DISCLAIMER

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Recipient Name: Southwest Research Institute

Final Technical Report (FTR)

Cover Page

a. Federal Agency	Department of Energy	
b. Award Number	DE-EE0009813	
c. Project Title	*Characterization of Inlet Guide Vane Performance for Discharge Compressor Operation near the Dome of an sCO_2 Pumped Heat Energy Storage*	
d. Recipient Organization	Southwest Research Institute	
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f. *Principal Investigator (PI)*	Jason C. Wilkes, Ph.D. Senior Manager – R&D jason.wilkes@swri.org 210.522.5130	
g. *Business Contact (BC)*	Liz Garcia Specialist, Contracts liz.garcia@swri.org 210.522.3179	
h. *Certifying Official (if different from the PI or BC)*	Same as PI	

Signature of Certifying Official — December 29,2025

Date

By signing this report, I certify to the best of my knowledge and belief that the report is true, complete, and accurate. I am aware that any false, fictitious, or fraudulent information, misrepresentations, half-truths, or the omission of any material fact, may subject me to criminal, civil or administrative penalties for fraud, false statements, false claims or otherwise. (U.S. Code Title 18, Section 1001, Section 287 and Title 31, Sections 3729-3730). I further understand and agree that the information contained in this report are material to Federal agency's funding decisions and I have any ongoing responsibility to promptly update the report within the time frames stated in the terms and conditions of the above referenced Award, to ensure that my responses remain accurate and complete.

1. Acknowledgement:

"This material is based upon work supported by the U.S. Department of Energy's Office of Energy Efficiency and Renewable Energy (EERE) Solar Energy Technologies Office (SETO) under the [DE-FOA-0002378] Award Number(s) [DE-EE0009813]."

2. Disclaimer:

3. Executive Summary:

Southwest Research Institute® (SwRI®) developed and tested a Variable Inlet Guide Vane (IGV herein) assembly on an integrally-geared sCO_2 compressor (IGC) to demonstrate compressor operation at both the compressor design point and near the dome and to define the operating limits of the compressor by monitoring for two-phase flow, flow turbulence from the IGVs, and compressor choke and surge as the CO_2 inlet temperature is varied. Performance testing was conducted on an existing integrally-geared, two-stage main compressor designed for near-critical-point operation with CO_2. This testing campaign validated the IGV design and operation, as well as improved the understanding and confidence in operating compressors and predicting performance characteristics near the critical point where fluid properties change rapidly with temperature. In addition to improving the robust operating limits of an sCO_2 compressor, the development of an IGV for the IGC system improved off-design compressor efficiency by 12%.

4. Table of Contents:

Contents

5. Background:

The nation's energy landscape is evolving, and will require exponentially more energy storage in the near future to achieve zero-carbon by 2035 [1]. While numerous proposed technologies exist, pumped thermal energy storage (PTES) shows promise to achieve a good balance of cost and flexibility while remaining agnostic to location. These technologies operate by charging a thermal reservoir using a heat pump when energy is cheap, and discharging the reservoir when energy is expensive. Among the proposed technologies for PTES, researchers are currently considering the use of air and sCO_2 as potential working fluids for these cycles. Air side equipment has a similar drawback to steam cycles, leading to low density fluid in parts of the cycle that require large machinery and associated piping that drive up cost, while sCO_2 systems are generally more compact, which may allow for a reduction in cost and footprint. Among the leading energy storage technologies, PTES is fast approaching commercialization with few technology gaps to still prove-out. One such gap that has not been fully explored is the application of IGV's on the discharge compressor of the system. This IGV is needed to accommodate variations in volume flow of the compressor due to inlet temperature changes, partial load operation, startup, and shutdown. While the conditions upstream of the discharge compressor are very near the dome similar to the conditions at the inlet of an air cooled CSP plant utilizing sCO_2 as the working fluid, an IGV has not been vetted for an axial flow machine at these pressures near the dome.

Proposed Technology

Variable IGVs allow for a wide compressor operating range and enhanced compressor performance by adjusting compressor inlet pre-swirl. During the DOE-EE0007114, a detailed IGV design was created to match the APOLLO compressor, but was never manufactured as it was not included in the scope of that project. This IGV mechanism uses a center body and annular vanes as shown in a cross section in Figure 1.

As shown in Figure 1, a center body is suspended in the center of the flow by struts. The center body creates an annular passage that contains the vanes that serve to turn the flow. The center body was manufactured with integral struts using additive manufacturing to reduce overall cost and lead time. The vanes are turned by a linkage system with actuator as shown in Figure 1. This figure also shows an external view of the IGV assembly and a cross section with the IGVs in the opened and closed positions. By adding positive or negative pre-swirl to the compressor inlet, IGVs can greatly expand the compressor operating range, which is critical for a PTES that is subject to variable discharge loads and ambient temperatures.

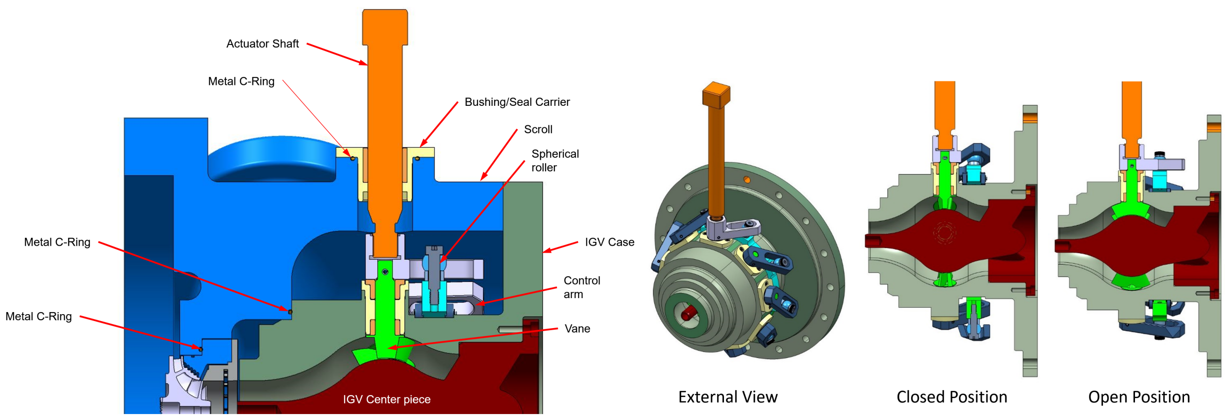


Figure 1. IGV Cross-Section Showing the IGV Assembly and First Stage Compressor (left); External View of the IGV along with a Cross Section Showing Opened and Closed Positions (right)

During testing, a range of inlet conditions was considered to provide feedback on the IGV performance as well as provide validation data to allow for optimization modeling for an sCO_2 PTES system. One of the challenges with modeling the performance of an air-cooled sCO_2 loop is that there will be potentially wide variations in the compressor inlet conditions. As the inlet conditions vary near the critical point, the performance of the compressor may vary dramatically. The performance of the compressor is a driving factor in the cycle performance that sets the flow and high side pressure of the cycle.

Variable IGV's provide necessary bulk flow control required by the cycle to respond to changes in heat load and cooling capacity. IGV's also change the aerodynamic characteristic of the flow at the inlet to the compressor, where the fluid is nearest the critical point and phase change is most likely. Detailed testing demonstrates the ability of the IGV's to deliver the expected cycle control and evaluate the interaction between the flow at the discharge of the IGV and the development of multiphase flow at the inlet to the compressor.

Utilizing the APOLLO Compander that is currently being tested at SwRI, provides a unique opportunity to test IGV performance in an sCO_2 environment at realistic operating conditions. This demonstrates the following:

1. How close to the dome the compressor can operate without subjecting the compressor to two-phase operation.
2. The mechanical reliability of the IGV assembly and its sealing capability would be verified.
3. Assess the impact of turbulence from the IGV has the potential to create unsteady flows in the compressor; thus, altering performance or life since this is one of the highest density IGVs ever tested in an axial compressor inlet.
4. How compressor range varies (choke-surge) as a function of compressor inlet temperature.

6. Project Objectives:

The objective of this project is to fabricate and install a variable IGV assembly on the inlet of an integrally-geared sCO_2 compressor. The testing campaign was conducted using sCO_2 at full process conditions to demonstrate how close to the dome the compressor can operate without subjecting the compressor to two-phase operation. The test campaign included a sweep of operating conditions to map compressor range (choke to surge) as a function of compressor inlet temperature. While testing, the project team evaluated the aerodynamic and mechanical performance of the IGV assembly. Furthermore, the collected data could be used to define a compressor control scheme to optimize compressor performance at given inlet conditions as well as monitor and control for surge.

Although an IGC is well suited to comprise an entire charge or discharge side of the sCO_2 system with increased efficiency, only the discharge compressor was tested in the current work. This advances the high-pressure sCO_2 IGV immediately upstream of a near-dome compressor from a technology readiness level (TRL) of approximately 5 to 8 and will provide confidence that placing IGVs immediately upstream of the compressor achieves acceptable performance across the range of inlet pressures and flows.

7. Project Results and Discussion:

Since the end of the DOE compressor testing campaign EE7114, the compressor was tested in a laboratory environment with additional instrumentation, beyond what is required by PTC-10, to minimize the uncertainty in the measured performance. Much of this material stems from a paper co-authored by the project team and presented at IGTI Turbo Expo 2021. Complete constant speed characteristics were collected at multiple supercritical points, operating at constant inlet conditions for each speed line covering a range of compressor inlet densities from 400 to 600 kg/m^3. Variations in the compressor stage efficiency and choke margin were observed, and the overall operability and stability of the compressor in response to changes in operating condition were also monitored. The following paragraphs are taken from work summarizing the testing performed commercially. This was presented at ASME Turbo Expo [5]. The compressor was shown to have excellent performance that closely matched the original design prediction. The performance at various inlet conditions showed minimal change in isentropic head coefficient at the design flow but did show some variation in efficiency and choke margin across the map. These changes in performance were observed to be minimal and did not affect the stable operation of the compressor. The results demonstrate that a commercial scale sCO_2 compressor can operate near the critical point and achieve the high levels of performance and stability required for power generation applications. The testing performed during the remainder of this test campaign will leverage the instrumentation added for this test campaign.

Instrumentation

The loop was originally instrumented following the basic specifications defined by ASME PTC-10 [18]. In addition to basic temperature and pressure measurements at the inlet and discharge of the stage, a Coriolis meter was installed at the inlet, and the motor power was measured on the VFD output. A Coriolis meter was added so that the compressor inlet state could be defined based on inlet pressure and density, which allows for much less uncertainty in performance calculations when compared to performance calculated using measured temperature and pressure, as suggested by [19], [20], and [21]. Additional flow measurements were conducted using an orifice flow meter on the main cooler inlet so its pressure drop would not add to the risk of producing liquids, and an annubar flow meter was added to measure stage 2 inlet flow. The addition of the inlet density measurement allowed for the inlet state to be resolved with greater accuracy using density and pressure as state points than could be achieved with the inlet temperature and pressure data. This is particularly true at 33-40 °C, where slight variations in temperature have a great influence on density, enthalpy, and thus gas power. Finally, the motor power was used in conjunction with lube oil temperature rise and lube oil flow to determine the overall gas power.

Test Procedure

The compressor inlet was operated at a range of supercritical inlet conditions with densities up to 600 kg/m^3. Figure 2 shows the compressor inlet conditions, on a P-d diagram, where steady performance measurements were taken. The design inlet condition is shown as a green triangle, at 85 barA and 37 °C corresponding to an inlet density of 512 kg/m^3. A full compressor map was measured at this point, as well as at 37 °C with higher and lower pressure resulting in inlet densities near 450 kg/m^3 and as high as 600 kg/m^3. Additional design head and flow coefficient test data were taken in the vicinity of the critical point and are indicated by purple circles.

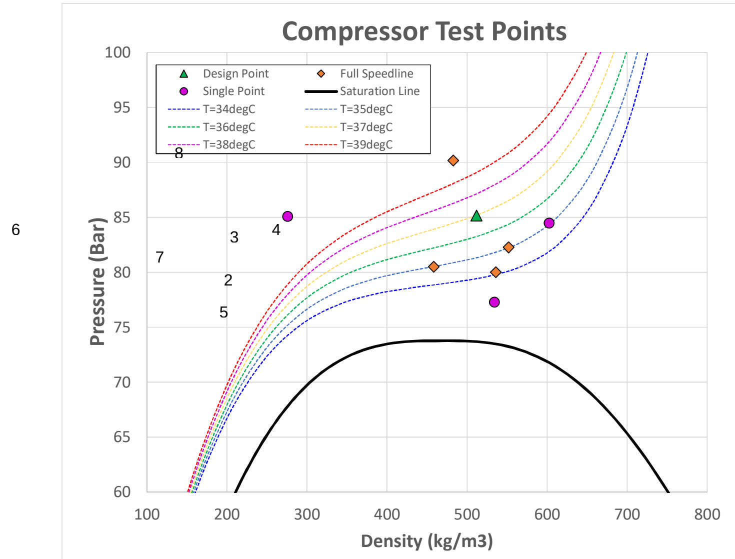


Figure 2. Compressor Inlet Conditions at Test Points

The stage performance was calculated with the inlet state defined from measured inlet pressure and density and the discharge state based on the measured temperature and pressure. NIST REFPROP version 10 [22] was used to calculate gas properties which uses the Span Wagner equation of state [23].

Previous Test Results

A full performance map, from choke to stall was collected at several different super-critical inlet conditions for a sCO_2 compressor designed for a 10MWe recompression Brayton cycle. In a machine operating with an ideal gas, the stage performance would be expected to collapse onto a single characteristic isentropic head and efficiency curve for the different inlet conditions. In this case, the real gas characteristics of the fluid are very pronounced near the design inlet conditions, since they are close to the critical point. The variability in gas properties means that complete aerodynamic similarity cannot be maintained and some deviation in the stage performance can be expected - and was observed - as the inlet conditions changed. The most significant variation in performance was found to be a reduction in calculated stage efficiency as the inlet conditions approached the saturation line.

Table 1 shows some of the test conditions taken at a constant inlet flow coefficient. The test conditions are all supercritical and vary in inlet temperature from 32-46 °C and inlet pressure from 80 to 90 barA. A reduction in inlet pressure and temperature of just 2 °C and 1 barA (Point 3) gives a reduction in Machine Mach number of 13% relative to the design point. The compressor was operated at a constant tip speed of 167 m/s since, in application, the unit operates at a constant speed with the compressors and expander stages coupled through the main bull gear. Therefore, the performance of the stage at a constant speed, but with variable inlet conditions best matches the expected operating state.

Table 1. Compressor Test Conditions

Test Point	Inlet Temp (degC)	Inlet Pressure (BarA)	Machine Mach Number	Inlet Density (kg/m3)	Reynolds No.
Design	37.0	85.2	0.84	612.9	1.42E+08
Cond. 2	34.1	80.0	0.87	629.6	1.41E+08
Cond. 3	35.0	82.3	0.82	642.1	1.40E+08
Cond. 7	35.0	80.5	0.92	569.3	1.44E+08
Cond. 8	40.2	90.2	0.81	591.0	1.42E+08
Cond. 4	35.0	84.5	0.73	680.4	1.35E+08
Cond. 5	32.8	77.3	0.92	622.7	1.41E+08
Cond. 6	45.8	85.1	0.78	398.2	1.25E+08

Test data is compared at the different operating conditions in non-dimensional maps of stage isentropic head coefficient and efficiency vs flow coefficient, where the stage flow coefficient is defined as:

$$\emptyset = \frac{Q}{\frac{\pi}{4}\times D_2^2\times U_2} \tag{1}$$

and isentropic head coefficient as:

$$\psi = \frac{\Delta h_s}{{}^1/_2 U_2^2} \tag{2}$$

At the design flow rate, the isentropic head coefficient is nearly constant as the inlet conditions change, Figure 3. The most pronounced variation in the head rise characteristic is evident at choke. The four test conditions presented in Figure 3 show that at certain operating conditions, the stage begins to choke sooner than others. The earlier onset of choke was expected based on the variation in the speed of sound at the different inlet operating conditions, Table 1. Test condition #2, in red, is operating at a Machine Mach number of 0.87, relative to the design condition map, in green, at a Machine Mach number of 0.82. Where Machine Mach number is defined as:

$$M_{U2} = \frac{U_2}{a_{00}} \tag{3}$$

In all cases, the shaft speed of the stage remained constant and only the sonic velocity of the gas was changed. Interestingly, condition #7 had the highest Machine Mach number at 0.92, but a similar choke point to the design condition which was at a lower Machine Mach number of 0.84.

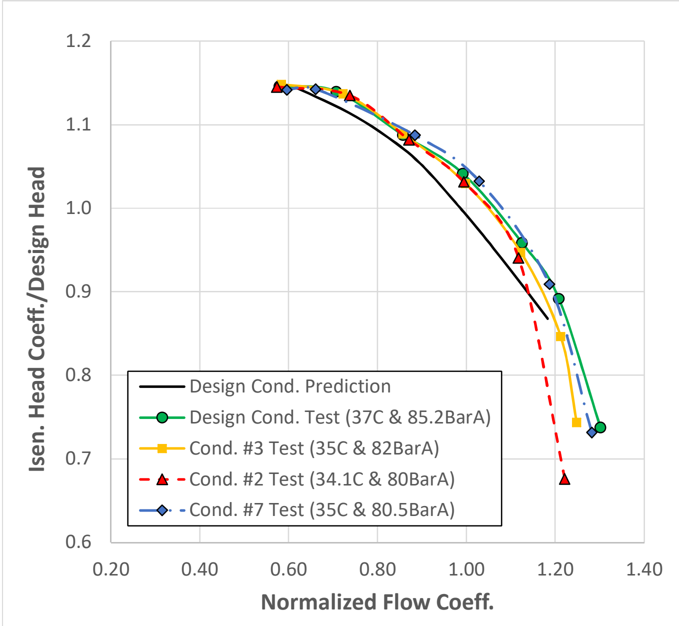


Figure 3. Compressor Isentropic Head/Flow Curve

The stage efficiency characteristic, Figure 4, also varies with inlet condition. The peak efficiency is reduced and shifts to lower flow coefficient as the inlet conditions are closer to the critical point. At lower flow rates, around 60% of the design flow, the efficiency for the maps at the different inlet conditions collapse to a nearly common value. The drop off in efficiency is most extreme at high flow, a flow rate in excess of 120% of design, in correlation with the onset of choke seen in Figure 3. This suggests that at higher flow rates where there is a greater suppression in the static conditions of the working fluid due to the higher fluid velocity that local regions of multi-phase flow may be developing and disrupting the gas path. The potential for the development of liquid formation at the inlet to this impeller was demonstrated numerically in advance of testing and reported separately [17]. This effect is more pronounced as the total inlet conditions are closer to the saturation line, which may explain why these operating conditions have a lower overall efficiency relative to those conditions further from the dome.

The casing treatment also has some effect on the efficiency characteristic of the stage. In this case, the casing treatment was optimized for the design inlet conditions of 37 °C and 85.2 barA. Typically, the performance effects would remain consistent as operating conditions change, and the stage characteristics would be unchanged. In this case, since the gas properties change so dramatically near the design point, it is possible that the casing treatment slot design becomes mismatched. This mismatch may contribute to some of the variation in the efficiency trend that was observed.

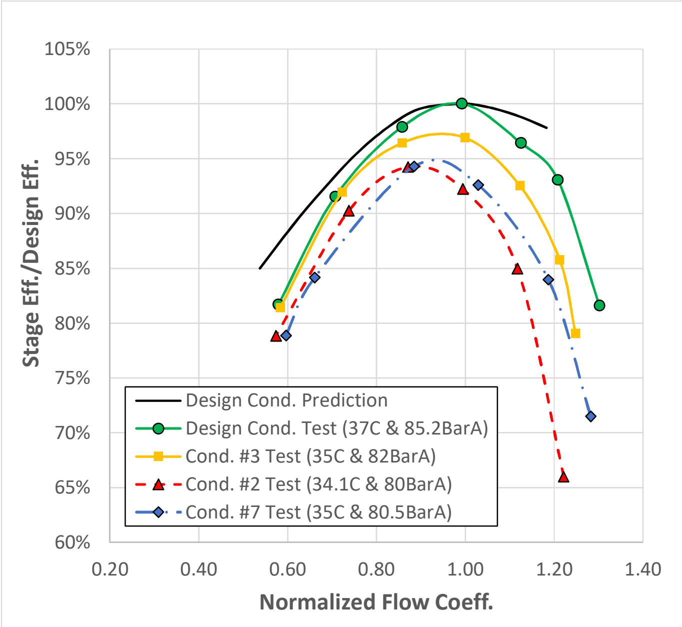


Figure 4. Compressor Efficiency at Various Operating Conditions

Figure 5 shows the stage efficiency at the design flow coefficient for a range of different supercritical inlet conditions. The testing shows that the stage is most efficient at inlet pressures of 85 barA or greater. As the inlet pressure was reduced closer to the saturation line, the stage efficiency decreased. The range and operability of the stage remained excellent, but a significant reduction in efficiency was observed. It is interesting to note that only the stage efficiency was reduced, while the stage isentropic head coefficient remained nearly constant, Figure 3.

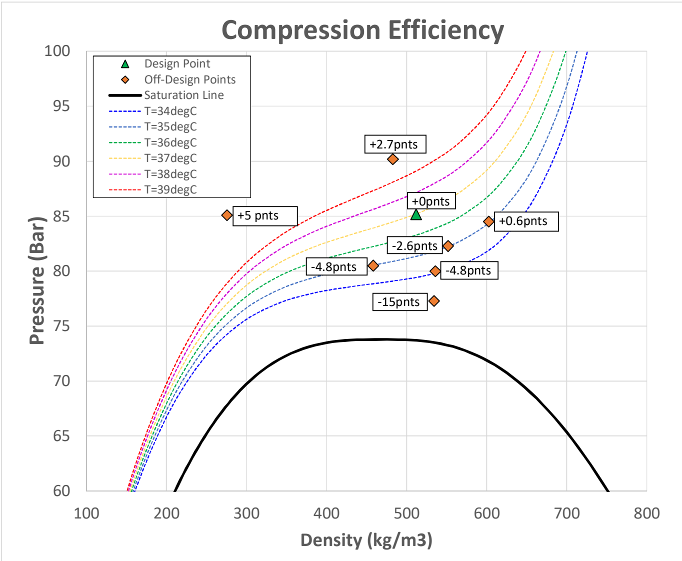


Figure 5. Design point efficiency vs operating point

Although the compressor efficiency may change at various inlet conditions, the variation in power is also critical to the overall cycle efficiency. Based on the measured compressor efficiency from these tests, the compression power was calculated assuming a constant mass flow rate and pressure ratio. Figure 6 shows how the compression power would change based on actual efficiency at the various compressor inlet conditions. The results show that the compression power is a minimum as the inlet conditions stay relatively dense, but do not approach too close to the saturation line. There is a significant increase in the required compression power as the inlet condition is reduced below an inlet density of 500 kg/m3, even if the compression efficiency increases, due to the additional work required to compress a less dense fluid.

The compression power of the cycle is also expected to increase for this compressor stage as the inlet pressure is reduced, moving closer to the saturation line. This is due to the fall off in efficiency that was observed at these conditions. It may be possible to design a stage that performs better near the saturation line. Minimizing any local areas of acceleration of the fluid in the inducer may be one way to help achieve this goal.

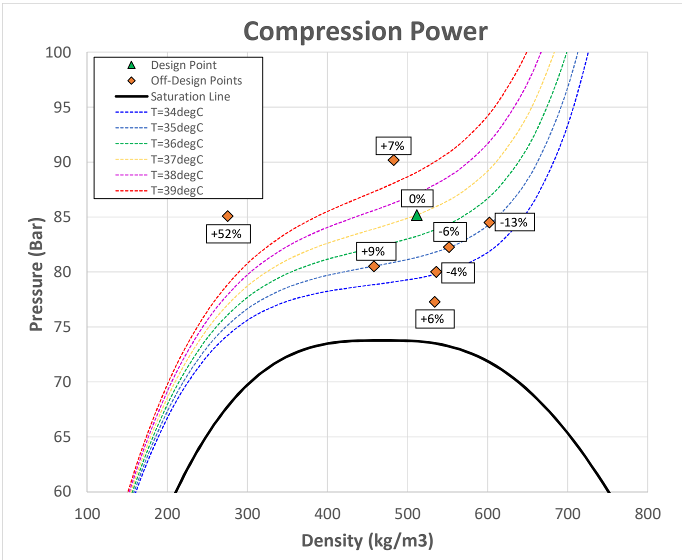


Figure 6. Compression Power at Various Inlet Conditions

The high rate of change of the fluid density relative to pressure and temperature fluctuations in the super-critical regime create unique challenges to the design and measurement of compressor performance - even more so as the critical point is approached. The preferred approach for assessing performance was based on measured inlet pressure and density to minimize the uncertainty in the performance calculations.

Test results show that the stage is most efficient at inlet pressures of 85 barA or greater. As the inlet pressure was reduced and the compressor was operated closer to the saturation line, the stage efficiency decreased. The range and operability of the stage remained excellent, but a significant reduction in efficiency was observed. It was observed that a significant increase in the required compression power for a cycle occurred as the inlet condition is shifted to a point where density is less than 500 kg/m3. Additionally, a significant reduction in compression work is expected for a sCO_2 Brayton cycle when operating at inlet densities higher than that at the critical point. It should be noted that sizing a compressor for nominal performance at cool inlet conditions, where the density is high, will result in a small stage with limited flow and isentropic head capacity at less dense inlet conditions.

IGV Design

During the current work, the machine used in the previous program was retrofitted with an inlet guide vane assembly.

Figure *7* shows an overall summary of the IGV used for the first stage compressor. The IGV will be externally mounted with a rotational actuator that penetrates the casing. The actuator turns a master vane, and the remainder of the vanes follow as the master vane translates a set of linkages that constrain the vanes.

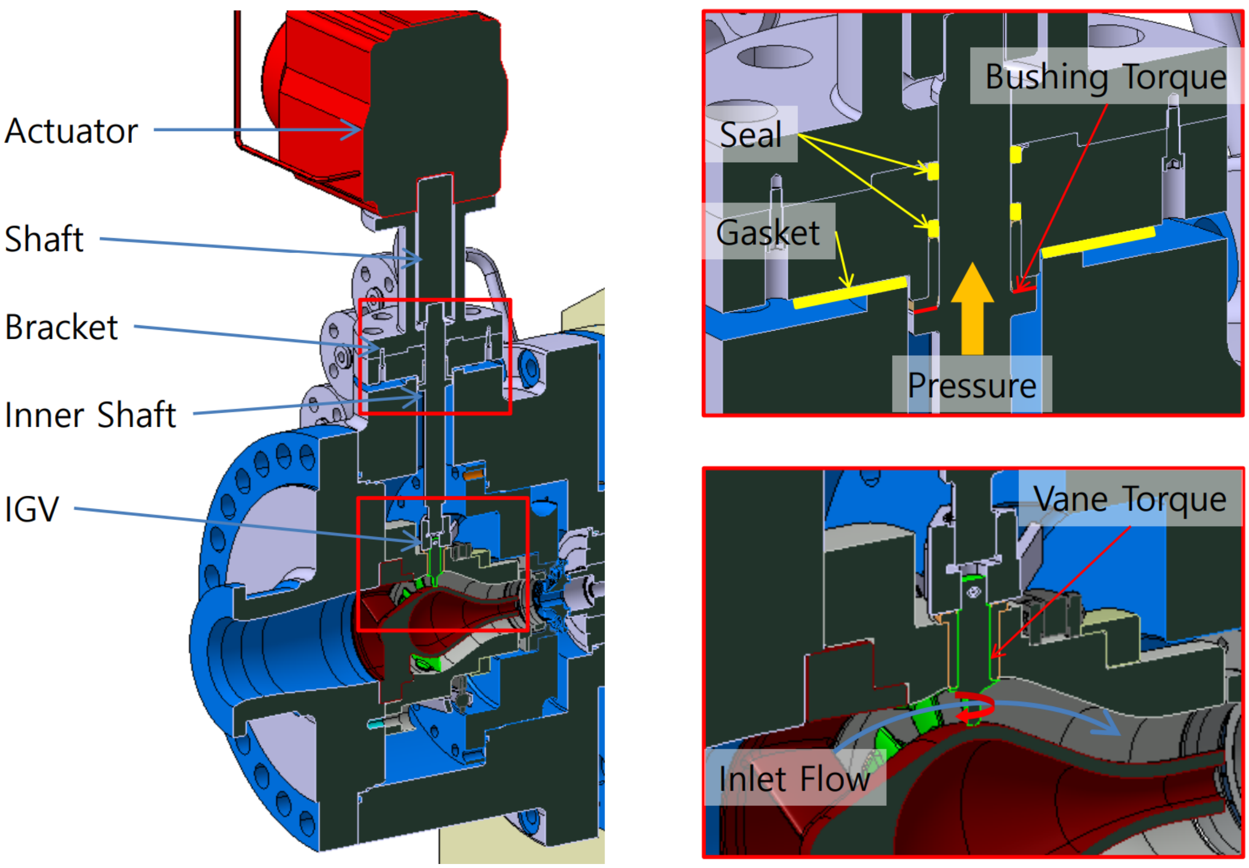
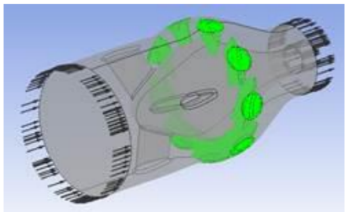


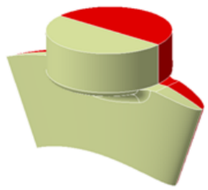
Figure 7. Overview of IGV Mechanism

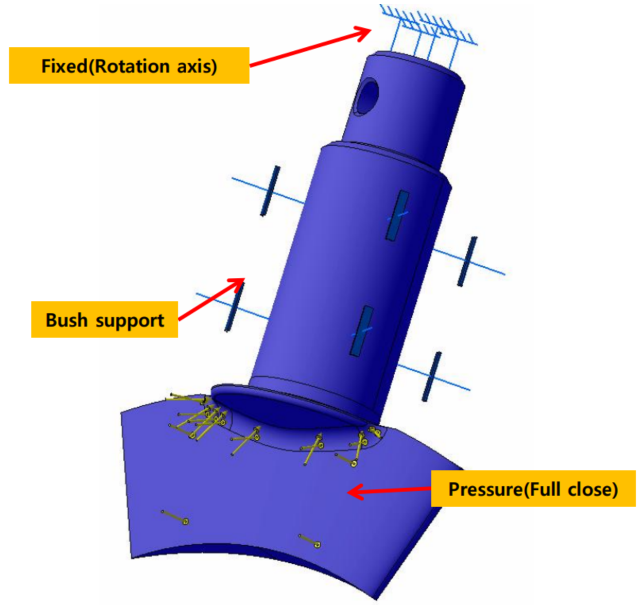
Table *2* shows the calculated vane forces at various operating angles and the relative DP that occurs at the vane. This table shows that the max force on the vane occurs at 70% flow at 20 degree vane angle. Figure 8 shows the FEA results for the vane angle with the highest stress. The stress criteria is satisfied at this condition. Additionally, Figure 9 shows the bushing torque required to maneuver the bushing into place. This feeds back on the actuator and the linkage requirements. The analysis shows that the actuator is sufficient to adequately locate the vanes. During testing, it is presumed that the angle of the IGV was within 1 degree of the desired value. This was verified with in-situ bump stop checks that were calibrated against a setup where the measurement could be taken with a protractor.

Table 2. IGV Design

Radial Vane force (N)		Flow(DP, %)					
		40	50	60	70	100	110
Opening Angle (deg)	20	51.31	78.61	115.67	156.04	-	-
	30	20.05	31.30	45.12	61.47	125.94	-
	40	9.35	14.63	21.06	28.65	58.58	70.90
	50	-	7.28	10.47	14.25	29.08	35.20
	110	-	-	-	-	7.18	8.71







◆ Worst case position

1) Opening Angel(deg) : 20°
2) Inlet Pressure(bar) : 86.12 (Re-comp)
3) Mass flow(kg/s) : 20.43 (Re-comp)

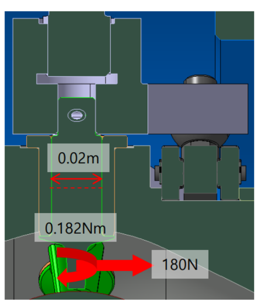
→ Max. Δ P(kPa) at Vane : 103.56

◆ Structural Analysis Result

1) Vane Material : 17-4PH(H1150)

Category	Criteria	Result	Satisfied
Stress	724MPa↓	6MPa	Yes
Max. Operating Torque (Each Vane)	NA	0.182Nm	NA

Figure 8. Vane Structural Analysis Summary



1) Vane Operating Torque : 0.182 Nm

2) Bushing Torque at Vane shaft : 0.72 Nm
(180N x 0.2(μ_max) x 0.01m(shaft_R) = 0.72 Nm)

3) Max. Vane Torque(Each Vane) : 0.902 Nm
(0.182 Nm + 0.72 Nm)

4) Safety Factor : 3

◆ Max. Total IGV Vane Torque for Actuator : 24.4 Nm
→ 0.902 Nm x 9(Vane Q'TY) x 3(Safety Factor) = 24.4 Nm

Figure 9. Analysis of Bushing Torque

Figure 10 shows a closeup of the straight vane configuration used during testing inserted in the anti-friction bushing, while Figure 11 and Figure 12 show the assembly from the impeller and inlet side of the machine, respectively.

Figure 10. IGV Vane and Vane installed in Brass Bushing

Figure 11. Impeller Side View of the IGV Assembly

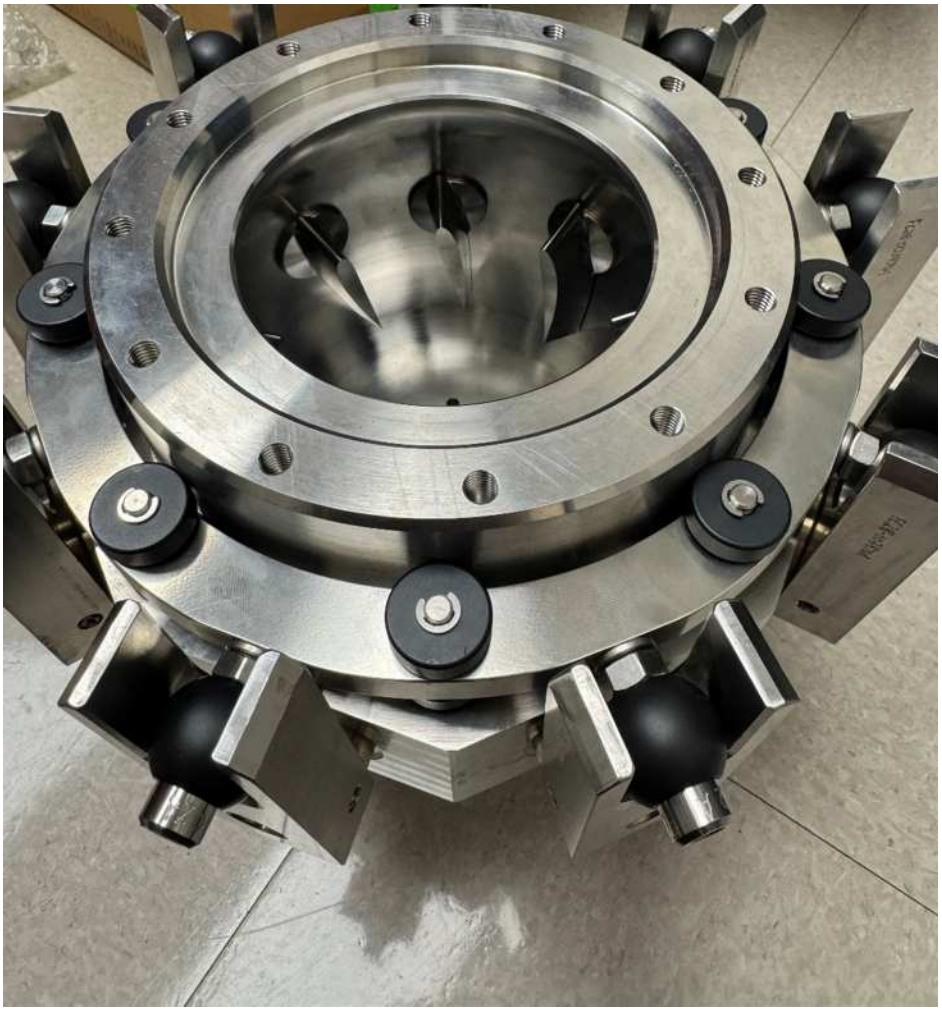


Figure 12. IGV Assembly Pre-assembly

Figure 13. IGV assembly with dynamic pressure probes shown on the inlet duct.

Summary of Test Results

Compressor Performance

SwRI took a total of 103 compressor performance test points across a multi-day campaign. The test plan was contrived to show the impact on performance (head and efficiency) due to both IGV setting but as well as fluid properties near the critical point of sCO_2. Figure 14 shows the suction conditions tested plotted on a pressure-enthalpy diagram. Loop mass and the amount of cooling were controlled to progressively bring the compressor nearer to the saturation dome, and there were even 8 test points taken with multiphase suction conditions.

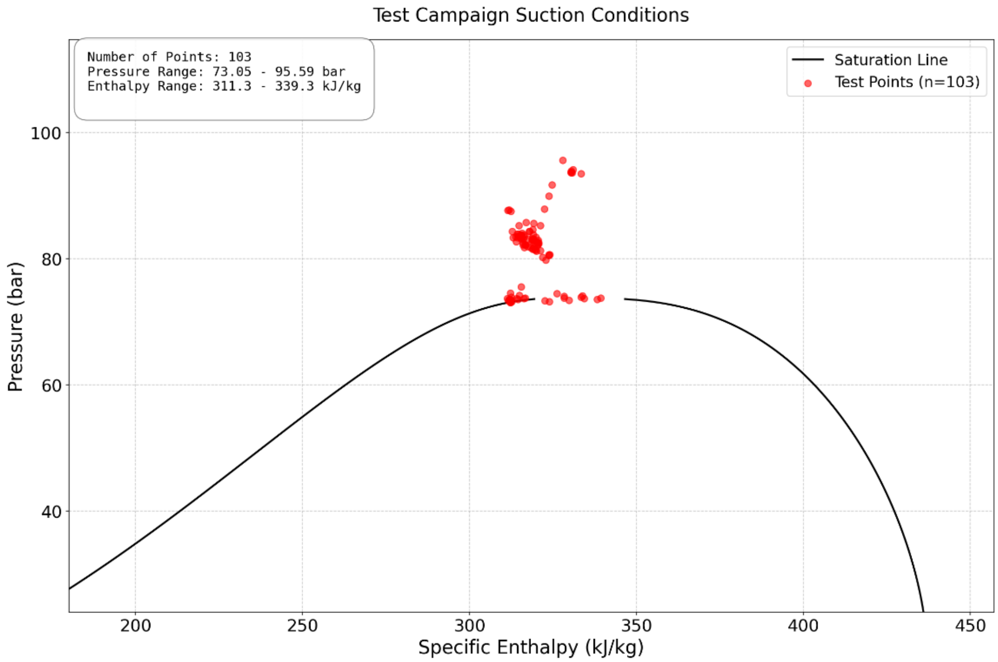


Figure 14. Compressor suction conditions during test campaign

Figure 15 and Figure 16 show the head coefficient and isentropic efficiency plotted against volumetric flow rate for the 103 test points. All test points were collected at the design speed of 1800 RPM. Notably there is a significant deviation in the compressor choke point depending on IGV setting, with the stage choking at 215 ACFM at an IGV setting of -60 degrees and 311 ACFM at 15 degrees. The difference is less pronounced for the surge line, with a minimum volume flow of 86 ACFM at -45 degrees and 123 ACFM for 15 degrees. A lower volume flow could have been achieved from the highly negative IGV settings, but there were three limiting factors. One is that machinery vibrations start to increase significantly at lower flow and the test operators chose to operate conservatively as to not risk machinery damage. It is possible that these vibrations were due to pre-stall behavior (inception of rotating stall cells), but the head rise characteristics at the lowest IGV settings had yet to fully flatten out or begin reducing. Isentropic efficiencies at these low flow test points did fall below 60%. To achieve the lowest flow conditions, the second stage compressor also had to operate near its surge line. For some cases the second stage compressor could not be conservatively brought nearer to stall and for other cases the head for the entire machine was too high and was limited by the loop pressure limit.

The compressor stage reached greater than 80% isentropic efficiencies for all IGV settings, with the -60 degree setting predictably having the lowest performance. A maximum isentropic efficiency of 86.2% was reached at the -15 degree setting, notably almost 2 points higher than the 0 degree IGV setting.

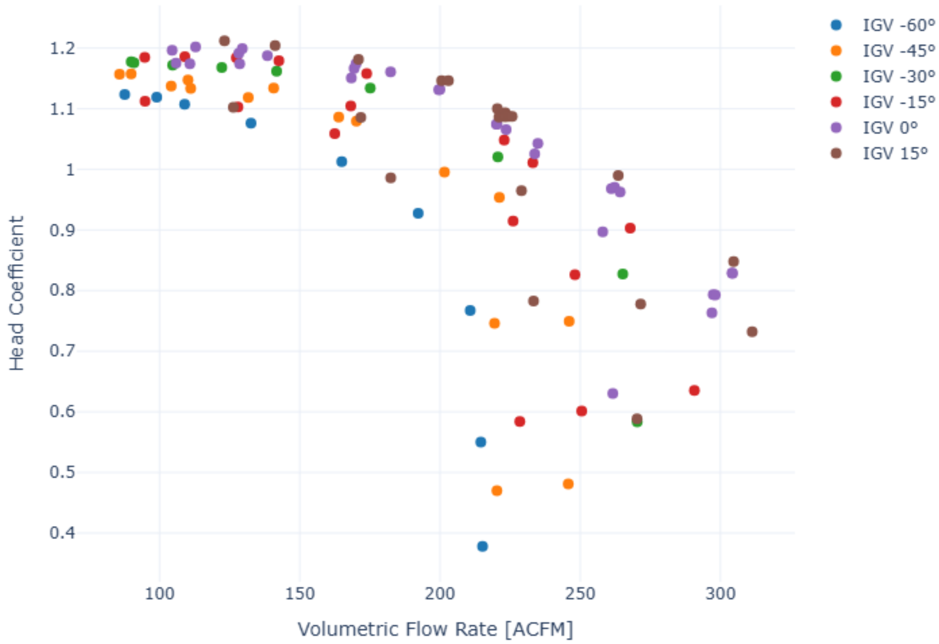


Figure 15. Head coefficient map of all campaign test points

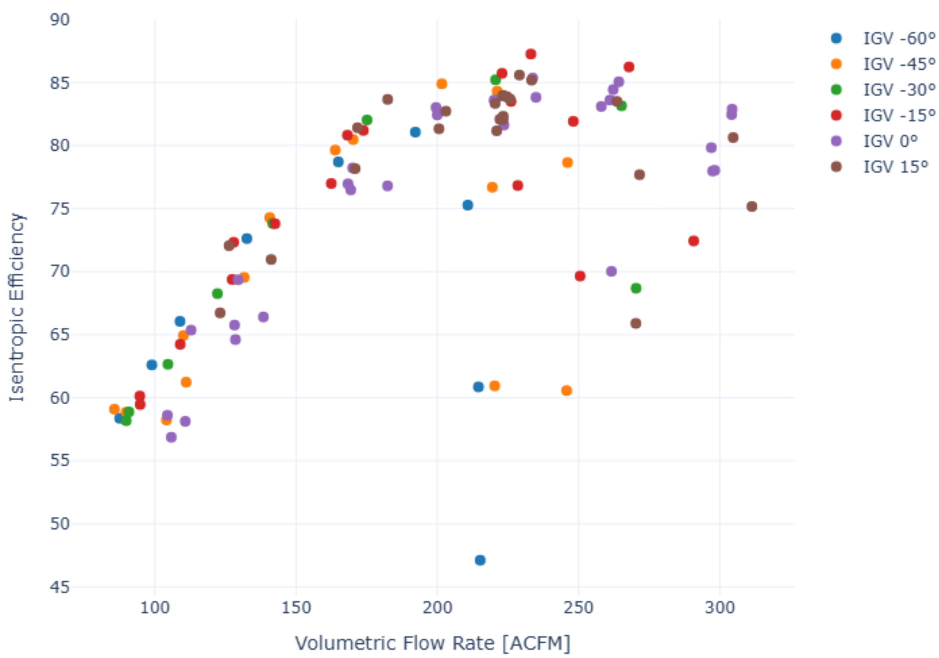


Figure 16. Isentropic efficiency map of all campaign test points

Figure 17 through Figure 20 show the sensitivity of isentropic efficiency to volume flow and suction conditions for various IGV settings. For all of these, there is a significant impact on the choke point. This behaves as expected, since the speed of sound is a function of temperature and the specific heat capacity ratio. At each IGV setting, there are distinct performance curves depending on the suction state conditions. For some IVG settings, such as -45 degrees, the performance difference is nominal and peak efficiencies are very close for both hot and cold suction conditions. The performance impact trends upwards with increasing IGV angle. At the 0 degree setting, for three points all at 129 ACFM there is an efficiency spread of 4.7 points, with the performance worsening nearer to the critical point. Interestingly at a setting of 15 degrees, the isentropic efficiency is 85.6% at 229 ACFM and ~88F but 82.1% at 222 ACFM and ~100 °F. Therefore, the performance is not universally better further away from the critical point.

The largest performance difference was found at the -15 degrees setting with a 10.4 point difference near the design volume flow. The maximum efficiency point was very near the critical point (89 °F, 1095 psia), but the low efficiency test point had multiphase suction conditions. Figure 21 shows the isentropic efficiency of all the multiphase test points captured. Note that there is some uncertainty regarding which suction conditions actually were multiphase, and there were likely many more points where the local conditions within the impeller dropped into the dome. While a significant performance impact seems evident due to multiphase operation, there were no obvious operational concerns in the middle of testing. Especially when operating in the middle of the map, there were no significant noise or vibrational signatures noticed.

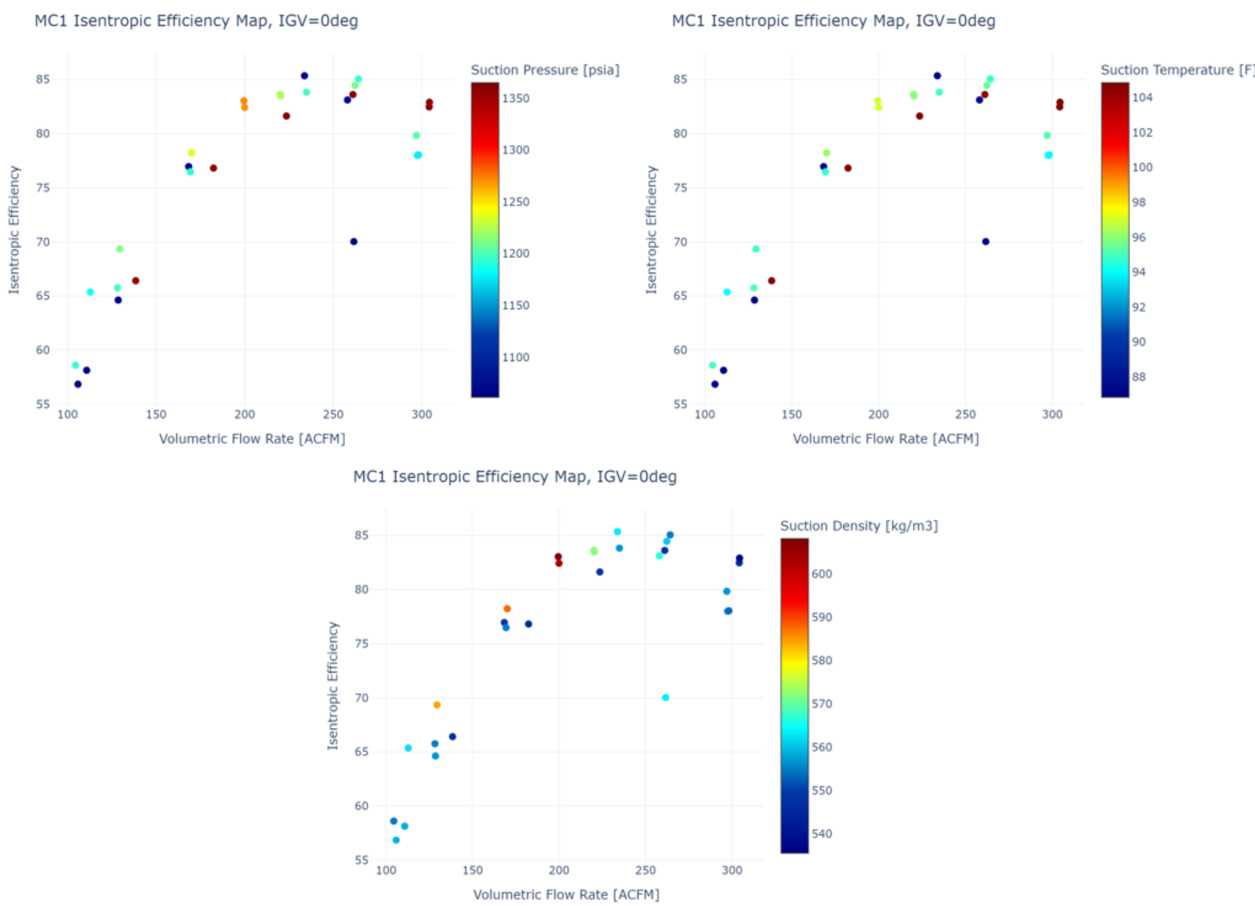


Figure 17. Sensitivity of compressor performance to suction thermodynamic properties at 0 degree IGV setting.

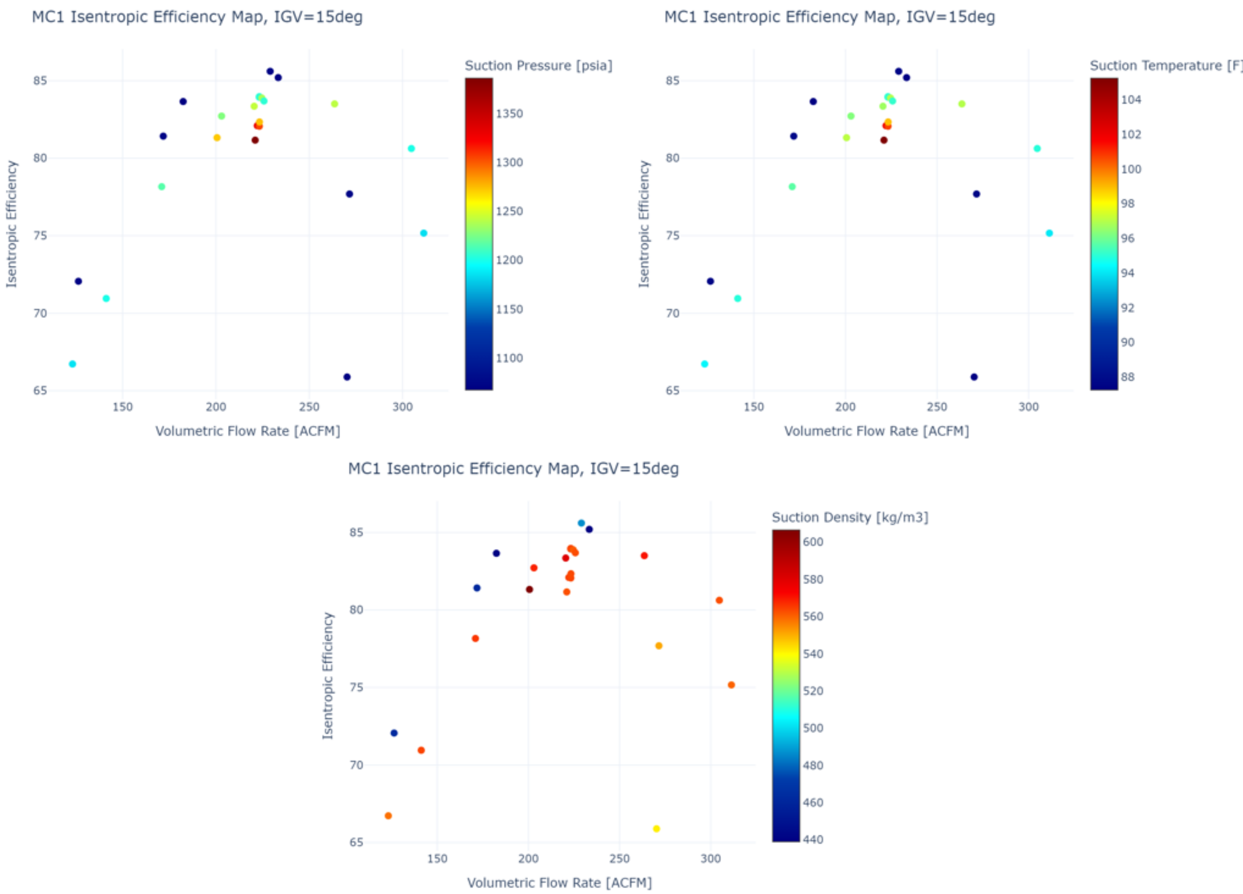


Figure 18.Sensitivity of compressor performance to suction thermodynamic properties at 15 degree IGV setting

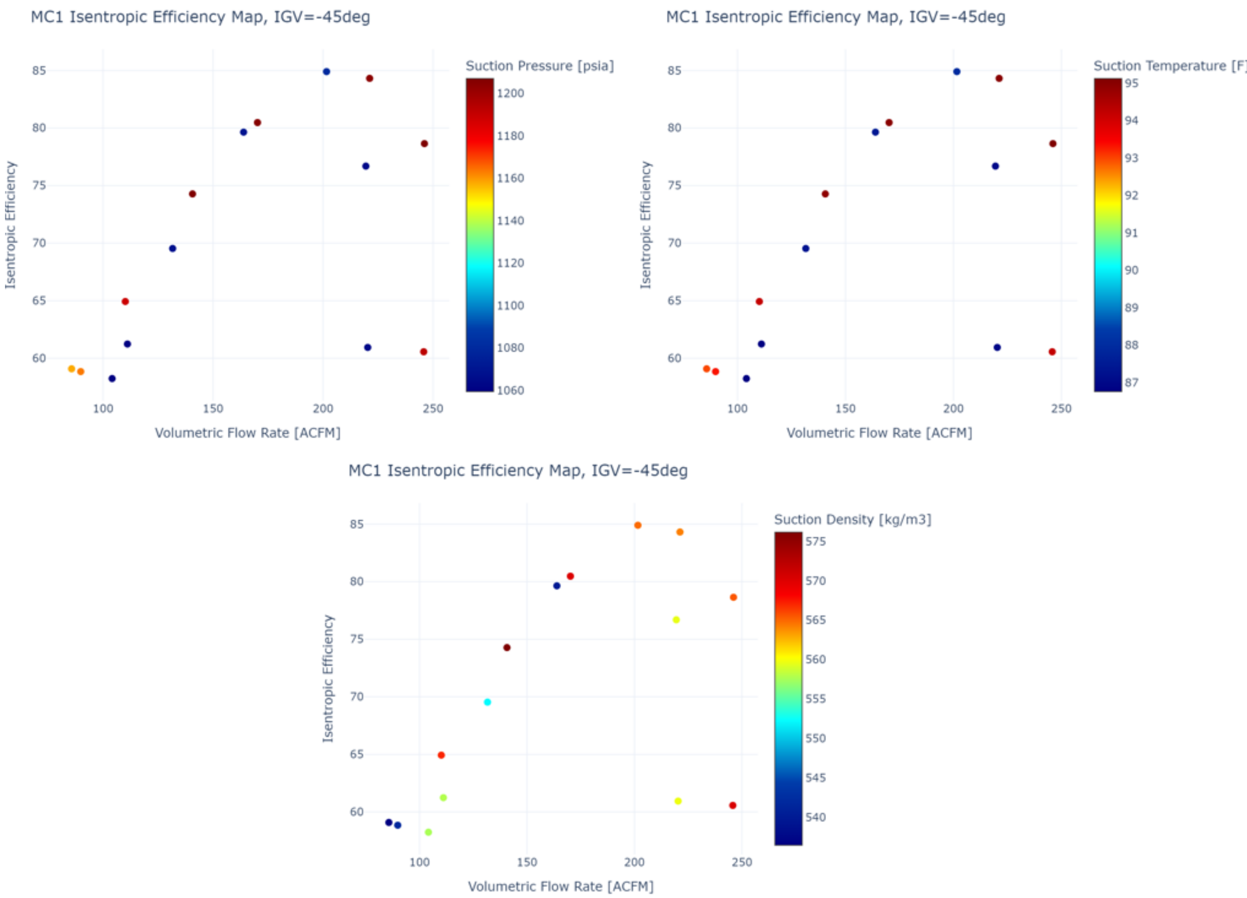


Figure 19. Sensitivity of compressor performance to suction thermodynamic properties at -45 degree IGV setting

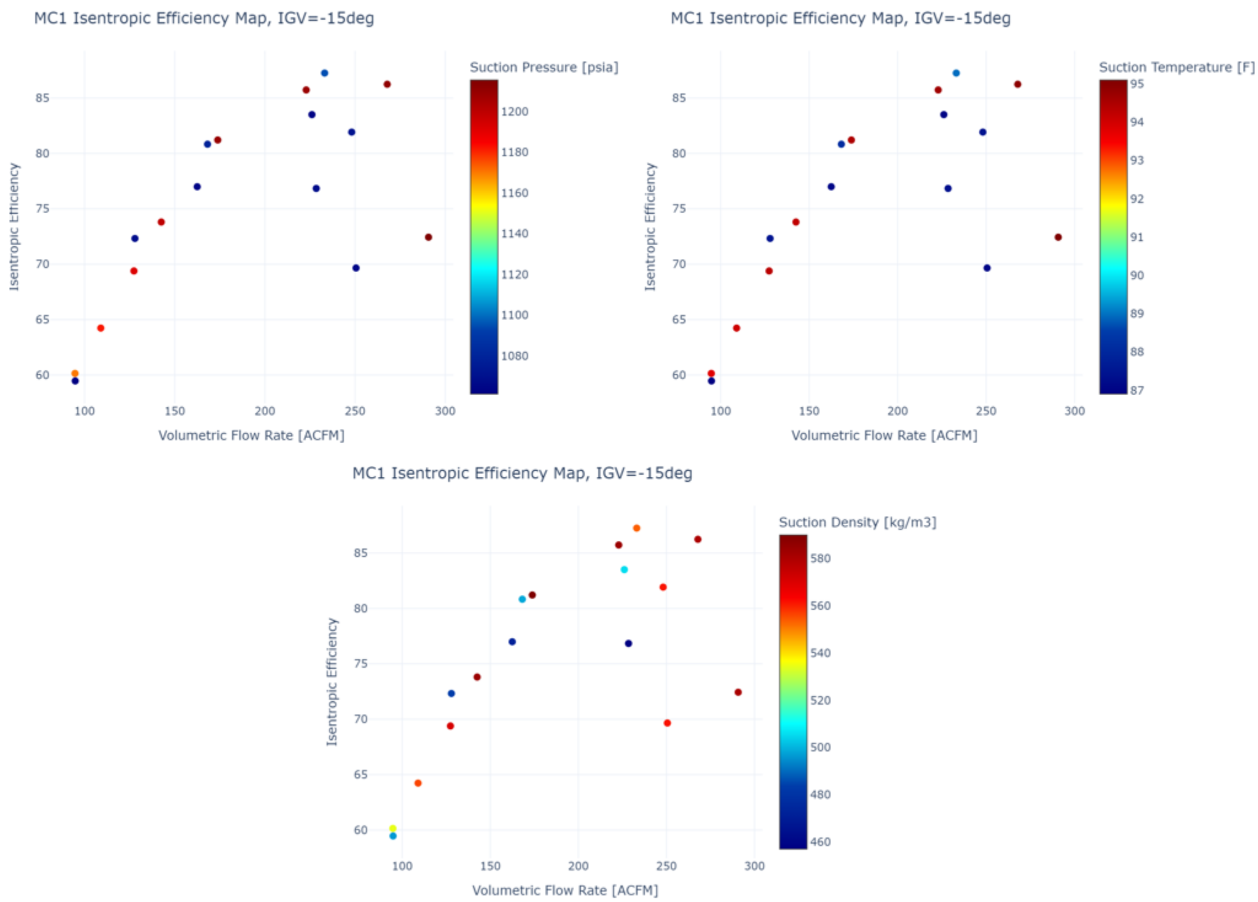


Figure 20. Sensitivity of compressor performance to suction thermodynamic properties at -15 degree IGV setting

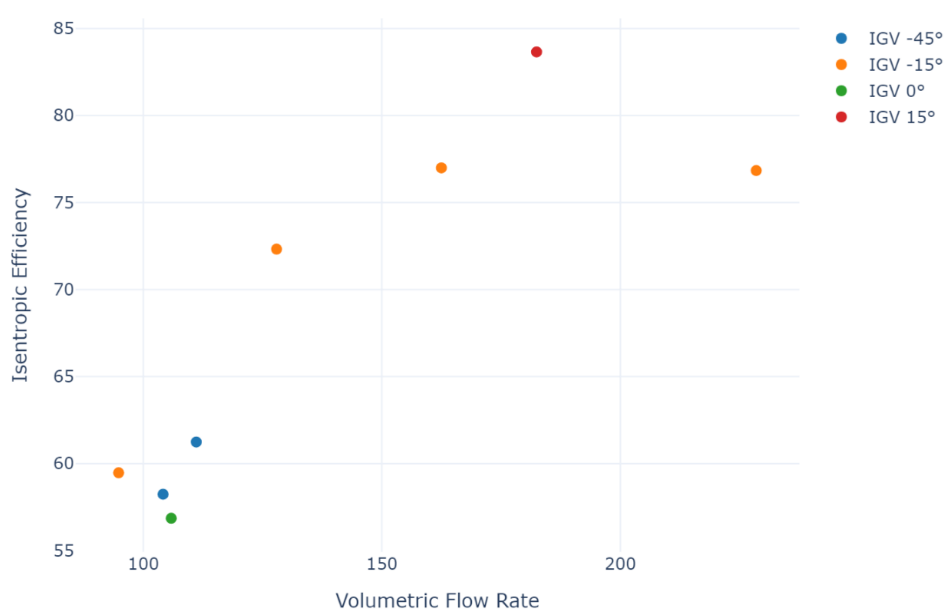


Figure 21. Multiphase test points

Throughout compressor operation, unsteady pressure probes (UPPs) measured dynamic pressure variation at the first stage compressor inlet (downstream of the IGV) and in the first stage diffuser. UPPs were placed at various circumferential locations around the impeller axis of rotation:

- IGV probes: 40°, 150°, and 310° from the vertical plane.
- Diffuser probes: 45°, 135°, and 295° from the vertical plane.

Figure 22 shows the locations of these probes in solid model sections.

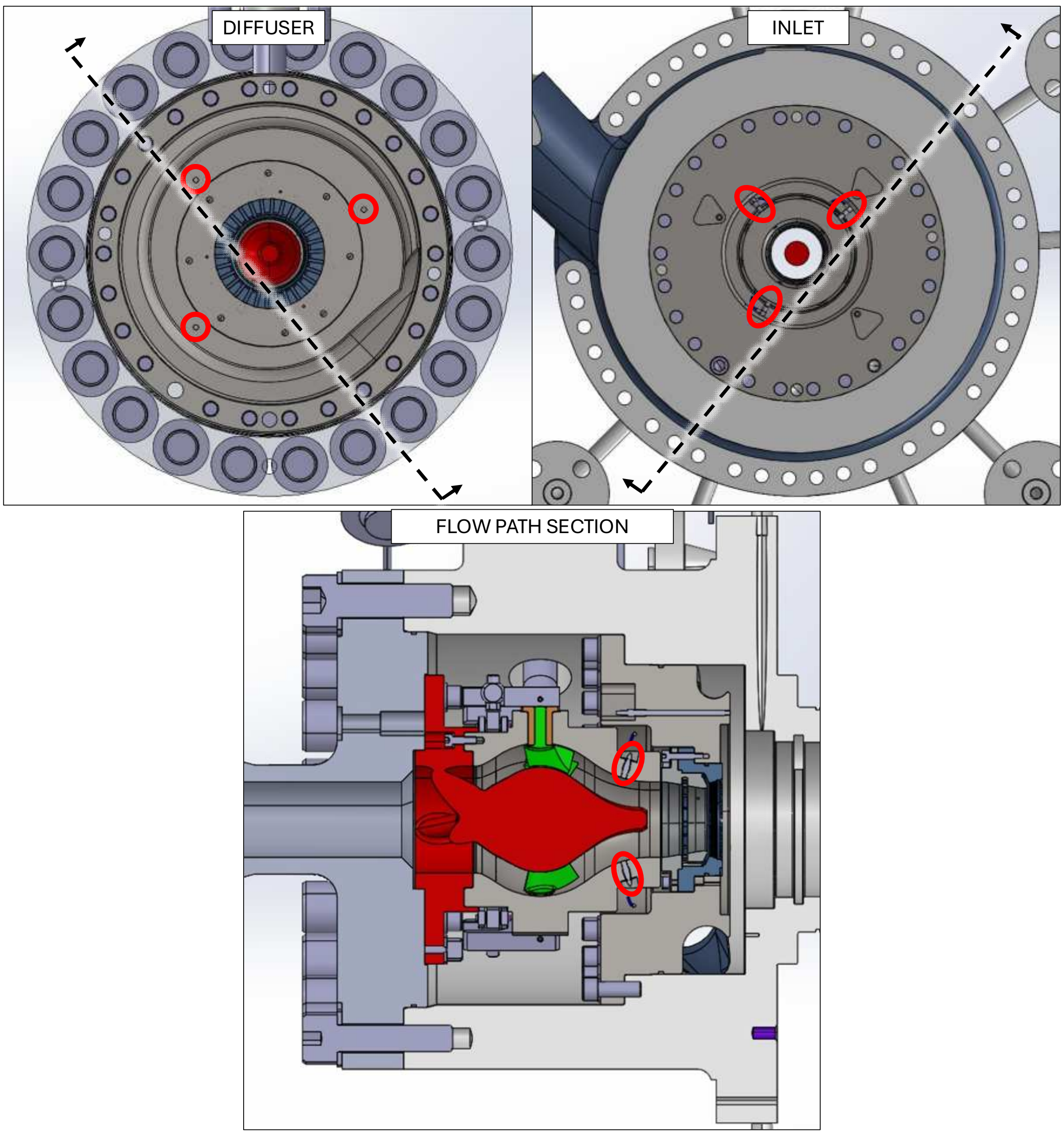


Figure 22. Stage one IGV and compressor housing solid model sections. UPPs are circled in red. Impeller is not shown.

Data was sampled at a rate of 25,600 Hz to ensure the capture of high frequency phenomenon (such as blade-pass frequencies) at a sufficient rate to prevent aliasing. Despite the large frequency range captured, the major function of the UPPs was to identify stall and surge phenomenon, which occur at sub-synchronous frequencies. As such, SwRI limited the frequency range of interest to 500 Hz (impeller speed: 27,600 RPM (460 Hz)). For reference, an arbitrary full frequency FFT spectrum (up to 10,000 Hz) from an IGV probe is shown in Figure 23 with annotations. In addition, only points recorded on the left side of the map were considered to investigate stall/surge phenomenon.

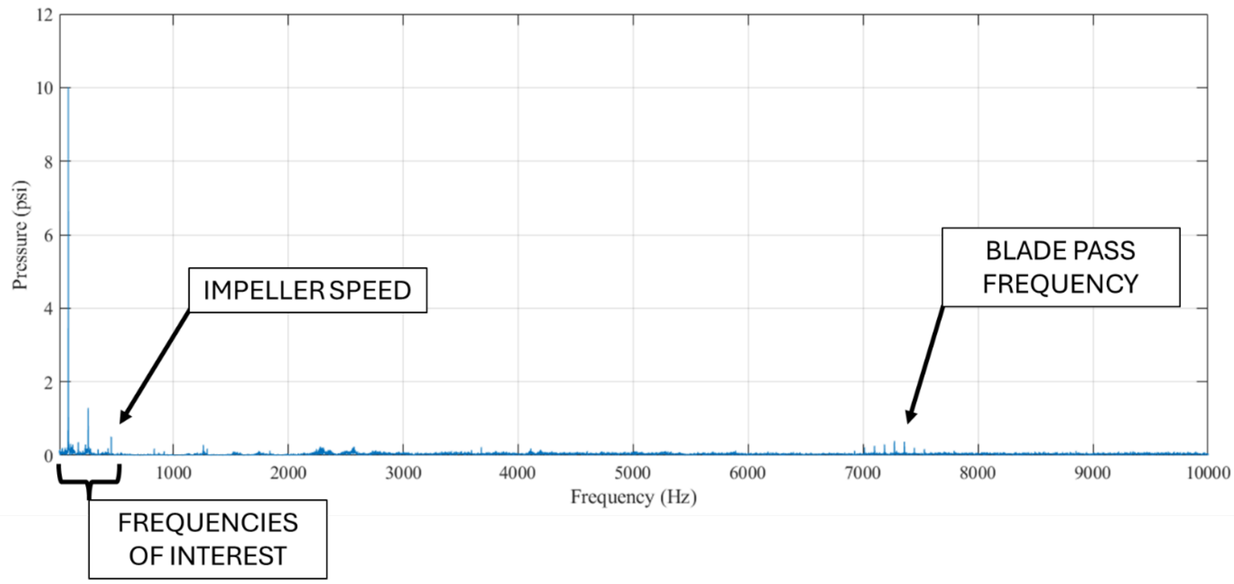


Figure 23. Full FFT spectrum of an IGV UPP.

To evaluate the effect of IGV angles, comparisons have been made between UPP data recorded at various IGV operating conditions. Figure 24 provides an overview waterfall of FFT spectrums. Data from the IGV and diffuser probes located at 150° and 135° from the vertical, respectively, are used in this figure and throughout this report to limit the repetition of similar data. It should be noted that in general, the IGV probes produce consistent and similar data irrespective of their circumferential location, however, data from probes in the diffuser vary in amplitude, likely due to non-axisymmetric geometry.

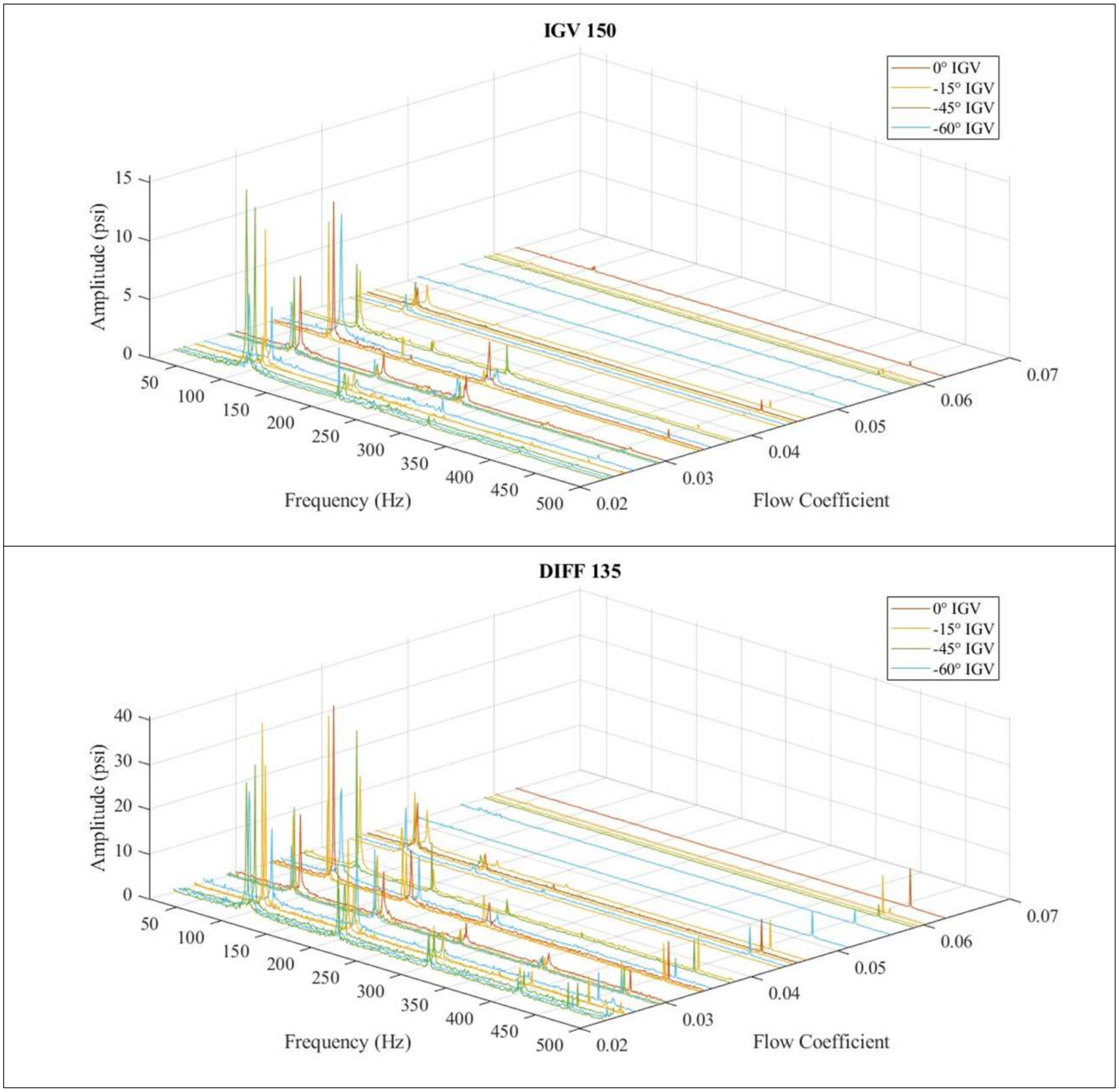


Figure 24. FFT spectrums vs flow coefficient for a diffuser and IGV UPP throughout range of recorded operating points.

Separating the data into groups by flow coefficient allows for the comparison of the effect of IGV angle on the dynamic data. Data from conditions near the best efficiency point and at a low flow rate are shown in Figure 25.

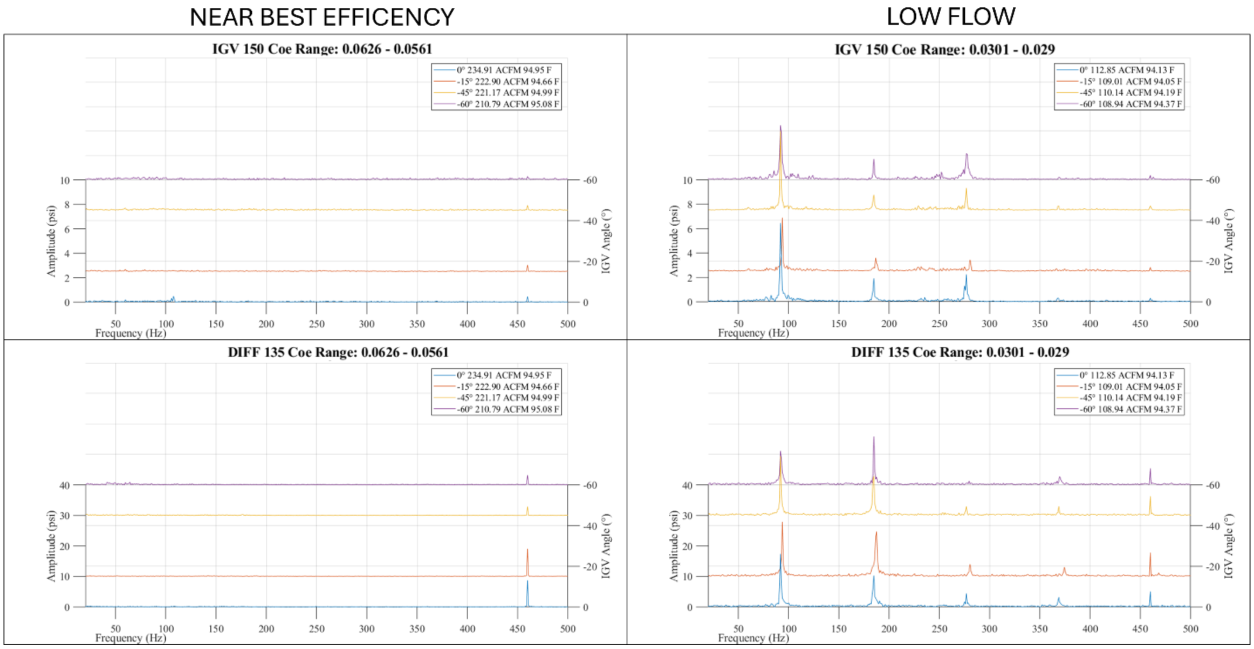


Figure 25. FFT spectrums of IGV and diffuser UPPs for various IGV angles at conditions with similar flow coefficients.

Despite suspected IGV stall at excessive IGV angles, no obvious differences are apparent in the IGV FFT spectrums. As a more sensitive means of evaluation, overall sub-synchronous fluctuation amplitude (20 to 455 Hz) has been estimated using a Euclidean norm. Plots of overall fluctuation for the same conditions as above are shown in Figure 26.

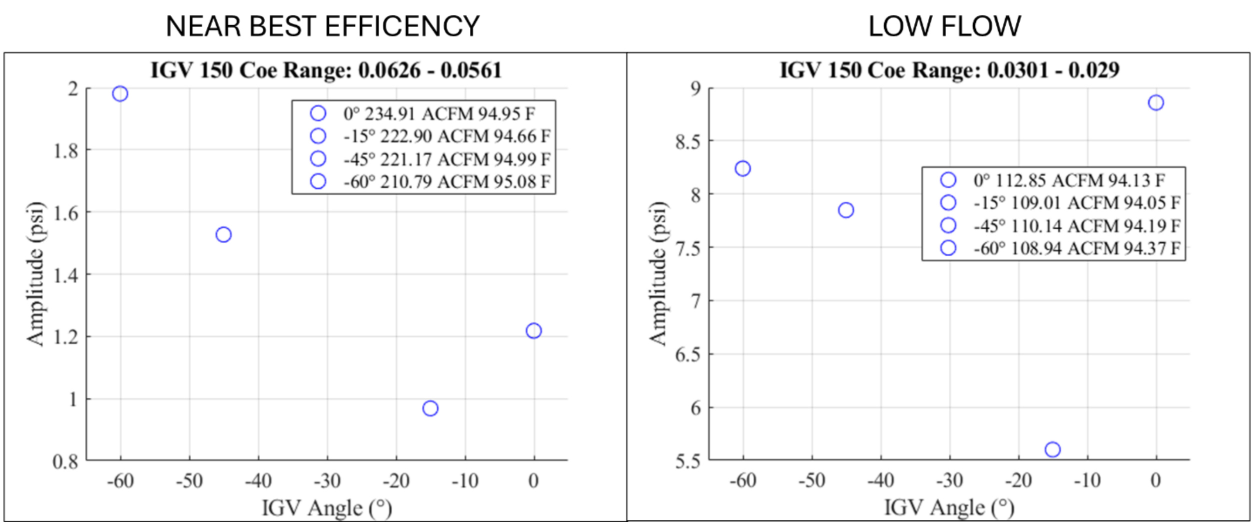


Figure 26. IGV and diffuser UPP overall sub-synchronous amplitude for various IGV angles at conditions with similar flow coefficients.

The trend shown for the IGV UPPs in Figure 26 shows the effect of decreasing the IGV angle. At less negative IGV angles, flow separation is not expected, and the result is coswirl with an improved incidence angle at the leading edge of the impeller. The smoother entry path results in lower amplitude flow disturbances detected at the IGV UPPs. As the IGV angle is decreased further, flow separation at the IGV is expected, and the turbulent wake is captured by the UPPs. Please note that the overall amplitude for the low flow condition also represents stall/surge phenomenon that appear in the FFT spectrum, so the plot is unlikely to represent IGV stall alone.

Figure 27 shows the effect of multiphase and near-multiphase flow on flow stability for a -15 IGV angle. At the IGV, FFT spectrums shows consistently reduced amplitudes for multiphase or near-multiphase flows at lower flow rates. This phenomenon can be better visualized in Figure 28, where overall sub-synchronous pressure fluctuation amplitudes are plotted. It is possible that the increased density of the multiphase and near-multiphase flows provided damping to the pressure fluctuations, reducing the fluctuation amplitudes. This trend is not apparent in the diffuser, where the flow is supercritical regardless of inlet conditions. Here, FFT spectrums for all plotted flows appear similar.

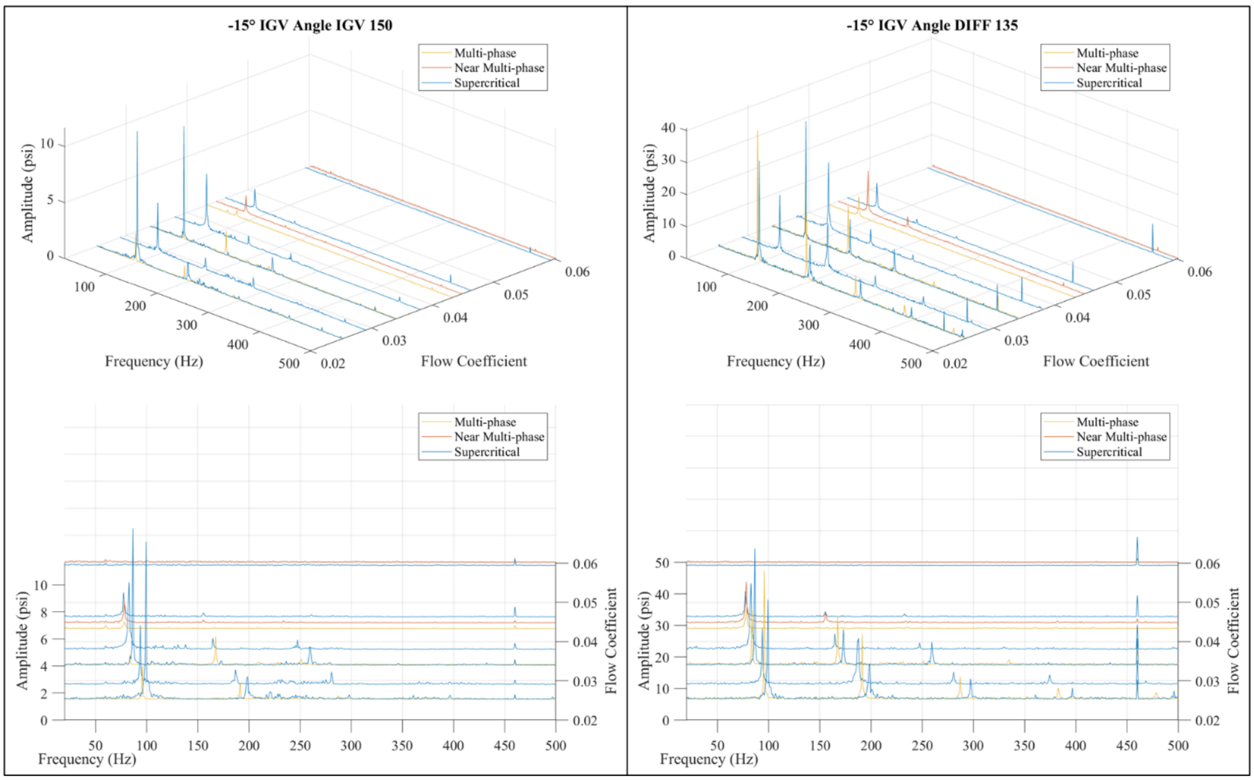


Figure 27. FFT spectrum vs flow coefficient for supercritical, near multi-phase, and multiphase inlet conditions at a -15° IGV angle.

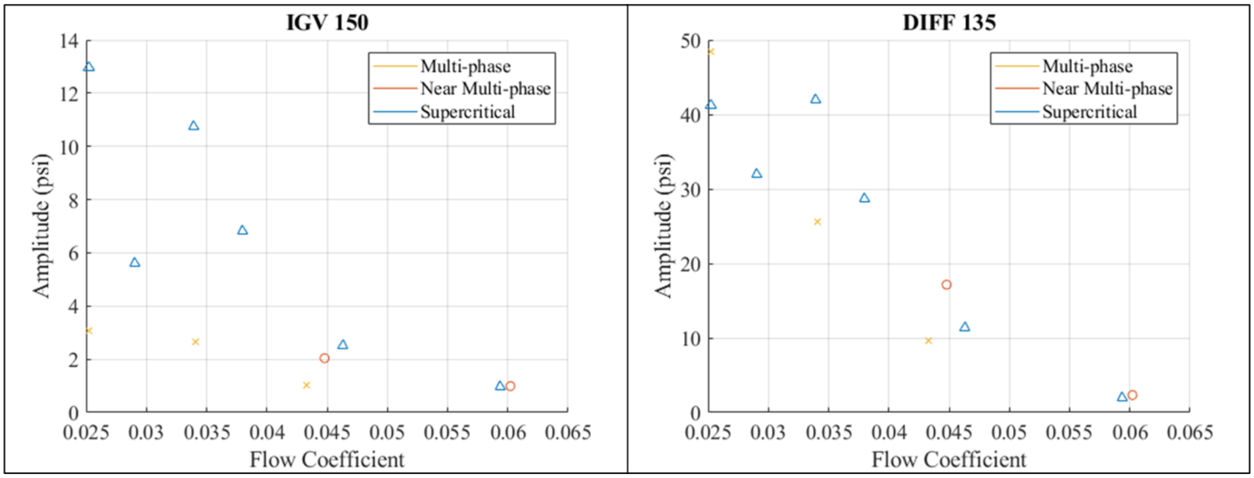
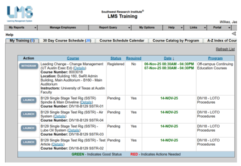


Figure 28. Overall sub-synchronous amplitude vs flow coefficient for supercritical, near multi-phase, and multiphase inlet conditions at a -15° IGV angle.

8. Significant Accomplishments and Conclusions:

Milestone #	Performance Metric	Success Value	Assessment Tool / Method of Measuring Success Value	Verification Process	Metric Justification, Additional Notes
1.1.1	IGV angle actuation at supercritical conditions	IGV angle actuated full scale	IGV angle can be controlled within 5% of desired angle	Figure 15	Proper IGV actuation must be verified before testing could continue. As is, no sCO2 compressor has incorporated variable IGVs.
1.2.1	Matrix of Q and IGV angle	Performance data collected at each map point	Yes/No, results accepted by DOE/sCO2 Operators Commission – Jeff Moore, Robert Pelton, and Joshua Kim	Figure 15	It is expected that the IGV will provide a more effective means of flow reduction over the non-IGV design at conditions close to the dome (at low inlet temperature conditions). To verify IGV operation in such conditions, the assembly should be fully actuated at such conditions.

Milestone #	Performance Metric	Success Value	Assessment Tool / Method of Measuring Success Value	Verification Process	Metric Justification, Additional Notes
1.2.2	V=Head*Flow, H=comp. head	V_actual = V_desired * 0.95 H_actual = H_desired * 0.95	Uncertainty error bars <= 11% via Monte Carlo analysis	This secondary goal was not met due to delays and a desire to obtain more test data that was of archival quality than to spend time working on control strategy.	A secondary overall project goal is to generate a control scheme which will inform IGV settings during off-design compressor operation to maximize efficiency. For this map to be useful, it must be deomonstrated that actual conditions can reasonably approach the predicted conditions.
1.3.1	LF=loss factor	LF = (ή_design - ή_actual)/ή_design	Uncertainty error bars <= 11% via Monte Carlo analysis	+- 2%	To better predict future design updates, analytical models have been built. By comparing test data to these models, we can better inform future analysis or performance predictions. A simple loss factor can be calculated and applied to future analytical results.

Milestone #	Performance Metric	Success Value	Assessment Tool / Method of Measuring Success Value	Verification Process	Metric Justification, Additional Notes
1.4.1	Meeting	Approve IGV test plan	Expert review and approval	Teams Meeting	To ensure that IGV testing efforts are being executed efficiently an advisory committee should be formed, consisting of industry stakeholders.
1.4.2	Meeting	Approve control scheme methodology	Expert review and approval	Teams Meeting	To ensure that the resulting compressor control scheme is useful to actual machinery operators, the sCO2 operators advisory committee should approve its effectiveness.
1.4.3	Publication	Conference/publication acceptance	Expert review and approval	Formal technical paper acceptance (This is in process)	To actualize the advancements associated with this research, the findings must be spread to the sCO2 machinery community through paper presentations.
1.5.1	Quantity of project staff current on annual training	100%	Yes/No, SwRI training management system		DEI activities bring awareness to the participants through training and support a pathway for growth and opportunity for underrepresented

Milestone #	Performance Metric	Success Value	Assessment Tool / Method of Measuring Success Value	Verification Process	Metric Justification, Additional Notes
					communities in STEM fields
1.5.2	Bids solicited from minority-serving vendors	>50% of applicable procurements	Yes/No	Report Q6	DEI activities bring awareness to the participants through training and support a pathway for growth and opportunity for underrepresented communities in STEM fields
1.5.3	Technical presentations conducted at minority-serving institutions	2 presentations per year	Yes/No	BOA Rocketry Training for Dual Language Students	DEI activities bring awareness to the participants through training and support a pathway for growth and opportunity for underrepresented communities in STEM fields

9. Conclusions and Path Forward:

Having completed testing on IGV's at near dome operation, the team would like to reiterate the primary accomplishments of the program and recommend pathways for future development of the current program and sCO_2 power cycles in general.

Key accomplishments of this program:

1. IGVs were designed, assembled, and tested in an sCO_2 compressor. For an axial inlet centrifugal compressor, this is the first instance of IGVs tested in an sCO_2 environment to the authors knowledge.
2. Compressor IGVs were tested successfully at an inlet density of up to 600 kg/m^3. This is one of the highest inlet densities and corresponding fluid momentums (ρV^2) ever tested for an IGV.
3. The compressor testing performed in this program includes one of the most comprehensive mappings of temperature, pressure, and IGV angle for two-phase, near dome, and supercritical CO_2 operation.
4. The compressor was operated for approximately 80 hours inside of the dome. Both kHz and MHz dynamic pressure data were recorded to learn more about the inlet characteristics of the flow into the compressor, wake and jet profiles, and aerodynamic stability.
5. The preliminary analysis of the compressor performance shows that the IGV does a suitable job of altering the head flow characteristics of the compressor and can be used to maximize efficiency or flexibility of the unit for fixed speed operation.

Compressor Performance and Efficiency:

- 103 performance test points were collected under varying inlet guide vane (IGV) settings (ranging from -60° to +15°) and conditions near the critical and multiphase states of CO_2.
- **Choke and Surge Points**:
  - Choke point volumetric flow varied significantly based on IGV setting: 215 ACFM at -60° versus 311 ACFM at +15°.
  - Surge line showed less variation, with minimum flow ranging from 86 ACFM (-45°) to 123 ACFM (+15°).
- **Efficiency**:
  - Peak isentropic efficiency was above 80% for all IGV settings, except at some low flow test points where it fell below 60%.
  - The best efficiency was 86.2% at a -15° IGV angle, outperforming the 0° setting by 2 points.

- **Suction Conditions**:
  - Efficiency was sensitive to fluid properties:
    - At a 15° IGV setting, efficiency ranged from 85.6% (88°F) to 82.1% (100°F).
    - Efficiency differences near the design volume flow reached up to 10.4 points, with the lowest performance linked to multiphase suction conditions.
  - Multiphase operation impact: While efficiency dropped during multiphase conditions, no significant operational concerns like excessive noise or vibrations were observed.

Dynamic Pressure Fluctuations (Unsteady Behavior):

- **UPP Data**:
  - Unsteady pressure probes (UPPs) monitored dynamic pressures at the first-stage inlet and diffuser to capture stall/surge behaviors.
  - Sub-synchronous fluctuation amplitude (20–455 Hz) increased at lower IGV angles, correlating with disturbance from flow separation and IGV stall.
- **IGV Behavior**:
  - Reduced IGV angles (-60°, -45°) led to increased flow turbulence due to separation and wake effects, as captured by UPPs.
  - At less negative IGV angles, smoother flow improved stability with minimal disturbances.
- **Multiphase Flows**:
  - Multiphase and near-multiphase inlet conditions at -15° IGV angles dampened sub-synchronous pressure fluctuation amplitudes near the impeller inlet, likely due to increased density.
  - In contrast, diffuser fluctuation amplitudes remained unaffected due to supercritical conditions regardless of inlet conditions.

Future recommendations for the advancement of sCO_2 cycles following the accomplishments from this program.

1. This program showed successful operation of an sCO_2 compressor at numerous locations inside and outside of the dome, and previous programs have optimized cycle performance using predicted head-flow curves. It is recommended that a future program focus on utilizing the real maps obtained in this program to close the loop and predict the performance of sCO_2 cycles (energy storage and power

generation) using real compressor maps. The result of this study would be improved confidence on the performance and economic viability of sCO_2 power cycles in real applications at various inlet conditions.

2. One of the interesting features from DE-EE0007114 was the shroud recirculation feature. It would be of interest to determine the effectiveness of this feature by testing a fully shrouded compressor wheel to determine the impact of the feature on both range and performance of the compressor.
3. Conduct a comprehensive teardown and forensic inspection of the compressor, IGV assembly, bearings, and seal systems to evaluate the condition of all rotating and stationary hardware after roughly 1,000 hours of sCO_2 operation. This effort should include detailed dimensional inspection, photographic wear mapping, and surface characterization, followed by metallurgical analysis (optical/SEM, hardness testing, fractography) on any components exhibiting wear, cracking, erosion, or material changes. In parallel, a full evaluation of the seals should be performed, including examination for thermal distress, contamination, or chemical interaction with high-density CO_2. Together, this project would identify the dominant degradation and failure mechanisms, validate component life predictions, and provide data-driven recommendations for improved materials, coatings, lubrication strategies, and maintenance intervals.
4. Undertake a detailed rotordynamics and vibration analysis of the compander system with specific emphasis on how IGV position—both fully in the dome and near-dome settings—affects shaft loading and stability. This effort should include updated Campbell diagrams, modal characterization, and high-speed vibration measurements across the operating envelope to identify any shifts in critical speeds, subsynchronous instabilities, or rub margins caused by IGV-induced aerodynamic forcing. Additional testing should evaluate bearing stiffness, seal cross-coupling effects, and the dynamic response during rapid IGV transitions or transient load changes. The results will quantify stability limits, validate analytical rotordynamic models, and establish recommended operating windows and hardware or control modifications needed to ensure robust vibration performance in future test campaigns.
5. Pursue an integrated system-level modeling and controls optimization effort that incorporates the newly generated compressor performance data, rotordynamic behavior, and heat-transfer characteristics from the compander loop. This project would update and validate dynamic plant models using measured aero maps, surge boundaries, IGV response characteristics, and transient datasets, ensuring that simulation tools accurately reflect real hardware behavior. With an improved model, the study should then refine control strategies—including IGV scheduling, surge avoidance logic, load-transition handling, and protective interlocks—to improve stability, efficiency, and operational flexibility. System integration tests using the physical loop would validate these control strategies under representative transients. The outcome will be a fully validated controls framework and a set of recommended operating and tuning practices that

enhance safety and performance in future DOE sCO_2 demonstration and scale-up efforts.

6. The results from this program show substantial deviations in head flow characteristics at different inlet conditions. Conventional surge control methodologies use these characteristics to protect the machine against surge. A future program should focus on analyzing the results obtained by the high-frequency dynamic pressure measurements from this program to develop a surge control scheme that uses dynamic pressure and flow instability to control surge in the machine as opposed to conventional methods such as head-flow characteristics. The anticipated result is increased allowable machine range in non-research applications of sCO_2 turbomachinery.
7. While CFD based maps of compressor performance were used to develop the design used in DE-EE0007114, the current program provides an interesting dataset that could be used to further validate the performance of CFD predictions for near-dome performance and compressor performance using IGVs. This could include modeling, mesh, and turbulence sensitivity studies, as well as two-phase vapor behavior and its impact on compressor performance with and without IGVs.
8. At the last section of FTR, please add a section on future work based on the data, to enable a robust control system using IGVs for broad range operation of the compander. Please add a discussion that would involve interviewing experts such as Jeff Moore, Hanwha, to provide directions to the DOE on future research on compressor inlet flow thermodynamics and performance/control. Please include at least two industry experts and one SWRI expert.
9. Please load the full dataset on experimental results to the PMC CLEARY marking proprietary. This dataset will not be issued to public. You may choose to load this to OSTI marking 5 year proprietary. This PMC load request is because we expect the present organization of SETO/CSP to go away and get reorganized.

10. Products:

[1] Wilkes, J., Bishop, N., Replogle, C., Brown, Z., Klaerner, J., 2026, “An Experimental Study on Inlet Guide Vane Effects in a Near-Critical sCO_2 Compressor, Part I: Aerodynamic Performance Characterization,” *Proc. ASME Turbo Expo 2026*, June 15-19, GT2026-178858 Milan Italy.

[2] Brown, Z., Wilkes, J., Bond, E., 2026, “An Experimental Study on Inlet Guide Vane Effects in a Near-Critical sCO_2 Compressor, Part II: Flow Stability and Compressor Health as Observed with Dynamic Instruments,” *Proc. ASME Turbo Expo 2026*, June 15-19, GT2026-177548, Milan Italy.

[3] Replogle, C., Wilkes, J., 2026, “An Experimental Study on Inlet Guide Vane Effects in a Near-Critical sCO_2 Compressor, Part III: Comparison and Uncertainty Evaluation of Efficiency Determination Methods,” *Proc. ASME Turbo Expo 2026*, June 15–19, GT2026-178879, Milan, Italy.

11. Project Team and Roles:

SwRI was the sole executor on this project.

12. References:

[1] United States, Executive Office of the President [Joseph Biden]. Executive Order on Tackling the Climate Crisis at Home and Abroad, 27 January 2021.

[2] Pelton, R., Bygrave, J., Wygant, K., Wilkes, J., Revak, T., Kim, K., 2022, "Near Critical Point Testing and Performance Results of an sCO_2 Compressor for a 10 MWe Brayton Cycle," Proc. of ASME Turbo Expo 2022, June 13-17, GT2022-83503, Rotterdam, The Netherlands.

[3] ASME PTC 10-1997, "Performance Test Code on Compressors and Exhausters", The American Society of Mechanical Engineers, New York, 1998.

[4] Wahl, G., "Efficiency Uncertainty of a Turbine Driven Compressor in a Supercritical CO_2 Brayton Cycle" sCO_2 Power Cycle Symposium, Troy, NY. 2009.

[5] Sandberg, M., Colby, G., 2013, "Limitations of ASMEPTC 10 in Accurately Evaluating Centrifugal Compressor Thermodynamic Performance" Turbomachinery & Pump Symposia, October 13, 2013

[6] Mortzheim, J., Hofer, D., Priebe, S., McClung, A., Moore, J., Cich, S., "Challenges with Measuring Supercritical CO_2 Compressor Performance When Approaching the Liquid-Vapor Dome," Proc. ASME Turbo Expo, Paper GT2021-59527, Online, 2021.

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[8] Span, R., Wagner, W., A New Equation of State for Carbon Dioxide Covering the Fluid Region from the Triple-Point Temperature to 1100 K at Pressures up to 800 MPa. J. Phys. Chem. Ref. Data 25, 1509. 1996

[9] Hosangadi, A., Weathers, T., Liu, T., Pelton, R., Wygant, K., Wilkes, J., Numerical Predictions of Mean Performance and Dynamic Behavior of a 10MWe sCO_2 Compressor with Test Validation. Proc. ASME Turbo Expo, Paper GT2022-82017, Rotterdam, The Netherlands, 2022.

13. Appendix A - Expert Review & Recommendations

Anonymous A:

I am aware that SwRI has been pursuing a DOE funded project to test an sCO_2 compressor stage with an IGV designed for these small, high power density stages. A basic overview of the program has been provided, but the complete details of the test geometry and results have not been shared. Based on the information provided by Dr. Wilkes it appears that the project was successful in completing a substantial amount of additional testing that will help better define the challenges in operating a practical sCO_2 compressor stage near the critical point. For IGC's, capacity control is primarily achieved through the use of IGVs. Unfortunately, there is little commercial experience with IGV's at these pressures and operating near the critical point. First, the program appears to have validated the mechanical integrity of the IGV used in the testing which gives confidence to the marketplace that robust mechanical designs are possible at these conditions. Second, there are substantial concerns and questions about operation near the dome. The data collected through this program showed that the compressor was able to operate with rotodynamic and aerodynamic stability across a very broad range of conditions including inside the dome.

Test data for compressor operation near and inside the dome is especially valuable to researchers and OEM's since simulations in this area is difficult to make due to the multiphase flow characteristics and have greater uncertainty than evaluating operation in single phase regions. This testing should help provide additional insight into the off-design operating characteristics of compressors. Dr. Wilkes indicated that substantial variation in the choke flow margin was observed. This was predicted based on bulk changes in the gas properties but was likely also impacted by local changes in phase. Understanding these dynamics will be critical to validating numerical tools used to design this type of equipment and in developing operating and controls strategies to ensure robust operation of the machinery. Previous researchers had shown that there may be some unusual characteristics of the choke line due to the changes in gas properties in the throat of the compressor. This test data may help validate those findings and the software tools used to generate them. It was noticed that small impacts on the stall margin were also observed. Stall in a violent dynamic event that control systems must be developed to avoid stall to protect the machinery from serious damage. Data that can help ensure systems can be designed to protect the machinery during typical commercial operation is critical and the results of this program should further enhance confidence in the technology and industry acceptance.

Anonymous B:

Future Recommendations:

1) Test across the entire dome since low ambient temperatures may want the cycle to perform at the left.
2) Consider reduced-speed tests. The STEP compressor's commercial warranty limits in-dome operation to low/idle speeds, and a detailed characterization of multiphase flow effects at this condition could be valuable to understanding and increasing the allowable speed for other similar situations.
3) Test another wheel geometry (not just a fully shrouded Hanwha impeller) and perhaps a radial inlet to understand the sensitivity of results to different designs.
4) Conduct testing with different piping arrangements and a separator to interrogate the effect of different multiphase flow regimes at the compressor inlet
5) Within your recommendation #5, I would be explicit that you will develop and validate a map-based surge control system optimized for near-dome compression (e.g. smooth interpolation between temperature-dependent maps or collapsing the maps to temperature-independent parameters). I don't know what specific ways you'd want to improve upon what's done at STEP, but I believe their setup has been limiting.
6) For recommendation #3, consider an initial/intermediate teardown and inspection rather than just one at the end.
7) I don't remember what inlet/exit measurements you used (I think there is a suction densitometer?). Is it worth recommending additional instrumentation schemes for performance?
8) Is there anything interesting in your high-freq dynamic pressure data that should be explored further?

Tim Held:

My general comments on the work (now that I know which program I’m talking about!):

- Once published, these will be valuable data sets to help derisk compressor operability and performance questions/issues.
- The relative insensitivity to near-dome and in-dome operation is surprising and will help with operational flexibility.
- The use of variable IGVs is a valuable addition to the system operation “toolkit” to help optimize system performance and reliability.
- The publication of actual dimensional results will be a welcome addition to the literature and should serve as an example for other projects.

Some thoughts on future activities:

- Post-test teardown and condition assessment.
- Expansion of the test window to lower inlet temperatures and pressures. The further the compressor can operate down and to the left on the PH diagram, the more efficient the power cycle becomes.
- Development / validation of compressor design tools based on the new data set.
- Investigation of using UPPs in commercial service as stall/surge precursors and active control systems.
- Combine the test experience with off-design system modeling, with a particular emphasis on part-load operation / efficiency improvement strategies.
- Transient modeling and control system development, validation and optimization using the initial data set. Further test conditions to be guided by the transient model to fill in any gaps.

Karl Wygant:

The DOE program on testing of supercritical CO_2 Inlet Guide Vanes (IGVs) within an axial-inlet radial-discharge centrifugal compressor provide valuable data to the turbomachinery industry as a whole. The team successfully operated the IGVs under challenging conditions, including near-dome, two-phase, and super-critical states, achieving one of the highest inlet densities (up to 600 kg/m3) and corresponding fluid momentums ever recorded for this type of component. Over 80 hours of near-dome operation provided a comprehensive data set of 103 performance test points, validating that the IGV's were highly effective at altering the compressor's head-flow characteristics. Preliminary analysis of the data sets confirms that IGVs are a viable control mechanism for fixed-speed operation, allowing operators to maximize unit efficiency or flexibility.

Analysis of the collected performance data confirmed peak isentropic efficiencies above 80% across nearly all IGV settings, with a maximum recorded efficiency of 86.2%, demonstrating a 2-point improvement over the baseline performance. Efficiency was shown to be acutely sensitive to fluid properties, dropping significantly, by up to 10.4 points, under multiphase suction conditions although the compressor maintained stable operation without excessive noise or vibration. Dynamic pressure measurements revealed that reducing the IGV angle increases flow turbulence and sub-synchronous fluctuations, while multiphase conditions unexpectedly dampened these instabilities near the impeller inlet due to higher fluid density. Based on these accomplishments, future development should be centered on leveraging the newly derived, real compressor maps to perform closed-loop system modeling, advancing specialized dynamic surge control methodologies, and conducting forensic hardware inspections to ensure the long-term reliability of sCO_2 power cycles.

The accomplishments of the testing are valuable to: developers of sCO_2 power cycles, carbon capture and sequestration machinery, enhanced oil recovery applications, and the turbomachinery industry in general.

Anonymous C:

First congratulations to you and the team for a successful test. I was talking with Evan about the high frequency measurements. I am interested to see some of the data when it is reduced. I would add the word “variable” when you mention IGV, since that is what is unique. Your data operating in the dome is very interesting and will be a great benchmark case. If I understood your configuration, you have both the casing treatment and variable IGVs. Are they complimentary? Your turn-down numbers look good. Did you do testing at hot conditions too (i.e. 50°C) to remove the multi-phase effects? I see you would like to test a fully shrouded configuration in the future. I think that is a good idea. You don’t mention blade leading edge modes, but I assume that you looked at them. That is one potential drawback of variable IGVs. Future tests with either strain gauges or tip timing to measure the blade response would be interesting.