Heat Pump Systems

Summary W7

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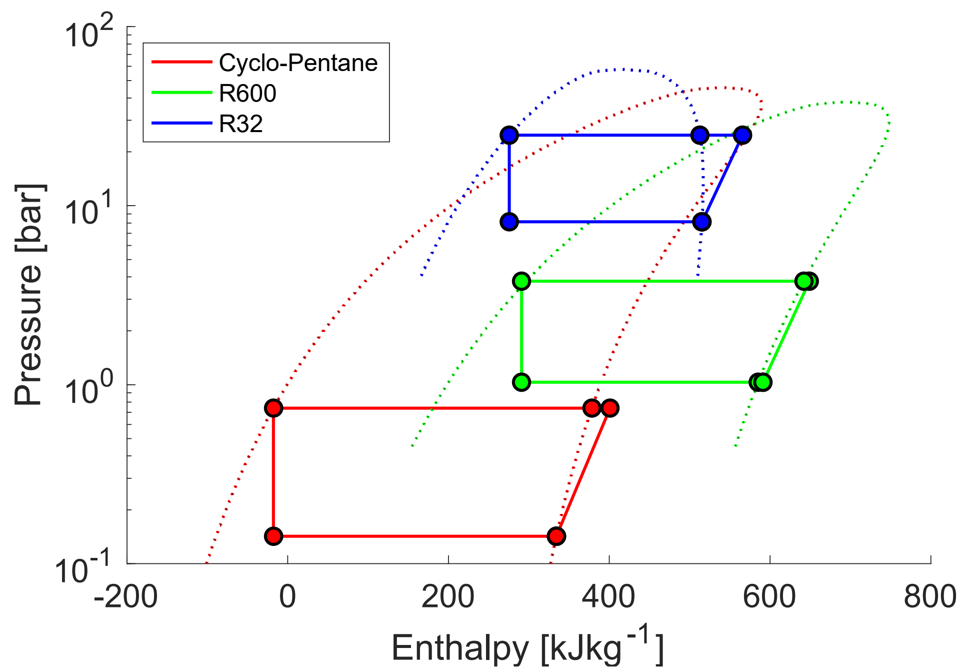
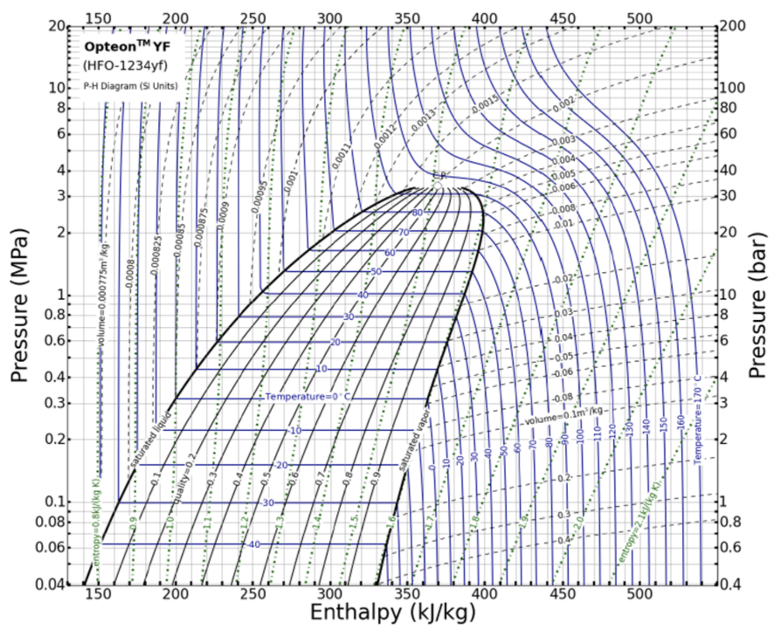
Working Fluid Selection

- Working fluid selection has important effect of cycle design
  - Design pressure ratio
  - Slope of saturation line
  - Slope of isentropic lines
  - Thermophysical properties
  - ODP & GWP
  - Compatibility with oil
  - Chemical stability
  - Toxicity
  - Flammability
  - Cost …



Working Fluid Modeling

- Cycle modeling requires accurate thermophysical properties
- Equation of state constructed on P_{Vap}, ρ_{Liq}, c_{p0}, P_{Crit}, T_{crit}
- Basis is cubic EOS → Refprop & CoolProp

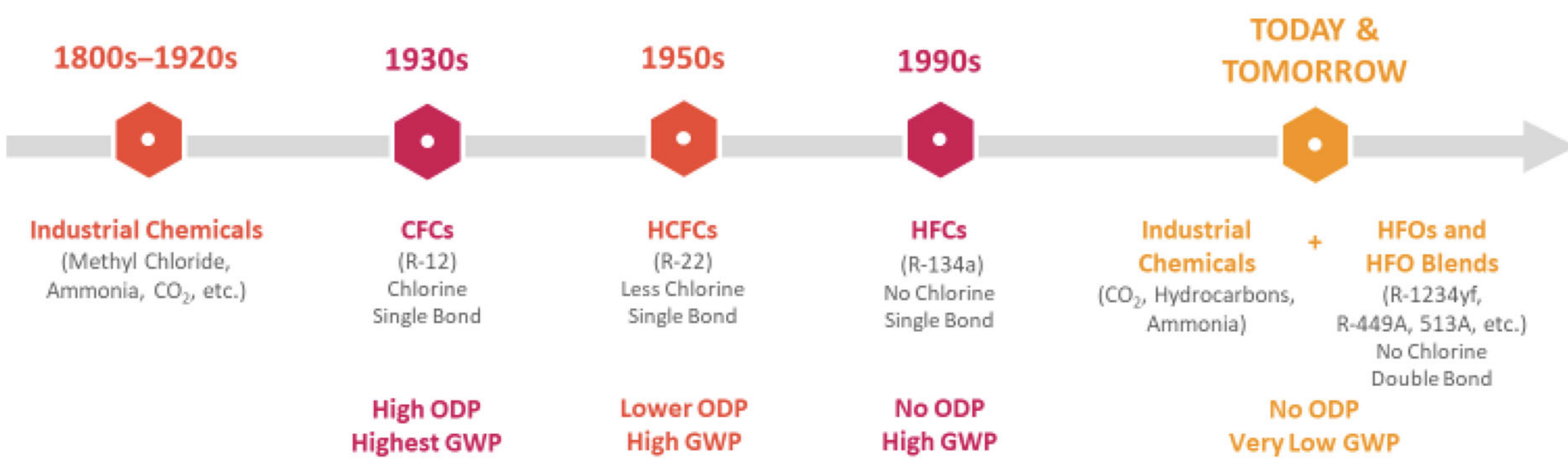


$$P = \frac{rT}{(v-b)} - \frac{a\alpha}{(v^2 + uvb + wb^2)}$$

EOS	u	w
Van der Waals (1873)	0	0
Redlich-Kwong (1949)	1	0
Soave-Redlich-Kwong (1972)	1	0
Peng-Robinson (1976)	2	-1

Evolution of Working Fluids

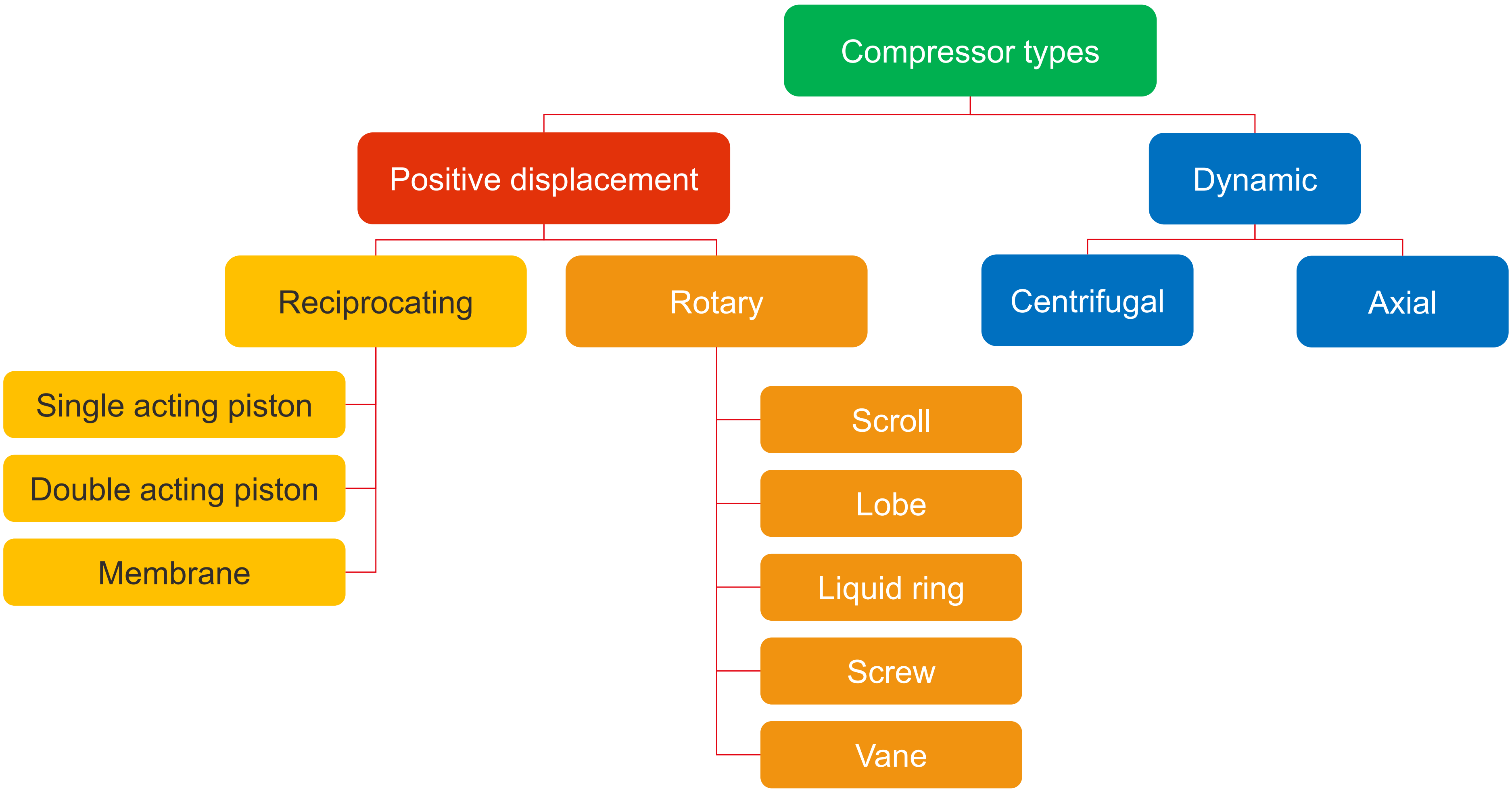
- Companies interested in selling synthetic fluids
- Natural working fluids pick up again
  - Carbon dioxide, water, ammonia, butane, propane



Role of Compressors

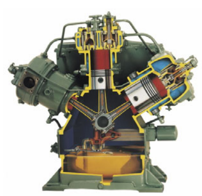
- Compressors play key role in driving heat pump cycles
- Compressor efficiency has direct impact on cycle performance
- Significant efforts are invested to improve compressor performance while increasing range, robustness
- Heat pump designers usually have a wide spectrum of machines to select from

Compressor Classification



Compressor Operational Areas

- Positive displacement machines preferred for small capacity, dynamic compressors traditionally for high capacities



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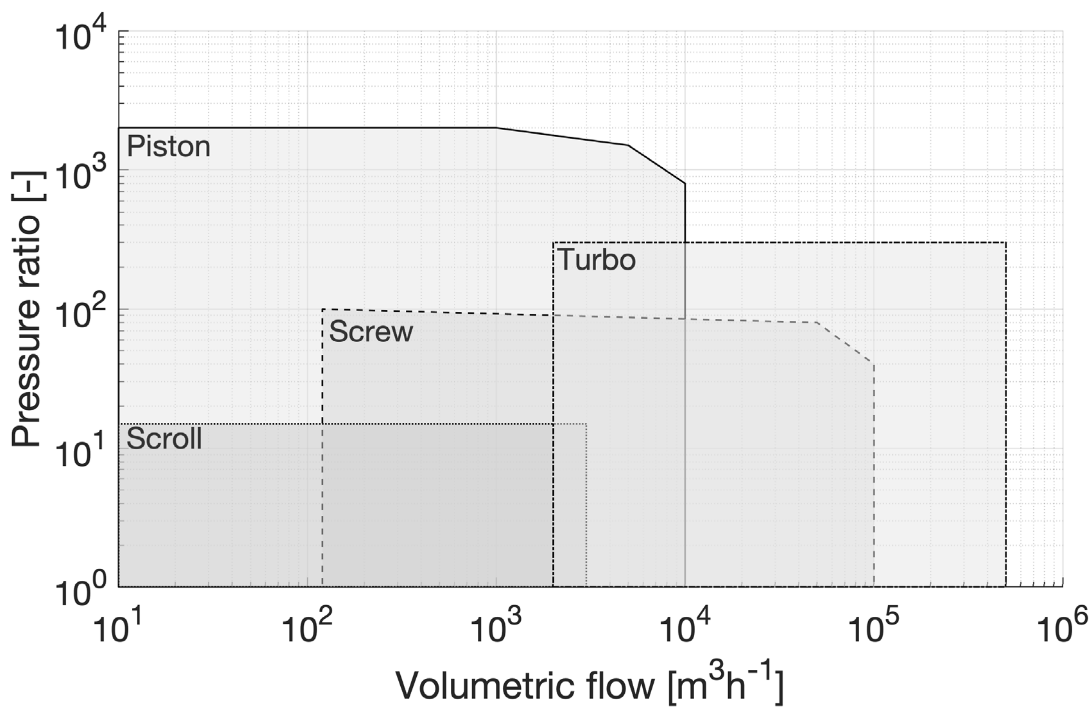

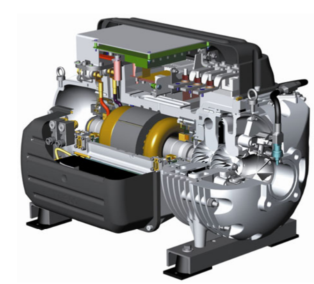

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Heat Pump Systems

Compressor Performance

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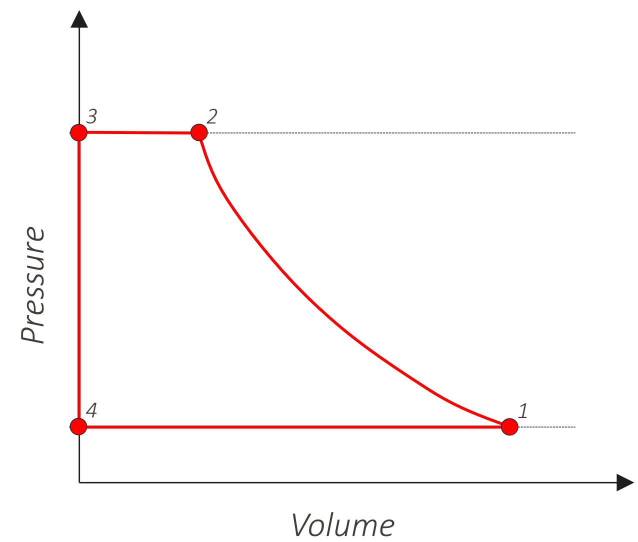
Ideal Compression

- Ideal compression process:
  - Isobaric suction (4 → 1)
  - Compression (1 → 2)
  - Isobaric discharge (2 → 3)
  - Expansion (3 →4)

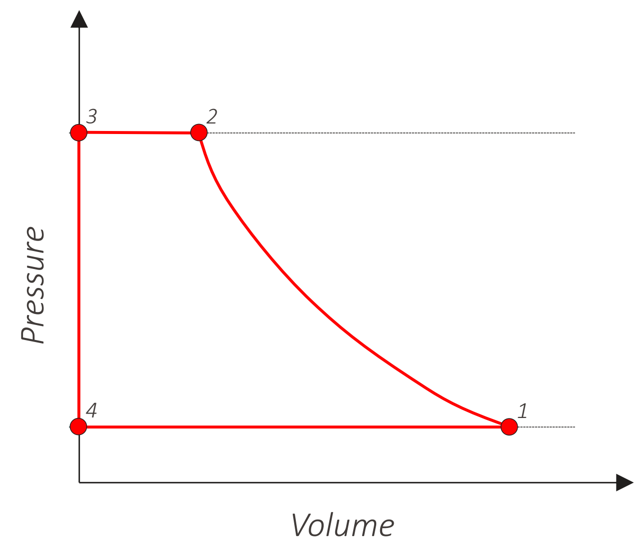


- Invested work for compression cycle

$$e^{+} = -\oint Pdv$$

$$e^{+} = -\int_{4}^{1} Pdv - \int_{1}^{2} Pdv - \int_{2}^{3} Pdv - \int_{3}^{4} Pdv$$

Ideal Compression



- Isobaric suction (4 → 1)

$$e^{+}_{4\to 1} = -\int_{4}^{1} P dv = -\left(v_1 - v_4\right) P_1$$

- Isobaric discharge (2 → 3)

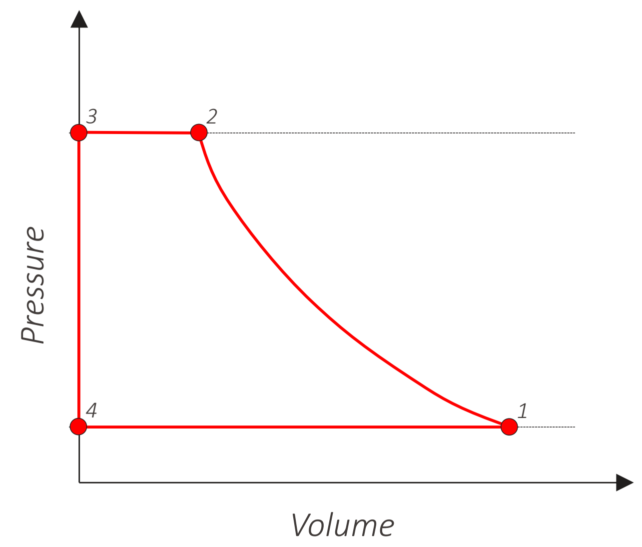
$$e^{+}_{2\to 3} = -\int_{2}^{3} P dv = -\left(v_3 - v_2\right) P_2$$

- Expansion (3 →4)

$$e^{+}_{3\to 4} = -\int_{3}^{4} P dv = 0$$

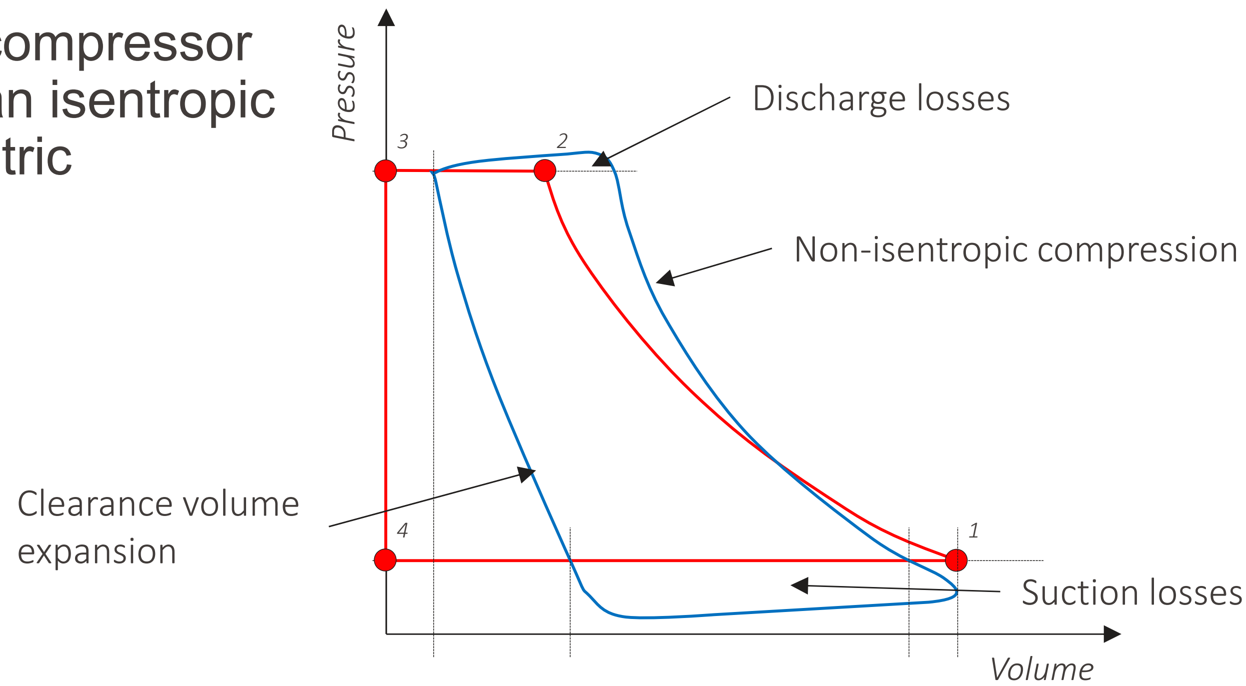
Ideal Compression

- Compression (1 → 2)



Real Compression Process

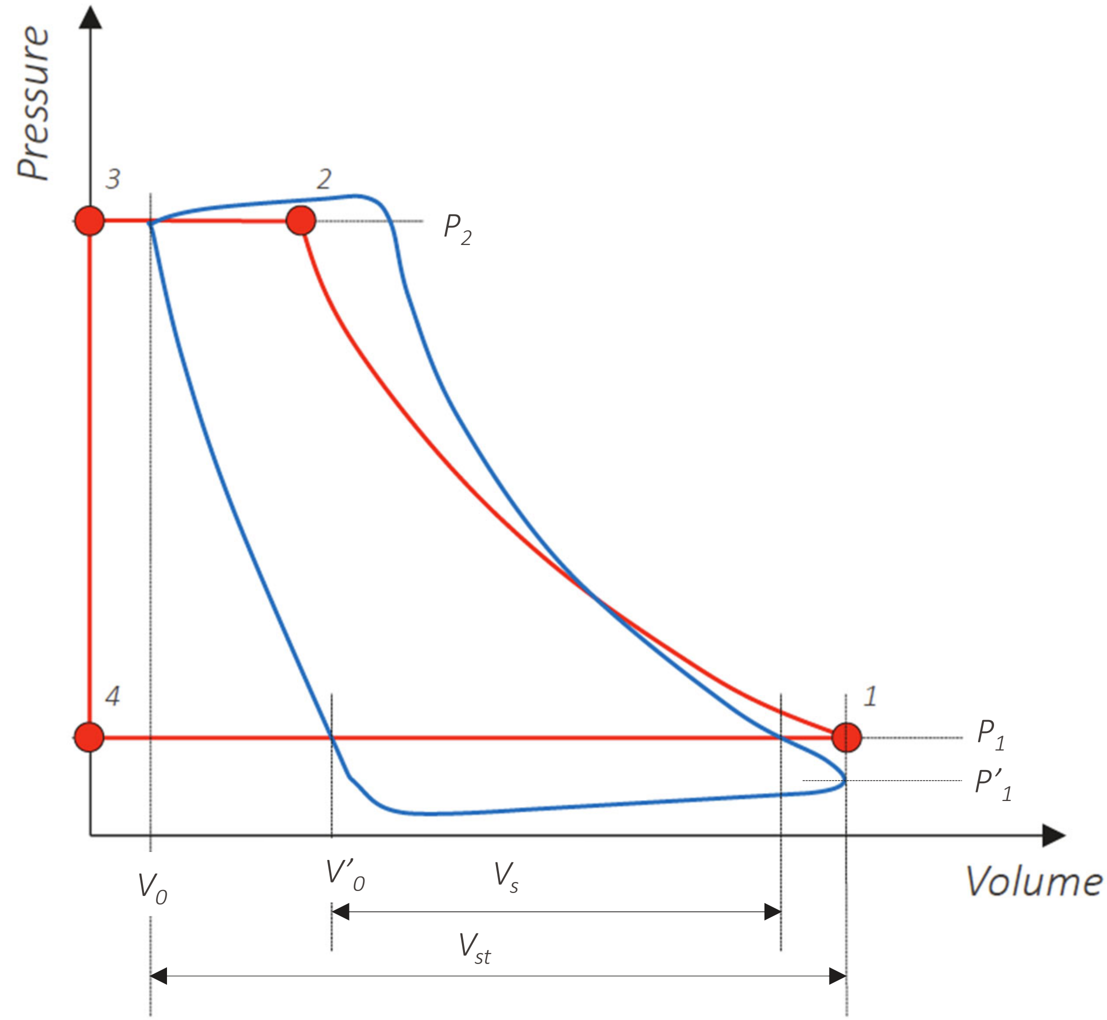
- Real compression deviates from ideal process
- Positive displacement compressor characterized through an isentropic efficiency and a volumetric performance metric



Volumetric Losses

- Clearance volume expands from discharge to suction & reduces suction volume
  - Assume isentropic expansion from P_2 to P_1
- Suction pressure drop leads to suction volume reduction
  - Assume isothermal compr. from P_1' to P_1
- Clearance volume ratio

$$\epsilon = V_0/V_{st}$$



Volumetric Efficiency

- Secondary effects related to mechanical design result in reduced suction volume

- Volumetric efficiency definition

$$\eta_v = \frac{V_s}{V_{st}} \approx \eta_{v1}\eta_{v2}\eta_{v3}$$

Suction losses

Leakage

Clearance volume expansion

$$\dot{V}_{st} = V_{st} f_c$$

Compressor drive frequency

Stroke volume

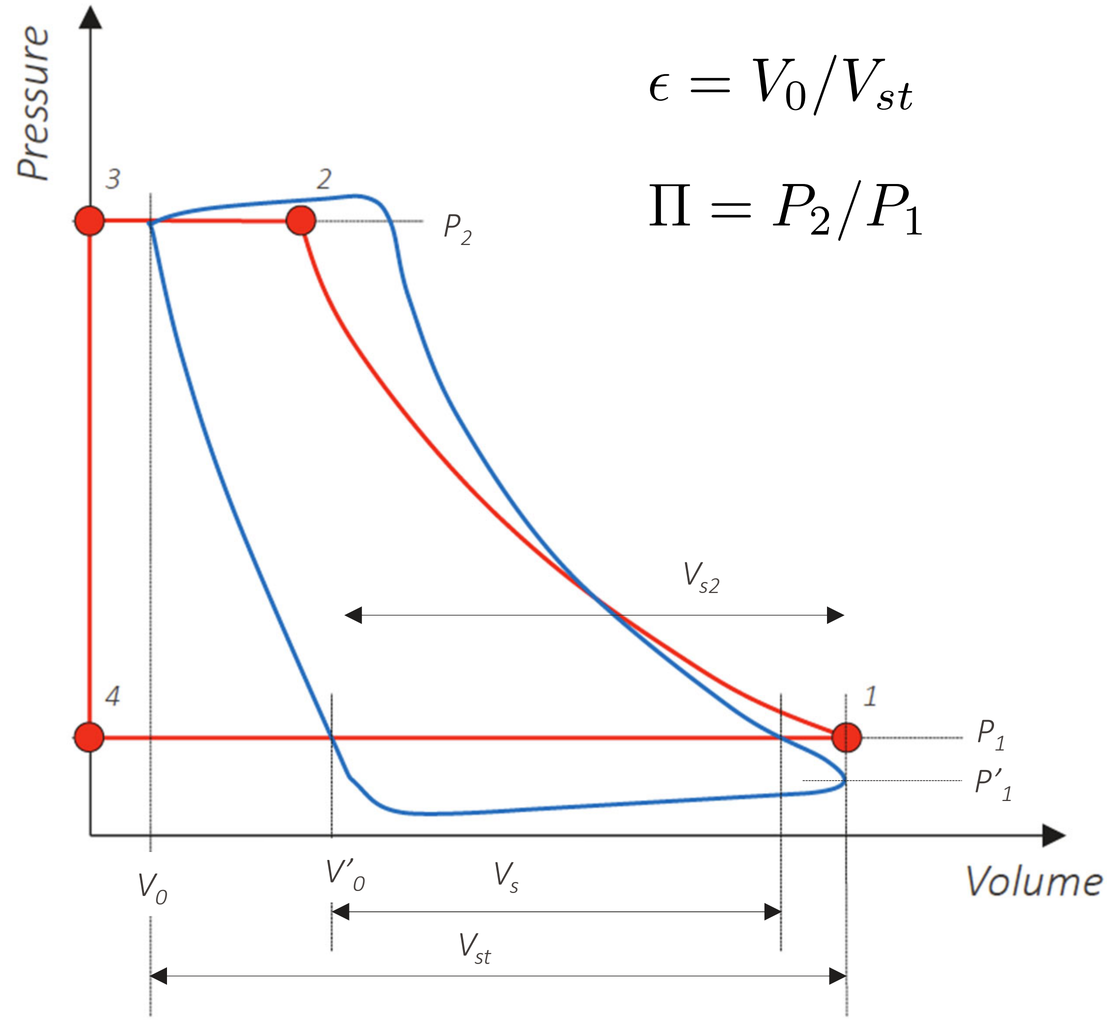
Volumetric Efficiency

$$P_1^{'}\left(V_{st}+V_0\right)=P_1\left(V_s+V_0^{'}\right)$$

$$V_s=\frac{P_1^{'}}{P_1}\left(V_{st}+V_0\right)-V_0^{'}$$

$$V_s=\frac{P_1^{'}}{P_1}V_{st}\left(1+\epsilon\right)-V_{st}\epsilon\Pi^{1/\kappa}$$

$$\eta_v=\frac{V_s}{V_{st}}=\frac{P_1^{'}}{P_1}\left(1+\epsilon\right)-\epsilon\Pi^{1/\kappa}$$



$$\epsilon=V_0/V_{st}$$

$$\Pi=P_2/P_1$$

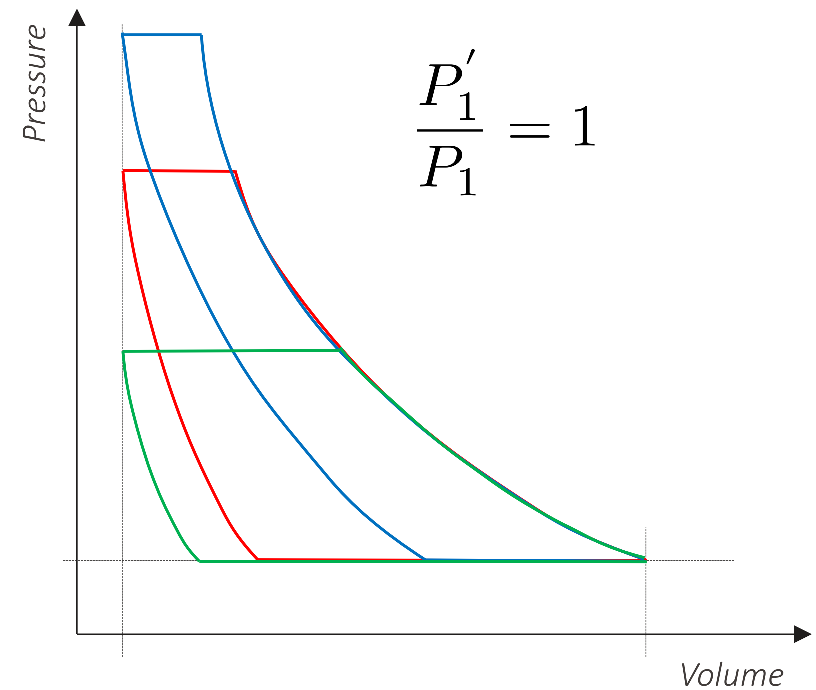
Effect of Clearance Volume Expansion
Isolated Losses on Clearance Volume

- Clearance volume expanding from discharge to suction decreases suction volume with increasing pressure ratio

$$\eta_{v2} = 1 - \frac{V_0}{V_{st}}\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{\kappa}} - 1\right]$$

- Critical pressure ratio above which no mass-flow is delivered → $\eta_{v2} = 0$

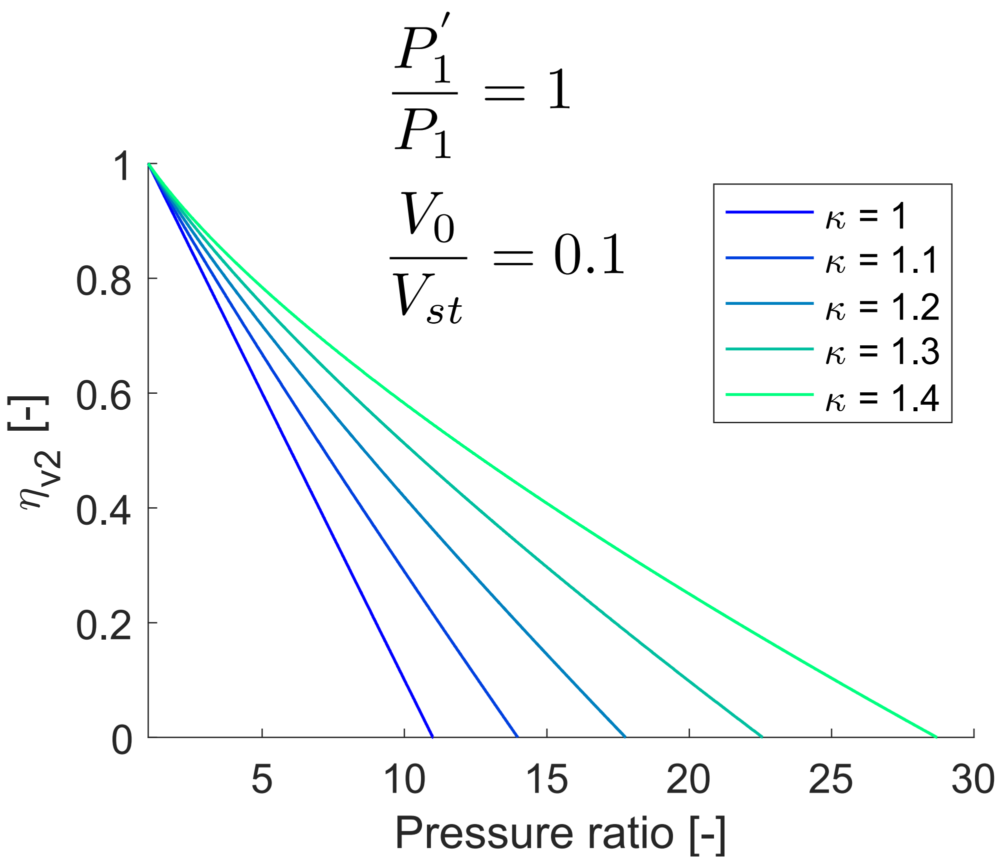
$$\left.\frac{P_2}{P_1}\right|_{Crit} = \left[\frac{V_{st}}{V_0} + 1\right]^{\kappa}$$



$$\frac{P_1'}{P_1} = 1$$

Effect of Clearance Volume Expansion

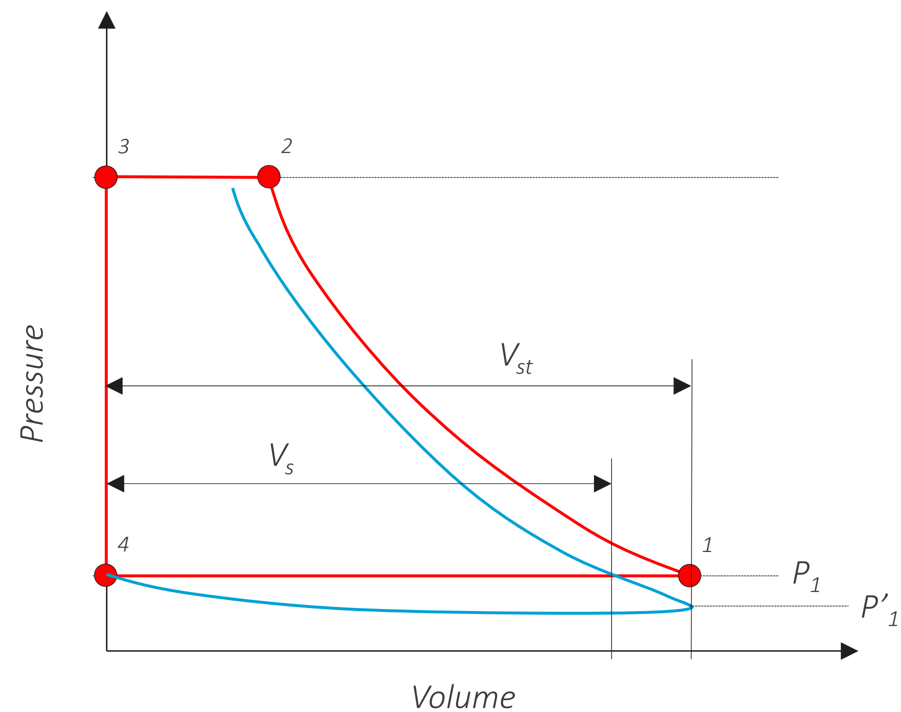
- Volumetric efficiency drops with increasing pressure ratio and with decreasing kappa
- Critical pressure ratio drops with decreasing kappa
- Volumetric efficiency needs to be considered when selecting compressor stroke volume for given application



Volumetric Efficiency
Isolated Losses on Suction Volume

- Suction pressure drop can be linked to suction volume reduction

$$\left.\frac{P_1'}{P_1}\right|_{T=cst} = \frac{V_s}{V_{st}} = \eta_{v1}$$



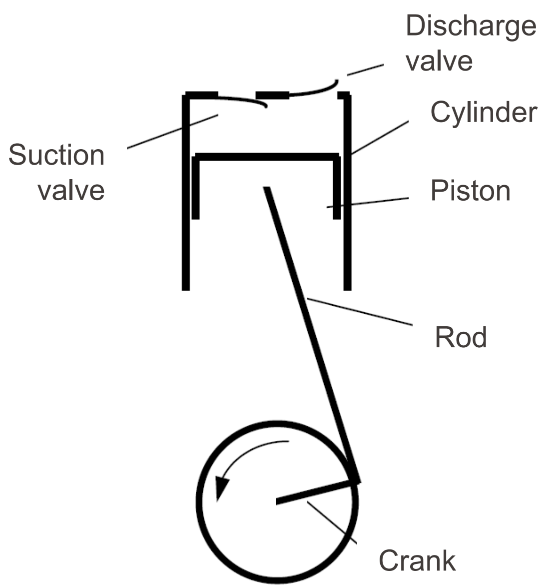
Volumetric Efficiency

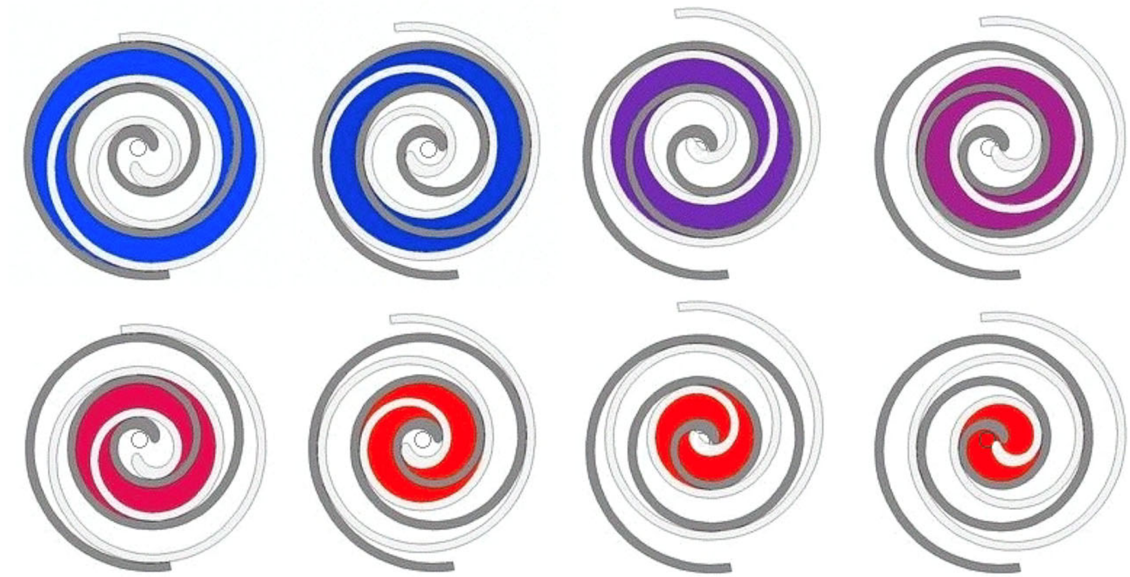
- Overall volumetric efficiency

$$\eta_v = \frac{V_s}{V_{st}} = \frac{P_1^{'}}{P_1}(1+\epsilon) - \epsilon \Pi^{1/\kappa} = \eta_{v1}(1+\epsilon) - \epsilon \Pi^{1/\kappa}$$

$$\eta_v = \eta_{v1}\left[1 - \epsilon\left(\frac{\Pi^{1/\kappa}}{\eta_{v1}} - 1\right)\right] \approx \eta_{v1}\eta_{v2}$$

Piston vs. Fixed Volume Ratio Compressors

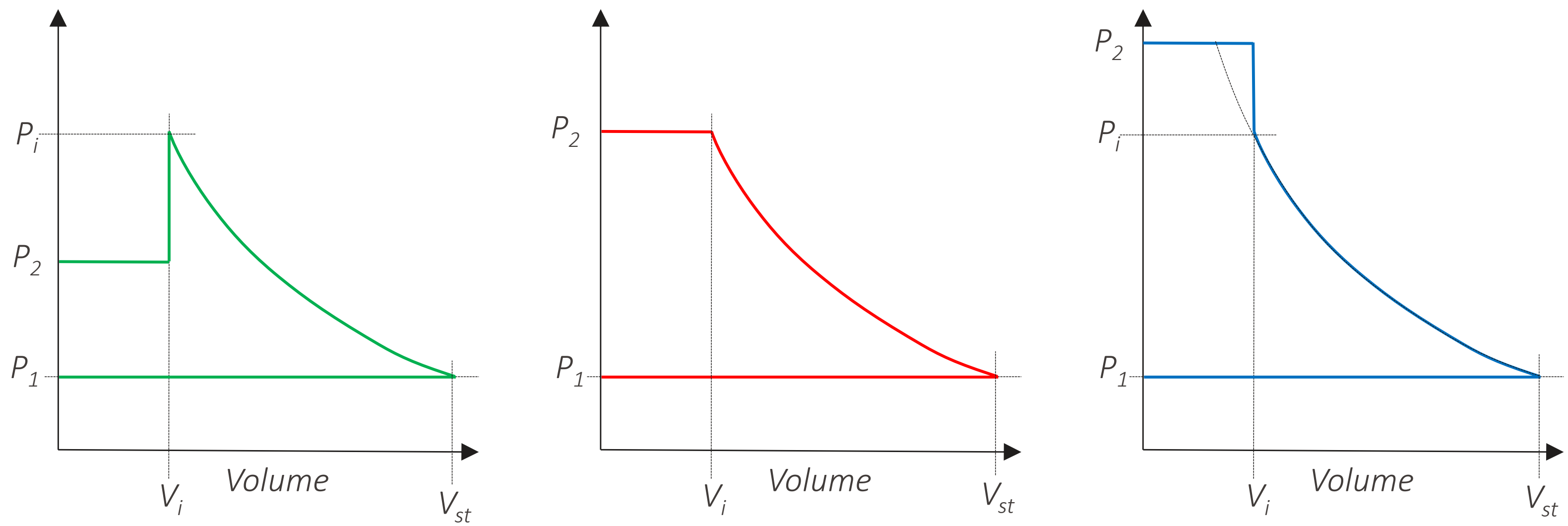




- In piston compressors discharge valve opens only when discharge pressure is achieved in working chamber
- In fixed volume ratio compressor over- & under-compression may occur

Over- & Under-Compression

- Occurs in built-in fixed volume ratio compressors → work surplus



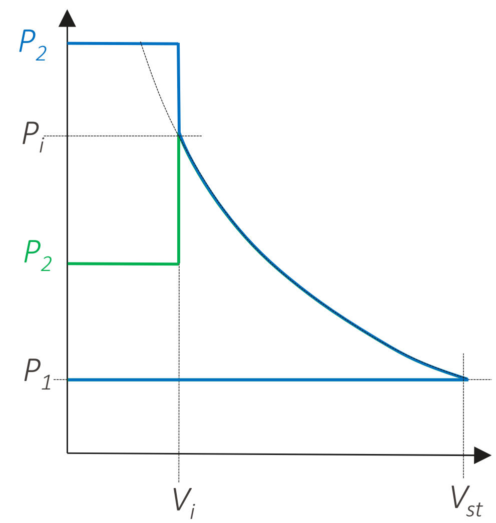
$$V_R = \frac{V_{st}}{V_i} \qquad \Pi = \frac{P_2}{P_1} \qquad \Pi_i = V_R^{\kappa}$$

Over- & Under-Compression

- Expression for theoretical compressor efficiency (perfect gas)

$$\eta_{th} = \frac{\frac{\kappa}{\kappa-1} R T_1 \left[\Pi^{\frac{\kappa-1}{\kappa}} - 1\right]}{\frac{\kappa}{\kappa-1} R T_1 \left[\Pi_i^{\frac{\kappa-1}{\kappa}} - 1\right] + (P_2 - P_i) V_i} = \frac{\kappa \left[\Pi^{\frac{\kappa-1}{\kappa}} - 1\right]}{V_R^{\kappa-1} - \kappa + (\kappa - 1) \frac{\Pi}{V_R}}$$

$$V_R = \frac{V_{st}}{V_i} \qquad \Pi = \frac{P_2}{P_1} \qquad \Pi_i = V_R^{\kappa}$$

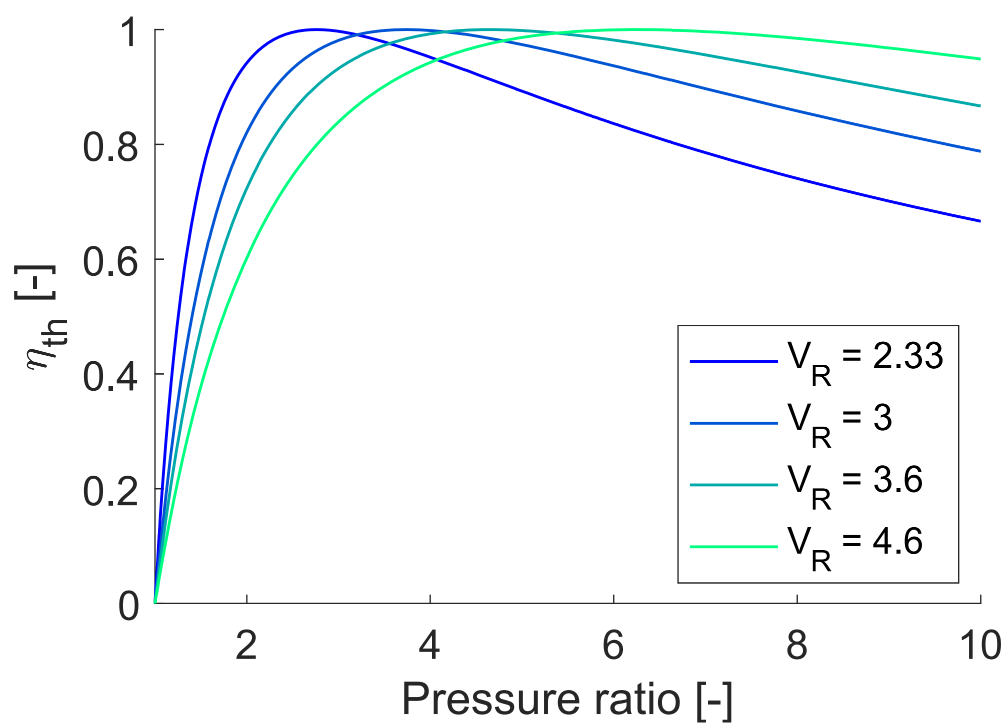


Theoretical Efficiency (Over- & Under-Compression)

- Efficiency drops rapidly in over-compression mode

- Built-in pressure ratio depends on built-in volume ratio and fluid

- Ideal compressor built-in volume ratio depends on operating envelope of heat pump cycle and on working fluid



Isentropic Efficiency

- Electrical efficiency $$\eta_{el} = \frac{e_m^+}{e_{el}^+}$$
- Mechanical efficiency $$\eta_m = \frac{e_f^+}{e_m^+}$$
- Internal isentropic efficiency $$\eta_{is-int} = \frac{e_{is}^+}{e_f^+}$$
- Isentropic efficiency $$\eta_{is} = \eta_{el}\eta_m\eta_{is-int} \approx \eta_{is-max}\eta_{th}$$

Comparison to Measured Data

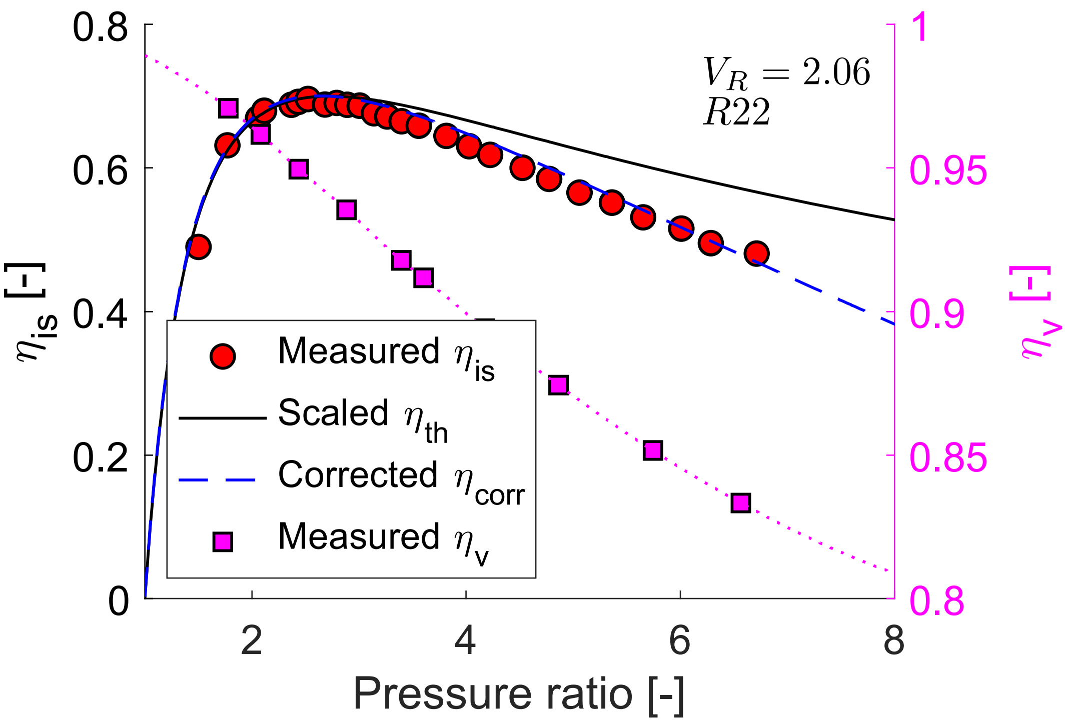
- Experimental scroll compressor data with $\eta_{is-max} = 0.71$

- Scaled theoretical efficiency overpredicted at high Π

- Correction to account for effect of leakage on isentropic efficiency

$$\eta_{is} = \eta_{is-max}\eta_{th} - c\left(1 - \eta_v\right)\Pi$$

$$c = 0.07$$



Summary

- Positive displacement compressors are characterized by a volumetric and an isentropic efficiency
- Volumetric efficiency is decreased by underfilling at suction, expansion of the clearance volume after discharge and by internal leakage
- Compressors with built-in volume ratio may suffer from over- and under-compression, which scaled with the maximum measured isentropic efficiency offers a good predictive model

Heat Pump Systems

Screw & Scroll Compressors

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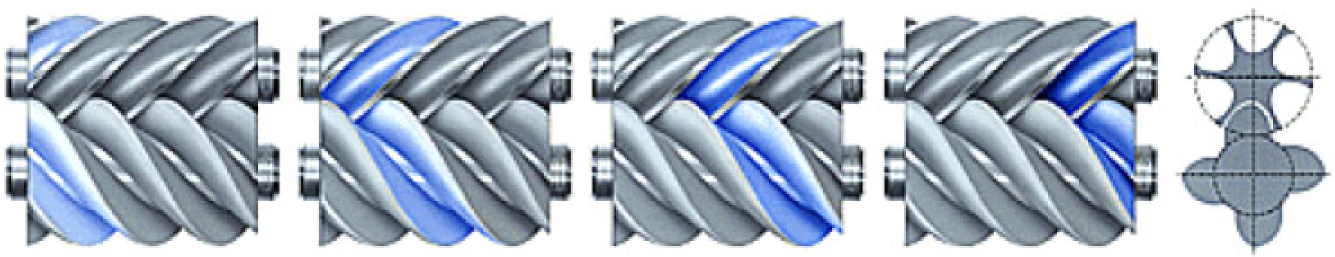
Varying Compressor Operating Conditions

- Heat pumps need to be able to operate under varying conditions
- Changing temperature levels in evaporator and condenser imply varying pressure ratios across compressor
- Changing heating and cooling power at constant temperature levels implies changing mass-flow / volumetric flow delivered by compressor
- Ideal positive displacement compressor should be able to change installed volume ratio and volume flow

Variable Screw Compressor Geometry

- In fixed geometry screw compressors volume ratio and suction volume are defined by geometry of screw rotors and on axial discharge port



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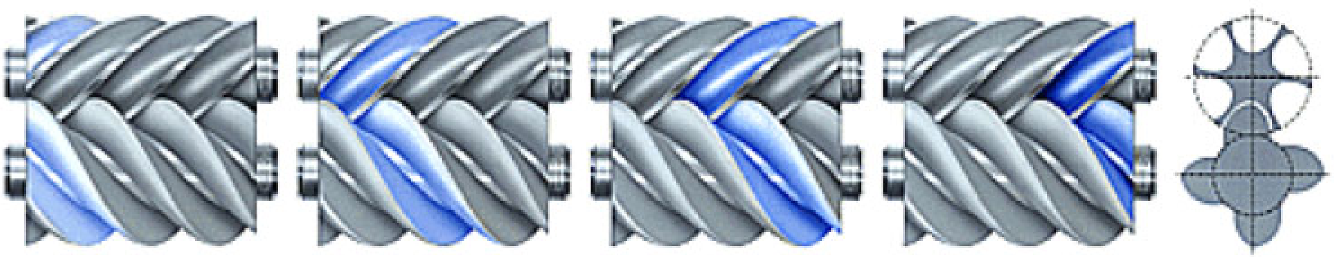

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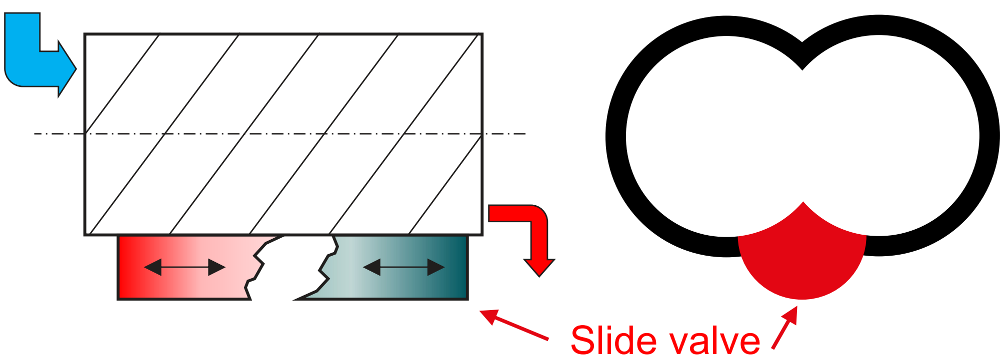
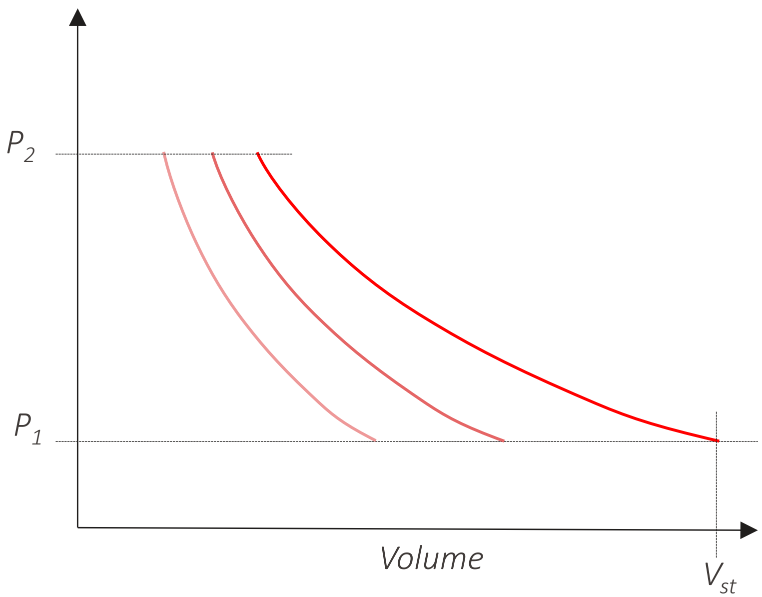
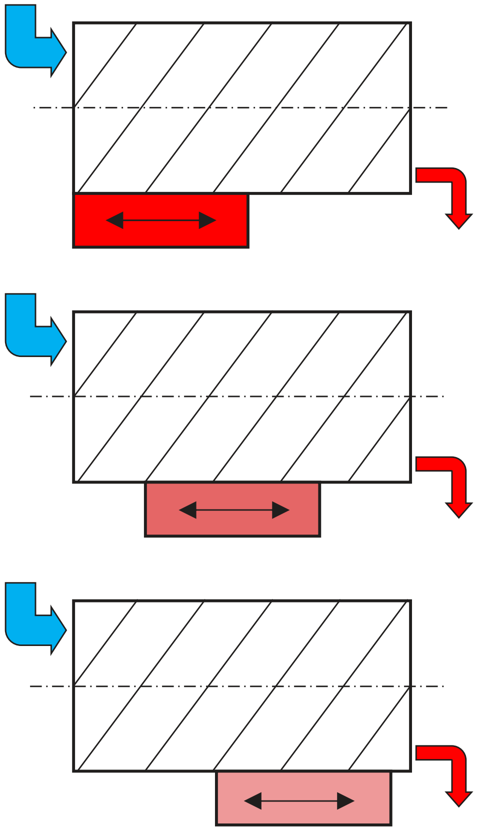


Suction Volume Modulation at Constant V_r

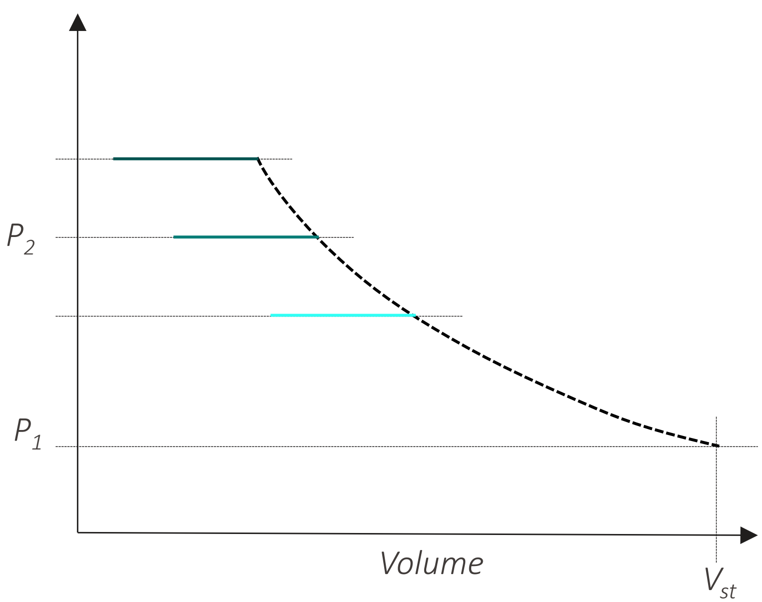
- Slide vane covering constant width yields constant volume ratio while modulating suction volume (→ power modulation)

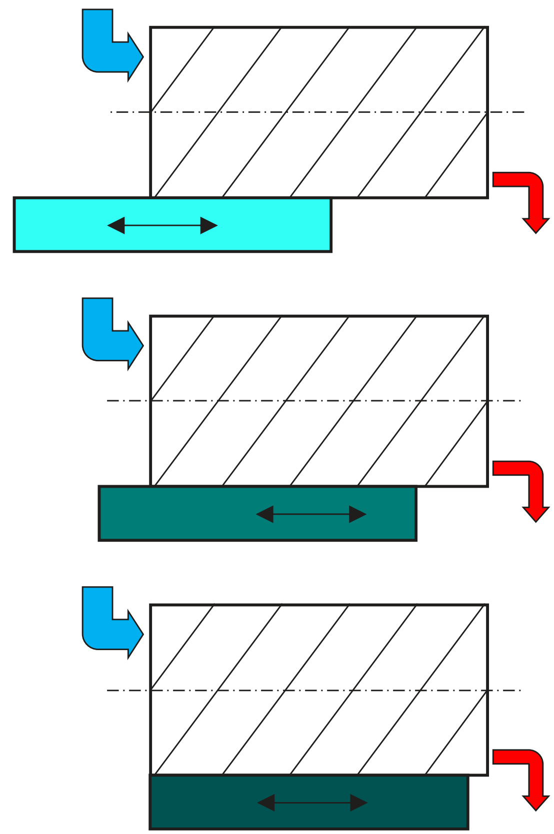




Volume Ratio Modulation at Constant Suction

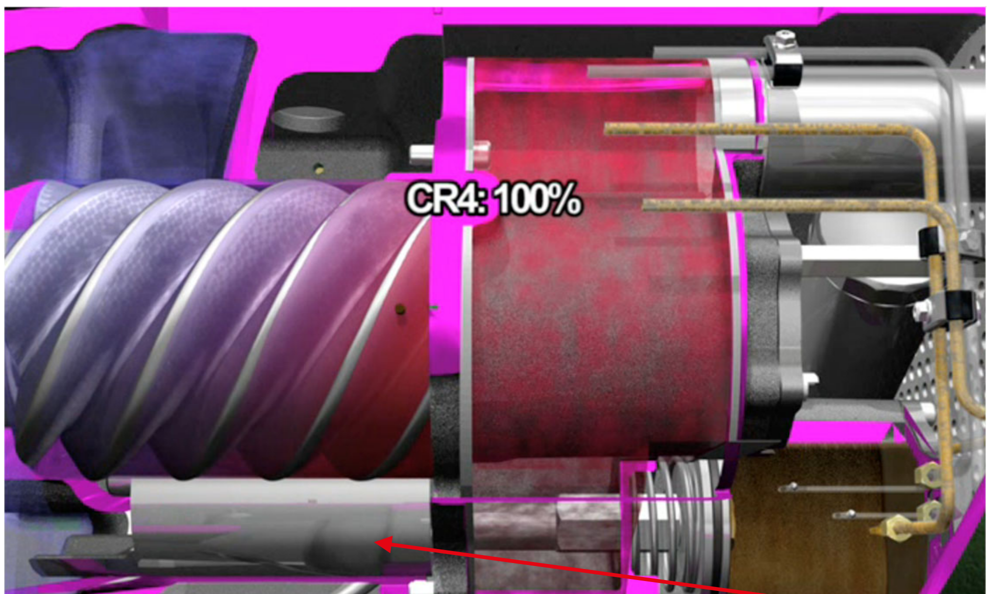
- Long slide valve to modify volume ratio while keeping suction volume constant





Volume Ratio and Suction Volume Modulation

- Long sliding vane with moving edge on the suction side only



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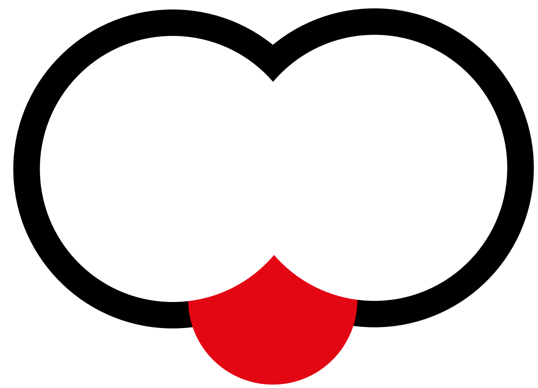
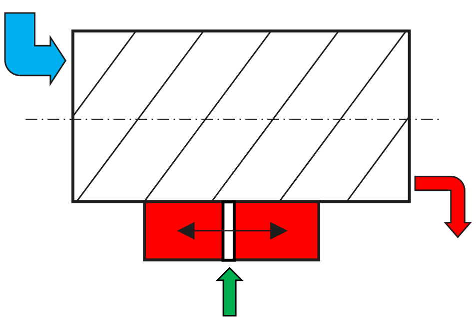
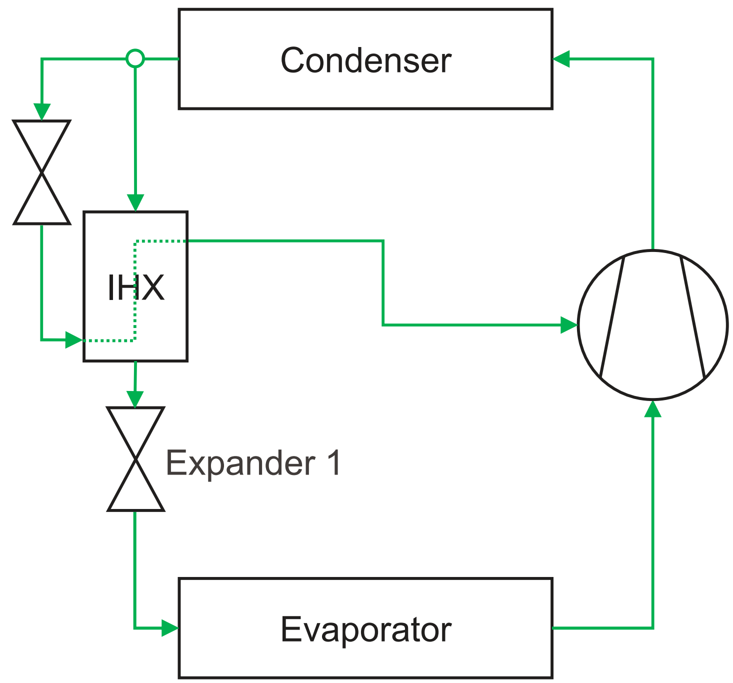


Injected Screw Compressor

- Use single stage compressor with intermediate injection port to avoid investment of two compressors

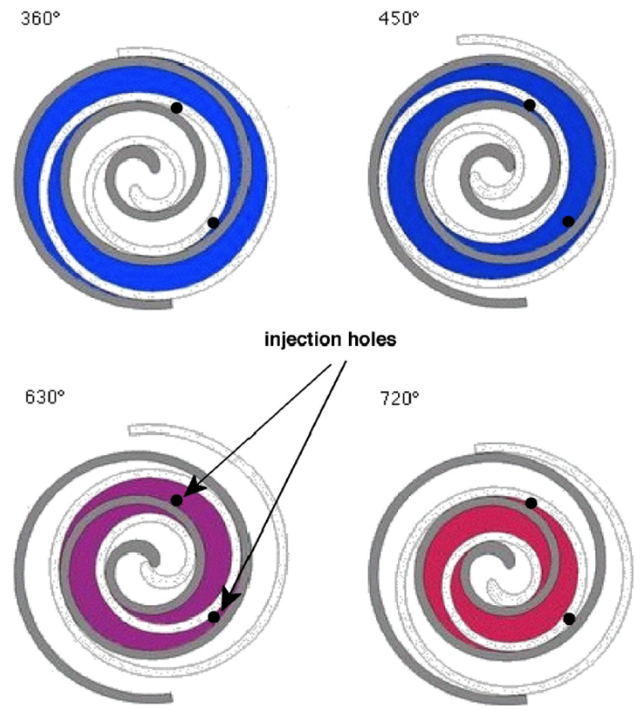
- In screw compressors intermediate injection can be implemented on sliding valve → economizer port





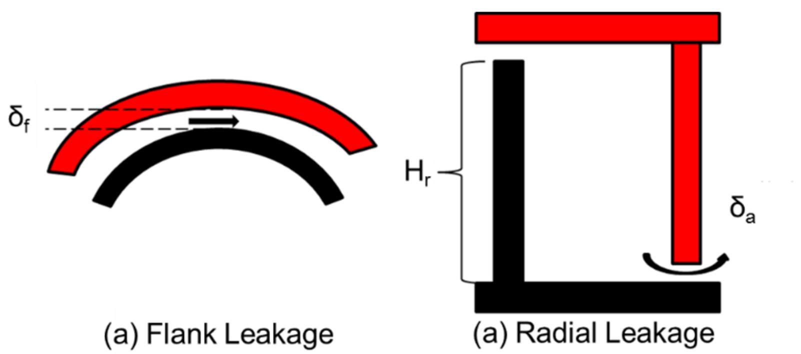
Injected Scroll Compressor

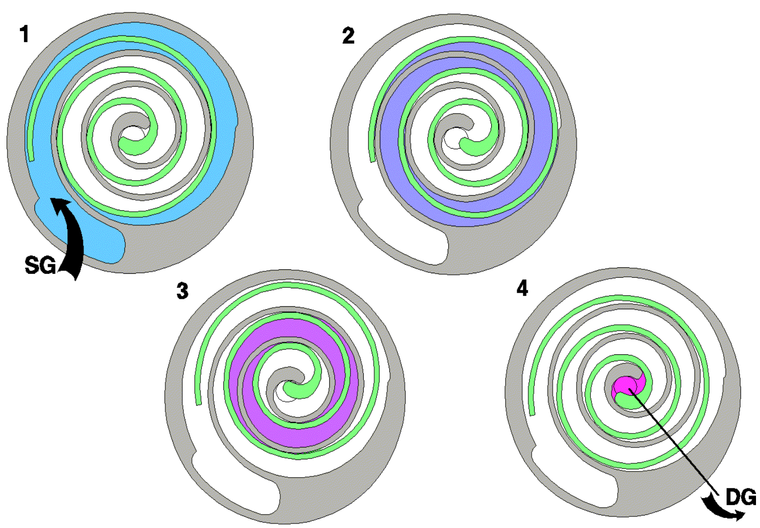
- Scroll compressors can also be designed with economizer ports
- Injection ports located in fixed scroll
- Injection occurs into closed compression chambers after compression initiated



Scroll Compressor Features

- Scroll compressor design prevents direct leakage from discharge to suction pressure → limited losses through internal leakage
- Feature small clearance volume compared to alternative designs





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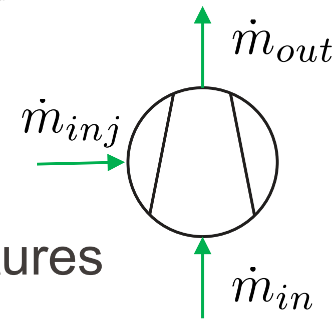
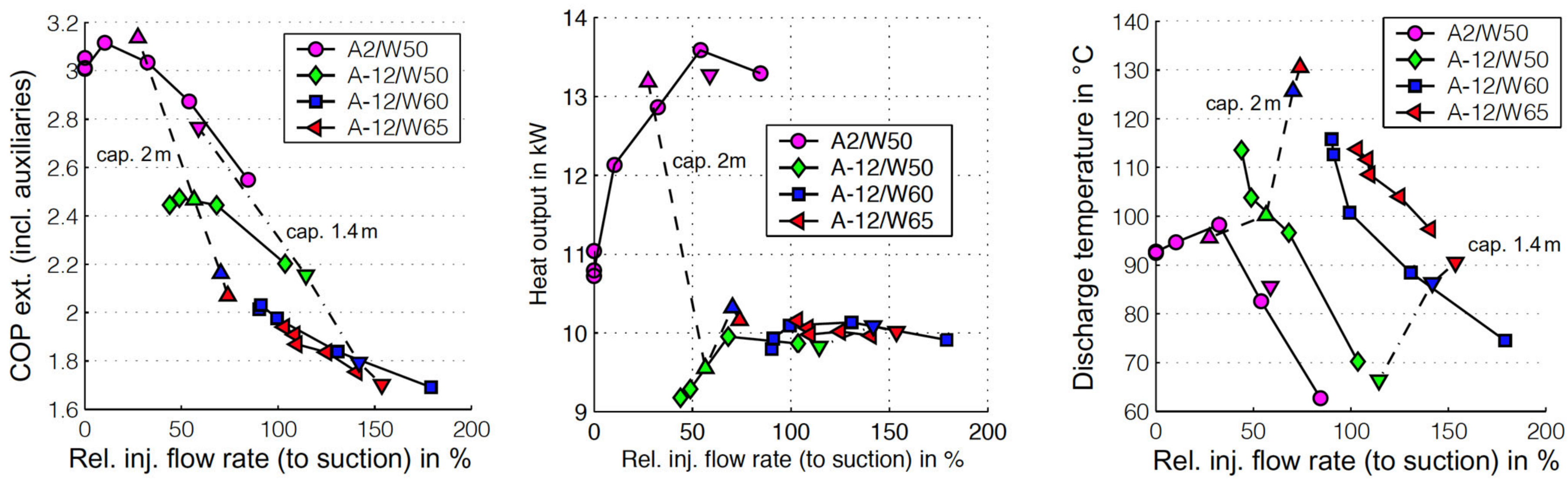
Test Results with Injected Scroll Compressor



- Lifts of 77K achieved with low compressor discharge temperatures
- COP drops with injection rate → injection losses
- Injection means to increase output power



EPFL-Thesis 2998, Michele Zehnder, Efficient air-water heat pumps for high temperature lift residential heating, including oil migration aspects

Summary

- Screw compressors with sliding vane offer added flexibility (variable volume ratio, variable suction volume, intermediate injection)
- Screw compressors preferred at large scale, scrolls for domestic scale
- Scrolls yield fixed volume ratio, mass-flow modulation is achieved with speed control
- For high temperature lifts scroll compressors can be equipped with intermediate economizer injection ports

Exercises W8

- Comprehension questions
- Compressor selection