Advances in ORC expander design

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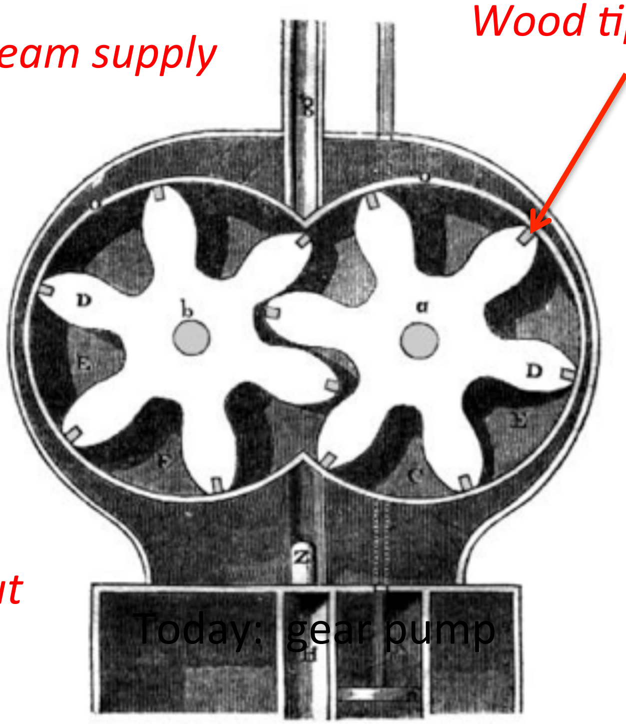
Introduction

Early years of positive displacement machines

The positive displacement machine is not a new idea…



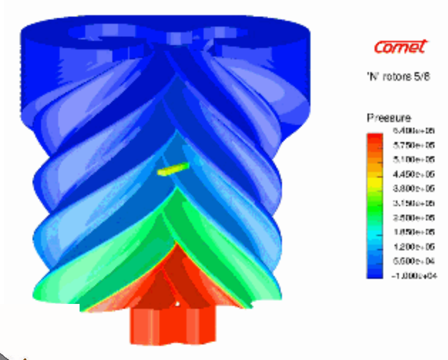
1799: Murdoch steam engine

Source: Museum of retrotechnology

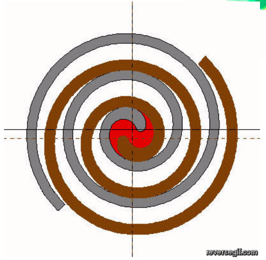
- 1588: vane-type water pump by Ramelli
- 1636: gear water pump in/by Pappenheim
- 1765: James Watts steam engine
- 1799: Murdoch steam engine, should deliver ½ HP but too many leakages
- …

« There is much friction in this kind of engine, much leakage, and but a small power realised in proportion to the size of the machine… » (Bourne, 1858)

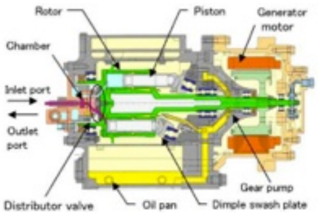
Introduction



- Today, regain of interest for small and medium scale expanders
  - Refrigeration (expansion valve replacement)
  - Stationary power production (steam cycles, ORC)
  - Heat recovery on engines



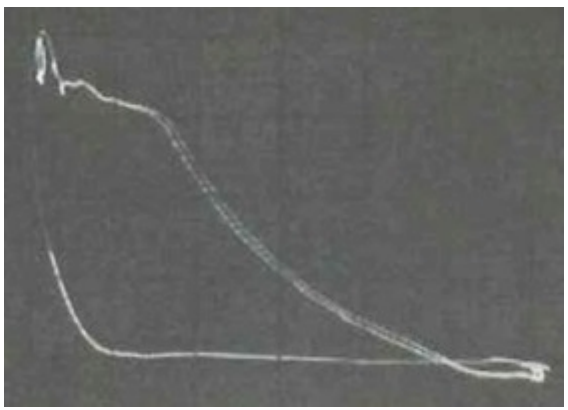
- Purpose of this presentation:
  - ✓ Describe technical constraints inherent to the design of positive displacement machines (piston, screw and scroll)
  - ✓ Present some modeling techniques
  - ✓ Discuss about the selection of expansion machines
  - ✓ Stress some relevant R&D trends related to positive displacement machines in order to improve the energy performance of both the expander and the ORC system



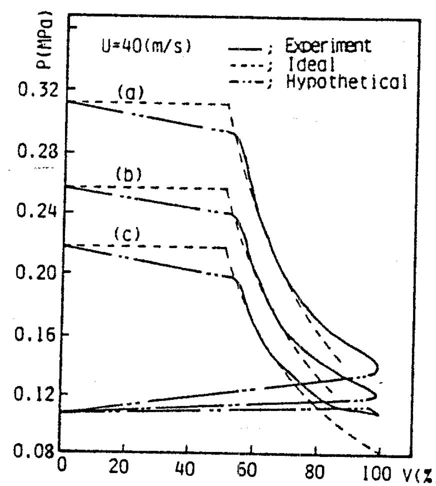
Technical constraints

Losses in positive displacement expanders

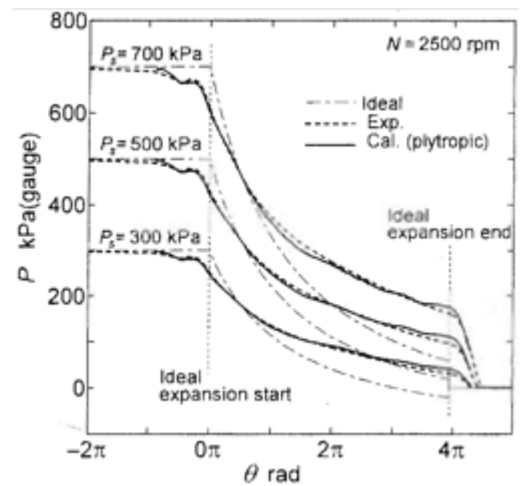
- Supply/discharge pressure losses, under-over expansion, leakages, heat transfer, (clearance volume), mechanical losses



Piston
(Doyle et al., 1970)



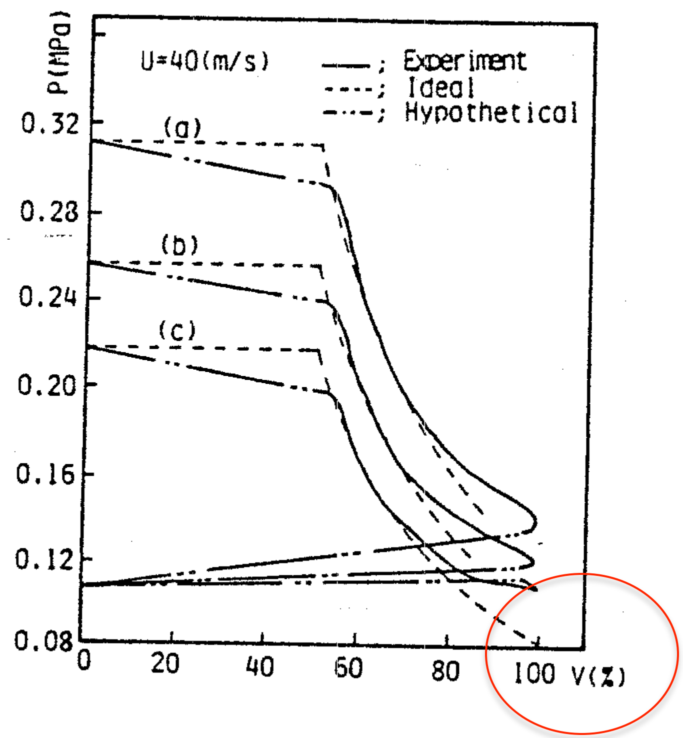
Screw
(Kaneko and Hirayama, 1985)



Scroll
(Yanagisawa et al., 2001)

Technical constraints

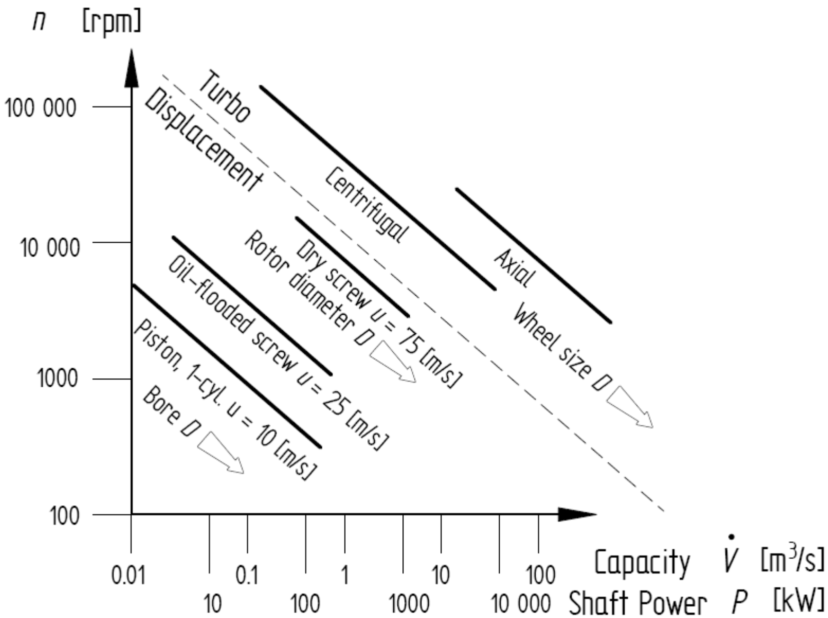
Capacity



displacement

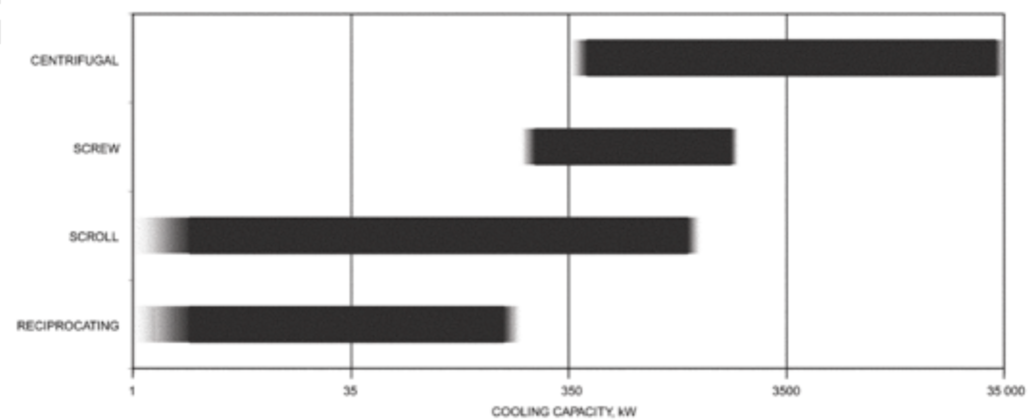
Technical constraints

Capacity



Source: Grip, 2009

- A category of machine is characterized by an optimal tip speed (m/s) fairly independent of its size (Persson, 1990).
- Turbines operate at larger tip speed than displacement machines.



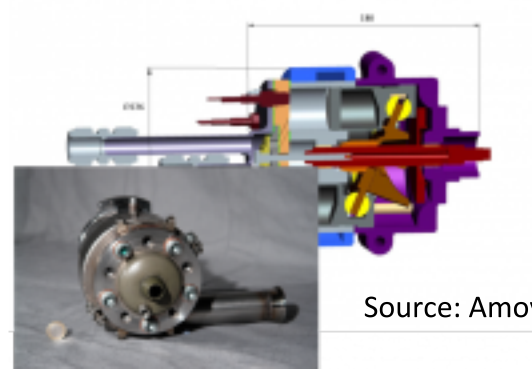
Approximate range of chiller cooling capacity range by compressor type (ASHRAE, 2008)

Technical constraints

Capacity

Piston expanders

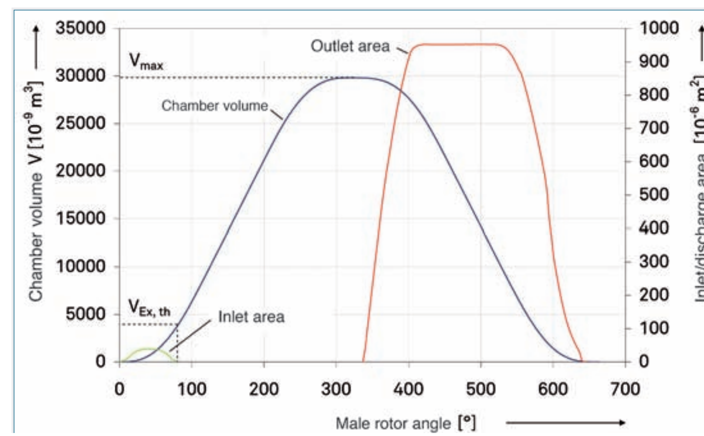
- Currently used for **small-scale** CHP and waste heat recovery on ICE (niche market tech.)
- **Axial** expanders: compact, low vibrations (MAC)
- Displacement: approx 1.25 to 75 l/s



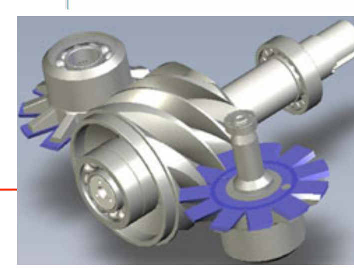
Source: Amovis, 2012

Screw expanders

- From 20 kWe to 1 MWe (In comp: 200W-2/3 MWe)
- Displacement: approx 25 to 1100 l/s
- Few records on micro-screw expanders (volumetric perf.!) $\varepsilon_{v,cp} \div 1 - \frac{cste}{N.D}$
- Mainly twin-screw (except BEP, Wang et al., 2011)
- Only rotating elements: **high rotational speeds** (21,000 rpm recorded)



Source: Brümmer, 2012



Source: BEP

Technical constraints

Capacity

Scroll expanders

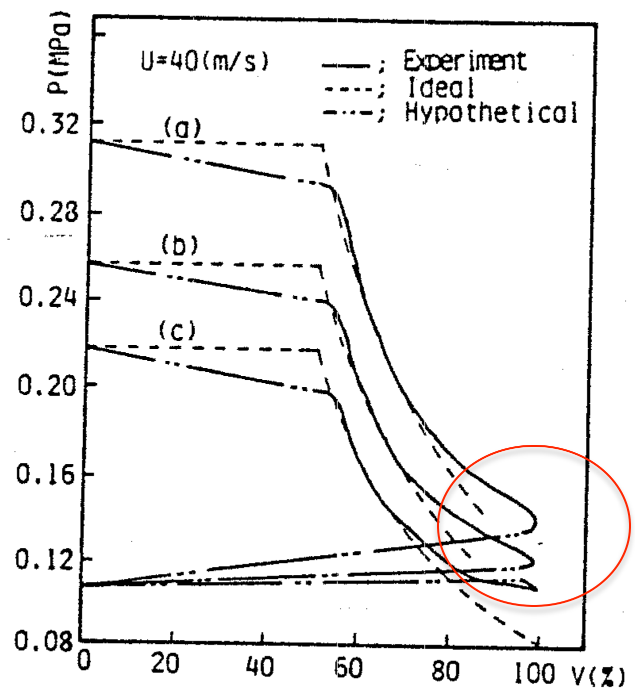
- From mini-scroll to very large machines
- Currently, the trend in the compressor industry is to **increase the size** (f.i. Emerson ZP725K with 158 kW cooling capacity)
- Large scroll compressors compete with small screw machines
- **Multi scroll ORCs** have not been investigated yet (except Eneftech): modulation, PL performance
- Speed:
  - Mobile A/C: up to 10,000 rpm
  - Permanent magnet generators: variak efficiency

60 HP 7.5 HP

Source: Emerson, 2012

Technical constraints

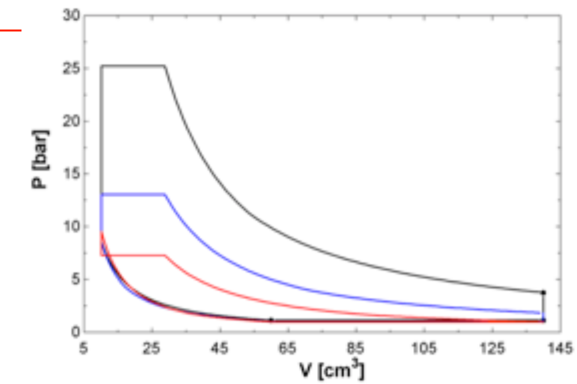
Built-in volume ratio



Over and under expansion losses

Technical constraints

Built-in volume ratio

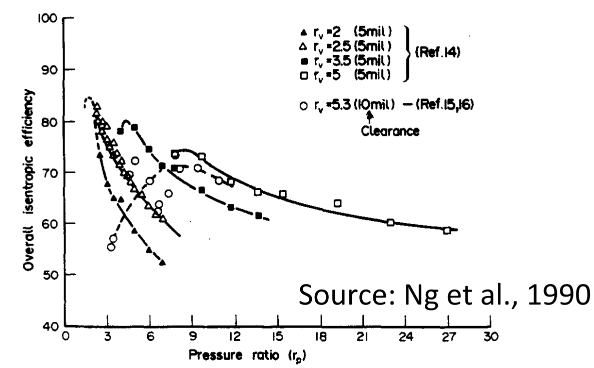


Piston expanders

- Could be large (f.i. internal combustion engines)
- Limited by the specific work of the machine (compactness).
- In practice, between **6-14**

Screw

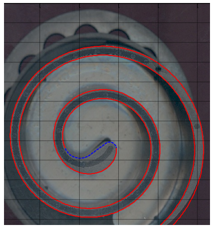
- Up to **5.0** in practice. 8.0 is recorded (Brümmer, 2012)
- Moderate volume ratio expanders show better performance
- Stronger impact of leakage in expander mode.



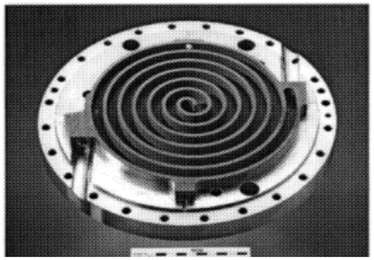
Source: Ng et al., 1990

Scroll

- From 1.5 to 3.5 (HVAC&R), **4.0** (air compressors)
- Results from cost and performance considerations:
  - Performance: number of pairs of sealing points limited to 2 or 3, friction
  - Cost: prohibitive scroll length, compactness



Source: Lemort, 2009

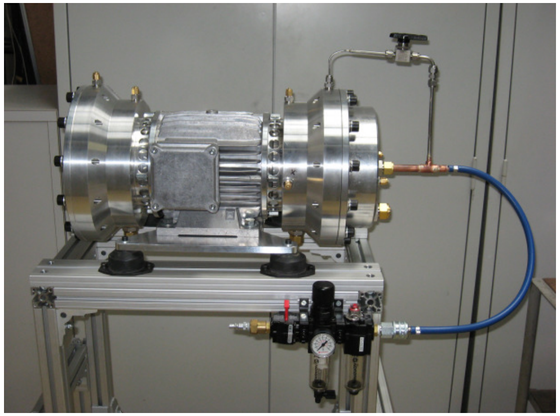


Source: Manzagol et al., 2002

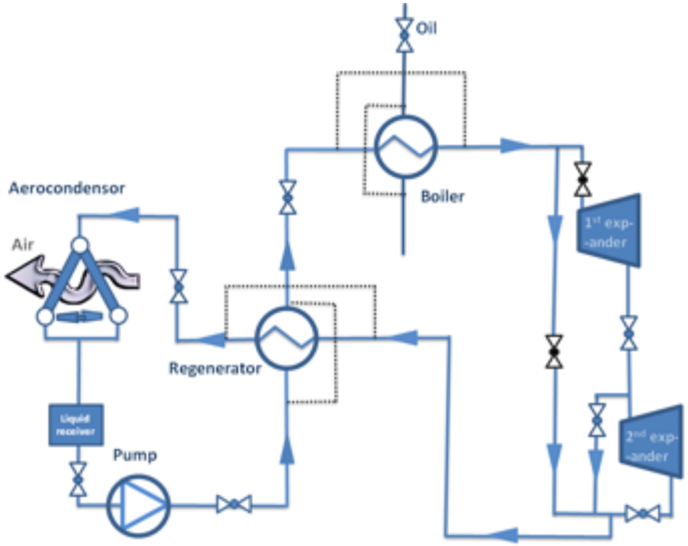
Technical constraints

Built-in volume ratio

- Increase volume ratio?
  - <u>Two-stage expansion</u>:
  - Experience from Kane et al.(2009): $r_{v,in}$: 4.2x3.0, high speed permanent magnet generator (6000 rpm), 5 kWe (steam, 250°C, 25/1 bar)
  - Optimal intermediate pressure (Quoilin, 2011)



Source: Kane et al., 2009

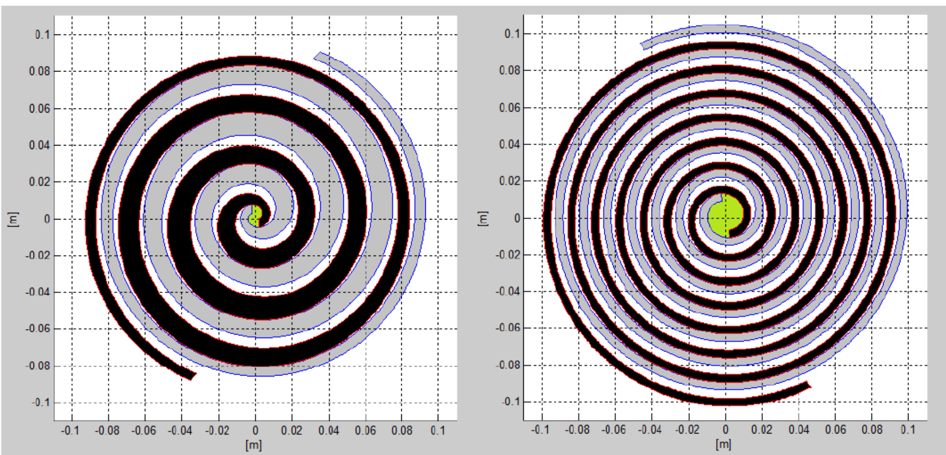


<u>2 hermetic scrolls in series:</u>
Net power: 3 kWe, efficiency ORC: 10-12%, Tev=141C and Tcd=35C

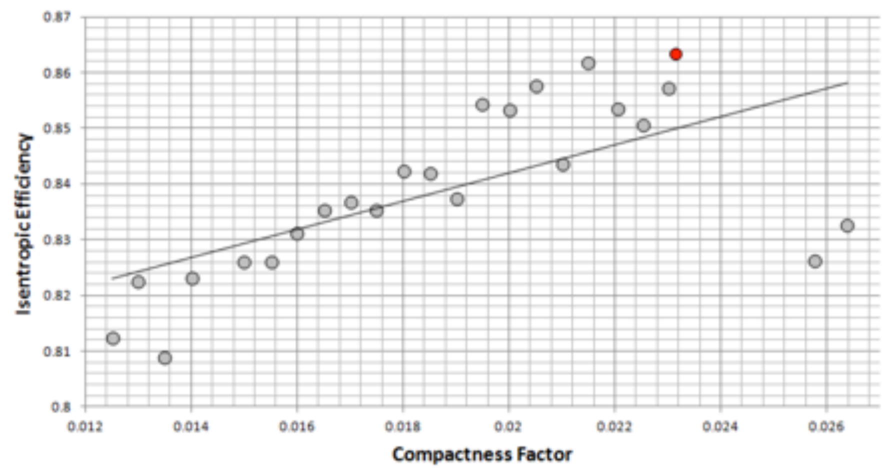
Technical constraints

Built-in volume ratio

➢ Variable wall thickness scrolls: allows to generate more compact geometries. Hence, higher volume ratio can be achieved in practice (MIT/ULg)



Comparison of a variable (left) and a constant (right) wall thickness geometries with the same volume ratio (r_v = 5)

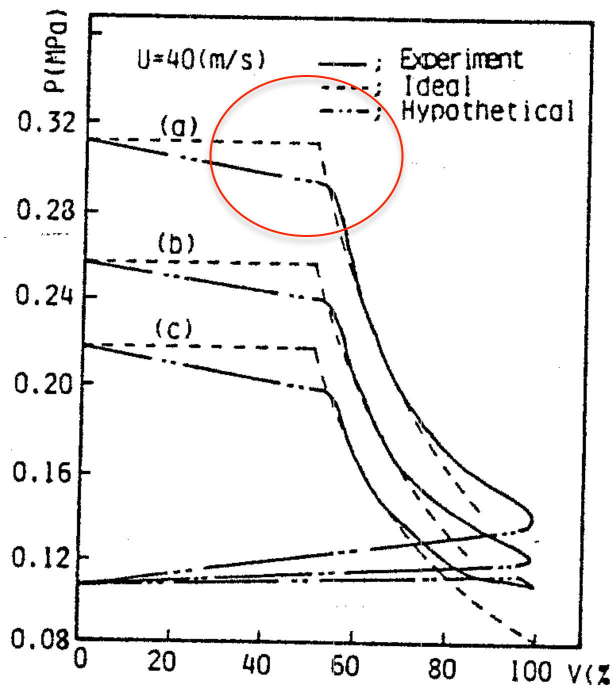


Isentropic efficiency as a function of the compactness factor

Source: Dechesne, 2012

Technical constraints

Supply pressure losses



Technical constraints

Pressure losses

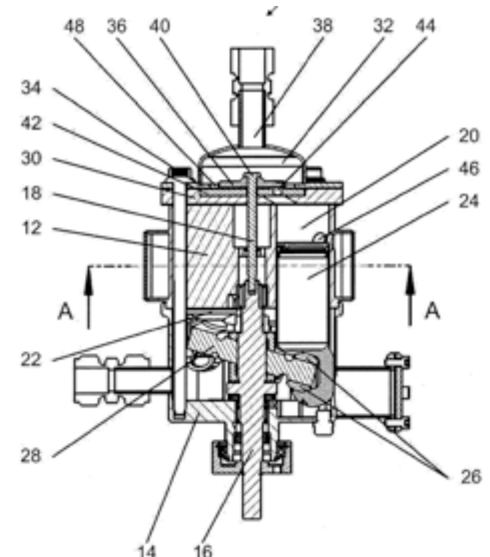
Piston expanders

Inlet valves:

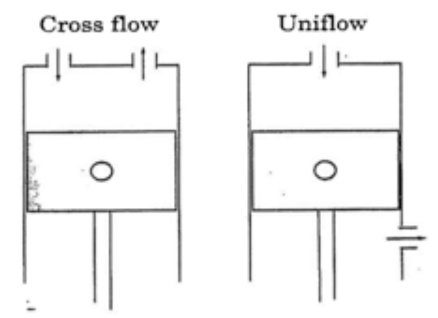
- Source of **pressure losses**
- different families of inlet valves: *Sliding valves*, *Rotating valves*, *Poppet valves*

Outlet valves:

- F.i., use of exhaust ports (vs valves) allows for
  - thermal "coherence" (cross flow vs uniflow)
  - Recover leakages through piston rings
- Use of exhaust ports will lead to larger compression work (fluid is recompressed earlier) and lower fluid mass flow.



Source: Amovis, 2006

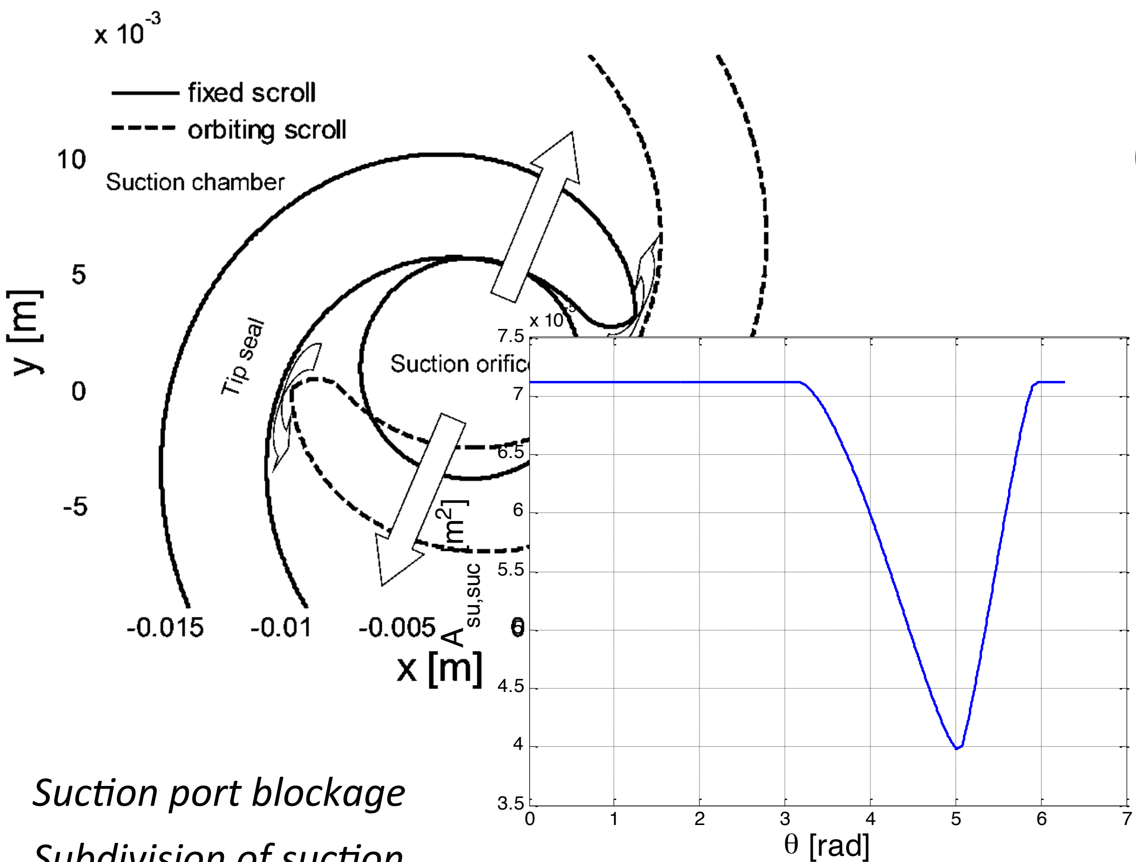


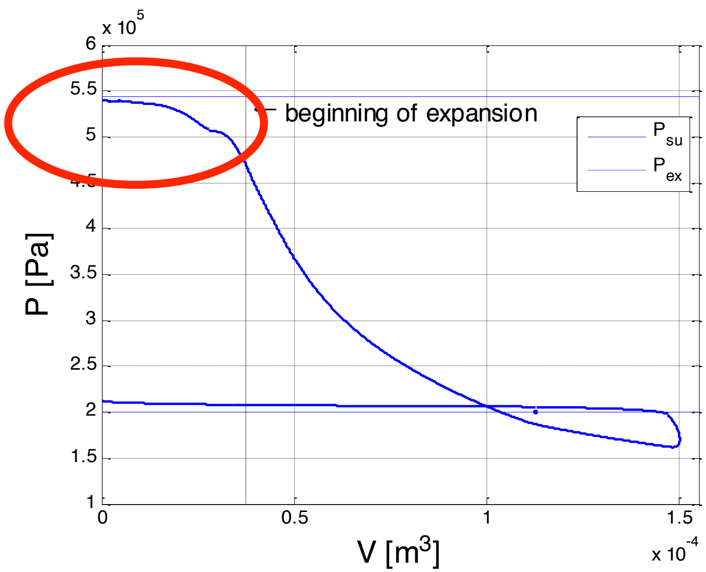
Source: Platell, 1993

Technical constraints

Supply pressure losses

Scroll expanders





Suction port blockage

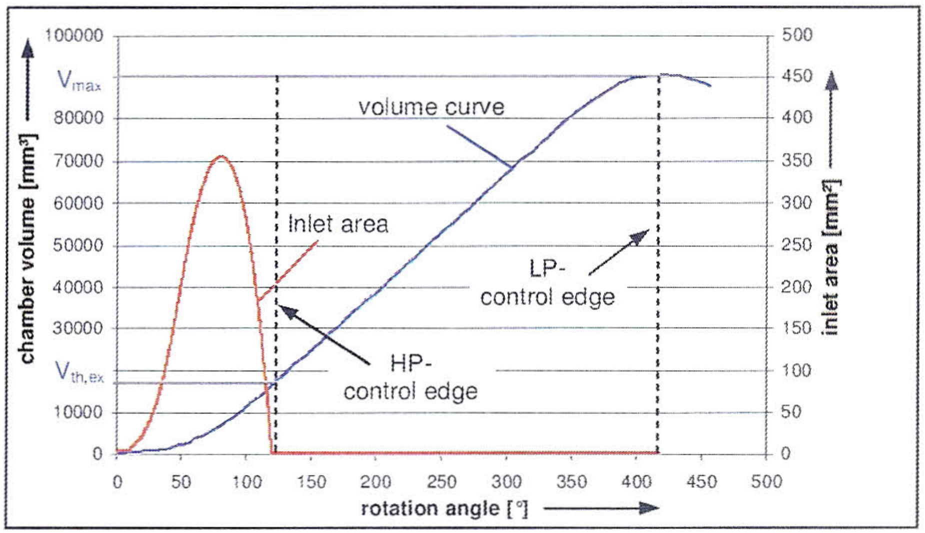
Subdivision of suction chamber (end of suction)

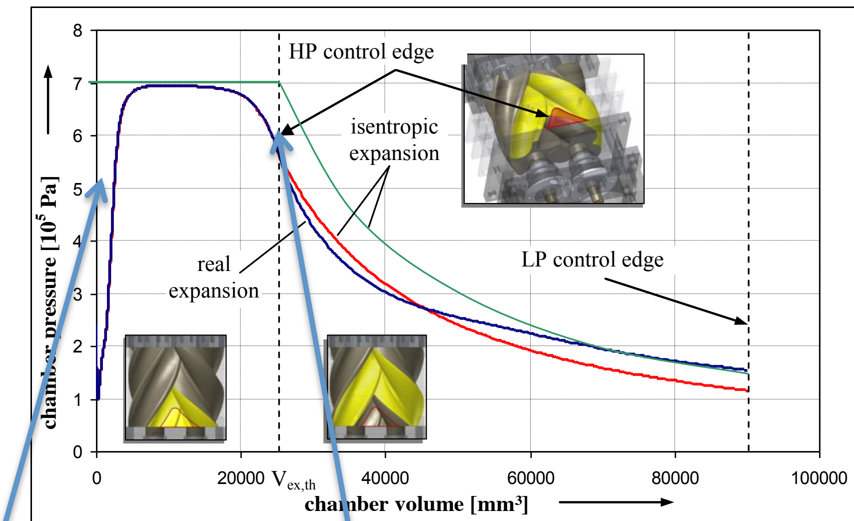
(Yanagisawa et al., 2001)

Technical constraints

Supply pressure losses

Screw expanders





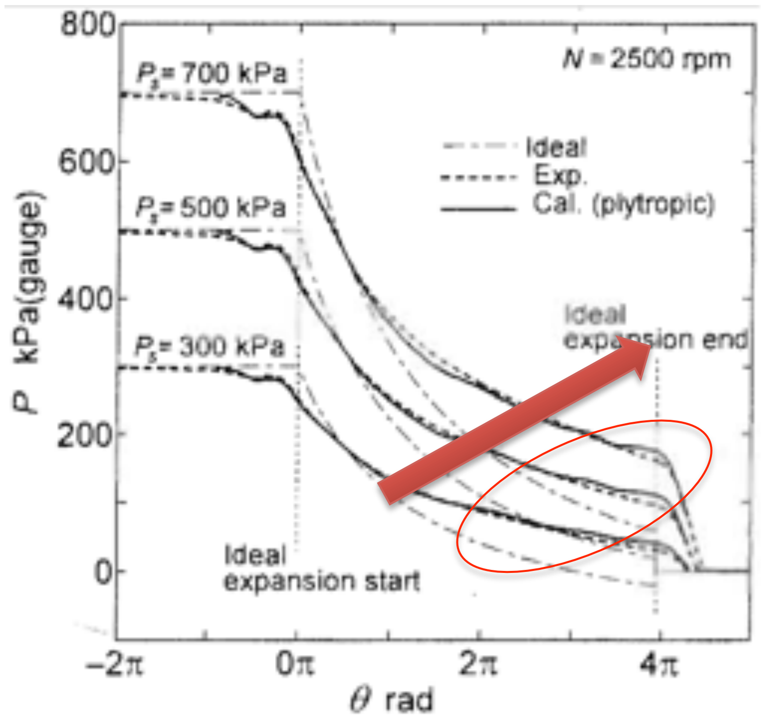
Source: Brümmer, 2011 and 2012

Supply port cross sectional area and volume start from zero (choked flow)

Supply port cross sectional area shrinks to zero and volume chamber is increasing

Technical constraints

Leakages



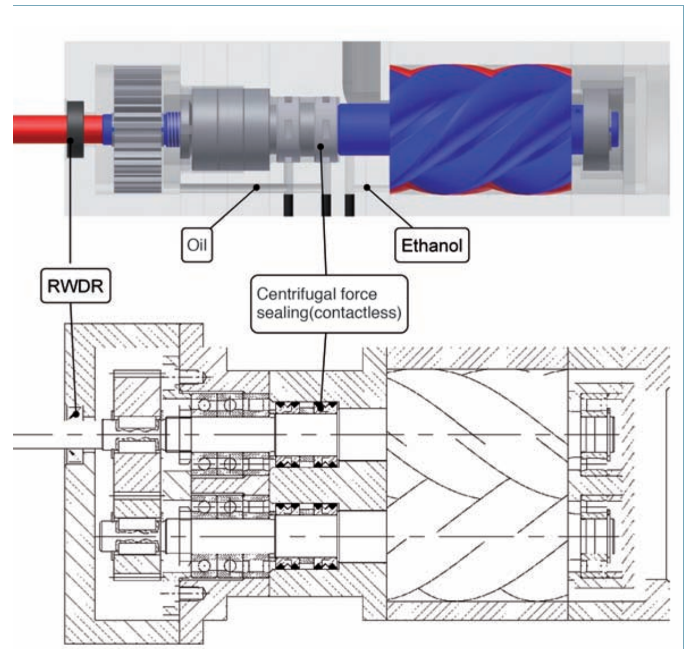
Deformation of P-V diagram

Technical considerations

Leakages and lubrication

Screw

- Oil used for lubrication and clearance gap sealing
- Synchronized machines can be oil-free
    - Unsynchronized
        - ✓ Approx. 90% of the sold screw compressors (Brümmer, 2012)
        - ✓ Oil injected for rotors and bearing lubrication (+sealing)
        - ✓ Better energy performance
        - ✓ Simpler, lighter, smaller size
    - Synchronized:
        - ✓ **Higher tip speed**: 60-120 m/s against 20-40 m/s for lubricated machines (Brümmer, 2012)
        - ✓ **No oil** (gears are greased lubricated)



Source: Brummer, 2012

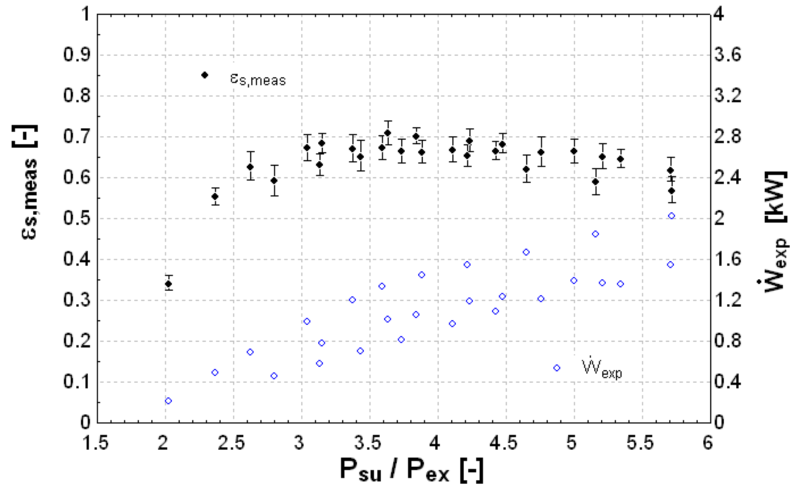
Technical considerations

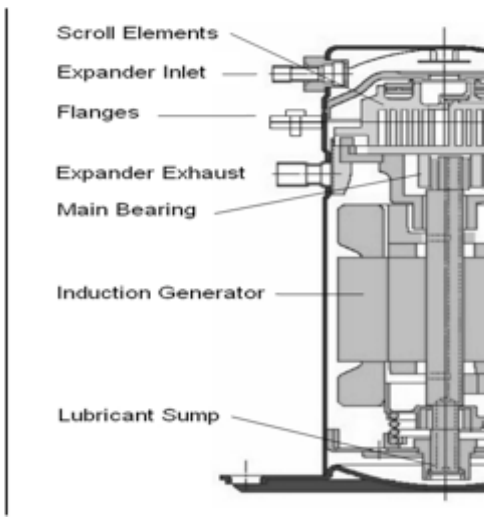
Leakages and lubrication

Scroll

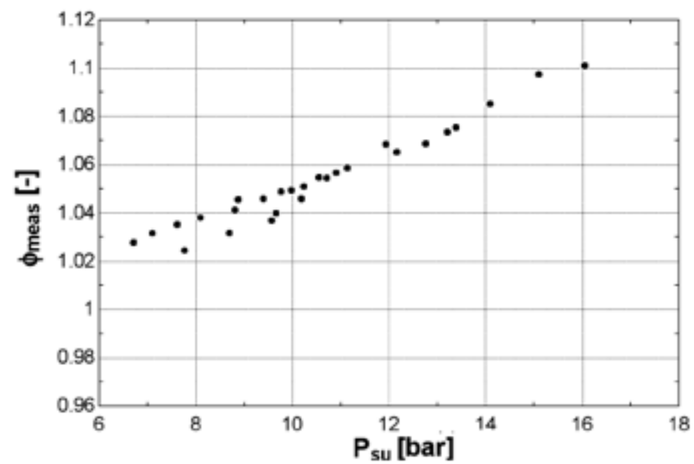
- <u>Refrigerant</u> compressors (hermetic or open-drive) are lubricated
- Few records on refrigerant oil-free machines.

$$\varepsilon_s = \frac{\dot{W}}{\dot{M}.(h_{su}-h_{ex,s})} = \frac{\dot{M}.(h_{su}-h_{ex})-\dot{Q}_{amb}}{\dot{M}.(h_{su}-h_{ex,s})} = \frac{h_{su}-h_{ex}}{h_{su}-h_{ex,s}} - \frac{\dot{Q}_{amb}}{\dot{M}.(h_{su}-h_{ex,s})}$$





$$\varphi = \frac{\dot{M}\,v_{su}}{\dot{V}_s}$$

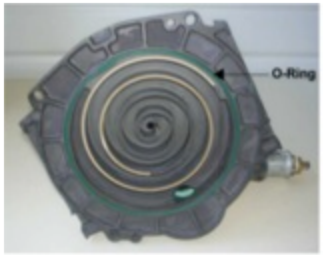


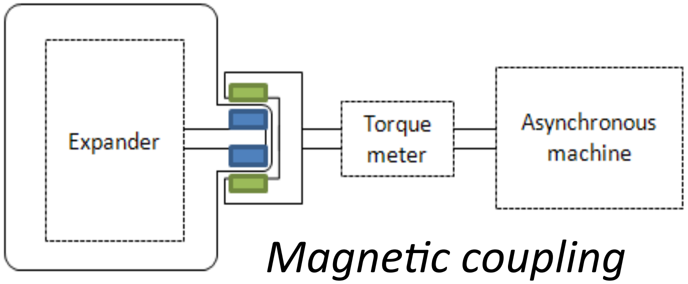
(source: Lemort et al., 2012)

Technical considerations

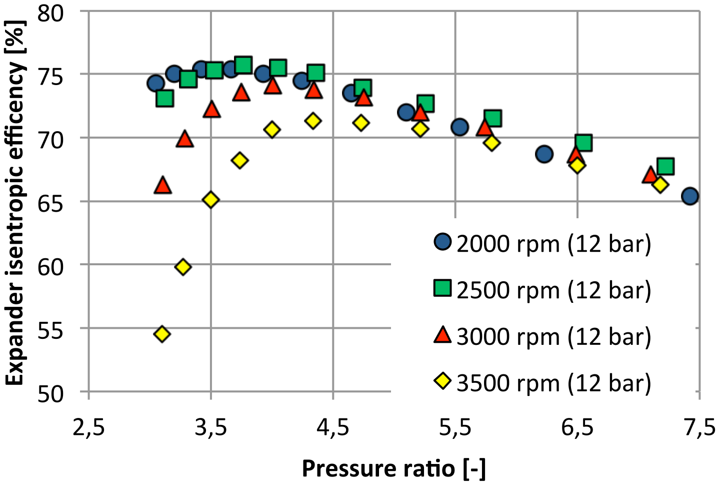
Leakages and ubrication

- <u>Air</u> compressors (open drive, kinematically rigid) can be oil-free. Use of a **tip seal**: limited operating life.
- When retrofitted as expanders: (-) tightness, volumetric performance





Magnetic coupling



Source: Declaye et al., 2012

- Solution proposed by Air Squared: 10 kWe, 155 cc/rev, rvin: 5.25, magnetic coupling, dry/lubricated

Open-drive oil-free machine (R245fa)

Technical considerations

Leakages and design

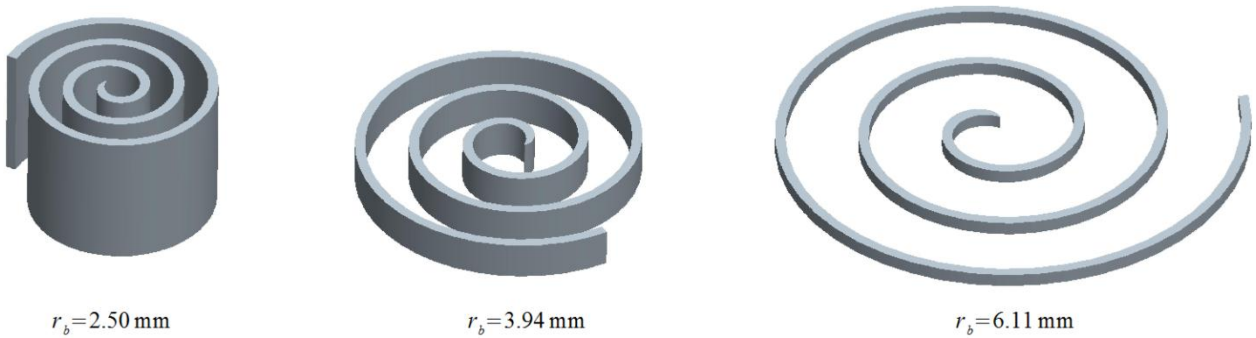
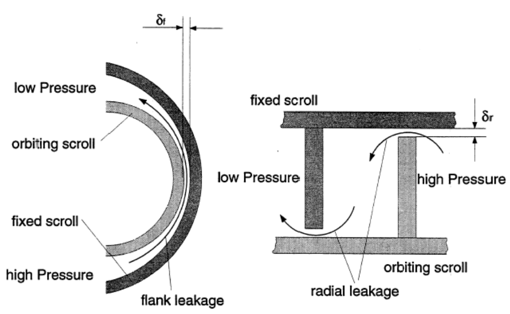
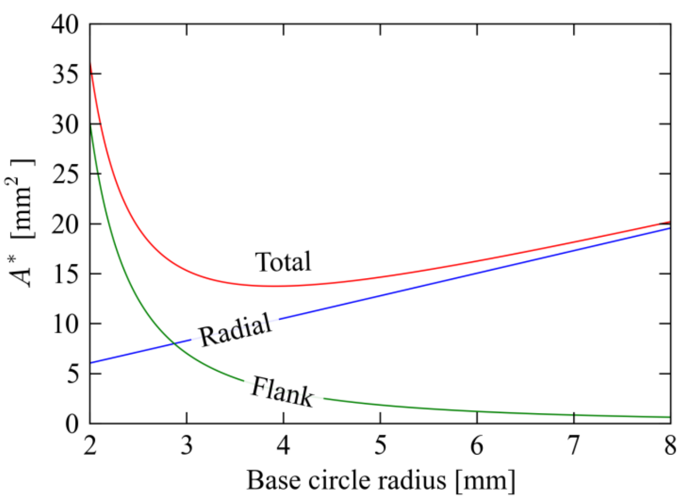


Figure 5: Family of scroll wraps for a volume ratio of 2.7, displacement of 104.8 cm^3, and wrap thickness of 4.66 mm

Source: Bell, et al. 2012





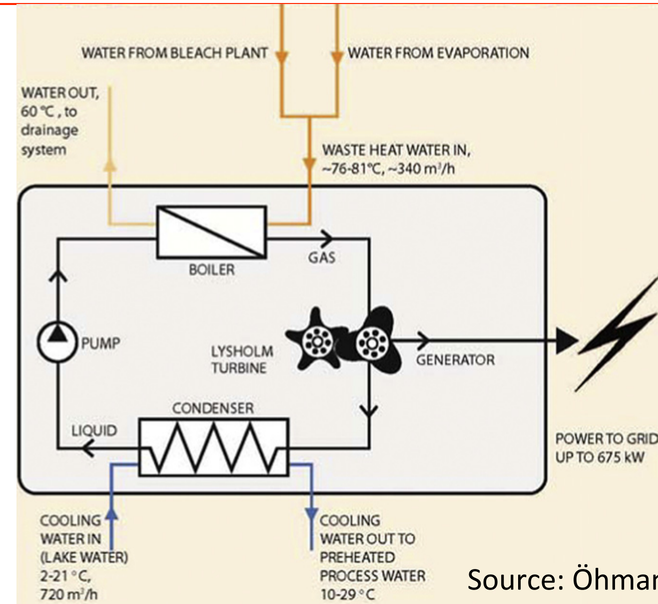
Source: Bel et al.l, 2012

Technical considerations

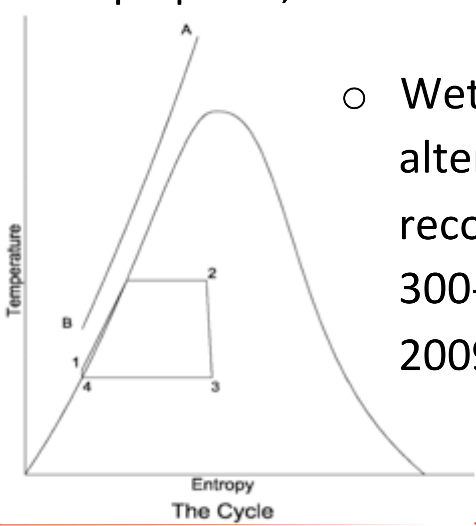
Two phase gas-liquid

Screw

- Largest expansion machine handling 2-phase flows: up to 85% in mass of liquid (Öhman, 2012).
- Use of ammonia has been reported recently (Öhman, 2012): Waste heat recovery ORC in a pulp mill, 750 kWe



Source: Öhman, 2012



Source: Smith 2009

- Wet steam cycle as an alternative to ORC for heat recovery in the range of 300-400°C (Smith et al., 2009)

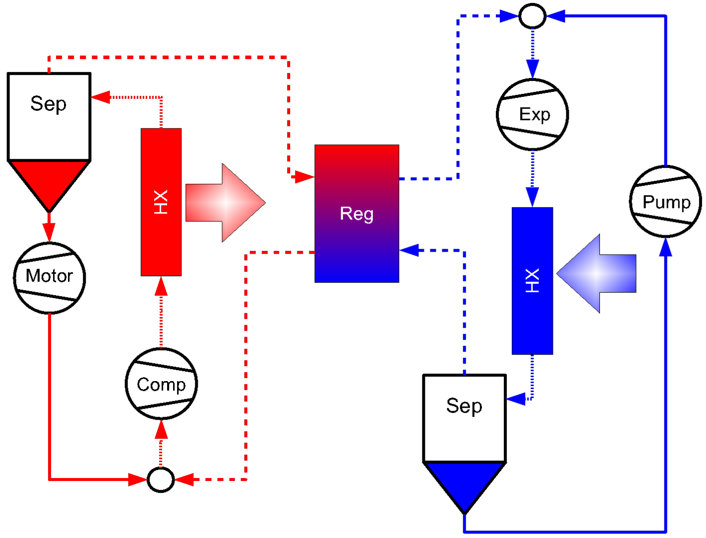
- Dortmund Univ. (Prof. Brümmer) is investigating liquid injection as a way to increase volumetric performance

Technical considerations

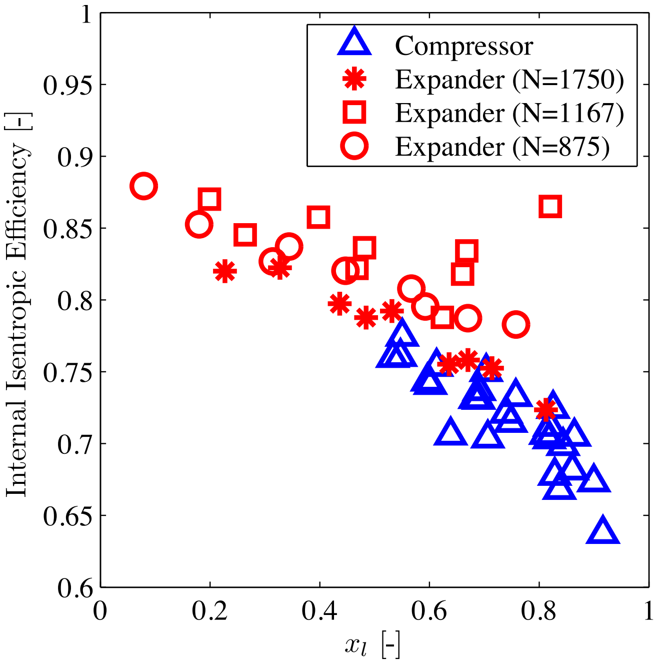
Two phase gas-liquid

Scroll

- No valves, no clearance volume => Flooding expansion is possible
- Flooding experienced on automotive compressors fed with a mixture of oil and nitrogen (Bell et al., 2012).



Ericsson cycle cooler (quasi-isothermal compression and expansion)



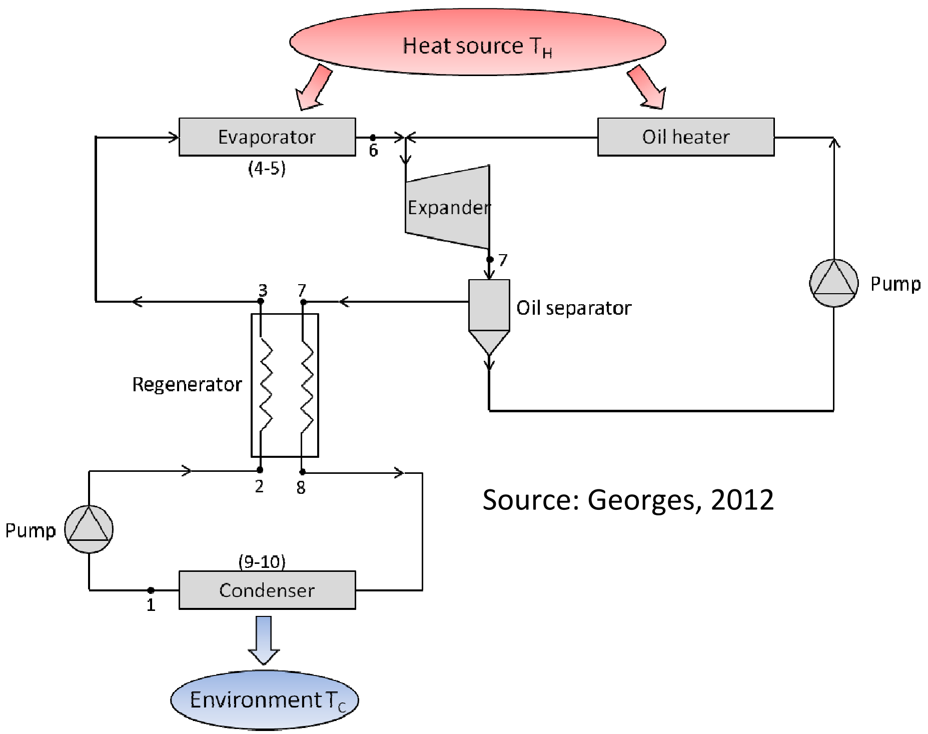
Expander fed with a mixture of nitrogen and oil (up to 80% in mass of oil)

Technical considerations

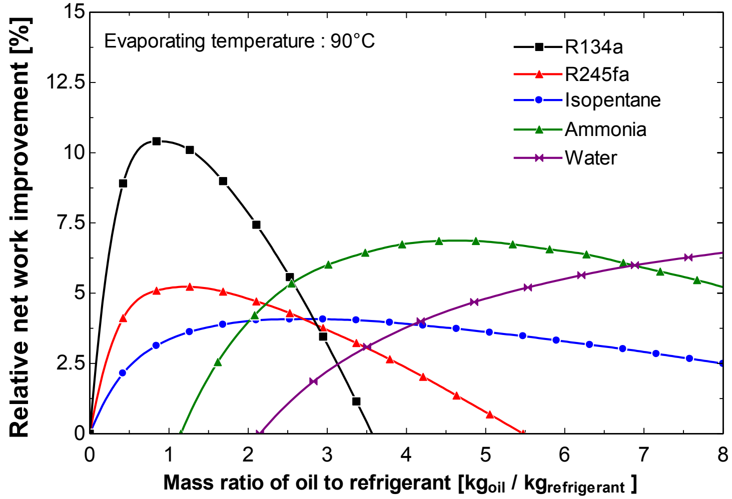
Two phase gas-liquid

Flooded ORC cycles:

Research at Purdue Universities on scroll expanders (Brandon Woodland from Groll's and Braun's team)



Source: Georges, 2012



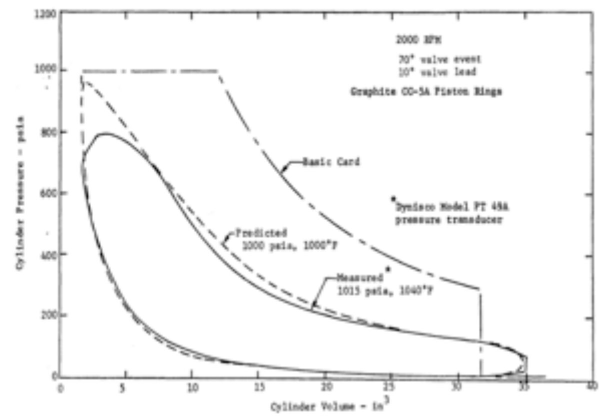
- ✓ Increase in expander power
- ✓ Increase in refrigerant temperature at expander outlet
- ✓ Ability to adjust the built-in volume ratio of the expander « seen » by the refrigerant
- ✓ Existence of an optimal mass ratio of oil

Technical constraints

Inlet temperature

Piston

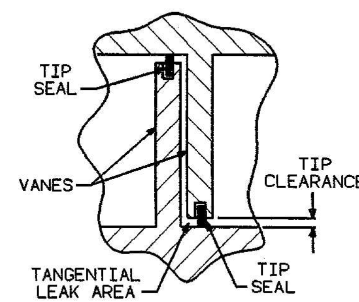
- High temperature allowed
- Could be used with high pressure/t° steam



(Source: Eckard and Brooks, 1973)

Scroll

- Hermetic compressors: discharge temperature limited to 145°C-160°C
- Temperature limited by thermal expansion and oil degradation
  - Potential advantage of tip seal vs compliant (clearance allows for dilatation)
- Currently a lot of development for high temperature heat pumps
  - f.i. Altereco project
- Records in literature:
  - Air expander: 165°C (Kane et al., 2009)
  - Steam expander: 215°C (Lemort et al., 2006)



(Source: Inaba et al., 1986)

Content of the presentation

1. Introduction
2. Technical constraints
3. **Modeling and Simulation**
4. Selection
5. Conclusions

Modeling and simulation

3 levels of models:

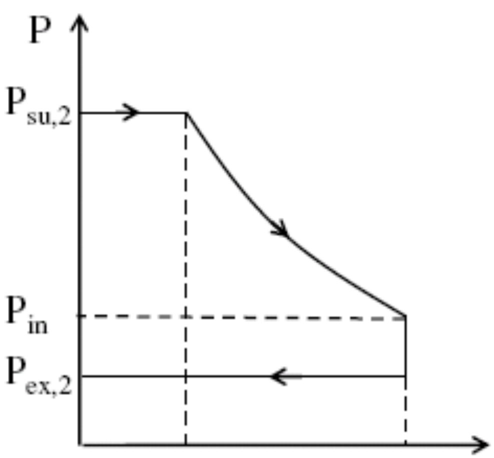
- **Empirical** or "black-box" models:
  - Very low computational time
  - Very robust
  - No extrapolation beyond calibration range
- **Semi-empirical** or "grey-box" models
  - Low computational time
  - Robust
  - Physical meaning of parameters
  - Partial extrapolation
- **Deterministic** or "white-box" models
  - Large computational time
  - Exact physical meaning of parameters

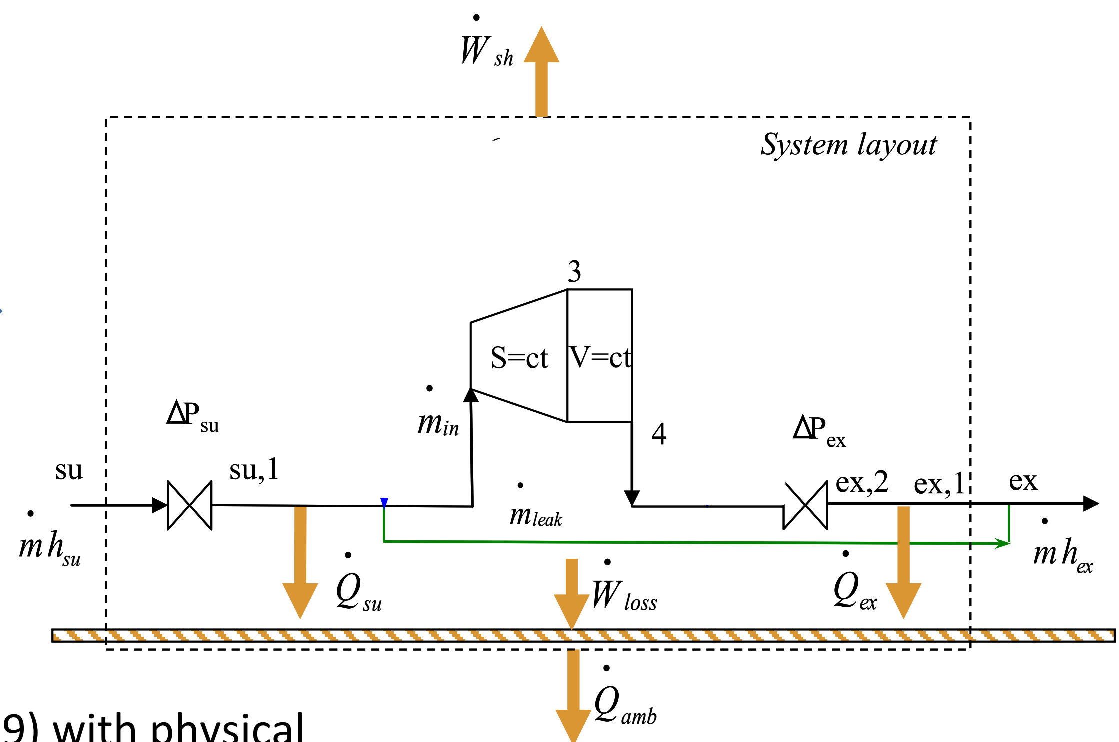
→ *Dynamic simulation of ORCs*

→ *Steady-state simulation of ORCs (design of ORC)*

→ *Design of expanders*

Scroll/screw expander





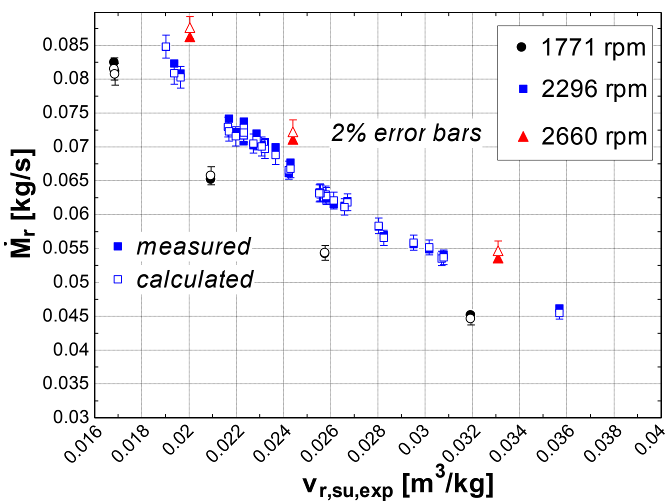
Limited number of parameters (9) with physical meaning to be identified based on performance points

Modeling and simulation

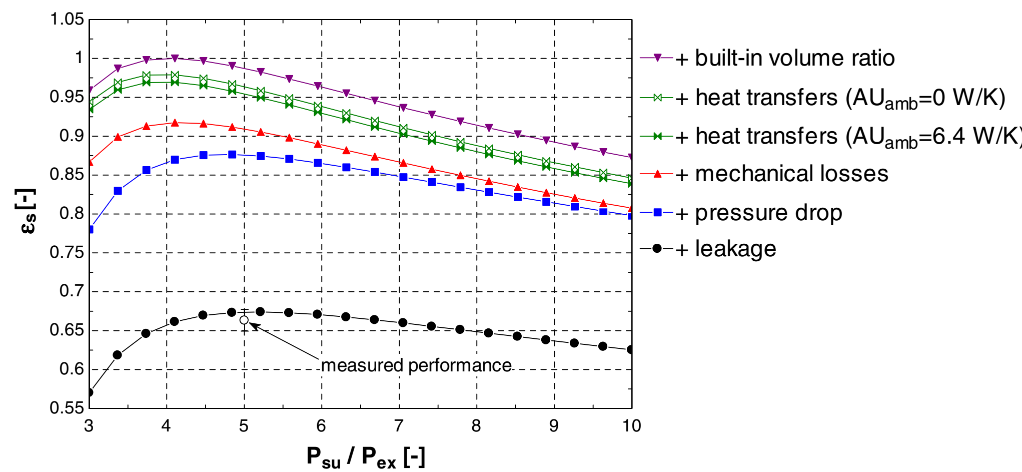
Semi-empirical

Open-drive oil-free scroll expander

AU_{amb} [W/K]	AU_{su} [W/K]	AU_{ex} [W/K]	$\dot{M}_n$ [kg/s]	A_{leak} [mm²]	$r_{v,in}$ [-]	V_s [cm³]	d_{su} [mm]	T_{loss} [N-m]
6.4	21.2	34.2	0.12	4.6	4.05	36.54	5.91	0.47



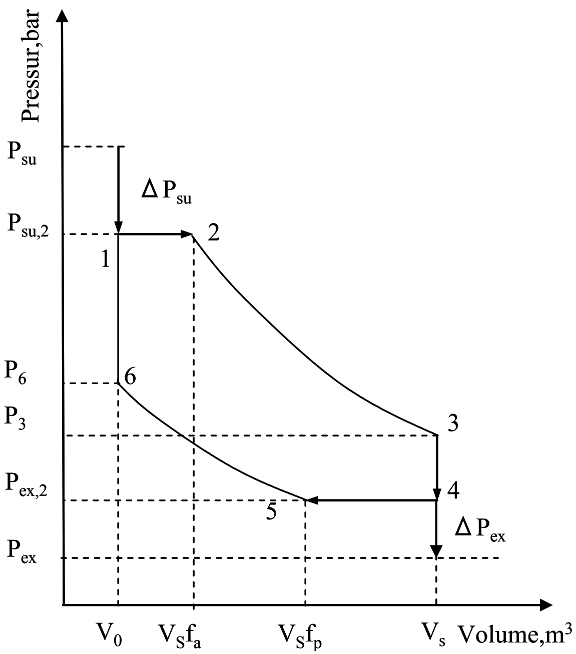
Prediction of the mass flow rate



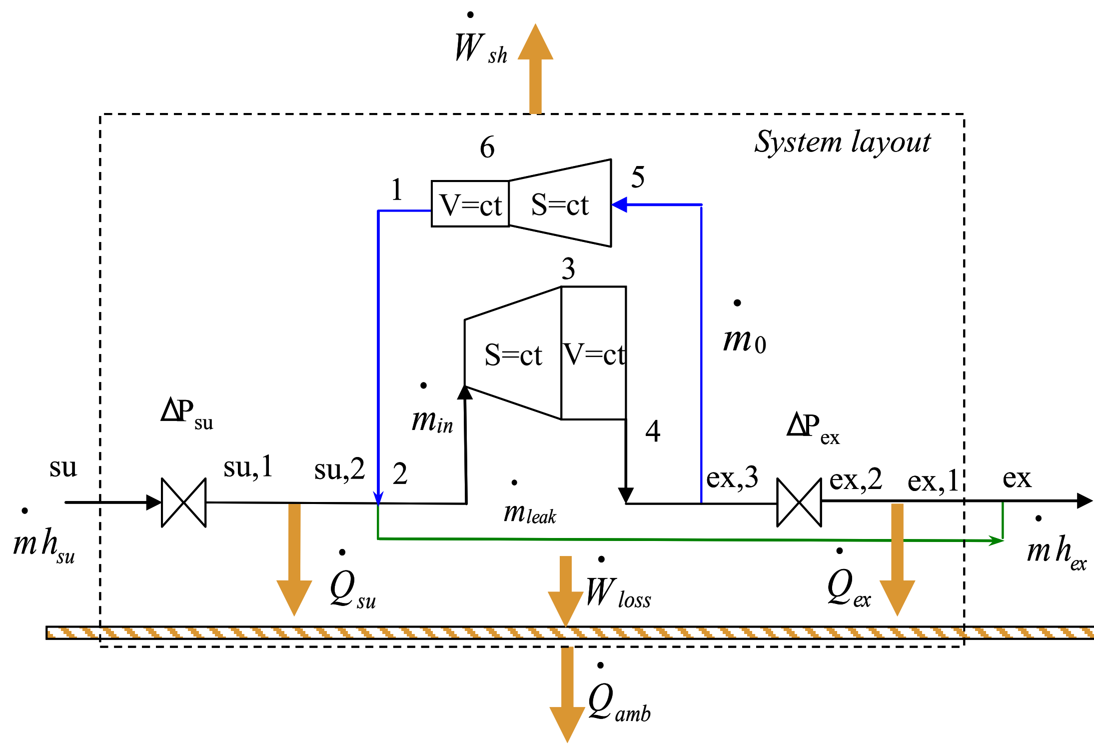
Impact of pressure losses, friction, leakages

Modeling and simulation
Semi-empirical

Piston expander



Re-compression of gas at the end of discharge phase



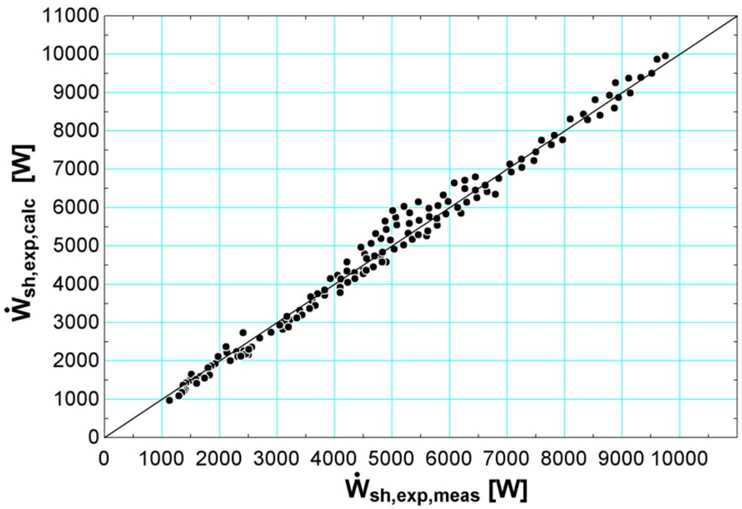
(source: Glavatskaya et al., 2012)

Modeling and simulation

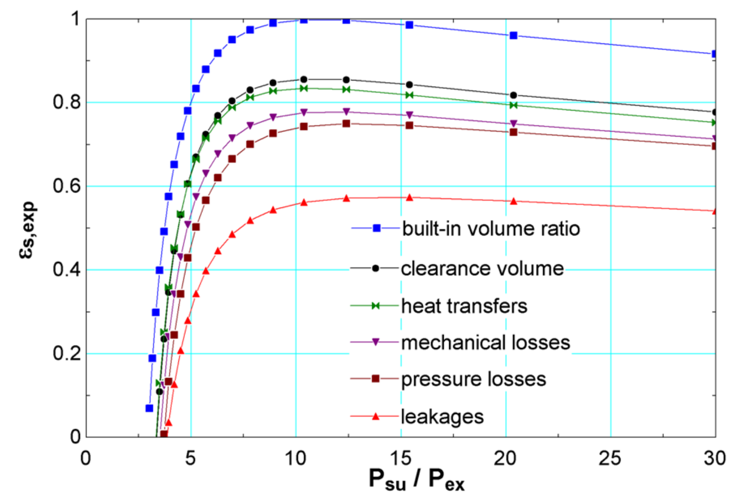
Semi-empirical

Axial piston expander

f_a	f_p	AU_{amb}, W/K	AU_{ex}, W/K	AU_{su}, W/K	A_{leak}, m^2	$d_{trh,su,n}$, m	$d_{thr,ex,n}$, m
0.25	0.419	2,5	45	22.2	$0.2386 \cdot 10^{-6}$	$6 \cdot 10^{-6}$	$11 \cdot 10^{-6}$



Prediction of the shaft power

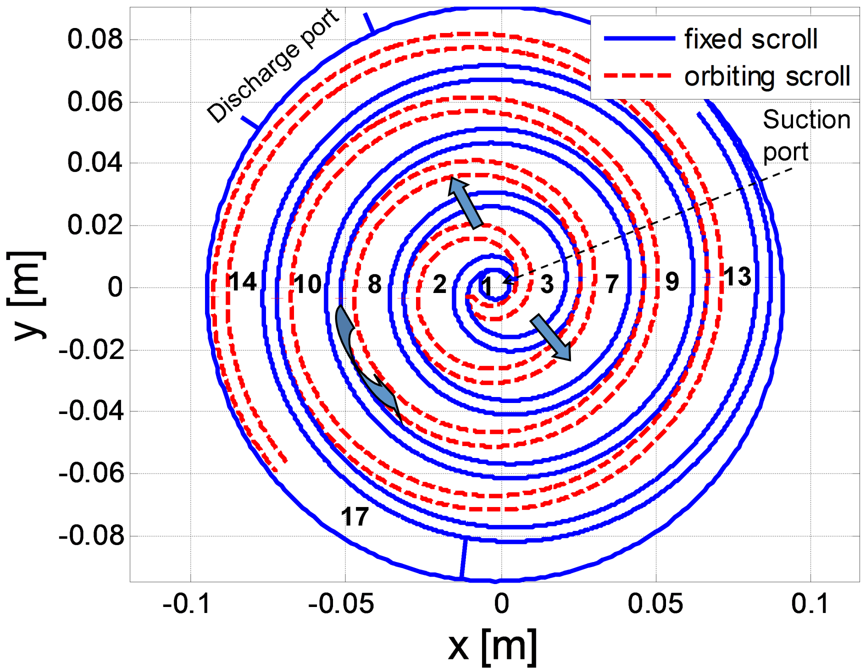


Isentropic effectiveness vs pressure ratio

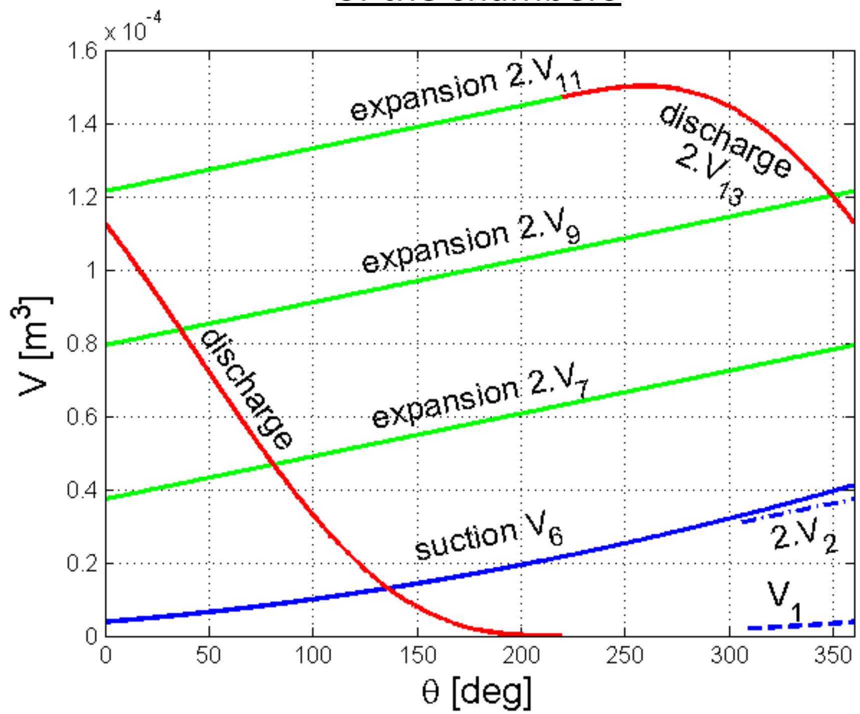
Modeling and simulation
Deterministic

Scroll

<u>Definition of the working chambers</u>



<u>Crank angle evolution of the volumes of the chambers</u>



Modeling and simulation
Deterministic

Scroll

- *Differential equations of Mass & Energy conservation*

$$\frac{dM}{d\theta} = \frac{1}{2 \cdot \pi \cdot N}\left(\sum \dot{M}_{su} - \sum \dot{M}_{ex}\right)$$

$$\frac{dU}{d\theta} = \frac{\dot{Q}}{2 \cdot \pi \cdot N} - P\frac{dV}{d\theta} + \frac{1}{2 \cdot \pi \cdot N}\sum \dot{M}_{su}.h_{su} - \frac{h}{2 \cdot \pi \cdot N}\sum \dot{M}_{ex}$$

Numerically solved for each chamber

- *Internal leakages between chambers*
- *Heat transfers between the fluid and the scroll wraps*
- *Simple mechanical loss model*

$$\frac{\mathrm{Nu}}{\mathrm{Nu}_{\mathrm{DB}}} = \mathrm{K_c} = \left(1.0 + 1.77\frac{\mathrm{D_h}}{\mathrm{R_c}}\right)$$

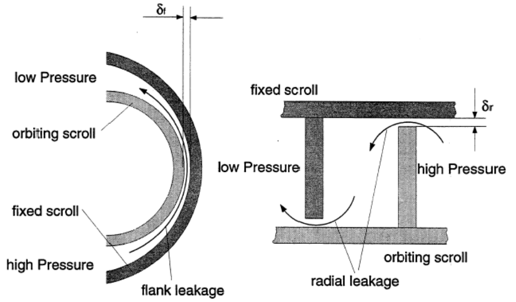
Modeling and simulation

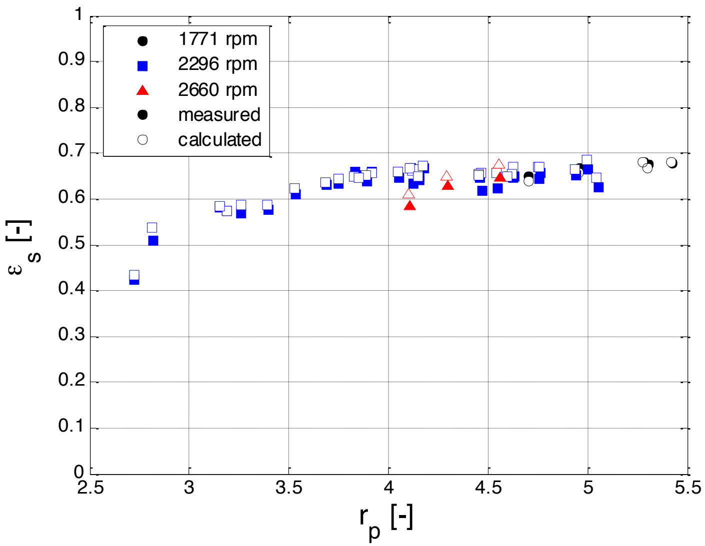
Deterministic

Scroll

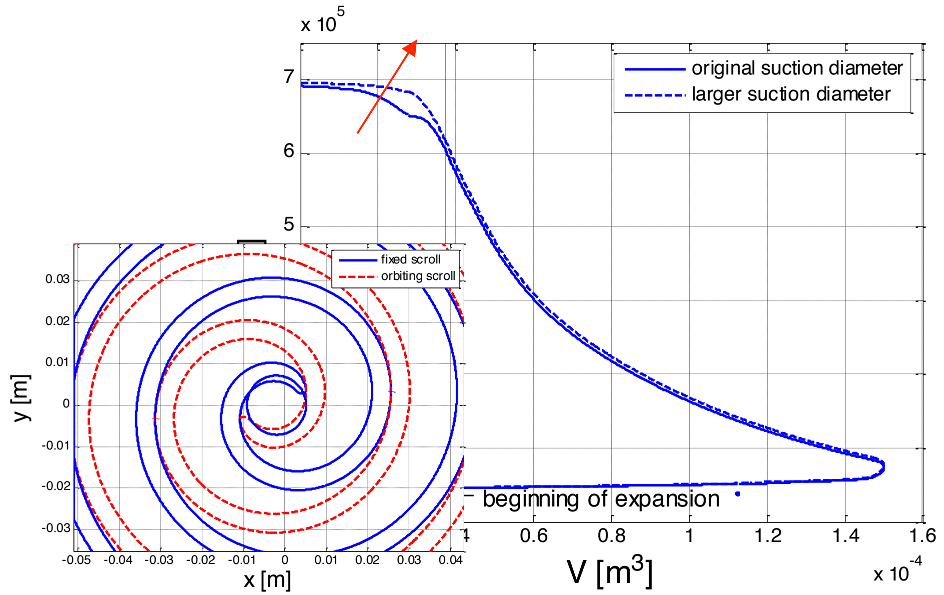
5 parameters

δ_f [µm]	δ_r [µm]	AU_{amb} [W/K]	T_{loss} [N.m]	C [-]
70	0	4	1.0	0.66





Prediction of the isentropic effectiveness



Content of the presentation

Selection

Limitations of positive displacement machines

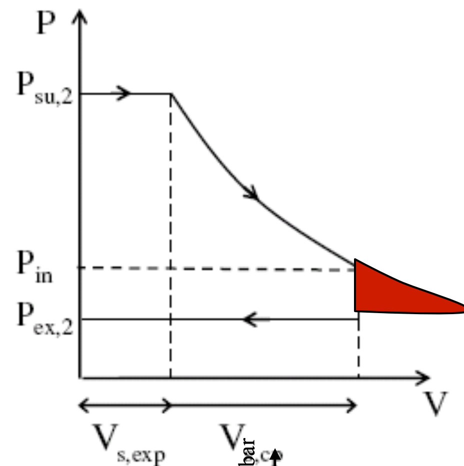
$$\varepsilon_s = \varepsilon_{in} . \eta_{mec}$$

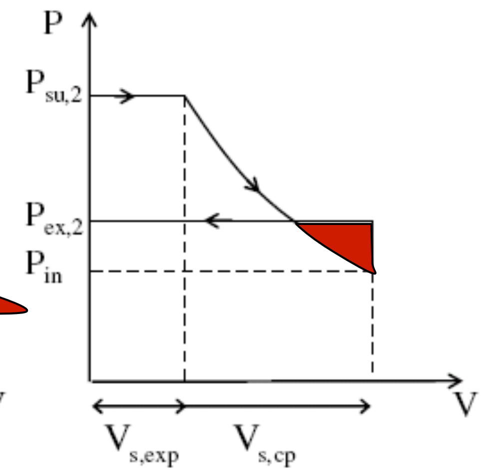
- Built-in internal volume ratios
  - $\Rightarrow$ Under and over-expansion
  - $\Rightarrow$ Recompression work (piston)
  - $\Rightarrow$ $r_{v,in,max}$: 4 (scroll), 5 (screw), 10 (piston)
- Volume coefficient (m^3/MJ)
- Range of volume flow rates

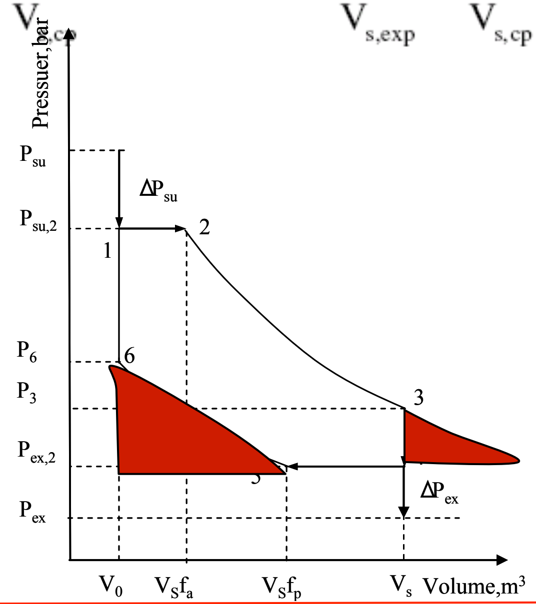
$$w_1 = h_{su} - h_{in}$$

$$w_2 = v_{in} \cdot (p_{in} - p_{ex})$$

$$\varepsilon_{in} = \frac{w_1 + w_2}{\Delta h_s}$$





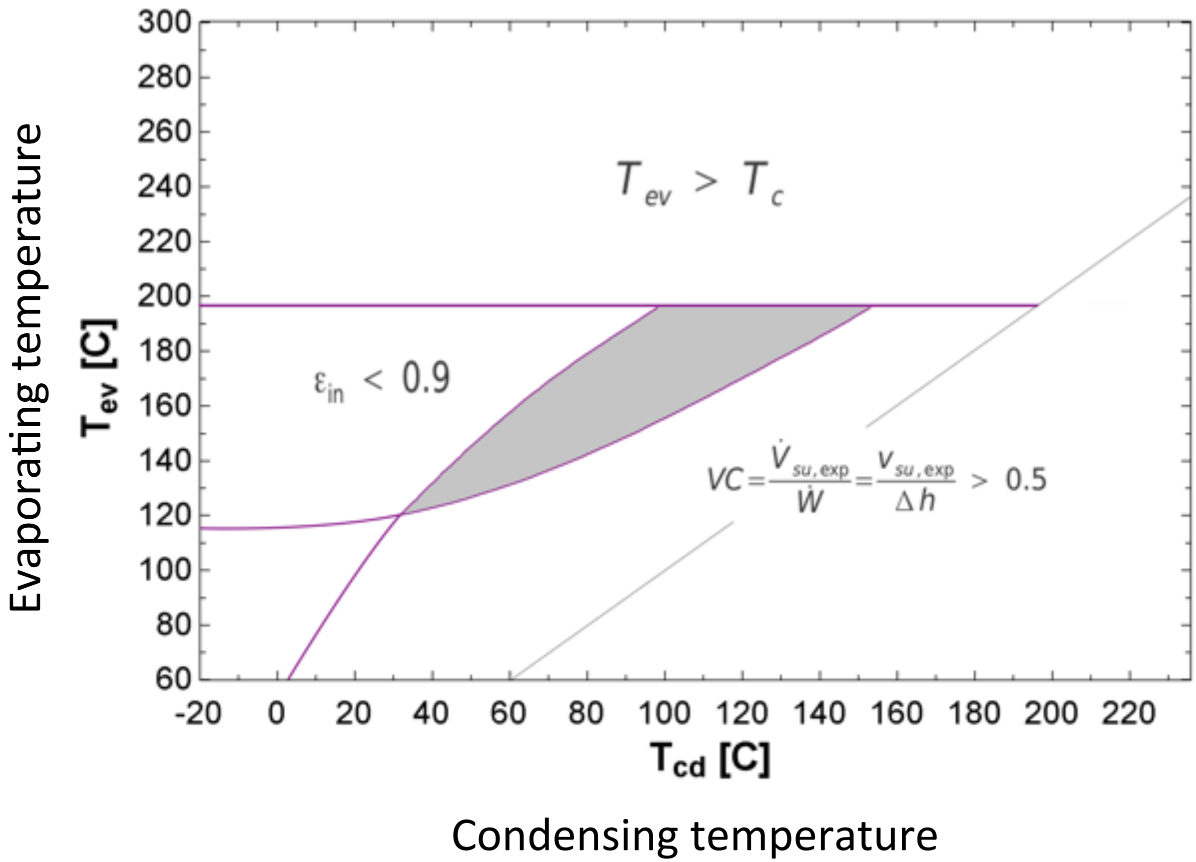


See Quoilin et al. (2012)

Selection

Operating maps

(screw+n-pentane)



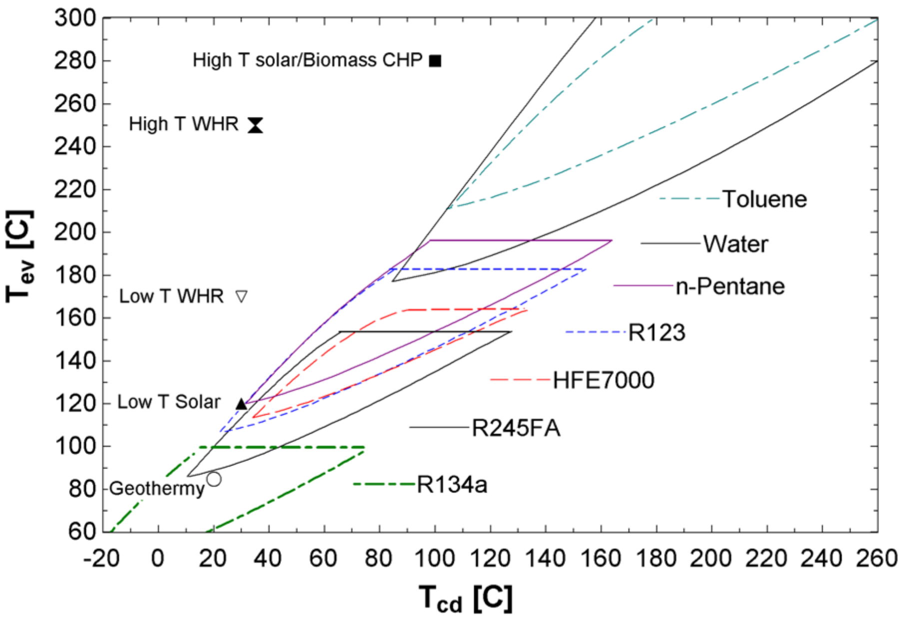
$$w_1 = h_{su} - h_{in}$$

$$w_2 = v_{in} \cdot (p_{in} - p_{ex})$$

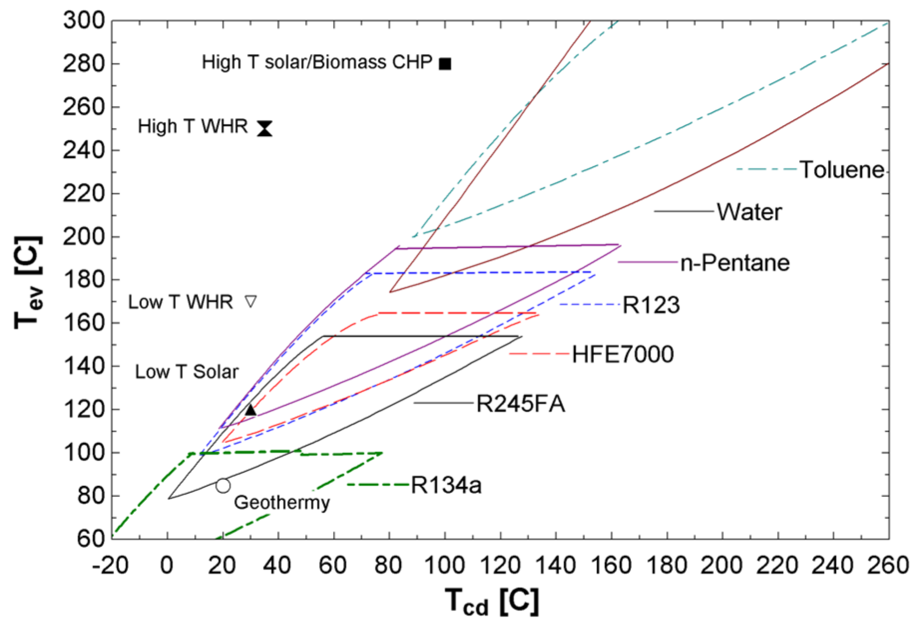
$$\varepsilon_{in} = \frac{w_1 + w_2}{\Delta h_s}$$

Selection

Operating maps



screw

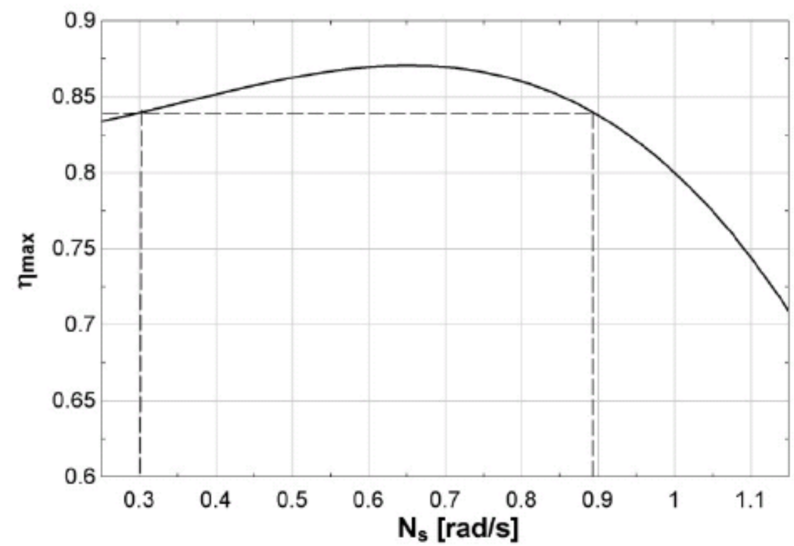


piston

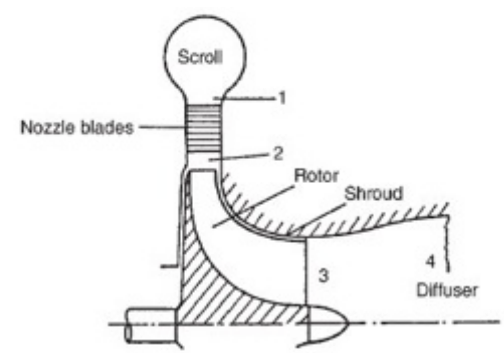
Selection

Limitations of turbines

- Min/max specific speed
- Maximum tip speed
- Maximum mach number in the nozzle
- Maximum mach number in the rotor
- Maximum rotating speed (depending on the bearings)



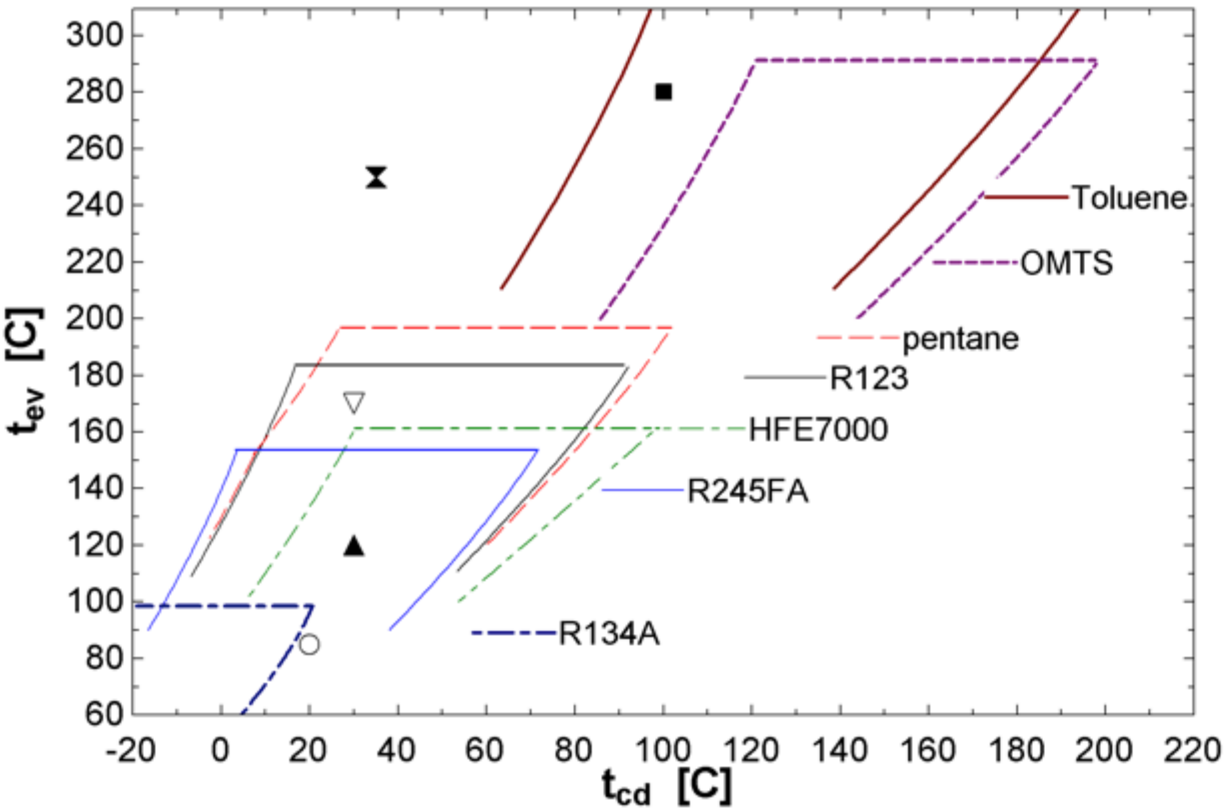
	Minimum Value	Maximum value
U_2		370 m/s
N_s	0.30	0.89
M_2		1.8
M_3		0.85
N		50000 rpm

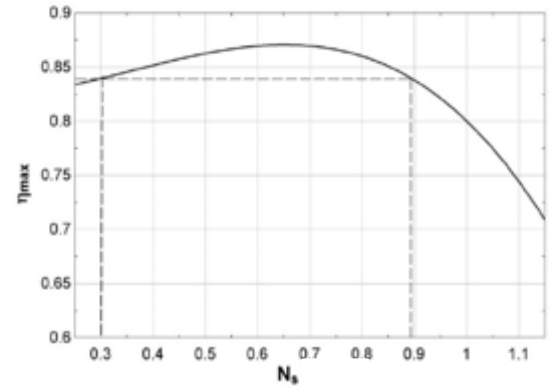


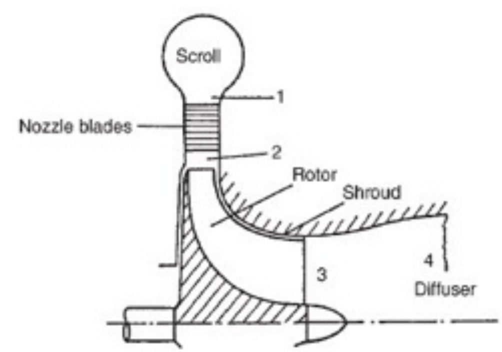
Selection

Operating maps

n-pentane:





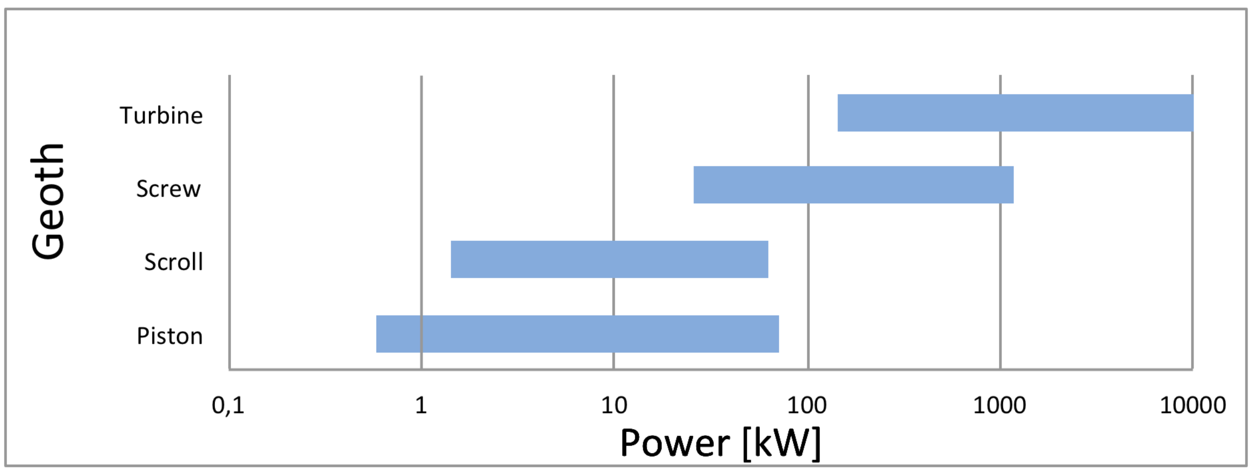


Selection

Power ranges

	$\dot{V}_{cp,min}$	$\dot{V}_{cp,max}$
Scroll	1.1 l/s	49 l/s
Screw	25 l/s	1100 l/s
Piston	1.25 l/s	75 l/s

rpm < 50000



Conclusions

- Positive displacement machines show some advantages over turbines, among them their ability to handle **a liquid phase**: lot of opportunities of research in 2-phase flows
- Larger **industrial maturity** of screw expanders vs scroll expanders, niche-market applications for piston expanders
- Developments on positive displacement expanders beneficiate from **developments of compressors**: increase of performance, reliability & extension of operating ranges (f.i. compressors for high temperature heat pumps)
- Different types of **modeling/simulation models** for different purposes
- The **selection of the expansion machine** depends on the working conditions and on the selected fluid
  - ✓ Operating maps of optimal working conditions for each combination of expander/fluid were presented
  - ✓ Useful tool for the preliminary selection of expanders and fluids for a given cycle.

Thank you for your attention!

Acknowledgments

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