Horizon 2020
European Union Funding
for Research & Innovation

sCO2flex

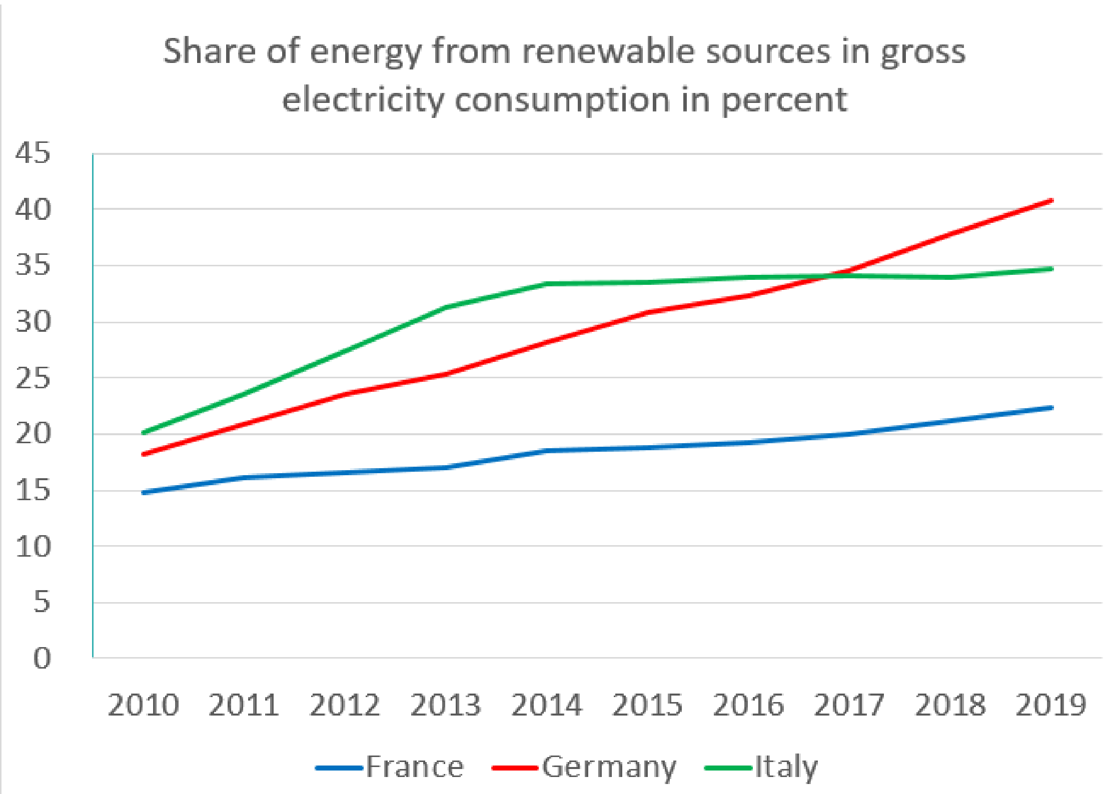
AN ATTEMPT FOR ESTABLISHING PRESSURE RATIO PERFORMANCE MAPS FOR SUPERCRITICAL CARBON DIOXIDE COMPRESSORS IN POWER APPLICATIONS

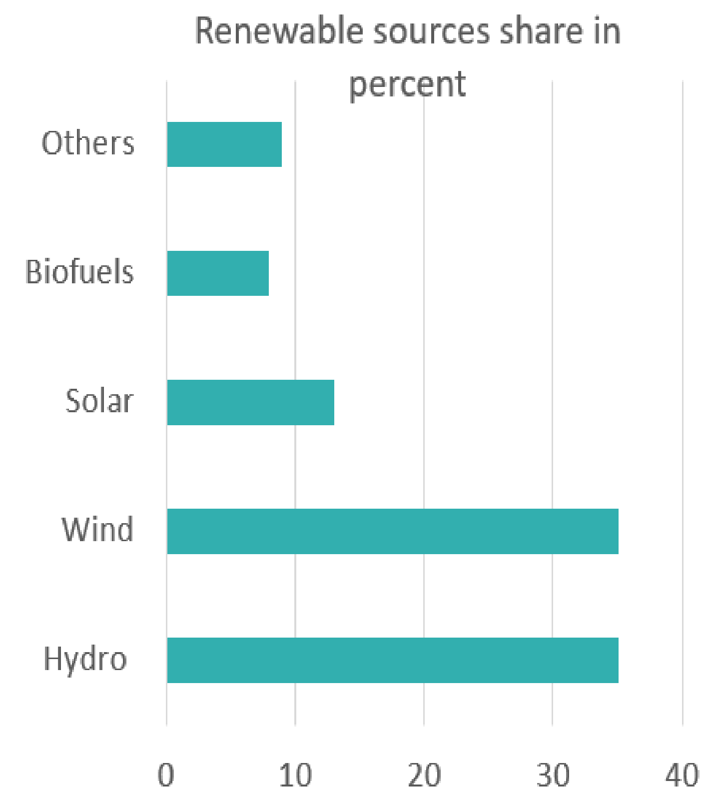
Ihab Abd El Hussein, M.Sc. ▪ 24-03-2021

sCO_2-flex motivation

UNIVERSITÄT DUISBURG ESSEN

Open-Minded





- **Steady increase in renewables share**
- **Solar and wind energy represent ~ 50 % of the renewable sources share**

⇒ **High intermittency in the power generation**

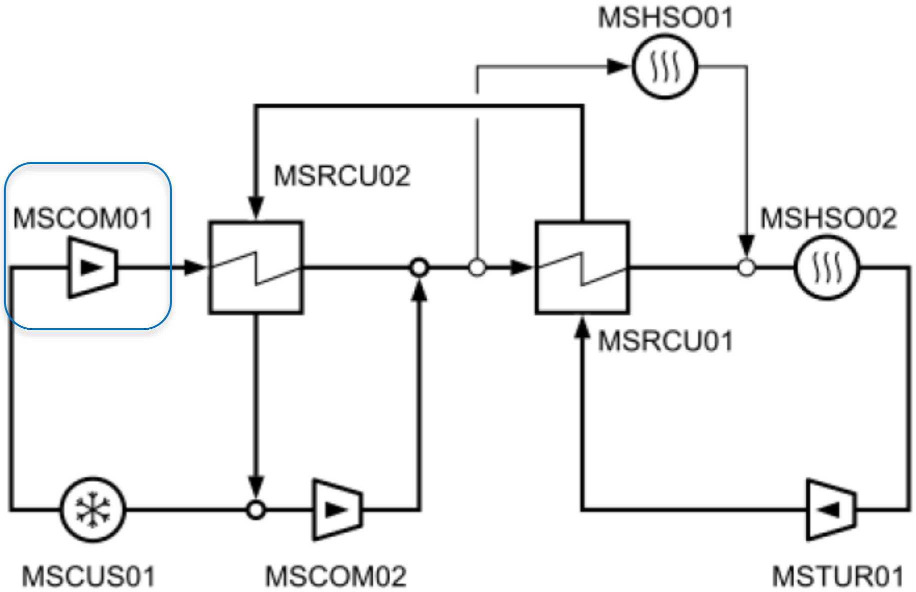
Goal:

Design a 25 MWe closed joule cycle with sCO_2 as working medium

- **Reach low off-design loads**
- **High load ramp-rate**
- **High efficiency at the off-design condition**
- **Bring the technology to a technical readiness level TRL 6**

sCO_2-flex cycle layout

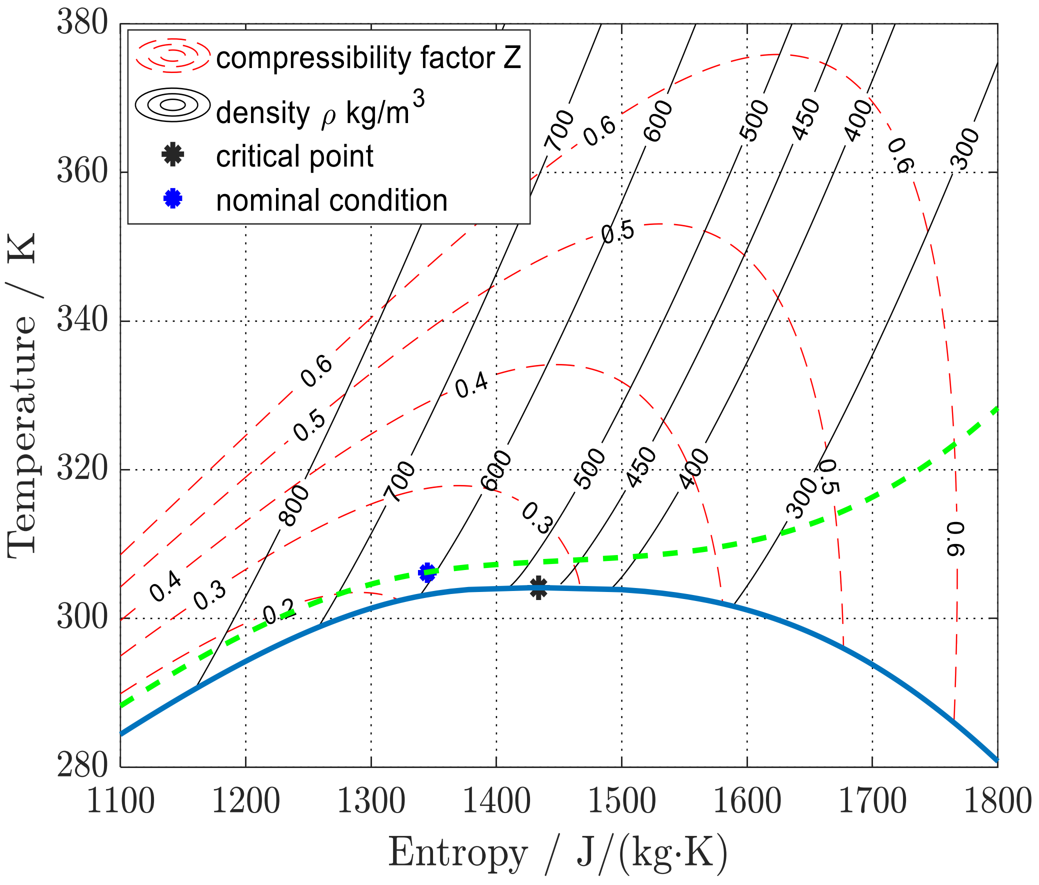
- **Recompression cycle**
- **High Temperature recuperator bypass**
- **Inlet condition 80 bar 306 K**



Main compressor with inlet condition near the critical point

- **High variation of CO_2 thermodynamic property**
- **An increase of only one Kelvin from the nominal condition leads to more than 10 percent change in density**

=>
Required: compressor performance map valid for any inlet condition



Buckingham Π-theorem

Compressor performance η, π, ψ_h function of four parameters

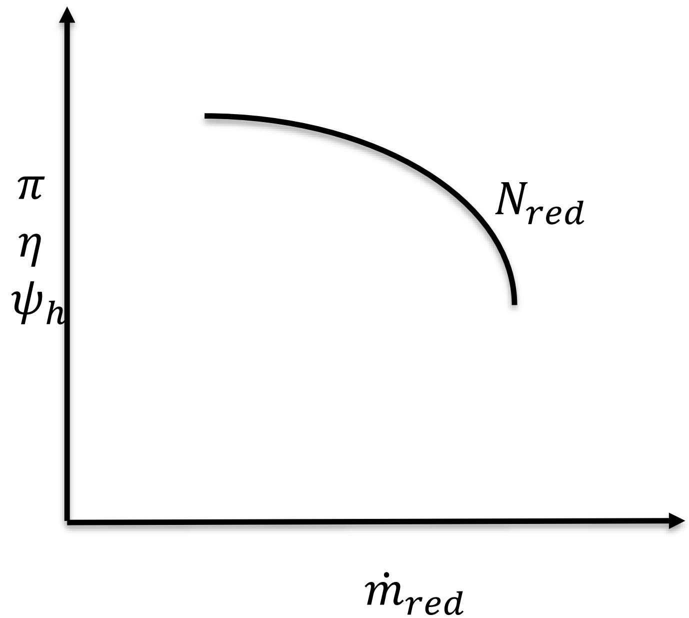
$$\dot{m}_{red} = \frac{\dot{m}}{\rho_{0t} \cdot a_{0t} \cdot D_2^2}$$

$$N_{red} = \frac{N \cdot D_2}{a_{0t}}$$

$$Re = \frac{\rho_{0t} \cdot N \cdot D_2 \cdot b_2}{\mu_{0t}}$$

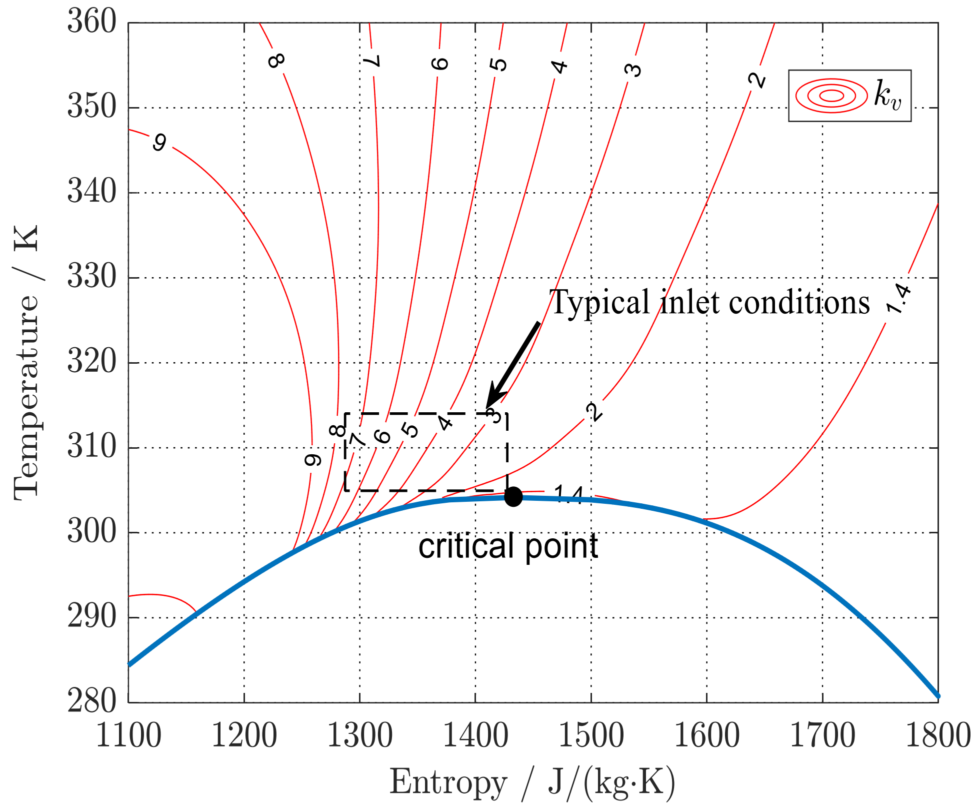
$$k_v = -\frac{v}{p} \cdot \frac{c_p}{c_v} \cdot \left(\frac{\partial p}{\partial v}\right)_T$$

=>



Isentropic exponent k_v for sCO_2

- **The isentropic exponent of sCO_2 can vary substantially**
- **The isentropic exponent can vary from around 3 to around 8**
- **Typical isentropic exponent value for fluid obeying the ideal gas law ranges between 1 to 2**



UNIVERSITÄT DUISBURG ESSEN

Open-Minded

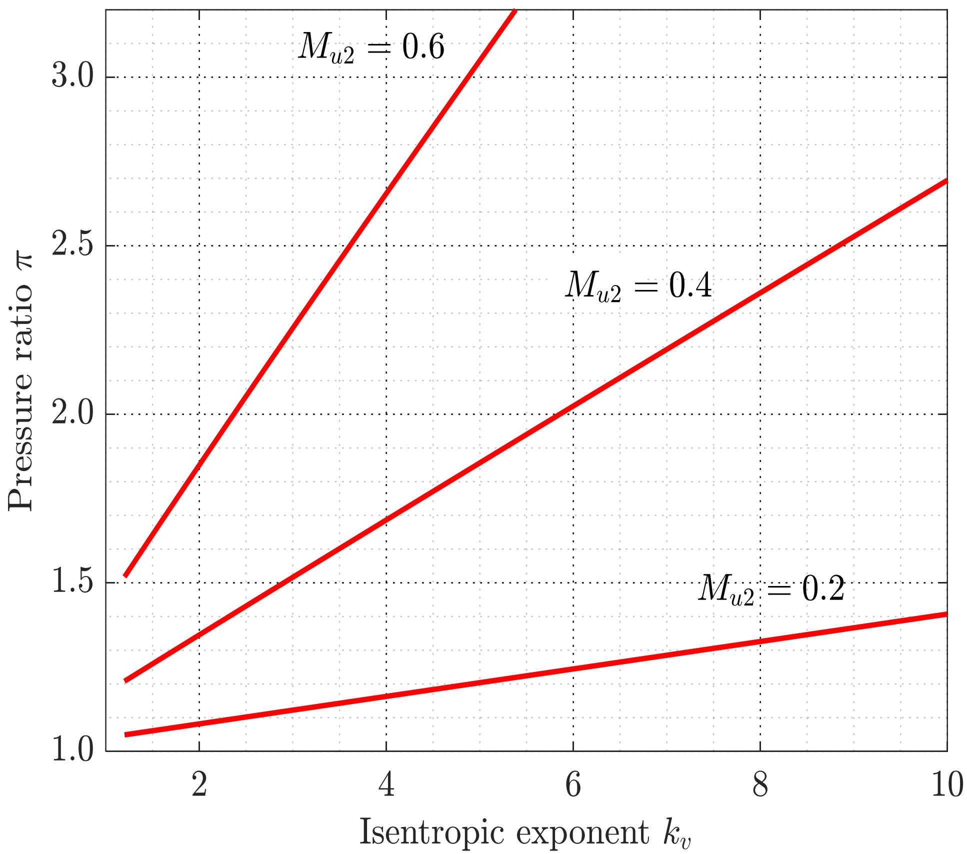
Assumption :

- **Isentropic compression of a radially bladed compressor**
- **Constant k_v along the compression path**

=>

$$\pi = \left[M_{u2}^2 \cdot (k_v - 1) + 1\right]^{\frac{k_v}{k_v - 1}}$$

Pressure ratio varies substantially with the isentropic exponent



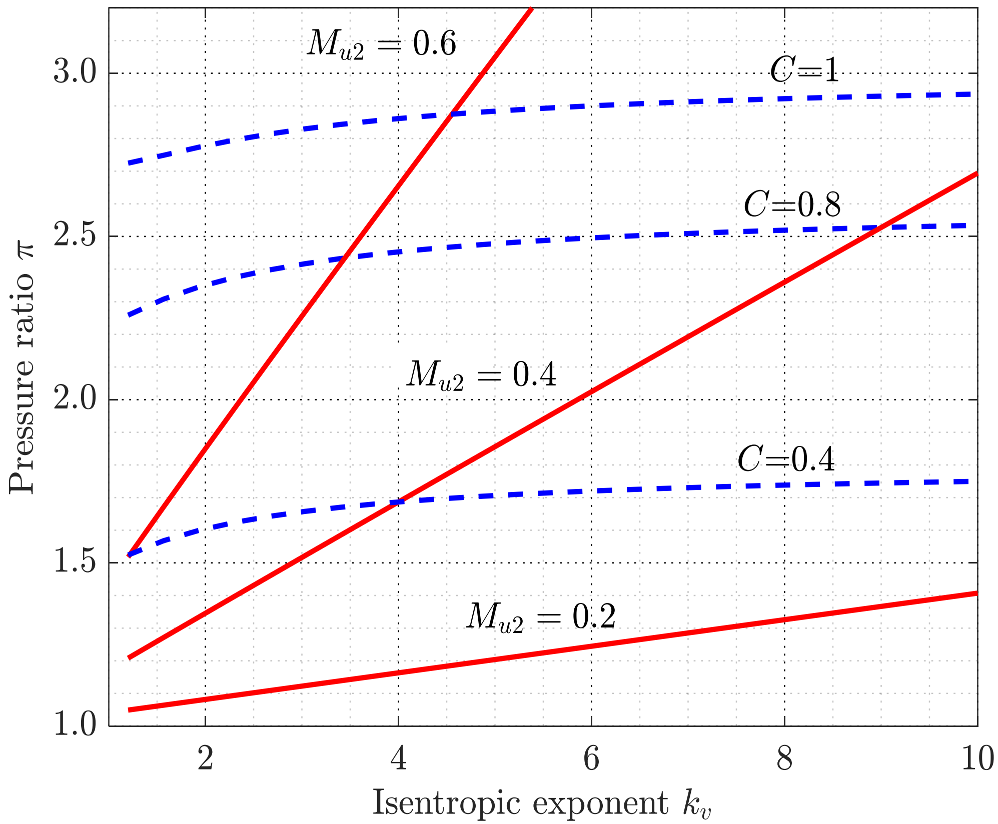
- **Reducing the rotational speed with the critical velocity**

$$N_{red} = \frac{N \cdot D_2}{v_{cr}} \propto \frac{\Delta h}{v_{cr}^2} = C$$

=>

Reduction of pressure ratio dependence on the isentropic exponent k_v

$$\pi = \left[2 \cdot C \cdot \frac{k_v - 1}{k_v + 1} + 1\right]^{\frac{k_v}{k_v - 1}}$$



local sound speed for Ma=1

$$v_{cr} = \sqrt{\frac{2 \cdot k_v}{k_v + 1} \cdot Z_{ot} \cdot R \cdot T_{ot}}$$

Glassman approach

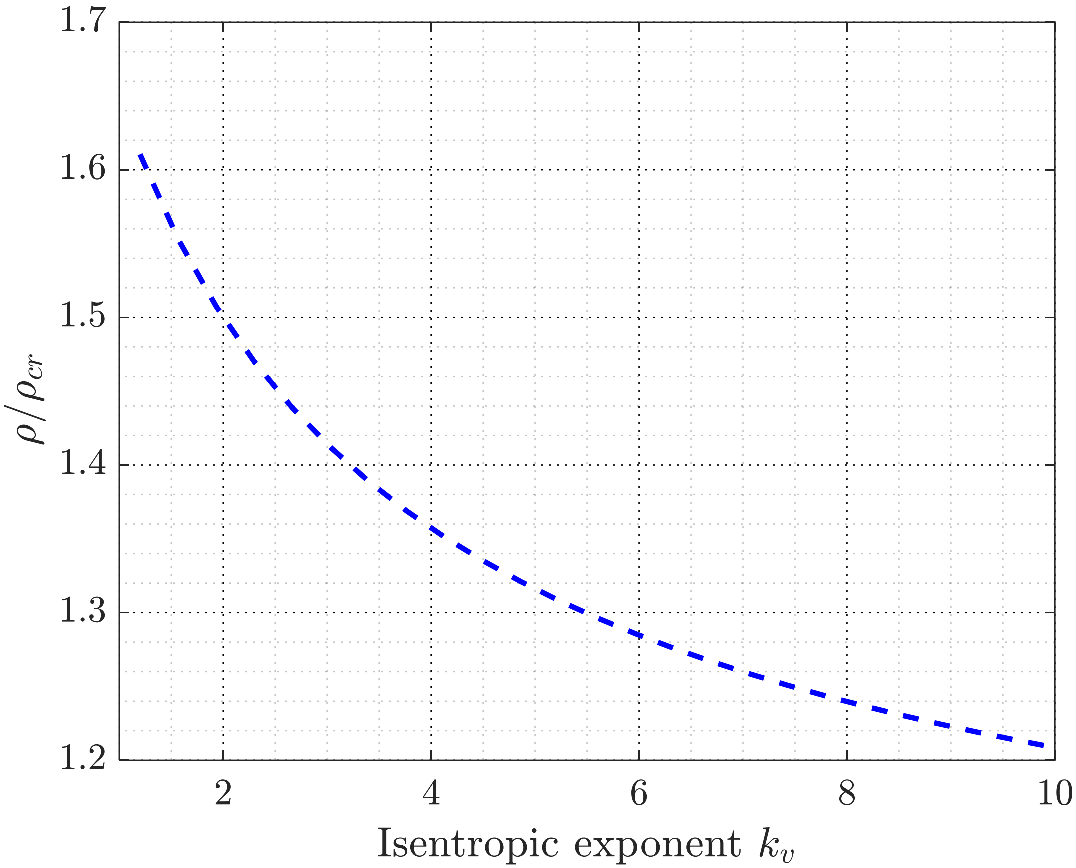
- **Mass flow rate is reduced with the critical mass flow rate in the original approach**

$$\dot{m}_{red} = \frac{\dot{m}}{\rho_{cr} \cdot v_{cr} \cdot D_2^2}$$

- **ρ/ρ_{cr} varies substantially in terms of k_v**

=>

$$\dot{m}_{red} = \frac{\dot{m}}{\rho_{0t} \cdot v_{cr} \cdot D_2^2}$$



Glassman approach

- **Additional term σ is added to match to volume flow ratio across the compressor**

- **For a polytropic compression**

$$pv^n = constant$$

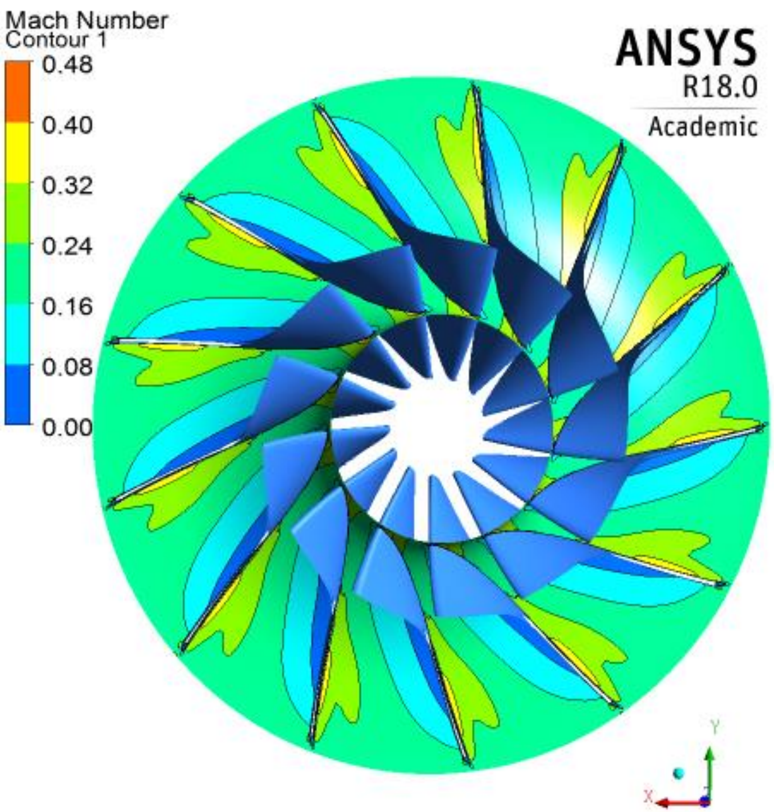
The polytropic work can be estimated

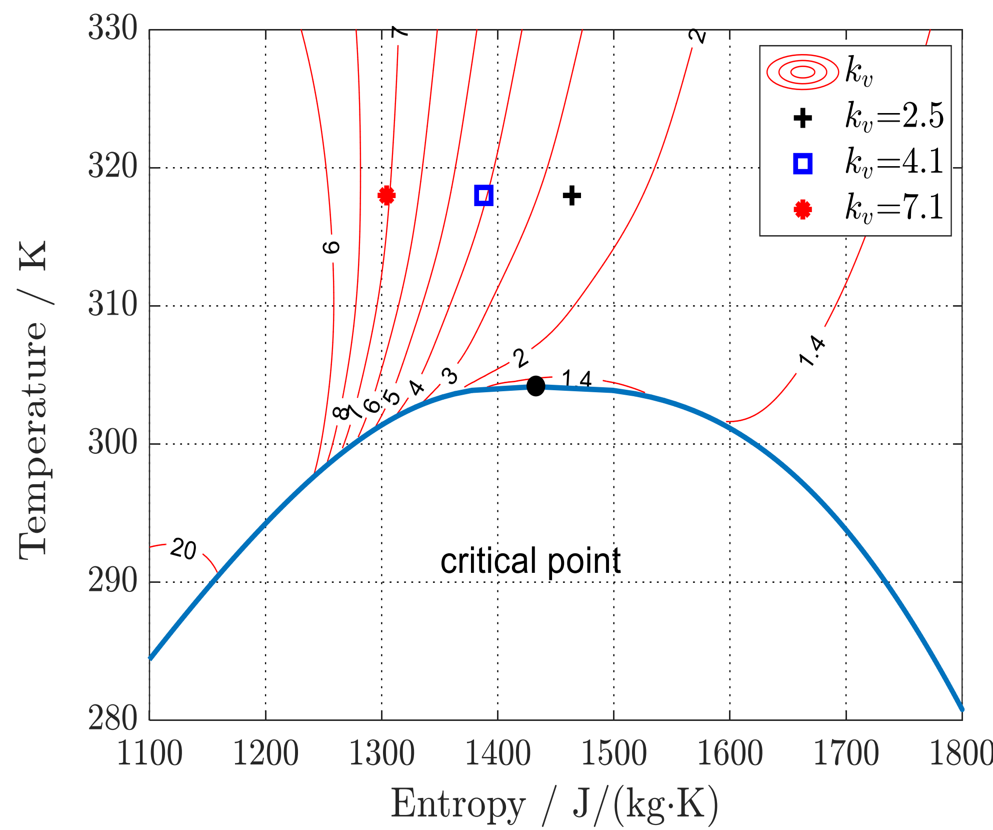
$$y = \frac{n}{n-1} \cdot Z \cdot R \cdot T \cdot (\pi^{\frac{n-1}{n}} - 1)$$

$$\Rightarrow \sigma = \frac{\dot{V}_{out}}{\dot{V}_{in}} = \pi^{-\frac{1}{n}} = \left(2 \cdot C \cdot \eta \cdot \frac{k_v}{k_v+1} \cdot \frac{n-1}{n} + 1\right)^{-\frac{1}{n-1}}$$

$$\Rightarrow \dot{m}_{red} = \frac{\dot{m} \cdot \sigma}{\rho_{0t} \cdot v_{cr} \cdot D_2^2}$$

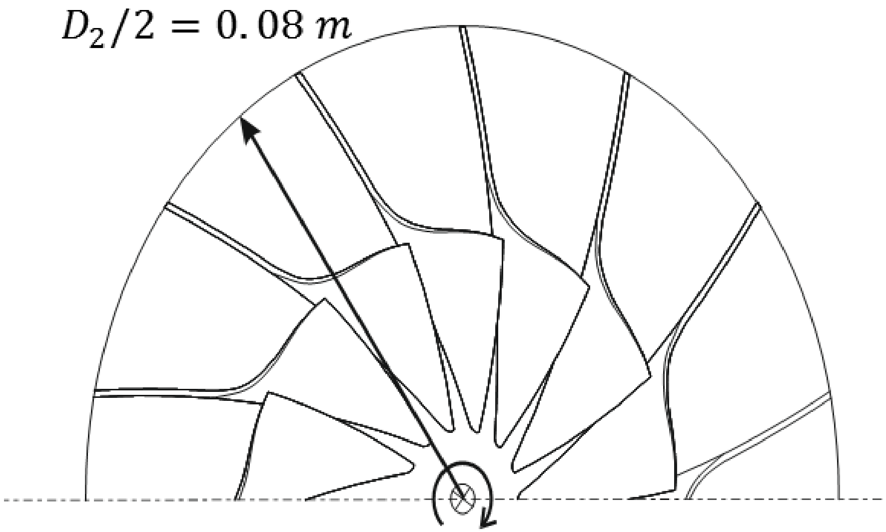
Methodology





Compressor geometry

- **In-house design tool for sCO2 radial compressors**
- **Typical specific speed and specific diameter**
- **Vaneless diffuser and rectangular volute**



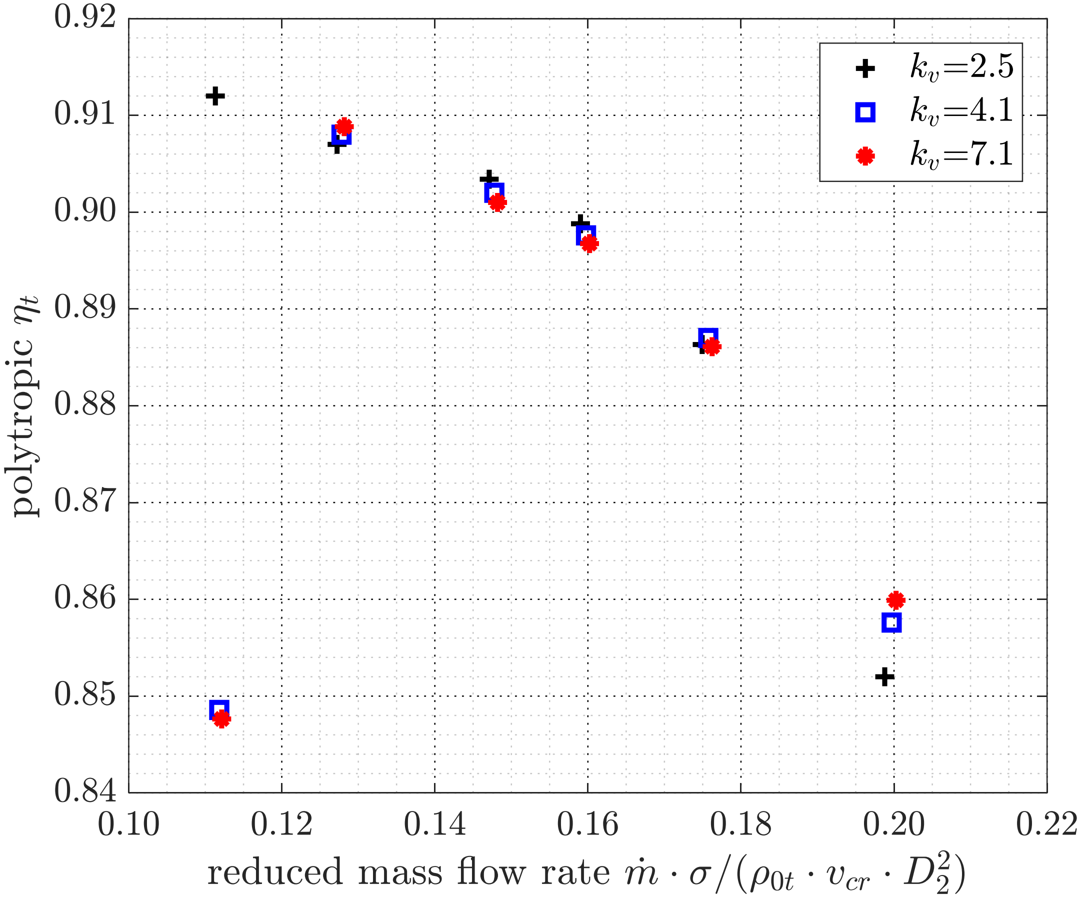
parameter	D_2	β_2	β_{1s}	Specific speed	Specific diameter
value	0.16 m	105°	170°	0.23	4.5

Efficiency comparison

- **Total efficiency**

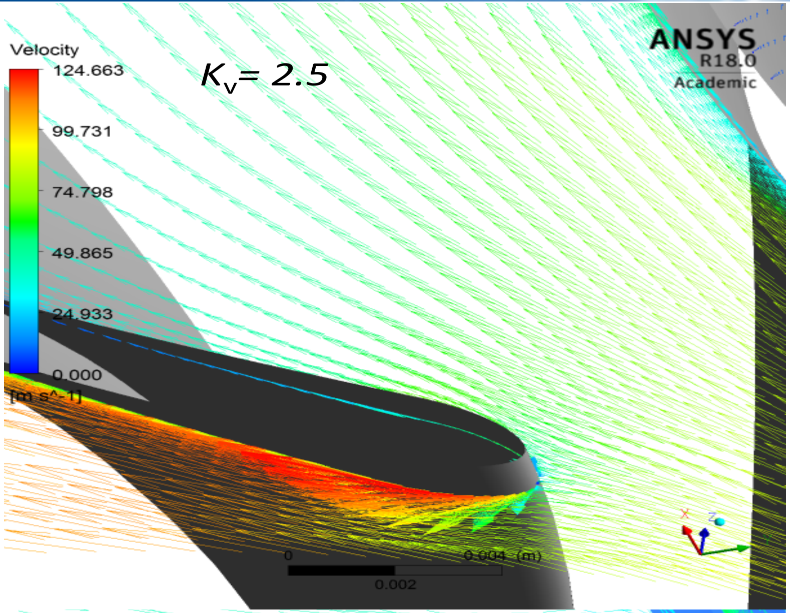
$$\eta_t = \frac{y_t}{\Delta h_t}$$

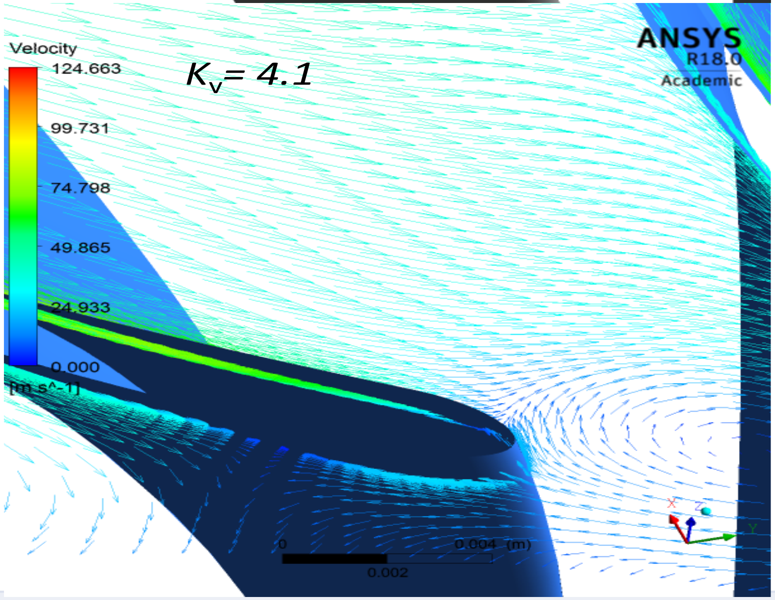
- **Overall good agreement of results**
- **Small difference at high flow rate**
- **Left to the BEP of the reference condition k_v=4.1 high difference in comparison with k_v=2.5**



Recirculation at the impeller inlet

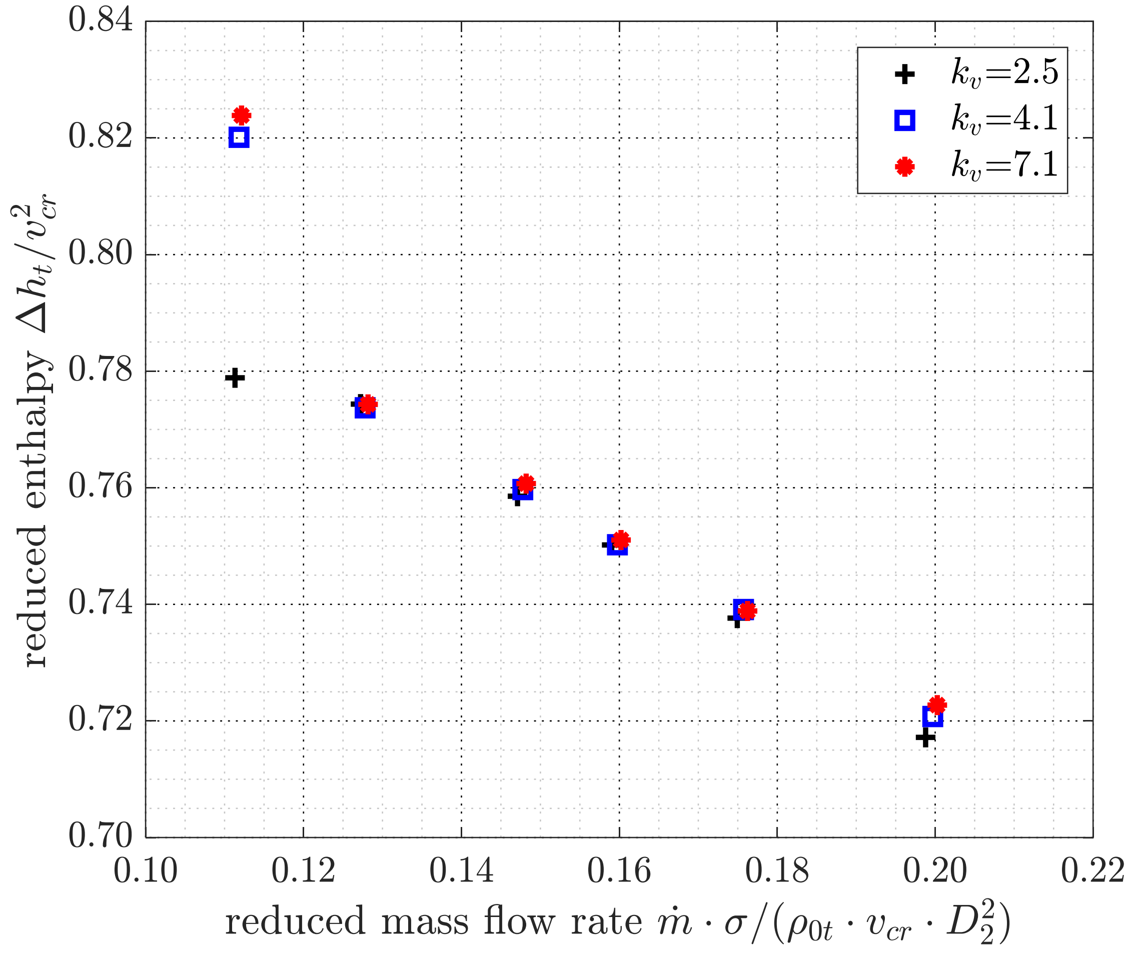
- **High recirculation takes place near the shroud for k_v=4.1**
- **Inlet flow coefficient for k_v=4.1 is smaller than for k_v=2.5**
- **The non-conventional steep blade angle near the shroud causes non-conventional steep drop in the efficiency**





Reduced enthalpy

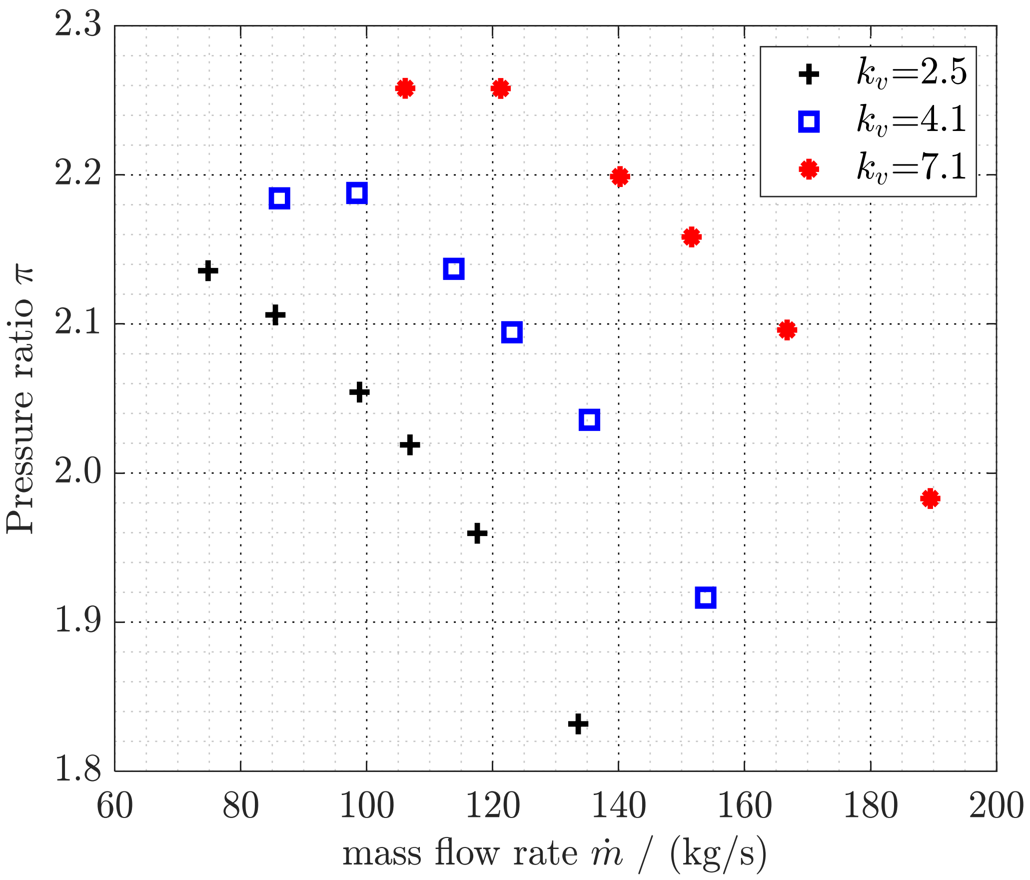
- **Overall, good match of the reduced enthalpy change**
- **High difference at low reduced mass flow rate**



Chair of Turbomachinery

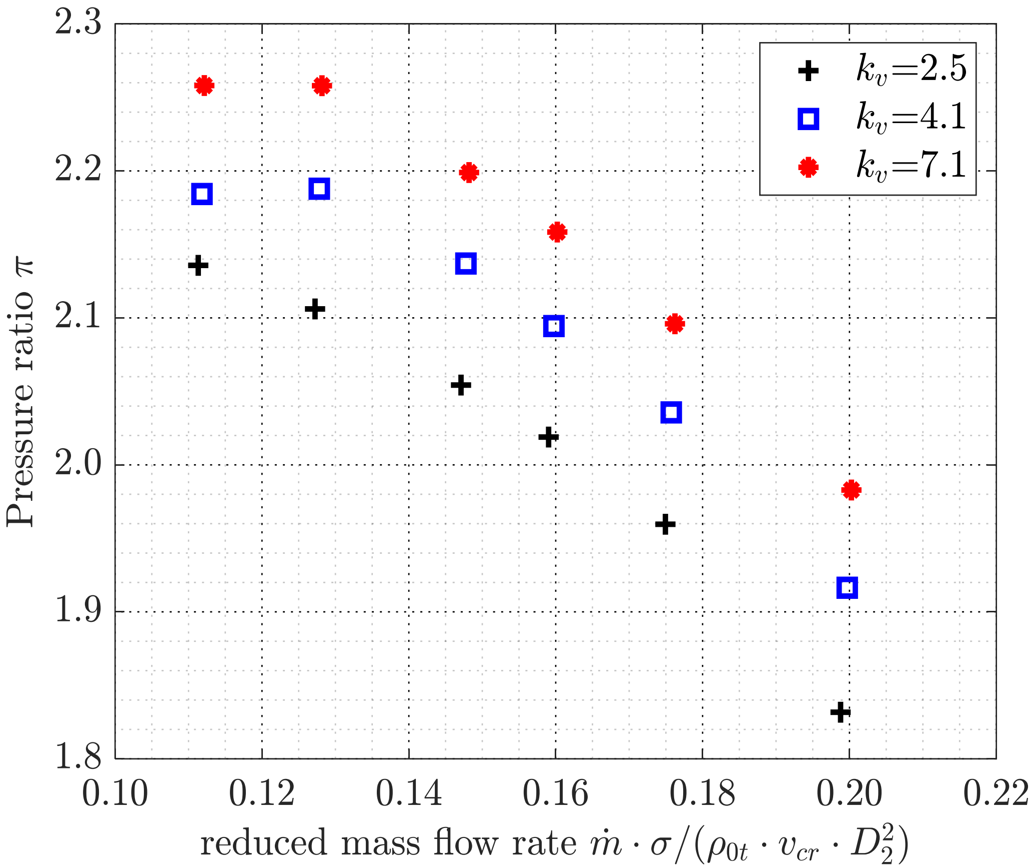
Pressure ratio map

- **Small variation in the pressure ratio along speed lines, β_2=105°**
- **Wide variation in mass flow rate and pressure ratio**



Pressure ratio map

- **Around 3 percent deviation between the pressure ratio for kv=4.1 and kv=7.1**
- **Deviation increase up to 8 percent between kv=2.5 and kv=7.1**
- **Results show consistent effect of the isentropic exponent on the pressure ratio**



Conclusion

- **An attempt to establish pressure ratio performance map is conducted**
- **Effect of the isentropic exponent on the pressure ratio is shown**
- **Implementation and modification of the so-called Glassman approach is introduced for sCO_2 compressors in order to reduce the pressure ratio dependency on the isentropic exponent**
- **Model verification conducted with help of CFD**
- **Deviation of the pressure ratio is limited to 3 percent for high isentropic exponent values (4.1 – 7.1)**
- **For isentropic exponent range (2.5 -7.1) pressure ratio deviation increases to 8 percent**

UNIVERSITÄT DUISBURG ESSEN

Open-Minded

Prof. Dr.-Ing. Dieter Brillert

Chair of Turbomachinery

Thank you for your attention!

Ihab.abd-el-hussein@uni-due.de