Design and Testing of CO_2 Compression Using Supersonic Shock Wave Technology

Final Report
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Principal Author(s):
Aaron Koopman

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DRESSER-RAND®

Seattle Technology Center
Dresser-Rand Company
11808 Northup Way, Suite W190
Bellevue, WA 98005

POC: Aaron Koopman
(425) 828-4919 ext. 235
E-mail: akoopman@dresser-rand.com

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Abstract

This report summarizes work performed by Ramgen and subcontractors in pursuit of the design and construction of a 10 MW supersonic CO_2 compressor and supporting facility. The compressor will demonstrate application of Ramgen's supersonic compression technology at an industrial scale using CO_2 in a closed-loop.

The report includes details of early feasibility studies, CFD validation and comparison to experimental data, static test experimental results, compressor and facility design and analyses, and development of aerodynamic tools.

A summary of Ramgen's ISC Engine program activity is also included. This program will demonstrate the adaptation of Ramgen's supersonic compression and advanced vortex combustion technology to result in a highly efficient and cost effective alternative to traditional gas turbine engines. The build out of a 1.5 MW test facility to support the engine and associated subcomponent test program is summarized.

1. Table of Contents

1. Executive Summary

Ramgen designed, built and tested three novel compressor demonstrators capable of providing high pressure ratio in a single stage. Two of the compressors were CO_2 compressors capable of 10:1 single-stage pressure ratio for CO_2 sequestration applications. The other was an air compressor for 6:1 pressure ratio as part of a novel engine configuration, the Integrated Supersonic Component Engine (ISCE). The project demonstrated that Ramgen could accurately model, design, build and test supersonic compression technology as part of a more efficient and less expensive compression option to other technologies available in the foreseeable future.

The largest portion of Ramgen's effort during the award period was devoted to advancing the design for the CO_2 Compressor. Design reviews were regularly held to review progress and confirm all system requirements were being met. Rotor manufacturing offered schedule challenges and delayed start of the test program. Ramgen improved and changed its design configuration for the CO_2 compressor to alleviate performance, manufacturing and schedule concerns. Engineering reviews were held in order to determine the feasibility of the new configurations and to determine if the overall design, budget and schedule goals would be met. Review and concurrence by the DOE was obtained for each significant change in configuration. Testing of the first build CO_2 Compressor proceeded at Ramgen's closed-loop test facility in Olean, NY in Q3 2012 and completed in Q4 2013. Ramgen achieved a peak pressure ratio of 9:1 in a single stage, and a peak discharge pressure of 1547 psia with 210 psia suction.

In 2014 and early 2015 the Build 2 HP CO_2 Compressor design was complete, parts were manufactured, assembled and installed in the Olean Test facility for test start in late March 2015. The latest configuration of the CO_2 compressor represents a developmental improvement that combines the advantages of supersonic flow and Dresser-Rand's commercial compression experience. Testing was successfully completed in May 2015. The test results matched analytical predictions and the DATUM-S configuration delivered the performance starting point predicted prior to test. The configuration will continue to evolve, with increase performance improvements, as the technology moves towards field deployment to a customer site.

Another major effort successfully executed during this award period, was the design, procurement, and commissioning of a 10 MW CO_2 closed-loop facility in Olean, NY. Civil construction progressed well and on schedule. The facility commissioning was not trouble free and required some time to bring to full working order. For such a complex facility this was not unexpected. The facility performed as designed to successfully monitor and control the test of the CO2 Compressor builds.

Ramgen worked closely with subcontractors and vendors to improve our analytical and simulation tools' speed and capabilities. Computational simulations and existing industry data were used to guide the analytical modeling efforts. A significant effort was expended to enable massively-parallel execution of our computational code on Oak Ridge National Laboratories' (ORNL) supercomputers. DOE Secretary Chu arranged for

Ramgen to receive a substantial allocation grant on ORNL supercomputers in conjunction with funding to improve the parallel operation of the code. Ramgen has also extended the functionality of this code to take advantage of the newly expanded Titan supercomputer at ORNL. Ramgen successfully executed two 8-hour runs using 100,000 computing cores and a single run using 240,000 computing cores on the Jaguar supercomputer while maintaining impressive scalability.

In parallel with the CO2 compressor development, Ramgen was simultaneously working on an Integrated Supersonic Component Engine (ISCE) power generation concept that makes use of shock compression technology. Coupled with Ramgen's demonstrated advanced vortex combustor, the technology enables the engine to run leaner and more efficiently than conventional turbomachinery. The shared technology between the ISC Engine program and that of the CO_2 compressor allowed Ramgen to make great strides in the program execution.

The Build 1 ISCE was completed in August 2013. The full flow path was not fully aerodynamically started. Part of the rotating flow path was independently tested. Compared to the analytical predictions, the test results showed good agreement with the predictions for the operating conditions. Due to Ramgen’s agreements with the DOE, budget limitations and resource allocations, all activities associated with the ISCE program had to be completed by the end of June 2014.

Coupled with Ramgen's shock compression technology, the advanced vortex combustor (AVC) enables the ISC engine to run leaner and more efficiently than conventional turbomachinery. The testing conducted in May and June, 2014 focused on the range of operation and the emissions measurements. In summary, the range of operation was quite large due to the flame stability. The emissions were higher than expected due to extra air cooling in comparison to the design intent. Although higher than expected, the NOx emissions are at a comparable level to commercial low NOx systems.

A conceptual turbine nozzle design for the ISC engine application was developed and optimized via CFD and thermal analysis. The test nozzle was configured to run in Ramgen’s Redmond Washington Test Facility. The manufacturing challenges were significant for the high temperature nozzle features like the strake, which required thin walls, complex internal cooling features, and a thin trailing edge. The knowledge gained from engaging and completing the manufacturing was valuable.

Another ISCE component Ramgen investigated was the turbo-expander. The configuration constraints of the turbo-expander were found to significantly limit performance potential. A mechanical and performance based design trade study was conducted to assess the potential of a single stage supersonic engine unconstrained by size (axial/radial) or speed, while still existing within the bounds of practical material selection.

This report documents work performed by Ramgen and subcontractors in pursuit of design and construction of two 10 MW supersonic CO_2 compressor builds, a supporting closed loop CO_2 test facility and the development of engine components for a novel engine configuration based on supersonic compressor technology.

2. Task 2.1 – Requirements and Large Machine Feasibility

Ramgen conducted a comprehensive configuration/feasibility study in late 2009, concluding with a large machine feasibility review. After closing out action items from this review, the Build 1 CO_2 compressor was declared feasible and authorization was given for engineering to proceed into preliminary design. All critical areas were reviewed and approved. Some immediate actions were assigned; these were quickly answered and closed out. The review agenda is included below.

In conjunction with the review, Ramgen selected a rotor configuration family known as SE 01 for the demonstration compressor. Ramgen chose to complete the aerodynamic and mechanical analyses necessary to perform the down selection during the configuration/feasibility portion of the program to reduce program risk and focus the remainder of the program effort on a single rotor family which has been shown capable of meeting our requirements.

To accomplish the design down selection, significantly more detailed work was required than usually expected in a feasibility study. A rotor feasibility study would typically include a simple meanline aerodynamic design analysis, a two-dimensional rotor aerodynamic geometry analysis using method of characteristics, general location and quantity estimates for boundary layer features, and mechanical rotor analyses using general stress formulae with stress concentration scalars applied. In contrast, this feasibility effort also included three-dimensional viscous, real gas Computational Fluid Dynamics (CFD) of each rotor flow path component.

By performing this level of evaluation and selecting a single rotor configuration family early in the program, the program's technical risk was significantly reduced. Dresser-Rand personnel were involved in the critical mechanical design and analyses efforts and provided valuable input regarding best commercial and corporate practice. Ramgen and D-R have developed a very good working relationship enabling access to the design and analysis expertise contained within D-R engineering.

The SE 01 rotor family was selected because it represents the best balance of performance capability and feasibility. Ramgen was now able to proceed into the preliminary and detailed design phases with significantly improved models, analysis techniques, and design tools developed during this effort.

In the review, the mechanical team presented design and analyses demonstrating feasibility for individual systems for the ~13,400 HP Build 1 CO_2 compressor. The remaining design work was significant but deemed achievable in the program schedule and budget. The critical issues were identified and tracked.

Among the concerns for scaling the compressor to 13,400 HP from 3,000 HP were the affordability of the electric motor and variable-frequency drive and the availability of a gearbox at the required speed and power. Working closely with D-R Supply Chain Management, Ramgen was able to show multiple options to meet our budget and schedule requirements. Offerings from Siemens, ABB, GE/Mitsubishi, Direct Drive Services, and Converteam were evaluated. Down selection to the ABB team (ABB and Laurence Scott) occurred shortly after the review.

Development contracts with multiple gearbox vendors produced feasible solutions for parallel-shaft and compound epicyclic gearbox approaches. Down selection to Allen Gears' compound epicyclic design occurred shortly after the review.

The mechanical agenda is presented to show the extent of issues and level of detail presented. After reviewing each system and resulting action items, each system was deemed feasible and ready to proceed into the next design phase.

Feasibility Review Agenda: Mechanical

Rotor Structure
- Stress results from SE 01 analysis, including pressure and centrifugal force (CF) loads
- Thermal analysis results
- Status of composite manufacturing development program and all-metal rotor effort
- Rotor start/stop, life, and safety margin pedigree to be used for design

Rotordynamics
- Results from SE 01 lateral rotordynamics and stability
- Critical factors in achieving satisfactory SE 01 rotordynamics

Seals
- Shaft seal configuration for SE 01 and resulting leakage rates
- Rotor seal configuration for SE 01 and resulting leakage rates

Static Structure Layout
- Journal and thrust bearing configuration for SE 01
- Pressure case, inlet ducting, and outlet ducting
- Variable Inlet Guide Vane (IGV) mounting and actuation, including subcontract approach
- Shock Wave Starting techniques and approach
- Boundary layer control systems

Facility
- Facility Front End Engineering Design (FEED) results and plant layout
- CO_2 closed-loop and Process Flow Diagrams (PFD)
- CO_2 makeup system
- Boundary layer control systems
- Leakage capture & recompression requirements and approach
- Lubrication system

Drivetrain
- Motor & Variable Frequency Drive (VFD) specifications
- Gearbox requirements, development status, fallback plans

High-speed coupling configuration
Controls & Instrumentation
Compressor control approach
Performance instrumentation approach
Diagnostic instrumentation approach
Maintenance and Access
Estimate time to access rotor during test

The Aerodynamic team then presented design and analyses demonstrating feasibility for the aerodynamic components. The remaining design work was significant but deemed achievable in the program schedule and budget. The critical issues were identified and tracked. After reviewing each system and resulting action items, each system was deemed feasible and ready to proceed into preliminary design.

The Aerodynamic agenda is presented to show the extent of issues and level of detail presented. Current supersonic ramp CFD models had advanced sufficiently to give confidence the design would achieve the necessary flow quality. More work was necessary to reduce flow distortion, control separation and minimize bleed but Feasibility goals had been met - further work was appropriate for the Preliminary and Final design phases.

Diffuser CFD models appeared to show sufficient performance to meet program goals. These models would be enhanced in future work as the detailed design progresses.

Feasibility Review Agenda: Aerodynamic

Inlet Guide Vane (IGV)
3D real gas properties CFD for SE.01 IGV (and others) with realistic inflow conditions
Rotor Performance
3D real gas properties CFD for shock compression, exducer, boundary layer features
Boundary layer control systems
Future optimization approach for SE.01 family
Exducer and Diffuser
3D real gas properties CFD for SE.01 exducer and diffuser with realistic inflow conditions
Michigan State University diffuser development/test plan
SPIT (System Performance Integration Tool)
High-level overview of SPIT function and approach
Current results for SE 01 and others
Starting
Analytical aerodynamic starting simulations and results/limitations
2D CFD aerodynamic starting simulations and results
Updated Demonstrator Spec
Present Demonstrator Spec with any updates available for Mechanical guidance
Lessons Learned for CFD Workflow Improvement
Workflow description, identify bottlenecks, plans for overcoming or reducing impact

3. Task 2.2, Task 2.7 & 3.2 – CFD Comparison / Shock Wave Boundary Layer Interaction (SWBLI) Investigation and Aerodynamic Tool Development

Ramgen has performed extensive numerical predictions of complex 3D shock wave / boundary layer interactions to test the ability of numerical algorithms to capture complex 3D turbulent boundary layer separation phenomena observed in experiments. These simulations were performed using linear or static configurations, rather than rotating, due to the availability of linear test data. Appendix 3.1 EUCASS Validation Paper contains a paper presented at the 3rd European Conference for Aerospace Sciences summarizing results from one of these validation efforts.

Static Test / SWBLI Investigation

Ramgen contracted with the Naval Postgraduate School (NPS) in Monterey, CA to investigate the effects of boundary layer control jet injection on shock-boundary layer interaction in a static test rig. Schlieren flow visualization was used to view the formation of a shock within a Mach 2.4 nozzle. A CFD model was created to confirm the measurements of a total pressure probe and end-wall static pressure within the test section. A shock generator was designed, constructed and installed to produce the desired shock profile. Schlieren images and total pressure profiles downstream of the slot injection were recorded at two shock generator positions and three injection pressures to ascertain the effect of injection on the separation bubble formed due to the shock reflection within the boundary layer.

Naval Postgraduate School (NPS) Rotating Air Test

Ramgen pursued a low-pressure inducer test with air as the working fluid, intended to obtain rotating test data prior to the main facility entering operation in 2011.

The Turbo Propulsion Laboratory at the naval Postgraduate School houses a number of experimental facilities for research and development related to turbines and compressors. The complex contains three cascade wind tunnels, a 3-stage axial research compressor, a transonic turbine rig, a compressor rig, a supersonic wind tunnel, two free jets, a shock tube, and a spin pit. The data acquisition system can accommodate 400 channels of steady state measurements and 32 channels of unsteady measurements at up to 200 kHz. The staff members have experience with Laser Dopler Velocimetry (LDV) systems, pressure sensitive paint, along with Schlieren and shadowgraph flow visualization techniques.

Figure 3.1 shows a schematic of the proposed transonic compressor test stand. The test stand was powered by an air turbine which connects to an overhung rotor with a spline shaft. The air to run the turbine comes from a compressor located in the same building. The power draw for this compressor limits testing to the morning hours. Rotors were bolted to the end of a shaft supported on ceramic bearings so that nose cones could be attached. If a nose cone was not desired, the rotor could be blanked off and an inlet hub surface could be included in the static structure. The pressure ratio across the rotor was set by throttling upstream, suppressing the inlet pressure. The exhaust was open to atmospheric pressure. The inlet mass flow was measured with a flow nozzle.

Instrumentation in the test cell consists of up to 48 thermocouples, 48 static pressures and 16 high speed channels set up for Kulite pressure transducers. NPS has extensive experience in characterizing rotors in this test stand. Typically, several speed lines are run up to stall and a compressor map is built which can be compared to CFD of the same. Stall could be detected with high speed pressure transducers, usually directly above the rotor, in additional to thermocouples mounted after the rotor. The pressure transducers show fluctuations that increase as the rotor approaches stall. The temperature rise behind the rotor also increases much faster than the pressure ratio and a significant drop in efficiency can be observed. A PoCoVD (Posterior Contacting Vibration Detector) system was employed to alert operators if the rotor had surged. Historically, only unshrouded rotors have been tested at this facility. NPS uses a machinable rubber compound above their rotors to create minimum tip clearances.

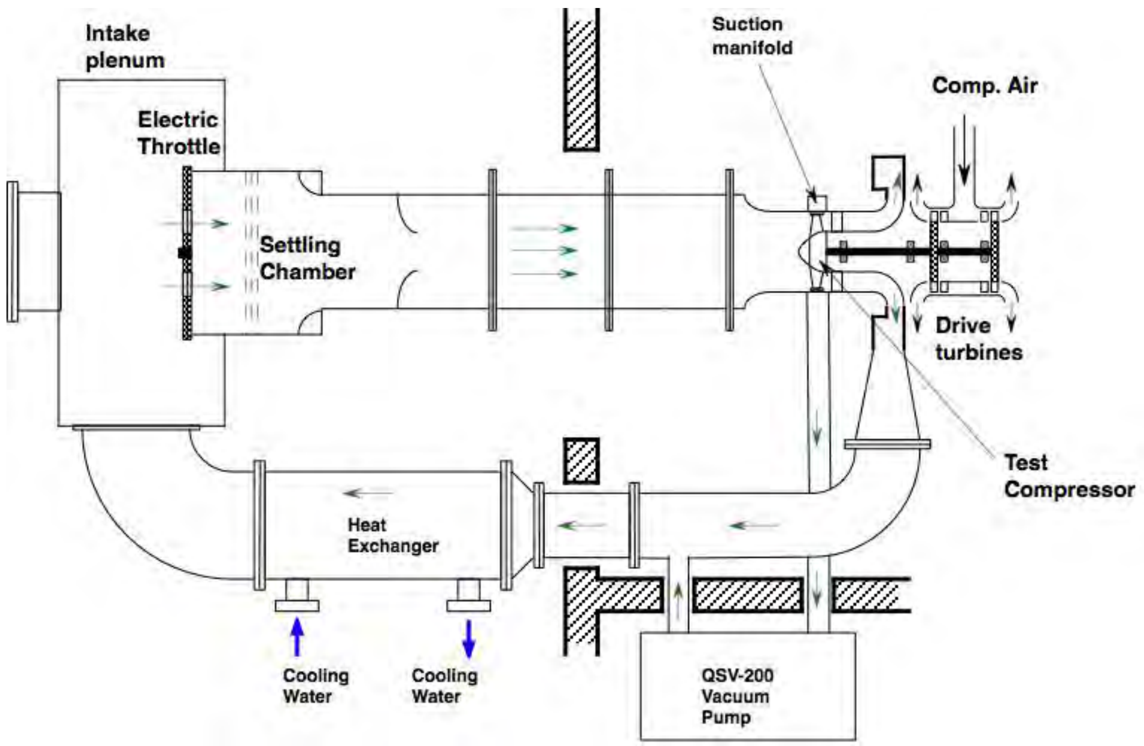


Figure 3.1: Proposed Closed-Loop Air Test Facility at NPS

High speed total pressure measurements can be made in the flow. The data acquisition system has a maximum sampling rate of 200 kHz per channel. This can either be done using several probes at different insertion depths or by using a single probe and a traverse mechanism. Typically, time accurate flow angle and velocity measurements are not made, though a technique to do this was developed by an individual. Flow angle and velocity measurements are made using both 90° and 45° probes at several different probe angles and the data are synchronized to the blade passing frequency. This can provide time accurate flow angle and velocity data during a blade passing event to give detailed information for comparison with CFD.

In late 2010 NPS informed Ramgen that they could not complete the test on the agreed to budget and schedule. Ramgen determined that based on the other testing to be completed for the ISCE program, investing in a test cell in our own facility would result in greater capability to accomplish the goals of the program than contracting the work to a remote facility.

In 2011, Ramgen significantly advanced its shock wave based compression aerodynamic design process by applying the incredible power of the Jaguar supercomputing cluster at Oak Ridge National Lab's National Center for Computational Sciences (NCCS). Two primary aerodynamic components of the shock compression technology were successfully modeled through a series of large design variation study 'database' runs. The first component study utilized a computational mesh consisting of 11 million grid cells each, and one thousand candidates were run. The database was accomplished by running ensemble jobs reaching the size of 90,000 computer cores for two hours. Subsequent iterative optimization runs based on the initial database yielded a design that demonstrated performance improvement and resulted in Ramgen choosing this design for manufacture and test.

In late 2011, very large database run was accomplished for a shock compression advanced concept design. Each mesh in this case contained 75 million grid cells, and 480 candidates were run. This database was completed through two eight hour 120,000 core runs on Jaguar. Analysis of the data from the run produced important benefits early in 2012. The Ramgen team observed designs that exhibited valuable aerodynamic characteristics. These advancements would not have been possible without the use of Jaguar. These runs represent a paradigm shift in achieving performance improvements for shock compression technology and establish a new model for improving turbomachinery.

Figure 7.10 displays a basic flow chart that describes the optimization process. The engineers select design variables they wish to modify in the study and the ranges for each. A large database of CFD simulations for a set of designs that reflect combinations of these variations over the specified ranges is generated. The resulting data is then used to construct an approximate model (or 'meta-model') of the multi-dimensional design and performance space. The result is a continuous interpolation that can be searched for designs predicted to offer high performance. We are specifically employing an Artificial Neural Network for the meta-model, and search is performed using a Genetic Algorithm. An iterative procedure is then run by predicting optima, running additional CFD on the results from the Genetic Algorithm search and repopulating the database, and looping until performance improvement is achieved. Figure 3.3 demonstrates the convergence of these performance predicted by the Artificial Neural Network and the actual optimized geometries.

The consequence of this approach was that extensive numbers of complex simulations needed to be run. If they were run one at a time, distributed over fewer processors, it would take many months to generate the same data that we have run during a single day on Jaguar. In addition, a sophistication of modeling can be achieved on Jaguar that is impossible on lesser computers. These capabilities were critically important to enabling innovation in aerodynamic design at Ramgen in a timeframe meeting DOE goals.

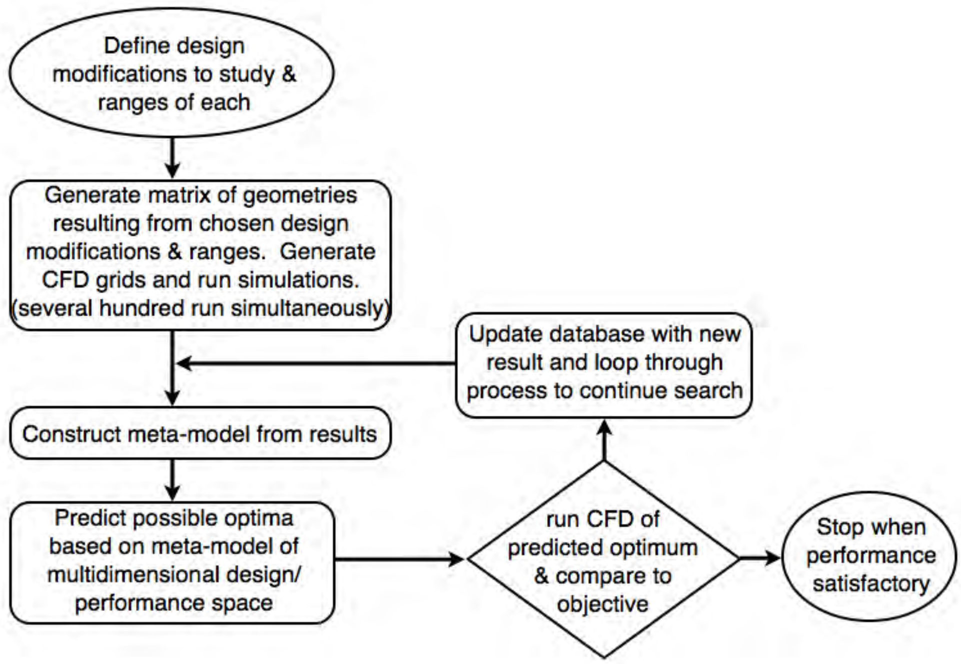


Figure 3.2: Optimization cycle flowchart

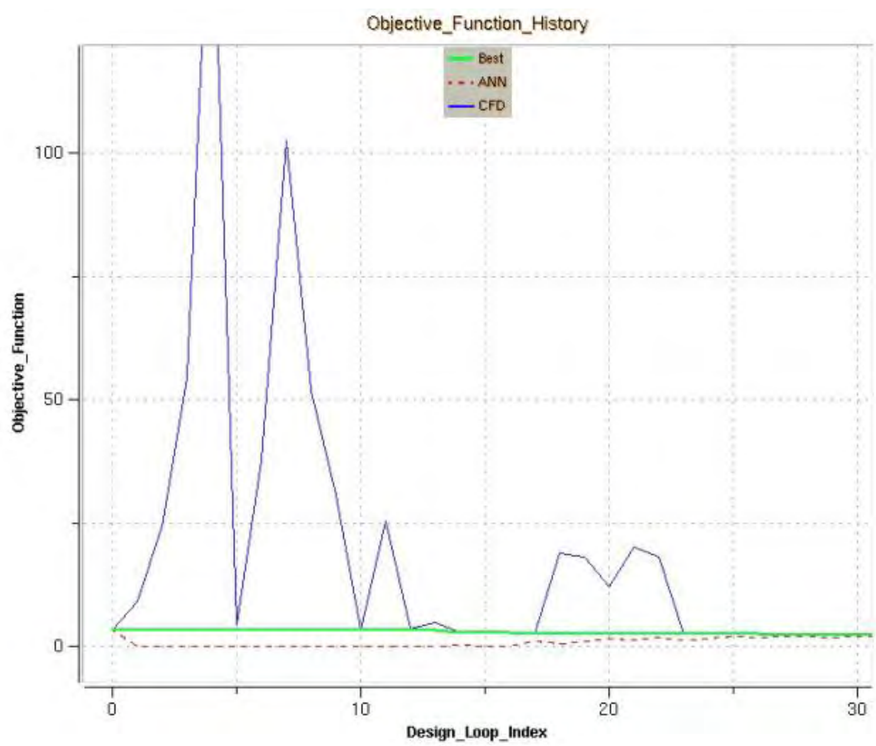


Figure 3.3: Example of an Optimization Cycle History Demonstrating

In order for an efficient optimization procedure to succeed, a complete chain of the entire workflow must be run automatically. Ramgen has spent years and a great deal of resources, including discussions with ORNL, towards improving individual modeling and analysis tools. Throughout 2011, Ramgen worked with our CFD vendor to continually improve the software. Parametric geometry generation capabilities were developed so that Ramgen's proprietary designs could be modified substantially based on user-controlled design variables and driven in batch mode parallel processes. Grid generation capabilities were developed to specifically handle Ramgen designs and automatically generate computational grids based on a template-driven process. The flow solver parallel implementation has been rewritten, now enabling simulations to be run in the thousands of computer cores range per simulation, and including input/output (I/O) acceleration. The result was the ability for Ramgen engineers to identify a large number of design variables they wish to study, and then execute runs at the OLCF that generate the geometries, grids, and then flow solutions for a database containing several hundred candidates in the space of a day.

During 2012, Ramgen continued its productive collaboration with the Oak Ridge National Lab's National Center for Computational Sciences (NCCS), extensively using the Jaguar supercomputer to perform intelligently-driven design optimization of the primary aerodynamic components of its shock wave based compression technology. Ramgen has run multiple ensemble jobs, encompassing more than seven thousand individual simulations and nearly 40 million CPU hours, with several of these jobs effectively utilizing 80% of Jaguar's available resources.

On June 14th, 2012, a large database run containing 2000 design candidates was completed on Jaguar. The results of this run formed the basis for subsequent optimization of the shock compression passage geometry and boundary layer flow control configuration. To date, over 1000 additional designs have been simulated during this optimization process. Analysis of these results has demonstrated significant performance enhancement over earlier designs and has been instrumental in improving Ramgen's understanding of the complicated relationship between the geometry, three-dimensional flow field, and performance of the system.

Over the course of 2013 Ramgen ran multiple ensemble jobs, encompassing more than eighteen thousand individual simulations and approximately 47.5 million CPU hours. Two-thirds of these hours were associated with jobs utilizing greater than 60% of Titan's available resources, and more than 12% were associated with jobs utilizing greater than 80% of Titan's available resources. This translated to a greater than 84% level of capability on Titan, as defined by NCCS as the percentage of hours spent utilizing more than 20% of the system.

Between March and August 2013, 13 large ensemble jobs, totaling more than 18,000 individual design simulations and representing 3 primary system components, were completed. Analysis of these results has demonstrated significant performance enhancement over earlier designs and has been instrumental in improving Ramgen's understanding of the complicated relationship between the geometry, three-dimensional flow field, and performance of the system.

During 2013, Ramgen started work to take advantage of the new capabilities of the Graphics Processing Unit (GPU) enabled Titan system. Previously, the parallel solver was predominantly a Message Passing Interface (MPI) code where the solution is partitioned into virtual blocks and distributed to the CPU cores, but work began in early 2013 to add solver acceleration via GPU through OpenACC directives. Our CFD vendor has developed an updated convergence acceleration algorithm. When using this new method the cost per iteration was multiplied by a factor of about three, but it enabled a stable solution at significantly higher Courant–Friedrichs–Lewy (CFL) condition numbers such that the total number of iterations required to reach convergence was reduced by about one order of magnitude. Such a combination lead to a reduction of the CPU time required to reach convergence by a factor of 3 to 4. The implementation utilizes intensive arithmetic, without interruption by I/O, memory reorganization, or any other system operation, making it an ideal candidate for the hardware acceleration offered by Titan's GPUs. Under contract to Ramgen, our CFD vendor began work in 2013 to restructure the implementation for multithreading.

Restructuring of the code yielded an additional factor of 2 speedup, and efforts to complete the implementations needed to off-load CPU. The speedup already reached by CPU Booster for a Ramgen computation example is shown in Figure 3.4.

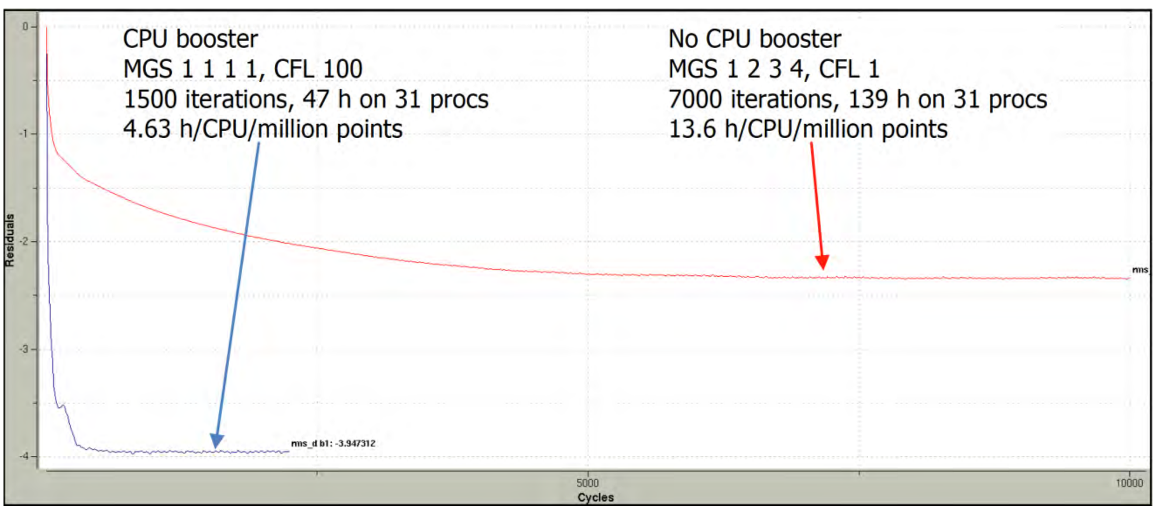


Figure 3.4: Example of convergence acceleration offered by CPU booster

4. Task 2.3 – Inlet Guide Vane Characterization

At program start, it was assumed that an Inlet Guide Vane (IGV) would be required, as was the case for the previous Rampressor-2 program. Task 2.3 was established to ensure that the flow coming from a new, untested vane shape was correctly predicted by CFD and matched our rotor inlet requirements.

After hiring an expert vane designer, Ramgen determined that standard National Advisory Committee for Aeronautics (NACA) vanes would be sufficient for our needs. Due to the well-characterized and tested performance of NACA vanes, the IGV test program was deemed redundant and the task budget was re-allocated to other portions of the program, with DOE concurrence.

During the second quarter of 2011 Ramgen evaluated the impact on the HP CO_2 compressor program to move to the advanced supersonic compressor configuration developed on the ISC Engine program. After all the schedule, cost, technical and manufacturing issues were assessed Ramgen's strong conclusion was that the alignment of the HP CO_2 program to the ISC Engine configuration would be the fastest route to the DOE contract goals. The analysis concluded that most of the existing hardware and design could be re-used. The components closest to the rotor will need to be redesigned, and it was determined that inlet guide vanes (IGVs) were no longer required.

5. Task 2.4 – Stationary Diffuser Characterization

The latter portion of the high-pressure ratio CO_2 compressor under development in early configurations for the CO_2 Compressor was referred to as the "Exducer", which consists of a centrifugal impeller and a vaned diffuser. Because of specific design features the flow conditions at the exit of the centrifugal impeller or diffuser inlet are challenging for the diffuser design and operation. Main challenges are high Mach number (i.e. Mach number >= 1.0) and high flow angle (i.e. flow angles approaching 80 degrees) at the inlet of the vaned diffuser. There are many centrifugal compressor designs (almost all high-pressure ratio centrifugal compressors) where the Mach number exceeds 1 at diffuser inlet, but typical range of centrifugal compressor diffuser inlet flow angle is 60 -73 degrees. High diffuser inlet flow angle increases total pressure loss and flow instability such as rotating stall. Based on a survey of the open literature, very limited information about centrifugal compressors with high inlet Mach number and flow angle conditions exists. Therefore it was decided to experimentally investigate vaned diffuser designs under high Mach number and high flow angle operating conditions. Because of time, schedule and cost considerations it was considered necessary to use an existing centrifugal compressor test rig and to modify the impeller in order to produce the diffuser inlet flow field conditions of interest. After contacting several universities and research institutions which have experience and test rigs for centrifugal compressor research it was decided to collaborate with the Turbomachinery Laboratory of the Michigan State University (MSU).

The research project at MSU consisted of two phases. Phase 1 was a feasibility study to demonstrate that the inlet and operating conditions with high Mach numbers (i.e. M > 1.0) and flow angles (i.e. flow angle > 80 degrees) to a vaned diffuser downstream of a centrifugal impeller could be produced at the existing MSU test rig. This part of the project was basically a new centrifugal impeller design, taking into account the existing test rig constraints at MSU Turbomachinery Laboratory, which would be capable of producing the range of diffuser test conditions of interest. Phase 2 was to be the actual vaned diffuser testing under high inlet flow angle and Mach number flow conditions. Different vaned diffuser designs were to be experimentally investigated at the operating conditions of interest (i.e. inlet Mach number range 0.8 – 1.1 and inlet flow angle range 78 – 85 degrees). The goal was to determine the performance (pressure recovery and losses) and operating range of the vaned diffusers.

During Phase 1, MSU Turbomachinery Laboratory designed a new centrifugal impeller and analyzed this impeller design using CFD. The effects of some impeller design parameters (i.e. inlet axial length) were also analyzed. Unfortunately the CFD analysis was carried out at only one operating point and did not cover the expected operating range of the diffuser. Based on this CFD analysis, it was shown that the designed centrifugal impeller can produce a maximum exit flow angle (diffuser inlet flow angle) of 80 degrees and maximum Mach number of 0.90. One major problem of the MSU impeller design was the danger of flow instability at impeller exit/diffuser inlet. Comparing the design characteristics (especially the exit width) of the impeller with the desired high flow angle and Mach number flow conditions, the proposed MSU impeller design was not stable according to the widely used SENOO stability criteria in the centrifugal compressor industry. The flow field information and related diagrams or plots provided in the design report are not conclusive about the flow instability at impeller exit but for example the flow angle plots at impeller exit clearly depicts flow separation and backflow regions. Considering that the MSU Phase 1 impeller design could only partially produce the high Mach number and flow angle conditions of interest and the concerns of flow instability, it was decided to not pursue Phase 2 of the project. The remaining task funds were re-allocated, with DOE concurrence.

6. Task 2.5 & Task 3.4 – CO_2 Compressor Design

General information for the CO_2 compressor is shown below:

- Suction: 220 psia / 100 F nominal
- Discharge: 2200 psia / TBD temperature (pending final performance CFD results)
- Water cooling temperature: 85 F (cooling tower return) nominal
- Suction flow rate: 86 lbm/sec nominal
- Gas composition: food-grade CO_2
- Rotor diameter: 11.408" maximum
- Rotor RPM: 31,000 design, 36,306 max mechanical speed
- Pressure case material: ASTM A350 LF2 Class 1

Due to the research nature of this compressor demonstrator, there were a large number of services and other connections needing to pass through the pressure case. High differential pressures between suction and discharge combined with high discharge temperatures create sealing and thermal management challenges. The pressure case final design review summarized the design and analyses performed on this component.

The radial inlet and IGVs condition and direct the suction gas into the supersonic compressor rotor. Efforts must be made to keep flow distortion and pressure loss at a minimum while providing the desired rotor inlet Mach number and flow angle.

Ramgen supersonic compression technology produces substantially higher pressure ratio per stage than conventional turbocompressors. As a result, differential pressure between the compressor discharge and suction or secondary flow passages require careful attention to sealing. The shrouded compressor configuration requires effective shroud seals to prevent discharge pressure from leaking back around the shroud to the inlet, where leakage back into the inlet would create flow distortion and reduce rotor performance. Additional seals must be used to isolate the boundary layer control secondary flows to ensure pressures and mass flows were kept to their design optimum. A combination of labyrinth and pocket damper seals was used to provide sufficient sealing, flow isolation, and damping. The final design review summarized the design and analyses associated with their design.

The combination of high rotor rotational speed and high discharge pressure result in potential for aerodynamic cross-coupled forces and resultant instability (Wachel forces). In addition, lateral rotordynamic stability was of critical importance in high-speed turbocompressors to avoid issues with tight-running seals and oil heating. To ensure trouble-free operation in test, Ramgen designed to meet API standards for vibration magnitude but adopted D-R's imbalance guidelines (4x to 16x the imbalance required by API standards). Successful results from these analyses were summarized in the final design review.

Providing oil lubricant for a high-speed turbocompressor and associated high-reduction gearbox required a redundant pump system with backup for power loss scenario. After evaluation of a shaft-driven pump and a gravity flow-down tank for fail-safe operation, Ramgen selected an electrical motor-driven pump with redundant backup along with an uninterruptible power supply (battery system) to enable lubricant delivery during power loss and subsequent compressor coast down.

The Gen 1 rotor design consisted of two separate supersonic inducers bolted to either side of a single back-to-back subsonic exducer. To manufacture the exducer, a Powder Metallurgy Hot Isostatic Press (PM/HIP) process was adapted to our specific geometry. A manufacturing demonstration exducer was fabricated and delivered to Ramgen in early 2011. The process showed good promise, but some additional manufacturing development and validation was still required before the rotor destined for the rig was ready to be fabricated.

The two supersonic inducers presented even greater challenges. To fabricate the integral shroud and bleed features at the scales required, complex 5-axis machining was required with tooling that pushed the limits of what was achievable (in terms of tool diameter versus the overall reach of the tool). In addition, Electric Discharge Machining (EDM) was also required to reach into the part and finish the machining operations that could not be achieved with the conventional tooling. After an exhaustive search for a capable supplier, a manufacturing demonstration inducer was attempted by the one and only supplier identified who was willing to attempt the part. In spite of their best efforts, this supplier was unable to demonstrate the manufacturing processes required to create the inducer geometry.

During the second quarter of 2011 Ramgen evaluated the impact on the HP CO_2 compressor program to move to the advanced supersonic compressor configuration developed on the ISC Engine program. After all the schedule, cost, technical and manufacturing issues were assessed Ramgen's strong conclusion was that the alignment of the HP CO_2 program to the ISC Engine configuration would be the fastest route to the technical goals. The end result was a program that was roughly comparable in schedule, with improved technical and programmatic risk reduction. The analysis concluded that most of the existing hardware and design could be re-used. The components closest to the rotor will need to be redesigned, and it was determined that inlet guide vanes (IGVs) were no longer required. The redesigned flow path has been designated internally as Generation (Gen) 2, Build 1.

The redesign and manufacturing of the revised static hardware were part of the critical path to a 2^{nd} quarter 2012 test target. Final design reviews were completed in January 2012, and final drawings were released in February 2012.

The modification of the flow path to the ISCE configuration modified the rotor design to that of a traditional impulse fan to accelerate flow into a non-rotating supersonic shock compression diffuser. Initial designs were generated by utilizing a NASA impeller blade design code. The rotor produces 12:1 total pressure ratio at design point. In order to achieve the desired rotor blade exit conditions, the final blade design was developed via an optimization process composed of 1000 database samples on Jaguar.

The rotor was designed as a single, solid axis with dovetail disc slots for blade attachments. Coverplates were used on either end of the blade slots to retain the blades' axial position. While the nominal static pressure difference across the rotor was low, basic sealing was necessary both across the top of the rotor as well as between the primary flow path and the rotor wheel space. This sealing requirement was in both cases addressed by maintaining minimum clearance between sets of labyrinth teeth and an abradable insert or coating on the adjacent surface.

The rotor Final Design Review (FDR) was completed on December 7th, 2011. A complete description of the rotor can be found in Appendix 6.1.

Work began in Q2 of 2011 on the modified diffuser following the decision to incorporate the ISCE flow path into the HP CO_2 design. In order to accommodate the facility as designed prior to the modification, both inlet and outlet designs remained largely the same.

In June, 2011 an analysis of primary flow loss mechanisms was initiated, with the goal to isolate various viscous effects and quantify their contribution to total pressure loss through the diffuser. The study identified multiple effects unique to the Gen 2 static diffuser as compared with the rotating supersonic component from previous Ramgen designs. The results from this study ultimately informed the design of the final diffuser structure and flow control features found in the current Gen 2 diffuser.

Down-selection to the final design of the static shock compression diffuser was initiated in August of 2011. Early design iterations were performed by Agilis Engineering working in concert with the aerodynamics team at Ramgen and utilizing Ramgen facilities. WIND-3D CFD software was used extensively during this portion of the design phase due to the rapid modeling capabilities of the bleed boundary layer control features. The aerodynamic team performed supersonic diffuser validation studies to successfully demonstrate sufficient modeling accuracy of our CFD tools as applied to the Gen 2 design (see Figure 6.1 and Figure 6.2). Simultaneously, the mechanical team began finalizing drawings of non-critical flow path components of the HP CO_2 test rig to support the manufacturing schedule. Critical flow path component preliminary design reviews (PDRs) were completed in December 2011 in support of the design schedule.

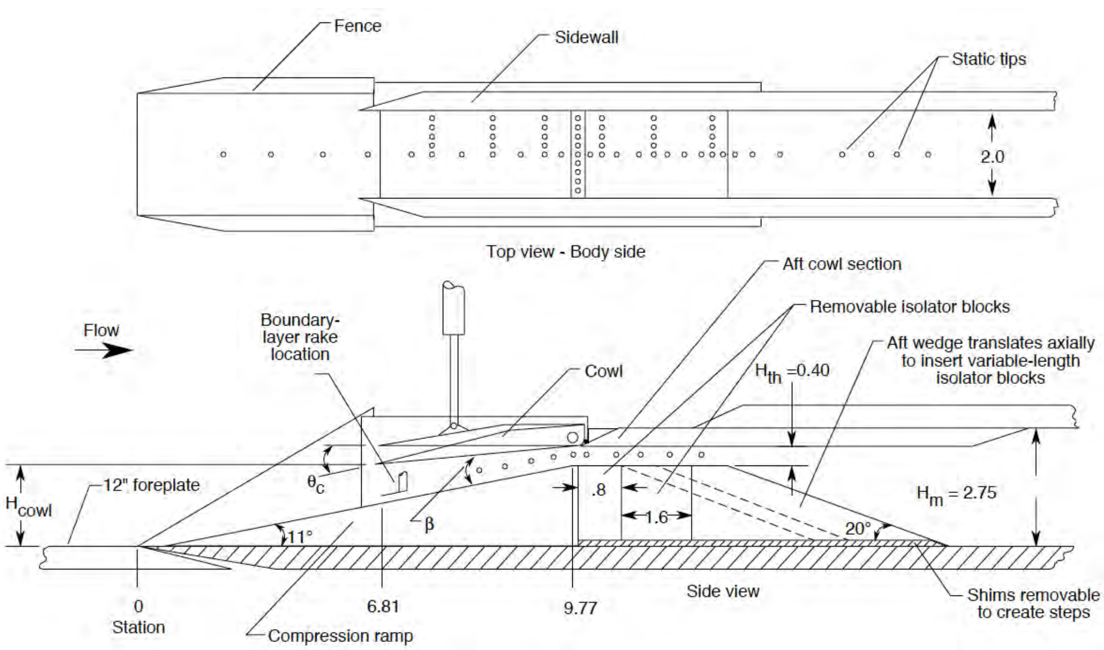


Figure 6.1: Experimental set-up referenced for validation study; see Emami, Trexler, Auslender, Weidner, 1995, "Experimental Investigation of Inlet-Combustor Isolators for a Dual Mode Scramjet at a Mach Number of 4"

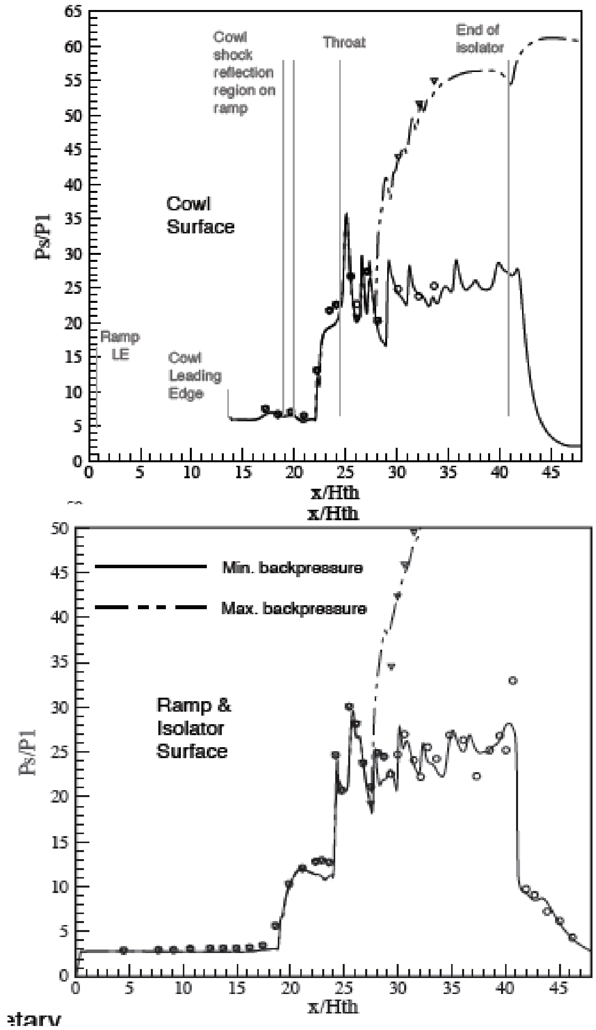


Figure 6.2: Experimental (points) and CFD results (lines) of the validation study. CFD showed excellent agreement with experimental results for both minimum and maximum backpressured cases.

The preliminary design review for the diffuser was completed on December 9th, 2011 with the final design review of the static diffuser components scheduled for January 16, 2012, in support of a Q2 2012 test date. Ramgen completed modeling and validation of primary flow path components and secondary systems, including: on and off-design system performance; validation of starting procedure; secondary flow routing and losses. Items identified for completion prior to the static diffuser FDR include thermal Finite Element Analysis (FEA); thermal and mechanical load FEA of actuation systems; completion of volute CFD analysis. A complete description of the diffuser and related systems can be found in corresponding Preliminary Design Review (PDR) in Appendix 6.2.

More than 2900 CFD simulations were completed on Jaguar, as part of an ongoing design optimization process. During the course of the optimization, Ramgen discovered several novel geometry modifications that showed significant promise towards increasing the overall performance of a supersonic compressor configuration. Estimates compiled at the beginning of Q3 2011 showed noticeable improvements over the currently employed Build 1 geometry, and further optimization moved Ramgen closer to its performance targets. The highest performing cases were passed to the mechanical design group for preliminary analysis and design work, while the facility impact of the Build 2 configuration was assessed. Though the designs were sufficiently different from the Build 1 configuration as to require replacement of a number of primary components, Ramgen worked to minimize the impact on budget and risk assessment of the Build 2 geometry by reusing a significant portion of the static structure. The pressure case did not require redesign, nor did the drive train components require modification.

The current Build 2 diffuser design required a redesign of the impulse blade, and in May 2012 Ramgen began preliminary design of a new impulse style rotor blade. Initial designs were developed using an in-house code based on the well-known streamline curvature method, and further developed in 3D steady-state viscous CFD. In August, 2012 Ramgen contracted to perform optimization design cycles on a parameterized blade, and a database of 1000 rotor samples were generated on the Oak Ridge Jaguar supercomputing system in September 2012. Highly detailed parallel post-processing of the results included high-resolution performance analysis through the blade passage. The result of these detailed analyses, in conjunction with the optimization cycles, ultimately resulted in a 3% increase in rotor efficiency, and a marked increase in rotor exit uniformity. The rotor system passed Conceptual Design Review (CDR) (Appendix 6.3).

While the design of Build 2 was progressing, the final design freeze was planned to be contingent on the Build 1 test schedule. A critical point of interest was the conformance of test results with CFD predictions made during the design process. The supersonic nature of Ramgen's technology requires detailed modeling of viscous interactions, more so than conventional turbomachinery designs, making such convergence of test data and CFD predictions a top priority when diagnosing the risk inherent in the Build 2 design.

The radial turn acts to further diffuse flow from the compressor exit to low subsonic velocities before discharge into the collection volute and recycling of the fluid. In order to support volute location and a changing diffuser length through Q2 and Q3, the radial turn underwent multiple iterations before aerodynamic freeze in September 2011. Ten vanes exist within the discharge flow path to accommodate bolts. The FDR for the radial discharge and volute was completed on 22 December 2011. The static diffuser FDR can be found in Appendix 6.4.

Ramgen uses a number of tools to perform performance evaluations and predictions. The initial predictions come from first- principal and loss-correlation spreadsheets. As the definition of the aerodynamic flow path was frozen for use by the mechanical design team, Computational Fluid Dynamic (CFD) codes were used to separately analyze the individual components of the flow path including:

Inlet
Inducer
Supersonic Diffuser

Subsonic Diffuser
Volute
Secondary flows (bleed, cooling, thrust balance, etc.)

These steps were completed prior to completing the mechanical design and manufacturing of the performance rotor. As the final performance rotor configuration manufacturing was being completed, the exact design of the flow path can be modeled in CFD. The fidelity and accuracy of the actual design requires a tremendous amount of computational resources. The level of sophistication and detail Ramgen has performed on the flow path are normally resource prohibitive by most aerospace companies. The level of analysis performed to date gives Ramgen the most accurate prediction possible prior to test. Once actual tests were conducted and test conditions were collected by instrumentation the performance models were further refined to match test results.

In Q3 2012, Ramgen began final facility preparations and assembly for testing of Build 1 for the HP CO_2 program. The final facility piping and instrumentation diagram (P&ID) was approved for safety and functionality, and was handed to contractors Mollenberg Betz so they could implement the final facility piping design. The full facility P&ID can be found in Appendix 6.5.

As part of good safety practice and our agreement with Dresser-Rand a test readiness and hazardous operations review was held with Dresser-Rand and the Olean facility test directors. The final review was part of a series of reviews conducted during the building and commissioning of the facility and the rig build up. The types of reviews that were held are listed here:

Design coordination meeting
Facility design review
Civil design review
Electrical design review
General status review
Facility hazop
Facility hazop follow up
Facility hazop follow up
Facility electrical hazop
Facility Gen-2 changes hazop
Compressor status review
Rotor mechanical review
Final design review
Bald rotor hazop
Fuzzy rotor hazop
Performance rotor hazop

The Agenda for the final review reflects the issues that were prepared and reviewed in detail:

1. **Review previous meetings held**
2. **Hardware description**
3. **Compressor hardware hazards**
4. **Rotordynamics**
5. **Starting door hardware**
6. **Rotor structural analysis**
7. **Performance rotor test description**

Action items were documented during the review and a closure plan was put in place for each item. Dresser-Rand safety officers and representatives had to concur that the item was closed before testing could begin. In addition to the readiness review itself there were a large number of operating and safety procedures to educate and guide the test personnel in the safe use of all equipment during testing including:

Ladder Use
Aux Compressor
Forklift
Building entry
Boom Lift
Fire Suppression
CO2 Supply
Compressor Test Rig Actuators
Compressor Test Rig Controls
Compressor Test Rig Cooling systems
Compressor Test Rig Lubrication
Compressor Test RigVibration monitoring
Personnel Protection
Plant Evacuation

The Test Director was responsible for ensuring all test personnel were familiar with the applicable procedures and competent to operate the equipment they were assigned to. There was a log to document the personnel that have reviewed each of the procedures.

In conjunction with facility work, Ramgen personnel were on site in Olean for final assembly of the performance rotor and bundle that was to be inserted into the pressure case for testing, see Figure 6.3 and Figure 6.4. Assembly took place over the course of 8 weeks as various pieces were independently assembled prior to final insertion of the completed rotor into the pressure case. The facility was simultaneously being prepared for the various instrumentation Ramgen required to adequately assess the performance of the design. Steel and nylon pressure tubes along with 24V wiring and thermocouple wires were routed to various programmable logic controllers (PLC) around the facility. Final instrumentation connections were made and the various instruments were checked for leaks and correct performance.

Figure 6.3: Ramgen personnel assembling the bundle.

Figure 6.4: Instrumentation work on inserted bundle.

Due to the complex nature of the Ramgen design, a large number of secondary flow systems were required. The process flow diagram in Figure 6. schematically illustrates the complexity of the facility piping design. Simultaneous control of the different process gas flows necessitated automated programming of the control valves to ensure safety, as well as to reduce the chance of operator error during test. Ramgen used these initial runs to simultaneously perform valve tuning as part of the facility commissioning stage. With the rotor accelerated from 1970 to 8860 rpm several times, the tuning of seven valves were performed to ensure proper operation. Once the valves responded acceptably, the process was repeated while accelerating the rotor from 8860 rpm up to 15,700 rpm and again from 15,700 rpm to 27,500 rpm. Examination of the vibration monitoring data revealed a new sub-synchronous vibration (SSV) not seen during previous rotor testing. The new SSV was seen with significant magnitude on the rotor driven end proximity probes, the high speed coupling proximity probes, and the gearbox input shaft. The frequency of the mode changed with speed but it was not a constant fraction of the input speed. After increasing loop pressure to 100 psia we saw a dramatic and sudden drop in the SSV which also corresponded with significantly cleaner and lower amplitude orbits at several locations on the drive train. The change was most apparent on the HS coupling proximity probes. Unfortunately, the elimination of the SSV was accompanied by a corresponding increase in rotor 1E vibration levels.

After careful examination of vibration and rotordynamic data, it was decided to increase suction pressure to 150 psia in an effort to reach higher loads and maximum continuous operating speed (MCOS). As the load on the drivetrain was increased with the step up in suction pressure, rotordynamics responded favorably. In the final week of November, 2012, Ramgen reached a major milestone when the build 1 performance rotor was successfully spun up to MCOS at a motor load of over 8 megawatts, see Figure 6.6.

After a final extensive review in the repeatability of the drivetrain performance (see Appendix 6.6), Ramgen began the aerodynamic portion of the build 1 performance rotor test. One of the biggest challenges with the Gen 2 design was the aerodynamic "starting" of the supersonic diffuser. Though many theories existed on what sequence of actions would start the diffuser, it was unknown which would be successful. A period of four weeks was allocated to the trial of the different possible paths to reach a started supersonic diffuser. On November 18th, 2012, aerodynamic data showed indication of a starting event in the supersonic diffuser, see Figure 6.7. The event was marked with a sharp drop in pressures in the diffuser, along with a decrease in the static pressure ratio across the rotor, both strong indicators of established supersonic flow. This was a huge milestone for the HP CO_2 program.

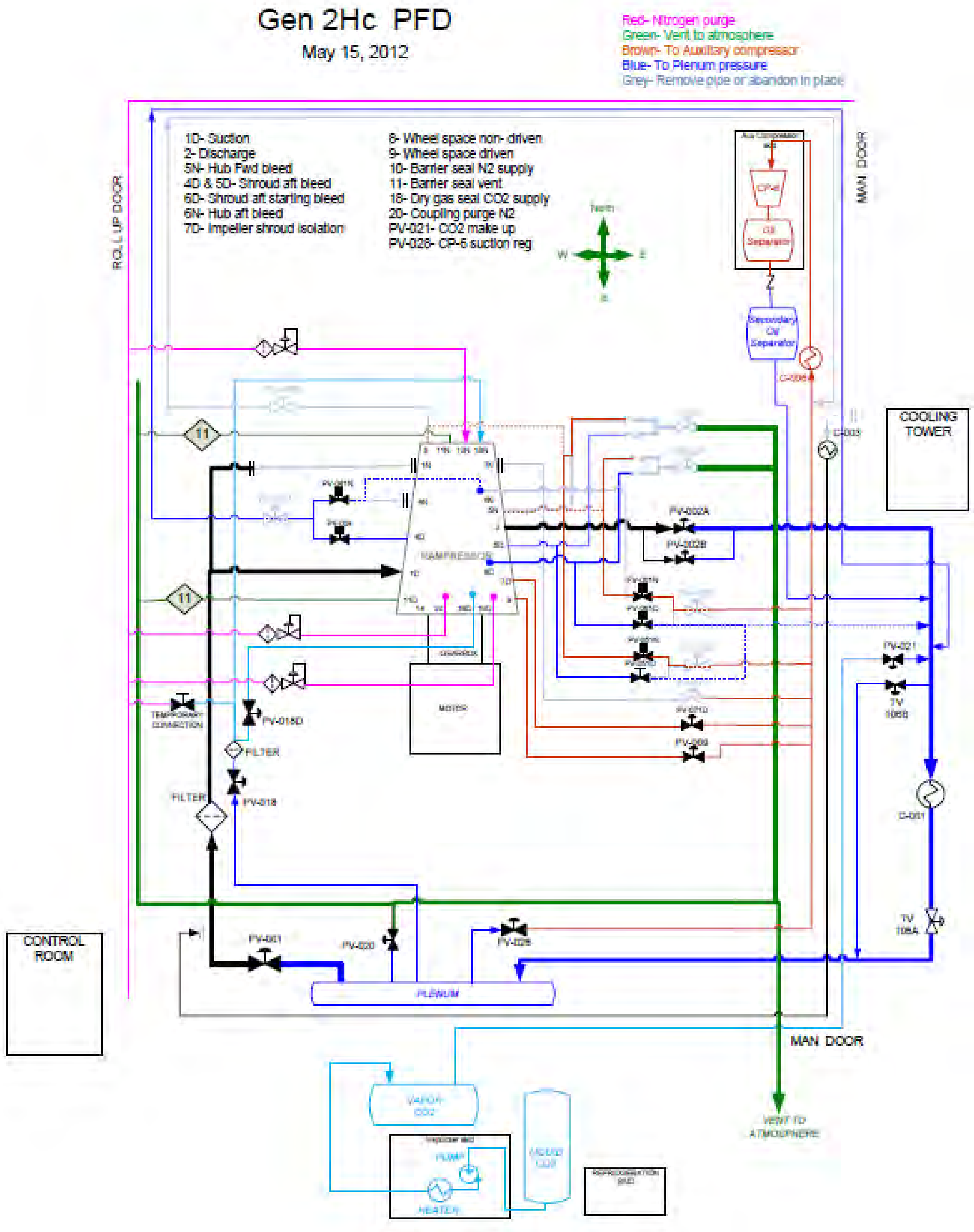


Figure 6.5: Gen 2 facility process flow diagram.

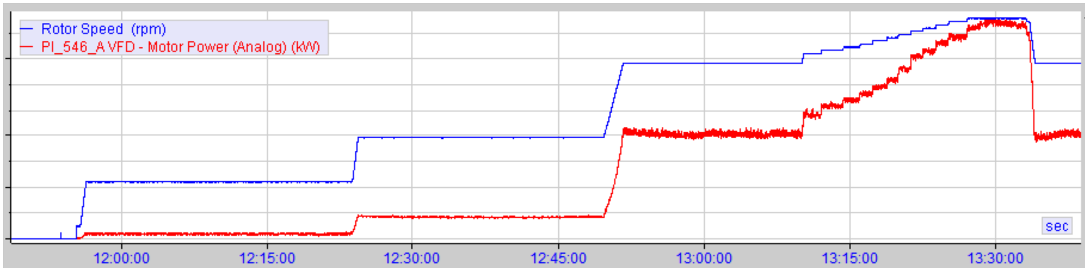


Figure 6.6: Rotor speed and power demand.

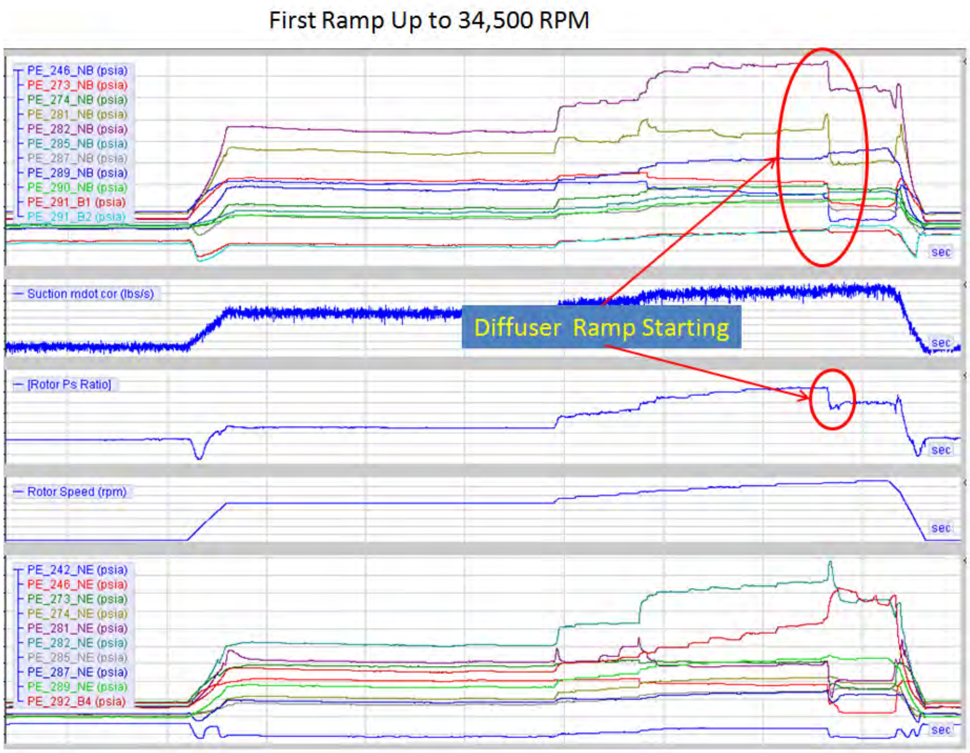


Figure 6.7: Evidence for a started diffuser.

Though the diffuser was started, the overall suction massflow was lower than Ramgen's pre-test prediction. Ramgen theorized that the rotor was operating in a stalled regime and thus ingesting less mass. Ramgen decided to finally increase the suction pressure to our design point of 210 psia. On December 7th, 2012, the test team observed a significant starting event as indicated by a reduction in rotor static pressure ratio to ~1 and a jump in power from 8.2 to over 9MW, see Figure 6.8. The aerodynamic state the rotor and diffuser were operating in matched pretest CFD predictions. All subsequent tests managed to successfully start the diffuser. The path to starting the Ramgen rig was deemed completed, another major technical goal for the Ramgen test team.

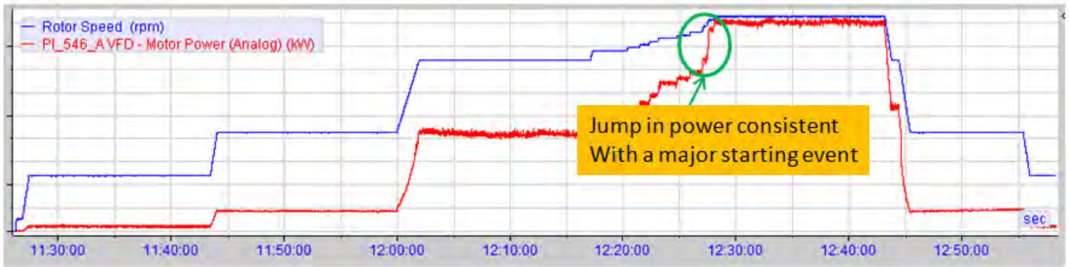


Figure 6.8: Spike in power draw supports the conclusion of a diffuser starting event.

Ramgen spent the rest of the December mapping performance targets. A major technical challenge in the Ramgen rig was the design and operation of the actuation system, critical for a high performing diffuser. It was discovered that in order to commission the actuation system, it was necessary to first apply backpressure to the rig. Though Ramgen's initial goal was to actuate the necessary components prior to backpressure, it was determined that the risk of increasing backpressure during door actuation was small. On the final test day of 2012, Ramgen's test team reached a backpressure of over 1000 psia, a static pressure ratio of 5.4 at MCOS. The concluding test of 2012 was a major step in proving the merit of Ramgen's supersonic compressor technology.

In Q1 of 2013, Ramgen continued performance testing of the Build 1 test rig. Following the "starting" recipe developed in Q4 of 2012, Ramgen was able to successfully start and exceed the maximum backpressure levels achieved in 2012. However, further testing demonstrated that the compressor unstarted and surged well before achieving the predicted pressure ratio. Multiple attempts to adjust secondary flow settings and repeat the tests to achieve higher pressure ratios were unsuccessful. After a number of surges, certain aspects of the actuation mechanism began to behave erratically. It was determined that a disassembly to inspect parts and locate the cause for this behavior was necessary. The disassembly showed damage to a number of pieces of hardware that would need to be repaired.

Ramgen's test operating procedures dictated 15 minute warm-up periods at 30%, 54%, and 94% of design speed. These warm-up periods were no longer deemed acceptable and Ramgen undertook the effort to make the necessary control systems and procedural changes, with related reviews, to ensure safe operation with a constant speed ramp up to maximum continuous operating speed (MCOS). Testing resumed in June of 2013 under the new operating procedures.

On September 6th 2013, Ramgen for the first time tested a fully started diffuser that was able to match pre-test predictions for peak backpressure. Over the course of the next several weeks, Ramgen continued to build on its success, matching predictions for peak backpressure not only at design speed, but at various speeds up to 115% of design speed. Ramgen's successes culminated in achieving a peak pressure ratio of 9:1 in a single stage, meeting or exceeding pre-test predictions (see Figure 6.9), and a peak discharge pressure of 1547 psia with a 210 psia suction. Ramgen also had good agreement with predicted efficiency, coming within 6% of predicted values (see Figure 6.10). The discrepancy in efficiency values was attributed to Ramgen's inability to turn down bleed levels in the diffuser to those predicted in CFD. Corrected test bleed values show a match within the uncertainty of the measurements. Unfortunately, a surge at particularly high backpressure caused another mechanical failure in the rig, but with the good match in data, the test was deemed a success and concluded.

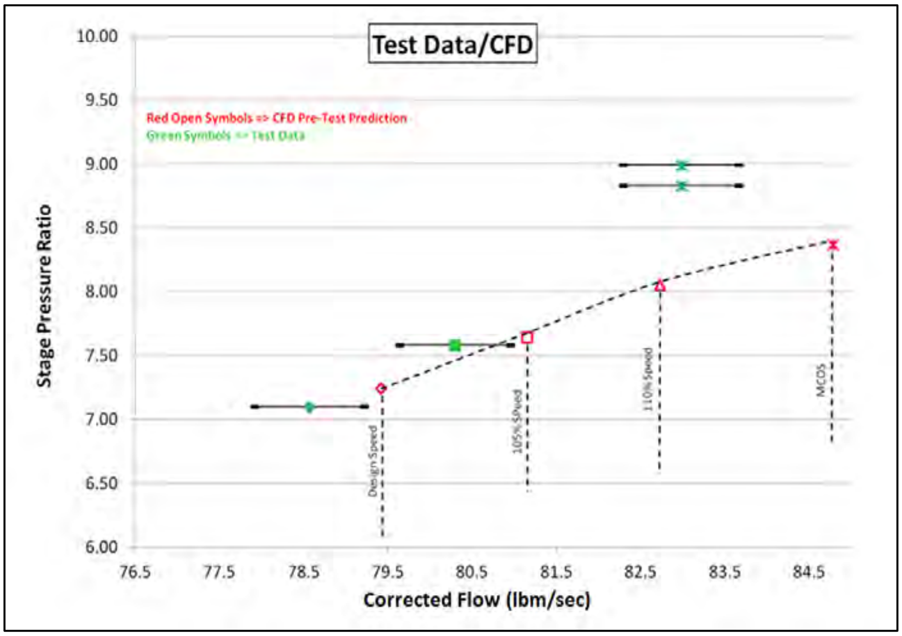


Figure 6.9: Comparison of test results to CFD predictions

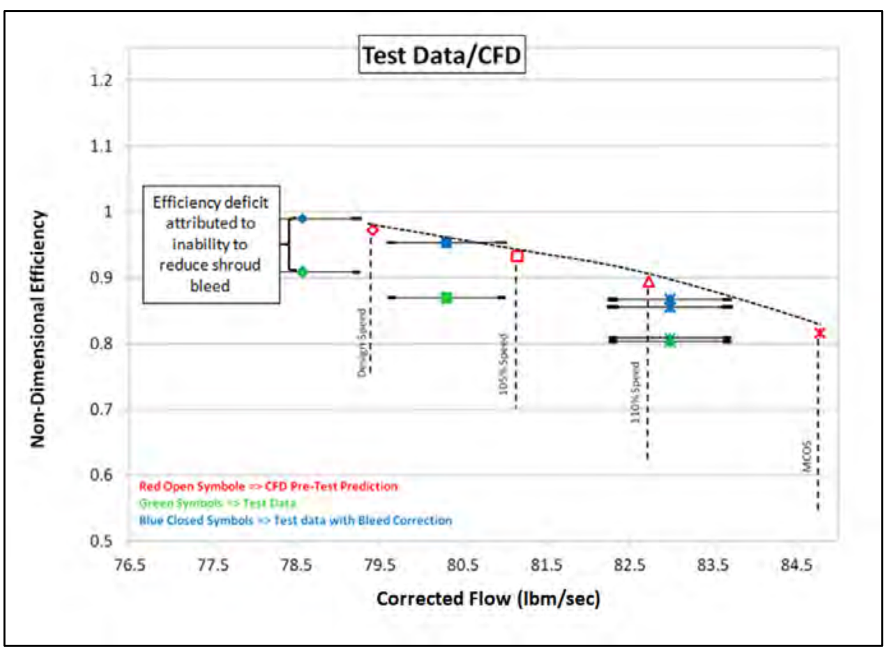


Figure 6.10: Comparison of test and predicted performance data

7. Task 2.6 & Task 3.1 – Test Facility Preparation

A comprehensive Front End Engineering Design (FEED) effort was performed by subcontractor ATSI to establish the demonstration unit test facility requirements, interfaces, and estimated costs. The FEED study studied the economics of main compressor drive using a steam turbine vs. electric motor. An industry-standard HAZOP and P&ID review was also performed to ensure the facility met safety standards.

Ramgen also completed an extensive study of performance instrumentation and measurements required to fully characterize compressor performance.

Steam vs. Electric Decision

After an extensive investigation of the costs and complexity for steam turbine drive (and associated infrastructure requirements) and electrical motor drive, ATSI concluded that electric motor drive was less expensive and better able to perform during upstate New York's cold winters. ATSI's summary report is included as Appendix 7.1 to this report.

Drivetrain

After completion of the steam versus electric trade study and the resulting selection of an electric drive motor/gearbox system for the compressor, multiple vendors were engaged to locate a motor and variable frequency drive (VFD) which could meet the technical requirements at the lowest cost. The VFD was required for motor control, specifically enable a "soft start" feature which limits the in-rush current the motor draws at start up, which is critical for motors in this power class. Vendors evaluated in this phase of the work included:

- ATB Lawrence Scott
- ABB
- Siemens
- Converteam
- DDS
- General Electric
- TMEIC

The gearbox vendor and design effort was also executed in parallel to the motor selection effort, and this ultimately had a major impact on the final motor configuration. Gearbox vendors evaluated as part of the drivetrain design effort included:

- Allen Gears
- BHS/Voith
- Lufkin
- Philadelphia Gears
- Flender

High speed motor vendors (specifically DDS) were initially considered as it was felt that by utilizing the highest possible motor output speed the lowest possible gearbox ratio could be utilized and still achieve the desired ~36,000 RPM rampressor speeed. Motors up to 10 MW have recently been developed by a few vendors that can achieve 10,000 RPM output speeds.

As the engineering effort on this system continued, it was discovered that increasing the motor output speed (and therefore the input speed to the gearbox) in an effort to reduce total gear ratio did not actually simplify the gearbox design. In fact, traditional parallel shaft gearboxes were unable to accommodate input speeds higher than 3600 RPM. The high speed motors that have recently become commercially available were targeted at applicatons in which the goal was to eliminate the gearbox completely. Unfortunately, given the relatively high rotational speed of the Rampressor, this was not a practical option.

Ultimately, a drivetrain design solution utilizing a high gear ratio compound epicyclic gearbox was selected that utilizes a 10 MW motor with an output speed of 3600 RPM.

Motor/VFD

ATB-Lawrence Scott was ultimately selected to supply the 10 MW motor after a competitive bid process. Figure 7.1, Figure 7.2 and Figure 7.3 show typical 10 MW class ATB-LS motors in various states of assembly. The motor was purchased and delivered to the test site in Olean, New York in November, 2010.

ABB teamed with ATB-LS to offer a complete motor/VFD/transformer package to Ramgen. The ABB VFD was actually a test unit that was in service at ABB for about one year. ABB refurbished this unit for use by Ramgen. The transformers were new units.

Figure 7.1: Partially Assembled Typical 10 MW Motor Showing Base Frame and Stator

Figure 7.2: 10 MW Motor Shaft with Windings Installed

Figure 7.3: Nearly Complete Typical 10 MW Motor with Outer Enclosure

Gearbox

A compound epicyclic gearbox was provided by Allen Gears (AG). AG has extensive experience manufacturing this type of gearbox and it is the core of their product line. However, the speed and power combination set by the CO_2 compressor will challenge AG to extend their design space. A preliminary design effort was conducted, with AG concluding the requirements could be met.

The other significant advantage to the compound epicyclic design solution was the compact nature of the gearbox. This enables direct mounting of the gearbox to the compressor case, thus providing the shortest possible high-speed coupling between the gearbox output shaft and the compressor shaft. This was critical to achieve rotordynamic stability given the high operating speeds of the compressor.

Figure 7.4 shows the power and torque available throughout the compressor RPM range.

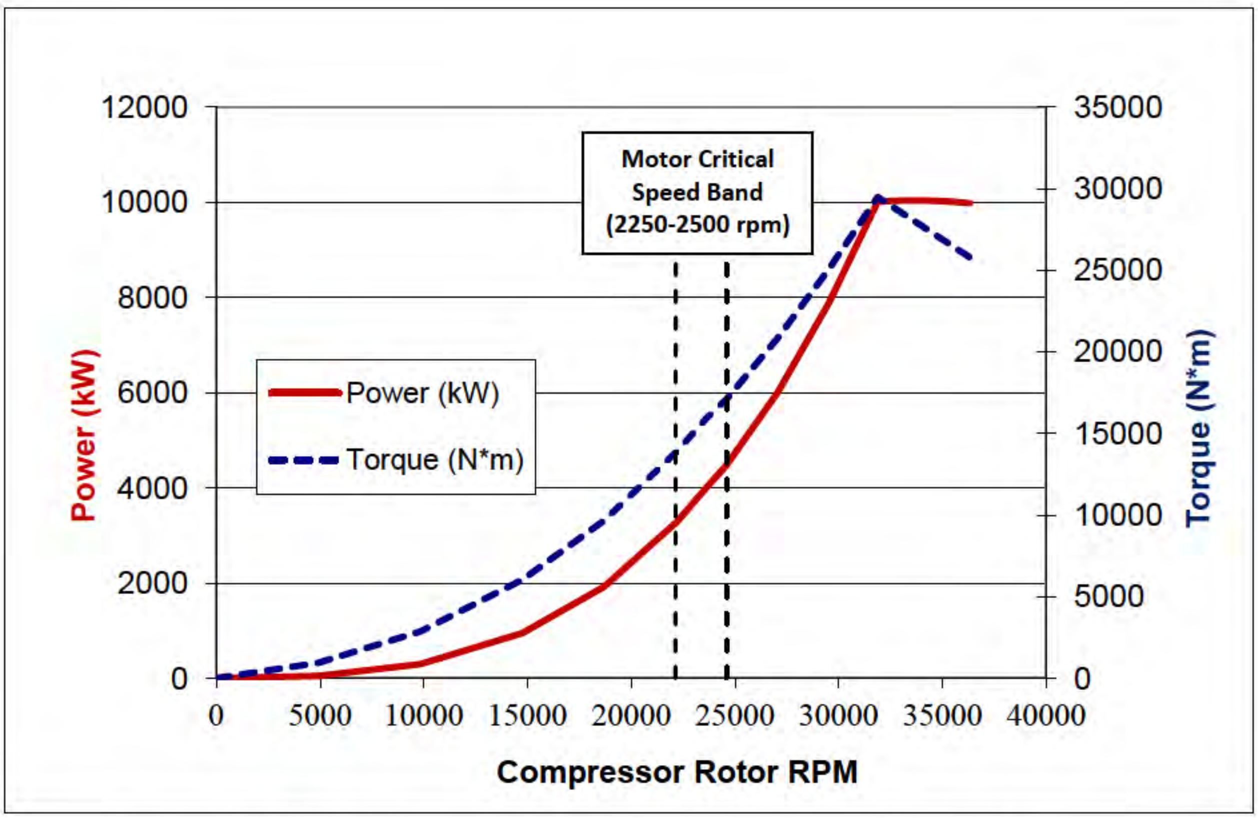


Figure 7.4: Motor Power and Speed vs. Compressor Rotor RPM

Facility Layout and Configuration

As the facility design progresses, 3D models have been generated to ensure adequate space for personnel and components and to enable group discussion of facility operation and construction. The following figures show the general layout of the facility building and major components, as well as preliminary piping connections.

Figure 7.5 shows a top (Plan) view of the facility buildings. The large building at center houses the compressor test operations and the closed-loop system. A yellow overhead crane can be seen near the top of this building. At right stands the cooling tower used to disperse the 10 MW heat of compression. The violet objects at far left are 34 kV transformers for the compressor electric motor. Between the transformers and the compressor test building stands the electrical/control building. Besides housing the Variable-Frequency Drive, this building houses the test operators and data acquisition systems.

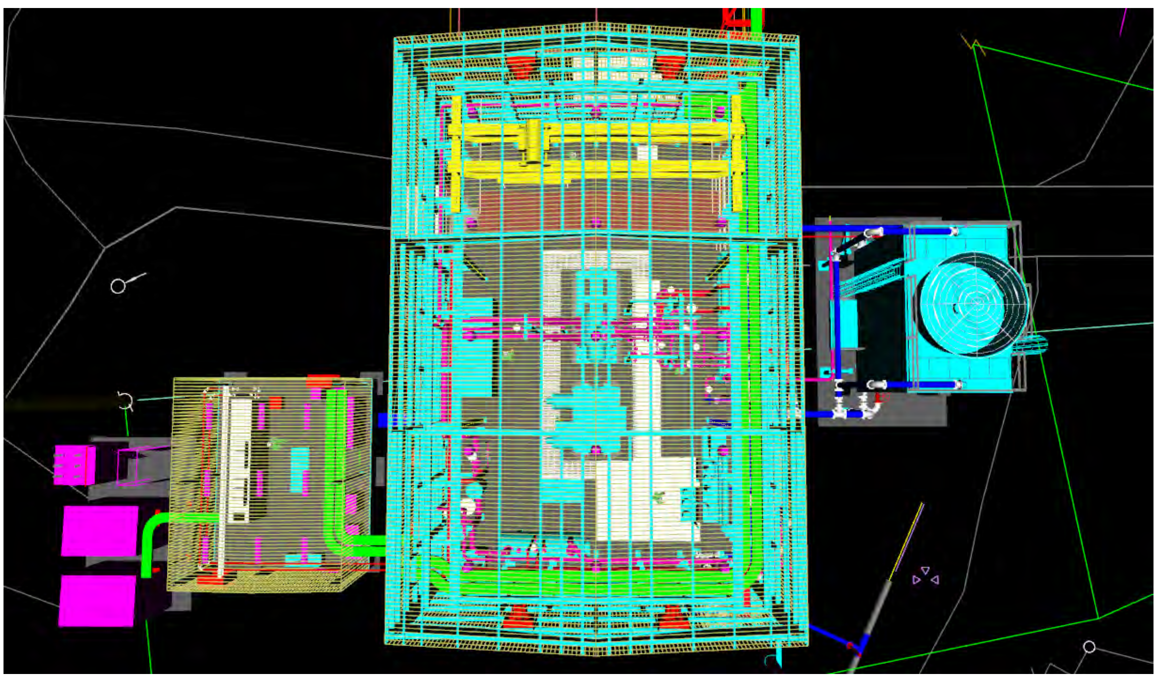


Figure 7.5: Top View of Facility Buildings and Major Components

Figure 7.6 is a closer view of the compressor and drive motor from above. The drive motor (with top-mounted cooling fans) sits on center just below the compressor/gearbox. Dual suction pipes (violet in color) connect to the left side of the compressor case as described elsewhere in this report. A smaller discharge pipe connects to the right side, along with other service connections.

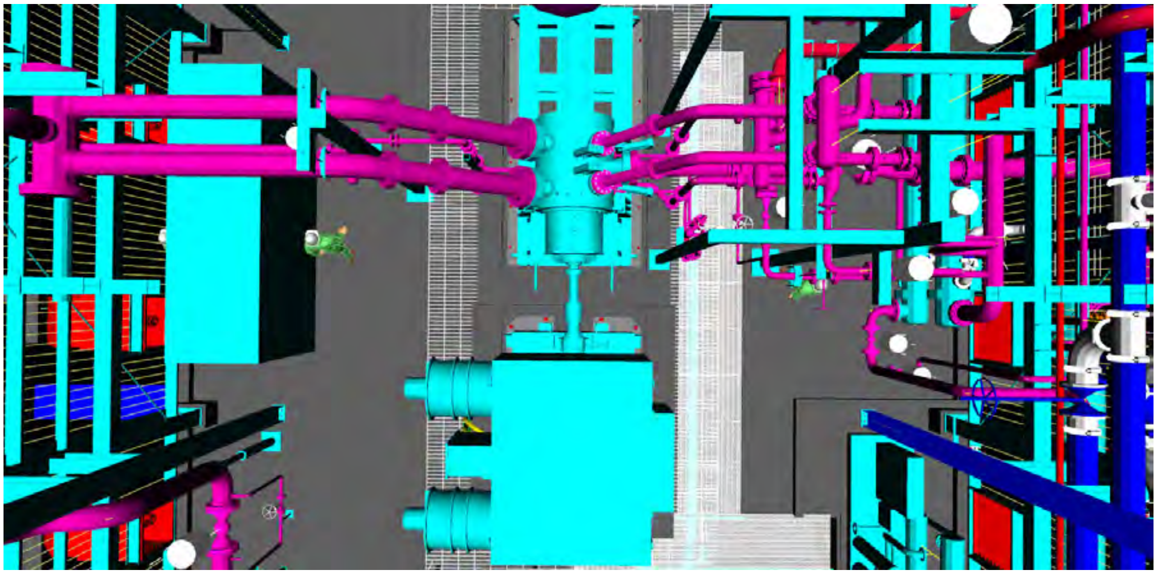


Figure 7.6: Close-Up of Motor/Compressor Arrangement

Figure 7.7 shows a view from the facility floor, looking at the compressor's non-driven end. This view would be looking down from the top of the page in the previous two figures. Service and closed-loop plumbing can be seen, particularly on the left side of the graphic. Large suction pipes connect on the right; smaller discharge pipe is on the left.

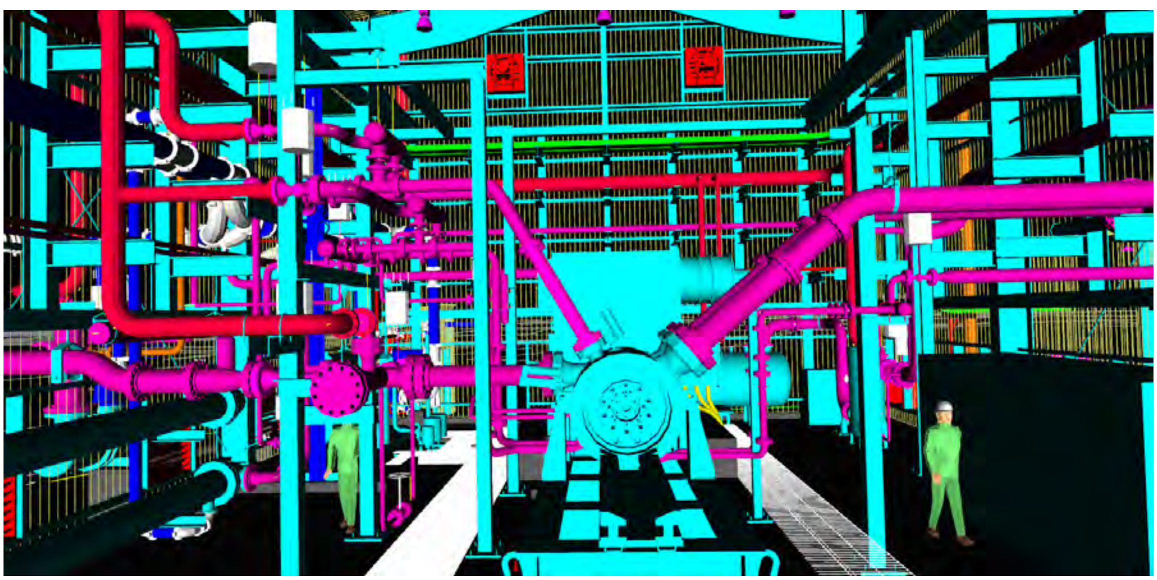


Figure 7.7: Side View of Facility Floor Area

HAZOP and P&ID Review

A combined HAZOP and P&ID review was led by an outside consultant to identify any unexpected facility hazards and provide means to reduce risk. This meeting proved highly useful, resulting in several actions which ATSI, Ramgen and D-R have undertaken to improve the facility design and operability. The entire contractor HAZOP report is included in this report as Appendix 7.2.

2011

In consultation with D-R, Ramgen has thoroughly reviewed existing industry standards (ASME, PTC, etc.) to ensure compliance with best measurement practices. Although a line-by-line listing of all instrument channels is beyond the scope of this report, a Data Analysis Plan (DAP) has been developed which documents the specific measurements and approaches needed to measure the unique characteristics of the supersonic compressor and ensure that compressor performance was quantified in an industry-approved manner.

Installation of the motor and VFD were completed by September 2011 in preparation for bald rotor spin, which had been slated for late September 2011. The test was delayed until December following two hardware problems. The first was an issue with the motor bearing system, which was not discovered until system checkout. The bearing required a redesign, which was completed and delivered to the Olean facility. The second delay was related to the gearbox designed by Allen Gears when the shipping container received significant damage during shipping. In order to assure the integrity of the gearbox, it was returned to Allen Gears for disassembly. Following checkout and reassembly by Allen Gears, the gearbox was shipped back to Olean for installation, and was delivered without incident.

On December 20th 2011, Ramgen began bald rotor testing. Initially, the rotor was accelerated to ~2000 rpm and then held at that speed while data was reviewed to verify instruments were reading properly. The rotor speed was increased in ~500 rpm increments with the new speed held for at least 30 seconds before changing speed again. Eventually a rotor speed of 9000 rpm was attained. This speed was held for approximately 20 minutes to allow the gearbox to reach equilibrium temperature per the manufacturer's instructions. The speed was then reduced in steps of 1000 rpm until the rotor was stopped. The duration of the test was over one hour and twenty minutes.

The lubrication supply conditions were within specifications during the test as was the bearing temperatures and lubrication return temperatures. The vibration data showed orbits under 0.5 mils for the compressor. The magnitude of the gearbox orbit was consistent with test data from Allen Gears. The test was stopped because there was a small oil leak present and there was concern about the amount of oil lost during the test. Post-test examination showed a loss of approximately 1 gallon.

Data from the test was analyzed during the report period. Particular attention was given to the data gathered by the vibration monitoring system. The maximum speed of the motor during the test was 920 rpm which was below the speed where concerning behavior was seen in motor only tests. As expected, the motor orbits were well behaved during the test. Vibration data from proximity probes monitoring the gearbox low speed input shaft are shown in Figure 7.8. The left figure shows the input shaft centerline orbit at the highest speed obtained during the test. The trace shows 20 shaft revolutions overlaid with a first order (1E) filter applied. At 2.09 mils pk-pk it was the largest orbit in the drive train, however, the orbit was very stable. The spectrum plot (right) shows a first order dominate character.

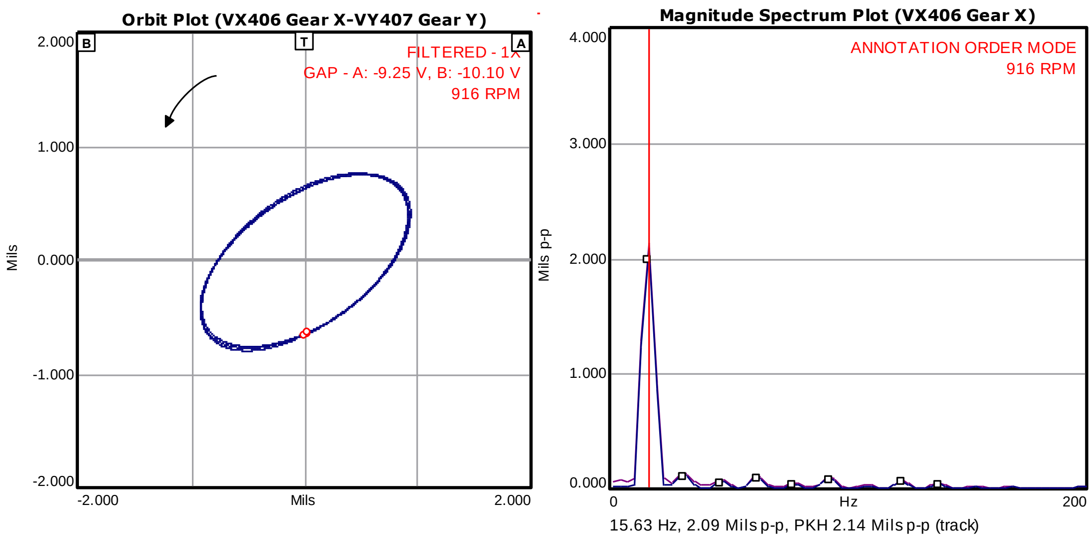


Figure 7.8: Twenty orbits of gearbox input shaft centerline with a first order filter applied (left) and magnitude spectrum plot showing first order dominance (right).

The compressor shaft orbits were well under 0.5 mils during the course of the test so the size of the orbits did not cause concern. However, the magnitude spectrum analysis of the driven end of the compressor (see Figure 7.9) showed a second order peak and some sub-synchronous noise with a fairly well formed peak at 18.75 Hz. Analysis of the non-driven end (see Figure 7.10) showed first order dominance and little sub-synchronous noise.

Gearbox temperature data from the test were compared with data obtained from Allen Gears. The difference in temperature between the bearings and the supply oil was found to be lower than seen by Allen Gears. This was attributed to the difference in test environment. Allen Gears performed their test in a heated test cell while the Ramgen test was conducted in an unheated building on a cold day with fans blowing cool air across the test article.

More rotordynamics analysis was performed to understand the behavior of the system and the different modes seen during testing.

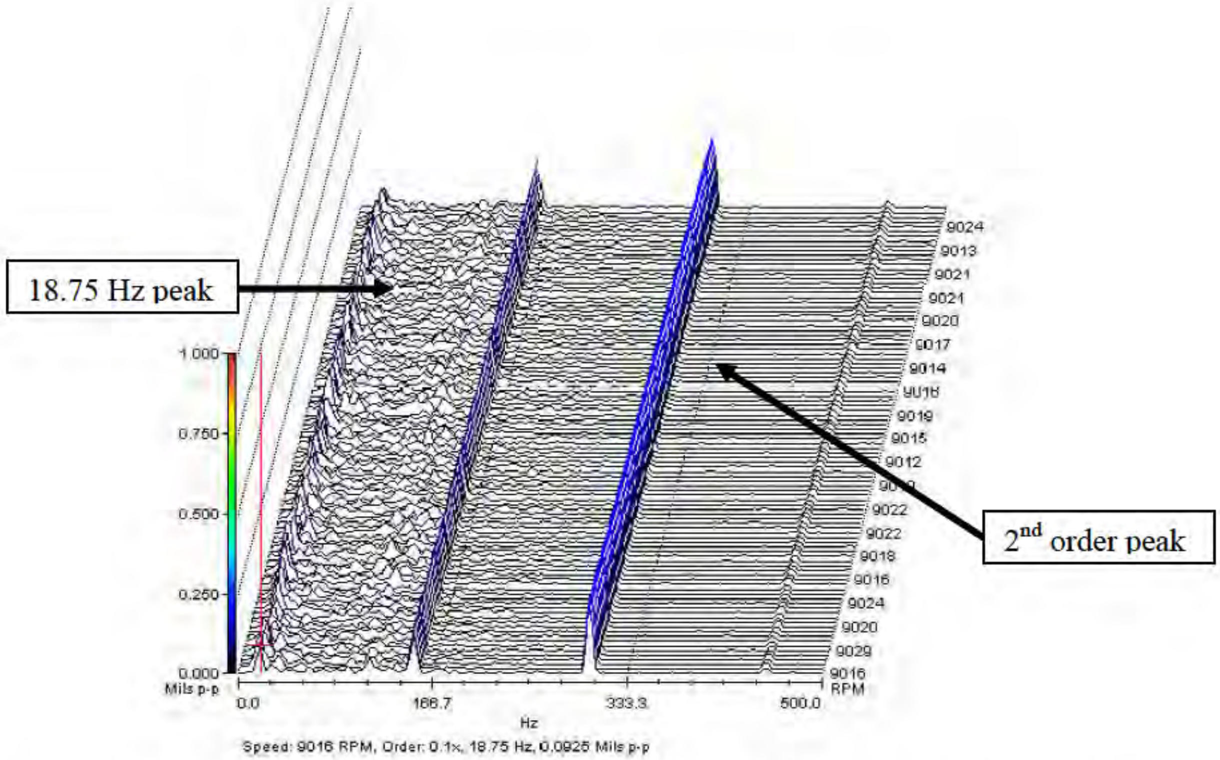


Figure 7.9: Magnitude spectrum plot of compressor driven end journal bearing showing a dominant second order peak and sub-synchronous noise with an 18.75 Hz peak.

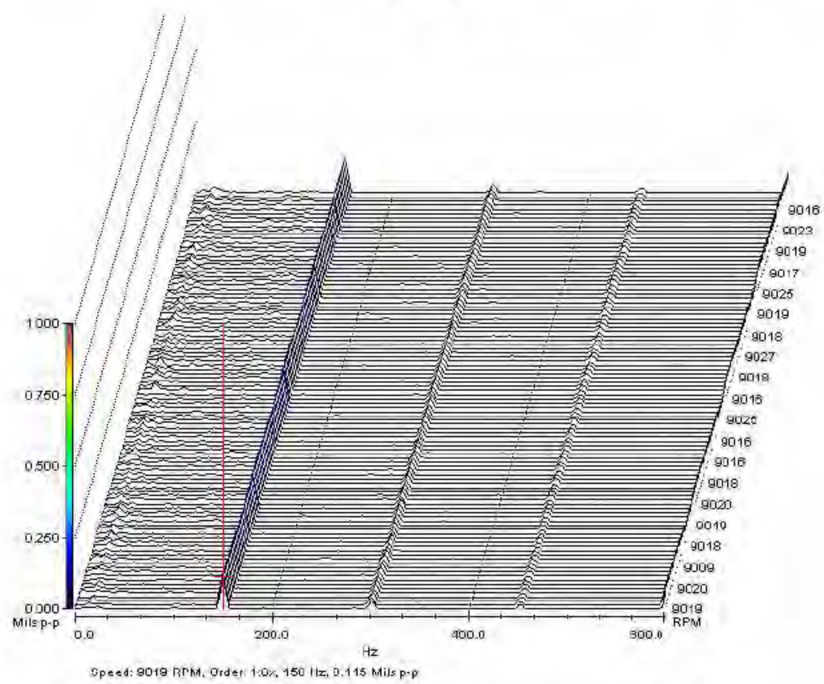


Figure 7.10: Magnitude spectrum plot of compressor non-driven end journal bearing showing first order dominance with very little sub-synchronous noise.

The new rotordynamic analysis predicted a mode at 36,000 rpm at the design point conditions. However, Allen Gears data indicated the stiffness of the high speed coupling to sun gear interface drops as the gearbox load was reduced. The analysis predicted a mode at 30,000 rpm, the approximate speed at which the mode was seen during test, if the stiffness was reduced by 92.5%. This level of stiffness was consistent with the Allen Gears data as the bald rotor had no aerodynamic surfaces and resulted in very little drag.

In an attempt to test the new model, it was decided to make a rotor with increased aerodynamic drag to provide a greater load to the gearbox to increase the drivetrain stiffness. To accomplish this, the bald rotor was modified to increase its aerodynamic drag. The resulting rotor was called the fuzzy rotor, seen in Figure 7.11. CFD analysis predicted a power requirement of 468 hp at 30,000 rpm with the pressure case at 212 psia.

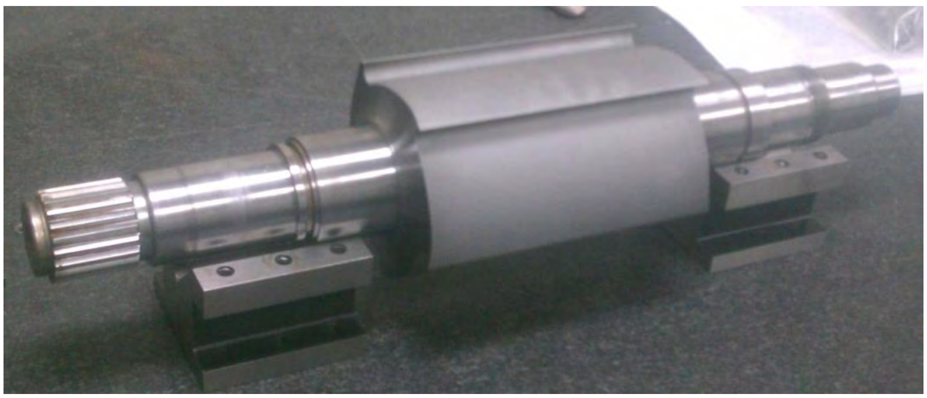


Figure 7.11: Photograph of fuzzy rotor.

The first test to be conducted used atmospheric pressure nitrogen as the working fluid. The purpose of this test was to verify the mode near 30,000 rpm was still present and to provide baseline data for comparison with higher pressure cases. The rotor was accelerated following previously described procedures with stops at three speeds to allow the gearbox to warm up. The first attempt used a final gearbox warm up speed of 2800 rpm at the motor. Once the gearbox was warm, the motor speed was increased to 2900 rpm for approximately 30 seconds. The motor speed was then increased to 3000 rpm for another 30 seconds. Finally, the motor speed was increased to 3100 rpm. After approximately 25 seconds, the drive train tripped offline due to the orbit exceeding 0.0025 inches.

Figure 7.12 shows a Bode plot of the compressor drive end X proximity probe from the test. The blue line is during acceleration and the green line is during deceleration. Note the increase in synchronous component magnitude while holding at a rotor speed of 30,500 rpm (motor speed of 3100 rpm) which resulted in a drive train trip. The corresponding increase in orbit size is clearly seen in the Bode plot.

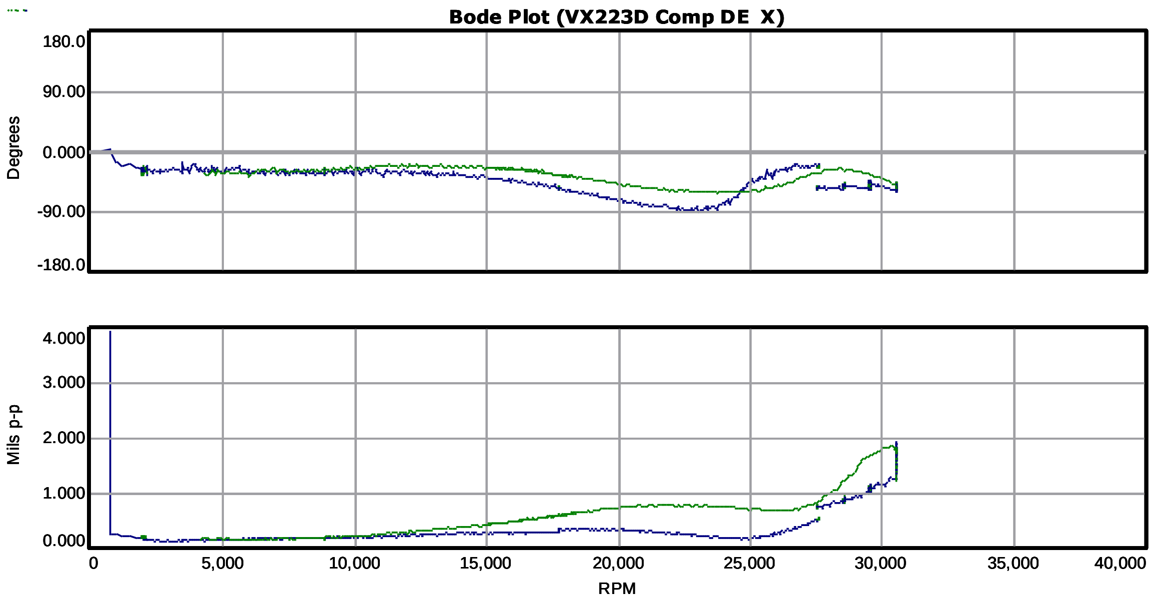


Figure 7.12: Bode plot of rotor driven end X probe during first fuzzy rotor test.

Figure 7.13 shows waterfall plots for the rotor driven end X and Y proximity probes during the deceleration after the trip. The large synchronous motion was clearly seen along with significant sub-synchronous noise, as well as noise between the first and second order components. The higher order components did not appear to change with speed indicative of electrical run out. As the synchronous component was the only mode of significance this additional data indicated the larger orbit magnitude was due to the rotor balance.

The tendency of the rotor driven end orbit size to increase while holding at rotor speeds between 30,000 and 33,000 rpm was also observed during bald rotor testing. By continuously accelerating from 30,000 rpm it was possible to achieve full speed. The test was repeated with this method employed in an attempt to reach full speed. The final gearbox warm up speed was with a motor speed of 2900 rpm. Once the gearbox was warm, a motor speed of 3600 rpm was commanded. The rotor driven end orbits increased with speed until the trip limit of 0.0025 inches was reached at a motor speed of 3390 rpm. As the rotor decelerated, the orbits continued to increase reaching 0.0041 inches at a rotor speed of 30,900 rpm.

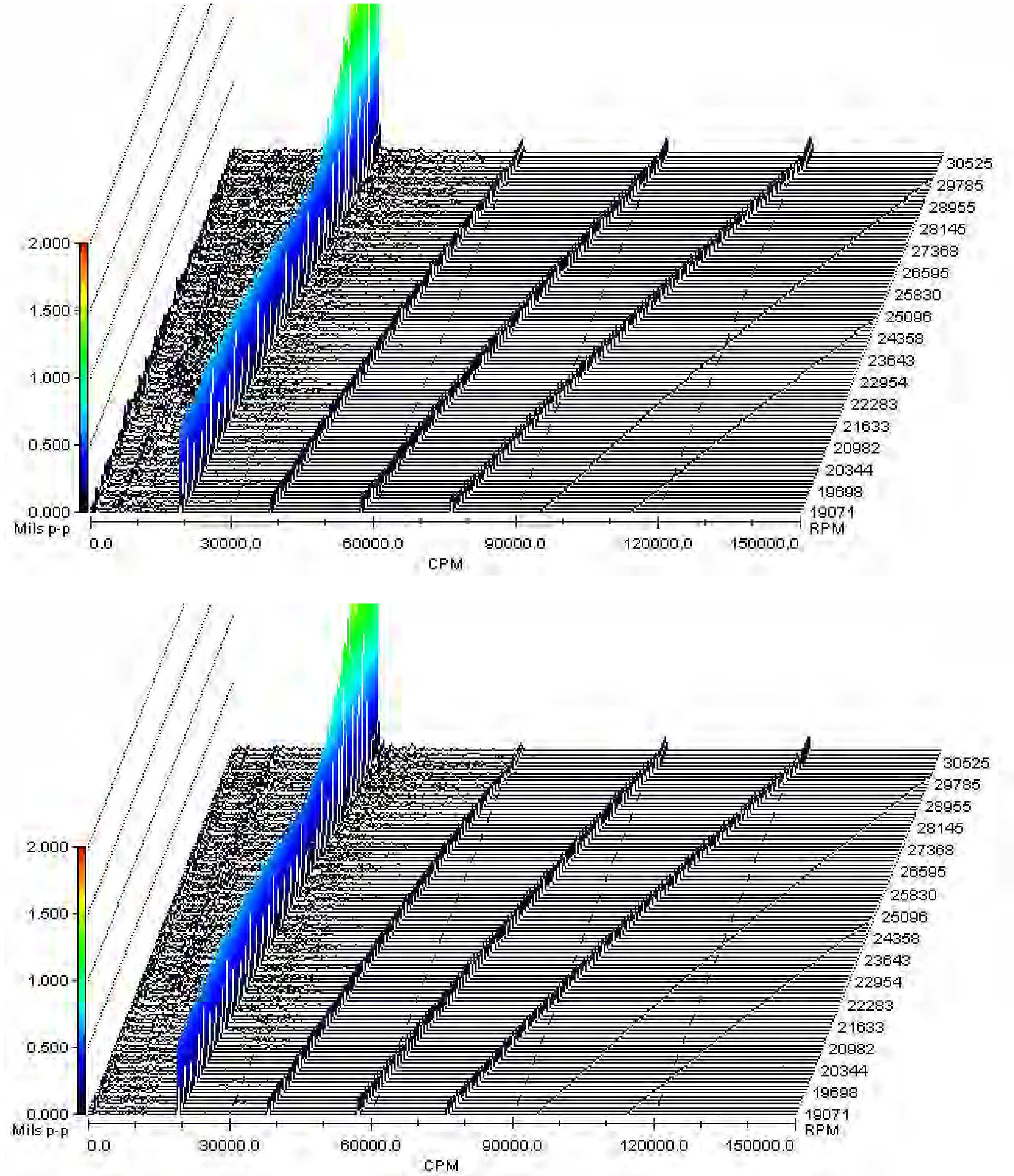


Figure 7.13: Waterfall plots of rotor driven end X (top) and Y (bottom) probes during the first fuzzy rotor test.

Figure 7.14 shows a Bode plot for the rotor driven end X proximity probe from the test. The acceleration phase shows the synchronous magnitude continuously increased to the trip point and clearly indicated that the system had not passed through the mode peak. The deceleration phase showed the large amplitude along with passage through a mode. Comparison with Figure 7.12 showed very similar behavior with larger amplitudes on the deceleration phase above 25,000 rpm. Both figures showed a mode peak during deceleration which implies the gearbox stiffness was lower during deceleration which was reasonable since the splines and gears mesh differently during acceleration and deceleration.

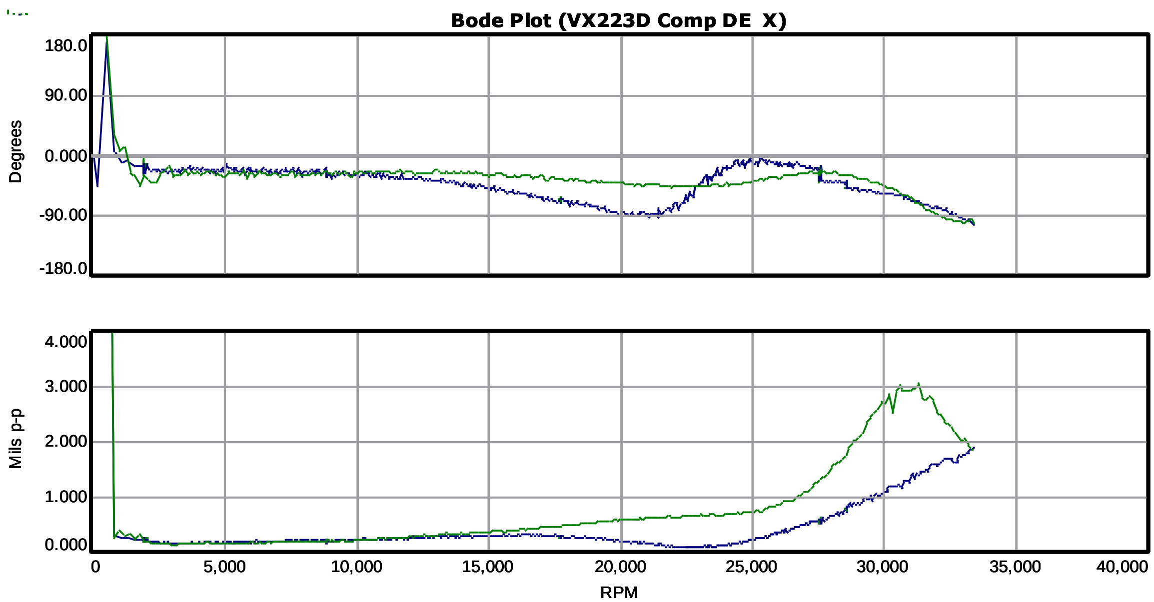


Figure 7.14: Bode plot of rotor driven end X probe during second fuzzy rotor test.

Figure 7.15 shows waterfall plots for the rotor driven end X and Y proximity probes during the deceleration after the trip. Again, the large synchronous motion was clearly seen along with significant sub-synchronous noise and noise between the first and second order components. Comparison with Figure 7.13 showed similar behavior but an obviously higher synchronous component and more noise present, particularly during the period with high orbit magnitudes.

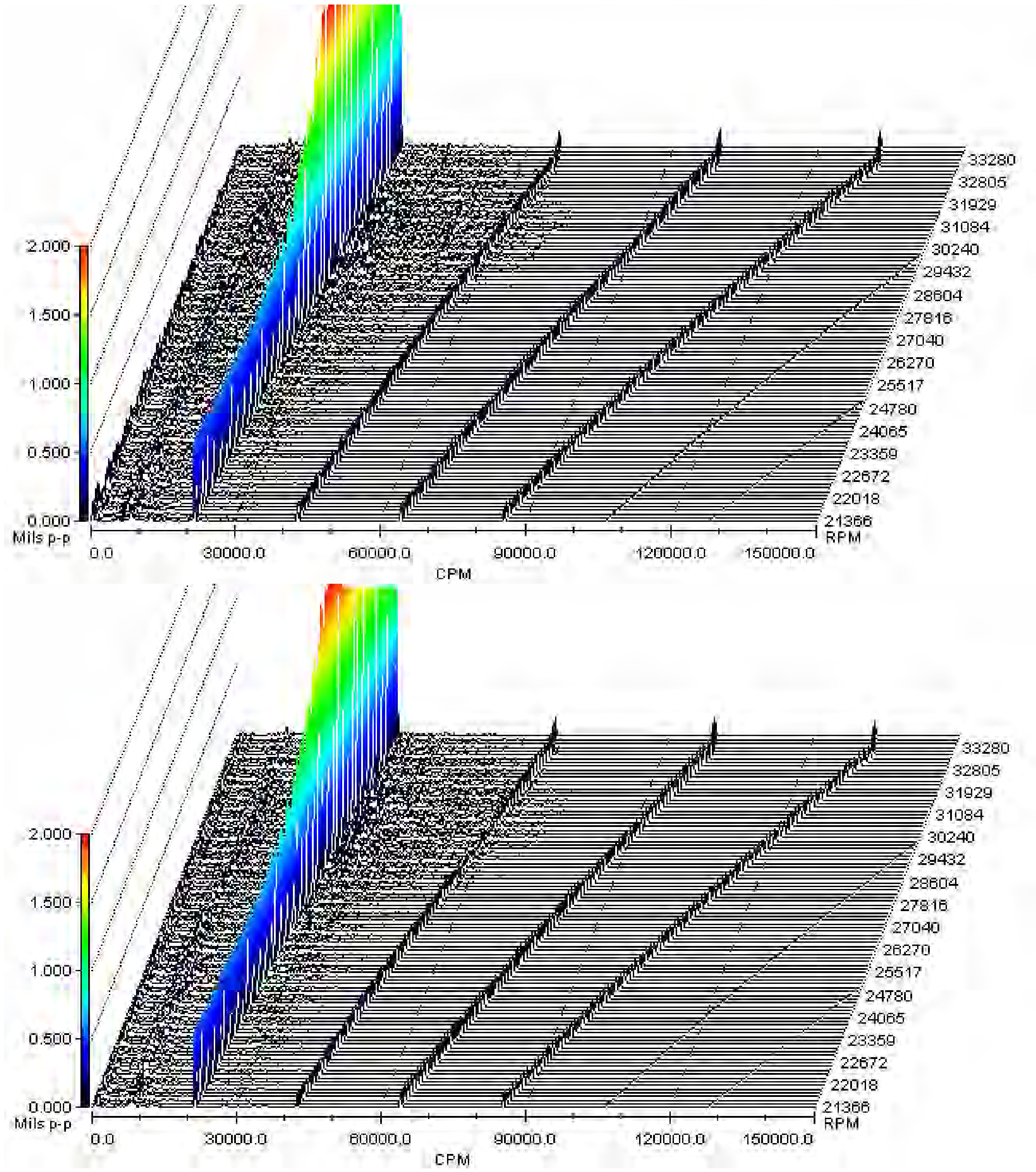


Figure 7.15: Waterfall plots of rotor driven end X (top) and Y (bottom) probes during the second fuzzy rotor test.

The atmospheric pressure tests failed to reach full speed and indicated that trim balancing of the rotor would be required. However, due to time constraints, it was decided that a pressurized test should be conducted to see if a shift in the mode seen during deceleration could be obtained. The test plan called for several steps between an atmospheric pressure test and a 200 psia test, the first being at 50 psia.

The flow loop was filled to 50 psia and the drive train was started. The standard gearbox warm up procedure was followed with the last gearbox warm up occurring at a motor speed of 2900 rpm. While at 2900 rpm the suction pressure had an average of 50.12 psia, a minimum of 49.85 psia, a maximum of 50.38 psia, and a standard deviation of 0.12 psia. Once the gearbox was warm the motor was commanded to a speed of 3600 rpm. The rotor drive end orbits increased with speed until the trip limit was reached at 3110 rpm.

A Bode plot for the test is shown in Figure 7.16. When compared with the previous two tests, Figure 7.12 and Figure 7.14, it can be seen that during acceleration the magnitude of the synchronous component was larger for speeds above 23,000 rpm. It is also interesting to note that the local minimum was at 21,000 rpm compared to 25,000 rpm for the first fuzzy rotor test and 23,000 for the second fuzzy rotor test.

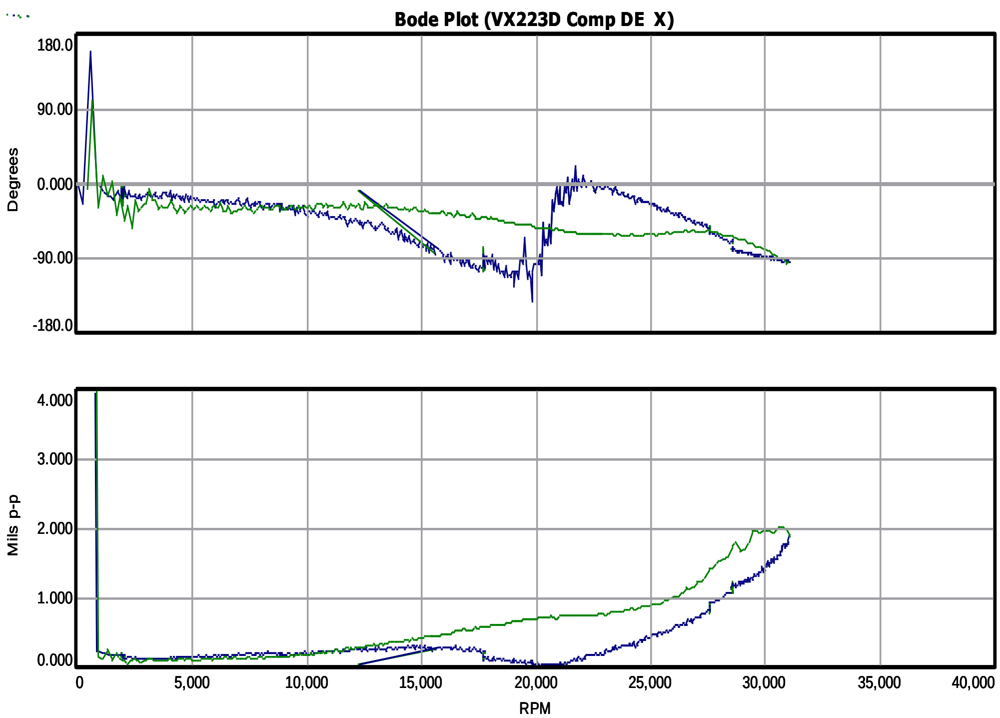
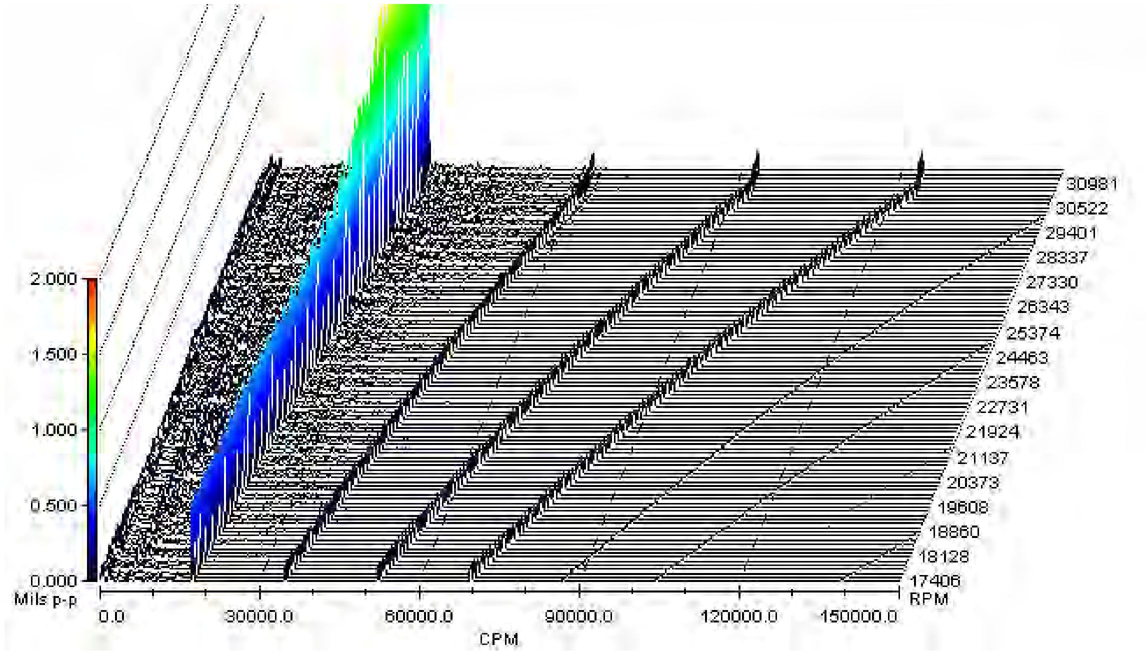


Figure 7.16: Bode plot of rotor driven end X probe during pressurized fuzzy rotor test.



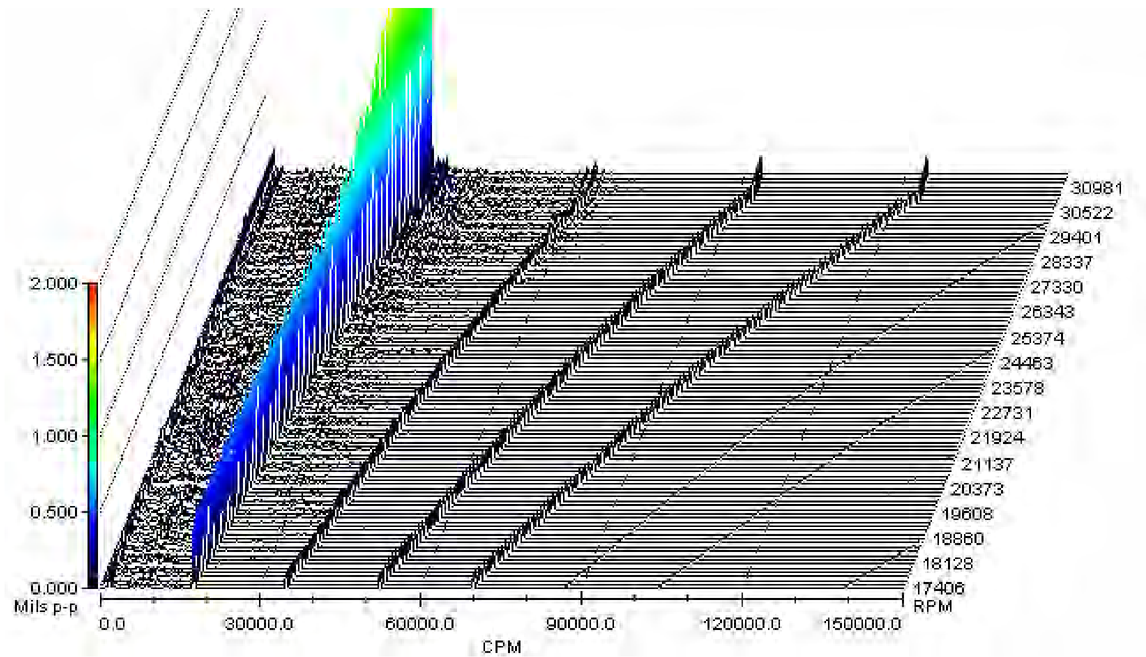


Figure 7.17: Waterfall plots of rotor driven end X (top) and Y (bottom) probes from the pressurized fuzzy rotor test.

Figure 7.17 shows waterfall plots for the rotor driven end X and Y proximity probes during the deceleration after the trip. The plots are very similar to the previous waterfall plots showing the large synchronous motion along with significant sub-synchronous noise and noise between the first and second order components. Unfortunately, the pressurized test did not attain a high enough speed to understand the location of the mode seen during deceleration. Higher vibration levels were seen during the pressurized test but remain unexplained. The fuzzy rotor testing was halted in order to keep performance rotor testing on track.

Testing on October 26 and 29, 2011 focused on determining the source of the sub-synchronous vibration (SSV) observed in earlier testing. The high speed (HS) coupling was inspected. Small axial wear marks on one end of the HS coupling spline were observed (Figure 7.18).

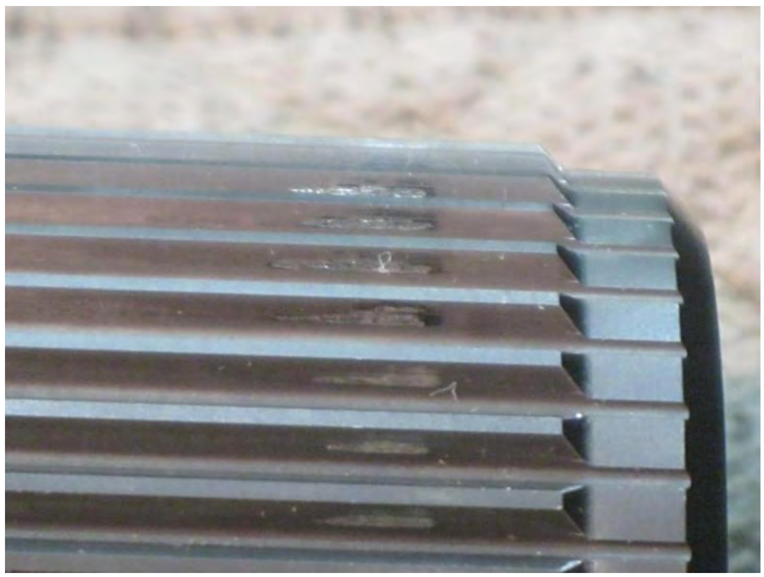


Figure 7.18: HS coupling spline wear marks

Allen Gear's opinion of the wear marks was that it was a normal phenomenon due to partial spline tooth surface engagement at lower loads. As to the SSV, AG recommended that we install an O-ring on the end of the HS coupling shaft to potentially dampen vibrations and better retain oil flowing between the teeth. We also discovered some vibration results indicating a need to re-balance the low speed (LS) coupling. Subsequent tests gave further credence to an LS coupling imbalance. The LS coupling was re-balanced and new instrumentation was added to track gearbox vibrations and accelerations.

On the next test the data indicated a successful LS coupling balance with a reduction of 1X amplitudes. Some positive impact on the SSV was observed which was attributed to the new O-ring on the HS coupling shaft. Due to continued presence of SSV after the LS balance we decided to increase the loop pressure from 50 to 100 psia.

After increasing loop pressure to 100 psia we saw a dramatic and sudden drop in the SSV which also corresponded with significantly cleaner and lower amplitude orbits at several locations on the drive train. The change was most apparent on the HS coupling proximity probes. Unfortunately, the elimination of the SSV was accompanied by a corresponding increase in rotor 1E vibration levels. The sudden change in orbits occurred at specific speeds, namely 2700-2800 rpm on acceleration and 900-1000 rpm on deceleration.

On November 9, 2011 we continued to explore this new behavior at 100 psia, making some changes to gear box oil flow rates and pushing speed higher. Speeds topped out at a motor speed of 3200 rpm. After some discussion and review of the data it was decided that a HS coupling balance should be performed to reduce the current 1E vibration level on the compressor DE. The loop pressure was increased to 150 psia, resulting in ~136 psia suction pressure. At this pressure, the rig achieved MCOS without tripping offline and achieved a peak power of ~6.4 MW. Other positive results from increasing power included reduction of the sub-synchronous drive train vibration levels at lower speeds than previously observed, and indications of a potential diffuser starting event.

8. Task 2.8 & Task 3.3 – Product Traceability

There were two major activities pursued by Ramgen to ensure the demonstration compressor was traceable to a product/production configuration. The first effort was to work with Dresser-Rand to understand how this new technology compressor will address many common product concerns like materials durability, operability, etc. In 2013, Ramgen's industrial partner Dresser-Rand analyzed the commercial characteristics of the Ramgen design and made a decision on the preferred configuration of the HP CO_2 compressor in 2014. The configuration was the Super Compressor configuration tested as the Build 2 HP CO_2 Compressor in 2015.

The second effort Ramgen undertook as part of this task was to analyze how the Ramgen compressor could be deployed in a coal power plant to best utilize heat of compression in the plant cycle. Because efficiently using the heat unique to Ramgen is essential to reducing the cost of CCS, Ramgen continues to seek ways to work with power plant engineering firms to define how the Ramgen compression process can be coordinated with the capture process to maximize reductions in plant operating costs. To date, Ramgen's efforts to integrate the Ramgen compressor into a power plant have provided encouraging indications of value.

- Ramgen can provide ∼275 Btu/lbm-CO_2 as heat of compression recovered at 100°F from the LP and HP discharge streams with discharge temperatures at ∼500°F.
- Conventional amine-based solvent regeneration requires 1530 Btu/lbm-CO_2
- Mitsubishi Heavy Industries offers their KS-1 advanced hindered amine which requires 1200 Btu/lbm-CO_2, and do utilize heat recovery from their inline compressor offerings discharging at ∼350 F. They quote a "net of heat recovery" regeneration heat requirement.
- Solvent regeneration heat is typically provided by steam drawn off the main steam turbine between the IP and the LP casings and can be as much as ½ of the LP

flow. The LP casing provides 50% of the power in conventional power plant architecture; therefore the power plant is de-rated by 25% to provide this heat.

- Regeneration occurs at 275 F and heat can be used down to 250 F without concern over solvent degradation
- The CO_2 exit pressure from amine-based solvents is typically 22 psia.
- Chilled ammonia has a lower regeneration heat requirement at 860 Btu/lbm-CO_2 and exits at 300 psia. Ramgen would apply only the HP stage of compression for this application.
- About half of Ramgen's heat of compression can be used to offset rebuilder duty; the other half can be used to replace a portion of the steam used for feed water heating. There are typically seven heaters in the feed water heater train. The specific feed water heater identified is quite site specific, but Ramgen's higher temperature will allow it to replace the higher value steam in the train.

It needs to be understood that a detailed integration analysis, including cost comparisons, requires significant funding and plant operator cooperation to complete. The feedback we gathered was that the effort required would be comparable to a power plant CCS FEED study. There were no standard methodologies to apply heat integration as it is site specific with too many variables to establish a best set of practices. Important site specific variables include the demographics of where the power plant is located, distances between power plant components, the ambient conditions including altitude, cooling medium available, design temperatures and hot to cold ranges thereof.

The type of fuel, type of power plant and type of capture system all affect the specific design of any heat recovery approach. The various solvent or other capture approaches each have their own unique set of requirements for regeneration, many of which remain trade secrets of their developers and not available to Ramgen or any other outside organization for review. In addition, new, greenfield power plants represent a very different set of issues than do existing power plants looking to retrofit a CCS system.

The review only cites steam raised to be used in the power cycle, and it is important to note that other uses for this heat of compression that Ramgen has considered include: Heat of compression CO_2 dryers; Coal Drying - with low rank fuels in common and growing use; Boiler air pre-heating – improves the boiler efficiency in certain situations; Flue Gas Reheating – improves the buoyancy of the flue gas after being subjected to various refrigerated level CCS processes; Feed water Heating – as mentioned, can be used as a secondary heat recovery of residual heat of compression following any of the applications above; and, Organic Rankine Cycle – Dresser-Rand and others are exploring the use of Organic Rankine Cycle to recover the heat of compression.

9. Task 3.7 – CO_2 Compressor Retrofit

Build 1 CO2 compressor testing was completed on the Olean NY test stand in October of 2013. After the test data anchored the analytical predictions the next build, Build 2, configuration was reviewed with Dresser-Rand for performance potential as well as commercial reliability. Dresser-Rand conducted a detailed internal review of the Ramgen technology and test results. The analysis yielded a configuration that combined the benefits of supersonic compression from Ramgen's successful testing with Dresser-Rand commercialization experience on CO2 and industrial compressors. This new configuration was originally called the SuperCompressor. It is now referred to as the DATUM-S. The Build 2 CO2 compressor was originally targeted to be completed by June 30, 2014. The DATUM-S configuration would involve a new case and flow path geometry. Dresser-Rand and Ramgen approached the DOE on the possibility of granting a No-Cost extension to complete the project by March 31, 2015. The No-Cost extension was granted in April 2014. The components that were replaced or significantly modified are discussed in this section.

9.1 - Flow path Aerodynamics

Compressor aerodynamic design work under this contract in 2014 was split between two design concepts. The first concept was managed by Ramgen Power systems as a continuation from the 2013 design through Q1 2014 and the second (DATUM-S) by Dresser-Rand from the middle of the second quarter of 2014 onward.

Following a detailed analysis of the Build 1 CO_2 compressor test completed under Task 3.4 of this contract, Ramgen identified several primary sources of aerodynamic losses in the supersonic flow path which could be improved in the next design. Additionally, operation of the Build 1 compressor required a complicated start-up procedure utilizing variable geometry and sensitive monitoring and manipulation of secondary flow-control features, all of which were deemed undesirable for an industrial compressor application. To address both high aerodynamic losses and complicated run characteristics, work on a two-stage compressor was begun in Q1 2013 Work continued on this design to mid-Q2 2014.

Following the acquisition of Ramgen by Dresser-Rand the DATUM-S concept was ultimately determined to be the better choice for Build 2 based on a commercialization evaluation conducted by the team. The DATUM-S design work was completed in 2014. Drawings were released in late 2014 for manufacturing in support of a March 2015 test date.

The DATUM-S is a single-stage supersonic compressor with transonic inlet flow and vaneless diffuser. When work on the design began under this contract, the majority of the aerodynamic flow path design had been completed. Focus was placed on detailed modeling of the secondary and seal flow paths to assess their impact on the compressor performance, as well as provide design feedback to the mechanical team. Detailed flange-to-flange pre-test performance predictions were modeled in CFD, and a pre-test performance design review was held in December 2014 to review the progress. Testing started in March 2015 with a range of geometric configurations.

Design work on the Build 2 DATUM-S was launched near the beginning of Q3. Based on the current compressor design, the Build 2 design is the result of an optimization of 30+ design parameters defining the geometry of the compressor and diffuser flow path. Methods utilized in this process were developed at Ramgen during the Build 1 design processes. Computations were run at the Oak Ridge Leadership Computing Facility at Oak Ridge National Laboratories utilizing the Titan supercomputing cluster under an ALCC grant. Initial results have shown significant performance gains, with the possibility of further gains if coupled to a vaned diffuser. Optimization work was completed in early Q1 2015.

9.2 Flow path Mechanical

In late 2013 the split-blisk Inducer design was released for manufacturing. However, prior to machining commencing, the configuration decision was put on hold pending the evaluation of the Build 2 CO_2 compressor design by Dresser-Rand. The result of Dresser-Rand's deliberations was to proceed with the DATUM-S, configuration. Once the strategic direction for the Build 2 configuration had been determined, with the compressor now envisaged as a single stage flow path, resources were redeployed onto the DATUM-S program.

A summary of the challenges faced with the flow path Design is provided in Appendix 9.2.1 Flow Path Design.

Design of the flow path proceeded and in August 2014 a Final Design Review (FDR) was held. See Appendix 9.2.2 Flow Path Final Design Review for the FDR material. At the rear of the flow path was a large cylindrical surface that provides sealing and rotordynamic damping capability. The diametral location of the seal was set such that at design point there was a small net thrust load away from the gearbox.

In late August a Production Readiness Review (PRR) was completed, the drawings released and manufacturing commenced. The PRR material is included as Appendix 9.2.3 Flow Path Production Readiness Review.

Milling, turning, shot peening and NDT of the flow path was completed in early December 2014 and profile machining and blade tip hard coating were completed in mid-Jan 2015 in support of the target March test date.

9.3 Static Structure

The DATUM-S configuration had a number of distinguishing characteristics. The configuration has a high total pressure ratio flow path with a sub-sonic inlet and supersonic exit. In this configuration flow exits the flow path and enters a diffuser. The rotational speeds needed to produce a high total pressure ratio require a smaller diameter flow path. Design review material of the static hardware design is included in Appendix 9.3.1 Static Hardware Design Reviews.

Rotordynamic stability was achieved in part by a rotor seal on the flow path. The rotor seal serves three main functions:

1) Aid in the rotordynamic stability of the unit,
2) Reduce the amount of high pressure leakage which was not directed through the diffuser flow path, and
3) Serve as a pressure control barrier for management of the drive train thrust loads.

Reduction of leakage from the main flow path and management of the flow path thrust loads were unfortunately in cross purpose to one another. A large pressure area was required to give adequate range for thrust balance control. However, leakage flow rates increase with a larger seal diameter, and high surface speeds result in more heat imparted to the fluid as it travels through the seal gap. A passive clearance control system allowed for near uniform radial growth of the structure at the rotor/stator interface, and small seal clearances could be achieved. Additionally, for the Build 2 CO_2 compressor test, an active rotor clearance control system was added for more flexibility. Seal clearance can be directly measured in test using proximity probes imbedded in the seal which target the flow path seal land, and clearance can be adjusted. Design review material for the rotor seal is included in Appendix 9.3.2 Rotor Seal.

Another challenge resulting from the high fluid temperatures at the exit of the rotor seal was cooling of the flow path and dry gas seal. This was addressed by a heat shield directly downstream of the rotor seal, as well as a heat shield/baffle plate next to the dry gas seal which helps block hot seal exit fluid from making direct contact with the dry gas seal and flow path wall. Both heat shields were cooled using dry gas seal supply flows, which eventually exit the compressor via the same passages used to capture the rotor seal leakage flow.

Another notable design feature of the static structure was the removable bundle. Instead of removing each piece part separately, the entire bundle can be removed from the pressure case as a single assembly. This was desirable from a maintenance standpoint because it gives easy access to the internals for cleaning, inspection, and replacement in the field if needed.

The thrust bearing and both journal bearings were contained within a bearing housing which was external to the pressure case, see Figure 9.3-1. This allowed for ease of removal and compressor maintenance. Design review material for the layout of the compressor is included in Appendix 9.3.3 Compressor Layout.

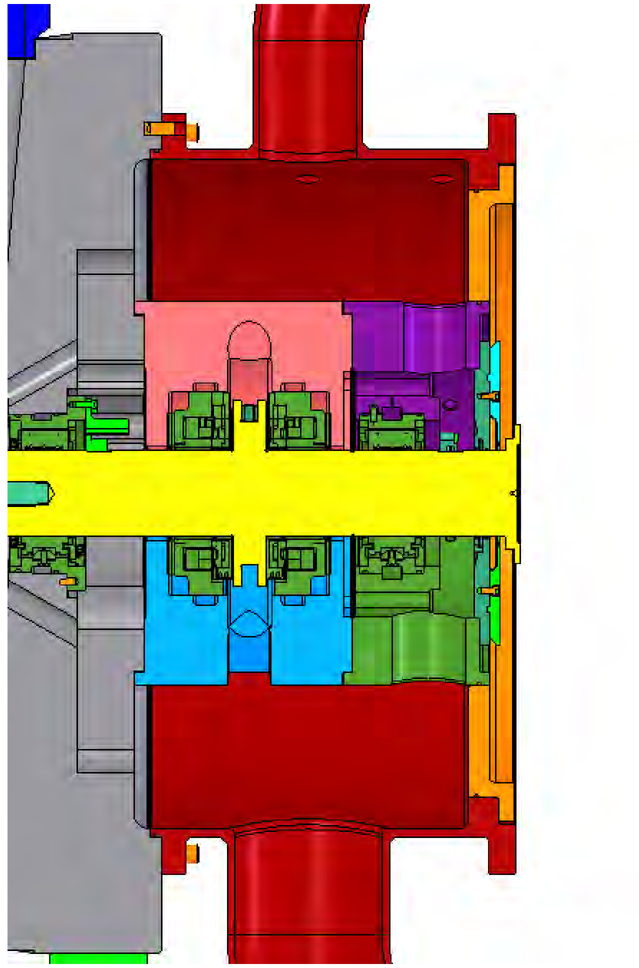


Figure 9.3-1: Bearing Housing

9.4 Facility/Drivetrain

Ramgen's existing 10 mega-watt closed loop CO_2 facility required modifications in order to accept and drive the Build 2 DATUM-S compressor. The existing drivetrain was capable of delivering 10 MW from a Lawrence Scott electric motor at a speed of 3240RPM, up to 3580 RPM. This power was transmitted to a compressor via a speed increasing Allen Gears epicyclic gearbox. The Build 2 compressor's design point operating speed was lower than for Build 1, and so modifications to the drivetrain were required in order to deliver the requisite power at the Build 2 design speed. Dresser-Rand decided a new Allen Gears epicyclic step-up gearbox was the most economical choice for the Build 2 DATUM-S compressor. It had proven, tested performance during the Ramgen test program and would allow the compressor to fit within the footprint of the existing concrete foundation in the Ramgen facility. The alternative option of linking multiple lower gear ratio gearboxes to achieve the desired overall ratio would have necessitated an overhaul of the existing lube oil system to accommodate the extra oil flow necessary to operate a second gearbox. Final Design Review (FDR) material for the drivetrain and overall baseplate layout is included in Appendix 9.4.1 Compressor Drivetrain and Skid FDR.

The new Allen Gears epicyclic gearbox was ordered in May of 2014, and carried a 44 week lead time. The Build 2 compressor test schedule required an interim solution to support a Q1 2015 test program. Dresser-Rand elected to have Allen Gears modify the existing gearbox in order to serve as the bridge between start of test and arrival of the new gearbox. The reworked gearbox was modified to include a single bearing supported high speed output shaft, rather than a splined hub, see Figure 9.4-1. This decision was made in order to increase the rotordynamic stability of the drivetrain to eliminate some risk of delay in collecting aerodynamic performance data due an inability to reach design speed. The increase in overall axial length of the reworked gearbox with the addition of the high speed output shaft more closely matches that of the new gearbox. It will allow for a speedy replacement for the new equipment once it's received on-site.

Figure 9.4-2 - New Allen Gears high speed output shaft

One area that remained unchanged with the reworked gearbox was the overall gear ratio. Thus, the drivetrain was not able to provide the required power at the design point operating speed. In order to allow for the test program to move continue on schedule, D-R elected to have the initial testing done at reduced suction pressure. This enabled the test team to operate in whatever power regime the drivetrain was capable of providing. These tests were still worthwhile as they provided critical aerodynamic performance data, while simultaneously allowing the test team to gather data and troubleshoot systems within the compressor.

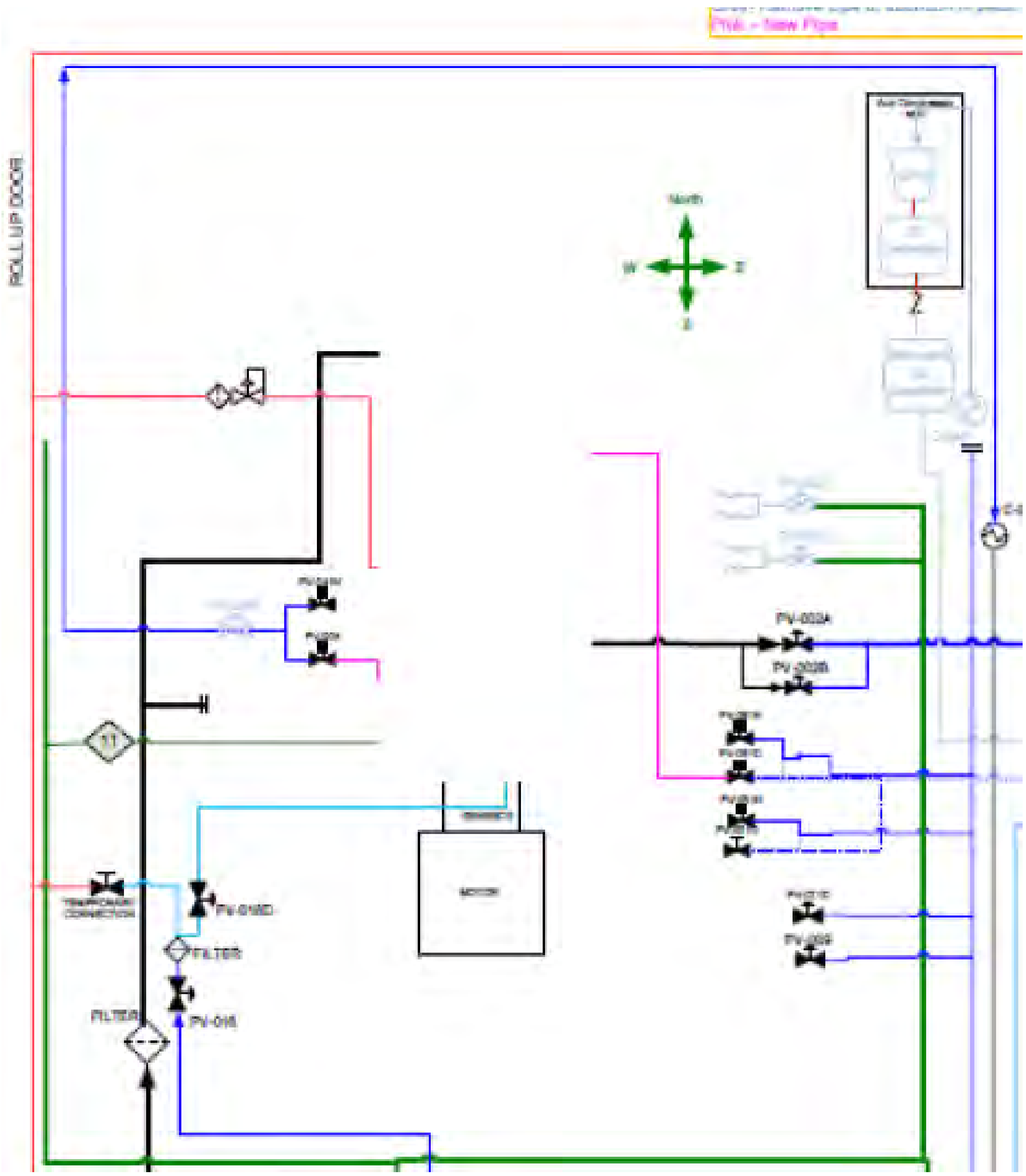


Figure 9.4-3 - Modified Process Flow Diagram

The complexity of the process gas system for the compressor decreased dramatically which allowed Dresser-Rand to simplify a large portion of the existing piping within the facility, see Figures 9.4-2. The figure above shows the updated process flow diagram, detailing new piping installed as well as the dead-headed piping legs. Major piping modifications included the inlet and discharge legs of the compressor. Final pipe fitting began once the pressure case was installed on the baseplate in January 2015. The reduced volume decreased test operating costs as well as limited the potential troubleshooting issues during the preliminary fills of the CO_2 loop. Concerns with contaminating the purity of the CO_2 in the loop were addressed by adding manual vent valves on each end of the newly created dead legs. The test team also procedurally checked the purity of the CO_2 with a gas analyzer while finalizing loop fill procedures.

The control system reduced the overall complexity of operation with fewer process gas systems to monitor during a typical test. Final Design Review (FDR) material for the facility layout is included in Appendix 9.4.2 Facility FDR.

Build 2 CO_2 Demonstrator Testing

The DATUM-S Build 2 CO_2 compressor started testing on March 27, 2015. The testing goals were to validate CFD on the latest DATUM-S configuration in CO_2. The first iteration of the DATUM-S was not designed to achieve optimum performance but it was predicted to achieve the same or greater single-stage pressure ratio as Build 1. After initial validation for CFD performance predictions, follow-on aerodynamic packages of Build 2 will target higher efficiency and operating range.

The test results compared to CFD predictions are plotted on Figures 9.4-3 and 9.4-4. The predicted and measured Pressure Ratio and Normalized Efficiency (ratio of data or CFD prediction vs. maximum efficiency for Build 2) are plotted against the inlet mass flow ratio of measured mass flow to predicted mass flow and maximum efficiency. The inlet mass flow ratio allows for a meaningful comparison of performance between several different configurations i.e. Build 1, Build 2, 2b and 2c.

As can be seen in the data, the Build 2 CO_2 compressor achieved higher pressure ratio than Build 1. Build 2 test data shows lower efficiency, but Build 2 demonstrated more range than Build 1. There is usually a trade-off between range and efficiency in turbo-machinery. Build 2 efficiency did not fall off as quickly over the operating range as did Build 1. Build 1 was not designed to demonstrate range. Future configurations of Build 2 i.e. 2b and 2c. are predicted to increase both pressure ratio and efficiency over a normal operating range, as shown in Figure 9.4-3 and 9.4-4.

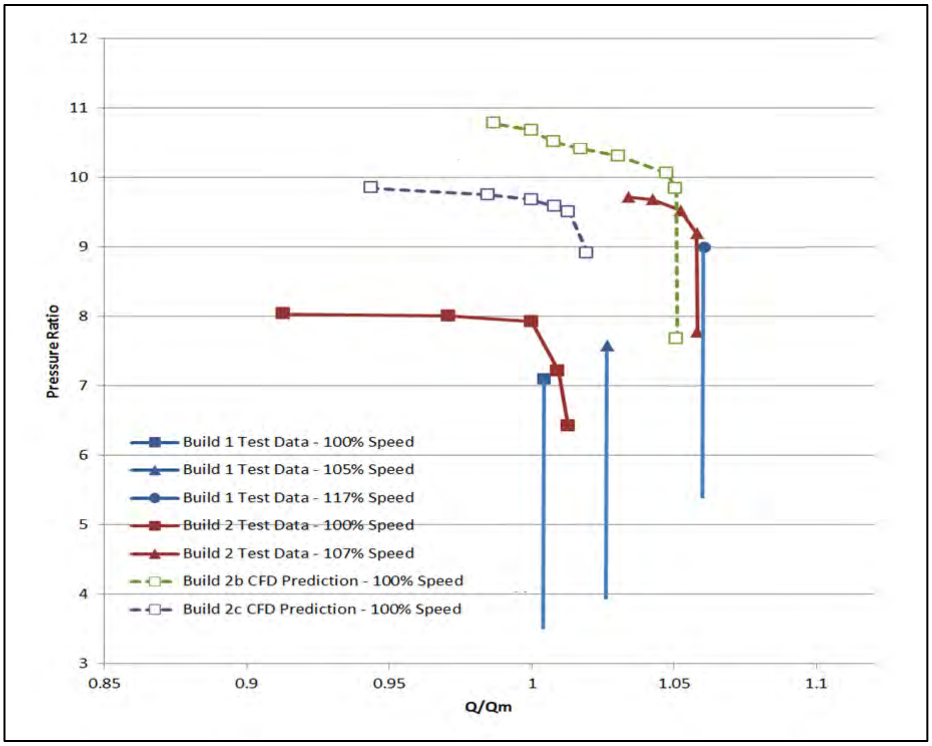


Figure 9.4-3: Comparison of test results to CFD predictions

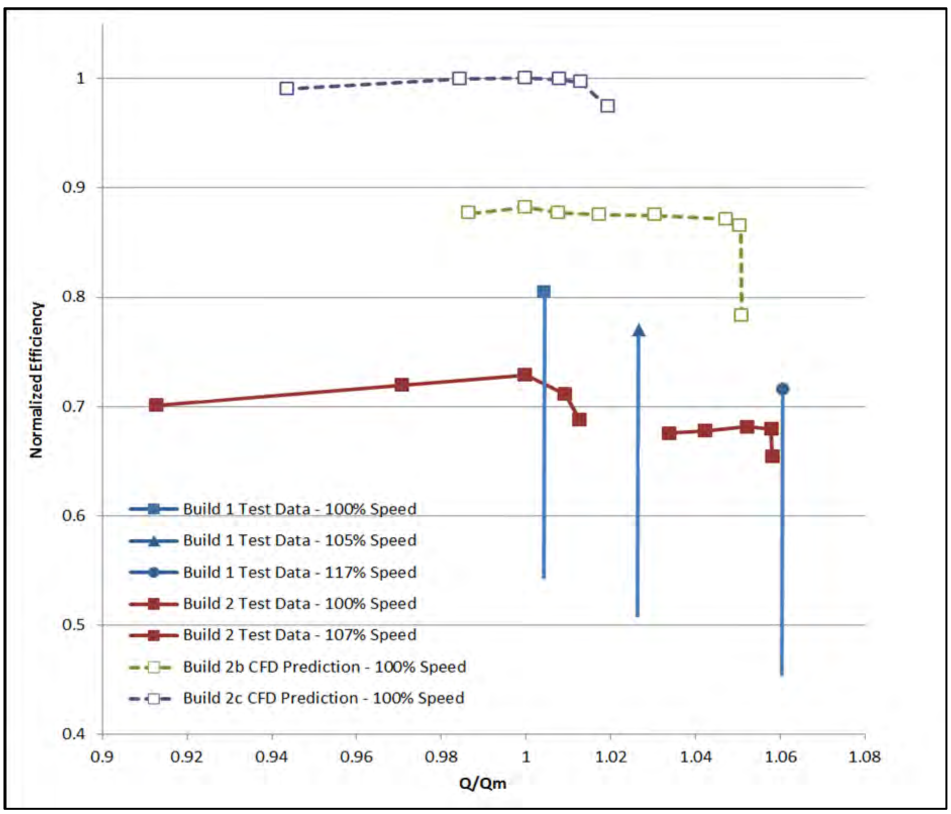


Figure 9.4-4: Comparison of test results to CFD predictions

10. Task 4.1 & Task 4.2 – Preliminary and Final Design and Testing of Integrated Supersonic Component Engine

In the first half of 2011, Ramgen completed a preliminary design review for the first design iteration of the Integrated Supersonic Component Engine (ISCE). The complete ISCE engine consists primarily of the inducer, diffuser, advanced vortex combustor, and multiple turbine stages.

Inducer

Prior to the review, Ramgen identified a large body of work that would need to be successfully completed in order to consider the preliminary design review a success. Ramgen choose to first complete the inducer blade design before advancing to the design of the diffuser. The inducer blade was a more traditional style of turbomachinery, and the larger body of prior work would allow for more rapid closure of the blade design. The design involved analytically solving aerodynamic equations for the flow between the blades which would achieve a total pressure ratio that would meet overall goals. The blades were then modeled in 3D inviscid and viscous CFD to give a better prediction of overall blade performance; see Figure 10.1. The inducer blade design was iterated based on CFD results until acceptable performance was achieved. Full Finite Element Analysis (FEA) analysis was performed on the blade to insure mechanical integrity as the blades will undergo large centrifugal forces.

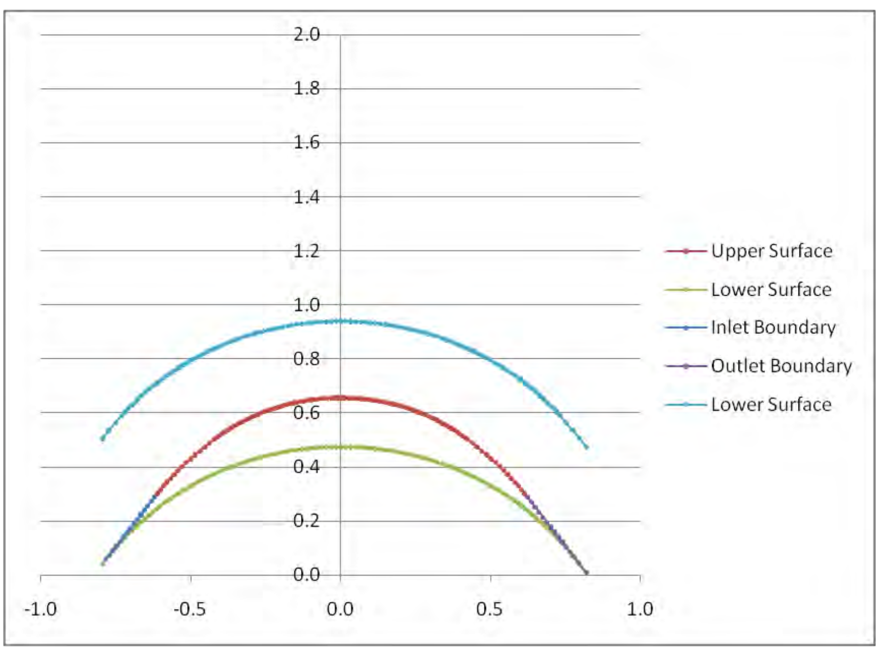


Figure 10.1: Sample Impulse Blade Section

Diffuser

Following satisfactory design closure of the impulse blade, Ramgen began concentrating on the design of the static diffuser which would convert the total pressure exiting the inducer blades to static pressure. A 2D method of characteristics routine was used to design the diffuser ramp which sets up the stable shock structure inside the diffuser. Ramgen initially iterated on the ramp design based on results from 2D viscous CFD simulations, as full 3D simulations are computationally expensive. Once the performance of the 2D simulations reached satisfactory levels, full 3D models were simulated to insure capturing of complex phenomena such as leading edge effects and shock wave boundary layer interactions. Starting simulations were performed in order to confirm the ability to start the diffuser during test. These simulations were performed in 2D, as a 3D starting simulation remains beyond current computational limits. Ramgen determined the 2D results provided sufficient margin such that additional complexities in the true 3D geometry would not inhibit the diffuser's ability to start.

It was determined during the review that the risk involved in the testing of a full engine design could be reduced by retrofitting a Solar Saturn engine to include the inducer blade and diffuser. Ramgen would then test the remaining components separately prior to the full scale test. This risk reduction strategy would still allow Ramgen to sufficiently validate some of the most critical components of the ISCE engine design while dramatically increasing the chance of success.

Work continued on iterating compressor stage designs that could be fit into the existing solar turbine configuration. Ramgen converged on a final inducer blade design that met design goals with acceptable performance levels. Characteristics for the final rotor blade are displayed below:

- Rotor hub radius: 5.500 inches
- Rotor tip radius: 6.770 inches
- Rotor mean radius: 6.135 inches
- Blade Height: 1.27 inches
- Discharge Mach: 1.334
- Inlet total pressure: 14.55 psia
- Inlet total temperature: 60° F
- Mass flow rate: 13.4 lbm/s
- Number of blades: 43
- Rotor RPM: 22,300 design
- Rotor power: 1.5 MW max mechanical

Ramgen's analysis on various components found the total pressure profiles exiting the inducer had a noticeable effect on the performance of the compressor. Specifically, the uniformity of the exit total pressure was determined to be a feature of considerable importance. Previous rotor designs that showed a lack of uniformity in the total pressure profile exiting the inducer had a significant impact on the stability of the boundary layer near the hub. To help circumvent this, the inducer was redesigned using similar methodology as was reported previously in an attempt to achieve total pressure uniformity. Details of the design are summarized in Appendix 10.1

The engine feeds the supersonic diffuser whose primary purpose was to convert total pressure to static pressure. The shock structure that develops within the compressor was influenced by the properties of the incoming flow, and most importantly the Mach number as the oblique shock angle was a function of only the turning generated by the ramp and the incoming Mach number. In order to achieve a uniform outflow total pressure profile, the Mach number near the hub of the diffuser was required to be larger than the Mach number at the shroud. This change in the inducer design required the diffuser ramp to be modified. Ramgen increased the complexity of its viscous 3D CFD analysis in order to try and capture the important interaction between the inducer and static diffuser. Analyzing a rotating and stationary component within the same computation introduces additional modeling complexities and necessitates an increase in model grid resolution. Ramgen iterated on diffuser ramp models, coupling its interaction with the inducer until the inducer/diffuser stage reached acceptable performance targets.

In parallel Ramgen was performing complete structural and thermal finite element analysis to ensure mechanical integrity of the aerodynamic designs. The combination of high rotor rotational speed and high discharge pressure results in the potential for aerodynamic cross-coupled forces and flow instabilities. In addition, lateral rotordynamic stability is of critical importance in high-speed turbocompressors to avoid issues with tight-running seals and oil heating. To ensure trouble-free operation in test, Ramgen designed to meet API standards for vibration magnitude. Successful results from these analyses were summarized in the final design review. Details of the conceptual design review can be found in Appendix 10.2

Redmond Facility

A multi-bay facility was leased in Redmond, Washington where the testing of the ISC Engine and its components took place. The facility required a multitude of upgrades before it will be fully prepared for testing. The master layout of the facility bays can be found in Figure 10.2. The facility layout was designed to support up to three separate test programs simultaneously - the advanced vortex combustor (AVC), the ISC Solar engine retrofit, and an LP CO_2 test. The LP CO_2 test was not completed before the end of the program.

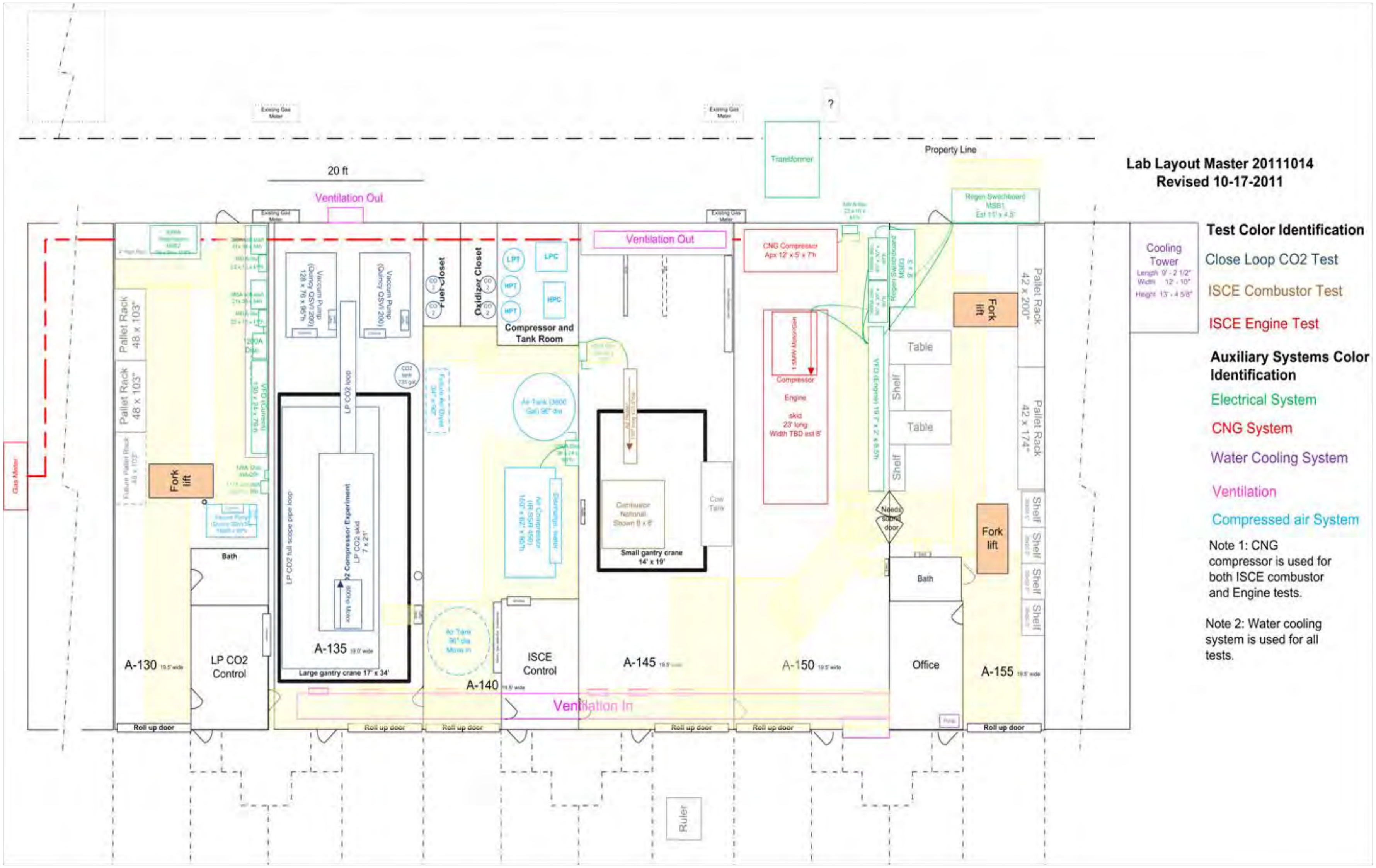


Figure 10.2: Full Redmond Facility Layout

Performance Instrumentation and Measurement

In consultation with D-R, Ramgen thoroughly reviewed existing industry standards (ASME, PTC, etc.) to ensure compliance with best measurement practices. A Data Analysis Plan (DAP) was developed which documented the specific measurements and approaches needed to measure the unique characteristics of both the supersonic compressor and engine to ensure that their performance was quantified in an industry-approved manner.

DOE Program Review and Report

In 2011 Ramgen met and presented current material with Tim Fouts and members of the DOE. The meeting purpose was to discuss Ramgen technology and progress at that time satisfies the requirements of this review. Presentation materials can be found in Appendix 10.3.

After completing the preliminary design phase of the ISC Engine program, work on aero-scaling algorithms continued in parallel with the design and testing of Ramgen's multiple programs. Ramgen continually investigated the potential of scaling the compression technology upward to increase variability in possible future product lines as well as increased marketability in various high compression applications.

Ramgen successfully completed the preliminary and final design phases of the ISC Engine program and held design reviews to identify any unexpected aerodynamic and mechanical issues leading up to the final design and start of procurement for the program. 3D viscous and inviscid CFD results were reviewed, along with relevant structural and mechanical work. Details of the design review are available in Appendices 10.4 and 10.5.

11. Task 4.3 – ISC Engine Subcomponent Test

With the decision to apply the supersonic compressor to the Saturn engine for retrofit the rotating combustor system definition as well as composite ring proof-of-concept definition, proposed in the initial design concept, were no longer applicable. This significantly reduced the risk of the overall design, and the cost and schedule risk associated with additional exploratory activities.

The subcomponents that were meaningful to study included:

- Non-rotating Combustor Test
- High Expansion Ratio Nozzle Test
- Turboexpander Test
- Turboexpander Module Checkout Test
- Primary Turboexpander Test

The following sections explain Ramgen's efforts to design, build and test these components as critical learning opportunities for the successful integration of a supersonic engine.

Non-Rotating Combustor Test

The ISCE subcomponent combustor design was the next generation of Ramgen's Advanced Vortex Combustor (AVC) in a fully annular configuration. The combustor was a dry lean premixed type that puts bulk fluid swirl into the flow to allow for less turning in the turbine first stage nozzle, and thus more efficient engine operation

In support of the ISCE design, work on the Annular Advanced Vortex Combustor (AVC) commenced in 2012. The AVC hardware was designed as a full scale combustor test of hardware representative of the ISCE Build 2 combustor, and was intended to provide efficient combustion with a highly-swirled exit flow, allowing for more efficient processing by the turbine. Ramgen sought to deliver this highly swirled flow with efficient combustion using trapped vortex cavities as the combustor pilot zone. This concept was a modification to previously designed trapped vortex combustor (see Figure 11.3) and utilizes many scaling and design lessons learned from that test program.

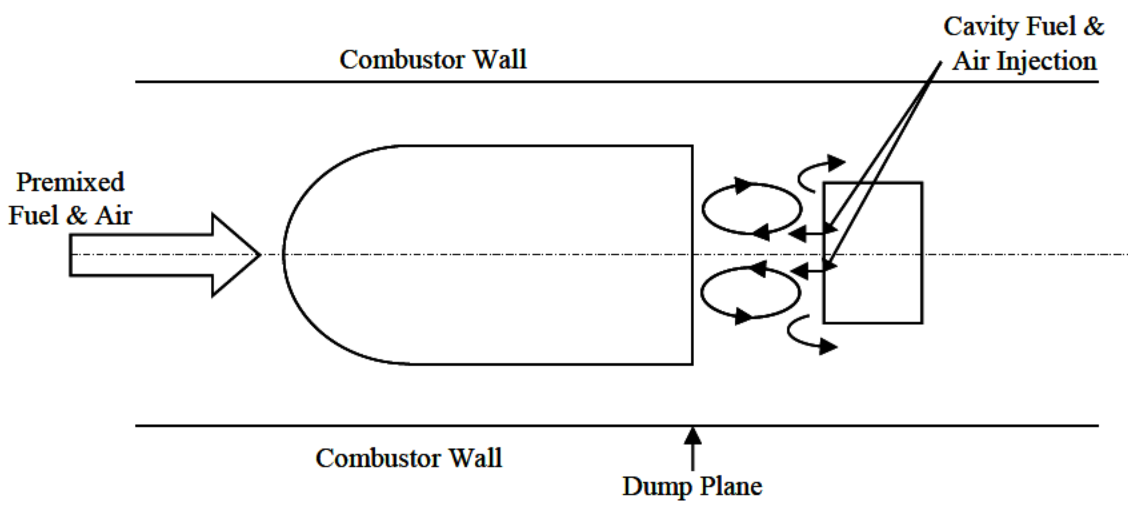


Figure 11.3: Schematic of the trapped vortex concept. From Edmonds, Steele, Williams, et al, "Ultra-low NO_X advanced vortex combustor." Proc. ASME Turbo Expo, 2006.

The hardware was intended to operate holding corrected massflow within the facility capabilities of the Ramgen combustor test cell. The designed geometry was run at a maximum pressure of 4 atm. Higher pressures were not attainable as they would require more airflow than the facility could deliver. The combustor test shall operate in both a pressure fed cavity condition and a plenum fed cavity condition. Specifically, pressure fed cavity operation means that there will be independent control of the cavity air flowrate. Plenum fed cavity operation means that the effective area of the injector holes in the combustor centerbody shall set the air flowrate to the cavity with the air being drawn in from the same plenum as the main air and cooling air.

The AVC program was intended to represent the ISCE Build 2 combustor, design, and the operating envelope. The AVC test conditions were based on the ISCE engine operating conditions of 290 psia inlet total pressure and 855 °F total temperature, and were run such that the corrected inlet massflow of the combustor test rig matches engine compressor exit flow. Ten percent (10%) of the compressor massflow was intended for turbine cooling. At this time, facility limitations impose a maximum inlet air massflow of 2.1 pounds per second with 58.8 psia and 650 °F total conditions.

The combustor initial geometry was based on known scaling rules, residence time goals and target velocities for the combustor based on prior testing. Once the baseline geometry was developed, CFD of the flow field was conducted using both non-reacting and reacting simulations. The combustor inlet centerbody that creates both bulk swirl and contains the trapped vortex pilot cavities was further optimized. One challenge identified early on was the need to get cooling air, cavity air, and cavity fuel into the centerbody and distributed. Correctly sizing the flow passages was key to achieve both stable vortex combustion and adequate cooling of the hardware. The conclusion reached was that the cavities should be split into a unique inner and outer cavity with one vortex pilot flame in each cavity, in order to provide services to the cavities.

An important design consideration for this program was how the bulk swirl would be added to the flow. Initially, it was thought that the compressor discharge flow would be left swirling upstream of the combustor, however upon further analyzing this it was found that the combustor cooling on the liner inner diameter (ID) would be starved of cooling air due to insufficient pressure drop to drive the cooling flow through the liner. This phenomenon was the result of conservation of angular momentum and the fact that the liner inner diameter was less than the combustor inflow annulus leading to a decrease in static pressure on the liner ID.

Arriving at the optimal combustor exit swirl involved consideration of multiple important factors. The concept of bulk swirl in combusting flows has been studied extensively in the past for afterburner or augmenters. What was found was the turbulent flame speeds are enhanced in swirling flows that induce a large amount of centrifugal g-loading on the flow, to a point. A sample of data obtained from this testing is shown in Figure 11.3, with this in mind the inlet radii and flow angle where carefully chosen for the combustor design in order to prevent extinction of the flame due to very high g-loadings. The final combustor geometry was set considering both the conservation of angular momentum from inlet to outlet for all flows and the combustor inlet g-loading was evaluated with the goal to keep it less than 3500 g's.

The combustor has been designed with optical access to visualize the bulk fluid swirl and confirm correct operation. The test article was instrumented for both temperature and pressure measurements to monitor combustor health, as well as with several combustor dynamic pressure transducers to monitor combustor acoustics. The combustor test rig has also been designed with four locations for exhaust plane measurements, which include exhaust gas emissions and temperature. The four mounting locations utilize a universal mounting interface so that any exhaust probe can be utilized in any location.

The final design of the combustor liner evaluated multiple shapes in an effort to optimize the liner cross-section for both cooling flow and desired combustor residence time. The final liner design was an impingement effusion type that additionally incorporates optical access through part of the outer liner to visualize the flow field bulk swirl. The combustor was instrumented with several thermocouples to monitor the health of the combustor hardware. The exhaust was monitored with an emissions rake and thermocouple rake.

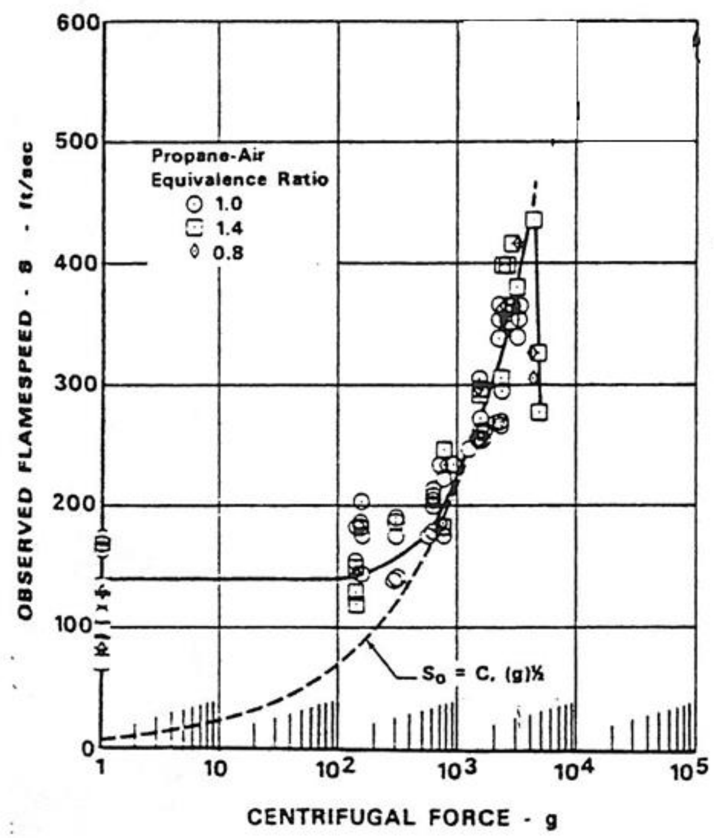


Figure 11.3: Flow field centrifugal effects on turbulent flame speed. Lewis, G. D., "Centrifugal-force effects on combustion." Proc. 14th Symposium (International) on Combustion, 1973, pp. 413-419.

Combustor Design Milestones

Significant design work was accomplished throughout 2012 by Ramgen employees as well as contractors at QuEST Global, including CFD and structural analysis of the primary combustor flow. The following design reviews were completed in 2012:

- Conceptual Design Reviews:
    - Pressure Vessel, May 2012
    - Fuel and Air Facility Delivery Systems, May 2012
    - Combustor Test Article, June 2012

- Preliminary Design Reviews:
  - Pressure Vessel, August 2012
  - Combustor Test Article Aerodynamic Review, August 2012
  - Combustor Test Article Aerodynamic Review, September 2012
- Final Design Reviews:
  - Pressure Vessel, November 2012
  - Combustor Test Article, December 2012 (Appendix 10.6)
- Combustor Test Article Long Lead Drawing Completion, May 2013
- Facility Air and Fuel Systems, January 2013
- Facility Air System, March 2013
- Facility Fuel System, March 2013
- Exhaust Water Cooling System, September 2013 (see Appendix 10.7)
- Facility Instrumentation, July 2013
- Combustor Test Hazop Complete, October 2013

In December 2012, the final design review was held on the ISCE combustor sub component. This review identified action items that had to be closed prior to release of hardware drawings for fabrication. During the first quarter of 2013, work to close these action items was completed and drawing creation was started. Additionally, during drawing creation, work began on finalizing the manufacturing plan for all combustor components. The modular nature of this design necessitated extra effort on developing the manufacturing plan for the hardware. Combustor component drawings were completed around the end of May 2013. Several dry fit checks were performed on the hardware after manufacture in an effort to identify and resolve assembly issues as early as possible.

<u>Combustion Facility Design Work</u>

The combustor test was conducted in an optically accessible pressure vessel. Safety considerations and local regulations required that this pressure vessel be an ASME stamped vessel. The pressure vessel initial design was done by Ramgen engineers with input from scientists at the Air Force Research Lab (AFRL) of Dayton, Ohio. Ramgen has previously done combustor testing at the AFRL High Pressure Combustion Research Facility (HPCRF) in Dayton and visited this facility again as part of this program to plan for the design of Ramgen's combustor test facility. The combustor pressure vessel was installed with the main axis horizontal (see Figure 11.4). The inlet plenum sits on rollers and can be pulled away from the instrumentation case for access to the combustor. Main inlet air enters through the inlet plenum and travels through a special inlet to create virtually quiescent inlet flow field at the combustor inflow plane. The combustor was cantilevered from the exhaust with all services and instrumentation entering through the instrumentation case.

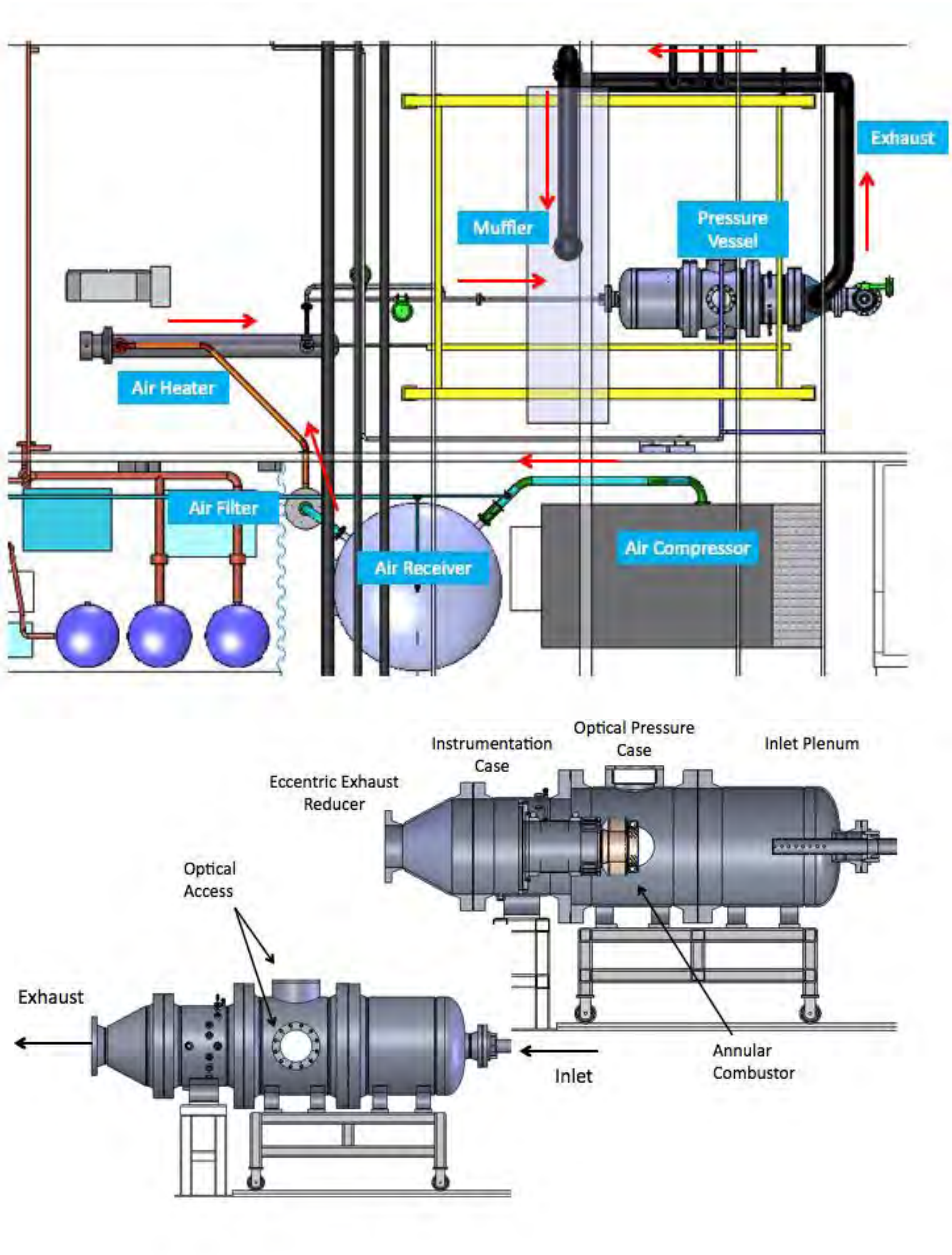


Figure 11.4: AVC facility layout.

Throughout the design process trips were made to Ohio to review the facility and combustor design with scientists in AFRL's combustion branch. Additionally, these trips have allowed for review and working meetings on the combustor design as much of the CFD and thermal design work has be performed by QuEST Global in Cincinnati, OH.

Combustion Test Facility Construction

For the current and future combustion tests, the facility requirement for air delivery was set at 2 lbm/sec, 200 PSIG, and 650 °F. Delivered air could be water saturated, but liquid water and any particulate matter was removed. Consistent delivered temperature control was critical and so the air was routed through an electric heater. A schematic of the overall compressed air system can be found in Figure 11.4.

Air Compressor/Receiver Tank

An Ingersoll-Rand HXPE450-2S air compressor was installed adjacent to the combustor test cell (Figure 11.5). This water-cooled 450 horsepower 2-stage screw compressor delivers 1739 SCFM (2.2 lbm/sec) at 200 psig, aftercooled to 90 °F. Inlet air was filtered down to 3 microns. Ducting was installed through the roof for both air intake and auxiliary cooling air exhaust.

Figure 11.5: Photograph of I-R air compressor

A large receiver tank, shown in Figure 11.6, was required to both minimize pressure fluctuations propagating downstream from the compressor and to provide reserve capacity in case of a power outage or compressor failure during test. Following Ingersoll-Rand guidance, a 3,800 gallon steel receiver tank was installed adjacent to the air compressor. An automatic condensate drain was installed to the tank bottom, which uses tank pressure to pump condensate to the facility water purification station. After removal of any compressor oil from the condensate, clean water was discharged.

Air Heater

To raise air temperature from the 90 °F compressor discharge to desired combustor inlet 650 °F, a 360 kW electric heater was installed between the receiver tank and the combustor test rig (Figure 11.7). Using redundant thermocouples to control output temperature, this heater provided continuous temperature adjustability and automatic response to changes in mass flow.

Figure 11.6: Photograph of air compressor receiver tank.

The pressure vessel was ASME code stamped and capable of 300 psi and 1000 °F operation with optical access. The vessel was delivered to Ramgen's Redmond, WA test facility in June 2013. Figure 11.7 shows the vessel prior to installation of the facility piping systems. Figure 11.8 shows the vessel after installation of some facility piping.

Figure 11.7: Combustor Test Pressure Installation

The combustor test facility has been designed with two independent air legs, main air and cavity air. The system was designed such that both air legs could be used or just the main air leg. Upstream of both control valves a pressure regulator was installed to set the system header pressure and remove any pressure fluctuations that might be induced by the air compressor. The main air utilized a proportional globe valve for control and a critical flow venturi for massflow metering. The cavity air utilized a globe valve for control and a venturi for massflow measurement.

The combustor test was conducted utilizing natural gas from the local utility in Redmond, WA. Gas was supplied to the facility at 10 psig and compressed up to 215 psig by a natural gas compressor that supplies fuel to two bays into the facility. Once the gas entered the combustor test bay it was divided into two legs, one leg was for fueling the combustor cavity region, the other for fueling the combustor main inlet flow. Both fuel legs measured massflow and were controlled by independent proportional globe valves.

Cooling water was utilized for exhaust cooling to protect the high temperature V-ball back pressure valve and exhaust piping. The back pressure valve was capable of withstanding 800 °F so cooling water was sprayed into the exhaust just downstream of the combustor exhaust annulus. Additionally, the closed loop water cooling was provided to the hardware at the combustor exhaust. The closed loop water supply served to cool the hardware and prevent boiling in the water supply before the water was injected into the exhaust stream.

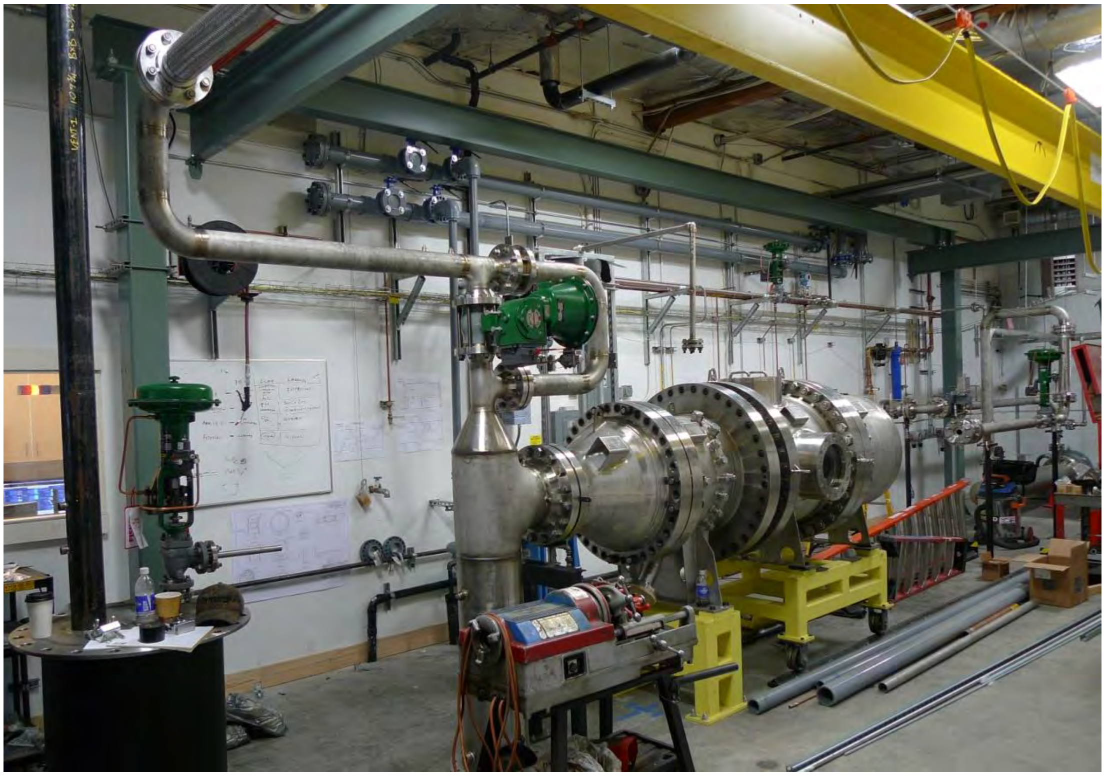


Figure 11.8: Combustor Test Bay nearing completion at Ramgen's Redmond, WA Test Facility.

11.1 Task 4.3.2 Combustor Test

In early 2014 combustor hardware was nearing the end of fabrication. During the last week of February, Ramgen personnel travelled to the machine shop that fabricated the test hardware and completed the hardware assembly of the centerbody and inner liner. Once this assembly was complete it was shipped to Ramgen's Redmond Lab facility for integration into the combustor test facility, see Figure 11.9

Prior to the arrival of the combustor test article, work was underway to complete the facility and install all of the necessary services to operate the combustor including:

- Two natural gas fuel legs
- Two heated air legs
- Closed loop combustor hardware water cooling
- Open loop combustor exhaust water cooling
- Control system programming

Once the test article arrived final integration began and checkout tests were conducted included fabrication and leak checking manifolds, instrumentation hookup and checkout, and control system verification, see Figures 10.10 and 10.11.

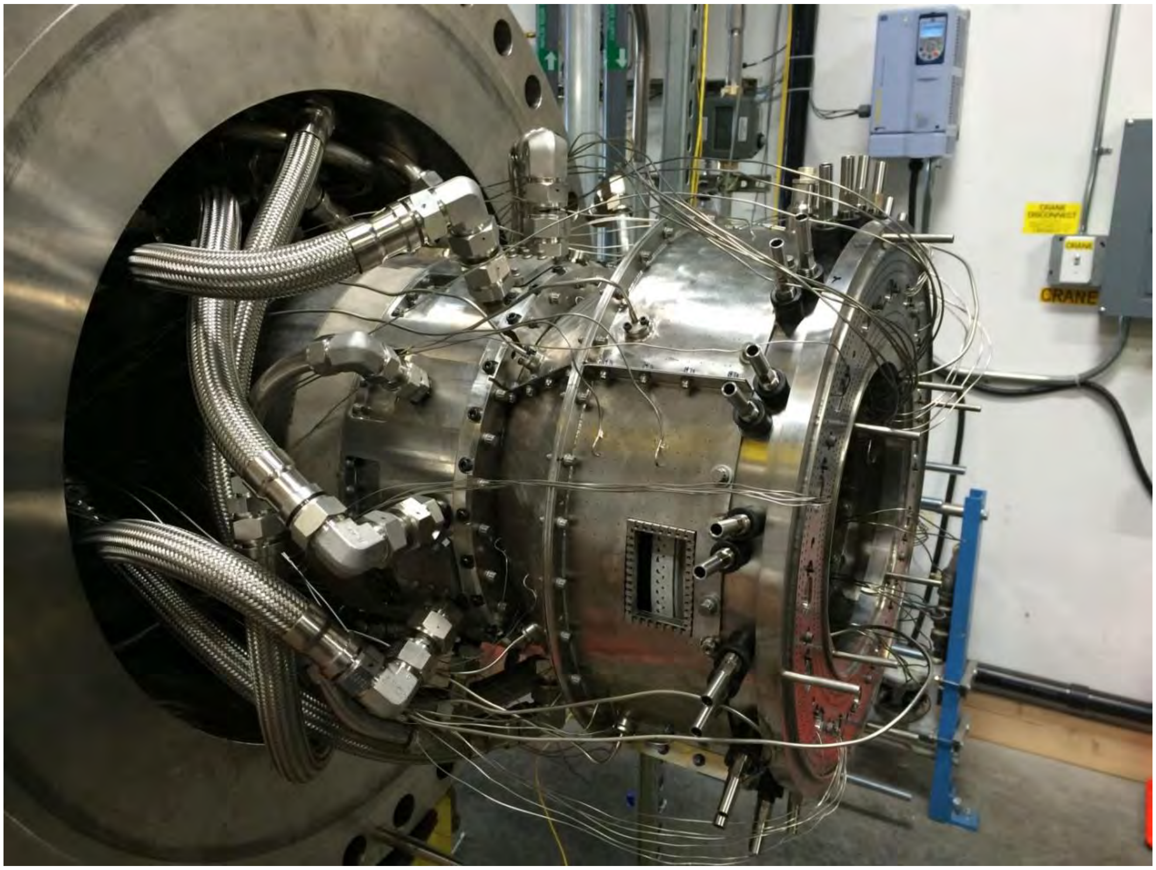


Figure 11.9: Combustor Assembly prior to hookup of fuel, air, water, and instrumentation in Redmond Lab.

Combustor test article health monitoring instrumentation included:

- 69 temperature measurements monitoring metal temperatures, fuel and air temperatures within the test article
- 14 dynamic pressure measurements use to monitor for potential combustor acoustic phenomena.
- 8 static pressure measurements monitoring combustor pressure drops.

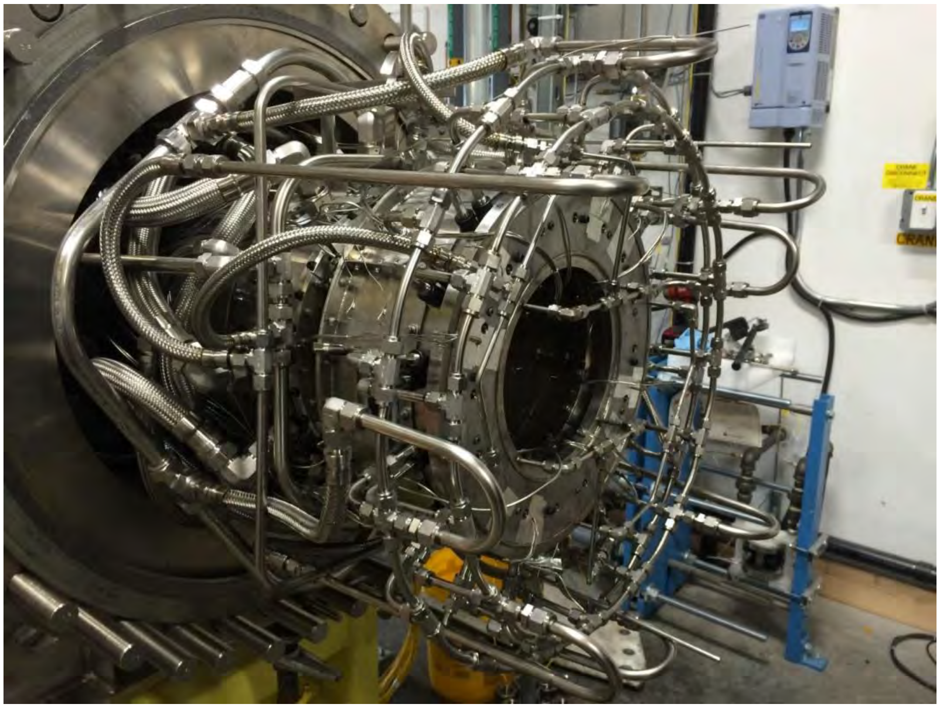


Figure 11.10: Combustor Test Article with all services hookup up and ready for liner air cooling flowchecks.

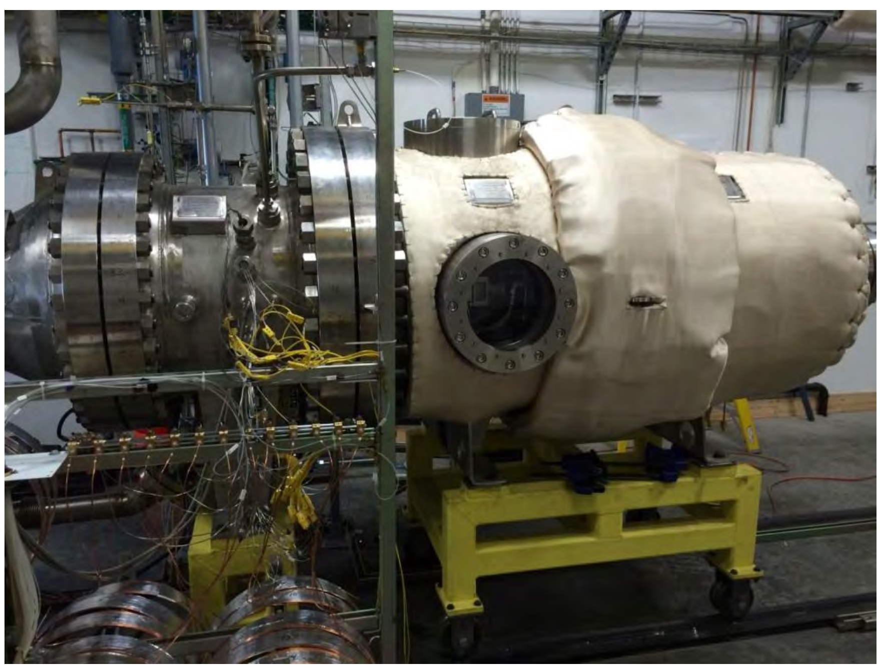


Figure 11.11: Combustor Pressure Vessel closed and ready for test.

Initial testing focused on verification of the PLC control software to insure safe facility operation. Testing focused on verification of fuel and air control valve operation, and combustor exhaust water cooling which was critical to prevent overheating of exhaust piping and the facility backpressure valve.

Once these checkout tests were complete the combustor was prepared for airflow checks. The purpose of these checks was to confirm that the hardware was flowing adequate amounts of cooling air. The experimental results were compared with the design intent to make sure that the combustor hardware would not overheat in operation. These tests were conducted by covering over the main flow passage to isolate the airflow to just the liner and allow for direct measurement of the combustor liner cooling flow. Hardware was also built to allow for determination of the centerbody cooling flow rate. After multiple tests and some troubleshooting these tests revealed that the combustor was flowing about 30% more cooling air than design intent. After performing necessary pressure vessel and hardware leak checks it was decided to proceed with testing to begin to understand the combustor operational characteristics.

Initial fueled combustor testing focused on ignition of the cavity region only. First ignition of the cavity was achieved at the end of April 2014. The combustor was designed to first light the cavity flame and then add main fuel to ignite the main flame. Figure 11.12 shows combustor ignition; the plot shows the rise in the cavity wall temperatures during an ignition event (y-axis: temperature, x-axis: time). Once repeatable ignition was demonstrated, the cavity operation limits were explored prior to lighting the main flame.

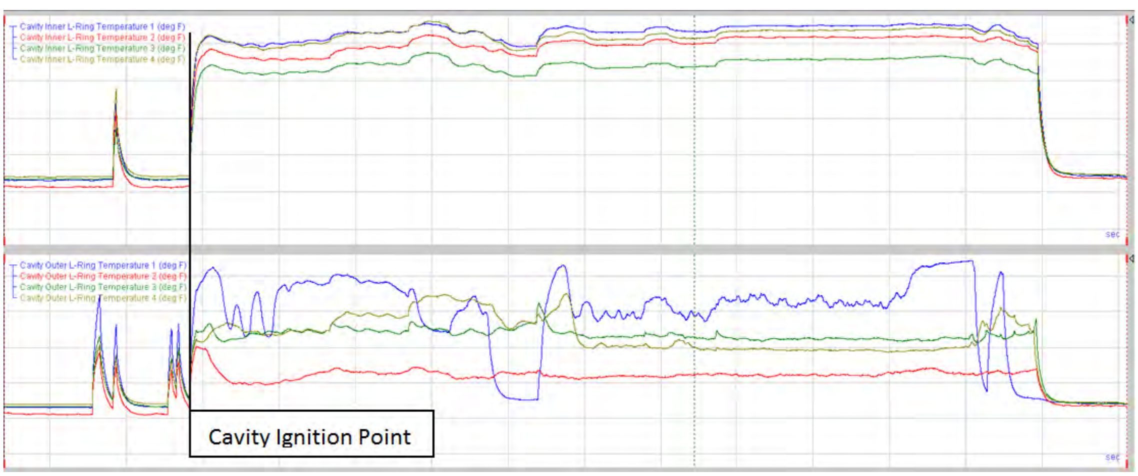


Figure 11.12: Cavity Ignition Data, 5/2/1014

In early May 2014, the main flame was lit and the combustor mapping was started. Combustor testing focused on understanding the operating limits of the combustor between 2 – 4 atmospheres. Temperature data from the inner liner are show in Figure 11.13 (y-axis: temperature, x-axis: time), this plot clearly shows the temperature rise caused by the cavity only ignition followed by the main flame ignition as main fuel was introduced to the combustor.

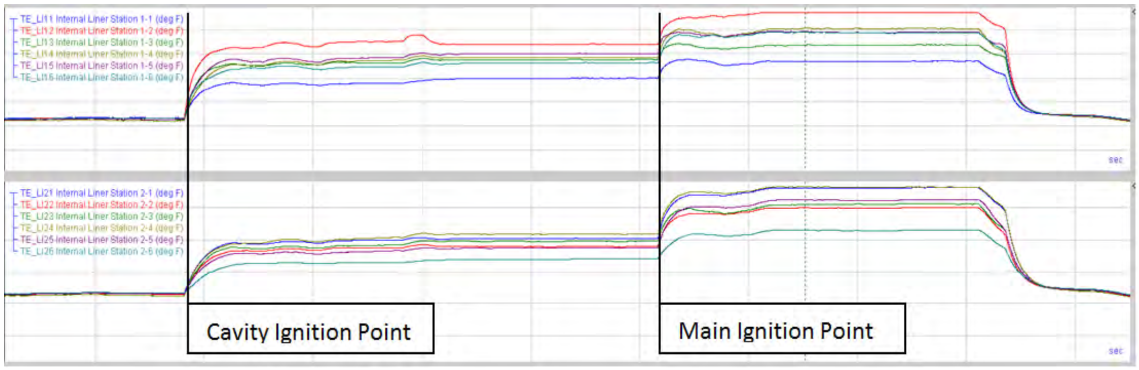


Figure 11.13: Main Ignition Data, 5/7/2014

The facility emissions equipment was being commissioned during June 2014 and initial emissions data was collected for NOX, CO, CO2, and O2. An Un-burnt Hydro Carbon (UHC) analyzer has also been installed in the facility, but was not available for use in June due to a delayed delivery by the analyzer vendor. Figure 11.14 shows a time trace (x-axis: time) of early data that were taken during commissioning of the gas analyzers. The top plot shows uncorrected (data not at 15% O2) gas analyzer output. The CO analyzer was not providing acceptable values due to an unacceptably cool combustor wall during this particular run. The middle plot shows the exhaust probe temperatures, and the bottom plot show exhaust flow angle. Once the initial system pressure variations settle the exhaust flow angle was well matched with our design target. The exhaust flow angle value was typically ±1°, this was because the flow angle probe was installed mid passage at our design exhaust flow angle of 55°.

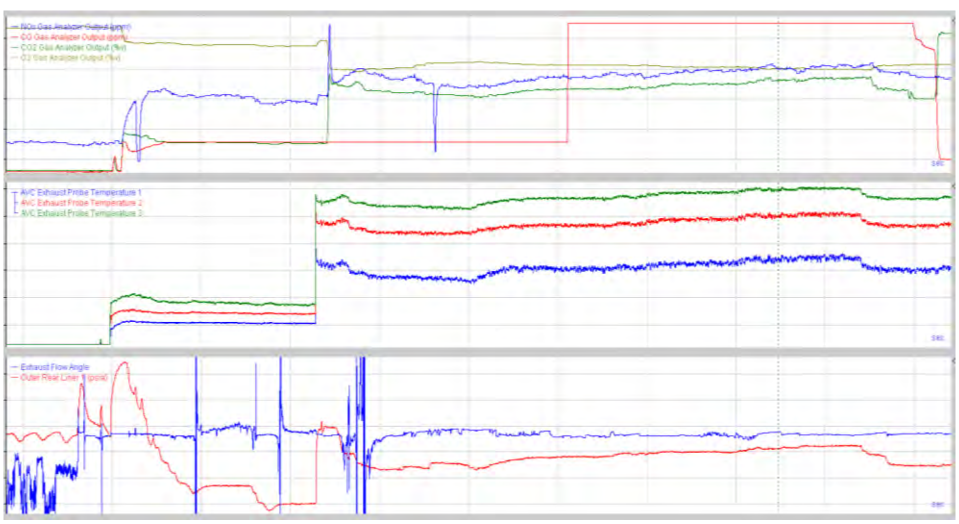


Figure 11.14: Uncorrected Emissions Data, Exhaust Temperatures, and Flow Angle

During this initial combustor testing repeatable ignition was demonstrated. The combustor also has a wide operating range as expected from previous AVC designs tested by Ramgen. The NO_X levels achieved were comparable to other state of the art

DLN combustors. However, the CO levels were unacceptably high due to excessive cooling air.

Continued testing was conducted in July 2014 by Dresser-Rand masking off some cooling holes to try and improve CO oxidation. After multiple tests to try and address the excess cooling flow rate, it was decided that the next appropriate step would be to dis-assemble the hardware and try to address the source of the excess cooling airflow.

Combustor Milestones

- Test Article Assembly Complete: February 2014
- Facility Integration Complete: March 2014
- Cooling Airflow Check Complete: April 2014
- First Ignition: April 29th , 2014
- Main/Full Load First Run: May 7th,2014

High Expansion Ratio Nozzle Test

The design and analysis of a partial admission nozzle test rig was executed to test the performance efficiency of a high expansion ratio, supersonic nozzle and to anchor CFD and thermal analysis modeling techniques. The aerodynamic and thermal design of the nozzle was modeled at full scale based on requirements for operation in the Ramgen Integrated Supersonic Compression Engine (ISCE). The mechanical design was limited to a 20% annular sector to match the heater and gas flow limitations present in the Ramgen test lab in Redmond, Washington. The 20% annular sector consisted of one full flow passage, and two partial flow passages – one on either side.

The nozzle design incorporated converging and diverging ramps on the hub and the shroud to achieve the target 10:1 supersonic area expansion ratio. The full annular nozzle was designed to have 10 vanes dividing the nozzle flow passages. The 20% sector of the test nozzle included 2 vanes which were each unique due to internal instrumentation features. To achieve the target operating temperatures, the ramp sections were designed to utilize a combination of back-side impingement cooling and surface film cooling. The vane sections incorporated backside impingement cooling on the leading edge, and internal pin-fin cooling features along its length. The nozzle was heavily instrumented to obtain gas temperature, metal temperature, and static pressure at multiple locations. Immediately downstream of the nozzle was a calibrated probe to measure gas temperature, total pressure, and flow direction. The probe was designed with the ability to perform sweeps of the flow passage in the radial and circumferential directions.

The gas supply system consisted of an industrial air compressor that could generate a mass flow rate of 2.2 lbm per second, matched with a 360 kW electric heater able to raise the gas temperature to 650°F; a flow uniforming section to break up the developed boundary layer; and inlet guide vanes to provide uniform flow into the nozzle. The gas exhaust system utilized vacuum compressors to lower the outlet pressure sufficient to achieve the desired flow rates and pressure ratio within the limits of the gas supply system, and a heat exchanger to reduce the gas temperature to within the allowable limits of the vacuum compressors.

The nozzle design for the target engine application was developed and optimized via CFD and thermal analysis. The aerodynamic and thermal design was based on a full annular nozzle for the target engine application. The engine nozzle was designed to accept the subsonic, highly-swirled exit flow directly from the AVC combustor and discharge the air supersonically directly into the turbine blades. A trade study of multiple nozzle configurations was performed using CFD analysis to determine the most aerodynamically efficient design on the basis of total pressure recovery. The number of strakes and the expansion configuration were evaluated.

Significant effort was expended to optimize the strake trailing edge for aerodynamic, thermal, mechanical, and manufacturing considerations. Aerodynamic efficiency improved as the trailing edge width was decreased. However, the need to pass cooling air through the strake trailing edge to maintain acceptable metal temperatures limited the minimum width of the strake trailing edge. Structural requirements, application requirements for thermal barrier coatings, and manufacturing tolerances also influenced the minimum width of the trailing edge. Several trailing edge configurations were evaluated to find the best balance of the multiple factors. Configuration variables included placement of bore cooling holes, whether to include film cooling holes, trailing edge width, cooling hole shape, wedge angle at the trailing edge, material choices and corresponding maximum allowed metal temperature, and thermal barrier coating placement and thickness. The result of these trade studies was that there was a benefit to utilize the higher temperature capability of castable, single crystal alloys for the strake material. Higher metal temperature allowances required less cooling air, and less cooling air required smaller air passages to eject the cooling air back into the primary flow, directly influencing the width of the trailing edge. It's worth noting that due to schedule and resource constraints, the potential benefits of film cooling along the length of the strake were not thoroughly evaluated and this is an area that may be revisited during subsequent design activities.

The test nozzle was designed to represent one full flow passage of the conceptual engine configuration, with a partial passage on either side as shown in Figure 11.15. This configuration allowed for a full scale nozzle test article that could operate within the 2.2 lbm per second air flow capacity and 360 kW heater capacity of Ramgen's Redmond test facility.

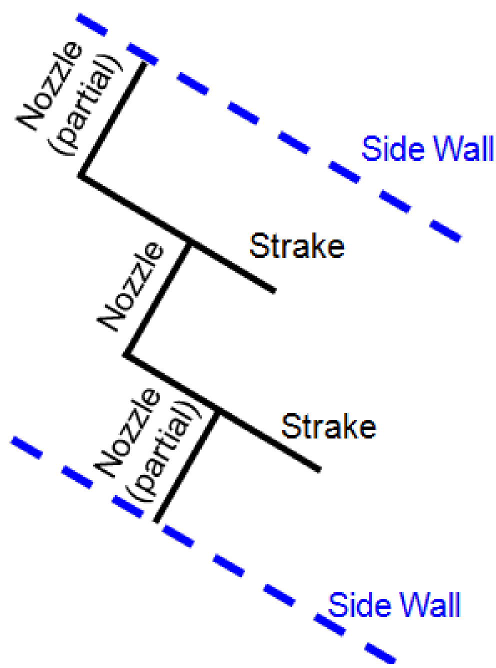


Figure 11.15 - Nozzle Test Configuration

The facility heater was rated to output 700°F so the nozzle test was designed for a nominal operating temperature of 650°F assuming some temperature loss in the inlet piping. Reynold's number similarity considerations were used to determine the target nozzle operating conditions. Partial or full similarity within the temperature and mass flow rate limitations of the test facility required sub-atmospheric nozzle exhaust pressures which would be achieved with the use of the test facility's vacuum system. A heat exchanger was implemented downstream of the nozzle to cool the air at the inlet to the vacuum pumps.

The test rig specific hardware included a flow uniformization section to transition from 4" diameter supply pipe, break up the developed boundary layer, and provide uniform flow. A contraction section was required to blend the flow from circular tubing to the partial annulus cross-section of the test nozzle. The nozzle test section included an inlet guide vane section to turn the flow and introduce swirl before encountering the nozzle. The test nozzle incorporated the cooling scheme anticipated in the engine configuration of the nozzle and consisted of 3 separate cooling channels to meter cooling air individually to the hub, shroud, and strake. The nozzle was heavily instrumented to obtain gas temperature, metal temperature, and static pressure at multiple locations. Immediately downstream of the nozzle was a calibrated probe to measure gas temperature, total pressure, and flow direction. The probe was configured with two actuators to provided automated sweep capability in the radial and circumferential directions. See Figure 11.16 for a block diagram of the test configuration.

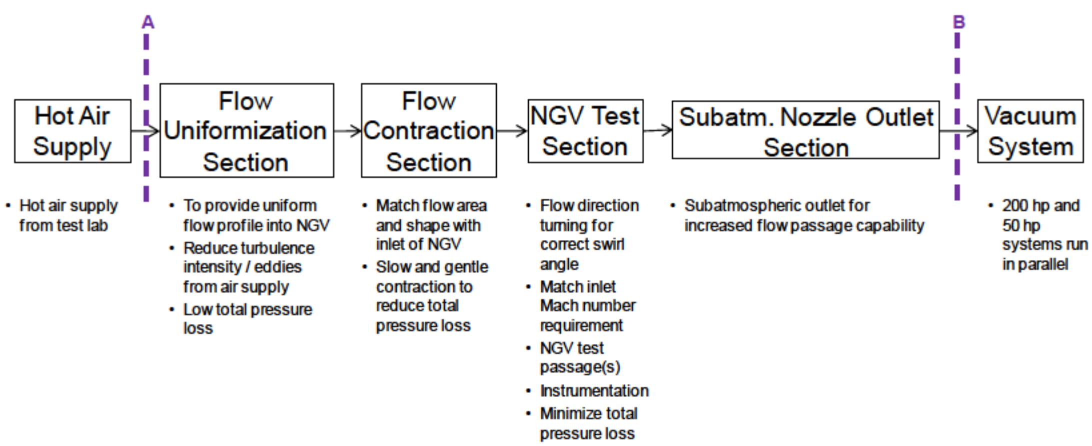


Figure 11.16 - Block Diagram of Test Configuration

Design activity for each section of the nozzle test rig was completed as follows. See Appendix 11.1.1 for a copy of the nozzle final design review.

- Kick-off Meeting (all sections), May 8, 2013
- Systems Requirements Review (all sections), May 22, 2013
- Conceptual Design Review (all sections), June 19, 2013
- Preliminary Design Review
    - All sections, August 2, 2013
    - Facility Piping (Delta), October 3, 2013
    - Downstream Pressure Measurement Section (Delta), November 6, 2013
- Final Design Review

- Flow Uniformization Section and Flow Converging Section, September 19, 2013
- Inlet Guide Vane, October 4, 2013
- Facility Piping, October 16, 2013
- Nozzle, November 4, 2013
- Downstream Pressure Measurement Section, October 14, 2013

- Production Readiness Review
  - Inlet Guide Vane, October 23, 2013
  - Nozzle, November 21, 2013
  - Downstream Pressure Measurement Section, December 6, 2013

Manufacturing began on October 9, 2013, when the order was placed for the flow uniformization section and the converging section. Installation of supply and exhaust piping in the test facility was completed in December, 2013. By the end of 2013 most major machined parts were on order.

Fabrication of the hardware required for the high expansion ratio nozzle test was completed in 2014. Fabrication of the nozzle vanes by metal laser sintering proved to be challenging and required substantial development efforts by the manufacturer. Issues that were experienced and subsequently mitigated included a sensitivity to build parameters, inspection and set-up challenges for post-machining, tooling development, and warping from residual stresses. The inlet guide vane segment, also fabricated by metal laser sintering, was another challenging part because of its thin trailing edges and leading edge airfoil contours. Both parts were eventually fabricated successfully and lessons learned have been identified within this report. The facility preparations for the nozzle test were completed in the first half of 2014. The flow uniformization section was installed, and nozzle cooling air circuits were completed. The nozzle test program was terminated before final assembly was completed.

Procurement activities for the major, fabricated parts began in October, 2013 and most purchase orders were placed by the end of 2013. Purchase orders for the downstream pressure measurement section were placed in January of 2014. The manufacturing schedule for the components for the nozzle test was aggressive to try and meet the schedule for start of test in May 2014. Most components for the nozzle test were straight forward to fabricate and were acquired without incident. Two parts that proved to be particularly challenging were the nozzle vanes, and the inlet guide vane segment. Both of these parts were laser sintered and fabricated by the same company.

The 861036-1 and 861036-2 nozzle vanes were about 8.50 inches long and 2.30 inches tall. They had an internal pin fin array for cooling, and walls as thin as .027" near the trailing edge for performance. See Figure 11.17 thru

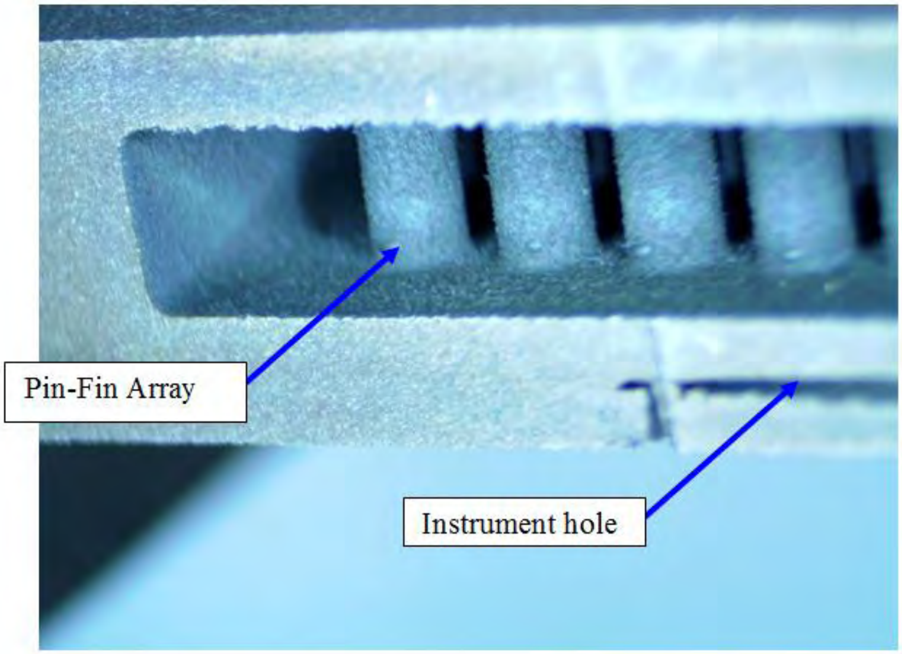


Figure 11.19 for representative pictures of the nozzle vane and its internal features. On typical turbine engines, internally cooled vanes such as these would be fabricated by investment casting with integral ceramic cores that would subsequently be dissolved. Discussions with casting suppliers revealed that the thin walls, length to width aspect ratio, length of the internal cavity, and internal pin-fin features would make this a challenging part to cast that would require a significant development activity. Given the performance goals of the nozzle test program, and the cost and schedule ambitions of the program, it was decided to fabricate the vanes by metal laser sintering.

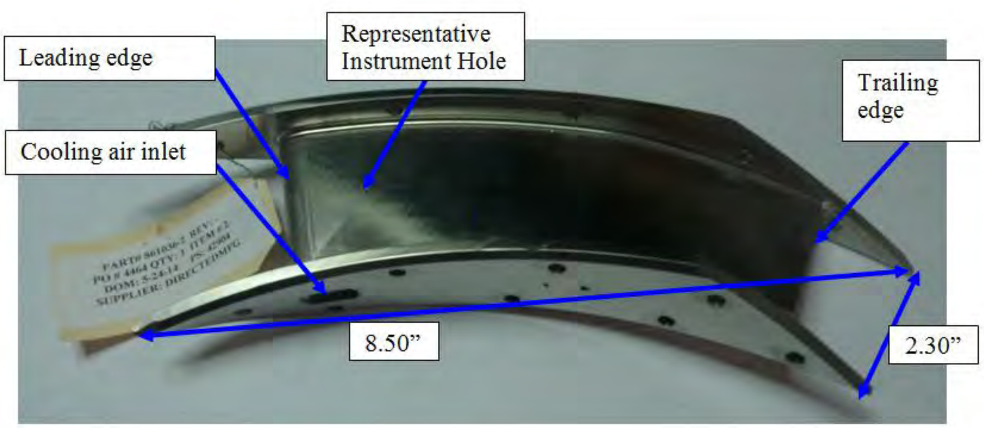


Figure 11.17 - Finished 861036-2 nozzle vane.

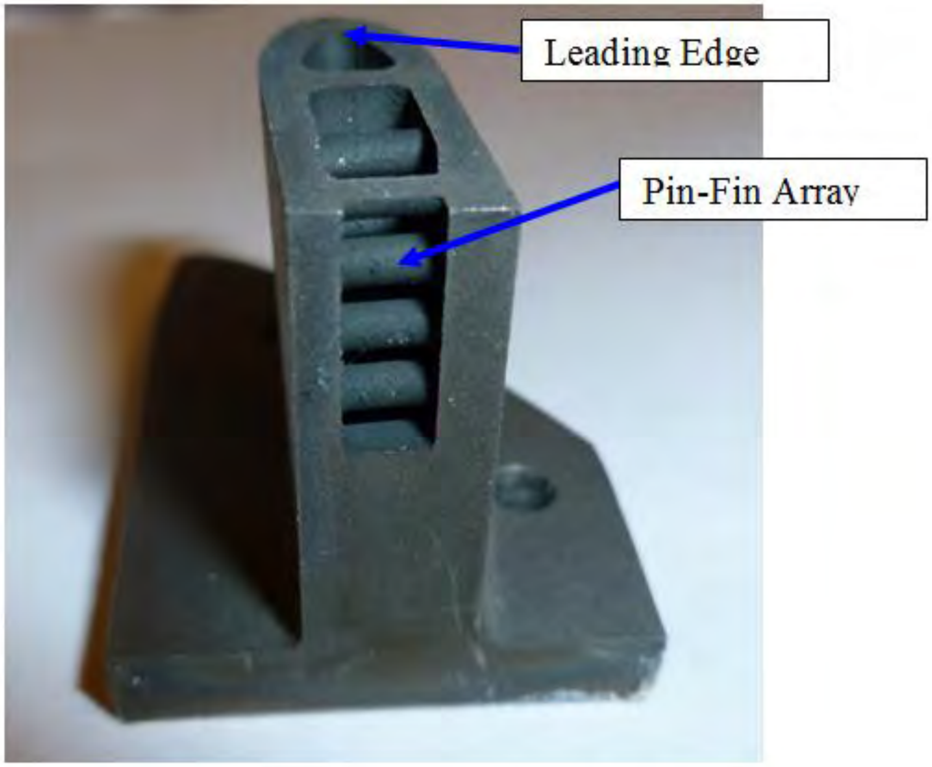


Figure 11.18 - Wire cut segment of vane leading edge from 2013 prototype vane, showing pin-fin array

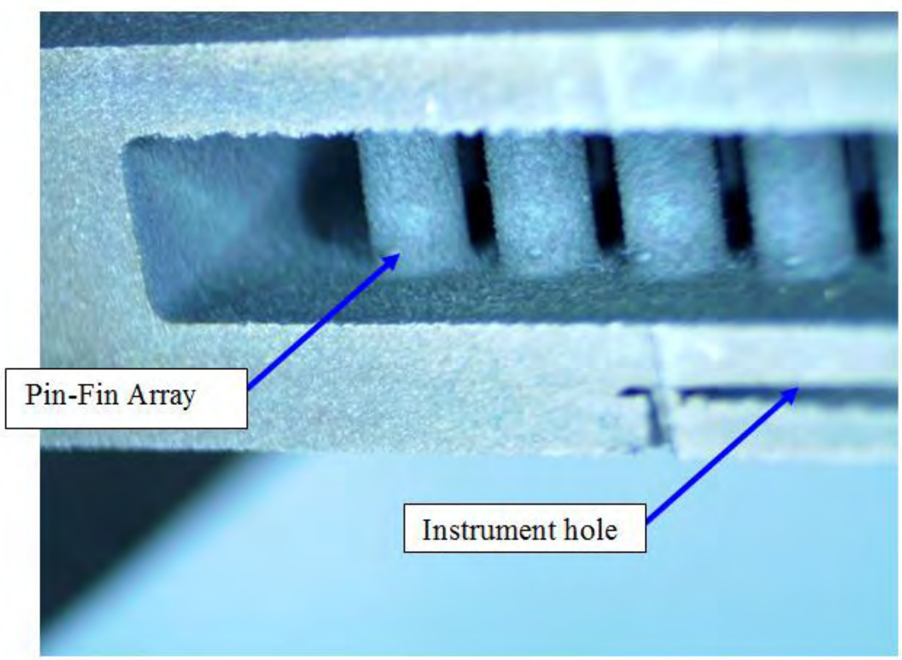


Figure 11.19 - Close-up of pin-fin array and instrument hole from wire cut segment of 2013 prototype.

A net-shaped, prototype vane was fabricated in late 2013, dimensionally inspected, and EDM wire sliced into cross-sections. Dimensional tolerances on the airfoil were acceptable, fabrication of the internal pin-fin features was excellent, and construction of the thin-wall sections near the trailing edge was very good. Dimensional tolerances on the mounting flange were not as tight as desired, so stock material was added to the mounting flanges of the production parts so the flanges could be post-machined.

Manufacturing challenges on the production nozzle vanes began almost immediately. The first attempt to build a production vane was unsuccessful and the laser sintering program aborted part way through as shown in Figure 11.20. The manufacturer had changed to a softer blade than they used for the 2013 prototype to distribute the metal powder and this resulted in uneven distribution of the powder and the defective build. Returning to the more rigid blade used for the 2013 prototype fixed this issue and a new part was fabricated as shown in Figure 11.21.

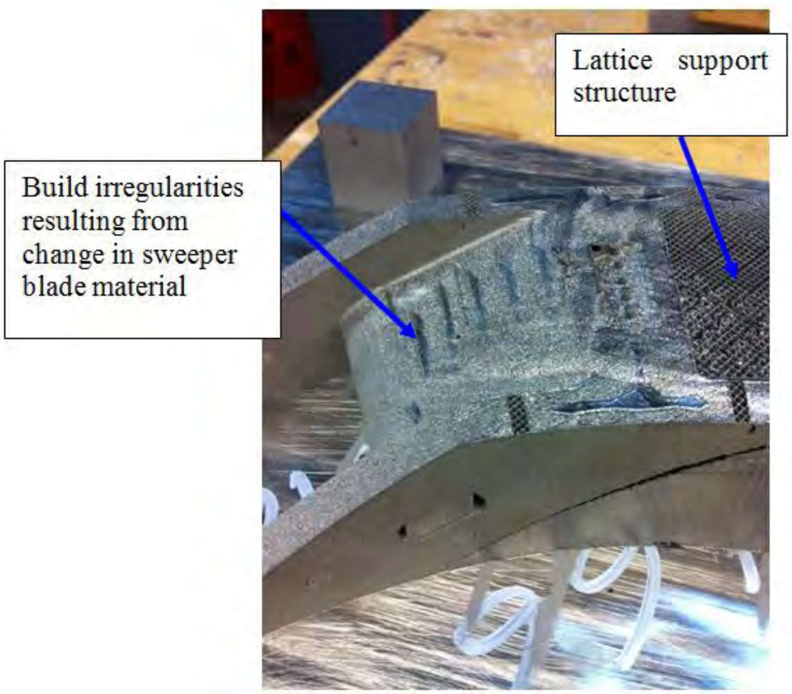


Figure 11.20. - Defective build due to change in sweeper blade material.

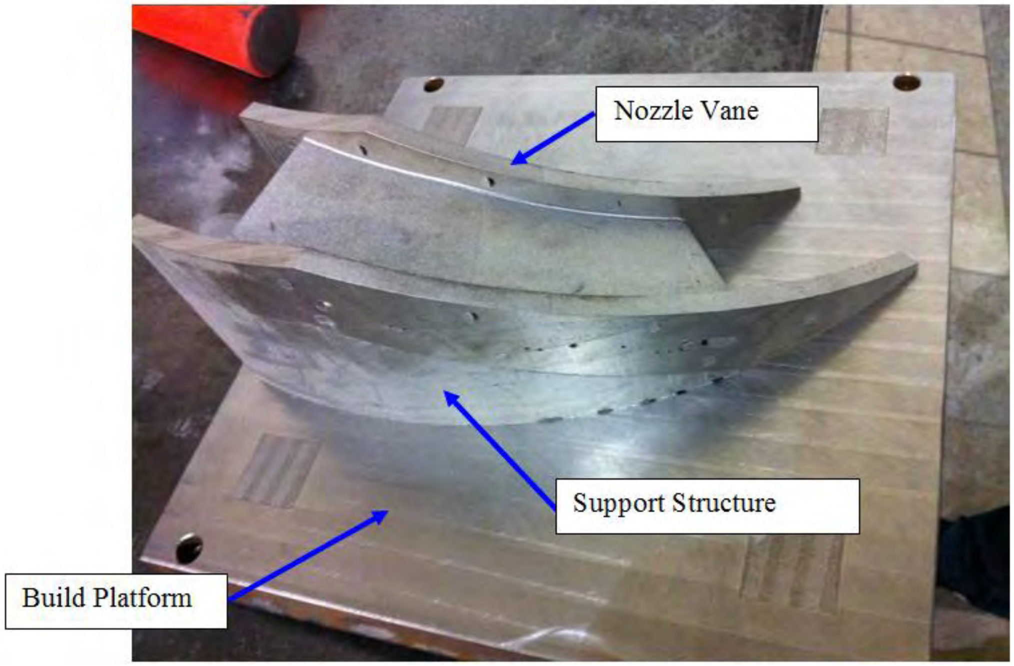


Figure 11.21. - As-designed, as-printed nozzle vane on build platform.

The next issue had to do with getting proper set-up position and orientation for post-machining the flanges on the nozzle vane. Once the measurements were made on the part, it was determined that the build platform was warped. The platform was machined flat so the part would set flush, and machining commenced once the set-up had been dimensional verified. A subsequent programming error related to the set-up caused the first 861036-1 part to be scrapped. Additional gage points were added to the nozzle vane and the build platform as shown in Figure 11.22 to make the dimensional set-up easier and help to validate the programming alignment. Stock material was added to the airfoil surfaces such that they could be post-machined with precision relative to the mounting flange.

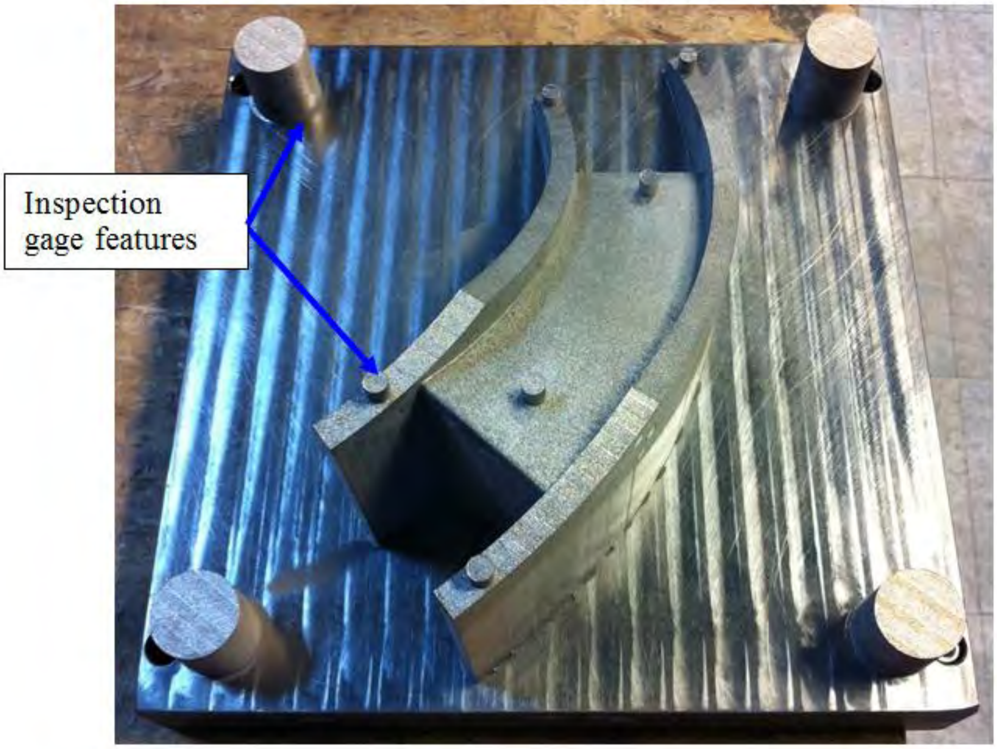


Figure 11.22 - Modified vane and build platform with additional gage features.

The 861036-1 part was successfully machined from the revised build configuration with the additional gage features and stock material. It was delivered to Ramgen in May, 2014. The results of a laser dimensional scan are included in Appendix 11.1.2.

Unfortunately, the 861036-2 part warped when it was removed from the build platform during manufacturing, even though it followed the same build sequence as the successful 861036-1 part. The manufacturer observed that the part was rocking when installed into the machining fixture. A subsequent laser dimensional scan, included as Appendix 11.1.3, confirmed that the part was twisted. The scan showed about .020" of variation across the machined flanges of the part with the scan aligned with the surface of the air foil.

Warping was a known risk due residual stresses from the sintering welds and the manufacturing process already included a stress-relief annealing operation to mitigate this issue. However, the stress-relief annealing operation appeared to be insufficient. Since finish machining had already been completed the exposed flange and vane surfaces prior to removing it from the build platform, it was not possible to perform the second machining operation and meet tolerance requirements necessary for proper fit-up to mating parts. Options to try and deform the vane back into shape were considered, but this would have a low probability of success so it was decided to start over and sinter another part.

The rebuilt -2 part manufacturing sequence was modified so that the part would be removed from the build platform before final machining of any surfaces. The logic was that it would be allowed to warp and then be finish machined in the warped state. This would ensure that the air foil and flange surfaces would be in precise position relative to each other on the finished part. The fixturing needed to be modified slightly for this approach. The rebuilt and final -2 part was received in July 2014 and is shown in Figure 11.17.

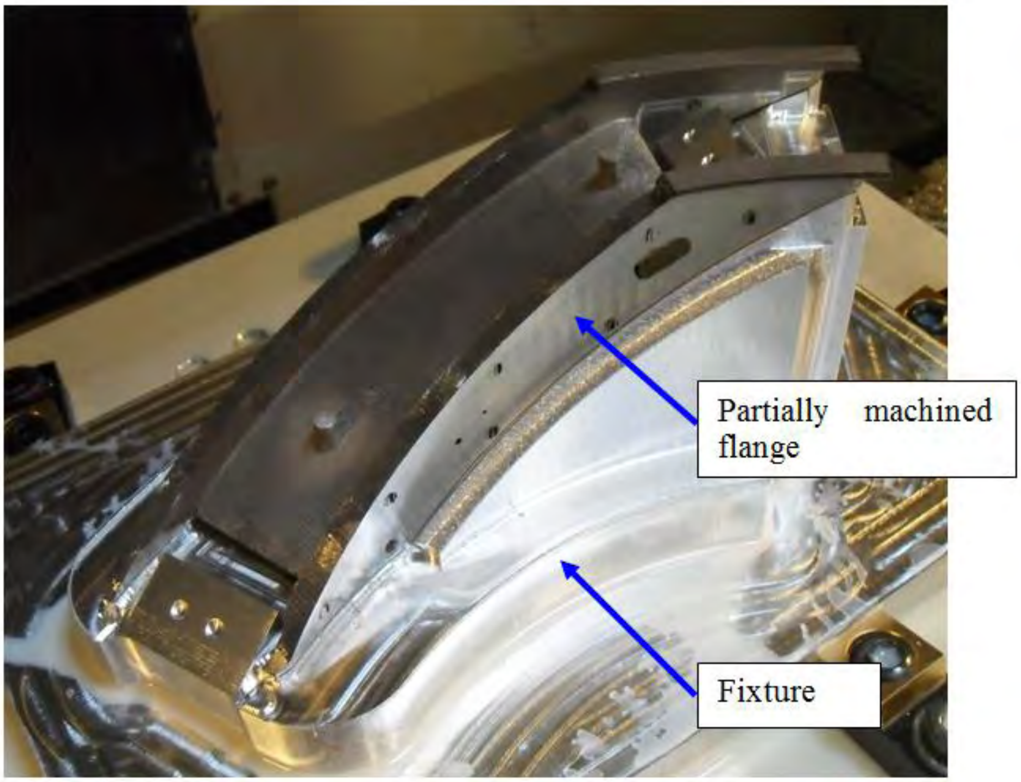


Figure 11.23 - Partially machined nozzle vane in the machining fixture.

The design of the nozzle vanes required slots to be fabricated in the trailing edge for cooling air running through the internal cooling cavity to be ejected back into the primary gas stream. Fabrication of these slots required secondary processing (EDM) at a separate supplier and this step was not completed

The 861010 inlet guide vanes were also fabricated by metal laser sintering. Metal laser sintering was chosen on the basis of cost and schedule over conventional machining. The challenge to laser sintering the inlet guide vanes was the thin trailing edges (.011" nominal thickness), and abnormalities created by the removal of the build structure necessary for its fabrication

The first attempt at fabricating the inlet guide vanes used 316 stainless steel powder based on the manufacturer's input that this material built small features the best and it was the most likely to give good results at the thin trailing edge. Unfortunately, the low strength of the 316 stainless steel allowed the trailing edges to be easily damaged and to sag. See Figure 11.24 through Figure 11.26 for images of the 316 SS inlet guide vane. Several of the trailing edges were distorted. It was not clear whether some distortion existed in the as-printed condition, or if it all occurred during subsequent post-processing and grit-blasting operations. It was also observed that manual clean-up operations to remove the latticed build structure required to support the powder during fabrication misshaped the airfoil contour on the leading edges.

Figure 11.24 - View of the leading edges on the 316 SS inlet guide vane.

Figure 11.25 - View of the trailing edges on the 316 SS inlet guide vane.

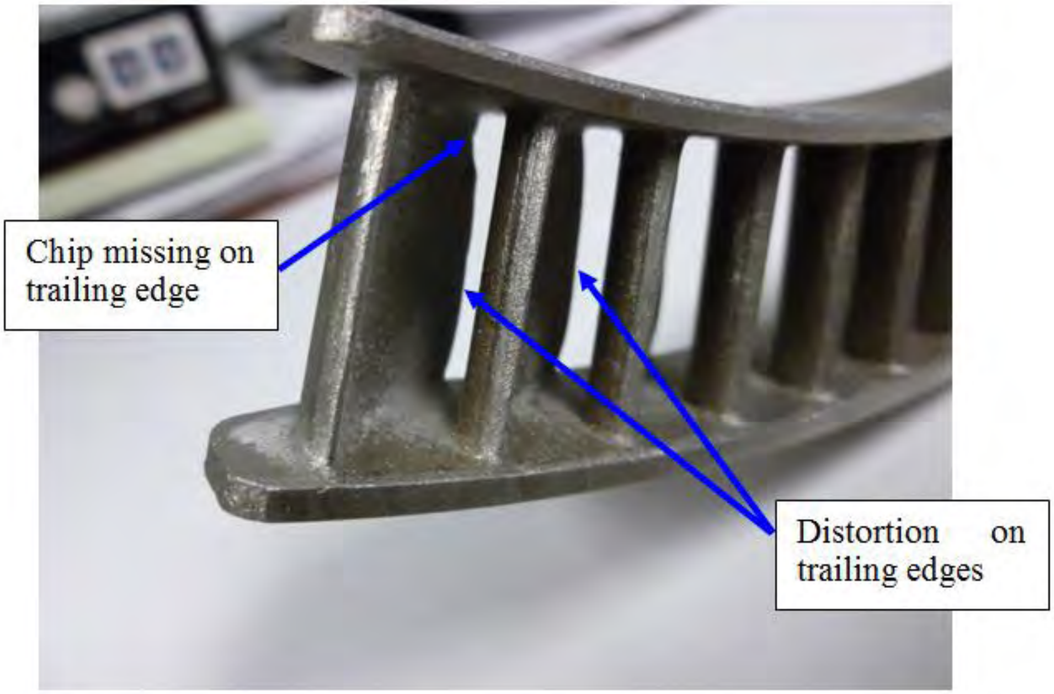


Figure 11.26 - View of the trailing edges on the 316 SS inlet guide vane.

The manufacturer suggested a change to cobalt chrome for the second build of the part. Cobalt chrome had significantly greater strength, but still built small features fairly well. An inspection gage was created to be used during the manual clean-up operations on the leading edge. . The grit-blasting operation was eliminated to reduce risk to the trailing edges.

The cobalt chrome material formed acceptable, straight trailing edges as shown in Figure 11.27. Figure 11.28 shows the leading edges in both the as-built condition after the removal of the support structure, and after the manufacturer attempted to clean up the leading edges to match the inspection gage. The leading edges cleaned up by the manufacturer tapered to more of a point than aerodynamically desired, so the manufacturer was asked to deliver the part as-is to Ramgen. Engineering staff at Ramgen finished cleaning up the leading edges and matched them to the inspection gage. Since the grit-blasting operation was eliminated, the part had more surface roughness than was aerodynamically desired so the parts were sent by Ramgen to a third party surface finishing operation. The finishing process was able to reduce the average area surface roughness from about 1000 micro inches, down to about 2 micro-inches. Figure 11.29 shows the surface finish on the as-printed part, and Figure 11.30 shows the polished, finished part.

Figure 11.27 - View of the as-printed trailing edges on the cobalt chrome replacement part.

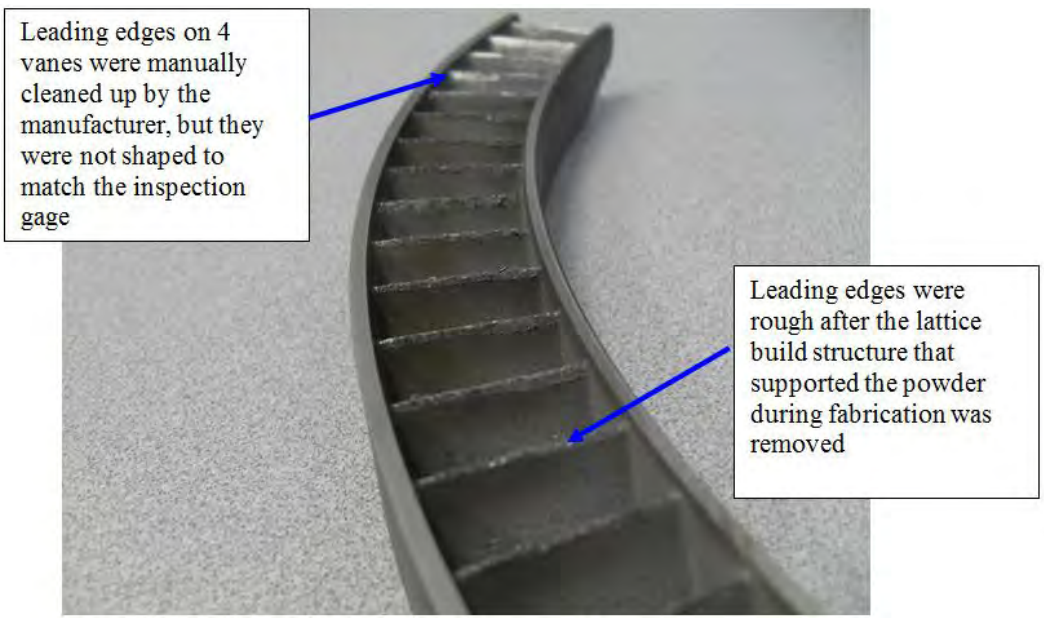


Figure 11.28 - View of the leading edges on the cobalt chrome replacement part.

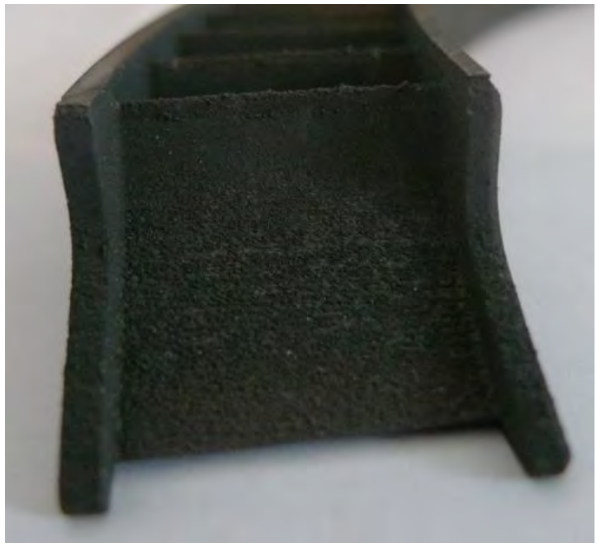


Figure 11.29 - As-printed surface finish on cobalt chrome replacement part.

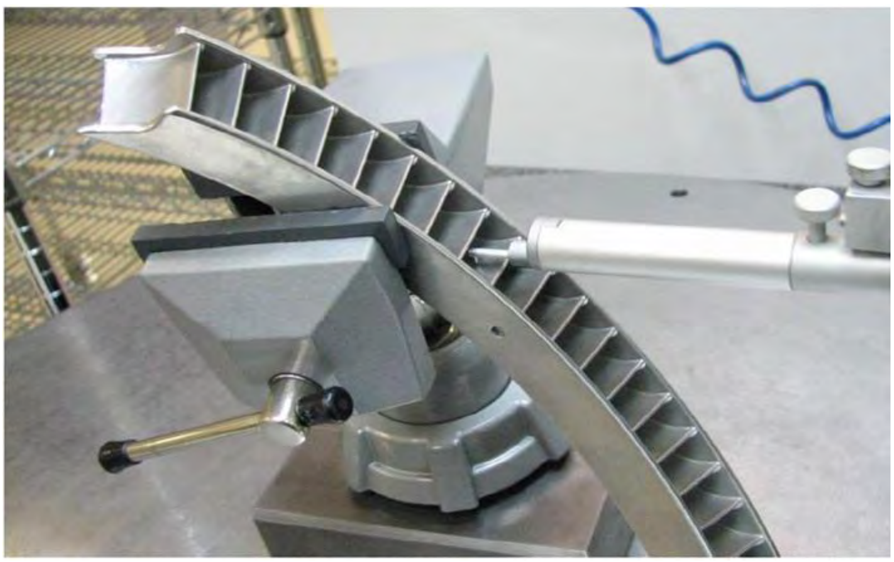


Figure 11.30 - Polished cobalt chrome inlet guide vane.

Original plans called for all nozzle components to be sent to a third party for instrumentation and assembly. However, the AVC test program had started by the time the last nozzle vane was received so resources were diverted from the nozzle task to the AVC test. Parts were kept in Redmond, Washington and assembly was completed by the engineering staff without instrumentation in order to provide a fit check on all parts. The assembled nozzle is shown in Figure 11.31 thru Figure 11.34, with key features labeled.

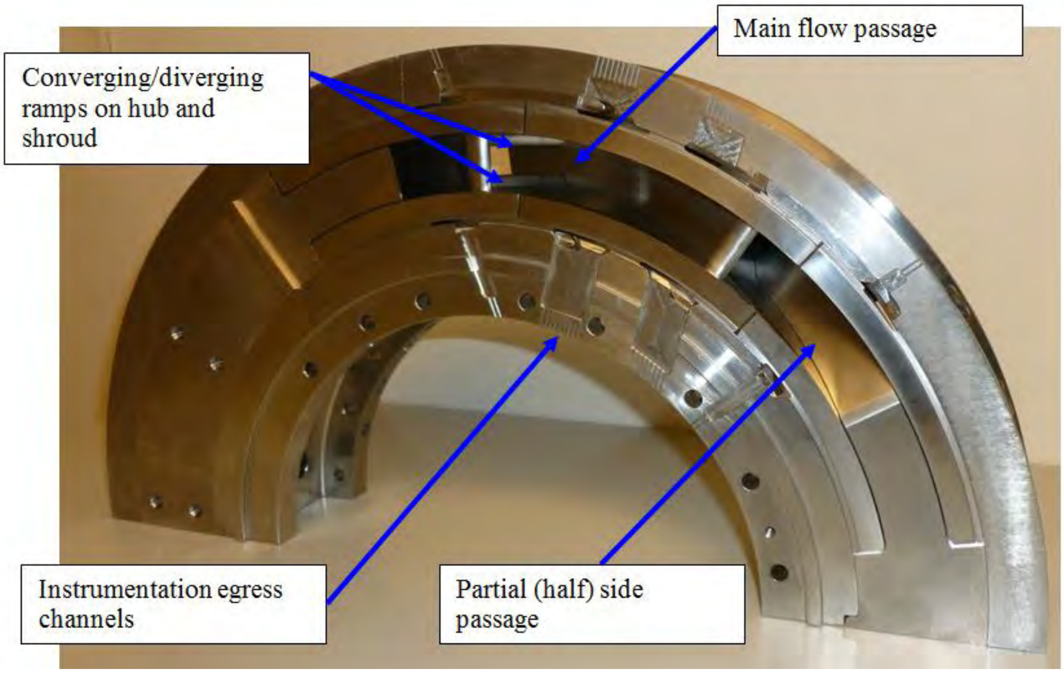


Figure 11.31 - Inlet side of assembled nozzle test article.

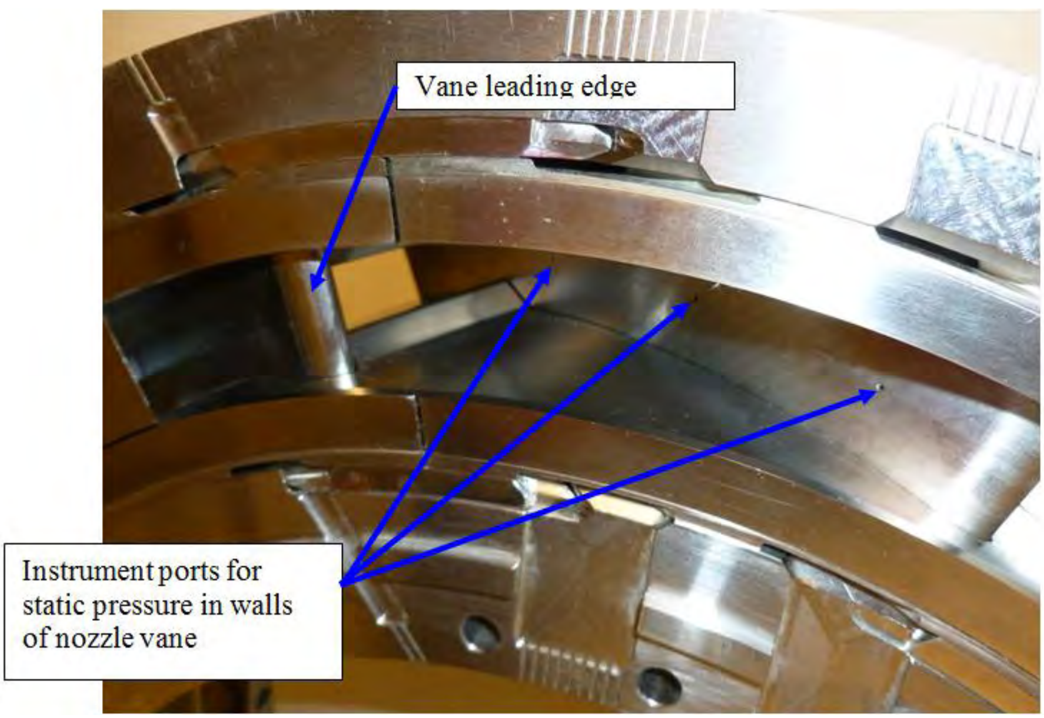


Figure 11.32 - Inlet of primary flow passage.

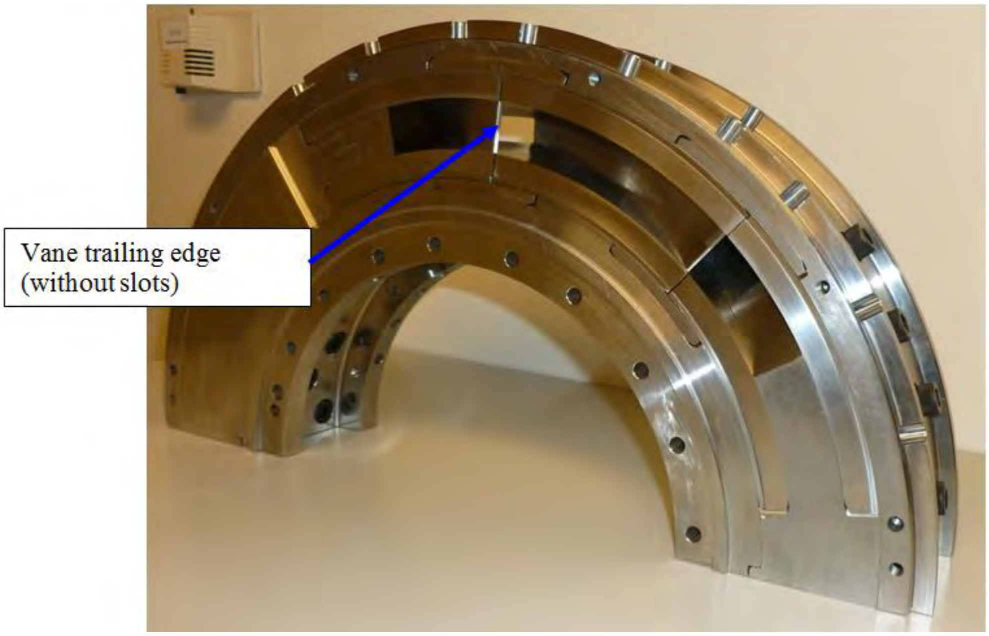


Figure 11.33 - Outlet side of assembled nozzle test article.

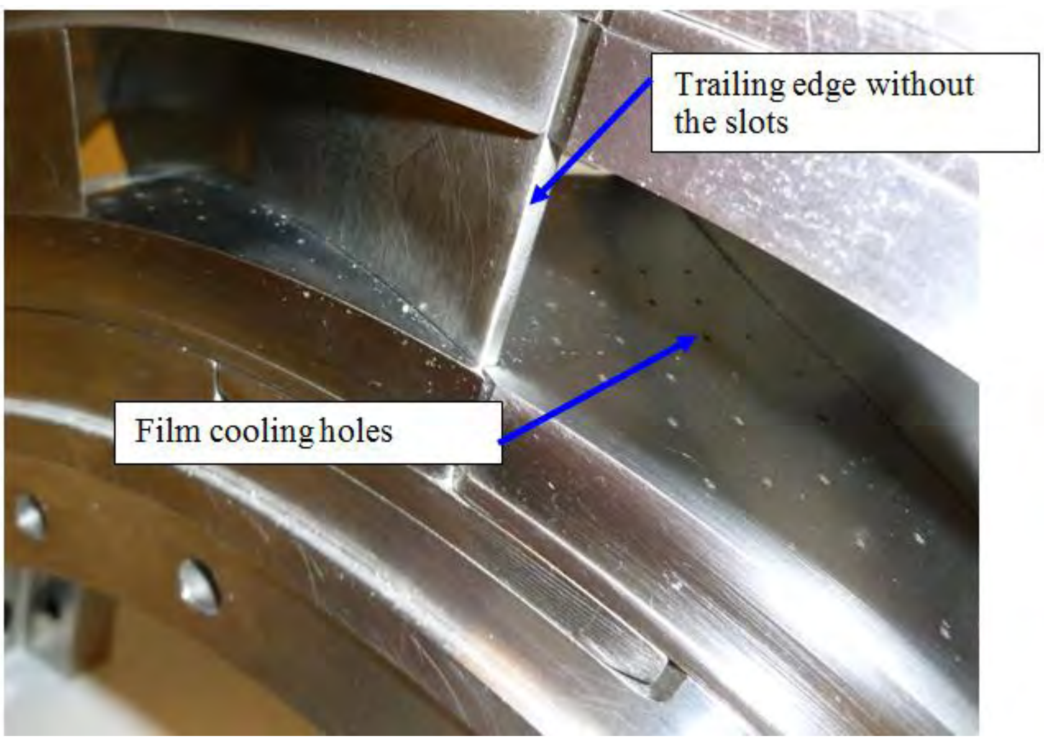


Figure 11.34 - Outlet side of nozzle showing film cooling holes on hub ramp.

During the first quarter of 2014, while parts were being fabricated, final preparations were underway in the facility to prepare for the test. The installation of supply and exhaust piping had already been completed in 2013. The flow uniforming section was installed as shown in Figure 11.35 and the control valves and flow meters for the nozzle cooling air were installed as shown in Figure 11.36 -. Both of these were done in accordance with the 8610002 Nozzle Test Piping drawings.

Figure 11.35 - Flow uniformization section installed in Redmond test lab.

Figure 11.36 - Nozzle cooling air control valves and flow meters installed in Redmond test lab.

Although the test was not completed, it was proven that the complex geometries of the nozzle vane and the inlet guide vane could be fabricated by metal laser sintering. Several key lessons learned along the way were:

a) Plan on a development process that includes all anticipated post-machining operations. Laser sintering is still an evolving technology with uncertainty.
b) Blade sweeper material has a significant impact on the build of the parts.
c) Complex geometries should include inspection gage features. The complete inspection and manufacturing sequence should be planned in advance to ensure that the inspection gage features are adequate.
d) Residual stress can cause robust parts to warp. Stress-relief annealing may not be sufficient to prevent warpage. Parts that may be prone to warpage should have all tightly toleranced features post-machined after removal from the build platform. Additional techniques to mitigate warpage, or a detailed review of the stress-relief parameters, may be required if warpage cannot be tolerated.
e) Cobalt-chrome was acceptable for building thin-walled guide vanes, where-as 316 SS sagged and was easily damaged.

11.2 Task 4.3.4 Turboexpander Test

The goal of the ISCE turboexpander subcomponent test was to demonstrate the operational capabilities of the coupled Ramgen supersonic nozzle and turbine rotor blade design, representing a single stage turboexpander. Ramgen's innovative supersonic nozzle concept fully expands the combustion gas, lowering the temperature such that the turbine rotor blades do not require cooling, while still allowing a competitive combustor firing temperature and best performance potential of the cycle. The turboexpander technology was to be demonstrated by modifying the ISCE Build 1 test engine. This approach was chosen as the best opportunity for Ramgen's turboexpander technology demonstration, while also leveraging existing hardware and facility capabilities from the previous ISCE Build 1 test engine for cost, cycle, and risk reduction.

As this testing was to specifically target the turboexpander subcomponents, the existing engine compressor section was to be removed. Compressed air was to be supplied via external compressors. This simplification eliminated technical risk associated with coupling the turbine and compressor sections together, while enabling more flexibility in providing the required compressed air flow rate and pressure to best suit the turboexpander design conditions. The existing engine combustor was to be replaced with the AVC combustor tested in Task 4.3.2. The AVC combustor was capable of operation to the required pressure ratio. The nozzle design was to be a complete annular version of the nozzle validated in the Task 4.3.3 sector test. The exhaust flow path downstream of the turbine blade was designed to mate with the existing engine exhaust components, minimizing redesign. The power generated would be appropriately off-loaded by the test facility in Redmond.

The testing was segmented into two parts. The first test was the Turboexpander Module Checkout Test. The goal of this test was to validate the facility subsystems through the operation of the existing ISCE Build 1 engine with minimal modifications, prior to the major engine rebuild with the Ramgen turboexpander for the primary test, designated as the 10:1 Turboexpander Test. The intermediate checkout test eliminated risk associated with the facility subsystems in parallel with the design and manufacture of the Ramgen turboexpander components. The subsequent sections discuss the two tests; checkout and primary, in further detail.

Turboexpander Module Checkout Test

The objective of the preliminary Turboexpander Module Checkout Test was to commission the test facility subsystems while gaining turbine operational experience, in parallel with the design and manufacture of the subsequent 10:1 Turboexpander Test hardware. A simplified schematic of the overall approach is shown in Figure 11.37 below.

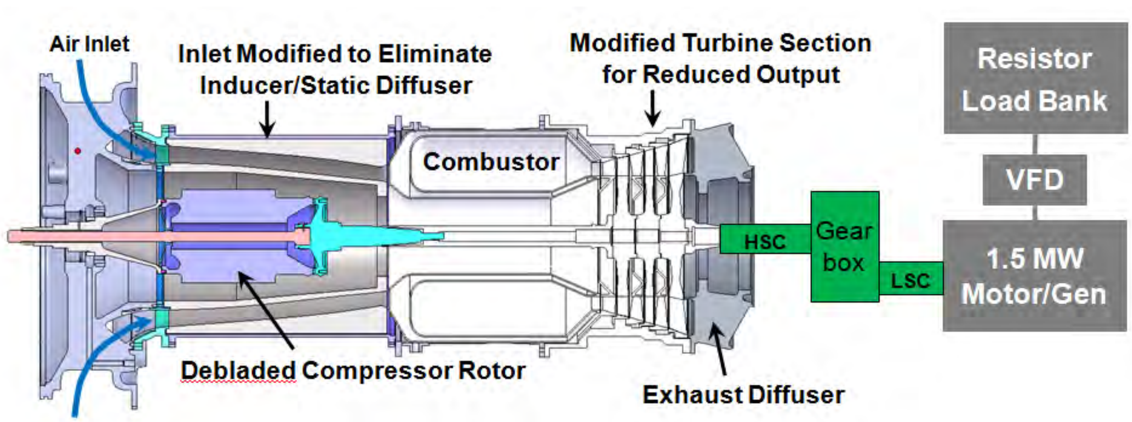


Figure 11.37 - ISCE B1 Engine Schematic, Modified for Turboexpander Module Checkout Testing

The test facility in Redmond consists of the following subsystems:

1. External air compressors, air supply piping and delivery, and control system is to be installed with the capability of providing up to 10 lbm/s flow rate at up to 150 psia (rated for the 10:1 Turboexpander Test conditions).
2. Existing natural gas compressor, fuel supply piping and delivery, and control system from the ISCE B1 engine test is to be reused.
3. Power output is to be dissipated with the existing VFD, 1.5 MW motor, and load resistors from the ISCE Build 1 test.

The existing ISCE B1 engine was to be reused for the preliminary test, with minimal modifications in support of the facility commissioning and operation. Modifications made focused on the turbine inlet to provide the compressed air flow to the combustor, and the turbine section to limit power output of the decoupled turbine (no compressor load) to within the capacity of the drive motor for power dissipation.

The design approach for the turbine inlet modification was to replace the rotating ISCE B1 compressor inducer blades with a smooth wall flow path static structure, fastening to existing structure and fitting over the debladed compressor rotating shaft. The diffuser portion of the compressor would be maintained. The new inlet components as well as the unmodified inlet components were capable of withstanding the pressure and thrust loadings for airflow delivery to the combustor at conditions needed for the 10:1 Turboexpander Test. The modified inlet structure was designed to maintain the stiffness of the original engine configuration. Additional details regarding the air inlet design can be found in the inlet FDR in Appendix 11.2.1.

The three stage turbine section of the ISCE B1 engine was designed to produce 2.4 MW of power, distributed between compressor drive and engine net output. With the decoupling of the compressor section, the full turbine output will would be dissipated by the motor and VFD. The existing motor and VFD system was only capable of dissipating 1.5 MW (2,000 hp) at design speed. The turbine third stage was debladed for power reduction, enabling the use of the same drive train.

Deblading of the turbine third stage was encompassed by three modifications, shown in the rough layout in Figure 11.38 -.

1. Removal of the turbine blades and machining the disk to a reduced diameter (blue).
2. Removal of the nozzle vanes and inner core structure, reusing the outer ring to preserve the flow path wall (blue).
3. Installation of a flow guide (gray), attached to the downstream exhaust diffuser (red) to preserve the inner flow path wall.

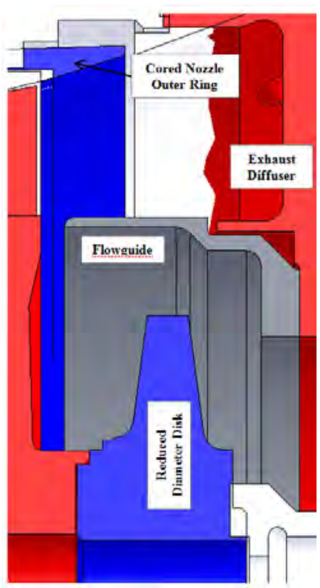


Figure 11.38 - Turboexpander Module Checkout Test Turbine Section

The reduced diameter turbine stage 3 disk is shown in Figure 11.39.

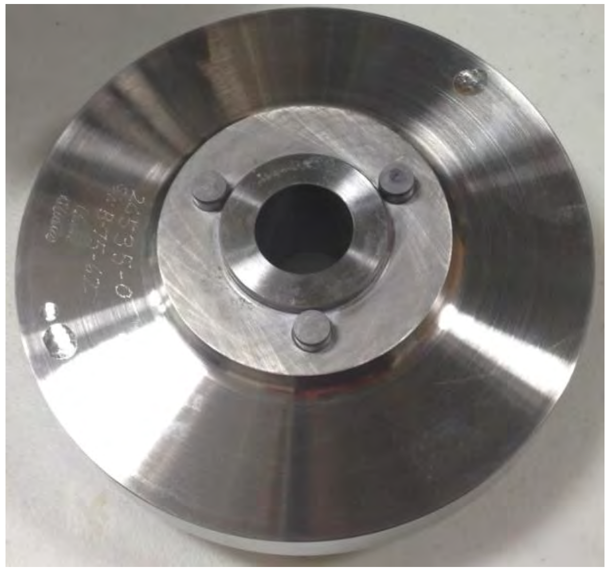


Figure 11.39 - Modified Turbine Stage 3 Disk

The inner flow guide, bolted to the exhaust diffuser is shown in Figure 11.40.

Figure 11.40 - Assembled flow guide

The assembled stage 3 disk and nozzle outer ring are shown in Figure 11.41.

Figure 11.41 - Reassembled Stage 3

Additional details regarding the turbine section modification design can be found in the FDR in Appendix 11.2.2.

In Q1 2014, preparations for a turboexpander checkout test were completed. To ensure safe operation, a series of tests were undertaken. First the engine would be run with cold flow – no combustion. Then the engine would be hot fired in which the engine combustor would be ignited and more power would be produced by the machine. Objectives included verifying that the turbine continued to operate properly after the de-blading operation, commissioning the engine-driven compressed air supply system, and commissioning the load resistors used to dissipate the electrical energy created by the turbine. Without the compressor power sink, the turbine could have been subject to speed runaway and potential failure in the event of a load resistor failure. Therefore careful attention was devoted to the overspeed detection and emergency shutdown system of the test facility.

By the second week of March, 2014 all systems had been satisfactorily commissioned and the test rig was deemed ready for air flow tests. A cold flow (low-energy) series of tests were undertaken to minimize the amount of energy driving the turbine in case of load resistor failure. Objectives included creating an unfired turbine map with the new de-bladed configuration and exercising the load resistors to a significant fraction of their total capability in advance of the high-energy fired test.

Compressed ambient-temperature air was introduced into the engine while rotating at speeds between 2,500 RPM and 15,000 RPM. Air mass flow ranged from 0.9 to 8.4 lbm/sec (max compressor capacity). The typical test procedure involved bringing the engine to low speed without air flow, then gradually increasing air flow to the desired set point. Mass flow sweeps at constant speed or speed sweeps at constant mass flow would then follow, with the test rig being stabilized at a given flow condition for at least 30 seconds before a data point was taken. A typical test profile is shown in Figure 11.42. Figure 11.43 - shows the turbine flow coefficient data obtained in this sequence.

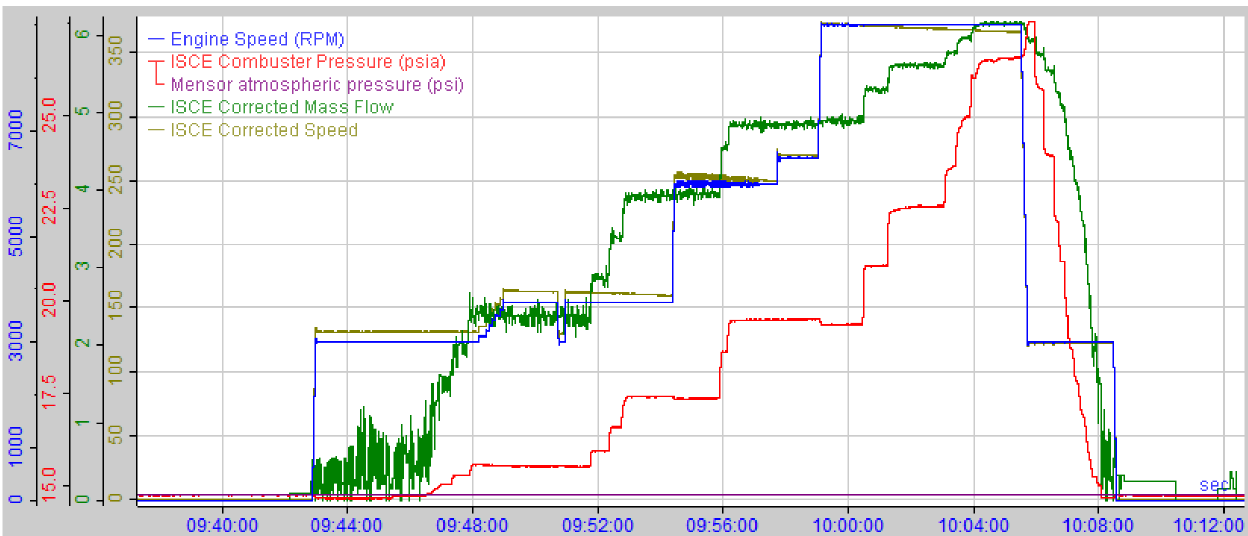


Figure 11.42 - Typical Turboexpander Test Sequence

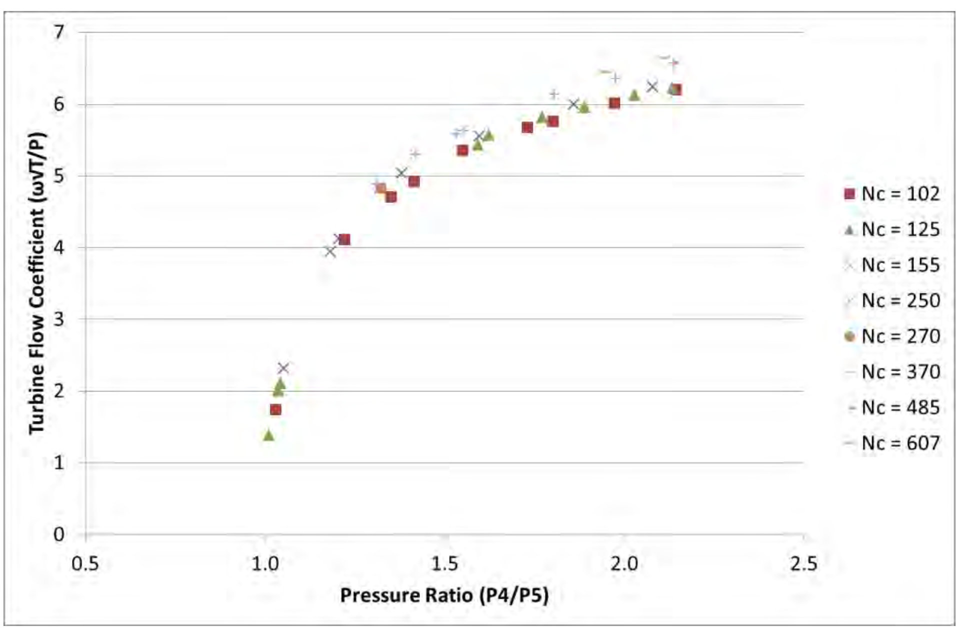


Figure 11.43 - De-Bladed Turbine Flow Coefficient

Following these tests, a full-speed speed ramp was performed with maximum airflow to ensure proper rotordynamic behavior and verify engine operation at full speed. In all respects, the cold flow test rig operated in a predictable, safe manner, laying the groundwork for the move to hot-fire testing.

The next step in the turboexpander checkout procedure was to run the engine hot by firing the combustor, generating more power and verifying the turboexpander test rig systems could dissipate the power being generated. Unfortunately, the program did not proceed on to the hot-fire test steps. By the time the rig was ready for hot-fire the AVC test program was also ready for testing. Although the two tests were in different cells of the test facility, both tests could not be running at the same time, due to facility constraints. The AVC testing was a higher priority as it was prepared to acquire new data for the test configuration and the turboexpander checkout test was a facility capability validation test. In the early spring, Ramgen moved all resources from the Turboexpander module checkout test to the AVC test effort.

10:1 Primary Turboexpander Test

The objective of the 10:1 Turboexpander Test was to demonstrate Ramgen turboexpander technology in a full annular configuration, at the most relevant conditions possible in terms of air flow rate, pressure ratio, temperature, and speed. This was all to be done by retrofitting the existing ISCE B1 engine, leveraging existing hardware and facility resources as much as possible to minimize risk, schedule and cost. The existing combustor and three stage turbine section were to be removed and replaced with Ramgen's higher pressure ratio capable AVC combustor, supersonic nozzle and, impulse turbine blade. Also within scope were outlet guide vanes (OGVs) and exhaust section flow path walls to achieve proper flow conditions mating with the existing exhaust structure, minimizing flow loss.

The engine design was segmented on a component basis: combustor, nozzle, turbine, and exhaust. At the end of 2013 the Preliminary Design phase was completed. In 2014 the planned tasks were the Final Design Review, drawing release, and procurement of components for the second rebuild. The more challenging engineering activities were the Combustor, Nozzle and Turbine. The preliminary turbine section layout cross section is shown in Figure 11.44. Direction of flow is left to right.

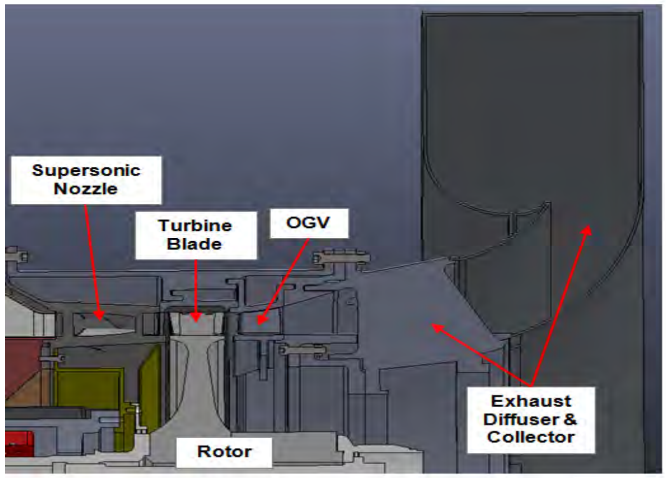


Figure 11.44 - 10:1 Turboexpander Test Combustor, Turbine, and Exhaust Layout

The initial combustor installed in the ISCE B1 engine (to be maintained for the preliminary module checkout testing) was to be removed as it was not rated for 10:1 pressure ratio operation. This included removal of the associated combustor casing, fuel lines, igniters, and seals. The full annular Ramgen technology AVC combustor would be integrated into the test article, with modifications to the exiting hub and shroud radii to accommodate the turbine inlet design annulus target. The AVC combustion hardware would require a thorough inspection to identify features for reuse, rework, and replacement after the AVC component testing. A new combustor casing would also be required, compatible with the engine and rated for 10:1 pressure ratio operation, with a front end mounting.

Technical challenges encountered during the AVC integration design included:

a. Establishing the combustor cavity cooling air flow rate and temperature to balance between:
    i. Insufficient cooling, increasing hardware metal temperatures, risking damage and rupture.
    ii. Excess cooling, generating large thermal gradients in the hardware, risking spalling of thermal barrier coating (TBC).
b. Minimizing flow loss in the flow path transition at combustor inlet. Limited axial space is available in this transition area. This is the same region in which the combustor will be mounted to the static casing.
c. Provision of fuel to the inner hub side cavity, if needed.
d. Design of a pressure casing large enough to enclose the combustor and rated for operation loading.

Ramgen's supersonic nozzle was downstream of the AVC combustor. The nozzle was designed and manufactured as a full annulus, using the same strake and converging-diverging hub and shroud profile design as used in the nozzle sector subcomponent test. Strake profile and count, and hub and shroud profiles were to be optimized for the flow conditions of this specific test.

Technical challenges encountered during the 10:1 supersonic nozzle design included:

e. Obtaining accurate thermal gradient predictions at steady state and transient startup/operation/shutdown conditions.
f. Design of the nozzle hub and shroud geometry to tolerate flow pressure and temperature gradient loading.
g. Design of the flexible nozzle mounting configuration preventing over restriction of thermal growths, inducing bending stress. Thermal stress management presents the greatest challenge of the nozzle component mechanical design.
h. Establishing the most effective nozzle manufacturing process. Casting was selected over machining due to EDM risks and cost, despite concern over finished flow path surface profile and tolerance. An initial full ring casting trial was completed, showing signs of local shrinkage and minor defects.

Additional details regarding the nozzle design can be found in Appendix 11.2.3.

The supersonic impulse turbine blade design would turn the nozzle exit flow approximately 120°. This angle would extract power to be dissipated by the motor and VFD. Power output through the drive train was to be limited to the 1.5 MW facility capacity. For design and manufacture simplicity, the turbine blade was envisioned as integral to the disk, as a "blisk" geometry. The ISCE B1 three stage turbine rotor was to be replaced by a blisk and spacer design, interfacing with the remaining shafting components to be reused. Several airfoil iterations were completed in progression towards an optimum aerodynamic mechanical design.

Technical challenges encountered during the 10:1 supersonic turbine rotor design included:

i. Design of an airfoil to achieve the performance target, fit into the existing turbine section envelope, without generating centrifugal load at speed in excess of the material based loading capability. Centrifugal loading evaluations include:
    i. Average section hoop stress in the disk (disk burst)
    ii. Average section radial stress in the vane
    iii. Peak stress in the vane root, disk web, and disk bore
j. Maintaining proper Campbell diagram resonance frequency margin.
k. Maintaining proper static to rotating component clearance during steady state and transient conditions.
l. Establishing acceptable wheel space purge flow rate and temperature to prevent excessive thermal gradients in the blisk.
m. Accurate steady state and transient temperature predictions. Thermal stress management again presents the greatest challenge of the turbine disk/blade mechanical design.

The rotor design is shown in Figure 11.45 below. Additional details regarding the turbine rotor design can be found in Appendix 11.2.4.

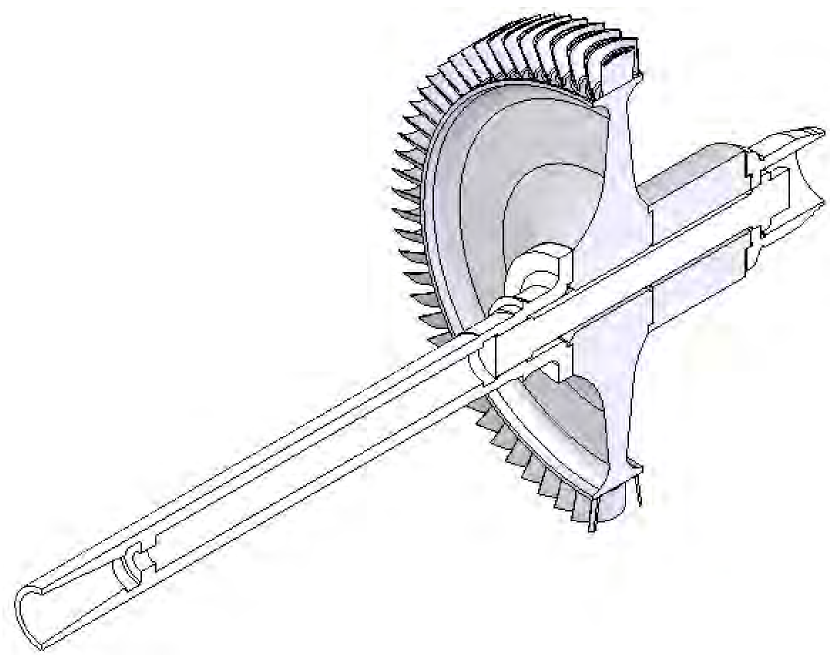


Figure 11.45 - 10:1 Turbo-Expander Rotor Assembly

The purpose of the exhaust portion of the turboexpander was to transition the flow from turbine blade exit to low velocity flow at pressure greater than atmospheric, for discharge out through the exhaust diffuser and collector. The turbine blade exit flow conditions did not meet these criteria, leading to the need for outlet guide vanes. Additionally, a new flow path surface at the hub, and potentially the shroud, was designed for more gradual expansion of the flow into the exhaust diffuser. The design of the turbine exhaust section to transition the flow with acceptable performance, but without significant and costly component rework/replacement proved to be mutually exclusive constraints. This eventually led to a performance study of a turbo expander design no longer limited to the envelope of the existing machine.

The physical constraints of fitting a high pressure ratio turbine into the existing ISCE Build 1 test rig made it difficult to reach the levels of performance we believed we could otherwise achieve. To understand the potential of the technology, we performed some comparative analyses and trade studies without the constraints imposed by the ISCE Build 1 test rig. Some of the variables that were liberated included:

a. Speed of rotation (RPM)
b. Flow path inner diameter
c. Flow path outer diameter
d. Blade geometry

e. Mach number of the flow
f. Blade loading

Some of the unconstrained configurations yielded improved performance over the constrained configurations. Changes in geometry yielded improved nozzle performance and reduced flow speed resulting in less pressure loss across the blades and diffuser. The optimum configuration for the conditions could not be found because the AVC test program, as well as the CO2 Compressor program, required more attention and company resources towards the end of the program – June 2014.

As the engineering team was working through detailed design closure a number of conditions were emerging that ultimately prevented us from building, assembling and testing the Ramgen technology Turboexpander:

- AVC component test start was delayed due to manufacturing delays
- AVC test article would not be available for the Turboexpander test before June 2014
- The Nozzle testing would not be complete and available for the Turboexpander test before June 2014.

Shortly after the FDR reviews were completed Ramgen discussed and agreed with the DOE that we would not pursue more design work on the Turboexpander test and concentrate our efforts on completing the AVC testing and the manufacturing development of the High Expansion Ratio Nozzle.

Supersonic Air Compressor Design

Development of the ISCE Build 2 supersonic compressor began in Q1 of 2013 with the conceptual design of a 20:1 air compressor. Based on successes in the HP CO_2 program in Q4 2012 and Q1 2013, preliminary geometry design and CFD was expected to progress quickly, but problems encountered during adaptation of the CO_2 blade design tool delayed development. Specifically, early attempts at ISCE compressor designs displayed poor performance compared to the tool predictions as well as the targeted requirement, and it was determined the higher blade loading requirement coupled with the lower operational Reynolds number of the ISCE engine contributed to poor designs compared to the HP CO2 program. While the use of optimization was envisioned to aid convergence on a final blade design, these initial results were considered unacceptably low such that they were unsuitable for seeding a study.

To generate an acceptable optimization seed, modifications of the design tool began in Q2, and resulted in complete redesign of the tool with the goal of developing a more robust supersonic compressor design tool applicable to both the ISCE and HP CO2 programs. Prior blade design methods resulted in compressor blades with little to no control of the static pressure ratio, and greater control of static pressure rise was desirable for the Build 2 phases of both the ISCE and HP CO2 programs. This control was marginally available using in-house tools at the end of 2012, but significant efforts were made during the tool redesign phase to simplify the design process and generate higher-performing geometries. A suite of loss models gathered from literature were added to increase the accuracy and relevance of performance estimates, allowing for faster design convergence. Initial CFD showed significant design and performance prediction improvements, and the majority of the tool re-work was completed by July 2013.

Preliminary geometries were generated by the updated tool in Q3, and initial CFD results matched well with performance predictions generated by the design tool. Multiple design iterations were completed during September and October, 2013 achieving 10:1 blades with 90% efficiency. Further design work was put on hold to support HP CO2 activities towards the end of the year.

Simultaneously, Ramgen developed and started running CFD on alternate supersonic compressor concepts where impulse or reaction wheels accelerate the flow to moderate supersonic velocities and new diffuser concepts are used to further compress and diffuse the flow to subsonic speeds.

12. Task 4.4 - 1.5 MW Proof-of-Concept Unit Build and Test

The design and goals of the Integrated Supersonic Component Engine test rig were described in Section 10. Task 4.1 and Task 4.2. This section describes the build and testing of the rig.

The manufacturing was competitively bid and distributed to fifteen manufactures and suppliers in Ramgen's supply chain throughout the United States. GLM in Kenai, AK, was responsible for modification and assembly of the Solar Saturn engine.

In parallel with engine component manufacturing, Ramgen's Redmond, WA test facility was augmented in anticipation of ISCE testing. The facility's electrical supply, fuel supply, control systems, and safety systems were significantly improved. New systems specific to the engine included air supply and exhaust, variable-speed drive (VFD) and motor, vacuum system, cooling system, and braking resistors. The improved facility has proved to be robust, safe, and extremely flexible in response to test needs.

A 2500 kVA transformer was installed behind the building to supply the 1.5 MW drive motor as well as other supporting equipment for the test. Two large motor control centers (MCCs) were placed inside the building, one to feed the 1.5 MW VFD/motor and one to feed the other facility supporting machinery. The 1.5 MW VFD was placed in the engine test cell to reduce conductor length - a significant cost consideration in light of copper prices (Figure 12.1 through Figure 12.4).

Figure 12.1 - 2500 kVA transformer for ISCE test facility.

Figure 12.2 - MCC used for 1.5 MW motor.

Figure 12.3 - MCC for facility supporting machinery.

Figure 12.4 - 1.5 MW Variable Frequency Drive (VFD).

At the engine's design point, it would generate excess power which must be dissipated or used to prevent the engine from over-speeding. To this end, a pair of 750 kW braking resistors was installed on the facility roof. These resistors would automatically turn excess electricity into heat and dissipate via fan-forced convection. The resistors were enclosed with an acoustic barrier to prevent fan noise from propagating into the surrounding neighborhood (Figure 12.5).

Figure 12.5 - Photograph of roof-mounted engine braking resistors.

At full speed operation, the ISCE would consume approximately 14 lbm/sec (11,000 SCFM) of air. A dedicated air inlet duct was constructed on the building roof to ensure the air flow was: clean, free from blockage, doesn't suffer from excessive flow losses, metered and measured and is silenced to prevent engine noise from propagating out to the surrounding neighborhoods (Figure 12.6 and Figure 12.7).

Figure 12.6 - Engine inlet duct (square duct on right).

Figure 12.7 - Engine inlet silencer (left) and exhaust silencer (right).

Hot exhaust needed to be collected from the engine and routed up and out of the building safely. A dedicated exhaust stack was constructed through the roof to ensure the exhaust flow was: free from blockage, free from excessive flow losses, measured, and silenced against engine noise (Figure 12.8). A no-loss stack design was utilized to minimize rain falling down into the engine exhaust collector while the engine was not running.

Figure 12.8 - Photograph of engine exhaust stack.

Natural gas supply of 300 SCFM at 200 PSIG was needed for hot-firing the engine. A 4" diameter, 10 PSIG supply was provided by the local utility, which feeds a 220 PSIG compressor bringing pressure up to the level required by the engine (Figure 12.9). The twin-screw compressor skid used the facility cooling system to after cool the compressed gas, allowing the fuel system to run at full speed in closed-loop mode until the engine was ready to light.

Figure 12.9 - ISCE natural gas supply skid.

Addition of the natural gas system required a significant augmentation of the buildings gas detection and ventilation systems. A networked system was implemented which monitored a variety of temperature, gas, and flame detectors throughout the facility (CO, CO_2, Natural Gas, and others) and signaled alarms as appropriate. The system also manipulated the facility's forced-air ventilation system appropriately in response to elevated temperature (blows fresh air for cooling), gas detection (actuates emergency room air purge), or fire (actuates room exhauster to remove smoke/heat, but does not blow fresh air in). The system interface display is shown in Figure 12.10.

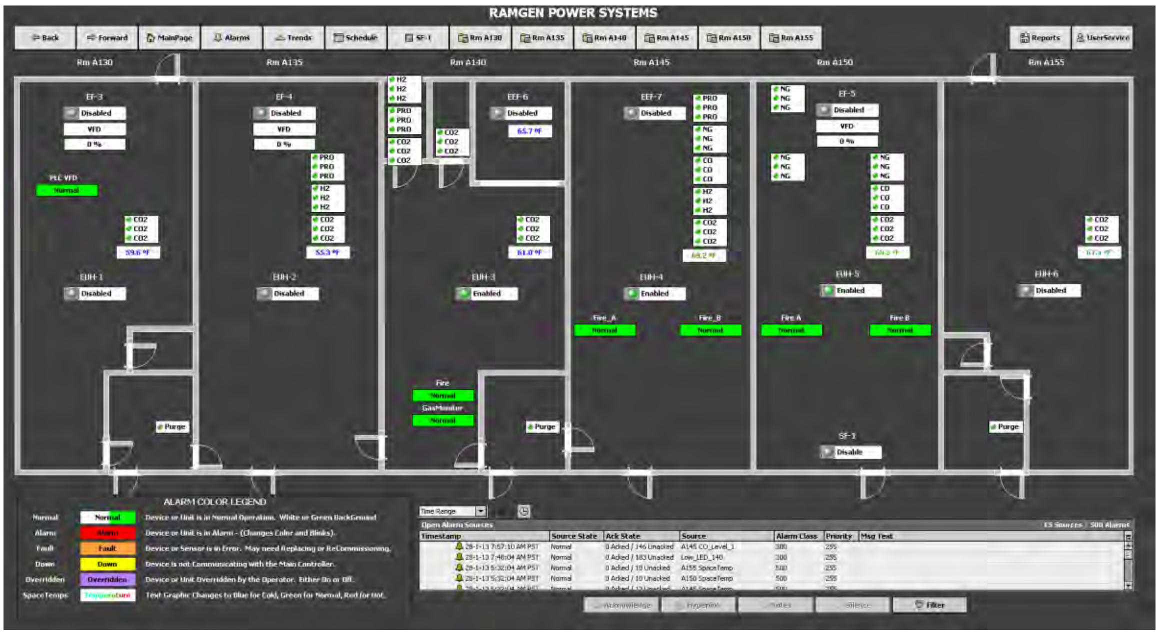


Figure 12.10 - Gas detection & ventilation system display sample

In order to dissipate the heat generated by air compressors, natural gas compressor, turbine oil cooling, and other heat loads, a facility-wide water cooling system was installed. Capable of moving 270 gallons per minute, the system can dissipate 1.3 MW of thermal energy continuously through an evaporative cooling tower located in the parking lot (Figure 12.11 and Figure 12.12).

The cooling system can move water through 4" diameter pipes across the ceiling of the facility, with built-in drops in each test cell for ease of access/use. Trim valves were installed in each cell to ensure that flow was balanced to all equipment being used. A separate 4" pipe returned the heated water to the cooling tower, again with drops in each cell.

Pure water was used as the working fluid, to avoid the environmental impact of chemical additives. A combination ultraviolet/sonic cleaning system was used to prevent the growth of organics. Screens and filters prevented debris from entering the system through the open cooling tower. Discharge water was completely clean and able to enter the city sewer without treatment.

Figure 12.11 - Facility cooling tower enclosure.

Figure 12.12 - Facility water cooling system plumbing, pumps, and cleaning systems.

Beginning in March, 2012 GLM assembled the ISC Engine at their facility in Kenai, AK. Ramgen personnel and contractors attended critical steps of the assembly on-site. The rotor was assembled and balanced prior to installation of the blades. The blades were individually weighed, then installed according to a mass-scattering program to provide neutral balance. The engine was then 'stacked' vertically to assemble the various external components around the rotor.

Assembly was halted at appropriate times to allow instrumentation contractors to install pressure tubing, proximity probes, and temperature sensors for engine health and performance monitoring. Instruments were threaded through the individual parts as the engine was 'stacked' to enable external monitoring of internal conditions. Instruments were coiled and secured until arrival in Redmond where they could be connected to the facility systems.

On May 9, 2012 a low-speed (5,000 rpm) spin was performed in Kenai using an electric starter motor to turn the engine. The engine lubrication system, vibrations, thrust balance, and other critical systems all were shown to be working well.

Following final integration and auxiliary system installation, the skid was boxed up and prepared for shipment to our Redmond facility. On May 18, the skid left Kenai aboard a freight ship headed for Seattle. It arrived in Redmond for installation on May 24, 2012 (Figure 12.13). No shipping damage had occurred.

Figure 12.13 - Boxed ISC engine skid arrival in Redmond.

After installation of the skid into the cell and motor mounting to the skid, final electrical connections were made and the control system completed. On July 26, 2012 a full-speed motor-only spin test was successfully conducted, completing the checkout process for electrical supply, transformer, MCC, VFD, motor, control system and instrumentation.

Following the motor-only test, the low-speed coupling was installed in preparation for engine spin tests. July 27, 2012 saw a low-speed engine test to 5,000 rpm, re-creating the test previously performed in Kenai. All systems were operating per design, with the exception of the control system, which required further code development before moving on to full-speed testing.

After working through the control system issues, engine speed was increased to 13,000 rpm on August 23, 2012 without incident. This corresponded to nearly 60% of the engine's rated design speed of 22,300 rpm. Difficulty obtaining a reliable high-speed tachometer signal delayed further testing until a solution was found. The high-speed tachometer was a critical component for monitoring engine rotordynamics and vibration in real time.

On August 27th, 2012 having solved the tachometer issues, we accelerated toward full engine speed. At approximately 16,000 rpm, a significant vibration was noted and the engine was slowed to a stop immediately. The vibration disappeared on the way down and there was no sign of any damage or change to the engine. This began a lengthy period of troubleshooting and debugging in an attempt to find and eliminate the source of vibration.

Rotordynamic models of the original Solar Saturn engine indicated that the #3 bearing was subject to instability at speeds close to where we encountered vibration. The original engine, however, has been in service at many installations for decades without encountering vibration of this magnitude and type. Rotordynamic models of our modified engine predicted a somewhat more benign response than the original production engine, so we did not expect to see any problems in test. Focus centered on ways this engine was different from the original Saturn and how that might have excited the instability in our test. The two main differences were a) replacement of traditional axial compressor with Ramgen supersonic compressor and b) motor driven rather than combustion-driven.

Although the Ramgen supersonic compressor differed significantly from the traditional axial Saturn compressor, high-frequency piezoelectric probes installed near the rotor did not indicate aerodynamic excitation which might have excited the engine's inherent instability. It did not appear that the compressor was causing the problem. Moving the high-frequency probes to the turbine section also failed to show any aerodynamic excitation from the turbine blades, as might have been caused by turbine off-design operation due to unfired operation of the engine.

During our extensive analytical and experimental investigation, our analyst discovered that the rotordynamic model was highly susceptible by bearing oil temperature. Small changes in temperature were predicted to make large changes in the onset of instability.

In an attempt to explore this effect, the oil heater set point was increased from 100 °F to 120 °F. The resulting test showed that vibration was encountered at much lower rpm, as predicted by the model. The oil heater was then turned off and oil temperatures were allowed to fall overnight for a test at 80F. On October 17, 2012 we successfully accelerated to full speed without incurring dangerous levels of vibration. The vibration was present on the speed ramp, but disappeared entirely at just over 20,000 rpm. Full speed was held for 30 seconds before ending the test.

Subsequent tests showed that vibration onset could be closely predicted by monitoring the bearing oil temperature supply temperature. At full speed without vibration, the temperature would rise slowly due to the engine adding heat to the oil sump. As supply temperature reached about 87F ± 1 °F, the vibration would begin and the engine would be stopped. This behavior was highly repeatable. Operation at full speed was only possible for about 2 minutes before oil sump temperature increased to this level - insufficient time to perform the necessary compressor characterization measurements.

Design changes were implemented to the oil system to allow forced cooling and therefore longer operation at full speed. On November 9, 2012 we tested the first upgrade and were able to dwell almost six minutes at full speed without vibration. An unrelated instrument problem caused the test to end, but based on the oil temperature measurements, the test could have lasted ten minutes before reaching the critical temperature - a reasonable amount of time for compressor characterization.

As part of a design review to assess operating the engine at off-design conditions, our analyst discovered a manufacturing problem with the compressor blades. The blades installed in the engine were not the final design blades that were intended to be built. Analysis of the installed blade showed unacceptable life and the possibility of crack formation. Engine testing was immediately halted for inspection and further investigation.

Partial disassembly of the engine was required to remove the rotor disc and blades. Fluorescent penetrant inspection (FPI) did not reveal any cracking or yielding on the blades. By removing some of the blade shroud mass, blade stresses were reduced to acceptable levels - a significant schedule and cost saver compared to making new blades. Static structure was modified to take up the space removed from the blades.

Full-speed testing before the shut-down had shown that the compressor system had not supersonically 'started', a necessary aerodynamic phenomenon to reach the pressure ratio and efficiency targets desired for the engine. Computational fluid dynamics analyses indicated that increasing the rotor Mach number incrementally would be sufficient to achieve starting. To achieve this Mach increase, inlet guide vanes (IGVs) have been designed to counter-swirl the flow toward the compressor rotor by 30°.

In July of 2013, the final engine tests were performed to survey the exit conditions of the inducer and validate the CFD predictions. The Build 1 ISCE was completed in August 2013. The full flow path was not started. Based on the CFD anchored by the HP CO_2 test results, Ramgen determined that the configuration changes and time required to modify the ISCE Build 1 flow path would be better applied to the next phase of the program. In August, 2013 a presentation (see Appendix 12.1) was given to DOE describing the experiment conclusions and decision not to modify the engine for further testing.

Design work for the second build of the ISC Engine focused on refining the engine cycle deck, on the preliminary design of an inducer blade for the compressor, on a preliminary analysis of the engine secondary flows and the cooling requirements of the turbine nozzle, and on the preliminary design of the engine turbine.

A reaction inducer blade was successfully designed which was able to meet the design total pressure rise requirement albeit at higher rotational speed than originally planned. The CFD analysis of this inducer blade showed that the inter blade passage was fully supersonic (started inducer) and that the inducer provided acceptable performance. Nonetheless a separation zone was observed on the blade suction side from 50% to 75% span which would prove detrimental for the diffuser and overall compressor performance standpoint. Based on these observations Ramgen decided to optimize the blade design to further improve its performance and reduced the separation region observed on the suction side. Generation of the inducer database was completed in 2012 and inducer optimization performed in 2013. In support of the inducer design process, a mechanical feasibility was done on an inducer blade design from the optimization database.

Appendix 12.2 summarizes the results of preliminary secondary flow analysis, turbine nozzle cooling scheme design and thermal analysis. The analysis shows that traditional turbine materials such as Haynes alloys can be utilized to fabricate the nozzle and that approximately 10% of the engine air mass flowrate would be sufficient to maintain metal surface temperatures within acceptable limits. Preliminary fabrication techniques such as direct laser metal sintering were also identified. The secondary flow analysis also identified requirement for all other the engine cooling and/or sealing flows.

Appendix 12.3 summarizes the ISCE Build 2 design concept. Two promising aerodynamic designs were initially identified for the turbine nozzle referred to as covered and uncovered turbine nozzle. CFD analysis performed on both sets of designs showed that the covered design provided the highest kinetic energy efficiency and provided insight on the nozzle design strategies to be followed to maximize nozzle performance. A first stage turbine rotor design was also generated and analyzed which unfortunately failed to provide a rotor with a fully supersonic inter blade passage, i.e. a started rotor. The design of the ISCE Build 2 engine was not completed prior to the end of the DOE award period in June 2014.

APPENDIX 3.1

EUCASS CFD Validation Paper

Numerical Prediction of 3D Shock-Induced Turbulent Flow Separation Surrounding Bodies of Revolution Adjacent to a Flat Surface

Allan D. Grosvenor,
Ramgen Power Systems, LLC. Bellevue WA, USA
Alexander A. Zheltovodov, Eugeny K. Derunov
Khristianovich Institute of Theoretical and Applied Mechanics SB RAS, Novosibirsk, Russia

ABSTRACT

Numerical predictions of complex 3D shock wave / turbulent boundary layer interactions (SWTBLI) are presented. The configuration studied is a set of two identical cylindrical bodies with conically shaped noses aligned parallel to a Mach 4 stream, adjacent to a flat plate. A series of distances between the bodies are analysed, to test ability of the numerical algorithms employed to capture complex 3D turbulent separation phenomena observed in experiments conducted at the Khristianovich Institute of Theoretical and Applied Mechanics. The numerical scheme employed solves the Reynolds Averaged Navier-Stokes equations, and turbulence closure is provided by the low Reynolds number Spalart-Allmaras one-equation model. Among other areas of concentration discussed herein are properties of the computational grid required to capture the flow physics sufficiently to reproduce the complex boundary layer separation patterns. The value of point-enrichment type adaptation over block-enrichment is demonstrated, and the shock capturing properties of 2[nd] order central and upwind discretisation schemes are compared.

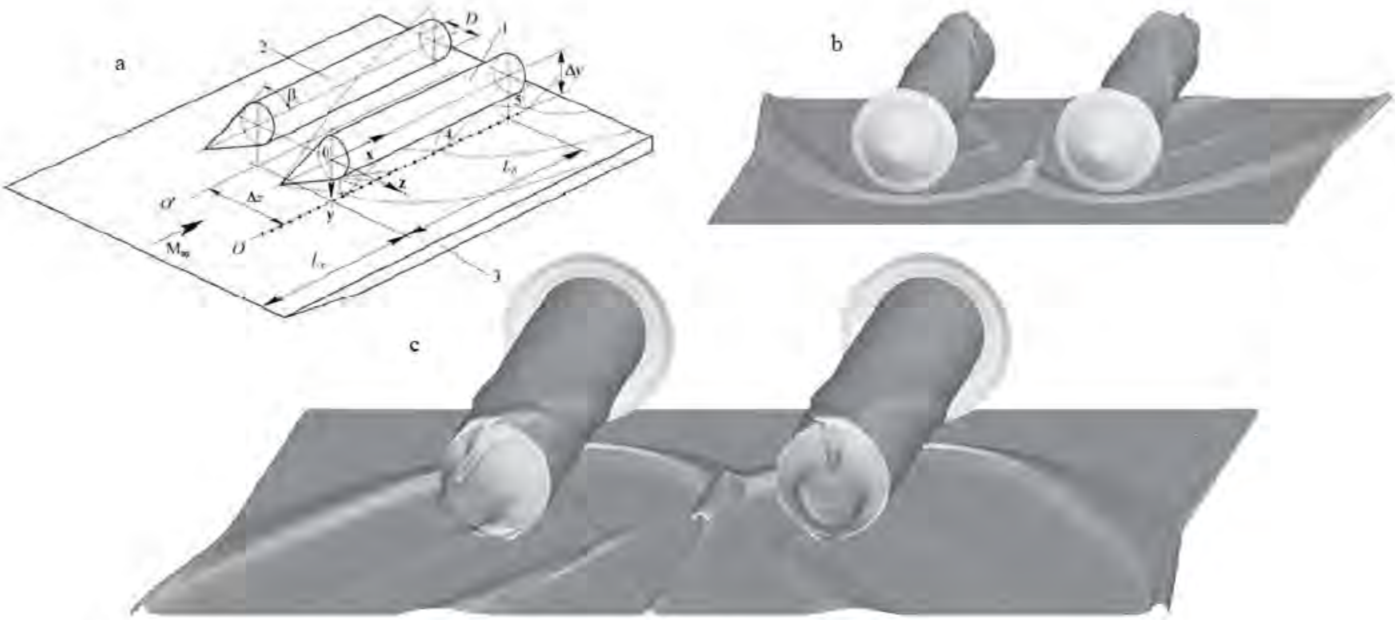


Figure 1. a) Schematic of experiment: 1 - first body, 2 - second body, 3 - flat plate, b) forward looking downstream - c) aft looking upstream - views of surfaces of constant velocity demonstrating 3D separating flow structures from bodies and plate at $\Delta z/D = 3$

INTRODUCTION

The predictions discussed herein represent experiments [1,2] studying the aerodynamic interference of two conically shaped cylindrical bodies of varying proximity aligned parallel to a Mach 4 stream, adjacent to a flat plate (Fig. 1a). As was demonstrated in the mentioned papers, crossing shock waves generated around the bodies, and their interaction with the turbulent boundary layers adjacent to surfaces of the flat plate and bodies initiates the emergence of complex three dimensional separated flow structures. Figure 1b,c indicate the regions of flow separation from the bodies and plate from one of the computations run for this study at $\Delta z/D = 3$. Surfaces of constant velocity are shown here to provide an overview of the 3D flowfield, where one can observe the forebody bow shock waves and their subsequent impingement on the plate and reflections with each other and the solid surfaces. The resulting raised areas of these surfaces indicate the displacement effect of 3D separation zones, which will be shown in greater detail below in terms of limiting surface streamlines. Topology of these separated flows as well as the shock structures, pressure distributions and aerodynamic forces depend on the distance between these bodies. The complex flowfield cannot be predicted based on the assumption of inviscid flow imposed by the solution of the Euler equations, but this simplified approach was employed for preliminary prediction of inviscid shock waves structure, surface pressure distributions and the bodies' aerodynamic forces and moment coefficients [2]. The present work was directed at validation of a RANS approach to predicting the complex 3D shock wave / boundary layer interaction physics of this

NOMENCLATURE

C	Saddle point
D	Diameter of body
F	Focus
L_b	Length of body from base of nose cone
N	Node
S	Line of separation
R	Line of attachment
x	Streamwise direction
y	vertical direction
y_1^+	perpendicular distance from wall to first grid node normalised by friction velocity and viscosity
z	cross-stream direction

flow. The Jameson-Schmidt-Turkel [3] type 2nd order finite volume scheme was primarily employed, and turbulence was closed using the Spalart-Allmaras one-equation model [4]. The low Reynolds number version of this model was applied, meaning that integration was conducted through the boundary layers down to the viscous sublayer.

EXPERIMENTAL TEST CONDITIONS

The supersonic tunnel conditions were set to produce Mach 4 freestream dried air flow at the unit Reynolds number of approximately $55x10^6$/m. Inflow total pressure and temperature were $1.074x10^6$ Pa and 291 K respectively. The adiabatic wall condition was realised in the experiment and the plate turbulent boundary layer thickness was 1.8 - 2mm immediately upstream of the impingement location of the generated bow shock waves on the plate. The body diameter was 50mm, the nose cone included angle ranged from 10-30 degrees (30 degrees is the focus herein - forebody angle = 60 degrees) and the body length measured from the nose cone base was 250mm (L_b/D = 5). The bodies were set at a vertical distance of 48mm from the plate to their centreline axes (*y/D* = 0.96). Distances between the bodies tested ranged from $\Delta z/D$ = 1.06 - 3. Four basic test configurations were analysed in the present study with the values of inter-body distances $\Delta z/D$ = 3, 1.8, 1.4 and 1.06 respectively. In the experimental work, several additional configurations were measured, but four were chosen for this study over the range of $\Delta z/D$ values tested. Measurements taken for comparison included static pressure taken with a dense set of static taps over the plate surface, schlieren photography of the shock system induced by the bodies, oil flow visualisation on the plate and body surfaces as well as balance measurements of aerodynamic forces and moments on one of the bodies.

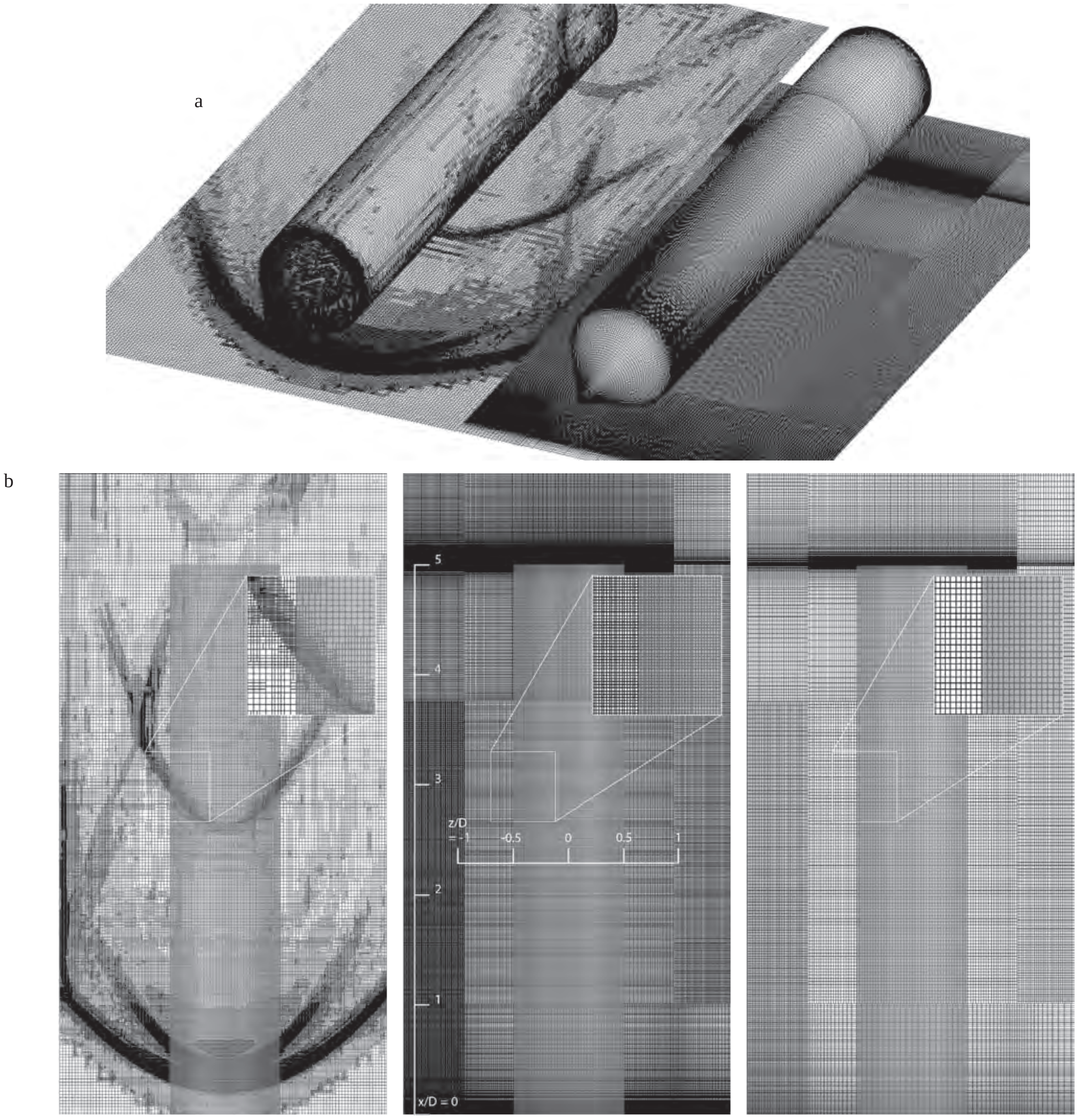


Figure 2. Views of structured and unstructured computational grids:
a) 3D view of structured multi-block grid (right) and solution-adapted hexahedral unstructured grid (left)
b) Plate computational grids - left = adapted unstructured 25 million cell, centre = structured 56 million cell, right = structured 7 million cell

NUMERICAL SCHEME

Two codes were employed in this study: FINE/Turbo and FINE/Hexa of Numeca (referred to hereafter as F/T and F/H respectively). Both solve the compressible form of the Reynolds averaged Navier-Stokes equations, and the Spalart-Allmaras one equation turbulence closure [4] was chosen. F/T and F/H employ a cell-centred finite-volume density based scheme modelled after the 4-stage Runge-Kutta time integration method of Jameson, Schmidt and Turkel [3] - commonly referred to as JST. While F/T conducts solutions on structured multi-block grids, F/H employs an unstructured hexahedral grid generated via octree-refinement. Agglomeration-type multigrid convergence acceleration is employed in both codes (in F/T 8 adjacent cells are agglomerated to one preserving the structured grid index system, and in F/H agglomeration is unstructured). F/H also offers solution-adaptive grid refinement. Further specifics of the F/T solver algorithm were supplied by Hakimi [5], previous CFD validation of F/T specific to SWBLI was given by Grosvenor [6] and details of F/H were provided by Patel [7] and Delanaye *et al.* [8].

NUMERICAL MODEL

Figure 2 displays views of both the structured multi-block and adapted unstructured computational grids employed for the $\Delta z/D = 3$ test configuration. Inflow velocity and static pressure and temperature were set to reproduce the experimental conditions of Mach 4 flow at a total pressure and temperature of $1.074\text{x}10^6$ Pa and 291 K respectively. Half of the physical domain was modelled by employing a zero-gradient Neumann boundary condition at the centreline surface. The opposite surface in the *z* direction was extended to the tunnel wall location using a coarser grid resolution. A slip boundary condition was set between the plate edge and tunnel wall, and the body and plate surfaces were set to no-slip. The remaining boundaries were set to supersonic outflow conditions where flow is assumed to be supersonic and extrapolated from the interior of the domain. Coarser versions of the structured grid were run, and sub-blocks were constructed and individually refined to reach a reasonably grid-independent solution (i.e., block-enrichment was conducted to maximize resolution of the relevant phenomena) - judged in terms of shock structures and limiting surface streamline patterns on the flat plate. Grid refinement was particularly employed in the blocks between the body and plate, and at the body's base - while coarser resolution was set above the body. Simultaneously, the first grid nodes from the plate and body solid surfaces were set such that average y_1^+ values did not exceed unity. The total number of cells in this final grid was 56 million - which might be considered a very high number for such a seemingly simple geometric configuration. For comparison, two coarser structured grids were run at 7 and 1 million grid cells. Details of the two finer structured grids are listed in Table 1, where # b.l. indicates number of cells in the plate boundary layer just upstream of the point of bow shock impingement. Computations were also performed using F/H employing solution-adaptive grid refinement, which lead to a grid resolution consisting of 25 million cells resolved to the 3D shock wave structures. As might be expected, solution-adaptive point-enrichment type grid refinement optimises the placement of grids cells to reduce the overall required grid size to adequately capture the highly three-dimensional concentrated SWBLI phenomena.

Table 1 Structured grid details

# total cells	y_1^+	# b.l.	# cells across shocks
$56\text{x}10^6$	0.15-1.2	40	3-6
$7\text{x}10^6$	0.25-2.5	20	1-3

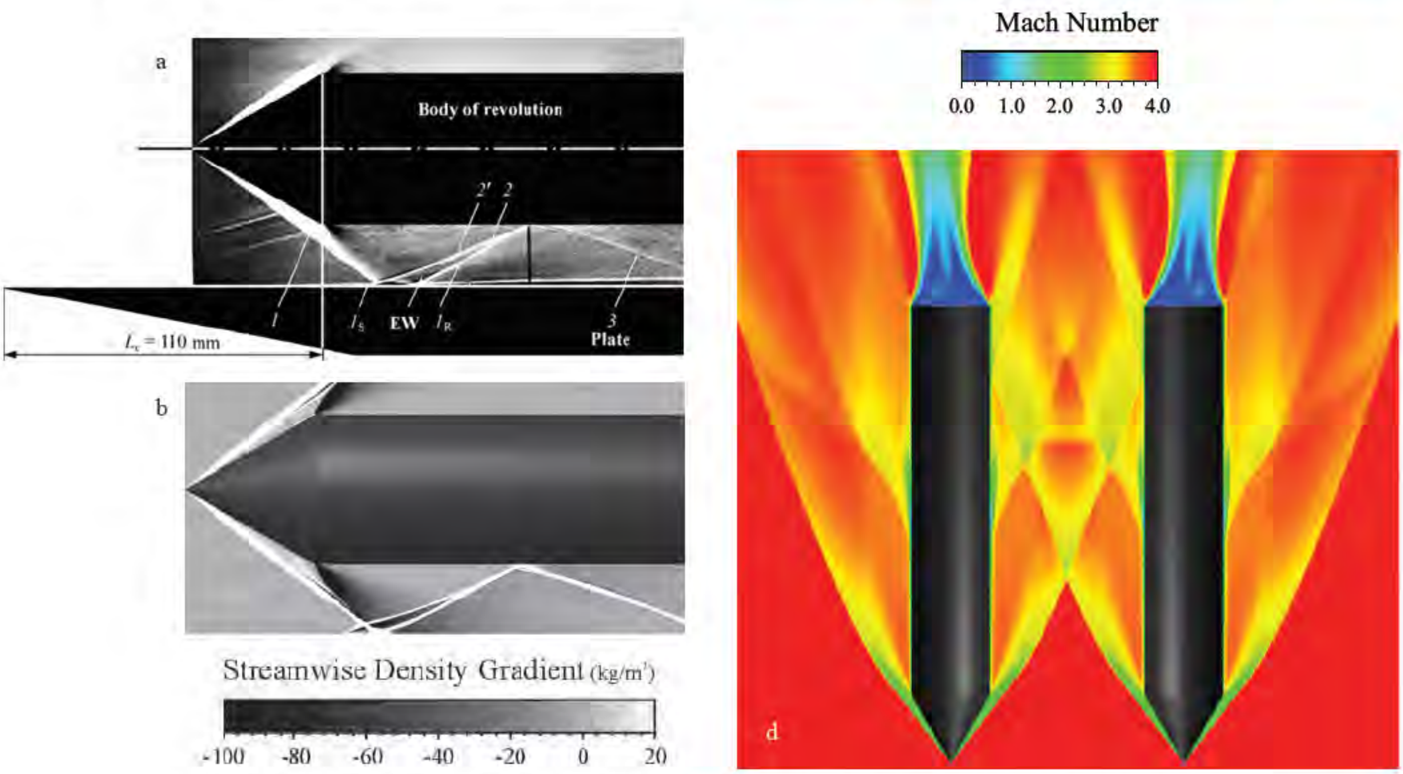


Figure 3. Views of flow field at $\Delta z/D = 3$: a) schlieren photograph taken over vertical plane (z-plane) at body centreline, b) predicted shock wave structure displayed in terms of streamwise density gradient, c) predicted Mach number contours over z-plane and d) a plane extending in cross-stream direction (y-plane) along body centreline

DISCUSSION OF PREDICTIONS AND COMPARISON OF MEASUREMENTS WITH CFD

Predictions of F/T and F/H follow, primarily employing the 2nd order central difference spatial discretisation JST scheme, and the Spalart-Allmaras one-equation turbulence closure. The 2nd order Roe Flux-Difference Splitting (FDS) upwind scheme [9] and Symmetric Total Variation Diminishing (STVD) scheme of Yee [10] in conjunction with the van Leer and superbee flux limiters were also selectively tested, and these results will be discussed additionally in the next section. Figure 3a displays a schlieren photograph indicating the shock wave structure over a vertical plane at the body centreline. In Figure 3b, the predicted streamwise density gradient for the same configuration is provided for comparison. One can observe that the results compare well with experiment in terms of shock angles and presence of both separation shock wave (1_S) and its continuation (2') above the conical bow shock (1), and terminal shock (1_R) from the reattachment region. Upon merging, shocks 2' and 1_R form a single shock wave 2, followed by the next downstream reflected shock (3). Figures 3c,d display the shock wave structure produced by the bodies spaced at $\Delta z/D$ = 3 in terms of predicted Mach number contours. Over the z-plane (Fig. 3c) one can observe the shock structure induced by the forebody and arising separation zone in the vicinity of the bow shock interaction with the boundary layer on the plate surface as well as subsequent shock reflections between the body and plate. As seen from the calculations, additional downstream reflected shocks (4,5) appear between the body and plate downstream of shock 3. The y-plane view (Fig. 3d) shows the shock structure between the bodies and the base separation zones downstream of them.

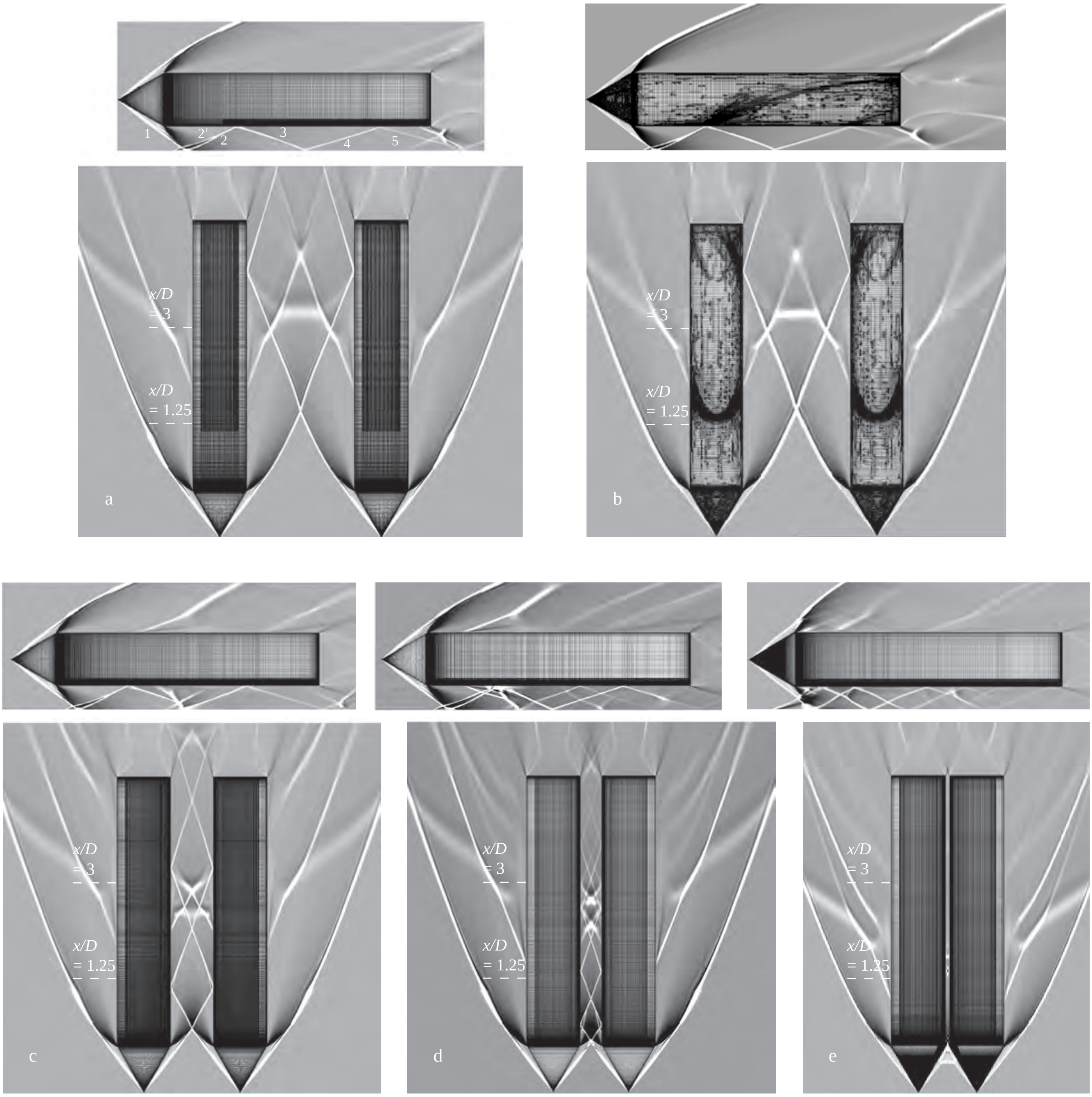


Figure 4. Cutting planes in three coordinates displaying predicted streamwise density gradient contours for x-plane & y-plane views taken at body centreline: a) $\Delta z/D$ = 3.0, structured 56 million cell; b) $\Delta z/D$ = 3.0, unstructured adapted 25 million cell; structured 56 million cell: c) $\Delta z/D$ = 1.8, d) $\Delta z/D$ = 1.4, e) $\Delta z/D$ = 1.06

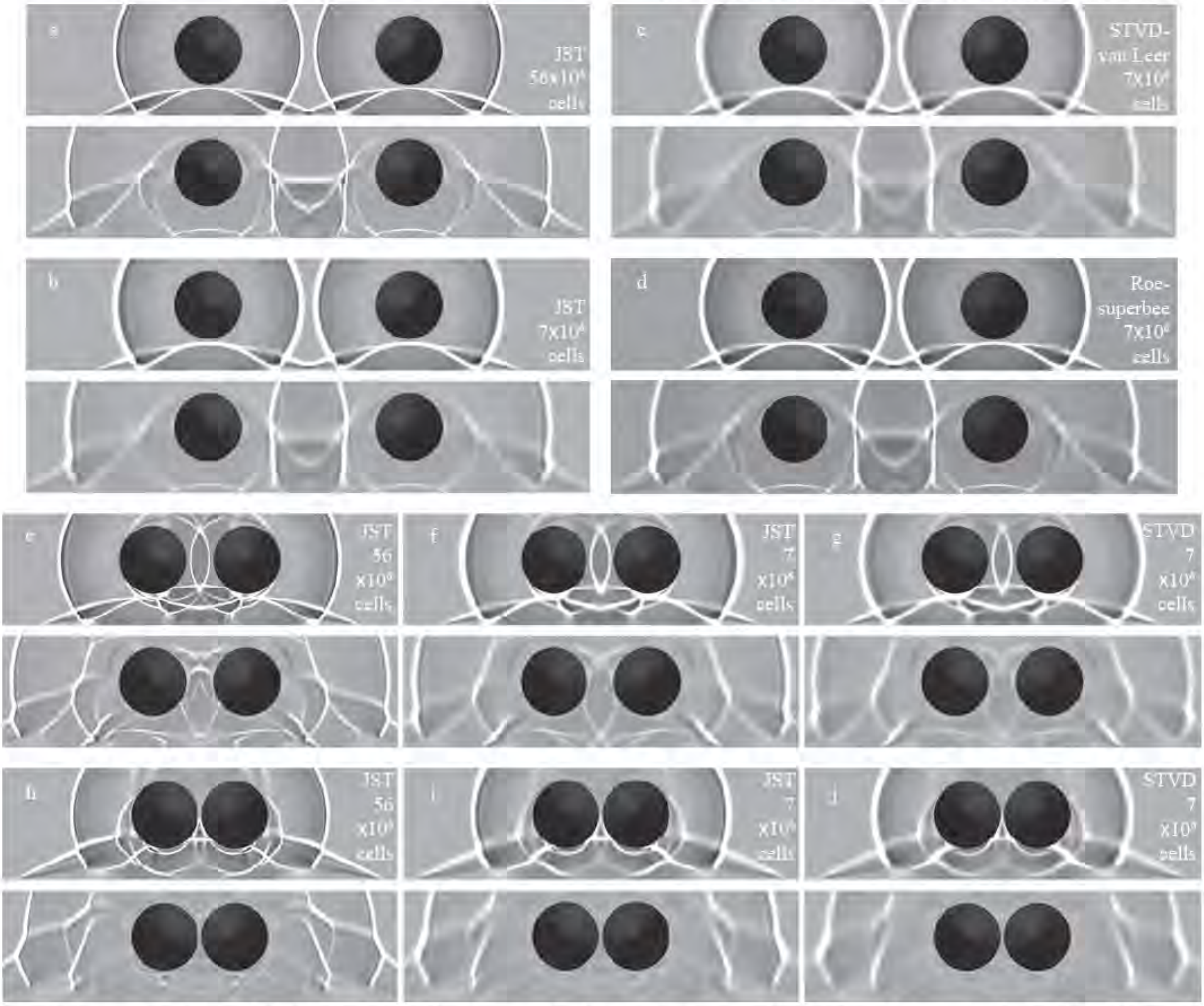


Figure 5. Cross-sectional views of shock system indicated by streamwise density gradient contours at x/D = 1.25 (upper figures a-j), and x/D = 3 (lower figures a-j) (a-d) $\Delta z/D$ = 3, (e-g) $\Delta z/D$ = 1.4, (h-j) $\Delta z/D$ = 1.06; (c,g,j) 2nd Order Upwind STVD-van Leer; (d) 2nd Order Upwind Roe-superbee; remaining (a,b,d,e,f,h,i) JST

The evolution of complex crossing shock patterns for the four configurations considered in this study with varying inter-body distance decreasing from $\Delta z/D$ = 3 to 1.06 is demonstrated in Figures 4 and 5a,e,h in accordance with the CFD predictions. The conical bow shocks induced by the forebodies are shown to merge and reflect in a complex 3D manner between the bodies and on the plate. Their structure differs markedly from the earlier predictions [2] that employed the Euler equations, due to the emergence of 3D secondary flows and separation zones in the boundary layer on the plate and bodies' surfaces described in detail earlier in accordance with the experiments [1,2]. Emerging separation and attachment zones and their interaction with a basic 'inviscid' shock wave structure change the flowfield significantly (further discussion of Fig. 5b,c,d,f,g,i,j is given in the next section in the context of comparing shock capturing strategies). As seen in Figure 6, the computations predict different stages of crossing bow shock interaction in the external flow between the bodies with decreasing distance $\Delta z/D$. The regular crossing shock structure at $\Delta z/D$=1.8 is followed by the appearance of a small Mach stem as the inter-body distance is reduced to $\Delta z/D$=1.4, and a distinct Mach stem is observed at $\Delta z/D$=1.06. This trend and the specific structures are predicted well in comparison with experiment.

One can observe a progressional shift in the complex 3D viscous/inviscid interaction of the shock waves and turbulent boundary layers as the distance between bodies, $\Delta z/D$, decreases (Figs. 4, 5). Strong adverse pressure gradient regions are subsequently induced at the plate surface, as shown in Figure 7. The experimental measurements are depicted on the left side of the figure (decreasing $\Delta z/D$ a-d), showing the two main pressure maximums (*A* & *B*) which correspond to attachment nodes (N_1 & N_2) under the bodies depicted in the limiting surface streamline plots in Figure 8 (left side of figure, decreasing $\Delta z/D$ a-d). The central peak (*C*) appears in the central zone (N_0) developing the central separation zone between the bodies, as described further in [1]. The experimentally indicated topology of surface streamlines with various marked singular points has been traced from oil flow visualisation and convergence (or separation) and divergence (or reattachment) lines are labelled as *S* and *R* respectively. The predicted limiting streamlines from CFD have been generated by computing velocity vector stream-traces projected to the planar cuts taken at the location of the first layer of grid nodes off the body and plate surfaces. Predicted static pressure coefficient distributions on the flat plate for each of the four configurations (Fig. 7, right) are compared with measurements in terms of pressure coefficient surfaces. The structured grid CFD prediction is seen to reproduce the pressure field quite well in terms of magnitudes and trends. Respectively, the computed surface flow patterns (Figs. 8 & 9) tend to follow experiment well. For instance, separation lines S_1, S_3 and S_5 on the plate at $\Delta z/D$ = 3.0 (Figs. 8a & 9) and their associated reattachment lines, are shown to compare well with experiment. These three separation lines can be seen to occur

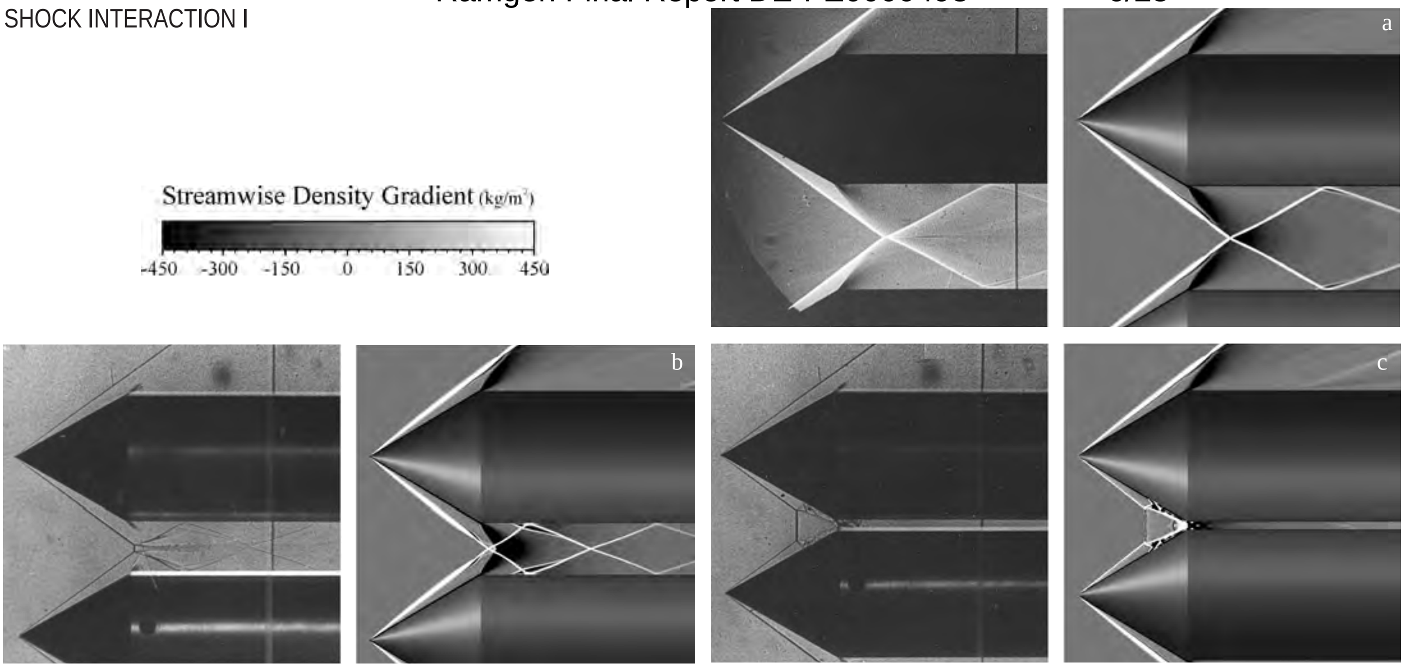


Figure 6. Comparison of experimental schlieren photographs (left figures) with predicted crossing shock wave structure (right figures) between the bodies: a – regular interaction of crossing bow shocks ($\Delta z/D$ = 1.8); b – incipience of small Mach stem ($\Delta z/D$ = 1.4); c – flow stage with distinct Mach stem ($\Delta z/D$ = 1.06). Note: legend applies to CFD figures

consistently with the first bow shock, and reflected second and third shock waves directed from the body to the plate intersecting the plate surface from left to right (see Fig. 3a,b). Separation line S_B is also well captured. It is perhaps one of the most complex separation locations, as it is due to essentially three dimensional interaction of crossing shock waves separating the plate boundary layer. One perhaps more complex location is the convergence region S_5 in the vicinity of the base flow, and this is also seen to be captured quite well. Conversely, secondary convergence line S_2 and divergence line R_2 are shown to be underpredicted in terms of definition and streamline angle. The underprediction here is largely due to the fact that grid resolution has been concentrated in the zones between bodies, and lower resolution has been chosen on the outside. Note that one can see a better resolved set of separation and reattachment lines S_3 and R_3 from the unstructured solution-adapted grid computation in Figure 9a. This zone exists in the region of x/D = 3, where it is shown in Figure 2 that the adapted grid employs the highest local resolution. One can observe a similar prediction accuracy demonstrated for each of the four configurations, and the impact of coarsening the grid is depicted in Figure 9 for $\Delta z/D$ = 3.0, which displays an expected trend of reducing definition as the grid is coarsened due to artificial diffusion. Note that from the experimentally indicated streamlines there appears to be some potential for flow asymmetry at $\Delta z/D$ = 1.8 (Fig. 8b). This may offer at least part of the explanation for the lack of focus nodes predicted for this case. One can see the predicted streamlines curling up in this region, but focus nodes are not present. In the experiment they seem to not be completely symmetric in size. The assumption of symmetry imposed on the computations obviously prevents the prediction of such phenomena. Another potential reason for disagreement is that the turbulence closure employed is predicting higher levels of turbulence locally than what is realistic. The asymmetry could potentially arise as a time-varying phenomena, which would further explain differences in prediction. An alternative explanation could be some geometric asymmetry of the test model caused by small deformations [1] which would not have been included in the numerical model. In general the separated flow patterns over both the bodies and plate for all cases are reproduced well in the predictions, and the impact of coarser grid resolution (e.g., the 1 and 7 million cell grids in Fig. 9) can be seen to be a loss of resolution of the key separating zones at $\Delta z/D$ = 3.0. In accordance with the Figures 7 and 8, the cardinal reconstruction of separated flow on plate surface occurs with decreasing the distance between the bodies to the minimal value $\Delta z/D$ = 1.06 at which forming the Mach stem (see Figs. 4e, 6c) indicates an 'unstart' phenomenon in the limited space between the bodies and plate.

Predicted static pressure coefficient distributions on the flat plate are compared quantitatively with experimental data for $\Delta z/D$ = 3.0 and $\Delta z/D$ = 1.4 in Figure 10. The structured grid CFD simulation is seen to reproduce the pressure field quite well in terms of magnitudes and trends. The largest differences appear to be in the vicinity of reflected shock 3, where the pressure coefficient predictions show a subsequent pressure rise slightly upstream of the true position. This is the previously mentioned region surrounding x/D = 3, and one can conclude from the comparison of finest structured grid and unstructured adapted grid that the difference between prediction and experiment is not due to grid resolution. It is expected that this is a region where the impact of different turbulence closure options would be of most interest. In addition to the highly three dimensional, non-equilibrium state of turbulence expected to be present here, the expansion downstream of the forebody may also be accelerating the flow such that local relaminarisation and subsequent transition occurs. The impact of higher levels of turbulence closure sophistication is being tested in a next phase of study.

As seen in Figures 4 and 5, a complex system of shock waves forming around the bodies interact also with the boundary layer on their surfaces. The computed surface flow pattern for the configurations shown tends to follow experiment well (Fig. 11). For instance at $\Delta z/D$ = 3.0 the number of the separation and attachment lines S_2, R_2, S_4, R_4 correspond to the influence of reflected shocks 2 and 4 from the plate to body (see, additionally, Fig. 3a,c), which penetrate and diffract around the bodies. The separation line S_1 arises from the conical bow shock wave penetrating from the second body to the surface of the first. Lines S_{2s} and S_{3s} indicate secondary separations. In accordance with experiment and computations, decreasing $\Delta z/D$ leads to a significant rise of separation zones on the body surface and in conditions of the 'unstart' phenomenon at $\Delta z/D$ = 1.06, they penetrate the surfaces of the conical forebodies. Note that local grid resolution between the forebodies was increased in this specific configuration to better capture the unstarted shock system.

Figure 12 compares predicted body force coefficients to balance measurements. One can see a significant increase in lift and lateral forces as the inter-body distance is decreased from $\Delta z/D$ = 3.0 down to $\Delta z/D$ = 1.06. Drag force (wave drag together with surface friction drag) is almost

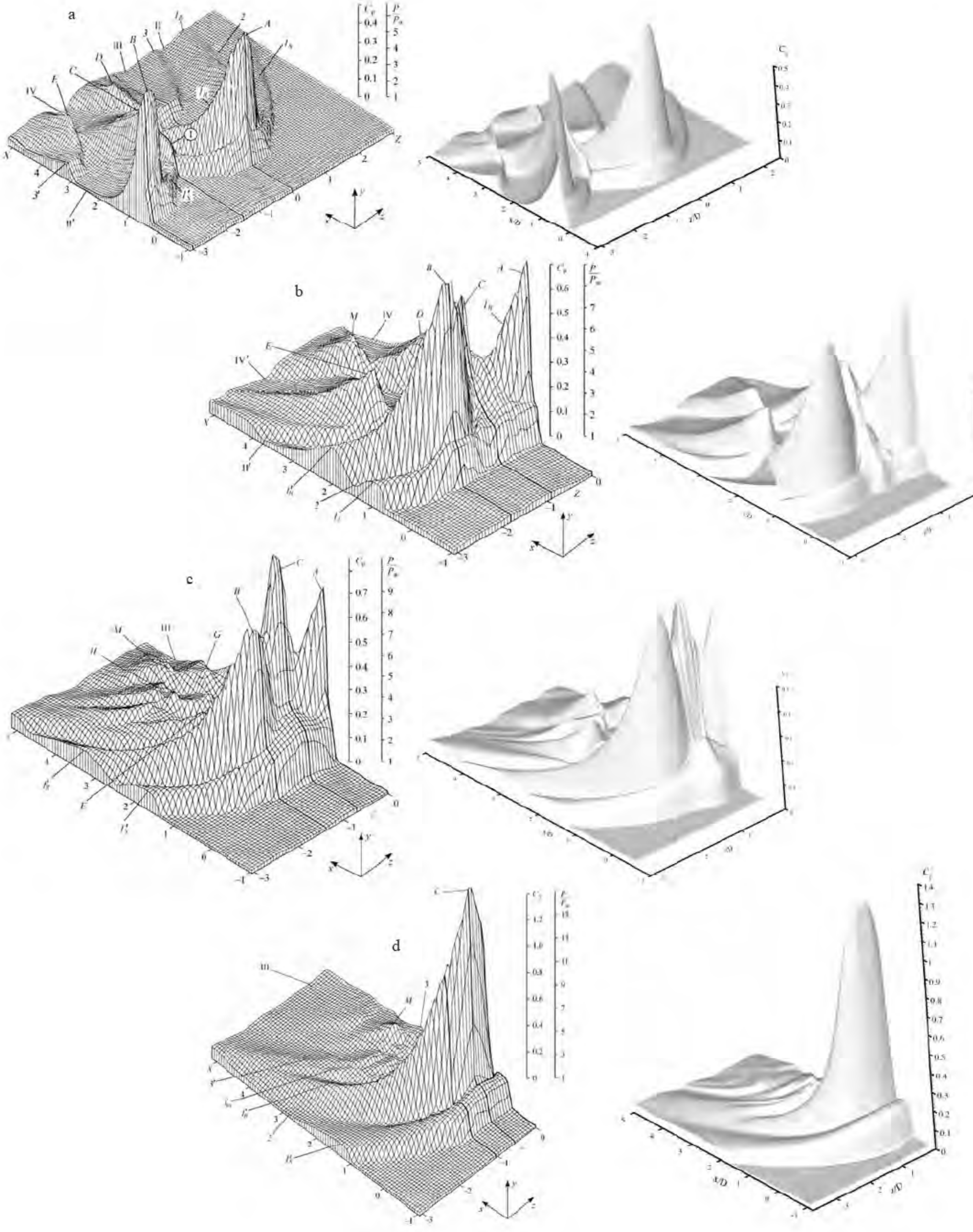


Figure 7. Axonometric views of plate static pressure coefficient surfaces comparing experimental measurements (left) with predictions (right)
a - $\Delta z/D$ = 3.0, b - $\Delta z/D$ = 1.8, c - $\Delta z/D$ = 1.4, d - $\Delta z/D$ = 1.06

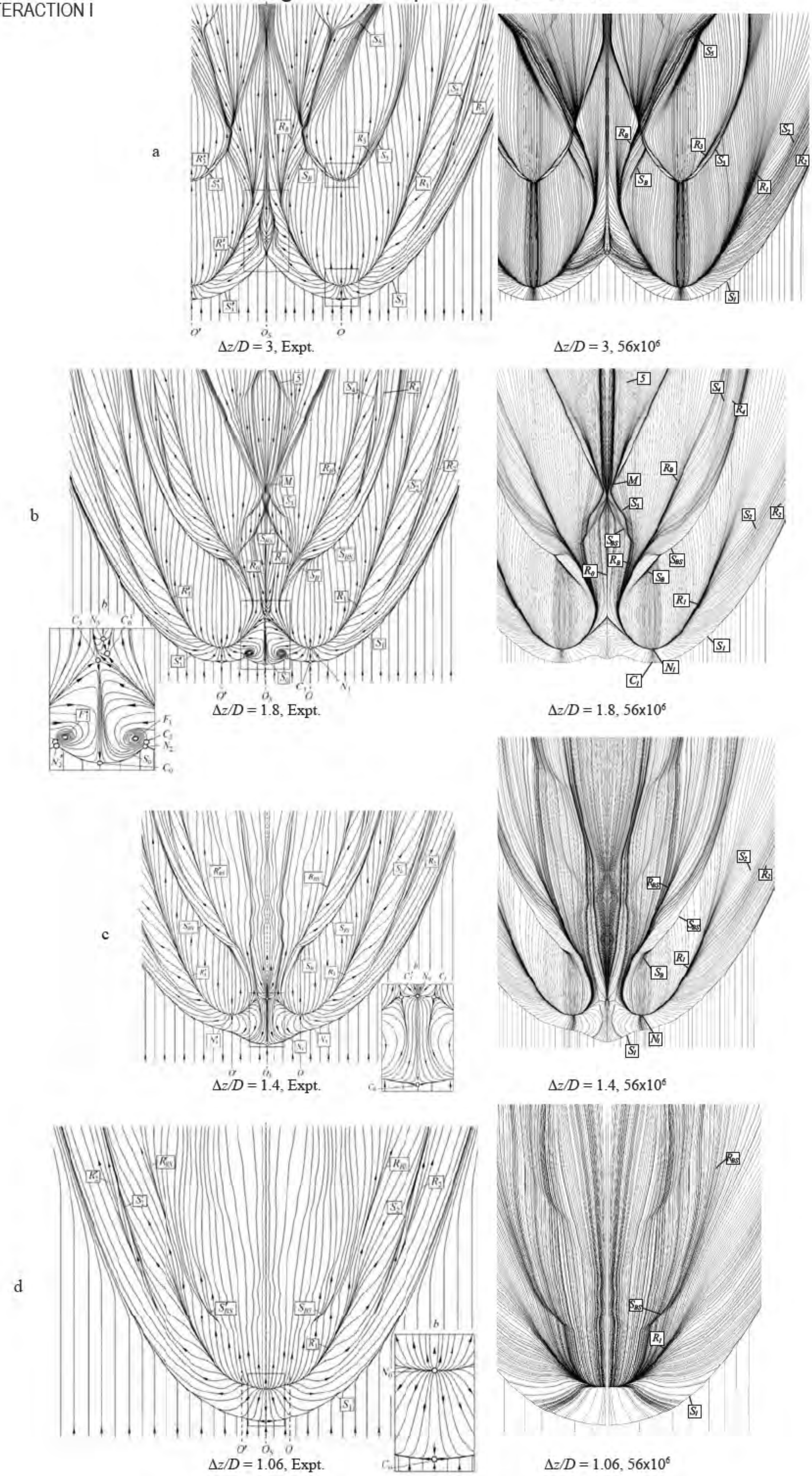


Figure 8. Predicted plate surface streamlines compared to experimental oil flow visualisation

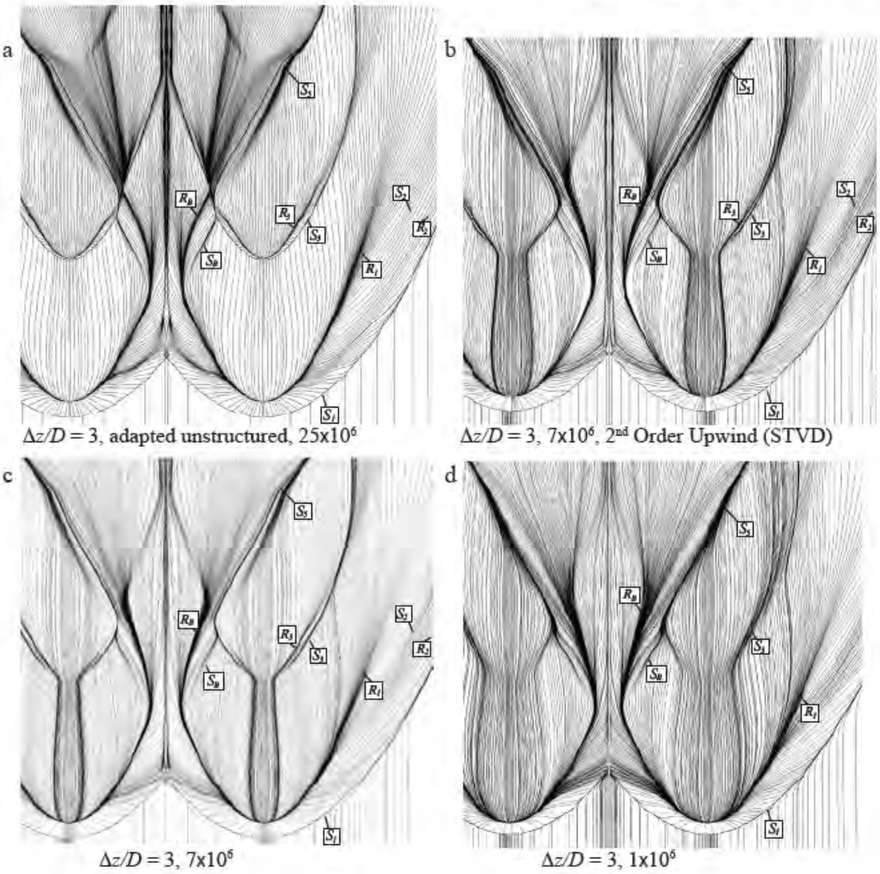


Figure 9. Predicted plate surface streamlines for $\Delta z/D = 3$ predicted with different computational grid topologies and resolutions

constant between $\Delta z/D$ = 1.4 - 3.0, and then suddenly increases by a significant margin as the distance between bodies is closed down to $\Delta z/D$ = 1.06. From the shock wave structures seen in the density gradient contours of Figure 4e and 6c and plate and body surface streamlines (Figures 8 and 11), the difference appears to be due to the shock system existing between the bodies and plate having coalesced into one stronger shock in the region between bodies and plate (similar to so-called 'unstart' phenomena as discussed above), thereby producing an increase in wave drag. Interestingly, the computations tend to predict the experimental lateral and lift forces well while there is a visible overprediction of drag force from $\Delta z/D$ = 1.4 - 3.0. The magnitudes of difference between computation and experiment are somewhat higher than the indicated experimental error. This difference is potentially due to the level of turbulence closure sophistication employed here for predicting surface friction force. Each of the configurations predicted in general have shown reasonable agreement between numerical simulation and experiment, so it must be acknowledged that the chosen scheme of applying JST with the Spalart-Allmaras turbulence model is quite effective in predicting this sort of complex 3D SWBLI flow. This observation is consistent with prior validation work that was performed [6].

SHOCK CAPTURING OPTIONS

The study presented herein concentrated on the assumption of steady, fully turbulent flow and predictions mostly employed the JST scheme and Spalart-Allmaras turbulence model. This prediction method has been shown to reproduce a range of subsonic and supersonic complex flows well [6] and here the correct 3D flow patterns and shock strengths were predicted, although exact shock positions, pressure and drag magnitudes exhibited some differences from the experimentally determined values.

Considering the results of the present study, it is interesting to look back on earlier work from over two decades ago such as the review of Woodward & Colella [11] who spoke of a set of shock capturing validation studies they considered at the time: "A problem of this nature in two dimensions is presently completely out of the question, as convergence for the most accurate scheme considered here would require a grid of a million zones. The one-dimensional test problem is useful in showing the performance of the schemes under extreme conditions not soon to be encountered in practical two dimensional calculations." Now that a solution based on grids containing 1 million - or even 100 million cells can be achieved, we can evaluate performance of numerical schemes and turbulence models practically in three dimensional calculations.

For the present calculations, convergence robustness was observed to be significantly higher for the JST scheme compared to the upwind schemes tested. It is worth reviewing some similarities and differences between the schemes. Venkateswaran and Merkle [12] stated: "Artificial dissipation is essential in computational fluid dynamics (CFD) algorithms in order to eliminate the high wavenumber modes in the solution. Artificial dissipation models may be broadly classified into two families. The first family is associated with central-differenced schemes (Jameson, Schmidt and Turkel), wherein the dissipation is added as a conscious, explicit step through the introduction of additional higher-order derivative terms. The second family is associated with upwind schemes, where the dissipation is an inherent part of the spatial discretisation." Practical issues in applying such schemes to highly three dimensional flows such as the one studied herein were pointed out by van Leer [13] where he stated that the Roe scheme should theoretically require only one grid cell across a shock wave to capture it, but that this was only true in practice when the shocks were aligned to the grid: "It can be shown that the upwind flux formula based on Roe's approximate Riemann solver yields a steady normal-shock structure (if aligned with the grid) that contains at most one internal cell. This property is lost for shocks oblique to the grid, which

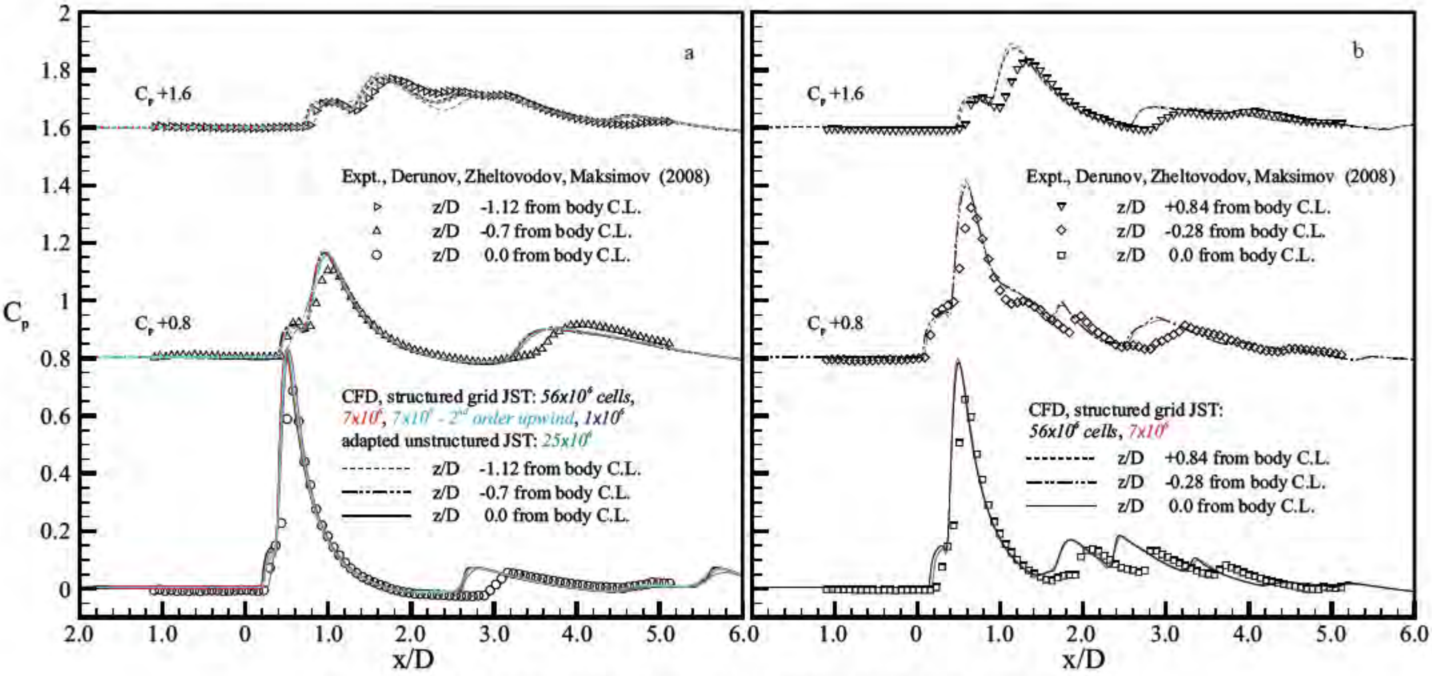


Figure 10. Predicted plate static pressure coefficients compared to experimental measurements
a) $\Delta z/D$ = 3: 7 million grid cell (red), 56 million grid cell (black) structured grids, 25 million cell adapted unstructured grid (green)
b) $\Delta z/D$ = 1.4: 7 million grid cell (red), 56 million grid cell (black) structured grids

serves as a motivation for the search of truly multi-D upwind methods..." In Figure 5 discussed earlier in the context of shock interactions arising between the bodies and plate, predictions of shock wave induced density gradient variation are given for both the JST 2nd order central scheme, and the 2nd order upwind schemes of Yee (STVD) and Roe (FDS) - employing two well know flux limiters (van Leer, and Roe's superbee). The finest (56 million cell) grid provided a range of 3-6 cells across the shock thickness in this highly three-dimensional supersonic flow. The next coarsest (7 million cell) grid placed 1-3 cells across the same shock thicknesses, so this grid resolution was chosen to compare the two schemes and determine whether the 2nd order upwind schemes would produce similar shock capture to JST on the finest grid. One can see no improvement in shock capture by the STVD scheme, and while minor improvements are visible with the Roe scheme employing the superbee limiter, the theoretical shock capture over one cell is not demonstrated when compared to the finest grid JST result. The earlier mentioned drawback of these schemes requiring shocks to be aligned with the grid is thus highlighted. In such a complex flow as the one considered, it is not practical to generate a grid aligned to each of the shocks and hence the difference in prediction between the 2nd order upwind schemes and the JST scheme should be expected to reduce compared to what might be seen in a primarily two-dimensional normal shock flow, for instance. Discussing the prediction of shock waves using the JST scheme, Jameson [14] stated "The JST scheme with scalar diffusive flux captures shock waves with about 3 interior points, and it has been widely used for transonic flow calculations because it is both robust and computationally inexpensive." Our findings have been consistent with this last point, from comparison of computational expense of running JST versus both the Roe FDS and Yee STVD schemes. Attaining convergence was much more straight forward with JST, and for instance the upwind schemes in this study needed to be initiated from a prior solution - making them less desirable in the context of design-cycle analysis. For all schemes tested, shock capture was not adequately demonstrated without a minimum of three grid cells across the shocks. Convergence stability of JST was observed to be significantly closer to monotone than the 2nd order upwind schemes tested. Jameson [14] discussed this problem and pointed to recent efforts to develop practical multidimensional upwind schemes such that "the upwind biasing is determined by properties of the flow rather than the mesh...preliminary results indicate the possibility of achieving high resolution of shocks and contact discontinuities which are not aligned with mesh lines." Thus both van Leer [13] and Jameson [14] pointed to this need for advanced multidimensional shock capturing schemes, and it appears that the present test case can be used as an example of an application where such schemes would be valuable. Jameson pointed to such work as Paillere and Deconinck [15] and van Leer additionally to these authors, highlighted the more recent work of Roe and others on so-called Residual-Distribution Schemes to achieve high order multi-dimensional shock capturing, stating that the "techniques are starting to look more and more like Discontinuous-Galerkin methods."

In the present study we have endeavoured to ensure accurate shock capture by employing very high grid resolution through block-enrichment that was achieved in a manual and iterative process using F/T, as well as through an automated point-enrichment (or really cell-enrichment) process in F/H. The next studies should now concentrate on the impact of transition prediction, higher order turbulence closure, and the potential unsteadiness of some configurations.

CONCLUSIONS

A set of RANS computations have been carried out for the complex three-dimensional case of two cylindrical bodies in Mach 4 flow, mounted over a flat plate at varying inter-body distances. The JST scheme with turbulence closure provided by the Spalart-Allmaras model was employed, and alternative 2nd order upwind schemes were also tested. It was found that when the different spatial discretisation schemes were tested on the same grid, affording a range of 1-3 grid cells across the shocks, that there were only slight differences in shock resolution. This was deemed likely to be due to the non-aligned nature of the highly three-dimensional shock system to the computational grid. Block-enrichment and point-enrichment type approaches were both employed to minimize grid sensitivity of the predictions. The JST scheme and S-A model produced

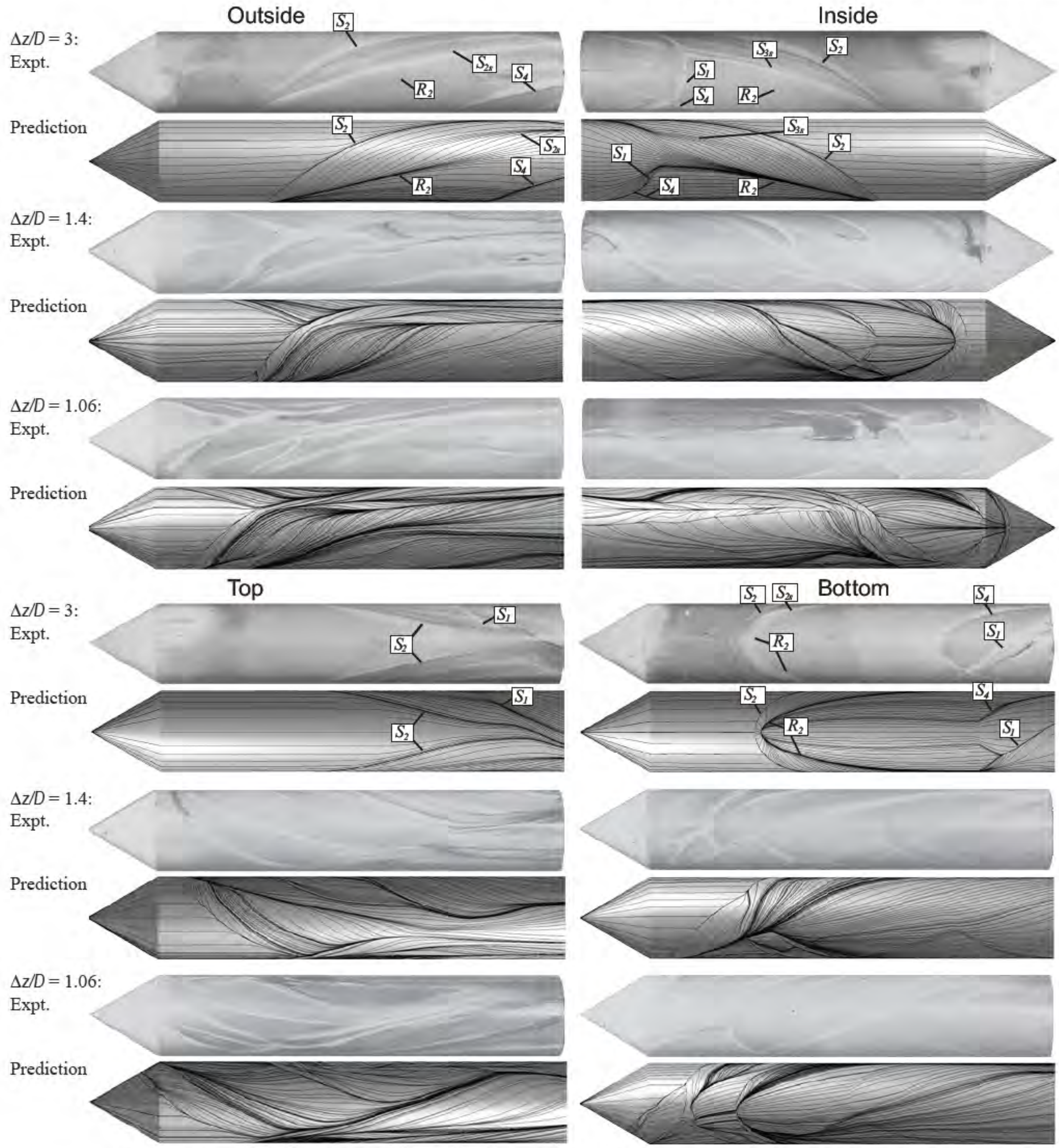


Figure 11. Comparison of predicted ($56x10^6$ cells) limiting surface streamlines with surface oil flow visualisation on body 1

surface flow separation topology and flowfield structure, static pressure coefficient distributions, and aerodynamic force coefficient predictions that compared well with the experimental values and trends. The large jump in body forces as the inter-body distance was decreased from $\Delta z/D$ = 1.4 to 1.06 was reproduced in the predictions. The chosen numerical scheme and turbulence model were demonstrated to provide acceptable prediction of complex 3D turbulent shock wave / boundary layer interaction. Nevertheless, some differences in predicted positions of separation zones in the vicinity of secondary (reflected) shock waves between the body and plate existed, which will represent the focus of the next stage of research. Grid convergence and shock capturing were shown to be well achieved in the study, and next studies will concentrate on better capturing the turbulence field through the employment of higher order closure, and investigation of potentially laminar/transitional zones, in addition to the potential for unsteadiness.

ACKNOWLEDGMENTS

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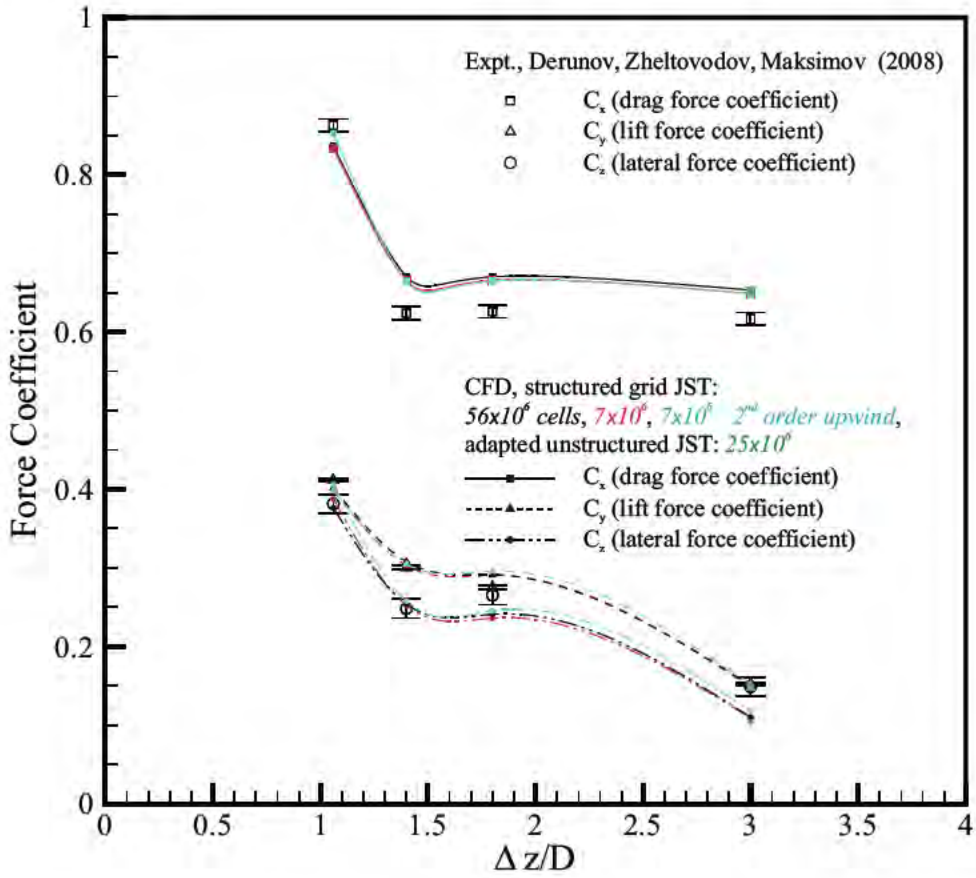

Figure 12. Comparison of predicted body force coefficients with balance measurements at $\Delta z/D$ = 1.06, 1.4, 1.8 and 3.0

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Appendix 6.1

HP CO_2 ROTOR FDR

Final Design Review

Gen-2 Inducer Assembly (Rev2)

Dec 7th, 2011

Agenda

Agenda

- **Intro (Dave)**
- **Budget & Schedule (Dave)**
- **Aero (Silvano)**
- **Mechanical (Dave)**
  - **Inducer Blade Mechanical**
  - **Rotor Mechanical**

System Definition and Scope

- **Rotating component providing the required inlet flow conditions to the static diffuser**
  - **Hence the change from Blade 7 to Blade 12**

- **The Inducer comprises 1 Blade row, disk / shafting and blade retention components.**

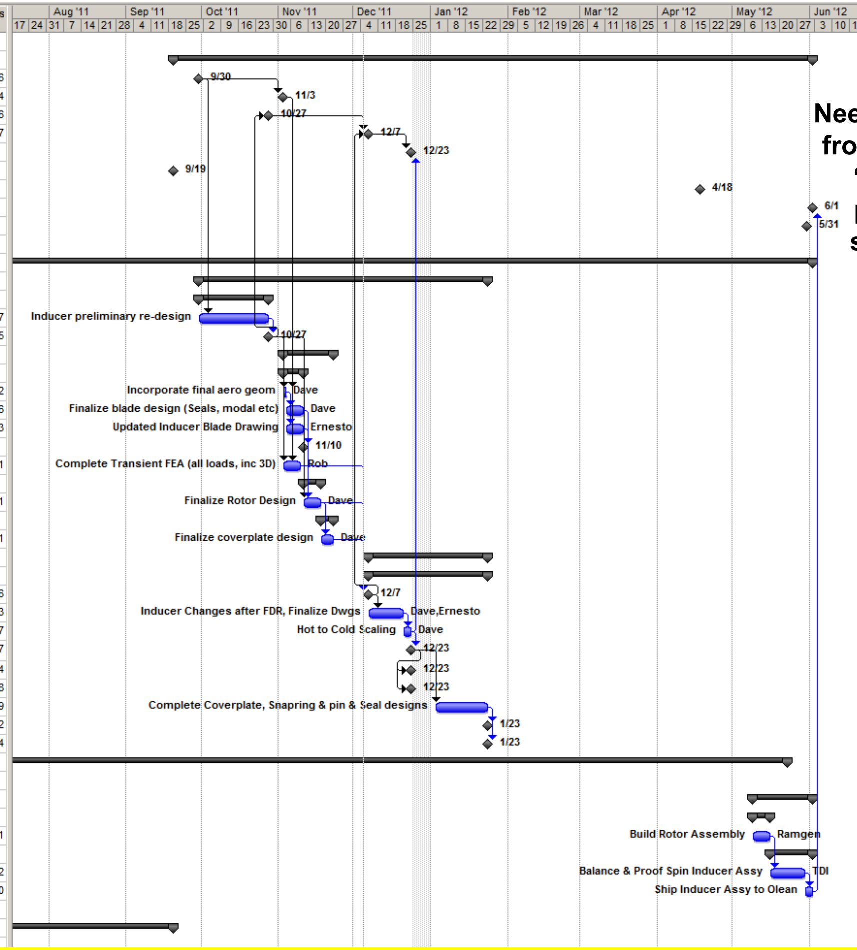
- **The Inducer Blade includes an integrated shroud. The Shroud interfaces via several seals with the static structure. The shaft is generally defined by driven and non driven ends. The shaft portion of the Inducer interfaces with several seals, bearings, thrust collars, and a coupling on the driven end.**

Schedule Overview

- **Current Inducer Blade & Rotor PRR & Drawing release date : Dec 23rd**
  - **Was Dec 6th so has slipped back just under 3 weeks with the move to Case 15 & Blade 12**
- **Rotor Assy available for installation into Rig: June 1st 2012**
  - **Was May 17th and has slipped 3 weeks**
  - **Blade is critical path**
    - **Turbocam (Europe) estimate aligns (just) with schedule**
    - **Need to get final design for firm quote**
- **Optimal program schedule from Sept shows Rotor Assy required April 18th**
  - **Adding 3 weeks for the Case15 / Blade12 change gives Rotor Assy required May 9th**
  - **Available June 1st so disconnect of 3 weeks still present**
    - **Will endeavor to release blade 1 week early**
    - **Need to recover 2 weeks from manufacturing schedule**
- **Question: What is the real need date for the rotor ???**
- **Detailed schedule follows**

Schedule – Inducer Design

		Task Name	Duration	Start	Finish	Predecessors	Successors
1							
2		**Key Dates**	**178 days**	**Mon 9/19/11**	**Fri 6/1/12**		
3		Inducer Gen-2 Rev-2 Prelim Aero Definition (from 9/30 schedule in 0900 folder)	0 days	Fri 9/30/11	Fri 9/30/11		+15 days,16
4		Inducer Gen-2 Rev-2Final Aero Definition (from 9/30 schedule in 0900 folder)	0 days	Thu 11/3/11	Thu 11/3/11	3FS+15 days	20,24
5		Inducer Rev-2 PDR	0 days	Thu 10/27/11	Thu 10/27/11	17	6
6		Inducer Rev-2 FDR	0 days	Wed 12/7/11	Wed 12/7/11	31,5	7
7		Inducer Rev-2 PRR	0 days	Fri 12/23/11	Fri 12/23/11	6,33	
8		Bald Rotor available	0 days	Mon 9/19/11	Mon 9/19/11	102	
9		Rotor Assy Required for install in Rig	0 days	Wed 4/18/12	Wed 4/18/12		
10		**Rotor Assy Available (Bladed Disk)**	**0 days**	**Fri 6/1/12**	**Fri 6/1/12**	**92**	
11		Gen-2 Build 1 Testing Starts (Optimal schedule)	0 days	Thu 5/31/12	Thu 5/31/12		
12							
13		**Inducer Assembly - Bladed Disk Configuration**	**262 days**	**Wed 5/25/11**	**Fri 6/1/12**		
14		**Inducer Assembly Design**	**76 days**	**Fri 9/30/11**	**Mon 1/23/12**		
15		**Inducer Assembly Preliminary Design**	**20 days**	**Fri 9/30/11**	**Thu 10/27/11**		
16		Inducer preliminary re-design	4 wks	Fri 9/30/11	Thu 10/27/11	3	17
17		Inducer PDR	0 days	Thu 10/27/11	Thu 10/27/11	16	20,24,26,5
18		**Inducer Assembly Detailed design**	**14 days**	**Thu 11/3/11**	**Tue 11/22/11**		
19		**Blade Detailed Design**	**6 days**	**Thu 11/3/11**	**Thu 11/10/11**		
20		Incorporate final aero geom	1 day	Thu 11/3/11	Thu 11/3/11	17,4	21,22
21		Finalize blade design (Seals, modal etc)	1 wk	Fri 11/4/11	Thu 11/10/11	20	26
22		Updated Inducer Blade Drawing	1 wk	Fri 11/4/11	Thu 11/10/11	20	23
23		Release Revised (Prelim) Inducer Blade Drawings -> vendor	0 days	Thu 11/10/11	Thu 11/10/11	22	
24		Complete Transient FEA (all loads, inc 3D)	1 wk	Thu 11/3/11	Wed 11/9/11	17,4	31
25		**Rotor (Disk / Shaft) Detailed Design**	**5 days**	**Fri 11/11/11**	**Thu 11/17/11**		
26		Finalize Rotor Design	1 wk	Fri 11/11/11	Thu 11/17/11	17,21	28,31
27		**Coverplate & Snap Ring detailed Design**	**3 days**	**Fri 11/18/11**	**Tue 11/22/11**		
28		Finalize coverplate design	3 days	Fri 11/18/11	Tue 11/22/11	26	31
29		**Inducer Assy Completion Activities**	**28 days**	**Wed 12/7/11**	**Mon 1/23/12**		
30		**Inducer FDR & PRR**	**28 days**	**Wed 12/7/11**	**Mon 1/23/12**		
31		Inducer Assy FDR	0 days	Wed 12/7/11	Wed 12/7/11	28,26,24	32,6
32		Inducer Changes after FDR, Finalize Dwgs	2 wks	Wed 12/7/11	Tue 12/20/11	31	33
33		Hot to Cold Scaling	3 days	Wed 12/21/11	Fri 12/23/11	32	34,7
34		**Inducer Assy PRR**	0 days	Fri 12/23/11	Fri 12/23/11	33	35,36,37
35		Inducer Blade Final Drawing Release (718012)	0 days	Fri 12/23/11	Fri 12/23/11	34	64
36		Inducer Finished Rotor Final Drawing Release (718014, 718002)	0 days	Fri 12/23/11	Fri 12/23/11	34	48
37		Complete Coverplate, Snapring & pin & Seal designs	3 wks	Tue 1/3/12	Mon 1/23/12	34	38,39
38		Coverplate Final Drawing Release	0 days	Mon 1/23/12	Mon 1/23/12	37	82
39		Snap Ring Final Drawing Release	0 days	Mon 1/23/12	Mon 1/23/12	37	84
40		**Inducer Assembly (718000) - Manufacturing**	**254 days**	**Wed 5/25/11**	**Tue 5/22/12**		
80							
87		**Inducer Assy: Build & Checkout**	**18 days**	**Wed 5/9/12**	**Fri 6/1/12**		
88		**Assembly, Balance & Spin Pit**	**5 days**	**Wed 5/9/12**	**Tue 5/15/12**		
89		Build Rotor Assembly	1 wk	Wed 5/9/12	Tue 5/15/12	54,82,71	91
90		**Inducer Assembly Spin Pit**	**13 days**	**Wed 5/16/12**	**Fri 6/1/12**		
91		Balance & Proof Spin Inducer Assy	2 wks	Wed 5/16/12	Tue 5/29/12	89	92
92		Ship Inducer Assy to Olean	3 days	Wed 5/30/12	Fri 6/1/12	91	10
93							
94		**Bald Rotor (716002)**	**80 days**	**Tue 5/31/11**	**Mon 9/19/11**		



Need dates are from the Sept 'optimal' program schedule

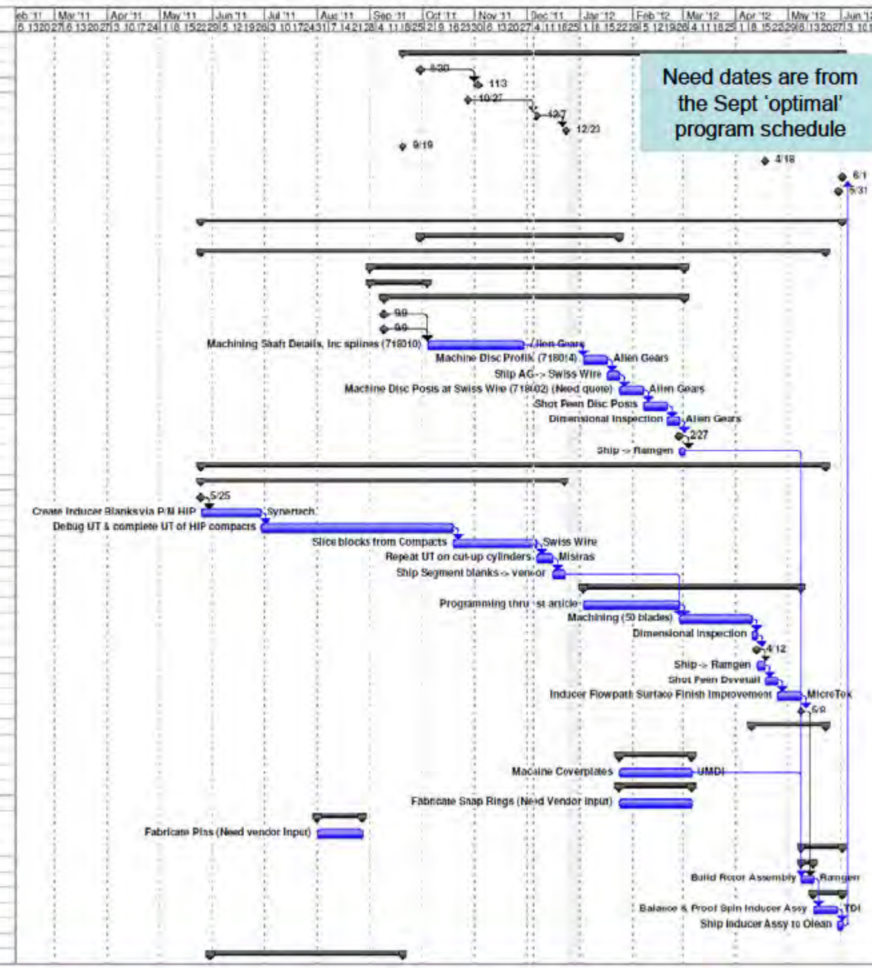
- **Blades are the critical path item on the Inducer Assembly**
- **Switch to blade 12 has added just under 3 weeks to schedule**
  - **Reflected in rotor assy availability moving from 5/7/2012 -> 6/1/2012**

Schedule – Inducer Manufacturing

6/25

ID	Task Name	Duration	Start	Finish	Predecessors	Successors	Resource Names
1							
2	**Key Dates**	179 days	Mon 9/19/11	Fri 6/1/12			
3	Inducer Gen-2 Rev-2 Prelim Aero Definition (from 9/30 schedule in 0900 folder)	0 days	Fri 9/30/11	Fri 9/30/11		S+15 days,16	
4	Inducer Gen-2 Rev-2Final Aero Definition (from 9/30 schedule in 0900 folder)	0 days	Thu 11/3/11	Thu 11/3/11	3FS+15 days	20,24	
5	Inducer Rev 2 PDR	0 days	Thu 10/27/11	Thu 10/27/11	17	6	
6	Inducer Rev 2 FDR	0 days	Wed 12/7/11	Wed 12/7/11	31,5	7	
7	Inducer Rev 2 PRR	0 days	Fri 12/23/11	Fri 12/23/11	633		
8	Bald Rotor available	0 days	Mon 9/19/11	Mon 9/19/11	102		
9	Rotor Assy Required for Install in Rig	0 days	Wed 4/18/12	Wed 4/18/12			
10	**Rotor Assy Available (Bladed Disk)**	**0 days**	**Fri 6/1/12**	**Fri 6/1/12**	92		
11	Gen-2 Build 1 Testing Starts (Optimal schedule)	0 days	Thu 5/31/12	Thu 5/31/12			
12							
13	**Inducer Assembly - Bladed Disk Configuration**	**262 days**	**Wed 5/25/11**	**Fri 6/1/12**			
14	**Inducer Assembly Design**	**76 days**	**Fri 9/30/11**	**Mon 1/23/12**			
40	**Inducer Assembly (718000) - Manufacturing**	**254 days**	**Wed 5/25/11**	**Tue 5/22/12**			
41	**Rotor (718002)**	**125 days**	**Thu 9/1/11**	**Thu 3/1/12**			
42	**Procure Rotor Raw Material (718029)**	**23 days**	**Thu 9/1/11**	**Mon 10/3/11**			
44	**Machine Rotor (718010 -> 718014 -> 718002)**	**119 days**	**Fri 9/9/11**	**Thu 3/1/12**			
45	Quote from Allen Gears & release P/O	0 days	Fri 9/9/11	Fri 9/9/11		47	
46	Release Semi Finished Rotor Drawing (718010)	0 days	Fri 9/9/11	Fri 9/9/11		47	
47	Machining Shaft Details, Inc splines (718010)	8 wks	Tue 10/4/11	Mon 11/28/11	46,45,43	48	Allen Gears
48	Machine Disc Profile (718014)	2 wks	Tue 1/3/12	Mon 1/16/12	47,36	49	Allen Gears
49	Ship AG -> Swiss Wire	1 wk	Tue 1/17/12	Mon 1/23/12	48	50	
50	Machine Disc Posts at Swiss Wire (718002) (Need quote)	2 wks	Tue 1/24/12	Mon 2/6/12	49	51	Allen Gears
51	Shot Peen Disc Posts	2 wks	Tue 2/7/12	Mon 2/20/12	50	52	
52	Dimensional Inspection	1 wk	Tue 2/21/12	Mon 2/27/12	51	53	Allen Gears
53	Rotor Complete	0 days	Mon 2/27/12	Mon 2/27/12	52	54	
54	Ship -> Ramgen	3 days	Tue 2/28/12	Thu 3/1/12	53	89	
55	**Blades (718012)**	**254 days**	**Wed 5/25/11**	**Tue 5/22/12**			
56	**Create HIP compacts for Blades**	**152 days**	**Wed 5/25/11**	**Thu 12/22/11**			
57	Release Synertech to HIP BNI HIP Compacts	0 days	Wed 5/25/11	Wed 5/25/11		58	
58	Create Inducer Blanks via P/M HIP	5 wks	Wed 5/25/11	Tue 6/28/11	57	59	Synertech
59	Debug UT & complete UT of HIP compacts	16 wks	Wed 6/29/11	Tue 10/18/11	58	60	
60	Slice blocks from Compacts	7 wks	Wed 10/19/11	Tue 12/6/11	59	61	Swiss Wire
61	Repeat UT on cut-up cylinders	7 days	Wed 12/7/11	Thu 12/15/11	60	62	Mistras
62	Ship Segment blanks -> vendor	5 days	Fri 12/16/11	Thu 12/22/11	61	65	
63	**Machine Inducer Blade (718012)**	**91 days**	**Tue 1/3/12**	**Tue 5/8/12**			
64	Programming thru 1st article	8 wks	Tue 1/3/12	Mon 2/27/12	35	65	
65	Machining (60 blades)	6 wks	Tue 2/28/12	Mon 4/9/12	64,62	66,73	
66	Dimensional Inspection	3 days	Tue 4/10/12	Thu 4/12/12	65	67	
67	All blades complete	0 days	Thu 4/12/12	Thu 4/12/12	66	68	
68	Ship -> Ramgen	3 days	Fri 4/13/12	Tue 4/17/12	67	69	
69	Shot Peen Dovetail	1 wk	Wed 4/18/12	Tue 4/24/12	68	70	
70	Inducer Flowpath Surface Finish Improvement	2 wks	Wed 4/25/12	Tue 5/8/12	69	71	MicroTek
71	Blades complete	0 days	Tue 5/8/12	Tue 5/8/12	70	89	
72	**Spare Inducer Blades**	**31 days**	**Tue 4/10/12**	**Tue 5/22/12**			
80							
81	**Coverplates (718005, 718006)**	**30 days**	**Tue 1/24/12**	**Mon 3/5/12**			
82	Machine Coverplates	6 wks	Tue 1/24/12	Mon 3/5/12	38	89	UMDI
83	**Snap Rings (718009)**	**30 days**	**Tue 1/24/12**	**Mon 3/5/12**			
84	Fabricate Snap Rings (Need Vendor Input)	6 wks	Tue 1/24/12	Mon 3/5/12	39		
85	**Anti Rotation Pins (122329 - Ram2)**	**20 days**	**Mon 8/1/11**	**Fri 8/26/11**			
86	Fabricate Pins (Need vendor Input)	4 wks	Mon 8/1/11	Fri 8/26/11			
87	**Inducer Assy: Build & Checkout**	**18 days**	**Wed 5/9/12**	**Fri 6/1/12**			
88	**Assembly, Balance & Spin Pit**	**5 days**	**Wed 5/9/12**	**Tue 5/15/12**			
89	Build Rotor Assembly	1 wk	Wed 5/9/12	Tue 5/15/12	54,82,71	91	Ramgen
90	**Inducer Assembly Spin Pit**	**13 days**	**Wed 5/16/12**	**Fri 6/1/12**			
91	Balance & Proof Spin Inducer Assy	2 wks	Wed 5/16/12	Tue 5/29/12	89	92	TDI
92	Ship Inducer Assy to Olean	3 days	Wed 5/30/12	Fri 6/1/12	91	10	
93							
94	**Bald Rotor (718002)**	**80 days**	**Tue 5/31/11**	**Mon 9/19/11**			

Need firm quote from Turbocam



'Optimal' program schedule does not align with actual design and manufacturing timescales

First Level Stress Calcs

RAMGEN POWER SYSTEMS ™

Analyses at 34500 RPM

Blade Version	**718012_v063**
Disc Attachment Geometry :	**718002_v037**
Skew Angle =	**0**

Part	Stress Type	Section	Width (in)	Axial Length (in)	CSA (in^2)	Load (lbf)	Average Stress (ksi)	Allowable Stress (ksi)				Stress Acceptable	Shed Speed (rpm)	% of disc burst Speed
								Yield	Ultimate	Creepstrain	Stress Rupture			
Blade	Tension	Segment Neck	0.1800	1.980	0.3564	9577	26.9	100.5	65.6	?	?	Yes	85,231	73%
	Shear	Segment Shear	0.1092	1.980	0.2162	5649	26.1	45.0	32.8	-	-	Yes	(Target < 90% burst speed)	
	Bearing	Bearing	0.0700	1.980	0.1386	3240	23.4	120.0	-	-	-	Yes		
Disc	Tension	DiscPost Neck	0.1307	1.980	0.2587	13694	52.9	97.2	69.6	?	?	Yes	60,730	52%
	Shear	DiscPost Shear	0.1376	1.980	0.2725	5649	20.7	43.5	34.8	-	-	Yes	(Target < 90% burst speed)	
	Bearing	Bearing	0.0700	1.980	0.1386	3240	23.4	116.0	-	-	-	Yes		

Neck Section 1 (Top)

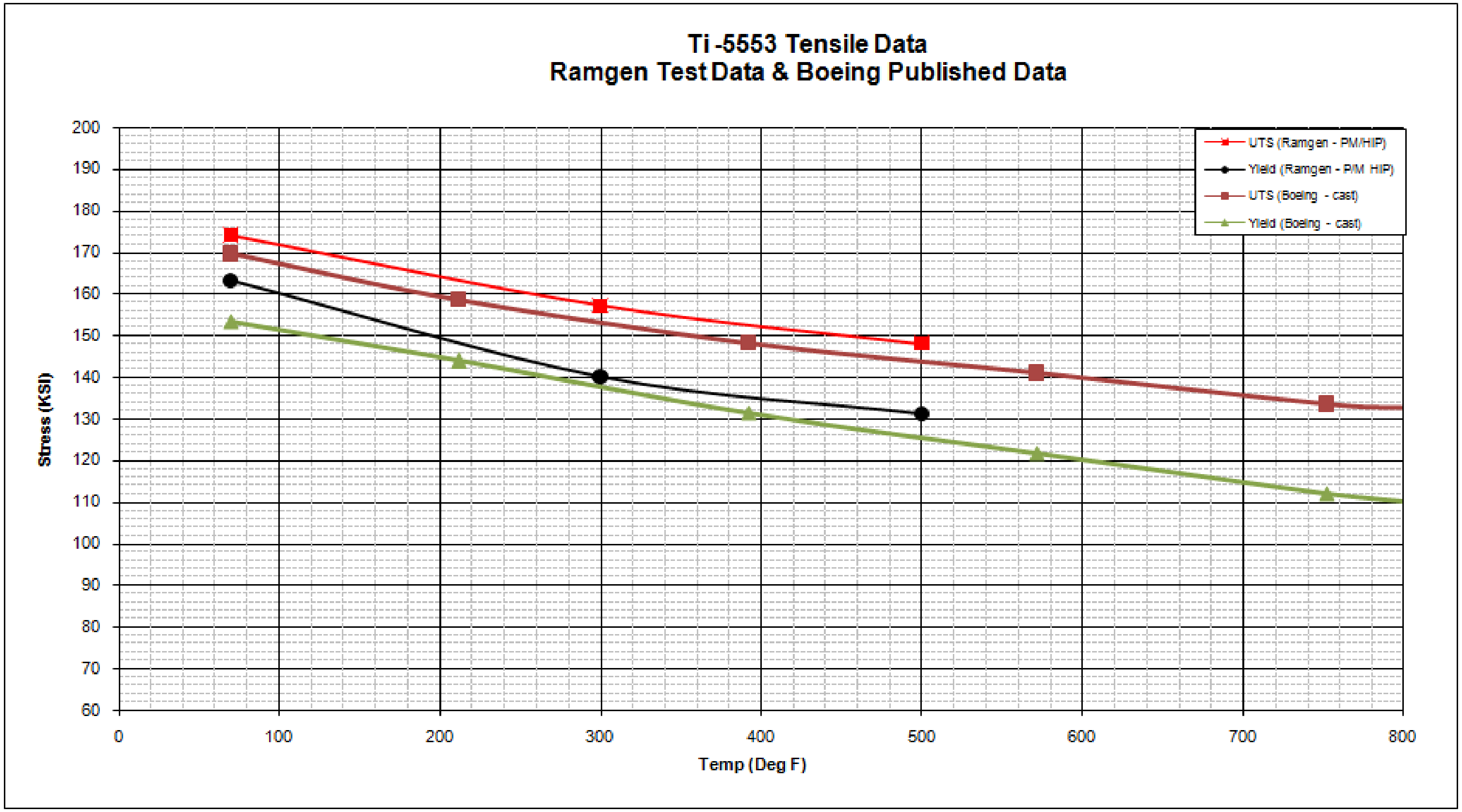
Segment Properties (Ti-5553)			Disc Properties (4340)		
Temperature =	200	deg F	Temperature =	**200**	deg F
UTS =	164	ksi	UTS =	174.0	ksi
YS =	150	ksi	YS =	145	ksi
1%ε_{cr} (60K hrs, T+0) <1> =	?	ksi	0.2%ε_{cr} (100K hrs, T+0) =	?	ksi
σ_{cr}(60K Hrs, T+50) =	?	ksi	σ_{cr}(100K Hrs, T+50) =	?	ksi

Item	Stress	Allowable Stress (ksi) based on the following					
		Yield		Ultimate		σ for 0.2%ε_{cr}	$\sigma_{rupture}$
Blade	Root/Firtree direct (ave)	67%	100.5	40%	65.6	?	?
	Firtree shear (ave)	30%	45.0	20%	32.8	-	-
	Firtree bearing	80%	120.0	-	-	-	-
Disc	Direct stress (ave)	67%	97.2	40%	69.6	?	?
	Shear stress (ave)	30%	43.5	20%	34.8	-	-
	Firtree bearing stress	80%	116.0	-	-	-	-

Disc Burst Speed (rpm) 116,310 Ref 718002_v037_DiscBurst_Run01_R130.wbpj

Basic stresses acceptable

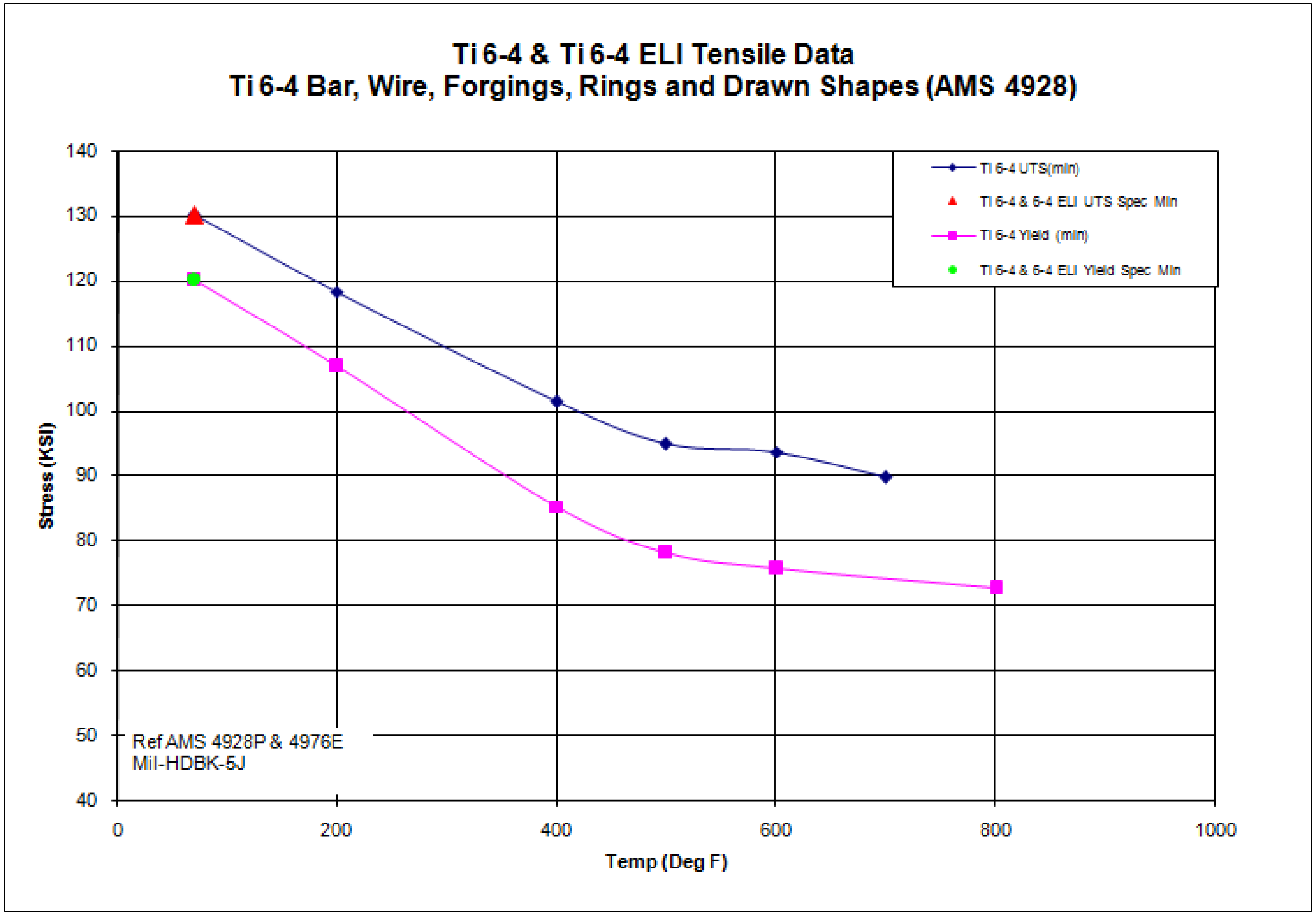
Ti-5553 Yield & Ultimate Data



At 200F
Min UTS = 164 ksi
Min Yield = 150 ksi

In addition, from Ramgen material testing < 135 ksi gives > 10,000 LCF cycles at 300F. See Gen1 PDR slide

Ti 6-4 Yield & Ultimate Data



At 200F
Min UTS = 118 ksi
Min Yield = 107 ksi

Ti 6-4 Fatigue Data

MIL-HDBK-5J
31 January 2003

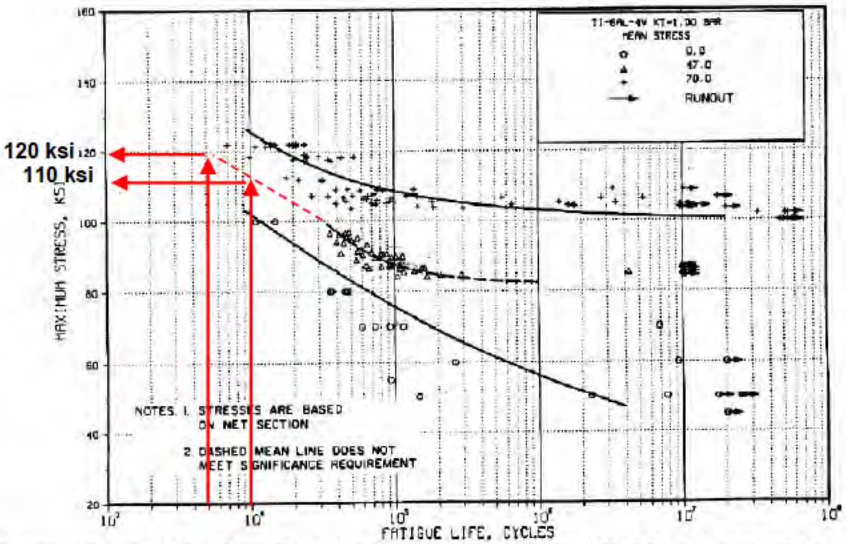


Figure 5.4.1.1.8(a). Best-fit S/N curves for unnotched Ti-6Al-4V annealed bar, longitudinal direction.

Data shown is for bar at room temp

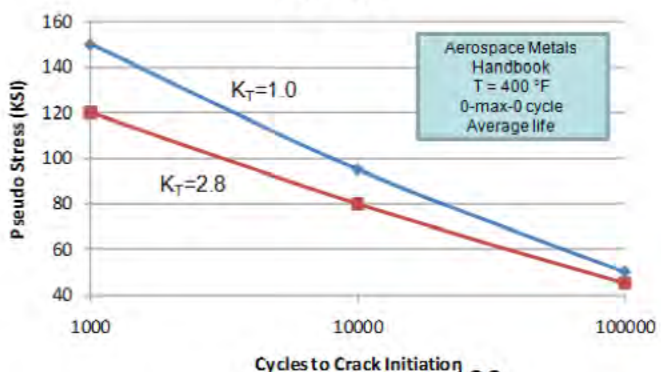
Using 10,000 cycle data, temperature knockdown:

- Mil5: 11% (ref tables 5.4.1.1.8(d) & 5.4.1.1.8(e)
- Aerospace handbook: 7% (Ref table 3.5.1.1)
- Both for RT -> 400F

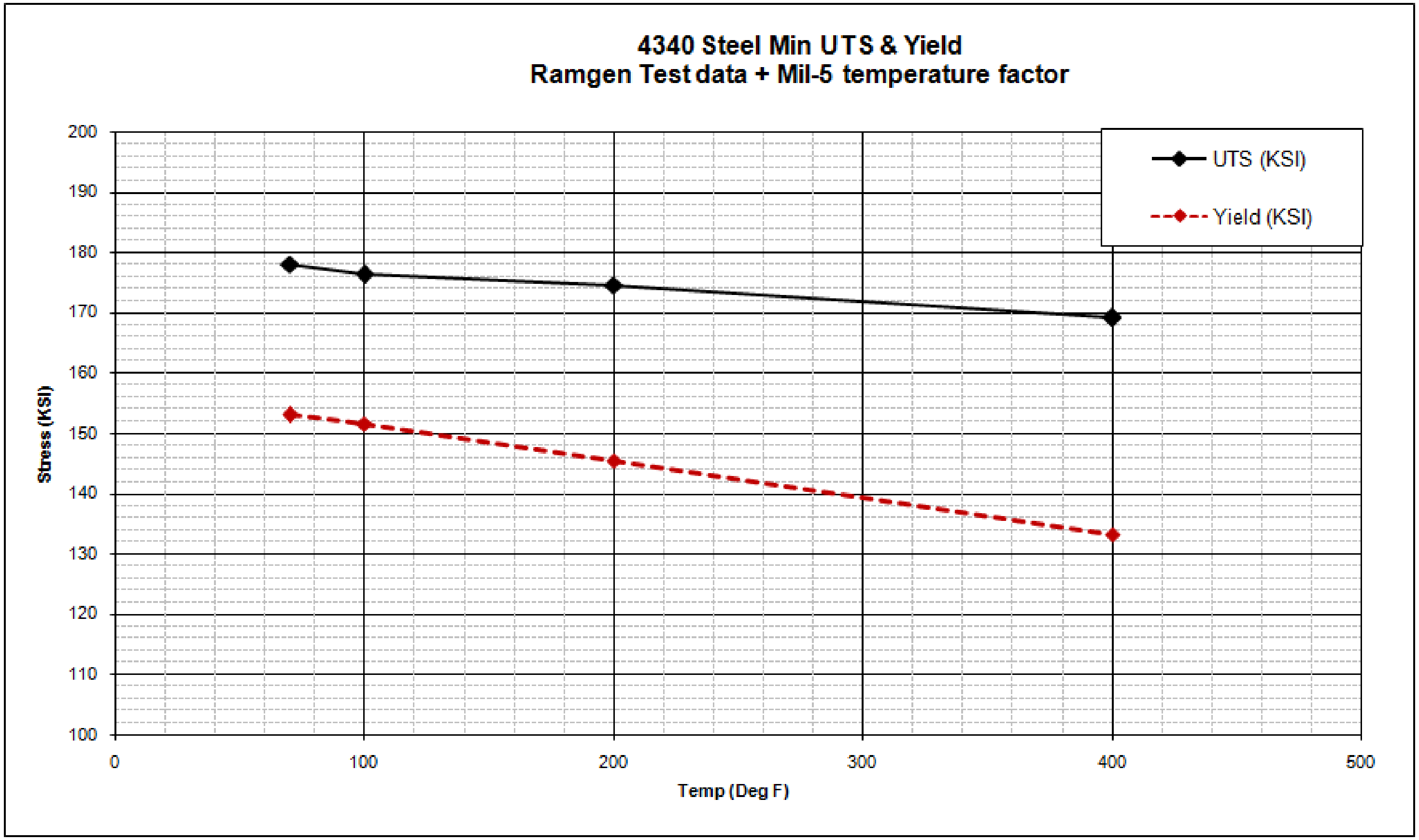
Assume 11% reduction in fatigue strength & using curve w/ mean stress of 47 ksi:

- 5,000 cycle stress limit at 200F = 120 ksi x .89 = 107 ksi
- 10,000 cycle stress limit at 200F = 110 ksi x .89 = 98 ksi

This is in line with Agilis analysis of LP CO_2
Prediction of 12,000 cycles at 94 ksi



4340 Yield & Ultimate Data



At 200F
Min UTS = 174 ksi
Min Yield = 145 ksi

Rotor Material Tensile Testing

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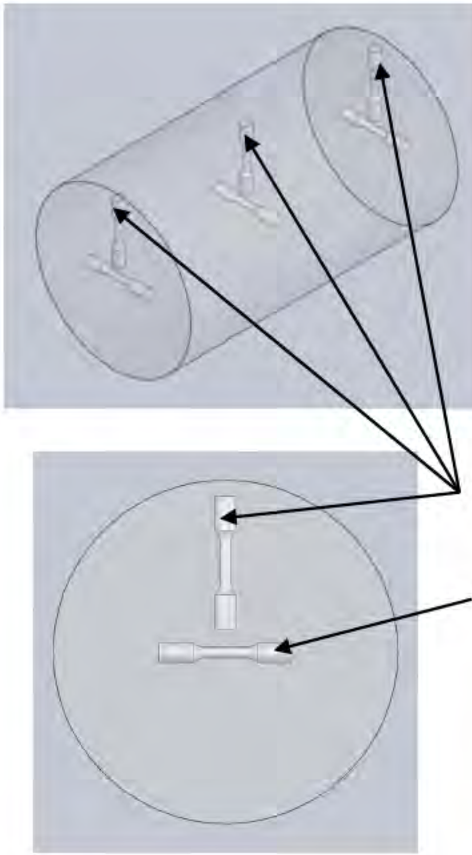


DICKSON TESTING COMPANY, INC

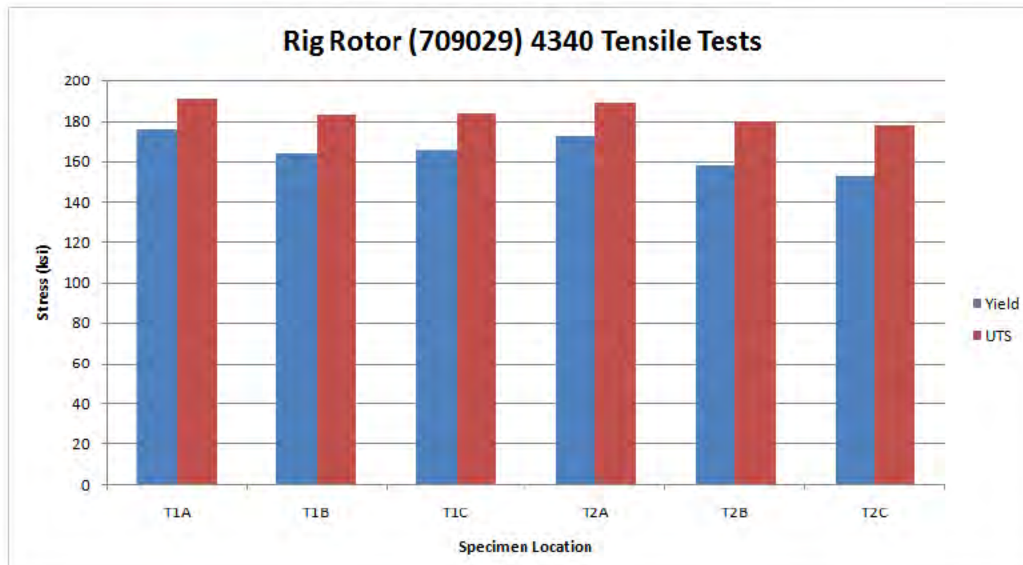
RAMGEN POWER SYSTEMS LLC
11808 NORTHUP WAY, SUITE W-190, BELLEVUE, WA 98005, USA

CERTIFIED TEST REPORT September 19, 2011
DTC LAB NUMBER **11099005-00**
PURCHASE ORDER **2995**

SIZE 7.5"DIA. X 13"LONG BAR MATERIAL 4340
SPECIFICATIONS CUSTOMER REQUIREMENT, ROTOR TENSILE TESTS RELEASED 8/18/11

TENSILE

SPECIMEN ID	DIR	TEST TEMP	INITIAL GAGE LENGTH	INITIAL GAGE DIM	OFFSET YIELD OFF	OFFSET YIELD LOAD	OFFSET YIELD STRENGTH	ULTIMATE LOAD	ULTIMATE STRENGTH	FINAL GAGE LENGTH	FINAL GAGE DIM	ELONG'N 4D	REDN IN AREA
		(°F)	(inch)	(inch)	(%)	(lbf)	(ksi)	(lbf)	(ksi)	(inch)	(inch)	(%)	(%)
LOC. T1A	TR	ROOM	1.000	0.252	0.20	8791	176	9550	191	1.160	0.1695	16.0	55
	YOUNG MODULUS: 31217 KSI												
LOC. T1B	TR	ROOM	1.000	0.250	0.20	8066	164	8998	183	1.124	0.204	12.4	33
	FRACTURED THROUGH 4D GAGE MARKS. YOUNG MODULUS: 30991 KSI												
LOC. T1C	TR	ROOM	1.000	0.252	0.20	8296	166	9184	184	1.136	0.205	13.6	34
	FRACTURED OUTSIDE MID-HALF OF GAGE SECTION. YOUNG MODULUS: 30786 KSI												
LOC. T2A	TR	ROOM	1.000	0.251	0.20	8538	173	9366	189	1.168	0.1705	16.8	54
	YOUNG MODULUS: 30630												
LOC. T2B	TR	ROOM	1.000	0.251	0.20	7794	158	8886	180	1.156	0.203	15.6	35
	YOUNG MODULUS: 30561 KSI												
LOC. T2C	TR	ROOM	1.000	0.252	0.20	7651	153	8867	178	1.132	0.209	13.2	31
	YOUNG MODULUS: 31819 KSI												



4340 Fatigue Data

MIL-HDBK-5J
31 January 2003

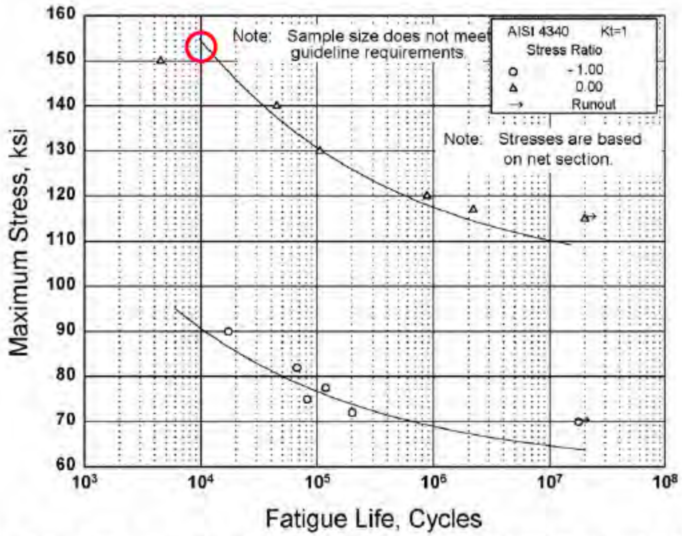


Figure 2.3.1.3.8(c). Best-fit S/N curves for unnotched AISI 4340 alloy steel bar, F_{tu} = 150 ksi, longitudinal direction.

MIL-HDBK-5J
31 January 2003

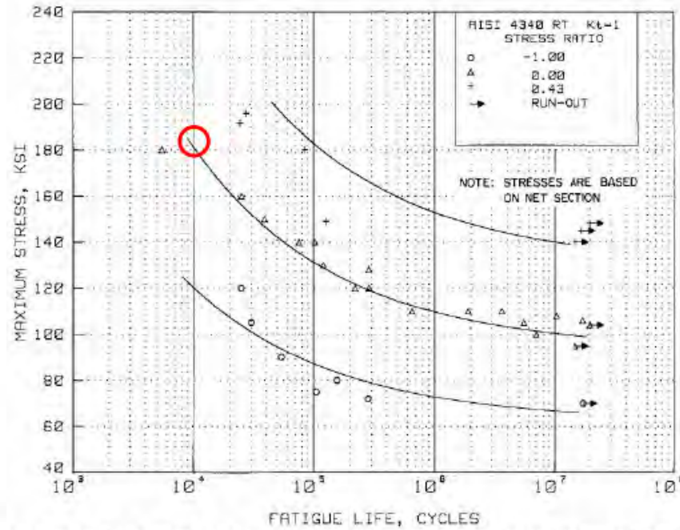
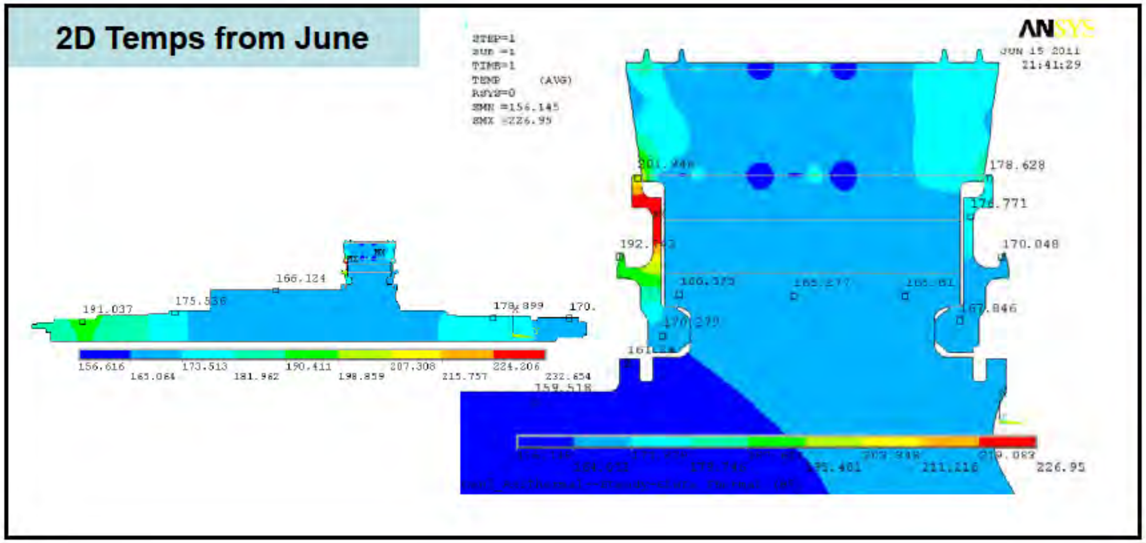
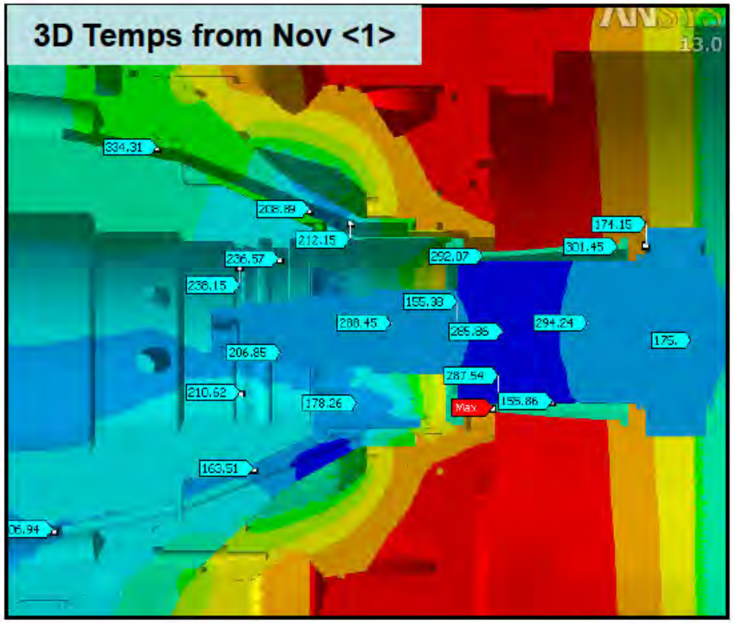


Figure 2.3.1.3.8(k). Best-fit S/N curves for unnotched AISI 4340 alloy steel bar and die forging, F_{tu} = 200 ksi, longitudinal direction.

- For 4340 w/ 150 ksi UTS: Stress < 155 ksi gives Life > 10,000 cycles
- For 4340 w/ 200 ksi UTS: Stress < 182 ksi gives Life > 10,000 cycles
- Ramgen Material UTS = 178 ksi (min)
  - Impose 150 ksi stress limit to enable > 10,000 cycle life
  - Consider fatigue testing of additional samples.

Temperatures

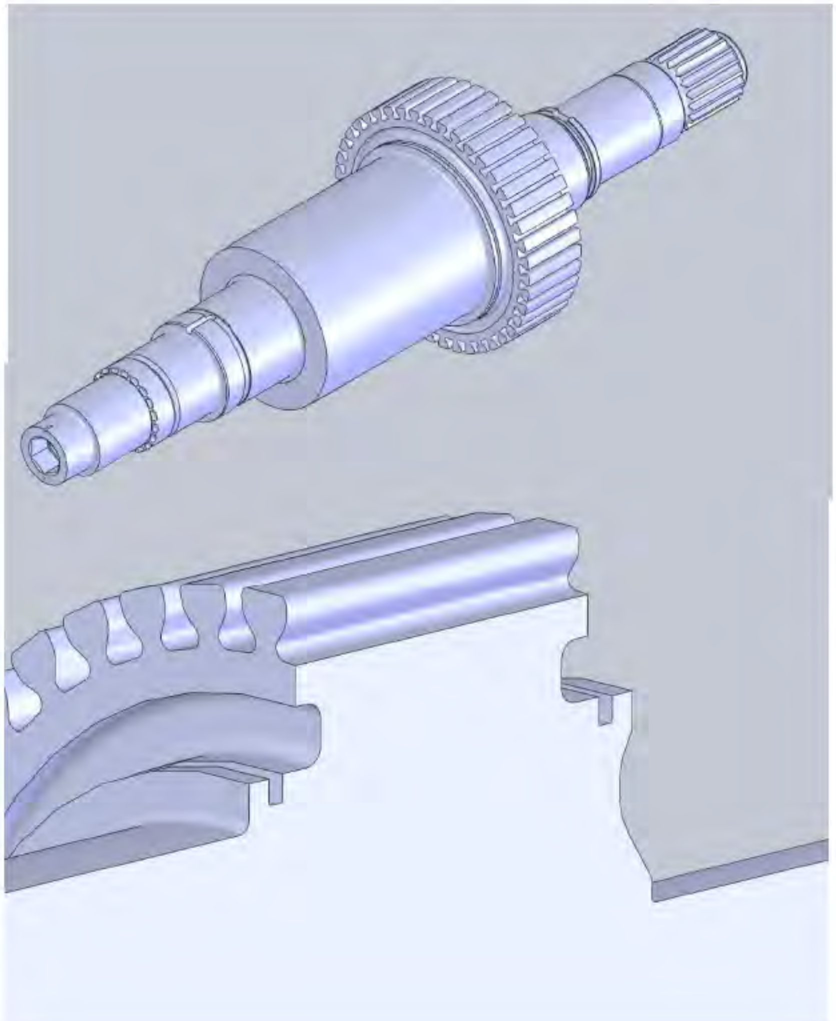




Rotor temperatures effectively unchanged since original 2D analysis

Ref: L:\CO2Program\Demo Unit Mechanical Systems\Static Structure\ThermalFEA\2DThermal_Gen2\Core Rig 3D Thermal FEA.pptx

Rotor Design



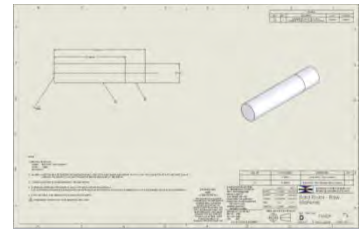
- Single Piece Rotor
- 4340 forged bar (Q & T to 180-200 ksi UTS)
- 43 disc slots – dovetail profile similar to Ram-2
- Large cylinder downstream of blade. Solid rotor maintains design flexibility whilst allowing rotor design and manufacture to proceed

- UT Inspection of forged bar
- Mag Particle & LPI of finished part
- 2 plane balance using disc posts & cylinder

Rotor Manufacturing

In an effort to move the rotor manufacturing along without defined blade geometry, the following approach has been adopted

716029 Raw material (complete)



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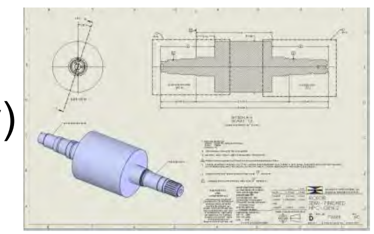
718010 Semi-Finished Rotor (in-progress. ECD end of Nov)



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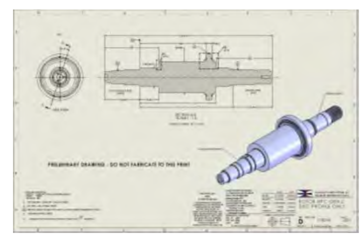
718014 Rotor Secondary Machining (2 weeks)



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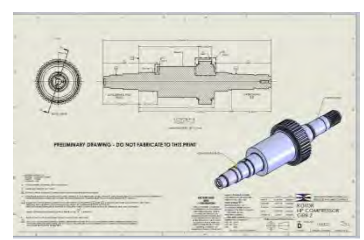
718002 Rotor Disc Slot Machining (2 weeks)



Rotor will be available ahead of when it is needed

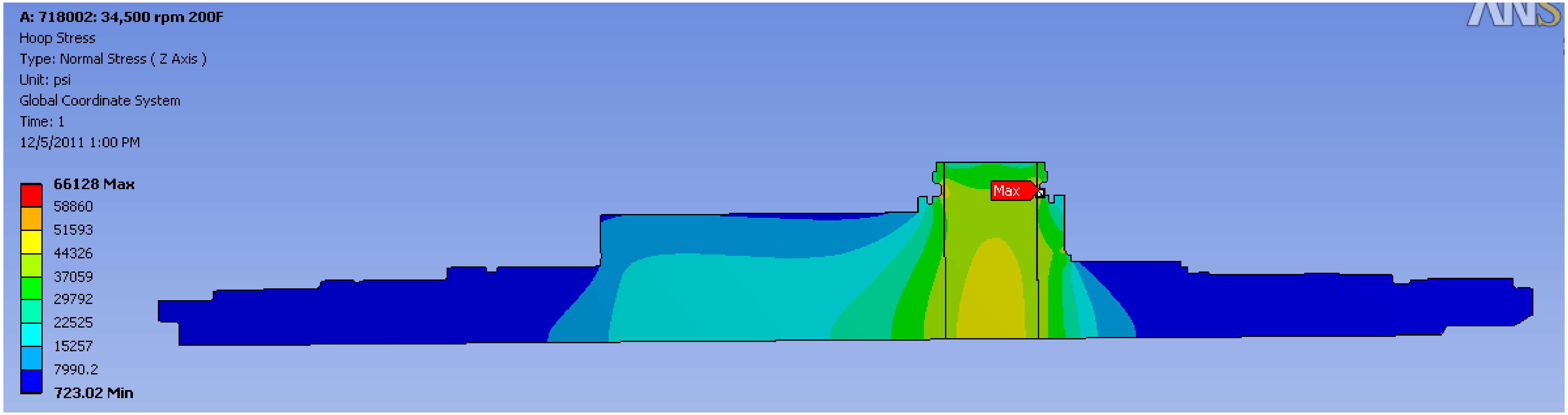
Rotor – Burst Margin

Steel Rotor

Stresses at MCOS (34,500rpm), 200F

Loads:

- **Rotational loads**
- **Live rim load of 588,859 lbf (24,833 psi)**
- **Coverplate load of 35,000 psi**



Material Properties at 200F

4340: Yield = 145 ksi, UTS = 174 ksi

Burst Speed in 4340 = 116,000 rpm (require > 43125rpm)
Burst margin = 3.37 (require > 1.25)

1. **Burst Margin acceptable**
2. **Local stresses to be discussed in Rotor 3D FEA**

Coverplate Design

- Ti 6-4 or 6242 Coverplate
- Similar to Ram-2 Design
- Pilots on Disc (0.001”-0.003” cold build interference)
- Retained by snap ring
- Anti-rotation via 2 pins (remake Ram-2 pins)
- Seals against end of blade platform via movement of OD section
- 0.001” cold build gap between Coverplate and blade
- NDE Coverplate likely different geometry as diffuser thermal growth < DE static structure, requiring < lab tooth radial movement.

Disk & Coverplate 3D FEA Model

- 1/43 sector model
- Run as coarse model & component sub-models
- Rotor, Blade, Coverplates, Snap rings & anti-rotation pins included

- Frictional between blade and disc post
- Frictional between Coverplate & disc pilot
- Frictionless contact between:
    - Blade & Coverplates

- 34,500rpm
- Radial temperature distribution
- Blade Pressures from CFD
- Shroud Pressures

- Axial restraint at thrust bearing
- Circumferential restraint to rotor and Coverplates

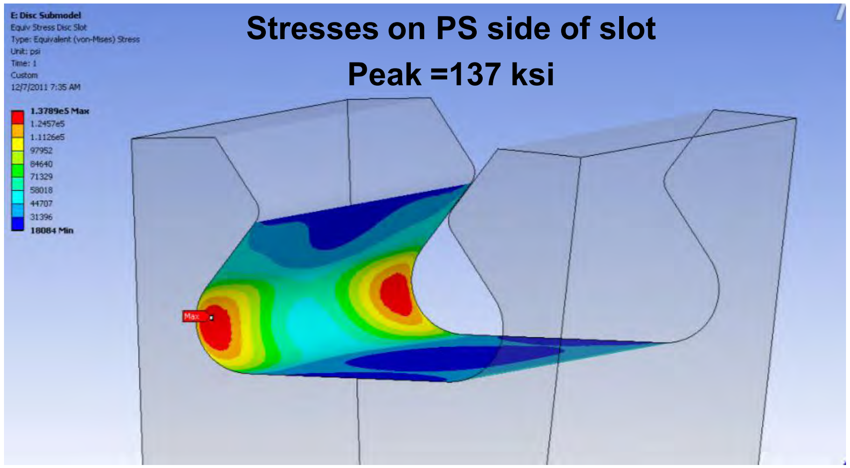
- Ti-5553 blade
- Ti 6-4 Coverplates
- 4340 shaft

Ref: 718000_v030_CoverPlateDisc_Run03_R130.wbpj

Disk Stresses

Disc Post Stresses

Disc submodel using disps from full assy coarse model

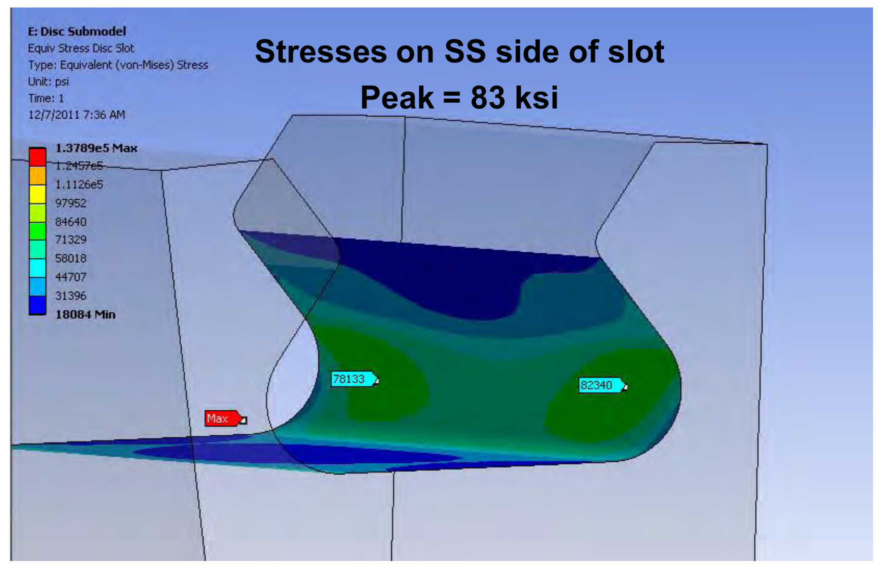


Peak Stress = 137 ksi

At 200F:

- **UTS = 174 ksi, Yield = 145 ksi**
- **Stress for 10,000 cycle life = 150 ksi**

Lean & camber of blade 12 prohibit optimum balance of blade which results in this biased stress distribution PS to SS



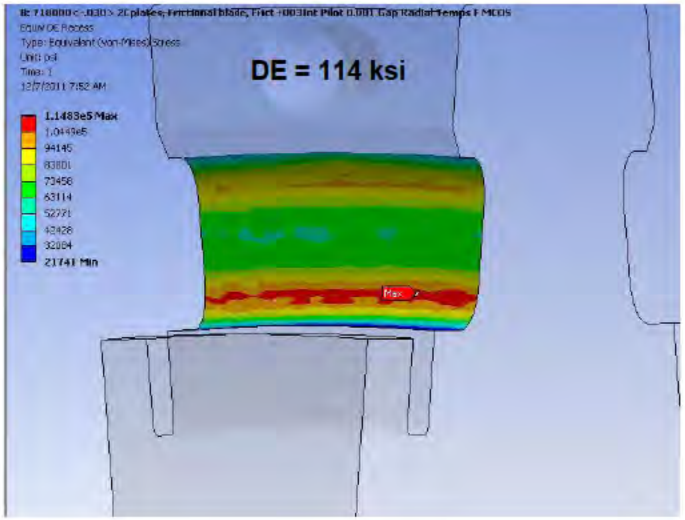
- **Disc Post stresses from FEA acceptable at MCOS**
- **Will attempt minor adjustments to blade to reduce bias**

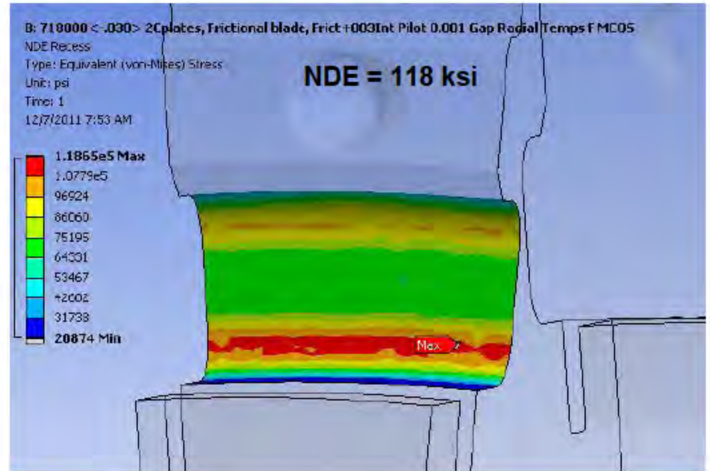
Ref: 718000_v030_CoverPlateDisc_Run03_R130.wbpj

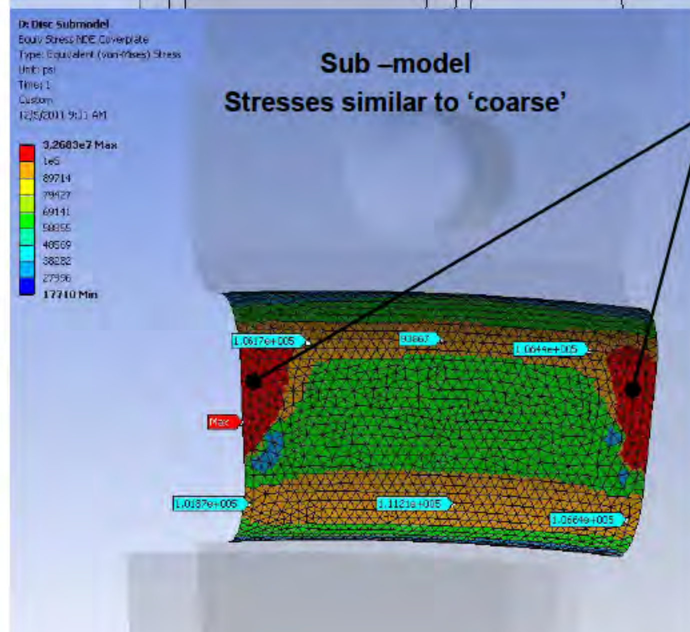
Disk Stresses

RAMGEN POWER SYSTEMS ™

Coverplate Recess







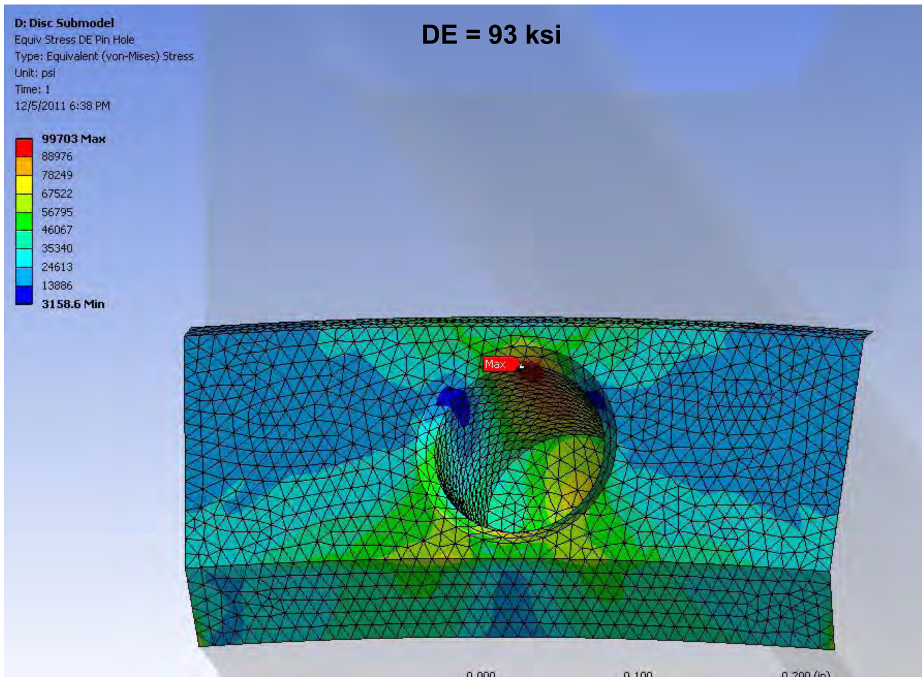
Note: These are edge effects from submodel cut boundary and are non-real

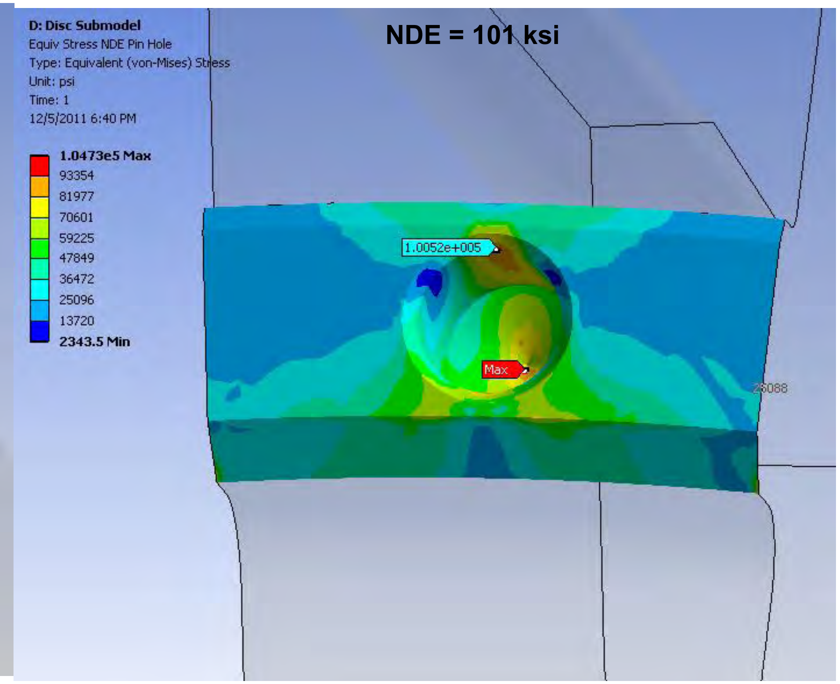
- **Recess stresses from FEA acceptable at MCOS**
- **Keep eye on recess stresses as NDE Coverplate changes in step with diffuser**

Ref: 718000_v030_CoverPlateDisc_Run03_R130.wbpj

Disk Stresses

Anti-Rotation Pin hole





<1> 'Singularity' at bottom of hole of 105 ksi ignored

Moving anti rotation hole out of radial loadpath (See Gen2 Rev1 PDR) results in acceptable hole stresses

Ref: 718000_v030_CoverPlateDisc_Run02_R130.wbpj

Coverplate Stresses

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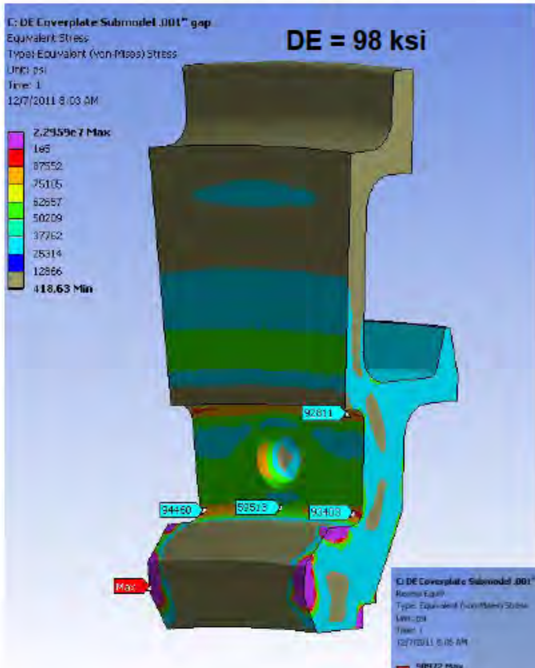
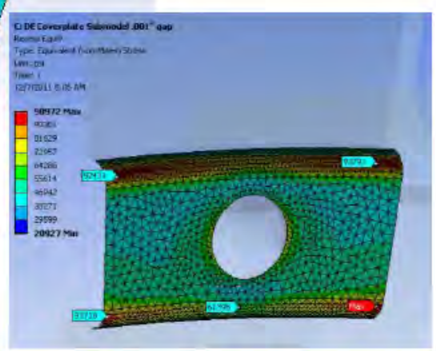
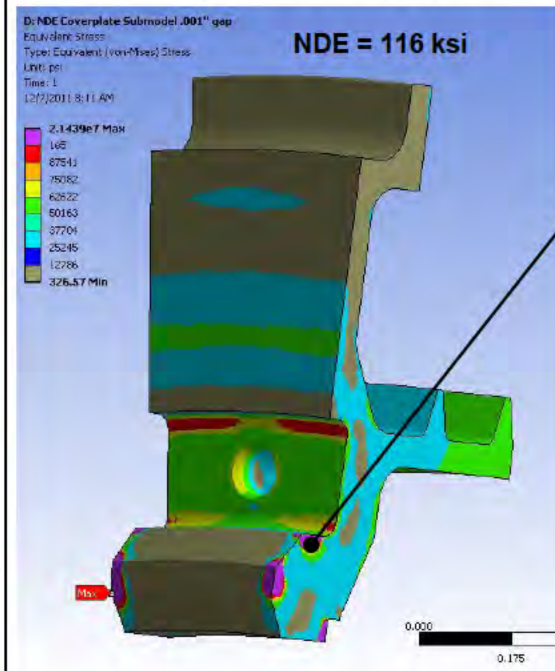
DE Coverplate

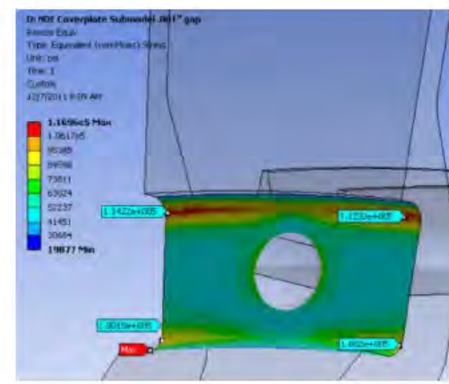




NDE Coverplate



- Local peaks shown are on sub model cut boundary & and non-real
- At 200F 6-4 Yield = 105 ksi

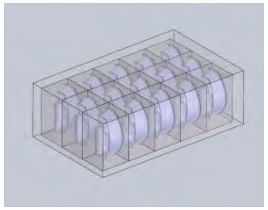


- DE Coverplate stresses acceptable at MCOS
- Keep eye on NDE Coverplate stresses as design changes in step with diffuser
    - May need to drop 2nd tooth or move to higher strength material…?

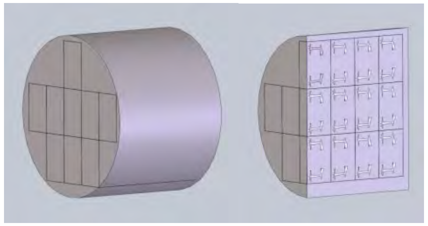
Ref: 718000_v030_CoverPlateDisc_Run03_R130.wbpj

Blade Raw Material Source

- **Use Weld Trial HIP Compact (716033) originally intended for Barber-Nichols Inc**
  - **Only yields 15 or so**
  - **HIP'ed, Heat Treated, Cut-up, UT'd & ready for machining**



- **Use Blisk HIP Compacts (716031) originally intended for BNI**
  - **Could yield 36-44 blades per compact. We have two.**
  - **Some wastage but not excessive**
  - **Cans were ready in May but not filled**
  - **HIP'ed, Heat Treated. Some indications on 2nd UT.**
    - **Cut-up and re-inspect**
      - **Cut up complete 12/1 (tbc)**



Cylinders & blocks at Redmond after UT

Inducer Assembly Build Sequence

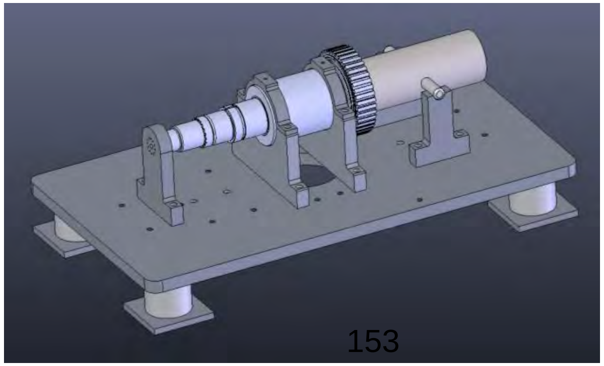
Inducer assembly tooling will likely utilize the methodology used in Ram-2

- **Segment Assy Tool**
  - **Aid in installation of interblade seals**
  - **Facilitate installation of blades into disc**
  - **See following slides**

- **Coverplate Assy Tool**
  - **Facilitate installation of coverplate and snap ring**
  - **Aid in removal of snap ring**
  - **See slides in *\\RP-FILE\engineering\Rampressor_2\Mech\Rotor Shaft\snap ring 03-09-05.ppt***

- **Co-ordinate with Chris who has a rotor build-up table & tool almost complete**

See PDR for additional details



Outstanding items / issues to resolve

1. Perform hot-to-cold conversion using design point data (30,000rpm)
2. Complete UT of remaining Ti-5553 material
3. Adjust shroud geom to increase raw material stock
4. Optimize cavities to increase 2nd Torsion : 12EO margin
5. Run 'safe diagram' analysis before PRR
6. Determine how many spare Inducer blades we require prior to placing an order
7. Order additional Ti-5553 HIP compacts for Build 2 after Build 1 PRR ?
8. Determine need for custom shipping crate for built up rotor assembly.

- **Target PRR for end of next week**

Appendix 6.2

HP CO_2 DIFFUSER PDR

Preliminary Design Review

Generation 2
Static Diffuser – Case 15H Design

System owner(s):

Mech: Geene Cevrero, Dave Taylor, Brian Massey, Rob Draper

Aero: Paul Brown, Ravi Shrinivasan

12/9/2011

Agenda

8:00-8:15	**System Definition and Scope**
8:15-8:30	**Functional Requirements**
8:30-9:30	**Aero Design/Analysis**
	Mechanical Design/Analysis:
9:30-11:00	– **Actuator System**
11:00-11:45	***Break for D-R Call/Lunch***
11:45-12:30	– **Starting Door**
12:30-1:15	– **Shroud**
1:15-2:00	– **Static Diffuser Hub**
2:00-2:15	**Work Plan/Analysis Tasks Remaining**
2:15-2:30	**Budget and Schedule**

System Definition and Scope

- **The Static diffuser comprises:**
  - **1 x Hub Component with 5 main Flowpaths & 10 Bleed Inserts (5 Fwd bleed, 5 Aft bleed)**
  - **1 x Shroud component to define the main flowpath outer annulus along with an integral bleed management system (aka door) and associated actuation hardware**
- **Static diffuser accepts flow exiting the inducer and converts total pressure to useful static pressure via a series of shock waves and gradual area changes while minimizing flow losses**
  - **Major components are diffuser hub, shroud, shroud door, and actuation system.**
- **Provides performance bleed through individual passages on hub and shroud (4 bleed circuits total: hub forward, hub aft, shroud bleed, door bleed)**
- **Shroud doors will simultaneously provide throat relief and additional aft bleed during diffuser starting. Shroud door will be attached to the shroud and have a flow path interface. Doors will have an external actuation system to provide required motion.**
- **External actuation system will provide necessary door motion for starting the system**

Functional Requirements

- Accept flow from inducer Blade 12
- Maximize conversion of flow velocity into static pressure
- Meet bleed flow requirements within the physical space available TBD
  - FWD Hub: (8%)
  - AFT Hub: (6%)
  - Shroud Bleed: (4%)
  - Door Bleed: (8%)
  - Starting Bypass: (26%)
- Implement features to allow starting of all flow paths
- Provide mechanical structure for the following Aero Definitions:
  - 1) Case 15H (as is)
  - 2) De-contracted Throat Case (Hub Only)
- Provide adequate sealing between flow path and bleed passages to avoid significant disruption of the main flow and/or performance loss
- Minimize leakage between bleed passages
- Minimize leakage between the flow paths

Functional Requirements (cont)

- **Structural integrity of components to allow for normal operation of pressure load schedule**
  - **As outlined in Gen2 Aero Spec and Released Data.xlsx**
  - **410 SS material in annealed condition assumed for all except the shroud, hub, and door components (as a starting point)**
    - **Yield strength: 35 ksi, UTS: 65 ksi, CTE: 5.5E-6 in/in-F**
  - **17-4PH SS material in H1100 condition assumed for the shroud, hub, and door components (as a starting point)**
    - **Yield strength: 132,000 psi, UTS: XX ksi, CTE: X.XE-6 in/in-F**

- **Component tolerance stack up to allow for ± 0.002" flow path dimensional tolerance in the supersonic section as outlined in Gen2 Aero Spec and Released Data.xlsx**

Preliminary Design Review

Generation 2
Door Actuation System.

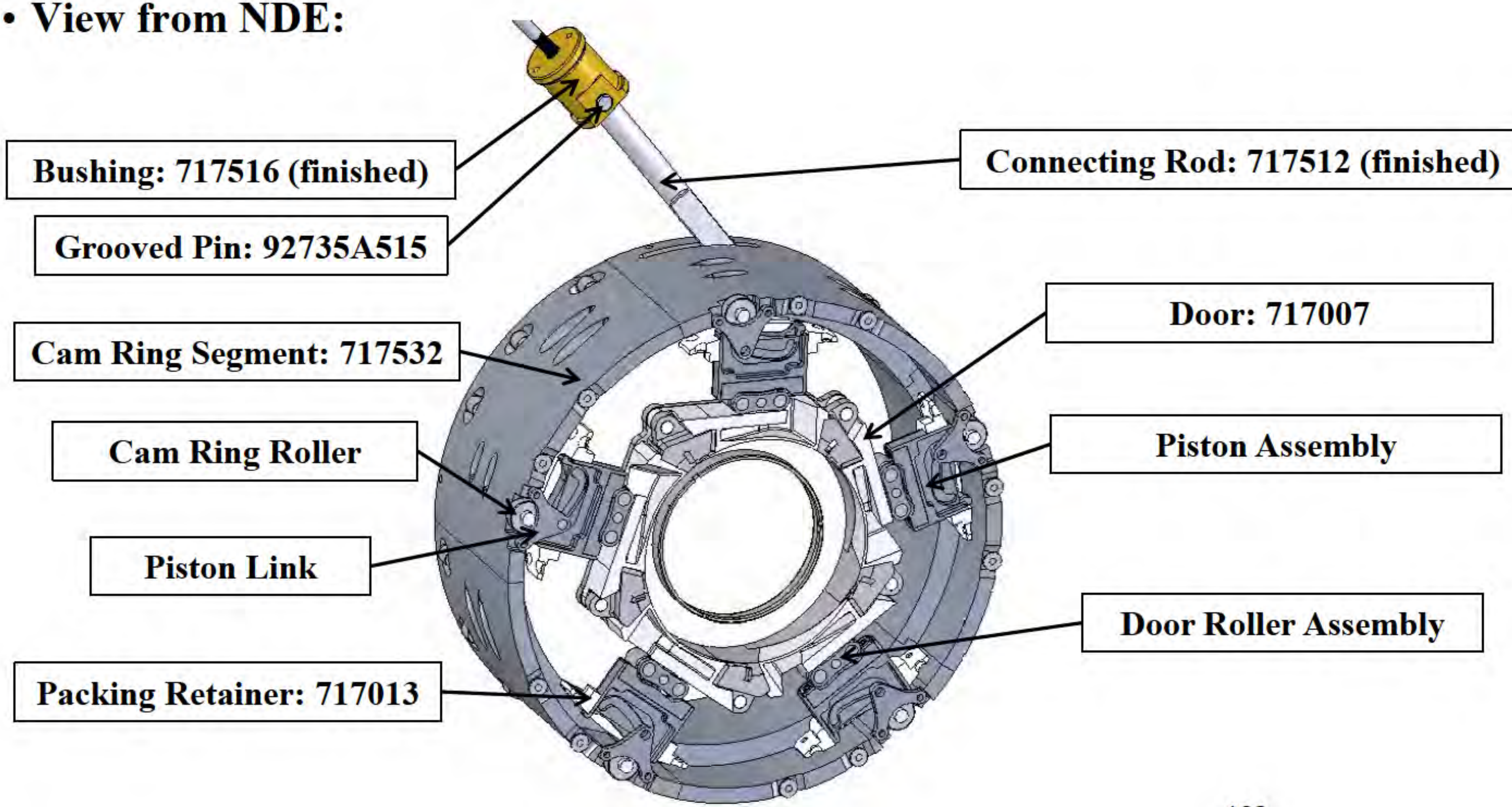
System owner(s):

Brian Massey, Rob Draper

12/9/2011

System Definition and Scope

- Simultaneously actuate all five, static diffuser doors to Aero defined open and closed positions during testing of CO2 compressor
- View from NDE:

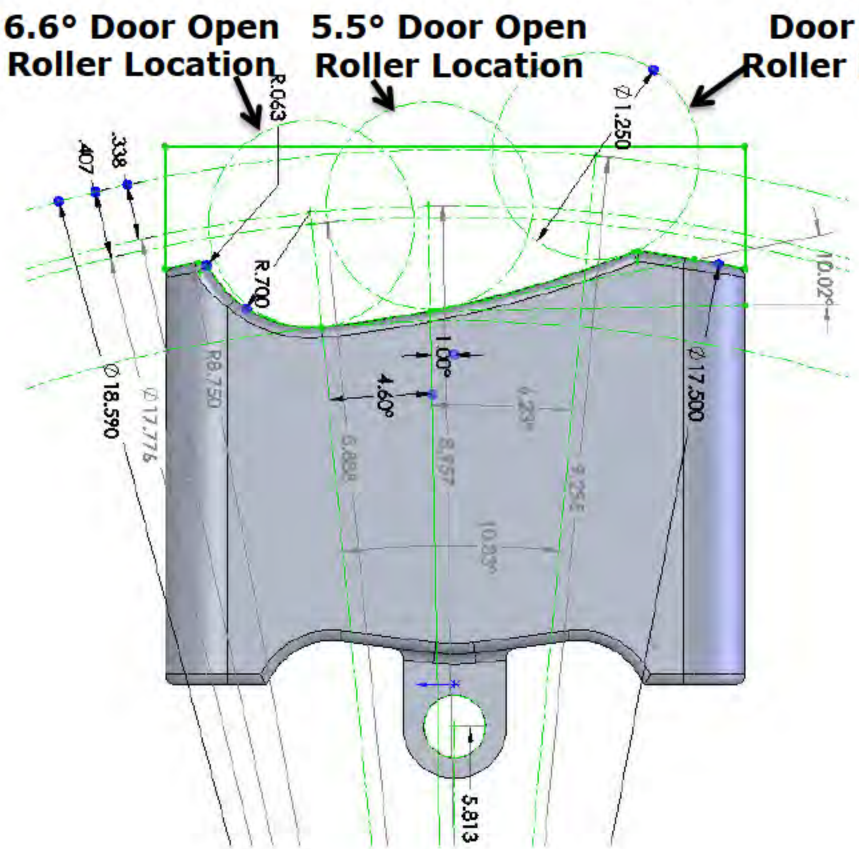


Functional Requirements

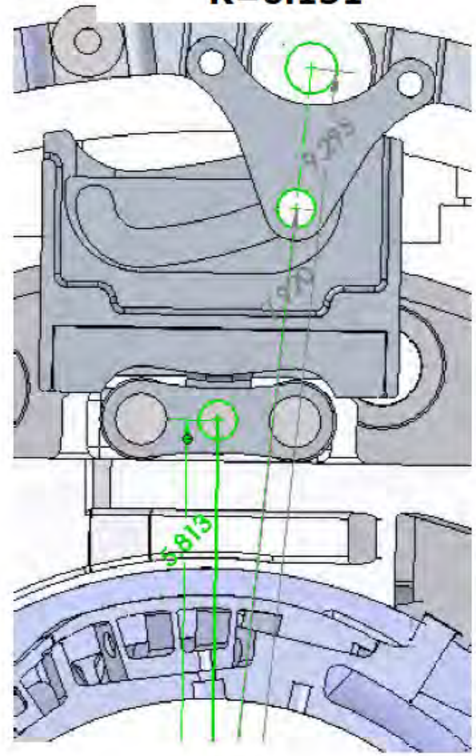
- Provide desired door travel schedule
- Structural integrity of components to allow for normal operation of pressure load schedule
  - As outlined in Gen2 Aero Spec and Released Data.xlsx
  - And refined in the following 2D, CFD starting analysis: HPCO2_Diffuser_StartingDoors_02November2011.pptx
  - 410 SS material in annealed condition assumed for all parts (as a starting point)
    - Yield strength: 35 ksi, UTS: 65 ksi, CTE: 5.5E-6 in/in-F
- Component tolerance stack up to allow for ± 0.002" flow path dimensional tolerance in the supersonic section as outlined in Gen2 Aero Spec and Released Data.xlsx
- Piston shall provide adequate sealing to created desired pressure load on actuation system

Door Travel Schedule & Cam Profile

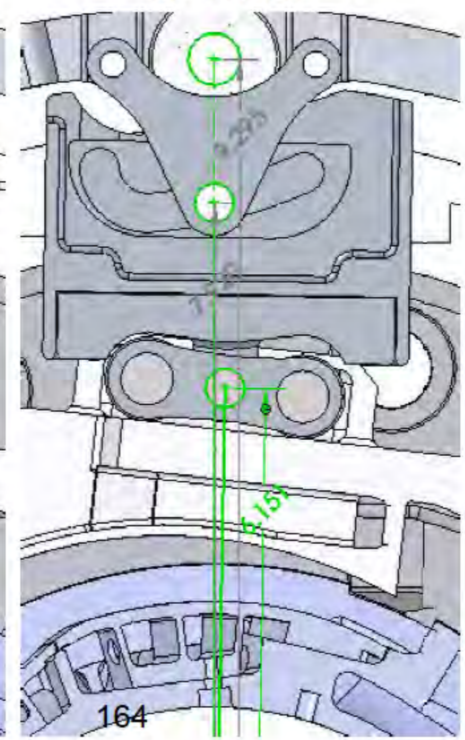
- **Door closed: piston ID roller at R5.813”**
- **Door opening locations:**
  - **5.5° door open: piston hinge at R6.151”, .338” radial move**
  - **6.6° door open: ID roller at R6.220”, .407” radial move**
- **Cam profile simplified to a single arc, provides desired Aero door travel**



Door closed:
R=6.151″



5.5° Door Open:
R=6.151″



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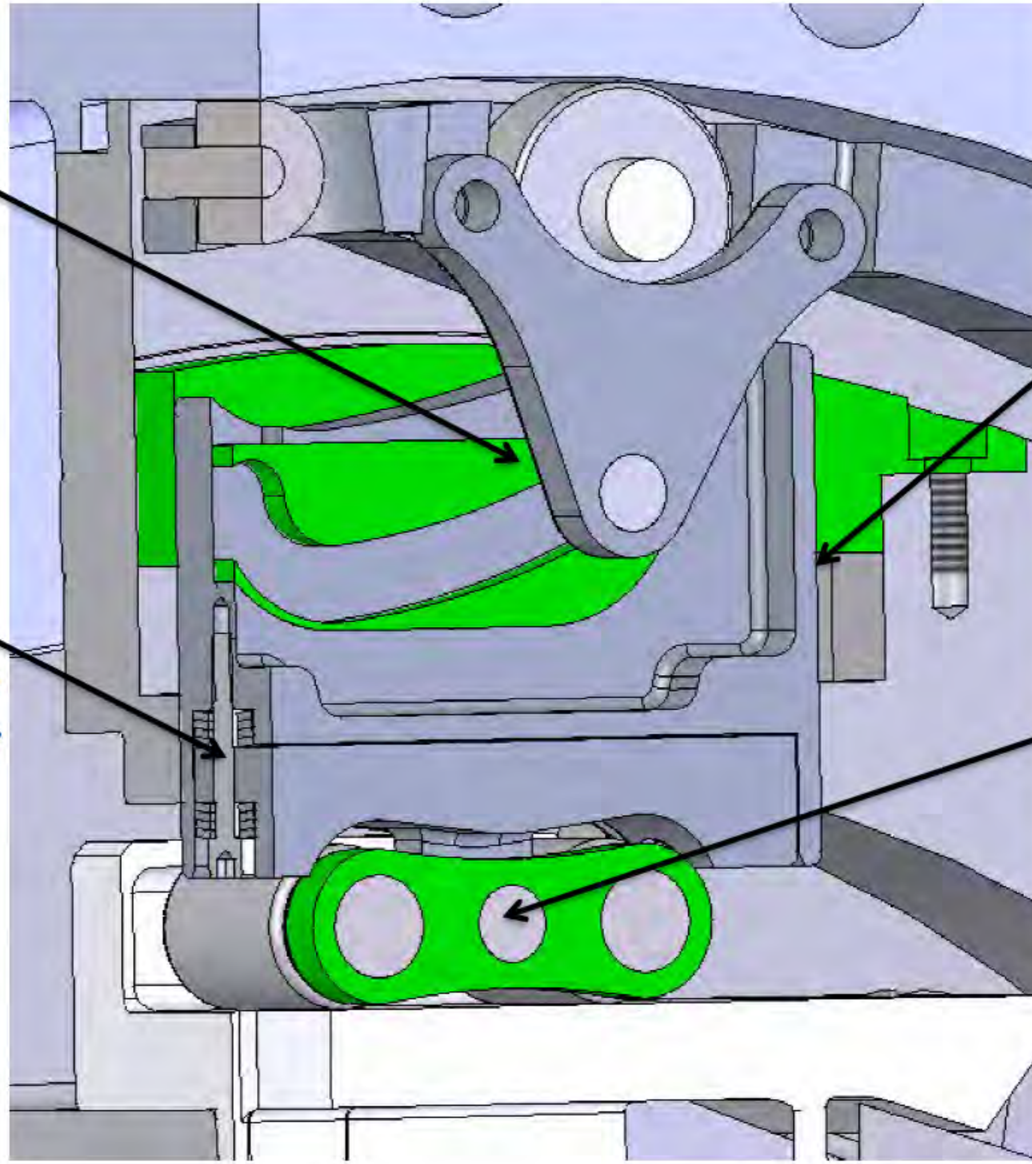
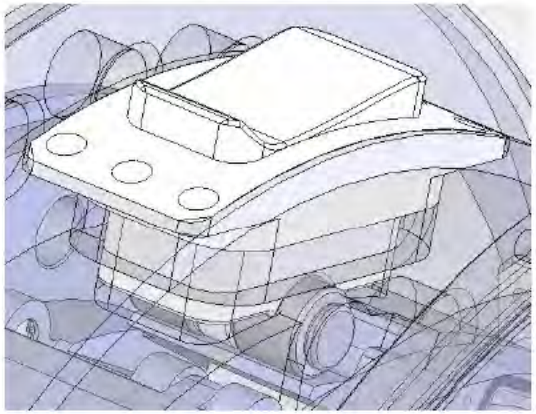
Piston Features & Functionality

- Slot plates with sliding linkage to cam ring ensures piston can overcome 1,700 lbf of seal sticktion upon door opening
- 2 – part piston with 4, inverted stacks of 4 parallel Belleville washers allow for ±.005" of radial slop in diffuser assembly
  - Not updated to diffuser 15H door loads

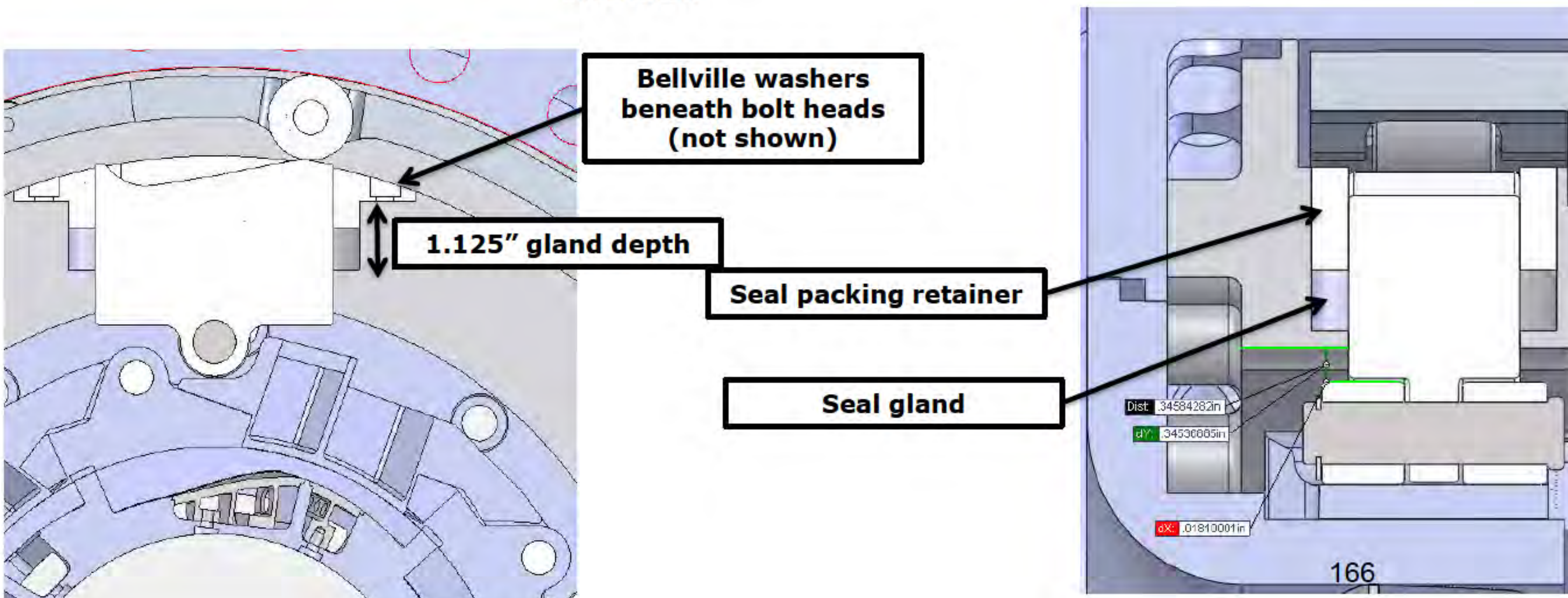


- Sealed translation reduces pressure load on cam ring rollers & actuator
- Piston hinge with roller truck accommodates 12,000 lbf door loads and 6.6° of door opening

Piston Sealing

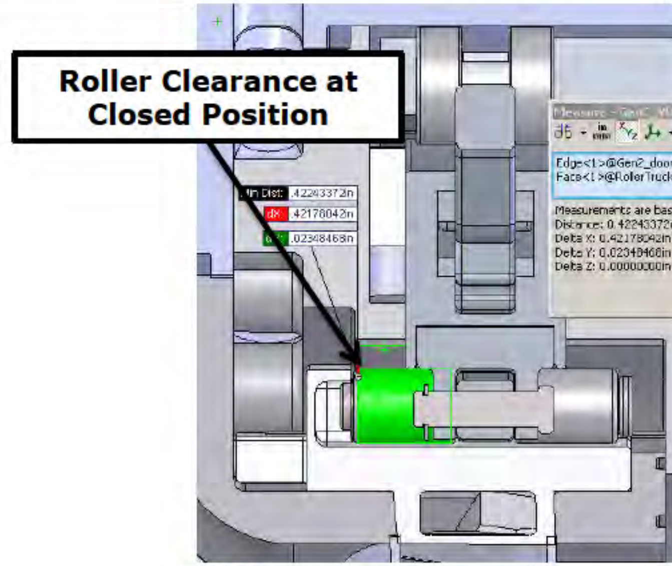


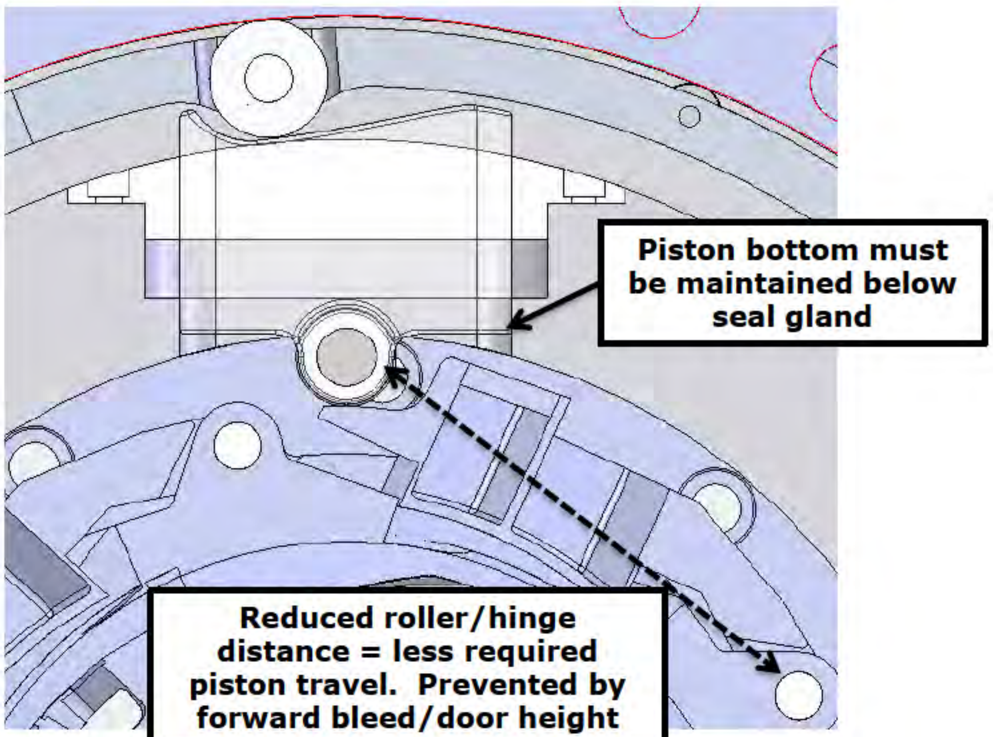
- Braided graphite die formed seal packing, similar to Ram-1 tip ring seal
- Initial conversation w/ John Crane: recommends 3x.375” cross section rings compressed ~30% (1.125” deep gland)
- Recommends bevel washers beneath bolt heads for live loading to maintain pre-load
- Predicted sticktion: 1,700 lbf (literature search, not from John Crane)
  - Endeavor to retrofit exposure test chamber to measure actual seal sticktion

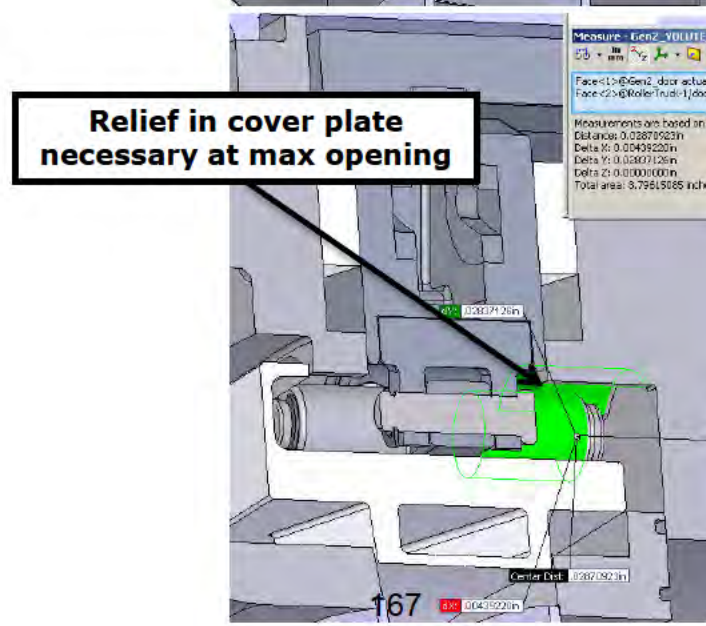


Piston Travel

- Required piston travel is determined by door rotation angle and distance from hinge
- Available radial space determined by door height, piston seal size, roller shroud clearance, roller diameter, and cam ring diameter

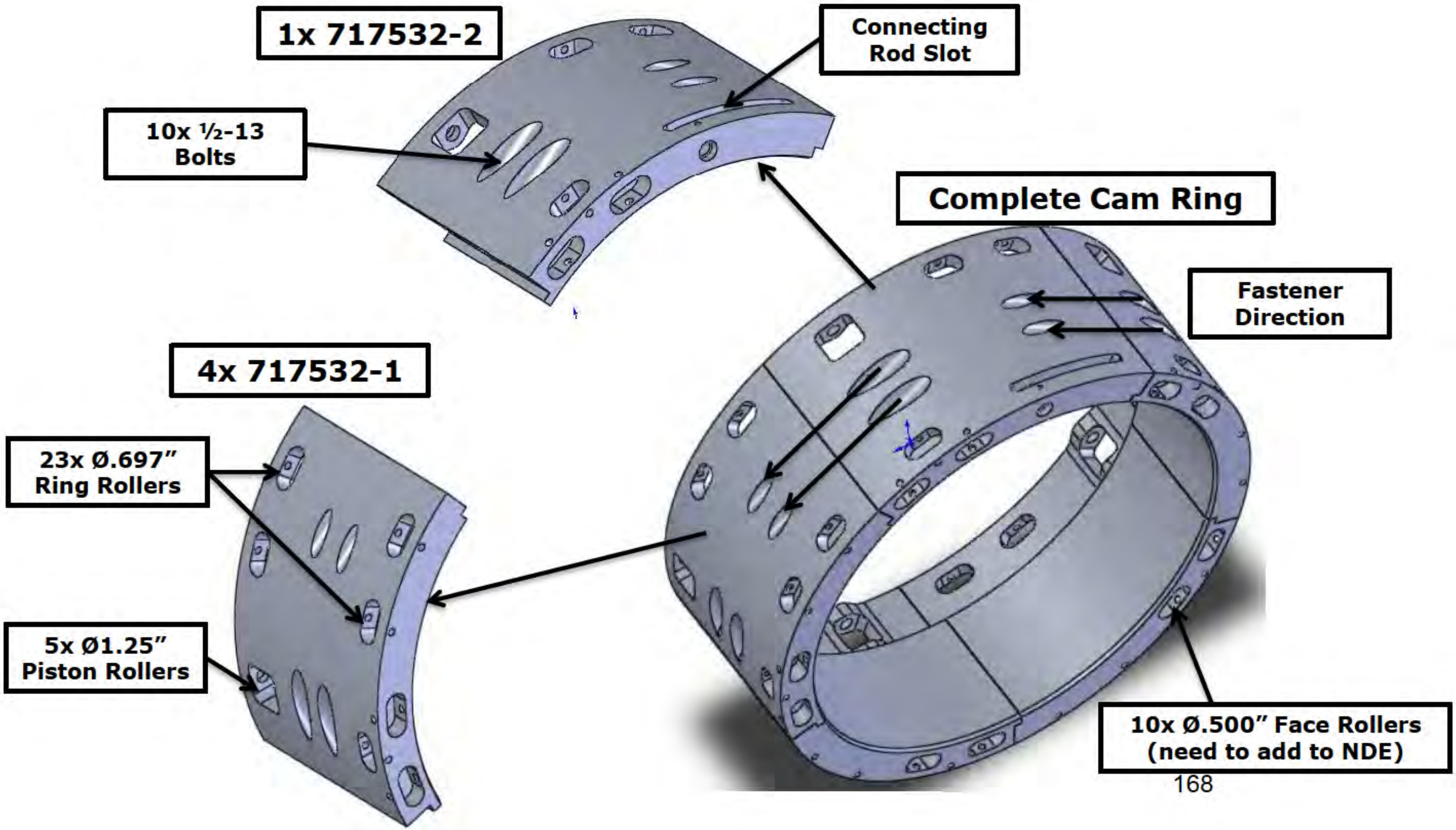






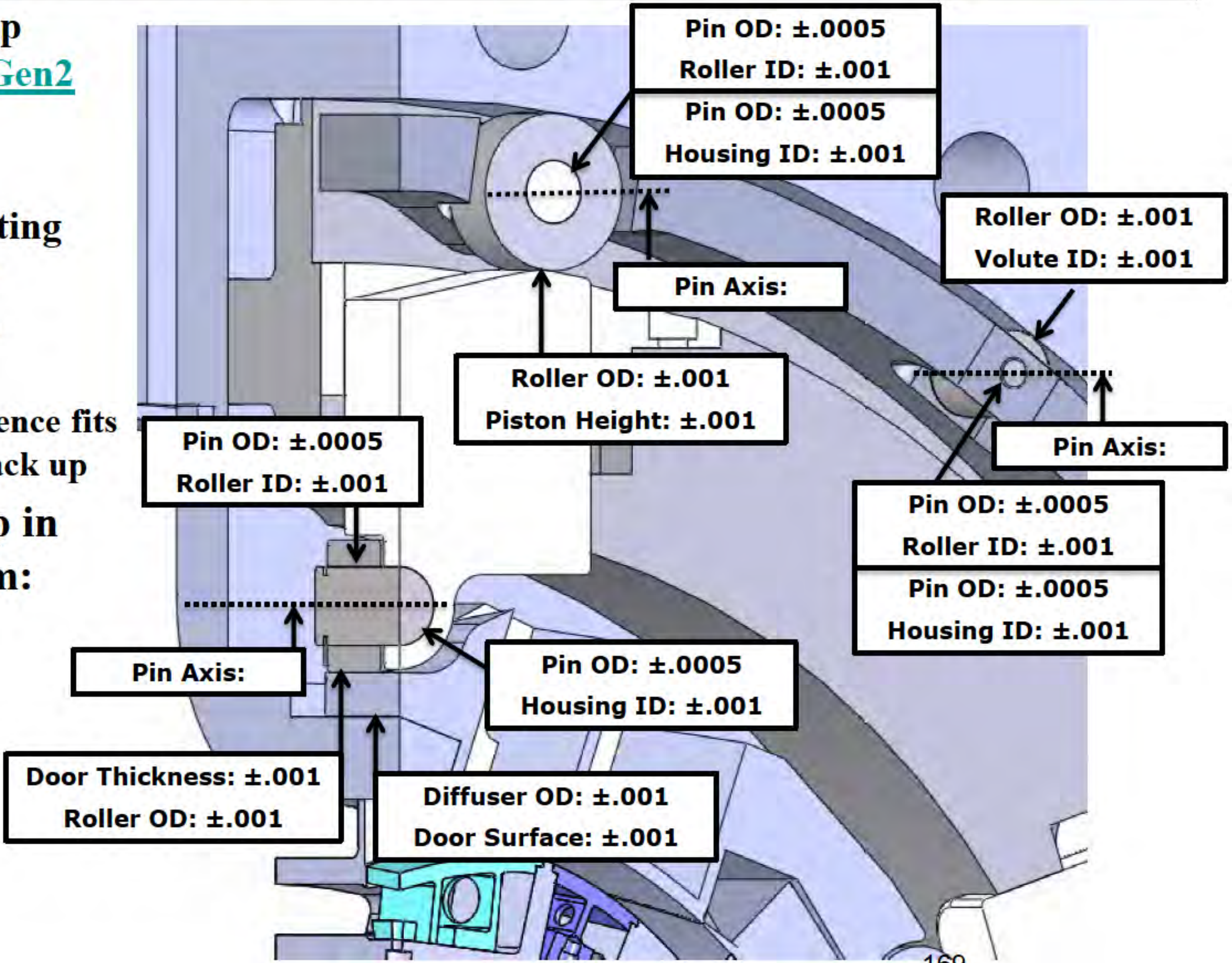
Segmented Cam Ring

- **717532 drawing linked here: ..\..\..\Demo Unit Mechanical Systems\Static Structure\Gen 2 material\component structural\Cam Actuator Ring\717532.PDF**

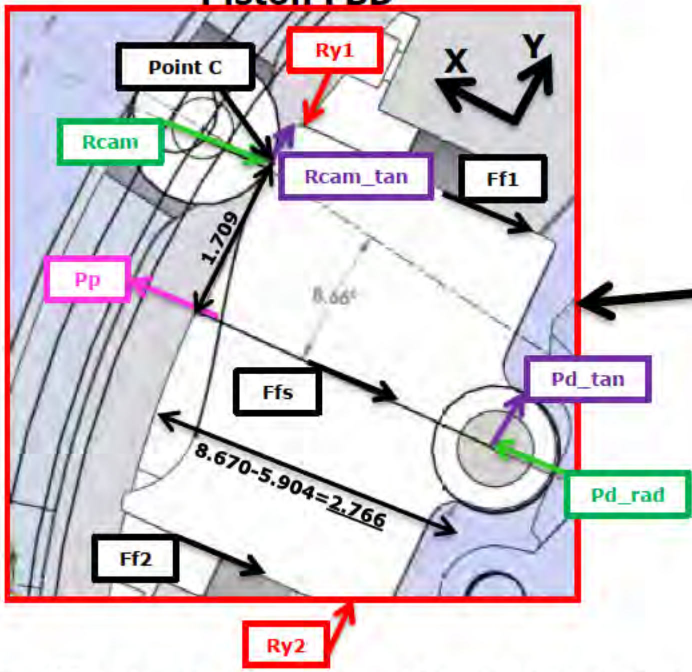


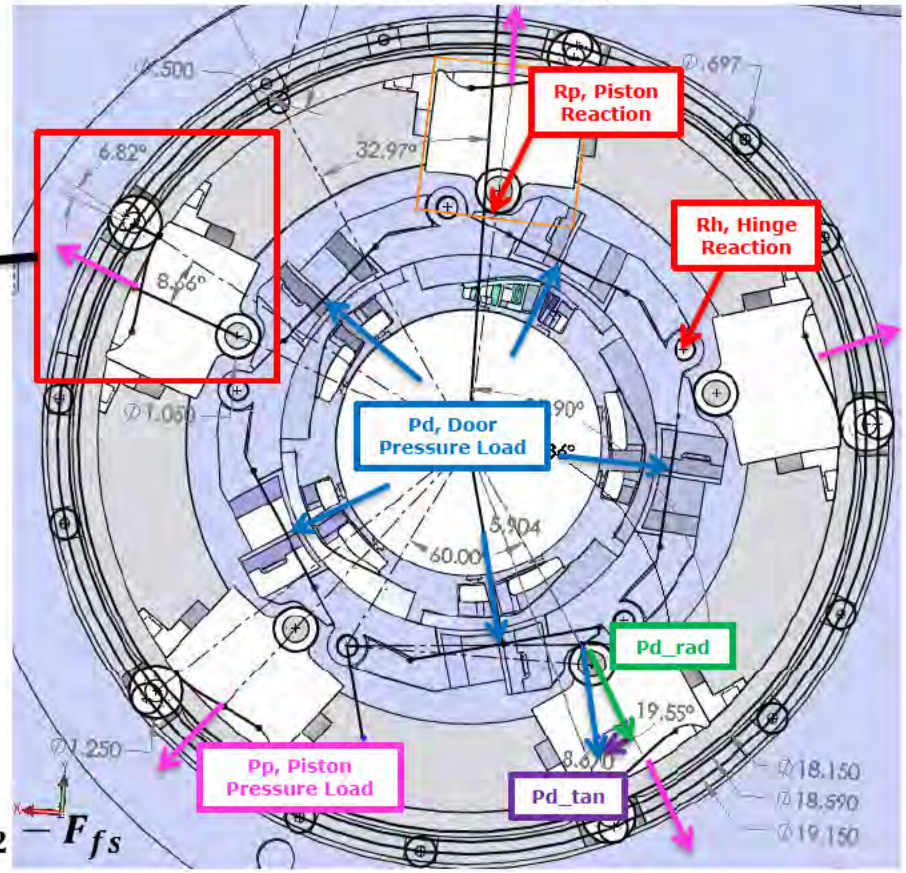
Actuation System Tolerance Stack Up

- Tolerance stack up calculated here: Gen2 Ramp Linkage Stresses.xlsx
- More than 10 mating features per door
  - Tight tolerances assumed
  - Assume interference fits do not add to stack up
- Total radial slop in actuation system:
  - Nominal: .006”
  - MMC: .002”
  - LMC: .010”



Piston & Door FBD: Door Closed





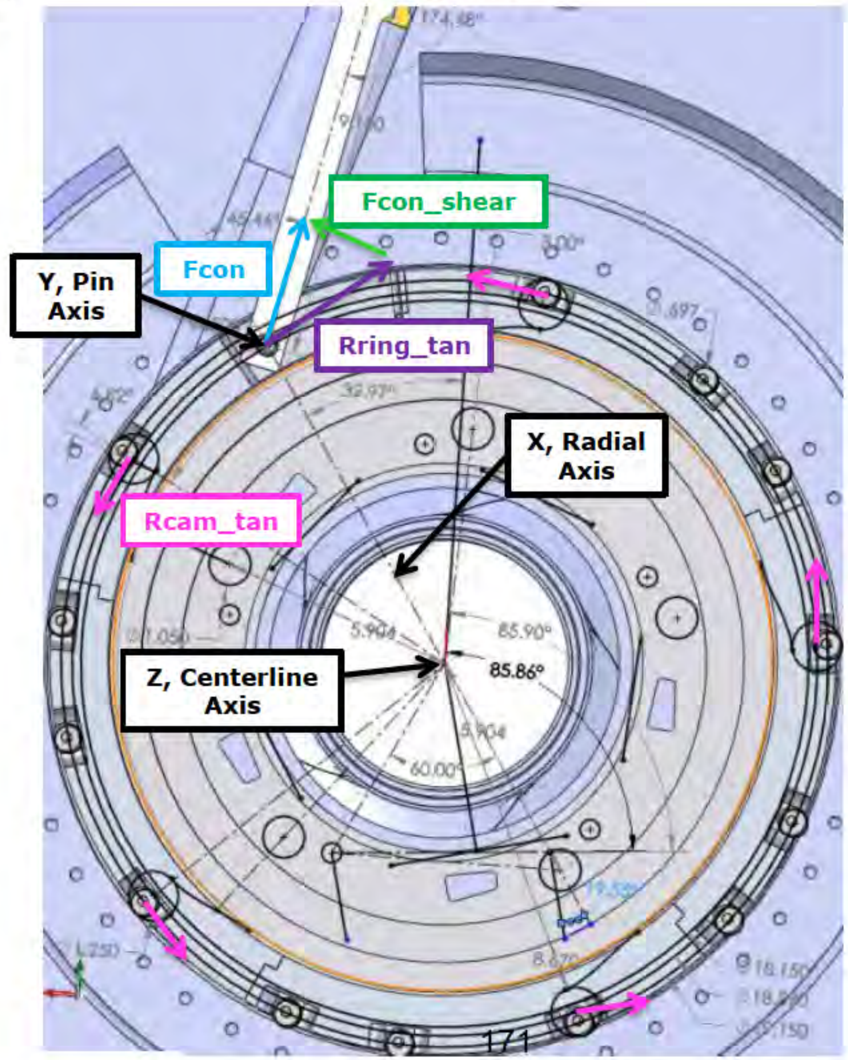
1. $\sum Fy = 0 = Pd_{tan} - Ry_1 + Ry_2 + Rcam_{tan}$
   - Where: $Rcam_{tan} = Rcam * tan(6.82°)$
   - $\therefore, Ry_1 = Pd_{tan} + Ry_2 + Rcam * tan(6.82°)$
2. $\sum Fx = 0 = Pd_{rad} + Pp - Rcam - F_{f1} - F_{f2} - F_{fs}$
   - Where: $F_{fi} = Ry_i * \mu$
   - $\therefore, Rcam = Pd_{rad} - Pp - \mu * (Ry_1 + Ry_2)$
3. $\sum M_C = 0 = -.713 * F_{f1} - .125 * Ry_1 - 1.709 * (Pd + Pd_{rad} - F_{fs}) + 2.766 * Pd_{tan} - 3.106 * F_{f2} + (2.766 - .25) * Ry_2$
   - Substitute 1. & 2, into 3, solving for Ry_2

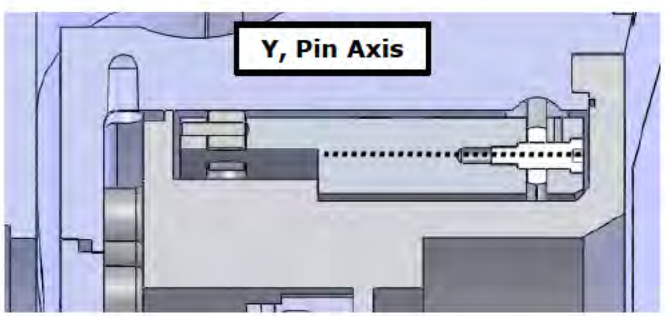
- $Pd = (P_{flowpath} - P_{bleed}) * A_{door}$
- $Pd_{rad} = Pd * cos(19.55°)$
- $Pd_{tan} = Pd * sin(19.55°)$
- $Pp = (P_{bleed} - P_{exhaust}) * A_{piston}$

Ring & Connecting Rod FBD: Door Closed

- Ringloading
  - $\sum M_Z = 0 = 5 * Rcam_{tan} * \frac{18.59}{2} - Rring_{tan} * \frac{18.15}{2}$
    - $\therefore, Rring_{tan} = 5 * Rcam_{tan} * \frac{18.59}{18.15}$
  - $\sum M_X = 0 = M_x - Rring_{tan} * (6.125 - .594)$
    - $\therefore, M_x = Rring_{tan} * (6.125 - .594)$
  - $\sum M_Y = 0 = M_Y + 5 * Rcam_{tan} * \left(\frac{18.59}{2} - \frac{18.15}{2}\right)$
    - $\therefore, M_y = -5 * Rcam_{tan} * \left(\frac{18.59}{2} - \frac{18.15}{2}\right)$
- **Connecting rod loading**
  - $Fcon = Rring_{tan} * cos(90 - 45.44)$
  - $Fcon_{shear} = Rring_{tan} * sin(90 - 45.44)$

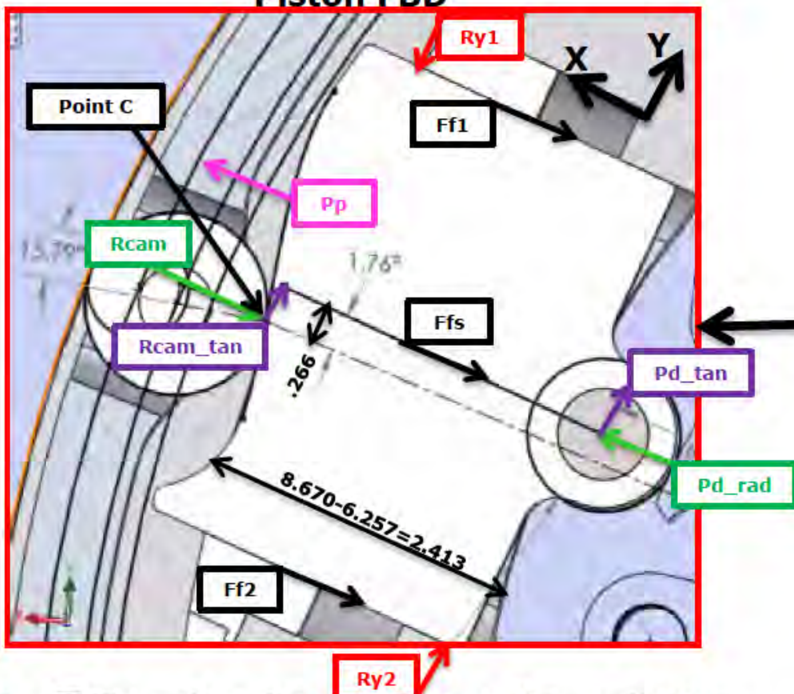
Ring FBD



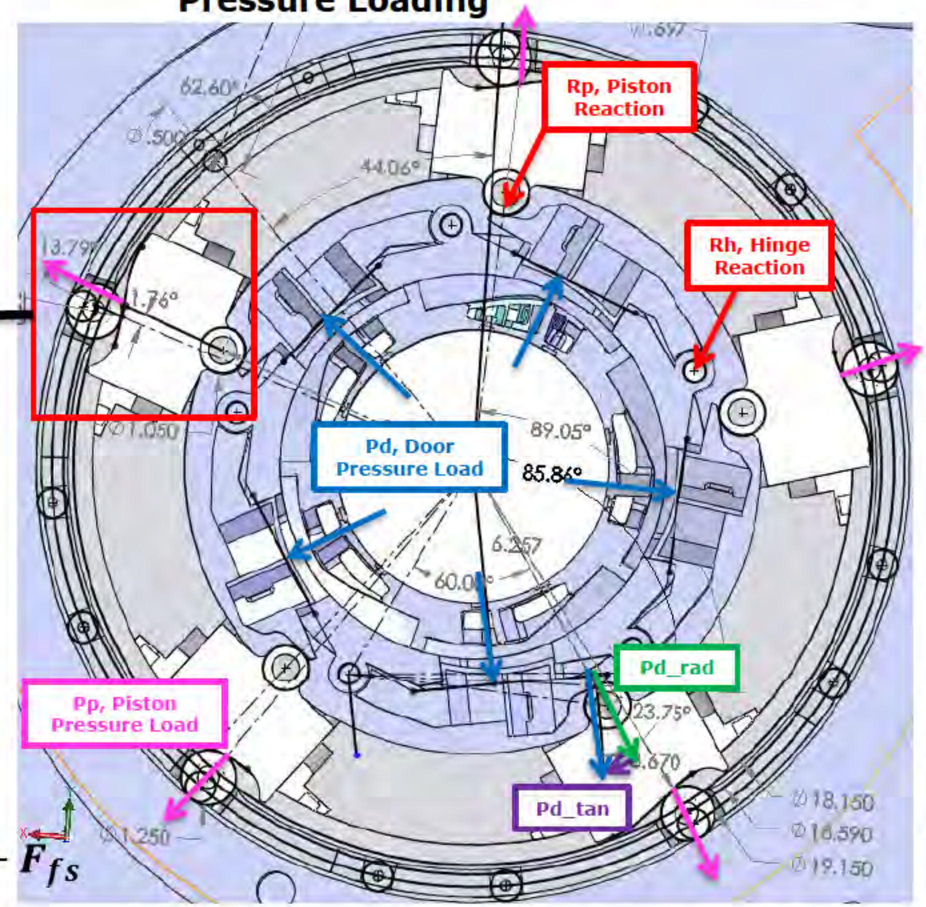


Piston & Door FBD: Door Open

Piston FBD



Pressure Loading



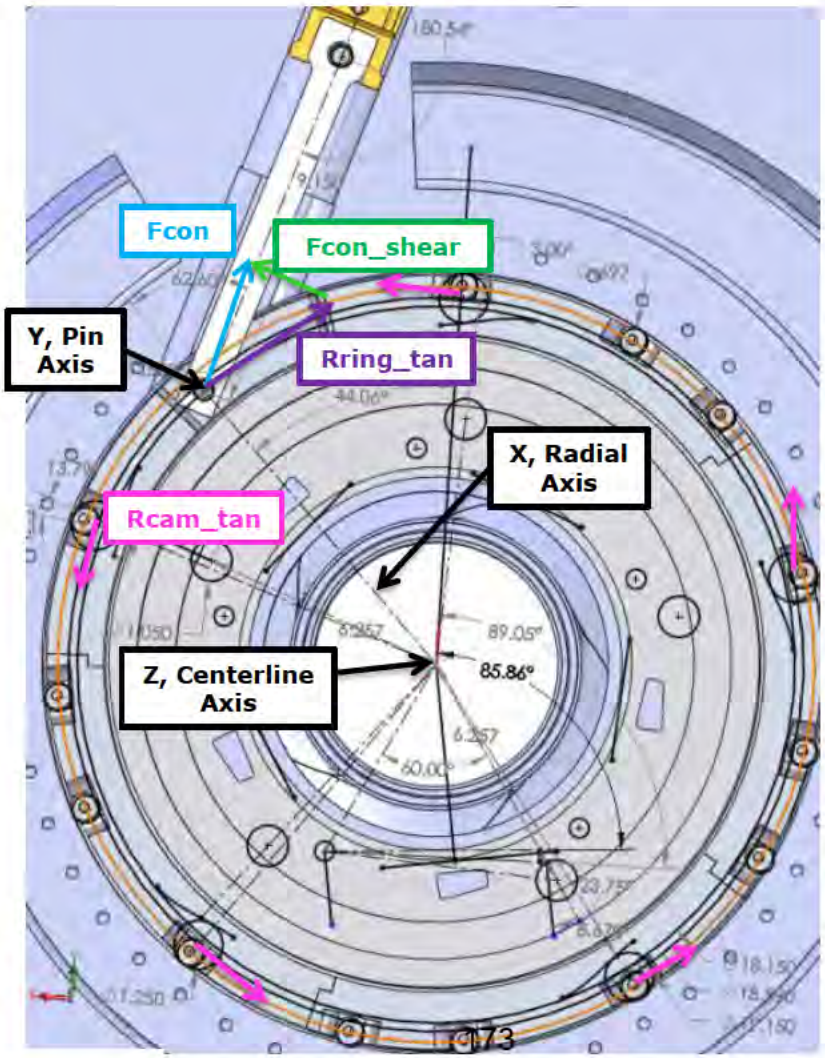
1. $\sum Fy = 0 = Pd_{tan} - Ry_1 + Ry_2 + Rcam_{tan}$
   - Where: $Rcam_{tan} = Rcam * tan(13.79°)$
   - $\therefore, Ry_1 = Pd_{tan} + Ry_2 + Rcam * tan(13.79°)$
2. $\sum Fx = 0 = Pd_{rad} + Pp - Rcam - F_{f1} - F_{f2} - F_{fs}$
   - Where: $F_{fi} = Ry_i * \mu$
   - $\therefore, Rcam = Pd_{rad} - Pp - \mu * (Ry_1 + Ry_2)$
3. $\sum M_C = 0 = -2.688 * F_{f1} - .125 * Ry_1 + .266 * (Pd + Pd_{rad} - F_{fs}) + 2.413 * Pd_{tan} + 1.131 * F_{f2} + (2.413 - .25) * Ry_2$
   - Substitute 1. & 2, into 3, solving for Ry_2

- $Pd = (P_{flowpath} - P_{bleed}) * A_{door}$
- $Pd_{rad} = Pd * cos(23.75°)$
- $Pd_{tan} = Pd * sin(23.75°)$
- $Pp = (P_{bleed} - P_{exhaust}) * A_{piston}$

Ring & Connecting Rod FBD: Door Open

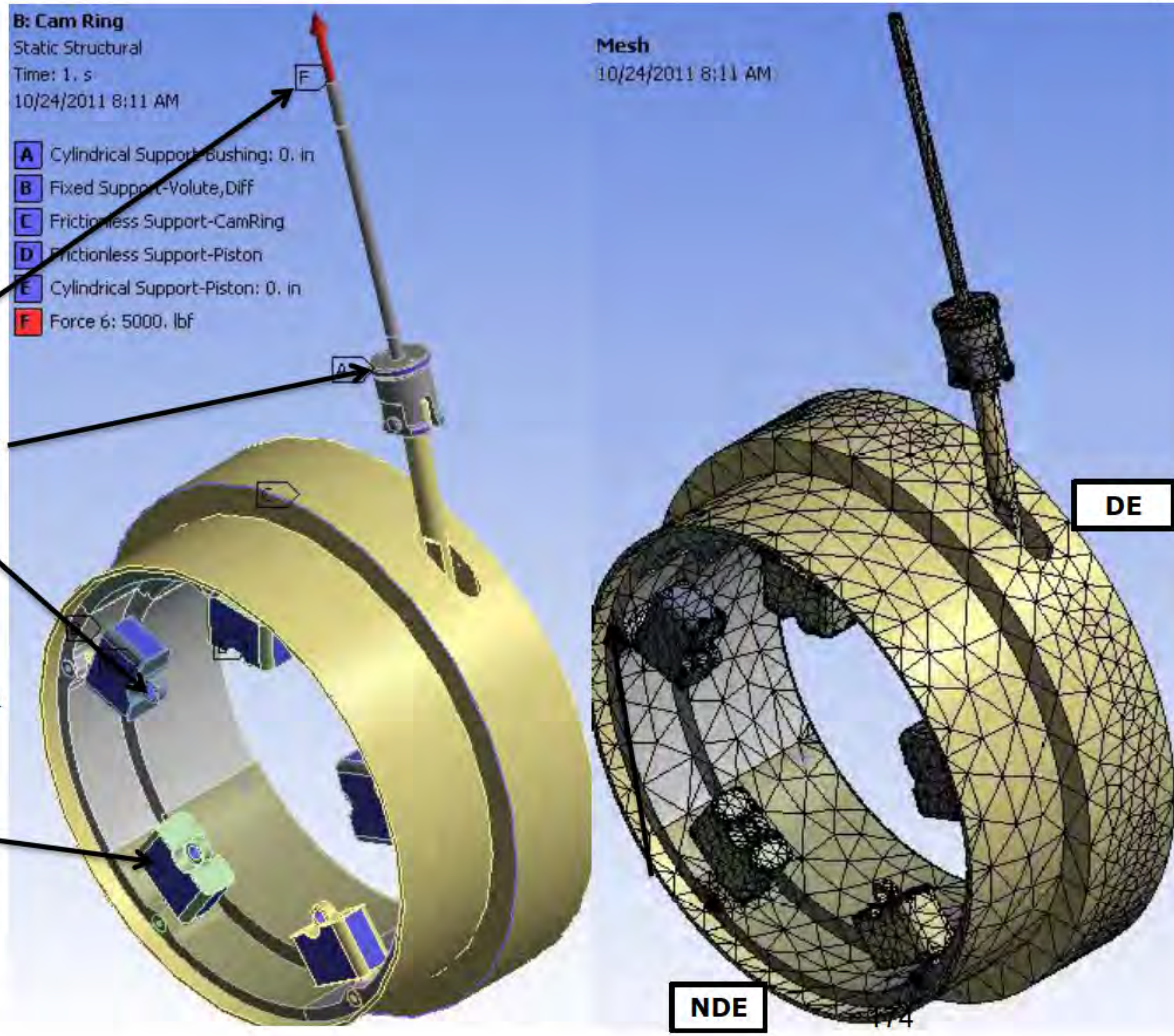
- Ring loading
  - $\sum M_Z = 0 = 5 * Rcam_{tan} * \frac{18.59}{2} - Rring_{tan} * \frac{18.15}{2}$
    - $\therefore, Rring_{tan} = 5 * Rcam_{tan} * \frac{18.59}{18.15}$
  - $\sum M_X = 0 = M_x - Rring_{tan} * (6.125 - .594)$
    - $\therefore, M_x = Rring_{tan} * (6.125 - .594)$
  - $\sum M_Y = 0 = M_Y + 5 * Rcam_{tan} * \left(\frac{18.59}{2} - \frac{18.15}{2}\right)$
    - $\therefore, M_y = -5 * Rcam_{tan} * \left(\frac{18.59}{2} - \frac{18.15}{2}\right)$
- Connecting rod loading
  - $Fcon = Rring_{tan} * cos(90 - 62.60)$
  - $Fcon_{shear} = Rring_{tan} * sin(90 - 62.60)$
- Ring travel
  - $Arc = R * \theta = \frac{18.15}{2} * (44.06° - 32.97°) * \frac{\pi}{180}$
  - θ = 11.06° (14.14° at 20% margin location)
  - Arc = 1.7565” (2.2396 “at 20% margin location)
- Actuator Axial Travel
  - 1.421” (1.858” at 20% margin location)

Ring FBD



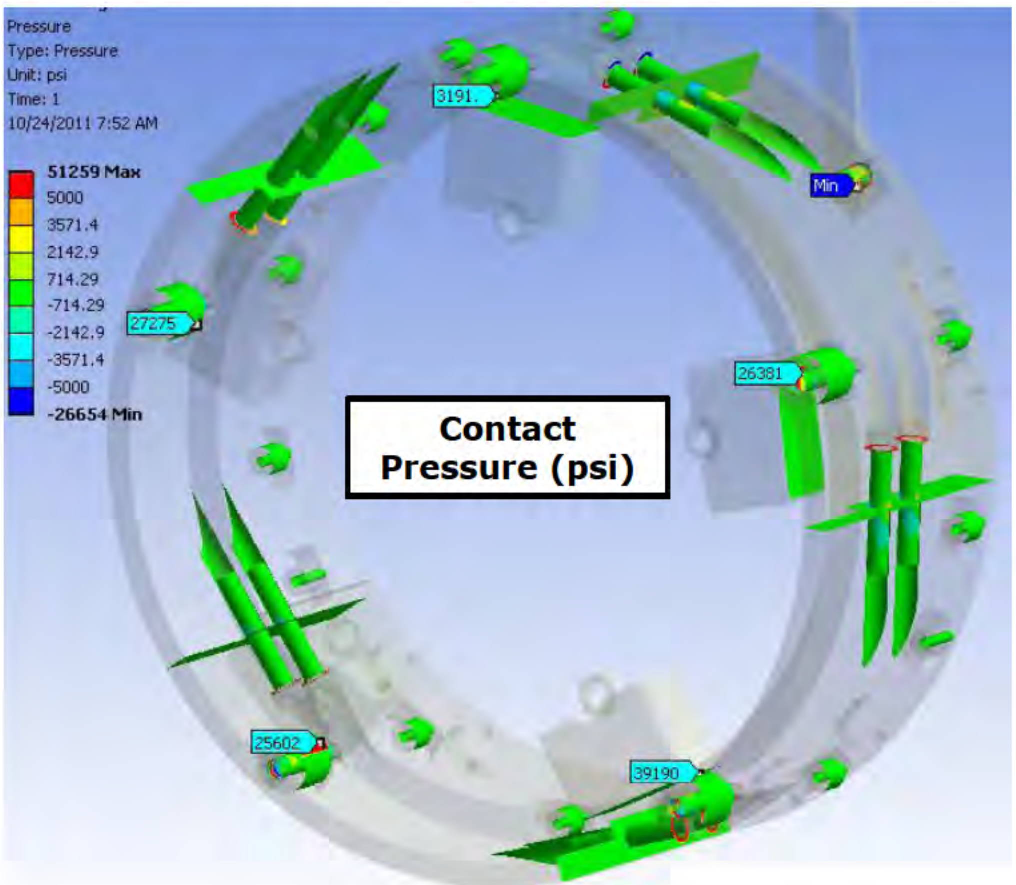
Actuation System: Kinematic Model & BC's

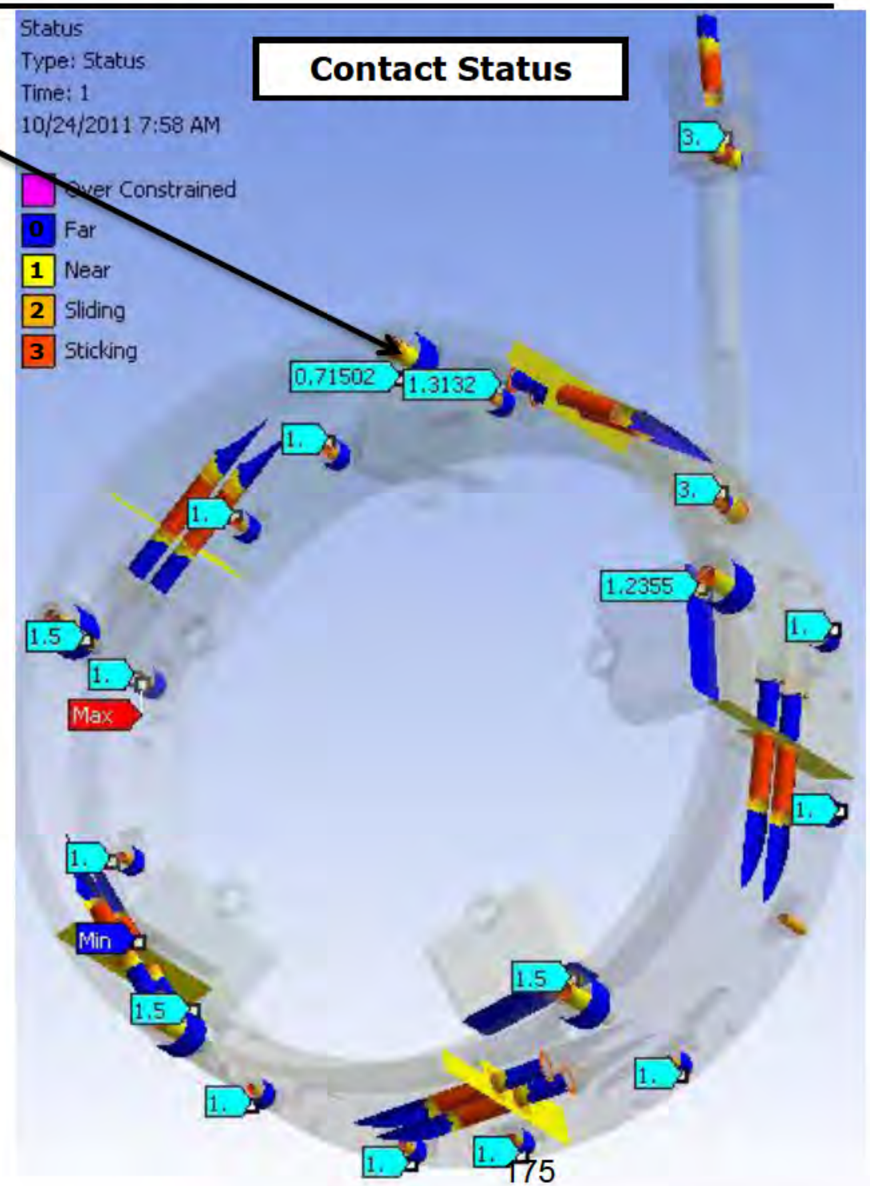
- Mesh (coarse)
  - 392,556 nodes
  - 174,436 elements
- BC's
  - Doors/pistons modeled in "closed" position
  - 5,000 lbf actuator "pull"
  - Bushing radial support
  - Pistons fixed at ID roller location
  - Frictionless contact on all roller surfaces
  - Frictionless contact b/n cam ring segments with bonded bolts
  - Piston walls held with frictionless supports
- This model used to check segmented cam ring loading



Segmented Cam Ring: Roller Contact

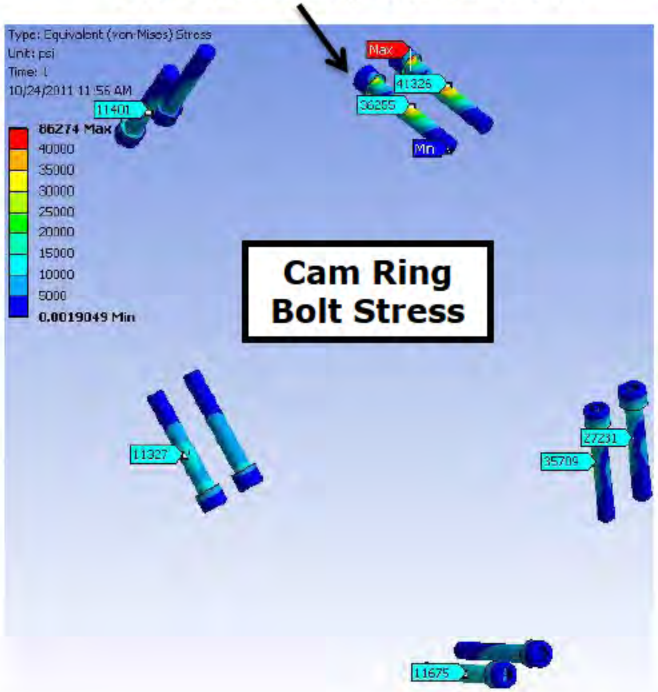
- TDC piston roller in weak contact
  - Contact pressure: 3 ksi, contact status < 1
- All other rollers in good contact
  - Contact pressure > 20 ksi, contact status ≥ 1

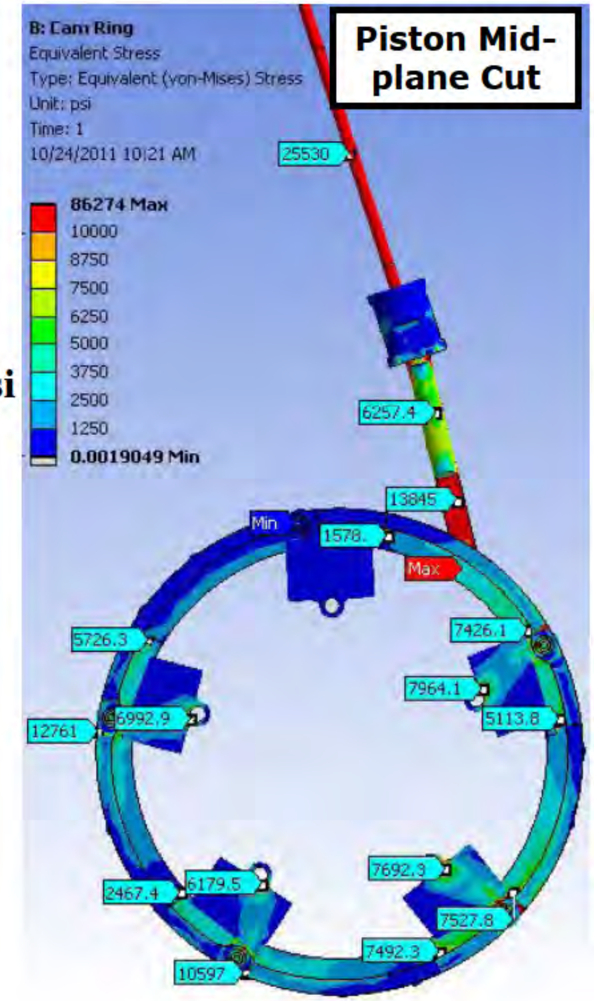




Segmented Cam Ring: Equivalent Stress (psi)

- TDC piston in weak contact
  - Piston, roller stress < 1 ksi
- All other pistons < 8 ksi
- All other rollers < 13 ksi
- Cam ring stress < 10 ksi
- Cam ring bolt stress < 45 ksi
- Con rod peak stress: 14 ksi, pin > 80 ksi
- Threaded rod peak stress: 25 ksi
  - TDC bolts carry the most load





Budget and Schedule

- **FDR date**
  - **January 13, 2011**
- **Drawing Release date (no later than)**
  - **717008, Piston, 2/3/2012**
  - **717013, Packing Retainer, 2/3/2012**
  - **717512, completed under Gen1**
  - **717532, Cam Ring Segment, 2/3/2012**
  - **Piston Link, 2/3/2012**
  - **Door Bogie, 2/3/2012**

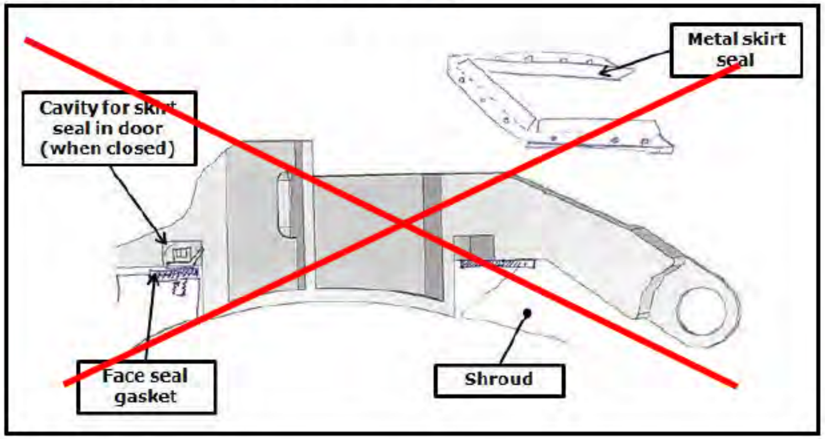
Current budgetary part cost estimates

717007	Diffuser shroud Door	$5,000.00
717008	Shroud Door Piston	$1,500.00
717508	Adapter bracket, diffuser bypass actuator	$576.00
717517	Adapter bracket swivel plate, diffuser bypass actua	$675.00
717524	Bypass actuator threaded rod	$400.00
717527	Bypass External Actuator Rod	$400.00
717532	Gen2 Ramp Actuator Ring	$18,000.00
717012	Load ring, actuator door	$350.00
717013	Piston seal preload plate	$500.00
717019	Load block, actuator door	$150.00
717020	Shroud door piston roller	$85.00
717021	Axle, shroud door piston	$85.00
717022	Shroud door, forward bleed seal	$250.00
717023	Shroud door piston seal	$250.00
717015	Shroud door hinge axle, NDE	$0.00
717016	Shroud door hinge axle, DE	$0.00

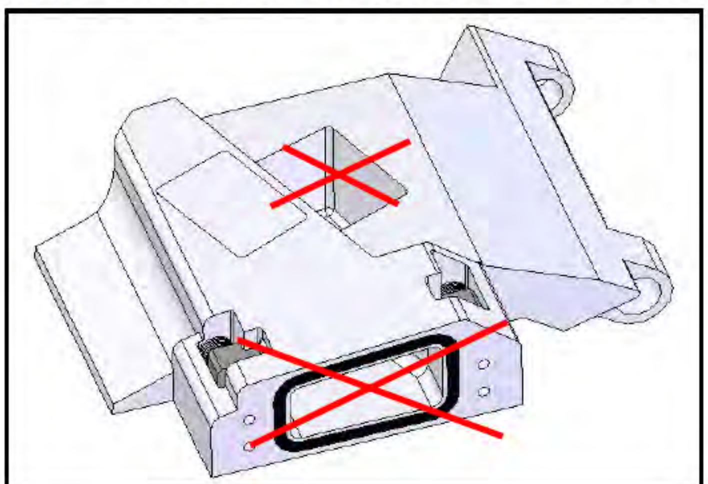
- **Estimated Manufacturing Time/Delivery date**
  - **March 27, 2012**
- **Is schedule achievable? Yes**
- **Current manufacturing lead time and cost/budget adequate?**
  - **Current estimates to be qualified based on preliminary drawing release post PDR. Current component complexity indicates 8 weeks of manufacturing is adequate**

Design Updates From Case 15 PDR

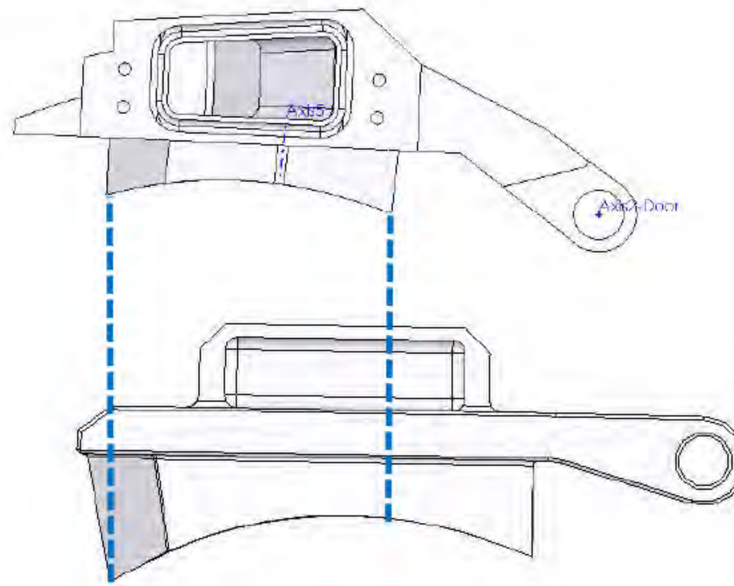
Door face seal unnecessary:



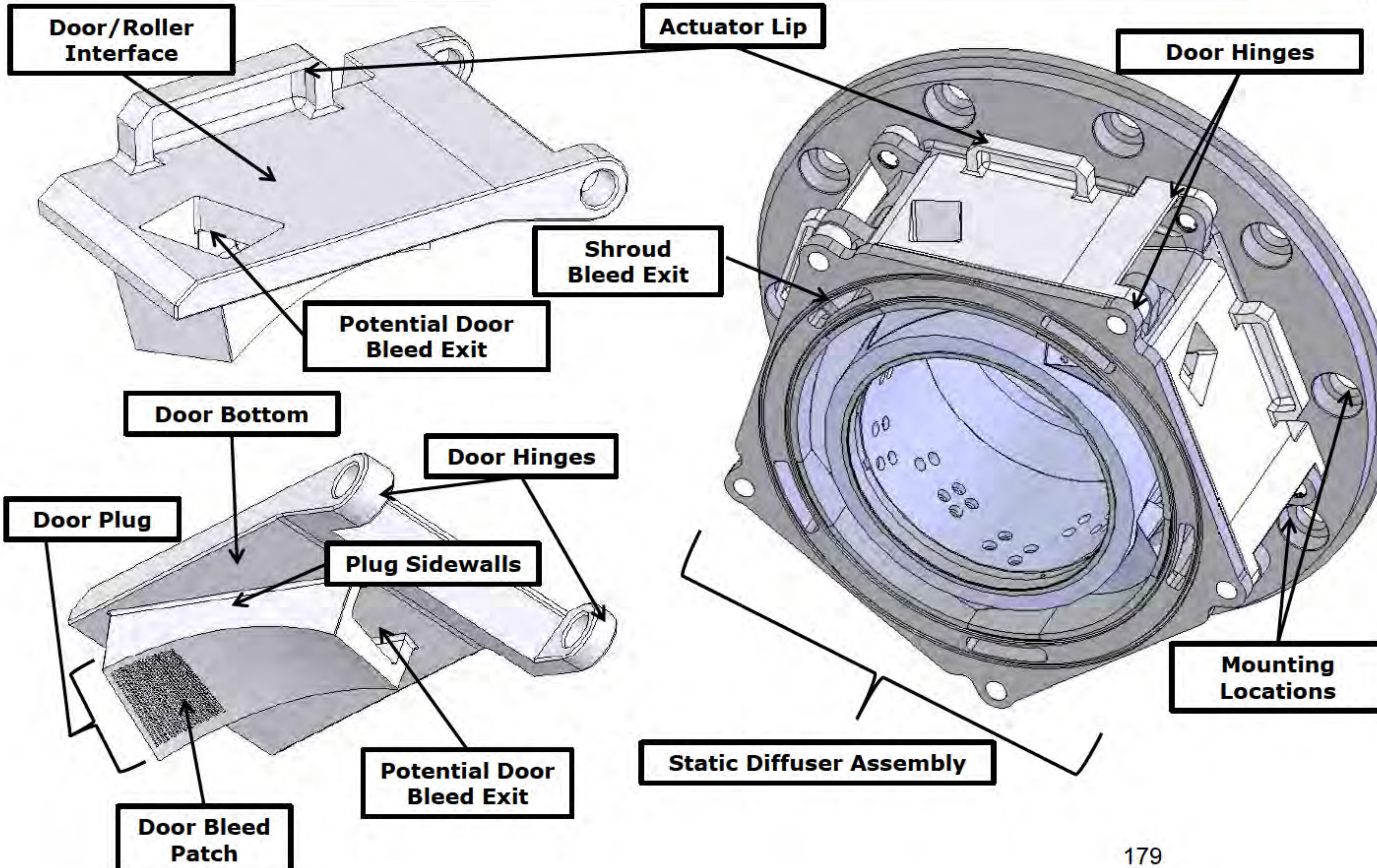
Aft bleed and forward bleed seal eliminated:



Large increase in door length and wetted flow path area:

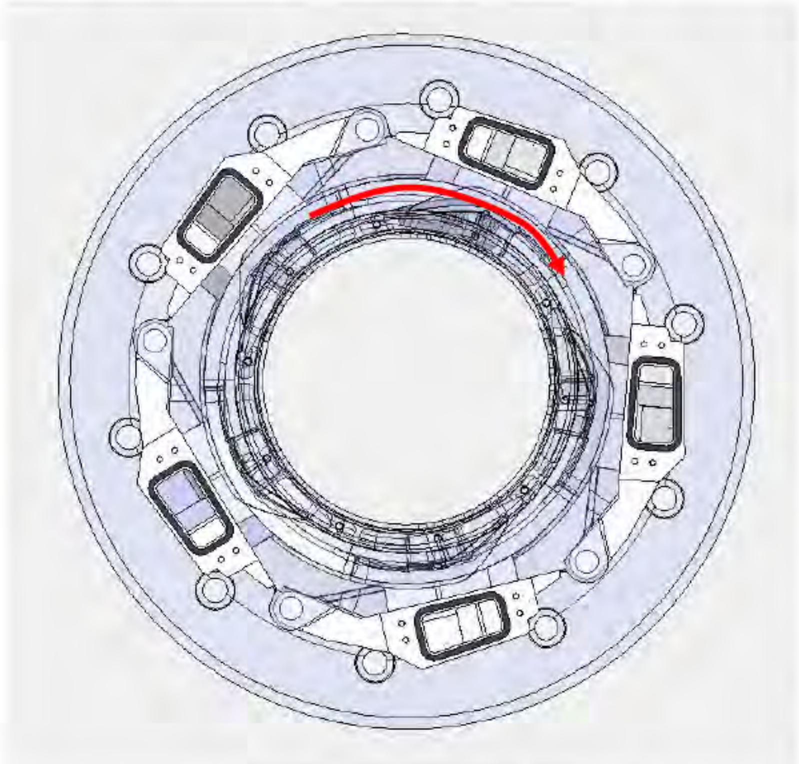


Shroud Starting Door

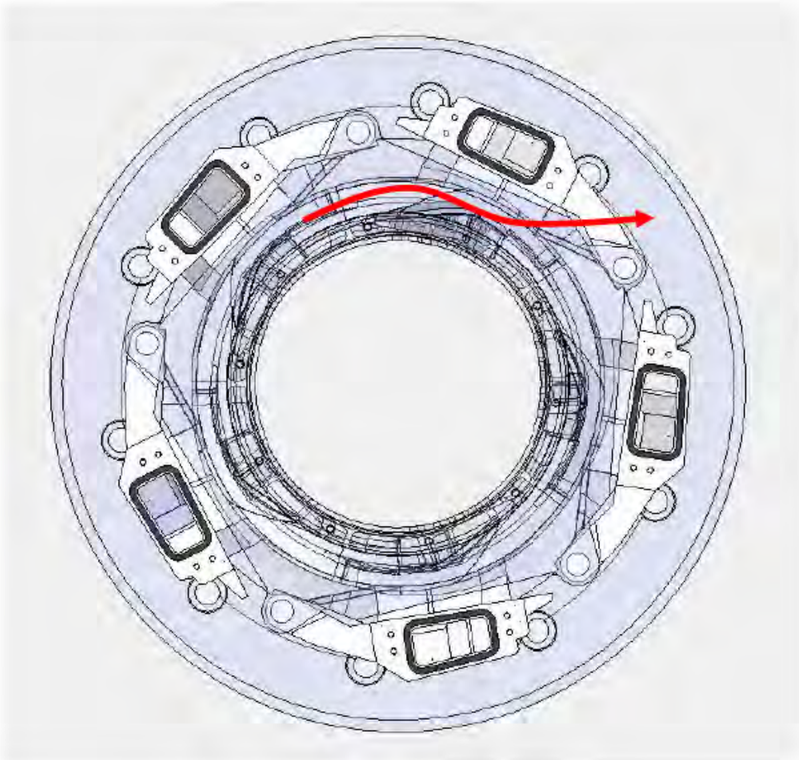


Shroud Starting Bleed

Note: Case 15 shroud and door depicted



Starting Doors Closed

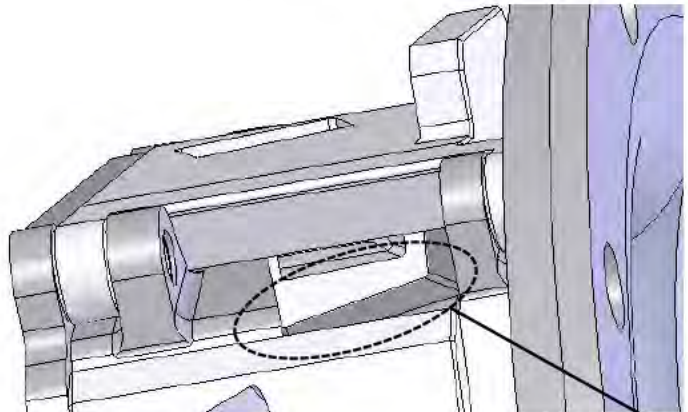


Starting Doors Open

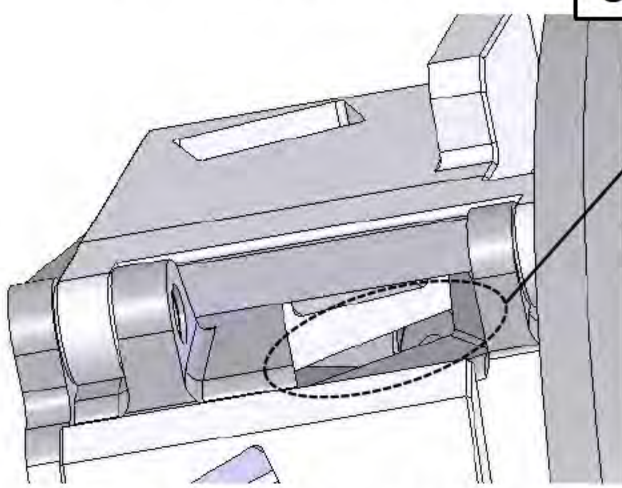
- **Target opening area (defined by aero) reached at ~5.5 deg of rotation for current config**
- **20% area margin at starting bleed exit reached at ~6.6 deg of rotation**

Shroud Starting/Aft Bleed

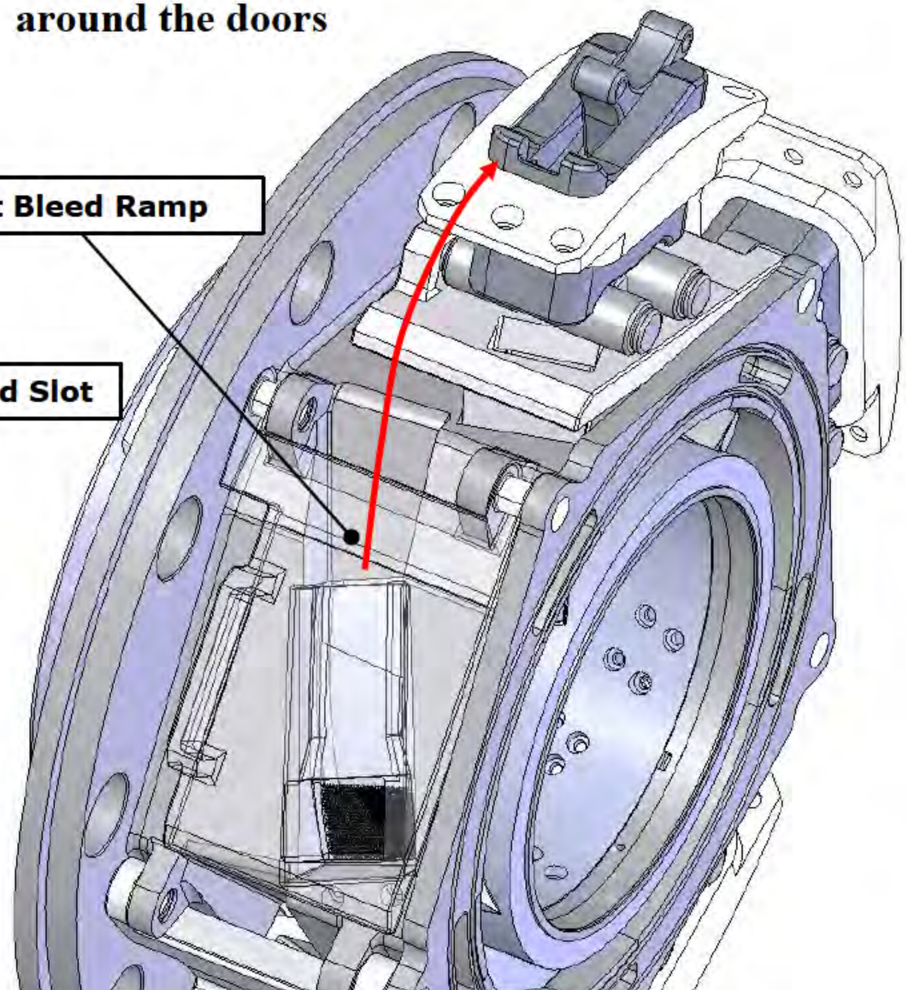
- **When doors open, starting bleed flows up the bleed ramp into an annulus region around the doors**



Starting Doors Closed

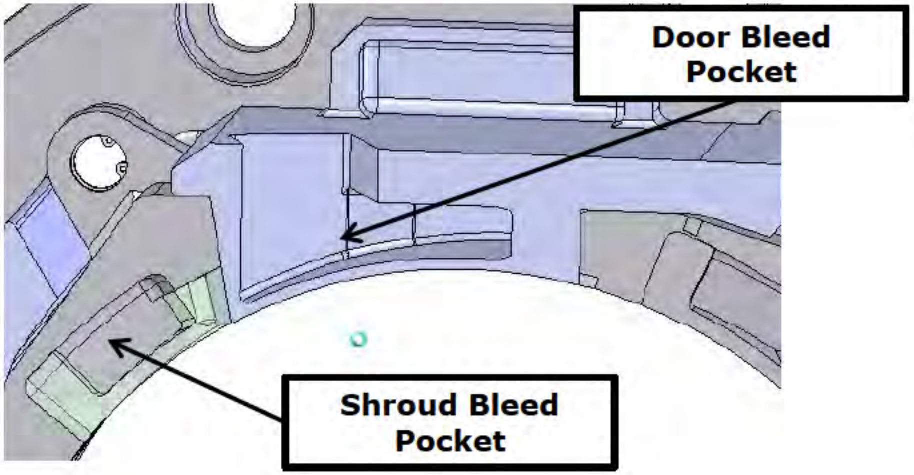


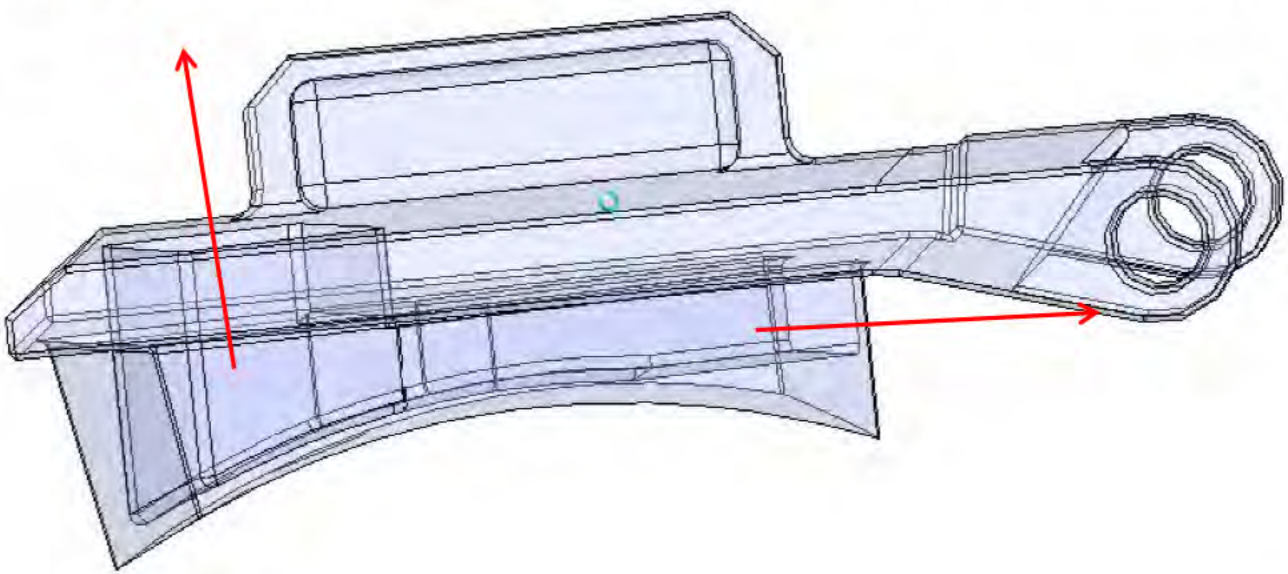
Starting Doors Open



Performance Bleed Routing in Door

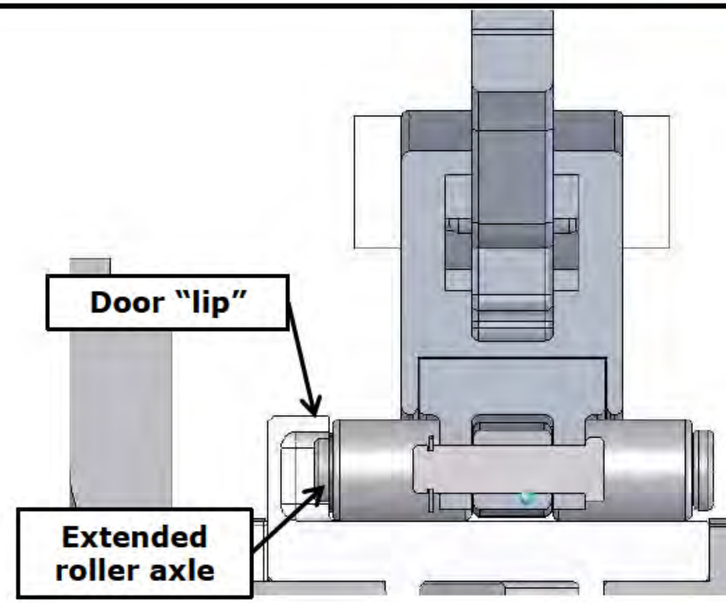
- Door bleed dumps into the annulus used for shroud starting bleed
- Currently pursuing two paths: out the top of the door or out the back via the starting bleed ramp
- Configuration determined by placement of door/piston rollers
- Shroud bleed is just upstream of the door LE

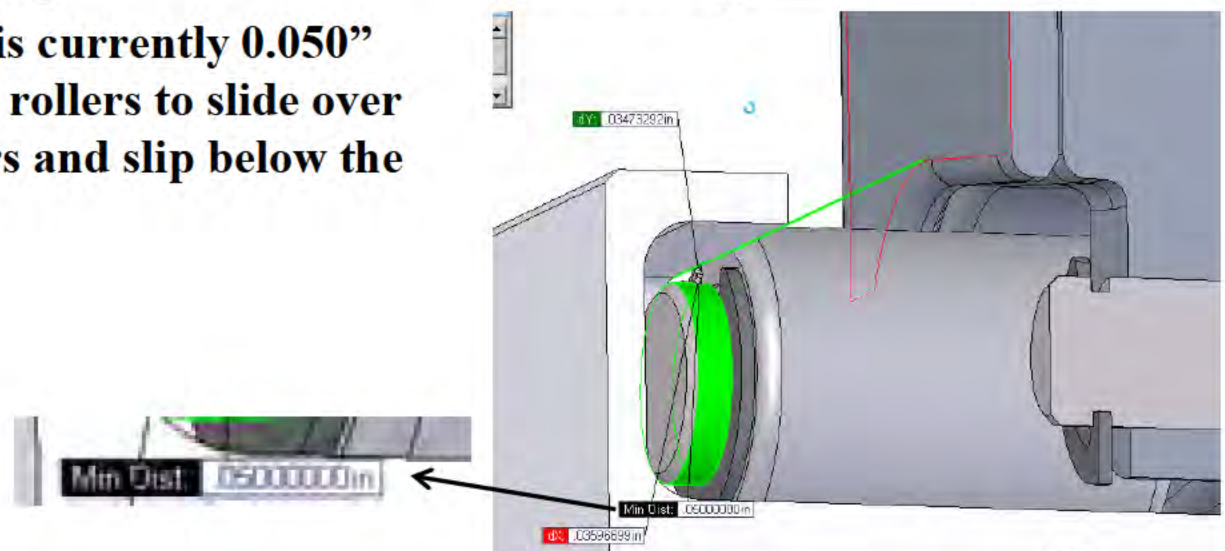




Door Pull Open Concept

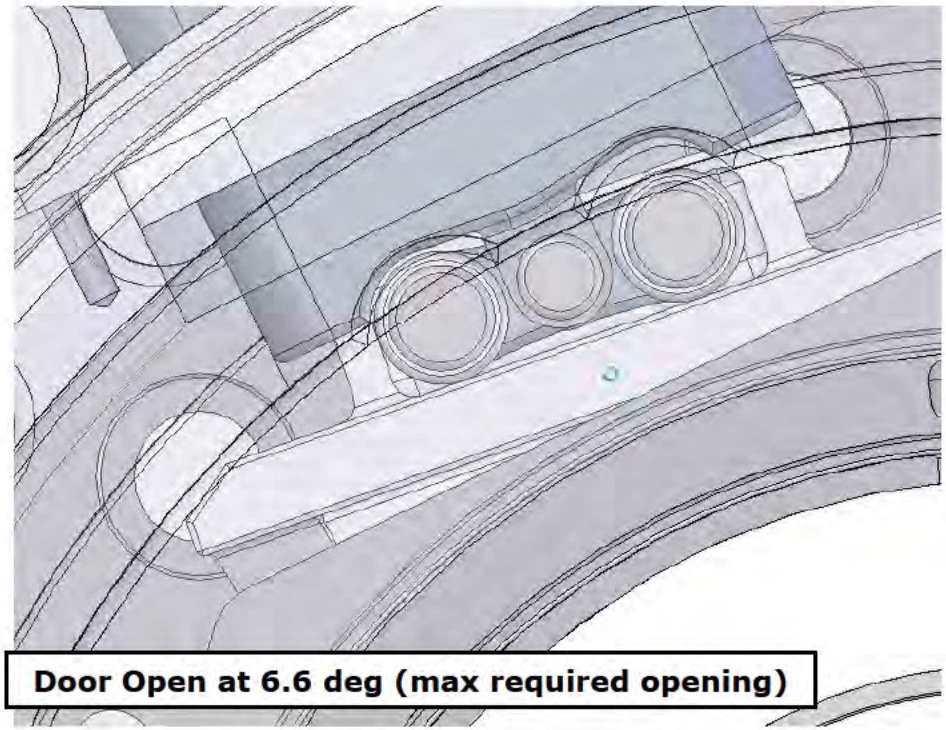
- Concept proposed to extend roller axle to engage with a lip that extends from the door and wraps around the axle
- Allows pistons to pull the doors radially outward
- Would require pistons to be "down" during assembly so pin could engage with door hook. This will make assembly challenging
- As modeled there is currently 0.050" total clearance for rollers to slide over the top of the doors and slip below the engagement lip





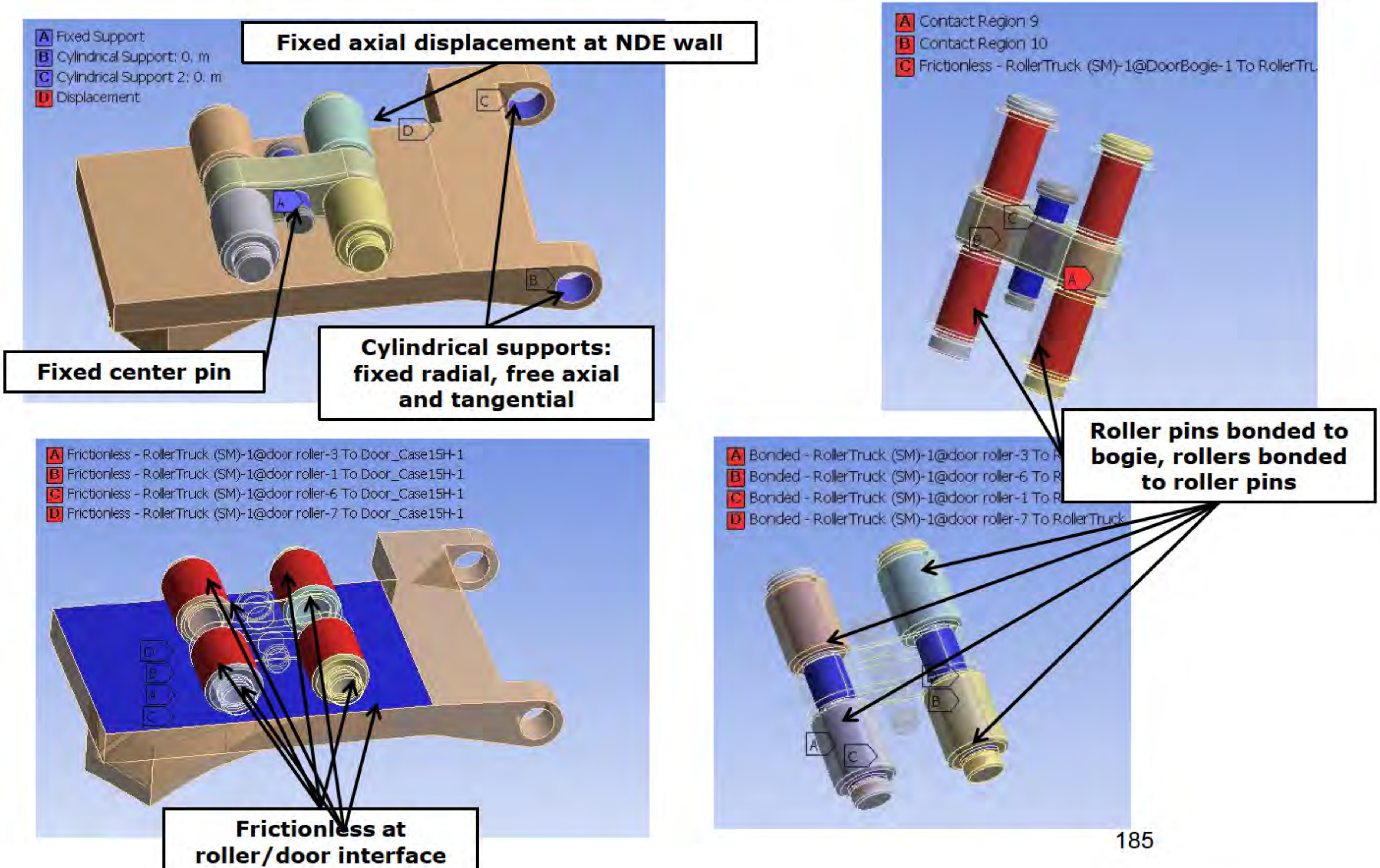
Currently 0.050" assembly clearance

Piston/Door Radial Clearance Issue

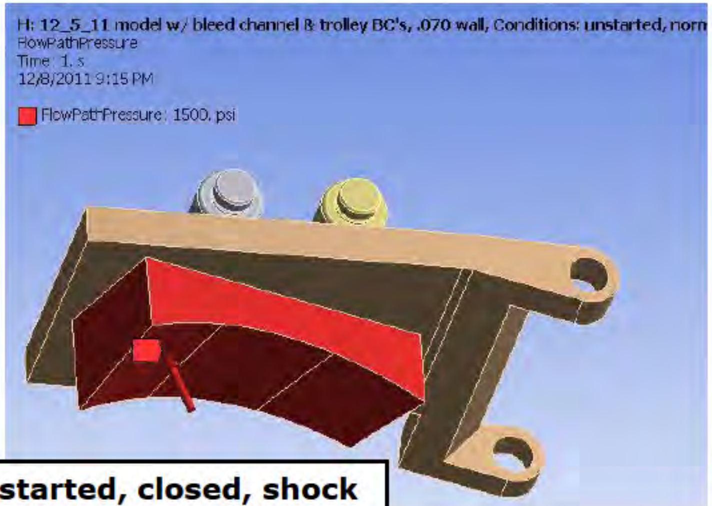

Door Open at 6.6 deg (max required opening)

- At max open condition clearance between door rollers and piston seal is minimal
- At door closed assembly is only 0.050"
- Reducing piston seal height is not currently recommended by John Crane
- Cannot reduce door thickness without violating shroud face seal region

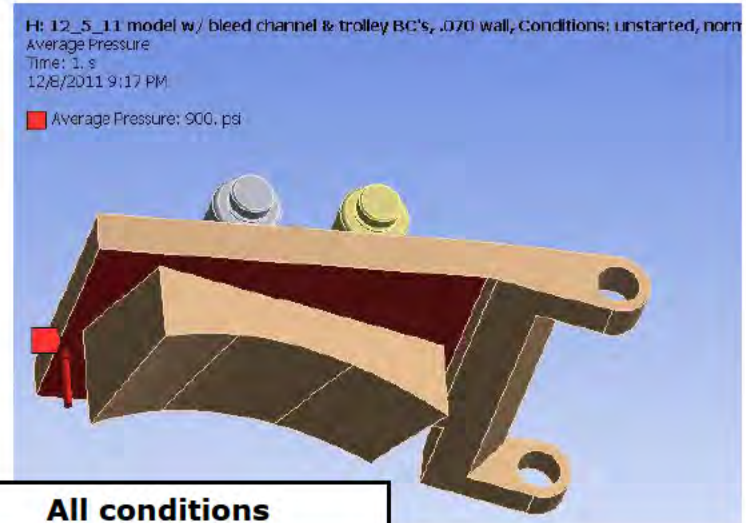
FEA Boundary Constraints and Contacts



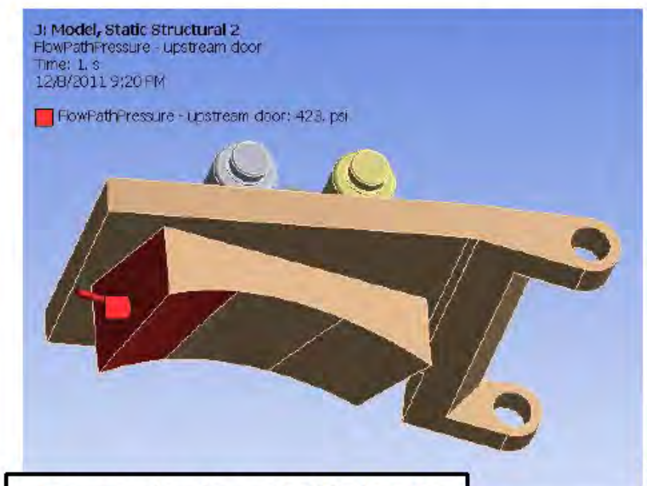
FEA Flow Path Pressure BC’s



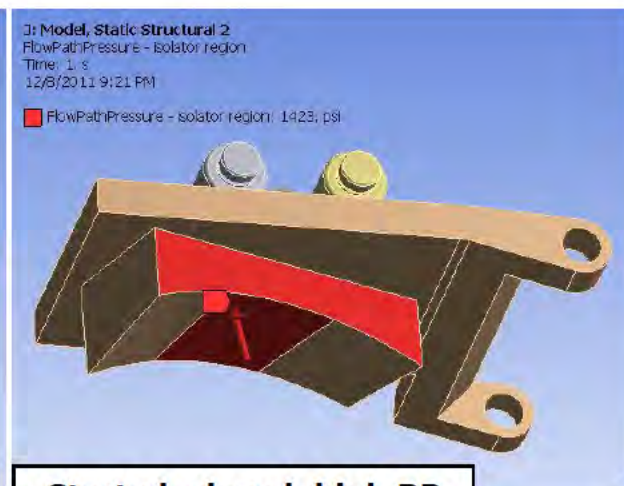
Unstarted, closed, shock upstream, high BP



All conditions



Started, closed, high BP (upstream – 423psia)



Started, closed, high BP (isolator – 1423 psia)

Started, closed, high BP (downstream – 1769 psia)

FEA Flow Path Pressure BC's

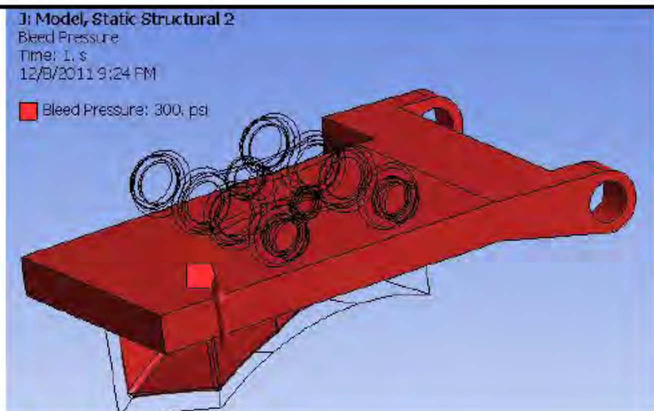
Door Bleed

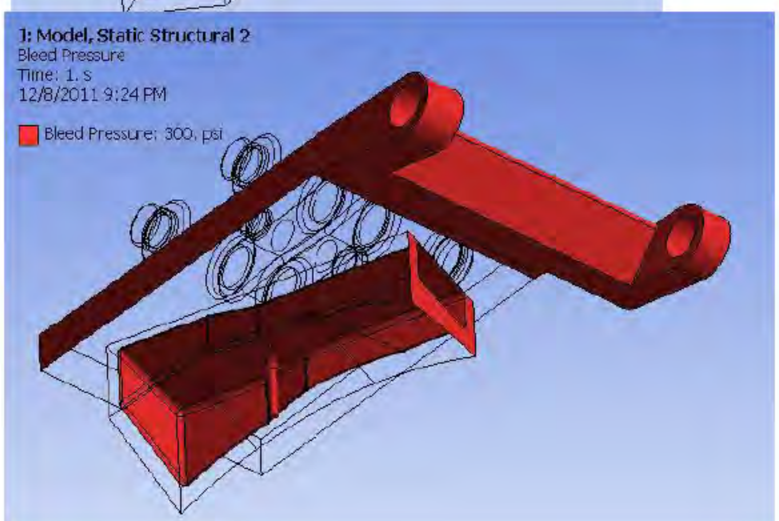
Started, closed, high BP:

300 psia

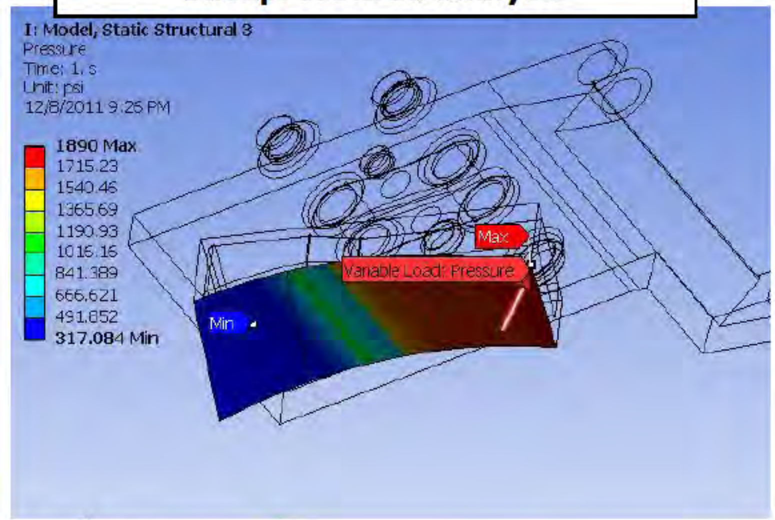
Unstarted, closed, shock upstream, high BP:

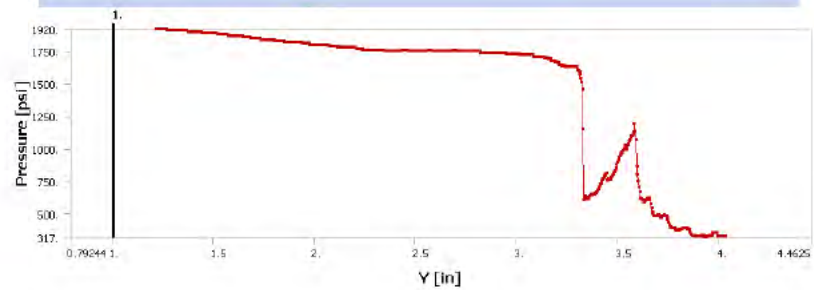
300 psia



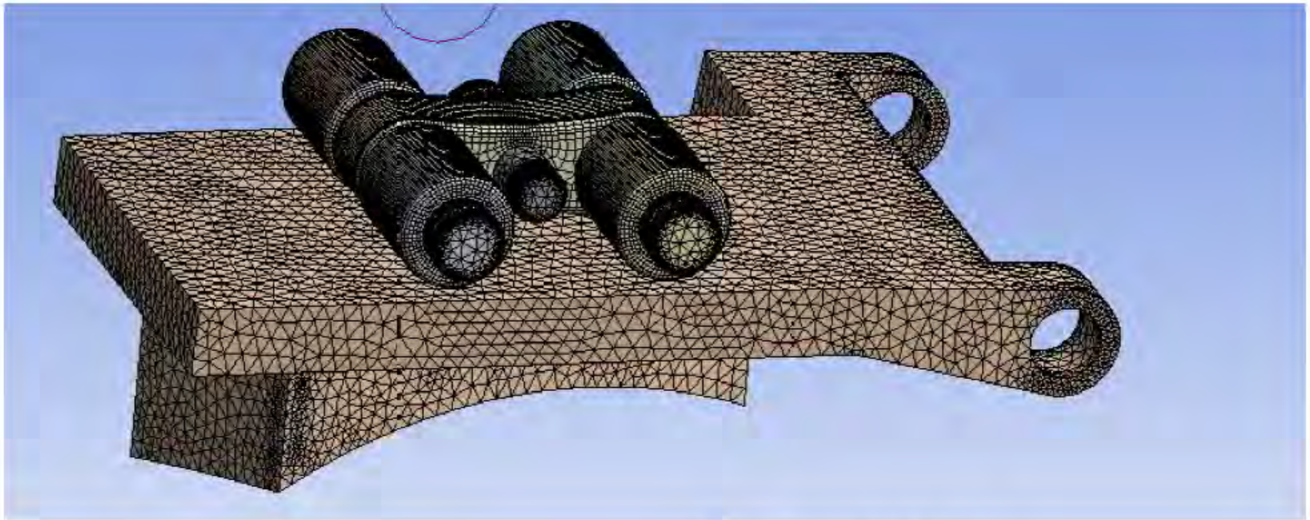


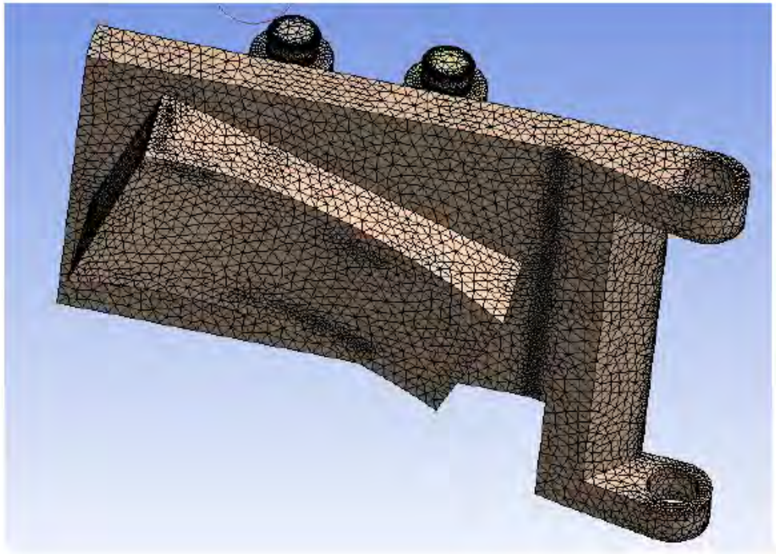
Pressure map from 2-D backpressured analysis



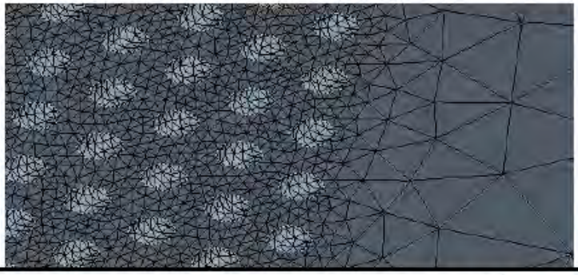


FEA Door Mesh



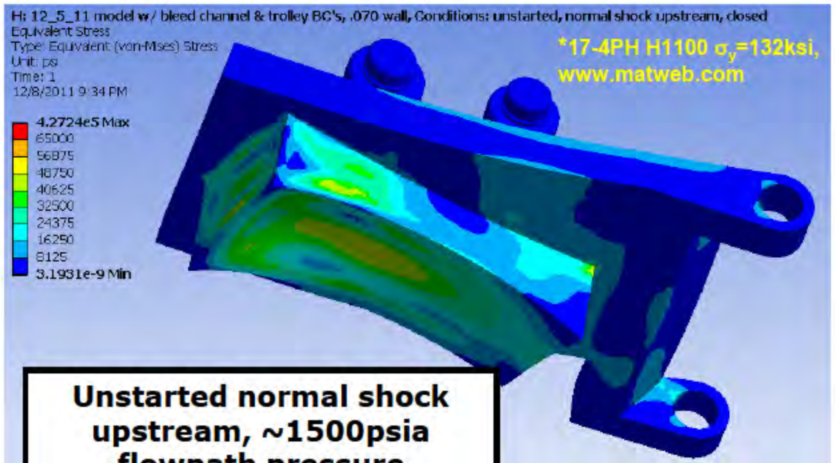


Nodes: 510,000
Elements: 220,000
-3 or more elements between all walls
-No bleed geometry



New Bleed Geometry Mesh (has not been analyzed yet)

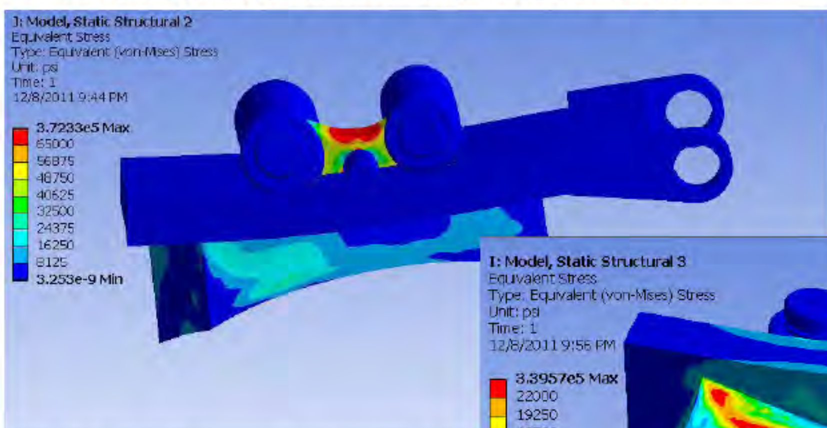
Door FEA Analysis

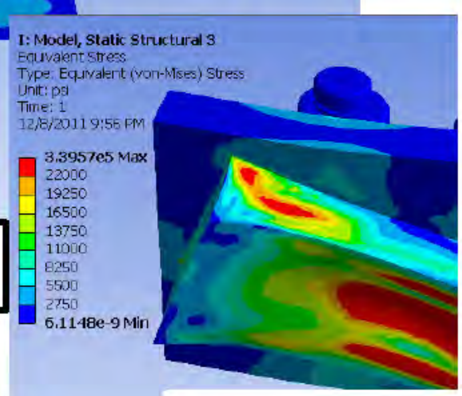


Unstarted normal shock upstream, ~1500psia flowpath pressure

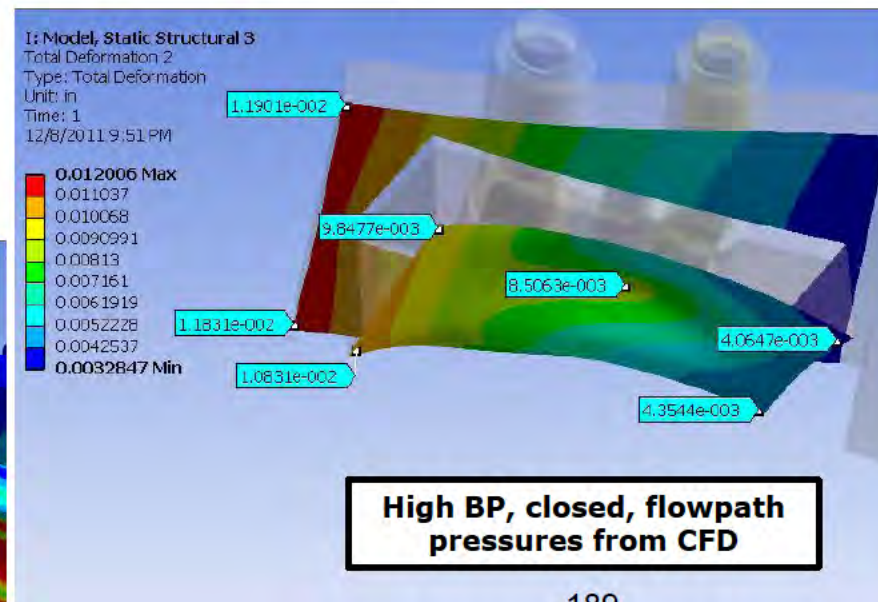
Unstarted normal shock upstream, ~1500psia flowpath pressure

Note: These results do not account for shroud deflections





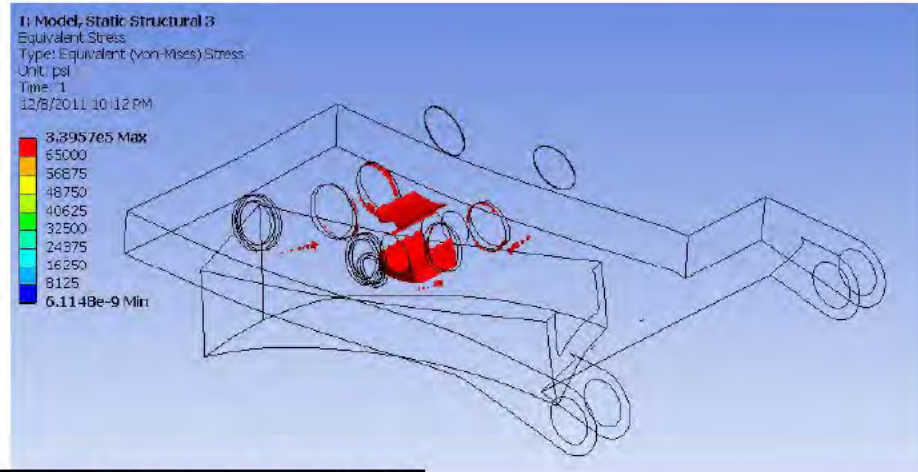
High BP, closed, flowpath pressures from CFD



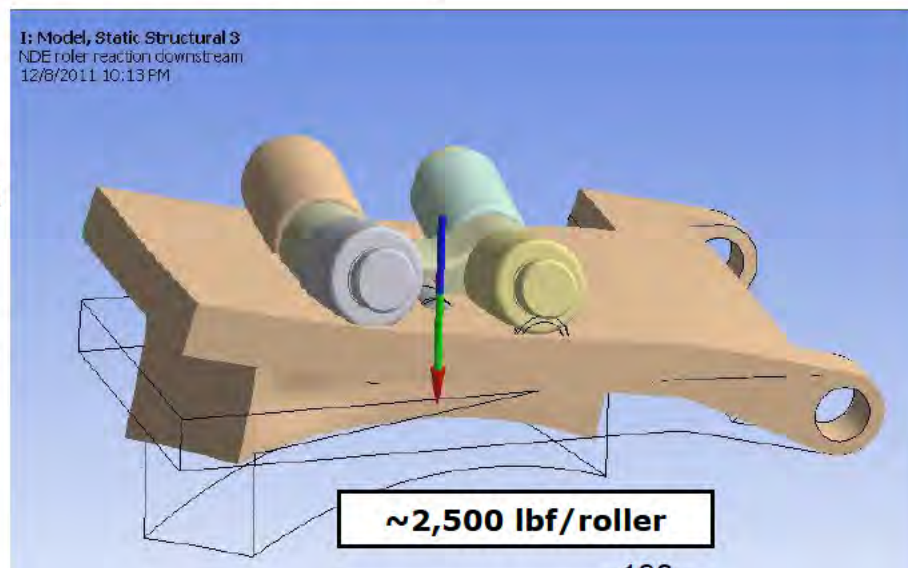
High BP, closed, flowpath pressures from CFD

Door FEA Analysis

- Door stresses appear to be acceptable
- Stresses appearred acceptable in initial Case 15 analysis of bleed holes for 0.75" wall thickness
- Deflections at flow path up to .011" at design point
- Roller reaction loads still a concern, deflections in actuator linkages a concern
- Hinge reactions not causing much stress in door
- No coupled shroud/door FEA yet



Red elements > 66ksi

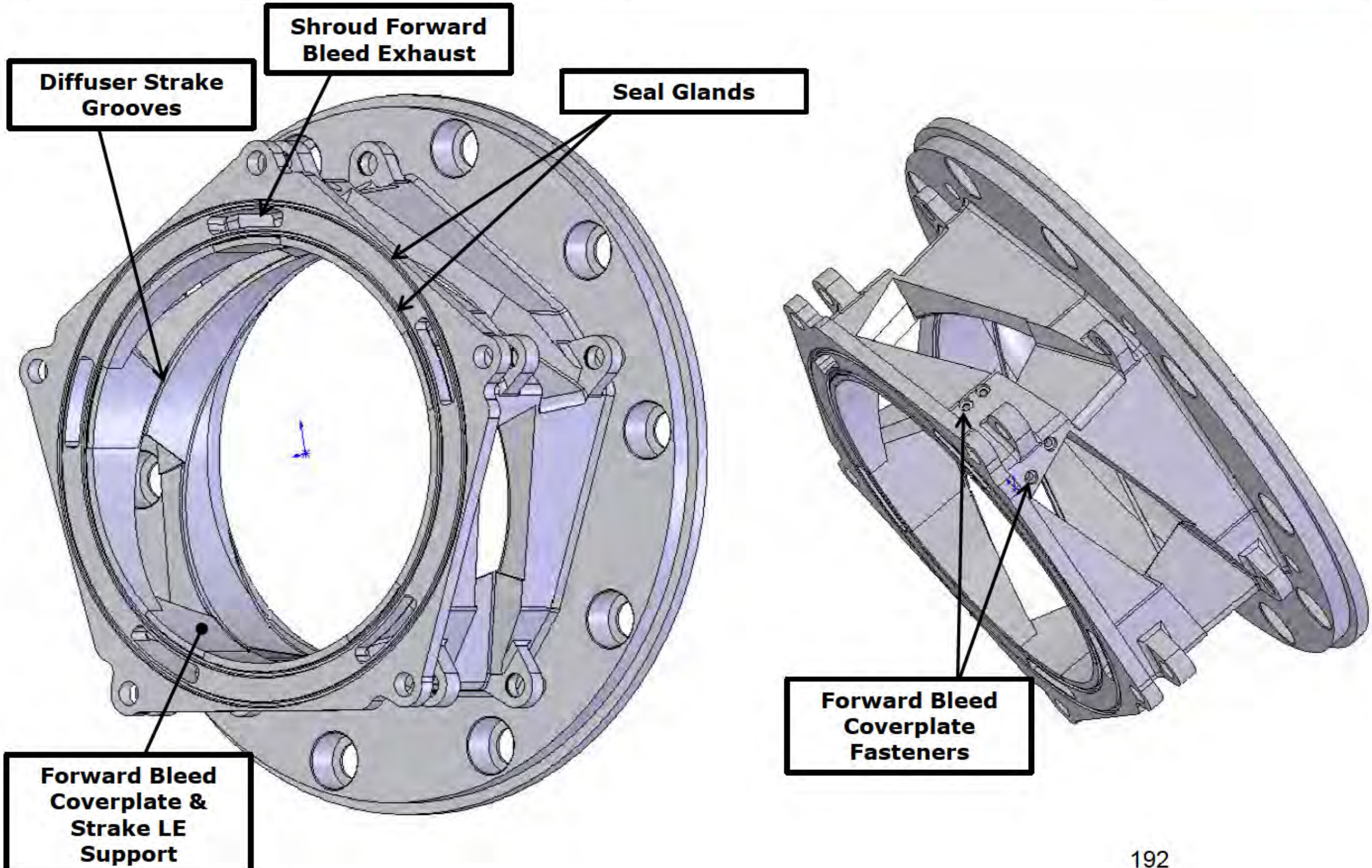


Static Diffuser Shroud

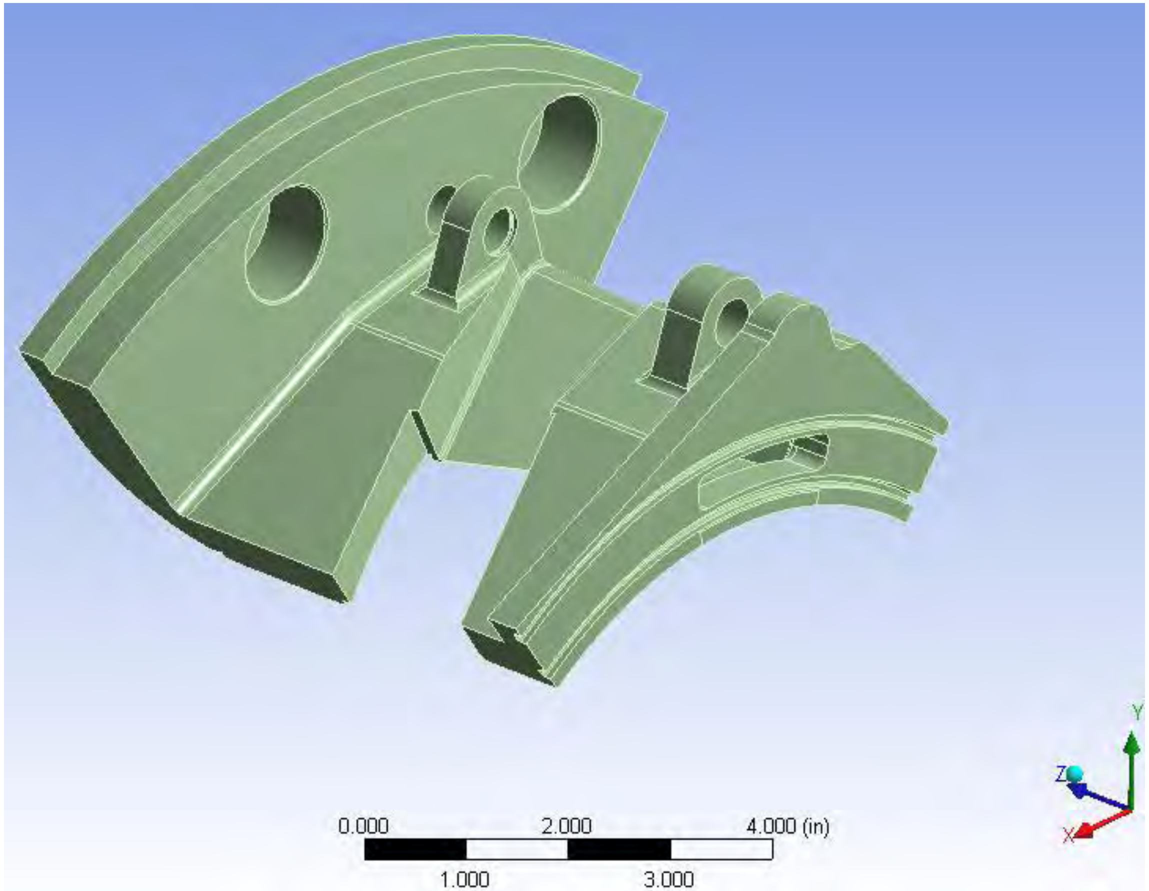
Ryan Edmonds

Original Presentation12/9/2011
Updated 12/12/2011

Current Shroud Geometry

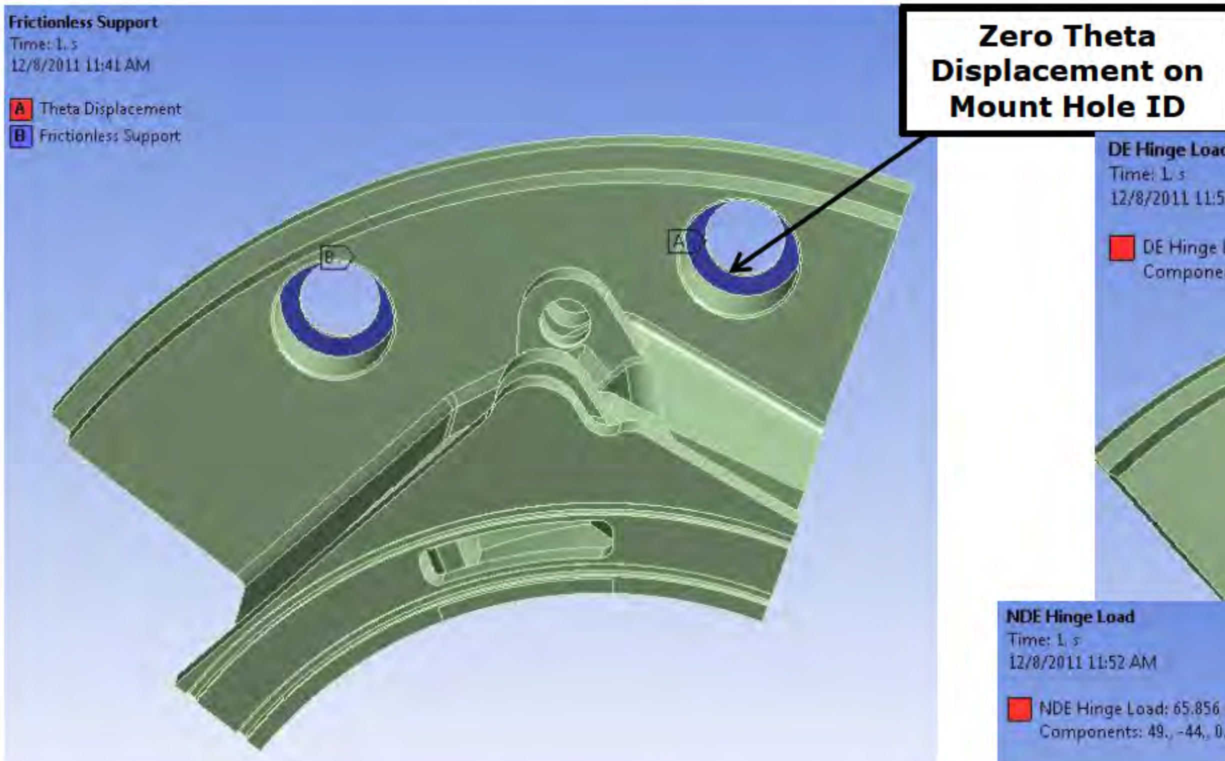


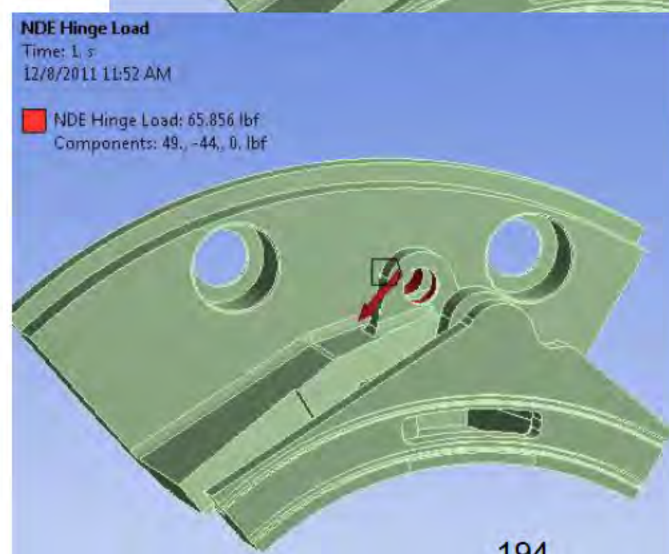
Updated Shroud Static Structural Model



- 1/5 sector model with shroud bleed coverplate
- Two conditions evaluated:
  - Design Point
  - Unstart

Design Point Boundary Conditions

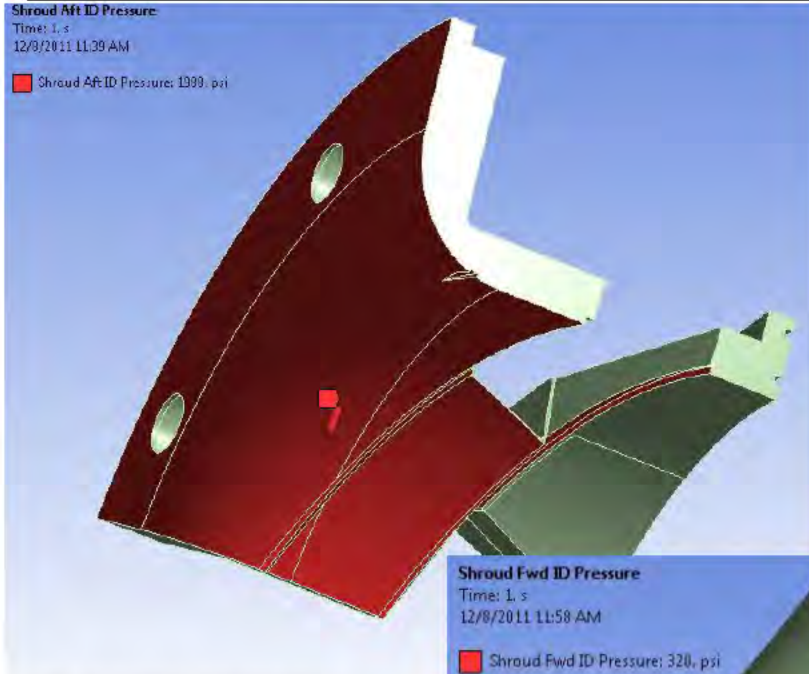


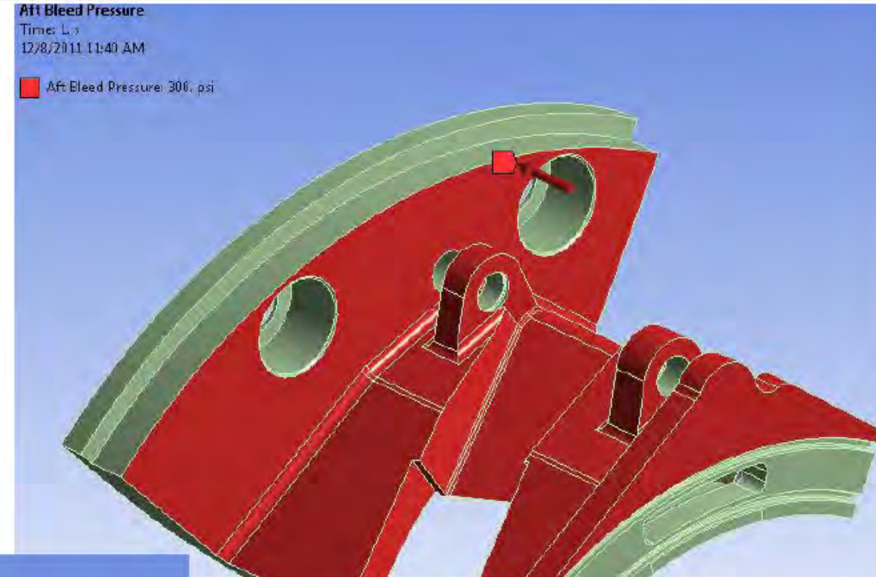


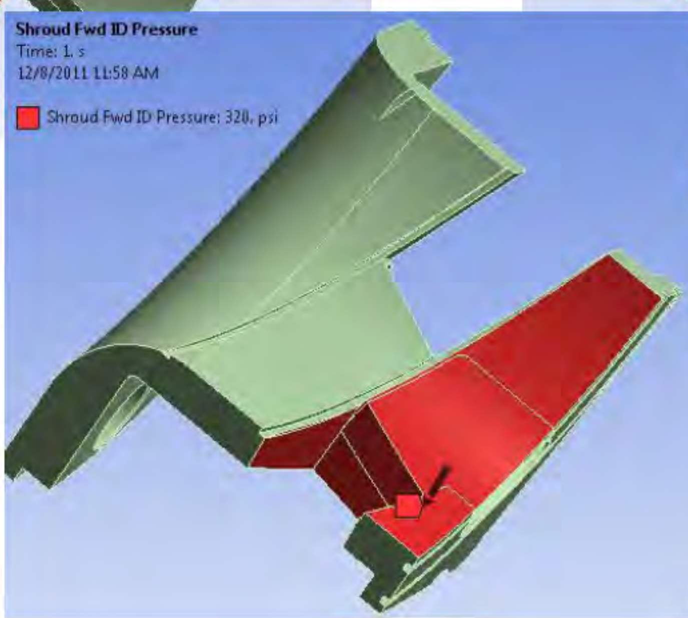
- **Door loads taken directly Case 15H Door.wbpj**

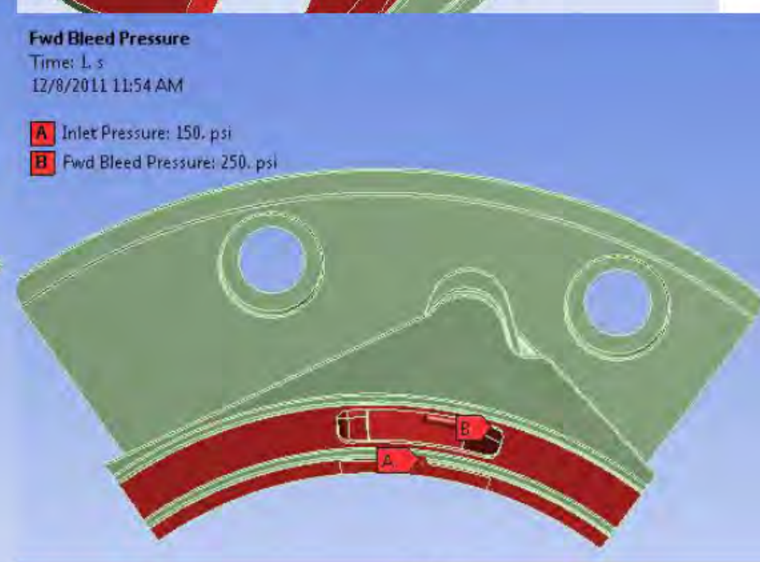
Shroud_Case 15H.SLDPRT
StaticDiffuser_15HBleedRamp.wbpj

Design Point Boundary Conditions



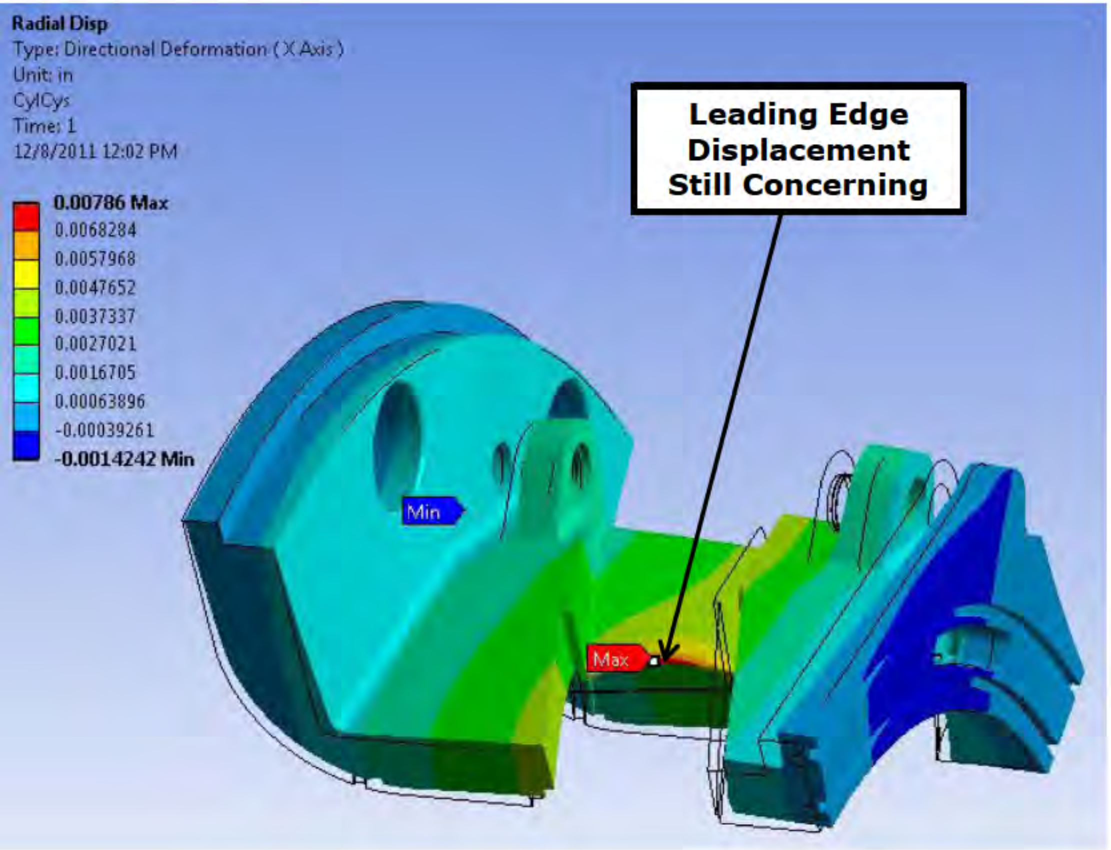






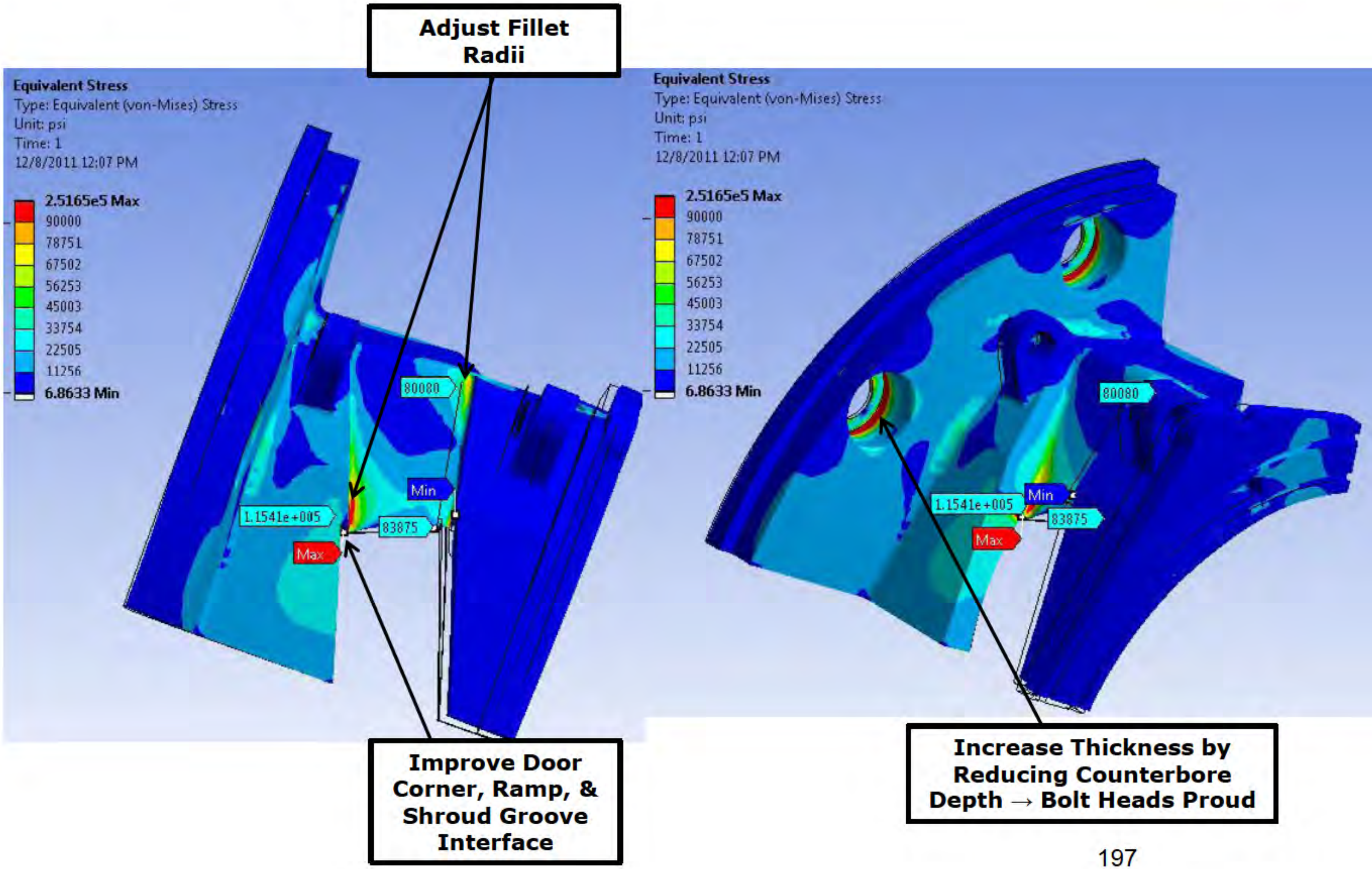
- **Exhaust pressure scaled from 2D CFD results**
- **Bleed pressures from Gen2 Aero Spec and Released Data.xlsx**

Design Point Radial Displacement Results

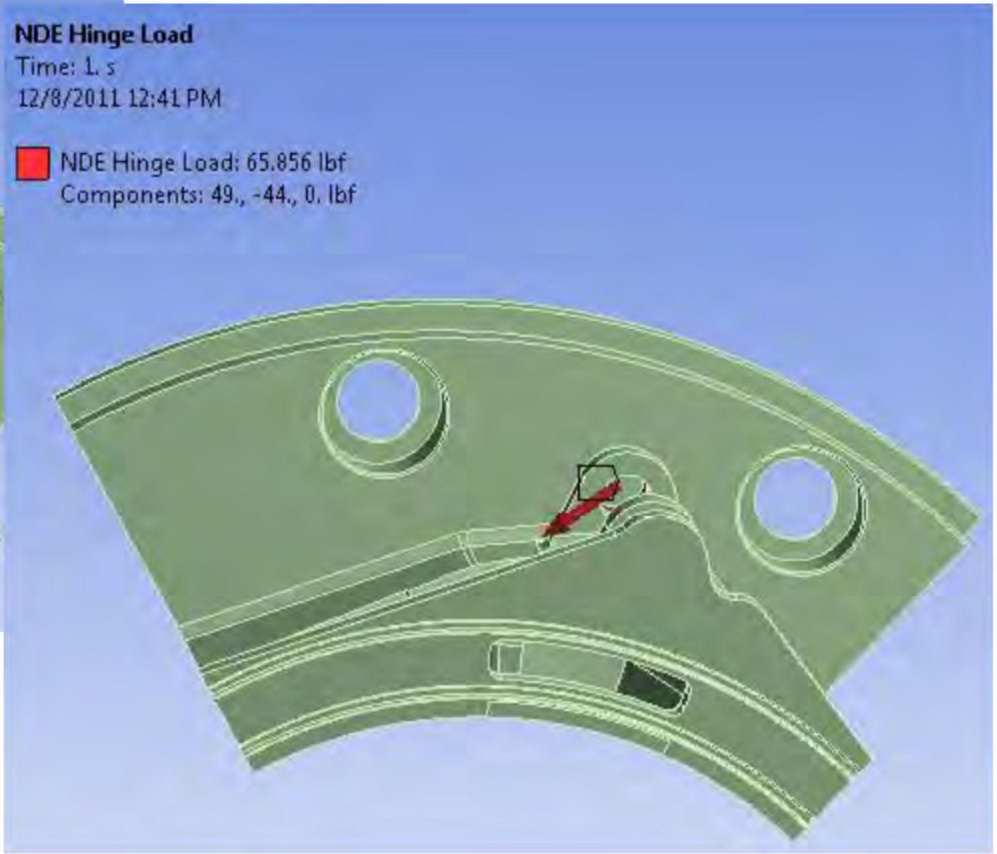


- LE of bleed ramp could be support by door as previously proposed
- Results may change substantially when door is included in a combined shroud & door structural model

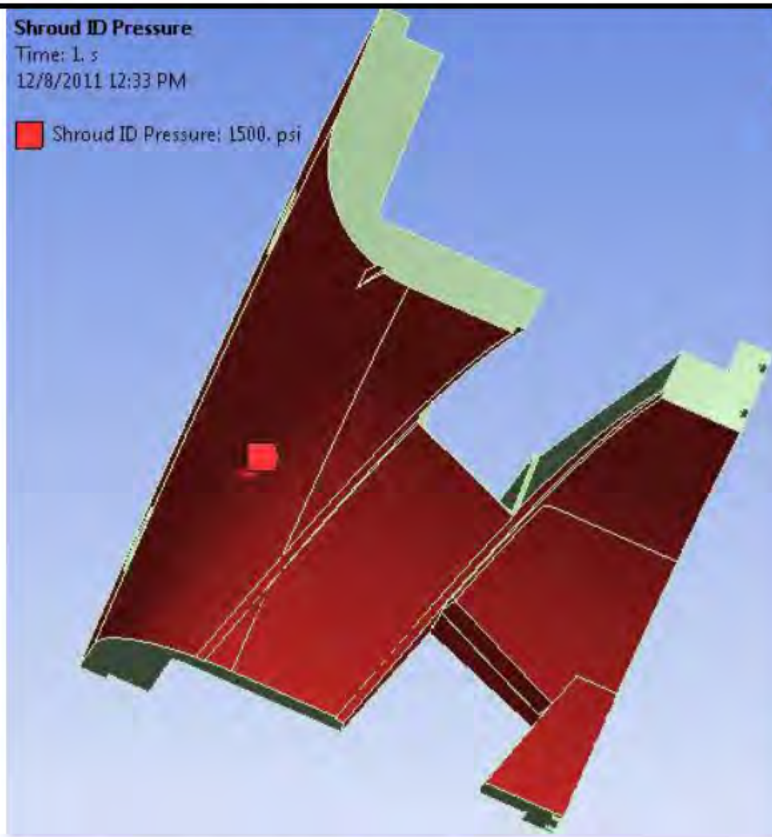
Design Point Equivalent Stress

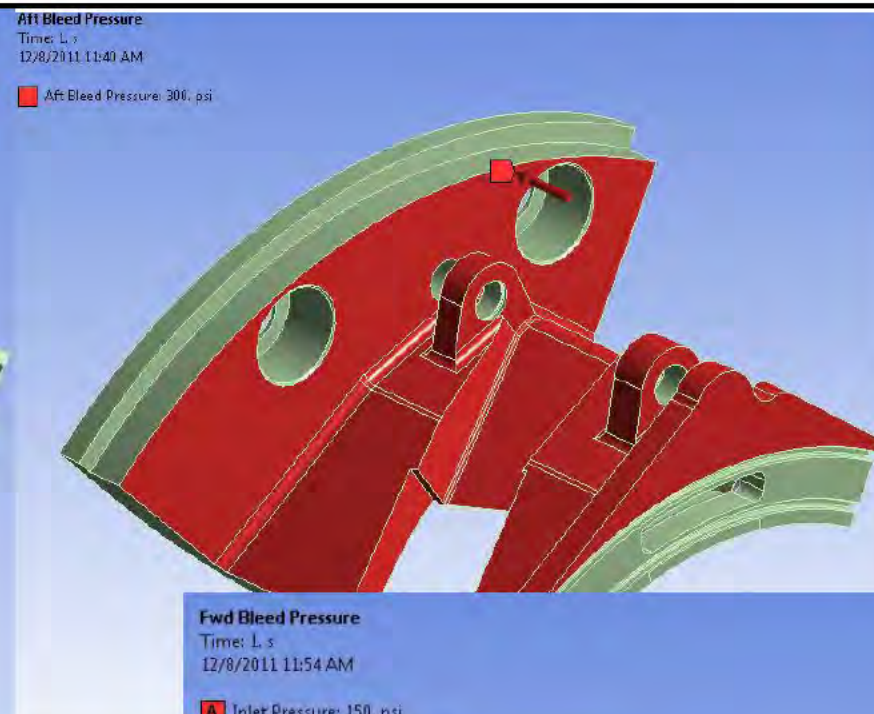


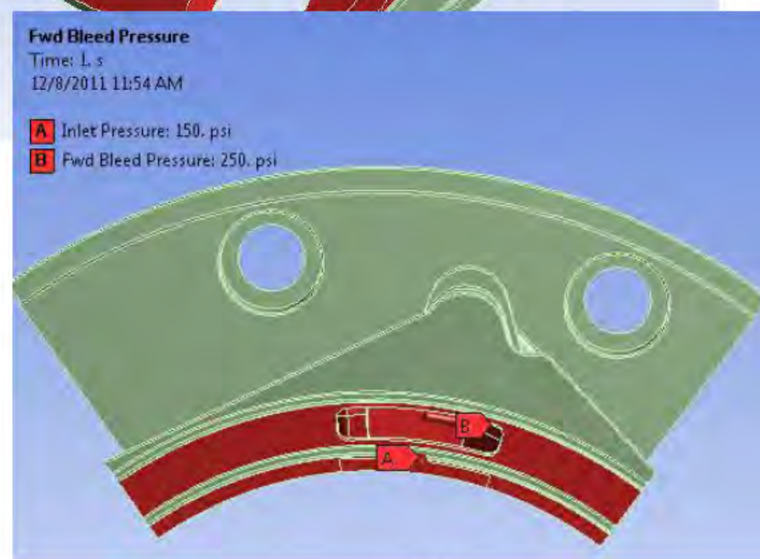
Unstarted Door Hinge Loads



Unstart Boundary Conditions

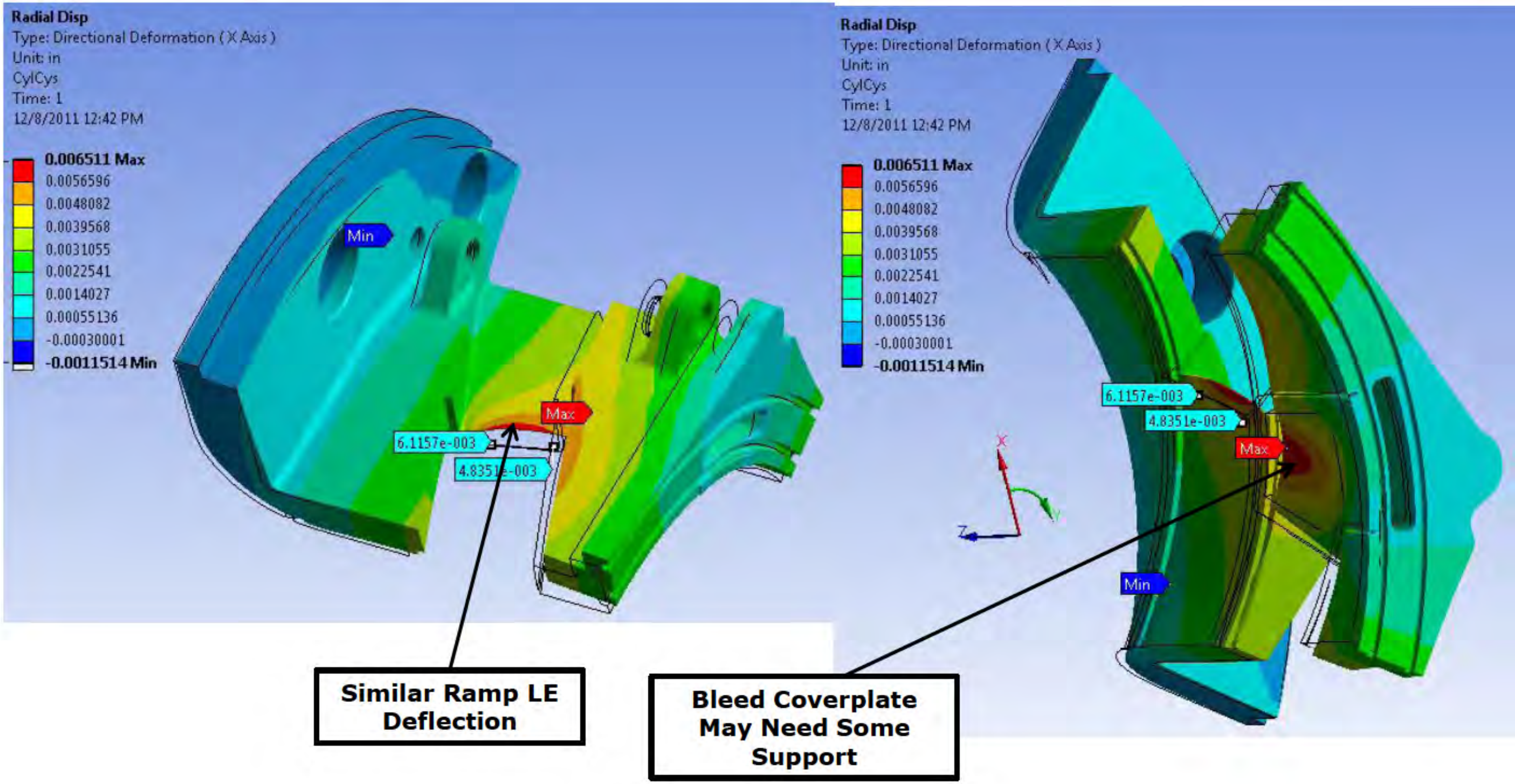




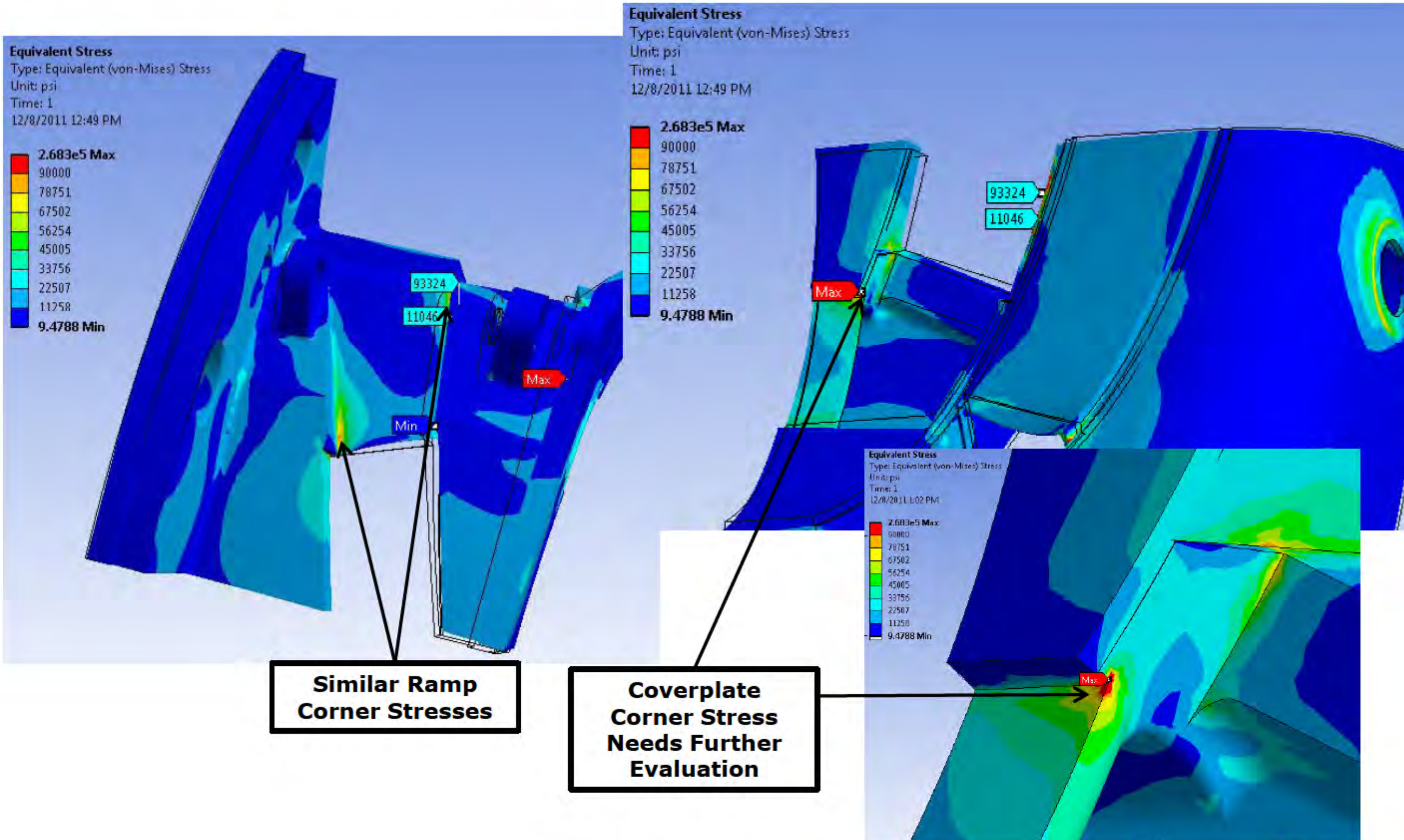


- **Uniform shroud unstart pressure**
- **Bleed pressures from Gen2 Aero Spec and Released Data.xlsx**

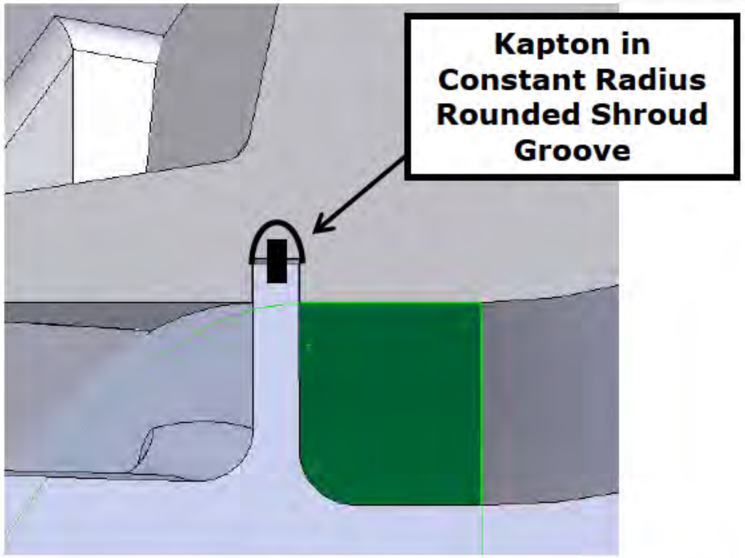
Unstart Radial Displacement Results



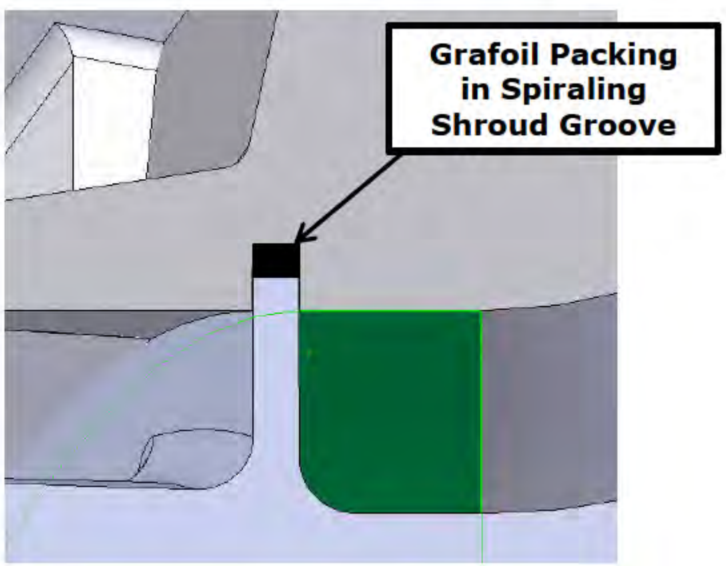
Unstart Equivalent Stress



Shroud Sealing

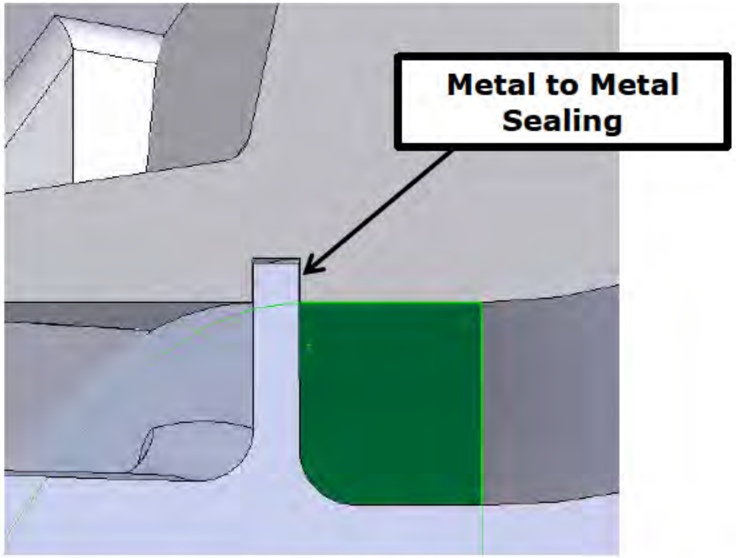


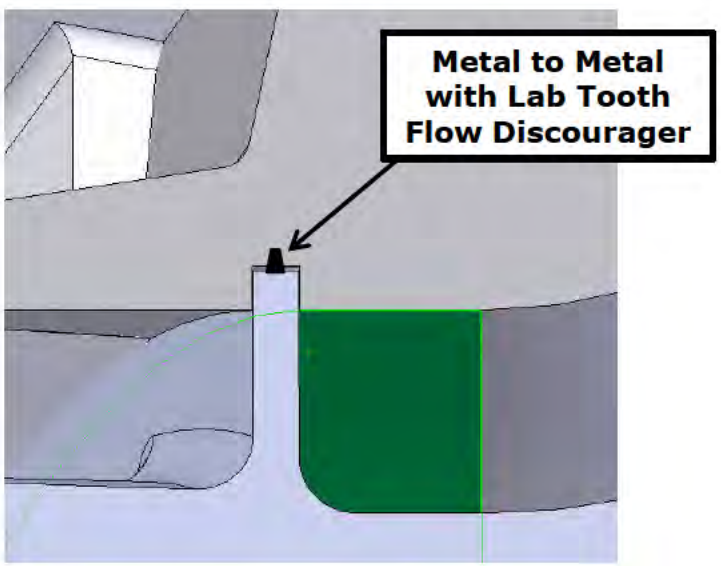
- **Kapton Issues**
  - Machining seal groove feature in strake tip
  - Kapton material would need to be long enough to flex and press against groove to seal, but not so long that it would be a challenge to install



- **Grafoil Issues**
  - Shroud groove would need to be cut as a spiral to give assembly clearance of sealing material
  - Strake would also be cut as a spiral/conical tip surface so that grafoil achieves full crush once the hub is fully "threaded" in.
  - Grafoil packing is fragile

Shroud Sealing





- **Issues**
  - **Consider modeling flow over strake tip assuming metal to metal sealing with very small gap?**
  - **Machining lab tooth like feature on strake tip challenging**

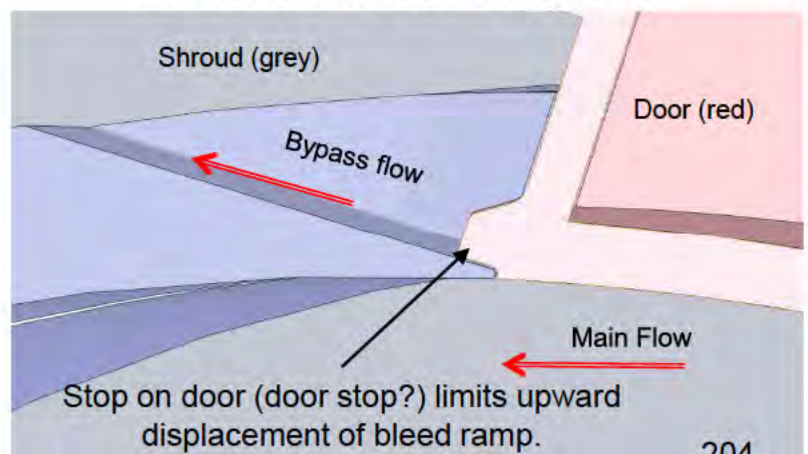
Note:

No sealing feature will be possible for first ~2" of strake tip due to 0.007" leading edge thickness and very shallow strake thickening angle.

Next Steps

- Cleanup interface issues between flat sided door and helical shroud grooves
- Include strake LE support cutout detail in shroud forward bleed coverplate
- Run shroud with door structural model
  - Also include thermal effects
- Work out shroud forward bleed seal interfaces
- Implement strake tip sealing scheme into shroud
- Do we add a door stop/deflection limiter for shroud bleed ramp?

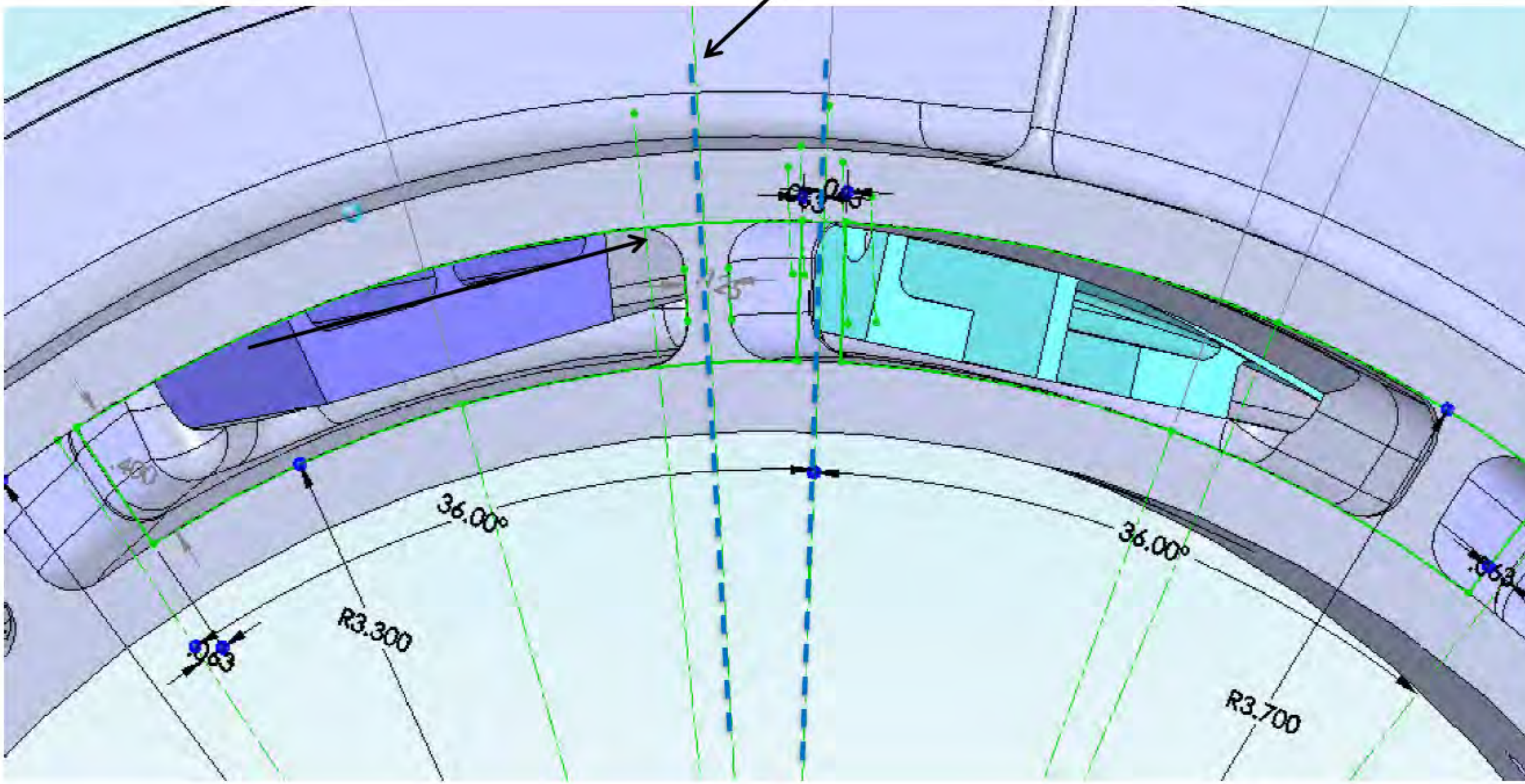
CAD Model from previous D. Taylor work



Case 15H Hub Geometry

Potential Bleed Area at Hub extension: .816 in ^2

Angular Clocking of Divider Wall to Match Hub Bleed Manifold



DE View
(from rear)

Case 15 "Dual Throat" Hub Geometry

Lofting Smoothness and Area distribution In Work

TOP View

Installing Inserts: (from case 15 PDR)

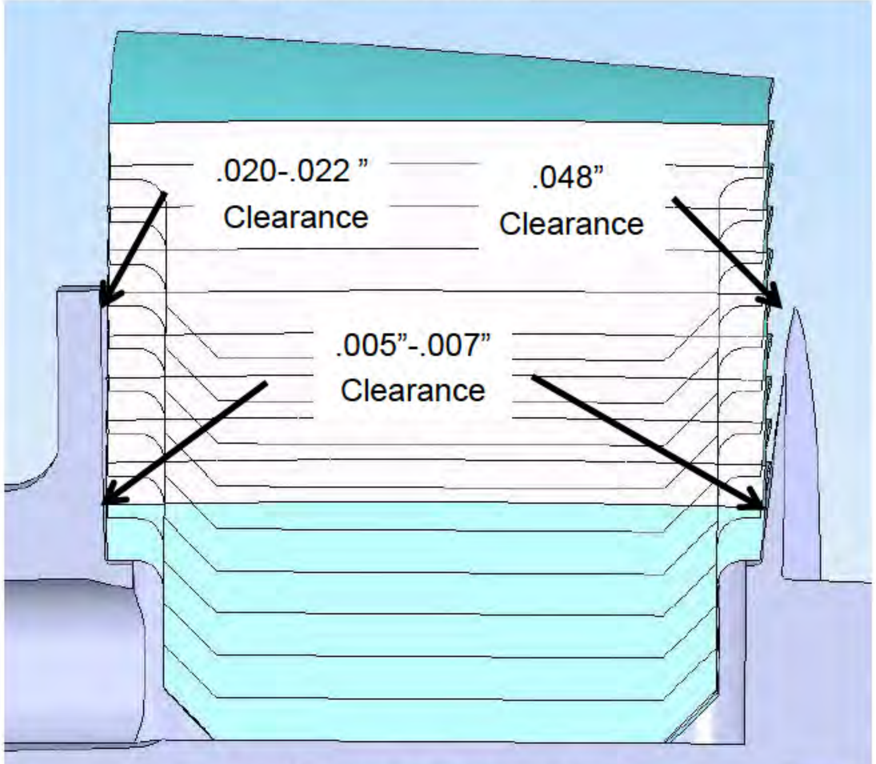
Step 1: Apply High temp sealant/gasket on bottom surface and Place Insert into position

Step 2: Shim around perimeter

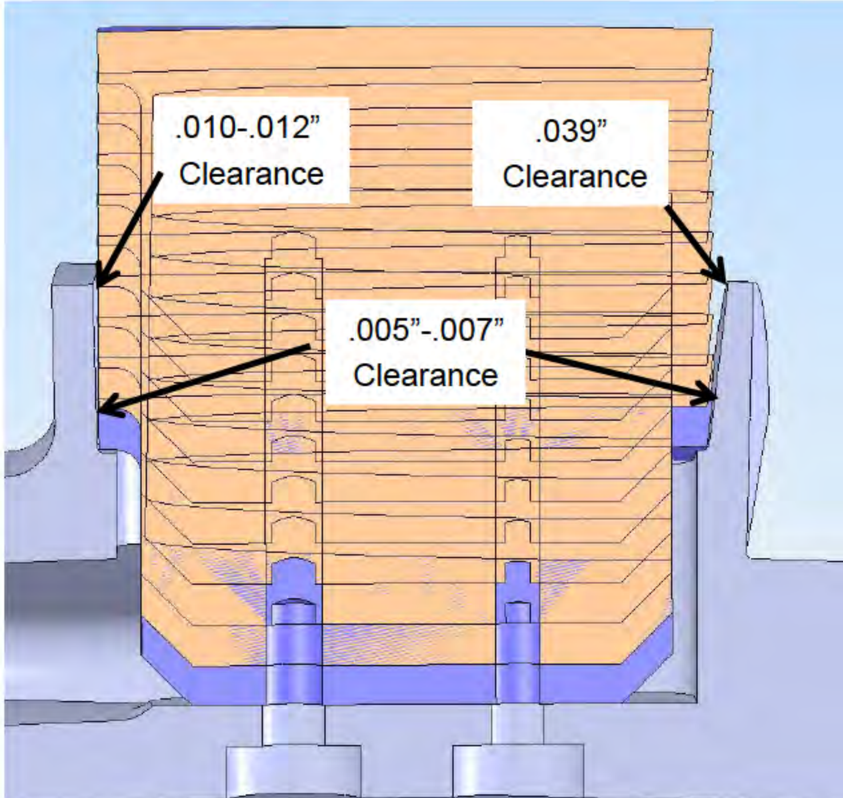
Step 3: Check FWD/AFT Steps (shim on bottom surface if necessary)

Step 4: Apply Lok-tight to Bolt and Torque

Step 5: Remove perimeter shims



Preliminary FWD Insert Install check via radial pattern



Preliminary Aft Insert Install check via radial pattern

Hub Manufacturing Discussion:

- **Static Diffuser Hub: 717025**
  - **Near Cast with hub bleed passages**
  - **Final Flowpath machining**
  - **Issues: list here**

- **Hub Inserts: 717017 & 717018**
  - **Combination of 5 axis milling and EDM wiring**
  - **Bleed Holes: Laser drilling or EDM**
  - **Do we need match machining with Hub?**
  - **Issues: list here**
- **Hub Bleed Manifold: 717534**
  - **5 axis milling**
  - **Vane Inserts, Seal Cavities secondary process(ie coatings)**
  - **Issues: list here**

Budget and Schedule

- FDR date
  - January 13, 2012
- Drawing Release date (no later than)
  - 717025 Static Diffuser Hub 2/3/2011
  - 717005 Static Diffuser Shroud 2/3/2011
  - ~~717009 Hub Bleed Extension 2/3/2011~~
  - 717026 Diffuser Hub, Casting 11/18/2011
  - 717011 Diffuser Shroud, rough 11/18/2011
  - 717012 Load Ring, Actuator Door 2/3/2011
  - 717027 Static Diffuser Fwd Hub Bleed Insert 2/3/2011
  - 717028 Static Diffuser Aft Hub Bleed Insert 2/3/2011
  - 717019 Load Block, Actuator Door 2/3/2011
  - 717022 Shroud Door, Forward Bleed Seal 2/3/2011
  - 717534 NDE Hub Bleed Manifold:2/3/2011

Current budgetary part cost estimates

717004	Static diffuser hub	$45,000.00
717005	Static diffuser shroud	$45,000.00
717010	717004 rough, diffuser hub	$0.00
717011	717005 rough, diffuser shroud	$0.00
717009	Hub bleed extension	$5,500.00
717012	Load ring, actuator door	$350.00
717017	Static diffuser fwd hub bleed insert	$1,250.00
717018	Static diffuser aft hub bleed insert	$1,250.00
717019	Load block, actuator door	$150.00
717022	Shroud door, forward bleed seal	$250.00
717014	Primary flowpath radial diffuser vane	$700.00

Budget and Schedule

- Estimated Manufacturing Time/Delivery date
  - Needs evaluation
- Is schedule achievable?
  - Design Schedule on track so far, any major changes to current config will result in schedule delays
  - Manufacturing needs evaluation
- Current manufacturing lead time and cost/budget adequate? ☺

Current budgetary part cost estimates

717004	Static diffuser hub	$45,000.00
717005	Static diffuser shroud	$45,000.00
717010	717004 rough, diffuser hub	$0.00
717011	717005 rough, diffuser shroud	$0.00
717009	Hub bleed extension	$5,500.00
717012	Load ring, actuator door	$350.00
717017	Static diffuser fwd hub bleed insert	$1,250.00
717018	Static diffuser aft hub bleed insert	$1,250.00
717019	Load block, actuator door	$150.00
717022	Shroud door, forward bleed seal	$250.00
717014	Primary flowpath radial diffuser vane	$700.00

Work Plan / Analysis Tasks : Mechanical

- **Mechanical Design & Additional Analysis**
  - **Hub**
    - **Continue FEA on Strake Fillet Stress reduction**
    - **More FEA on Insert as well as remodeling web stiffeners to accommodate bleed holes better**
    - **Modal Analysis on Hub and Inserts**
    - **More Conclusive Flange Analysis between Bleed manifold and Shroud**
    - **Evaluate Thermal effects and axial stack-up effects, and implement thermal into FEA**
    - **Implementing Manufacturing Plans into design**
  - **Shroud/Door**
    - **Finalize door/shroud helical vs straight sidewall interfaces**
    - **Include strake LE support cutout detail in shroud forward bleed coverplate**
    - **Determine if door starting ramp limiter is required/possible to implement**
    - **Coupled Shroud/Door FEA analysis**
    - **Bleed hole/pocket geometry FEA and optimization FEA study**
    - **Additional modeling to address door/shroud/actuator assembly issues**
    - **Address shroud/inducer abradable sealing issues**
    - **Define sealing method to mitigate strake tip leakage**
    - **Determine method for minimizing shroud groove downstream of strake TE**
    - **Determine acceptable aero-mechanical design for bypass ramp**
  - **Complete system**
    - **Hot to cold analysis, thermal loads incorporated into FEA**

Budget and Schedule

- FDR date
  - January 13, 2012
- Drawing Release date (no later than)
  - ~~717004~~ Static Diffuser Hub 2/3/2011
    - » New P/N 717025
  - ~~717005~~ Static Diffuser Shroud 2/3/2011
    - » New P/N 717029
  - ~~717009~~ Hub Bleed Extension 2/3/2011
    - » Part of Static Diffuser Hub now
  - ~~717010~~ Diffuser Hub, rough 11/18/2011
    - » New P/N 717026
  - 717011 Diffuser Shroud, rough 11/18/2011
  - ~~717012 Load Ring, Actuator Door 2/3/2011~~
    - » No longer necessary
  - 717017 Static Diffuser Fwd Hub Bleed Insert 2/3/2011
    - » New P/N 717017

Budget and Schedule

- **Drawing Release date (no later than)**
  - **717018 Static Diffuser Aft Hub Bleed Insert 2/3/2011**
  - ~~**717019 Load Block, Actuator Door 2/3/2011**~~
    - » **No longer necessary**
  - ~~**717022 Shroud Door, Forward Bleed Seal 2/3/2011**~~
    - » **No longer necessary**
  - **717031 Shroud Forward Bleed Insert 2/3/2011**

Budget and Schedule

- **Estimated Manufacturing Time/Delivery date**
  - **Needs evaluation**
- **Is schedule achievable?**
  - **Case 15H FDR is 3.5-4 working weeks from previous now (previously, there was 4.5 weeks between PDR and FDR).** ***This schedule may not be achievable***
  - **Manufacturing needs evaluation**
- **Current manufacturing lead time and cost/budget adequate? ☺**

Current budgetary part cost estimates

717004	Static diffuser hub	$45,000.00
717005	Static diffuser shroud	$45,000.00
717010	717004 rough, diffuser hub	$0.00
717011	717005 rough, diffuser shroud	$0.00
717009	Hub bleed extension	$5,500.00
717012	Load ring, actuator door	$350.00
717017	Static diffuser fwd hub bleed insert	$1,250.00
717018	Static diffuser aft hub bleed insert	$1,250.00
717019	Load block, actuator door	$150.00
717022	Shroud door, forward bleed seal	$250.00
717014	Primary flowpath radial diffuser vane	$700.00

Appendix 6.3

Rotor Assembly CDR

Conceptual Design Review

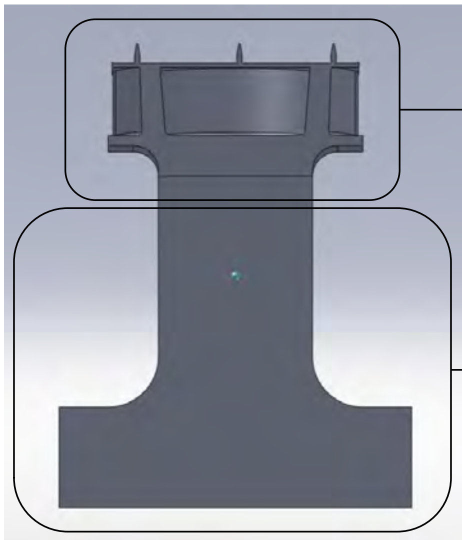
Gen-2 Build-2 Rotor Assembly

System owner(s):

Dave Taylor, Ravi Srinivasan

Nov 8th, 2012

Inducer Overview



Investigated various concepts for Flowpath region – Blisk with Integral shroud, Blisk with attached shroud, bladed-disc.

Investigated both solid and pierced disks

Inlet annulus lines as build-1, Shroud exit as build-1

Hub rad = 3.69” inlet, 4.12” exit
Tip rad = 4.374”
Inlet blade ht = 0.68”, Exit blade ht = 0.25”
Axial Chord approx 2.4”

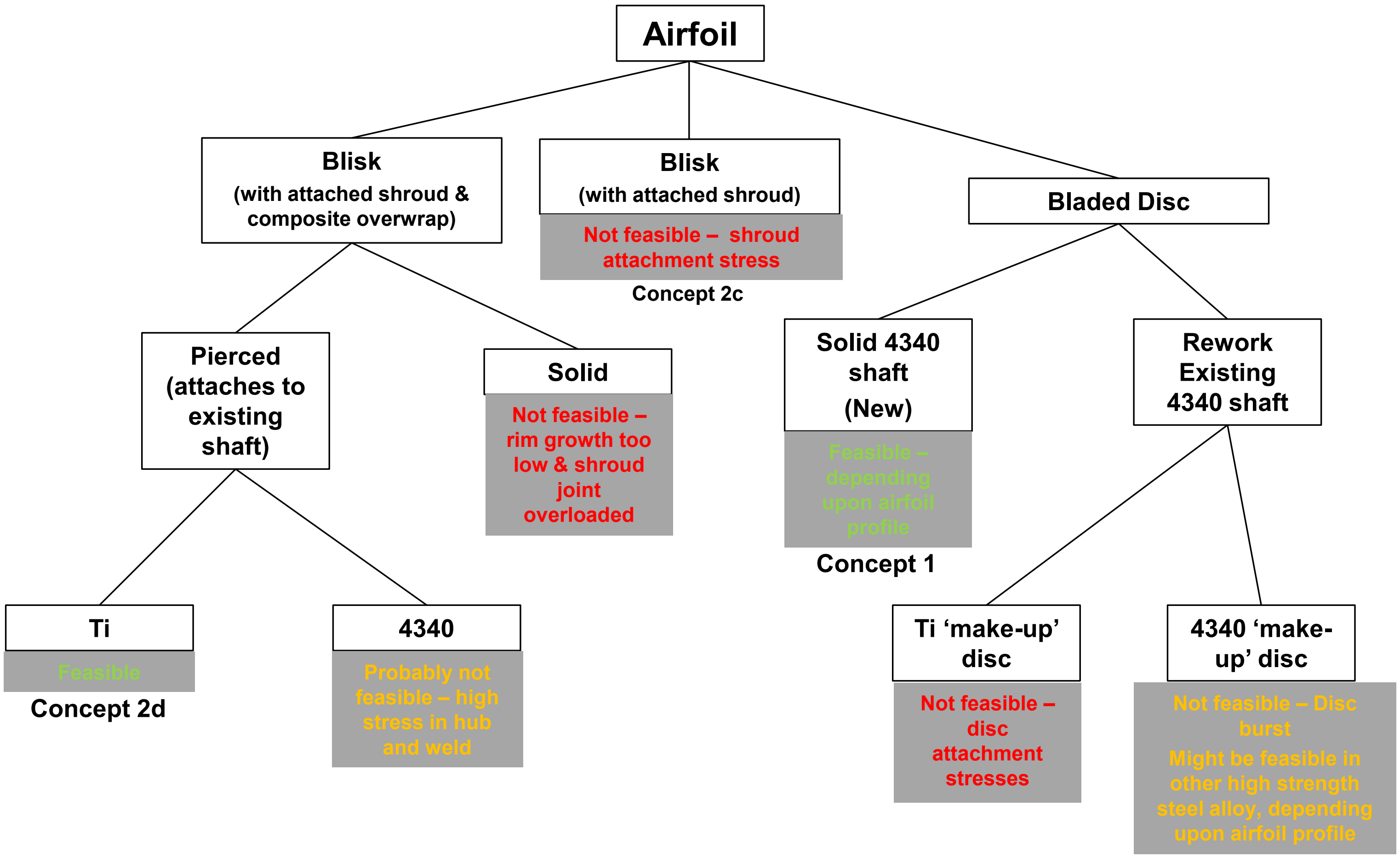
Blade count = 61
Design pt speed = 33,000 rpm, MCOS = 36,306rpm

Tip speed
At 30,000 rpm = 1145 ft/sec
33,000 rpm = 1259 ft/sec
36,306 rpm = 1386 ft/sec

HPC - Gen 2 Build 2 Inducer : Summary

- Looked at 4+ blade profiles
  - Reaction Blade 6, Reaction Blade 6 with TE cutback ~.150” to give .010” TE and 3 airfoils from the optimization database: D29, D507 and D783 (all with .010” LE and TE)
- Conventional bladed disc approach appears feasible.
- Blisk Configuration (Solid & pierced blisk variants)
  - Structurally feasible. Requires composite overwrap if separate shroud used
  - Manufacturing
    - With Integral shroud:
      - Milling - not possible (feedback from C & A and Turbocam)
      - EDM – not possible
      - ECM – not possible
      - DMLS – Not possible (feedback from C & A and Morris)
      - P/M HIP – may be possible. Iterative, Long lead-time, high cost. Not pursuing.
    - With separately fabricated blisk and shroud:
      - Can be conventionally machined but will require welding & composite overwrap to retain shroud.
      - Composite appears feasible – Feedback from composites vendor (Mentis) positive

Mechanical Feasibility of Manufacturable Inducer Options



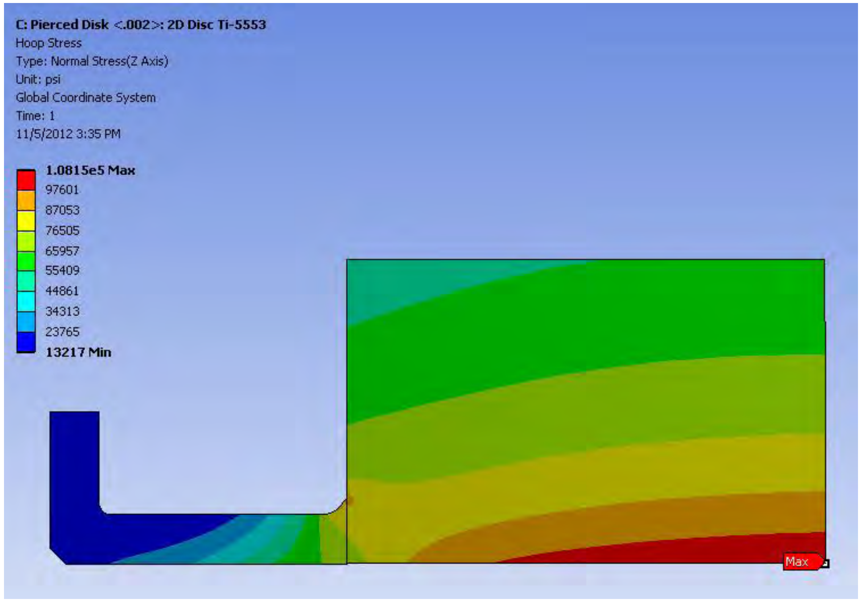
2D Analysis for Burst Margin (Ti)

- **Geometry:**
  - **Bladed Disc, 2.95" ID, Ti blade, Ti-5553 disc**
  - **Analysis at 36,306 rpm**
  - **Live Rim load of 965,978 lbf (20,936 psi)**
- **Results:**

```
*GET  VOLUME    FROM  SSUM  ITEM=ITEM VOLUME    VALUE=  53.0977412
*GET  PRDUCTI   FROM  SSUM  ITEM=ITEM PRDUCTI   VALUE=  3713461.05
PARAMETER STRSAVI =     69936.32805
PARAMETER UTS70 =     164000.0000
PARAMETER MAT_UTIL_FACTOR =    0.9500000000
PARAMETER OMEGA =     3802.000000
PARAMETER BURSTMARGIN =     1.492561770
PARAMETER BURSTSPEED =     5674.719849
```

- **Burst Speed = 5674 rad/s = 54,100 rpm**
- **Burst Margin (rel 36,306 rpm) = 1.49**
  - **Criteria is >= 1.25 x redline**
- **Peak Hoop stress (Bore) = 108 ksi**
  - **Yield stress = 150 ksi**
  - **UTS = 164 ksi**

Hoop Stress



Disc burst margin acceptable for Ti-5553 disc
Bore Hoop stress < yield

Bulk stresses acceptable (but attachment stresses likely not feasible as shown previously)

Ref:Gen2Build2-InducerDisc_v002_2D.SLDPRT
Gen2B2-HPC_BladedDisc_RxnBlade6_2DBurst_Run01.wbpj

Gen2 Build 2 Inducer : Bladed Disc Summary

Findings So far

- **Blade + disk configuration feasible with conventional attachment design**
  - **Airfoil 'wrap' (camber & stagger) makes locating airfoil on attachment a challenge.**
    - **Requires complex root profile, weight reduction features and careful balancing.**
    - **Some airfoil designs from optimization database may not be mechanically feasible**
  - **Utilized blade with .010" LE and TE**
  - **Requires new, solid 4340 disc/shaft same as Build-1. Cannot use spare rotor**
  - **Modal characteristics – don't know yet.**
  - **Can be conventionally machined**

Some brief comments on ECM

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Pros/Cons from Build-1 applicable here……

- **Pros**
  - **Mechanical Design of Blisk with Integral shroud somewhat simpler in the flowpath**
  - **No blade-to-blade gaps**
  - **2nd part quick and low cost (3 weeks, $10K ?)**

- **Cons**
  - **Est $250,000-$300,000 for first part**
  - **Schedule approx (insert duration based on GKM Gen2 Build-1 feedback)**
  - **Highly cambered blade not ideal for ECM**
    - **Process development required**
    - **Post ECM machining of all Leading and Trailing Edges (86 operations)**
    - **ECM from both sides – potential for bump or step in flowpath**
    - **Request for significant bow in blade to facilitate stiffer tool.**
    - **Tolerances outside requirements**

ECM not being actively pursued

Gen-2 Build-2: Rotor Build-up

- **If Bladed Disc- Similar procedure to Build-1**
    - **Ensure all tooling designed & ordered in time**

- **If Blisk – No blade/coverplate/snapring install required**
    - **Shaft Hardware identical to build-1**

Schedule: Bladed Disc (Concept 1)

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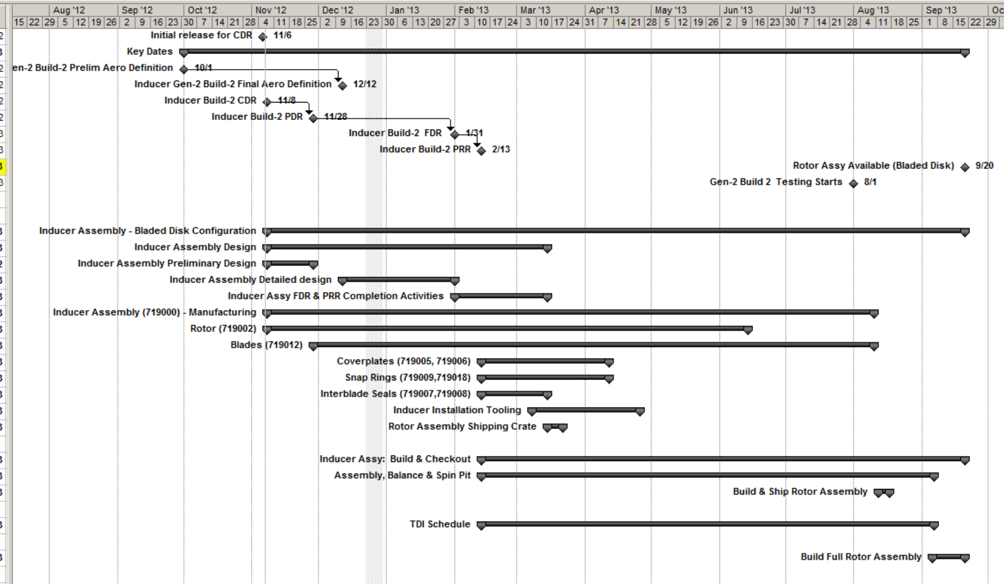
	Task Name	Duration	Start	Finish
1	Initial release for CDR	0 days	Tue 11/6/12	Tue 11/6/12
2	Key Dates	43.5 days	Mon 10/1/12	Fri 9/20/13
3	Inducer Gen-2 Build-2 Prelim Aero Definition	0 days	Mon 10/1/12	Mon 10/1/12
4	Inducer Gen-2 Build-2 Final Aero Definition	0 days	Wed 12/12/12	Wed 12/12/12
5	Inducer Build-2 CDR	0 days	Thu 11/8/12	Thu 11/8/12
6	Inducer Build-2 PDR	0 days	Wed 11/28/12	Wed 11/28/12
7	Inducer Build-2 FDR	0 days	Thu 1/31/13	Thu 1/31/13
8	Inducer Build-2 PRR	0 days	Wed 2/13/13	Wed 2/13/13
9	**Rotor Assy Available (Bladed Disk)**	**0 days**	**Fri 9/20/13**	**Fri 9/20/13**
10	Gen-2 Build 2 Testing Starts	0 days	Thu 8/1/13	Thu 8/1/13
11				
12				
13	Inducer Assembly - Bladed Disk Configuration	15.5 days	Thu 11/8/12	Fri 9/20/13
14	Inducer Assembly Design	84 days	Thu 11/8/12	Thu 3/14/13
15	Inducer Assembly Preliminary Design	15 days	Thu 11/8/12	Wed 11/28/12
18	Inducer Assembly Detailed design	30 days	Wed 12/12/12	Thu 1/31/13
28	Inducer Assy FDR & PRR Completion Activities	30 days	Thu 1/31/13	Thu 3/14/13
45	Inducer Assembly (719000) - Manufacturing	187 days	Thu 11/8/12	Fri 8/9/13
46	Rotor (719002)	148 days	Thu 11/8/12	Thu 6/13/13
59	Blades (719012)	172 days	Thu 11/29/12	Fri 8/9/13
73	Coverplates (719005, 719006)	42 days	Wed 2/13/13	Thu 4/11/13
76	Snap Rings (719009,719018)	42 days	Wed 2/13/13	Thu 4/11/13
79	Interblade Seals (719007,719008)	22 days	Wed 2/13/13	Thu 3/14/13
82	Inducer Installation Tooling	35 days	Fri 3/8/13	Thu 4/25/13
85	Rotor Assembly Shipping Crate	5 days	Fri 3/15/13	Thu 3/21/13
87				
88	Inducer Assy: Build & Checkout	53.5 days	Wed 2/13/13	Fri 9/20/13
89	Assembly, Balance & Spin Pit	43.5 days	Wed 2/13/13	Fri 9/6/13
90	Build & Ship Rotor Assembly	5 days	Mon 8/12/13	Fri 8/16/13
93				
94	TDI Schedule	43.5 days	Wed 2/13/13	Fri 9/6/13
131				
132	Build Full Rotor Assembly	11 days	Thu 9/5/13	Fri 9/20/13



- **Balanced Rotor Assembly (bladed disc) available late Sept 2013 – 3 months later than required**
  - **Driven by:**
  - **Aero design, Mechanical design w/ complex airfoil & blade manufacturing**
    - **Lead times based on achieved lead times on Build-1**

Ref: Gen2Build2_Inducer-BladedDisk_2012-11-06.mpp

Schedule: Blisk (Concept 2d)

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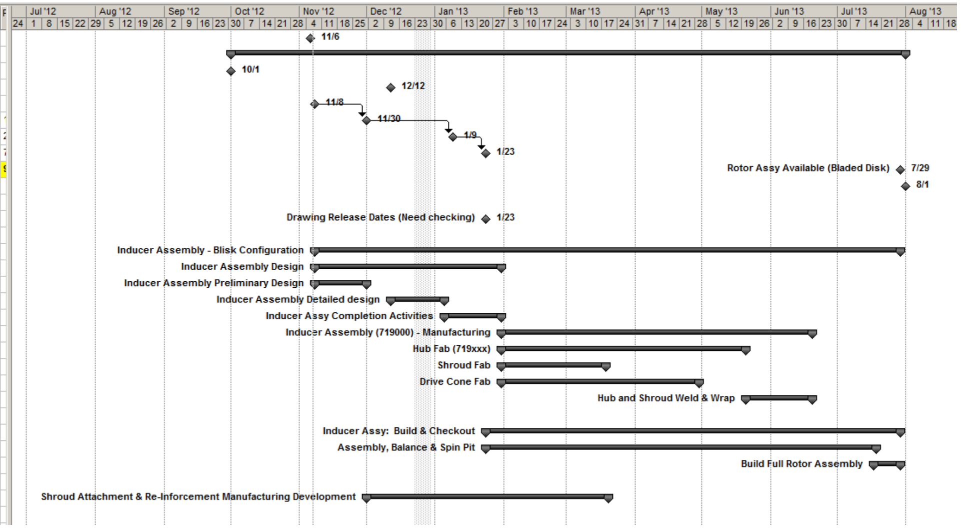
		Task Name	Duration	Start	Finish
1		Initial release for CDR	0 days	Tue 11/6/12	Tue 11/6/12
2		⊟ Key Dates	206 days	Mon 10/1/12	Thu 8/1/13
3		Inducer Gen-2 Build-2 Prelim Aero Definition	0 days	Mon 10/1/12	Mon 10/1/12
4		Inducer Gen-2 Build-2 Final Aero Definition	0 days	Wed 12/12/12	Wed 12/12/12
5		Inducer Build-2 CDR	0 days	Thu 11/8/12	Thu 11/8/12
6		Inducer Build-2 PDR	0 days	Fri 11/30/12	Fri 11/30/12
7		Inducer Build-2 FDR	0 days	Wed 1/9/13	Wed 1/9/13
8		Inducer Build-2 PRR	0 days	Wed 1/23/13	Wed 1/23/13
9		Rotor Assy Available (Bladed Disk)	0 days	Mon 7/29/13	Mon 7/29/13
10		Gen-2 Build 2 Testing Starts	0 days	Thu 8/1/13	Thu 8/1/13
11					
12		⊞ Drawing Release Dates (Need checking)	0 days	Wed 1/23/13	Wed 1/23/13
14					
15		⊟ Inducer Assembly - Blisk Configuration	175.5 days	Thu 11/8/12	Mon 7/29/13
16		⊟ Inducer Assembly Design	51 days	Thu 11/8/12	Wed 1/30/13
17		⊞ Inducer Assembly Preliminary Design	15 days	Thu 11/8/12	Fri 11/30/12
20		⊞ Inducer Assembly Detailed design	11 days	Wed 12/12/12	Fri 1/4/13
26		⊞ Inducer Assy Completion Activities	18 days	Fri 1/4/13	Wed 1/30/13
36		⊟ Inducer Assembly (719000) - Manufacturing	99 days	Thu 1/31/13	Wed 6/19/13
37		⊞ Hub Fab (719xxx)	78 days	Thu 1/31/13	Mon 5/20/13
41		⊞ Shroud Fab	33 days	Thu 1/31/13	Mon 3/18/13
44		⊞ Drive Cone Fab	63 days	Thu 1/31/13	Mon 4/29/13
47		⊞ Hub and Shroud Weld & Wrap	21 days	Tue 5/21/13	Wed 6/19/13
52					
53		⊟ Inducer Assy: Build & Checkout	129.5 days	Wed 1/23/13	Mon 7/29/13
54		⊞ Assembly, Balance & Spin Pit	122.5 days	Wed 1/23/13	Thu 7/18/13
91		⊞ Build Full Rotor Assembly	8 days	Wed 7/17/13	Mon 7/29/13
97					
98		⊞ Shroud Attachment & Re-Inforcement Manufacturing Development	70 days?	Fri 11/30/12	Tue 3/19/13



- **Balanced Rotor Assembly (Blisk) available late July 2013 - 1 month later than required**
  - **Driven by Final Aero release, Final Mechanical Design and Manufacturing**
    - **Lead times based on Build-1 design schedule, ROM quote & build-1 lead times**

Ref: Gen2Build2_Inducer-Blisk_2012-11-06.mpp

Pros/Cons: Bladed Disc Concept 1

Pros

- **Known configuration (too early to say proven)**
- **Damaged airfoils readily replaced**

Cons

- **Cost**
- **Schedule**
- **Gaps & steps in flowpath**
- **Will not readily accept all airfoil designs. Some may not be possible**

Pros/Cons: Blisk Concept 2d

Pros

- **Cost compared to bladed disc**
  - **Especially if manufacturing development taken out**
- **Schedule compared to bladed disc**
- **Less intensive mechanical design – free up resource earlier ?**
  - **May be offset by support required for Manufacturing Development**
- **No interblade gaps**
- **More consistent flowpath from airfoil to airfoil**
- **May be possible to allow smaller airfoil fillets at hub**
- **No airfoil fillet required (or possible) at shroud**

Cons

- **Composite Overwrap requires development – it's not an 'off the shelf' solution**
- **FOD to one airfoil could scrap entire blisk**
- **Vibration could be an issue – lack of mechanical damping**
- **Unable to provide as many lab tooth seals on shroud as bladed disc. 2 or 3 max.**
- **Adds additional joint in rotor – between blisk and shaft**

'Stacked' Attachment Concept

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Attachment stresses - basic

Analyses at 34500 RPM

Blade Version	**Gen2B2_Inducer_RxnBlade6_Blade.SLDPRT<-.009>**
Disc Attachment Geometry :	**TBD**
Skew Angle =	**0**

Part	Stress Type	Section	Width (in)	Axial Length (in)	CSA (in^2)	Load (lbf)	Average Stress (ksi)	Allowable Stress (ksi)				Stress Acceptable	Shed Speed (rpm)	% of disc burst Speed
								Yield	Ultimate	Creepstrain	Stress Rupture			
Blade	Tension	Blade neck	0.0900	2.347	0.2112	7314	34.6	100.5	65.6	?	?	Yes	75,085	#VALUE!
	Shear	Blade Shear	0.0790	2.347	0.1854	4218	22.8	45.0	32.8	-	-	Yes	(Target < 90% burst speed)	
	Bearing	Bearing	0.0500	2.347	0.1174	2420	20.6	120.0	-	-	-	Yes		
Disc	Tension	DiscPost Neck	0.0960	2.347	0.2253	10384	46.1	97.2	69.6	?	?	Yes	65,079	#VALUE!
	Shear	DiscPost Shear	0.1000	2.347	0.2347	4218	18.0	43.5	34.8	-	-	Yes	(Target < 90% burst speed)	
	Bearing	Bearing	0.0500	2.347	0.1174	2420	20.6	116.0	-	-	-	Yes		

Stresses acceptable based on proportionate load sharing

Ref: Gen2B2_Inducer_RxnBlade6_Blade.SLDPRT<-.009>

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Ti-5553 Powder Usage

Updated 5/26/2011

Update

	lbs
Total Used to date	377
Forecast Usage	129
Total Requirement	506
Initial Powder Stock	820
Powder remaining after Forecast Usage	314

Material Already Used & Weighed

Part No.	Description	Solidworks Vol (in^3)	Density	Wt (lb)	Weighed Wt Used (lb)	Weighed as % of CAD model	Notes
709029	Pancakes & Test 'half moons'	397	0.164	65	87	134%	Praewest Pancakes
709020 hub	Hub for 709020 (Exducer 1A)	185	0.164	30	40	132%	
709020 hub	Hub for 709020 (Exducer 1B	185	0.164	30	40	132%	
-	Fill stems	-	0.164		2		

Material Already Used but not yet weighed

Part No.	Description	Solidworks Vol (in^3)	Density	Wt (lb)	Predicted Weighed Wt (lb)	Notes
709020 flowpath	Flowpath for Exducer 1A	54	0.164	9	12	
716009 Rev D	Exducer 1C	294	0.164	48	64	
716005	HIP compact for Inducer 1B (qty2+2)	467	0.164	77	101	Sent to Barber Nichols. Current Status ?
-	UT Blocks	143	0.164	24	31	
			Total used to date		377	

Forecast Usage

Part No.	Description	Solidworks Vol (in^3)	Density	Wt (lb)	Predicted Weighed Wt (lb)	When required	Powder remaining (lbs)	
716031	BNI Blisk HIP Compact (qty 1)	445	0.164	73	97	1-Oct-10	346	Use for blade material ? Could yield 36-44 blades
716033	BNI Weld Block	149	0.164	24	32	1-Jan-11	314	Use for blade material ? Could yield 15 blades
			Forecast Future Usage		129			

Usage as % of Cad model (Average) 132%

Appendix 6.4

Static Diffuser FDR

Final Design Review

Generation 2
Static Diffuser – Case 15H Design
Starting Door Actuation System
Secondary Flows

System owner(s):

Mech: Geene Cevrero, Dave Taylor, Brian Massey, Rob Draper, Chris Braman, Jonathan Bucher, Kirk Lupkes, Ryan Edmonds

Aero: Paul Brown, Ravi Shrinivasan, Mark Krzystopik, Silvano Saretto, Ryan Edmonds

1/16/2012

Agenda

9:00-9:15	**System Definition and Scope**
9:15-9:30	**Functional Requirements**
9:30-10:00	**Review Action Items**
	Aero Design/Analysis
10:00-10:45	**– Static Diffuser/Radial Diffuser Results**
10:45-11:00	**– Performance Roll-Up**
11:00-12:00	**– Secondary Flows/Starting**
12:00-12:30	***Lunch Break 30min***
12:30-1:00	**– (cont) Secondary Flows/Starting**
	Mechanical Design/Analysis
1:00-2:30	**– Actuator System**
2:30-3:30	**– Starting Door**
	15 min Break
3:45-4:30	**– Shroud**
4:30-5:30	**– Static Diffuser Hub**
5:30-5:45	**Outstanding Work Plan/Analysis Tasks**
5:45-6:00	**Budget and Schedule**

System Definition and Scope

- **The Static diffuser comprises:**
  - **1 x Hub Component with 5 main Flowpaths & 10 Bleed Inserts (5 Fwd bleed, 5 Aft bleed)**
  - **1 x Shroud component to define the main flowpath outer annulus along with an integral bleed management system (aka door) and associated actuation hardware**

- **Static diffuser accepts flow exiting the inducer and converts total pressure to useful static pressure via a series of shock waves and gradual area changes while minimizing flow losses**
  - **Major components are diffuser hub, shroud, shroud door, and actuation system.**

- **Provides performance bleed through individual passages on hub and shroud (3 bleed circuits total: hub forward, hub aft, door bleed)**
  - **Provisions for future shroud forward bleed intact**

- **Shroud doors will simultaneously provide throat relief and additional aft bleed during diffuser starting. Shroud door will be attached to the shroud and have a flow path interface. Doors will have an external actuation system to provide required motion.**

- **External actuation system will provide necessary door motion for starting the system**

17-4PH Material Specs: continued

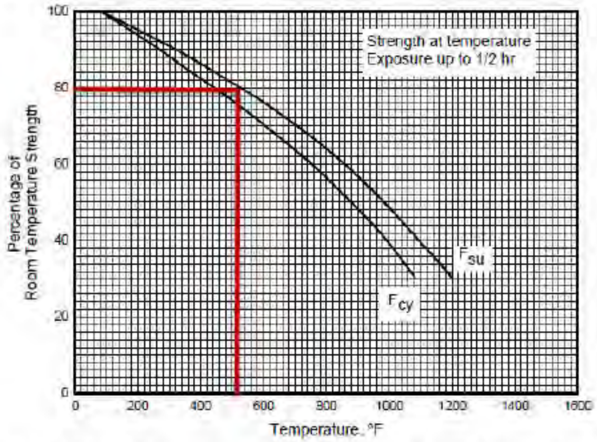


Figure 2.6.9.1.2. Effect of temperature on the compressive yield strength (F_{cy}) and the shear ultimate strength (F_{su}) of 17-4PH (H900) stainless steel bar and forging.

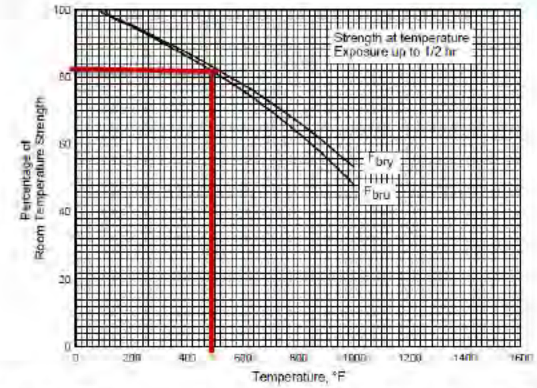


Figure 2.6.9.1.3. Effect of temperature on the bearing ultimate strength (F_{bru}) and the bearing yield strength (F_{bry}) of 17-4PH (H900) stainless steel bar and forging.

Design Point Temperatures are estimated to be approximately 500F which will render properties at approximately 80% of room temp.

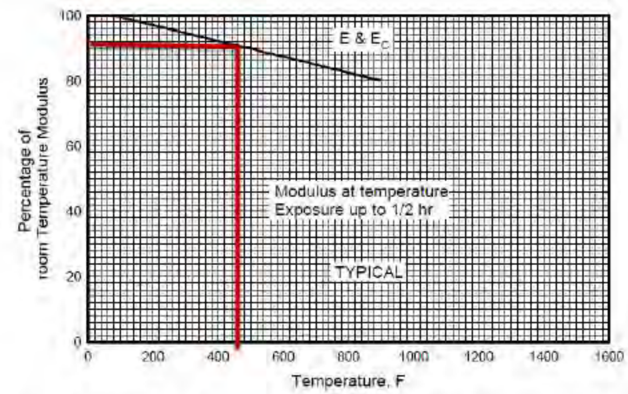


Figure 2.6.9.1.4. Effect of temperature on the tensile and compressive moduli (E and E_c) of 17-4PH (H900) stainless steel bar and forging.

Gen 2 Rampressor Aero Analysis
Final Design Review (CDR)

Paul M. Brown, Ravi Srinivasan, Mark Krzysztopik, Sabri Deniz, Logan Sailer, Silvano Saretto

16 January 2012

Agenda

- **Rampressor CFD Analysis**
  - **Geometry and CFD Model Description**
  - **Design Point CFD Analysis Results**
  - **Off-Design Starting CFD Analysis Results**
  - **Off-Design Started Low Backpressure CFD Analysis Results**
- **Rampressor Performance Rollup**
- **Conclusions**
- **Future Work**

Objectives

- Perform Coupled Simulation Of Inducer (Blade 12), Diffuser Case 15H Bleed 4b, And Subsonic Radial Turn Diffuser ("Medium Turn") With 70 Deg Vanes
- Provide Analysis Showing That Selected Rampressor Design Can Start and Remained Started Under Foreseen Operating Conditions
- Provide Performance Estimate (Flange-to-flange) Of Rampressor Stage
- Provide Mechanical Team With Sealing Requirements, And Pressure And Temperature Loads

Design Point Simulation of Case 15H
Inducer-Diffuser-Radial Turn

R. Srinivasan, Paul M. Brown

Grid Details (000 Level)

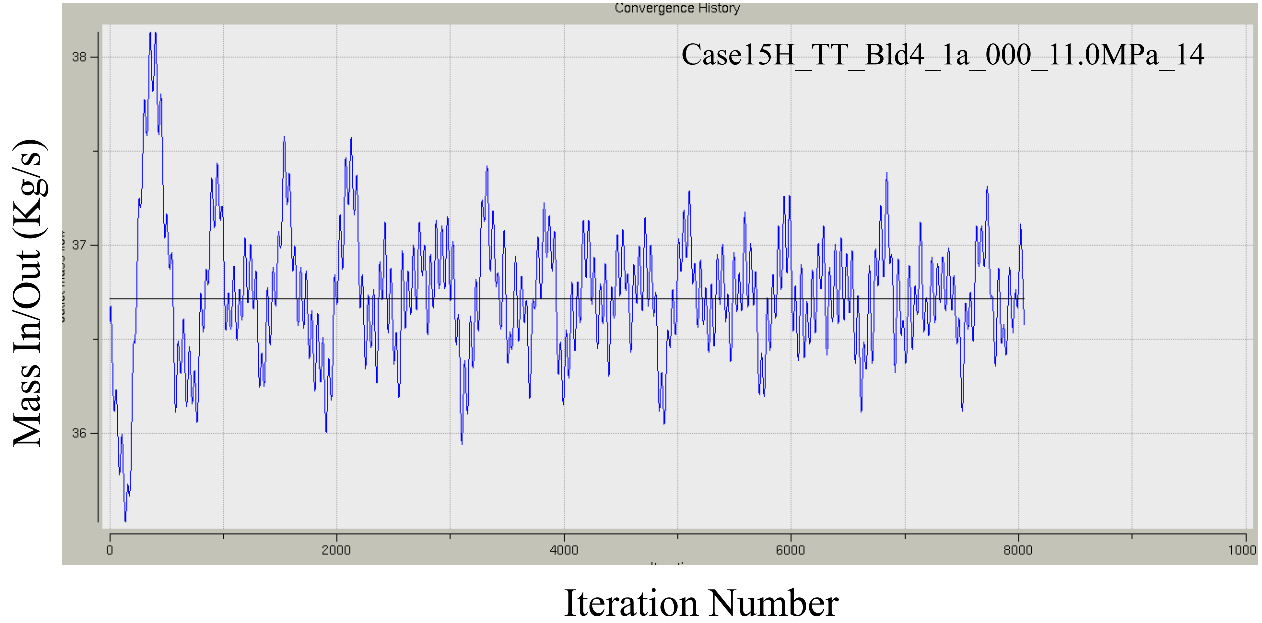
- Inducer Grid Generated Based On Numeca's Template With Modifications To Accommodate Axial Separation Between Inducer And Diffuser.
  - Number Of Grid Points Is Approximately 9M
- Diffuser Grid Generated Using Ramgen Topology
  - Includes Part Of The Subsonic Radial Turn
  - Number Of Grid Points Is Approximately 63M
- Radial Turn With Vanes
  - Grid Includes Two Passages To Ensure Proper Transfer Of Data To The Diffuser Blocks.
  - Number Of Grid Points Is Approximately 14M (Includes Both Passages)
- Bleed/Shroud Gap Grid Was Adapted From Existing Bleed 4b Model.

Total Number Of Grid Points In The Model Is ~ 119M. The Wall Distance Of The First Cell Center In The Diffuser And Radial Turn Is 1.3e-7 m.

CFD Simulation Details

- Simulations Performed Using Fine Turbo Condensable CO2 Model
- SA Turbulence Model
- Total Pressure And Total Temperature Specified At The Inlet Along With Velocity Direction.
  - Pt = 210 Psia (1447899 Pa)
  - Tt = 100 ^{O}f (311 K)
  - Vz/|V| = 1
  - Uniform Inlet Profile
- Average Pressure Specified At The Outlet
  - Radial Turn Exit – Varying Pressure Values
  - Shroud And Aft Hub Bleed Cavities – 300 Psia (2068000 Pa)
  - Fwd Bleed Cavity – 120 Psia (827370 Pa)
- Rotation
  - Inducer Walls At 29400 RPM

Mass Convergence



- **Outlet Mass Flowrate Exhibits Larger Fluctuations Than Diffuser Only Simulations**
- **Fluctuations Are Likely Caused By Unsteady Separated Regions Behind Radial Turn Vanes and In Bleed Cavities**
- **Convergence is Deemed Acceptable For Performance Prediction**

Off-Design Starting CFD Analysis Results
Inducer-Diffuser-Radial Turn

Paul M. Brown

Flowpath Starting Background

- **Current Approach Utilizes A Door Hinged At The Back For Starting**
- **Opening Of The Door Provides Both Throat Area Relief As Well As A Bypass Flow To Allow Starting Of The Supersonic Diffuser**
  - **Door LE Selected Based On 1D Self Starting Area At M=2.4 Including Effect Of Forward Bleed**
  - **Door TE And Door Opening Selected Based On 1D Self Starting Area At M=2.4 Including Effect Of Diffuser Bleed**
- **Bypass Flow is Re-injected Into Flow Loop Intermediate Pressure Volume To Alleviate Auxiliary Compressor Flowrate Requirements**

Flowpath Started, Door Open 3D CFD Analysis

- **3D CFD Analysis Of The Supersonic Diffuser With The Starting Door Open At Full Speed Has Been Conducted**
- **Analysis Shows That Flowpath Is Started And That Bypass Circuit Backpressure Is Compatible With Flow Reinjection In Into Flow Loop Intermediate Pressure Volume**
- **At Door Nominal Opening Angle Bypass Flowrate Is Higher Than Target**
- **Details Of The Analysis Will Be Covered As Part Of Secondary Flow Aero Analysis**

2D And 3D CFD Analyses Indicate That Selected Starting Strategy Allows Starting Of The Supersonic Diffuser With Margin

Off-Design Started Low Backpressure CFD Analysis Results

Inducer-Diffuser-Radial Turn

Sabri Deniz

Background And Approach

- Starting Of The Supersonic Diffuser Is Currently Envisioned To Be Accomplished With Minimum Backpressure (~300 Psia)
- At This Condition Flow Entering The Radial Turn Is Highly Supersonic
- Once The Starting Door Are Closed The Entire Mass Flowrate Processed By The Inducer Minus The Bleed And Leakage Flows Must Pass Through The Radial Turn
- Excessive Blockage From The Radial Turn + Volute Might Prevent Starting Of The Supersonic Diffuser
- A Coupled Inducer-diffuser-radial Turn Low Backpressure Simulation Was Performed To Determine Performance Of The Radial Turn Under These Conditions

Off-design Inducer + Diffuser + Radial Turn Inlet and Exit Conditions

	INLET	EXIT
m [kg/sec]	36.72	30.28 (+ bleed flow)
M [-]	0.57	0.71
Pt [psia]	210.0	477.15
Pst [psia]	171.5	328.81
Tt [K]	311	522.9
Tst [K]	296.7	489.3
Flow Angle [º]		26.8
V [m/s]		237.6
Vt [m/s]		99.2
Vr [m/s]		193.0

Exit Mass Flow Oscillating Due To Unsteady Flow Separation, Vortex Shedding In The Radial Turn, Around The Vanes

Conclusions

- Simulation Results Indicate, That The Radial Turn Passes The Mass Flow At Off-design Conditions (High, Supersonic Inlet Mach Number, Low Exit Static Pressure) During The Starting
- The Flow Is Unsteady, With Large Separation Regions Around The Vanes
  - Shock Induced Separation Occur On The Vanes Surface (The Location Of The Separation And The Size Of The Separated Region Depend On The Exit Pressure!)
- Inducer + Diffuser + Radial Turn Cfd Simulation At Low Back-pressure (300 Psia) Is Not Converging (Exit Mass Flow Oscillating)
- Unsteady Flow, With Large Separated Regions (Jet-wake Type Flow) Discharging To The Volute
- Well Designed Volutes Can Handle Non-uniform, Separated Inflow Conditions, But There Is No Information Or Data Available For High Inlet Mach Number Flows
- Similar Concern For Flow Angle Change At Volute Inlet (± 90deg Change In Flow Direction)
- Should Consider Including Volute In The Model (Inducer, Diffuser, And Radial Turn) And Run An Unsteady CFD Analysis

Appendix 6.5

Facility P&ID

DRESSER-RAND OLEAN, N.Y.

HIGH PRESSURE CO2 TEST LOOP

FLOW DIAGRAMS & P& I DIAGRAMS DRAWING LIST

REV	DESCRIPTION	MADE BY	APP BY	DATE
4	UPDATED DWG REVISIONS	GAW	MAS	6/9/11
3	UPDATED DWG REVISIONS	TD	MAS	3/21/11
2	UPDATED DWG REVISIONS	MGR	MAS	12/11/10
1	UPDATED DWG REVISIONS	MGR	MAS	10/22/10
0	ISSUED FOR CONSTRUCTION	DAW	MAS	10/22/10

REVISIONS

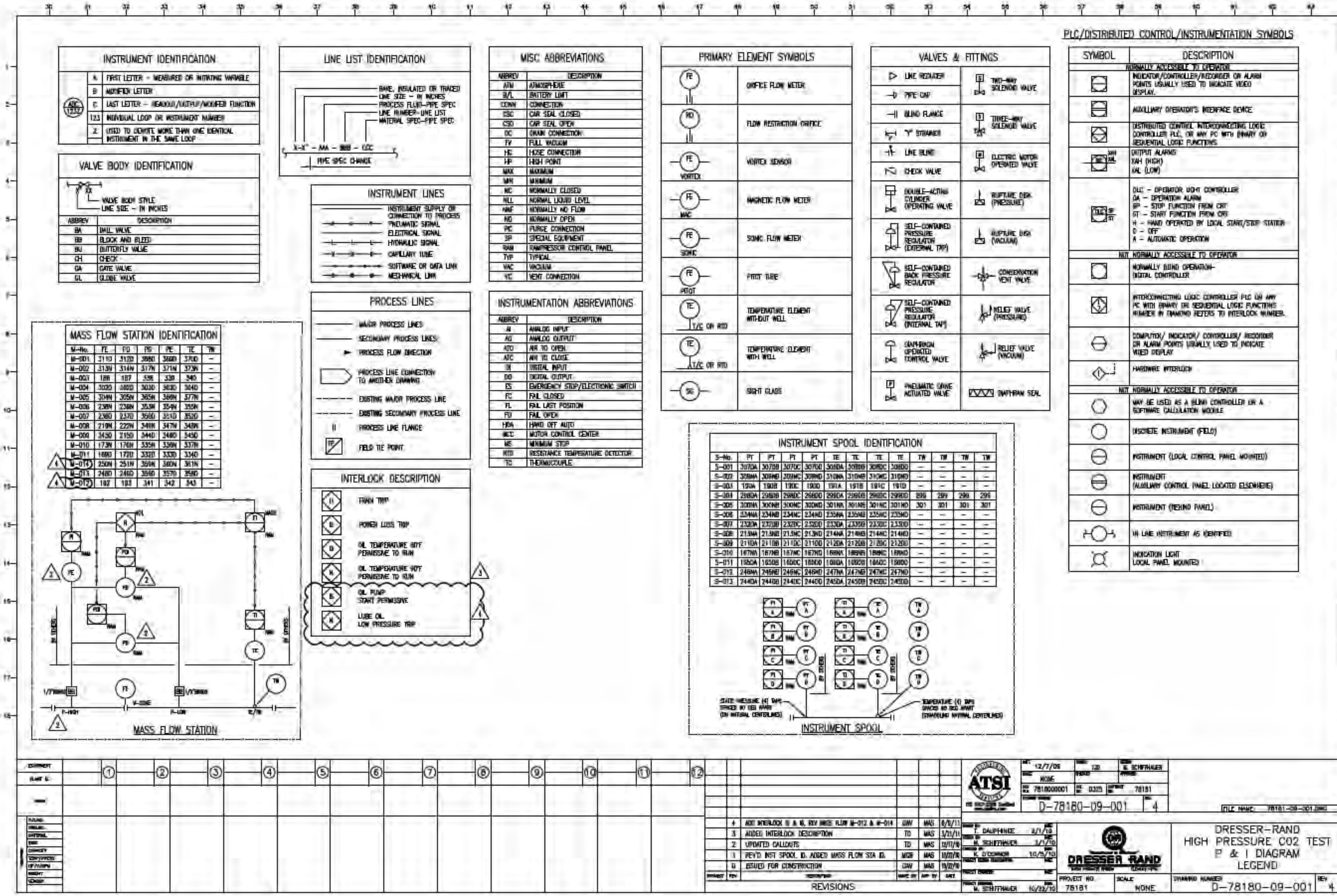
T. DAUPHINEE 1/29/10
M. SCHIFFHAUER 1/29/10
K. O'CONNOR 10/5/10
M. SCHIFFHAUER 10/22/10

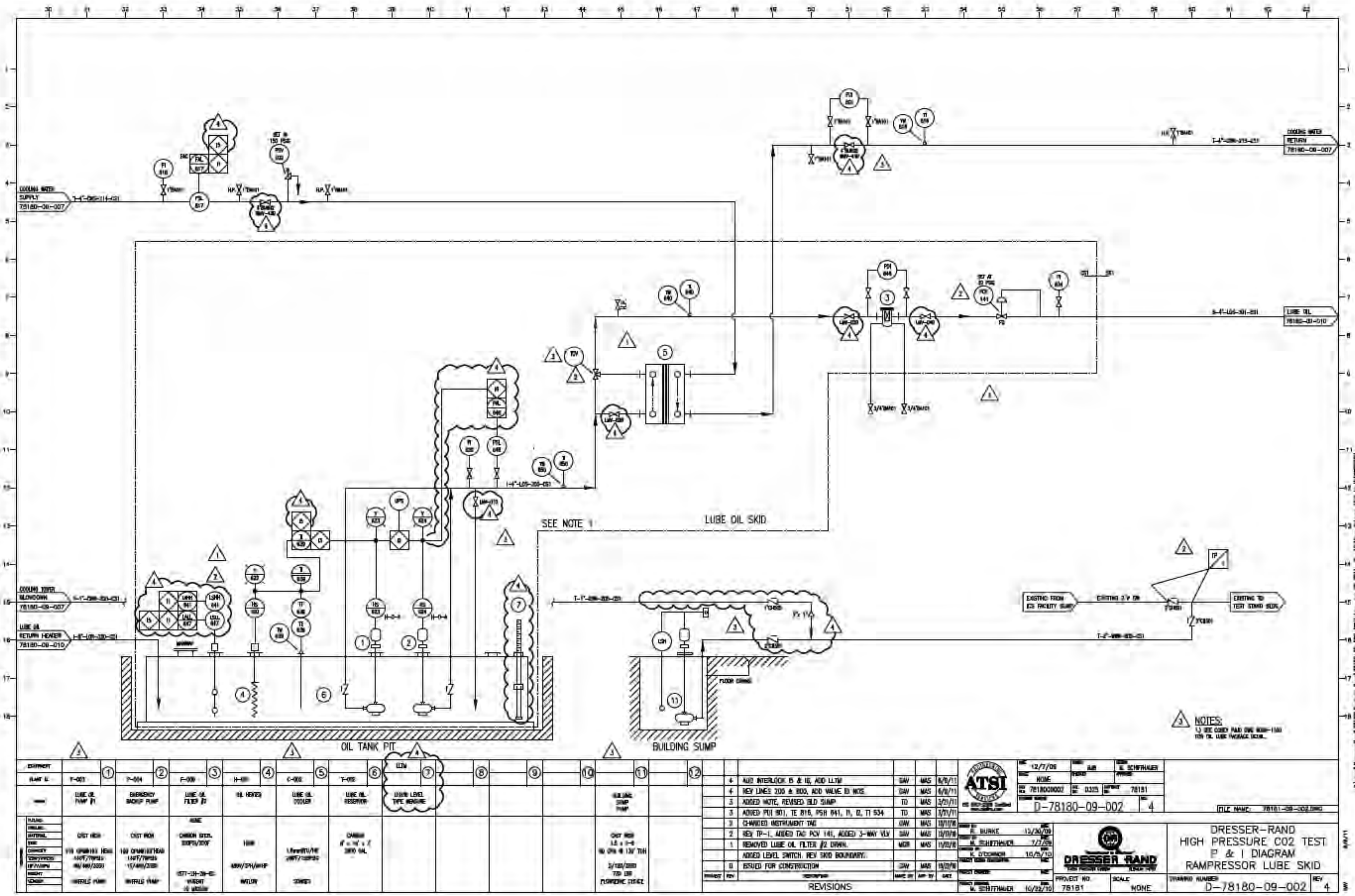
1/29/10 — TJD — M. SCHIFFHAUER
NONE
7818000000 — 78180
4

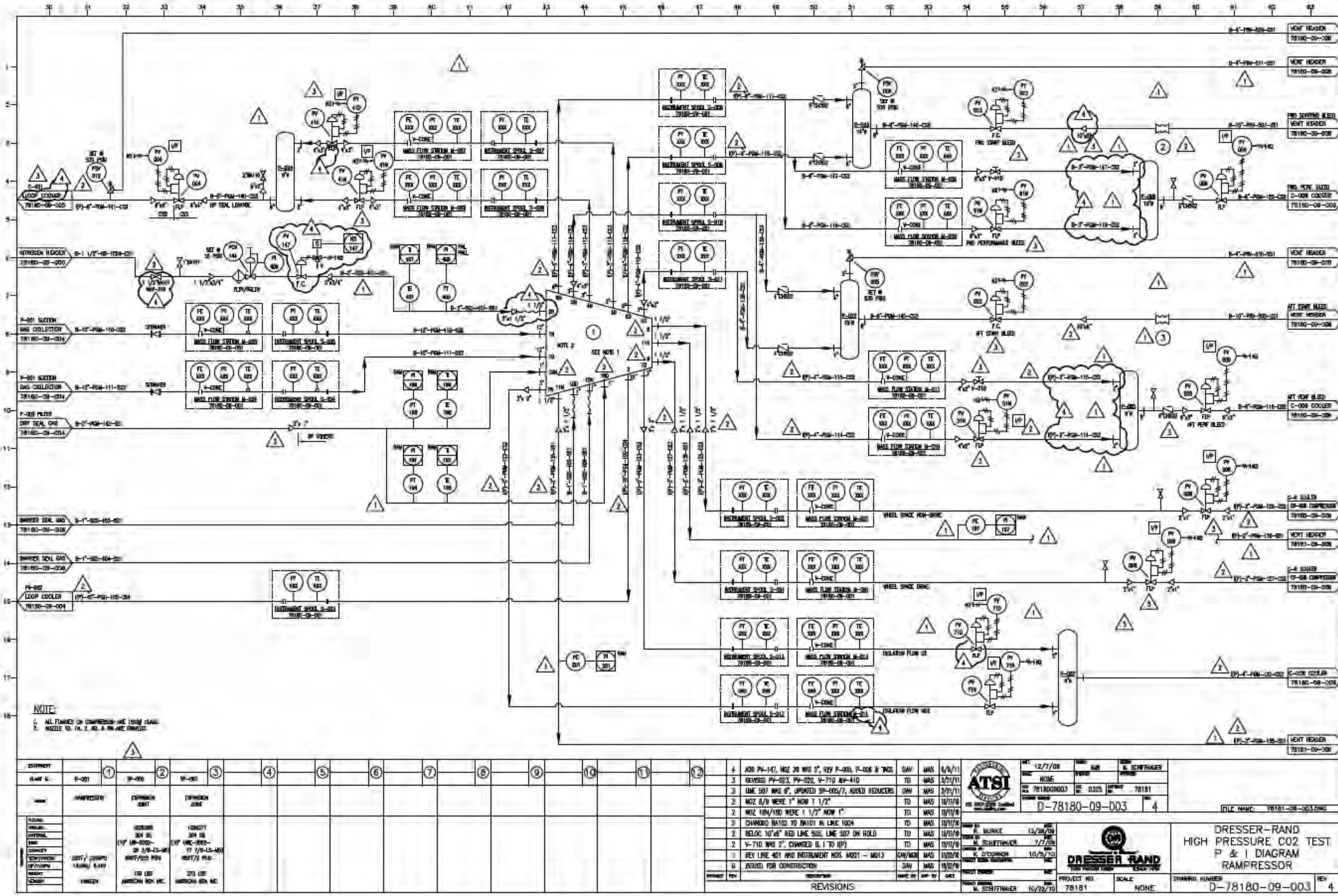
FILE NAME: 78180-09-000.DWG

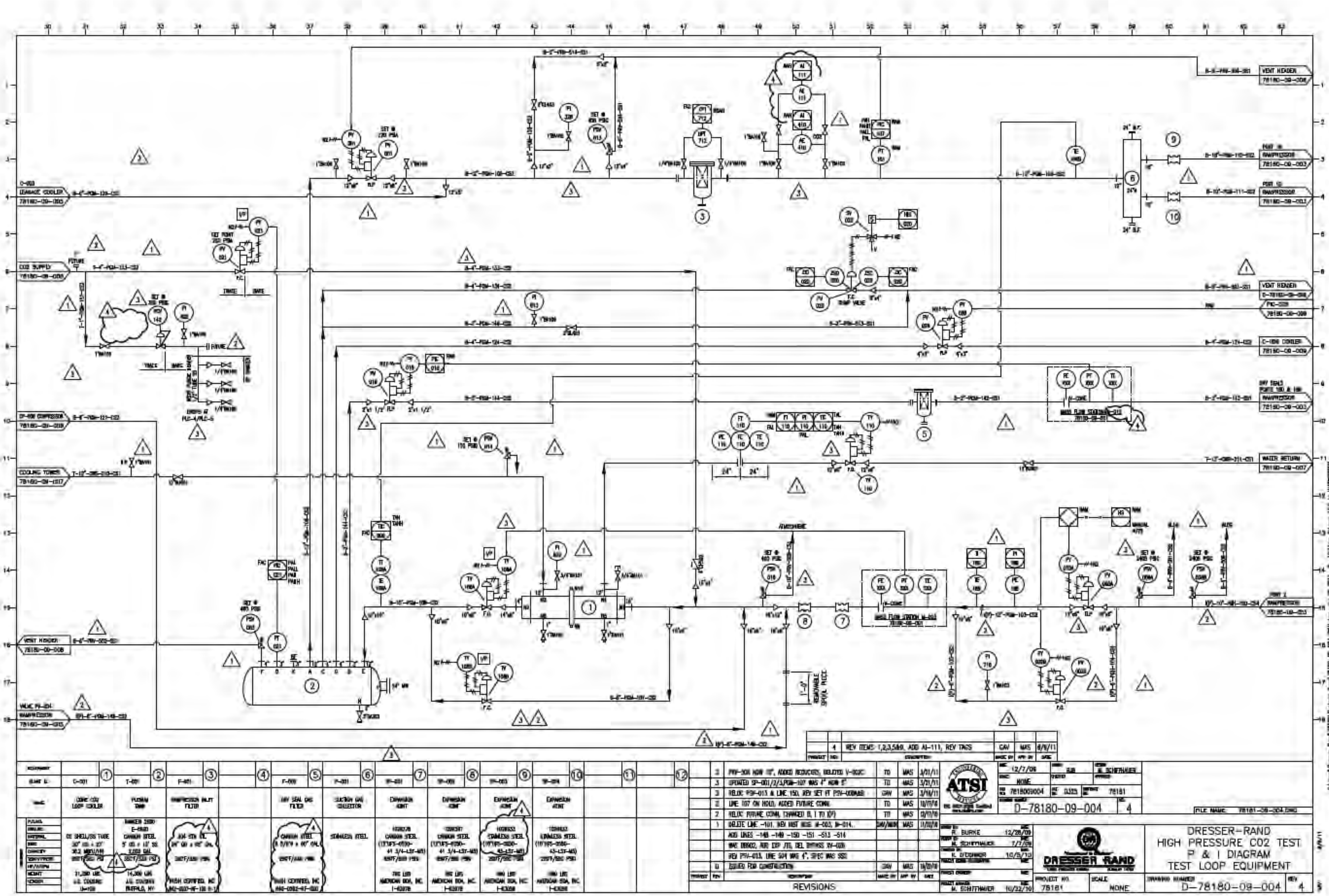
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
COVER SHEET

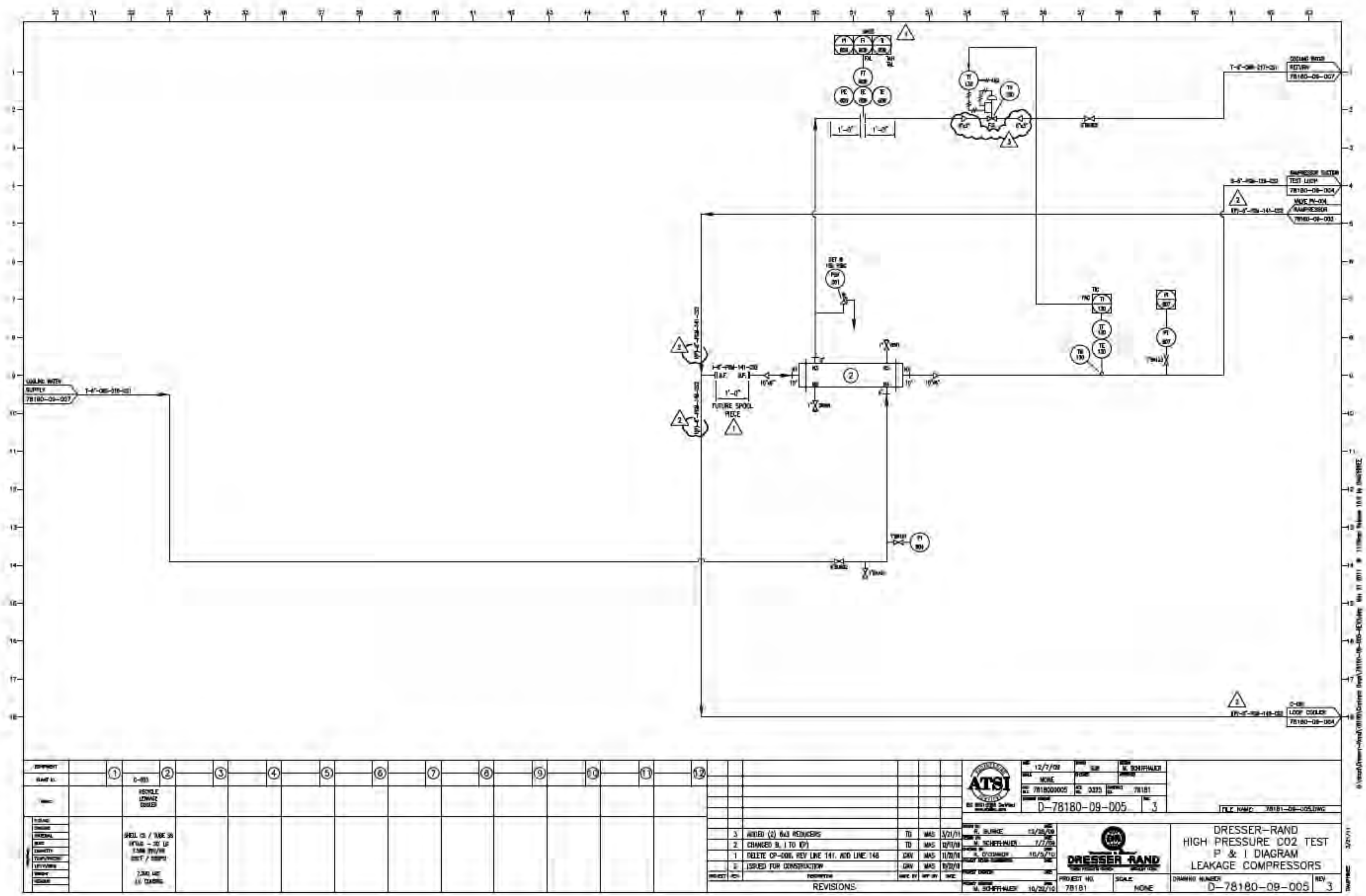
PROJECT NO. 78180 — SCALE NONE — DRAWING NUMBER D-78180-09-000 — REV 4

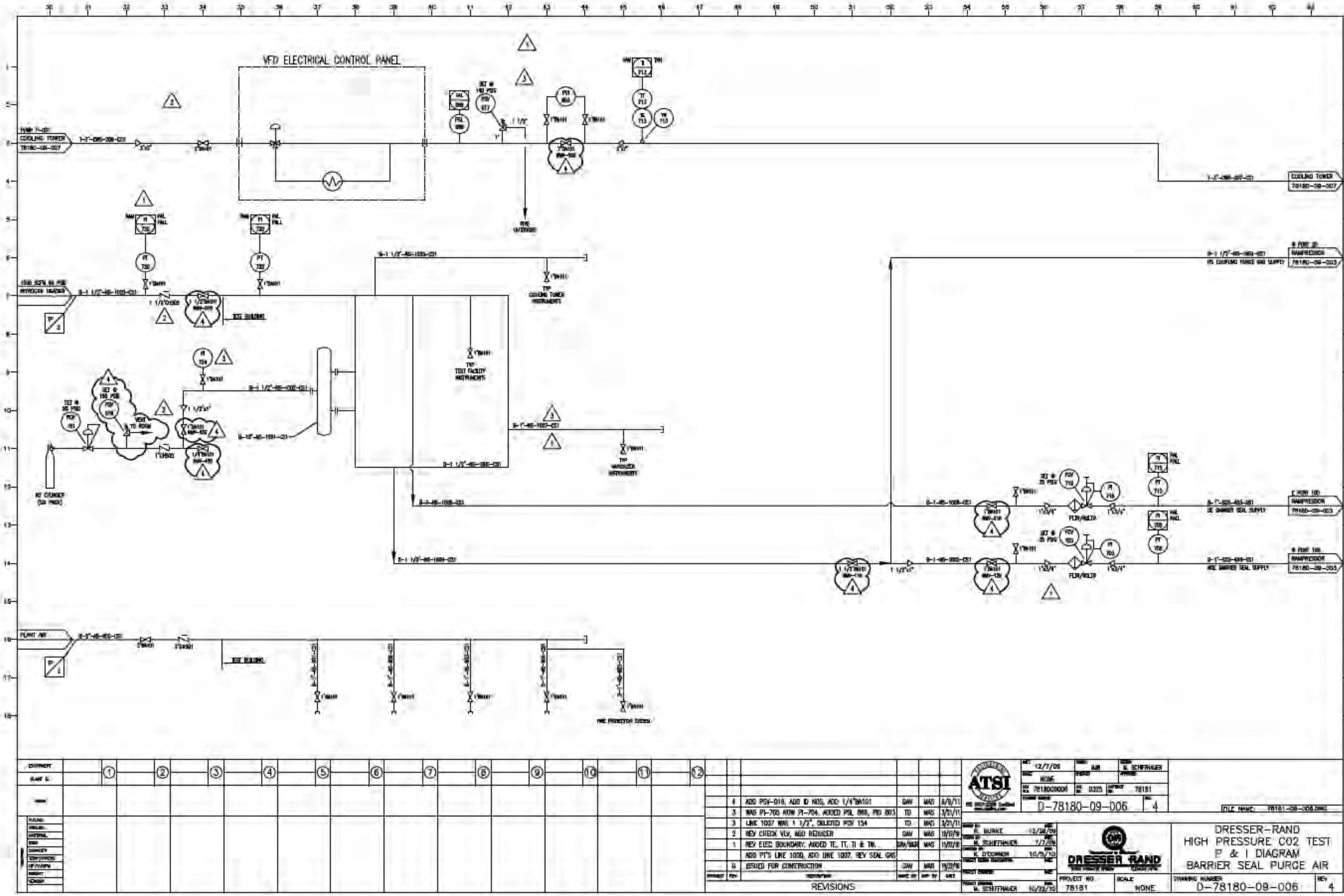

PLC/DISTRIBUTED CONTROL/INSTRUMENTATION SYMBOLS
SYMBOL
DESCRIPTION
INSTRUMENT IDENTIFICATION
LINE LIST IDENTIFICATION
MISC ABBREVIATIONS
PRIMARY ELEMENT SYMBOLS
VALVES & FITTINGS
VALVE BODY IDENTIFICATION
INSTRUMENT LINES
PROCESS LINES
INSTRUMENTATION ABBREVIATIONS
MASS FLOW STATION IDENTIFICATION
INTERLOCK DESCRIPTION
INSTRUMENT SPOOL IDENTIFICATION
MASS FLOW STATION
INSTRUMENT SPOOL
REVISIONS
ATSI
D-78180-09-001
DRESSER-RAND
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
LEGEND
D-78180-09-001

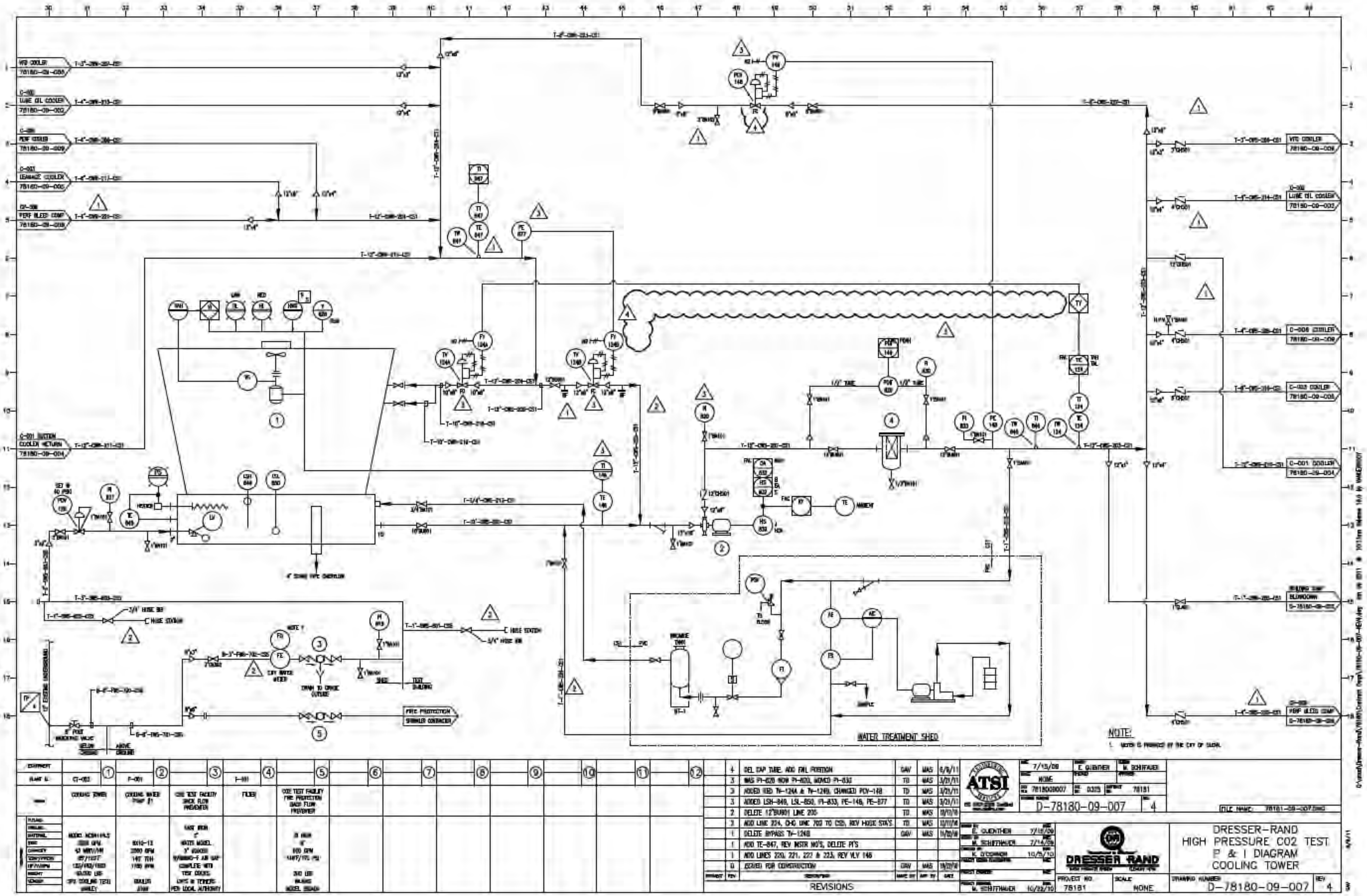

SEE NOTE 1
LUBE OIL SKID
OIL TANK PIT
BUILDING SUMP
NOTES:
REVISIONS
4 ADD INTERLOCK 15 & 16, ADD LLTM
4 REV LINES 200 & 800, ADD VALVE ID NOS.
3 ADDED NOTE, REVISED BLD SUMP
3 ADDED PDI 901, TE 816, PSH 841, 11, 12, TI 534
3 CHANGED INSTRUMENT TAG
2 REV TP-1, ADDED TAG PCV 141, ADDED 3-WAY VLV
1 REMOVED LUBE OIL FILTER #2 DRAIN.
ADDED LEVEL SWITCH, REV SKID BOUNDARY.
0 ISSUED FOR CONSTRUCTION
ATSI
12/7/09
NONE
D-78180-09-002
4
R. BURKE
M. SCHIFFHAUER
K. O'CONNOR
DRESSER-RAND
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
RAMPRESSOR LUBE SKID
PROJECT NO. 78181
SCALE NONE
DRAWING NUMBER D-78180-09-002
REV 4

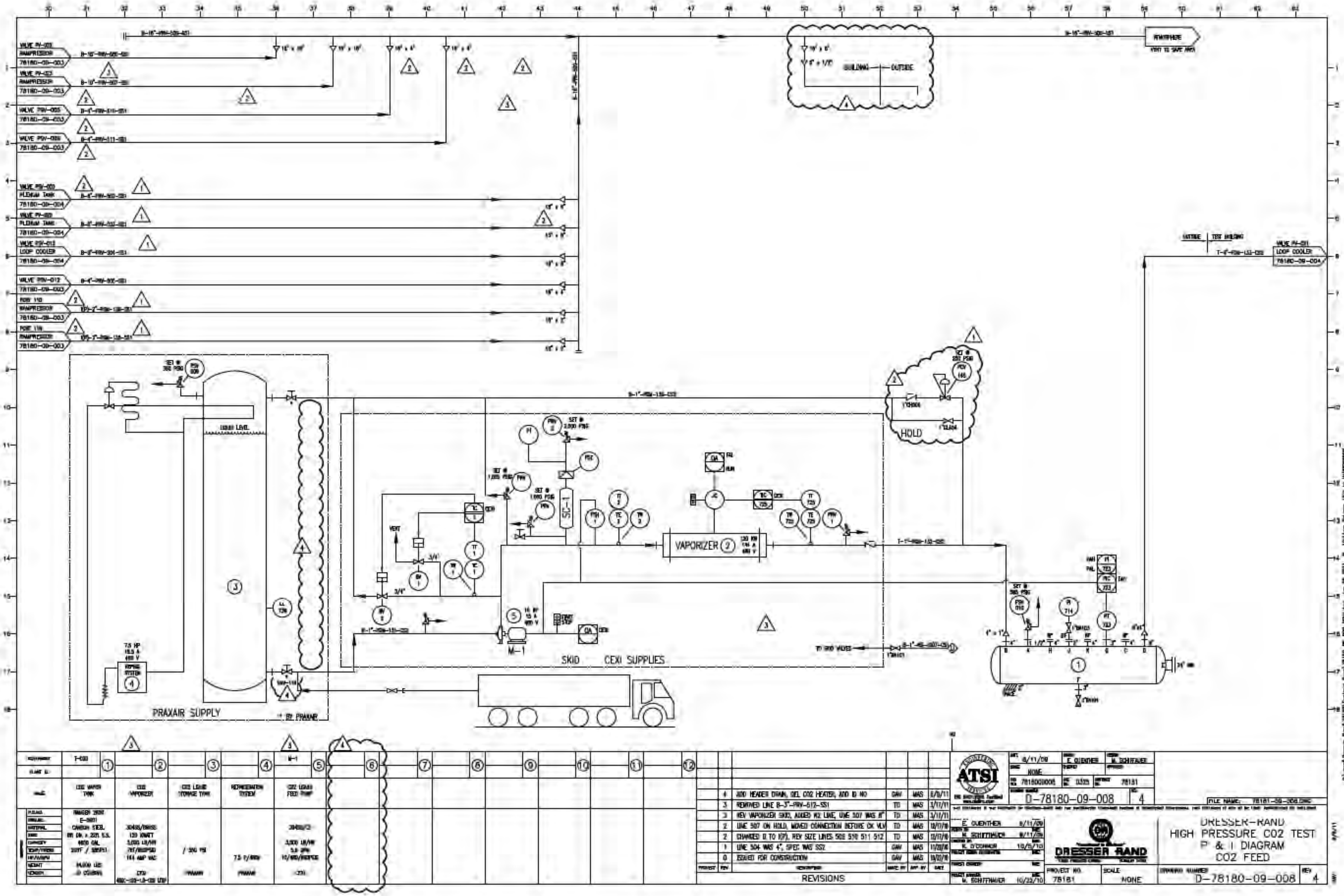

NOTE:
REVISIONS
ATSI
D-78180-09-003
DRESSER-RAND
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
RAMPRESSOR
D-78180-09-003
4

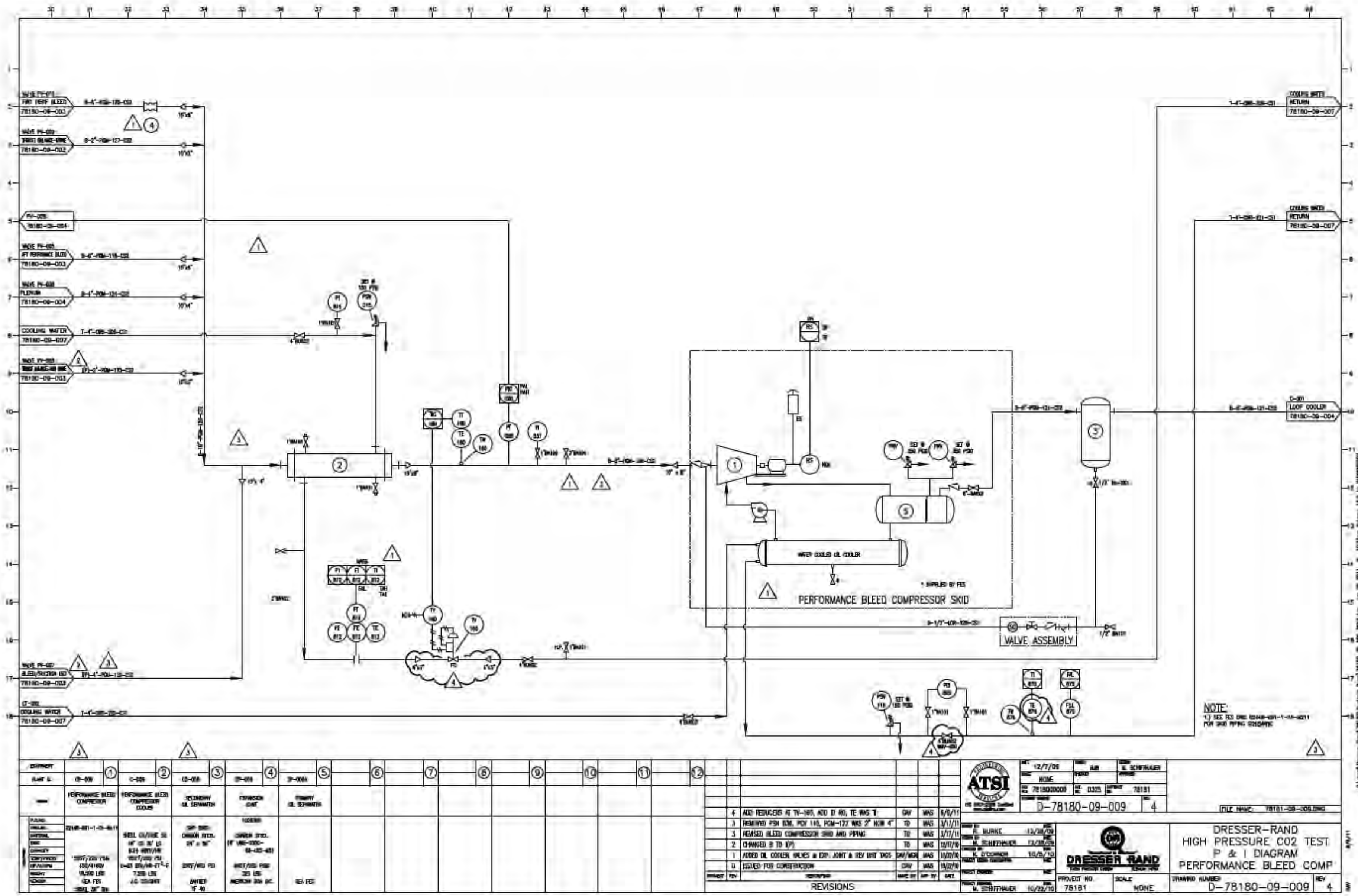

DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
TEST LOOP EQUIPMENT
D-78180-09-004
REVISIONS

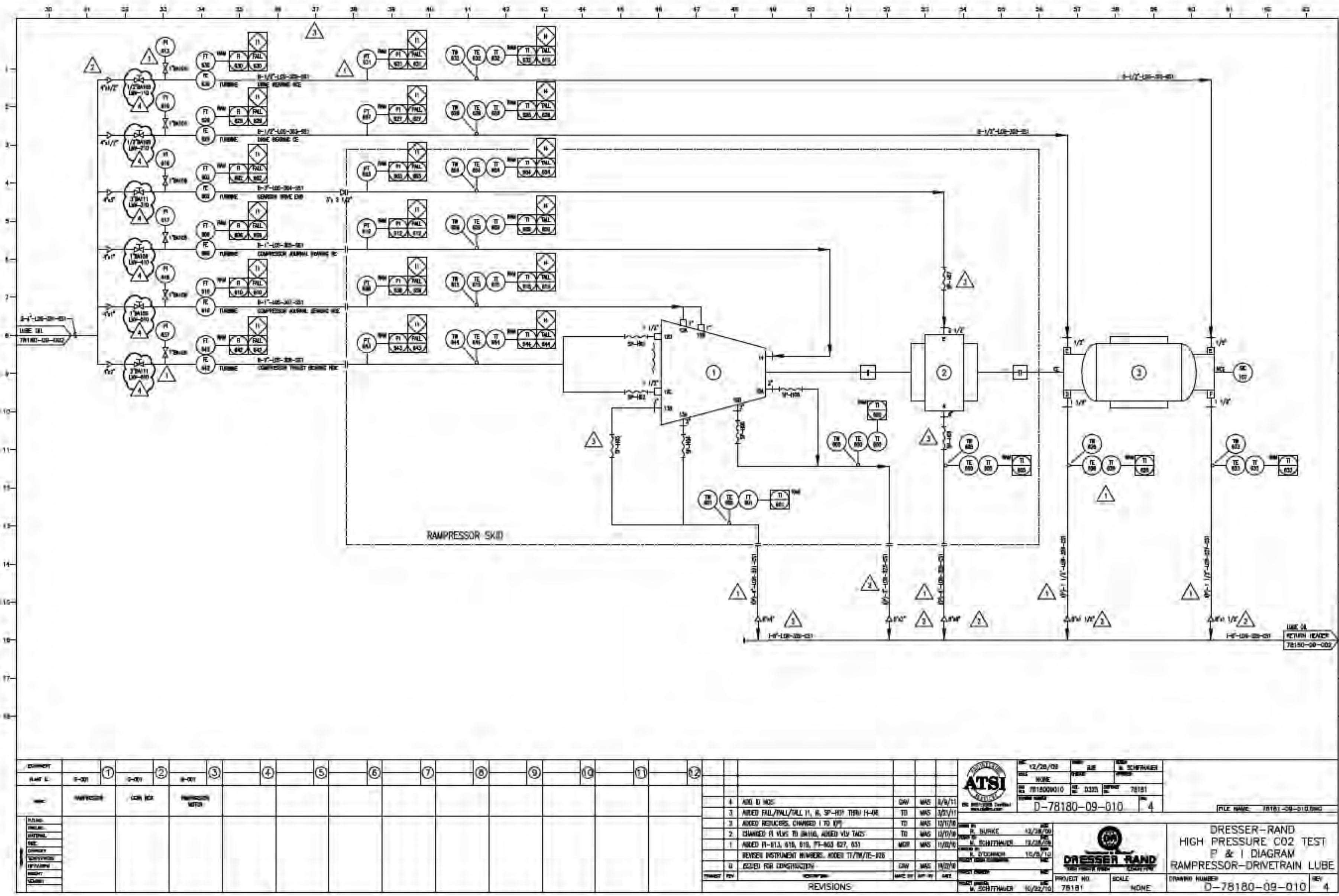

DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
LEAKAGE COMPRESSORS
D-78180-09-005
REVISIONS
FUTURE SPOOL PIECE


VFD ELECTRICAL CONTROL PANEL
COOLING TOWER
78180-09-007
N2 CYLINDER
PLANT AIR
REVISIONS
4 ADD PSV-010, ADD ID NOS, ADD 1/4"BA101
3 WAS PI-705 NOW PI-704, ADDED PSL 860, PID 803
3 LINE 1007 WAS 1 1/2", DELETED PDV 154
2 REV CHECK VLV, ADD REDUCER
1 REV ELEC BOUNDARY, ADDED TE, TT, TI & TN
ADD PT'S LINE 1000, ADD LINE 1007, REV SEAL GAS
0 ISSUED FOR CONSTRUCTION
ATSI
D-78180-09-006
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
BARRIER SEAL PURGE AIR
D-78180-09-006


WATER TREATMENT SHED
NOTE:
REVISIONS
ATSI
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
COOLING TOWER
D-78180-09-007
4


VAPORIZER
SKID
CEXI SUPPLIES
PRAXAIR SUPPLY
HOLD
M-1
SC-1
REVISIONS
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
CO2 FEED
D-78180-09-008
ATSI
DRESSER-RAND


PERFORMANCE BLEED COMPRESSOR SKID
VALVE ASSEMBLY
NOTE:
REVISIONS
D-78180-09-009
DRESSER-RAND
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
PERFORMANCE BLEED COMP
D-78180-09-009


RAMPRESSOR SKID
REVISIONS
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
RAMPRESSOR-DRIVETRAIN LUBE
D-78180-09-010

Appendix 6.6

Drivetrain Repeatability

Drive Train Repeatability Assessment

HP CO_2 Compressor Development Program

Brian Massey

Report Date: 6 December 2012

11/16/2012 and 11/17/2012 Nyquist Plot Comparisons

The following data is for runs with 100psia loop pressure on 11/16/2012 and 11/17/2012. The Nyquist plots are generated at specific steady state dwell points (listed in motor rpm). Each plot contains 10-15s of data shortly after arrival at the dwell speed. No changes in drive train components are made between the two days. Differences in aerodynamic configuration (bleed cavity pressure, wheel space pressure, bleed mass flow, etc) are neglected in the comparisons. The aerodynamic configuration may or may not have an effect on drive train vibrations behavior.

Previous observations noted that sub-synchronous vibrations were present in the signal at lower speeds. The sub-synchronous vibrations reduce sharply in amplitude once motor speed reaches 2800rpm. For this reason, it is expected that day to day repeatability will be better at speeds above 2800rpm versus 900rpm and 1600rpm. Figures 1-16 contain comparisons between the two days at several locations on the drive train for 900-1600rpm motor speed (8856-15744rpm compressor speed).

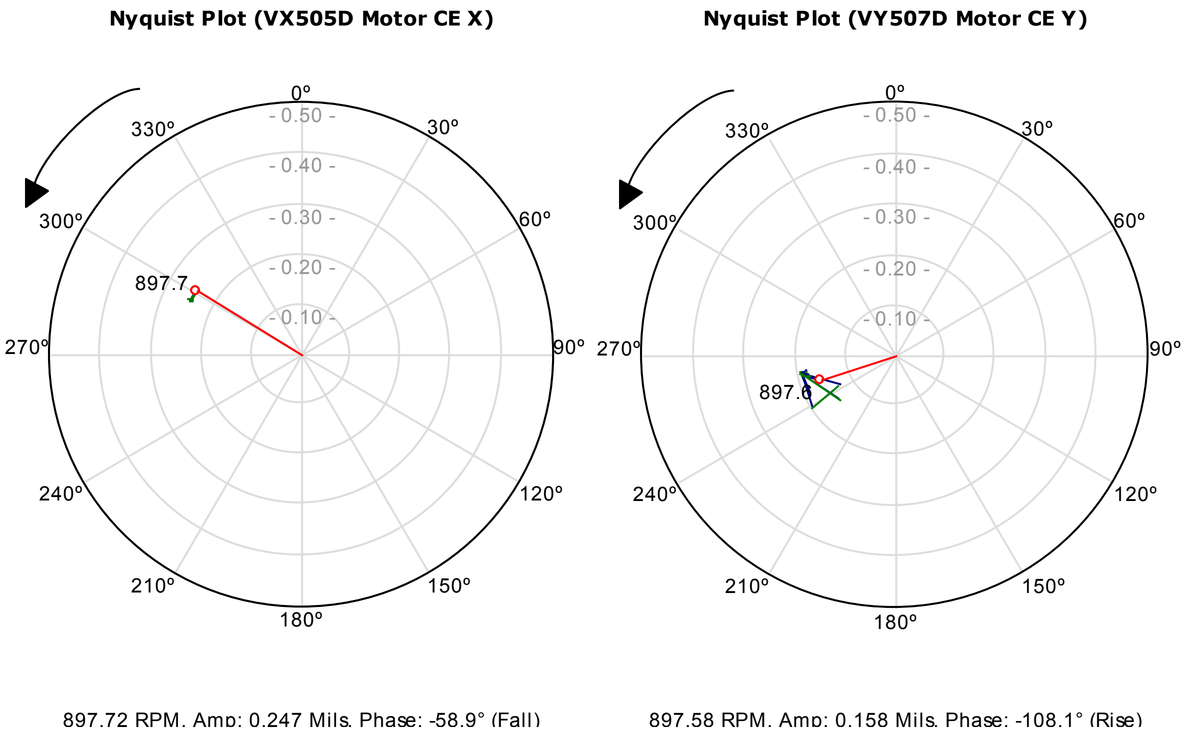


Figure 1: 11/16/2012, 900rpm, 100psi, Motor CE X and Y

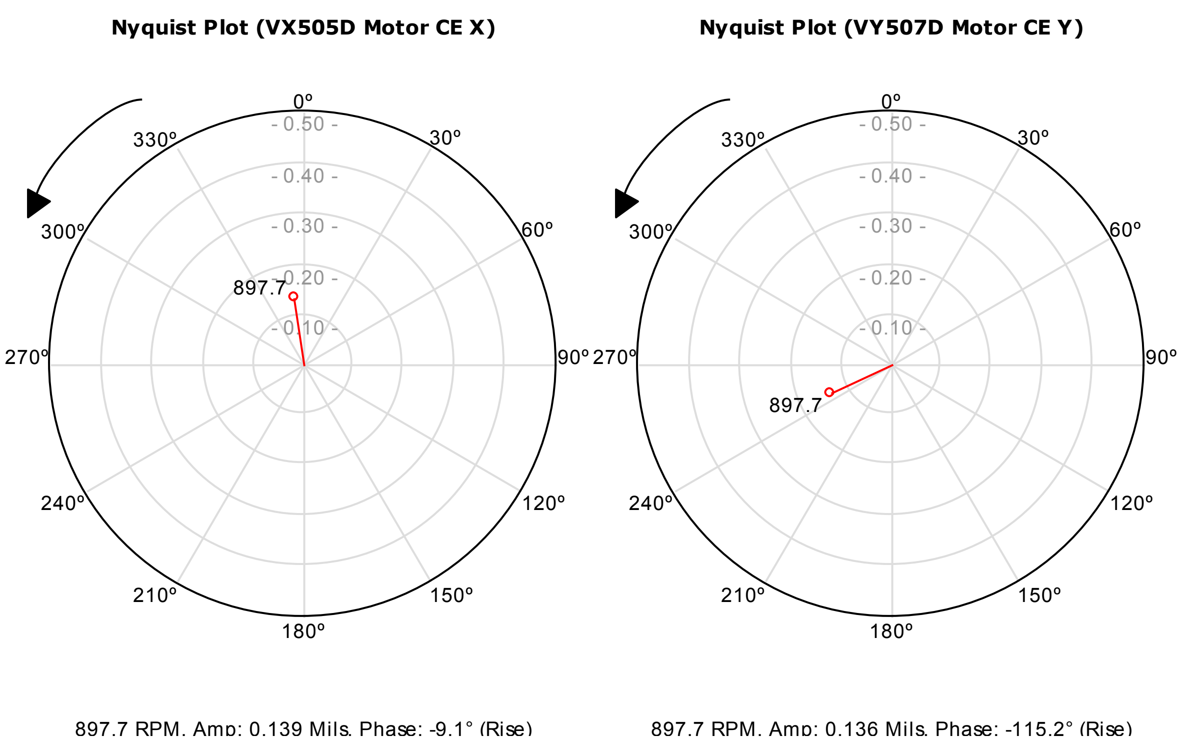


Figure 2: 11/17/2012, 900rpm, 100psi, Motor CE X and Y

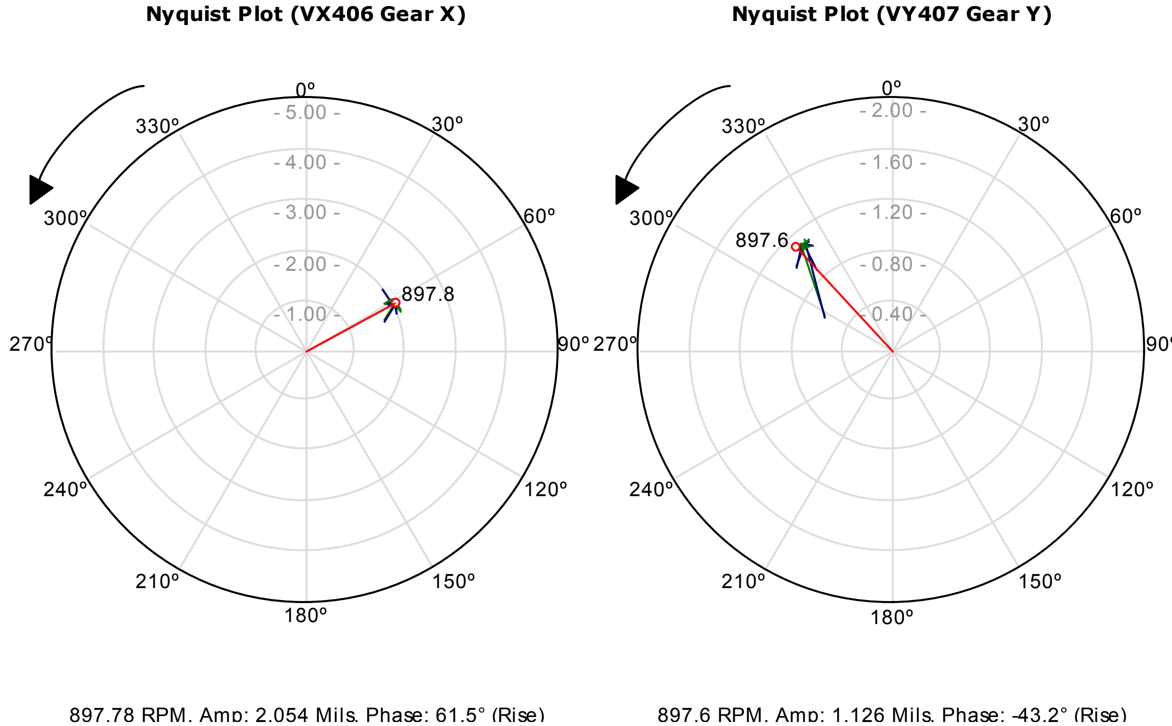


Figure 3: 11/16/2012, 900rpm, 100psi, Gearbox input X and Y

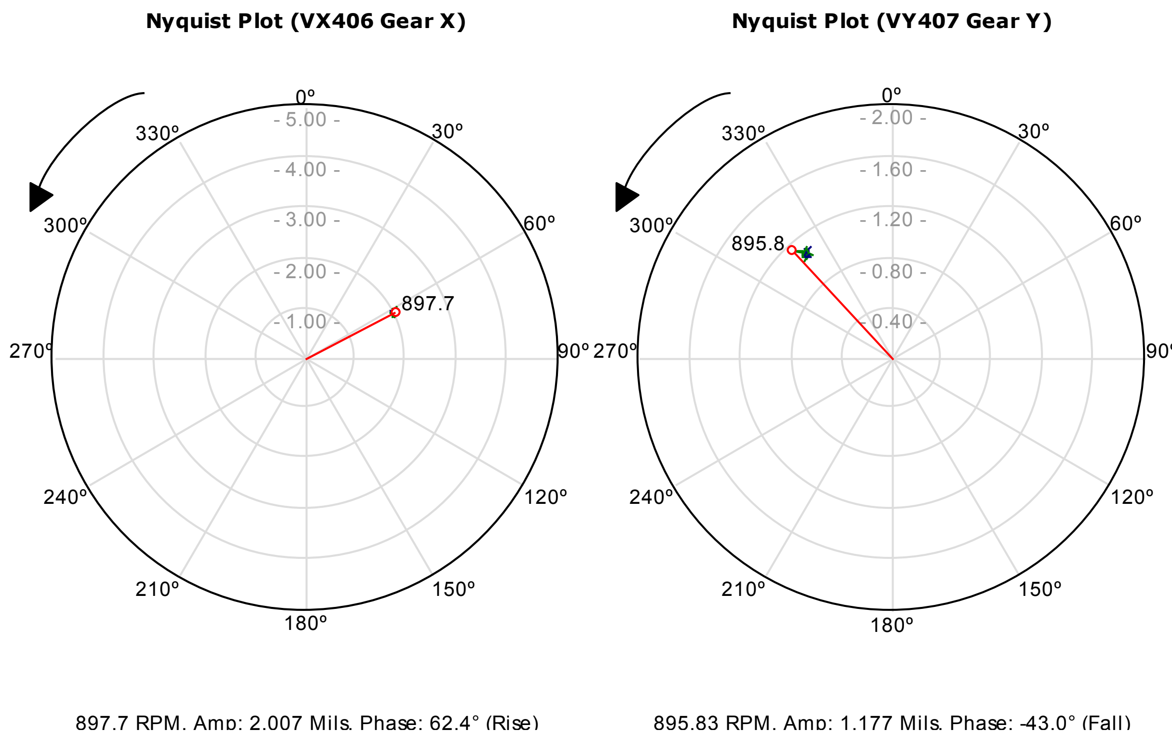


Figure 4: 11/17/2012, 900rpm, 100psi, Gearbox input X and Y

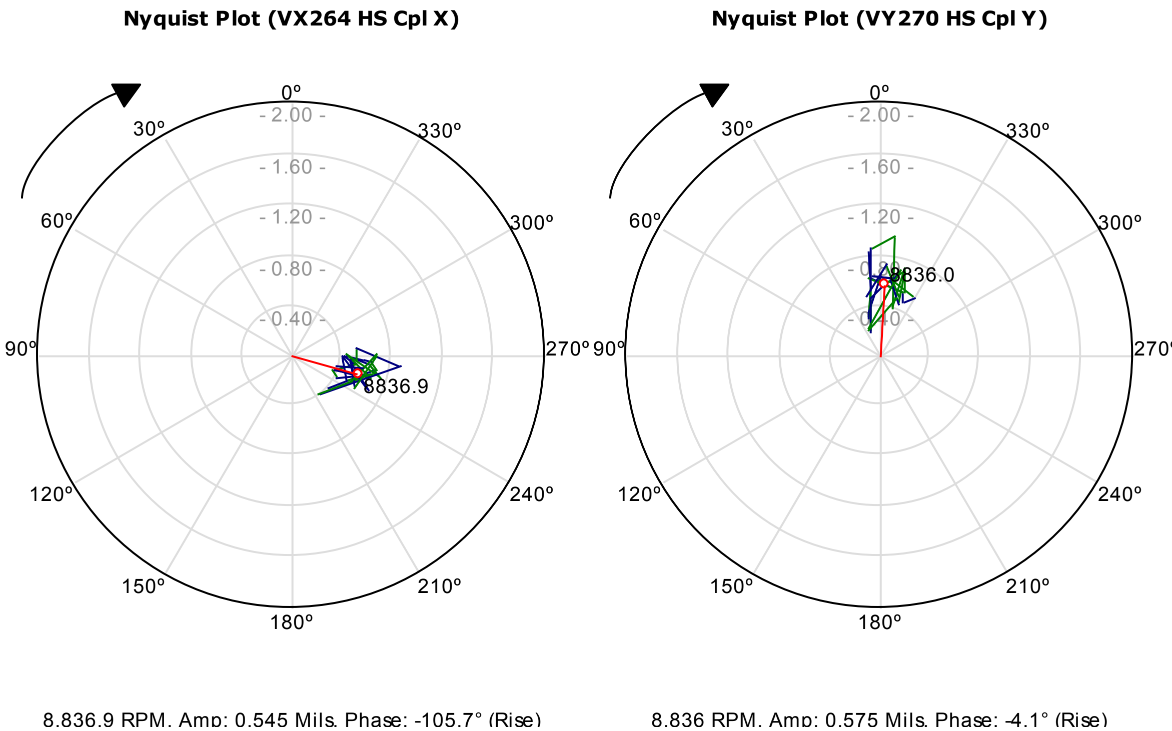


Figure 5: 11/16/2012, 900rpm, 100psi, HS coupling X and Y

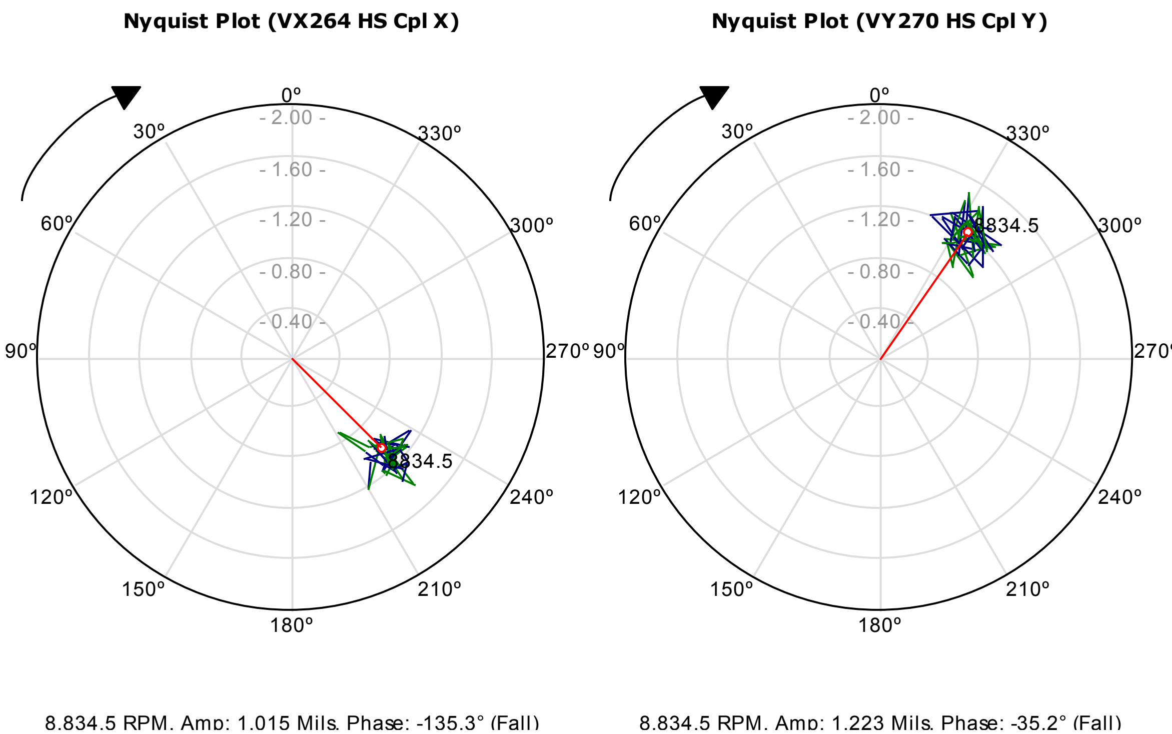


Figure 6: 11/17/2012, 900rpm, 100psi, HS coupling X and Y

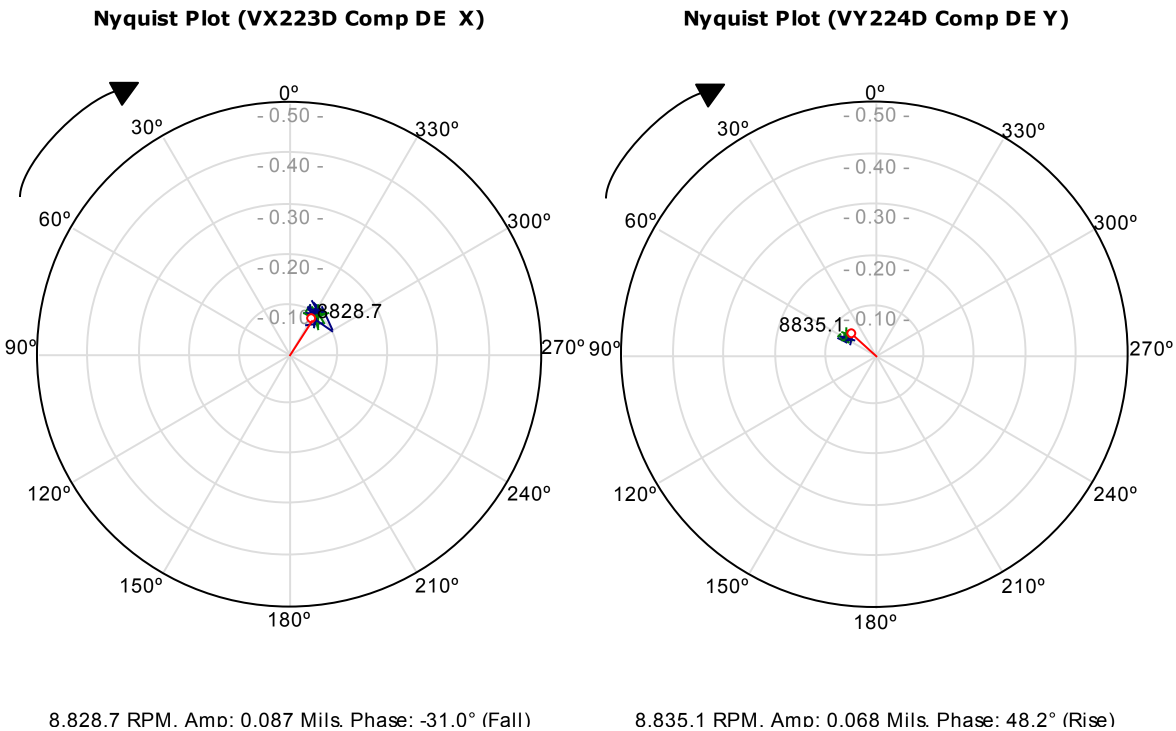


Figure 7: 11/16/2012, 900rpm, 100psi, Compressor DE X and Y

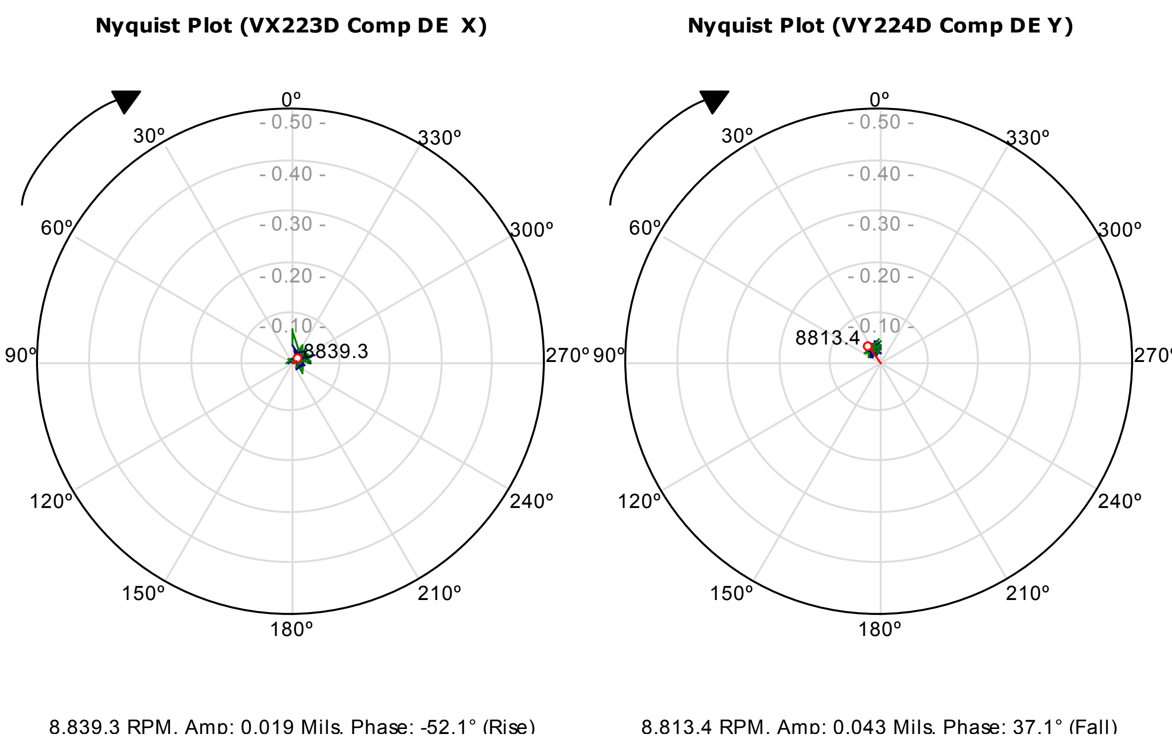


Figure 8: 11/17/2012, 900rpm, 100psi, Compressor DE X and Y

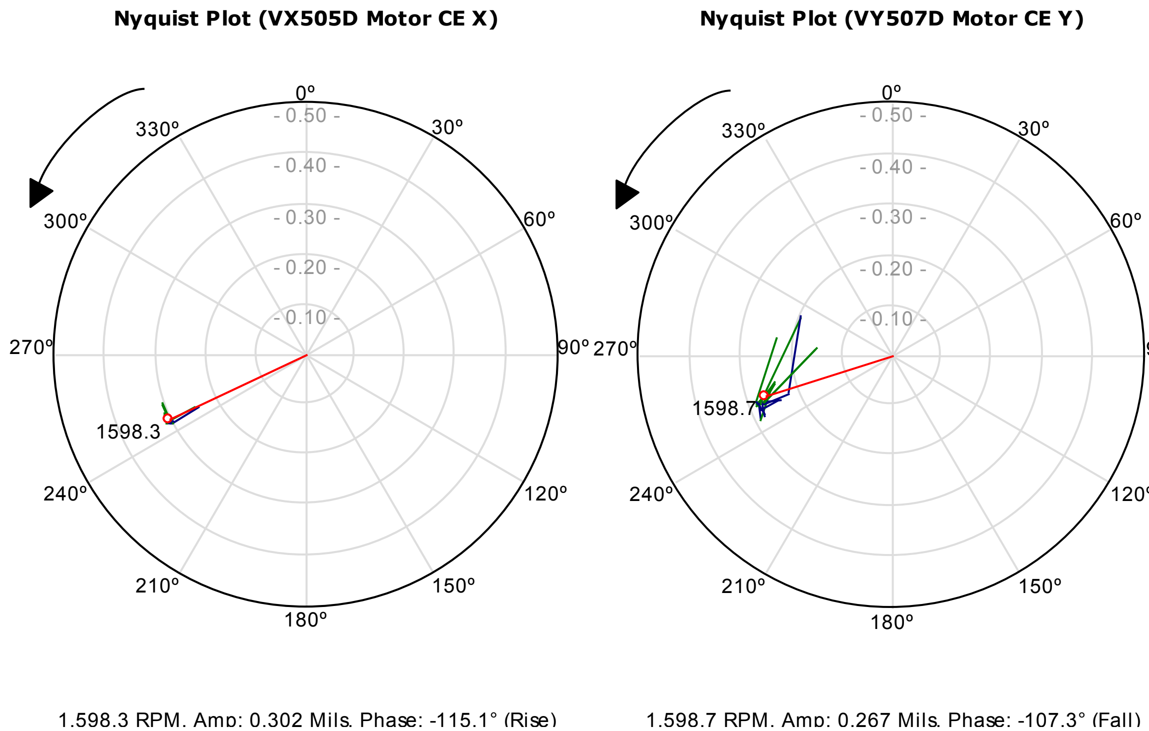


Figure 9: 11/16/2012, 1600rpm, 100psi, Motor CE X and Y

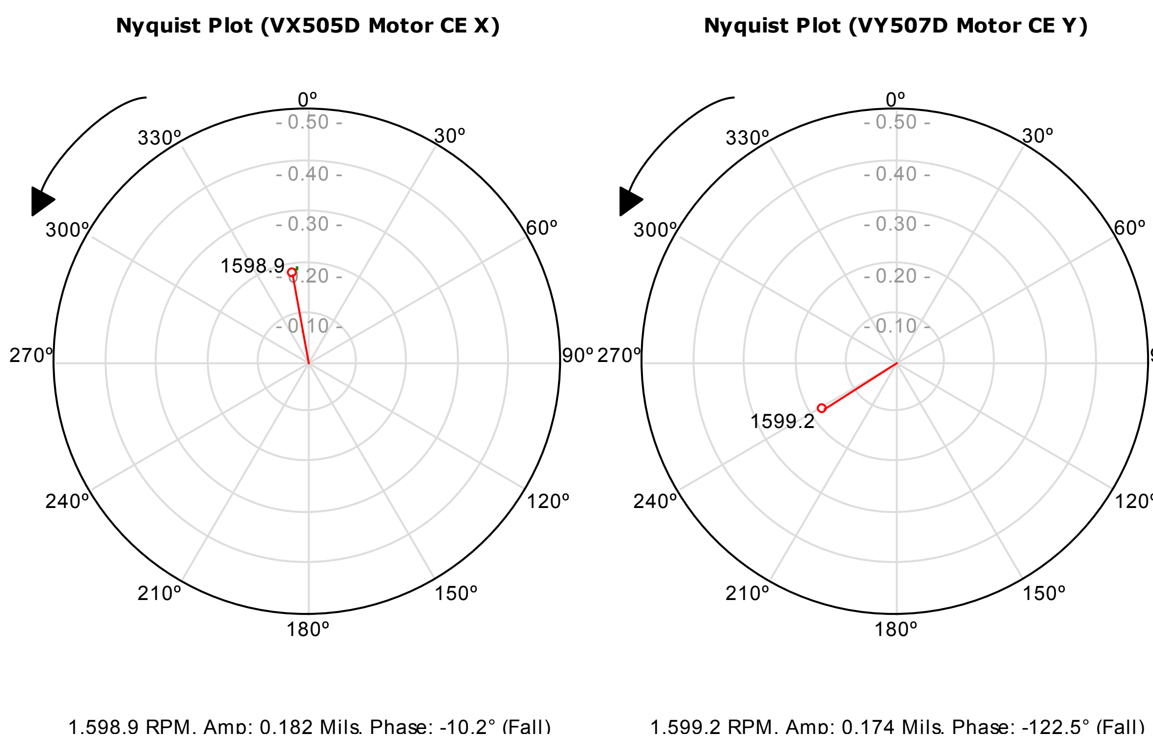


Figure 10: 11/17/2012, 1600rpm, 100psi, Motor CE X and Y

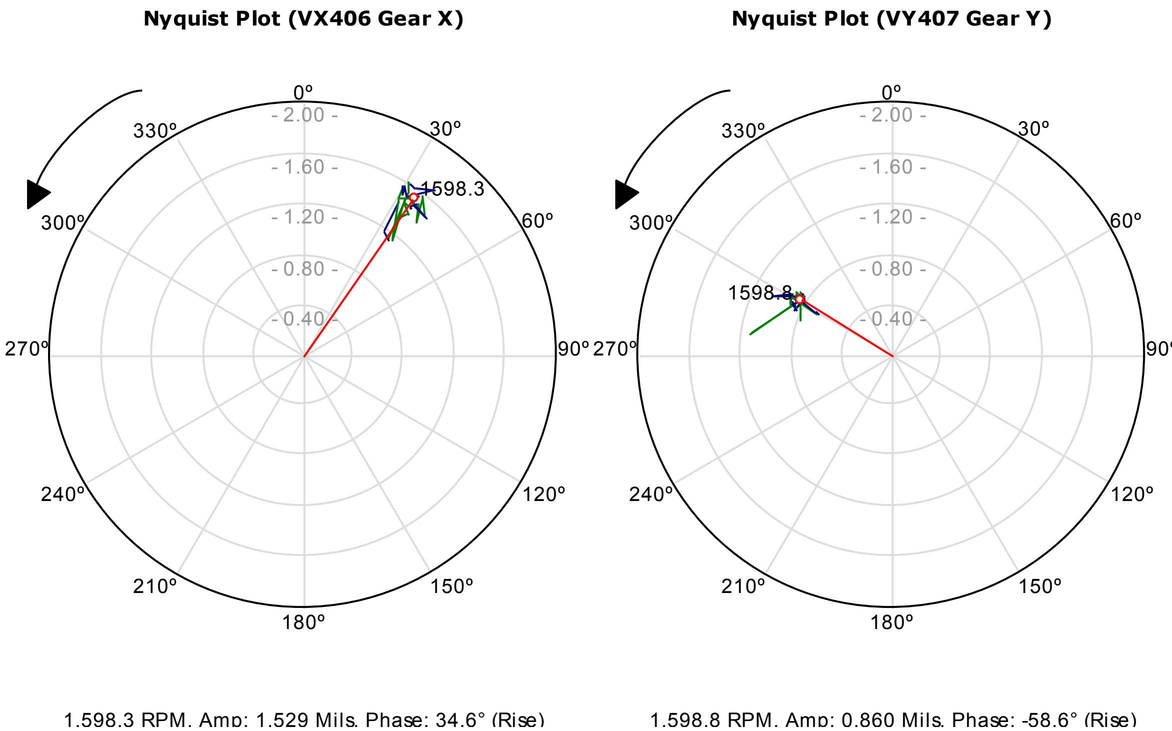


Figure 11: 11/16/2012, 1600rpm, 100psi, Gearbox input X and Y

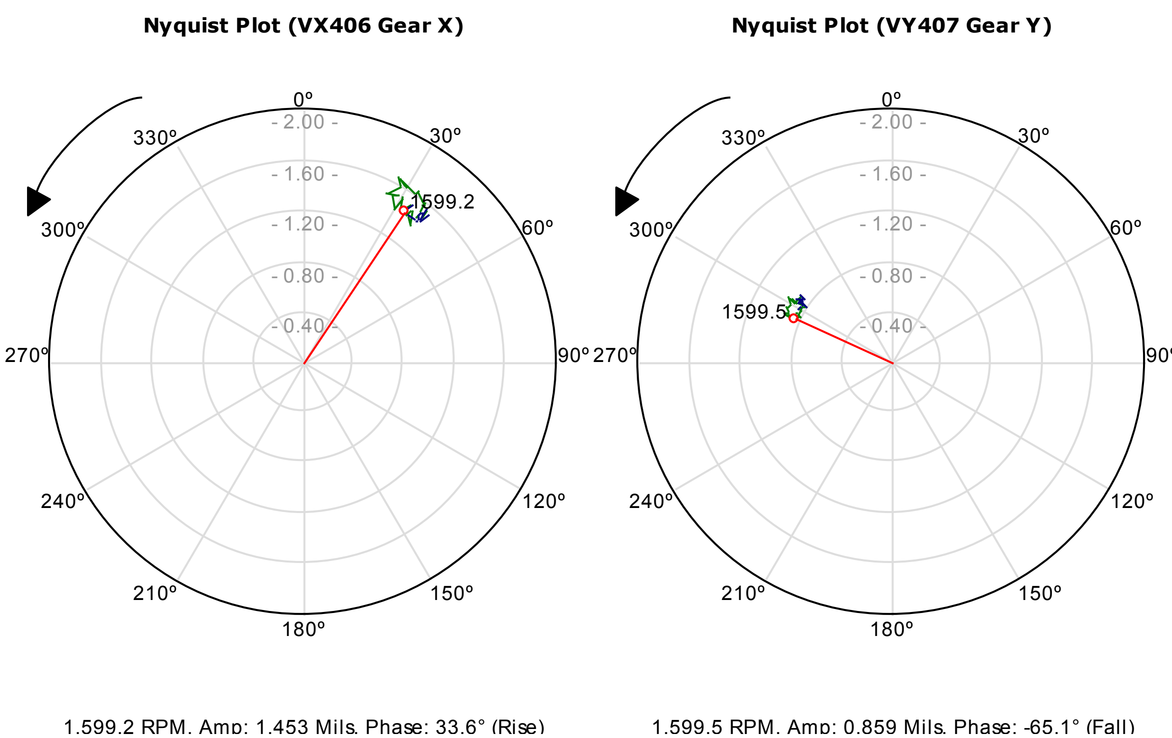


Figure 12: 11/17/2012, 1600rpm, 100psi, Gearbox input X and Y

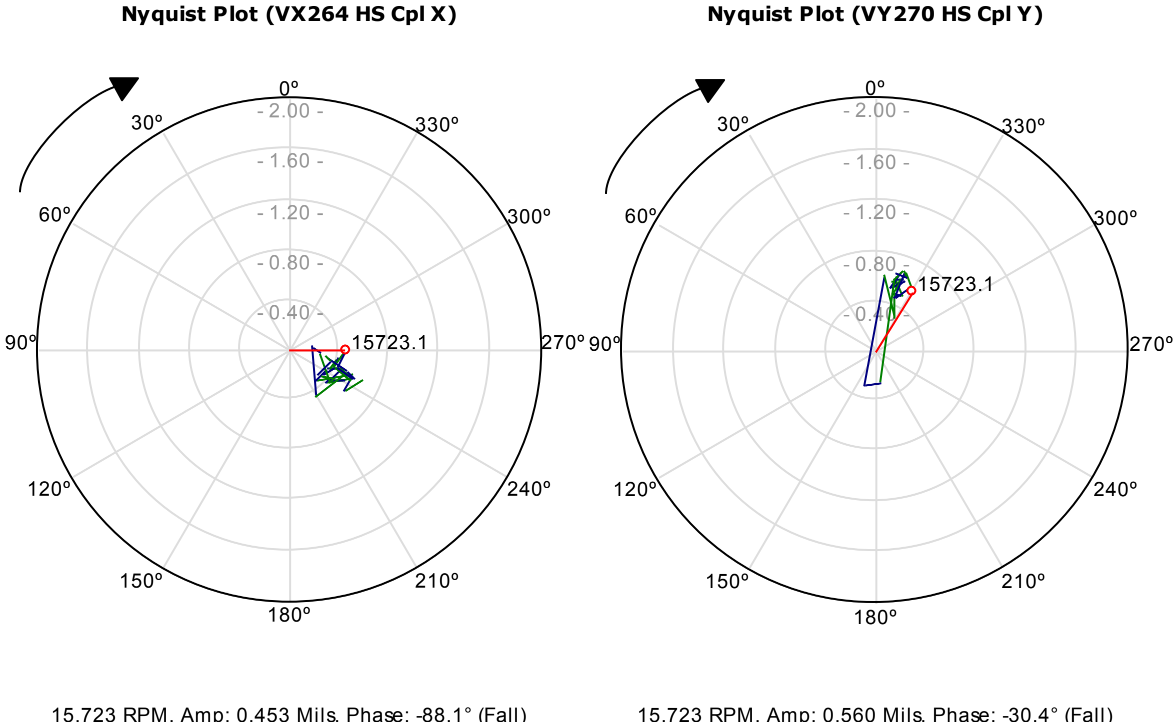


Figure 13: 11/16/2012, 1600rpm, 100psi, HS coupling X and Y

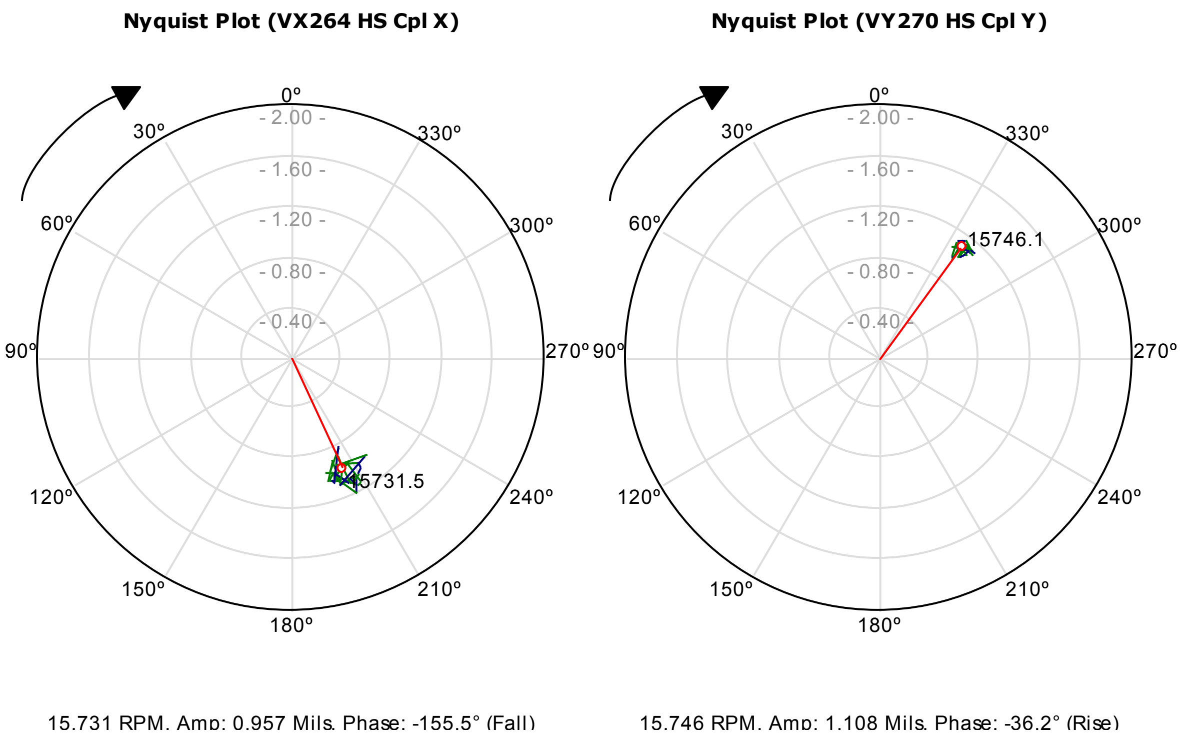


Figure 14: 11/17/2012, 1600rpm, 100psi, HS coupling X and Y

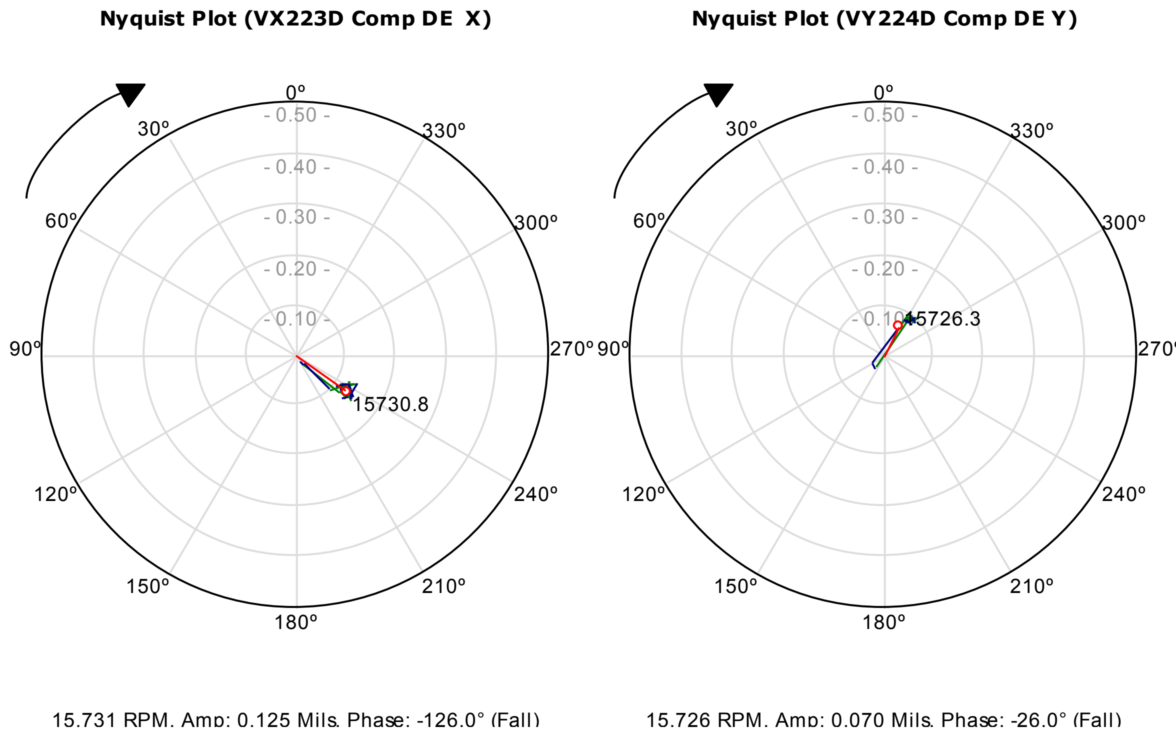


Figure 15: 11/16/2012, 1600rpm, 100psi, Compressor DE X and Y

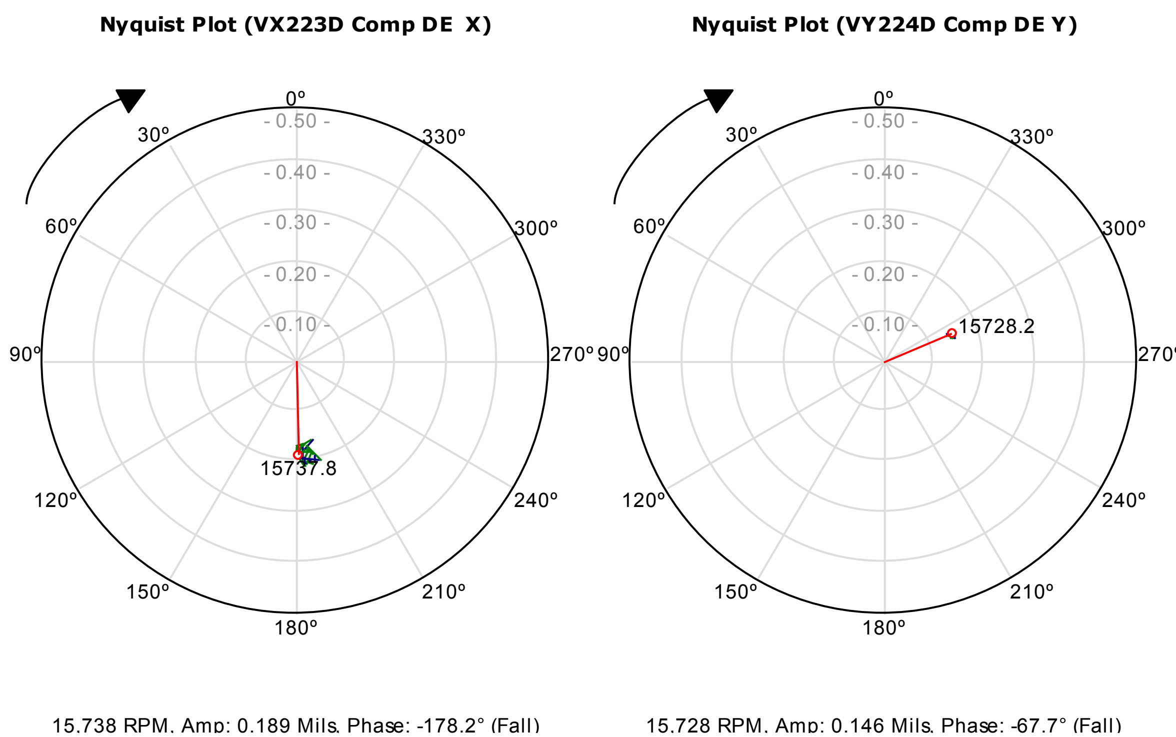


Figure 16: 11/17/2012, 1600rpm, 100psi, Compressor DE X and Y

At lower speed ranges there is fairly good phase and amplitude repeatability between the two days at the gearbox input shaft. As expected, there are some notable variations in the phase and amplitude at the high speed coupling (HS) and compressor driven end (DE) for these speeds. Again, this is expected due to the presence of sub-synchronous vibrations at the lower speeds.

When comparing data phase and amplitude data on 11/16 to data acquired on 11/17, the data is very repeatable at 2800 and 3200rpm motor speed. The phase angle variation is generally within 10deg save a couple outliers which are closer to 15 deg, and the amplitude variation is typically within 10 or 20%. The only notable exception to this observation is the Motor CE x-probe, which appears to undergo a ~180deg phase shift between 11/16 and 11/17. This does not seem to have any effect on the gearbox input phase and amplitude.

Figures 17-32 contain 11/16 and 11/17 comparisons at several locations on the drive train for 2800 and 3200rpm motor speed (27,552 and 31,488rpm compressor speed).

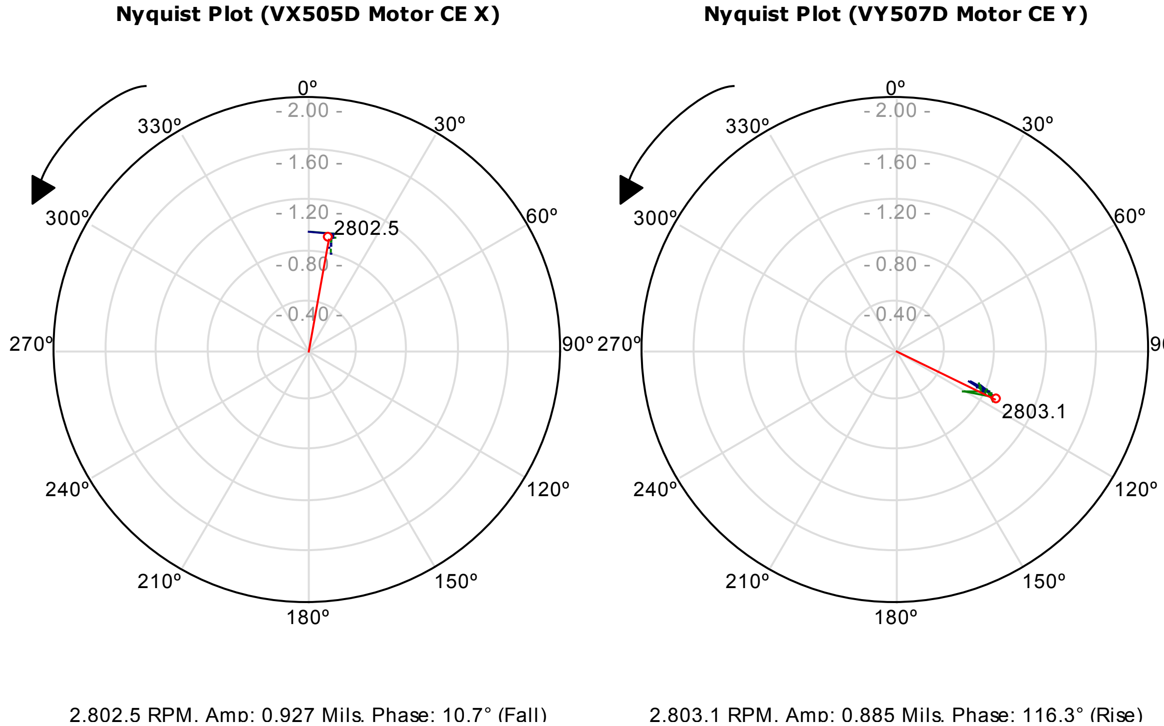


Figure 17: 11/16/2012, 2800rpm, 100psi, Motor CE X and Y

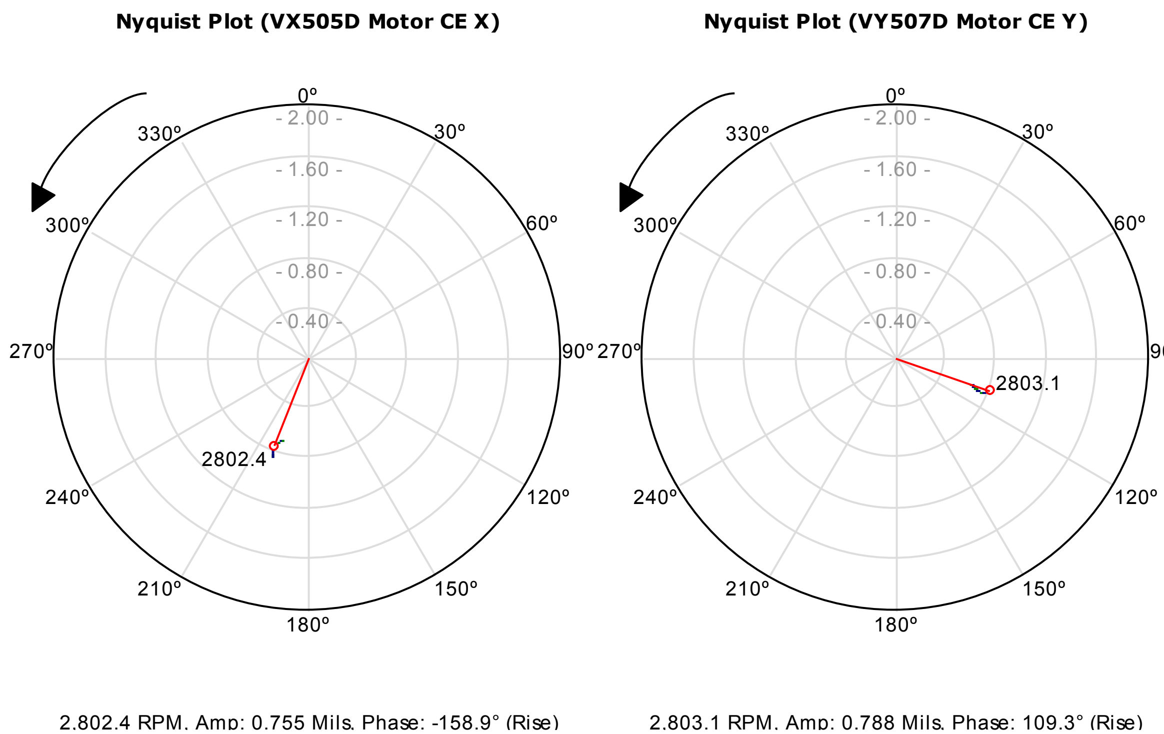


Figure 18: 11/17/2012, 2800rpm, 100psi, Motor CE X and Y

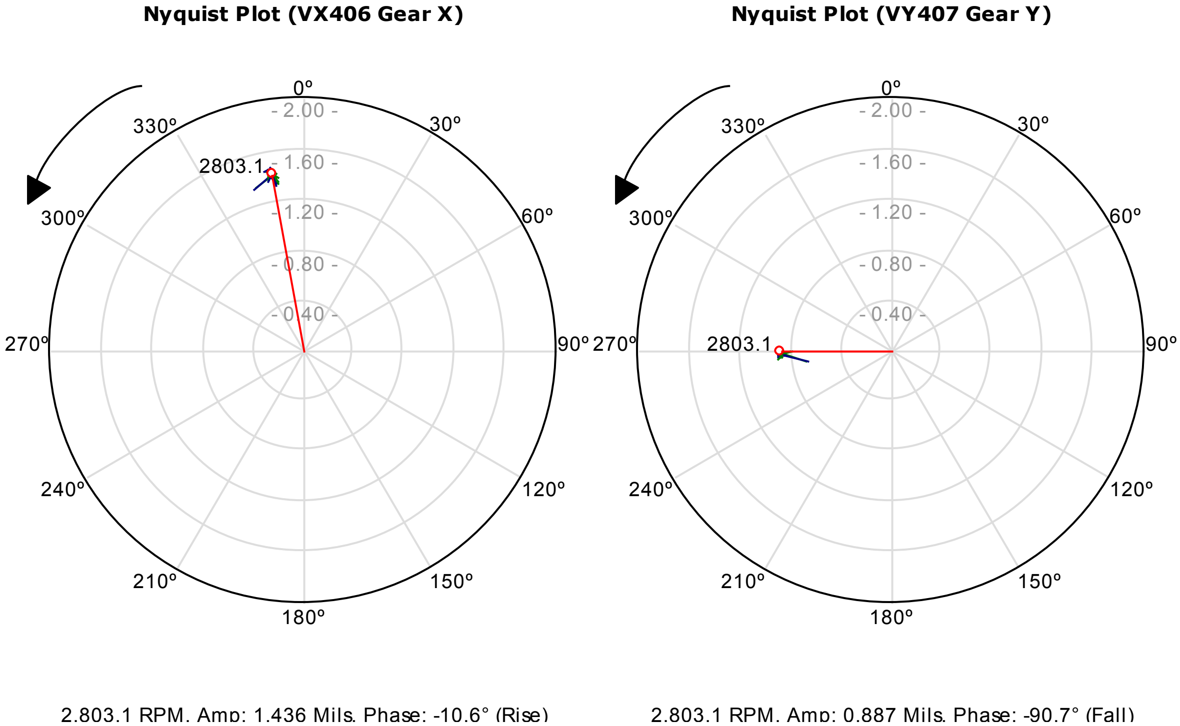


Figure 19: 11/16/2012, 2800rpm, 100psi, Gearbox input X and Y

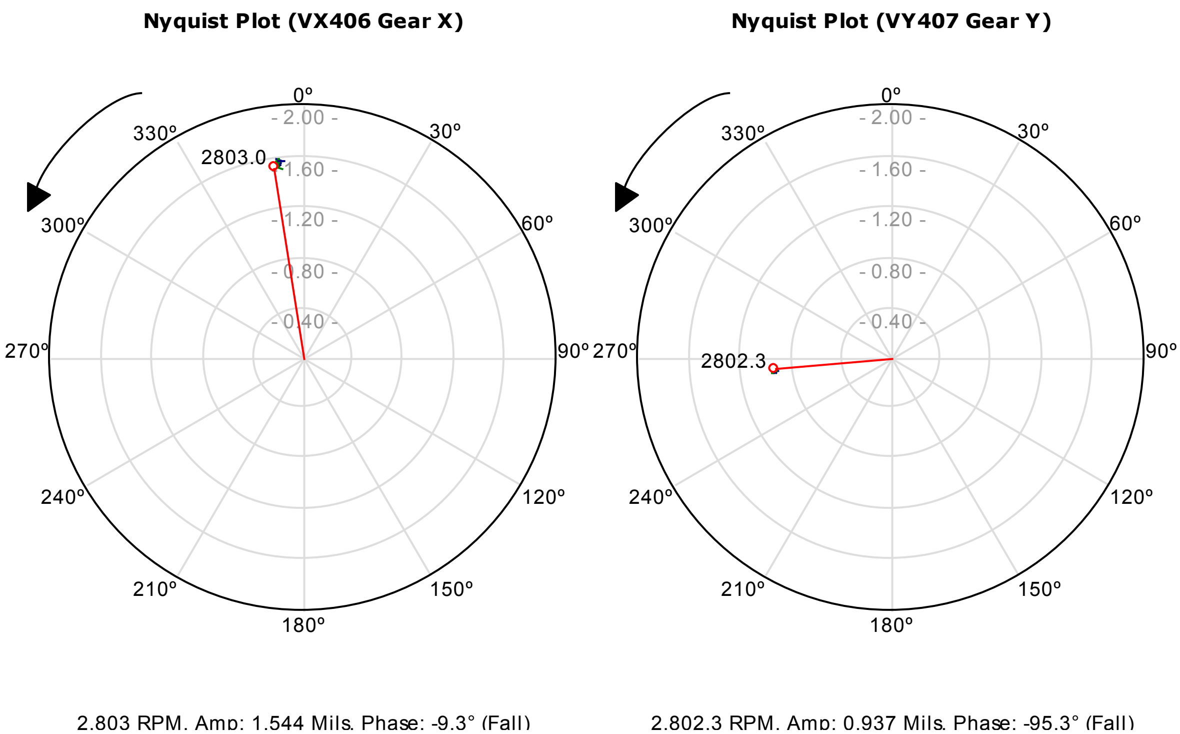


Figure 20: 11/17/2012, 2800rpm, 100psi, Gearbox input X and Y

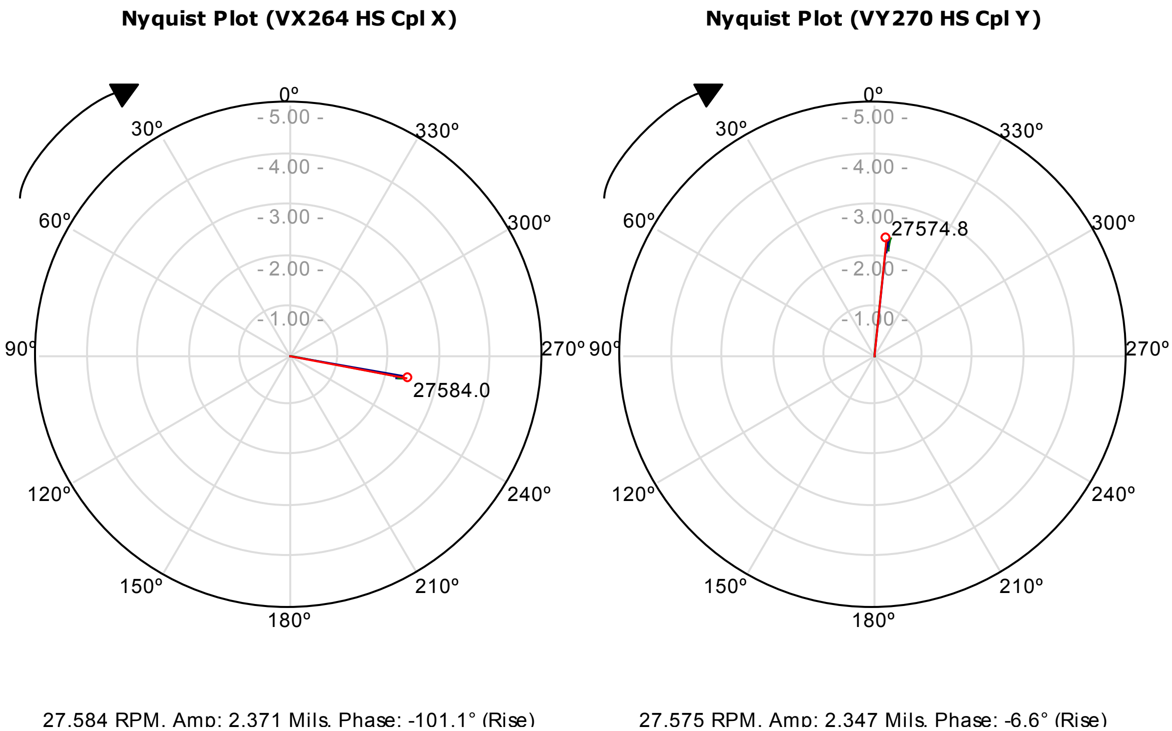


Figure 21: 11/16/2012, 2800rpm, 100psi, HS coupling X and Y

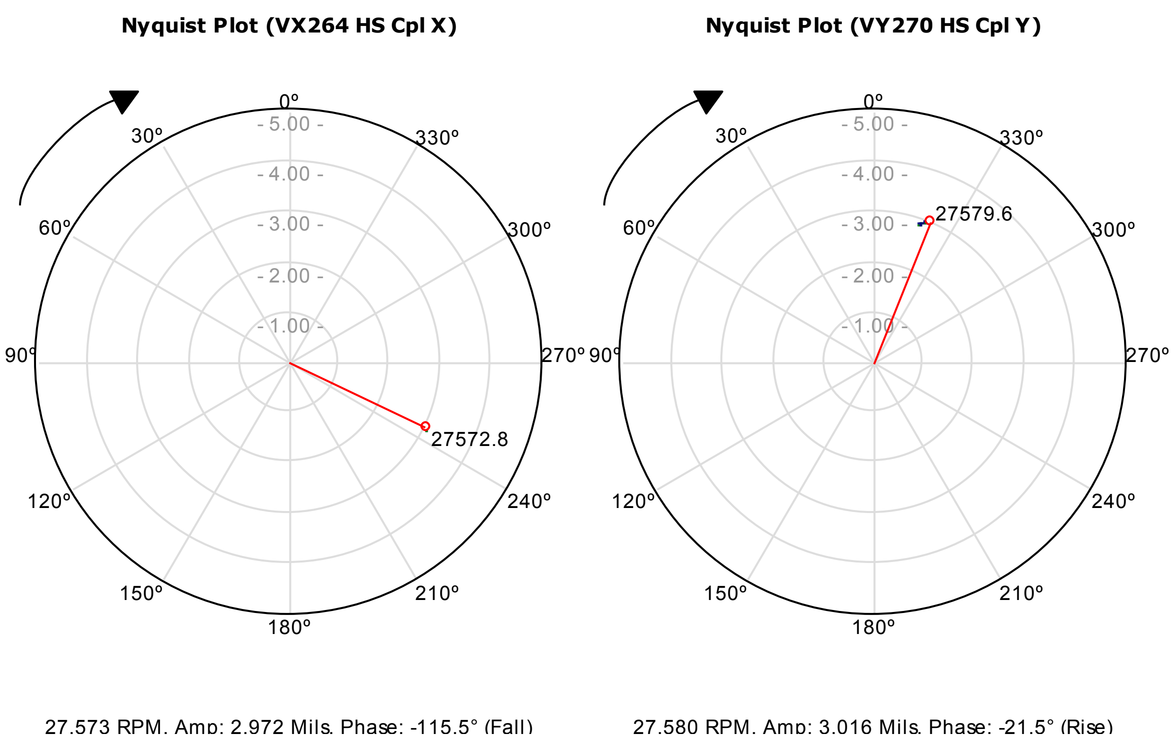


Figure 22: 11/17/2012, 2800rpm, 100psi, HS coupling X and Y

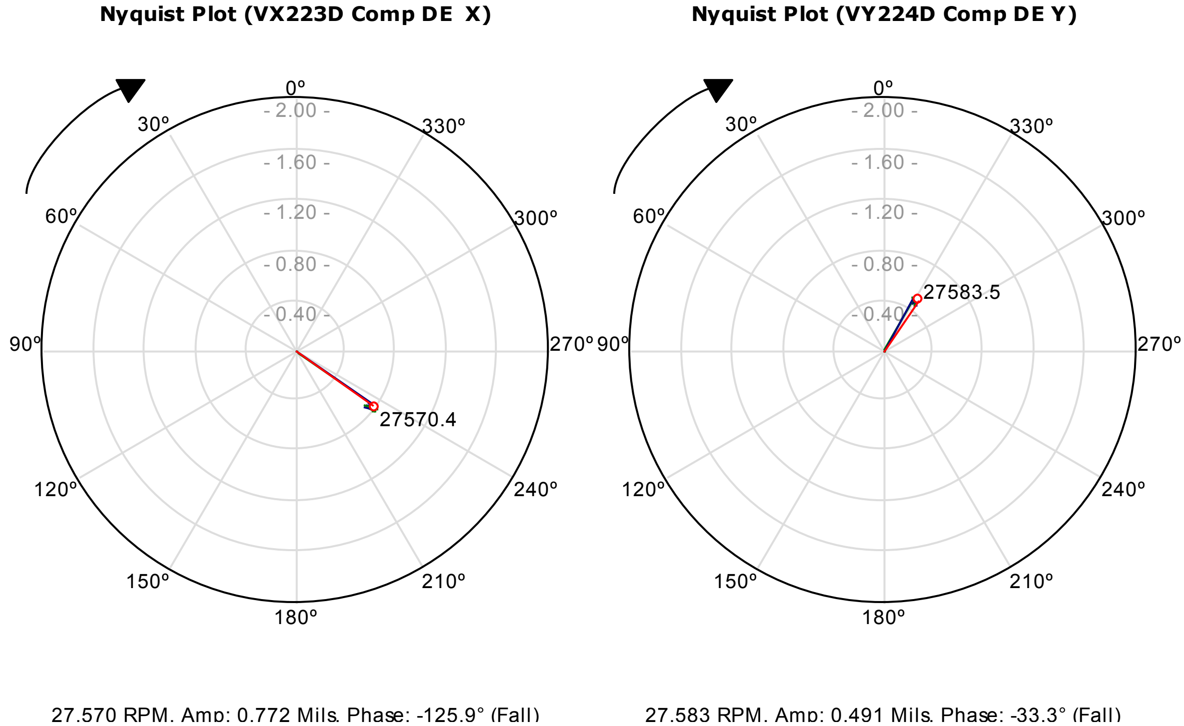


Figure 23: 11/16/2012, 2800rpm, 100psi, Compressor DE X and Y

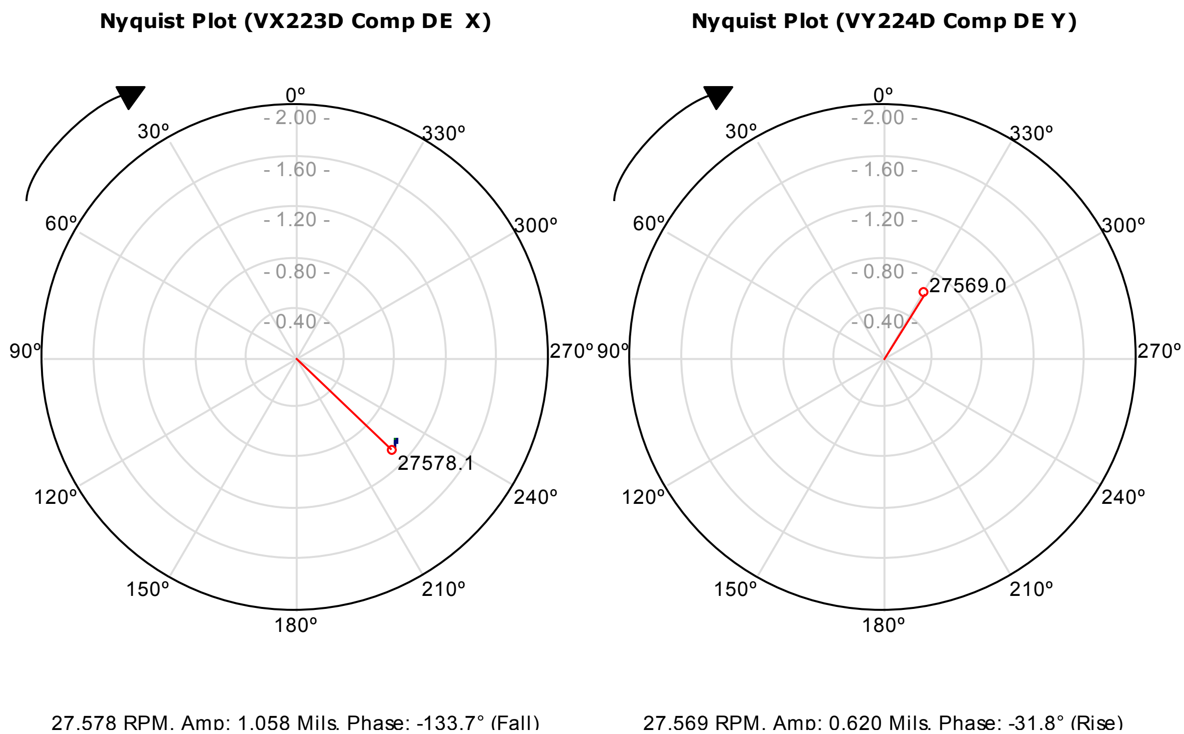


Figure 24: 11/17/2012, 2800rpm, 100psi, Compressor DE X and Y

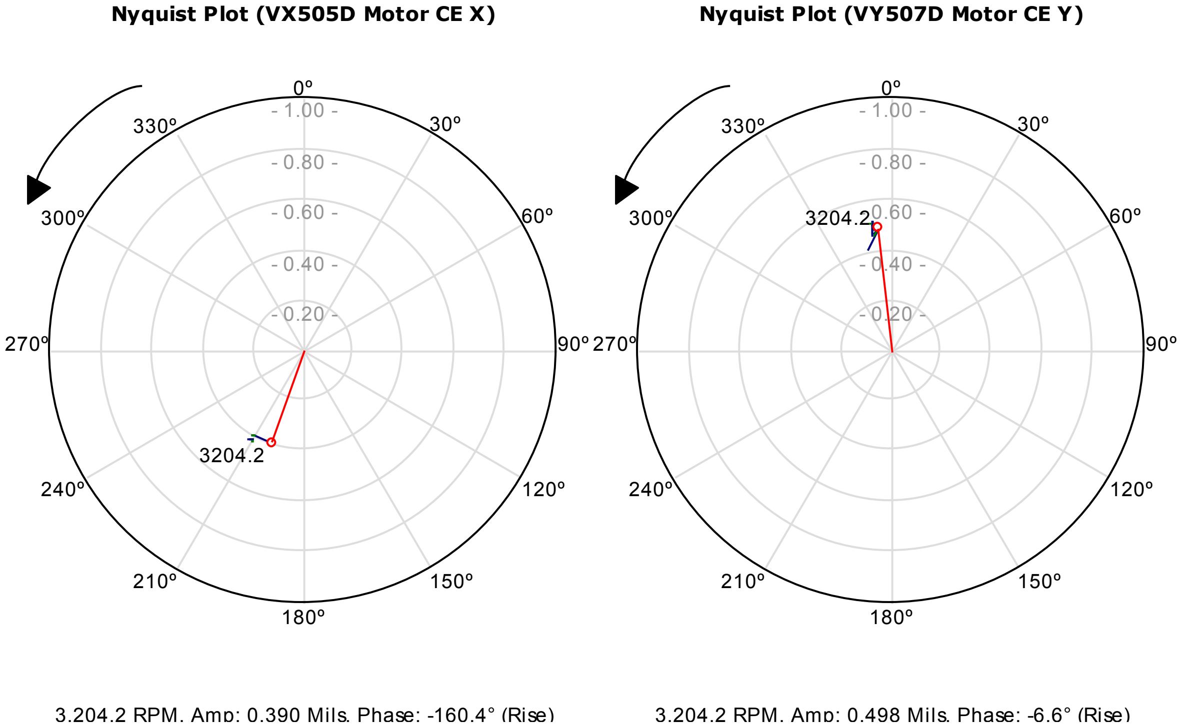


Figure 25: 11/16/2012, 3200rpm, 100psi, Motor CE X and Y

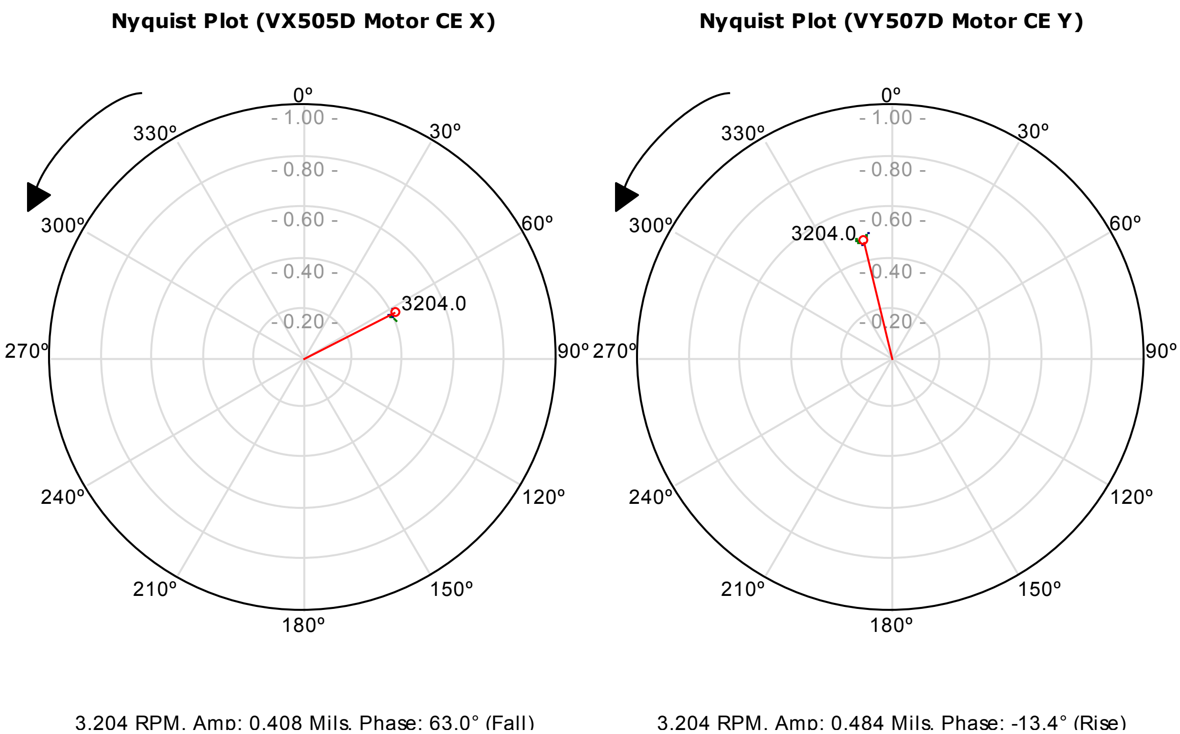


Figure 26: 11/17/2012, 3200rpm, 100psi, Motor CE X and Y

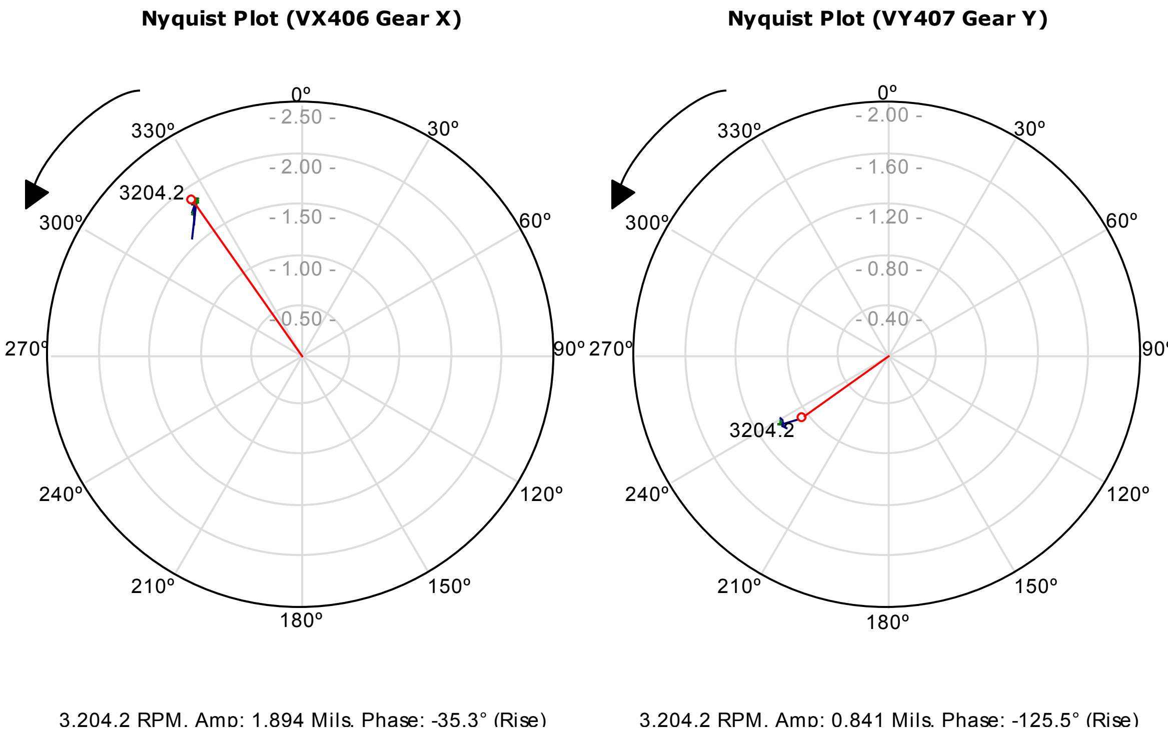


Figure 27: 11/16/2012, 3200rpm, 100psi, Gearbox input X and Y

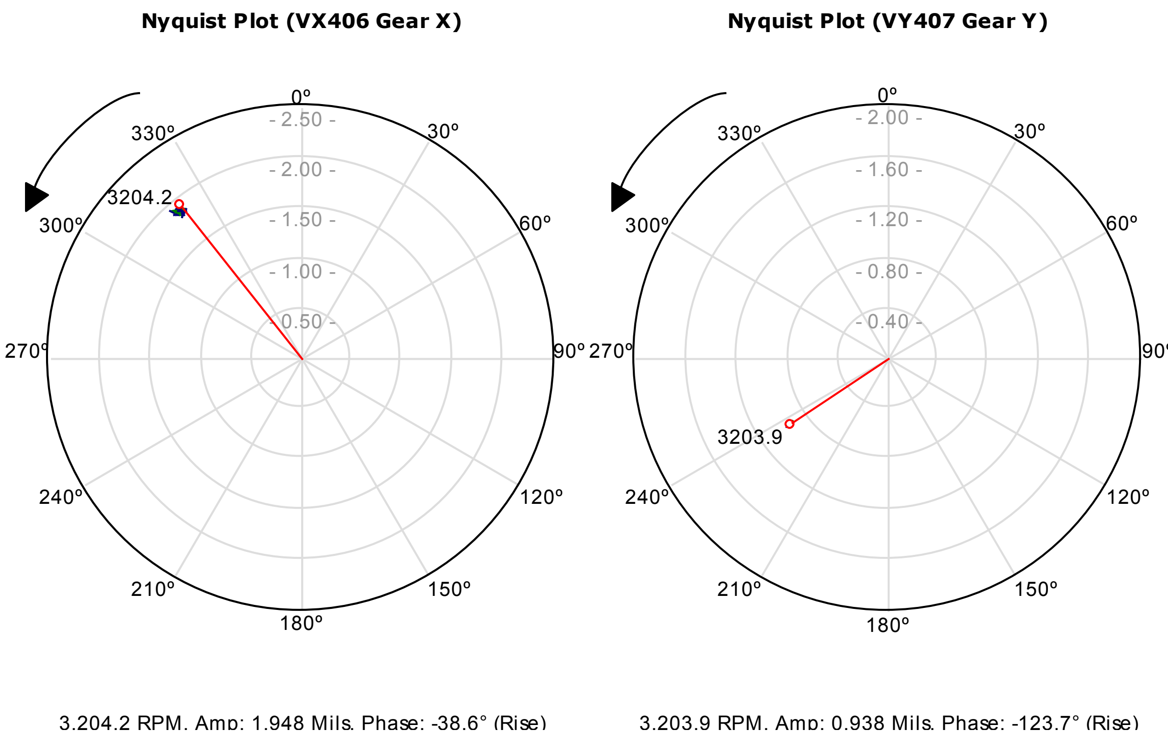


Figure 28: 11/17/2012, 3200rpm, 100psi, Gearbox input X and Y

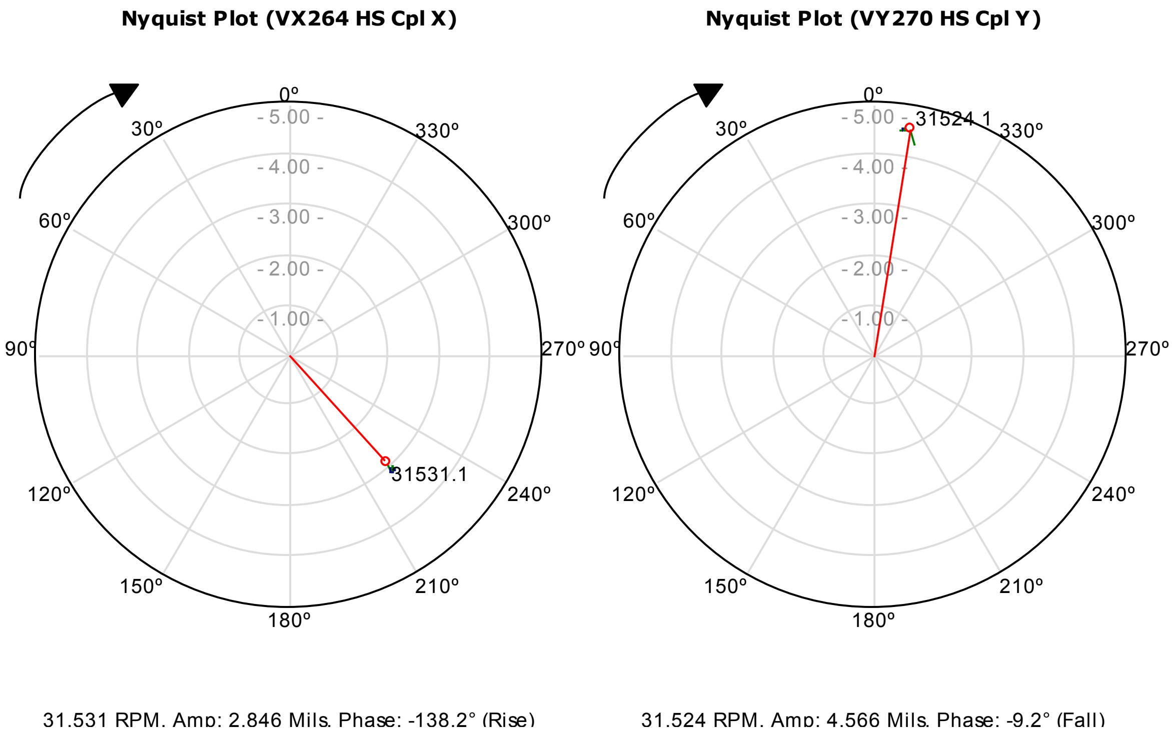


Figure 29: 11/16/2012, 3200rpm, 100psi, HS coupling X and Y

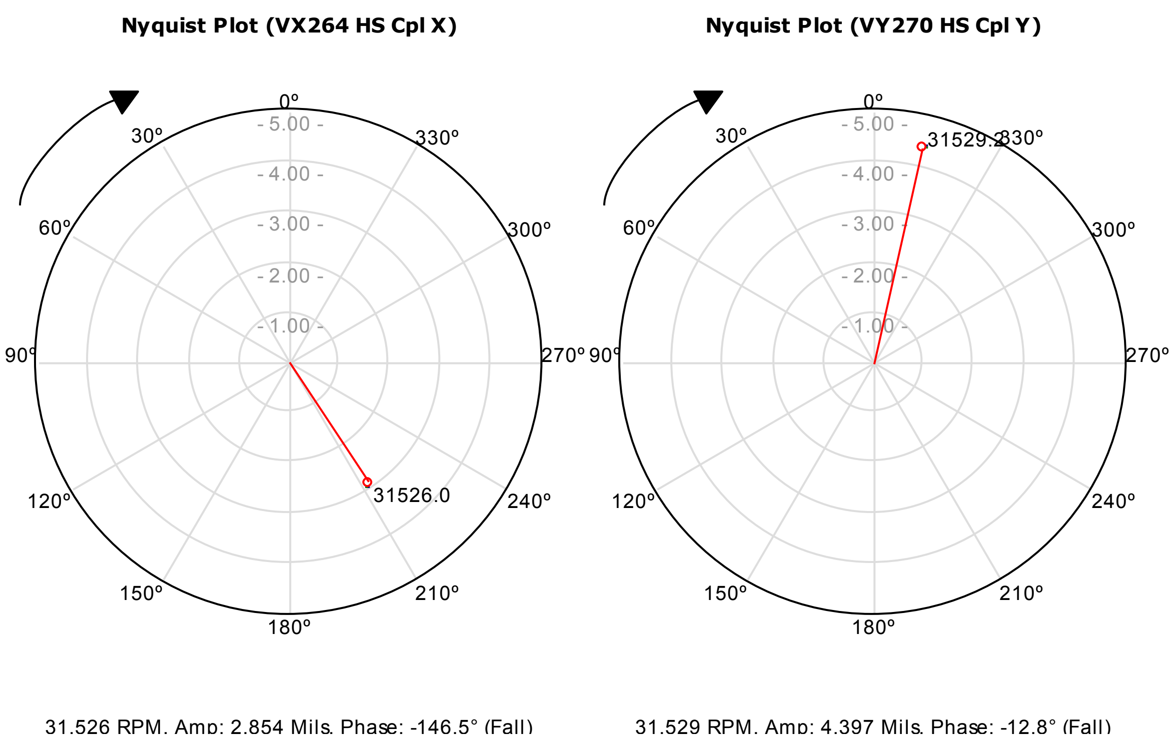


Figure 30: 11/17/2012, 3200rpm, 100psi, HS coupling X and Y

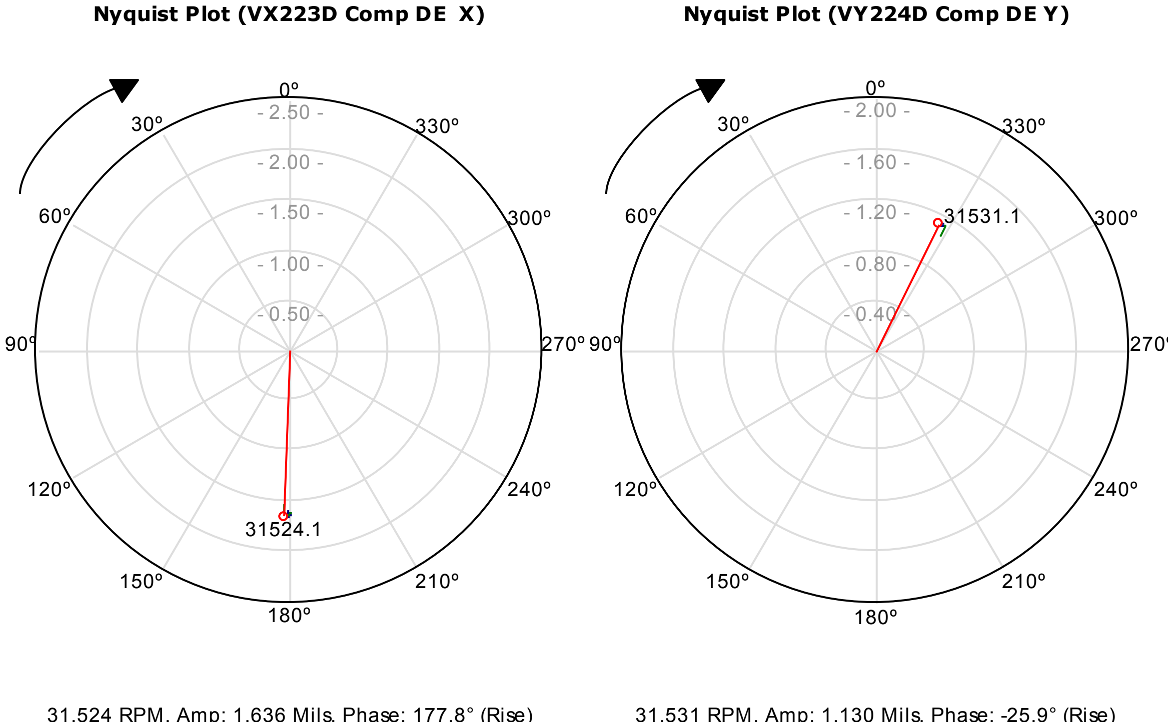


Figure 31: 11/16/2012, 3200rpm, 100psi, Compressor DE X and Y

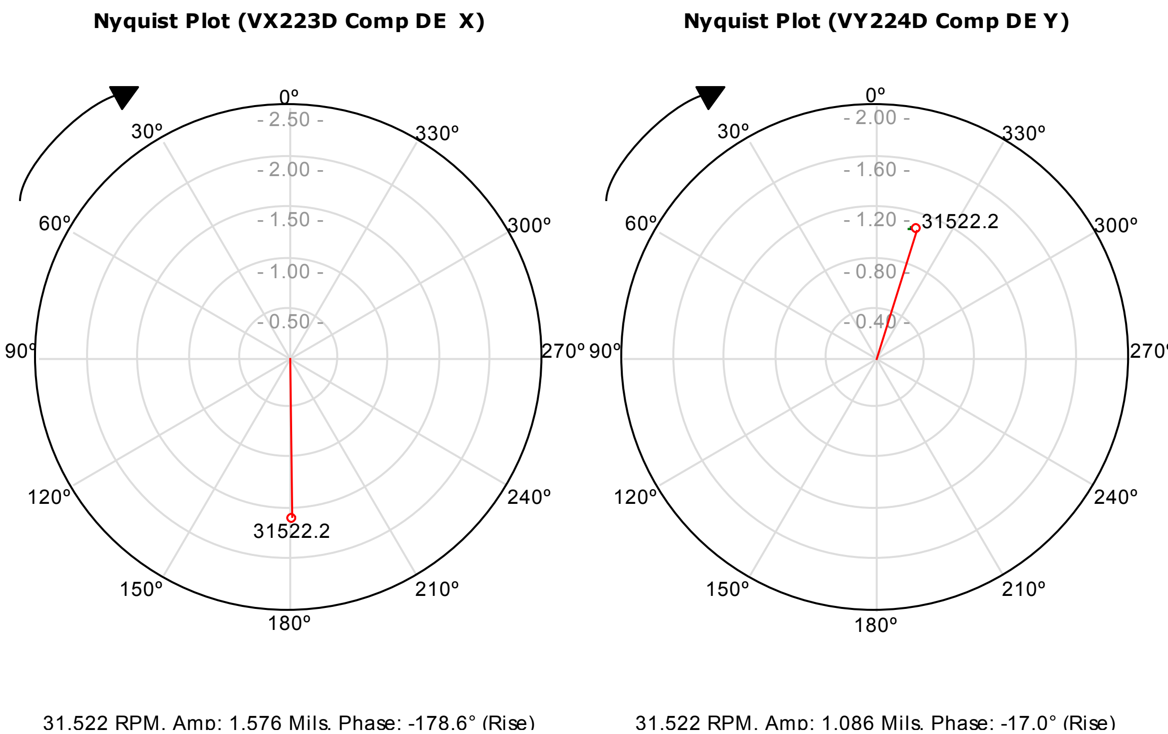


Figure 32: 11/17/2012, 3200rpm, 100psi, Compressor DE X and Y

11/28/2012 to 11/30/2012 Tabulated Data and Nyquist Plots

The following data is for runs at 200psia loop pressure on 4 separate days. Data is gathered in the same method outlined above for 11/16 and 11/17. No changes to the drive train were made between 11/28, 11/29, and 11/30, although wheel space cavity pressures were varied significantly between tests in an effort to manage thrust loads, among other things. The changes to bleed flows and cavity pressures are not accounted for in the data. On 12/4 the gearbox was removed and the HS coupling balance was altered in an effort to reduce compressor DE and HS coupling vibration amplitudes. Tabulated below is a condensed set of data at 2800rpm which contains vibration phase angle, amplitude, shaft centerline data, and shaft Z-prox DC offset.

				Motor NCE X			Motor NCE Y		
	speed	mdot cor	power, kW	DC	phase	amp	DC	phase	amp
11/28 accel	2800		4207.5	55.7	178	1.22	56.6	100	0.76
11/29 AM accel	2800	56.3	4272.2	55.9	-178	0.98	56.7	100	0.6
11/30 accel 11-18-52	2800	56.6	4283.5	56.2	-171	0.76	57	105	0.42
12/4/12**	2800	56.4	4272	56.2	-178.4	1.19	57	100.8	0.716

Table 1: Motor NCE X and Y Phase, Amplitude, and Shaft DC Offset Comparisons

				Motor CE X			Motor CE Y		
	speed	mdot cor	power, kW	DC	phase	amp	DC	phase	amp
11/28 accel	2800		4207.5	58.3	151	0.52	62.9	59	0.49
11/29 AM accel	2800	56.3	4272.2	58.5	143	0.41	63.5	49	0.38
11/30 accel 11-18-52	2800	56.6	4283.5	58	26	0.28	63.4	53	0.24
12/4/12**	2800	56.4	4272	59.8	151.9	0.58	65.2	60	0.47

Table 2: Motor CE X and Y Phase, Amplitude, and Shaft DC Offset Comparisons

				Gearbox LS bearing_X			Gearbox LS bearing_Y		
	speed	mdot cor	power, kW	DC	phase	amp	DC	phase	amp
11/28 accel	2800		4207.5	46.8	-21	1.83	49.6	-103	1.28
11/29 AM accel	2800	56.3	4272.2	47.1	-20	1.28	50.4	-98	0.92
11/30 accel 11-18-52	2800	56.6	4283.5	46.3	-27	1.19	49.3	-108	0.86
12/4/12**	2800	56.4	4272	47.1	-24.5	1.32	49.8	-103	0.97

Table 3: Gearbox Input X and Y Phase, Amplitude, and Shaft DC Offset Comparisons

				HS coupling shaft_X			HS coupling shaft_Y		
	speed	mdot cor	power, kW	DC	phase	amp	DC	phase	amp
11/28 accel	2800		4207.5	60.4	-160	1.36	55.2	-60	1.44
11/29 AM accel	2800	56.3	4272.2	60.9	-151	1.28	54	-48	1.35
11/30 accel 11-18-52	2800	56.6	4283.5	60.8	-160	1.3	54.1	-51	1.34
12/4/12**	2800	56.4	4272	59.7	-154	1.13	58.8	-47	1.209

Table 4: HS Coupling X and Y Phase, Amplitude, and Shaft DC Offset Comparisons

				Comp DE X			Comp DE Y		
	speed	mdot cor	power, kW	DC	phase	amp	DC	phase	amp
11/28 accel	2800		4207.5	49.8	162	0.78	45.3	-69	0.35
11/29 AM accel	2800	56.3	4272.2	49.9	171	0.8	45.2	-55	0.33
11/30 accel 11-18-52	2800	56.6	4283.5	49.8	173	0.75	45.2	-68	0.37
12/4/12**	2800	56.4	4272	50.1	176	0.599	45.5	-64	0.32

Table 5: Compressor DE X and Y Phase, Amplitude, and Shaft DC Offset Comparisons

				Comp NDE X			Comp NDE Y			Comp Z
	speed	mdot cor	power, kW	DC	phase	amp	DC	phase	amp	DC
11/28 accel	2800		4207.5	38.8	6	0.29	31.6	144	0.39	57.2
11/29 AM accel	2800	56.3	4272.2	38.8	14	0.29	31.5	159	0.38	57
11/30 accel 11-18-52	2800	56.6	4283.5	38.8	6	0.25	31.7	142	0.32	56.6
12/4/12**	2800	56.4	4272	38.9	9	0.27	31.9	137	0.33	55.2

Table 6: Compressor NDE X and Y Phase, Amplitude, and Shaft DC Offset Comparisons

The phase, amplitude, and DC offset values are repeatable between the first three days with the most variation occurring on 11/29. The only major changes in phase angle are observed on the motor CE and NCE, which is consistent with the data from 11/16 and 11/17. No unexpectedly large changes in phase or amplitude are observed on 12/4/12 after the HS coupling balance has been altered. There was a decrease in amplitude on the HS coupling and compressor DE prox probes, which was the intended consequence of altering the HS coupling balance.

Data has not been tabulated for higher than 2800rpm for these cases, although Nyquist plots exist at 3100rpm motor speed for 11/28 and 11/29. Figures 34-41 give Nyquist plot comparisons on 11/28 and 11/29 at 3100rpm motor speed (30,504rpm compressor speed). This speed is very close to design point speed 30,400rpm.

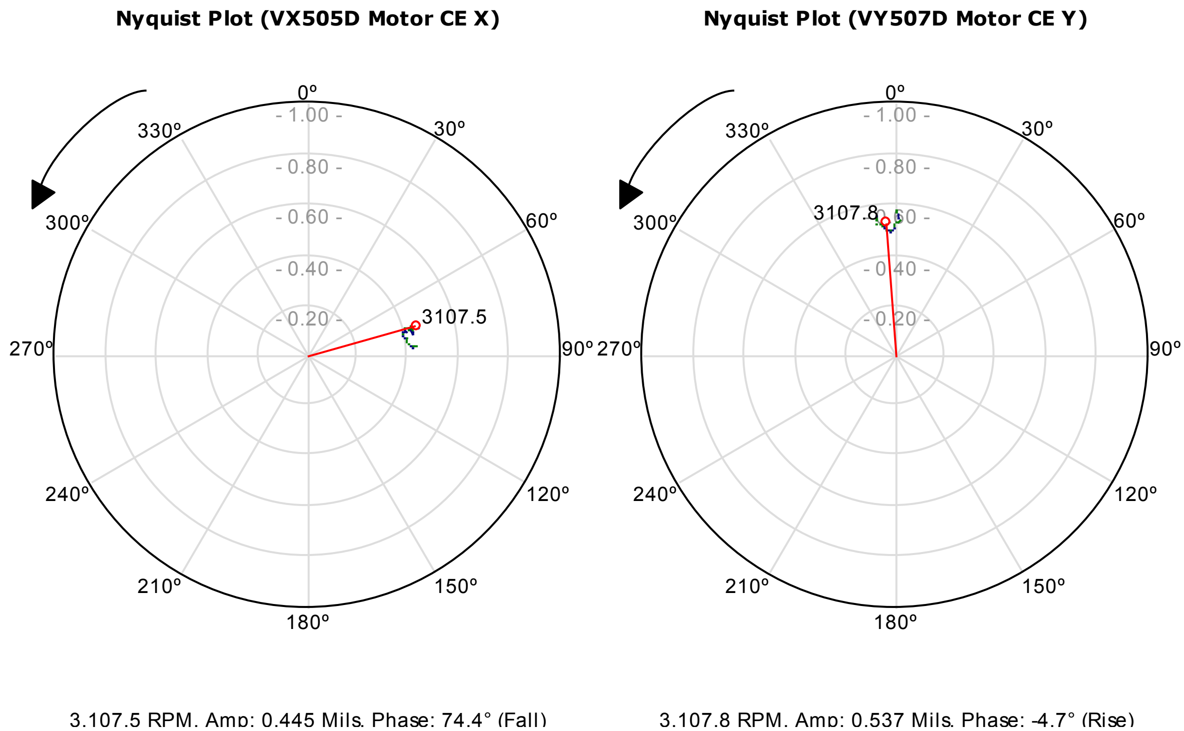


Figure 34: 11/28/2012, 3100rpm, 200psi, Motor CE X and Y

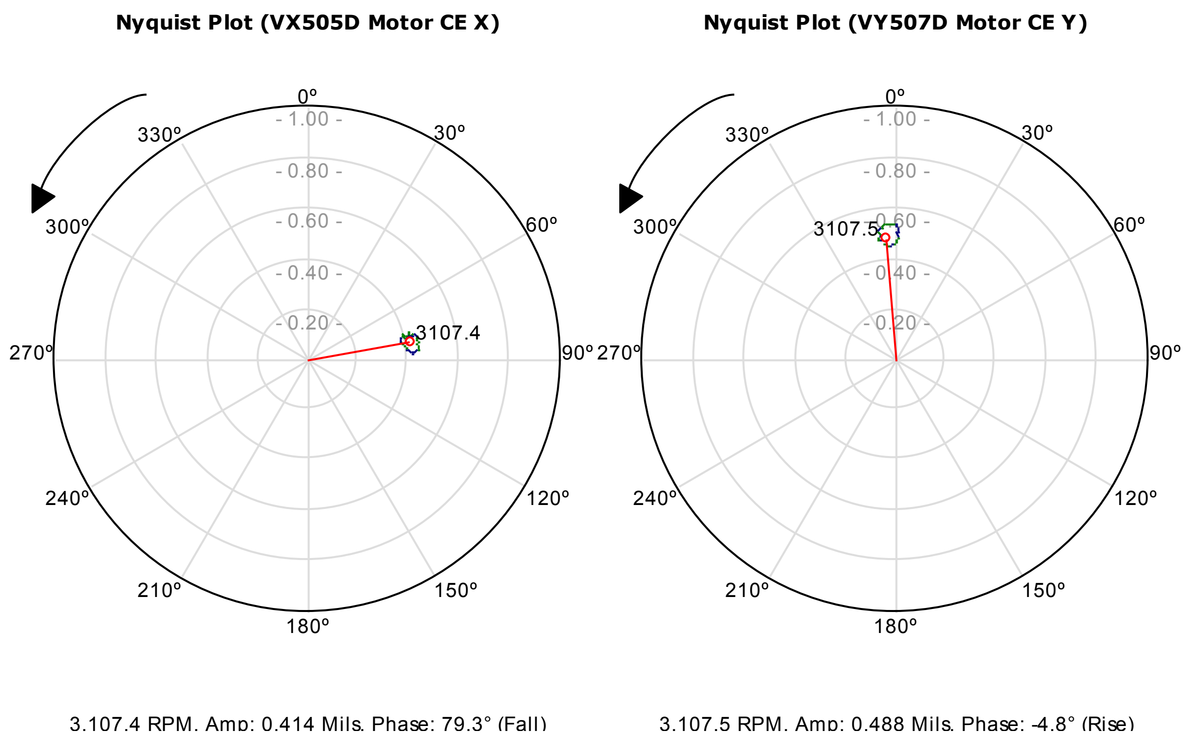


Figure 35: 11/29/2012, 3100rpm, 200psi, Motor CE X and Y

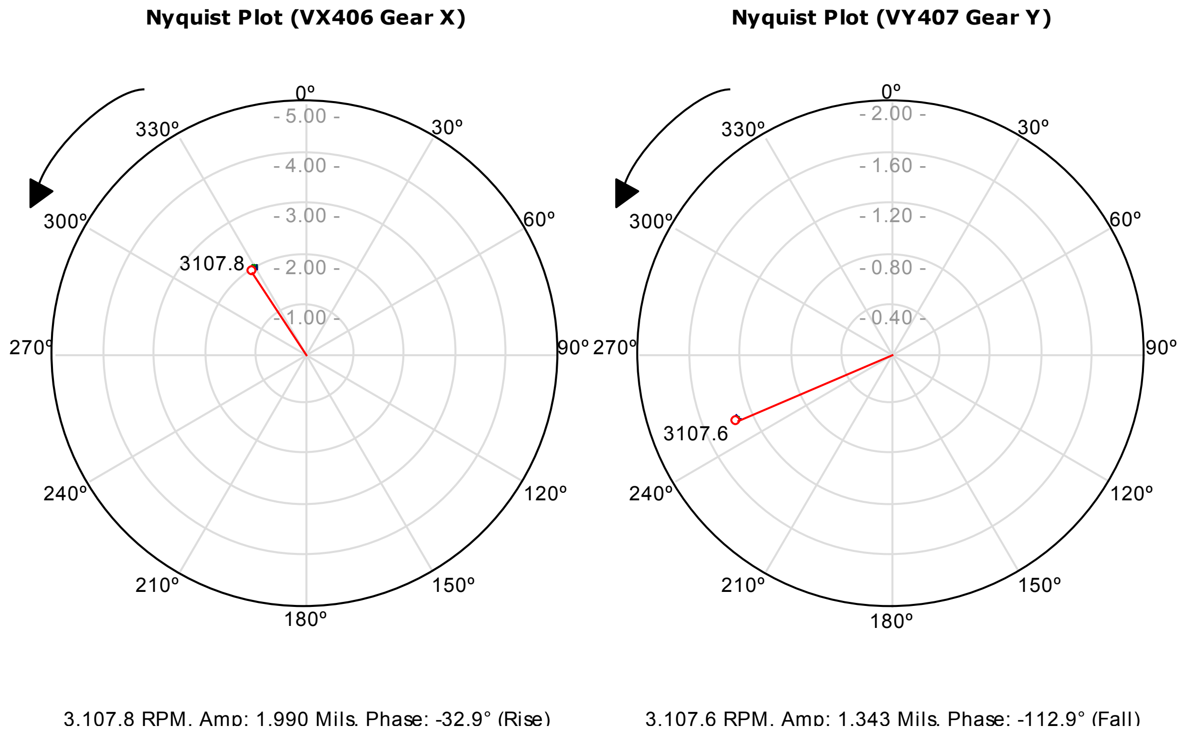


Figure 36: 11/28/2012, 3100rpm, 200psi, Gearbox input X and Y

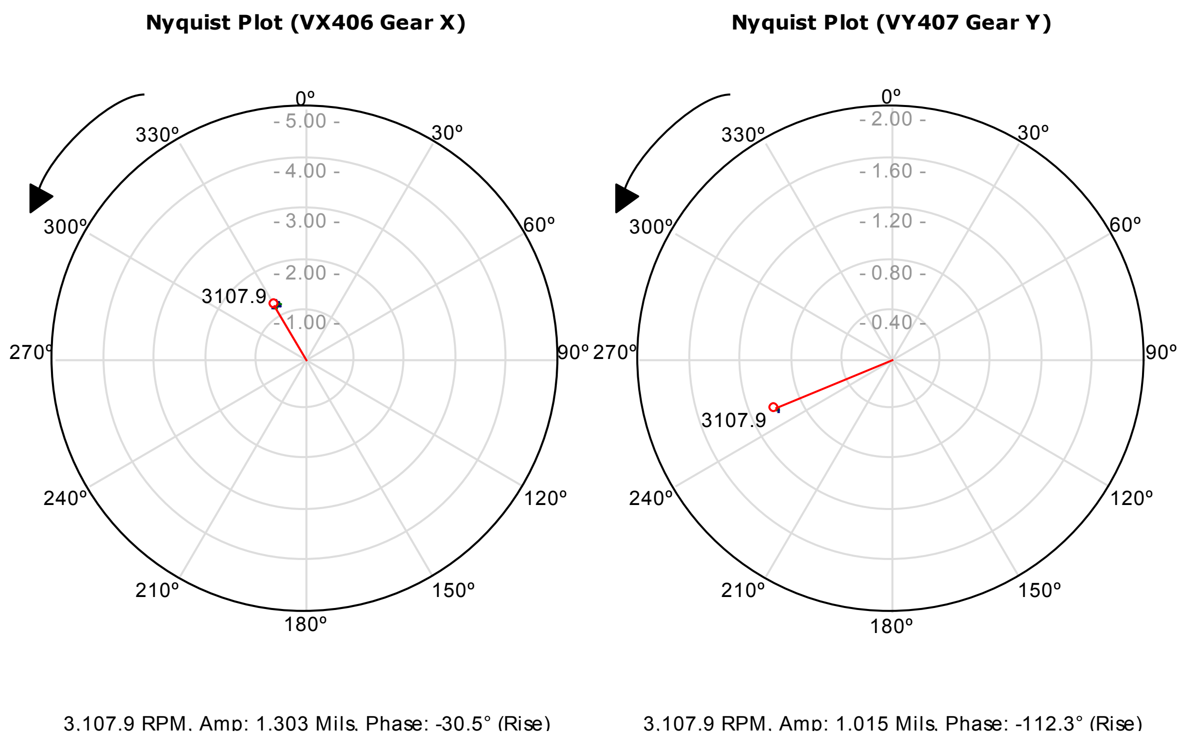


Figure 37: 11/29/2012, 3100rpm, 200psi, Gearbox input X and Y

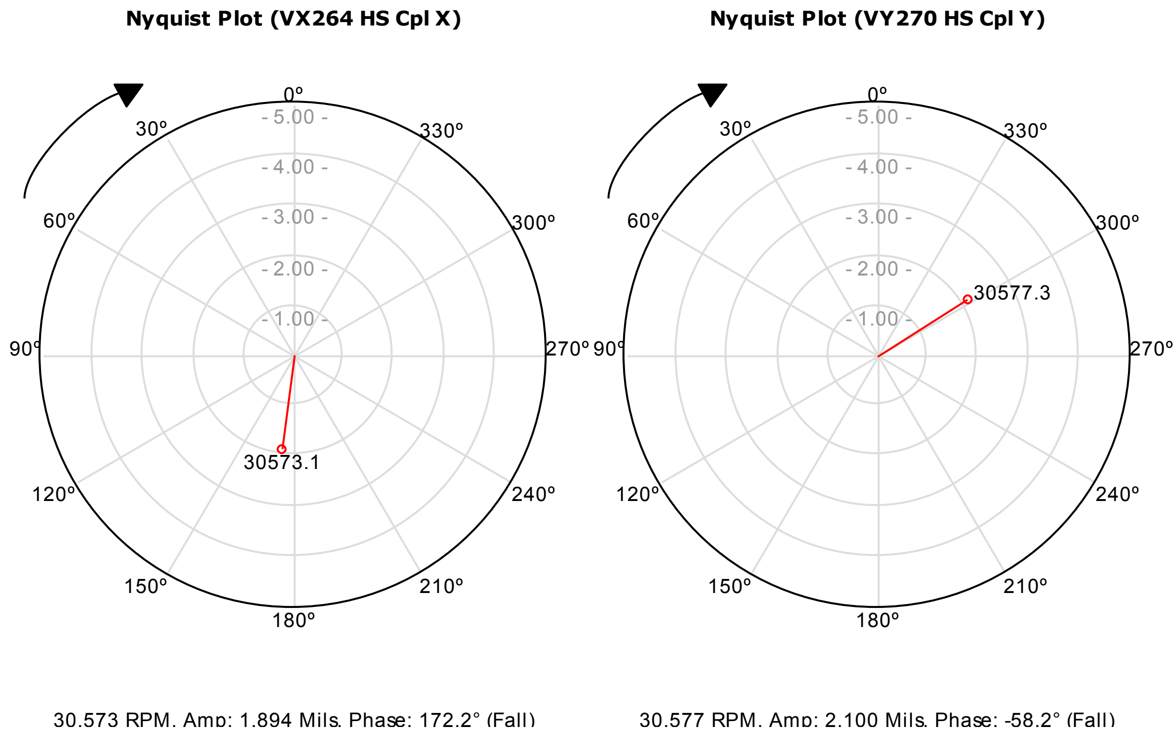


Figure 38: 11/28/2012, 3100rpm, 200psi, HS coupling X and Y

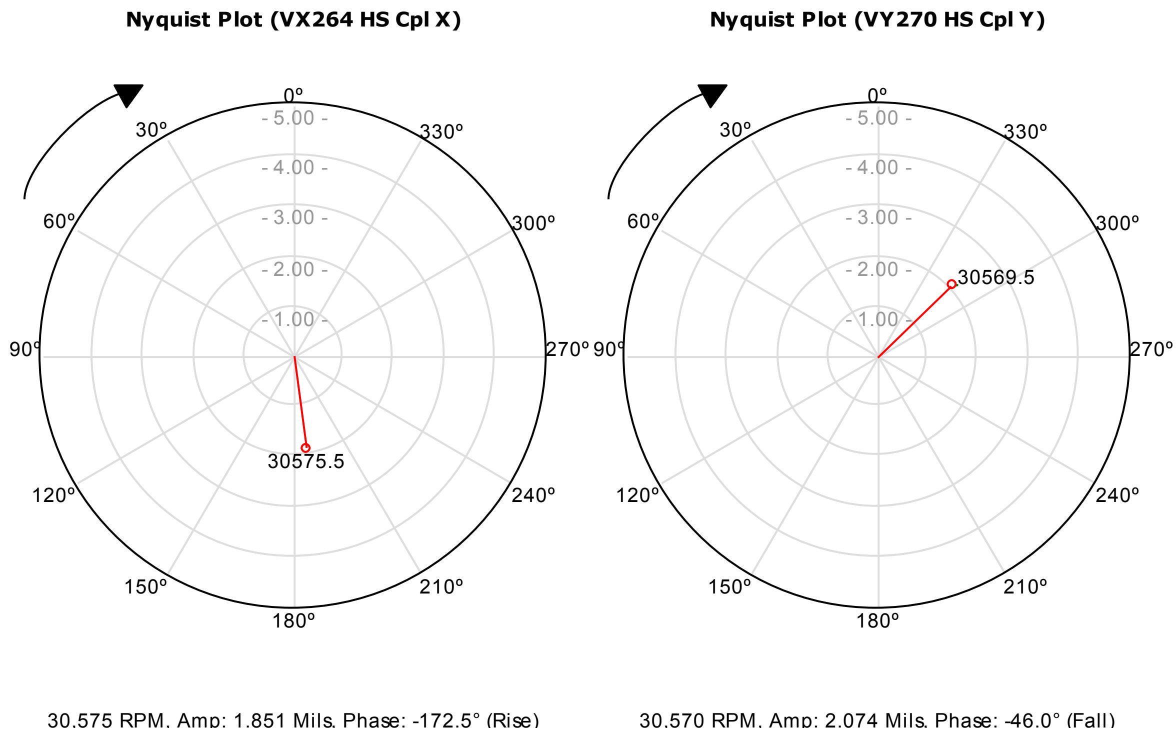


Figure 39: 11/29/2012, 3100rpm, 200psi, HS coupling X and Y

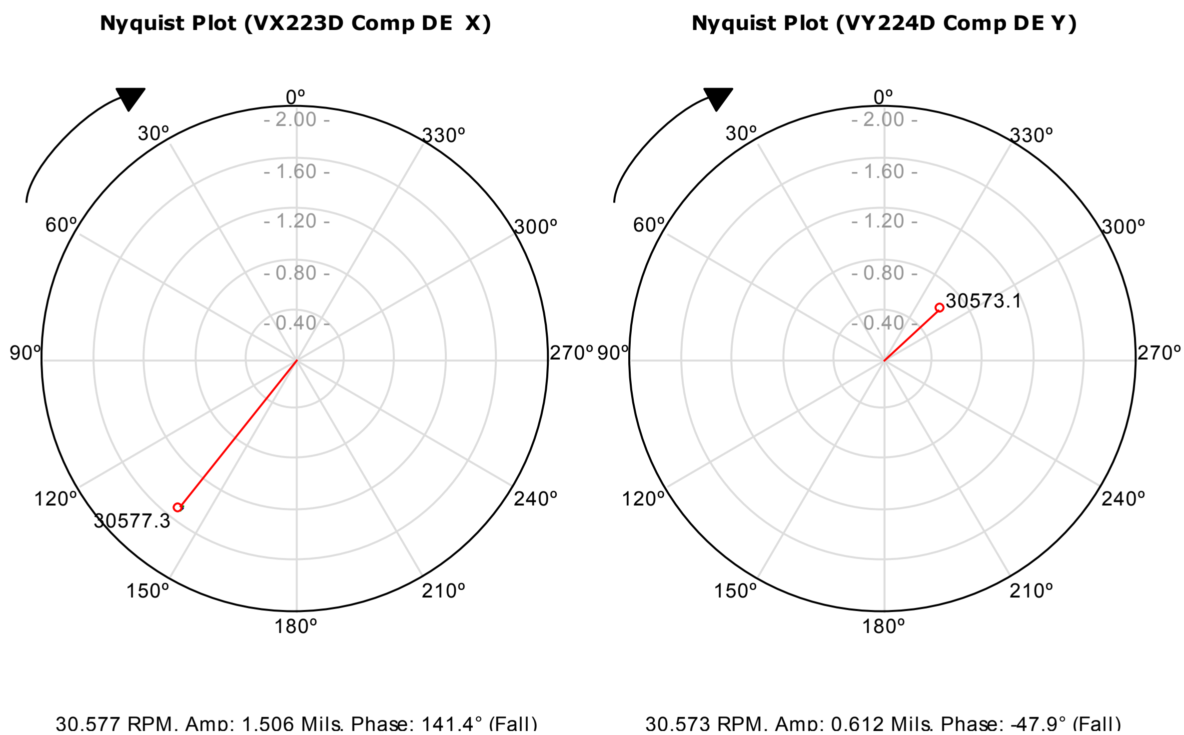


Figure 40: 11/28/2012, 3100rpm, 200psi, Compressor DE X and Y

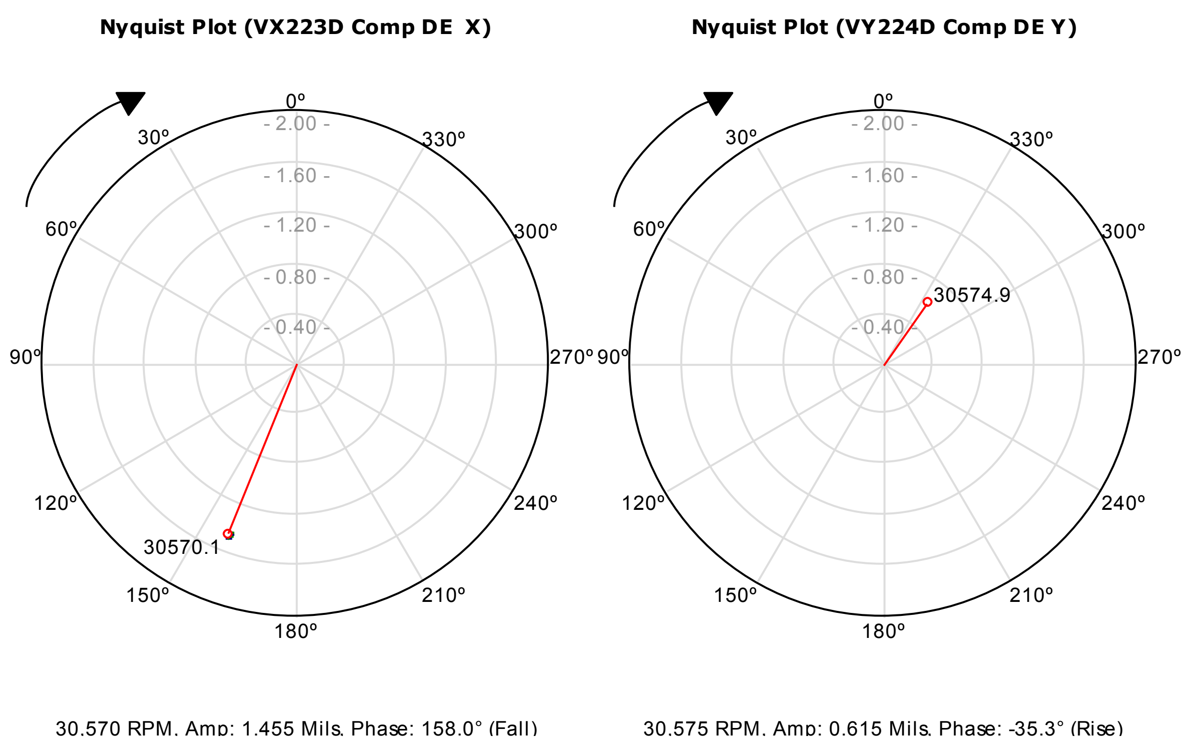


Figure 41: 11/29/2012, 3100rpm, 200psi, Compressor DE X and Y

In general, comparison of Nyquist plots on 11/28 and 11/29 at 3100rpm (30,504rpm compressor speed) shows good repeatability between the two days.

Conclusion:

Drive train vibration data appears to be repeatable between test days when comparing runs at the same power and speed level, and when the drive train is kept at constant speed. This is apparent from Nyquist plots comparing 100psi runs on 11/16 and 11/17. It is also apparent in tabulated data and Nyquist plots for 200 psi runs on 11/28, 11/29, and 11/30. Additionally, tabulated data from 12/4 shows good agreement with the runs on 11/28-11/30 even after the gearbox was removed and the high speed coupling balance altered. There is a drop in vibration amplitude at the HS coupling and compressor DE locations after the HS coupling re-balance indicating that there was some positive effect of adjusting the balance.

Appendix 7.1

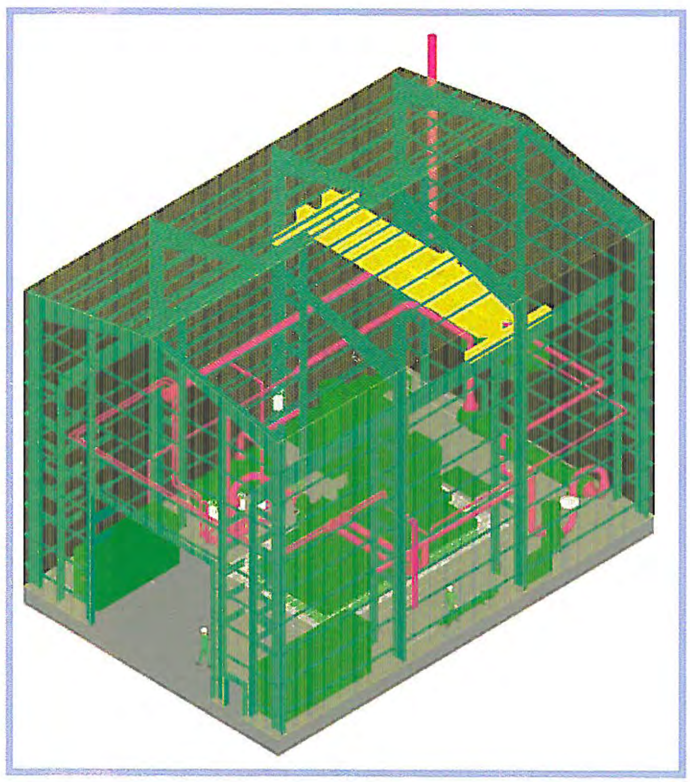
Steam Drive vs. Electric Drive

ATSI, Inc.

Front End Engineering Design
High Pressure CO2 Test Loop

Steam Drive
Vs.
Electrical Drive

Steam Turbine
Flow Diagram
Equipment Layout
Equipment Estimates
Cost Estimates



415 Commerce Dr
Amherst, New York
716-691-9200
WWW.ATSI.COM

Steam Drive verses Electrical Drive Cost Evaluation

Dresser Rand Olean Test Facilities Upgrade Project
High-Pressure CO_2 Test Loop – Phase I FEED

ENGINEERING ATSI SERVICES
ISO 9001:2000 Certified

Steam Turbine Requirements

Steam Turbine:

A steam turbine producing 13.4 khp is required for normal operation of the Rampressor. This turbine requires 100,000 lbs/hr of superheated steam at a temperature of 725 degrees F and a pressure of 550 psig. The turbine discharge will be saturated steam at a temperature of 125 degrees F and a pressure of 4" Hg absolute. The steam turbine will be designed and built by Dresser Rand in Wellsville, NY.

Steam supply:

The current boiler at Dresser Rand's Olean facility has an output of 160,000 lbs/hour at a temperature of 725 degrees F and a pressure of 600 psig, which has enough capacity to drive the steam turbine. It is assumed that the Rampressor's steam turbine will be the only load on this boiler during testing and operation of the Rampressor.

The boiler piping tie-in is located approximately 600 feet away from the new test loop, so steam supply lines and condensate return lines will connect the existing boiler piping to the new steam turbine piping via the overhead pipe bridge. The size of the steam supply lines will be 8" pipe, and the condensate return lines will be 4" pipe. There will be a 50-psi pressure drop of the supply steam due to the long length of the supply piping. The steam supply piping will include several expansion loops, steam trap stations, and a large steam separator to maintain the quality of the steam into the turbine. The steam supply lines will require insulation and electric tracing to maintain the quality of the steam into the turbine. The condensate return lines will also require insulation and electric tracing for freeze protection during the winter months.

Steam Separator:

An in-line flanged steam separator will be required to improve the quality of the steam into the steam turbine. The capacity of the steam separator will be 100,000 lbs/hour at 725 degrees F and 600 psig. The steam separator will be located close to the turbine's inlet T&T valve.

Surface Condenser:

The surface condenser will change the state of the turbine's discharge steam from saturated vapor to saturated liquid. This will require 6,000 gpm of cooling water, which will be supplied from a stand-alone 106 MBTU/hr cooling tower. The condensate from the surface condenser's discharge piping will then be returned back to the boiler at a temperature of 115 degrees F and a pressure of 30 psig. The surface condenser has been sized using a 35 degree F temperature difference, where the cooling water enters the condenser at 85 degrees F and exits at 120 degrees F.

Condensate Return Pumps:
There will be two (2) condensate return pumps located below the surface condenser. There will be one pump operational and one pump standby. These pumps will provide 250 gpm and 140' TDH with 23' NPSHa and will return the condensate back to the boiler's tie-point located at the pipe bridge. The condensate return lines will be 4" pipe. A 30 psi pressure drop is assumed through this closed system. There will be no filters in the condensate return piping.

Cooling Tower:
The cooling tower for the Turbine's surface condenser will be a stand-alone unit capable of rejecting 106 MBTU/hr of heat from the surface condenser. This stand-alone cooling tower will be located approximately 100 feet North of the North wall of the new CO_2 test facility. The cooling water supply and return lines from the surface condenser to the tower will be 20" pipe with an approximate run length of 150 feet. This cooling water will be filtered with two 20" filters in parallel before entering the surface condenser.

Cooling Water Pumps:
The cooling tower water will be circulated to the surface condenser via three (3) cooling water pumps. There will be two pumps operational and one pump standby. These pumps will provide 3,000 gpm at 140' TDH with 23' NPSHa and will circulate the cooling water through the surface condenser and back to the cooling tower. The supply and return lines will be 20" pipe. A 30 psi pressure drop is assumed through this cooling system. There will be two (2) 20" flanged filters in the condensate return piping to filter the cooling water into the surface condenser.

Reviewed: File in Wnd 6/25

Aspen ICARUS

IPM

Overall Project Summary
Account Basis

Project summary (direct and indirect costs). Direct costs presented at an account level. Indirect costs presented at a summary level.

Project Title: HP CO2 Test Loop

Project Name: DR
Scenario Name: Steam Turbine Drive
Project Location: Olean, NY
Estimate Date: 19AUG09 10:22:45

Job No: 78180
Prep. By: MAS

Project Title:	HP CO2 Test Loop	***Prepared By:***	MAS
Project Location:	Olean, NY	***Currency:***	DOLLARS USD
Estimate Date:	19AUG09 10:22:45		

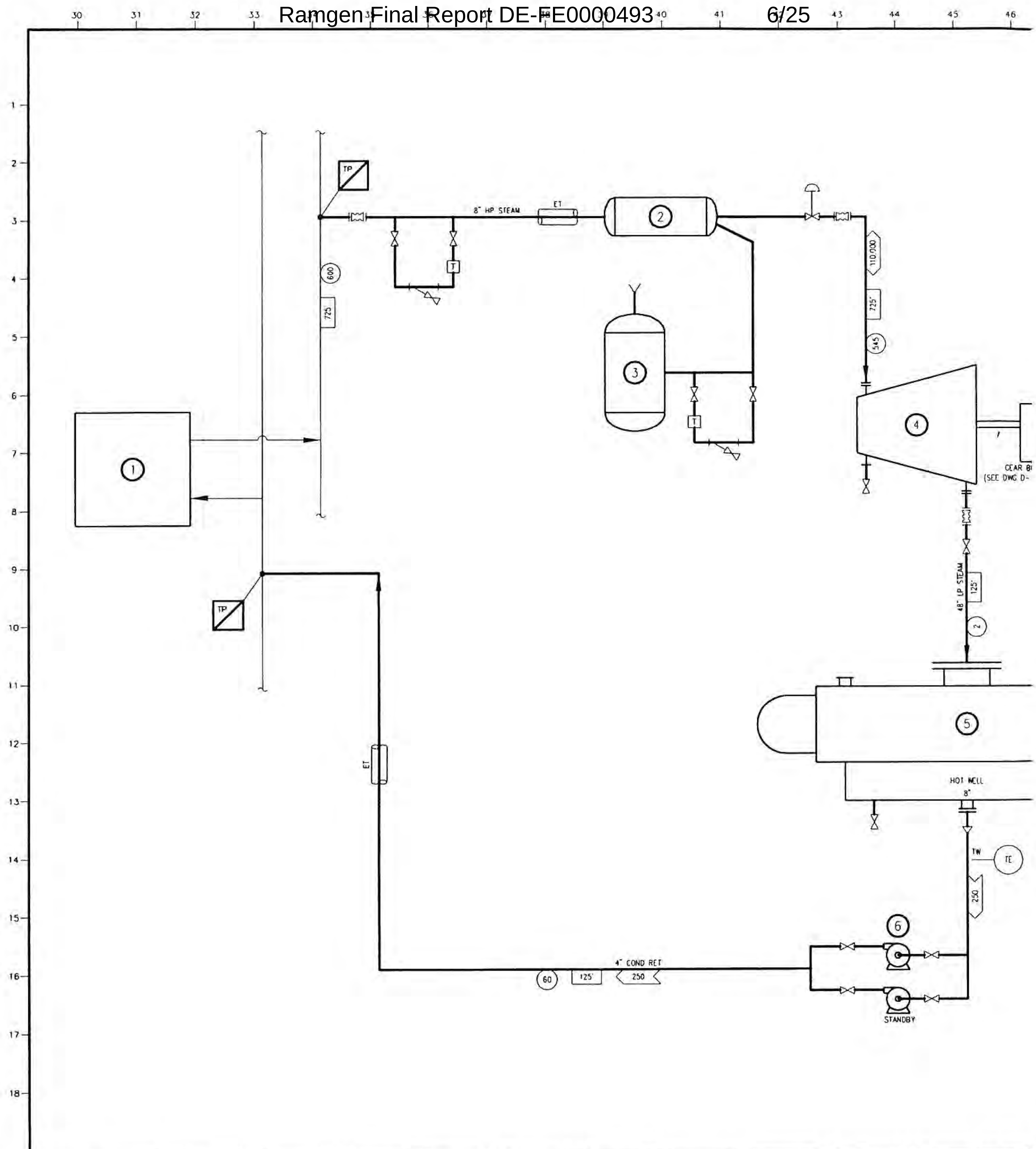
Project Notes

Report Printed At
10:24 am 8/19/09

Project Title:	HP CO2 Test Loop		
Project Location:	Olean, NY	***Prepared By:***	MAS
Job No:	78180	***Est. Class:***	Budget
Estimate Date:	19AUG09 10:22:45	***Currency:***	DOLLARS USD

Overall Project Summary - Account Basis

Account	MH	Labor Cost	Matl Cost	Total Cost
(2) Equipment	918	48,680	2,299,100	2,347,780
(3) Piping	5,002	280,225	340,123	620,347
(4) Civil	769	33,906	21,622	55,528
(5) Steel	193	8,924	17,426	26,351
(6) Instruments	104	6,141	13,254	19,395
(7) Electrical	1,742	95,171	106,691	201,862
(8) Insulation	1,929	93,376	71,408	164,783
(9) Paint	639	26,721	5,902	32,623
Direct Totals	**11,297**	**593,144**	**2,875,525**	**3,468,669**
Const Equip & Indirects				42,700
Const Mgt, Staff, Supv				41,520
Freight				0
Taxes and Permits				0
Engineering				520,300
Other Project Costs				0
Contingency				610,978
Indirect Totals				**1,215,498**
Project Totals:	**11,297**	**593,144**	**2,875,525**	**4,684,167**



[illegible]	1	2	3	4	5	6	7	8	9	10	11
[illegible]	BOILER	STEAM SEPARATOR	BLOWDOWN TANK	STEAM TURBINE	SURFACE CONDENSER	CONDENSATE RETURN PUMPS	COOLING TOWER	COOLING TOWER PUMPS	FILTERS		
[illegible]	160,000 LB/HR 725°F/600 PSIA			110,000 LB/HR 725°F/545 PSIA	110,000 LB/HR	250 GPM 125°F/75 PSIA	105 MBTU/HR 35°F ΔT/	3,000 GPM EACH 85°F/75 PSIA	14"		

ATSI Engineering Services

Steam Turbine Equipment List

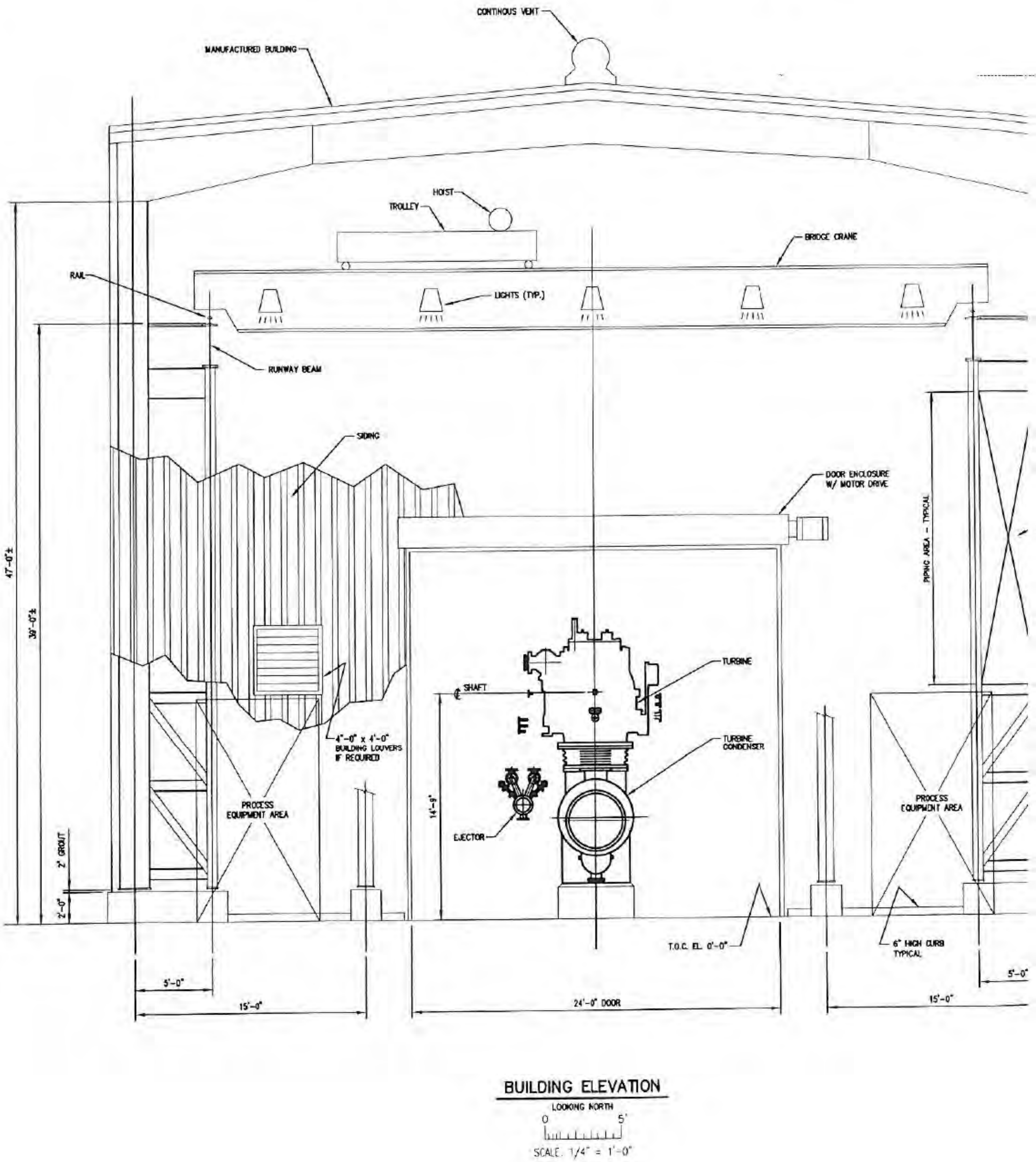
PROJECT NUMBER: 5-78180
PROJECT NAME: HP CO2 Test Loop
CLIENT: Dresser-Rand and Ramgen

Revision: A
Date: 8/3/2009

P&ID -010 rev.A (Rampressor - Steam Turbine for Rampressor)

					Dimensional Data (inches)							Connection Data					
Item	Qty.	Tag	Description	Size	Length	Width	Height	Diameter	flg to flg	Material	Wt (lbs)	Process in	Process out	CW in	CW out	Unit cost	Total cost
2	1		Steam Separator	8"	---	---	49	---	42	CS	972	8"-600#	8"-600#	---	---	$10,000	$10,000
3	1		Blowdown tank											---	---	$10,000	$10,000
4	1		Dresser-Rand Steam Turbine	---	180	112	114	---	---	---	28,000	8"-600#	48"-150#	---	---	$1,382,100	$1,382,100
5	1		Graham Surface Condenser	---	437	60	81	60	---	---	20,000	48" x 53"	8"-150#	20"-150#	20"-150#	$475,000	$475,000
6	2		Condensate return pumps	250 gpm												$10,000	$20,000
7	1		Cooling tower	105 MBTU	536	317	326	---	---	---	138,426			(6) 10"	(3) 10"	$292,000	$292,000
8	3		Cooling tower pumps	3,000 gpm												$15,000	$45,000
9	2		Cooling water filters	14"			95	48	70	CS		---	---	14"-150#	14"-150#	$16,750	$33,500
			Piping:														
	720 lf		Turbine inlet HP steam pipe, ASTM A-106 Grade B	8" Sch 80													
	24		LR 90 deg. elbows. ASTM A-106 Grade B	8" Sch 80													
	8 lf		Turbine exhaust LP steam pipe, ASTM A-106 Grade B	48" STD.													
	600 lf		Condensate return piping, ASTM A-106 grade B	4" Sch. 40													
	600 lf		Cooling tower makeup water piping, ASTM A-106 Grade B	4" Sch. 40													
	200 lf		Surface condenser cooling water piping, ASTM A-106 Grade B	20" Sch. 40													

Page 1 of 1


CONTINOUS VENT
MANUFACTURED BUILDING
HOIST
TROLLEY
BRIDGE CRANE
RAIL
LIGHTS (TYP.)
RUNWAY BEAM
SIDING
DOOR ENCLOSURE W/ MOTOR DRIVE
PIPING AREA - TYPICAL
47'-0"±
39'-0"±
TURBINE
SHAFT
TURBINE CONDENSER
4'-0" x 4'-0" BUILDING LOUVERS IF REQUIRED
PROCESS EQUIPMENT AREA
14'-9"
EJECTOR
2" GROUT
2'-0"
T.O.C. EL. 0'-0"
6" HIGH CURB TYPICAL
5'-0"
15'-0"
24'-0" DOOR
BUILDING ELEVATION
LOOKING NORTH
0
5'
SCALE: 1/4" = 1'-0"

ATSI Engineering Services

PROJECT NUMBER: 5-78180
PROJECT NAME: HP CO2 Test Loop
CLIENT: Dresser-Rand and Ramgen

Steam Turbine Equipment List

Revision: A
Date: 8/3/2009

P&ID -010 rev.A (Rampressor - Steam Turbine for Rampressor)

					Dimensional Data (inches)							Connection Data					
Item	Qty.	Tag	Description	Size	Length	Width	Height	Diameter	flg to flg	Material	Wt (lbs)	Process in	Process out	CW in	CW out	Unit cost	Total cost
2	1		Steam Separator	8"	---	---	49	---	42	CS	972	8"-600#	8"-600#	---	---	$10,000	$10,000
3	1		Blowdown tank											---	---	$10,000	$10,000
4	1		Dresser-Rand Steam Turbine	---	180	112	114	---	---	---	28,000	8"-600#	48"-150#	---	---	$1,382,100	$1,382,100
5	1		Graham Surface Condenser	---	437	60	81	60	---	---	20,000	48" x 53"	8"-150#	20"-150#	20"-150#	$475,000	$475,000
6	2		Condensate return pumps	250 gpm												$10,000	$20,000
7	1		Cooling tower	105 MBTU	536	317	326	---	---	---	138,426			(6) 10"	(3) 10"	$292,000	$292,000
8	3		Cooling tower pumps	3,000 gpm												$15,000	$45,000
9	2		Cooling water filters	14"			95	48	70	CS		---	---	14"-150#	14"-150#	$16,750	$33,500
			Piping:														
	720 lf		Turbine inlet HP steam pipe, ASTM A-106 Grade B	8" Sch 80													
	24		LR 90 deg. elbows. ASTM A-106 Grade B	8" Sch 80													
	8 lf		Turbine exhaust LP steam pipe, ASTM A-106 Grade B	48" STD.													
	600 lf		Condensate return piping, ASTM A-106 grade B	4" Sch. 40													
	600 lf		Cooling tower makeup water piping, ASTM A-106 Grade B	4" Sch. 40													
	200 lf		Surface condenser cooling water piping, ASTM A-106 Grade B	20" Sch. 40													

Page 1 of 1

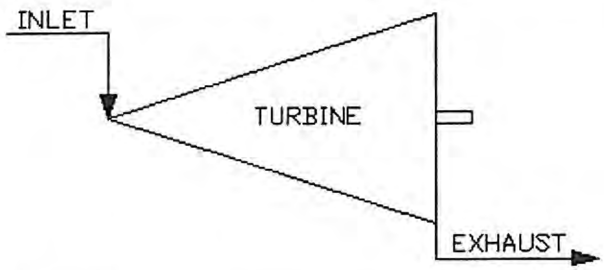
Steam Turbine Equipment Quotes

DRESSER-RAND

3800 West Avenue
Burlington, IA 52601
Ph: 319-753-5431
FAX: 319-752-1616

STEAM TURBINE BUDGET PROPOSAL

TO Hesse-Reynolds ATTN: Neil Noble
EMAIL: neil@hessereynolds.com DATE: July 22, 2009
SUBJECT: ATSI
SHEET: 1 of 2 INCLUDING THIS SHEET
SIGNED: Tim Jameson tjameson@dresser-rand.com D-R REF: T27475



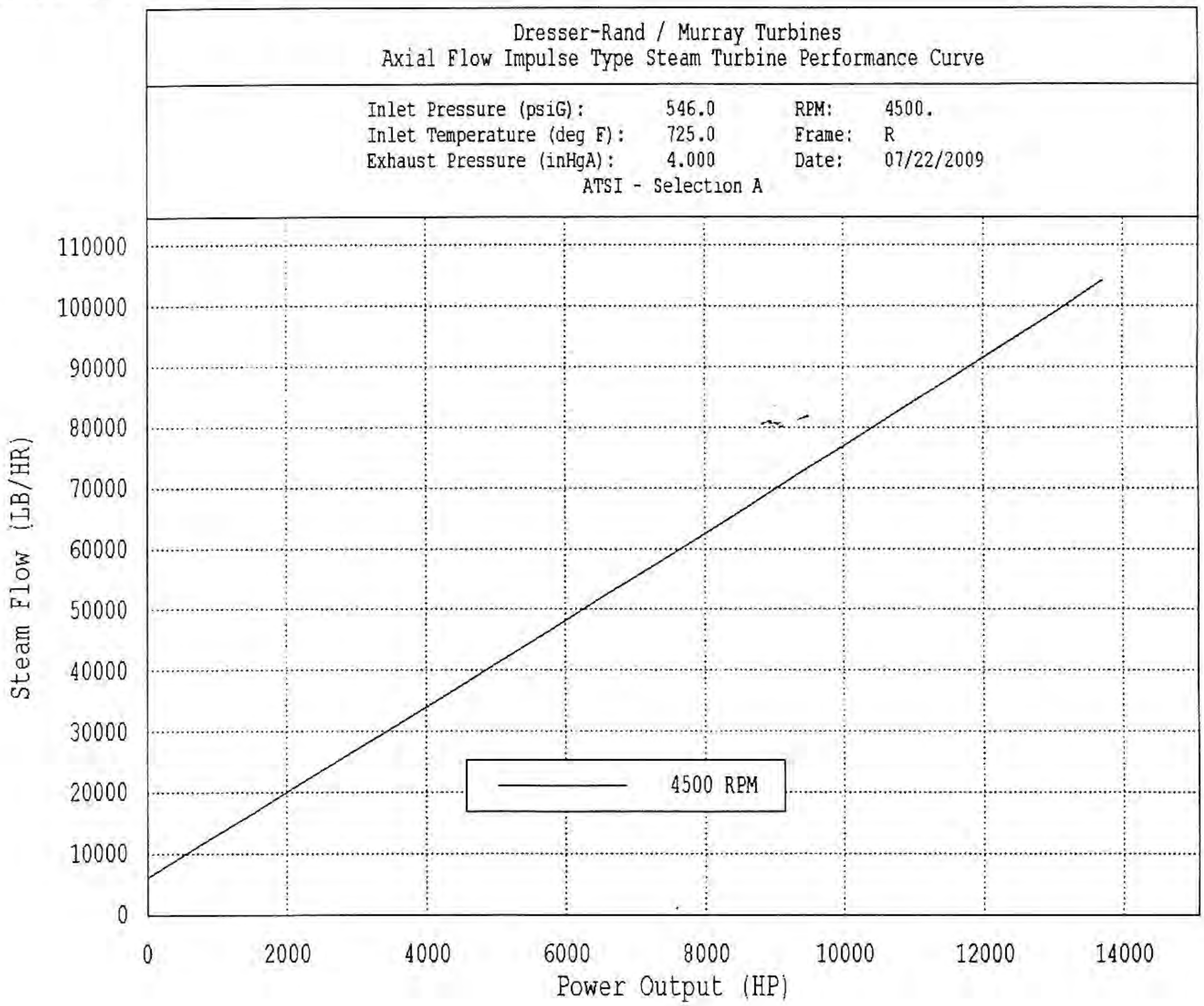
TURBINE DATA	
SELECTION	**A**
TURBINE FRAME	R
NUMBER OF STAGES	9
INLET VALVES	MULTIPLE
INLET SIZE/RATING	8"/ 600# Left or Right
EXHAUST SIZE/RATING	48"/ 125# Up or Down
PERFORMANCE DATA	
INLET PRESSURE (PSIA)	546
INLET TEMPERATURE (DEG F	725
EXHAUST PRESSURE (INHGA)	4
TURBINE SPEED (RPM)	4,500
INLET FLOW (LB/HR)	101,858
EXHAUST TEMPERATURE (DEG F)	125
EXHAUST ENTHALPY (BTU/LB)	1,033
EXHAUST STEAM QUALITY	89.8%
TURBINE OUTPUT (HP)	13,410
COMMERCIAL DATA	
SHIPMENT (WEEKS)	36
TURBINE PRICE (USD)	$ 1,382,100
Price Addition:	
Water Cooled Surface Condenser (Budget)	$ 525,000

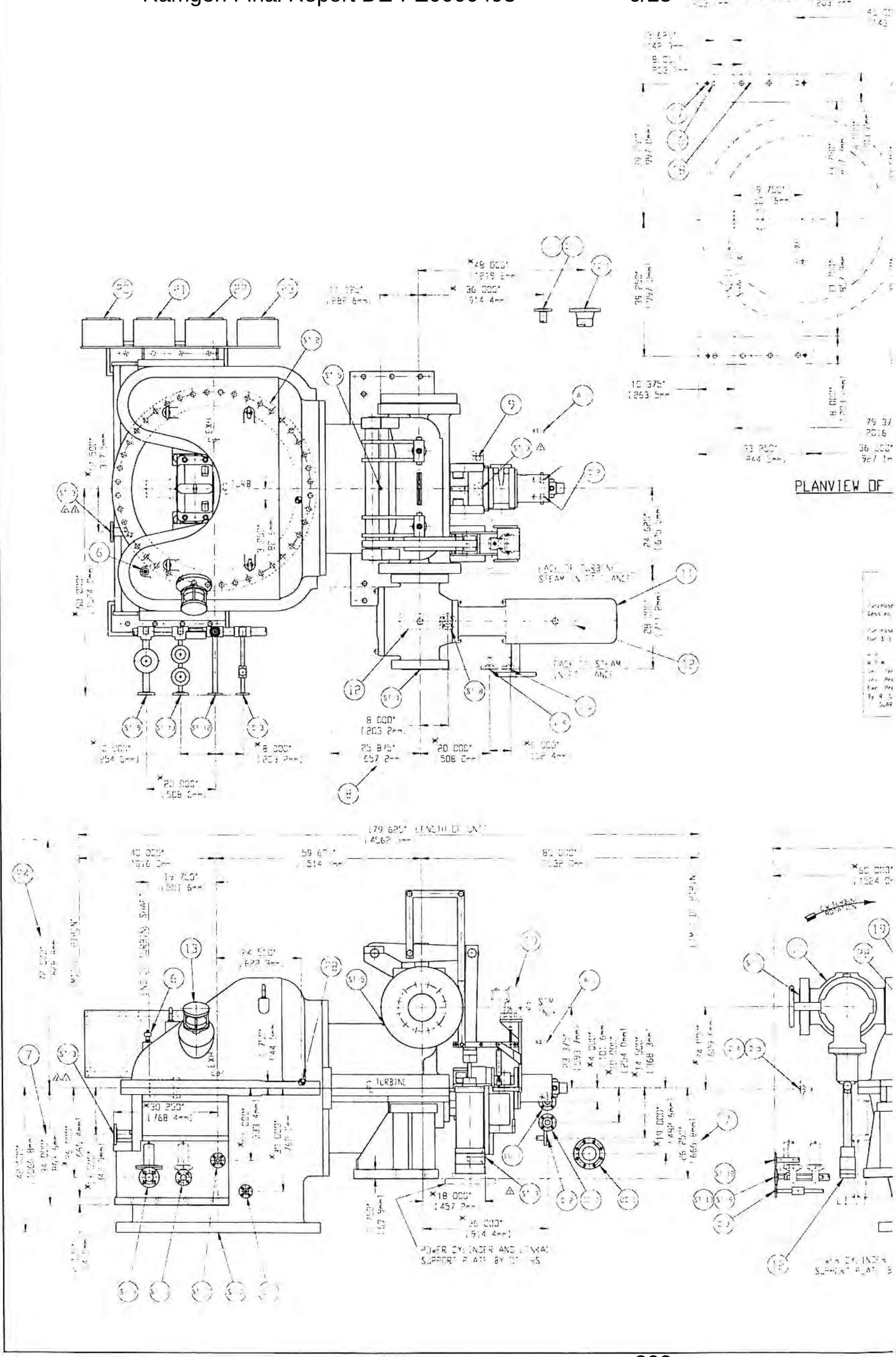
DRESSER-RAND

3800 West Avenue
Burlington, IA 52601
Ph: 319-753-5431
FAX: 319-752-1616

PRICE INCLUDES TURBINE, GLAND CONDENSER AND EJECTOR, LABYRINTH SEALS, ELECTRONIC OVERSPEED TRIP, WOODWARD DG505 GOVERNOR WITH HYDRAULIC ACTUATOR, OIL OPERATED GIMPEL T&T VALVE, MANIFOLD OIL PIPING, BEARING DRAIN LINE THERMOMETERS AND SIGHT FLOW INDICATORS, SARCO AUTOMATIC CASING DRAIN PUMP (FOR UP EXHAUST), HYDRAULIC VACUUM BREAKER, BLANKET INSULATION WITHOUT SHEET METAL LAGGING, GAUGEBOARD & TACHOMETER, VIBRATION PROBES, AXIAL PROBES, RTD'S, PRICE FCA BURLINGTON, IOWA.

PRICE DOES NOT INCLUDE SURFACE CONDENSER, INTERCONNECTING PIPING BETWEEN TURBINE AND LUBE SYSTEM, OR BETWEEN STEAM TURBINE AND CONDENSER, FREIGHT, OR START-UP SUPERVISION.




PLANVIEW OF

GOULDS PUMPS ESTABROOK CORP

ATSI Engineering

August 3, 2009

INQ NO: Dresser Rand Pilot Plant Project
Proposal No: KM09-08-03 01
Item No: ITEM 002

Attn: Kevin O'Connor

MODEL:3410 M SIZE:8x10-14 QTY: 3

PRICES in USD	
Pump Unit	14,789
Driver	7,108
Subtotal 3 Units	65,691
Boxing	
Testing	
Freight	
Accessories	
Total 3 Units	**65,691**

Operating conditions

SERVICE	
LIQUID	Water Temp. 85.0 deg F, SP.GR 1.000, Viscosity 1.000 cp, rated / max. suction pressure 0.0 / 0.0 psi g
CAPACITY Rated	3,000.0 gpm
HEAD	140.0 (ft)

Performance at 1785 RPM

PUBLISHED EFFY	85.0% (CDS)
RATED EFFY	84.0% with contract seal
RATED POWER	126.3 hp (incl. Mech. seal drag 0.52). (Run out 137.2 hp)
NPSHR	12.6 ft (available NPSH is 23.0 ft)
DISCH PRESSURE(R)	61.8 psi g (79.4 psi g @ Shut off) Based on 0.0 psi g Suc.press
PERF. CURVE	3843-1 (Rotation CW viewed from coupling end)
SHUT OFF HEAD	183.5 ft
MIN. FLOW	Continuous Stable: 969.3 gpm Hydraulic: 969.3 gpm Thermal: N/A

Materials

CONSTRUCTION	Bronze fitted
CASING	Cast iron (max.casing.pres. @ rated temp. 175.0 psi g)
CASING WEAR RING	Bronze
IMPELLER	Bronze - Enclosed (13.1250 in rated, max=14.0000 in, min=10.0000 in)
CASING GASKETS	Non asbestos
SHAFT MATERIAL	SAE 4140
SHAFT SLEEVE	Bronze
LUBRICATION	Regreasable bearings
SEAL CHAMBER	Enlarged bore
GLAND	Bronze Flush
BEARINGS	SKF 6207 (Inboard) SKF 5306 A/C3 (Outboard)
COUPLING	Falk - T10 1090T-S.F. 1.00
COUPLING GUARD	Carbon steel
BASEPLATE	Cast iron D00091A

Sealing Method

MECHANICAL SEAL	John Crane 8-1T XF(51)1XO(10)1 (Carbon vs Ceramic) - (Conventional - Single)

Flanges

125# flat face

Liquid end features
1/4in bronze casing vent valve

Frame features
Labyrinth oil seals - Inpro VBX
Single extended shaft

Piping
Copper bypass tubing

Testing
Non witnessed casing hydrostatic-test

Painting
Goulds Blue standard painting

Optional Features:

Impeller wear ring
Bronze add 424

Casing connections
Tapped suction and discharge gauge (4 taps) add 316

All above optional adders are per unit in (USD)

Noise level Data
Maximum predicted sound pressures level pump only in Decibels (db) Re 0.0002 microbars measured 3ft horizontally and 5ft from the floor per QCP 580

Noise Level	31.5	63	125	250	500	1k	2k	4k	8k	A
Pump	68.0	66.0	71.0	81.0	77.0	79.0	78.0	76.0	71.0	84.0

Driver : Electric motor Manufacturer : Pump mfg`s Choice

FURNISHED BY	Pump mfg	MOUNTED BY	Pump mfg
RATING	150.0 hp (111.9 KW)	ENCLOSURE	Severe Duty/Mill and Chemical Premium Efficiency
PHASE/FREQ/VOLTS	3/60 Hz/460	SPEED	1800 RPM
INSULATION/SF	F/1.15	FRAME	445T

Weights and Measurements

TOTAL NET UNIT WEIGHT / VOLUME	3,283.0 lb / 44.5 ft^3
TOTAL GROSS UNIT WEIGHT / GROSS VOLUME	3,623.0 lb / 68.0 ft^3

Program Version 1.30.0.0

Our offer does not include specific review and incorporation of any Statutory or Regulatory Requirements and the offer is limited to the requirements of the design specifications. Should any Statutory or Regulatory requirements need to be reviewed and incorporated then the Customer is responsible to identify those and provide copies for review and revision of our offer.

Our quotation is offered in accordance with our conditions of Sale.

Click here to download the pump Bulletin

PUMPSMART FLOW ECONOMY ESTIMATES

FIXED SPEED

25.3
gpm/kW

Expected range for typical operation 20.0 to 29.9 gpm/kW

PUMPS

37.1
gpm/kW

Expected range for typical operation 29.2 to 44.1 gpm/kW

Click Here To Learn More!

Estimated Annual Savings 16,134 USD

Are you aware of PumpSmart Process Systems?

PumpSmart is a system that utilizes a standard process pump in conjuction with ITT 's unique and patented PumpSmart Control System and Software. The software, which resides on the controller microprocessor chip, allows the pump to monitor and react to any system condition.

PumpSmart

* Eliminates control valves, flow meters, and recirculation lines.
* Significantly reduces energy costs.
* Significantly increases MTBF.

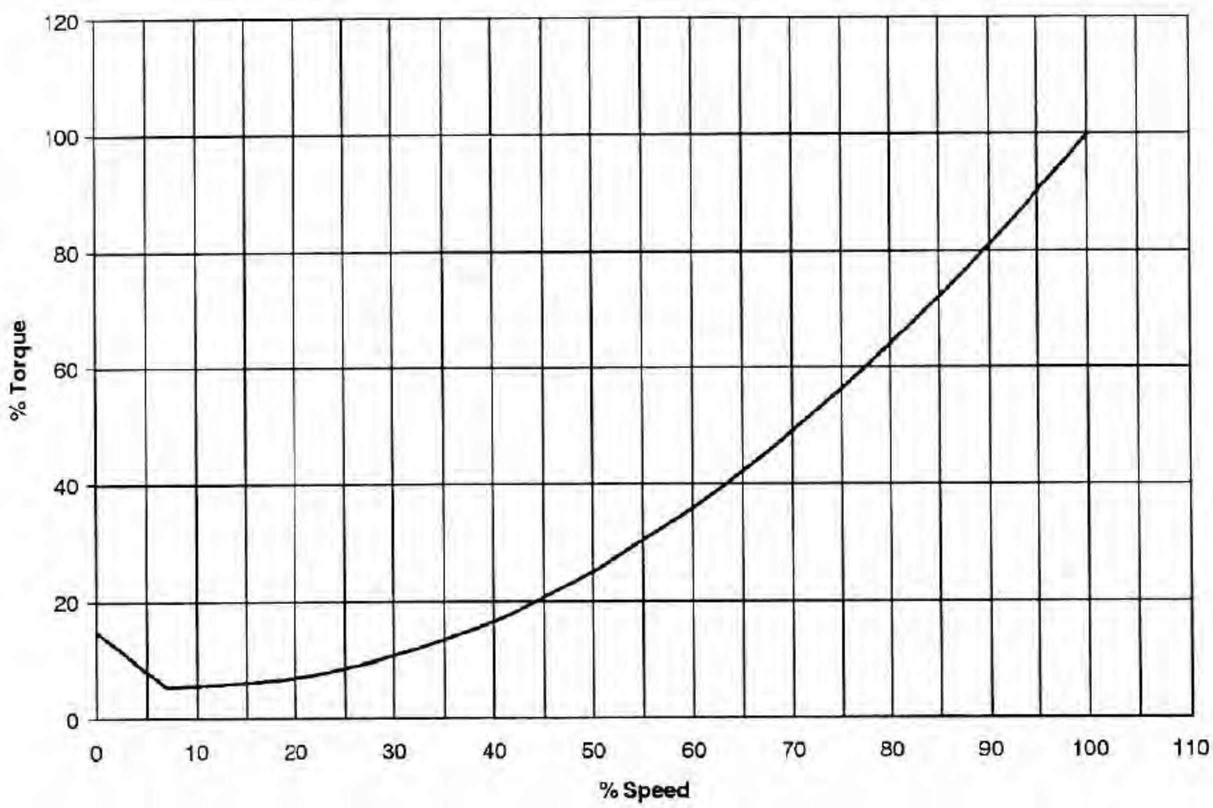
Please contact your local Goulds Pumps representative for details and a demonstration CD-ROM. You may also contact us at www.gouldspumps.com or e-mail pumpsmart@fluids.ittind.com.

GOULDS PUMPS

Speed - Torque Curve

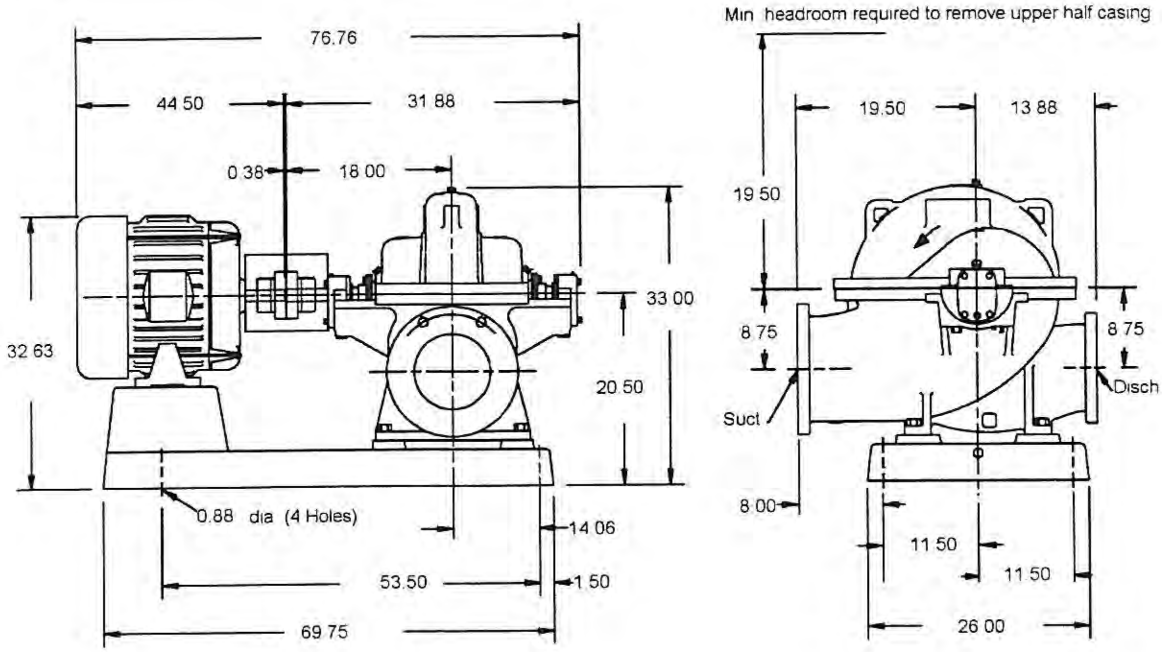
S.O. No.	
Customer :	ATSI Engineering
P.O. No. :	
Proposal No :	KM09-08-03 01
Item No. :	ITEM 002 3410 / M / 8x10-14
Speed :	1785 in rpm.
100% Torque :	189.70 in Feet*lb force At Shut off (closed valve)
100% Torque :	403.50 in Feet*lb force At Run out (open valve)
100% Torque :	371.50 in Feet*lb force At Rated Point
WR2 of pump :	

Speed-Torque of the Pump



Size: 8x10-14
Rev. No: 0
Date: 08/03/2009

GOULDS PUMPS **OUTLINE DRAWING** **Model 3410M** **Size 8x10-14** ITT



Pump specification

SUCT.FLANGE SIZE *10"*	DRILLING *ANSI 125 #*	FACING *FF*	FINISH *SMOOTH*
DISCH.FLANGE SIZE *8"*	DRILLING *ANSI 125 #*	FACING *FF*	FINISH *SMOOTH*
PUMP ROTATION (LOOKING AT PUMP FROM MOTOR) *CW*			
TYPE OF LUBRICATION *REGREASABLE BEARINGS*		COOLED *NO*	
TYPE OF STUFFING BOX *ENLARGED BORE*		COOLED *NO*	
TYPE OF SEALING *MECHANICAL SEAL*			

Weights and Measurements

PUMP	996.0 lb
MOTOR/CPLG	1,650.0/57.0 lb
BASEPLATE	580.0 lb
TOTAL	3,283.0 lb
GR.VOLUME w/BOX	68.0 ft³
GR.WEIGHT w/BOX	3,623.0 lb

Motor specification

MOTOR BY *PUMP MFG*	MOUNT BY *PUMP MFG*	MFG. *PUMP MFG'S CHOICE*
FRAME *445T*	POWER *150.0 hp*	RPM *1800*
PHASE *3*	FREQUENCY *60 HZ*	VOLTS *460*
INSULATION *F*	S.F. *1.15*	
ENCLOSURE *SEVERE DUTY/MILL AND CHEMICAL PREMIUM EFFICIENCY*		

Notes and References

- MTR DIMENSIONS ARE APPROXIMATE
- INSTALL FOUNDATION BOLTS IN PIPE SLEEVES
- ALLOW FROM 0.75 to 1.50in. FOR GROUTING. SEE INSTRUCTION BOOK FOR DETAILS.

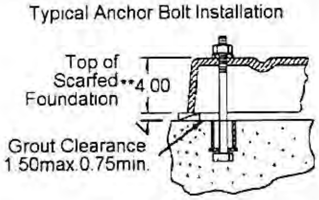
** Foundation bolt grip thickness

FOR PUMP TAPPED OPENINGS REFER TO DWG.: TKM09-08-03 01 / ITEM 002

Auxiliary specification

COUPLING BY *PUMP MFG*	CPLG TYPE *FALK T10 1090T*
CPL GUARD BY *PUMP MFG.*	CPLG GUARD MATL *CARBON STEEL*
BASEPLATE *CAST IRON D00091A*	
MECH SEAL *JOHN CRANE 8-1T XF(51)1XO(10)1 (CARBON VS CERAMIC)*	

Typical Anchor Bolt Installation



DRAWING IS FOR REFERENCE ONLY.
NOT CERTIFIED FOR CONSTRUCTION UNLESS SIGNED.

Customer: ATSI Engineering
Serial No:
Customer P.O. No:
Item No: ITEM 002
End User: DRESSER RAND OLEAN
Service:

All dimensions are in inches.
Drawing is not to scale
Weights (lbs) are approximate

DRAWING NO KM09-08-03 01/ITEM 002

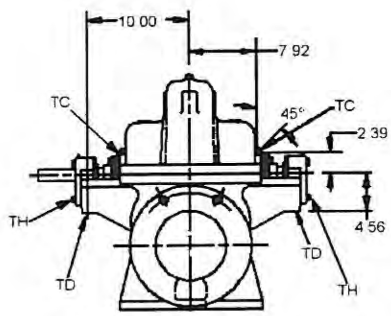
FORM # ED0188 N/A

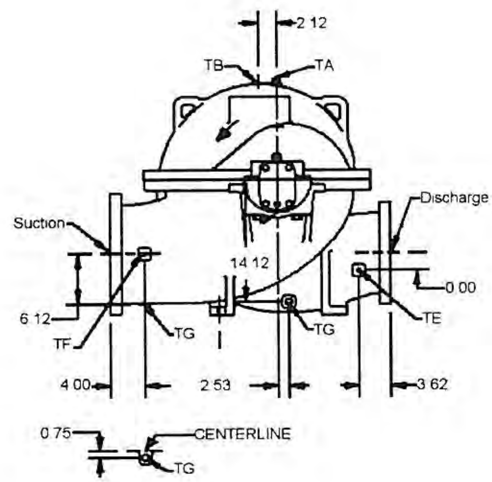
Program Version 1.30.0.0

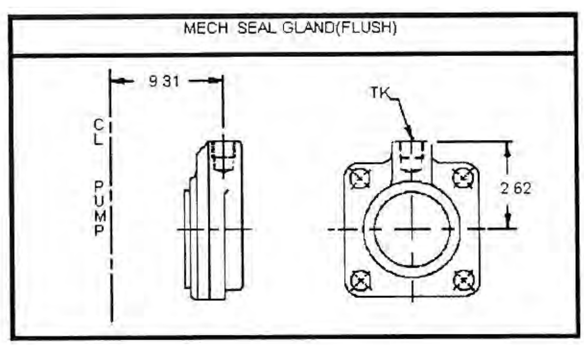
GOULDS PUMPS **TAPPED OPENINGS** **Model 3410 M Size 8x10-14** ITT

TAPPED OPENINGS MODEL 3410 M 8x10-14

NO.	SIZE	QTY.	PURPOSE	FURNISHED YES/NO	NO.	SIZE	QTY.	PURPOSE	FURNISHED YES/NO
TA	3/8	1	CASING VENT	YES	TH	1/4	2	OIL DRAIN	YES
TB	3/4	1	CASING PRIME CONN	YES	TJ	1/4	4	BEARING COOLING CONN	NO
TC	3/8	2	STUFF. BOX SEAL RING CONN	YES	TK	3/8	2	GLAND FLUSH CONN	YES
TD	3/4	2	STUFF.BOX OVERFLOW CONN	YES	TL	1/4	2	GLAND VENT CONN	NO
TE	3/8	1	DISCH. GAUGE CONNECTION	YES	TM	3/8	2	GLAND DRAIN CONN	NO
TF	3/8	1	SUCTION GAUGE CONNECTION	YES	TN	1/4	4	GLAND QUENCH CONN	NO
TG	3/4	2	CASING DRAIN CONN	YES					







DRAWING IS FOR REFERENCE ONLY.
NOT CERTIFIED FOR CONSTRUCTION UNLESS SIGNED.

Customer: ATSI Engineering
Serial No:
Customer P.O. No:
Item No: ITEM 002
End User: DRESSER RAND OLEAN
Service:

All dimensions are in inches.
Drawing is not to scale

DRAWING NO KM09-08-03 01/ITEM 002

FORM # ED0189 +850 Program Version 1.30.0.0

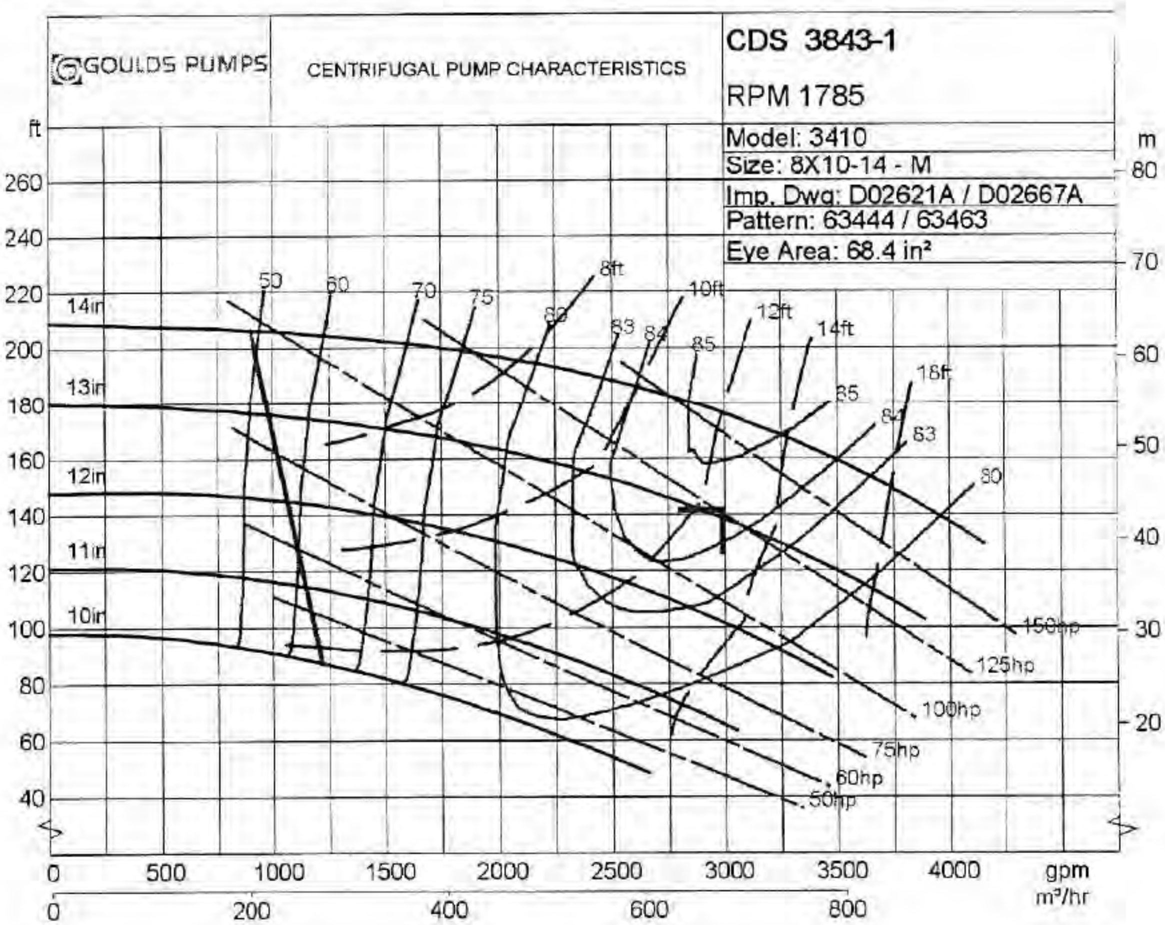
Model: 3410	Size: 8x10-14	Group: M	60Hz	RPM: 1785	Stages: 1

Job/Inq.No. :	Dresser Rand Pilot Plant Project				
Purchaser :	ATSI Engineering				
End User :	DRESSER RAND OLEAN	Issued by :	Kevin Mckenzie		
Item/Equip.No. :	ITEM 002	Quotation No. :	KM09-08-03 01	Date :	08/03/2009
Service :					
Order No. :		Certified By :		Rev. :	0

Operating Conditions / **Pump Performance**

Operating Conditions		Pump Performance			
Liquid:	Water	Published Efficiency:	85.0 %	Suction Specific Speed:	10,812 gpm(US) ft
Temp.:	85.0 deg F	Rated Pump Efficiency:	84.0 %	Min. Hydraulic Flow:	969.3 gpm
S.G./Visc.:	1.000/1.000 cp	Rated Total Power:	126.3 hp	Min. Thermal Flow:	N/A
Flow:	3,000.0 gpm	Non-Overloading Power:	137.2 hp		
TDH:	140.0 ft	Imp. Dia. First 1 Stg(s):	13.1250 in		
NPSHa:	23.0 ft	NPSHr:	12.6 ft		
Solid size:		Shut off Head:	183.5 ft		
% Susp. Solids (by wtg):		Vapor Press:			
Max. Solids Size:	1.0600 in				

Notes: 1. Power and efficiency Losses are not reflected on the curve below.
2. Elevated temperature effects on performance are not included.



Model: 3410	Size: 8x10-14	Group: M	60Hz	RPM: 1785	Stages: 1

Job/Inq.No. : Dresser Rand Pilot Plant Project
Purchaser : ATSI Engineering
End User : DRESSER RAND OLEAN
Rev. : 0
Issued by : Kevin Mckenzie
Item/Equip.No. : ITEM 002
Quotation No. : KM09-08-03 01
Date : 08/03/2009
Service :
Order No. :
Certified By :

Operating Conditions

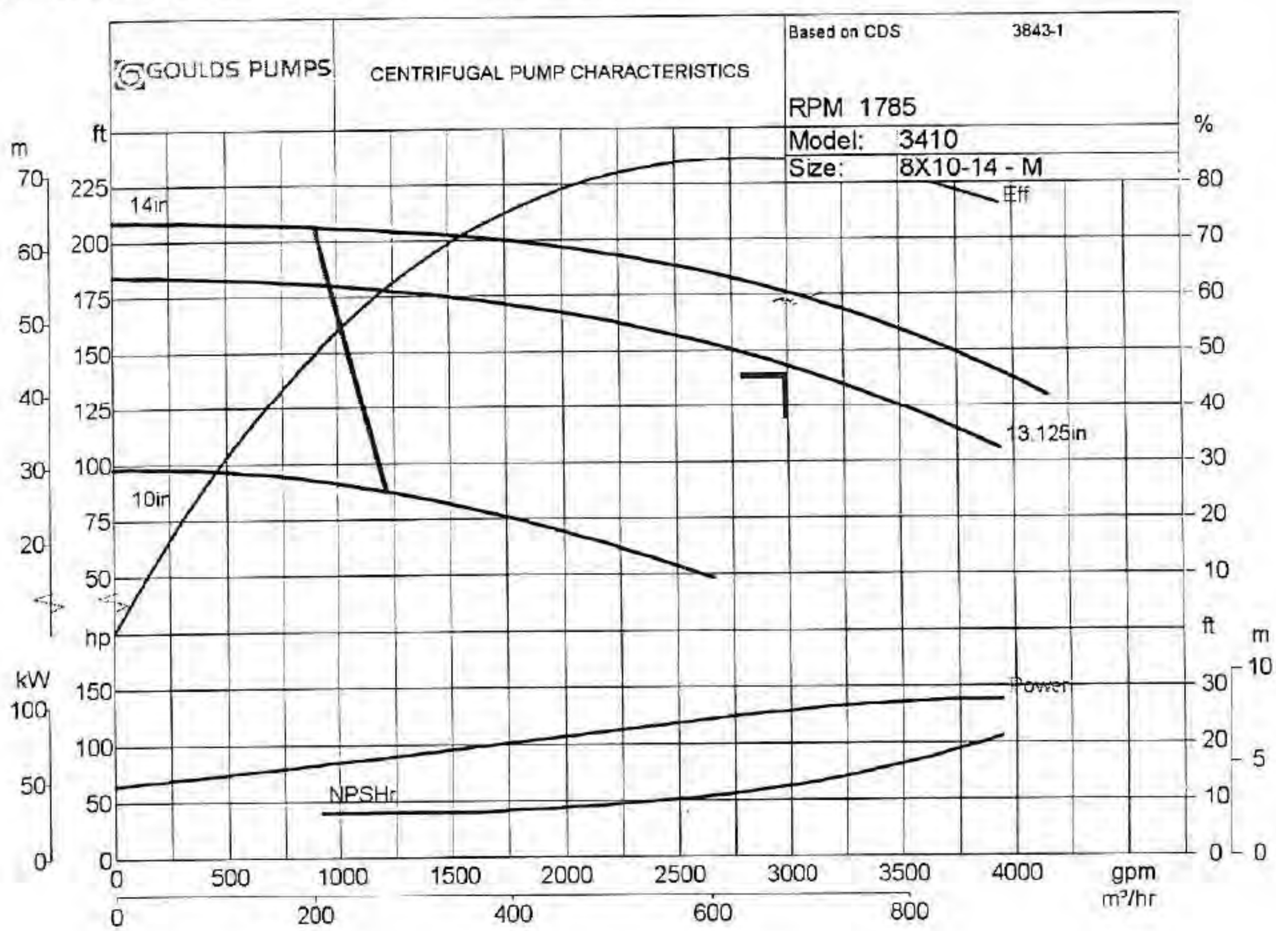
Liquid:	Water
Temp.:	85.0 deg F
S.G./Visc.:	1.000/1.000 cp
Flow:	3,000.0 gpm
TDH:	140.0 ft
NPSHa:	23.0 ft
Solid size:	
% Susp. Solids (by wtg):	
Max. Solids Size:	1.0600 in

Pump Performance

Published Efficiency:	85.0 %
Rated Pump Efficiency:	84.0 %
Rated Total Power:	126.3 hp
Non-Overloading Power:	137.2 hp
Imp. Dia. First 1 Stg(s):	13.1250 in
NPSHr:	12.6 ft
Shut off Head:	183.5 ft
Vapor Press:	

Suction Specific Speed:	10,812 gpm(US) ft
Min. Hydraulic Flow:	969.3 gpm
Min. Thermal Flow:	N/A

Notes: 1. Elevated temperature effects on performance are not included.



New Product Announcement

ONBOARD INTELLIGENCE BECOMES STANDARD ON WORLD'S BEST-SELLING PROCESS PUMP

On July 1, 2008 ITT Goulds Pumps proudly began production of the next generation ANSI pump - the **3196 i-FRAME ™**.

Marking an industry first, the new 3196 i-FRAME provides operations personnel, maintenance managers, and reliability engineers - the people responsible for monitoring and repairing rotating equipment - with early warning of trouble so that changes to the process or machine can be made before failure occurs.

The 3196 i-FRAME's patent-pending condition monitor is nested securely atop the power end to measure critical vibration and temperature readings. Variations in temperature or vibration that exceed preset parameters will activate the early warning system by displaying flashing red lights easily recognized during routine walk-arounds.

The Goulds Model 3196 is already the best-selling process pump in the world and now we've made it even better. This increased reliability and condition monitoring intelligence gets to the heart of our most important customer requirement - reduced downtime and equipment Life Cycle Cost.

In addition to the condition monitor built into the pump, this innovative new design incorporates many other standard features designed to increase reliability and the life of the pump. They include:

- Premium severe duty thrust bearings which increase fatigue life by 2 to 5 times that of standard bearings
- Dual stainless-steel, bronze-bearing isolators for improved corrosion resistance and contaminant exclusion
- An optimized sump design to improve heat transfer and collect and concentrate contaminants away from the bearings, resulting in longer bearing life

So in addition to providing great additional value, buying the 3196 i-FRAME gives you peace of mind in knowing that you have an authentic Goulds pump, designed and manufactured to our quality standards to minimize Life Cycle Costs and maximize uptime.

We're so confident this is the most reliable and intelligent product on the market, we will back every 3196 i-FRAME pump with a 5-Year Warranty as standard.

The Goulds 3196 pump continues to be recognized as a workhorse in chemical, oil and gas, petrochemical, pulp and paper, and other industrial processes, making it the most popular process pump in the world. It is available in 29 different sizes with a wide range of features for handling challenging applications.

ITT

Industrial Process

Terms & Conditions of Sale

WARRANTY
(a) Company warrants that on the date of shipment the goods are of the kind and quality described herein and are free of non-conformities in workmanship and material. This warranty does not apply to goods or parts delivered by Company but manufactured by others. (b) Buyer's exclusive remedy for nonconformity in any item of the goods shall be the repair or the replacement (at Company's option) of the item and any affected part of the goods. Company's obligation to repair or replace shall be in effect for a period of one (1) year from initial operation of the goods but not more than eighteen (18) months from Company's shipment of the goods, provided Buyer has sent written notice within that period of time to Company that the goods do not conform to the above warranty. Repaired and replacement parts shall be warranted for the remainder of the original period of notification set forth above, but in no event less than 12 months from repair or replacement. At its sole expense, Buyer shall remove and ship to Company any such nonconforming goods and shall reinstall the repaired or replaced goods or parts. Buyer shall grant Company access to the goodsnat all reasonable times in order for Company to determine any nonconformity in the goods. Company shall have the right of disposal of items replaced by it. If Company is unable or unwillingnto repair or replace, or if repair or replacement does not remedy the nonconformity, Company and Buyer shall negotiate an equitable adjustment in the order price, which may include a full refund of the order price for the nonconforming goods. (c) COMPANY HEREBY DISCLAIMS ALL OTHER WARRANTIES, EXPRESS OR IMPLIED, EXCEPT THAT OF TITLE. SPECIFICALLY, IT DISCLAIMS THE IMPLIED WARRANTIES OF MERCHANTABILITY, FITNESS FOR A PARTICULAR PURPOSE, COURSE OF DEALING AND USAGE OF TRADE. (d) Buyer and successors of Buyer are limited to the remedies specified in this article and shall have no others for a nonconformity in the goods. Buyer agrees that these remedies provide Buyer and its successors with a minimum adequate remedy and are their exclusive remedies, whether Buyer's or its successors' remedies are based on contract, warranty, tort (including negligence), strict liability, indemnity, or any other legal theory, and whether arising out of warranties, representations, instructions, installations, or non-conformities from any cause. Buyer shall assume all responsibility and expense for removal, reinstallation and freight in connection with these remedies. (e) Company neither assumes, nor authorizes any person to assume for it, any other obligation in connection with the sale of its goods. This warranty shall not apply to any goods that: (1) have been repaired or altered outside of Company's factories or authorized service centers, in any manner; or (2) have been subjected to misuse, negligence or accidents, or (3) have been improperly stored or handled or used in a manner contrary to Company's instructions or recommendations; or (4) have design errors due to inaccurate or incomplete information supplied by Buyer or its agents.

LIMITATION OF LIABILITY
NEITHER COMPANY, NOR ITS SUPPLIERS SHALL BE LIABLE, WHETHER IN CONTRACT, WARRANTY, FAILURE OF A REMEDY TO ACHIEVE ITS INTENDED OR ESSENTIAL PURPOSES, TORT (INCLUDING NEGLIGENCE), STRICT LIABILITY, INDEMNITY OR ANY OTHER LEGAL THEORY, FOR LOSS OF USE, REVENUE OR PROFIT, OR FOR COSTS OF CAPITAL OR OF SUBSTITUTE USE OR PERFORMANCE, OR FOR INDIRECT, SPECIAL, LIQUIDATED, INCIDENTAL OR CONSEQUENTIAL DAMAGES, OR FOR ANY OTHER LOSS OR COST OF A SIMILAR TYPE, OR FOR CLAIMS BY BUYER FOR DAMAGES OF BUYER'S CUSTOMERS. COMPANY'S MAXIMUM LIABILITY UNDER THIS CONTRACT SHALL BE THE CONTRACT PRICE. BUYER AND COMPANY AGREE THAT THE EXCLUSIONS AND LIMITATIONS SET FORTH IN THIS ARTICLE ARE SEPARATE AND INDEPENDENT FROM ANY REMEDIES WHICH BUYER MAY HAVE HEREUNDER AND SHALL BE GIVEN FULL FORCE AND EFFECT WHETHER OR NOT ANY OR ALL SUCH REMEDIES SHALL BE DEEMED TO HAVE FAILED OF THEIR ESSENTIAL PURPOSE.

GENERAL
(a) Company will comply with all laws applicable to Company during manufacture and sale of the goods. Purchaser will comply with all laws applicable to Purchaser during operation or use of the goods. (b) The laws of the State of New York shall govern the validity, interpretation and enforcement of any order of which these provisions are a part, without giving effect to any rules governing the conflict of laws. The application of the United Nations Convention on Contracts for the International Sale of Goods shall be excluded. (c) Assignment may be made only with written consent of both parties, provided, however, Company may assign to its affiliate without Buyer's consent. (d) Buyer shall be liable to Company for any attorney's fees and costs incurred by Company in enforcing any of its rights hereunder. This document and any other documents specifically referred to as being a part hereof, constitute the entire contract on the subject mater, and it shall not be modified except in writing signed by both parties. Unless otherwise specified, any reference to Buyer's order is for identification only.

ACCEPTANCE
The determination of compliance with performance guarantees will be based on results of factory tests under controlled conditions with calibrated instruments and tested per standards of the Hydraulic Institute, ISO standards, API standards, or other nationally recognized accreditation standards.

STATUTE OF LIMITATIONS
To the extent permitted by applicable law, any lawsuit for breach of contract, including breach of warranty, arising out of the transactions covered by this order, must be commenced not later than twelve (12) months from the date the cause of action accrued.

SHIPMENT
The term "shipment" means delivery to the initial carrier in accordance with the delivery terms of this order. Company may make partial shipments. Company shall select method of transportation and route, unless terms are f.o.b. point of shipment and Buyer specifies the method and route and is to pay the freight costs in addition to the price. When terms are f.o.b. destination or freight allowed to destination, "destination" means common carrier delivery point -within the continental United States, excluding Alaska- nearest the destination. For movement outside the United States, company shall arrange for inland carriage to port of exit and shall cooperate with Buyer's agents in making necessary arrangements for overseas carriage and preparing necessary documents.

SPECIAL SHIPPING DEVICES
On shipments to a destination in the continental United States or Canada, Company has the right to add to the invoice, as a separate item, the value of any special shipping device (barrel, reel, tarpaulin, cradle, crib and the like) used to contain or protect the goods invoiced, while in transit. Full credit will be given on the return to Company of the device in a reusable condition, f.o.b. destination, freight prepaid.

DELAYS
If Company suffers delay in performance due to any cause beyond its control, including but not limited to act of God, war, act or failure to act of government, act or omission of Buyer, fire, flood, strike or labor troubles, sabotage, or delay in obtaining from others suitable services, materials, components, equipment or transportation, the time of performance shall be extended a period of time equal to the period of the delay and its consequences. Company will give Buyer notice in writing within a reasonable time after Company becomes aware of any such delay.

NONCANCELLATION
Buyer may not cancel or terminate for convenience, or direct suspension of manufacture, except with Company's written consent upon terms agreed to by Company.

STORAGE
Any item of the goods on which manufacture or shipment is delayed by causes within Buyer's control, or by causes which affect Buyer's ability to receive the goods, may be placed in storage by Company for Buyer's account and risk and Buyer shall pay all charges for storage and shipping and incidental expenses.

TITLE AND INSURANCE
Title to the goods and risk of loss or damage shall pass to Buyer at the f.o.b. point, except that a security interest in the goods and proceeds and any replacement shall remain in Company, regardless of mode of attachment to realty or other property, until the full price has been paid in cash. Buyer agrees to do all acts necessary to perfect and maintain said security interest, and to protect Company's interest by adequately insuring the goods against loss or damage from any external cause with Company named as insured or co-insured.

INSPECTIONS / EXPEDITING
The Company restricts access to its facilities and requires seventy two (72) hours notice prior to each visit. Company requires that its agents or employees accompany inspectors/expeditors on their visit(s).

TERMS OF PAYMENT
Unless otherwise stated, all payments shall be by Letter of Credit or Net Thirty (30) Days and in United States dollars, and a pro rata payment shall become due as each shipment is made. If shipment is delayed by Buyer, date of readiness for shipment shall be deemed to be date of shipment for payment purposes. If at any time in Company's judgment Buyer may be or may become unable or unwilling to meet the terms specified, Company may require satisfactory assurances or full or partial payment as a condition to commencing or continuing manufacture or making shipment, and may, if shipment has been made, recover the goods from the carrier, pending receipt of such assurances.

GOODS RETURN
Goods can be returned for credit only after receiving Company's written authorization and shipping instructions. Consignor's name and address must be plainly written on the shipping tag. Special goods fabricated to order are not returnable under any conditions.

PATENTS
Company shall pay costs and damages finally awarded in any suit against Buyer or its vendees to the extent based upon a finding that the design or construction of the goods as furnished, infringes a United States patent (except infringement occurring as a result of incorporating a design or modification at Buyer's request), provided that Buyer promptly notifies Company of any charge of infringement, and Company is given the right at its expense to settle such charge and to defend or control the defense of any suit based upon such charge. Company shall have no obligation hereunder with respect to claims, suits or proceedings, resulting from or related to, in whole or in part, (a) the use of software or software documentation, (b) compliance with Buyer's specifications, (c) the combination with, or modification of, the goods after delivery by Company, or (d) the use of the goods, or any part thereof, in the practice of a process. THIS ARTICLE SETS FORTH COMPANY'S ENTIRE LIABILITY WITH RESPECT TO PATENTS.

BUYER DATA
Timely performance is contingent upon the Buyer supplying to the Company, when needed, all required technical information, including drawing approval, and all required commercial documentation.

NUCLEAR
Buyer represents and warrants that the goods covered by this order shall not be used in or in connection with a nuclear facility or application.

PRICES
The prices stated herein will remain firm for the period up to the stated date of shipment providing the shipment is not delayed by the Buyer. If shipment is delayed by the Buyer beyond the shipment date quoted herein, the prices will be based on the prices in effect at time of shipment, including storage and material handling costs. In no event shall the adjusted price be less than the original order price, including change orders. Prices are F.O.B. Shipping Point, unless otherwise specified. When price includes transportation and other charges pertaining to the shipment of goods, any increase in transportation rates and other charges will be for the account of the Buyer. There will be an extra charge for any test other than that which may be normally run by the Company, or for any test performed to suit the convenience of the Buyer. Any applicable duties or sales, use, excise, value added or similar taxes will be added to the price and invoiced separately.

CONTROLLING PROVISIONS
These terms and conditions shall control with respect to any purchase order or sale of the Company's goods. No waiver, alteration or modification of these terms and conditions whether on Buyer's purchase order or otherwise shall be valid unless the waiver, alteration or modification is specifically accepted in writing and signed by an authorized representative the of Company.

EXPORT
If this transaction involves EXPORT, the following additional terms and conditions shall apply.

- Compliance is required for ALL applicable US export laws, and the export laws of the country from where the goods are exported.
- **PACKING**
  When packing is available, equipment will be packed, boxed or crated in accordance with the Company's standard commercial practice, for underdeck export shipment, unless otherwise agreed.
- **LETTER OF CREDIT**
  Unless otherwise specified in writing, payment shall be made by irrevocable letter of credit in form acceptable to Company, confirmed by a major USA bank, acceptable to the Company and providing for payment in full in United States dollars against presentation of United States inland shipping documents and invoices, such letter of credit to be established prior to Company's acceptance of the order. The letter of credit shall also provide that in the event Company is, for any reason beyond its control, prevented from making shipment from Company's factory or delivery at the port of embarkation, a certificate of manufacture of the whole or any part of the goods shall constitute delivery of such whole or any part of the goods and payment in full of any and all drafts drawn against the letter of credit for the goods so "delivered" shall be made upon presentation of such certificates of manufacture in lieu of United States inland shipping documents. In the event that Company is prevented by law, or otherwise, from making shipment from Company's factory or delivery at port of embarkation of the goods or any part thereof, on completion of manufacture, Company reserved the right to place the goods in storage for the Buyer's account and risk. Any charges incurred in this connection will be for the account of the Buyer at cost and will be payable upon demand. In regions where Letters of Credit are not available, surety bonds will be utilized in lieu of the bank guarantee.
- **COMPANY AS AGENT**
  If Company makes or arranges for ocean shipment, Company shall act as agent for the Buyer and reserves the right to procure full insurance coverage, including war risk insurance, at the expense of the Buyer. All expenses incurred in this connection will be payable upon demand to the Company. If Company as agent applies for or secures manufacturing, financing, exporting or other licenses required by the United States Government, or any department thereof, Company shall make such applications or secure such licenses solely as agent for the Buyer, and assumes no responsibility therefore.

CANCELLAT N SCHEDULE

Planned Shipment (weeks)	Elapsed Time - Date of Order to Date of Cancellation (weeks)																	
	0 to 2	2.01 to 4	4.01 to 6	6.01 to 8	8.01 to 12	12.01 to 16	16.01 to 20	20.01 to 24	24.01 to 28	28.01 to 32	32.01 to 36	36.01 to 40	40.01 to 44	44.01 to 48	48.01 to 52	52.01 to 56	56.01 to 60	60.01 64
Up to 8	20	50	75	100														
8.01 to 12	15	40	60	80	100													
12.01 to 16	10	25	45	60	85	100												
16.01 to 20	10	15	25	45	65	85	100											
20.01 to 24	10	10	20	25	50	70	90	100										
24.01 to 28	10	10	15	20	25	50	70	90	100									
28.01 to 32	10	10	10	15	20	35	60	75	90	100								
32.01 to 36	10	10	10	15	20	25	50	60	85	95	100							
36.01 to 40	10	10	10	10	15	25	50	60	70	85	95	100						
40.01 to 44	10	10	10	10	15	25	45	55	65	80	90	95	100					
44.01 to 48	10	10	10	10	15	25	45	55	60	65	80	90	95	100				
48.01 to 52	10	10	10	10	15	20	40	50	55	60	70	85	90	95	100			
52.01 to 56	10	10	10	10	15	20	35	50	55	60	70	80	85	90	95	100		
Above 56	*	*	*	*	*	*	*	*	*	*	*	*	*	*	*	*	*	*

Note: The above cancellation rates apply to manufactured equipment only. Any sub-supplier cancellation charges are not reflected above, and would apply accordingly.

* to Be Assigned

GRAHAM
ENGINEERING ANSWERS

PROPOSAL

TYPE OF EQUIPMENT	Surface Condenser with Steam Jet Air Ejector And Typical Accessories
PURCHASER	ATSI Engineering Services
PURCHASER'S INQUIRY	-
DESIGN STANDARDS	ASME Sec. VIII Div. 1 & HEI 10th Edition
ALTERNATES	20 °F cooling water rise

FOR FURTHER INFORMATION REGARDING THIS PROPOSAL
PLEASE CONTACT

REPRESENTATIVE

F J Guppenberger Inc
56 Harvester Ave
Box 436
Batavia, NY 14021-0436
phone: 585-343-0115
fax: 585-343-0121
email: guppinc@verizon.net

DATE July 23, 2009 Graham Reference EG 708 BU 09

GRAHAM CORPORATION
20 Florence Avenue, Batavia, NY 14020

Customer	ATSI Engineering Services		**Graham Corporation**
EG:	708 BU 09	Date	July 23, 2009
		Engr	AMC

TABLE OF CONTENTS

Customer	ATSI Engineering Services		**Graham Corporation**
EG:	708 BU 09	Date	July 23, 2009
		Engr	AMC

PRICING

Item Description: **Base Option (35 °F Cooling Water Rise)**	Price Each ex-works Factory US dollars
Surface Condenser Model 43 60 / 31.75 TALTD Steam jet ejector air removal model 2-4A2-128-2/3H plus indicated accessories as components, and domestic packaging.	$475,000.

Estimated Shipping Weight 49,300 lbs

Item Description: **Alternate Option (20 °F Cooling Water Rise)**	Price Each ex-works Factory US dollars
Surface Condenser Model 40 57 / 21.75 TALTD Steam jet ejector air removal model 2-4A2-128-2/3H plus indicated accessories as components, and domestic packaging.	$395,000.

Estimated Shipping Weight 35,000 lbs

OPTIONS OFFERED

Freight extra	Later
Price Extra for Tube Removal/Replacement Tools	$2,560 / set

VALIDITY

This quotation is budgetary. We will be happy to firm the pricing as the project progresses. Our firm quotations are valid for 14 days.

SCHEDULE

Drawings and data for approval will be submitted 8 - 10 weeks after receipt of an order. Return of approved drawings is required 2 weeks after submission
Shipment from our factory will be 32-34 weeks after receipt of approved drawings.

TERMS OF PAYMENT

Graham's offering is based on the following progress payments:

15% payable at time of order
10% payable upon submittal of general arrangement drawing for approval
40% payable on receipt of major materials at Graham as evidenced by tubing
35% payable net 30 days after date material is shipped or is reported ready for shipment

Final purchase order acceptance, including commercial and payment terms, is subject to review by our Sales and Credit Departments. The Buyer will be notified in writing of any changes in the commercial and/or payment terms for a mutually agreed upon contract between both parties.

This quotation is based on our General Conditions of Sale, Form GMC 1002-E Rev. No. 10 3/06.

Customer	ATSI Engineering Services		**Graham Corporation**
EG:	708 BU 09	Date	July 23, 2009
		Engr	AMC

COMMENTS, CLARIFICATIONS AND EXCEPTIONS TO THE SPECIFICATIONS TECHNICAL

The condensers offered have been designed with ¾" diameter, 18 BWG Admiralty tubes and naval rolled brass tubesheets. The purchaser is responsible for review of the materials quoted to determine their suitability for the site conditions.
Graham does not assume responsibility for selection or suitability of materials of construction as we do not maintain a metallurgical staff nor do we have as much familiarity with the site, service and environmental conditions, or preventative maintenance procedures and schedules as the Purchaser or their representatives.

As a clarification the condenser is designed with non-divided waterboxes. Divided waterbox design (operation with ½ tube bundle at reduced performance) is available as a price extra upon request.

Graham's scope of supply with regard to accessory items was not fully described in the specifications. Please be aware that scope of supply offered by Graham is as described by the Graham technical data sheets included in this proposal.

A steam duct with expansion joint has been provided. Since its height was not specified an overall height of 34" has been assumed. Any additions or deletions from length will result in an increase or decrease in price. Excessive forces, moments and long duct runs have not been considered and may result in a price addition. When available please advise us the required height and Graham will update our bid accordingly.

GRAHAM
ENGINEERING ANSWERS

SURFACE CONDENSER SPECIFICATION SHEET

Customer:	ATSI Engineering Services	EG.No.:	708 BU 09
Customer Ref:		Date:	07/23/09
Location		Engineer:	AMC/BU708A
Item:			

PERFORMANCE				
Absolute Pressure @ Steam Inlet	inHgA			4
Steam Condensed	lb/hr			110000
Heat Rejected	Btu/hr			104500000 104,500,000
Circulating Water Fresh Water	gpm			5971.4 6,000 GPM
Water Specific Gravity / Specific Heat	Btu/lb °F			1 / 1
Water Inlet / Outlet	°F			85 / 120 ✓
Water Pressure Loss:	ft H20 /psi			28.17 / 12.21
Percent Clean				85
Tube Velocity	ft/sec			7.99
DESIGN				
Model:				43 60 / 31.75 TALTD
Surface Area Total / Effective	sqft			8952.2 / 8845.3
Number Of Water Passes				2
Number Of Tubes				1436
Outside Diameter (in) - BWG				0.75 - 18 AW
Total Tube Length	ft			31.75
Design / Test Pressure	psig	Shell	FV&	15 / 19.5
		Tubes		75 / 97.5
Design Temperature	°F	Shell		250
		Tubes		150
Hotwell: bathtub Supply		min		1
Steam Inlet (rectangular) (in) (LB)				53. x 48.
Water Connections(in)				2 - 20
Condensate Outlet (in)				1 - 8

MATERIALS		
Shell	(SA-516-70)	Carbon Steel
Air Cooling Shrouds	(SA-516-70)	Carbon Steel
Tube Support Plates	(SA-516-70)	Carbon Steel
Tube Sheets	(SB-171-464)	Naval Rolled Brass
Tubes	(SB-111-443)	Admiralty
Water Boxes	(SA-516-70)	Carbon Steel
Water Box Covers	(SA-516-70)	Carbon Steel

REMARKS: DESIGN PER HEI TENTH EDITION

Construction and Stamp per ASME Sect. VIII , Div. 1

Steam Inlet Impingement Protection Included

Ejector Package Mounted on the Main Condenser

Customer to confirm turbine exhaust size and connection type

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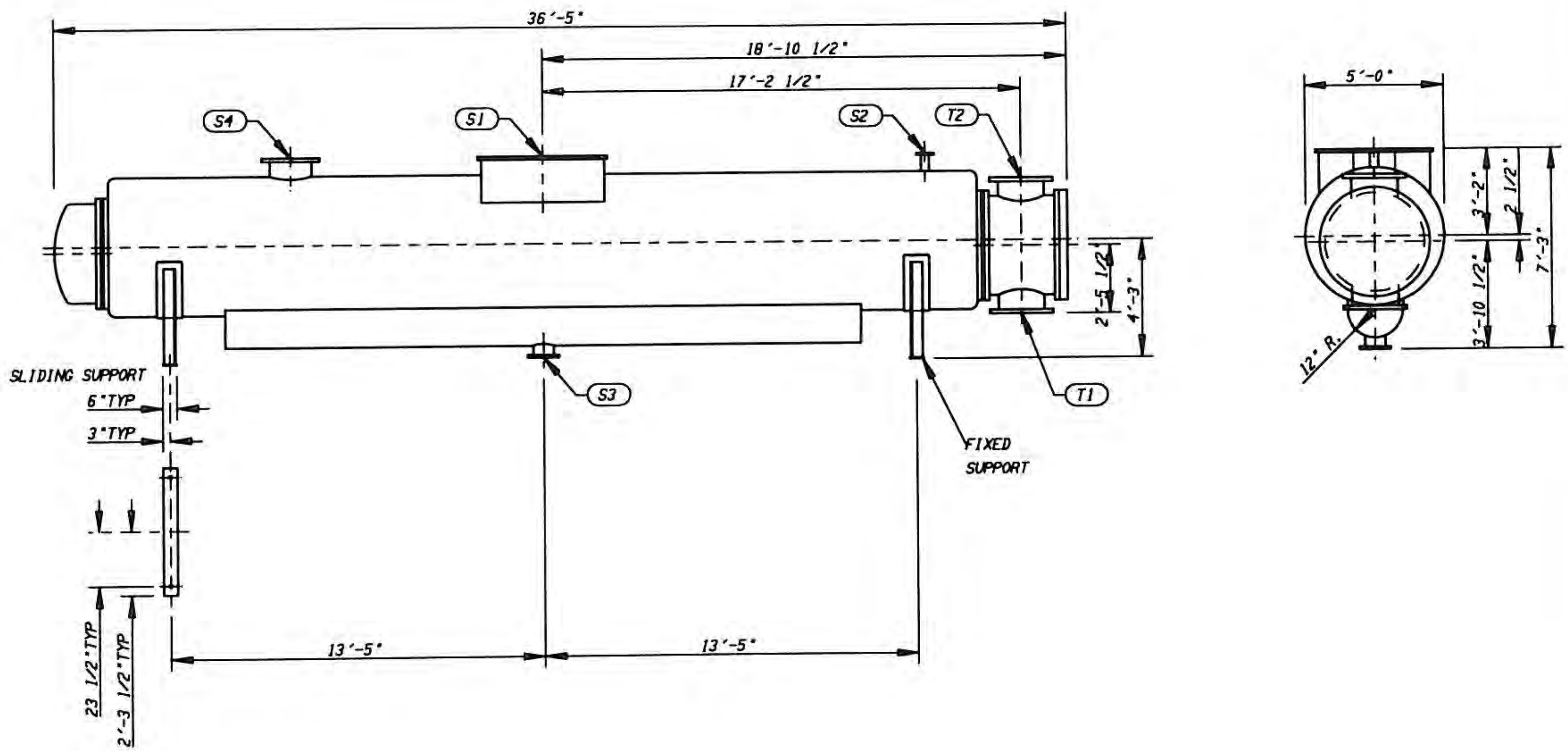
*** PRELIMINARY DRAWING NOT CERTIFIED FOR CONSTRUCTION ***



NOTE:
CONNECTION FLANGES ARE ANSI STANDARD DRILLING AND THICKNESS UNLESS OTHERWISE NOTED.
CUSTOMER TO SPECIFY COOLING WATER NOZZLE ARRANGEMENT.
SLIDING SUPPORT: (2) 1 1/8" x 2" SLOTTED HOLES.
FIXED SUPPORT: (2) 1-1/8" DIA. HOLES.

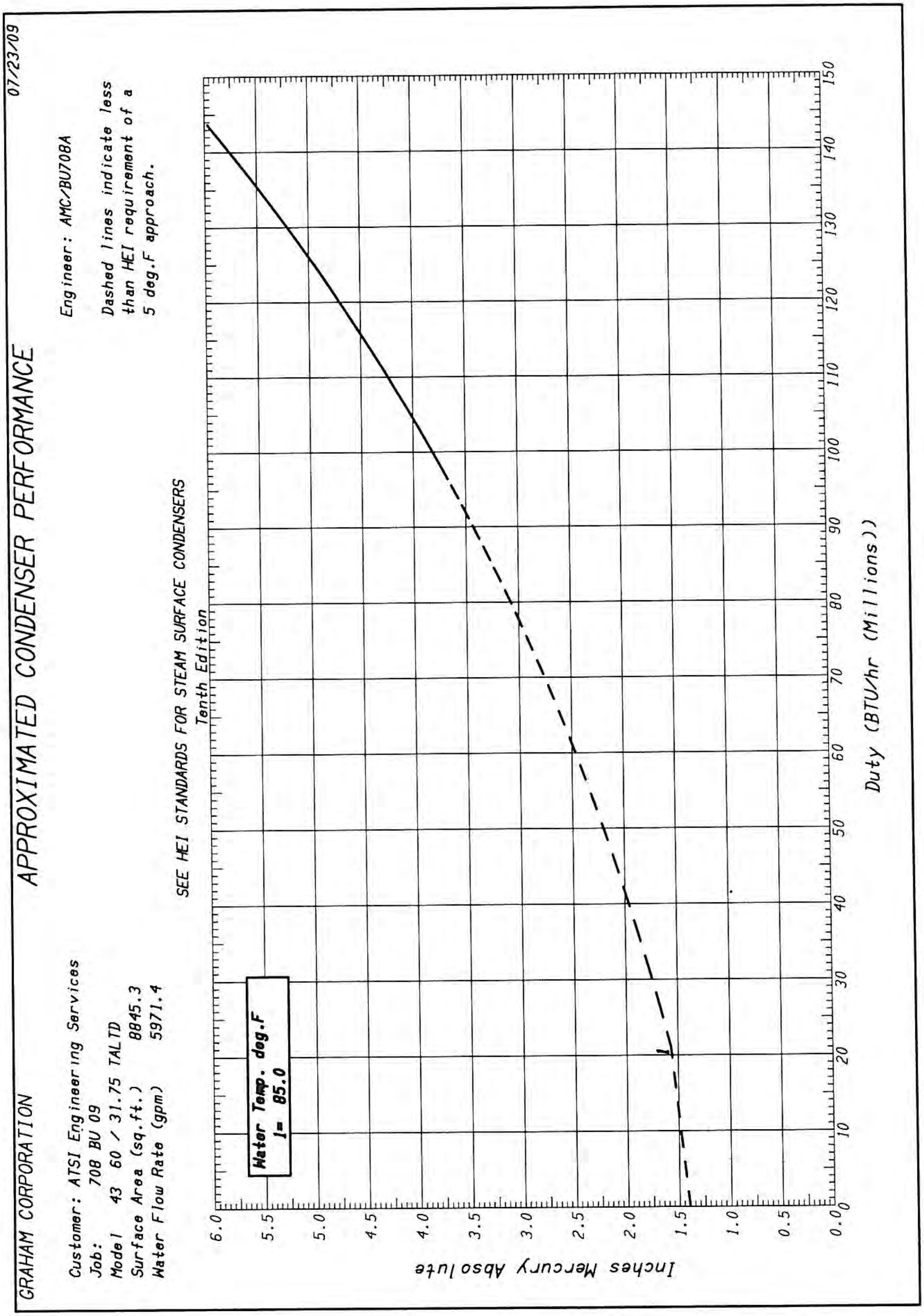
T1	20"	150# RF COOLING WATER INLET
T2	20"	150# RF COOLING WATER OUTLET
S1	53" x 48"	LB STEAM INLET
S2	3"	150# RF AIR OFF TAKE
S3	8"	150# RF CONDENSATE OUTLET
S4	18"	150# RF RELIEF VALVE

CUSTOMER	ATSI Engineering Ser		
CUSTOMER REF.			
MODEL	4360/31.75TALTD		
	M.A.W.P.	TEST PRESS.	DESIGN TEMP
SHELL SIDE	FV 15 PSIG	20 PSIG	250 F
TUBE SIDE	75 PSIG	98 PSIG	150 F

GRAHAM CORPORATION
20 FLORENCE AVE, BATAVIA N.Y.

PRELIMINARY SURFACE CONDENSER DRAWING

SCALE	MADE	CHKD	APPD	DATE	DWG. NO.	REV.
1/74	AMC			07/23/09	A708BU09-1	


GRAHAM CORPORATION
APPROXIMATED CONDENSER PERFORMANCE
07/23/09
Customer: ATSI Engineering Services
Job: 708 BU 09
Model 43 60 / 31.75 TALTD
Surface Area (sq.ft.) 8845.3
Water Flow Rate (gpm) 5971.4
Engineer: AMC/BU708A
Dashed lines indicate less than HEI requirement of a 5 deg.F approach.
SEE HEI STANDARDS FOR STEAM SURFACE CONDENSERS
Tenth Edition
Water Temp. deg.F
1= 85.0
Inches Mercury Absolute
Duty (BTU/hr (Millions))
0.0
0.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0
4.5
5.0
5.5
6.0
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150

GRAHAM
ENGINEERING ANSWERS

SURFACE CONDENSER SPECIFICATION SHEET

Customer:	ATSI Engineering Services	EG.No.:	708 BU 09
Customer Ref:		Date:	07/23/09
Location		Engineer:	AMC/BU708
Item:			

PERFORMANCE

Absolute Pressure @ Steam Inlet	inHgA	4
Steam Condensed	lb/hr	110000
Heat Rejected	Btu/hr	104500000
Circulating Water Fresh Water	gpm	10450
Water Specific Gravity / Specific Heat	Btu/lb °F	1 / 1
Water Inlet / Outlet	°F	85 / 105
Water Pressure Loss:	ft H20 /psi	11.08 / 4.8
Percent Clean		85
Tube Velocity	ft/sec	8

DESIGN

Model:				40 57 / 21.75 TALTD
Surface Area Total / Effective	sqft			5363.9 / 5257.1
Number Of Water Passes				1
Number Of Tubes				1256
Outside Diameter (in) - BWG				0.75 - 18 AW
Total Tube Length	ft			21.75
Design / Test Pressure	psig	Shell	FV&	15 / 19.5
		Tubes		75 / 97.5
Design Temperature	°F	Shell		250
		Tubes		150
Hotwell: bathtub	Supply	min		1
Steam Inlet (rectangular) (in) (LB)				53. x 48.
Water Connections(in)				2 - 24
Condensate Outlet (in)				1 - 8

MATERIALS

Shell	(SA-516-70)	Carbon Steel
Air Cooling Shrouds	(SA-516-70)	Carbon Steel
Tube Support Plates	(SA-516-70)	Carbon Steel
Tube Sheets	(SB-171-464)	Naval Rolled Brass
Tubes	(SB-111-443)	Admiralty
Water Boxes	(SA-516-70)	Carbon Steel
Water Box Covers	(SA-516-70)	Carbon Steel

REMARKS: DESIGN PER HEI TENTH EDITION

Construction and Stamp per ASME Sect. VIII , Div. 1
Steam Inlet Impingement Protection Included
Ejector Package Mounted on the Main Condenser
Customer to confirm turbine exhaust size and connection type

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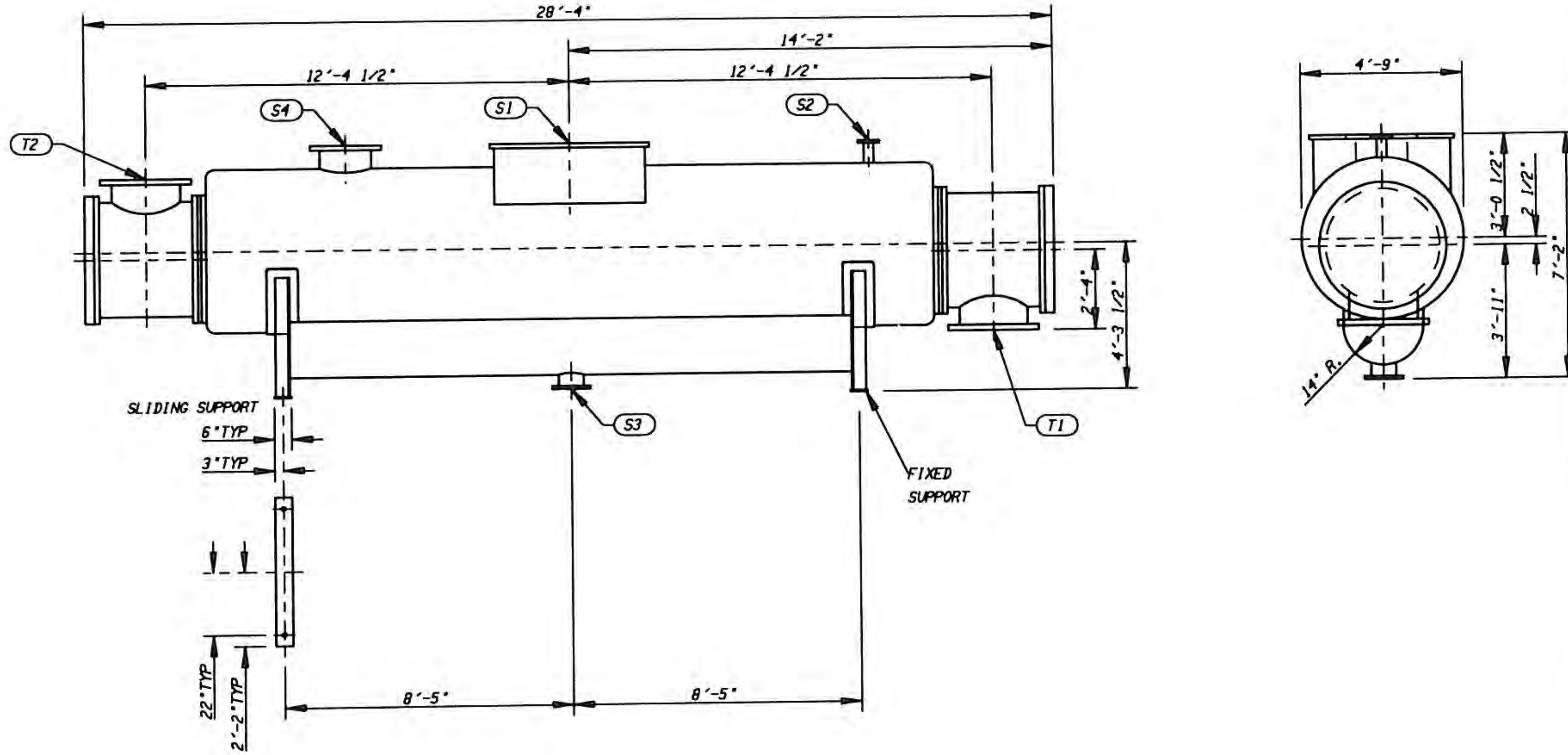
*** PRELIMINARY DRAWING NOT CERTIFIED FOR CONSTRUCTION ***



NOTE:
CONNECTION FLANGES ARE ANSI STANDARD DRILLING AND THICKNESS UNLESS OTHERWISE NOTED.
CUSTOMER TO SPECIFY COOLING WATER NOZZLE ARRANGEMENT.
SLIDING SUPPORT: (2) 1 1/8" x 1 3/4" SLOTTED HOLES.
FIXED SUPPORT: (2) 1-1/8" DIA. HOLES.

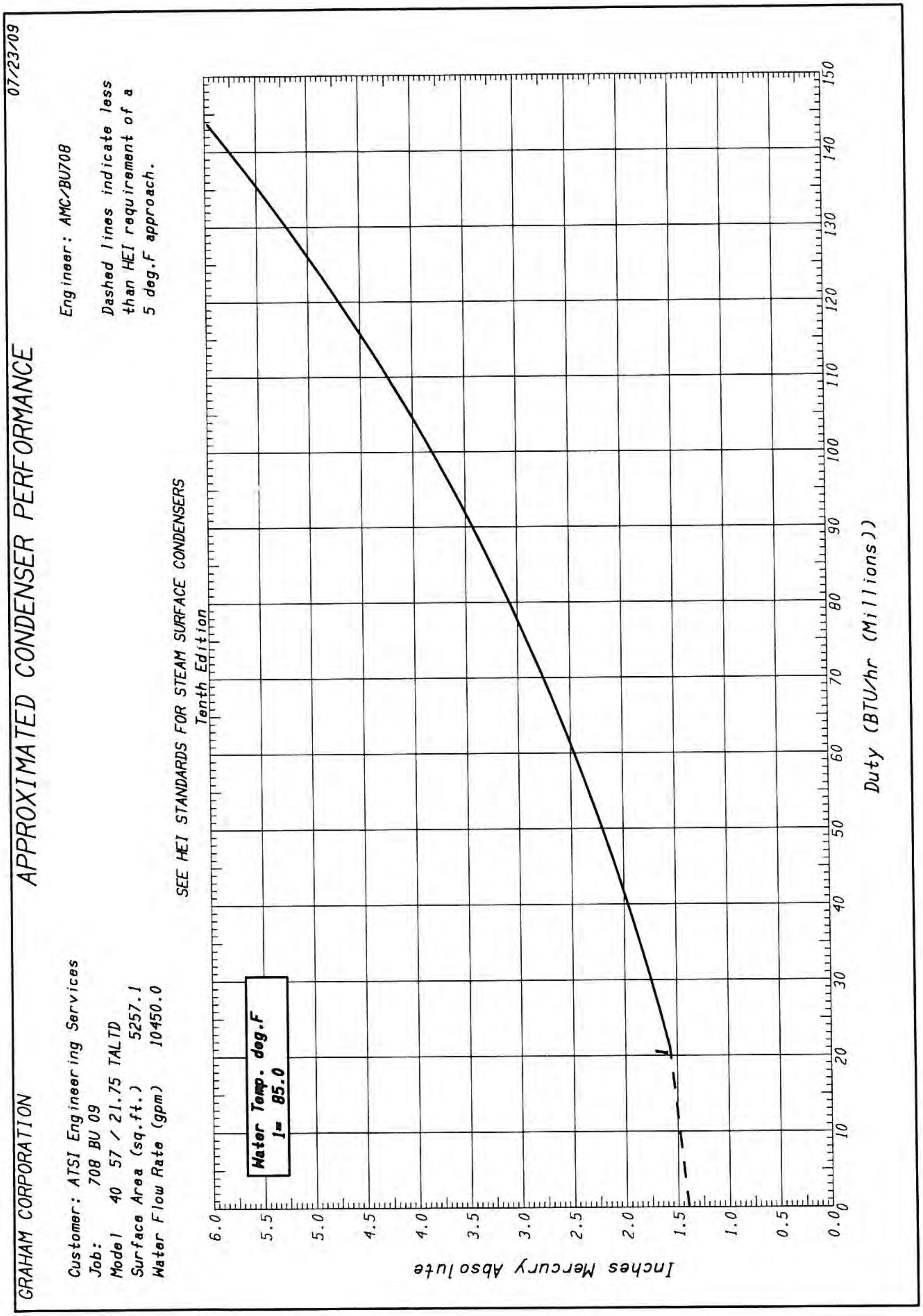
T1	24"	150# RF COOLING WATER INLET
T2	24"	150# RF COOLING WATER OUTLET
S1	53" x 48"	LB STEAM INLET
S2	3"	150# RF AIR OFF TAKE
S3	8"	150# RF CONDENSATE OUTLET
S4	18"	150# RF RELIEF VALVE

CUSTOMER	ATSI Engineering Ser		
CUSTOMER REF.			
MODEL	4057/21.75TALTD		
	M.A.W.P.	TEST PRESS.	DESIGN TEMP
SHELL SIDE	FV 15 PSIG	20 PSIG	250 F
TUBE SIDE	75 PSIG	98 PSIG	150 F

GRAHAM CORPORATION
20 FLORENCE AVE, BATAVIA N.Y.

PRELIMINARY SURFACE CONDENSER DRAWING

SCALE	MADE	CHKD	APPD	DATE	DWG. NO.	REV.
1/58	AMC			07/23/09	A708BU09-1	


GRAHAM CORPORATION
APPROXIMATED CONDENSER PERFORMANCE
07/23/09
Customer: ATSI Engineering Services
Job: 708 BU 09
Model 40 57 / 21.75 TALTD
Surface Area (sq.ft.) 5257.1
Water Flow Rate (gpm) 10450.0
Engineer: AMC/BU708
Dashed lines indicate less than HEI requirement of a 5 deg.F approach.
SEE HEI STANDARDS FOR STEAM SURFACE CONDENSERS
Tenth Edition
Water Temp. deg.F
1= 85.0
1
Inches Mercury Absolute
0.0
0.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0
4.5
5.0
5.5
6.0
Duty (BTU/hr (Millions))
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150

GRAHAM
ENGINEERING ANSWERS

CONDENSER AIR REMOVAL PACKAGE

Customer:	ATSI Engineering Services	EG.No.:	708 BU 09
Customer Ref:		Date:	07/23/09
Location		Engineer:	AMC
Item:			

STEAM JET EJECTOR PERFORMANCE

Pressure maintained	inHgA	1.0
Total Fluid Evacuated	lb/hr	108.1
Dry Air Evacuated	lb/hr	33.8
Motive Steam Required per Element	lb/hr	400.0
Operating Steam Pressure	psig	100.0
Operating Steam Temperature	°F	338.0
Inter Condenser Cooling Water Temp.	°F	CONDENSATE
Inter Condenser Cooling Water Required	gpm	CONDENSATE
Cooling water pressure drop thru I/A condenser	psi	1.7

STEAM JET EJECTOR DESIGN

Model Designation		2-4A2-128-2/3H		
Number of Stages		TWO		
Number Of Elements for Parallel Operation		TWO		
Material of Diffuser and suction chamber		SA-216-WCB		
Material of Steam Nozzles		316L		
Type of inter and after condenser		IN-LINE		
Material of Inter and After Condenser		SA-53-B		
Tube Sheets		SA-240-304L		
Tubes	0.7500 - 20 BWG AW	SA-249-TP304		
M. A. W. P / Test Pressure	psig	Shell	15 / 19	
		Tubes	75 / 97	
Design Temperature	°F	Shell	250	
		Tubes	150	

APPURTENANCES INCLUDED

Steam Strainer		Included
Interconnecting steam piping		Included
Air leakage meter		Included
Priming ejector - Size		3H (14C)
Steam consumption	lb/hr	842
Drainers or Traps		Included
Design per HEI Construction of I/AC per ASME Sec VIII Div I		
Isolation valve(s) at 1st stage suction and discharge		Included
Isolation valve(s) at 2nd stage suction		Included
Isolation valve at hogger suction		Included
Suction Manifold		Included
Hogging Ejector silencer		Included
Motive steam stop valve for each jet		Included
Pressure relief valve on each first stage ejector		Included

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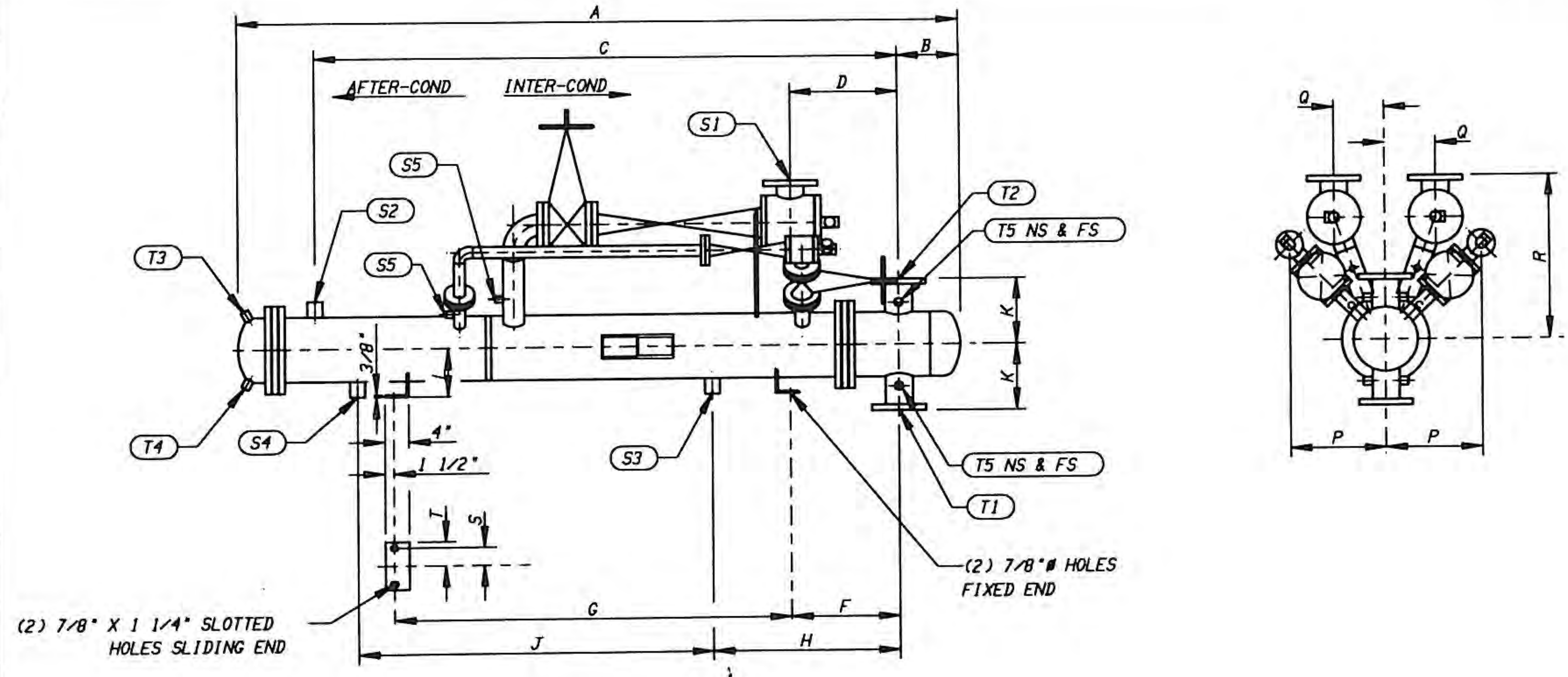


NOTE:
CONNECTION FLANGES ARE ANSI STANDARD DRILLING AND THICKNESS UNLESS OTHERWISE NOTED.

CUSTOMER ATSI Engineering Services
CUSTOMER REF.
MODEL 2-4A2-128-2

CONNECTION SCHEDULE S=SHELL SIDE T=TUBE SIDE

SYMBOL	SIZE	TYPE	SERVICE
S1	4"	150# ANSI (FF)	VAPOR INLET
S2	1 1/2"	NPT	VAPOR OUTLET
S3	1 1/2"	NPT	CONDENSATE OUTLET
S4	1 1/2"	NPT	CONDENSATE OUTLET
S5	1/2"	NPT	TEST CONN.
T1	4"	150# ANSI (RF)	WATER INLET
T2	4"	150# ANSI (RF)	WATER OUTLET
T3	3/4"	NPT	VENT
T4	3/4"	NPT	DRAIN
T5	3/4"	NPT	TEST CONN.

DIMENSIONS

A	B	C	D	F	G	H	J
10'-3"	12 3/16"	8'-1"	6 7/8"	18"	5'-6"	2'-7"	4'-11"
K	L	P	Q	R	S	T	
12"	9"	16"	8 1/2"	2'-3 1/2"	3 1/2"	4 1/2"	

	SHELL SIDE	TUBE SIDE
M.A.W.P (PSIG)	15	75
TEST PRESS (PSIG)	19	97
DESIGN TEMP (°F)	250	150

GRAHAM CORPORATION
20 FLORENCE AVE, BATAVIA N.Y.

PRELIMINARY EJECTOR PACKAGE

SCALE	MADE	CHKD	APPD	DATE	DWG. NO.	REV.
NONE	AMC			07/23/09	A708BU09-1	

GRAHAM
ENGINEERING ANSWERS

Customer:	ATSI Engineering Services	Page	1
Customer Ref:		EG.No.:	708 BU 09
Location		Date:	7/23/2009
Item:	Accessories	Engineer:	AMC/bu708

Vendors and model numbers are listed to illustrate construction features.
Graham reserves the right to substitute equipment of an equal type and quality by other vendors.
Any items omitted from this scope of supply list are excluded from this quotation at this time and will not be furnished

Pressure / Vacuum Indicator
(1) Ashcroft 1279, bronze bourdon tube

Pressure Indicator/ Motive steam inlet
(1) Ashcroft 1279, bronze bourdon tube

Pigtail Siphon-motive steam inlet
(1) Ashcroft 1098S, Steel

Temperature Indicator
(2) Ashcroft 50EI, SS, bi-metallic

Thermowell
(2) Ashcroft 75W-260C, 304 SS, 3/4" NPT

Level Gauge- surface cond. hotwell
(1) Conbraco #20-207, bronze valves, NPT, tubular

Air Leakage Meter-AC discharge
(1) Graham standard, brass/steel, NPT, orifice type

Pressure Relief Valve-1st stage jets
(2) Kunkle 910BEDM03-LE0015, 3/4" x 1 1/4" NPT 15 psig set pressure, Steel body

Drain Trap- inter condenser
(1) ITT-Hoffman FT015H-1.5", Float trap Max press = 175 psig, Max temp = 377 degF
CI body & cover w/ SS internals

Drain Trap-after condenser
(1) ITT-Hoffman FT015H-1", Float trap CI body & cover w/ SS internals
Max press = 250 psig, Max temp = 406 degF

Turbine to Condenser Ex.Joint & Duct
(1) 43" x 60" x 34" OAH, SS bellows w/liner,FLG x WE Drawing later

Level Controller
(1) Masoneilan 12000, Steel, 1.5" NPT, Pneumatic, 14" cage

Control Valve- Overboard
(1) Masoneilan 3"-35-35112, 150#, CS body with fail open actuator

Control Valve- Recycle
(1) Masoneilan 2"-35-35212, 150#, CS body with fail close actuator

GRAHAM
ENGINEERING ANSWERS

Customer:	ATSI Engineering Services	Page	2
Customer Ref:		EG.No.:	708 BU 09
Location		Date:	7/23/2009
Item:	Accessories	Engineer:	AMC/bu708

Vendors and model numbers are listed to illustrate construction features.
Graham reserves the right to substitute equipment of an equal type and quality by other vendors.
Any items omitted from this scope of supply list are excluded from this quotation at this time and will not be furnished.

Atmospheric Relief Valve

(1) Viking 18V, Vertical, 18"-150#(RF) conn's Carbon Steel construction

Silencer

(1) Vanec Model 412-03A-102, 2 1/2" inlet x 3" outlet, 150#(FF) conns, pressure drop = 0.44 psi, 85 dBA at 3 feet fits jets 8C-16C with 2.5" discharge

Customer	ATSI Engineering Services		**Graham Corporation**
EG:	708 BU 09	Date	July 23, 2009
		Engr	AMC

GENERAL COMMERCIAL COMMENTS

Payment – Quoted prices are net to Graham in USA funds and do not include any brokerage, export or import fees, duties or licenses, General Sales Tax, Value Added Tax, or other taxes beyond those normally applicable to Graham as an equipment manufacturer in Batavia, NY, USA. Customs clearance and all expenses are to the customer's account.

All payments are net 30 days after submittal of invoice

Release for Purchase - Release to purchase critical materials is required at time of receipt of an order. Equipment delivery is subject to availability of materials at time of order entry.

Documentation - Graham Corporation will provide engineering procedures and drawing documentation in electronic format. The drawings will be provided in PDF, DXF or TIF format and all other documentation in PDF format. An email address for submission of documentation is to be provided when a purchase commitment is made. Documentation can be provided in hardcopy format for a contract extra.

Return of approved drawings is required within 2 weeks after their submittal. Should return exceed 2 weeks after submittal, schedules will need to be reviewed to determine date for shipment.

Purchaser's Delays - Should the Purchaser cause Graham to delay processing of the order by fourteen or more days cumulatively, it is cause for possible price adjustments based on material escalation. If the delay extends to greater than ninety days cumulatively, the project will be considered canceled for Convenience of Buyer and appropriate cancellation charges will be applicable.

Should termination / cancellation become necessary, cancellation charges shall include costs of materials received, cancellation charges incurred for materials on order, costs for work completed or in progress, and normal overhead burden, administration charges and a reasonable profit on that portion of the work completed.

Graham has offered a fabrication cycle time and ship date based on current sub-supplier material leadtimes and shop loading for the quoted equipment. Due to material supply volatility, firm ship dates will be provided after release for manufacture. This is also based on return of approval drawings and release for manufacture in two (2) weeks, or the agreed upon time frame as reflected in the Purchase Order, after submission of the approval drawings. Failure to return the approval drawings and provide release for manufacture in this time frame will result in a review and possible change of the fabrication cycle time based on current sub-supplier material leadtimes and shop loading for the equipment. Graham will make every attempt to hold the cycle time, however, any change in the cycle time and ship date will be communicated by Graham after receipt of the approval drawings and release for manufacture. Any delay in shipment will not necessarily be day for day based on actual drawing approval return time.

Fabrication Location - On occasion to maintain schedule, Graham Corporation finds it necessary to complete certain fabrication in facilities outside of our own in Batavia, New York. Therefore, we request that permission be granted to use other facilities for these operations at time of order. Graham retains full responsibility for all work under contract with the purchaser.

Site Storage - The equipment is prepared suitable for normal 12 month site or 6 months outdoor storage. The equipment should be stored in a warm, dry, suitable environment away from on-going work and subjected to routine preventative maintenance inspections and procedures to maintain the equipment in its factory shipped condition. A price extra can be offered for a nitrogen purge and blanket, if requested. Additional long term storage procedures should be reviewed and determined by the Purchaser based upon the site environmental conditions and preventative maintenance procedures.

Customer ATSI Engineering Services
EG: 708 BU 09

Graham Corporation
Date July 23, 2009
Engr AMC

GENERAL CONDITIONS OF SALE

1 This offer to sell is expressly conditioned on Buyer's acceptance of all terms and conditions hereof, which shall take precedence over any inconsistent, contradictory or additional terms and conditions contained in any request for quotation, purchase order or other document furnished by Buyer in connection with this transaction whether such documents are exchanged simultaneously with this offer or prior or subsequent thereto, and Buyer's acceptance and receipt of the goods shipped hereunder shall constitute acceptance of such terms and conditions contained herein No acceptance by Seller shall be deemed contained herein except upon Buyer's express written consent to all terms and conditions set forth herein additional to or different from those of Buyer

All price and delivery quotations shall expire thirty (30) days from date thereof and in the meantime may be changed or withdrawn at any time

The beneficiary named on any purchase order or similar form furnished by Buyer should be "Graham Corporation" c/o the name and address of the local sales office through which Buyer's order is placed

2. SHIPPING DATE - FORCE MAJEURE: Shipment dates are from the date of receipt of Buyer's order with complete manufacturing information or from the date of approval of drawings, when required It is understood that Buyer will accept this equipment, at an earlier date if Seller is able to ship it sooner than such specified shipment date Seller may ship any portion of the equipment contingent upon good freight cost practices, as soon as it is completed and payment therefor shall be in accordance with agreed terms of payment If shipment is delayed at buyer's request or by reason of other causes beyond Seller's control, payment shall become due under the terms of payment from the date equipment is reported ready for shipment, and Buyer further agrees to pay appropriate storage charges in the event Seller is compelled to store the equipment Storage of such goods will be at Buyer's risk

Seller shall not be liable for any loss or damage for delay or non-delivery due to governmental acts or regulations or any civil or military authority, acts of Buyer or by reason of any force majeure, which shall be deemed to mean all other causes whatsoever not reasonably within the control of Seller, including but not limited to acts of God, war, riot or insurrection, blockades, embargoes, sabotage, epidemics, storms, floods, earthquakes, labor disputes, lockouts or other industrial disturbances, delays of carriers, interruption of power, and inability to secure materials Any delay resulting from any such cause shall extend shipping dates correspondingly Seller shall in no event be liable for any special, indirect or consequential damages arising from delay or non-delivery irrespective of the reason therefor, and receipt by Buyer shall constitute acceptance of goods and waiver of any claims due to delay

3. CANCELLATION OR TERMINATION: If Buyer shall cancel or terminate this order, such cancellation or termination shall only be upon written notice to Seller, and in such event, Buyer shall pay to Seller Seller's reasonable charges, including but not limited to, a quantity price adjustment for any goods delivered, and all other costs incurred and committed for by Seller, and Seller's pro rated profit thereon

4. SUSPENSION: If Seller's performance of the work is delayed for a period of more than six (6) months by reason of any cause set forth in paragraph two (2), above, upon removal of the cause of any such delay, performance shall be resumed, delivery rescheduled, and the purchase price shall be subject to any price increase in effect at the time of resumption of performance If Buyer is unwilling to accept such adjusted purchase price and such rescheduled delivery date, it shall cancel its order as provided in paragraph three (3) above

5. TERMS OF PAYMENT Unless otherwise specified, the equipment offered herein is quoted FOB Seller's plant The terms of payment are quoted in U S Funds, payable net 30 days after date material is shipped or is reported ready for shipment These terms are applicable to partial as well as complete shipments A 1½% SERVICE CHARGE per month will be applicable to outstanding balances past 30 days

If applicable, progress payments will be stated in the proposal

When in the opinion of Seller the financial conditions of Buyer renders it appropriate, Seller may require cash payment or satisfactory security before each shipment

6. WARRANTY AND LIABILITY LIMITATION: THE FOLLOWING IS IN LIEU OF ALL WARRANTIES OF SELLER EXPRESSED OR IMPLIED AND ALL IMPLIED WARRANTIES, INCLUDING BUT NOT LIMITED TO ANY IMPLIED WARRANTY OF MERCHANTABILITY OR FITNESS FOR A PARTICULAR PURPOSE, AND/OR ANY OTHER OBLIGATION ON THE PART OF THE SELLER ARE HEREBY EXCLUDED Seller, except as otherwise provided, warrants goods of its own manufacture against faulty workmanship or the use of defective materials, under normal use and service, and that such goods will conform to mutually agreed upon written specifications, drawings, and is guaranteed to meet specified performance requirements, for a period of twelve (12) months from date of shipment of the goods from the factory

Seller assumes no responsibility for deterioration of the equipment due to corrosion, erosion, or flow induced tube vibration, or for fouling, maintenance problems or any other causes not specifically covered under the foregoing warrant The sole remedy of Buyer with respect to any part not conforming to any warranty of Seller shall be the Repair or, at Seller's option, replacement of any defective part at the point of manufacture, Buyer assuming all costs of removal, shipping and reinstallation, provided that immediate written notice of the defect has been given to Seller, and Seller shall not be liable for any other expenses incurred because of failure of any part to meet Seller's warranty, nor for any special, indirect or consequential damages Material returned to Seller's factory without its written consent will not be accepted No back charges will be honored without Seller's advance approval of the work to be performed Seller's liability on any claim of any kind, including negligence, for any loss or damage arising out of, connected with, or resulting from this transaction, or the design, manufacture, sale, delivery, resale, installation, technical direction of installation, inspection, repair, operation, or use of any equipment covered by or furnished hereunder shall in no case exceed the price paid by Buyer for the equipment Seller also disclaims all liability, whether in contract, tort, warranty, or otherwise, to any party other than the Buyer

7. DRAWINGS AND DESIGN CHANGES: Proposal drawings submitted with Seller's offer are intended only to show the general style, arrangement and approximate dimensions of the equipment and are not certified for field installation Only when specifically requested by Buyer will Seller submit plans or certified drawings for Buyer's approval Shop detail drawings shall not be furnished under any circumstances since they are proprietary

Should Buyer request changes Seller shall have the option of adjusting contractual delivery dates and increasing original purchase price for design and material changes required to comply with Buyer's changes, however, Seller shall notify Buyer of such additional charge and schedule change prior to proceeding with the modification Should Buyer approve plans and/or drawings without change, Seller shall then proceed with fabrication of the equipment in accordance with such approval Should Buyer subsequently request changes after fabrication has commenced, Seller shall notify Buyer of schedule and cost impacts and upon approval, Buyer shall be responsible for Seller's additional charges

8 TITLE-RISK OF LOSS-FREIGHT-ROUTING: Unless otherwise agreed in writing, the equipment purchased hereunder shall be delivered F.O B Seller's place of manufacture Title, possession and risk of loss from any damage or casualty to the equipment, regardless of cause, shall be upon Seller until Seller has delivered the equipment to the carrier Buyer agrees that Seller shall retain and Buyer hereby grants to Seller a security interest in the equipment only until the purchase price has been paid and Buyer agrees to perform all acts necessary to perfect and assure Seller's security interest

In the event of any loss or damage or shortage in transit on a sale where it is expressly agreed in writing that Seller is responsible for the freight, and/or that the F.O B is other than Seller's place of manufacture, Buyer must make notation on the carrier's delivery receipt of said loss or damage or shortage, and make this document available to Seller and as further provided in paragraph 9

Buyer shall pay to Seller, in addition to the purchase price, any amount by which transportation charges may be increased, by reason of increased transportation rates, between dates of proposal and the actual shipping date

Seller may ship or route as Seller deems reasonable in the circumstances and is authorized to ship the goods by carrier Should premium transportation be required such as air or exclusive use of truck, Buyer agrees to reimburse Seller the difference between normal and premium transportation costs on all orders sold with freight included

9. SHORTAGES No claims for shortages, errors or breakage will be recognized by Seller unless made in writing within 30 days after receipt of goods at destination, accompanied by transportation bill with notation thereon

10. INSURANCE: No insurance coverage shall be provided by Seller unless by special agreement expressly consented to by Seller

11. TAXES: Any tax imposed by any present or future law on the sale of equipment described herein shall be added to the purchase price stated herein and is the responsibility of Buyer Seller is registered to collect sales tax in the States of California, Connecticut, Florida, Indiana, Michigan, Missouri, New Jersey, New York, North Carolina, Texas, and Washington and Buyer shall provide, if applicable, acceptable certification that it is exempt from such taxes, and in all other jurisdictions Buyer shall reimburse the proper tax authorities

12 WAIVER OR MODIFICATION: The terms and conditions stated herein constitute the entire agreement between the parties relating to this transaction and no addition to or modification of any provision hereof shall be binding upon Seller unless made in writing and signed by a duly authorized representative of Seller No waiver by Seller of any provision set forth herein shall constitute a waiver of any other provision

13. APPLICABLE LAW: The validity, performance and construction of any agreement between Buyer and Seller shall be governed by the laws of the State of New York

14. ESCALATOR CLAUSE: Form GMC-1002-E Rev No 3 Supplement 4/80 dated ______________ is attached hereto and is made part of the General Conditions of Sale Escalator is/is not applicable

15. CREDIT CARDS: The payment will be processed on the purchased date Note that all shipping and handling costs will be charged at that time

GOULDS PUMPS ESTABROOK CORP

August 3, 2009

WEIGHTS & VOLUMES

Proposal No: KM09-08-03 01
App.Engineer: Kevin Mckenzie

Item No	Qty	Model	Size	Bare Pump Weight	Cplg. Weight	Base. Weight	Driver Weight	Unit Net Weight	Unit Gross Weight	Total Gross Weight	Unit Net Volume	Unit Gross Volume	Total Gross Volume
ITEM 001	2	3196 -MTi	2x3-13	295 lb	8.2 lb	155.0 lb	270.0 lb	728.2 lb	845.2 lb	1690.4 lb	13.4 ft³	23.4 ft³	46.8 ft³
ITEM 002	3	3410 -M	8x10-14	996 lb	57.0 lb	580.0 lb	1,650.0 lb	3,283.0 lb	3,623.0 lb	10869.0 lb	44.5 ft³	68.0 ft³	204.0 ft³
	5									**12,559.4**			**250.8**

ITT

GOULDS PUMPS ESTABROOK CORP

August 3, 2009

CONSTRUCTION DETAILS - MATERIALS

Customer : ATSI Engineering
Proposal No: KM09-08-03 01
Inq.No.: Dresser Rand Pilot Plant Project
App.Engineer: Kevin Mckenzie

Item No	Product Service	Liquid Name	Qty	Model	Size	Casing Material	Impeller Material	Shaft Material	Sleeve material	Casing wear ring	Impeller wear ring	Seal Chamber. Mat.
ITEM 001		Water	2	3196 -MTi	2x3-13	Ductile iron	Ductile iron	SAE 4140	316SS			Ductile iron
ITEM 002		Water	3	3410 -M	8x10-14	Cast iron	Bronze	SAE 4140	Bronze	Bronze		

ITT

GOULDS PUMPS ESTABROOK CORP

ATSI Engineering

August 3, 2009

INQ NO: Dresser Rand Pilot Plant Project
Proposal No: KM09-08-03 01
Item No: ITEM 001

Attn: Kevin O'Connor

i-FRAME™

MODEL:3196 MTi SIZE:2x3-13 QTY: 2

PRICES in USD	
Pump Unit	6,924
Driver	1,287
Subtotal 2 Units	16,422
Boxing	
Testing	
Freight	
Accessories	
Total 2 Units	**16,422**

Operating conditions

SERVICE	
LIQUID	Water Temp. 125.0 deg F, SP.GR 1.000, Viscosity 1.000 cp, rated / max. suction pressure 0.0 / 0.0 psi g
CAPACITY Rated	250.0 gpm
HEAD	140.0 (ft)

Performance at 1770 RPM

PUBLISHED EFFY	59.0% (CDS)
RATED EFFY	58.0% with contract seal
RATED POWER	15.2 hp (incl. Mech. seal drag 0.22). (Run out 20.7 hp)
NPSHR	4.2 ft (available NPSH is 23.0 ft)
DISCH PRESSURE(R)	61.4 psi g (66.3 psi g @ Shut off) Based on 0.0 psi g Suc.press
PERF. CURVE	3832-4 (Rotation CW viewed from coupling end)
SHUT OFF HEAD	153.1 ft
MIN. FLOW	Continuous Stable: 58.8 gpm Hydraulic: 58.8 gpm Thermal: N/A

Materials

CONSTRUCTION	Ductile iron
CASING	Ductile iron (max.casing.pres. @ rated temp. 245.6 psi g)
ST.BOX COVER	Ductile iron
IMPELLER	Ductile iron - Open (12.1250 in rated, max=13.0000 in, min=9.0000 in)
CASING GASKETS	Aramid Fiber with EPDM Rubber
IMPELLER O-RING	Teflon
SHAFT MATERIAL	SAE 4140
SHAFT SLEEVE	316SS
LUBRICATION	Flood oil
SEAL CHAMBER	Taper bore plus with axial ribs
GLAND	316SS Flush high performance
GLAND O-RING	Teflon jacketed O-ring gasket
BEARINGS	6309 (Inboard) 3309 (Outboard)
COUPLING	Rexnord - Omega Rex Elastomer- ES-5 (standard orange element)-S.F. 1.00
COUPLING GUARD	Carbon steel
BASEPLATE	Cast iron camber top D00057A

Sealing Method

MECHANICAL SEAL	John Crane 8-1T X(1)F(51)1XO(58)1 (316) (Carbon vs Silicon Carbide) - (Conventional - Single)

Flanges
150# flat face

Liquid end features
Impeller balance holes
Impeller balanced to ISO 1940 G6.3

Frame Connections
Bearing frame drain
Bottle oiler connection
Frame cooler access
Oil fill connection

Frame features
Condition Monitor
Ductile iron frame adapter
Inpro VBXX-D Hybrid Bearing Isolators
Premium Severe Duty Thrust Bearings

Testing
Non witnessed casing hydrostatic-test

Painting
Goulds Blue standard painting

Warranty
5 Year Extended Warranty (All the components, manufactured by ITT Goulds pumps, in the liquid end and power end are covered).

Noise level Data
Maximum predicted sound pressures level pump only in Decibels (db) Re 0.0002 microbars measured 3ft horizontally and 5ft from the floor per QCP 580

Noise Level	31.5	63	125	250	500	1k	2k	4k	8k	A
Pump	70.0	72.0	72.0	71.0	73.0	74.0	73.0	72.0	71.0	79.5

Driver : Electric motor Manufacturer : Pump mfg`s Choice

FURNISHED BY	Pump mfg	MOUNTED BY	Pump mfg
RATING	20.0 hp (14.9 KW)	ENCLOSURE	Severe Duty/Mill and Chemical - Epact Efficient
PHASE/FREQ/VOLTS	3/60 Hz/230/460	SPEED	1800 RPM
INSULATION/SF	F/1.15	FRAME	256T

Weights and Measurements

TOTAL NET UNIT WEIGHT / VOLUME	728.2 lb / 13.4 ft³
TOTAL GROSS UNIT WEIGHT / GROSS VOLUME	845.2 lb / 23.4 ft³

Program Version 1.30.0.0

Our offer does not include specific review and incorporation of any Statutory or Regulatory Requirements and the offer is limited to the requirements of the design specifications. Should any Statutory or Regulatory requirements need to be reviewed and incorporated then the Customer is responsible to identify those and provide copies for review and revision of our offer.

Our quotation is offered in accordance with our conditions of Sale.

Click here to download the pump Bulletin

Click here to learn more about the new *i-FRAME*™

PUMPSMART FLOW ECONOMY ESTIMATES

FIXED SPEED		PUMPSMART
17.3 gpm/kW		**24.0** gpm/kW
Expected range for typical operation 13.3 to 20.8 gpm/kW		Expected range for typical operation 20.3 to 26.2 gpm/kW

Click Here To Learn More!

Estimated Annual Savings 1,766 USD

Are you aware of PumpSmart Process Systems?

PumpSmart is a system that utilizes a standard process pump in conjuction with ITT 's unique and patented PumpSmart Control System and Software. The software, which resides on the controller microprocessor chip, allows the pump to monitor and react to any system condition.

PumpSmart

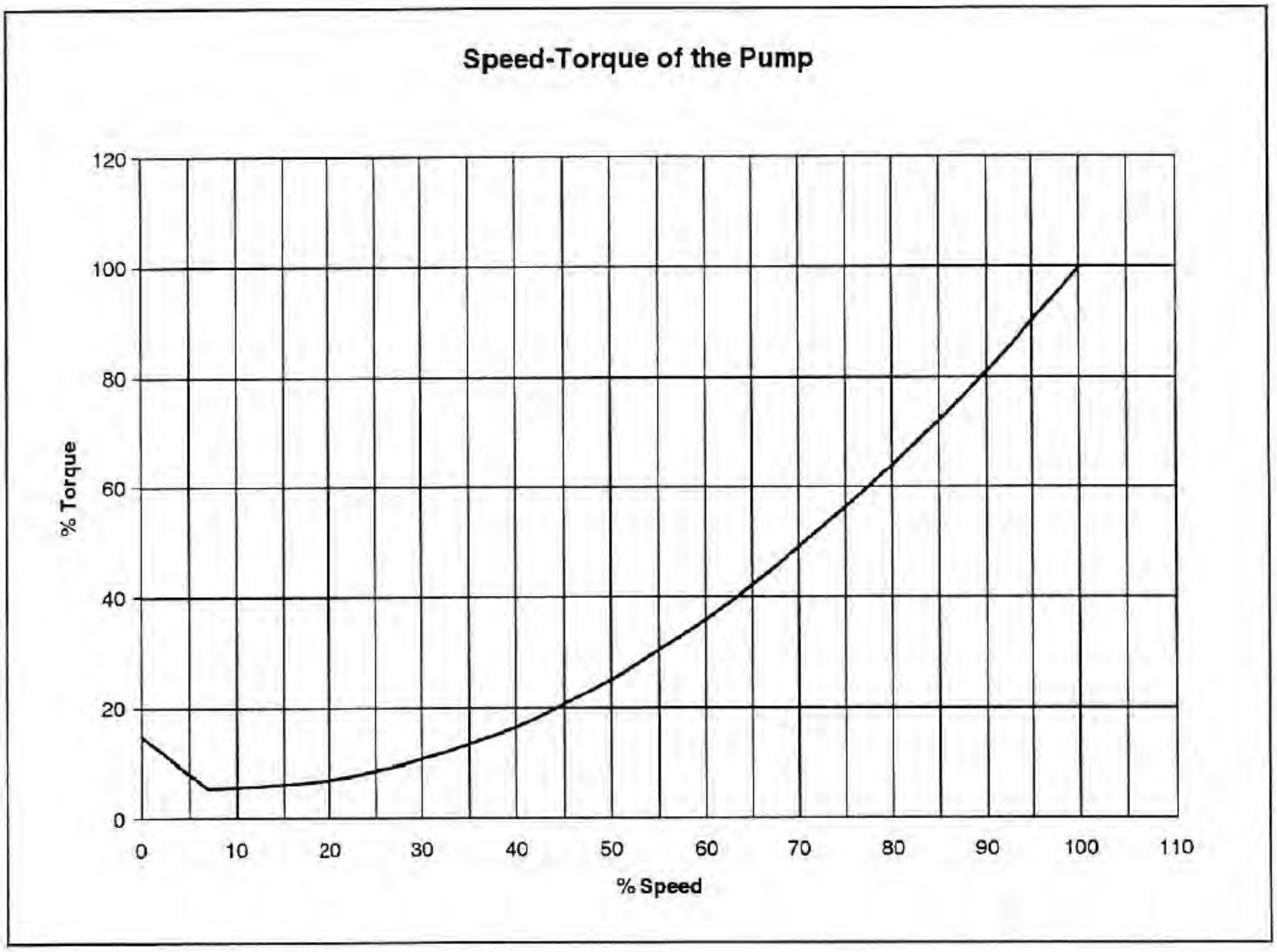
- Eliminates control valves, flow meters, and recirculation lines.
- Significantly reduces energy costs.
- Significantly increases MTBF.

Please contact your local Goulds Pumps representative for details and a demonstration CD-ROM. You may also contact us at www.gouldspumps.com or e-mail pumpsmart@fluids.ittind.com.

GOULDS PUMPS

Speed - Torque Curve

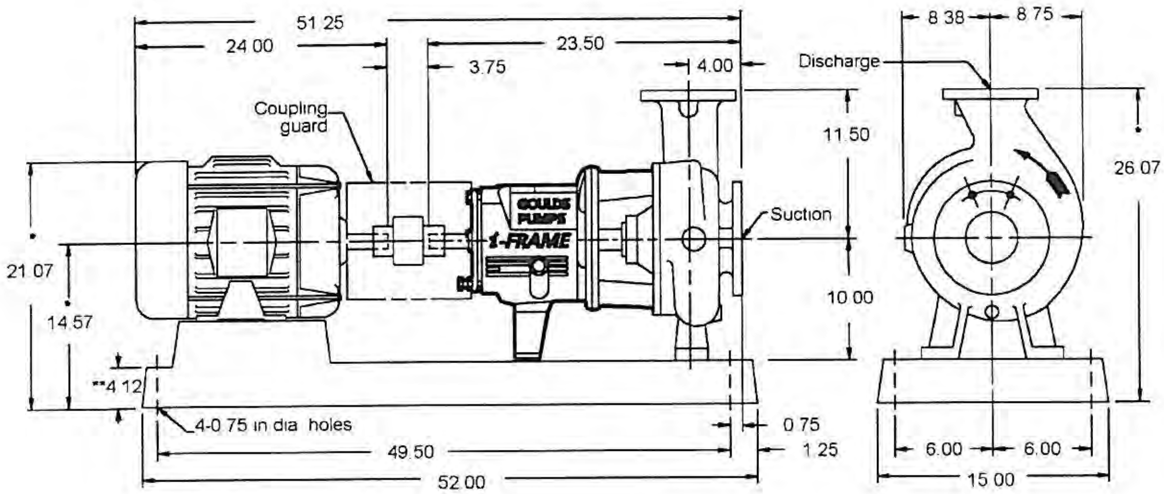
S.O. No.	
Customer :	ATSI Engineering
P.O. No. :	
Proposal No :	KM09-08-03 01
Item No. :	ITEM 001 3196 / MTi / 2x3-13
Speed :	1770 in rpm.
100% Torque :	25.70 in Feet*lb force At Shut off (closed valve)
100% Torque :	61.30 in Feet*lb force At Run out (open valve)
100% Torque :	45.10 in Feet*lb force At Rated Point
WR2 of pump :	1.4 in [Lbs Ft²]



Size: **2x3-13** Date: 08/03/2009

Rev. No: **0**

GOULDS PUMPS **OUTLINE DRAWING** **Model 3196 MTi** **Size 2x3-13** ITT



Pump specification

SUCT.FLANGE SIZE *3"*	DRILLING *ANSI 150#*	FACING *FF*	FINISH *SERRATED*
DISCH.FLANGE SIZE *2"*	DRILLING *ANSI 150#*	FACING *FF*	FINISH *SERRATED*
PUMP ROTATION (LOOKING AT PUMP FROM MOTOR) *CW*			
TYPE OF LUBRICATION *FLOOD OIL*			COOLED *NO*
TYPE OF STUFFING BOX *TAPER BORE PLUS WITH AXIAL RIBS*			COOLED *NO*
TYPE OF SEALING *MECHANICAL SEAL*			

Weights and Measurements

PUMP	295.0 lb
MOTOR/CPLG	270.0/8.2 lb
BASEPLATE	155.0 lb
TOTAL	728.2 lb
GR.VOLUME w/BOX	23.4 ft³
GR.WEIGHT w/BOX	845.2 lb

Motor specification

MOTOR BY *PUMP MFG*	MOUNT BY *PUMP MFG*	MFG. *PUMP MFG'S CHOICE*
FRAME *256T*	POWER *20.0 hp*	RPM *1800*
PHASE *3*	FREQUENCY *60 HZ*	VOLTS *230/460*
INSULATION *F*	S.F. *1.15*	
ENCLOSURE *SEVERE DUTY/MILL AND CHEMICAL - EPACT EFFICIENT*		

Notes and References

- MTR DIMENSIONS ARE APPROXIMATE
- INSTALL FOUNDATION BOLTS IN PIPE SLEEVES
- ALLOW FROM 0.75 to 1.50in. FOR GROUTING. SEE INSTRUCTION BOOK FOR DETAILS.

*Tolerance is +0 -0.5

** Foundation bolt grip thickness

FOR PUMP TAPPED OPENINGS REFER TO DWG.. TKM09-08-03 01 / ITEM 001

Auxiliary specification

COUPLING BY *PUMP MFG*	CPLG TYPE *REXNORD OMEGA REX ELASTOMER- ES-5 (STANDARD ORANGE ELEMENT)*
CPL GUARD BY *PUMP MFG.*	CPLG GUARD MATL *CARBON STEEL*
BASEPLATE *CAST IRON CAMBER TOP D00057A*	
MECH.SEAL *JOHN CRANE 8-1T X(1)F(51)1XO(58)1 (316) (CARBON VS SILICON CARBIDE)*	

DRAWING IS FOR REFERENCE ONLY.
NOT CERTIFIED FOR CONSTRUCTION UNLESS SIGNED.

Customer: ATSI Engineering
Serial No:
Customer P.O. No:
Item No: ITEM 001
End User: DRESSER RAND OLEAN
Service:

All dimensions are in inches.
Drawing is not to scale
Weights (lbs) are approximate

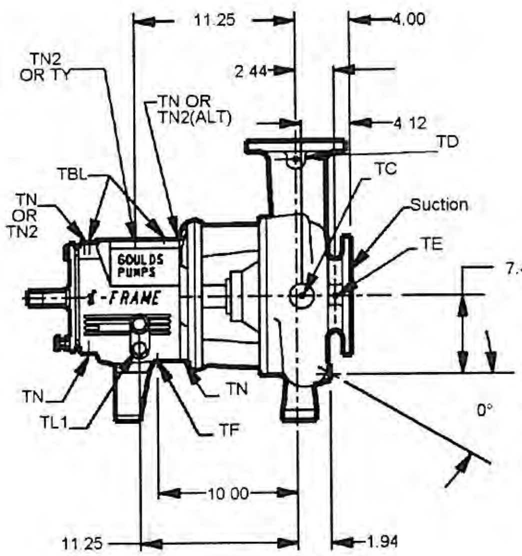
DRAWING NO KM09-08-03 01/ITEM 001

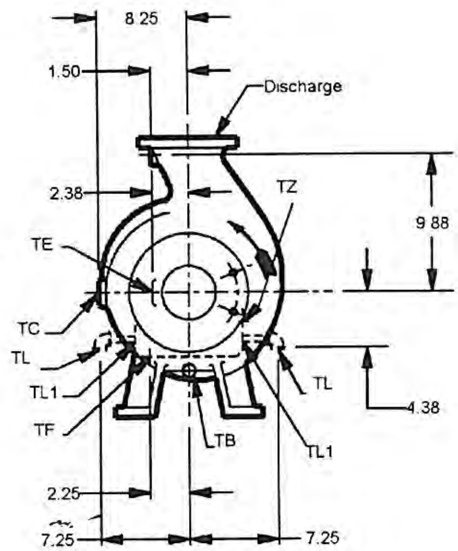
FORM # ED0048 Program Version 1.30.0.0

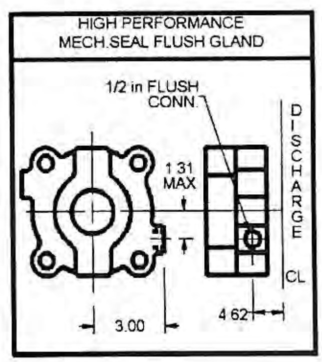
GOULDS PUMPS **TAPPED OPENINGS** **Model 3196 MTi Size 2x3-13** ITT

TAPPED OPENINGS MODEL 3196 MTi 2x3-13 ANSI NO. A30

NO.	SIZE	QTY.	PURPOSE	FURNISHED YES/NO	NO.	SIZE	QTY.	PURPOSE	FURNISHED YES/NO
TB	1/2	1	CASING DRAIN	NO	TL1	---	2	FRAME COOLER ACCESS	YES
TC	1/2	1	BY-PASS CONNECTION	NO	TN	1/4	4	GREASE FITTING	NO
TD	3/8	1	DISCH. GAUGE CONNECTION	NO	TN2	1/4	2	OIL MIST INJECTION PORT	NO
TE	3/8	1	SUCTION GAUGE CONNECTION	NO	TY	3/4	1	OIL FILL	YES
TF	1/2	1	BEARING FRAME DRAIN	YES	TZ	1/4	1	BOTTLE OILER CONNECTION	NO
TL	1/2	2	FRAME COOLING CONNECTION	NO	TBL	1/4	2	VIB./TEMP. CONNECTION	YES







DRAWING IS FOR REFERENCE ONLY.

NOT CERTIFIED FOR CONSTRUCTION UNLESS SIGNED.

Customer: ATSI Engineering
Serial No:
Customer P.O. No:
Item No: ITEM 001
End User: DRESSER RAND OLEAN
Service:

All dimensions are in inches.

Drawing is not to scale

DRAWING NO KM09-08-03 01/ITEM 001

FORM # ED0274/5 +87 Program Version 1.30.0.0

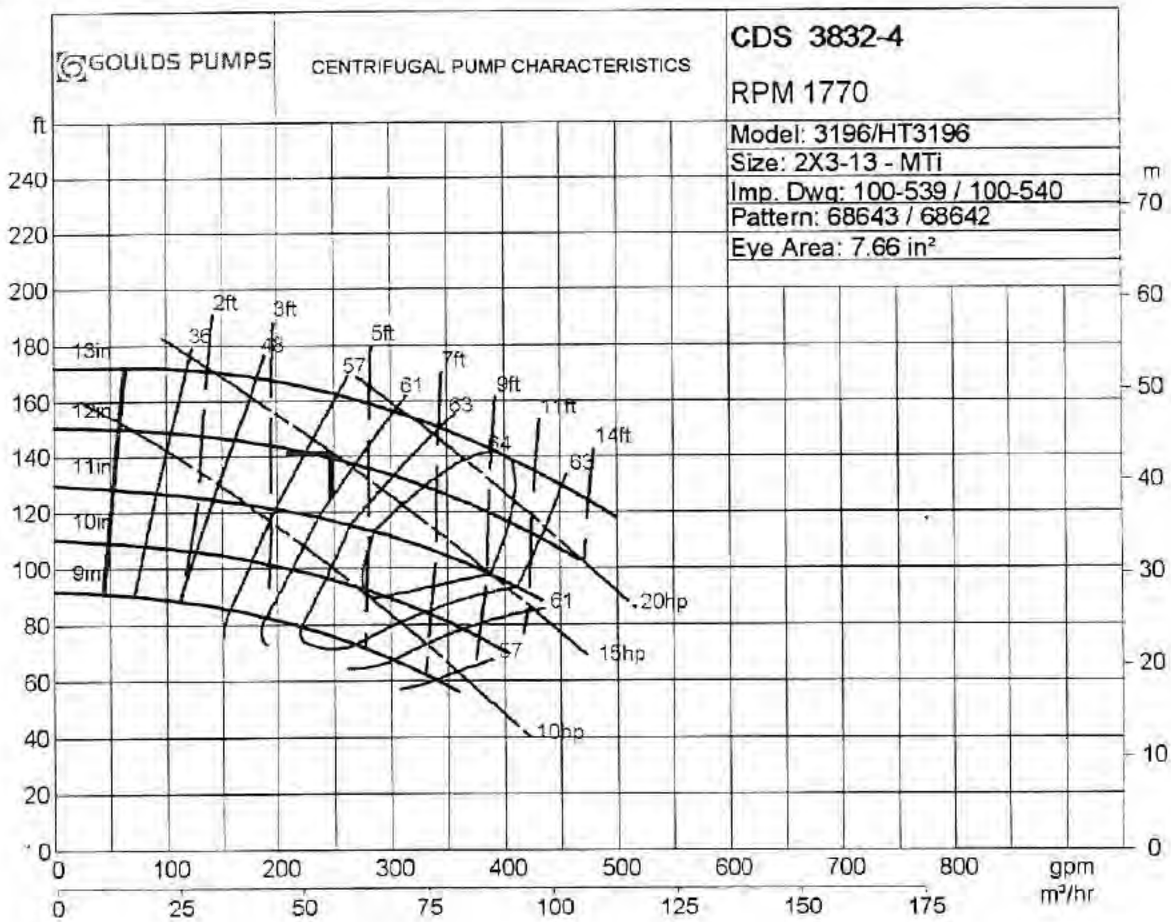
Model: 3196	Size: 2x3-13	Group: MTi	60Hz	RPM: 1770	Stages: 1

Job/Inq.No. :	Dresser Rand Pilot Plant Project				
Purchaser :	ATSI Engineering				
End User :	DRESSER RAND OLEAN	Issued by :	Kevin Mckenzie		
Item/Equip.No. :	ITEM 001	Quotation No. :	KM09-08-03 01	Date :	08/03/2009
Service :					
Order No. :		Certified By :		Rev. :	0

Operating Conditions / **Pump Performance**

Liquid:	Water	Published Efficiency:	59.0 %	Suction Specific Speed:	6,330 gpm(US) ft
Temp.:	125.0 deg F	Rated Pump Efficiency:	58.0 %	Min. Hydraulic Flow:	58.8 gpm
S.G./Visc.:	1.000/1.000 cp	Rated Total Power:	15.2 hp	Min. Thermal Flow:	N/A
Flow:	250.0 gpm	Non-Overloading Power:	20.7 hp		
TDH:	140.0 ft	Imp. Dia. First 1 Stg(s):	12.1250 in		
NPSHa:	23.0 ft	NPSHr:	4.2 ft		
Solid size:		Shut off Head:	153.1 ft		
% Susp. Solids (by wtg):		Vapor Press:			
Max. Solids Size:	0.3750 in				

Notes: 1. Power and efficiency Losses are not reflected on the curve below.
2. Elevated temperature effects on performance are not included.



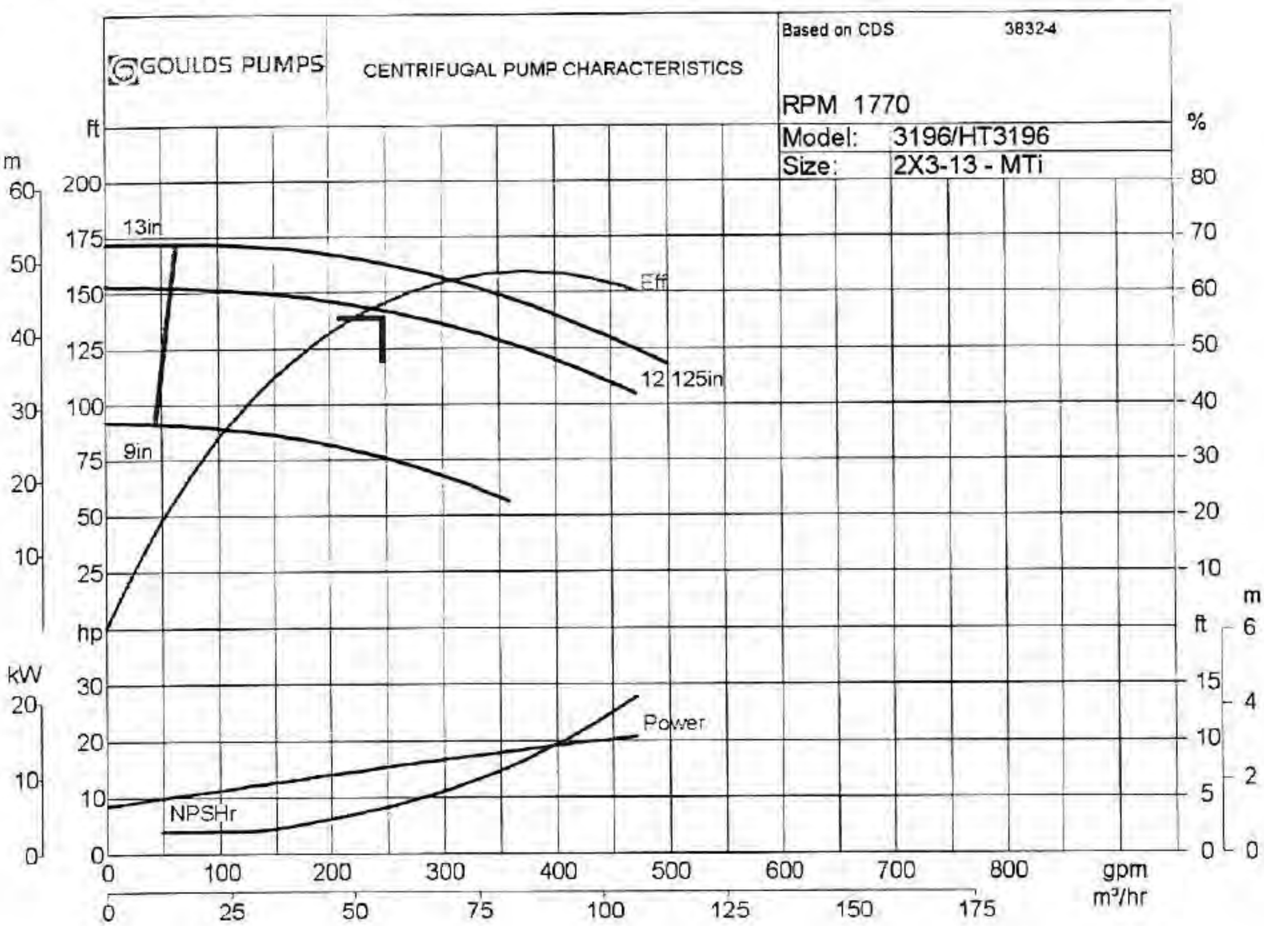
Model: 3196	Size: 2x3-13	Group: MTi	60Hz	RPM: 1770	Stages: 1

Job/Inq.No. : Dresser Rand Pilot Plant Project
Purchaser : ATSI Engineering
End User : DRESSER RAND OLEAN
Rev. : 0
Issued by : Kevin Mckenzie
Item/Equip.No. : ITEM 001
Quotation No. : KM09-08-03 01
Date : 08/03/2009
Service :
Order No. :
Certified By :

Operating Conditions		**Pump Performance**			
Liquid:	Water	Published Efficiency:	59.0 %	Suction Specific Speed:	6,330 gpm(US) ft
Temp.:	125.0 deg F	Rated Pump Efficiency:	58.0 %	Min. Hydraulic Flow:	58.8 gpm
S.G./Visc.:	1.000/1.000 cp	Rated Total Power:	15.2 hp	Min. Thermal Flow:	N/A
Flow:	250.0 gpm	Non-Overloading Power:	20.7 hp		
TDH:	140.0 ft	Imp. Dia. First 1 Stg(s):	12.1250 in		
NPSHa:	23.0 ft	NPSHr:	4.2 ft		
Solid size:		Shut off Head:	153.1 ft		
% Susp. Solids (by wtg):		Vapor Press:			

Max. Solids Size: 0.3750 in

Notes: 1. Elevated temperature effects on performance are not included.



SPX
COOLING TECHNOLOGIES

Johnston Equipment Company
39 Saginaw Drive
ROCHESTER, NY 14623-3146
585 244 3338 /
email: guyward@jecoroc.com

Proposal to:
Kevin O'Connor
ATSI, Inc.
415 Commerce Drive
Amherst, NY 14228

Project:
Dresser-Rand
6000 GPM CT RFQ Rv1
Olean, NY

Engineer:
ATSI, Inc.
Amherst, NY

Opportunity / Quote No. (Ver): GUY WARD_090730_133923952 / GUY WARD_090730_134020071 (1)
Rep Quote No.:

July 30, 2009

Marley NC8400 Tower

TOWER MODEL	PERFORMANCE CONDITIONS	MOTOR DATA	TOWER DIMENSIONS	WEIGHTS
Quantity of (1) Marley NC Class model **NC8413WAS** factory assembled 3-Cell crossflow cooling tower	Per 3-cell tower: 6,000 gpm 120.0 °F Hot Water 85.0 °F Cold Water 78.0 °F Entering WB	NEMA 75 HP 1 speed / 1 wind 3 phase / 60 Hz / **230/460v** 1.15sf / TEFC 1800 RPM Premium Efficiency Inverter duty nameplated	Each cell: (without options) Length 11' - 10 3/4" Width 22' - 5" Height 27' - 1 1/8" Per 3-cell tower: (with options) Length 44' - 7 3/4" Width 26' - 5" Height 27' - 1 1/8"	Per cell: Shipping: 22,122 lb Operating: 46,142 lb Per 3-cell tower: Shipping: 66,366 lb Operating: 138,426 lb

Quantities shown below are per tower.

Base Tower Construction/Equipment:

Galvanized Steel casing.
Galvanized Steel structure.
Stainless Steel collection basin.
Galvanized Steel distribution basin.
All stainless steel is series 300.
Anchorage design selected to meet customer specified design requirements for wind load of 30.0 psf.
Low Sound fan with aluminum blades.
Marley designed Geareducer® with 5-year warranty.
15 mil PVC film fill with integral louvers and drift eliminators designed and manufactured by Marley.
Drift rate guaranteed to be no greater than .005% of the design flow rate.
CTI certification per STD-201.
Factory Mutual Approval
For multi-cell towers Purchaser must ensure that the number of cells includes consideration of cell outage due to fire damage, mechanical failure and preventive maintenance such that sufficient cooling capacity is available to enable normal business and manufacturing operations to continue throughout the year.
Fiberglass fan stack.

Collection Basin Connections and Accessories:

(3) 10 in (254 mm) diameter depressed side sump outlet(s) with trash screen(s) and anti-vortex plate(s).
Stainless steel interconnecting flume(s) for water flow and equalization between cells
Interconnecting flume weir plate(s) for cell isolation
4 in (102 mm) diameter combination drain and overflow in each cell
(3) 2 in (50.8 mm) water make-up float valve(s)
24 kW per cell 480/3 volt/phase electric immersion heater for freeze protection of the collection basin during cold weather system shutdown
Includes heater elements, water temperature sensor probe and control box
Heater system disconnect switch

Distribution Basin Inlet and Accessories:

(2) 10 in (254 mm) diameter top inlet connections per cell.
(2) 10 in (254 mm) horizontal flow control valves per cell.

Maintenance & Maintenance Access Features:

Tower is designed in accordance with OSHA safety standards.
This quotation includes features that will allow safe access on the fan deck while the fan is still operating.
External lube line with dipstick
Full face horizontally mounted air inlet screens for easy access to collection basin

Johnston Equipment Company
39 Saginaw Drive
ROCHESTER, NY 14623-3146
585 244 3336 /
email: guyward@jecoroc.com

Convenient access to the collection basin and plenum area is provided via a large access door located on each endwall
(2) Access door platforms
Stainless Steel plenum walkway in each cell
Internal mechanical equipment access platform in each cell
Easy fitting perimeter guardrail, kneerail & toeboard
(2) Cased face ladders
Easy fitting ladder safety cage(s)
Self closing safety gate(s) included at the top of the access ladder(s)

Control Systems:
Metrix 5550-111-01 SPDT vibration cutoff switch
(1) field installed control panel per cell
ABB VFD Premium NEMA 3R
Field installed Single Speed UL NEMA 3R safety switch with interlock
VFD startup expenses included (no vibration test).

Motor Specials:
3 @ 75 HP NYSERDA 95.4% NEMA Nominal Efficient - Inverter Duty

Field Installed Equipment:
The field installed portion of the equipment will require approximately 143.3 man-hours of installation time after the tower arrives at the jobsite (based on USA experienced crew). The price to install these components is NOT included in the total price.

Total Sell Price ....(US Dollar) $ 292,000.00
(Freight included. Installation labor not included. Taxes not included.)

Labor Warranty, 18 months, FAP Galvanized Steel Tower $ 6,400.00

Payment Terms: NET 30 days from date of shipment (subject to credit approval)
Freight Terms: F.O.B. SPX plant(s) Olathe, Kansas, USA with freight prepaid
Shipment Lead-Time After Drawing Approval: 30 business days

Please advise if the drawing type you need has not been supplied. These are the available drawing types:
1). PDF 2D documents - These documents display the tower geometry with dimensions, notes and annotations.
2). DWG 2D AutoCAD layouts – This 2D layout is a full scale electronic representation of the tower to insert into your own AutoCad layout. The .dwg contains no text so should be accompanied by the PDF files.
3). JT 3D solid model files - These lightweight 3D solids may be used by solid model programs such as NX (Unigraphics), I-DEAS, Solid Edge, Catia, Pro/Engineer, and Autodesk Inventor 2009, among others. JT is relatively new technology and will be adopted by more programs in the coming months. A free JT viewer can be found at www.jt2go.com. JT is not compatible with Revit.
Revit – Revit files are not yet available. We are working towards developing a lightweight, simple Revit file. In the meantime, the .dwg 2D file can be used in Revit.

Notes:

- **Any purchase orders should be made out to SPX Cooling Technologies.**
- **This is a proposal and not a contract. When signed by the Purchaser in the space provided below, this form will be considered a purchase order from the Purchaser to SPX Cooling Technologies ("SPX Cooling"). Purchaser's order is not binding on SPX Cooling until the order is accepted by SPX Cooling as indicated by a signature of an officially authorized employee of SPX Cooling in the space provided below. The SPX Cooling Sales Representative providing this proposal has no authority to bind SPX Cooling to any contract, or contract terms or conditions in contradiction to those stated herein.**
- **Offer to purchase is valid for thirty (30) days from the proposal date. Shipment must be made within ninety (90) days of original purchase date. For requested shipments beyond ninety (90) days, pricing must be approved prior to receipt of order.**
- All sales, use or excise taxes payable by SPX Cooling, or to be collected by SPX Cooling from Purchaser, in connection with the sale, installation, or use of the proposed equipment shall be added to the prices quoted above at time of shipment. Any purchase order submitted for a tax exempt project must be accompanied by the Purchaser's valid tax exemption certificate for the state to which the goods are to be delivered. It is the Purchaser's responsibility to provide this documentation. SPX Cooling will not be liable for any failure of Purchaser to pay sales tax without said documentation.
- SPX Cooling's responsibility for delivery is limited to date of shipment. Carrier can be requested to give a maximum of 24 hours notice of delivery. Shipments involving more than one truck may arrive at a job site at different times.

UPDATE™ Version 4.12.4
Product Data: 7/22/2009 (Current)

7/30/2009 1:36:10 PM

Job Information
Dresser-Rand 6000 GPM CT RFQ RV1
ATSI Engineering Services
Amherst, NY 14228-2304

Selected By
Johnston Equipment Company
39 Saginaw Drive
Rochester, NY 14623-3146
guyward@jecoroc.com
guy ward
Tel 585-244-3336
Fax 585-244-3695

Marley NC Steel Cooling Tower Specification

Options:
Stainless Steel Cold Water Basin
Factory Mutual Approved
Air Inlet Screens
Extended Lube Line
Flow Control Valves
Basin Heaters
Ladder and Guardrail
Ladder Extension
Ladder Safety Cage
Ladder Safety Gate
Access Door Platform
Plenum Walkway
Interior Mechanical Equipment Access Platform
Vibration Switch
Marley "Premium" Variable Speed Drive using ABB Series 800 Drive
Premium Efficiency Motor
Marley Single-Point TF Terminal Box

1.0 Base:

1.1 Provide an induced draft, crossflow type, factory assembled, film fill, industrial duty, galvanized steel cooling tower situated as shown on the plans. The limiting overall dimensions of the tower shall be 22.417 ft wide, 36.271 ft long, and 27.087 ft high. Total operating horsepower of all fans shall not exceed 221.27 Hp, consisting of 3 @ 75 Hp motor(s). Tower shall be similar and equal in all respects to Marley Model NC8413WAS3.

2.0 Thermal Performance:

2.1 The tower shall be capable of cooling 6000 gpm of water from 120 °F to 85 °F at a design entering air wet-bulb temperature of 78 °F, and its thermal rating shall be Certified by the Cooling Technology Institute.

2.2 The tower shall be capable of a minimum _____ GPM/hp efficiency per ASHRAE Standard 90.1.

3.0 Performance Warranty:

3.1 CTI Certification notwithstanding, the cooling tower manufacturer shall guarantee that the tower supplied will meet the specified performance conditions when the tower is installed according to plan. If, because of a suspected thermal performance deficiency, the owner chooses to conduct an on-site thermal performance test under the supervision of a qualified, disinterested third party in accordance with CTI or ASME standards during the first year of operation; and if the tower fails to perform within the limits of test tolerance; then the cooling tower manufacturer will pay for the cost of the test and will make such corrections as are appropriate and agreeable to the owner to compensate for the performance deficiency.

4.0 Design Loading:

4.1 The tower structure, anchorage and all its components shall be designed by licensed structural engineers per the International Building Code to withstand a wind load of 30 psf, as well as a .3g seismic load. The fan deck and hot water basin covers shall be designed for 50 psf live load or a 200 lb. concentrated load. Guardrails, where specified, shall be capable of withstanding a 200 lb. concentrated live load in any direction, and shall be designed in accordance with OSHA guidelines.

5.0 Construction:

5.1 Except where otherwise specified, all components of the cooling tower shall be fabricated of heavy-gauge steel, protected against corrosion by G-235 galvanizing. The tower shall be capable of withstanding water having a pH of 6.5 to 8.0; a chloride content (NaCl) up to 300 ppm; a sulfate content (SO4) up to 250 ppm; a calcium content (CaCO3) up to 500 ppm; silica (SiO2) up to 150 ppm; and design hot water temperatures up to 125°F. The circulating water shall contain no oil, grease, fatty acids or organic solvents.

5.2 The specifications, as written, are intended to indicate those materials that will be capable of withstanding the above water quality in continuing service, as well as the loads described in paragraph 4.1. They are to be regarded as minimum requirements. Where component materials peculiar to individual tower designs are not specified, the manufacturers shall take the above water quality and load carrying capabilities into account in the selection of their materials of manufacture.

5.3 The tower shall include all design and material modifications necessary to meet the fire rating requirements of Factory Mutual. The product proposed shall be listed in the FM Approval Guide, latest edition.

6.0 Mechanical Equipment:

6.1 Fan(s) shall be propeller-type, incorporating wide-chord aluminum alloy blades and galvanized hubs. Blades shall be individually adjustable. Maximum fan tip speed shall be 13,000 ft/min. Fan(s) shall be driven through a right angle, industrial duty, oil lubricated, geared speed reducer that requires no oil changes for the first five (5) years of operation. The gearbox bearings shall be rated at an L10A service life of 100,000 hours or greater.

An external oil level dipstick shall be located adjacent to the motor at the fan deck surface and shall be accessible from a portable maintenance ladder.

6.2 Motor(s) shall be 75 Hp maximum, TEFC, 1.15 service factor, variable torque, and specially insulated for cooling tower duty. Speed and electrical characteristics shall be 1800 rpm, single-winding, ___ phase, 60 Hz, ____ volts. Motor shall operate in the shaft-horizontal position, and nameplate horsepower shall not be exceeded at design operation.

6.3 The complete mechanical equipment assembly for each cell shall be supported by a rigid steel structural support that resists misalignment between the motor and the gear reducer. The mechanical equipment assembly shall be warranted against any failure caused by defects in materials and workmanship for no less than five (5) years following the date of tower shipment. This warranty shall cover the fan, speed reducer, drive shaft and couplings, and the mechanical equipment support. The electric motor shall carry a manufacturer's warranty of at least one year.

6.3 The complete mechanical equipment assembly for each cell shall be supported by a rigid steel structural support that resists misalignment between the motor and the gear reducer. The mechanical equipment assembly shall be warranted against any failure caused by defects in materials and workmanship for no less than five (5) years following the date of tower shipment. This warranty shall cover the fan, speed reducer, motor, drive shaft and couplings, and the mechanical equipment support.

6.5 A complete UL listed Variable Speed Drive system in a NEMA 12 indoor or NEMA 3R outdoor enclosure shall be provided. The VFD shall use PWM technology with IGBT switching and integrated bypass design. VFD output switching shall not cause mechanical issues with gearbox teeth or drive shafts. The VFD shall catch a fan spinning in the reverse direction without tripping. The panel shall include a main disconnect with short circuit protection and external operating handle, lockable in the off position for safety. The system shall include a solid state, PI temperature controller to adjust frequency output of the drive in response to the tower cold-water temperature. The temperature of the cold water and set point shall be displayed on the door of the control panel. The bypass shall include a complete magnetic bypass circuit with capability to isolate the VFD when in the bypass mode. Transfer to the bypass mode shall be automatic in the event of VFD failure or for specific trip conditions allowing safe transfer of utility voltage to the motor. Automatic bypass with an earth ground condition is not allowed. The bypass contactor shall be cycled on and off while operating in bypass, to maintain the set-point temperature of the cold water. The drive design shall be operated as a stand-alone system without the need for a BMS system. Operator controls shall be mounted on the front of the enclosure and shall consist of start and stop control, bypass/VFD selector switch, Auto/Manual selector switch, manual speed control, and solid-state temperature controller. An emergency bypass selector switch internal to the panel allowing the cooling tower fan motor to be run at full speed shall be furnished. To prevent heating problems in the cooling tower fan motor the VFD system shall de energize the motor once 25% motor speed is reached and cooling is no longer required. The VFD shall include de-icing logic with auto canceling and adjustable time. Speed in De-Ice mode shall not exceed 50 % motor speed. The cooling tower manufacturer shall supply VFD start-up assistance. Tower vibration testing throughout the speed range is required to identify and lockout any natural frequency vibration levels which may exceed CTI guidelines.

6.6 A vibration limit switch shall be installed on the mechanical equipment support assembly and wired into the control panel. The purpose of this switch will be to interrupt power to the motor in the event of excessive vibration. It shall be adjustable for sensitivity, and shall require manual reset.

6.7 An externally mounted and wired terminal box shall be provided for each cell providing a single access location to the internal wiring. Terminate wiring for fan motor and options such as vibration switch, oil level switch and water level probes to the terminal box. The terminal box shall be built to UL508 standards and all terminal points marked for ease of connection in the field. The enclosure shall be NEMA 4X fiberglass. Entry points shall be into and out of the bottom of the enclosure preventing water collection in the enclosure.

7.0 Fill, Louvers and Drift Eliminators:

7.1 Fill shall be film type, thermoformed of 15 mil thick PVC, with louvers formed as part of each fill sheet. Fill shall be suspended from hot dip galvanized structural tubing supported from the tower structure, and shall be elevated above the floor of the cold water basin to facilitate cleaning. Air inlet faces of the tower shall be free of water splash-out. Fill shall be capable of withstanding a hot water temperature of 125°F.

7.2 Drift eliminators shall be PVC, triple-pass, and shall limit drift losses to 0.005% or less of the design water flow rate.

8.0 Hot Water Distribution System:

8.1 Two open basins (one above each bank of fill) shall receive hot water piped to each cell of the tower. These basins shall be installed and sealed at the factory, and shall be equipped with removable, galvanized steel covers capable of withstanding the loads described in paragraph 4.1. The water distribution system shall be accessible and maintainable during tower fan and water operation.

Heavy-duty flow-regulator valves shall be provided at the hot water inlet connections. These valves shall be disc-type, with cast iron bodies and stainless steel operating stems. There shall be a locking handle to maintain the valve setting in any position. Valves shall be right-angle configuration, precluding the need for inlet elbows.

8.3 The water distribution system shall be accessible and maintainable while tower is operating.

9.0 Casing, Fan Deck and Fan Guard:

9.1 The casing and fan deck shall be heavy-gauge galvanized steel, and shall be capable of withstanding the loads described in paragraph 4.1. The top of the fan cylinder shall be equipped with a conical, non-sagging, removable fan guard, fabricated of welded 5/16" and 7 gauge rods, and hot dip galvanized after fabrication. Fan cylinders 5'-0" in height and over shall not be required to have a fan guard.

10.0 Access:

10.1 A large galvanized, rectangular access door shall be located on both end panels for entry into the cold water basin. Doors shall provide access to the fan plenum area to facilitate inspection and allow maintenance to the fan drive system.

9.2 The air inlet faces of the tower shall be covered by 1" mesh hot-dipped galvanized welded wire screens. Screens shall be secured to removable galvanized U-edge frames. Screens shall be designed so bottom half can be removed for easy access to the cold water basin.

10.2 The top of the tower shall be equipped with a sturdy guardrail, complete with kneerail and toeboard, designed according to OSHA guidelines and factory welded into subassemblies for ease of field installation. Posts, toprails and kneerails shall be 1.5 " square tubing. The guardrail assembly shall be hot dipped galvanized after welding and capable of withstanding a 200 pound concentrated live load in any direction. Posts shall be spaced on centers of 8'-0" or less. A 1'-6" wide aluminum ladder with 3" I-beam side rails and 1.25" diameter rungs shall be permanently attached to the endwall casing of the tower, rising from the base of the tower to the top of the guardrail.

Provide a ladder extension for connection to the foot of the ladder attached to the tower casing. This extension shall be long enough to rise from the roof (grade) level to the base of the tower. The installing contractor shall be responsible for cutting the ladder to length; attaching it to the foot of the tower ladder; and anchoring it at its base.

10.3 Ladder Safety

A heavy gauge aluminum safety cage shall surround the ladder, extending from a point approximately 7'-0" above the foot of the ladder to the top of the guardrail.

A galvanized steel, self-closing gate shall be provided at the guardrail level of the ladder.

10.4 There shall be an access platform at the base of the tower extending from the vertical ladder to the endwall access door. The platform shall be surrounded by a guardrail, kneerail, and toeboard.

10.5 Provide a factory-installed, walkway extending from one endwall access door to the other endwall. This walkway shall be supported by a steel framework, and the top of the walkway shall be at or above the cold water basin overflow level. Walkway and framework to be equivalent material to tower basin.

An internal ladder shall extend upward from the plenum walkway to an elevated fiberglass bar grating platform convenient to the care and maintenance of the tower's mechanical equipment. The platform shall be surrounded by a sturdy guardrail and kneerail system.

11.0 Cold Water Collection Basin:

11.1 The collection basin shall be heavy-gauge S300 stainless steel, and shall include the number and type of suction connections required to accommodate the outflow piping system shown on the plans. Suction connections shall be equipped with stainless steel debris screens. A factory-installed, float-operated, mechanical make-up valve shall be included. An overflow and drain connection shall be provided in each cell of the cooling tower. The basin floor shall slope toward the drain to allow complete flush out of debris and silt which may accumulate. Towers of more than one cell shall include stainless steel flumes for flow and equalization between cells. The basin shall be accessible and maintainable while water is circulating. All steel items which project into the basin (columns, diagonals, anchor clips, etc.) shall also be made of stainless steel.

11.2 Provide a system of electric immersion heaters and controls for each cell of the tower to prevent freezing of water in the collection basin during periods of shutdown. The system shall consist of one or more stainless steel electric immersion heaters installed in threaded couplings provided in the side of the basin. A NEMA 4 enclosure shall house a magnetic contactor to energize heaters; a transformer to provide 24-volt control circuit power; and a solid-state circuit board for temperature and low water cut-off. A control probe shall be located in the basin to monitor water level and temperature. The system shall be capable of maintaining 40°F water temperature at an ambient air temperature of _____ °F.

SPEC NC-09, 2/4/09

Aspen ICARUS

IPM

Overall Project Summary
Account Basis

Project summary (direct and indirect costs). Direct costs presented at an account level Indirect costs presented at a summary level

Project Title: HP CO2 Test Loop

Project Name: DR
Scenario Name: Steam Turbine Drive
Project Location: Olean, NY
Estimate Date: 26AUG09 19:24:33

Job No: 78180
Prep. By: MAS

Project Title:	HP CO2 Test Loop	**Prepared By:**	MAS
Project Location:	Olean, NY	**Currency:**	DOLLARS USD
Estimate Date:	26AUG09 19:24:33		

Project Notes

Report Printed At
7:25 pm 8/26/09

Project Title:	HP CO2 Test Loop		
Project Location:	Olean, NY	***Prepared By:***	MAS
Job No:	78180	***Est. Class:***	Budget
Estimate Date:	26AUG09 19:24:33	***Currency:***	DOLLARS USD

Overall Project Summary - Account Basis

Account	MH	Labor Cost	Matl Cost	Total Cost
(2) Equipment	918	48,680	2,299,100	2,347,780
(3) Piping	5,002	280,225	340,123	620,347
(4) Civil	769	33,906	21,622	55,528
(5) Steel	193	8,924	17,426	26,351
(6) Instruments	104	6,141	13,254	19,395
(7) Electrical	1,742	95,171	106,691	201,862
(8) Insulation	1,929	93,376	71,408	164,783
(9) Paint	639	26,721	5,902	32,623
Direct Totals	**11,297**	**593,144**	**2,875,525**	**3,468,669**
Const Equip & Indirects				42,700
Const Mgt, Staff, Supv				41,520
Freight				0
Taxes and Permits				0
Engineering				520,300
Other Project Costs				0
Contingency				610,978
Indirect Totals				**1,215,498**
Project Totals:	**11,297**	**593,144**	**2,875,525**	**4,684,167**

Aspen ICARUS

IPM

Direct Accounts
Code of Account Totals

Total direct cost (TDC) summary by Code of Account. Direct costs presented with key quantities at a sub-account level

Project Title:	HP CO2 Test Loop		
Project Name:	DR		
Scenario Name:	Steam Turbine Drive		
Project Location:	Olean, NY	**Job No:**	78180
Estimate Date:	26AUG09 19:24:33	**Prep. By:**	MAS

Project Title:	HP CO2 Test Loop		
Project Location:	Olean, NY	*Prepared By:*	MAS
Job No:	78180	*Est. Class:*	Budget
Estimate Date:	26AUG09 19:24:33	*Currency:*	DOLLARS USD

Direct Accounts - Code of Account Totals

Account	Code of Account	MH	Labor Cost	Matl Cost	Total Cost	Weight LBS
(2) Equipment						
	100 - Equipment - General	143	7,450		7,450	
	113 - Vertical Vessels	20	1,042	10,000	11,042	
	161 - Centrifugal Pumps	99	5,215	47,000	52,215	2,000
	172 - Steam Turbines	508	27,137	1,382,100	1,409,237	101,800
	192 - Filters	24	1,250	33,000	34,250	
	200 - Fluid Separation Equip.	20	1,042	10,000	11,042	
	261 - Shell & Tube Exchangers	38	2,100	525,000	527,100	6,800
	265 - Cooling Towers	66	3,444	292,000	295,444	56,700
	Account Totals	**918**	**48,680**	**2,299,100**	**2,347,780**	**167,300**
(3) AG Pipe						
	304 - Special Equip. Piping	20	1,122	10,000	11,122	
	306 - Piping System Testing	231	13,139		13,139	
	307 - Prefab Pipe Rework	62	3,511		3,511	
	311 - CS Field Mat'l			99,673	99,673	53,095
	312 - CS Field Shop Fab	15	888		888	
	313 - CS Remote Shop Mat'l			55,776	55,776	56,714
	314 - CS Remote Shop Fab			25,718	25,718	
	315 - CS Valves: Flanged			73,325	73,325	20,025
	316 - CS Valves: Non-Flanged			44,188	44,188	2,474
	317 - CS Pipe Erection	4,287	240,379		240,379	
	366 - Pipe Hangers, Shoes Etc.	387	21,186	13,475	34,661	1,595
	368 - Mechanical Connections			17,968	17,968	4,391
	Account Totals	**5,002**	**280,225**	**340,123**	**620,347**	**138,294**
(4) Concrete						
	444 - Concrete			12,676	12,676	
	446 - Concrete Pour And Finish	144	5,799	306	6,105	
	447 - Excavation	43	1,647		1,647	
	451 - Rebar	184	8,578	7,181	15,759	16,700
	452 - Foundation Accessories	14	665	405	1,070	300
	454 - Formwork Materials			1,055	1,055	
	455 - Field Fabricate Formwork	128	5,850		5,850	
	456 - Install Formwork	185	8,270		8,270	
	457 - Strip & Clean Formwork	64	2,881		2,881	
	458 - Backfill	6	215		215	
	Account Totals	**769**	**33,906**	**21,622**	**55,528**	**17,000**
(5) Steel						
	511 - Equipment Support Steel	171	7,922	16,050	23,972	10,000
	512 - Ladders	12	536	1,376	1,912	360
	591 - Steel Unload & Handling	10	466		466	
	Account Totals	**193**	**8,924**	**17,426**	**26,351**	**10,360**
(6) Instrumentation						
	613 - Pressure Instruments	4	255	333	588	10
	614 - Temperature Instruments	34	2,125	1,525	3,651	46
	621 - Control Center Panels			1,988	1,988	40

Report Printed At 7:25 pm 8/26/09

Project Title: HP CO2 Test Loop
Project Location: Olean, NY
Job No: 78180
Estimate Date: 26AUG09 19:24:33
Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Direct Accounts - Code of Account Totals

Account Code of Account	MH	Labor Cost	Matl Cost	Total Cost	Weight LBS
622 - Control Ctr Connections	4	234		234	
631 - Air Supply Piping	14	800	252	1,052	12
632 - Instrument Piping	7	419	1,251	1,670	60
633 - Terminations	2	106	57	162	3
634 - Pneumatic Tubing	2	101	6	106	
636 - Instrument Signal Wiring	1	50	26	76	1
642 - Conduit & Fittings	22	1,193	218	1,411	133
659 - Other Instr. Electrical	1	28	865	894	260
681 - Control Valves			6,733	6,733	1,628
691 - Instrument Testing	14	831		831	
Account Totals	**104**	**6,141**	**13,254**	**19,395**	**2,193**
(7) AG Electrical					
711 - Wire/Cable - LV	194	10,537	8,427	18,964	2,527
713 - Pilot Light	18	957	633	1,590	190
714 - Push Button Station	9	479	558	1,037	168
715 - Terminators/Connectors	13	722	252	974	76
718 - Wire/Cable - CV	15	805	194	998	58
721 - Conduit	579	31,971	8,757	40,728	8,068
722 - Conduit Fittings	226	12,459	3,507	15,966	507
733 - Motor Control Center-LV	364	19,351	51,365	70,715	2,569
739 - Disconnect Switch	4	200	315	515	16
752 - Lighting Fixtures	19	949	1,358	2,308	476
755 - Misc. Small Transformers	11	575	969	1,543	261
756 - Panelboards	19	1,022	767	1,789	268
757 - Wire/Cable - Lighting	2	120	29	149	10
791 - Electrical Circuit Tstng	22	1,237		1,237	
792 - Electrical Tracing	249	13,790	29,559	43,349	5,911
Account Totals	**1,742**	**95,171**	**106,691**	**201,862**	**21,105**
(8) Pipe Insulation					
811 - Pipe Insulation	1,929	93,376	71,408	164,783	43,446
Account Totals	**1,929**	**93,376**	**71,408**	**164,783**	**43,446**
(9) Paint					
912 - Paint - Piping	328	13,842	5,583	19,425	
922 - Surface Prep - Piping	311	12,880	319	13,198	
Account Totals	**639**	**26,721**	**5,902**	**32,623**	

Project Title: HP CO2 Test Loop

Project Location: Olean, NY
Job No: 78180
Estimate Date: 26AUG09 19:24:33

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Direct Accounts - Code of Account Totals

Account	Code of Account	MH	Labor Cost	Matl Cost	Total Cost	Weight LBS
Project Direct Totals		11,297	593,144	2,875,525	3,468,669	399,698

Aspen ICARUS

IPM

Direct Accounts
Code of Account Totals - Area

Total direct cost (TDC) summary by Code of Account by area. Direct costs presented with key quantities at a sub-account level

Project Title: HP CO2 Test Loop

Project Name: DR
Scenario Name: Steam Turbine Drive
Project Location: Olean, NY
Estimate Date: 26AUG09 19:24:33

Job No: 78180
Prep. By: MAS

Project Title:	HP CO2 Test Loop		
Project Location:	Olean, NY	Prepared By:	MAS
Job No:	78180	Est. Class:	Budget
Estimate Date:	26AUG09 19:24:33	Currency:	DOLLARS USD

Direct Accounts - Code of Account Totals - Area

Area / Account / Code of Account	MH	Labor Cost	Matl Cost	Total Cost	Weight LBS
Main Area : Main Area Area					
(2) Equipment					
100 - Equipment - General	143	7,450		7,450	
113 - Vertical Vessels	20	1,042	10,000	11,042	
161 - Centrifugal Pumps	99	5,215	47,000	52,215	2,000
172 - Steam Turbines	508	27,137	1,382,100	1,409,237	101,800
192 - Filters	24	1,250	33,000	34,250	
200 - Fluid Separation Equip.	20	1,042	10,000	11,042	
261 - Shell & Tube Exchangers	38	2,100	525,000	527,100	6,800
265 - Cooling Towers	66	3,444	292,000	295,444	56,700
Account Totals	**918**	**48,680**	**2,299,100**	**2,347,780**	**167,300**
(3) AG Pipe					
304 - Special Equip. Piping	20	1,122	10,000	11,122	
306 - Piping System Testing	231	13,139		13,139	
307 - Prefab Pipe Rework	62	3,511		3,511	
311 - CS Field Mat'l			99,673	99,673	53,095
312 - CS Field Shop Fab	15	888		888	
313 - CS Remote Shop Mat'l			55,776	55,776	56,714
314 - CS Remote Shop Fab			25,718	25,718	
315 - CS Valves: Flanged			73,325	73,325	20,025
316 - CS Valves: Non-Flanged			44,188	44,188	2,474
317 - CS Pipe Erection	4,287	240,379		240,379	
366 - Pipe Hangers, Shoes Etc.	387	21,186	13,475	34,661	1,595
368 - Mechanical Connections			17,968	17,968	4,391
Account Totals	**5,002**	**280,225**	**340,123**	**620,347**	**138,294**
(4) Concrete					
444 - Concrete			12,676	12,676	
446 - Concrete Pour And Finish	144	5,799	306	6,105	
447 - Excavation	43	1,647		1,647	
451 - Rebar	184	8,578	7,181	15,759	16,700
452 - Foundation Accessories	14	665	405	1,070	300
454 - Formwork Materials			1,055	1,055	
455 - Field Fabricate Formwork	128	5,850		5,850	
456 - Install Formwork	185	8,270		8,270	
457 - Strip & Clean Formwork	64	2,881		2,881	
458 - Backfill	6	215		215	
Account Totals	**769**	**33,906**	**21,622**	**55,528**	**17,000**
(5) Steel					
511 - Equipment Support Steel	171	7,922	16,050	23,972	10,000
512 - Ladders	12	536	1,376	1,912	360
591 - Steel Unload & Handling	10	466		466	
Account Totals	**193**	**8,924**	**17,426**	**26,351**	**10,360**
(6) Instrumentation					
613 - Pressure Instruments	4	255	333	588	10
614 - Temperature Instruments	34	2,125	1,525	3,651	46
621 - Control Center Panels			1,988	1,988	40
622 - Control Ctr Connections	4	234		234	

Project Title:	HP CO2 Test Loop		
Project Location:	Olean, NY	*Prepared By:*	MAS
Job No:	78180	*Est. Class:*	Budget
Estimate Date:	26AUG09 19:24:33	*Currency:*	DOLLARS USD

Direct Accounts - Code of Account Totals - Area

Area / Account / Code of Account	MH	Labor Cost	Matl Cost	Total Cost	Weight LBS
Main Area : Main Area Area					
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631 - Air Supply Piping	14	800	252	1,052	12
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633 - Terminations	2	106	57	162	3
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636 - Instrument Signal Wiring	1	50	26	76	1
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Account Totals	**104**	**6,141**	**13,254**	**19,395**	**2,193**
(7) AG Electrical					
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715 - Terminators/Connectors	13	722	252	974	76
718 - Wire/Cable - CV	15	805	194	998	58
721 - Conduit	579	31,971	8,757	40,728	8,068
722 - Conduit Fittings	226	12,459	3,507	15,966	507
733 - Motor Control Center-LV	364	19,351	51,365	70,715	2,569
739 - Disconnect Switch	4	200	315	515	16
752 - Lighting Fixtures	19	949	1,358	2,308	476
755 - Misc. Small Transformers	11	575	969	1,543	261
756 - Panelboards	19	1,022	767	1,789	268
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792 - Electrical Tracing	249	13,790	29,559	43,349	5,911
Account Totals	**1,742**	**95,171**	**106,691**	**201,862**	**21,105**
(8) Pipe Insulation					
811 - Pipe Insulation	1,929	93,376	71,408	164,783	43,446
Account Totals	**1,929**	**93,376**	**71,408**	**164,783**	**43,446**
(9) Paint					
912 - Paint - Piping	328	13,842	5,583	19,425	
922 - Surface Prep - Piping	311	12,880	319	13,198	
Account Totals	**639**	**26,721**	**5,902**	**32,623**	
Area Totals	**11,297**	**593,144**	**2,875,525**	**3,468,669**	**399,698**

Project Title:	HP CO2 Test Loop		
Project Location:	Olean, NY	*Prepared By:*	MAS
Job No:	78180	*Est. Class:*	Budget
Estimate Date:	26AUG09 19:24.33	*Currency:*	DOLLARS USD

Direct Accounts - Code of Account Totals - Area

Area / Account / Code of Account	MH	Labor Cost	Matl Cost	Total Cost	Weight LBS
Project Direct Totals	11,297	593,144	2,875,525	3,468,669	399,698

Aspen ICARUS

IPM

Project Direct Totals
Overall Item Summary

Total direct cost (TDC) summary
Direct costs presented with key quantities at an item commodity level.

Project Title: HP CO2 Test Loop
Project Name: DR
Scenario Name: Steam Turbine Drive
Project Location: Olean, NY
Estimate Date: 26AUG09 19:24:33

Job No: 78180
Prep. By: MAS

Project Title: HP CO2 Test Loop
Project Location: Olean, NY
Job No: 78180
Estimate Date: 26AUG09 19:24:33

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Project Direct Totals - Overall Item Summary

Account / Code of Account / Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(2) Equipment										
100 - Equipment - General										
Tower Install		1 EACH	143.0	143.0	7,450	0.00		7,450		
Code of Account Subtotal				143	7,450			7,450		
113 - Vertical Vessels										
Blowdown Tank		1 EACH	20.0	20.0	1,042	10,000	10,000	11,042		
Code of Account Subtotal				20	1,042		10,000	11,042		
161 - Centrifugal Pumps										
Cooling Water Pump		3 EACH	8.0	24.0	1,250	15,000	45,000	46,250		
EQUIPMENT & SETTING	2 ITEM(S)		37.6	75.2	3,965	1,000	2,000	5,965	CS	2,000
Code of Account Subtotal				99	5,215		47,000	52,215		2,000
172 - Steam Turbines										
EQUIPMENT & SETTING	1 ITEM(S)		507.8	507.8	27,137	1,382,100		1,409,237	A285C	101,800
Code of Account Subtotal				508	27,137		1,382,100	1,409,237		101,800
192 - Filters										
Cooling Water Filter		2 EACH	12.0	24.0	1,250	16,500	33,000	34,250		
Code of Account Subtotal				24	1,250		33,000	34,250		
200 - Fluid Separation Equip.										
Steam Separator		1 EACH	20.0	20.0	1,042	10,000	10,000	11,042		
Code of Account Subtotal				20	1,042		10,000	11,042		
261 - Shell & Tube Exchangers										
EQUIPMENT & SETTING	1 ITEM(S)		38.3	38.3	2,100	525,000	525,000	527,100	A 214	6,800

Project Title: HP CO2 Test Loop
Project Location: Olean, NY
Job No: 78180
Estimate Date: 26AUG09 19:24:33

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Project Direct Totals - Overall Item Summary

Account / Code of Account / Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(2) Equipment										
Code of Account Subtotal				38	2,100		525,000	527,100		6,800
265 - Cooling Towers										
EQUIPMENT & SETTING	1 ITEM(S)		65.7	65.7	3,444	292,000	292,000	295,444	GALV	56,700
Code of Account Subtotal				66	3,444		292,000	295,444		56,700
Account Total - (2) Equipment				918	48,680		2,299,100	2,347,780		167,300
(3) AG Pipe										
304 - Special Equip. Piping										
Expansion Joints		2 EACH	10.0	20.0	1,122	5,000	10,000	11,122		
Code of Account Subtotal				20	1,122		10,000	11,122		
306 - Piping System Testing										
PIPE TESTING			231.4	231.4	13,139	0.00		13,139		
Code of Account Subtotal				231	13,139			13,139		
307 - Prefab Pipe Rework										
REPAIR & ADJ PREFAB PIPE			15.4	61.8	3,511	0.00		3,511		
Code of Account Subtotal				62	3,511			3,511		
311 - CS Field Mat'l										
BOLTS & NUTS 0.625 IN		1,376 EACH				0.53	733	733	A 53	
BOLTS & NUTS 0.875 IN		1,068 EACH				0.91	969	969	A 53	
ELBOW 4 IN SCH 40		5 EACH				28.90	144	144	A 53	47
ELBOW 20 IN SCH 40		12 EACH				1,769	21,231	21,231	A 53	4,120
GASKET 4 IN 150 CLASS		172 EACH				8.58	1,476	1,476	GRAPH	

Project Title: HP CO2 Test Loop
Project Location: Olean, NY
Job No: 78180
Estimate Date: 26AUG09 19:24:33

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Project Direct Totals - Overall Item Summary

Account / Code of Account / Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(3) AG Pipe										
311 - CS Field Mat'l										
GASKET 8 IN 300 CLASS		89 EACH				18.84	1,677	1,677	GRAPH	
PIPE 4 IN SCH 40	80 FEET					8.27	662	662	A 53	863
PIPE 20 IN SCH 40	360 FEET					160	57,477	57,477	A 53	44,320
PIPE 48 IN 0.38 IN THK	8 FEET					181	1,450	1,450	A 53	1,526
PIPE 0.75 IN SCH 80	2 FEET					3.20	6	6	A 106	3
REDUCR 20 IN SCH 40		2 EACH				1,562	3,124	3,124	A 53	204
TEE 4 IN SCH 40		1 EACH				49.12	49	49	A 53	15
TEE 20 IN SCH 40		6 EACH				1,766	10,597	10,597	A 53	1,993
THREADOLETS		8 EACH				9.67	77	77	CS	4
Code of Account Subtotal							**99,673**	**99,673**		**53,095**
312 - CS Field Shop Fab										
THREADOLETS		8 EACH	1.9	15.5	888	0.00		888	A 53	
Code of Account Subtotal				**15**	**888**			**888**		
313 - CS Remote Shop Mat'l										
ELBOW 4 IN SCH 40		87 EACH				15.45	1,345	1,345	A 106	744
ELBOW 4 IN SCH 40		6 EACH				15.46	93	93	A 53	51
ELBOW 8 IN SCH 80		82 EACH				86.94	7,129	7,129	A 106	5,701
FLG SO 4 IN 150 CLASS		172 EACH				11.49	1,977	1,977	CS	1,977
FLG WN 8 IN 300 CLASS		89 EACH				82.23	7,318	7,318	CS	6,307
PIPE 4 IN SCH 40	600 FEET					8.27	4,964	4,964	A 106	6,474
PIPE 4 IN SCH 40	10 FEET					8.20	82	82	A 53	108
PIPE 8 IN SCH 80	720 FEET					33.23	23,924	23,924	A 106	31,239
REDUCR 4 IN SCH 40		36 EACH				13.51	486	486	A 106	129

Project Title: HP CO2 Test Loop
Project Location: Olean, NY
Job No: 78180
Estimate Date: 26AUG09 19:24:33

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Project Direct Totals - Overall Item Summary

Account Code of Account Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(3) AG Pipe										
313 - CS Remote Shop Mat'l										
REDUCR 8 IN SCH 80		29 EACH				51.42	1,491	1,491	A 106	628
TEE 4 IN SCH 40		46 EACH				35.91	1,652	1,652	A 106	574
TEE 4 IN SCH 40		4 EACH				35.91	144	144	A 53	50
TEE 8 IN SCH 80		34 EACH				147	5,007	5,007	A 106	2,722
THREADOLETS		17 EACH				9.67	164	164	CS	10
Code of Account Subtotal							**55,776**	**55,776**		**56,714**
314 - CS Remote Shop Fab										
PIPE 4 IN SCH 40		9,901 LBS				0.75	7,457	7,457	A 106	
PIPE 4 IN SCH 40		210 LBS				6.45	1,354	1,354	A 53	
PIPE 8 IN SCH 80		46,603 LBS				0.36	16,907	16,907	A 106	
Code of Account Subtotal							**25,718**	**25,718**		
315 - CS Valves: Flanged										
BALL V 4 IN 150 CLASS		18 EACH				448	8,070	8,070	CS	1,962
BALL V 8 IN 300 CLASS		6 EACH				2,383	14,299	14,299	CS	3,168
BFLY V 4 IN 150 CLASS		12 EACH				326	3,914	3,914	CS	449
BFLY V 8 IN 300 CLASS		6 EACH				1,156	6,938	6,938	CS	1,087
CHEK V 4 IN 150 CLASS		14 EACH				257	3,592	3,592	CS	1,347
CHEK V 8 IN 300 CLASS		6 EACH				970	5,823	5,823	CS	2,796
GLOB V 4 IN 150 CLASS		24 EACH				407	9,779	9,779	CS	2,693
GLOB V 8 IN 300 CLASS		12 EACH				1,743	20,911	20,911	CS	6,523
Code of Account Subtotal							**73,325**	**73,325**		**20,025**
316 - CS Valves: Non-Flanged										

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Project Title: HP CO2 Test Loop
Project Location: Olean, NY
Job No: 78180
Estimate Date: 26AUG09 19:24:33

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Project Direct Totals - Overall Item Summary

Account Code of Account Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(3) AG Pipe										
316 - CS Valves: Non-Flanged										
BFLY V 4 IN 150 CLASS		1 EACH				390	390	390	CS	13
BFLY V 20 IN 150 CLASS		8 EACH				5,424	43,392	43,392	CS	2,298
GATE V 4 IN 150 CLASS		1 EACH				304	304	304	CS	107
GATE V 0.75 IN 800 CLASS		4 EACH				25.32	101	101	CS	56
Code of Account Subtotal							**44,188**	**44,188**		**2,474**
317 - CS Pipe Erection										
BOLT UP CONNECTIONS		229 EACH	1.6	364.8	20,633	0.00		20,633	A 106	
ERECT SHOP FAB PIPE		1,320 FEET	0.58	764.4	41,304	0.00		41,304	A 106	
ERECT SHOP FAB PIPE		10 FEET	0.41	4.1	222	0.00		222	A 53	
ERECT STRAIGHT RUN PIPE		448 FEET	0.43	190.9	10,261	0.00		10,261	A 53	
ERECT VALVE		98 EACH	2.2	217.6	11,893	0.00		11,893	A 106	
ERECT VALVES & FITTINGS		37 EACH	5.5	203.5	10,936	0.00		10,936	A 53	
FIELD X-RAY		124 EACH	1.2	148.8	8,498	0.00		8,498	A 106	
FIELD X-RAY		2 EACH	3.3	7.3	417	0.00		417	A 53	
INSTR. PIPE,FITTINGS		2 FEET	3.8	7.7	439	0.00		439	A 106	
VICTAULIC COUPLING		78 EACH	0.51	39.7	2,245	0.00		2,245	A 53	
WELDING		781 EACH	2.9	2,248.4	128,418	0.00		128,418	A 106	
WELDING		11 EACH	8.1	89.5	5,114	0.00		5,114	A 53	
Code of Account Subtotal				**4,287**	**240,379**			**240,379**		
366 - Pipe Hangers, Shoes Etc.										
ERECT PREFAB PIPE SUPP.		123 EACH	3.1	386.6	21,186	110	13,475	34,661		1,595
Code of Account Subtotal				**387**	**21,186**		**13,475**	**34,661**		**1,595**

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Project Title: HP CO2 Test Loop
Project Location: Olean, NY
Job No: 78180
Estimate Date: 26AUG09 19:24:33

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Project Direct Totals - Overall Item Summary

Account / Code of Account / Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(3) AG Pipe										
368 - Mechanical Connections										
VIC CP 4 IN		17 EACH				24.55	417	417	CS	94
VIC CP 20 IN		61 EACH				288	17,550	17,550	CS	4,297
Code of Account Subtotal							17,968	17,968		4,391
Account Total - (3) AG Pipe				5,002	280,225		340,123	620,347		138,294
(4) Concrete										
444 - Concrete										
READY-MIX CONC. - TYPE B		6.3 CY				81.60	514	514		
READY-MIX CONC. - TYPE C		124.2 CY				97.92	12,162	12,162		
Code of Account Subtotal							12,676	12,676		
446 - Concrete Pour And Finish										
POUR AND FINISH CONCRETE	130.50 CY		1.0	130.8	5,254	0.00		5,254		
SEAL SLAB - TYPE A CONC.		4.7 CY	2.9	13.6	545	65.28	306	851		
Code of Account Subtotal				144	5,799		306	6,105		
447 - Excavation										
HAND EXCAVATION		4.8 CY	3.6	17.2	592	0.00		592		
MACHINE EXCAVATION		235.2 CY	0.11	25.9	1,055	0.00		1,055		
Code of Account Subtotal				43	1,647			1,647		
451 - Rebar										
REBAR INSTALLATION		8.4 TONS	22.1	184.4	8,578	860	7,181	15,759		16,700
Code of Account Subtotal				184	8,578		7,181	15,759		16,700

Project Title: HP CO2 Test Loop
Project Location: Olean, NY
Job No: 78180
Estimate Date: 26AUG09 19:24:33

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Project Direct Totals - Overall Item Summary

Account / Code of Account / Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(4) Concrete										
452 - Foundation Accessories										
ANCHORS AND EMBEDMENTS		300.0 LBS	0.05	13.5	665	1.35	405	1,070		300
Code of Account Subtotal				**14**	**665**		**405**	**1,070**		**300**
454 - Formwork Materials										
BRACING FORMWORK		2,480.0 BD FT				0.20	496	496	WOOD	
CONTACT SURFACE FORMWORK		1,270.0 SF				0.44	559	559	WOOD	
Code of Account Subtotal							**1,055**	**1,055**		
455 - Field Fabricate Formwork										
FORMWORK FABRICATION		1,270.0 SF	0.10	127.8	5,850	0.00		5,850		
Code of Account Subtotal				**128**	**5,850**			**5,850**		
456 - Install Formwork										
FORMWORK INSTALLATION		1,270.0 SF	0.15	184.8	8,270	0.00		8,270		
Code of Account Subtotal				**185**	**8,270**			**8,270**		
457 - Strip & Clean Formwork										
STRIP AND CLEAN FORMWORK		1,270.0 SF	0.05	64.4	2,881	0.00		2,881		
Code of Account Subtotal				**64**	**2,881**			**2,881**		
458 - Backfill										
EXCAVATED SOIL		125.0 CY	0.05	6.3	215	0.00		215		
Code of Account Subtotal				**6**	**215**			**215**		
Account Total - (4) Concrete				**769**	**33,906**		**21,622**	**55,528**		**17,000**

Project Title: HP CO2 Test Loop
Project Location: Olean, NY
Job No: 78180
Estimate Date: 26AUG09 19:24:33

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Project Direct Totals - Overall Item Summary

Account / Code of Account / Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(5) Steel										
511 - Equipment Support Steel										
18.00 LB/FT	3.00 TONS		28.4	85.3	3,944	3,054	9,162	13,106	A 36	6,000
9.00 LB/FT	2.00 TONS		43.0	86.0	3,977	3,444	6,889	10,866	A 36	4,000
Code of Account Subtotal				**171**	**7,922**		**16,050**	**23,972**		**10,000**
512 - Ladders										
VERT. LADDER WITH CAGE	0.18 TONS	20.0 FEET	64.4	11.6	536	7,644	1,376	1,912		360
Code of Account Subtotal				**12**	**536**		**1,376**	**1,912**		**360**
591 - Steel Unload & Handling										
UNLOAD/HANDLE MISC STEEL		5.2 TONS	2.0	10.4	466	0.00		466		
Code of Account Subtotal				**10**	**466**			**466**		
Account Total - (5) Steel				**193**	**8,924**		**17,426**	**26,351**		**10,360**
(6) Instrumentation										
613 - Pressure Instruments										
PI GAUGE	4 EACH		1.0	4.0	255	83.34	333	588		10
Code of Account Subtotal				**4**	**255**		**333**	**588**		**10**
614 - Temperature Instruments										
THERMOWELL	3 EACH		3.2	9.5	534	112	335	870	SS316	10
TI DIAL	2 EACH		1.0	2.0	127	92.31	185	312		6
TT/TC HEAD SM XMTR	1 EACH		23.0	23.0	1,463	1,006	1,006	2,469		30
Code of Account Subtotal				**34**	**2,125**		**1,525**	**3,651**		**46**
621 - Control Center Panels										

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Project Direct Totals - Overall Item Summary

Account / Code of Account / Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(6) Instrumentation										
621 - Control Center Panels										
PANEL INSTRUMENTS FOP		1 EACH				1,988	1,988	1,988		40
Code of Account Subtotal							**1,988**	**1,988**		**40**
622 - Control Ctr Connections										
TIC		1 EACH	4.2	4.2	234	0.00		234		
Code of Account Subtotal				**4**	**234**			**234**		
631 - Air Supply Piping										
0.25 IN DIA BALL VALVE		2 EACH	0.40	0.80	51	10.21	20	71	BRASS	2
0.25 IN DIA NIPPLE XS		1 EACH	0.35	0.35	22	2.40	2	25	GALV	
0.25 IN DIA SWAGE NIP.		1 EACH	0.35	0.35	22	4.81	5	27	GALV	
0.75 IN DIA ELBOW		2 EACH	0.35	0.70	40	4.16	8	48	GALV	
0.75 IN DIA GATE VALVE		1 EACH	0.45	0.45	26	22.50	23	48	BRASS	1
0.75 IN DIA NIPPLE XS		1 EACH	0.35	0.35	20	3.21	3	23	GALV	
0.75 IN DIA PIPE		20 FEET	0.35	7.0	396	1.52	30	426	GALV	2
0.75 IN DIA SWG NIPPLE		1 EACH	0.35	0.35	20	6.75	7	27	GALV	
0.75 IN DIA TEE,REDUCER		1 EACH	1.5	1.5	85	8.63	9	94	GALV	
FILTER REGULATOR		1 EACH	2.1	2.1	119	145	145	264	BRASS	7
Code of Account Subtotal				**14**	**800**		**252**	**1,052**		**12**
632 - Instrument Piping										
0.50 IN DIA GLOBE VALVE		4 EACH	0.45	1.8	102	263	1,053	1,155	SS	52
0.50 IN DIA NIPPLE XS		4 EACH	0.35	1.4	79	6.31	25	105	SS	2
0.50 IN DIA PLUG		4 EACH	0.35	1.4	79	4.51	18	97	SS	
0.50 IN DIA TEE		4 EACH	0.35	1.4	79	14.86	59	139	SS	2

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Project Location: Olean, NY
Job No: 78180
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Currency: DOLLARS USD

Project Direct Totals - Overall Item Summary

Account / Code of Account / Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(6) Instrumentation										
632 - Instrument Piping										
0.75 IN DIA SWG NIP. XS		4 EACH	0.35	1.4	79	23.79	95	174	SS	4
Code of Account Subtotal				**7**	**419**		**1,251**	**1,670**		**60**
633 - Terminations										
1 PAIR CABLE TERM		4 EACH	0.48	1.9	106	14.17	57	162	RIGID	3
Code of Account Subtotal				**2**	**106**		**57**	**162**		**3**
634 - Pneumatic Tubing										
0.25 IN DIA MNPT TUB CN		2 EACH	0.54	1.1	69	1.78	4	72	BRASS	
0.25 IN DIA TUBE		5 FEET	0.10	0.50	32	0.44	2	34	CU	
Code of Account Subtotal				**2**	**101**		**6**	**106**		
636 - Instrument Signal Wiring										
1 PAIR INST CABLE		120 FEET	0.01	0.90	50	0.22	26	76	RIGID	1
Code of Account Subtotal				**1**	**50**		**26**	**76**		**1**
642 - Conduit & Fittings										
0.75 IN DIA BUSHING		8 EACH	0.30	2.4	133	1.16	9	142	GALV	1
0.75 IN DIA CONDUIT		100 FEET	0.13	13.0	718	1.29	129	847	GALV	120
0.75 IN DIA COUPLNG		4 EACH	0.12	0.48	27	1.09	4	31	GALV	1
0.75 IN DIA ELBOWS		2 EACH	0.12	0.24	13	2.61	5	18	GALV	3
0.75 IN DIA FITTNGS		2 EACH	0.70	1.4	77	8.12	16	94	GALV	2
0.75 IN DIA SEALS		4 EACH	0.60	2.4	133	11.31	45	178	GALV	4
0.75 IN DIA UNIONS		4 EACH	0.42	1.7	93	2.10	8	101	GALV	2
Code of Account Subtotal				**22**	**1,193**		**218**	**1,411**		**133**

Project Title: HP CO2 Test Loop
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Prepared By: MAS
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Project Direct Totals - Overall Item Summary

Account / Code of Account / Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(6) Instrumentation										
659 - Other Instr. Electrical										
STD. E/P POSITIONER		1 EACH	0.50	0.50	28	865	865	894		260
Code of Account Subtotal				**1**	**28**		**865**	**894**		**260**
681 - Control Valves										
BU/P V 14 IN 150 CLASS	1 EACH					6,733	6,733	6,733	CS	1,628
Code of Account Subtotal							**6,733**	**6,733**		**1,628**
691 - Instrument Testing										
INSTRUMENT TESTING		1 EACH	14.0	14.0	831	0.00		831		
Code of Account Subtotal				**14**	**831**			**831**		
Account Total - (6) Instrumentation				**104**	**6,141**		**13,254**	**19,395**		**2,193**
(7) AG Electrical										
711 - Wire/Cable - LV										
12 AWG 600 V	600 FEET					0.12	72	72		21
2 AWG 600 V	2,145 FEET					0.68	1,468	1,468		440
250 KCMIL 600 V	1,890 FEET					3.04	5,754	5,754		1,726
6 AWG 600 V	1,962 FEET					0.37	725	725		218
8 AWG 600 V	1,890 FEET					0.22	407	407		122
PULL WIRE IN CONDUIT		8,487 FEET	0.02	193.6	10,537	0.00		10,537		
Code of Account Subtotal				**194**	**10,537**		**8,427**	**18,964**		**2,527**
713 - Pilot Light										
PILOT LIGHTS		6 EACH	3.0	18.0	957	105	633	1,590		190

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Project Direct Totals - Overall Item Summary

Account / Code of Account / Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(7) AG Electrical										
Code of Account Subtotal				**18**	**957**		**633**	**1,590**		**190**
714 - Push Button Station										
PUSHBUTTONS		6 EACH	1.5	9.0	479	93.08	558	1,037		168
Code of Account Subtotal				**9**	**479**		**558**	**1,037**		**168**
715 - Terminators/Connectors										
12 AWG LV-TERM		24 EACH	0.08	1.8	98	1.31	31	129		11
14 AWG CV-TERM		48 EACH	0.08	3.6	196	1.31	63	259		19
2 AWG LV-TERM		24 EACH	0.08	1.9	104	1.31	31	136		9
250 KCMIL LV-TERM		18 EACH	0.13	2.3	127	3.54	64	191		19
6 AWG LV-TERM		30 EACH	0.08	2.3	122	1.31	39	162		11
8 AWG LV-TERM		18 EACH	0.08	1.4	73	1.31	24	97		7
Code of Account Subtotal				**13**	**722**		**252**	**974**		**76**
718 - Wire/Cable - CV										
14 AWG 600 V	2,120 FEET					0.09	194	194		58
PULL WIRE IN CONDUIT		2,120 FEET	0.01	14.8	805	0.00		805		
Code of Account Subtotal				**15**	**805**		**194**	**998**		**58**
721 - Conduit										
0.75 IN DIA CONDUIT		1,710 FEET				1.29	2,209	2,209	GALV	2,055
1.00 IN DIA CONDUIT		910 FEET				1.63	1,485	1,485	GALV	1,474
1.25 IN DIA CONDUIT		675 FEET				2.24	1,514	1,514	GALV	1,377
2.50 IN DIA CONDUIT		600 FEET				5.92	3,549	3,549	GALV	3,162
INSTALL CONDUIT		3,895 FEET	0.15	578.9	31,971	0.00		31,971		

Project Title: HP CO2 Test Loop
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Project Direct Totals - Overall Item Summary

Account / Code of Account / Description		Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(7) AG Electrical											
Code of Account Subtotal					579	31,971		8,757	40,728		8,068
722 - Conduit Fittings											
0.75 IN DIA	BUSHING		40 EACH	0.30	12.0	663	1.16	46	709	GALV	3
0.75 IN DIA	COUPLNG		59 EACH	0.12	7.1	391	1.09	64	455	GALV	13
0.75 IN DIA	ELBOWS		19 EACH	0.12	2.3	126	2.61	50	176	GALV	26
0.75 IN DIA	FITTNGS		16 EACH	0.70	11.2	619	8.12	130	749	GALV	15
0.75 IN DIA	SEALS		20 EACH	0.60	12.0	663	11.31	226	889	GALV	18
0.75 IN DIA	UNIONS		38 EACH	0.42	16.0	881	2.10	80	961	GALV	15
1.00 IN DIA	BUSHING		44 EACH	0.35	15.4	851	1.74	77	927	GALV	7
1.00 IN DIA	COUPLNG		33 EACH	0.15	5.0	273	1.45	48	321	GALV	10
1.00 IN DIA	ELBOWS		15 EACH	0.15	2.3	124	3.92	59	183	GALV	31
1.00 IN DIA	FITTNGS		15 EACH	0.80	12.0	663	12.04	181	843	GALV	16
1.00 IN DIA	SEALS		19 EACH	0.75	14.3	787	14.87	282	1,069	GALV	21
1.00 IN DIA	UNIONS		30 EACH	0.47	14.1	779	4.13	124	903	GALV	15
1.25 IN DIA	BUSHING		16 EACH	0.40	6.4	353	2.47	39	393	GALV	3
1.25 IN DIA	COUPLNG		24 EACH	0.35	8.4	464	1.81	44	507	GALV	10
1.25 IN DIA	ELBOWS		7 EACH	0.35	2.5	135	5.37	38	173	GALV	22
1.25 IN DIA	FITTNGS		7 EACH	1.1	7.7	425	18.13	127	552	GALV	13
1.25 IN DIA	SEALS		7 EACH	1.0	7.0	387	17.99	126	513	GALV	13
1.25 IN DIA	UNIONS		14 EACH	0.52	7.3	402	7.54	106	508	GALV	12
2.50 IN DIA	BUSHING		12 EACH	0.55	6.6	365	9.50	114	479	GALV	7
2.50 IN DIA	COUPLNG		21 EACH	0.80	16.8	928	8.34	175	1,103	GALV	30
2.50 IN DIA	ELBOWS		6 EACH	0.80	4.8	265	18.86	113	378	GALV	68
2.50 IN DIA	FITTNGS		6 EACH	2.2	13.2	729	75.42	453	1,182	GALV	51
2.50 IN DIA	SEALS		6 EACH	2.0	12.0	663	53.88	323	986	GALV	51

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Project Title: HP CO2 Test Loop
Project Location: Olean, NY
Job No: 78180
Estimate Date: 26AUG09 19:24:33

Prepared By: MAS
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Project Direct Totals - Overall Item Summary

Account / Code of Account / Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(7) AG Electrical										
722 - Conduit Fittings										
2.50 IN DIA UNIONS		12 EACH	0.79	9.5	524	40.32	484	1,007	GALV	37
Code of Account Subtotal				**226**	**12,459**		**3,507**	**15,966**		**507**
733 - Motor Control Center-LV										
480 V		3 SPACES	7.7	23.0	1,217	570	1,711	2,928		86
480 V 200.0 HP		18 SPACES	7.5	135.0	7,179	1,463	26,332	33,511		1,317
480 V 50.00 HP		12 SPACES	7.7	92.0	4,892	834	10,013	14,906		500
480 V 75.00 HP		15 SPACES	7.6	114.0	6,062	887	13,309	19,371		666
Code of Account Subtotal				**364**	**19,351**		**51,365**	**70,715**		**2,569**
739 - Disconnect Switch										
PANEL DISCONNECT SWITCH		1 EACH	3.6	3.6	200	315	315	515		16
Code of Account Subtotal				**4**	**200**		**315**	**515**		**16**
752 - Lighting Fixtures										
175 W M.V.		2 EACH	1.4	2.7	147	451	902	1,049		316
MAST		2 EACH	8.0	16.0	802	228	457	1,259		160
Code of Account Subtotal				**19**	**949**		**1,358**	**2,308**		**476**
755 - Misc. Small Transformers										
15 KVA 480 V		1 EACH	10.8	10.8	575	969	969	1,543		261
Code of Account Subtotal				**11**	**575**		**969**	**1,543**		**261**
756 - Panelboards										
100 A 8 CIRCUIT		1 EACH	6.5	6.5	360	331	331	692		116
12 CIRCUIT TRACER PANEL		1 ITEM(S)	12.0	12.0	661	436	436	1,097		152

Project Title: HP CO2 Test Loop
Project Location: Olean, NY
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Project Direct Totals - Overall Item Summary

Account / Code of Account / Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(7) AG Electrical										
Code of Account Subtotal				19	1,022		767	1,789		268
757 - Wire/Cable - Lighting										
12 AWG 600 V	240 FEET					0.12	29	29		10
PULL WIRE IN CONDUIT		240 FEET	0.01	2.2	120	0.00		120		
Code of Account Subtotal				2	120		29	149		10
791 - Electrical Circuit Tstng										
ELECTRICAL TESTING		1 EACH	22.3	22.3	1,237	0.00		1,237		
Code of Account Subtotal				22	1,237			1,237		
792 - Electrical Tracing										
BREAKER		5 EACH	0.75	3.8	208	127	637	845		126
CONTACTOR - 3 POLE		2 EACH	1.5	3.0	166	255	510	676		102
MISC. FITTINGS,HARDWARE		4 EACH				252	1,007	1,007		201
POWER CONNECT. KIT & JB		5 EACH	3.5	17.5	970	81.32	407	1,377		81
TAPE,LABEL,STRAP		4 EACH				64.40	258	258		53
THERMOSTAT-AMB. SENSING		2 EACH	3.0	6.0	333	255	510	842		102
TRACER CABLE - 5P	2,712 FEET		0.08	216.9	12,029	9.65	26,180	38,209		5,236
TRACER END SEAL KIT		5 EACH	0.30	1.5	83	10.59	53	136		10
Code of Account Subtotal				249	13,790		29,559	43,349		5,911
Account Total - (7) AG Electrical				1,742	95,171		106,691	201,862		21,105
(8) Pipe Insulation										
811 - Pipe Insulation										
CLADDING		18,829 SF				0.88	16,650	16,650	AL	4,252

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Project Direct Totals - Overall Item Summary

Account / Code of Account / Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
(8) Pipe Insulation										
811 - Pipe Insulation										
FASTENING BAND		12,804 FEET				0.07	894	894	AL	227
INSTALLATION PIPE		1,867 FEET	0.42	782.5	37,871	0.00		37,871		
INSTALLATION V/F		3,449 FEET	0.33	1,146.9	55,505	0.00		55,505		
PIP 2.0 IN TH 4.00 IN DI	11 FEET					5.57	58	58	CASIL	41
PIP 2.0 IN TH 6.00 IN DI	714 FEET					7.25	5,175	5,175	CASIL	3,739
PIP 2.0 IN TH 24.00 IN DI	378 FEET					21.67	8,192	8,192	CASIL	6,432
PIP 3.0 IN TH 8.00 IN DI	756 FEET					10.56	7,983	7,983	MWOO	5,443
PIP 3.0 IN TH 48.00 IN DI	8 FEET					53.61	450	450	CASIL	421
V/F 2.0 IN TH 4.00 IN DI	42 FEET					5.57	234	234	CASIL	165
V/F 2.0 IN TH 6.00 IN DI	1,990 FEET					7.25	14,422	14,422	CASIL	10,418
V/F 2.0 IN TH 24.00 IN DI	214 FEET					21.67	4,642	4,642	CASIL	3,645
V/F 3.0 IN TH 8.00 IN DI	1,203 FEET					10.56	12,706	12,706	MWOO	8,663
Code of Account Subtotal				**1,929**	**93,376**		**71,408**	**164,783**		**43,446**
Account Total - (8) Pipe Insulation				**1,929**	**93,376**		**71,408**	**164,783**		**43,446**
(9) Paint										
912 - Paint - Piping										
1 FINAL COAT BY SPRAY	3,112 SF		0.02	63.5	2,676	0.36	1,105	3,781		
1 PRIMER COAT BY SPRAY	10,360 SF		0.03	264.9	11,165	0.43	4,478	15,644		
Code of Account Subtotal				**328**	**13,842**		**5,583**	**19,425**		
922 - Surface Prep - Piping										
COMMERCIAL SAND BLAST		10,360 SF	0.03	310.8	12,880	0.03	319	13,198		
Code of Account Subtotal				**311**	**12,880**		**319**	**13,198**		

Project Title: HP CO2 Test Loop
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Project Direct Totals - Overall Item Summary

Account Code of Account Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
Account Total - (9) Paint				639	26,721		5,902	32,623		

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Project Direct Totals - Overall Item Summary

Account Code of Account Description	Key Qty	Other Qty	Unit MH	MH	Labor	Unit Mat	Matl Cost	Total Cost	Matl	Weight LBS
Project Direct Totals				11,297	593,144		2,875,525	3,468,669		399,698

Aspen ICARUS

IPM

Overall Project Summary
Account Basis

Project summary (direct and indirect costs). Direct costs presented at an account level. Indirect costs presented at a summary level.

Project Title: DR-Ramgen

Project Name: DR

Scenario Name: Electric Drive Option

Project Location: US

Estimate Date: 7AUG09 11:16:20

Job No: 78180

Prep. By: MAS

Project Title: DR-Ramgen
Project Location: US
Job No: 78180
Estimate Date: 7AUG09 11:16:20
Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Overall Project Summary - Account Basis

Account	MH	Labor Cost	Matl Cost	Total Cost
(2) Equipment	40	2.114	80.000	82.114
(3) Piping	97	5.649	1.094	6.743
(4) Civil	707	32.263	24.867	57.130
(7) Electrical	3.220	53.711	2.260.634	2.314.344
(9) Paint	14	585	100	685
Direct Totals	**4,077**	**94,321**	**2,366,694**	**2,461,015**
Const Equip & Indirects				14.700
Const Mgt, Staff, Supv				6.602
Freight				0
Taxes and Permits				0
Engineering				369.152
Other Project Costs				0
Contingency				427.720
Indirect Totals				**818.174**
Project Totals:	**4,077**	**94,321**	**2,366,694**	**3.279.189**

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Direct Accounts
Code of Account Totals

Total direct cost (TDC) summary by Code of Account. Direct costs presented with key quantities at a sub-account level.

Project Title:	DR-Ramgen		
Project Name:	DR		
Scenario Name:	Electric Drive Option		
Project Location:	US	**Job No:**	78180
Estimate Date:	7AUG09 11:16:20	**Prep. By:**	MAS

Project Title: DR-Ramgen
Project Location: US
Job No: 78180
Estimate Date: 7AUG09 11:16:20
Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Direct Accounts - Code of Account Totals

Account	Code of Account	MH	Labor Cost	Matl Cost	Total Cost	Weight LBS
(2) Equipment						
	260 - Heat Exchangers	40	2,114	80,000	82,114	
	Account Totals	**40**	**2,114**	**80,000**	**82,114**	
(3) AG Pipe						
	306 - Piping System Testing	5	301		301	
	311 - CS Field Mat'l			490	490	309
	312 - CS Field Shop Fab	31	1,822		1,822	
	315 - CS Valves: Flanged			350	350	94
	317 - CS Pipe Erection	46	2,649		2,649	
	366 - Pipe Hangers, Shoes Etc.	15	876	254	1,131	8
	Account Totals	**97**	**5,649**	**1,094**	**6,743**	**411**
(4) Bldg - Arch						
	472 - Bldg Structure/Finishes	177	8,131	9,545	17,676	
	473 - Building Furnishings	75	3,883	6,336	10,219	
	Account Totals	**252**	**12,014**	**15,881**	**27,895**	
(4) Concrete						
	444 - Concrete			4,509	4,509	
	446 - Concrete Pour And Finish	135	5,650	257	5,907	
	447 - Excavation	26	1,112		1,112	
	451 - Rebar	80	3,750	2,086	5,836	4,850
	452 - Foundation Accessories	3	124	67	192	50
	454 - Formwork Materials			786	786	
	455 - Field Fabricate Formwork	97	4,478		4,478	
	456 - Install Formwork	79	3,591		3,591	
	457 - Strip & Clean Formwork	30	1,335		1,335	
	458 - Backfill	0	4		4	
	Account Totals	**449**	**20,044**	**7,705**	**27,749**	**4,900**
(4) Other Civil						
	404 - Special Equip. Civil			1,000	1,000	
	Account Totals			**1,000**	**1,000**	
(4) Roads / RR						
	421 - Grade,Compact Base	5	205	281	486	
	Account Totals	**5**	**205**	**281**	**486**	
(7) AG Electrical						
	700 - Electrical - General	200	11,704	415,000	426,704	110,000
	711 - Wire/Cable - LV	4	208	65	272	19
	713 - Pilot Light	7	373	211	584	63
	714 - Push Button Station	3	187	186	373	56
	715 - Terminators/Connectors	4	237	73	310	22
	717 - Wire/Cable - MV	20	1,158	1,879	3,036	564
	718 - Wire/Cable - CV	2	107	22	129	7
	721 - Conduit	52	3,033	894	3,927	805
	722 - Conduit Fittings	49	2,885	971	3,856	144
	731 - Variable Frequency Drive	250	14,630	1,430,000	1,444,630	74,000

Project Title: DR-Ramgen
Project Location: US
Job No: 78180
Estimate Date: 7AUG09 11:16:20

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Direct Accounts - Code of Account Totals

Account	Code of Account	MH	Labor Cost	Matl Cost	Total Cost	Weight LBS
	733 - Motor Control Center-LV	51	2,860	3,422	6,282	172
	734 - Transformers - MV	2,317	2,341	389,950	392,291	19,498
	791 - Electrical Circuit Tstng	1	39		39	
	Account Totals	**2,959**	**39,761**	**2,242,673**	**2,282,434**	**205,350**
(7) UG Electrical						
	712 - Wire/Cable - HV	115	6,613	14,555	21,169	4,366
	715 - Terminators/Connectors	3	149	98	247	30
	721 - Conduit	50	2,922	1,968	4,890	1,115
	722 - Conduit Fittings	21	1,229	432	1,661	74
	761 - Electrical Trenching	72	3,036	907	3,943	45
	Account Totals	**261**	**13,950**	**17,961**	**31,910**	**5,630**
(9) Paint						
	912 - Paint - Piping	11	466	97	563	
	922 - Surface Prep - Piping	3	119	3	121	
	Account Totals	**14**	**585**	**100**	**685**	

Project Title:	DR-Ramgen		
Project Location:	US	*Prepared By:*	MAS
Job No:	78180	*Est. Class:*	Budget
Estimate Date:	7AUG09 11:16:20	*Currency:*	DOLLARS USD

Direct Accounts - Code of Account Totals

Account	Code of Account	MH	Labor Cost	Matl Cost	Total Cost	Weight LBS
Project Direct Totals		4,077	94,321	2,366,694	2,461,015	216,291

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IPM

Project Direct Totals
Installation Details - Overall

Total direct cost (TDC) details. Direct costs presented with key quantities per component and function/location.

Project Title: DR-Ramgen
Project Name: DR
Scenario Name: Electric Drive Option
Project Location: US
Estimate Date: 7AUG09 11:16:20

Job No: 78180
Prep. By: MAS

Project Title:	DR-Ramgen
Project Location:	US
Job No:	78180
Estimate Date:	7AUG09 11:16:20
Prepared By:	MAS
Est. Class:	Budget
Currency:	DOLLARS USD

Installation Details - Overall

Area / Component / Source / Account / Location / Code of Account	Tag No. / Description	Key Qty	Other Qty	Unit MH	MH	Labor Cost	Unit Mat	Matl Cost	Total Cost
Project / Area - AREA		ID:							
GENERAL									
AG 733 Motor Control Center-LV	480 V 20.00 HP		3.00 SPAC	8.52	25.56	1,430.09	570.36	1,711.09	3,141.18
791 Electrical Circuit Tstng	ELECTRICAL TESTING		1.00 EACH	0.67	0.67	39.44			39.44
733 Motor Control Center-LV	480 V 20.00 HP		3.00 SPAC	8.52	25.56	1,430.09	570.36	1,711.09	3,141.18
306 Piping System Testing	PIPE TESTING			5.06	5.06	301.48			301.48
Component / Source Totals					**57**	**3,201**		**3,422**	**6,623**
14,000 hp Cable(ID:23)	**Cable-01**	ID: 23							
GENERAL									
UG 722 Conduit Fittings	5.00 IN DIA ELBOWS		2.00 EACH	1.24	2.47	144.95	61.03	122.06	267.01
721 Conduit	INSTALL BURIED CONDUIT		150.00 FEET	0.14	21.33	1,252.18			1,252.18
721 Conduit	5.00 IN DIA CONDUIT		150.00 FEET				5.62	843.64	843.64
715 Terminators/Connectors	750 KCMIL HV-TERM		6.00 EACH	0.22	1.30	74.65	8.17	49.03	123.68
712 Wire/Cable - HV	PULL WIRE IN CONDUIT		480.00 FEET	0.10	49.61	2,859.86			2,859.86
712 Wire/Cable - HV	750 KCMIL 15000 V	480.00 FEET					13.11	6,294.21	6,294.21
722 Conduit Fittings	5.00 IN DIA COUPLNG		5.00 EACH	1.33	6.67	391.31	15.67	78.33	469.64
Component / Source Totals					**81**	**4,723**		**7,387**	**12,110**
14,000 hp Motor(ID:26)	**M-001**	ID: 26							
GENERAL									
AG 700 Electrical - General	14,000 hp Motor		1.00 ITEM(	200.00	200.00	11,704.14	415,000.00	415,000.00	426,704.14
Component / Source Totals					**200**	**11,704**		**415,000**	**426,704**
15MVA Transformer(ID:18)	**XFMR-01**	ID: 18							
GENERAL									
AG 722 Conduit Fittings	3.00 IN DIA BUSHING		4.00 EACH	0.67	2.67	156.52	12.84	51.35	207.87
452 Foundation Accessories	ANCHORS AND EMBEDMENTS		50.00 LBS	0.05	2.50	124.44	1.35	67.45	191.89
AG 717 Wire/Cable - MV	PULL WIRE IN CONDUIT		105.00 FEET	0.07	7.03	405.15			405.15
734 Transformers - MV	15MVA Transformer		1.00 ITEM(	2,276.90			389,950.00	389,950.00	389,950.00
715 Terminators/Connectors	350 KCMIL MV-TERM		6.00 EACH	0.17	1.00	57.57	4.07	24.40	81.97
722 Conduit Fittings	3.00 IN DIA FITTNGS		1.00 EACH	2.83	2.83	166.31	97.76	97.76	264.07
721 Conduit	3.00 IN DIA CONDUIT		25.00 FEET				7.68	192.07	192.07
721 Conduit	INSTALL CONDUIT		25.00 FEET	0.24	6.11	358.70			358.70
722 Conduit Fittings	3.00 IN DIA ELBOWS		1.00 EACH	1.11	1.11	65.22	30.31	30.31	95.53

Project Title: DR-Ramgen
Project Location: US
Job No: 78180
Estimate Date: 7AUG09 11:16:20

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Installation Details - Overall

Area / Component / Source / Account / Location / Code of Account	Tag No. / Description	Key Qty	Other Qty	Unit MH	MH	Labor Cost	Unit Mat	Matl Cost	Total Cost
15MVA Transformer(ID:18)	**XFMR-01**	**ID: 18**							
AG 722 Conduit Fittings	3.00 IN DIA UNIONS		2.00 EACH	1.06	2.11	123.91	61.07	122.13	246.04
717 Wire/Cable - MV	350 KCMIL 5000 V	105.00 FEET					6.82	716.00	716.00
722 Conduit Fittings	3.00 IN DIA COUPLNG		1.00 EACH	1.11	1.11	65.22	10.88	10.88	76.10
722 Conduit Fittings	3.00 IN DIA SEALS		2.00 EACH	2.78	5.56	326.09	67.88	135.76	461.85
734 Transformers - MV	15MVA Installation		1.00 EACH	40.00	40.00	2,340.83			2,340.83
BASIN									
446 Concrete Pour And Finish	SEAL SLAB - TYPE A CONC.		2.50 CY	3.61	9.02	364.64	65.28	163.20	527.84
446 Concrete Pour And Finish	POUR AND FINISH CONCRETE	26.25 CY		2.51	65.83	2,678.70			2,678.70
454 Formwork Materials	CONTACT SURFACE FORMWOR		875.00 SF				0.44	385.00	385.00
454 Formwork Materials	BRACING FORMWORK		1,625.00 BD FT				0.20	325.00	325.00
455 Field Fabricate Formwork	FORMWORK FABRICATION		875.00 SF	0.09	80.82	3,739.06			3,739.06
451 Rebar	REBAR INSTALLATION		1.75 TONS	30.34	53.09	2,491.92	860.00	1,505.00	3,996.92
444 Concrete	READY-MIX CONC. - TYPE C		26.25 CY				97.92	2,570.40	2,570.40
456 Install Formwork	FORMWORK INSTALLATION		875.00 SF	0.07	64.56	2,920.75			2,920.75
457 Strip & Clean Formwork	STRIP AND CLEAN FORMWORK		875.00 SF	0.03	26.44	1,196.45			1,196.45
EXCAV									
447 Excavation	HAND EXCAVATION		0.50 CY	4.56	2.28	79.15			79.15
447 Excavation	MACHINE EXCAVATION		24.50 CY	0.13	3.27	134.64			134.64
Component / Source Totals					**2,654**	**17,795**		**396,347**	**414,142**
6.6 KV Feeder Cable(ID:14)	**T-3 Cable**	**ID: 14**							
GENERAL									
UG 721 Conduit	5.00 IN DIA CONDUIT		200.00 FEET				5.62	1,124.85	1,124.85
AG 717 Wire/Cable - MV	4/0 AWG 5000 V	255.00 FEET					4.56	1,162.51	1,162.51
722 Conduit Fittings	2.50 IN DIA BUSHING		4.00 EACH	0.61	2.44	143.48	9.50	38.00	181.48
722 Conduit Fittings	2.50 IN DIA COUPLNG		3.00 EACH	0.89	2.67	156.52	8.34	25.02	181.54
722 Conduit Fittings	2.50 IN DIA SEALS		2.00 EACH	2.22	4.44	260.87	53.89	107.77	368.64
722 Conduit Fittings	2.50 IN DIA UNIONS		2.00 EACH	0.88	1.76	103.04	40.32	80.64	183.68
722 Conduit Fittings	2.50 IN DIA FITTNGS		1.00 EACH	2.44	2.44	143.48	75.42	75.42	218.90
722 Conduit Fittings	2.50 IN DIA ELBOWS		1.00 EACH	0.89	0.89	52.17	18.86	18.86	71.03
721 Conduit	INSTALL CONDUIT		75.00 FEET	0.22	16.67	978.27			978.27
721 Conduit	2.50 IN DIA CONDUIT		75.00 FEET				5.92	443.63	443.63
UG 712 Wire/Cable - HV	PULL WIRE IN CONDUIT		630.00 FEET	0.10	65.12	3,753.57			3,753.57
AG 717 Wire/Cable - MV	PULL WIRE IN CONDUIT		255.00 FEET	0.05	13.05	752.42			752.42
UG 712 Wire/Cable - HV	750 KCMIL 15000 V	630.00 FEET					13.11	8,261.15	8,261.15
404 Special Equip. Civil	Backhoe		1.00 EACH				1,000.00	1,000.00	1,000.00

Project Title: DR-Ramgen
Project Location: US
Job No: 78180
Estimate Date: 7AUG09 11:16:20

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Installation Details - Overall

Area / Component / Source / Account / Location	Code of Account	Tag No. / Description	Key Qty	Other Qty	Unit MH	MH	Labor Cost	Unit Mat	Matl Cost	Total Cost
6.6 KV Feeder Cable(ID:14)		**T-3 Cable**	**ID: 14**							
447	Excavation	Excavation		1.00 EACH	16.00	16.00	725.24			725.24
UG 761	Electrical Trenching	BACKFILL & COMPACT		44.44 CY	0.32	14.44	574.25			574.25
761	Electrical Trenching	CONDUIT ENVELOPE		11.11 CY	4.01	44.56	1,903.46	81.61	906.67	2,810.13
761	Electrical Trenching	EXCAVATE TRENCH		44.44 CY	0.30	13.46	558.40			558.40
722	Conduit Fittings	5.00 IN DIA COUPLNG		7.00 EACH	1.33	9.33	547.83	15.67	109.66	657.49
722	Conduit Fittings	5.00 IN DIA ELBOWS		2.00 EACH	1.24	2.47	144.95	61.03	122.06	267.01
721	Conduit	INSTALL BURIED CONDUIT		200.00 FEET	0.14	28.44	1,669.58			1,669.58
715	Terminators/Connectors	750 KCMIL HV-TERM		6.00 EACH	0.22	1.30	74.65	8.17	49.03	123.68
AG 715	Terminators/Connectors	4/0 AWG MV-TERM		6.00 EACH	0.13	0.78	45.23	1.98	11.86	57.09
Component / Source Totals						**240**	**12,587**		**13,537**	**26,125**
6.9kV VFD Drive(ID:21)		**VFD-01**	**ID: 21**							
GENERAL										
AG 714	Push Button Station	PUSHBUTTONS		2.00 EACH	1.67	3.33	186.53	93.08	186.16	372.69
260	Heat Exchangers	VFD Heat Exchanger		1.00 EACH	40.00	40.00	2,113.79	80,000.00	80,000.00	82,113.79
AG 731	Variable Frequency Drive	6.9kV VFD Drive		1.00 ITEM(	250.00	250.00	14,630.18			
713	Pilot Light	PILOT LIGHTS		2.00 EACH	3.34	6.67	373.07	105.49	210.98	584.05
EXCAV										
458	Backfill	EXCAVATED SOIL		2.00 CY	0.06	0.11	3.86			3.86
447	Excavation	MACHINE EXCAVATION		19.60 CY	0.14	2.65	109.15			109.15
447	Excavation	HAND EXCAVATION		0.40 CY	4.55	1.82	63.32			63.32
LINE 1		**Cooling for VFD**								
AG 312	CS Field Shop Fab	HANDLE & WELD PREP OPER.		75.00 FEET	0.17	13.02	767.69			767.69
912	Paint - Piping	2 FINAL COATS BY SPRAY	85.18 SF		0.09	7.30	310.61	0.71	60.49	371.10
AG 312	CS Field Shop Fab	FIELD SHOP X-RAY		2.40 EACH	0.99	2.37	139.75			139.75
317	CS Pipe Erection	WELDING		8.00 EACH	1.43	11.45	664.26			664.26
311	CS Field Mat'l	PIPE 2 IN SCH 40	75.00 FEET					4.82	361.55	361.55
312	CS Field Shop Fab	BEVELING PIPE		12.00 EACH	0.09	1.10	65.41			65.41
312	CS Field Shop Fab	WELDING		12.00 EACH	1.15	13.74	799.98			799.98
317	CS Pipe Erection	ERECT SHOP FAB PIPE		75.00 FEET	0.38	28.33	1,636.17			1,636.17
315	CS Valves: Flanged	BALL V 2 IN 150 CLASS		2.00 EACH				174.85	349.69	349.69
317	CS Pipe Erection	FIELD X-RAY		1.60 EACH	1.08	1.72	99.64			99.64
312	CS Field Shop Fab	CUTTING PIPE		7.00 EACH	0.12	0.83	49.06			49.06
317	CS Pipe Erection	ERECT VALVE		2.00 EACH	0.56	1.11	65.21			65.21
317	CS Pipe Erection	BOLT UP CONNECTIONS		4.00 EACH	0.78	3.11	184.05			184.05
912	Paint - Piping	1 PRIMER COAT BY SPRAY	85.18 SF		0.04	3.65	155.30	0.43	36.82	192.12

Report Printed At 11:17 am 8/7/09

Project Title: DR-Ramgen
Project Location: US
Job No: 78180
Estimate Date: 7AUG09 11:16:20

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Installation Details - Overall

Area / Component / Source / Account / Location / Code of Account		Tag No. / Description	Key Qty	Other Qty	Unit MH	MH	Labor Cost	Unit Mat	Matl Cost	Total Cost
6.9kV VFD Drive(ID:21)		**VFD-01**	**ID: 21**							
AG 311	CS Field Mat'l	BOLTS & NUTS 0.625 IN		16.00 EACH				0.53	8.52	8.52
922	Surface Prep - Piping	COMMERCIAL SAND BLAST		85.18 SF	0.03	2.84	118.67	0.03	2.62	121.29
AG 311	CS Field Mat'l	REDUCR 2 IN SCH 40		2.00 EACH				9.92	19.83	19.83
311	CS Field Mat'l	ELBOW 2 IN SCH 40		6.00 EACH				6.84	41.04	41.04
311	CS Field Mat'l	FLG WN 2 IN 150 CLASS		4.00 EACH				12.84	51.36	51.36
311	CS Field Mat'l	GASKET 2 IN 150 CLASS		4.00 EACH				1.88	7.53	7.53
366	Pipe Hangers, Shoes Etc.	ERECT PREFAB PIPE SUPP.		8.00 EACH	1.93	15.46	876.29	31.79	254.34	1,130.63
MOTOR										
AG 715	Terminators/Connectors	10 AWG LV-TERM		12.00 EACH	0.08	1.00	57.64	1.31	15.72	73.36
722	Conduit Fittings	0.75 IN DIA SEALS		4.00 EACH	0.67	2.67	156.52	11.31	45.25	201.77
711	Wire/Cable - LV	10 AWG 600 V	360.00 FEET					0.18	64.71	64.71
711	Wire/Cable - LV	PULL WIRE IN CONDUIT		360.00 FEET	0.01	3.60	207.52			207.52
722	Conduit Fittings	0.75 IN DIA COUPLNG		4.00 EACH	0.13	0.53	31.30	1.09	4.35	35.65
722	Conduit Fittings	0.75 IN DIA UNIONS		4.00 EACH	0.47	1.87	109.57	2.10	8.41	117.98
722	Conduit Fittings	0.75 IN DIA FITTNGS		2.00 EACH	0.78	1.56	91.30	8.12	16.24	107.54
722	Conduit Fittings	0.75 IN DIA ELBOWS		2.00 EACH	0.14	0.27	15.65	2.61	5.22	20.87
721	Conduit	INSTALL CONDUIT		100.00 FEET	0.14	14.44	847.83			847.83
721	Conduit	0.75 IN DIA CONDUIT		100.00 FEET				1.29	129.18	129.18
722	Conduit Fittings	0.75 IN DIA BUSHING		8.00 EACH	0.33	2.67	156.52	1.16	9.28	165.80
PUSHB										
AG 718	Wire/Cable - CV	14 AWG 600 V	240.00 FEET					0.09	21.91	21.91
722	Conduit Fittings	0.75 IN DIA FITTNGS		2.00 EACH	0.78	1.56	91.30	8.12	16.24	107.54
722	Conduit Fittings	0.75 IN DIA COUPLNG		4.00 EACH	0.13	0.53	31.30	1.09	4.35	35.65
722	Conduit Fittings	0.75 IN DIA BUSHING		8.00 EACH	0.33	2.67	156.52	1.16	9.28	165.80
722	Conduit Fittings	0.75 IN DIA SEALS		4.00 EACH	0.67	2.67	156.52	11.31	45.25	201.77
722	Conduit Fittings	0.75 IN DIA UNIONS		4.00 EACH	0.47	1.87	109.57	2.10	8.41	117.98
722	Conduit Fittings	0.75 IN DIA ELBOWS		2.00 EACH	0.14	0.27	15.65	2.61	5.22	20.87
721	Conduit	INSTALL CONDUIT		100.00 FEET	0.14	14.44	847.83			847.83
721	Conduit	0.75 IN DIA CONDUIT		100.00 FEET				1.29	129.18	129.18
718	Wire/Cable - CV	PULL WIRE IN CONDUIT		240.00 FEET	0.01	1.86	107.22			107.22
715	Terminators/Connectors	14 AWG CV-TERM		16.00 EACH	0.08	1.33	76.86	1.31	20.96	97.82
SLAB GRD										
451	Rebar	REBAR INSTALLATION		0.68 TONS	39.43	26.81	1,258.25	853.68	580.50	1,838.75
446	Concrete Pour And Finish	SEAL SLAB - TYPE A CONC.		1.44 CY	4.09	5.89	238.08	65.28	94.00	332.08
444	Concrete	READY-MIX CONC. - TYPE C		19.80 CY				97.92	1,938.82	1,938.82
446	Concrete Pour And Finish	POUR AND FINISH CONCRETE	19.80 CY		2.74	54.24	2,368.79			2,368.79

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Project Title: DR-Ramgen
Project Location: US
Job No: 78180
Estimate Date: 7AUG09 11:16:20

Prepared By: MAS
Est. Class: Budget
Currency: DOLLARS USD

Installation Details - Overall

Area / Component / Source / Account / Location / Code of Account	Tag No. / Description	Key Qty	Other Qty	Unit MH	MH	Labor Cost	Unit Mat	Matl Cost	Total Cost
6.9kV VFD Drive(ID:21)	**VFD-01**	**ID: 21**							
454 Formwork Materials	CONTACT SURFACE FORMWOR		90.00 SF				0.44	39.60	39.60
454 Formwork Materials	BRACING FORMWORK		180.00 BD FT				0.20	36.00	36.00
455 Field Fabricate Formwork	FORMWORK FABRICATION		90.00 SF	0.18	15.97	738.76			738.76
457 Strip & Clean Formwork	STRIP AND CLEAN FORMWORK		90.00 SF	0.03	3.06	138.45			138.45
421 Grade,Compact Base	GRADE AND COMPACT BASE		18.00 CY	0.29	5.20	204.90	15.60	280.80	485.70
456 Install Formwork	FORMWORK INSTALLATION		90.00 SF	0.16	14.81	670.09			670.09
Component / Source Totals					**592**	**32,296**		**1,515,120**	**1,547,416**
Electrical Building Mods(ID:15)	**BLDG-E**	**ID: 15**							
GENERAL									
472 Bldg Structure/Finishes	SUPERSTRUCTURE TOTAL		1.00 ITEM(	177.42	177.42	8,131.16	9,545.00	9,545.00	17,676.16
473 Building Furnishings	FITTING&FURNISHING TOTAL		1.00 ITEM(	74.88	74.88	3,882.99	6,336.00	6,336.00	10,218.99
Component / Source Totals					**252**	**12,014**		**15,881**	**27,895**
Project Direct Totals					**4,077**	**94,321**		**2,366,694**	**2,461,015**

Report Printed At 11 17 am 8/7/09

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PPENDIX 7.2

HAZOP and P&ID Review

Hazards Review

For the

High Pressure CO_2 Rampressor Test Facility

Dresser-Rand
Olean, New York

This Hazards Review has been prepared by Ronald J. O’Mara, P.E., P.C.

21-June-2010
Ref. Job Number C-1023

5813 Main Street . Williamsville, NY 14221 . Tel. (716) 634-9736 . Fax (716) 634-4912

Introduction

In order to compile this Hazards Review on the new High Pressure CO_2 Rampressor Test Facility, Dresser-Rand of Olean, New York contracted with the Ronald J. O'Mara Corporation of Williamsville, New York to lead the review and generate this report. The Hazards Review was done collaboratively between Ramgen of Washington State, ATSI Engineering Services of Amherst, New York and Dresser-Rand engineers and operating / maintenance personnel. A team was formed to facilitate and develop the Hazards Review of the new High Pressure CO_2 Rampressor Test Facility for Dresser-Rand of Olean, New York. Dresser-Rand and Ronald J. O'Mara, P.E., P.C. agreed on the Hazards Review assessment objective and approach.

The new High Pressure CO_2 Test Facility is designed to test the new Ramgen carbon dioxide rampressor.

A Hazards Review was deemed appropriate by Dresser-Rand for this new installation in order to mitigate the potential for undesirable consequences (e.g., personal injuries, environmental impacts, or catastrophic equipment damage).

From the Hazards Review, it is hoped that critical points in the design, operation and maintenance of the new high pressure CO_2 test facility will be apparent. By providing attention to these points and addressing the action items, risks may be minimized and the overall safety of the system improved.

The recommendations from this Hazards Review are listed in the section of this report entitled "Recommendations".

Scope of study

The scope of this review included the new process piping and equipment that begins with the CO2 truck unloading station and includes the following equipment and systems: the CO2 vapor tank and test loop, the vent system to atmosphere, instrument air to the rampressor and through to the atmosphere, the cooling tower and all its support piping and equipment, the leakage compressor and the performance bleed compressor. The scope of this review does not cover the rampressor itself, items outside of the test skid boundary or future items.

Hazop risk analysis method

The risk analysis used the Hazop method and covered all the P&ID generated nodes. The Hazop risk analysis follows a general scheme that can be described as follows:

- Describe the system under analysis.
- Identify loss scenarios (i.e. sequences of events leading up to potential or actual losses, incidents or accidents) in the form of hazards, potential productivity interruptions, asset damage events, environmental issues etc.
- Evaluate the risks of each loss scenario by determining the relative likelihood of each event, and the relative consequence of each event.
- Evaluate the currently planned controls, barriers and safeguards.
- Identify additional, potential controls, barriers and safeguards.

Note: Two sections, facility siting and human factor, do not have P&ID's and were reviewed using a freer-flowing question/response strategy.

In the current exercise, a select team from Dresser-Rand, Ramgen and ATSI accomplished these steps:

Define the operational system

The Hazards Review was scoped to review risks related to the new High Pressure CO_2 Rampressor Test Facility at the Dresser-Rand plant in Olean, New York. The Piping and Instrumentation Drawings were used and sections of the flow sheets were divided into operational "nodes" that were then reviewed for possible hazards. The process nodes were selected based on their "fit" in the operation. Node "size" was an important criterion. It was advantageous to have nodes that were small enough in size to allow for a clear understanding and large enough to reduce redundancies.

Identify the possible system hazards

This step postulated the maximum reasonable consequence of loss scenarios or failures (i.e. of circumstances leading up to or resulting in hazards). The consequences were classified as losses to equipment and health and safety of personnel.

Determine the level of risks

Risks associated with each step in the operational process were considered. This is achieved by considering the event frequency or probability, and the event severity or consequence.

The ranking system used is described below:
Risk is defined as the product of probability and consequence.

Probability categories

Probability categories were defined as follows:

F1 = Once Every 1000 years

F2 = Once Every 100 years

F3 = Once Every 10 years

F4 = Once Every year

Risk categories

Risk categories were defined as follows. Note that the Risk Categories can be a result of equipment damage (causing adverse economic impact), or injury to people.

S1= Very minimal damage, less than $10,000 damage, self treated injury

S2= Minimal damage, between $10,000 to $100,000 damage, reportable injury

S3= Damage causing less than one month downtime, between $100,000 to $1,000,000 damage, disabling injury

S4= Damage causing more than one month shutdown, between $1,000,000 damage, fatality or permanent disability

Risk categories

Risk categories were defined by combining the probability and consequence categories above according to a matrix of risk ranking as follows.

Safety/Environmental Category Names Cell Names	S1	S2	S3	S4
F4	**Optional (2)**	**Action (3)**	**Action (3)**	**Action (3)**
F3	**Optional (2)**	**Optional (2)**	**Action (3)**	**Action (3)**
F2	**None (1)**	**Optional (2)**	**Optional (2)**	**Optional (2)**
F1	**None (1)**	**None (1)**	**None (1)**	**Optional (2)**

Define and describe the system safeguards

This step identified existing controls and barriers, which could be used to manage the operational risk. Controls and barriers include engineering devices, operational methods and practice, management action and principles that the team agrees appropriate to consider.

Assess the adequacy of the controls

The adequacy of the nominated controls in terms of design devices, management and operational practices was reviewed by the team to ensure that additional scope for risk reduction has not been overlooked. If the controls are considered inadequate, recommendations to improve the situation are made.

Conclusions and Recommendations

The report is presented so that the Dresser-Rand Inc., Ramgen and ATSI can review and implement the recommendations established through the risk analysis.

Hazards Review Schedule

The risk analysis was conducted on June 10, 2010 and June 11, 2010 at Dresser-Rand offices with the selected risk review team participating in the exercise. Participants are listed below:

Participants

First Name	Last Name	Job Title	Company
Jerry	Williams	Test Engineer	Dresser-Rand
Don	Wehlage	Manager Test Engineer	Dresser-Rand
Mike	Weimer	Construction Manager	Ramgen
Matthew	Weeks	Co-op	Dresser-Rand
H. Allan	Kidd	Emerging Tech Director	Dresser-Rand
Tony	Giardini	Test Engineer	Dresser-Rand
Mike	Johnson	Maint / Fac Manager	Dresser-Rand
Donna	McIntyre	HSE Manager	Dresser-Rand
Jim	Wilson	Facilitator	Ronald J. O'Mara
Karl	Guntheroth	Engineer	Ramgen
George	Talabisco	Principal Engineer	Dresser-Rand
John	Beers	PM	Ramgen
Susie	Shimamoto	Program Manager	Dresser-Rand
Bruce	Hudson	Supply Chain Manager	Dresser-Rand
Kyle	Whiteside	Product R&D Engineer	Dresser-Rand
Mark	Schiffhauer	ATSI - PM	ATSI
Joe	Williams	Chief Engineer (via telecom)	Ramgen

First Name	Last Name	Job Title	Company
Moulay	Belhassan	Aero Supervisor	Dresser-Rand
Greg	Stubbs	Manager - HSE	Dresser-Rand
Charles	Rohrs	Product Design Engineer	Dresser-Rand

Recommendations

Type	No.	Action	Responsibility	Due Date
Recomm	1.2.1	Transmitters to be added to P&ID with a shut-down on low pressure	ATSI/MAS	7/30/10
Recomm	1.2.2	Document operational procedures (pre-start checklists, etc.)	Ramgen/JB	12/31/10
Recomm	1.2.3	PCV-610 to be designed to fail open	ATSI/MAS	7/30/10
Recomm	1.2.4	F-006 to have PDI at a minimum	ATSI/MAS	7/30/10
Recomm	1.2.5	Add emergency back-up LO pump to P&ID	ATSI/MAS	7/30/10
Recomm	1.4.1	Add to lube oil reservoir LSL and LSLL with shut-down.	ATSI/MAS	7/30/10
Recomm	1.4.2	Add local level indicator accessible to operator	ATSI/MAS	7/30/10
Recomm	1.4.3	Add to SOPT (Standard Operating Procedures and Training) that the operator must check reservoir level between tests.	Ramgen/JB	12/31/10
Recomm	1.6.1	Show existing thermostat on temperature control loop	ATSI/MAS	7/30/10
Recomm	1.6.2	Add to P&ID permissive-to-start notations (for pump and compressor)	Ramgen/JB	12/30/10
Recomm	1.6.3	SOPT will not allow running with oil temperature less than 60 degrees F	Ramgen/JB	12/30/10
Recomm	1.7.1	Develop SOPT commissioning procedures	Ramgen/JB	12/30/10
Recomm	1.7.2	Update P&ID to show capacity of lube oil tank	ATSI/MAS	7/30/10
Recomm	1.7.3	Add local level indicator accessible to operator	ATSI/MAS	7/30/10
Recomm	1.9	Heat exchanger C002 to be designed to 150 PSIG	ATSI / Mark Schiffhauer	7/30/10
Recomm	1.11	Select hoses based on design guides	Ramgen/JB	12/31/10
Recomm	2.2.1	Perform Control Objectives Analysis including Control Philosophy	Ramgen/KG	10/1/10
Recomm	2.2.2	Review Control Objectives Analysis	D-R/GT, AG, and DW	10/1/10
Recomm	2.6	Motor that drives the pump that pressurizes the cooling water must have winter logic.	ATSI/MAS	10/1/10
Recomm	2.9	Check consequences with Ramgen Aerodesigners.	Ramgen/KG	7/30/10
Recomm	2.11	Institute final inspection prior to initial start-up (SOPT).	Ramgen/JB	12/31/10
Recomm	2.14.1	Consider adding relief valve downstream of PV004 (due to concern with seeing discharge pressure on low-pressure side of PV004).	Ramgen/KG and ATSI/MAS	7/30/10
Recomm	2.14.2	Review control function of PV004	Ramgen/KG and ATSI/MAS	7/30/10
Recomm	3.3	Update P&ID to add check valves upstream of valves 5 and 6	ATSI/MAS	7/30/10
Recomm	3.9	SOPT to include training and warnings against inadvertent valve closing	Ramgen/JB	12/31/10
Recomm	3.9.1	SOPT to include training to address CO2 leakage concerns	Ramgen/JB	12/31/10
Recomm	3.11	Consider interface fitting only compatible with CO2 and unload signage	Ramgen/JB	12/31/10
Recomm	4.5.1	Add to HS106 operator indicator of pump status on DCS	ATSI/MAS	7/30/10
Recomm	4.5.2	Consider more preventive maintenance, redundancy, and having spare parts on-hand (particularly a spare pump)	Ramgen/JB	12/31/10
Recomm	4.6	Incorporate winter logic into motor/heater operation	ATSI/MAS	7/30/10
Recomm	4.7.1	Add level indicator	Ramgen/JB & ATSI/MAS	7/30/10
Recomm	4.7.2	Add to SOPT to periodically inspect tower	Ramgen/JB	12/31/10
Recomm	4.7.3	Add to Maintenance Manual to annually replace float switch	Ramgen/JB	12/31/10

Type	No.	Action	Responsibility	Due Date
Recomm	4.8.1	Add to HSL/H106 operator indicator of pump status on DCS	ATSI/MAS	7/30/10
Recomm	4.9.1	Consider more preventive maintenance, redundancy, and having spare parts on-hand (particularly a spare pump)	Ramgen/JB	12/30/10
Recomm	4.9.2	Add low amp cut-off in pump	ATSI/MAS	7/30/10
Recomm	4.9.1.1	Consider adding relief valve downstream of PV004 (due to concern with seeing discharge pressure on low pressure side of PV004)	Ramgen/KG and ATSI/MAS	7/30/10
Recomm	4.11.1	Implement existing D-R test stand procedures into Ramgen test facility	Ramgen/JB and D-R/AG	12/31/10
Recomm	4.11.2	Analyze source of contamination in ICS Test Facility	D-R/DW	7/30/10
Recomm	4.11.3	Review selection of particulate filter	Ramgen/JB	7/30/10
Recomm	5.1	Incorporate pressure transmitter into design	ATSI/MAS	7/31/10
Recomm	5.2	Incorporate pressure transmitter into design	ATSI/MAS	7/31/10
Recomm	6.2	Acoustic testing will be performed.	Ramgen/JB	4/1/2011
Recomm	6.3	Review and approve safety procedures associated with the test gas of CO2.	Ramgen/JB and D-R/HAK	1/31/2011
Recomm	6.6.1	Construction plot plan is needed to show details.	ATSI / MAS	6/11/10
Recomm	6.8	Review and comment on plan for CO2 delivery system and tankage.	Ramgen/KG	6/11/10
Recomm	6.9.1	Provide parking and access to the facility	Dresser-Rand/Ed Wilber	8/1/10
Recomm	6.11	Review potential hazards from plant design prior to commissioning.	GC/tbd, Ramgen/JB & MW, and D-R/AG, GS, DM	12/31/10
Recomm	6.11.1	Periodically assess compliance of Construction contractors adherence to D-R's HSE policy.	GC/tbd, Ramgen/JB & MW, and D-R/AG	6/11/10 start date and on-going
Recomm	6.12	Upgrade emergency response plans to indicate escape routes.	D-R/DM	7/30/10
Recomm	6.16	Include transmission of summary alarm(s) to the central station. Review team to meet to make specific design recommendations.	D-R/AG	7/30/10
Recomm	6.17.1.1	Ensure compliance with applicable exposure limit	ASTI/MAS	8/31/10
Recomm	6.17.1.2	Ensure communication of potentially hazardous situation	D-R/DW	4/1/11 and on-going
Recomm	6.17.1.3	Consider scheduling facility open house to mitigate curious	Ramgen/JB	4/1/11
Recomm	6.18	Make sure the contractors have tool box safety discussions that include lessons learned from D-R	D-R/AG & Ramgen/MEW	Start on 7/01/2010
Recomm	7.1	Schedule follow-up discussion and consider updating P&IDs.	Ramgen/JB	2/1/2011
Recomm	7.2	Review all alarms and S/Ds and categories	Ramgen/JB	9/1/2010
Recomm	7.3	Develop commissioning procedures	Ramgen/JB	10/1/2010
Recomm	7.4	Develop S/D and S/U procedures	Ramgen/JB	9/1/2010
Recomm	7.5	Show control schemes on the P&ID's	Ramgen/JB	9/1/2010
Recomm	7.6	Schedule separate design review of back-up electrical design system.	Ramgen/JB	9/1/2010
Recomm	7.8	Consider insulating water meter box against low ambient temperatures	ATSI/MAS	7/1/10
Recomm	7.9.1	Ramgen to identify hardware requiring LOTO.	Ramgen/KG	12/31/10
Recomm	7.9.2	D-R to provide Ramgen with LOTO procedure	D-R/DM to provide to AG for transmittal to Ramgen	6/15/10
Recomm	7.9.3	Ramgen to review D-R procedure for LOTO	Ramgen/KG	12/31/10

Type	No.	Action	Responsibility	Due Date
Recomm	7.11	Finalize sparing philosophy.	Ramgen/JB	10/10/10
Recomm	7.12	Update P&ID Sheets 3 and Sheet 8, vent system and relief valves need to be sized.	ATSI/MAS	8/31/10

Node and Section Divisions

Item No.	Description	Drawings
1	Lube oil from the lube skid through the rampressor, the gearbox and the common motor. Lube oil returns back to the tank skid.	D-78180-09-002 Rev G and D-78180-09-010 Rev C.
2	From CO2 truck unloading station, through the CO2 vapor tank and into the test loop equipment. Covers the test loop equipment and the bulk of the vent system to atmosphere.	D-78180-09-004 Rev J, D-78180-09-008 Rev C.
3	Covers from plant instrument air, to the rampressor and through to the vent system to atmosphere.	D-78180-09-003 Rev J, D-78180-09-006 Rev H and D-78180-09-008 Rev C.
4	Cooling tower, cooling water supply (CWS) system, CWR system and the leakage compressors.	D-78180-09-005 Rev J, D-78180-09-007 Rev G.
5	Performance Bleed Compressor.	D-78180-09-009 Rev C.
6	Facility Siting.	N/A.
7	Human Factor.	N/A.

INSTRUMENT IDENTIFICATION

(ABC 123Z)

A	FIRST LETTER – MEASURED OR INITIATING VARIABLE
B	MODIFIER LETTER
C	LAST LETTER – READOUT/OUTPUT/MODIFER FUNCTION
123	INDIVIDUAL LOOP OR INSTRUMENT NUMBER
Z	USED TO DENOTE MORE THAN ONE IDENTICAL INSTRUMENT IN THE SAME LOOP

INSTRUMENT IDENTIFICATION LETTERS

	FIRST LETTER	MODIFIER LETTER	LAST LETTER
A	ANALYSIS		ALARM
B	BURNER		
C	CONDUCTIVITY	CLOSED	CONTROL
D	DENSITY	DIFFERENTIAL	
E	VOLTAGE/EMF		PRIMARY ELEMENT
F	FLOW	RATIO/BIAS	
G	GAGING		GLASS
H	HAND		HIGH
I	CURRENT	INDICATE	INDICATE
J	POWER	SCAN	
K	TIME	TIME RATE OF CHANGE	CONTROL STATION
L	LEVEL		LOW
M	MOISTURE/HUMIDITY	MOMENTARY	MIDDLE/INTERMEDIATE
N			
O	OPACITY	OPEN	ORIFICE/RESTRICTION
P	PRESSURE/VACUUM		POINT CONNECTION
Q	QUANTITY/EVENT	INTEGRATE/TOTALIZE	
R	RADIOACTIVITY		RECORD
S	SPEED/FREQUENCY	SAFETY	SWITCH
T	TEMPERATURE		TRANSMITTER
U	MULTI-VARIABLE	MULTI-FUNCTION	MULTI-FUNCTION
V	VIBRATION		VALVE
W	WEIGHT		WELL
X		X AXIS	CONTROL SET POINT
Y	EVENT/STATE	Y AXIS	COMPUTE
Z	POSITION STATUS	Z AXIS	DRIVER/ACTUATOR

VALVE BODY IDENTIFICATION

X"-XX
- VALVE BODY STYLE
- LINE SIZE – IN INCHES

ABBREV	DESCRIPTION
BA	BALL VALVE
BB	BLOCK AND BLEED
BU	BUTTERFLY VALVE
CH	CHECK
GA	GATE VALVE
GL	GLOBE VALVE

LINE LIST IDENTIFICATION

X-X" – AAA – BBB – CCC

- BARE, INSULATED OR TRACED
- LINE SIZE – IN INCHES
- PROCESS FLUID–PIPE SPEC
- LINE NUMBER–LINE LIST
- MATERIAL SPEC–PIPE SPEC

PIPE SPEC CHANGE

INSTRUMENT LINES

- INSTRUMENT SUPPLY OR CONNECTION TO PROCESS
- PNEUMATIC SIGNAL
- ELECTRICAL SIGNAL
- HYDRAULIC SIGNAL
- CAPILLARY TUBE
- SOFTWARE OR DATA LINK
- MECHANICAL LINK

PROCESS LINES

- MAJOR PROCESS LINES
- SECONDARY PROCESS LINES
- PROCESS FLOW DIRECTION
- PROCESS LINE CONNECTION TO ANOTHER DRAWING
- EXISTING MAJOR PROCESS LINE
- EXISTING SECONDARY PROCESS LINE
- PROCESS LINE FLANGE
- TP FIELD TIE POINT

MISC ABBREVIATIONS

ABBREV	DESCRIPTION
ATM	ATMOSPHERE
B/L	BATTERY LIMIT
CONN	CONNECTION
CSC	CAR SEAL CLOSED
CSO	CAR SEAL OPEN
DC	DRAIN CONNECTION
FV	FULL VACUUM
HC	HOSE CONNECTION
HP	HIGH POINT
MAX	MAXIMUM
MIN	MINIMUM
NC	NORMALLY CLOSED
NLL	NORMAL LIQUID LEVEL
NNF	NORMALLY NO FLOW
NO	NORMALLY OPEN
PC	PURGE CONNECTION
SP	SPECIAL EQUIPMENT
RAM	RAMPRESSOR CONTROL PANEL
TYP	TYPICAL
VAC	VACUUM
VC	VENT CONNECTION

INSTRUMENTATION ABBREVIATIONS

ABBREV	DESCRIPTION
AI	ANALOG INPUT
AO	ANALOG OUTPUT
ATO	AIR TO OPEN
ATC	AIR TO CLOSE
DI	DIGITAL INPUT
DO	DIGITAL OUTPUT
ES	EMERGENCY STOP/ELECTRONIC SWITCH
FC	FAIL CLOSED
FL	FAIL LAST POSITION
FO	FAIL OPEN
HOA	HAND OFF AUTO
MCC	MOTOR CONTROL CENTER
MS	MINIMUM STOP
RTD	RESISTANCE TEMPERATURE DETECTOR
TC	THERMOCOUPLE

PRIMARY ELEMENT SYMBOLS

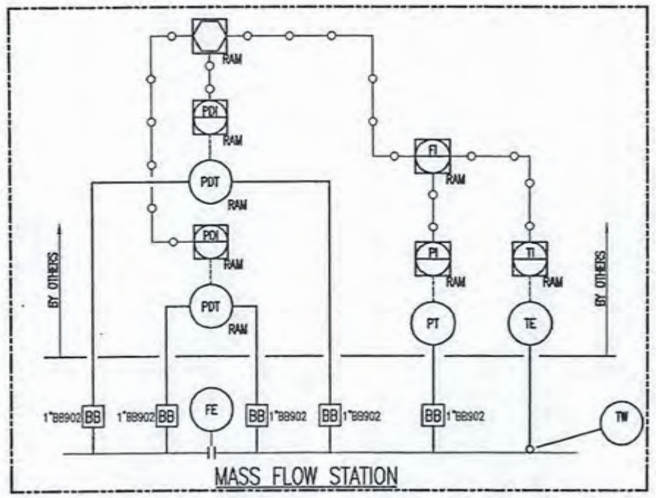
SYMBOL	DESCRIPTION
FE	ORIFICE FLOW METER
RO	FLOW RESTRICTION ORIFICE
FE VORTEX	VORTEX SENSOR
FE MAG	MAGNETIC FLOW METER
FE SONIC	SONIC FLOW METER
FE PITOT	PITOT TUBE
TE T/C OR RTD	TEMPERATURE ELEMENT WITHOUT WELL
TE T/C OR RTD	TEMPERATURE ELEMENT WITH WELL
SG	SIGHT GLASS

VALVES & FITTINGS

LINE REDUCER	TWO-WAY SOLENOID VALVE
PIPE CAP	
BLIND FLANGE	THREE-WAY SOLENOID VALVE
"Y" STRAINER	
LINE BLIND	ELECTRIC MOTOR OPERATED VALVE
CHECK VALVE	
DOUBLE-ACTING CYLINDER OPERATING VALVE	RUPTURE DISK (PRESSURE)
SELF-CONTAINED PRESSURE REGULATOR (EXTERNAL TAP)	RUPTURE DISK (VACUUM)
SELF-CONTAINED BACK PRESSURE REGULATOR	CONSERVATION VENT VALVE
SELF-CONTAINED PRESSURE REGULATOR (INTERNAL TAP)	RELIEF VALVE (PRESSURE)
DIAPHRAGM OPERATED CONTROL VALVE	RELIEF VALVE (VACUUM)
PNEUMATIC DRIVE ACTUATED VALVE	DIAPHRAM SEAL

PLC/DISTRIBUTED CONTROL/INSTRUMENTATION SYMBOLS

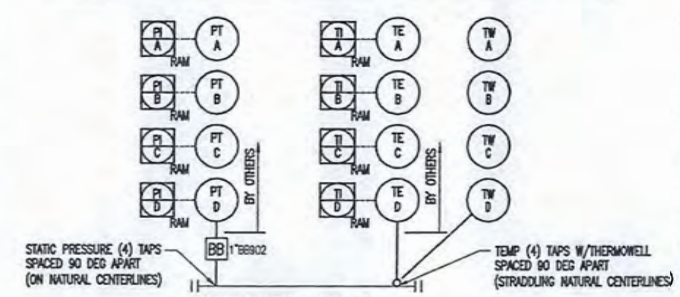
SYMBOL	DESCRIPTION
	NORMALLY ACCESSIBLE TO OPERATOR
	INDICATOR/CONTROLLER/RECORDER OR ALARM POINTS USUALLY USED TO INDICATE VIDEO DISPLAY.
	AUXILLIARY OPERATOR'S INTERFACE DEVICE
	DISTRIBUTED CONTROL INTERCONNECTING LOGIC CONTROLLER PLC, OR ANY PC WITH BINARY OR SEQUENTIAL LOGIC FUNCTIONS.
XIC XAH XAL	OUTPUT ALARMS XAH (HIGH) XAL (LOW)
OLC SP ST	OLC – OPERATOR LIGHT CONTROLLER OA – OPERATION ALARM SP – STOP FUNCTION FROM CRT ST – START FUNCTION FROM CRT H – HAND OPERATED BY LOCAL START/STOP STATION O – OFF A – AUTOMATIC OPERATION
	NOT NORMALLY ACCESSIBLE TO OPERATOR
	NORMALLY BLIND OPERATION– DIGITAL CONTROLLER
I 7	INTERCONNECTING LOGIC CONTROLLER PLC OR ANY PC WITH BINARY OR SEQUENTIAL LOGIC FUNCTIONS NUMBER IN DIAMOND REFERS TO INTERLOCK NUMBER.
	COMPUTER/ INDICATOR/ CONTROLLER/ RECORDER OR ALARM POINTS USUALLY USED TO INDICATE VIDEO DISPLAY
	HARDWIRE INTERLOCK
	NOT NORMALLY ACCESSIBLE TO OPERATOR
	MAY BE USED AS A BLIND CONTROLLER OR A SOFTWARE CALCULATION MODULE
	DISCRETE INSTRUMENT (FIELD)
	INSTRUMENT (LOCAL CONTROL PANEL MOUNTED)
	INSTRUMENT (AUXILIARY CONTROL PANEL LOCATED ELSEWHERE)
	INSTRUMENT (BEHIND PANEL)
	IN LINE INSTRUMENT AS IDENTIFIED
	INDICATION LIGHT LOCAL PANEL MOUNTED



MASS FLOW STATION

INSTRUMENT SPOOL IDENTIFICATION

SP- No.	PT	PT	PT	PT	TE/TW	TE/TW	TE/TW	TE/TW
SP-001	134A	134B	134C	134D	134A	134B	134C	134D
SP-002	135A	135B	135C	135D	135A	135B	135C	135D
SP-003	104A	104B	104C	104D	104A	104B	104C	104D
SP-004	105A	105B	105C	105D	105A	105B	105C	105D
SP-005	106A	106B	106C	106D	106A	106B	106C	106D
SP-006	107A	107B	107C	107D	107A	107B	107C	107D
SP-007	108A	108B	108C	108D	108A	108B	108C	108D
SP-008	109A	109B	109C	109D	109A	109B	109C	109D
SP-009	110A	110B	110C	110D	110A	110B	110C	110D
SP-010	111A	111B	111C	111D	111A	111B	111C	111D
SP-011	112A	112B	112C	112D	112A	112B	112C	112D
SP-012	113A	113B	113C	113D	113A	113B	113C	113D
SP-013	114A	114B	114C	114D	114A	114B	114C	114D



INSTRUMENT SPOOL

PRELIMINARY
UNCHECKED
NOT TO BE USED
FOR CONSTRUCTION

REV	DATE	DESCRIPTION	BY
C			
B			
A	3/2/10	REVIEW AND COMMENT	ATSI

REVISIONS PRIOR TO CONSTRUCTION RELEASE

EQUIPMENT ① ② ③ ④ ⑤ ⑥ ⑦ ⑧ ⑨ ⑩ ⑪ ⑫
PLANT ID.
NAME
P.O.NO. / DWG.NO. / MATERIAL / SIZE / CAPACITY / TEMP/PRESS / HP/V/RPM / WEIGHT / VENDOR

PROJECT	REV.	DESCRIPTION	MADE BY	APP BY	DATE
–	–	WORK IN PROGRESS	–	–	–

REVISIONS

ATSI ENGINEERING SERVICES, www.atsiinc.com, ISO 9001:2008 Certified

DATE 12/7/09 · DRAWN TJD · DESIGN M. SCHIFFHAUER · SCALE NONE · CHECKED · APPROVED · CAD FILE 7818009001 · ACT. NO. 0325 · CONTRACT NO. 78181 · DRAWING NUMBER D-78180-09-001 · REV. A

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DRAWN BY: T.DAUPHINEE DATE: 2/1/10
DESIGN BY: 2/1/10 DATE: 2/1/10
CHECKED BY: · PROJECT DESIGN COORDINATOR · PROJECT ENGINEER · PROJECT MANAGER

FILE NAME: 7818109001.DWG

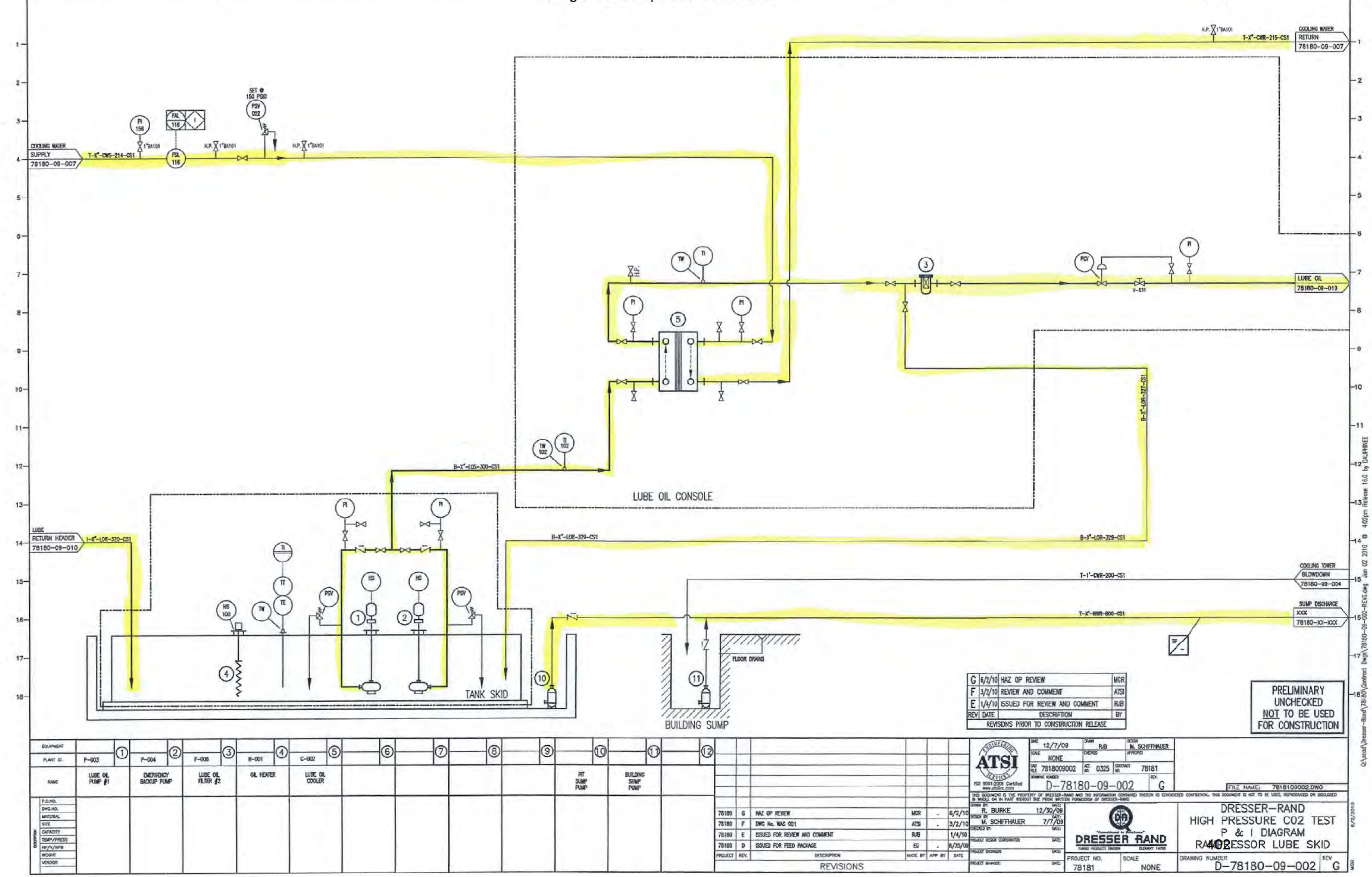
DRESSER-RAND, TURBO PRODUCTS DIVISION, OLEAN, NY 14760

DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
LEGEND

PROJECT NO. 78181 · SCALE NONE · DRAWING NUMBER D-78180-09-001 · REV A

Q:\acad\Dresser-Rand\78180\Contract Dwgs\78180-09-001-REVA.dwg Jun 02 2010 @ 4:02pm Release 16.0 by DAUPHINEE

3/4/10 DAUPHINEE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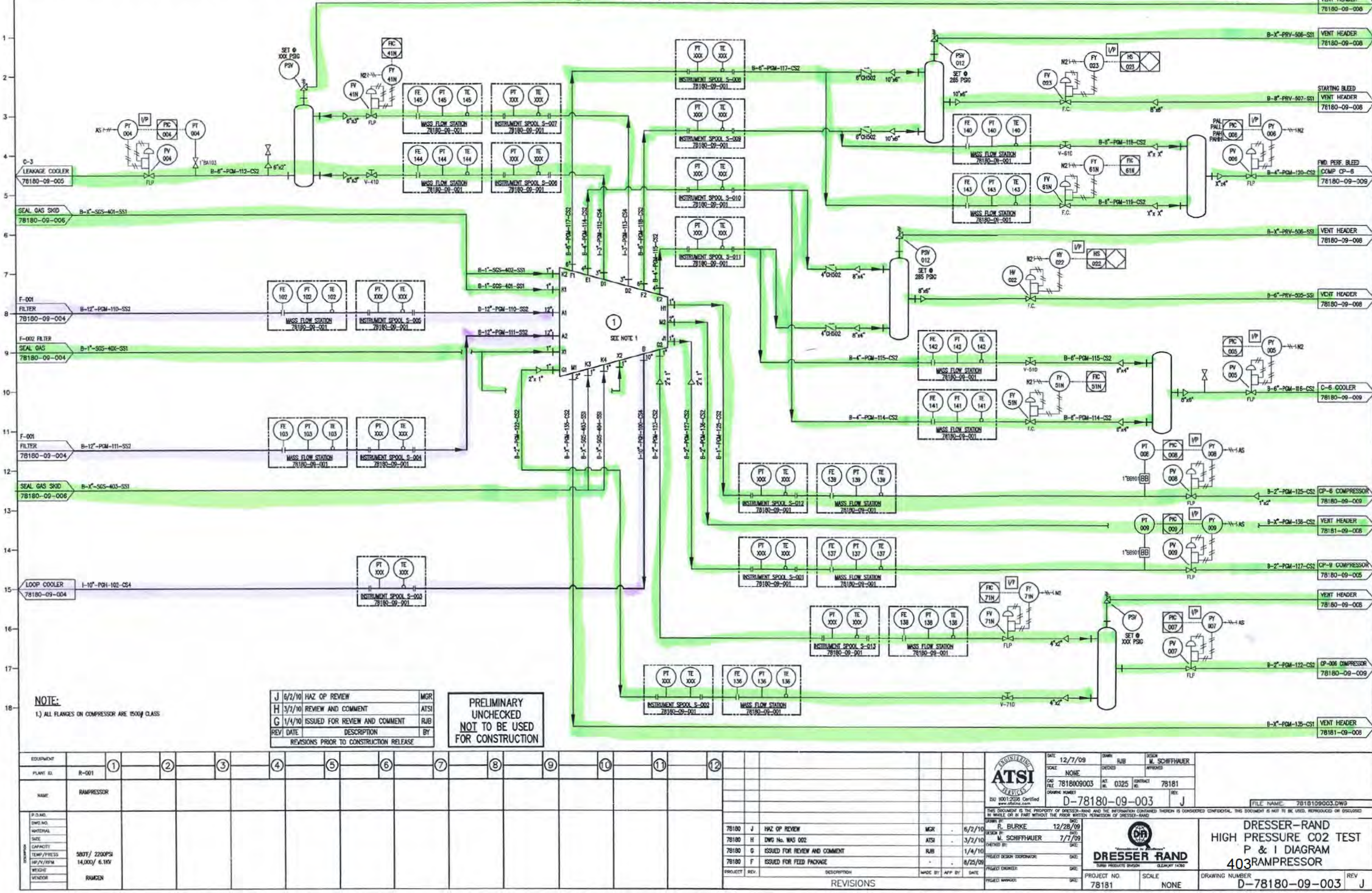
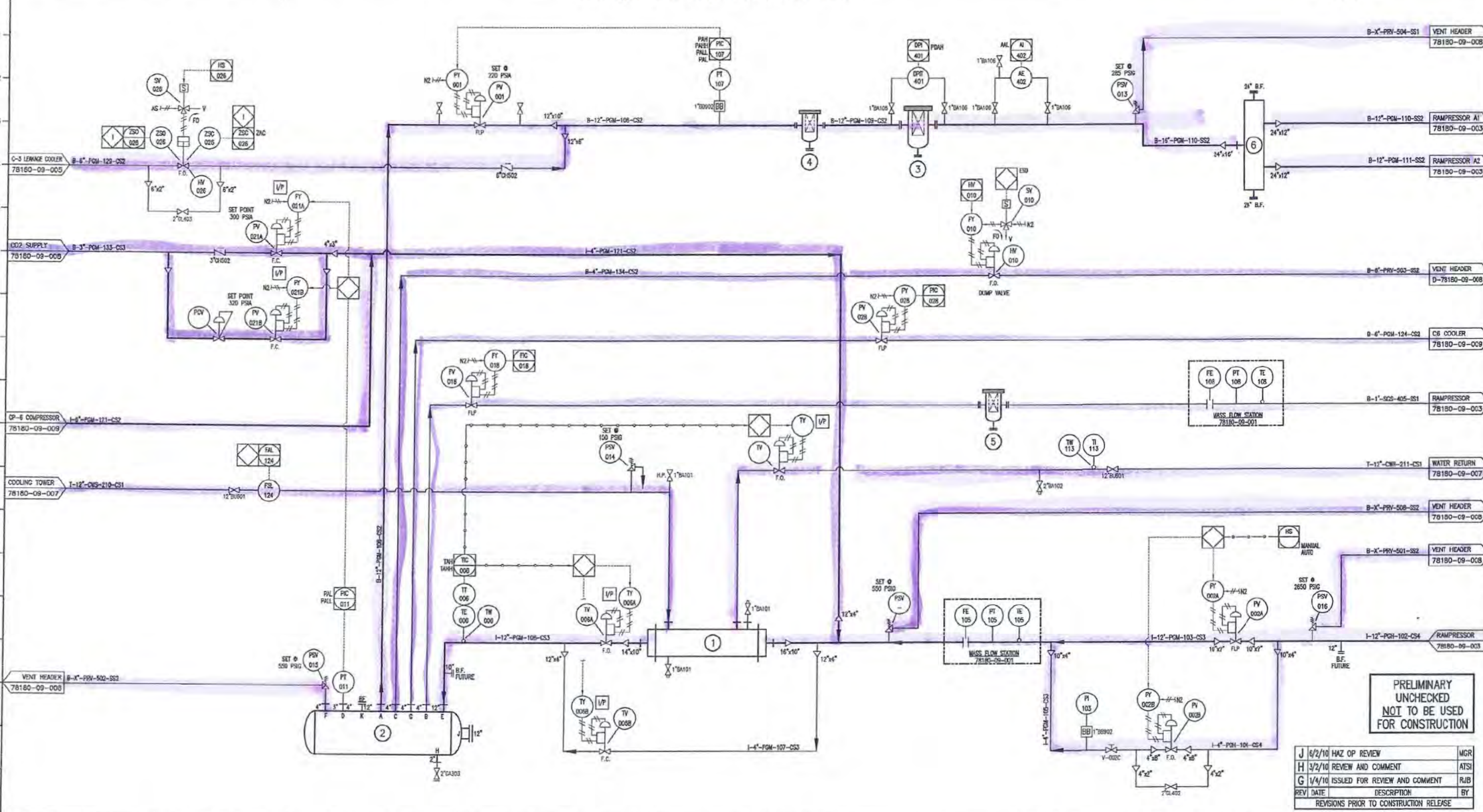
T-X"-CWR-215-CS1
COOLING WATER SUPPLY 78180-09-007
T-X"-CWS-214-CS1
SET @ 150 PSIG
PSV 022
FAL 116
FSL 116
PI 156
LUBE OIL 78180-09-010
LUBE OIL CONSOLE
B-X"-LOS-300-CS1
B-X"-LOR-327-CS1
B-X"-LOR-329-CS1
LUBE RETURN HEADER 78180-09-010
I-6"-LOR-320-CS1
TANK SKID
COOLING TOWER BLOWDOWN 78180-09-004
T-1"-CWR-200-CS1
SUMP DISCHARGE XXX 78180-XX-XXX
T-X"-WWR-800-CS1
FLOOR DRAINS
BUILDING SUMP
G 6/2/10 HAZ OP REVIEW MCR
F 3/2/10 REVIEW AND COMMENT ATSI
E 1/4/10 ISSUED FOR REVIEW AND COMMENT RJB
REV DATE DESCRIPTION BY
REVISIONS PRIOR TO CONSTRUCTION RELEASE
PRELIMINARY UNCHECKED NOT TO BE USED FOR CONSTRUCTION
EQUIPMENT
PLANT ID. P-003 P-004 F-006 H-001 C-002
NAME LUBE OIL PUMP #1 EMERGENCY BACKUP PUMP LUBE OIL FILTER #2 OIL HEATER LUBE OIL COOLER PIT SUMP PUMP BUILDING SUMP PUMP
78180 G HAZ OP REVIEW MCR 6/2/10
78180 F DWG No. WAS 001 ATSI 3/2/10
78180 E ISSUED FOR REVIEW AND COMMENT RJB 1/4/10
78180 D ISSUED FOR FEED PACKAGE EG 8/25/09
REVISIONS
ATSI
12/7/09 RJB M. SCHIFFHAUER
NONE
7818009002 0325 78181
D-78180-09-002 G
R. BURKE 12/30/09
M. SCHIFFHAUER 7/7/09
DRESSER-RAND
FILE NAME: 7818109002.DWG
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
RAMPRESSOR LUBE SKID
PROJECT NO. 78181
SCALE NONE
DRAWING NUMBER D-78180-09-002
REV G


NOTE:
1.) ALL FLANGES ON COMPRESSOR ARE 1500# CLASS
PRELIMINARY
UNCHECKED
NOT TO BE USED
FOR CONSTRUCTION
J 6/2/10 HAZ OP REVIEW MGR
H 3/2/10 REVIEW AND COMMENT ATSI
G 1/4/10 ISSUED FOR REVIEW AND COMMENT RJB
REV DATE DESCRIPTION BY
REVISIONS PRIOR TO CONSTRUCTION RELEASE
LEAKAGE COOLER 78180-09-005
SEAL GAS SKID 78180-09-006
FILTER 78180-09-004
LOOP COOLER 78180-09-004
VENT HEADER 78180-09-008
STARTING BLEED
C-6 COOLER 78180-09-009
CP-6 COMPRESSOR 78180-09-009
CP-9 COMPRESSOR 78180-09-005
MASS FLOW STATION 78180-09-001
INSTRUMENT SPOOL
SEE NOTE 1
PLANT ID. R-001
NAME RAMPRESSOR
580T/ 2200PSI
14,000/ 6.1KV
RAMGEN
ATSI
D-78180-09-003
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
RAMPRESSOR
DRAWING NUMBER D-78180-09-003 REV J
FILE NAME: 7818109003.DWG
REVISIONS



REV	DATE	DESCRIPTION	BY
J	6/2/10	HAZ OP REVIEW	MGR
H	3/2/10	REVIEW AND COMMENT	ATSI
G	1/4/10	ISSUED FOR REVIEW AND COMMENT	RJB

REVISIONS PRIOR TO CONSTRUCTION RELEASE

EQUIPMENT	1	2	3	4	5	6	7	8	9	10	11	12
PLANT ID.	C-001	T-001	F-001	S-001	F-002	P-001						
NAME	CORE CO2 LOOP COOLER	PLENUM TANK	RAMPRESSOR INLET FILTER	RAMPRESSOR INLET STRAINER	SEAL GAS FILTER	SUCTION GAS COLLECTOR						
P.O. NO.												
DWG. NO.												
MATERIAL	CS SHELL/SS TUBE	CARBON STEEL				STAINLESS STEEL						
SIZE	30" OD x 22'	5' OD x 12' SS										
CAPACITY	36.2 MBTU/HR	2,000 GAL										
TEMP/PRESS	650F/550 PSI	350F/550 PSI										
HP/V/RPM												
WEIGHT	21,350 LBS	12,150 LBS										
VENDOR	J.D. COUSINS	J.D. COUSINS										
	U-106											

PROJECT	REV.	DESCRIPTION	MADE BY	APP BY	DATE
78180	J	HAZ OP REVIEW	MGR	.	06/2/10
78180	H	DWG No. WAS 003	ATSI	.	03/02/10
78180	G	ISSUED FOR REVIEW AND COMMENT	RJB	.	01/04/10
78180	F	ISSUED FOR FEED PACKAGE	EG	.	8/25/09

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DATE 12/7/09	DRAWN RJB	DESIGN M. SCHIFFHAUER	
SCALE NONE	CHECKED	APPROVED	
CAD FILE 7818009004	ACL NO. 0325	CONTRACT NO. 78181	
DRAWING NUMBER D-78180-09-004			REV. J

FILE NAME: 7818109004.DWG

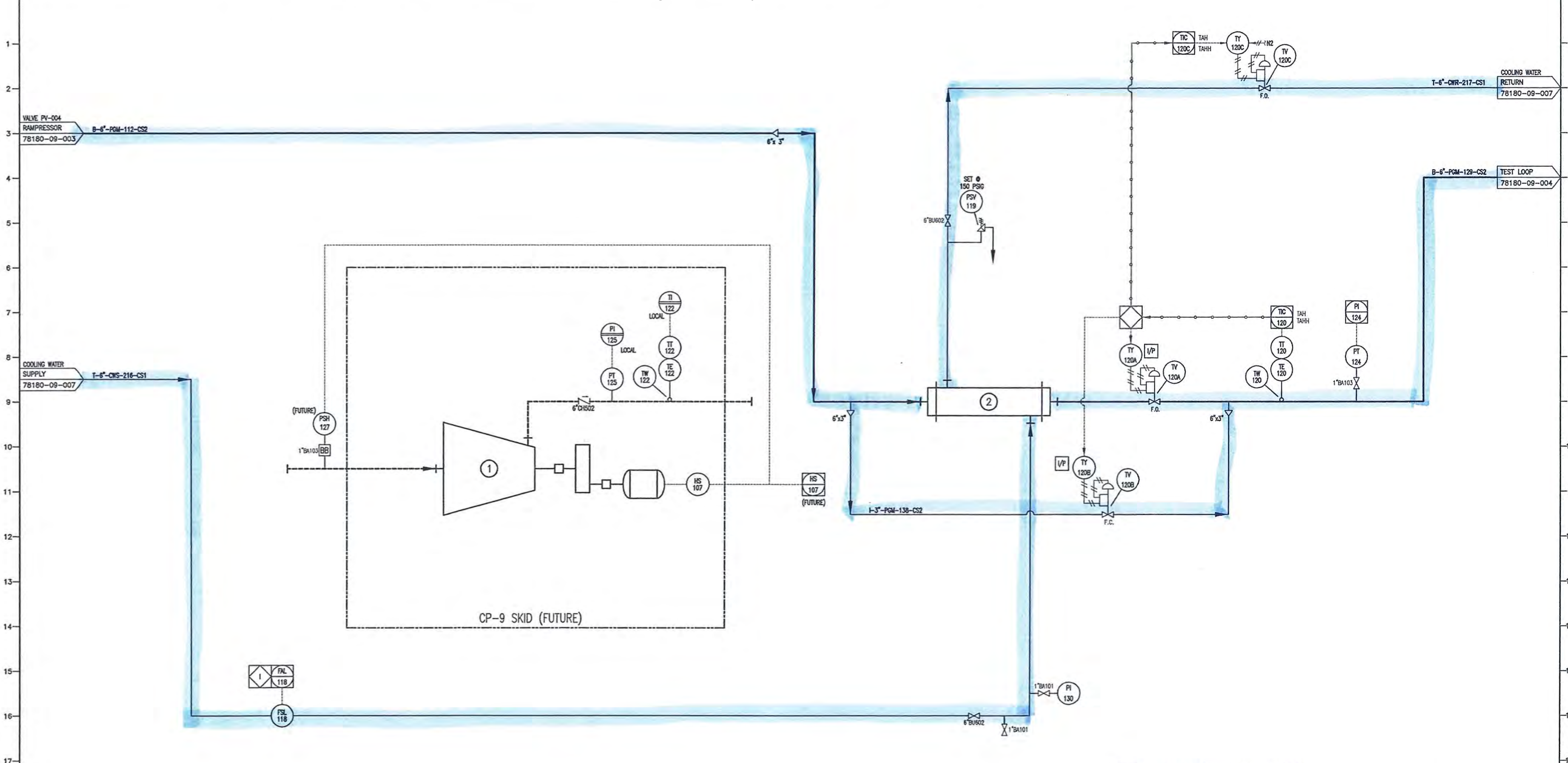
DRAWN BY: R. BURKE	DATE: 12/28/09
DESIGN BY: M. SCHIFFHAUER	DATE: 7/7/09
CHECKED BY:	DATE:
PROJECT DESIGN COORDINATOR:	DATE:
PROJECT ENGINEER:	DATE:
PROJECT MANAGER:	DATE:

DRESSER-RAND — TURBO PRODUCTS DIVISION, OLEAN NY 14760

PROJECT NO. 78181 — SCALE NONE

DRESSER-RAND HIGH PRESSURE CO2 TEST P & I DIAGRAM TEST LOOP EQUIPMENT

DRAWING NUMBER D-78180-09-004 — REV J



REV	DATE	DESCRIPTION	BY
J	6/2/10	HAZ OP REVIEW	MGR
H	3/2/10	REVIEW AND COMMENT	ATSI
G	1/4/10	ISSUED FOR REVIEW AND COMMENT	RJB
F	8/25/09	ISSUED FOR FEED PACKAGE	ATSI

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EQUIPMENT	①	②
PLANT ID.	CP-009	C-003
NAME	THRUST BALANCE LEAKAGE COMPRESSOR	RECYCLE LEAKAGE COOLER
P.O.NO.		-
DWG.NO.		-
MATERIAL		SHELL CS / TUBE SS
SIZE		18"O.D. - 20' LG
CAPACITY		7.56M BTU/HR
TEMP/PRESS		650'F / 550PSI
HP/V/RPM		-
WEIGHT		7,200 LBS
VENDOR		J.V. COUSINS

PROJECT	REV.	DESCRIPTION	MADE BY	APP BY	DATE
78180	J	HAZ OP REVIEW	MGR	.	6/2/10
78180	H	DWG No. WAS 004	ATSI	.	3/2/10
78180	G	ISSUED FOR REVIEW AND COMMENT	RJB	.	1/4/10
78180	F	ISSUED FOR FEED PACKAGE	EG	.	8/25/09

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DATE 12/7/09	DRAWN RJB	DESIGN M. SCHIFFHAUER
SCALE NONE	CHECKED	APPROVED
CAD FILE 7818009005	ACT. NO. 0325	CONTRACT NO. 78181
DRAWING NUMBER D-78180-09-005		REV. J

FILE NAME: 7818109005.DWG

	DATE
DRAWN BY: R. BURKE	12/28/09
DESIGN BY: M. SCHIFFHAUER	7/7/09
CHECKED BY:	
PROJECT DESIGN COORDINATOR:	
PROJECT ENGINEER:	
PROJECT MANAGER:	

"Commitment to Excellence"
DRESSER RAND
TURBO PRODUCTS DIVISION OLEAN,NY 14760

DRESSER-RAND
HIGH PRESSURE C02 TEST
P & I DIAGRAM
LEAKAGE COMPRESSORS

PROJECT NO. 78181 | SCALE NONE | DRAWING NUMBER D-78180-09-005 | REV J

6/2/2010 MGR

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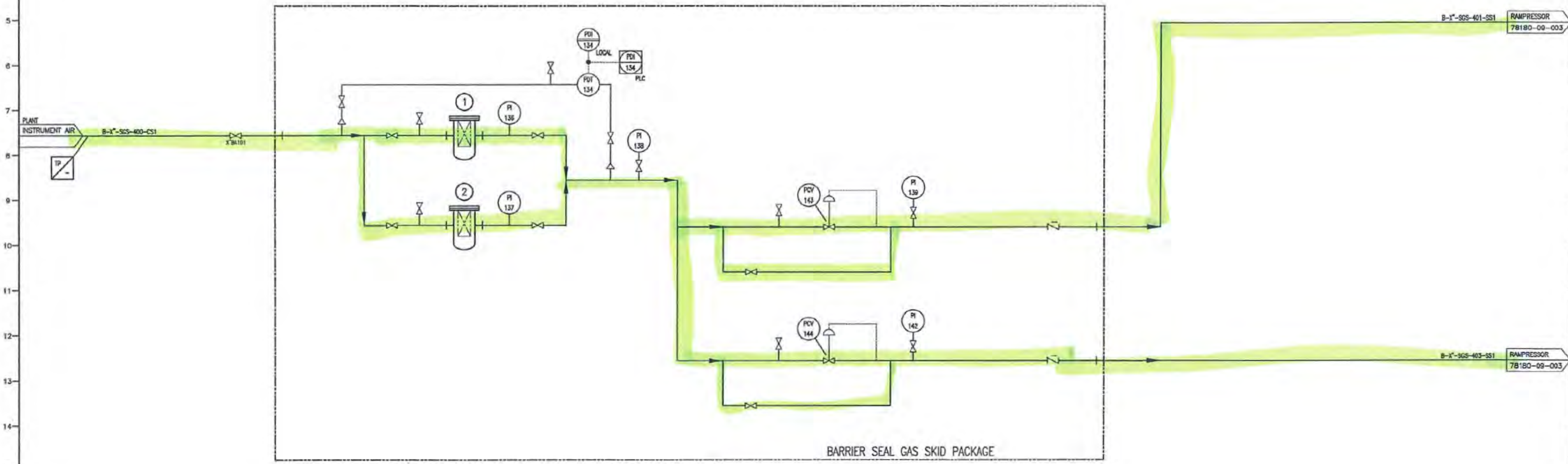

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G	1/4/10	ISSUED FOR REVIEW AND COMMENT	RJB
F	8/25/09	ISSUED FOR FEED PACKAGE	ATSI

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FOR CONSTRUCTION

EQUIPMENT	1	2	3	4	5	6	7	8	9	10	11	12
NAME	SEAL GAS INLET FILTER NO. 1	SEAL GAS INLET FILTER NO. 2										

PROJECT	REV.	DESCRIPTION	MADE BY	APP BY	DATE
78180	H	DWG No. WAS 005	-	.	
78180	G	ISSUED FOR REVIEW AND COMMENT	RJB	.	1/4/10
78180	F	ISSUED FOR FEED PACKAGE	EG	.	8/25/09

REVISIONS

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DATE 12/7/09 — DRAWN RJB — DESIGN M. SCHIFFHAUER — SCALE NONE — CAD FILE 7818009006 — ACT NO. 0325 — CONTRACT NO. 78181 — DRAWING NUMBER D-78180-09-006 — REV. H

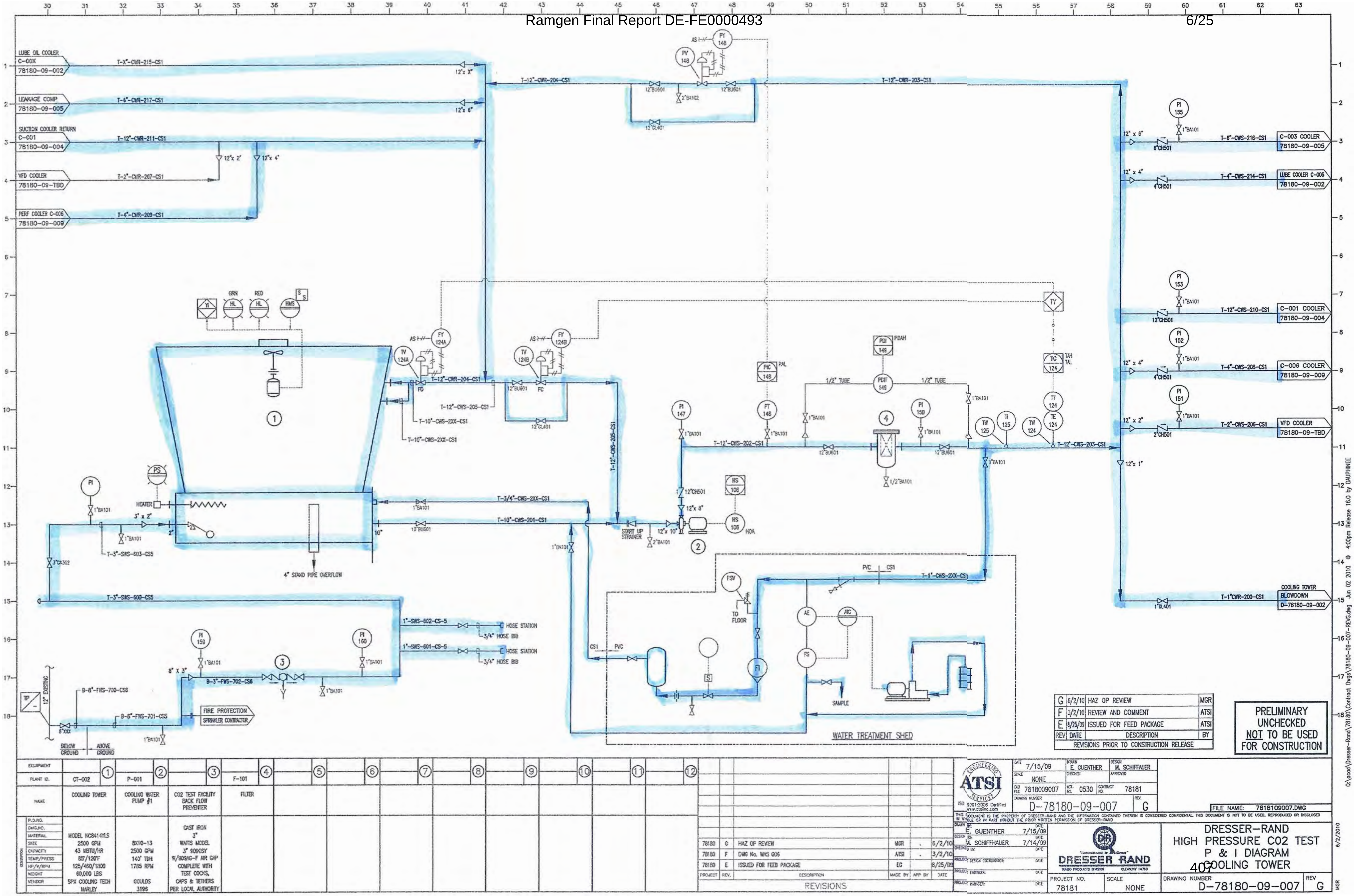
FILE NAME: 78181O9006.DWG

DRAWN BY: R. BURKE — DATE: 12/28/09
DESIGN BY: M. SCHIFFHAUER — DATE: 7/7/09

DRESSER RAND

DRESSER-RAND
HIGH PRESSURE C02 TEST
P & I DIAGRAM
BARRIER SEAL PURGE AIR

PROJECT NO. 78181 — SCALE NONE — DRAWING NUMBER D-78180-09-006 — REV H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LEAKAGE COMP
SUCTION COOLER RETURN
VFD COOLER
PERF COOLER C-006
C-003 COOLER
C-001 COOLER
C-006 COOLER
COOLING TOWER BLOWDOWN
4" STAND PIPE OVERFLOW
HOSE STATION
3/4" HOSE BIB
FIRE PROTECTION SPRINKLER CONTRACTOR
START UP STRAINER
SAMPLE
TO FLOOR
WATER TREATMENT SHED
BELOW GROUND
ABOVE GROUND
COOLING TOWER
COOLING WATER PUMP #1
CO2 TEST FACILITY BACK FLOW PREVENTER
FILTER
MODEL NC8414YLS
2500 GPM
43 MBTU/HR
85°/120°F
125/460/1800
60,000 LBS
SPX COOLING TECH MARLEY
8X10-13
2500 GPM
140' TDH
1785 RPM
GOULDS 3196
CAST IRON
3"
WATTS MODEL 3" 909OSY
W/909AG-F AIR GAP
COMPLETE WITH TEST COCKS, CAPS & TETHERS PER LOCAL AUTHORITY
G 6/2/10 HAZ OP REVIEW MGR
F 3/2/10 REVIEW AND COMMENT ATSI
E 8/25/09 ISSUED FOR FEED PACKAGE ATSI
REVISIONS PRIOR TO CONSTRUCTION RELEASE
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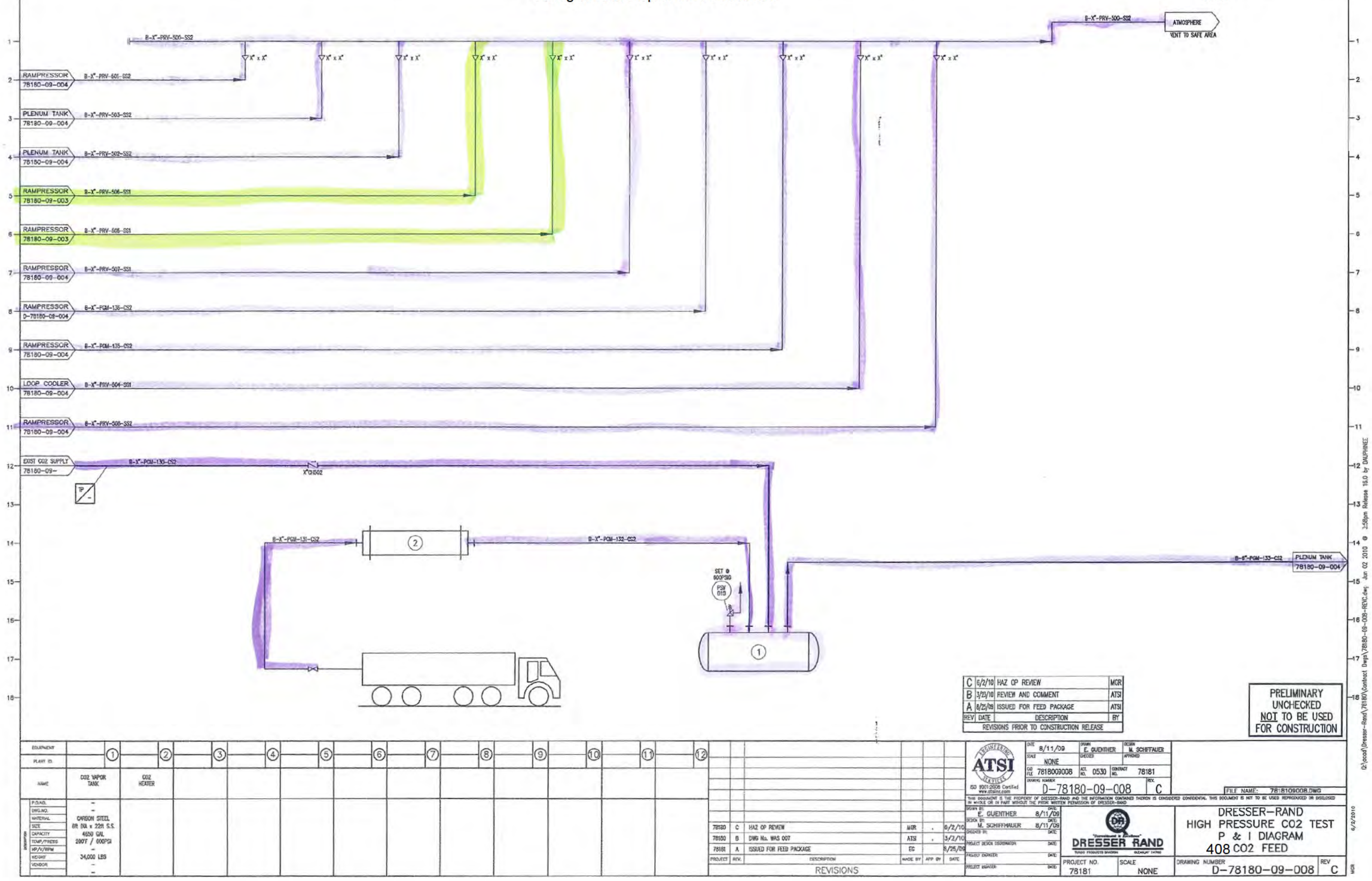
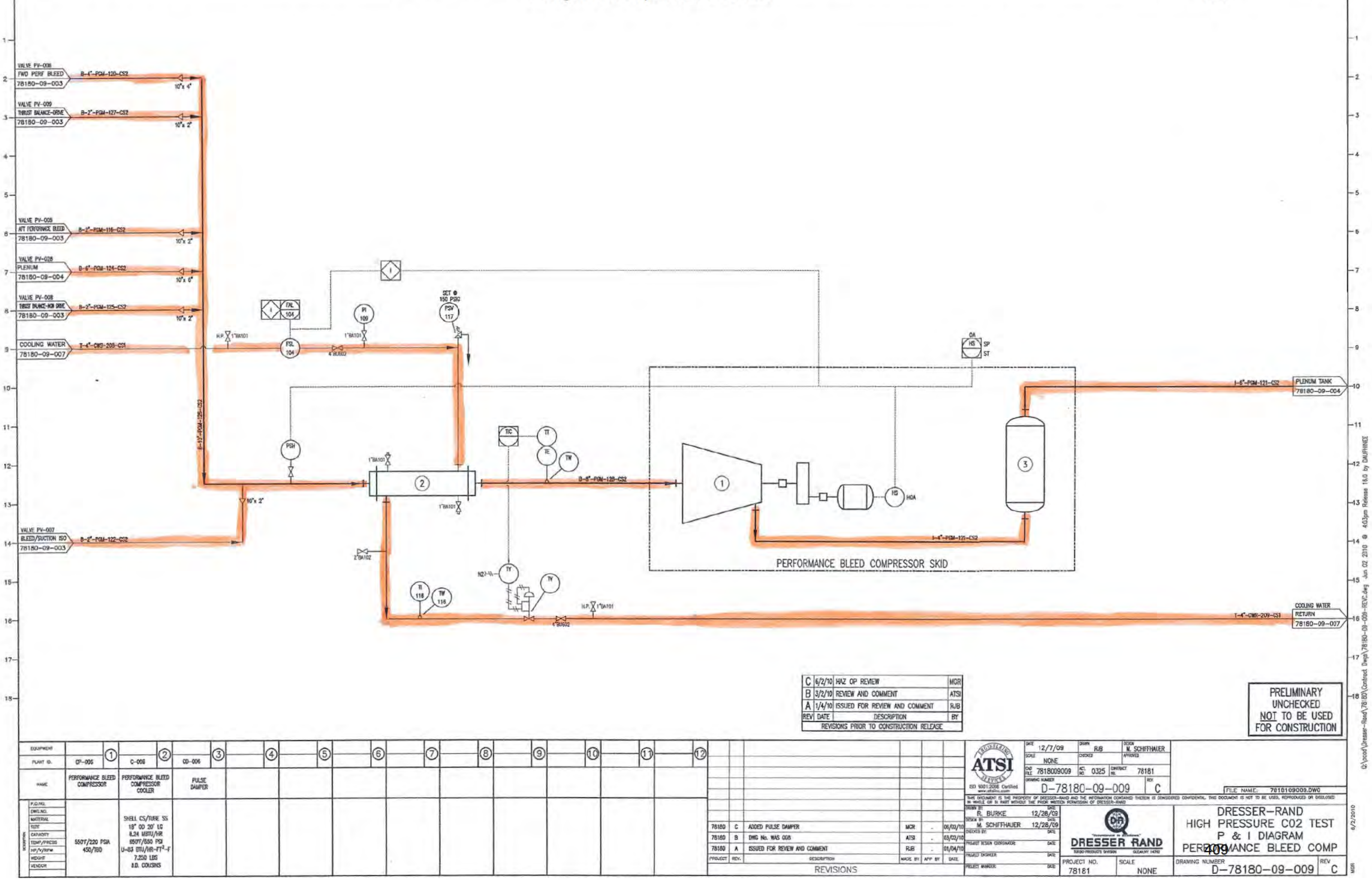
FOR CONSTRUCTION
REVISIONS
D-78180-09-007
FILE NAME: 7818109007.DWG
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
COOLING TOWER
PROJECT NO. 78181
SCALE NONE
DRAWING NUMBER D-78180-09-007
REV G


ATMOSPHERE
VENT TO SAFE AREA
RAMPRESSOR 78180-09-004
PLENUM TANK 78180-09-004
RAMPRESSOR 78180-09-003
LOOP COOLER 78180-09-004
EXIST CO2 SUPPLY 78180-09-
PLENUM TANK 78180-09-004
SET @ 600PSIG
PSV 010
C 6/2/10 HAZ OP REVIEW MGR
B 3/3/10 REVIEW AND COMMENT ATSI
A 8/25/09 ISSUED FOR FEED PACKAGE ATSI
REVISIONS PRIOR TO CONSTRUCTION RELEASE
PRELIMINARY UNCHECKED NOT TO BE USED FOR CONSTRUCTION
CO2 VAPOR TANK
CO2 HEATER
CARBON STEEL
4650 GAL
34,000 LBS
REVISIONS
ATSI
D-78180-09-008
FILE NAME: 7818109008.DWG
DRESSER-RAND
DRESSER-RAND HIGH PRESSURE CO2 TEST P & I DIAGRAM 408 CO2 FEED
PROJECT NO. 78181
SCALE NONE
DRAWING NUMBER D-78180-09-008
REV C


VALVE PV-006
FWD PERF BLEED
78180-09-003
VALVE PV-009
THRUST BALANCE-DRIVE
VALVE PV-005
AFT PERFORMANCE BLEED
VALVE PV-028
PLENUM
78180-09-004
VALVE PV-008
COOLING WATER
78180-09-007
VALVE PV-007
BLEED/SUCTION ISO
SET @ 150 PSIG
PERFORMANCE BLEED COMPRESSOR SKID
PLENUM TANK
COOLING WATER RETURN
C 6/2/10 HAZ OP REVIEW MGR
B 3/2/10 REVIEW AND COMMENT ATSI
A 1/4/10 ISSUED FOR REVIEW AND COMMENT RJB
REV DATE DESCRIPTION BY
REVISIONS PRIOR TO CONSTRUCTION RELEASE
PRELIMINARY
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FOR CONSTRUCTION
CP-005 PERFORMANCE BLEED COMPRESSOR
C-006 PERFORMANCE BLEED COMPRESSOR COOLER
CD-006 PULSE DAMPER
SHELL CS/TUBE SS
18" OD 20' LG
550°F/220 PSIA
450/180
7,250 LBS
78180 C ADDED PULSE DAMPER MGR 06/02/10
78180 B DWG No. WAS 008 ATSI 03/02/10
78180 A ISSUED FOR REVIEW AND COMMENT RJB 01/04/10
REVISIONS
ATSI
12/7/09
RJB
M. SCHIFFHAUER
NONE
D-78180-09-009
C
R. BURKE 12/28/09
M. SCHIFFHAUER 12/28/09
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
PERFORMANCE BLEED COMP
PROJECT NO. 78181
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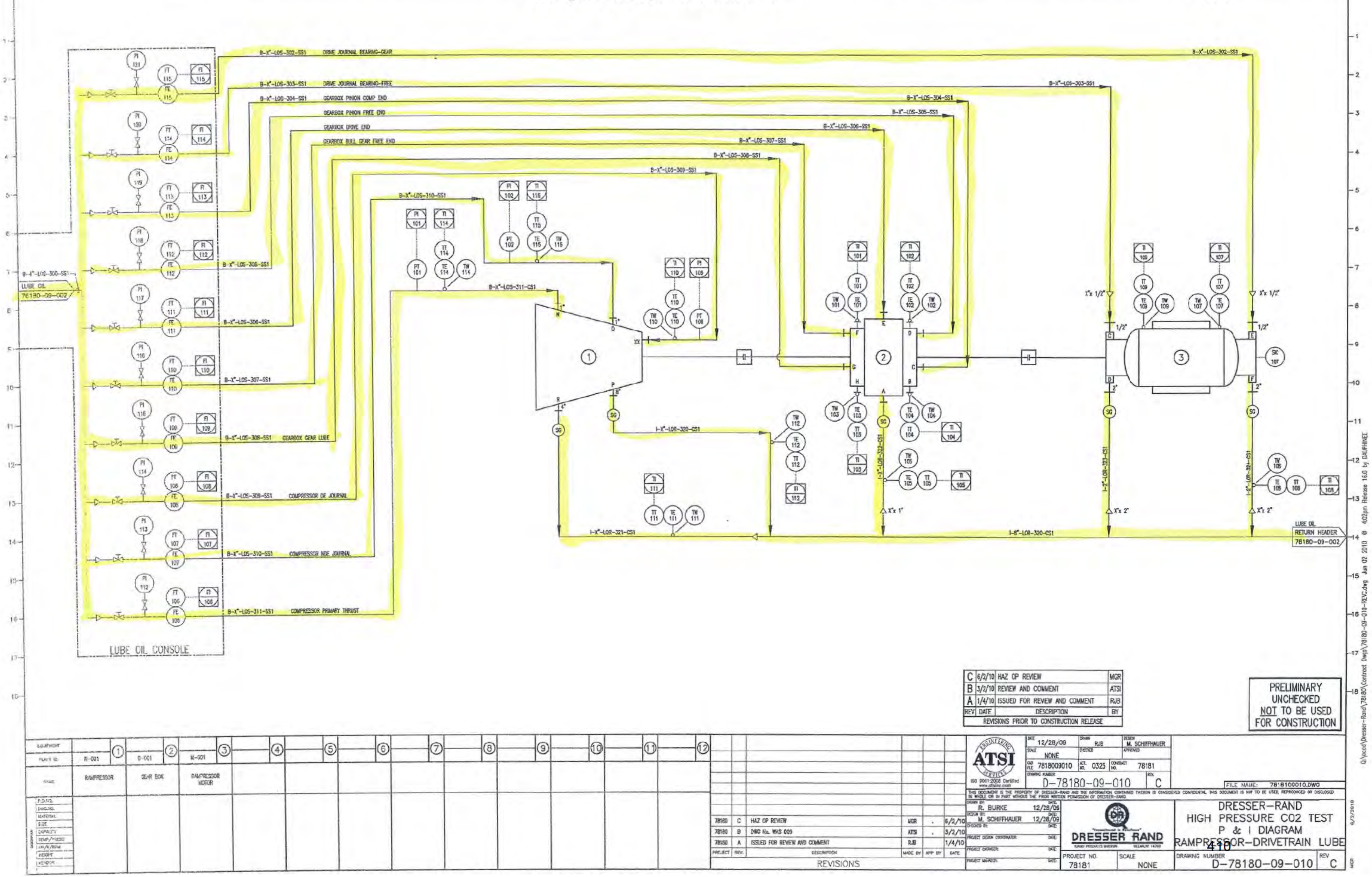

LUBE OIL CONSOLE
DRIVE JOURNAL BEARING-GEAR
DRIVE JOURNAL BEARING-FREE
GEARBOX PINION COMP END
GEARBOX PINION FREE END
GEARBOX DRIVE END
GEARBOX BULL GEAR FREE END
GEARBOX GEAR LUBE
COMPRESSOR DE JOURNAL
COMPRESSOR NDE JOURNAL
COMPRESSOR PRIMARY THRUST
LUBE OIL 7E180-09-002
LUBE OIL RETURN HEADER 78180-09-002
C 6/2/10 HAZ OP REVIEW MGR
B 3/2/10 REVIEW AND COMMENT ATSI
A 1/4/10 ISSUED FOR REVIEW AND COMMENT RJB
REVISIONS PRIOR TO CONSTRUCTION RELEASE
PRELIMINARY UNCHECKED NOT TO BE USED FOR CONSTRUCTION
R-001 RAMPRESSOR
G-001 GEAR BOX
M-001 RAMPRESSOR MOTOR
REVISIONS
ATSI
D-78180-09-010 C
FILE NAME: 78180090010.DWG
R. BURKE 12/28/09
M. SCHIFFHAUER 12/28/09
DRESSER-RAND
HIGH PRESSURE CO2 TEST
P & I DIAGRAM
RAMPRESSOR-DRIVETRAIN LUBE
PROJECT NO. 78181
SCALE NONE
DRAWING NUMBER D-78180-09-010
REV C

Appendix 9.2.1

Flow Path Design

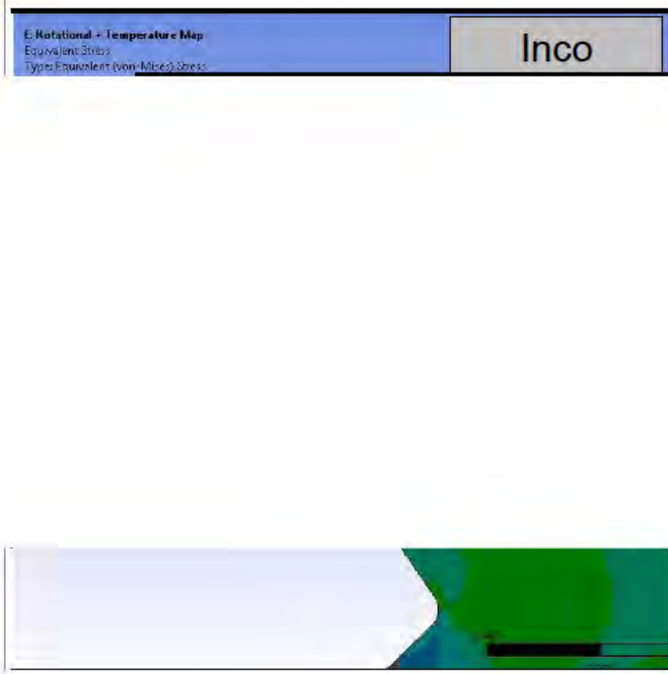
Supercompressor [redacted]
Structural Assessment

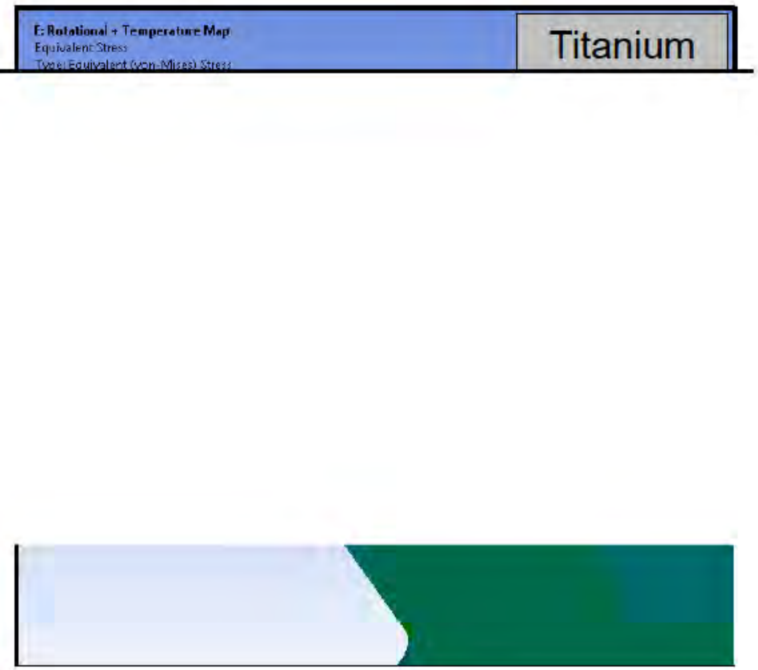
[redacted]

[redacted]

Stress in Blade-Hub Region

Stress > Yield **shown** in red





Appendix 9.2.2
Flow Path Final Design Review

Supercompressor Build-1 [redacted]

[redacted]

Final Design Review

[redacted]

DRESSER-RAND

Action Items from PDR 7/22/14

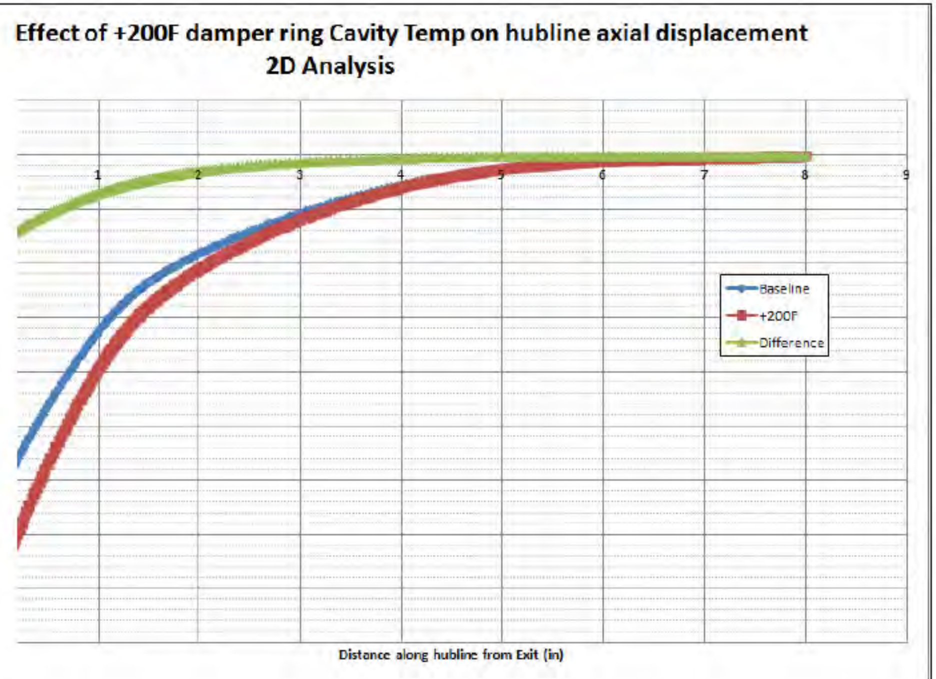
- *Look at potential for adding seal below damper ring* [redacted]
  - Incorporated
- *Check thread direction on tie-bolt and nut* [redacted] *Look into nut retention*
  - In discussion with hydraulic nut vendor.
- [redacted] ?
  - [redacted]
- *"No serrations in flowpath"*
  - Flowpath surface finish callout is 32 micro-inches. Note added to drawing regarding minimization of machining marks on flowpath

SuperCompressor – [redacted]

Layout

Thermal analysis – Axial displacements at DP

Potential impact of increased damper cavity temperatures



SuperCompressor – [redacted] Mechanical

Thrust Loads

3lbm/sec leakage flow

Axial Thrust Loads with 11.050" OD Damper ring

220 & 110 psi suction

Damper ring at OD = 11.05"

ANSYS + Hand Calc

	Location	Zone	Area (in^2)	220 psi suction		110 psi suction <1>	
				Pressure	Load lbf	Pressure	Load lbf
Aft Acting	Flowpath (P1)	P1	-	ANSYS	-95,276	ANSYS	-47,638
	Inlet Hub (P2)	P2	3.490	200	-698	100	-349
	Dry Gas Seal (P3)	P3 <2>	7.502	function	-1,497	function	-749
Fwd Acting	Back Face above Damper Ring (P4)	P4 <2>	62.469	function	78,936	function	39,468
	Backface between Damper ring -> Isolation seal (P5)	P5 <2>	62.767	function	13,140	function	6,570
	Backface between Isolation seal -> Shaft (P6)	P6 <2>	27.720	function	5,308	function	2,654
	Atmospheric loads on shaft (P7)	P7	12.914	14.7	190	14.7	190
	Net Thrust (+ve towards inlet)			-	**102**	-	**146**

<1> Scaled from 220 psi results on inlet total pressure
<2> From CFD analysis of back cavity. Input as function of radius

Net thrust loads reduced to close to zero with 11.050" OD Damper ring Analysis

3lbm/sec leakage flow

Effect of 10% change in P1 & P4 (with Damper OD at 11.050")

Damper ring at OD = 11.05"

ANSYS + Hand Calc

	Location	Zone	ID (in)	OD (in)	Area (in^2)	220 psi suction		110 psi suction <1>	
						Pressure	Load lbf	Pressure	Load lbf
Aft Acting	Flowpath (P1)	P1			-	ANSYS	-85,748	ANSYS	-42,874
	Inlet Hub (P2)	P2			3.490	200	-698	100	-349
	Dry Gas Seal (P3)	P3 <2>			7.502	function	-1,497	function	-749
Fwd Acting	Back Face above Damper Ring (P4)	P4 <2>			62.469	function	71,042	function	35,521
	Backface between Damper ring -> Isolation seal (P5)	P5 <2>			62.767	function	13,140	function	6,570
	Backface between Isolation seal -> Shaft (P6)	P6 <2>			27.720	function	5,308	function	2,654
	Atmospheric loads on shaft (P7)	P7			12.914	14.7	190	14.7	190
	Net Thrust (+ve towards inlet)					-	**1736**	-	**963**

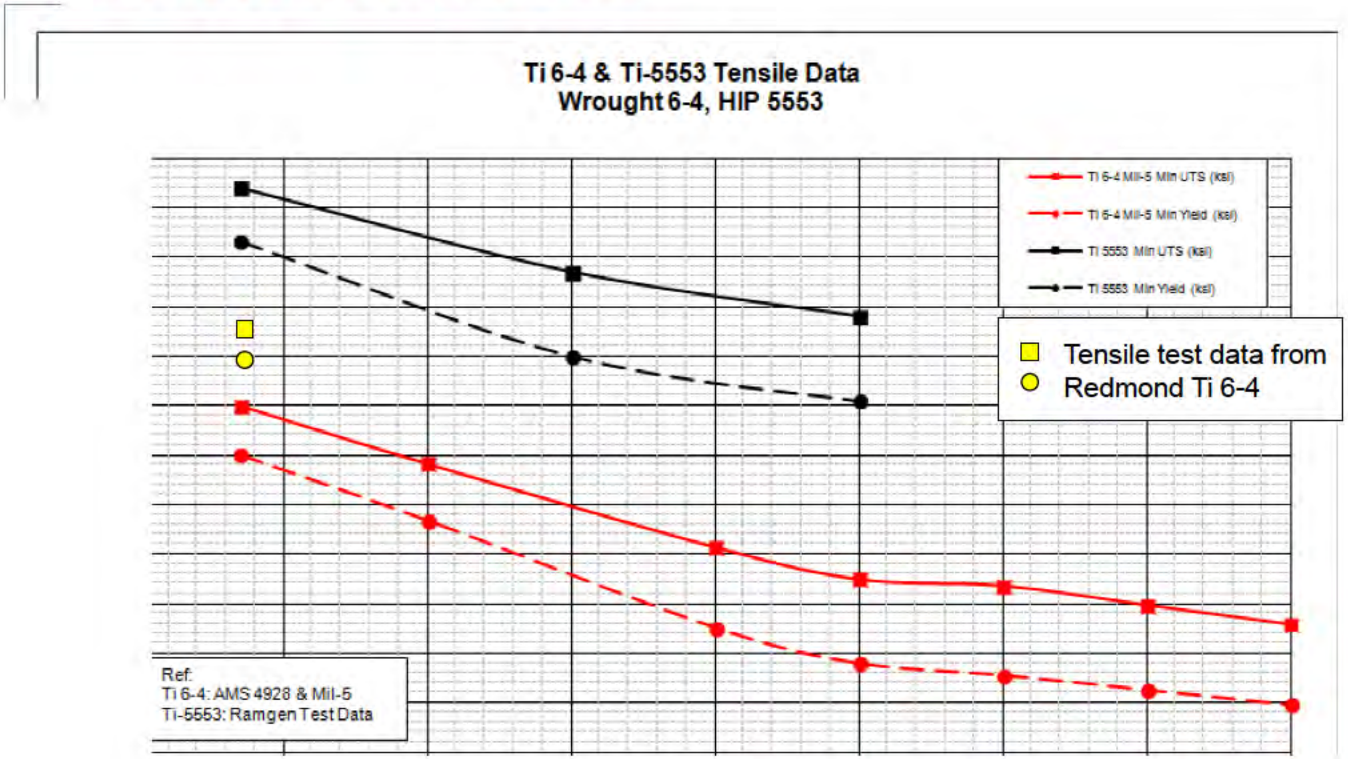
Net load:
@ 220 psi suction: increases from 100 lbf to 1700 lbf
@ 110 psi suction: increases from 150 lbf to 960 lbf

Change in P1 somewhat mitigated by commensurate change in P4

SuperCompressor

Stresses

Ti 6-4 & Ti 5553 Tensile Data



Redmond Ti 6-4 R/T strength ~ 15% above spec min
Ti-5553 has significantly more strength, particularly at elevated temp

3D Analysis at 28,980 rpm, Includes pressure loads & Thermal Loads

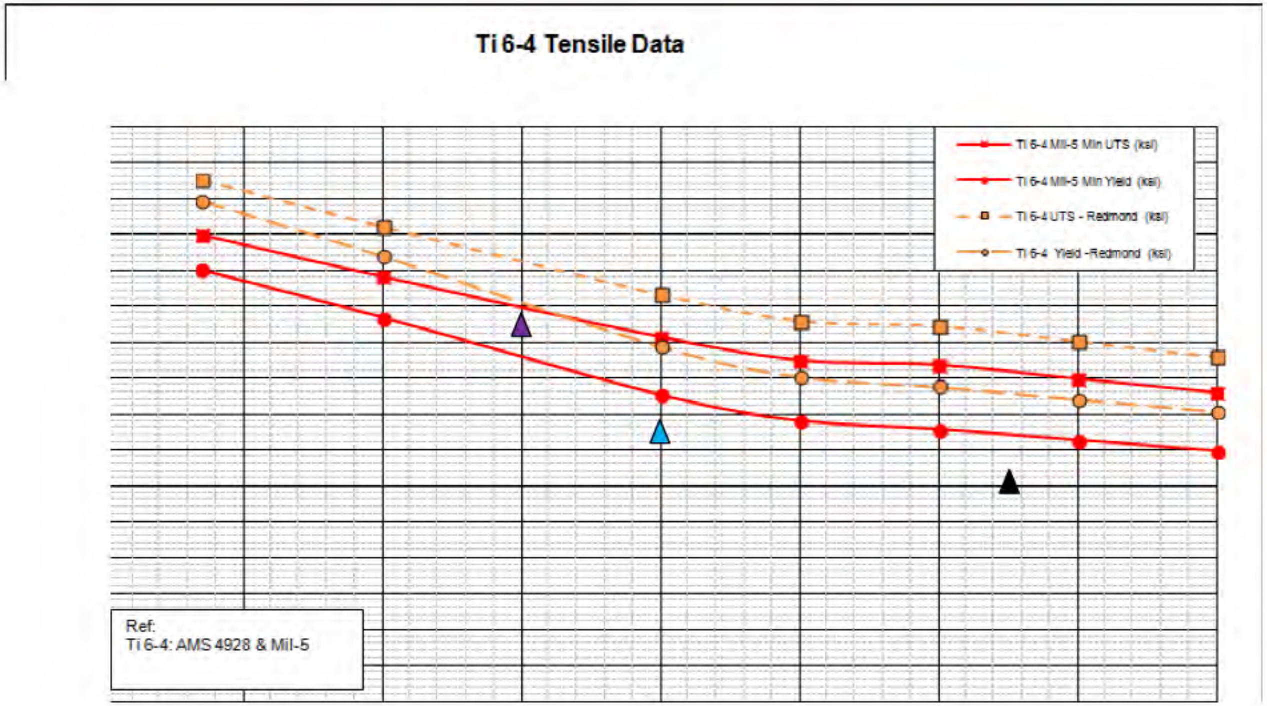
Boundary Conditions : 3D FEA

Temperatures
(Extrapolated from 2D of Closed [redacted]

Mechanical Loads
(Flowpath Loads modified from closed)

Update with revised damper ring cavity profile

Stress in selected regions compared to tensile data



All stresses < yield based on Redmond Ti 6-4 test data at R/T
Initiate material testing at elevated temp ?

3D Analysis at 28,980 rpm, Includes pressure loads & Thermals

Factor of Safety (Yield): Ti

3D FEA with rotational (MCOS), Pressure & Thermal loads

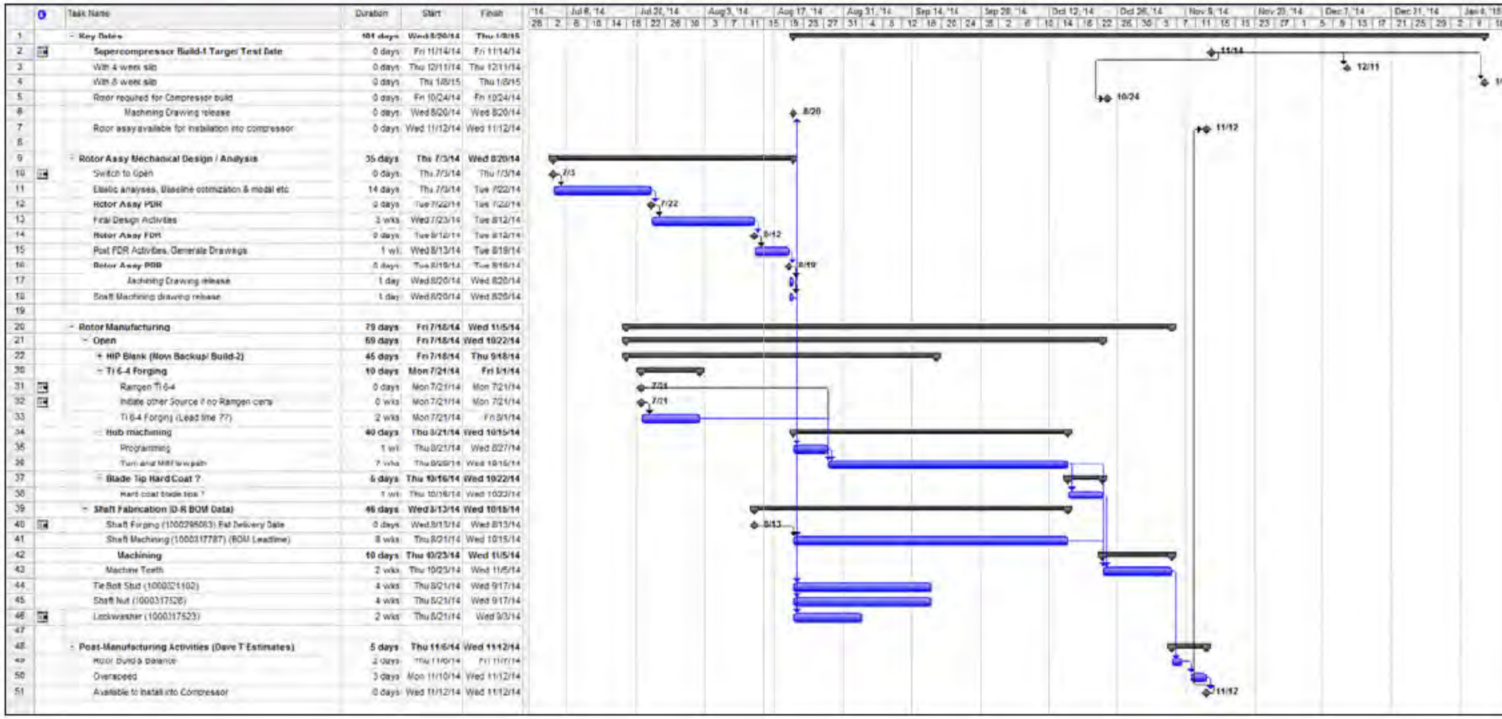
Using Von Mises Failure theory:
Ti 6-4 Min Safety Factor = 0.99
See plot on left

Note: This uses Material Min yield from AMS spec.
Test data for Redmond based raw material indicates yield at room temp may be 19 ksi above spec min.

Using test data then min safety factor > 1.0

For reference:
Ti-5553 Min Safety Factor ~ 1.3

Build-1 Rotor: Schedule



Ti 6-4 Rotor assembly available for installation Nov 12th

With 3 weeks for compressor build, does not meet Nov 14th test date. Meets 4 week slip date

Shaft Machining critical path

Build-1 Rotor: Cost Estimate

Supercompressor

Item	Part number	Source
Rotor Assembly (Sub-System 8)		
In-718 Hub and Cover Forgings	1000294957,8	On order. Includes tooling etc. No longer required
Ti 6-4 Forgings		Inventory from Ramgen
Ti-5553 Powder		on order
HIP Compact, Heat Treat		on order
lowpath Machining		quote 7/21/14 (Everything except Hirth)
Tooth machining		AMC quote
Shaft Forging	1000295063	quote (?)
Shaft Machining	1000317787	Estimate - out for quote 8/13
Tiebolt, nut, washer	1000321102 et a	Estimate. Assume IN718
Subtotal: Rotor Assembly (		
Spares		
Hub and Cover Forgings	1000294957,8	On order, no longer required
Near net shape HIP Compact		Included above
Hub Flowpath Machining		quote 7/21/14
Hirth Tooth machining		AMC quote
	Su	
Rotor Assembly Checkout		
Balancing	-	in-house
Spin Pit testing		in-house
Post Spin Inspection	-	in-house
Subtotal: Rotor Asser		
HIP Compact Material Validation		
Tensile Testing	-	Estimate based on prior work at Dickson, Mistras
Subtotal: HIP Compact Mate		
Build 1 total		

Tasks / Analyses to be performed

- Before Drawing Release
  - Update hot-to-cold with prelim temperatures from 3lbm/s cavity analysis (~ 800F damper surface)
  - Complete run-up check of cold geometry

- After Drawing Release
  - Roll in 'final' temps from H/T analysis. Check acceptability of increased temperature on damper ring
  - Document modal analysis
    - Dynamic Audit ?
  - Add teeth to 3D FEA and check stresses at steady state
    - Has been run as 'submodel' with part of disc and acceptable. Add to full
  - Update Structural analysis with:
    - Temperatures from [redacted] Heat Transfer analysis.
  - LCF Life assessment
    - [redacted] body & eeth following change to Ti 6-4
  - Assess rub loads from flowpath abradable
  - NLH structural analysis of [redacted] (After drawing rel ?)
  - Transient structural (start-stop cycle) with Speed, Pressure & Temperature

Appendix 9.2.3

Flow Path Production Readiness Review

[redacted]

DATUM-S [redacted] Production Readiness Review

[redacted]

Dave Taylor

[redacted]

DRESSER-RAND

SuperCompressor – Drawing

Drawing 1000339773

- Bellevue drawing and CAD model to be released to vendor
- Sensitivity level for Penetrant inspection to be confirmed.
- Aero acceptance on flowpath tolerance / surface finish required
- Copy included in the PRR folder for reference

Drawing almost ready for sign-off

SuperCompressor – [redacted] Mechanical

Hot-to-Cold Analysis

Hot-to-Cold : Runup of cold geom - Radial

Radial Displacements

Run-up of cold geom (v020): Displacements of selected points at design Point 220 psi suction pressure + temperatures)

[redacted]

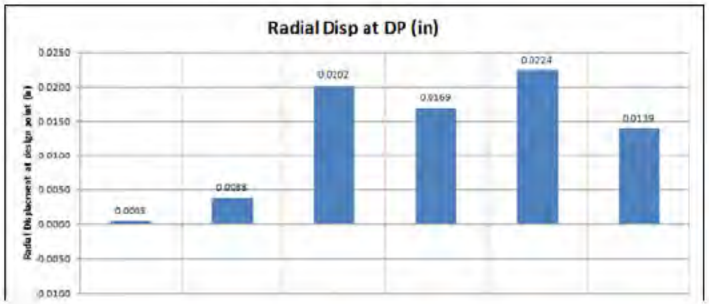
Radial Displacment

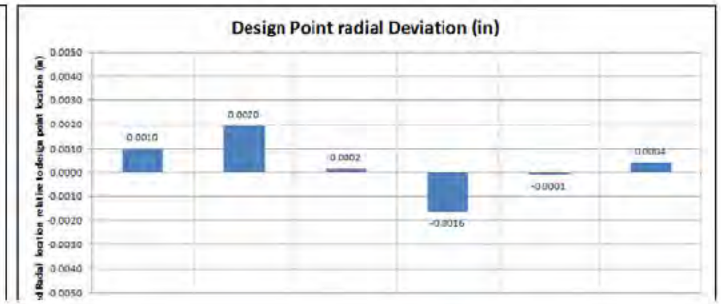
Location	Radial Disp at DP (in)	Cold Dia (in)	Cold radius (in)	Achieved Dia at DP (in)	Target Dia at DP (in) <1>	Design Point Diametral Deviation (in)	Design Point radial Deviation (in)	Radial Disp at MCOS (in)	Radial Displacement Delta Rel DP (in)
[redacted]nlet	0.0005					0.0020	0.0010	0.0005	0.0000
[redacted]Tip LE	0.0038					0.0039	0.0020	0.0041	0.0003
[redacted]Exit	0.0202					0.0003	0.0002	0.0210	0.0008
Damper Ring Fixed End OD	0.0169					-0.0032	-0.0016	0.0176	0.0007
Damper Ring Free End OD	0.0224					-0.0002	-0.0001	0.0234	0.0010
Isolation Ring ID	0.0139					0.0008	0.0004	0.0147	0.0008

<1> I.e the 'as-designed' or 'Hot' geom

Hot-to-Cold : Runup of cold geom. Radial Displacements

Radial Displacements





Fixed end of pocket damper ring grows 17 thou and free end grows 22 thou

With the exception of the fixed end of the pocket damper ring, all locations shown return to desired location within .002” or better.

Pocket Damper ring machined cylindrical, which gives 5 ½ thou of convergence in the pocket damper. Target was 4 thou.

Hot-to-Cold looks acceptable

Hot-to-Cold : Runup of cold geom - Axial

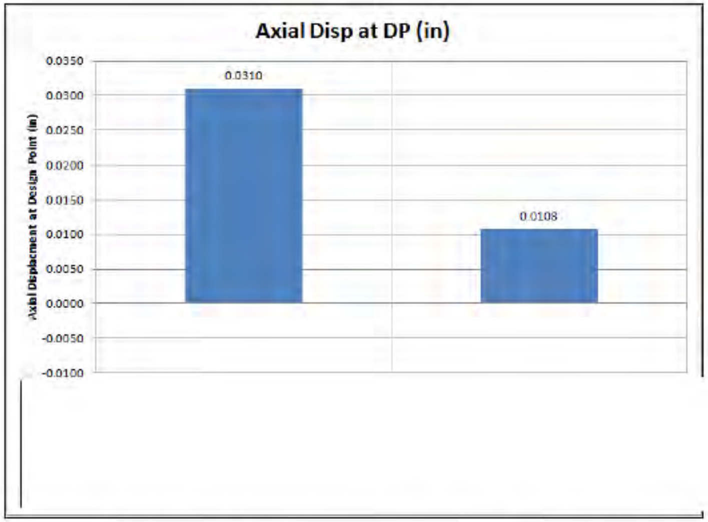
Axial Displacements

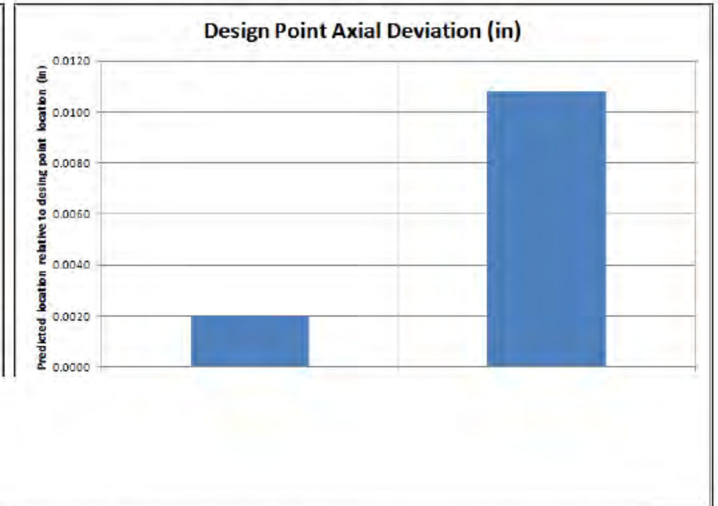
Axial Displacment							
Location	Axial Disp at DP (in)	Cold Axial Coord (in)	Achieved Axial Loc at DP (in)	Target Axial Loc at DP (in) <1>	Design Point Axial Deviation (in)	Axial Disp at MCOS (in)	Axial Displacement Delta Rel DP (in)
[redacted] Exit	0.0310	-0.029	0.002	0.000	0.0020	0.0313	0.0003
Damper Ring Fixed End	0.0108	-2.500	-2.489	-2.500	0.0108	0.0108	0.0001

See charts on next slide

Hot-to-Cold : Runup of cold geom - Axial Displacements

Axial Displacements





[redacted] exit grows 31 thou towards [redacted] inlet (fixed at hub aft side)

[redacted] Exit returns to desired location to within 2 thou. No correction applied to Damper Ring axial location. Need to ensure adequate clearance as it moves fwd.

Hot-to-Cold looks acceptable

Hot-to-Cold : Runup of cold geom.

Comparing runup geom with 'hot' (Design point) geom

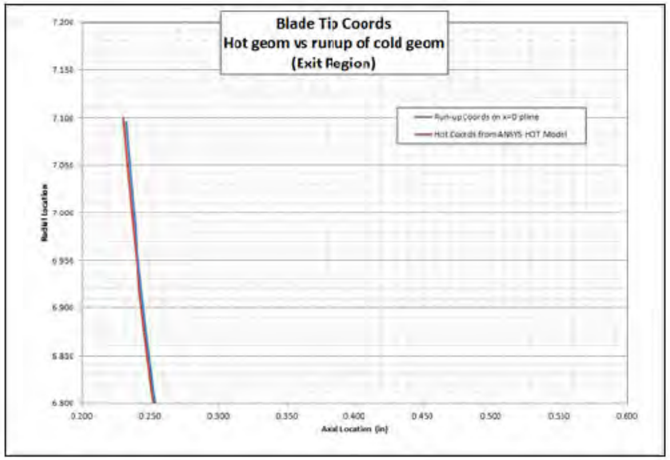
Largest axial discrepancy occurs in exit region but values are small as shown.

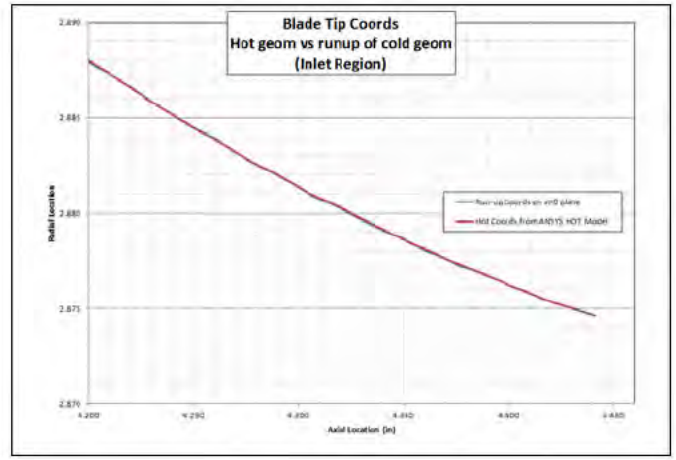
For example, 0.5" down from [redacted] exit, the blade tips runup to the desired location (i.e the hot, as designed location) to within .001"

Hot-to-Cold looks acceptable

Hot-to-Cold : Runup of cold geom.

Comparing runup geom with 'hot' (Design point) geom – [redacted] line





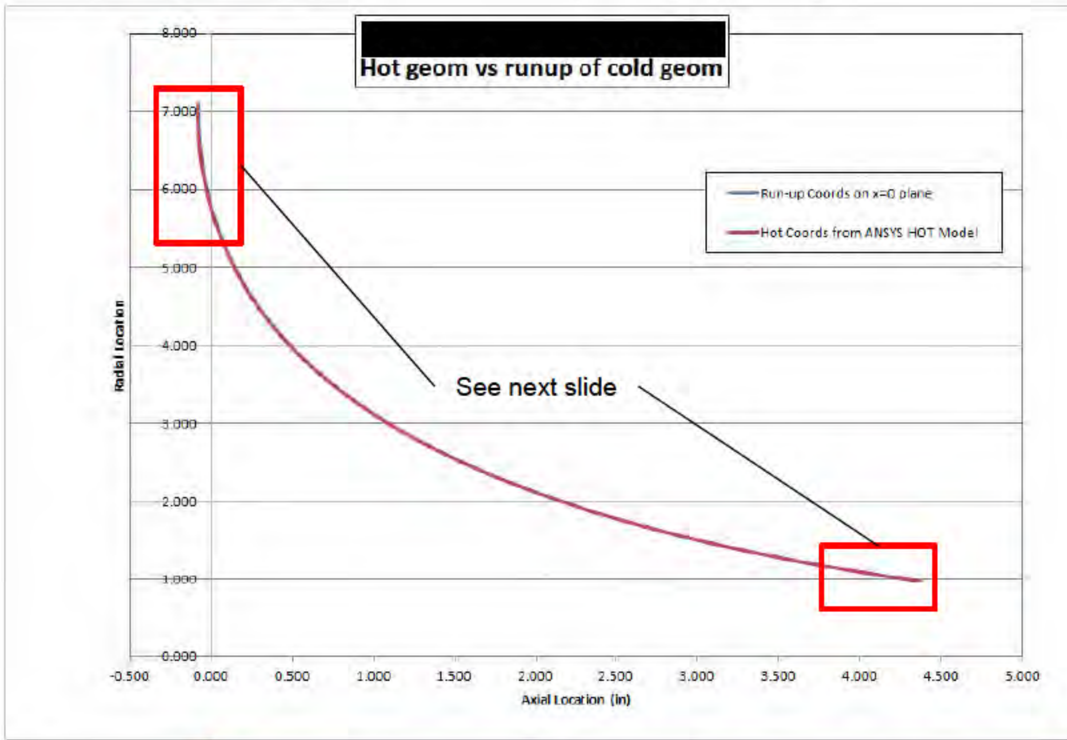
Largest deviation between hot geometry and run-up of cold geometry occurs in the exit region. However, max deviation is .002" as shown previously

Inlet region virtually no deviation

Hot-to-Cold acceptable

Hot-to-Cold : Runup of cold geom.

Comparing runup geom with 'hot' (Design point) geom – Hubline

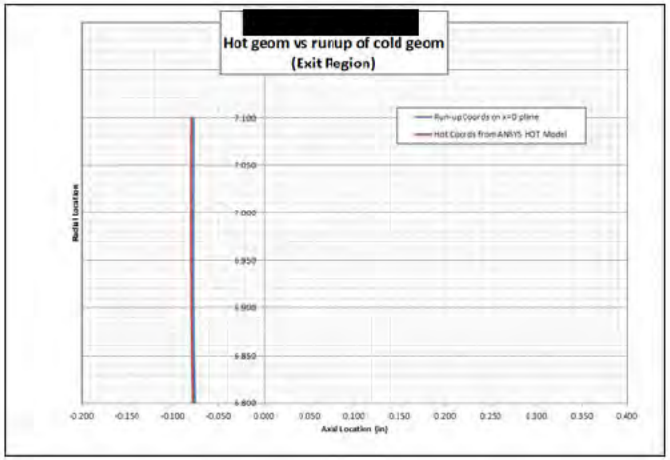


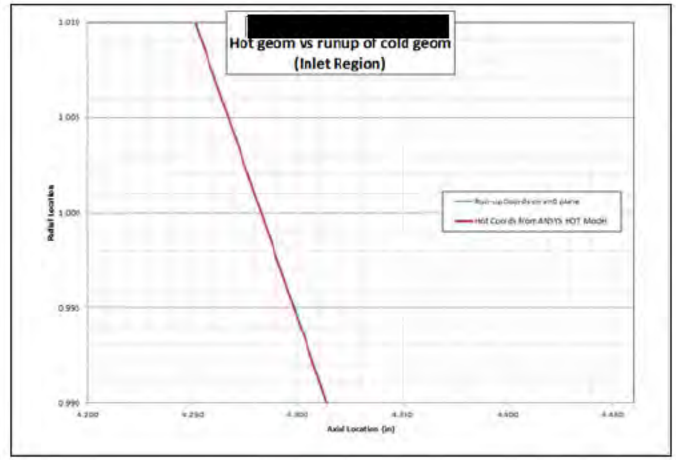
<1> profile assessed is .080" offset from hubline for ease of analysis in ANSYS

Hot-to-Cold looks acceptable

Hot-to-Cold : Runup of cold geom.

Comparing runup geom with 'hot' (Design point) geom – Hub line





Largest deviation between hot geometry and run-up of cold geometry occurs in the exit region.
Max deviation is .0026"

Inlet region virtually no deviation

Hot-to-Cold acceptable

Axial Thrust Loads with 11.125" OD Damper ring

220 & 110 psi suction

Damper ring at OD = 11.125" (Hot)

ANSYS + Hand Calc

	Location	Zone	ID (in)	OD (in)	Area (in^2)	220 psi suction		110 psi suction <1>	
						Pressure	Load lbf	Pressure	Load lbf
Aft Acting	Flowpath (P1)	P1			-	ANSYS	-95,276	ANSYS	-47,638
Aft Acting	Inlet Hub (P2)	P2			3.490	200	-698	100	-349
Aft Acting	Dry Gas Seal (P3)	P3 <2>			7.502	function	-1,497	function	-749
Fwd Acting	Back Face above Damper Ring (P4)	P4 <2>			61.162	function	77,384	function	38,692
Fwd Acting	Backface between Damper ring -> Isolation seal (P5)	P5 <2>			64.073	function	13,435	function	6,718
Fwd Acting	Backface between Isolation seal -> Shaft (P6)	P6 <2>			27.720	function	5,308	function	2,654
Fwd Acting	Atmospheric loads on shaft (P7)	P7			12.914	14.7	190	14.7	190
	Net Thrust (+ve towards inlet)					-	**-1155**	-	**-483**

<1> Scaled from 220 psi results on inlet total pressure
<2> From CFD analysis of back cavity. Input as function of radius

Net thrust load just over 1000lbf towards gearbox with revised damper ring OD

SuperCompressor : Tie-bolt clearance (Revised)

NDE (A->B)				
	+	-	MMC	LMC
	0.0010	0.0000	1.0100	1.0110
Tie Bolt	0.0000	0.0005	1.0090	1.0085
		Clearance (dia)	0.0010	0.0025
		Clearance (Radial)	0.0005	0.0012

NDE (B-> C)					
	Dia	+	-	MMC	LMC
		0.0050	0.0000	1.0100	1.0150
Tie Bolt		0.0000	0.0005	1.0090	1.0085
			Clearance (dia)	0.0010	0.0065
			Clearance (Radial)	0.0005	0.0032

DE				
	+	-	MMC	LMC
	0.0010	0.0000	1.0630	1.0640
Tie Bolt	0.0000	0.0005	1.0620	1.0615
		Clearance (dia)	0.0010	0.0025
		Clearance (Radial)	0.0005	0.0012

- Tolerance & runout applied over specific zones

Appendix 9.3.1

Static Hardware Design Reviews

Super Compressor [redacted]
Abradable [redacted] FDR

Kyle Badeau

[redacted]

DRESSER-RAND

Agenda

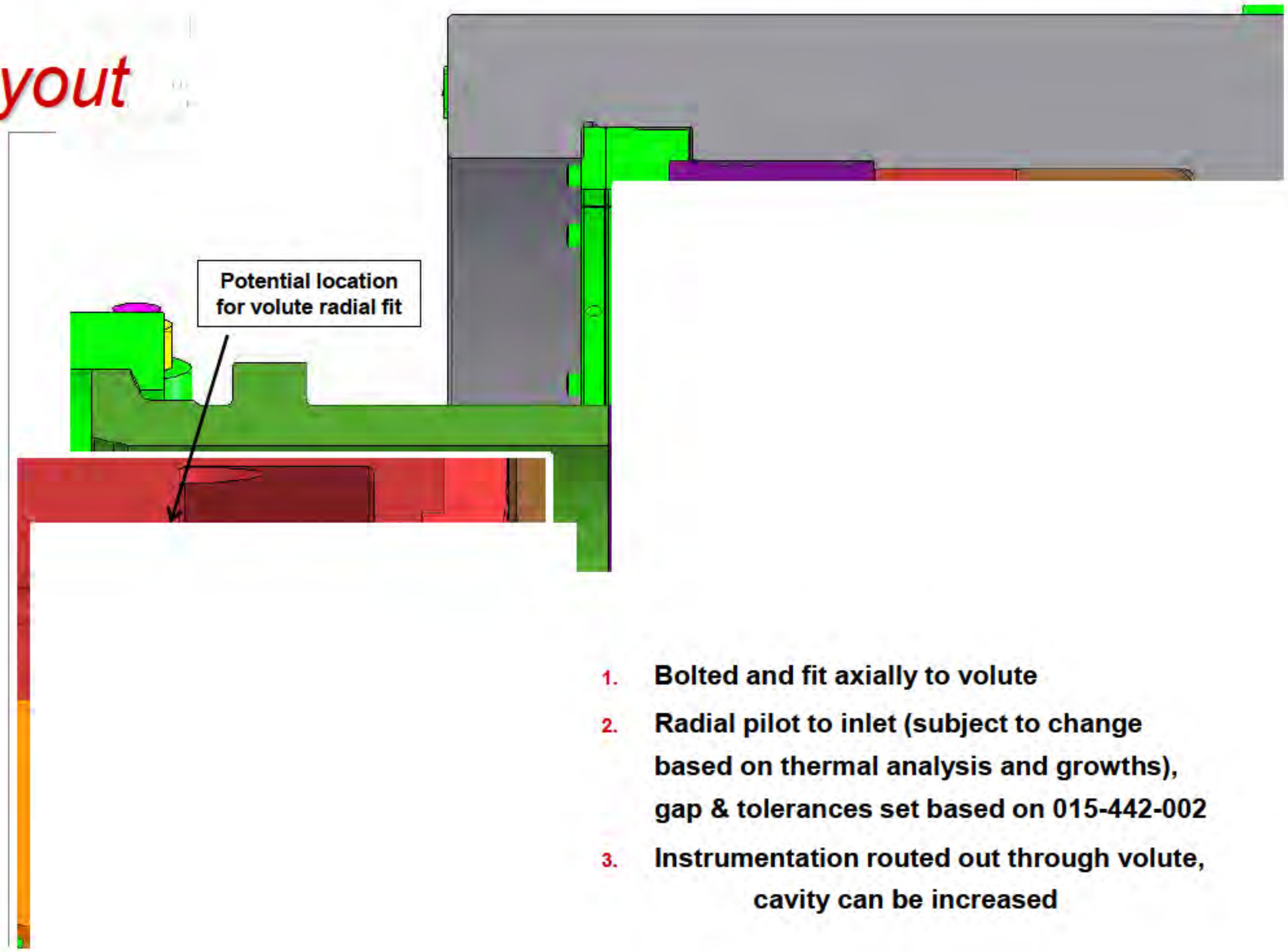
1. Component requirements
2. Layout, assembly, & geometry definition
3. Materials, processing, cost & cycle
4. Production order of operations
5. Structural analysis
   - Temperature & pressure profiles
   - Boundary conditions
   - Stress results
   - Growths
6. Instrumentation definition & routing
7. Remaining work

Abradable [redacted] Design Requirements

Key Objectives/Requirements:

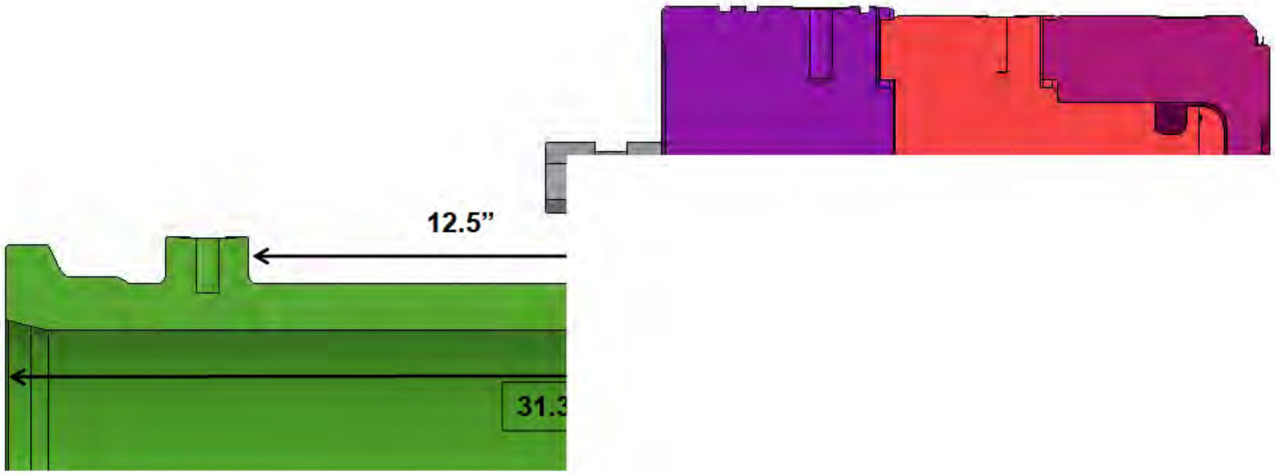
1. **Features for axial, radial, and circumferential retention**
2. **Continuous [redacted] flowpath profile from inlet to diffuser pinch transition (i.e. no fastening features exposed to flowpath)**
3. **Provide instrumentation routing**
4. **Ability to remove [redacted] for repair or replacement of abradable without breaking bundle**
   - More critical for test rig due to risk of instrumentation damage
   - Less critical for product, depending on customer disassemble/reassemble time requirements or expected abradable replacement interval, without instrumentation to get in the way
5. **Meet structural, fatigue, and frequency margin criteria**
6. **Minimize [redacted] to [redacted] gap at operating conditions**

Layout



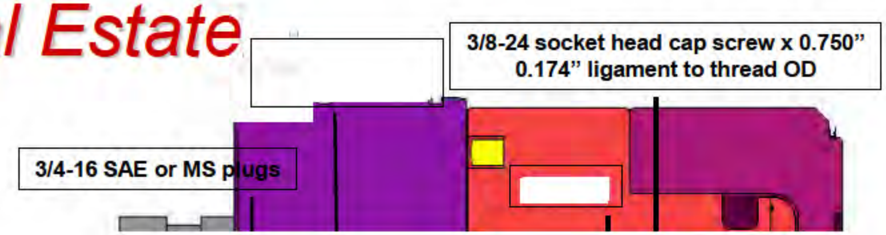
1. Bolted and fit axially to volute
2. Radial pilot to inlet (subject to change based on thermal analysis and growths), gap & tolerances set based on 015-442-002
3. Instrumentation routed out through volute, cavity can be increased

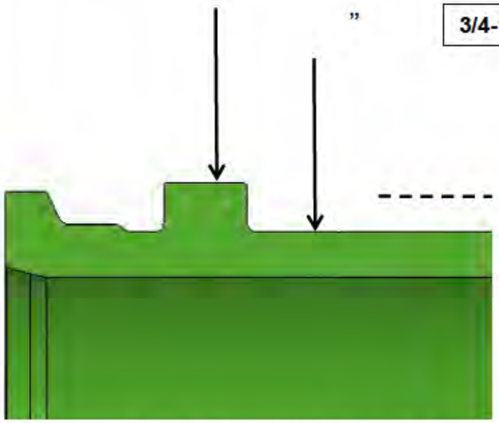
Assembly Approach: Outer Cavity Bolting



- [redacted] assembly tooling needs to reach ~10”
- Bolt retention w/ snap rings to keep bolts inside volute once bundle is assembled, preventing them from falling into cavity
- Hole through pressure head machined, threaded, and plugged
- Crowded cavity space linkages, bolting, instrumentation)
- Bolts do clear variable linkages, need to be positioned at ~0° for assembly clearance

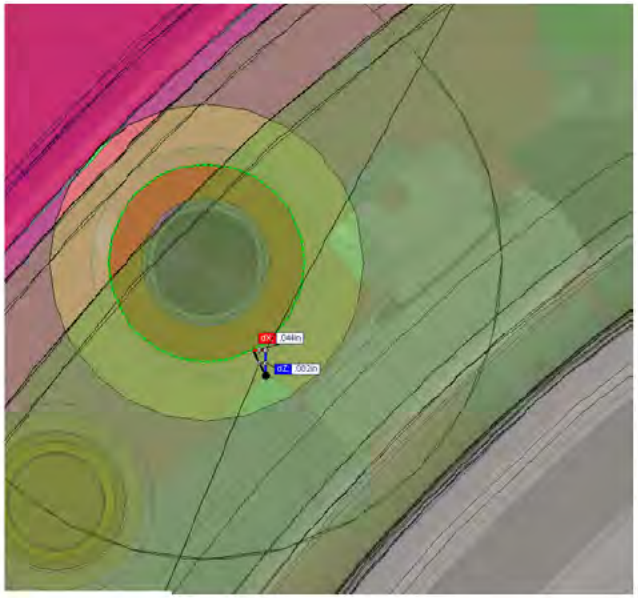
Cavity Real Estate



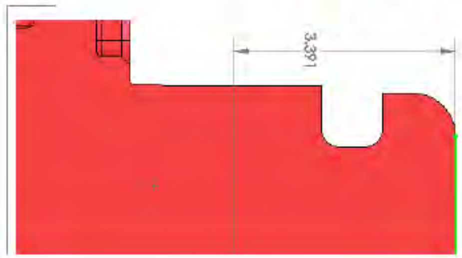


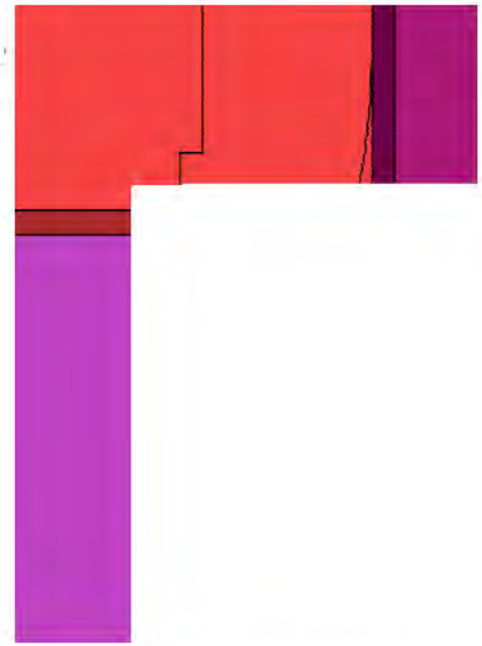
- bolting diameter, clocked 13° from TDC
- 32.7° bolt spacing, 10 bolts, 1 blank due to actuator arm
- Potential to make 2 screws jacking screws
- Washers under screw heads?
- Pins needed for clocking or assembly guides?

Assembly:



Hot Aero Profile





No change since 8/17
Reflected in UG layout

Material & Processing

Base Metal:

1. **Cast iron (potentially Ni-Resist)**
   - **Used in [redacted] for its higher internal damping characteristics for reduced modal deflection**
   - **Casting envelope will probably need to be reduced for reduced risk of porosity**
   - **Ni-Resist has improved corrosion resistance, but contains graphite, which may be difficult to bond abradable to (need to confirm)**

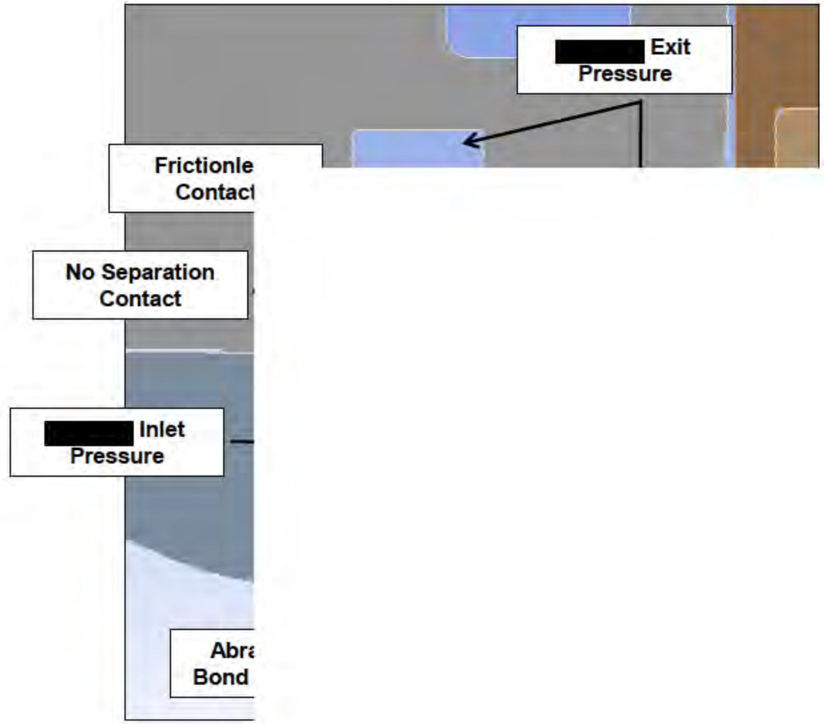
Envelope sent to SCM for vendor discussions and ROM quotes

Production Sequence

1. Raw material (Vendor 1 TBD)
2. Machine (Vendor 2 TBD)
3. Machine TC ports
4. Machine static pressure ports, backside only
5. Weld TCs
6. Abradable coat
7. Machine (EDM?) 0.020" diameter static pressure taps abradable side
8. Abradable coating heat treat to 1050°F, carbon precip risk?
9. Install pressure tubes, can this be done before heat treat?
10. Machine radial pilot surface (???)

Need to examine bundling of steps to reduce overall cycle

BCs



"Structural steel" used for both abradable and base metal in thermal and structural analyses

Stress Results

- **Pressure loads constitute <5ksi of total stress**
- **[redacted] stress primarily driven by thermal loading**
- **Thermal stress primarily oriented in the hoop direction**
  - **Compressive in the hot, exit region**
  - **Tensile in the cool, inlet region**
  - **T_{bulk} of 312°F**
- **Carbon steel RT YS ~36 ksi, higher strength material selection**

Potential to reduce thermal stress with thermal mass reduction (carving out outer interface with volute)

Material Candidate Strength Assessment

- Carbon steel lacks strength
- Low-alloy to stainless steel
  1. 304 SS lacks strength (30 ksi @ RT)
  2. 410 SS 60 ksi @ RT
  3. AISI 4130 plate, bar, or forging (normalized, tempered, stress relieved)
     - 70 ksi YS @ RT
     - 56 ksi YS @ 600°F
  4. AISI 4340 100 ksi @ RT
  5. 17-4 PH SS (plate or forging)
     - 105 – 170 ksi YS @ RT
     - 84 – 136 ksi YS @ 600°F
  6. 4330V bar or forging (quench & temper)
     - 185 ksi YS @ RT
     - 148 ksi YS @ 600°F
- Cast Iron
  1. ASTM A48 grey cast iron up to 60 ksi @ RT
  2. Ni Resist strengths up < 40 ksi @ RT

Growth

- **Radial Growth**
  1. **2 mils outward**
  2. **16 mils outward**
- **Axial Growth**
  1. **36 mils to the left**
  2. **33 mils to the left**

- **Structural steel CTE 6.67E-6 °F^{-1} (constant)**
  - **410SS: 6.11E-6 °F^{-1} @ 600°F**
  - **17-4 PH: 6.4E-6 °F^{-1} @ 600°F**
  - **4130: 7.33E-6 °F^{-1} @ 600°F**
  - **4340: 7.58E-6 °F^{-1} @ 600°F**
  - **Ni Resist: 5-9E-6 °F^{-1} @ 600°F (could be selected for CTE if stress can be reduced)**

Approach: Higher CTE is Better

Remaining Work

- Select material
- Add remaining assembly features & components
- Finalize 2D profile
- Finalize instrumentation placement, prox probe?
- Clearance strategy
- Hot to cold conversion
- Modal analysis
- Fatigue assessment
- Drawings & BOM

Super Compressor [redacted] FDR II

Kyle Badeau

[redacted]

DRESSER-RAND

Agenda

1. Component requirements
2. Layout, assembly, & geometry definition
3. Materials, processing, cost & cycle
4. Production order of operations
5. Structural analysis
    - Temperature & pressure profiles
    - Boundary conditions
    - Stress results
    - Growths
    - Modal analysis
6. Instrumentation definition & routing
7. Remaining work

Abradable [redacted] Design Requirements

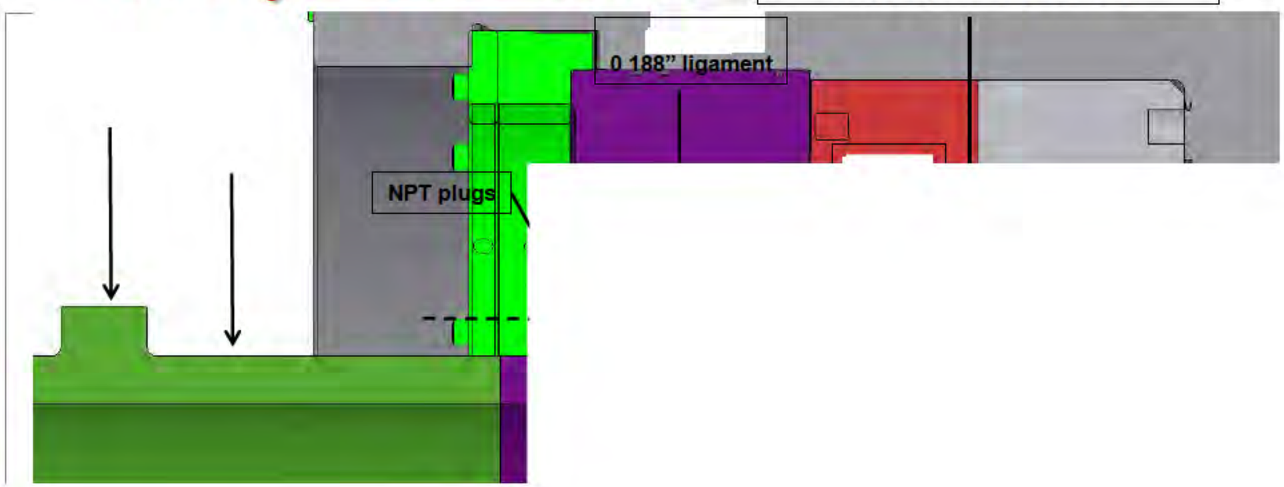
Key Objectives/Requirements:

1. **Features for axial, radial, and circumferential retention**
2. **Continuous [redacted] flowpath profile from inlet to diffuser pinch transition (i.e. no fastening features exposed to flowpath)**
3. **Provide instrumentation routing (thermocouples, static pressure taps, prox probes)**
4. **Ability to remove [redacted] for repair or replacement of abradable without breaking bundle**
   - **More critical for test rig due to risk of instrumentation damage**
   - **Less critical for product, depending on customer disassemble/reassemble time requirements or expected abradable replacement interval, without instrumentation to get in the way**
5. **Meet structural, fatigue, and frequency margin criteria**
6. **Minimize [redacted] to [redacted] gap at operating conditions**

Assembly Approach: Outer Cavity Bolting

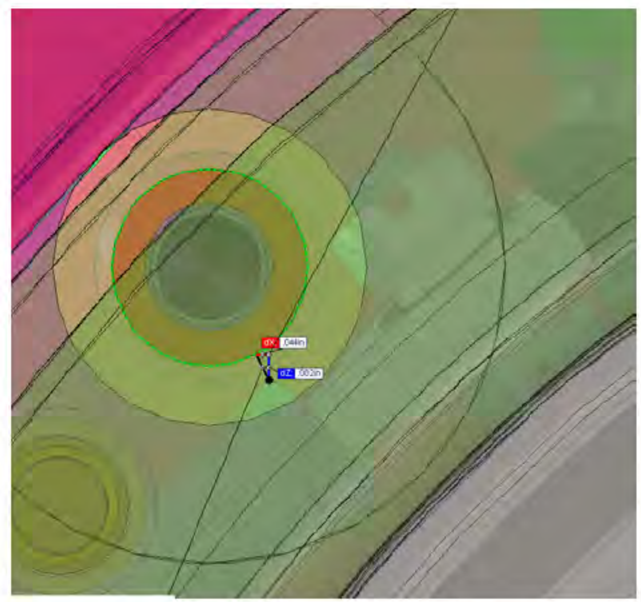
- [redacted] assembly tooling needs to reach
- Hole through pressure head machined, threaded, and plugged with NPT (NPT not shown here)
- Bolts do clear variable linkages, need to be positioned at ~0° for assembly clearance
- Bolt retention w/ snap rings to keep bolts inside volute once bundle is assembled, preventing them from falling into cavity

Cavity Real Estate

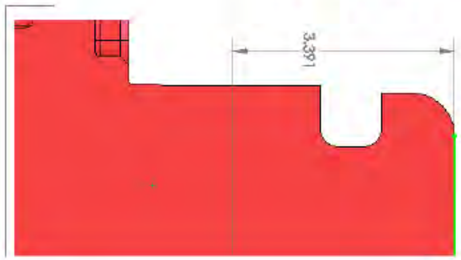


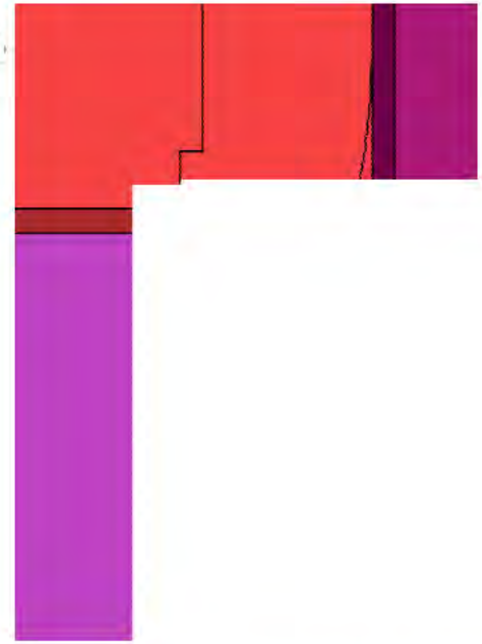
- bolting diameter, clocked 13° from TDC, 32.7° bolt spacing, 8 bolts, 2 jacking screws, 1 blank due to actuator arm blockage
- A286 used as a belt and suspenders approach to reduce risk of galling (if bolts can’t be removed, bundle disassembly would be required)
- Nord-Lock to prevent bolts from backing out
- Anti-sieze
- Pin needed for clocking

Assembly: VIGV 0°



Hot Aero Profile





No change since 8/17
Reflected in UG layout

Production Sequence

Step	Description
1	4330 Forging, ship
2	Rough Machine
3	Oil Quench & Temper
4	Finish Machine
5	Machine & weld TC's, secure leads
6	Machine blind static pressure ports, backside
7	Abradable coat, finish machine, bake
8	Machine static pressure taps abradable side
9	Install pressure tubes

Volute/Inlet Temperature Profile

- Steady state thermal analysis
- Volute hotter than [redacted] ([redacted] will not grow into volute)
- Inlet and [redacted] comparable temperatures at location of potential fit
- [redacted] exit region of hollow [redacted] has better thermal soak than solid [redacted] design and shallower thermal gradient
- [redacted] hotter in contact region with volute for side cut out profile

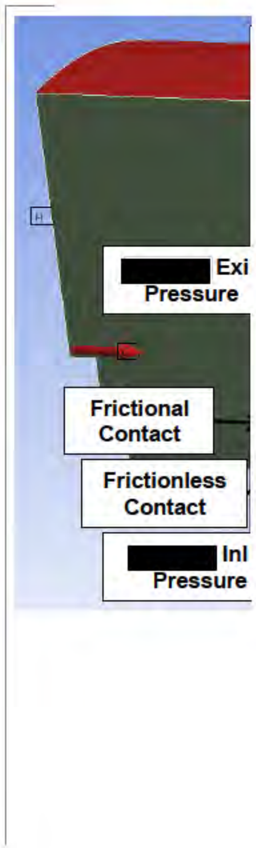
[redacted] $T_{bulk} = 345°F$

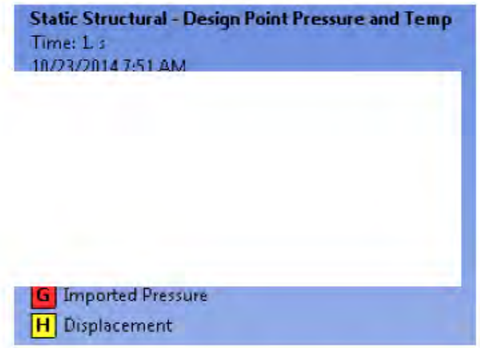
Pressure Profile

- Pressure map input from aero
- ~200 psi [redacted] inlet pressure
- ~1500 psi [redacted] exit pressure
- 3D pressure map used in 3D analysis

Axial Pressure Reaction Force = 112,000 lbf

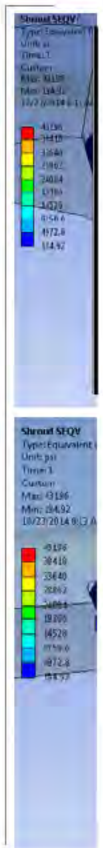
BCs

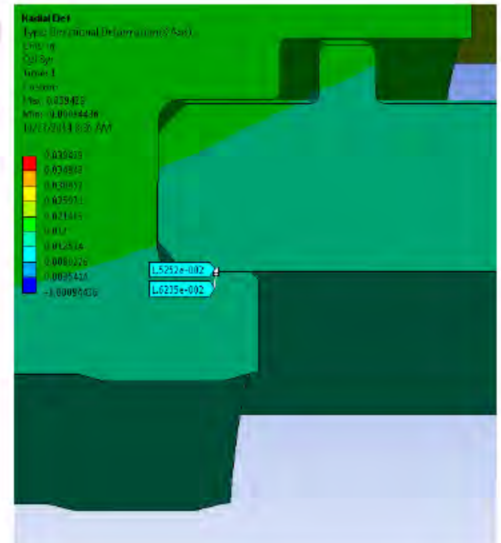




- 3D analysis used to capture non axisymetric [redacted] bolt hole effects
- 1/11th cyclic sector (11 bolt holes)
- Simplified, revolved volute geometry
- [redacted] and volute temperatures mapped
- Gaspath pressures mapped
- Frictional contact at bolted interface needed for convergence
- Volute A-36 material
- [redacted] 4340 low alloy steel (assumed similar to 4330)
- Volute front face constrained in axial and tangential directions (typical of cyclic sector analysis)

Structural Analysis

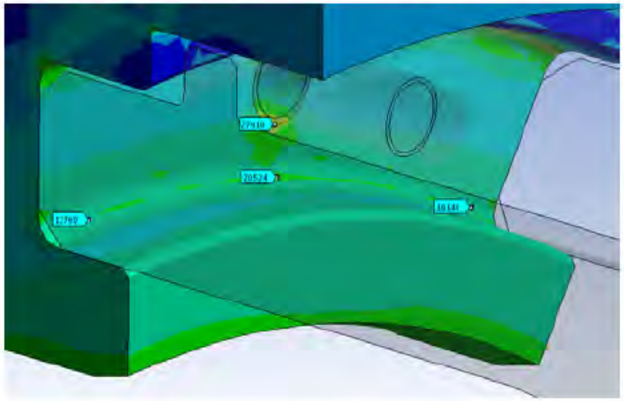




- [redacted] stress well below yield strength (~100 ksi) for steady state thermal loading
- Nominal/max pilot gap of 1 mil (shown)
- Volute radial growth 1 mil more than [redacted] (gap closes for contact at S.S. conditions)
- Peak [redacted] stress unaffected for 0.5 mil pilot max interference

[redacted] Stress Below Yield Strength

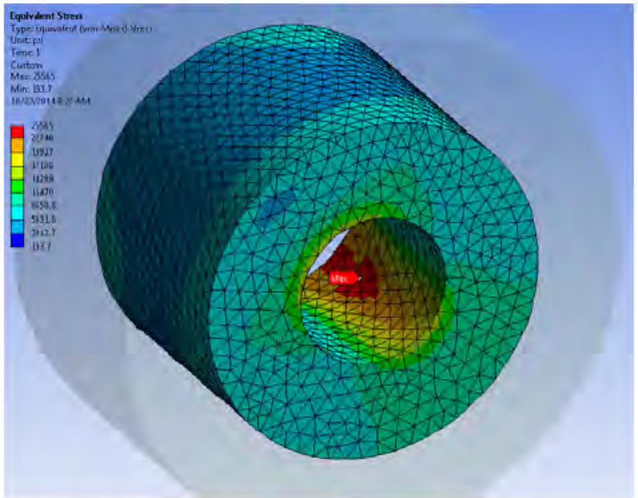
Volute Interface Structural Analysis

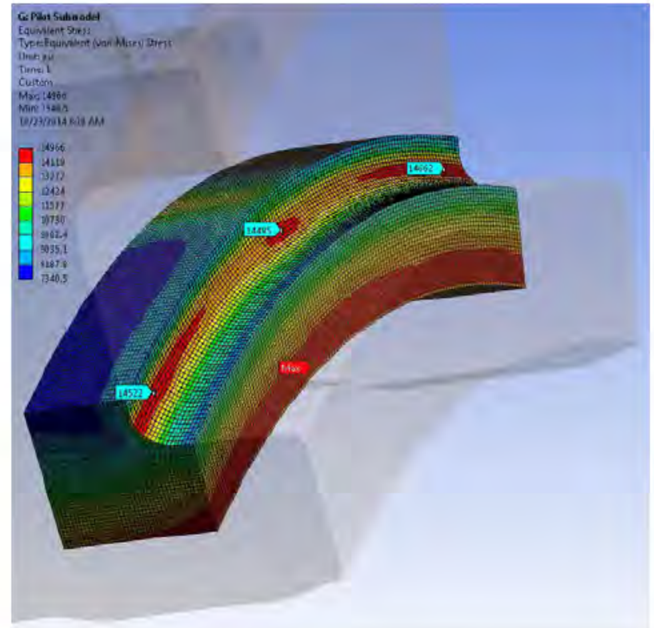


Volute Interface @ Yield

- Volute stress approaching yield (~30 ksi) for steady state thermal loading
- Bolt hole and pilot fillet stress verified with sub model (next slide)
- Nominal/max pilot gap of 1 mil (shown)
- 45 ksi stress at instrumentation pocket not representative of full geometry (reference lute FDR)

Volute Sub Model



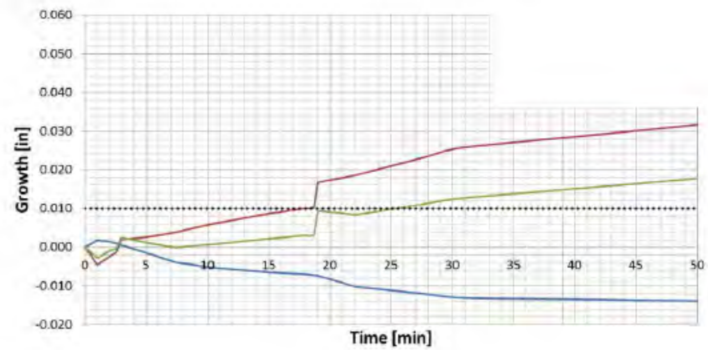


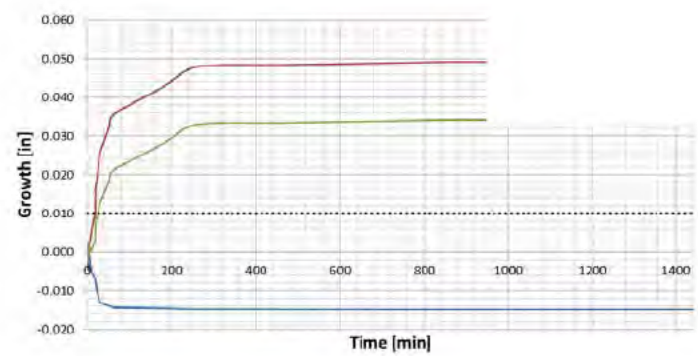
Volute Interface @ Yield

- **Bolt hole and pilot fillet stress verified with sub model, reference Volute FDR for accuracy with fully detailed model**
- **Nominal/max pilot gap of 1 mil (shown)**
- **Bolt hole stress unaffected at 0.5 mil max pilot interference conditions**
- **Pilot fillet stress increases to 24 ksi at 0.5 mil max pilot interference conditions**

Volute Relative Axial Growth

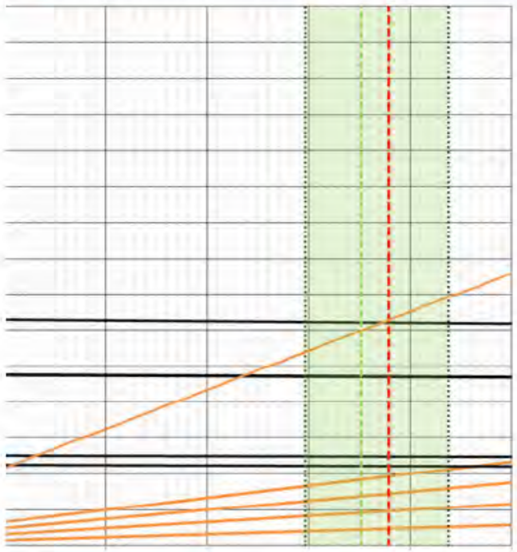
- [redacted] reaches maximum axial growth at ~50 mins
- At 50 mins all tip gaps are within 12 mils
- Volute continues to move away from [redacted] another 17 mils over the next ~200 mins, pulling [redacted] with it as static structure continues to heat up
- [redacted] grows additional 10 mils after 50 mins, reaching maximum axial growth at about the same time as static structure





Modal Analysis

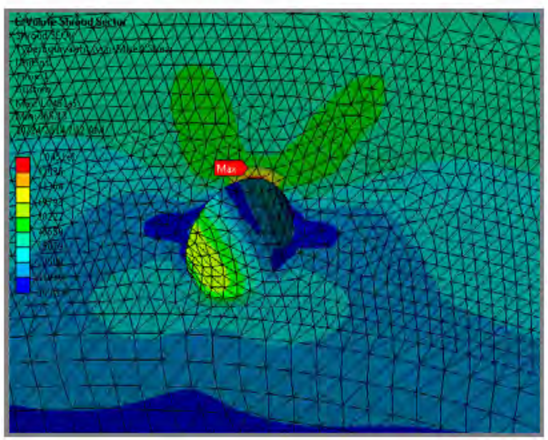
- [redacted]
- [redacted]
- No true resonance crossings (paired /rev stimulus with corresponding nodal diameter mode) within operating range for the first four nodal diameters
- Mode 1 ND 13 crossing with 13/rev (blade passing frequency) at 25,725 Hz (significantly outside of operating speed range)
- Presence of abradable has neglegable impact on frequencies
- Hollow [redacted] design maintains sufficient stiffness to keep resonance outside of operating range

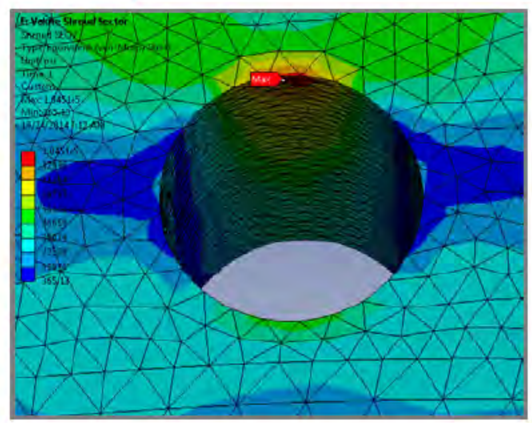


Instrumentation: 3/8" Prox Probe

- **3/8" prox probe hole creates Kt of 3 and stress approaching yield for S.S. conditions**
- **Items to explore:**
    1. **Mesh refinement/sub model for stress calculation accuracy**
    2. **Changing prox probe angle and where it projects through [redacted]**
    3. **Changing [redacted] profile for reduced thermal mass**
    4. **Shielding holes or slots to interrupt hoop stress field (ugly)**
    5. **E-P and/or fatigue analysis**

Stress ~Yield for Prox Probe





Fylde High Frequency FM Tip Clearance System For Interrupted Surfaces (Blade Tips)

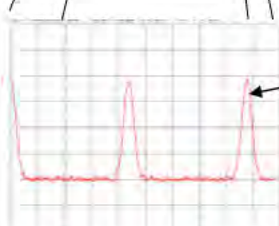
Engine enclosure

Triax Cable, 4m

BNC Cable

BNC Cable

BNC Cable



Main components

Capacitance Probe: design and manufacture

Oscillator and demodulator: supply

FE-497-OSC Oscillator, 1 per probe

RMS signal - DC level for mean clearance

P-P signal - Instantaneous output for individual tip clearance.

FE-457-CDT Demodulator, 1 per probe (units will be in 3U ½ rack)

Blade passing event. Peak inst voltage inversely proportional to distance from probe

Note: Slide courtesy of Engineering

Tip Clearance Components

- Two channels of high speed tip clearance equipment available at Redmond Lab
  - FE-497 Oscillator
  - FE-457 Demodulator
- Plan to install three prox probes into the rig and actively monitor two at a time.

Remaining Work

- 2D profile: Needs to change to enable instrumentation routing clearance to volute and space for axial shim
- Layout/Assembly:
  - [redacted] add clocking pin hole, remove two bolt holes for surface to which jacking screws will push off of
  - Volute: Add slots for lock ring (& washer?) for [redacted] bolt retention, add jacking screw threading to two holes, increase through hole diameter for Nord-lock clearance
  - Hardware: locator pin, bolts, Nord-locks (Rick has model), bolt retention (lock ring or lock ring & washer), jacking screws (same as shroud bolts?)
  - Try to place [redacted] bolt position blocked by actuator arm and two jacking screw locations ~120° apart
  - Nominal shim
  - Switch to E seal if current seal does not compress sufficiently with shim added
- Finalize instrumentation placement
  - TC: 3 locations along [redacted] flowpath arc length, 3 locations circumferentially (3x3)
  - Static pressure taps: 4 locations along [redacted] surface arc length, 3 locations circumferentially (4x3)
    1. 0.100" in front of [redacted] vane LE
    2. 0.100" in front of splitter vane LE
    3. [redacted] exit
    4. Halfway between locations 2 and 3
  - Prox probe
- Hot to cold conversion
- Fatigue assessment
- Drawings & BOM
- Tooling
  1. [redacted] fit up to bundle
  2. [redacted] bolting
- [redacted] FDR AI's… S:\ProjectManagementSystem\DevelopmentProjects_129416 D-R Super Compressor\Design Reviews\4_FDR

Material Candidate Strength Assessment

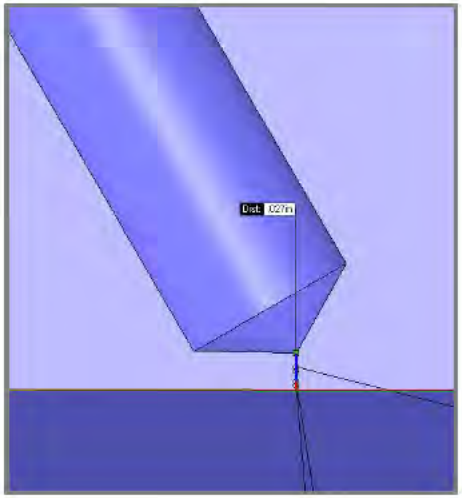
- Carbon steel lacks strength
- Low-alloy to stainless steel
  1. 304 SS lacks strength (30 ksi @ RT)
  2. 410 SS 60 ksi @ RT
  3. AISI 4130 plate, bar, or forging (normalized, tempered, stress relieved)
     - 70 ksi YS @ RT
     - 56 ksi YS @ 600°F
  4. AISI 4340 100 ksi @ RT
  5. 17-4 PH SS (plate or forging)
     - 105 – 170 ksi YS @ RT
     - 84 – 136 ksi YS @ 600°F
  6. 4330V bar or forging (quench & temper)
     - 185 ksi YS @ RT
     - 148 ksi YS @ 600°F
- Cast Iron
  1. ASTM A48 grey cast iron up to 60 ksi @ RT
  2. Ni Resist strengths up < 40 ksi @ RT

- Structural steel CTE 6.67E-6 $°F^{-1}$ (constant)
  - 410SS: 6.11E-6 $°F^{-1}$ @ 600°F
  - 17-4 PH: 6.4E-6 $°F^{-1}$ @ 600°F
  - 4130: 7.33E-6 $°F^{-1}$ @ 600°F
  - 4340: 7.58E-6 $°F^{-1}$ @ 600°F
  - Ni Resist: 5-9E-6 $°F^{-1}$ @ 600°F (could be selected for CTE if stress can be reduced)

Need to Finalize Material

Pressure Tap Ports

- **3 pressure tap port positions, each with 3 circumferential repetitions 120º apart (9 total)**



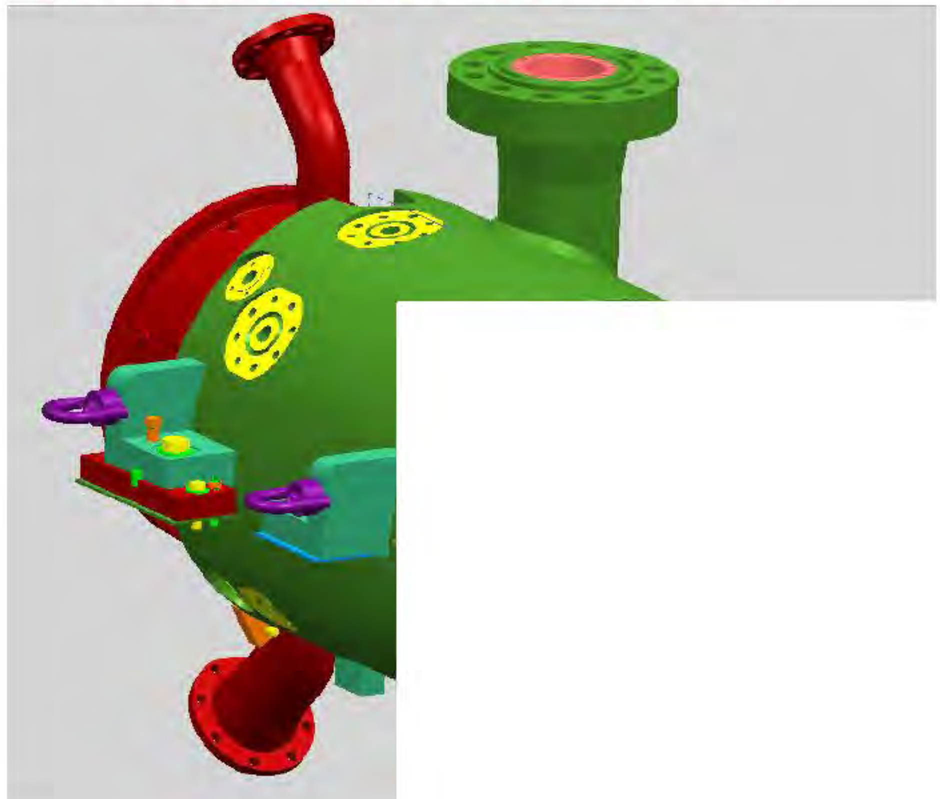

SUPERCOMPRESSOR
CASING DESIGN REVIEW
[redacted]
[redacted]
D. J. Peer
26 June 2014
[redacted]
DRESSER-RAND

Forged/Fabricated Case Construction

Rating:

2685 psig

650 °F



Case Design & Hydro Pressure

Split	Radial Split		Allowable Membrane Stress		
Material	D-R MAT'L #	Industry Designation	Room Temp	650°F	Hydro
Casing	090-598	ASTM A266, Cl 4	24.0	17 8	4525
Split Line Flange (Axial split only)					
Split Line Flange Studs (Axial split only)					
Split Line Flange Nuts (Axial split only)					
Discharge Nozzle	090-598	ASTM A266, Cl 4	24.0	17 8	4525
Discharge Flange	090-598	ASTM A266, Cl 4	24	17 8	4525
Inlet Head	091-762	ASTM A516 Grade 70, UT Quality	25.3	18 8	4517
Inlet Nozzle	090-598	ASTM A266, Cl 4	24.0	17 8	4525
Inlet Flange	090-598	ASTM A266, Cl 4	24	17 8	4525
Shear Rings/Retaining Rings	091-546	G4340			

- MAWP: 2685 psig
- Hydro Pressure: 4525 psig
  - 1.5 x MAWP:
    1.5 x 2685 = 4028 psig
  - 1.25 x (S_T/S) x MAWP:
    1.25 x (24/17.8) x 2685 = 4525 psig

PRIOR EXPERIENCE [redacted]

[redacted]

[redacted]

DRESSER-RAND

Case Design

- Leverage the DATUM D08 2995 psig / 500°F rating:
    -
    - D8 Shear Rings & Retaining Rings
    - 37 units built and tested
- Verbatim use of D8 2995 Case end, shear ring, and retaining ring within the D8 application envelope. The design is grandfathered and not included in the analysis.

Case Design, Cont'd

- Extrapolation to 650°F:
    - Allowable Stress (A266 Cl 4) @ 500°F: 19.6 ksi
    - Allowable Stress (A266 Cl 4) @ 650°F: 17.8 ksi
    - Allowable Rating @ 650°F:

      2995 x 17.8 / 19.6 = 2720 psig
- Inlet Flange:
    - Grayloc 10H84 Stainless steel ring carbon steel clamp and hubs.
    - Published Ratings:
        - 4059 psig @ 100°F
        - 3567 psig @ 650°F

Case Design, Cont'd

- Disch Flange Rating:
  - ANSI B16.5 Material Group 1.1
  - Class 1500 @ 650°F
  - Working Pressure = 2685 psig


DESIGN CRITERIA [redacted]
[redacted]
[redacted]
DRESSER-RAND

Case Design Criteria:

Generally in accordance with

ASME BPVC SEC VIII DIV 2 2013

1. *Protection against plastic collapse – qualified by elastic plastic analysis*
2. *Protection against local plastic failure – qualified by elastic plastic analysis*
3. *Protection against failure from cyclic loading – qualified by elastic analysis*

Protection Against Plastic Collapse Criteria:

1. *Equilibrium solution required for elastic-plastic material model for the following minimum load cases:*
   - *Design: 2.4 x MAWP = 6444 psig @ 650°F*
   - *Hydro: 2 x (S_T/S) x MAWP = 7240 psig @ 70°F*

Protection Against Local Plastic Failure

CODE Criterion:

$$\varepsilon_L = \varepsilon_{Lu} \cdot \exp\left[-\left(\frac{\alpha_{sl}}{1+m_2}\right)\left(\left\{\frac{(\sigma_1+\sigma_2+\sigma_3}{3\sigma_e}\right\}-\frac{1}{3}\right)\right]$$

$$\varepsilon_{peq} + \varepsilon_{cf} \le \varepsilon_L$$

- ε_L triaxial strain limit
- ε_{LU} uniaxial strain limit (use m_2 if EL or RA are not specified)
  - = 2 ln(1+EL/100)
  - = ln(100/(100-RA))
- α_{SL} material factor for the multiaxial strain limit
- m_2 uniaxial strain limit =.6(1-S_Y/S_U)
- ε_{peq} equivalent plastic strain
- ε_{cf} forming strain = 0 after stress relief

From Table 5.7 for ferritic steel

Protection Against Local Plastic Failure

- ***Load Case:***

  1.7 x MAWP (4565 psig)

  & Rated Temperature

- ***Criterion:***

  $\ln(\varepsilon_p)<(\ln(m_2)-[(\alpha_{SL}/(1+m_2))*((S1+S2+S3)/3/SEQV-1/3)])$

- ***User Defined Result:***

2.302585*(log10(EPPLEQV_RST) -log10(0.3569))+((S1+S2+S3)/3/SEQV-1/3)*2.2/1.3569 < 0

  - *Positive value indicates local failure*
  - *No value indicates no plastic strain*

Protection Against Failure from Cyclic Loading:

- Elastic Stress Analysis
- Load Case: MAWP & Rated Temperature
- S_A = Maximum Equivalent Stress / 2
- Minimum Number of Load Cycles = 1000

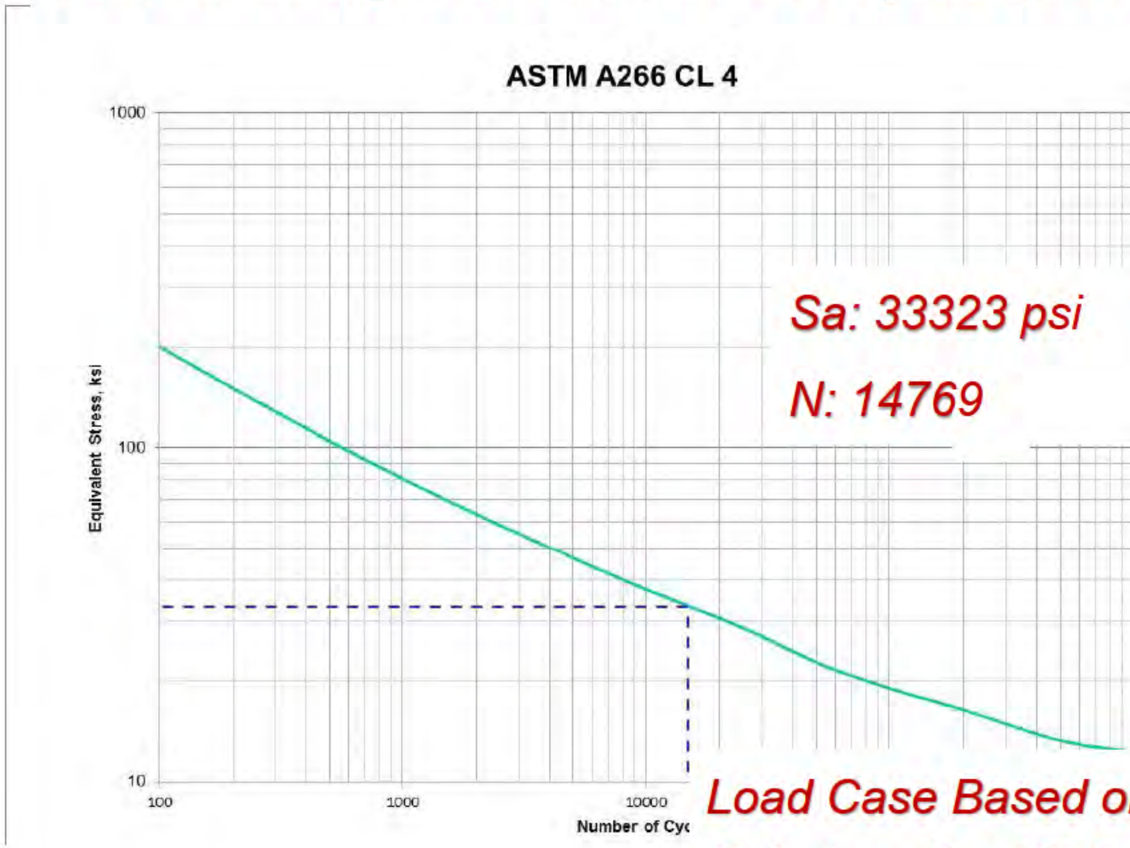
  (Weekly for 20 years)
- Fatigue Curve per ASME BPVC SEC VIII DIV 2 Annex 3F

CASE [redacted]

[redacted]

[redacted]

Protection Against Failure from Cyclic Loading

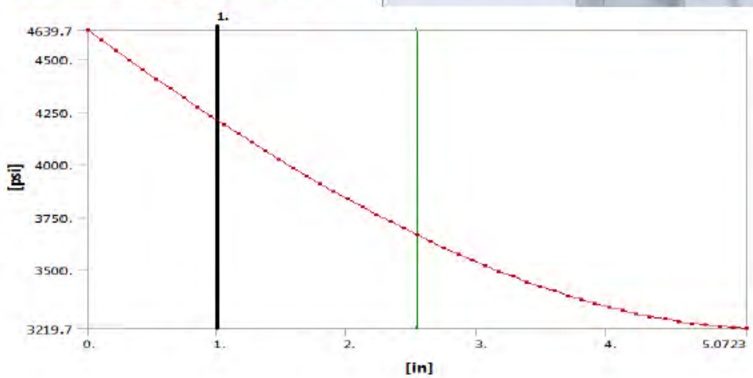


Load Case Based on Design Criterion: N > 1000

Linearized Equivalent Elastic Stress @ 2685 psig & 650°F

Allowable

- *Membrane: 17.8*
- *Membrane plus Bending: 26.7*



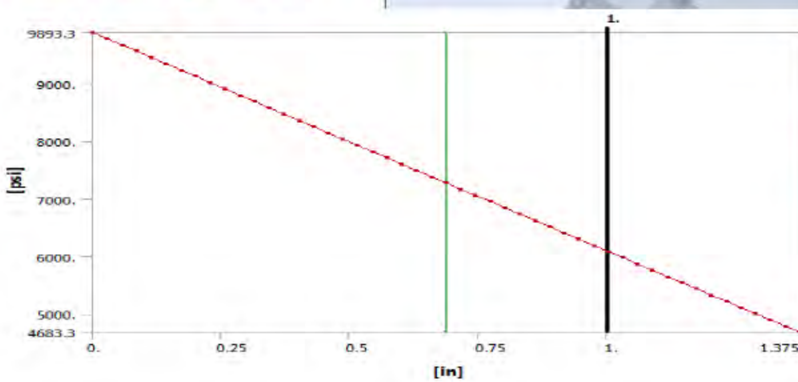
Actual

Membrane plus Bending = 4.6 ksi

Linearized Equivalent Elastic Stress @ 2685 psig & 650°F

Allowable

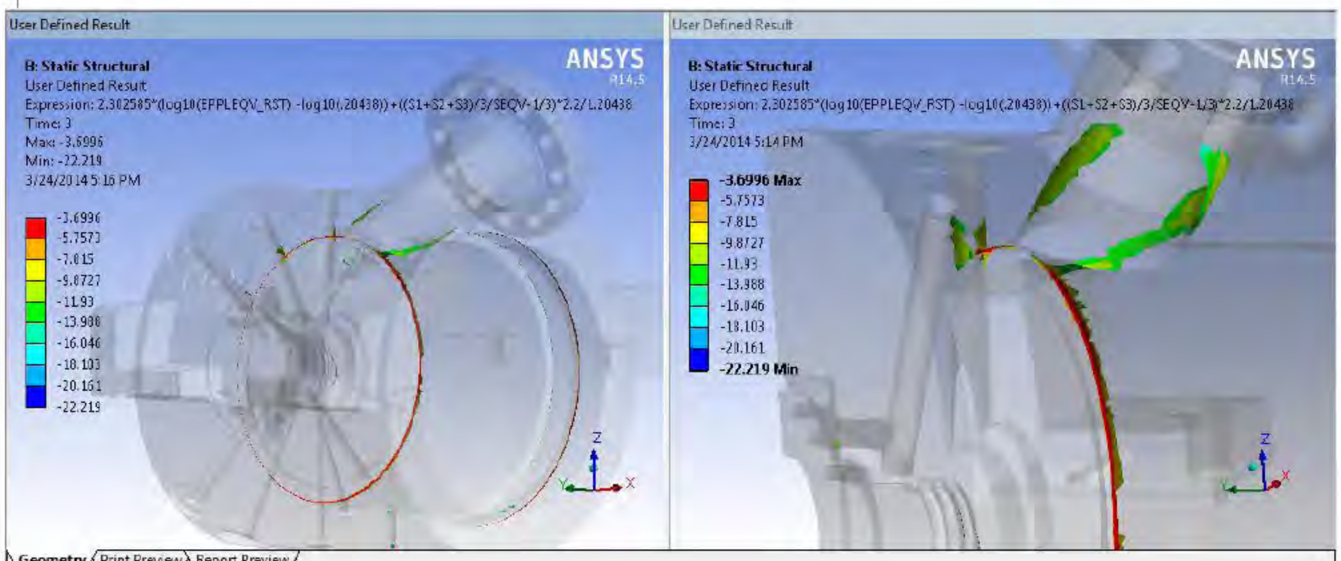
- *Membrane: 17.8*
- *Membrane plus Bending: 26.7*



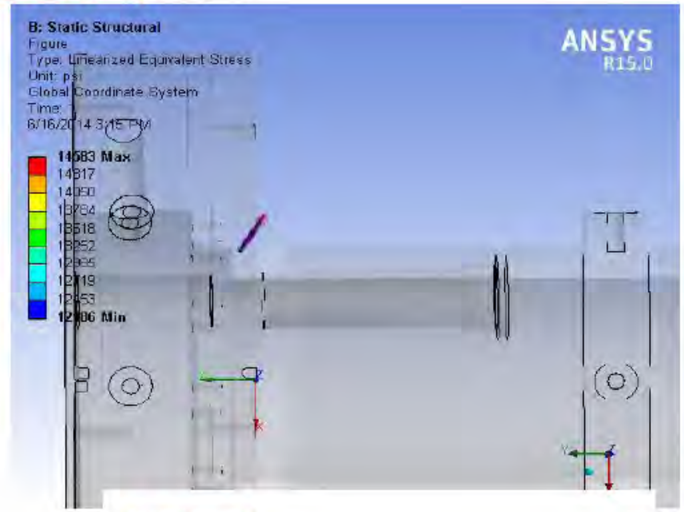
Actual

Membrane plus Bending = 9.9 ksi

Protection Against Local Plastic Failure: Equivalent Plastic Strain Parameter < 0 @ 1.7 x Design Pressure & 650°F

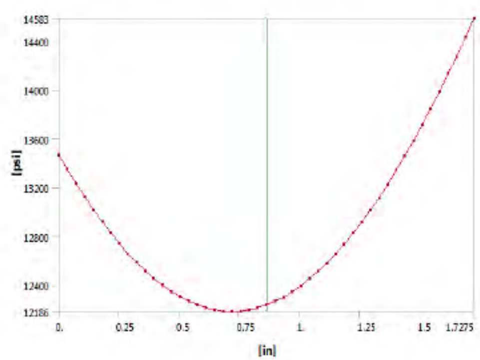


Linearized Equivalent Elastic Stress @ 2685 psig & 650°F



Allowable

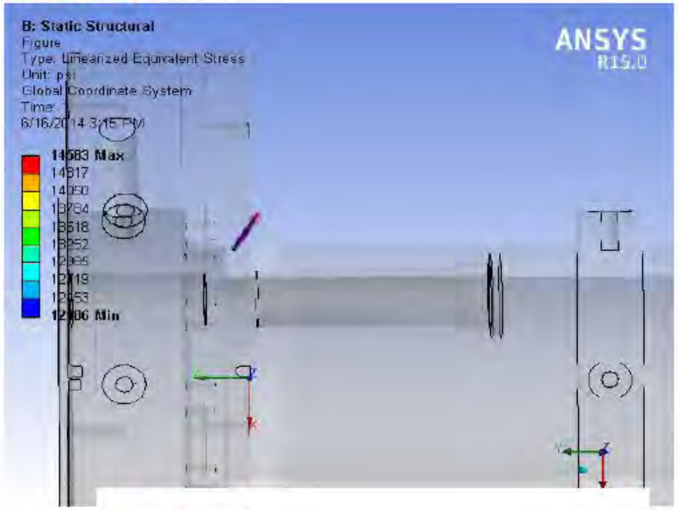
- *Membrane: 17.8 ksi*
- *Membrane plus Bending: 26.7 ksi*



Actual

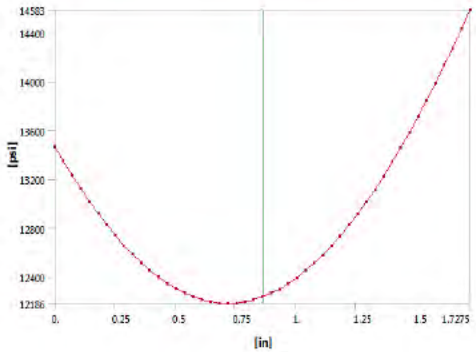
- *Membrane: 12.4 ksi*
- *Membrane plus Bending: 14.6 ksi*

Linearized Equivalent Elastic Stress @ 2685 psig & 650°F



Allowable

- *Membrane: 17.8 ksi*
- *Membrane plus Bending: 26.7 ksi*



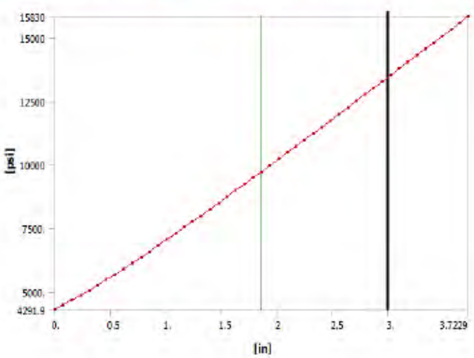
Actual

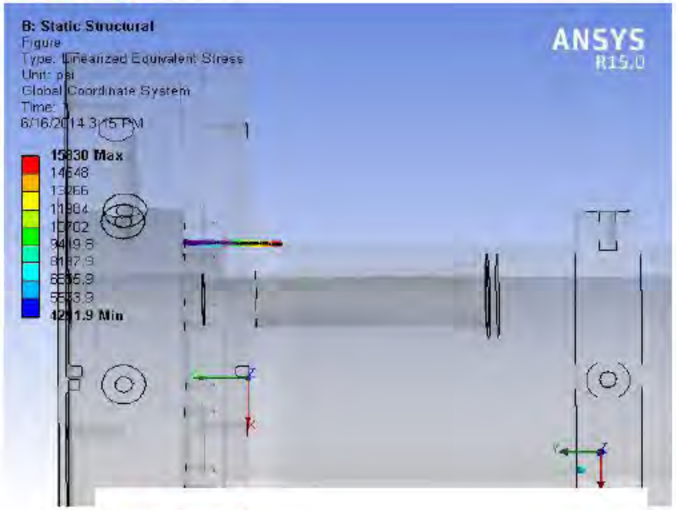
- *Membrane: 12.4 ksi*
- *Membrane plus Bending: 14.6 ksi*

Linearized Equivalent Elastic Stress @ 2685 psig & 650°F

Allowable

- *Membrane: 17.8 ksi*
- *Membrane plus Bending: 26.7 ksi*





Actual

- *Membrane: 9.8 ksi*
- *Membrane plus Bending: 15.8 ksi*

[redacted]

Diffuser & Volute

[redacted]

Dave Peer

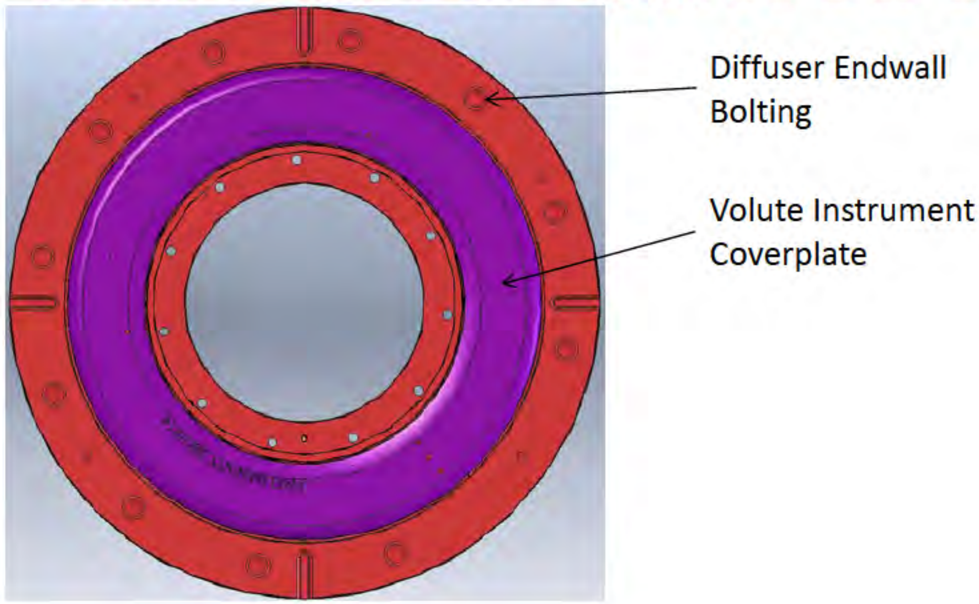
Ryan Edmonds

Dave Taylor

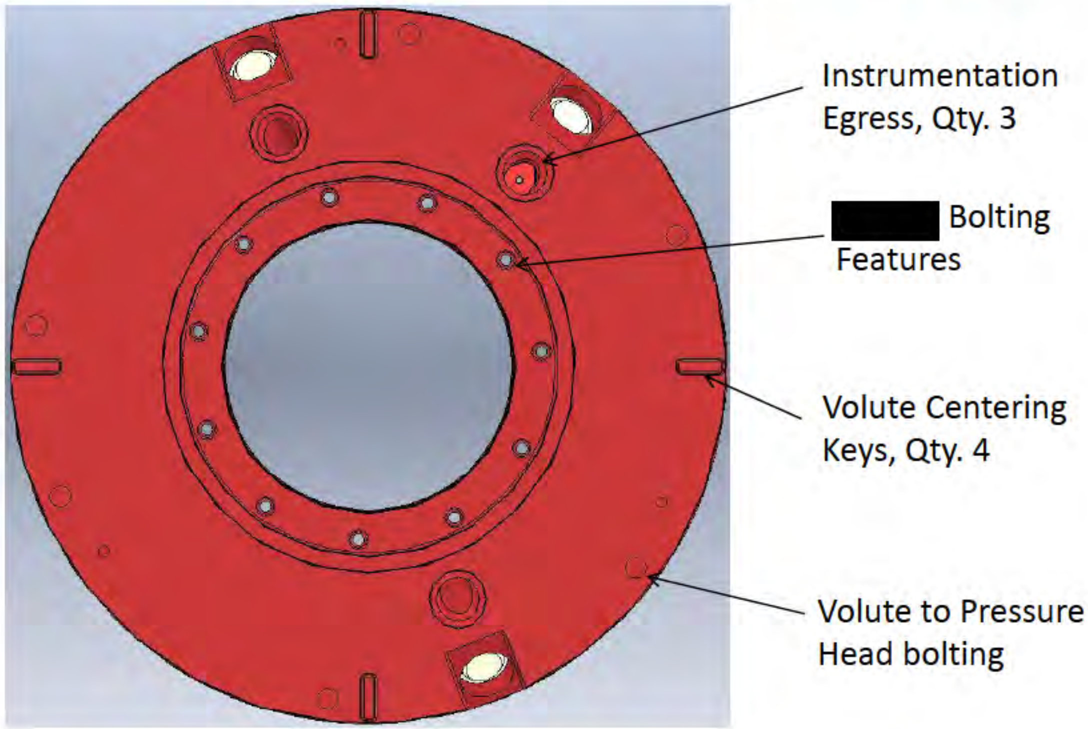
[redacted]

DRESSER-RAND

Volute Diffuser Features DE Side



Volute Diffuser Features NDE Side



Volute Coverplate Features

Metal Seal Gland

Static Pressure ports
12 total, 4 X 3
locations

Pressure Recess
Pocket

Blind Tapped Holes
for Mounting

Rig Instrumentation List

DATUM-S Rig Instrumentation List				Priority 10 = Mandatory 7 = Preferred 5 = Nice To Have 3 = Optional 1 = Wishful Thinking
Location	**Type of Probes**	**Details**	**Quantity**	
Inlet spool	Total pressure probes	Kiel head probes	4	10
	Static pressure taps		4	10
	Total temperature probes	Type J or T thermocouples	4	10
[redacted] eye	Static pressure taps		3	10
	Dynamic pressure transducers		3	7
	Total pressure probes	1/8" Kiel head probes	3	5
	~~Total temperature probes~~	~~Type J or T thermocouples~~	~~3~~	~~3~~
[redacted] exit	Static pressure taps		3	10
	Dynamic pressure transducers		3	10
End of diffuser pinch	Static pressure taps		3	10
	Dynamic pressure transducers		~~3~~ 2	7
Mid-radius diffuser	Static pressure taps		3	5
Diffuser exit	Static pressure taps		3	10
	Dynamic pressure transducers		~~3~~ 1	10

Rig Instrumentation List

Volute throat	~~Total pressure probes~~	~~Kiel head probes~~	~~3~~	~~7~~
	~~Static pressure taps~~		~~3~~	~~10~~
	~~Total temperature probes~~	~~Type J or T thermocouples~~	~~3~~	~~3~~
[redacted] Recess [redacted] side)	Static pressure taps	~~2 taps at 6 different location along the cover~~ 4 taps along [redacted] 3 circumferential positions	12	5
[redacted] Recess (hub side)	Static pressure taps	~~2 taps at 2 (or 3) different locations along the disk~~	3 taps ~~4 (or 6)~~	5
(downstream of BP)	Static pressure taps	Outside Pressure Case	2	5
	Temperature probe	Thermocouples	2	5
Exit spool	Total pressure probes	Kiel head probes	4	10
	Static pressure taps		4	10
	Total temperature probes	Type J or T thermocouples	4	10
FLOW MEASUREMENTS				
Inlet flow	~~Orifice run?~~ Cone Meter	Typical instrumentation		
	~~Orifice run?~~ Cone Meter	Typical instrumentation		
	Metal Temperature	Type K TC	3	
	Seal gap	Capacitance Proximity probe	3	

Volute/Diffuser Axial Forces

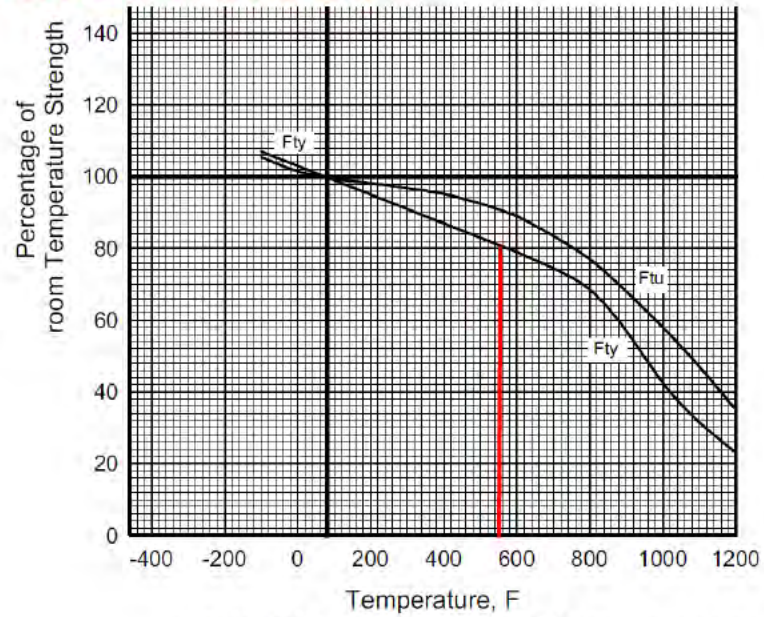
	P (psi)	Ri (in)	Ro (in)	F (lbf)	Notes:
F_1	220			21118	
F_2	2200			856884	
F_3	varies			398542	Integrated from CFD
F_4	varies			-366978	Integrated from CFD
F_5	1390			-301694	
F_6	2200			-449213	
F[redacted]p	varies	-	-	-111997	Net force, integrated from CFD
F[redacted]nlet	220			15457	
			NET FORCE	***62118***	"+" force is towards driven end

9/22/2014 Corrected Radius in yellow highlighted cell.

Net Force Still Toward Driven End.

Volute/Diffuser Axial Forces

- F_{bolts} = 427000 lbs
- 1-8 SHCS have combined proof load capability:
  - 899,800 lbs at room temperature.
  - 719,840 lbs at operating temperature.



Note: Data from Mil Handbook 5, for low alloy steels

Fasteners between Volute and Diffuser Endwall have Adequate Load Capacity.

DATUM-S Volute Structural Analysis

- Analysis Performed using
  - 3D model of Volute + Volute endwall
  - 2D Temperatures mapped to 3D model
  - 'bulk' pressures mapped to relevant surfaces
    - Inlet pressure of 220 psi
    - [redacted] Exit pressure of 1390 psi
    - Diffuser pressure radially varying
    - Volute gaspath pressures of 2200 psi
- Results
  - Pressure loads only
    - Displacements appear reasonable & stresses are < 20 ksi
  - Thermal Loads only
    - Results suspect due to over constraint at interface with Diffuser – need to add to model

DATUM-S Volute : Boundary Conditions

Frictionless restraint

Compression only at interface with diffuser <1>

- Material: 090-601 (A36 Plate)
- At 600F:
  - UTS = 58 ksi
  - YS = 24.5 ksi

Elastic support epresenting Belleville tack

Frictionless on sides of keys (4)

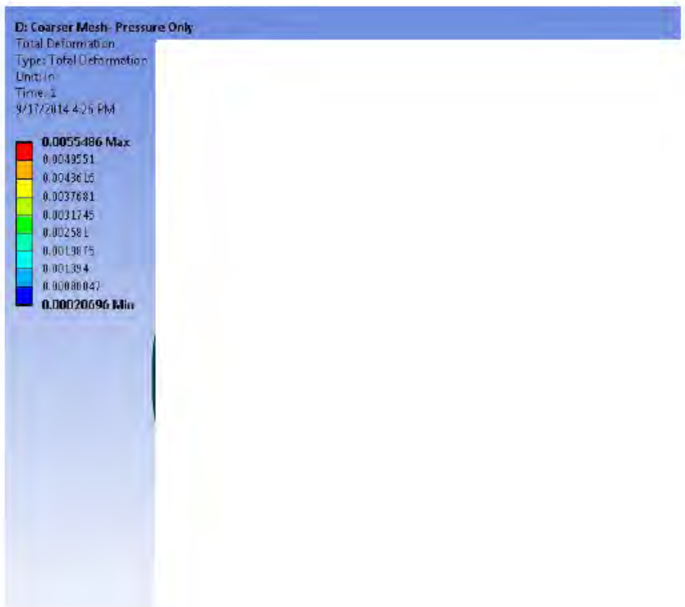
Compression only at bolt head

<1> This leads to over-constraining the model.
Diffuser being added

DATUM-S Volute Displacements – Thermal Loads only

Add Diffuser to analysis model

DATUM-S Volute Displacements – Pressure Loads Only



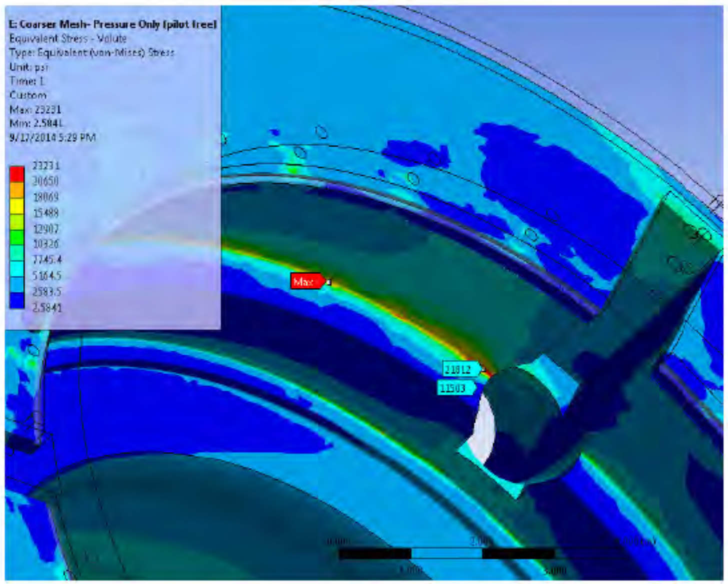
- Displacements due to pressure loads unaffected by radial constraint at interface with diffuser

Displacements show some variation around volute

Volute run01.wbpj

DATUM-S Volute Stress – Pressure Loads Only

Equivalent Elastic Stress

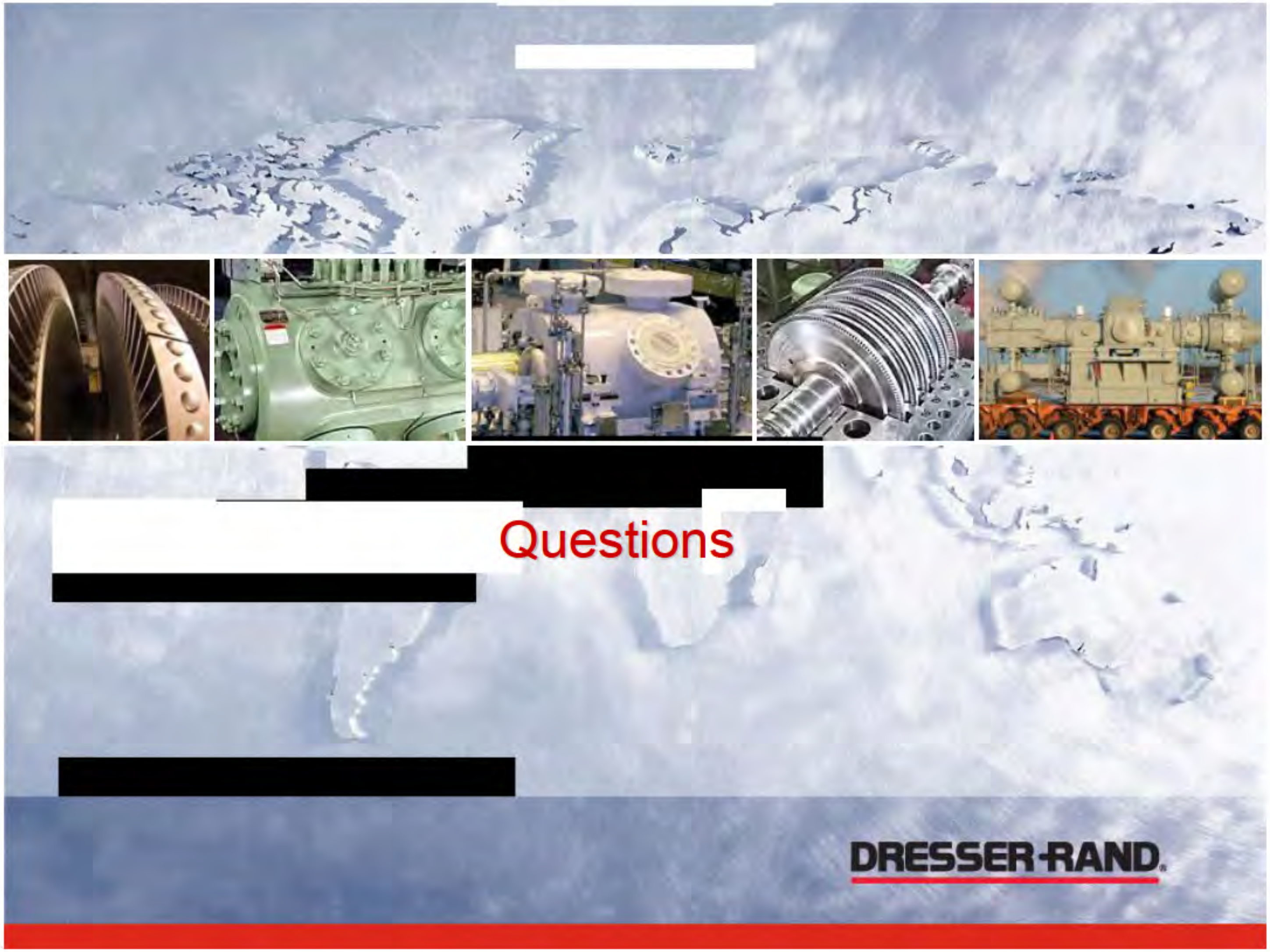


- Pressure loads only:
  - Highest stress in volute approx 23 ksi (local)
  - Peak temp = 565F
  - At 600F yield stress = 24.5 ksi

Local Stresses approaching yield with pressure loads alone.
Add Diffuser and thermal loads, improve mesh and assess.
May need to consider material change ?

Remaining Work/Analysis Tasks

- Add Diffuser to FEA model for more representative modeling of radial 'pilot'
- Add 3D pressures from CFD to FEA model
- Assess elastic analysis results and if problematical run elastic-plastic FEA analysis
- If still problematical, consider material change. Or change material anyway ?
- Finalize volute and diffuser endwall pilot or centering feature.
- Confirm all instrument sizes and routing.
- Design instrument bayonet with metal seals.
- Finalize key features between diffuser endwall, volute, and pressure head.


[redacted]
Questions
[redacted]
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DRESSER-RAND

[redacted]

DATUM-S Volute FDR - Structural Analysis

[redacted]

Dave Taylor

[redacted]

DRESSER-RAND

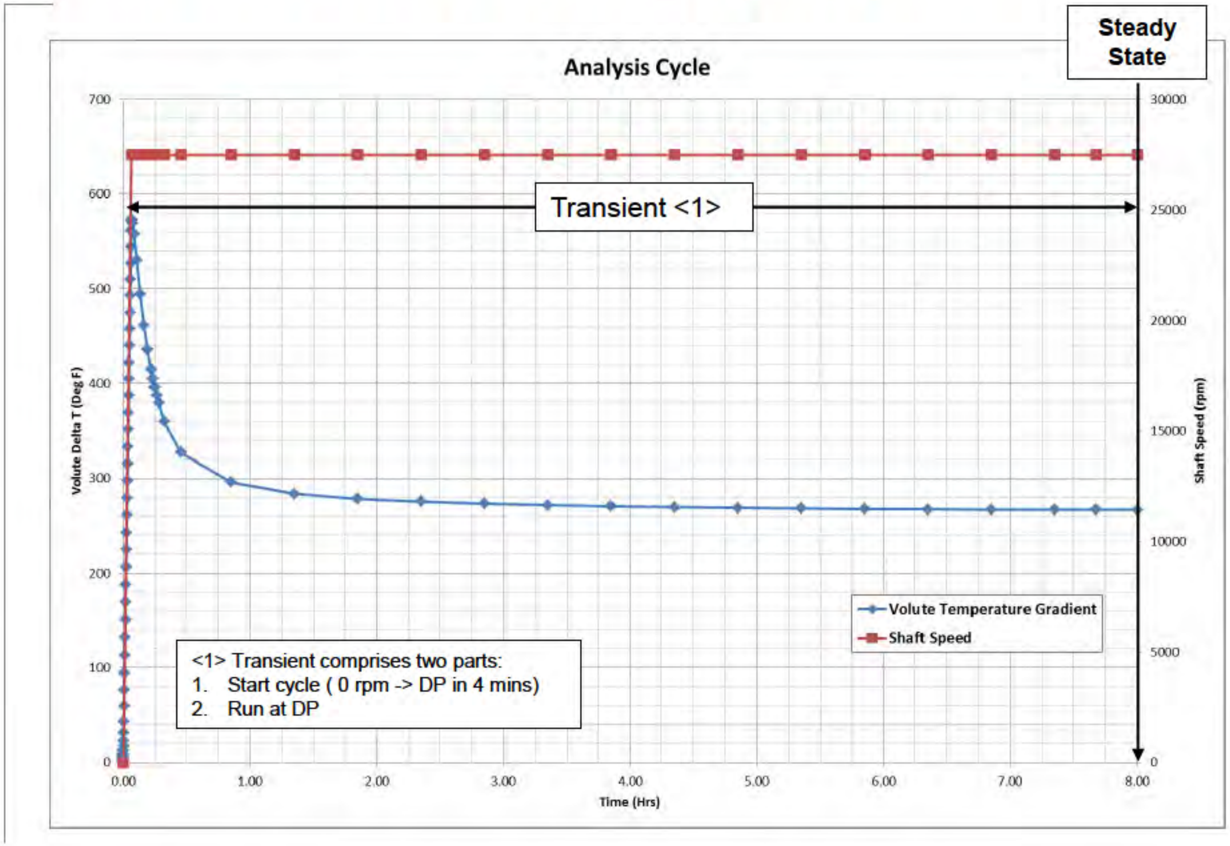
DATUM-S Volute Structural Analysis

- At Design Review on 9/19 preliminary results for volute structural analysis presented
  - Indicated some high stress in part but boundary conditions needed improving
- Model has been updated
  - Added Diffuser wall and de-featured pressure case
    - Subsequently determined only Volute
  - Added 3D pressure map from CFD
- Analyses shown here:
  - Steady State & Start Cycle
  - 2D Model of compressor
  - 3D model of Volute + Volute
    - 2D Temperatures mapped to 3D model
    - Inlet pressure of 220 psi, [redacted] Exit pressure of 1390 psi
    - Diffuser pressure radially varying
    - Volute & Diffuser gaspath pressures from CFD

DATUM-S Volute Structural Analysis : Results Summary

- At steady state condition
  - Stresses in volute are generally < yield
    - A couple of regions where local peak stresses exceed yield:
      - 39 ksi at [redacted] fastener thru hole (YS = 31 ksi and UTS = 58 ksi)
      - 42 ksi at instrumentation cutout
- At 'transient' conditions (i.e before temperatures have soaked)
  - Stresses in volute exceed UTS
    - At 4 mins (end of start ramp) peak stress = 80 ksi
- Work outstanding
  - Re-profile Instrumentation cutout to further reduce steady state stress
  - Structural analysis with modified start cycle
    - Iterate 2D to obtain acceptable cycle profile then confirm with 3D

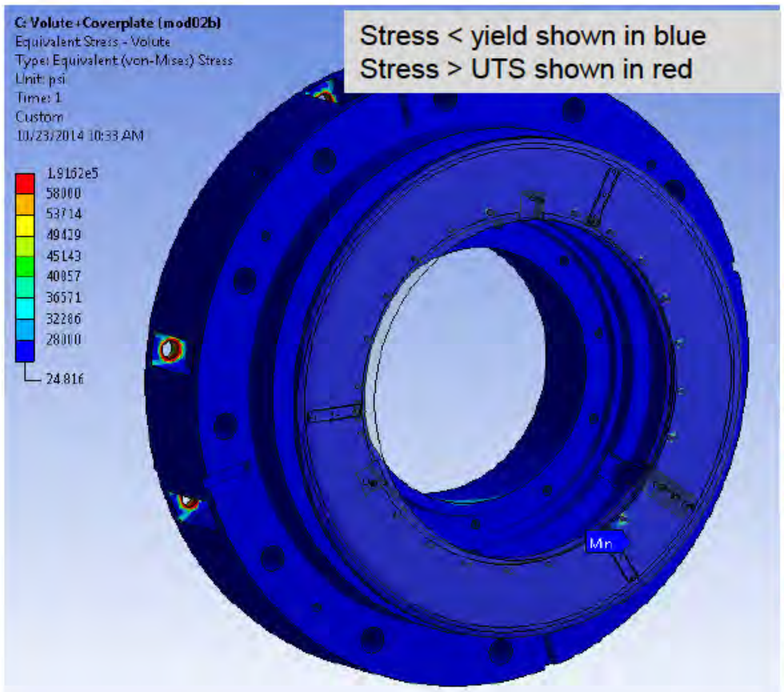
DATUM-S Volute Structural Analysis Cycle



DATUM-S Volute Structural Analysis

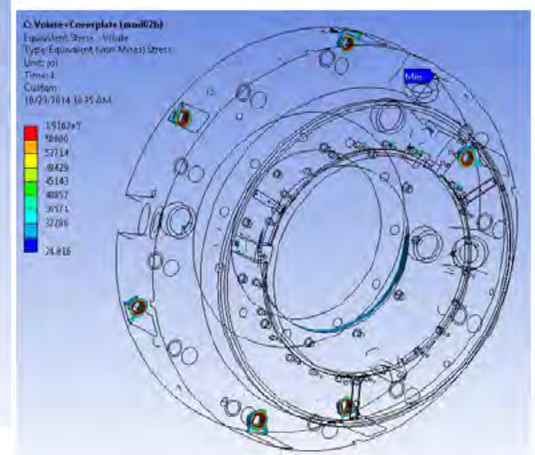
Steady State

DATUM-S Volute : Equivalent Stress at Design Pt, S/S



Almost all of volute < yield

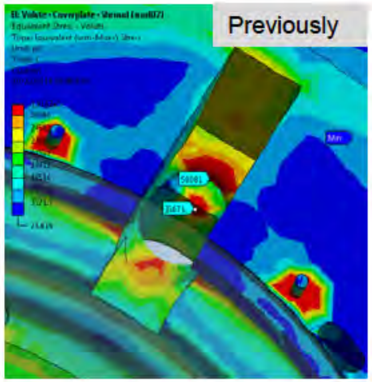
Areas around instrumentation cutouts exceed yield locally. See next slide



See next slide for more details

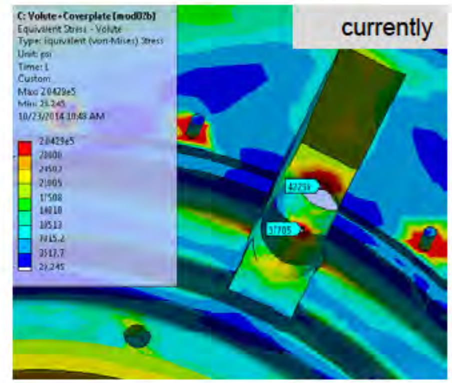
Volute run04.wbpj

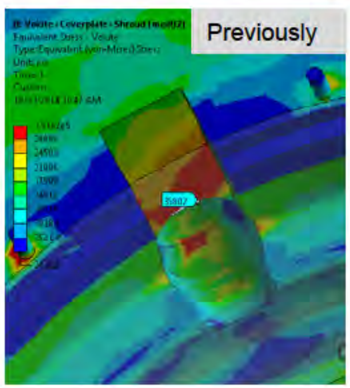
DATUM-S Volute : Equivalent Stress (Design Pt, S/S)



Cutout at 2 o'clock

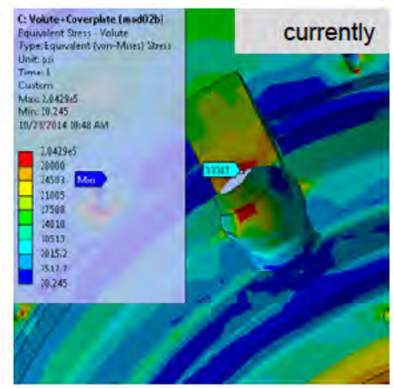
Reduces 50ksi -> 42 ksi
When 'roof' moved in .1"





Cutout at 10 o'clock

Peak reduces 35ksi -> 33 ksi and region above yield is more localized when 'roof' moved in .1"

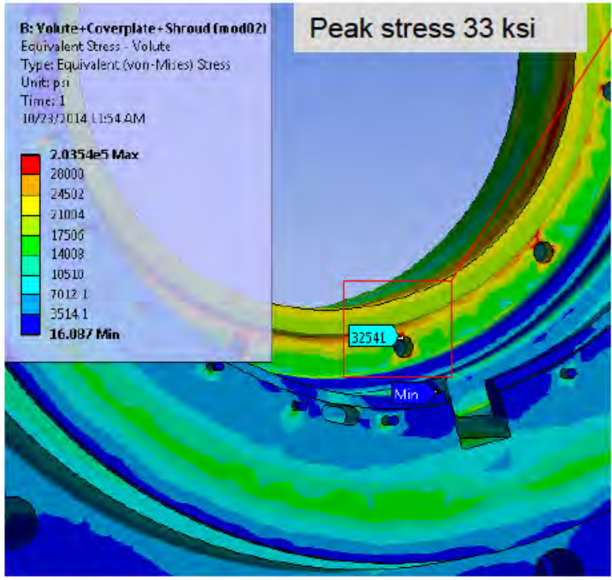


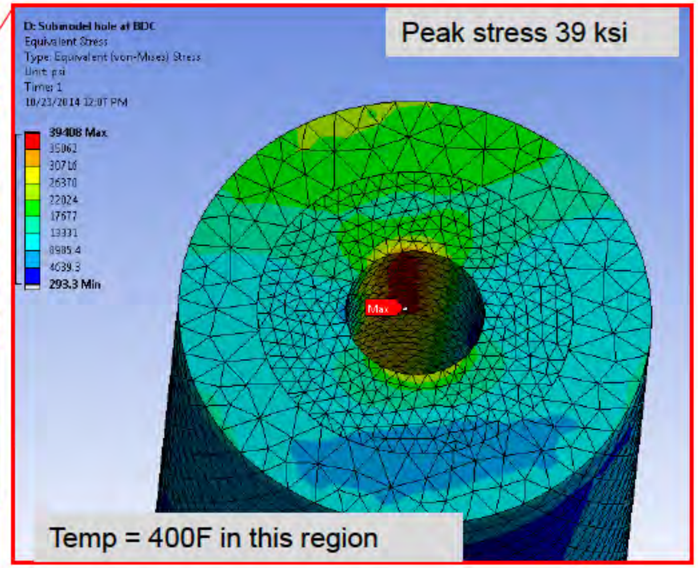
Some local regions at instrumentation cutouts > yield
May be possible to reduce further

Volute run04.wbpj

DATUM-S Volute Stress at thru hole (Design Pt, S/S)

Submodel of thru hole – simplified volute, Steady state (4hr?) pressures and temperatures





Volute is A36:
At 400F, UTS = 58 ksi, Yield = 31 ksi

- Submodel gives slightly higher stress than 'master' model
- Stress at thru hole is above yield, below UTS - Localized

Volute run04.wbpj

DATUM-S Volute Structural Analysis

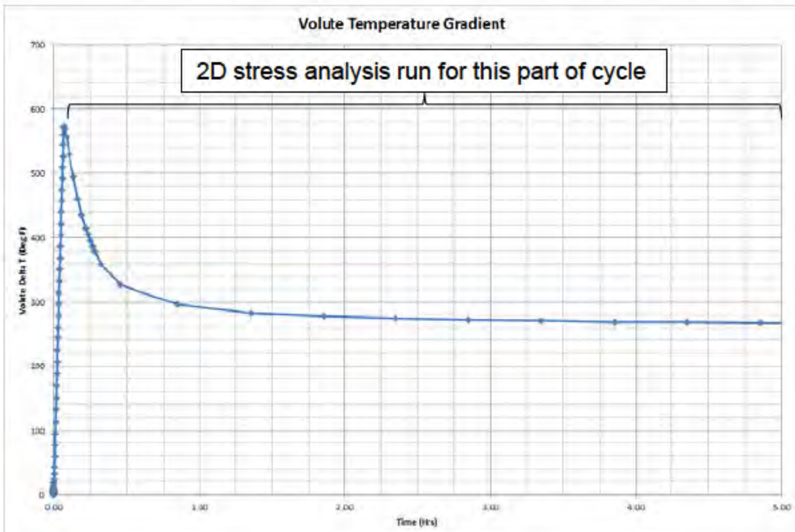
Transient

DATUM-S Volute : Transient Temperatures

Temperatures obtained from 4-min accel to 100% speed, then dwell to 15 mins

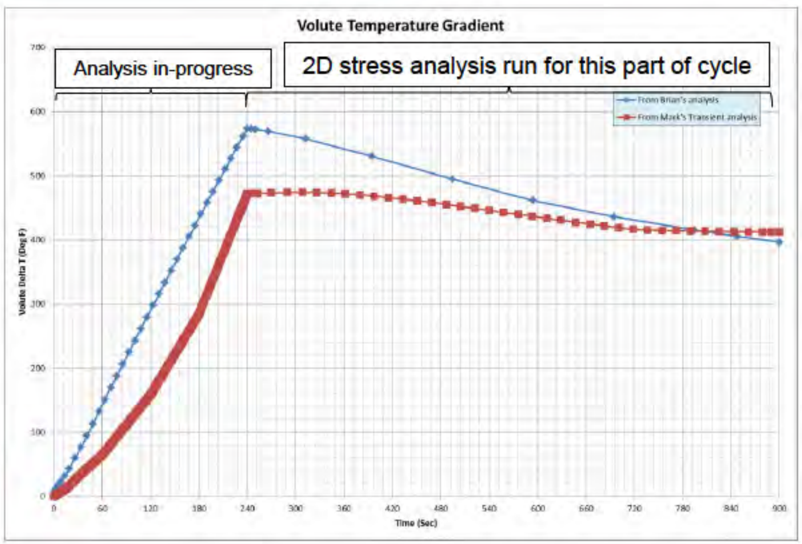
DATUM-S Volute : Temperature gradient

Volute reaches thermal equilibrium around 1-5 hrs



DATUM-S Volute : Temperature gradient

Stresses shown on next slide.
However, based on temperature gradient, stresses at 240s (4 min) should be the largest.



Although 2D temperatures available for 0 – 4min portion of start cycle, structural analysis not run.

Updated cycle being analyzed, using most current geometry and structural analysis will be performed using temperatures from that.

Analyses shown differ because of way part speed BC's calculated (blue line is linear interpolation, red line calculated at 25%, 50%, 75% and 100% speed).

DATUM-S Volute : Transient Stress from 3D FEA

Map 2D transient temps to 3D model, No pressure loads <1>

<1> Note: Ran temperatures only so that coverplate not required (runs 5x quicker without)

Analysis at 240s with End-of-Ramp temperatures + DP pressures shown on next slide

3D model shows stresses peaking at end of start ramp (same as 2D)

DATUM-S Volute : Stress from 3D FEA

Analysis at End of Ramp (240s)– Pressure & Temp

3D model shows stress peak in similar location to simplified (no coverplate) 3D model with temperatures only
Stresses exceed Yield in many locations and UTS in places

DATUM-S Volute : Stress from 3D FEA

Steady State Condition – Pressure & Temp

3D model shows stress peak in similar location to 2D model
but localized at cutout for instrumentation

[redacted]

Questions

[redacted]

Dave Taylor

dataylor@dresser-rand.com

(425) 828-4919 x230

425-229-0138

[redacted]

DRESSER-RAND

Appendix 9.3.2

Rotor Seal

Agenda

- Review Requirements
- Design Overview
- Heater Selection and Electrical Assy
- Material Review and Radial Gap Predictions
- Instrumentation Details
- Work to Do
- *Supporting Data*

Requirements

- [redacted]
- [redacted]
- Ensure free-running assembly in cold condition and no interference with the during operation
- Size heaters to achieve 750ºF
- Target rise time to 750ºF of less than 20 minutes
- No requirements on cool off/contraction time
- Incorporate instrumentation defined in controls and instrumentation design reviews

Assumptions

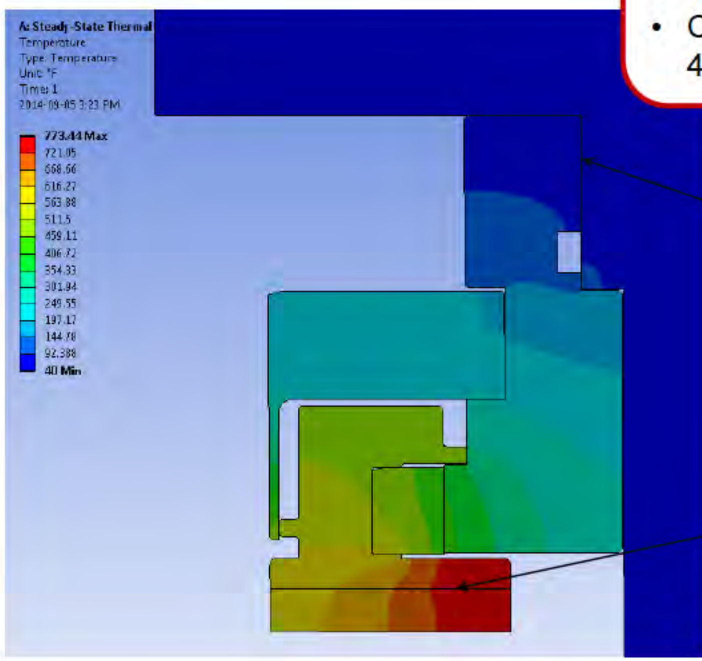
- Speed of compressor can be incrementally ramped up to keep pace with the temperature rise of the to maintain gap control as required during start-up
- Precision gap control is not required during shut-down
  - i.e. rotor stability does not require control of the gap during shut-down

DESIGN OVERVIEW

HEATER SELECTION & ELECTRICAL ASSY

Preliminary Steady-State Thermal Model Results

- Estimates are that ~3000W is required to achieve target temperature of ~750°F
- 4500W reaches 730°F within ~20 min
- Current heater configuration provides 4680W at room temperature



Primary heat flux is out

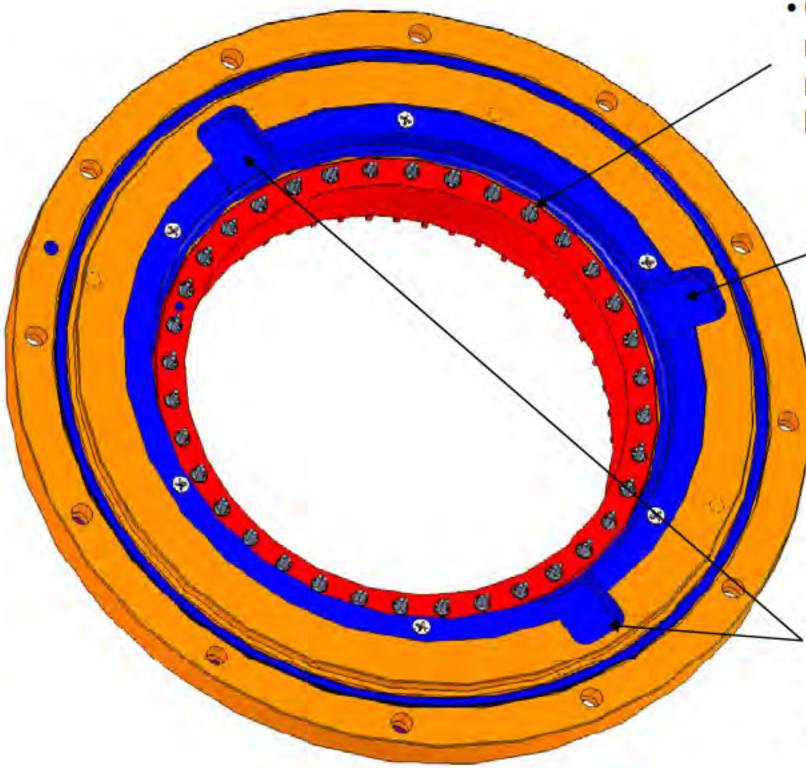
3000W applied on a portion of this line.

Need approximately 4500W to heat to over 730°F in 20 min.

Cartridge Heaters

- Ø.375 x 1.50 cartridge heaters were selected
    - 130W each at 240Vac supply
    - Qty: 36
    - Readily available
    - Supply target power
    - Large surface are for small power density
    - Uniform heating
    - Range of available power levels (100-250W per heater)
- Also considered
    - Coil heater
    - Interference fit construction heaters

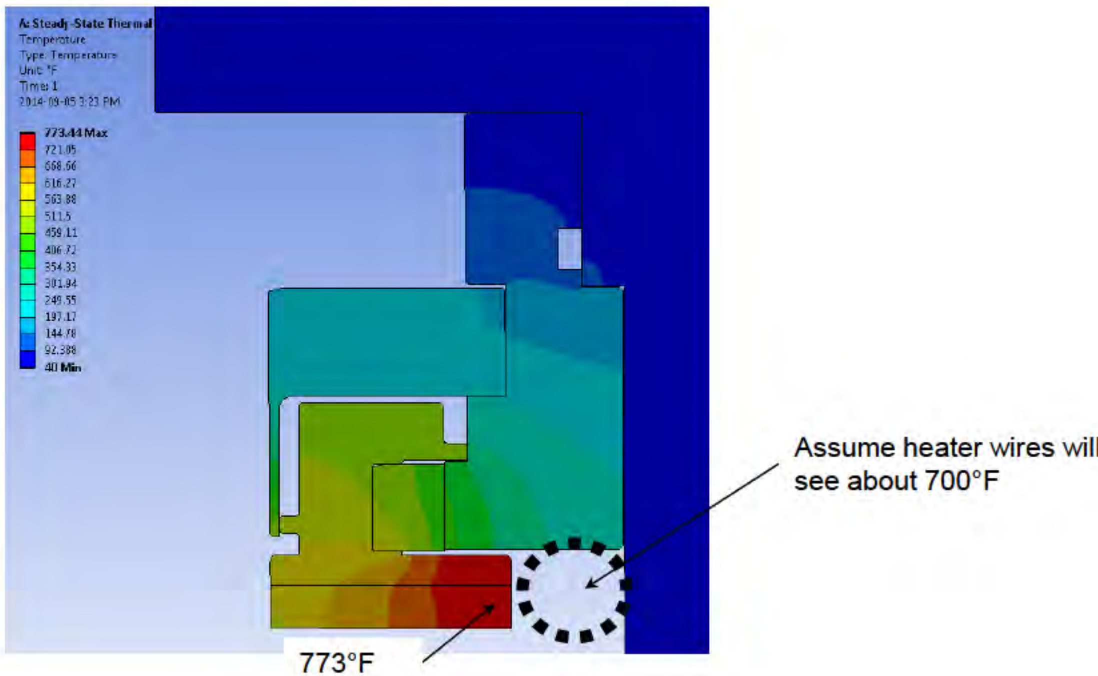
Routing Scheme



- Challenge with the large number of cartridge heaters is managing the connecting wire bundle
- This port reserved for proximity probes
- Planning on 6 heater circuits to distribute power, incorporate redundancy, and minimize wire sizes, using these 2 ports
- Need to lay out details

Thermal Considerations for Wiring

Source: Steady-State Thermal Model, 9/5/2014



Splice Candidates

TE connectivity

STRATO-THERM* Terminals, Splices, and Spare Wire Caps

Application Specification
114-2162
17 JUN 11 Rev D

NOTE: *All numerical values are in metric units [with U.S. customary units in brackets]. Dimensions are in millimeters [and inches]. Unless otherwise specified, dimensions have a tolerance of ±0.13 [±.005] and angles have a tolerance of ±2°. Figures and illustrations are for identification only and are not drawn to scale.*

1. INTRODUCTION

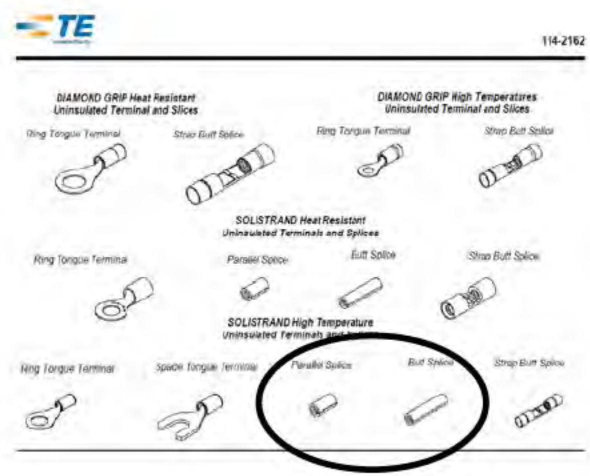
This specification covers the requirements for application of STRATO-THERM terminals, splices, and spare wire caps for high temperature applications. There are two types of these terminals, spices, and spare wire caps. One type is insulated, which consist of PIDG* (pre-insulated diamond grip) ring tongue terminals, splices, and spare wire caps insulated with polytetrafluoroethylene (PTFE), post-insulated, or pre-insulated, and the other type is uninsulated which consist of DIAMOND GRIP* (insulation support) ring tongue terminals, spade tongue terminals, and splices with SOLISTRAND' (non insulation support). The uninsulated are available in heat resistant and high temperature. The terminals, splices, and spare wire caps accept solid or stranded wire for single applications.

The pre-insulated terminals, splices, and spare wire caps are designed for reliable performance at maximum temperatures of 288°C [550°F] and at 260°C [500°F] for silver-plated post-insulated terminals and splices. The uninsulated heat resistant terminals and splices operate at maximum temperature of 343°C [650°F], and at 649°C [1200°F] for high temperature terminals and splices.

The terminals and splices are color-coded to provide a visual reference applicable to the wire size range suitable for the terminal or splice. In addition, terminals are marked on the tongue with the wire size range. The serrations or dimples inside the wire barrel provide maximum contact and tensile strength after crimping. The terminals are suitable for mounting and accept stud sizes M2.5 [4] through M8 [.375] (inside diameter range of 2.95 through 9.80 [.116 through .386]). The terminals, splices, and spare wire caps are available in loose-piece for terminating with manual and pneumatically-powered hand-held tools, and in tape-mounted form for terminating with semi-automatic hand-held tools and electrically-powered machines.

When corresponding with personnel, use the terminology provided in this specification to facilitate your inquiries for information. Basic terms and features of this product are provided in Figure 1.

- Expected temperature at heaters is in the range of 700°F
- Uninsulated, high temperature, Strato-Therm crimp splices from Tyco are rated to 1200°F

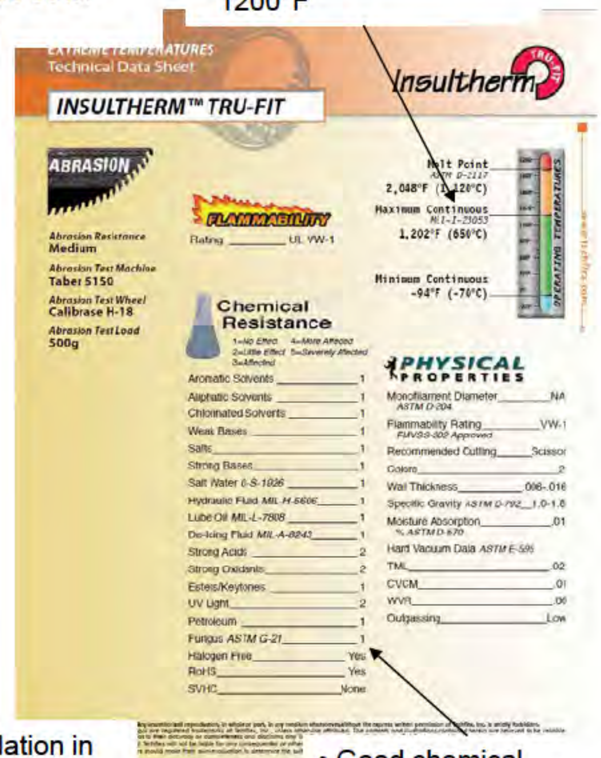


Insulation Candidate

- 464° rating is where acrylic outer coating starts to burn off

- Used as primary insulation in various heated products
- Acrylic binder prevents fraying during assy

- Sleeving can be used at continuous temps up to 1200°F



- Good chemical resistance

Use fiberglass lacing tape to secure insulation

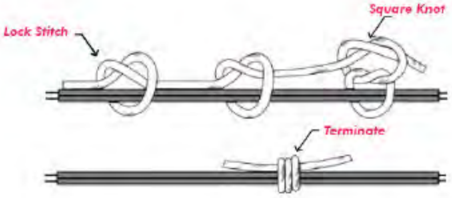
- Fiberglass lacing tape is rated to 800°F

Braided Fiberglass Lacing Tape

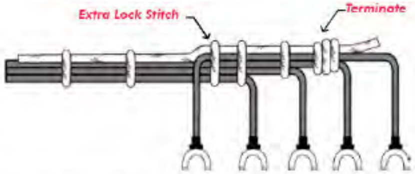
A-A-52083 TYPE IV

- Extremely high temperature performance
- Very low elongation
- Minimal fiber to fiber abrasion
- Produced from continuous filament electrical grade glass (E-glass)
- Available colors: Natural (NT)
- Thermal endurance: -67°F to 800°F; Melting point: 2,102°F
- Finishes: D-TFE-Fluorocarbon

Part #	Mil Spec Size	Mil Spec Finish	Width Min	Width Max	Thickness Min	Thickness Max	Break Strength	Bulk Spool	Lbs/ Spool
LT4-S1-FD-NT	1	D	.203"	.248"	.013"	.019"	200lbs	750'	1.40
LT4-S2-FD-NT	2	D	.099"	.121"	.013"	.019"	100lbs	750'	1.10
LT4-S3-FD-NT	3	D	.077"	.094"	.013"	.019"	75lbs	1,500'	1.70
LT4-S4-FD-NT	4	D	.054"	.066"	.013"	.019"	50lbs	1,500'	1.20



When you are at the end of the run, simply make another square knot to secure the lacing. Also you can put a dab of glue on the knot, which ensures the lacing will not come undone.



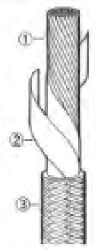
Once you master your lacing, you can try your skill at breaking out branches of the wiring harness along the lacing.

An alternate method is to make discrete "ties" where a start type knot is made every 1/2", and cut off so that each knot is independent. With this method, there is no chance of the lacing unraveling, but it is much more labor intensive.

www.LacingTape.com

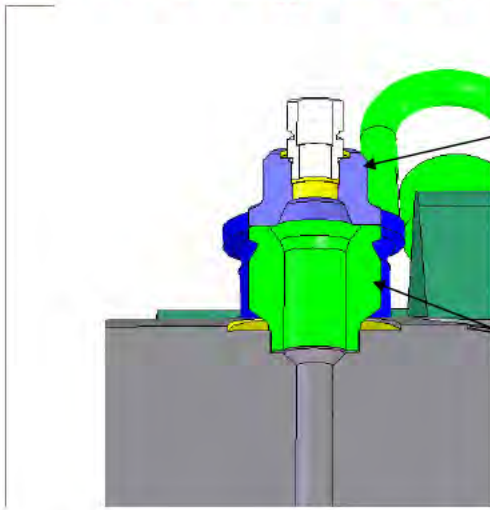
Wire

- Plan to use MGT wire rated for use at 450°C (842°F)
- Size based on deratings for temperature (X .34) and number of conductors in bundle (~X .5, to be reviewed)
- 4680W at 240 Vac requires 19.5A total at room temp
- With knock-down factors, need to size supply for 19.5A / .34 / .5 = ~115A
    - 6 circuits is ~ 19A each
    - In the range of 6-8 AWG supply wires
    - Needs further review, and evaluation of power as a function of temperature



AWG	Base Temp Rating (40°C / 104°F)	Approx. current rating at 618-662°F (X.49)	Approx. current rating at 708-752°F (X.34)	Conductor Diameter (in)
22	5.6A	2.7 A	1.9 A	Ø.025
20	8 A	3.9 A	2.7 A	Ø.032
18	11 A	5.4 A	3.7 A	Ø.040
16	14 A	6.9 A	4.8 A	Ø.051
14	21 A	10.3 A	7.1 A	Ø.064
12	26 A	12.7 A	8.8 A	Ø.081
10	35 A	17.2 A	11.9 A	Ø.102
8	49 A	24.0 A	16.7 A	Ø.129
6	65 A	31.9 A	22.1 A	Ø.162
4	76 A	37.2 A	25.8 A	Ø.204
3	85 A	41.7 A	28.9 A	Ø.229
2	99 A	48.5 A	33.7 A	Ø.258
1	110 A	53.9 A	37.4 A	Ø.289
1/0	126 A	61.7 A	42.8 A	Ø.325
2/0	141 A	69.1 A	47.9 A	Ø.365
3/0	159 A	77.9 A	54.1 A	Ø.410

Wire Egress, 3 Places



- Out Conax compression fitting mounted in Grayloc blind
- Blind can be easily replaced, or revised if damaged or design changes

- Route through Grayloc hub
- Provides some thermal isolation

MATERIAL REVIEW AND RADIAL GAP PREDICTIONS

Discussion

- Reviewed a wide range of materials and material families to identify candidate seal materials based on
  - Ability to achieve target seal gaps
  - Hardness compared to titanium
  - Strength at operating temperature
  - Wear properties
- Reviewed inputs from
  - CDM 500-5-1, Design Guidelines
  - CDM 500-6-4, General Labyrinth Design – Toothed
- Solicited input from Byron Mohr was to consider Ni-Resist
  - Recommended ni-resist

Performed Radial Gap Analysis for Each Candidate Material

- Started with assumption that the target radial gap in the operating conditions would be .009" on the upstream end of the seal, and .006" on the downstream end of the seal
- Used [redacted] operating diameters provided by Dave Taylor
- Correlated analytical model to FEA predictions from Paul Brown and Brian Massey
- Backed out the radial clearance (or interference) for each material at static, room temperature conditions necessary to achieve the target operating gaps of .009 and .006

Spreadsheet Inputs

	Fixed End	Free End	Comments
Axial Position (in):			
Cold [redacted] Nominal Diameter (in):			
Room Temperature (°F):	70	70	
Minimum Acceptable Room Temp Gap (in):			
From PRR [redacted] Analysis:			
Radia [redacted] Displacement (in):			From [redacted] PRR dated 8/29/2014; combined effect of thermal, pressure, and rotational loads
Resultant [redacted] OD in operation (in):			
[redacted] temperature (°F):	772	772	Approximated from plot
Target SS radial gap:	0.0090	0.0060	
Target hole damper ID at SS:			
From Seal Prelim FEA			
radial displacement (in):			3000W steady-state, 410 SS
temperature (°F):	580	773	Approximated from plot
410 SS Evaluation - Compared to FEA			
temperature (°F):	580	773	Approximated from plot
Material CTE:	6.1000E-06	6.1000E-06	At 600°F
Unrestrained Diametrical thermal growth:	0.0345	0.0475	
Unrestrained Radial thermal growth:	0.0172	0.0238	
Prelim FEA % of unrestrained thermal growth:	87.0%	88.4%	Seems reasonable given restricted growth from cooler material at larger diameters

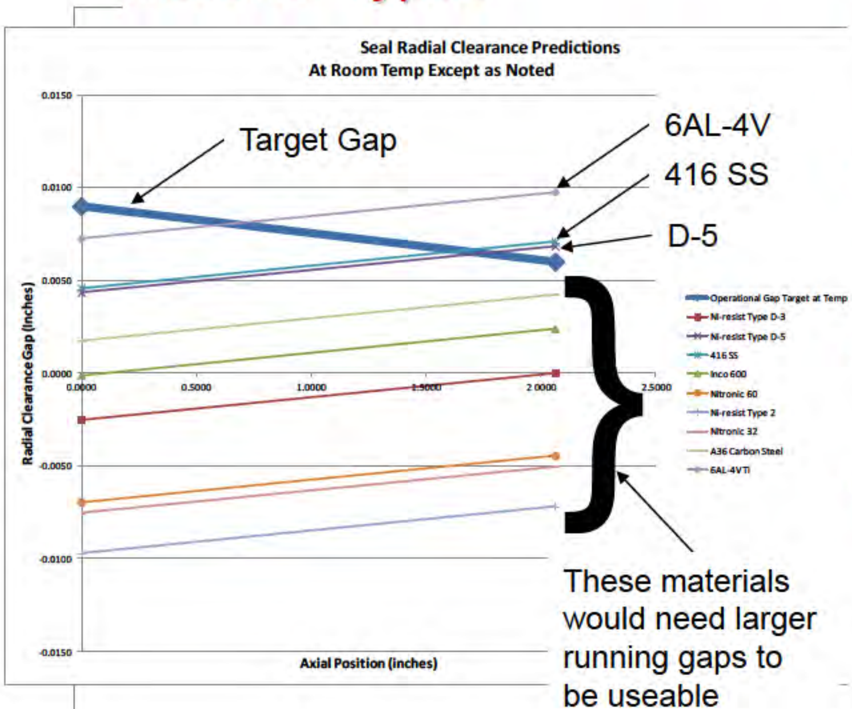
Spreadsheet Output – Ni-Resist Type D-5

	Fixed End	Free End	Comments
Ni-resist Type D-5			
SS temperature (°F):	772	772	Assume temperatures similar to the [redacted]
Material CTE (in/in/°F):	6.3000E-06	6.3000E-06	2.8 ppm/°F at 200°F; CTE is 6.3 ppm/°F at 773°F
Unrestrained diametrical thermal growth at SS (in):			
Diametrical Growth Restrained Adjustment Factor:			Assume 88% of theoretical to account for restricted growth from cooler material at larger diameters
Restrained diametrical growth at SS (in):			
Restrained Radial thermal growth at SS(in):			
design point ID at room temp (in):			
Room temp diametrical seal gap (in):			
Room temp radial seal gap (in):			Comfortable start up gaps, even without pre-heat;
Start-up Step 1:			
SS temperature (°F):	580	773	Assuming 3000W steady-state start-up condition, pre-heated before start of test
Unrestrained diametrical thermal growth at SS (in):			
Restrained diametrical growth at SS (in):			
Restrained Radial thermal growth at SS(in):			
Radial seal gap at start of rotation (in):			

Spreadsheet Output – 416 SS

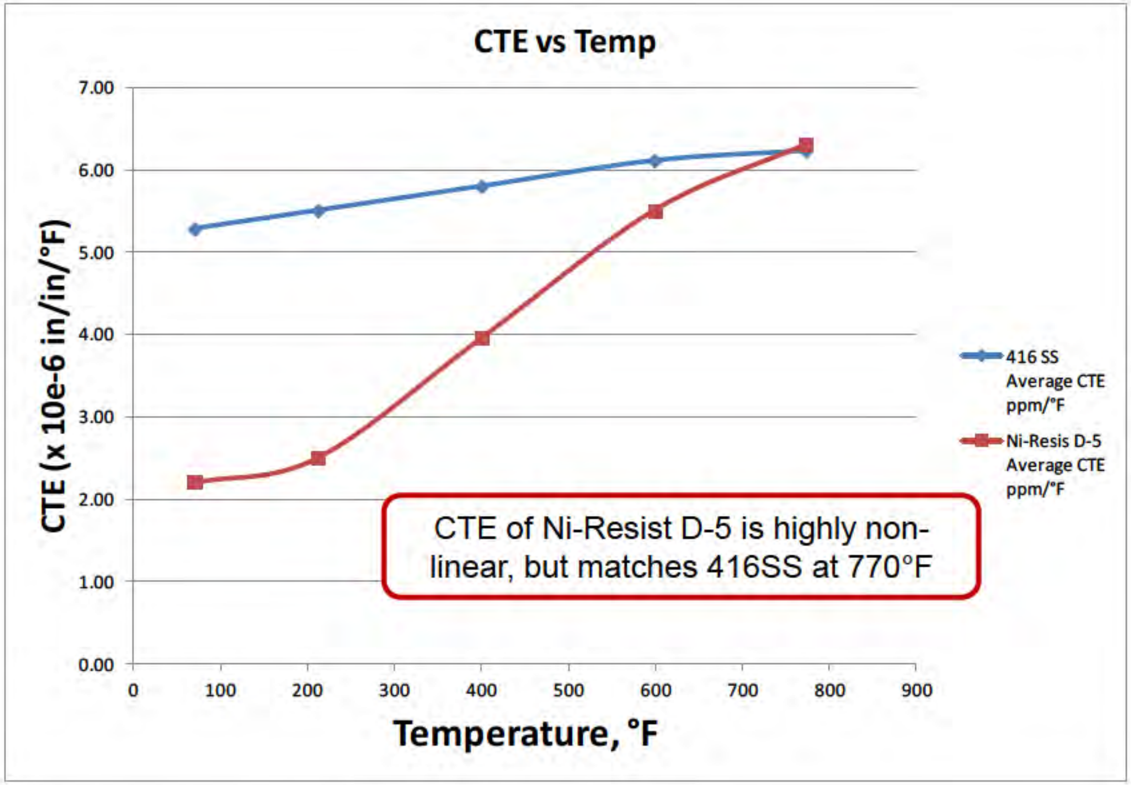
	Fixed End	Free End	Comments
416 SS			
SS temperature (°F):	772	772	Assume temperatures similar to the [redacted]
Material CTE (in/in/°F):	6.2300E-06	6.2300E-06	Est. CTE at 773°F
Unrestrained diametrical thermal growth at SS (in):			
Diametrical Growth Restrained Adjustment Factor:	0.8800	0.8800	Assume 88% of theoretical to account for restricted growth from cooler material at larger diameters
Restrained diametrical growth at SS (in):			
Restrained Radial thermal growth at SS(in):			
design point ID at room temp (in):			
Room temp diametrical seal gap (in):			
Room temp radial seal gap (in):			Tolerable start-up gap; would like to increase slightly
Start-up Step 1:			
SS temperature (°F):	580	773	Assuming 3000W steady-state start-up condition, pre-heated before start of test
Unrestrained diametrical thermal growth at SS (in):			
Restrained diametrical growth at SS (in):			
Restrained Radial thermal growth at SS(in):			
Radial seal gap at start of rotation (in):			

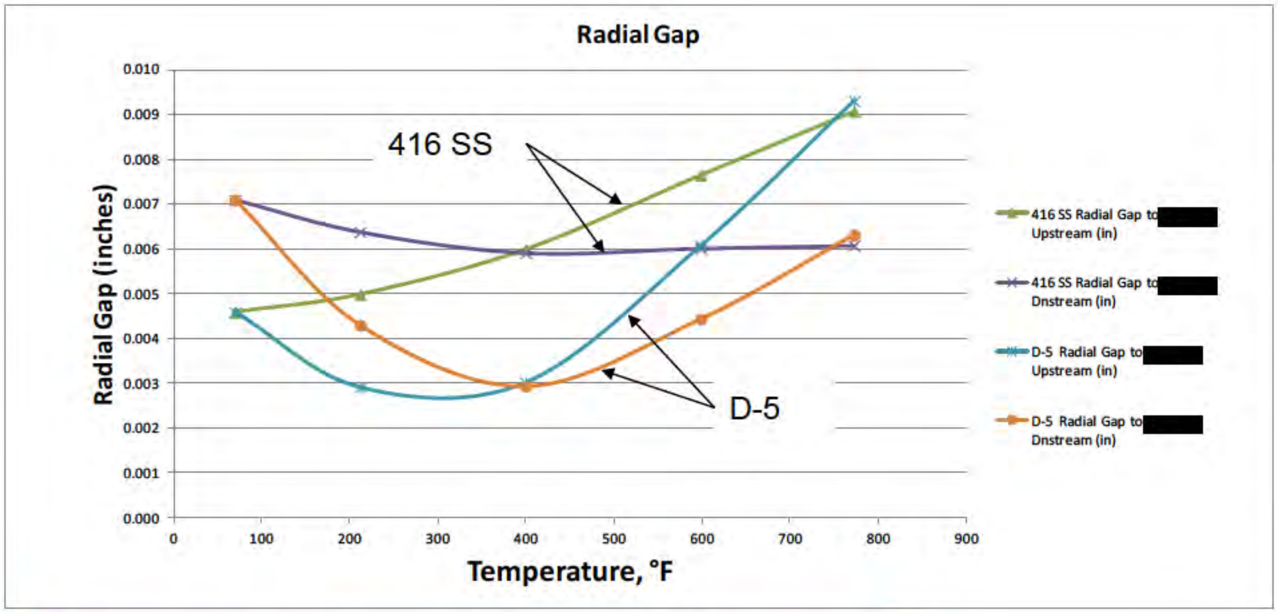
Room Temp Gap Requirements for Different Material Types



- To achieve .009" &.006" radial running gaps, 416 SS needs room temperature assy gaps of .0046" & .0071", respectively
- To achieve .009" &.006" radial running gaps, Type D-5 ni-resist needs room temperature assy gaps of .0043" & .0068", respectively
- Titanium, shown for reference, would need room temperature assy gaps of .0072" & .0097", respectively
- Ni-resists Types 2 & D-3, Inco 600, Nitronics 32 & 60, and A36 carbon steel, and similar product families, cannot simultaneously achieve the target running clearances and free-running cold clearances
  - Larger running gaps would be required (i.e. shift all the curves up)

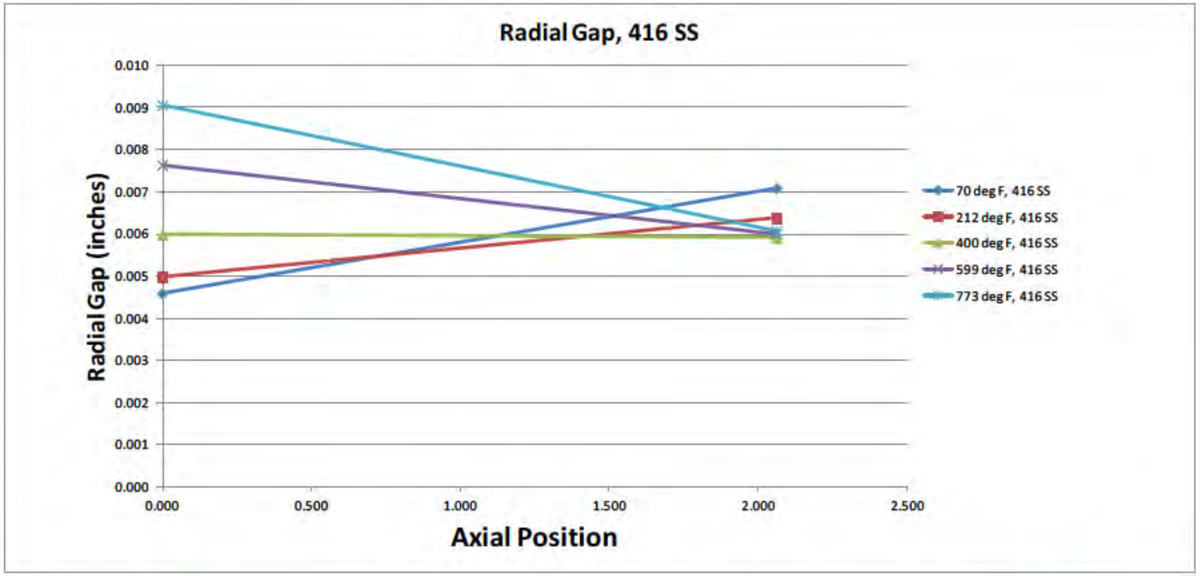
CTE Comparison of 416 SS and Ni-Resist D-5





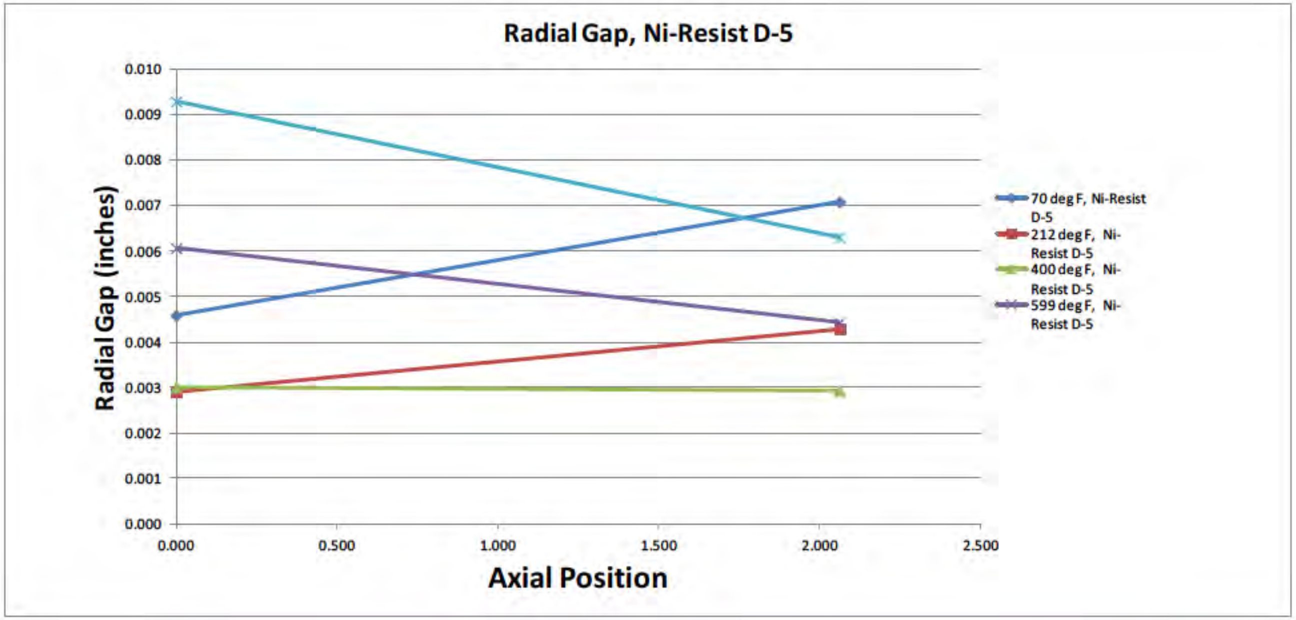
Assuming linear growth of the [redacted] as a function of temperature, 416 SS maintains gaps larger than 4.6 mils, while ni-resist D-5 gaps squeeze to less than 3 mils without supplemental heating

Radial Seal Gap for 416 SS vs Axial Position



Minimum gap with 416 SS is at start-up conditions

Radial Seal Gap for Ni-Resist D-5 SS vs Axial Position



Minimum gap with Ni-Resist D-5 is near 400°F

Material Conclusions

- The only metals evaluated able to achieve the target running gaps of .009" upstream and .006" downstream in steady-state operating conditions (without active cooling), and free-running assembly in the cold condition, are titanium (6AL-4V), 400-series stainless steel (416 SS), and ni-resist Type 5 or D-5
- Of these, the most suitable materials for the target operating range for an actively heated 416 SS and Ni-Resist Type D-5
- 416 SS will behave linearly with respect to temperature; Ni-resist Type D-5 will behave non-linearly; ni-resist has much better wear properties
- Both materials show promise of being able to be used as a seal without supplemental heating, depending on the non-linear ramp-up characteristics of the [redacted]
- Other ni-resists types, Inco 600, Nitronics, and A36 carbon steel, and similar product families, cannot achieve the target operational gap with free running assembly
- Ni-resist Type D-5 is the preferred material, with 416SS a close second depending on availability, cost, lead-time, and the non-linear characteristics of the [redacted]
- Other materials could be used if running gaps are increased by .003" to .016"

INSTRUMENTATION DETAILS

Radial Proximity Probe Candidate

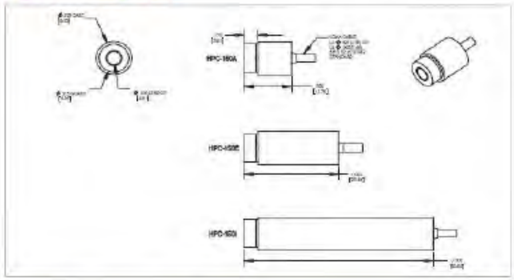
- Capacitor-based probes are the only probe technology able to handle the temperatures
- The HPT-75 from Capacitec is the smallest available probe
- Requires electrically grounded target for reliable reading (i.e. grounded shaft)

Axial Proximity Probe Candidate

HPC-150

Standard Temperature Probes

Temperature Ranges A & E



Linear Range		0.100" [2.54]
Maximum Range		0.150" [3.81]
Material (Case, Guard and Sensor)		303 stainless steel
Standard Probe Lengths	A	0.500" [12.70]
	E	1.000" [25.40]
	I	2.000" [50.80]
	Custom	Custom lengths [illegible] available B .625" [15.88], C .750" [19.05], D .875" [22.23], F 1.250" [31.75], G 1.500" [38.10], H 1.750" [44.45], please consult factory
Temperature Ranges	A	32°F (0°C) to 302°F (150°C), please consult factory for cryogenic
(Continuous Use)	E	-100°F (-73°C) to 400°F (205°C), please consult factory for cryogenic
Sensor OD		0.150" [3.81]
Case OD		0.375" [9.53]
Connector Options		Microdot, BNC or MCX (microdot and MCX not compatible with L2 cable)
Cable Options OD		L2 0.090" [2.26], L3 0.063" [1.60] or L7 0.125" [3.18], all cable must not be cut or spliced
Dip Coating		Optional protective epoxy dip coating available
Calibration	NIST Traceable	10 point archived calibration record of nominal displacement versus voltage standard Several sensitivities available
Mounting Hole		0.375" +0.001/-0.000, [9.53 +0.05/-0.000]

 HPC-150REVC www.capacitec.com

- HPC-150 probe has linear range of .100" (more clearance to [redacted]
- .375" OD

Work To Do

- Tolerance analysis (axial and radial, including coupling)
- Lay out electrical and instrumentation routing details
- Detail Conax fitting requirements
- Evaluate heat power vs temperature, and sizes supply wires accordingly
- Update 2D thermal and structural analysis based on material selections
- Perform 3D thermal and structural analysis
- Drafting drawings of major parts to supply chain for quote/lead time
- Complete and release drawings

SUPPORTING DATA

Heater Information

- Ø.375 x 1.50 cartridge heater
  - 240 Vac, single phase
  - 130 Watts at 70°F per heater
  - 36 heaters per assy
  - 4680 Watts total
- Available sources:
  - Watlow
    - P/N G1J-15313 (15 days at $23 per heater per quote, custom part)
  - Chromalox
    - CIR-20152/240 (160979) (15 days at $43 per heater on-line price, standard part)
  - Omega
    - CIR-20152/240V (private labeled from Chromalox) (5 weeks at $47 per heater on-line price, standard part)
- *Need to circle back to detailed wire routing, and characterize heater power as a function of temperature – Action for Rick*

UPDATE

Chromalox Specs & Design Input

Chromalox
PRECISION HEAT AND CONTROL

Search:

Product Catalog | Custom Solutions | Industries We Serve | Capabilities & Expertise | Case Studies | Res

Home > Product Catalog > Component Technologies > Cartridge Heaters > CIR

CIR

1/4 - 3/4" Dia. Cartridge Heater

- 1/4, 3/8, 1/2, 5/8, and 3/4" Dia.
- 50 - 5,000 Watts
- 120 and 240 Volt
- 1-1/4 - 48" Sheath Lengths
- INCOLOY® Sheath
- Third-party certifications: UL Recognized, CSA

Next | Zoom

Features

Leads can be bent at right angle near the heater without exposing bare wire, therefore eliminating electrical shorts.

Sheath Material — Type CIR cartridge heaters are made with a high-temperature INCOLOY® sheath material.

High Temperature Leadwire — Up to 842°F (450°C)

Lead Length — Type CIR cartridge heaters are stocked with 14" long leads. Longer lead lengths can be readily spliced on.

Description

Chromalox CIR industrial heaters are available in a variety of commercial heating applications: dies and molds, packing machinery, laminating/adhesives, hot glue melting, lead melting, medical extruding dies, and stamping and marking machines. Type CIR includes several significant advances in cartridge heater technology. Its high performance characteristics have been proven not only in the laboratory, but also on customers' equipment in selected problem applications, at 1500°F and higher operating temperatures.

Resources

- CIR Catalog Page
- Cartridge Heaters Design Guide
- CIR Installation Manual
- Cartridge Heaters Specification Data Sheet

Cartridge Heaters
Application Guidelines

- Up to 1.297" Dia.
- Up to 60" Lengths
- Up to 11,500 Watts
- 120 and 240 Volt
- Up to 1400°F Max. Working Temp.
- Modification Available to Fit Custom Applications

Type CIR cartridge heaters are most frequently used for heating metal parts by insertion into drilled holes. For easy installation, the heaters are made slightly undersize relative to their nominal diameter.

Determining Fit — At high watt densities, a close fit is important. The fit is the difference between the minimum diameter of the heater and the maximum diameter of the hole. For example, 1/2" diameter Type CIR cartridge heater is actually 0.498" plus 0.000" minus 0.005". If this heater is placed in a hole which has been drilled and reamed to a diameter of 0.503", then the fit would be 0.01" (0.503" - 0.493" = 0.01").

A close fit is more important for high-watt density applications.

Determining Watt Density — Watt density refers to the heat flow rate or surface loading. It is the number of watts per square inch of heated surface area. For calculation purposes, CIR stock cartridge heaters have 1/4" unheated length at each end. Thus, for a 1/2 x 12" heater rated 1,000 watts, the watt density calculation would be as follows:

$$\text{Watt density} = \frac{W}{\pi \times D \times HL}$$

Where:
W = wattage = 1,000 W
D = diameter = 0.5 in.
HL = heated length = 11.5 in.

$$\text{Watt density} = \frac{1{,}000}{3.14 \times 0.5 \times 11.5} = 55\ W/in^2$$

Selecting Sizes and Ratings — The calculation of total heat requirements for an application is outlined in the Technical section of this catalog.

Determining Quantity, Size and Rating — Once total heat requirements are established, the quantity, size and rating of cartridge heaters can be decided. Plan for enough heaters to permit even temperatures through the part during heat-up and operation. The sensor for the temperature control should be placed close to the working surface for accurate control.

Calculate Watt Density and Fit — After the wattage for each heater has been established, the watt density and fit must be calculated. Then, use Graph G-235 to be sure that the watt density is within allowable limits. For example, a 1/2 x 12" CIR heater rated 1000 watts has a watt density of 55 W/in². If it were used in a part with an operating temperature of 1000°F with a fit of 0.01", the allowable watt density from the graph would be 90 W/in². Thus, the actual watt density of 55 W/in² is well below the maximum allowed. A substantial safety margin would exist and high reliability can be expected.

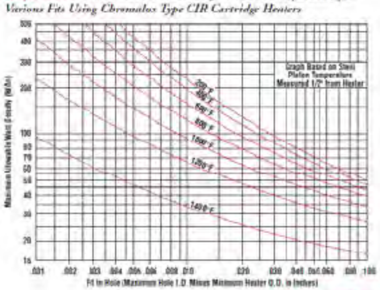
If the heater selected had a watt density higher than that allowed by the graph, consider the following changes:

1. Using more heaters of lower watt density.
2. Using longer or larger diameter heaters.
3. Improving the fit.
4. Reducing heat requirements by reducing heat losses or by allowing for longer heat-up time.

Using the Maximum Allowable Watt Density Graph — This graph is useful for choosing Type CIR cartridge heaters. The curves should be considered as guides and not precise limits.

The graph is based on a 1600°F resistance wire temperature inside the cartridge heater, when the heater is installed in an oxidized mild steel block. Watt density values from the graph should be lowered by about 10% or more when other materials are used which have a lower thermal conductivity or lower emissivity than oxidized mild steel. Contact your Local Chromalox Sales office.

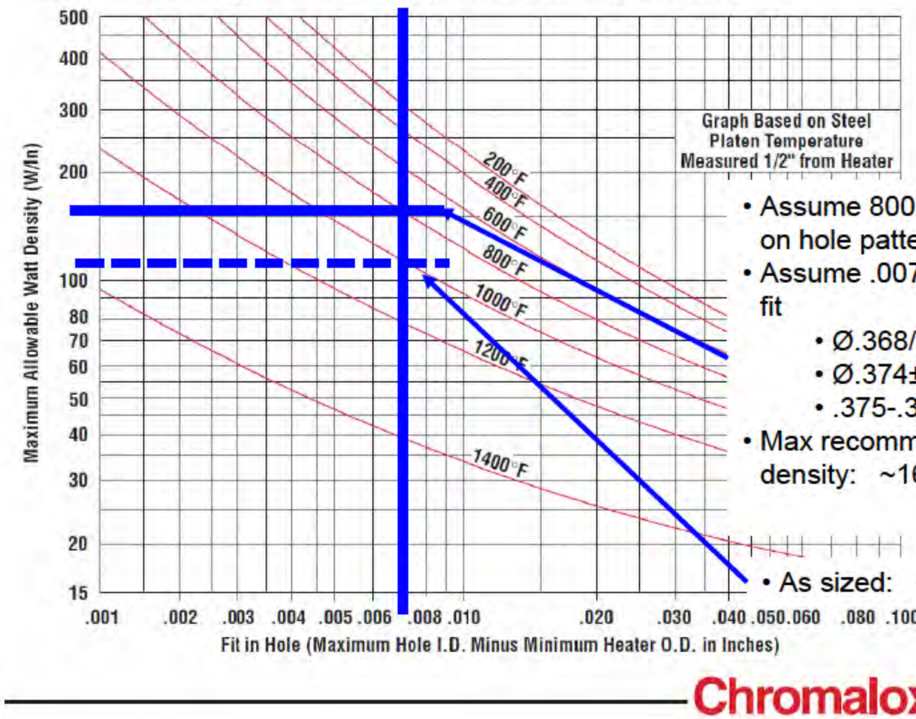
Graph G-235 — Maximum Watt Density vs. Platen Temperature for Various Fits Using Chromalox Type CIR Cartridge Heaters



A-92 Chromalox®

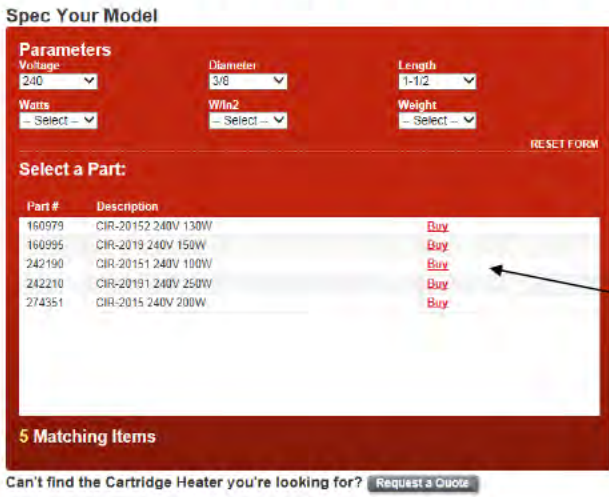
Heater Sizing

Graph G-235 — Maximum Watt Density vs. Platen Temperature for Various Fits Using Chromalox Type CIR Cartridge Heaters



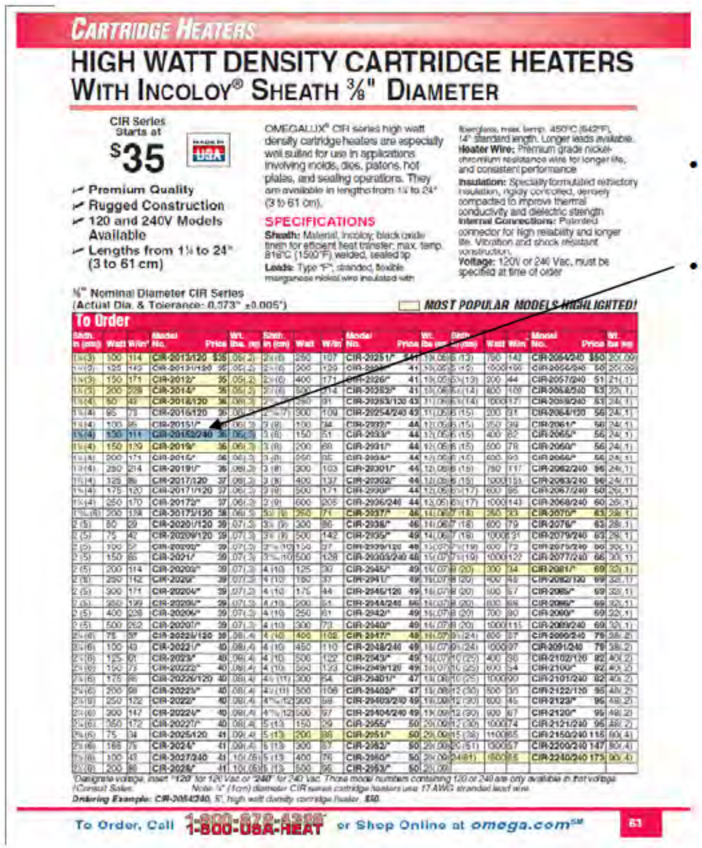
- Assume 800°F max metal temp on hole pattern seal
- Assume .007 max diametric hole fit
  - Ø.368/.373 heater diameter
  - Ø.374±.001 hole
  - .375-.368 = .007 fit
- Max recommended power density: ~160 W/in²
- As sized: ~111 W/in²

Chromalox Heaters in Ø.375 x 1.50



- Available power levels
  - 100W
  - 130W
  - 150W
  - 200W
  - 250W
- Select 130 W
  - Gives 4680 watts total power in 36 heater configuration
  - Could step up to higher power during testing if necessary

Omegalux Heaters in Ø.375



- Same CIR-20152/240V, 130W heater available direct from Chromalox
- Power density is 111 W/in^2
  - Comfortably below recommended 160 W/in^2 maximum

Watlow Quote

- Had requested quote for 125 W heater
- Revise to 130 W
    - Should have minimal effect on price

Quotation # 247280 Rev: 0
Please reference quote and revision when ordering

Quote Date: 12-Sep-2014 **Account:** 070626
Expiration: 09-Dec-2014

WATLOW
Order Email: orderentry@watlow.com
Tel: 1-800-WATLOW2

Dresser Rand - Bellevue
Attn. Rick Wiedener
11808 Northup Way
Suite W 190
Bellevue, WA 98005
4258959205
bjilka@dresser-rand.com

Sales Office Contact
Eagley, Jeff
Watlow Seattle
425-222-4090

Item #	Plant	Part Number / Description Line Comments	Qty	UOM	Price/Unit USD	Ext. Price USD	Estimated Lead Time Business Days
1	STL	Part#: E1J-12897	36.00	EA	Net 24.412	878.83	15
Description:		Firerod 1/4"					
Comments:		6 pc minimum order					
Config Details:		Diameter .246" +/- .002" ACTUAL DIA. Length 1.500 Volts 240 Watts 125.0 Lead End No Heat 0 Termination Style......*: S = Swaged in Leads Lead Type.............*: F = Fiberglass - Standard Leads Lead Length............ 18 Sleeving N = No Selection Ground Lead Length...... 0 Mounting Option........*: No Selection Potted Cavity ....*: N = No Selection Sheath Material........*: A = Incoloy Max Length (Y/N)......... No					
2	STL	Part#: G1J-15313	36.00	EA	Net 23.287	838.33	13
Description:		Firerod 3/8"					
Comments:		5 pc minimum order					
Config Details:		Diameter .371" +/- .002" ACTUAL DIA. Length 1.500 Volts 240 Watts 125.0 Lead End No Heat 0 Termination Style S = Swaged in Leads Lead Type H = High Temperature leads Lead Length 18 Sleeving N = No Selection Ground Lead length 0 Mounting Option No Selection Potted Cavity N = No Selection Sheath Material A = Incoloy Max Length No					

[redacted]

FDR Updates [redacted]

R. Wiederien

Paul Brown

Date: 10/24/2014

[redacted]

DRESSER-RAND

Safe Harbor Disclosure

The Private Securities Litigation Reform Act of 1995 provides a "safe harbor" for certain forward-looking statements so long as such information is identified as forward-looking and is accompanied by meaningful cautionary statements identifying important factors that could cause actual results to differ materially from those projected in the information.

The use of words such as "may", "might", "will", "should", "expect", "plan", "outlook", "anticipate", "believe", "estimate", "appear", "project", "intend", "future", "potential" or "continue", and other similar expressions are intended to identify forward-looking statements.

All of these forward-looking statements are based on estimates and assumptions by our management that, although we believe to be reasonable, are inherently uncertain. Forward-looking statements involve risks and uncertainties, including, but not limited to, economic, competitive, governmental and technological factors outside of our control, that may cause our business, industry, strategy or actual results to differ materially from the forward-looking statements.

These risks and uncertainties may include those discussed in the Company's most recent filings with the Securities and Exchange Commission, and other factors which may not be known to us. Any forward-looking statement speaks only as of its date. We undertake no obligation to publicly update or revise any forward-looking statement, whether as a result of new information, future events or otherwise, except as required by law.

Agenda

- Action Item Updates
- Instrumentation Update
- Heater Update
- Routing Details
- Thermal Structural Results
- Work to Do
- Supporting Data

ACTION ITEMS

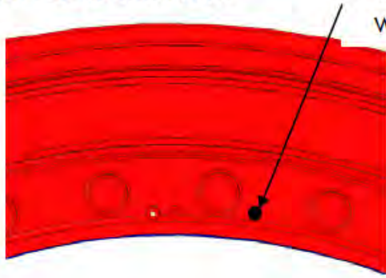
Action Items

Damper Seal

1. **Run rotordynamics model with largest thermal transient gap to verify worst case stability (Badeau/Peer)**
2. **Check bolt material for high temperature properties (Wiederien)**
3. **Evaluate silver solder as electric heating wiring joint (vs. mechanical crimps) (Wiederien)**

Metal Temperature (3 places, ~120° apart)

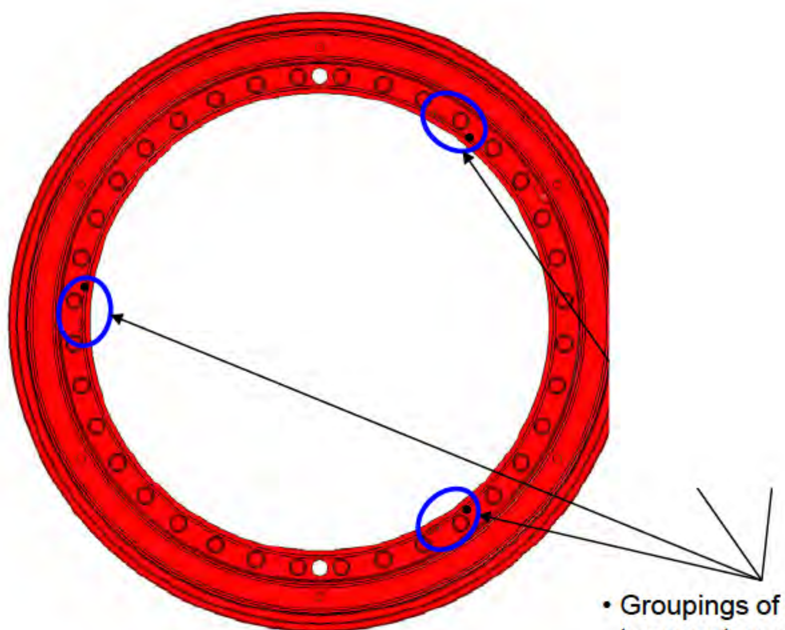
- Ø.062 Thermal couple welded to surface
- Ø.067 through-hole
- Ø.125 counterbore



- Thermal couple welded to surface

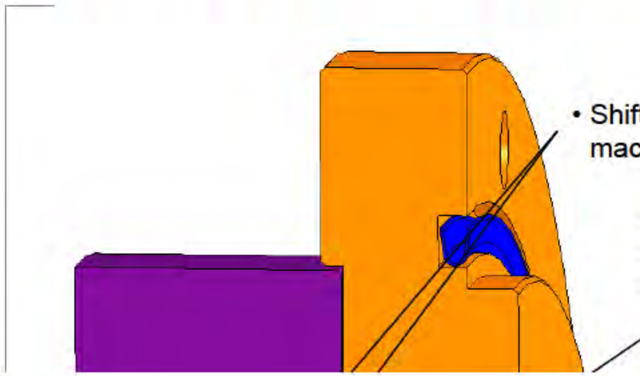
- Thermal couple bottomed out in well and bonded
- Is this one really needed?
    - Viewed as “nice-to-have”, so blind well is acceptable

Metal Temperature



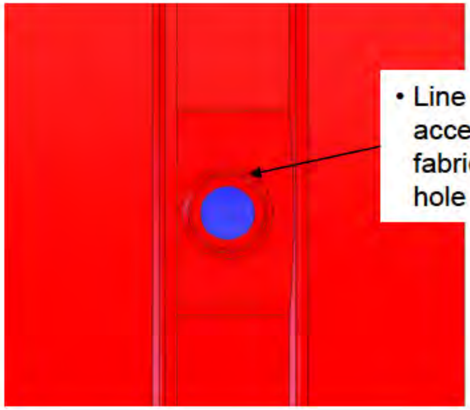
Radial Prox Probe – 7/16-20 UNF Threaded Sleeve

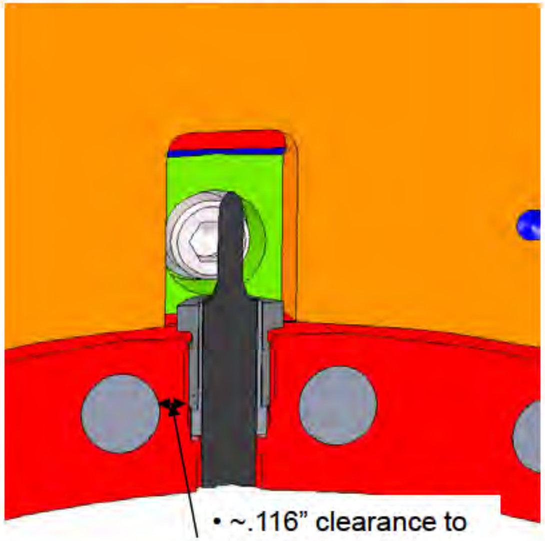
- Detailed probe design



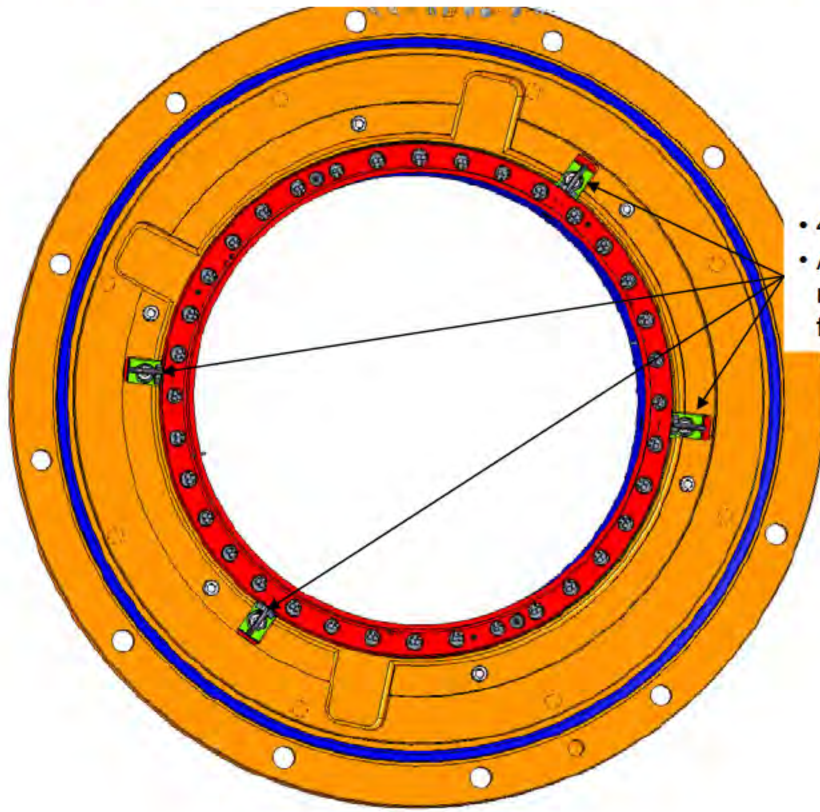
- Shifted seal location .10” axially for machining clearance
- 3 mm triax cable
- R.25 bend radius
- confirmed the bend can be made, once
- Triax cable can touch metal, does not need extra insulation, can be easily sealed to, and protects signal integrity
- Routing through pressure case to be validated with rapid prototype models
- Stretched face of seal ring forward .10” for wall thickness
- 7/16-20 UNF threaded sleeve
- Copper washer
- ¼” diameter probe

Radial Prox Probe – 7/16-20 UNF Threaded Sleeve



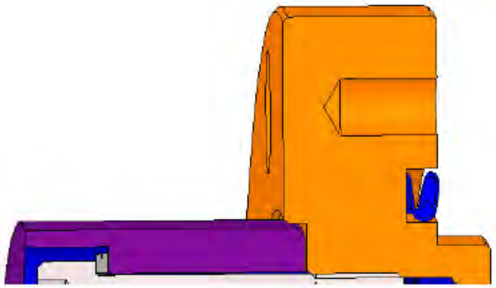


Radial Prox Probe – 7/16-20 UNF Threaded Sleeve



- 4 radial prox probes
- Aligned with ring bolt locations for egress

Axial Prox Probe – 7/16-20 UNF Threaded Sleeve



- Detailed probe design is part of scope
- Need confirmation that the 7/16 thd is acceptable, or diameters need to be adjusted to make more room

- 7/16-20 UNF threaded sleeve

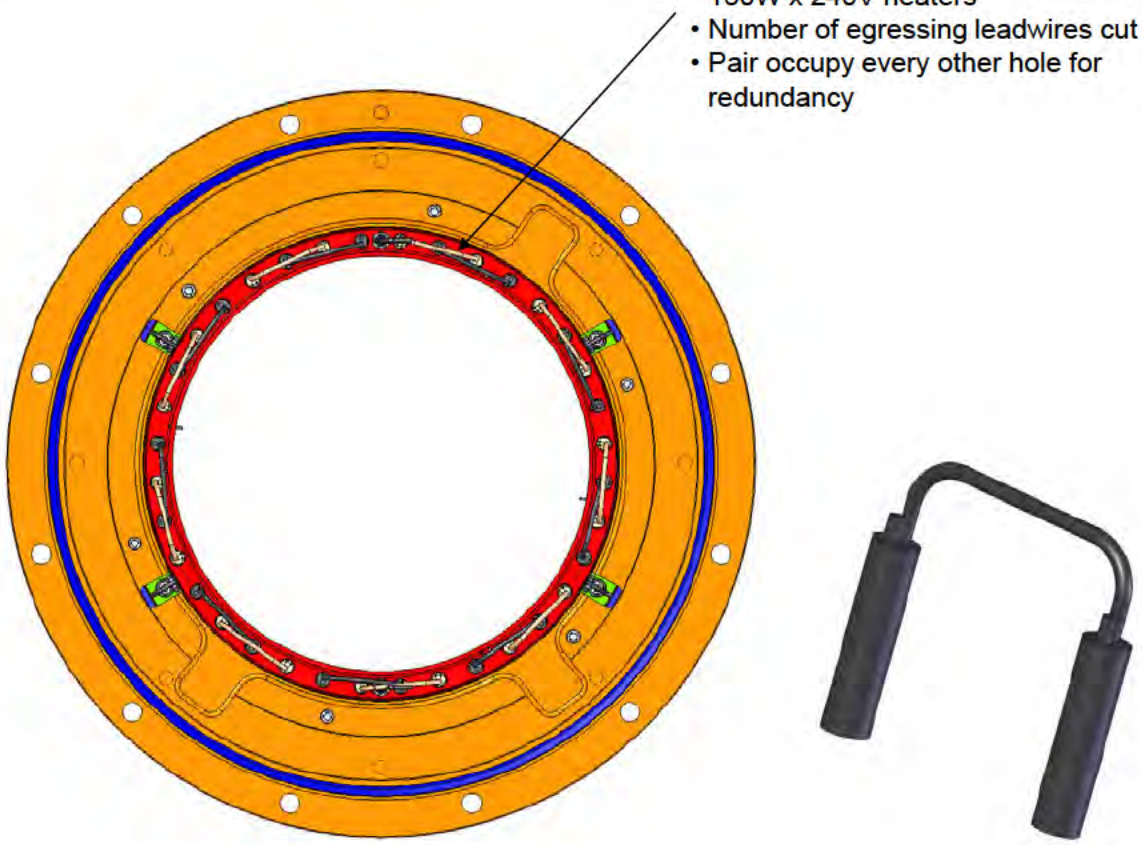
- 3 mm triax cable
- R.25 bend radius
- Egress is recessed within sleeve for more room for ben

- Same geometry as radial probe, except longer

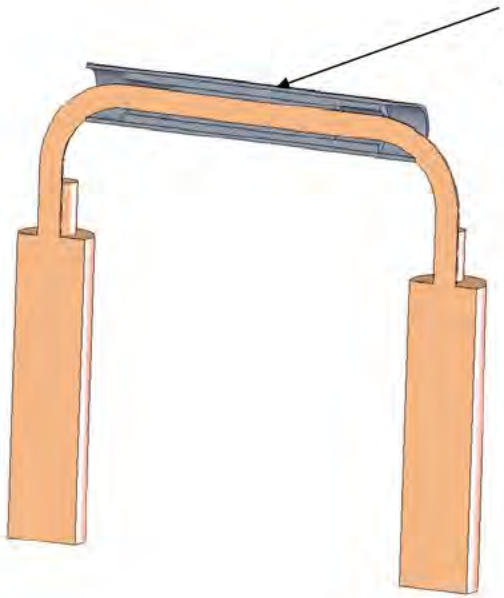
HEATER UPDATES

Heaters

- Reconfigure heaters for 2 x ~130W x 120V heaters in series, instead of individual 130W x 240V heaters
- Number of egressing leadwires cut by half
- Pair occupy every other hole for redundancy



Splice Configuration



- Tyco C-322325 "Stratotherm" splice (or similar)
    - 1200°F Temp Rating
- Nextel ceramic insulation braided sleeve
    - Excellent insulator
- Fiberglas braided outer sleeve
    - For secondary insulation, and protection of the Nextel
- Fiberglass lacing tape spiral wound around bundle to hold it together
- Recommend advanced electrical fab demonstration
    - Alternatives, if there are issues
        - Eliminate half of the heaters and downgrade power capability
        - Match every 3rd heater in a pair

Heaters

Spec Your Model

Parameters

Voltage: 120
Diameter: 3/8
Length: 1-1/2
Watts: -- Select --
W/In2: -- Select --
Weight: -- Select --

RESET FORM

Select a Part:

Part #	Description	
140011	CIR-20141 120V 30W	Buy
242202	CIR-20191 120V 250W	Buy
274335	CIR-2016 120V 85W	Buy
274343	CIR-2015 120V 200W	Buy
274520	CIR-2018 120V 50W	
274538	CIR-2019 120V 150W	
279638	CIR-20151 120V 100W	

7 Matching Items

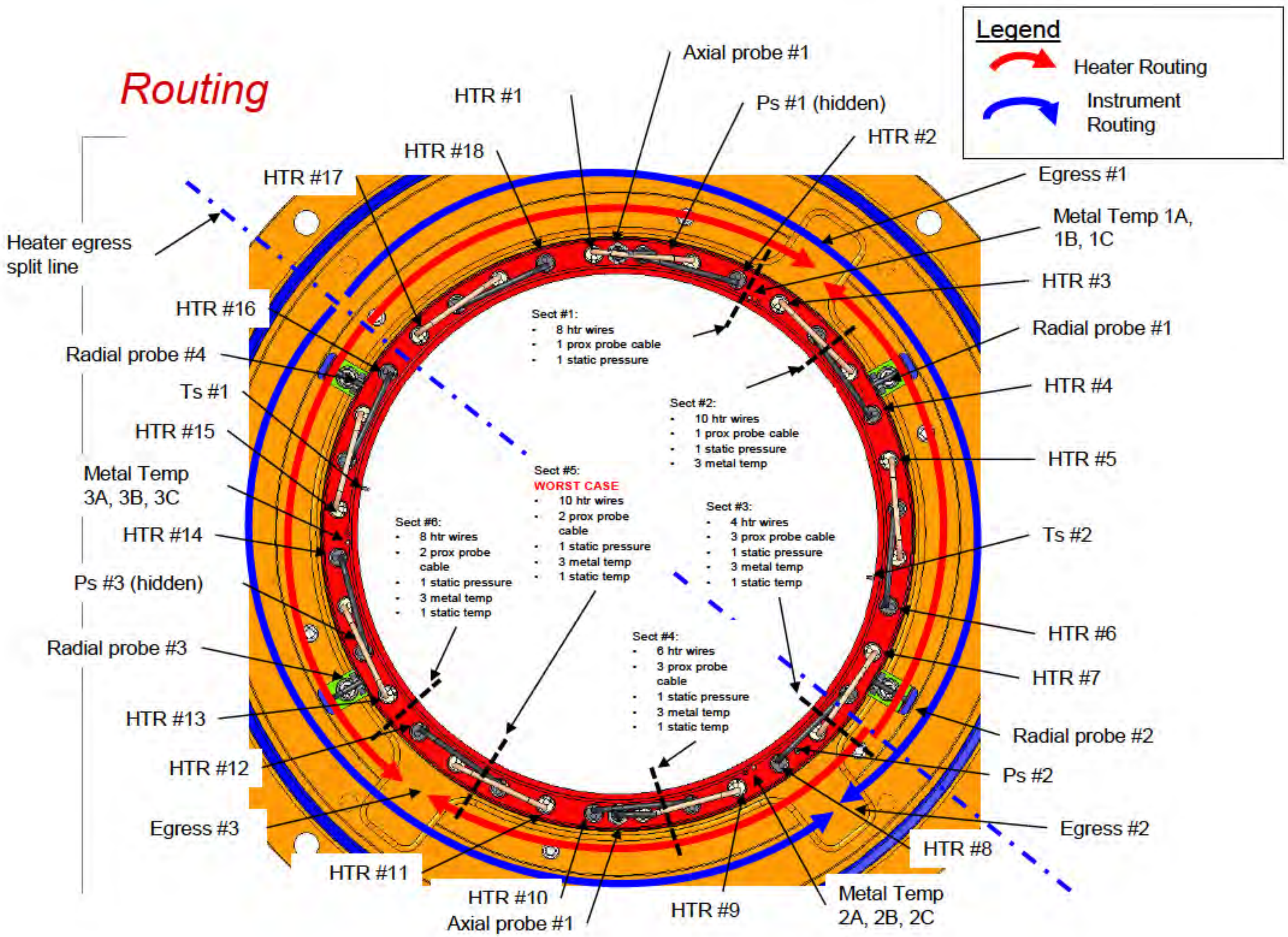
- Available 120V heaters from Chromalox cover range from 30W to 250W off the shelf
- Custom heater required for 130W, with 72” leadwire
  - Lead time ~4 wks per Chromalox
- Leadwire has UL rating of 842°F (450°C), non-UL rating of 1000°F (538°C)
  - 18 AWG wire confirmed in stock by Chromalox

Wire Power Rating Review

- Current per heater circuit at room temperature
  - 250W / 240V = 1.04 A
- Heaters can be supplied with 17 AWG or 18 AWG MGT leadwires
- 18 AWG heater leadwires rated for 11 Amps (ProHeat)
- Derating factor at 842°F (450°C) is ~0.18 (extrapolated)
- Derating factor for 10-20 wire bundle is 0.50 (NFPA 70)
- Current rating for 18 AWG, MGT wire at 842°F in 18 wire bundle is 0.99A
  - Barely under 1.04A full power current
  - Current will decrease natural with temperature
  - Variable voltage control will limit power

- 18 AWG MGT wire has sufficient capability for desired power and temperature

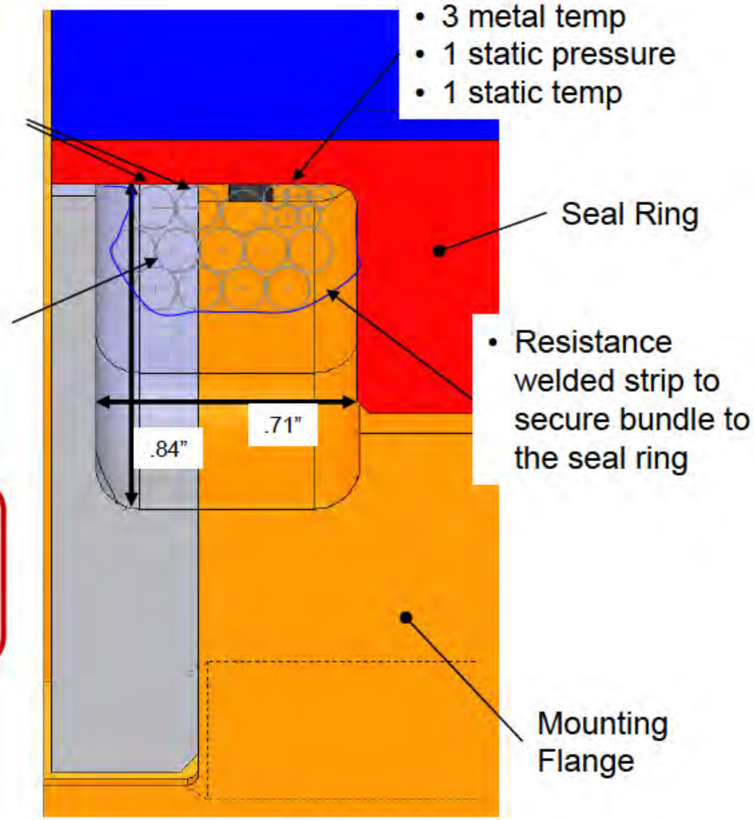
ROUTING DETAILS


Routing
Legend
Heater Routing
Instrument Routing
Heater egress split line
HTR #1
HTR #2
HTR #3
HTR #4
HTR #5
HTR #6
HTR #7
HTR #8
HTR #9
HTR #10
HTR #11
HTR #12
HTR #13
HTR #14
HTR #15
HTR #16
HTR #17
HTR #18
Axial probe #1
Ps #1 (hidden)
Egress #1
Metal Temp 1A, 1B, 1C
Radial probe #1
Ts #2
Radial probe #2
Ps #2
Egress #2
Metal Temp 2A, 2B, 2C
Axial probe #1
Egress #3
Radial probe #3
Ps #3 (hidden)
Metal Temp 3A, 3B, 3C
Ts #1
Radial probe #4
Sect #1:
8 htr wires
1 prox probe cable
1 static pressure
Sect #2:
10 htr wires
1 prox probe cable
1 static pressure
3 metal temp
Sect #3:
4 htr wires
3 prox probe cable
1 static pressure
3 metal temp
1 static temp
Sect #4:
6 htr wires
3 prox probe cable
1 static pressure
3 metal temp
1 static temp
Sect #5:
WORST CASE
10 htr wires
2 prox probe cable
1 static pressure
3 metal temp
1 static temp
Sect #6:
8 htr wires
2 prox probe cable
1 static pressure
3 metal temp
1 static temp

Egress Summary

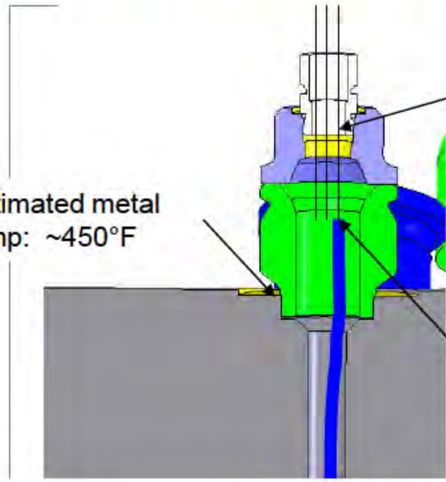
	Egress #1	Egress #2	Egress #3
Heater Circuits	• #17, 18, 1, 2, 3, 4, 5, 6, 7 • Wire Size: 17 AWG, MGT Style, ~Ø.120 (TBR) • Fiberglass braid insulation • Convert to 20 AWG, 250°C leadwire for passage through Conax gland	• 4 radial prox probes (Ø.118) • 2 axial prox probes (Ø.118) • 9 metal temperatures (Ø.062) • 2 static temperatures (Ø.062) • 3 static pressures(Ø.062)	• #8, 9, 10, 11, 12, 13, 14, 15, 16 • Wire Size: 17 AWG, MGT Style, ~Ø.120 (TBR) • Fiberglass braid insulation • Convert to 20 AWG, 250°C leadwire for passage through Conax gland

Cross-Section #5 Simulation



- Bundle will fit in available egress, with room for extra sleeving on heater leadwires and margin for variability

Heater Leadwire Egress (#1 & #3)



- Estimated metal temp: ~450°F

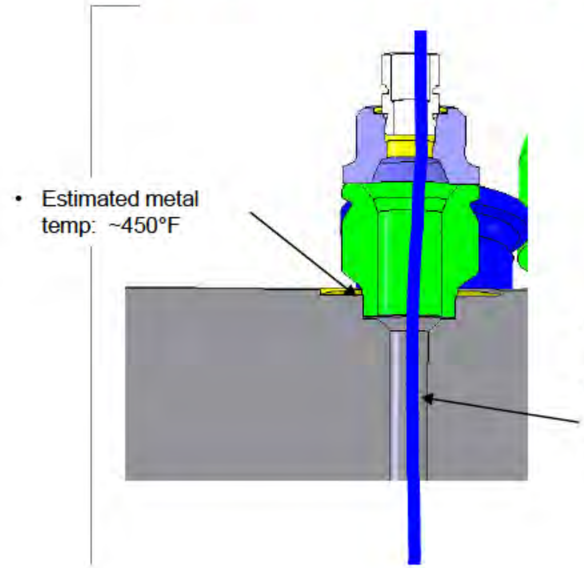
- Pass 18, 18 or 20 AWG, 250°C (482°F) leadwires through Conax

- Gland details needs review with test team for serviceability

- Join with pre-insulated, high temperature splices
- Tyco Strato-Therm or similar, rated for 550°F
- Apply secondary, high temperature shrink sleeving for added strain relief protection (Polytetrafluoroethylene shrink sleeve is rated for 500°F)
- Wire assembly can be completed before the seal assembly is installed

- Bundle of 18, 17 AWG (TBR), MGT-style heater leadwires
- Leadwires have fiberglass braided outer insulation which will be difficult to seal to

Instrument Egress (#2)



- Gland details needs review with test team for serviceability

- 6 x Ø.118 (3 mm) triax cables (tubes) for proximity probes
- 14 x Ø.062 tubes for pressure and temperature measurements
- Combination of 6 x Ø.118 and 14 x Ø.062 confirmed feasible by Conax

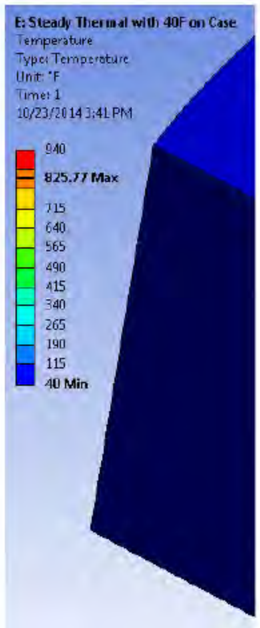
THERMAL STRUCTURAL RESULTS

3D MODEL

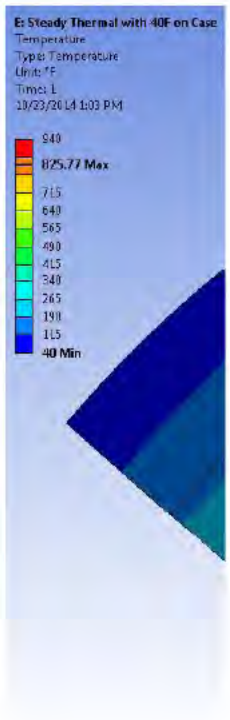
Updated 3D Thermal Model

- Thermal contact reduced from 5000 to 500 BTU/(hr-ft^2-F) between:
  - seal ring
- Case exterior set to 40 °F since the case won't have time to heat up significantly
  - Simulates a worst case scenario with the seal ring hot and the case cold
- Changed material of mounting flange and retaining ring to 17-4 for increased strength
- Heaters set to 65 W each for a total of 2340 W

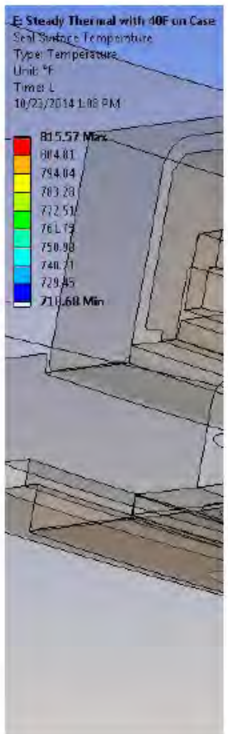
Temperature Overview



17-4 Mounting Flange Temp Map



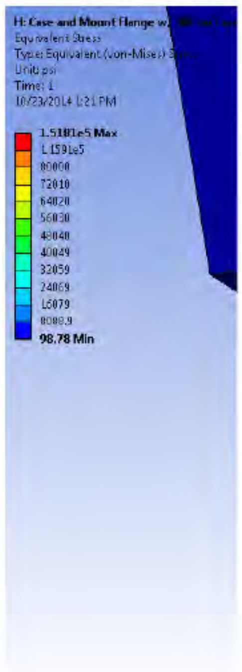
Sealing Surface Temperature Map



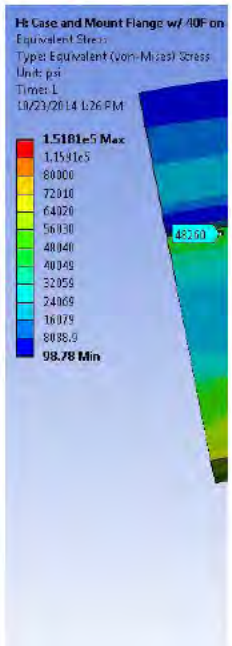
Updated 3D Structural Model

- Mesh refined around teeth to accurately capture peak stress
- Sub-model shows high stresses in mounting flange driven by thermal expansion of the mounting flange.
  - Mounting flange and case can be modeled on their own to increase speed
- BC between flange and case is no-separation BC for increased speed

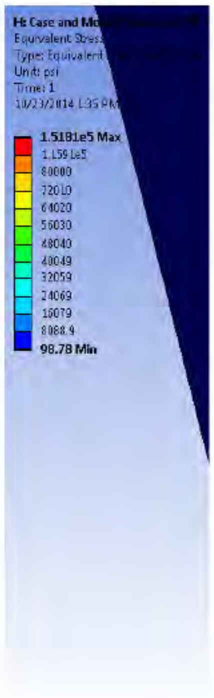
Equivalent Stress – Overview



Equivalent Stress – Mounting Flange



Equivalent Stress - Case



-
  substantial area
- Thermal expansion of mounting flange causing high stress
- Need some compliance in the system

Conclusions

- Two trouble spots remain
  - teeth root stress over 140 ksi
  - Case step over 100 ksi
- Will need to find a way to add radial compliance to the system
  - Can be investigated with 2D model
- Will need to consider alternate teeth profiles to reduce stresses and/or limit preheat temperatures
- Need to run full 3D mechanical model
  - Find displacements on the sealing surface
  - Find stresses in Ring teeth
- Clamping force on retaining ring is 4900 lbf when assembled
  - Linear seal force could be up to 53 lbf/in, corresponding to 2430 lb
  - Clamping ring force is sufficient to compress seal

2D MODEL

Discussion

- Evaluated multiple configurations to establish effects of
  - Thermal stand-off between seal ring
  - Thermal stand-off between mounting flange and pressure case
  - Length of cylindrical engagement
  - 416 SS vs 17-4 SS for retaining ring, mounting flange, and wire cover
  - Pressure load on top of thermal loads
- All results had a strong temperature dependence; heater power was adjusted with each trade to maintain a maximum temperature near 766°F for comparison

Material Properties

Material	410 SS, 1125°F temper	416 SS, 1100°F Temper	17-4 H1025
Component	None-reference	Seal ring; hirth ring	Retaining ring; wire cover
Elastic Modulus (psi)	29e6 psi	29e6 psi	28.5e6 ksi
Poison's Ratio	.28	.28	0.27
Yield Strength at Room Temp	104 ksi	100 ksi	145 ksi
Ult Strength at Room Temp	123 ksi	125 ksi	155 ksi
Yield Str at 800°F	82 ksi	78 ksi*	109 ksi (75%)
Ult Str at 800°F	95 ksi	96 ksi*	121 ksi (78%)
Hardness	241 Brinell (23 HRC)	262 Brinell (26 HRC)	35-43 HRC
CTE (ppm/°F)	5.3e-6 (68-392°F) 6.5e-6 (68-1112°F)	5.7e-6 (68-392°F) 6.7e-6 (68-1200°F)	6.2e-6 (68-392°F) est. 6.9e-6 (68-1200°F) est.
Elongation at Break	17%	17%	12%
Thermal Conductivity	14.4 Btu/(hr-ft-°F)@212°F	14.4 Btu/(hr-ft-°F)@212°F	10 Btu/(hr-ft-°F)@212°F
Notes:	Allegheny Ludlum datasheet; Carpenter datasheet; Aerospace Structural Mat'l Hdbk	Carpenter datasheet; Aerospace Structural Mat'l Hdbk	MIL-HDBK-5; AK Steel datasheet

* Properties of 416 SS at temperature for heat treated condition are estimated with scale factors from 410 SS; use with comfortable safety factors to account for uncertainty

Conclusions

- Adding thermal stand-off between seal ring and ring results in slight power reductions (-4%)
  - Trade-off is slightly higher stresses in ring, which stays cooler
- Adding thermal stand-off between mounting flange and pressure case results in slight power reduction (-5%), and reduced stresses at teeth (-10%) as the temperature gradient in the mounting flange is reduced
  - Trade-off is increased stresses at the pressure case (+13%)

Conclusions

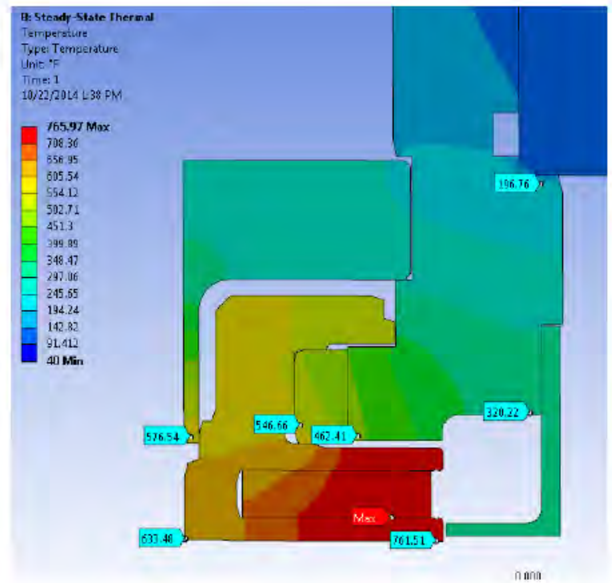
- Reducing the length of the cylindrical engagement between the ring and seal ring reduces the power requirement (-8%). Stress in the mounting flange teeth decrease (-7%).
- Changing the retaining ring from 416 SS to 17-4 SS resulted in small changes in power requirements and stresses. Higher strength of 17-4 will allow greater clamping force and compliance for tolerance variation.

Conclusions

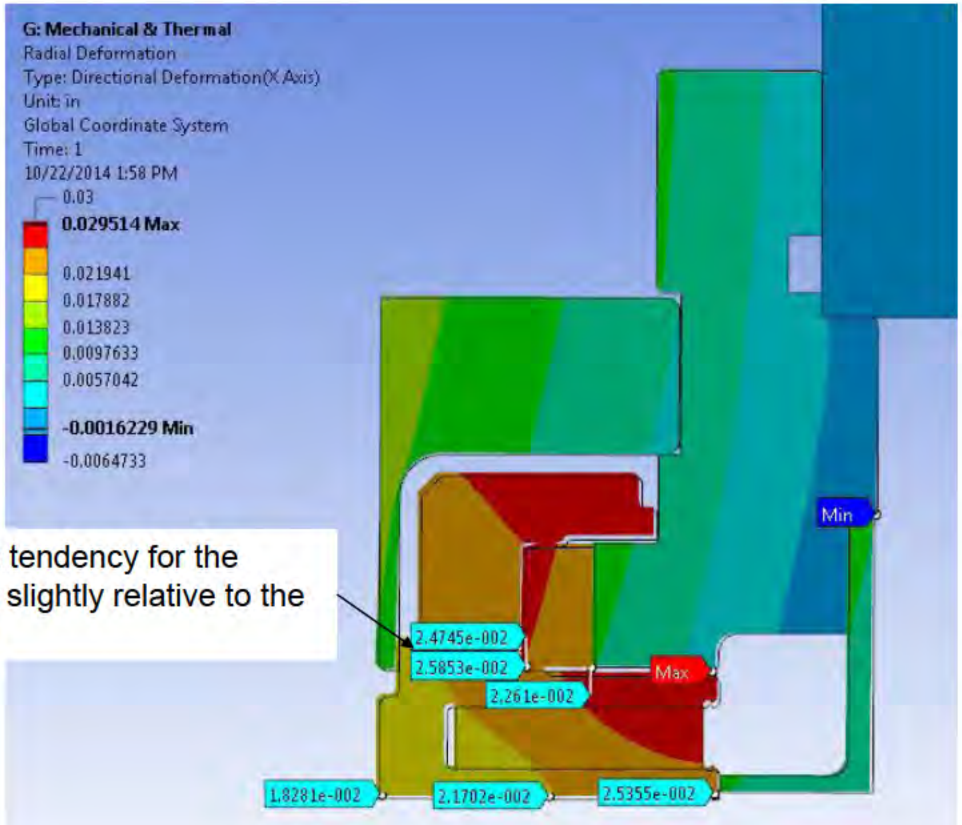
- Changing the mounting flange from 416 SS to 17-4 SS reduces the power requirements (-15%)
  - Increases stress in mounting flange teeth (+13%)
    - Increase is more than offset by substantial increase in yield strength of 17-4 SS
  - Trade of is reduced wear properties
    - Should use some type of lubricant between teeth (anti-seize, graphite, molybdenum, etc.)
- Pressure loads increase stresses near pressure case and on the mounting flange teeth and the mounting flange deforms
  - Effect at teeth needs to be quantified with 3D analysis

Temperature Plot for 416 mounting flange in current configuration

- 416 SS mounting flange, seal ring, and hirth ring
- 17-4 SS retaining ring and wire cover
- 2260W heater power, corresponds to 766°F max temp
- “Soft” thermal contact at seal ring to hirth ring
- “Soft” thermal contact at mounting flange to pressure case

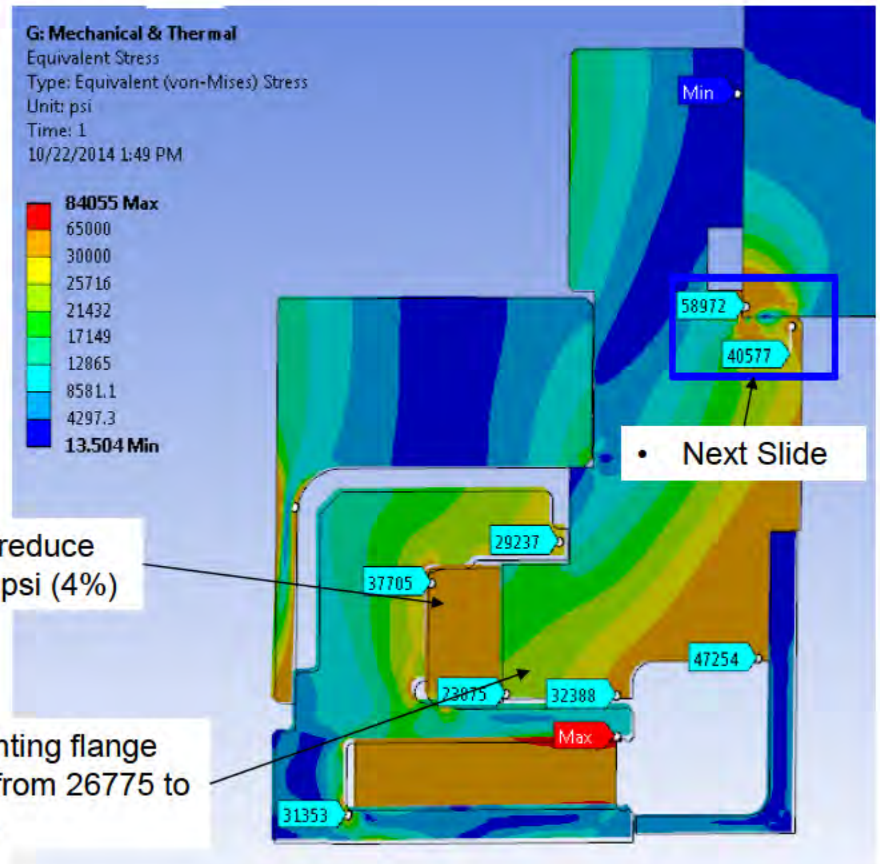


Radial Deformation



- There's a tendency for the ring to tilt slightly relative to the seal ring

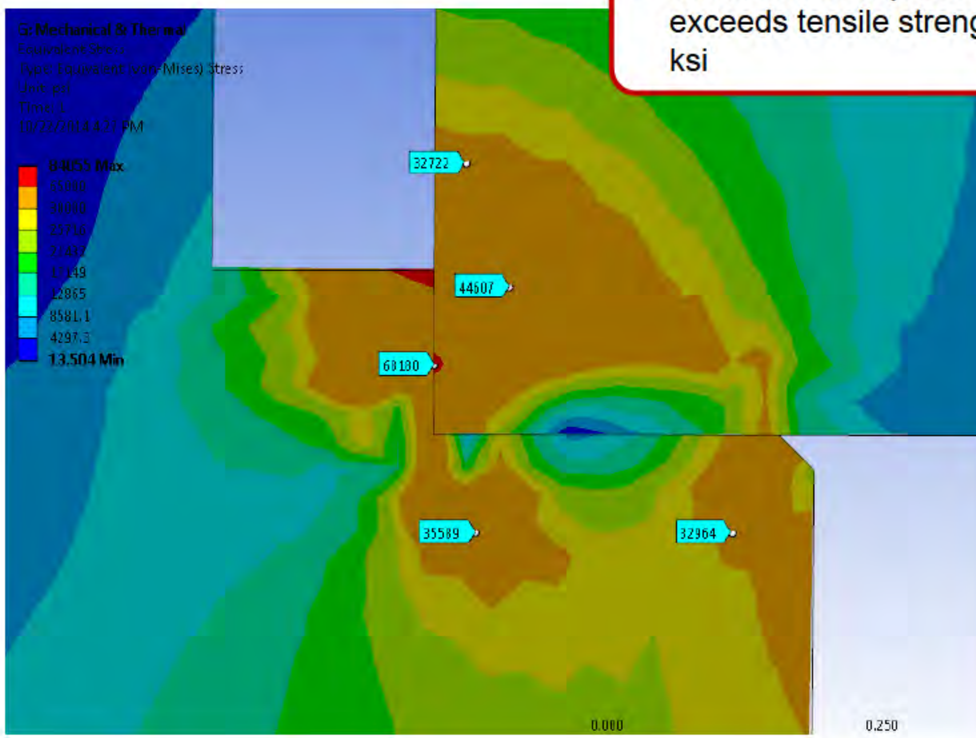
Equivalent Stresses



- Stresses in ring reduce from 39420 to 37705 psi (4%)
- Stresses in mounting flange by teeth reduce from 26775 to 23875 psi (11%)

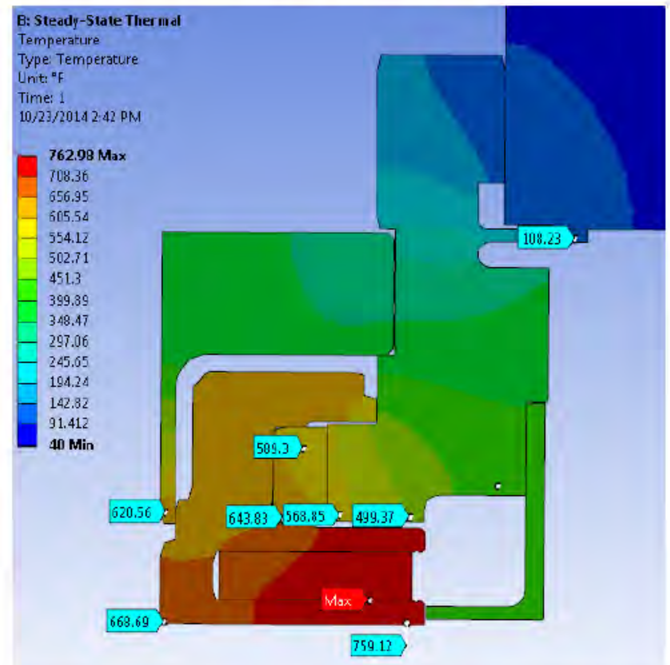
Equivalent Stresses

- 68 ksi stress in pressure case exceeds tensile strength of 36 ksi

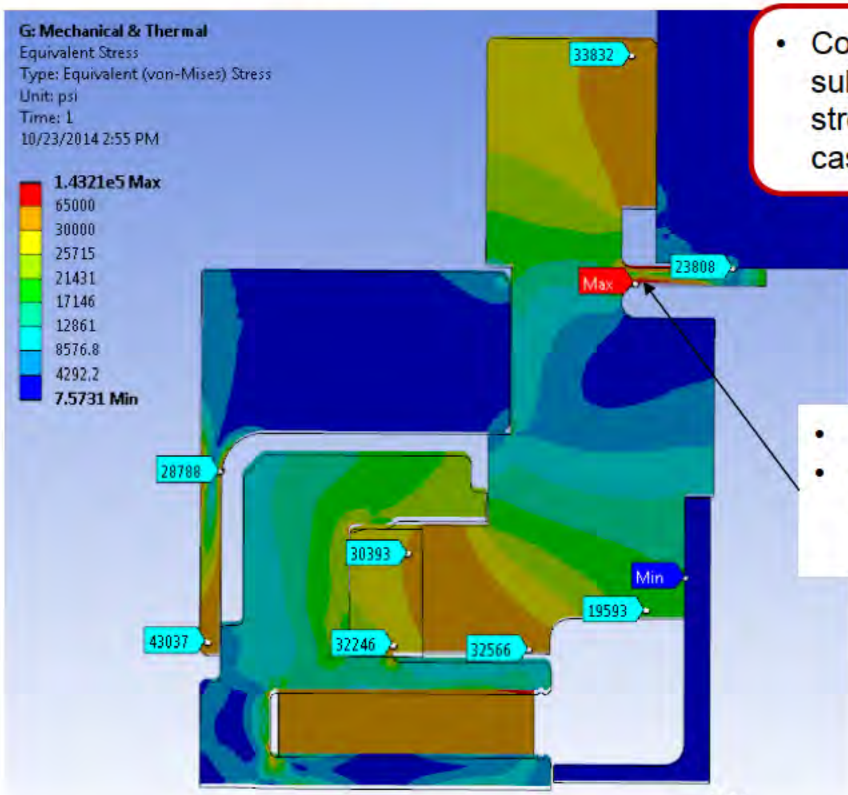


Added compliant tongue on mounting flange

- 416 SS mounting flange, seal ring, and ring
- 17-4 SS retaining ring and wire cover
- 1800W heater power
- Hard thermal contact at seal ring to ring
- Hard thermal contact at mounting flange to pressure case
- Lip extends .30” and and is .10” thick



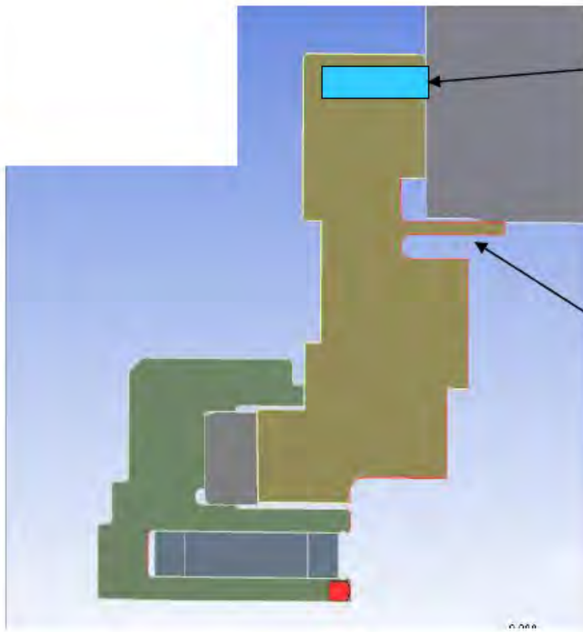
Equivalent Stresses



- Compliant tongue substantially reduces stresses in pressure case

- 143 ksi stress on lip
- Can dial in stress to by adjusting thickness or length of the lip

Compliant Tongue Discussion

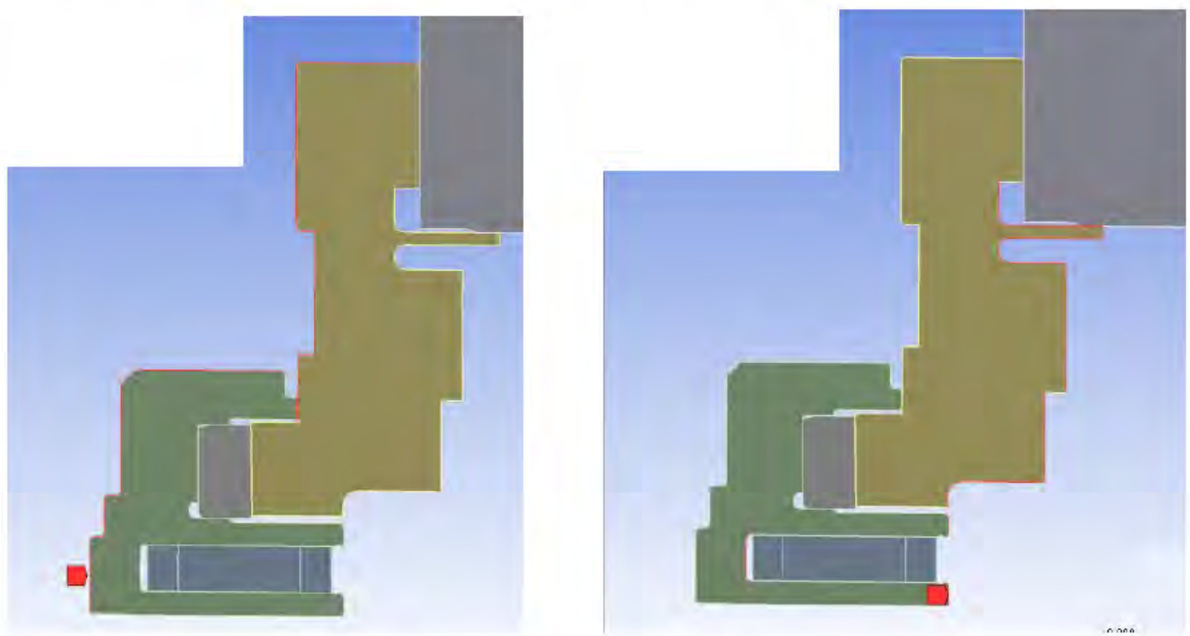


Potential alternate location for compliant member

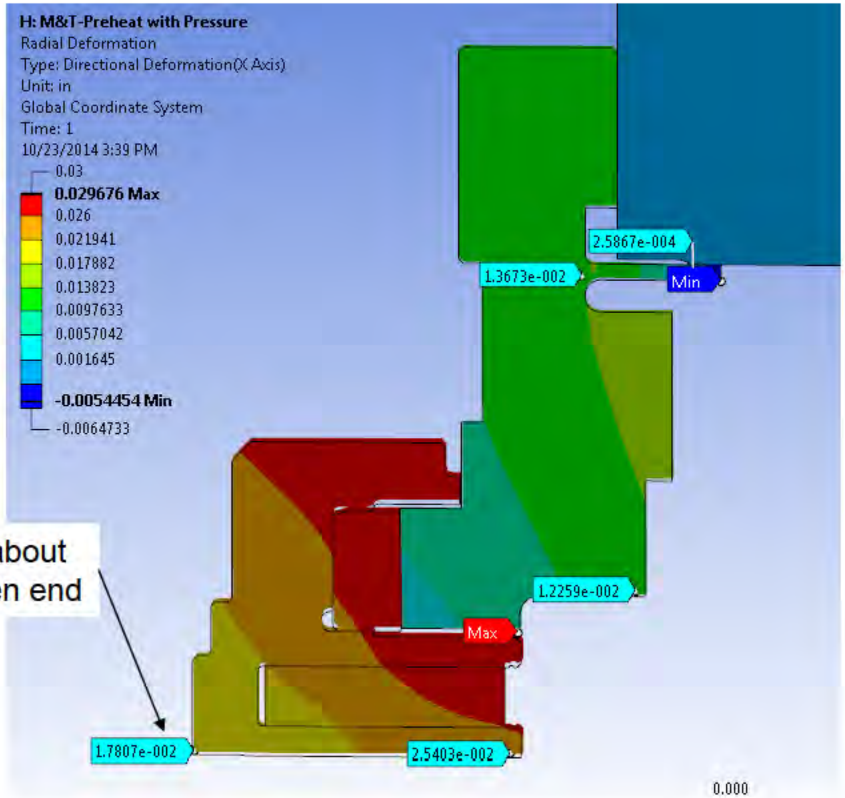
- Extension into existing pressure case envelope creates issues for routing vent ports

Operational Pressure Loads, Plus SS pre-heat Thermal

- 1390 psia [redacted] exit pressure
- 150 psia cavity pressure
- Applied withpre-heat thermal loads

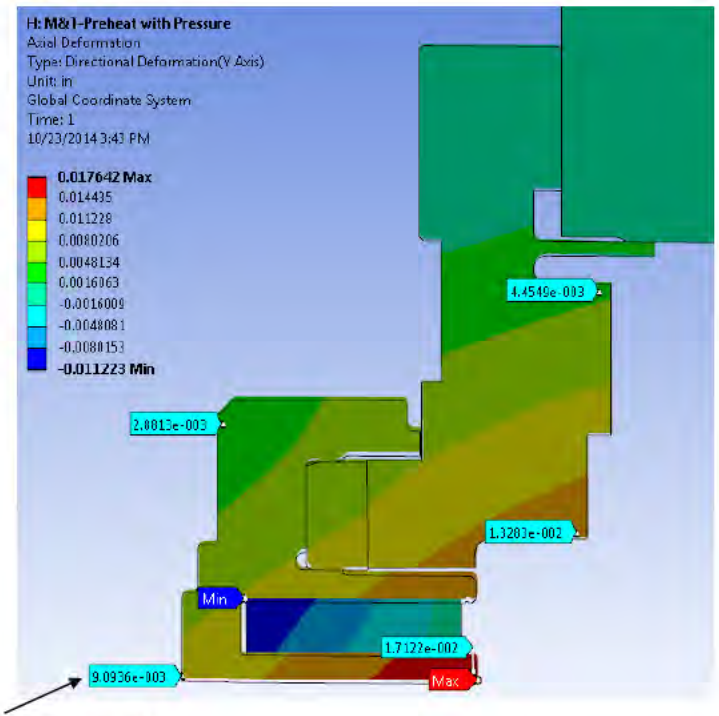


Radial Displacement



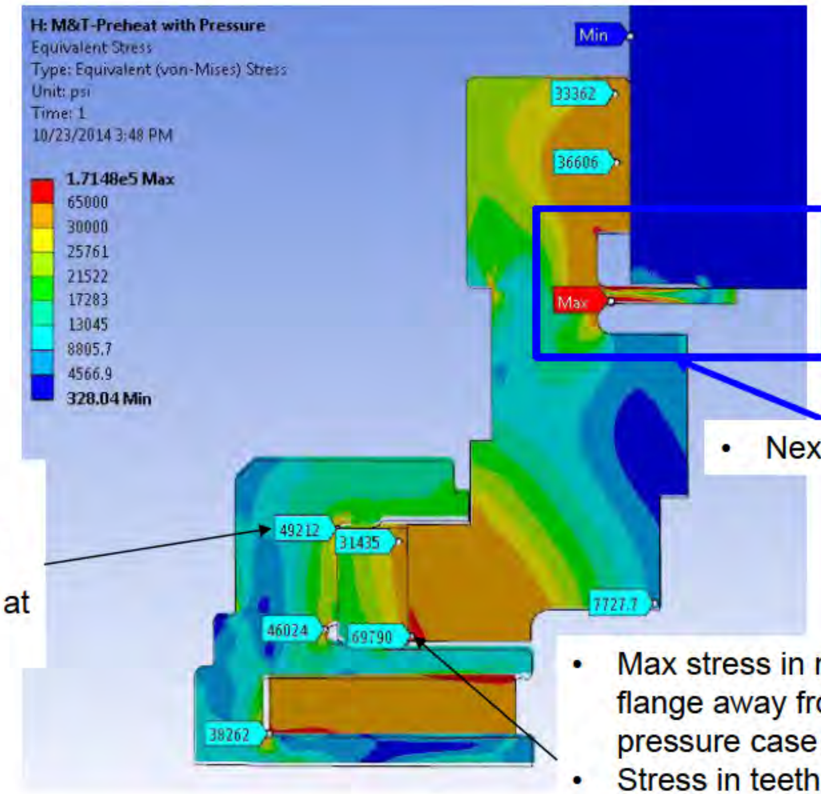
- Additional tilt by about .003 on non-driven end

Axial Displacement



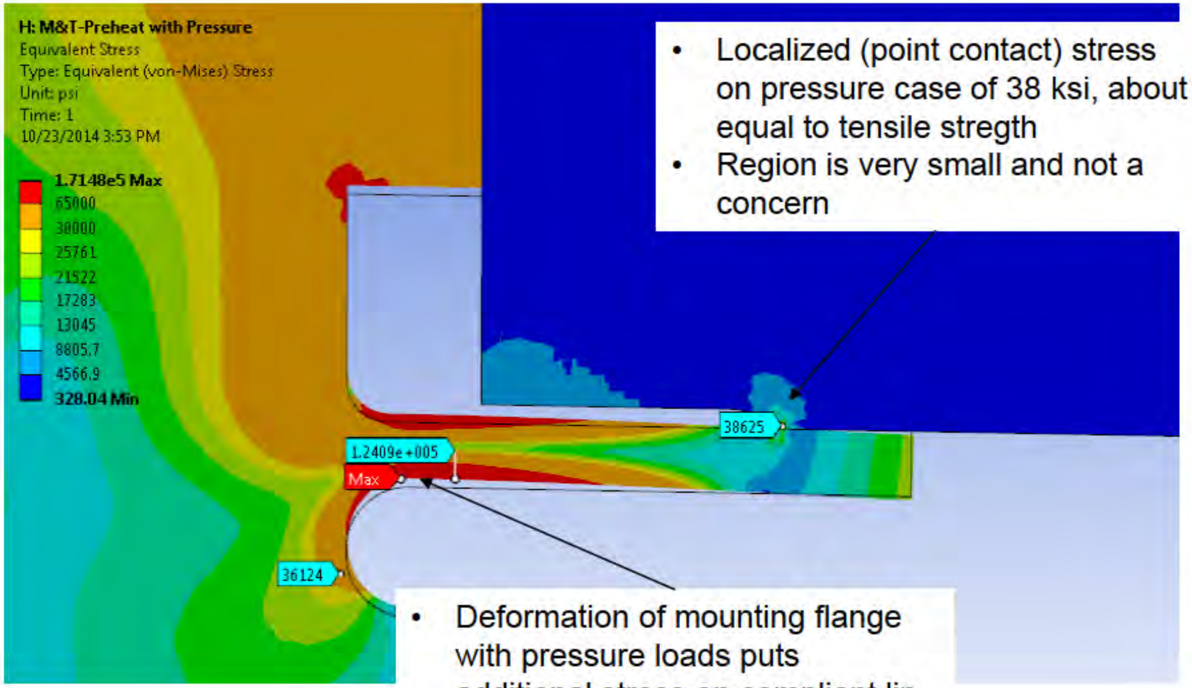
- Axial displacement of .009" on non-driven end

Equivalent Stress



- Next Slide
- Stresses in Seal Ring is 49 ksi and below est yield strength of 78 ksi at 800°F
- Max stress in mounting flange away from pressure case is 70 ksi
- Stress in teeth could be 2-3X, or 140-210 ksi and needs review

Equivalent Stress



- Localized (point contact) stress on pressure case of 38 ksi, about equal to tensile stregth
- Region is very small and not a concern

- Deformation of mounting flange with pressure loads puts additional stress on compliant lip
- Stress at lip is 171 ksi and exceeds 145 ksi RT yield strength of 17-4 SS

SUPPORT DATA

Appendix 9.3.3

Compressor Layout

[redacted]

Supercompressor Layout Preliminary Design Review

[redacted]

Paul Chilcott

Rob Draper

[redacted]

DRESSER-RAND

Outline

- Steady state thermal FEA
  - 2D (Paul Chilcott)
  - 3D (Rob Draper)
- Steady state structural FEA
  - 2D (Rob Draper)
- Transient thermal FEA
  - 2D Start Up (Paul Chilcott)
  - 2D Shutdown (Paul Chilcott)

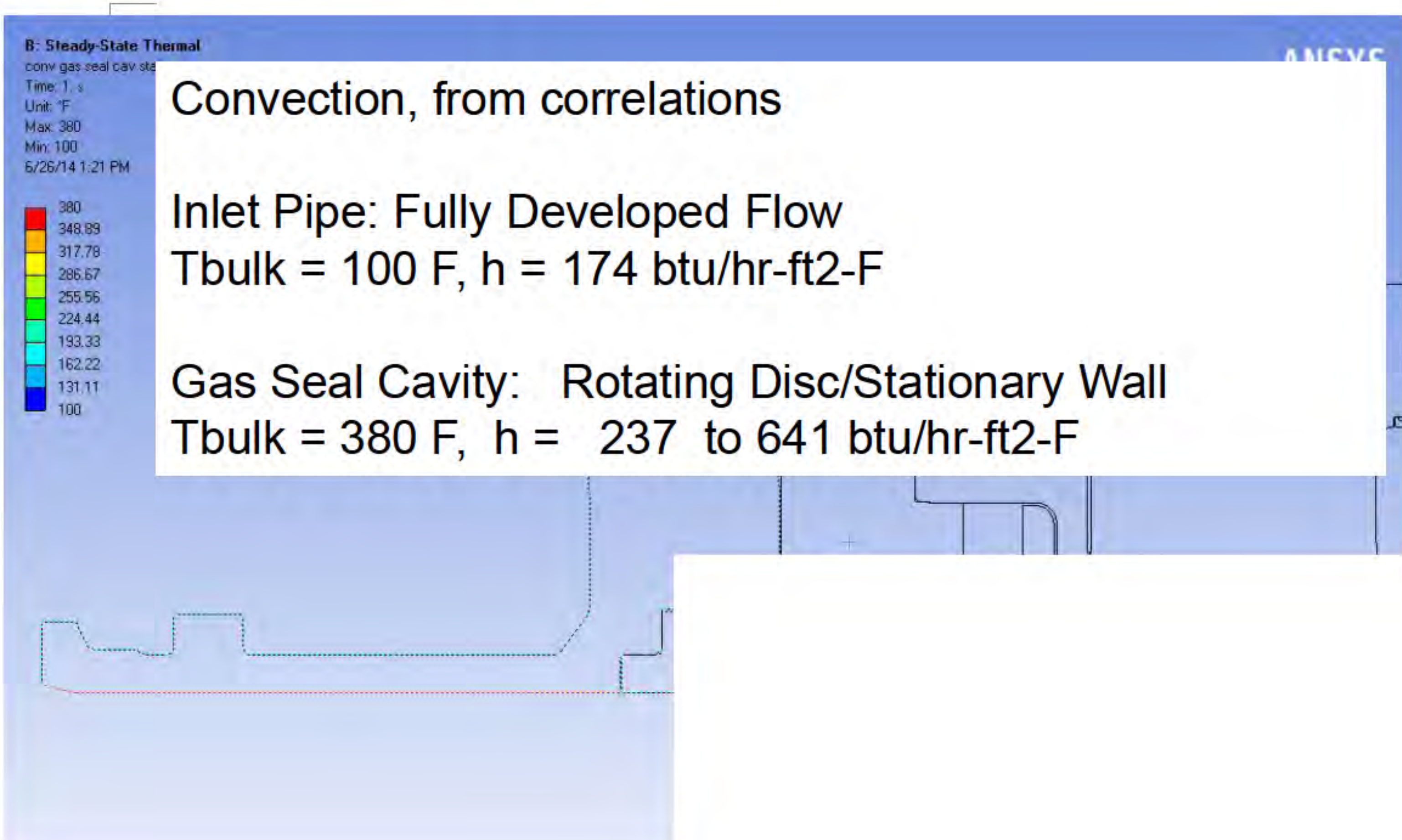
System Requirements

- Assumptions:
  - Utilize conjugate CFD for all [redacted] thermal loads
  - Compressor layout is primarily axi-symmetric in form
- Goals:
  - Establish hot condition for compressor rig
  - Ensure adequate running clearances and desired component fits
  - Assess transient startup/shutdown clearances

2D Steady State Thermal FEA

- Reviewed by analysis team on 6/26/2014
- Revisions completed by Paul Chilcott on 6/27/2014

2D Thermal Model



Convection, from correlations

Inlet Pipe: Fully Developed Flow
Tbulk = 100 F, h = 174 btu/hr-ft2-F

Gas Seal Cavity: Rotating Disc/Stationary Wall
Tbulk = 380 F, h = 237 to 641 btu/hr-ft2-F

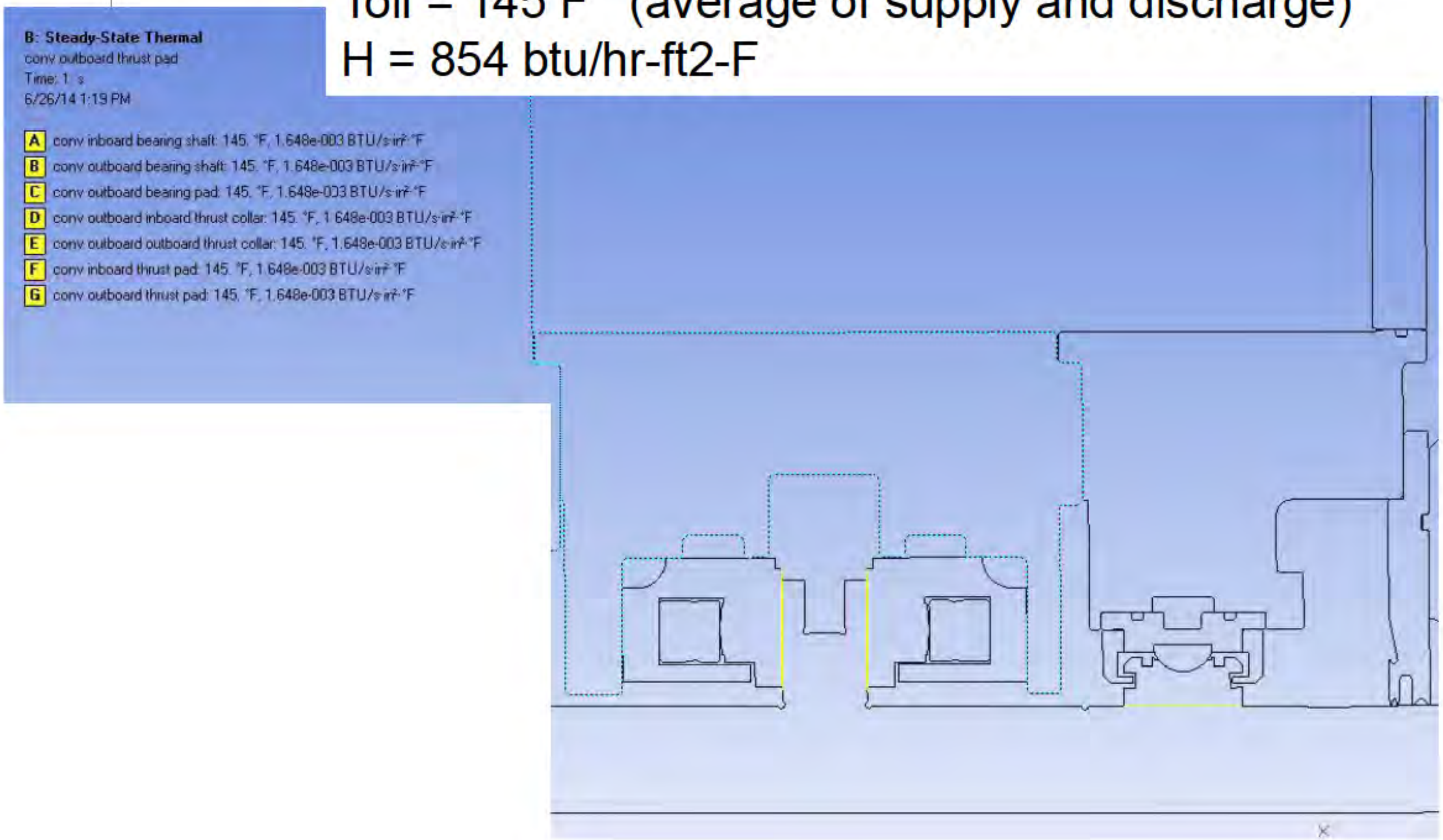
2D Thermal Model

Oil Convection

Gazely, Rotating Cylinder and Annulus

Toil = 145 F (average of supply and discharge)

H = 854 btu/hr-ft2-F



2D Thermal Model
Steady State Results

2D Thermal Model
Steady State Results

Component	Max Temperature (F)
[redacted] Hub	
[redacted]	
Stator Ring	
Volute	
Seal Housing	
Casing	
Gas Seal	
[redacted] End Journal Hsg	
Coupling Guard Adapter	
Tiebolt	
Shaft	
Thrust Hsg	
Inlet Head	
Coupling End Journal Hsg	

3D Steady State Thermal FEA

- Reviewed by analysis team on 7/2/2014

3D Thermal Model

- 2D temperature distribution interpolated on bundle
- Corresponding contacts and loads applied to case, coupling guard adapter, bearing housing
- Coarse Mesh:
  - 1.083M nodes, 609K elements
  - To run higher density mesh if time permits (~3M nodes)

3D Temperature Distribution

- Discharge nozzle insert and volute gaps isolate case from exhaust
- 20°F variation around case at discharge plane

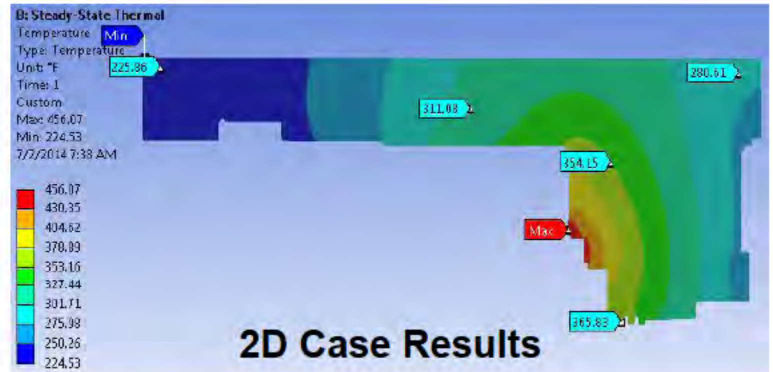
2D/3D Comparison

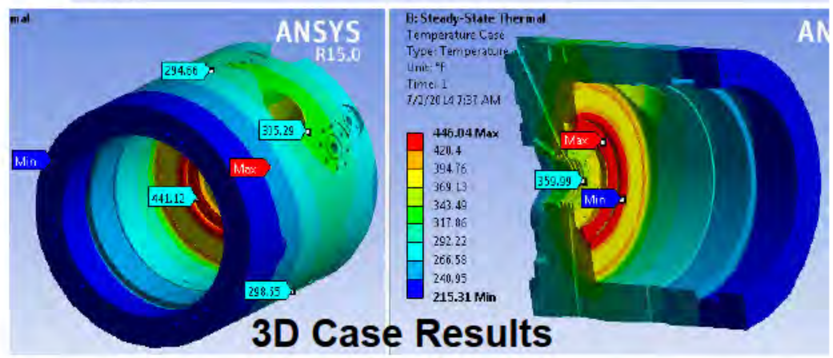
- Good correlation between 2D & 3D

Move forward with 2D structural and thermal-transient FEA to better appreciate startup and actual test conditions

3D Case Temperature Distribution

- Maximum case temperature is 450°F
- Case loses ~ 5 HP to surroundings
  - Reaches thermal equilibrium after 20 hours of operation





Steady state temperature is expected to be 200°F below pressure vessel code rating

2D Steady State Structural FEA

- Reviewed by analysis team on 7/2/2014

2D Structural Model

- Contacts similar to thermal (suppressed thermal gaps)
- Loads: 2D temperatures, internal pressure, centrifugal
- Mesh (same as thermal): 53k nodes, 49K elements

Mesh

Loads

2D Structural: Axial Deformation

Volute/Head Gap:
Cold: .060”, **Hot: .053”**
Worst Case Trans. (hot volute/cold head): **-.008”** (interf.)

Eye [redacted]
Cold: .246”, **Hot: .263”**
Worst Case Trans.
(hot [redacted] cold inlet): .224”

Eye Hub:
Cold: .190”, **Hot: .218”**
Worst Case Trans.
(hot [redacted] cold inlet): .175”

Thrust
Cold: .047”, **Hot: .050”**
Worst Case Trans.
(hot shaft/cold bearing): .047”

2D Structural: Radial Deformation

Case/Volute Gap:
Cold: .030”, **Hot: .018”**
Worst Case Trans. (hot volute/cold case): **-.007”** (interf.)

[redacted] Tip Seal:
Cold: .105”, Hot: .095”
Worst Case Trans.
(hot [redacted] cold inlet): **.074”**

[redacted]:
Cold: .010”, **Hot: .008”**
Worst Case Trans. (hot
[redacted] cold inlet): **-.001”** (interf.)

Damper Sea
Cold: .010”,
Worst Case Trans. (hot
[redacted] cold inlet): **-.007”** (interf.)

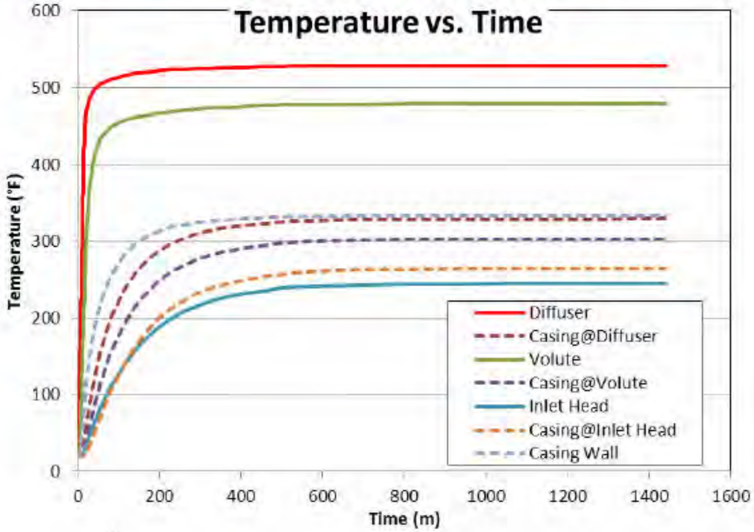
2D Transient FEA, Startup

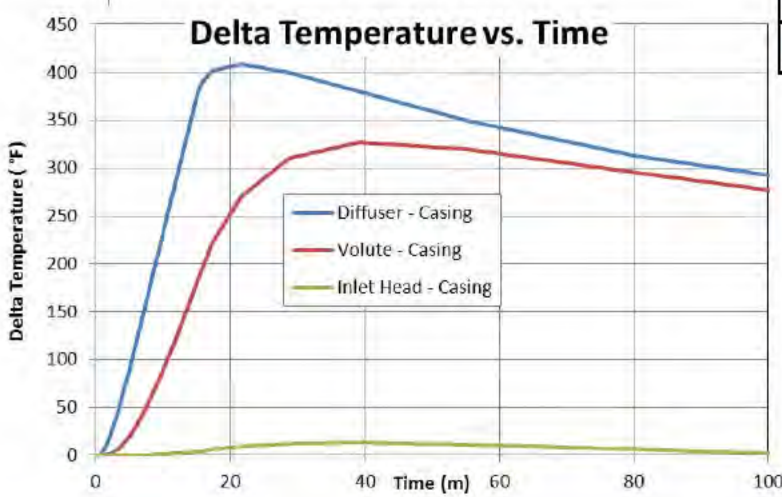
Assumptions:

1) 20 °F Ambient
2) Initial Condition Uniform 20 °F
3) 15 minute Ramp to T_{in} = 100 °F
4) 24 Hr. Dwell
5) Contact Status & Gap Conductance Not Updated

Load Step	End Time	RPM	Gas Inlet Temp. (F)	Bearing Journal Oil Temp. (F)	Oil Discharge Temp. (F)
1	1 s	0	20	120	120
2	15 m		100	145	160
3	8 hr		100	145	160
4	24 hr		100	145	160

2D Transient FEA, Startup





Max Delta Temperatures

		Temperature (F)		
Part 1 / Part 2	Time (m)	Part 1	Part 2	Delta
Diffuser / Casing	21.7	469	61	**408**
Volute / Casing	39.1	408	81	**327**
Inlet Head / Casing	39.1	61	48	**14**

Steady State Temperatures (24 hrs)

	Temperature (F)		
Part 1 / Part 2	Part 1	Part 2	Delta
Diffuser / Casing	528	329	**199**
Volute / Casing	479	303	**176**
Inlet Head / Casing	245	264	**-20**

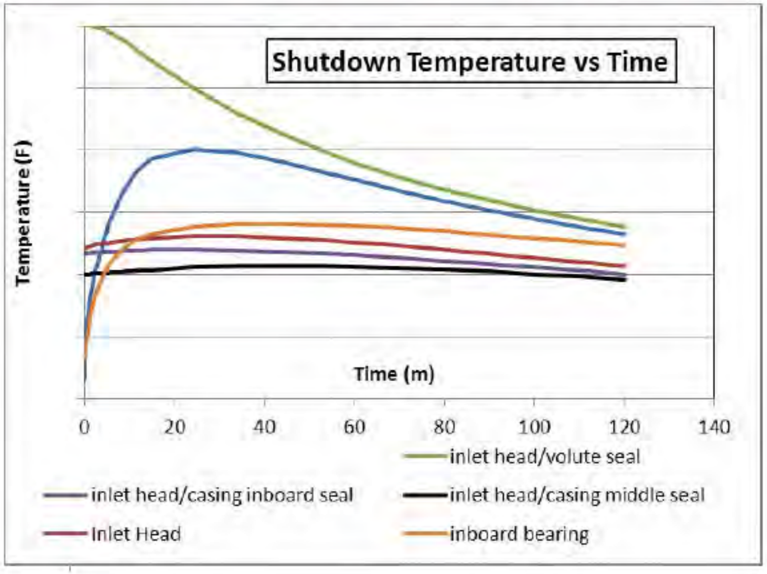
2D Transient, Shutdown

Assumptions:

1) Initial Condition = Steady State, 70 °F Ambient
2) 2 Hr. Dwell
3) Blow down Cooling Effect Negligible
4) No post shutdown lube oil
5) Contact Status & Gap Conductance Not Updated

Thermal BC	Steady State	Shutdown
External Convection	✓	✓
External Radiation	✓	✓
Conduction Across Gaps	✓	✓
Flowpath Convection	✓	
Recess & Seal Leakage Convection	✓	
Oil Convection	✓	

Shutdown Temperatures



Summary

- Compressor layout modeling completed for configuration at design point conditions
  - 2D layout model validated by 3D thermal results
  - Steady state, thermal/structural baseline is established
  - Transient startup and shutdown, thermal analyses complete
- Remaining Work
  - Update compressor layout for configuration
  - Transient structural analyses to evaluate clearances and stress
  - Run off design conditions as needed

Steady State Axial Deflections in Region

- Inlet bolts to the pressure head which rests against the shear ring. Pressure case grows between shear ring and DE case wall
- rests on volute, which grows from DE case wall. grows toward from volute wall.

[redacted]

Supercompressor Layout
Preliminary FDR Results

[redacted]

Brian Massey
Paul Chilcott
Mark Krzysztopik

[redacted]

DRESSER-RAND

Major Design Differences From Last 2D Thermal Analysis

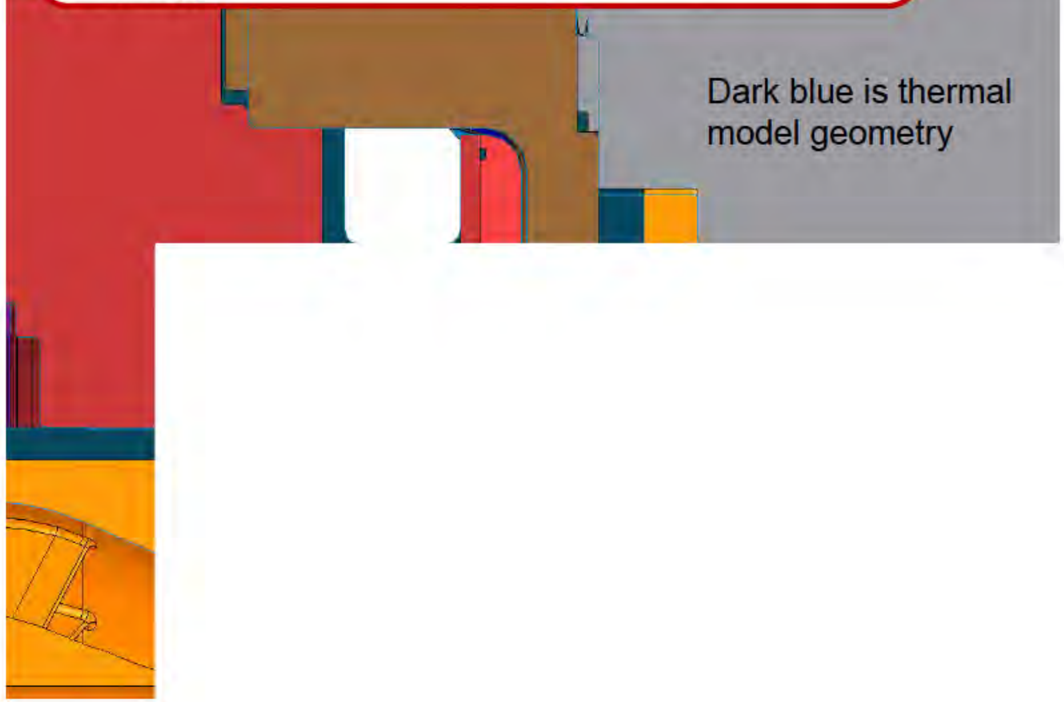
- Using an rather than a
- seal changes
  - Radius increase resulting in ~70% higher surface speeds
  - geometry rather than labyrinth tooth
  - New design which allows stationary seal to grow radially
- Incorporated a baffle plate/heat shield to protect the dry gas seal from hot seal exit gas

Major Technical Differences From Last 2D Thermal Analysis

- CFD results are from FineTurbo simulations rather than ANSYS CFX
- Heat transfer coefficients (HTC's) manually calculated using flat plate correlations and CFD flow quantities, versus extracting HTC's directly from the CFD results as we did before
- Bulk temperatures in gas seal cavity come from CFD model, no longer assumed to be constant 350F

Simulation vs Current Geometry

Thermal model geometry is a snapshot in time, a few changes have been made recently which will need to be reflected in the model (e.g. [redacted] profile and seal geometry)



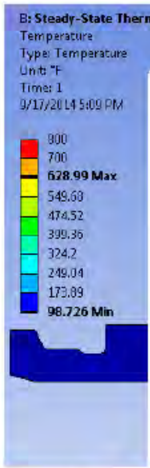
Simulation vs Current Geometry

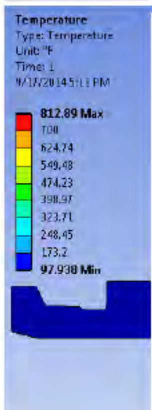
Similarily, results from CFD do not perfectly match thermal model geometry

Black lines are CFD data, colored lines are extrapolations to thermal geometry

2D Steady State Temperature Distribution Comparison

Previous Result

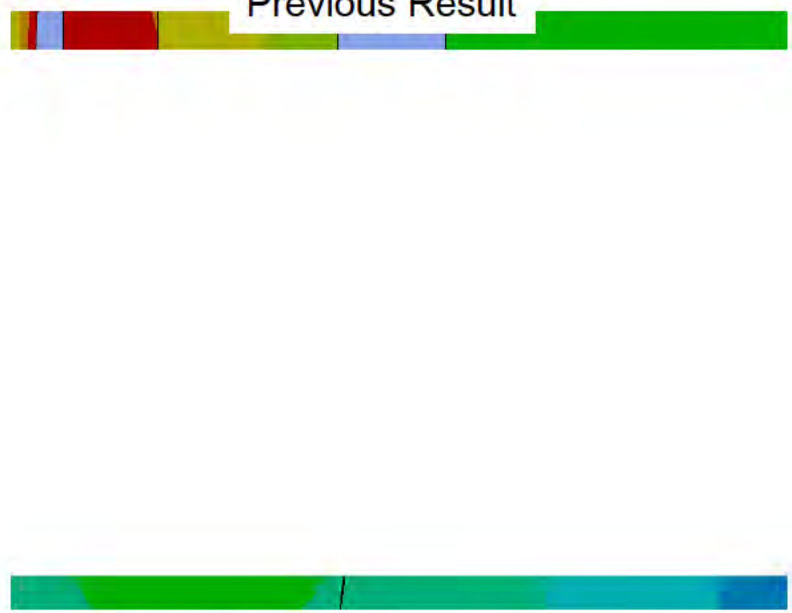




DGS/Seal Cavity Region Case Temperatures

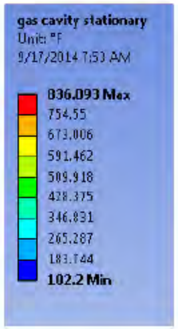
Current Result

Previous Result



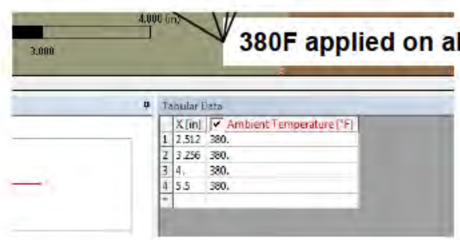
DGS temperatures have increased significantly

Seal Gas Cavity BC's



Current BC's

Previous BC's



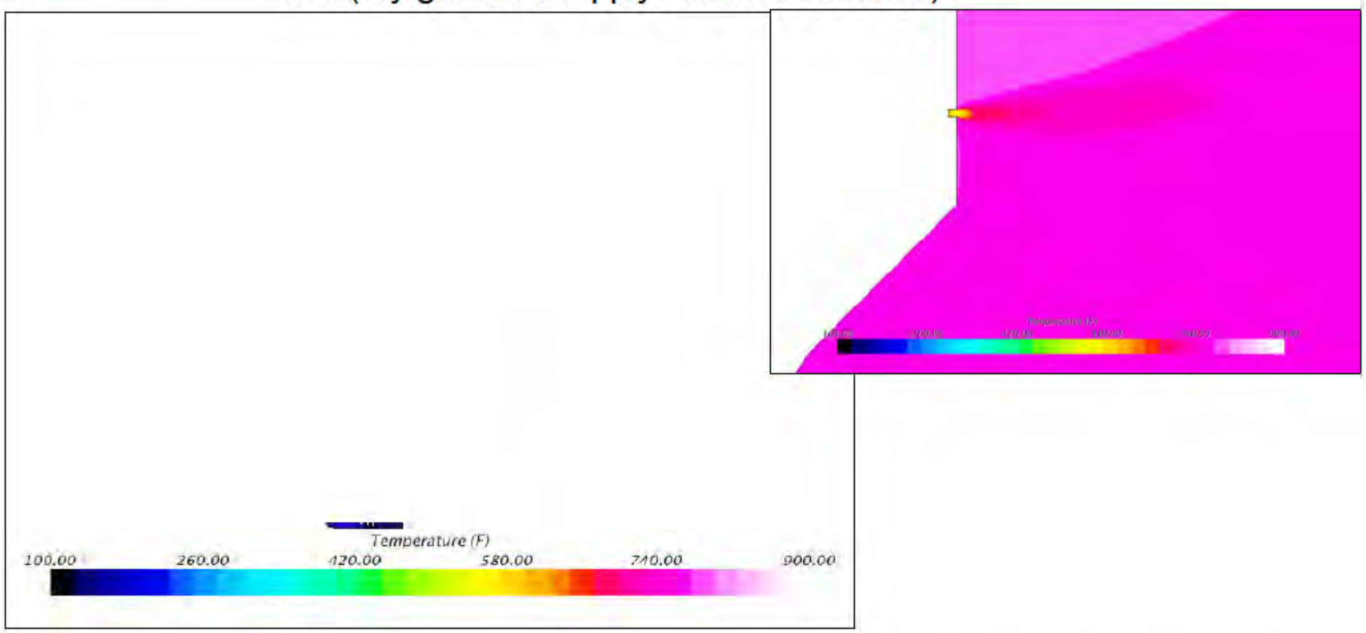
380F applied on all walls

Wall temperatures from CFD results

Temp BC's can explain the added heat into the pressure case near the DGS

Dry Gas Seal Region CFD – Case 2

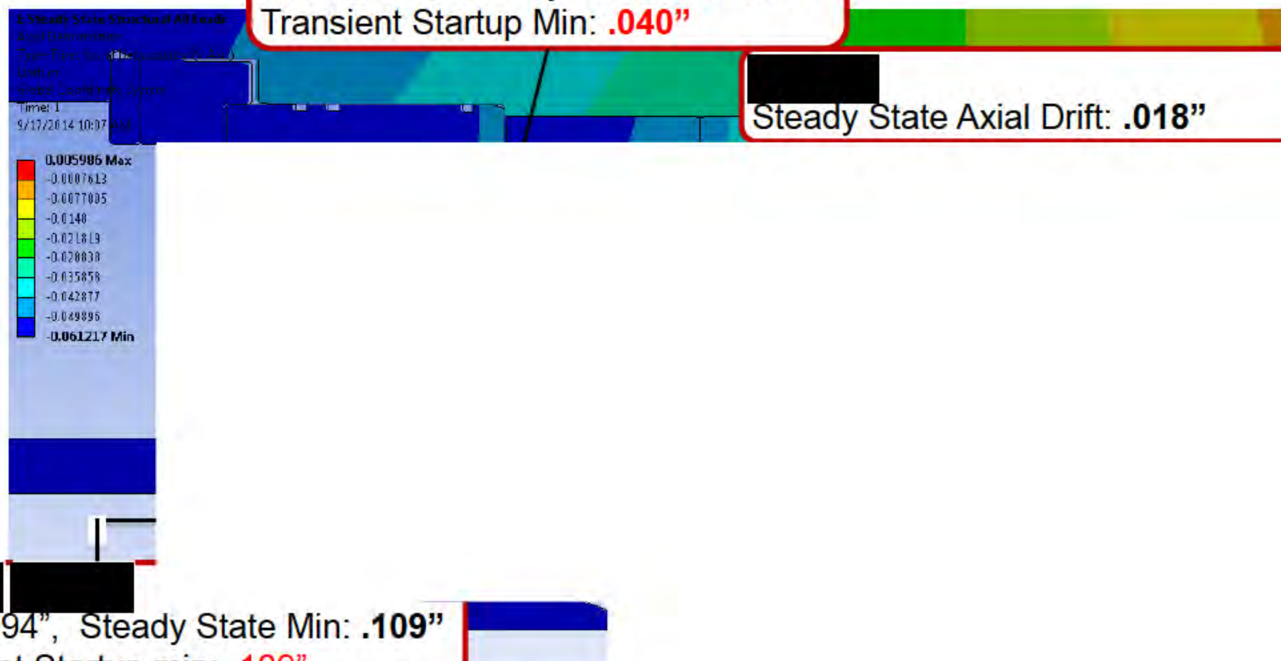
- Includes domain downstream of seal exit
- Includes seal leakage exit and region from dry gas seal supply through the seal (dry gas seal supply 0.1 lbm/s at 100F)



*Used for current thermal analysis

2D Structural: Axial Deformation

Volute/Head Gap:
Cold: .060”, Steady State Min: **.051”**
Transient Startup Min: **.040”**



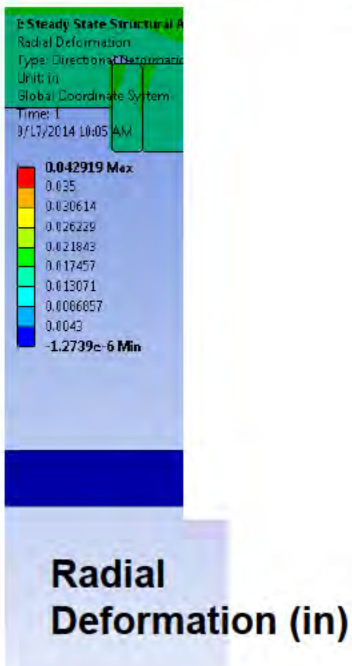
[redacted]
Steady State Axial Drift: **.018”**

[redacted]
Cold: .094”, Steady State Min: **.109”**
Transient Startup min: .109”

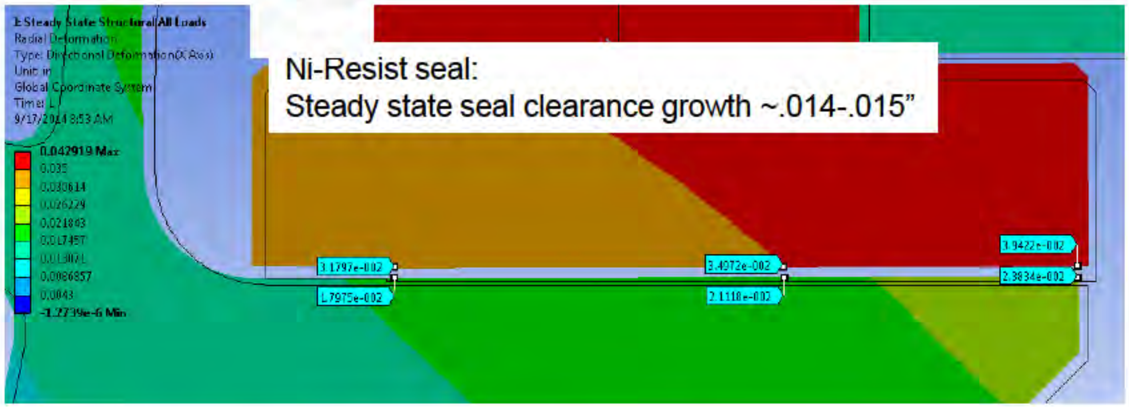
2D Structural: Radial Deformation

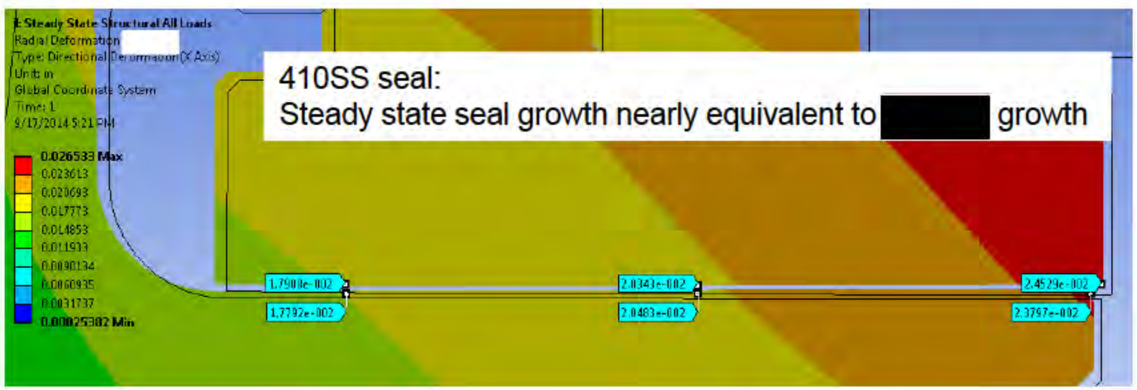
Case/Volute Gap:
Cold: .030”, Steady State Min**: .021”**
Startup Transient Min: **.020”**

[redacted] Tip:
Cold: .105”, Steady State Min**:** .102”
Startup Transient Min: **.084”**



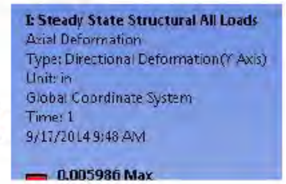
Steady State Clearance Seal





Steady State [redacted] *2D Axial Deflections*

- Note: [redacted] blades are represented as axisymmetric and may not give accurate deflection results. 3D analysis will be used to confirm
- Based on 2D analysis [redacted] has moved .012-.022" away from [redacted] in this region



2D Transient FEA, Startup

Assumptions:

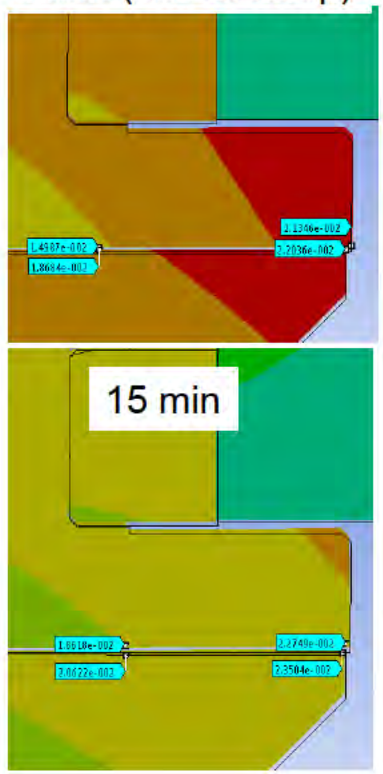
1) 20 °F Ambient
2) Initial Condition Uniform 20 °F
3) 4 minute Ramp to
4) 24 Hr. Dwell
5) Contact Status & Gap Conductance Minimally Updated from Previous analysis

Load Step	End Time
1	1s
2	4m
3	15m
4	8hr
5	24hr

Gas Inlet Temp. (F)	Bearing Journal Oil Temp. (F)	Oil Discharge Temp. (F)
20	120	120
100	145	160
100	145	160
100	145	160
100	145	160

Seal Transient Deflections – 410SS

4 min (end of ramp)



- At 4 minutes seal gap has closed 3-4 mil in front half of seal
- At 15 min seal clearance getting better
- Preliminary results indicate heaters may not be necessary

Summary

- No major impacts on head, inlet, volute, , or [redacted] design
- Case temperatures near seal cavity are very high, dry gas seal temperatures very high, this region requires more design
- Making good progress on managing seal clearance, may not require startup heaters
- [redacted] is much cooler than before but moving away from [redacted] clearance needs to be maintained at ~.010” for performance, requires more design
- [redacted] Diffuser misalignment due to thermal growths is not insignificant, may need to be addressed

Work To Be Done…

- Continued refinement of thermal model and post-processing
- Thermal isolation design in gas seal cavity to reduce heat going into pressure case
- Detail work/sensitivity analysis on seal
- Sensitivity analysis of BC's on [redacted] thermal growth, management of [redacted] gap
- Potential 3D CFD of hole with geometry
- 3D transient thermal analysis with blade deflections (for [redacted]
- Re-run FEA analysis at test conditions (i.e. 150psia inlet, reduced mass flow)
- Re-calculate HTC's using "CFX" method for apples to apples comparison to previous thermal analysis

[redacted]

Questions

[redacted]

Contact Rob Draper:

bmassey@dresser-rand.com

[redacted]

Transient Model Update

- Thermal model did not include mid-ramp transients. First "time-step" investigated @ 4 minute mark.
- HTC and temperature data taken from CFD, scaled pressure, temperature and velocity data for part speed transient thermal model.
  - 25%, 50%, 75%, 100% Speed

$$T \propto RPM^2$$
$$\text{Vel} \propto RPM$$
$$(P/Pin)^{\wedge}((k-1)/k) - 1 \propto RPM^2$$

- Recalculated flat plate correlations for part speed conditions

Appendix 9.4.1

Compressor Drivetrain and Skid FDR

Final Design Review [redacted]

Drive Train/Lube Oil System

System owner(s):

Mark Krzysztopik

6/13/2014

[redacted]

DRESSER-RAND.

Requirements

1. Utilize Ramgen test facility for all builds of Supercompressor test program.
2. Deliver 10MW from existing Laurence Scott Motor at RPM
3. Support start of test in November 2014.
4. Maintain fixed position of pressure case for duration of test program.
5. Minimize re-work required when installing ‘production’ gearbox
6. Ensure facility lube oil system is sufficient to meet drive train requirements
7. Rotordynamic stability throughout operating range.

Existing Lube Oil Skid

- Lube oil skid and piping designed for 219 GPM of oil flow.
  - Adequate for current design
- Additional capacity to control lube oil temperature possible with addition of three way valve has been incorporated

	Flow (gpm)	Heat Load (BTU/Hr)	FHP
Motor (ramgen) - ISO-32	3.68	34116.4	13.4
Existing Allen Gears - ISO 32	75	471139	185.00
Proposed Allen Gears - ISO 32	80.4	529714	208
Compressor Thrust Bearing:	97	392192	154.00
Compressor NDE Journal:	13.3	198642.5	78.00
Compressor DE Journal	13.3	198642.5	78.00
TOTAL	279	1790329.6	703.00
Total using Allen proposed Option =	207.68	1353307	531.4
Available at Ramgen	219	1.8	MBTU/Hr

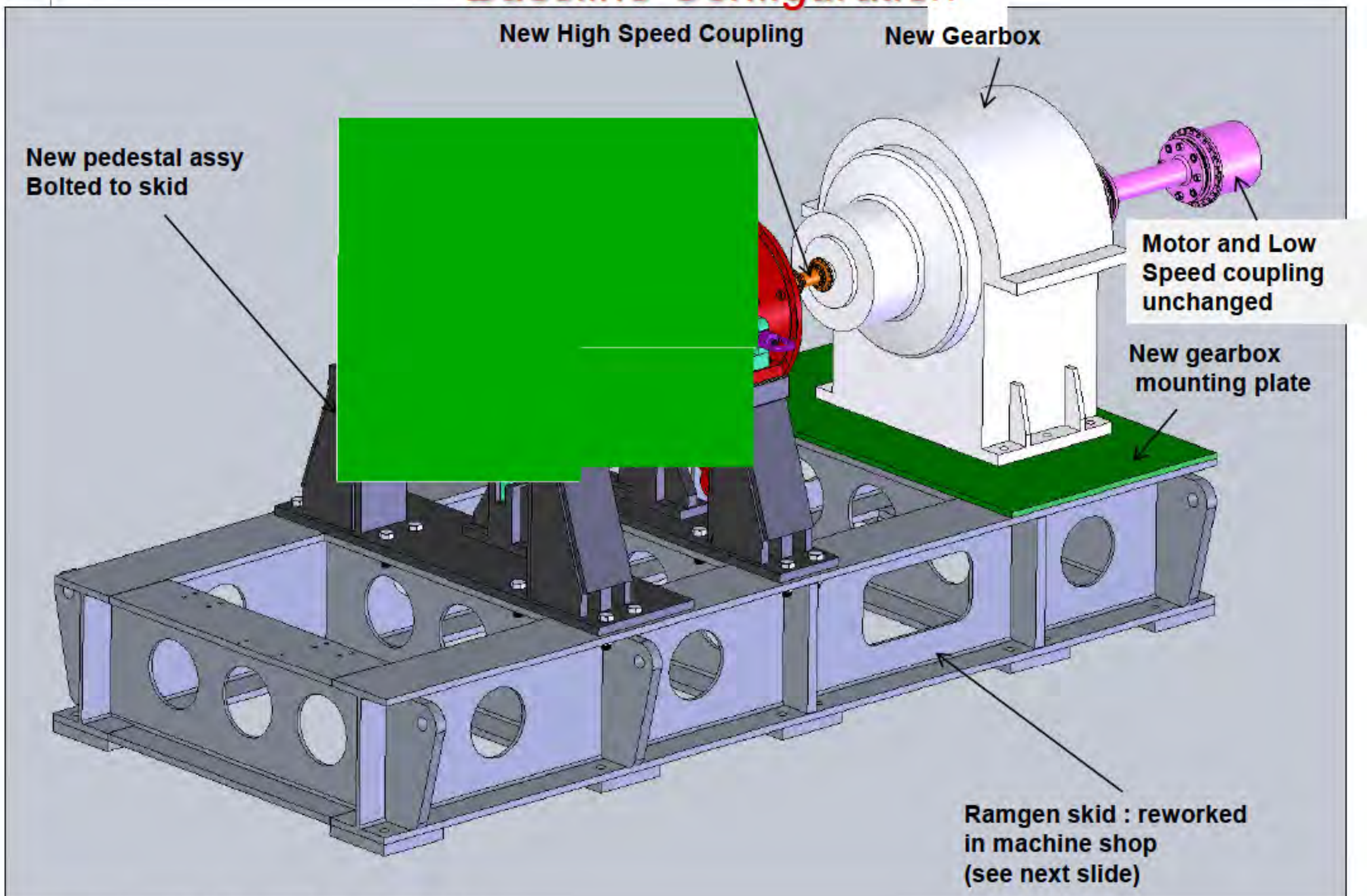
Supercompressor Integration

Baseline Configuration

Supercompressor
+
15" Kopflex High Speed Coupling
+
New Allen-Gears Gearbox
+
Skid unchanged from 'earlier-to-test' configuration
(rework of Ramgen Skid)

Note: Case image for reversed direction of rotation

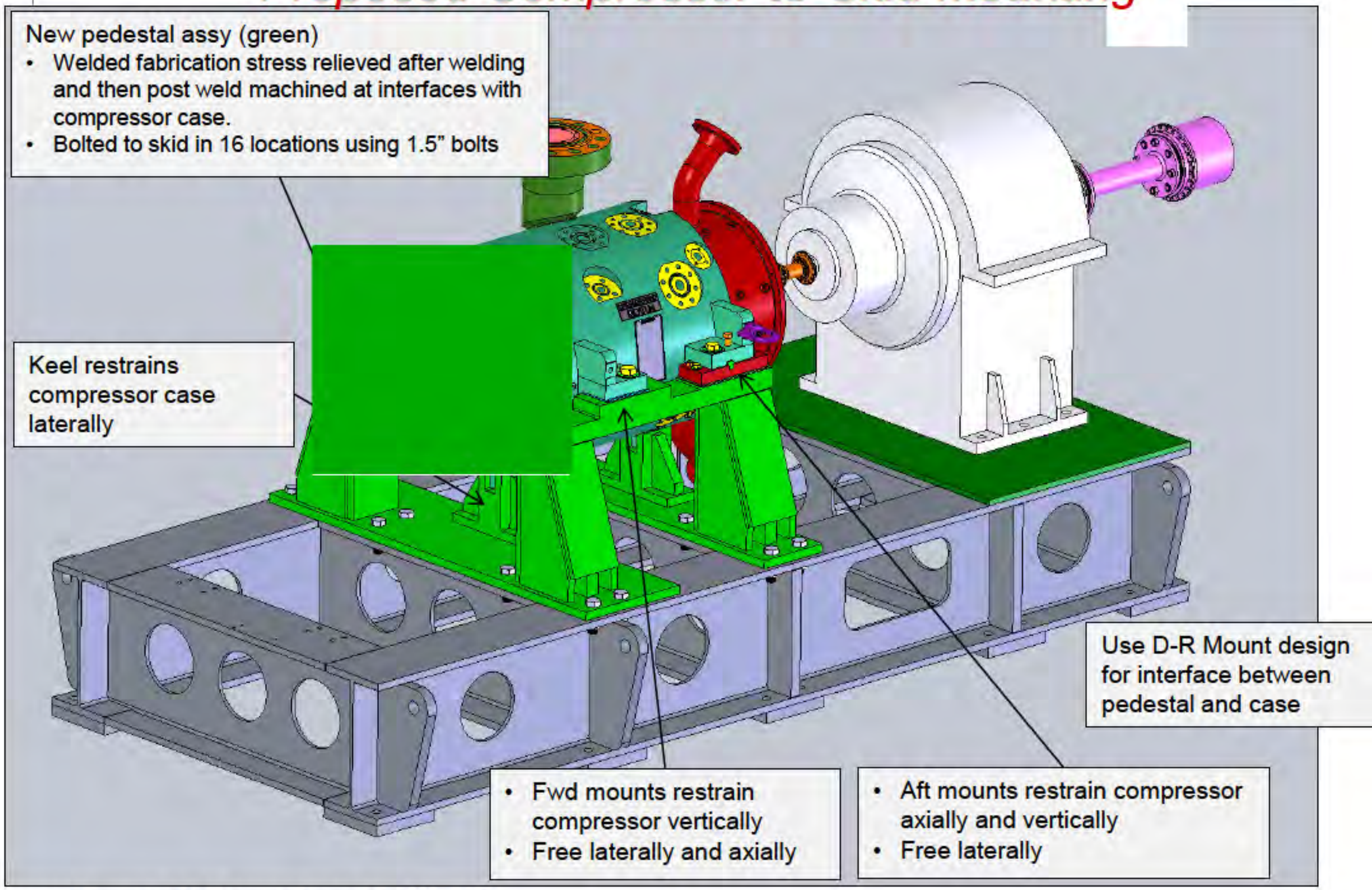
Baseline Configuration



Ref: Supercompressor Conceptual Test Assembly-Option1b.SLDASM

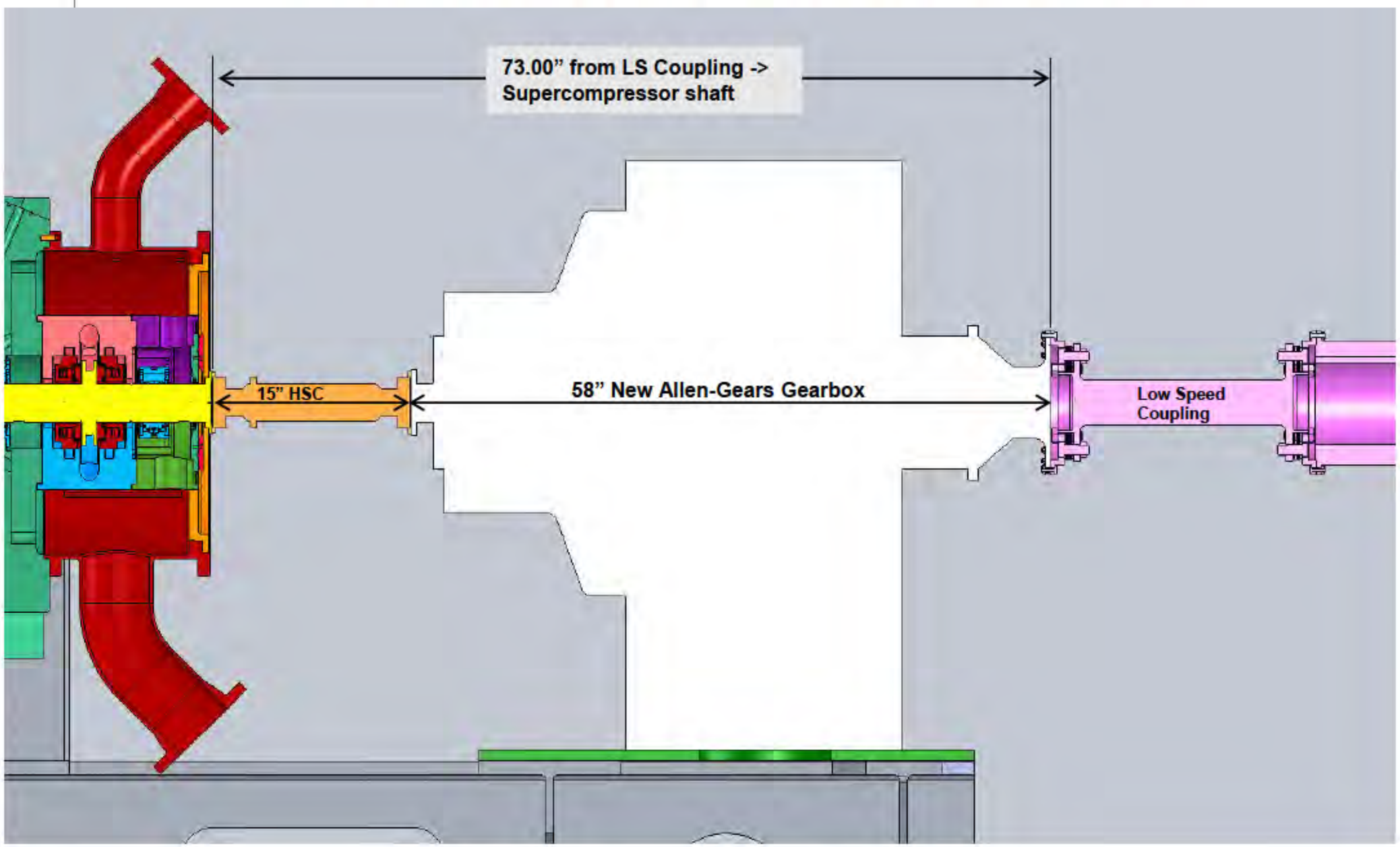
Note: Case image for reversed direction of rotation

Proposed Compressor-to-Skid mounting



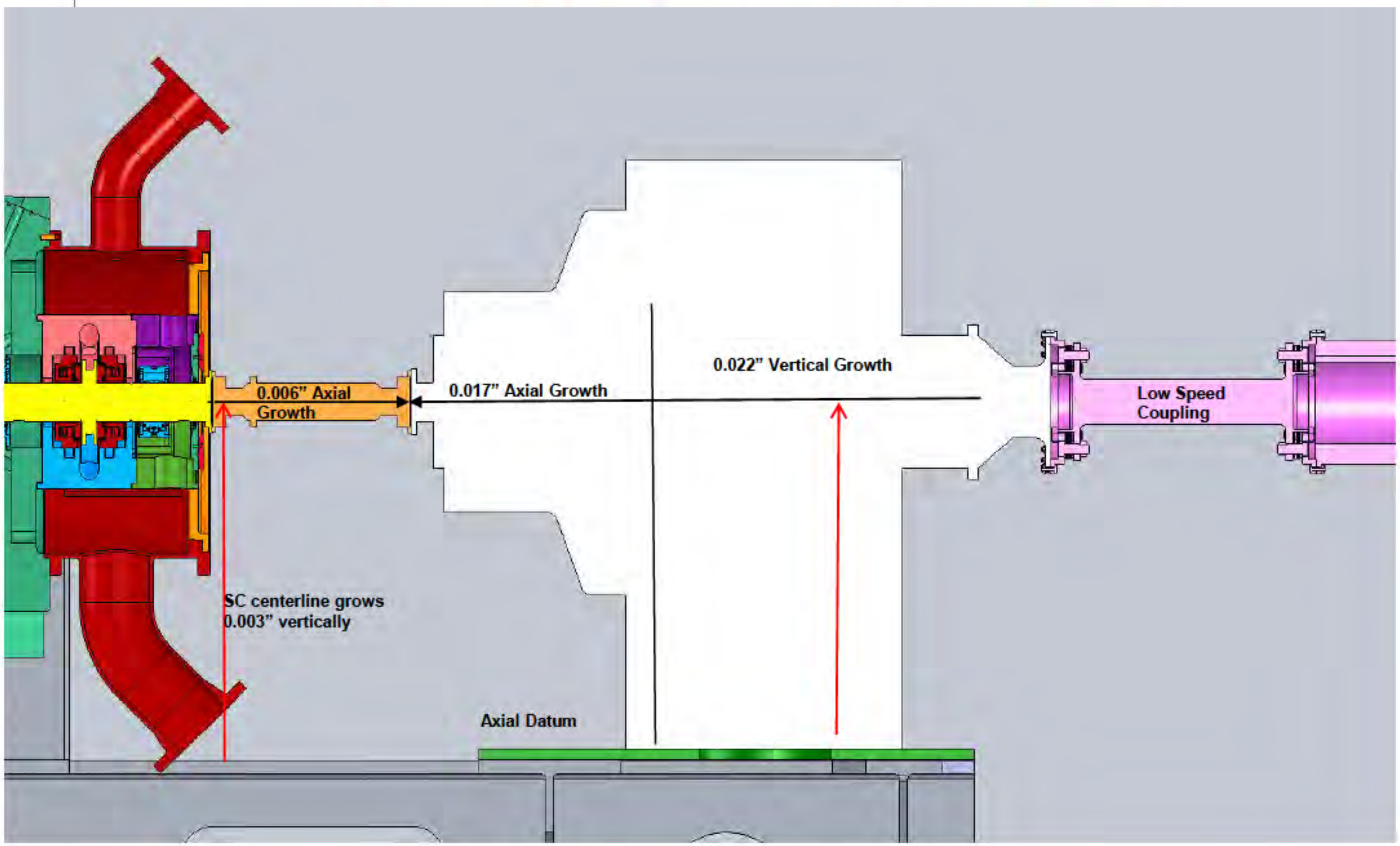
Supercompressor Conceptual Test Assembly-Option1b.SLDASM<-.007>

Baseline Configuration : Axial Stack



Ref Supercompressor Conceptual Test Assembly-Option1b.SLDASM

Baseline Configuration Thermal Growth



Ref Supercompressor Conceptual Test Assembly-Option1b.SLDASM

Baseline Configuration Thermal Growth

- Flex elements on coupling can withstand 0.040” under max continuous torque and misalignment values.
- Pre-stretch coupling elements so at design point flex elements are not stretched.
- Shim supercompressor so vertical misalignment is maximum at cold/start-up conditions.
- Flex element can handle 0.2° degree misalignment.

Gearbox Testing

<table>
<tr><th></th><th>Description</th><th>Allen Gears</th></tr>
<tr><td>16.1</td><td>Service Factor:</td><td>Service Factor confirmed as 1.6.</td></tr>
<tr><td>16.2</td><td>Predicted Bearing Temperatures:

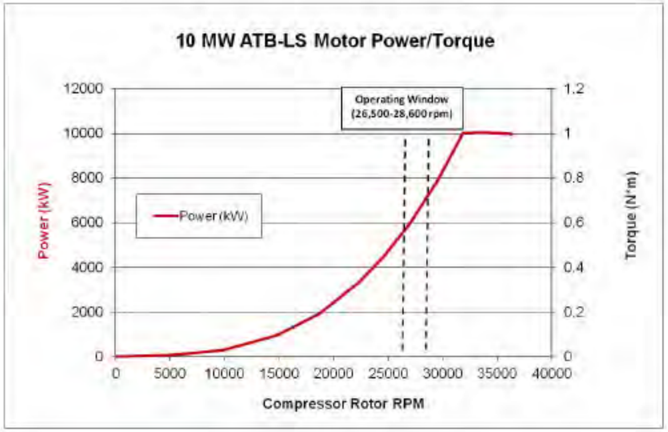
Bearing Temperature Alarm & Trip Settings:</td><td>
<table>
<tr><th>Pressure bar</th><th>Temp Deg C</th><th colspan="2">Speed 33000 (110%)</th><th colspan="2">Speed 30000 (100%)</th></tr>
<tr><td></td><td></td><td colspan="2">Bearing Clearance</td><td colspan="2">Bearing Clearance</td></tr>
<tr><td></td><td></td><td>Min</td><td>Max</td><td>Min</td><td>Max</td></tr>
<tr><td rowspan="2">1.38 (20 psi)</td><td>40</td><td>98</td><td>89</td><td>95</td><td>86</td></tr>
<tr><td>55</td><td>101</td><td>103</td><td>98</td><td>91</td></tr>
<tr><td rowspan="2">1.00 (15 psi)</td><td>40</td><td>98</td><td>89</td><td>95</td><td>86</td></tr>
<tr><td>55</td><td>101</td><td>104</td><td>99</td><td>91</td></tr>
</table>
Allen Gears will maintain AG standard bearing metal temperature alarm and trip limits (115°C & 120°C) for the No Load Gearbox Test. Once consistent temperatures at stable conditions have been recorded during the string tests, Allen Gears would advise lowering the Alarm & Trip limits to +10°C nominal temperature for Alarm, and ±15°C nominal for Trip.</td></tr>
<tr><td>16.3</td><td>Oil Inlet Pressure to Gearbox - Normal, Alarm & Trip values:</td><td>Allen Gears preference is as follows:-
Normal = 20 psi.
Alarm = 17 psi.
Trip = 15 psi
If more margin is required between the normal and alarm setting, AG could accept an increase in pressure to say 22 psi.</td></tr>
<tr><td>16.4</td><td>Allen Gears to add weight to HS Shaft end to simulate effects of HS Coupling, during No Load Gearbox Test:</td><td>Agreed - will be included in Test Procedure for Gearbox.</td></tr>
<tr><td>16.5</td><td>During No Load Gearbox Test - speed to be increased in 10% speed increments:</td><td>Agreed - will be included in Test Procedure for Gearbox.</td></tr>
<tr><td>16.6</td><td>During No Load Gearbox Test - Gearbox to undergo a 4 hour run, once oil inlet conditions have stabilised:</td><td>Agreed - will be included in Test Procedure for Gearbox.</td></tr>
<tr><td>16.7</td><td>Alarm & Trip settings:</td><td>See 16.2, 16.3 & 16.10 for expected Alarm & Trip values for No Load Gearbox Test. AG will perform Rotor Dynamics checks, as part of the design verification process.</td></tr>
<tr><td>16.8</td><td>AGQS410 to be sent to Ramgen:</td><td>Copy sent to Ramgen 14th May 2014.</td></tr>
<tr><td>16.9</td><td>Ramgen/Dresser-Rand require to witness the Gearbox Running Test</td><td>Agreed - will be included in Test Procedure for Gearbox. AG will provide 10 working days notice of scheduled Test Date.</td></tr>
<tr><td>16.10</td><td>Expected XY probe readings for HS and LS Shafts:

Gearcase Vibration:</td><td>XY Probes:
LSS - XY : No Load Test Limit = 1.8mils
HSS - XY: No Load Test Limit = 1.7mils - The HSS is only supported in a bearing at one end (gear mesh at the other end), and the API limit cannot be met. There is backlash in the Gear Mesh between the Sunwheel and the Compound Gears, which will allow a radial movement of 1.7mils at the probe position on the Sunwheel.

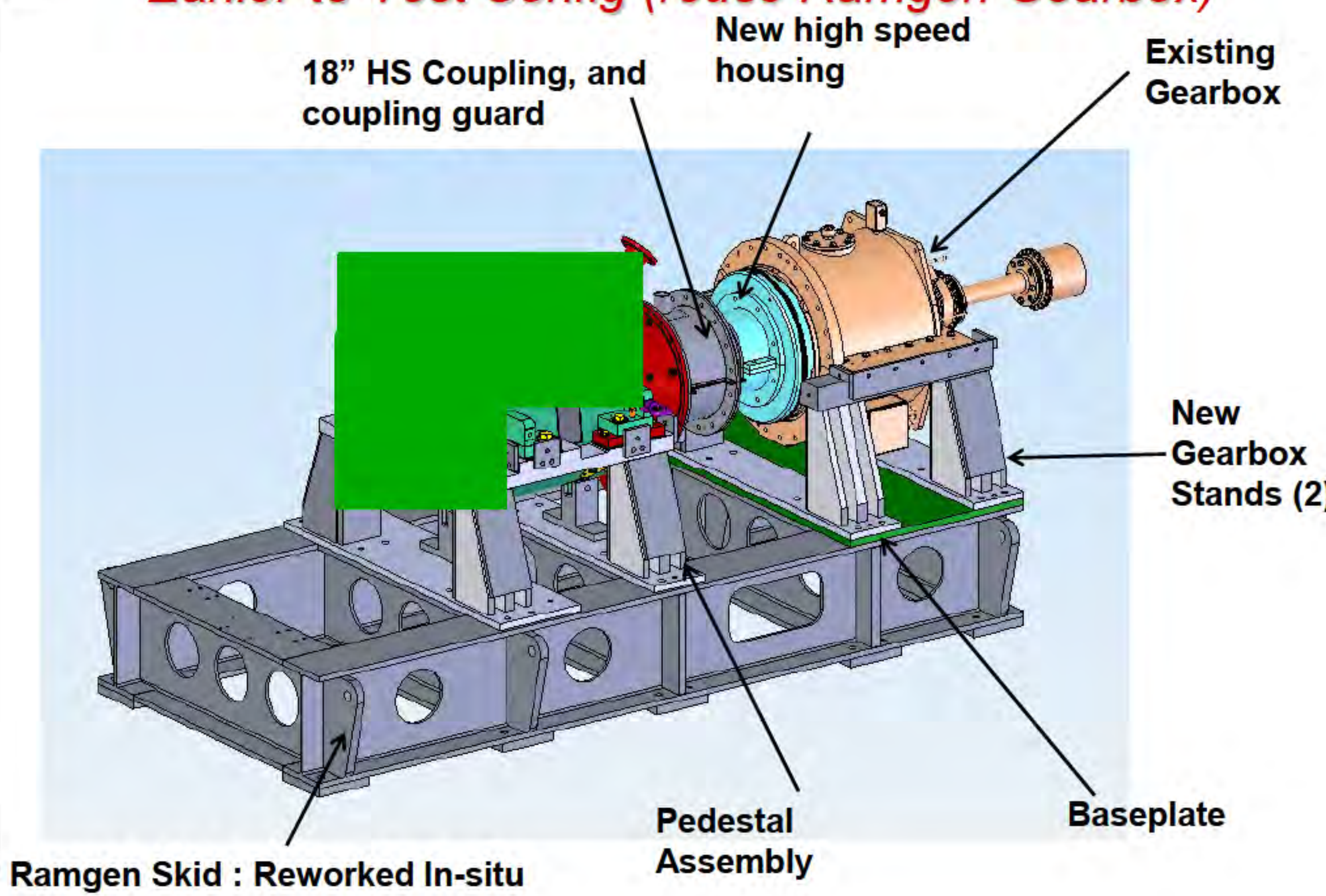
For this type of Gearbox Allen Gears experience has shown that a realistic value for overall casing vibration on No Load Test is 3mm/s.</td></tr>
</table>

"Early-to-Test" Configuration

- New Allen Gear lead time ~40 weeks.
  - Estimated delivery Mid-March 2015.
- Requires additional solution to meet November test window.
- Current gearbox has ~6MW available at supercompressor design speed,
- Use this configuration as an "aero" test to characterize performance of [redacted] and validate CFD design tools.
- Suppress suction pressure to reduce power requirement

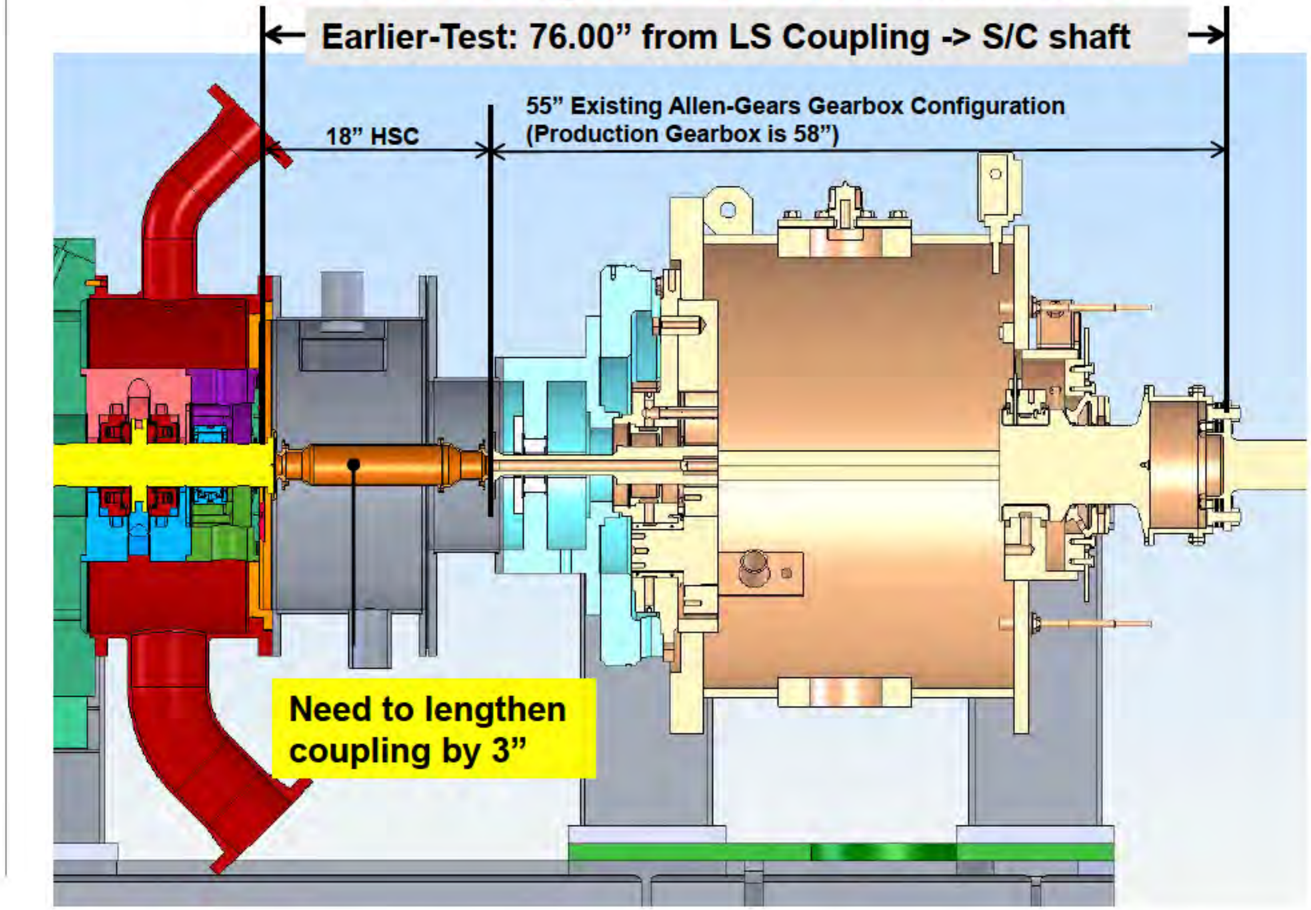


Earlier-to-Test Config (reuse Ramgen Gearbox)



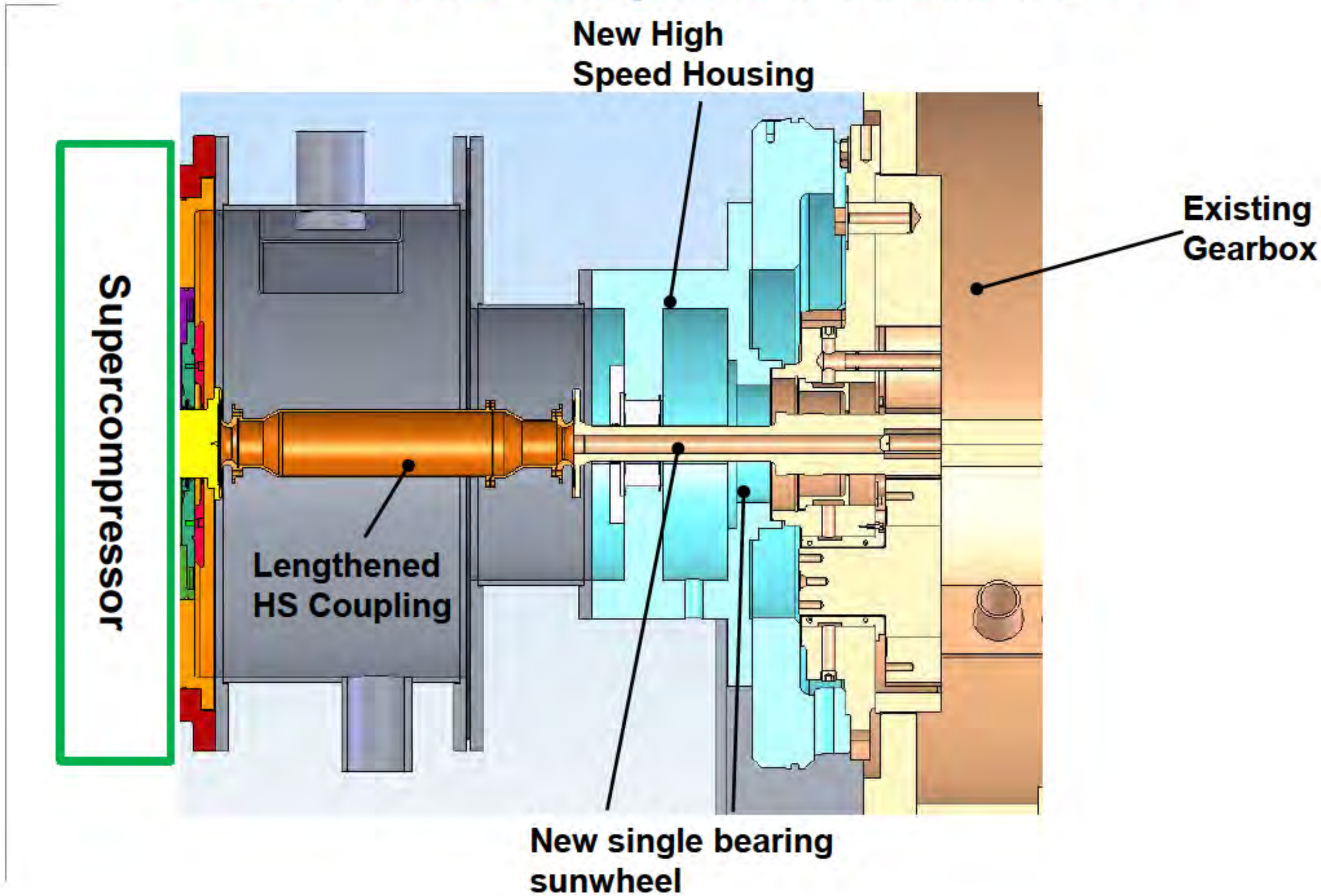
Ref: Supercompressor Conceptual Test Assembly-Option1c.SLDASM

Re-use Ramgen Gearbox: Axial Stack



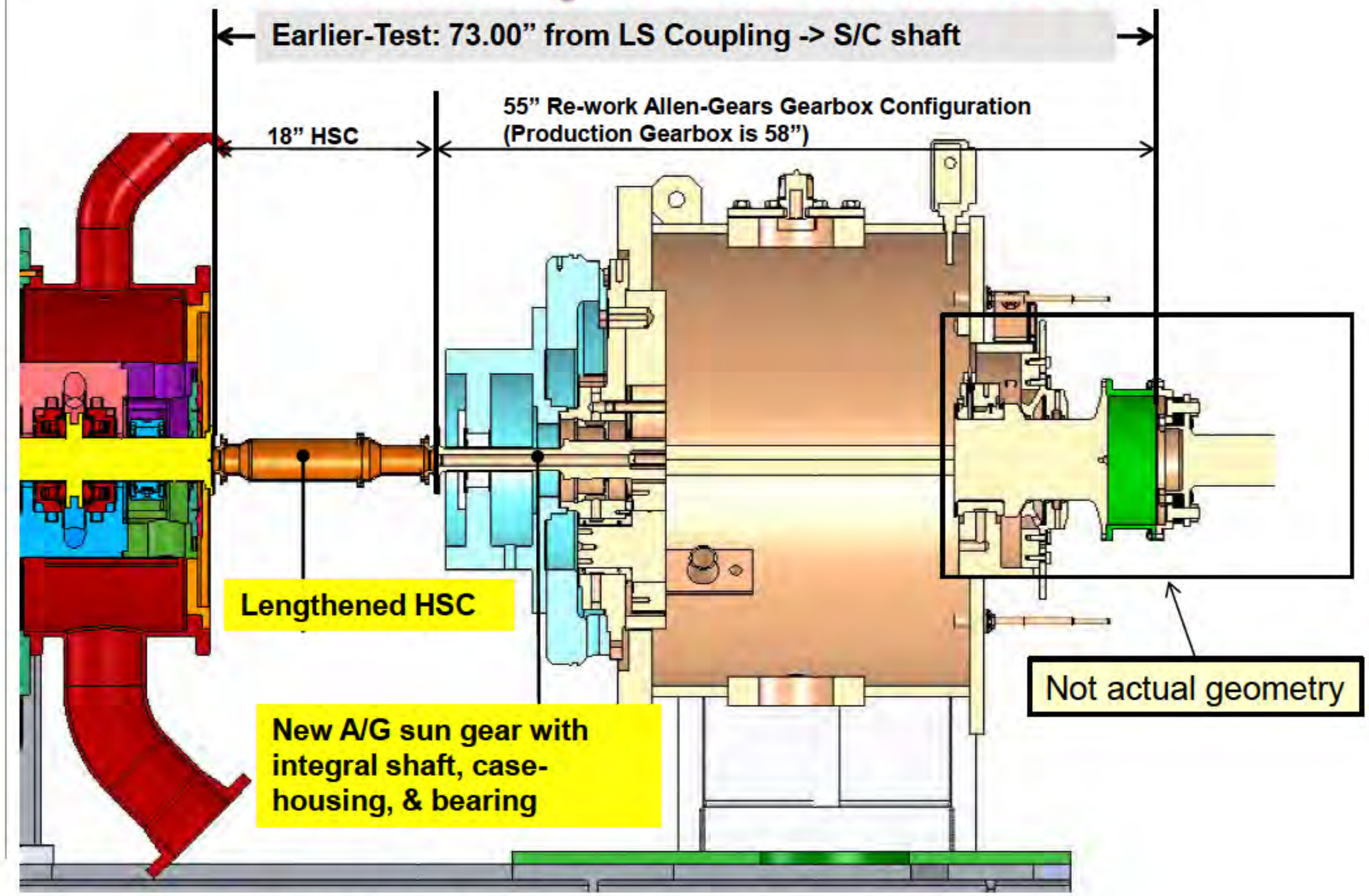
Ref: Supercompressor Conceptual Test Assembly-Option1c.SLDASM

Earlier-to-Test Configuration: Modifications



Ref: Supercompressor Conceptual Test Assembly-Option1c.SLDASM

Re-work Ramgen Gearbox: Axial Stack



Ref: Supercompressor Conceptual Test Assembly-Option1c-Aftermarket.SLDASM

Super Compressor Rotordynamic Evaluation

System Definition:

This system encompasses the Super Compressor drive train (all rotating components) for the early to test configuration

1. Super Compressor rotor
2. FWD high speed coupling
3. bearing shaft
4. AFT high speed coupling (existing)
5. Allen Gears gearbox (existing/reworked)
6. Low speed coupling (existing)
7. Laurence Scott motor (existing)

Components AFT of Super Compressor and FWD of motor subject to change based on final early to test configuration

Rotordynamic Evaluation Objective:

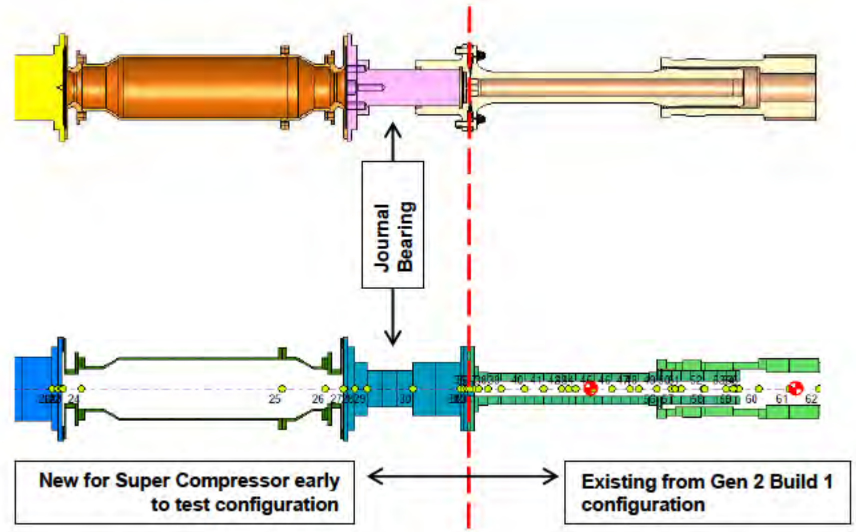
Demonstrate a rotordynamically stable Super Compressor early to test configuration, and identify any at risk components or operating conditions/speeds.

Approach:

Leverage existing analysis of the Ramgen Gen 2 Build 1 rotordynamics as a basis for stability or relative stability. This approach will only include the high speed operating components (Super Compressor thru sun gear). Acceptability will be based on a combination of API and D-R standards for the following:

1. Lateral response to imbalance, shaft bow, and bearing misalignment
2. Stability margin
3. Rotor torsional response (?)

Super Compressor Rotordynamic Geometry/BC's



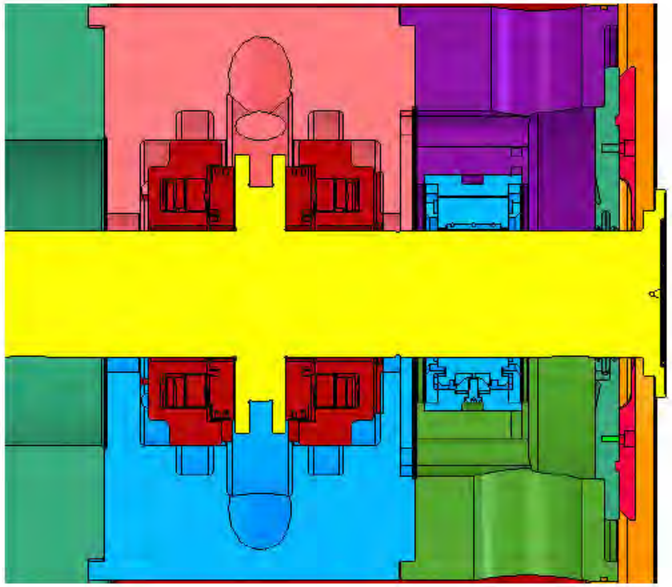
***Rotor train configuration subject to change

Software: DyRoBeS (continue with this software?)

<u>Geometry:</u>
Element L/D ratio < 0.5
Element layout to be reworked based on final early to test configuration

<u>Input BC's:</u>
1. **[redacted] aerodynamic cross coupling (need input from D-R)**
2. **Journal bearing characteristics (need input from DR on Supercompressor bearings)**
3. **High speed coupling interface characteristics**
4. **Aft HSC interface with sun gear**
5. **Establish imbalance, shaft bow, and misalignment magnitudes and locations**
6. **Axial and torsional connectivity between shaft segments**

Layout



Driveline Cost Summary

Driveline Costs

Item

New Gearbox

Gearbox Rework

High Speed Coupling

Lube Oil Skid Rework

High Speed Coupling Guard

Schedule

Driveline

Item	Need by order date
New Gearbox	13th June, 2014
Gearbox Rework	21st October, 2014
High Speed Coupling	27th June, 2014
Lube Oil Skid Rework	22nd August, 2014
High Speed Coupling Guard	-----

Items in red already purchased

[redacted]

Super Compressor Skid FDR

[redacted]

Kyle Badeau

[redacted]

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Skid System Definition

System Definition:

1. **Base weldment**
2. **Super Compressor support pedestals and keel blocks**
3. **Gearbox support pedestals**
4. **Gearbox baseplate**
5. **Skid to ground fastening**
6. **Super Compressor and gearbox to corresponding pedestals fastening**

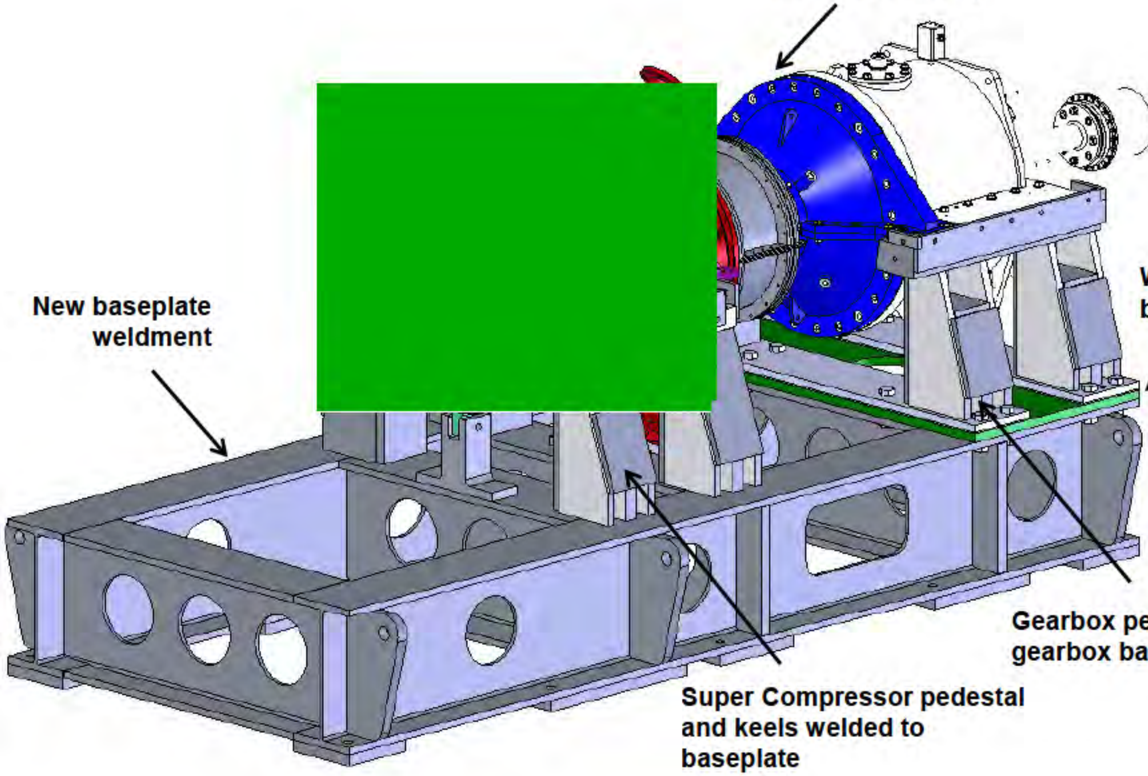
Rework of existing skid vs. new skid:

- **Rework skid estimated at $38k cost and 10 week lead time**
- **New skid estimated at +$10k and 12 week lead time (estimates to be refined with next release of drawings to vendors)**
- **Decision made to pursue new skid, based on similar cost/cycle and ability to customize skid to Super Compressor drivetrain**

Skid System Requirements

1. **Support Super Compressor and gearbox weight, torque, and piping loads**
2. **Position Super Compressor and gearbox in vertical, lateral, and axial directions**
3. **Allow for minor adjustments of Super Compressor and gearbox positioning during installation for drive train alignment**
4. **Allow for replacement of reworked (early to test) gearbox with new A.G. gearbox**
5. **Provide fastening/mounting/alignment features**
6. **Anchor into floor, maintaining as many of the existing anchor bolt locations as possible**
7. **Lifting requirements**

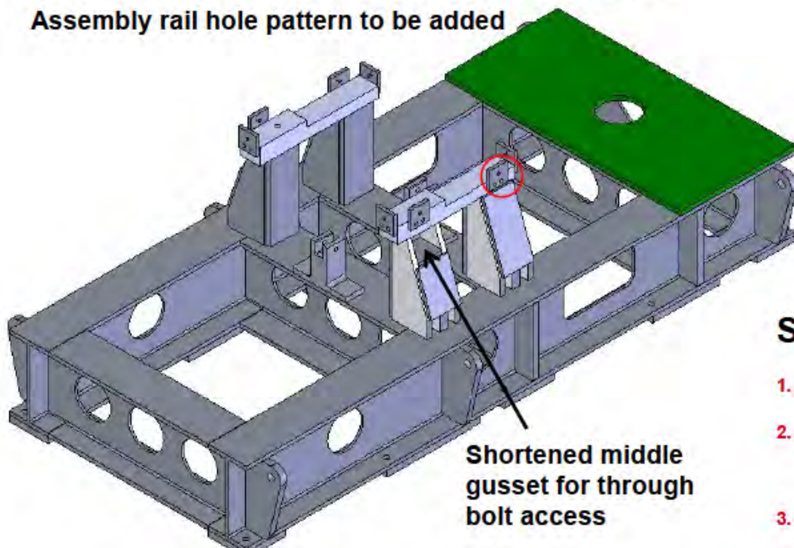
Current Configuration – Early to Test



Weldment Sub Assembly

Base:

1. Skid lengthened 3” for latest drive train stackup
2. Cross beams and stiffeners located axially underneath S.C. and gearbox mounting
3. Maintained position of 8/10 anchor bolts (foot pads lengthened accordingly)
4. Welded gearbox baseplate, fits new gearbox location
5. Assembly rail hole pattern to be added

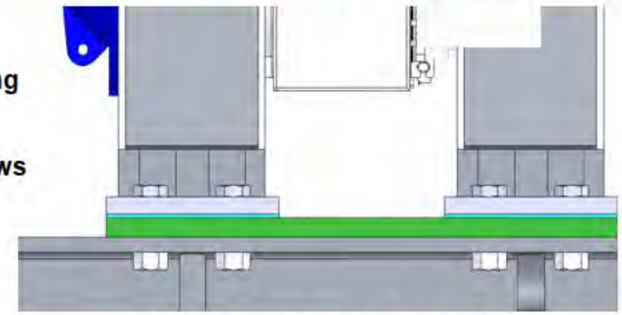


Super Compressor Pedestal

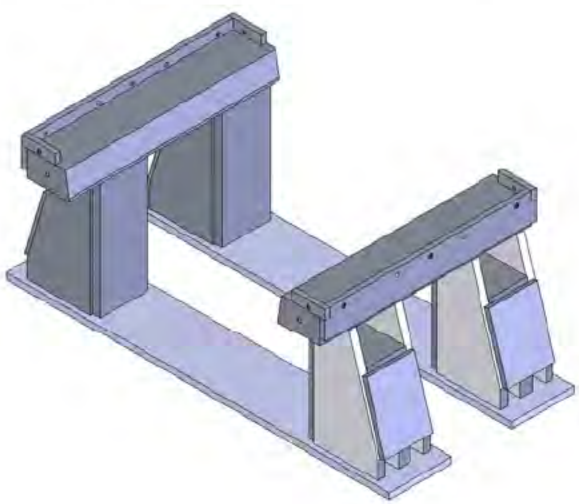
1. Welded to base
2. Pedestal legs axially located under S.C. mounting locations
3. Plates fastened to pedestal top piece for lateral & axial jacking screws (red)

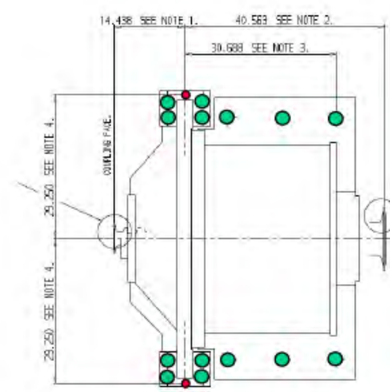
Gearbox Pedestal Sub Assembly

1. Bolted to gearbox base plate
2. Pedestal legs axially located under gearbox mounting locations
3. Plates welded to pedestal top piece for jacking screws
4. Shortened middle gusset for through bolt access
5. Top plate bulky (~550 lbs), can examine reduced volume design with FEA



¼" shims for vertical adjustment (blue)



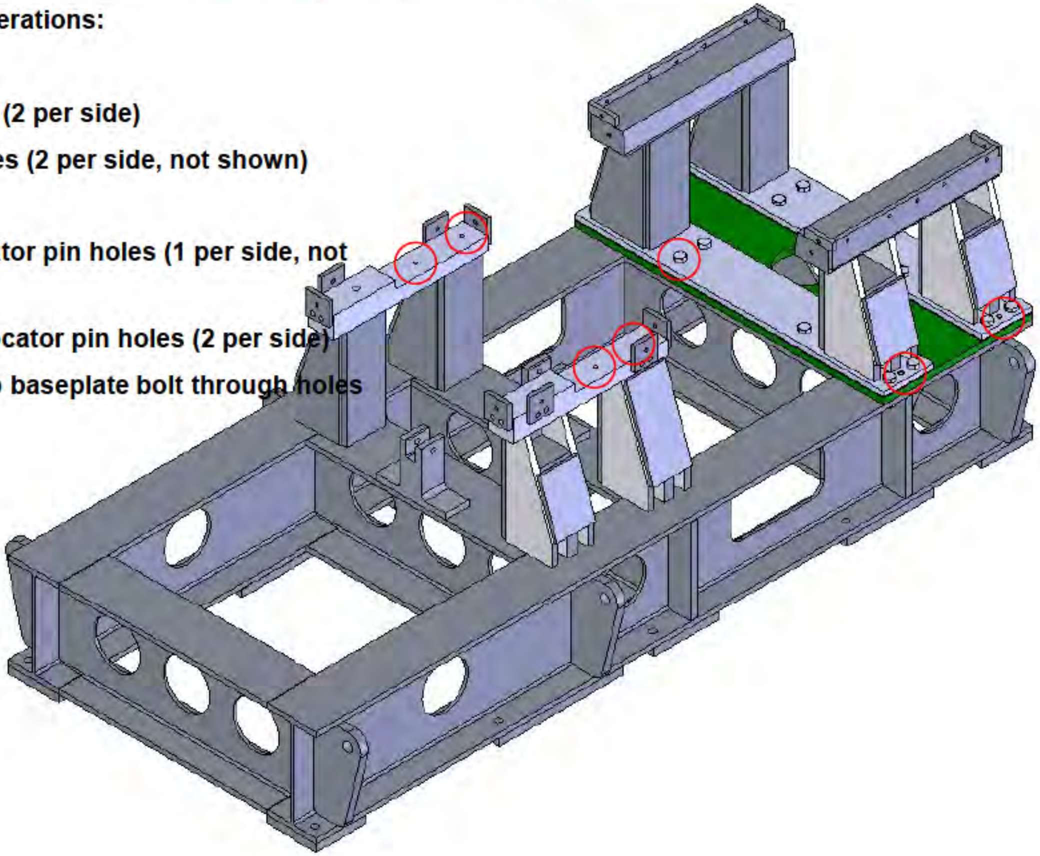


Through bolt hole pattern (green) and locator pin hole (red) to be added to skid top plate

Skid Top Level Assembly

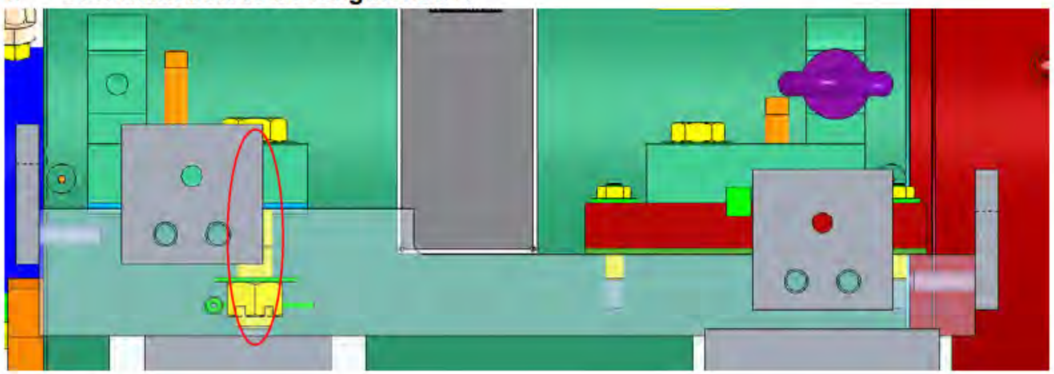
Top Level Machining Operations:

1. **S.C. threaded holes (2 per side)**
2. **S.C. locator pin holes (2 per side, not shown)**
3. **Gearbox flange locator pin holes (1 per side, not shown)**
4. **Gearbox pedestal locator pin holes (2 per side)**
5. **Gearbox pedestal to baseplate bolt through holes (8 per side)**

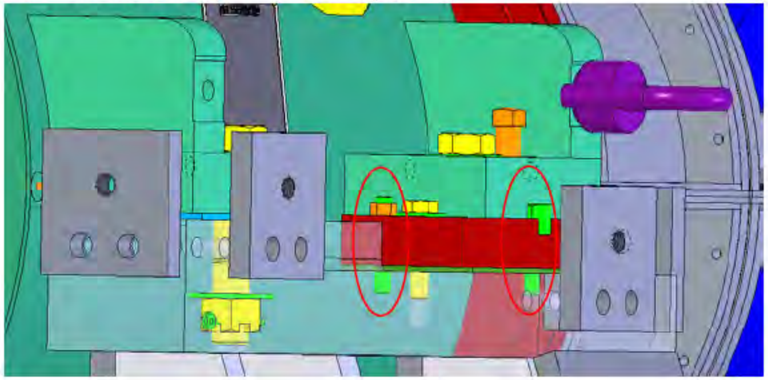


Skid Top Level Assembly

1. **S.C. main bolt to be lengthened 2"**



2. **Ream for tapered dowel pins**



Gearbox Pedestal Loads

Star Epicyclic Gear Foundation Loads

Contract: RAMGEN RE-WORK
Date: 08/05/2014 **Wk. No.**

Basic Data

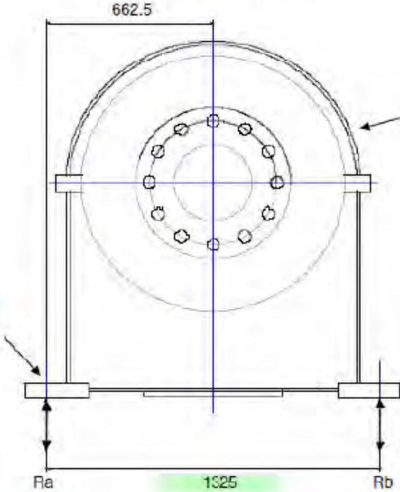
Power	kW
LS Speed	rpm
LS Torque	Nm
HS Speed	rpm
HS Torque	Nm
Total Torque	Nm
Rotation of LS Shaft	ACW
LS Shaft Connected To	Driver

Bolt Detail & Stress

Bolt Size	M24	mm
Bolt Root Dia	24.00	mm
Bolt Root Area	452	mm²
Max Tensile Load	58628.6	N
Tensile Stress	130	N/mm²
Proof Load Stress (Gr8.8)	571	N/mm²

Number of Bolts Each Side 4

View Looking On Low Speed Shaft



Total Gearbox Mass 2300 kg

Load Sense
Positive Vertically Down (+ ↓
Negative Vertically Up (- ↑

Foot Loads at Rb
Static Force (per side)
Static Force (per bolt)
Torque Force at FLT (per side)
Torque Force at FLT (per bolt)
Bolt Load at FLT
Bolt Load at 10 x FLT

Maximum Compressive Load (Positive is Down)

Total Load	35,861	N	8,061.58	lbf
Load Per Bolt (4 each)	8,965	N	2,015	lbf

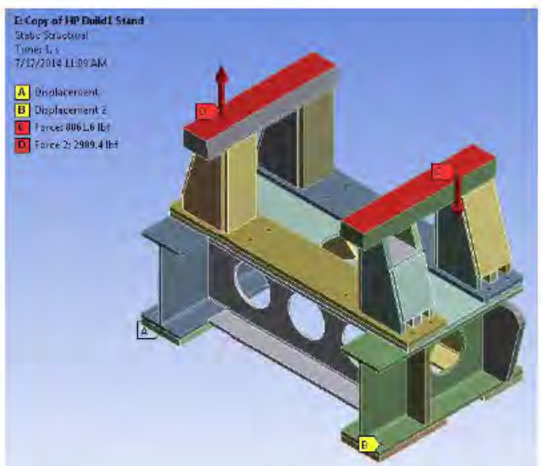
Maximum Tensile Load (Negative is Up)

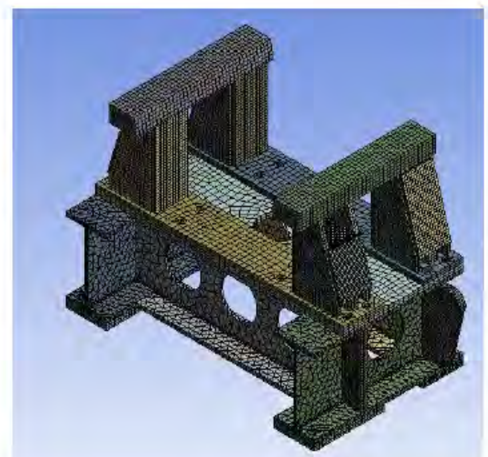
Total Load	13,298	N	-2,989.42	lbf
Load Per Bolt (4 each)	-3,325	N	-747	lbf

Gearbox Pedestal FEA

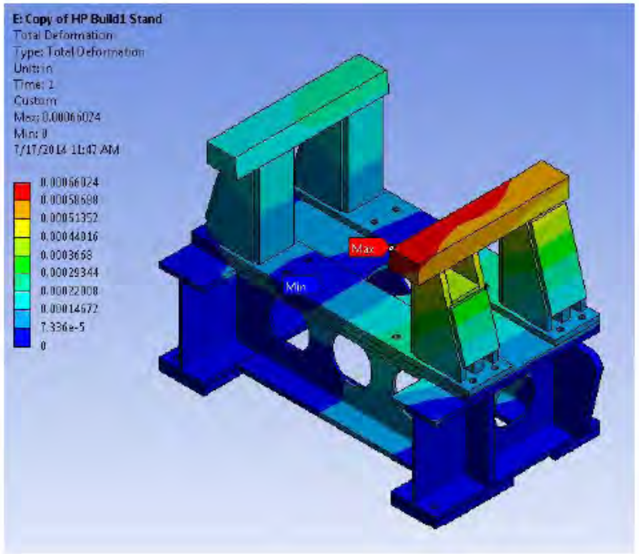
Boundary Conditions

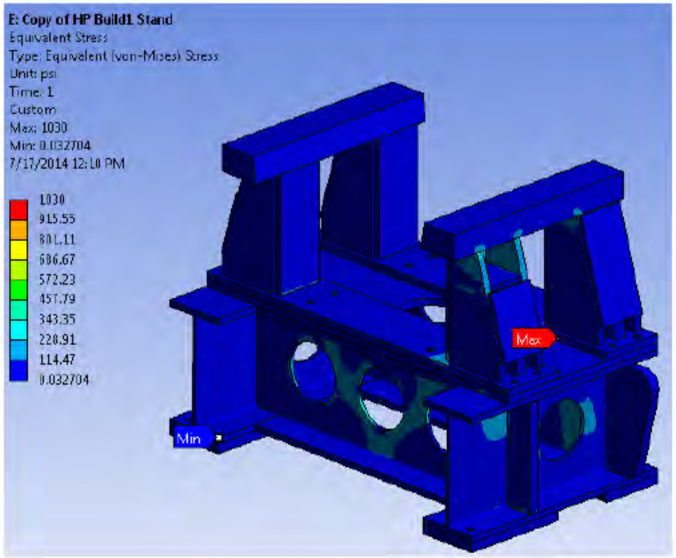
1. **All contacts bonded (bolted model does not converge yet)**
2. **Vertical fixed displacement at foot pads**
3. **Lateral and axial fixed displacement at anchor bolt holes**
4. **Pedestal torque & weight loads applied**





Gearbox Pedestal FEA



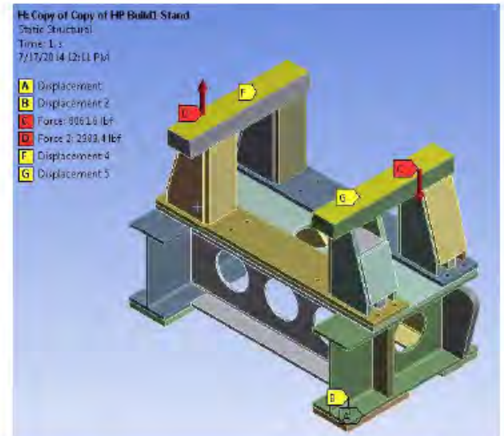


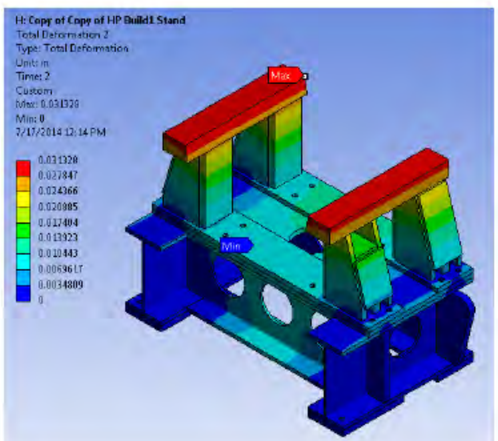
Less than 1mil of deflection
~1ksi SEQV

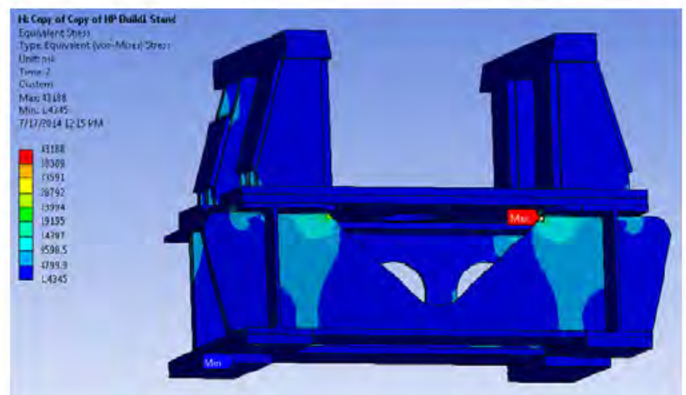
Gearbox Pedestal FEA

Boundary Conditions

1. All contacts bonded (bolted model does not converge yet)
2. Vertical fixed displacement at foot pads
3. Lateral and axial fixed displacement at anchor bolt holes
4. Pedestal torque & weight loads applied
5. Each pedestal displaced 30 mils laterally (estimated thermal growth displacement)







Work Remaining

1. **Complete mounting definition, ensuring features are provided at appropriate part/assembly levels**
2. **Attempt to solve assembly with gearbox pedestal bolt preload**
3. **Release baseplate weldment and gearbox pedestal weldment drawings**
4. **Obtain cost and cycle quotes**
5. **Release skid PO**
6. **Additional FDR Action Items**

[redacted]

Questions

[redacted]

[redacted]

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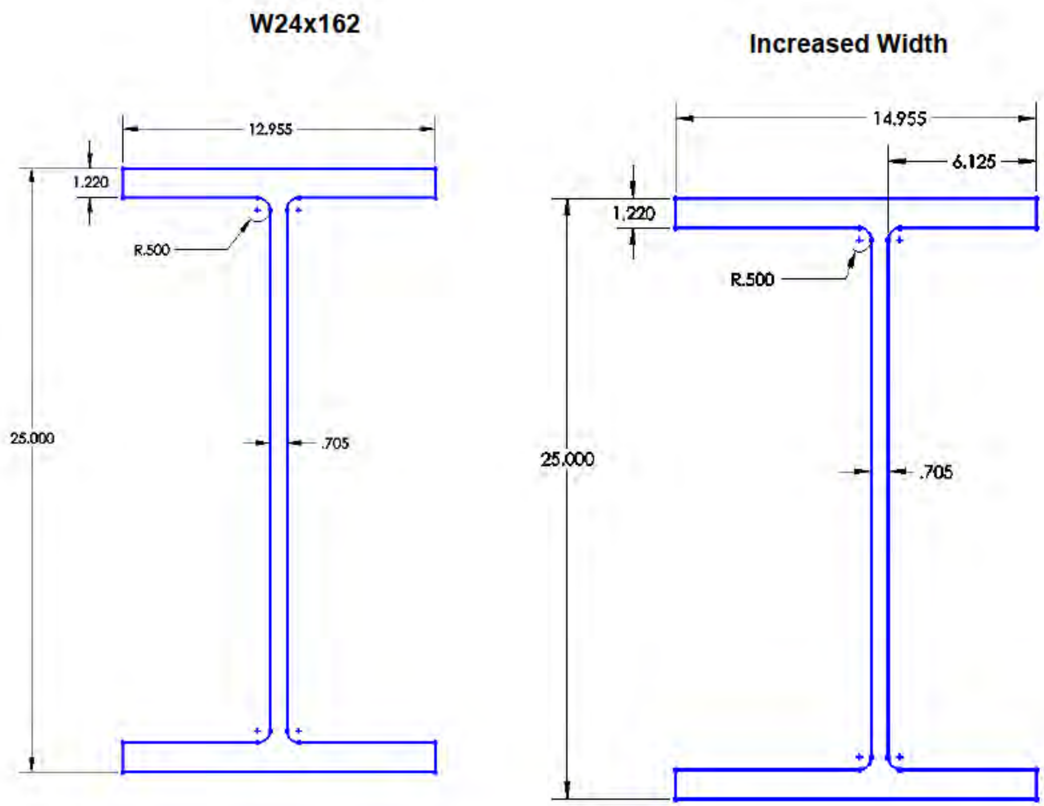
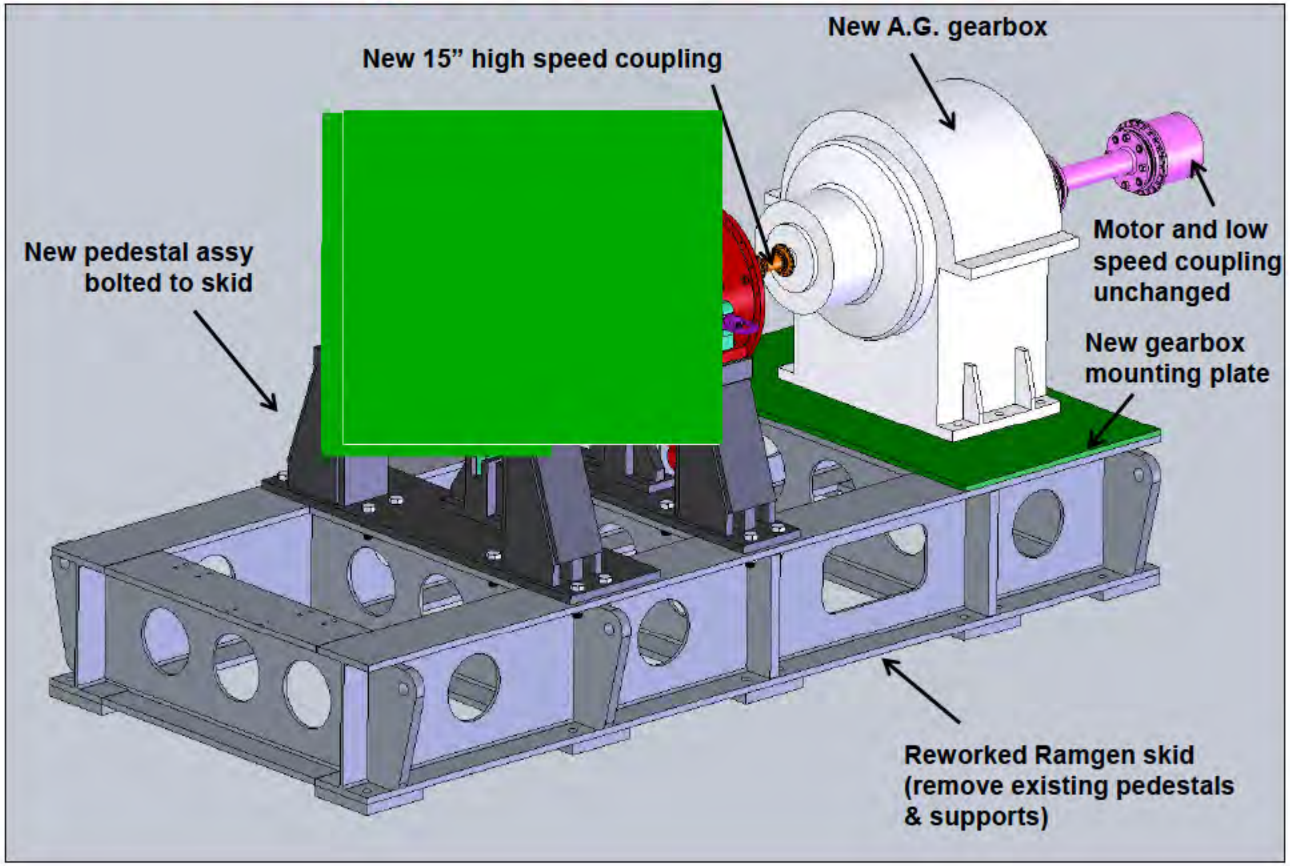
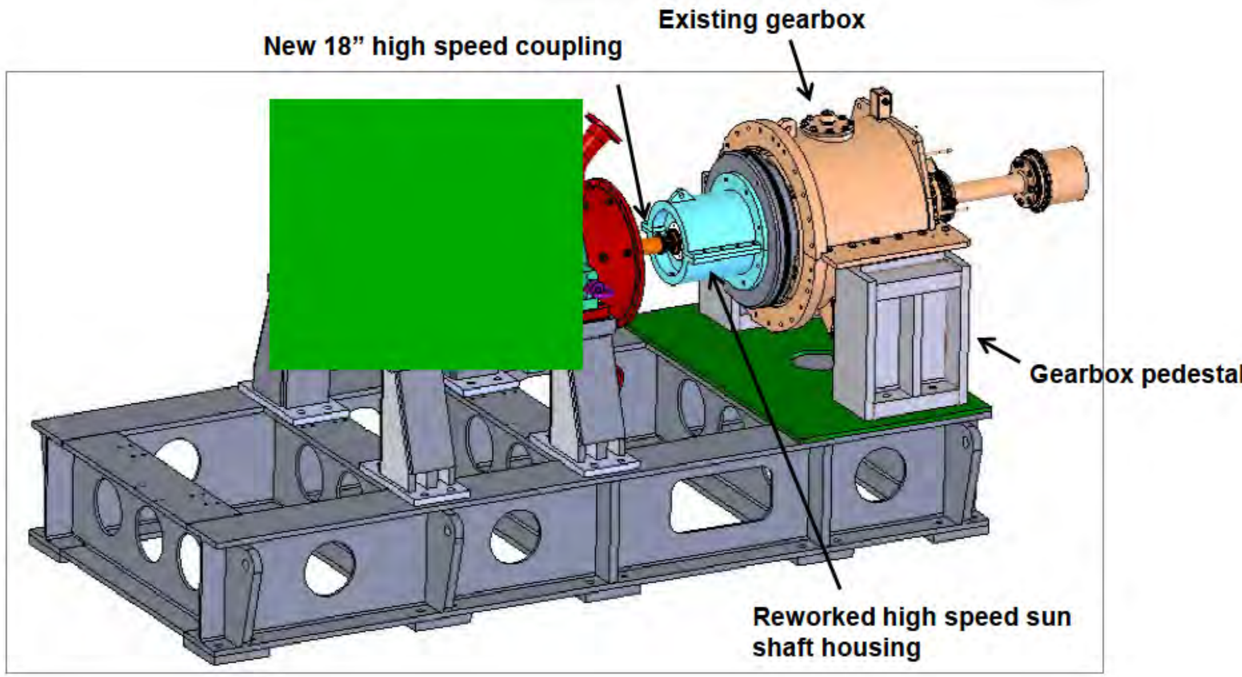
Cross Beam


W24x162
12.955
1.220
R.500
25.000
.705
Increased Width
14.955
6.125
1.220
R.500
25.000
.705

PDR Configuration - Baseline

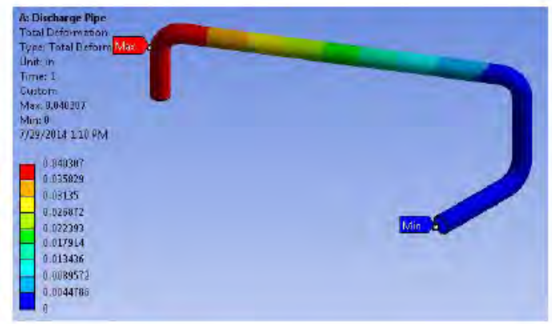


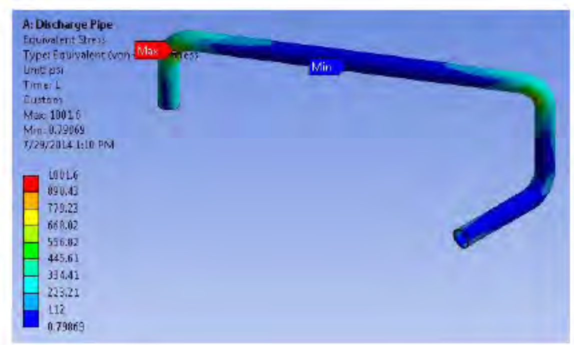
PDR Configuration – Early to Test



Compressor Discharge Pipe Loads

- 0.040” vertical displacement (compressor thermal growth) applied to inlet end of pipe
- Exit end of pipe fixed





Max discharge pipe stress ~1 ksi,
117 lbf to be reacted

Appendix 9.4.2
Facility FDR

Facility Modifications [redacted]

[redacted]

Mark Krzysztopik

[redacted]

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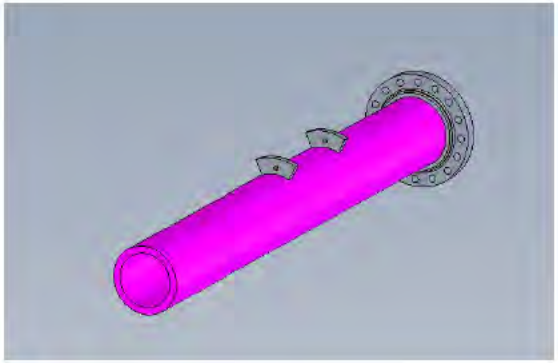
Systems Overview

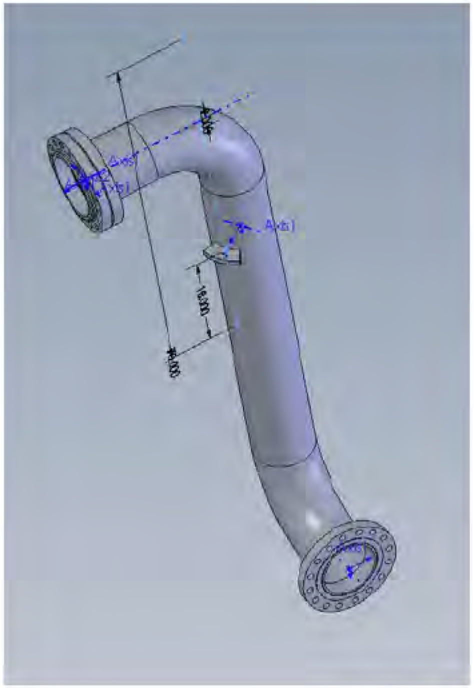
Facility

1. Modify existing closed loop CO2 system to **safely** and successfully run Supercompressor test program
2. Provide adequate lube oil flow to drivetrain components

PDR Action Items

1. Add lifting lugs on inlet pipe spool pieces for ease of disassembly



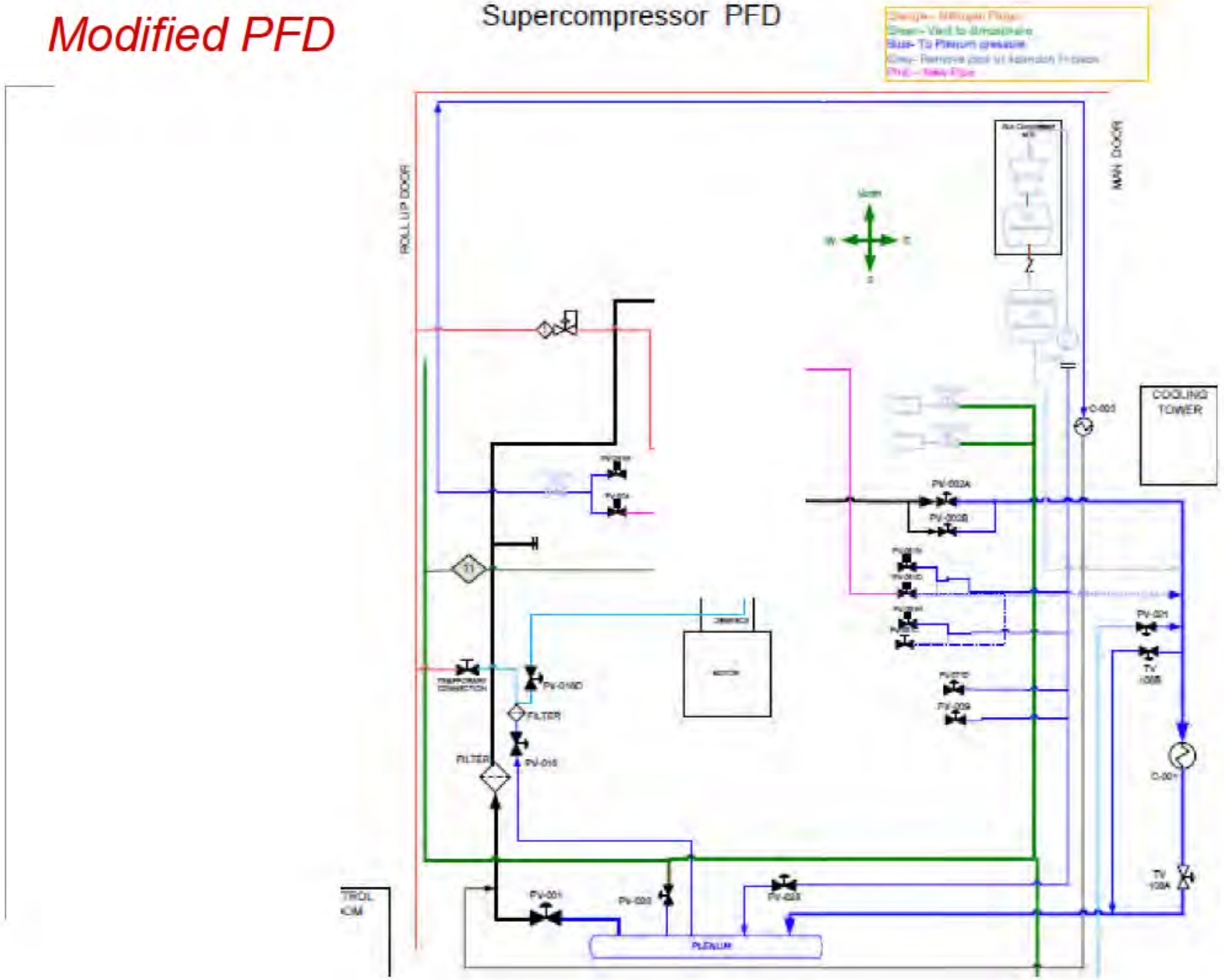


PDR Action Items

- 2. Determine whether a coating/electroless nickel plating is an acceptable alternative to using a 10' spool of 10" schedule 160 SS pipe
  - Coat of Epo-Phen FF Hi-Temp Coating
    - 7-9 mils thick
    - Up to 450°F
    - $1300 to coat spool piece
  - Electroless Nickel Plating
    - RFQ sent out 8/21/2014 for both inlet spool piece and existing dry gas seal filter
  - 10" Sch. 160 Stainless Steel pipe > $700 per foot

Modified PFD

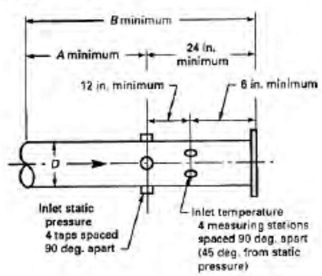
Supercompressor PFD



Inlet Piping Assembly

- IP1, IP2, IP3 all 10" schedule 40 stainless steel
- Abrupt transition to 10" schedule 160 carbon steel pipe between IP-3 and IP-4.
- IP-4 89" long

ASME PTC 10-1997

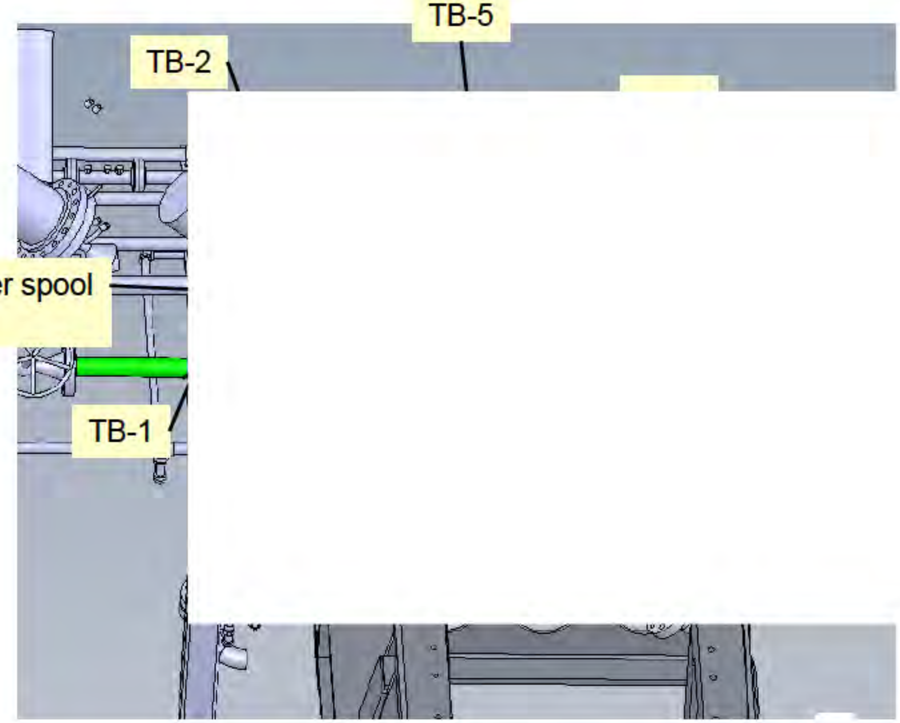


Inlet Opening Preceded By	Minimum Dimension A	Minimum Dimension B
Straight run	2D	3D
Elbow	2D	3D
Reducer	3D	5D
Valve	8D	10D
Flow device	3D	5D

For open inlet, see Fig. 4.2.

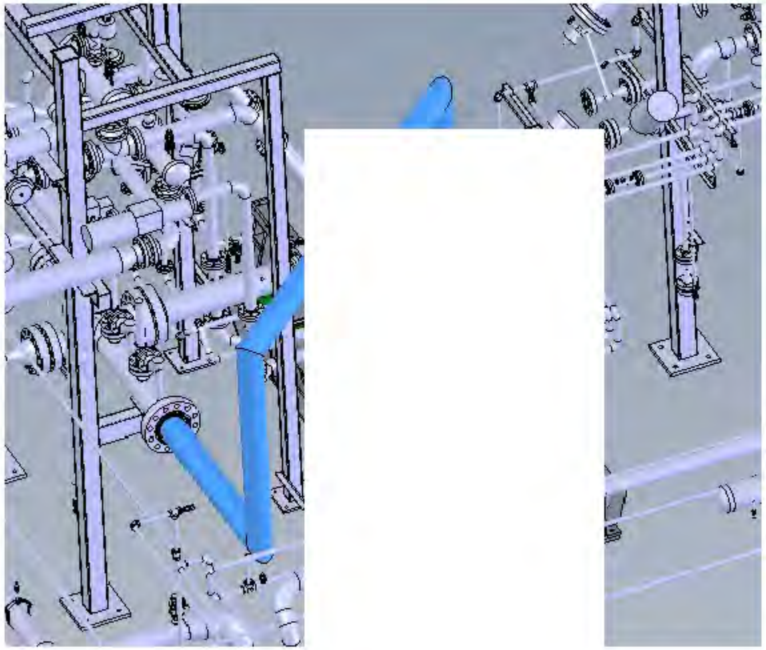
For vortex producing axial inlet, see Fig. 4.3.

Thrust Balance Piping



Discharge Piping

- Change in discharge pipe routing
- Mike Weimer has identified existing material from ICS acceptable for use as sections of discharge pipe
- Most flanges also identified



Other Gas Services

Service	Line Details
Seal Gas Reference/Thrust Balance Gage	1" Tubing
Seal Gas Supply	1.5" Tubing
Primary Seal Vent	1" Tubing
Separation Gas Injection	1" Tubing
4x Lube Oil Supply	1.5" Hose Connection

Flow Meter Calibration

- $19,000 total for all 4 in service cone flowmeters
  - ~$16,000 comes from inlet and discharge flowmeter
- CEESI rep mentioned measuring throat and inlet, if good agreement with cal sheets, we are *probably okay*.
- Onus is on us to determine how best to meet our own quality standards

[redacted]

Questions

[redacted]

[redacted]

[redacted]

Assembly Document

[redacted]

[redacted]

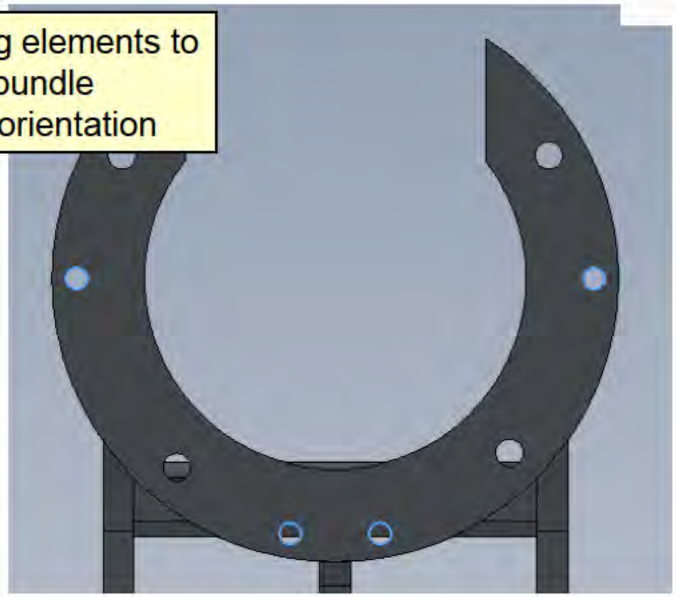
DRESSER-RAND

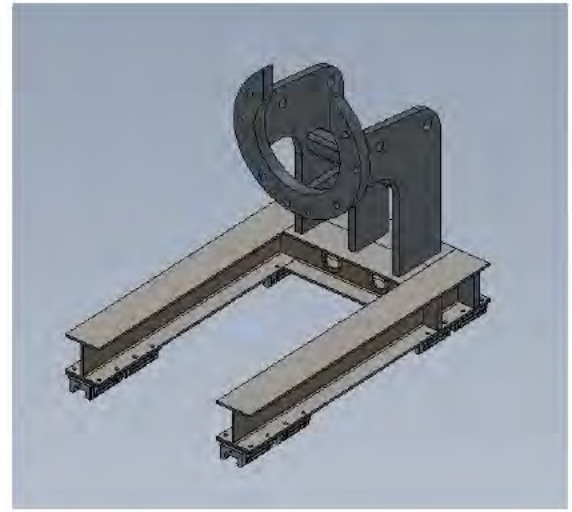
NDE Assembly

1. Dry Gas Seal
2. Diffuser Hub
3. assembly
4. Tie Bolt and [redacted]
5. **Remaining parts assembled as a single bundle**

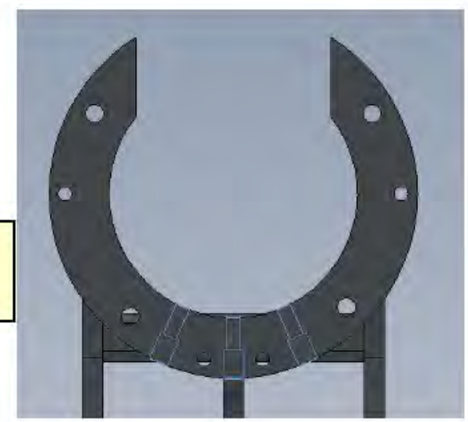
Carriage Assembly

Pitching elements to adjust bundle "nose" orientation

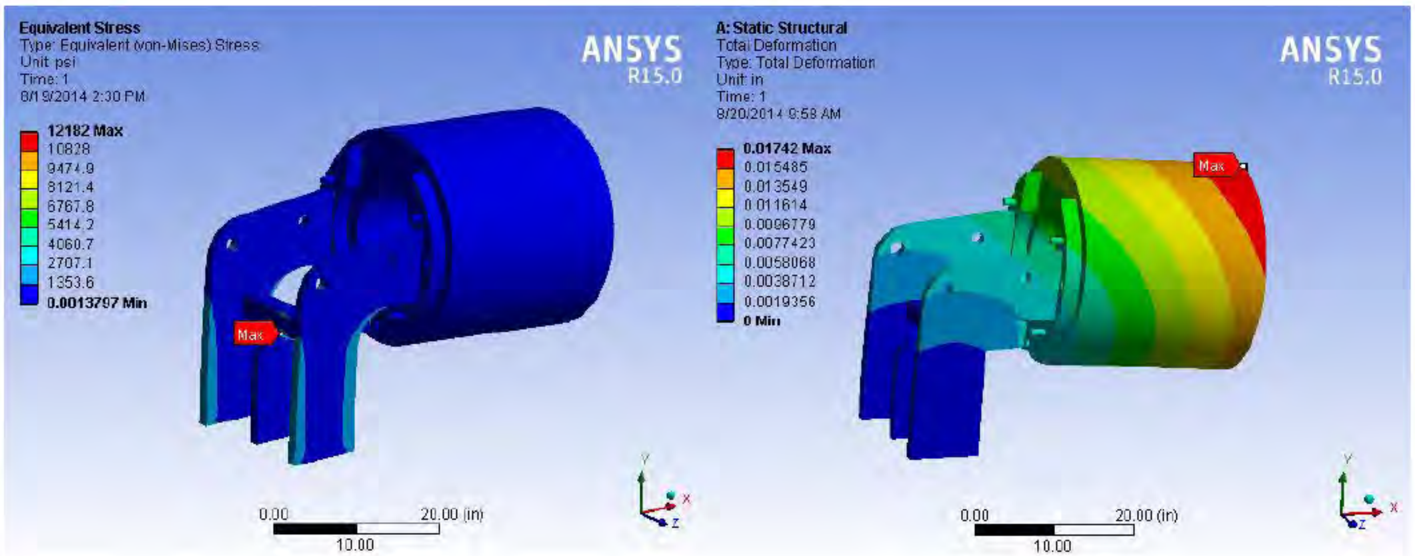




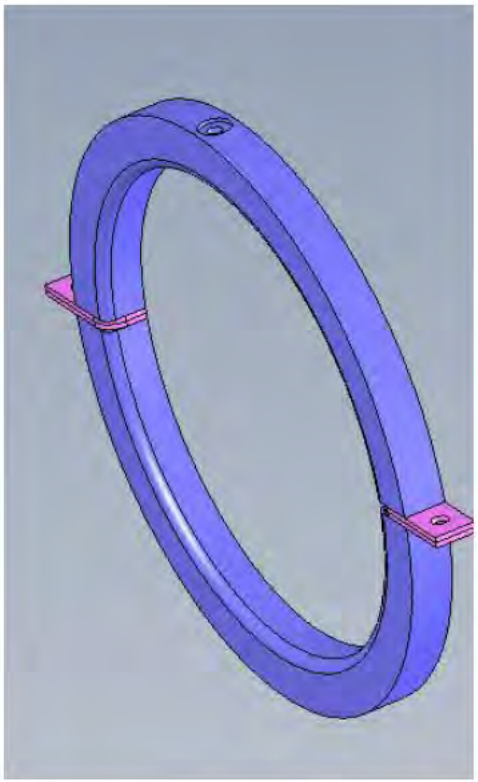
Radial Support Threads



Carriage Structural



[redacted] Clamshell

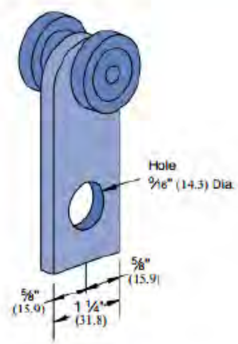


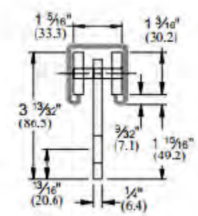
[redacted] *Installation*

TROLLEY ASSEMBLIES
FOR 1⅝" (41 MM) WIDTH SERIES CHANNEL

P2949

TROLLEY ASSEMBLY



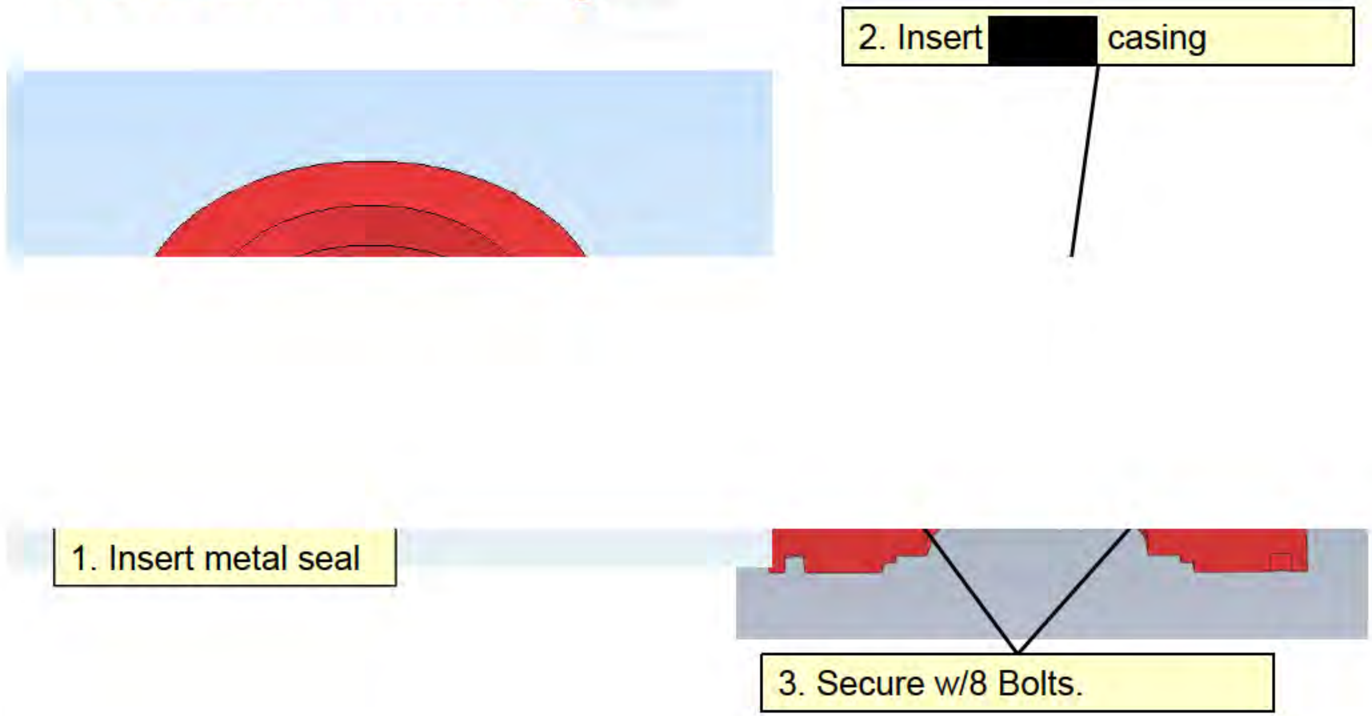


RPM	Design Load In P1000	
	Lbs	kN
600	150	.7
300	225	1.0
100	437	1.9

Wheel bearings are stainless steel. Do not lubricate.

Wt/C 46 Lbs (20.9 kg)

Volute Assembly



Stationary [redacted] Assembly

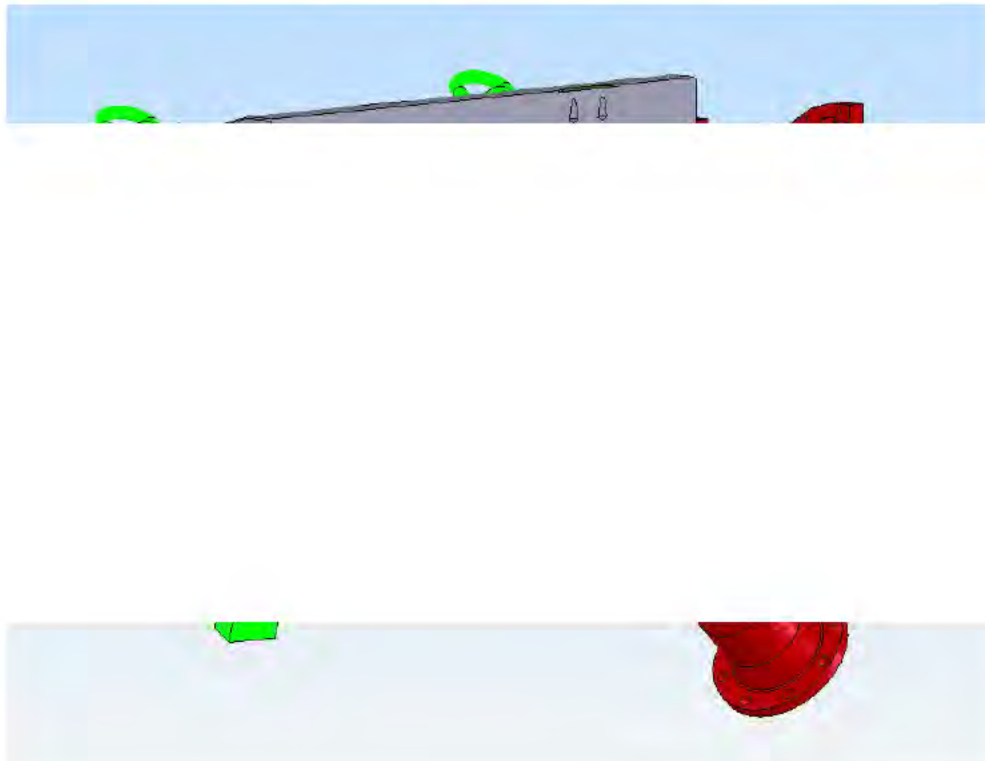
- Still unclear how abradable [redacted] will connect to [redacted] casing
  - Quick removal of [redacted] abradable an assembly requirement

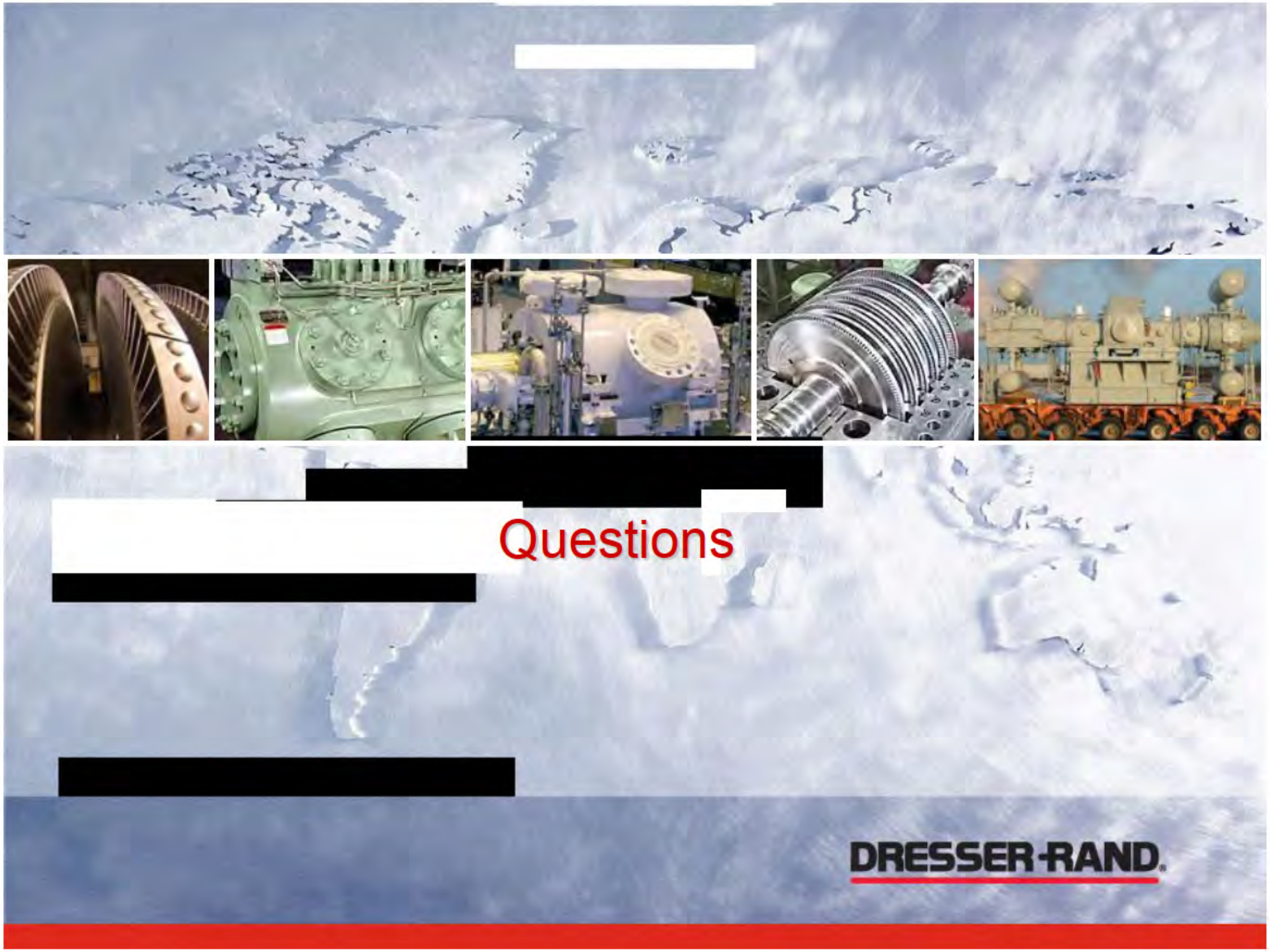
Diffuser Wall to Volute

- supported on installation table, volute assembly dropped atop
  - Lifting features need to be added to NDE face

Qty. 11 bolts to secure volute assembly to diffuser wall

Final installation




[redacted]
Questions
[redacted]
[redacted]
DRESSER-RAND

Test Plan Stages. SC Build 1

1. Assembly – 4 weeks
2. Installation – 2 weeks
3. Debug, Rotordynamics and Controls Checkout – 3 weeks
4. Compressor Map – 1 weeks
5. Turndown Characterization – 1 weeks (if applicable)
6. New Vaned Radial Diffuser – 3 Weeks (including reassembly)

Items 4-5 repeated for rebuilds of diffuser/[redacted] Rotordynamics checkout will need to be repeated once new gearbox is installed

Testing Constraints

- Eight minute minimum acceleration time (0-
- Rotordynamic and Modal Stayout Zones
  - UPDATE
- Intermediate plenum pressure: min 40 psia > suction pressure
  - Provide adequate flow through dry gas seal
- Estimated max inlet suction pressure
  - Pt = 125psia

Assembly

Purpose:

Resolve assembly issues and then assemble the components of the compressor to create a complete assembly and install the required instrumentation

Approach:

A team of test engineers will assemble the DATUM-S compressor. During the assembly process instrumentation wiring and tubing will also be installed. Alignment of skid components will be done at this time.

Duration: 4 week

Installation

Purpose:

Install the assembled compressor into the closed loop facility and connect the required instrumentation and control lines to the data acquisition and control systems.

Approach:

The local test team in Olean will assemble the DATUM-S compressor. During the assembly process instrumentation wiring and tubing will also be installed.

Duration: 2 week

Debug, Checkout, Prerun

Purpose:

Spin the rotor to MCOS at low-suction conditions to characterize rotordynamic performance. Checkout modifications to controls system

Approach:

Increase rotor speed in 25% increments (adhering to all modal and rotordynamic stay-out zones) to warm-up gearbox and verify proper operation. Use warm-up time to tune suction control valves and verify operation of CO2 make-up system. Gather aerodynamic data for preliminary review. Determine maximum suction pressure for compressor map testing due to drivetrain power limitations.

Duration: 3 weeks

Compressor Map

Purpose:

Create a compressor performance map by generating numerous speed lines.

Inlet Pt = maximum allowable determined during prerun.

Inlet Tt = 100°F

Approach:

1. Take a full speed line of data at design point operation Minimum of 8 data points from choke to stall.
2. Take a full speed line of data at MCOS . Minimum of 8 data points from choke to stall.
3. Take a full speed line of data at 95% design speed Minimum of 8 data points from choke to stall

Duration: 1 week

Appendix 10.1

ISCE Diffuser

Geminus Technology Development

ISCE Program: Inducer-Diffuser Simulations

John Hinkey

June 22, 2011

NACA RM-L56J01 Stage Simulations

- **Goals:**
  - Simulate The Geometry and Operating Conditions Shown In NACA RM-LM56J01
  - Determine Where Their Design Got It Right and/or Wrong
- **Objectives:**
  - Perform Mean Line/2D Simulation To See The Mean Line Design (Quick)
    - + *Specifically The Impulse Blade Design: How Well The MOC Did*
  - Generate Their Stator and Rotor Geometries Based On Coordinates Called Out In The NACA Report
  - Perform On-Design Speed Simulations Using Ideal FREON As the Working Gas
    - + *Compare Results To Experiment Measurements*
  - Perform On-Design Speed Simulations Using Ideal AIR As The Working Gas
    - + *Determine Any Differences/Similarities and Lessons To Be Learned For ISCE Impulse Inducer Design*

NACA RM-L56J01: Test Apparatus & Geometry

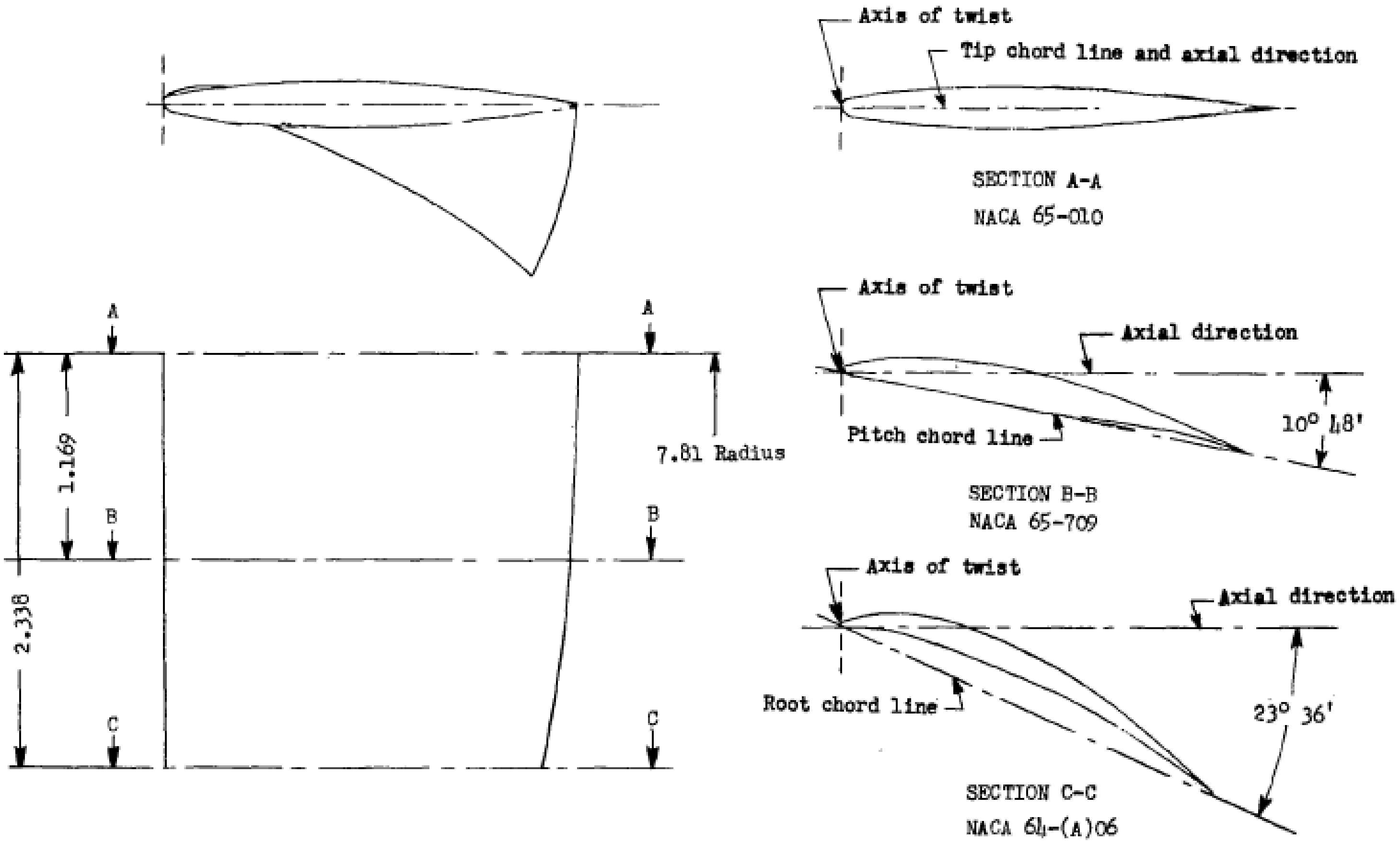
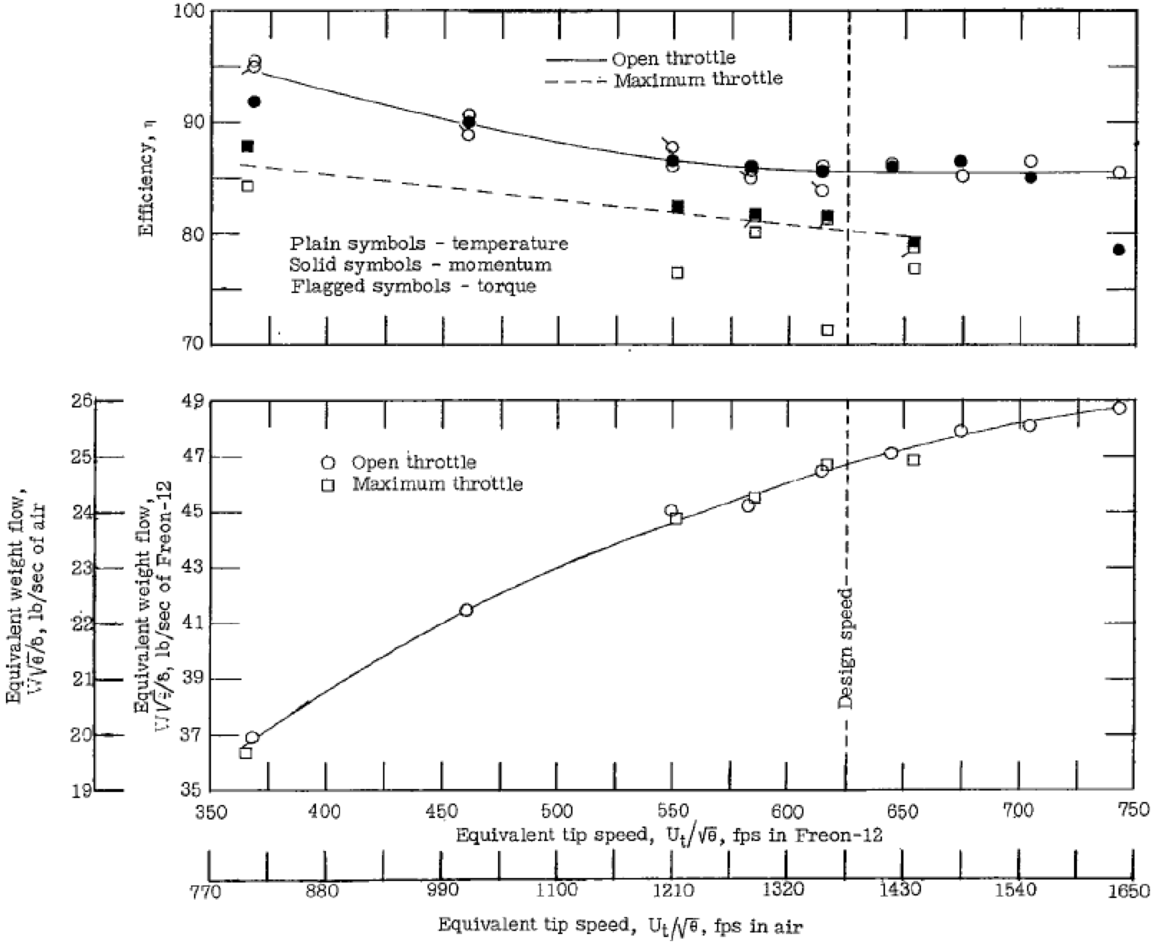


Figure 7.- Guide-vane details. All dimensions are in inches.

NACA RM-L56J01: Operating Conditions



- **Designed For Air (γ = 1.40), But Operated On Freon (γ =1.125)**

Design #1 3D CFD: Conclusions

- **Flow Looks Pretty Decent Overall**
  - No Hub Separation As Occurred In NACA Rotor
  - Decent Performance
- **0.007” Tip Gap Does Not Degrade Performance That Much**
  - May Want To Run An Equivalent Shrouded Simulation To Check This

1-19-2011 Update

- **Many Many Simulations Performed**
  - 88 Blade Model w/6mil radius LE
    - + *10 and 12 psia back pressure*
  - 44 Blade Model w/6mil rad. LE
    - + *100%, 105%, 110%*
  - 44 Blade Model w/12mil rad. LE
    - + *100%, 105%, 110%*
  - 80 Blade Model w/6mil LE
    - + *10 psia back pressure*
- **Really Really Need Automated Post-Processing**
  - Huge backlog – Mark working on scripts
  - Network MATLAB Licenses ASAP

What's Next For This Design?

- **Trying To Get A Stable Un-Back Pressured Solution**
  - May need to back pressure it a little bit
- **Installation of Preliminary Bleed Patches**
  - Hub & Shroud
  - Via Built-In Bleed Hole Modeling Capabilities
  - Low and High Back Pressure
- **Slip Wall Simulation**
  - Check To See If Inviscid Shock Structure Is OK
  - Grid In Progress
- **Other Issues Hindering Things**
  - Auto Splitting Version of FINE/Turbo and/or IGG Needed To Turn Solutions Around Faster
    - + *Currently Grid Topology Only Allows Use of 4 Processors*
    - + *Current IGG-based Splitting Does Not Maintain Periodic Connections*

April 08, 2011 Update

- **Back-Pressured Non-Slip Wall Solution Results**
- **Back-Pressured Slip Wall Results**
  - Viscous Inflow Profile Used (see previous slides)

Ramp Design #1: Back-Pressured Results

- **Fully Viscous Model Back Pressured To 45 psia (mass averaged)**
  – Could not get a low back pressure simulation to converge: Outflow Problems

Appendix 10.2

ISCE CDR

ISCE Phase I CDR

Compressor Inducer Rotor

April 15, 2011

Outline

- Inducer blade geometry
- Rotor geometry
- Blade alone modal analysis
- Campbell Diagram
- Thermal Analysis
- Stress Analysis

Appendix 10.3

ISC Engine Presentation

ISC Engine Configuration Advances

July 13, 2011

Ramgen Power Systems

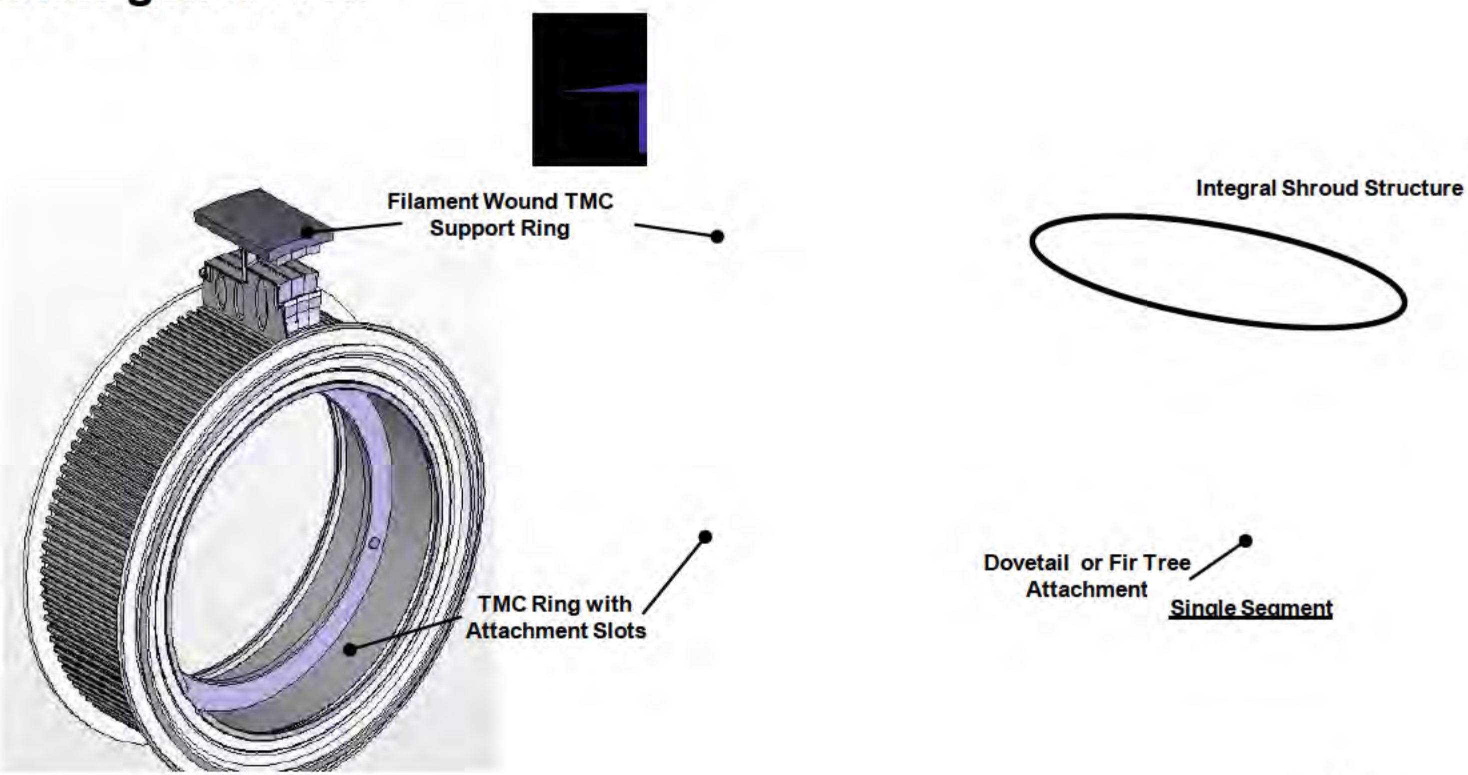
0900-01388

Agenda

- Introductions
- Technology Advances
- Performance potential
- Advances that enable risk reduction
- Program schedule acceleration
- Budget impact

Technology Advances – Initial Proposal

- Initial proposal focused on ring rotor based shock compression and power generation



Program Technical Risk Reduction

- Program risk and cost reduction achieved by utilizing existing IGT combustor and turbine – avoids design and check-out of these proven components
- Accelerates demonstration and validation of compressor performance, "Gen-2" compressor will be retrofit onto existing IGT with power output consistent with testing at Ramgen facility in Redmond
- Saturn T1200 selected – PR=6:1, Airflow = 13 pps, Power Output = 1,141 hp (850 kW)
- Mechanical design/fabrication proceeding with goal of having hybrid unit available for demonstration in October time frame

Performance Potential New ISCE Configuration

- "Gen-2" compressor configuration (supersonic Inducer + Stationary Shock Diffuser) Capable of Increased Compression Ratios Compared to Original Rotor Based System

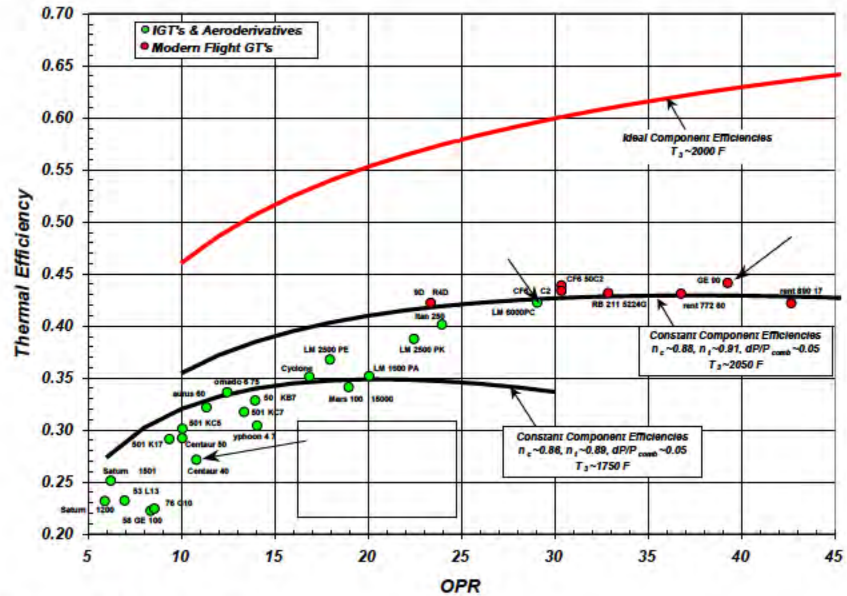
- Preliminary Analysis Indicates PR~20:1 Possible with Conventional Shrouded Inducer in 1,500 kW System

- Potential Performance for 1,500 kW at PR~20:1 with Proven AVC Combustor Characteristics in mid 40% thermal efficiency Range

- System Retains All Capabilities Originally Proposed
  - Rapid Load Following
  - Ability to Burn Low Pressure Dilute Fuels

Dramatic Range of Applications

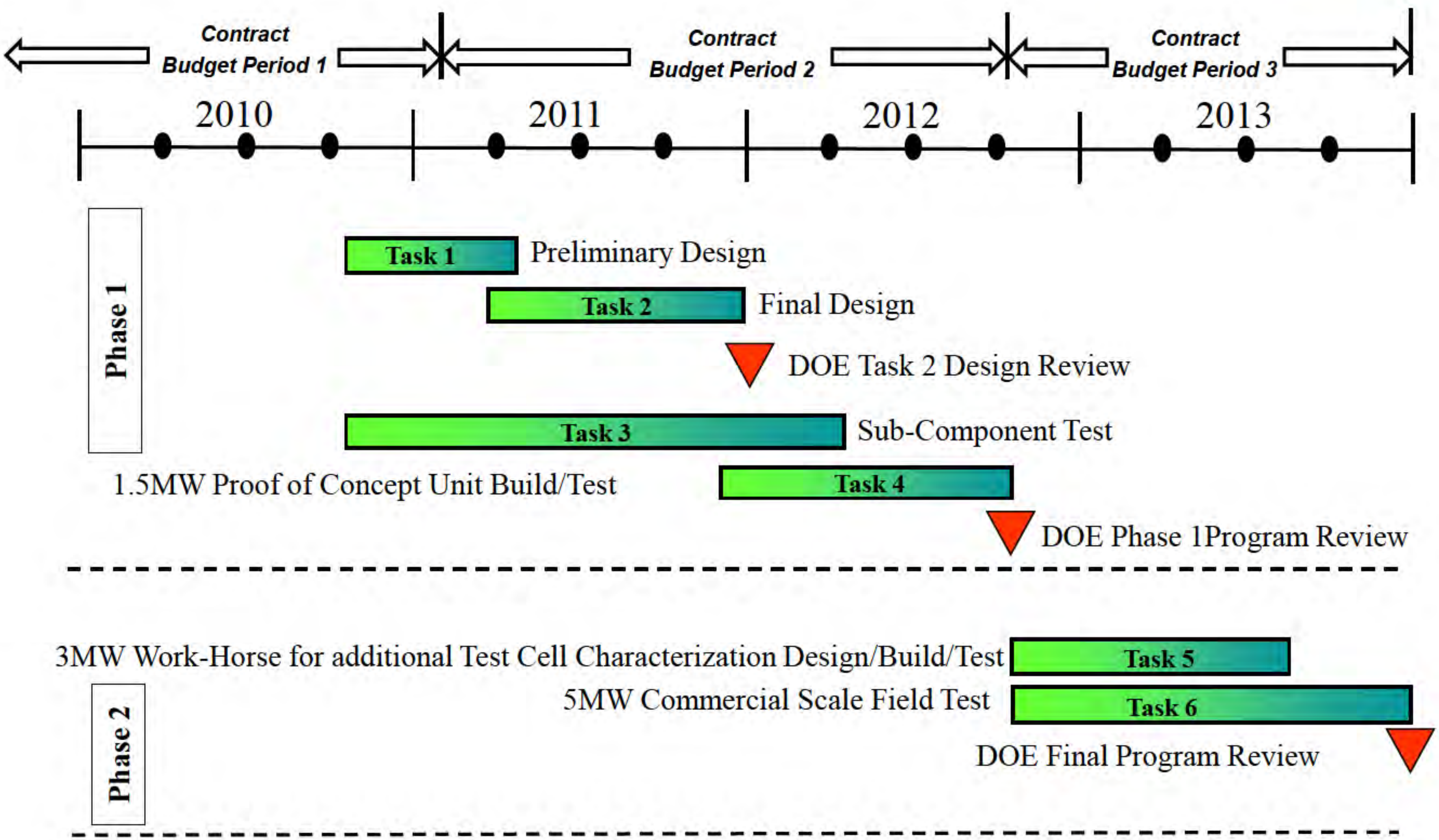
- **Program structure fundamentally demonstrates scalability of technology**
  - **High pressure CO2 – Olean (10MW)**
  - **Low pressure multi-gas compressor rig – Redmond (800 kW)**
  - **ISCE (1,500 kW)**



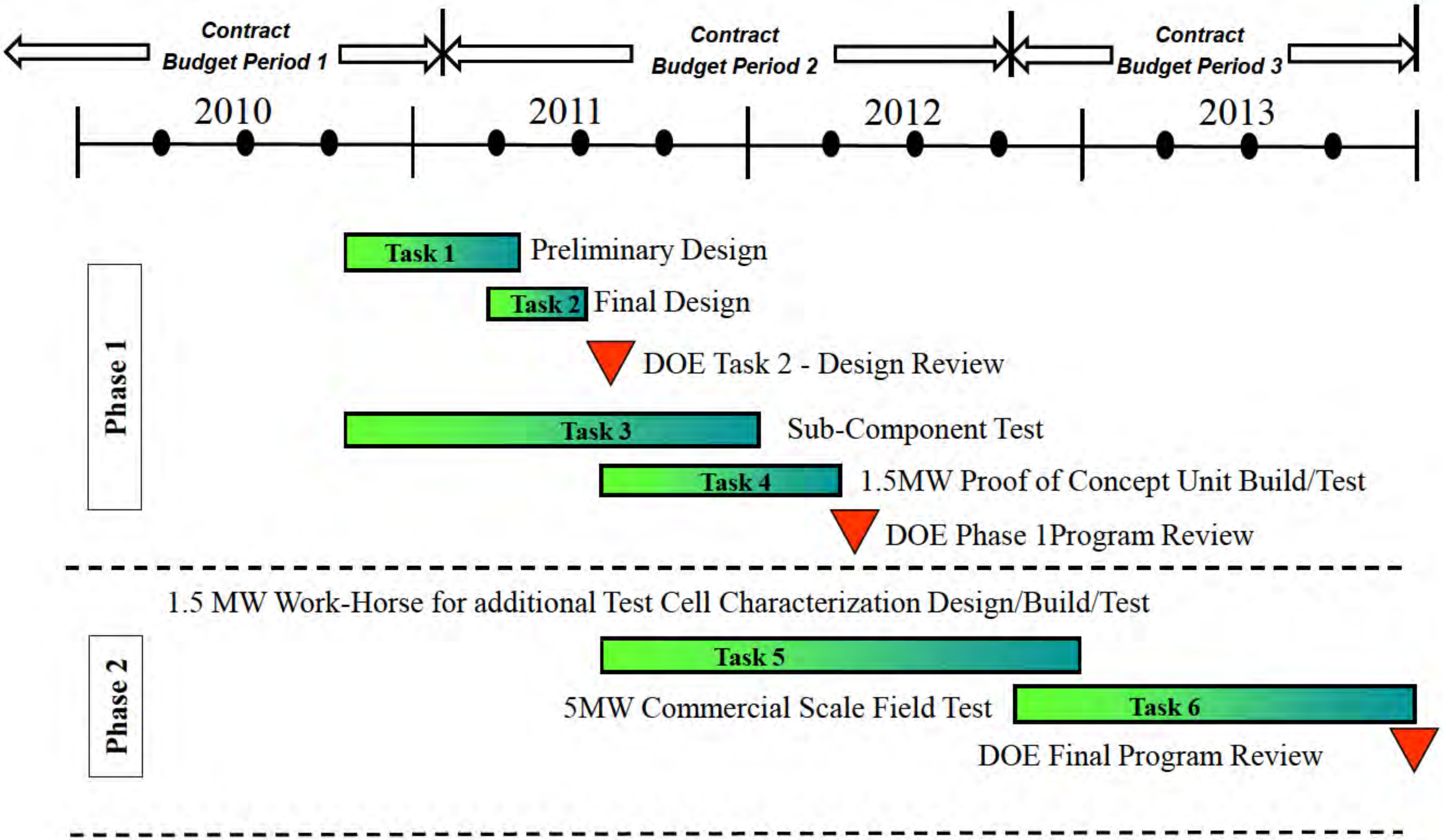
- **Wide range of applications**
  - **Coal mine/landfill/Waste Fuel applications**
  - **Distributed generation/grid stabilization**
  - **Base load power generation**
  - **Combined heat and power**

RAMGEN POWER SYSTEMS

Original ISC Engine Schedule – Ring/Gen 1



Accelerated ISC Engine Schedule – Gen 2



CO_2 Schedule Impact

- **Gen 2 configuration is being incorporated into CO_2 compressor**
  - **Exceptional synergy between programs realized**
  - **All resources working on best configuration**
  - **Significant reduction in ultimate commercialization risk**

- **Test start delayed by 3 months – from Dec 2011 to Feb 2012**
  - **Manufacturing delays of Gen 1 configuration created comparable start dates**
  - **No significant Facility changes required for Gen 2**
  - **Longest lead/biggest parts will work for Gen 2**
  - **Ramgen Schedule Summary 2011.07.7.xlsx**

Installed Pressure Case

Bearing Housing

Budget Impact

- ISC Engine
  - No new funding required
  - Overlap of BP3 Task 5 with BP2 would result in accelerated spending of funding
- CO_2
  - No new DOE funding required – additional private funding being secured
  - Accelerated spending of DOE funding
    - Currently Ramgen splits cost share each month, not by end of Budget Period
    - Request change to cost share at end of Budget Period concurrent with ISC Engine Program

APPENDIX 10.4

Fuel and Air Facility Delivery Systems

Conceptual Design Review

AVC Fuel and Air Delivery Systems

System owner(s):

Brian Massey

5/4/2012

Agenda

- System Definition and Scope
- Functional Requirements/Design Goals
- Interfaces
- Air System Conceptual Design
- Fuel System Conceptual Design
- Exhaust Back Pressure Valve
- Budget and Schedule

System Definition and Scope

- **System includes air piping from the filter outlet flange through the air heater and to the combustor**
- **System includes natural gas piping from the skid to the combustor**
- **Includes flow meters, CFV's, and control valves necessary to meter the flow for fuel and air systems**
- **Also will touch on the exhaust system back pressure valve**

Functional Requirements/Design Goals

- **Air system must maintain choked flow when back pressure is 5atm or above**
- **Fuel system must maintain choked flow at all times**
- **Stoichiometry variation must be continuous over entire operating range (i.e. no binary operation)**
- **Main air must be capable of delivering from 0.336 to 1.681 lbm/s at ~650F**
  - **Min flow based on 2 atm ignition, max flow based on roughly 1/5th sector of full scale ISCE AVC combustor**
  - **Required turndown ratio 5:1**
  - **Air supply is max 2.1 lbm/s at 200-210psig, target combustor back pressure at design point is 150 psia (i.e. need to maintain choked flow with 65-75 psid from supply to point of use)**
- **Cavity air must deliver from 4-12% of main air flow across entire operating range**
  - **Results in min .0084 lbm/s at ignition to max .1375 lbm/s at design point**
  - **Required turndown ratio 16:1**

Functional Requirements/Design Goals

- **Main fuel delivery pressure to combustor must be 50 psi above combustor pressure at design point and 2x above at ignition**
  - **Results in 200psia delivery at design point, 60 psia delivery at ignition**
  - **Natural gas compressor currently delivers at 208 psig**
- **Main fuel must control stoichiometry from ϕ=0.4 to 0.7 over design point operating range and maintain ϕ=0.7 at ignition**
  - **Results in min .008 lbm/s at ignition and max .0408 lbm/s at design point**
  - **Required turndown ratio 5:1**
  - **If ϕ=0.4 to 0.7 variability is required at ignition, min is .0047 lbm/s and turndown is 9:1**
- **Cavity fuel must deliver from 4-12% of main fuel flow and control stoichiometry ϕ=0.7 to 2.2 across entire operating range (6% of main at ignition)**
  - **Results in min .00052 lbm/s at ignition to max .01749 lbm/s at design point**
  - **Required turndown ratio 34:1**

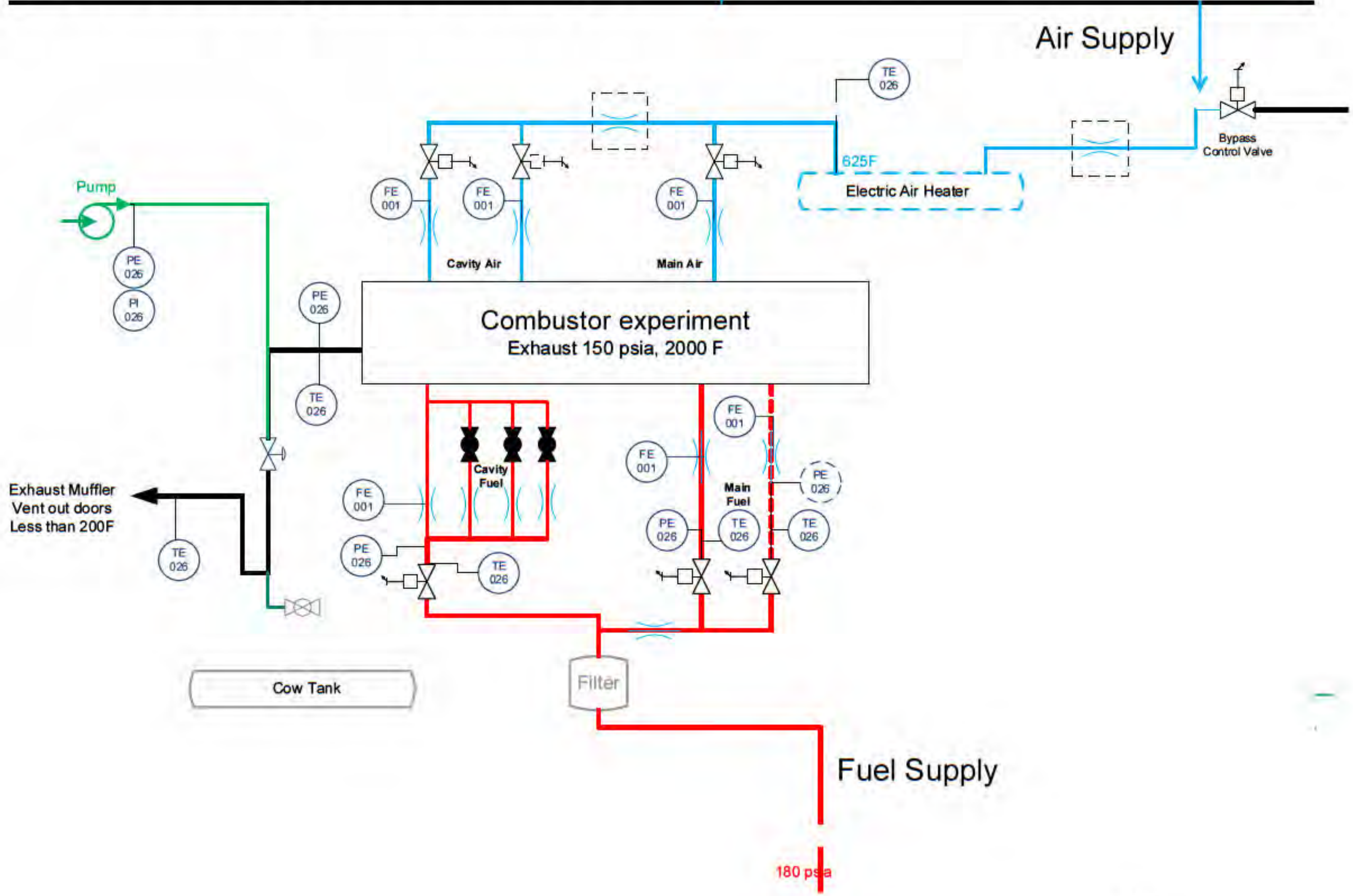
Important Interfaces

- Electrical
  - Instrumentation
  - Control valve feedback
- Mechanical
  - 3” pipe downstream of air filter inlet
  - Outlet is combustor controlled by backpressure valve in exhaust, 2-10 atm back pressure range
- Fluids
  - Air
    - 2.1 lbm/s rotary screw compressor @ max 210psig with 60% turndown capability
  - Fuel
    - ~0.2 lbm/s natural gas skid @ max 208 psig, turndown unknown at this time
  - Cooling Water
    - Estimate of 10 GPM

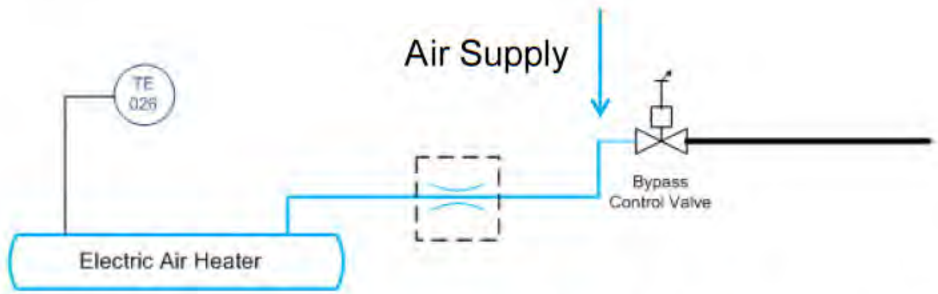
Design Considerations

- **Because supply pressures are so close to point of use pressures, CFV's are required to maintain choked flow at these conditions**
  - **Pressure ratio across CFV is ~1.2**
- **Because such large turndown with continuous variability is required, CFV's will not remain choked at the lower flow rates**
  - **Control valves must be sized to choke before CFV becomes unchoked**
  - **Additional upstream flow meter will be required if accurate flow measurement is desired when CFV is not choked**
- **In some situations the turndown required is so large we may have to resort to binary control (i.e. small CFV's with downstream isolations so we can run high pressures at low flow rates, increase flow by opening isolations)**

Current Proposed Concept



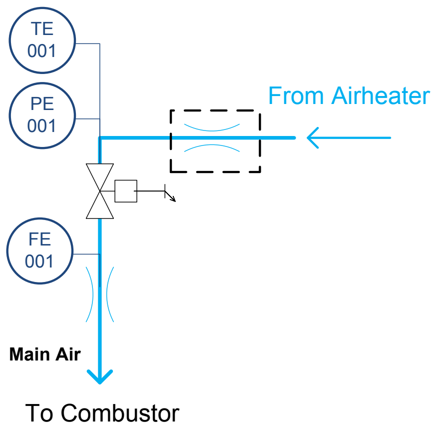
Bypass Valve



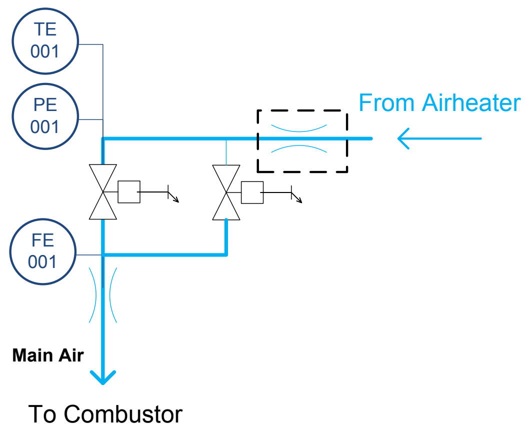
- **Current air turndown requirement is 5:1 (1.68lbm/s to .34 lbm/s)**
- **Based on my interpretation of the 450hp rotary screw compressor, in modulation mode the inlet valve will automatically meter incoming air to reduce mass flow and maintain pressure**
  - **Minimum compressor output is 60% of full output (1.2 lbm/s), otherwise will unload by closing inlet valve and re-circulating**
- **AFRL method is to run compressor at full load and control header pressure by dumping flow overboard with a bypass valve**

Main Air Concepts (5:1 turndown, 200psig supply)

Concept 1:



Concept 2:



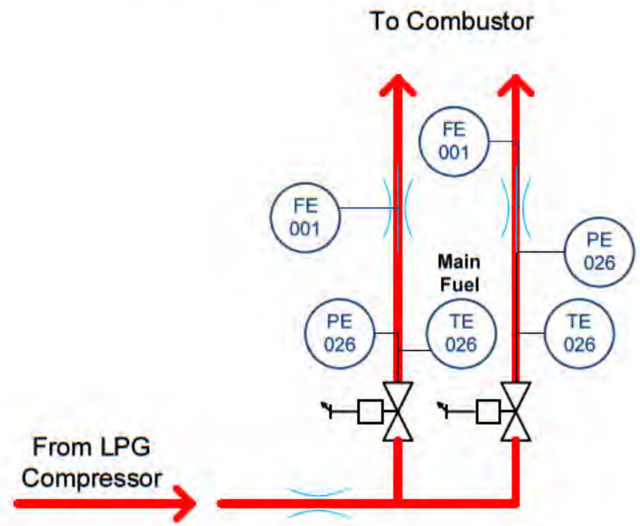
- Single CFV sized for 150 psia back pressure at max flow (185 psia req upstream of CFV)
- Single flow control valve, choke point may jump back to control valve at ignition (37psia/31psia respective up/downstream CFV pressure at ignition)
- Can use upstream sub-sonic venturi flow meter if necessary for more accurate flow measurement when venturi is unchoked

- Same as concept 1 except using a course and trim control valve for better control at low flow rates
- This is how they run in both labs we say at AFRL
- Another variation is to run parallel control valves and parallel CFV's

Main Fuel Concepts (5:1 turndown, 208psig supply)

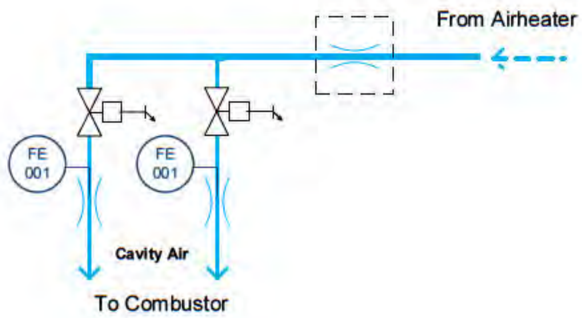
- Similar turndown to main air, but several differences:
  - Target injection pressure 200psia at design point
  - CFV must be sized for full back pressure from ϕ=0.4 to 0.7
  - Needs good cavity control at low ignition flow rate ranges
- Because of supply pressure, cannot achieve desired injection pressure. Will assume 240 psia for this design
- Current working concept is two equal sized CFV's sized for ½ max flow at design point, independent control valves
- Single CFV can do min flow rate with 104psia upstream pressure (ignition at ϕ=0.7)
- Upstream flow meter can be used to measure flow when CFV's are unchoked
- If we had 460psia supply we could do full flow range w/ single CFV and control valve

Leading Concept (requires 240 psia supply):



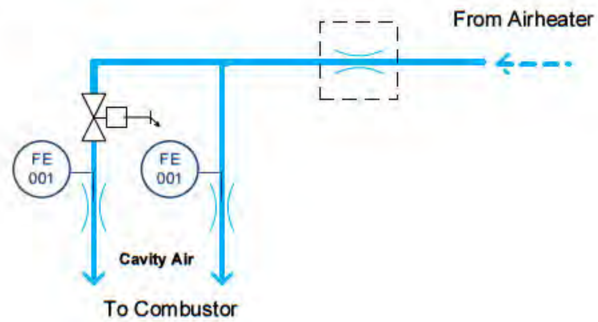
Cavity Air Concepts (16:1 turndown, 200psig supply)

Concept 1:



- CFV1 sized for 4-12% main flow at ignition while remaining choked (downstream pressure above 2 atm ignition target for 4%)
- CFV2 sized for 150 psia back pressure at max flow (185 psia req upstream of CFV, CFV1 will remain on as well)
- Can use upstream sub-sonic venturi flow meter if necessary to meter flow in region where CFV2 is not choked

Concept 2:

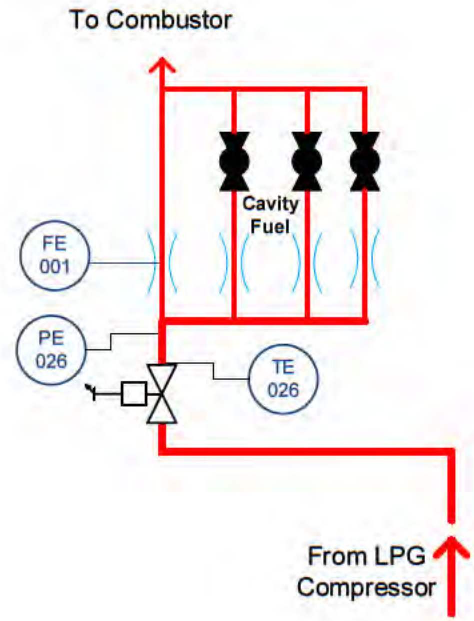


- Size CFV1 for 4% main flow at ignition, 185 psia upstream pressure
- Size CFV2 to make up from 4% main at ignition to 12% main at max flow rate
- Use upstream flow meter when CFV2 is unchoked

Cavity Fuel Concepts (34:1 turndown, 208psig supply)

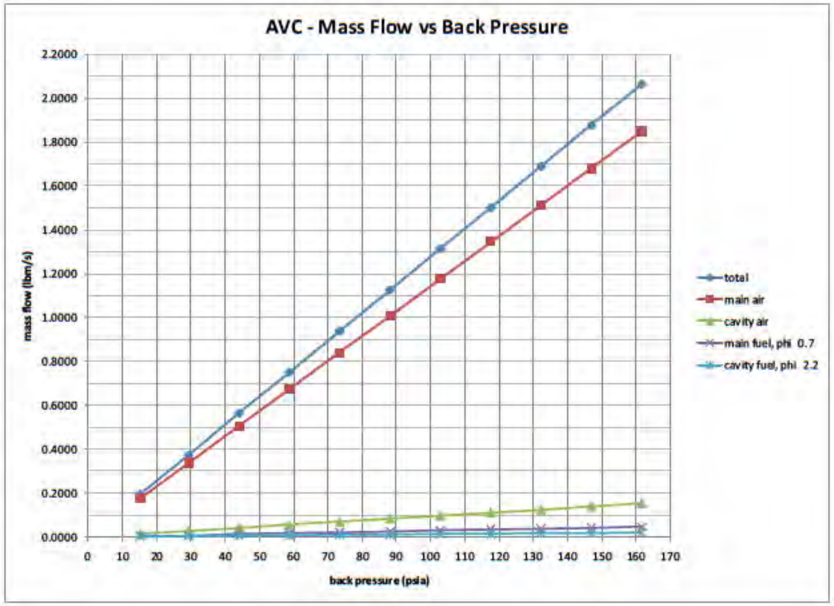
- Widest flow range of all systems at very low flowrates
- Propose using binary flow, i.e. small CFV's or orifice's in parallel with single control valve
- Most likely at least two CFV's will be very small at the same size
- Flow is metered by opening/closing CFV isolations and increasing/decreasing header pressure
- Because of supply pressure, cannot achieve desired injection pressure. Will assume 240 psia for this design

Leading Concept (requires 240 psia supply):



Exhaust Back Pressure Valve

- Back pressure ranges from 2 atm (ignition) to 10 atm from 0.4 to 2.1 lbm/s total mass flow (neglecting water mass flow)
- Preliminary sizing shows 2" Fischer V200 would have to be open ~50 degrees from ignition to full back pressure. A 4" valve would need to be open ~20 degrees
- Valve would have to withstand very high temperatures (trim exists for up to 800F)



Control Valve Selection

- **Preliminary Cv ranges calculated and compared against Fischer V-ball selection available**
- **2” size is the largest we could use on delivery systems, but generally would prefer 1” (.01 < Cv < 35), 1 ½” (.014 < Cv < 76) sizes, or 1” micro Cv sizes (.014 < Cv < 5.23)**
- **2-4” required for back pressure valve**
- **No sizing on bypass valve yet, expecting 2” will be about the size we want**
- **Need to consider acoustic damping trim to reduce our noise levels**
- **Need to look at smaller non-Fischer valves for cavity fuel and maybe cavity air**

- **What we own:**

pipe size	class	QTY	location	make	model
inch					
2	150	1	A160	Bauman	24588SVFEB
4	150/300	Javier 1	A160	Fisher	V200
4	150/300	1	Olean	Fisher	V200
2	150/300	Javier 2, 1 availabl	A160	Fisher	V200
6	150/300	Javier 1	A160	Fisher	V200

Work Plan / Analysis Tasks

- Resolve natural gas supply shortcomings
- Higher fidelity delta-P analysis between compressor and test cell
- Determine when and where choked flow is necessary
- Continue to trade concepts for delivery final down select on concepts
- Detailed control valve, CFV, and sub-sonic flow meter selection, spec all remaining line sizes and components and do detailed delta-P analysis
- Work toward goal of cold flow check out of air system before combustor is installed

Budget and Schedule

- PDR date – targeted with combustor sector PDR, 7/13/2012
- FDR date – targeted with combustor sector FDR, 9/1/2012
- Drawing Release date – P&ID needs to be ready for facility installation, 11/1/2012
- Estimated Manufacturing Time/Delivery date
- Any differences from master schedule? - No
- Is schedule achievable? – Schedule appears achievable for facility hardware and is largely dependent on combustor design effort.

- Current budget
  - Facility - $100k
  - AVC Hardware - $40k
- Is current budget adequate?
  - 7 control valves x $4k = $28,000 (ROM)
  - 11 flow meters x $2k = $22,000 (ROM)
  - $90,000 remaining for GM piping, instrumentation, valves, misc gas system components, etc

APPENDIX 10.5

AVC Test Article

Final Design Review

AVC Test Article

System owner(s):
Ryan Edmonds, Rob Draper, Michael Crayton, Chris Braman
Quest: Paul Vitt, Steve Koester, Chris Mento

12/14/2012

Agenda

- Aero (Quest – Steve/Paul)
  - Burnout Zone Optimization Models
  - Cavity Injection Analyses/Cavity Premixer Models
  - Reacting Flow Model without cooling air
  - Cavity Only Lit Model
  - Main Premixer
  - Cooling Air/Dilution Model of Entire Combustor
- Heat Transfer Models (Quest- Chris)
  - Centerbody
  - Liner
  - Summary
- Mechanical
  - Centerbody Thermal Structural Model (Rob)
  - Liner Structural Model (Quest)
  - Design Details (Michael/Ryan)
    - Liner
    - Centerbody
    - Window
    - Injection of Main and Cavity Fuel
- Instrumentation & Assembly (Ryan)

Functional Requirements/Design Goals

- **Run holding corrected flowrate to match engine conditions.**
- **Must utilize existing Redmond Combustor Test Facility Capabilities.**
  - **Main limitation is air delivery: Capability is 2.1 lbm/s at 150 psia in test rig, Approx. 700 °F maximum preheat temperature.**
  - **Fuel compressor capability: ~0.2 lbm/s natural gas, at max 208 psig on compressor discharge.**
- **Monitor combustor acoustic pressure fluctuations.**
- **Measure combustor exhaust products.**
  - **No specific NO_x or CO requirement for this demonstration.**
- **Record high speed video of combustor in operation.**
  - **Direct radial viewing if possible, or periscope style camera through liner OD.**
- **Demonstrate cavity jets with just plenum fed air letting geometry and pressure set cavity flowrate.**
  - **Design must also allow for separate cavity air feed to vary cavity flowrate as has been done in past designs.**

Important Interfaces

- Electrical
  - Igniter
  - Instrumentation
- Mechanical
  - Mounting connection at combustor exit
- Fluid
  - Main air & cooling air
  - Main fuel
  - Cavity air and fuel
  - Exhaust quench water supply

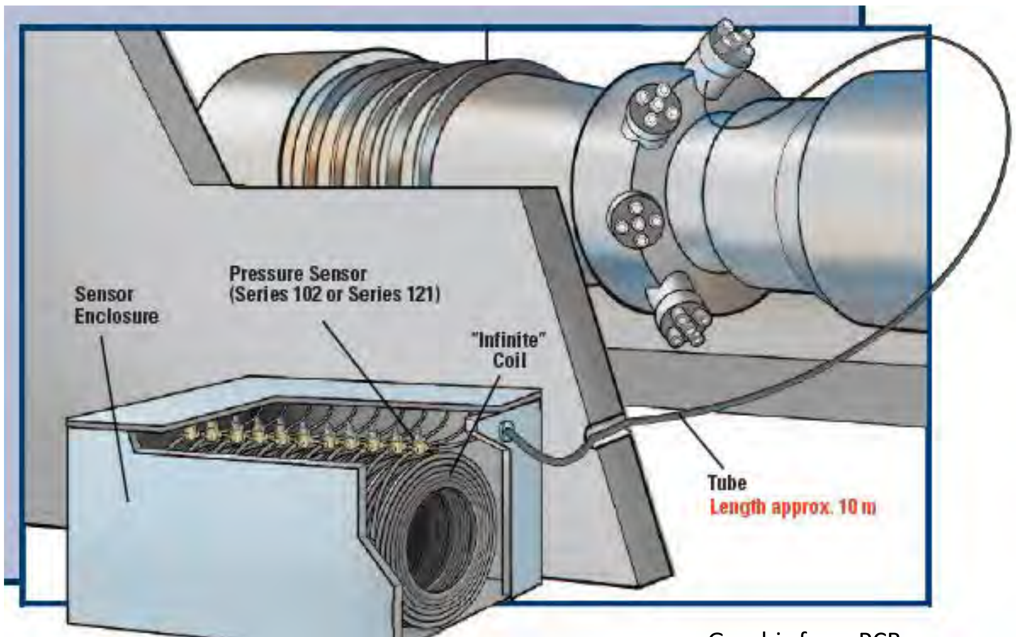
External Presentations

- Aero – Quest, Paul/Steve
- Heat Transfer – Quest, Chris Mento
- Centerbody Structural Model – Rob
- Liner Structural Model – Quest, Chris Craighill
- Liner Design – Michael/Ryan

Mechanical Design Notes

- Split impingement shields closed with nichrome strip?
- Champion igniter feedback
  - The CH34419 has a limited tip temperature due to tungsten material used
- L-ring central cooling pattern line of sight issues to cooling holes

Instrumentation – Dynamic Pressures



Graphic from PCB

- **Plan to remote mount dynamic pressure transducers outside pressure vessel with T upstream of "infinite" coil.**
- **Internal high temperature transducers that can mount on liner range from $3300 - $4600 each are not within the budget.**

Remaining Work Plan / Analysis Tasks

- **Update centerbody thermal/structural model with cooling slot change**
- **3D heat transfer structural modeling:**
  - **Igniter**
  - **Window Region**
  - **Exhaust Flange**
- **Finalize cooling features for combustor optical windows/probes**
- **Detailed review of all cooling hole patterns during drawing creation**

Exhaust Region*

- **Combustor Exhaust Diagnostics:**
  - **Emissions Probe**
  - **Temperature Rake**
  - **Flow Angle Probe?**
- **Design exhaust water quench**
- **Finalize extension barrel that connects combustor to the pressure vessel**

*Not part of this review

Schedule

- **Drawing Release Completion – 2/15/2013?**
- **Estimated Manufacturing Time/Delivery date – 12-14 weeks**
  - **Targeting material order for centerbody by 12/21/2012 to expedite schedule**
  - **Subject to revision as Major Tool receives final hardware models/drawings**
- **Any differences from master schedule?**
  - **Current dates are in line with master schedule**
- **Is schedule achievable?**
  - **Dependent on personnel availability with two active test programs**

Final Design Review

Annular AVC Test Article

System owner(s):

Ryan Edmonds
Quest: Paul Vitt, Steven Koester, Chris Mento, Chris Craighill

12/14/2012

Burnout Zone Optimization

Background:

- There was concern that the centrifugal forces associated with thermal gradients in the swirling combustor flow would disturb the outer liner flowfield with a peaked combustor, so the OD was made cylindrical
- A second concern was that the separation region behind the aft body needed to be closed out well before the turbine inlet, so several variations on the aft wall slope were examined to see if the vortex region could be made smaller

Objective:

- Evaluate the impact of the wall slope changes on recirculation zone size and residence time (goal of between 12 & 18 ms with the new configurations)

Approach:

- Use reacting, steady state CFD analysis on three burnout region designs and compare the results.

Premix Supply CFD and Vortex Cavity Performance CFD

Background:
- Vortex cavity pre-mix supply system was design to mix the fuel and air on-board the centerbody, so that the same primary hardware could be used for both pressure-fed and plenum-fed conditions
- Detailed model required to verify mixing, pressure drops and flow distribution

Objective:
- Determine mixing, pressure drop and flow splits through the premix delivery system.
- Size the vortex cavity injection holes for pressure and plenum supply conditions.

Approach:
- Steady-state multi-component mixing CFD analysis was used to assess the premix delivery system

Premix Supply CFD

Model History

Transfer Tube

- Initial analysis showed the transfer tube was the highest loss contributor.
- Diffuser elements were added to aid in pressure recovery.
- Other design changes allowed the transfer to diameter to increase to the current size (as large as poss ble).

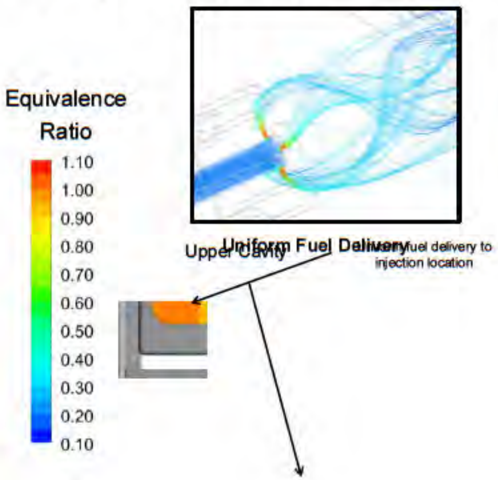
Primary Mixer

- Shaped mill-out in centerbody to introduce swirl in the mixing chamber and transfer tube
- Preliminary models were used to shape the chamber for reduced pressure loss

Current Design

- Driven by both previous studies (large transfer tube, primary mixer) and packaging to allow fuel and air to be fed through the combustor struts

Multiple Fuel Injection Locations



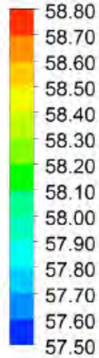
Primary Mixer Design Study

	Model 2	Model 3	Model 4	Model 5	Model 6
Model Description	Deep radius Full mixer	Shallow radius Full mixer	Shallow radius Filleted mixer	Deep radius Squared mixer	Medium radius Squared mixer
Outlet Total Pressure (psia)	284.8	285.6	286.6	281 8	285.1
Pressure Drop (psi)	5.2	4.4	3.4	8.1	4 9
Pressure Drop %	1.8%	1.5%	1.2%	2.9%	1.7%
Fuel Mixing	Good	OK	Bad	Good	Bad

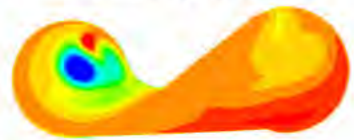
Premix Delivery CFD

Total Pressure Loss

Total Pressure (psia)

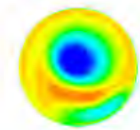


Primary Mixer



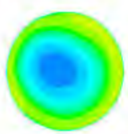
Pt = 58.61 psia

Transfer Tube Inlet



Pt = 58.32 psia

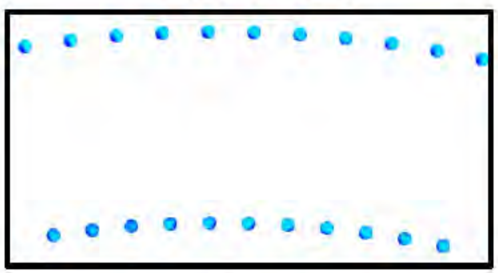
Transfer Tube Outlet



Pt = 58.14 psia

Mixing Plenum

Injection Outlet



Pt = 57.8 psia

- Total pressure drop is 1 psi or 1.7%.
- The largest pressure drop (0.34 psi) was in the mixing plenum

Cooling Addition Model CFD

Status and Recommendations

Cooling Addition Model Status

- Two CFD models with varying simplifications were created for the combustor with centerbody and liner cooling included to get initial estimates for combustor exit temperature and profile factor.
- In both models, the simplifications may be overshadowing the desired results, indicating the need for a more complex model

Recommendations

- Increase the fidelity of the cooling in the reacting flow model that includes radiation and a reasonable estimate of wall TBC temperature (from the thermal analysis).

Detailed Model CFD

Objective:

- Combine all components of the combustor into a single reacting model.
- Verify component performance as a system.
- Verify cooling assumptions.

Approach:

- Use steady state CFD analysis at rig conditions.
- Combine previous models: Cooling flow supply, premix delivery, main fuel injection, inner and outer liner cooling and combustor aero.
- First model will not include impingement or effusion cooling geometry to ensure model operation.
- Second model will incorporate effusion and impingement geometry using results from previous model for initialization.
- Use flow visualization and measurement plane to verify mass flow rates, pressures, and cavity, mixing performance.

Aero Design

Summary and Remaining Work

Analysis Summary

- Burnout zone was re-sized for improved flow patterns (to minimize impact on turbine inlet flowfield)
- Vortex cavity premix delivery system operation was verified, and the injection holes were sized for plenum and pressure fed conditions
- Updated combustor model (with two changes above) showed expected performance
- Internal centerbody cooling system modeled and delivered close to the expected results (not shown in the current presentation)
- Achieving acceptable mixing levels at the dump plane has proven to be challenging, with the current design delivering 45% mixing in the primary stream
- Combustor cooling models were assembled, but the simplifying assumptions made to enable the required schedule have reduced the usefulness of the results

Next Steps

- Continue to investigate improvements to the main fuel mixer
- Upgrade the fidelity of the cooled combustor model and assess the turbine inlet temperature profile
- Coupled internal-external flow combustor model to verify operation of the assembled components

Aft Body 2D Thermal FEA Analysis

Aft Body Thermal Analysis

Liner 2D Thermal FEA Analysis

Liner Thermal Analysis

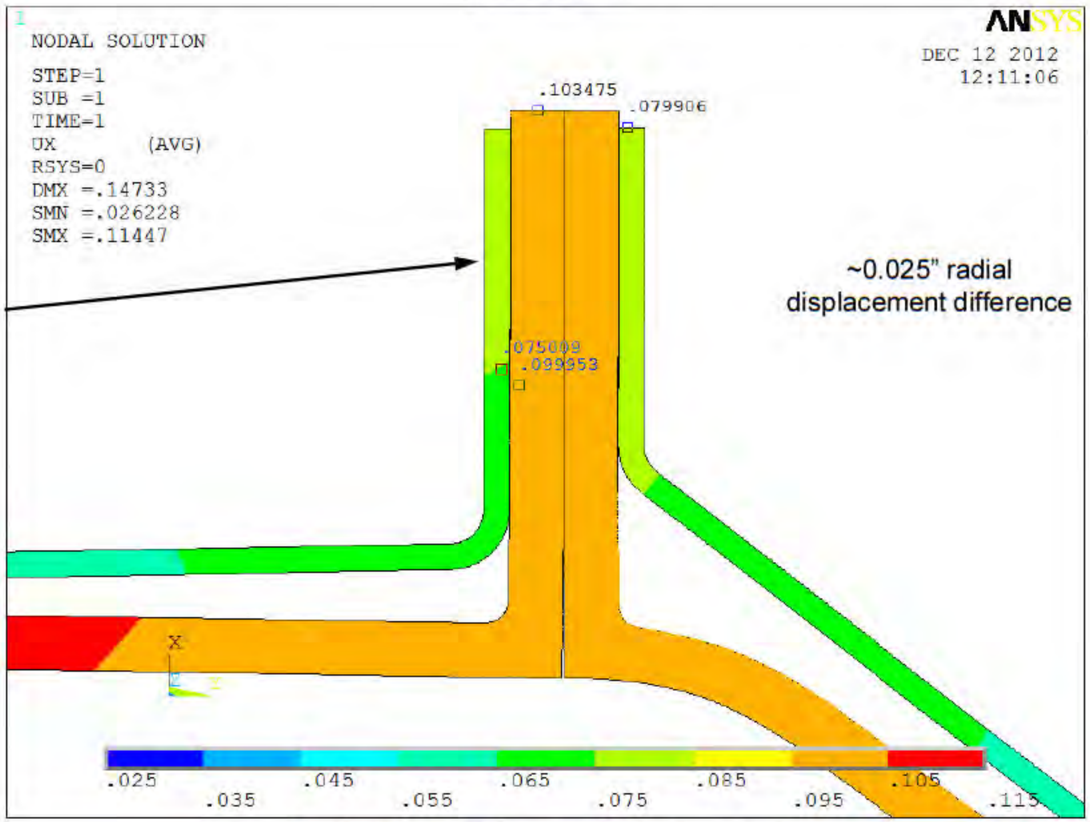
Liner 2D Structural Analysis

2D Liner Structural Model

Outer Flanges - Radial displacement

Updated Model

Radial slotted holes on cold impingement plates will allow them to slide as hot liner grows



~0.025" radial displacement difference

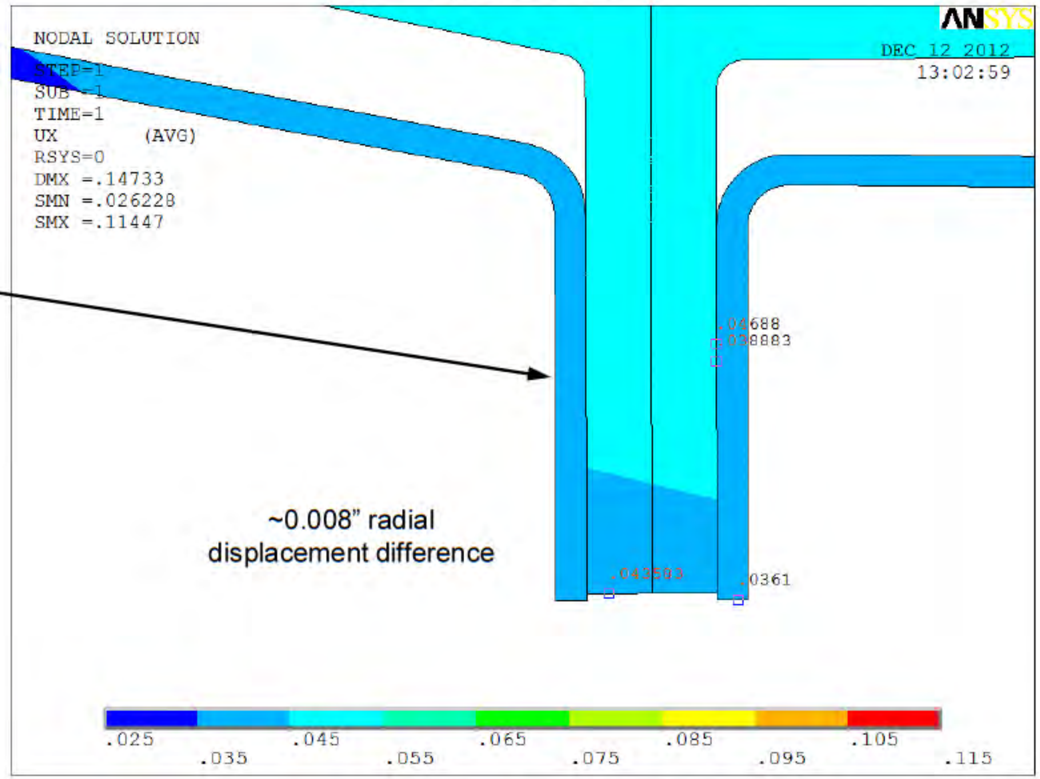
Displaced geometry – radial displacements (inches)

2D Liner Structural Model

Inner Flanges – Radial Displacement

Radial slotted holes on cold impingement plates will allow them to slide as hot liner grows

Updated Model



Displaced geometry – radial displacements (inches)

ISCE AVC Test

3D Sector FEA

Final Design Review

December 14, 2012

Haynes 230 Temperature Degradation

- **RT Ultimate strength: 114 ksi**
  - **900°F : 84% of RT = 96 ksi**
  - **1200°F: 78% of RT = 86 ksi**
  - **1500°F: 58% of RT = 66 ksi**
  - **1800°F: 28% of RT = 32 ksi**
- **RT Tensile yield strength: 49 ksi**
  - **900°F: 72% of RT = 35 ksi**
  - **1200°F: 68% of RT = 33 ksi**
  - **1500°F: 68% of RT = 33 ksi**
  - **1800°F: 36% of RT = 18 ksi**

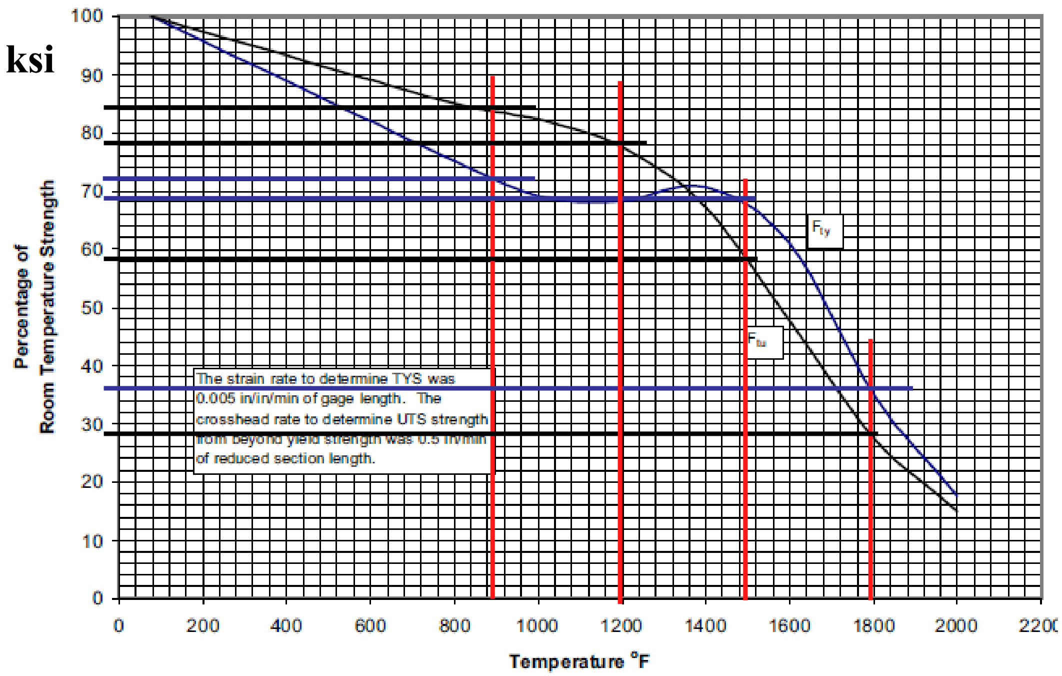


Figure 6.3.9.1.1(b). Effect of temperature on tensile properties of HAYNES 230

F6.X FEA: Aft Cap

- Aft cap surfaces range from 1000°-1500°F
  - Bolt attachment: 1000°F: UTS = 96 ksi, YS = 35 ksi
  - Majority of part: 1500°F: UTS = 66 ksi, YS = 33 ksi
- Need to improve temperature mapping process at bolt attachment and rib regions for better stress assessment
- Apart from bolt attachment and rib sections, rest of aft cap is below yield strength

Remaining Work on Structural Model

- Rebuild model with latest geometry and temperature map
- Fix structural mapping issues on manifold, manifold plate, and aft cap
- Ensure loading meets Milam gasket manufacturer recommendations
- Fix contacts as needed to obtain realistic component peak stresses
- Run sub-models as needed on selected components

Backup Slides

Seal Plate to Centerbody Gasket

Milam Mica Laminate

PRINTER FRIENDLY VERSION

« Back to Previous Page

Milam Mica PSS 130

- Type PSS
- High Quality Mica Sheet
- Laminated on Pegged Stainless Steel Insert

All Milam laminates are suitable for use in hot, dry gas applications such as exhaust manifolds, turbines, turbo chargers and air heat exchangers.

Compressibility ASTM F36A	12-15%
Recovery ASTM F36A	38-45%
Ignition Loss DIN 52911	<5%
Stress Relaxation DIN 52913 50 MPa/300°C	4,786 psi
Stress Relaxation DIN 52913 40 MPa/300°C	4,061 psi
Tang Insert Stainless Steel Thickness	AISI 316 .004"
Continuous Service Temperature	1,652°F maximum
Gas Leakage DIN 3535/6	>100 ml/min
Stress	14,503 psi maximum
Thickness	.051"
Thickness Tolerance	+/- 5% of nominal thickness
Sheet Sizes (Nominal)	47" x 40"
Color	Metallic Tan

For Additional Information Call Toll Free 1.800.990.7325 or email info@thermosealinc.com.

Haynes 230 Temperature Degradation

- **RT Ultimate strength: 114 ksi**
  - **900°F : 84% of RT = 96 ksi**
  - **1200°F: 78% of RT = 86 ksi**
  - **1500°F: 58% of RT = 66 ksi**
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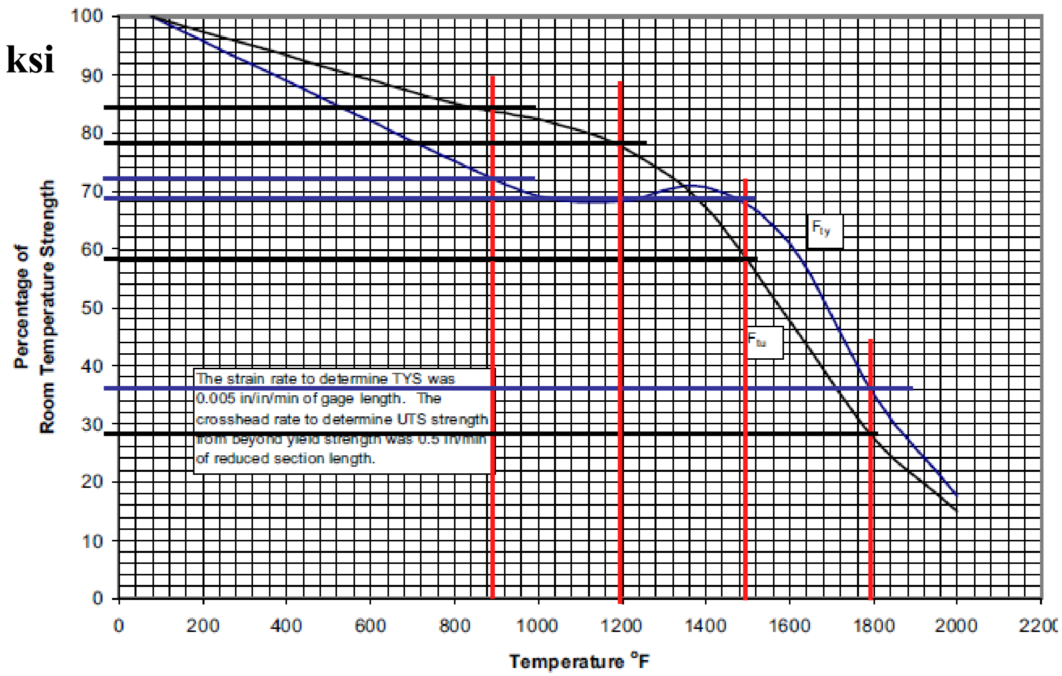


Figure 6.3.9.1.1(b). Effect of temperature on tensile properties of HAYNES 230

Hastelloy X Temperature Degradation

- UTS = 102 ksi at RT
  - 84% of RT at 950°F = 86 ksi
  - 30% of RT at 1560°F = 31 ksi
- Syt = 44 ksi at RT
  - 84% of RT at 900°F = 37 ksi
  - 46% of RT at 1560°F = 20 ksi

MIL-HDBK-5H
1 December 1998

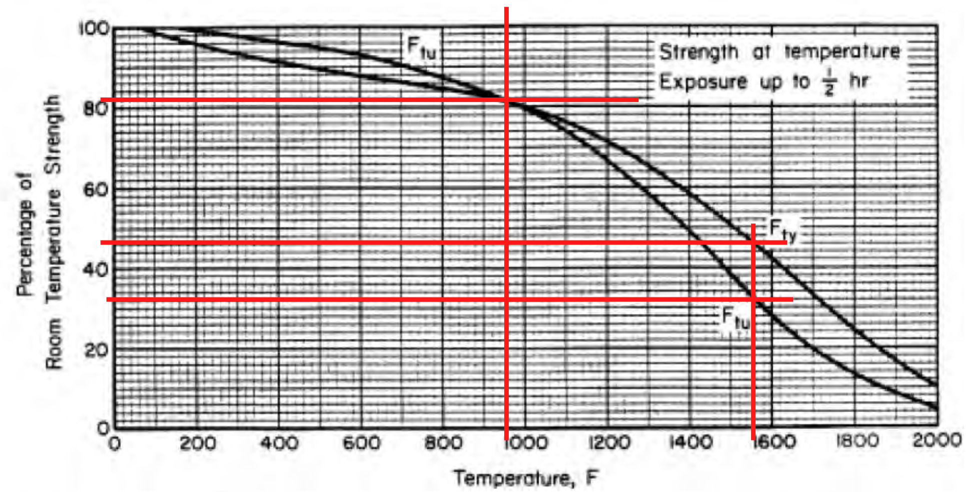
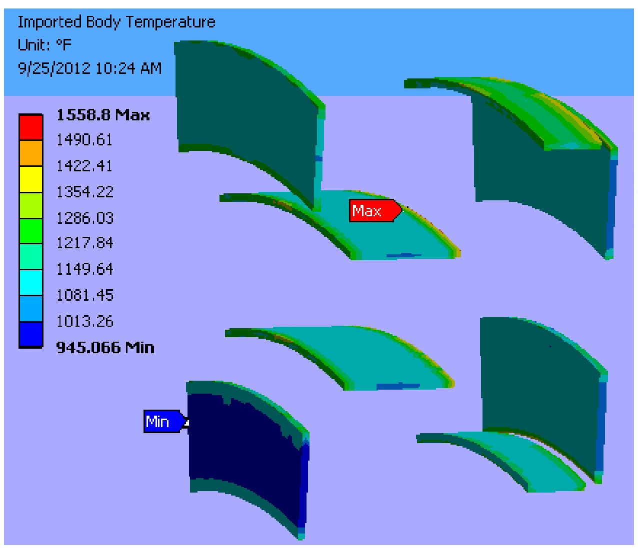


Figure 6.3.1.1.1. Effect of temperature on the tensile ultimate strength (F_{tu}) and the tensile yield strength (F_{ty}) of Hastelloy X sheet.



Haynes 230 Temperature Degradation

- RT Ultimate strength: 114 ksi
  - 900°F : 84% of RT = 96 ksi
  - 1200°F: 78% of RT = 86 ksi
  - 1500°F: 58% of RT = 66 ksi
  - 1800°F: 28% of RT = 32 ksi
- RT Tensile yield strength: 49 ksi
  - 900°F: 72% of RT = 35 ksi
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  - 1500°F: 68% of RT = 33 ksi
  - 1800°F: 36% of RT = 18 ksi

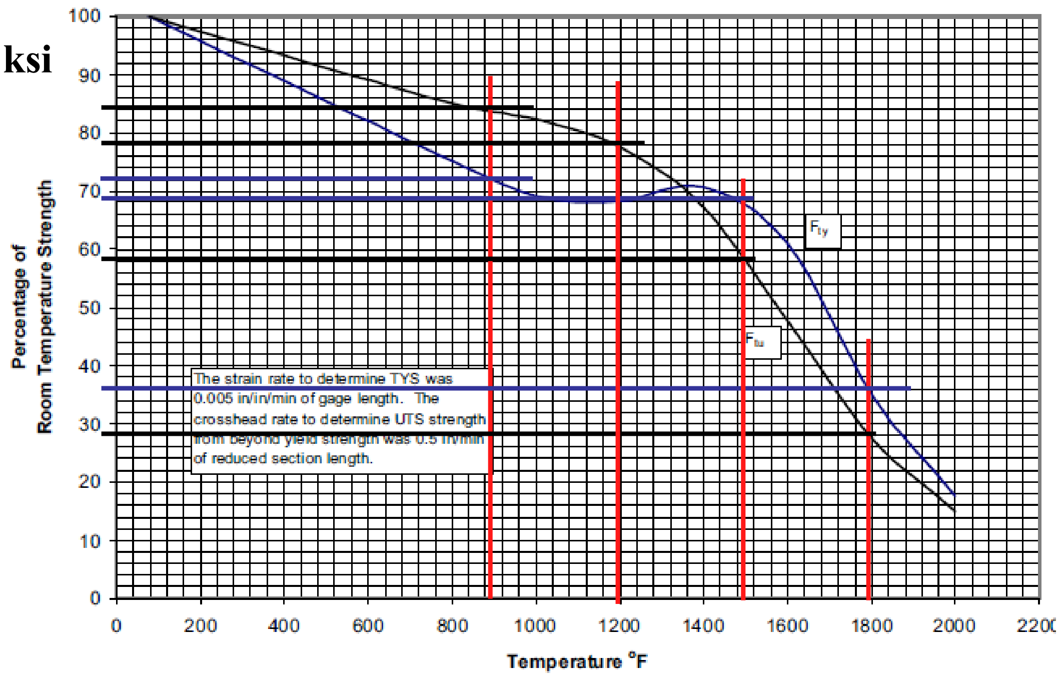


Figure 6.3.9.1.1(b). Effect of temperature on tensile properties of HAYNES 230

Haynes 188 Temperature Degradation

- Ultimate strength = 125 ksi at RT
  - 78% of RT at 900°F = 98 ksi
  - 24% of RT at 1800°F = 30 ksi
- Tensile yield strength = 57 ksi at RT
  - 66% of RT at 900°F = 38 ksi
  - 34% of RT at 1800°F = 19 ksi
- Comp. yield strength = 55 ksi at RT
  - 66% of RT at 900°F = 36 ksi
  - 20% of RT at 1800°F = 11 ksi ????

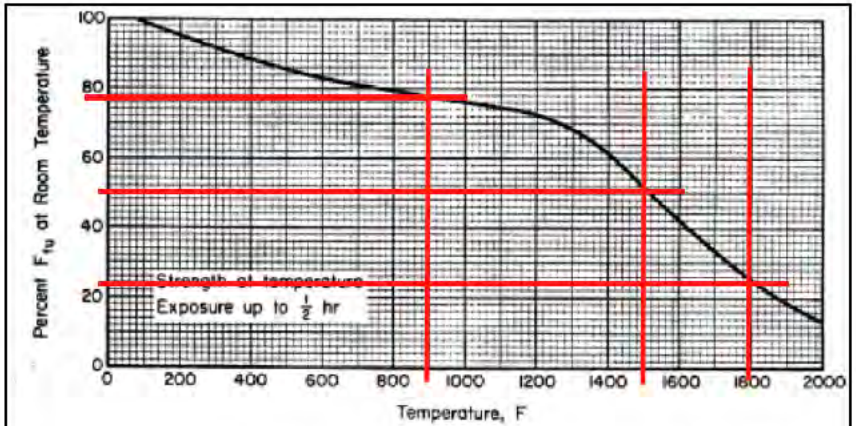


Figure 6.4.2.1.1(a). Effect of temperature on tensile ultimate strength (F_{tu}) of HS 188 sheet.

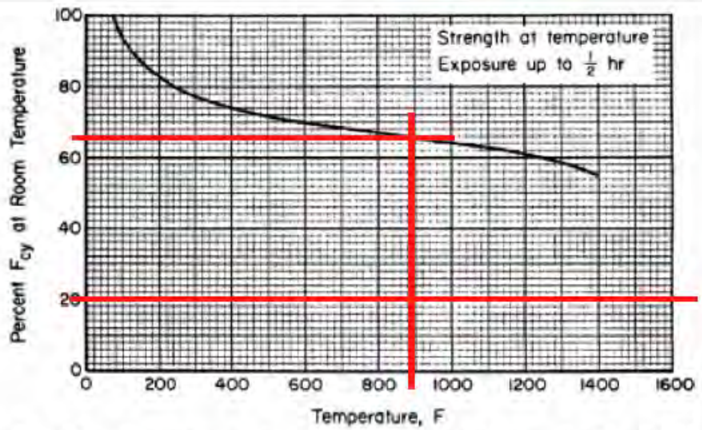


Figure 6.4.2.1.2. Effect of temperature on compressive yield strength (F_{cy}) of HS 188 sheet.

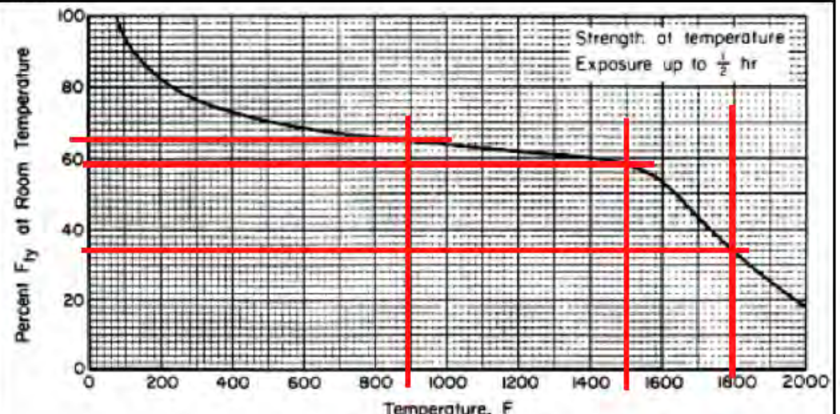


Figure 6.4.2.1.1(b). Effect of temperature on tensile yield strength (F_{ty}) of HS 188 sheet.

19

Mechanical Design Notes

- Igniter
- Pressure Ports
- Updated Liner
- Cooling Holes
- Seals
- Hanging Issues

APPENDIX 10.6

Combustor Test Article

Preliminary Design Review

AVC Combustor Pressure Vessel

System owner(s):

Chris Braman, Brian Massey, Ryan Edmonds, Michael Crayton

08-02-12

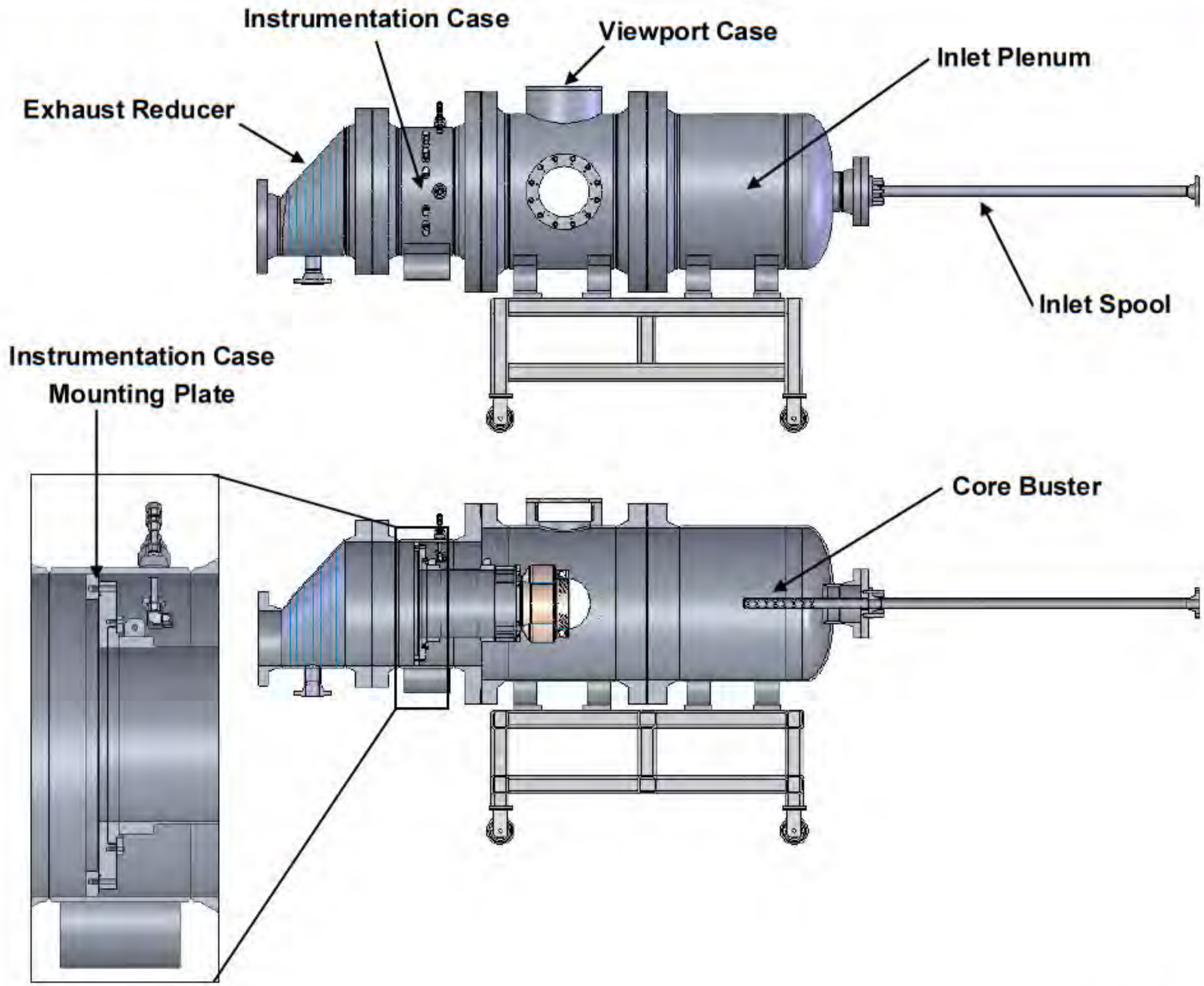
Purpose

- The purpose of today's PDR is to review the large long lead time pressure vessel components before they go out to Alaskan Copper and other vendors for quote
- Some internal and external case components may be discussed but will be reviewed in more detail later and are not part of the pressure vessel quote

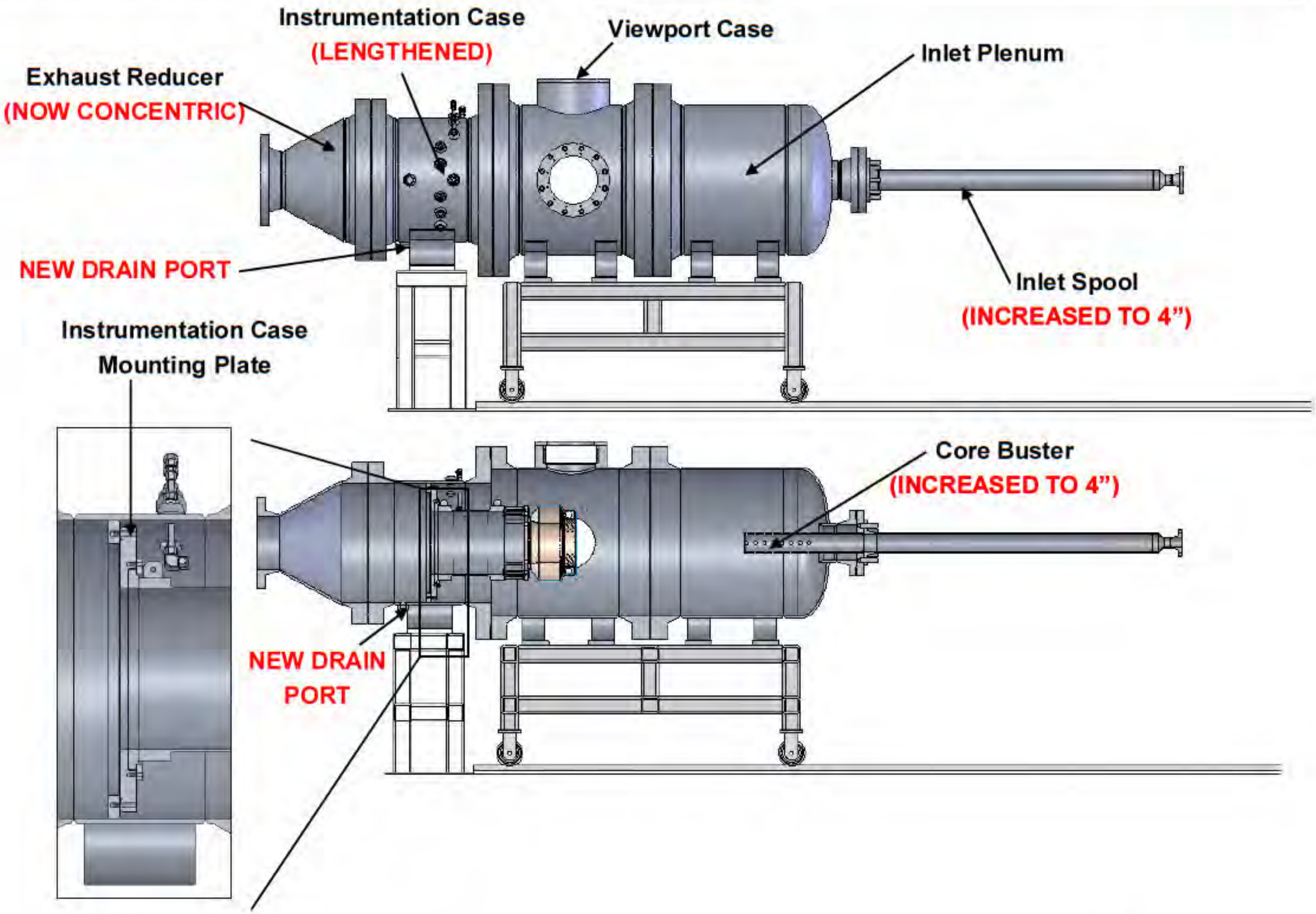
System Definition and Scope

- **Pressure vessel enclosure for ISCE AVC test that shall simulate combustor operation at high pressure and temperature while maintaining corrected mass flow (Mach number similarity).**
- **Pressure vessel starts downstream of inlet valve and flow meters and ends at pipe spool upstream of backpressure valve**
  - **Comprised of 5 pieces: inlet pipe spool, inlet plenum, view port case, instrumentation case, and exhaust reducer**
- **The pressure vessel does not include the case stand, exhaust system components, the extension box, or other internal combustor mounting hardware**

System Definition and Scope



System Definition and Scope – UPDATE 8/10



Functional Requirements / Design Goals

- NOTE: New or altered items in red
- Combustor to be tested at 60 psia (from 150 psia), 600-650 °F
- Air mass flow 1.92 lbm/s (from 1.68 lbm/s)
- Pressure Vessel Maximum Operating Conditions:
  - 300 psia, 1000 °F
    - Maximum pressure chose for future capability, maximum temperature based on ISCE current T_2 of 855 °F that might be run if vitiator were to be re-built.
- Pressure Vessel shall support the combustor components.
- Pressure Vessel shall allow for combustion component changes with minimal down time.
- Pressure Vessel shall be optically accessible.
  - Optical viewing port minimum diameter, 10”.
    - Rectangular window can also be used to achieve similar viewing area.
  - Pressure Vessel can be de-rated to 150 psia with windows installed.

Functional Requirements / Design Goals

- **Pressure Vessel shall be ASME boiler and pressure vessel code certified.**
  - **Requires review and sign off by a PE.**
  - **Appears to be required in Washington State.**
  - **Removes Ramgen's liability with insurance company if certified.**
  - **Must have a pressure relief system.**
- **Pressure vessel shall be designed to allow for flexibility to add future capability in terms of flow rates, pressures, temperatures, and combustor types. No specific guidelines set, but generally trying to build hardware the same size as what AFRL uses**

Important Interfaces

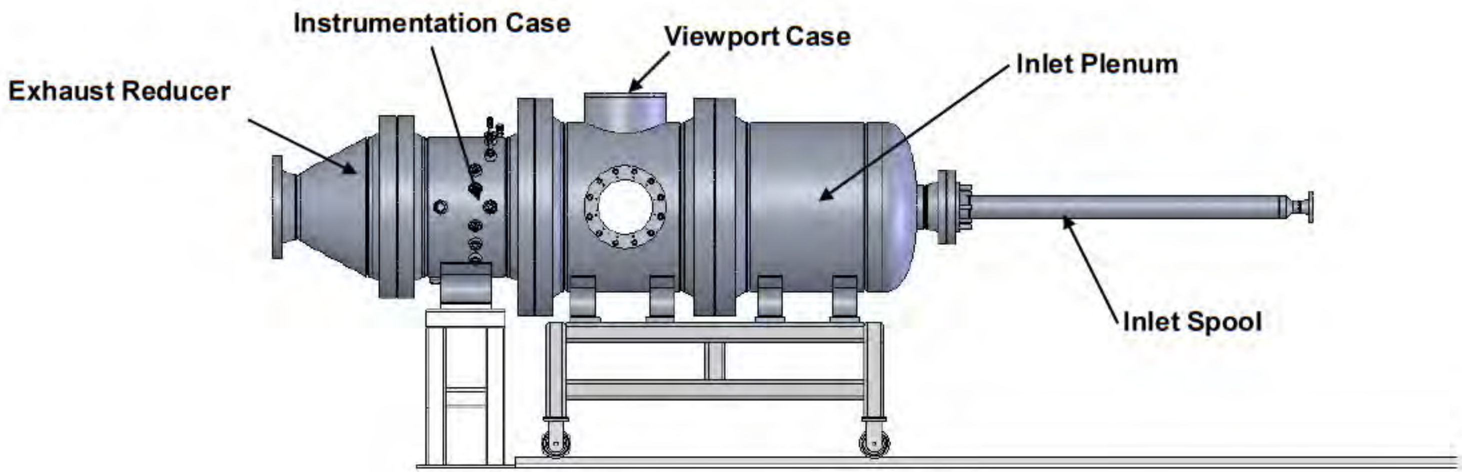
- Electrical
  - Instrumentation interfaces (TC's, pressure transducers)
  - Igniter system interface
- Mechanical
  - Mounting of the combustor test components.
  - Maintenance and change out of the combustor test components.
  - Exhaust/back pressure system interfaces
  - Main air, cavity air, main fuel, cavity fuel
- Fluid
  - Air
    - Design Point Maximums
      - » Main Flow – 1.92 lbm/s
      - » Cavity Flow – 0.19 lbm/s
      - » Max compressor airflow capability is ~2.1 lbm/s
  - Fuel
    - Design Point Maximums
      - » Main Flow – 167.9 lbm/hr (from 147 lbm/hr)
      - » Cavity Flow – 85.9 lbm/hr (from 63 lbm/hr)
  - Cooling Water
    - Estimate of 10 GPM
- Most of these interfaces will take place through the instrumentation case.

CDR Action Items

- Test section or Test rig naming is confusing come up with consistent naming convention.
  - Nomenclature is as follows, the combustor is the "test article", test section will not be used we will refer to this as the pressure vessel, which will be comprised of the "inlet plenum, view port case, and instrumentation case. Ryan (8/1/2012)
- Add graphic at beginning of slides to show scope for design review.
  - Added to PDR slides. Ryan (8/1/2012)
- Burst disk or PRV should be tied to fuel delivery system to trigger fuel shutoff if pressure vessel experiences pressure relief.
  - This will be addressed in the fuel and air delivery PDR Ryan (8/1/2012)
- Is a window required on top of the test section for future diagnostic work? Check on cost delta to add this to the rig?
  - Window has been added to TDC, cost delta will be captured in updated quotes that will be obtained after PDR. Ryan (8/1/2012)
- Can the window design be inverted? Round boss into "pipe" rather than welded on outside of pipe.
  - This is not recommended since we are likely planning to build a full annulus combustor that will be round and have window protrude into the section will be space prohibitive. Ryan (8/1/2012)

Design Concept #2 selected from CDR

- **Make a more conventional pressure case.**
- **Developed in conjunction with potential vendor, Alaskan Copper.**
- **Inlet pipe spool, Inlet plenum, view port case, instrumentation case, and exhaust reducer are weldments, using standard pipe components, rolled plate, and forged flanges.**
- **All materials stainless steel, most likely 304 stainless steel.**

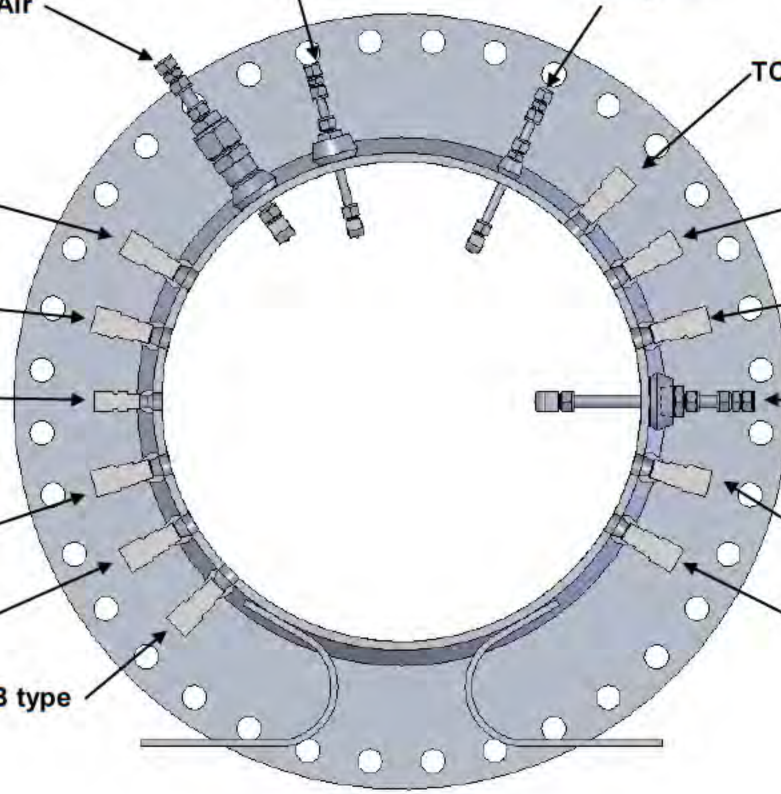


Updated Items from CDR

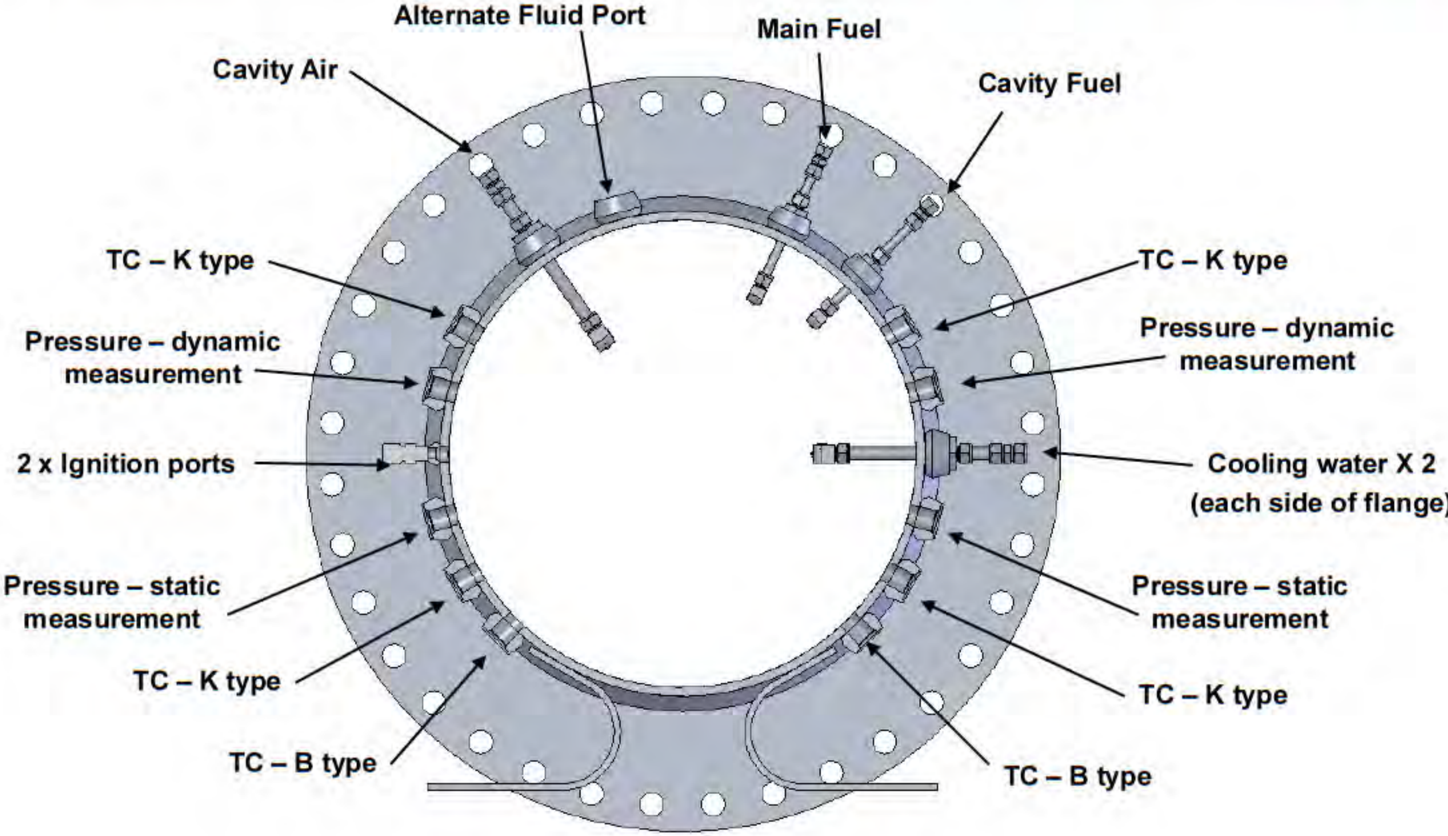
- Settled on full annular full scale combustor at reduced pressure/mass flow rather than 1/5th full scale sector
  - Affects combustor mounting to inside of case
- Increased mass flow and significantly decreased fuel/air delivery pressures
  - Need to update fuel/air system design, will result in increased inlet spool and core buster diameter
- Added third view port to top of pressure vessel
- Altered reducing exhaust spool to be eccentric and added a water drip leg
- Added some preliminary exhaust connections
- Assigned preliminary locations and connection types for services entering the case via the instrumentation case assembly
- Inlet spool length increased from 4ft to 6ft

See model for details

Instrumentation Case Ports



Instrumentation Case Ports – UPDATE 8/10



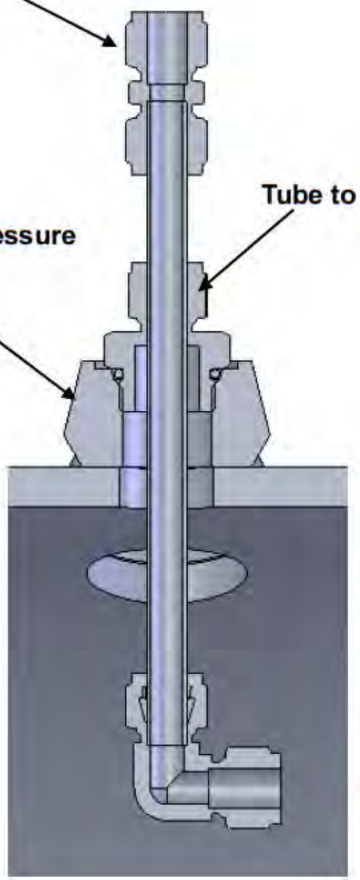
Instrumentation List

- Type K thermocouples (1/16” diameter)
  - AVC test requirement: 30
  - Max allowable: 64
- Type B thermocouples (1/16” diameter)
  - AVC test requirement: 8
  - Max allowable: 32
- High speed pressure (30kHz, 1/16” or 1/8” diameter)
  - AVC test requirement: 8
  - Max allowable: 16-32 depending on diameter
- Low speed pressure (1Hz, 1/16” or 1/8” diameter)
  - AVC test requirement: 8
  - Max allowable: 16-32 depending on diameter

Instrumentation Porting

- Conex MHM5 fittings with weld neck adapter (butt or socket weld)
- Igniter uses smaller

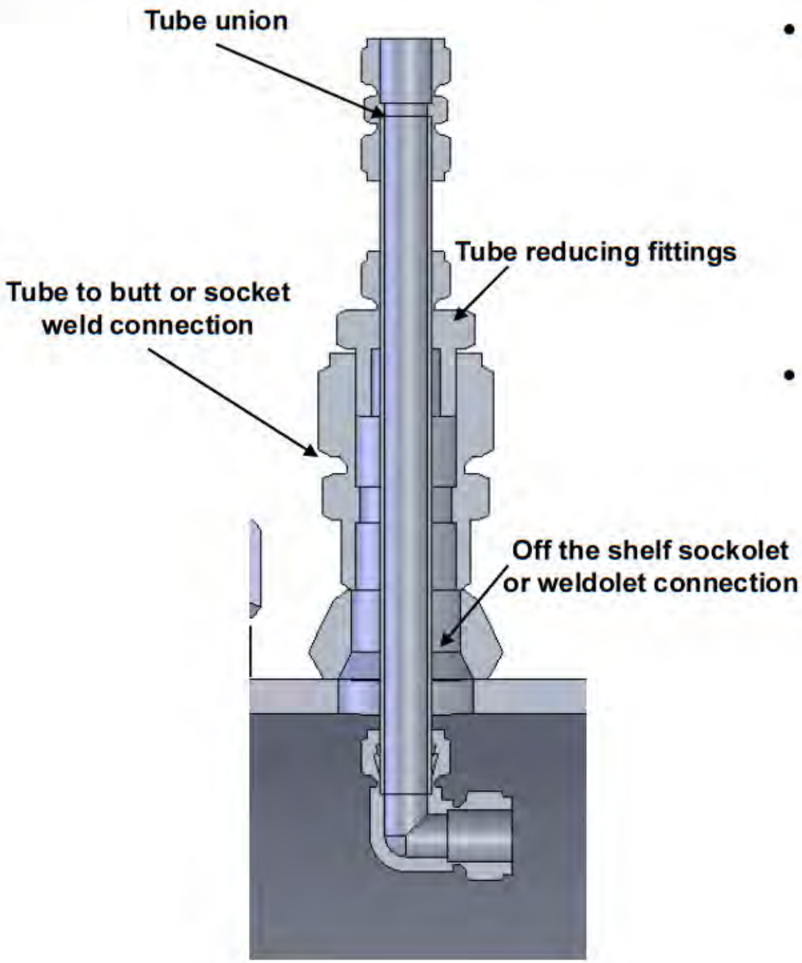
Fluid Porting Option #1: SAE fitting



- Pros
  - Easy removal of external case connections
  - Internal connection will pass through case port
- Cons
  - Long lead (~14 weeks) needed to get fittings of right size to pass internal tubing nut through external port
  - Need to get metal o-ring replacement seals for SAE fitting
  - External weld fitting is not off the shelf, will need to be designed/coordinated with pressure vessel vendor

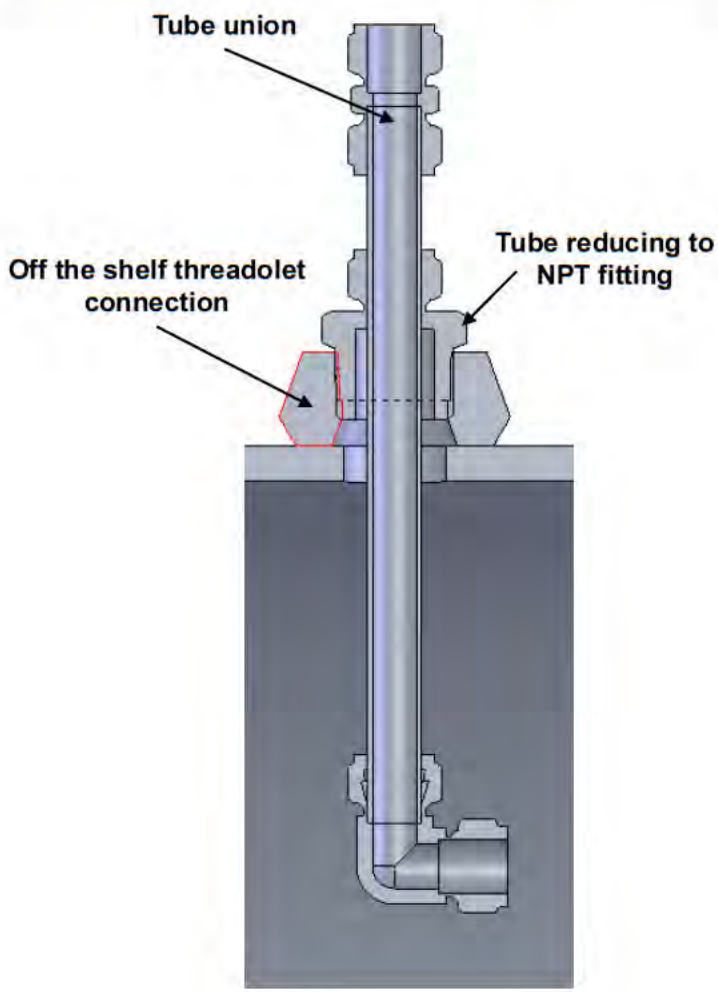
UPDATE 8/10/12:
OPTION 1 CHOSEN MOVING FORWARD

Fluid Porting Option #2: Butt or Socket Weld



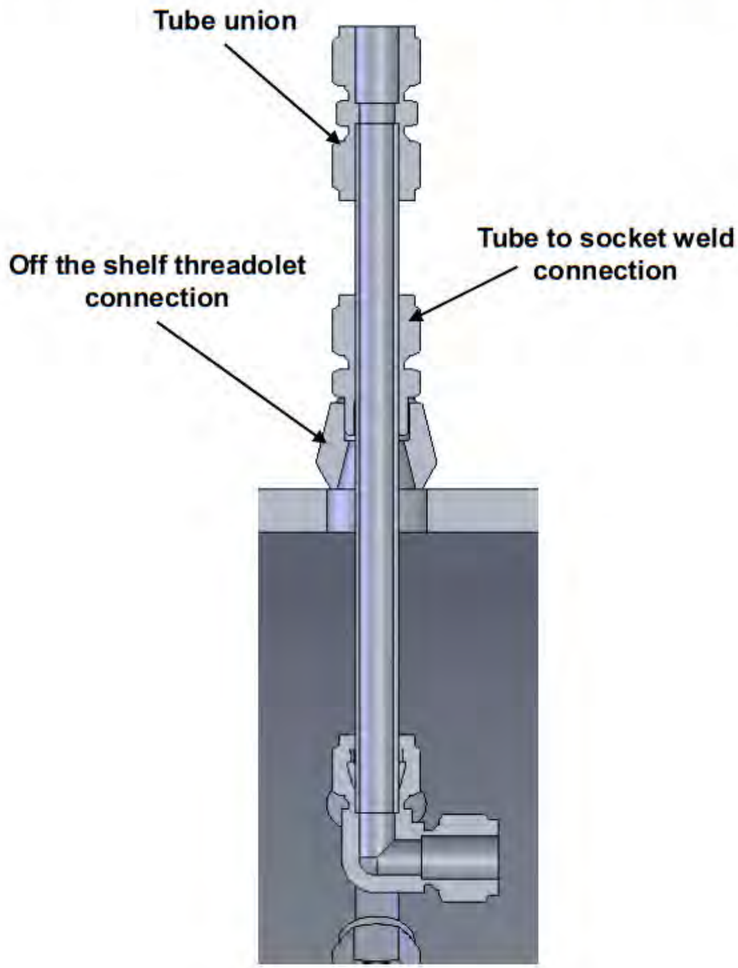
- Pros
  - Easy removal of external case connections
  - Internal connection will pass through case port
  - Standard external pressure rated case ports exist (weldolet, sockolet, etc)
- Cons
  - Very large fittings needed to pass internal fittings
  - Long lead (~14 weeks) needed to get fittings of right size to pass internal tubing nut through external port
  - Reducing fitting is also long lead

Fluid Porting Option #3: NPT fitting



- Pros
  - Relatively easy removal of external case connections, but not as good as straight thread or tube connection
  - Internal connection will pass through case port
  - Standard external pressure rated case ports exist (weldolet, sockolet, etc)
- Cons
  - NPT is not a good fitting to use for SS on SS where connections are broken often
  - Long lead (~14 weeks) needed to get fittings of right size to pass internal tubing nut through external port

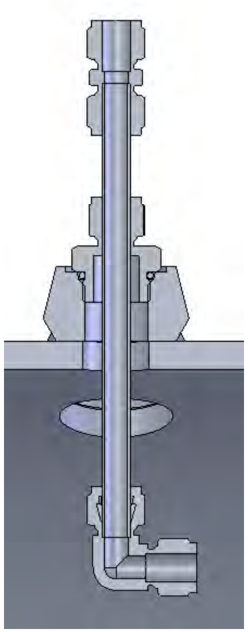
Fluid Porting Option #4: Small socket weld



- Pros
  - Smallest fitting option available
  - No long lead fittings required
- Cons
  - Internal fittings will need to be made up in the case (tubing nut cannot pass through external port)
  - Removal of combustor will require cutting tubing inside pressure case

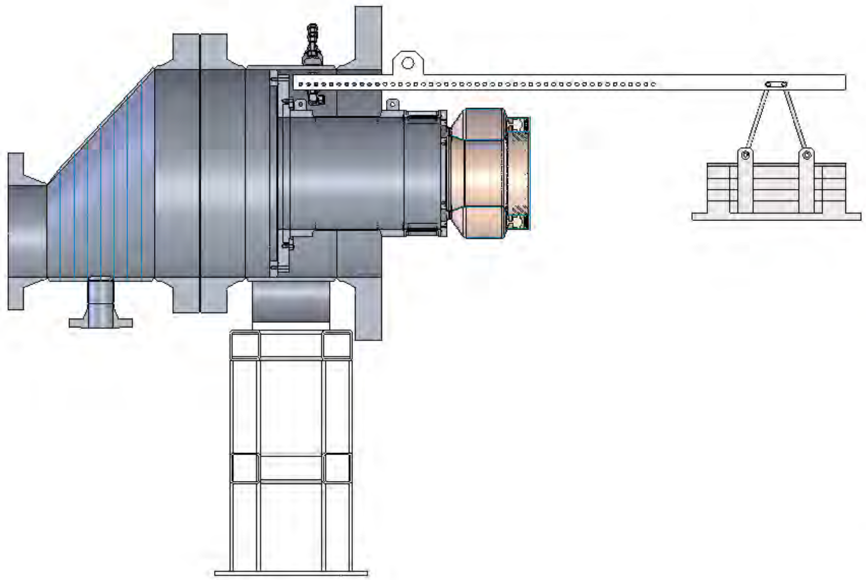
Fluid Porting Options

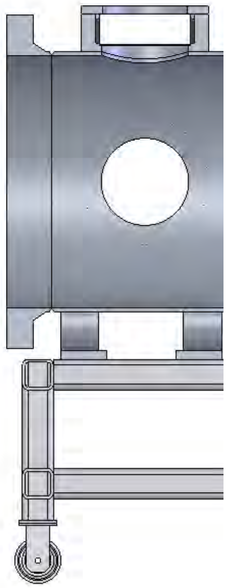
- Option 1 (SAE fitting) chosen as best option to allow ease of access and flexibility without risk to instrumentation case vessel
- If case stamping is a concern due to ports having to be made to fit the SAE fitting we can use one of the socket weld or NPT option with standard external ports (this will need to be coordinated with pressure vessel vendor)



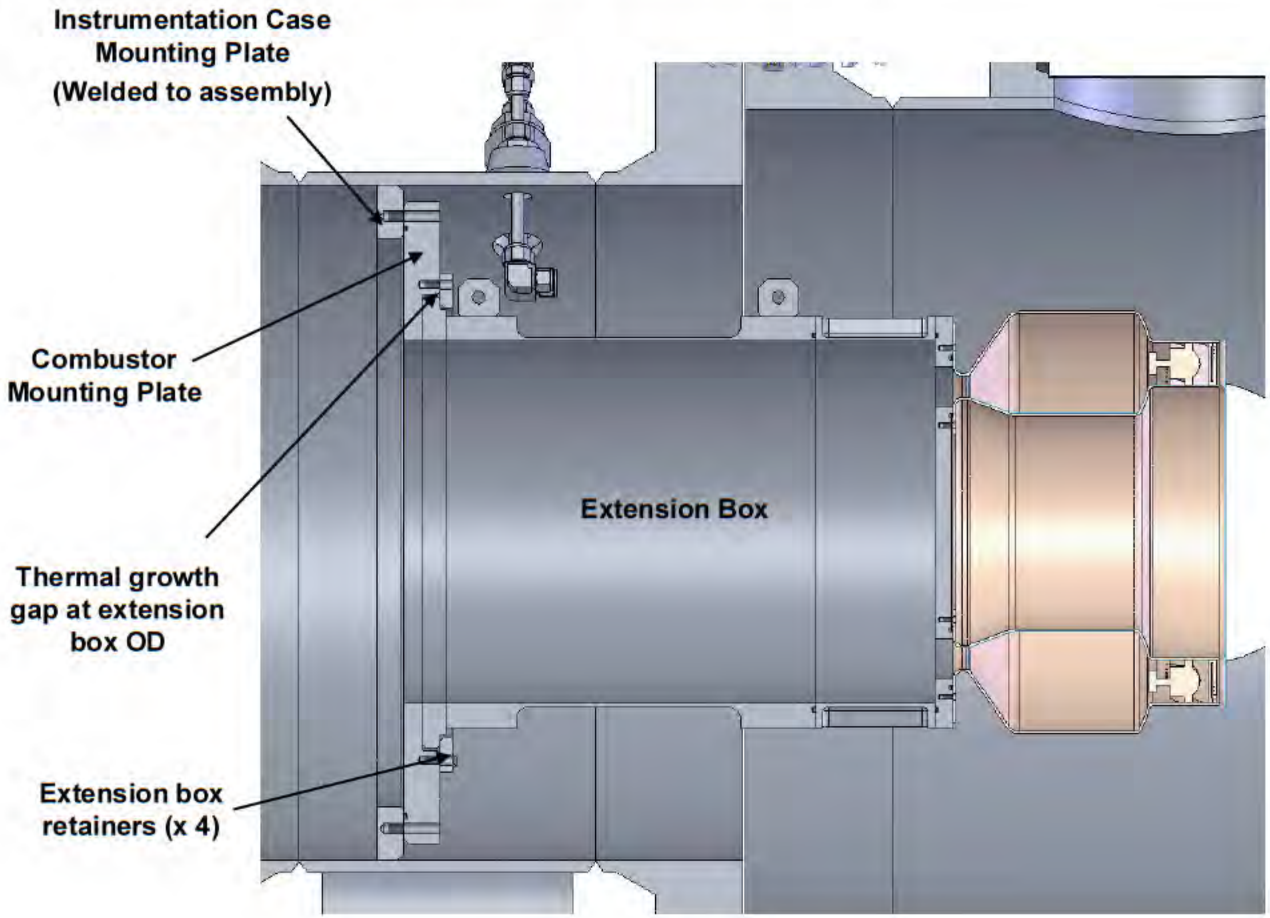
Combustor Installation/Removal

- **Combustor test piece changes.**
  - **Remove inlet pipe spool.**
  - **Unbolt flange between test section and instrumentation case.**
    - **Test section and inlet plenum will be mounted on a wheeled cart.**
    - **Cart will be most likely be a weldment using rectangular steel channel.**
    - **Wheeled cart may require track.**
  - **Roll test section and inlet plenum away from instrumentation case to allow access to combustor test pieces.**
    - **Could use a pneumatic system to push / pull the cart.**
  - **Use lifting bar and counterweight to install/remove extension box and combustor from pressure vessel**

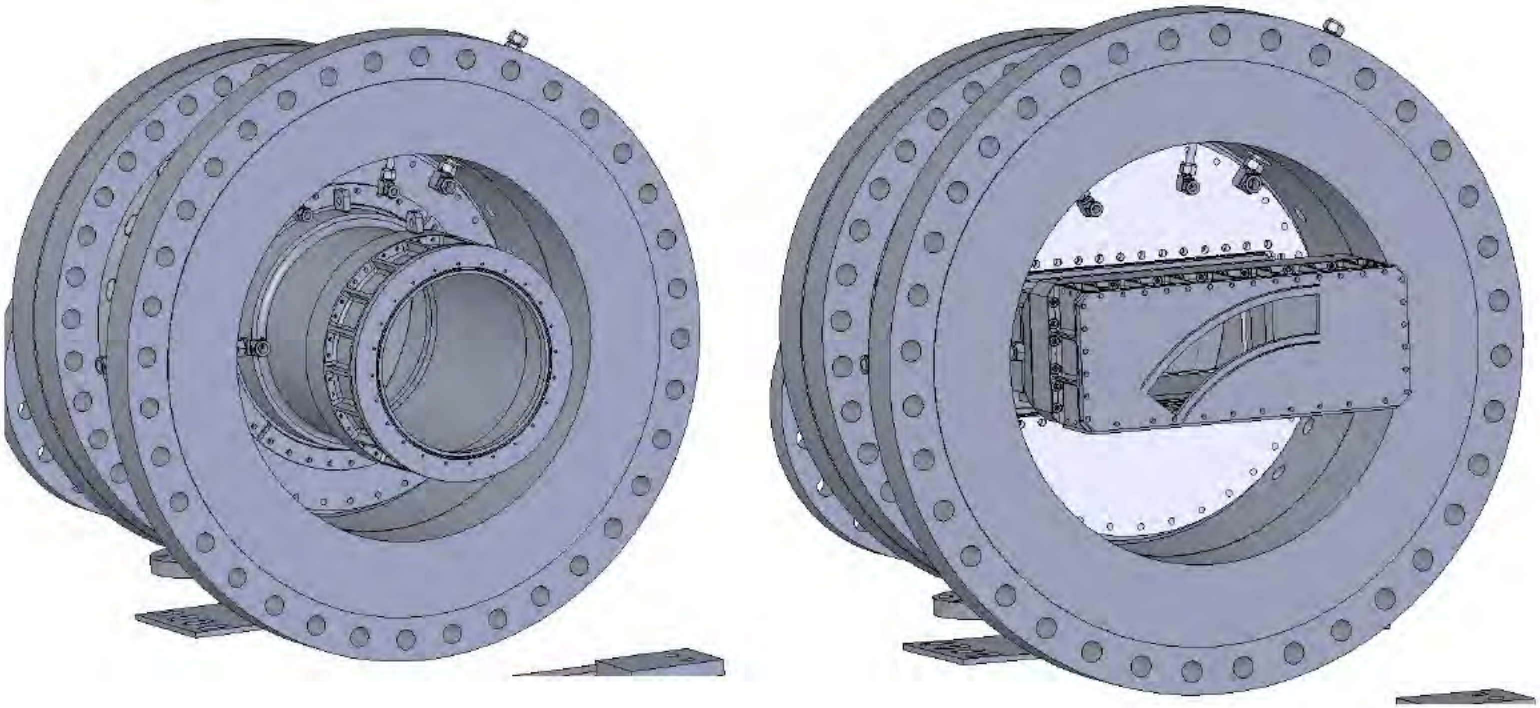




Combustor Mounting Hardware



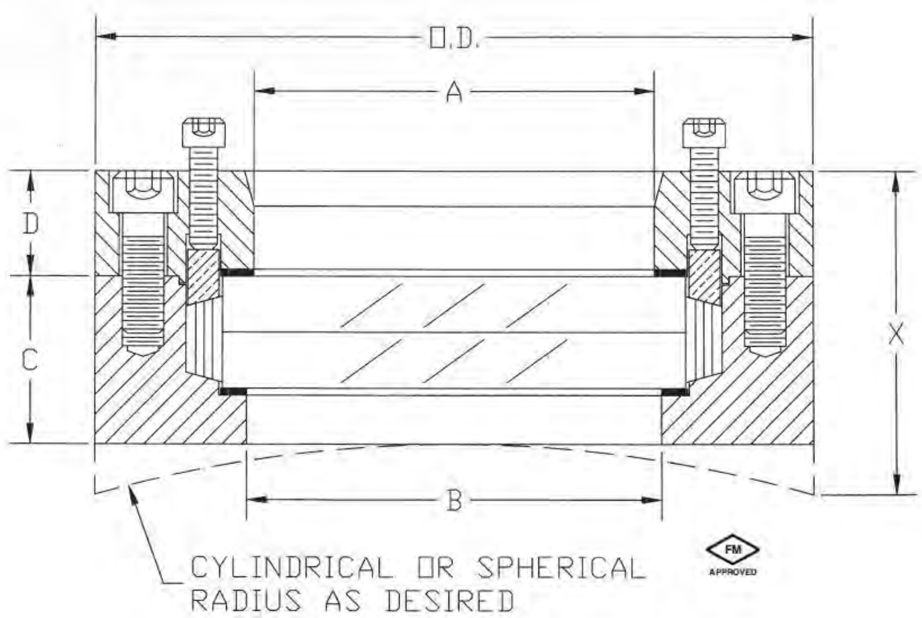
Combustor Mounting Hardware



Having a welded mounting plate inside the instrumentation case allows ease of changing the combustor mounting plate to fit different style combustors

Window Design (no update from CDR)

- **Pressure Products Window**
  - **Model B Welded Sight Glass**
  - **Fused silica or quartz**
  - **Similar in concept to AFRL design, glass is sandwiched between two metal frames with gasket material on both sides.**



Design Criteria (no change from CDR)

$$t = \frac{P * R}{S * E - 0.6 * P}$$

- **Example wall thickness calculation**
  - **t – Required wall thickness (in)**
  - **P – MAWP (psi)**
  - **R – Inner radius of wall (in)**
  - **S – Allowable stress per the boiler pressure code (psi)**
  - **E – Joint efficiency factor, dependent on weld inspections**
- **Design criteria**
  - **P = 300 psi**
  - **R = 16.500 in (radius of inlet plenum, largest diameter component)**
  - **S = 14,000 psi (@ 1000 F for ASTM A240 Gr 304)**
  - **E = 1 (dependent on weld test procedure, need to confirm with vendor)**
  - **t = 0.358 in (all vessels currently designed with 0.500 in thick walls)**
    - **0.500 in wall thickness satisfies this equation if E = 0.85.**

Design Criteria (no change from CDR)

$$t = d * \sqrt{\frac{5 * P}{S}}$$

- **Example window thickness calculation**
  - **This equation seems to be fairly standard for the industry.**
  - **t – Required window thickness (in)**
  - **d – Diameter of window (in)**
  - **P – MAWP (psi)**
  - **S – Tensile strength of glass (psi)**
- **Design criteria**
  - **P = 300 psi**
  - **d = 12 in (desired target, would accept 10”)**
  - **S = 10,000 psi (value was given by vendor some other literature suggests 7,000)**
  - **t = 4.650 in**
    - **This is much thicker (~2X) than what is currently in use at AFRL.**
    - **A 12” window is roughly 20% more area than the AFRL window.**
    - **AFRL window designed to 300 psi and 1100 F, although has only been used up to 250 psi.**
    - **This thickness is outside of the proposed suppliers history, although window material in this thickness is available.**
    - **Even limiting the design to 150 psi would produce a much thicker window than AFRL (3.300”).**
    - **Need to resolve this difference.**

Alaskan Copper Quote from CDR

- **Budgetary Quotation**
  - **Quoted by Don Rosen (Ramgen shareholder) at Alaskan Copper**
  - **Inlet Spool – $2,000**
  - **Inlet Plenum - $26,000**
  - **Test Section – $45,000 (includes estimate of $5,000 per sight glass)**
    - **Updated to $100,000 in later email**
  - **Instrumentation Case - $18,000 (includes provision for 17 passages through pressure wall)**
  - **Exhaust Reducer - $12,000 (design will most likely become eccentric, concentric shown, per verbal feedback eccentric design should not have a significant affect on cost)**
  - **Assembly - $8,000 (includes bolting, gaskets, hydro test, and ASME documentation)**
  - **Total - $111,000**
    - **No physical vessel support material / labor is currently included.**
  - **Lead time – Estimated at 20 weeks after final design.**
    - **Lead time estimate includes a 16 week guess on sight glass delivery.**

Work Plan / Analysis Tasks

- **Update air/fuel system design and re-size core buster hardware**
- **Finalize instrumentation case ports per review feedback**
- **Send pressure vessel package out for quote to 3 vendors**
- **Pursue final design based on feedback from chosen vendor**
- **Work internal components and bring to PDR level**
- **Work/analyze pressure case stand and bring to PDR level**
- **FEA**
  - **Investigate actual window thickness requirements.**
  - **Ideally vendor will complete actual structural design, need to review methods and cross check with our own calculations**
- **CAD**
  - **Model all of the air, fuel, water interfaces.**
  - **Model more realistic combustor component support and duct work based on the actual potential combustor geometry.**

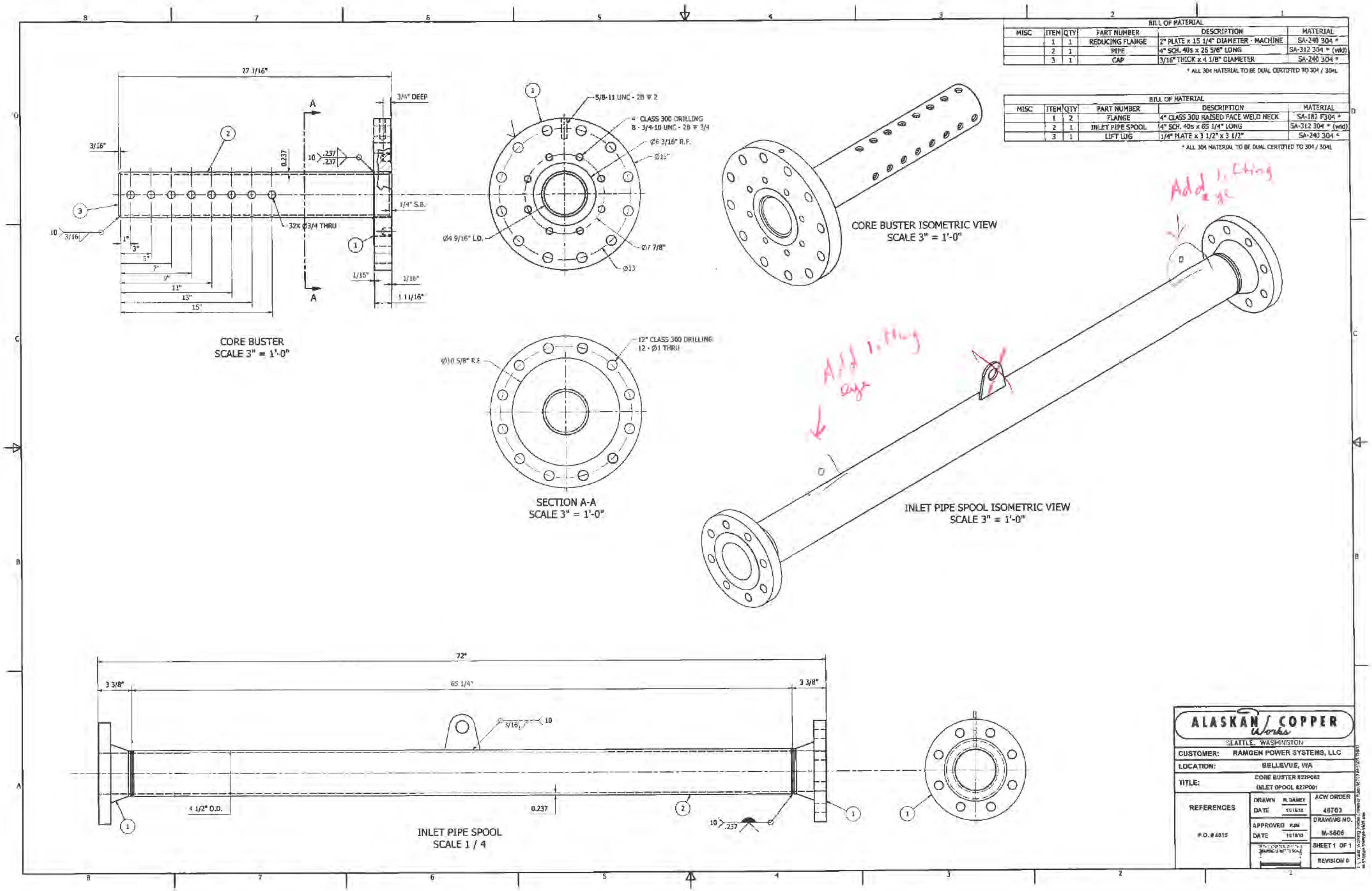
Schedule

- Vendor Selection – 8/31/2012, or earlier if possible
- FDR date – 9/15/2012
  - Alaskan Copper estimates ~ 2 weeks from PO for final drawing review.
  - Subject to vendor feedback as final design work is likely to be completed by vendor.
- Drawing Release date – Vendor driven
- Estimated Manufacturing Time/Delivery date – 5 months manufacturing time, 1/1/2012 delivery to Redmond, WA
- Any differences from master schedule? - No
- Is schedule achievable? Yes, assuming some overlap work completed in August.

Budget

- Current budget - $290K
- Alaskan Copper ROM

– Inlet Plenum	$26,000.00
– Test Section	$100,000.00
– Instrumentation Case	$18,000.00
– Exhaust Reducer	$12,000.00
– Assembly	$8,000.00
TOTAL	$164,000.00



BILL OF MATERIAL

MISC	ITEM	QTY	PART NUMBER	DESCRIPTION	MATERIAL
	1	1	REDUCING FLANGE	2" PLATE x 15 1/4" DIAMETER - MACHINE	SA-240 304 *
	2	1	PIPE	4" SCH. 40s x 26 5/6" LONG	SA-312 304 * (wld)
	3	1	CAP	3/16" THICK x 4 1/8" DIAMETER	SA-240 304 *

* ALL 304 MATERIAL TO BE DUAL CERTIFIED TO 304 / 304L

BILL OF MATERIAL

MISC	ITEM	QTY	PART NUMBER	DESCRIPTION	MATERIAL
	1	2	FLANGE	4" CLASS 300 RAISED FACE WELD NECK	SA-182 F304 *
	2	1	INLET PIPE SPOOL	4" SCH. 40s x 65 1/4" LONG	SA-312 304 * (wld)
	3	1	LIFT LUG	1/4" PLATE x 3 1/2" x 3 1/2"	SA-240 304 *

* ALL 304 MATERIAL TO BE DUAL CERTIFIED TO 304 / 304L

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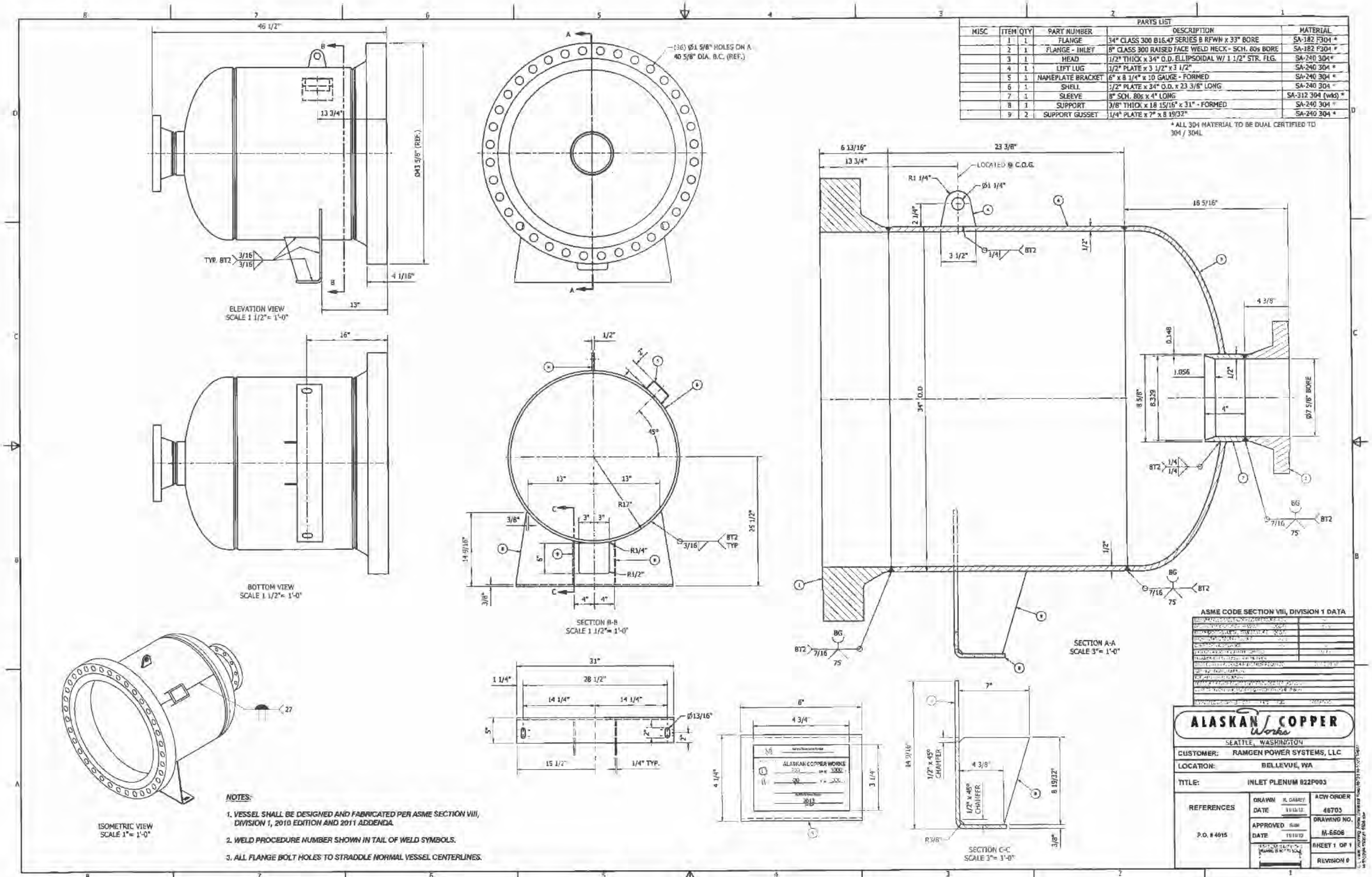


MISC	ITEM	QTY	PART NUMBER	DESCRIPTION	MATERIAL
	1	1	FLANGE	34" CLASS 300 B16.47 SERIES B RFWN x 33" BORE	SA-182 F304 *
	2	1	FLANGE - INLET	8" CLASS 300 RAISED FACE WELD NECK - SCH. 80s BORE	SA-182 F304 *
	3	1	HEAD	1/2" THICK x 34" O.D. ELLIPSOIDAL W/ 1 1/2" STR. FLG.	SA-240 304 *
	4	1	LIFT LUG	1/2" PLATE x 3 1/2" x 3 1/2"	SA-240 304 *
	5	1	NAMEPLATE BRACKET	6" x 8 1/4" x 10 GAUGE - FORMED	SA-240 304 *
	6	1	SHELL	1/2" PLATE x 34" O.D. x 23 3/8" LONG	SA-240 304 *
	7	1	SLEEVE	8" SCH. 80s x 4" LONG	SA-312 304 (wld) *
	8	1	SUPPORT	3/8" THICK x 18 15/16" x 31" - FORMED	SA-240 304 *
	9	2	SUPPORT GUSSET	1/4" PLATE x 7" x 8 19/32"	SA-240 304 *

PARTS LIST

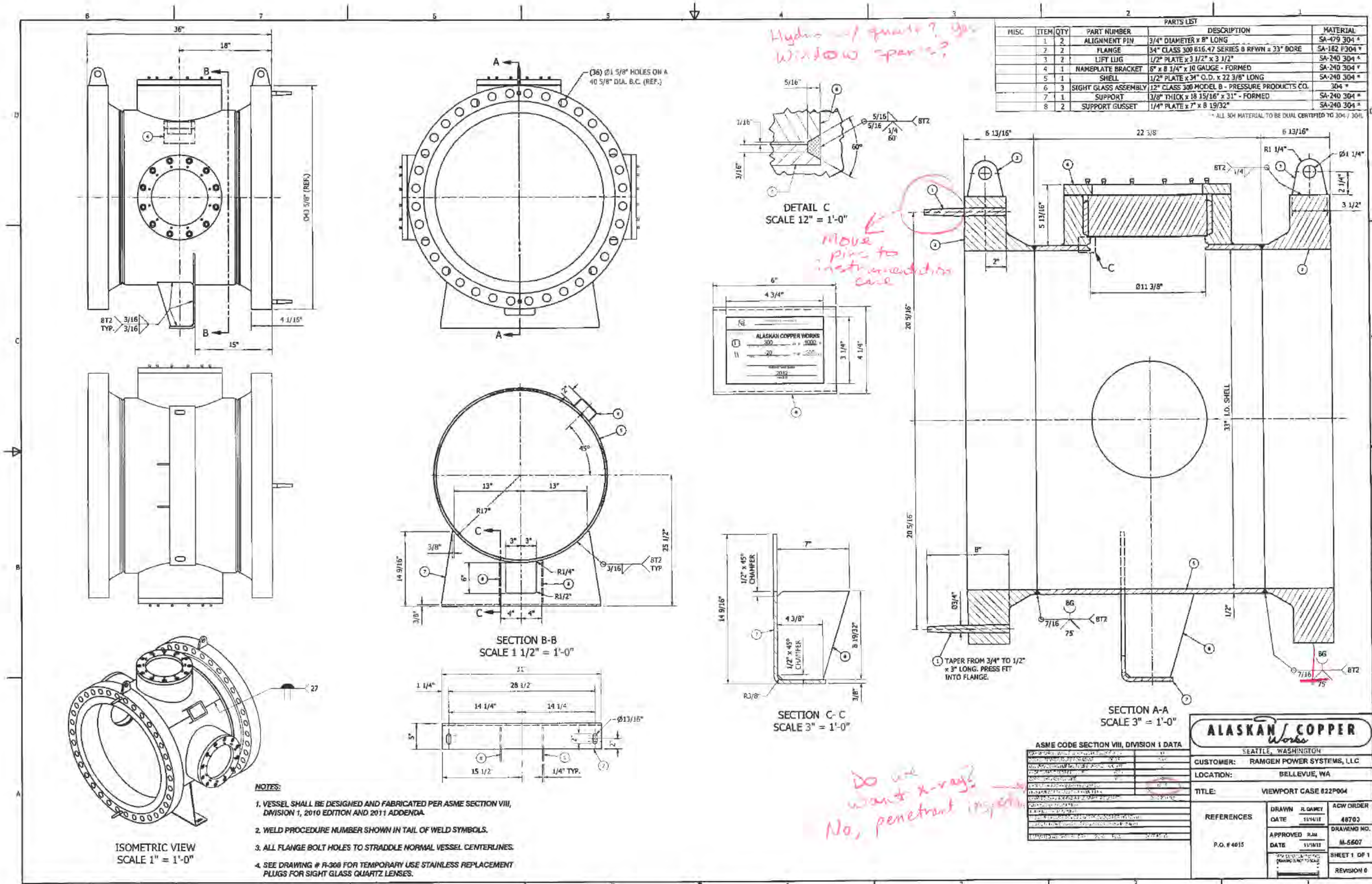
* ALL 304 MATERIAL TO BE DUAL CERTIFIED TO 304 / 304L

NOTES:

1. VESSEL SHALL BE DESIGNED AND FABRICATED PER ASME SECTION VIII, DIVISION 1, 2010 EDITION AND 2011 ADDENDA.
2. WELD PROCEDURE NUMBER SHOWN IN TAIL OF WELD SYMBOLS.
3. ALL FLANGE BOLT HOLES TO STRADDLE NORMAL VESSEL CENTERLINES.

CUSTOMER: RAMGEN POWER SYSTEMS, LLC
LOCATION: BELLEVUE, WA
TITLE: INLET PLENUM 822P003
REFERENCES
P.O. # 4015
DRAWN K. DAMEY
APPROVED
ACW ORDER 48703
DRAWING NO. M-6506
SHEET 1 OF 1
REVISION 0

G32



PARTS LIST

MISC	ITEM	QTY	PART NUMBER	DESCRIPTION	MATERIAL
	1	2	ALIGNMENT PIN	3/4" DIAMETER x 8" LONG	SA-479 304 *
	2	2	FLANGE	34" CLASS 300 B16.47 SERIES B RFWN x 33" BORE	SA-182 F304 *
	3	2	LIFT LUG	1/2" PLATE x 3 1/2" x 3 1/2"	SA-240 304 *
	4	1	NAMEPLATE BRACKET	6" x 8 1/4" x 10 GAUGE - FORMED	SA-240 304 *
	5	1	SHELL	1/2" PLATE x 34" O.D. x 22 3/8" LONG	SA-240 304 *
	6	3	SIGHT GLASS ASSEMBLY	12" CLASS 300 MODEL B - PRESSURE PRODUCTS CO.	304 *
	7	1	SUPPORT	3/8" THICK x 18 15/16" x 31" - FORMED	SA-240 304 *
	8	2	SUPPORT GUSSET	1/4" PLATE x 7" x 8 19/32"	SA-240 304 *

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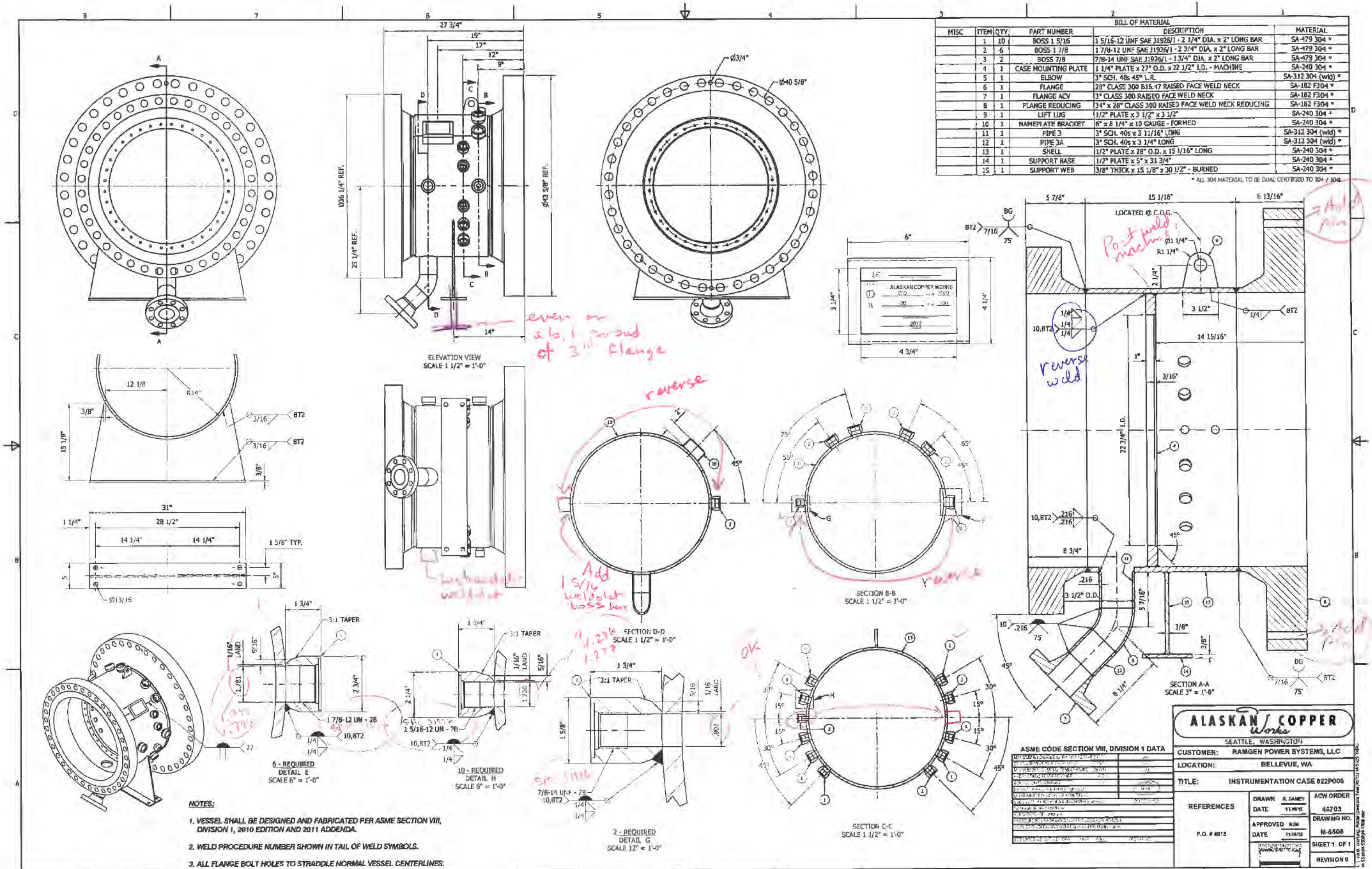
4. SEE DRAWING # R-308 FOR TEMPORARY USE STAINLESS REPLACEMENT PLUGS FOR SIGHT GLASS QUARTZ LENSES.



MISC	ITEM	QTY	PART NUMBER	DESCRIPTION	MATERIAL
	1	10	BOSS 1 5/16	1 5/16-12 UNF SAE J1926/1 - 2 1/4" DIA. x 2" LONG BAR	SA-479 304 *
	2	6	BOSS 1 7/8	1 7/8-12 UNF SAE J1926/1 - 2 3/4" DIA. x 2" LONG BAR	SA-479 304 *
	3	2	BOSS 7/8	7/8-14 UNF SAE J1926/1 - 1 3/4" DIA. x 2" LONG BAR	SA-479 304 *
	4	1	CASE MOUNTING PLATE	1 1/4" PLATE x 27" O.D. x 22 1/2" I.D. - MACHINE	SA-240 304 *
	5	1	ELBOW	3" SCH. 40s 45° L.R.	SA-312 304 (wld) *
	6	1	FLANGE	28" CLASS 300 B16.47 RAISED FACE WELD NECK	SA-182 F304 *
	7	1	FLANGE ACV	3" CLASS 300 RAISED FACE WELD NECK	SA-182 F304 *
	8	1	FLANGE REDUCING	34" x 28" CLASS 300 RAISED FACE WELD NECK REDUCING	SA-182 F304 *
	9	1	LIFT LUG	1/2" PLATE x 3 1/2" x 3 1/2"	SA-240 304 *
	10	1	NAMEPLATE BRACKET	6" x 8 1/4" x 10 GAUGE - FORMED	SA-240 304 *
	11	1	PIPE 3	3" SCH. 40s x 3 11/16" LONG	SA-312 304 (wld) *
	12	1	PIPE 3A	3" SCH. 40s x 3 1/4" LONG	SA-312 304 (wld) *
	13	1	SHELL	1/2" PLATE x 28" O.D. x 15 1/16" LONG	SA-240 304 *
	14	1	SUPPORT BASE	1/2" PLATE x 5" x 31 3/4"	SA-240 304 *
	15	1	SUPPORT WEB	3/8" THICK x 15 1/8" x 30 1/2" - BURNED	SA-240 304 *

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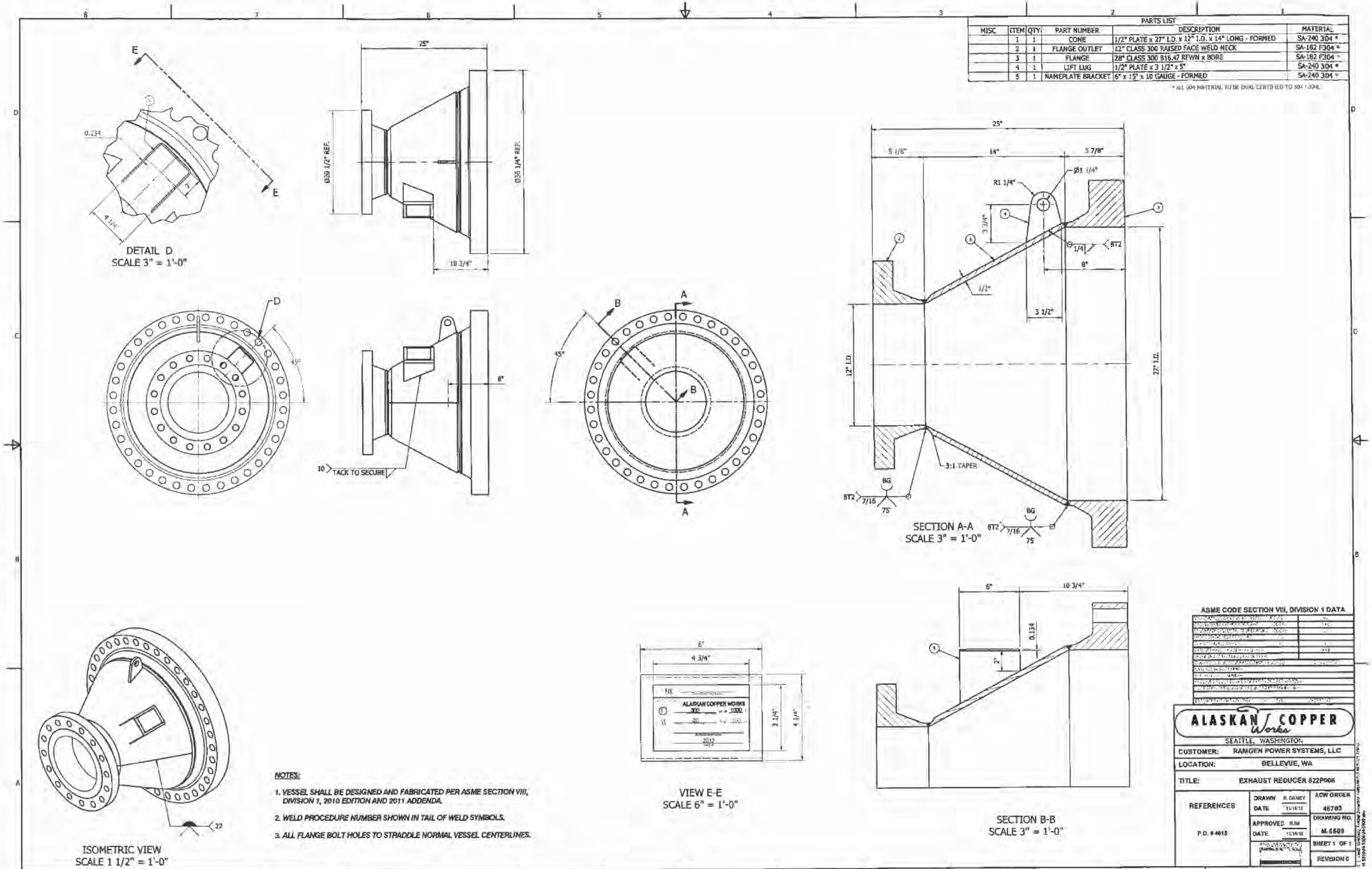
ALASKAN COPPER WORKS — SEATTLE, WASHINGTON

CUSTOMER:	RAMGEN POWER SYSTEMS, LLC
LOCATION:	BELLEVUE, WA
TITLE:	INSTRUMENTATION CASE 822P006
REFERENCES	P.O. # 4015
DRAWN	R. DAMEY
APPROVED	RJM
ACW ORDER	46703
DRAWING NO.	M-6508
SHEET	1 OF 1
REVISION	0

G34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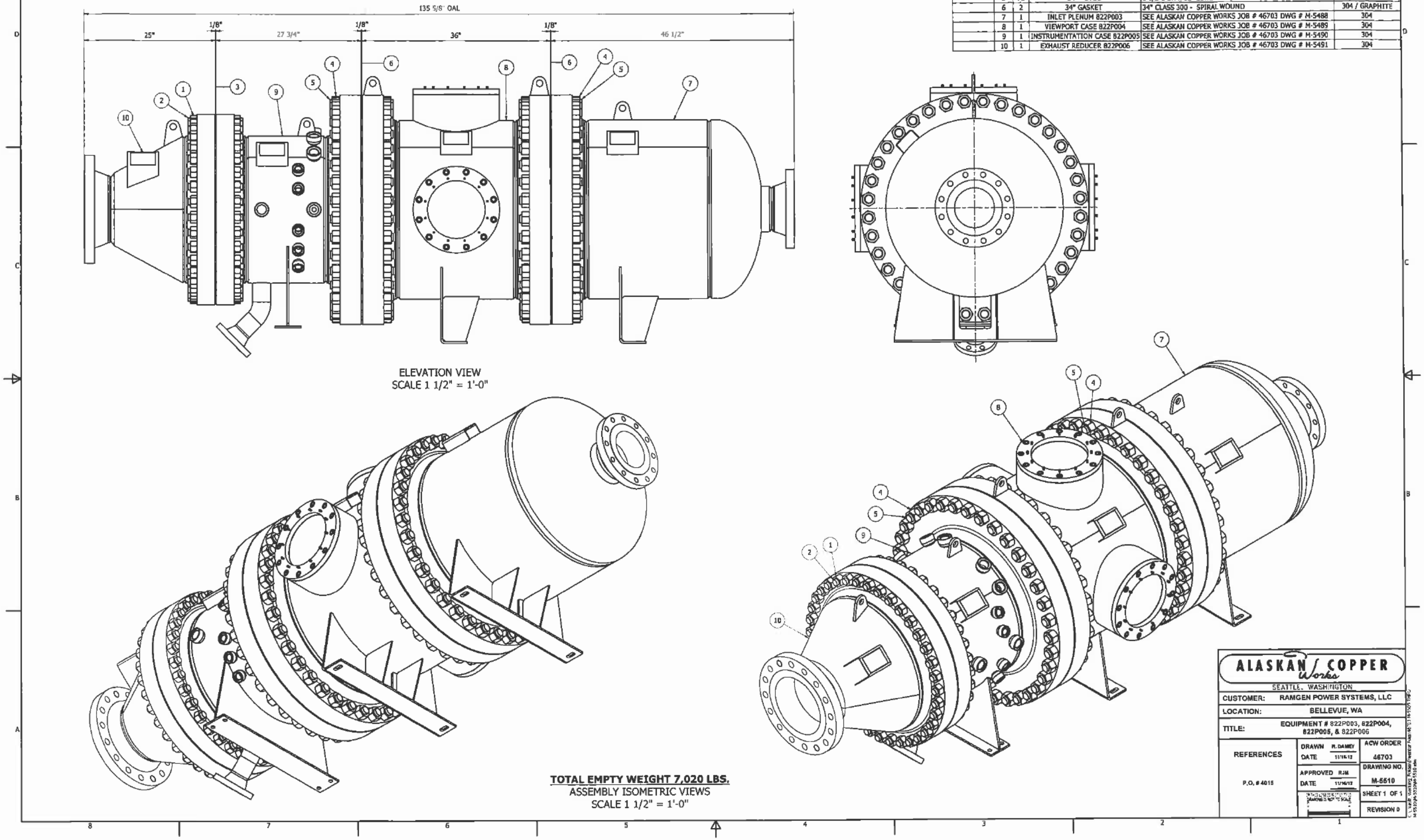
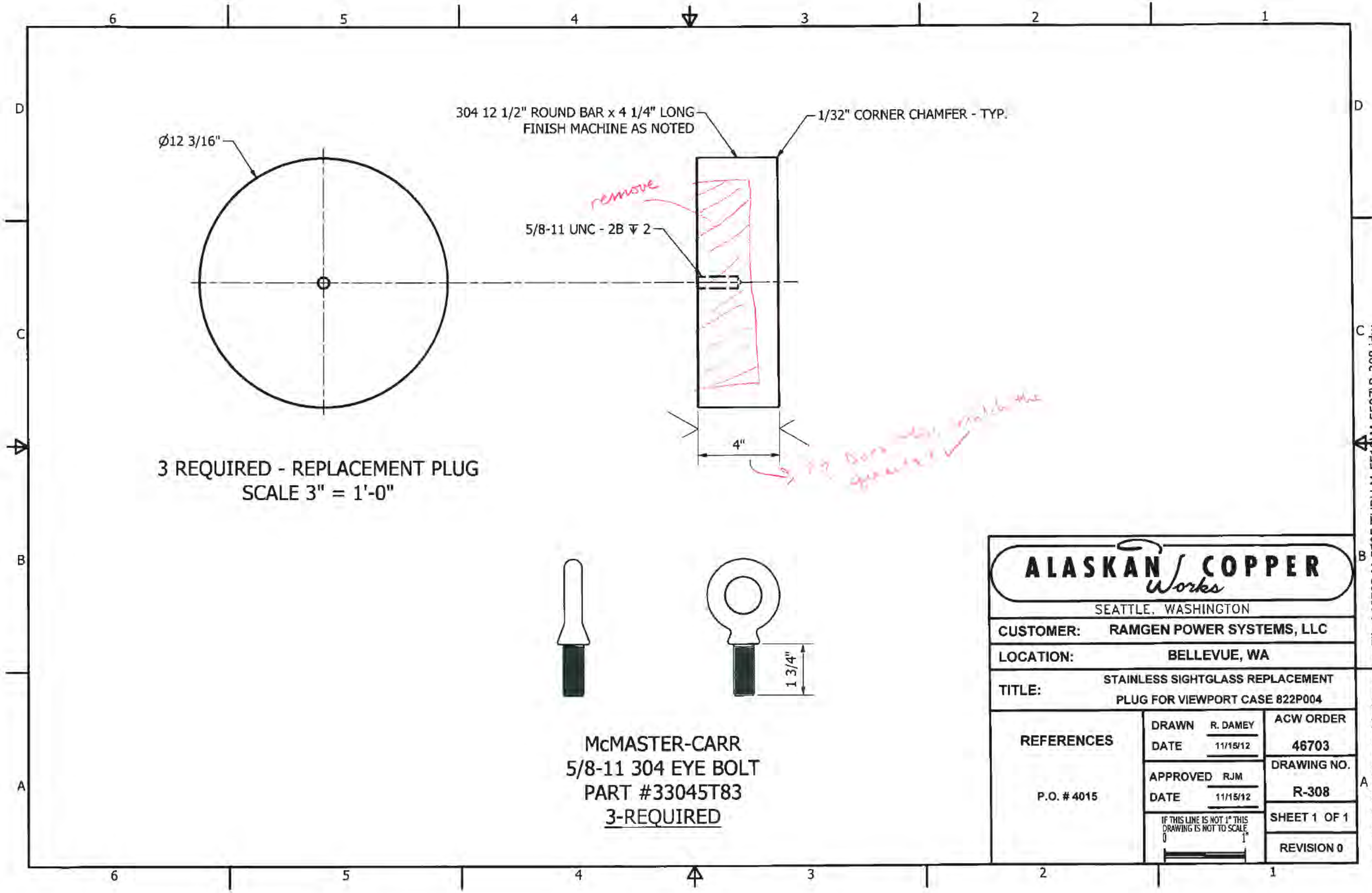
1 | 1 | CONE | 1/2" PLATE x 27" I.D. x 12" I.D. x 14" LONG - FORMED | SA-240 304 *
2 | 1 | FLANGE OUTLET | 12" CLASS 300 RAISED FACE WELD NECK | SA-182 F304 *
3 | 1 | FLANGE | 28" CLASS 300 B16.47 RFWN x BORE | SA-182 F304 *
4 | 1 | LIFT LUG | 1/2" PLATE x 3 1/2" x 5" | SA-240 304 *
5 | 1 | NAMEPLATE BRACKET | 6" x 15" x 10 GAUGE - FORMED | SA-240 304 *
DETAIL D
SCALE 3" = 1'-0"
SECTION A-A
SCALE 3" = 1'-0"
TACK TO SECURE
3:1 TAPER
VIEW E-E
SCALE 6" = 1'-0"
SECTION B-B
SCALE 3" = 1'-0"
ISOMETRIC VIEW
SCALE 1 1/2" = 1'-0"
ASME CODE SECTION VIII, DIVISION 1 DATA
NOTES:
1. VESSEL SHALL BE DESIGNED AND FABRICATED PER ASME SECTION VIII, DIVISION 1, 2010 EDITION AND 2011 ADDENDA.
2. WELD PROCEDURE NUMBER SHOWN IN TAIL OF WELD SYMBOLS.
3. ALL FLANGE BOLT HOLES TO STRADDLE NORMAL VESSEL CENTERLINES.
ALASKAN COPPER Works
SEATTLE, WASHINGTON
CUSTOMER: RAMGEN POWER SYSTEMS, LLC
LOCATION: BELLEVUE, WA
TITLE: EXHAUST REDUCER 822P006
REFERENCES
P.O. # 4015
DRAWN R. DANEY
APPROVED RJM
ACW ORDER 46703
DRAWING NO. M-5509
SHEET 1 OF 1
REVISION 0

G35

BILL OF MATERIAL

MISC	ITEM	QTY	PART NUMBER	DESCRIPTION	MATERIAL
	1	72	28" - NUT	1 1/4-8 UNC HEAVY HEX	SA-194 2H
	2	36	28" - STUD	1 1/4-8 UNC x 10 1/2" LONG	SA-193 Gr. B7
	3	1	28" GASKET	28" CLASS 300 - SPIRAL WOUND	304 / GRAPHITE
	4	144	34" - NUT	1 1/2-8 UN HEAVY HEX	SA-194 Gr. 2H
	5	72	34" - STUD	1 1/2-8 UN x 12" LONG	SA-193 Gr. B7
	6	2	34" GASKET	34" CLASS 300 - SPIRAL WOUND	304 / GRAPHITE
	7	1	INLET PLENUM 822P003	SEE ALASKAN COPPER WORKS JOB # 46703 DWG # M-5488	304
	8	1	VIEWPORT CASE 822P004	SEE ALASKAN COPPER WORKS JOB # 46703 DWG # M-5489	304
	9	1	INSTRUMENTATION CASE 822P005	SEE ALASKAN COPPER WORKS JOB # 46703 DWG # M-5490	304
	10	1	EXHAUST REDUCER 822P006	SEE ALASKAN COPPER WORKS JOB # 46703 DWG # M-5491	304




Ø12 3/16"
304 12 1/2" ROUND BAR x 4 1/4" LONG
FINISH MACHINE AS NOTED
1/32" CORNER CHAMFER - TYP.
remove
5/8-11 UNC - 2B ↧ 2
4"
3 REQUIRED - REPLACEMENT PLUG
SCALE 3" = 1'-0"
1 3/4"
McMASTER-CARR
5/8-11 304 EYE BOLT
PART #33045T83
3-REQUIRED
ALASKAN COPPER Works
SEATTLE, WASHINGTON
CUSTOMER: RAMGEN POWER SYSTEMS, LLC
LOCATION: BELLEVUE, WA
TITLE: STAINLESS SIGHTGLASS REPLACEMENT PLUG FOR VIEWPORT CASE 822P004
REFERENCES
P.O. # 4015
DRAWN R. DAMEY
DATE 11/15/12
APPROVED RJM
DATE 11/15/12
IF THIS LINE IS NOT 1" THIS DRAWING IS NOT TO SCALE
ACW ORDER 46703
DRAWING NO. R-308
SHEET 1 OF 1
REVISION 0
C:_Vault_Working_Folder\Inventor Files\46703 M-5505 THRU M-5510\M-5507\R-308.idw

PPENDIX 10.7

Exhaust Water Cooling System

SYMBOL LEGEND

SYMBOL	DESCRIPTION
	GAS PIPING
	CHILLED WATER SUPPLY PIPING
	CHILLED WATER RETURN PIPING
	PIPE DOWN
	PIPE UP
	BRANCH - TOP CONNECTION
	BRANCH - BOTTOM CONNECTION
	BRANCH - SIDE CONNECT ON
	CAP END OF PIPE
	FLOW DIRECTION
	GATE VALVE
	GLOBE VALVE
	BUTTERFLY VALVE
	PRESSURE REDUCING VALVE
	CHECK VALVE
	FLOW CONTROL VALVE
	TEMPERATURE & PRESSURE RELIEF VALVE
	BALL VALVE
	FLOAT VALVE
	PETES PLUG
	GAS COCK
	PUMP
	SUCTION DIFFUSER
	BALANCING VALVE
	FLEXIBLE PIPE CONNECTION
	BACKFLOW PREVENTER ASSEMBLY
	REDUCER
	FUNNEL DRAIN
	STRAINER W/ BLOWDOWN VALVE & 3/4" HOSE CONNECTION
	UNION
	3-WAY CONTROL VALVE
	3-WAY CONTROL VALVE
	DEMOLITION SCOPE OF WORK
	POINT OF CONNECTION TO E ISTING

SYMBOL	DESCRIPTION
	FLEXIBLE DUCT
	EXHAUST/RETURN AIR DUCT UP
	OUTSIDE AIR DUCT UP
	VOLUME DAMPER
	MOTORIZED DAMPER
	EXHAUST/RETURN AIR DUCT DOWN
	OUTSIDE AIR DUCT DOWN
	SOUNDLINED DUCT
	CEILING AIR TERM NAL ROUND)
	CEILING AIR TERM NAL SQUARE)
	BACKDRAFT DAMPER
	FLEXIBLE DUCT CONNECTION
	TURNING VANES
	SQUARE TO ROUND TRANSITION
	CONICAL DUCT TAKEOFF
	5" DUCT BRANCH TAKEOFF
12X12 CD-1 / 100 25	DIFFUSER CALLOUT (SIZE TYPE CFM 50 FPM THROW)
12/12 DL	DOOR LOUVER W/ S ZE
	HYDROGEN SENSOR
	CARBON DIOXIDE (CO2) SENSOR
	CARBON MONOXIDE (CO) SENSOR
T	THERMOSTAT
	TEMPERATURE SENSOR
	TEMPERATURE GAUGE
	PRESSURE SENSOR
	PRESSURE GAUGE
	FLOW SWITCH
M	MOTOR CONNECTION
	LOW WATER SENSOR
	NATURAL GAS SENSOR
	PROPANE SENSOR
VFD	VARIABLE FREQUENCY DRIVE
	RETURN/EXHAUST/TRANSFER AIRFLOW
	SUPPLY AIRFLOW
	LEVEL SWITCH

ABBREVIATION LIST

°F	DEGREES FAHRENHEIT
(E)	EXISTING
AABC	ASSOCIATED AIR BALANCE COUNCIL
AFF	ABOVE F NISHED FLOOR
AFUE	ANNUAL FUEL UTILIZATION EFFIC ENCY
ASSY	ASSEMBLY
AUX	AUXILIARY
BTU	BRIT SH THERMAL UNIT
BTUH	BRIT SH THERMAL UNITS PER HOUR
CFM	CUBIC FEET PER MINUTE
COP	COEFF CIENT OF PERFORMANCE
CPVC	CHLORINATED POLYVINYL CHLORIDE
CV	COEFF CIENT OF FLOW
CW	COLD WATER
DB	DRY BULB
DEG	DEGREE (FAHRENHEIT)
D A	DIAMETER
DN	DOWN
EAT	ENTERING AIR TEMPERATURE (DEGREES FAHRENHE T)
EC	ELECTRICAL CONTRACTOR
EDB	ENTERING DRY BULB TEMPERATURE (DEGREES FAHRENHE T)
EER	ENERGY EFFICIENCY RATIO
ESP	EXTERNAL STATIC PRESSURE
EWB	ENTERING WET BULB TEMPERATURE (DEGREES FAHRENHEIT)
EXH	EXHAUST
F	FAHRENHEIT
FIN	FINISHED
FLR	FLOOR
FPM	FEET PER MINUTE
FT	FEET
GAL	GALLON
GPM	GALLON PER MINUTE
GC	GENERAL CONTRACTOR
HP	HORSEPOWER
HSPF	HEAT NG SEASON PERFORMANCE FACTOR
IBC	INTERNATIONAL BUILDING CODE
D	INSIDE DIAMETER
MC	INTERNATIONAL MECHANICAL CODE
N	INCH
IPLV	INTEGRATED PART LOAD VALUE
KW	KILOWATT
LB	POUND
LBS	POUNDS
LF	L NEAR FEET
MAX	MAXIMUM
MBH	THOUSAND BRIT SH THERMAL UNITS PER HOUR
MC	MECHANICAL CONTRACTOR
MCA	MINIMUM CIRCUIT AMPACITY
MFR	MANUFACTURER
M N	MINIMUM
MOCP	MAXIMUM OVERCURRENT PROTECTION
N A	NOT APPLICABLE
NC	NO SE CRITER A
NEBB	NATIONAL ENVIRONMENTAL BALANCING BUREAU
NEC	NATIONAL ELECTR CAL CODE
NFPA	NATIONAL FIRE PROTECTION ASSOCIATION
N C	NOT IN CONTRACT
OD	OUTSIDE DIAMETER
OSA	OUTSIDE AIR
PCF	POUNDS PER CUB C FOOT
PD	PRESSURE DROP
PH	PHASE
PSI	POUNDS PER SQUARE INCH
PSIG	POUNDS PER SQUARE INCH GAUGE
PVC	POLYVINYL CHLORIDE
RPM	REVOLUT ONS PER MINUTE
SCFM	STANDARD CUBIC FEET PER MINUTE
SEER	SEASONAL ENERGY EFFIC ENCY RAT O
SMACNA	SHEET METAL AND AIR CONDITION NG CONTRACTORS NATIONAL ASSOCIATION
SP	STATIC PRESSURE
SQ FT	SQUARE FEET
TEMP	TEMPERATURE
TYP	TYP CAL
VD	VOLUME DAMPER
VFD	VARIABLE FREQUENCY DRIVE
VTR	VENT THROUGH ROOF
W	WATT
W	WITH
WB	WET BULB

GENERAL NOTES

1 SYSTEMS PROVIDED UNDER THIS CONTRACT ARE CLASSIFIED AS SIMPLE SYSTEMS PER THE WASHINGTON STATE NON-RESIDENTIAL ENERGY CODE PROVIDE COMMISSIONING PER SECTION 1 16 OF THAT CODE

2 ALL DUCTWORK SHALL BE 2 00 PRESSURE CLASS UNLESS SPEC FICALLY NOTED OTHERWISE IN THESE DRAWINGS OR IN THE SPECIFICATIONS

3 PROVIDE TURNING VANES IN ALL MITERED RECTANGULAR DUCT ELBOWS & TEES

PROVIDE BALANCING DEVICES ON ALL NEW AIR TERMINALS & OUTSIDE AIR DUCT SECTIONS

5 INSULATE ALL HEATING & COOLING DUCTWORK PER WASHINGTON STATE RESIDENTIAL ENERGY CODE REQU REMENTS

6 INSULATE ALL DUCTWORK SYSTEMS PER WASHINGTON STATE NON-RES DENT AL ENERGY CODE REQUIREMENTS

7 PROVIDE R-7 INSULATION FOR ALL OUTSIDE AIR DUCTWORK FROM OUTSIDE AIR DAMPER TO OUTS DE AIR INTAKE AT BUILDING EXTERIOR

8 LOCATE ALL VALVES TRAPS TEST PORTS DAMPERS CONTROLS CLEANOUTS ETC TO BE ACCESSIBLE FOR MA NTENANCE ADJUSTMENT & TESTING PROVIDE ACCESS PANELS FOR ALL CONCEALED DEV CES

9 PROVIDE 8W/FT HEAT TRACE FOR ALL HYDRON C PIP NG & P-TRAPS IN UNHEATED SPACES

10 INSULATE PIPING PER WASHINGTON STATE NON-RESIDENTIAL ENERGY CODE REQUIREMENTS

11 ALL ELECTRICAL WORK SHALL BE PERFORMED IN ACCORDANCE WITH NATIONAL ELECTRICAL CODE STANDARDS AND LOCAL REQUIREMENTS

12 ALL FIELD WIRING SHALL REQUIRE AN ELECTRICAL PERMIT AND SHALL BE PERFORMED BY A LICENSED ELECTR CIAN

13 THE F RE PROTECTION CONTRACTOR SHALL BE RESPONSIBLE FOR SYSTEM LAYOUT DESIGN AND COORDINATING PIPE PLACEMENT AND ROUTING OF PIPING W TH OTHER TRADES SHOP DRAWINGS SHALL COMPLY WITH NFPA REQUIREMENTS DUCTWORK AND OTHER P PING SYSTEMS HAVE ROUTING PR ORITY OVER F RE PROTECTION PIPING

1 SLOPE HOR ZONTAL FIRE PROTECTION PIPING TO FACILITATE DRAINAGE TO SYSTEM LOW POINTS PROVIDE DRAIN VALVES AT ALL SYSTEM LOW POINTS

15 PROVIDE COMPLETE CONDENSATE DRAINAGE SYSTEM FOR ALL PTAC UNITS FIELD ROUTE DRAINAGE PIP NG FROM EQUIPMENT TO NEAREST DRA N LOCATION (SUCH AS SERVICE SINK FUNNEL DRAIN ETC) SLOPE NON-PRESSURIZED DRAIN PIPING TO DRAIN LOCATION P PING & FITT NGS SHALL BE PVC OR CPVC USE CPVC FOR ALL PLENUM APPLICATIONS OR WHERE NONCOMBUST BLE P PING IS REQU RED MINIMUM PIPE SIZE SHALL BE 3/

16 TOTAL STATIC PRESSURE NOTED IN SCHEDULES SHALL BE ASSUMED TO NCLUDE DUCT SYSTEM TERMINAL UNITS F LTERS CO LS ETC

17 ALL RETURN/EXHAUST OUTSIDE AIR FILTERS SHALL BE MERV-7 RATED

18 AIR TERMINAL SIZES SHOWN ON PLANS ARE NECK SIZES PROVIDE ADDITIONAL PANS HARDWARE ETC REQUIRED TO INSTALL A R TERMINAL IN CEIL NG SYSTEM

19 WATER P PE CONNECTIONS TO HEAT EXCHANGERS SHALL BE MANUFACTURED & CONNECTED IN COUNTER-FLOW CONFIGURATION

20 AIR TERMINALS IN UNFINISHED SPACES OR OPEN CEILING AREAS SHALL BE INSTALLED AT 10 AFF UNLESS OTHERWISE NOTED ON THESE DRAWINGS

21 ALL PRESSURES LISTED ARE GAGE PRESSURES UNLESS OTHERWISE NOTED

22 SEISMIC PROVIS ONS ARE REQUIRED - REFER TO SPECIFICATIONS

23 THE FIRST FIGURE OF DUCT SIZE CALLOUTS INDICATES DIMENSION OF FACE SHOWN OR INDICATED DUCT S ZES ARE NET INSIDE DIMENSIONS

2 PROVIDE AUTOMATIC A R VENTS AT ALL LOCAL HIGH POINTS FOR COOLING WATER SYSTEMS

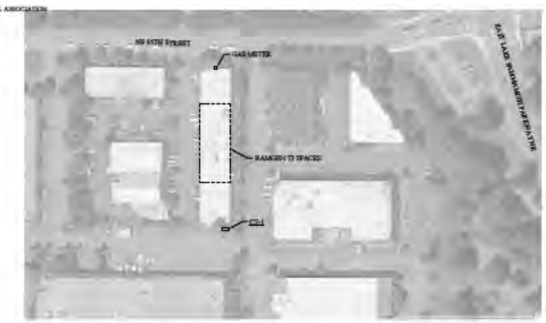
25 A SHORT DASH IN A SCHEDULE TABLE CELL INDICATES THAT THE COLUMN HEADING S NOT USED OR NOT APPLICABLE TO THAT SCHEDULED ITEM

26 COORDINATE LOCATIONS OF ALL CEILING-MOUNTED EQUIPMENT DUCTS A R TERMINALS ETC WITH L GHT F XTURE LOCATIONS

27 MAINTA N AND RESTORE (IF INTERRUPTED) ALL CONDUITS & CONDUCTORS PIPING & DUCTWORK PASSING THROUGH RENOVATED AREAS AND SERVICING UNDISTURBED AREAS

28 REMOVE ALL ABANDONED DUCTWORK PIP NG CONTROLS W RING ETC WHERE ACCESSIBLE IN RENOVATED AREAS

29 WHERE CONTROLS ARE DEMOLISHED REMOVE WIRING BACK TO NEAREST CONTROL PANEL OR JUNCTION BOX REMOVE ACCESSIBLE CONDUIT JUNCTION BOXES ETC



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RO ECT IT E

RAMGEN POWER SYSTEMS

REDMOND RESEARCH AND DEVELOPMENT CENTER

RO ECT ADDRESS

7825 65TH STREET NE
SUITE A- 55
REDMOND WA 98052

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LEGEND, ABBREV AT ONS LIST, & GENERAL NOTES

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EXISTING EQUIPMENT SCHEDULE

SYMBOL	TYPE	MANUFACTURER	MODEL	TYPE	SERVICE	DESCRIPTION DATA	WORK SCOPE	NOTES
EUH 1	GAS UNIT HEATER	REZNOR			SPACE HEATING		EX STING TO REMAIN	
EUH 2	GAS UNIT HEATER	REZNOR			SPACE HEATING		EX STING TO REMAIN	
EUH 3	GAS UNIT HEATER	DAYTON			SPACE HEATING		EX STING TO REMAIN	
EUH 4	GAS UNIT HEATER	REZNOR			SPACE HEATING		EX STING TO REMAIN	
EUH 5	GAS UNIT HEATER	REZNOR			SPACE HEATING		EX STING TO REMAIN	
EUH 6	GAS UNIT HEATER	REZNOR			SPACE HEATING		EX STING TO REMAIN	
EEF 1	EXHUAST FAN			CEILING MOUNTED	BATHROOM EXHAUST		DEMO	
EEF 2	EXHUAST FAN			CEILING MOUNTED	BATHROOM EXHAUST		DEMO	
EEF 3	EXHUAST FAN			CEILING MOUNTED	BATHROOM EXHAUST		DEMO	
EEF 4	EXHUAST FAN			CENTR FUGAL HUNG	GAS STORAGE EXHAUST		DEMO	
EEF 5	EXHUAST FAN			CENTR FUGAL HUNG	GAS STORAGE EXHAUST		DEMO	
EEF 6	EXHUAST FAN			WALL MOUNT	AREA EXHAUST		EX STING TO REMAIN	
EEF 7	EXHUAST FAN			CENTR FUGAL HUNG	ARE A EXHAUST		DEMO	
EEF 8	EXHUAST FAN			WALL MOUNT	ARE A EXHAUST		EX STING TO REMAIN	
EAD 1	AIR DRYER				COMPRESSED AIR			
EC 1	AIR COMPRESSOR				COMPRESSED AIR			
EC 2	AIR COMPRESSOR				COMPRESSED AIR			
EL 1	LOUVER			WALL MOUNT	VENTILATION			
EL 2	LOUVER			WALL MOUNT	VENTILATION			
EL 3	LOUVER			WALL MOUNT	VENTILATION			
EAHU 1	AIR HANDLER				VENTILATION			
EPTAC 1	PTAC UNIT			WALL MOUNT	OFFICE CONDIT ONING			
EPTAC 2	PTAC UNIT			WALL MOUNT	OFFICE CONDIT ONING			
ED 1	DIFFUSER/GRILL			DUCT MOUNT/ CEIL NG	VENTILATION			

NOTES

HEAT EXCHANGER SCHEDULE

SYMBOL	MANUFACTURER	MODEL	SERVICE	HEAT LOAD (BTUH)	SHELL SIDE SYSTEM	GPM	EWT	LWT	PD (FT)	TUBE SIDE SYSTEM	LBS/HR	EWT	LWT	PD (FT)	NOTES
HEX001	RP ADAMS CO	AR 18014 1 NS	AFTERCOOLER	2 237 870	WATER	149	80	110	3 50	CO2	21 600	346	100	2 00	1
HEX002	RP ADAMS CO	AR 12022 1 NS	AFTERCOOLER	1 186 852	WATER	158	80	95	1 10	CO2	11 520	346	100	0 49	1
HEX003	RP ADAMS CO	SAF 105 NS	AFTERCOOLER	186 518	WATER	26 6	80	94	1 00	CO2	1 800	346	100	0 60	1
HEX004	RP ADAMS CO	SAF 105 NS	AFTERCOOLER	186 518	WATER	26 6	80	94	1 00	CO2	1 800	346	100	0 60	1

NOTES
1 CONSTRUCT PER ASME SECTION V II DIV SION 1

ELECTRIC UNIT HEATER SCHEDULE

SYMBOL	MANUFACTURER	MODEL	FAN CFM	FAN HP	FAN W	HEATER KW	ELECTRICAL VOLT	PHASE	FLA	MOC	WEIGHT (LBS)	NOTES
UH 1	INDEECO	MCI	100			1 125	208	1	6 8	15	24	1

NOTES
1 UNIT OPERATES MA NTA NS A SHED TEMPERATURE OF 40 DEGREES FAHRENHEIT

ROOF HOOD SCHEDULE

SYMBOL	MANUFACTURER	MODEL	PERFORMANCE	NOTES
GH 1	GREENHECK	FGI	5000 CFM @ 0 09 SP	1
GH 2	GREENHECK	FGI	5000 CFM @ 0 09 SP	1
GH 3	GREENHECK	FGI	5000 CFM @ 0 09 SP	1
RC 1	GREENHECK	RCC	7 THROAT	
RC 2	GREENHECK	RCC	7 THROAT	

NOTES
1 30 X36 THROAT

AIR TERMINAL SCHEDULE

SYMBOL	MANUFACTURER	MODEL	MOUNTING TYPE	STYLE	NOTES
ED 1			DUCT MOUNTED	EXISTING	
RD 1	TITUS	50RL	DUCT MOUNTED	RETURN GRILLE	
SD 1	TITUS	301RS	DUCT MOUNTED	SUPPLY GRILLE	

NOTES
PROVIDE AIR BALANCING DEVICE AT EACH AIR TERMINAL

PUMP SCHEDULE

SYMBOL	MANUFACTURER	MODEL	SERVICE	TYPE	FLOW (GPM)	HEAD (FT)	RPM	ELECTRICAL HP	VOLT	PHASE	STARTER VFD	MFO	EC	N/A	MOUNTING	WEIGHT (LBS)	NOTES
P 1	GOULDS	7BF1F5B0	HYDRON C	CENTRIFUGAL	275	230	3525	25	460	3	X				BASE	255	1

NOTES
1 CLOSED COUPLED END SUCT ON CONFIGURATION

FAN SCHEDULE

SYMBOL	MANUFACTURER	MODEL	AREA SERVED	CFM	SP (IN)	RPM	SOUND POWER (UNITS)	MOUNTING	ELECTRICAL HP	W	VOLT	PHASE	STARTER VFD	MFO	EC	N/A	WEIGHT (LBS)	NOTES
EF 1	GREENHECK	SP B70	BATHROOM	50	0 25	615	2	CE LING		37 8	120	1		X			8	2
EF 2	GREENHECK	SP B710	BATHROOM	100	0 25	980	2	CE LING		80 2	120	1		X			10	2
EF 3	GREENHECK	CUBE 220	A130	4100	0 75	755	14 5	ROOF	1		460	3	X				174	1 3 4
EF 4	GREENHECK	CUBE 300	A135	8700	1	765	20	ROOF	3		460	3	X				373	1 3 4
EF 5	GREENHECK	CUBE 300	A150	9800	1	765	20	ROOF	3		460	3	X				373	1 3 4
EF 6	GREENHECK	BSQ 70 4	A140	100	0 5	1313	9 2	CE LING	0 25		115	1					76	
EF 7	GREENHECK	BSQ 70 4	A140	100	0 5	1313	9 2	CE LING	0 25		115	1					76	

NOTES
1 VFD SHALL CONTROL MOTOR SPEED TO MAINTAIN TEMPERATURE BELOW 100 DEGREES FAHRENHEIT IN TEST ROOMS
2 INTERLOCK FAN WITH LIGHT SW TCH
3 PROVIDE WITH INTEGRAL BACKDRAFT DAMPER
4 PROV DE FAN WITH 12 STANDARD FACTORY CURB

UNITARY HEAT PUMP SCHEDULE

SYMBOL	MANUFACTURER	MODEL	DESIGN CONDITIONS SUMMER DB	SUMMER WB	WINTER DB	CAPACITY SUMMER TOTAL	SUMMER SENSIBLE	WINTER TOTAL	AUX HEATER KW	EER	SEER	IPLV	COP	HSPF	SUPPLY FAN CFM	ESP	OSA	ELECTRICAL VOLT	PHASE	MCA	MOC	WEIGHT (LBS)	NOTES
PTAC 1	GE	AZ61H12DA	86	65	22	11 8	7 7	10 6	2 32	12 1			3 7		316		45	208	3	11	15	113	1
PTAC 2	GE	AZ61H12DA	86	65	22	11 8	7 7	10 6	2 32	12 1			3 7		316		45	208	3	11	15	113	1
PTAC 3	GE	AZ61H12DA	86	65	22	11 8	7 7	10 6	2 32	12 1			3 7		316		45	208	3	11	15	113	1

NOTES
1 PROVIDE WITH RAK204D1SP SUB BASE AND RAKR57 DIRECT CONNECT ON KIT

COOLING TOWER SCHEDULE

SYMBOL	MANUFACTURER	MODEL	GPM	TOTAL HEAT LOAD (MBH)	EWT	LWT	SOUND (DBA)	ELECTRICAL VOLT	PHASE	MCA	MOC	WEIGHT (LBS)	NOTES
CT 1	MARLEY	NC8401HLN	234	2331 4	105F	80F	65	460	3			9 333	1

NOTES
1 PROVIDED WITH A VFD

AIR SEPARATOR SCHEDULE

SYMBOL	MANUFACTURER	MODEL	GPM	PD (FT)	LINE SIZE (IN)	SERVICE	WEIGHT (LBS)	NOTES
AS 1	B&G	R 4F	255	1 00	4	HYDRONIC	126	

NOTES

EQUIPMENT SCHEDULE

SYMBOL	MANUFACTURER	MODEL	TYPE	SERVICE	DESCRIPTION DATA	NOTES
VP 1	QUINCY	QSVI 200	VACUUM PUMP	CO2 EXPERIMENT	200 HP 30 HG VACUUM	
VP 2	QUINCY	QSVI 200	VACUUM PUMP	CO2 EXPERIMENT	200 HP 30 HG VACUUM	
VP 3	QUINCY	QSVI 50	VACUUM PUMP	VACUUM LINES	50 HP 238 ACFM	
M 1			MOTOR	GAS TURBI NE	1 5MW MOTOR/ GENERATOR	
M 2			MOTOR	CO2	800 HP SERVING CO2 EXPERIMENT	
C 1	INGERSOLL RAND	SSR-H3P5450	AIR COMPRESSOR	COMBUSTOR EXP	2 STAGE 1730 ACFM @ 200 PSIG	
AH 1	WATLOW		AIR HEATER	COMBUSTOR EXP		
EX 1			COMBUSTOR EXPERIMENT	EXPERIMENTATION		
EX 2			GAS TURBINE EXPER MENT	EXPERIMENTATION		
NGC 1	GLM	SATURN	NATURAL GAS COMPRESSOR	EXPERIMENTATION	ROTARY SCREW 100 HP 460V3PH 0 PSI INLET 200 PSI OUTLET	
T 1	INGERSOLL RAND	CUSTOM TANK	AIR STORAGE TANK	COMBUSTOR EXP	3800 GALLONS 72 D AMETER VERTICAL	

NOTES

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RO ECT TIT E

RAMGEN POWER SYSTEMS

REDMOND RESEARCH AND DEVELOPMENT CENTER

RO ECT ADDRESS
7825 65TH STREET NE
SUITE A- 55
REDMOND WA 98052

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VALVE SCHEDULE

SYMBOL	MANUFACTURER	MODEL	S ZE (IN)	TYPE	SERVICE	CONSTRUCT ON	CONFIGURATION 2 WAY	3 WAY	ACTUATOR	CV	PD (FT)	ELECTRICAL LINE	0 10V	4 20mA	AMPS	NORMAL POSITION	FAIL POSITION	NOTES
CNG 001	N BCO		4 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		1
CNG 002	N BCO		4 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		
CNG 003			4 00	MOTOR ZED VALVE	COMP NAT GAS	STAINLESS STEEL	X		PNEUMATIC/ELECTRIC					X		OPEN	CLOSED	5
CNG 004	N BCO		4 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		2
CNG 005			4 00	LOW PRESSURE CUTOFF VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		2
CNG 006			4 00	CHECK VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		2
CNG 007	N BCO		4 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		1
CNG 008			1 00	MOTOR ZED VALVE	COMP NAT GAS	STAINLESS STEEL	X		PNEUMATIC/ELECTRIC					X		OPEN	CLOSED	5
CNG 009	N BCO		1 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		1
CNG 010	N BCO		1 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									CLOSED		5
CNG 011	N BCO		1 00	PRESSURE RELIEF VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		2
CNG 012	N BCO		1 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		1
CNG 013	N BCO		1 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		1
CNG 014	N BCO		1 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		1
CNG 015	N BCO		1 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		1
CNG 016	N BCO		1 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		1
CNG 017	N BCO		1 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		
CNG 018	N BCO		1 00	CHECK VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		
CNG 019	N BCO		1 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									CLOSED		
CNG 020	N BCO		1 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		1 5
CNG 021	N BCO		1 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		
CNG 022	N BCO		1 00	CHECK VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		
CNG 023	N BCO		1 00	GATE VALVE	COMP NAT GAS	STAINLESS STEEL	X									CLOSED		
PRV 001			4 00	PRESSURE REGULATING VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		
PRV 002			1 00	PRESSURE REGULATING VALVE	COMP NAT GAS	STAINLESS STEEL	X									OPEN		
PRV 003	DRESSER	FLOWGRID	4 00	PRESSURE REGULATING VALVE	COMP A R	STAINLESS STEEL	X			173 00						OPEN		3
CA 001	N BCO		4 00	GATE VALVE	COMP A R	STAINLESS STEEL	X									OPEN		
CA 002	N BCO		4 00	GATE VALVE	COMP A R	STAINLESS STEEL	X									CLOSED		
CA 003	N BCO			GATE VALVE	COMP A R	STAINLESS STEEL	X									OPEN		
CA 004	N BCO			GATE VALVE	COMP A R	STAINLESS STEEL	X									OPEN		
CA 005	N BCO		4 00	GATE VALVE	COMP A R	STAINLESS STEEL	X									OPEN		
CA 006	N BCO		4 00	GATE VALVE	COMP A R	STAINLESS STEEL	X									OPEN		
CA 007	N BCO		1 50	GATE VALVE	COMP A R	STAINLESS STEEL	X									CLOSED		
CWS 001	N BCO		1 00	BALL VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 002			1 50	MOTOR ZED VALVE	COOL NG WATER	STAINLESS STEEL	X		ELECTRIC					X		OPEN	CLOSED	6
CWS 003			1 00	PRESSURE REGULATING VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 004	N BCO		1 00	BALL VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 005			1 00	FLOAT VALVE	COOL NG WATER	STAINLESS STEEL	X									CLOSED		4
BFP 001	WATTS		1 00	DOUBLE CHECK VALVE	COOL NG WATER	BRASS	X									OPEN		
CWS 006	N BCO		4 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 007	N BCO		5 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 008	N BCO		0 50	BALL VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 009	N BCO		0 50	BALL VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 010	N BCO		4 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 011	N BCO		4 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 012			0 50	MOTOR ZED VALVE	COOL NG WATER	STAINLESS STEEL	X		ELECTRIC					X		CLOSED	CLOSED	6
CWS 013	N BCO		0 50	BALL VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 014	N BCO		0 50	BALL VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 015	N BCO		0 50	BALL VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 016	N BCO		4 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 017	N BCO		0 50	BALL VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 018	N BCO		4 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 019	N BCO		4 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 020	N BCO		3 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 021	N BCO		3 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 022	N BCO		2 00	BALL VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 023	N BCO		2 00	BALL VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 024	N BCO		4 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 025	N BCO		4 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 026	N BCO		3 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 027	N BCO		3 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 028	N BCO		2 00	BALL VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 029	N BCO		2 00	BALL VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 030	N BCO		4 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 031	N BCO		4 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 032			4 00	MOTOR ZED VALVE	COOL NG WATER	STAINLESS STEEL		X								OPEN		
CWS 033			6 00	BALANCING VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		
CWS 034	N BCO		6 00	BUTTERFLY VALVE	COOL NG WATER	STAINLESS STEEL	X									OPEN		

NOTES
1 PROVIDE WITH LOCK NG HANDLE
2 PROV DED WITH NATURAL GAS COMPRESSOR
3 SELF CONTAINED PILOT OPERATED TYPE
4 PROV DED WITH COOL NG TOWER
5 PNEUMATIC/ELECTRIC ACTUATOR WITH 4 20MA CONTROL SIGNAL AND 4 20MA POSITION FEEDBACK
6 ELECTRIC ACTUATOR WITH 4 20MA CONTROL SIGNAL AND 4 20MA POSITION FEEDBACK

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7 20

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REDMOND RESEARCH AND DEVELOPMENT CENTER

RO ECT ADDRESS
7825 65TH STREET NE
SUITE A- 55
REDMOND WA 98052

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M0.3
SHEET O

PERMIT SET

VALVE SCHEDULE (CONTINUED)

SYMBOL	MANUFACTURER	MODEL	SIZE (IN)	TYPE	SERVICE				ACTUATOR	CV	PD (FT)	ELECTRICAL LINE	0-10V	4-20mA	AMPS	NORMAL POSITION	FAIL POSITION	NOTES
CWS 035			6.00	BALANCING VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 036	N BCO		6.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 037	N BCO		4.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 038			4.00	MOTOR ZED VALVE	COOLING WATER	STAINLESS STEEL	X		ELECTRIC							OPEN	OPEN	6
CWS 039	N BCO		4.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 040			4.00	MOTOR ZED VALVE	COOLING WATER	STAINLESS STEEL	X		ELECTRIC							OPEN	OPEN	6
CWS 041	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 042			2.00	MOTOR ZED VALVE	COOLING WATER	STAINLESS STEEL	X		ELECTRIC							OPEN	OPEN	6
CWS 043	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 044	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 045	N BCO		4.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 046	N BCO		4.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 047	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 048	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 049	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 050	N BCO		4.00	CHECK VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 051	N BCO		4.00	CHECK VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 052	N BCO		2.00	CHECK VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 053			1.50	CIRCUIT SETTER VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 054			1.50	CIRCUIT SETTER VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 055			1.50	CIRCUIT SETTER VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 056	N BCO		2.00	CHECK VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 057	N BCO		3.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 058	N BCO		3.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 059			3.00	MOTOR ZED VALVE	COOLING WATER	STAINLESS STEEL	X		ELECTRIC							OPEN	OPEN	6
CWS 060	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 061	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 062	N BCO		3.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 063	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 064	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 065	N BCO		3.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 066	N BCO		3.00	CHECK VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 067	N BCO		4.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 068			2.00	MOTOR ZED VALVE	COOLING WATER	STAINLESS STEEL	X		ELECTRIC							OPEN	OPEN	6
CWS 069	N BCO		3.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 070	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 071	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS-072	N BCO		2.00	CHECK VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 073	N BCO		1.50	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 074	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 075	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 076	N BCO		4.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 078	N BCO		2.50	CHECK VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 079	N BCO		2.50	CHECK VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN	OPEN	
CWS 080	N BCO		2.50	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 081			3.00	MOTOR ZED VALVE	COOLING WATER	STAINLESS STEEL	X		ELECTRIC							OPEN	OPEN	6
CWS 082	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 083	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 084	N BCO		3.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 085	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 086	N BCO		2.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 087	N BCO		3.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 088	N BCO		3.00	CHECK VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 089			0.50	CIRCUIT SETTER VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 090	N BCO		4.00	BUTTERFLY VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 091	N BCO		1.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 092			1.00	MOTOR ZED VALVE	COOLING WATER	STAINLESS STEEL	X		ELECTRIC					X		OPEN	CLOSED	6
CWS 093			1.00	PRESSURE REGULATING VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
CWS 094	N BCO		1.00	BALL VALVE	COOLING WATER	STAINLESS STEEL	X									OPEN		
BFP 002	WATTS		1.00	DOUBLE CHECK VALVE	COOLING WATER	BRASS	X									OPEN		

NOTES
1 PROVIDE WITH LOCK NG HANDLE
2 PROVIDED WITH NATURAL GAS COMPRESSOR
3 SELF CONTAINED PILOT OPERATED TYPE
4 PROVIDED WITH COOLING TOWER
5 PNEUMATIC/ELECTRIC ACTUATOR WITH 4-20MA CONTROL SIGNAL AND 4-20MA POSITION FEEDBACK
6 ELECTRIC ACTUATOR WITH 4-20MA CONTROL SIGNAL AND 4-20MA POSITION FEEDBACK

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RAMGEN POWER SYSTEMS
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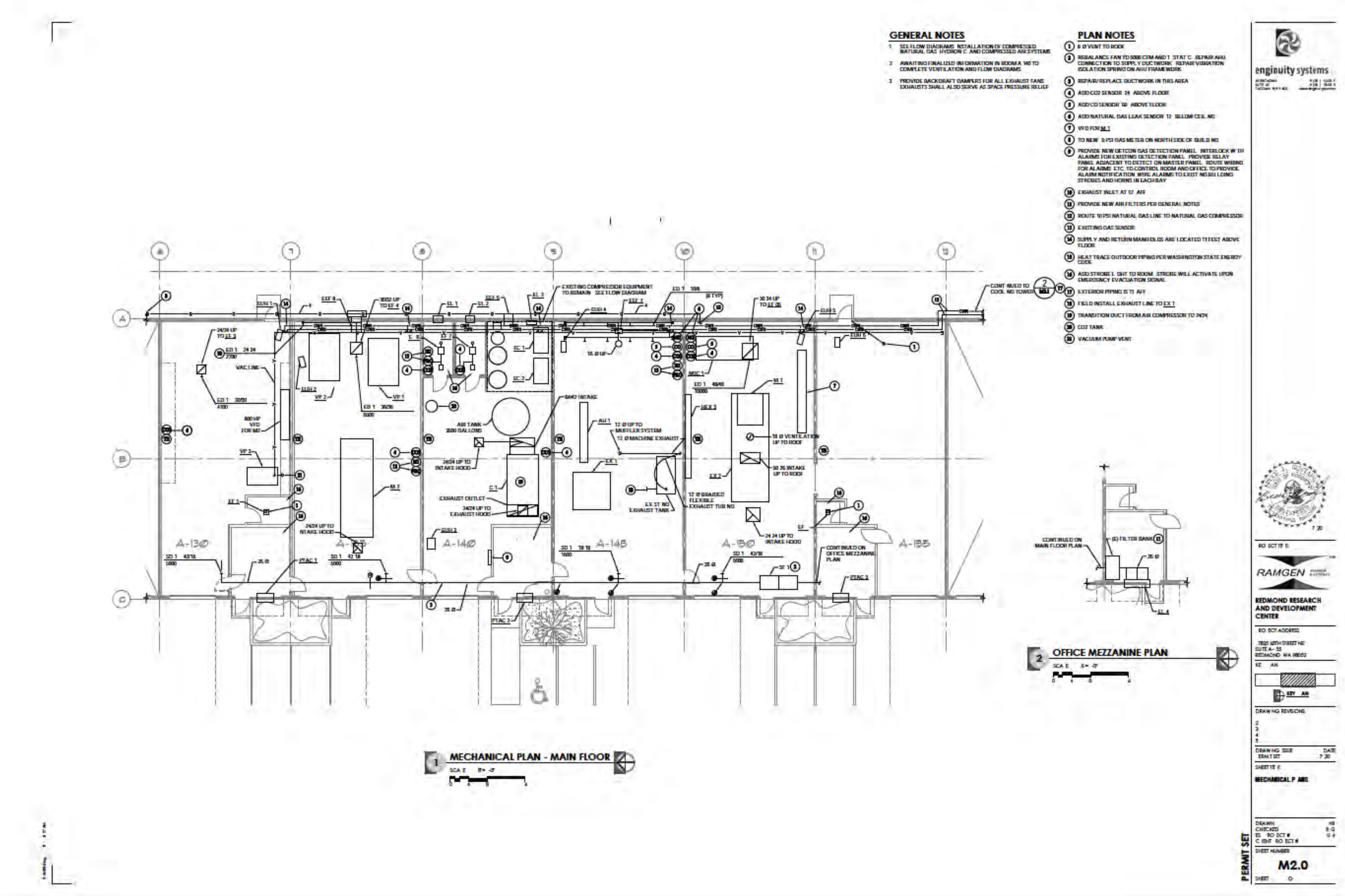
PROJECT ADDRESS
REDMOND WA
KEY PLAN
DRAWING REVISIONS
PERMIT SET
SHEET TITLE
SCHEDULES
SHEET NUMBER
M0.4


GENERAL NOTES
PLAN NOTES
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RAMGEN
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MECHANICAL PLAN - MAIN FLOOR
OFFICE MEZZANINE PLAN
PERMIT SET
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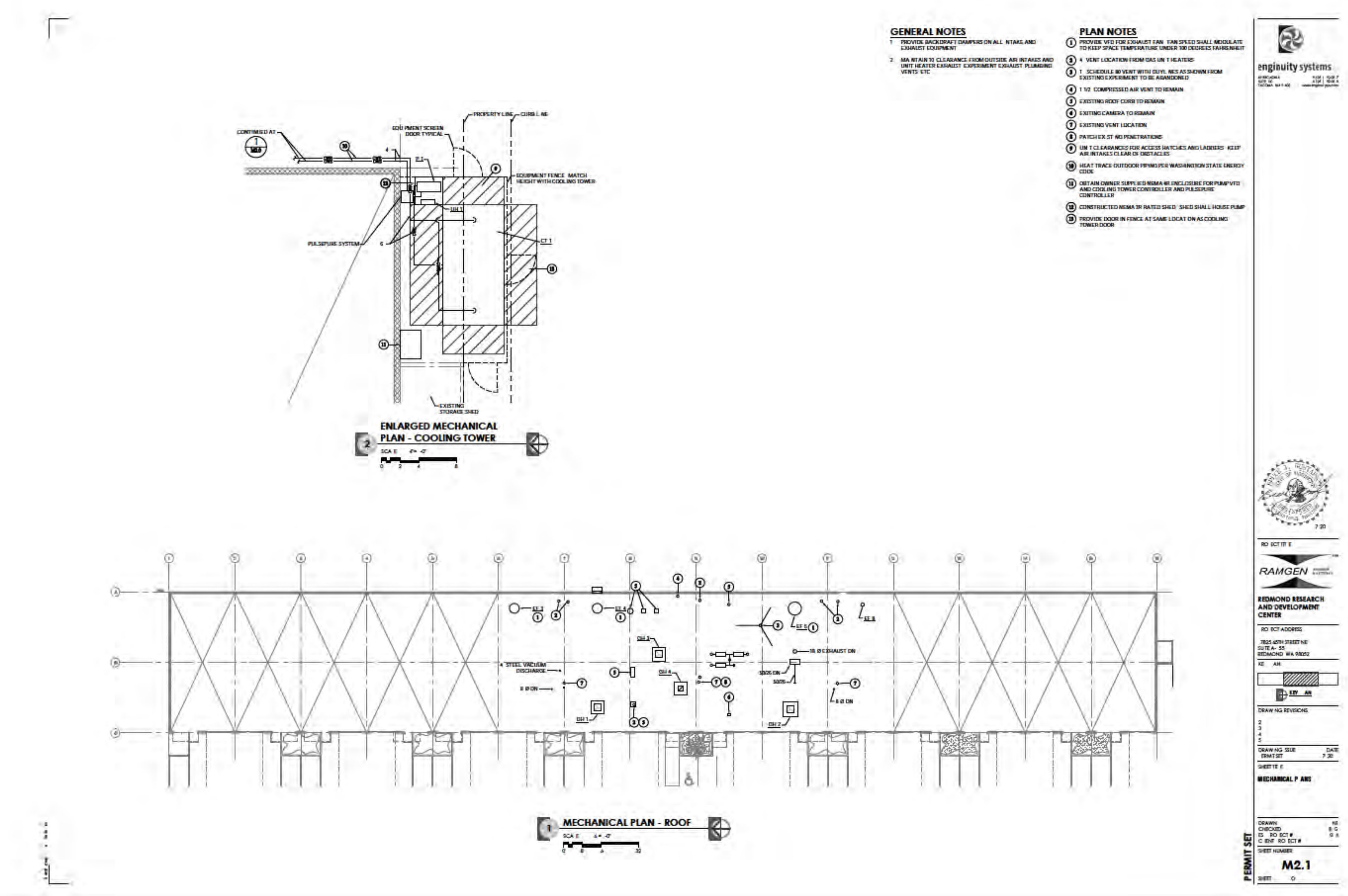
GENERAL NOTES
1 PROVIDE BACKDRAFT DAMPERS ON ALL INTAKE AND EXHAUST EQUIPMENT
2 MAINTAIN 10 CLEARANCE FROM OUTSIDE AIR INTAKES AND UNIT HEATER EXHAUST EXPERIMENT EXHAUST PLUMBING VENTS ETC
PLAN NOTES
1 PROVIDE VFD FOR EXHAUST FAN FAN SPEED SHALL MODULATE TO KEEP SPACE TEMPERATURE UNDER 100 DEGREES FAHRENHEIT
2 4 VENT LOCATION FROM GAS UNIT HEATERS
3 1 SCHEDULE 80 VENT WITH DUYL NES AS SHOWN FROM EXISTING EXPERIMENT TO BE ABANDONED
4 1 1/2 COMPRESSED AIR VENT TO REMAIN
5 EXISTING ROOF CURB TO REMAIN
6 EXITING CAMERA TO REMAIN
7 EXISTING VENT LOCATION
8 PATCH EXIST NG PENETRATIONS
9 UNIT CLEARANCES FOR ACCESS HATCHES AND LADDERS KEEP AIR INTAKES CLEAR OF OBSTACLES
10 HEAT TRACE OUTDOOR PIPING PER WASHINGTON STATE ENERGY CODE
11 OBTAIN OWNER SUPPLIED NEMA 4R ENCLOSURE FOR PUMP VFD AND COOLING TOWER CONTROLLER AND PULSEPURE CONTROLLER
12 CONSTRUCTED NEMA 3R RATED SHED SHED SHALL HOUSE PUMP
13 PROVIDE DOOR IN FENCE AT SAME LOCATION AS COOLING TOWER DOOR
ENLARGED MECHANICAL PLAN - COOLING TOWER
MECHANICAL PLAN - ROOF
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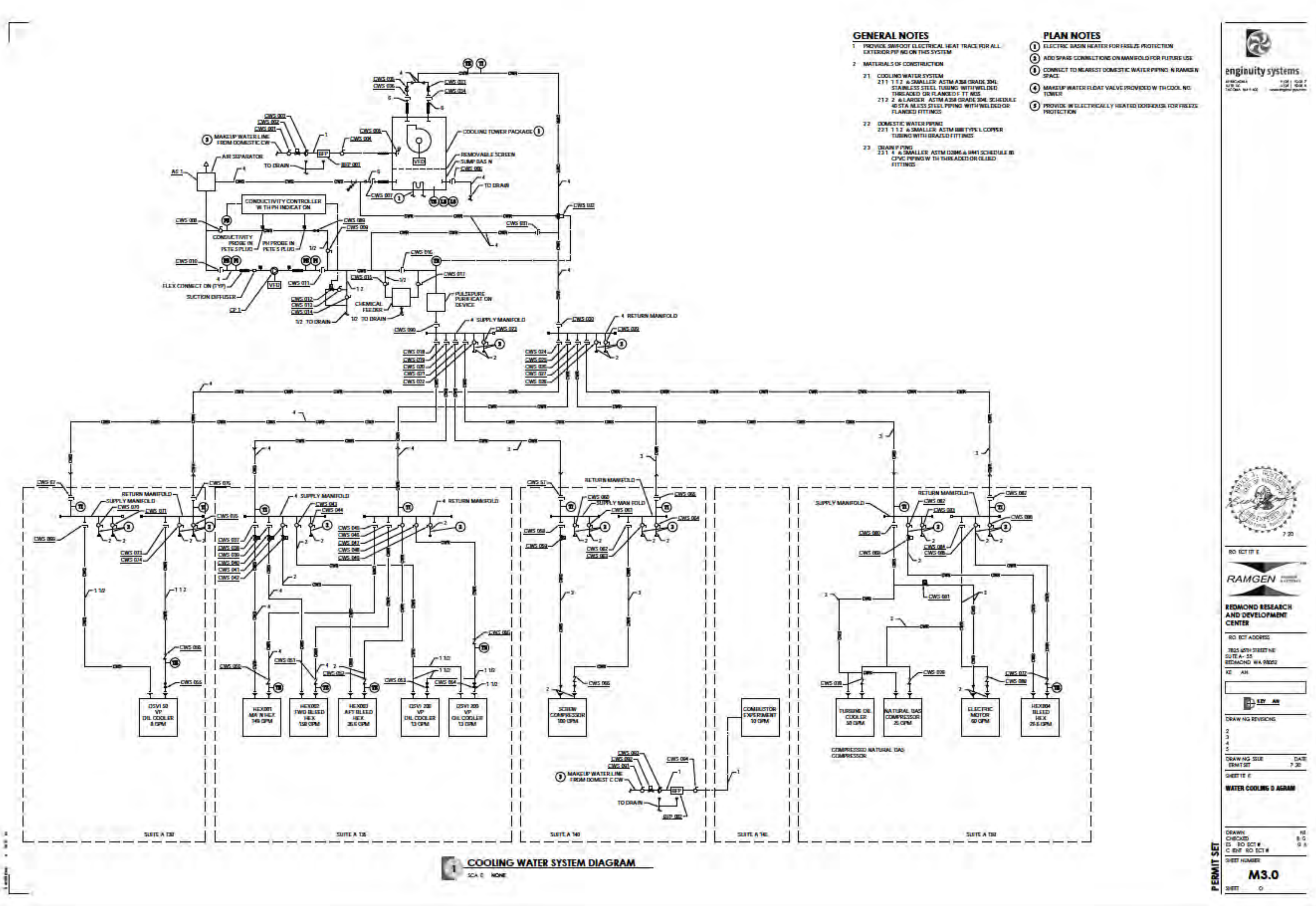
MECHANICAL PLANS
PERMIT SET
M2.1



COOLING WATER SYSTEM DIAGRAM

GENERAL NOTES

1 NATURAL GAS COMPRESSOR INCLUDES MANY OF THE COMPONENTS NEEDED FOR NORMAL OPERATION. OTHER COMPONENTS/POINTS OF THE NATURAL GAS COMPRESSOR INCLUDE:

- 1.1 SUCTION PRESSURE AND TEMPERATURE TRANSDUCERS
- 1.2 DISCHARGE PRESSURE AND TEMPERATURE TRANSDUCERS
- 1.3 OIL INLET PRESSURE TRANSDUCER
- 1.4 OIL MANIFOLD TRANSDUCER
- 1.5 OIL SEPARATOR TEMPERATURE TRANSDUCER
- 1.6 OIL INJECTION TEMPERATURE TRANSDUCER

2 MATERIALS OF CONSTRUCTION

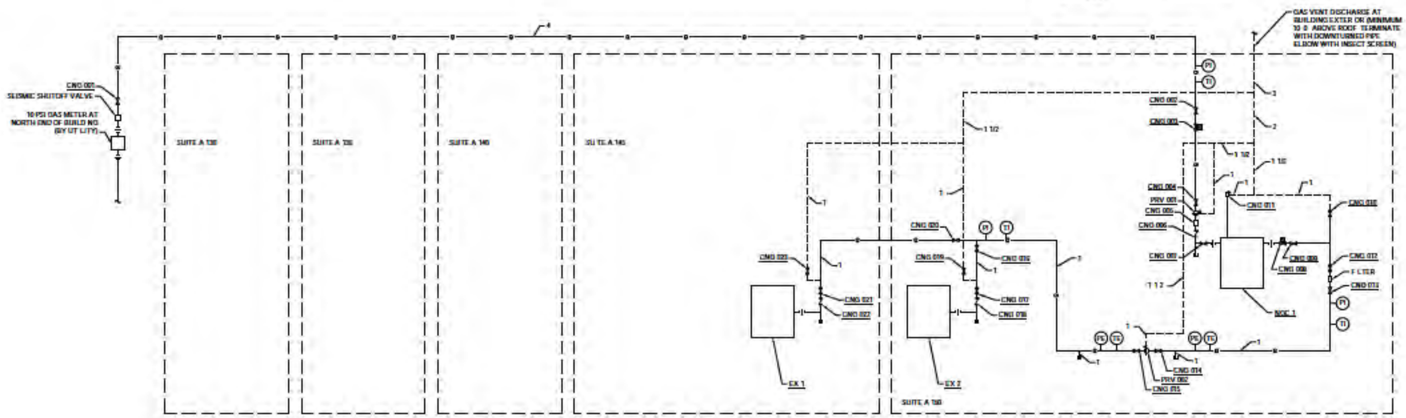
- 2.1 NATURAL GAS PIPING (UPSTREAM OF COMPRESSOR)
  - 2.1.1 2" & LARGER: ASTM A53 GRADE B, SCHEDULE 40 STEEL WITH WELDED OR FLANGED FITTINGS. PAINT ALL EXTERIOR GAS PIPING
- 2.2 NATURAL GAS PIPING (DOWNSTREAM OF COMPRESSOR)
  - 2.2.1 1 1/2" & SMALLER: ASTM A358 GRADE 304L SCHEDULE 40 STAINLESS STEEL PIPING WITH THREADED, WELDED, OR FLANGED FITTINGS
- 2.3 GAS VENT PIPING
  - 2.3.1 3" & SMALLER: ASTM A53 GRADE B, SCHEDULE 40 STEEL WITH THREADED FITTINGS

3 PAINT ALL EXTERIOR GAS PIPING FOR CORROSION RESISTANCE

PLAN NOTES

(1) LOCATION METER ON NORTH SIDE OF BUILDING ON BUILDING'S EXTERIOR

(3) INCLUDED WITH GAS COMPRESSOR



COMPRESSED NATURAL GAS SYSTEM DIAGRAM

SCALE: NONE

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PERMIT SET

COMPRESSED NATURAL GAS DIAGRAM

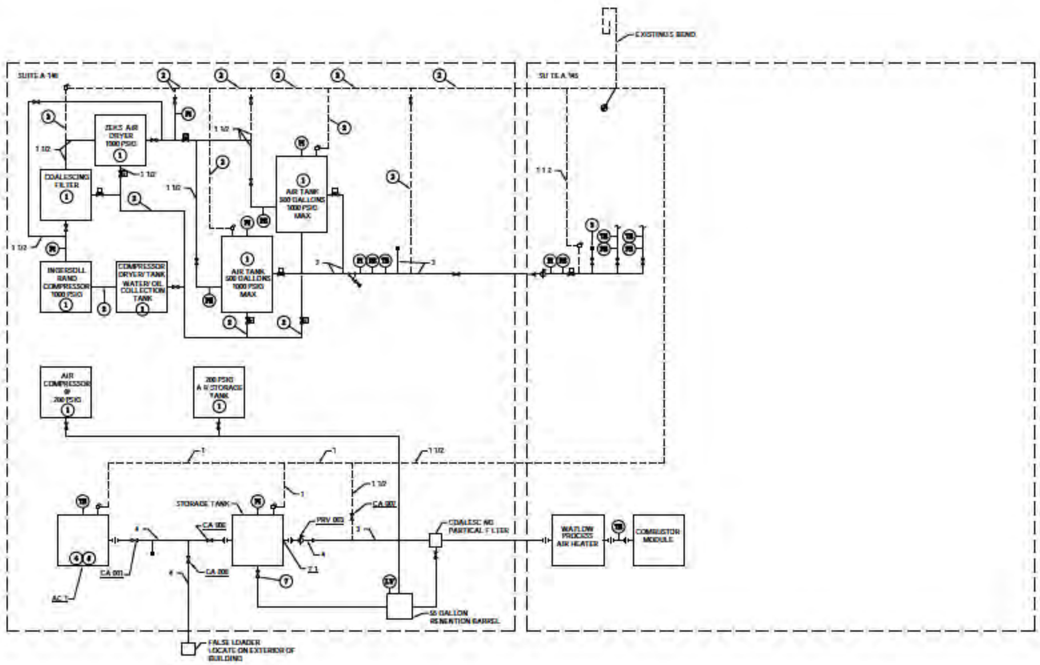
M3.1

GENERAL NOTES

1. EXISTING COMPRESSED AIR SYSTEMS IN BUILDING ARE TO REMAIN IN PLACE FOR REUSE. INCLUDED SHOP AIR SYSTEM MEDIUM PRESSURE & HIGH PRESSURE SYSTEMS
2. MATERIALS OF CONSTRUCTION
   - 2.1 COMPRESSED AIR PIPING (0-450 PSIG)
     - 2.1.1 1 1/2" & SMALLER: ASTM B88 TYPE L COPPER TUBING WITH SOLDERED FITTINGS OR ASTM A269 GRADE 304L STAINLESS STEEL TUBING WITH THREADED, WELDED, OR FLANGED FITTINGS
     - 2.1.2 2" & LARGER: ASTM A358 GRADE 304L STAINLESS STEEL PIPING WITH WELDED OR FLANGED FITTINGS
   - 2.2 COMPRESSED AIR VENT PIPING
     - 2.2.1 2" & SMALLER: ASTM A53 GRADE B STEEL PIPE WITH THREADED, WELDED, OR FLANGED FITTINGS

PLAN NOTES

1. EXISTING EQUIPMENT
2. PIPING IS 1/2" OR LESS
3. EMERGENCY VALVE VENTS TO ROOM
4. INSTALL TEMPORARY INLET SCREEN. RUN TO REMOVE WELD BEADS AND SCALE LEFT OVER DURING INSTALLATION. REMOVE AFTER HOURS OF USE.
5. BRACE PIPING TO KEEP WEIGHT OF PIPING OFF OF THE COMPRESSOR
6. COMPRESSOR INCLUDES PREFILTER
7. CONDENSATE DRAIN PROVIDED WITH COMPRESSED AIR STORAGE TANK. PIPE TO NEARBY RETENTION BARRELS



1 COMPRESSED AIR SYSTEM DIAGRAM
SCALE: NONE

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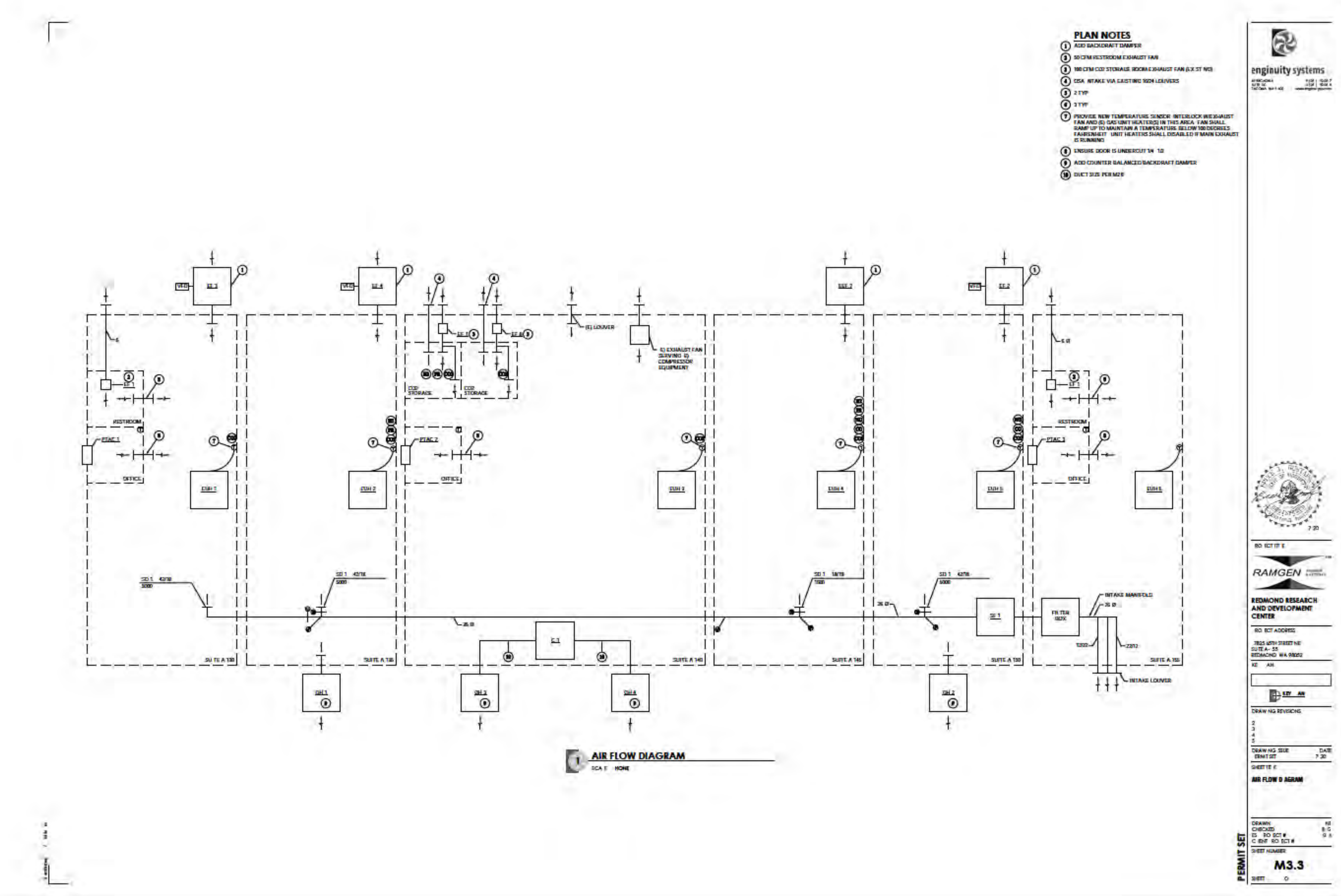
COMPRESSED AIR DIAGRAM

PERMIT SET

M3.2


PLAN NOTES
ADD BACKDRAFT DAMPER
2 TYP
3 TYP
ADD COUNTER BALANCED BACKDRAFT DAMPER
AIR FLOW DIAGRAM
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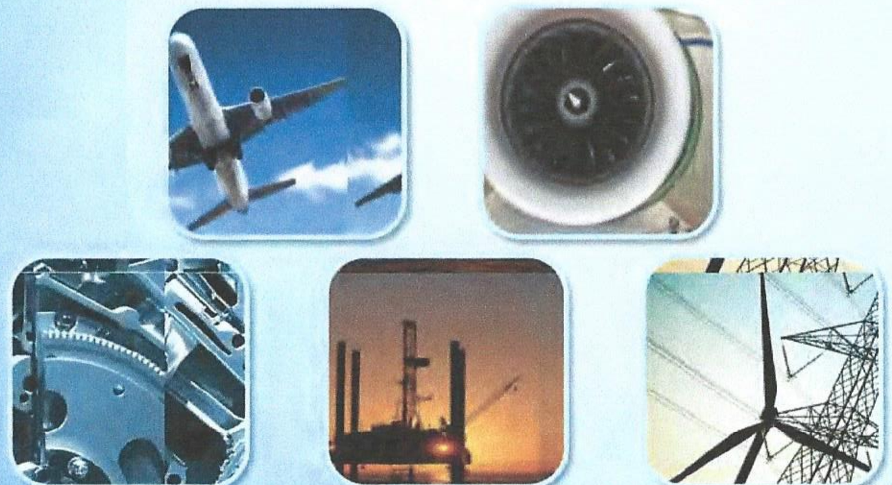
AIR FLOW DIAGRAM
PERMIT SET
M3.3

APPENDIX 11.1.1

Nozzle Final Design Review

QUEST

BORN TO ENGINEER



Carbon Canyon Design Support – Task 1

Paul Vitt, Steven Koester, Bryan Arko

10 October 2012

Carbon Canyon Design Support—Task 1

Nozzle Design Status

QUEST
Preliminary Document

- **Current Status**
  - Design point analysis completed for 7 concepts:
    - Covered design Q-1
    - Uncovered design Q-2
    - Covered design Q-3.2
    - Q-4: uncovered nozzle with increased overlap
    - Q-5: uncovered nozzle with decreased overlap
    - Q-6: uncovered nozzle with low solidity (hybrid of Q-2 and Q-4)
    - Q-7: uncovered nozzle ~same as Q-2 with an elliptical LE
  - Both covered designs were predicted to have high ID losses, caused by secondary flow roll-up of the boundary layer in that region
    - High solidity of the airfoils contributed to the problem
  - Uncovered designs have reduced losses relative to the covered designs (lower solidity airfoils in general)
  - Overlap is a driver – no overlap developed high losses
  - Similar results for circle-bevel and elliptical LE concepts with the same level of overlap
    - Sonic line shape was improved with the elliptical LE

Carbon Canyon Design Support—Task 1

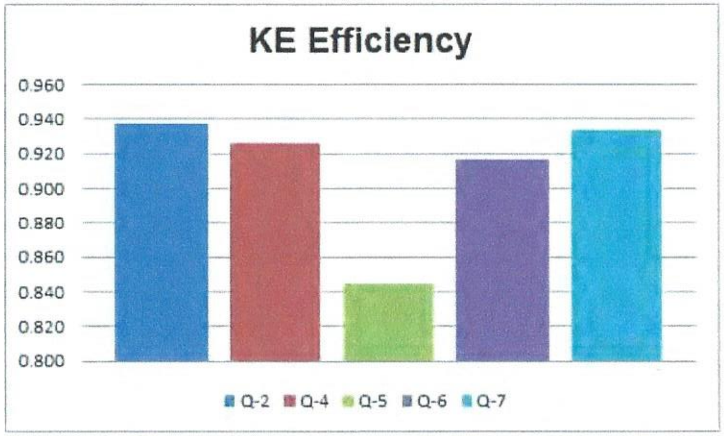
1-D Performance Comparison

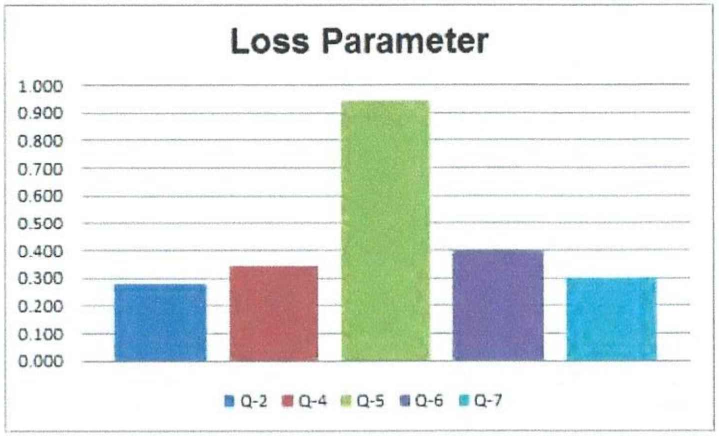
QUEST
Preliminary Document

	Q-2	Q-4	Q-5	Q-6	Q-7
Massflow [pps]	9.53	9.49	10.56	9.12	9.49
Ptx, abs [psi]	226.86	217.23	154.20	209.10	224.05
Psx [psi]	18.03	17.28	9.33	16.71	18.06
Gamma (exit)	1.32	1.32	1.33	1.32	1.32
Delta Pt / Pt-in	0.204	0.238	0.459	0.266	0.214
Nozzle Loss (ΔPt/Qx)	0.280	0.342	0.941	0.398	0.298
KE Efficiency	0.937	0.926	0.844	0.917	0.934
Abs Flow Angle [deg]	69.82	66.82	55.56	68.50	69.66
Deviation [deg]	1.47	2.00	18.59	5.05	1.64

Q-2: Base uncovered design
Q-4: uncovered nozzle with increased overlap
Q-5: uncovered nozzle with decreased overlap
Q-6: uncovered nozzle with low solidity (hybrid of Q-2 and Q-4)
Q-7: uncovered nozzle ~same as Q-2 with an elliptical LE

Note Ptx is a mixed value (used for the eff and loss calcs)





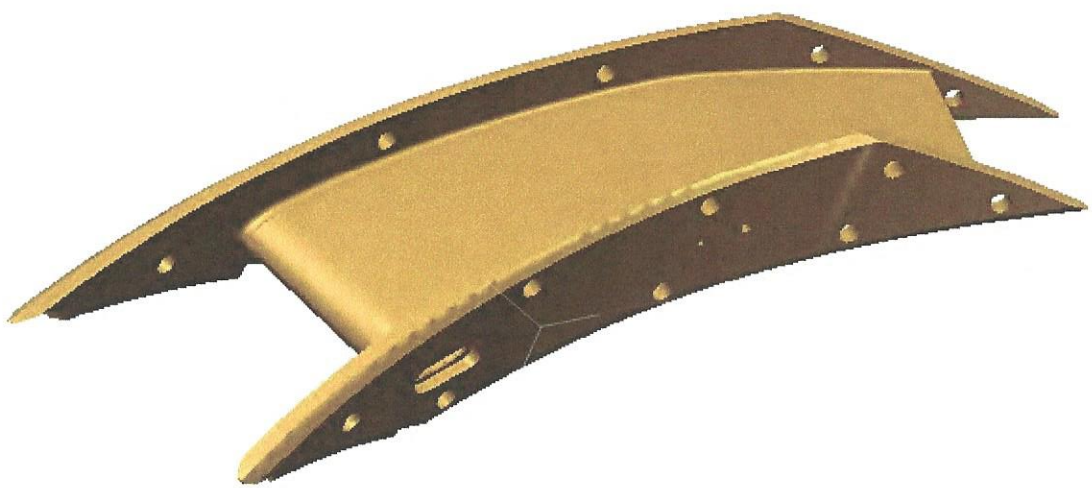
Appendix 11.1.2

Laser Scan Results

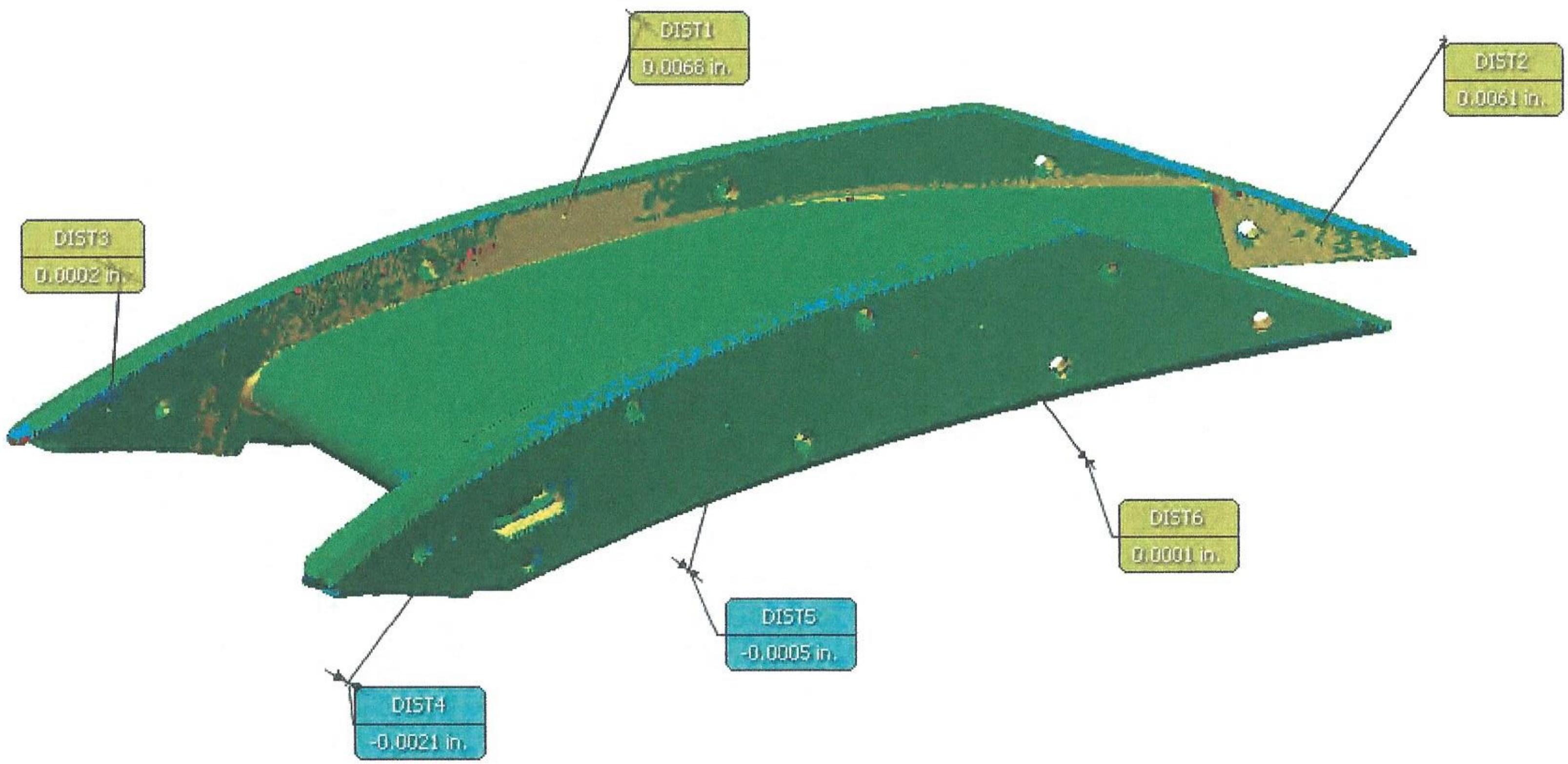
ABSOLUTE Geometries

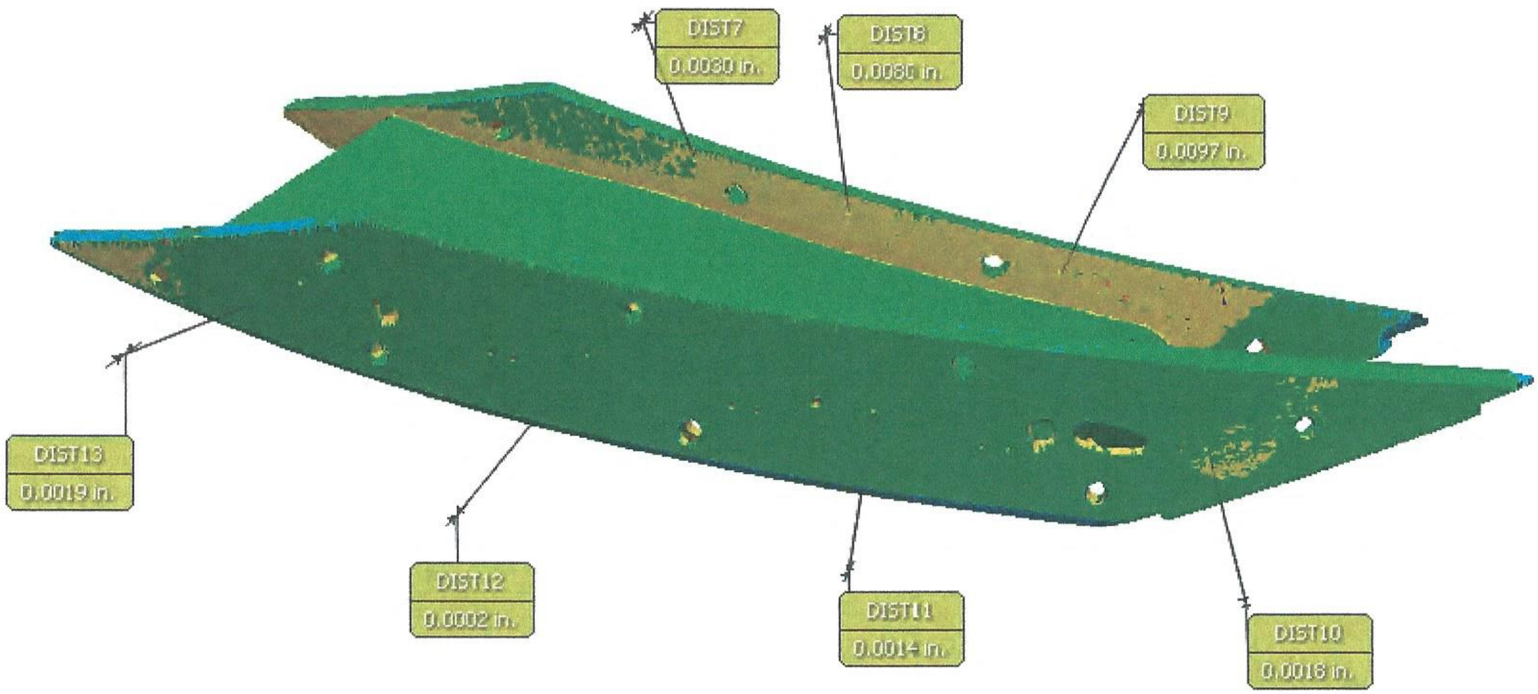
Customer:	Directed MFG
Part Name:	Vane
Part Number:	861036-1
SN:	1
Tolerance:	.005"
Inspector:	Durham
Date:	5-6-14

CAD



ABSOLUTE Geometries

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DIST8
DIST9
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DIST13
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DIST12
0.0002 in.
DIST11
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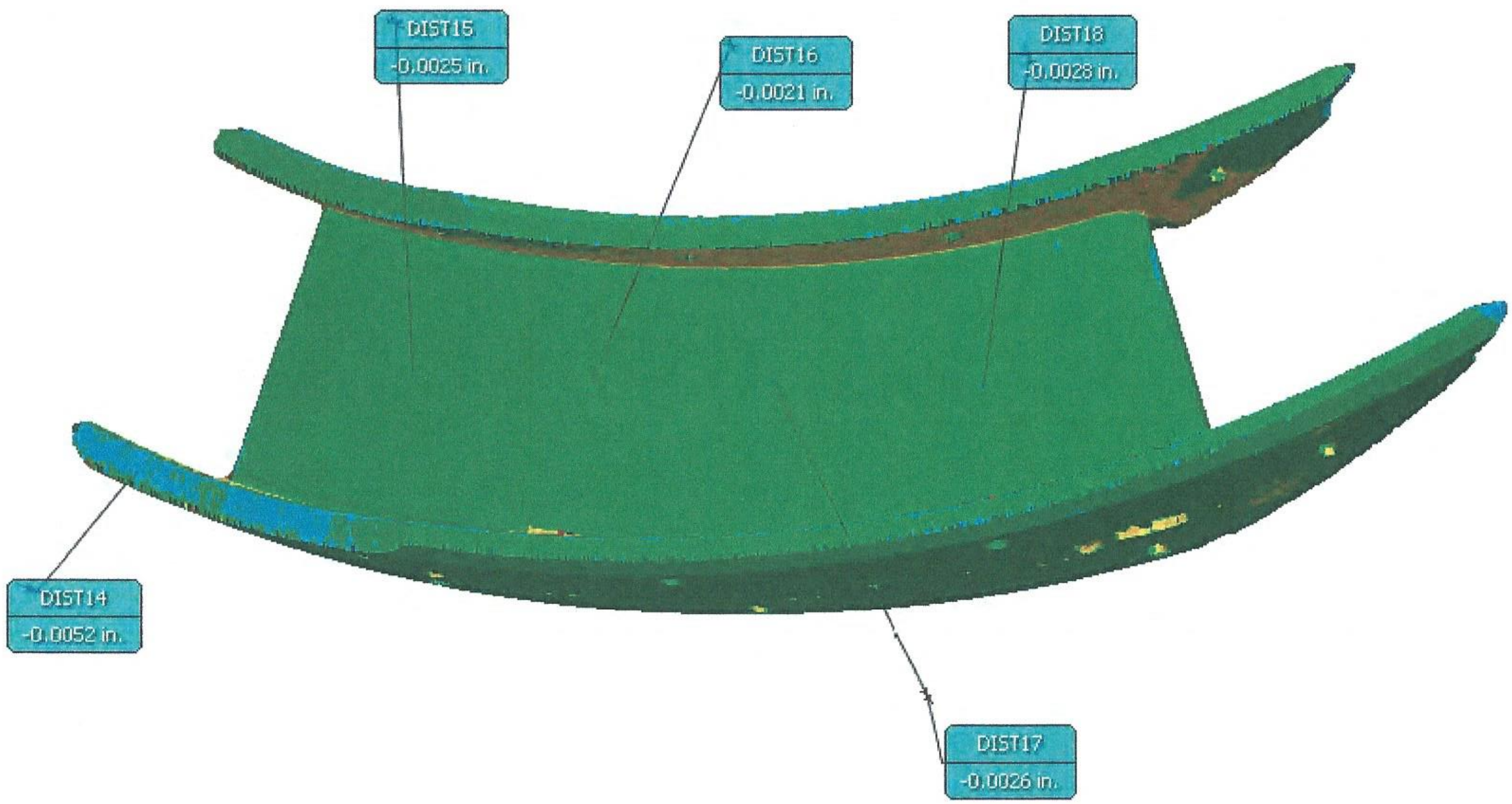
DIST10
0.0016 in.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DIST16
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DIST18
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DIST14
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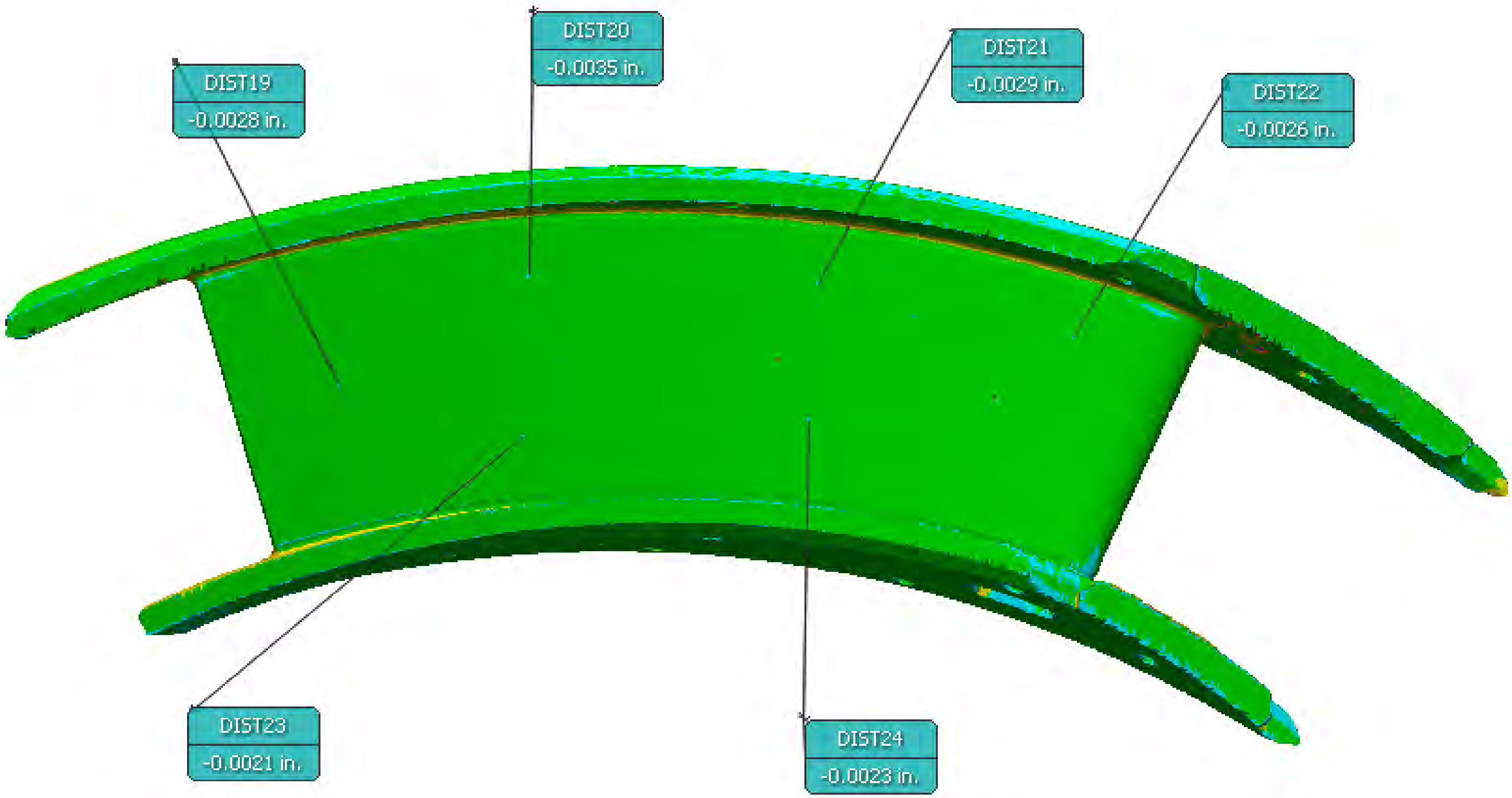
DIST17
-0.0026 in.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DIST24
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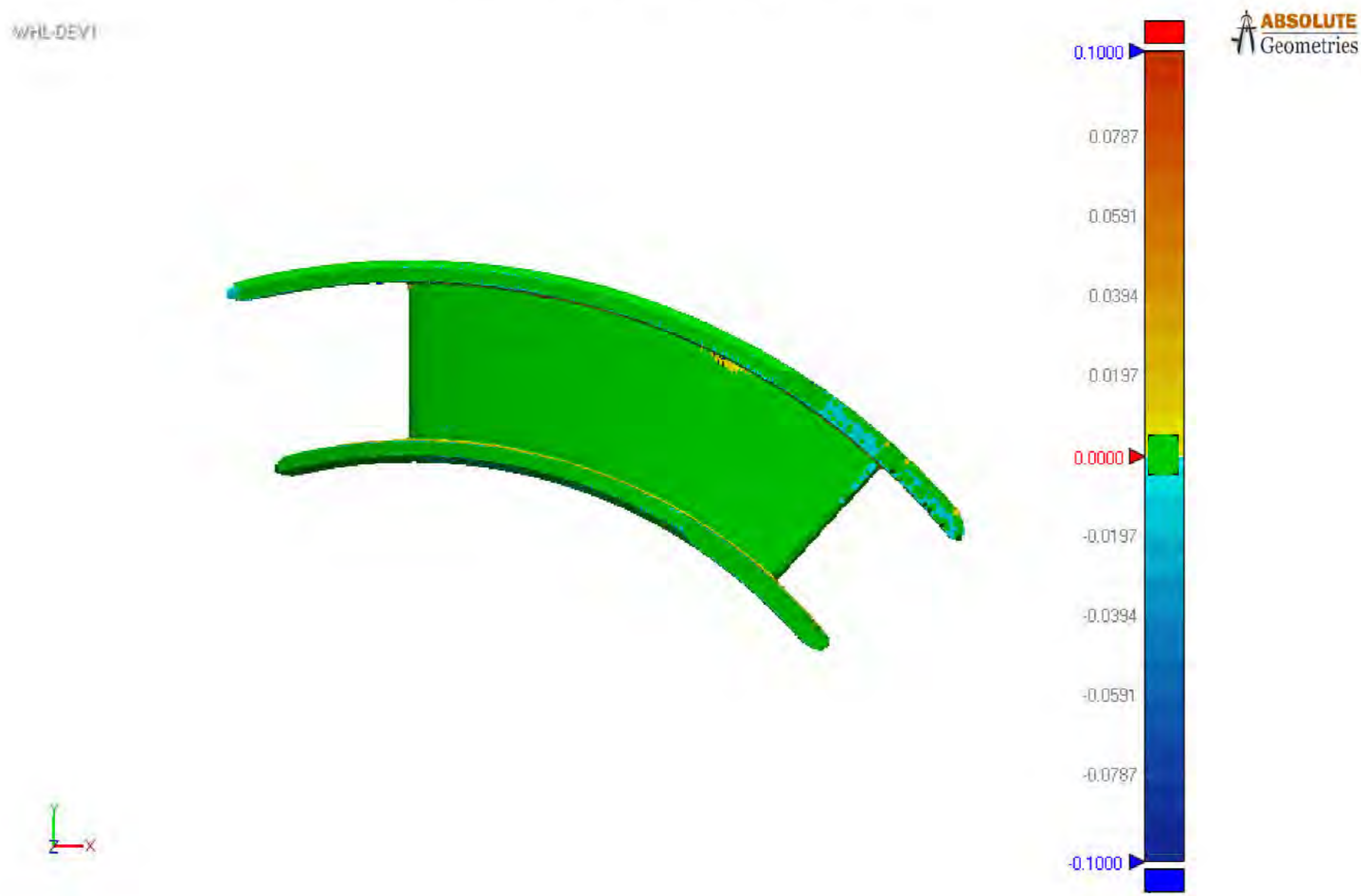
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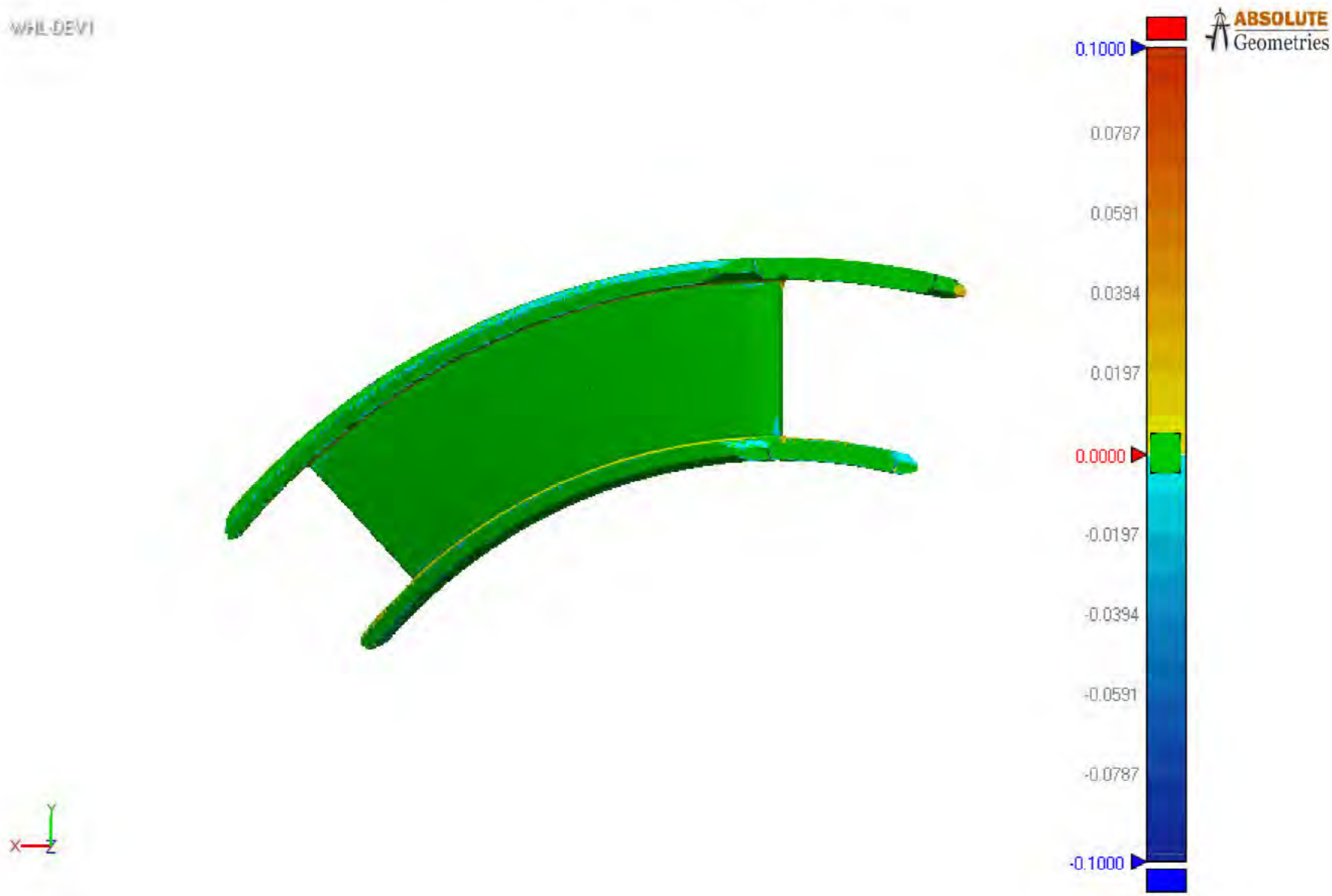
Y
Z
X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ABSOLUTE
Geometries
Y
X
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WHL-DEV1
ABSOLUTE Geometries
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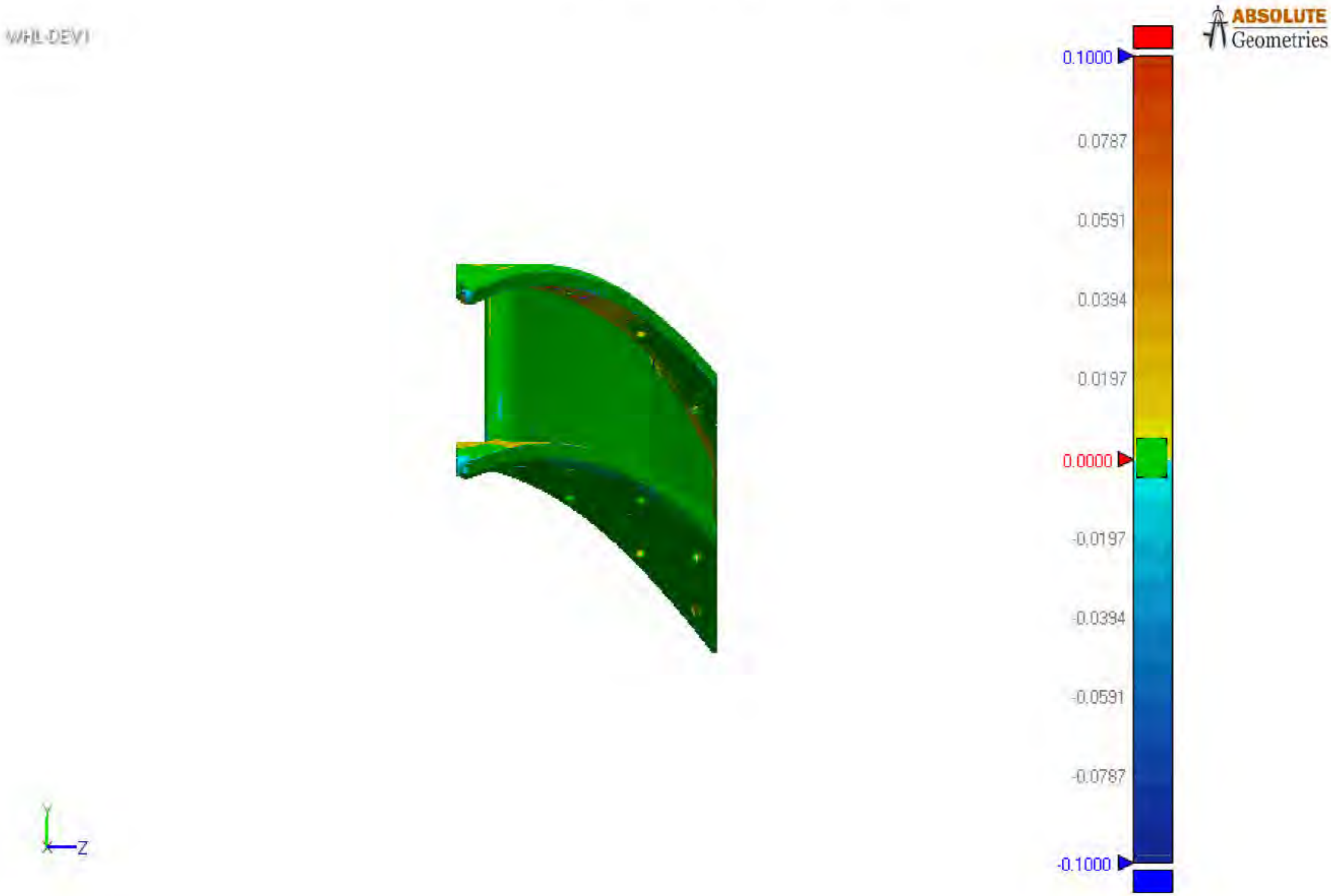
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Y
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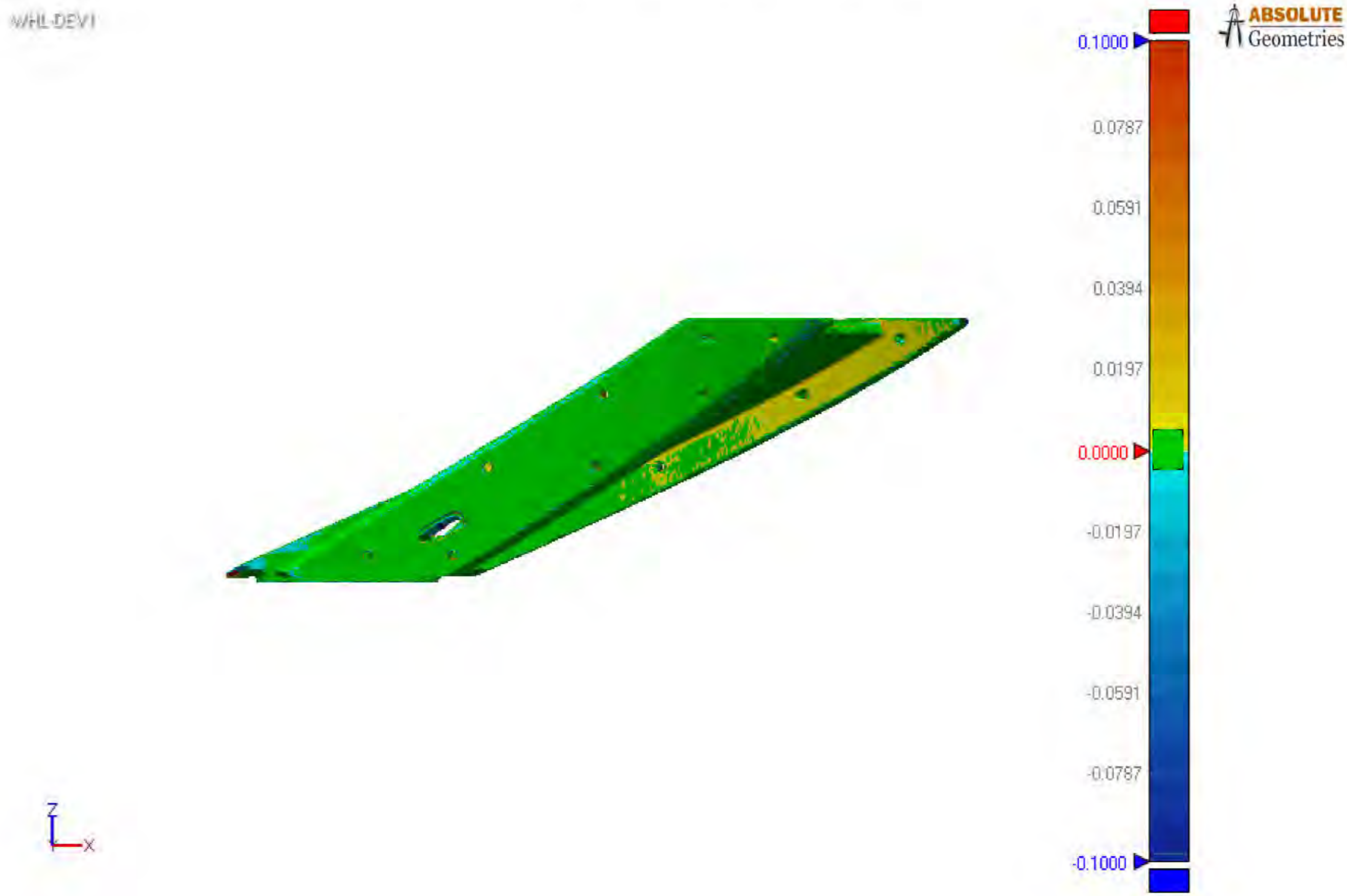
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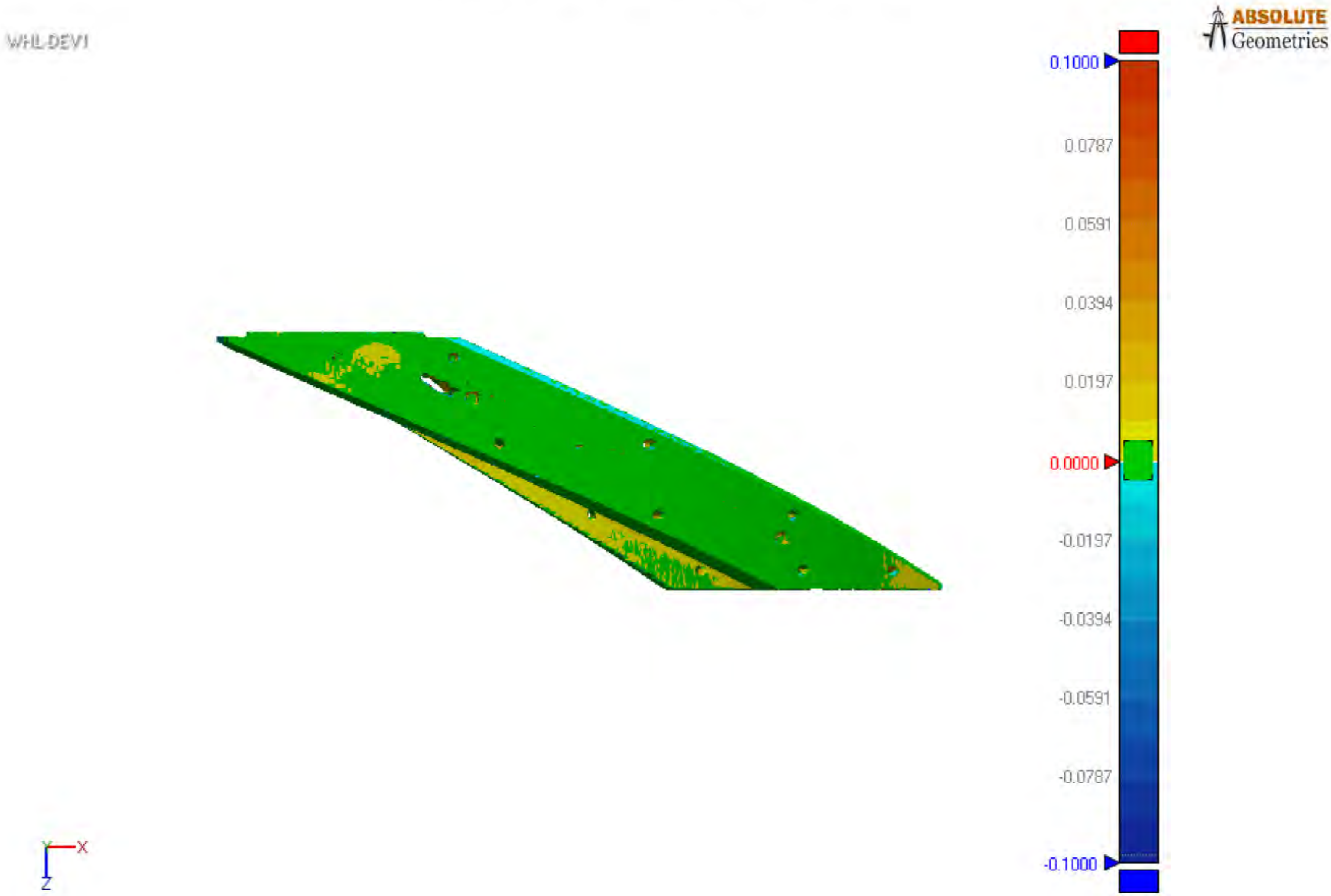
ABSOLUTE
Geometries
Z
X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-0.0591
-0.0787
-0.1000
Y
X
Z

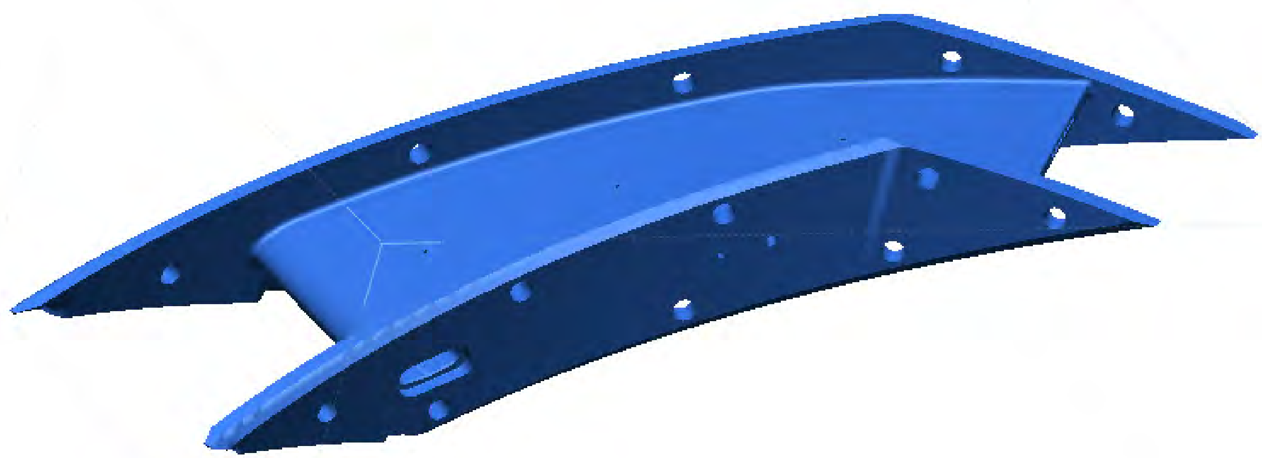
Appendix 11.1.3

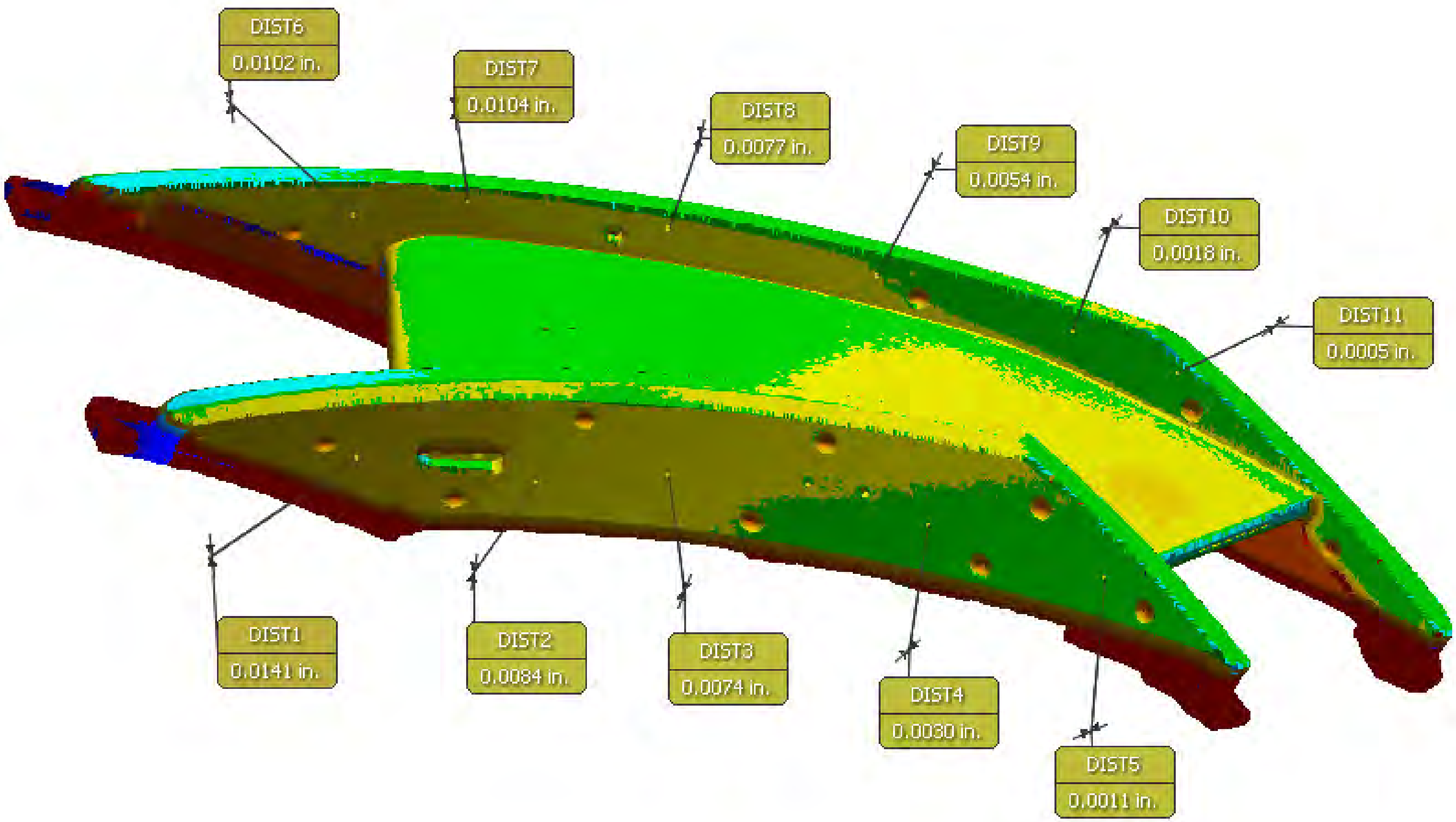
Laser Scan of Warped Part

ABSOLUTE Geometries

Customer:	Directed MFG
Part Name:	Vane
Part Number:	861036-2
SN:	1
Tolerance:	.005"
Inspector:	Durham
Date:	5-7-14

CAD

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DIST7
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DIST8
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DIST9
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DIST10
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DIST11
0.0005 in.
DIST1
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DIST2
0.0084 in.
DIST3
0.0074 in.
DIST4
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DIST5
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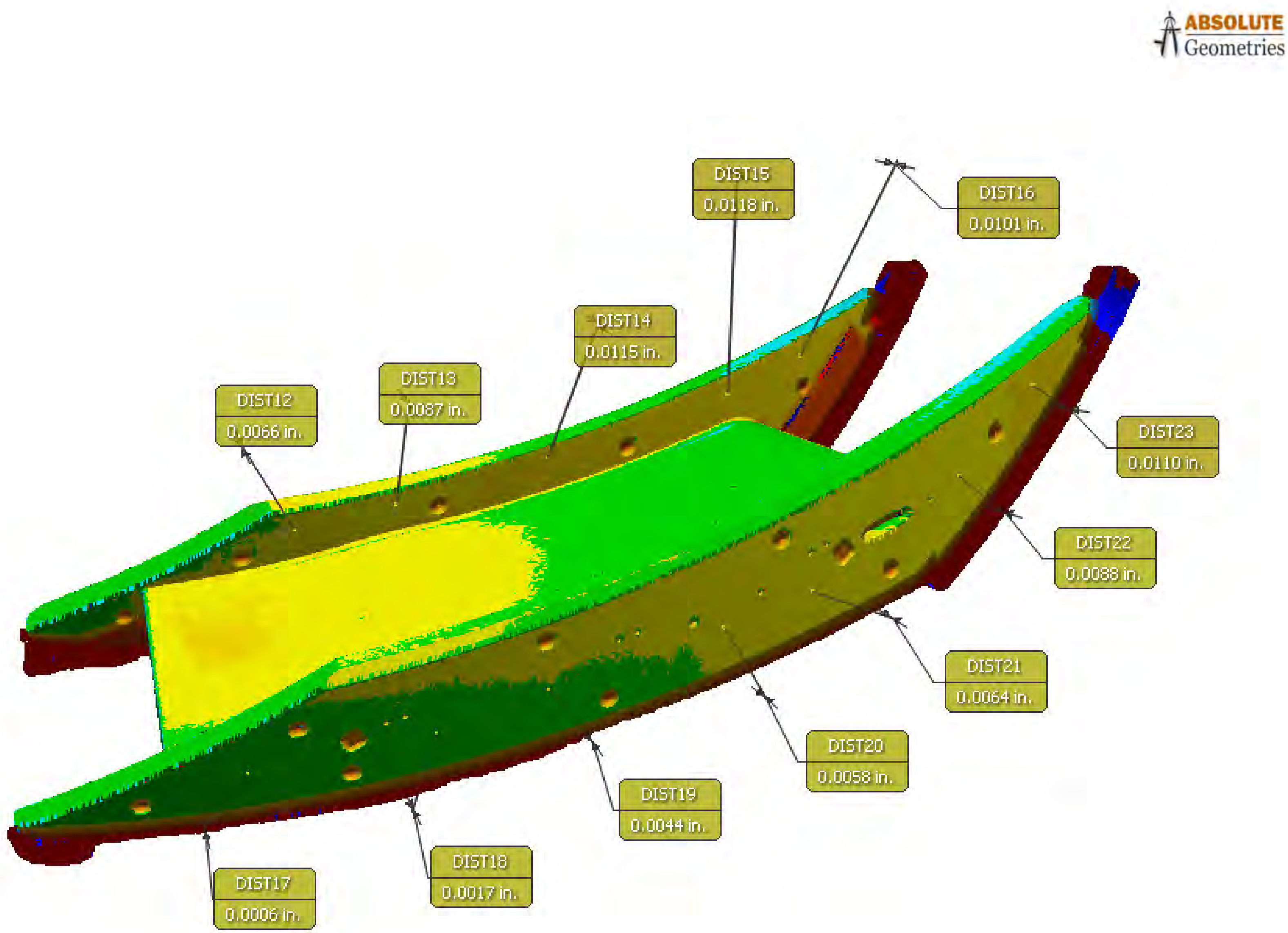
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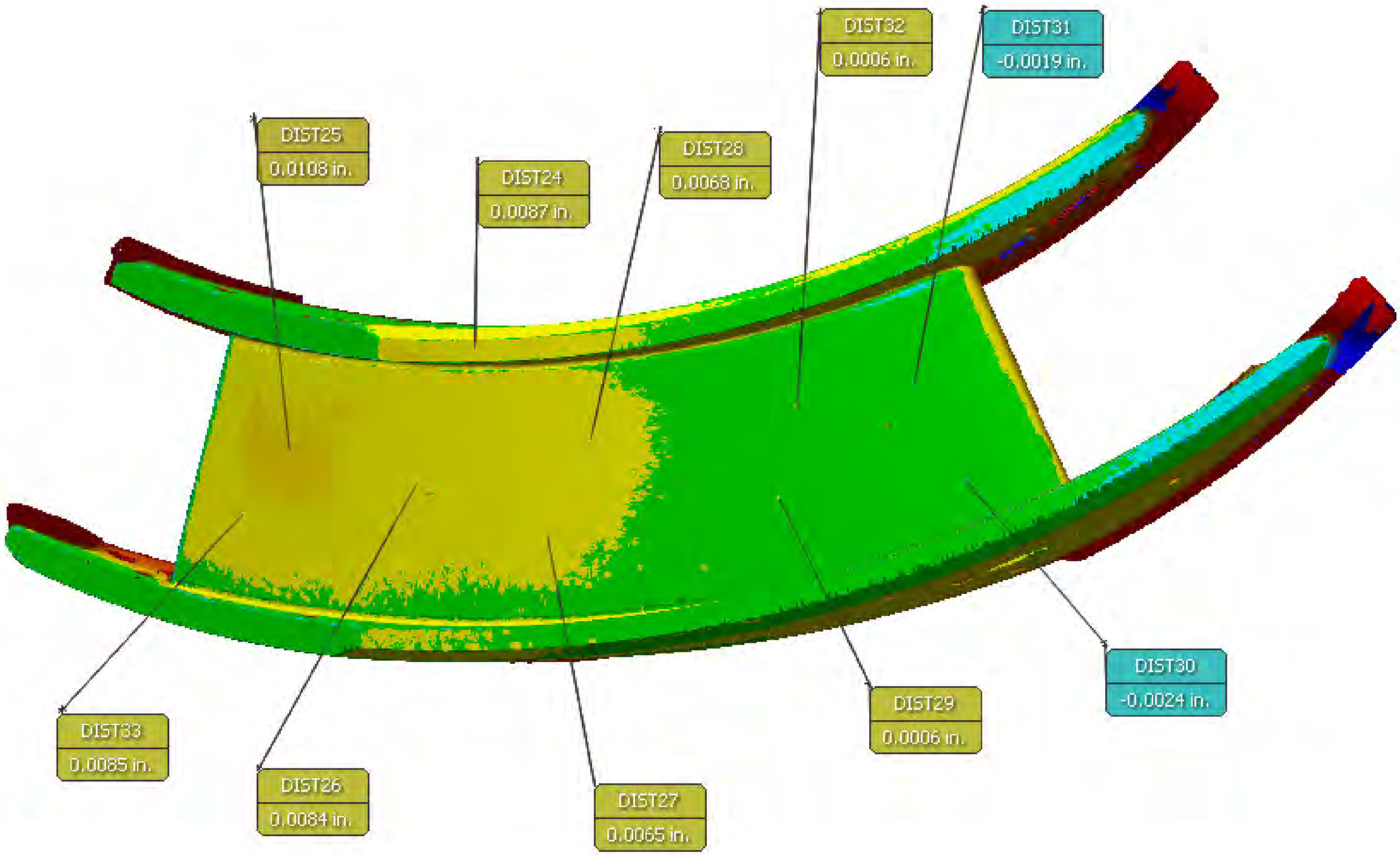
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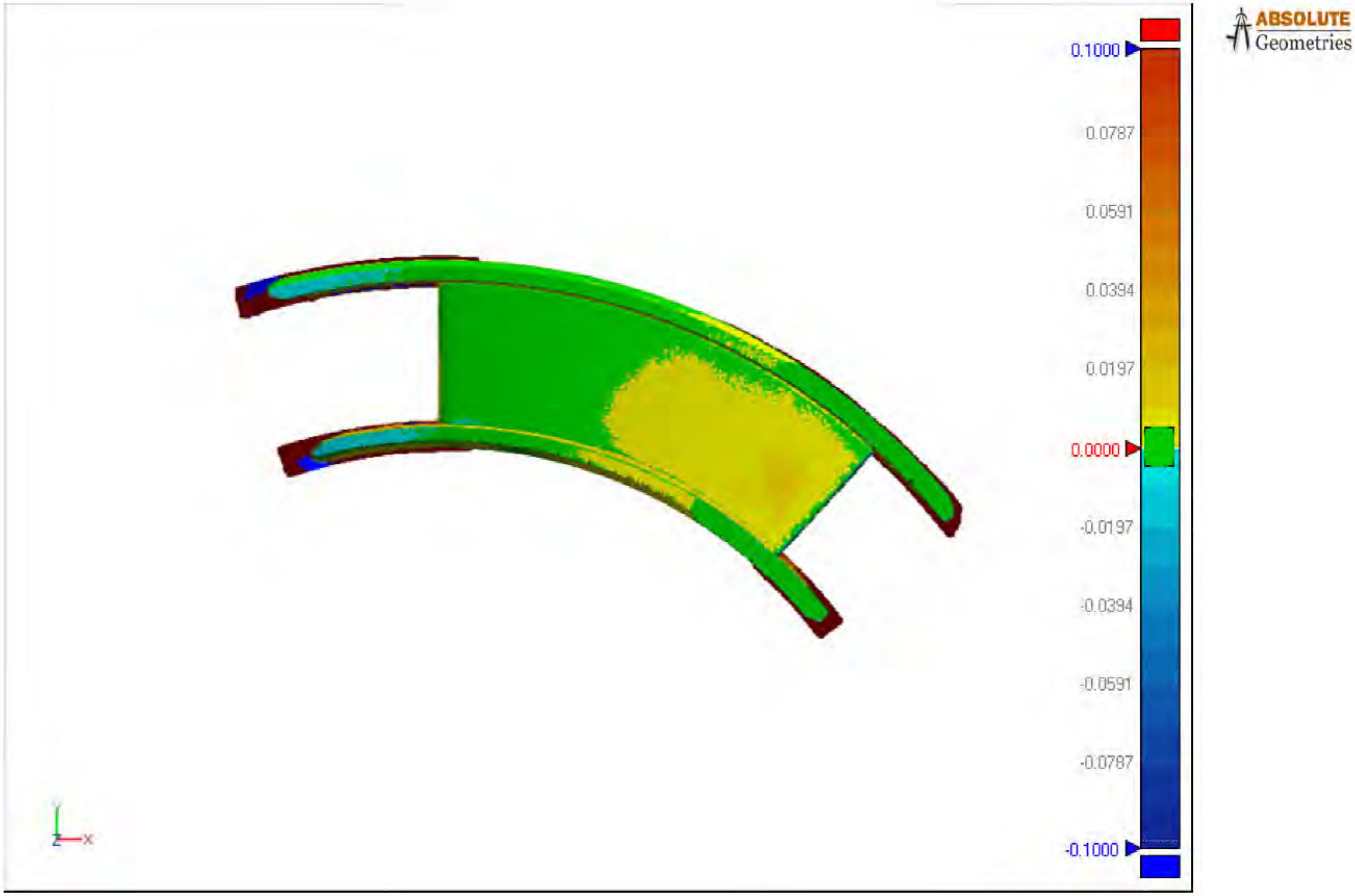
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-0.1000
ABSOLUTE Geometries
Y
Z
X


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0.0591
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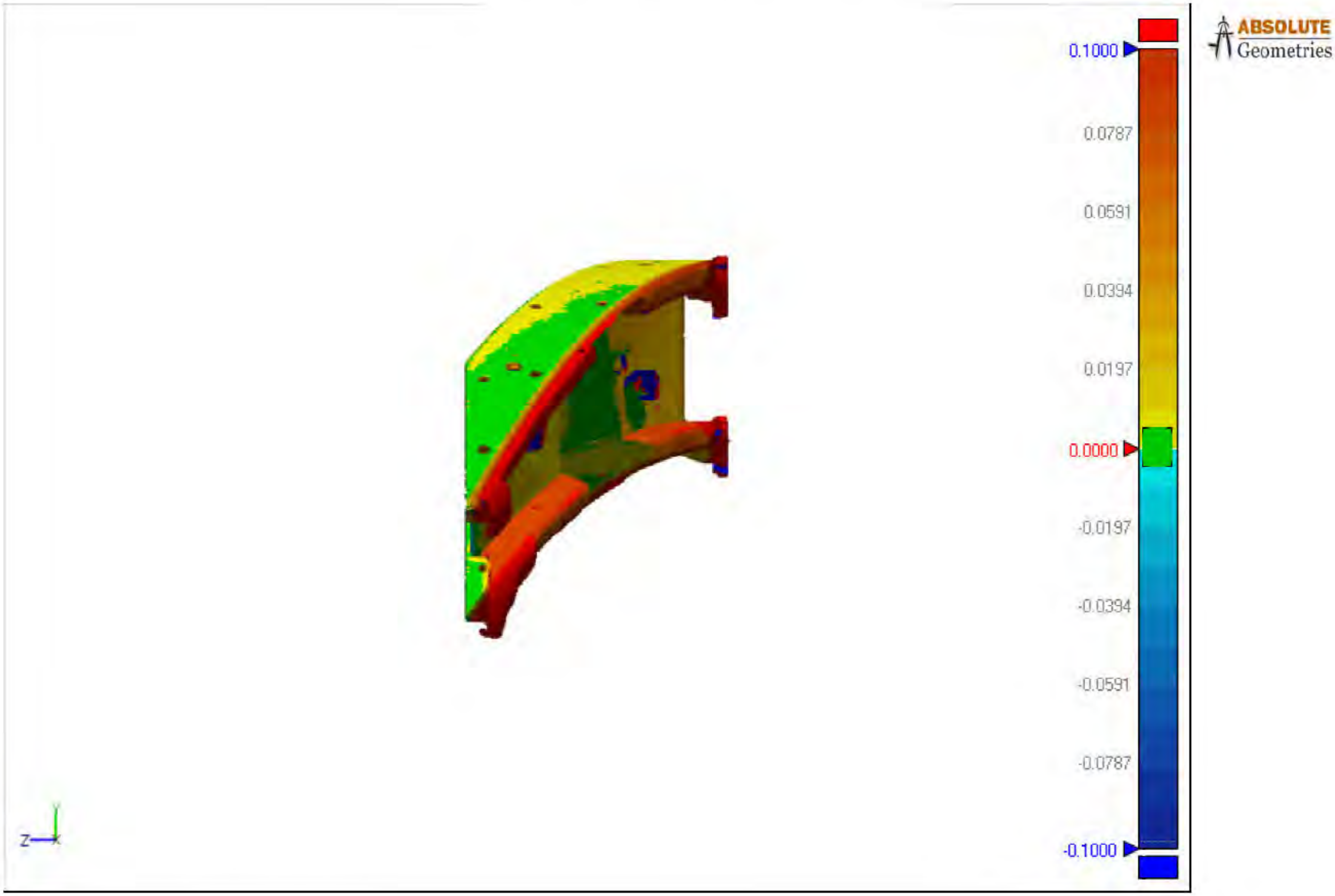
ABSOLUTE
Geometries
Y
X
Z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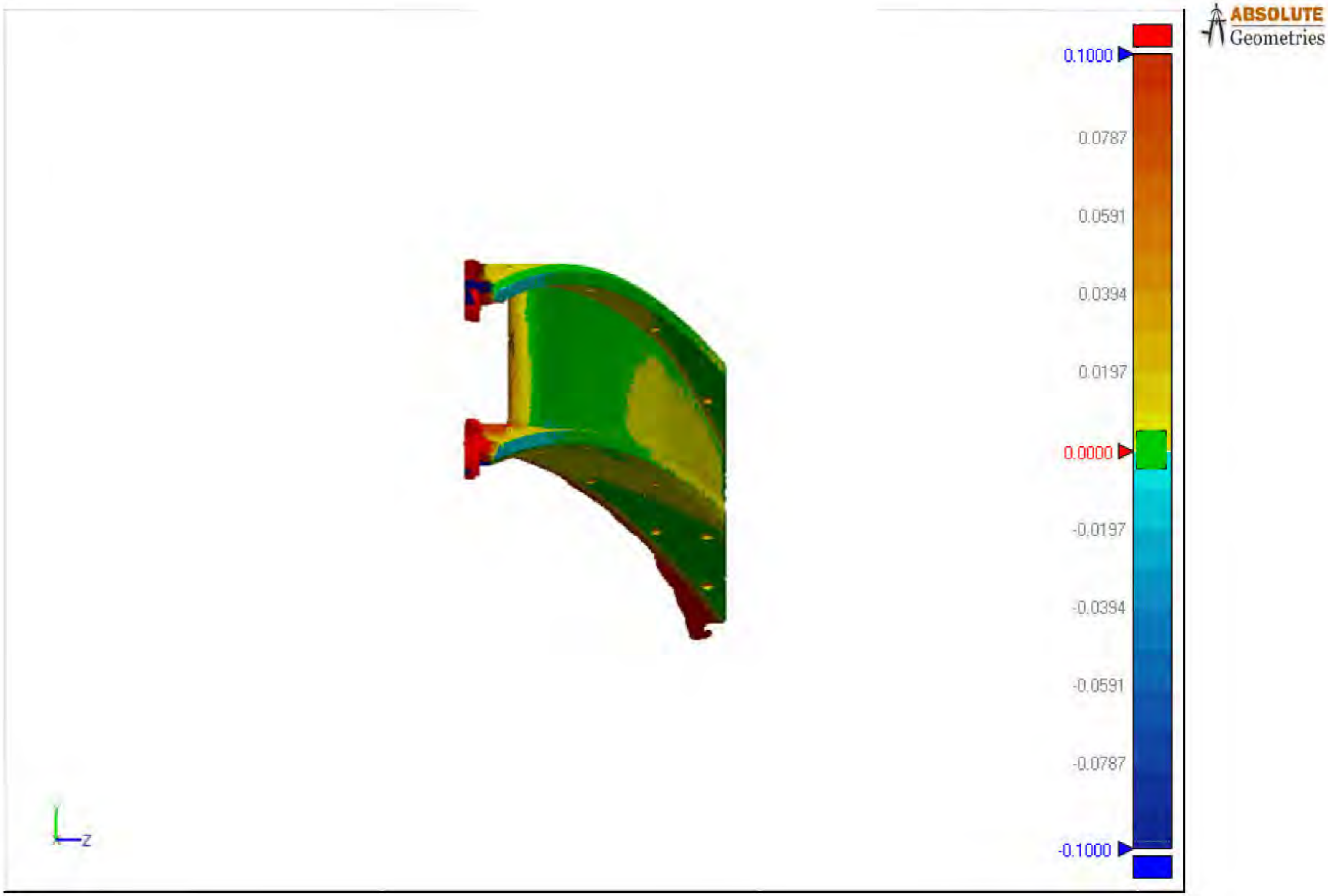
ABSOLUTE
Geometries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Geometries
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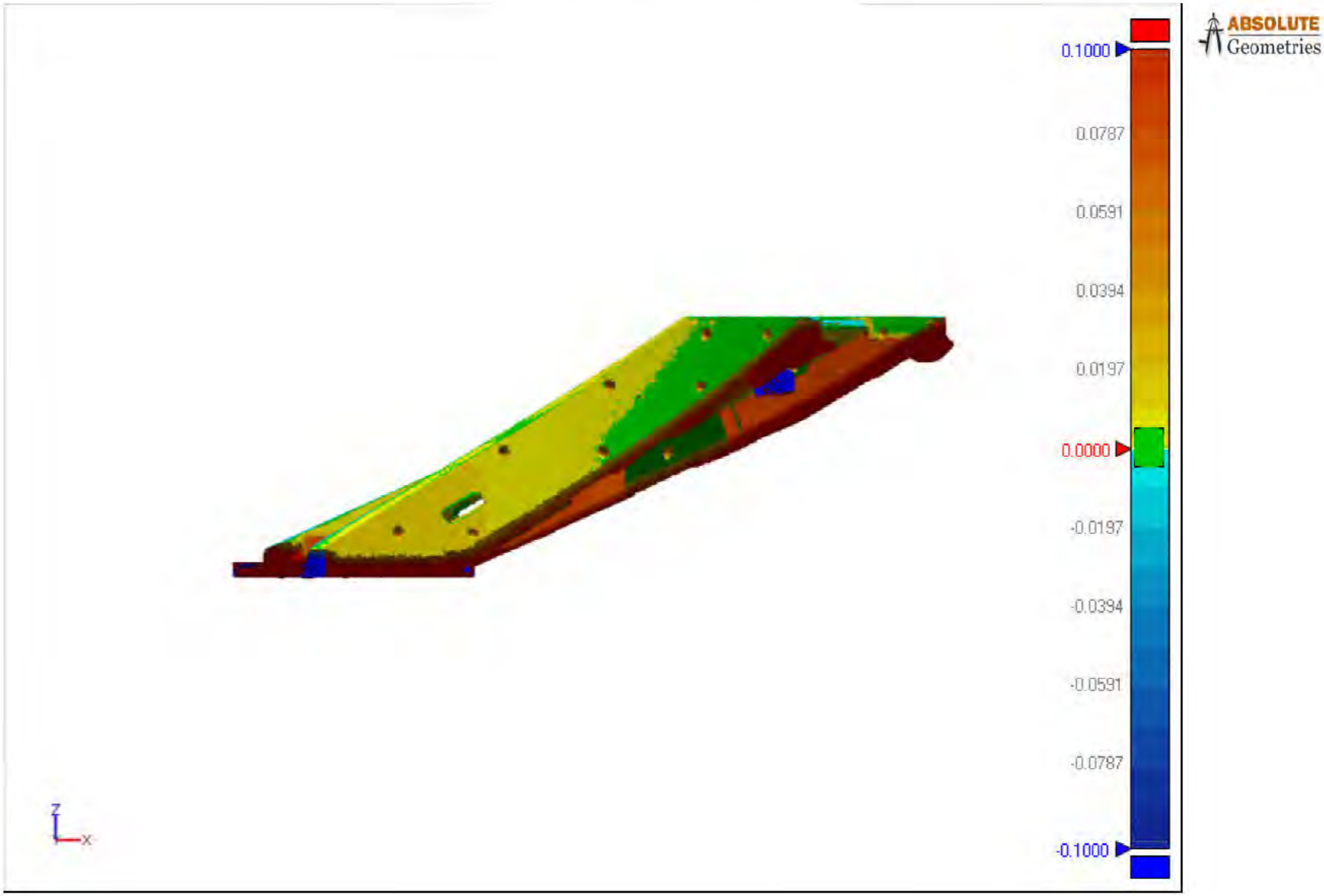
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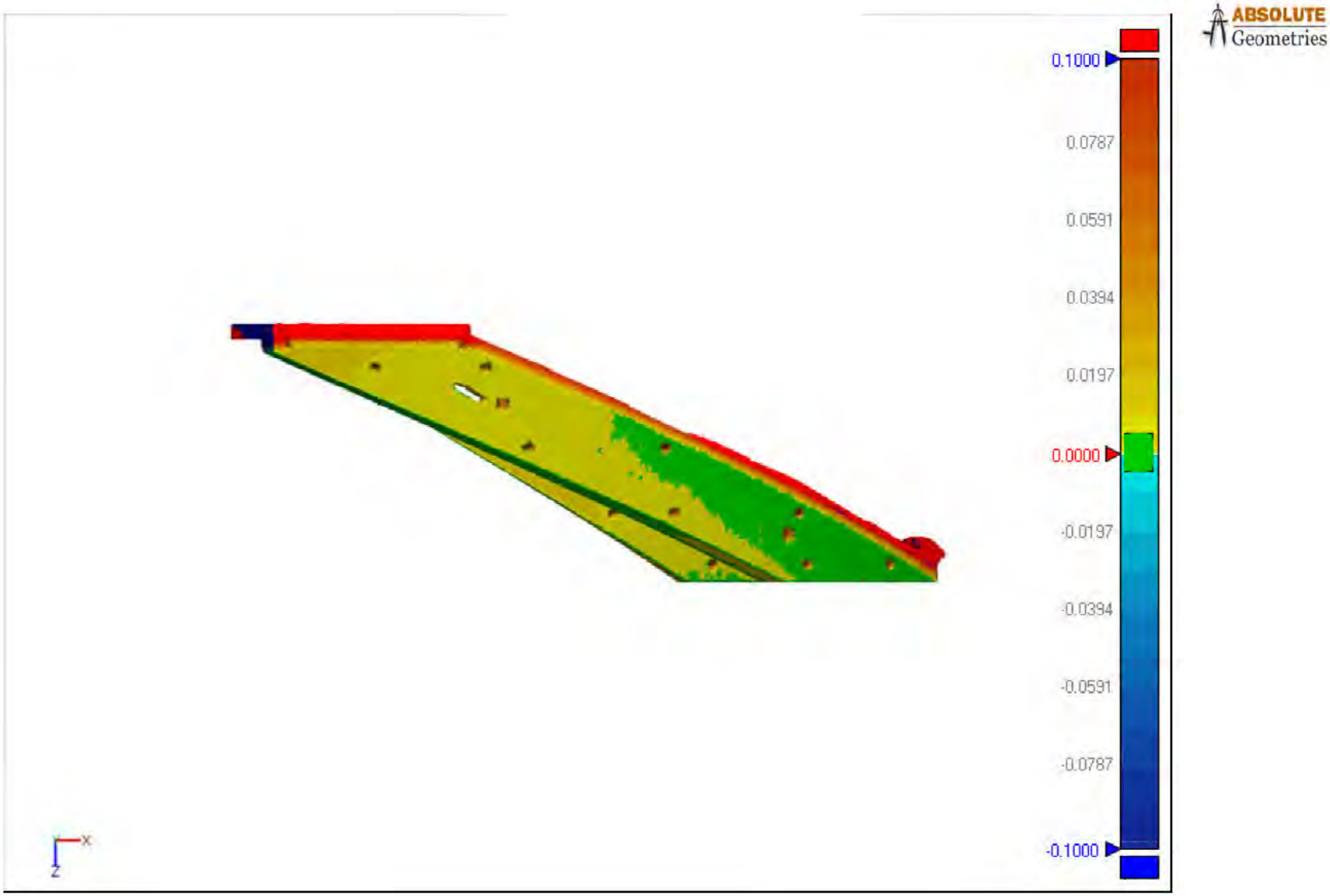

ABSOLUTE
Geometries
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-0.0394
-0.0591
-0.0787
-0.1000
X
Z

APPENDIX 11.2.1

Inlet FDR

ISCE Build 2 6:1 Checkout Inlet FDR

Aaron Salzbrun

2013-10-25

Updated System Requirements

1. Deliver 87 (and/or 147) psi, 180 F air at 10 lb/s to static combustor with equivalent flow uniformity to existing OGV output
2. Accommodate up to 200 psi internal pressure (SF)
3. Admit high-pressure air from auxiliary air system
4. Not interfere with existing seal and thrust flows
5. Not interfere with existing bearings
6. Maintain identical or greater stiffness relative to existing components
7. Operate with internal environments up to 325 F
8. Operate safely
9. Accommodate additional axial load due to internal pressure
10. Accommodate thermal and axial growth of rotor section.
11. Accommodate additional vertical load from Air inlet system

Action Items

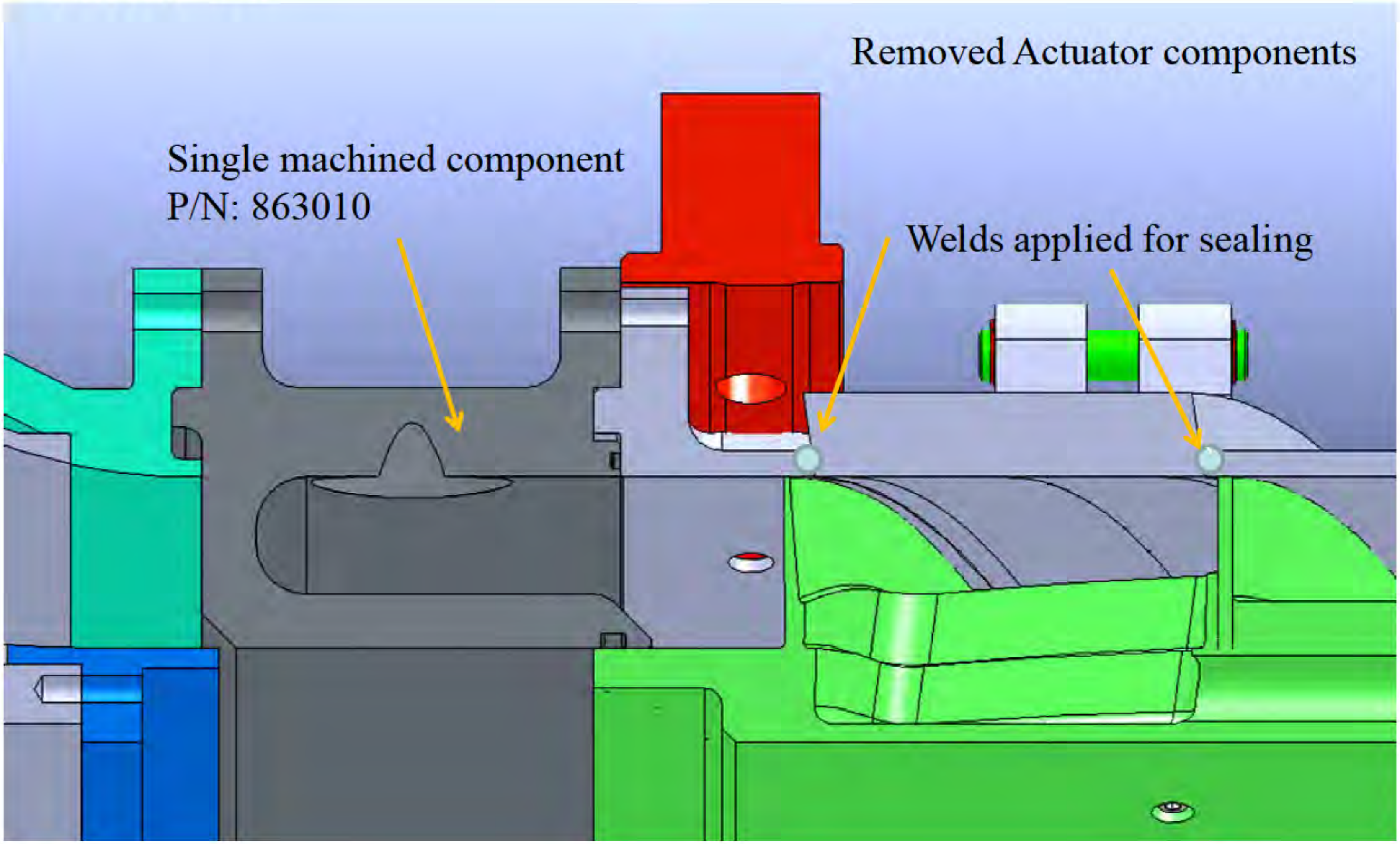
- **Assess Instrumentation requirements**
  - No new provisions needed
- **Determine need to reinstall bleed covers**
  - Bleed covers determined not be to a requirement but ideal, an opportunity to reinstall given tear-down was seized upon.
- **Spec O-ring seals to ensure proper dimensions**
  - Completed, see slide 11-12
- **Stager inlet tubes**
  - Completed, to be assembled on site and modifications required of off-the-self-part only
- **Perform analysis of Diffuser and shroud structure to ensure strength**
  - Completed, see analysis slide
- **Change inlet material to carbon steel**
  - Completed, see material slide 14.
- **Analyze impact on engine structure of loads due to pressurized inlet**
  - Completed, see analysis slide(s)

Final System Boundaries and Interfaces

- Aux air system under design (see air system CDR). It is anticipated that a close aboard header will be provided to link to.
- If this varies additional hoses and/or pipes can be used to reach source.
- System will necessarily apply structural loads to other systems in the rig (detailed in analysis section.)

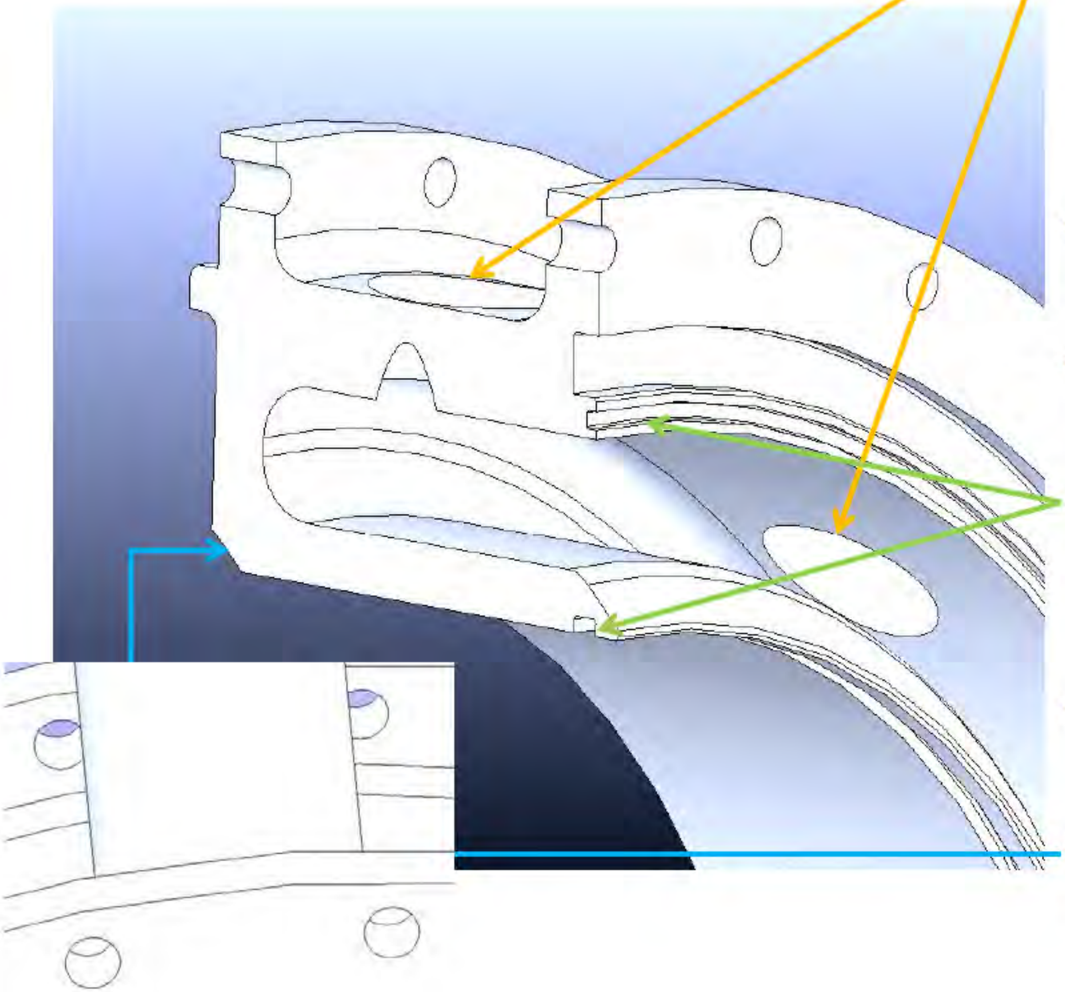
System Design

- **Overview :**



System Design

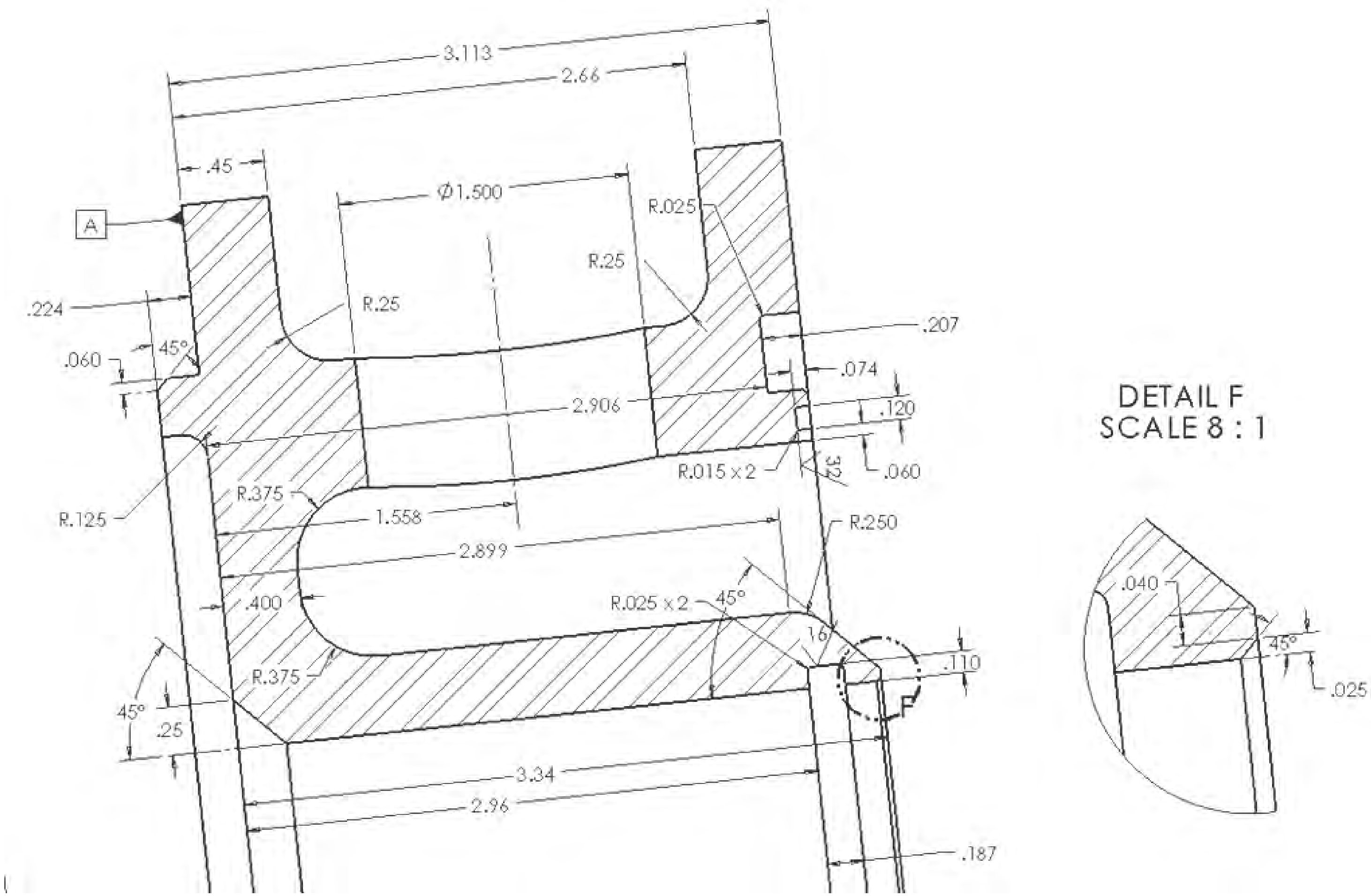
P/N 863010 Features



- 10 alternating 1 and 1.25 NPT threaded holes at regular radial allied intervals (6° offset centerline to avoid bolt features)
- 30 bolt holes fwd and aft (match existing components)
- Pilots fits fwd and aft (match existing components)
- Inner and outer glands to accommodate O-ring seals (Parker spec, see slide 11-12)
- Designed to accommodate easy tool access for machining
- Chamfer allows thrust-flow to escape.

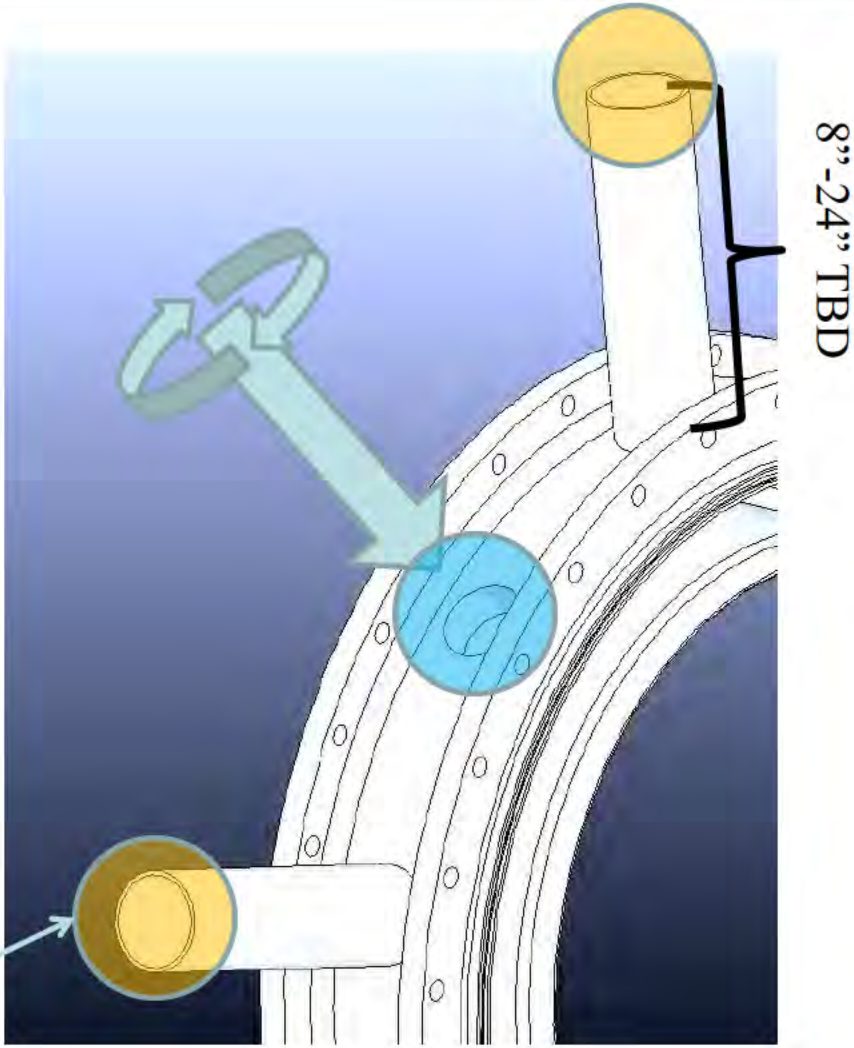
System Design

- **P/N ######## Dimensions**



P/N 863010

- **Once in position five 1.5 inch and five 1.25 inch tubes can be affixed to the inlet cap via NPT nipples and pipe to tube adapters.**
- **Tubes will vary in height to accommodate working area. On-site modifications will be needed to ensure ease-of-access.**
- **Tubes ultimately connect to air system via existing bleed hoses (with extensions and fittings needed see budget slide)**



RAMGEN POWER SYSTEMS™

System Design

Aero

- Table 1 shows the anticipated Mach numbers for the inlet during operation of the two experiments
- Maximum internal Mach number in the system projected to be 0.314 at ramp constriction.
- Loss of kinetic head at Inlet due to radial impingement to induce a tolerable pressure loss on the order of 10% with M=0.31

Hole Dia [in]	1.25	1.5
ID	1.01	1.26
Area [in^2]	0.801	1.247
Flow [lb/s]	10	

	6 to 1	10 to 1
Psi	87	147
Temp	120	120
p [lb/ft^3]	0.405	0.685

Experiment	6 to 1	Velocity	
Pipe	1.25	[ft/s]	M
hole #	10	443.52	0.37

Experiment	10 to 1	Velocity	
Pipe	1.25	[ft/s]	M
hole #	10	262.40	0.22

Experiment	6 to 1	Velocity	
Pipe	1.5	[ft/s]	M
hole #	10	284.98	0.24

Experiment	10 to 1	Velocity	
Pipe	1.5	[ft/s]	M
hole #	10	168.60	0.14

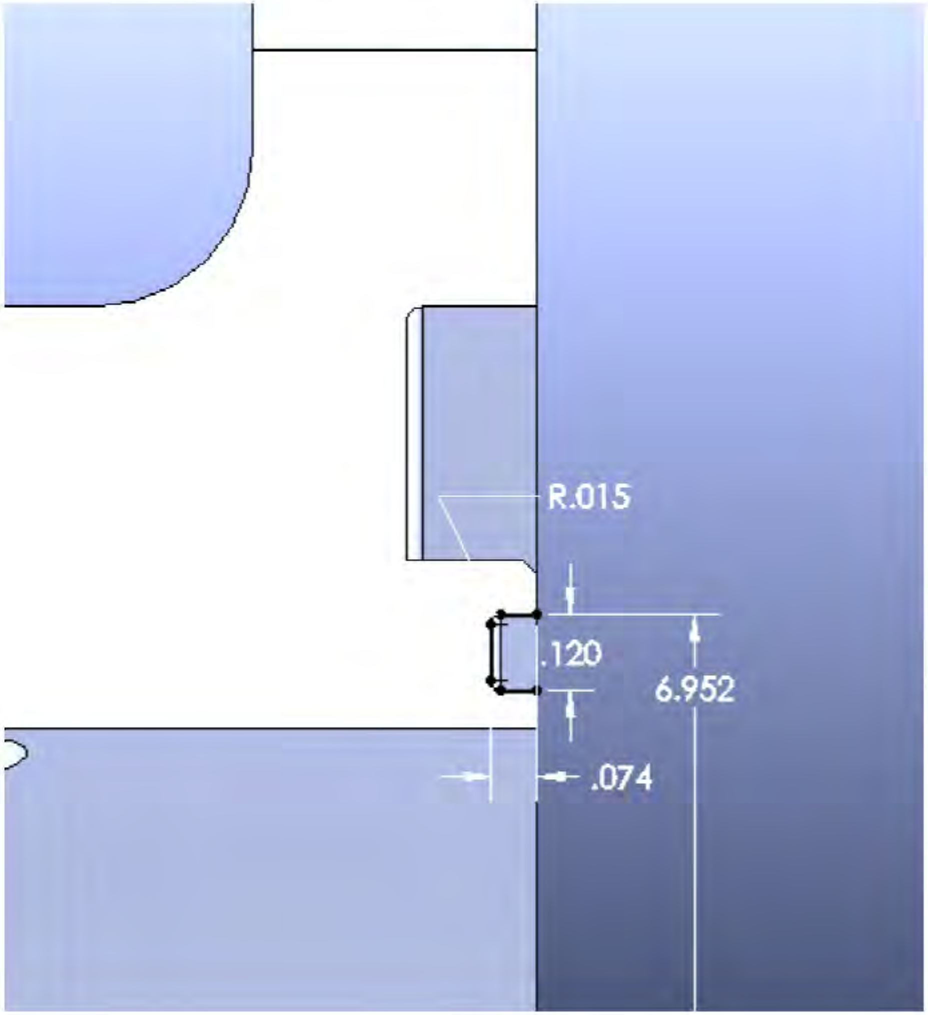
M	0.31

M	0.18

Table 1: Effect of inlet Tube diameter axial inlet velocity.

Average M given 5 of each tube size

System Design



Sealing

- **Face seal gland must seal the shroud chamber from the flow path due to drilled holes.**
- **Standard O-ring size (3/32") used with non-typical diameter (13.9" dia) for face seal.**

O-Ring Face Seal Glands These dimensions are intended primarily for face type O-ring seals and low temperature applications.

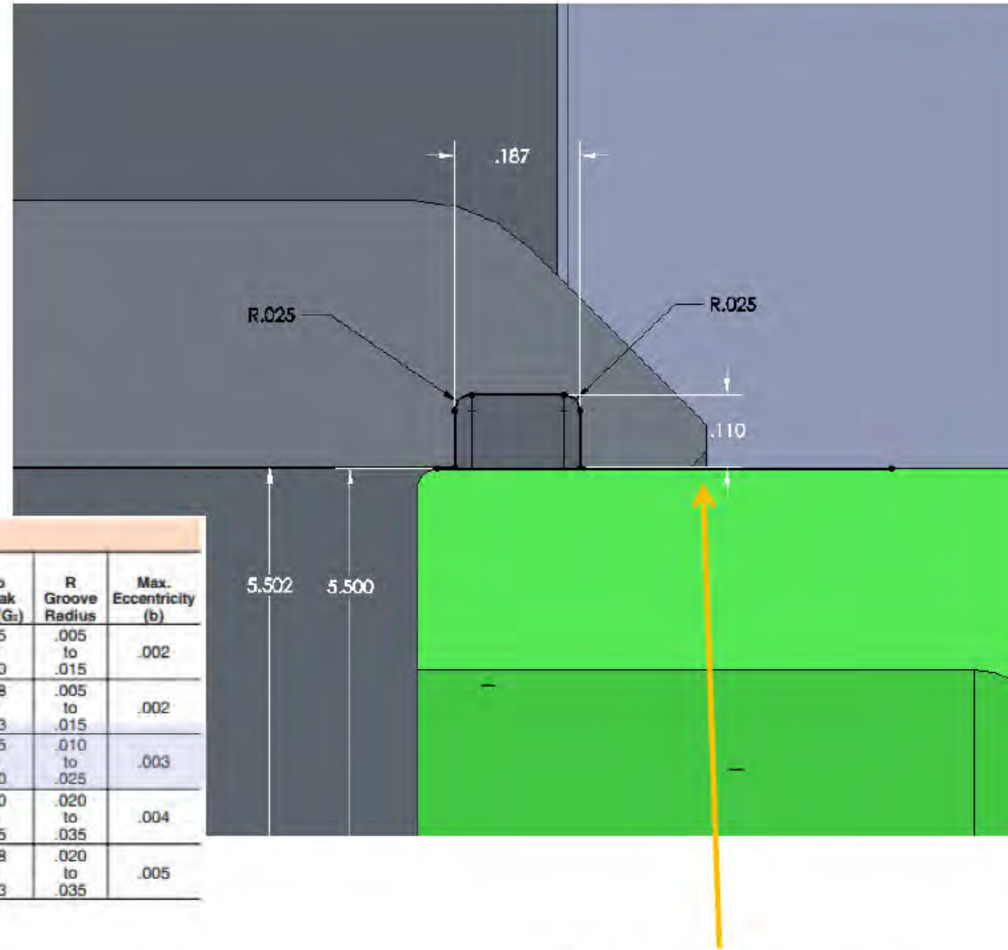
O-Ring Size Parker No. 2	W Cross Section Nominal	W Cross Section Actual	L Gland Depth	Squeeze Actual	Squeeze %	G Groove Width Liquids	G Groove Width Vacuum and Gases	R Groove Radius
004 through 050	1/16	.070 ±.003 (1.78 mm)	.050 to .054	.013 to .023	19 to 32	.101 to .107	.084 to .089	.005 to .015
102 through 178	3/32	.103 ±.003 (2.62 mm)	.074 to .080	.020 to .032	20 to 30	.136 to .142	.120 to .125	.005 to .015
201 through 284	1/8	.139 ±.004 (3.53 mm	.101 to .107	.028 to .042	20 to 30	.177 to .187	.158 to .164	.010 to .025
309 through 395	3/16	.210 ±.005 (5.33 mm)	.152 to .162	.043 to .063	21 to 30	.270 to .290	.239 to .244	.020 to .035
425 through 475	1/4	.275 ±.006 (6.99 mm)	.201 to .211	.058 to .080	21 to 29	.342 to .362	.309 to .314	.020 to .035
Special	3/8	.375 ±.007 (9.52 mm)	.276 to .286	.082 to .106	22 to 28	.475 to .485	.419 to .424	.030 to .045
Special	1/2	.500 ±.008 (12.7 mm)	.370 to .380	.112 to .138	22 to 27	.638 to .645	.560 to .565	.030 to .045

Design Chart 4-3: Design Chart for O-Ring Face Seal Glands

System Design

Sealing

- Piston seal gland seals the rotor chamber from the flow path
- Standard O-ring (1/8” x 11”) and groove used. Standard clearance used.



Forward Chamfer

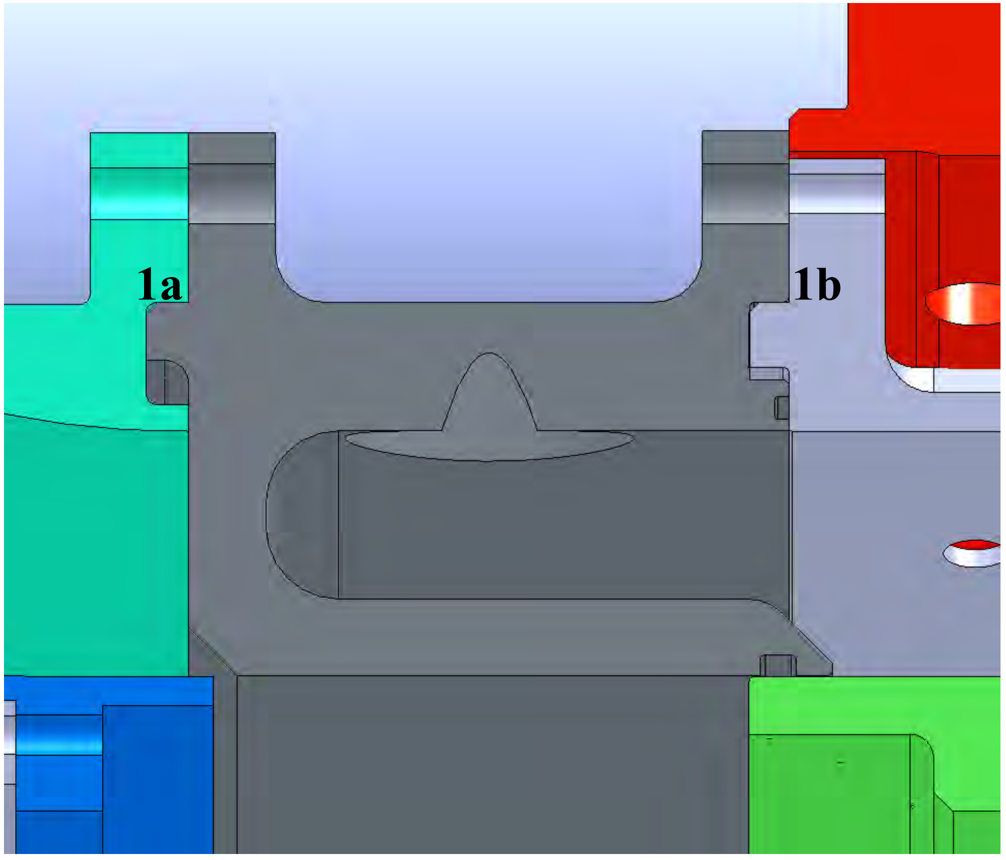
Industrial O-Ring Static Seal Glands

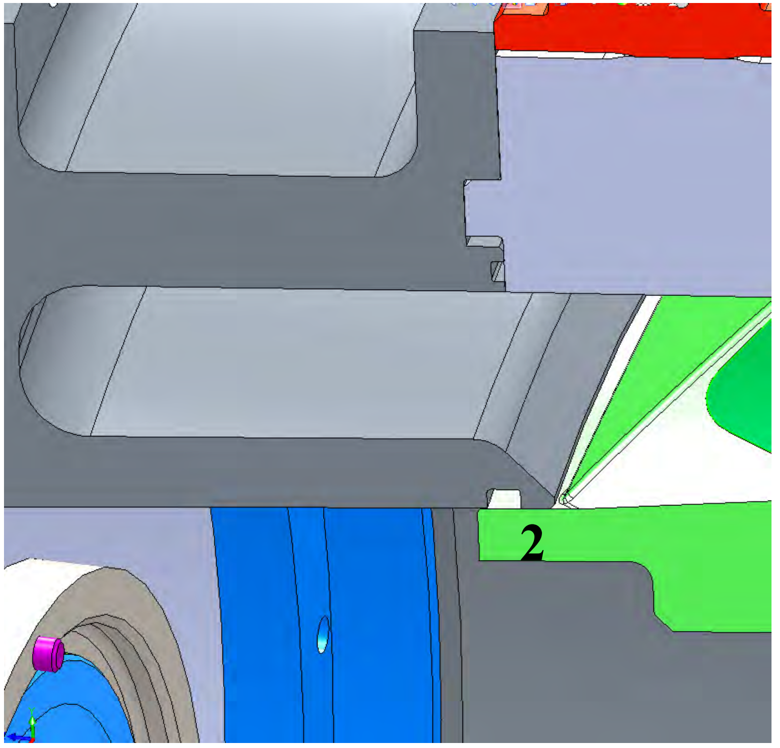
O-Ring 2-Size AS568B-	W Cross-Section Nominal	W Cross-Section Actual	L Gland Depth	Squeeze Actual	Squeeze %	E(a) Diametral Clearance	G - Groove Width: No Parbak Ring (G)	G - Groove Width: One Parbak Ring (G1)	G - Groove Width: Two Parbak Ring (G2)	R Groove Radius	Max. Eccentricity (b)
004 through 050	1/16	.070 ±.003 (1.78 mm)	.050 to .052	.015 to .023	22 to 32	.002 to .005	.093 to .098	.138 to .143	.205 to .210	.005 to .015	.002
102 through 178	3/32	.103 ±.003 (2.62 mm)	.081 to .083	.017 to .025	17 to 24	.002 to .005	.140 to .145	.171 to .176	.238 to .243	.005 to .015	.002
201 through 284	1/8	.139 ±.004 (3.53 mm)	.111 to .113	.022 to .032	16 to 23	.003 to .006	.187 to .192	.208 to .213	.275 to .280	.010 to .025	.003
309 through 395	3/16	.210 ±.005 (5.33 mm)	.170 to .173	.032 to .045	15 to 21	.003 to .006	.281 to .286	.311 to .316	.410 to .415	.020 to .035	.004
425 through 475	1/4	.275 ±.006 (6.99 mm)	.226 to .229	.040 to .055	15 to 20	.004 to .007	.375 to .380	.408 to .413	.538 to .543	.020 to .035	.005

(a) Clearance (extrusion gap) must be held to a minimum consistent with design requirements for temperature range variation.
(b) Total indicator reading between groove and adjacent bearing surface.
(c) Reduce maximum diametral clearance 50% when using silicone or fluorosilicone O-rings.
(d) For ease of assembly, when Parbaks are used, gland depth may be increased up to 5%.

Design Chart 4-2: For Industrial O-Ring Static Seal Glands

System Design





Details of critical junctions, note pilot fits [1] and proximity to inducer strake [2] (.031”). Pilot fit clearances will be identical to current tip ring as per its production drawing

System Design

- **Materials**
  - **Carbon Steel**
    - 12L14, 60 ksi yield
  - **Electroless nickel plate, AMS 2404 on the order of .0005-.001” thickness**
    - “…This deposit has been used typically to provide a uniform build-up on intricate shapes, to improve wear and/or corrosion resistance, or to improve solderability on or for selected materials, but usage is not limited to such applications. The deposit has been used in service up to 1000 °F (540 °C) although wear and/or corrosion resistance may degrade as service temperature increases.” ~ SEA
  - **Stainless Steel welding for diffuser case.**
    - Confirmed with welder that with removal of components as shown in slide 6, and assuming unfavorable weld properties of substrate it would be possible.

Analyses Performed

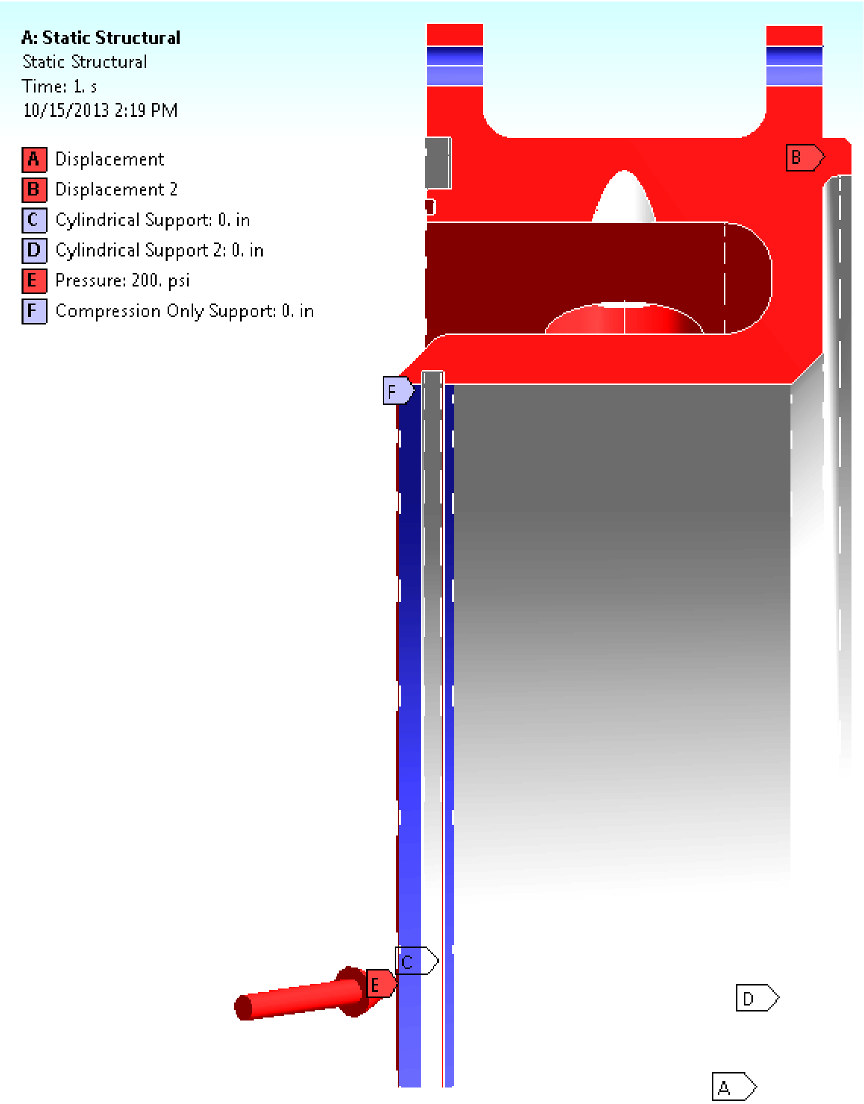
- CFD on OGV for the ISCE B1 under M<1
- Thermal-structural analysis on rotating rotor to establish rotor clearance under maximum allowable speed.
- Structural analysis on Inlet section to establish performance under maximum allowable load.
- Structural analysis on diffuser hub and shroud to establish performance under load and monitor the effect of reaction forces from Inlet section.
- Structural analysis on OGV shroud section as per above.
- Structural analysis on combustor casing section as per above.
- Structural analysis on turbine shroud section as per above
- Qualitative projections on overall bolt loading.

Analyses Results

- **CFD on OGVs:**
  - Aero review confirms adequate flow into combustor sections given sub-sonic inlet conditions (see requirement #1 slide 2 , and slide 10).
- **Dynamic thermal-structural loading of rotor:**
  - Induces .015” radial growth of rotor under maximum loading conditions minimally impacting rotor clearance gap to new Inlet section (see PRR reference slides [attached]).

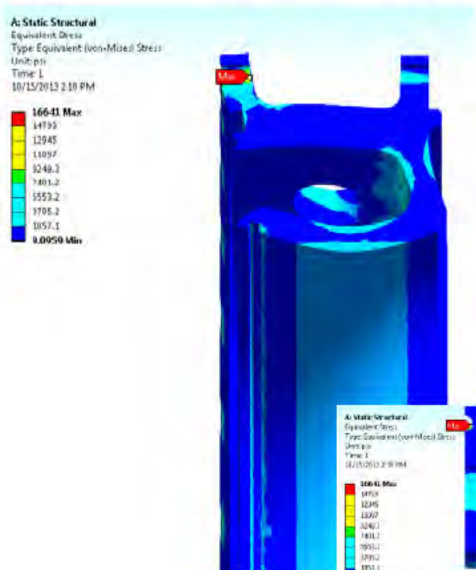
Analyses Results

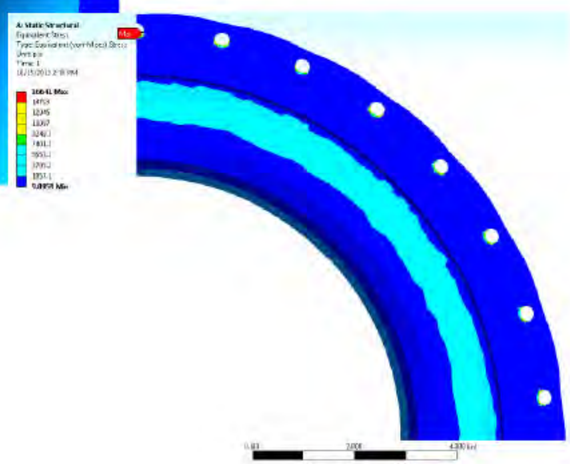
- **Structural analysis on Inlet**
  - ANSYS FEA constraints:
    - 200 Psi internal load vs. vacuum
    - X-Y displacement constraints along ¼ cut
    - Axial and tangentially fixed cylindrical support on aft bolt circle, tangential only on fwd bolt circle.
    - Compression only support on [F](contact with inducer hub)

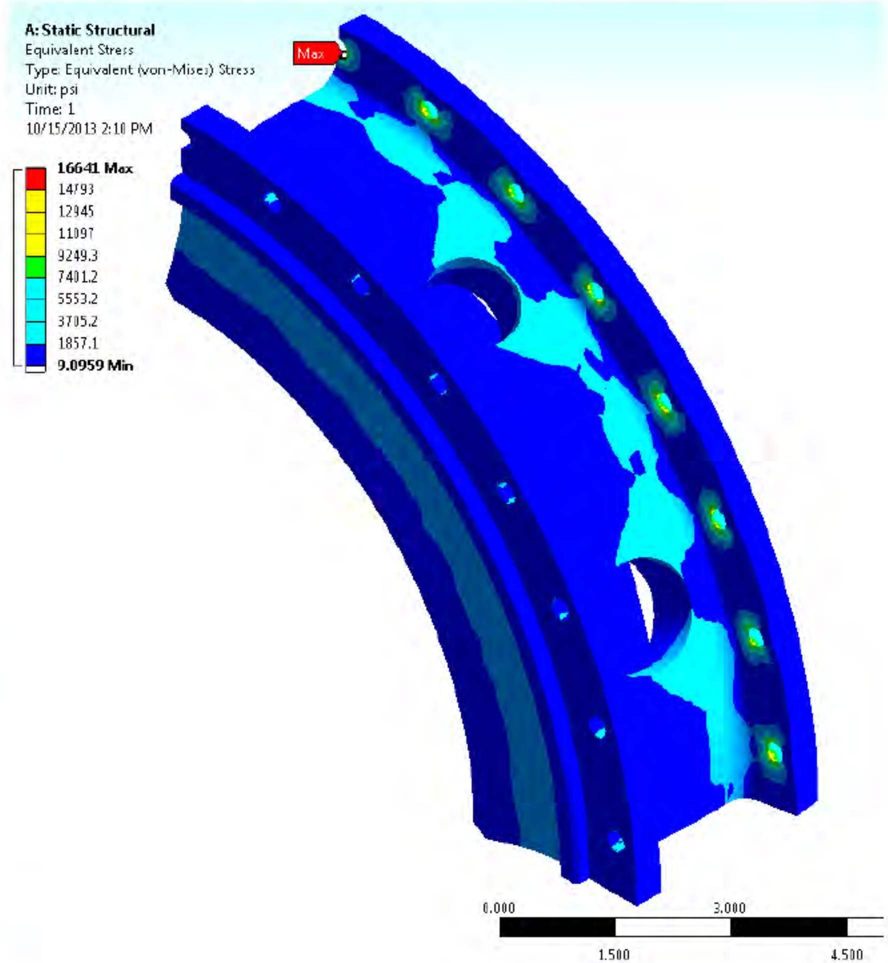


Analyses Results

- **Structural analysis on Inlet**
  - Stress
    - 16 ksi max (von-mises)

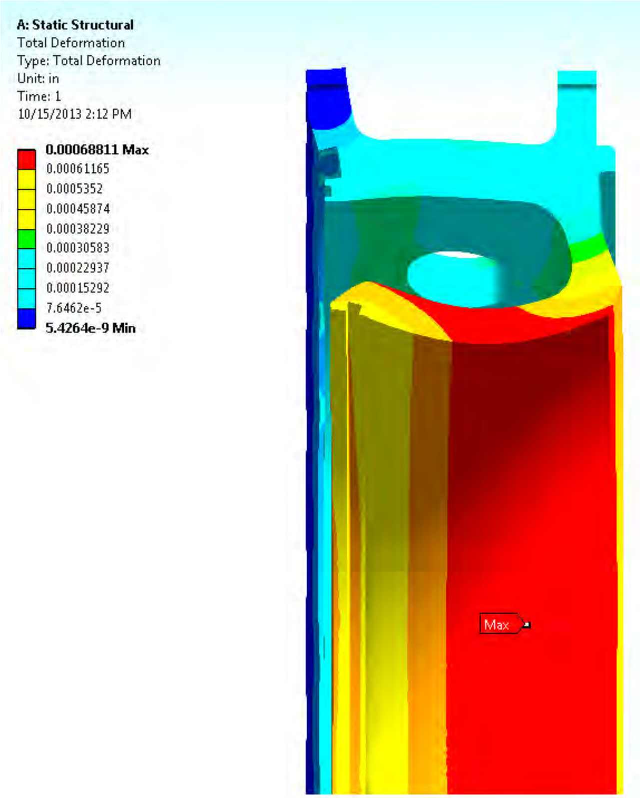


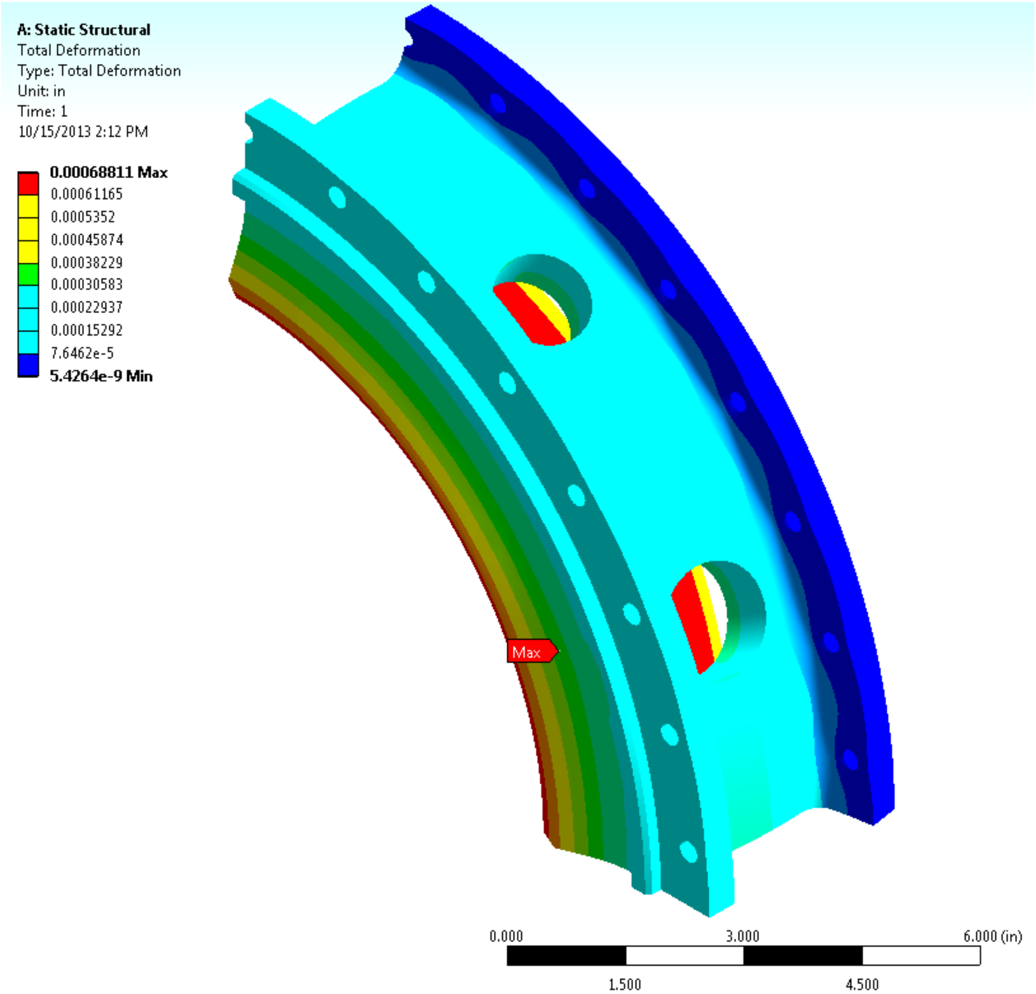




Analyses Results

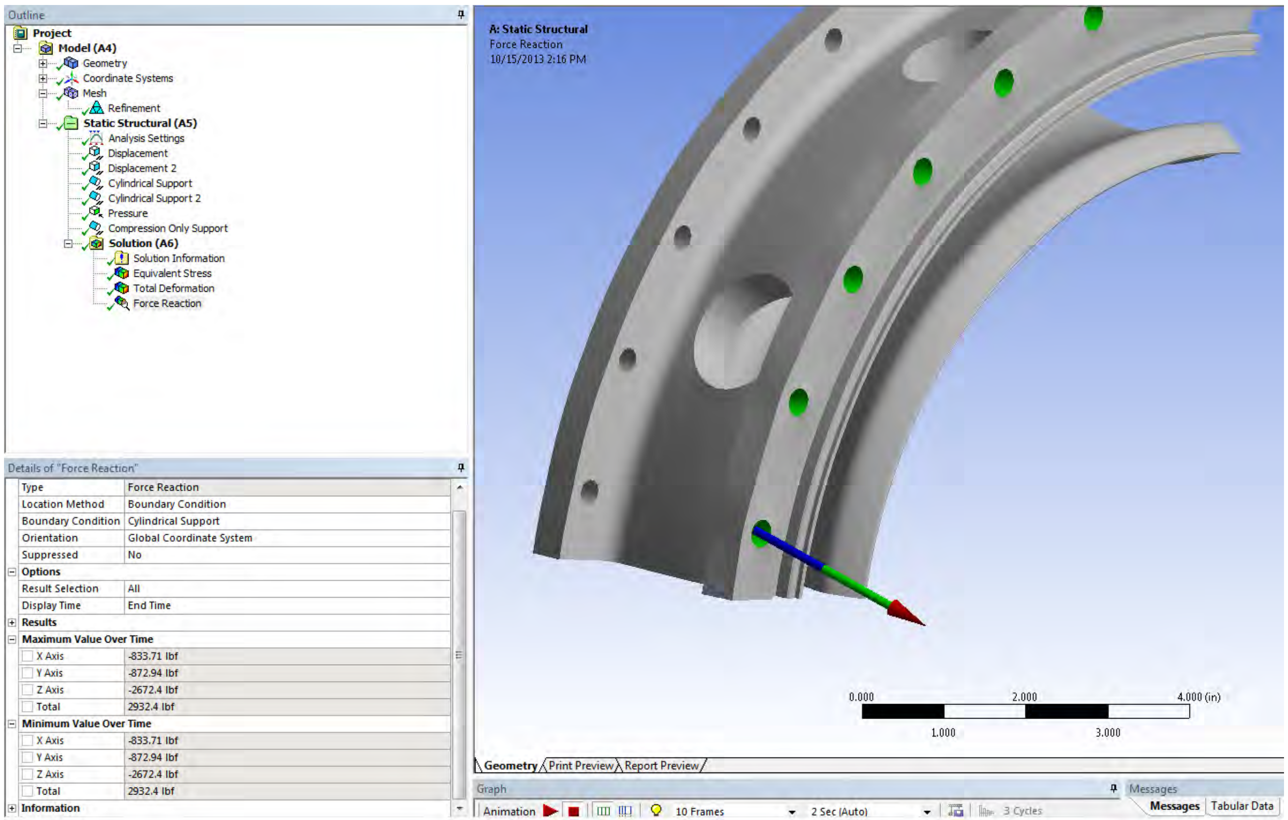
- **Structural analysis on Inlet**
  - Deformation
    - .0007" max





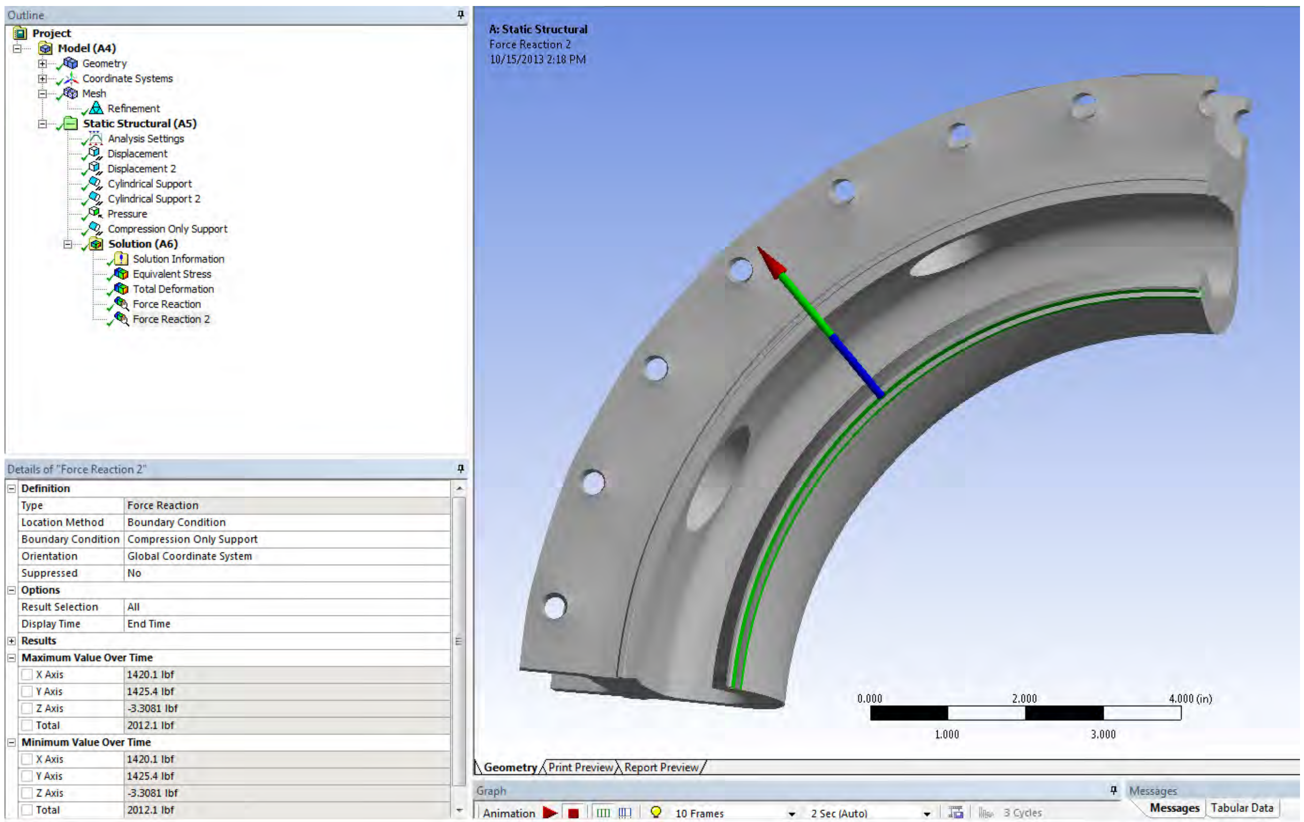
Analyses Results

- **Structural analysis on Inlet**
  - Reaction
    - 2672 lbf reaction in the axial direction for the ¼ piece yielding a total load on the order of 10,700 lbf



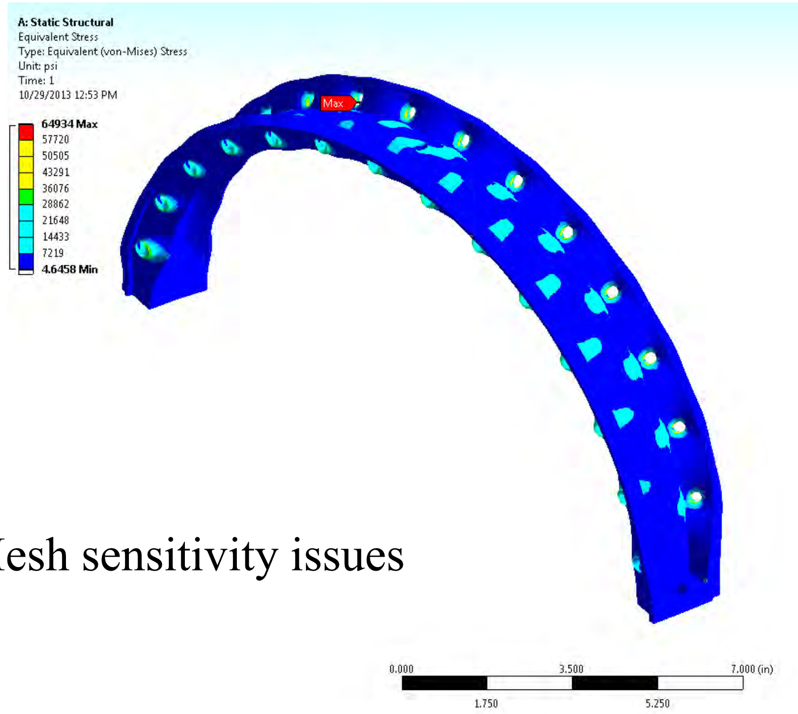
Analyses Results

- **Structural analysis on Inlet**
  - Compression only reaction
    - 2012 lbf reaction in the radial direction for the ¼ piece. Resting on the diffuser hub.

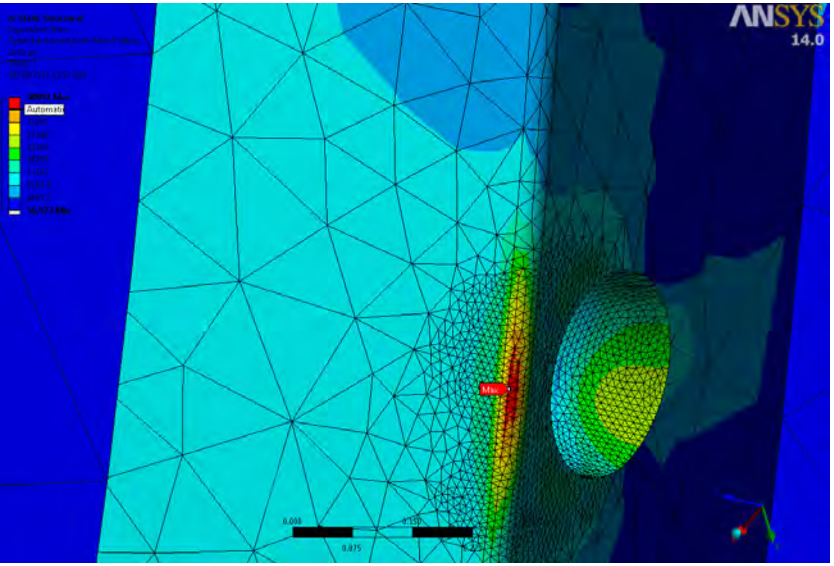


Analyses Results

- **Structural analysis on OGV shroud section**
  - Stress
    - <u>41 ksi Max</u>
    - Internal load of 200 psi
    - axial constraints on bolt holes
    - Reaction force of ~5300 lbf applied through bolt pattern



Mesh sensitivity issues

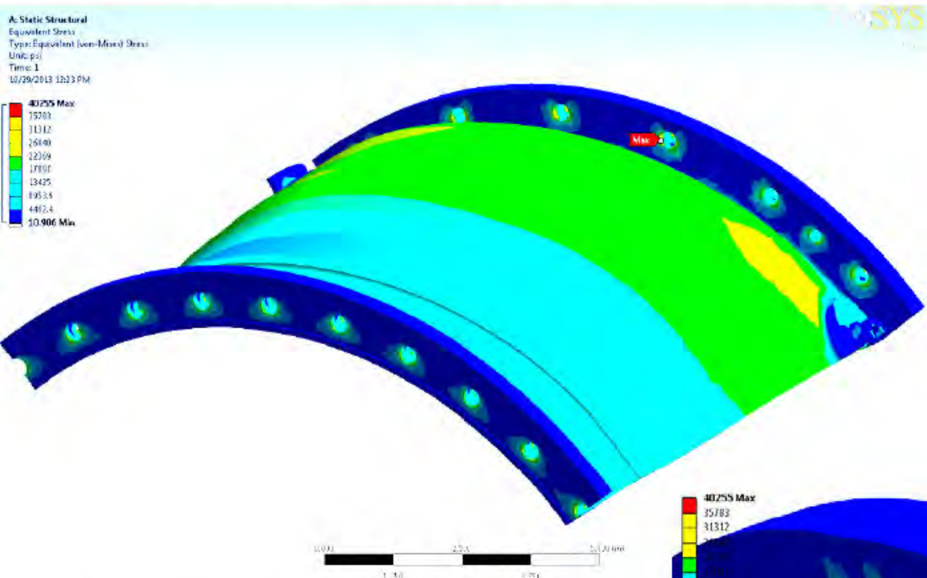


Constructed of annealed 410 SS with yield strength of <u>45 ksi</u>
<u>Tempering to 1200F can raise hardness up to 90 ksi</u>

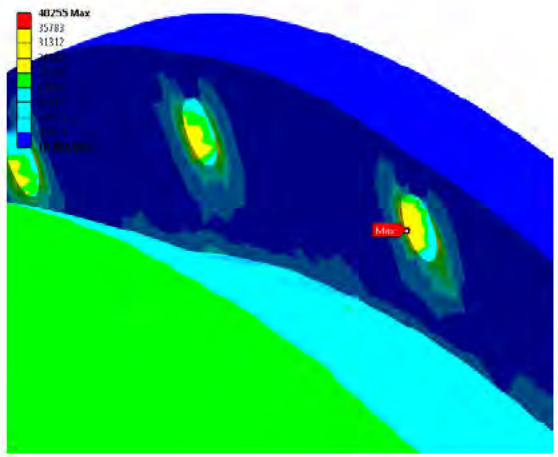
Analyses Results

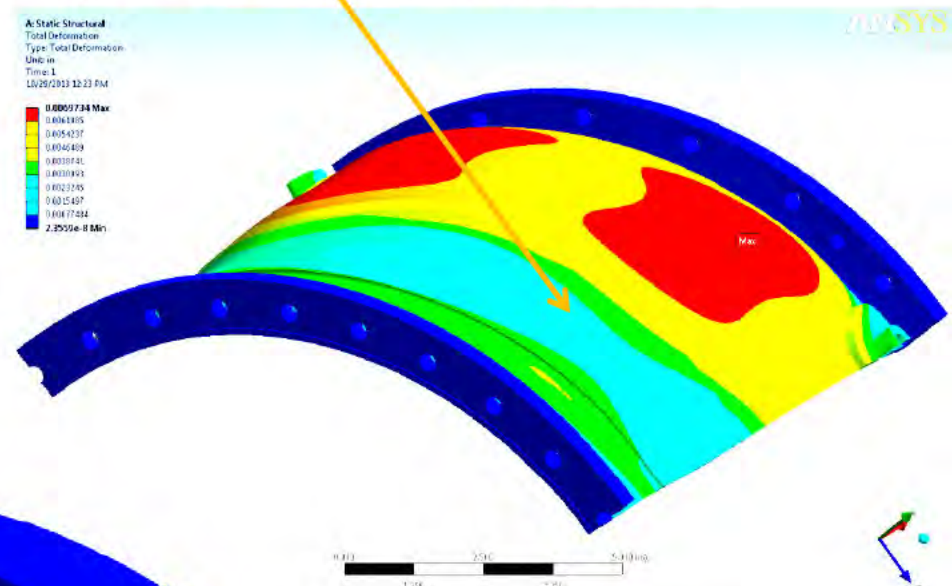
Welded strip to seal case

- **Structural analysis on combustor casing section (minimal mods)**



Stress [40.2 ksi max]





Deformation [.006" max]

15-5 PH

Analyses Results

- **15-5 PH**

Mechanical Properties:						
F_{tu}, ksi:						
L	190	170	155	145	140	135
T	190	170	155	145	140	135
F_{ty}, ksi:						
L	170	155	145	125	115	105
T	170	155	145	125	115	105

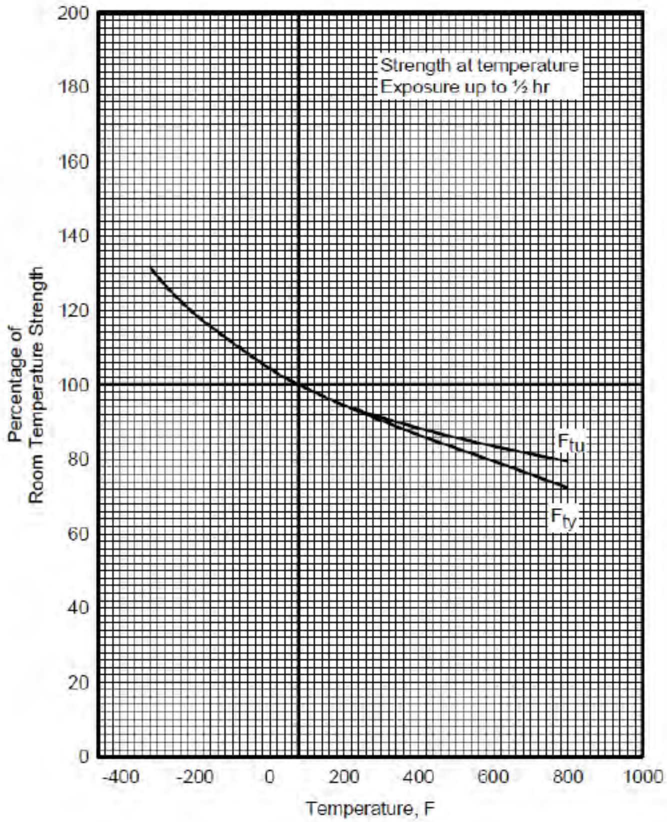
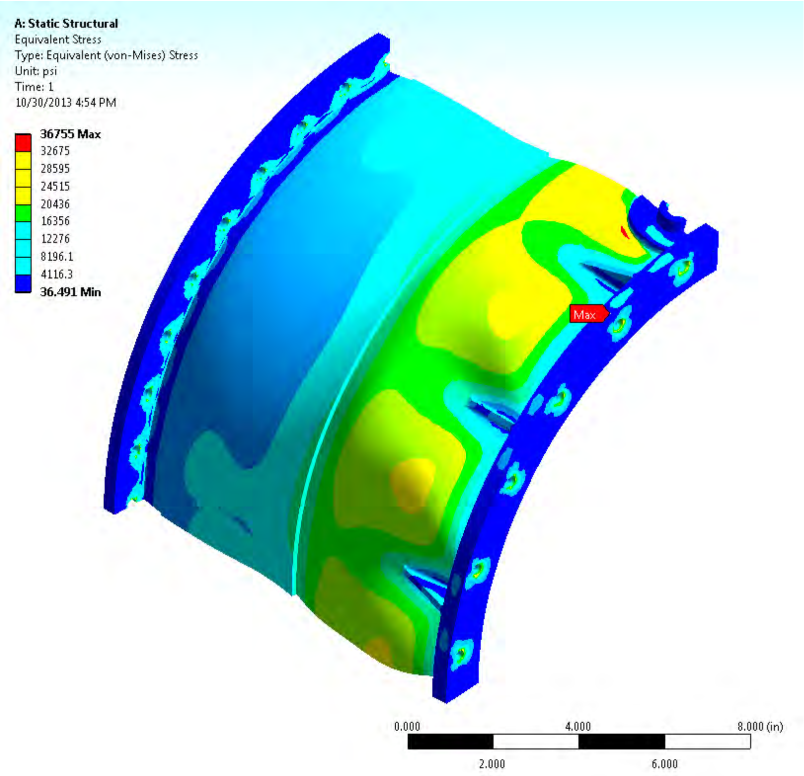


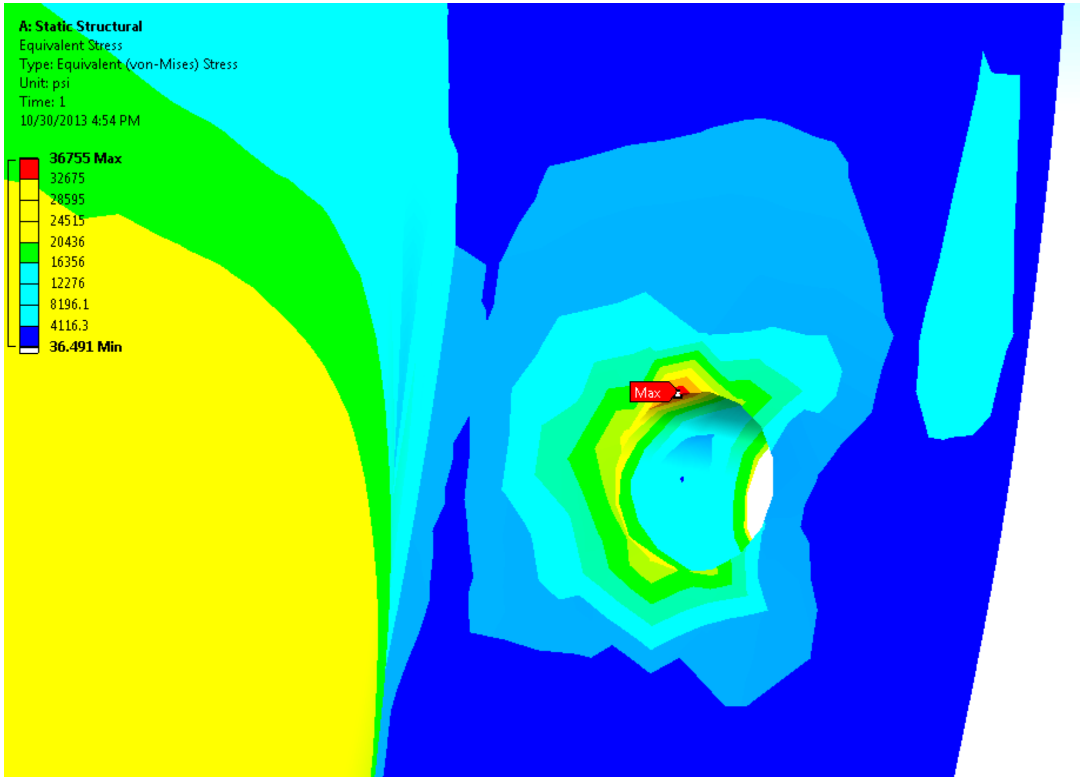
Figure 2.6.7.1.1. Effect of temperature on the tensile ultimate strength (F_{tu}) and the tensile yield strength (F_{ty}) of 15-5PH (H925, H1025, and H1100) stainless steel bar.

Lowest yield found to be 105 ksi providing significant strength up to 800 F (30% loss)

Analyses Results

- **Structural analysis on combustor casing section (with gussets)**

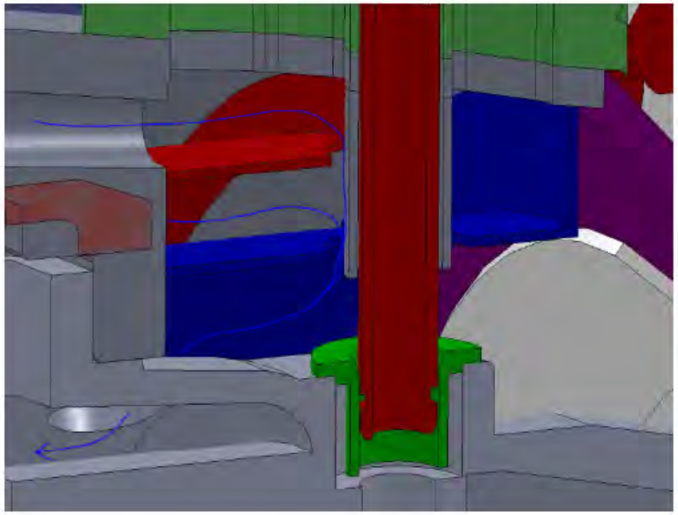


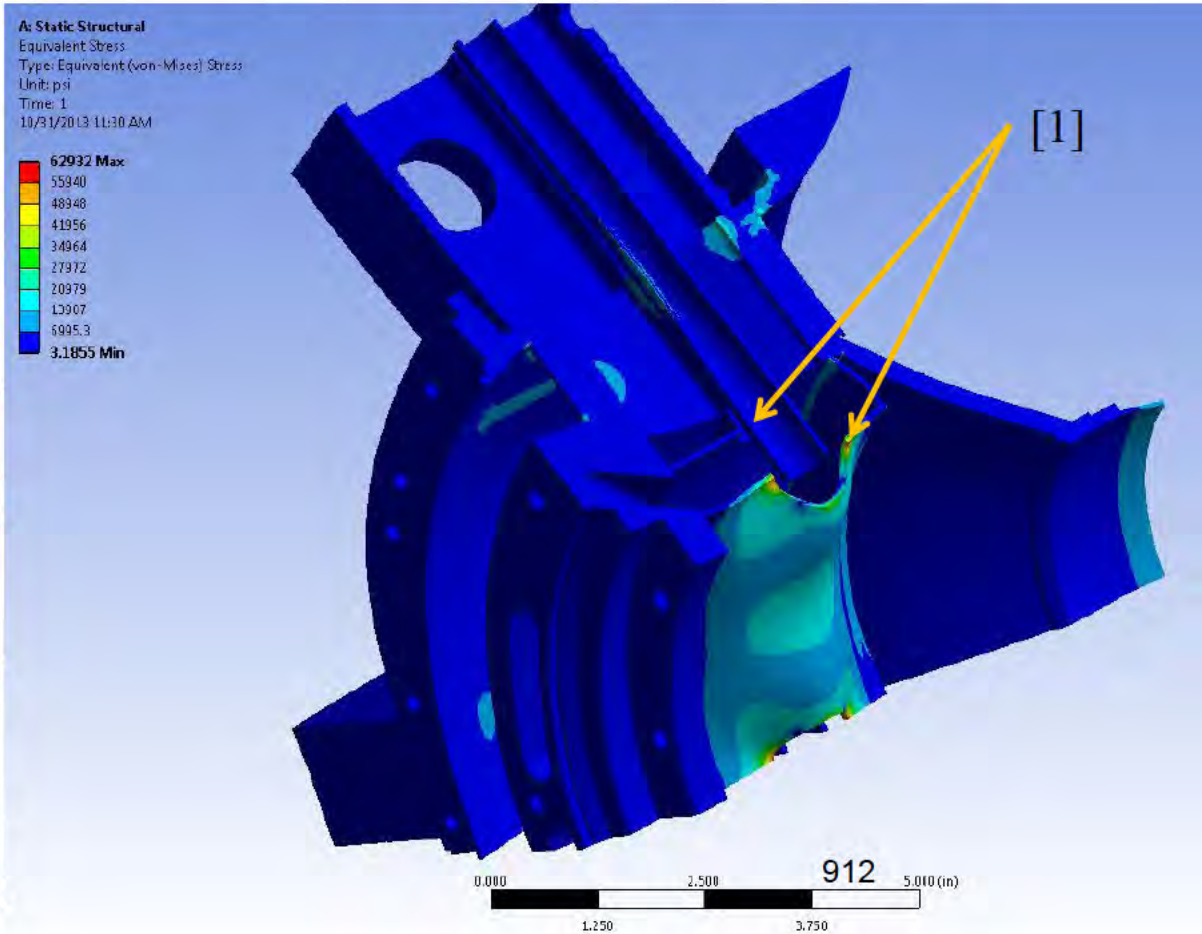


Stress [36.7 ksi max]

Analyses Results

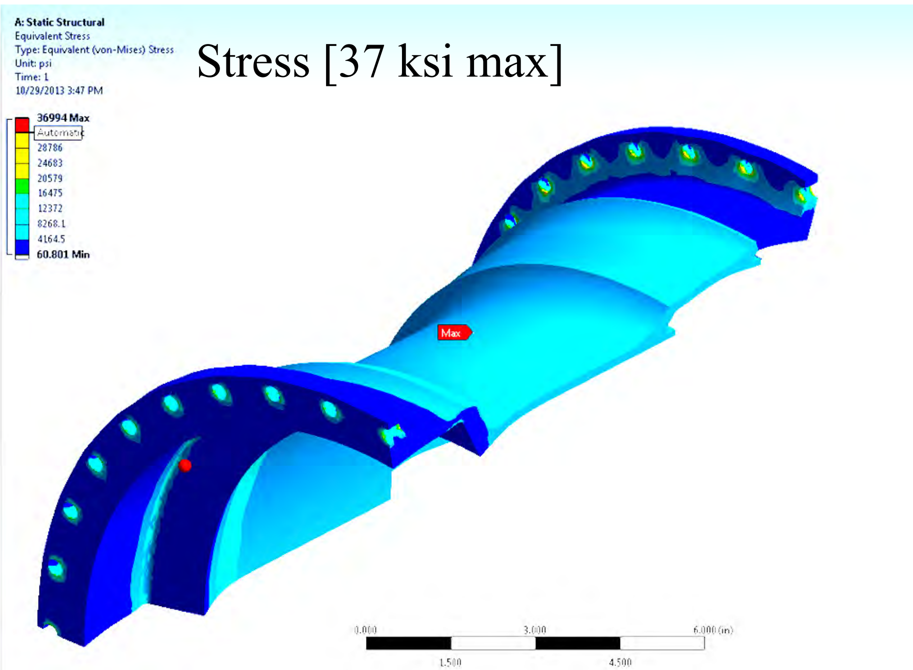
- **Structural analysis on Discharge Duct (P/N 11110)**
  - Constructed of 17-4 ph Steel with yield strength of 105 ksi (mil-spec)
  - Inputs form Steve confirm the model as presented in not as-built and welds seal bleed annulus [1]
- **Constraints**
  - Displacement constraints at secured locations
  - Internal 200 psi

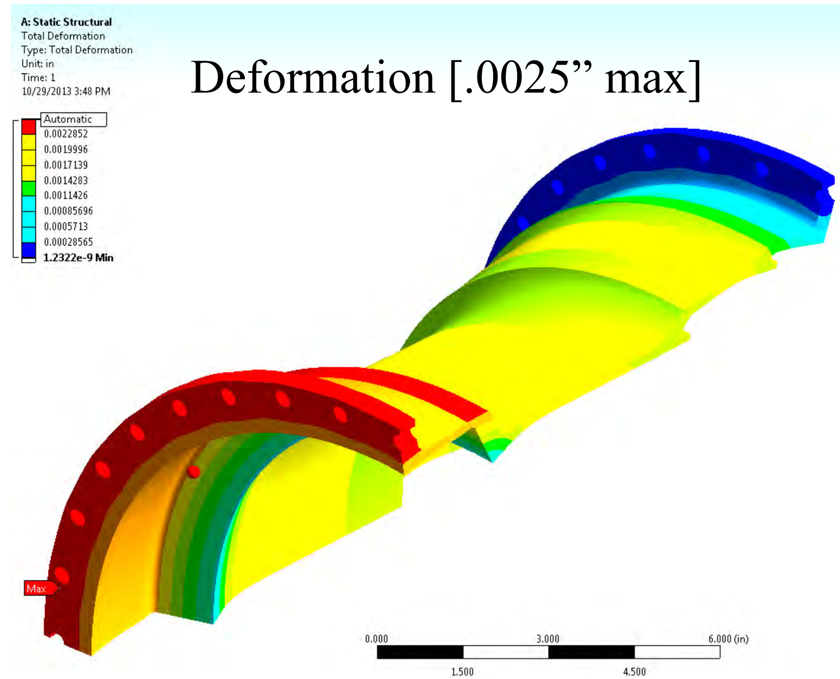




Analyses Results

- **Structural analysis on turbine shroud**





From GLM answers document 9/9/13:
“Turbine section structural casing
Believed to be 420 SS, maybe 430 SS”.

Min for 420 annealed: 50 ksi*
Min for 430: 30 ksi

Analyses Results

- **Overall bolt loading**
  - Simple division yields approximate tension on each bolt

[36] 5/16-24
294 lbf each
3.8 ksi

[36] 5/16-24
294 lbf
3.8 ksi

[24] 5/16-24
441 lbf
5.7 ksi

2x[28] 1/4-20
378 lbf
7.7 ksi

Replace *all* bolts during reassembly with high grade.

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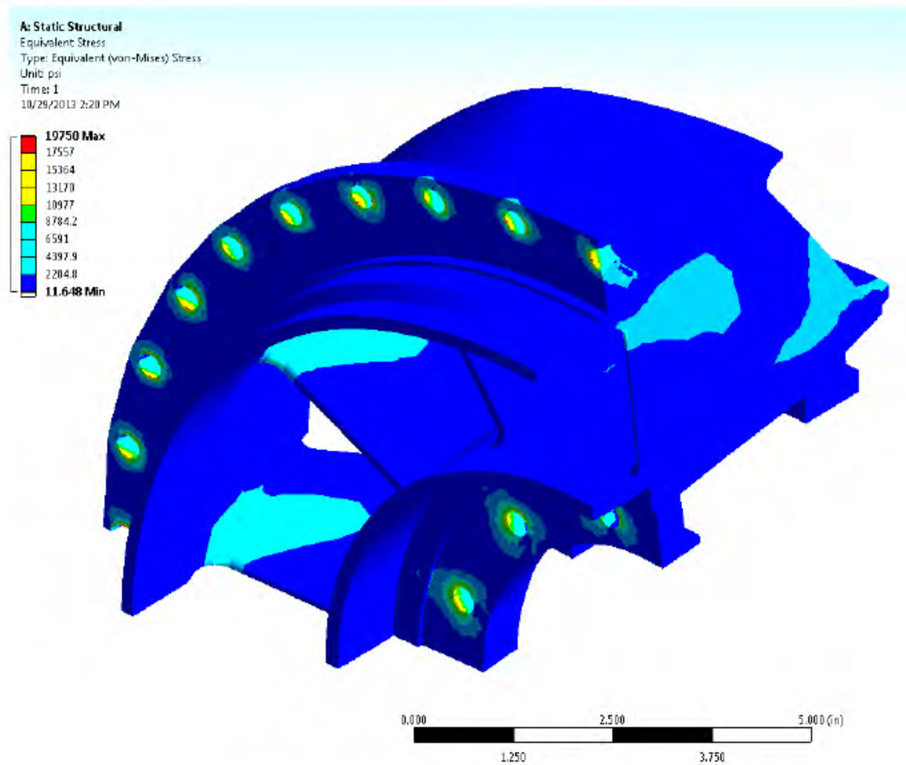
Analyses Results

[12] 3/8-24
883 lbf
7.9 ksi



20 ksi max stress

Budget and Schedule

- **Budget allocation of $40,000**

- **Main inlet component**
  - United Machine & Design [includes plate]: **$13,700 @ 6 weeks**
  - Mueller [confirmation on plate pending] : **$6,320 @ 8 weeks**
- **Stainless steel tube .120 wall thickness, cut on site**
  - Grainger **$800 @ 2 days** (off-the-shelf)
- **Pipe nipples and Swagelok fittings**
  - **$2000 estimate @ 2 weeks** (off-the-shelf)
- **New fittings**
  - **$400**
- **Welding**
  - **$5000**

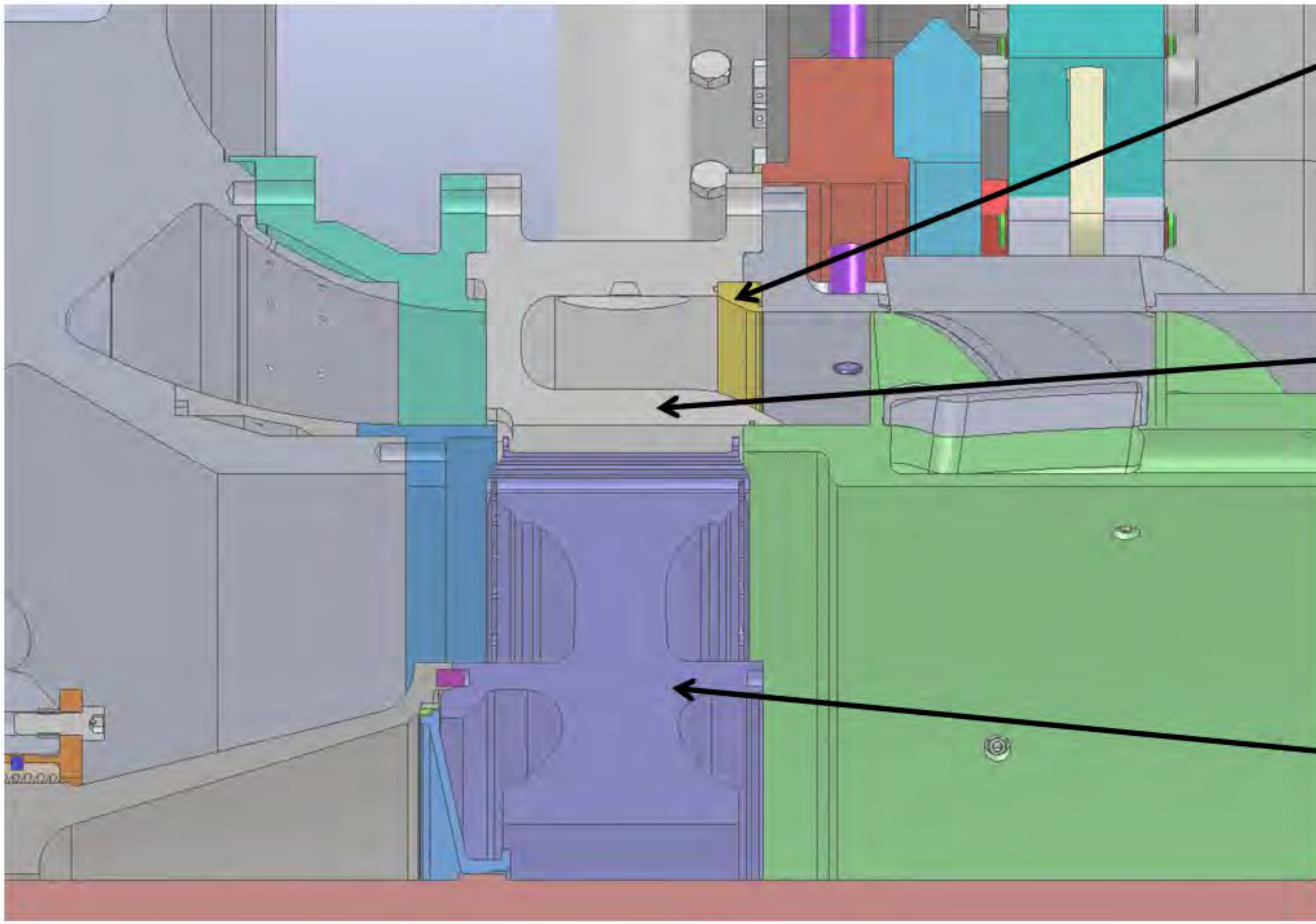
- **$21,900 expected total.**

Conclusion

- **Rig appears to be able to accommodate inlet design, recommend welding of additional gussets to combust case to ensure sufficient strength, and further review required for turbine shroud.**
- **Thank you for your time and input.**
  - PDR reference slides to follow

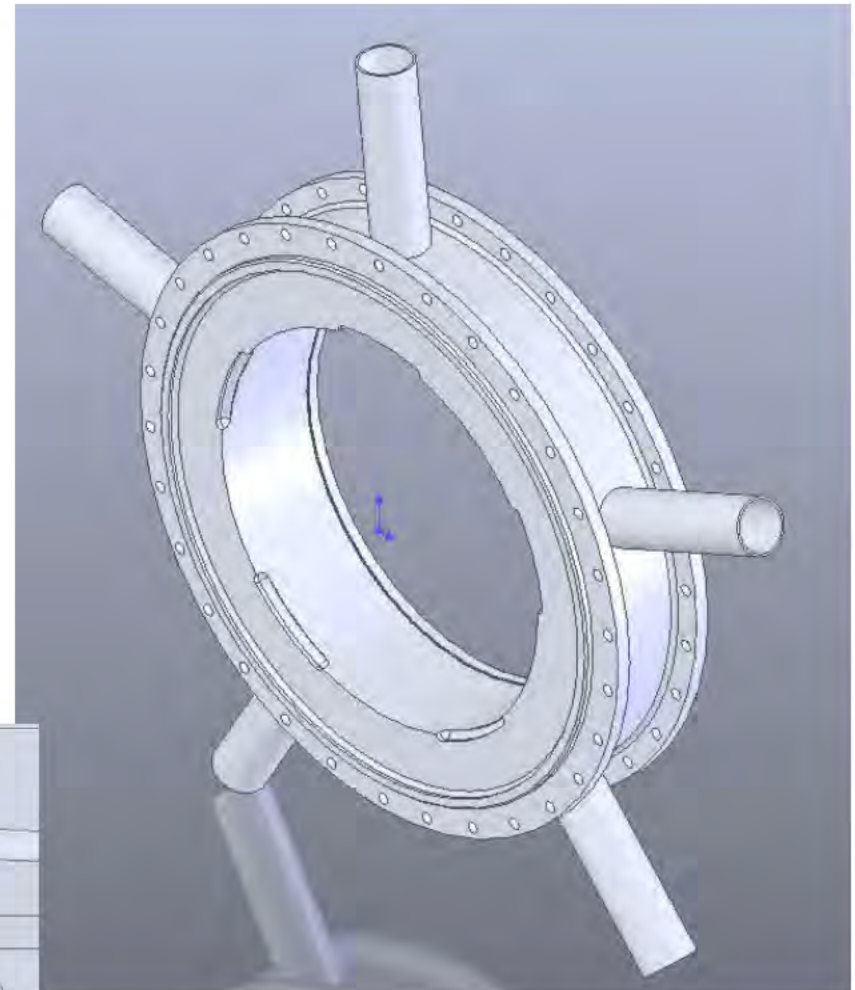
Rev 01

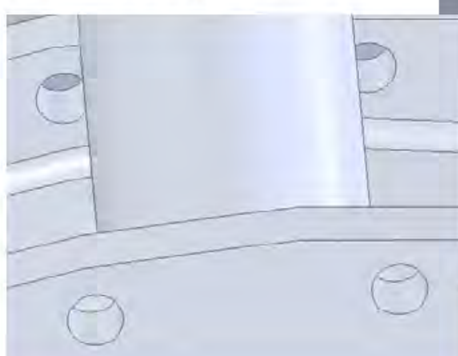
Two component inlet as viewed in current assembly



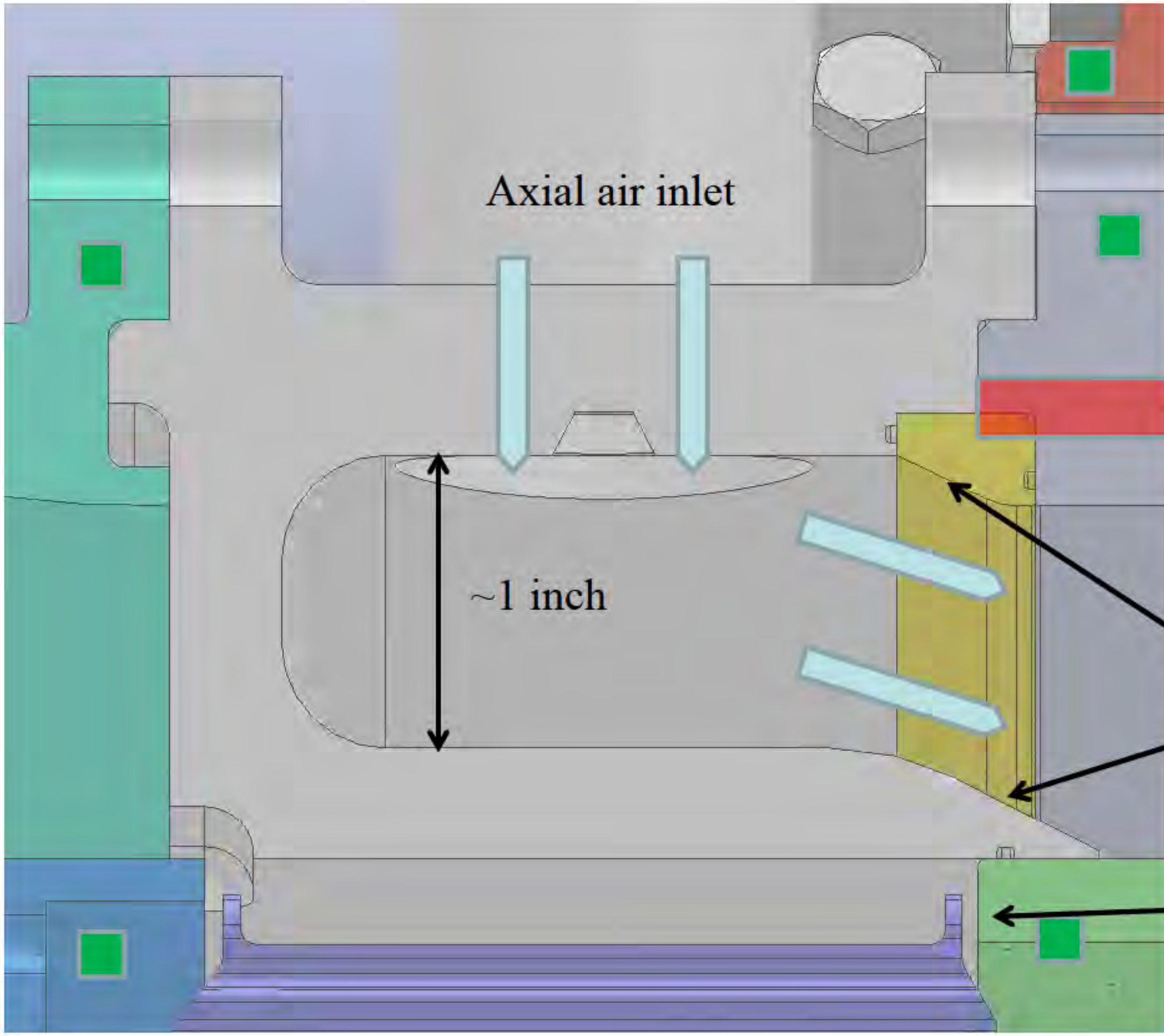
Rev 01

- **Inlet/cap**
  - Constructed of weld-friendly SS (410-316 TBD)
  - Designed to maximize tool access for ease of manufacture
  - Inlet horns accommodate existing methods of air delivery and ease of assembly (number of horns is variable; see later slides)
  - Horns are clocked 6 degrees out of phase with center line to dodge bolt pattern





Rev 01



With this design all components surrounding the cap and sealing ring are unmodified and in identical orientation to the prior configuration.

Through-hole to door space (must be sealed)

Flats/radius provide for smother flow

.125 clearance to rotation component

= Existing

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Requirements

Hole Dia [in]	1.25	1.5
ID	1.12	1.37
Area [in^2]	0.985	1.474
Flow [lb/s]	10	

	6 to 1	10 to 1
Psi	87	147
Temp	180	180
p [lb/ft^3]	0.367	0.620

Experiment	6 to 1	Velocity		Experiment	10 to 1	Velocity	
Pipe	1.25	[ft/s]	M	Pipe	1.25	[ft/s]	M
hole #	5	796.86	0.64	hole #	5	471.77	0.38
	6	664.05	0.53		6	393.14	0.32
	7	569.18	0.46		7	336.98	0.27
	8	498.04	0.40		8	294.86	0.24
	9	442.70	0.36		9	262.09	0.21
	10	398.43	0.32		10	235.88	0.19

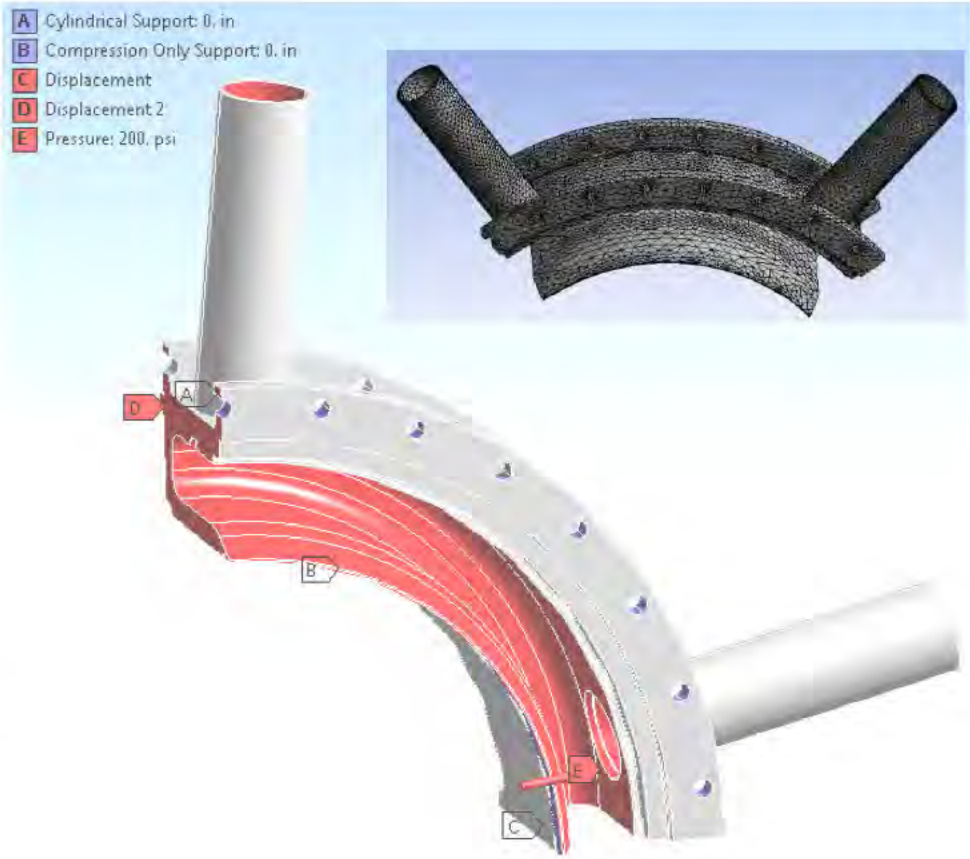
Experiment	6 to 1	Velocity		Experiment	10 to 1	Velocity	
Pipe	1.5	[ft/s]	M	Pipe	1.5	[ft/s]	M
hole #	5	532.57	0.43	hole #	5	315.30	0.25
	6	443.81	0.36		6	262.75	0.21
	7	380.41	0.31		7	225.21	0.18
	8	332.86	0.27		8	197.06	0.16
	9	295.87	0.24		9	175.17	0.14
	10	266.28	0.21		10	157.65	0.13

Table 1: Effect of inlet pipe diameter and number on axial inlet velocity for Rev 01

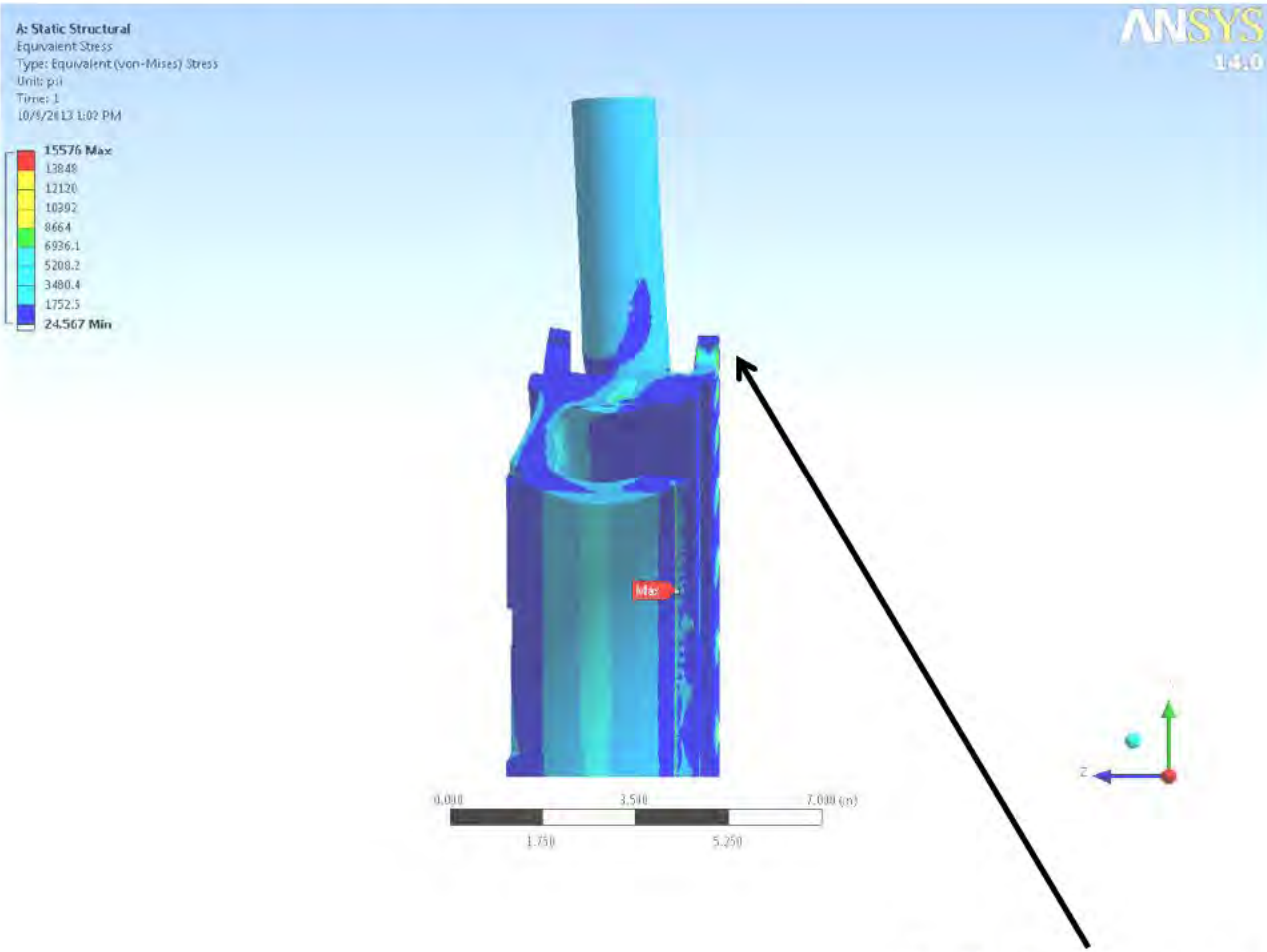
- **Deliver 87 (and/or 147) psi, 180 F air at 10 lb/s to static combustor with equivalent flow uniformity to existing OGV output**
  - Design can deliver air flow, and flow uniformity is accomplished with existing OGVs
  - Mach numbers can be managed with increase of pipes (table 1) and max mach number experienced in static diffuser is .314 independent of inlet configuration.

Requirements

- **Accommodate up to 200 psi internal pressure**
  - ANSYS analysis of the Inlet performed with an internal pressure condition of 200 psia to establish maximum anticipated stress and deformation.
  - Potential fabrication materials have a minimum yield stress of 45 Ksi (410 annealed) temperature with a variations of no more than 10% at max considered temperature of 325 F.
  - Main constraint for model was cylindrical supports allowing for radial growth on aft bolt circle anticipating forward load.



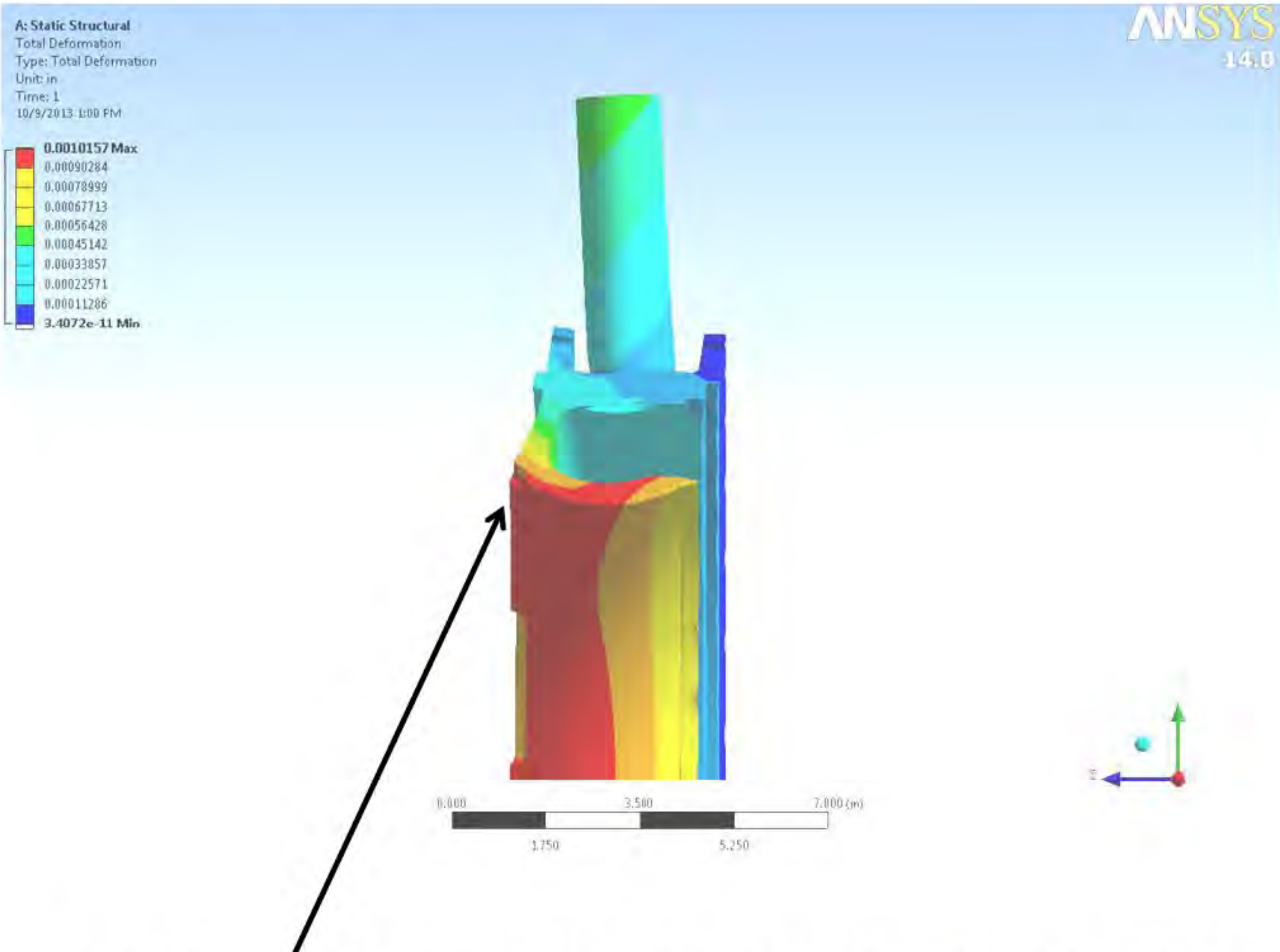
Requirements



Cylindrical support

- Max stress of 15 Ksi experienced along compression only support of front lip
  - SF of over 2.5 at temp.

Requirements

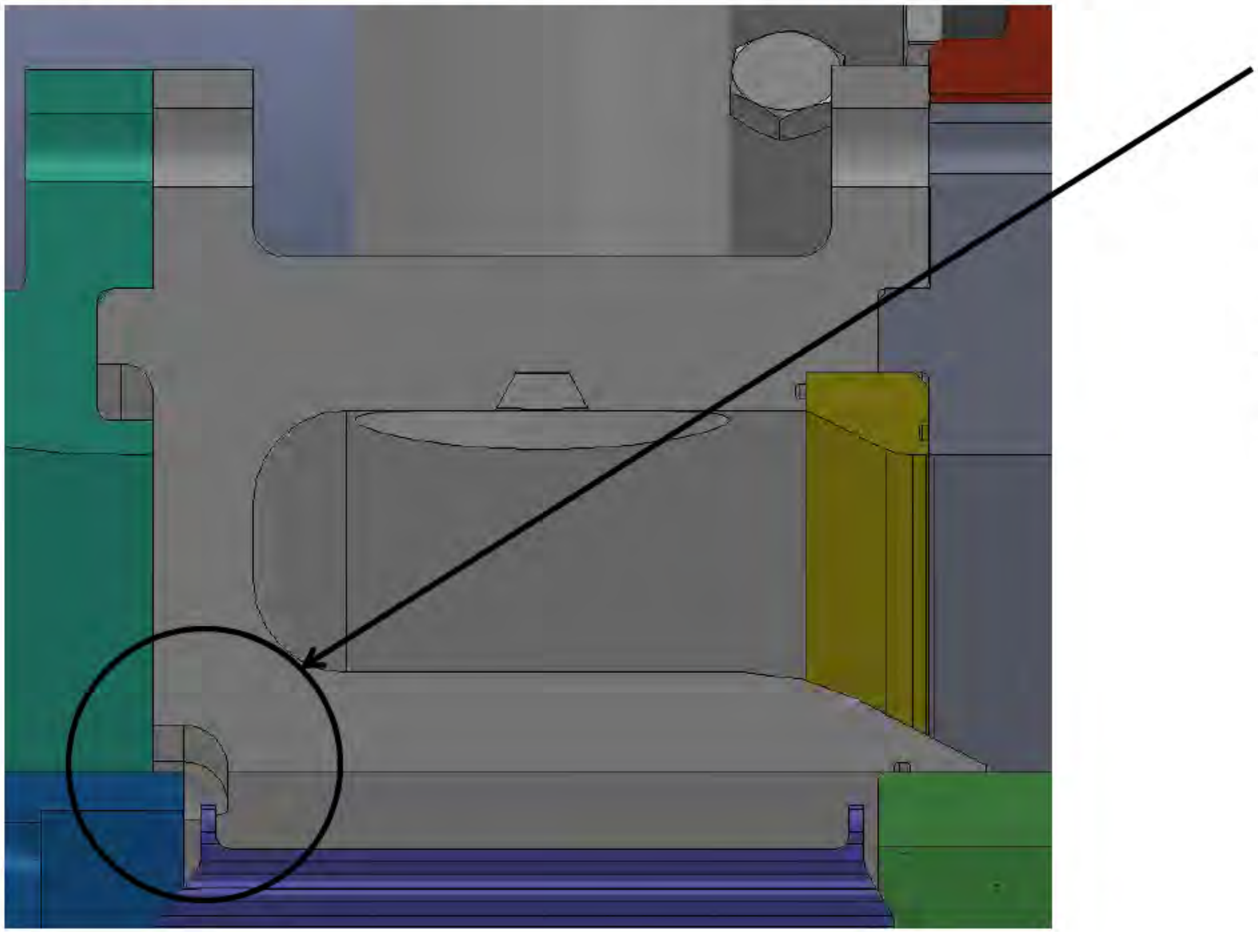


Unsupported deformation of .001 max experienced

Negative radial movement minimal, closing gap to rotor to .124 while stationary.

Requirements

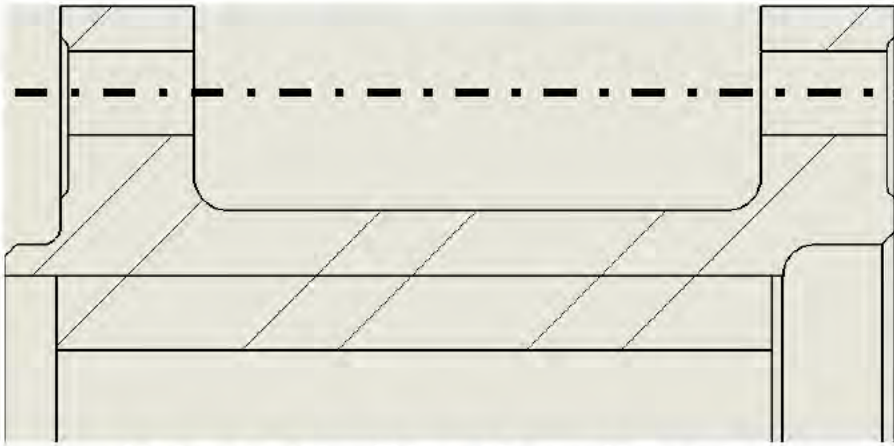
- **Not interfere with existing seal and thrust flows**



5 equally spaced .125 inch high cavity vents provide flow path for thrust and bearing air the exist much like current design.

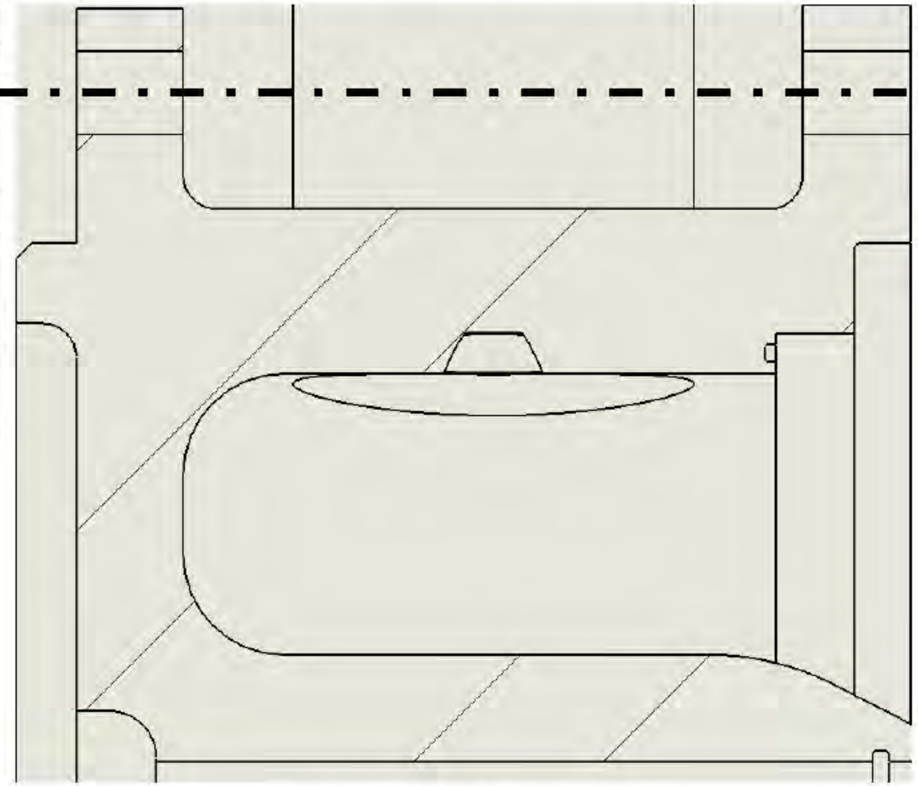
Requirements

- **Maintain identical or greater stiffness relative to existing components**



Existing Tip-Ring Cross-Section

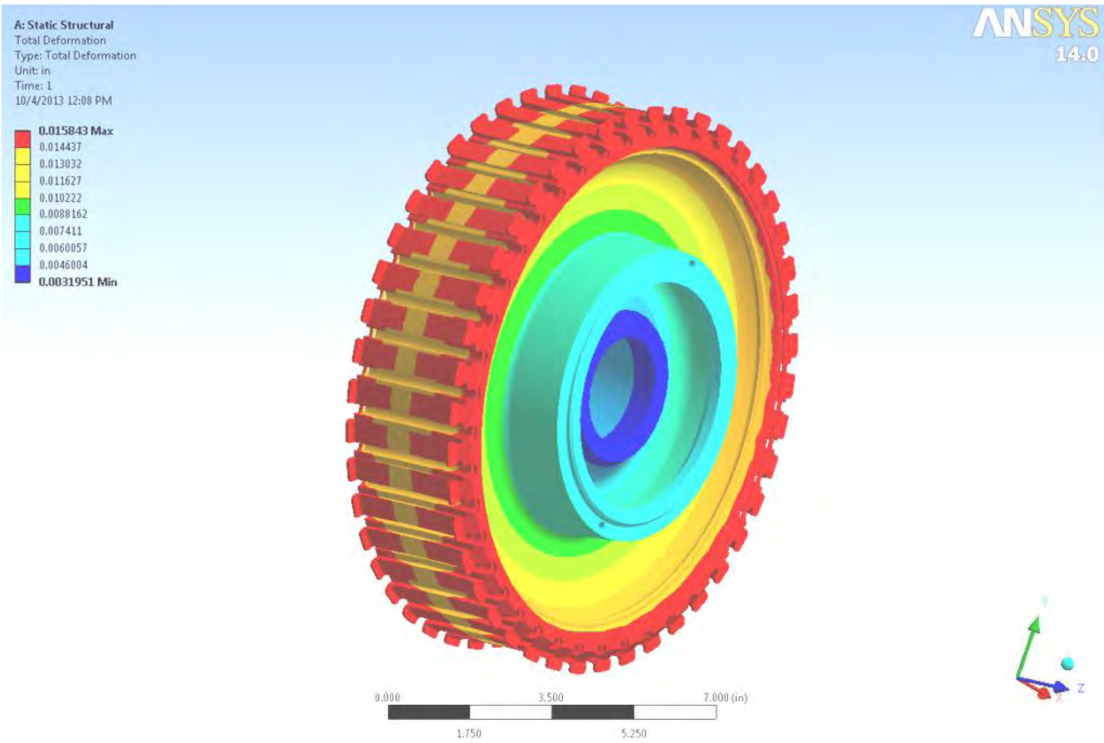
- **Structure Comparison**
  - Identical scale
  - Engages the same pilot fits
  - Only significant structural replacement
  - Stiffness maintained



Design Concept

Requirements

- **Accommodate thermal and axial growth of rotor section.**



Brief FEA analysis of rotor at design speed (24,000 rpm) and at 325 F could grow as much as .015

This narrows the gap between the internal inlet surface under full operation conditions to .1 inch.

ENDIX 11.2.2

Turboexpander FDR

Turboexpander Module Checkout Final Design Review: Solar Turbine Section Rework

Kyle Badeau

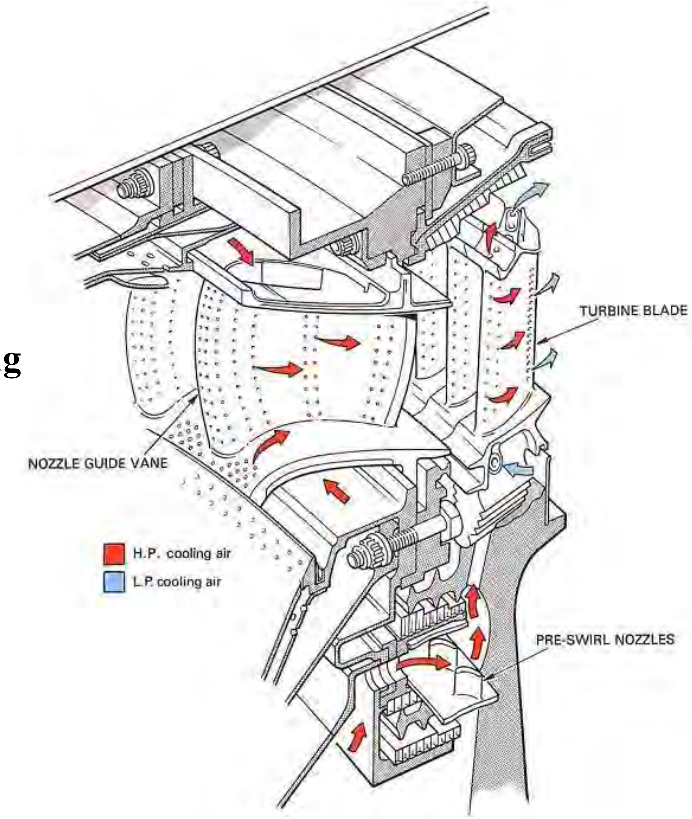
October 24, 2013

Updated 10/29/13, adding bolting calculation

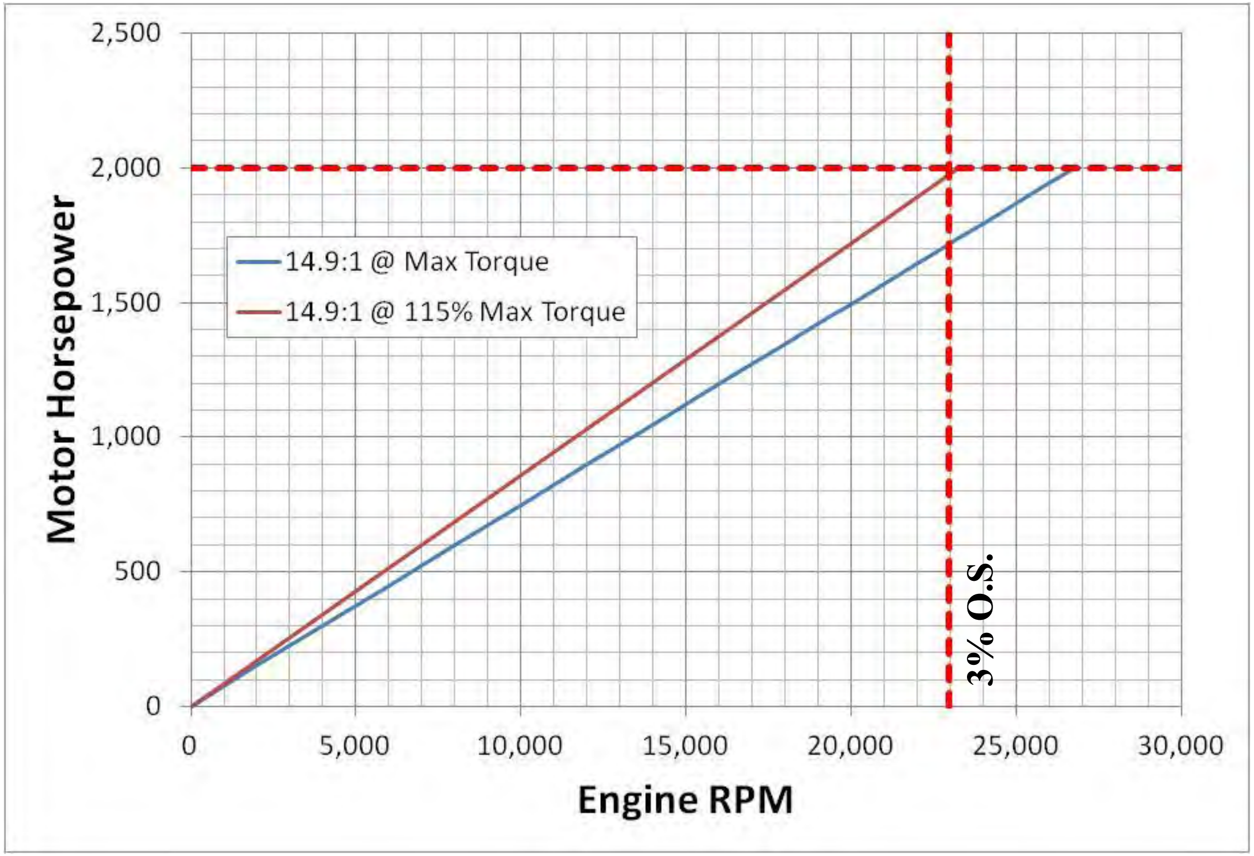
Updated 11/5/12, adding Ramgen Solar Saturn measurements

Introduction

- **The original Solar Saturn engine turbine section was designed for the following output (estimates from thermodynamic cycle reconstruction based on provided Solar data sheets):**
  1. **1382 kW for compressor drive (first two stages)**
  2. **1014 kW for power production (third stage)**
- **The turboexpander module checkout test will not have a compressor, requiring the full turbine output to be dissipated by the motor and VFD**
- **Goal: Enable the turboexpander checkout test at Redmond lab with minimal impact to existing Solar turbine, rotor train, and power dissipation system**
- **Objective: Operate the Solar turbine, limiting power produced according to the drive train power curve by either:**
  1. **Change of operating conditions (reduced speed, mass flow, firing temperature, pressure ratio, or combination)**
  2. **Reconfigure of engine components (deblading)**



Redmond Lab Drive Train Power Limits



- **14.9:1 gearbox (50Hz)**
- **60Hz, 1791 RPM motor rated for 2000 hp (5683 ft-lb torque limit)**
- **ISCE B1 was operated up to 15% over max torque**
- **Target blue power curve limit, up to 3% overspeed limit**

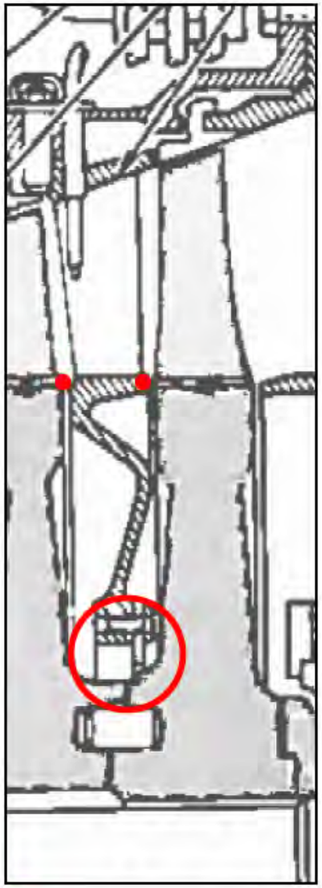
Turbine Operation Requirements

- **Operation of the Solar Saturn T1200 engine for the turboexpander module checkout test should not exceed the following conditions**

1. **1450F average combustor firing temperature limit**
    - **Ensures the material strength of turbine components is not reduced**
    - **Maintains steady state rotating to static component thermal driven clearances**
    - **Checkout test firing temperature targeted to be 1110F**
2. **Exhaust temperatures equal to or less than 860F**
    - **Lower efficiency operation and/or deblading can lead to increased exhaust temperatures**
    - **Solar component operating temperature design margin unknown**
    - **Exhaust diffuser bears structural loading, made of 17-4 PH steel…17-4 PH strength significantly reduces beyond ~900F**
    - **Checkout test exhaust temperature targeted to be 800F**
3. **22,300 RPM maximum steady state speed**
    - **Ensures average and peak stresses of components do not increase**
    - **Maintains blade to shroud centrifugal driven clearances**
    - **Maintains margin to ISCE B1 unvalidated rotordynamic resonance predicted above MCOS**
    - **Operation at speeds below 22,300 RPM will need acceptable rotordynamic margin**
    - **Speed margin to turbine blade resonance crossings is unknown. If blade profiles can be obtained, Campbell diagram predictions can potentially be made to mitigate blade HCF failure. Turbine startup cycle map shows speed hold at 80% speed, indicating potential blade frequency margin.**

Turbine Reconfiguration Requirements

1. **Maintain inner and outer flowpath walls**
   - **Enables smooth flow through annulus, eliminates risk of windage heating (such as in the disk dovetails)**
2. **Maintain wheel space sealing and purge flow**
   - **Maintains purge flow effectiveness**
   - **Rotor to stator axial seals located at blade platform**
   - **Additional radial labyrinth seals located at disk inner diameter to prevent stage to stage leakage**
   - **Purge flow controlled at disk hub hole passages**
3. **Maintain turbine rotor balance**
4. **Provide rotordynamics assessment input (mass/stiffness/size) of final rotor configuration**
5. **Maintain axial flow leaving last turbine stage**
   - **If turbine blade is removed, remove upstream stator blade to prevent swirl going into exhaust diffuser)**



● **Seal Locations**

Turbine Section Reconfiguration

Concept:

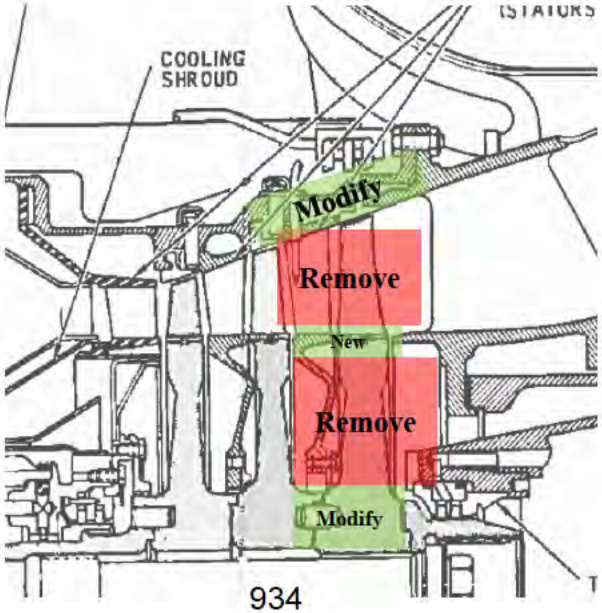
1. Remove vanes and inner structure from 3rd stage stator by either:
    - Modifying existing components
    - Replacing with GLM inventoried components, preserving existing
2. Deblade 3rd stage disk and machine to reduced diameter (by similar means as stator modifications)
    - Rebalance turbine section rotor assembly (three disks, turbine shaft, turbine tie bolt, end seal)
3. Install static inner hub flowguide, fastening to exhaust diffuser with existing bolting, extending to 2nd stage disk

Pros:

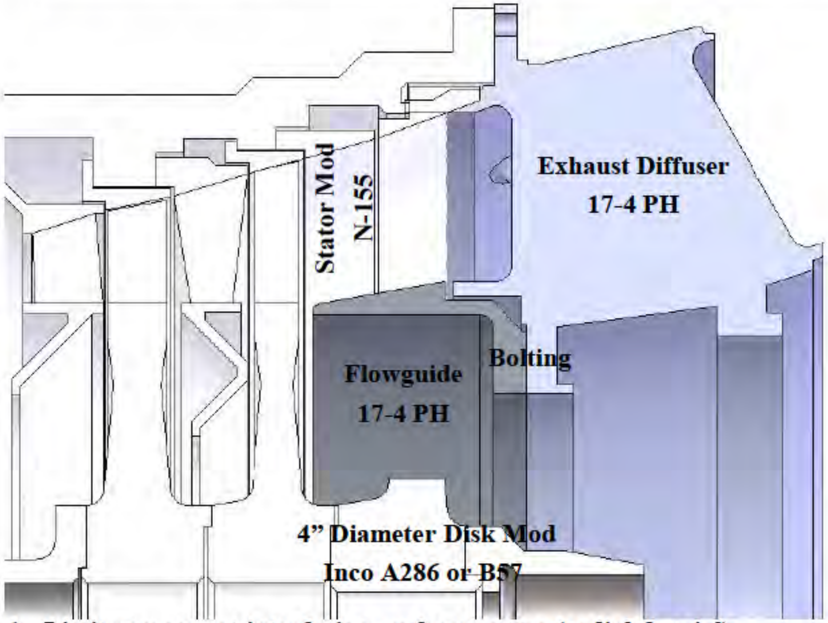
- Potential for disassembly/rebuild in Redmond
- 3rd stage static and rotating vanes removed from flowpath
- Inner hub flowpath formed by static component
- Existing bolt configuration on diffuser can be used for fastening flowguide with longer bolts
- No new rotating components and associated forged material lead times
- Hardware exchange potential with GLM for cost reduction
- Possibility of no original hardware destruction

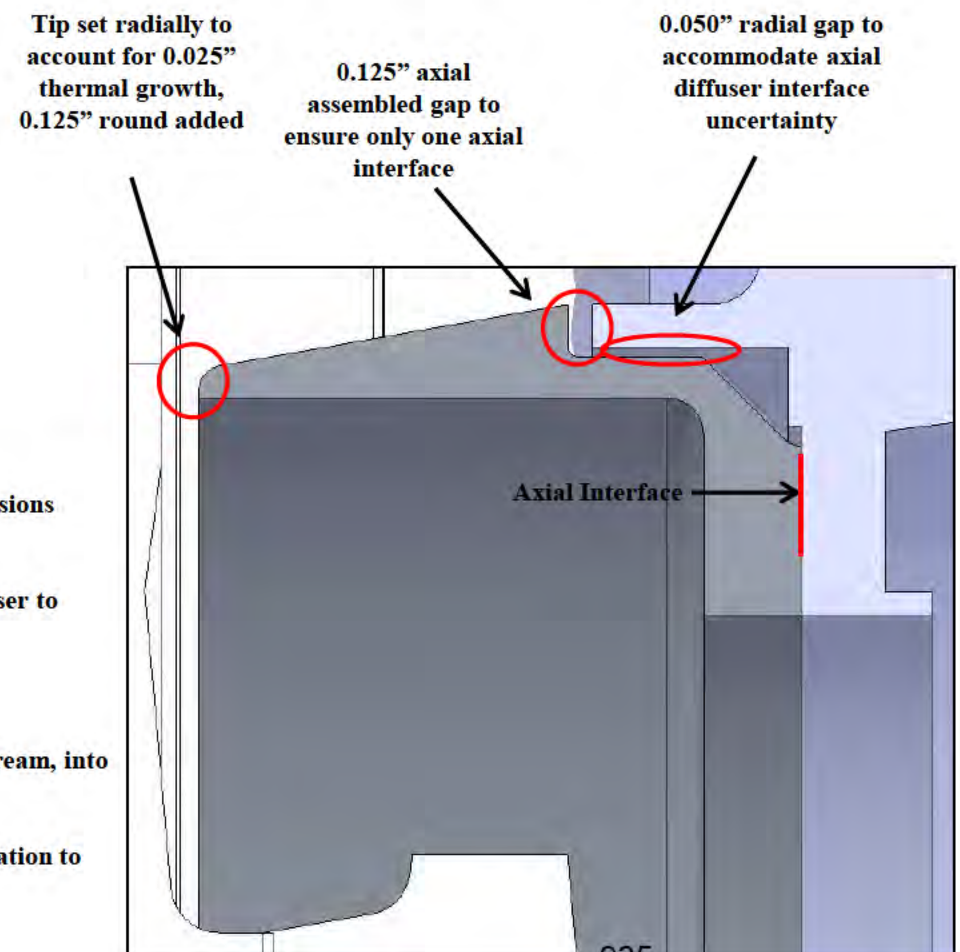
Cons:

- Rotor will need to be disassembled for rebalance, turbine disassembly up to stage 1 nozzle
- Flowguide interface with exhaust diffuser are not controlled surfaces



Turbine Section Layout

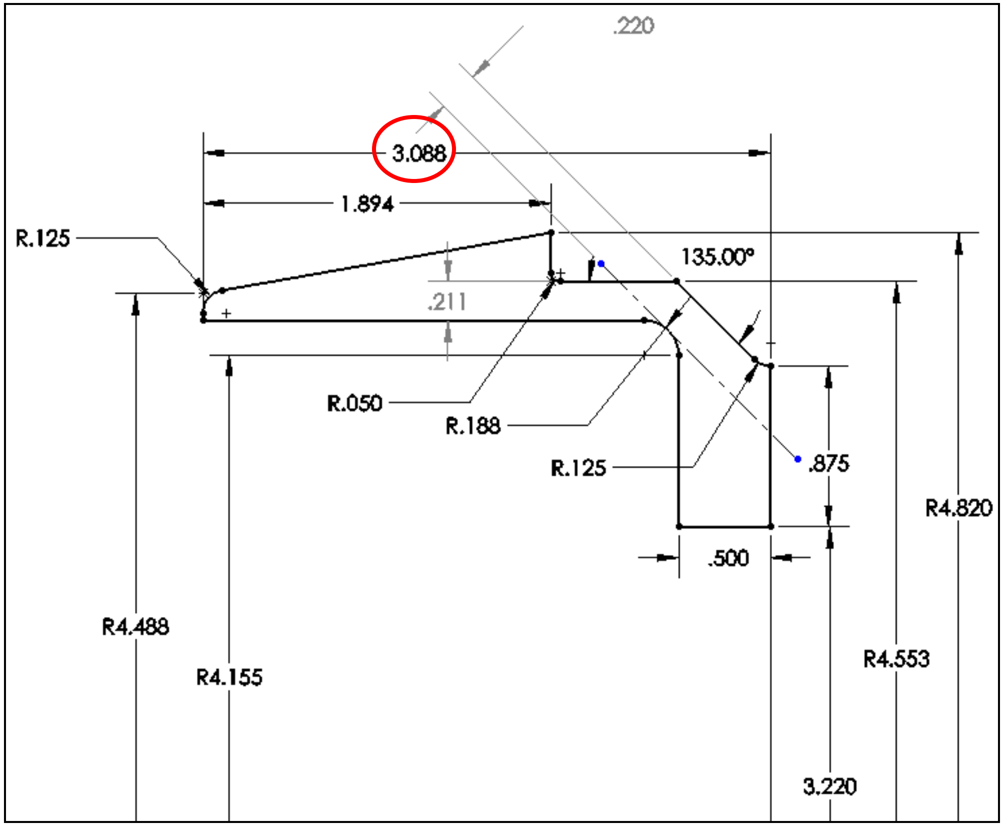




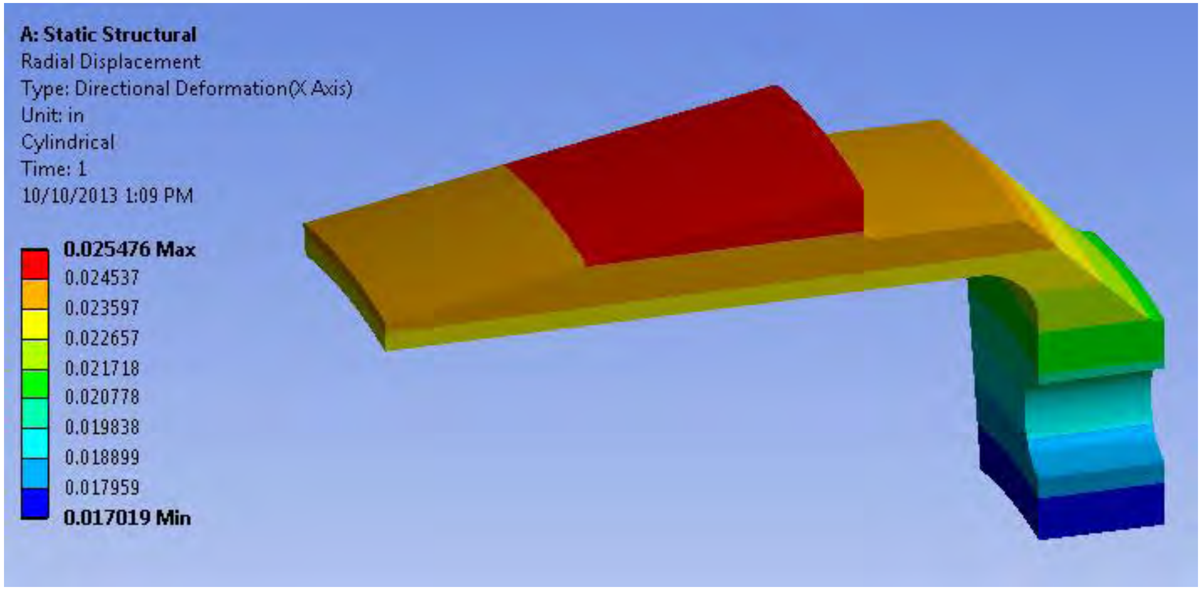
1. Limit component interfacing surfaces to two (radial & axial)
2. Underside faces of Exhaust Diffuser inner ring are uncontrolled, rough dimensions provided by GLM, reflected in layout
3. Diffuser is welded to exhaust collector (downstream) making removal of diffuser to machine mating faces difficult
4. Ability to shim radial interface at 0.050" gap
4. Maintains current purge flow configuration (stage 3 disk upstream & downstream, into wheelspace, into flowpath)
5. Axial gap to stage 2 disk TBD, need measurements of current engine configuration to finalize, max material condition to be specified in initial drawing release
6. Preliminary drawings out to GLM for disk & stator mod quote

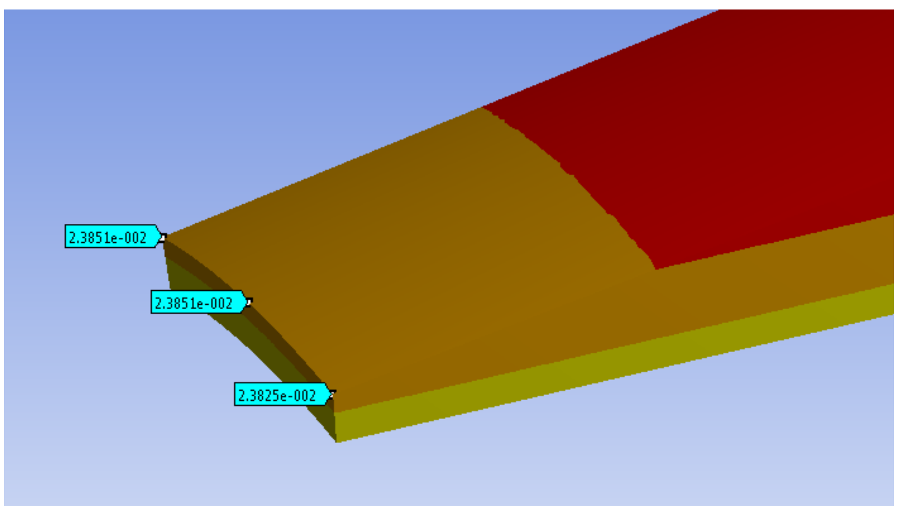
Flowguide Geometry

1. **Overall axial length (3.088) to be finalized for gap to 2nd stage disk, (existing engine measurements needed)**
   - **To be set to match existing sum of T2-N3, N3-T3, T3-ED gaps for purge flow purposes**
2. **Hub radius at interface with 2nd stage disk set to accommodate estimated growths**
   - **Stage 2 disk radius ~ 4.513"**
   - **Stage 2 disk only centrifugal growth ~ 0.003" (neglected)**
   - **Flowguide tip max thermal growth at uniform 860F ~ 0.025"**
   - **Radius set to (4.513" – 0.025") 4.488"**
3. **Bolting**
   - **12 bolts equal spaced on 7 3/8" bolt hole circle**
   - **3/8" UNF hex head cap screw, 1" length (existing), drilled for tie wire**
   - **0.5" length adder for flowguide attachment**
   - **Bolt material: ASTM A193 B6**
   - **0.406" through hole in flowguide and exhaust diffuser**
4. **Flowguide material: 17-4 PH 1025 (same as diffuser)**
5. **UMDI quote for $5.6K, 7 weeks delivered (waiting on two more quotes)**



Flowguide Max Thermal Growth

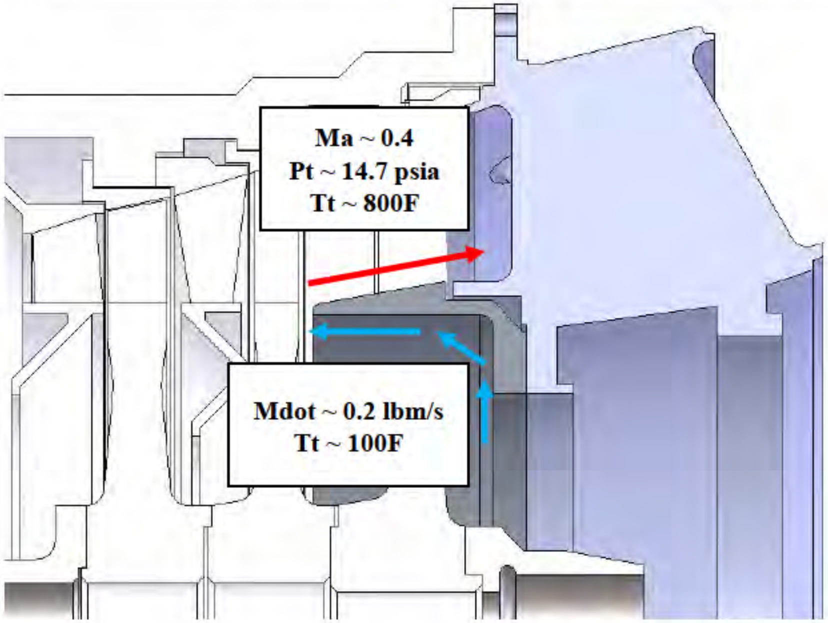




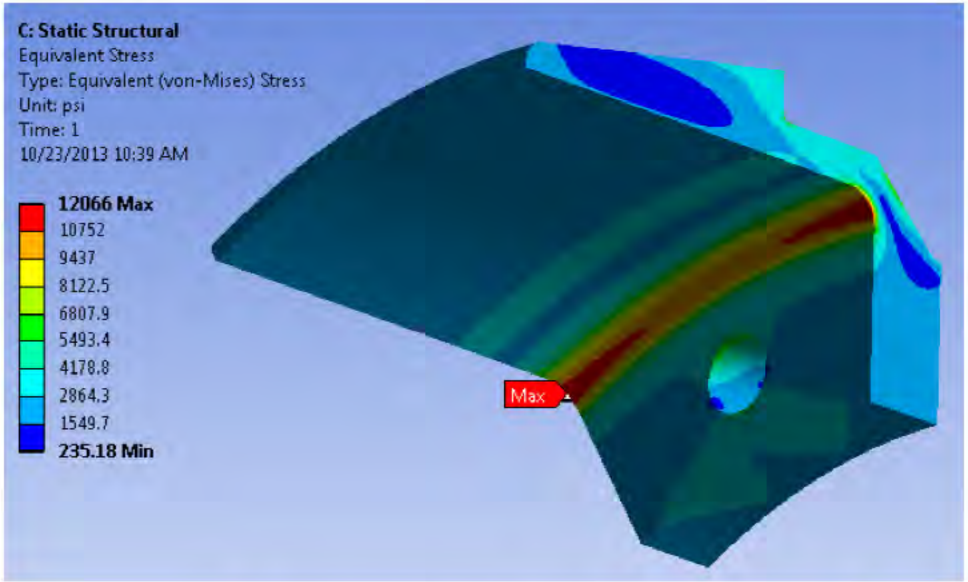
- **Uniform T=860°F**
- **Alpha=6.703E-6 F^-1**
- **25 mil max radial thermal growth at outer radius**
- **24 mil radial growth at tip**
- **Stage 2 disk only centrifugal growth ~3 mils (negligible)**

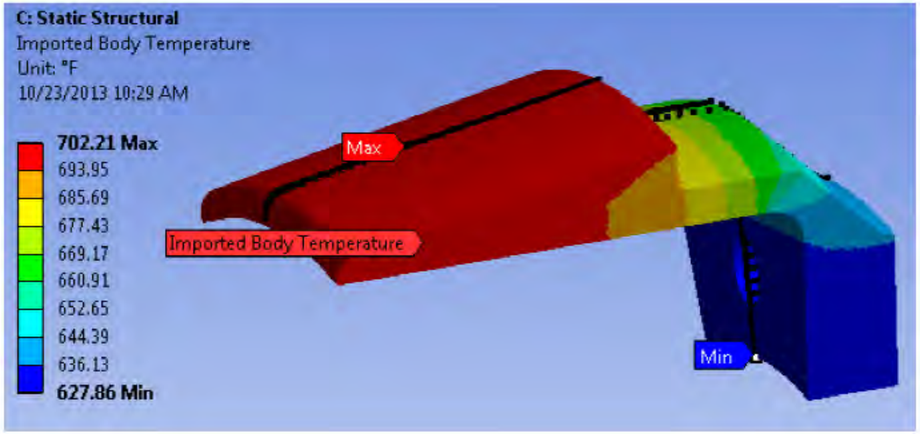
Ensure tip thermal growth is less than disk growth by adjusting cold radius

Flowguide Steady State Thermal Analysis



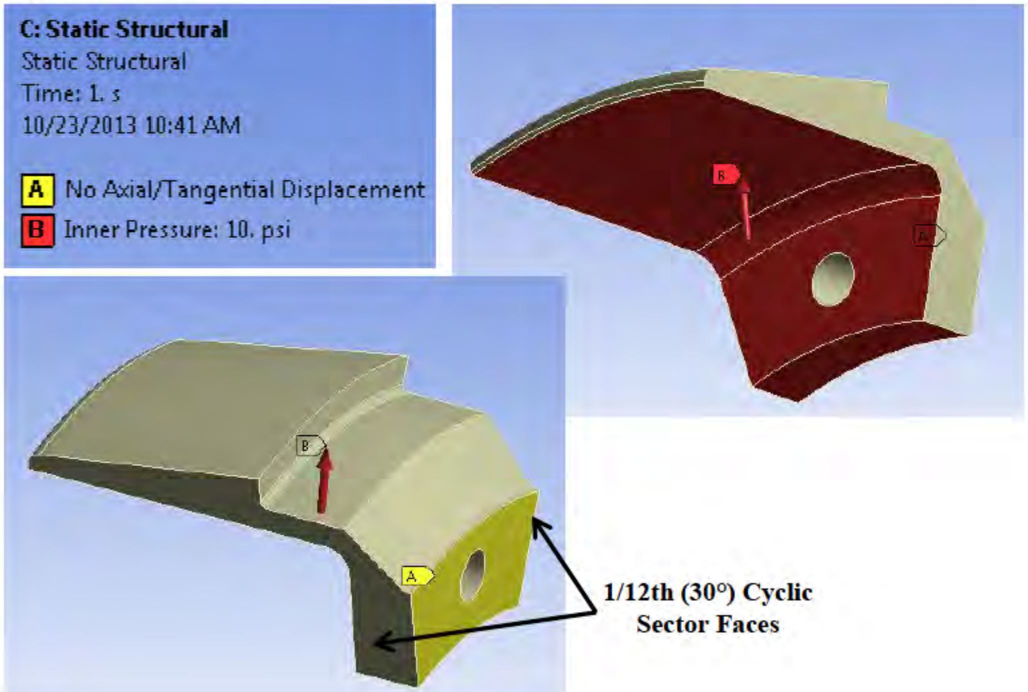
- Convection used on top surface w/ film coefficient of 4.3831e-004 BTU/s·in²·°F
- Fluid pump elements used in wheel space





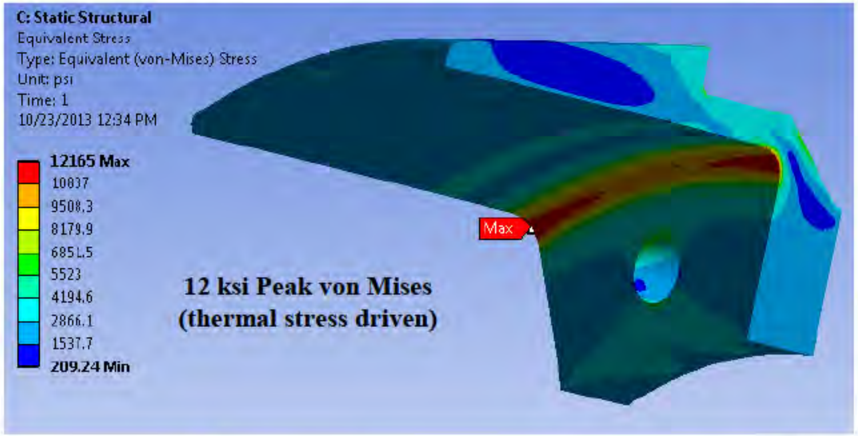
12 ksi max Steady State Thermal Stress

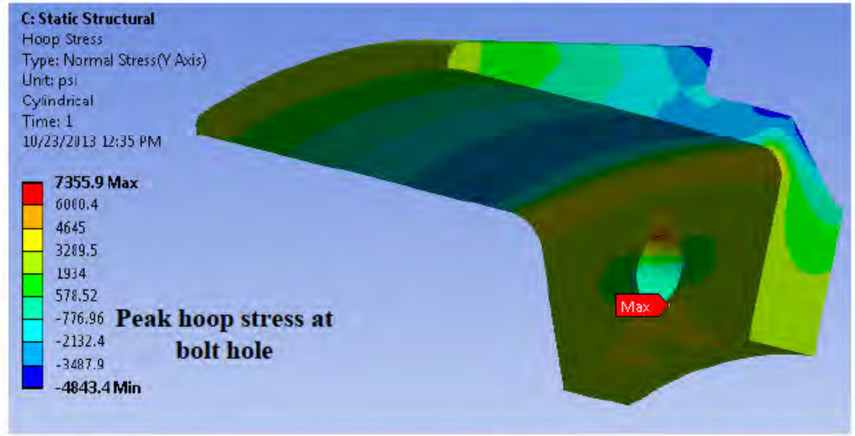
Flowguide Static Analysis – BC's

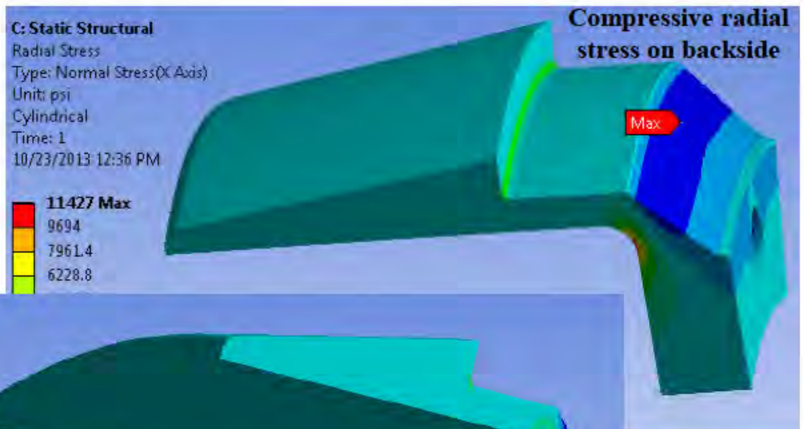


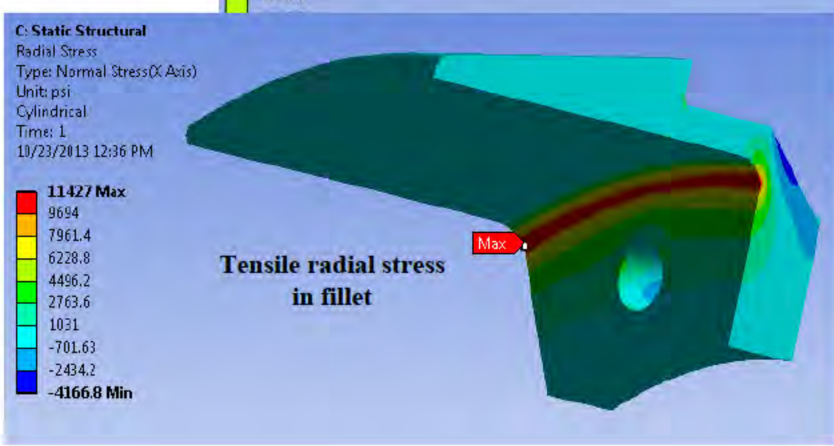
- **Thermal stress coupled with flowpath to wheelspace (radial) pressure delta**
- **Assumption: diffuser grows with flowguide or freely (no thermally induced stress on diffuser, flowguide, or bolting from component growth mismatch)**
    - **Radial gap between diffuser and flowguide**
    - **15 mil radial gap around bolt**

Flowguide Static Analysis – Stress Results









- Room temperature yield strength of 17-4 1025 PH 145 ksi
- 72% reduction at 860°F
- 104 ksi yield strength at temperature

12 ksi radial pressure delta induced stresses well below yield

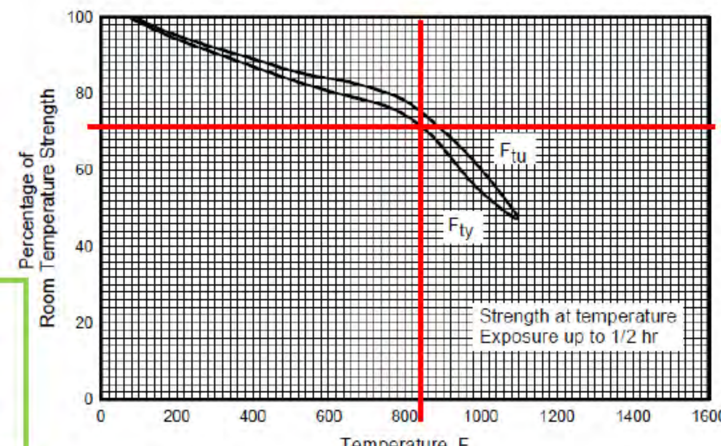
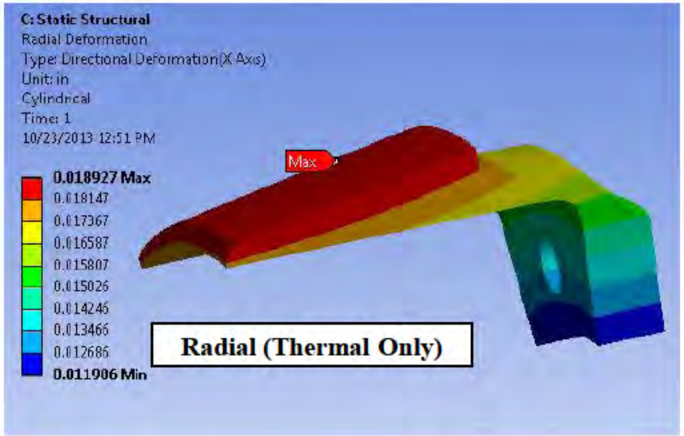
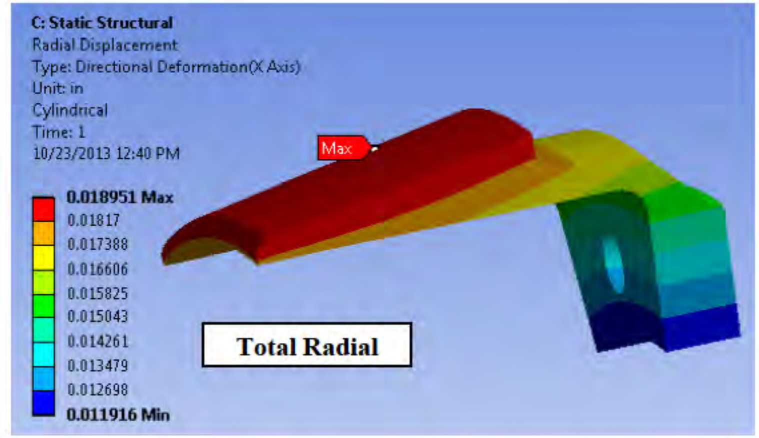
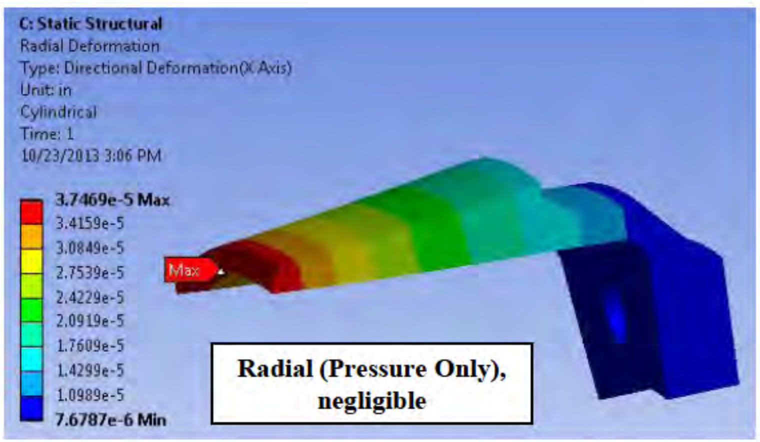


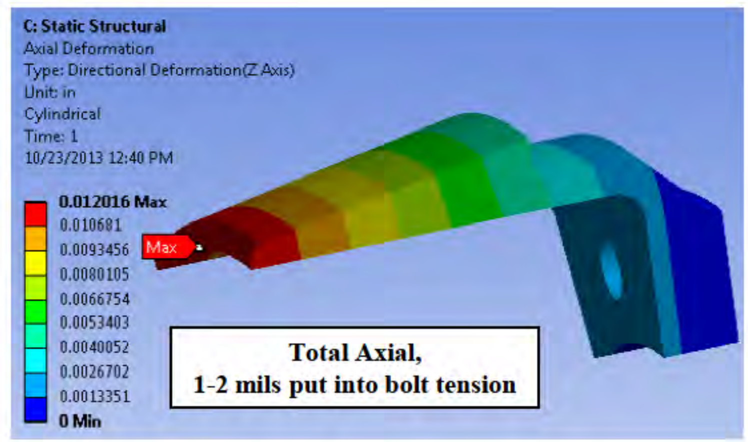
Figure 2.6.9.2.1. Effect of temperature on the tensile ultimate strength (F_{tu}) and the tensile yield strength (F_{ty}) of 17-4PH (H900, H925, H1025, and H1075) stainless steel bar.

Flowguide Static Analysis – Growth









Radial Growth Primarily Thermal Driven

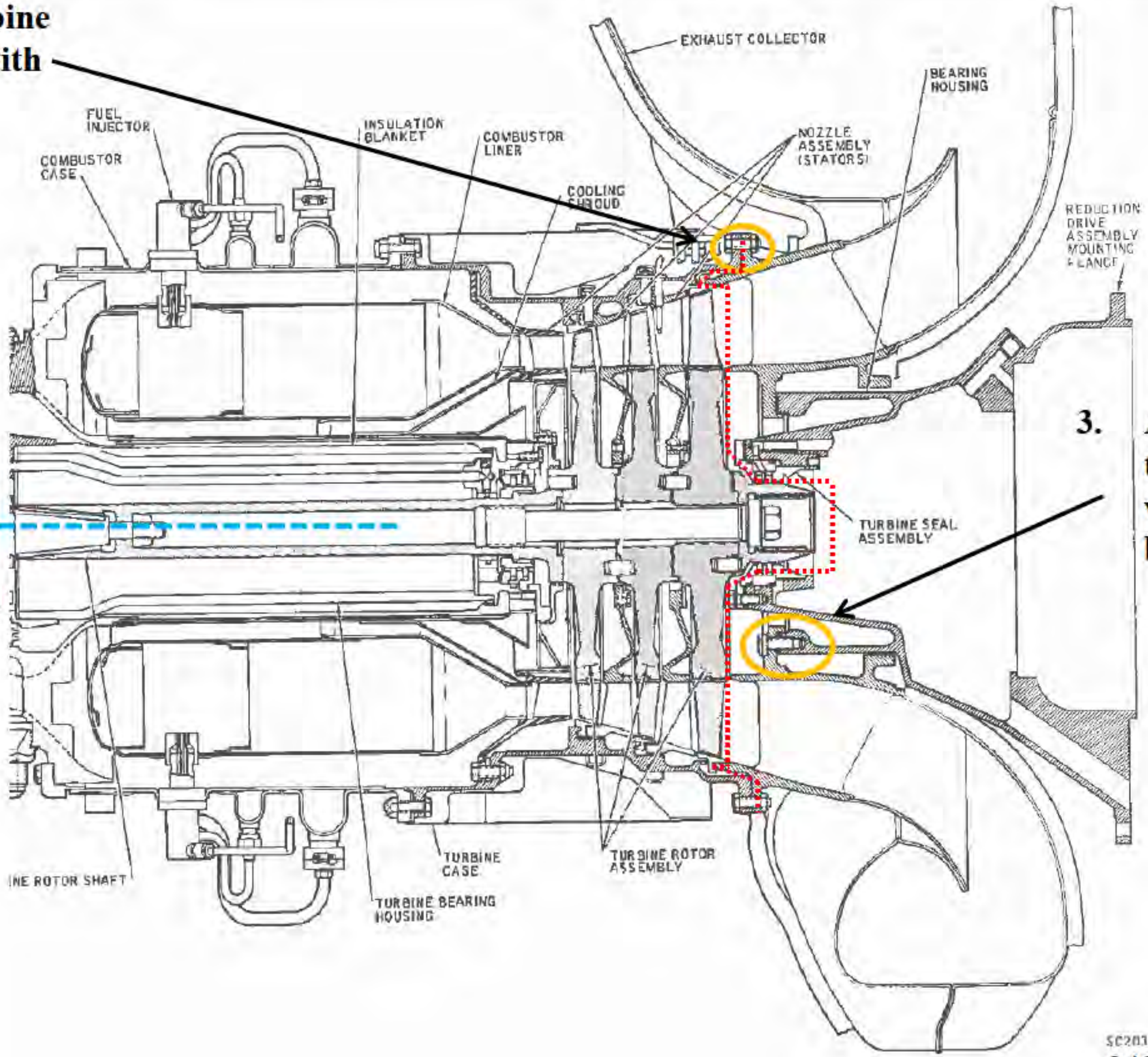
Solar Saturn Disassembly – High Level

Disassemble Solar Saturn engine at turbine end to enable the following…

1. **Shipment of turbine section rotating components (shafts, disks, blades, etc.) to GLM for 3rd stage disk modifications and assembled turbine rotor rebalance**
   - **Mitigates risk associated with 3rd stage disk only rebalance**
2. **Modifications to remaining engine systems in Redmond in parallel**
   - **Turbine inlet for compressor air provision**
   - **Combustor casing hole repair and thermocouple placement**
   - **Rework of exhaust diffuser for new flow guide fit up**

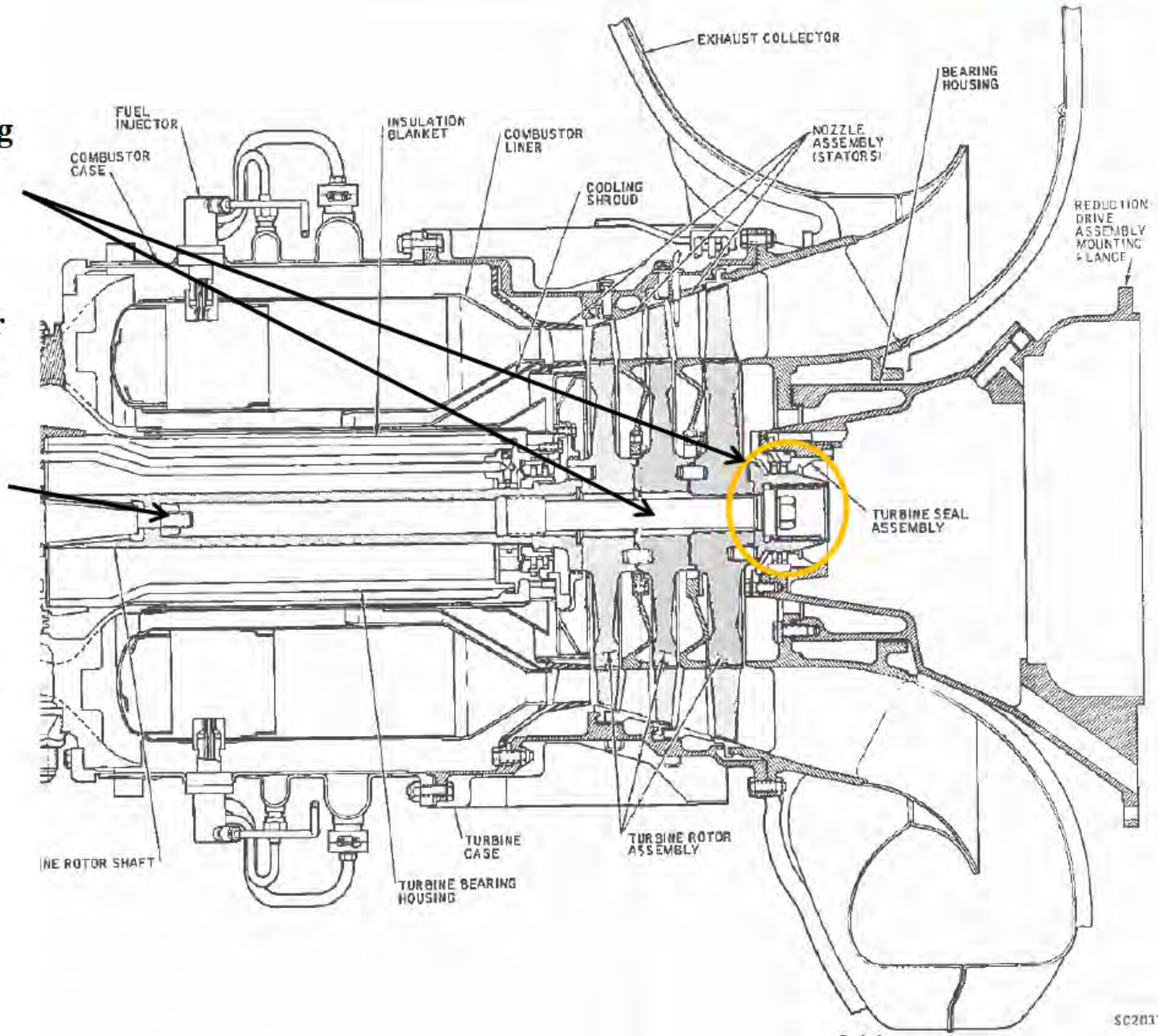
Sequence

1. **Disassemble turbine section at joint with exhaust diffuser**
2. **Move engine towards compressor end**
3. **Attach flow guide to exhaust diffuser with lengthened bolting**



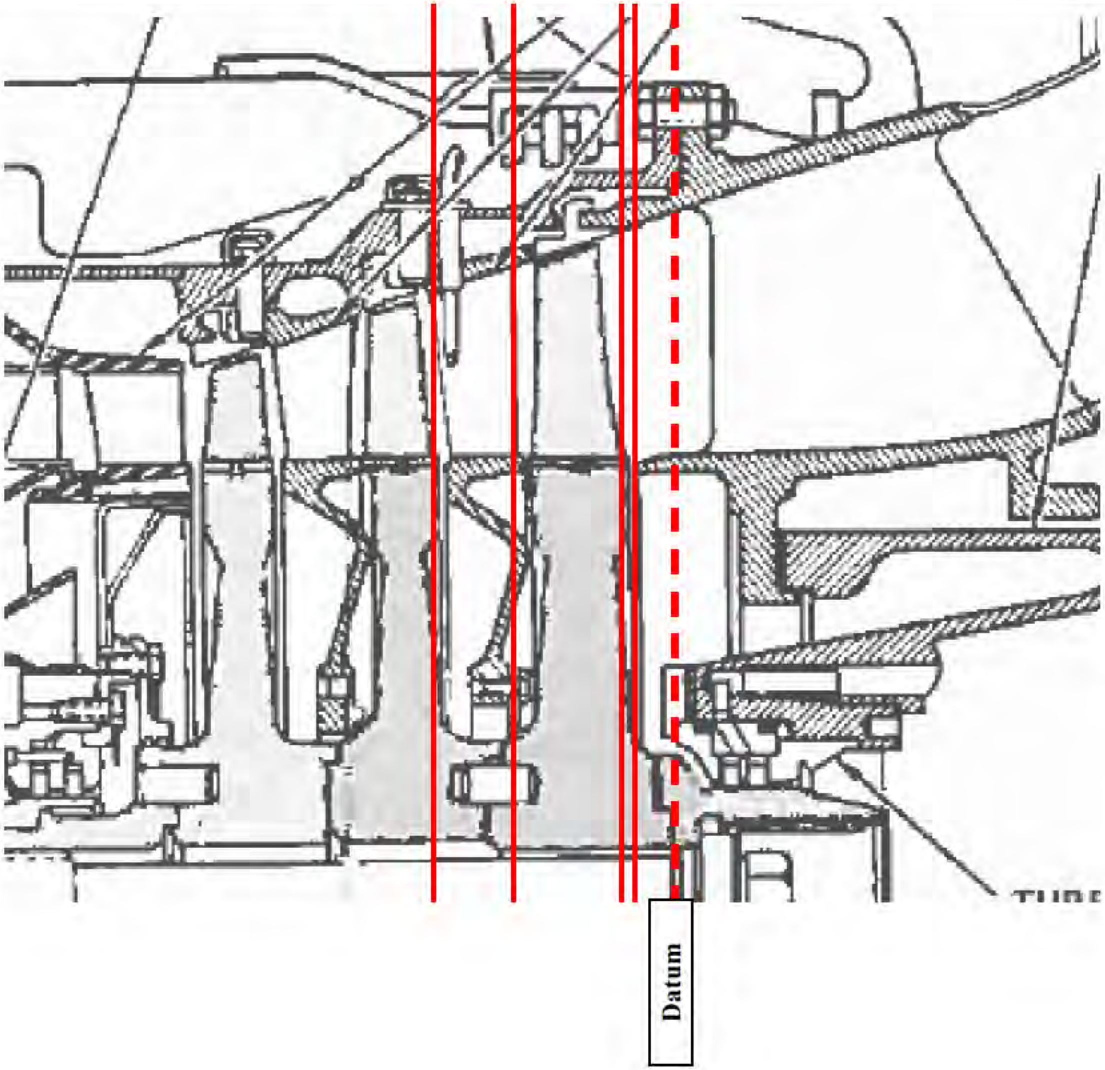
Sequence, cont'd

4. **Stand engine up vertically**
5. **Remove rotor end seal components and tie bolt, exposing stage 3 disk**
6. **Remove stages 3, 2, 1 disks, dowels, nozzles, shrouds, casing components in appropriate order (stage 1 nozzle remains in place), exposing turbine shaft**
7. **Remove turbine shaft at press fit**
8. **Ship rotating components to GLM for stage 3 disk mod and rotor rebalance as system**
9. **Ship stage 3 nozzle to GLM (cut for outer ring reinstall only)**



RAMGEN POWER SYSTEMS

Measurements Needed



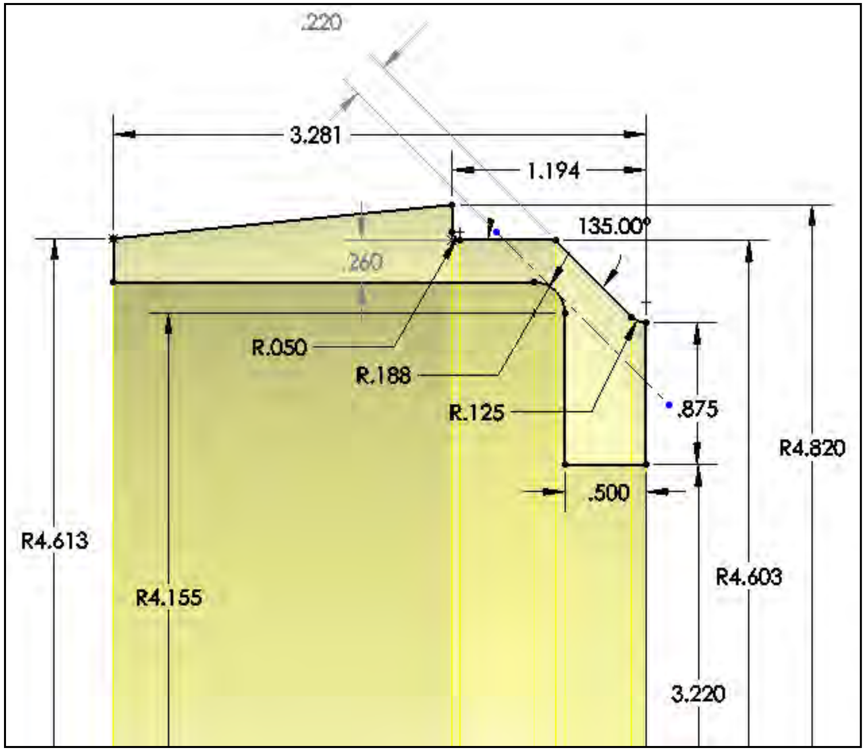
Using flange as a datum, measure axial distances to:

1. **Exhaust diffuser landing (from flange right side)**
2. **Stage 3 disk downstream face**
3. **Stage 3 stator downstream face**
4. **Stage 2 disk downstream face**

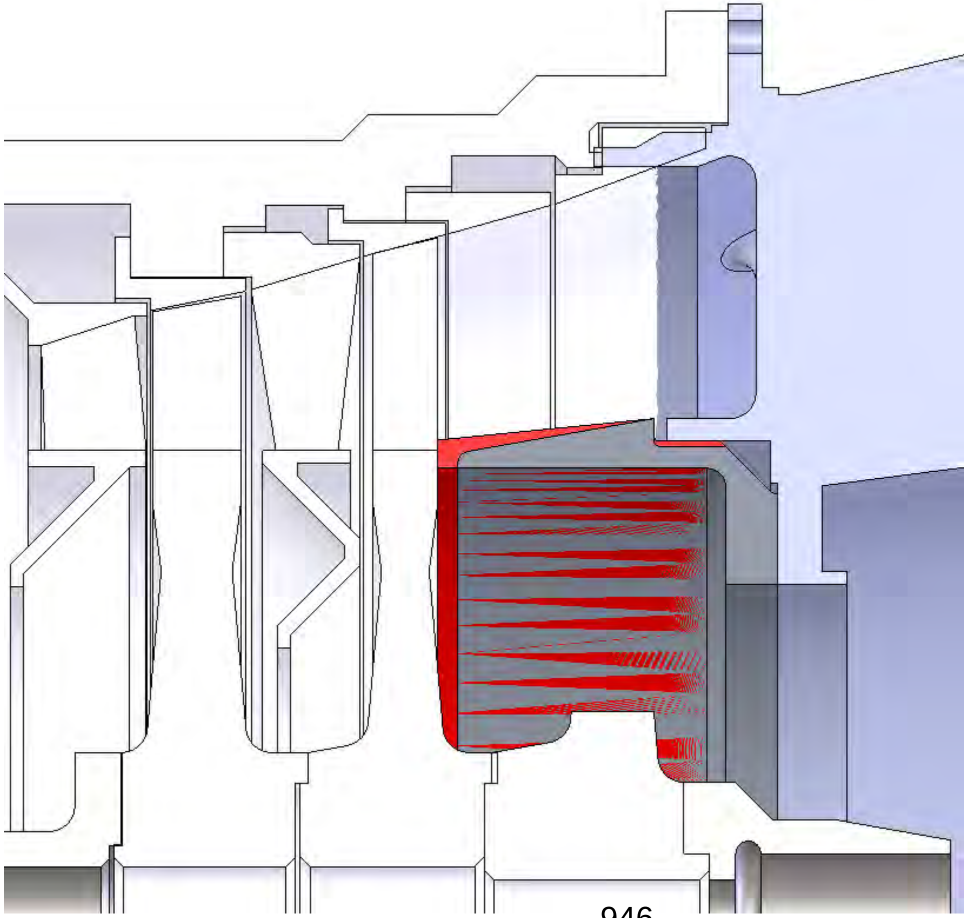
Additional measurements needed:

- **Stage 3 disk width**
- **Stage 3 stator width**
- **Stage 3 disk diameter**
- **Stage 2 disk diameter**
- **Exhaust diffuser interface with flowguide**

Drawing Release – Max Material Condition



- Max material in red
- Extend axially to meet Stage 2 Disk
- Extend radially to meet diffuser
- Add material radially to tip for disk mismatch



Instrumentation System Requirements

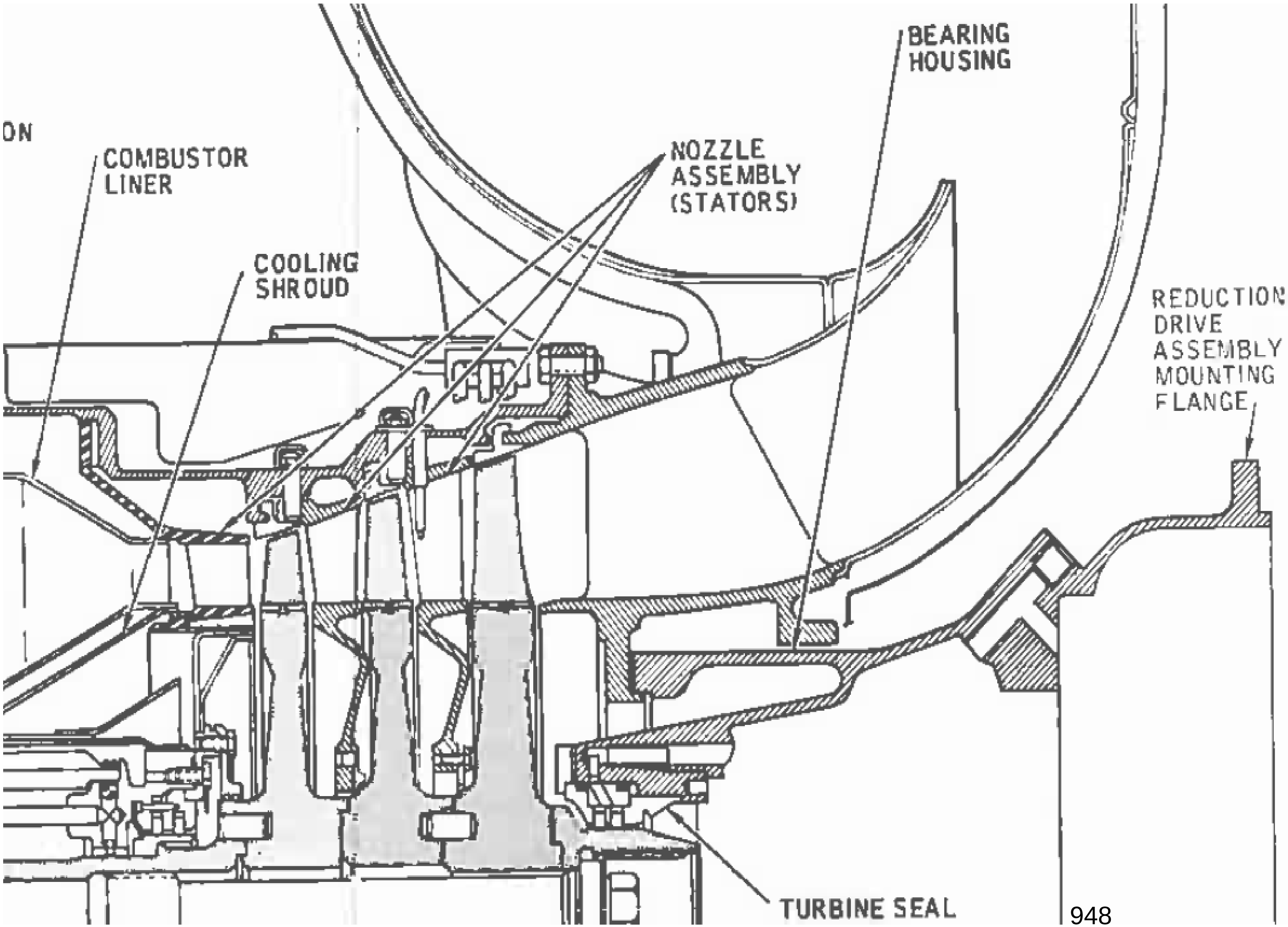
1. **Establish instrumentation hardware and locations to measure flange to flange turbine performance**

   - **Shaft speed and VFD power (torque to be inferred from VFD, accounting for upstream drive train losses)**
   - **Mass flow at inlet (cooler end)**
   - **Combustor firing temperature**
   - **T5 (stage 2 exit, now exhaust location) and T7 (exhaust stack) measurements**
   - **Exhaust pressure measurement at T7 location**
   - **Compressor discharge pressure**

2. **Rig health**

   - **Shaft motion (radial prox probes, axial prox probe, accelerometers)**
   - **Bearings (oil supply pressure, temperature, mass flow; drain pressure, temperature)**
   - **Cooling and buffer flows (supply pressure, temperature, mass flow, internal passage pressure)**

Appendix: Solar Saturn Cross Section



Appendix: Saturn Design Data from GLM

Materials

- **Disks: A286 or B57 Inconel**
- **Blades: IN738 (all three stages)**
- **Shouds: 600 Inconel**
- **Shaft: TBD**
- **Nozzles**
  1. **FSX-414**
  2. **N-155**
  3. **N-155**
- **Case: 420SS (maybe 430)**
- **Exhaust Diffuser: 17-4 PH**

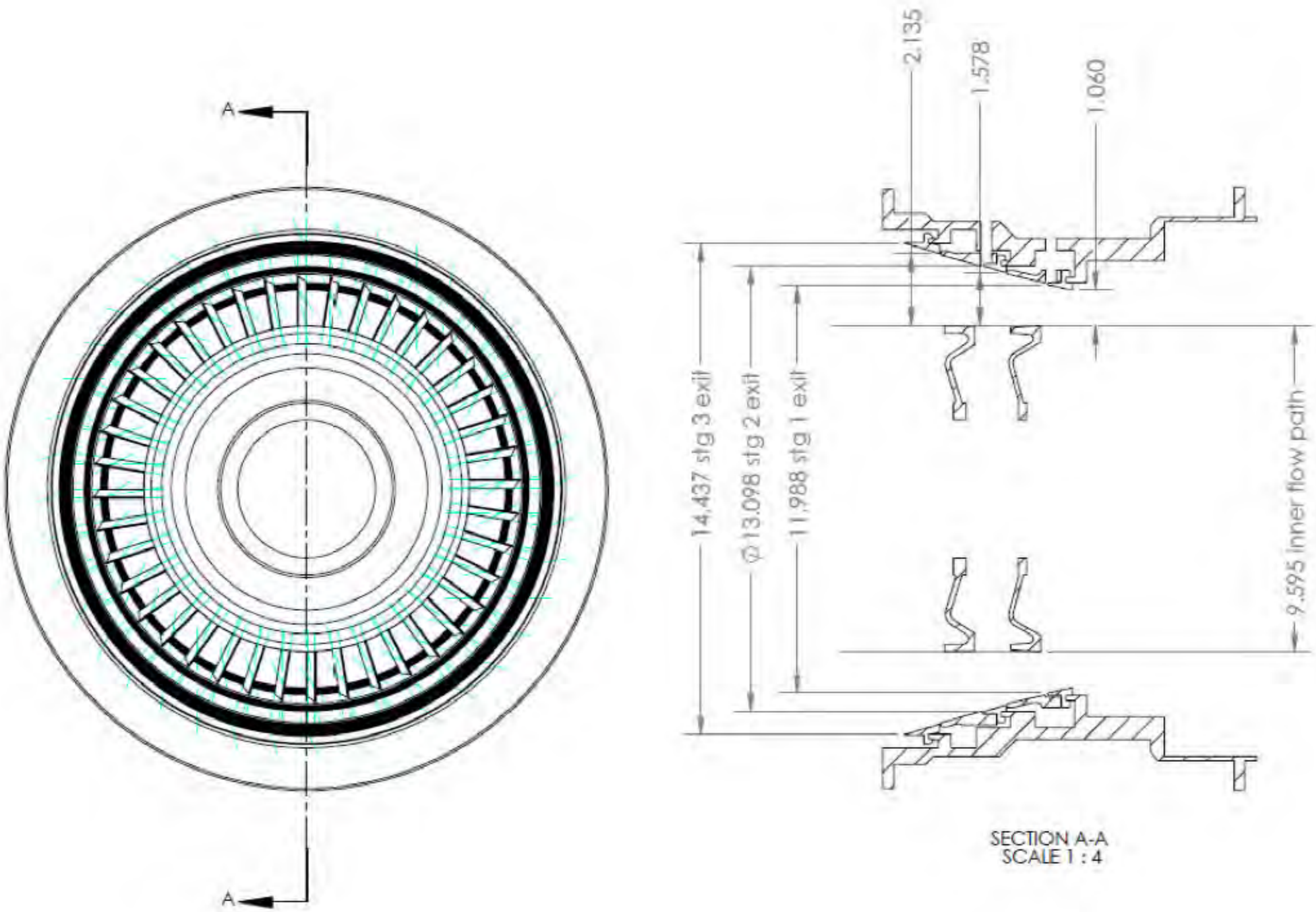
Geometry

- **Disk weight**
  1. **17 lb**
  2. **17.5 lb**
  3. **26 lb**

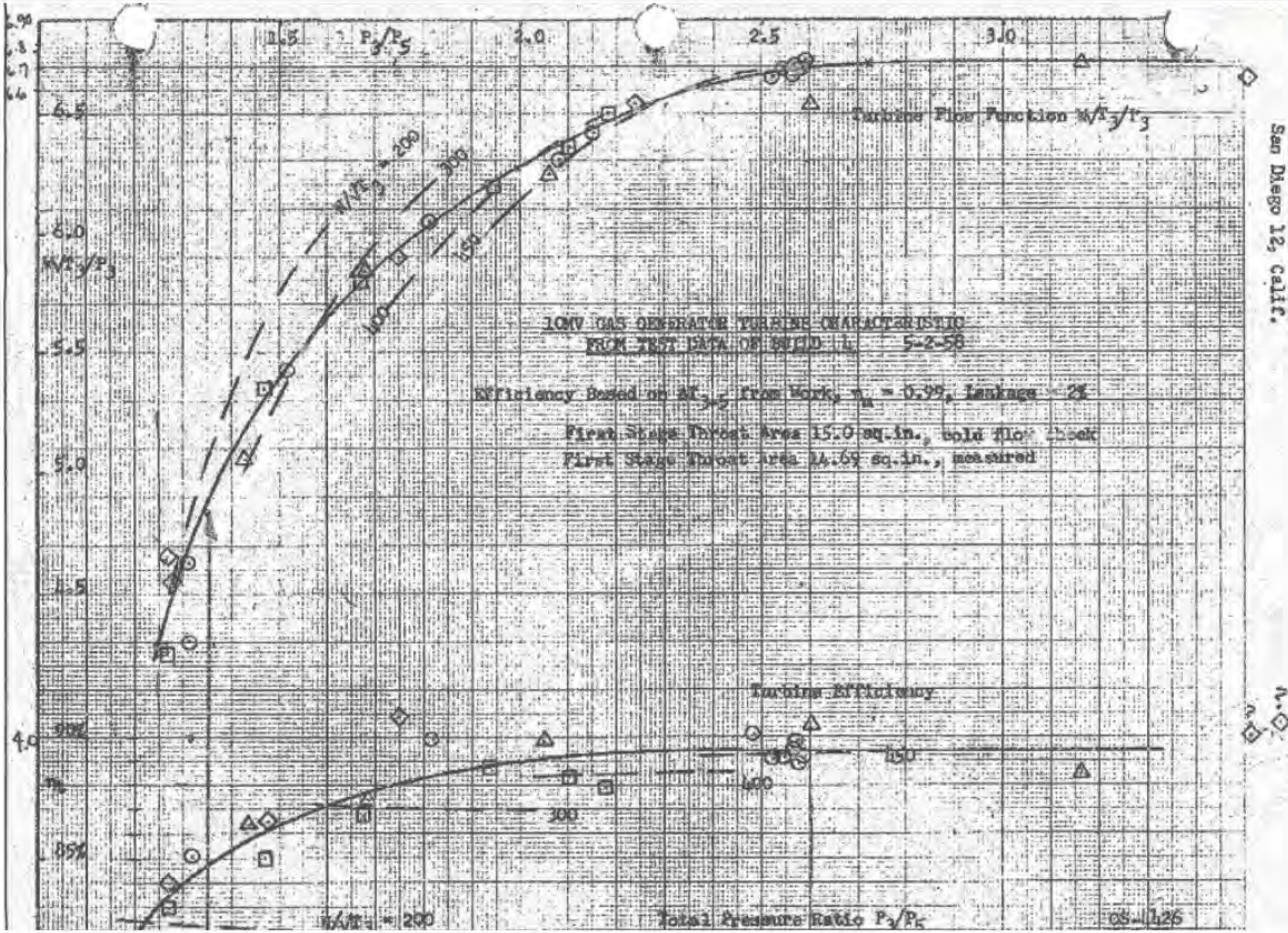
Airfoil Count

- **N1: 27**
- **T1: 58**
- **N2: 41**
- **T2: 58**
- **N3: 43**
- **T3: 52**

Appendix: GLM Flowpath Dimensions



Appendix: Saturn 10 T1400 Data Sheet 1



RAMGEN POWER SYSTEMS™

Appendix: Saturn 10 T1400 Data Sheet 2

Engine	Units	SATURN 10 T1263
Match	—	90 °F
MPDC	—	TSB-2 REV. 0.0
Output Power After Gearbox	Hp	1141
Output Torque After Gearbox	Ft-Lbf	269
Specific Fuel Consumption	BTU/Hp-Hr	11040
Cycle Efficiency	%	23.05%
Compressor Efficiency	%	83.88%
Combustor Efficiency	%	97.63%
Gas Producer Turbine Efficiency	%	89.81%
Power Turbine Efficiency	%	76.72%
Overall Turbine Efficiency	%	85.24%
Gearbox Efficiency	%	100.00%
Output Shaft Mechanical Efficiency	%	108.11%
Power Turbine Speed	Rpm	22300
Optimum Power Turbine Speed	Rpm	22300
100% Power Turbine Speed	Rpm	22300
Corrected Optimum Power Turbine Speed	%	103.00
Mechanical Limit Power Turbine Speed	%	100.00
Gas Producer Turbine Speed	Rpm	22300
100% Gas Producer Turbine Speed	Rpm	22300
Mechanical Limit Gas Producer Turbine Speed	%	100.00
Corrected Gas Producer Turbine Speed	%	100.00
Fuel Flow	Millions BTU/Hr	12.60
Inlet Airflow	Lbm/Sec	13.20
Engine Exhaust Flow	Lbm/Sec	13.35
Ambient Temperature	°F	80
Compressor Diffuser Exit Temperature	°F	507
Gas Producer Turbine Inlet Temperature	°F	1450
Power Turbine Inlet Temperature	°F	1075
Compensated Power Turbine Inlet Temperature	°F	1075
Engine Exhaust Temperature	°F	860
Compressor Diffuser Exit Static Pressure	psig	72.8

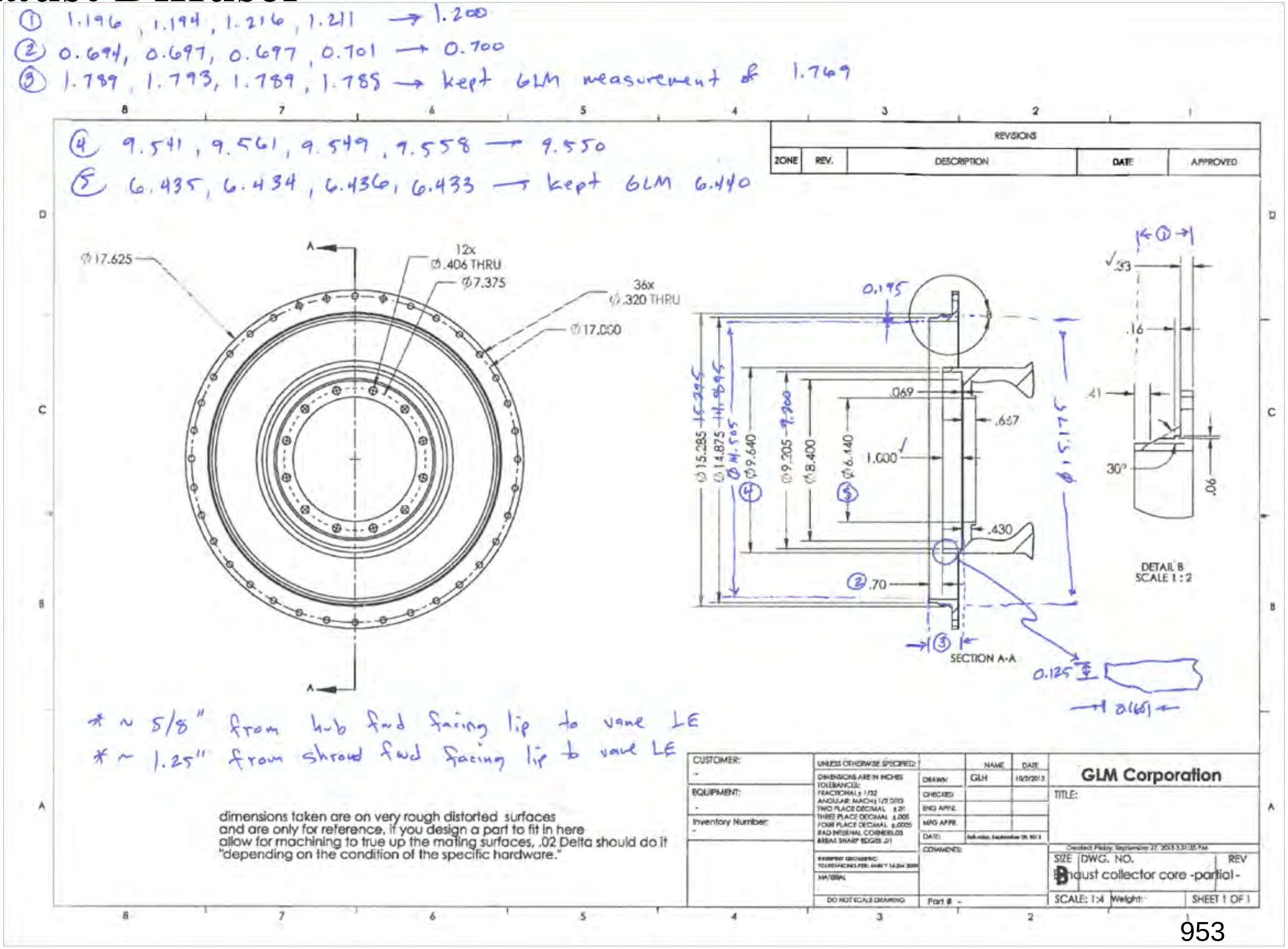
Compressor Diffuser Exit Static Pressure	psig	72.8
Compressor Pressure Ratio, Rc=P2/P1	—	6.10
PCD Factor, P2=PCD/PCD Factor	—	0.976
Combustor Delta P/P	%	5.00%
Compressor Stages	—	8
Compressor Variable Geometry Stages	—	0
Gas Producer Turbine Stages	—	2
Power Turbine Stages	—	1
Customer/Start Bleed Exit Compressor Stage	—	6
Cooling Bleed Compressor Exit Stage	—	8
Compressor Inlet Critical Area	Sq. In.	39.4
Gas Producer Turbine Inlet Critical Area	Sq. In.	13.0
Power Turbine Inlet Critical Area	Sq. In.	31.8
Auxiliary Horsepower Limit	Hp	50.0
Compressor Bleed Limit for Customer Bleed	Lbm/Sec	0.0283
Total Cooling (Turbine Cooling, Sump Flow, Leakage)	% Wc Comp	2.43%
1st Stage Turbine Cooling	% Wc Comp	0.00%
2nd Stage Turbine Cooling	% Wc Comp	1.27%
3rd Stage Turbine Cooling	% Wc Comp	0.98%
4th Stage Turbine Cooling	% Wc Comp	0.00%
Oil Sump Vent Flow + Leakage	% Wc Comp	0.18%
Work Factor for 1st Stage Turbine Rotor Cooling	%	89.00%
Work Factor for 2nd Stage Turbine Rotor Cooling	%	89.00%
Work Factor for 3rd Stage Turbine Rotor Cooling	%	85.60%
Mechanical Efficiency (Bearings, Etc) GP Turbine	%	100.00%
Mechanical Efficiency (Bearings, Etc) Power Turbine	%	100.00%
MPDC	—	TSB-2 REV. 0.0
Match	—	80 °F
Engine	—	SATURN 10 T1200

Inlet Pressure Loss = 0.00%
Exhaust Pressure Loss = 0.00%
Relative Humidity = 60%
Margin on Wi = 0%, exc. T5800 (-1%)
All CS/MD exc. T1601
Nominal Engines, for Minimum Engine Hp Subtract = 3%

Access
Custom
Gearb
Gener
Fuel, SD Nat

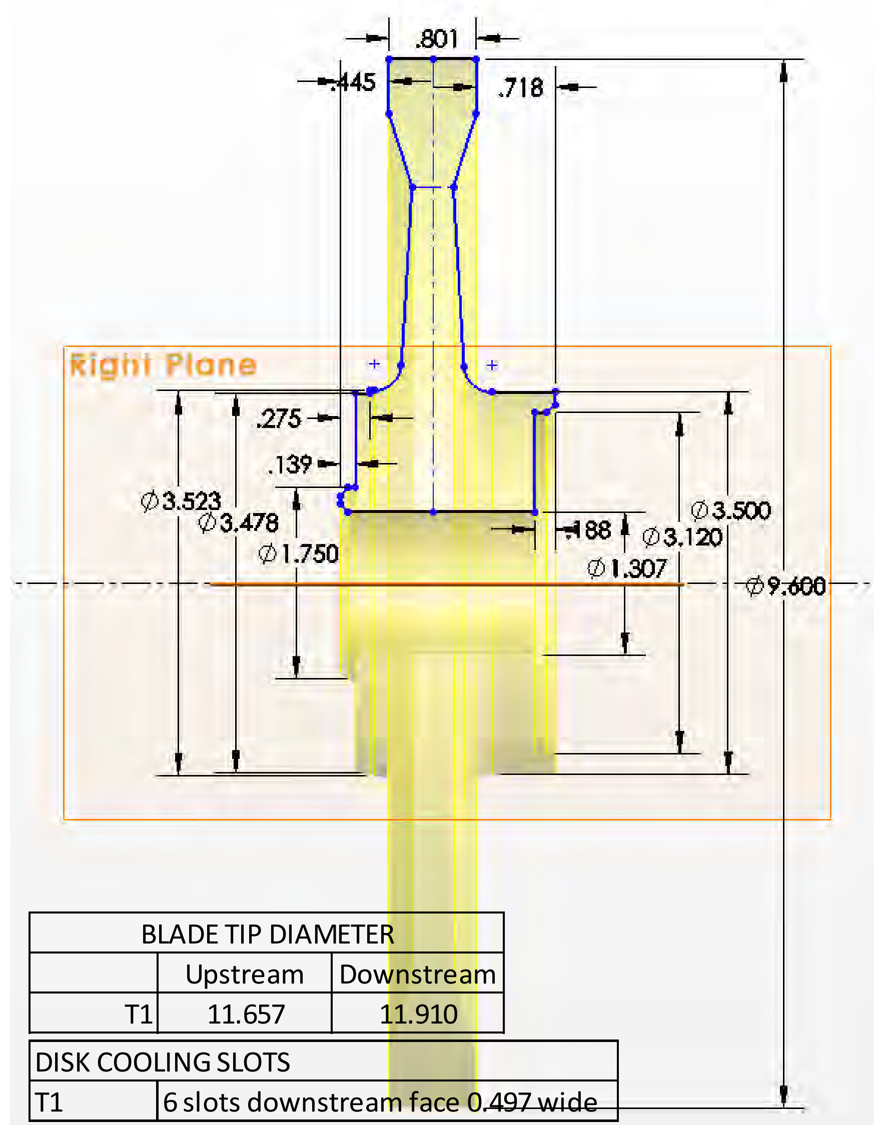
Appendix: Ramgen Solar Saturn Measurements

Exhaust Diffuser



Appendix: Ramgen Solar Saturn Measurements

Stage 1

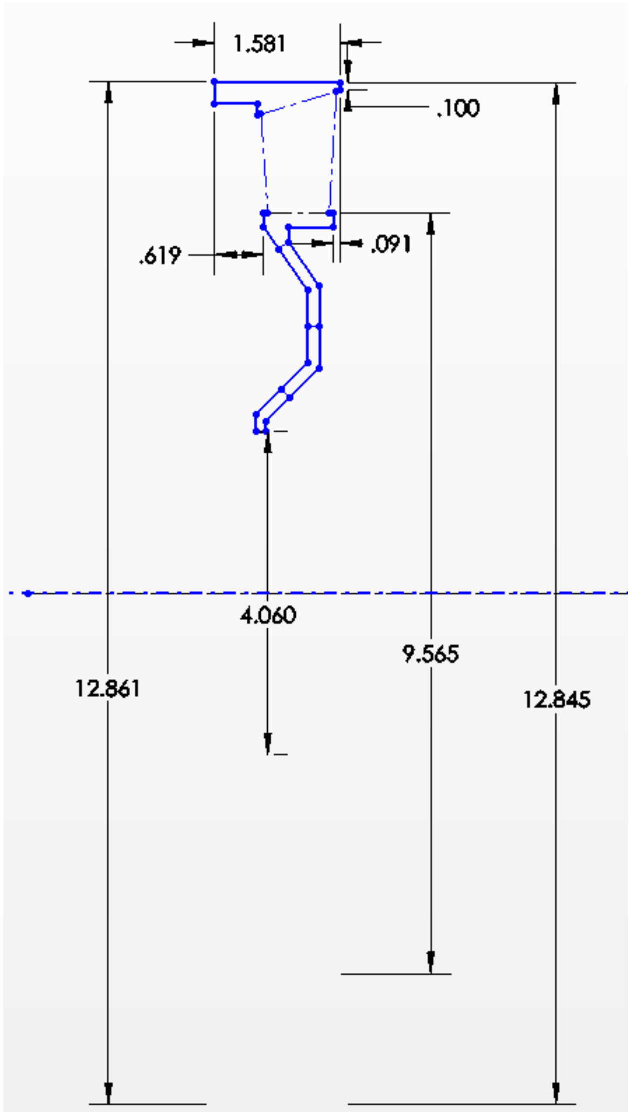


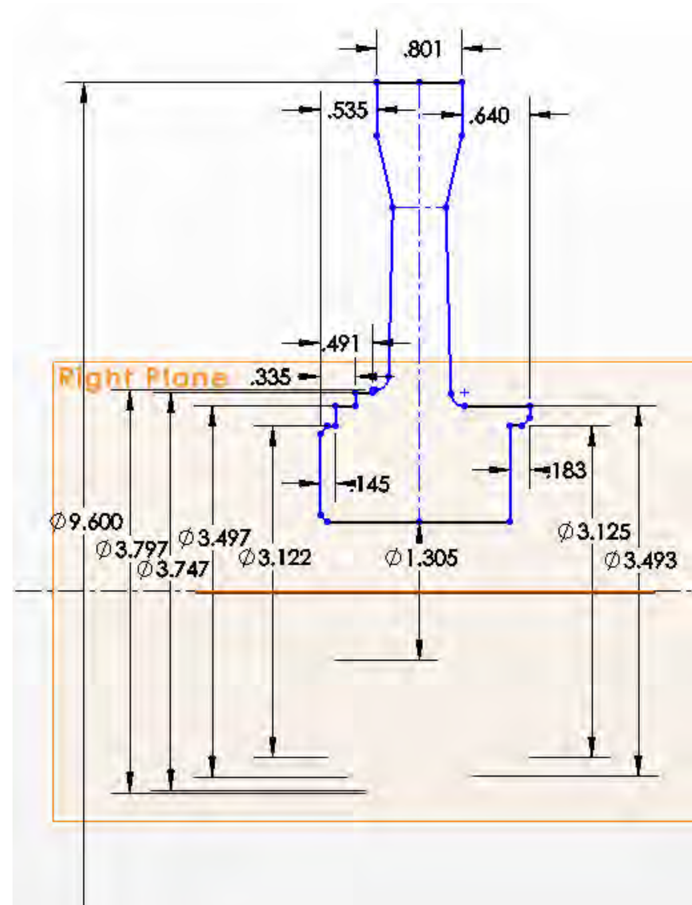
BLADE TIP DIAMETER		
	Upstream	Downstream
T1	11.657	11.910

DISK COOLING SLOTS	
T1	6 slots downstream face 0.497 wide

Appendix: Ramgen Solar Saturn Measurements

Stage 2



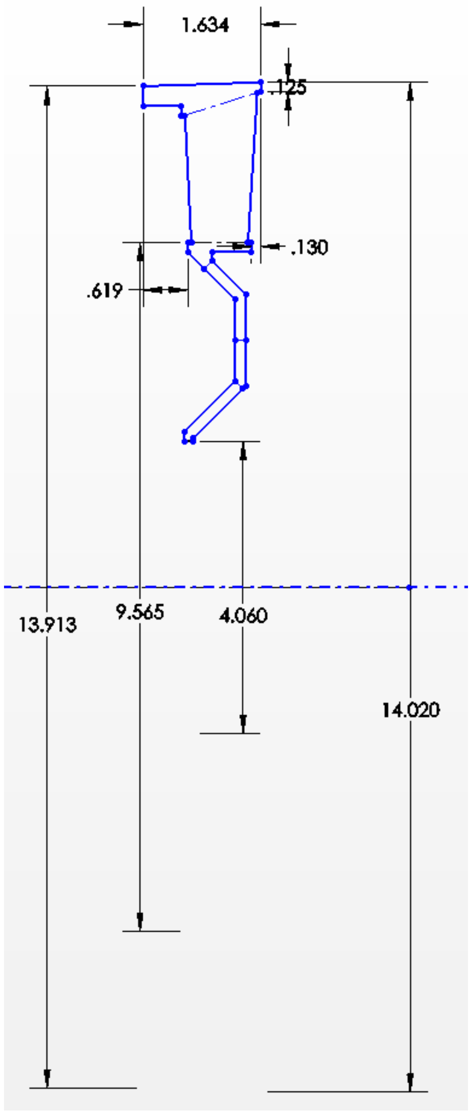


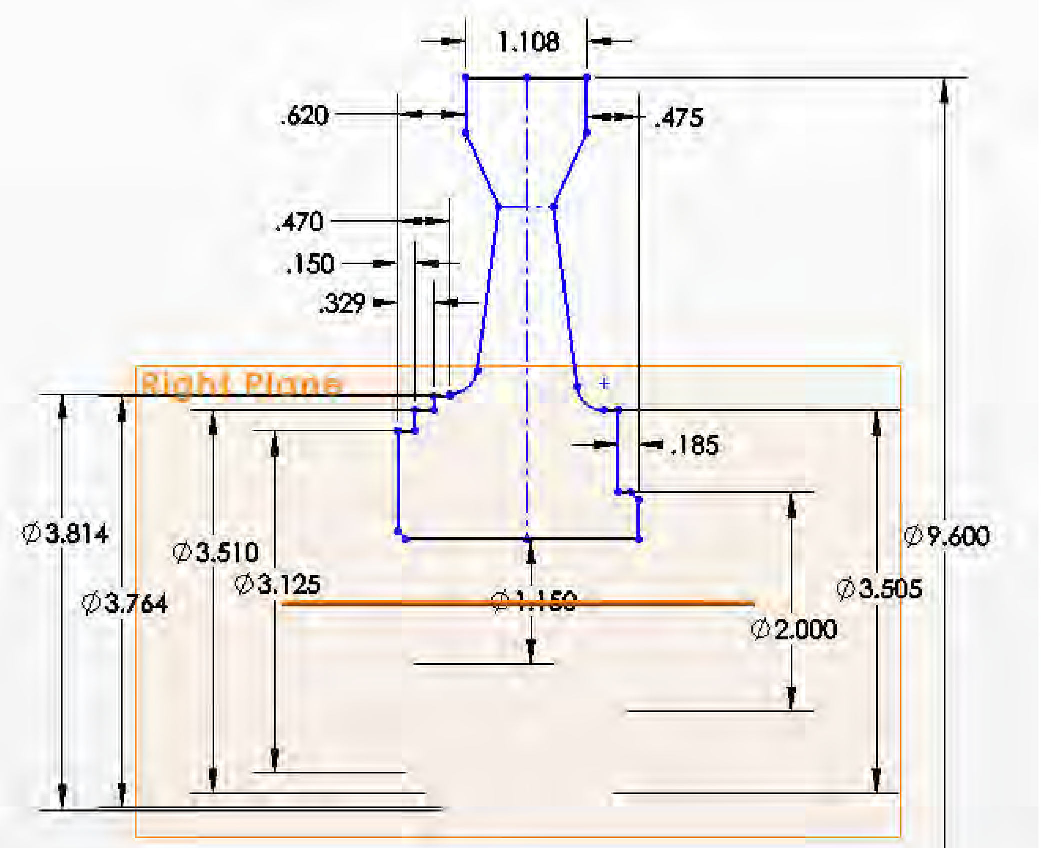
BLADE TIP DIAMETER		
	Upstream	Downstream
T2	12.697	12.974

DISK COOLING SLOTS	
T2	2 slots downstream face 0.373 wide

Appendix: Ramgen Solar Saturn Measurements

Stage 3





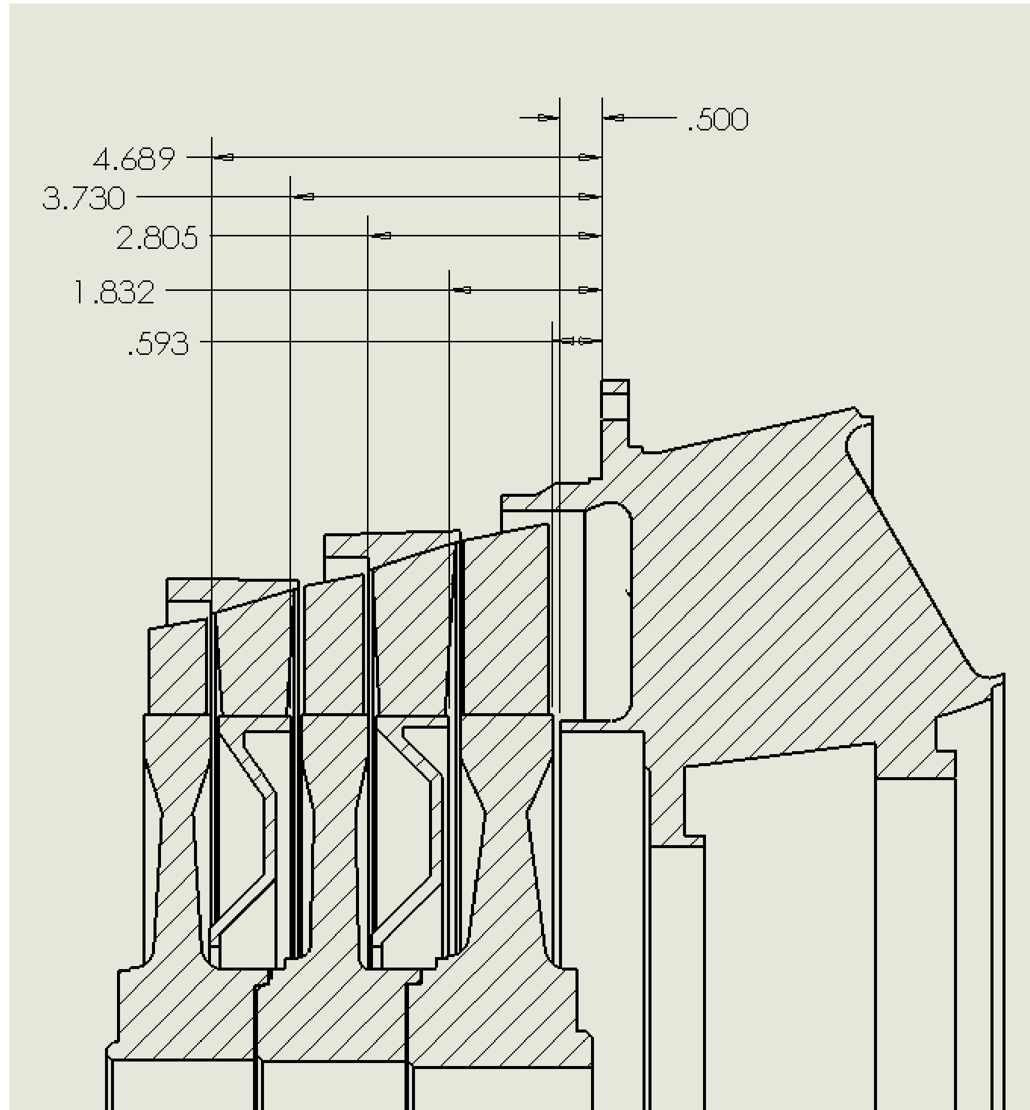
BLADE TIP DIAMETER		
	Upstream	Downstream
T3	13.787	14.194

DISK COOLING SLOTS	
T3	no cooling slots

Appendix: Ramgen Solar Saturn Measurements

Stackup

AXIAL STACKUP	
F to ED	0.500
GAP	0.093
F to T3	0.593
T3	1.108
GAP	0.131
F to N3	1.832
N3	0.885
GAP	0.088
F to T2	2.805
T2	0.801
GAP	0.124
F to N2	3.730
N2	0.871
GAP	0.088
F to T1	4.689
T1	0.801
GAP	0.147
F to N1	5.637



Appendix: Thermal Growth Bolt Loading

Component	Material	Ligament (in)	alpha (in/in/degF)	Starting Temp (F)	Max Temp (F)	Thermal Growth (in)
Flowguide	17-4 PH	0.5	6.70E-06	70	860	0.00265
Bolt	410 SS	0.5	6.50E-06	70	860	0.00257

Delta Growth	0.0000802	in
Bolt Length	1.5	in
Tensile Area	0.0878	in^2
Modulus	2.90E+07	psi
Force	136	lbf
Stress	1550	psi

Bolting

- **12 bolts equal spaced on 7 3/8" bolt hole circle**
- **3/8" UNF hex head cap screw, 1" length (existing), drilled for tie wire**
- **0.5" length adder for flowguide attachment**
- **Bolt material: ASTM A193 B6**
    1. **70F yield strength: 85 ksi**
    2. **70F tensile strength: 110 ksi**
    3. **410 SS Coefficient of Thermal Expansion: 6.5E-6 in/in/°F (32-1200°F)**
    4. **410 SS Static Modulus: 29E6 psi**
- **0.406" through hole in flowguide and exhaust diffuser**

Flowguide

- **0.5" thick at joint**

Appendix 11.2.3

Turboexpander FDR - Nozzle Assy

10:1 Turboexpander Final Design Review: Nozzle

Geene Cevrero

TBD, 2014

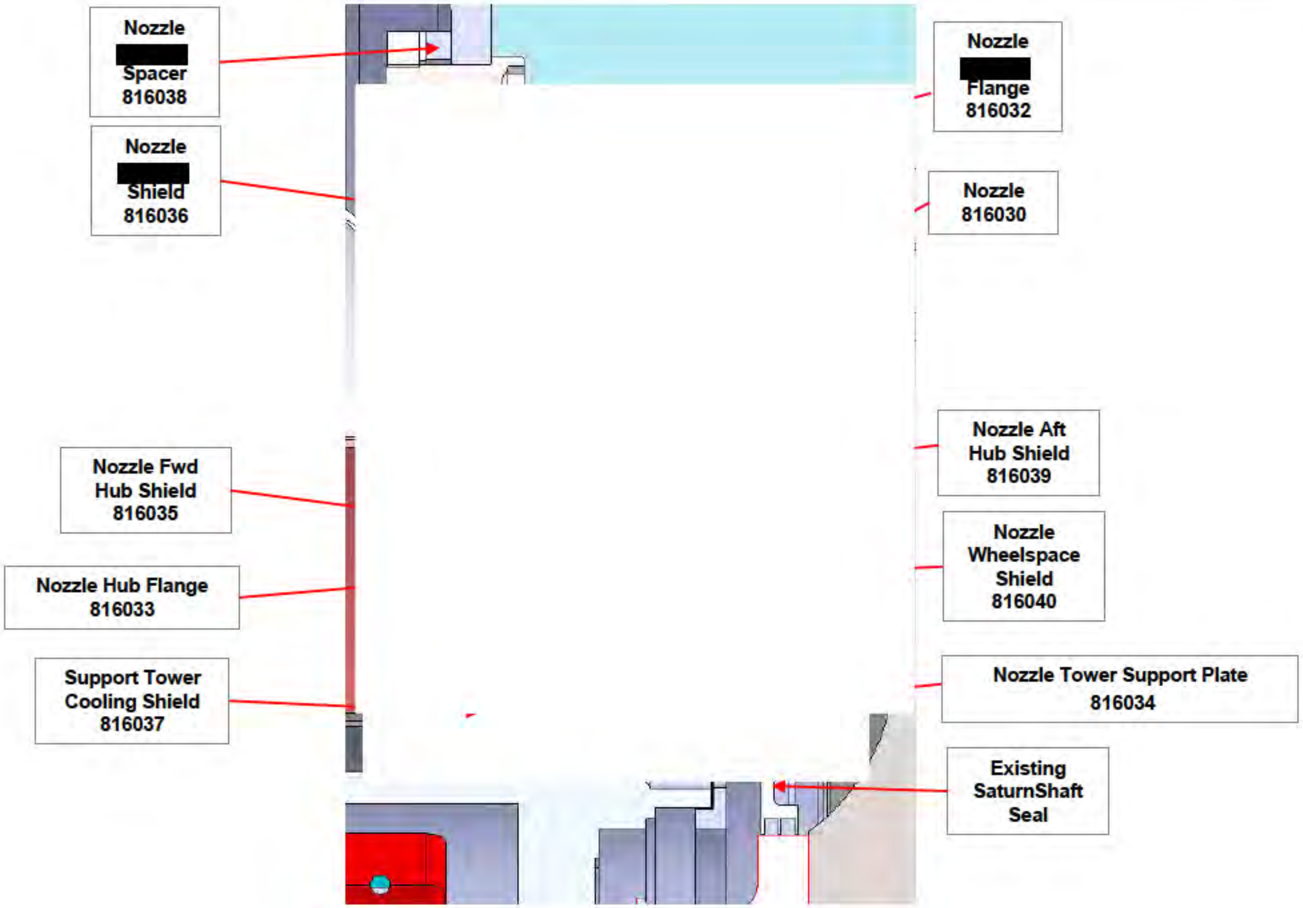
FDR Agenda

1. **Current Nozzle 015**
2. **Layout/Assembly/Manufacturing**
3. **Material Selection**
4. **Heat Transfer**
5. **Nozzle Structural**
6. **Flange Structural**
7. **Nozzle Modal**
8. **LCF/HCF/Creep**
9. **Growths & Clearances**

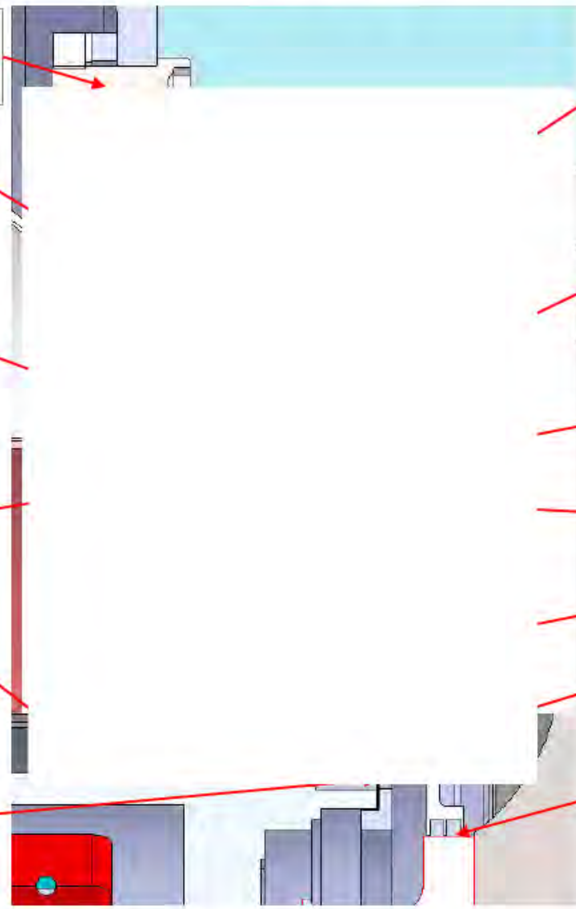
		Analysis	
Design Criteria	Heat Transfer Steady State	X	X
	Heat Transfer Transient		
	Peak Stress	X	X
	Flange Analysis	X	X
	LCF	X	X
	HCF	X	X
	NLH	X	X
	Crack Propagation		
	Creep	X	X
	Growths/Clearances	X	X
	Heat Fatigue		
	Hot to Cold	X	
	Modal Analysis	X	
	Nozzle Assembly	X	X

Nozzle Assembly: 11 Parts

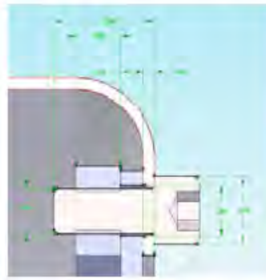
Nozzle Assy Components



Nozzle Interfaces: Overview



Hub Fasteners: *SHCS 1/4-28 UNF x 0.75"*



ALLOY 310 FASTENERS

Temp F	Strength	Yield Strength
958	79771	44672
1070	68941	38607
1138	62366	34925
1318	40611	22742
1498	26107	14620
1600	23366	13085
1678	13053	7310
1858	7252	4061

1070F

Assumes ΔP=128 psi @ Do=16", Di=12"

Although Bolt takes no external Load

1070F Bolt Preload and Clamping Calculations

Bolt Preload % -	0%	10%	20%	30%	40%	50%	60%	70%	80%	90%	100%	
F_i -	0	140	281	421	562	702	843	983	1,123	1 264	1 404	lb
F_i -	0	3 861	7 721	11,582	15,443	19,303	23,164	27,025	30,886	34 746	38 607	psi
N -	22	22	22	22	22	22	22	22	22	22	22	# of bolts
P -	509	509	509	509	509	509	509	509	509	509	509	lb
F_{BOLT} -	211	352	492	633	773	913	1,054	1,194	1,335	1 475	1 616	lb - Load per bolt based on joint stiffness
$S_{BOLTTENSION}$ -	5 810	9 670	13 531	17,392	21,252	25,113	28,974	32,834	36,695	40 556	44 417	psi - Stress in bolts
F_{MEMBER} -	298	157	17	-124	-264	-404	-545	-685	-826	-966	-1 107	lb - Clamping load in members based on joint stiffness
n_{BOLT} -	6.65	5.98	5.32	4.65	3.99	3.32	2.66	1.99	1.33	0.66	0.00	- Bolt load factor >1 means bolt stress will be below yield.
n_{CLAMP} -	0.00	0.47	0.94	1.42	1.89	2.36	2.83	3.30	3.77	4.25	4.72	- Clamp load factor >1 means joint should not gap.
TCOEFF -	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	- torque coefficient
T -	0	7	14	21	28	35	42	49	56	63	70	inlb
T	0	1	1	2	2	3	4	4	5	5	6	ft-lb
T	0	7	14	21	28	35	42	49	56	63	70	in-lb

1600F

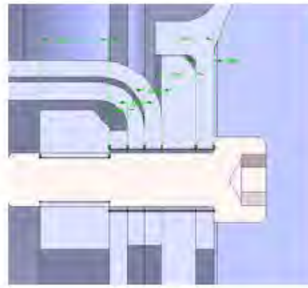
Assumes ΔP=55psi @ Do=16", Di=12"

Although Bolt takes no external Load

1600F Bolt Preload and Clamping Calculations

Bolt Preload % -	0%	10%	20%	30%	40%	50%	60%	70%	80%	90%	100%	
F_i -	0	48	95	143	190	238	286	333	381	428	476	lb
F_i -	0	1 308	2 617	3,925	5,234	6,542	7,851	9,159	10,468	11 776	13 085	psi
N -	22	22	22	22	22	22	22	22	22	22	22	# of bolts
P -	221	221	221	221	221	221	221	221	221	221	221	lb
F_{BOLT} -	92	139	187	235	282	330	377	425	473	520	568	lb - Load per bolt based on joint st ffness
$S_{BOLTTENSION}$ -	2 524	3 832	5 141	6,449	7,758	9,066	10,375	11,683	12,992	14 300	15 608	psi - Stress in bolts
F_{MEMBER} -	129	82	34	-13	-61	-109	-156	-204	-251	-299	-347	lb - Clamping load in members based on joint stiffness
n_{BOLT} -	5.18	4.67	4.15	3.63	3.11	2.59	2.07	1.56	1.04	0.52	0.00	- Bolt load factor >1 means bolt stress will be below yield.
n_{CLAMP} -	0.00	0.37	0.74	1.10	1.47	1.84	2.21	2.58	2.94	3.31	3.68	- Clamp load factor >1 means joint should not gap.
TCOEFF -	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	- torque coefficient
T -	0	2	5	7	10	12	14	17	19	21	24	inlb
T	0	0	0	1	1	1	1	1	2	2	2	ft-lb
T	0	2	5	7	10	12	14	17	19	21	24	in-lb

Support Tower Fasteners:
SHCS .190-28 UNF x .750"



ALLOY 310 FASTENERS

Temp F	Strength psi	Yield psi
70	90649	50763
470	85749	48019
958	79771	44672
1070	68941	38607
1138	62366	34925
1318	40611	22742
1498	26107	14620
1600	23366	13085
1678	13053	7310
1858	7252	4061

470F

1600F Bolt Preload and Clamping Calculations

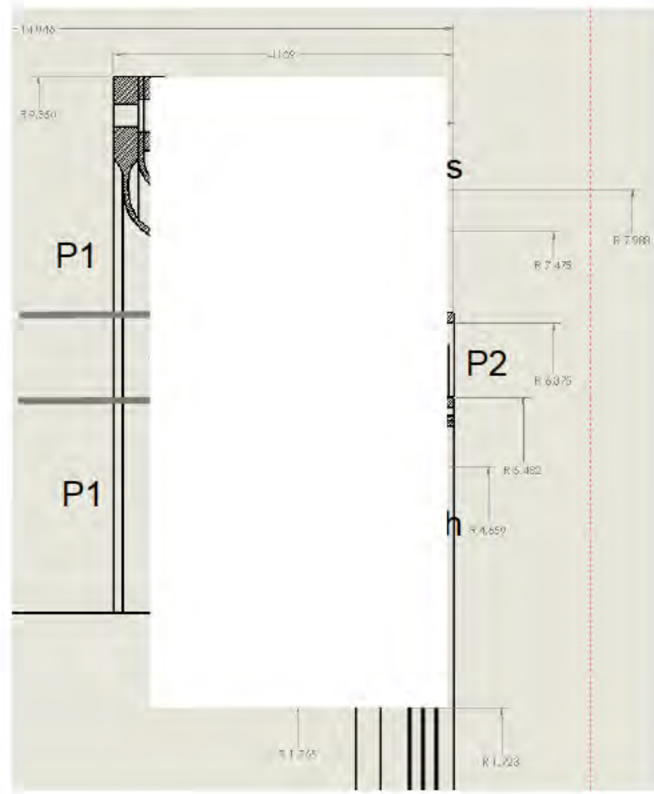
Bolt Preload % -	0%	10%	20%	30%	40%	50%	60%	70%	80%	90%	100%	
F_i -	0	48	95	143	190	238	286	333	381	428	476	lb
F_i -	0	1 308	2 617	3,925	5,234	6,542	7,851	9,159	10,468	11 776	13 085	psi
N -	22	22	22	22	22	22	22	22	22	22	22	# of bolts
P -	221	221	221	221	221	221	221	221	221	221	221	lb
F_{BOLT} -	92	139	187	235	282	330	377	425	473	520	568	lb - Load per bolt based on joint stiffness
$S_{BOLTTENSION}$ -	2 524	3 832	5 141	6,449	7,758	9,066	10,375	11,683	12,992	14 300	15 608	psi - Stress in bolts
F_{MEMBER} -	129	82	34	-13	-61	-109	-156	-204	-251	-299	-347	lb - Clamping load in members based on joint stiffness
n_{BOLT} -	5.18	4.67	4.15	3.63	3.11	2.59	2.07	1.56	1.04	0.52	0.00	- Bolt load factor >1 means bolt stress will be below yield.
n_{CLAMP} -	0.00	0.37	0.74	1.10	1.47	1.84	2.21	2.58	2.94	3.31	3.68	- Clamp load factor >1 means joint should not gap.
TCOEFF -	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	- torque coefficient
T -	0	2	5	7	10	12	14	17	19	21	24	inlb
T	0	0	0	1	1	1	1	1	2	2	2	ft-lb
T	0	2	5	7	10	12	14	17	19	21	24	in-lb

Assumes ΔP=55psi @ Do=16", Di=12"

Although Bolt takes no external Load

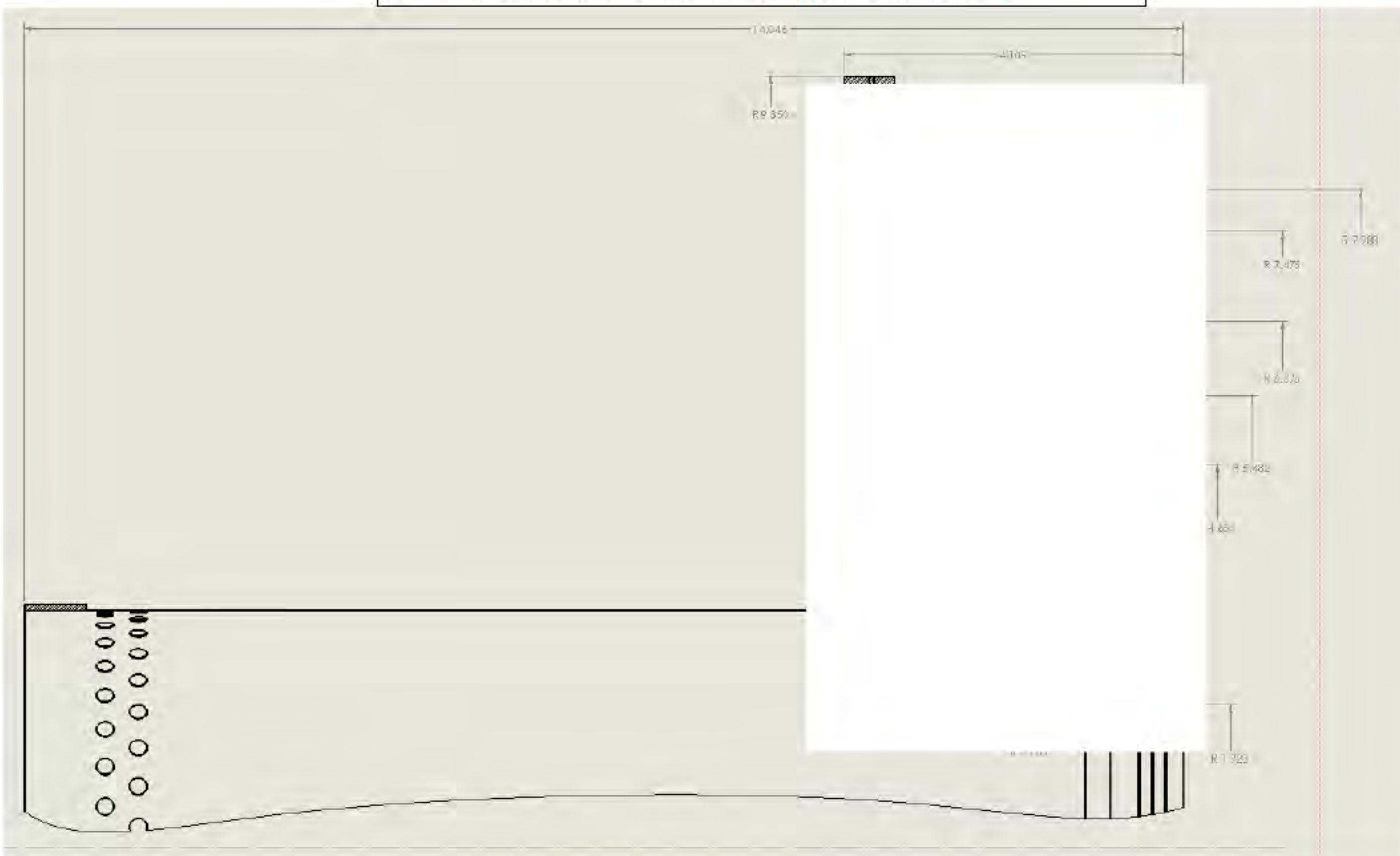
Nozzle Assembly DP Pressure Conditions

1) Thermal Map 1070F, 1600F
2) Design Point Pressures
   P1=147psi
   P2s=20 psi
   P2=14.7 psi
   P2h=20psi



Nozzle Assembly Dimensions

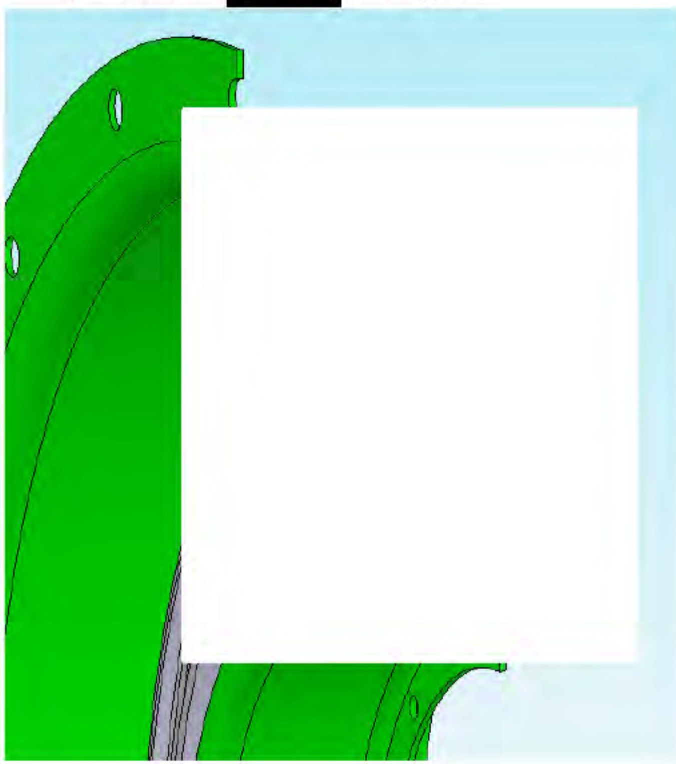
10:1 Nozzle and components assemble with reuse of Solar shaft Seal



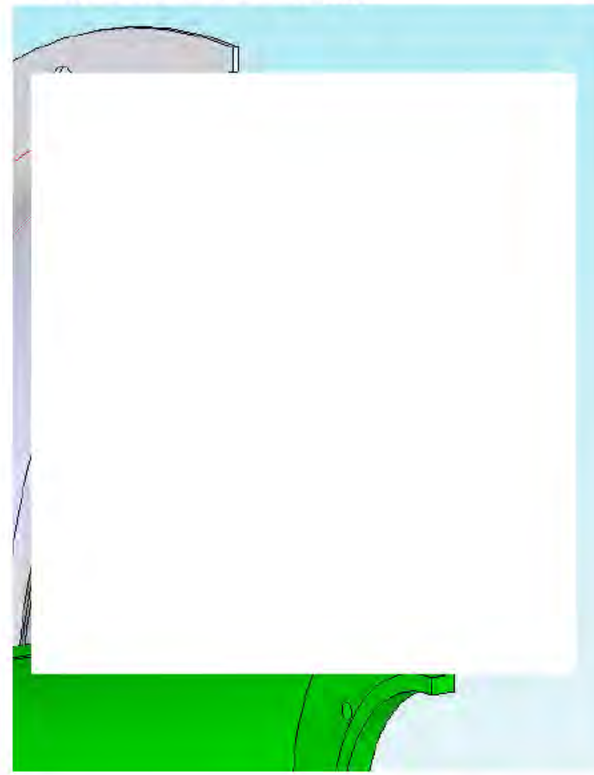
Nozzle Assembly

10:1 Nozzle and components assemble with reuse of Solar shaft Seal

1) Hub and [redacted] Flanges



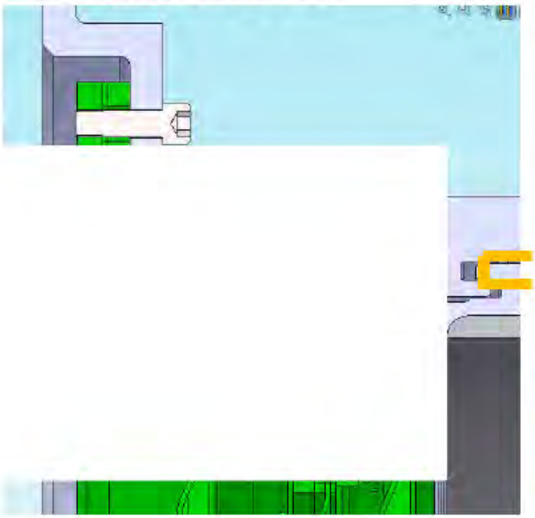
2) Lower Hub Flange



RAMGEN POWER SYSTEMS™

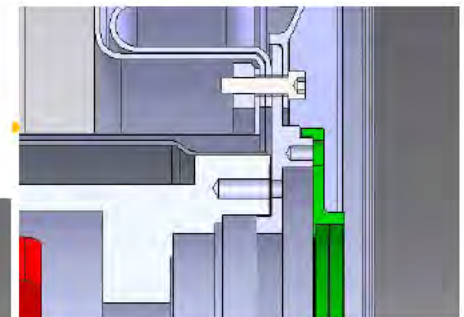
Nozzle Assembly

3) Turbine Case



4) Support Tower

5) Shaft Seal



(10:1 reuse of Solar shaft Seal)

NOZZLE 015 AERO PARAMETERS

Strake Layout		
Thickness	0.19	[in]
Elliptical LE	0.1	[fraction of chord]
TE Wedge	5.750	[deg]
TE Thick	0.03	[in]
Strake Lth	5.3246	[in]

Nozzle 015: Parameters and Requirements

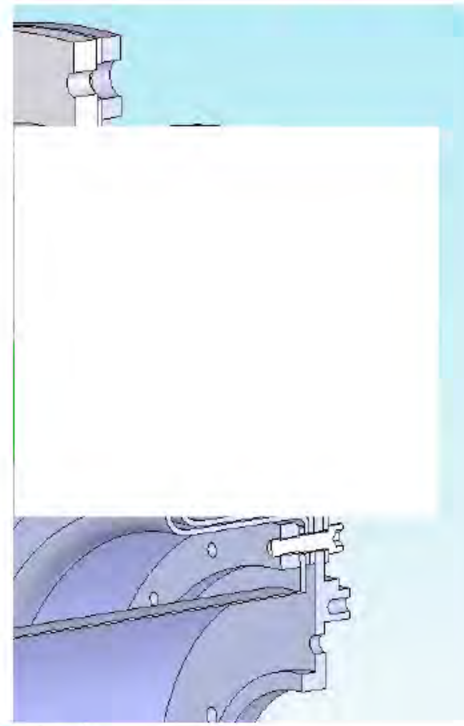
Hot Inlet Area	33.793	in^2
Cold Inlet Area	33.264	in^2
Cold Ramp Inlet	0.820	in^2
Cold Ramp Throat	0.392	in^2
Cold Exit	0.821	in^2

Variable Fillet on Hub & [redacted]

RAMGEN POWER SYSTEMS ™

Manufacturability

- 10:1 Turboexpander Nozzle is a fastened Assy with various Flange and Heat Shield Parts
- Nozzle Casting (BESCAST)
  1. Bescast(Casting chosen over machining due to EDM risks and Cost)
     - In738LC Vacuum pour Cast (improved mechanical capability over IN718)
     - Development Process underway, 18 wk lead to produce 1 Unit
     - Cast per AMS5410 and Inspection per AMS2175, done after Final Machining
     - Inspection per AMS2175
     - Development Cost: $20.5K
     - PO 4551 released 3/10/14
- Nozzle Casting Final Machining
  1. Bescast(Casting chosen over machining due to EDM risks and Cost)
     - No Drawings/Models given to Vendor
     - Type of Machining: Turning and either Gundrill/EDM for 44 Holes
     - Issues: Leadtime/Cost oon Instrumentation holes
     - Delivery?
     - Cost?
- Flange and Heat Shield Machining: In738LC
  1. Vendor XXX
     - No Drawings/Model given to Vendor
     - Type of Machining?
     - Issues?
     - Delivery?
     - Cost?
- Bolts or Fastener Ring
  1. Ring drawings needed



BUDGET:

Turbine Hardware					
Modified AVC disharge components			$	60,000	
Nozzle			$	200,000	
Turbine Shaft			$	25,000	
Turbine case			$	20,000	
Turbine disk/blades			$	150,000	
Turbine OGV's			$	75,000	
Assy Balance & Spin Pit Proof Test	$	30,000			
Assembly tooling			$	3,000	
Task total	$	30,000	$	533,000	$

Other Manufacturing items

- **10:1 Turboexpander Nozzle is a fastened Assy with various Flange and Heat Shield Parts**
- **Flange and Heat Shield Machining: In738LC**
  1. **Vendor XXX**
     - **Drawings:**
     - **Type of Machining?**
     - **Issues?**
     - **Delivery?**
     - **Cost?**

- **Bolts or Fastener Ring**
  1. **High Temp Bolts: Quote received from XXX**
  2. **Fastener Ring: Drawings needed for quote**

BUDGET:

Turbine Hardware					
Modified AVC disharge components			$	60,000	
Nozzle			$	200,000	
Turbine Shaft			$	25,000	
Turbine case			$	20,000	
Turbine disk/blades			$	150,000	
Turbine OGV's			$	75,000	
Assy Balance & Spin Pit Proof Test	$	30,000			
Assembly tooling			$	3,000	
Task total	**$**	**30,000**	**$**	**533,000**	**$**

High Temp Bolts

ALLOY 310 S

Designation and Specifications		Sheet/strip		Tube/pipe	
USA	Designation , UNS	S 31008			
	ASTM (B) ASME (SB)	167/240		213/249/312	
GB	Designation , BS	310 S 24		-	
	BS	1449		-	
F	AFNOR	Z 12 CN 25.20			
D	Designation	X 12 CrNi 25 21			
	Werkstoff-Nr.	1.4845			
	DIN	-		-	
	SEW	470		-	
	VdTUV data Sheet	-		-	
Chemical composition (%)					
Nickel		19 – 22			
Chromium		24 – 26			
Iron		Balance			
Silicon		0.2 – 0.7			
Aluminium		-			
Titanium		-			
Carbon		Max. 0.08			
Others		RE: max. 0.08 (Rare Earths)			
Mechanical properties (N/mm², %)					
Temperature (°C)		**Rp 0.2**	**Rp 1.0**	**Rm**	**A5**
20		min. 210	250	min. 500	min. 35
100		190	220	-	-
200		170	195	-	-
300		160	185	-	-
400		150	175	-	-
500		140	165		
Creep properties (N/mm²)					
Temperature (°C)		Rp 1.0/10^4	Rm /10^4	Rp 1.0/10^5	Rm10^5
600		105	160	-	80
700		37	40	-	18
800		12	18	-	7
900		5.7	8.5	-	3
Physical properties at room temperature or as indicated					
Density	g/cm³	7.9			
Specific heat	J/kg K	500			
Thermal conductivity	W/m K	14			
Electrical resistivity	μ Ω cm	85			
Thermal expansion 20-300°C	10^{-6}/K	16.4			
Modulus of elasticity	kN/mm²	200			
Fabrication characteristics					
Formability		Good			
Weldability		Good			
Welding products					
Filler wire		1.4842			
Covered electrode		1.4842			
		EL-25 20			

http://www.extreme-bolt.com/Materials-Hastelloy-Fasteners.html

http://www.alloy-fasteners.com/nickel-alloy-grades.html

Article on High Temp Fasteners:

http://www.azom.com/article.aspx?ArticleID=1175

Inco 625: 1400F

Inco 718: 1600F

Alloy 310: 1800F (Furnace applications)

High Temp Bolts….continued

SX 310 is a highly alloyed austenitic stainless steel used for high-temperature applications. The high chromium and nickel contents give the steel excellent oxidation resistance as well as high strength at high temperatures. This grade is also very ductile, and has good weldability enabling its widespread usage in many applications.
SX 310S is the low carbon version of SX 310 and is suggested for applications where sensitisation, and subsequent corrosion by high temperature gases or condensates during shutdown may pose a problem.
SX 310 is manufactured in accordance with ASTM A 167 and SX 310S to ASTM A 240.

Typical Applications

SX 310/310S find wide application in all high-temperature environments where scaling and corrosion resistance, as well as high temperature strength and good creep resistance, are required.

Chemical Composition

SX	C	Mn	P	S	Si	Cr	Ni
310	0.25 max	2.0 max	0.045 max	0.030 max	1.5 max	24.0 - 26.0	19.0 - 22.0
310S	0.08 max	2.0 max	0.045 max	0.030 max	1.5 max	24.0 - 26.0	19.0 - 22.0

Typical Properties in the Annealed Condition
The properties quoted in this publication are typical of mill production and unless indicated should not be regarded as guaranteed minimum values for specification purposes.

1. Mechanical Properties at Room Temperature

	SX 310		SX 310S	
	Typical	Minimum	Typical	Minimum
Tensile Strength, MPa	625	515	575	515
Yield Stress (0.2 % offset), MPa	350	205	290	205
Elongation (Percent in 50mm)	50	40	50	40
Hardness (Brinell)	172	-	156	-
Endurance (fatigue) limit, MPa	260	-	260	-

2. Properties at Elevated Temperatures
The values quoted are those for SX 310.
Enquire for data on 310S .

Short Time Elevated Temperature Tensile Strength

Temperature, °C	550	650	750	850	950	1050
Tensile Strength, MPa	550	430	280	180	90	50

Creep data
Stress to develop a creep rate of 1% in the indicated time at the indicated temperature.

Time	Temperature°C	550	600	650	700	750	800
10 000 h	Stress MPa	110	90	70	40	30	15
100 000 h	Stress MPa	90	75	50	30	20	10

Creep Rupture Stress

Time	Temperature °C	600	700	800	900	1000
1 000 h	Stress MPa	190	110	50	35	15
10 000 h	Stress MPa	170	70	35	20	10
100 000 h	Stress MPa	110	55	25	10	2

Recommended Maximum Service Temperature
(Oxidising Conditions)
Continuous 1150°C
Intermittent 1035°C

Thermal Processing
1. **Annealing.** Heat from 1050 to 1150°C and water quench. This treatment ensures that all carbides are in solution.
2. **Hot working**
   Initial forging and pressing: 1150 - 1200°C
   Finishing temperature: above 950°C

Note: Soaking times to ensure uniformity of temperature are up to 12 times that required for the same thickness of mild steel.

Casting Issues

Casting Manufacturers	Chromalloy	Bescast	Comments
Partsize:			
Open Air Pou	15"	up to 30" Dia	Must be Uniaxis Materials: Inco 625 (common), Hastalloy X(very reactive time/temp)
Vacuum Pou	15"	limited to 20-25lbs on actual Part	Must be Uniaxis Materials: Inco 718, 738, 909 & MAR-M-247, CM 247LC, CM 186LC
Flowpath Fillet radius:			
Min	prefer 040-.060R min	0.031R	Bescast seemed confident with experience
Profile Tolerance:			
Strake to Strake	+/-.010"	+/-.010"	Similar tolerances, although Bescast seemed more confident due to experience
Inlet/Exit Diameters	+/-.045-.060"	+/- 040	
Min Throat Height:	>.400"	> 500"	Shell Integrity 4 layers per side, 4 layer= .180-.200". Bescast said if smaller than .500 than a ceramic core s needed.
Surface Finish:	Ra=75avg	Ra=85avg	chromalloy uses aluminum grip
Trailing Edge Min thickness	Prefers .050-.060"	.010"	Bescast seemed confident and has experience doing smaller TE thickness (ie .021"TE on 14" Blade length 2.5" blade thickness)
Instrumentation Holes	possible	possible	Chromalloy more hesitant than Bescast, although both recommonded post machining
Inspection process	forgot to ask	AMS2175	
Secondary In house processes	no	no	Bescast has a Sweko vibration machine for small single blades

Min/Max Casting Tolerances

Expected Casting Tolerances in Cold Condition

DIMS	**Nominal**	**Max**	**Min**	**Max Tol**	**Min Tol**	
Inlet Do	11.052	11.092	11.012	0.04	-0.04	in
Inlet Di	10.964	11.004	10.924	0.04	-0.04	in
Nozzle Inlet W	0.927	0.937	0.917	0.01	-0.01	in
Nozzle Inlet H	0.894	0.904	0.884	0.01	-0.01	in
Nozzle Throat W	0.924	0.934	0.914	0.01	-0.01	in
Nozzle Throat H	0.427	0.437	0.417	0.01	-0.01	in
Nozzle Exit W	0.928	0.938	0.918	0.01	-0.01	in
Nozzle Exit H	0.894	0.904	0.884	0.01	-0.01	in
Exit Do	11.052	11.092	11.012	0.04	-0.04	in
Exit Di	10.964	11.004	10.924	0.04	-0.04	in

AREAS				Max % Tol	Min % Tol	
Inlet	1.522	1.527	1.516	0.362	-0.365	in^2
Nozzle Inlet	0.829	0.847	0.811	2.162	-2.234	in^2
Nozzle Throat	0.395	0.408	0.381	3.334	-3.518	in^2
Nozzle Exit	0.830	0.848	0.812	2.160	-2.233	in^2
Exit	1.522	1.527	1.516	0.362	-0.365	in^2

* no fillets

Inco 718 vs 738C: Tensile Properties

Inco 718

- RT Ultimate strength: 180 ksi
  - 900°F : 90% of RT = 162 ksi
  - 1200°F: 82% of RT = 148 ksi
  - 1400°F: 50% of RT = 90 ksi
  - 1500°F: 32% of RT = 58 ksi
  - 1600°F: 38% of RT = ??
- RT Tensile yield strength: 145 ksi
  - 900°F: 90% of RT = 131 ksi
  - 1200°F: 82% of RT = 119 ksi
  - 1400°F: 52% of RT = 75 ksi
  - 1500°F: 38% of RT = 55 ksi
  - 1600°F: 38% of RT = ??

Inco 738 C

- RT Ultimate strength: 180 ksi
  - 900°F : = 155 ksi
  - 1200°F: = 153 ksi
  - 1400°F: = 140 ksi
  - 1500°F: = 126 ksi
  - 1600°F: = 112 ksi
- RT Tensile yield strength: 145 ksi
  - 900°F: = 133 ksi
  - 1200°F: = 132 ksi
  - 1400°F: = 115 ksi
  - 1500°F: = 97.5 ksi
  - 1600°F: 38% of RT = 80 ksi

TABLE VII
Short-Time Elevated Temperature Tensile Properties of Alloy IN-738

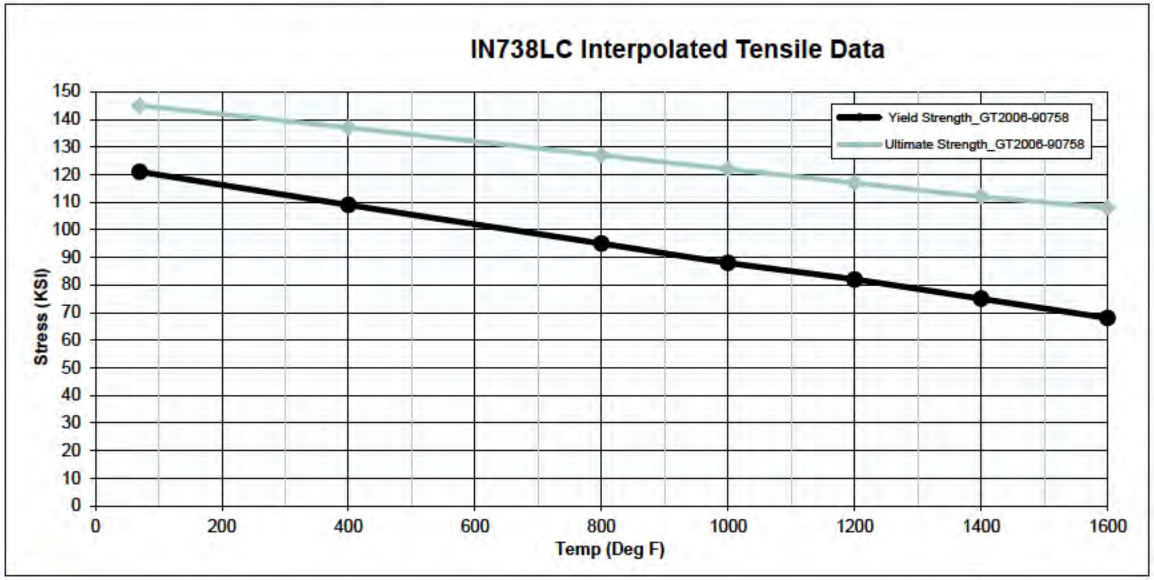
Temperature °F	Yield Strength (0.2% Offset) psi	Tensile Strength psi	Elongation (2 in.) %	Reduction of Area %
70	138,000	159,000	5.5	5
1200	132,000	153,000	7	7
1400	115,000	140,000	6.5	9
1600	80,000	112,000	11	13
1800	50,000	68,000	13	15

TABLE VIII
Stress-Rupture Properties of Alloy IN-738C

Temperature °F	Stress psi	Life hr	Elongation %	Reduction of Area %
1350	90,000	212	6	8
1500	40,000	3314	5	5
1700	33,000	95	8	14
1800	22,000	66	12	18

Casting Properties are will be IN738LC

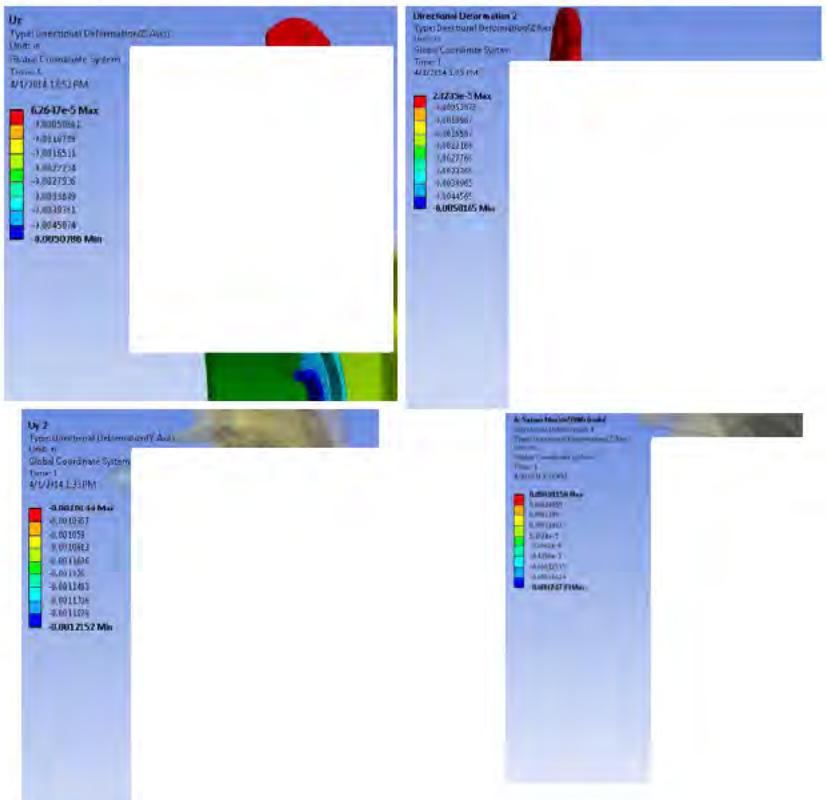
Material Strength: IN738LC

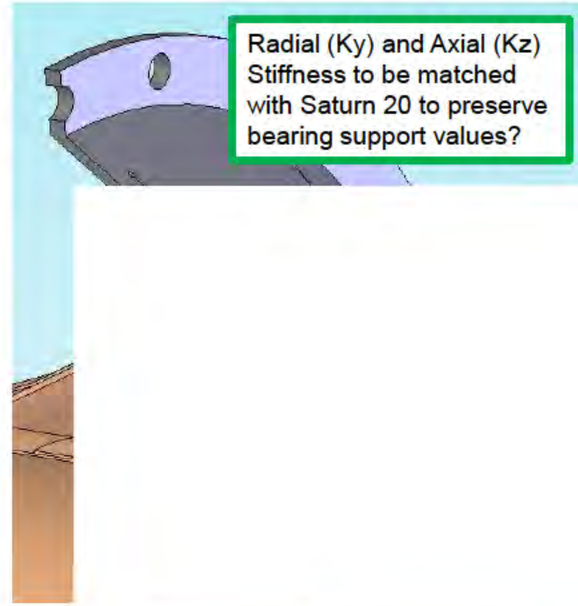


IN738											
IN738 LC(Low Carbon) From IN738 Technical Data by The International Nickel Company			IN738 LC (Carbon) Fatigue Life Paper_GT2006-90758			IN738 C (Carbon) From IN738 Technical Data by The International Nickel Company			Engineering Specification (Sundstrand Turbomach)		
Temp (Deg F)	UTS (Ksi)	Yield (ksi)	Temp (Deg F)	UTS (Ksi)	Yield (ksi)	Temp (Deg F)	UTS (Ksi)	Yield (ksi)	Temp (Deg F)	UTS (Ksi)	Yield (ksi)
70	150	130	70	145	121	70	159	138	70	125	108
1200			400	137	109	1200	153	132	400		
1400			800	127	95	1400	140	115	800		
1600			1000	122	88	1600	112	80	1000		
1800			1200	117	82	1800	66	50	1200	125	91
			1400	112	75				1400		
			1600	108	68				1600		
			1800						1800		

Saturn 20 Nozzle: Stiffness Comparison

	ISCE B2 (Model 026)		Saturn		
	Ky	Kz	Ky	Kz	
F	500	500	500	500	lb/in
x	0.0011	0.0051	0.0001	0.0050	in
k	4.565E+05	9.845E+04	9.752E+06	9.967E+04	lb/in
Moment	238		863		lb-in





Radial (Ky) and Axial (Kz) Stiffness to be matched with Saturn 20 to preserve bearing support values?

Nozzle SS Heat Transfer Analysis

MODEL

- All engine rotor components modeled
- Nozzle components modeled as IN738
- Compressor rotor components modeled as SS410
- Nozzle modeled with new element type (152) for improved HTC accuracy
- Nozzle contact surfaces modeled withvarying contact thermal conductivity (see conductivity Map)
- Remaining gaps modeled with thermal conductivity per rules of thumb, based on gap size
- Fluid elements across turbine blade
  1. 1075°F: Ttrel=685F, 8.15 lbm/s
  2. 1600°F: Ttrel=1080F, 7.0 lbm/s
- Fluid pumping elements along upstream and downstream rotor faces
  - 100°F for 1075°F case (at inlet)
  - 500°F for 1600°F case (at inlet)
  - 0.08 lbm/s (each side)

RESULTS

- Thermal gradient generates excessive TE stresses due to small thickness and greater Temperature difference due to the Wheelspace Flow
- 500°F wheel space cooling air needed for 1600°F case to reduce T_{bulk} to T_{max} difference
- New blade element type increases HTC making blade hotter than previous results
- Applying wheel space purge flow at actual inlet locations results in 150°F temperature drop by the time flow enters actual wheel space for 1600°F case
- Disk rim compressive thermal stress higher

1600°F / 27,000 RPM Rotor Thermal Map (assumed same for 1600°F / 23,000 RPM case)

1100°F / 23,000 RPM Rotor Thermal Map

Contact info: Conductivity Values

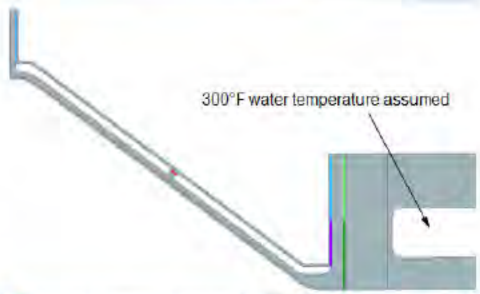
	Type of Contact	Gap, Compression in	Gap, Compression m	HTC BTU/s-in^2-F	K BTU/hr-ft^2-F	K W/m^2-K	K BTU/hr-ft-F
1	Large Gap	0.0900	0 0023	9.65E-06	5 00	28.37	0 0375
2	.025" Gap Contac	0.0250	0 0006	3.47E-05	18.00	102.14	0 0375
3	.010" Gap Contact	0.0100	0 0003	8.49E-05	44.00	249.67	0 0367
4	Loose Gasket (loose on either side)			6.36E-05	32.95	186.95	0.1400
5	Tight Gasket (tight on either side)			1.49E-04	77.33	438.78	0 2835
6	Loose Contact (rule of thumb)	0.0044	0 0001	1.93E-04	100.00	567.44	0 0367
7	Tight Contact (rule of thumb)	0.0009	0 0000	9.65E-04	500.00	2,837.20	0 0367
8	Welded	0.0004	0 0000	0.001929012	1,000.00	5,674.40	0 0367
9	Treated Essentially as 1 Piece of metal	0.0001	0 0000	0.009645062	5,000.00	28,372.01	0 0367

2D Liner Aft Segment Model

Contacts

QUEST Preliminary Document

- Inserted 1D gasket effect
- Assumed tight gasket connection along outer radius, loose gasket along inner radius due to liner flexion
- Tight gasket
  - Tight contact (HTC = 500) on both sides of gasket
  - Gasket thickness x = 0.043"
  - Gasket HTC = K/x = 114
  - HTC = $(\frac{1}{500}+\frac{1}{114}+\frac{1}{500})^{-1} = 78$
- Loose gasket
  - Loose contact (HTC = 100) on both sides of gasket
  - Gasket thickness x = 0.051"
  - Gasket HTC = 96
  - HTC = $(\frac{1}{100}+\frac{1}{96}+\frac{1}{100})^{-1} = 33$



Contact Color	Contact Type	H (BTU/hr-ft²-F)
purple line	0.025" Gap	18
red line	0.01" Gap	44
green line	Loose Gasket	33
light green line	Tight Gasket	78
blue line	Tight	500

Nozzle Assy Thermal Conductivity Map

Nozzle_036
4/11/2014 4:26 PM

A Nozzle_001
B Nozzle_002
C Nozzle_003
D Nozzle_004
E Nozzle_005
F Nozzle_006
G Nozzle_007
H Nozzle_008
I Nozzle_009
J Nozzle_010
K Nozzle_011
L Nozzle_012
M Nozzle_013
N Nozzle_014
O Nozzle_015
P Nozzle_016
Q Nozzle_017
R Nozzle_018
S Nozzle_019
T Nozzle_020
U Nozzle_021
V Nozzle_022
W Nozzle_023
X Nozzle_024
Y Nozzle_025
Z Nozzle_026
AA Nozzle_027
AB Nozzle_028
AC Nozzle_029
AD Nozzle_030
AE Nozzle_031
AF Nozzle_032
AG Nozzle_033
AH Nozzle_034
AI Nozzle_035
AJ Nozzle_036

K Value

1	28372 W/m^2-K
2	2837
3	2837
4	2837
5	2837
6	2837
7	2837
8	567
9	567
10	2837
11	2837
12	2837
13	2837
14	2837
15	2837
16	567
17	2837
18	2837
19	249
20	567
21	28
22	2837
23	2837
24	28
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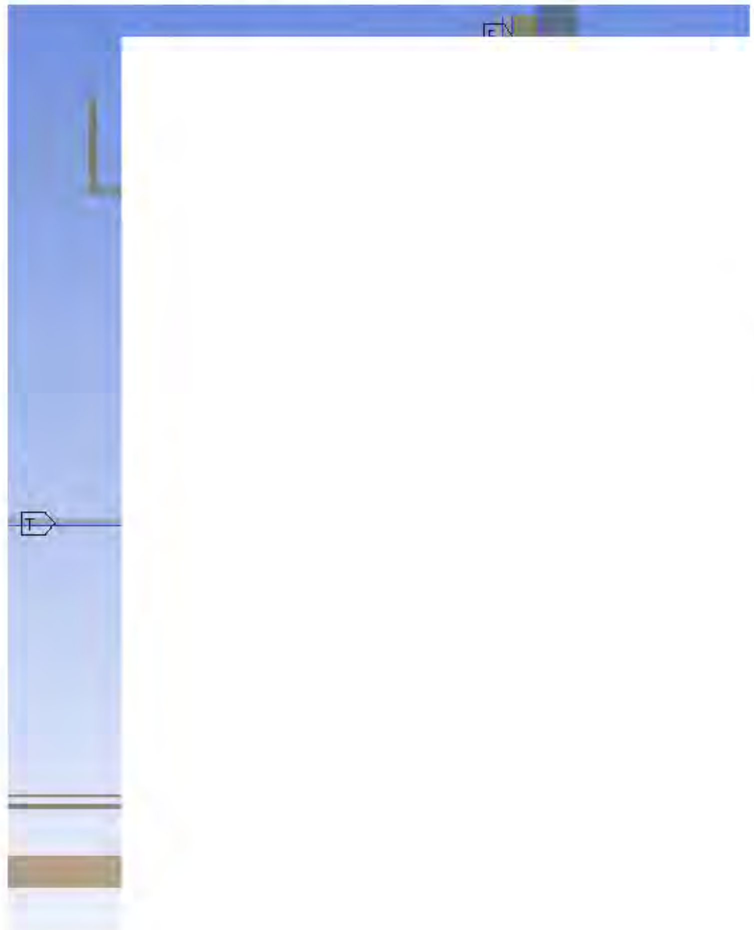
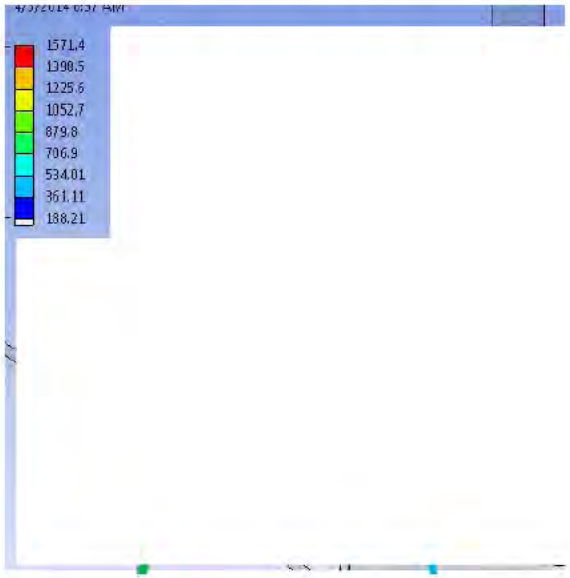
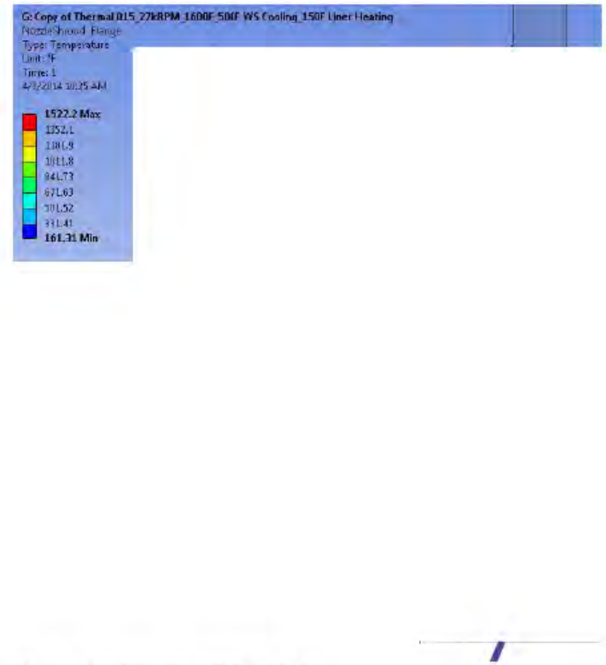
34	28
35	2837
36	28



[redacted] Seal to lengthen Temp gradient in [redacted] Flange

Thermal Map 26A & 26B: [redacted]

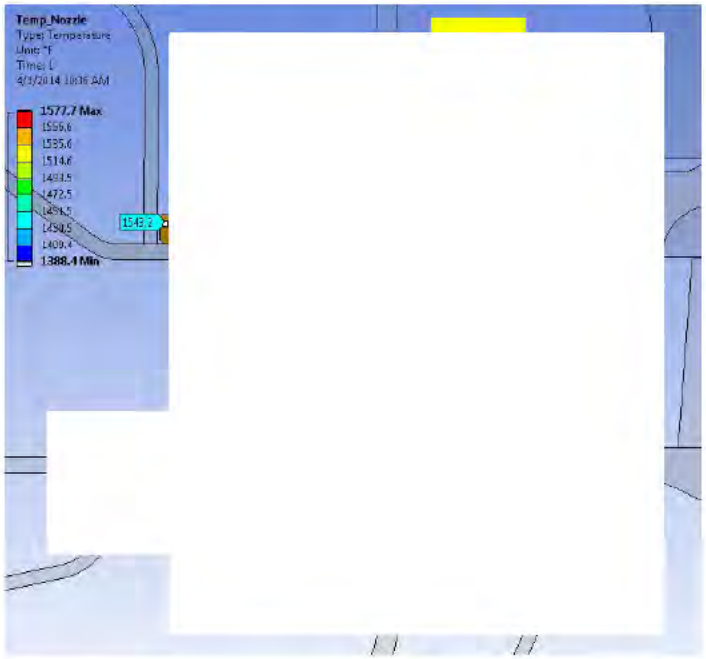


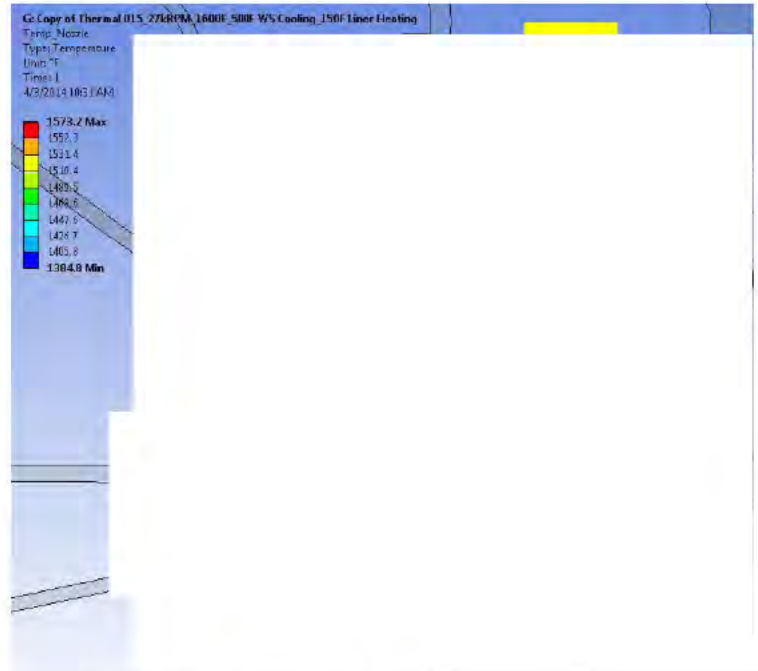


Contact Conductance 1: 28 W/m^2-K
Contact Conductance 2: 20 W/m^2-K

Temp Gradient has more length in 26B

Thermal Map 26: Nozzle Effects



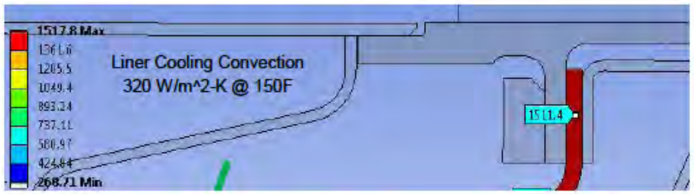


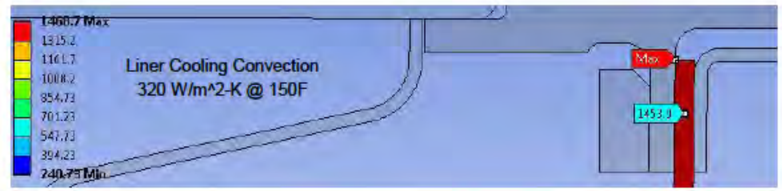
Biggest Effect of eliminating Exit Flow [redacted]
Convection is at the [redacted] and Hub Flanges

[redacted] 1570F to 1515 F (ΔT=55F)

HUB: 1522 F to 1472 F (ΔT=50F)

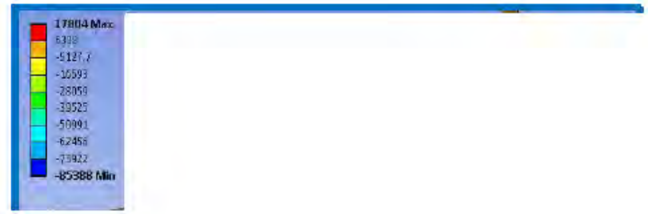
Thermal Map 26: Hub



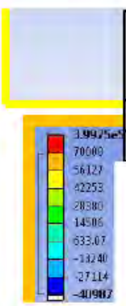


Temp Gradient has more length in 26B

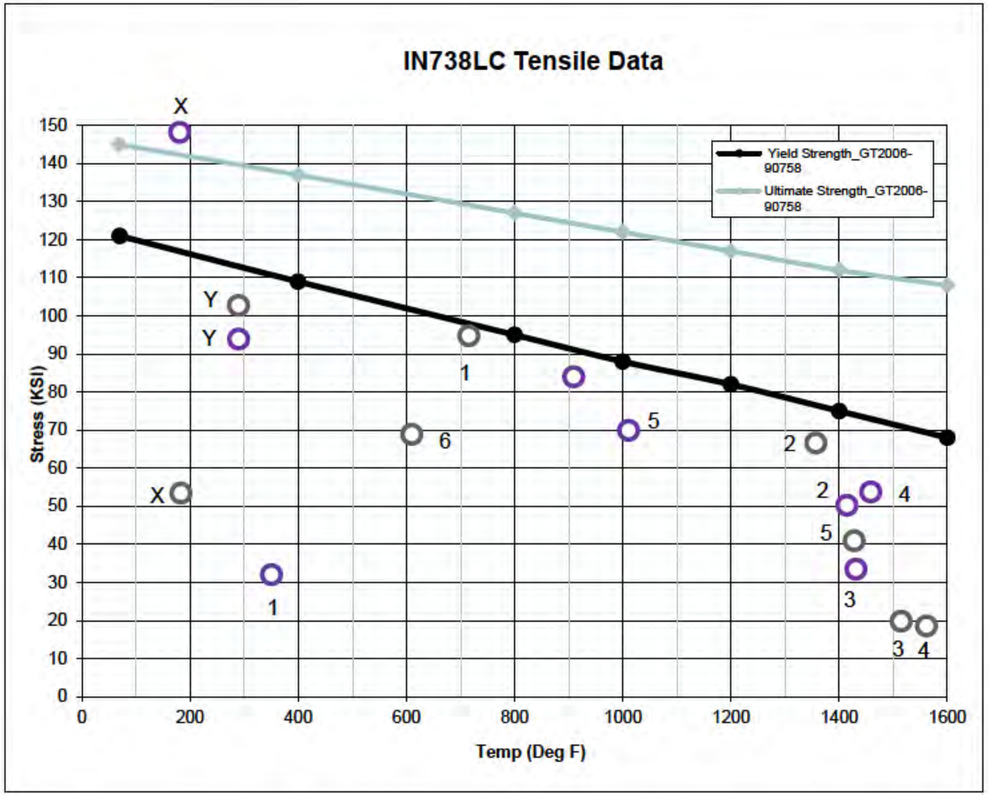
2D FEA Comparison



Geometry	Loading Types	[redacted] Flange Smax	[redacted] Flange Smin	[redacted] Flange Seq	Hub Flange Smax	Hub Flange Smin	Hub Flange Seq	Uz @ [redacted]
26	Combined	70	-96	88	69	-37	61	-0.050
26B	Combined	62	-85	76	70	-54	62	-0.058



2D Flange FEA: 1600F & DP Pressure

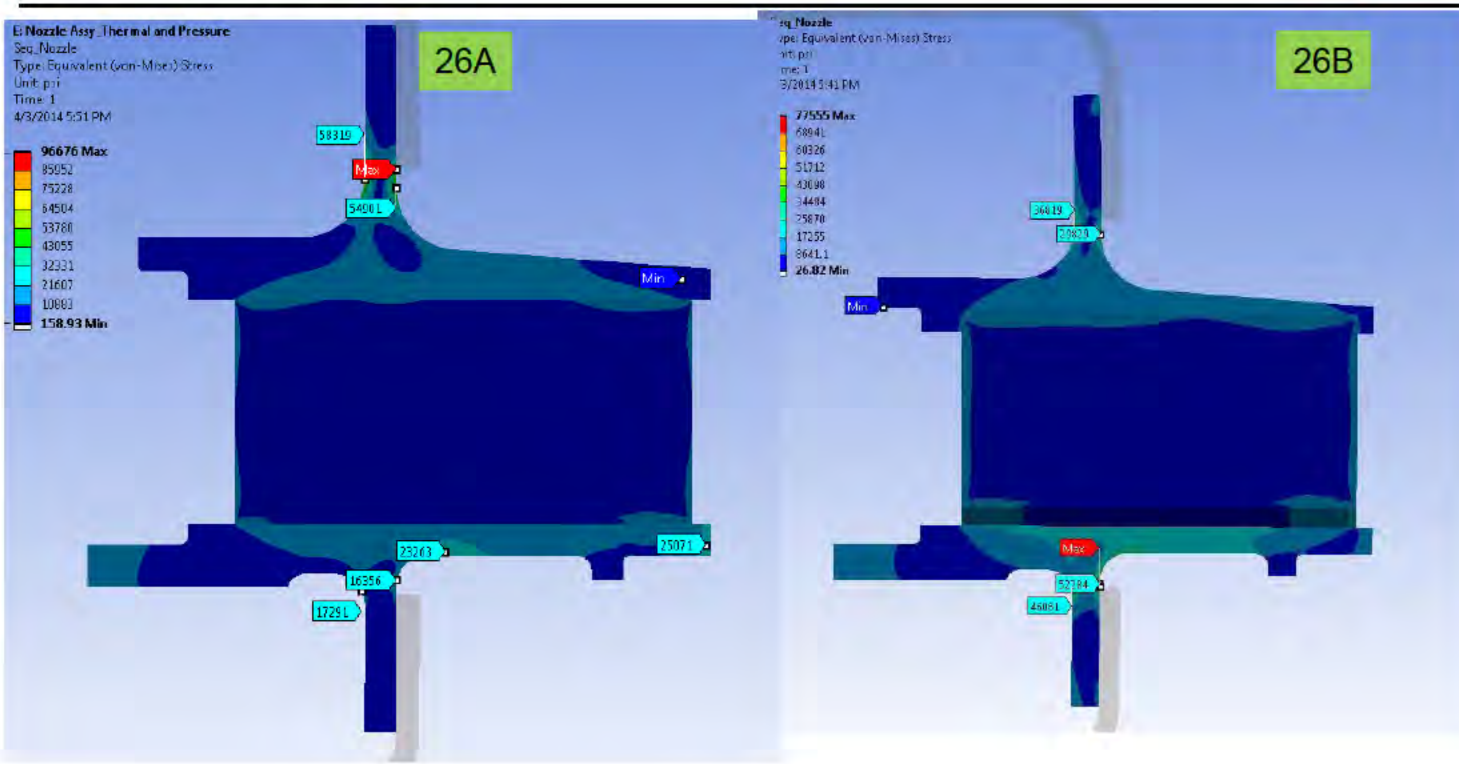


○ 26A Load Case

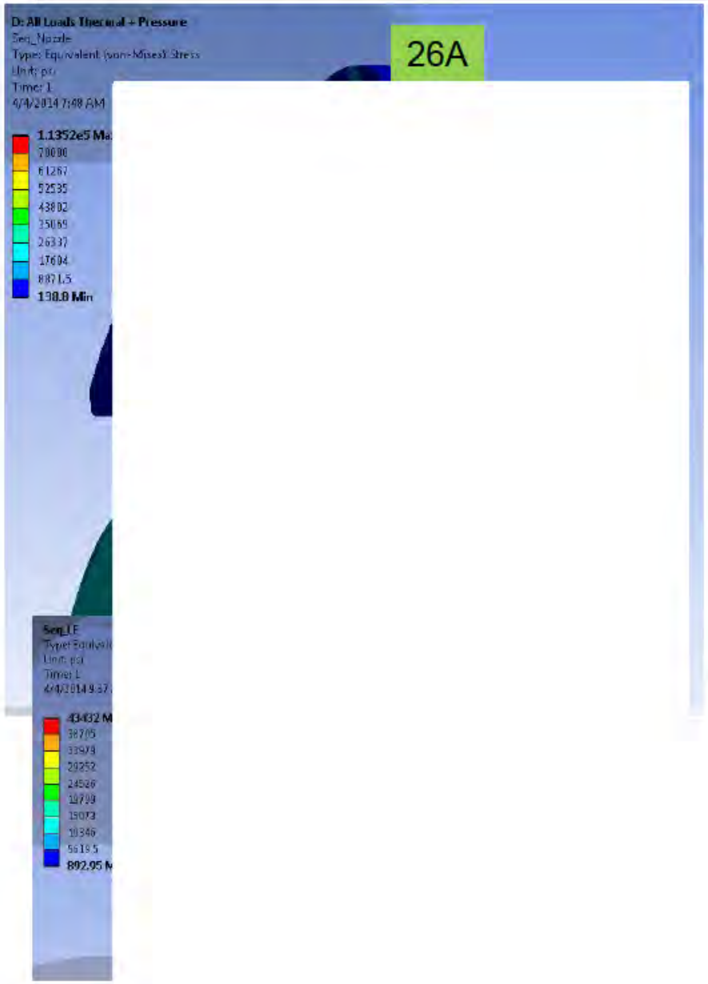
○ 26B Load Case

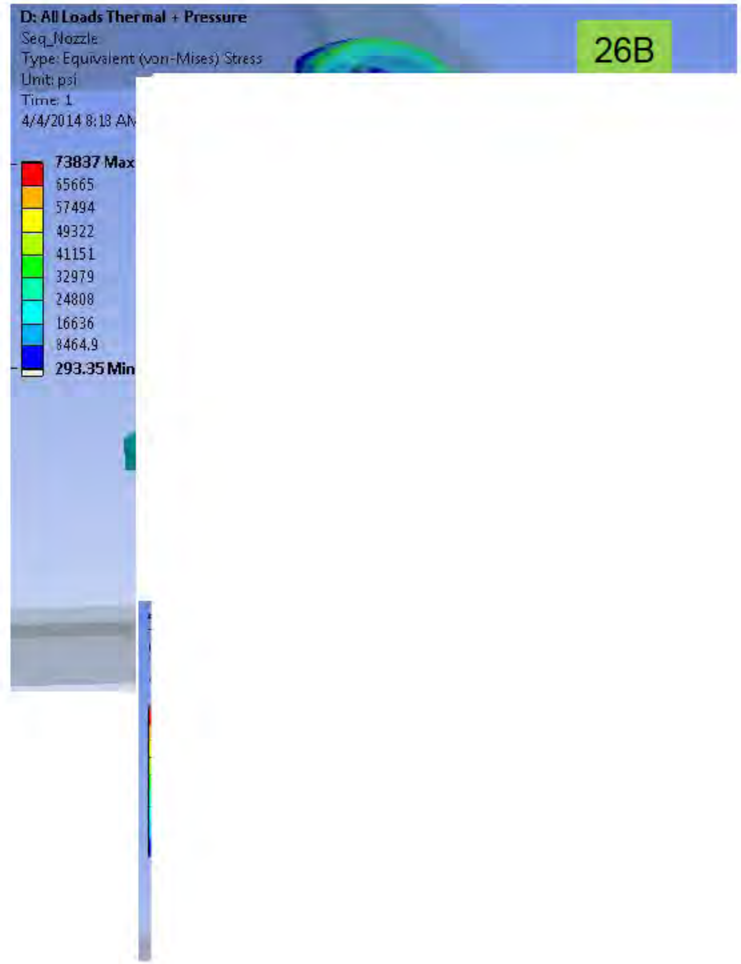
Location X for 26A exceeds ultimate strength

2D Nozzle FEA



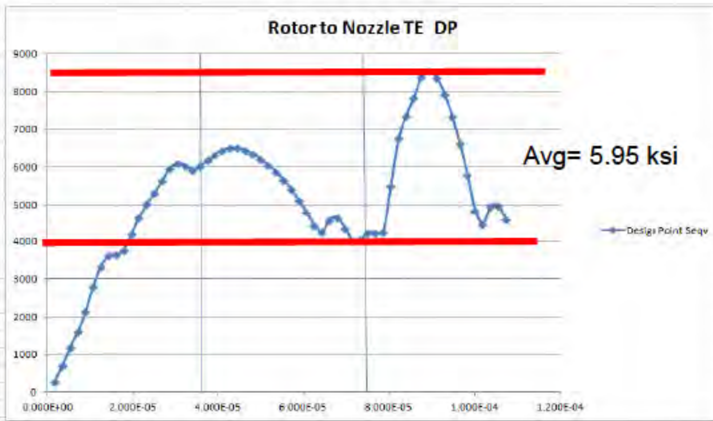
3D Nozzle FEA





3D FEA: Flanges 1600F+Pressure

HCF:Transient Results

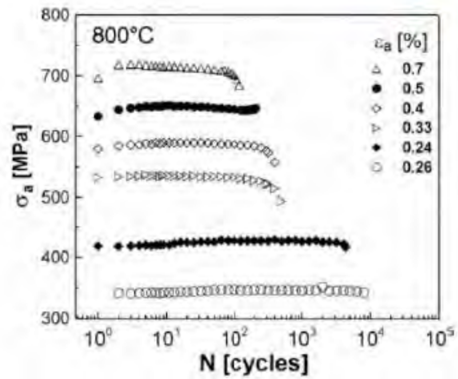


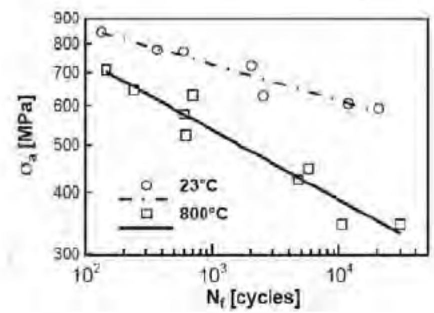
SEQV: 23,000 RPM w/ 1600°F, 84% of YS

SEQV: 23,000 RPM w/ 1100°F, 80% of YS

- TE Fillet stress can Vary from 72 ksi with Radial definition
- To 36 ksi with scallop definition

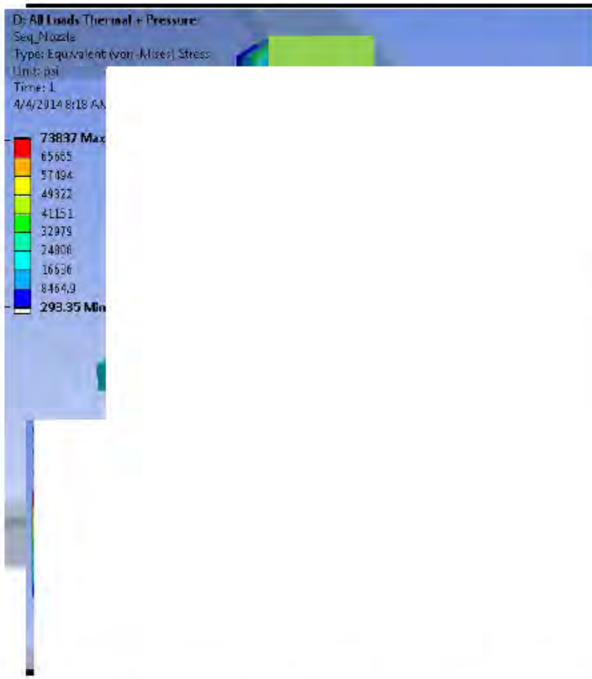
incomplete

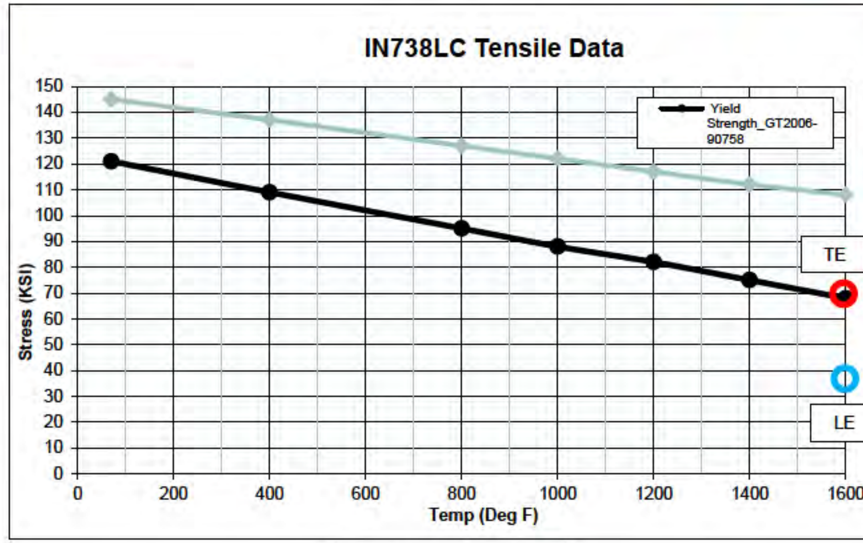




IN738LC Fatigue Data from:
Inconel738LC Fatigue_302_Tobias_J-FT pdf
(L:\ISCE\ISCE Build 2\Turboexpander\Mech\Materials and documents\Materials)

Creep Rupture: 1600F

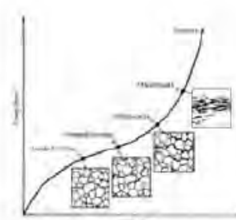




SEQV: 1600°F,50 % of YS

SEQV: 1600F, 100% of YS
Scallop is 50% of YS
Hub Heat Shield optional

incomplete



Test Data

Sundstrand Turbomach

ESR 0151
RSV B

3.6.1.1 Cast-to-Size Bar Minimum Tensile Properties

Test Temperature	Room	1200°F
Yield Strength (0.2%)	108 ksi	91 ksi
Ultimate Tensile Strength	125 ksi	125 ksi
Elongation in 4D	4.0%	4.0%
Reduction of Area	5.0%	5.0%

3.6.1.2 Cast-to-Size Bar Minimum Stress-Rupture Properties

Test Temperature	1400°F	1700°F
Stress	88 ksi	33 ksi
Life	30 hours	30 hours
Elongation in 4D	4.0%	5.0%
Reduction of Area	5.0%	5.0%

Tensile testing shall be conducted in accordance with ASTM E8 and E21, and stress-rupture testing shall be conducted in accordance with ASTM E139.

Creep Rupture: 1600F

	Inputs	
	Temp (deg F)	IN 738 LC
Comments		Candidate for cast nozzle vane
Room Temp Ultimate Strength (KSI)		150
Room Yield Strength (KSI)		130
Room Temp Percent Strain at Failure		7
Young's Modulus (10^3 KSI), Room Temp		29.1
Density (lbm/in^3)		0.293
Temp for good oxidation resistance (deg F)		1800
Typical temp limit (deg F)		1800
CTE at 650 deg F (in/in/deg F)		7.30E-06
CTE at 1100 deg F (in/in/deg F)		7.92E-06
CTE at 1600 deg F (in/in/deg F)		8.60E-06
Thermal conductivity (Btu/[(hr(ft^2)(F)/ft] at 650 deg F		8.2
Thermal conductivity (W/mK)] at 650 deg F		14.2
Weldability		Poor
Yield Strength Derating at 650 deg F		98.00%
Yield strength at 650 deg F (KSI)		127.4
Temp for steep decline in strength (deg F)		1400
Larson-Millar Parameter at 40 KSI		44.2
Rupture Life (hrs)	1000	18791982647
Rupture Life (hrs)	1100	215443469
Rupture Life (hrs)	1200	4231614
Rupture Life (hrs)	1300	129908
Rupture Life (hrs)	1400	5800
Rupture Life (hrs)	1500	356
Rupture Life (hrs)	1600	29
Rupture Life (hrs)	1700	3
Rupture Life (hrs)	1800	0
Rupture Life (hrs)	1900	0
Larson-Millar Parameter at 20 KSI		50
Rupture Life (hrs)	1000	176431181504442
Rupture Life (hrs)	1100	1125335582601
Rupture Life (hrs)	1200	13197203931
Rupture Life (hrs)	1300	256502091
Rupture Life (hrs)	1400	7615806
Rupture Life (hrs)	1500	323746
Rupture Life (hrs)	1600	18700
Rupture Life (hrs)	1700	1407
Rupture Life (hrs)	1800	133
Rupture Life (hrs)	1900	15
Larson-Millar Parameter at 10 KSI		53
Rupture Life (hrs)	1000	20015657595671600
Rupture Life (hrs)	1100	94266845511788
Rupture Life (hrs)	1200	846663511319
Rupture Life (hrs)	1300	12990813969
Rupture Life (hrs)	1400	312337159
Rupture Life (hrs)	1500	10985411
Rupture Life (hrs)	1600	534756
Rupture Life (hrs)	1700	34438
Rupture Life (hrs)	1800	2827
Rupture Life (hrs)	1900	287

incomplete

40 ksi

20 ksi

Appendix 11.2.4

Turboexpander FDR

10:1 Turboexpander Final Design Review: Turbine Rotor

Kyle Badeau

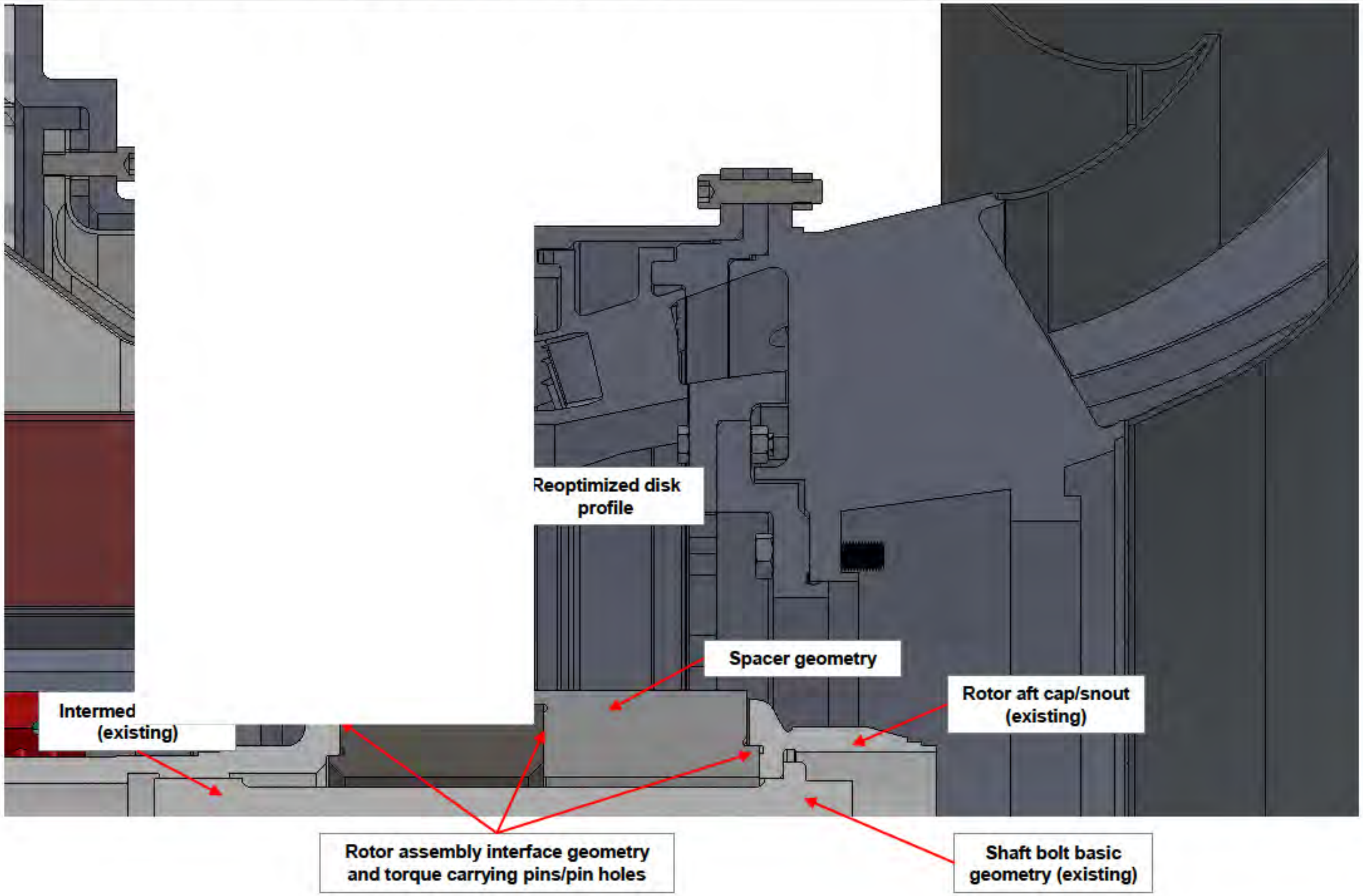
March 3rd, 2014

FDR Agenda

1. **Current Blade 018**
2. **Layout/Assembly/Manufacturing**
3. **Material Selection**
4. **Heat Transfer**
5. **Blisk Structural**
6. **Blisk Modal**
7. **LCF/HCF/Creep**
8. **Growths & Clearances**

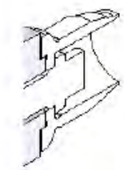
		Analysis		
		1100F/23,000RPM	1600F/23,000RPM	1600F/27,000RPM
Design Criteria	Heat Transfer	X	Uses 27,00 RPM	X
	Disk Burst			X
	Blade P/A			X
	Peak Stress	X	X	X
	LCF	X	X	
	HCF	X	X	
	Crack Propagation	TBD	TBD	
	Creep	X	X	
	Growths/Clearances	X	X	
	Hot to Cold	TBD		
	Modal Analysis	X	X	X
	Rotor Assembly	TBD	TBD	

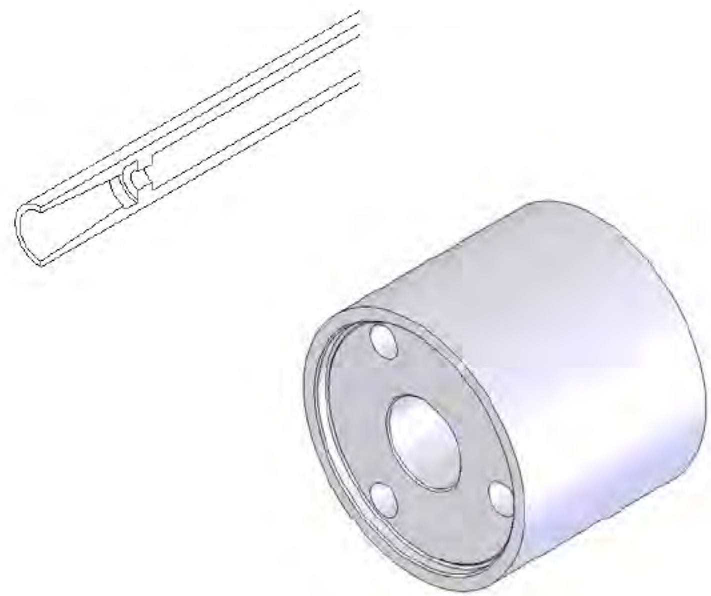
FDR Rotor Layout (2/25/14)



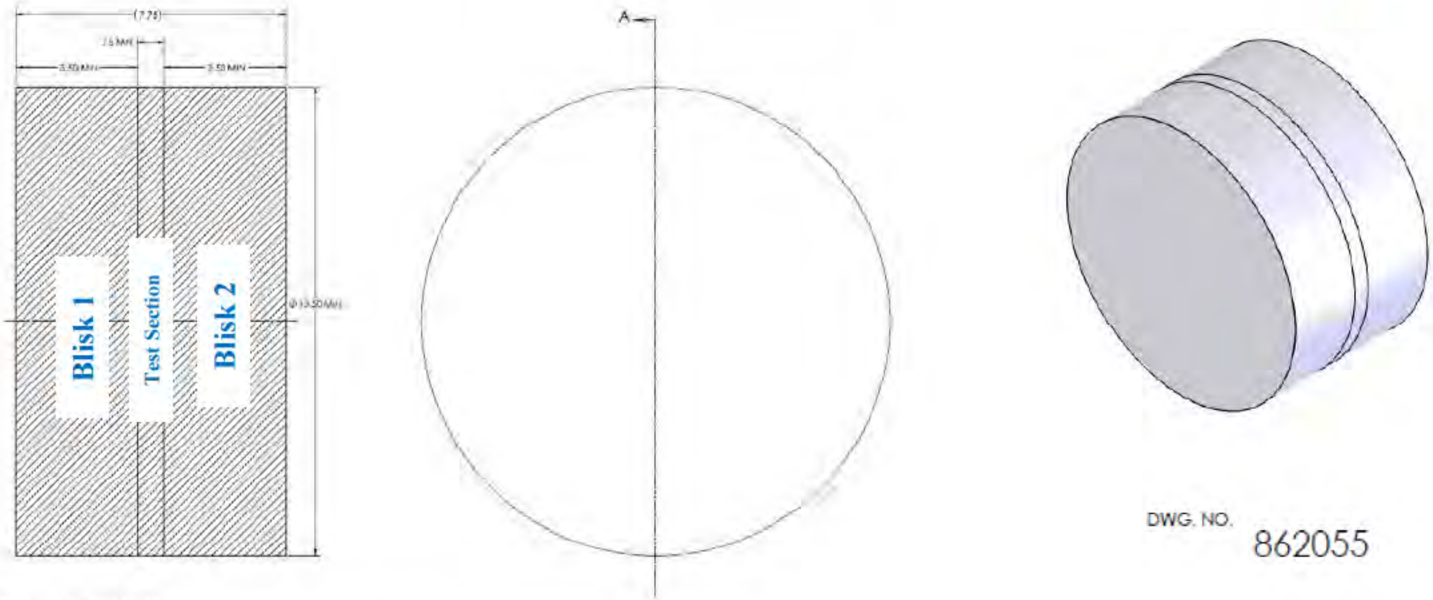
Rotor Assembly

10:1 turbine blisk and spacer components assemble with reuse of Solar intermediate shaft, shaft aft cap, and shaft bolt





Synertech HIP'ed Billet



1. DRAFTING BASED ON:
   MODEL: 862055 ISCE Build2 10-1 Turboexpander Blisk Raw Material Envelope
   PART#: 862055
   REVISION: -.008

Drawing & PO Released

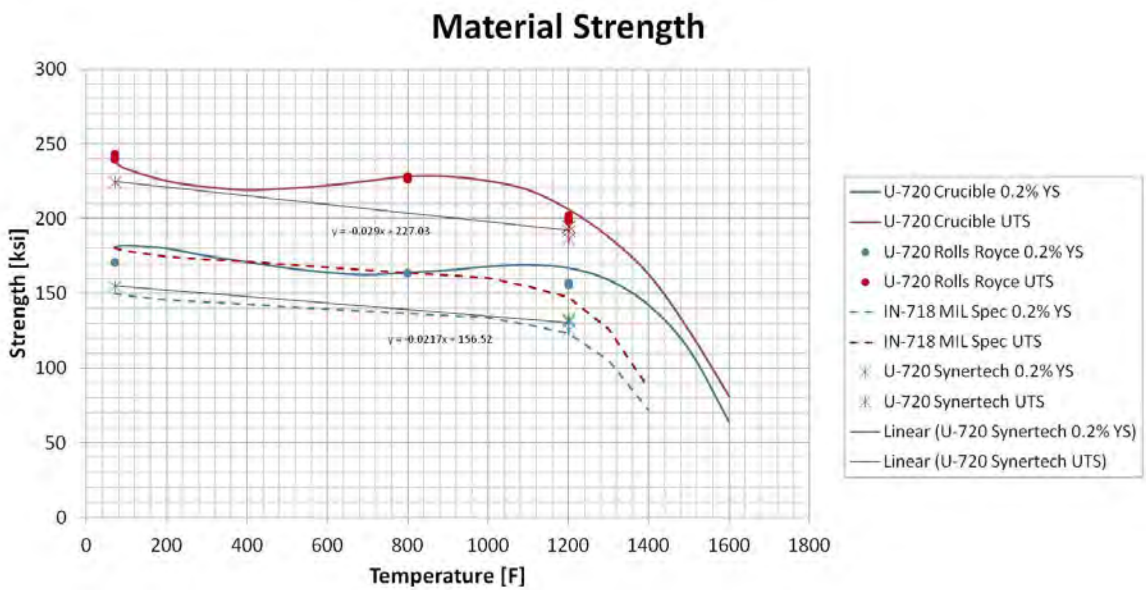
2. MATERIAL: U-720 POWDER (-140 MESH SIZE)
   HIP FOR MINIMUM ROOM TEMPERATURE PROPERTIES OF 190 KSI UTS, 145 KSI 0.2% YS,
   14% ELONGATION, 14% REDUCTION OF AREA, AND MINIMUM 1200°F STRESS RUPTURE LIFE AT 100 KSI OF 100 HOURS.

3. POWDER MESH ANALYSIS METHOD PER ASTM B214

4. ULTRASONIC INSPECTION PER AMS-2154, MAX. FLAW SIZE ALLOWABLE IS CLASS AAA.
   ANY FLAWS DETECTED SHOULD BE COMMUNICATED TO RAMGEN AS THE PART MAY BE ACCEPTABLE IF DISCREPANT FLAW IS IN A REGION THAT IS TO BE MACHINED OFF.

5. PERMANENTLY IDENTIFY WITH PART NUMBER AND SERIAL NUMBER OF END FACE OF EACH CYLINDER

Material Strength: U-720 vs. IN718



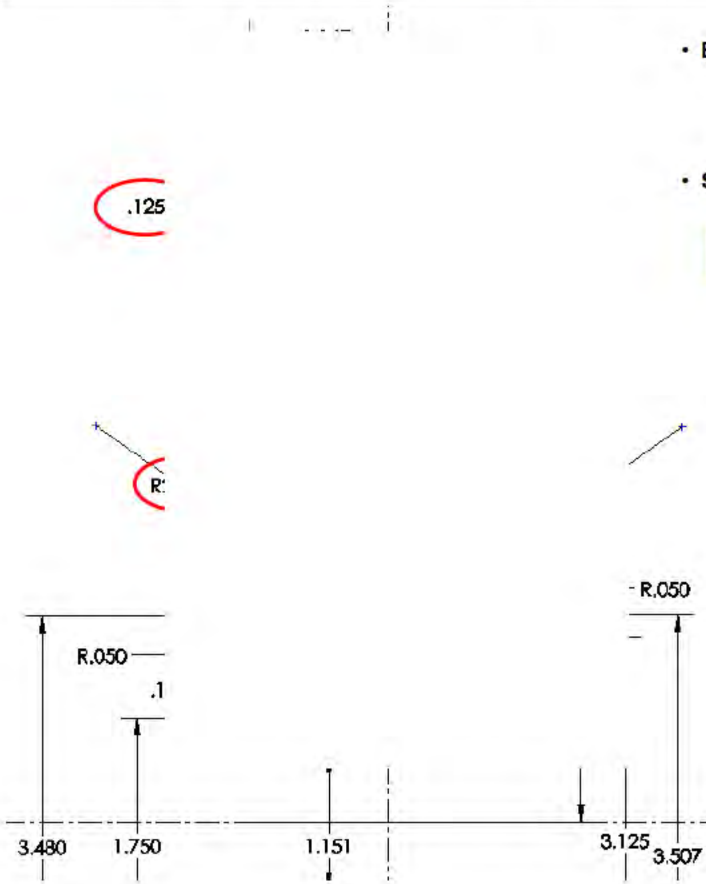
- **U-720 shows superior strength to IN-718 at all temperatures (also shows better LCF and Creep capabilities)**
- **All U-720 data shown is for P/M HIP'ed and subsolvus processed**
- **Synertech tested U-720 strength lower than source curves (Crucible and Rolls Royce)**
  - **Linear curve fit (shown) used to interpolate for U-720 strength criteria**
- **Synertech minimum RT strength: 190 ksi UTS & 145 ksi 0.2% YS (lower than available Synertech tested data)**
- **All structural analysis conducted with IN-718 elastic modulus (U-720 elastic modulus unknown, but will be obtained from batch tensile testing)**

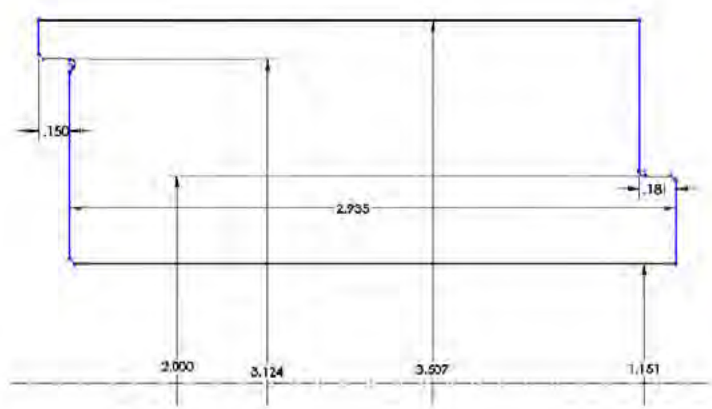
HIP'ed U-720 selected as material over Forged IN-718 for Improved Strength/Temperature Capability

Disk and Spacer Profile for Blade 018

- Blade 018 disk profile optimized for
    1. Minimum bore hoop stress
    2. Minimum web fillet radial stress
    3. Maximum burst stress
- Six variable optimization (circled)

Optimized Disk Profile





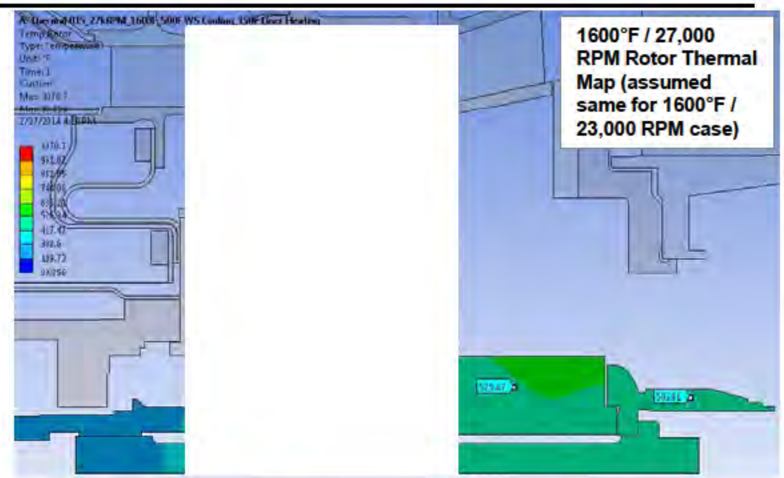
Rotor SS Heat Transfer Analysis

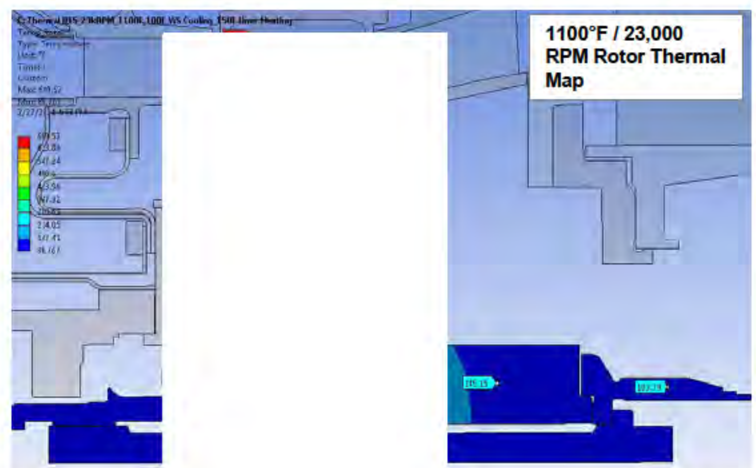
MODEL

- All engine rotor components modeled
- Turbine rotor components modeled as IN718
- Compressor rotor components modeled as SS410
- Blade modeled with new element type (152) for improved HTC accuracy
- Rotor shaft contact surfaces modeled with tight contact thermal conductivity due to bolt compression
- Remaining gaps modeled with thermal conductivity per rules of thumb, based on gap size
- Fluid elements across turbine blade
  1. 1075°F: Ttrel=685F, 8.15 lbm/s
  2. 1600°F: Ttrel=1080F, 7.0 lbm/s
- Fluid pumping elements along upstream and downstream rotor faces
  - 100°F for 1075°F case (at inlet)
  - 500°F for 1600°F case (at inlet)
  - 0.08 lbm/s (each side)

RESULTS

- Thermal gradient generates compressive stress at rim, tensile stress at bore
- 500°F wheel space cooling air needed for 1600°F case to reduce T_{bulk} to T_{max} difference
- New blade element type increases HTC making blade hotter than previous results
- Applying wheel space purge flow at actual inlet locations results in 150°F temperature drop by the time flow enters actual wheel space for 1600°F case
- Disk rim compressive thermal stress higher





Disk Structural: 27,000 RPM, Speed Only

- Max Von Mises stress @ bore 88% of U-720 RT yield strength
- Thermal loading not included (next slide)
  1. Bore surface stress will increase
  2. Reduced strength at temperature
- With material utilization factor of 0.83, 46% O.S. burst margin is achieved (acceptable)
  - Burst calculation uses 1200°F UTS of 187 ksi
- AVG hoop stress in disk ~73ksi…39,347 RPM burst speed (71% O.S. to 23,000 RPM)

Disk Burst Speed Margin Acceptable

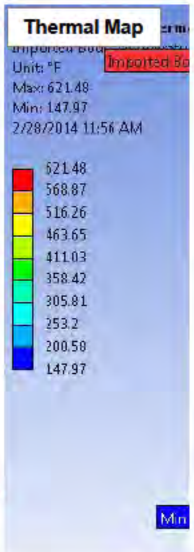
Disk Structural: 27,000 RPM, 1600°F

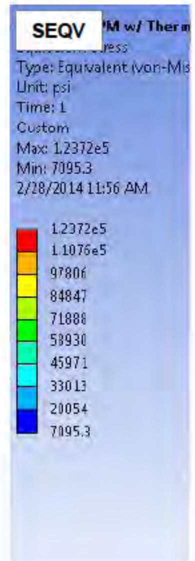
- Max disk temperature from 1600°F / 27,000 RPM thermal map ~1000°F at rim
  - 350 to 500°F bore surface temperature
- Maximum Von Mises stress percentage of yield strength is 114% @ bore
  - All nodal temperature/stress combinations evaluated
- Bore stress exceeds material strength at corresponding temperature, operation at 27,000 RPM AND 1600°F firing temperature not acceptable
  - Acceptance could change based on Ramgen batch of U-720 strength
  - If speed turned down to 23,000 RPM with same thermal map, maximum Von Mises stress percentage of yield strength reduces to 89%
- With thermal stress included, burst speed increases to ~40,500 RPM (67% O.S.)
  - Burst speed is an average hoop stress based calculation
  - Thermal loading puts outer radial region of disk into hoop compression

Disk Burst Speed Margin Acceptable,
Bore Hoop Stress Exceeds Material Capability

Disk Structural: 23,000 RPM, 1100°F

SPEED & THERMAL

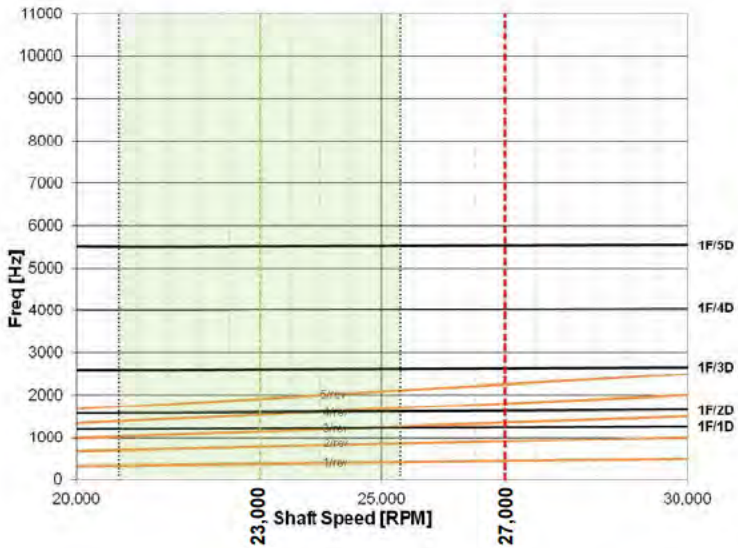




- Max disk temperature from 1100°F / 23,000 RPM thermal map ~620°F at rim
  - ~150°F bore surface temperature
- Maximum Von Mises stress percentage of yield strength is 81% @ bore
  - All nodal temperature/stress combinations evaluated
- 23,000 RPM AND 1100°F firing temperature acceptable

Bore Hoop Stress Within Material Capability

Modal Analysis: 1100°F

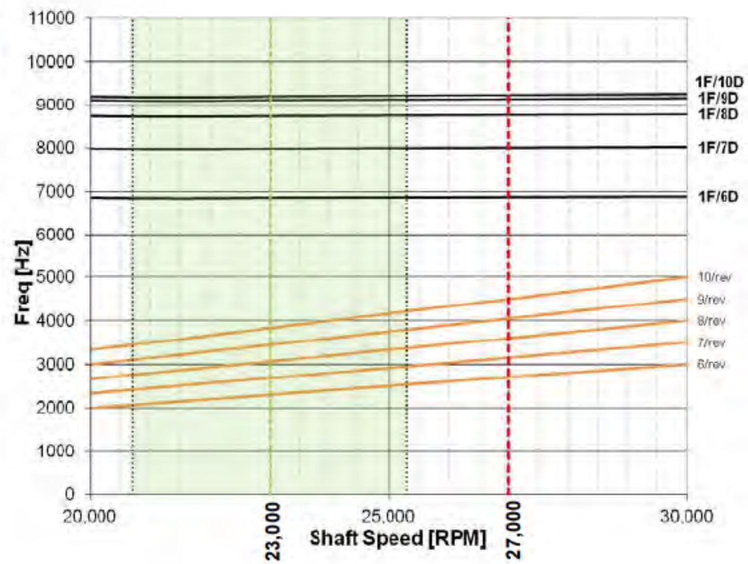


No resonance crossing (intersection between ND and /rev line at or near running speed) below 10/rev

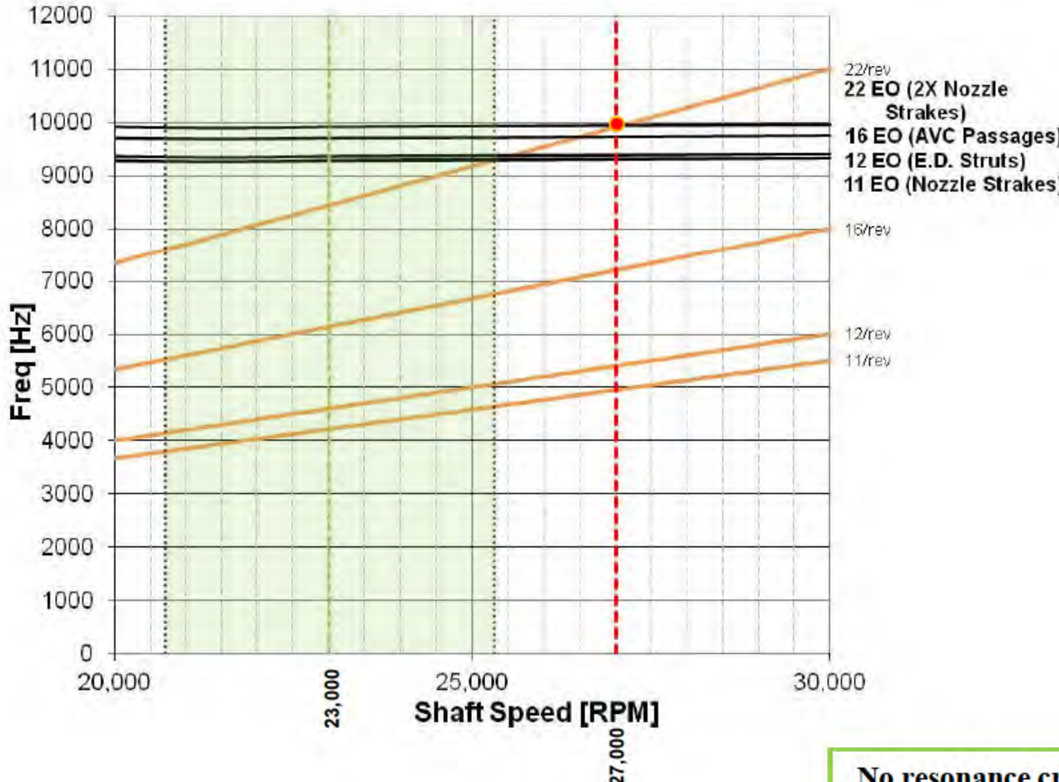
Four modal analysis conducted to form nodal diameter lines through speed range of interest

1. **70°F, 0RPM**
2. **1100°F, 20,700 RPM (-10%)**
3. **1100°F, 23,000 RPM (design speed)**
4. **1100°F, 25,300 RPM (+10%)**

- **27,000 RPM frequency margin assessed with 1600°F cases in subsequent slides**

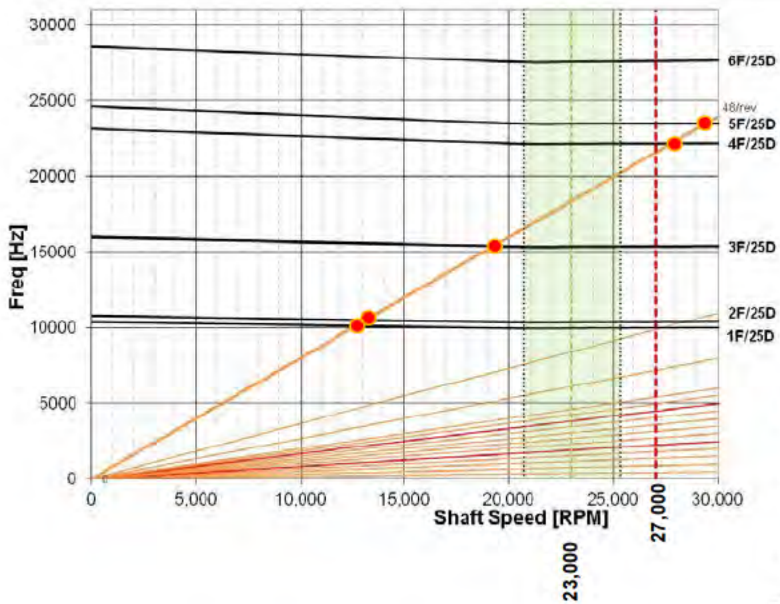


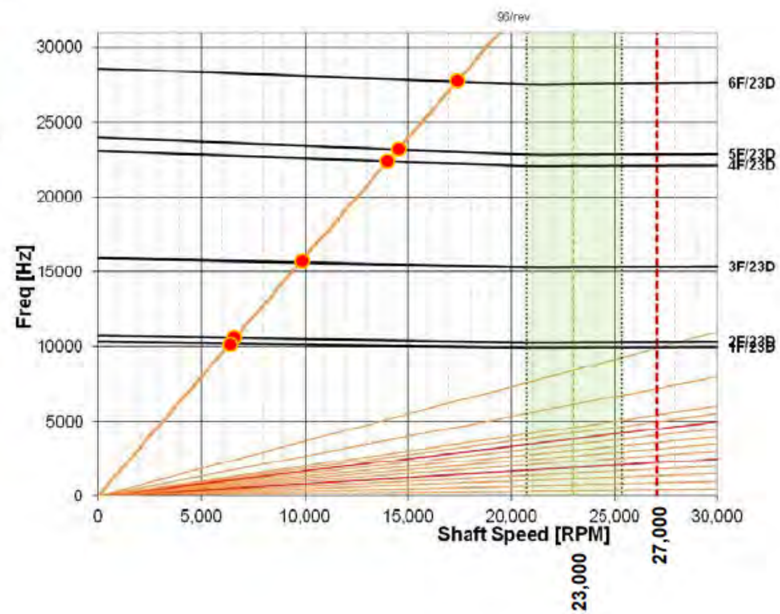
Modal Analysis: 1100°F



No resonance crossing to be excited by 12, 14, 16, or 22/rev stimulus, within +/- 10% speed range of 23,000 RPM

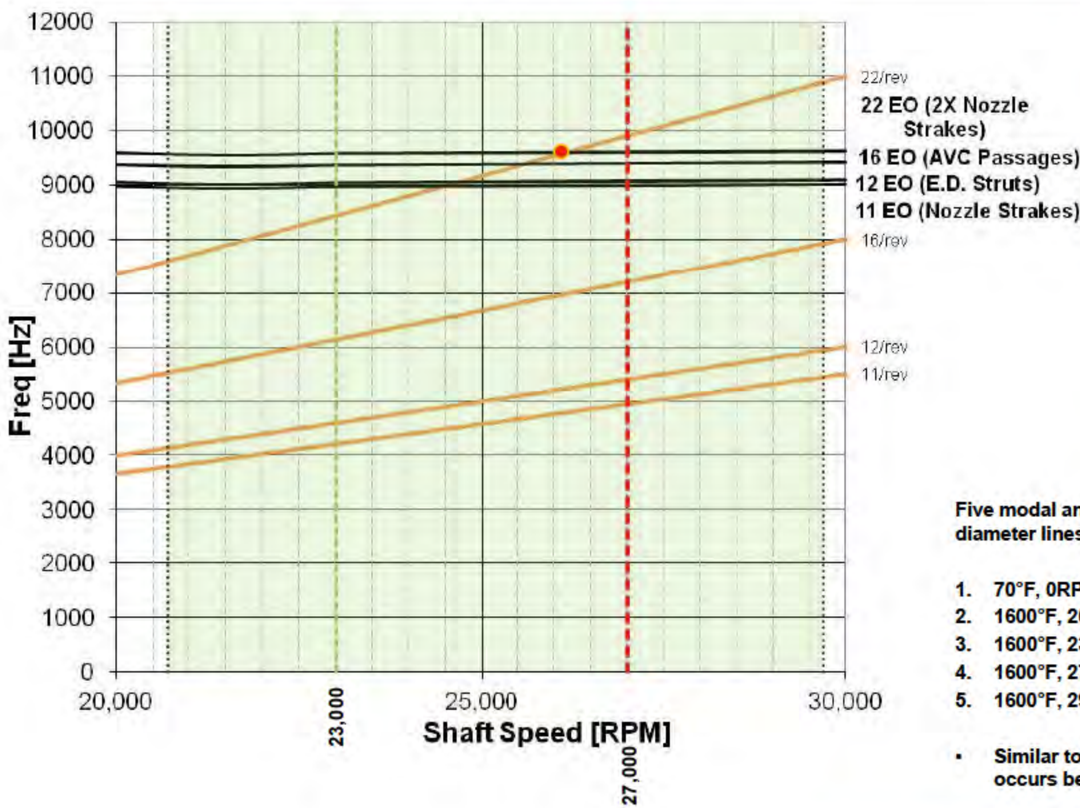
Modal Analysis: 1100°F





No resonance crossing to be excited by 48/rev OGV or 96/rev 2X OGV stimulus, within +/- 10% speed range of 23,000 RPM

Modal Analysis: 1600°F



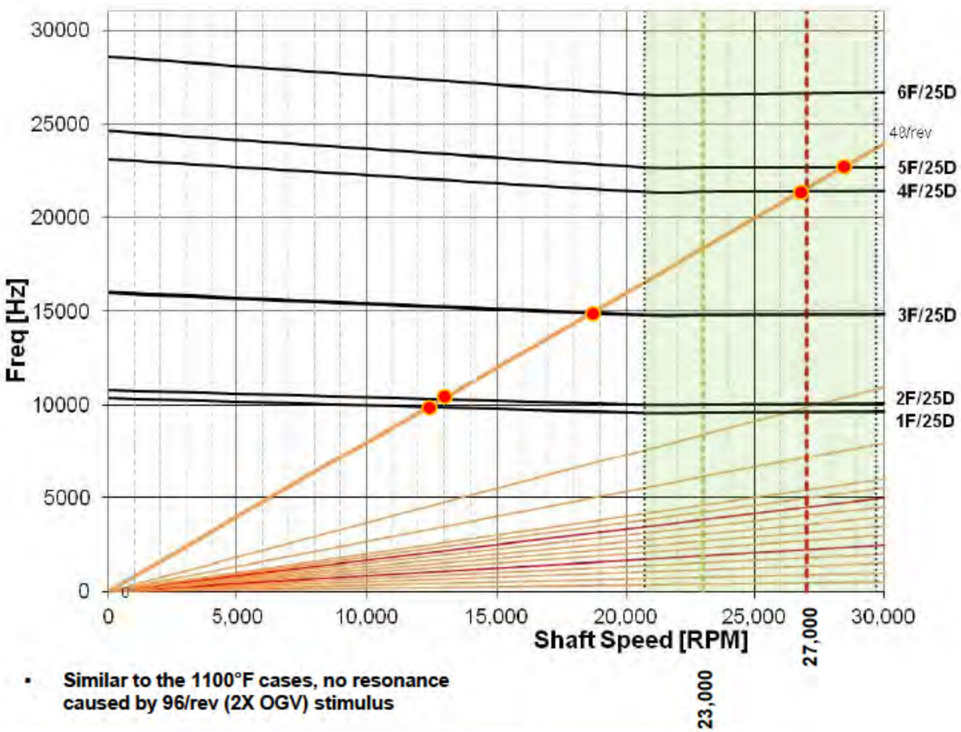
Five modal analysis conducted to form nodal diameter lines through speed range of interest

1. **70°F, 0RPM**
2. **1600°F, 20,700 RPM (-10%)**
3. **1600°F, 23,000 RPM (design speed 1)**
4. **1600°F, 27,000 RPM (design speed 2)**
5. **1600°F, 29,700 RPM (+10%)**

- **Similar to the 1100°F cases, no resonance occurs below 10/rev**

2X nozzle strake (M1 22/rev) stimulus occurs at ~26,250 RPM (2.8% margin)

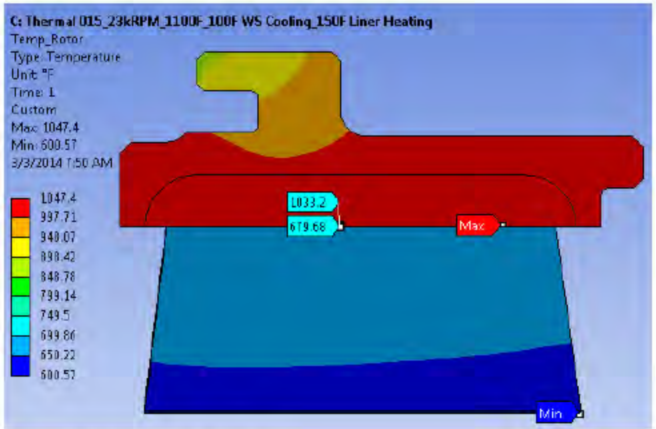
Modal Analysis: 1600°F



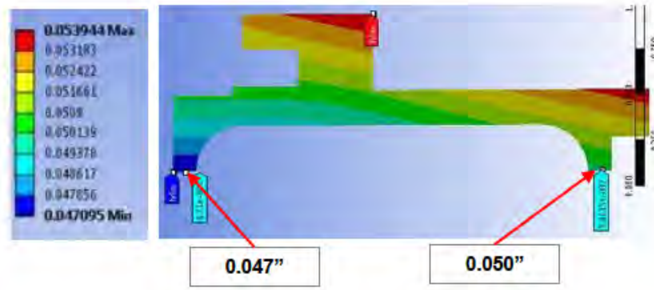
- Similar to the 1100°F cases, no resonance caused by 96/rev (2X OGV) stimulus

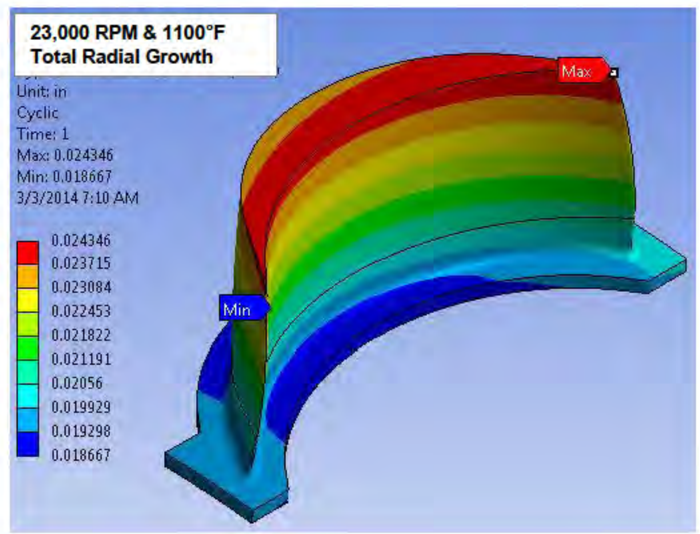
1. OGV (M4 25ND) stimulus occurs at ~26,900 RPM (0% margin)
2. OVG (M5 25ND) stimulus occurs at ~ (2.2% margin)

Blade to [redacted] Radial Clearance 1100°F



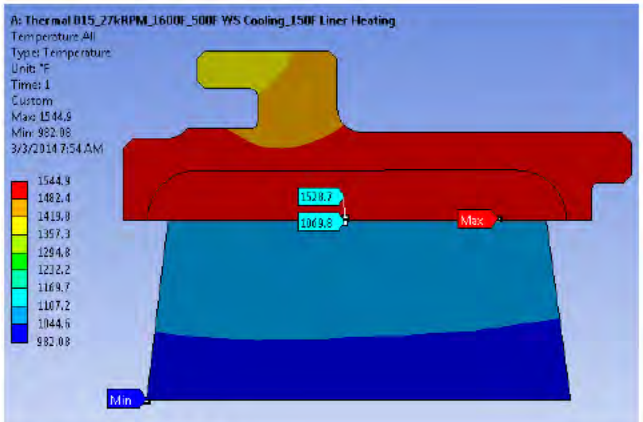
- 10 mil initial gap



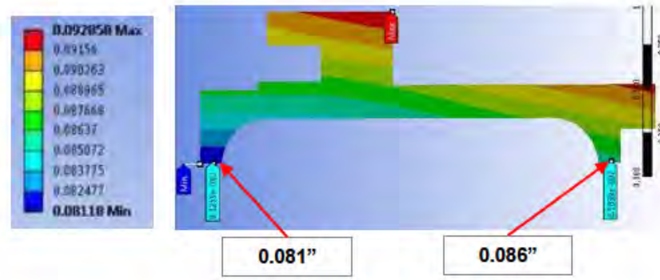


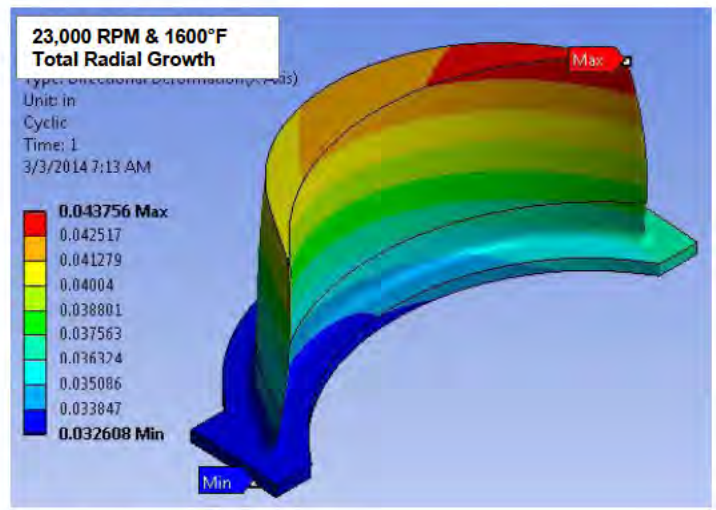
1100°F conditions lead to 353°F blade to [redacted] temperature delta and 26 mil gap

Blade to [redacted] Radial Clearance 1600°F



- **10 mil initial gap**





1600°F conditions lead to 459°F blade to [redacted] temperature delta and 42 mil gap

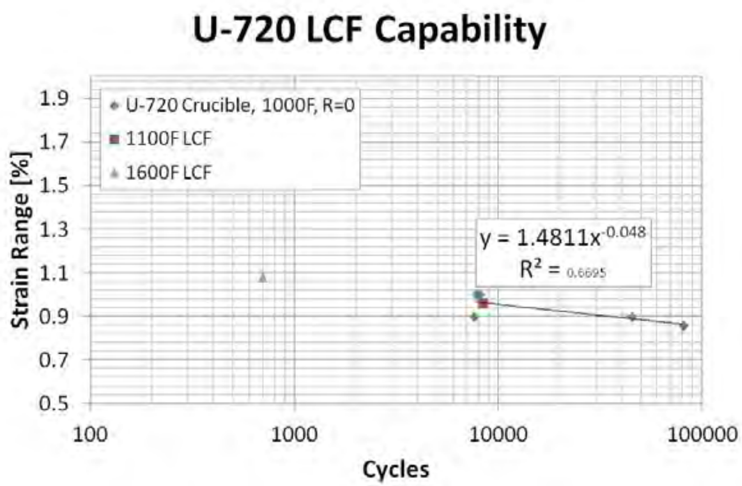
Rotor Assembly Analysis

Rotor Axial Growth

LCF: 23,000 RPM

- At S.S. conditions, when centrifugal and thermal loading coincide, maximum equivalent stress remains within yield strength for both 1100°F and 1600°F cases (at 23,000 RPM)
- If at any point during startup/shutdown cycle the centrifugal and thermal loading are out of phase (probable), 0 RPM/70°F and S.S. conditions may no longer be the minimum LCF life defining time points
- Highest accuracy LCF life calculation evaluates the strain range at all nodes across all time point combinations with mission mix (we do not have transient analyses)
- Estimated LCF life calculated by assuming thermal only and centrifugal only loading occur at different times, used for equivalent strain range calculation
- Both stress ranges less than twice the 0.2% yield strength, indicating that stress/strain behaves linearly, and psuedo-stress method is applicable
- Only R=0 (zero to max) loading LCF curve available, conservative due to mean stress effects (R=-1, fully reversed LCF curve more applicable here)

1. 1100°F strain range of 0.96%...8,379 cycles (within LCF data)
2. 1600°F strain range of 1.082%...693 cycles (extrapolated)



Blisk LCF estimates acceptable for 1100°F conditions, unacceptable for 1600°F conditions

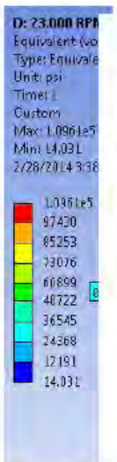
<u>Fundamentals of Metal Fatigue Analysis by Bannantine, Comer, Handrock</u>

Equivalent strain range criterion. In the LCF regime, an equation similar to Eq. (7.13) has been developed in terms of strain and is used in an ASME code procedure [12]. This criterion requires the calculation of an equivalent strain range, $\Delta\epsilon_{eq}$.

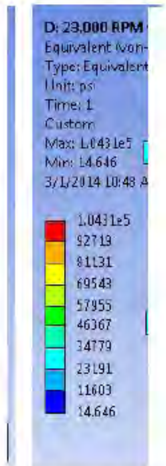
$$\Delta\epsilon_{eq} = \text{value of } \left\{ \frac{\sqrt{2}}{3} [(\Delta\epsilon_{11} - \Delta\epsilon_{22})^2 + (\Delta\epsilon_{22} - \Delta\epsilon_{33})^2 + (\Delta\epsilon_{33} - \Delta\epsilon_{11})^2 + 6(\Delta\epsilon_{12}^2 + \Delta\epsilon_{23}^2 + \Delta\epsilon_{31}^2)]^{1/2} \right\} \text{ maximized with respect to time} \tag{7.14}$$

where $\Delta\epsilon_{ij} = \epsilon_{ij}(t_1) - \epsilon_{ij}(t_2)$ are strain differences
$\epsilon_{ij}(t_1)$ = components of the strain tensor at time t_1
$\epsilon_{ij}(t_2)$ = components of the strain tensor at time t_2

HCF:



SEQV: 23,000 RPM w/ 1600°F, 84% of YS



SEQV: 23,000 RPM w/ 1100°F, 80% of YS

- Max vane root fillet stress
    1. 110 ksi for 1600°F: minimal HCF capability
    2. 104 ksi for 1100°F: +/-19 ksi dynamic stress capability for 10 hours
- NLH data to be assessed
- Need to locate U-720 HCF curves

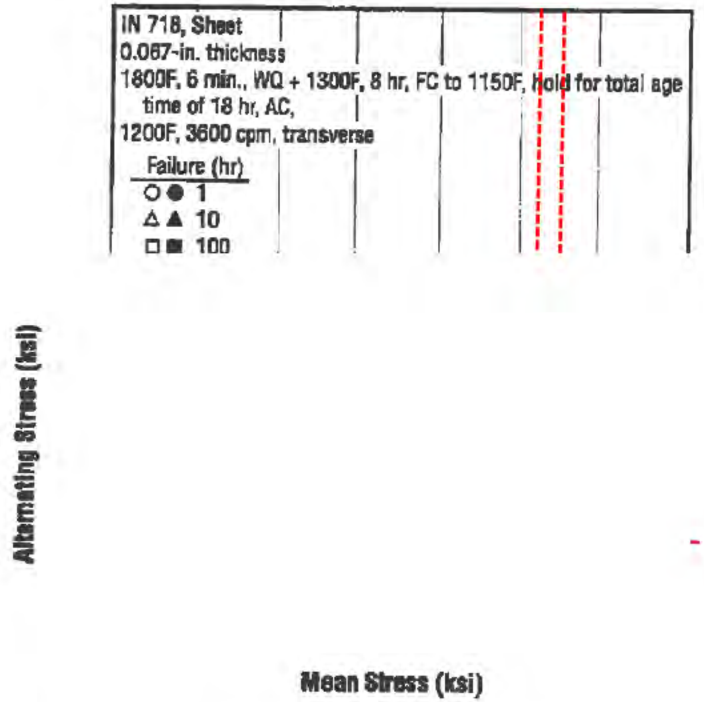
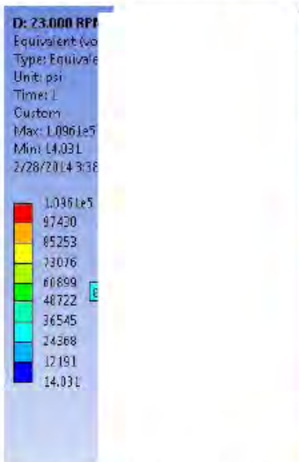
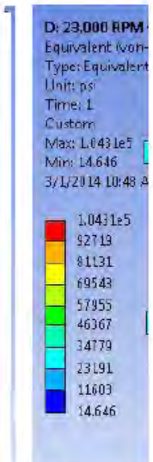


Fig. 3.5.1.11 Stress-range diagram for notched and smooth sheet at 1200F (Ref. 42)

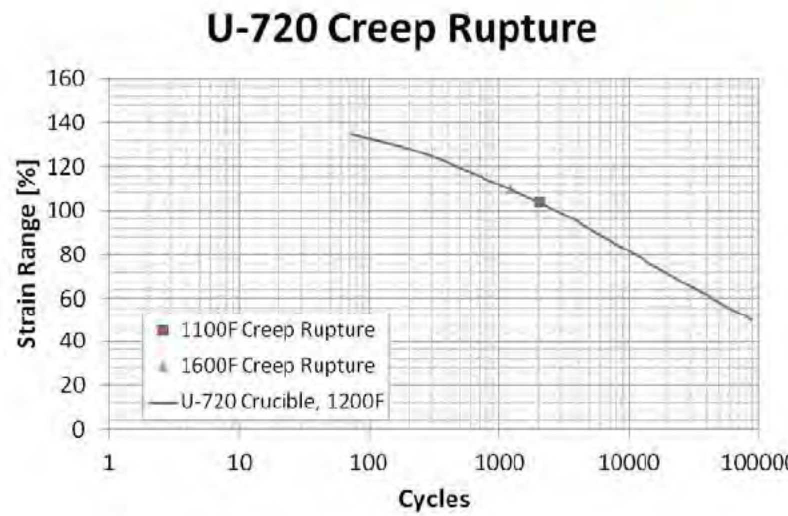
Creep Rupture: RPM



SEQV: 23,000 RPM w/ 1600°F, 84% of YS



SEQV: 23,000 RPM w/ 1100°F, 80% of YS



Synertech Test Data

*** STRESS RUPTURE TEST RESULTS

TEMP	dia. (in.)	AREA (in.²)	STRESS (ksi)	TIME (hrs)
1200F	.2500	.0491	130.0	**85.1**
1200F	.2490	.0487	130.0	100.0
REQUIREMENTS				
1200F			130.0	100.0

1. **1100°F creep rupture: 2,000 hours**
2. **1600°F creep rupture: 1,200 hours**

APPENDIX 12.1

ISCE Build 1 DOE Presentation

ISCE Program

Task 4.4
ISC Engine Build 1

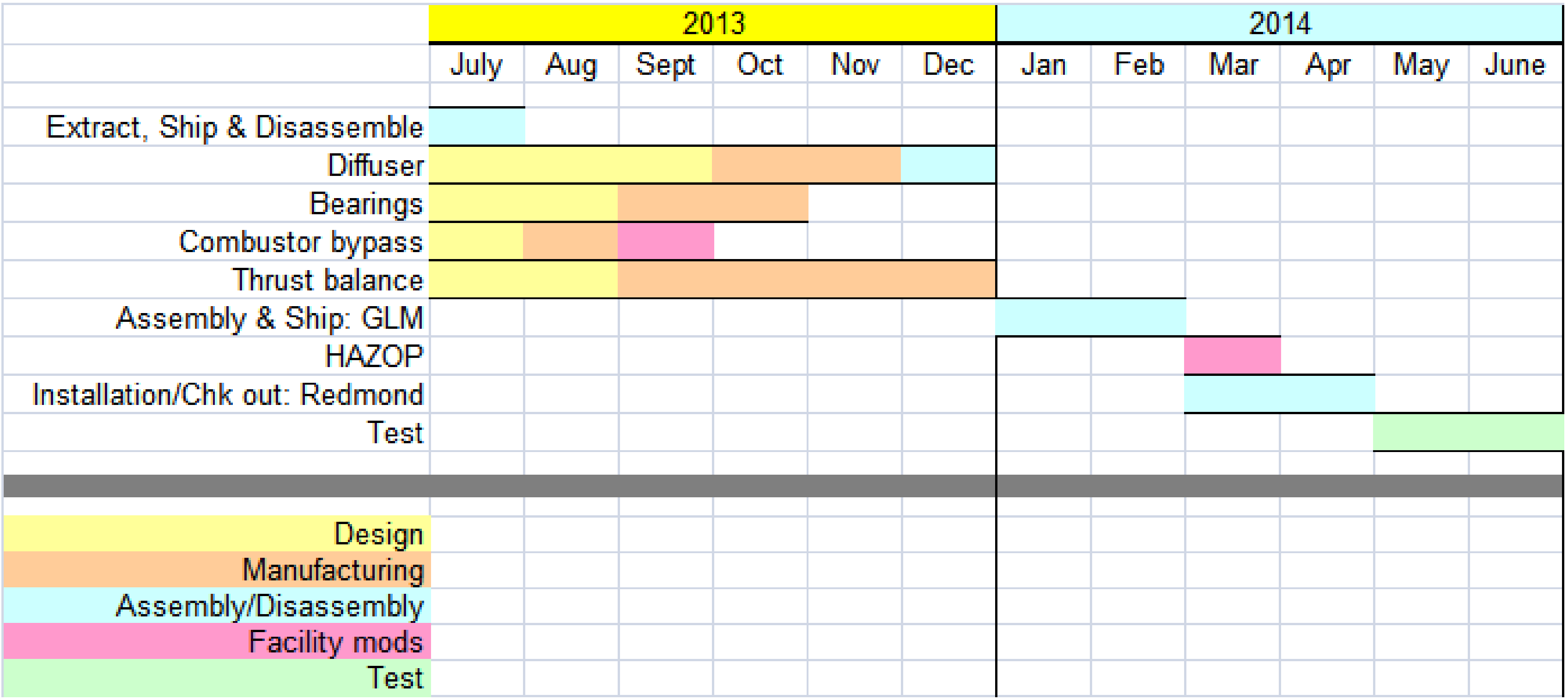
August 30, 2013

ISCE B1 Issues to Overcome

- Diffuser over-contraction during starting
- Insufficient pre-throat ramp & shroud bleed
- Insufficient bearing damping
- Excessive diffuser back-pressure
- Insufficient thrust balance system flexibility
- Insufficient motor power

6/25

Estimated B1 Redesign/Rebuild Schedule



Schedule shows test May ’14,
but may be overly aggressive

Conclusions

- ISCE B1 was conceived as a fast / easy proof of concept demonstrator - the complexities of transient behavior and starting requirements for the integrated engine were not fully appreciated during design which resulted in systems being operated beyond their design envelope
- Although the ISCE B1 could be rebuilt in an attempt to reach its original objectives, the cost, schedule, and technical risk are significant and it is not recommended
  - Re-engineering a 50+ year-old design doesn't appear to be the best path
- Independent mapping of the compressor and turbine systems should be performed prior to any follow on integrated engine activity

ISC Engine Comparison Of Test Data And CFD Simulations

Bellevue, August 30, 2013

CFD Data Extraction Process

- The supersonic probe was incorporated in the test article CAD model and its tip location tracked as function of the insertion depth
- CFD model did not include the probe
- The flow quantities reported in the next slides where then extracted from the three-dimensional CFD solution on the line described by the probe tip
  - At any insertion depth, in the plots reported in the next few slides this profile is referred to as CFD – 20,650 RPM
- To account for the high degree of three dimensionality of the flowfield and the finite size of the probe tip itself, for any given flow quantity the maximum and minimum values within a square region 0.2in x 0.2in in size around the probe tip location are reported as well
  - At any insertion depth, in the plots reported in the next few slides these profiles are referred to as CFD Min and CFD Max, respectively

Conclusions

- **CFD simulations predict a partially started system and are therefore capable of qualitatively predicting the overall flowfield behavior**
- **CFD underpredicts corrected mass flowrate processed by the compressor by 4.2%. Actual discrepancy is larger due to the fact that the test was run at a lower corrected speed than the CFD simulations.**
- **Survey of total pressure and temperature show reasonable match with CFD**
- **Survey of Mach number shows a lower value than predicted by CFD and survey of static pressure show a higher value than predicted by CFD analysis**
  - **Likely due to higher blockage in test with respect to the CFD model due to geometric simplifications introduced in the model to maintain its size within reasonable limits (e.g. wheel spaces, bleed and mass takeoff cavities) and to differences in corrected speed between CFD and test**
  - **Measured and CFD profile shapes show good agreement**
- **Survey of flow angle shows poor agreement with CFD prediction**
  - **Probe alignment during test was questionable and could explain the shift in measured angle observed between CFD and experimental data**
  - **Measured and CFD profile shapes show good agreement**
- **Differences between the CFD predictions and the measured data could be attributed to slightly different operating conditions and to the effects of the supersonic probe on the flowfield**
- **Overall the comparison indicates that our CFD tool can be used to predict the inducer behavior, and to improve its design and performance**

APPENDIX 12.2

Nozzle Analysis

Final Design Review

Nozzle Test Configuration

System owner:

Rick Wiederien

Oct 31 & Nov 4, 2013

R6, as presented

R7, 11/11/2013-11/21/2013:

- ***Updated to-do list for each component***
- ***Updated slide 3 (day 2 attendance)***
- ***Insert new slide 164 showing stress at fillet to pin-fin within strake***

Attendees – Day 1

- Lorie Krois
- Kirk Lupkes
- Silvano Saretto
- Frank Lu
- Paul Brown
- John Beers
- Chris Braman
- Rob Draper
- Karl Guntheroth
- Steve Amsbaugh
- Bill Ward

Attendees – Day 2

- **Lorie Krois**
- **Kirk Lupkes**
- **Silvano Saretto**
- **Frank Lu**
- **Paul Brown**
- **Ravi Srinivasan**
- **Chris Braman**
- **Rob Draper**
- **Karl Guntheroth**
- **Steve Amsbaugh**
- **Bill Ward**

Agenda

Day 1

- Action items from PDR
- Mechanical Design Overview
- System Flow Passage
- Mechanical Details
  - Planar Section Views
  - Strake (Includes CFD and thermal summaries) (Actually Stopped here)

  - Ramps and Impingement Plates
  - Remaining Pieces
- Instrumentation

Day 2

- Structural Analysis
- Thermal Analysis
- Tolerance Analysis / Hot-Cold variation
- Interfaces
- Requirements Compliance
- Schedule
- Budget
- Conclusions
- Summary of action items
- Backup Items
  - Notes from Day 1
  - Remaining Tasks

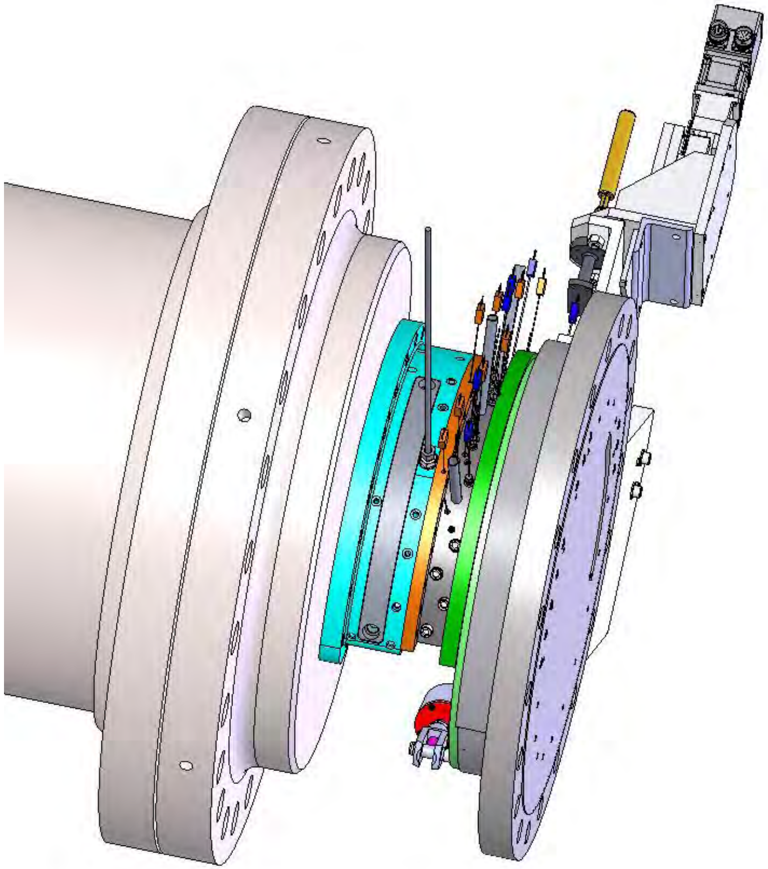
Ramgen FDR Expectations

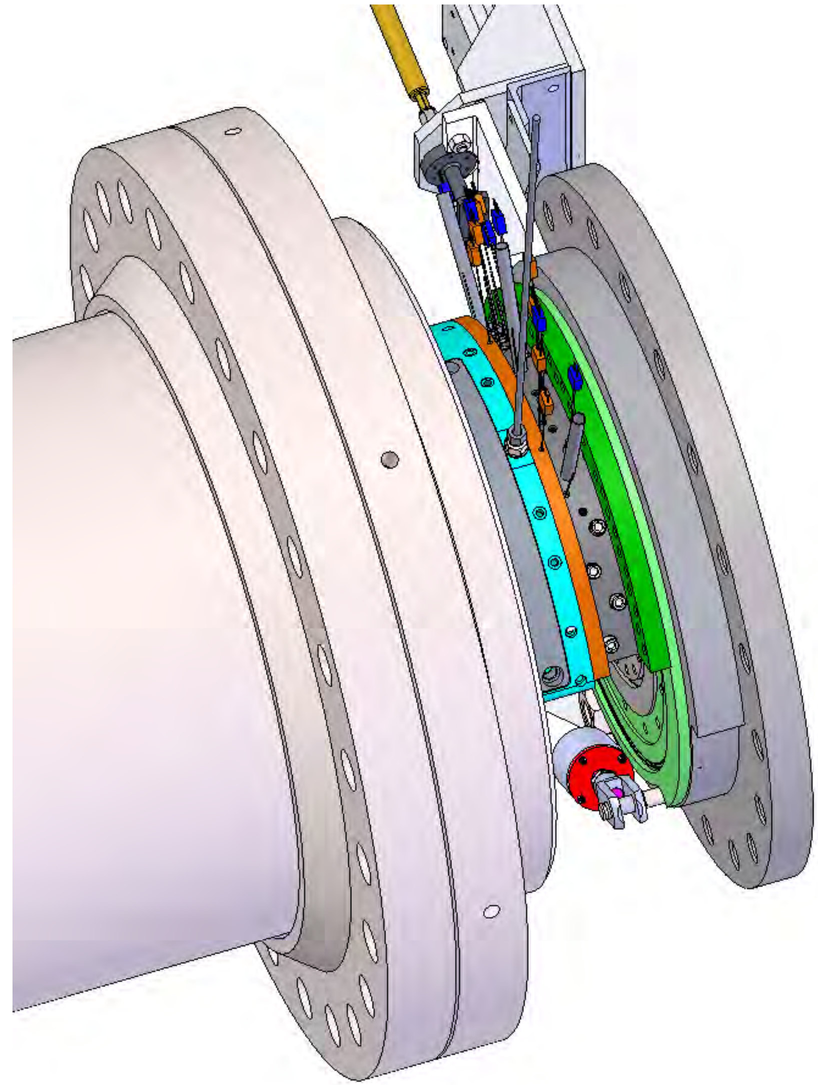
- Final System Requirements with Values
- Final boundaries of the system
- Final interfaces with other systems
- System design
- Final list of analyses performed
- Results of analyses performed
- Review Budget
- Review Schedule

All items will be covered

Mechanical Design Overview

Integrated Test System





Materials of Construction

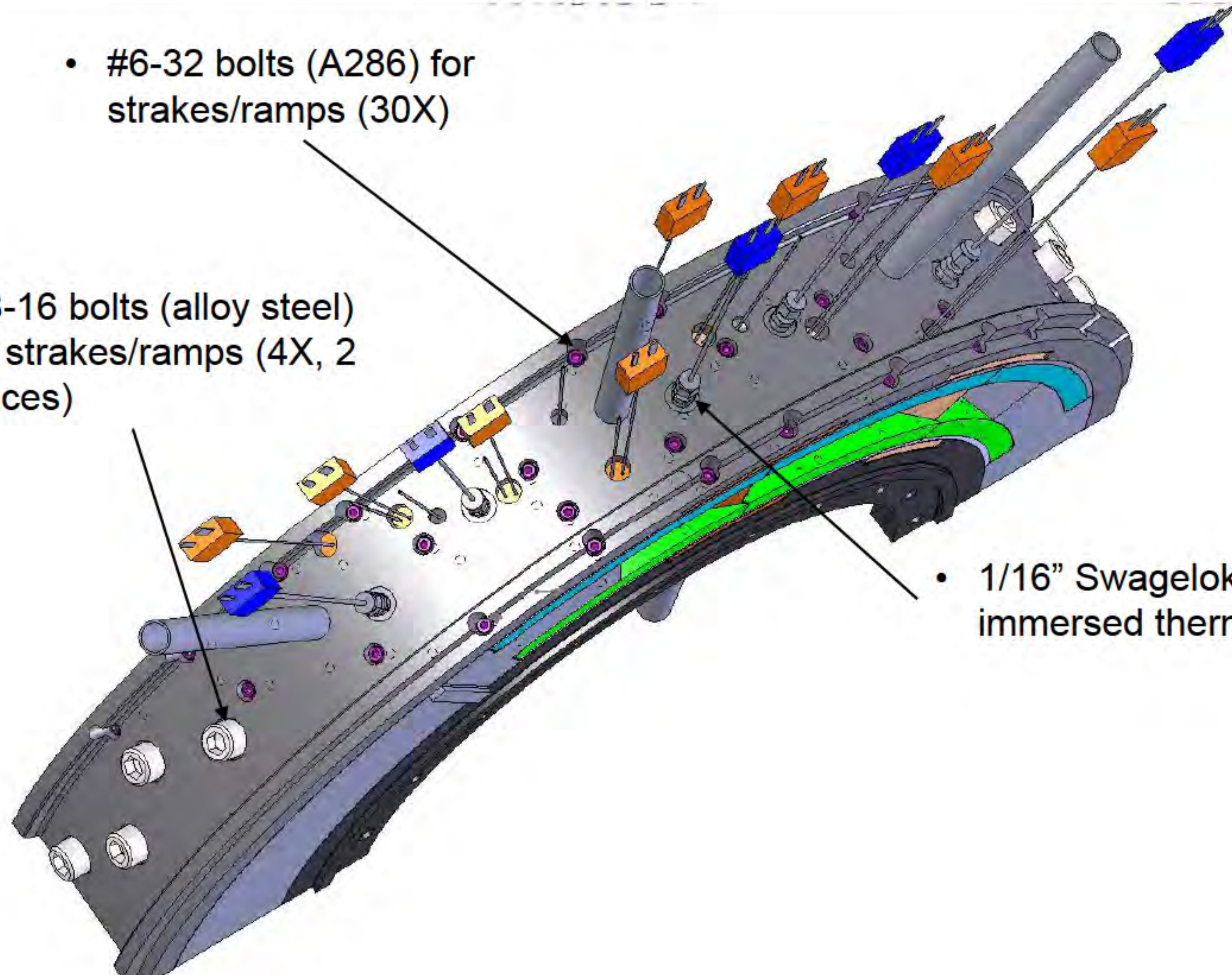
- **IN 718 (DMLS)**
  - **Strake**
  - **Selection criteria**
    - **Nickel-based alloy for simularity to ultimate engine configuration**
    - **Commonly available DMLS material**
    - **Good building DMLS material**
- **INCO 625**
  - **Strake insert**
  - **Selection criteria**
    - **Similar CTE to IN 718**
    - **Nickel-based alloy for simularity to ultimate engine configuration**

Materials of Construction

- 17-4 PH, Condition H1150
  - Ramps
  - Impingement Plates
  - Hub
  - Shroud
  - Endwalls
  - Shroud Clamps, Upstream and Downstream
  - Selection criteria
    - CTE closely resembles IN 718 and INCO 625
    - Condition H1150 has the best machinability and adequate strength
    - Availability (large diameter bar and plate)
- Bolts
  - #6 screws for ramps/strakes: A286
    - Selection criteria: Good strength at temp, corrosion resistant, ductility
  - All other screws
    - A 574 (grade 8) alloy steel
    - Selection critieria: Strength, cost, availability

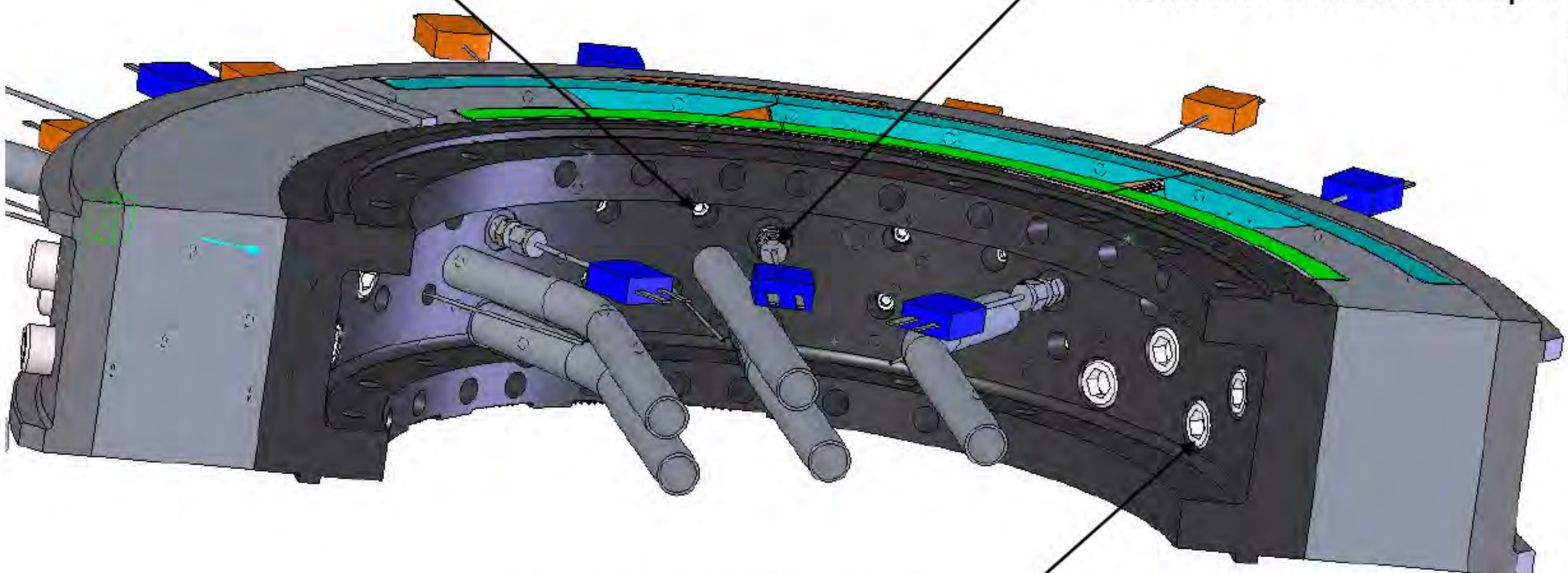
Shroud without Flanges

RAMGEN POWER SYSTEMS ™



Shroud without Flanges

- #6-32 bolts (A286) for strakes/ramps (30X)
- 1/16” Swagelok fitting for immersed thermocouples



- 3/8-16 bolts (alloy steel) for strakes/ramps (4X, 2 places)

Sealing Method

Toll Free: 87 PERMATEX
(877-376-2839)

10 Columbus Blvd., Hartford, Connecticut 06106

6875 Parkland Boulevard, Solon Ohio 44139

Technical Data Sheet

Permatex® Ultra Copper® RTV Silicone Gasket

AAM Revised 09/06

PRODUCT DESCRIPTION **S.I.N.: 834-300**

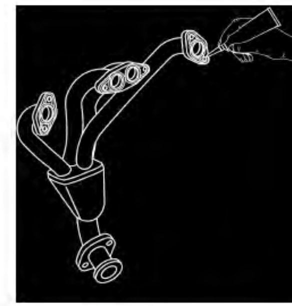
Permatex® Ultra Copper® is a single component, room temperature vulcanizing gasketing compound designed to provide reliable "formed-in-place" gaskets for mechanical assemblies. This material cures on exposure to moisture in the air to form a tough, flexible, silicone rubber gasket. The product resists aging, weathering and thermal cycling without hardening, shrinking or cracking. Designed for the higher temperature environments encountered in 4-cylinder, turbocharged, and high performance engines. Permatex® Ultra Copper® is the most advanced high performance, high temperature (up to 700°F intermittent) RTV gasket available. OEM Specified.

PRODUCT BENEFITS
- High temperature resistance
- Sensor safe, non-corrosive
- Superior adhesion and flexibility
- Replaces most cut gaskets
- Improved oil resistance
- Can be used as a gasket maker or dressing
- Non-flammable, Non-toxic
- Low odor

TYPICAL APPLICATIONS
- Exhaust manifolds
- Thermostat housings
- Valve covers
- Timing gear covers
- Water pumps
- Differential covers

DIRECTIONS FOR USE

For assembly as form-in-place gasket
1. Remove all previous material from mating surfaces. Permatex® Silicone Stripper or Gasket Remover is recommended for most materials.
2. For best results, clean and dry all surfaces with a residue-free solvent, such as Permatex® Brake and Parts Cleaner.
3. Cut nozzle to desired bead size, 1/16 " to 1/4 " in diameter. An 1/8" bead is usually sufficient for most applications.
4. Remove cap, puncture tube or cartridge seal and attach extension nozzle.
5. Apply a continuous and even bead of silicone to one surface, first tracing the internal areas of the gasket configuration, then all surrounding bolt holes as shown below:



6. Assemble parts immediately while silicone is still wet. Secure or tighten to recommended torque specs.
7. Re-torque will not be necessary after the product has cured.

For assembly as gasket dressing
1. Repeat steps 1 thru 4 as in previous section.
2. Apply a thin film of silicone to one surface to be sealed.
3. Place the pre-cut gasket onto silicone film.
4. Apply a second thin to pre-cut gasket surface.
5. Remove any excess and assemble parts immediately.
Note: Product not recommended for use as a cylinder head gasket or head gasket sealant.

Instructions for PowerBead™
1. Clean and dry all flange surfaces to be sealed.
2. Remove black cap from top of extension nozzle.
3. Turn nozzle extension one complete turn (360°) counterclockwise.
4. Depress finger trigger and apply a continuous 1/16 inch to 1/8 inch PowerBead™ to one surface.
5. Assemble parts immediately while silicone is still wet.
6. Finger tighten flange only until material begins to seep out the sides of the flange.
7. Allow to set for at least two hours and re-torque at least one quarter to one half turn.
8. For best results, allow to cure overnight.
9. To close, turn extension nozzle clockwise until tight (about one full turn). Wipe off excess material from nozzle and replace black cap.

System Flow Passage

Test Arrangement CFD Analysis

- CFD analysis was completed for the primary flow passage, including the total pressure probe
- See linked presentation for more details
  - Test_Section_FlowPath_CFD_Results_10-15-2013.pptx
  - Excerpts follow
- Conclusions:
  - Shock wave from probe propagates upstream, but will not intefere with the test

- CFD of test configuration with cooling air has not been completed and is planned to proceed in parallel to part fabrication

Objective

(From "Test_Section_FlowPath_CFRD_Results_10-15-2013.ppt"

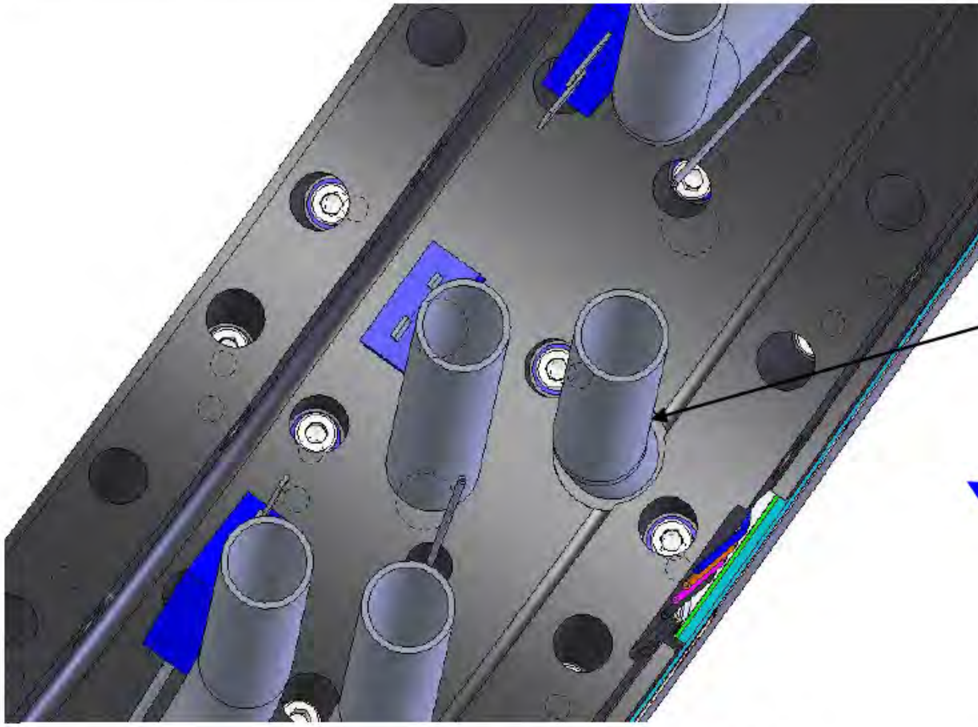
- **Evaluate flow behavior in the geometry designed for the ISCE Build 2 Nozzle Test**
  - **Simulate flow in the test domain with probe in location 1**
    - **Location 2 is at 0.45" downstream of the nozzle strake TE, aligned with the camber angle of the strake (left) and aligned in the flow direction.**
  - **Evaluate the differences between the two flow-path test section and the test section with the probe**
    - **Both simulations to be run at the 10 psi back pressure**

Mechanical Design Details

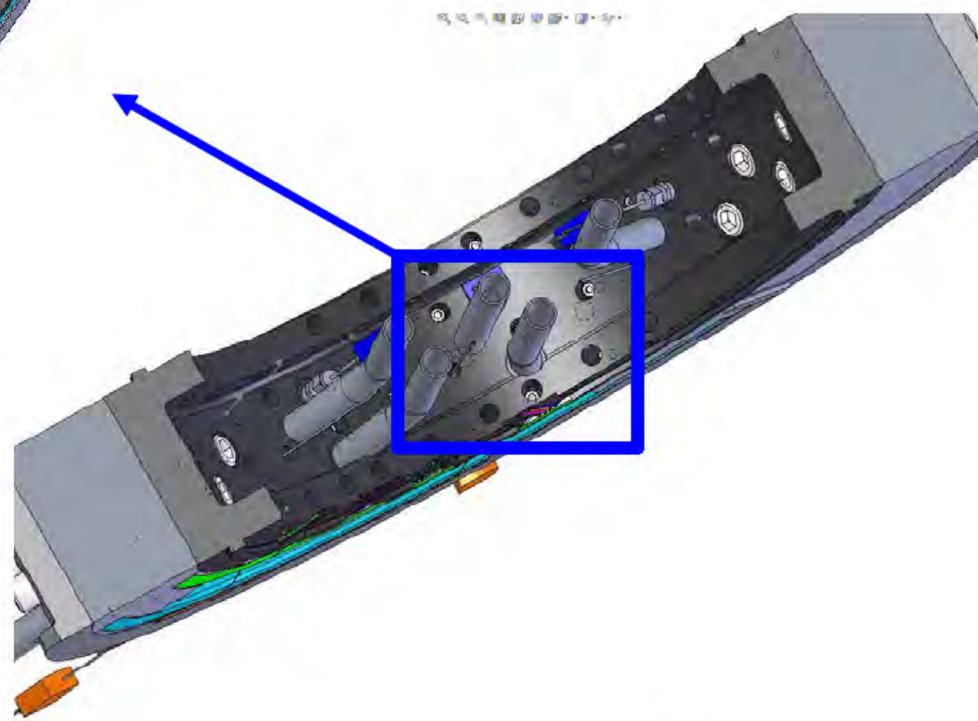
Planar Section Views

Air Inlet to Strake

RAMGEN POWER SYSTEMS ™



- Air inlet to strake



- Strake cooling @ 85 psia delivery, two final branches (See Piping FDR slides for details)
  - 1/2" tube, 3%, Mach 0.11, <u>0.4 psi loss /ft</u>
  - 1/2" tube, 1.5%, Mach 0.05, 0.1 psi loss/ft

Strake

Strake Thermal Design and Analysis– QuEST

–Detailed thermal design and analysis was performed by Jim Bruns at QuEST

–Focus was on the engine configuration

–Detailed report for the engine level analysis is linked

–Ramgen Nozzle heat transfer design summary Oct 13.pptx

–1D calculations done for test configuration

–Excerpts follow

–For details, see Ramgen_Turbine_HT_Nozzle_test_1D_HT_9Oct13.pdf

Strake CFD Results – Engine Config

- Excerpts from CFD analysis completed 10/17/2013 follows
- See linked presentation for more details
  - Strake_Internal_Flow_Field_Revised_Strake_10_16_2013_rev2.pptx

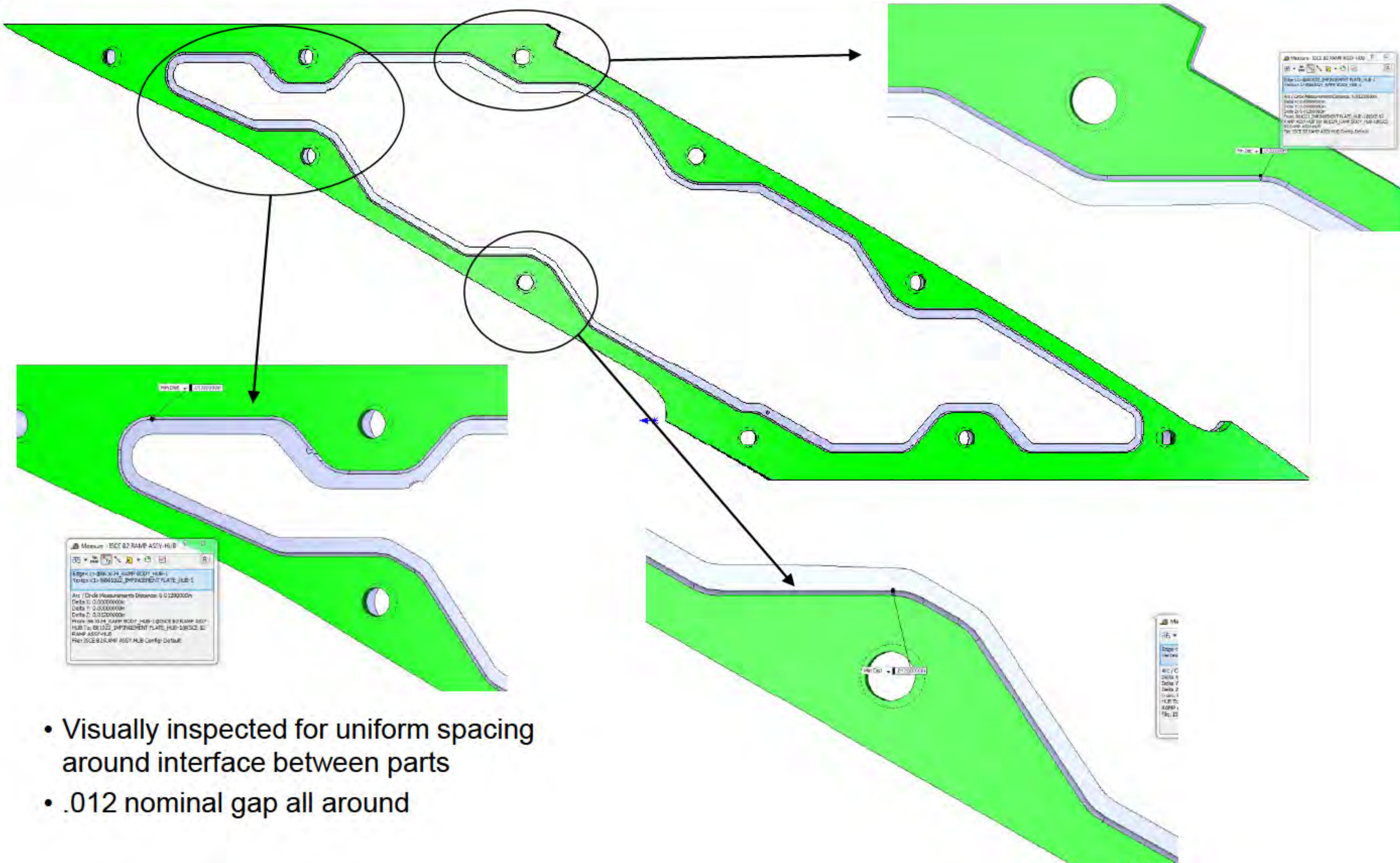
Mass Flow Rates

lb/s					
Inlet	2.09E-02	LE_1	7.49E-04	LE_L_1	2.07E-04
Outlet	2.09E-02	LE_2	6.94E-04	LE_L_2	1.99E-04
Error	3.15E-02	LE_3	6.48E-04	LE_L_3	1.98E-04
		LE_4	6.21E-04	LE_L_4	1.98E-04
		LE_5	5.99E-04	LE_L_5	1.98E-04
		LE_6	5.84E-04	LE_L_6	1.98E-04
		LE_7	5.72E-04	LE_L_7	2.01E-04
		TOTAL	4.47E-03	TOTAL	1.40E-03

LE_LL_1	3.66E-04	LE_R_1	1.90E-04	LE_RR_1	3.64E-04
LE_LL_2	3.66E-04	LE_R_2	2.00E-04	LE_RR_2	3.44E-04
LE_LL_3	3.57E-04	LE_R_3	1.99E-04	LE_RR_3	3.45E-04
LE_LL_4	3.58E-04	LE_R_4	1.97E-04	LE_RR_4	3.39E-04
LE_LL_5	3.54E-04	LE_R_5	1.96E-04	LE_RR_5	3.36E-04
LE_LL_6	3.38E-04	LE_R_6	1.96E-04	LE_RR_6	3.31E-04
LE_LL_7	3.29E-04	LE_R_7	1.98E-04	LE_RR_7	3.24E-04
LE_LL_8	3.19E-04	LE_R_8	1.93E-04	LE_RR_8	3.15E-04
TOTAL	2.79E-03	TOTAL	1.57E-03	TOTAL	2.70E-03

Ramps and Impingement Plates

Hub Ramp Assembly Circular Cross Section

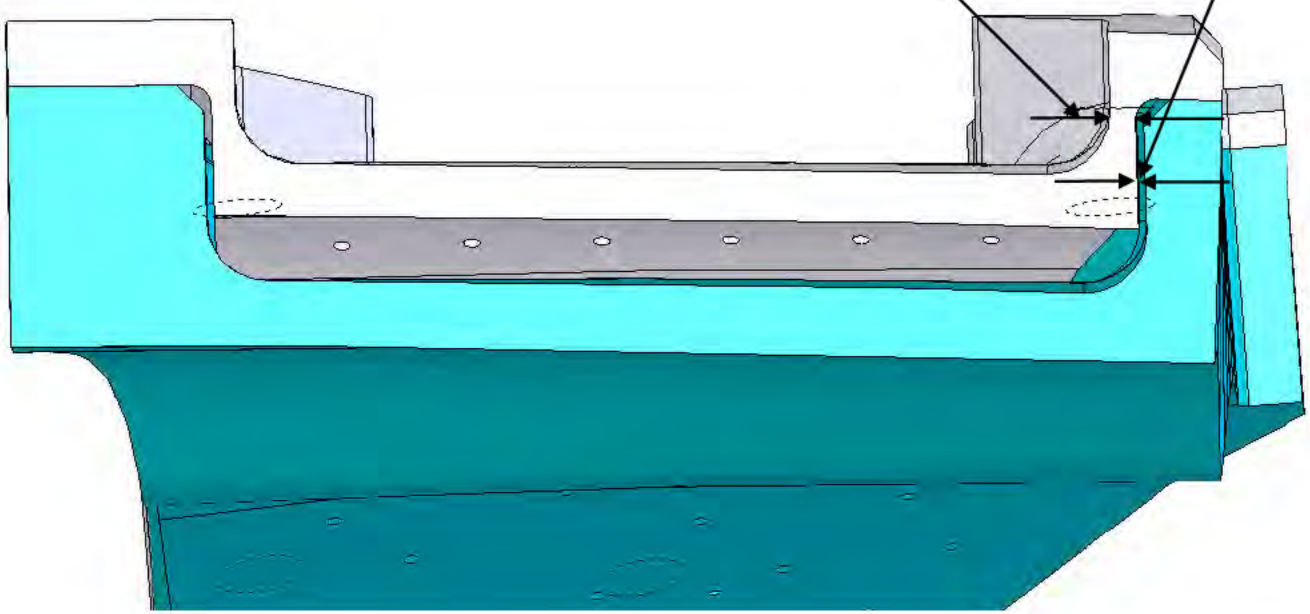


- Visually inspected for uniform spacing around interface between parts
- .012 nominal gap all around

Shroud Ramp Assembly

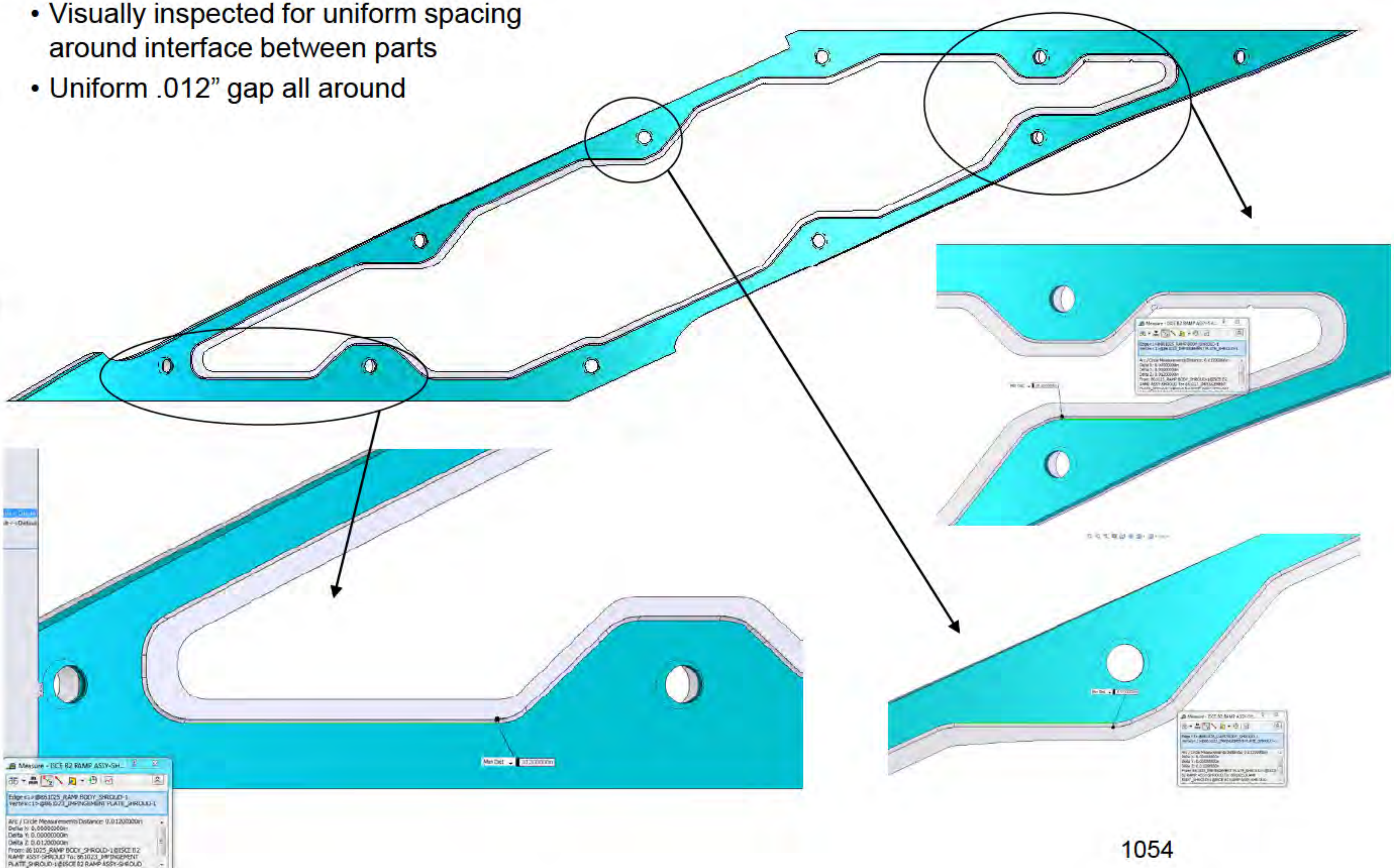
- .038 nominal wall all around
  - .008 surface profile tolerance default (±.004)
  - Minimum wall thickness: 0.030

- .012 nominal gap all around
  - .008 surface profile tolerance default (±.004)
  - Minimum gap: 0.004



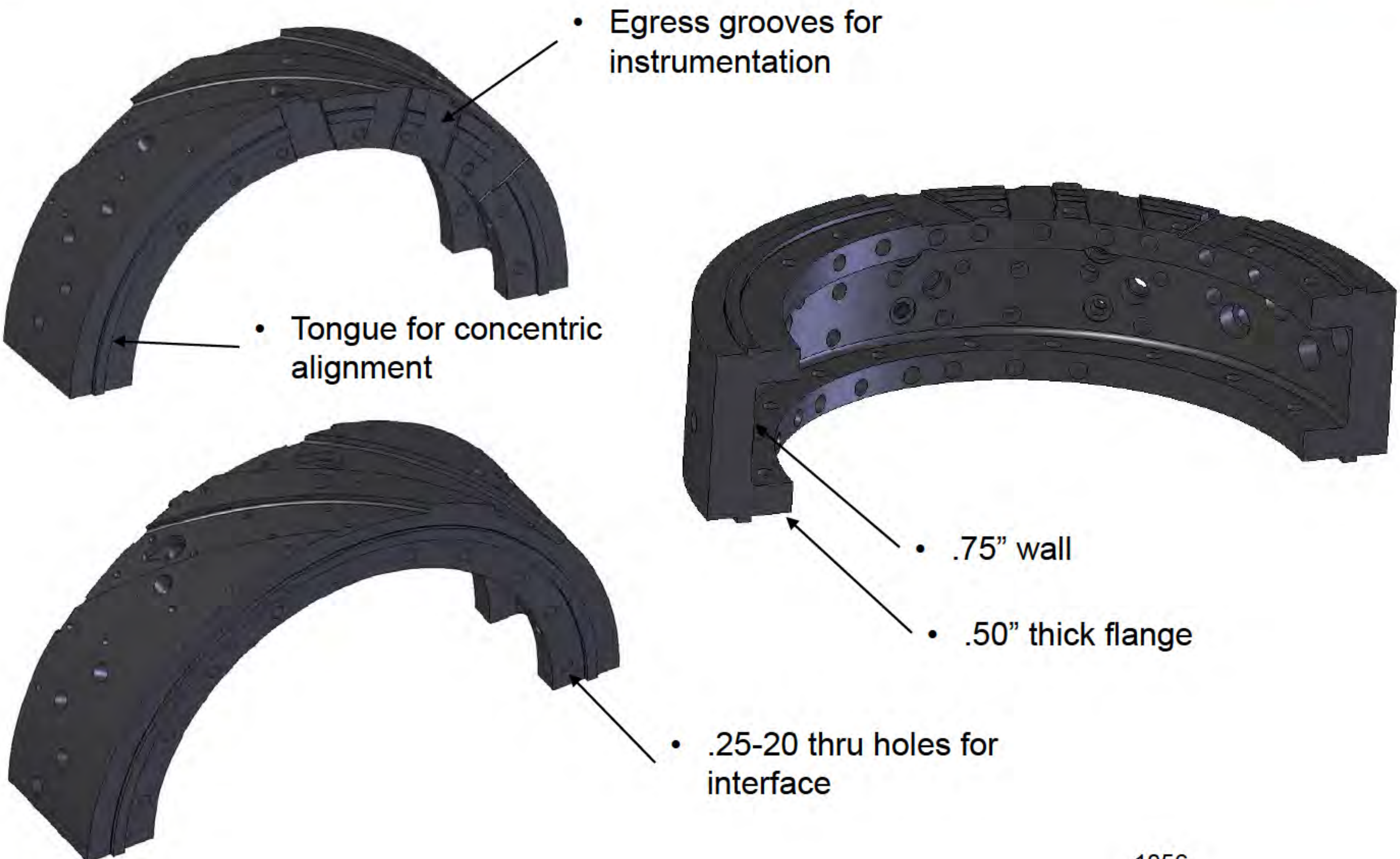
Shroud Ramp Assembly Circular Cross Section

- Visually inspected for uniform spacing around interface between parts
- Uniform .012" gap all around

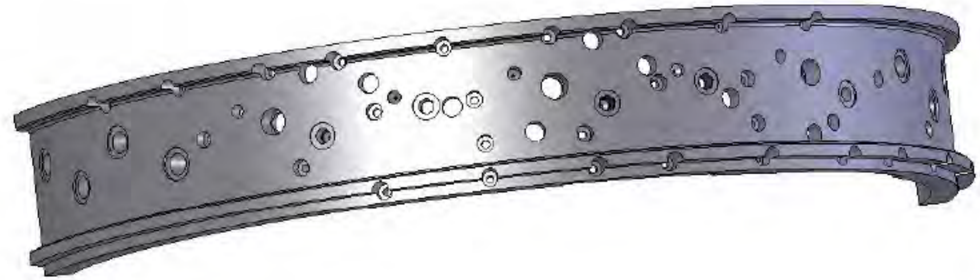


Remaining Pieces

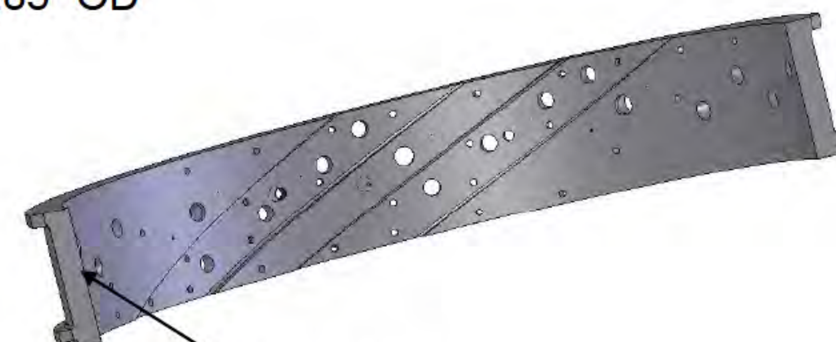
Hub


• Egress grooves for instrumentation
• Tongue for concentric alignment
• .75” wall
• .50” thick flange
• .25-20 thru holes for interface

Shroud

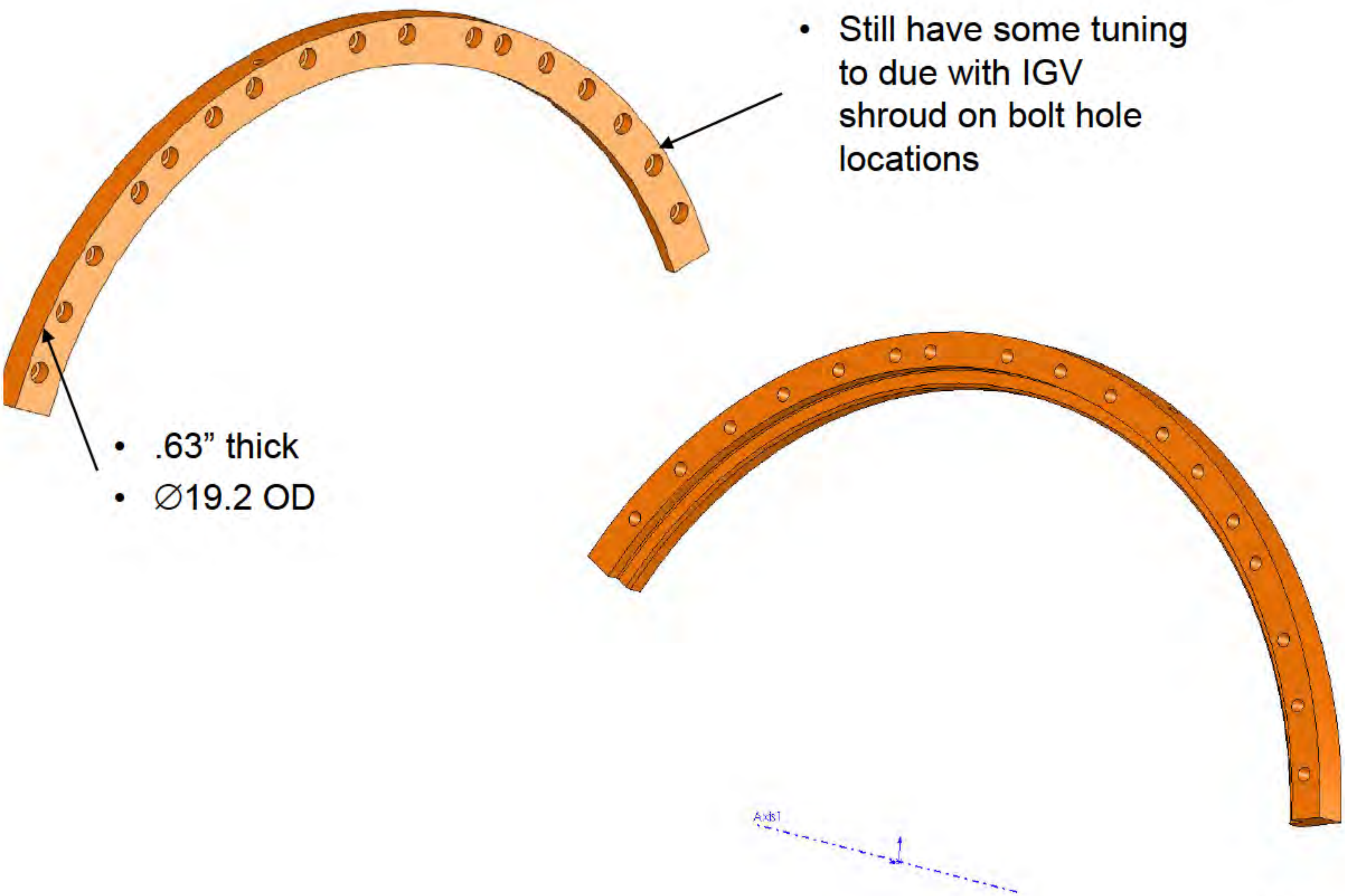


- 17.85” OD



- .50” wall

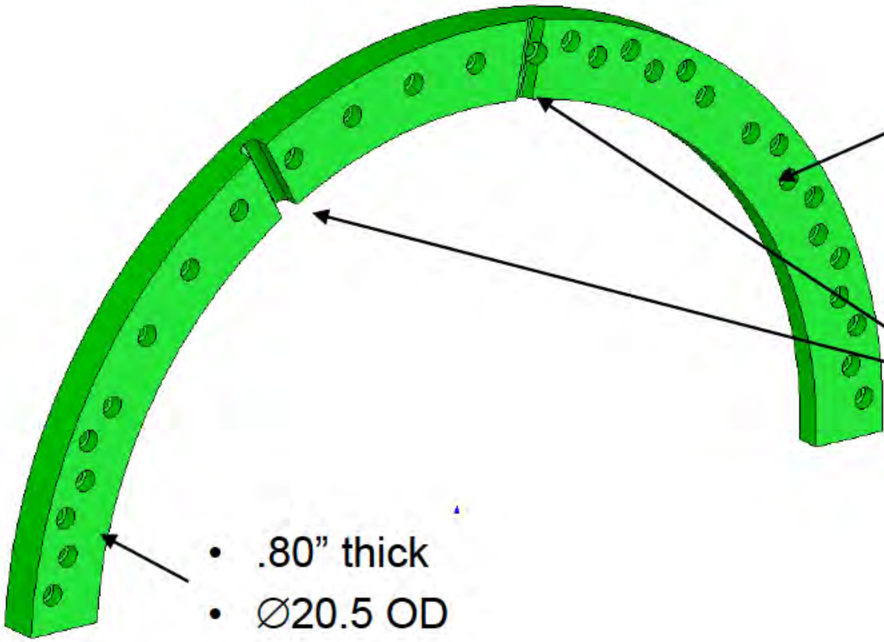
Shroud Clamp, Upstream



- Still have some tuning to due with IGV shroud on bolt hole locations

- .63” thick
- ∅19.2 OD

Shroud Clamp, Upstream



- Still have some tuning to due with measurement section on bolt hole locations
- Notches around instrumentation
- .80” thick
- ∅20.5 OD

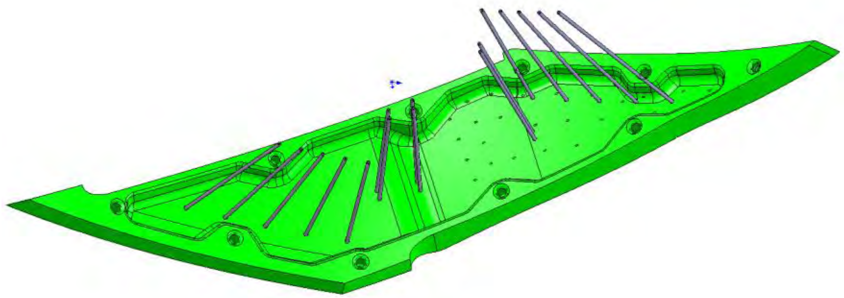
Instrumentation

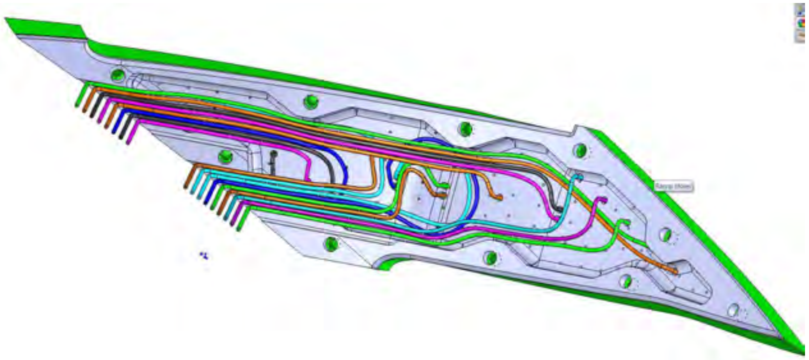
Instrumentation Summary

- **Internal Instrumenation**
  - **Static pressure: 42**
  - **Subsurface (wall) temperature: 30**
  - **Immersed gas temperature: 8**

Ramp Assembly Instrumentation Sequence

1. Bond instrument tubing to sockets in ramp
    - Thermally conductive antisieze on thermocouple tips
    - Epoxy stake around perimeter
2. Insert impingement plate
    - Will require flexing of instrumentation tubing
3. Bend and laying down tubing, starting on the forward side and working aft
    - Epoxy stake around perimeter as proceeding





Ramp Assembly Details

- See linked slides for detailed assembly sequence with instrumentation for ramp assembly
  - 69 Ramp Instrument Egress Details 10-23-2013a.pptx

Additional Instrumentation

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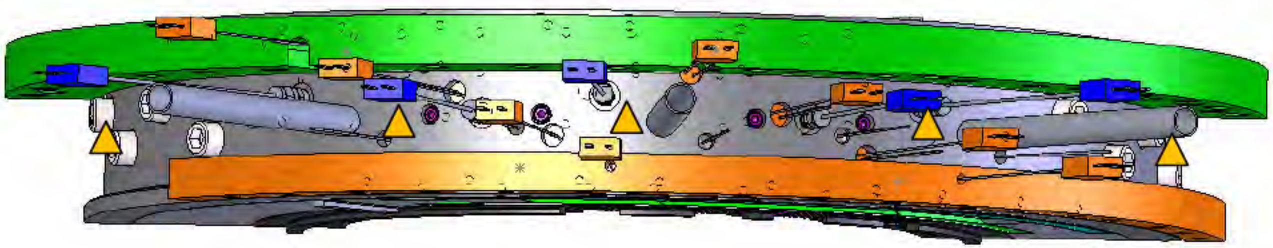
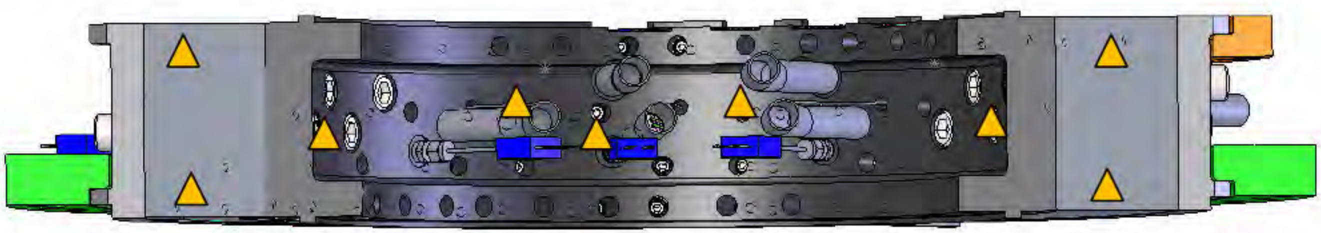
Legend

△ Surface temperature

- **Need to add external thermal couples to instrumentation plan for nozzle and other components**
- **Propose adding several surface metal thermocouples to exterior surfaces**
  - **Hub: 5X**
  - **Shroud: 5X**
  - **Endwalls: 2X each (4 total)**





Structural Analysis

Structural Analysis Conclusion

- Stresses in all components are comfortably below the yield strength of the material away from the bolt hole
- Constraint method at bolt holes (line contacts) results in some elevated stresses at the edge of the bolts, but even these are comfortably below the material yield strength
- Structural integrity is dependant on the bolt strength
  - All tensile loads are well below bolt rated breaking strength
  - Shear loads, if preload is lost, are below bolt material shear strength
    - Shear loads are discussed separately where appropriate

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Material Properties

Material	IN 718 (DMLS), as-built	17-4 H1150
Component	Strake	Ramps, Impingement plates, Hub, Shroud, Clamps, Endwalls
Elastic Modulus	23,000 ksi	28,500 ksi
Poison's Ratio	0.297	0.27
Yield Strength at Room Temp	113 ksi in X/Y 92 ksi in Z (can be heat treated to 180)	105 ksi
Ult Strength (ksi) at Room Temp	154 ksi in X/Y 142 ksi in Z (can be ht trtd to 203)	135 ksi
Yield Str at 650°F	84.6 ksi in Z (92%)	93 ksi (89%)
Ult Str at 650°F	131 ksi in Z (92%)	112 ksi (83%)
CTE	7.78 ppm/°F @ 650°F	7.15 ppm/°F @ 650°F
Elongation at Break	27% in X/Y 31% in Z	8% for thicknesses < .186" (relevant to ramps/impingement plates)
Notes:	Source: MIL-HDBK-5H, EOS datasheet	Dimensional contraction during heat treat – requires post machining – H1150 easiest to machine Sources: MIL-HDBK-5H

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IN 718 Datasheet Excerpts

eos

Material data sheet

EOS NickelAlloy IN718

EOS NickelAlloy IN718 is a heat and corrosion resistant nickel alloy powder which has been optimized especially for processing on EOSINT M systems.

This document provides information and data for parts built using EOS NickelAlloy IN718 powder (EOS art.-no. 9011-0020) on the following system specifications:

- EOSINT M 270 Installation Mode *Xtended*
  with PSW 3.4 and default job IN718_020_default.job
- EOSINT M 270 Dual Mode
  with PSW 3.5 and EOS Original Parameter Set IN718_Surface 1.0
- EOSINT M 280
  with PSW 3.5 and EOS Original Parameter Set IN718_Surface 1.0

Description

Parts built from EOS NickelAlloy IN718 have chemical composition corresponding to UNS N07718, AMS 5662, AMS 5664, W.Nr 2.4668, DIN NiCr19Fe19NbMo3. This kind of precipitation-hardening nickel-chromium alloy is characterized by having good tensile, fatigue, creep and rupture strength at temperatures up to 700 °C (1290 °F).

This material is ideal for many high temperature applications such as gas turbine parts, instrumentation parts, power and process industry parts etc. It also has excellent potential for cryogenic applications.

Parts built from EOS NickelAlloy IN718 can be easily post-hardened by precipitation-hardening heat treatments. In both as-built and age-hardened states the parts can be machined, spark-eroded, welded, micro shot-peened, polished and coated if required. Due to the layerwise building method, the parts have a certain anisotropy - see Technical Data for examples.

EOS GmbH – Electro Optical Systems
Robert-Stirling-Ring 1
D-82152 Krailling / München
Telephone: +49 (0)89 / 893 36-0
Telefax: +49 (0)89 / 893 36-285
Internet: www.eos.info

eos

Material data sheet

Mechanical properties of parts at 20 °C (68 °F)

	As built	Heat treated per AMS 5662 [5]	Heat treated per AMS 5664 [6]
Tensile strength [7]			
- in horizontal direction (XY	typ. 1060 ± 50 MPa (154 ± 7 ksi)		
- in vertical direction (Z)	typ. 980 ± 50 MPa (142 ± 7 ksi)	n. 1241 MPa (180 ksi) yp. 1400 ± 100 MPa (203 ± 15 ksi)	min. 1241 MPa (180 ksi) typ. 1380 ± 100 MPa (200 ± 15 ksi)
Yield strength (Rp 0.2 %) [7]			
- in horizontal direction (XY	typ. 780 ± 50 MPa (113 ± 7 ksi)		
- in vertical direction (Z)	typ. 634 ± 50 MPa (92 ± 7 ksi)	n. 1034 MPa (150 ksi) yp. 1150 ± 100 MPa (167 ± 15 ksi)	min. 1034 MPa (150 ksi) typ. 1240 ± 100 MPa (180 ± 15 ksi)
Elongation at break [7]			
- in horizontal direction (XY	typ. (27 ± 5) %		
- in vertical direction (Z)	typ. (31 ± 5) %	min. 12 % typ. (15 ± 3) %	min. 12 % typ. (18 ± 5) %
Modulus of elasticity [7]			
- in horizontal direction (XY	typ. 160 ± 20 GPa (23 ± 3 Msi)		
- in vertical direction (Z)		170 ± 20 GPa 24.7 ± 3 Msi	170 ± 20 GPa 24.7 ± 3 Msi
Hardness [8]	approx. 30 HRC approx. 287 HB	approx. 47 HRC approx. 446 HB	approx. 43 HRC approx. 400 HB

[5] Heat treatment procedur
1. *Solution Anneal* at 980 °C (1800 °F) for 1 hour, air (/argon) cool.
2. *Ageing treatment*: hold at 720 °C (1330 °F) 8 hours, furnace cool to 620 °C (1150 °F) in 2 hours, hold at 620 °C (1150 °F) 8 hours, air (/argon) cool.

EOS GmbH – Electro Optical Systems
Robert-Stirling-Ring 1
D-82152 Krailling / München

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IN 718MIL-HDBK-5 Excerpts

MIL-HDBK-5H
1 December 1998

Table 6.3.5.0(d). Design Mechanical and Physical Properties of Inconel 718 Investment Castings

Specification	AMS 5383
Form	Investment Casting
Condition	ST
Location within casting	Any
Thickness, in.	≤0.500
Basis	S
Mechanical Properties:	
F_{tu}, ksi	120
F_{ty}, ksi	105
F_{cy}, ksi	105
F_{su}, ksi	88[a]
F_{bru}[b], ksi:	
(e/D = 1.5)	202
(e/D = 2.0)	248
F_{bry}[b], ksi:	
(e/D = 1.5)	161
(e/D = 2.0)	188
e, percent	3
RA, percent	8
E, 10^3 ksi	29.4
E_c, 10^3 ksi	30.9
G, 10^3 ksi	11.4
μ	0.29
Physical Properties:	
ω, lb/in.^{3}	0.297
C, K, and α	See Figure 6.3.5.0

a Determined in accordance with ASTM Procedure B769.
b Bearing values are "dry pin" values per Section 1.4.7.1.

6-54

Investment cast properties shown for reference as comparable with DMLS properties

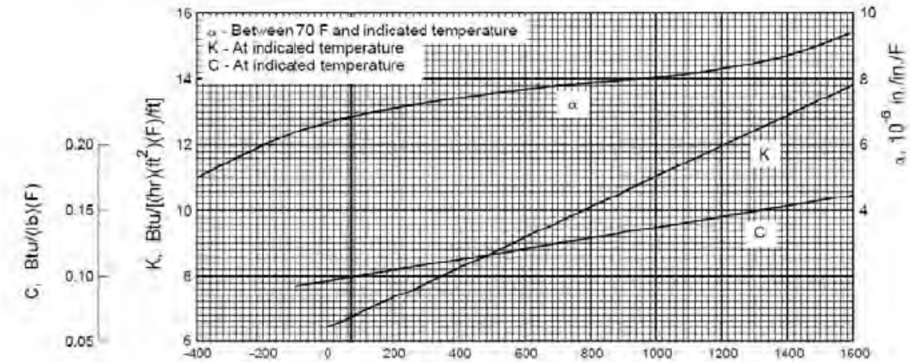

Figure 6.3.5.0. Effect of temperature on the physical properties of Inconel 718.

6-51

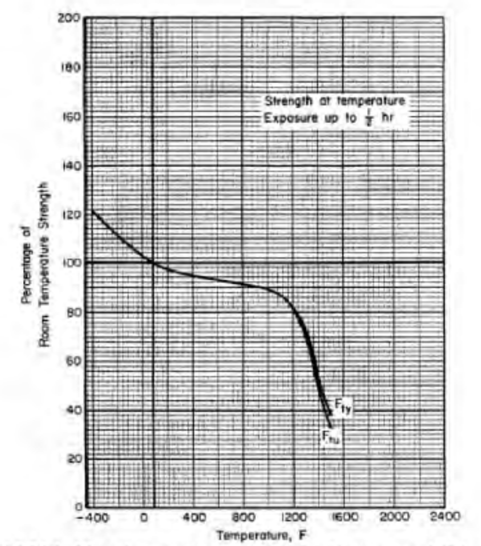

Figure 6.3.5.1.1. Effect of temperature on the tensile ultimate strength (F_{tu}) and tensile yield strength (F_{ty}) of solution-treated and aged Inconel 718.

17-4 PH MIL-HDBK-5 Excerpts

Table 2.6.8.0(b). Design Mechanical and Physical Properties of 17-4PH Stainless Steel Sheet, Strip, and Plate

Specification	AMS 5604					
Form	Sheet, strip[a], and plate					
Condition	H900	H925	H1025	H1075	H1100	H1150
Thickness, in.	≤ 4.000					
Basis	S	S	S	S	S	S
Mechanical Properties:						
F_{tu}, ksi:						
L	...	...	...	...	...	...
LT	190	170	155	145	140	135
F_{ty}, ksi:						
L	...	...	...	...	...	...
LT	170	155	145	125	115	105
F_{cy}, ksi:						
L	...	...	...	...	...	...
LT	...	...	...	...	...	...
F_{su}, ksi	...	...	...	...	...	...
F_{bru}, ksi:						
(e/D = 1.5)	...	...	...	...	...	...
(e/D = 2.0)	...	...	...	...	...	...
F_{bry}, ksi:						
(e/D = 1.5)	...	...	...	...	...	...
(e/D = 2.0)	...	...	...	...	...	...
e, percent:						
LT	b	b	b	b	b	b
E, 10^3 ksi	28.5					
E_c, 10^3 ksi	30.0					
G, 10^3 ksi	11.2					
μ	0.27					
Physical Properties:						
ω, lb/in.^{3}	0.282 (H900), 0.283 (H1075), 0.284 (H1150)					
C, K, and α	See Figure 2.6.8.0					

a Test direction longitudinal for widths less than 9 inches; long transverse for widths 9 inches and over.
b See Table 2.6.8.0(c).

Table 2.6.8.0(c). Minimum Elongation Values for 17-4PH Sheet, Strip, and Plate

	e, percent (LT)					
Thickness	H900	H925	H1025	H1075	H1100	H1150
0.015 through 0.186	5	5	5	5	5	8
0.187 through 0.625	8	8	8	9	10	10
0.626 through 4.000	10	10	12	13	14	16

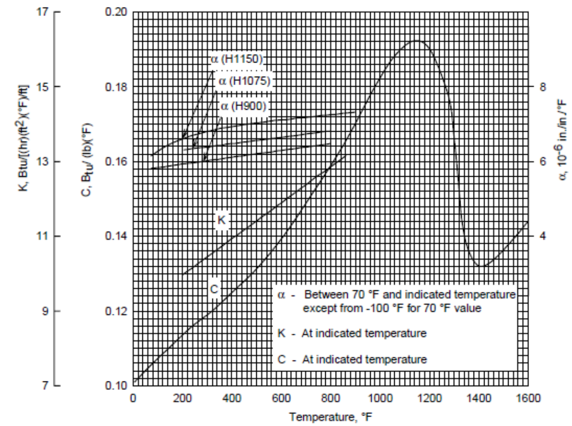

Figure 2.6.8.0. Effect of temperature on the physical properties of 17-4PH stainless steel.

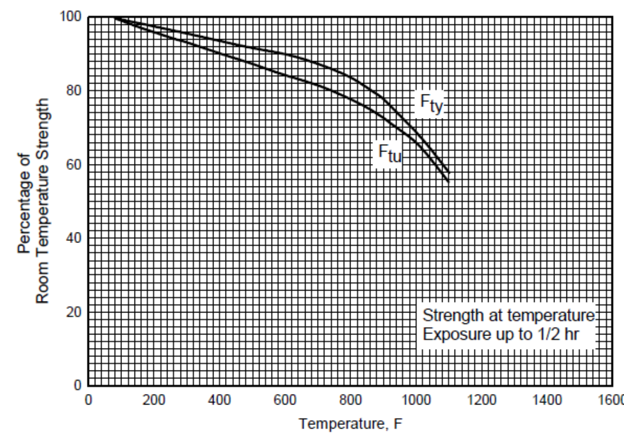

Figure 2.6.8.6.1. Effect of temperature on the tensile ultimate strength (F_{tu}) and the tensile yield strength (F_{ty}) of 17-4PH (H1150) stainless steel bar.

Screws

Component	Strake/Ramp Screws (Reference)	Strake/Ramp Screw	Hub/Shroud	Endwalls
Material	A286	Alloy Steel (A 574)	Alloy Steel (A 574)	Alloy Steel (A 574)
Size	#6-32 (.138-32)	#6-32 (.138-32)	¼-20	.375-16
Minimum Breaking Strength (per NAS 1352)	1458 lbf	1640 lbf	5700 lbf	13900 lbf
Estimated Breaking Strength at 650°F	1203 lbf (88%)	1476 lbf (90% assumed from generic low alloy steels)	5130 lbf (78% assumed from generic low alloy steels)	12510 lbf (78% assumed from generic low alloy steels)
Tensile Strength	160 ksi	180 ksi	180 ksi	180 ksi
Yield Strength	95 ksi (Unbrako handbook)	153 ksi	153 ksi	153 ksi
Proof Strength		140 ksi	140 ksi	140 ksi
Ultimate Shear Strength at Room Temp	99 ksi (.55* Ftu)	108 ksi (.6 * Ftu)	108 ksi (.6 * Ftu)	108 ksi (.6 * Ftu)
Ultimate Shear Strength at 650°F	87 ksi (88%)	97 ksi (90%)	97 ksi (90%)	97 ksi (90%)
CTE	9.4 ppm/°F (650°F)	7.5 ppm/°F (650°F)	7.5 ppm/°F (650°F	7.5 ppm/°F (650°F
Elongation at Break	12-15%	10%	10%	10%
Notes:				

Discussion on Galling Action Item

- Question during Day 1 of FDR was to consider galling
- Could not find direct data for A286 on 17-4
- However, found quantitative data that 17-4 and A286 are both independently prone to galling
  - Surmise that the combination of the 2 would also be prone to galling
- Further tabulation of properties showed that the CTE of alloy steel screws more closely matches CTE of IN 718 and 17-4PH
- Go forward plan is to use alloy steel screws through-out, rather than A286 for the #6-32 screws as previously presented
- Will use high-temperature thread lubricant on all screws to minimize potential for galling
  - Molybdenum disulfide
    - Rated for temperatures up to 725°F

Galling

- **Source: "Review of the Wear and Galling Characteristics of Stainless Steels," A Designers' Handbook Series, Committee of Stainless Steel Producers,**

TABLE XI

GALLING RESISTANCE OF STAINLESS STEELS[10]

Block Material	Condition & Nominal Hardness (Brinell)	Button Material									
		410	416	430	440C	303	304	316	S17400	Nitronic 32	Nitronic 60
Type 410	Hardened & Stress Relieved (352)	3	4	3	3	4	2	2	3	46	50+
Type 416	Hardened & Stress Relieved (342)	4	13	3	21	9	24	42	2	45	50+
Type 430	Annealed (159)	3	3	2	2	2	2	2	3	3	36
Type 440C	Hardened & Stress Relieved (560)	3	21	2	11	5	3	37	3	50+	50+
Type 303	Annealed (153)	4	9	2	5	2	2	3	3	50+	50+
Type 304	Annealed (140)	2	24	2	3	2	2	2	2	30	50+
Type 316	Annealed (150)	2	42	2	37	3	2	2	2	3	38
S17400	H 950 (415)	3	2	3	3	2	2	2	2	50+	50+
Nitronic 32	Annealed (235)	46	45	8	50+	50+	30	3	50+	30	50+
Nitronic 60	Annealed (205)	50+	50+	36	50+	50+	50+	38	50+	50+	50

Values shown are unlubricated threshold galling stress (10^3 psi) for the "button and block" galling test. Condition and hardness apply to both the button and the blank material. Tests were terminated at 50 x 10^3 psi, so values given as 50+ indicate the samples did not gall.

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GALLING RESISTANCE OF ALLOYS[14]

Metals in Contact			Threshold Galling Stress (KSI)
Silicon Bronze	(200) vs. Silicon Bronze	(200)	4
Silicon Bronze	(200) vs. Type 304	(140)	44
A286	(270) vs. A286	(270)	3
AISI 4337	(484) vs. AISI 4337	(415)	2
AISI 1034	(415) vs. AISI 1034	(415)	2
Waukesha 88	(141) vs. Type 303	(180)	50+
Waukesha 88	(141) vs. Type 201	(202)	50+
Waukesha 88	(141) vs. Type 316	(200)	50+
Waukesha 88	(141) vs. S17400	(405)	50+
Waukesha 88	(141) vs. 20Cr-80Ni	(180)	50+
Type 201	(202) vs. Type 201	(202)	15
Type 201	(202) vs. Type 304	(140)	2
Type 201	(202) vs. S17400	(382)	2
Type 201	(202) vs. Nitronic 32	(231)	36
Type 301	(169) vs. Type 416	(342)	3
Type 301	(169) vs. Type 440C	(560)	3
Type 410	(322) vs. Type 420	(472)	3
Type 416	(342) vs. Type 416	(372)	13
Type 416	(372) vs. Type 410	(322)	4
Type 416	(342) vs. Type 430	(190)	3
Type 416	(342) vs. 20Cr-80Ni	(180)	7
Type 440C	(560) vs. Type 440C	(604)	11
S17400	(311) vs. Type 304	(140)	2
S17400	(380) vs. Nitronic 32	(401)	13
S17400	(435) vs. Type 304	(140)	2
S17400	(400) vs. S17700	(400)	3
S17400	(435) vs. S17700	(435)	2

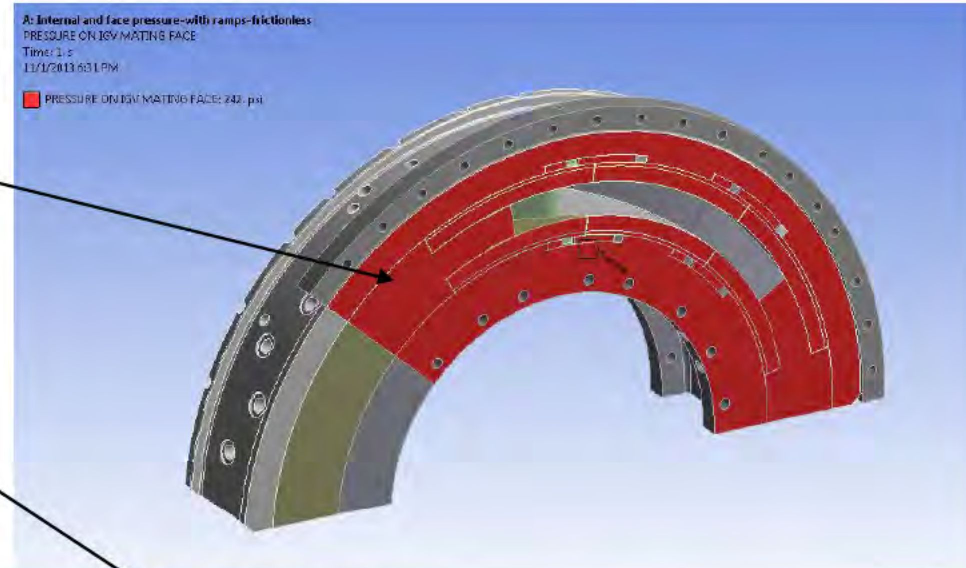
The lubricating properties of molybdenum disulfide are unaffected by temperatures up to 725ºF (400ºC), since, being solid, viscosity changes with temperature changes do not occur as with normal oils and greases.

Load Case #1, Worst Case Pressure Load

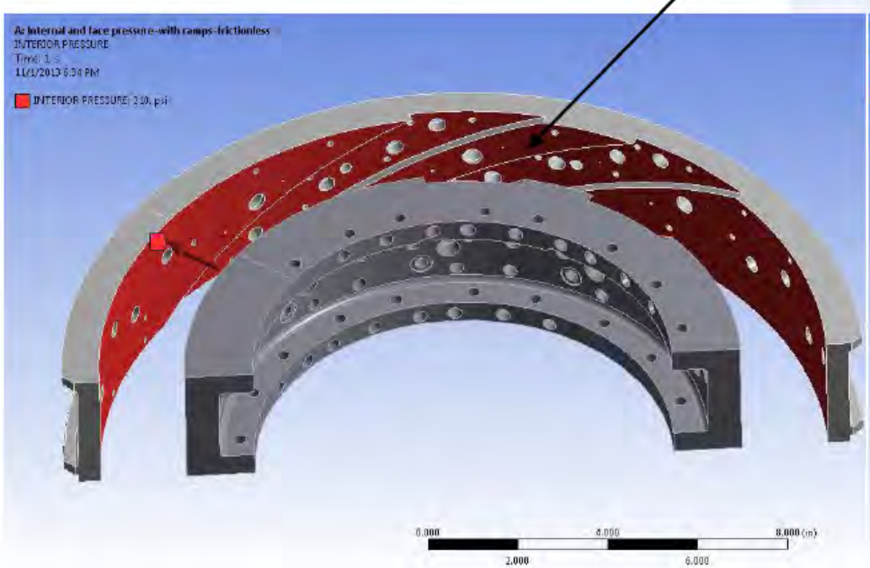
- **210 psi on all hub and shroud surfaces**
  - **Assumes regulator fails wide open**
  - **Assumes throat gets blocked (only way full compressor pressure is reached given the CFV in the system)**
  - **Assumes all seals leak and pressure propagates through all surfaces**
- **Analysis done with room temperature properties**
  - **Results compared to derated properties at 650°F**
- **Objective**
  - **Stress distribution in hub, shroud, strakes, shroud clamp, and endwalls (load bearing elements)**
  - **Bolt loads**
- **Assumptions**
  - **Ignored tongue & grove interface to IGV and Measurement Ring**
  - **Ignored measurement ring**

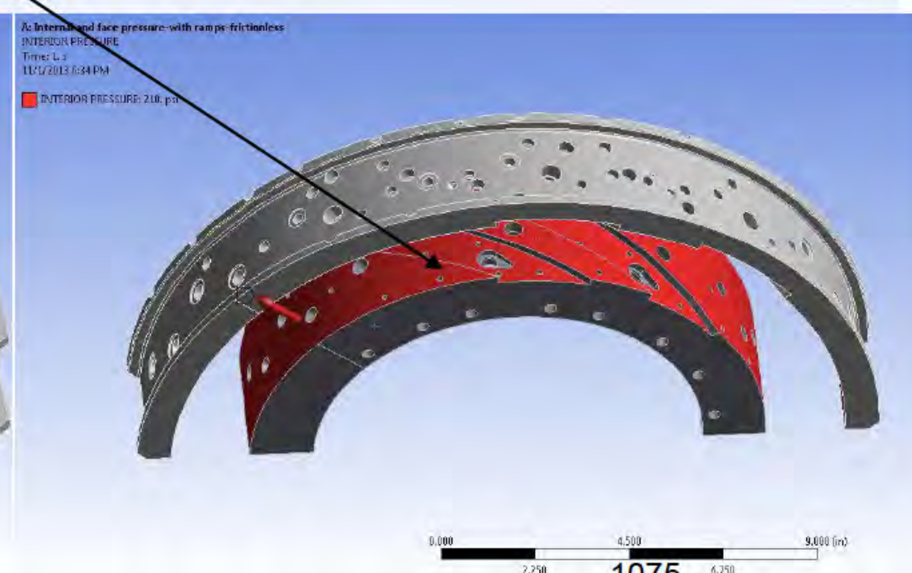
Loads

- 242 psi applied across the entire mating surface with the IGV
- Based on 210 psi * (selectable surface area / total area with throat blockage)

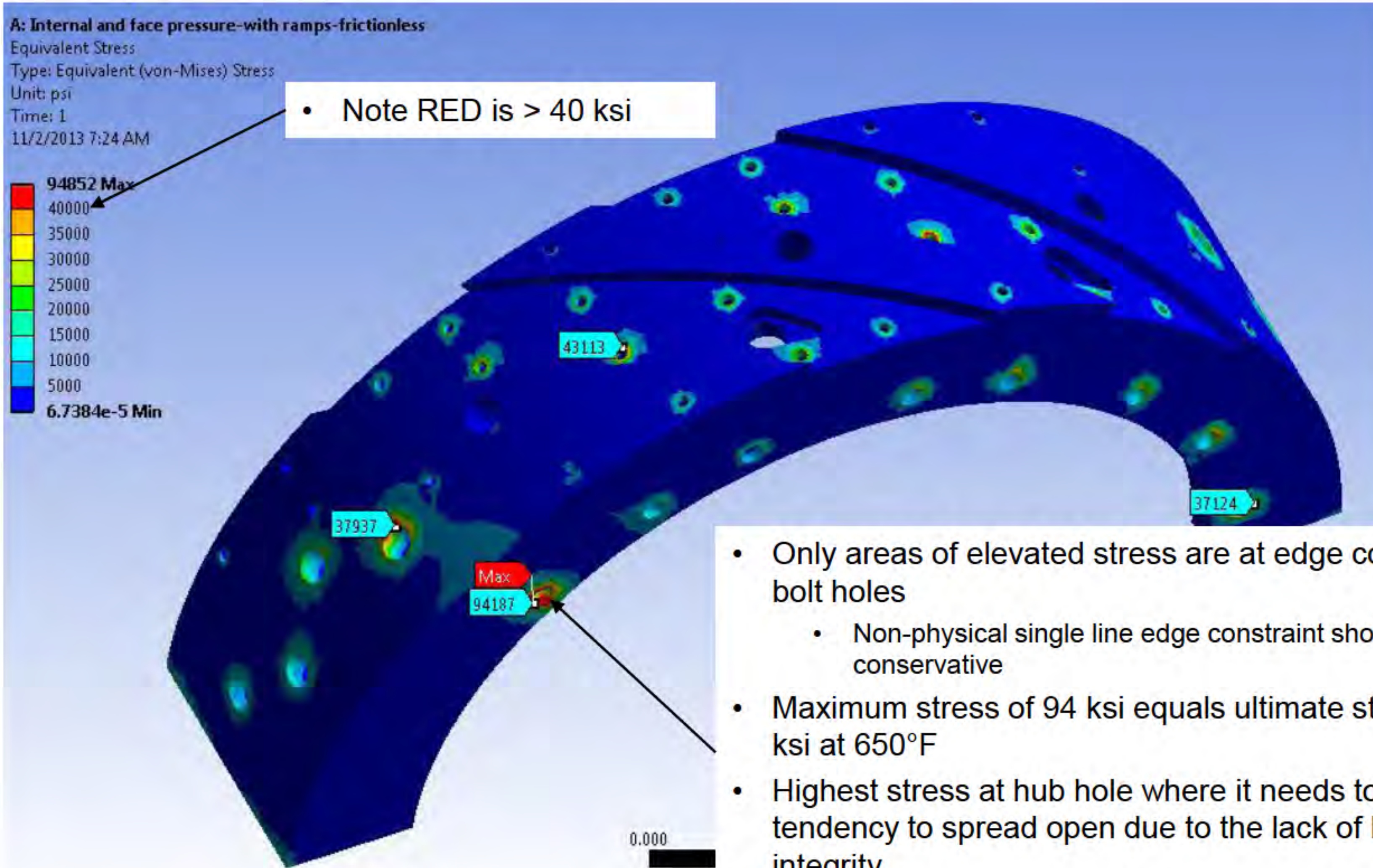


- 210 psi applied to interior surface of hub and shroud





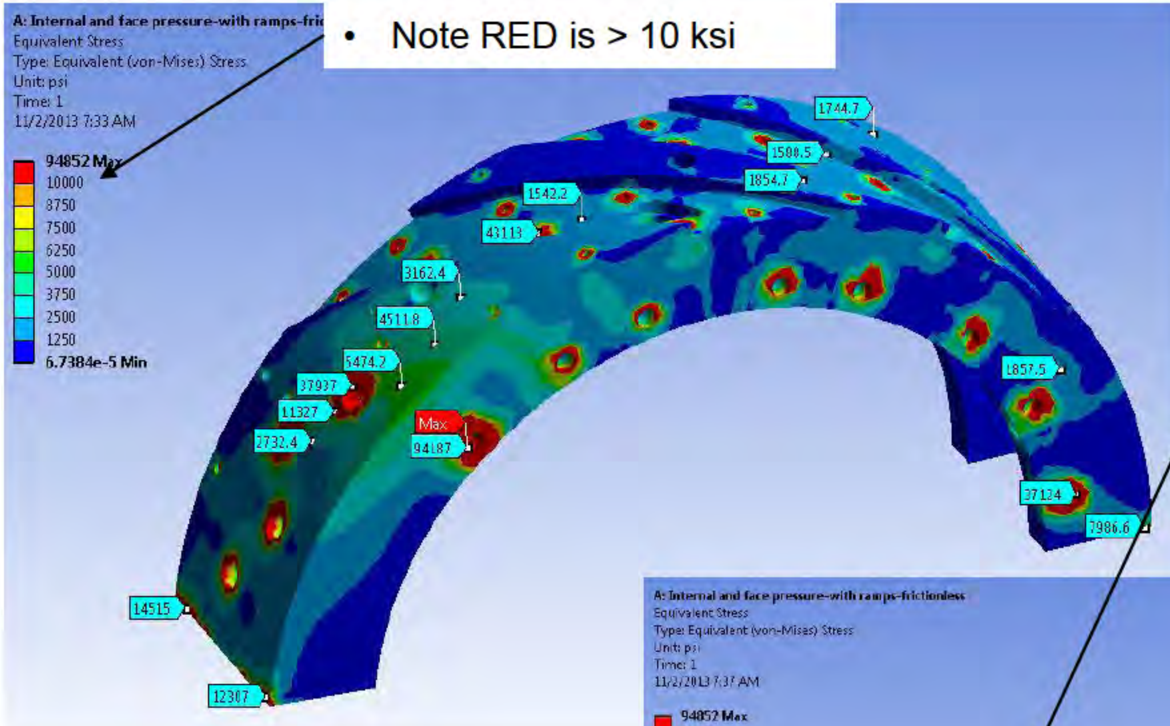
Equivalent Stress - Hub



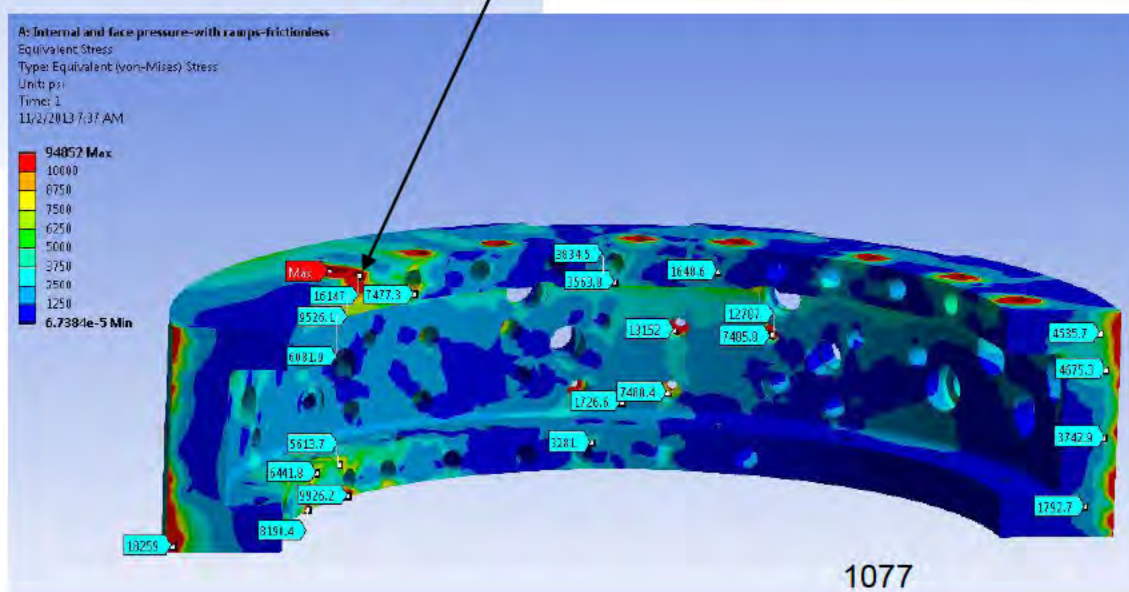
- Note RED is > 40 ksi

- Only areas of elevated stress are at edge constraints on bolt holes
  - Non-physical single line edge constraint should be conservative
- Maximum stress of 94 ksi equals ultimate strength of 93 ksi at 650°F
- Highest stress at hub hole where it needs to resist the tendency to spread open due to the lack of hoop integrity
- Ignores beneficial effect of tongue engagement to IGV and extra set of bolts to Measurement Ring

Equivalent Stress - Hub



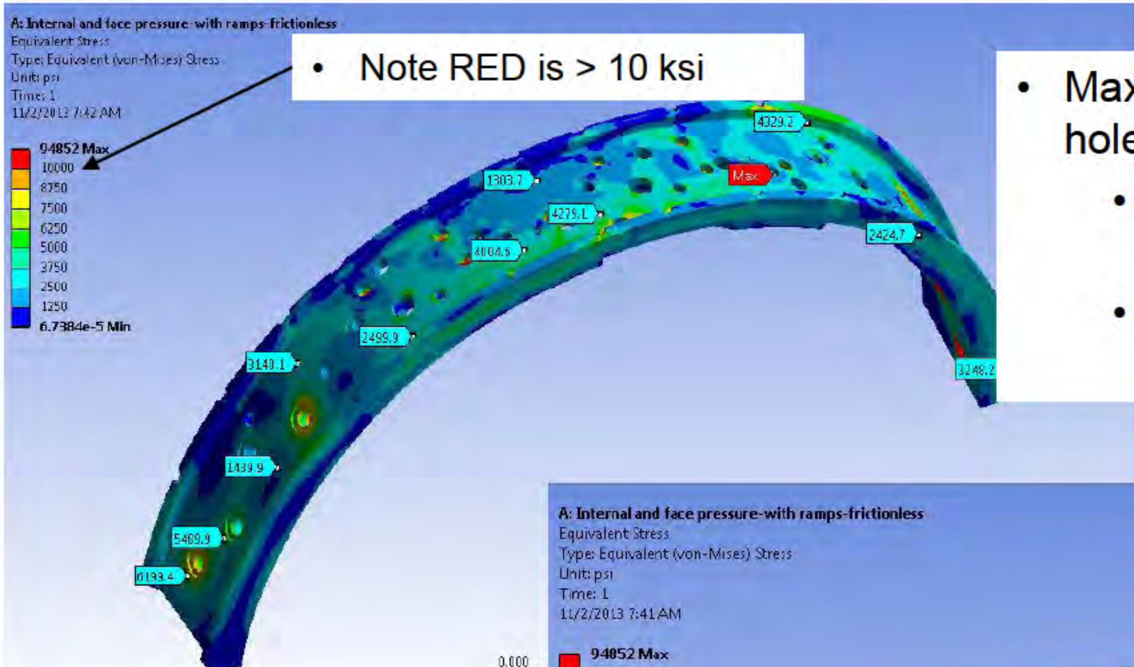
- Note RED is > 10 ksi



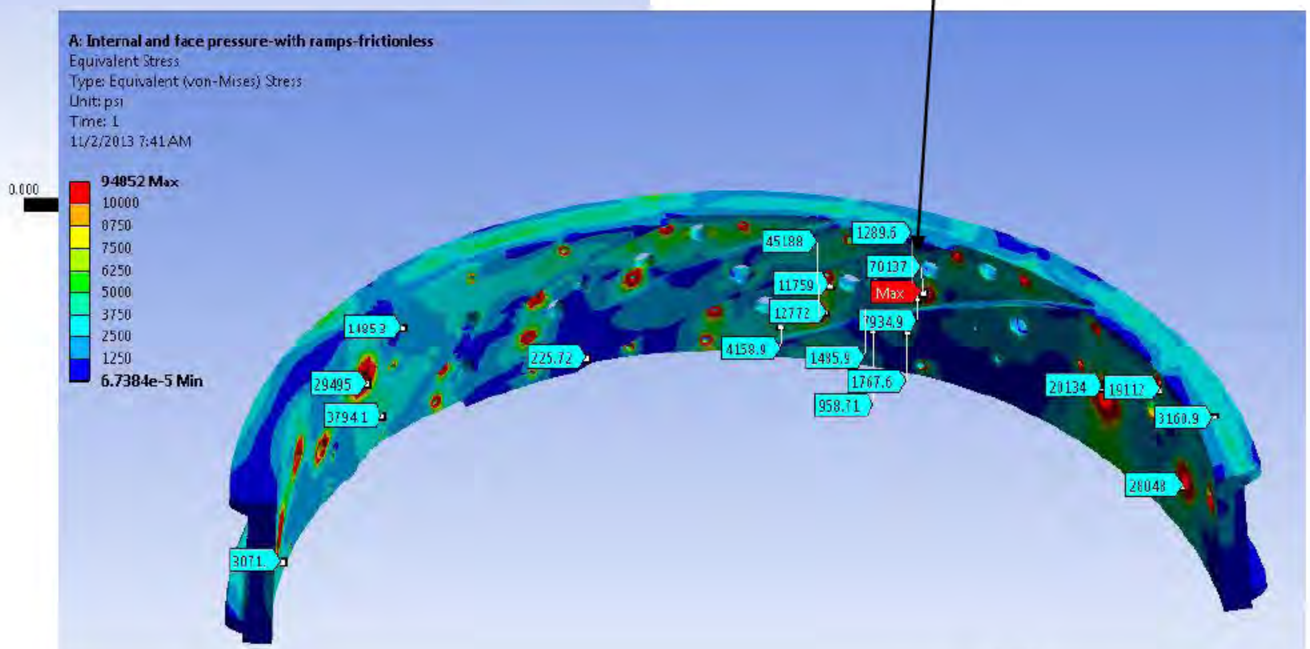
- Away from bolt holes and edge constraints, stress quickly becomes less than 16 ksi
- Most of hub is less than 10 ksi

Equivalent Stress - Shroud

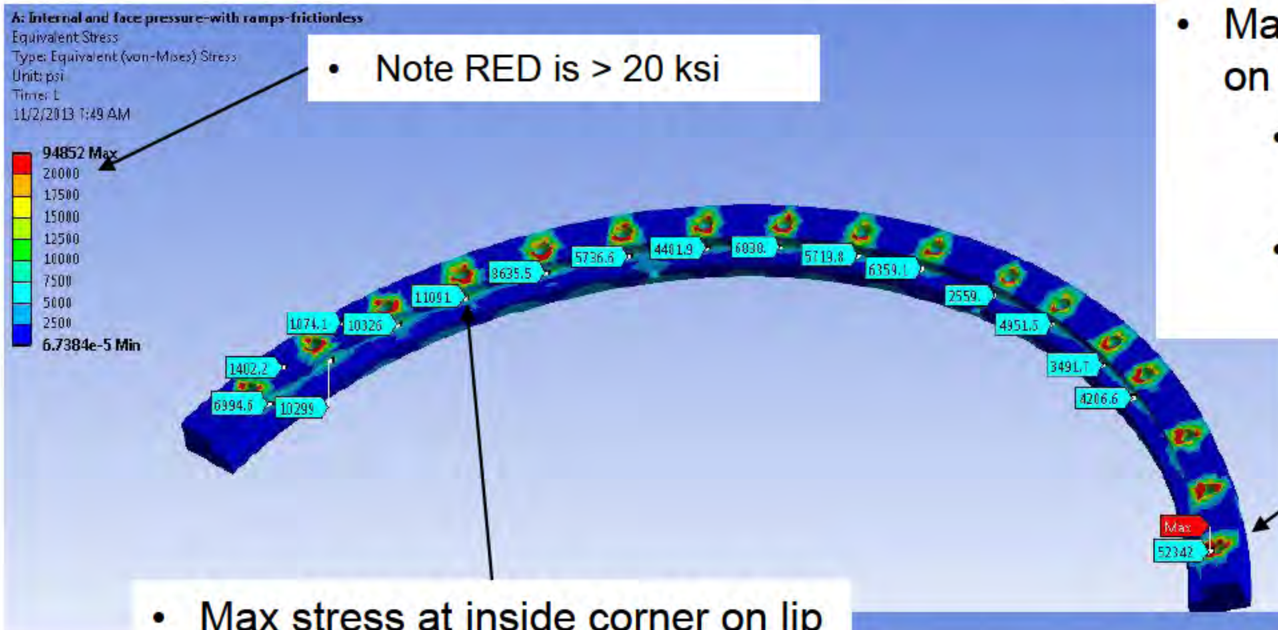
- Note RED is > 10 ksi



- Max stress at edge constraints on bolt holes is 70 ksi
  - Non-physical stress result due to contact constraint
  - Below 93 ksi strength of 17-4 at 650°F



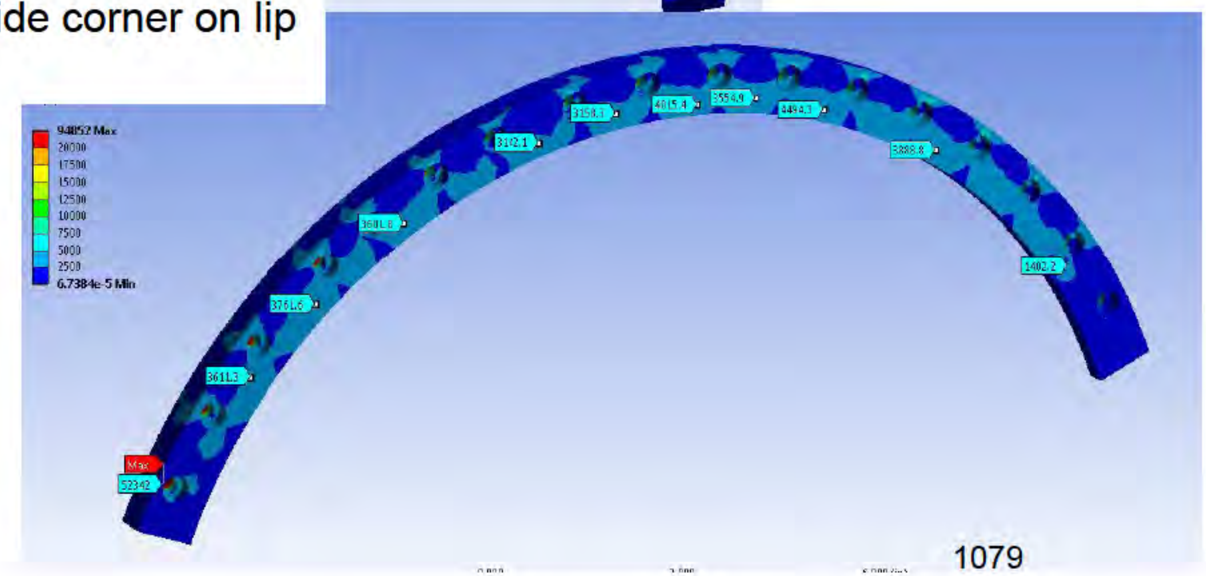
Equivalent Stress – Shroud Clamp



- Note RED is > 20 ksi

- Max stress at edge constraints on bolt holes is 52 ksi
    - Non-physical stress result due to contact constraint
    - Below 93 ksi strength of 17-4 at 650°F

- Max stress at inside corner on lip is 11 ksi



Strake Bolt Loads - Tabulation

Bolt Size:	.138-32		
Material:	A574		
Nominal Size:	0.138	in	
Torque Coefficient:	0.2		
Recommended Installation Torque:	24.5	lbf-in	(Unbrako Technical Brochure)
Resultant Preload at Room Temp:	887.7	lbf	
Estimated Elastic Modulus Derating at 650°F:	87%		(MIL-HDBK-5, AISI low-alloy steels)
Estimated Preload at 650°F:	772.3		
Room Temp Breaking Strength:	1640.0	lbf	
Breaking strength at 650°F:	1476.0	lbf	
Shear strength at 650°F:	108.0	ksi	
Section Area at Minor Diameter:	0.00745	in^2	(Mark's Standard Handbook)
Friction coefficient:	0.16		(Lubricated and greasy steel to steel)

Bolt ID	X (lbf)	Y (lbf)	Z (lbf)	Total (lbf)	Tensile Magnitude	Shear Force (X & Z) (lbf)	Tension % of Breaking Strength at 650°F:	Tension % of Preload at 650°F:	Shear Stress (psi)	Shear % of Shear Strength at 650°F:
SB1	20.2	232.4	-33.9	235.7	232.4	39.5	16%	30%	5297	5%
SB2	-28.6	535.0	105.1	546.0	535.0	108.9	36%	69%	14620	14%
SB3	-65.8	395.9	-70.4	407.5	395.9	96.4	27%	51%	12935	12%
SB4	1.6	154.1	-19.8	155.4	154.1	19.9	10%	20%	2666	2%
SB5	-51.1	254.7	-42.4	263.2	254.7	66.4	17%	33%	8913	8%
SB6	-80.0	454.7	-28.3	462.6	454.7	84.9	31%	59%	11390	11%
SB7	-54.2	521.5	-58.2	527.5	521.5	79.5	35%	68%	10675	10%
SB8	19.0	140.8	-88.6	167.4	140.8	90.6	10%	18%	12163	11%
SB9	20.1	294.9	-81.4	306.6	294.9	83.8	20%	38%	11254	10%
SB10	-59.6	471.6	-135.2	494.2	471.6	147.8	32%	61%	19833	18%
SB11	-58.8	361.0	-108.8	381.6	361.0	123.7	24%	47%	16600	15%
SB12	-13.6	152.4	-45.2	159.5	152.4	47.2	10%	20%	6336	6%
SB13	-59.9	207.6	-71.4	227.6	207.6	93.2	14%	27%	12510	12%
SB14	-147.9	464.3	-50.5	489.9	464.3	156.3	31%	60%	20978	19%
SB15	-97.3	604.9	-69.8	616.6	604.9	119.7	41%	78%	16073	15%
SB16	81.4	178.4	-122.0	230.9	178.4	146.7	12%	23%	19686	18%
HB1	26.9	-396.3	-125.4	416.5	396.3	128.3	27%	51%	17215	16%
HB2	69.9	-588.6	-99.0	600.9	588.6	121.2	40%	76%	16267	15%
HB3	95.9	-486.1	-14.9	495.7	486.1	97.1	33%	63%	13027	12%
HB4	60.4	-212.0	29.6	222.4	212.0	67.3	14%	27%	9029	8%
HB5	41.7	-326.4	28.2	330.3	326.4	50.3	22%	42%	6757	6%
HB6	11.6	-337.7	-12.7	338.1	337.7	17.2	23%	44%	2309	2%
HB7	-21.2	-356.9	-73.3	365.0	356.9	76.3	24%	46%	10242	9%
HB8	3.3	-132.5	-106.0	169.7	132.5	106.1	9%	17%	14235	13%
HB9	31.4	-377.0	-126.9	399.0	377.0	130.7	26%	49%	17547	16%
HB10	82.0	-580.0	-128.6	599.7	580.0	152.5	39%	75%	20472	19%
HB11	105.8	-466.6	-35.6	479.8	466.6	111.6	32%	60%	14984	14%
HB12	89.6	-186.4	-1.9	206.8	186.4	89.6	13%	24%	12030	11%
HB13	42.8	-267.7	-2.7	271.1	267.7	42.9	18%	35%	5756	5%
HB14	13.2	-369.1	-35.9	371.1	369.1	38.2	25%	48%	5134	5%
HB15	-31.1	-433.8	-92.0	444.5	433.8	97.1	29%	56%	13035	12%
HB16	-4.1	-178.4	-123.0	216.7	178.4	123.1	12%	23%	16519	15%
Max	**105.8**	**604.9**	**105.1**	**616.6**	**604.9**	**156.3**	**41%**	**78%**	**20977.7**	**19%**
Min	-147.9	-588.6	-135.2	155.4	132.5	17.2	9%	17%	2308.8	2%

- Results / Conclusions
  - SB15 has the highest total load
  - SB14 has the highest shear load
  - Maximum bolt tensile load of 605 lbf
    - 41% of 1476 lbf breaking strength
    - Less than 772 lbf preload gives positive clamping force
  - Maximum shear force of 156 lbf results in stress of 21 ksi, which is 19% of screw shear strength
- Conclusions
  - Strake screws will handle worst case pressure loads
  - Shear force on SE14 is 20% of initial preload. This is greater than typical friction of .16. Slippage is likely.
- *The following were not considered in this analysis given the margins, but could be added if desired*
  - *Combined tensile, shear, and bending moment*
  - *Remaining grip force under load*

Endwall Bolt Loads - Tabulation

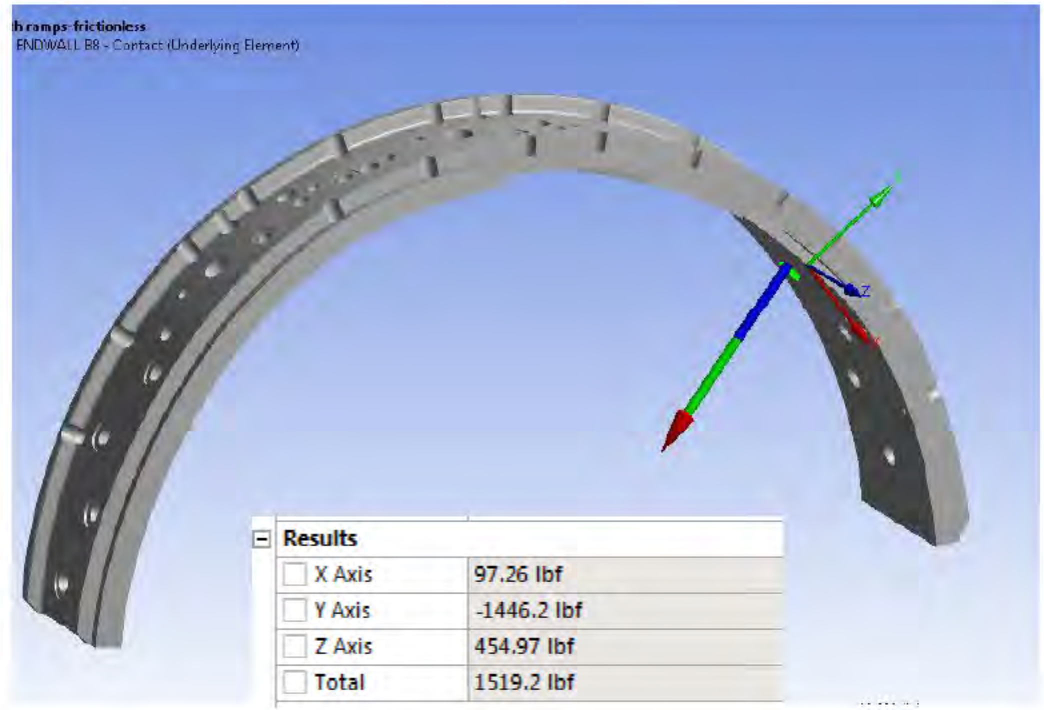
Bolt Size:	.375-16		
Material:	A574		
Nominal Size:	0.375	in	
Torque Coefficient:	0.2		
Recommended Installation Torque:	46.8	lbf-ft	(Unbrako Technical Brochure)
Resultant Preload at Room Temp:	7488.0	lbf	
Estimated Elastic Modulus Derating at 650°F:	87%		(MIL-HDBK-5, AISI low-alloy steels)
Estimated Preload at 650°F:	6514.6		
Room Temp Breaking Strength:	13900.0	lbf	
Breaking strength at 650°F:	12510.0	lbf	
Shear strength at 650°F:	108.0	ksi	
Section Area at Minor Diameter:	0.06780	in^2	(Mark's Standard Handbook)
Friction coefficient:	0.16		(Lubricated and greasy steel to steel)

Bolt ID	X (lbf)	Y (lbf)	Z (lbf)	Total (lbf)	Tensile Magnitude (lbf)	Shear Force (X & Z) (lbf)	Tension % of Breaking Strength at 650°F:	Tension % of Preload at 650°F:	Shear Stress (psi)	Shear % of Shear Strength at 650°F:
SE1	-91.0	-614.0	-289.0	684.7	614.0	303.0	5%	9%	4469	4%
SE2	-287.0	-1401.0	103.0	1433.8	1401.0	304.9	11%	22%	4497	4%
SE3	-233.0	-89.0	209.0	325.4	89.0	313.0	1%	1%	4617	4%
SE4	-312.0	-1731.0	128.0	1763.5	1731.0	337.2	14%	27%	4974	5%
SE5	115.0	-14.3	-233.0	260.2	14.3	259.8	0%	0%	3832	4%
SE6	47.0	-752.0	-79.0	757.6	752.0	91.9	6%	12%	1356	1%
SE7	134.0	-521.0	6.0	538.0	521.0	134.1	4%	8%	1978	2%
SE8	97.3	-1446.0	455.0	1519.0	1446.0	465.3	12%	22%	6863	6%
SE9	81.4	539.0	302.0	623.2	539.0	312.8	4%	8%	4613	4%
SE10	178.0	611.0	179.0	661.1	611.0	252.4	5%	9%	3723	3%
SE11	56.0	854.0	99.0	861.5	854.0	113.7	7%	13%	1678	2%
SE12	164.0	957.0	-88.0	974.9	957.0	186.1	8%	15%	2745	3%
SE13	36.0	610.0	-95.0	618.4	610.0	101.6	5%	9%	1498	1%
SE14	-131.0	678.0	167.0	710.4	678.0	212.2	5%	10%	3131	3%
SE15	167.0	969.0	505.0	1105.4	969.0	531.9	8%	15%	7845	7%
SE16	-148.0	1133.0	1405.0	1811.0	1133.0	1412.8	9%	17%	20837	19%
Max	**178.0**	**1133.0**	**1405.0**	**1811.0**	**1731.0**	**1412.8**	**14%**	**27%**	**20837.4**	**19%**
Min	-312.0	-1731.0	-289.0	260.2	14.3	91.9	0%	0%	1355.8	1%

- Results / Conclusions
  - SE4 has the highest tensile load
  - SE16 has the highest shear load
  - Maximum bolt tensile load of 1731 lbf
    - 14% of 12510 lbf breaking strength
    - Less than 6515 lbf preload gives positive clamping force
  - Maximum shear force of 1413 lbf results in stress of 21 ksi, which is 19% of screw shear strength
- Conclusions
  - Endwall screws will handle worst case pressure loads
  - Shear force on SE16 is 22% of initial preload. This is greater than typical friction of .16. Slippage is likely without shear feature.
- *The following were not considered in this analysis given the margins, but could be added if desired*
  - *Combined tensile, shear, and bending moment*
  - *Remaining grip force under load*

Endwall Bolt Loads – Max Shear Force at Shroud

- Maximum shear force on the shroud occurs at bolt SE8 and is 465 lbf
- Bolt load is mostly tensile at this location
- Shear load is 7% of 6514 lbf initial preload
  - Frictional force (4 * .16 * 6514 = 4168 lbf) of the 4 bolts on each end of the shroud should be sufficient to resist shear force
- Pin could be added to take shear if desired
  - Would need to be a match-drilled, serial operation after initial dry assembly of nozzle
  - Doesn't seem necessary given multiple failure scenarios necessary to create this condition



Hub Bolts to IGV- Tabulation

Bolt Size	.250-20		
Material	A574		
Nominal Size	0.250	in	
Torque Coefficient	0.2		
Recommended Installation Torque	153.0	lbf-in	(Unbrako Technical Brochure)
Resultant Preload at Room Temp	3060.0	lbf	
Estimated Elastic Modulus Derating at 650°F	87%		(MIL-HDBK-5, AISI low-alloy steels)
Estimated Preload at 650°F	2662.2		
Room Temp Breaking Strength	5700.0	lbf	
Breaking strength at 650 F	5130.0	lbf	
Shear strength at 650 F	108.0	ksi	
Section Area at Minor Diameter	0.02690	in^2	(Mark's Standard Handbook)
Friction coefficient	0.16		(Lubricated and greasy steel to steel)

Bolt ID	X (lbf)	Y (lbf)	Z (lbf)	Total (lbf)	Tensile Magnitude (lbf)	Shear Force (X & Y) (lbf)	Tension % of Breaking Strength at 650°F:	Tension % of Preload at 650°F:	Shear Stress (psi)	Shear % of Shear Strength at 650°F:
IGVH1	962.0	83.0	-1169.0	1516.2	1169.0	965.6	23%	44%	35895	33%
IGVH2	530.0	-46.0	-1255.0	1363.1	1255.0	532.0	24%	47%	19777	18%
IGVH3	339.0	-219.0	-1234.0	1298.3	1234.0	403.6	24%	46%	15003	14%
IGVH4	443.0	-186.0	-1120.0	1218.7	1120.0	480.5	22%	42%	17861	17%
IGVH5	-20.0	-442.0	-992.0	1086.2	992.0	442.5	19%	37%	16448	15%
IGVH6	56.0	-275.0	-785.0	833.7	785.0	280.6	15%	29%	10433	10%
IGVH7	-99.0	-149.0	-798.0	817.8	798.0	178.9	16%	30%	6650	6%
IGVH8	-1893.0	-1089.0	-1474.0	2634.8	1474.0	2183.9	29%	55%	81185	75%
Max	**962.0**	**83.0**	**-785.0**	**2634.8**	**1474.0**	**2183.9**	**29%**	**55%**	**81185.5**	**75%**
Min	-1893.0	-1089.0	-1474.0	817.8	785.0	178.9	15%	29%	6650.2	6%

- Results / Conclusions
  - IGVH8 has the highest total load and highest shear load
  - Maximum bolt tensile load of 1474 lbf
    - 29% of 5130 lbf breaking strength
    - Less than 2662 lbf preload gives positive clamping force
  - Maximum shear force of 2183lbf results in stress of 81 ksi, which is 75% of screw shear strength
- Conclusions
  - Shear component too high to consider bolted interface to IGV only
  - Bolted interface to measurement section should cut bolt loads by about half
  - Tongue/groove engagement required take majority of shear load
  - Bolts will take the tensile load with substantial margin
- *The following were not considered in this analysis given the margins, but could be added if desired*
  - *Combined tensile, shear, and bending moment*
  - *Remaining grip force under load*

Hub and Shroud Bolts to IGV- Tabulation

Bolt Size	.250-20		
Material	A574		
Nominal Size	0.250	in	
Torque Coefficient	0.2		
Recommended Installation Torque	153.0	lbf-in	(Unbrako Technical Brochure)
Resultant Preload at Room Temp	3060.0	lbf	
Estimated Elastic Modulus Derating at 650°F	87%		(MIL-HDBK-5, AISI low-alloy steels)
Estimated Preload at 650°F	2662.2		
Room Temp Breaking Strength	5700.0	lbf	
Breaking strength at 650 F	5130.0	lbf	
Shear strength at 650 F	108.0	ksi	
Section Area at Minor Diameter	0.02690	in^2	(Mark's Standard Handbook)
Friction coefficient	0.16		(Lubricated and greasy steel to steel)

Bolt ID	X (lbf)	Y (lbf)	Z (lbf)	Total (lbf)	Tensile Magnitude (lbf)	Shear Force (X & Y) (lbf)	Tension % of Breaking Strength at 650°F:	Tension % of Preload at 650°F:	Shear Stress (psi)	Shear % of Shear Strength at 650°F:
IGVS1	-89.0	-231.0	-631.0	677.8	631.0	247.6	12%	24%	9203	9%
IGVS2	-89.0	-259.0	-248.0	369.5	248.0	273.9	5%	9%	10181	9%
IGVS3	-90.0	-67.0	-316.0	335.3	316.0	112.2	6%	12%	4171	4%
IGVS4	-69.0	77.0	-379.0	392.8	379.0	103.4	7%	14%	3844	4%
IGVS5	-42.0	47.0	320.0	326.1	320.0	63.0	6%	12%	2343	2%
IGVS6	-9.0	90.0	280.0	294.2	280.0	90.4	5%	11%	3362	3%
IGVS7	-42.0	55.0	221.0	231.6	221.0	69.2	4%	8%	2573	2%
IGVS8	-88.0	43.0	-274.0	291.0	274.0	97.9	5%	10%	3641	3%
IGVS9	-56.0	62.0	-342.0	352.1	342.0	83.5	7%	13%	3106	3%
IGVS10	-45.0	94.0	-428.0	440.5	428.0	104.2	8%	16%	3874	4%
IGVS11	-3.0	83.0	-352.0	361.7	352.0	83.1	7%	13%	3088	3%
IGVS12	-28.0	92.0	-333.0	346.6	333.0	96.2	6%	13%	3575	3%
IGVS13	7.0	93.0	-430.0	440.0	430.0	93.3	8%	16%	3467	3%
IGVS14	48.0	75.0	-432.0	441.1	432.0	89.0	8%	16%	3310	3%
IGVS15	139.0	15.0	-436.0	457.9	436.0	139.8	8%	16%	5197	5%
IGVS16	210.0	-103.0	-293.0	374.9	293.0	233.9	6%	11%	8695	8%
IGVS17	245.0	-156.0	-317.0	429.9	317.0	290.4	6%	12%	10797	10%
Max	**245.0**	**94.0**	**320.0**	**677.8**	**631.0**	**290.4**	**12%**	**24%**	**10797.4**	**10%**
Min	-90.0	-259.0	-631.0	231.6	221.0	63.0	4%	8%	2343.2	2%

- Results / Conclusions
  - IGVS1 has the highest total load
  - Maximum bolt tensile load of 1631 lbf
    - 12% of 5130 lbf breaking strength
    - Less than 2662 lbf preload gives positive clamping force
  - Maximum shear force of 290 lbf results in stress of 11 ksi, which is 10% of screw shear strength

Conclusions

- Bolts have adequate margin for worst case loads

The following were not considered in this analysis given the margins, but could be added if desired

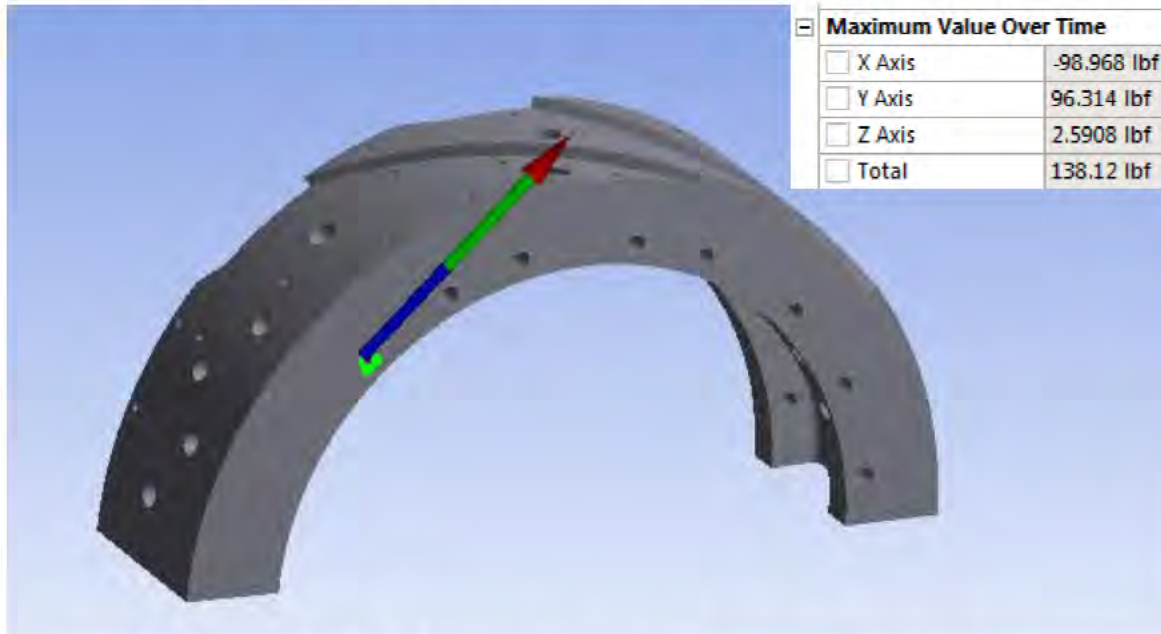
- *Combined tensile, shear, and bending moment*
- *Remaining grip force under load*

Load Case #2, Nominal Pressure Load

- 83.4 psi on flow passage ramp surfaces
  - Simulates structural load on structure of 83.4 psi cooling air in impingement cavities
- Analysis done with room temperature properties
  - Results compared to derated properties at 650°F
- Objective
  - Nominal loads on highest load bolts identified in worst case-analysis
- Assumptions
  - Ignored tongue & grove interface to IGV and Measurement Ring
  - Ignored measurement ring

Hub Bolts

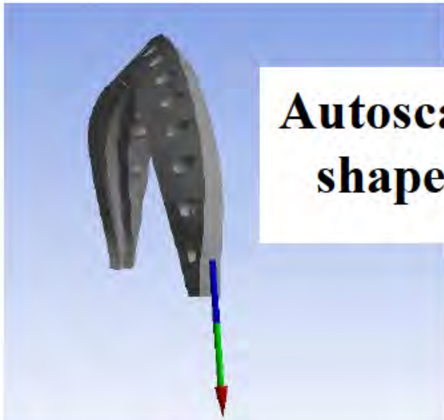
IGVH8 (had highest worst case load of 2184 lbf)



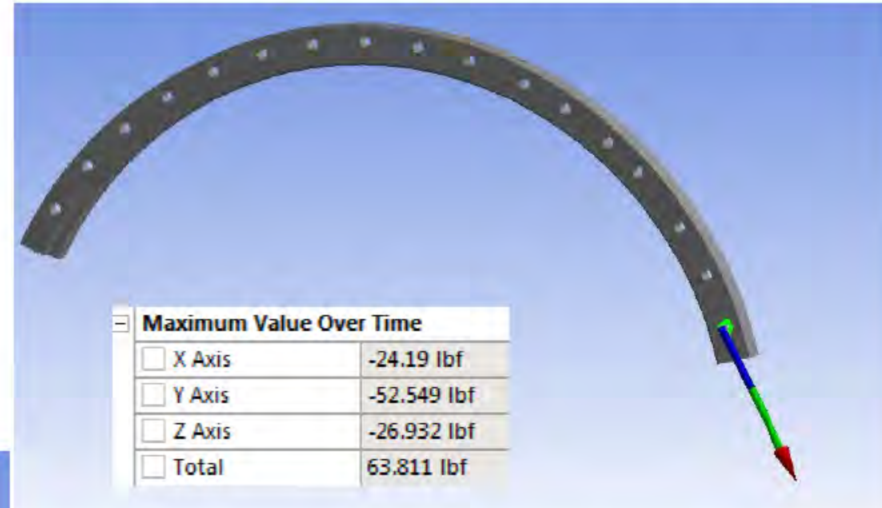
Maximum Value Over Time	
X Axis	-98.968 lbf
Y Axis	96.314 lbf
Z Axis	2.5908 lbf
Total	138.12 lbf

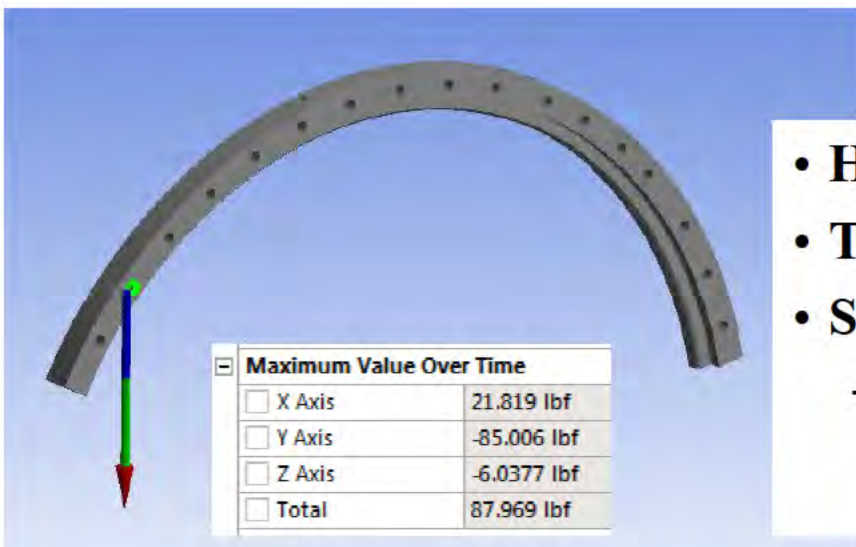
- **Highest total load remains on IGVH8**
- **Total reaction force on bolt is 138 lbf**
- **Shear component is 138 lbf**

Shroud Bolts



IGVS8 (had highest worst case load of 678 lbf)





- **Highest total load is on IGVS16**
- **Total reaction force on bolt is 88 lbf**
- **Shear component is 88 lbf**
  - **Due to twisting from lack of hoop integrity**

Load Case #3, Strake Only

- 210 psi on all internal strake surfaces
  - Assumes regulator fails wide open (will be set at about 190 psia)
    - Note: Nominal strake supply pressure should be ~85 psia max
  - Assumes strake cooling air supply valve is set wide open
- Analysis done with room temperature properties
  - Results compared to derated properties at 650°F
- Objective
  - Stress distribution
- Assumptions
  - No pressure loss along the length (only happens if TE slots get plugged)
  - Uniform pressure distribution
  - Vacuum conditions in primary flow path

Load Case #4, Ramp Only

- 210 psi on all internal ramp surfaces
  - Assumes regulator fails wide open (will be set at about 190 psia)
    - Note: Nominal ramp supply pressure should be ~85 psia max.
  - Assumes ramp cooling air supply valve is set wide open
- Analysis done with room temperature properties
  - Results compared to derated properties at 650°F
- Objective
  - Stress distribution
- Assumptions
  - Uniform pressure distribution
  - Vacuum conditions in primary flow path

Thermal Analysis

– **See <u>FDR Nozzle Test Section - Thermal_R1.pptx</u>**

Tolerance Analysis / Hot-Cold Variation

Throat Tolerance Anlaysis

- **Performed worst-case and statistical tolerance analysis on throat and exist size variation and resultant variation in expansion ratio**
- **Assumed aggressive 0.002" surface profile tolerance on critical interface features in the radial direction**
  - **Consistent with prior Ramgen experience for critical features**
- **Assumed .020" surface profile tolerance on DMLS'd strake vanes**

Predicted Throat (and exit) Height Variation

	Dimensions in inches		
Height Variation (Throat and Exit)			
ID	**Profile Tolerances**		**Assumptions**
1	OD of hub slot for strake flange	0.002	Mating surfaces are machined
2	OD of hub-side strake flange	0.002	Mating surfaces are machined
3	OD of hub-side impingement plate flange	0.002	Mating surfaces are machined
4	OD of hub-side ramp	0.002	Mating surfaces are machined
5	ID of shroud-side strake flange	0.002	Mating surfaces are machined
6	ID of shroud-side impingement plate flange	0.002	Mating surfaces are machined
7	ID of shourd-side ramp	0.002	Mating surfaces are machined
	INSERT ROWS ABOVE		
	Total Profile Tolerance - Worst Case	0.014	
	Bi-lateral tolerance- Worst Case (+/-)	0.007	
	Root Sum Square (RSS) Profile Tolerance:	0.005	
	RSS Bi-lateral tolerance- Worst Case (+/-)	0.003	
	Modified RSS (MRSS) Profile Tolerance		
	Multiplier Factor (historical)	1.4	
	MRSS Profile Tolerance:	0.007	
	MRSS Bi-lateral tolerance- Worst Case (+/-)	0.004	

- Worst case predicted variation is ±.007”
- Statistically likely variation is ±.004”

Predicted Throat (and exit) Width Variation

Width Variation (Throat and Exit)			
ID	**Profile Tolerances**		**Assumptions**
A	Profile tolerance on strake width, Side 1	0.020	DMLS as-built assuming Z direction worst-case build
B	Profile tolerance on strake width, Side 2	0.020	DMLS as-built
C	Positional Variation	0.020	Estimate (+/- .010)
	INSERT ROWS ABOVE		
	Total Profile Tolerance - Worst Case	0.060	
	Bi-lateral tolerance- Worst Case (+/-)	0.030	

- Worst case predicted variation is ±.030” on throat width

Differential Thermal Expansion

- Two possible approaches to deal with differential thermal expansion
  - Size all gaps to tolerance worst-case differential thermal expansion created by rapid heat-up
    - 580°F temperature increase on ramps/strakes while hub/shroud remain at room temperature
    - The gaps will close during heat-up, then reappear when thermal equilibrium is reached
  - Size gaps to tolerance a smaller thermal gradient and heat up slowly
- Recommendation
  - Ramp up slowly and maintain less than a 100°F gradient across the nozzle at all times
  - Propose ramp up cycle
    - 6 steps to get from room temperature to full 650°F operating temperature (97°F each)
    - 45 minute dwell at each step
      - » 580°F step reached near equilibrium in about 60 minuts
      - » Assumes the smaller temperature step will reach equilibrium faster
      - » Can manually monitor temperatures through-out nozzle and make real-time adjustments

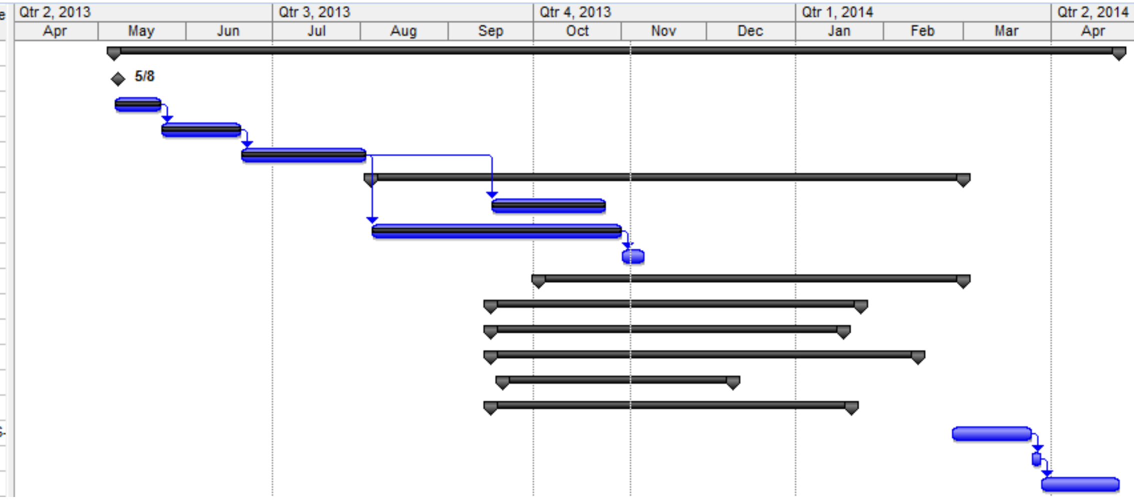
Interfaces

Requirements

See link: ..\..\..\Specification-Compliance\Nozzle Requirements and Compliance Matrix 20131003.xlsx

Schedule

Task Name	Duration	Start	Finish	e
Nozzle Sector Component Characterization (8610)	238.5 days?	Mon 5/6/13	Thu 4/24/14	
Kick off meeting	0 days	Wed 5/8/13	Wed 5/8/13	
System Requirements Review (SRR)	2.5 wks	Mon 5/6/13	Wed 5/22/13	
Conceptual Design Review (CDR)	4 wks	Thu 5/23/13	Wed 6/19/13	
Preliminary Design Review (PDR)	6.2 wks	Thu 6/20/13	Fri 8/2/13	
Nozzle (Rick)	136 days	Mon 8/5/13	Fri 2/28/14	
Structural analysis, thermal design & draft dwgs	6 wks	Mon 9/16/13	Fri 10/25/13	
Final Design Review (FDR)	12.6 wks	Mon 8/5/13	Thu 10/31/13	
Production Readiness Review (PRR)	1.2 wks	Fri 11/1/13	Fri 11/8/13	
Manufacturing	94 days	Thu 10/3/13	Fri 2/28/14	
Flow Uniformization and Converging Section (Aaro	82 days	Mon 9/16/13	Thu 1/23/14	
Inlet Guide Vane Section (Chris)	78 days?	Mon 9/16/13	Fri 1/17/14	
Downstream Pressure Measurment Section (Rob)	96 days	Mon 9/16/13	Wed 2/12/14	
Facility (Karl)	55 days	Fri 9/20/13	Mon 12/9/13	
Control Software Development (Jilka)	79 days	Mon 9/16/13	Mon 1/20/14	
Assembly & Instrumentation	4 wks	Tue 2/25/14	Mon 3/24/14	S-
Checkout	3 days	Tue 3/25/14	Thu 3/27/14	
Test	4 wks	Fri 3/28/14	Thu 4/24/14	



- It took about 2 months longer than planned to get to nozzle FDR
- This has pushed delivery of components to the end of Feb 2014
- With assembly and instrumentation through-out the month of March, testing is now scheduled to start at the end of March
  - Slipping into April is likely unless manufacturing schedules can be improved
- Plan to prioritize release of ramp & impingement plate drawings (ECD 11/8)
- Then follow 11/15 with PRR and release of remaining drawings

Budget

RAMGEN POWER SYSTEMS ™

TURBINE NOZZLE COMPONENT TEST

WBS		ODC	Equipment	Travel	Notes
	ODC Design Support				
	Design				
	Aero/thermal Support	$ 20,000	$ -	$ -	Occasional travel for consultation/manufacturing/procurement support
	Static Hardware				
	Drafting support	$ 9,600	$ -	$ -	
	Assembly				
	Test rig assembly	$ 10,000		$ -	Aerodyne instrumentation support & assembly tooling
	Shipping				
	General Shipping	$ -	$ -	$ -	
	Task total	**$ 39,600**	**$ -**	**$ -**	
	B2 Engine Compressor Module Static Hardware				
	Fabricated Components				
	Nozzle test section		$ 166,582		
	Support hardware		$ 79,860		
	Instrumentation		$ 56,906		
	Spares		$ 21,500		
	Shipping	$ 5,000	$ -	$ -	
	Task total	**$ 5,000**	**$ 324,848**	**$ -**	
	Facility				
	Facility air system modifications	$ 22,000	$ 28,000		
	Task total	**$ 22,000**	**$ 28,000**	**$ -**	
	Compressor Module Test				
	Electricity Costs			$ -	4 week test program assumed
	Energy charge ($.06523/kW-Hr)	$ 5,464			30 hrs/week assumed (compressor, heater and vacuum system running)
	Demand charge ($6.08/kW)	$ 5,375			Monthly demand charge set by running 338 kW compressor, 360 kW heater and 186 kW Vacuum system s
	Task total	**$ 10,838**	**$ -**	**$ -**	
	Total of all tasks	$ 77,438	$ 352,848	$ -	
		$ 430,286	**Program Grand Total**		

- Based on ROM quotes for ramps, impingement plates, and strakes, and PR's submitted to date (Settling chamber, Converging Section) costs are tracking the budget
- Better fidelity available within the next few weeks has PR's get submitted for the majority of the hardware

Piece Part Drawing Status

Nozzle Piece Part Dwgs			
Drawing No	Description	Draft Complete	Target Release Date
861022	Impingement Plate, Hub	Y	11/8/2013
861023	Impingement Plate, Shroud	Y	11/8/2013
861024	Ramp Body, Hub	Y	11/8/2013
861025	Ramp Body, Shroud	Y	11/8/2013
861034	Hub Body	Y	11/15/2013
861036	Nozzle Vane, Machined	Y	11/15/2013
861037	Nozzle Vane, As Built (DMLS)	Y	11/15/2013
861039	Nozzle Vane Insert	In work	11/15/2013
861040	Shroud Body - 1 Pc	Y	11/15/2013
861041	Endwall, Left	In work	11/15/2013
861042	Endwall, Right	In work	11/15/2013
861043	Clamp, Shroud, Upstream	In work	11/15/2013
861048	Clamp, Shroud, Downstream	In work	11/15/2013

- **Required assembly drawings (tubing brazements, instrumentation) to follow by 12/15/2013**

Conclusions

- **Design closes and meets program requirements**
- **Ready to proceed with final design tweaks and detailed drawing completion pending the results of this review**

Backup Items

Notes from Day 1

- **Several action items were take. See Action Item spreadsheet for details.**
- **Paul indicated that copper RTV was used on the Ram2 program. It tends to be a little more "crumbly" in the cured state than the red RTV, but worked.**
- **There was discussion on the tolerances for impingement holes in the strake leading edge insert. Since these holes do not meter the air, Aero (Silvano) agreed that we'll take what we get. Presumably the tolerance will be ±.002.**
- **The strake leading edge insert will likely be fabricated by plunge EDM due to the aspect ratio**

Open Tasks for Each Major Part

Design Details to be Completed

- **Strake**
  - **Update/review flange chamfer size – DONE 11/7. FROM .020 TO .075**
  - **Update diameter of thermocouple holes in strake based on prototype results – Per input from Directed Manuf, increase hole dia to .040.**
  - **Add cross-passage in strake wall at bottom of thermocouple passage**
    - **Leading edge too?? - CANCELLED 11/15/2013. DRAWING TO SPECIFY A MINIMUM HOLE DIAMETER TO BE VERIFIED**
  - **Review and update tube counter-bore diameters and depth in strake based on tube size – DONE. Target size for all instrument counterbores ∅.046±.002 x .10 deep. ∅.050 hole in prototype was good fit to .042 tubing. Size left at ∅.050.**
  - **Update radius from strake to flange from R.030 to R.047 DONE 11/11/2013**
  - **Update flange width for clearance – DONE 11/18/2013 (Increased from .010 to .020 nominal clearance)**
  - **Update interface to strake insert for clearance and error-proof assy – DONE 11/18/2013**
  - **Update as built (DMLS) configuration, post-machining req't (flange only), and datums – DONE 11/20/2013**

Design Details to be Completed

- **Impingement Plate (Hub and Shroud) – DONE 11/13/2013**
  - **Adjust impingement holes and instrument holes – DONE 11/11/2013**
  - **Set-back forward and aft face by .002” for .007” set-back from hub and shroud – DONE 11/7/2013**
  - **Review and update tube counter-bore diameters based on tube size – DONE 11/11/2013**

Design Details to be Completed

- Ramp (hub and shroud) – DONE 11/13/2013
  - Adjust film holes and instrument holes – DONE 11/11/2013
  - Set-back forward and aft face by .002” for .007” set-back from hub and shroud – DONE 11/7/2013
- Hub
  - Update instrumentation egress on hub – Done 11/6/2013
  - Add interlocking lip to endwall – Done 11/16/2013
  - Route tubing to avoid total pressure measurement section components (does not affect hub)
  - Add jacking screws to remove from IGV section - Done 11/15/2013
  - Design lifting platform or hanger to move nozzle (~80 lbm’s) (does not affect hub)
- Endwalls
  - Add interlocking lip to hub – Done 11/16/2013
  - Set-back forward and aft face by .002” for .007” set-back from hub and shroud – CANCELLED 11/7/2013. NOT SUBJECT TO DIFFERENTIAL THERMAL GROWTH LIKE STRAKE/RAMPS.

APPENDIX 12.3

ISCE Build 2 Design

Updated System Requirements

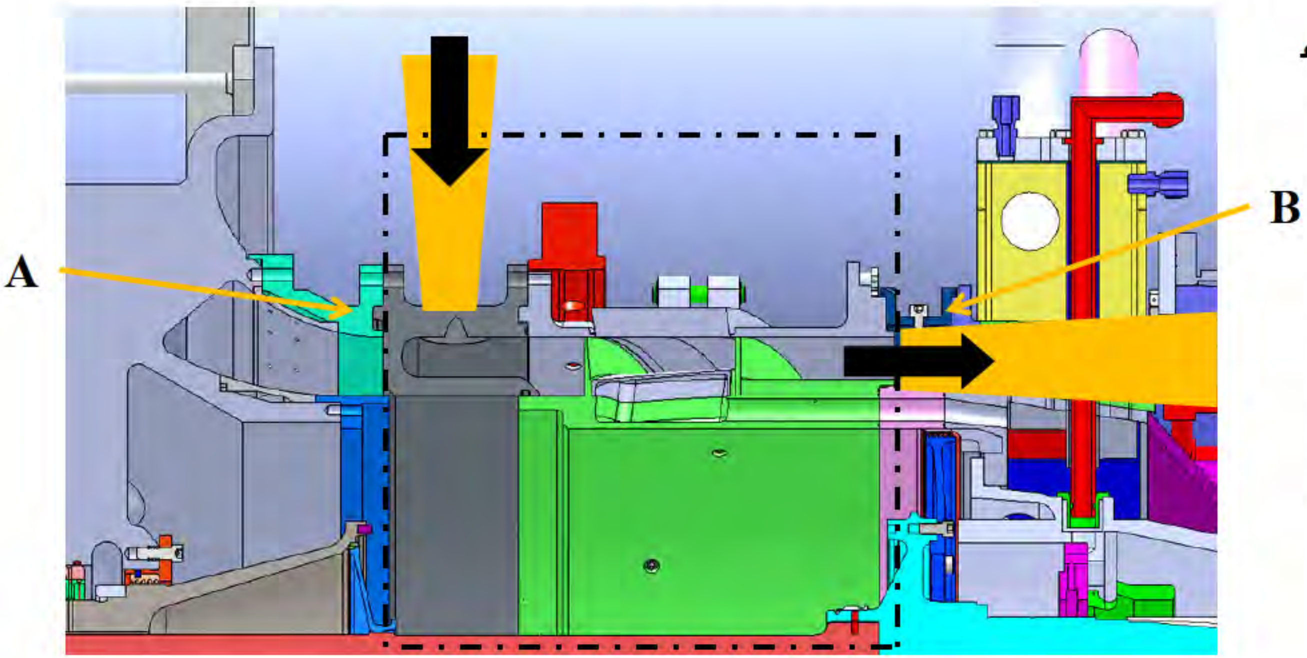
1. **Deliver 87 air at 10 lb/s to static combustor with equivalent flow uniformity to existing OGV output**
2. **Accommodate up to 200 psi internal pressure (SF)**
3. **Admit high-pressure air from auxiliary air system**
4. **Not interfere with existing seal and thrust flows**
5. **Not interfere with existing bearings**
6. **Maintain identical or greater stiffness relative to existing components**
7. **Operate with internal environments up to 325 F**
8. **Operate safely**
9. **Accommodate additional axial load due to internal pressure**
10. **Accommodate thermal and axial growth of rotor section.**
11. **Accommodate additional vertical load from Air inlet system**

Action Items

- **Assess Instrumentation requirements**
  - No new provisions needed
- **Determine need to reinstall bleed covers**
  - Bleed covers determined not be to a requirement but ideal, an opportunity to reinstall given tear-down was seized upon.
- **Spec O-ring seals to ensure proper dimensions**
  - Completed, see slide 11-12
- **Stager inlet tubes**
  - Completed, to be assembled on site and modifications required of off-the-self-part only
- **Perform analysis of Diffuser and shroud structure to ensure strength**
  - Completed, see analysis slide
- **Change inlet material to carbon steel**
  - Completed, see material slide 14.
- **Analyze impact on engine structure of loads due to pressurized inlet**
  - Completed, see analysis slide(s)

Final System Boundaries and Interfaces

- System boundaries fully defined from existing aluminum inlet addition[A] to existing OGV cover[B] and all components contained therein; this enables inclusion of components that will need to be modified to provide adequate sealing for the internal pressure.
- System interfaces with aux air system (which provides adequate air to meet requirement #1) and annular entry to combustor.



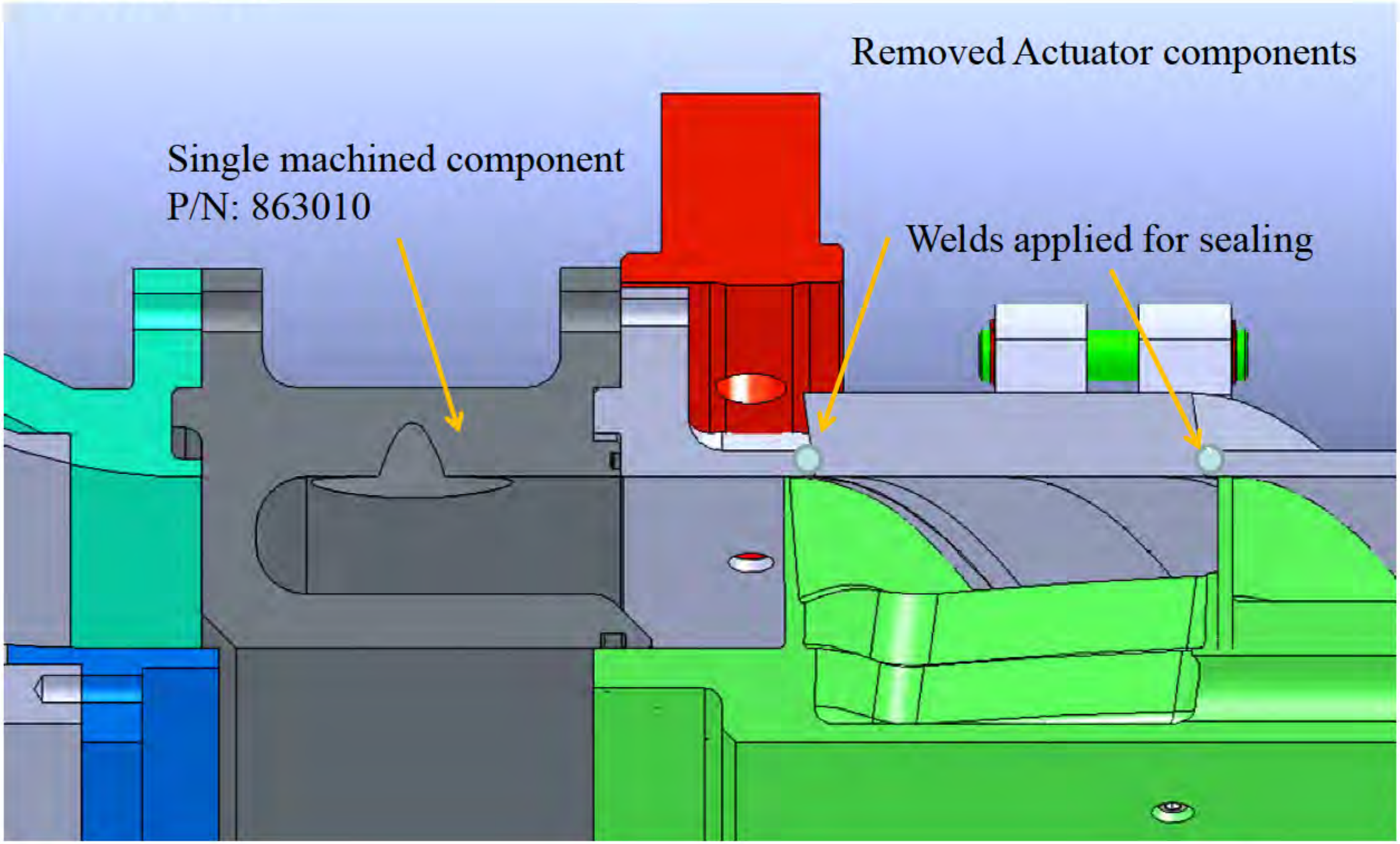
- ~~No additional instrumentation required, but existing lines in system boundary can be re-utilized (i.e. hub static pressure ports etc.)~~

Final System Boundaries and Interfaces

- Aux air system under design (see air system CDR). It is anticipated that a close aboard header will be provided to link to.
- If this varies additional hoses and/or pipes can be used to reach source.
- System will necessarily apply structural loads to other systems in the rig (detailed in analysis section.)

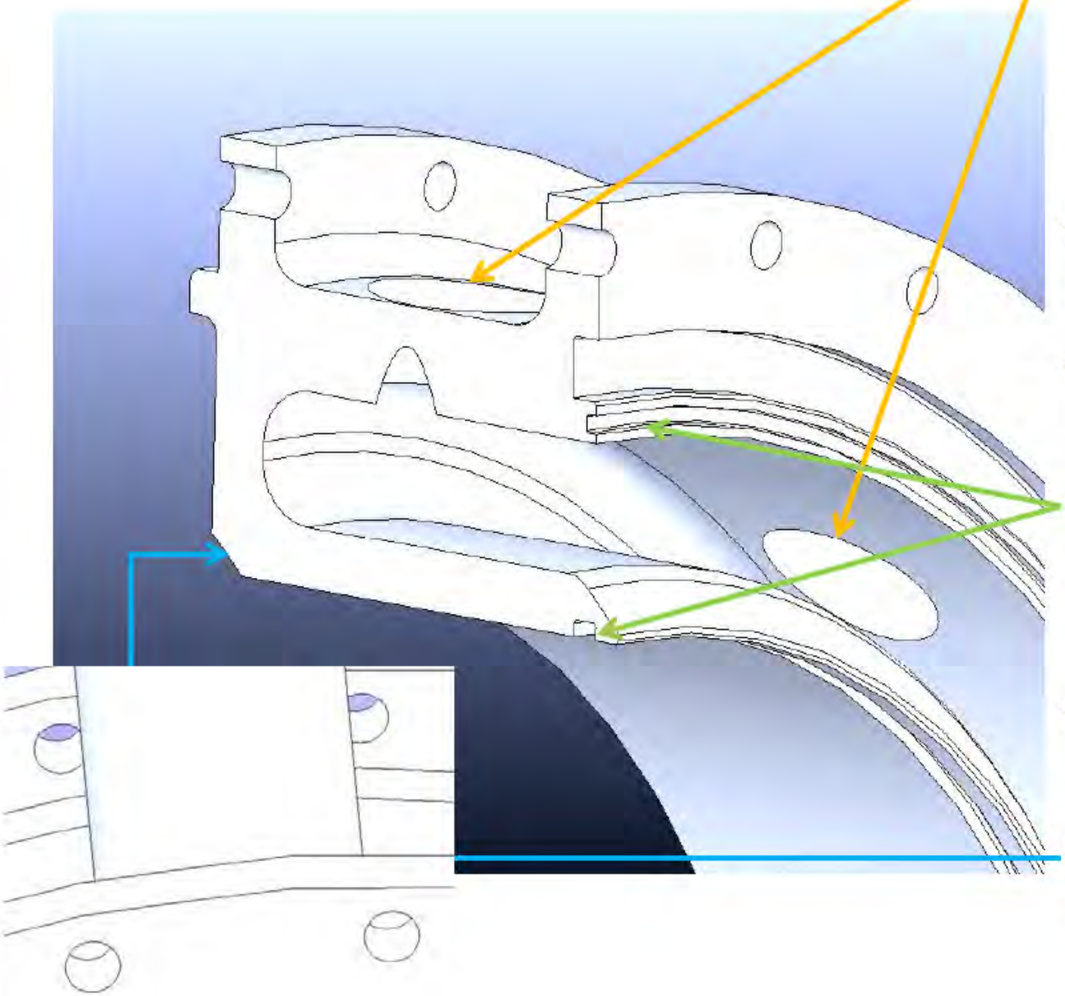
System Design

- **Overview :**



System Design

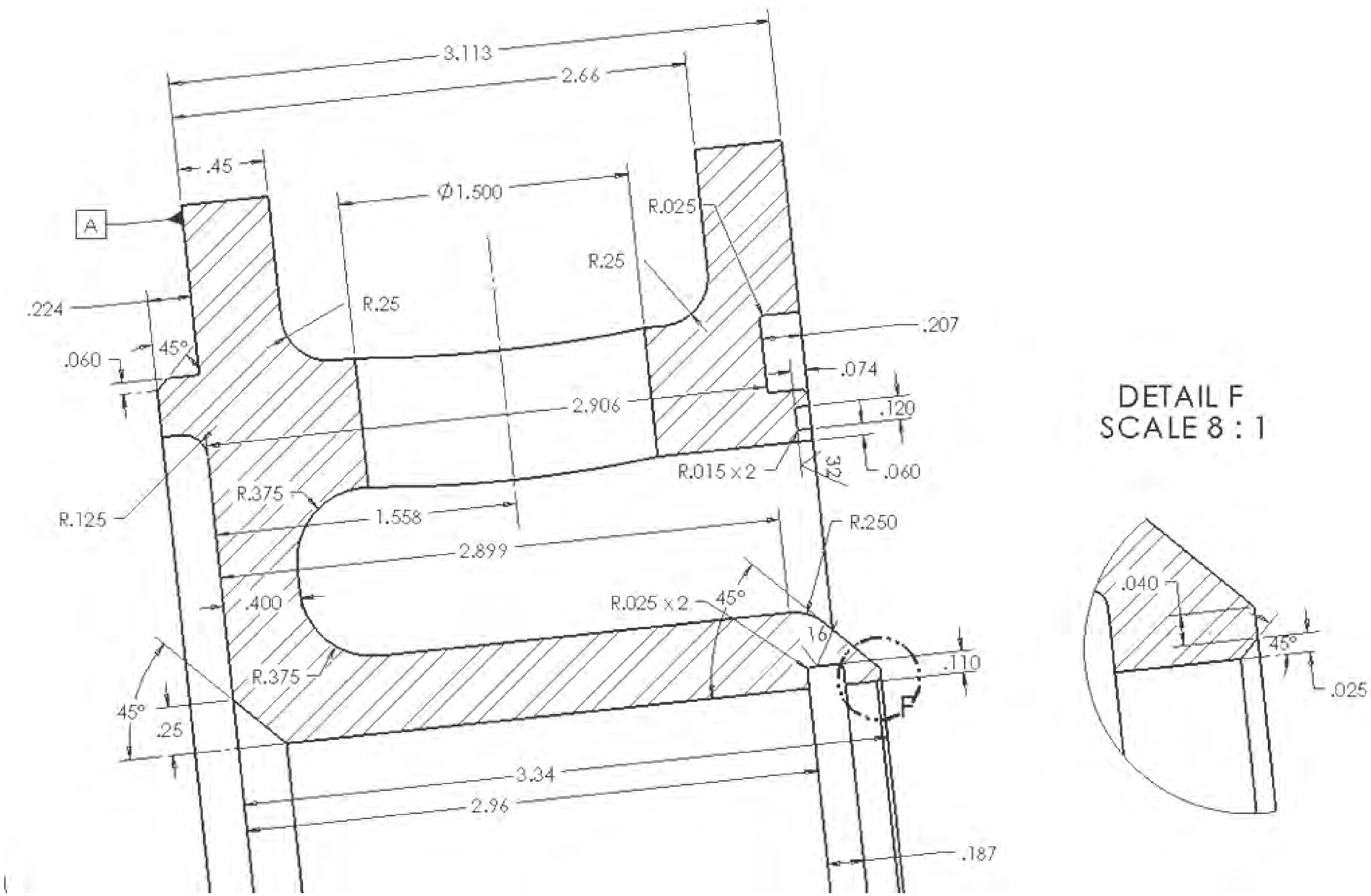
P/N 863010 Features



- 10 alternating 1 and 1.25 NPT threaded holes at regular radial allied intervals (6° offset centerline to avoid bolt features)
- 30 bolt holes fwd and aft (match existing components)
- Pilots fits fwd and aft (match existing components)
- Inner and outer glands to accommodate O-ring seals (Parker spec, see slide 11-12)
- Designed to accommodate easy tool access for machining
- Chamfer allows thrust-flow to escape.

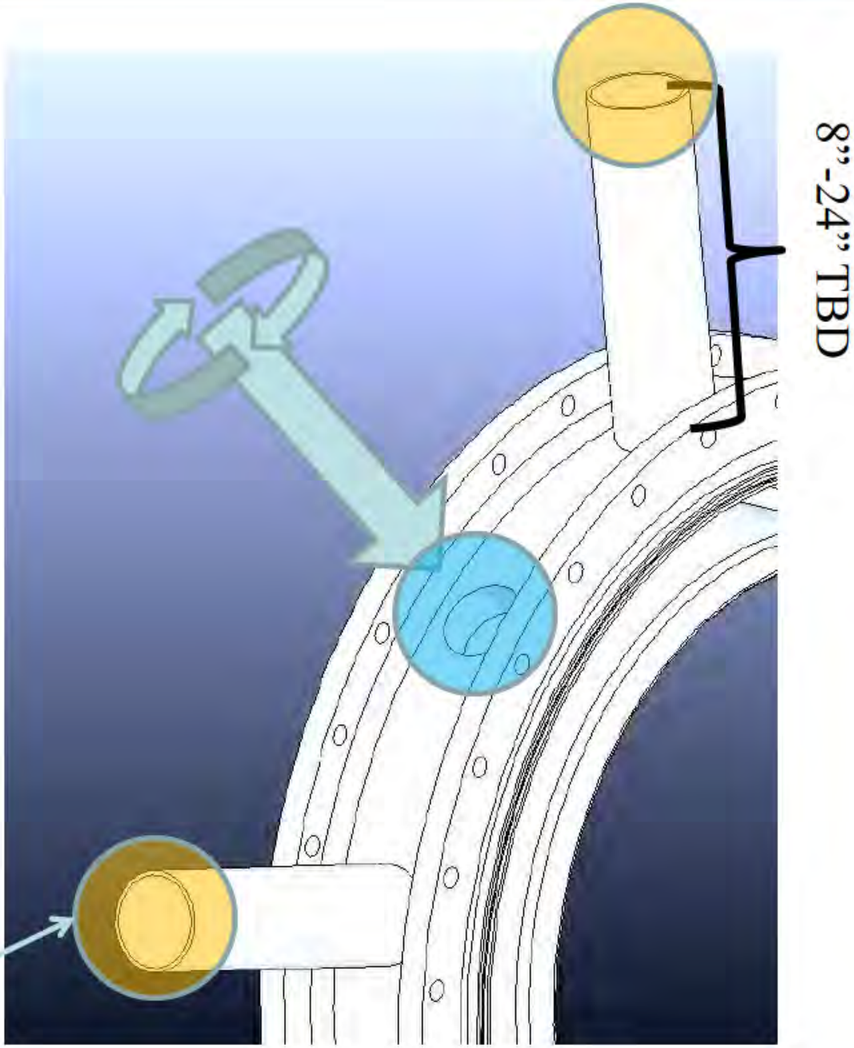
System Design

- **P/N ######## Dimensions**



P/N 863010

- **Once in position five 1.5 inch and five 1.25 inch tubes can be affixed to the inlet cap via NPT nipples and pipe to tube adapters.**
- **Tubes will vary in height to accommodate working area. On-site modifications will be needed to ensure ease-of-access.**
- **Tubes ultimately connect to air system via existing bleed hoses (with extensions and fittings needed see budget slide)**



System Design

Aero

- **Table 1 shows the anticipated Mach numbers for the inlet during operation of the two experiments**
- **Maximum internal Mach number in the system projected to be 0.314 at ramp constriction.**
- **Loss of kinetic head at Inlet due to radial impingement to induce a tolerable pressure loss on the order of 10% with M=0.31**

Hole Dia [in]	1.25	1.5
ID	1.01	1.26
Area [in^2]	0.801	1.247
Flow [lb/s]	10	

	6 to 1	10 to 1
Psi	87	147
Temp	120	120
p [lb/ft^3]	0.405	0.685

Experiment	6 to 1	Velocity	
Pipe	1.25	[ft/s]	M
hole #	10	443.52	0.37

Experiment	10 to 1	Velocity	
Pipe	1.25	[ft/s]	M
hole #	10	262.40	0.22

Experiment	6 to 1	Velocity	
Pipe	1.5	[ft/s]	M
hole #	10	284.98	0.24

Experiment	10 to 1	Velocity	
Pipe	1.5	[ft/s]	M
hole #	10	168.60	0.14

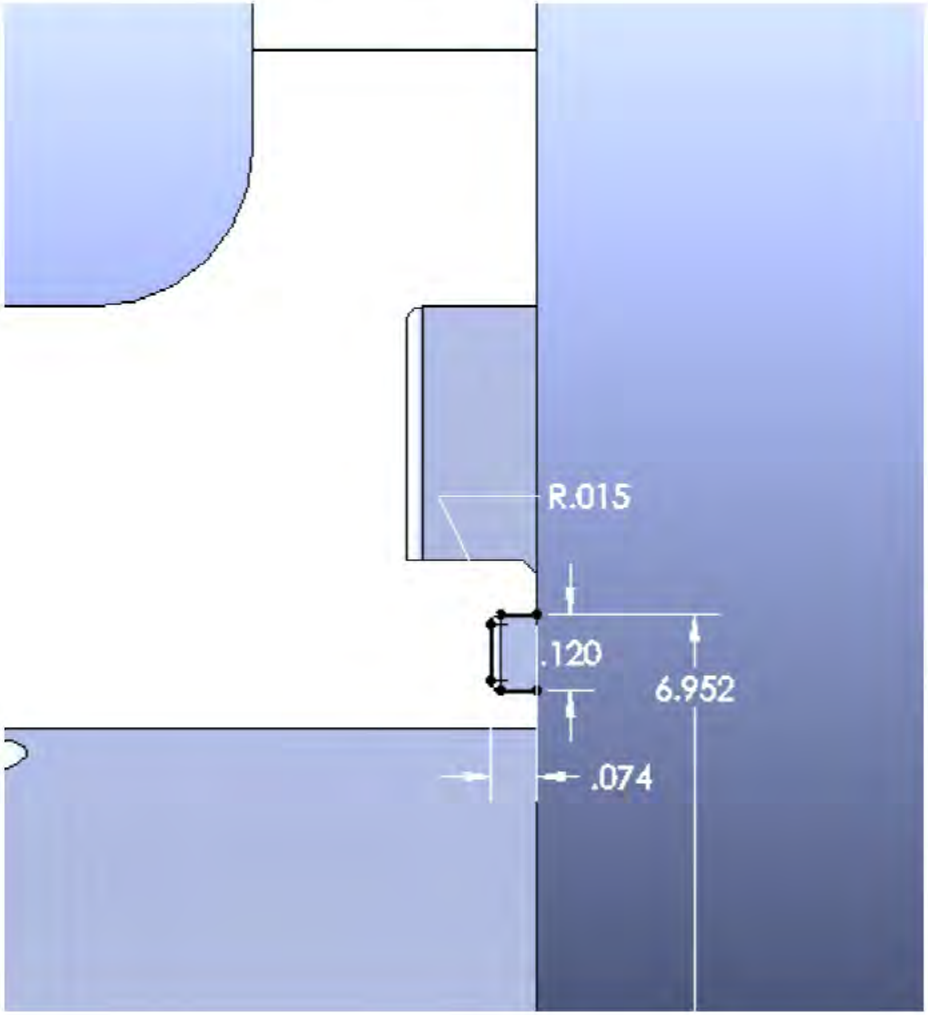
M	0.31

M	0.18

Table 1: Effect of inlet Tube diameter axial inlet velocity.

Average M given 5 of each tube size

System Design



Sealing

- **Face seal gland must seal the shroud chamber from the flow path due to drilled holes.**
- **Standard O-ring size (3/32") used with non-typical diameter (13.9" dia) for face seal.**

O-Ring Face Seal Glands These dimensions are intended primarily for face type O-ring seals and low temperature applications.

O-Ring Size Parker No. 2	W Cross Section Nominal	W Cross Section Actual	L Gland Depth	Squeeze Actual	Squeeze %	G Groove Width Liquids	G Groove Width Vacuum and Gases	R Groove Radius
004 through 050	1/16	.070 ±.003 (1.78 mm)	.050 to .054	.013 to .023	19 to 32	.101 to .107	.084 to .089	.005 to .015
102 through 178	3/32	.103 ±.003 (2.62 mm)	.074 to .080	.020 to .032	20 to 30	.136 to .142	.120 to .125	.005 to .015
201 through 284	1/8	.139 ±.004 (3.53 mm	.101 to .107	.028 to .042	20 to 30	.177 to .187	.158 to .164	.010 to .025
309 through 395	3/16	.210 ±.005 (5.33 mm)	.152 to .162	.043 to .063	21 to 30	.270 to .290	.239 to .244	.020 to .035
425 through 475	1/4	.275 ±.006 (6.99 mm)	.201 to .211	.058 to .080	21 to 29	.342 to .362	.309 to .314	.020 to .035
Special	3/8	.375 ±.007 (9.52 mm)	.276 to .286	.082 to .106	22 to 28	.475 to .485	.419 to .424	.030 to .045
Special	1/2	.500 ±.008 (12.7 mm)	.370 to .380	.112 to .138	22 to 27	.638 to .645	.560 to .565	.030 to .045

Design Chart 4-3: Design Chart for O-Ring Face Seal Glands

System Design

Sealing

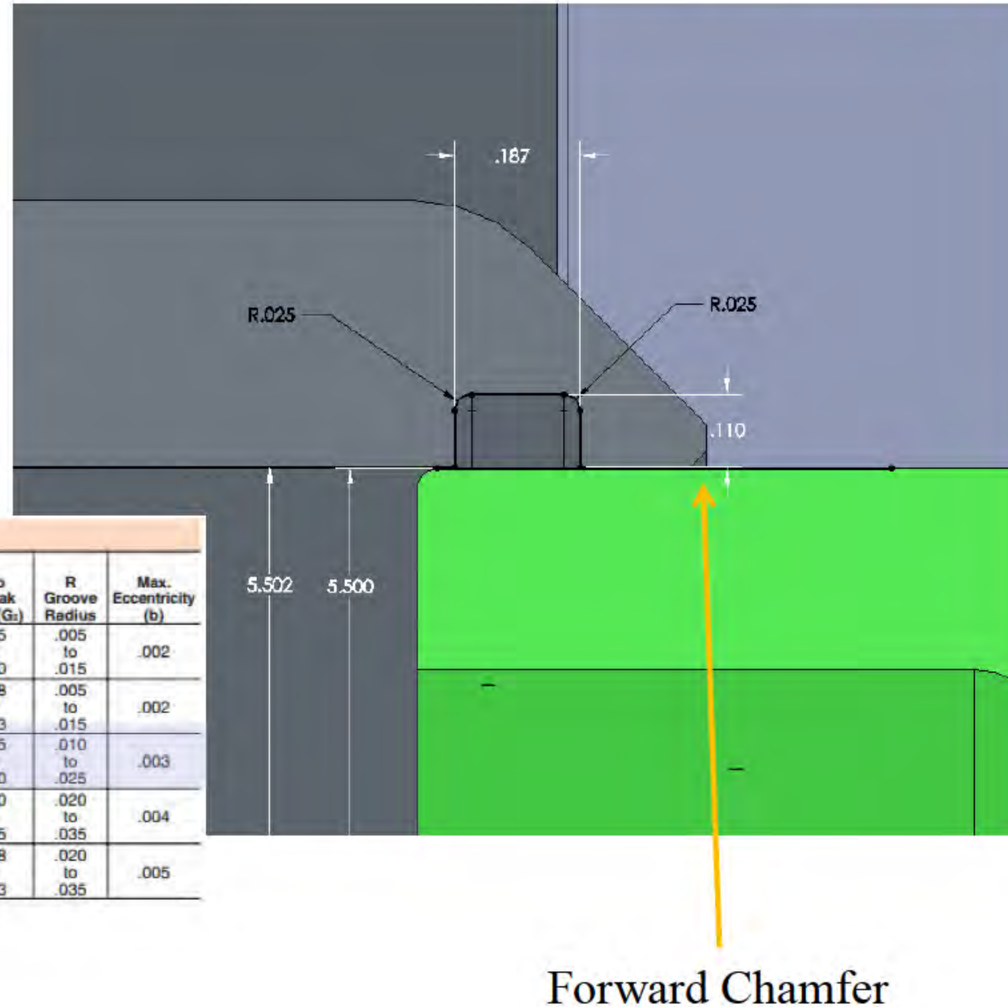
- **Piston seal gland seals the rotor chamber from the flow path**
- **Standard O-ring (1/8" x 11") and groove used. Standard clearance used.**

Industrial O-Ring Static Seal Glands

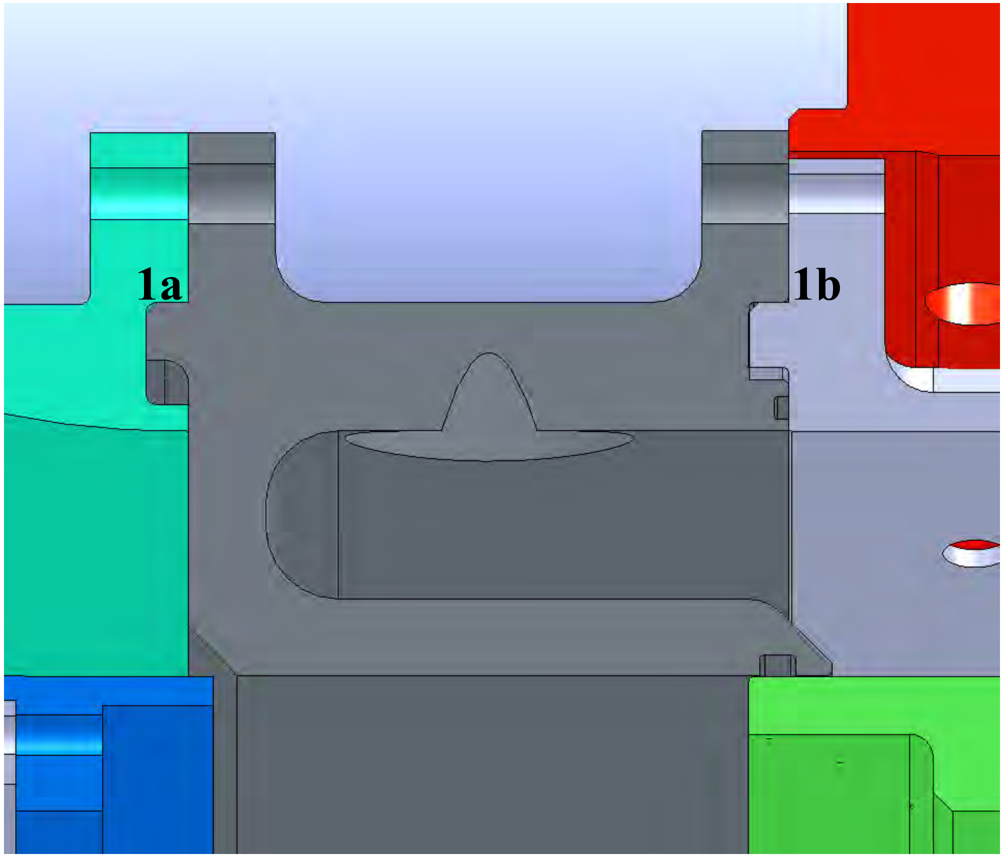
O-Ring 2-Size AS568B-	W Cross-Section Nominal	W Cross-Section Actual	L Gland Depth	Squeeze Actual	Squeeze %	E(a) Diametral Clearance	G - Groove Width: No Parbak Ring (G)	G - Groove Width: One Parbak Ring (G1)	G - Groove Width: Two Parbak Ring (G2)	R Groove Radius	Max. Eccentricity (b)
004 through 050	1/16	.070 ±.003 (1.78 mm)	.050 to .052	.015 to .023	22 to 32	.002 to .005	.093 to .098	.138 to .143	.205 to .210	.005 to .015	.002
102 through 178	3/32	.103 ±.003 (2.62 mm)	.081 to .083	.017 to .025	17 to 24	.002 to .005	.140 to .145	.171 to .176	.238 to .243	.005 to .015	.002
201 through 284	1/8	.139 ±.004 (3.53 mm)	.111 to .113	.022 to .032	16 to 23	.003 to .006	.187 to .192	.208 to .213	.275 to .280	.010 to .025	.003
309 through 395	3/16	.210 ±.005 (5.33 mm)	.170 to .173	.032 to .045	15 to 21	.003 to .006	.281 to .286	.311 to .316	.410 to .415	.020 to .035	.004
425 through 475	1/4	.275 ±.006 (6.99 mm)	.226 to .229	.040 to .055	15 to 20	.004 to .007	.375 to .380	.408 to .413	.538 to .543	.020 to .035	.005

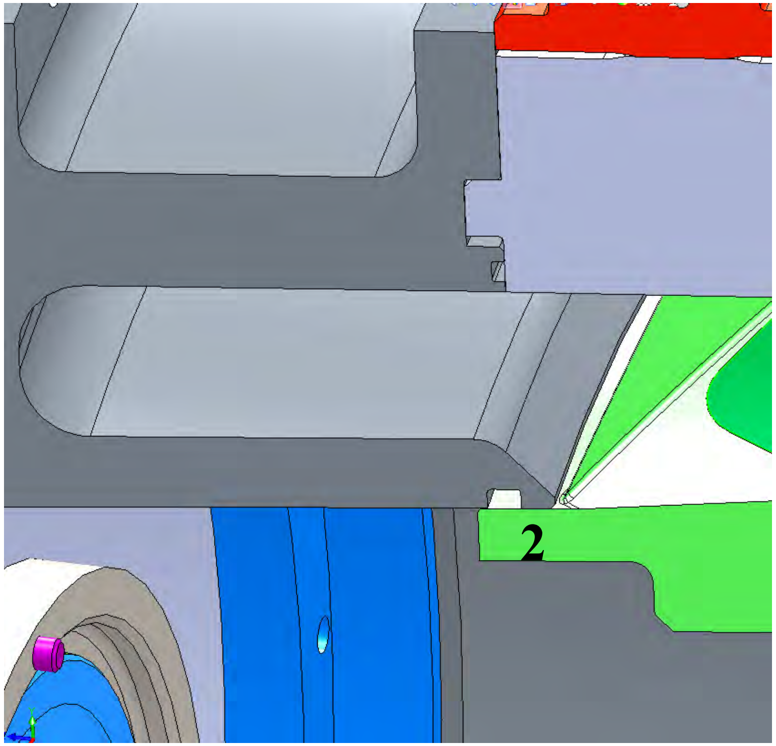
(a) Clearance (extrusion gap) must be held to a minimum consistent with design requirements for temperature range variation.
(b) Total indicator reading between groove and adjacent bearing surface.
(c) Reduce maximum diametral clearance 50% when using silicone or fluorosilicone O-rings.
(d) For ease of assembly, when Parbaks are used, gland depth may be increased up to 5%.

Design Chart 4-2: For Industrial O-Ring Static Seal Glands



System Design





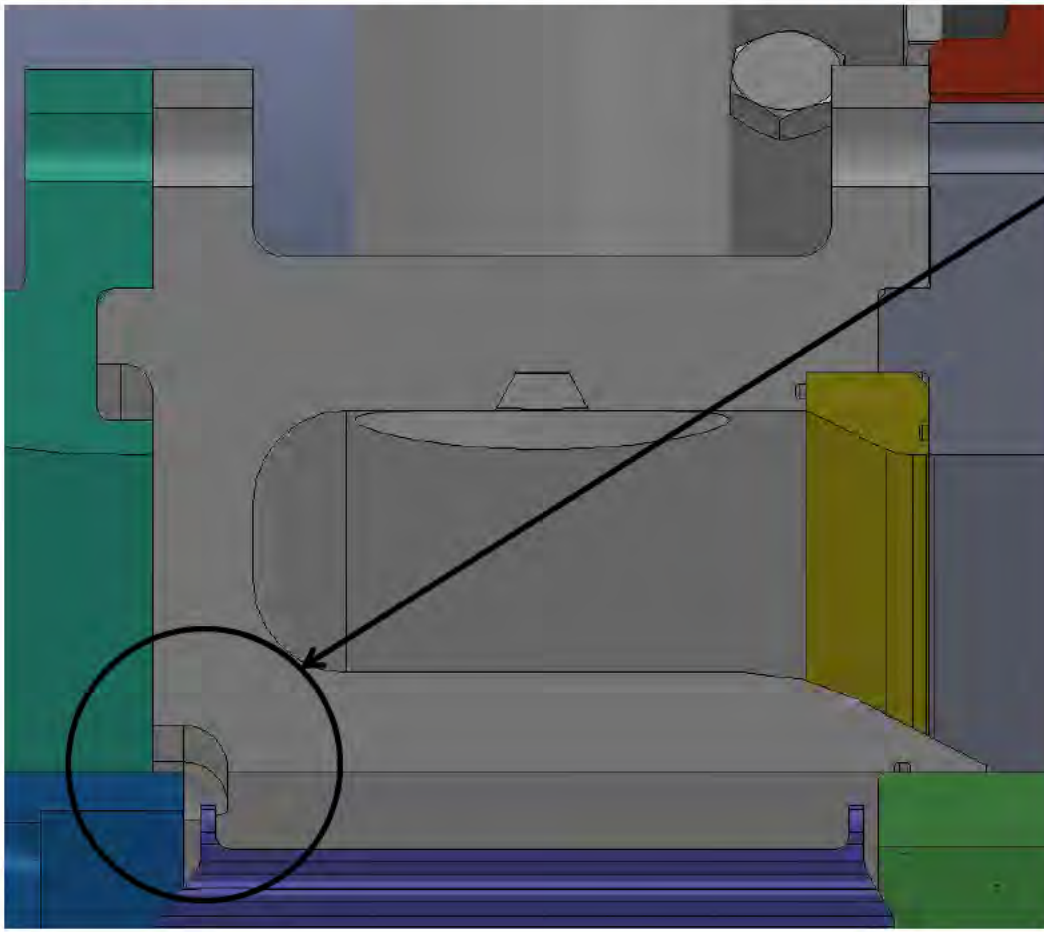
Details of critical junctions, note pilot fits [1] and proximity to inducer strake [2] (.031”). Pilot fit clearances will be identical to current tip ring as per its production drawing

System Design

- **Materials**
  - **Carbon Steel**
    - 12L14, 60 ksi yield
  - **Electroless nickel plate, AMS 2404 on the order of .0005-.001" thickness**
    - "…This deposit has been used typically to provide a uniform build-up on intricate shapes, to improve wear and/or corrosion resistance, or to improve solderability on or for selected materials, but usage is not limited to such applications. The deposit has been used in service up to 1000 °F (540 °C) although wear and/or corrosion resistance may degrade as service temperature increases." ~ SEA
  - **Stainless Steel welding for diffuser case.**
    - Confirmed with welder that with removal of components as shown in slide 6, and assuming unfavorable weld properties of substrate it would be possible.

Requirements

- **Not interfere with existing seal and thrust flows**



5 equally spaced .125 inch high cavity vents provide flow path for thrust and bearing air the exist much like current design.

Analyses Performed

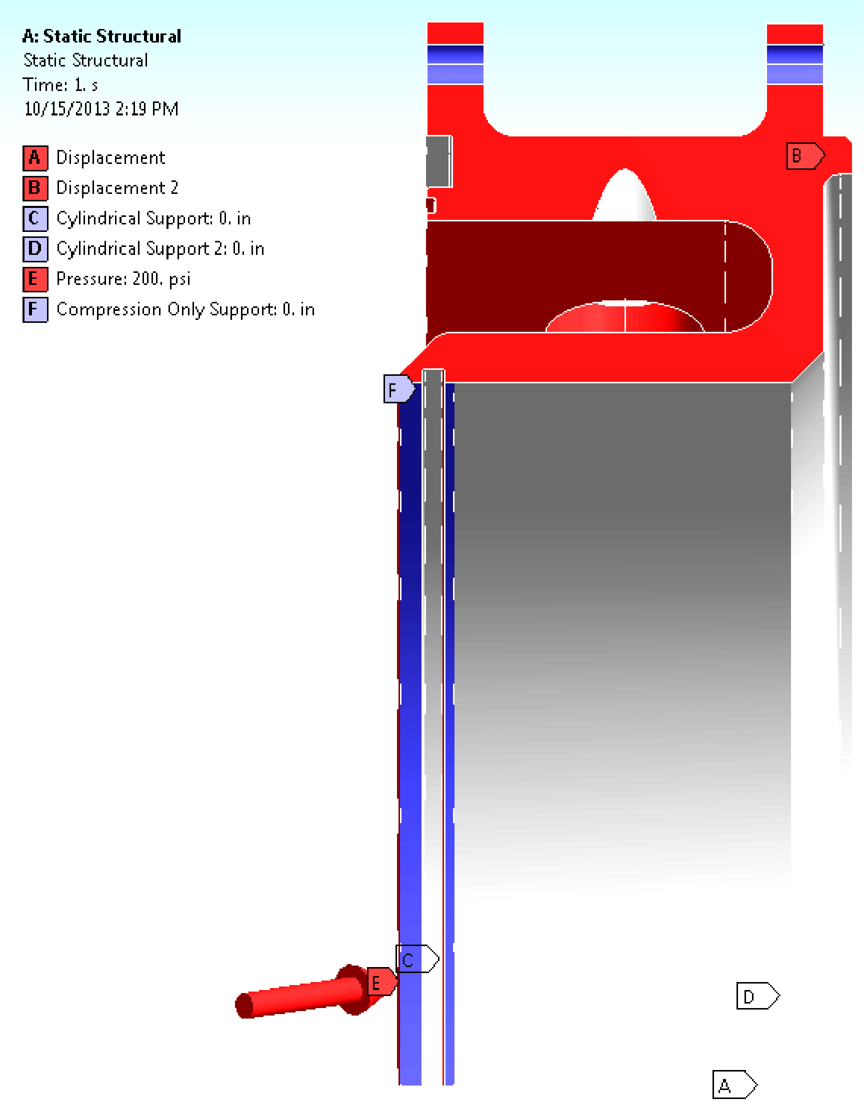
- CFD on OGV for the ISCE B1 under M<1
- Thermal-structural analysis on rotating rotor to establish rotor clearance under maximum allowable speed.
- Structural analysis on Inlet section to establish performance under maximum allowable load.
- Structural analysis on diffuser hub and shroud to establish performance under load and monitor the effect of reaction forces from Inlet section.
- Structural analysis on OGV shroud section as per above.
- Structural analysis on combustor casing section as per above.
- Structural analysis on turbine shroud section as per above
- Qualitative projections on overall bolt loading.

Analyses Results

- **CFD on OGVs:**
  - Aero review confirms adequate flow into combustor sections given sub-sonic inlet conditions (see requirement #1 slide 2 , and slide 10).
- **Dynamic thermal-structural loading of rotor:**
  - Induces .015” radial growth of rotor under maximum loading conditions minimally impacting rotor clearance gap to new Inlet section (see PRR reference slides [attached]).

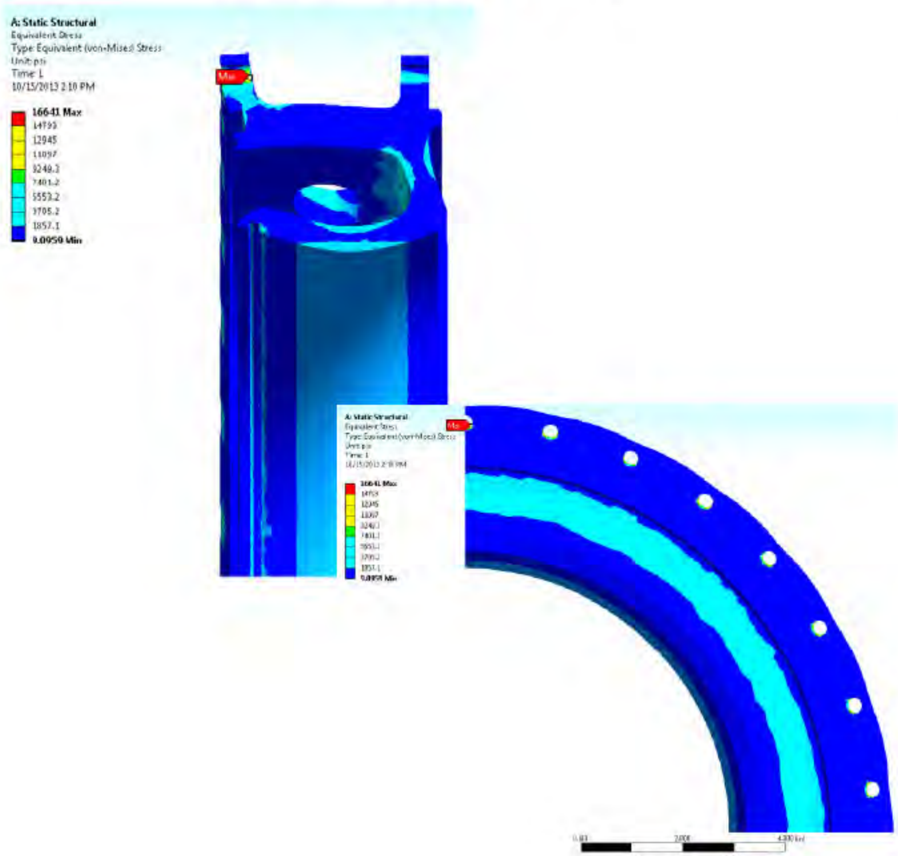
Analyses Results

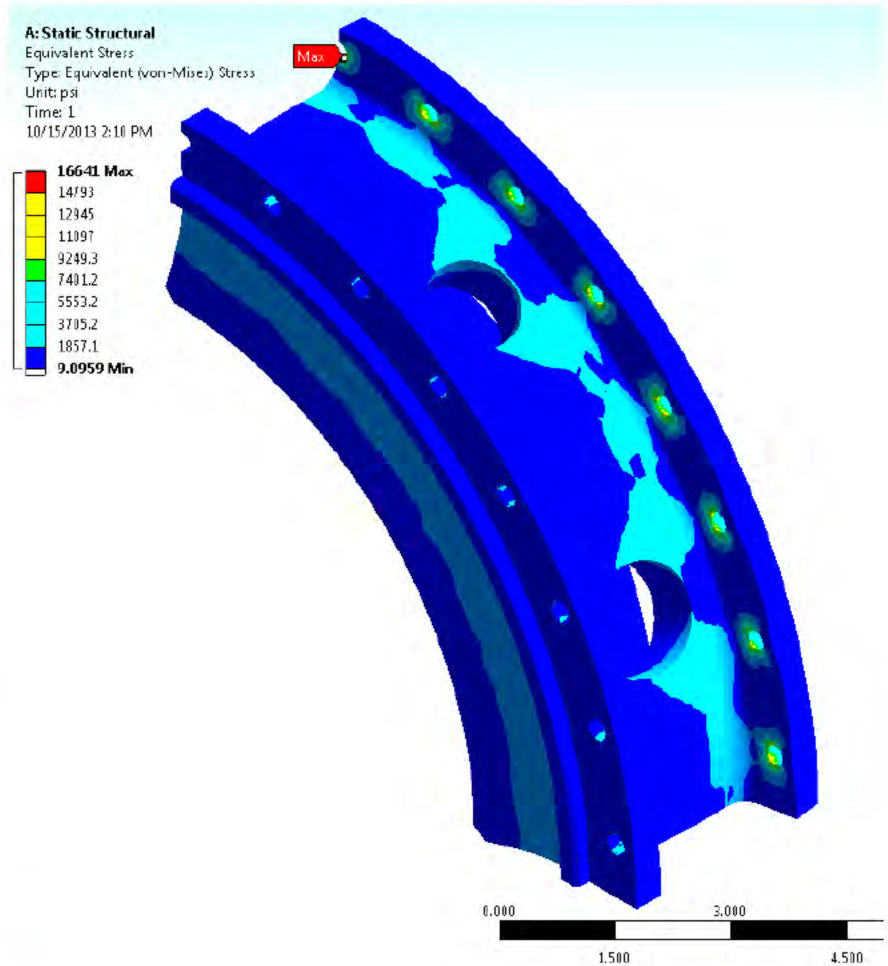
- **Structural analysis on Inlet**
  - ANSYS FEA constraints:
    - 200 Psi internal load vs. vacuum
    - X-Y displacement constraints along ¼ cut
    - Axial and tangentially fixed cylindrical support on aft bolt circle, tangential only on fwd bolt circle.
    - Compression only support on [F](contact with inducer hub)



Analyses Results

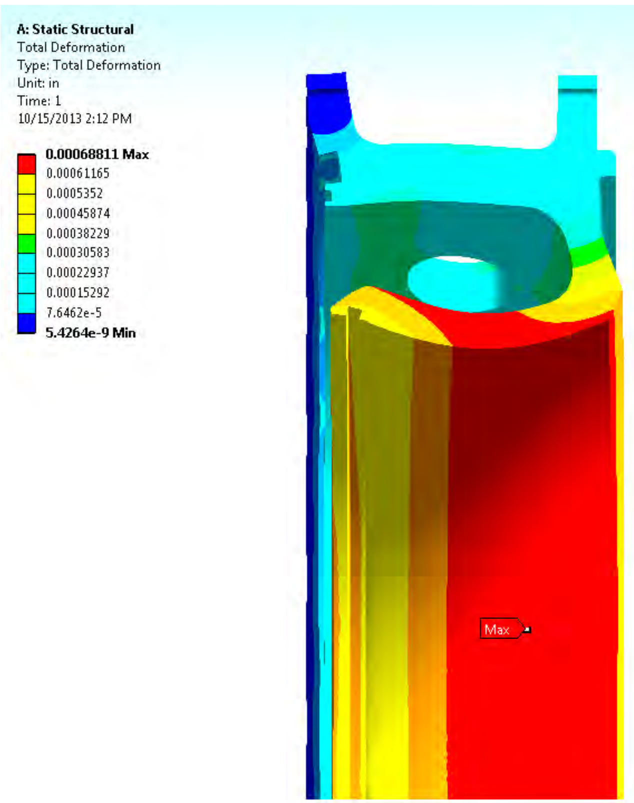
- **Structural analysis on Inlet**
  - Stress
    - 16 ksi max (von-mises)

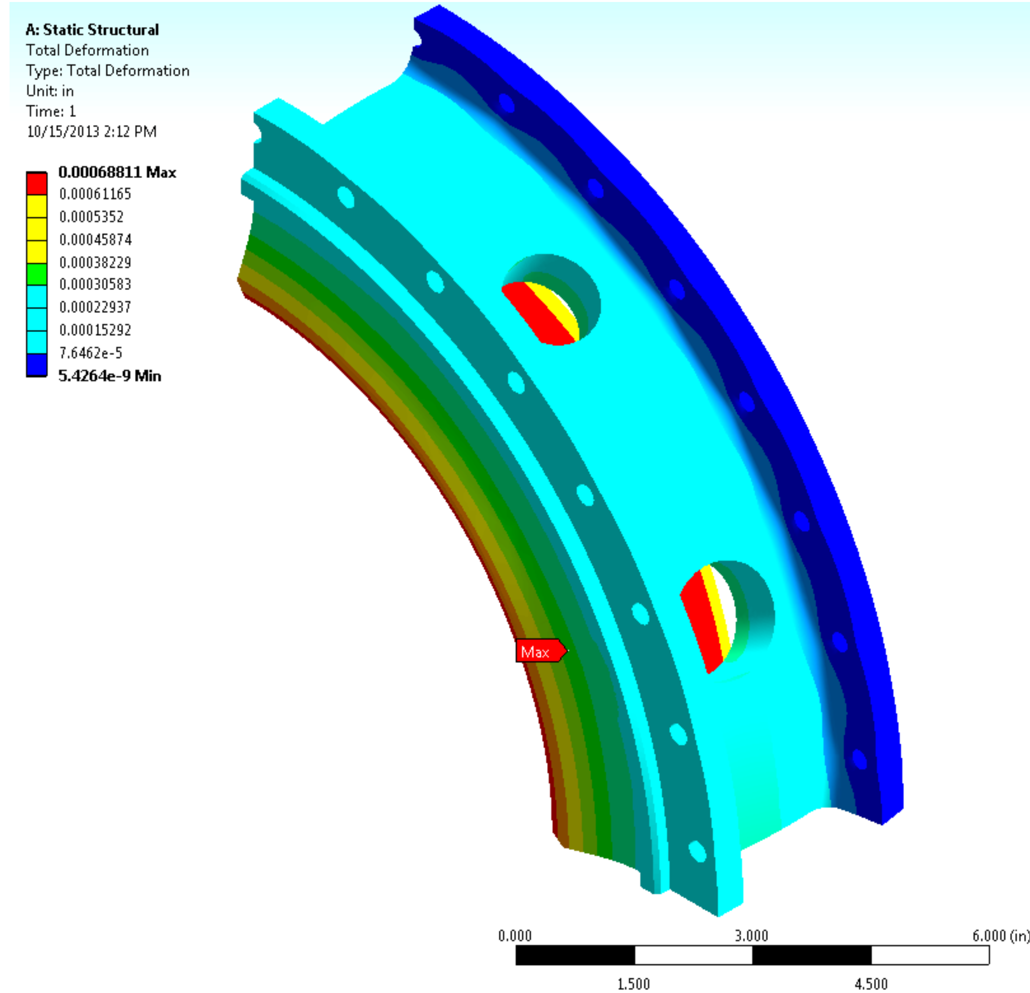




Analyses Results

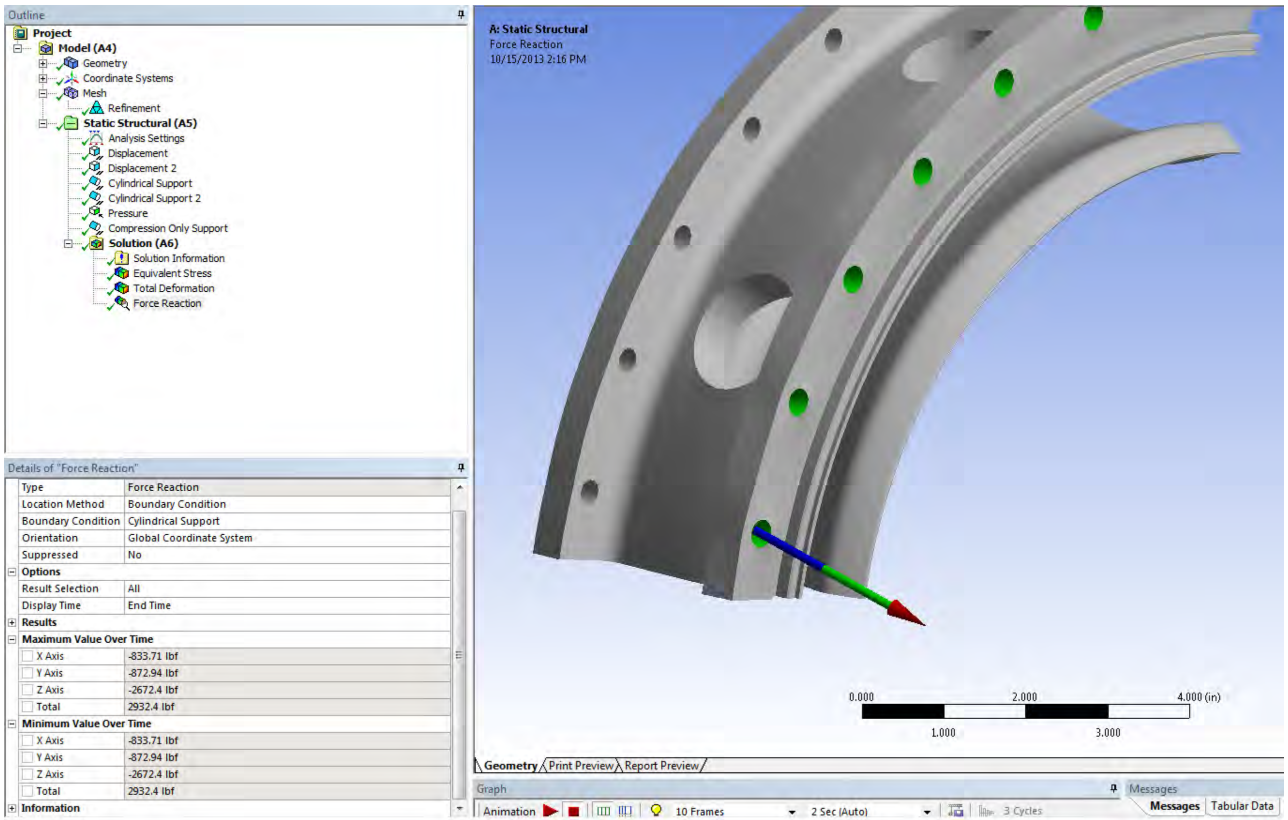
- **Structural analysis on Inlet**
  - Deformation
    - .0007" max





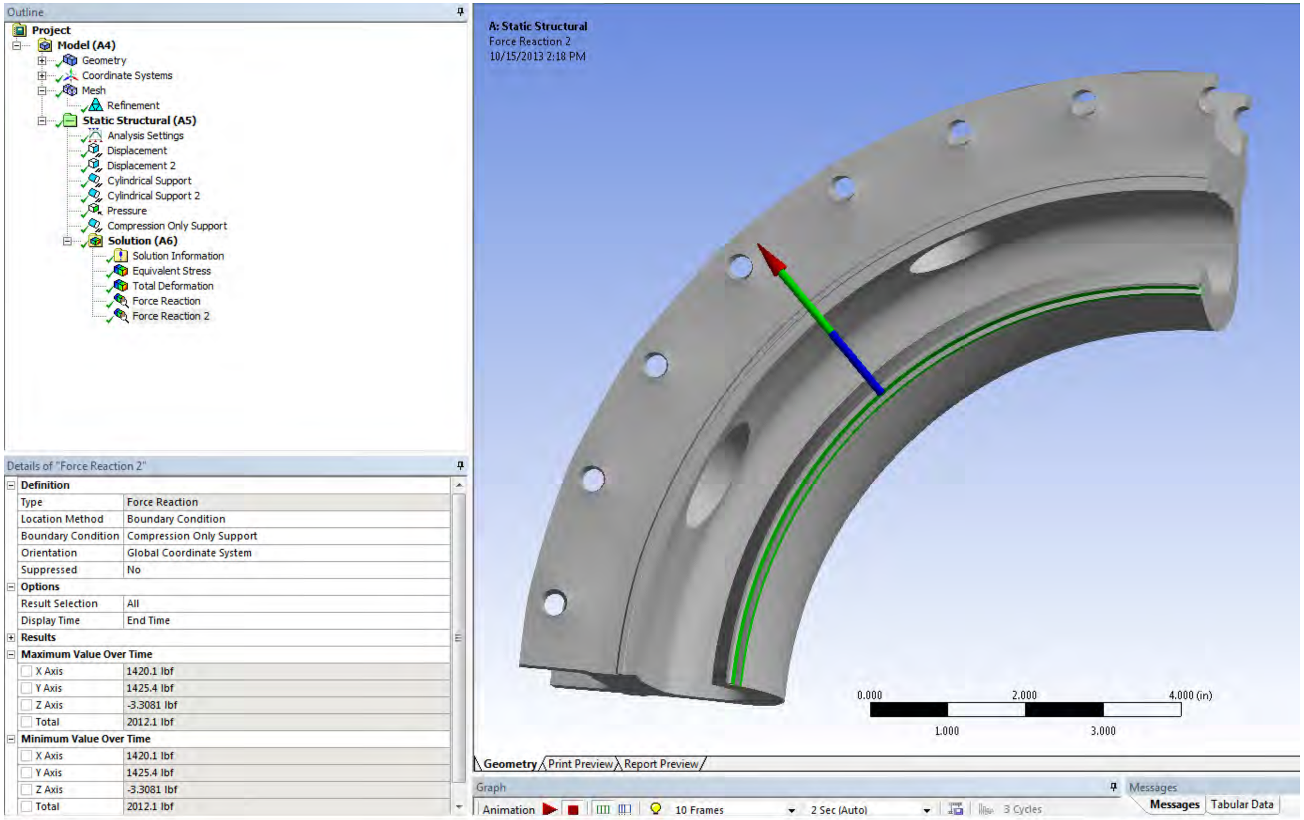
Analyses Results

- **Structural analysis on Inlet**
  - Reaction
    - 2672 lbf reaction in the axial direction for the ¼ piece yielding a total load on the order of 10,700 lbf



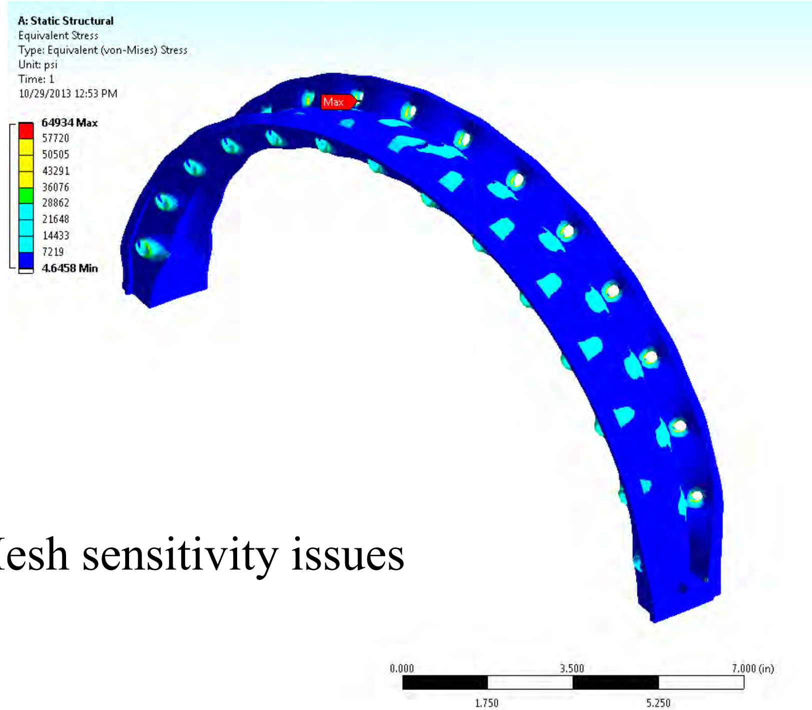
Analyses Results

- **Structural analysis on Inlet**
  - Compression only reaction
    - 2012 lbf reaction in the radial direction for the ¼ piece. Resting on the diffuser hub.

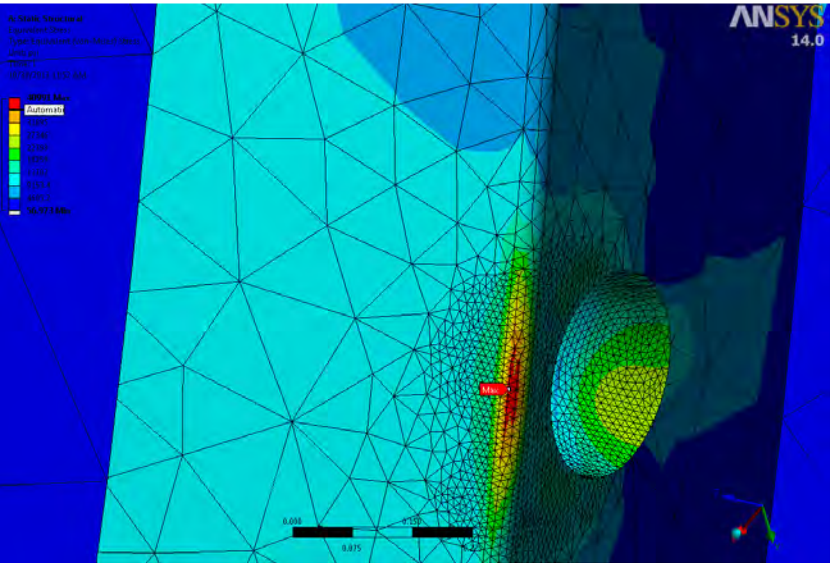


Analyses Results

- **Structural analysis on OGV shroud section**
  - Stress
    - 41 ksi Max
    - Internal load of 200 psi
    - axial constraints on bolt holes
    - Reaction force of ~5300 lbf applied through bolt pattern



Mesh sensitivity issues

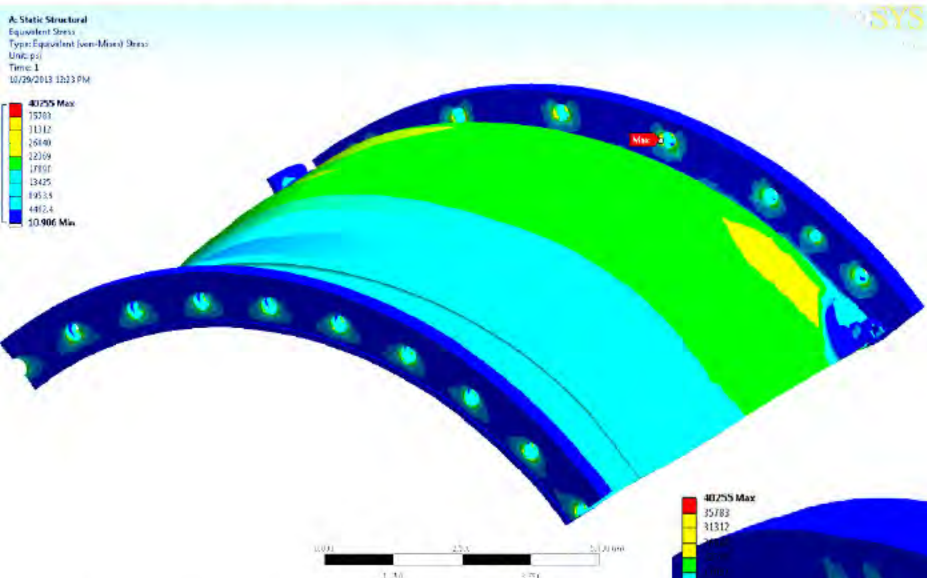


Constructed of annealed 410 SS with yield strength of 45 ksi
Tempering to 1200F can raise hardness up to 90 ksi

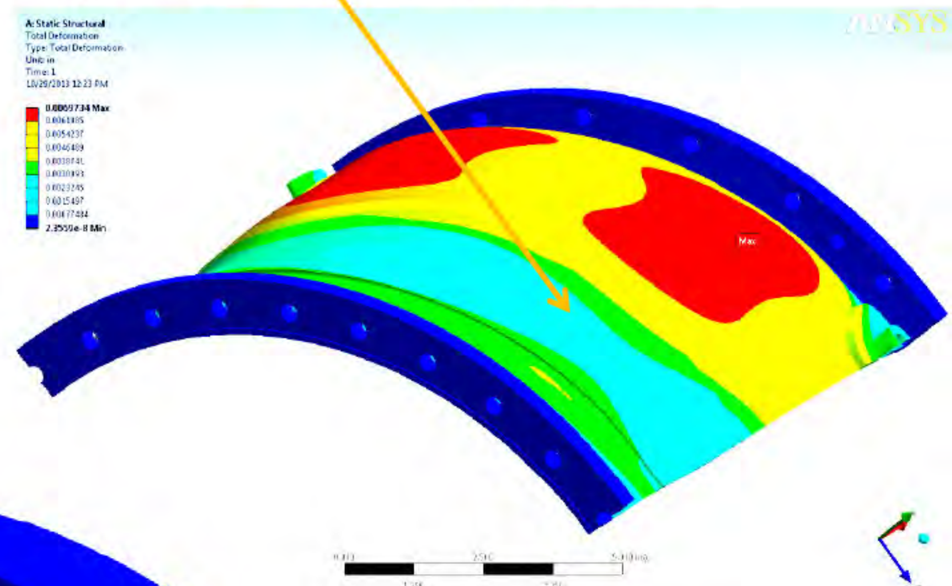
Analyses Results

Welded strip to seal case

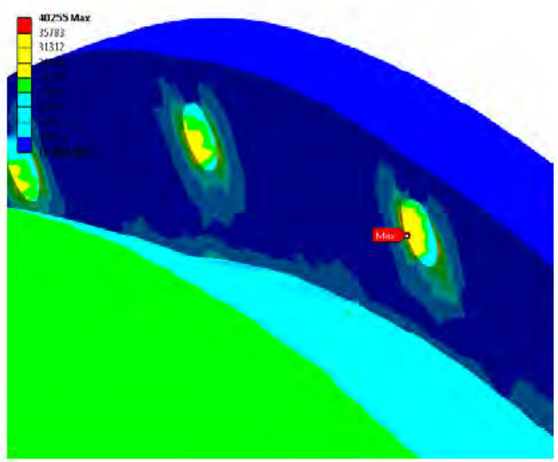
- **Structural analysis on combustor casing section (minimal mods)**



Stress [40.2 ksi max]



Deformation [.006" max]



15-5 PH

Analyses Results

- **15-5 PH**

Mechanical Properties:						
F_{tu}, ksi:						
L	190	170	155	145	140	135
T	190	170	155	145	140	135
F_{ty}, ksi:						
L	170	155	145	125	115	105
T	170	155	145	125	115	105

Lowest yield found to be 105 ksi providing significant strength up to 800 F (30% loss)

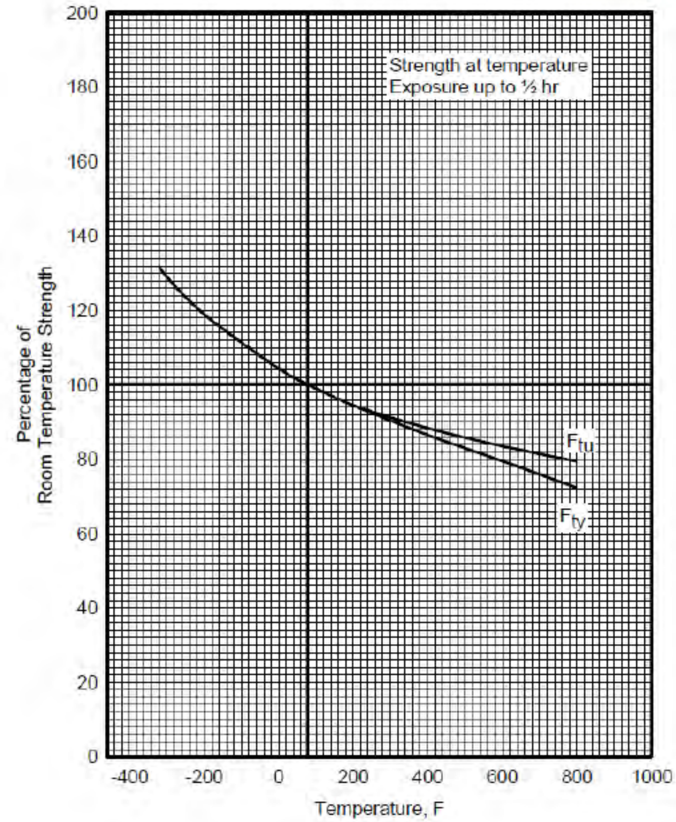
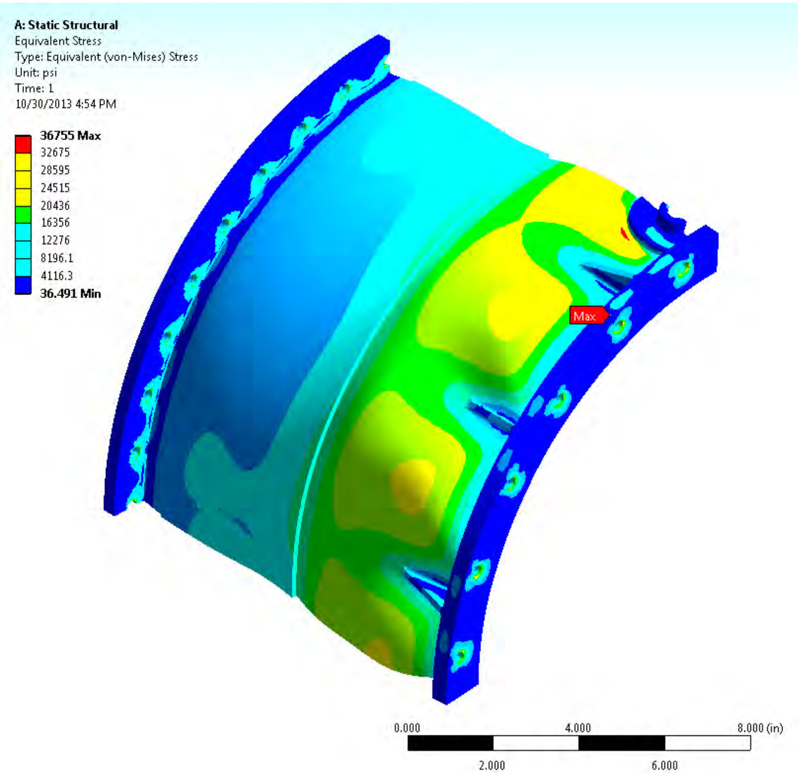
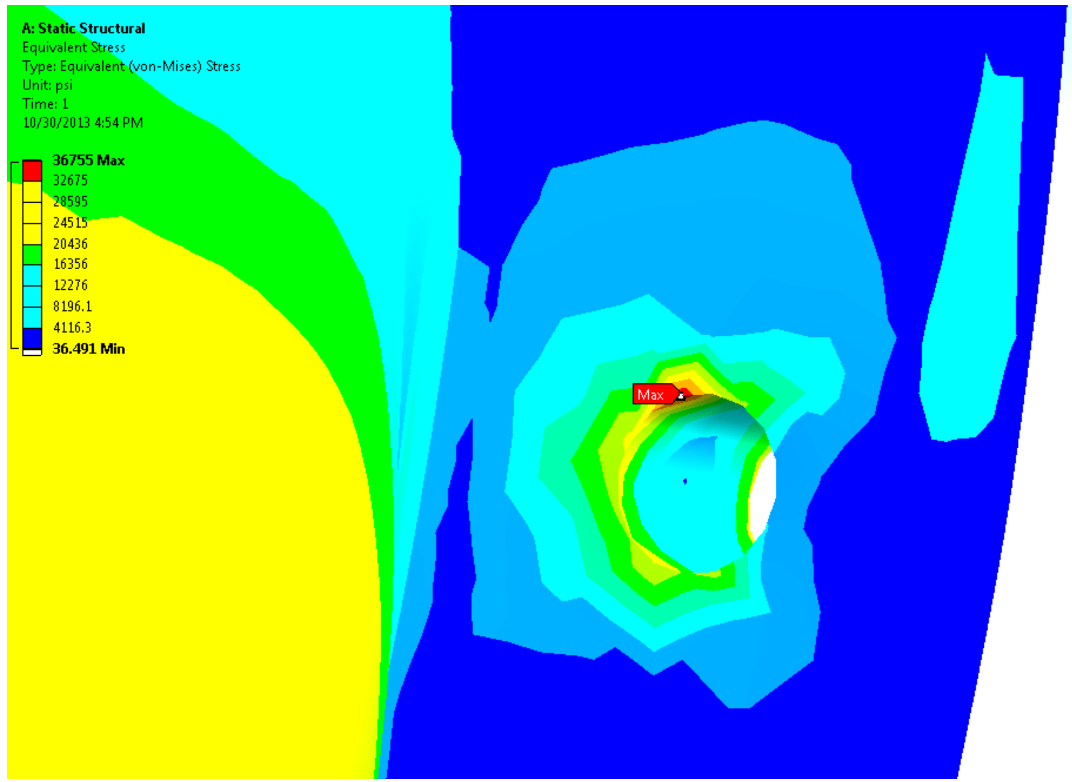


Figure 2.6.7.1.1. Effect of temperature on the tensile ultimate strength (F_{tu}) and the tensile yield strength (F_{ty}) of 15-5PH (H925, H1025, and H1100) stainless steel bar.

Analyses Results

- **Structural analysis on combustor casing section (with gussets)**

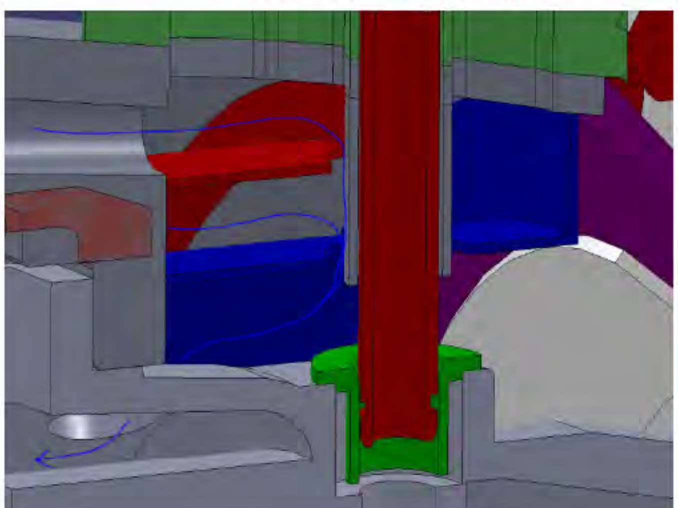


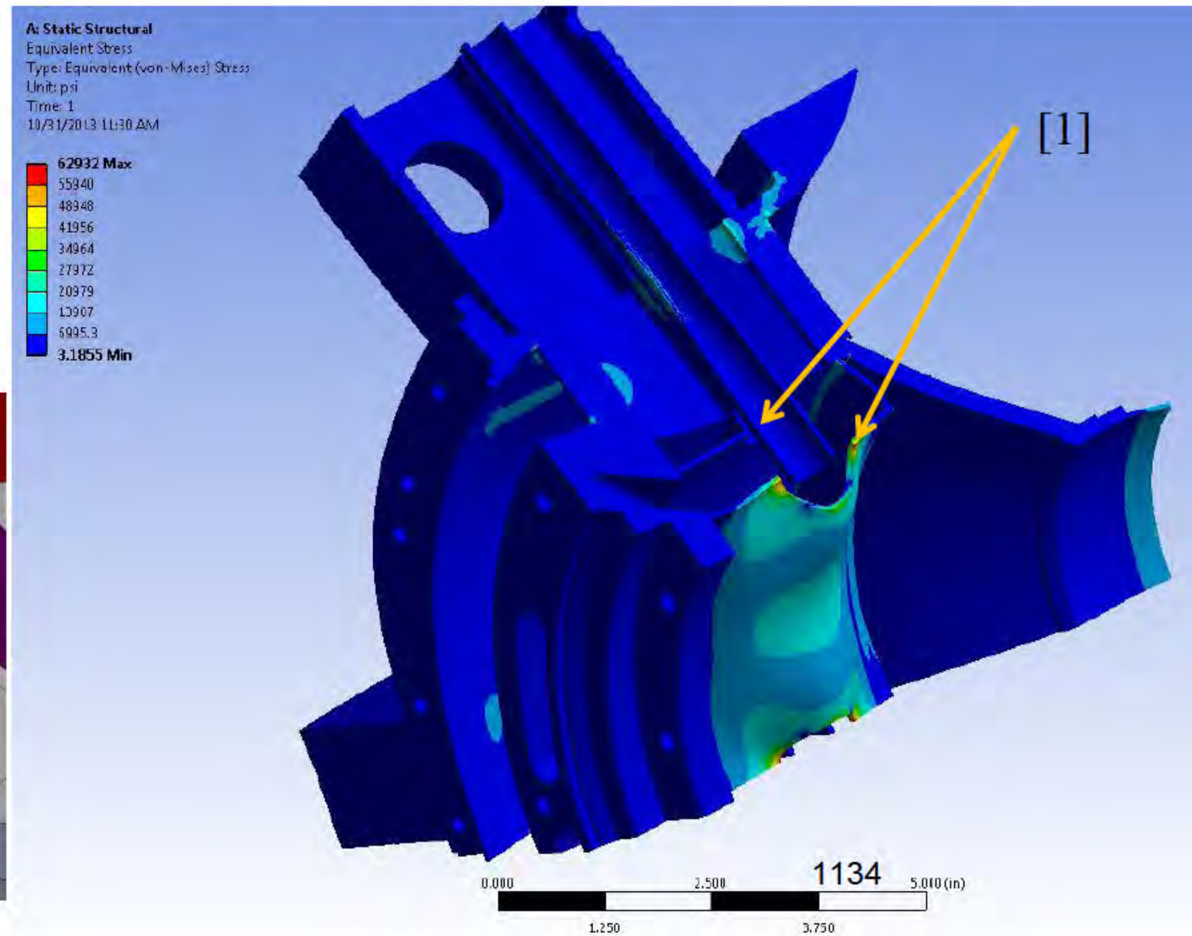


Stress [36.7 ksi max]

Analyses Results

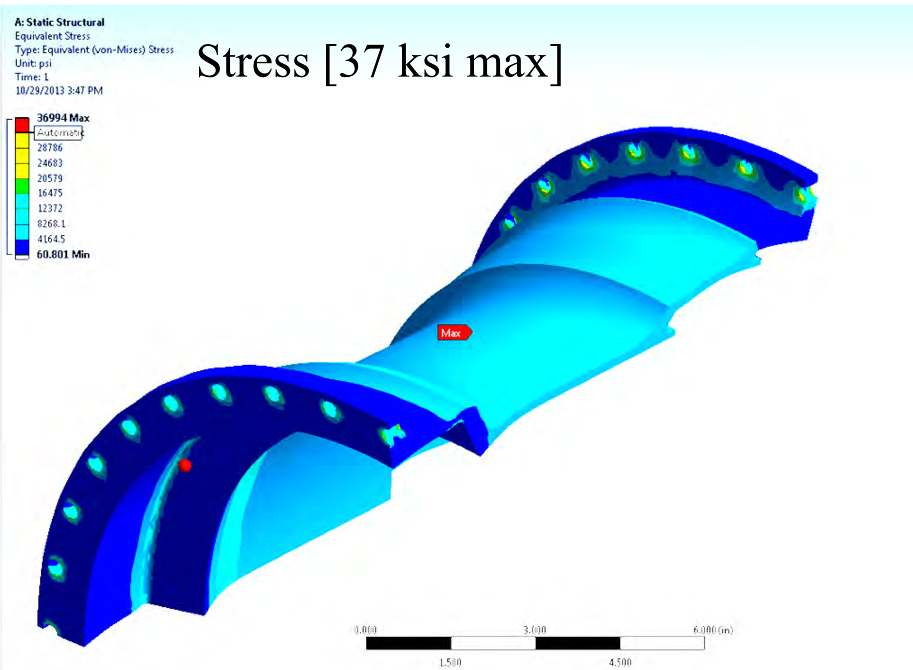
- **Structural analysis on Discharge Duct (P/N 11110)**
  - Constructed of 17-4 ph Steel with yield strength of 105 ksi (mil-spec)
  - Inputs form Steve confirm the model as presented in not as-built and welds seal bleed annulus [1]
- **Constraints**
  - Displacement constraints at secured locations
  - Internal 200 psi

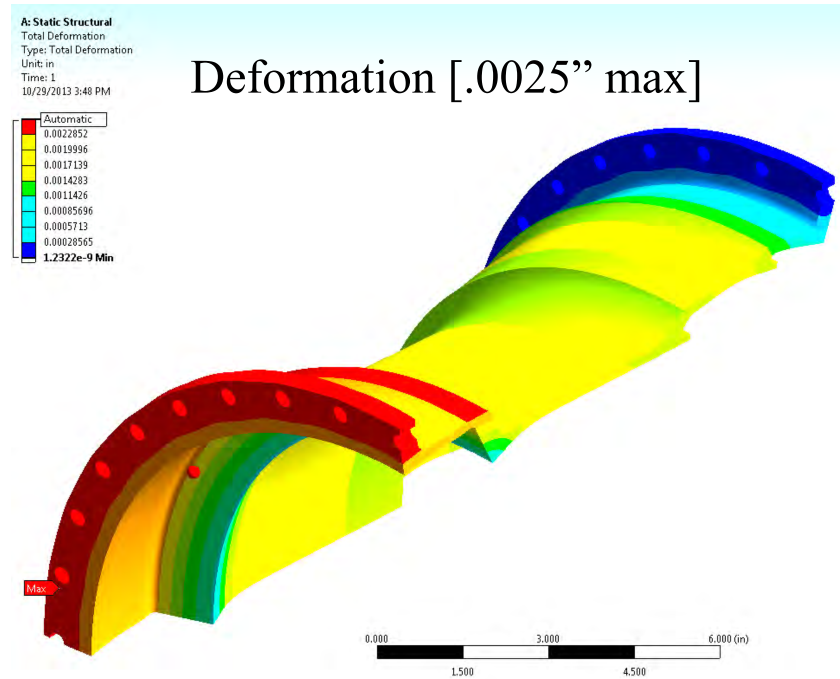




Analyses Results

- **Structural analysis on turbine shroud**





From GLM answers document 9/9/13:
"Turbine section structural casing
Believed to be 420 SS, maybe 430 SS".

Min for 420 annealed: 50 ksi*
Min for 430: 30 ksi

Analyses Results

- **Overall bolt loading**
  - Simple division yields approximate tension on each bolt

[36] 5/16-24
294 lbf each
3.8 ksi

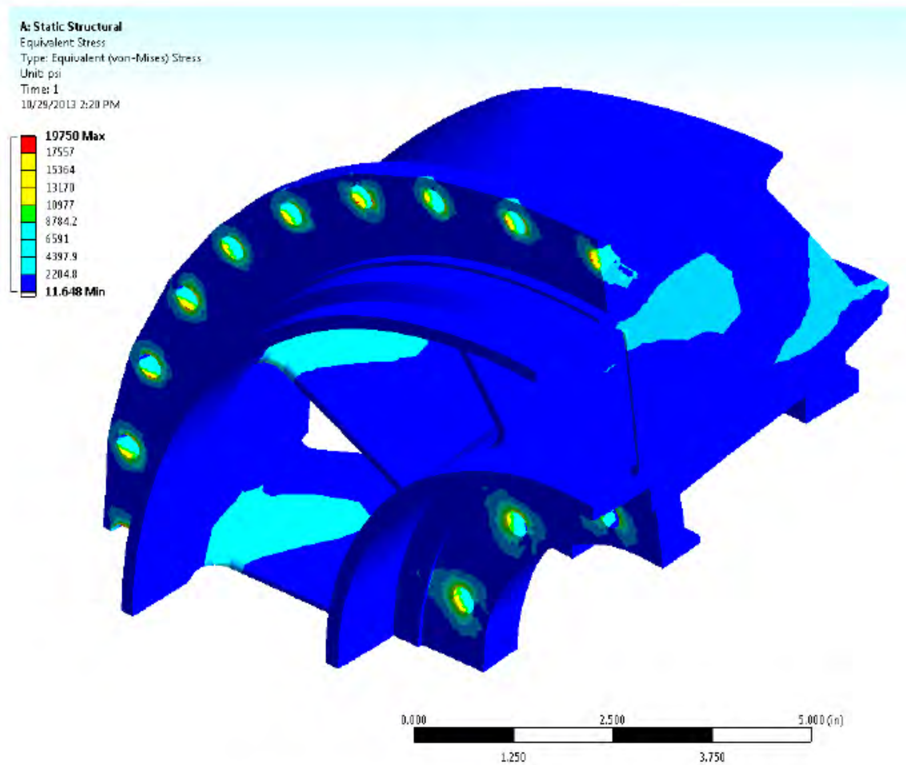
[36] 5/16-24
294 lbf
3.8 ksi

[24] 5/16-24
441 lbf
5.7 ksi

2x[28] 1/4-20
378 lbf
7.7 ksi

Analyses Results

[12] 3/8-24
883 lbf
7.9 ksi



20 ksi max stress

Budget and Schedule

- **Budget allocation of $40,000**

- **Main inlet component**
  - United Machine & Design [includes plate]: **$13,700 @ 6 weeks**
  - Mueller [confirmation on plate pending] : **$6,320 @ 8 weeks**
- **Stainless steel tube .120 wall thickness, cut on site**
  - Grainger **$800 @ 2 days** (off-the-shelf)
- **Pipe nipples and Swagelok fittings**
  - **$2000 estimate @ 2 weeks** (off-the-shelf)
- **New fittings**
  - **$400**
- **Welding**
  - **$5000**

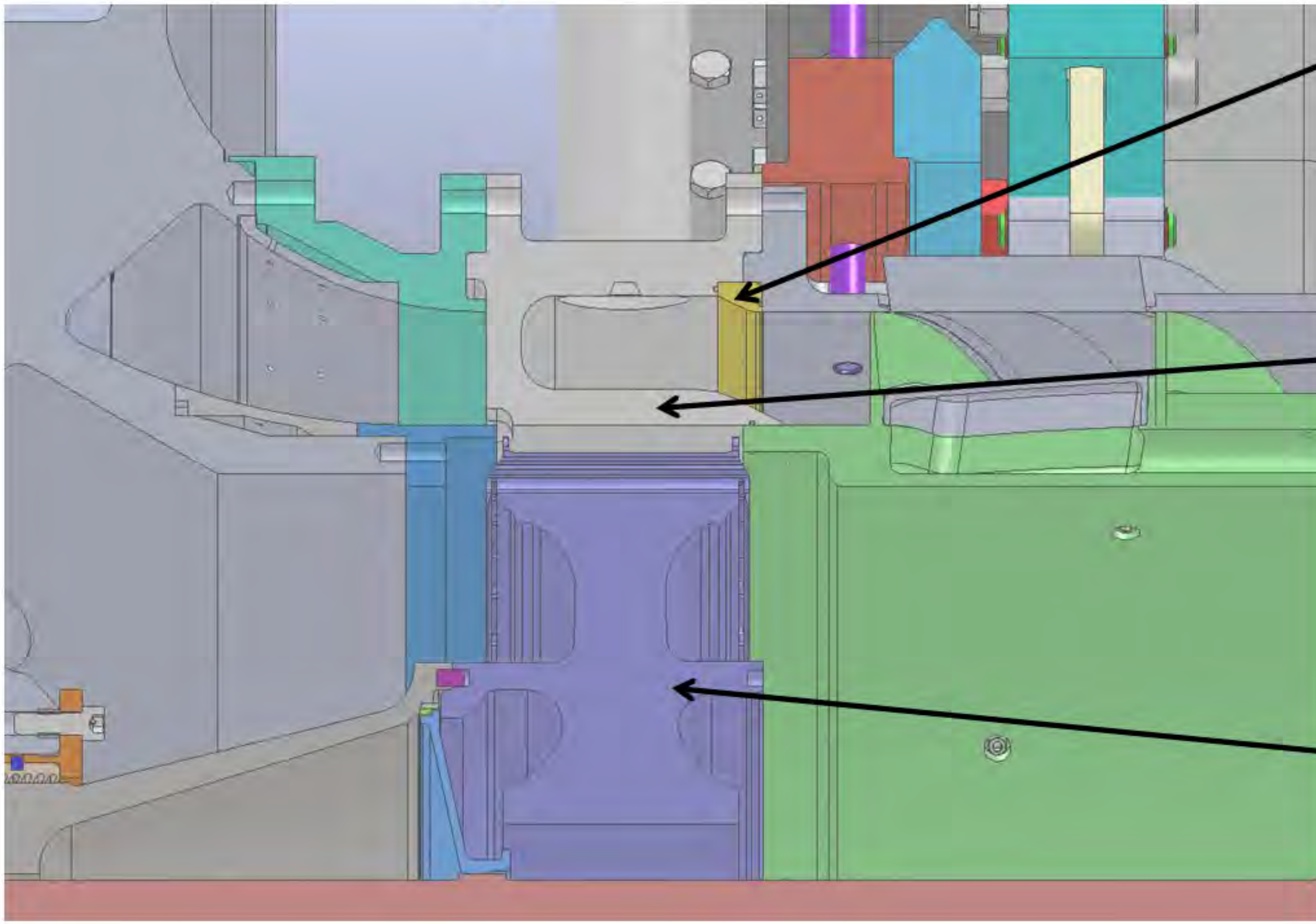
- **$21,900 expected total.**

Conclusion

- **Rig appears to be able to accommodate inlet design, recommend welding of additional gussets to combust case to ensure sufficient strength, and further review required for turbine shroud.**
- **Thank you for your time and input.**
  - PDR reference slides to follow

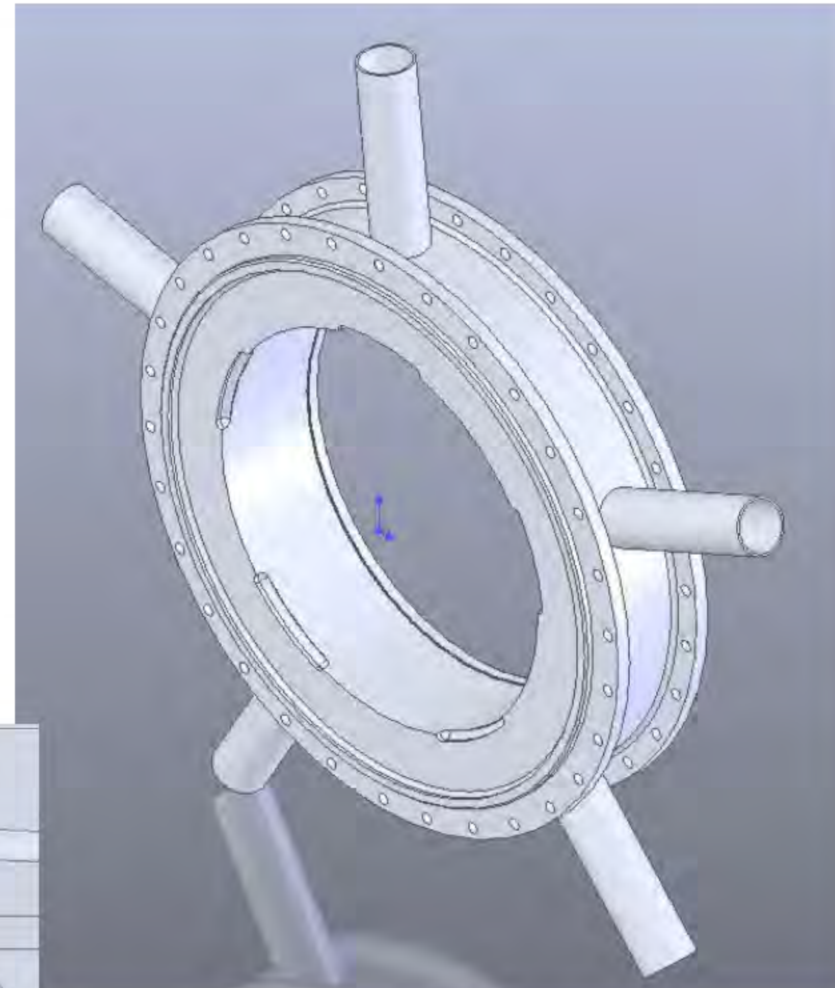
Rev 01

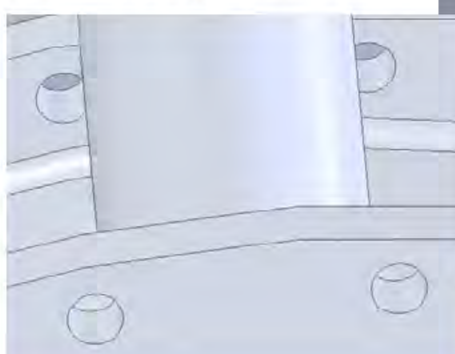
Two component inlet as viewed in current assembly



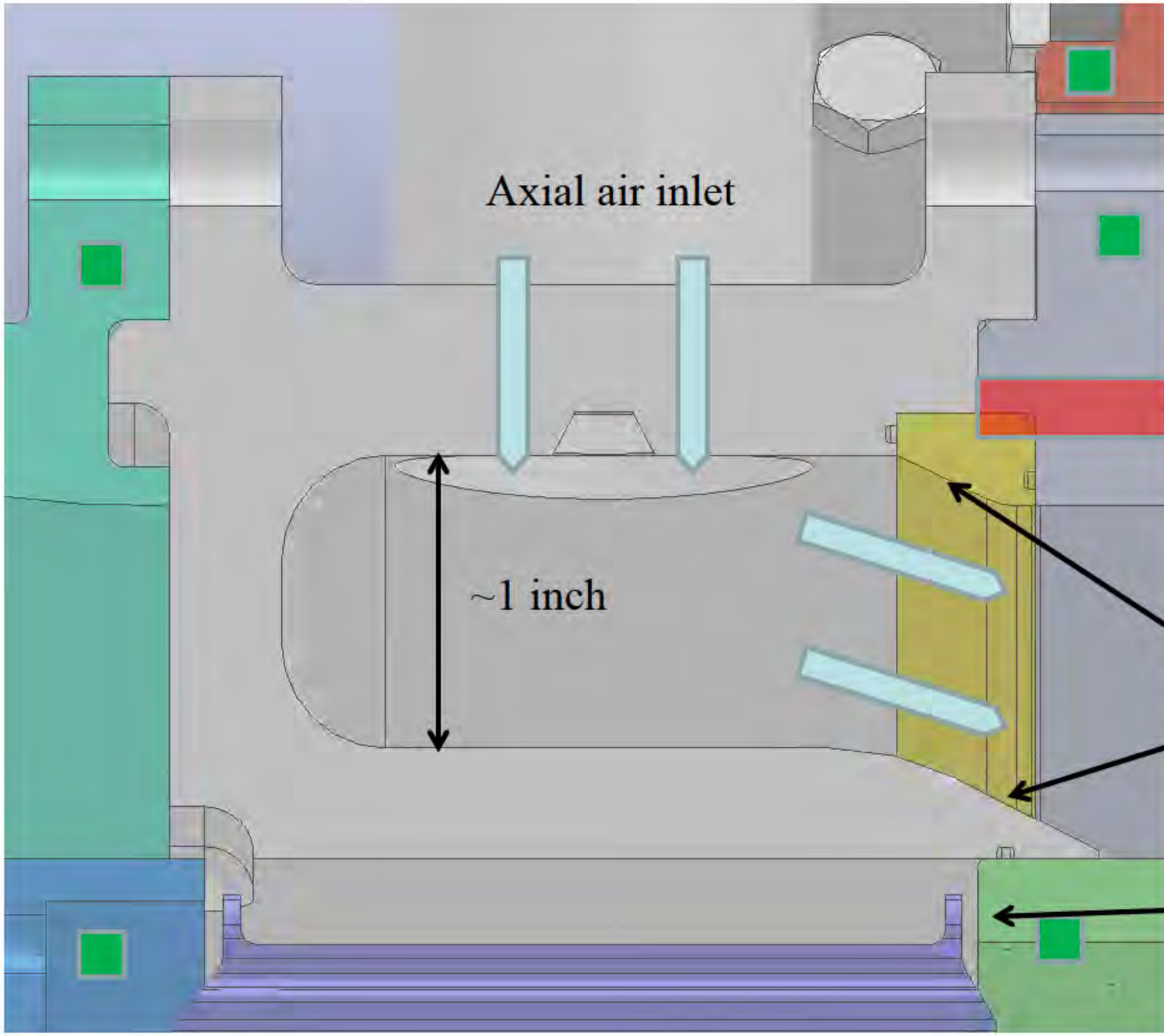
Rev 01

- **Inlet/cap**
  - Constructed of weld-friendly SS (410-316 TBD)
  - Designed to maximize tool access for ease of manufacture
  - Inlet horns accommodate existing methods of air delivery and ease of assembly (number of horns is variable; see later slides)
  - Horns are clocked 6 degrees out of phase with center line to dodge bolt pattern





Rev 01



With this design all components surrounding the cap and sealing ring are unmodified and in identical orientation to the prior configuration.

Through-hole to door space (must be sealed)

Flats/radius provide for smother flow

.125 clearance to rotation component

= Existing

Requirements

Hole Dia [in]	1.25	1.5
ID	1.12	1.37
Area [in^2]	0.985	1.474
Flow [lb/s]	10	

	6 to 1	10 to 1
Psi	87	147
Temp	180	180
p [lb/ft^3]	0.367	0.620

Experiment	6 to 1	Velocity		Experiment	10 to 1	Velocity	
Pipe	1.25	[ft/s]	M	Pipe	1.25	[ft/s]	M
hole #	5	796.86	0.64	hole #	5	471.77	0.38
	6	664.05	0.53		6	393.14	0.32
	7	569.18	0.46		7	336.98	0.27
	8	498.04	0.40		8	294.86	0.24
	9	442.70	0.36		9	262.09	0.21
	10	398.43	0.32		10	235.88	0.19

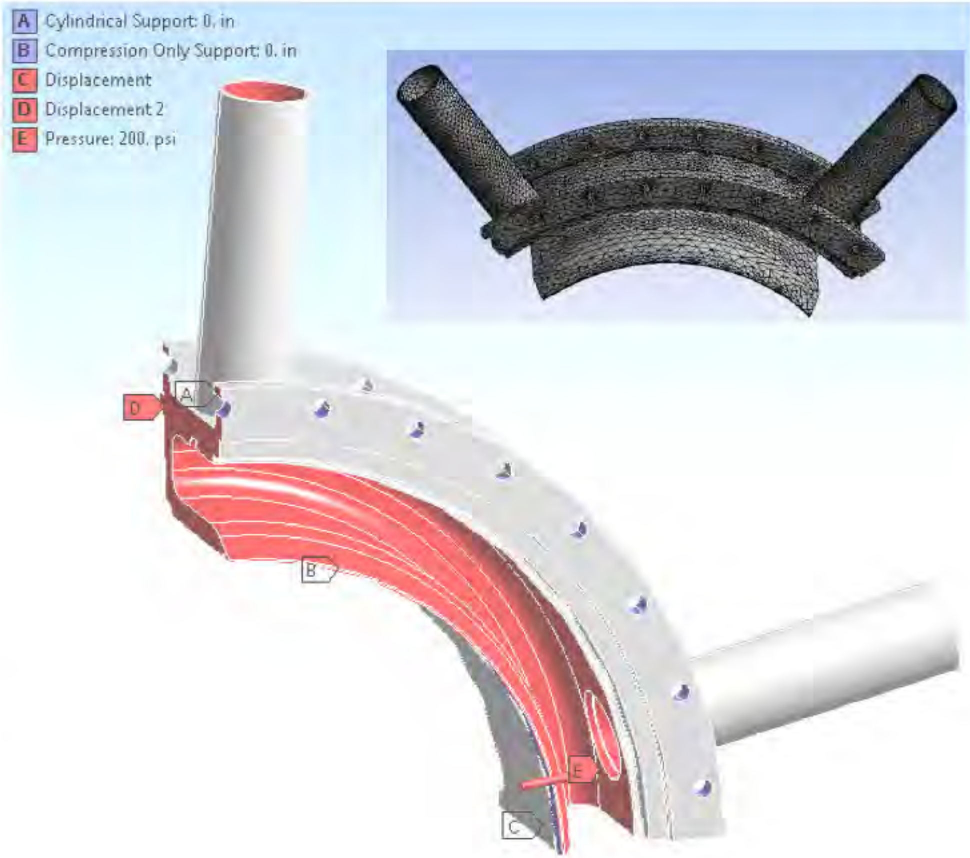
Experiment	6 to 1	Velocity		Experiment	10 to 1	Velocity	
Pipe	1.5	[ft/s]	M	Pipe	1.5	[ft/s]	M
hole #	5	532.57	0.43	hole #	5	315.30	0.25
	6	443.81	0.36		6	262.75	0.21
	7	380.41	0.31		7	225.21	0.18
	8	332.86	0.27		8	197.06	0.16
	9	295.87	0.24		9	175.17	0.14
	10	266.28	0.21		10	157.65	0.13

Table 1: Effect of inlet pipe diameter and number on axial inlet velocity for Rev 01

- **Deliver 87 (and/or 147) psi, 180 F air at 10 lb/s to static combustor with equivalent flow uniformity to existing OGV output**
  - Design can deliver air flow, and flow uniformity is accomplished with existing OGVs
  - Mach numbers can be managed with increase of pipes (table 1) and max mach number experienced in static diffuser is .314 independent of inlet configuration.

Requirements

- **Accommodate up to 200 psi internal pressure**
  - ANSYS analysis of the Inlet performed with an internal pressure condition of 200 psia to establish maximum anticipated stress and deformation.
  - Potential fabrication materials have a minimum yield stress of 45 Ksi (410 annealed) temperature with a variations of no more than 10% at max considered temperature of 325 F.
  - Main constraint for model was cylindrical supports allowing for radial growth on aft bolt circle anticipating forward load.



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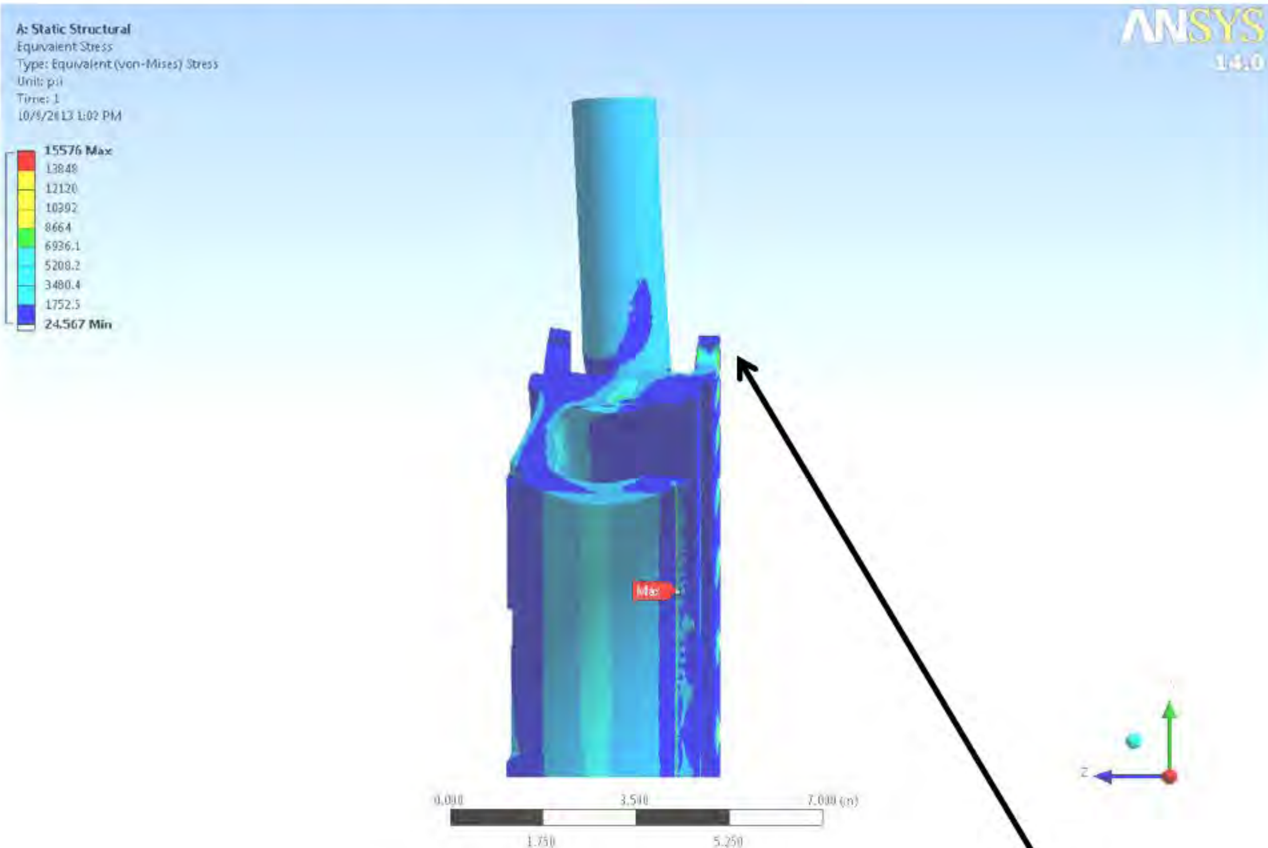
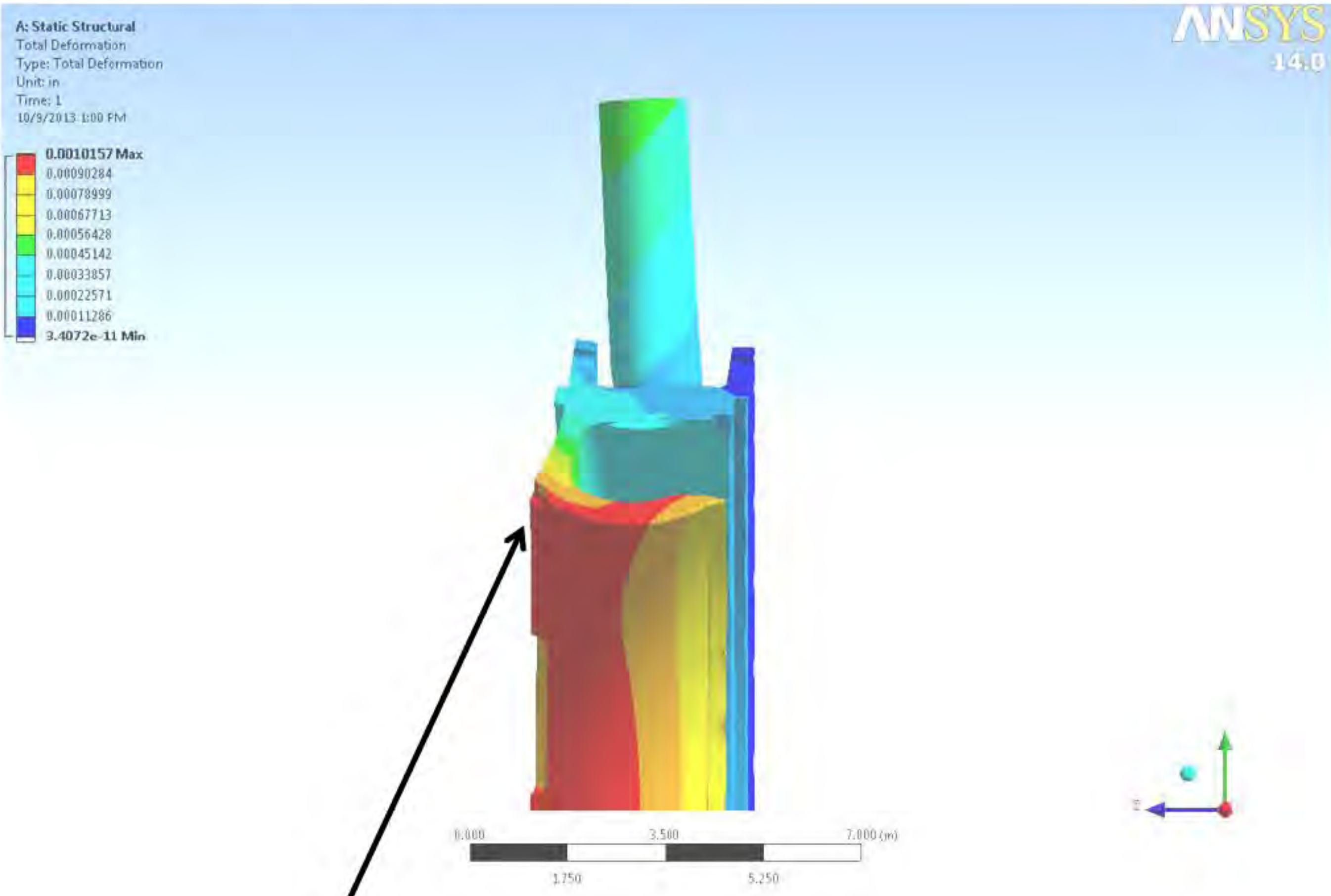
Requirements



- Max stress of 15 Ksi experienced along compression only support of front lip
  - SF of over 2.5 at temp.

Cylindrical support

Requirements

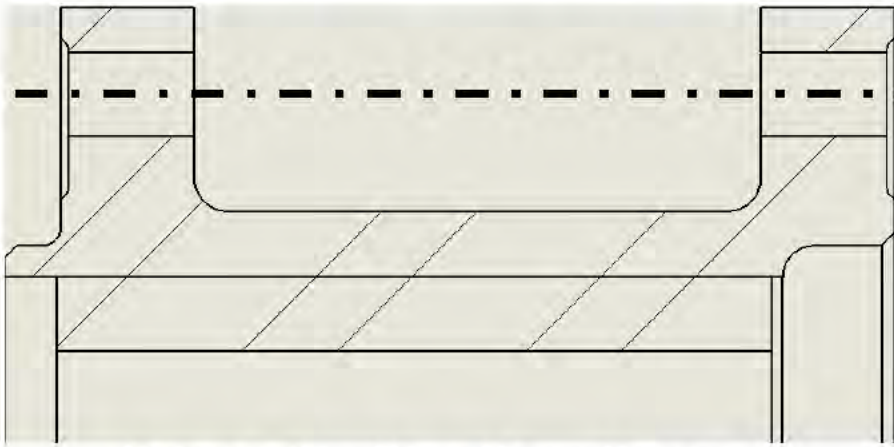


Unsupported deformation of .001 max experienced

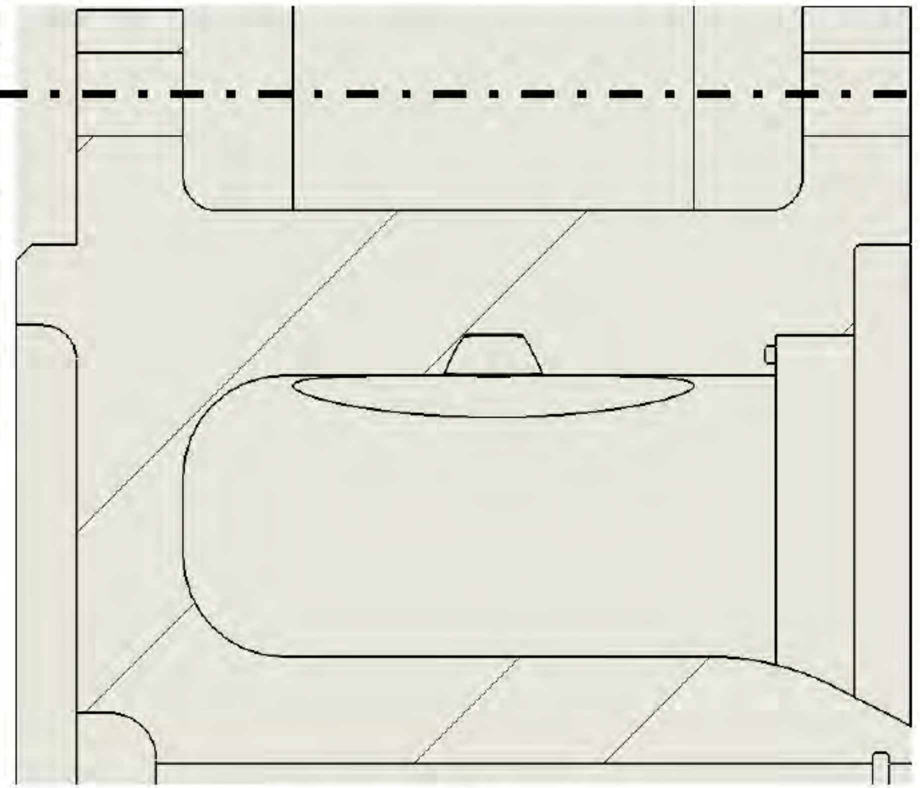
Negative radial movement minimal, closing gap to rotor to .124 while stationary.

Requirements

- **Maintain identical or greater stiffness relative to existing components**



Existing Tip-Ring Cross-Section

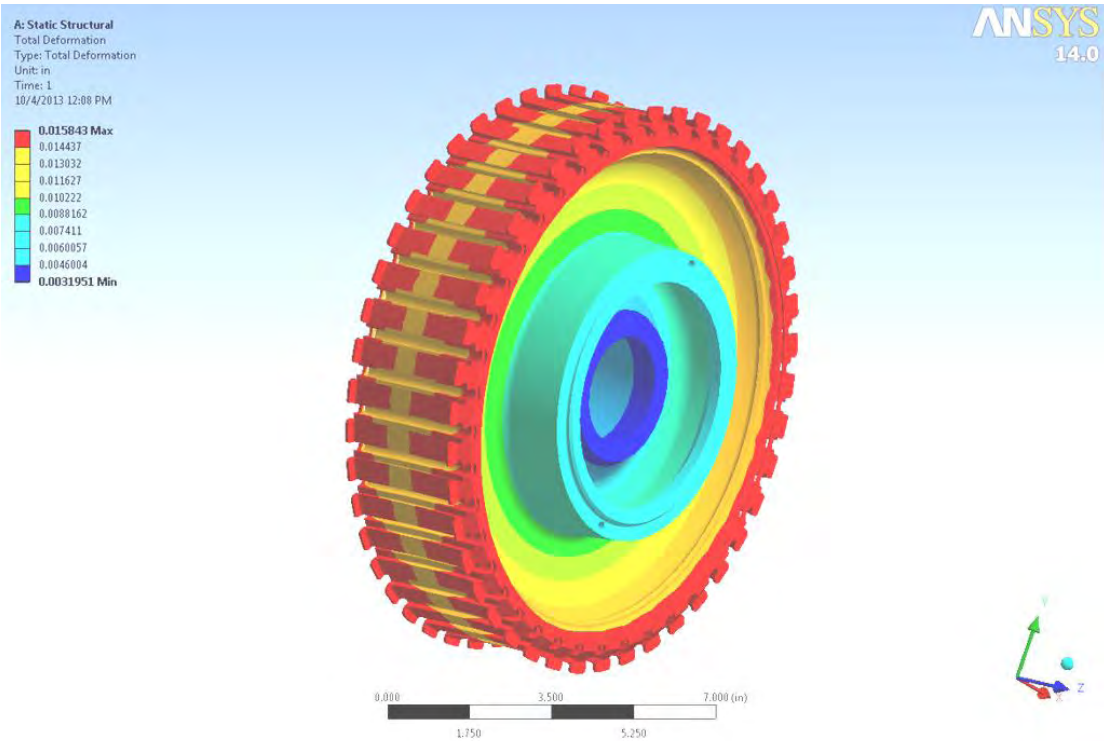


Design Concept

- **Structure Comparison**
  - Identical scale
  - Engages the same pilot fits
  - Only significant structural replacement
  - Stiffness maintained

Requirements

- **Accommodate thermal and axial growth of rotor section.**



Brief FEA analysis of rotor at design speed (24,000 rpm) and at 325 F could grow as much as .015

This narrows the gap between the internal inlet surface under full operation conditions to .1 inch.