LP CO_2 Compressor
Final Technical Report

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Submitted by:

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Abstract

This report summarizes work performed by Dresser-Rand, A Siemens Business and subcontractors, under DOE award number DE-FE0026727, in pursuit of the design, construction and test of a 10:1 pressure ratio “Low Pressure” (LP) supersonic CO_2 compressor suitable for use as a first stage in a two stage 100:1 compression system. This LP compressor, in conjunction with a corresponding 10:1 high pressure compressor, provide the overall 100:1 compression ratio required to compress CO_2 from 22 psia (as generated by a CO_2 capture system) to the 2215 psia required to transport CO_2 via a pipeline to a suitable sequestration site. The compressor demonstrates application of Dresser-Rand’s supersonic compression technology at an industrial scale using CO_2 in a closed-loop testing facility.

This scope of this technical report is to summarize all phases of the program executed from August 5, 2015 to the June 30, 2018. This program included the final testing of a “high pressure” (HP) compressor that had previously been built by Dresser-Rand and the design, fabrication, assembly and test of a corresponding LP compressor. In summary, the LP compressor exceeded the target pressure ratio and efficiency goals for the program at CO_2 flowrates suitable for those expected in coal fired power plant carbon capture systems.

Initial and Final Techno-Economic Analysis reports were generated to show the value this compression solution brings to carbon capture and sequestration (CC&S) systems for the Case B12B coal fired power plant model. Integration of the DATUM S compression technology into a carbon capture and sequestration system was shown to result in a 27% reduction in the Cost of Electricity for the compression system, as well as a 21,500 gallon per hour reduction in required cooling water flow for the plant.

Background

The project was divided into nine separate tasks across three budget periods (BP) as follows:

Task 1: Project Management and Planning, BP1, BP2, BP3

Task 2: HP Compressor Test, BP1

Task 3: Initial Detailed TEA, BP1

Task 4: LP Compressor Design and Analysis, BP1

Task 5: Testing Facility Preparation, BP2

Task 6: Manufacturing and Assembly, BP2

Task 7: Test Plan and HAZOP, BP3

Task 8: Testing and Results Analysis, BP3

Task 9: Final Detailed Techno-Economic Analysis, BP3

The total project budget was $8M, with a 50:50 cost share between Dresser-Rand and the DOE. A summary report was provided as a milestone deliverable at the conclusion of Tasks 2 through 9.

Task 1: Project Management and Planning

Management of the program and engineering design was coordinated by the Olean, NY Research and Development group within Dresser-Rand. The majority of the design work was executed by D-R resources located at the Bellevue, WA and Olean, NY sites. Compressor assembly and test was executed at the Olean, NY site. Component fabrication was primarily subcontracted to US based 3rd party companies. Approximately 40 different Dresser-Rand employees contributed to the program throughout the different phases of execution.

The formal program start date was January 1, 2016 with a target end date of March 31, 2018. Pre-award costs were allowed back to August 5, 2015 to include the HP compressor test executed in that time frame. The total contract schedule was therefore 138 weeks. A no-cost extension of twelve weeks was requested and awarded at the beginning of 2018, shifting the end of BP3 to June 30, 2018. The program was successfully completed by June 30, with a final period of execution of 150 weeks, with a schedule performance index of 1.086 (i.e. 8.6% behind schedule relative to the initial target).

From a budget perspective, the total spending on the program was $8,131,107. Relative to the $8,000,000 initial budget target, the cost performance index was 1.016 (i.e. 1.6% over the initial budget target). The $131,107 cost overrun was funded by Dresser-Rand.

Task 2: HP Compressor Test

In March of 2015, Dresser-Rand began a series of tests with the first single stage supersonic inducer design, 10:1 pressure ratio "DATUM S" high pressure (HP) compressor, referred to as DS-01-01 (DS = DATUM S, first two digit number refers to a unique inducer design, second two digit number refers to a unique static hardware configuration). This compressor was designed to compress 220 psia CO_2 (as generated by a carbon capture system and first stage CO_2 compressor) to 2215 psia (suitable for delivery to a CO_2 pipeline). Following the successful completion of the DS-01-01 test, an additional test campaign was executed, DS-02-01, utilizing an improved supersonic inducer with a supersonic vaned diffuser design. The DS-02-01 final compressor configuration is shown in Figure 1 installed in the dedicated Dresser-Rand high pressure CO_2 test facility in Olean, New York.

Figure 1 – DATUM S HP Compressor (DS-02-01) Installed in High Pressure CO_2 Facility

HP Test Results

Testing of DS-02-01 commenced on September 15, 2016. Test data at various MIGV setting angles were taken. As expected, the DS-02 geometry outperformed the DS-01 geometry. The increase in overall performance was attributed to the optimization work done using the Oakridge Leadership Computing Facility TITAN supercomputer. The results further validated the merit of using large ensemble jobs to perform intelligently-driven optimization. Test data (solid lines, closed symbols) and pre-test predictions (dotted lines, open symbols) can be seen in Figure 2 and Figure 3 below.

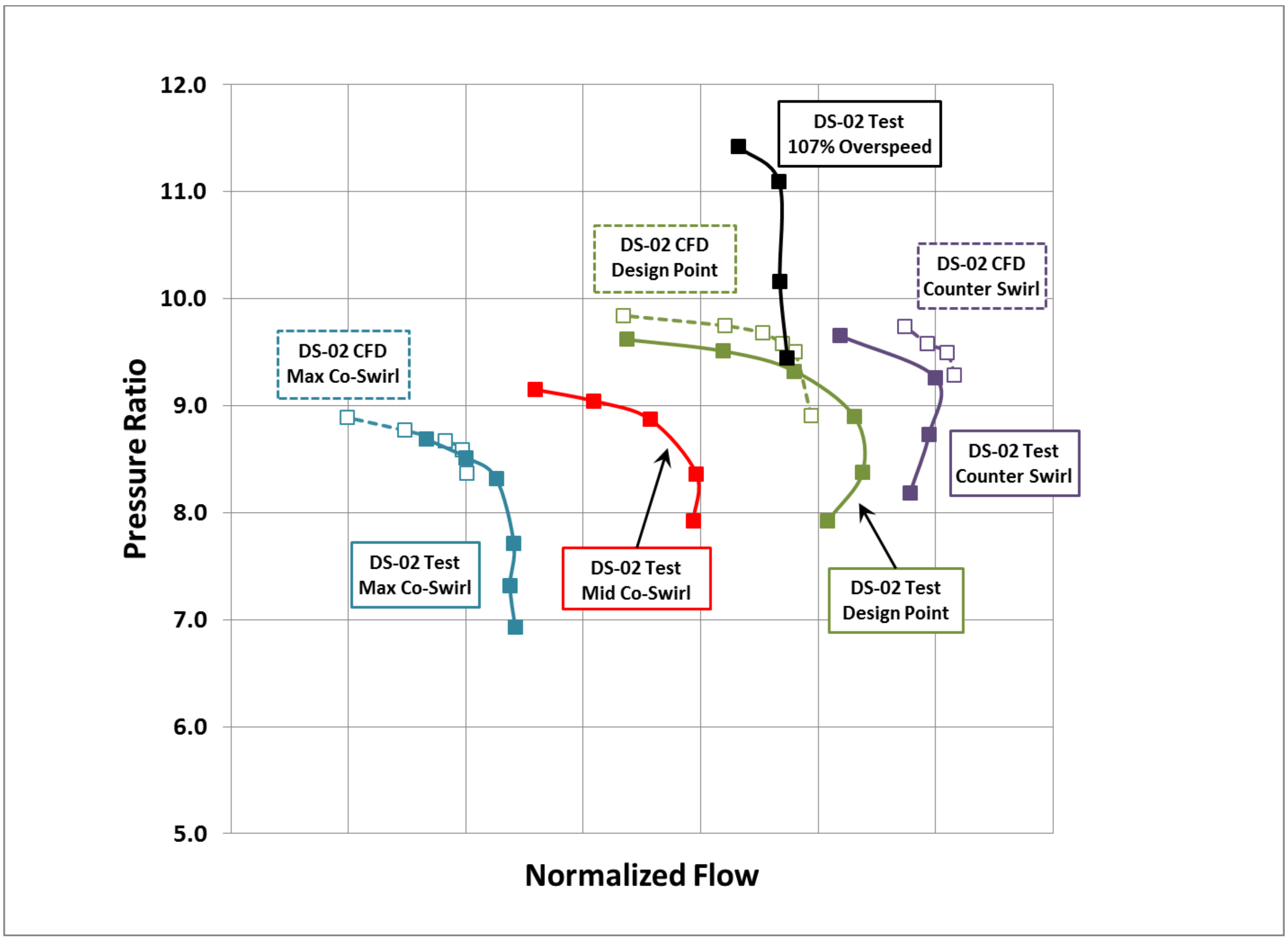

Figure 2 – Stage Performance: Pressure Ratio

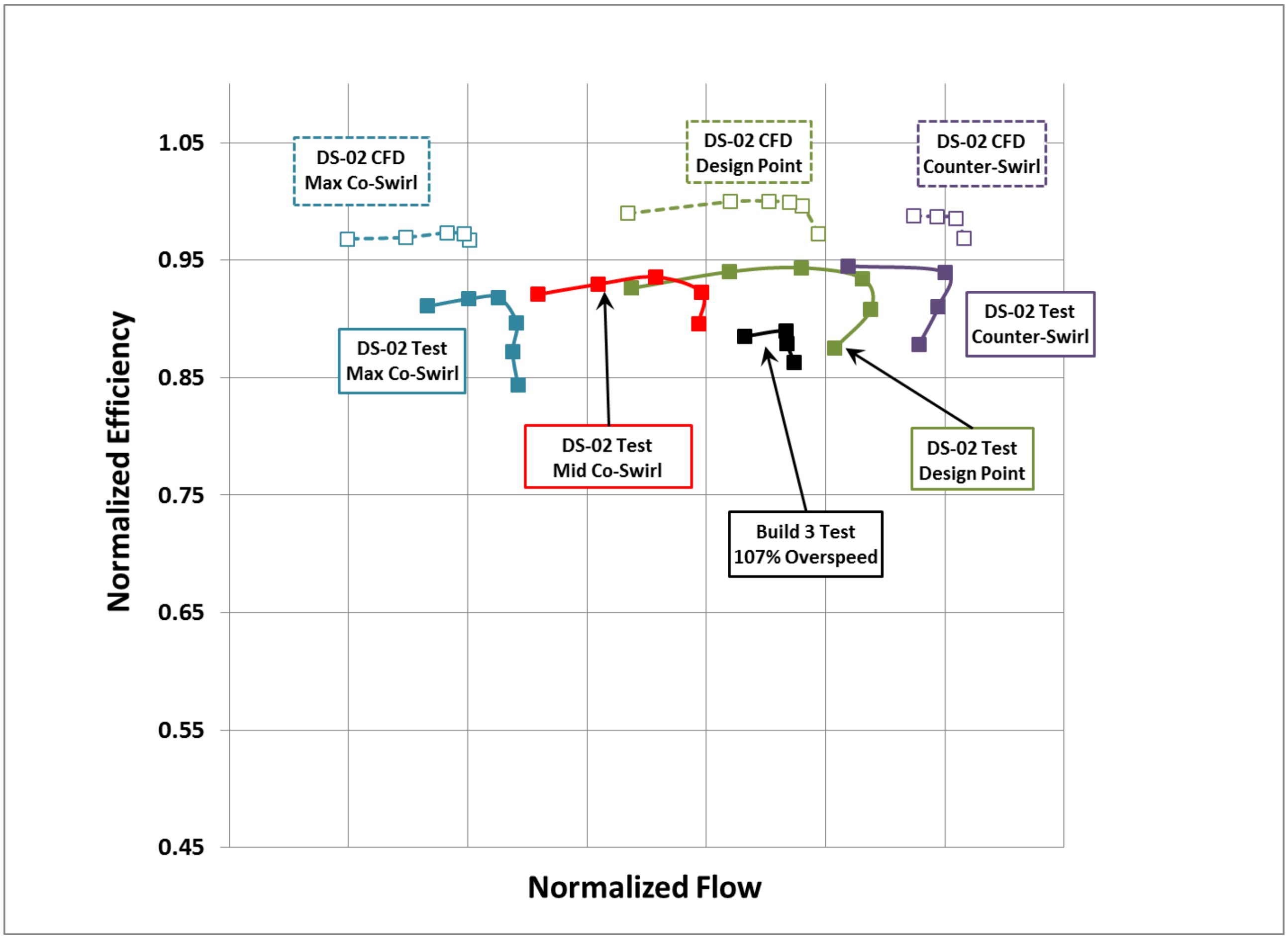

Figure 3 – Stage Performance: Normalized Efficiency

The DS-02-01 data showed improved agreement with the pre-test performance predictions compared to the DS-01-01 results demonstrating that the analytical methods were improving. The pressure ratio and range test results showed excellent agreement with the predictions. The measured efficiency was slightly over-predicted by the pre-test models, but the reason for this is understood and later analytical models captured this effect.

HP Testing Conclusions

The DS-02-01 compressor test program successfully demonstrated the operation of a single stage 10:1 pressure ratio CO2 compressor with discharge temperatures of approximately 550 °F suitable for utilization in a Carbon Capture and Sequestration systems. Testing demonstrated that unit was able to achieve a peak pressure ratio of over 11:1 while efficiency was slightly lower than targeted. Further mechanical and aerodynamic improvements were identified as a result of lessons learned in this test program that were incorporated into the Low Pressure (LP) compressor design/build/test efforts that followed in later tasks.

Task 3: Initial Detailed TEA

The key capability of DATUM-S is the ability to achieve a high (10:1) pressure ratio in a single stage with a compact overall footprint and good efficiency. By combining multiple stages of compression into a single supersonic high pressure ratio stage, DATUM-S generates a significant amount of heat in the compressed gas which can be effectively integrated into a thermal power plant. The purpose of the initial Techno-Economic Analysis (TEA) was to explore the use and benefit of this waste heat to reduce the cost penalty of CO_2 separation and sequestration.

NETL Baseline Case B12B from Cost and Performance Baseline for Fossil Energy Plants, Volume 1a, Revision 3 was used as a benchmark and reference, depicting a supercritical pulverized coal power plant with CO_2 separation and compression. Based on Dresser-Rand knowledge of the industry and experience with CO_2 compressors, the Case B12B CO_2 compressor appeared to have optimistic aerodynamic performance. An integrally-geared compressor selection was made which appeared to confirm this observation.

Case B12B shows an overall performance improvement compared to the previous Case 12 version, and was configured in a way that allowed the DATUM-S compressor to contribute some of its full potential. A DATUM-S compressor selection was made to conform to the existing CO_2 drying pressure level, but it is expected that greater economic benefits will be achievable with co-optimization of the various CO_2 separation, dehydration, and compression systems.

An Excel-based tool was created to model the heat integration changes to Case B12B. The model calculated rigorously conserved energy and mass to ensure accurate analysis, but did not perform a full system optimization.

A two-body, three-stage DATUM-S compressor was selected, utilizing an electric motor to drive the single compressor train. The low- and intermediate-pressure (LP and IP) stages were arranged in a back-to-back configuration in one supersonic compressor body, while the high-pressure (HP) stage occupied the second supersonic compressor body. Significant footprint, cost, piping, and foundation costs reductions were achieved with this compressor train configuration, compared to the baseline which utilized two eight-stage integrally-geared compressors.

Waste heat from the DATUM-S stage discharges was used to heat the Cansolv amine solution and boiler feedwater, thereby reducing the amount of steam diverted from the low-pressure steam turbine for this purpose. The displaced steam was therefore available to pass through the LP turbine, generating additional electrical power. In addition, heat integration reduced the circulating cooling water flowrate by over 21,000 gallons per hour compared to the baseline and eliminated nearly 67 MW of thermal energy being rejected by the cooling tower.

The Case B12B plant with DATUM-S heat integration generated 3.5 MWe more net electricity than the B12B baseline. Total plant cost was also reduced by almost $15M due to reduced compressor capital and installation cost, and savings in the boiler feedwater system and the cooling tower. These factors combined to reduce the cost of electricity (COE) by $1.17/MWh, a decrease that represents 12% of the baseline COE which is attributable to CO_2 compression. The initial TEA demonstrated the significant benefits which can be obtained through DATUM-S CO_2 compression and heat integration.

Task 4: LP Compressor Design and Analysis

The design effort for DATUM S LP Compressor was started 'at-risk' by Dresser-Rand in January 2016. The DOE contract (DE FE0026727) supporting this effort was formally awarded on March 1, 2016, with a corresponding increase in deployment of engineering design resources to the project occurring at that time.

Conceptual Design (Task 4.1)

The Conceptual Design Phase was concluded on schedule at the end of March and presented to NETL April 8, 2016 at the formal program kick off meeting. A single stage, centerhung inducer configuration with a radial inlet was presented. These design elements deviated from those in the utilized in the HP compressor (overhung with axial inlet), but the aerodynamic design of the supersonic inducer and diffuser are the same. This enables the LP stage to be a drive through unit, where the LP can drive the HP unit on the same drive line.

Preliminary Design (Task 4.2)

The Preliminary Design Phase began on April 1, 2016. The following major systems were evaluated in detail to develop different design options.

- Drivetrain
- Pressure Case
- Compressor Layout/Thermal Mapping
- Shaft/Bearings/Rotordynamics
- Variable Inlet Guide Vanes
- Supersonic Inducer
- Supersonic Diffuser
- Instrumentation

Each system went through a design review to down select the preferred option/approach prior to approval to move into the Final Design phase.

Aerodynamic Design (Task 4.3)

The aerodynamic design work associated with the optimization of the supersonic inducer was captured in this task which was executed in parallel with and in support of both the preliminary and final design phase.

The initial aerodynamic design of the LP inducer followed a classic approach. Preliminary sizing was performed using one-dimensional (1D) bulk flow analyses using the Vista CFD code. Though the performance models embedded in the code had not been tuned for the pressure ratios and Mach number required in this design, the preliminary numbers obtained were sufficiently accurate to provide adequate seed information for the remainder of the design process. The preliminary analyses aimed at designing an inducer that could produce a sufficiently high pressure ratio so that an overall 10:1 flange to flange pressure ratio could be achieved. The design also aimed at limiting the Mach number entering the stationary diffuser and the absolute inducer exit flow angle.

Keeping these design goals in mind, a matrix of inducer geometries was generated across a range of total pressure ratios. Designs that provided the best combinations of aerodynamic characteristics were used to develop full 3-D geometries (i.e., hub, shroud and blade profiles). These 3-D geometries were next assessed using coarse grid CFD runs to fine-tune the meridional and blade geometries and to “weed out” poor designs prior to embarking on a full 3-D CFD simulations.

Computational fluid dynamics (CFD) played a key role in the tuning of the inducer aerodynamic design. Analyses were conducted using the Numeca Fine/Turbo code and Ansys CFX. Typical CFD simulations were conducted using domain meshes containing 10 to 80 Million grid points depending on the simulated domain size and accuracy required. For the higher fidelity simulations grid size and wall clustering were selected to ensure that the CFD simulations provide accurate results for the near-wall effects; i.e., turbulence and boundary layer effects.

As the aerodynamic design of the inducer and diffuser often required satisfying conflicting design requirements advanced optimization techniques were used in the design of the compressor. An overview of the optimization process is illustrated in Figure 4. Starting at the left in the figure, the image describes various steps accomplished during the process of optimization including: a) decisions on variable selection and range specification, b) using scripts and templates to generate geometry, grids and setup files for simulation execution, c) launching simulations on thousands of computer cores to evaluate the performance of all geometries, d) evaluation of responses for each geometry with respect to various input parameters, and e) generation of optimized samples and evaluation of responses. The process is repeated until the optimization converges. These processes are explained in more detail below.

The process of optimizing the compressor geometry begins with the generation of a parametric model of the flow path where 30-50 parameters are used to describe the entire inducer and diffuser flow path geometry. This initial parametrized geometry is then perturbed to generate a large database consisting of thousands of samples of derived geometries that are analyzed using CFD tools. The perturbed samples are selected to uniformly cover the design space.

For each perturbed sample several CFD simulations were conducted to determine their aerodynamic performance from choke to surge. The simulations were collected in large ensembles which were launched in parallel on the Oak Ridge Leadership Computation Facility (OLCF) Titan supercomputer. Typically the database generation required running tens of thousands of CFD simulations. Ensemble simulations were of such magnitude that in some cases they required utilization of 50%-90% of the 290,000 cores available on the Titan supercomputer.

Post-processed information from these simulations were analyzed using surrogate functions describing the response of key compressor design parameters; e.g. efficiency and range. Different sets of goals and constraints were specified to focus the optimization process in areas of interest in the design space. An iterative process was then used, which generated new geometry predictions via genetic optimization techniques to be evaluated using the same procedure as the initial database samples. Each iteration of the database analysis generated hundreds of additional geometries. Information from the original database, along with results of samples from optimization iterations, was re-analyzed at each intermediate step to refine the surrogate function accuracy and to predict a new series of optimized geometries. The optimization process was terminated when sufficient convergence had been reached. In the high-dimensional setting, based on the team's experience with prior optimization projects, a typical minimum number was 10-20 design iterations, i.e. 10-20 updates of the database.

Data processing scripts were developed to automate execution of the large number of simulations required for database generation. The inputs to these scripts were perturbed values of parameters of the geometry model and any associated simulation inputs, e.g. RPM. The scripts enabled generation of database geometry and grids in parallel using thousands of computer cores. Solution execution was accomplished in stages, given system and time constraints. A typical execution would be a launch of thousands of runs comprising of perturbed geometries at different operating points. Each simulation was run in parallel on hundreds of cores. Successful solutions were automatically identified and post-processing of these results was executed in parallel. Offline optimization and refinement of the database was performed as outlined in the previous paragraph. Both inducer efficiency and inducer range are improved by the optimization process.

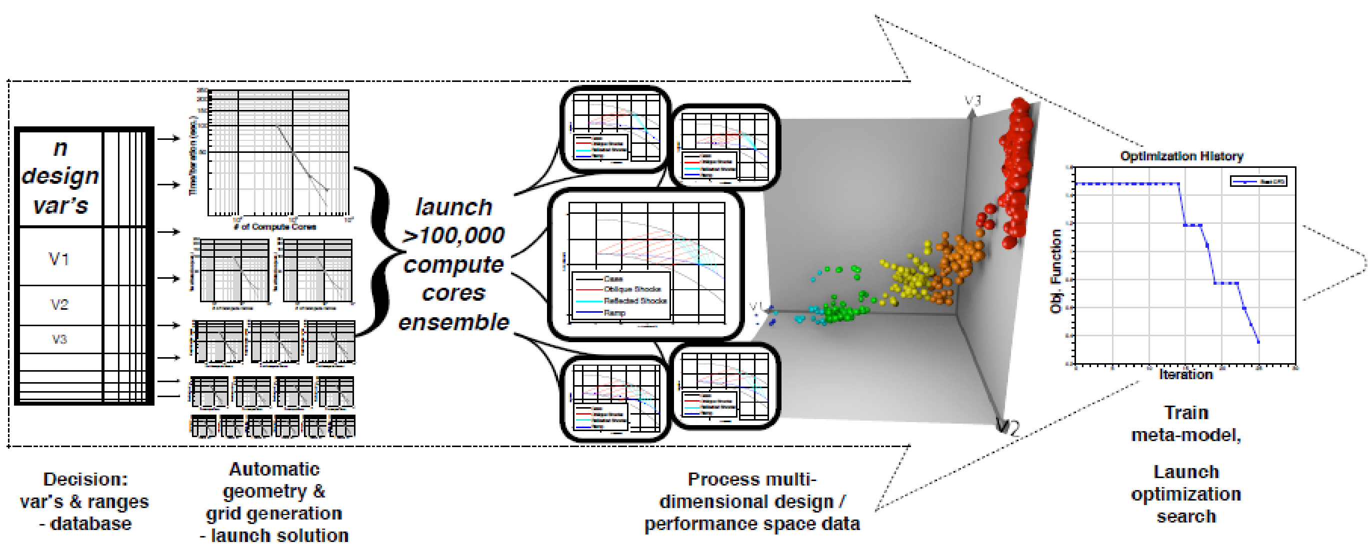

Figure 4 - Workflow of Database Ensemble Computations, Supporting Design Optimization

The flowfield data generated by CFD simulation of the inducer was then used to design a vaned, radial diffuser. Particular attention was paid to the design of the diffuser inlet region to accept the elevated absolute flow Mach number at the inducer exit and to efficiently slow down the flow to subsonic velocities prior to entering the volute. The last component of the aerodynamic flowpath is a radial volute. Standard aerodynamic design practices were followed in the design of this component.

Figure 5 shows pressure ratio and normalized efficiency predicted via CFD for the compressor design. The flange to flange performance estimates presented include a correction factor to account for differences between CFD prediction and measured test data observed during the HP (DS-02-01) compressor testing campaign. The effects of flow leakage form the inducer balance piston are accounted for as well through a leakage term. As these results include separate CFD estimates form individual components, interaction between the components and their effects are not captured in this results. Prior to testing, Dresser-Rand conducted a flange to flange coupled CFD simulation for the entire compressor starting from the inlet, and including inducer, vaned diffuser, volute, and secondary flow passages. This simulation provided the pre-test performance prediction estimates to be compared to experimental test data.

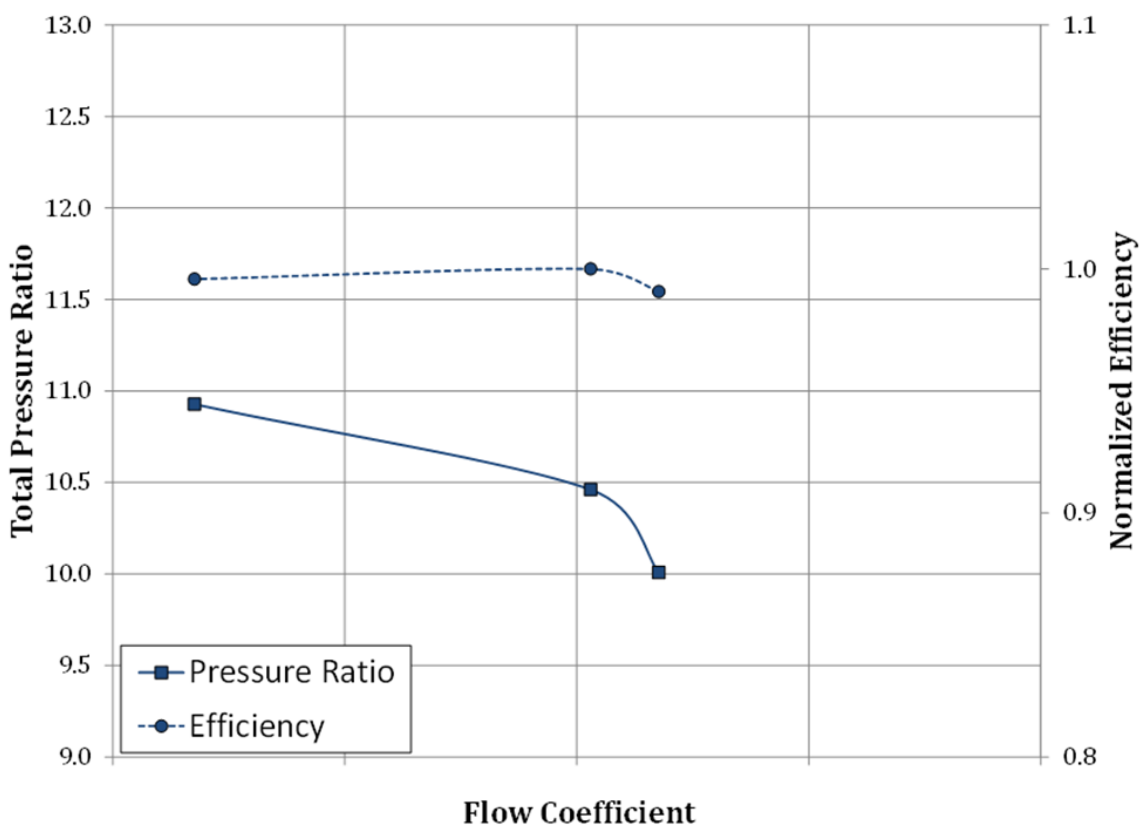

Figure 5 – Predicted Flange-to-Flange Pressure Ratio and Normalized Efficiency

Final Design (Task 4.4)

The Final Design Phase began on October 1, 2016. With the down select of the preferred design options completed as part of the Preliminary Design Phase, the team was able to focus on the more detailed design and analysis tasks for each subsystem and individual components. CFD simulations using FineTurbo, coupled structural/thermal analyses using ANSYS and rotordynamics analyses using internal codes have all been completed to validate the overall design.

Pressure Case

A "D18" casing (using standard Dresser-Rand nomenclature) with stainless steel construction for wet CO_2 compatibility has been designed (see Figure 6). Externally the pressure case uses industry standard mounting connections and nozzle connections to piping.

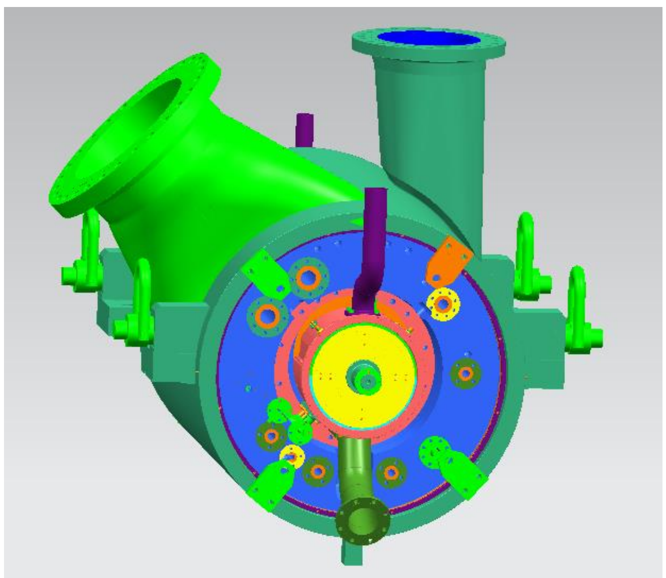

Figure 6 – Pressure Case CAD Model (Inlet Nozzle at 45 deg, Discharge Nozzle Vertical)

Compressor Layout/Thermal Mapping

A "centerhung" compressor configuration (with the inducer supported between journal bearings) was selected to maximize rotordynamic stability and enable "drive through" configurations of the compressor (such that the LP and HP stage could be driven in series by a single driver as part of an overall 100:1 compression system). Detailed thermal maps of the compressor were developed by applying fluid velocity and temperature conditions from the CFD simulations to coupled structural/thermal models. These simulations were important to validate the overall layout of the components and their ability to maintain the desired clearances between rotating and static components.

Variable Inlet Guide Vanes

Moveable inlet guide vanes (MIGV), shown in Figure 7 and Figure 8, were incorporated to change the angle of the gas flow into the inducer and extend the operating range of the compressor. The moveable vanes in the inlet guide are actuated via a connecting rod attached to a unison ring. The vane assembly was designed for a range of gas co-swirl and counter-swirl. The unison ring is rotated by an actuator rod that penetrates the pressure case and connects to a linear actuator. The actuator is calibrated to correlate the position of the rod to specific vane setting angle.

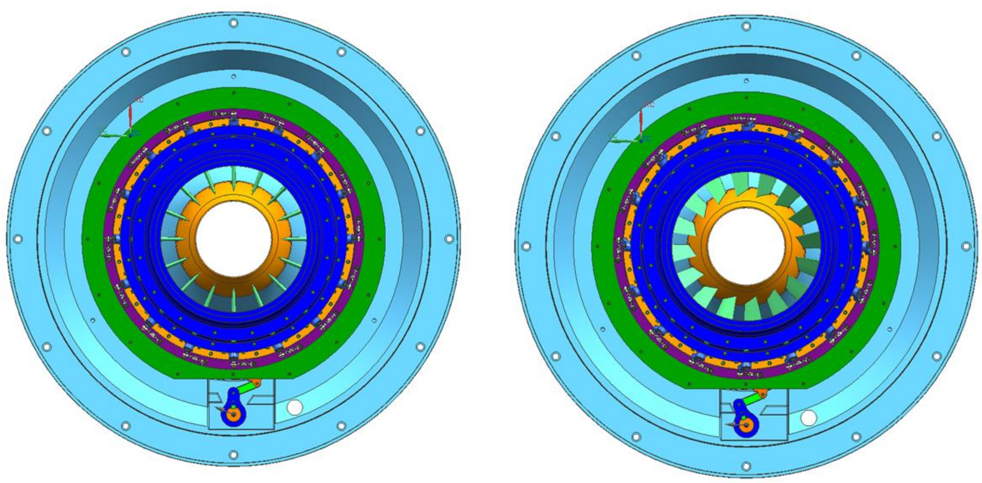

Figure 7 – Inlet Guide Vane Assembly

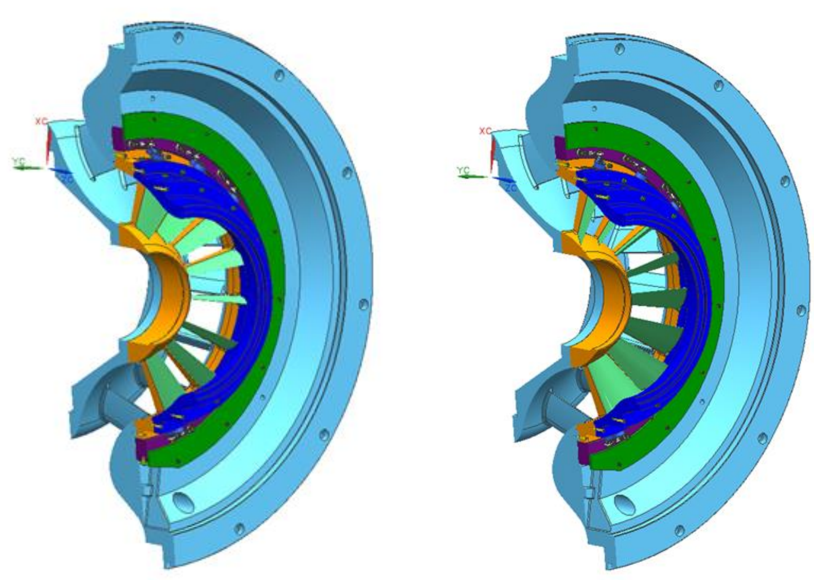

Figure 8 – Inlet Guide Vane Assembly (Isometric Cut-Away View)

Shaft/Bearings/Rotordynamics

The compressor shaft has a two piece construction, an approximate 9 inch diameter and be made from 4340 steel to maximize stiffness for rotordynamic stability. Journal (qty 2) and thrust (qty 1) bearings will be of largely conventional tilt pad, oil lubricated design. The rotor/bearing system is expected to be compliant with API (American Petroleum Institute) rotordynamic requirements which will greatly aid customer acceptance of a DATUM S commercial product line which features higher speed operation than conventional industrial compressors.

Supersonic Inducer

See the Aerodynamic Design (Task 4.3) section for a description of the design process utilized in the aerodynamic design of this component. Once the aerodynamic design was established, structural and thermal analyses were completed to validate acceptable stress levels and deflections. Hot-to-cold conversions were executed to account for the centrifugal and thermal growth of the inducer under steady state operation. This was important to achieve the required running clearances between rotating and static components. Modal analyses were also completed to evaluate system natural frequencies and forcing function.

Supersonic Diffuser

See the Aerodynamic Design (Task 4.3) section for a description of the design process utilized in the aerodynamic design of this component. Once the aerodynamic design was established, structural and thermal analyses were completed to validate acceptable stress levels and deflections.

Instrumentation

The compressor featured an extensive suite of instrumentation to monitor pressures, temperatures, displacements and vibrations internal to the compressor pressure case and rotor shaft system. In all, 137 separate instruments internal to the compressor were monitored during testing.

The main flow loop and associated secondary flows comprising the test loop were also instrumented with pressure, temperature and mass flow instrumentations. The test instrumentation were in compliance with the ASME (American Society of Mechanical Engineers) PTC-10 (Performance Test Code on Compressors and Expanders) which is the industry accepted standard for industrial compressor testing. In all, an additional 61 separate instruments on the flow loop were monitored.

Task 5: Test Facility Preparation

The DATUM S HP test facility was designed specifically for testing high pressure compressors in the 10 MW power range with discharge pressures up to 2400 psia. Designed as a corresponding “low pressure” stage with 22 psia suction pressure, the DS-03-01 compressor had lower nominal discharge pressure of 220 psia. The pipe sizing of the HP test facility was determined to be undersized for use with the higher volumetric flow rate of the DS-03-01 compressor compared to the prior DS-02-01 compressor that was tested. Therefore, a commercial production test stand (with associated drivers, gearboxes, loop piping and heat exchangers) was identified in Dresser-Rand's Olean, NY based facility to test the DS-03-01 compressor.

Although the test loop had to be built (from existing components within the Olean commercial operations inventory) to specifically meet the test requirements of the compressor, this was effectively the same approach D-R takes with all the in-house testing of commercial compressors prior to delivery to a customer. Assembly of the test loop began in December, 2017 and was completed in March, 2018. A photo of the test facility assembly as of early February 2018 is shown in Figure 9.

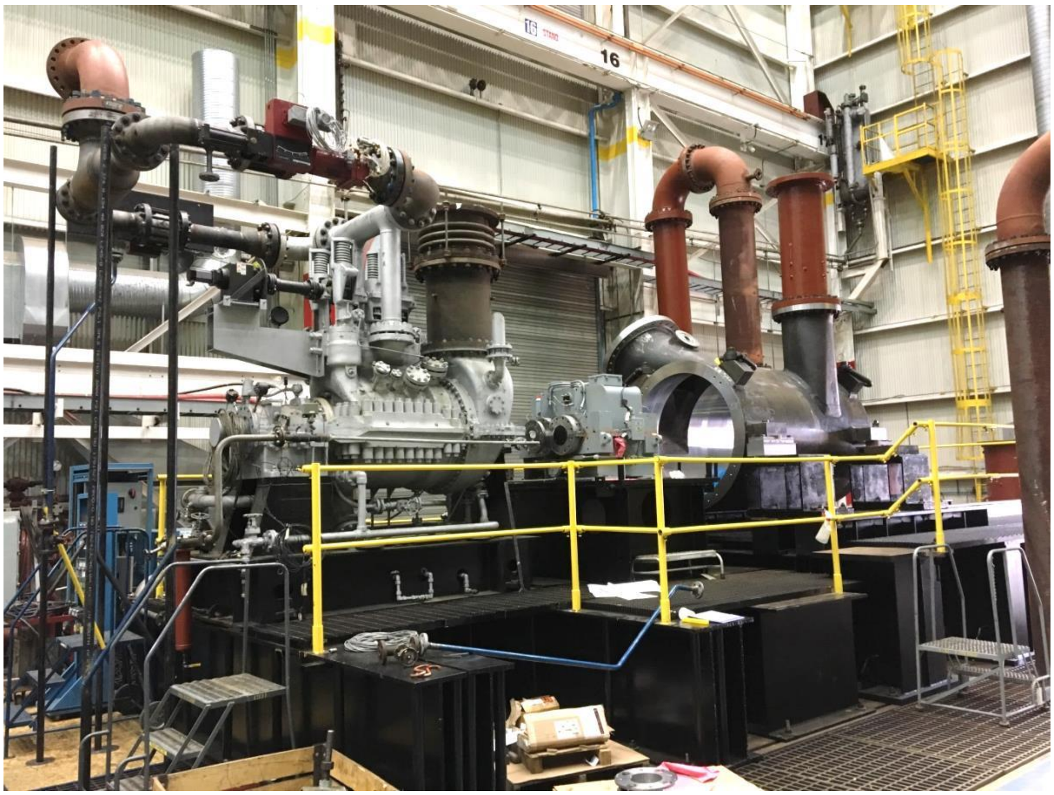

Figure 9 – DS-03-01 Compressor Test Stand

The compressor was driven by a steam turbine driver as can be seen in the left center of the photograph. A gearbox to increase the driver speed from the steam turbine output shaft speed to the required compressor input shaft speed is in the center of the image. The compressor case (with a portion of the discharge piping attached vertically) is shown in the right center of the image. The compressor "bundle" was still being assembled and had not yet been installed into the pressure case at the time of this photo.

Drivetrain

A 20,000 hp (14.9 MW) steam turbine driver (see Figure 10) capable of operating in the 3,000 - 6,200 RPM speed range has been selected. This driver in conjunction with a speed increasing parallel shaft gearbox provides the necessary compressor shaft speed.

Figure 10 – Steam Turbine Driver and Gearbox

Test Loop

The process flow diagram for the compressor test loop is shown in Figure 11. The CO_2 was recirculated in a closed loop from the compressor to a cooler to reject the waste heat of compression (C-001), then through a back pressure throttling valve (PV-001A) and back to compressor suction. Numerous branch circuits provided control of leakage flows, thrust balance control, loop filling/venting and coarse/fine throttling valve control. Secondary flow control systems for bearing oil, water cooling and dry gas shaft seal systems are also shown.

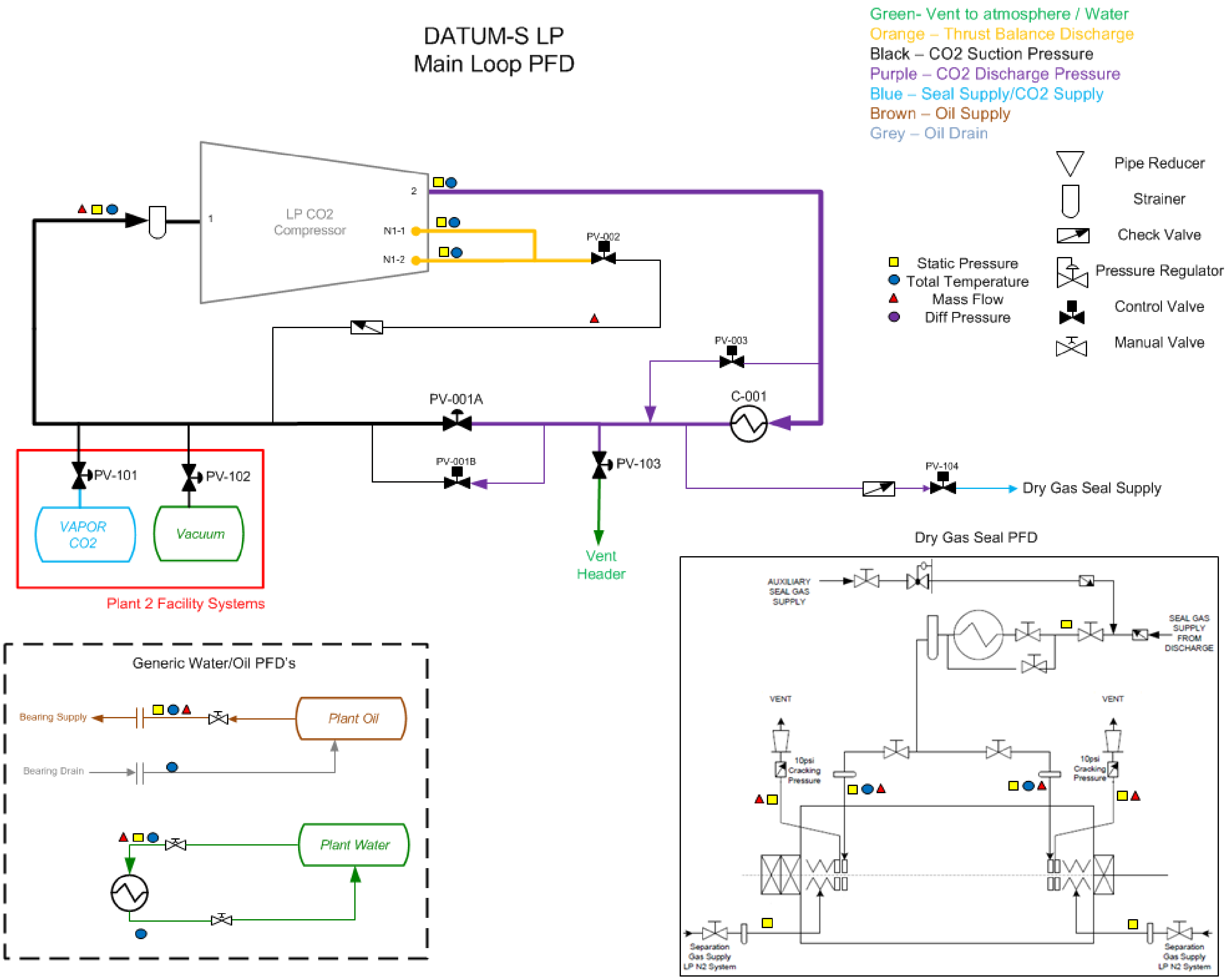

Figure 11 – Test Loop Process Flow Diagram

As previously mentioned, to handle the large volumetric flow of the DS-03-01 compressor, large pipe diameters are required in the flow loop. The 30" diameter suction piping assembly is shown in Figure 12. As a standard practice, this piping assembly was installed after the compressor bundle has been test fit into the case. Figure 13 shows other loop components awaiting final compressor assembly completion before installation: the instrumentation spool for the suction line (vertical pipe), a section of discharge pipe (horizontal) and the high pressure discharge throttling valve with actuator (green).

Figure 12 – Suction Piping Assembly

Figure 13 – Suction Line Instrumentation Spool; Discharge Piping and Discharge Throttling Valve

The loop cooler (C-001) was sourced from existing Olean Operations test equipment inventory. Hydro testing was completed to ensure structural integrity and to confirm no leaks between the CO_2 gas side and water side of the internal heat exchanger assembly. The cooler is shown in Figure 14 installed one elevation level below the main compressor test stand.

Figure 14 – Flow Loop Cooler

Task 6: Manufacturing and Assembly

Compressor Fabrication

The design of the compressor proceeded through Conceptual, Preliminary and Final Design phases with engineering work concluding in Q4 FY17. Final drafting work continued into Q1 FY18 on remaining short lead items. Initial ordering of raw material for long lead items (most notably the pressure case and supersonic inducer) began in Q2 FY17 before the Final Design phase was completed in order to hold the overall program schedule.

Compressor Subsystems

The following major systems were designed and procured to support compressor assembly:

- Pressure Case
- Moveable Inlet Guide Vanes
- Shaft/Bearings/Seals
- Instrumentation
- Supersonic Inducer and Diffuser

- Bundle Assembly and Tooling

Dresser-Rand's network of suppliers was utilized extensively for fabrication of all components and for sourcing commercial-off-the-shelf (COTS) items. In total, over 700 total parts were ordered, with 508 custom fabricated items and 238 COTS components. As a test prototype, the compressor has features and instrumentation that a commercially offered compressor would not have, therefore a compressor offered for commercial sale would have a lower part count.

Pressure Case

A "D18" casing (using standard Dresser-Rand nomenclature) with stainless steel construction for wet CO_2 compatibility was procured. Figure 15 shows as-received pressure case components. Externally the pressure case used industry standard mounting and nozzle connections to suction and discharge piping.

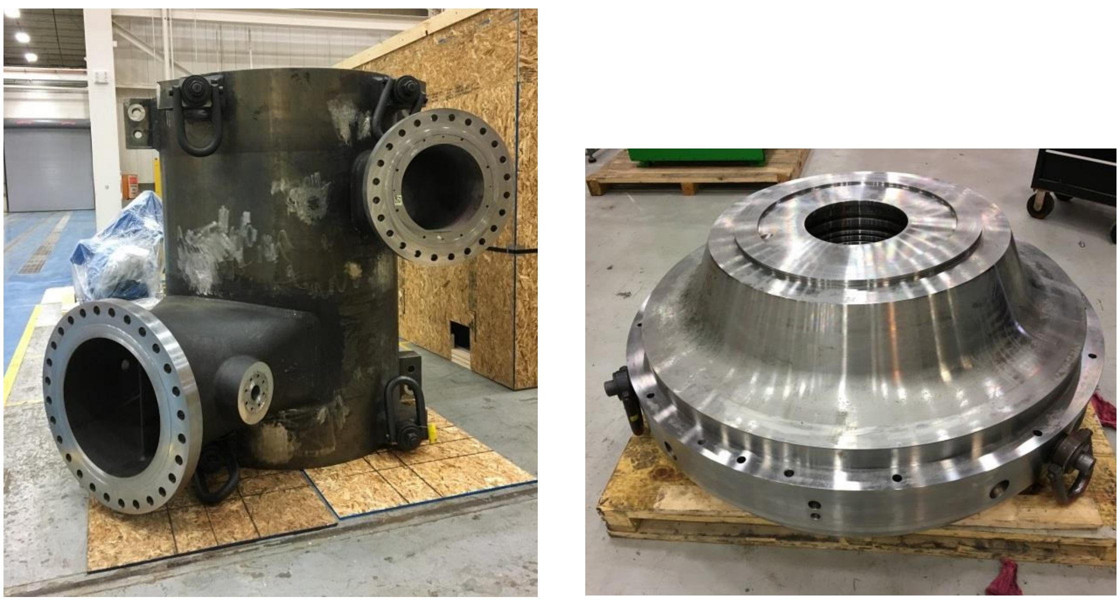

Figure 15 – Pressure Case and Drive End Head as Received from Supplier

Movable Inlet Guide Vanes

Moveable inlet guide vanes (MIGV's), shown in Figure 16, were incorporated to change the angle of the gas flow into the inducer and extend the operating range of the compressor. The moveable vanes in the inlet guide are actuated via a connecting rod attached to a unison ring. The vane assembly was designed for a range of gas co-swirl and counter-swirl. The unison ring is rotated by an actuator rod that penetrates the pressure case and connects to a linear actuator. The actuator is calibrated to correlate the position of the rod to specific vane setting angle.

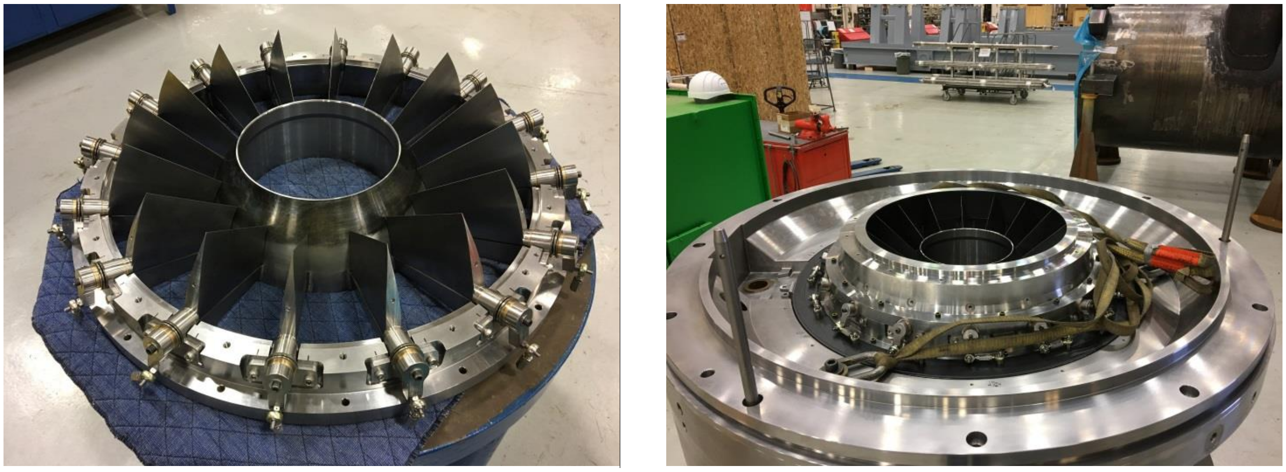

Figure 16 – Inlet Guide Vane Assembly

Rotor Shaft

The compressor shaft has a two piece construction, each with approximate 9 inch diameter. They are made from 4340 steel to maximize stiffness for rotordynamic stability. The drive end shaft is shown in Figure 17. The rotor/bearing system is compliant with API (American Petroleum Institute) rotordynamic requirements

Figure 17 – Compressor Drive End Shaft

Supersonic Inducer and Diffuser

The inducer was mounted to the main compressor shaft. Structural integrity of the inducer was validated via overspeed testing in a spin pit. The compressor shaft/inducer assembly was balanced installed into the compressor bundle. The diffuser was instrumented and installed in the compressor bundle.

Instrumentation

The compressor has a suite of instrumentation to monitor pressures, temperatures, displacements and vibrations internal to the compressor pressure case and rotor shaft system. In all, 137 separate instruments internal to the compressor were monitored during testing. Figure 18 shows the in-progress installation of typical pressure and temperature sensors in a compressor bundle component.

Figure 18 – Component Instrumentation Installation

The main flow loop and associated secondary flows comprising the test loop was instrumented with pressure, temperature and mass flow instrumentations. The test instrumentation were in compliance with the ASME (American Society of Mechanical Engineers) PTC-10 (Performance Test Code on Compressors and Expanders) which is the industry accepted standard for industrial compressor testing. In all, an additional 61 separate instruments on the flow loop were monitored.

Bundle Assembly and Tooling

The bundle was assemble with standard and customized assembly tooling that was designed to meet the specific requirements of the LP CO2 compressor. Figure 19 shows the completed bundle assembly in March 2018.

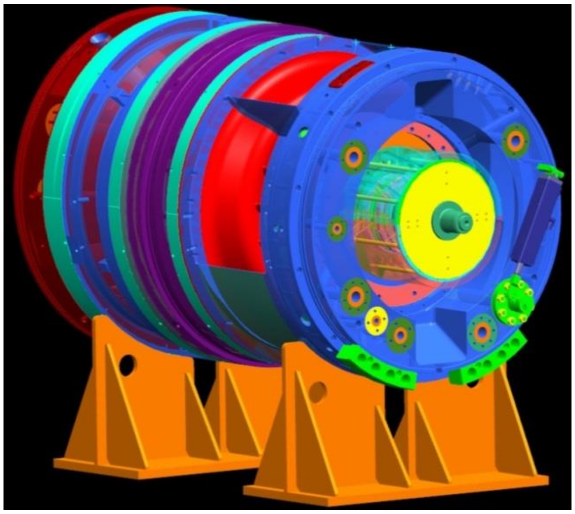

Figure 19 – Completed Bundle Assembly

Task 7: Test Plan and HAZOP

As the final stages of the compressor and test loop assembly (see Figure 20) were being completed, the engineering team developed a detailed test plan and also executed a HAZOP (Hazard and Operability Analysis) workshop to evaluate and mitigate equipment and/or operator risks in the execution of the test.

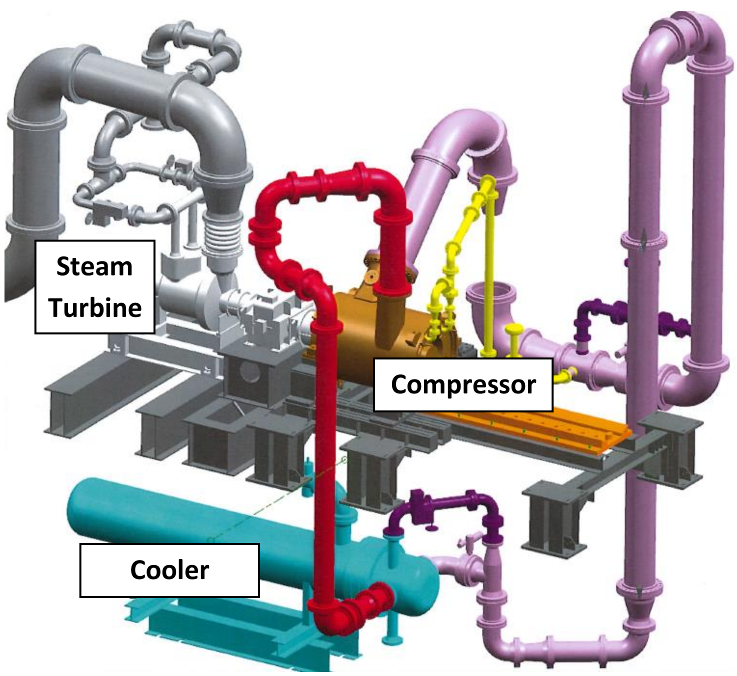

Figure 20 – DATUM S LP Compressor (DS-03-01) Test Setup Model

Compressor Test Plan

Test Objectives

The primary objective of the test was to fully characterize the performance of the LP compressor. The test gas was 100% CO_2 (44 mole weight). The critical calculated aerodynamic performance parameters were the pressure ratio (target of 10:1), the polytropic efficiency and range. As part of establishing the overall aerodynamic performance map of the compressor, multiple operating speeds and inlet guide vane (IGV) setting angles were explored. Per standard operating procedures for large scale industrial compressor testing, the test team monitored numerous internal and external temperature measurements to ensure thermal equilibrium was achieved before collecting performance data.

In addition to the primary test goal of quantifying the overall aerodynamic performance of the compressor, data was also be collected on a number of mechanical test goals (typically to assess the mechanical and/or aerodynamic characteristics of sub-system/component level compressor features). This included data collected on the performance of sealing systems, the rotordynamic stability and behavior of the compressor and the performance of the variable inlet guide vane system.

Test Schedule

A two week test schedule was planned once the final compressor assembly and control/data end-to-end checks were complete. The first two days were a "pre-run" phase in which the test loop is pressurized with CO2 and leak checked. The steam turbine was operated at low speeds in initial check out runs. Numerous data streams were monitored in the flow loop and compressor to ensure test parameters were within expected limits.

Once the initial pre-run testing was complete and the test team has concluded that all systems are operating as expected, the speed was increased to the full design speed. Once thermal equilibrium is achieved, a full aerodynamic performance curve was collected by varying the discharge pressure while holding speed constant. In addition, the inlet guide vanes were actuated to verify mechanical operation. The next phase of testing involved operating the compressor at full design speed and collecting aerodynamic performance curves at a total of four different inlet guide vane settings ranging from co-swirl to counter-swirl conditions.

HAZOP

On February 15, 2018 a HAZOP (Hazard and Operability Analysis) workshop was held to review physical hazards to personnel and equipment associated with the testing of the compressor. As the compressor was tested on a commercial production test stand, the test crew is well versed in the hazards associated with testing full scale compressors at the gas flowrates and power levels similar to those of the DS-03-01 compressor. In total the team identified 32 unique testing hazards, which is typical for most HAZOP workshops with this class of machinery. The hazards were ranked in terms of likelihood and severity and safeguards/mitigation plans were identified for each hazard. Mitigation steps generally included hardware changes (such as adding insulation to shield hot surfaces from operators), procedural changes (ensuring proper ordering of test procedure steps) and automated control system updates (such as identifying and setting alarm set point limits).

Task 8: Testing and Results Analysis

Test Overview

The primary objective of the test was to fully characterize the performance of the LP compressor with 100% CO_2 (44 mole weight) as the test gas. The key calculated aerodynamic performance parameters were pressure ratio (target of 10:1), polytropic efficiency and range. As part of establishing the overall aerodynamic performance map of the compressor, multiple operating speeds and inlet guide vane (IGV) setting angles were explored.

Per standard operating procedures for large scale industrial compressor testing, the test team monitored numerous internal and external temperature measurements to ensure thermal equilibrium was achieved before collecting aerodynamic performance data.

In the first phase of testing the test loop was pressurized with CO_2 and leak checked. The steam turbine driver was operated at low speeds in initial check out runs. Numerous data streams were monitored in the flow loop and compressor to ensure test parameters were within expected limits.

Once the initial test phase was completed the test team concluded that all systems were operating as expected and proceeded to the final testing phase.

Test Results

The test team was able to achieve 100% speed on the second full day of testing. Thermally stabilized aerodynamic performance curves were acquired at 100% and 104% speed as shown in Figure 21. At both operating speeds the 10:1 pressure ratio goal was exceeded. Increasing the speed by 4% demonstrated pressure ratios in excess of 12:1, greatly exceeding the goal.

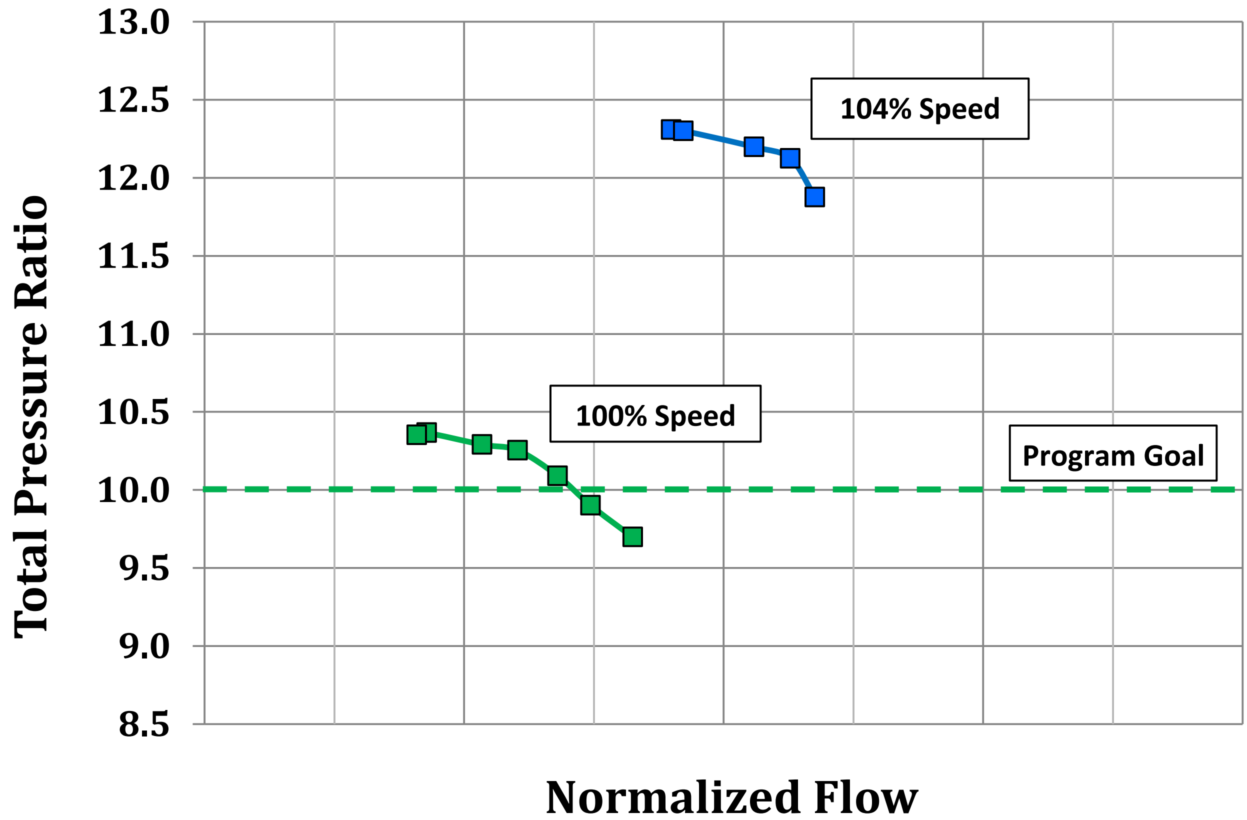

Figure 21 – Stage Performance: Pressure Ratio

The normalized efficiency for the 100% and 104% performance curves are shown in Figure 22. The peak efficiency at 100% speed exceeded the program goal (normalized to 1.00 in the figure) while simultaneously meeting the 10:1 pressure ratio goal. At the 104% speed condition, the efficiency dropped slightly below the goal level, but this level of efficiency drop is consistent with D-R compressor testing experience as the operating speed moves away from the design speed (i.e. more pressure ratio can be achieved through overspeed at the expense of slightly reduced efficiency).

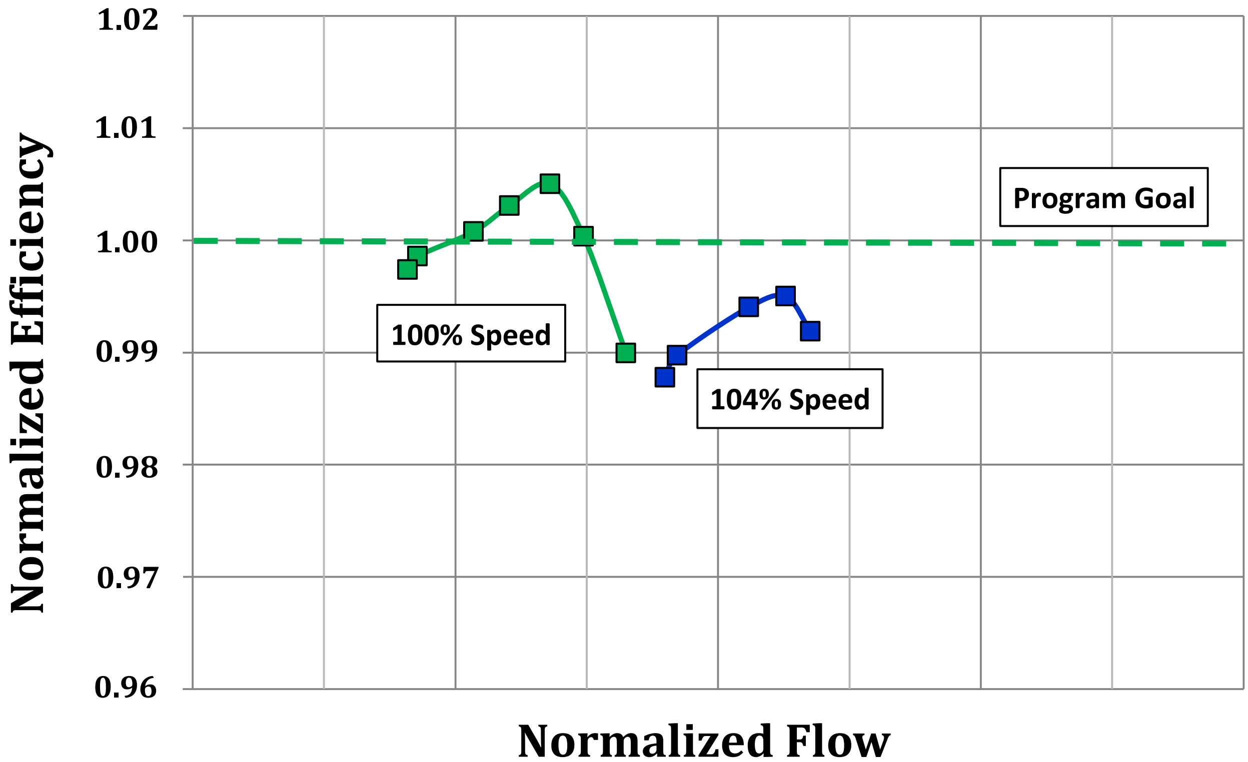

Figure 22 – Stage Performance: Normalized Efficiency

The next phase of testing involved collecting aerodynamic performance curves at a total of four different inlet guide vane settings ranging from the baseline 0 degree setting to 15 degrees of co-swirl. This test was conducted at 104% speed, but could have very well been conducted at 100% speed as well. Figure 23 shows the data collected relative to the program pressure ratio goal. The shift in normalized flow from the 0 to 15 degree IGV setting angle matched well with that expected in the pre-test prediction.

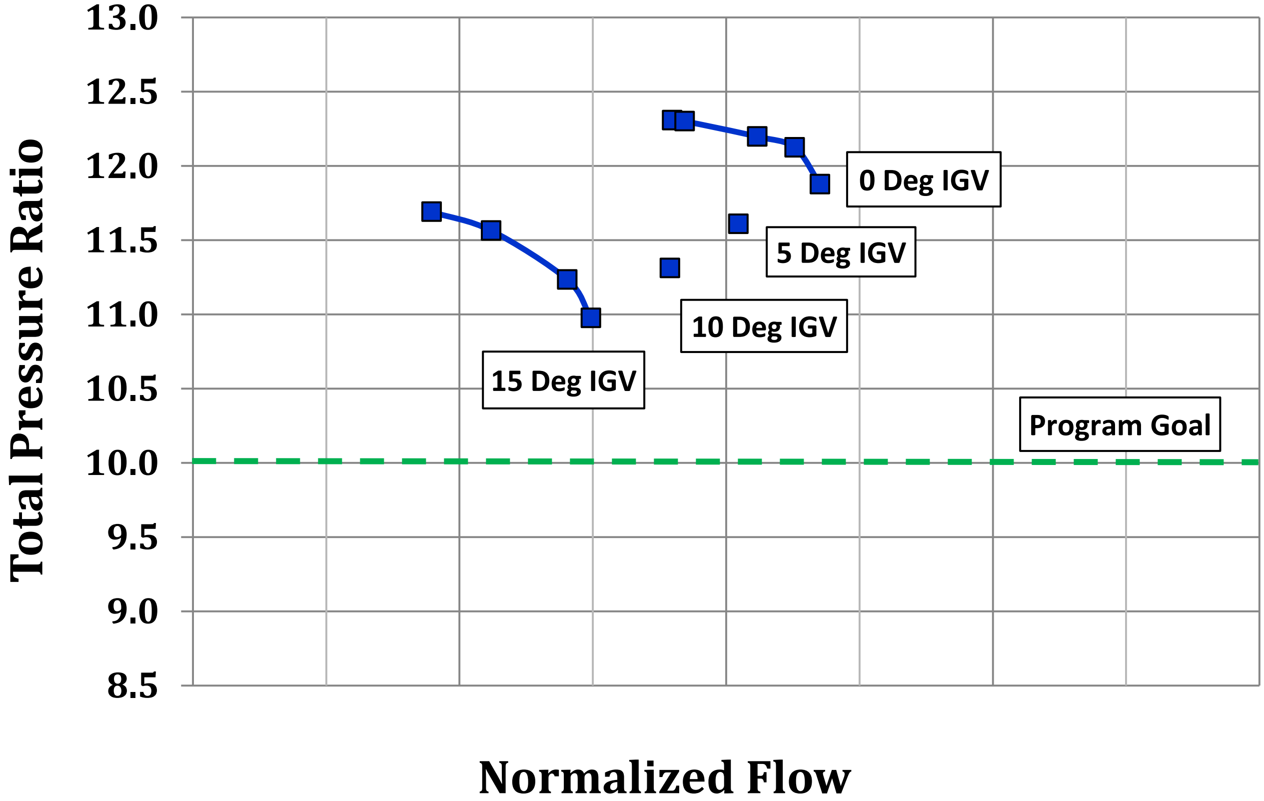

Figure 23 – Stage Performance at 104% Speed: Pressure Ratio at Varying IGV Setting Angles

Task 9: Final Detailed Techno-Economic Analysis

As described in the Initial TEA, the key capability of DATUM-S is the ability to achieve a high (10:1) pressure ratio in a single stage with a compact overall footprint and good efficiency. As a byproduct of combining multiple stages of compression into a single high pressure ratio stage, DATUM-S generates a significant amount of waste heat in the compressed gas which can be beneficially integrated into a thermal power plant. The purpose of the final Techno-Economic Analysis (TEA) was to explore the use and benefit of DATUM-S and waste heat to reduce the cost penalty of CO2 separation, compression and sequestration.

NETL Baseline Case B12B from Cost and Performance Baseline for Fossil Energy Plants, Volume 1a, Revision 3 was used as a benchmark and reference, depicting a supercritical pulverized coal power plant with CO_2 separation and compression to pipeline pressure. Based on Dresser-Rand knowledge of the industry and experience with CO_2 compressors, the Case B12B CO_2 compressor appeared to have an optimistic shaft power specification. An integrally-geared compressor selection was made which confirmed this observation. After discussion of this issue, NETL directed Dresser-Rand to prepare two versions of the analysis presented in this report – one assuming the original baseline B12B, and another assuming B12B's CO_2 compressor is replaced with the commercial selection. The new case is referred to as the Modified Baseline.

Two models were created to simulate the heat integration – one in Excel using NIST REFPROP9 state calls, and one in Thermoflex v27. Thermoflex is a commercial code which can be used to model power plants and novel energy cycles – ideal for thermodynamic analysis of both the baseline and heat integration configurations. Both Excel and Thermoflex models rigorously conserved energy and mass to ensure accurate analysis. Close agreement between the two models was achieved, but Thermoflex model outputs were used for the purposes of this report.

A two-body, three-stage DATUM-S compressor as shown in Figure 24 was selected for the heat integration cases, utilizing an electric motor to drive the single compressor train. The low- and intermediate-pressure (LP and IP) stages were arranged in one supersonic compressor body, while the high-pressure (HP) stage occupied the second supersonic compressor body. Significant footprint, cost, piping, and foundation costs reductions were achieved with this compressor train configuration, compared to the baseline which utilized two eight-stage integrally-geared compressors.

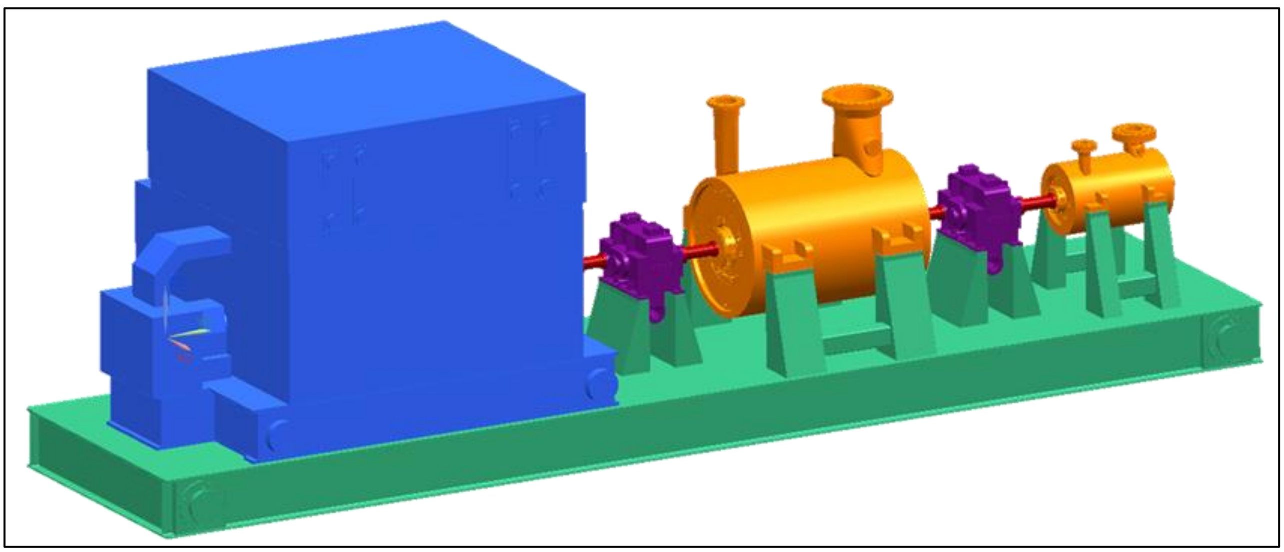

Figure 24 – DATUM S Compressor Train Configuration

Waste heat from the DATUM-S stage discharges was used to heat the Cansolv amine solution at higher temperature followed by boiler feedwater heating at lower temperature, thereby reducing the amount of steam diverted from the low-pressure steam turbine for these processes. The displaced steam was therefore available to pass through the LP turbine, generating additional electrical power. Heat integration reduced the circulating cooling water flowrate significantly compared to the baseline and reduced the plant's overall heat rejection from the cooling tower, which also helped result in a reduction in cooling water flow of 21,500 gallons per hour.

The plant with DATUM-S and heat integration generated more net electricity. Total plant cost was lowered due to reduced compressor capital and installation cost and also from savings in the boiler feedwater system and cooling tower. These factors combined to reduce the cost of electricity (COE). In total, DATUM-S compressors with heat integration reduced the portion of the COE attributable to CO_2 compression by 27% compared to the Modified Baseline B12B Case. This result is summarized in Figure 25 below.

	Case B12B	Case B12B alternate	DATUM S w/ heat integration		
	Baseline	Lower IG comp. perf.	Case DR1	Delta to baseline	Delta to alternate
Gross Power, MWe	641.5	641.5	656.3	14.8	14.8
Aux Load, MWe	91.3	99.7	104.7	13.4	5.0
Net Power, MWe	550.2	541.8	551.6	1.4	9.8
HHV Net Plant Eff., %	32.5%	32.0%	32.6%	0.1%	0.6%
HHV Net Plant Heat Rate, Btu/kWh	10,508	10,672	10,482	-26	-190
COE w/o T&S, \$/MWh	133.17	135.40	132.13	-1.04	-3.27
$\Delta COE/COE_{comp}$, %	—	—	—	-10%	-27%

Figure 25 – Summary of Benefits of DATUM S Integration to Case B12B

Summary and Conclusions

Testing of a 10:1 pressure ratio compressor suitable for CO_2 compression from 22 PSIA to 220 PSIA was successfully completed. The program was executed within 1.6% of the original target budget and 8.6% of the original target schedule.

The measured aerodynamic performance exceeded program goals in both efficiency and pressure ratio. Usable range of the compressor was achieved via actuation of a variable inlet guide vane system while maintaining over 10:1 pressure ratio across all setting angles.

Integration of the DATUM S compression technology into a carbon capture and sequestration system was shown to result in a 27% reduction in the Cost of Electricity for the compression system, as well as a 21,500 gallon per hour reduction in required cooling water flow for the plant.