EXPLORING THE DESIGN SPACE OF THE SCO2 POWER CYCLE COMPRESSOR

B. Monge, D. Sánchez, T. Sánchez

Dep. Energy Engineering

University of Seville (Spain)

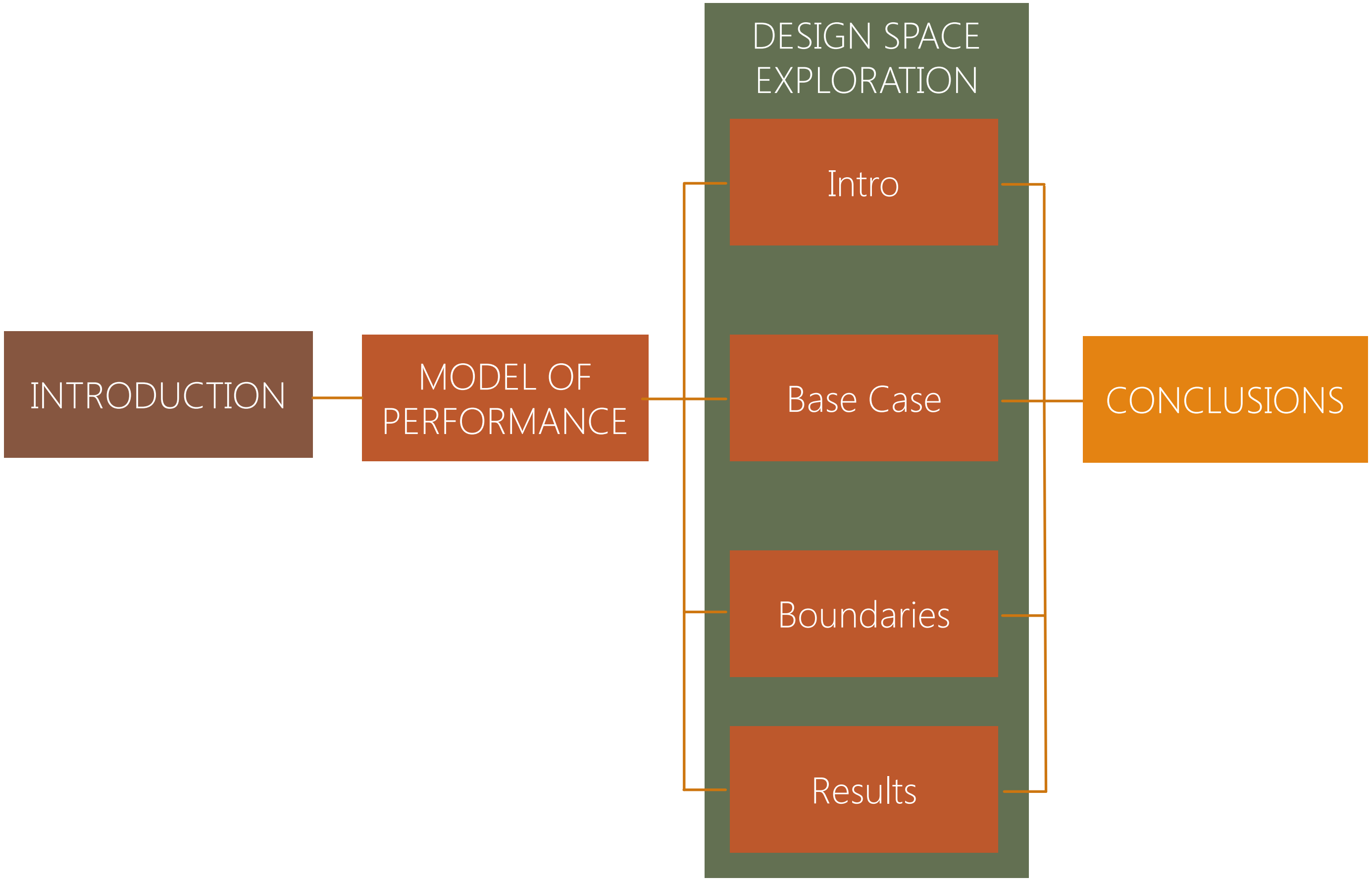
M. Savill, P. Pilidis

Dep. Power and Propulsion

Cranfield University (UK)

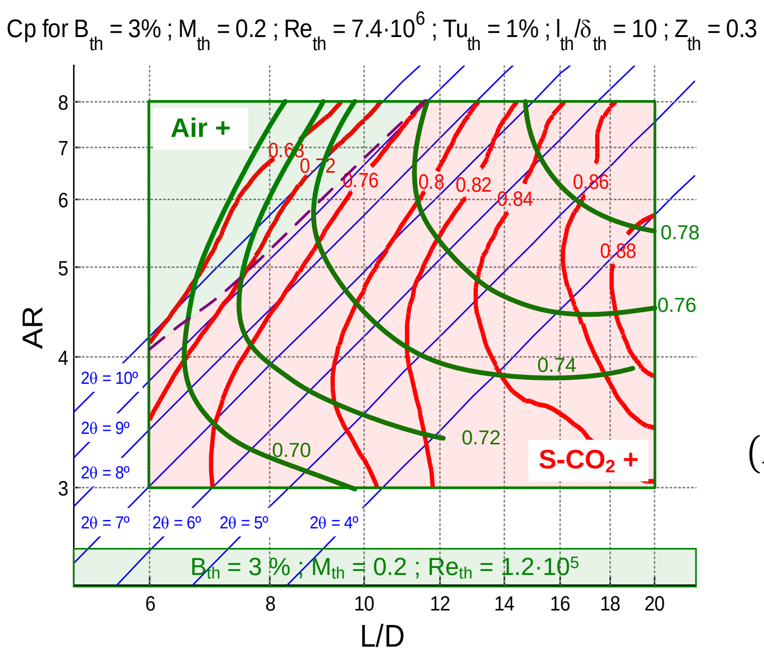
0. CONTENTS

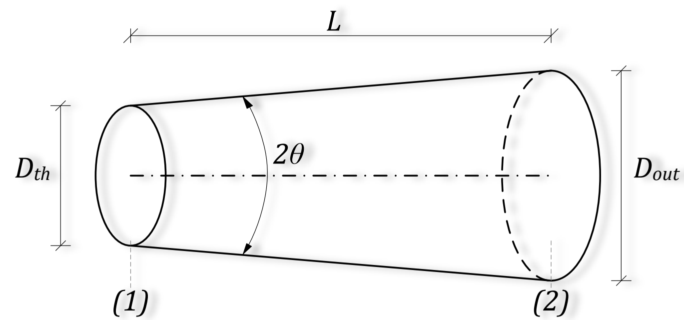
- FLOW DIAGRAM OF PRESENTATION



1. INTRODUCTION

- 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 sCO2 diffusers during the last five years





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

1. INTRODUCTION

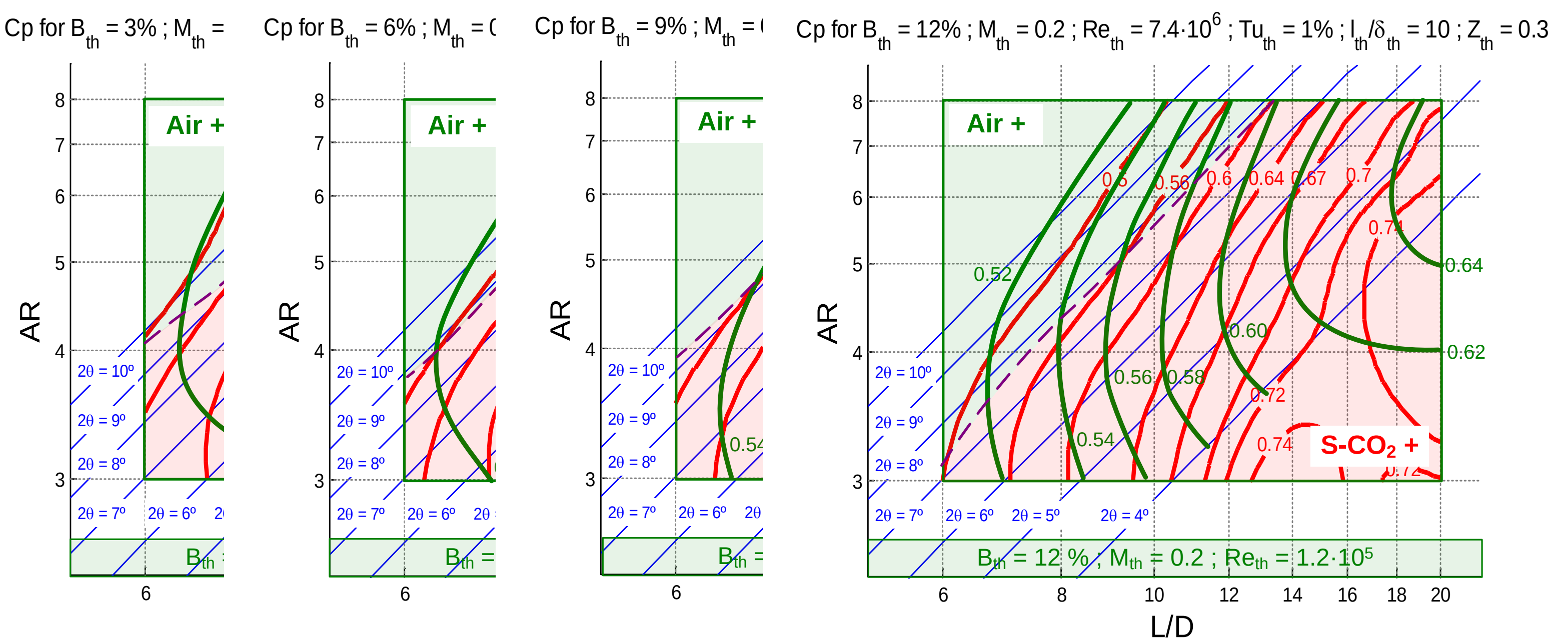
- REVIEW OF PAST WORK
  - The Thermal Power Group at the University of Seville and the Department of Power & Propulsion at Cranfield University have researched the performance of sCO_2 diffusers during the last five years
  - Objectives:
    - Main:

      To understand the differential features of sCO_2 & air in typical operating conditions of the corresponding Brayton cycles → influence on design/performance
    - Secondary:

      To develop the modeling tools and to gain confidence in their application to design processes

1. INTRODUCTION

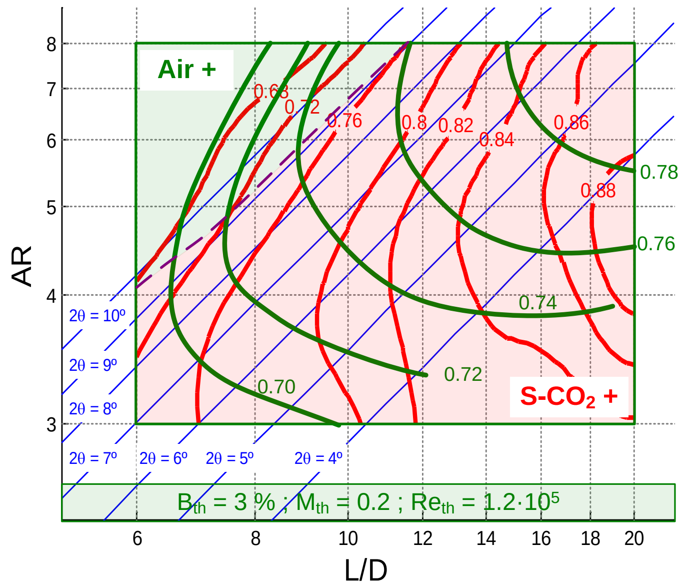
- REVIEW OF PAST WORK



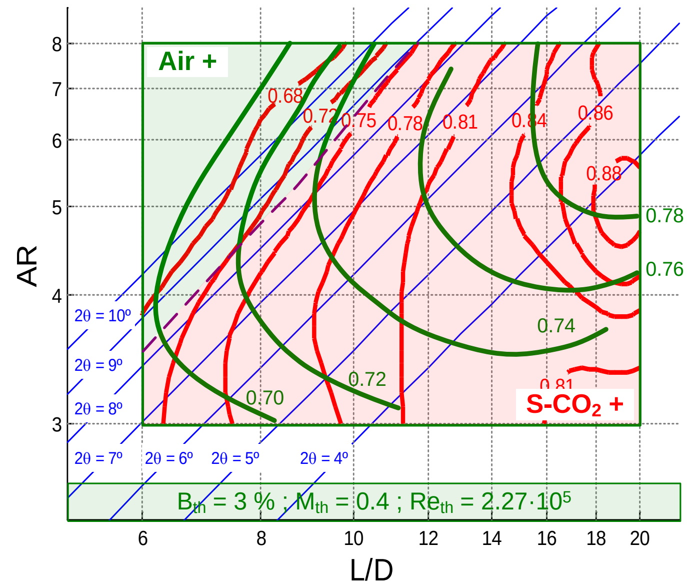
1. INTRODUCTION

❑ REVIEW OF PAST WORK

Cp for B_{th} = 3% ; M_{th} = 0.2 ; Re_{th} = $7.4 \cdot 10^6$; Tu_{th} = 1% ; l_{th}/δ_{th} = 10 ; Z_{th} = 0.3



Cp for B_{th} = 3% ; M_{th} = 0.4 ; Re_{th} = $7.4 \cdot 10^6$; Tu_{th} = 1% ; l_{th}/δ_{th} = 10 ; Z_{th} = 0.3



1. INTRODUCTION

- 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 sCO2 diffusers during the last five years
  - Results available in the public domain
    - B. Monje et al., Comparing the pressure rise of air and supercritical carbon dioxide in conical diffusers, Paper GT2012-69895, Turbo Expo 2012, Copenhagen
    - A. López et al., Effect of turbulence intenstity 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 fo conical diffusers operating with air and supercritical carbon dioxide, International Journal of Heat and Fluid Flow 44 (2013) 542-553

1. INTRODUCTION

- 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 sCO2 diffusers during the last five years
  - Second step: development of a 1D & 3D parallel analysis

1. INTRODUCTION

- REVIEW OF PAST WORK (1D & 3D parallel analysis)

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

Impeller + Diffuser
ANSYS Fluent® → 3D

Draft design VS. *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

1. MODEL OF PERFORMANCE (1D)

- 1D MODEL: 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

$$h_{0,i} = h_{0,i-1} \quad \text{(Stationary)}$$

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

Deviation

Impeller slip factor, σ (Wiesner)

Diffuser: deviation angle, δ (Howell)

Pressure losses

(Stationary elements)

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

(Impeller)

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

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

Work input coefficients (secondary)

Disk friction

Leakage

Recirculation

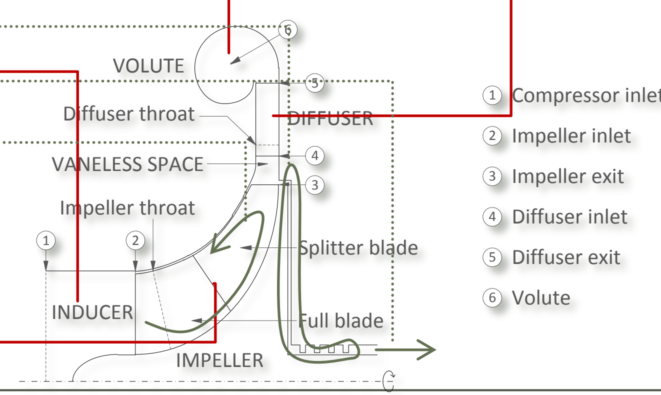
$$\frac{\Delta H}{u_2^2} = I_B + I_{DF} + I_L + I_R$$

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

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

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

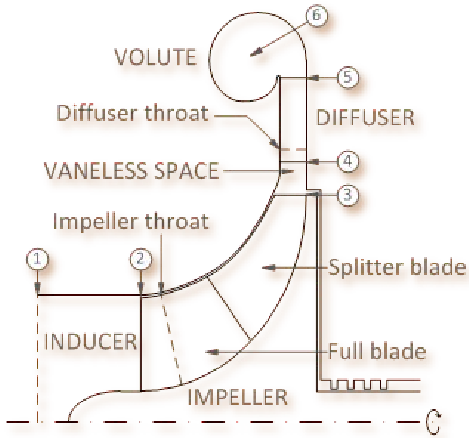
Impeller
- *Friction*
- *Aerodynamic loading*
- *Mixing processes*
- *Clearance gap flow*
- *Local shock waves on suction surfaces*



1. MODEL OF PERFORMANCE (1D)

- 1D MODEL – Validation against data from SANDIA

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	310.09	310.06	-0.01
59584	308.33	82.24	2.609	102.60	104.89	2.23	112.28	118.5	5.54	321.64	321.88	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.

2. DESIGN SPACE EXPLORATION - INTRO

- STEPWISE APPROACH TO COMPRESSOR DESIGN
  - Accuracy of 1D model already confirmed
  - Parallel 1D-3D simulations confirm existence of local flow features to be taken into account
  - Next step: assessing the influence of main design parameters → DESIGN SPACE EXPLORATION
  - Exploring the Design Space allows for comprehensive design decisions

- OBJECTIVE
  - Comprehensively evaluate trade-offs between opposing design decisions

2. DESIGN SPACE EXPLORATION - INTRO

PARAMETERS STUDIED - STANDARD

- Flow coefficient
- Load coefficient
- Degree of reaction

$$\phi = \frac{v_{x1}}{u_2}$$

$$\psi = \frac{\Delta h_{0s}}{u_2^2}$$

$$R = \frac{h_2 - h_1}{\Delta h_0}$$

PARAMETERS STUDIED – SPECIFIC TO sCO2

- Compressibility factor
- Acceleration Margin to Condensation (AMC)

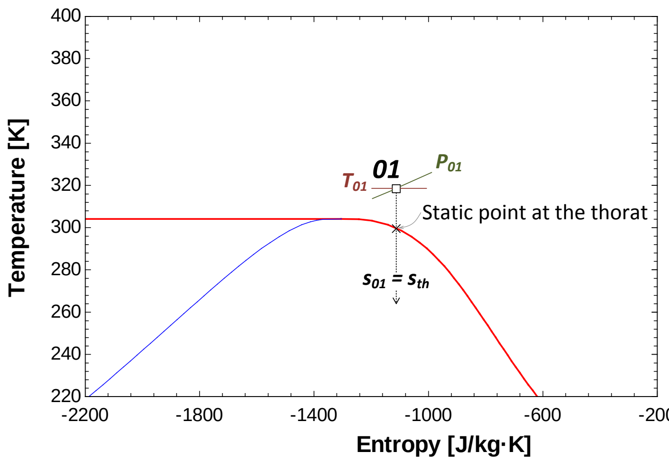
$$Z_{01} = \frac{p_{01} \cdot v_{01}}{R \cdot T_{01}}$$

$$h_{01} = h_{th} + \frac{v_{th}^2}{2}$$

$$If\ x_{th} = 1 \rightarrow AMC = M_{th} = \frac{v_{th}}{a_{th}}$$

Supersaturation

$$\dot{P} = -\frac{1}{P}\frac{dP}{d\tau} = -\frac{c_{ax}}{P}\frac{dP}{da}$$

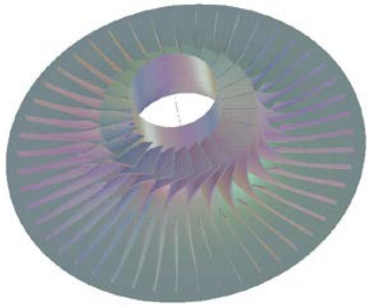


3. DESIGN SPACE EXPLORATION – BASE CASE

- REFERENCE CSP APPLICATION – Recuperative (non-condensing) Brayton cycle
  - Output @ generator terminals: 10 MWe
  - Compressor inlet: 40ºC & 75 bar
  - Mass flow rate: 73 kg/s
  - Total pressure ratio: 3.33

- MODELING ASSUMPTIONS
  - Isentropic efficiency of compressor/turbine (guess): 80 / 90%
  - Recuperator effectiveness: 95%
  - Pressure drop in the hot/cold side of the recuperator: 1.5 / 0.5%
  - Pressure drop in the heater: 2.0%
  - Pressure drop in the cooler: 1.0%

Inlet total temperature:	40ºC
Inlet total pressure:	85 bar
Static pressure at impeller outlet:	187.3 bar
Number of full/splitter blades:	24/24
Splitter-full blade length ratio:	0.5
Blade thickness:	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:	49.9/64.7º
Blade angle at impeller exit (backswept):	37.8º



3. DESIGN SPACE EXPLORATION – BASE CASE

COMPRESSOR DESIGN ASSUMPTIONS

- Isentropic inlet flow with no-swirl: purely axial flow with no blockage
- Incidence angle at impeller inlet: 0°
- Inlet hub radius (guess): 25 mm (double-checked for mechanical integrity)
- Constant meridional velocity
- Splitter to full blade length ratio: 0.5
- Number of splitter/full blades based on loading coefficient

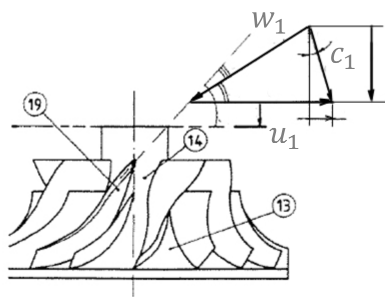
$$\frac{2\Delta w}{w_1 + w_2} \leq 0.9 \qquad \Delta w = \frac{2\pi D_2 u_2 v_{u2}}{z_{FB} L_{FB} + z_{SB} L_{SB}}$$

4. DESIGN SPACE EXPLORATION – BOUNDARIES

BOUNDARIES OF THE DESIGN SPACE

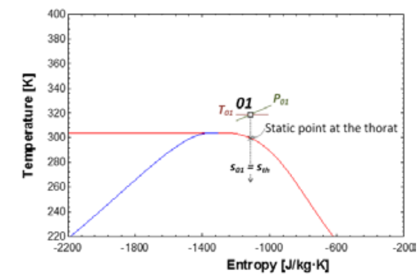
- SUPERSONIC FLOW:

$$M'_1 = \frac{w_1}{a_1} < 1$$



- OPERATION IN THE TWO-PHASE REGION:

$$M_{th} < AMC$$



- (PSEUDO)SURGE AND SECONDARY INEFFICIENCIES:

$$B = \varpi_{fr} \frac{P_{01} - P_1}{P_{02} - P_2} \sqrt{\frac{s_1\, \bar{d}_H}{s_2\, b_2}} + \left[0.3 + \frac{b_2^2}{L_B^2}\right] \frac{A_R^2\, \rho_2\, b_2}{\rho_1\, L_B} + \frac{\delta_{CL}}{2\, b_2} < 30\%$$

Aspect ratio: b_2/L_B
Clearance: δ_{cl}
Area ratio: A_R
Channel width: s_2

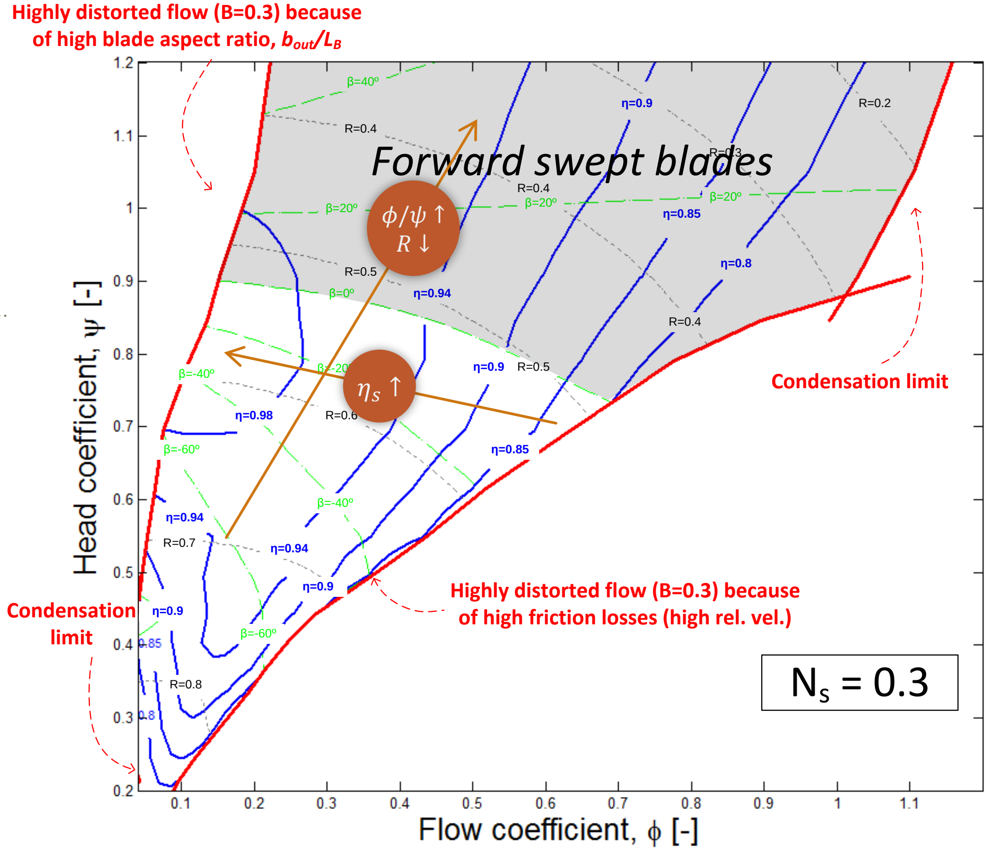
- SHAFT INTEGRITY

$$d_{axis,min} = \sqrt{\frac{16\dot{W}_m}{\omega\pi\tau_m}}$$

Shaft power: $\dot{W}_m$
RPM: ω
Adm. shear stress: τ_m

5. DESIGN SPACE EXPLORATION – RESULTS

- GENERAL FEATURES AND OBSERVATIONS
  - Specific speed Ns=0.3:
  - General pattern as expected
    - Forward-swept blades rise ϕ and ψ simultaneously…
    - …drastic drop in efficiency for increasing β_2
    - Highest efficiency for $R\sim0.6$ → impeller-loaded



$$\phi = \frac{v_{x1}}{u_2} \qquad \psi = \frac{\Delta h_{0s}}{u_2^2} \qquad R = \frac{h_2 - h_1}{\Delta h_0} \qquad N_s = \frac{\omega\sqrt{Q}}{\Delta h_s^{3/4}}$$

5. DESIGN SPACE EXPLORATION – RESULTS

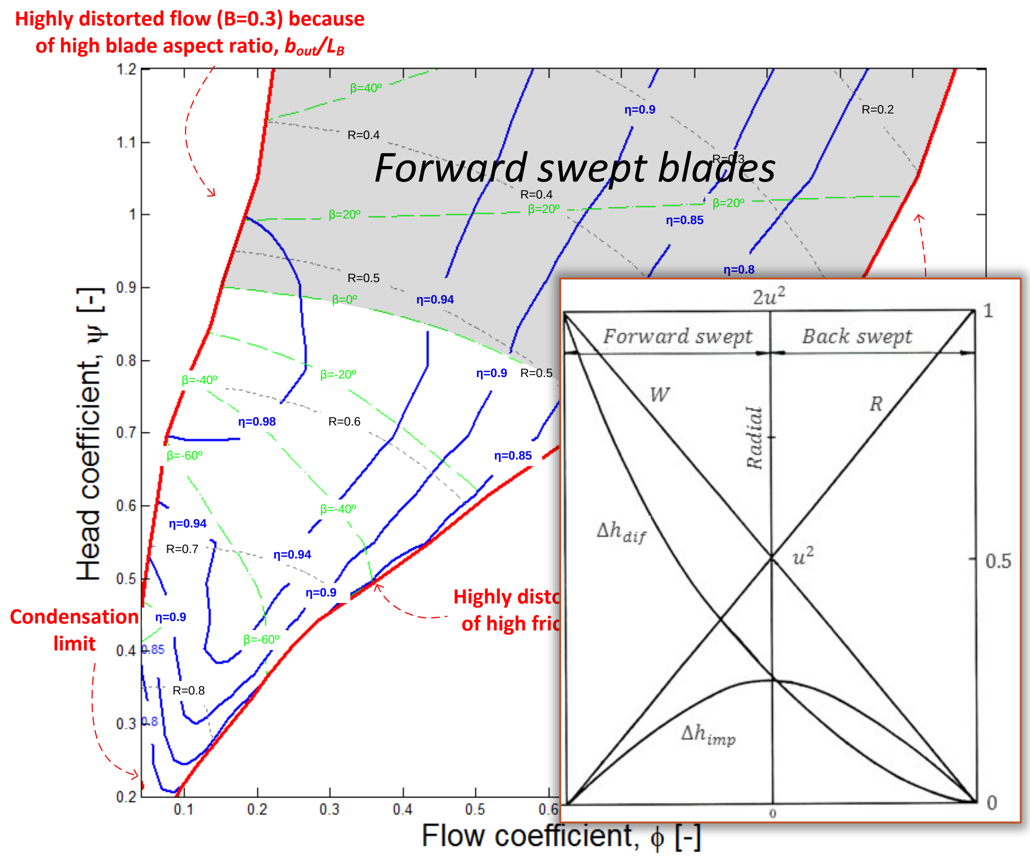
- GENERAL FEATURES AND OBSERVATIONS
  - Effect of β_2 on the main design parameters

$$W = u_2(u_2 + c_{r2}\tan\beta_2)$$

$$\Delta h_{dif} = \frac{1}{2}(u_2 + c_{r2}\tan\beta_2)^2$$

$$\Delta h_{imp} = \frac{1}{2}(u_2^2 - c_{r2}^2\tan^2\beta_2)^2$$

$$R = \frac{1}{2}\left(1 - \frac{c_{r2}}{u_2}\tan\beta_2\right)$$



$$\phi = \frac{v_{x1}}{u_2} \qquad \psi = \frac{\Delta h_{0s}}{u_2^2} \qquad R = \frac{h_2 - h_1}{\Delta h_0} \qquad N_s = \frac{\omega\sqrt{Q}}{\Delta h_s^{3/4}}$$

5. DESIGN SPACE EXPLORATION – RESULTS

❑ GENERAL FEATURES AND OBSERVATIONS

- Effect of β_2 on the main design parameters

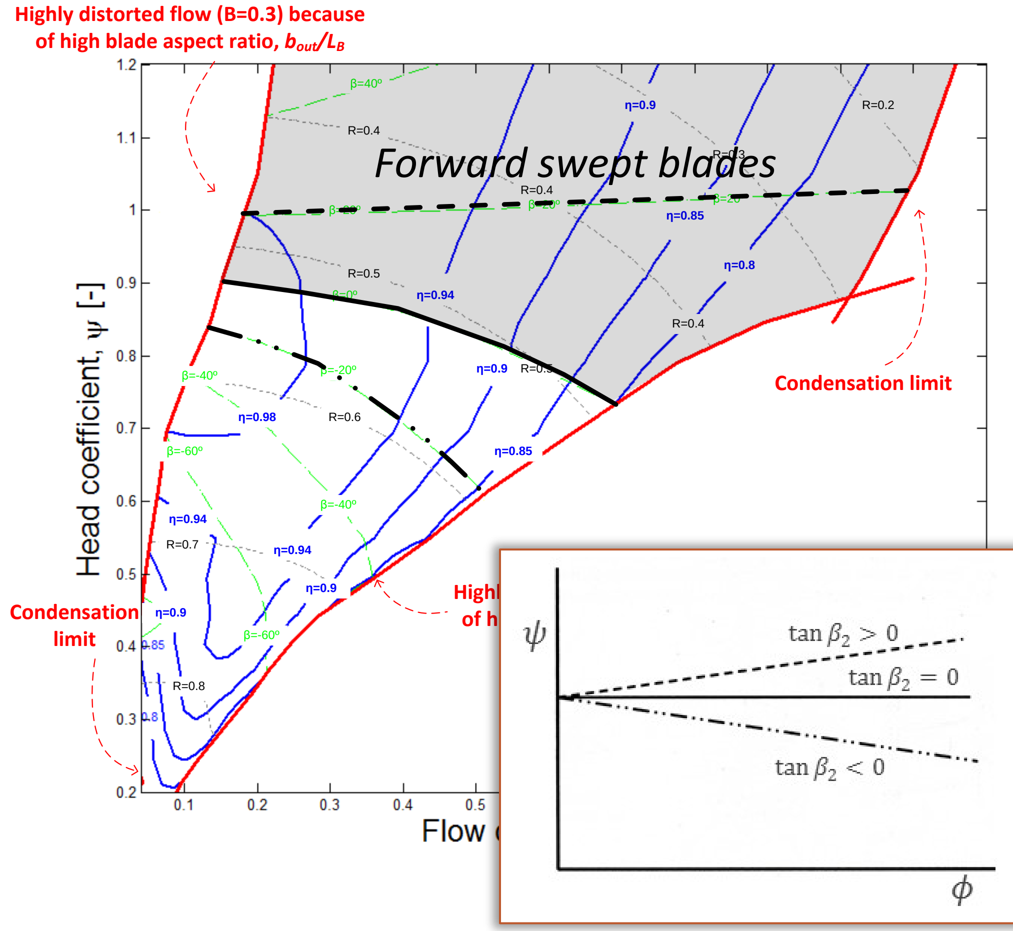
$$W = u_2(u_2 + c_{r2}\tan\beta_2)$$

$$\Delta h_{dif} = \frac{1}{2}(u_2 + c_{r2}\tan\beta_2)^2$$

$$\Delta h_{imp} = \frac{1}{2}(u_2^2 - c_{r2}^2\tan^2\beta_2)^2$$

$$R = \frac{1}{2}\left(1 - \frac{c_{r2}}{u_2}\tan\beta_2\right)$$

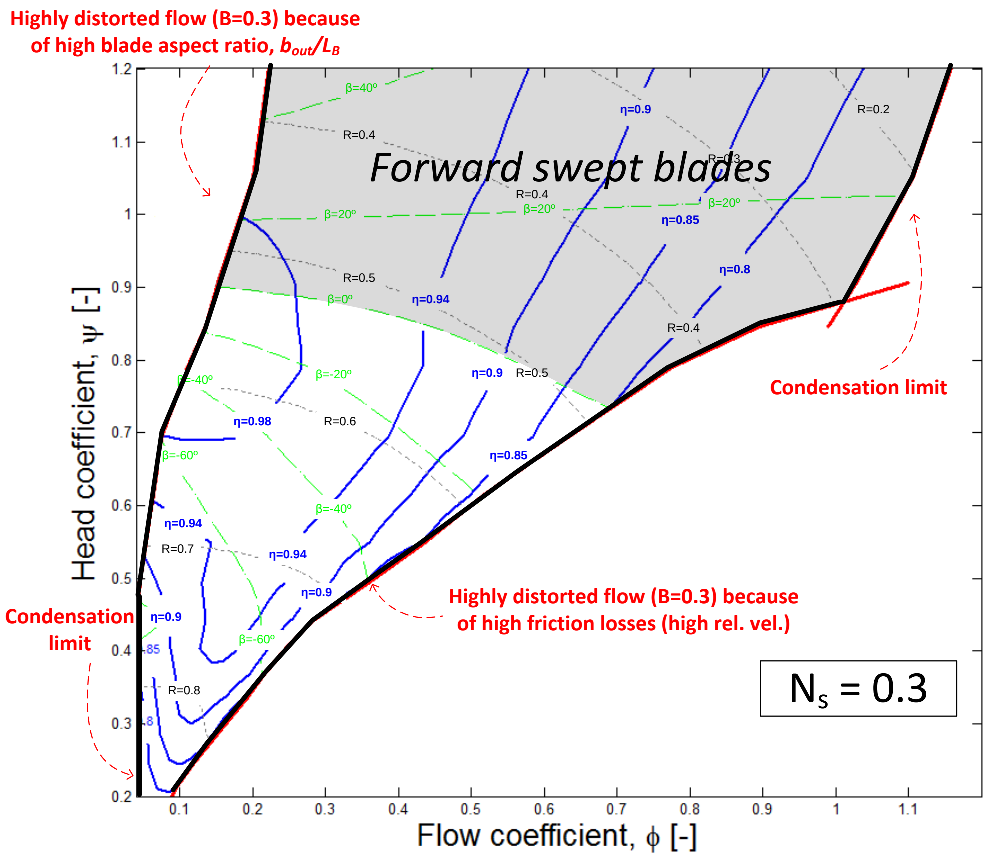
- Similarity with an ideal compressor



$$\phi = \frac{v_{x1}}{u_2} \qquad \psi = \frac{\Delta h_{0s}}{u_2^2} \qquad R = \frac{h_2 - h_1}{\Delta h_0} \qquad N_s = \frac{\omega\sqrt{Q}}{\Delta h_s^{3/4}}$$

5. DESIGN SPACE EXPLORATION – RESULTS

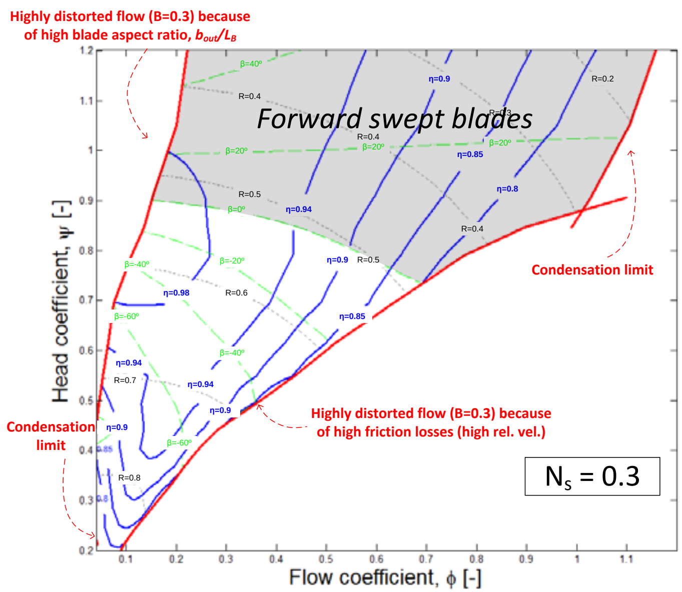
- SPACE BOUNDARIES
  - Low flow and high head: pseudo-surge due to high aspect ratio b_2/L_B
  - Low flow and low head: two-phase region
  - High flow and high head: two-phase region
  - Low to mid flow and low head: low efficiency due to aerodynamic (friction) losses

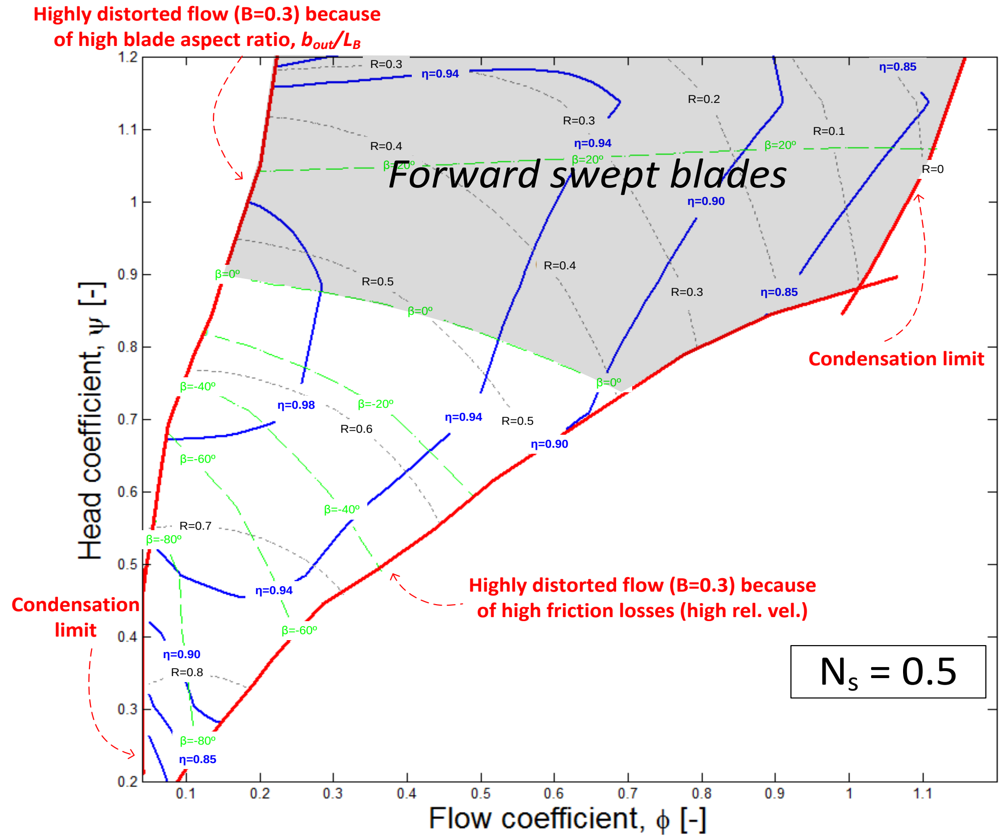


$\phi = \frac{v_{x1}}{u_2}$ $\psi = \frac{\Delta h_{0s}}{u_2^2}$ $R = \frac{h_2 - h_1}{\Delta h_0}$ $N_s = \frac{\omega\sqrt{Q}}{\Delta h_s^{3/4}}$

5. DESIGN SPACE EXPLORATION – RESULTS

- Ns increased to 0.5
  - Same pattern
  - Same location of space boundaries
  - Smoother efficiency decay at higher ϕ

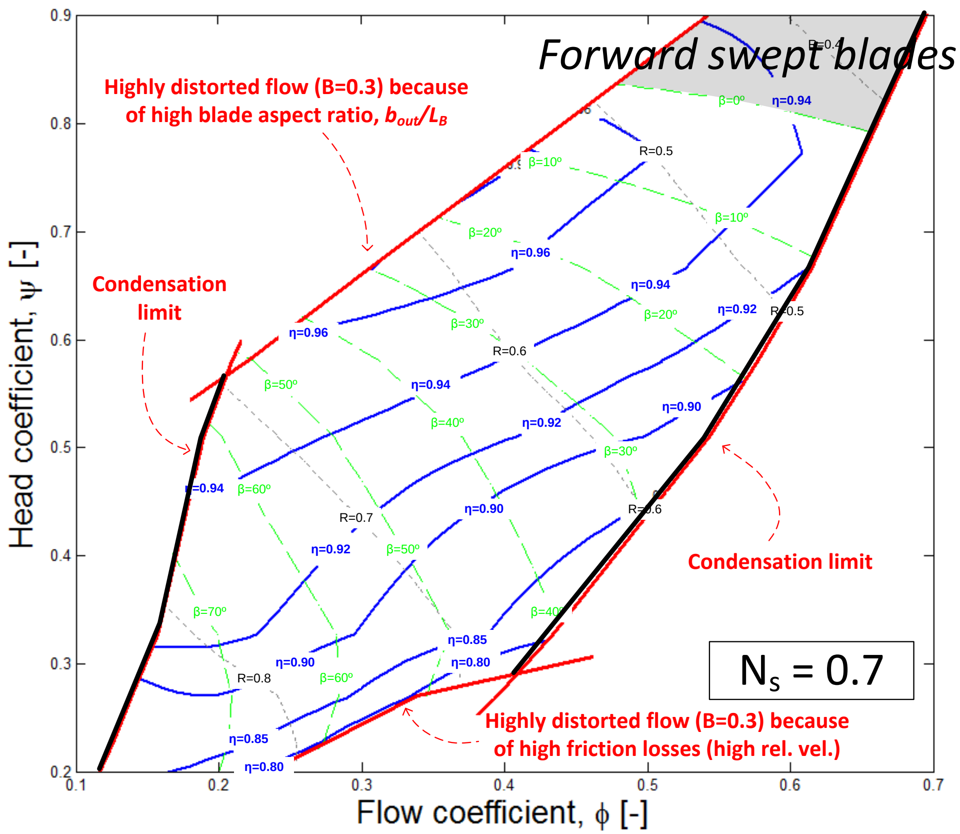




$$\phi = \frac{v_{x1}}{u_2} \qquad \psi = \frac{\Delta h_{0s}}{u_2^2} \qquad R = \frac{h_2 - h_1}{\Delta h_0} \qquad N_s = \frac{\omega\sqrt{Q}}{\Delta h_s^{3/4}}$$

5. DESIGN SPACE EXPLORATION – RESULTS

- Ns increased to 0.7
  - Map largely reduced
  - High and low two-phase flow limits more restrictive → higher flow velocity…
  - …boundaries due to two-phase flow region more restrictive than flow distortion (pseudo-surge) → not the case at low Ns
  - Forward swept blades not an option
  - Peak efficiency at higher ϕ but similar R and β_2



$$\phi = \frac{v_{x1}}{u_2} \qquad \psi = \frac{\Delta h_{0s}}{u_2^2} \qquad R = \frac{h_2 - h_1}{\Delta h_0} \qquad N_s = \frac{\omega\sqrt{Q}}{\Delta h_s^{3/4}}$$

6. CONCLUSIONS

- As expected, general pattern of design space similar to similar turbomachinery for standard applications (for instance, influence of swept angle on efficiency and degree of reaction)
- Vicinity of critical point sets more restrictive boundaries.
  - Low flow coefficient → blade aspect ratio too low
  - High flow coefficient → risk to enter saturation dome (irrelevant in small loops but to be explored in large applications)
- Best performance found for specific speeds somewhat lower than Balje's ($N_s \sim 0.4$)
- Best performance found for the expected flow coefficients ($\phi|_{\eta_{opt}} \sim 0.1$) except for higher specific speed ($N_s \geq 0.6$)

❑ NEXT STEPS

- So far only impeller performance explored → next step to incorporate diffuser
- Breakdown of losses in off-design operation to elucidate best design recommendations

EXPLORING THE DESIGN SPACE OF THE SCO2 POWER CYCLE COMPRESSOR

B. Monge, D. Sánchez, T. Sánchez

Dep. Energy Engineering

University of Seville (Spain)

M. Savill, P. Pilidis

Dep. Power and Propulsion

Cranfield University (UK)