The 6th International Supercritical CO_2 Power Cycles Symposium
March 27 - 29, 2018, Pittsburgh, Pennsylvania

DESIGN OF A SUPERCRITICAL CO_2 COMPRESSOR FOR USE IN A 10 MWe POWER CYCLE

Stefan D. Cich
Research Engineer
Southwest Research Institute
San Antonio, TX

Jeff Moore, PhD
Institute Engineer
Southwest Research Institute
San Antonio, TX

Jason P. Mortzheim
Senior Engineer, Mechanical
GE Global Research
Niskayuna, NY

Douglas Hofer, PhD
Senior Principle Engineer
GE Global Research
Niskayuna, NY

ABSTRACT

An enabling technology for a successful deployment of the sCO_2 close-loop recompression Brayton cycle is the development of a compressor that can maintain high efficiency for a wide range of inlet conditions due to large variation in properties of CO_2 operating near its dome. One solution is to develop an internal actuated variable inlet guide vane (IGV) system that can maintain high efficiency in the main and bypass compressor with varying inlet temperature. A compressor with this system is not currently available in the marketplace for sCO_2 applications. This compressor was developed with funding from the US DOE Apollo program and industry partners. The lower thermal mass and increased power density of the sCO2 cycle, as compared to steam-based systems, enables the development of compact, high-efficiency power blocks that can respond quickly to transient environmental changes and frequent start-up/shut-down operations. While being able to make something that is much smaller than conventional compressors in the industry is good from material cost and weight perspective, it also presents a challenge in packaging all the needed components and features in a smaller envelope. This paper describes the detailed design of a back to back compressor including pressure containment, rotordynamics, sealing, variable IGVs, rotor layout, and packaging. It will also look into the challenge of balancing requirements from these competing disciplines and how it resulted in an optimized design for this application.

INTRODUCTION

The recompression Brayton cycle (RCBC) is an attractive cycle for sCO_2 that could meet the relative efficiencies of steam based Rankine cycles. To meet these efficiencies, the cycle requires low and high temperature recuperators, turbine inlet temperatures between 600-760^{o}C, a main compressor, and a bypass compressor. There is an ideal balance between the flow split on the main compressor. While the lower temperature flow requires less power to compress, the higher temperature will increase the effectiveness of the recuperation in the loop [1]. Other US DOE projects have focused on the cost and effectiveness of the recuperators and the design and development of high temperature sCO2 turbine. The US DOE Apollo program's main focus is on the design and testing of the main compressor for this RCBC. Figure [1] shows the overall cycle model and design conditions that are looked at for all of these components:

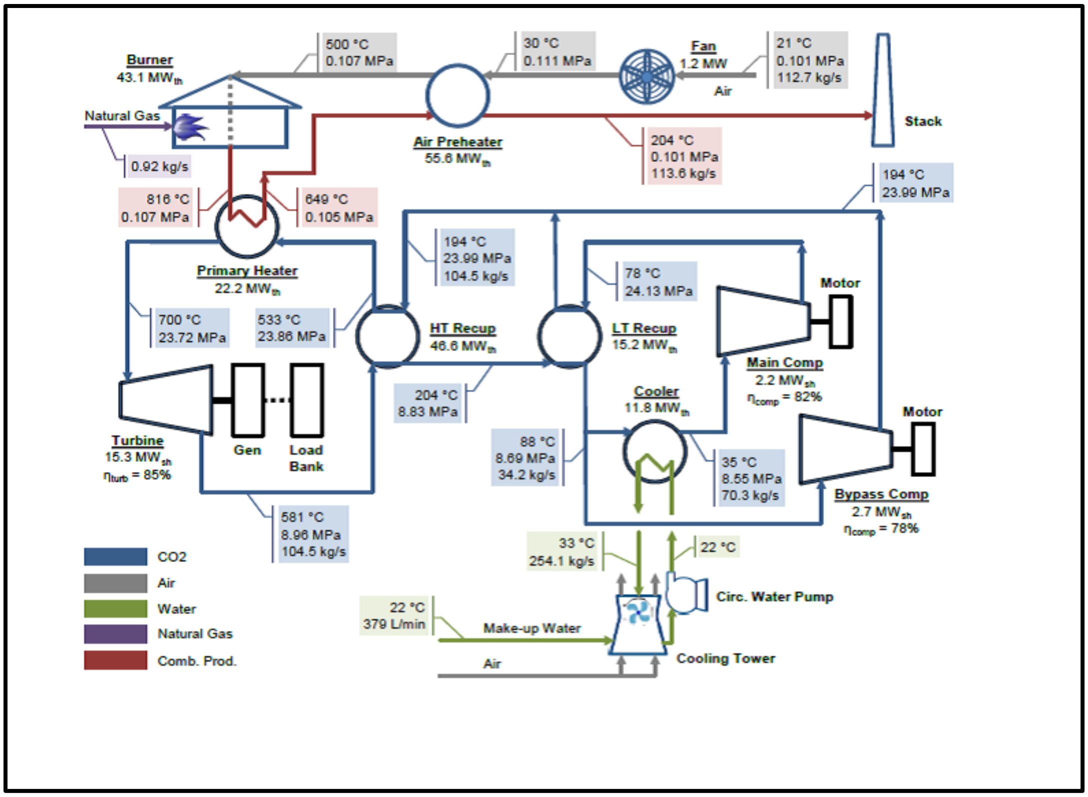

Figure 1 – sCO_2 Closed-Loop Re-compression Brayton Cycle Model

With its higher efficiencies and compact design, sCO_2 cycles have gained interest in various forms of power generation: solar, coal, natural gas, and waste heat recovery to name a few. The focus of this paper will be on the advantages and challenges of designing a compact compressor for use in all of these applications.

While limiting the overall foot print might be more beneficial in solar and waste heat applications, having less equipment, materials, and components is always going to be advantageous when designing and building a power generation system. However, smaller does not always mean easier. With compact turbomachinery, like sCO_2 compressors and turbines, packing all of the necessary features becomes a challenge. In addition to packaging, it is also important to find an optimal balance between key disciplines for turbomachinery design: rotordynamics, aero dynamics, and mechanical design.

To take full advantage of the compact design, it is ideal to combine the main compressor and bypass compressor into a single casing, and also have the compressor directly coupled to the turbine. This reduces the overall foot print and also prevents the need for additional motors, gearboxes, couplings, and skids. This does present a challenge to the overall design of the compressor. Since the compressor will be on a single train with the turbine and the generator, there will be no variable speed control on the compressor to maintain efficiency with varying fluid properties [2].

At the design conditions of 35^{o}C and 8.55 MPa, the sCO_2 is operating close to its property dome, Figure [2]. The blue and red lines represent the nominal inlet temperature to the main compressor, 35C, and bypass compressor, 88C, respectively, and the black lines show the inlet pressure, 8.69 MPa.

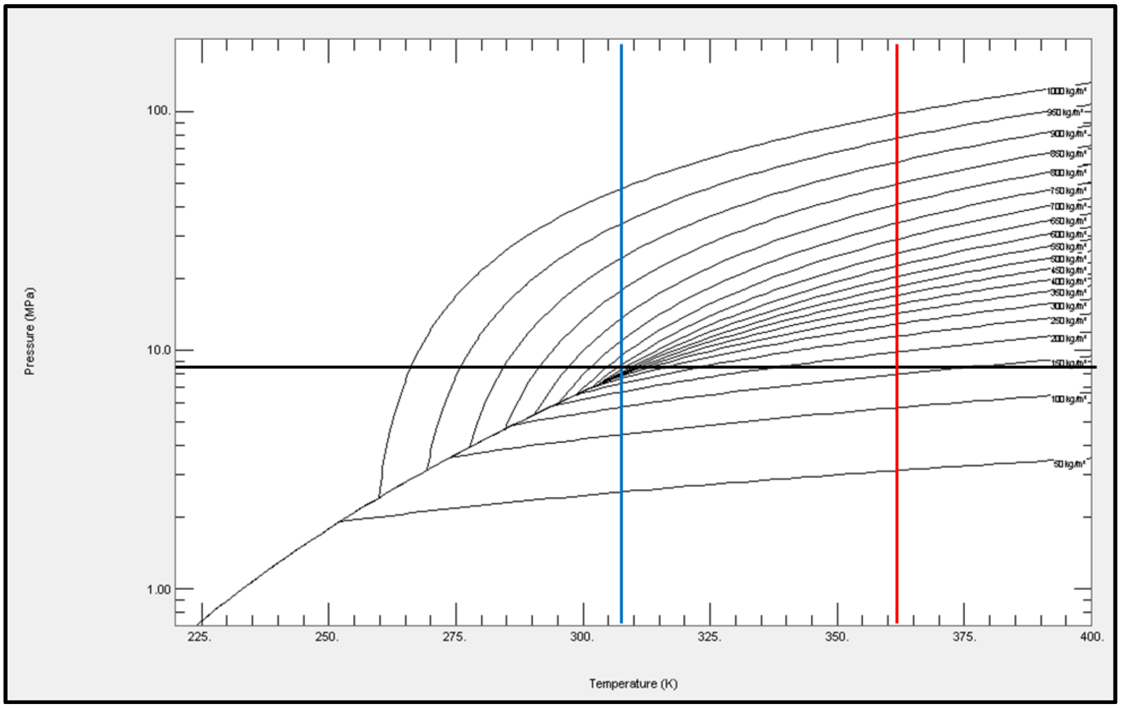

Figure 2 – CO_2 p-T Map with Density Lines [REFPROP]

With a variation of +/- 5°C at 8.55 MPa, the density of the fluid can range from 728 kg/m^3 to 365 kg/m^3, which will create a large variation in volume flow and will affect the compressor performance. The inlet conditions to the main compressor can vary based on the atmospheric temperature. It is important to operate the compressor as close to the dome as possible since this will reduce the power required to compress the flow. Nominal design conditions will be based on average operating temperatures, and off design conditions will have to be accounted for. While the bypass compressor will not see the density swing like the main compressor, it will be critical to maintain an exit pressure equal to the main compressor to avoid surge [3]. Since the compressor will be on a single train with the turbine, the best solution to maintaining high efficiency is utilizing variable IGVs [2].

Implementing a variable IGV system requires space inside the compressor, which is already limited. The rest of this paper will focus on the limitations in packaging of a compact sCO_2 compressor. While the focus of this paper is on compressor design, many of the design rules will also apply to turbine mechanical design in sCO_2 power cycles.

RESULTS AND DISCUSSION

When designing a compressor, it is important to understand the balance between three key disciplines: aerodynamics, rotordynamics, and mechanical design. Each has their own design goals, which will conflict with the goals of the other disciplines. It is true for all turbomachinery, but it is even more of a challenge in sCO_2 turbomachinery where space is extremely limited due to the small size of the flow paths. Figure [3] shows some of the important goals of each of the disciplines and how they relate to each other:

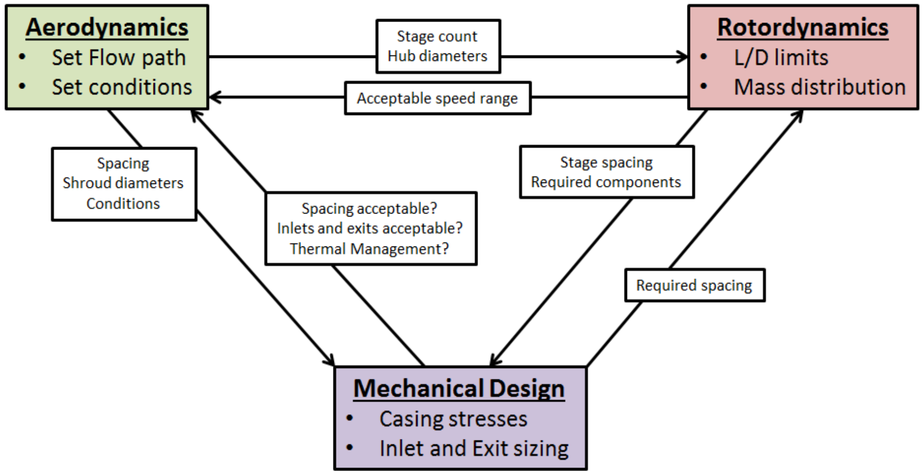

Figure 3 – Compressor Design Disciplines

Aerodynamics focuses on the flow path of the process fluid through the machine. This includes inlet and exit volutes, inlet guide vanes, compressor blades, and diffusers. In order to increase aerodynamic performance, the aero design will look to more stages, longer axial span, smaller hub diameters, and larger inlets.

Rotordynamics focuses on the overall stability of the shaft. This includes looking at bearing stiffness, damping coefficients from seals and dampers, and ensuring that the shaft has significant margin to critical modes at operating speeds to prevent damage to the rotor and the rest of the machine. To improve rotordynamic stability, rotor design will look to decrease the axial span, increase the hub diameters, and add axial space for damping features.

Mechanical design focuses on the stresses in the casing and the shaft. This includes hoop stresses in the casing, blade and hub stresses in the shaft due to rotation and temperature, fatigue and creep life, and the overall packaging of necessary components and features. To reduce stresses and improve the mechanical life of the machine, mechanical design leads to smaller inlets, longer axial stages, fewer stages, and smaller diameters.

As can be seen by the various design goals of each discipline, there are contradictions. Depending on the goal configuration of the system, certain design goals can be met more than others. For a system in which the main compressor and bypass compressor are not in a single casing, more stages could be designed for each compressor and the hub diameters could be decreased to improve the overall efficiency of each compressor. This would also mean fewer components to package and shorter axial span since only one compressor would have to be contained. From a design perspective, this is a much simpler solution; however, from an overall cost and foot print, this is not ideal. By having two separate compressor systems, this means two more skids, at least two more gearboxes, two more motors, and an additional high pressure casing. All of this leads to more piping, a larger required footprint, and will end up leading to higher costs and performance loss due to power required for additional bearings, gearboxes, and motors.

By combining the main and bypass compressor into a single casing and directly coupling it to the turbine, the overall footprint and cost of the system is decreased. The complexity of the compressor design is increased significantly, and the balance of the three disciplines will have to be considered even more. For this compressor design, the main goals will be:

1) Fewer compressor stages to reduce the overall axial span
2) Larger hub diameters that will meet bearing span to diameter goals set by rotordynamics
3) Pushing the rotordynamic limits by balancing stage count, axial span, and key diameters to meet the overall design goals set by the cycle

To begin the layout and design, it is important to know the necessary components inside the compressor. As with the 14 MW turbine designed under the Sunshot project, this compressor will run on integral squeeze film damper tilting pad bearing [4]. Because sCO_2 is the operating fluid, dry gas seals will be required to limit CO_2 losses. Multiple studies are looking at the application of CO_2 gas bearings that would remove the need for oil systems and dry gas seals, but that technology is not available yet at this power level. With the challenges of installing both compressors and actuated IGVs, it is important to use as much existing technology as possible to limit the risks. Thrust bearings and a balance piston will also be required to manage any axial loading acting on the rotor. While the nominal design will have balanced thrust between the two compressors, off design cases will have to be looked at to ensure that the thrust bearings will not be overloaded. This compressor will also require a coupling on the drive end and a balance drum on the opposite end. Figure [4] shows an example of a compressor layout:

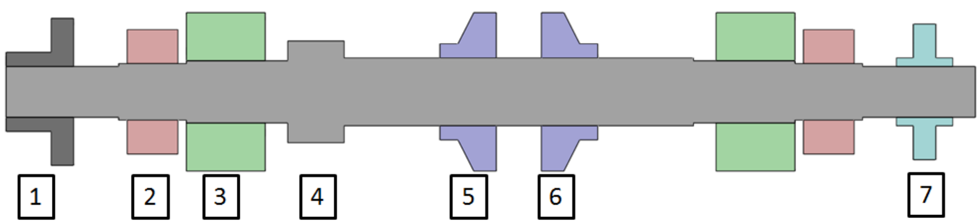

Figure 4 – Initial Apollo Rotor layout with necessary components

Standard locations for internal components:

1) Coupling
2) Journal bearings
3) Dry gas seals
4) Balance Piston
5) Main Compressor / Bypass Compressor
6) Bypass Compressor / Main Compressor
7) Thrust Collar

This compressor will be designed according to API 617 and ASME Boiler and Pressure Vessel Code Section VIII Division II (ASME VIII-2). API 617 covers many of the critical components on the shaft and stators and lists out the required analysis and studies to use for rotordynamics, mechanical design, and manufacturing. ASME VIII-2 is utilized to look at the pressure containment of the casing. Since the temperatures for this compressor are not pushing material limits, materials recommended by both of these codes will be used to limit additional design risk.

Component sizing starts with the furthest outboard component, which is the coupling on the drive end of the shaft, which will see the smallest diameter and also peak stresses from torsion. For assembly, it is important that the diameters step up from the coupling diameter to allow for ease of install and also prevent the damaging of critical surfaces. As mentioned before, one of the big advantages of sCO_2 is the compact flow design compared to more conventional power turbines. With its high density, the airfoils on the impellers can be made relatively small and reduce the overall size of the machine. However, the power generated is still the same, which means the shaft has to be able to handle the torque. Depending on the source of the torque, certain safety factors are required. If this were a motor driven compressor, it would need to account for start-up transients that could lead to torques 5-10X greater than the max continuous torque of the compressor [5]**.** Since this is driven by a turbine, the transient torsional stresses will be limited and safety factors can be limited to 5X to yield. From Ameridrive data, Figure [5] shows shaft speed and power vs shaft diameter:

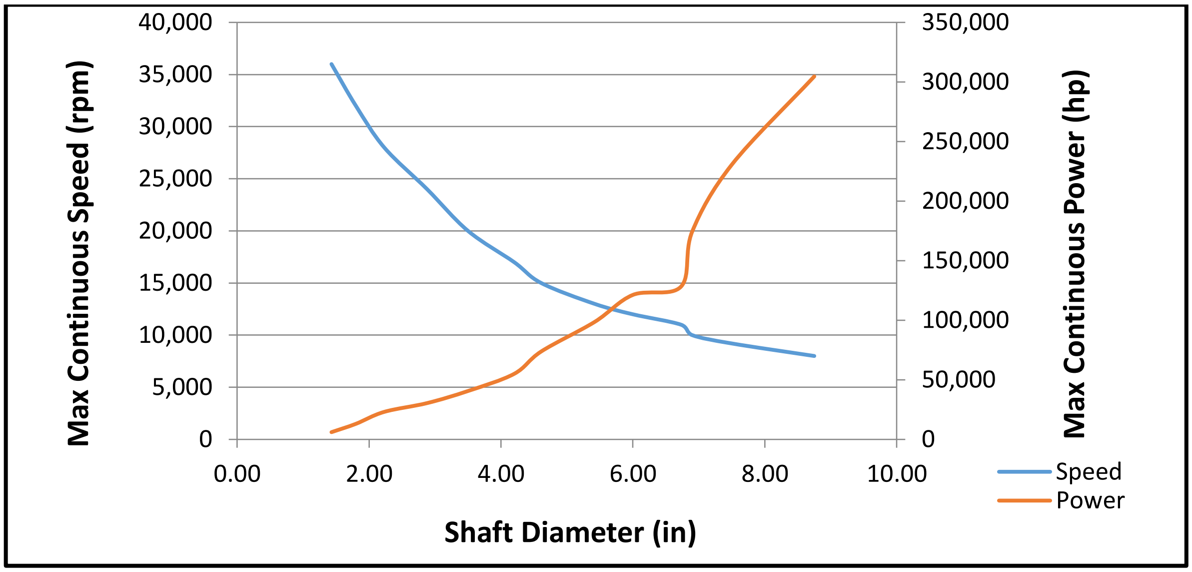

Figure 5 – Coupling Speed and Power Ratings - Ameridrive

For this compressor, the design power is at 4.9 MW (6,570 hp) at 28,350 rpm, 5% above operating speed of 27,000 rpm [4]. This limits the max coupling size to around 2.25 in, size 6135/8135 high performance disc coupling. A coupling of this size has a max continuous torque 42,000 in-lb, a max peak torque of 52,000 in-lb, and a max short circuit torque of 67,000 in-lb. The torque from this compressor:

$$T = 63{,}025\frac{P}{w} = 63{,}025\frac{6570}{27000} = 15{,}336 \qquad [1]$$

Where: T = Torque, in-lb; P = Power, hp; and w = Speed, rpm. Peak shear stress acting on the shaft is:

$$\tau = \frac{Tr}{J} = \frac{15336\ x\ 1.125}{2.52} = 6{,}857\ psi \qquad [2]$$

$$J = \frac{\pi r^4}{2} = 2.52\ in^4 \qquad [3]$$

Where: τ = Shear Stress, psi; r = radius, in; and J= Polar moment of inertia, in^4. Since this compressor will be driven by a turbine, the peak shear stress from torque is far below standard shaft material, AISI 4140, which has a yield stress of 95 ksi. In order to meet the 5X safety factor to yield, the minimum shaft diameter is 1.6” based on torque and maximum of 2.25” based on coupling speed limits. Due to torsional requirements for the turbine, a 2.25” coupling will be used to allow for adequate safety factor.

With a 2.25” coupling, the minimum bearing diameter is 2.5”. This is due to a required step up in diameter to allow for installation of the bearing. Without this step, the bearing could slide across the coupling surface and be damaged. In most cases, a minimum diameter step of 1/4” is required. With the small size of flow path and shaft, the bearings will not be seeing high loads due to the mass of the rotor like gas compressors.

After the coupling and bearings have been sized, the next step involves determining the hub diameter of the impellers. Since the impellers are in the middle of the shaft, the hub diameter will be critical in determining the max bearing span of the rotor. For compressors like this, a max length to bearing diameter ratio of 10 is recommended for conceptual design before a full rotordynamic analysis can be completed.

The minimum and maximum hub diameters are limited based on torsional and rotating stress requirements respectively. Earlier, it was determined that the minimum solid shaft diameter is 1.6”. If a built up shaft is chosen, a tie bolt is required to hold all of the joints together. The tie bolts diameter will be chosen based on pressure loads and rotational moments from the impellers. A larger tie bolt will lead to larger hub diameters to maintain minimum cross sectional areas.

For rotating stresses, the limit will be 50% of yield at the inner diameter. It is important to note that this is only looking at the peak stress in the hub. Finite element analysis (FEA) will be required when looking at peak stresses in the blade geometry. Localized peak stresses can be higher than yield. The peak stress at the ID of the impellers is:

$$\sigma = \frac{3+v}{8}\rho\omega^2\left[r^2 + 2R^2 - \frac{1+3v}{3+v}r^2\right] \qquad [4]$$

Where: σ is peak stress at the smallest diameter, psi; v is poisson’s ratio; ρ is material density, lbm/in^3; ω is rotating speed, rps; r is the inner diameter, in; and R is the outer diameter, in. At 450 rps (27,000 rpm), the max diameter is 6.26” for a solid shaft and 6.18” for a shaft with a 2” tie bolt. This outer diameter is an initial assumption on the max diameter of any impellers, balance pistons, and thrust collars. Since the impellers will have the blade geometry, their diameters can be larger. This is a hard limit for balance pistons and thrust collars since their geometry is a solid cylinder.

Aerodynamic detailed design will determine an appropriate hub diameter to impeller exit diameter ratio for peak performance. If this ratio is 0.5, the hub diameter would be around 3” which would allow for a bearing span of 30”, see Figure [6]:

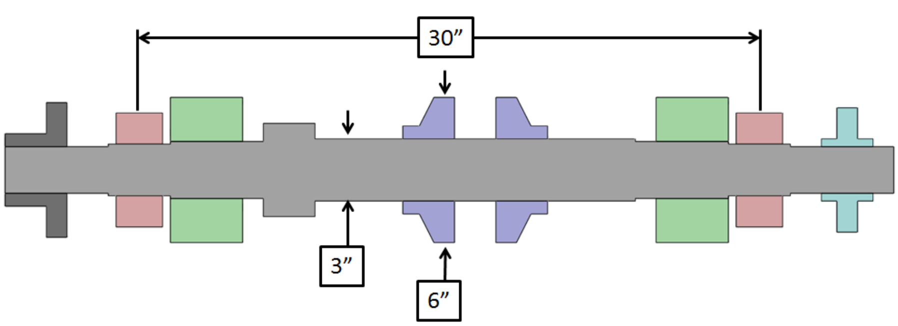

Figure 6 – Initial Apollo Rotor bearing span and hub diameters

Depending on the final tie bolt size, the minimum hub diameter will be determined by the torsional requirements. In Equation [2], replace J with:

$$J = \frac{\pi}{2}(R^2 - r^2) \quad [5]$$

Table 1 – Tie Bolt Diameter vs Minimum Hub Diameter

Tie Bolt Diameter	Min. Hub Diameter	Bearing Span
in	in	in
0.50	1.61	16.1
0.75	1.65	16.5
1.00	1.72	17.2
1.25	1.82	18.2
1.50	1.97	19.7
1.75	2.15	21.5
2.00	2.34	23.4

A larger tie bolt means higher clamp load which is good for keeping joints in contact. This also leads to longer bearing span, but it does limit the minimum hub diameter. Torsional requirements set the minimum shaft diameter and rotational stresses set the maximum diameter. Within these bearing spans of 16" to 30", the optimal design and layout for the Apollo compressor will be determined.

With minimum and maximum spans based on required shaft sizes known, the next step is to determine required axial spacing based on stator components. Stator components of concern are the pressure containing end caps, diaphragms, and inlet and exit plenums for both compressors. End caps are sized to ensure that they can contain the full design pressure of the compressor. To allow for off design performance, compressor casings are designed to contain 125% of max operating pressure. For the case of the Apollo compressor this means the case and end caps will be designed to meet ASME Section VIII Division 2 at 400F and 4,350 psi. For initial sizing and axial spacing requirements, the end caps can be designed based on simple hand calculations show in Figure [7]:

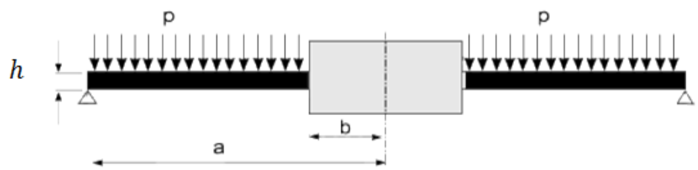

Figure 7 – Simply supported pressurized plate [6]

$$\sigma_{max} = \frac{k_1 P a^2}{h^2} \quad [6]$$

$$w_{max} = \frac{k_2 P a^4}{E h^3} \quad [7]$$

Table 2 – K factors for diaphragm thickness calculations

a/b	Simply Supported		Fixed Support	
	k1	k2	k1	k2
1.25	0.592	0.184	0.105	0.002
1.50	0.976	0.414	0.259	0.014
2.00	1.440	0.664	0.480	0.058
3.00	1.880	0.824	0.657	0.130
4.00	2.080	0.830	0.710	0.162
5.00	2.190	0.813	0.730	0.175

Where: σ_{max} is the peak stress in the diaphragm, psi; w_{max} is the max displacement at the ID of the diaphragm, in; a is the outer diameter, in; b is the inner diameter, in; P is the pressure, psi; h is the required thickness, in; E is modulus of elasticity, psi. σ_{max} is based on the allowable stress of the material at design temperature as specified by ASME Section II Part D. w_{max} is limited based on the maximum displacement allowed in the diffuser of the compressor. The goal is to keep this displacement at <10% of the diffuser width. End caps will usually be sized with stress as the limiting factor while diaphragms will be sized with displacement as the limiting factor. The pressure acting on the diaphragm is the pressure difference between inlet and exit of each compressor stage.

Analyzing the diaphragms and end caps based on the simply supported plate is a conservative approach. There are similar K factors for a fixed supported plate which are not conservative. Based on how the diaphragms are supported by adjacent parts, the actual displacement and stresses will fall in between simply supported and fixed support plates. Before a design can be finalized, the diaphragms and end caps must be confirmed with FEA according to ASME Section VIII Division 2. See Figure [8] for example of simply supported plate. Final geometry will require supporting features and rough contacts to accurately simulate how each diaphragm and housing is supported. End caps will be affected greatly by how the pressure is contained, whether it is bolts or shear faces.

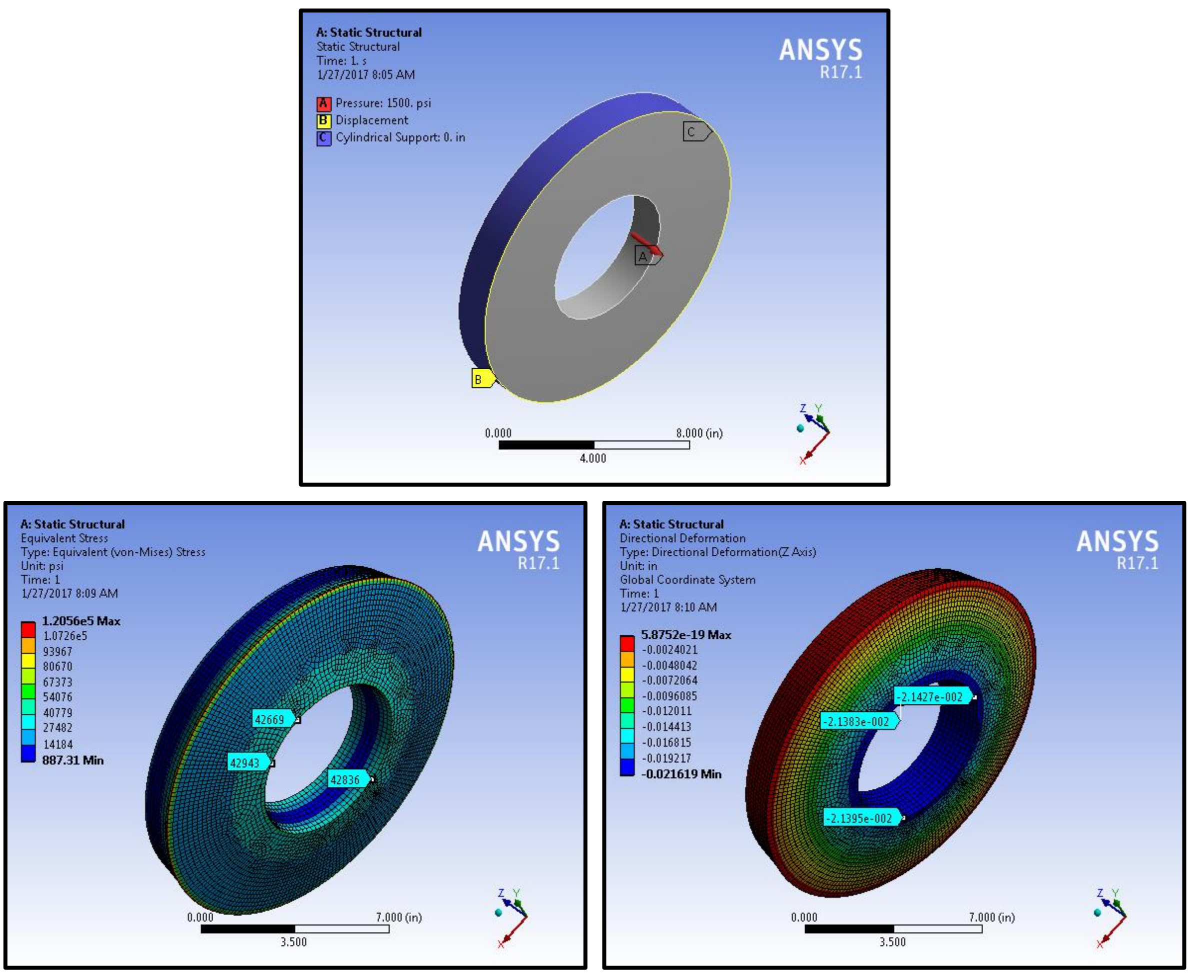

Figure 8 – Example FEA of simply supported plate

Based on these equations, it is seen that as the outer diameter (a) is increased, the required thickness to maintain the required stress and displacement must also increase. This will limit the maximum outer diameter of the compressor bundle based on allowable axial span. Larger outer diameters allow for more radial space, including: shear rings, seals, bundle bolts, inlet and exit volutes, etc. With the high speeds and small diameters, axial space is limited for the internal components required. This leads to packaging constraints and design exceptions that have to be made. Figure [9] shows an example compressor layout with diaphragms, end caps, and outer casing:

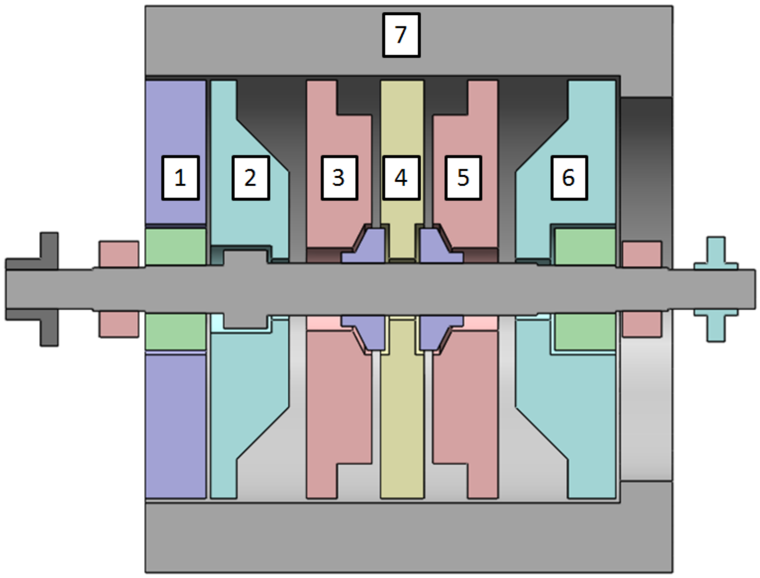

Figure 9 – Compressor layout with stator components

Items (1) and (6) are the two end caps. These will be designed to contain the full design pressure of the compressor. The axial thickness at the OD is h in the stress and displacement calculation. Items (2) and (6) contain the inlet for the respective compressors. One houses the dry gas seal and the other houses the balance piston seal. This seal housing will require pressure references to the exit flow to allow for a pressure differential across the seal which will allow for damping and thrust control. Items (3) and (5) are the main and re compressor diaphragms. These will house the exit plenums and also the actuated IGV system. Item (4) is the division wall between both compressors. This will house a seal and should see zero pressure load. Item (7) is the outer casing. For a larger machine, this would be an axial split casing. For a smaller compressor like this, a sold barrel is preferred. The internal bundle containing the diaphragms and rotor should be able to be removed while the outer casing is installed on the operating skid.

For overall bearing span, the key features in addition to required diaphragm thicknesses will be in the inlet and exit flow paths. In the bundle, these flow paths will be scheduled volutes and plenums, but coming into the case will be standard piping connections, and to get to the bundle will require circular holes through the wall. The size of these holes is based on flow velocity limitations. ASME B31.1, Table IV-5.2 lists higher erosion rates for flows over 10 ft/s for water and 150 ft/s for steam. Based on density, sCO_2 is between water and steam, which leads to an average flow velocity of 80 ft/s and can be used for initial sizing of inlet and exit piping. Lower velocities also reduce pressure losses. There is a balance between piping size, cost, and pressure loss. For longer lengths of pipe, the pressure loss is more significant, and for shorter lengths in the case of compressor inlet and exit nozzles, the pressure loss can be increased without affecting the overall pressure drop of the system significantly. Table [3] shows recommended flow path sizes for the inlet and exits of the main and bypass compressor. Table [4] show axial spans for diaphragms and end caps based on the outer diameter:

Table 3 – Inlet and Exit diameters based on maximum flow velocity

Section		Pressure		Temperature		Density	Mass Flow		Max Vel.	Min Dia.
		Mpa	psi	C	F	lbm/in^3	kg/s	lbm/s	ft/s	in
Main	Inlet	8.55	1,240	35	95	0.0224	70.3	155.0	80	3.03
	Exit	24.13	3,500	78	172	0.0247	70.3	155.0	80	2.88
Bypass	Inlet	8.69	1,260	88	190	0.0061	34.2	75.4	80	4.04
	Exit	23.99	3,479	194	381	0.0116	34.2	75.4	80	2.94
Main	Inlet	8.55	1,240	35	95	0.0224	70.3	155.0	150	2.21
	Exit	24.13	3,500	78	172	0.0247	70.3	155.0	150	2.11
Bypass	Inlet	8.69	1,260	88	190	0.0061	34.2	75.4	150	2.95
	Exit	23.99	3,479	194	381	0.0116	34.2	75.4	150	2.15

Table 4 – Diaphragm and end cap axial span based on outer diameter of bundle

OD	Material (Allowable Stress)			
in	20 ksi	30 ksi	40 ksi	50 ksi
10	11.13	9.15	7.98	7.21
12	14.92	12.27	10.69	9.77
14	18.54	15.25	13.29	12.27
16	22.06	18.15	15.81	14.72
18	25.54	21.01	18.36	17.16
20	29.01	23.87	21.00	19.63
22	32.49	26.73	23.67	22.13
24	35.89	29.53	26.33	24.63
26	39.06	32.14	28.85	26.99
28	41.67	34.29	30.96	28.96
30	43.15	35.50	32.09	29.98

Calculations for bundle span are based on the pressure difference between inlet and exit on items (3) and (5) in Figure [9], all end caps, items (1) and (6), containing the full design pressure of the compressor casing, the division wall, item (5), containing the worst case pressure difference between main and bypass compressor, and the balance piston diaphragm, item (2), which will see the pressure difference between inlet and exit. Bearing span must include axial spacing for the inlet and exit nozzles, the required thicknesses for all the diaphragms, axial allowance for the bearings, dry gas seals, and the balance piston. The bearing span is roughly equivalent to the bearings diameter, 2.5”, and axial span for the balance piston diaphragm will be ignored since it can be included with the inlet or end cap. Dry gas seals are also contained within the end caps since this helps reduce space significantly. In a machine with more axial allowance, the dry gas seals could have their own housing, but that would require roughly 25% of max allowable axial span for this compressor.

If designed to 80 ft/s flow velocity, the required span for flow paths is 12.9” and another 2.5” for the bearings, allows 14.6” for diaphragm axial span. This limits the maximum OD to around 16” with 50 ksi allowable material. With a design velocity of 150 ft/s, the required flow path span is 9.4” and allows for 18.1” for diaphragm axial span. This allows for up to 18” OD with 50 ksi allowable material. Realistically, any pressure containing material will not have an allowable stress greater than 30 ksi, which limits this design to 16” bundle OD, a 30” bearing span, and a hub diameter of 3.00”. Figure [10] shows the relative size and layout for this conceptual compressor layout:

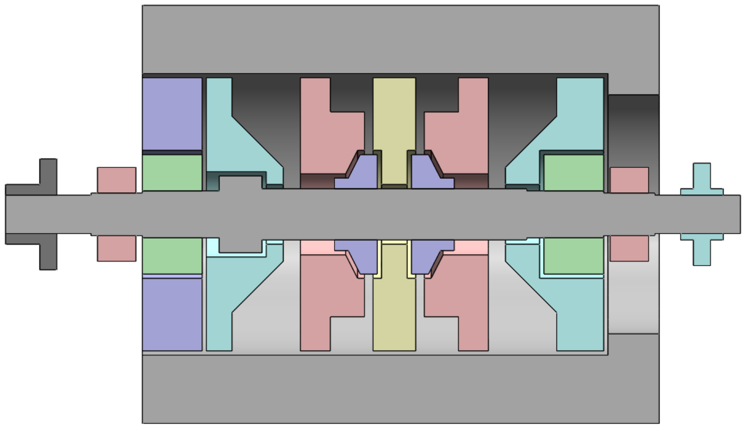

Figure 10 – Conceptual compressor layout for Apollo

With the overall design envelope established, packaging of key internal components can begin. Current axial spacing already allows for bearings, dry gas seals, diaphragms, inlets, exits, and pressure containment. The last component, mainly for this design application, is the variable IGVs. As mentioned, earlier, variable IGVs will be required for the main and bypass compressor. Looking back at figure [9], the variable IGVs will need to be packaged either in item (2) or (3) for the main compressor and item (5) or (6) for the bypass compressor.

Based on the diaphragm thickness calculations for the main and bypass compressor, the required thickness for each diaphragm is 3.00”, items (2), (3), and (5), and the end cap is 3.88”, item (6). Ideally, the actuator system could be placed in the end caps, where more room in available, but due to requirements for balance pistons, dry gas seals, and inlet plenums, the actual available space is much less. Balance pistons require flow capacity for the supply CO_2. Dry gas seals require porting for buffer air, venting, and supply CO_2. Inlet plenums require more flow area due to the lower pressures. This means the best location for actuator systems is in the compressor diaphragms between inlet and exit. The less space the actuator system requires, the more stages that can be added to increase compressor efficiency. If a single actuated vane system requires the full span of the diaphragm, this means each compressor can only have a single stage. If the span is half the required thickness, a 2nd stage could be packaged for each section.

There are two options for actuated inlet guide vanes: axial or radial as shown in Figure [11] and [12] respectively:

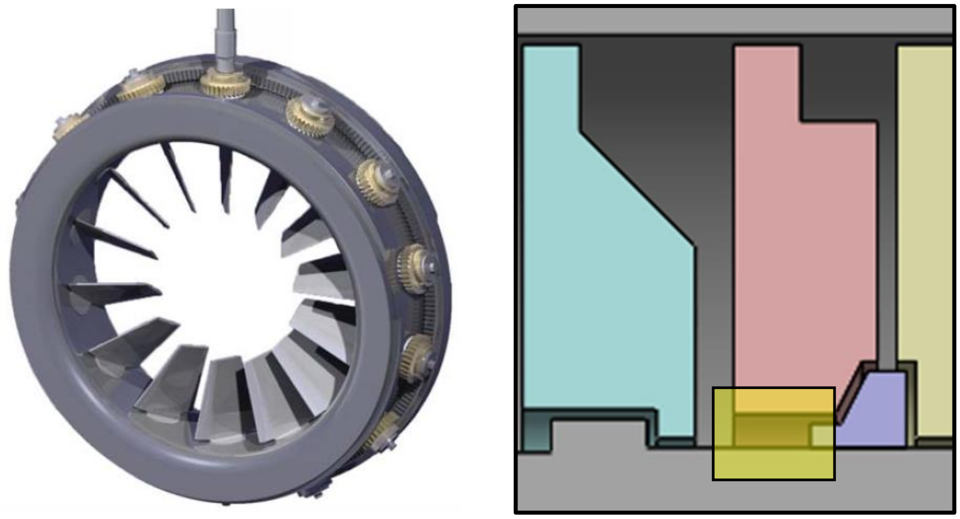

Figure 11 – Axial IGVs for a radial compressor [7]

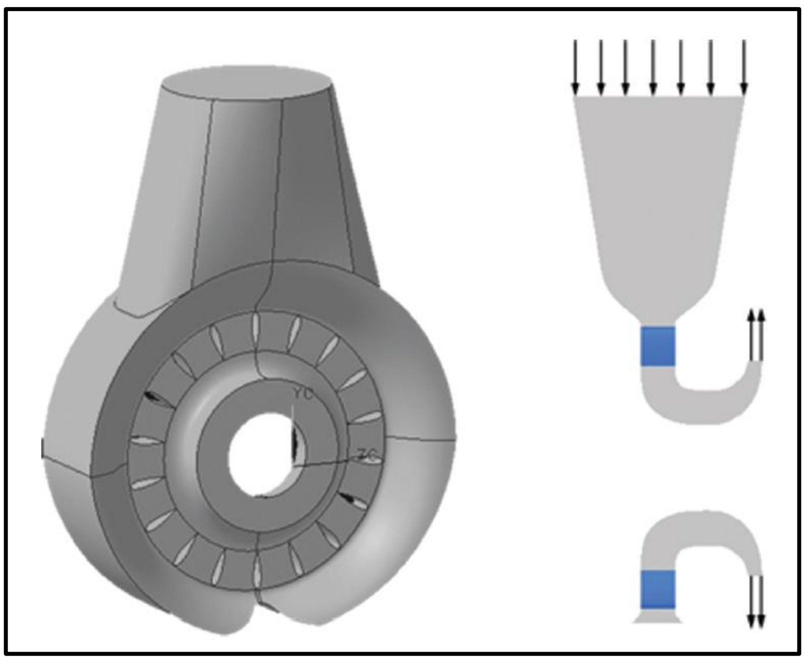

Figure 12 – Radial IGVs for a radial compressor [8]

Based on rotor layout, each IGV has its own advantage. For an overhung compressor where axial space on the stators at the end of the shaft doesn't affect rotor spans, axial IGVs would be used. They are also used in axial compressors where less axial space is required to contain an actuated IGV compared for a fixed IGV. When axial space is limited between the bearings, radial IGVs will be preferred. An actuated radial IGV system will require an external actuator that will penetrate through the pressure boundary of the case and connect with a pivot or gear system that can use a tangential load to rotate the IGVs radially. If a pivot system is use, each joint requires bearings. Operating in a CO_2 environment with limited space, the bearings will have to be unlubricated and will need to handle the loading from the actuator and the aerodynamic loading from the IGVs. The max load will be calculated based on the largest angle from the radial line to the center of the axis. Figure [13] shows the difference between incidence angles on an IGV:

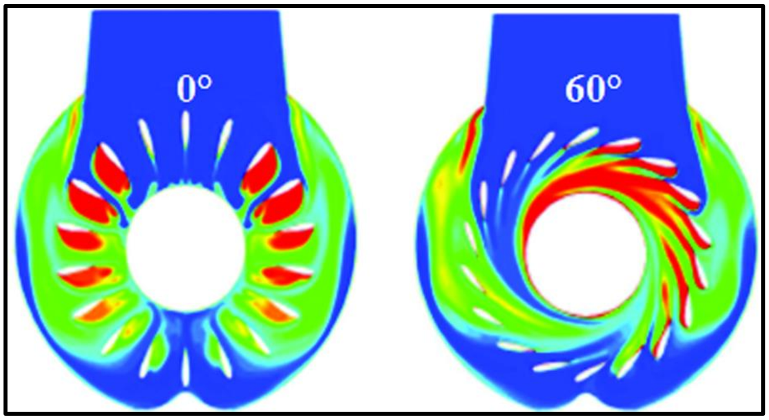

Figure 13 – Perfectly Radial IGVs vs Large Incidence Angle IGVs [8]

Based on the above image, the aerodynamic force acting on the 60° case would produce a much larger load on the IGV. Figure [14] shows how these aerodynamic loads create a total load on the actuator:

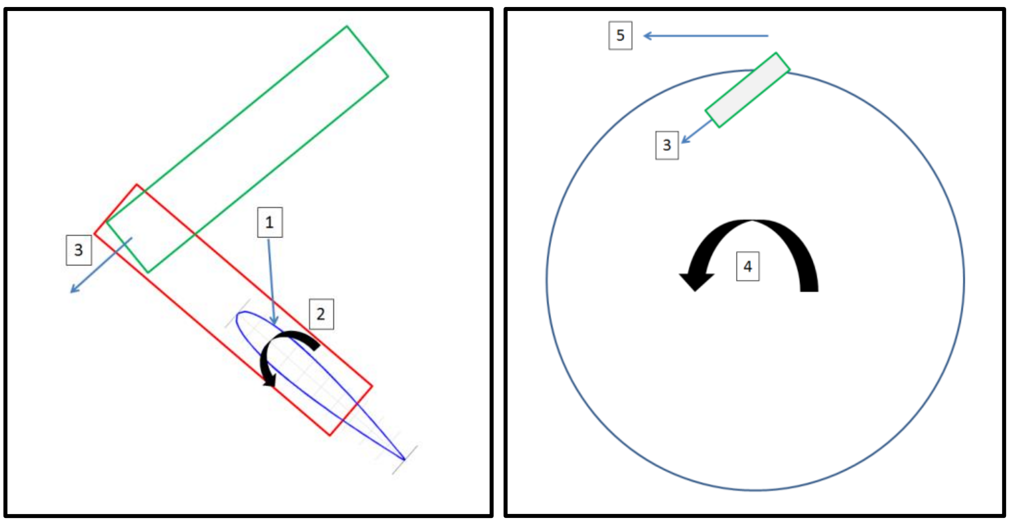

Figure 14 – Aerodynamic Loads acting on Actuated IGVs

The aerodynamic load (1) will create a torque (2) that will cause a load between the two pivot joints (3). This load is then transmitted to a rotating ring that will contain all of the pivot joints and create a torque (4) that will lead to a combined axial load on the actuator (5) from all of the IGVs. Since lubricated bearings will not work in this environment, non-lubricated bearings will be used and they will have a higher friction coefficient that leads to a larger load required by the actuator to not only maintain position but also rotate the IGVs during operation. This system will require one or two bearings at each joint based on the loads that are acting. One of the smallest bearing available is a ¼" shaft teflon sleeve bearing. These bearings can only handle around 21 lbs. As the shaft diameter steps up, so does the length of the bearing. Length and diameter are key in determining how much axial and radial space is required to house the actuator. With a 3.00" axial span and max OD of 16", the actuator envelope must be smaller, while also allowing space for seals, bolts, and assembly/disassembly features. Figure [15] shows how the actuated system will affect the compressor diaphragm and how much space is required. This is assuming ¼" bearings:

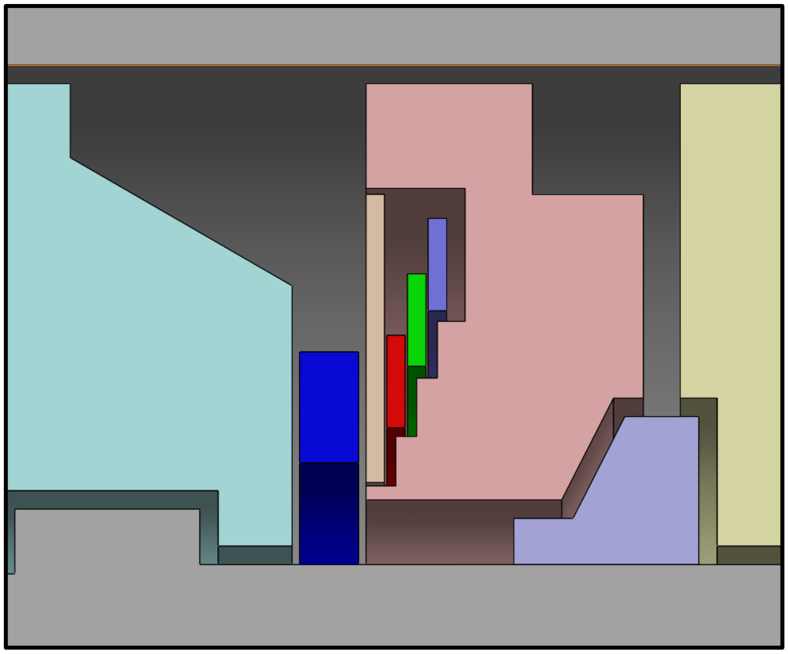

Figure 15 – Actuated IGV system design envelope in compressor diaphragm

Conclusion

It is important to note that all of the calculations presented in this paper represent a conservative approach to performing initial sizing calculations on a compressor. All of the mechanical calculations are based on uniform cross sections. With the amount of features required in the diaphragms for actuators and flow paths, they will not end up having uniform shapes. Complicated geometry will require FEAs to determine if stress and displacement limits are met. The basic rules do apply in that larger diameters require thicker diaphragms, increase stresses, and require more bearing span. This process is very helpful when needing quick answers to establish an overall build envelope that can be used to establishing rotordynamic and aerodynamic design limits. From detailed analysis, the bearing span of the rotor could be increased or decreased based on actual modes. It could also lead to a more aggressive design and thinner diaphragms to allow for more space for extra stages or increased flow paths.

Full analysis is required for API 617 and ASME Section VIII Division 2 to complete the design. The rest of the design process will be focused on the rest of the packaging details. This includes: lubrication systems, dry gas seal porting, balance piston porting, bolting, seals, and instrumentation. All of this will require axial and radial space and it can affect the design greatly if it is not account for early on. Within the limits of this particular design application, it is important to utilize available space for conceptual design to limit any design changes that could affect the overall performance. Exceptions have to be made, and it is important to understand the balance between rotordynamics, aerodynamics, and mechanical design in order to complete a design that can meet the minimum established requirements of each discipline.

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ACKNOWLEDGEMENTS

This material is based upon work supported by the Department of Energy, Office of Energy Efficiency and Renewable Energy (EERE), under Award Number DE-EE0007109. This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.