EXPLORING THE DESIGN SPACE OF THE SCO2 POWER CYCLE COMPRESSOR

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Tomás Sánchez graduated from the Polytechnic University of Madrid where he later earned his PhD degree in turbomachinery analysis and optimisation applied to steam turbines. He has served at the Polytechnic University of Madrid, the National Distance-Education University and the University of Seville, where he is currently Head of the Department of Energy Engineering. In all these institutions, Prof. Sánchez has developed research for a wide variety of companies and R&D programmes, both national and international. It is noted that he has recently completed a four year project aimed at developing design guidelines for supercritical carbon dioxide radial compressors whose main results are now available in the public domain.

ABSTRACT

In the last five years, the Universities of Seville (Spain) and Cranfield (United Kingdom) have developed joint research in the field of turbomachinery for supercritical carbon dioxide power cycles. In particular, the research interest of the former group has focused on the design of radial compressors to be integrated in low and mid-scale systems, below 100 MWe, whereas Cranfield University has concentrated on oxycombustion applications to enable capture and storage of pipeline-ready CO_2. As a consequence of this work, a number of tools to design and analyse these power cycles at system and component levels have been developed. Amongst these, those specifically addressing turbomachinery design are worth noting.

In particular, an algorithm to combine 1D (meanline) and 3D (CFD) codes has been developed at the University of Seville to explore the potential of simple tools to produce a tentative design from which an optimisation process can be launched. Such initial estimate of the compressor design, if accurate, can save up substantial time and thus allow for accelerated design processes.

One of the activities within this process of setting up the aforecited optimisation tool is exploring the design space of radial compressors for supercritical carbon dioxide applications. In other words, when approaching a new design of such machine, it is crucial to know the range in which the principal design parameters vary and the impact of such variations on compressor performance. This task is commonly known as exploration of the design space and is presented in this work for a particular application (though the conclusions can be regarded as qualitatively general).

This design space exploration is done with a 1D design tool developed by the authors and implemented in Matlab®. The code is based on conservation laws and empirical correlations for pressure losses which are adapted to the fluid of interest. Once validated, it is applied to perform sensitivity analyses for each design parameter and assuming certain constraints to avoid the following limiting operating conditions: (i) supersonic flow, (ii) two-phase flow (or, more precisely, operation in saturated conditions), (iii) surge, (iv) secondary inefficiencies and (v) technological barriers (i.e. mechanical limitations).

The paper begins with a similarity analysis from which the main parameters used later are extracted. Special emphasis is put on the identification and characterisation of saturation conditions. Then a case-study is defined based on a sCO2 power cycle application, whose compressor is the object of the investigation, under the constraints specified. Finally, the main results from this analysis are presented in the form of design maps which can potentially be used for optimisation purposes.

PRELIMINARY CONCEPTS

Similarity rules traditionally applied to turbomachinery suggest that the performances (basically isentropic efficiency, η_s, and pressure ratio, PR) of any piece of turbomachinery are mainly functions of the specific speed (N_S) and diameter (D_s), as well as the Mach and Reynolds numbers [Balje]. As widely known, the effect of the latter parameter can be disregarded above a critical threshold, $Re > Re_{cr}$, typically in the order of $2 \cdot 10^5$ for air machinery. This value is actually well exceeded when supercritical CO_2 is used due to its much higher density (two orders of magnitude higher than air) and in spite of the characteristic compactness of this machinery (note that $Re \propto \rho \cdot D$). With regard to Mach number, it has a large influence on the design and performance of the machine as it affects the compressibility of the inlet flow. Nonetheless, this known influence is now complemented by additional features brought about by the proximity of the inlet stagnation conditions to the saturation curve, as a consequence of which the inlet Mach number has a strong influence on whether or not the flow enters the two-phase flow region (or, more precisely, whether or not the saturation temperature at a given pressure is met). This has been already termed by different authors as the condensation risk of a particular compressor (this effect is, obviously, not present in turbines).

In effect, even though the total inlet temperature is supercritical, subcritical conditions within the saturation dome can still be met if the flow is subjected to an expansion (acceleration) process, see figure 1. Based on this plot, the Acceleration Margin to Condensation (AMC) is defined as the Mach number for which, given the actual total inlet conditions, the static pressure/temperature would lie on the saturation line (i.e., would correspond to dry saturated vapor, $x = 1$). From a practical point of view and assuming that condensation takes place in equilibrium conditions, two-phase flow is avoided as long as the absolute Mach number at the throat (M_{th}) is below this value.

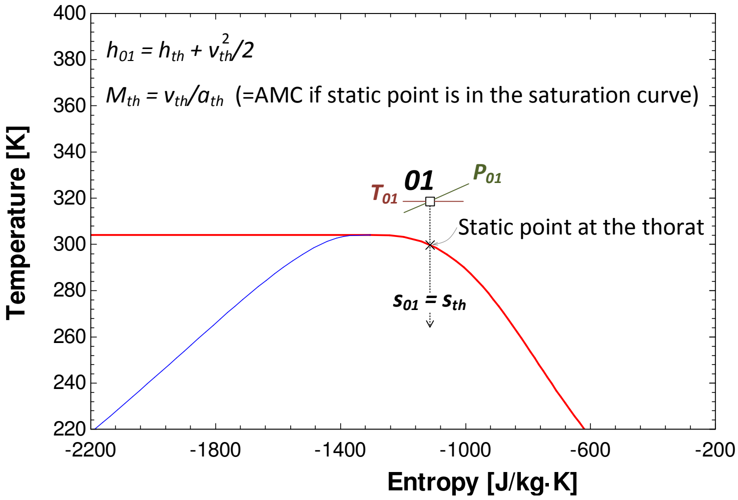

Figure 1. Definition of the Acceleration Margin to Condensation (AMC)

For the sake of formality, and even if it is out of the scope of the paper, it is here noted that a certain degree of supercooling must be expected in expansion flows entering the two-phase region. This supercooling is defined as the temperature difference between the saturation temperature at the condensation pressure and the temperature at which the flow is actually condensing ($\Delta T_s = T_{sat}|_{P_{cd}} - T_{cd}$). The same phenomenon can be referred to pressures rather than temperatures by the so-called supersaturation. Whichever term is used, non-equilibrium condensation is largely influenced by the speed of expansion (s^{-1}):

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

In practice, the loci of real condensation within the saturation dome are typically defined by constant quality lines (Wilson lines) in the two-phase region, whose location depends on $\dot{P}$. For instance, for a condensing steam flow, real condensation takes place at $x = 0.977$ if $\dot{P} = 10\ s^{-1}$ or $x = 0.963$ if $\dot{P} = 10000\ s^{-1}$.

The theoretical analysis of supersaturation in sCO2 turbomachinery is currently being developed by the authors as this effect is expected to be relevant in the upscale process. It has been briefly discussed here just to evidence that the condition $M_{th} < AMC$ is conservative in terms of avoiding two-phase flow.

In addition to the specific parameters already cited, the following non-dimensional parameters universal to turbomachinery analysis are used throughout this work: flow (ϕ) and load (ψ) coefficients and degree of reaction (R) [Dixon & Hall].

$$\phi = v_{x1}/u_2$$
$$\psi = \Delta h_{0s}/u_2^2$$
$$R = \frac{h_2 - h_1}{\Delta h_0}$$

Finally, owing to the vicinity of the critical point, the compressibility factor at the inlet to the compressor (Z_{01}) must also be considered to account for the real fluid behaviour. This is a new feature that is particular to this application and has historically been disregarded in standard turbomachinery based on the ideal gas assumption (Z_{01}=1).

BASE CASE

The base case is a 10 MWe power plant based on a simple recuperative sCO2 Brayton cycle [Angelino, Feher] and used in a Concentrated Solar Power plant. In this field of application, reasonable near term operating conditions are 1100 K turbine inlet temperature and 250 bar maximum cycle pressure. Therefore, under this constraints and the assumptions in table 1 [Muñoz de Escalona], the inlet conditions to the compressor are set to 40 ºC and 75 bar, resulting in a mass flow rate of 73 kg/s and a total pressure ratio of 3.33. To overcome this pressure ratio, the compressor is split into several stages owing to the need to reduce shaft speed to avoid condensation risks. The multi-stage approach is common in the power range under consideration [Fuller et al.] and, in this case, gives place to three stages with an average stage pressure ratio of about 1.5 (note that it is stage work which is constant and not pressure ratio) when a 2% pressure drop is assumed along the return channel.

Thus, for the reference case described in the paragraph above, this paper investigates the design space of a radial compressor pumping 73 kg/s of carbon dioxide at 75 bar and 40 ºC with a pressure ratio of 1.5.

Table 1. Geometrical design choices.

Isentropic efficiency of turbine/compressor	90/80 %
Recuperator effectiveness	95 %
Pressure drops 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 %

DESIGN CONSTRAINTS

Radial compressor design implies a highly complex process involving several parameters whose values are determined based on a number of design choices relying on the designer's experience. Accordingly, in addition to the aforementioned inputs, the following design specifications have to be considered:

- The flow at inlet to the impeller is supposed to be isentropic and with no-swirl velocity, meaning that blockage is null (the boundary layer has not grown yet) and the flow is completely axial.
- Based on past design experience by the authors in the power range considered, the hub radius at the impeller inlet is set to 25 mm. This value is expectedly consistent with mechanical integrity if a maximum shear stress of 400 MPa is considered.
- Constant meridional velocity is assumed ($v_{m1} = v_{m2}$ [Dixon & Hall]) while incidence is set to 0º.
- Same number of splitter and full blades, calculated based on a maximum aerodynamic loading parameter of 0.9 [Aungier]:

  $$\frac{2\Delta w}{w_1 + w_2} \leq 0.9$$

  where, for this specific case:

  $$\Delta w = \frac{2\pi D_2 u_2 v_{u2}}{z_{FB} L_{FB} + z_{SB} L_{SB}}$$

- Finally, the following table complements the required design choices that affect the compressor in terms of manufacturability :

Table 2. Geometrical design choices.

Impeller type	Uncovered
Splitter-to-full blades length ratio	0.5
Thickness of impeller blades	1 mm
Angle between mean streamline and rotation axis at impeller inlet section	0º
Angle between mean streamline and rotation axis at impeller inlet section	90º

DESCRIPTION OF DESIGN SPACE LIMITS

Any design requires the specification of three dimensionless parameters. Therefore, the results are plotted in ϕ-ψ planes for which specific speed is kept constant and where iso-curves of the remaining variables are drawn; i.e. efficiency, degree of reaction, blade angles, etc. Additionally, the following limits are analysed and plotted when necessary: (i) supersonic flow, (ii) two-phase flow, (iii) surge and secondary inefficiencies and (iv) shaft integrity.

Supersonic flow

Given that the absolute velocity at the inlet is axial, the relative velocity at this station is always higher than the absolute one. When this velocity exceeds the sonic limit a shock wave is formed, incurring important losses which result in undesirable compressor design due to poor performance. Moreover, even if the relative Mach number is substantially higher than one, shock waves are still not normal to the leading edge, bringing about their reflection. Modelling this phenomenon is complex and does not give added value from a design standpoint. Therefore, this limit is identified in the model as one of the boundaries of the design space even if such phenomenon cannot be modelled.

Two-phase flow

As stated before, the flow enters the two-phase region in $M_{th} \geq AMC$, which is achieved by extreme values, both high and low, of the flow coefficient. Low ϕ are associated with low meridional velocity and high peripheral speed, giving place to pronounced leading edge angles (with respect to the meridional direction) after which the flow is highly accelerated down to the throat exceeding the AMC. On the other hand, high ϕ imply a high absolute velocity itself with little margin for acceleration before the cited conditions are met ($M_{th} \geq AMC$).

Surge and secondary inefficiencies

Surge limits in a compressor are identified by the flow conditions at diffuser inlet inasmuch as this phenomenon typically takes place due to generalized stall operation in a number of diffuser channels. Thus, based on the fact that stall is favoured by high velocity and distorted flow conditions at impeller outlet, those designs associated with high aerodynamic blockage at impeller exit are identified as undesirable. In particular, considering that blockages in the order of 10% (distortion factor around 1.11) are usual in efficient designs, the surge limit is established at 30% blockage[1].

This can be elucidated by the expression employed for blockage, which accounts for three effects: friction (ϖ_{fr} being the skin friction coefficient), aspect ratio (b_2/L_B) and clearance (δ_{CL}).

[1] Note that even if this might seem arbitrary, this limit is tentatively set to explore the design space and can be revisited later in the design process. Actually, the three-dimensional analysis subsequent to this 1D design will typically double check whether or not surge is taking place for a particular design.

$$B = \varpi_{fr} \frac{P_{01} - P_1}{P_{02} - P_2} \sqrt{\frac{w_1 \, \overline{d}_H}{w_2 \, b_2}} + \left[0.3 + \frac{b_2^2}{L_B^2}\right] \frac{A_R^2 \, \rho_2 \, b_2}{\rho_1 \, L_B} + \frac{\delta_{CL}}{2 \, b_2}$$

Where w stands for channel width, b for blade height and A_R for the area ratio of the impeller.

Shaft integrity

The hub radius at the inlet was previously set to 25 mm and, at the end of the design process, has to be double checked for mechanical integrity. To this aim, the following equation is employed:

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

If, upon the application of this equation, the shaft diameter happened to be too low, it would have to be recalculated and the corresponding design spaces (shown later) re-plotted.

MAIN RESULTS

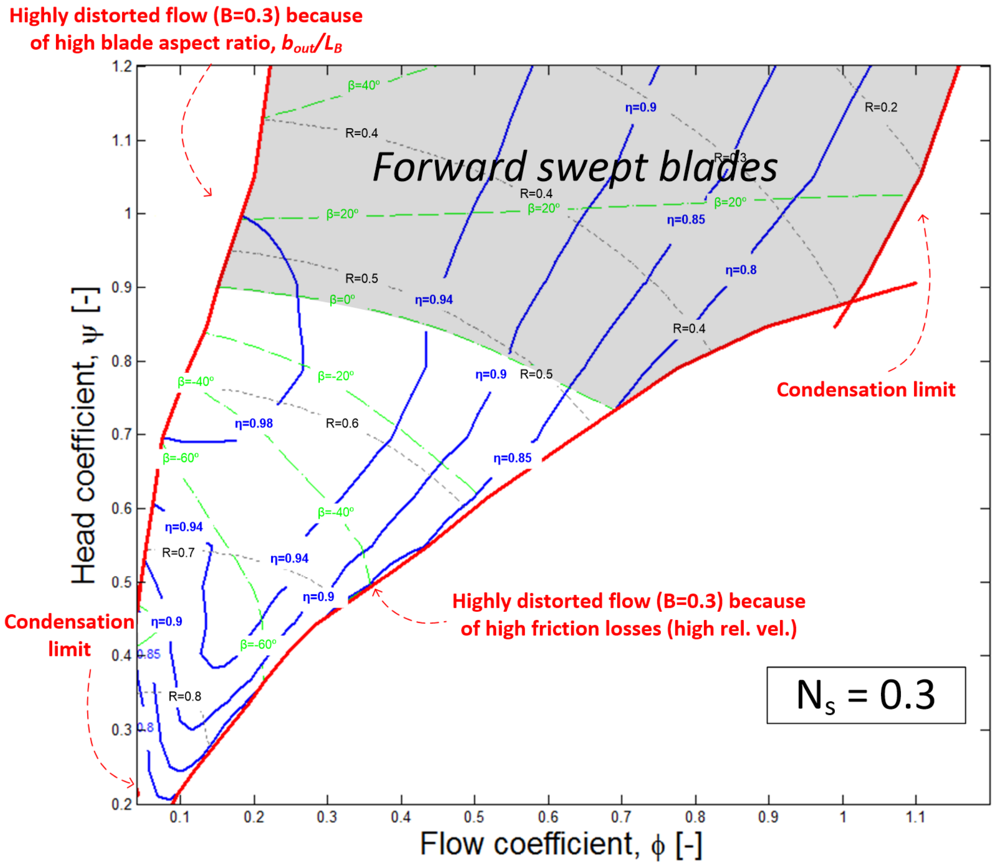

Figure 2. Design space map for Ns=0.3

Three different design maps are presented in this section, corresponding to three different values of specific speed: $N_s = 0.3, N_s = 0.5$ and $N_s = 0.7$ (figures 2, 3 and 4)[2]. In all of them, the two limits corresponding to static conditions within the two-phase flow region (namely the condensation limits for high and low flow

[2] Note that the efficiency quoted in these plots is impeller efficiency only as a similar set of maps is available for the diffuser only and for the entire compressor.

coefficientd) have been identified as well as the boundary indicating very high blockage. Additionally, sonic flow was found at the throat for those designs lying on the condensation boundary.

The dark region in the maps corresponds to those designs resulting in impellers with forward-swept blades, an option that though possible is not usually adopted due to its very narrow operating range. As expected, this region is characterised by a low degree of reaction: as the blades are swept forward, a larger fraction of the compression process takes place in the diffuser. Hence, radial blades ($\beta_2 = 0º$) and $R = 0.5$ lie very close to one another in figure 2 and the degree of reaction decreases down to about $R = 0.2$ at the top right corner of the map. These designs correspond to heavily loaded stages with a very high flow coefficient which, even if possible, yield rather low efficiency.

As a consequence of this observation, more stringent (tighter) boundaries could be set in order to obtain efficient designs. These limits would be the following:

- Radial impellers ($\beta_2 = 0º$), thus discarding those designs with forward-swept blades.
- Extremely swept-back impellers ($\beta_2 > 60º$). These designs present three major shortcomings:
    - Reduced flow capacity due to the very low radial velocity at impeller outlet (low ϕ).
    - Increased distortion at impeller outlet which increases the risk of unstable operation of the diffuser.
    - Little compression in the diffuser (very high degree of reaction $R > 0.7$).

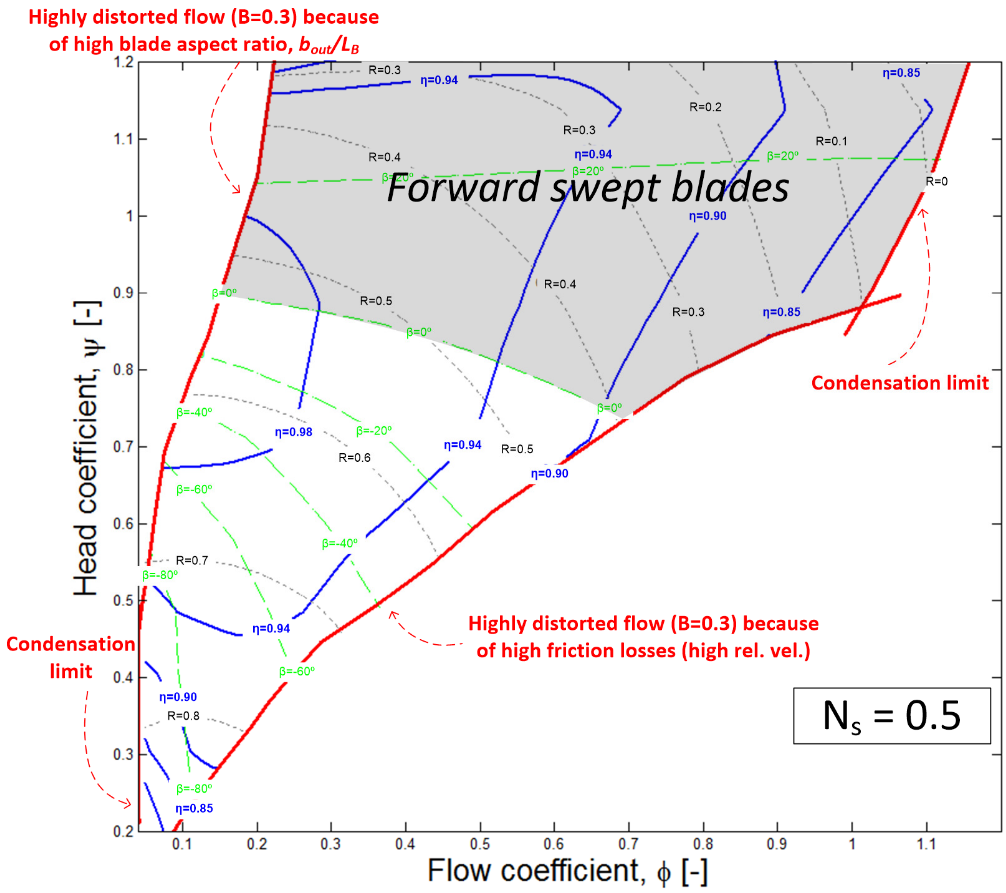

Figure 3. Design space map for Ns=0.5

The design space maps for $N_s = 0.3$ and $N_s = 0.5$ are fairly similar and only a steeper drop in efficiency when the design parameters (ϕ-ψ) move away from the peak efficiency point for $N_s = 0.3$ is worth noting. Other than this, the peak efficiency points is similar and is found in a similar location in the ϕ-ψ map. When looked into with more detail, this highest efficiency area corresponds to impellers where compression takes place mostly due to the centrifugal effect, which is dominant to the relative flow diffusion (which hardly has any influence on compression). In this region of interest, the degree of reaction is in the order of $R = 0.55 - 0.60$ and the blades are swept back some 20º ($\beta_2 \approx 20º$).

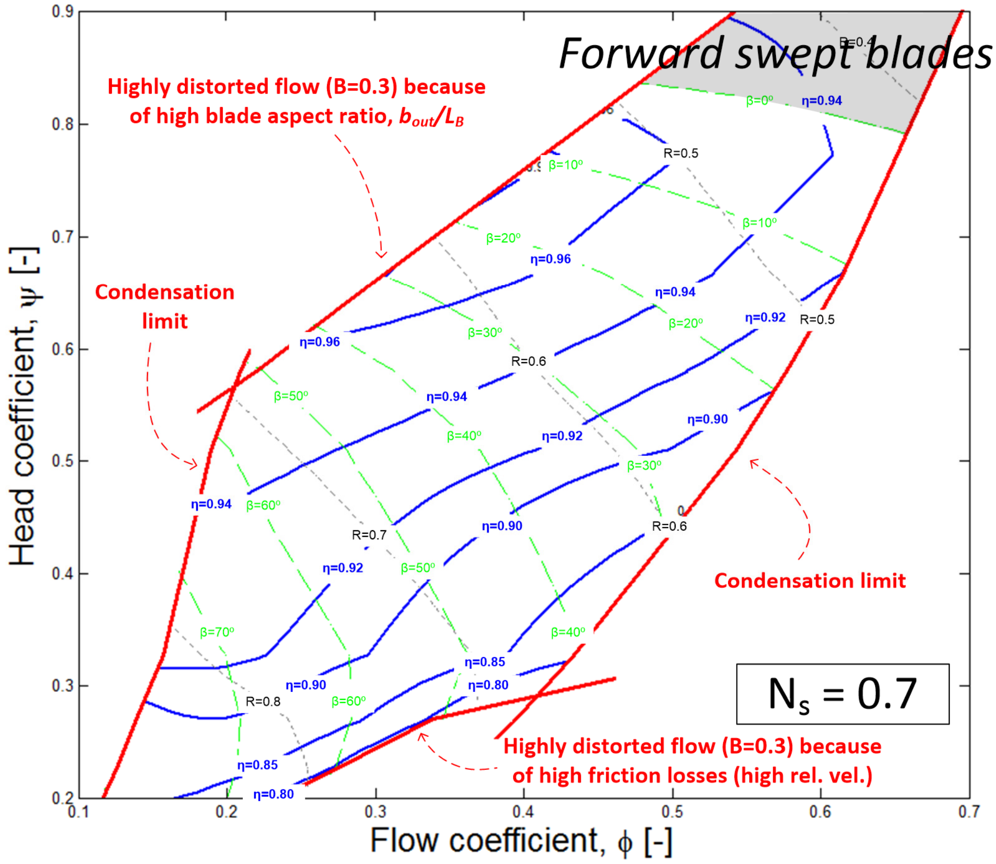

Figure 4. Design space map for Ns=0.7

Figure 4 illustrates the design space map for $N_s = 0.7$, corresponding to a shaft speed of about 15000 rpm, which is notably distorted with respect to the previous cases. This distortion applies to both the condensation boundaries and the boundaries due to aerodynamic blockage. For a given blade (peripheral) speed at the outlet, a higher angular speed (rpm) requires higher relative flow angles at the impeller inlet which, in turn, cause a higher expansion at the throat (M_{th}). This higher risk to enter the two-phase flow region is easily observed in figure 3 where the condensation limits come closer to one another, thus narrowing the design space available.

Geometrically, this throttling yields impellers with higher area ratios which shift the design space towards lower load coefficients (ψ). At the same time, the higher throat Mach number yields (for constant meridional velocity) higher friction which moves the boundary due to high blockage factor slightly towards higher flow coefficients (though this is more a visual effect coming from the reallocation of the boundaries rather than an appreciable quantitative difference).

All this into account, the design space for the higher specific speed is largely reduced as illustrated in figure 4. Moreover, should the forward-swept blades be discarded/banned, the feasible design space would be even narrower given that the line corresponding to radial impellers ($\beta_2 \approx 0º$) lies now in a region of lower load coefficient (ψ) and degree of reaction ($R \approx 0.45$). As expected, this reduced availability of the design parameters is accompanied by a general drop in efficiency due to higher losses.

CONCLUSIONS

The main conclusions drawn from this work are listed below:

- Selecting the specific speed to design a sCO2 compressor is a critical step in the design process given the influence of this parameter on compressor performance.
- The value typically employed in air compressors to achieve highest efficiency ($N_s = 0.7$) does not seem to be optimal when it comes to supercritical carbon dioxide turbomachinery. Not only is peak efficiency lower, but the design space (loci of potential load-flow coefficient combinations) is reduced as well.
- Theoretically speaking, and considering the very conservative assumption that condensation takes place in equilibrium, it does not seem possible to design for specific speeds higher than $N_s = 0.9$ since the flow enters the two phase region regardless of the ϕ-ψ combination of choice.

FUTURE WORK

Future work by the authors aims to complement the information presented here by:

- Extend the design space maps to take into consideration other design parameters like, very importantly, compressibility factor. The influence of the total inlet conditions on compressibility is deemed crucial in these applications and therefore deserves a separate analysis in detail.
- Develop specific design space maps of the diffuser, including vaneless and vaned diffusers, which can later be combined with those of the impeller to yield more complex design space maps for the entire compressor.
- Incorporate mechanical integrity considerations in the maps and, when necessary, add the corresponding mechanical boundaries.

NOMENCLATURE (except if noted in the text)

1	=	Impeller inlet
2	=	Impeller outlet
AMC	=	Acceleration Margin to Condensation
CR	=	Critical
FB	=	Full blade
PR	=	Pressure Ratio
SB	=	Splitter blade
TH	=	Throat
a	=	Streamwise coordinate
b	=	Blade height
c_{ax}	=	Streamwise (meridional) velocity
D_s	=	Specific diameter
h	=	Enthalpy
M	=	Mach number
N_s	=	Specific speed
L	=	Blade length
$\dot{P}$	=	Speed of expansion
R	=	Degree of reaction
Re	=	Reynolds number
u	=	Blade speed
v_u	=	Tangential component of absolute velocity
v_x	=	Streamwise component of absolute velocity
$\dot{W}_m$	=	Shaft power

x = Quality
z = Number of blades
Z = Compressibility factor
β = Blade angle
η_s = Isentropic efficiency
ϕ = Flow coefficient
ψ = Load coefficient
ρ = Density
τ = Time
τ_m = Maximum torque stress
τ_m = Maximum torque stress
ω = Shaft speed

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