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Hydraulic Shock Event and Analysis During Simple Cycle Testing at STEP Demo Facility

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ABSTRACT

In 2024, the 10 MWe Supercritical Transformational Electric Power (STEP) Demo pilot plant in San Antonio, Texas, successfully completed its Simple Recuperated Brayton Cycle (Simple Cycle) testing, demonstrating operability and efficiency of the supercritical carbon dioxide (sCO2) power cycle. However, during commissioning and testing, multiple hydraulic shock or 'fluid hammer' events occurred. A fluid hammer event is a phenomenon that occurs in piping systems when a sudden change in fluid velocity results in a momentary increase in pressure. This typically occurs with liquids due to their incompressibility; however, sCO2 when used as a high-density supercritical fluid is susceptible and concerns must be mitigated. These events were caused by the fast closing of the Turbine Stop Valve (TSV). Large displacements were experienced. Measurements and monitoring of the piping system's vibrations allowed the severity of these events to be assessed and analyzed. GTI utilized Flownex SE to model the sCO2 flow and pressure increase, while SwRI utilized CAESAR II to assess the loads and displacements experienced by the piping system. With both operational data and the models, the team implemented mitigations to reduce the severity of future fluid hammer events and have re-evaluated conditions that will be experienced when the STEP Demo facility is brought online in the Recompression Brayton Cycle (RCBC) configuration. This paper will discuss the fluid hammer event, the analysis to assess and mitigate the severity of the fluid hammer events, and implications for RCBC conditions. This important learning from STEP Demo operation should be considered in designing future sCO2 power systems.

INTRODUCTION

The STEP (Supercritical Transformation of Electrical Power) program is sponsored by the U.S. Department of Energy (DOE) and several commercial partners for the development and expansion of sCO2 power cycles. A 10 MWe pilot power plant facility has been constructed at the Southwest Research Institute (SwRI) in San Antonio, TX, in collaboration with GTI Energy and GE Vernova. The primary goals of the project include the development of a first-of-a-kind sCO2 Brayton cycle and testing of the component technologies. This encompasses the development of novel turbomachinery, piping, valves, and heat exchangers required for the cycle. Previous work, which documented development on the program, discussed the design and goals in detail [1][2][3][4][5].

Figure 1 STEP FACILITY IN SAN ANTONIO TX

The cycle is designed to flow sCO2 at temperatures and pressures up to 715°C and 252 bar, respectively, at the turbine inlet [5]. The project is envisioned to operate within two configurations of increasing complexity to reduce the program risk during commissioning and testing. The simple cycle (SC) configuration includes the turbine, compressor, a cooler, a heater, and a high-temperature recuperator (HTR). The simple cycle configuration has a maximum temperature of 500°C while operating at full pressure. The second configuration is the Recompression Closed Brayton Cycle (RCBC), which adds a low-temperature recuperator (LTR) and a bypass compressor (BC). The RCBC configuration will operate at the full 715°C temperature condition, demonstrating a pathway to elevated cycle efficiencies and higher power production. The facility has operated in a compressor recirculation configuration for the commissioning of the main compressor and many supporting systems in the plant. The simple cycle configuration testing has been completed after an extended testing campaign. The facility's efforts are now shifting towards the completion of the RCBC configuration.

In addition to the demonstration of component technologies, an operational goal of the project is to understand the system dynamics during startup, shutdown, transients, and steady-state operation. sCO2 presents unique challenges for flow loop operation and control due to rapidly changing density and speed of sound near the critical point [6]. Response and tuning of control valves, tuning and detuning of system acoustic natural frequencies, and system dynamics such as fluid hammer are all challenges for sCO2 systems [7].

Due to the high temperature, pressure, and fluid conditions, the facility has the largest loop of Inconel 740H piping in the world [5]. During testing, it was important to continuously monitor the dynamic response to allow real-time decision about piping component health and safety of the system. By monitoring the vibrations and dynamics of the piping loop, the testing team can prevent failures, identify conditions to be avoided based on mechanical or acoustic resonances, and find locations that require additional support. In the event of a turbine trip leading to a fast TSDV closure, dynamic monitoring allowed the team to gauge the severity of the state of the equipment and make modifications for future testing.

SwRI operators used FieldDAS, an in-house National Instruments (NI) Labview tool, for collection of accelerometers, dynamic pressures, and microphone data used to ensure the health of the piping loop and the equipment. More details on the monitoring set-up and implementation can be found on [9].

SIMPLE CYCLE TESTING OVERVIEW

Figure 2 shows a picture of the facility in the simple cycle configuration, while a simplified diagram of the simple cycle configuration is shown in Figure 3. Major equipment incorporated includes the main compressor, main cooler, the high-temperature recuperator, gas-fired heater, and turbine. sCO2 is heated in the heater to desired conditions for that experimental case. The sCO2 passes through the turbine stop and control valve (TSV) and is then expanded through the turbine. Heat is recovered in the high temperature recuperator (HTR) before the flow enters the main cooler and main compressor. The main compressor is driven by a Variable Frequency Drive (VFD) electric motor to control the speed of the compressor. Prior to entering the main compressor, the sCO2 is cooled in the main cooler. A bypass loop, which includes a flow path from the compressors to the recycle valves and through the coolers, allows for the compressors to be started independently, as well as to provide anti-surge protection. The exit stream of the main compressor flows through the HTR and then re-enters the heater.

The simple cycle configuration was constructed in October 2023. Commissioning and testing took place between February 2024 through October 2024. Three testing periods occurred during this time: February, May, and September/October. Specific component issues caused pauses in between the testing periods.

Figure 2 HIGH LEVEL OVERVIEW OF FACILITY

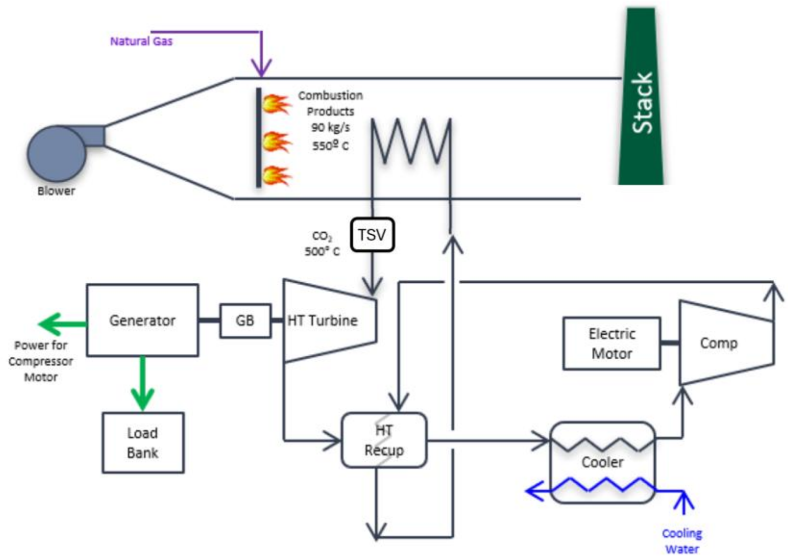

Figure 3 CYCLE OVERVIEW FOR SIMPLE CYCLE CONFIGURATION

PIPING DESIGN

Previous work describes the design, supply chain development and fabrication of the piping loop [5]. The process piping at the STEP facility is divided into two sections, high and low temperature sections. The high temperature section includes pipes 10" and 8" in diameter made of the nickel super alloy Inconel 740H and is found between the heater and the recuperators. The low-temperature section is made of P91, a chrome moly steel alloy. In addition to the process, piping for the cooling water and lube oil drains use carbon steel. The lube oil supply and tanks are stainless steel [5]. Standard ANSI type flanges are used on the low-temperature sections of the test loop, and clamp styled flanges such as Grayloc and Wireloc are used on the high-temperature pipe due to stress concerns at the elevated temperatures. During operation, due to the relatively accessible critical conditions for CO2, there are times when all the process piping is in contact with the fluid in supercritical form.

One of the hardest challenges in designing the piping system was accommodating thermal growth while ensuring equipment loads are within allowable limits. These design considerations conflict with one another since the high temperatures result in large thermal growth, but due to the high pressure and temperatures, the small sCO2 machinery packages have low allowable loads requiring minimal displacement. The solution was to include large expansion loops, which are mostly used on the connection to the heater and to the higher-temperature recuperator. The length of these expansion loops is increased for this facility as it is intended to have a piping arrangement that can be easily reconfigured to accommodate future testing of other equipment. The section between the turbine and HTR specifically had to be designed to best reduce the stress on equipment over a relatively small space, leading to various turns in what could have been a short section. The high-temperature sections of the loop are supported using spring handers and pedestals to accommodate the thermal growth needed. No buffer volumes for pulsations were added to the facility to conserve costs, especially on the high-temperature sections on the loop where the supply chain for Inconel 740H was already being pushed beyond what was readily available [5].

During the design and commissioning phase, the piping system was analyzed using CAESAR piping software, Strouhal Analysis and modal testing. The CAESAR model focused on thermal stresses and equipment allowable loads, while the Strouhal analysis and modal tests obtained data to predict how the system would react to changing fluid conditions, results are further discussed in [9]. Further details of the design analysis of the piping system have been reported in [9].

DATA COLLECTION METHODS DESCRIPTION

The software used for data gathering is a SwRI in-house developed package called FieldDAS. The software package was developed using LabVIEW and works with National Instruments hardware. The software is designed for customizability based on the dynamic needs of field studies. The flexibility of the system has been especially useful during testing campaigns by allowing rapid modification of the test points to better capture changing points of interest. The system allows for both frequency and time domain data tracking and display. In addition, it is possible to perform in-line numerical integration or differentiation on the data, which lets the user have flexibility on the recorded measurement units. Raw data can also be stored for post-processing at a later time, a useful feature if a correction factor is determined. Data was collected at high frequencies in the thousands of Hz; frequency varied depending on the number of test points being collected. This frequency, although higher than what was required, helped ensure that all transient data was collected. Usually, the frequency range being actively monitored during testing would be between 0-1,000 Hz, ensuring that all machinery related frequencies (~450 Hz at full speed) and piping frequencies were gathered. The equipment used for data collection on the loop consisted of the following:

- 8 Chassie DAQ
- 6 NI-9234 Cards (BNC connections)
- 2 NI-9229 Cards (BNC connections)
- 8 Tri axis accelerometers (PCB 356A33)
- 3 dynamic pressure probes (PCB 112B05 with a PCB 422E01 charge converter)
- 1 One-axis accelerometer (PCB308M87)
- 1 Microphone

Refer to [9] for further details of the software and hardware setup.

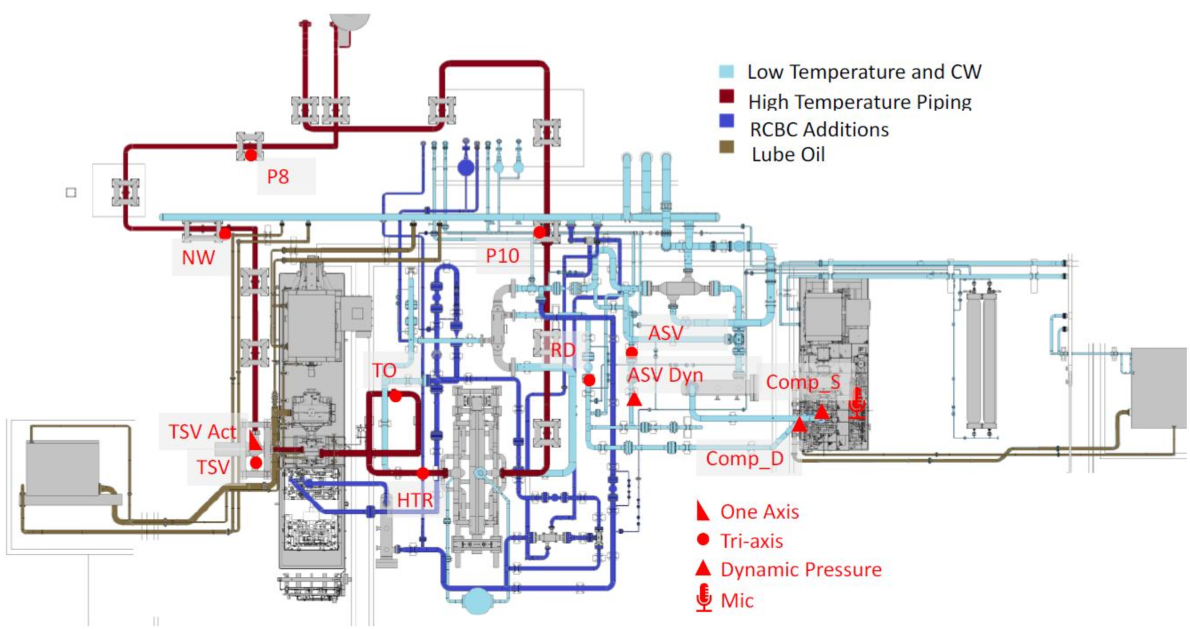

Figure 4 SENSOR MAP FOR SIMPLE CYCLE TESTING [9]

HYDAULIC SHOCK

One area of concern during simple cycle testing was a hydraulic shock or “fluid hammer” effect caused by rapidly closing the turbine stop and control valves (TSV and TCV). In this scenario a pressure wave is created and reflected through the system by a sudden stopping of the flow. The turbine stop valve is in place to protect the turbine from an overspeed event. The elevated densities of sCO_2 make this a complex problem since they typically increase the severity when compared to other gases and other fluids (such as water). The entire hot section of piping rests on spring hangers due to its high thermal expansion and the low nozzle allowable loads of the equipment. This support does not offer a lot of stiffness, adding to the risk of a sudden TSV closure, which can close within 200 milliseconds.

During commissioning and testing, an emergency stop occurred in which a hydraulic shock event took place on February 8, 2024. The data was collected, and the team was able to analyze and assess the severity of the event. Large displacements were experienced during the events. Figure 5 shows a video screenshot with an overlay indicating the estimated displacement observed. Figure 6 shows the severity of the hydraulic shock using a piping severity chart as described in [10], the figure has indications of acceptable vibrations in accordance with API 618. This emergency stop occurred with the turbine operating at low speeds.

A second emergency stop occurred on May 8, 2024, while the turbine was operating at full speed. Piping system modifications were implemented because of the February 2024 event, but excessive levels were still experienced and measured in specific locations. Additional analysis and design modifications were implemented due to the May 2024 event.

Lastly, a controlled emergency stop occurred on September 27, 2024, while the turbine was operating at full speed and partially loaded. All the design modifications implemented reduced vibration severity at those locations.

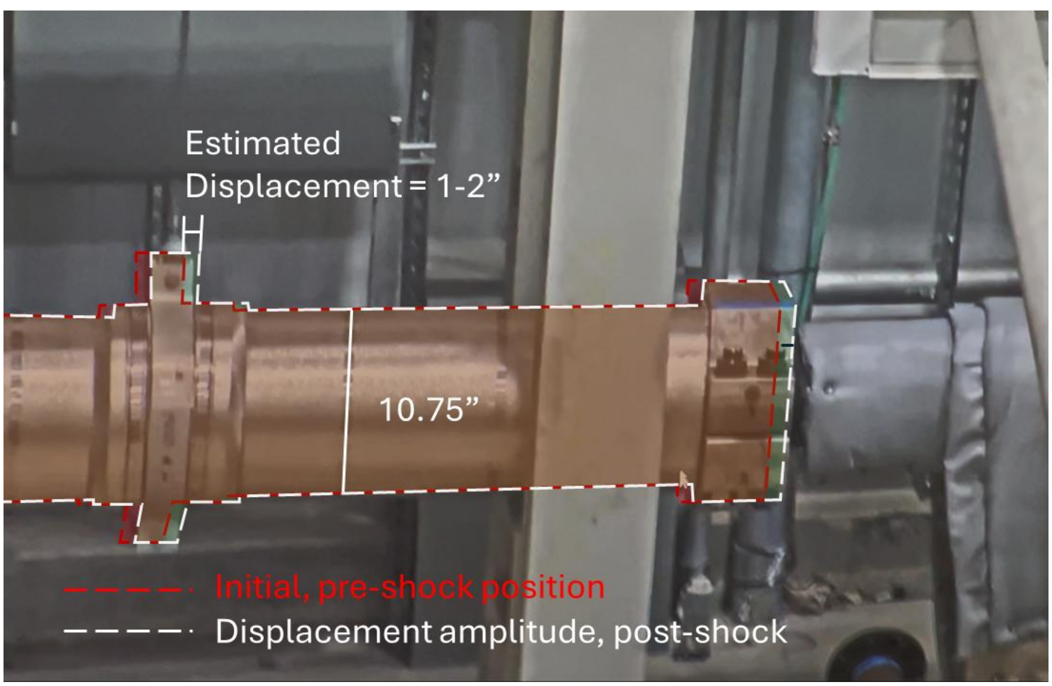

Figure 5 VIDEO STILL OF DISPLACEMENT EXPERIENCED DURING FEBRUARY HYDRAULIC SHOCK EVENT

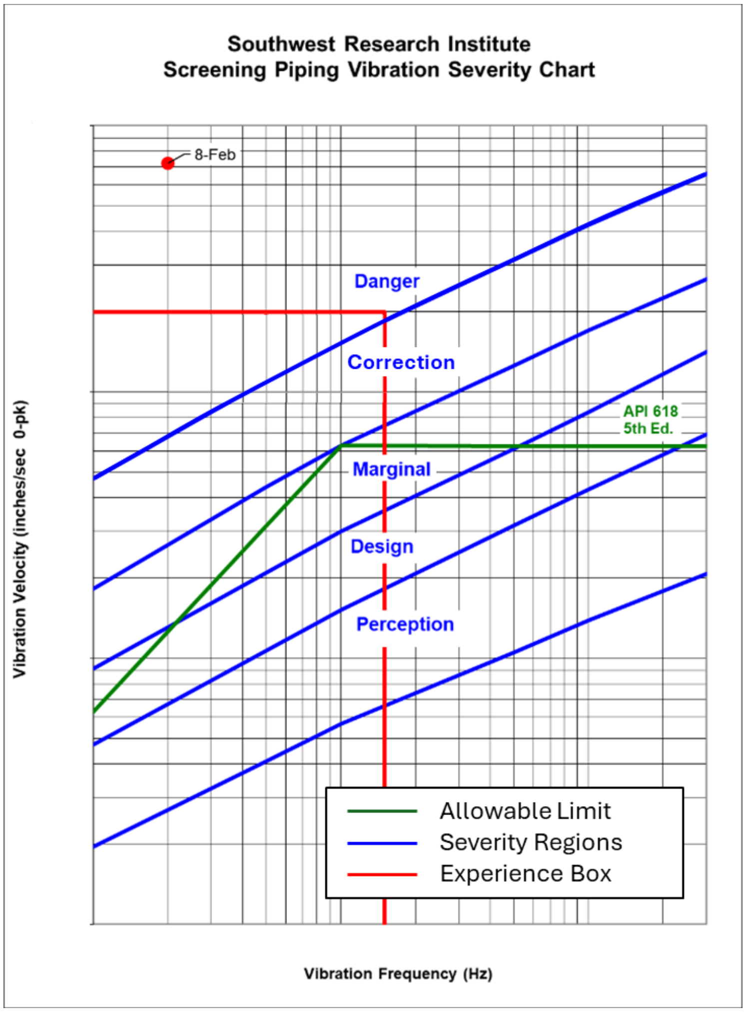

Figure 6 ASSESSED SEVERITY OF THE "FLUID HAMMER" EVENT ON FEB 8, 2024 [9]

HYDRAULIC SHOCK MITIGATION ANALYSIS

Figure 7 shows a simplified flow diagram of the actions taken to recover from the hydraulic shock event from the time of the actual emergency trip event during testing to the implementation of the design modifications. The vibrations experienced in the hot section of the piping system exceeded allowable limits and modifications were needed to reduce the severity of future events. Bump stops, shown in Figure 8, were identified as a possible solution since bump stops allow for thermal growth of the system, while mitigating the effects of displacement, and consequently the reaction loads, from a fluid hammer event. This section will focus on the design analysis performed to identify the locations of additional bump stops implemented to the piping system. Bump stops were incorporated in the original piping system design to control loads on the turbine due to thermal expansion. One bump stop is located upstream of the TSV and the second bump stop is downstream of the turbine, prior to the low pressure HTR inlet. Figure 9 shows the location of the bump stops in the original piping system design, prior to the first fluid hammer event.

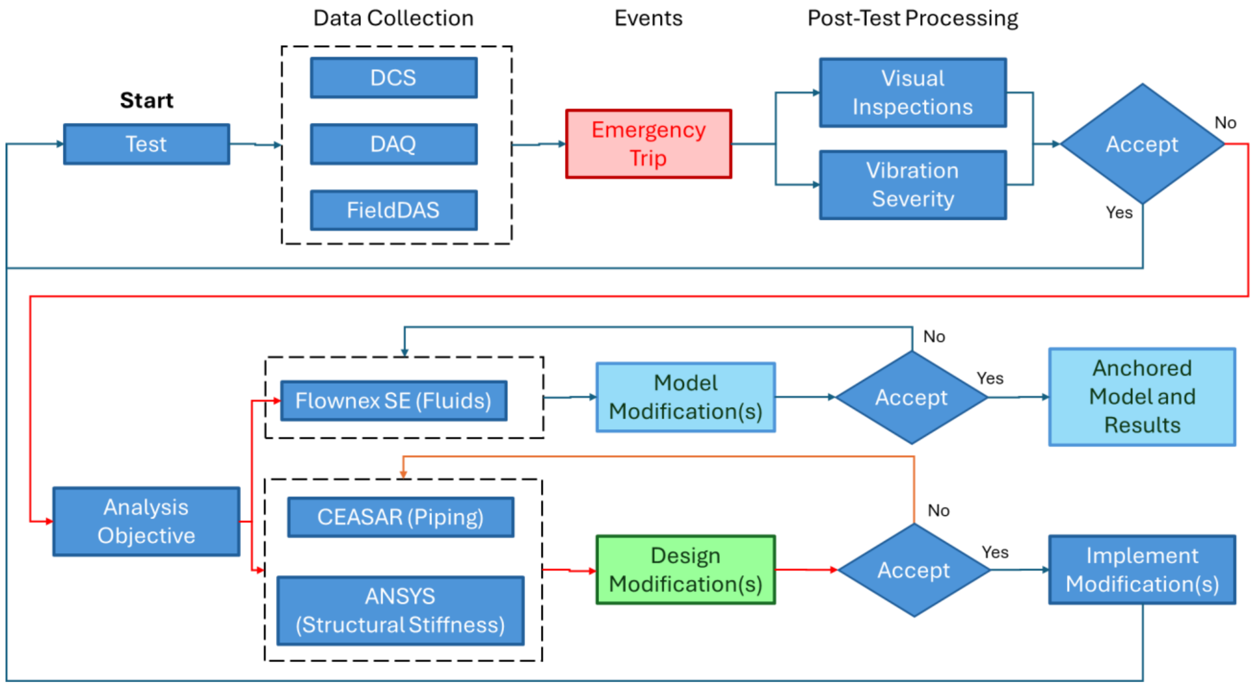

Figure 7 FLOW DIAGRAM OF THE ACTIONS TAKEN TO IMPLEMENT DESIGN MODIFICATIONS TO THE PIPING SYSTEM

Figure 8 BUMP STOP EXAMPLE

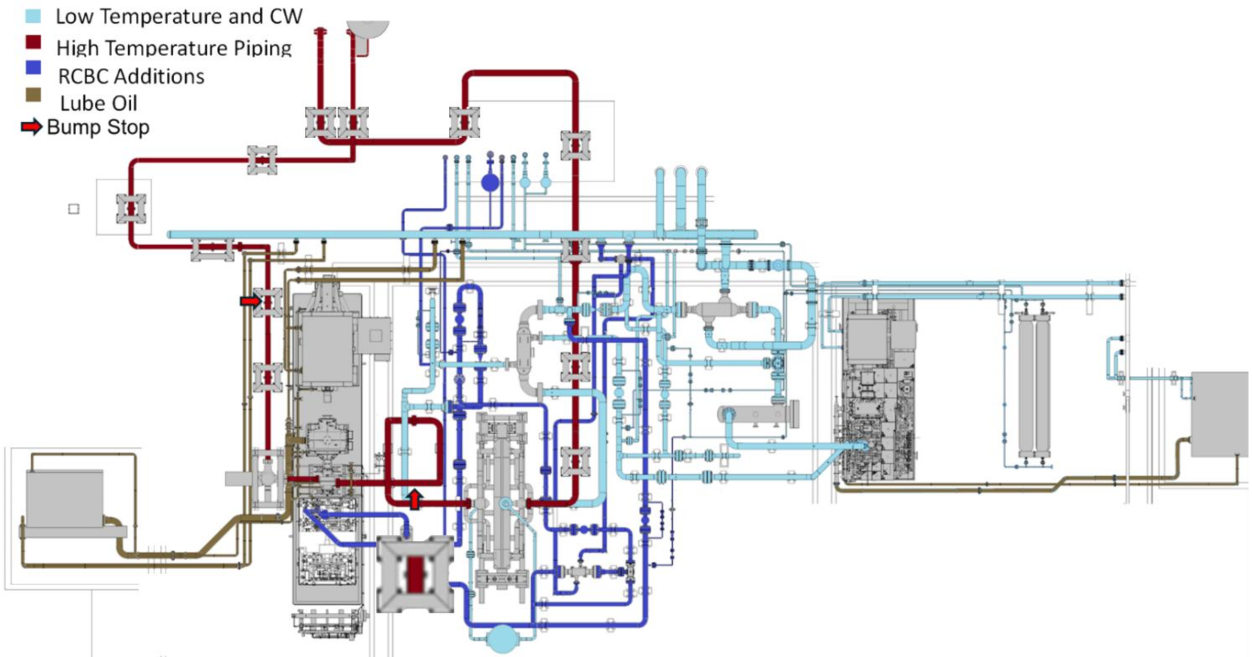

Figure 9 ORIGINAL BUMP STOP LOCATIONS IN PIPING SYSTEM DESIGN [9]

ANALYSIS APPROACH

Due to the limitation of the data recording speed, the Flownex SE dynamic model was used to verify the magnitude of the pressure pulse, as well as verify the model with test data. Flownex SE is an object-oriented simulation environment developed and verified in the nuclear industry that simulates 1D fluid networks with heat transfer and controls capabilities. GTI Energy has generated models for the simple cycle and RCBC configurations that will be tested in the facility [8]. SwRI performed a dynamic piping stress analysis utilizing CAESAR II. A static model was generated during the initial design phase to understand thermal growth and equipment nozzle loading. A dynamic model was developed once the fluid hammer event occurred to predict the force and displacement induced by the fast closure of the TSV. This model was the primary analysis tool to determine bump stop locations to be added. To supplement the CAESAR dynamic model, ANSYS Mechanical models were used to verify bump stop stiffnesses used in the CAESAR dynamic model.

FLOWNEX SE ANALYSIS

After the fluid hammer event occurred, the team analyzed the data to assess the severity. Figure 10 shows the recorded measurements from the test: turbine and compressor speeds, associated system pressures, turbine inlet temperature and TSV and ASV positions at the time of the emergency trip. The Flownex dynamic model was used to assess the peak pressure experienced during the event and to assess worst case scenarios: i.e. system at maximum operational conditions. Figure 11 shows the comparison of test data to model results for the Feb 8, 2024, event.

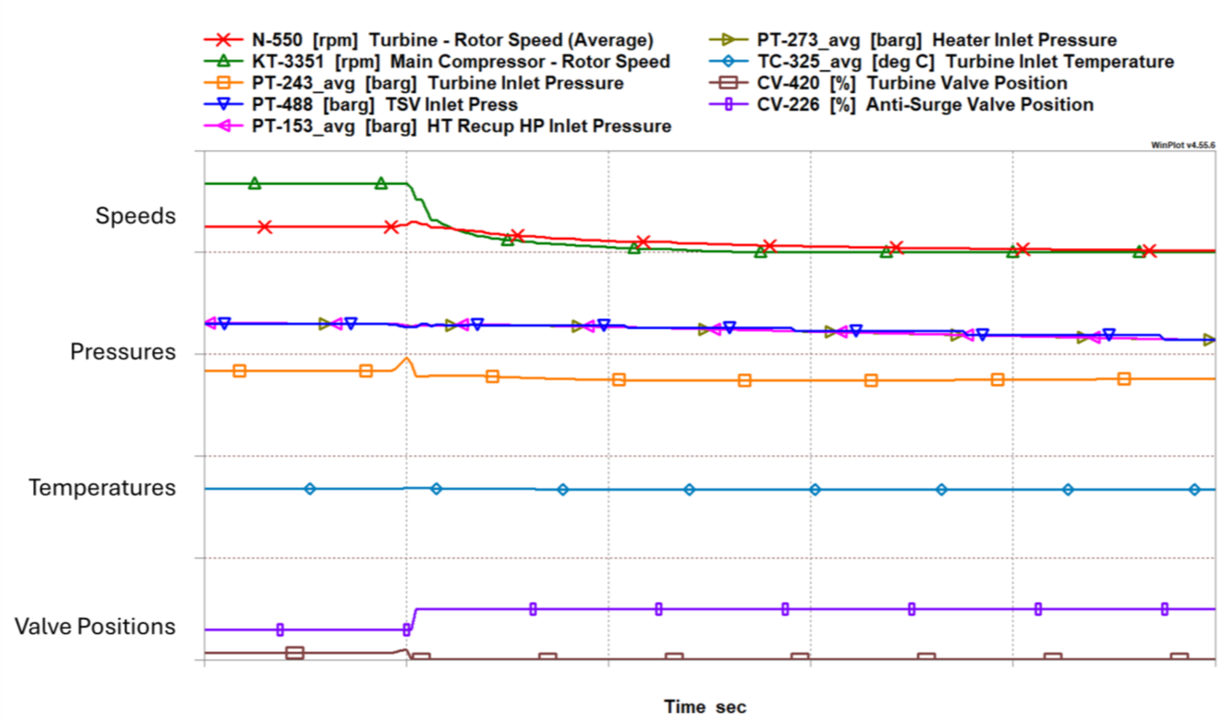

Figure 10 RECORDED SYSTEM MEASUREMENTS FROM FEB 8, 2024, FLUID HAMMER EVENT

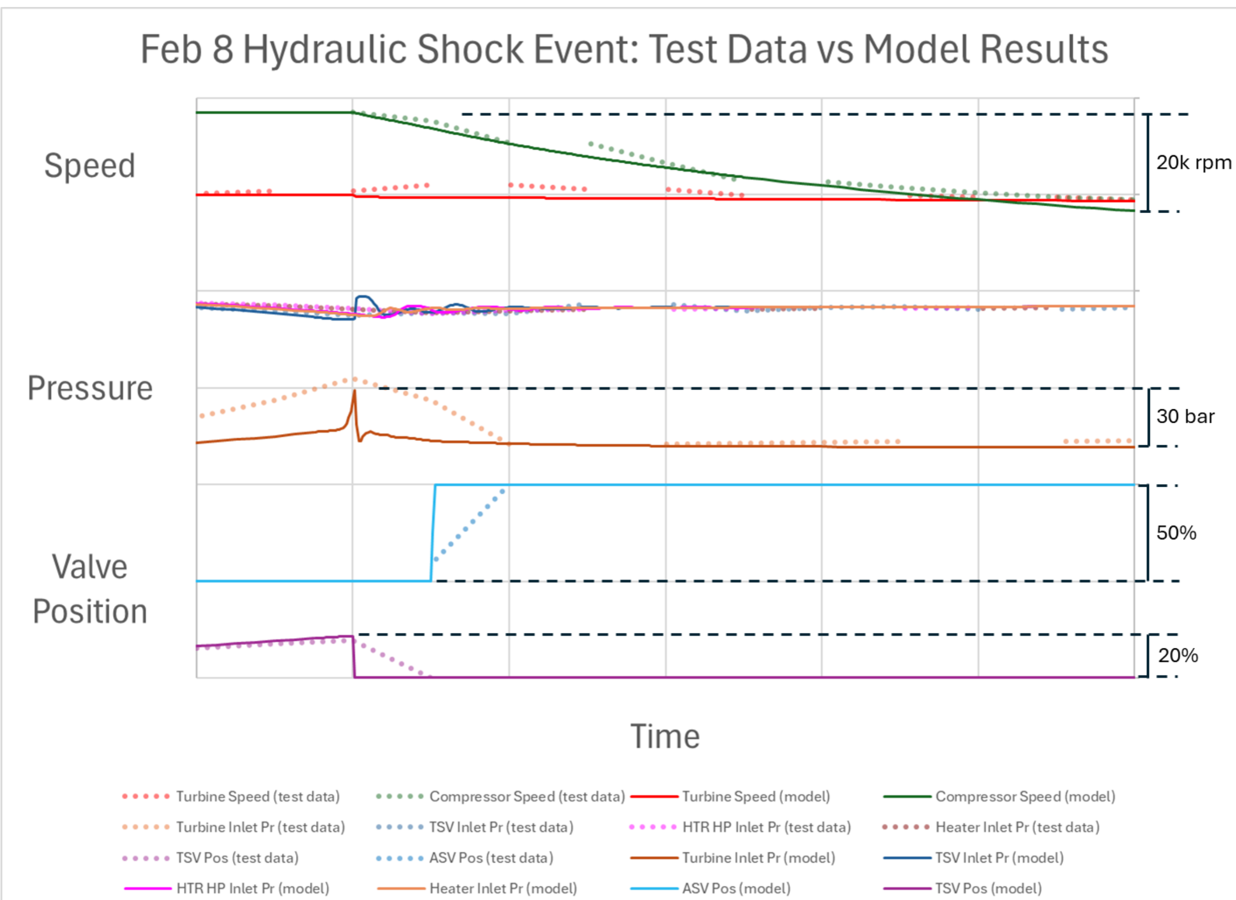

Figure 11 COMPARISON OF FLOWNEX SE MODEL RESULTS TO TEST DATA

The results of the model were very comparable to the recorded data. The model results provided additional insight into the pressure pulse, since the model can produce results with higher fidelity through shorter time steps than can be detected by the instrumentation records. The test data was mostly recorded at 1Hz due to data storage concerns with some variables with recorded at a higher rate. This reduced the ability to observe quick time events, similar to the pressure pulse of the fluid hammer. These results verified the model predictions and there were no additional concerns from the test to assess.

The following analysis was performed to assess the worst-case scenario for a fluid hammer event during simple cycle testing with max power operating conditions and a fast TSV closure time. During the fluid hammer event in February 2024 the system was operating at low turbine speed, reduced temperature and pressure. The analysis modeled max power conditions for simple cycle, with TSV closure times of instantaneous, 200 milliseconds, and a long closure time (approximately 2 seconds). The purpose was to assess the worst-case pressure pulse (max conditions with an instantaneous TSV closure), while understanding the TSV closure time effects on the peak pressure. Figure 12 shows the results while Table 1 displays peak pressure changes for the various closure times. One item to note, for the simulation of the February 2024 fluid hammer event, the closure time is very quick due to the valve position being approximately 10% open at the time of the emergency trip. Therefore, the closure time is much faster than the 200-millisecond closure rated time from full open. The results of the assessment indicate the initial pressure spike magnitude correlates with a faster closure time, which is nonexistent at a closure time of 200 msec. However, the primary fluid hammer pressure wave is not significantly impacted by the fast closure time.

Table 1 RESULTS SUMMARY OF TSV CLOSURE TIME VS PEAK PRESSURE PULSE

Case	Description	Closure Time sec	Peak Press Increase %
1	Max Conditions: Instantaneous Closure	0.002	10.4%
2	Max Conditions: Rated TSV Closure	0.200	9.5%
3	Max Conditions: Long Closure	1.940	0.0%
Test Data	Actual: Feb 8, 2024 Test Conditions	0.025	6.5%

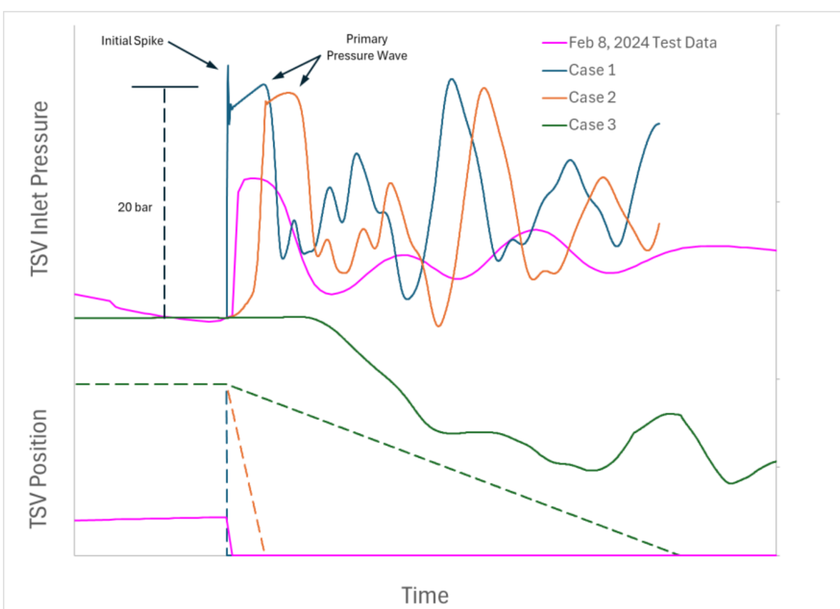

Figure 12 FLOWNEX MODEL RESULTS OF TSV CLOSURE TIME VS PEAK PRESSURE PULSE

BUMP STOP DESIGN METHODOLOGY: DYNAMIC CAESAR ANALYSIS

The dynamic CAESAR model was used to evaluate the force-displacement effects of the pressure pulse from the fluid hammer. Bump stops were incorporated into the model to minimize stresses and equipment nozzle loads. The allowable locations were limited due to the original design of the piping system and material constraints at high temperatures. The new bump stops would need to allow the pipe to grow during thermal expansion to keep pipe stress low. Figure 13 shows the HTR-to-Heater section of the model. Multiple bump stop configurations were evaluated to minimize loads on the HTR, Heater and piping connection in the middle of this section. Bump stop configuration variables included, but not limited to, location, number of stops, gaps (thermal growth allowance). Table 2 shows the max load compared to the allowable load for each component and each bump stop configuration.

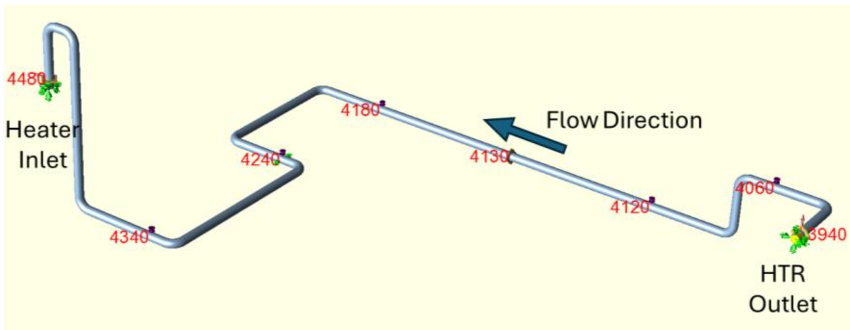

Figure 13 CAESAR PIPE SECTION: HTR TO HEATER

Table 2 CAESAR DYNAMIC MODEL RESULTS FOR BUMP STOP CONFIGURATIONS

		Highest Load-to-Allowable %			
Case	**Description**	**HTR Out**	**Greylock**	**Heater In**	**Comments**
1	Baseline Condition: Feb 8, 2024	151%	161%	419%	
2	Bump Stop	83%	64%	300%	
3	2 Bump Stops, No Gap	53%	97%	114%	Bump Stop supports cannot withstand thermal growth
4	2 Bump Stops, Gap 50%	53%	90%	155%	
5	2 Bump Stops, Gap 75%	52%	88%	188%	As compliance gets larger, max load on Heater increases
6	Final Config: 2 Bump Stops, Gap 75%	57%	85%	137%	Updated Allowables, Approved for use

The final configuration did not meet all allowable load limits, but approval on prior nozzle loads exceeding the allowable limits were obtained at similar values. Approval for use was obtained.

The analysis illustrated was just for the high-pressure HTR outlet-to-Heater piping section. This analysis was completed for all hot-pipe sections. Once the analysis was completed for all hot sections of piping, several bump stops were added at the inlet and outlet of the heater. As mentioned previously, the two bump stops noted on the bottom of the image were already existing before the trip to control loads on the turbine due to thermal expansion. A modification was performed for the bump stop between the turbine and HTR after the turbine trip in May 2024. The bump stop was changed to be adjustable like the one shown in Figure 8, previously. All bump stops were welded to existing support structures, and Ansys modeling was conducted to verify that the stiffnesses were comparable to the CAESAR model. The addition of the bump stops allowed for the thermal expansion of the system while mitigating the effect of the fluid hammer event and minimizing the loads on the equipment nozzles. Figure 14 shows all bump stops installed after February 2024; three newly installed bump stops, while two were installed as part of the original design.

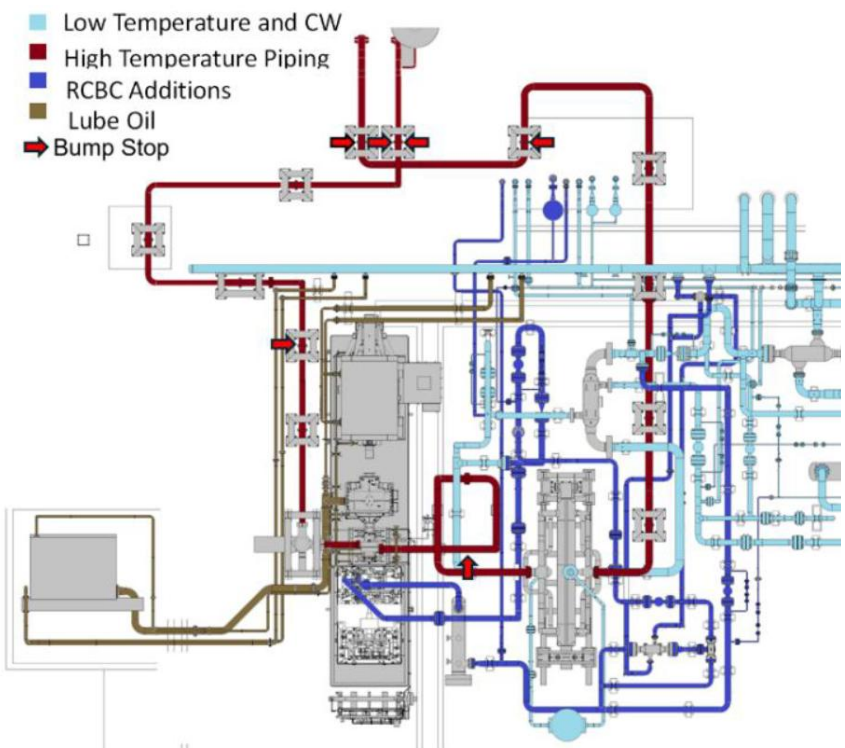

Figure 14 BUMP STOP PLACEMENT MAP [9]

ANSYS STRUCTURAL STIFFNESS VERIFICATION

The CAESAR piping model was the primary tool used to design the piping system, evaluating pipe stresses, thermal growth and reactive equipment nozzle loads. However, to accurately predict loads and displacement, all inputs into the model must be as accurate as possible. This includes supports, structures and major equipment. These boundary conditions are typically simplified to reduce simulation run time but need to provide enough detail and accuracy for the model to generate representative results. Therefore, the SwRI team supplemented the CAESAR model with ANSYS models to verify boundary conditions' stiffnesses, or displacement caused by a load. One example of an ANSYS model stiffness verification is illustrated using the turbine frame. Figure 15 shows an example of the turbine frame simplified into boundary conditions using beam elements and constraints.

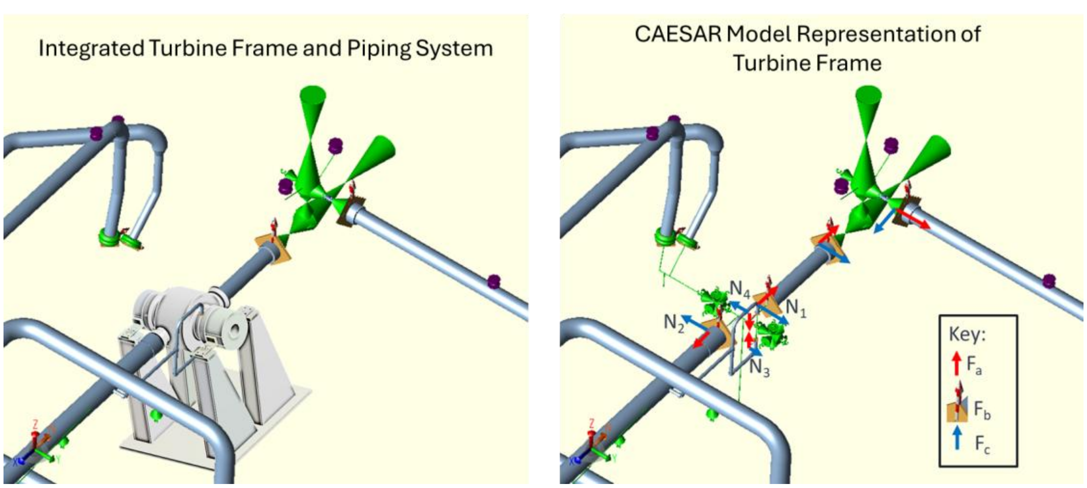

Figure 15 CAESAR MODEL REPRESENTATION OF STRUCTURAL FRAMES AND SUPPORTS

To accurately represent the turbine casing and turbine frame within the piping model, the turbine casing is modeled as beam elements, while the frame is modeled as support constraints. The supports have stiffness, or spring element, applied to replicate how the frame will deflect under loads. To ensure accurate stiffness is applied to the piping model, an ANSYS 3D static structural model was generated and used to calculate a stiffness, or spring constant, in each direction. A load is applied in each direction and the corresponding displacement is calculated by the model. This load-to-displacement ratio, or spring constant, is the calculated stiffness of the turbine frame for that direction. Figure 16 shows a displacement contour for the rotational direction. Table 3 shows the calculated directional stiffnesses, which would be incorporated into the CAESAR models.

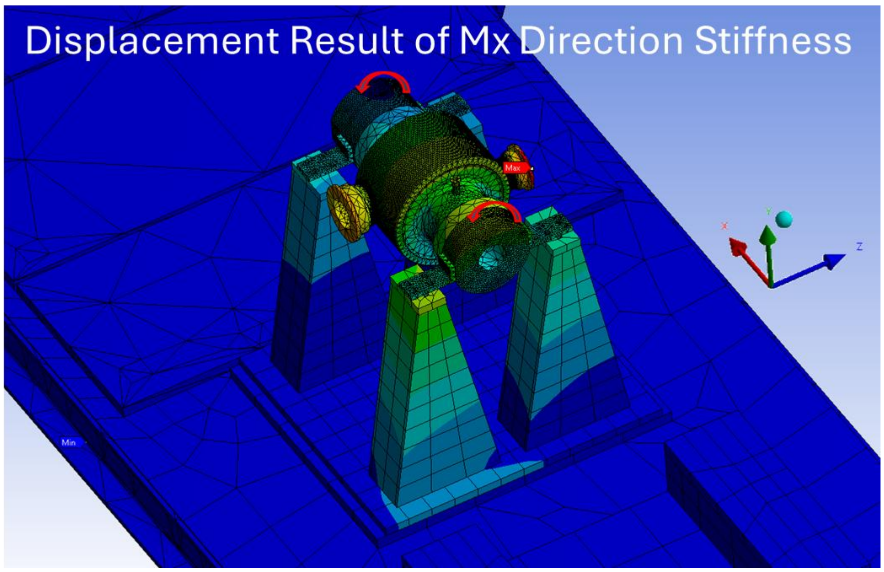

Figure 16 ANSYS MODEL OF TURBINE AND FRAME DISPLAYING DISPLACEMENT CONTOUR FOR STIFFNESS CALCULATIONS

Table 3 CALCULATED STIFFNESSES FOR THE TURBINE FRAME FROM ANSYS MODEL

Direction --	**Stiffness** lbf/in
X	1.60E+06
Y	5.00E+06
Z	1.80E+07
RX	5.00E+06

SNUBBER DESIGN METHODOLOGY

A second turbine trip occurred on May 8, 2024. The bump stops were implemented after the February 2024 trip, which reduced vibration levels; however, excessive levels were still experienced and measured. Therefore, the team looked into incorporating a snubber, or damper, to the turbine exhaust piping. The purpose of the snubber is to modify the natural frequency of the pipe section, thus shifting away from an excitation resonance domain. This section experienced low vibrations during testing and the emergency shutdown sequence. This section was particularly challenging; the changes in mass flow made by the actuation of the ASV and TSV would have ripple effects on the piping lines and excite acoustic and mechanical modes. Turbine operation would also play a factor with the turbine exciting the flow on the line. Because this is a high-temperature section and because of the low allowable on the equipment, the line could not be overly constrained and consists of several twists and turns.

A similar design process was executed in determining the location of the snubber and the overall impact to the piping system. Table 4 lists the maximum location resultant nozzle loads to allowable loads ratio, as a percentage. To reduce the vibrations on the line while still ensuring it had the required degrees of freedom for expansion, the snubber shown in Figure 17 was added.

Table 4 SNUBBER DESIGN RESULTS FROM CAESAR DYNAMIC MODEL

	Highest Load-to-Allowable %					
Case	**Heater Out**	**Turbine In**	**TSV Out**	**TSV In**	**Greylock**	**Comments**
1	148%	80%	100%	132%	112%	Configuration approved for installation
2	147%	192%	102%	139%	95%	
3	149%	190%	85%	127%	90%	
4	197%	64%	79%	96%	101%	
5	144%	51%	102%	117%	110%	

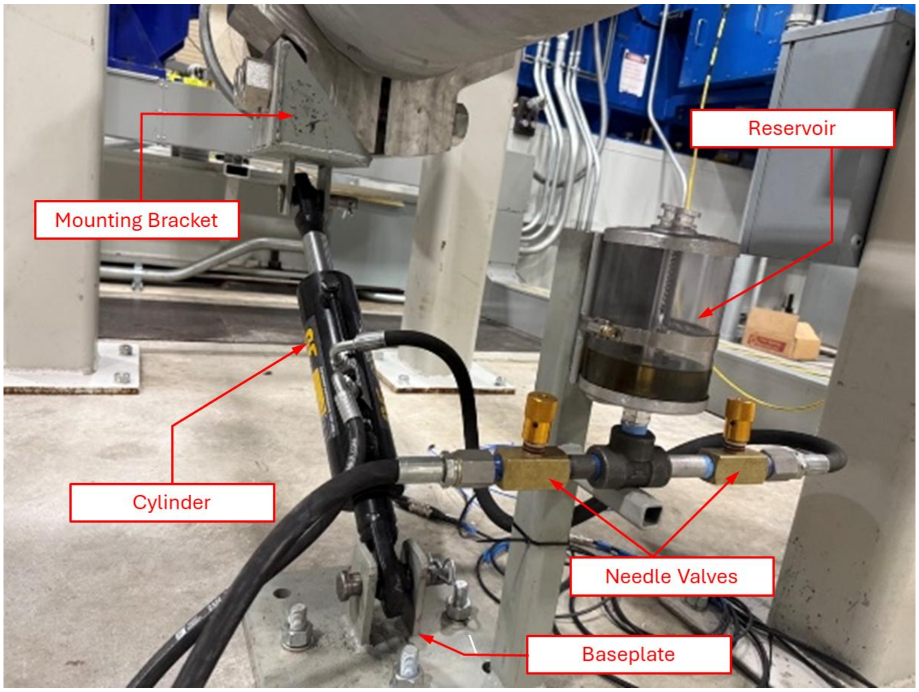

Figure 17 SNUBBER ADDED TO DAMPEN VIBRATIONS [9]

The designed snubber uses a hydraulic cylinder attached to a fluid reservoir; it uses two needle valves to modulate the amount of fluid that will go into the cylinder; there is also a check valve to prevent backflow. The snubber was tuned to shift the natural frequency of the piping system by modifying the piston. A damping coefficient increase of up to three percentage points was achievable with this design. To tune and determine the damping, two velocimeters were added, and the pipe was excited by pushing on it until it started oscillating as described in [9]. The damping ratio was calculated based on the decay over time using the logarithmic decrement method. After the snubber was installed, a similar trip from full speed occurred in September. A comparison of the vibration peak amplitudes as measured in May and September shows a notable improvement of up to 3.6x in the total acceleration at the location of the snubber.

DESIGN MODIFICATION IMPACTS

In total, 3 fluid hammer events occurred during the simple cycle test campaign: February, May and September 2024. The bump stops were implemented after the February event, while the snubber was installed after the May event. The February fluid hammer was the first time data was obtained and analyzed to assess the severity of the fluid hammer effect. The operational speed of the turbine was low at the time of the event, but the severity of the vibrations the piping system experienced were high. The second fluid hammer event occurred in May 2024, where the turbine was at full speed, but the overall severity of the event was reduced. Lastly, in September 2024, the last fluid hammer occurred in a controlled rapid emergency stop with the turbine at full speed and partially loaded. Figure 18 shows the impact of the bump stops and snubber on the severity of the vibrations experienced by the piping system. Remember, the February emergency trip occurred at a low turbine speed, while the May and September events occurred at full turbine speeds.

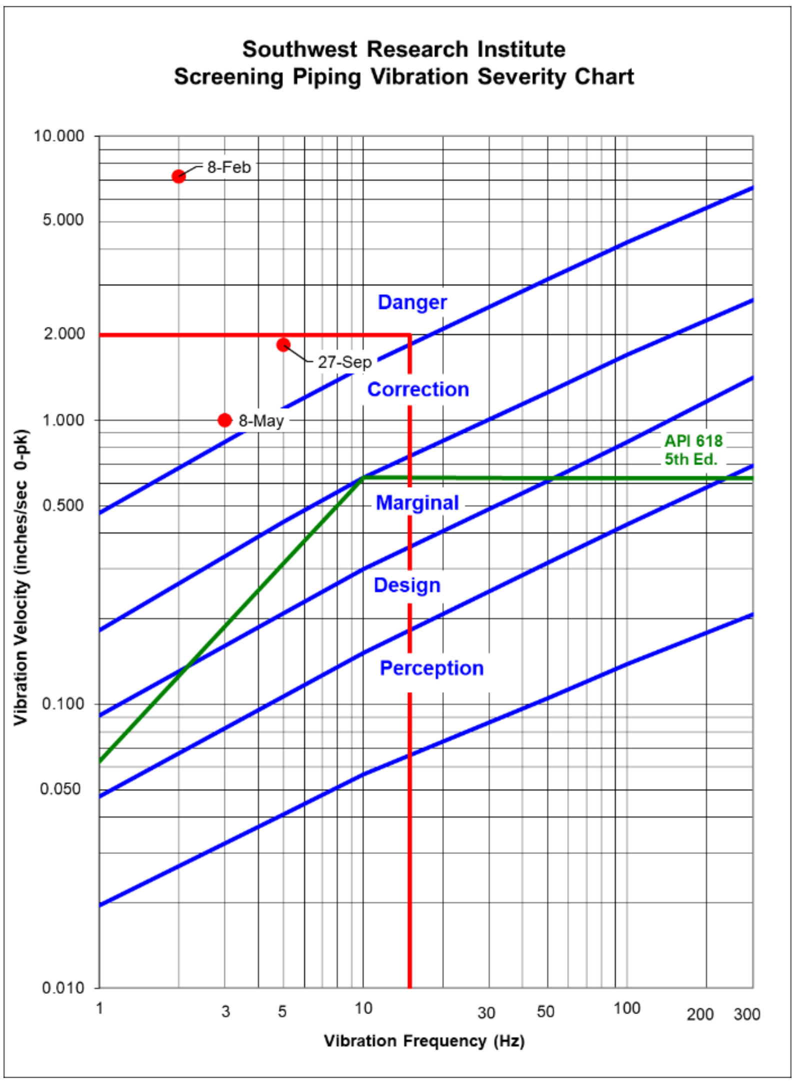

Figure 18 SEVERITY IMPACT OF DESIGN MODIFICATION IMPLEMENTED BY THE STEP TEAM [9]

CONCLUSION

A lot of experience and knowledge has been gained through the simple cycle testing campaign. The wide changes in both density and speed of sound for sCO2 added new levels of complexity. The initial design of the piping system focused on thermal growth, equipment nozzle loads, system acoustics and Strouhal frequencies which helped defined potentially high vibration locations within the system. These design analyses set the foundation to assess real operational data.

The operational data obtained during the first fluid hammer event on February 8, 2024, proved to be invaluable. This event provided one validation for the results from the Flownex SE dynamic model, which is used to predict cycle performance at various conditions and transients. Additionally, the Flownex dynamic model was used to assess the pressure pulse for the worst-case fluid hammer event.

Using the data collected during the fluid hammer event led to developing a new dynamic piping analysis utilizing CAESAR, which informed appropriate modifications to be implemented that helped mitigate the vibration amplitudes and the risk for failure: bump stops and snubbers. To supplement the CAESAR models, ANSYS Mechanical models were utilized to evaluate appropriate stiffness parameters as boundary conditions to the CAESAR models. These design methodologies were validated when additional fluid hammer events occurred and the severity levels were reduced, even though the turbine was operating at higher speeds than the February 2024 event.

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DISCLAIMER

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