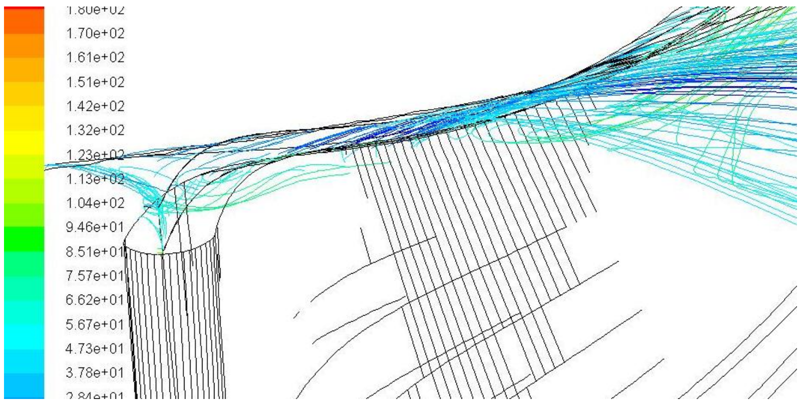
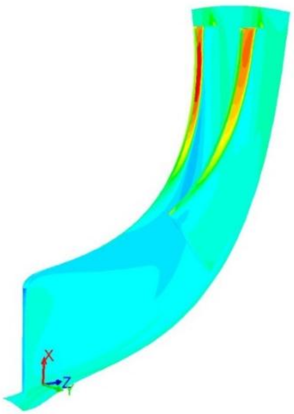
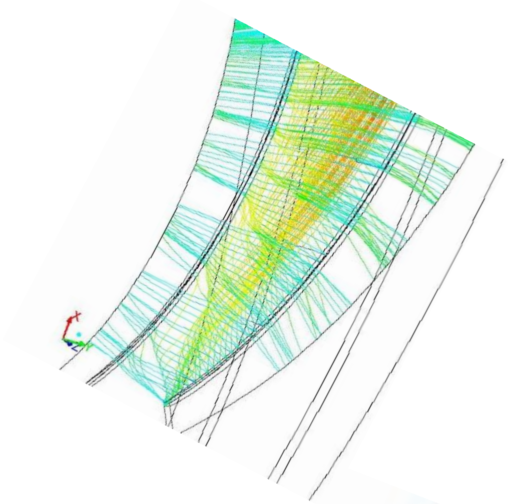
1D AND 3D TOOLS TO DESIGN SUPERCRITICAL CO_2 RADIAL COMPRESSORS: A COMPARISON







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7. SUMMARY AND CONCLUSIONS

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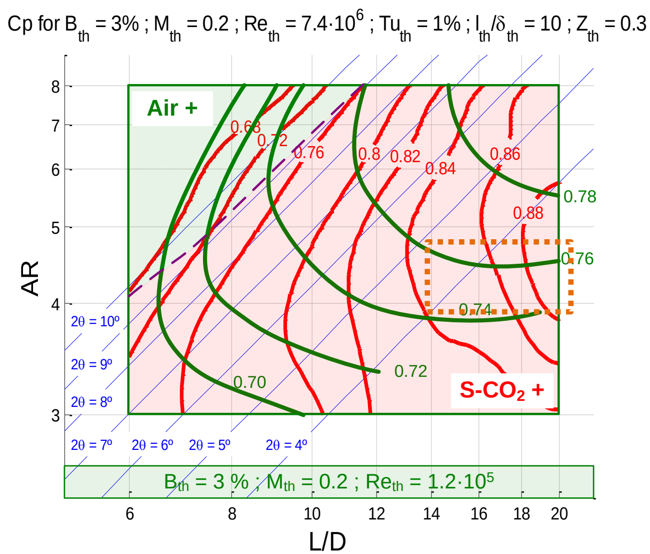
1. INTRO – Review of past work

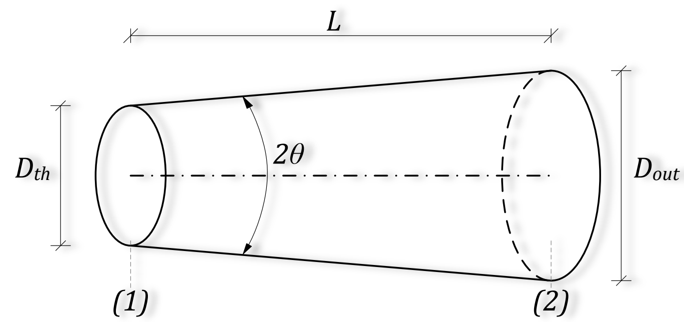
- The **Thermal Power Group at the** **University of Seville** and the **Department of Power & Propulsion** at **Cranfield University** have researched the performance of S-CO_2 diffusers.

B. Monje et al., Comparing the pressure rise of air and supercritical carbon dioxide in conical diffusers, Paper GT2012-69835, Turbo Expo 2012, Copenhagen.

A. López et al., Effect of turbulence intensity and flow distortion on the performance of conical diffusers operating on supercritical carbon dioxide, Paper GT2013-94009, Turbo Expo 2013, San Antonio.

B. Monje et al., Aerodynamic analysis of conical diffusers operating with air and supercritical carbon dioxide, International Journal of Heat and Fluid Flow, In press (2013).





$$(K, C_p) = \mathcal{F}(AR, L/D_{th}, Re, M, B_{th}, T_u, l_{tu}/\delta^*, \alpha_{dist}, \alpha_{swirl}, Z, \gamma)$$

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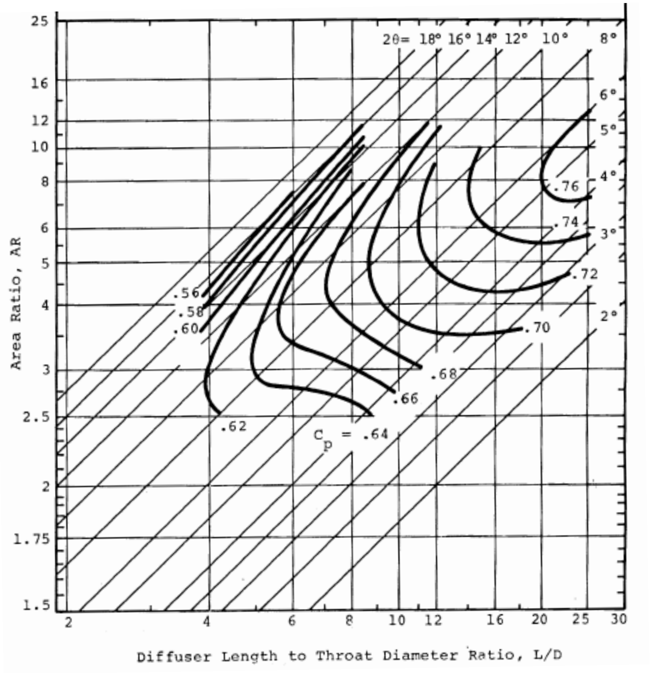
- The **Thermal Power Group at the** **University of Seville** and the **Department of Power & Propulsion** at **Cranfield University** have researched the performance of S-CO_2 diffusers.

- Outcome: understanding S-CO_2 turbomachinery design gained by exploring the design space, as done by other researchers in the past (for air).



F.X. Dolan, P.W. Runstadler Jr., Pressure recovery performance of conical diffusers at high subsonic Mach numbers, NASA CR 2299, 1973.

David Japikse, Turbomachinery diffusers design technology. Concepts ETI, Inc., Norwich, VT.

1. INTRO – This work

- The **Thermal Power Group at the University of Seville** and the **Department of Power & Propulsion** at **Cranfield University** have researched the performance of S-CO_2 diffusers.
- Outcome: understanding S-CO_2 turbomachinery design gained by exploring the design space, as done by other researchers in the past (for air).
- The work presented today aims to

  1. *Assess the suitability of a particular 1D model to design efficient **S-CO_2 compressors**.*
  2. *Use 3D-CFD analysis to determine if **local flow phenomena** are likely to be overlooked by a 1D tool: condensation, shock waves, choke…*

1. INTRO – This work

STEP 1 **Diffusers:** Air vs. S-CO2 → 2D CFD (ANSYS Fluent®)

STEP 2 **(Radial) Compressor:** Air vs. S-CO2

0) TBD: Is the standard design practice valid for S-CO_2?

STEP 3 **Experimental validation of diffusers**

STEP 4 **Experimental validation of compressors**

Matlab® → 1D

Draft design

VS.

Impeller + Diffuser
ANSYS Fluent® → 3D

3D design
(not optimized)

1) Validation and/or comparison → Design guidelines/correlations still valid?

2) If not, **how do they change?**

2.1) How far from expected efficiency?

2.2) New design guidelines (hopefully) developed

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2. THE ONE-DIMENSIONAL MODEL

- 1D Mean streamline code based on **Conservation Laws** and **Empirical Correlations**:

Mass conservation

$$\dot{m} = \rho_i v_{m,i} A_i (1 - B_i)$$

2nd principle

$$\Delta s_0 \geq 0$$

Energy conservation

$$h_{0,i} = h_{0,i-1} \qquad \textit{(Stationary elements)}$$

$$h_{0,i} + \frac{w_i^2}{2} - \frac{u_i^2}{2} = h_{0,i-1} + \frac{w_{i-1}^2}{2} - \frac{u_{i-1}^2}{2} \qquad \textit{(Rotating elements)}$$

Deviation

Impeller: slip factor, σ (Wiesner)

Diffuser: deviation angle, δ (Howell)

Volute
- *Friction*
- *Meridional losses*
- *Tangential losses*

Diffuser
- *Incidence*
- *Choking at the throat*
- *Friction*
- *Aerodynamic loading*
- *Mixing processes*

Pressure losses

(Stationary elements)

$$P_{0,i} = P_{0,i-1} - (P_{0,i-1} - P_{i-1}) \sum \bar{\omega}$$

(Impeller)

$$P'_{03} = P'_{02} - f_c (P'_{02} - P_2) \sum \bar{\omega}$$

$$f_c = \begin{cases} (\rho'_{03} T'_{03})/(\rho'_{02} T'_{02}) & \textit{(Aungier's proposal)} \\ P'_{03}/P'_{02} & \textit{(Used in this work)} \end{cases}$$

Work input coefficients (secondary)

Disk friction

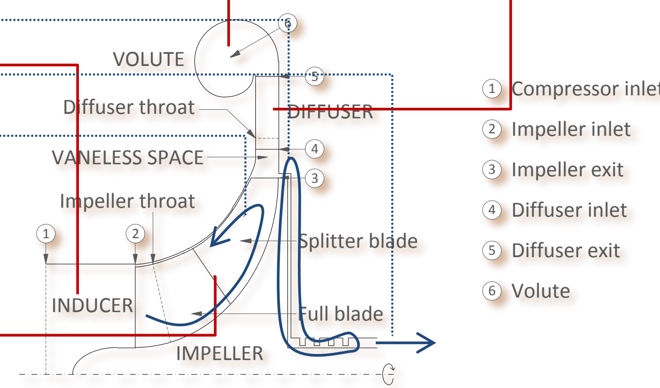
Leakage

Recirculation

$$\frac{\Delta H}{u_2^2} = I_B + I_{DF} + I_L + I_R$$

Inducer
- *Contraction*
- *Incidence*
- *Diffusion from inlet to throat*
- *Choking at the throat*

Impeller
- *Friction*
- *Aerodynamic loading*
- *Mixing processes*
- *Clearance gap flow*
- *Local shock waves on suction surfaces*



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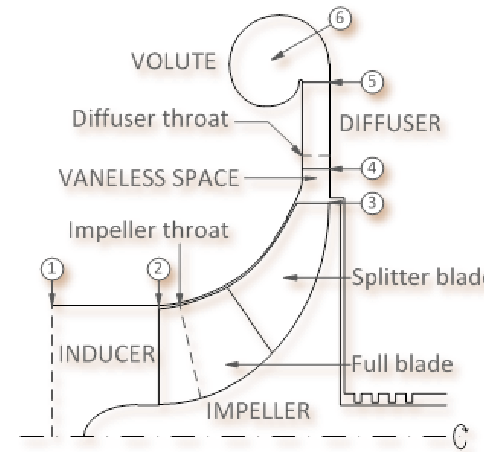
3. VALIDATION

- Validation for S-CO_2: Centrifugal compressor at **SANDIA N.L.**



N (rpm)	T_{01} (K)	P_{01} (bar)	$\dot{m}$ (kg/s)	P_3 (bar)			P_{05} (bar)			T_{05} (K)		
				Exp.	Mod.	Dev. (%)	Exp.	Mod.	Dev. (%)	Exp.	Mod.	Dev. (%)
10000	305.5	76.76	0.454	76.76	77.40	0.83	79.79	77.74	1.28	-	-	-
20000	305.5	76.76	0.771	78.54	79.49	1.21	80.69	81.11	0.53	-	-	-
28000	305.5	76.76	1.134	82.11	82.06	-0.06	85.33	85.19	-0.16	-	-	-
39000	305.6	77.11	1.451	85.68	87.88	2.54	92.82	94.55	1.86	-	-	-
49000	306.3	78.54	1.816	94.25	95.46	1.28	106.39	106.18	-0.20	-	-	-
55000	306.4	78.90	2.043	99.96	100.53	0.57	113.53	114.41	0.78	-	-	-
56000	306.6	78.26	2.088	101.04	101.54	0.49	114.96	115.85	0.77	-	-	-
60000	306.9	79.97	2.225	102.11	105.69	3.51	121.39	122.37	0.81	-	-	-
64900	307.9	82.11	2.406	108.53	111.98	3.18	129.24	131.59	1.82	-	-	-
64384	308.71	82.86	2.860	106.7	108.94	2.10	119.4	108.94	4.30	323.82	324.17	0.11
29888	306.78	79.20	1.315	82.64	84.86	2.69	85.68	88.01	2.72	310.094	310.056	-0.01
59584	308.33	82.24	2.609	102.60	104.89	2.23	112.28	118.5	5.54	321.64	321.884	0.08

S.A. Wrigth et al., Operation and Analysis of a Supercritical CO2 Brayton Cycle, Report SAND2010-0171, 2010.

R.B. Vilim, A One-Dimensional Compressor Model for Super-Critical Carbon Dioxide Applications, Proceedings of ICAPP'10, Paper 10156.

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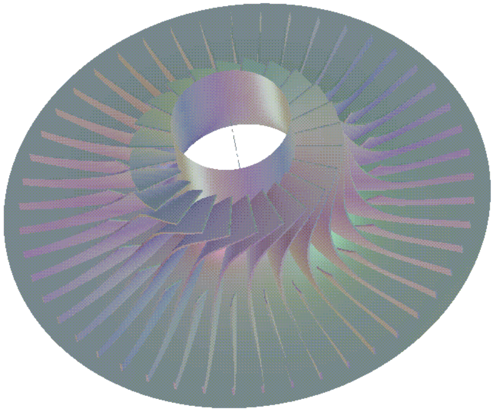
4. CASE STUDY

- **10 MW S-CO2** power cycle for a CSP application
  - ✓ Compressor inlet: 85 bar & ~ 40 ºC
  - ✓ Mass flow rate: 71.55 kg/s
  - ✓ Pressure ratio: 3:1

Common design inputs

- **Design inputs:** M'_{2S}, Z_{FB}, β_3
- **Optimization** of η_{tt} in the ranges Z_{FB}=[18,30], β_3=[30,45] and M'_{2S}=[0.6,0.75]
- **Design results:** Z_{FB}=24, β_3=37.8º and M'_{2S}=0.676



Inlet total temperature: 40 ºC
Inlet total pressure: 85 bar
Static pressure at impeller outlet: 187.27 bar
Number of full/splitter blades: 24/24
Splitter-full blade length ratio: 0.5
Blade thickness: constant, 1 mm
Hub/Shroud inlet radius: 25/44.6 mm
Impeller exit radius: 94.8 mm
Impeller exit blade height: 5.1 mm
Hub/Shroud blade angle at inlet: 49.9/64.71º
Blade angle at impeller exit: 37.79º

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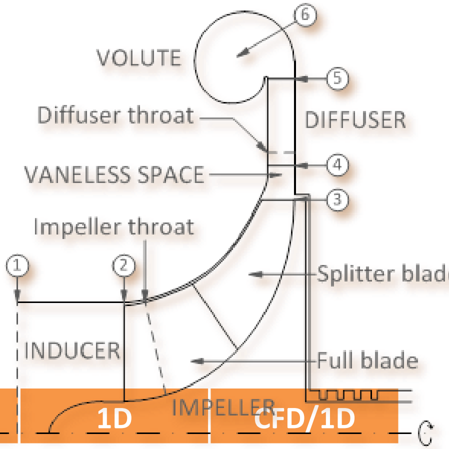
- **Design results:** Z_{FB}=24, β_3=37.8º and M'_{2s}=0.676
- **Computation model:**
  - ✓ Single Reference Frame (SRF) → Moving walls.
  - ✓ Boundary conditions: Mass flow inlet, static pressure outlet and shaft speed
  - ✓ Highly turbulent flow (Re ~ $2\cdot10^7$) → boundary layer refinement
  - ✓ Given the limitation in the computational capacity, the realizable k-ε model is used → No refinement in the near wall region

4. CASE STUDY

- **Results (impeller)**



Parameter	1D	CFD/1D	1D	CFD/1D	1D	CFD/1D	1D	CFD/1D
m (kg/s)	43.6	B.C.	46.5	B.C.	35.8	B.C.	44.0	B.C.
N (rpm)	25087	B.C.	25087	B.C.	12544	B.C.	15052	B.C.
P_{01}/P_{04} (bar)	85.0/202.4	0.91/1.06	85.0/223.9	1.08/1.05	85.0/123.0	0.96/1.02	85.0/145.8	0.97/1.03
T_{01}/T_{04} (K)	333.1/408.9	B.C./1.04	328.1/408.4	B.C./1.04	313.1/335.8	B.C./0.73	313.1/345.9	B.C./1.02
P_1/P_4 (bar)	82.4/138.6	0.90/B.C	82.3/151.0	0.93/B.C.	84.0/104.0	0.96/B.C.	83.5/115.9	0.96/B.C.
T_1/T_4 (K)	330.7/376.6	0.98/1.03	325.7/376.1	1.00/1.02	312.5/325.5	1.00/1.01	312.1/332.1	1.00/1.01
v_1/v_4 (m/s)	48.9/197.9	1.18/1.06	48.7/199.3	1.13/1.05	23.8/96.9	1.22/1.23	29.3/116.7	1.15/1.12
ρ_1/ρ_4 (kg/m^3)	207.8/287.8	0.85/0.94	222.7/326.0	0.89/0.92	351.4/392.6	0.85/0.87	350.0/415.0	0.87/0.89

4. CASE STUDY

- **Results (impeller)**

  ✓ Differences between 5-10 % in P&T

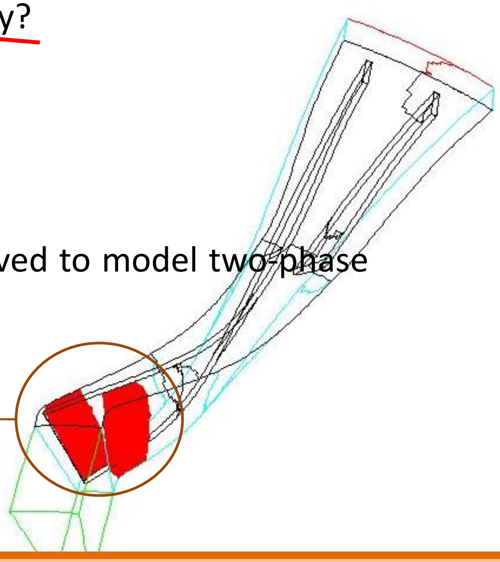
  - Underestimation at inlet & overestimation at outlet

  ✓ Higher discrepancies for velocities (inaccurate blockage factor?)

  ✓ Inlet pressure overpredicted by CFD suggests that losses are underestimated, which can be explained by:

  - ↓ Computation capacity → ~~Refined grids + UDRGM simultaneously?~~
  - Realizable *k-ε* instead of *k-ω SST* → inaccurate friction loses
  - **No gap in the impeller model but…**
  - …from 1D: friction + tip losses = 85-90 % of overall impeller losses
  - Condensation likely to take place → UDRGM needs to be improved to model two-phase flow.

Results for high speed (~86 %) confirm the existence of a saturated wet CO_2 vapour region at the throat of the impeller.



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5. FURTHER DISCUSSION

- **A very recent contribution by MIT: Nikola Baltadjiev MSc Thesis, 2012**
  - ✓ Influence of real-gas effects on compressor performance (mean streamline code)
  - ✓ Gap: analysis of condensation under non-equilibrium conditions

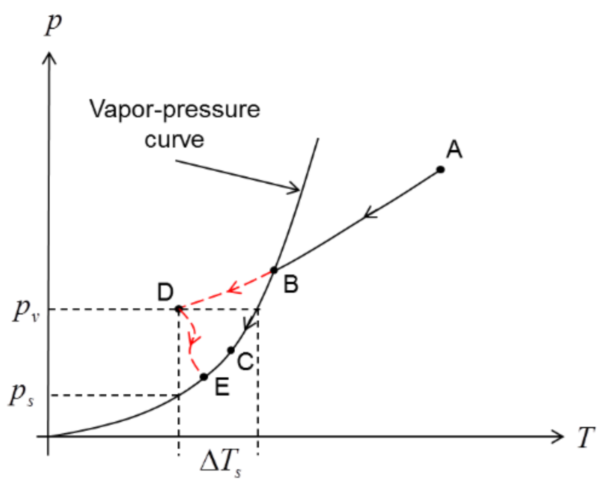
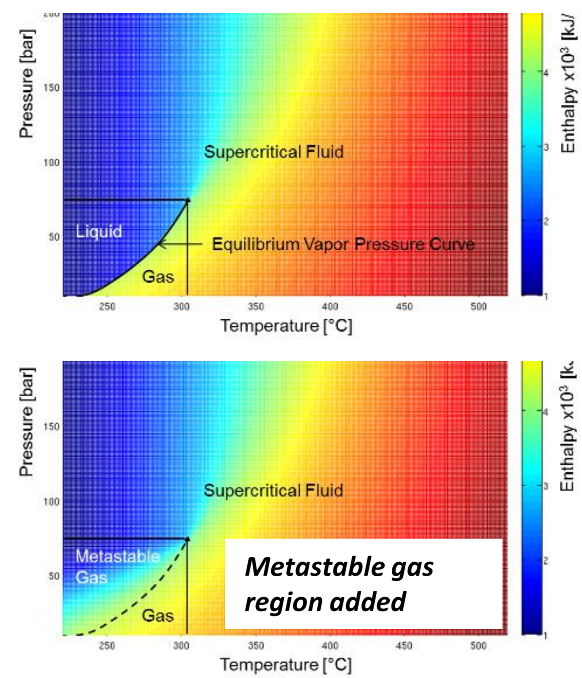


Figure 5-1: Equilibrium (ABC) and non-equilibrium (ADE) flow expansion through the vapor-pressure curve.



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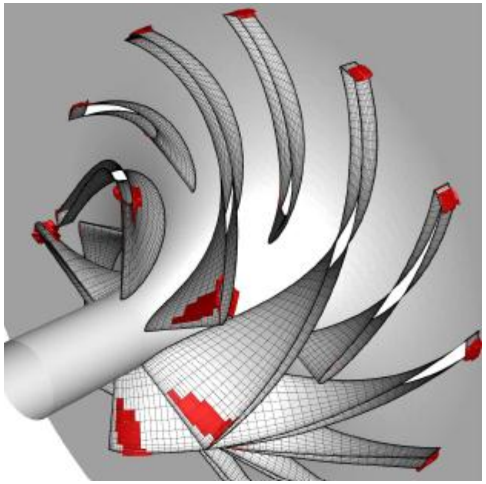
- **The work confirms the observations by Pecnik et al. (TU Delft): saturation conditions are met around the leading edge of the impeller**



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- **The work confirms the observations by Pecnik et al. (TU Delft): saturation conditions are met around the leading edge of the impeller**
- **Proposal: a parameter is suggested to evaluate whether non-equilibrium condensation will take place:**

$$If\ \frac{t_n}{t_r} = \frac{time\ for\ nucleation\ to\ take\ place}{residence\ time\ in\ saturation\ region} < 1 \rightarrow Condensation\ occurs$$

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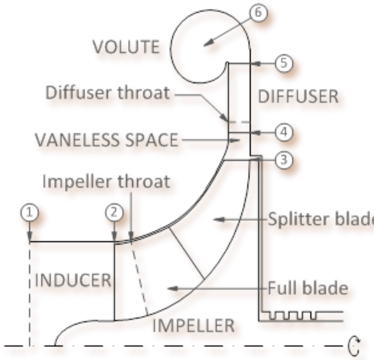
- **It is not conclusive with regard to whether or not condensation will actually take place in a S-CO_2 compressor with close to critical inlet conditions → most likely it will (?)**

5. FURTHER DISCUSSION



- **With respect to the gap**

Parameter	1D	No Gap + No BL	Gap + No BL	No Gap + BL	Gap + BL
m (kg/s)	73.04	B.C.	B.C.	B.C.	B.C.
N (rpm)	15184	B.C.	B.C.	B.C.	B.C.
P_{01}/P_{04} (bar)	75.00/141.16	74.70/141.41	75.09/142.68	74.25/140.93	74.94/143.00
T_{01}/T_{04} (K)	313.15/361.40	B.C./362.47	B.C./362.53	B.C./362.95	B.C./362.83
P_1/P_4 (bar)	73.07/100.89	71.17/B.C.	71.61/B.C.	70.66/B.C.	71.44/B.C.
T_1/T_4 (K)	308.79/335.75	309.6/336.5	309.7/336.0	309.5/337.1	309.7/336.1
v_1/v_4 (m/s)	61.28/160.08	55.9/161.5	55.22/163.1	56.88/161.4	55.53/163.8
ρ_1/ρ_4 (kg/m³)	221.6/281.4	221.4/276.5	224.5/278.9	218.0/274.0	223.3/278.8
Slip factor, σ	0.9075	0.8414	0.8491	0.8486	0.8459
Total pressure loss coefficient, $\Sigma\varpi$	0.3982	0.7282	0.7078	0.6932	0.5976

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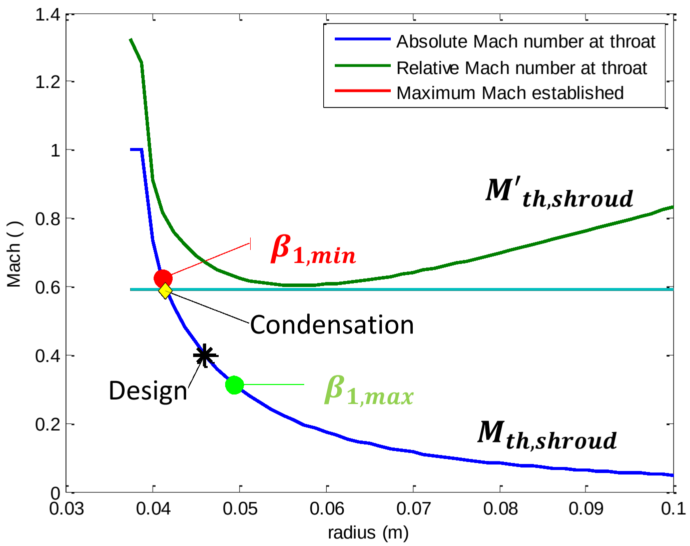
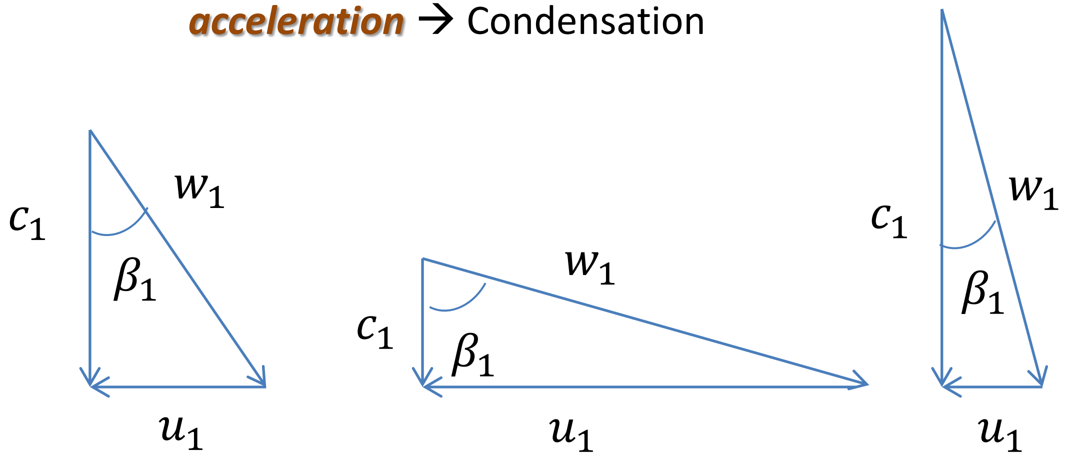
6. APPLICATIONS OF NEW GUIDELINES

- **Novel contributions to the design methodology**

  ✓ Upper-bounded absolute/relative Mach number M_1/M'_1 → ***Avoiding condensation***

  ✓ Upper bound of pressure ratio → ***Avoiding acceleration*** → Condensation





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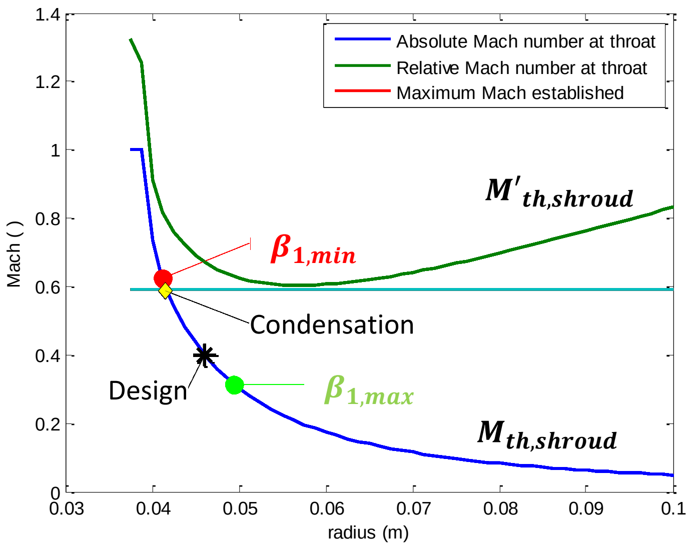
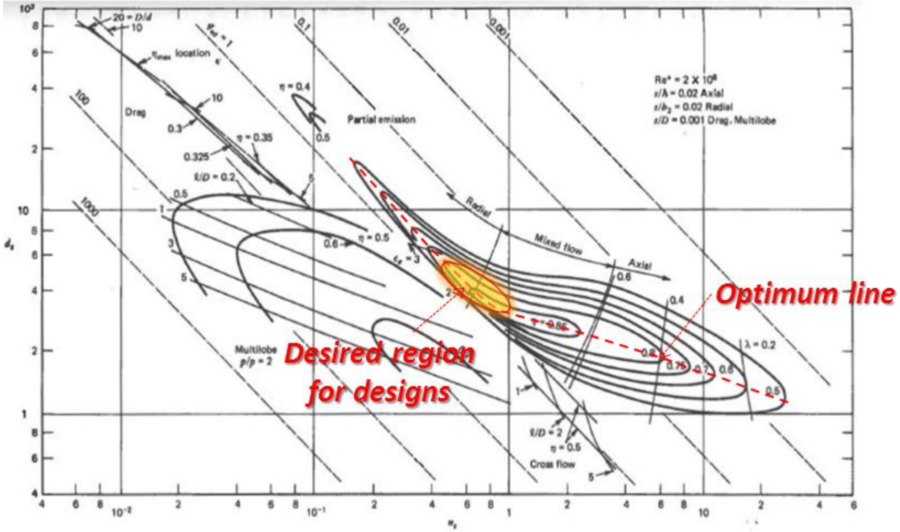




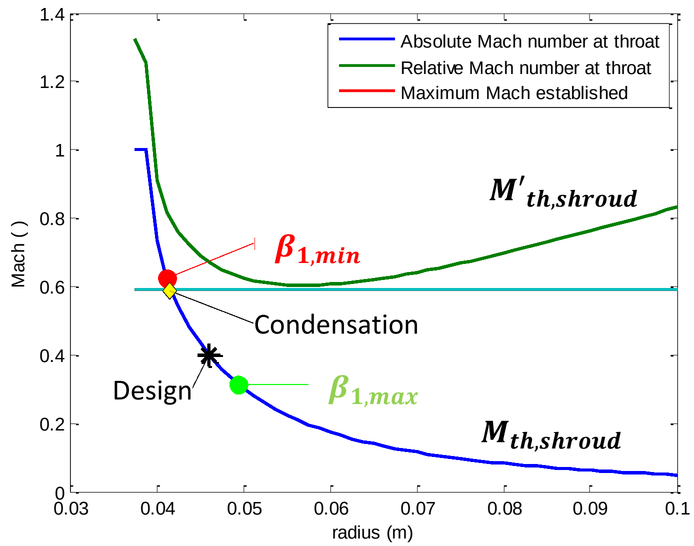
$$N_s = \frac{N\sqrt{Q}}{\Delta H^{3/4}}$$

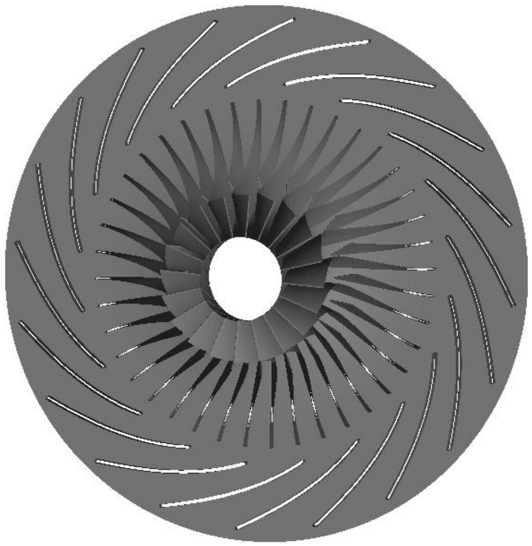
$$D_s = \frac{D \cdot \Delta H^{1/4}}{\sqrt{Q}}$$

6. APPLICATIONS OF NEW GUIDELINES

- **Novel contributions to the design methodology**

  - ✓ Upper-bounded absolute/relative Mach number M_1/M'_1 → ***Avoiding condensation***
  - ✓ Upper bound of pressure ratio → ***Avoiding acceleration*** → Condensation
  - ✓ Need to increase the ***number of blades due*** to high aerodynamic loading
  - ✓ Need to add ***splitter blades*** due to high aerodynamic loading





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7. SUMMARY AND CONCLUSIONS

- The analysis of air vs. S-CO2 conical diffusers shows substantial differences between both fluids

- A 1D tool for centrifugal compressor design is implemented and used to produce a reference S-CO2 compressor → validation to the extent possible

- 1D vs 3D differences are large enough to suggest that standard design rules might not be applicable if efficiencies comparable to contemporary air compressors are sought.

- Some complements to ANSYS Fluent® are necessary in order to properly simulate two phase flow (likely to happen in supercritical CO2 turbomachinery) when pushing the supercritical features to the limit.

1D AND 3D TOOLS TO DESIGN SUPERCRITICAL CO_2 RADIAL COMPRESSORS: A COMPARISON

Thank you – Q&A

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