Robust Centrifugal Compressor Stage Design for Industrial Process Super-Critical CO_2 Applications

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ABSTRACT

Process compressor design practices are well established for fluids such as air and many hydrocarbons. Modern compressor stages for these applications operate with a wide margin for mechanical reliability and achieve broad aerodynamic operability. There is growing demand for process compressors designed for supercritical carbon dioxide (sCO2) to address global net-zero

carbon emission goals. Carbon sequestration, district heating heat pumps, and advanced closed-loop Brayton power systems are some of the applications for which sCO2 is an ideal working fluid. Efficient and robust compression systems for sCO2 push the traditional boundaries of aerodynamic and mechanical design of process compressors. To meet these challenges, the mechanical and aerodynamic design criteria must be revisited to achieve stage designs that meet the technical demands of these new applications.

The mechanical and aerodynamic design challenges of sCO2 compressor stages push changes to traditional configurations to meet fatigue, rotor-dynamics, material mechanical limitations, and adherence to manufacturing limitations. A few specific challenges are the impact of high density of sCO2 on rotor-dynamic instability, and sCO2 is particularly susceptible to causing explosive decompression in O-ring seals. This paper presents a detailed overview of an approach for improving process compressor design practices to address these multifaceted challenges, integrating aerodynamic and mechanical aspects to optimize across disciplines concurrently. This methodology aims to enhance the efficiency, reliability, and performance of sCO2 compressor stages, contributing to the advancement of cutting-edge technology in the field.

INTRODUCTION

1.1 Background

Process compressors have been at the heart of industry for decades, ever being improved to provide more range, efficiency, and power in a single unit. Decarbonization has become the modern driving call for innovation of every industrial component. Numerous applications involving supercritical CO_2 (sCO_2) compression are presently being developed in support of global decarbonization goals. Applications such as pipeline transport, waste heat recovery, direct power generation, and CO_2 sequestration are all active areas of product development. Many of the challenges specific to turbomachinery in sCO_2 cycles arise from the high-power density that exists at elevated pressures of CO_2 and the operational requirements across diverse applications.

Multistage process compressors can be configured as either an inline machine, Figure 1, or an integrally geared compressor (IGC), Figure 2. This paper focuses only on IGC application into sCO2 service.

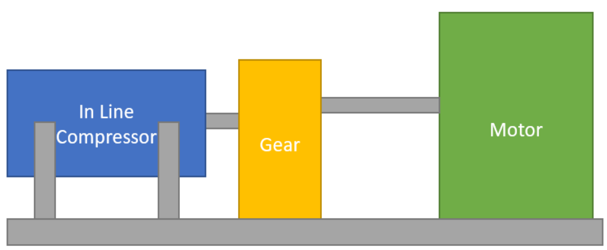

Figure 1: Simple Inline Machine Configuration

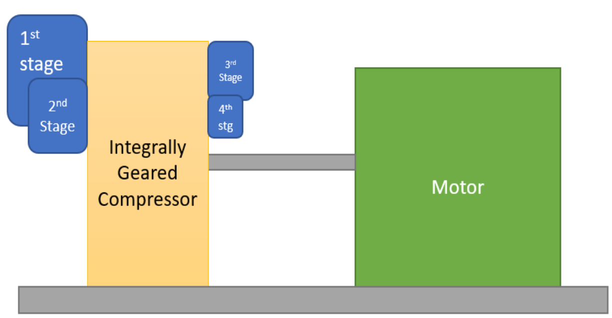

Figure 2: Simple IGC Machine Configuration

In an inline compressor the driver is coupled to a speed increasing gearbox then to an independent compressor body where all the stages operate on a common rotor at a single speed and similar diameters. This has the advantage of minimizing the number of seals and bearings. The driver in an IGC is coupled to a single large bull-gear driving multiple pinions at different speeds. Each of these pinions typically has an overhung impeller stage on each end. This configuration offers improved aerodynamic matching of speed and stage sizing as well as easier intercooling. However, IGCs require more seals, bearings and piping, resulting in more complex packages.

Compressing CO_2 into its supercritical phase causes the volumetric flow rate within a compressor to decrease drastically from stage to stage as the density increases. This requires significant size reduction and speed increase of the downstream stages to maintain optimal performance. Each IGC stage has its own casing and thus can be more closely sized for the pinion speed and flow coming from upstream stages of compression allowing for higher efficiencies. Since inline centrifugal compressors operate with all the stages on a single rotor, the flow coefficient of the downstream stages can become very small and lead to poor overall performance. This can be mitigated by splitting the process into multiple bodies which allows for intercooling and speed increasing. This adds cost and complexity to the inline configuration such as more seals and bearings, eliminating some of the advantages that an inline machine may have over an IGC. This puts the IGC design at the forefront of sCO_2 applications with its capacity for easy intercooling, and stages with speeds and sizes optimized for maximum efficiency.

IGC compressor development has made tremendous progress since its inception in the 1960s, increasing its suction capacity by four times over forty years [1]. Concurrent development of centrifugal compressors for carbon dioxide has been ongoing. Carbon Capture and Storage (CCS) are crucial technologies for achieving the goal of reaching net-zero human-made greenhouse gas emissions by 2050 [2]. Independent of the method used to capture the CO_2, the process to compress carbon dioxide for liquefaction, transportation, and storage remains central to CCS [3]. The CO_2 stream exiting almost any capture technology is in the gas phase, but it will be transported as a sub-cooled liquid for ship-based transportation or a dense-phase gas for pipeline-based transportation. Inline centrifugal compressors and integrally geared compressors are compared for large scale CO_2 compression applications [3]. Wacker, et. al. [4] gives an overview of advanced and proven technologies for CO_2 compression. The authors list relevant design challenges concerning thermodynamical and mechanical design in regard to topics arising when compressing CO_2 in the supercritical region. These topics include the pronounced real gas behavior of supercritical CO_2 and the close-coupled design process of integrally geared compressors.

Brun [5] describes the link and technical challenges between the hydrogen and CCS based economies. The author writes that the trend towards a carbon free energy economy requires significant investment into technologies to efficiently produce, transport, store, and utilize hydrogen. As the hydrogen market increases its capacity to produce green hydrogen, most hydrogen entering the market over the next 15-20 years will still be natural gas derived blue hydrogen. Consequently, there will be a need for added new compression of carbon dioxide (CO_2) for separation, pipeline, and storage injection. About 10 kg of CO_2 is produced for every kg of blue hydrogen that is produced through steam reforming or partial oxidation gasification. This highlights that technical challenges face not only hydrogen compression but CO_2 compression as well.

1.2 Design Challenges

There are several machine design challenges due to the unique properties of sCO2 which push the limits of modern compressor aerodynamic and mechanical design. Both perspectives have to be considered simultaneously when pushing the boundaries of conventional process compressor design.

While CO_2 is not the heaviest gas found in process compressors, with a mole weight of 44, it is unique in that when compressed to its supercritical state, its density becomes closer to that of a liquid than a gas. This introduces challenges in an inline style compressor not typically found when compressing fluids that perform in a fully gaseous state. The main challenge is the volume reduction of the gas from inlet to discharge is so great that the first stage is nearly six times larger in diameter than the last stage. This immense reduction in volume makes it nearly impossible to compress CO_2 to a supercritical state within a single inline style machine.

Figure 3 shows the relative size of the stages in a 7 stage CCS IGC compressor. In this example the final stage is about 15% the size of the first stage which intakes ambient CO_2. This aligns with the compression process shown in Figure 4 which highlights a nearly 3-order of magnitude increase in the density of the process fluid in this sample 7 stage process.

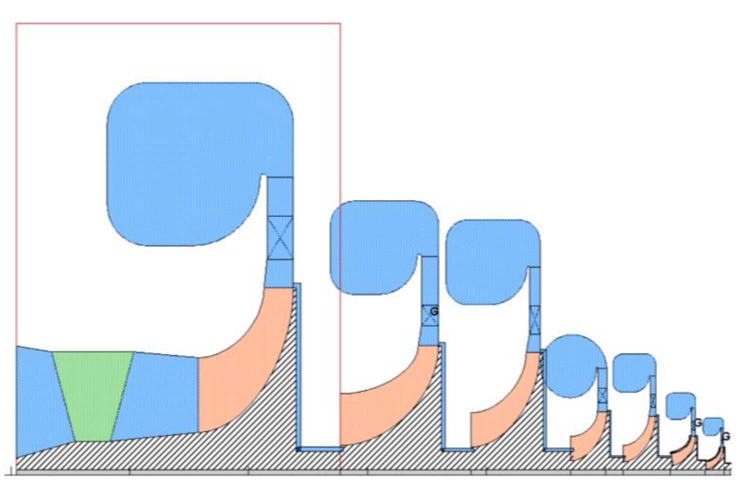

Figure 3: Typical 7-Stage CCS IGC Compressor Stage Size Comparison

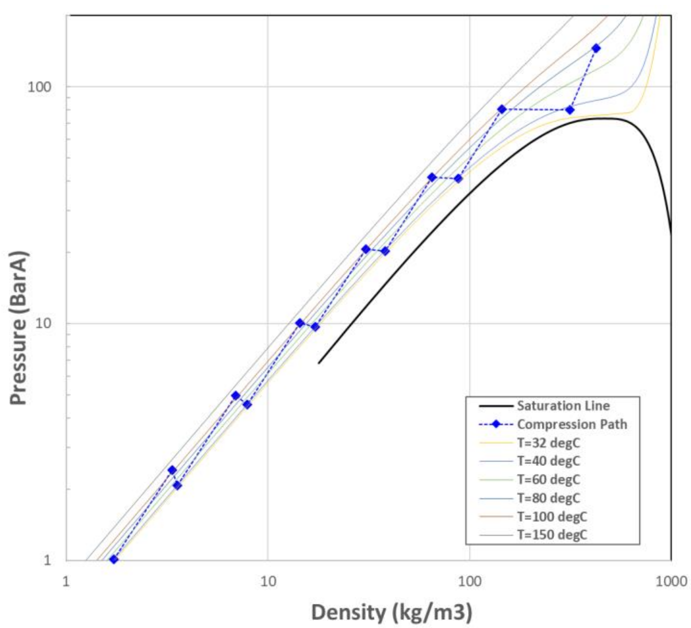

Figure 4: PH diagram for a 7 stage CO2 Compression Process with Intercooling on Each Stage

Understanding this, the focus for sCO_2 process compressor development in this case study is on IGCs rather than inline style machines. The development of a sCO_2 IGC compressor revealed that rotor-dynamic instability, specifically due to aerodynamic loading of high-density CO_2, is one of the major design challenges to address.

The preliminary rotor-dynamic analyses for this case study sCO_2 IGC machine's high-pressure pinion rotor showed points in Region B of critical speed ratio (CSR) plots as divided and described in API 617 [6], Figure 5. Rotor-dynamic CSR is the ratio of the first Prohl speed to the maximum continuous speed where the first Prohl speed is the rotor's undamped natural frequency assuming infinitely stiff bearings. CSR can be plotted against many mechanical values and is typically plotted against average gas density. API [6] divides this specific plot combination into the two regions shown in Figure 5. The two regions show combinations of CSR and average gas density, or other mechanical results one wishes to review that are of different levels of risk and requiring different levels of analyses.

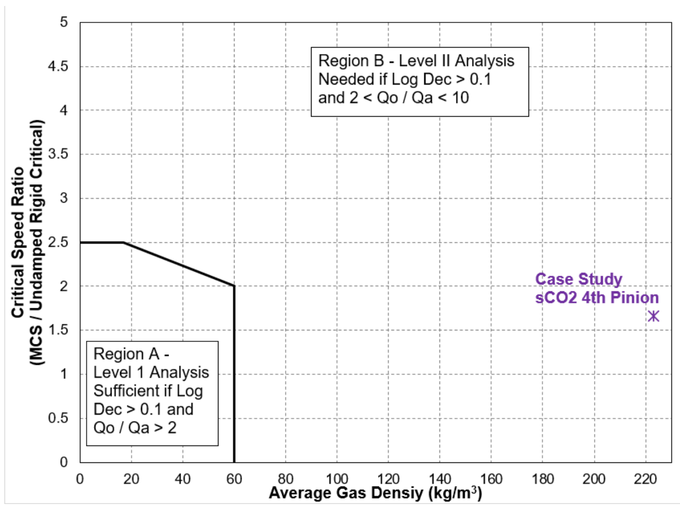

Figure 5: Critical Speed Ratio Vs. Average Gas Density

Region A has experience showing there is lower risk when the criteria are met within the region and thus only requires an API Level I rotor-dynamic analysis. Region B requires a higher level of rotor-dynamic analysis due to the higher risk and uncertainty of rotor stability when operating with that combination of CSR and gas density. Ideally rotating machinery is designed towards Region A with various changes to the rotor that help bring the CSR down, or possibly decrease the gas density of a specific pinion in the case of an IGC. Average gas density is often a fixed parameter defined by the compression process. One method to influence the CSR is to modify the stiffness of the rotor. Equation 2 shows how to calculate the stiffness of a rotor section.

$$K = \frac{48 * E * I}{L^3} \qquad (2)$$

Diameter impacts stiffness more than length since the area moment of inertia term has a diameter to the fourth power term while the stiffness equation has a length cubed. However, if the diameter change limit is reached before achieving the rigidity necessary, modifying the length of parts of the rotor is also a way to affect the CSR. This trade-off is often referred to as the length over diameter (L/D) ratio. Lower L/D ratios indicate higher stiffness shafts.

The bearing span is also an important feature to consider when evaluating the stiffness of a shaft. IGC pinion rotors have overhung impellers with each stage having an axial inlet and its own scroll resulting in a shorter bearing span since the bearings have to be outside of the process gas but should be as close as possible to the impeller to limit the overhung weight. Inline machines have much longer bearing spans due to the need to have the aerodynamic stages together while keeping the bearings outside of the process gas. Figure 6 and Figure 7 show basic cross-sections of IGC and inline machines. While not to scale, the example second IGC pinion has similar stage diameters as first few stages in the inline machine example. The example fourth IGC pinion stages here show the large volume reduction from inlet to discharge being around one quarter of the size of the second pinion stages. Bearings are represented by the boxed x’s.

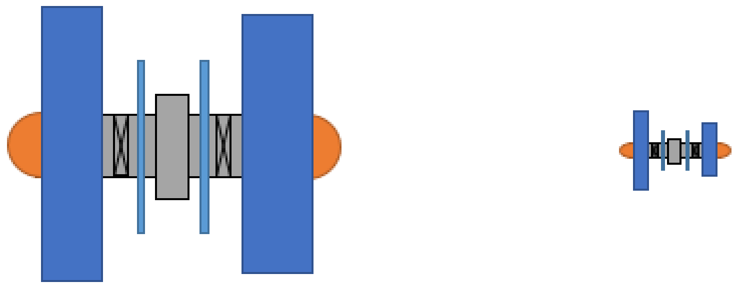

Figure 6: Basic Cross-Section of a Second (Left) and Fourth IGC Pinion Rotor (Right)

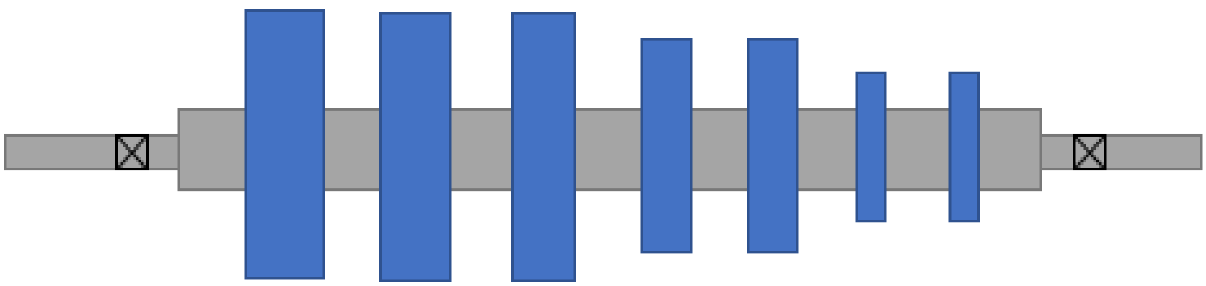

Figure 7: Basic Cross-Section of a Seven Stage Inline Rotor

Sealing the casing faces against the gearbox is a problem unique to supercritical fluid compression, specifically sCO_2. IGC casings are designed with multiple pieces fitting together building from the gearbox out toward the inlet of the stage. The main casing body is installed over the shaft, the impeller is installed, then the final pieces to the stage are installed in front of the impeller. These fit together tightly but not tight enough to prevent gas from passing between them so O-rings are used to seal the spaces between these final stage components. Figure 8 below shows a cross section of an IGC stage with the major casing components numbered in rough assembly order. O-rings work very well to seal gas but can be at risk of rapid gas decompression failure (RGDF) especially in high-pressure, high-density fluid environments. RGDF happens when the pressurized gas is absorbed into the O-ring, diffused throughout it, then desorbed. This can cause the O-ring to swell. When the system pressure is lowered the gas trapped in the O-ring can rip / damage the O-ring when it diffuses faster than the surface area can allow based on the permeability of the material. Mo et al [7] explains that RGDF in polymer chain materials, such as what O-rings are made of, has two main mechanisms. These are chain breakage, the splitting of polymer chains due to plastic loading, and cross-linking point failure, where the rubber network starts to pull apart under high stress. Higher durometer O-ring materials resist gas entrapment and the potential of RGDF better. Typically, compressor applications use 70 durometer O-rings but for cases where RGDF is a concern, higher durometer O-rings such as 90 are used.

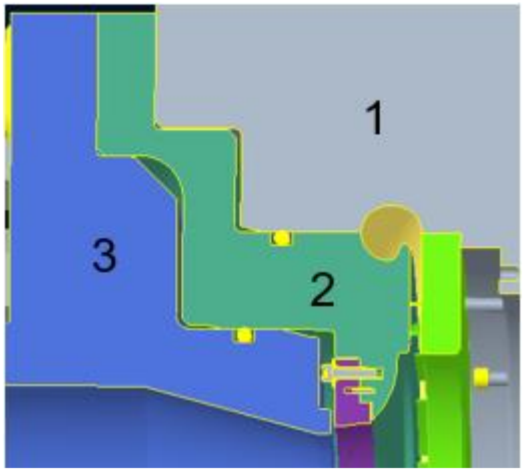

Figure 8: Example IGC Stage Build Up Components And O-Rings

Many high-pressure gasses can cause O-ring RGDF but CO_2 is known to be a higher risk for RGDF than other gasses such as nitrogen. Chen et al [8] explains that CO_2 is a linear molecule that is non-polar due to this shape causing its two dipoles to cancel each other out. This allows CO_2 to permeate into both polar and non-polar materials with ease. This is further increased when CO_2 is in its supercritical state due to the high pressure and temperature combination that exacerbates the absorption and diffusion of gas into permeable materials such as polymer O-rings.

RESULTS AND DISCUSSION

2.1 Mechanical Configuration

Rotor-dynamic stability is imperative for any rotating machinery. It is an evaluation of how the natural frequencies of a machine's rotor interact with respect to its running speed and critical speeds. This is used to ensure the machine does not self-excite and cause excessive vibration and damage.

Figure 9 shows an overlay of lateral rotor-dynamic stability of various carbon dioxide compressor stages compared to an inline H2 compressor and an IGC H2 compressor. The lateral rotor dynamic stability of compressor configurations can be separated into various API [6] categories as mentioned previously. More complex rotor configurations require additional analytics to be qualified. These are shown as Region B in the plot. By contrast, simpler configurations are characterized in Region A. Notice how, for comparison, the increasing number of stages on a single rotor of an inline hydrogen compressor has little effect on gas density, but quickly increases CSR moving into Region B after only four stages. A two stage H_2 IGC pinion overhung on both stages narrowly reaches into Region B. The CO_2 IGC stages, by contrast, only move along the average gas density and actually decrease in CSR since the overhung weight reduces due to the smaller downstream stages.

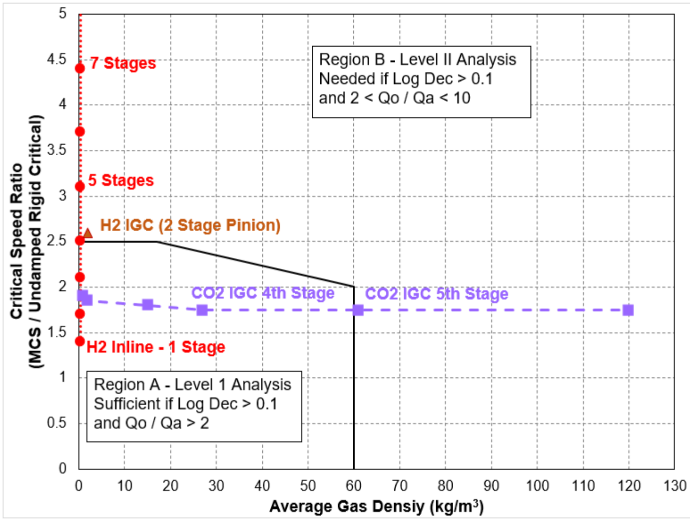

Figure 9: Rotor-Dynamic Stability Comparison of H2 And CO2 Compressors

CO_2 compression for CCS and pipeline transport is most readily done by 8 stage IGC's. CO_2 compressors are driven into Region B not by increasing CSR, but by increasing gas density. After the 5th stage (3rd pinion) a level II analysis is required. A fourth pinion must deal with very high gas densities that will tend to drive lateral instabilities. Therefore, stability mitigation efforts are essential for the 3rd and 4th pinions. Wygant [9] describes the evolution of integrally geared compressors from plant air applications to more challenging applications such as CO_2 and hydrogen compression and describes many of the other rotor-dynamic challenges.

Bearing designs fall into similar challenges for hydrogen and sCO_2 compressors as the lateral rotor-dynamics. Vannini [10] shows, through a dedicated rotor-dynamic study, how the Flexure Pivot journal bearing with Integral Squeeze Film Dampers enable a rotor-dynamic design that fully complies with the requirements and moreover offers some new experimental data to confirm the validity of the design.

Bearing surface velocity is also an important factor in rotor-dynamic stability. A larger shaft makes for a more rigid rotor but means larger bearings which results in higher bearing surface velocity. There is a limit where bearing surface velocity negatively affects the rotor stability. However, while a smaller shaft reduces the bearing surface speed, it also decreases the rigidity of a rotor. With both inline and IGC machines, the rotor is between two bearings and this bearing span is a crucial factor in the rotor-dynamic stability of the rotor. The overhung masses of the impellers on IGC rotors also influence rotor-dynamic stability.

The rotating components undergo stress and strain evaluations at an overspeed rotational value defined by API [6] as at least 115% of max continuous operating speed. This evaluation is to determine the mechanical limitations of an impeller design. Additionally, there are modal checks performed on both the impeller and the diffuser, especially when it is a vaned diffuser. This is used to determine if there is any risk of resonance between the impeller blades and the downstream blades further ensuring machine operational stability.

It is also necessary to consider how the impeller is connected to the shaft. IGC impellers are overhung style, typically small bore, fitting over a stud in one end of the shaft which is tensioned with a hydraulic puller and held in place by a nut. Since IGC are a series of single or double overhung stages and volutes on a common bull gear, having an overhung impeller reduces the number of the stationary components that need to be removed during overhaul to only those that prevent access to pulling the impeller off of the small shaft in the upstream axial direction. This may also include the some of the piping connecting each stage with inter-coolers. Since it is designed around the components and it can become very complicated.

The IGC rotor-dynamic issue is dominated by the mechanical aspects of the machine but is driven by the high density of CO_2 in the super-critical phase in the last few stages of the machine. This is depicted in a critical speed versus average gas density plot in Figure 5. Because the gas density cannot be reduced and there is not much to be done with respect to the pinion design to further reduce the CSR. Consequently, an API level 2 [6] analysis was performed on the case study design. The results of this analysis showed there is sufficient stability margin and the machine is acceptable in its current stability Region B state.

2.2 Aerodynamics

Centrifugal compressor stages operate with a near constant head coefficient, Equation 3, generating a near consistent enthalpy rise at a given speed and flow even with different working fluids. Depending on the gas properties this constant enthalpy rise does not result in the same level of stage pressure ratio. The chart below shows that heavy gases easily achieve a high-pressure ratio while very little pressure rise is achieved at the same tip speeds when operating in low molecular weight gases, Figure 10. For CO_2 the achievable pressure ratio is even greater when operating at supercritical conditions.

$$\Psi = \frac{\triangle h}{U_2^2} \tag{3}$$

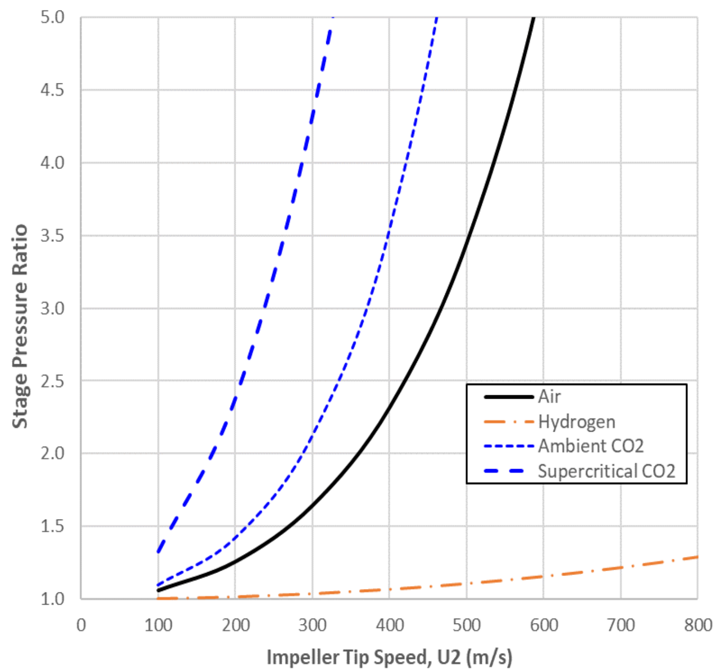

Figure 10: Stage Pressure Ratio Vs. Impeller Tip Speed for Various Gases

Therefore, achieving a very high-pressure ratio for a CO_2 application can be accomplished in fewer stages than a light MW gas. For example, a 2000 psi (13.79 mpa) CO_2 injection compressor with an overall pressure ratio greater than 130:1 may have just 7-8 stages.

There are practical limits to the performance of each stage from both aerodynamic and mechanical standpoints. First, mechanical stresses increase with the square of tip speed. Therefore, conventional steel alloy impellers in process compressors typically must operate at tip speeds below 1640 ft/s (500 m/s) to stay within mechanical limits. This restricts the achievable stage head generated since head varies directly with the impeller tip speed. It becomes increasingly difficult to find materials with the required mechanical properties to operate faster than conventional steel alloys can handle. Additionally, the achievable aerodynamic efficiency and operating range of a stage begins to fall as the machine Mach number increases beyond 1.2. These constraints combine to define the practical design space for compressors in terms of tip speed and machine Mach number shown in Figure 11. This shows that for CO_2 applications the machine Mach number limits constrain the design rather than mechanical stresses associated with high speeds. While compressors designed for lighter gases, such as hydrogen, will be constrained by the achievable impeller tip speed for the chosen material. Accordingly, for most sCO2 applications, the centrifugal stress in the impeller will be fairly moderate allowing for flexibility in the mechanical design.

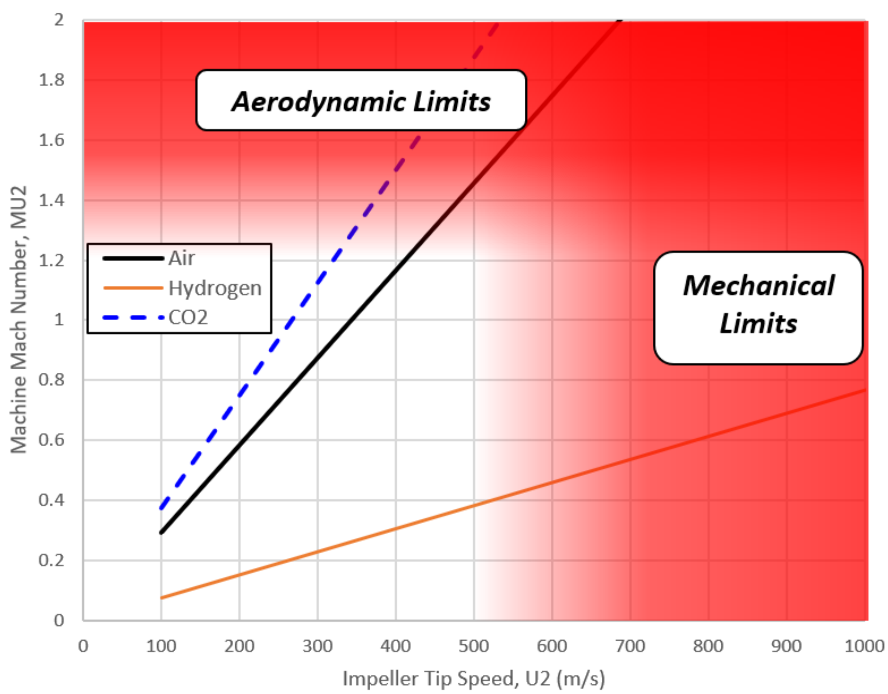

Figure 11: Achievable Aerodynamic Efficiency and Operating Range

2.3 O-rings

O-rings are critical to sealing any process gas compressor. sCO2 application compressor casings should be designed in such a way to reduce the number of O-rings and to allow for smaller than typical cross-section O-rings to be used. Both of these reduce the risk of RGDF. Designing the casing to reduce the number of O-rings reduces the number of possible RGDF occurrences. Reducing the O-ring cross section limits how much gas the O-rings can absorb, also reducing the risk of RGDF. Parker O-ring [11] suggests a cross-section of 0.210" or smaller as well as designing the groove so that the gland fill is close to but not exceeding 90% the maximum recommended for a given O-ring. Gland fill is how much of the O-ring groove, or gland, that the O-ring takes up.

Material selection is key to ensuring sCO2 compressors do not experience RGDF. Higher durometer O-ring materials resist gas absorption better than low durometer materials. Typically, compressor applications use 70 durometer O-rings but for cases where RGDF is of concern, such as sCO2, 90 durometer or higher are used. There are also specific materials which are more resistant to RGDF than others, specifically designed to be better in these failure modes. AFLAS is a common O-ring material used for compressors with applications where this type of failure is a risk.

Spring energized C-seals are good alternatives to O-rings if the fit location allows it. These seals are typically made of Teflon, which does not allow the gas to be absorbed into the seal. Therefore, RGDF cannot occur, because there is no high-pressure gas entrained in the seal. These types of seals would require different types of grooves, and mating conditions between components. They would be an option for any face seals present, but would be complicated to use at radial seal locations.

One last, very important consideration, is startup and shut down procedures. Since RGDF occurs when a fluoro-elastomeric material, which absorbed a gas at high pressure and temperature, suddenly experiences a large pressure drop, it is important to control the material's rate of desorption. By controlling the machine rate of startup and shutdown so that the O-rings do not see pressure drops too quickly, RGDF can be avoided. Parker O-ring [11] recommends pressure drops of no more than 200 psi (1.38 mpa) per minute but this is dependent on the material and specific application conditions. API 617 also states that when RGDF is of concern, the "supplier shall state the maximum allowable depressurizing rate of the compressor casing required to prevent explosive decompression of the O-ring material supplied" [6]. This allows the gas time to desorb at a safe rate without tearing the polymer chains apart causing O-ring failure.

3.1 Conclusion

The case study sCO2 IGC machine will have a prototype that will validate the analyses completed and design changes to address the unique concerns of sCO_2 compression to show that it can operate successfully despite rotor-dynamic stability concerns of Region B, large stage reduction due to high compression ratio and CO_2 phase change, and O-ring RGDF.

Designing process compressors for super critical CO_2 applications to address global net-zero carbon emissions goals shows unique design constraints that are best addressed simultaneously during the product design. Although the processes have unique properties and solutions, this paper described how designing for turbomachinery to operate in areas with limited experience requires continual communication between the mechanical and aerodynamic design engineers. sCO_2 IGC compressors may always have rotor-dynamic results in Region B due to the gas density and pinion design constraints, risks of RGDF, but they can be evaluated, designed around, and accepted.

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ACKNOWLEDGEMENTS

The authors appreciate the support of Ebara Elliott Energy in the creation of this paper.