Heat Pump Systems

Summary W10

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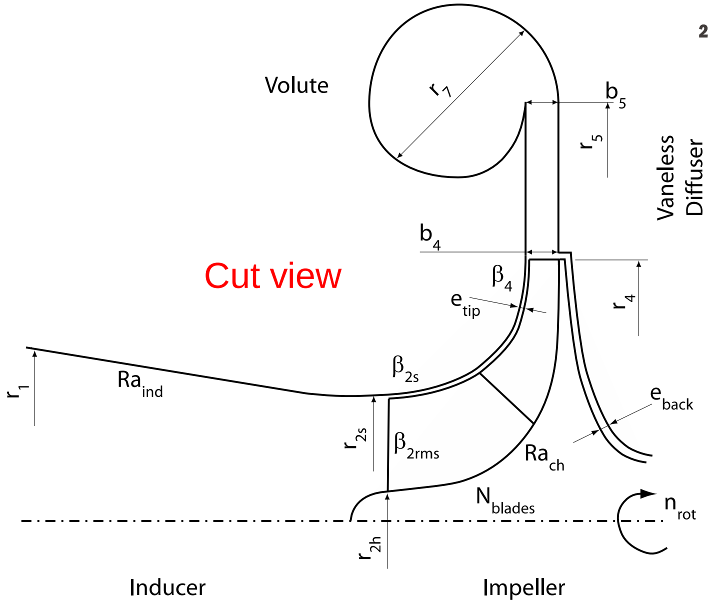
Radial Compressor Geometry

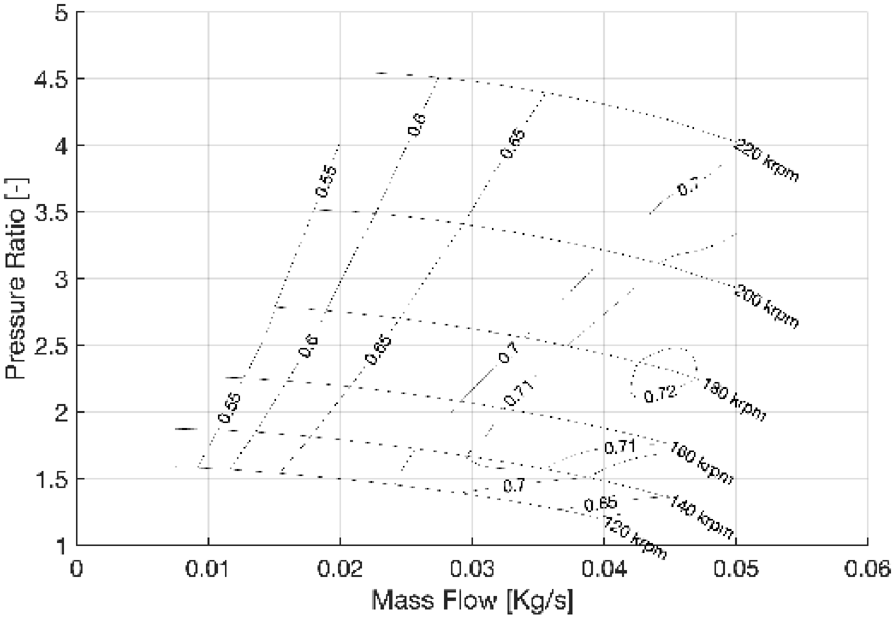
- Typical geometric features
  - Impeller tip width ratio
    $$0.02 < b_4/r_4 < 0.2$$
  - Relative impeller tip clearance
    $$e_{tip}/b_4 < 0.05$$
  - Number of blades
    $$12 < N_{blades} < 32$$
  - Axial length
    $$L/r_4 \approx 0.35$$
  - Inlet diameter ratio
    $$0.3 < r_{2s}/r_4 < 0.7$$
  - Blade angles
    $$\beta_{2s} = -60^\circ \qquad \beta_4 \approx -40^\circ$$

Cut view



Typical Compressor Map

- Map limited towards lower mass flows by surge and towards higher mass by choke



Compressor at Choke

- Choke defines upper mass flow for each speed line
  - Reducing back pressure increases velocity until it reaches the speed of sound in smallest passage
  - Further reduction in back pressure yields no increase in mass flow
  - Speed line is vertical after choking has occurred
  - Choking can occur at inlet, impeller, or diffuser
  - Higher speeds shift choke to higher mass flows until inlet is choked
  - Throat usually near impeller inlet or diffuser inlet

Compressor at Surge

- Surge line marks the low flow limit of region of stable operation
  - Caused by increase in loading as mass flow is reduced (lower mass flow → increased incidence → increased loading → separation)
  - Depends on compressor itself & system configuration

- Instability can have several forms
  - Rotating Stall: separation in blade rows which jumps from one blade to next → mass flow nearly constant with small high-frequency pressure fluctuations
  - Mild Surge: Pulsations in mass flow and pressure without backflow
  - Deep Surge: Strong periodic backflow through compressor with large pressure and mass flow variations → should be avoided

?

Heat Pump Systems

On the Modeling of Centrifugal Compressors

Prof. J. Schiffmann

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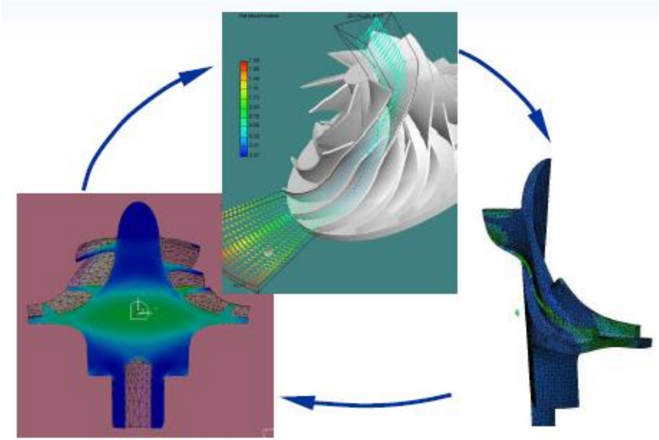
Design Process of Turbomachinery

- Design of turbocompressor is interdisciplinary task
  - Aerodynamics
  - Mechanical stress
  - Dynamics
  - Thermal

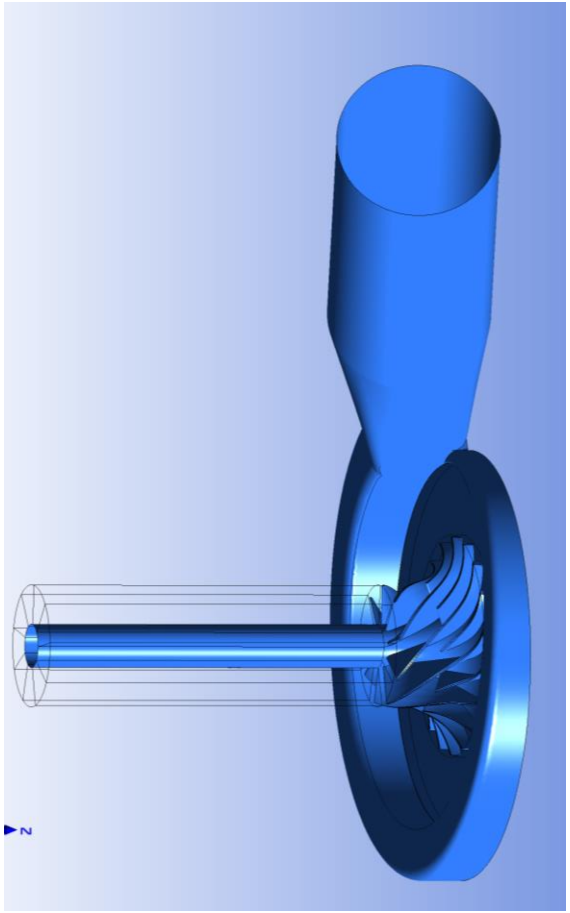


- Flow in turbomachinery is highly complex

3D – CFD I

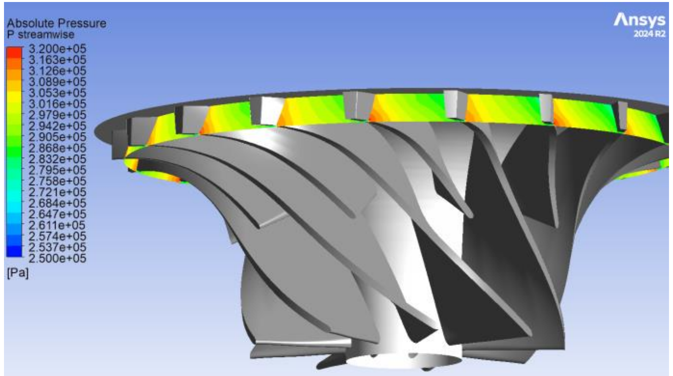
- Mesh of 30 million nodes
  → 15h computational time for one operating point → computational resources

- Not well suited for design

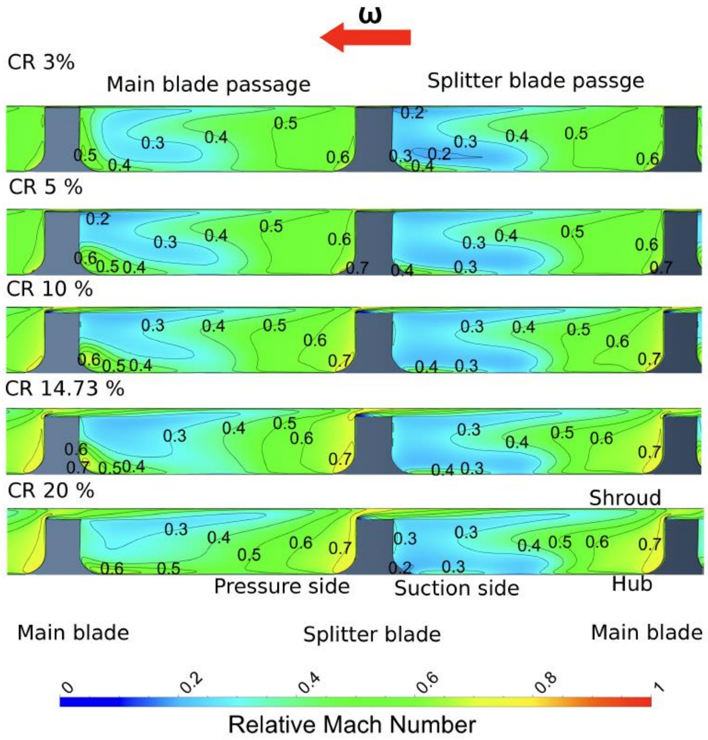


3D – CFD II

- Great for visualizing detailed flow features and analysis



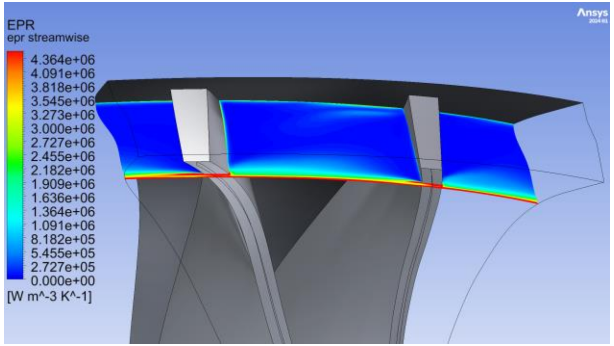
Pressure on pressure side larger than on suction side



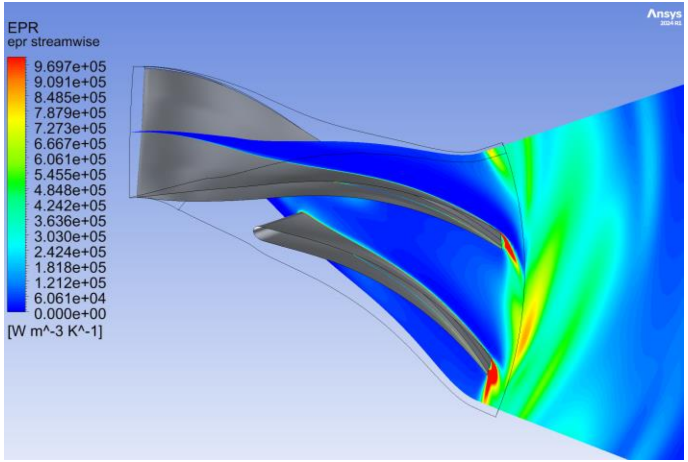
Accumulation of low momentum fluid close to suction side → jet-wake pattern → mixing losses into diffuser

3D – CFD III

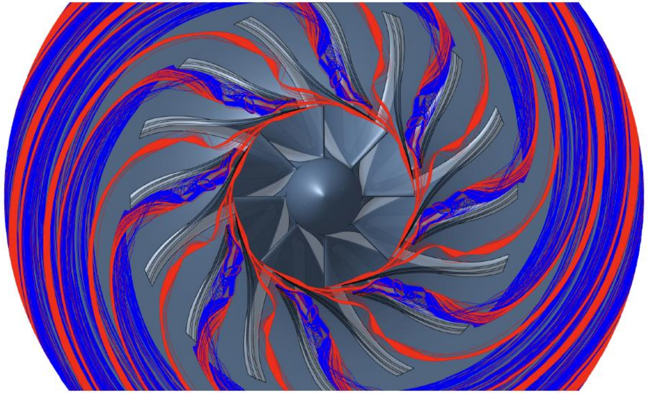
- Great for visualizing detailed flow features and analysis



Entropy production rate highest in tip gap and on surfaces → skin friction



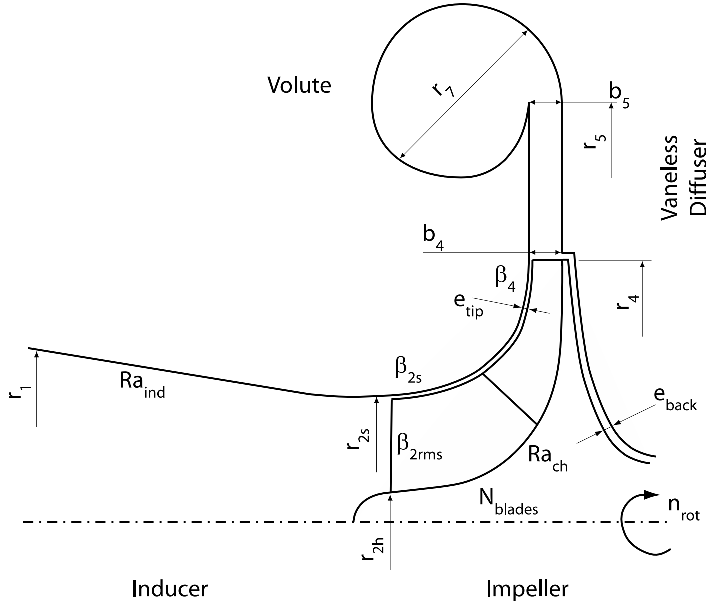
Trailing edge mixing into diffuser



Trajectories of main (red streamlines) and splitter blade (blue streamlines) tip leakage vortices

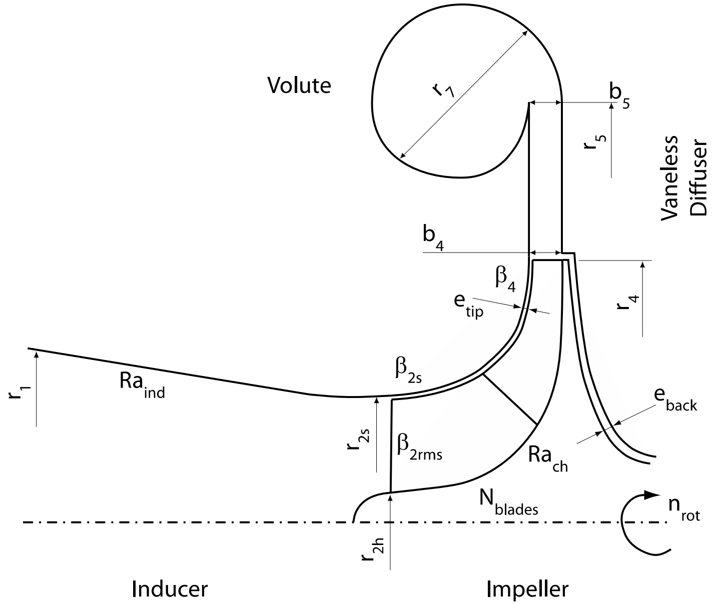
1D – Meanline Model

- Modeling along flow direction, component by component
- Based on velocity triangles, mass- & energy conservation, h-s-diagram
- Representation of losses through empirical correlations



1D – Meanline Model

- Proven Empirical Loss Correlations
  - Skin Friction } Inducer
  - Incidence
  - Skin Friction
  - Diffusion
  - Clearance
  - Disk friction
  - Recirculation } Impeller
  - Skin Friction
  - Trailing edge mixing } Diffuser



1D – Meanline Model

- Further effects
  - Slip → flow does not follow blade angle
    → causes a decrease in work input

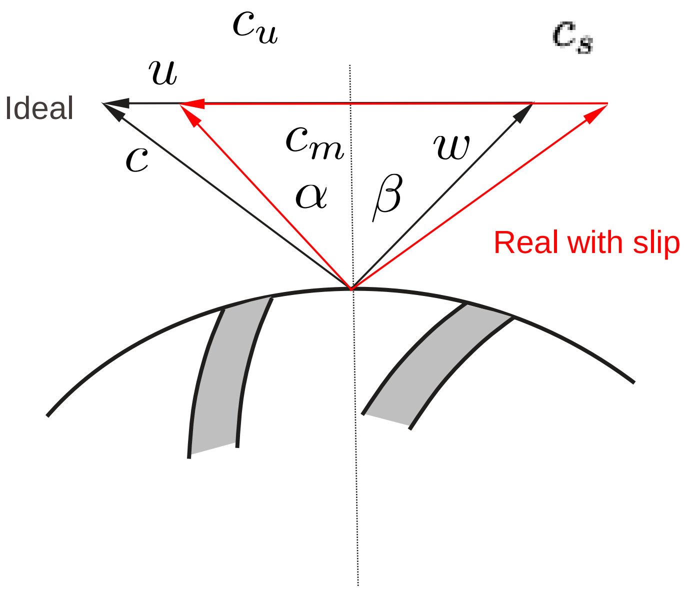
$$\frac{c_s}{u_4} = \frac{\sqrt{\cos\beta_{4_{bl}}}}{N_{bl}^{0.7}}$$

  - Aerodynamic blockage

$$B_i = A_{\text{eff}}/A_{\text{geom}}$$

  - Surge in vaneless diffuser

$$\alpha_{crit} = f\left(M_4, b_4/r_4, r_5/r_4\right)$$



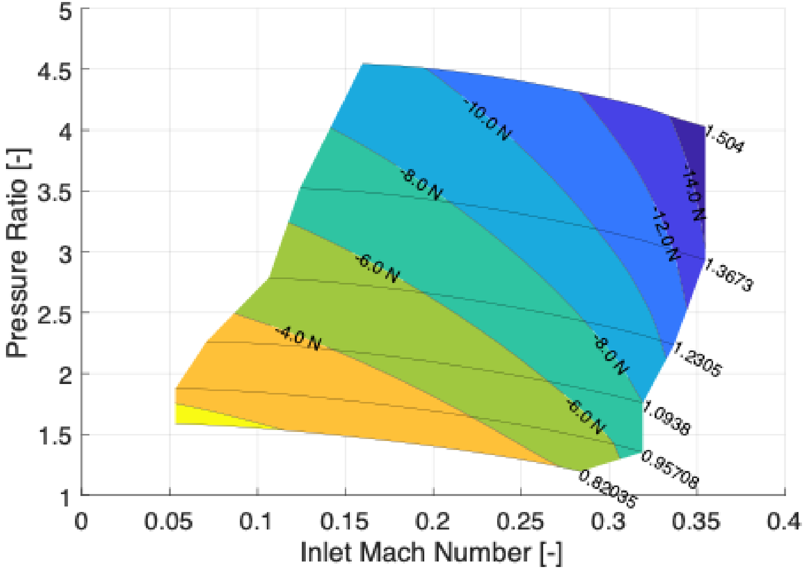
EPFL

1D – Meanline Model

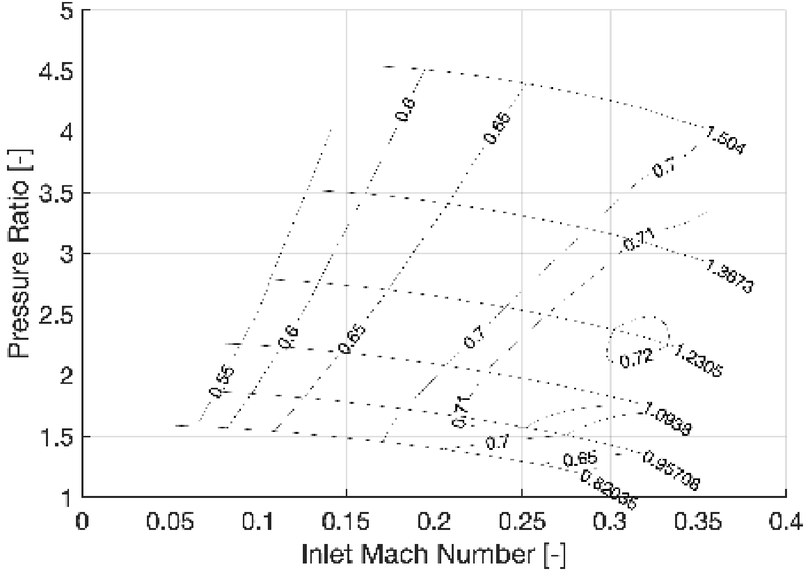
$$[PR, \eta_{is}, \text{surge}, \text{choke}] = f\left(\dot{m}, N_{rot}, P_{01}, T_{01}, \text{Fluid}, \text{Geom}\right)$$

- Fast compressor map prediction → great for design
- No capturing of 3D flow patterns

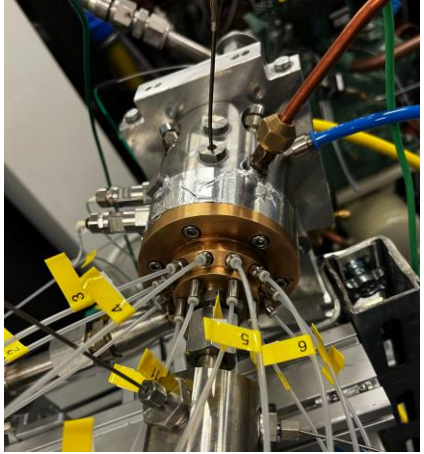
Thrust force contours



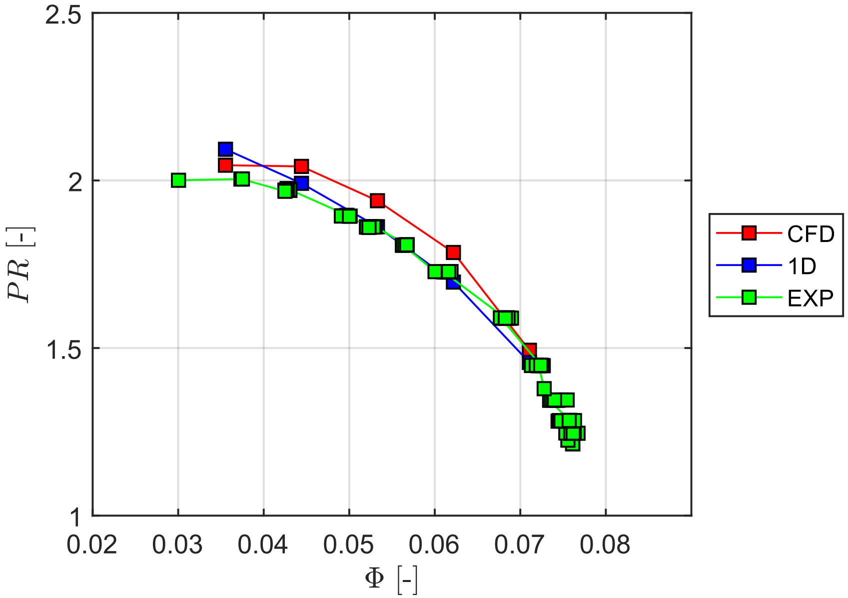
Isentropic efficiency contours

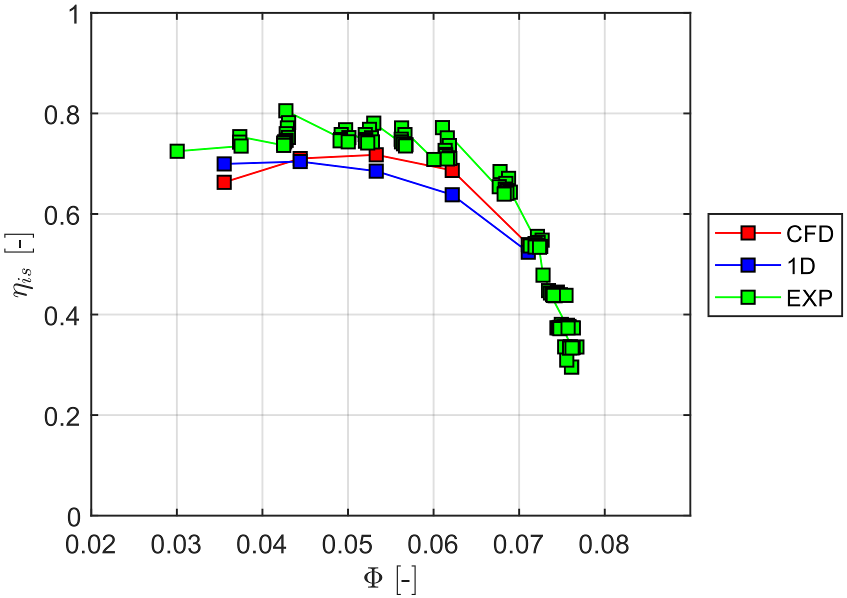


Comparison CFD – Meanline Model



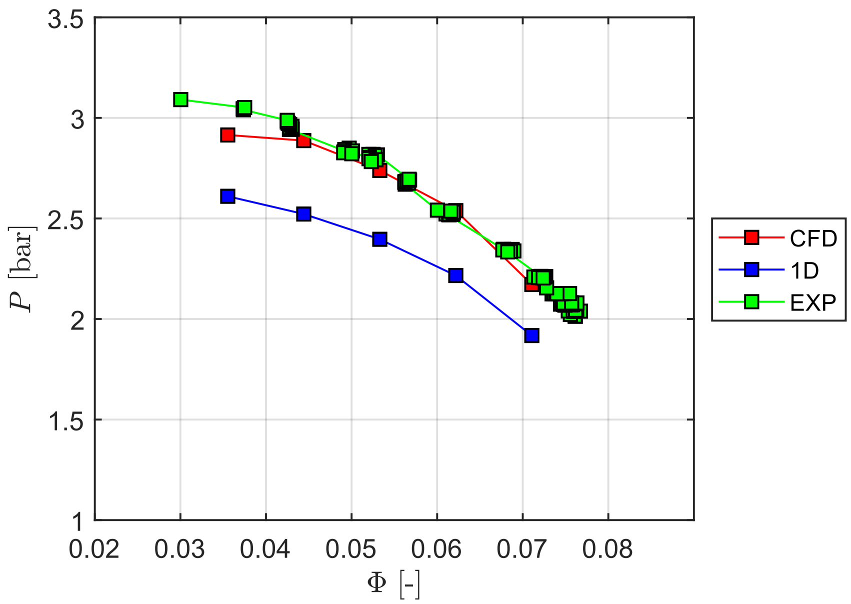
- Excellent overall (out-in) agreement between experimental data, CFD, and meanline-model



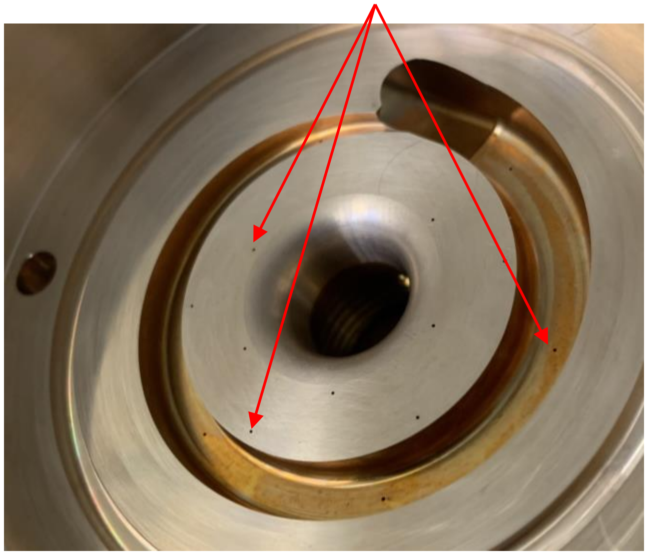


Comparison CFD – Meanline Model

- Inconsistency between meanline model, CFD & experiments at component level Impeller pressure ratio

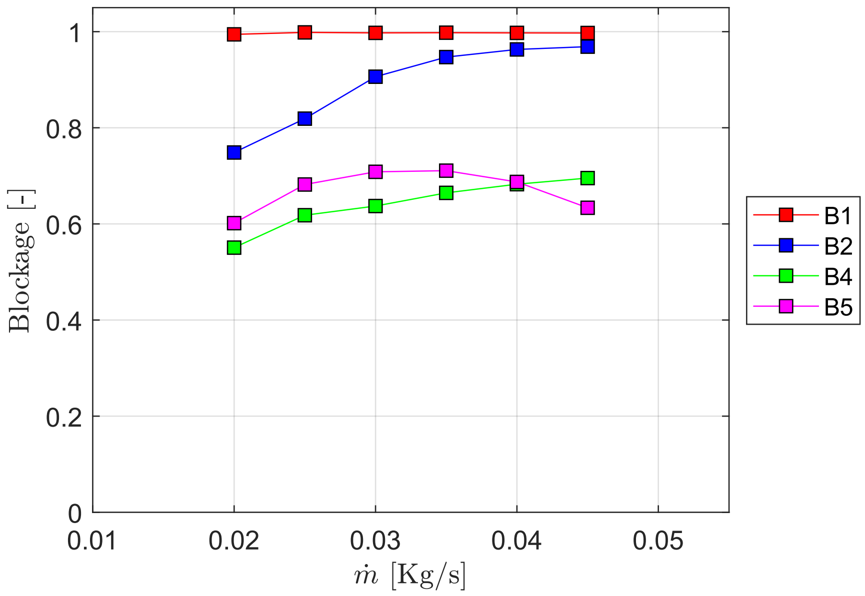


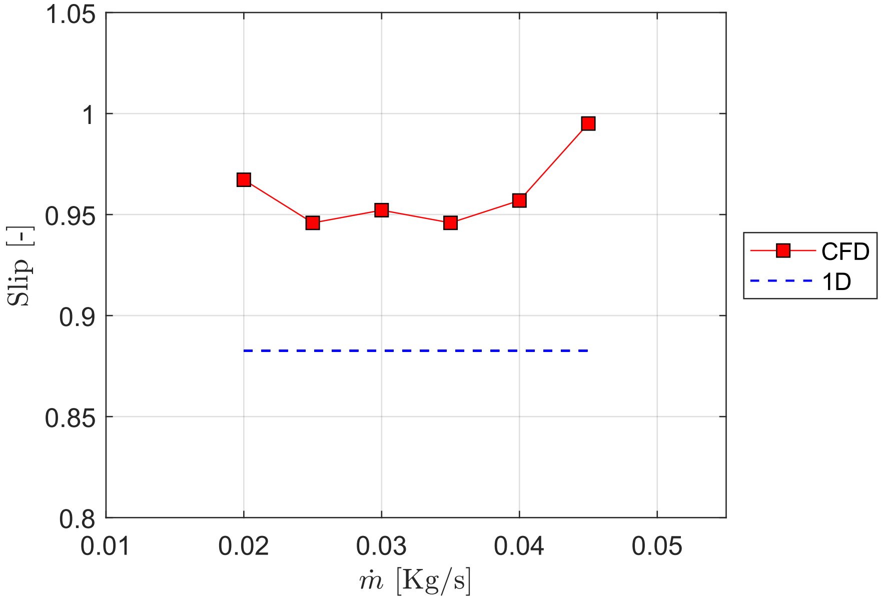
Pressure measurements



Meanline Model Calibration

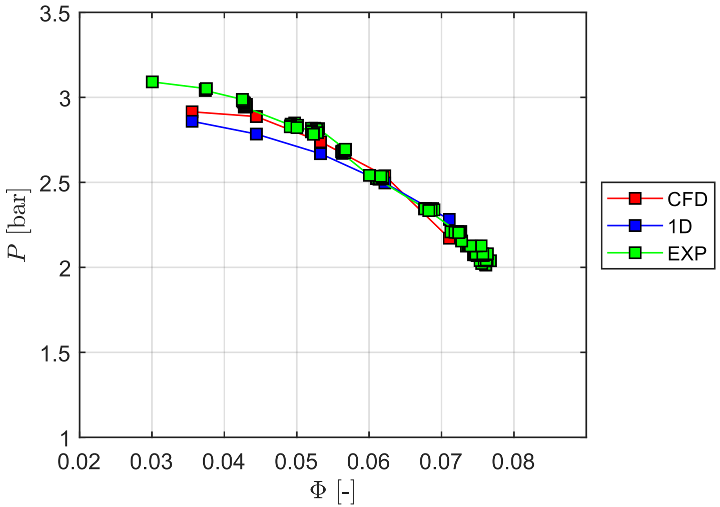
- Correction of aerodynamic blockage factors, slip, and losses based on CFD data

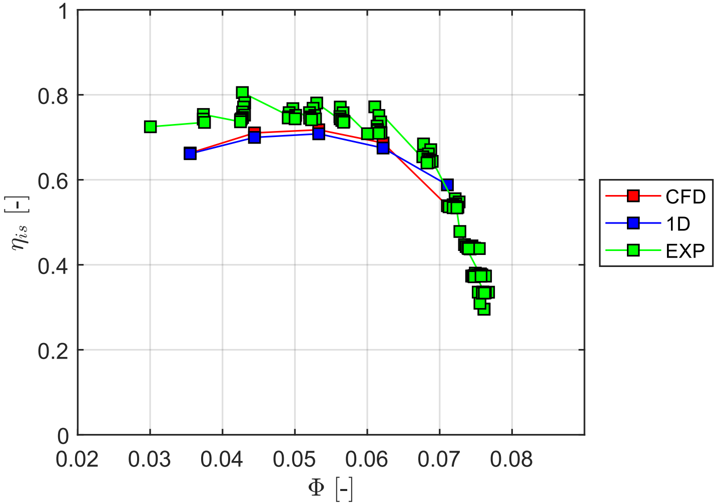


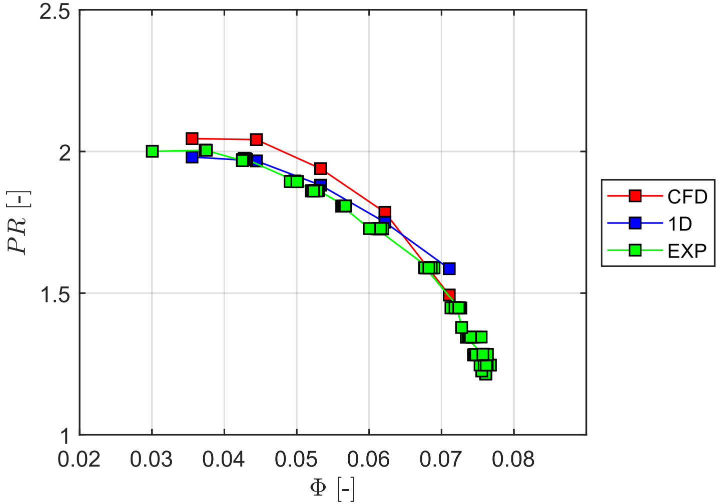


Calibrated Meanline Model

- Good agreement at a component level and overall
- Meanline models are very efficient but may need calibration



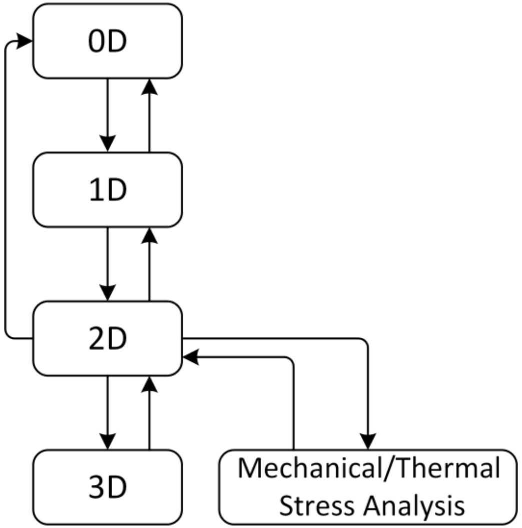




Design Process of Turbomachinery

- Iterative combination of increasingly complex models
  - 0D: Overall dimensions – based on empirical performance maps
  - 1D: Details of inlet and exhaust areas → meanline model
  - 2D: Definition of blade geometry
  - 3D: Assessment of detailed flow patterns

- Accurate starting point (0D) key for efficient design process
  - Requires design maps



Dimensional Analysis

- Parameters influencing compressor performance

$$[PR, \eta_{is}, \text{surge}, \text{choke}] = f\left(\dot{m}, N_{rot}, P_{01}, T_{01}, \text{Fluid}, \text{Geom}\right)$$

- Using dimensional analysis

$$[\lambda, \eta_{is}] = f\left(\phi_{01}, M_{u4}, Re, \text{Fluid}, \text{Geom}\right)$$

$$M_{u4} = \frac{u_4}{a_{01}}$$ Machine Mach number

$$\phi_{01} = \frac{\dot{m}}{\rho_{01} D_4^2 u_4^2}$$ Flow coefficient

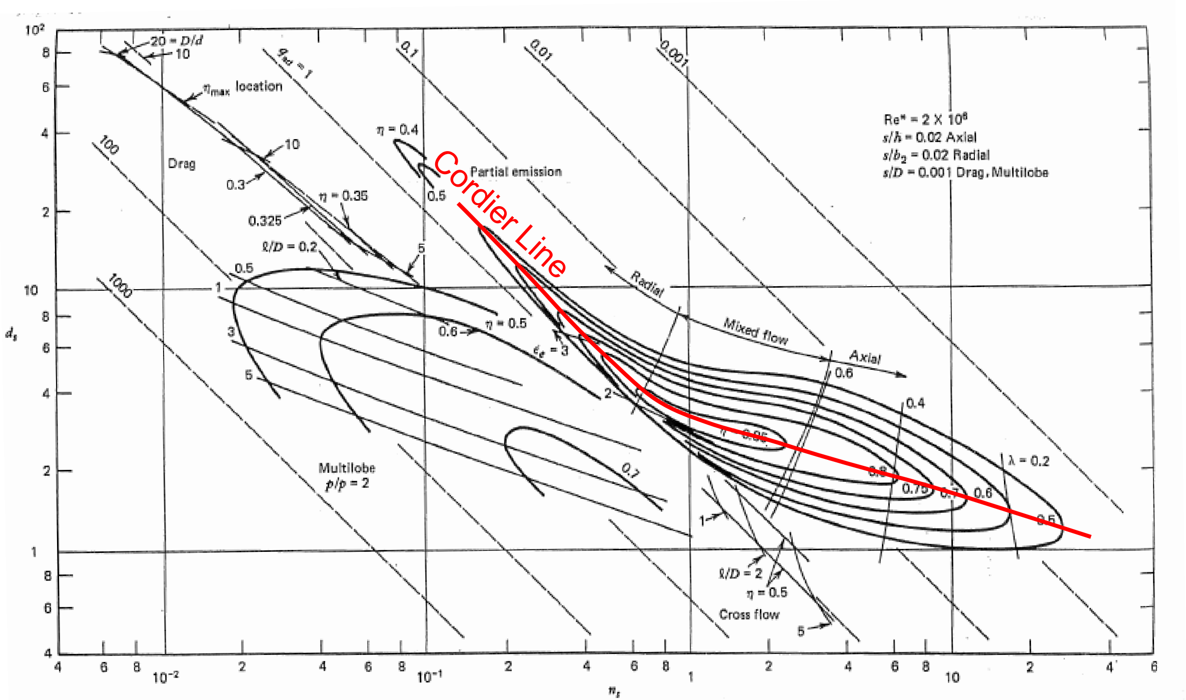
$$\lambda = \frac{h_{04} - h_{01}}{u_4^2}$$ Work input coefficient

Compressor Design Maps

- Specific speed and diameter alternative set of dimensionless parameters

$$n_s = \omega \frac{\dot{V}_{01}^{0.5}}{\Delta h_{is}^{0.75}}$$

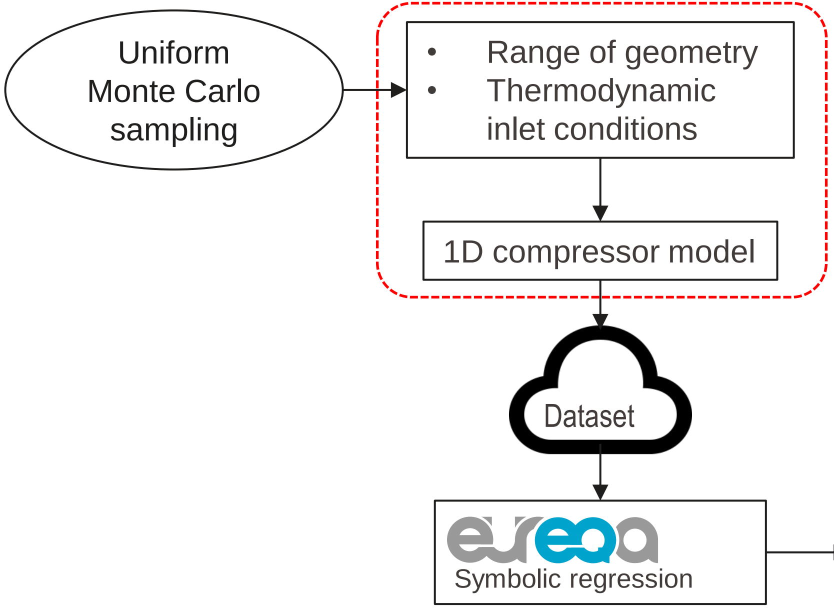
$$d_s = d_4 \frac{\Delta h_{is}^{0.25}}{\dot{V}_{01}^{0.5}}$$



O. Balje, Turbomachines: A guide to design, selection and theory, 1981, Wiley

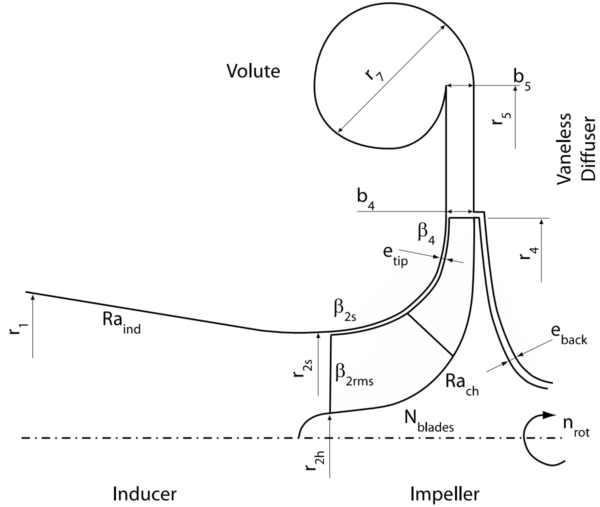
Updated Compressor Design Maps

- Experimentally validated tool used to generate surrogate model



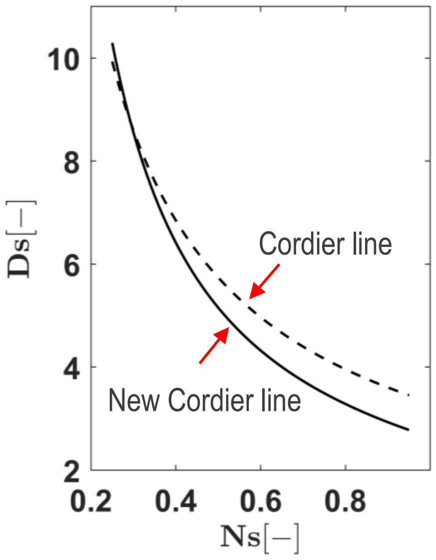
https://www.nutonian.com/products/eureqa/

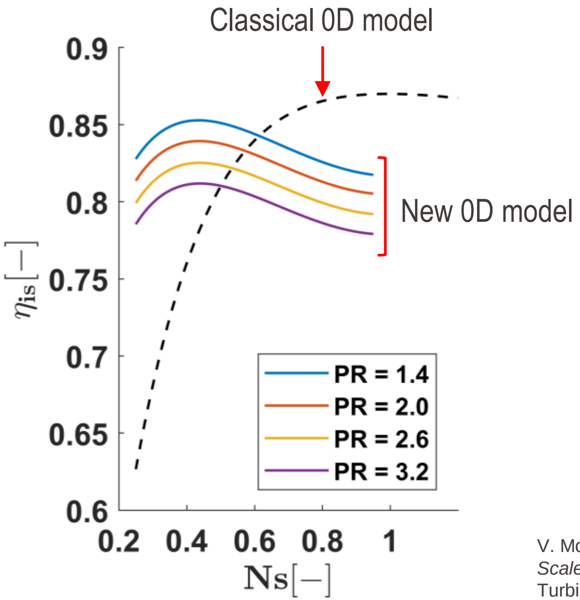
$$\eta_{is,0D} = f(Ns, Ds, PR, \overline{b_4}, \overline{r_{2s}})$$

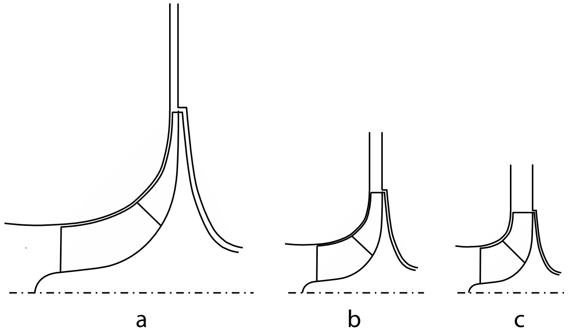


Updated Design Maps

- Updated model for reduced scale compressors deviates from literature
- New model includes additional design variables → more information



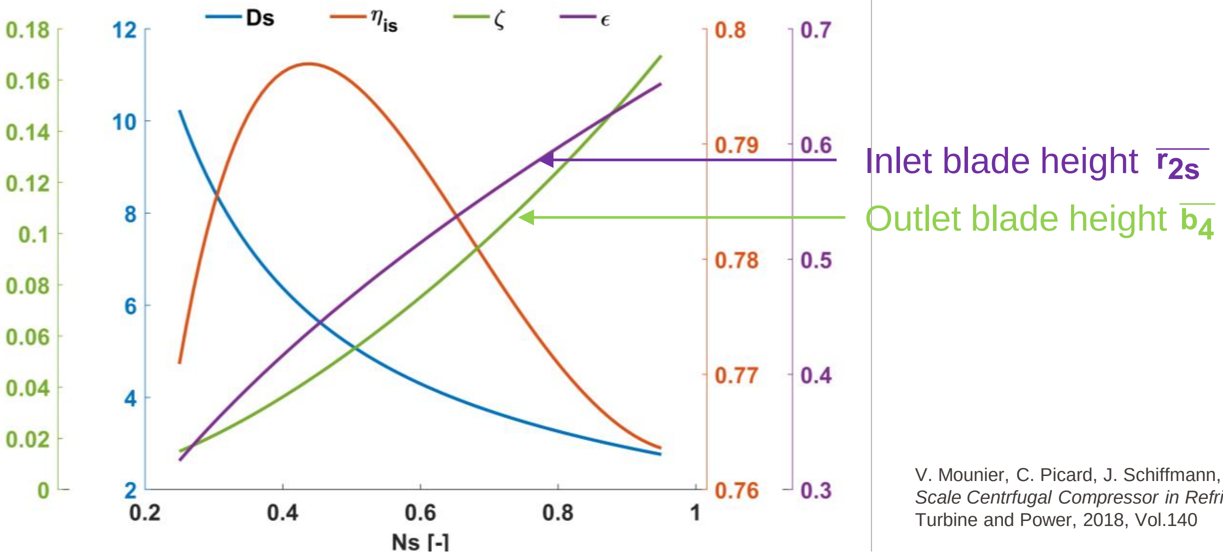




V. Mounier, C. Picard, J. Schiffmann, *Data-Driven Predesign Tool for Small-Scale Centrfugal Compressor in Refrigeration*, Journal for Engineering of Gas Turbine and Power, 2018, Vol.140

Impact of Updated Design Maps

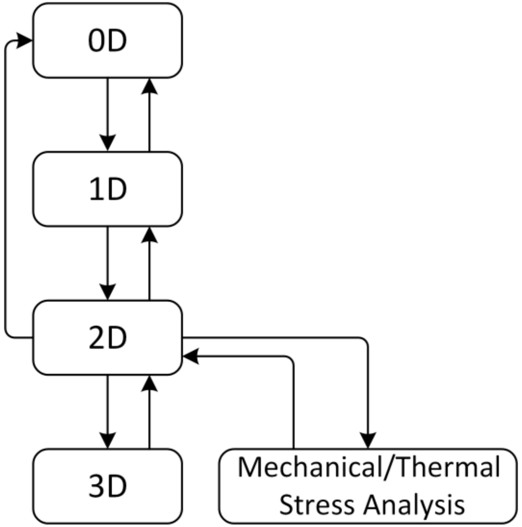
- Compared to classical tools new model yields more design information
- Data driven tool improves starting point and bypasses 1D design loop
- New tool is 1’500 x faster than meanline model with similar accuracy



V. Mounier, C. Picard, J. Schiffmann, *Data-Driven Predesign Tool for Small-Scale Centrfugal Compressor in Refrigeration*, Journal for Engineering of Gas Turbine and Power, 2018, Vol.140

Turbocompressor Design

- Design of radial compressor is iterative process
- Involves aerodynamic, mechanical, and thermal aspects
- Design starts with “first guess” and evolves by engaging increasingly complex models
- Research efforts
  - Improve meanline models
  - Increase speed of high-fidelity models
  - Surge prediction
  - Behavior of turbomachinery at small scale



Heat Pump Systems

On Small-Scale Turbocompressors for Heat Pumps

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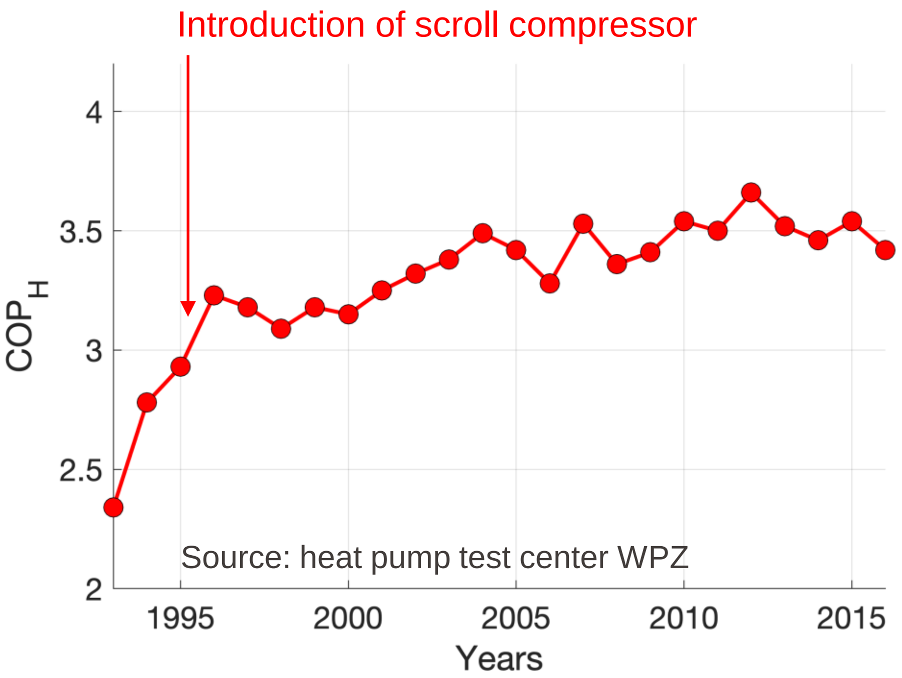
Historic Evolution of COP Trends

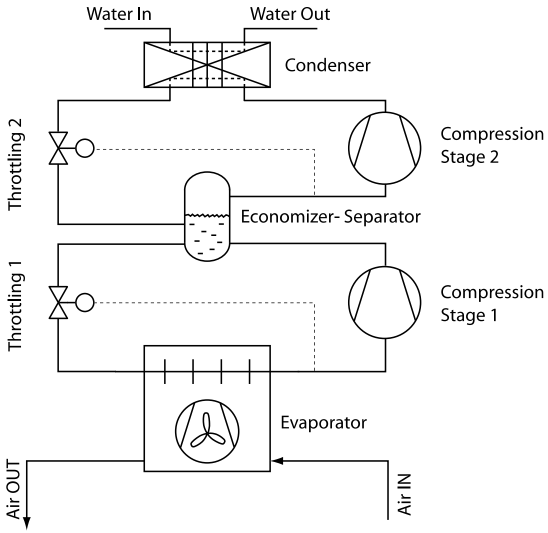


$$COP_H = \frac{q_h^-}{e^+}$$

- Since introduction of scroll compressors, COP has been rising slowly
- Key question: can another step-change be achieved and if so, how?

Key Challenges

- Assessment of losses in heat pump cycles
  - Compression 50%
  - Expansion 30%
  - Heat transfer 20%
- Possible ways to reduce losses
  - Increase compressor efficiency
  - Use oil-free technology
  - Implement multistage cycles
- Potential enabler
  - Turbocompressors on oil-free bearings



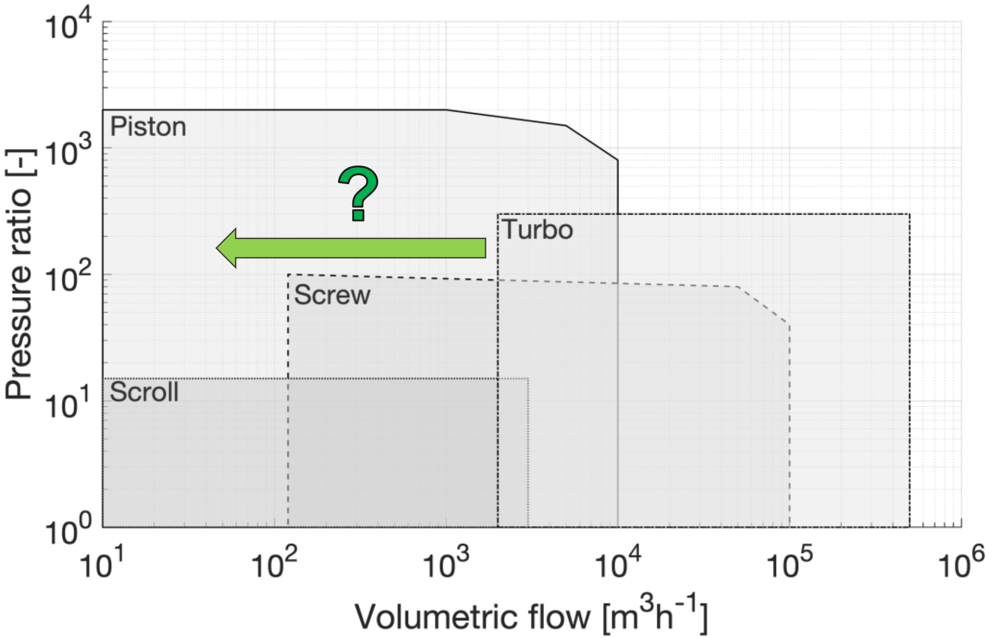
Compressor Technology Selection

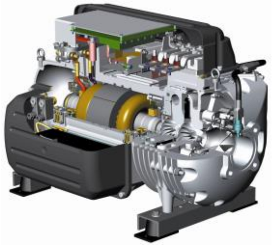
- Positive displacement machines preferred for small capacity, dynamic compressors traditionally for high capacities


weiku.com


bitzer.de


atlascopco.ch



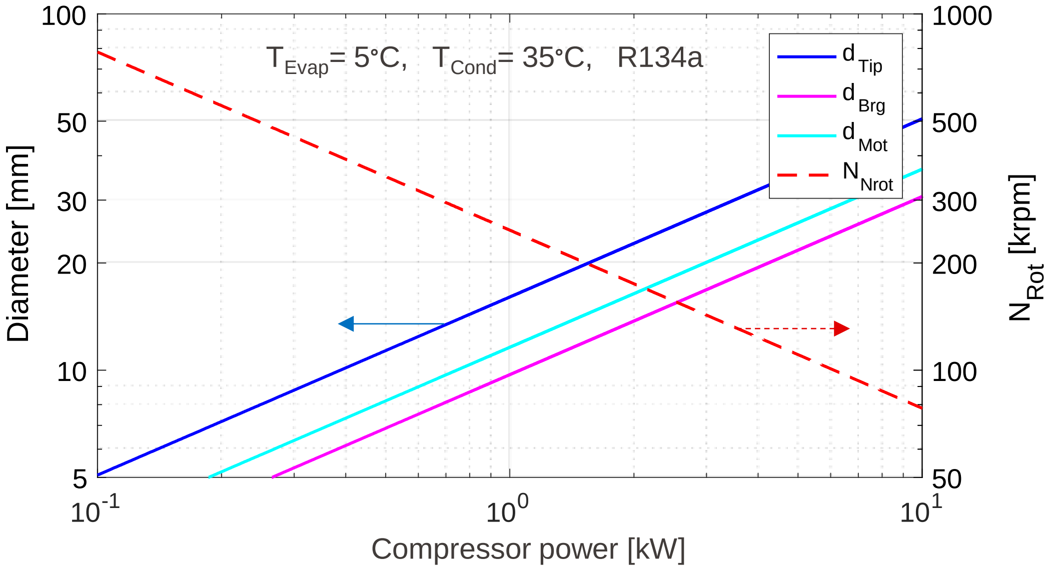

turbocor.com

Turbocompressor Scaling Analysis

- Downscaling power increases N_{Rot} and decreases size
  - Constant tip speeds → no mechanical issues

- Aerodynamic challenges
  - Increased friction due to lower Re-number and higher roughness
  - Larger relative tip-clearance
  - Non-adiabatic operation

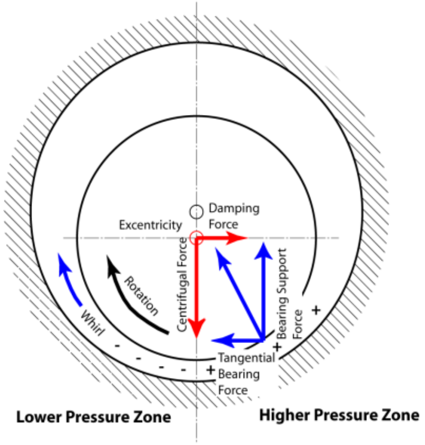
- Bearings
  - Need to support high speed and high lifetime



High Speed Bearing Classes

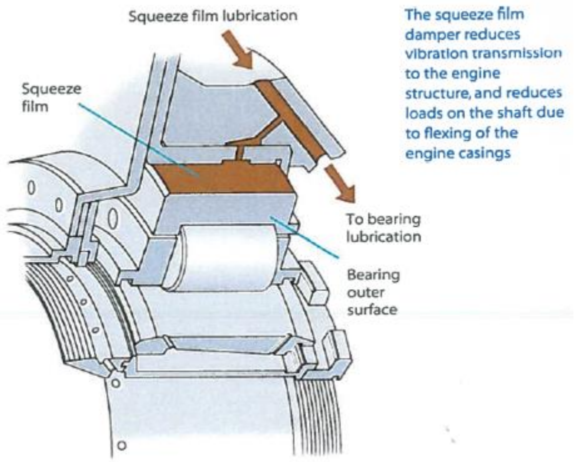
- Rolling element bearings
- Magnetic bearings
- Fluid film bearings


Courtesy of Mecos Traxler AG



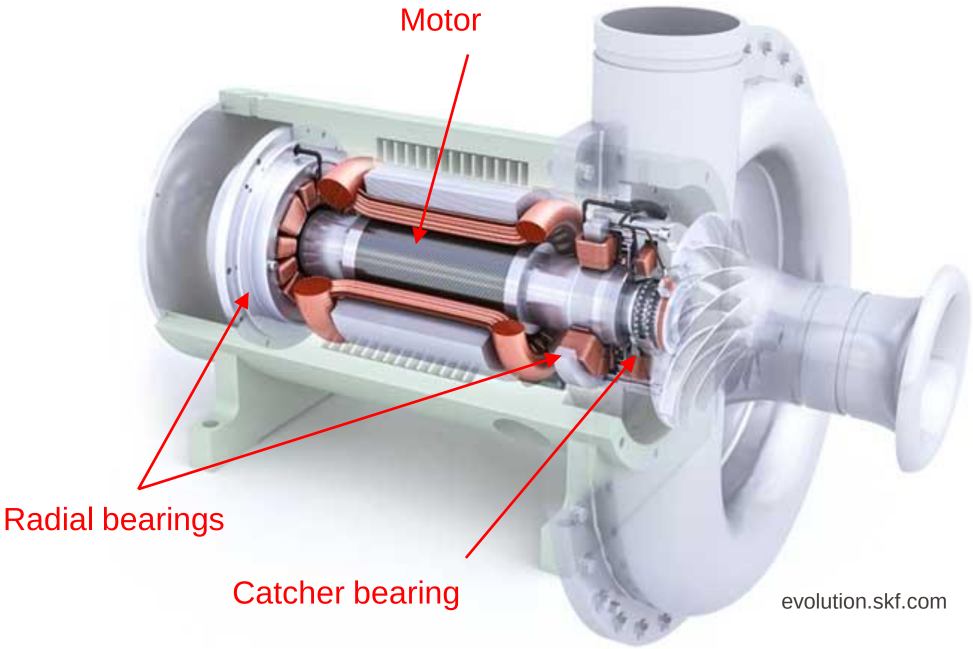
Rolling Element Bearings

- Standardized and robust technology
- Needs controlled lubrication
- Offers little damping
- Limited lifetime at high speeds (inertial forces)



Magnetic Bearings

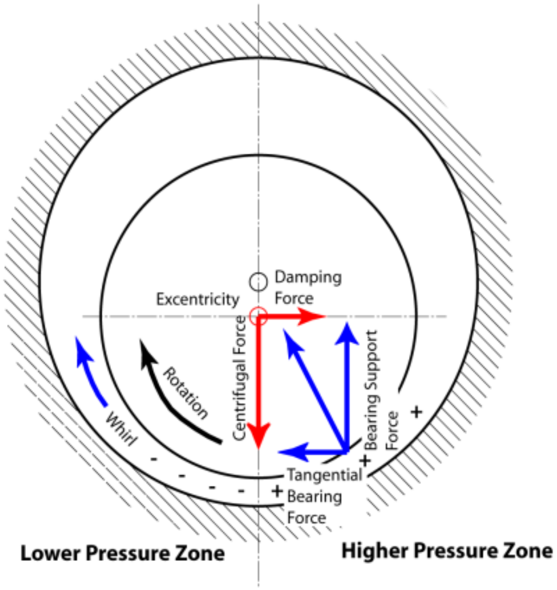
- No mechanical contact
- Work in vacuum
- Requires no lubrication
- Needs probes and controller
- Expensive and bulky
- Requires catcher bearings



evolution.skf.com

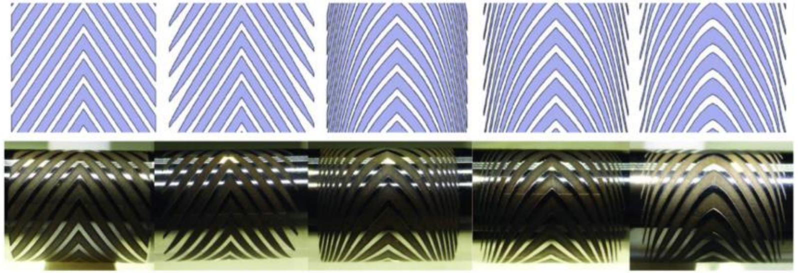
Fluid Film Bearings

- Compressible and incompressible lubricants
- Externally pressurized or dynamic
- Very simple geometry → ease of downscaling
- No wear after liftoff
- Low mechanical losses
- No cycle contamination
- Low specific load capacity and damping
- Rotordynamic stability issues



Dynamic Gas Lubricated Bearings

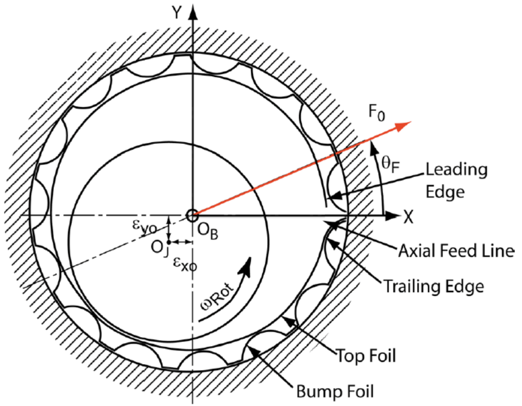
- Herringbone groove journal bearings
  - Rigid bearing bushings with v-shaped grooves
  - Tight clearances
  - Perfect alignment required
  - Accurate rotor position → low tip clearances possible
  - Very high stability threshold
  - Enhanced grooves suggested to improve performance

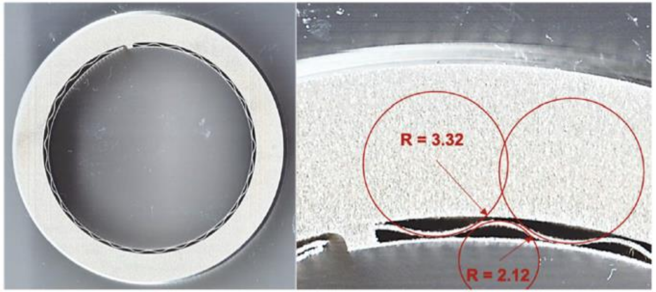


Bättig, P. K., Wagner, P. H., and Schiffmann, J. A. (March 18, 2022). "Experimental Investigation of Enhanced Grooves for Herringbone Grooved Journal Bearings." ASME. *J. Tribol*. September 2022; 144(9): 091801.

Dynamic Gas Lubricated Bearings

- Foil bearings
  - Soft support structure makes outer fluid film surface compliant
  - Tolerant to misalignment and thermal gradients
  - Friction between foils generates external damping
  - Rotor needs large orbits to generate damping → large tip clearances → large losses
  - Highly non-linear behavior
  - Prone to unstable behavior
  - Repeatable and reproducible manufacturing challenging
  - 3D-printable





Shalash, K., and Schiffmann, J. "Experimental Assessment of a 3D-Printed Stainless Steel Gas Foil Bearing." ASME. *J. Tribol*. August 2020; 142(8): 081802.

Dynamic Gas Lubricated Bearings

- Tilting pad bearings
  - Composed of several pads, each with tilting degree of freedom
  - Tilting avoids cross-coupled stiffness → ultra-high stability
  - Lower load capacity
  - Expensive & time-consuming manufacturing
  - Complex dynamic behavior

https://www.bearingsplus.com

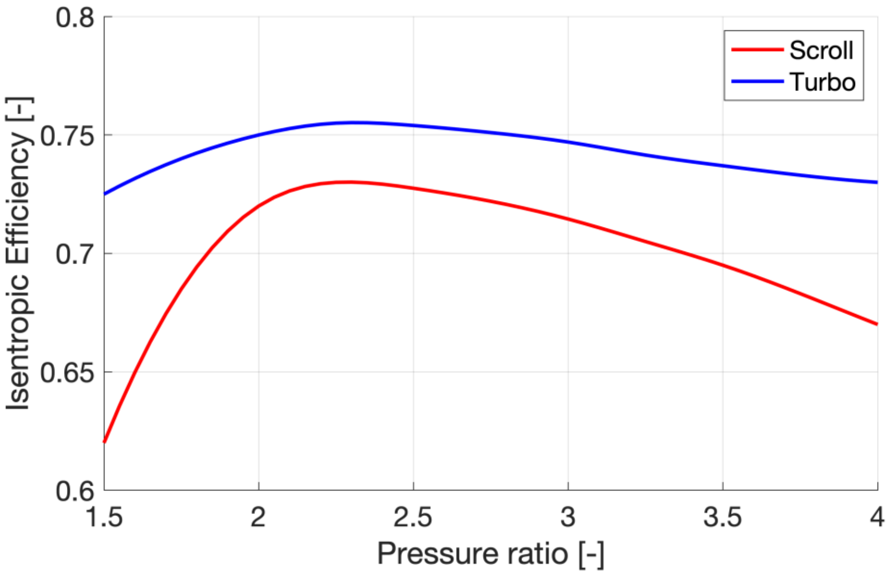
https://www.bearingsplus.com

Proof of Concept
Single Stage Compressor

- Experimental demonstration of small-scale turbocompressor
  - ø20 mm impeller, 210 krpm, 2 kW, Π 3.3, $\eta_{\text{IS-TT}}$ 0.8
  - Oil-free, R134a-lubricated bearings
  - Herringbone grooved journal bearings
  - Increased specific power compared to scroll (x10)
  - 500’000 stop & go

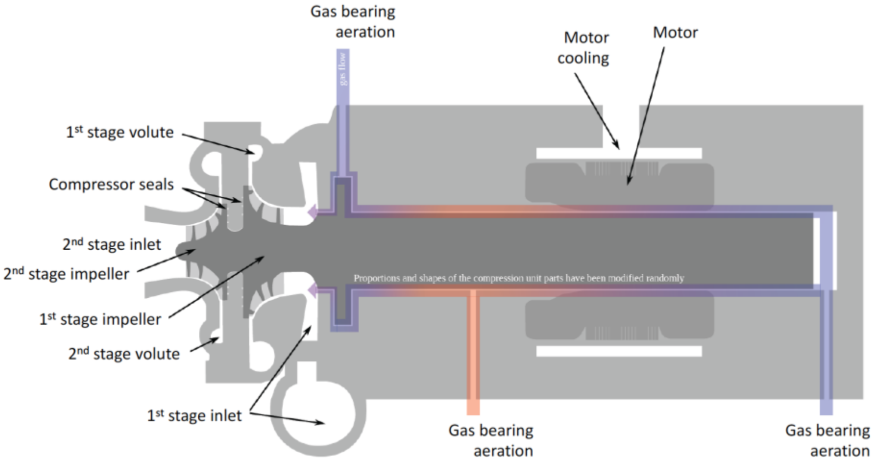
Proof of Concept Comparison

- Turbocompressor achieves higher peak efficiency than positive displacement compressor and improves off-design operation



Twin Stage Heat Pump Compressor

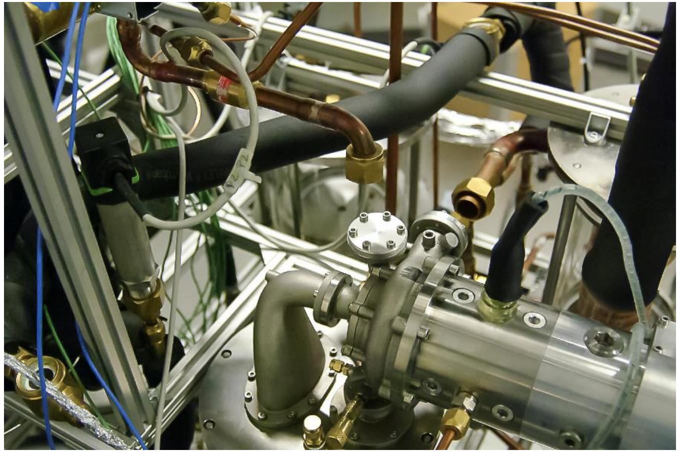
- Experimental investigation of twin stage turbocompressor for high temperature lift heat pumps
  - Both impellers on same rotor, 180 krpm, 6 kW
  - Oil-free, R134a-lubricated bearings

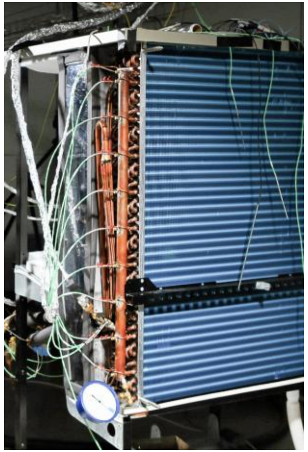


EPFL-Thesis 6764, Jean-Baptiste Carré, Experimental investigation of domestic heat pumps equipped with a twin-stage oil-free radial compressor

Twin Stage Heat Pump Compressor

- Air-water heat pump
- Twin-sage cycle with open flash tank
- Challenging flash tank design
- Competitive performance a A-7W35

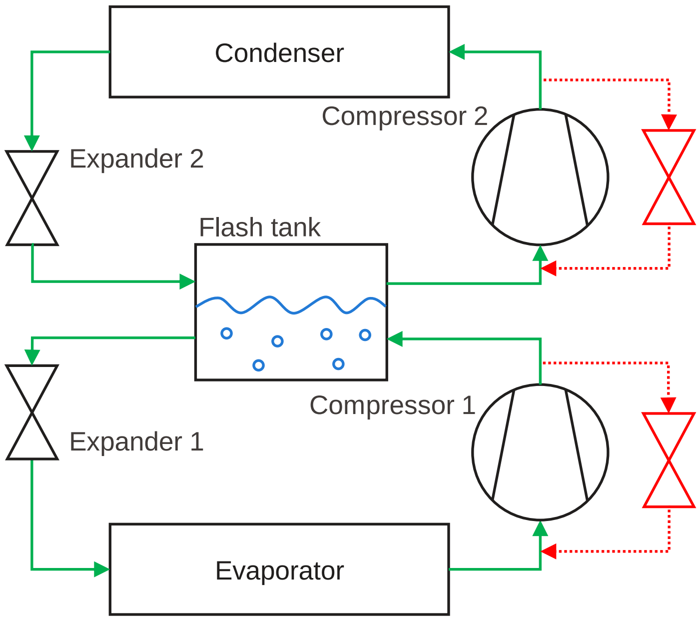




EPFL-Thesis 6764, Jean-Baptiste Carré, Experimental investigation of domestic heat pumps equipped with a twin-stage oil-free radial compressor

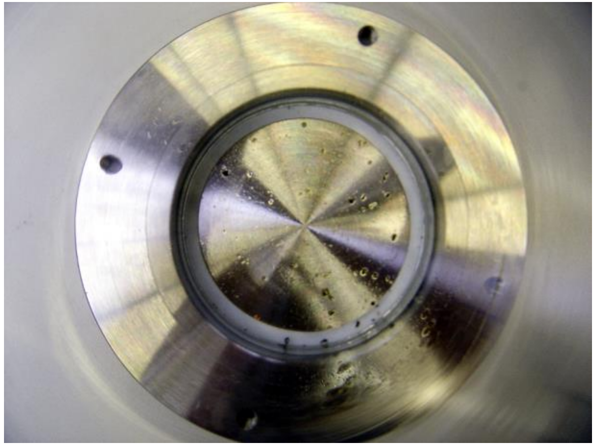
Twin Stage Heat Pump Compressor Experimental Challenges

- Thrust forces & surge
  - Compressor surge perturbs the complete cycle
  - Compressor by-pass required for hot start & stop to avoid compressor surge
  - Additional safety for compressor



Twin Stage Heat Pump Compressor Experimental Challenges

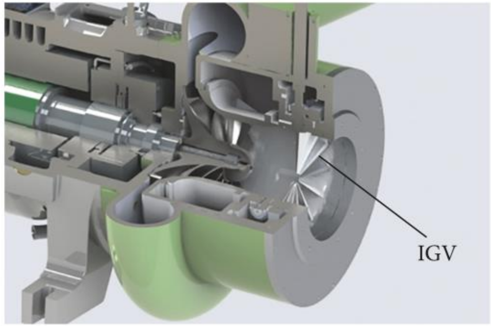
- Cycle pollution
  - Cycle contamination caused by heat exchanger and valve manufacturing process or polluted working fluid
  - Pollution may lead to bearing failure



EPFL-Thesis 6764, Jean-Baptiste Carré, Experimental investigation of domestic heat pumps equipped with a twin-stage oil-free radial compressor

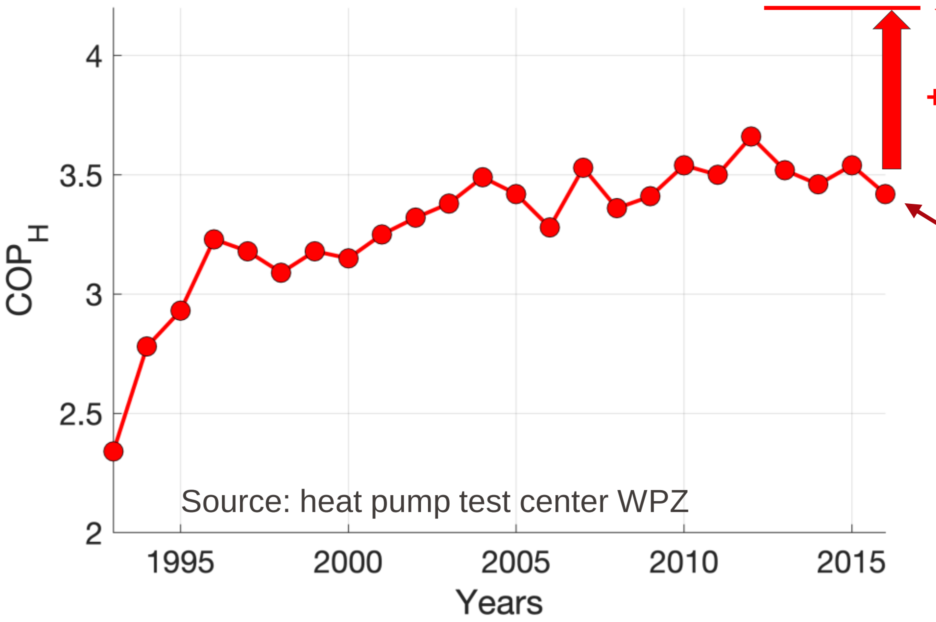
Twin Stage Heat Pump Compressor Experimental Challenges

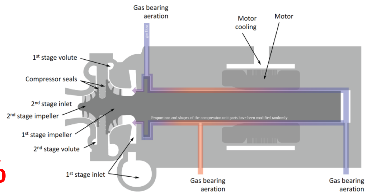
- Compressor map matching
  - Energy balance across economizer driven by mass-flow-ratio
  - Accurate compressor matching required to achieve stable economizer pressure
  - Additional control degree of freedom needed



Sun, Q., Ji, C., Fang, J., Li, C., Zhang, X., Optimization Design of IGV Profile in Centrifugal Compressor, *Mathematical Problems in Engineering*, 2017, 8437325, 9 pages, 2017

Potential of Oilfree Turbocompressors on COP





- Turbocompressor
- Multi-stage
- Oilfree

- Scroll compressor
- Single stage
- Oil lubricated

- Key question: can another step-change be achieved and if so, how?
- With multi-stage heat pump cycles driven by oilfree turbocompressors

Summary of Challenges of Reduced-Scale Turbomachinery

- Gas lubricated bearings require small clearances to achieve stable operation → manufacturing cost → industrialization challenging
- Low bearing clearances require clean working fluids
- Reduced scale compressors efficiency suffers from Re-number effects, increased relative surface roughness and large tip clearances
- Reduced scale turbomachinery more sensitive to heat fluxes
- Leakage between stages is increased due to scaling effects

Outlook W12

- Heat exchanger design

Exercises W11

- Comprehension questions
- Centrifugal compressor analysis
- Flow through a turbine runner