}$$



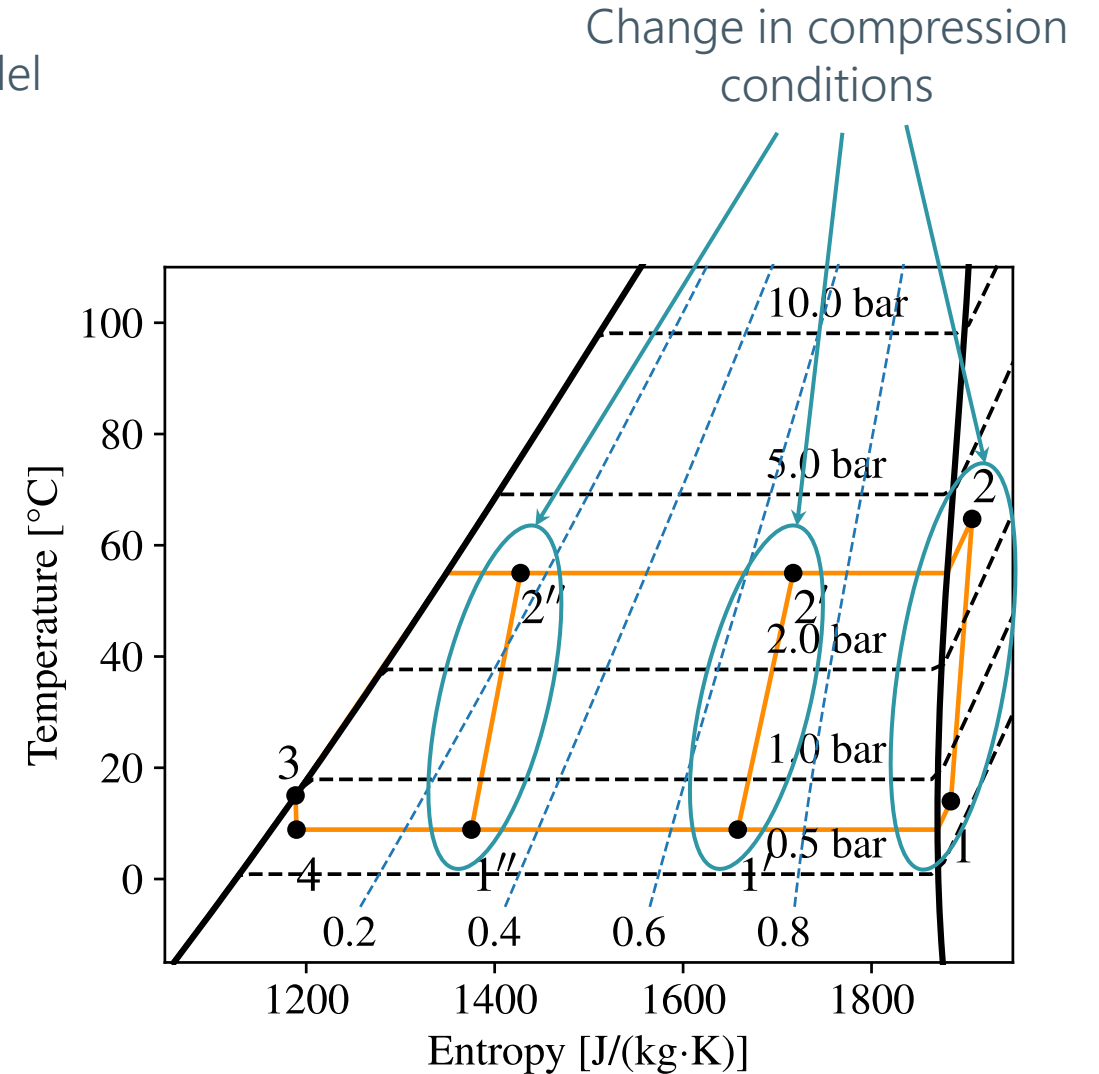
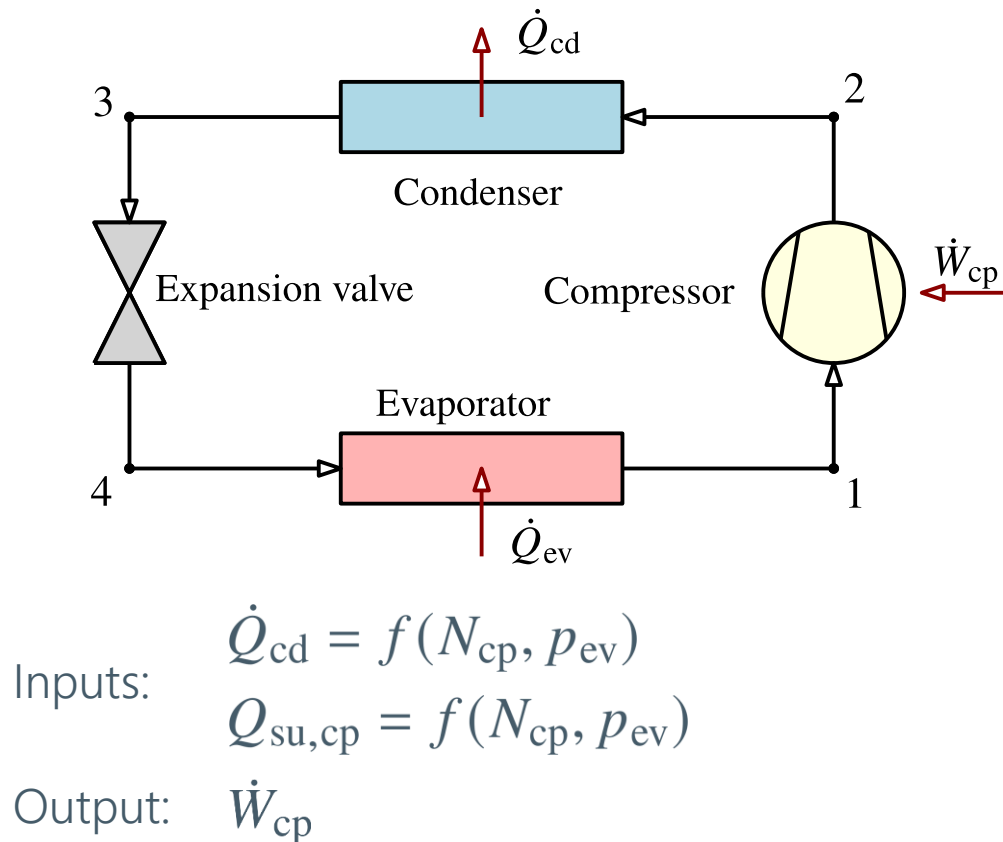
Danger zone



Safe zone

Two-phase compression in thermodynamic cycles

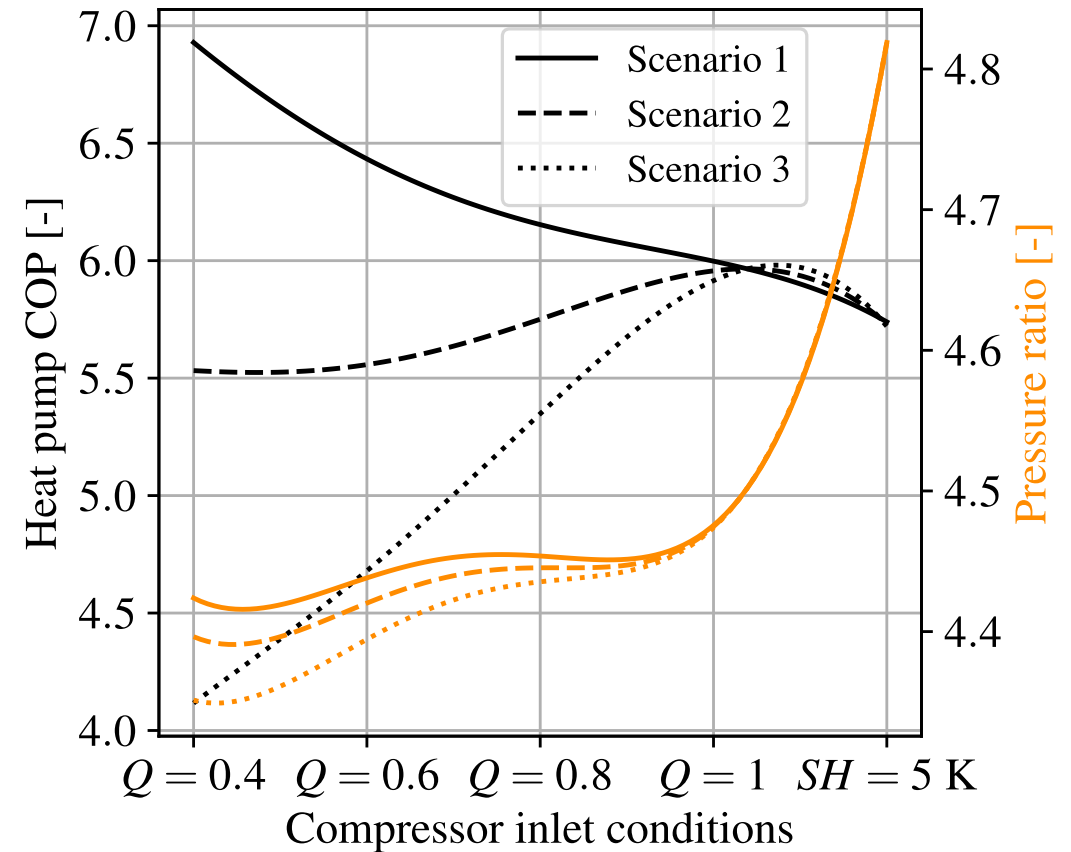
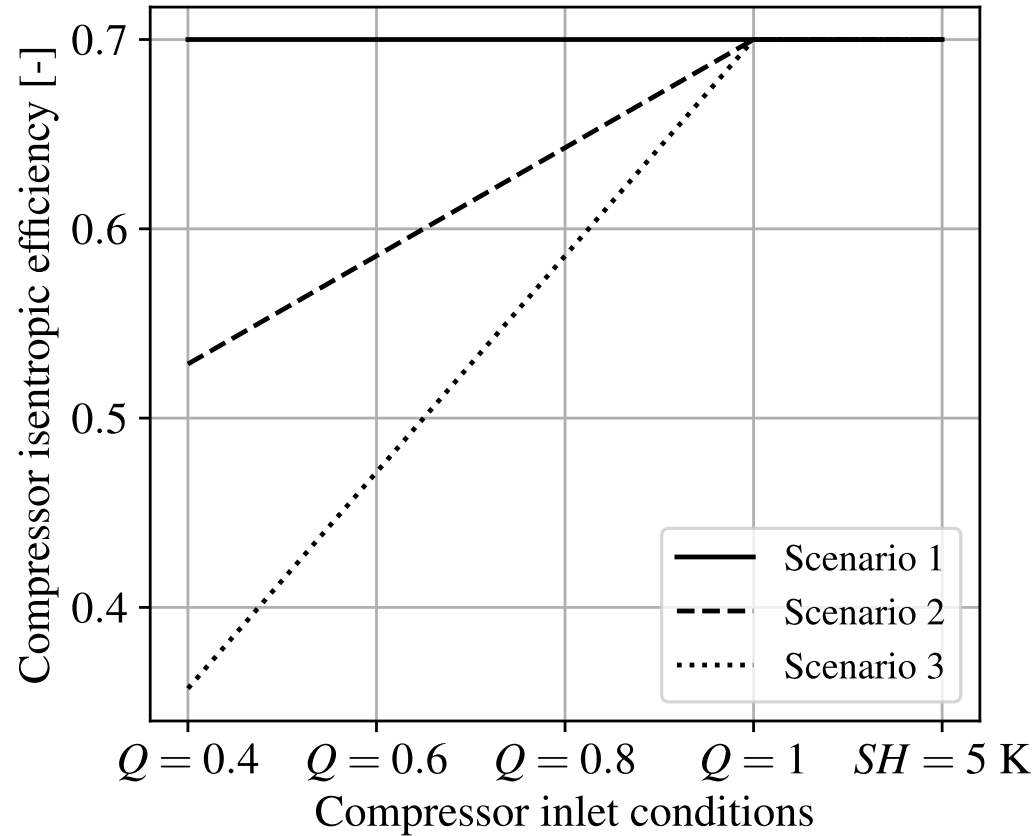
- Modeling of a heat-pump water heater
- Heat exchanger modeled with a moving-boundaries model
- Assumption on the compressor efficiency



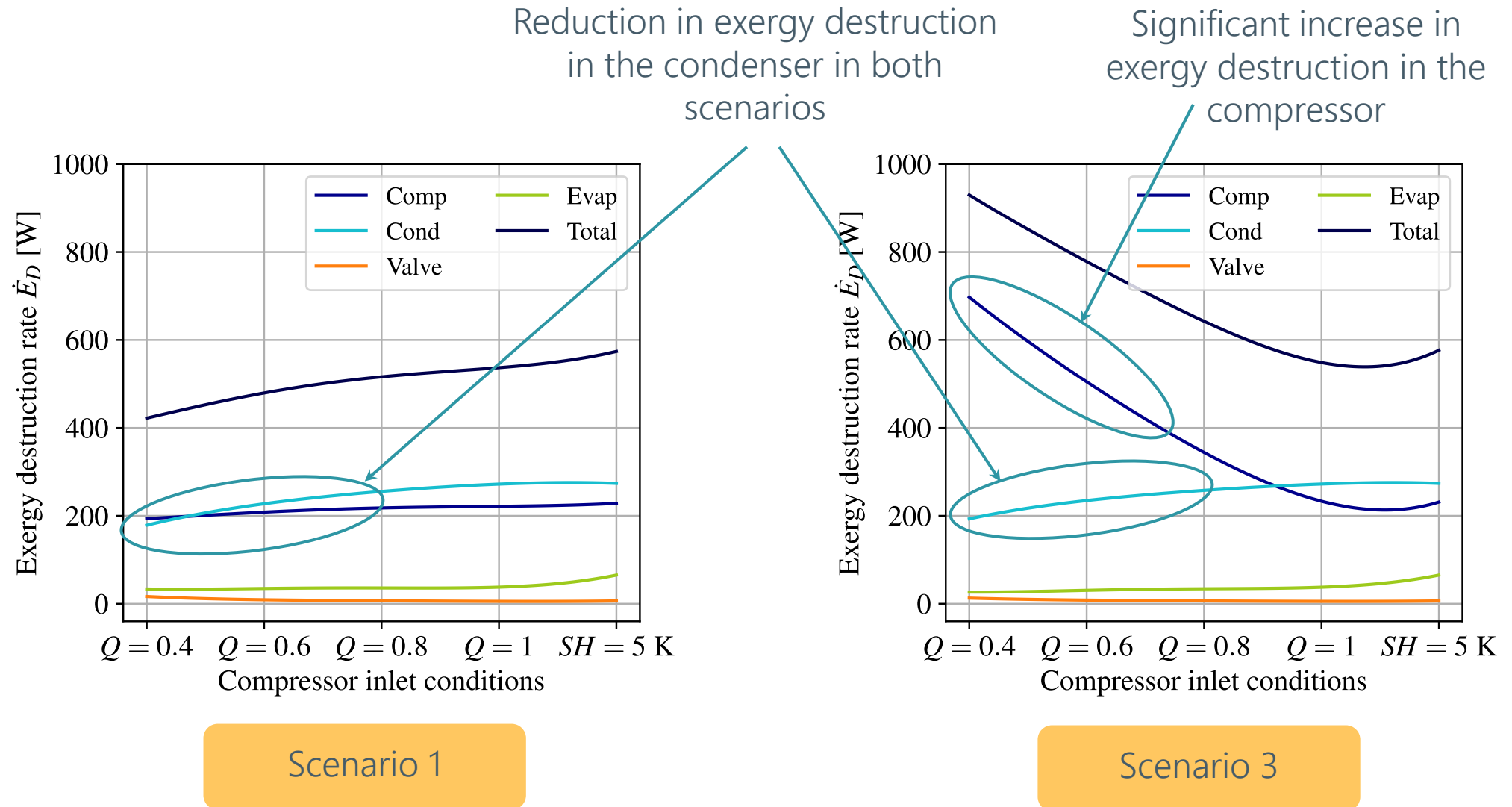
Two-phase compression in thermodynamic cycles

$$\varepsilon_{\text{is}} = \frac{\dot{m} (h_{\text{ex,cp,is}} - h_{\text{su,cp}})}{\dot{W}_{\text{cp}}}$$

$$COP = \frac{\dot{Q}_{\text{cd}}}{\dot{W}_{\text{cp}}}$$

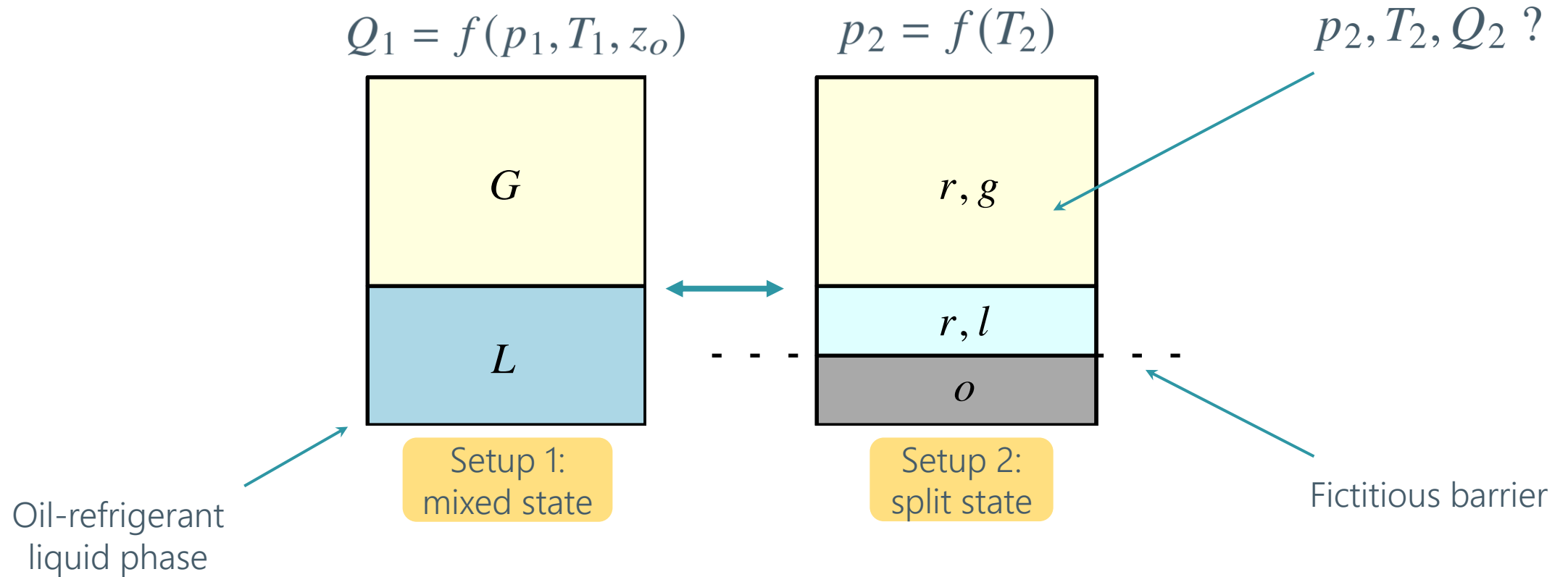


Two-phase compression in thermodynamic cycles



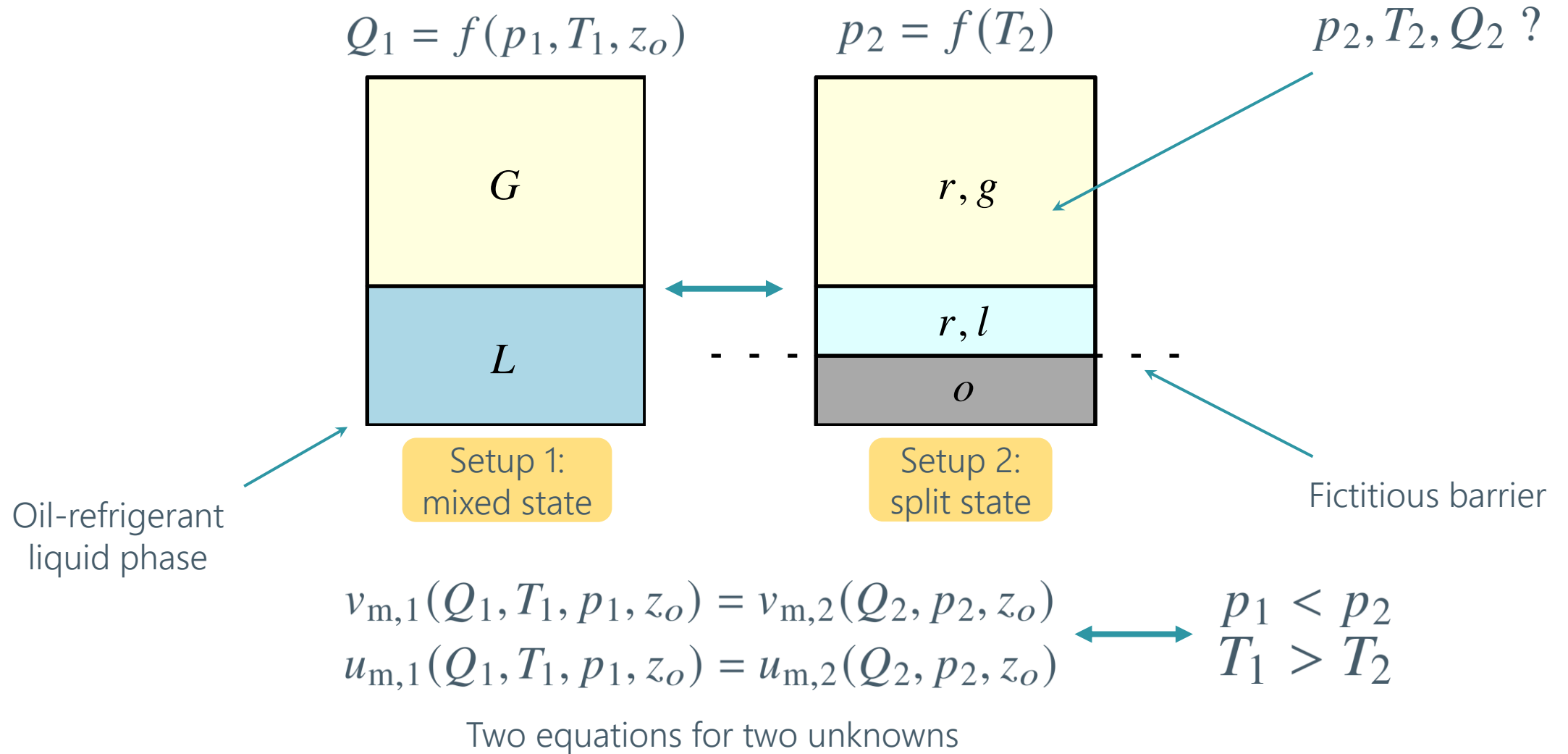
2. Oil-refrigerant mixture Modeling

Desolubilization of the mixture

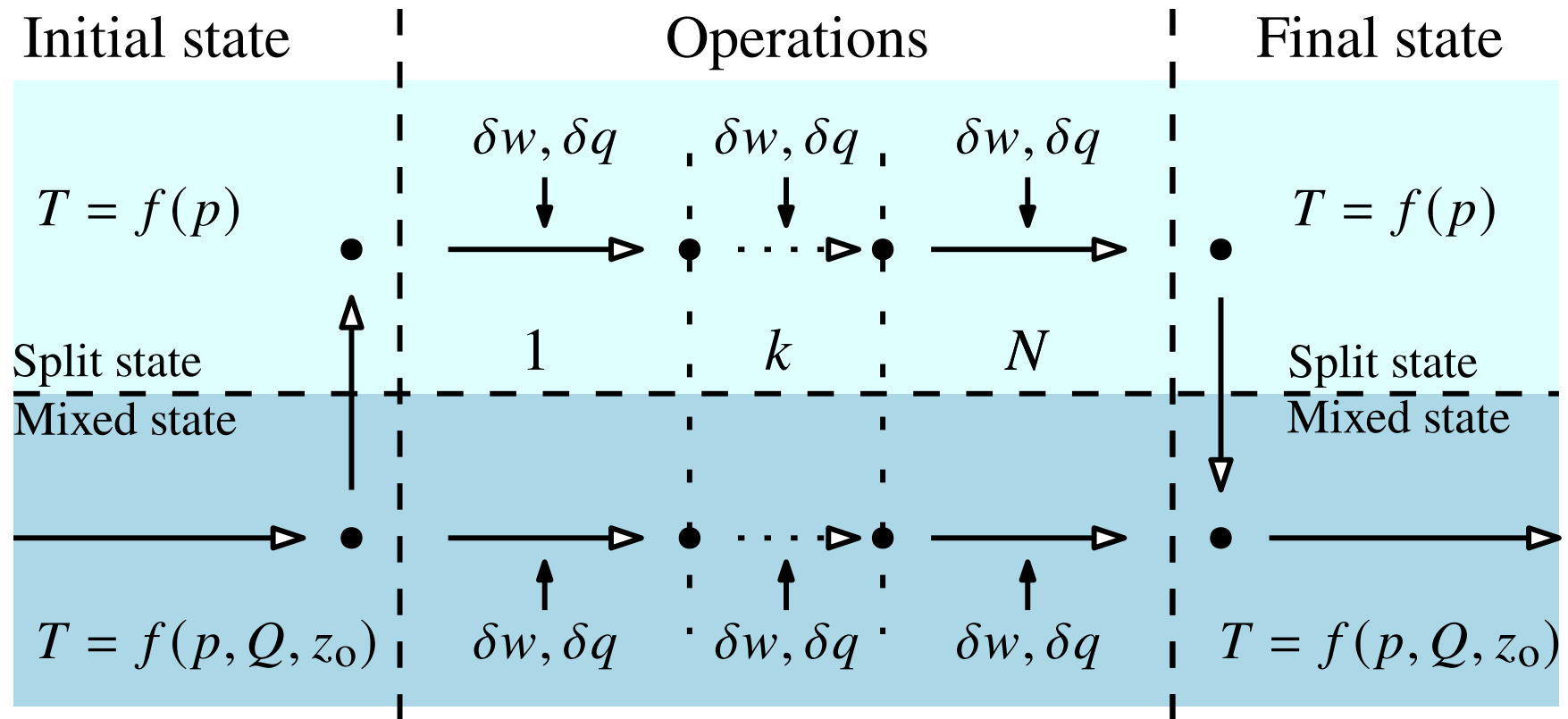


What could be the state of the refrigerant (temperature, pressure, vapor quality) and the oil to be mixed together to obtain the state of setup 1?

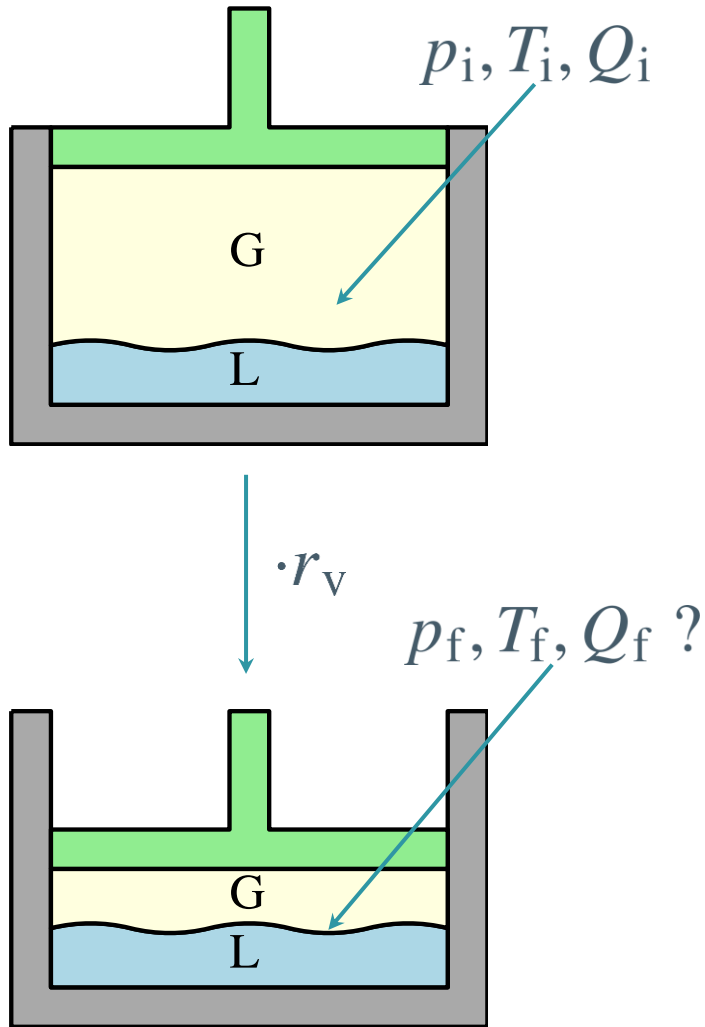
Desolubilization of the mixture



Simplified resolution process



Compression example



Mixed state resolution

$$s_{m,f}(p_f, T_f, Q_f) = s_{m,i}$$

$$v_{m,f}(p_f, T_f, Q_f) = v_{m,i} \cdot r_v$$

$$p_f = f(T_f, Q_f)$$

2 residuals

Split state resolution

$$s_{m,f}(p_f, T_f, Q_f) = s_{m,i}$$

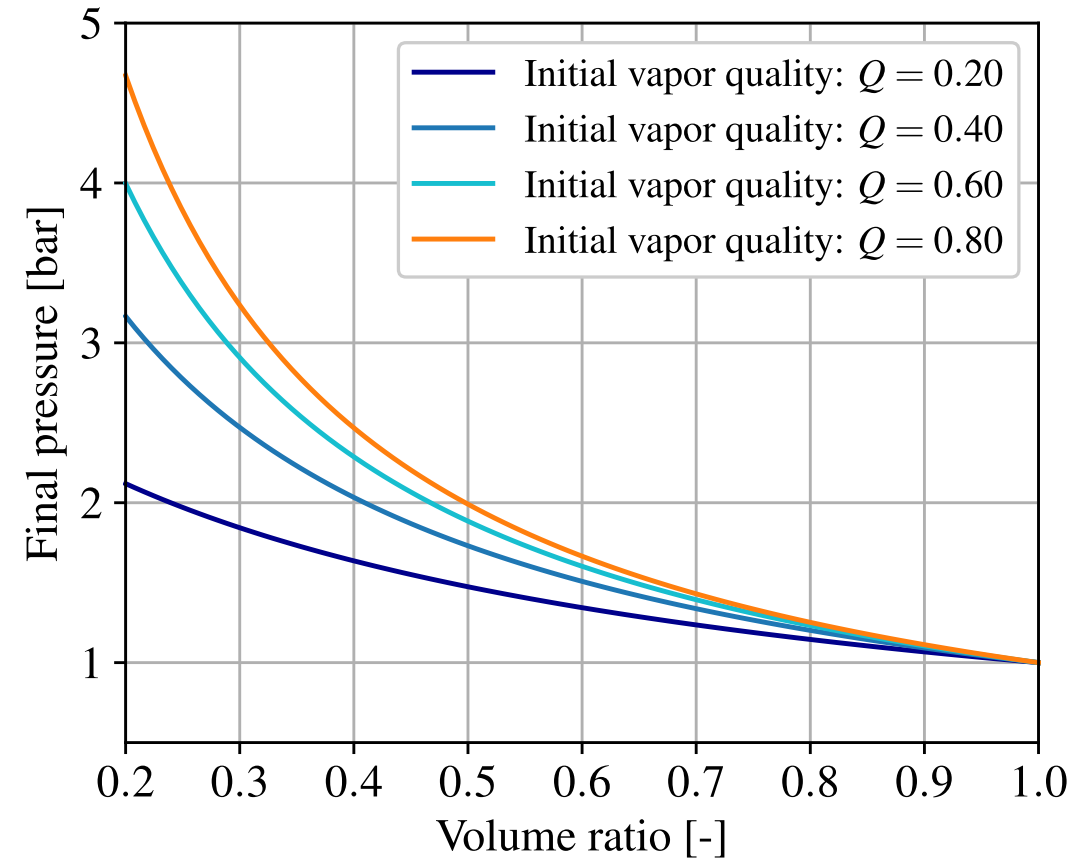
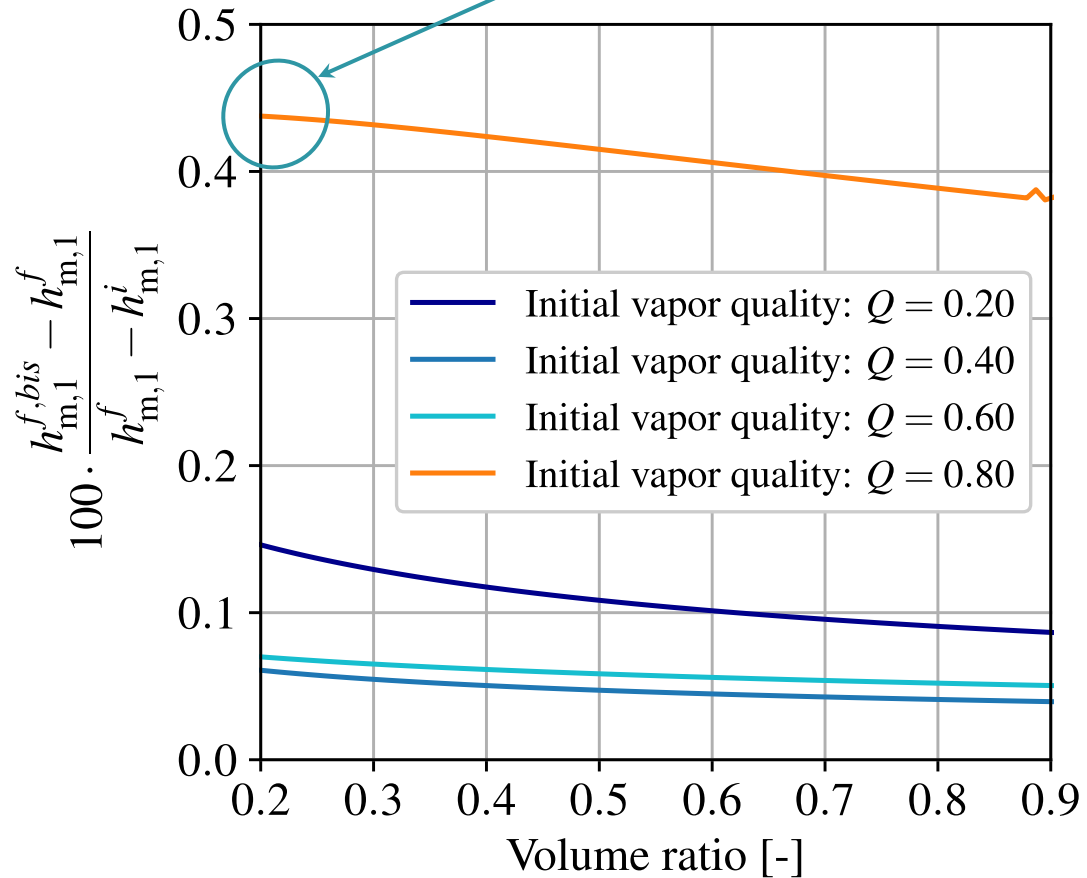
$$v_{m,f}(p_f, T_f, Q_f) = v_{m,i} \cdot r_v$$

$$p_f = f(T_f)$$

1 residual

Compression example

Deviation inferior to 0.5%



3. Experimental investigations

Definition of the performance indicators (Change this slide !!)

Volumetric efficiency: $\varepsilon_v = \frac{\dot{m}}{\dot{m}_{th}} = \frac{\dot{m}_{tot}}{N_{cp} V_{disp} \rho_{su}}$

Known

Determined from measurement

Isentropic efficiency: $\varepsilon_{is} = \frac{\dot{W}_{is}}{\dot{W}} = \frac{\dot{m}_{tot} (h_{ex,is} - h_{su})}{\dot{W}_{shaft,cp}}$

$h_{ex,is} ?$

2 unknowns

2 equations

$$\begin{cases} s_{ex,is} = s_{su} \\ Q_{is} = f(p_{ex}, T_{is}) \end{cases}$$

$$h_{ex,is} = Q_{is} h_{r,g}(p_{ex}, T_{is}) + z_o h_o(T_{is}) + (1 - Q_{is} - z_o) h_{r,l}^{\sigma}(T_{is})$$

Post-processing using GPExp

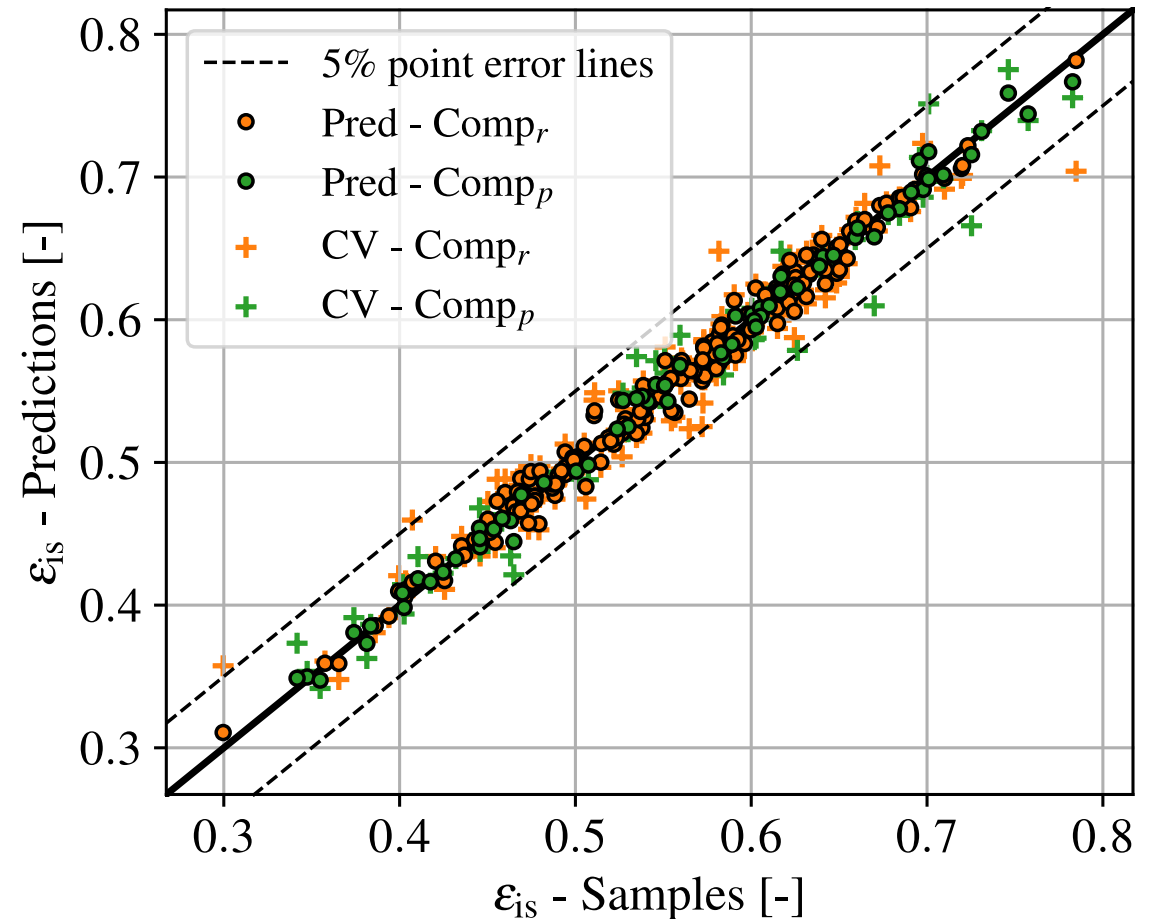
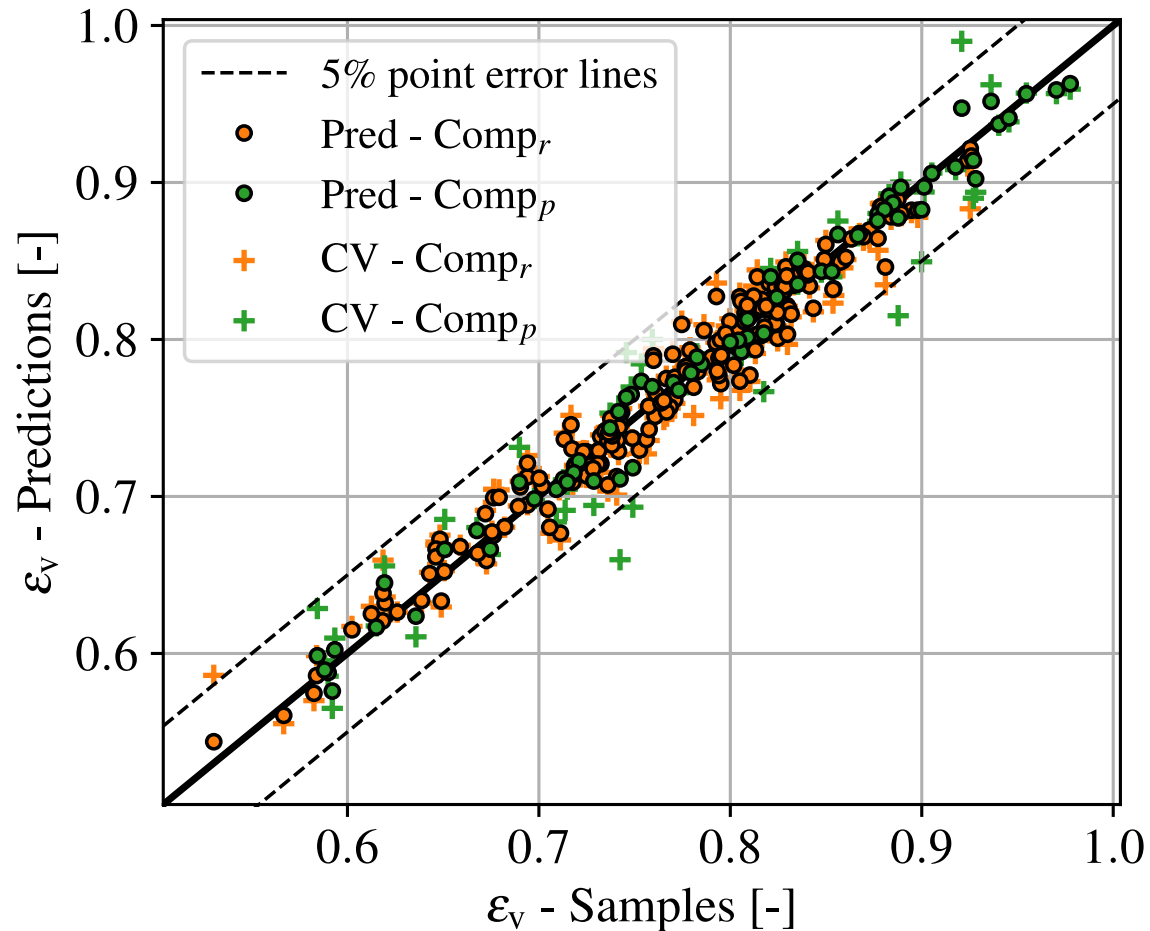
Interpolation tool

$$AAD_{CV,r} = 3.41\%$$

$$AAD_{CV,p} = 4.77\%$$

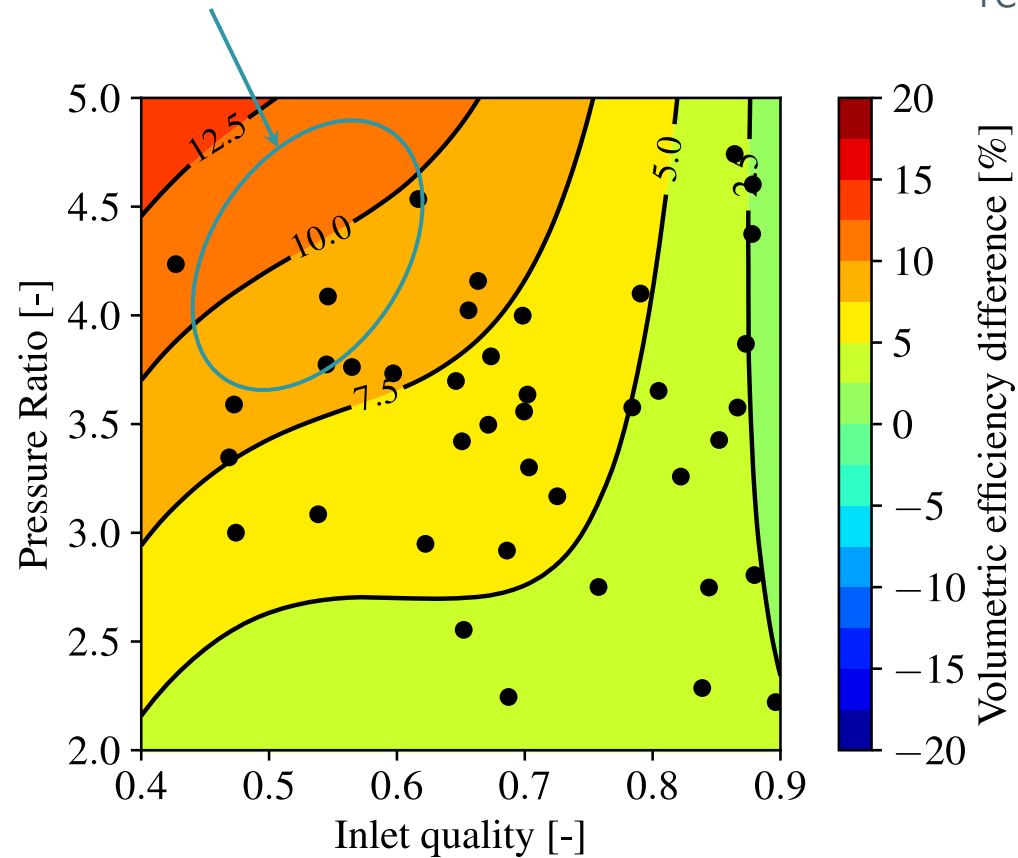
$$AAD_{CV,r} = 2.56\%$$

$$AAD_{CV,p} = 3.45\%$$

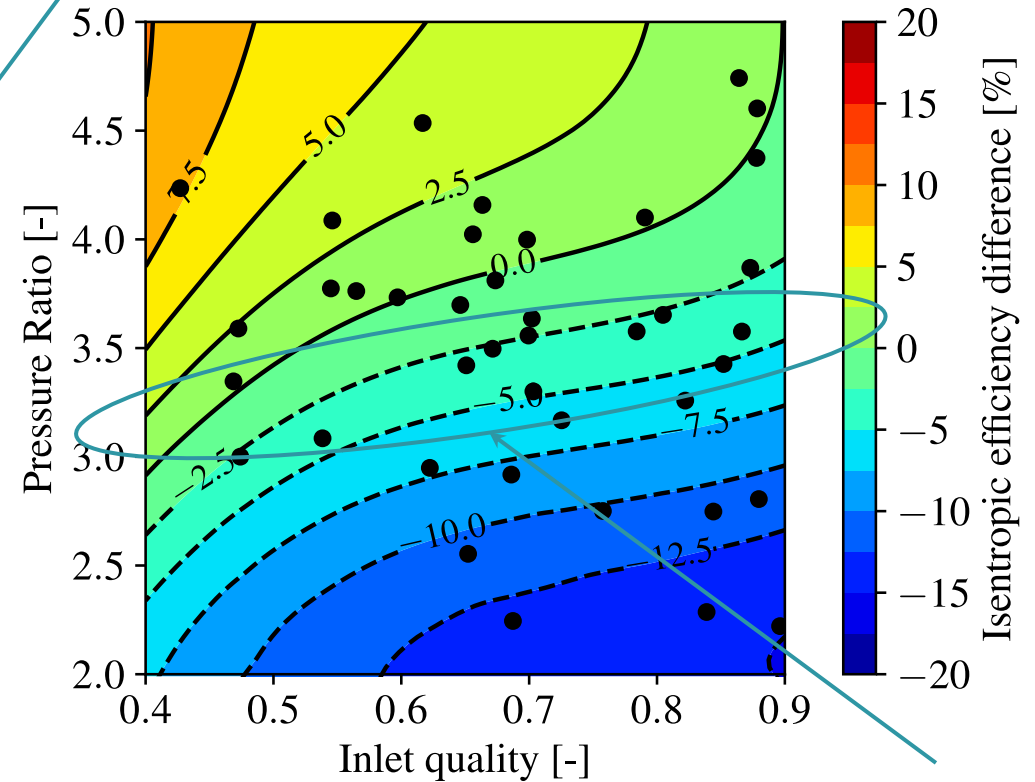


Comparison with the lab-scale prototype results

Better volumetric efficiency



Efficiency difference with the retrofitted compressor



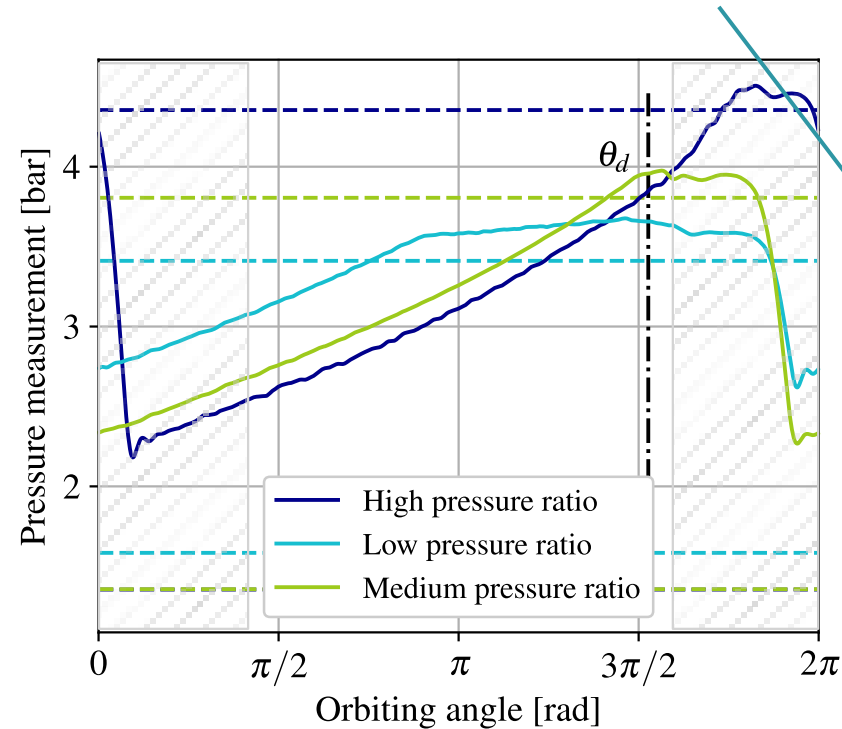
$$p_{\text{su}} = 1.5 \text{ [bar]} \quad OCR = 5 \text{ [%]} \quad N = 2500 \text{ [RPM]}$$

Impact of the higher built-in volume ratio

Dynamic pressure sensor results

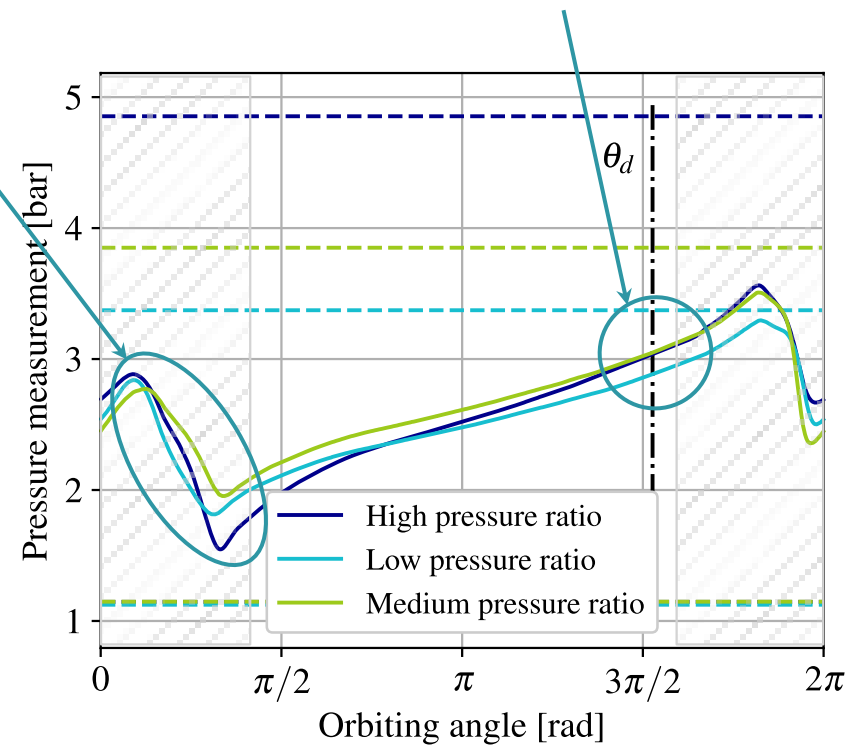
One less plot

Unexpected behavior when the scroll passes the port



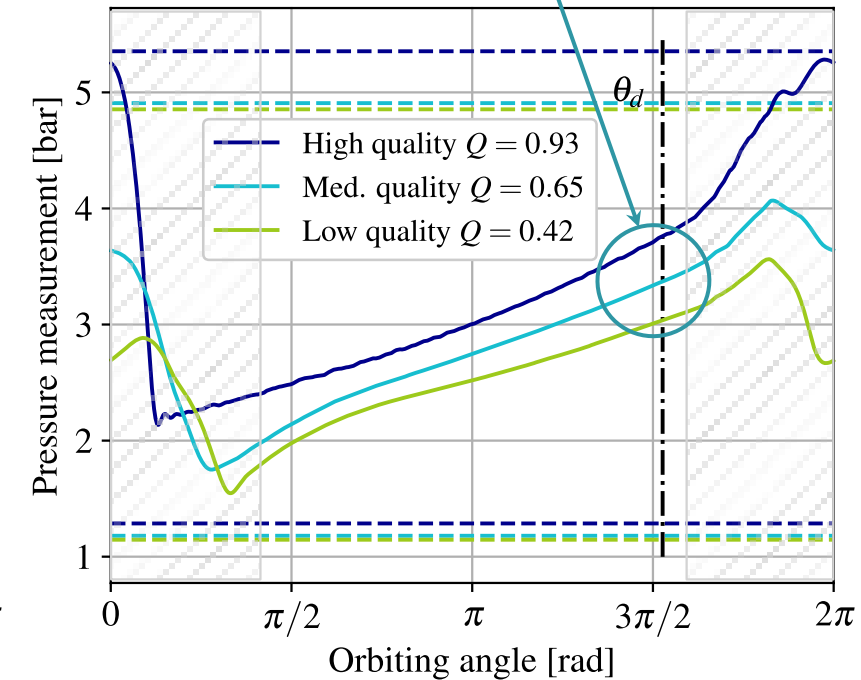
High vapor quality
 $Q_{su} = 0.8 \pm 0.05$ [-]

Undercompression trend for all three points



Low vapor quality
 $Q_{su} = 0.4 \pm 0.05$ [-]

Lower pressure at a low vapor quality



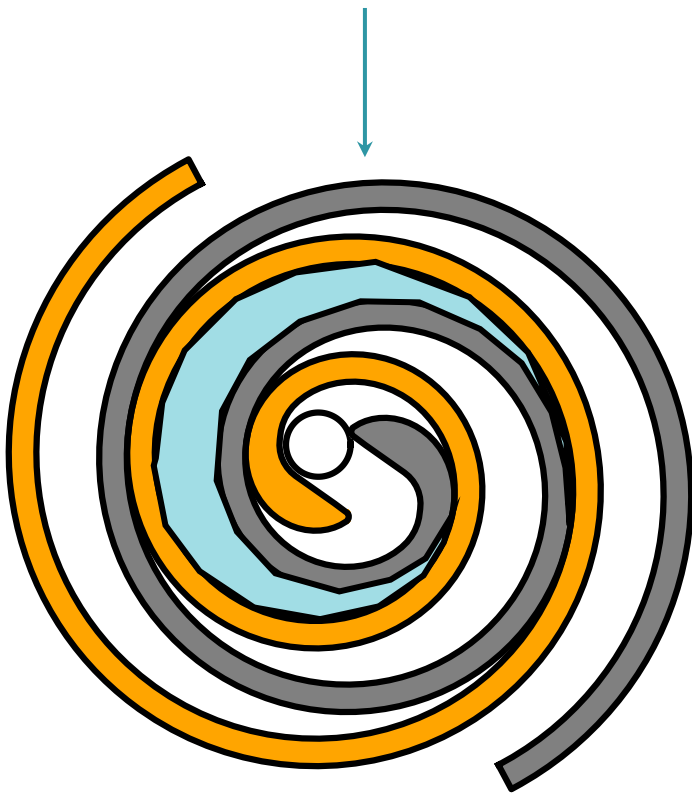
$p_{su} = 1.25 \pm 0.15$ [bar]
 $r_p = 4.0 \pm 0.25$ [-]

4. Modeling and validation

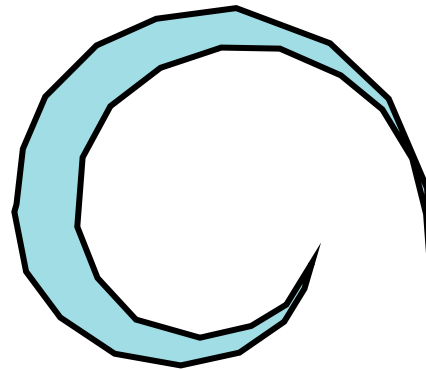
Geometrical model

Green theorem?

Curve equations



Polygon of N
segments



Volume

$$V = \frac{1}{2} h_s \sum_{i=1}^{N-1} (x_i y_{i+1} - x_{i+1} y_i)$$

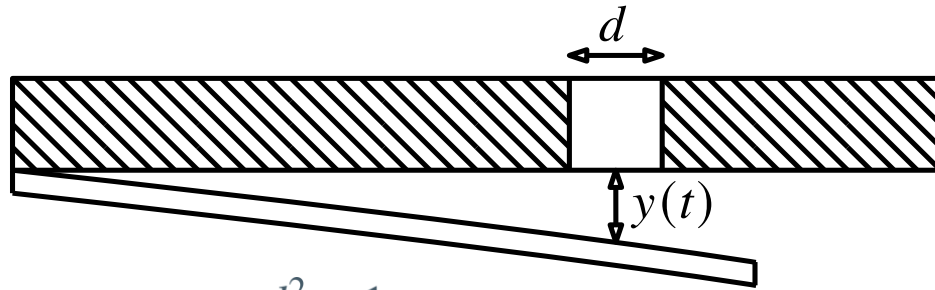
Interpolation $\downarrow$ $N = 1000$
 $\theta = 1, 2, \dots, 360^\circ$

$$V(\theta) = c_6 \theta^5 + c_5 \theta^4 + c_4 \theta^3 + c_3 \theta^2 + c_2 \theta + c_1$$

$$\frac{dV(\theta)}{d\theta} = 5c_6 \theta^4 + 4c_5 \theta^3 + 3c_4 \theta^2 + 2c_3 \theta + c_2$$

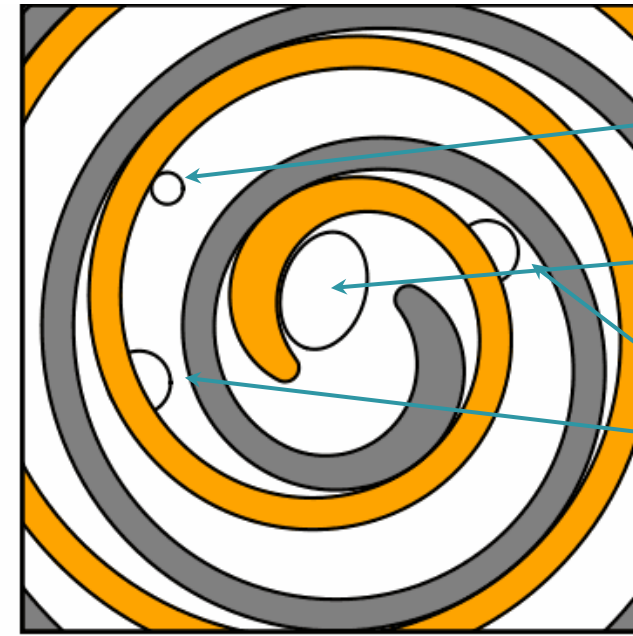
Maximum error of 0.8%

Discharge valves



$$y = \pi \frac{d^2}{4} \frac{1}{C_{\text{valve}}} \min(0, p - p_{\text{ex}})$$

$$y \leq y_{\text{stopper}}$$

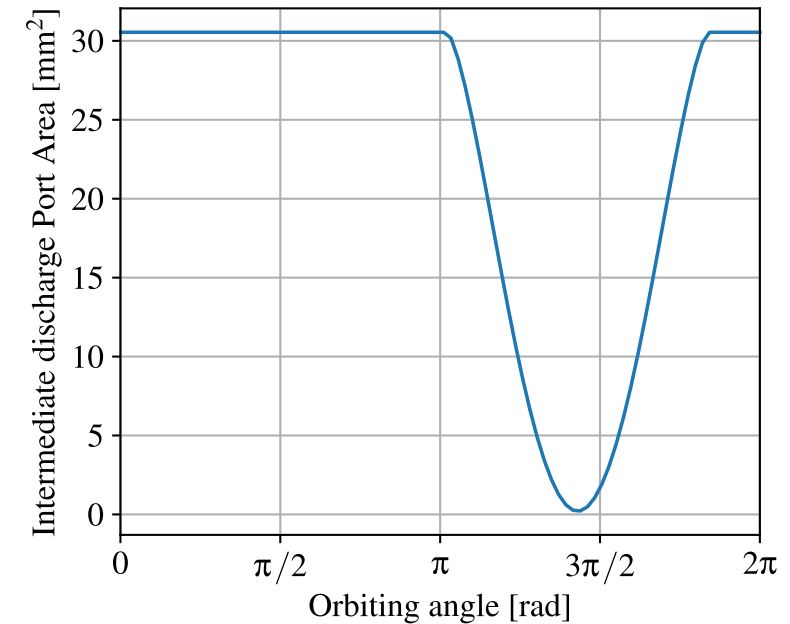


Total discharge mass flow rate:

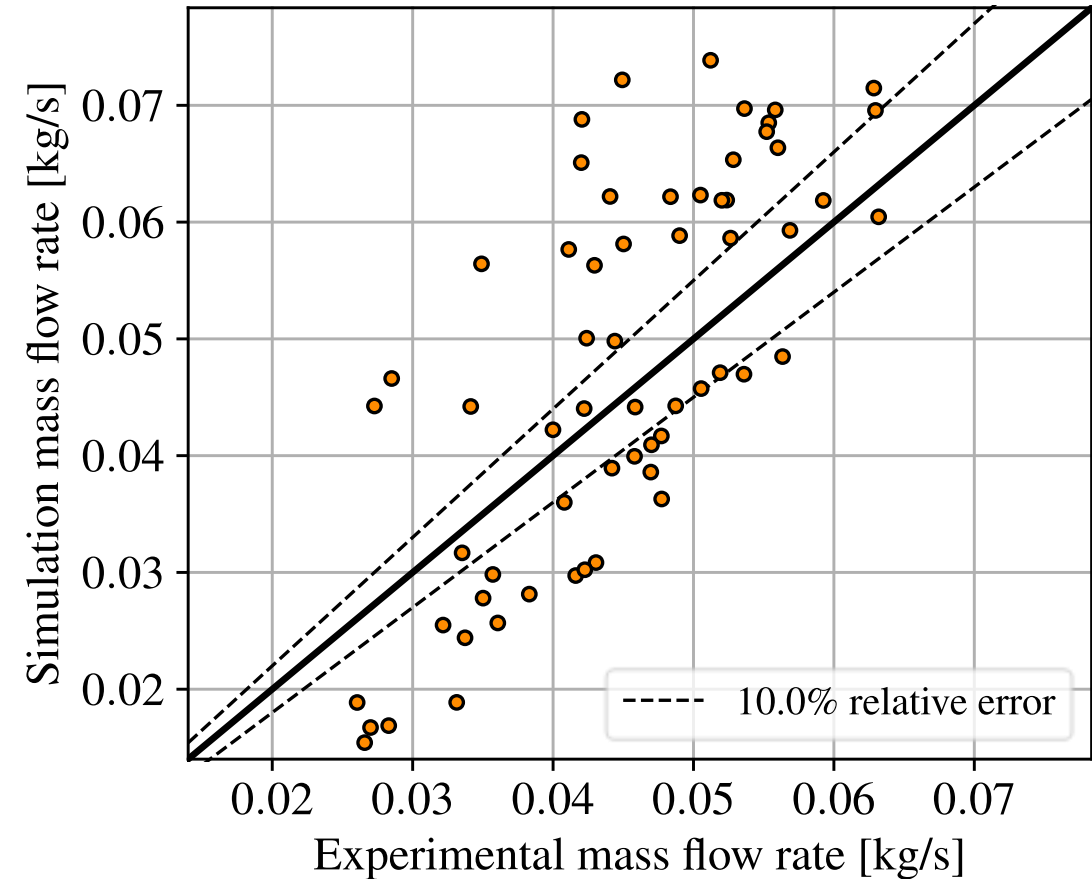
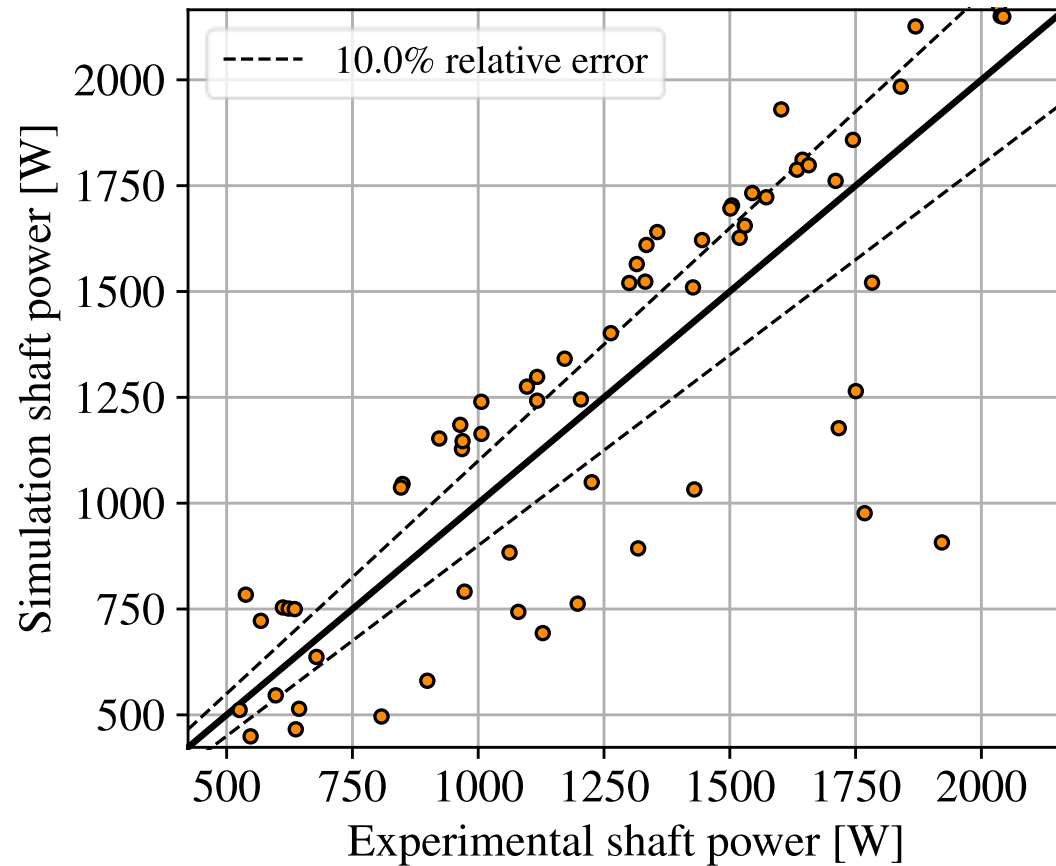
$$\dot{m}_{\text{dis}} = \frac{1}{T} \int_0^T \dot{m}(t) dt = \frac{1}{2\pi} \int_0^{2\pi} \dot{m}(\theta) d\theta$$

Exhaust enthalpy for ϕ (r,g - r,l - o):

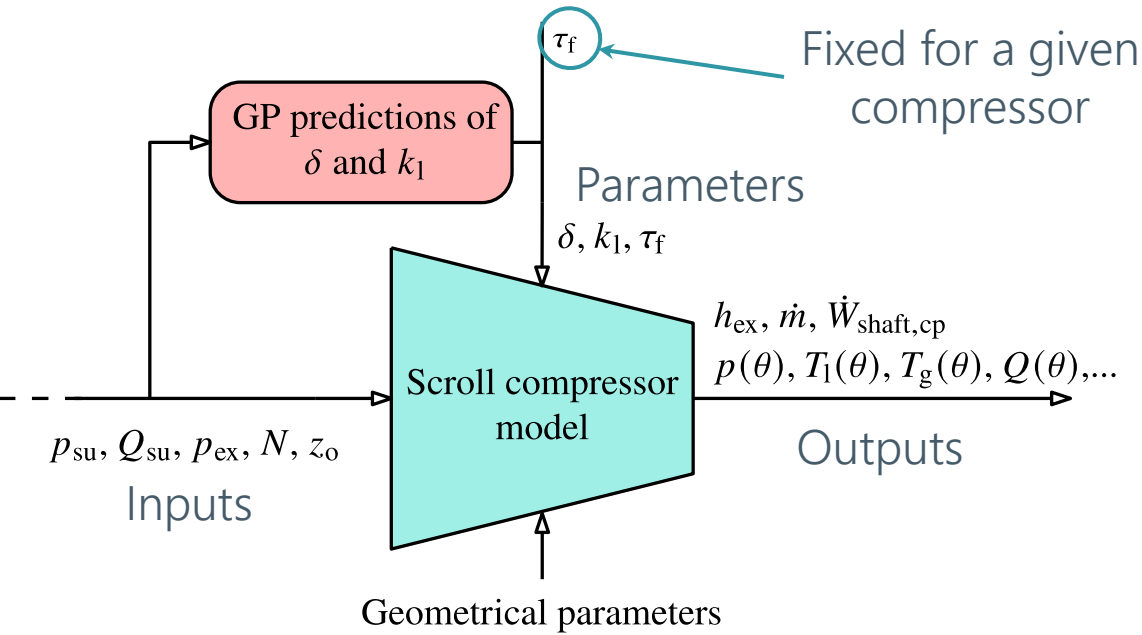
$$h_{\phi,\text{dis}} = \frac{\int (\dot{m}_{\phi,\text{main}} h_{\phi,\text{main}} + \dot{m}_{\phi,\text{int}} h_{\phi,\text{int}}) d\theta}{\int (\dot{m}_{\phi,\text{main}} + \dot{m}_{\phi,\text{int}}) d\theta}$$



Constant parameters – $\tau = 2.6 \text{ Nm}$



Validation – varying parameters



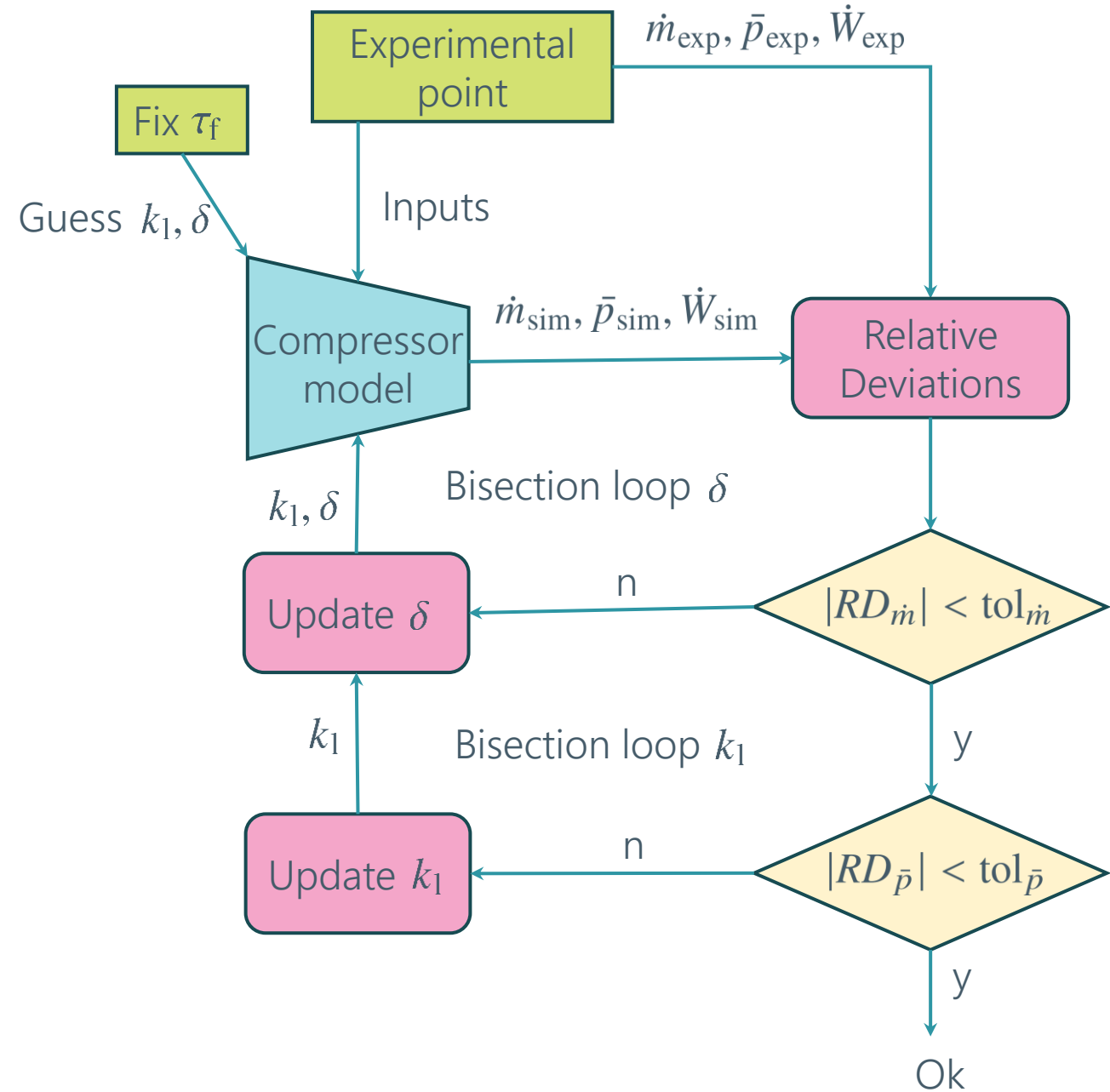
For each data point find the perfect

(δ, k_1)

Interpolation

$$\delta = f(p_{su}, Q_{su}, p_{ex}, N, z_o)$$

$$k_1 = f(p_{su}, Q_{su}, p_{ex}, N, z_o, \delta)$$

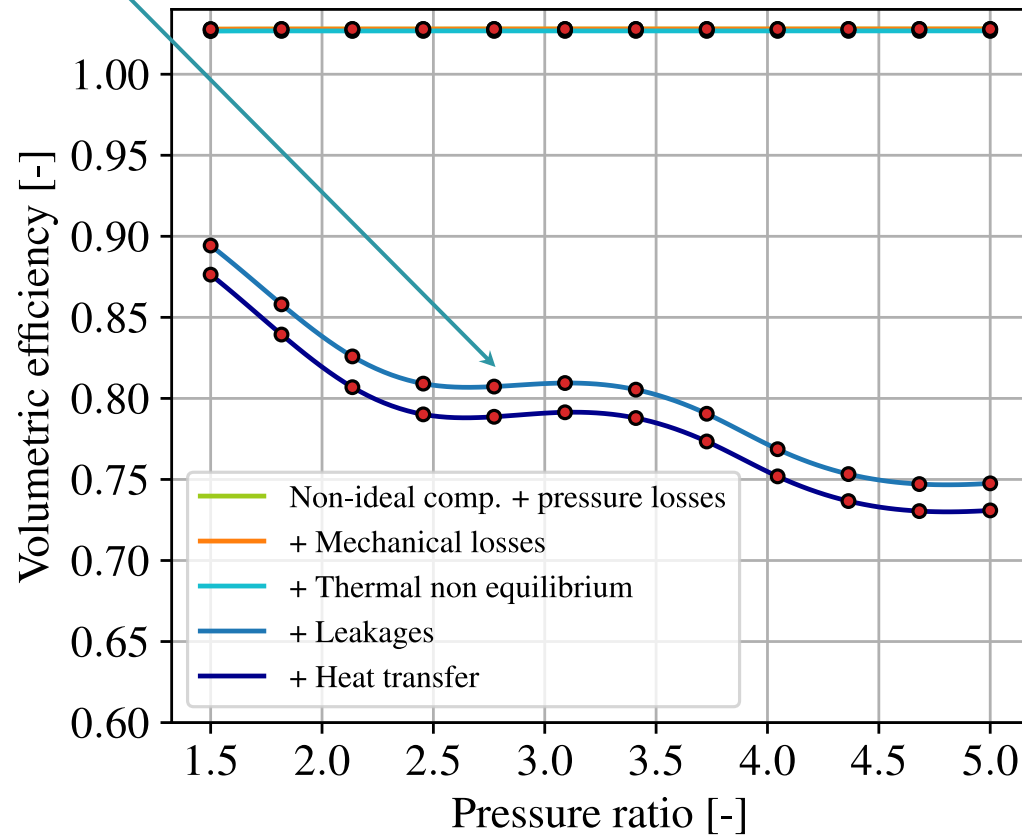


5. Results and conclusions

Sensitivity analysis: volumetric efficiencies

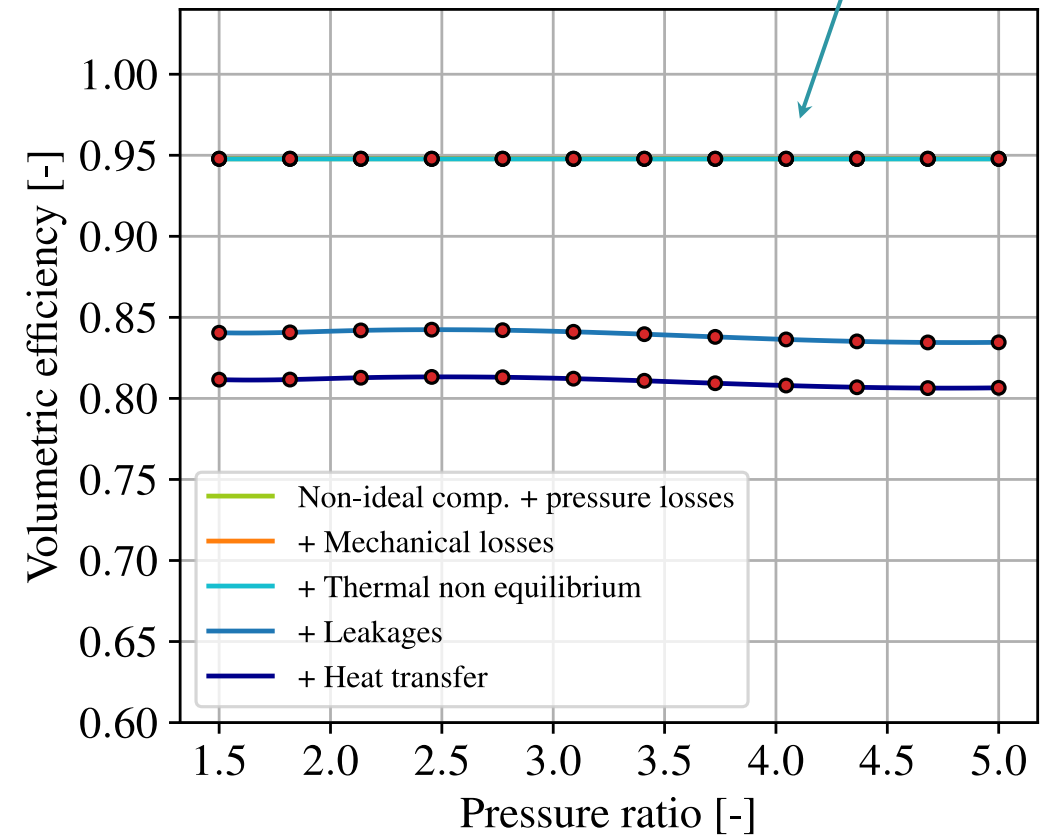
Leakage depends on the pressure ratio

Retrofitted compressor



High suction pressure losses

Lab-scale prototype



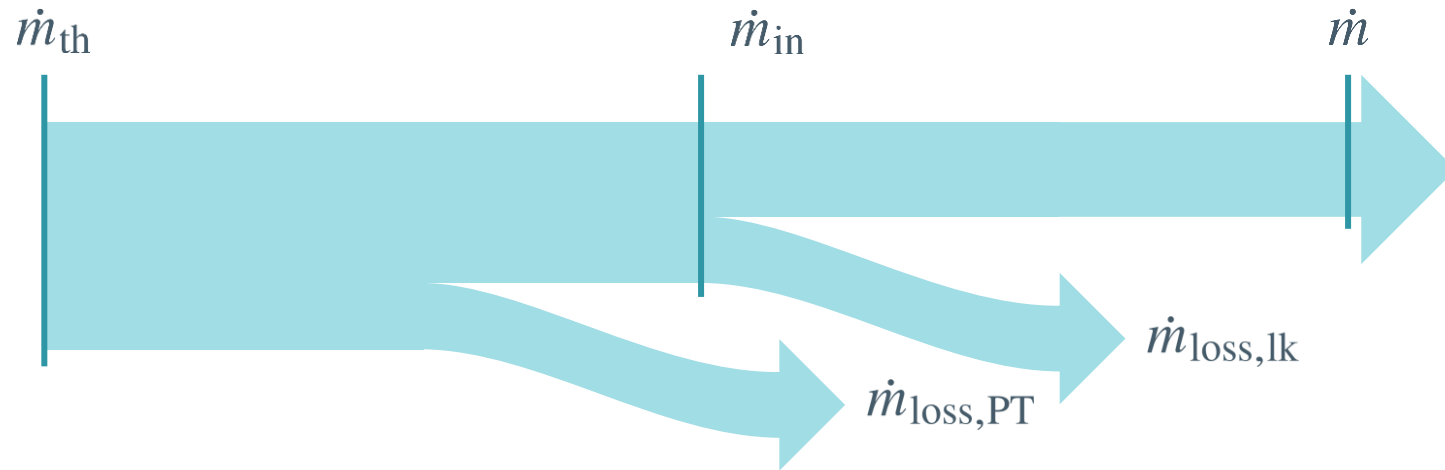
$$Q_{su} = 0.5 [-]$$

$$N = 4000 [\text{RPM}]$$

$$p_{su} = 1.5 [\text{bar}]$$

$$OCR = 5 [\%]$$

Impact of the speed and machine size



Speed [RPM]	$\dot{m}_{th}$ [g/s]	$\dot{m}_{loss,PT}$ [g/s]	$\dot{m}_{loss,lk}$ [g/s]	$\dot{m}$ [g/s]	ε_v [-]
1000	55	3.6	11.3	40.1	73%

Increasing the machine size also allows to reduce the impact of losses