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FAILURE ANALYSIS OF HIGH PRESSURE TEST FACILITY CONTROL VALVES

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ABSTRACT

In support of a series of upcoming Hydrocarbon Boost (HCB) Tests, facility operation demonstrations were run to validate the Air Force Research Laboratory (AFRL) Test Stand 2A (TS2A) Liquid Oxygen (LOX) and Gaseous Nitrogen (GN2) propellant systems were fully functional and able to meet HCB test requirements. While facility activation, blowdown, and system checkout tests were in progress, several anomalies were detected. First, it became apparent after a few tests, substantial audible leaks had developed in two control valves critical to tank pressurization. Second, anomalies in the data from the control system suggested that valve performance had changed in a manner suggesting leakage. These two issues caused TS2A to suspend operations until an investigation could be performed to determine the cause of the irregularities.

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This paper will describe the nature of the failure of the valves, and the impact on their capability to operate effectively. It will provide a failure analysis and lessons learned so that future procurements of high-pressure, high-performance control valves can avoid the mistakes made in the design and procurement of these kinds of valves.

INTRODUCTION

The Experimental Demonstration Branch at the AFRL Aerospace Systems Directorate at Edwards AFB, CA began system activation activities of the TS2A LOX system in early 2012 to support component testing for the HCB program. The LOX system is simple and comprised of four main components: GN2 pressurization, pressurization control valves (referred to as XCVs), LOX run tank, and LOX distribution piping. The GN2 pressurization system and XCVs are rated to 10,000 psig, and the LOX run tank to 8,500 psig. After activation of facility mechanical, control, and instrumentation systems, we began performing a series of blowdown tests using Liquid Nitrogen (LN2) for system shakedown and facility performance verification. The initial test was performed using 500 psig LOX run tank pressurization, with subsequent test pressures being increased incrementally up to the required test pressure of 6,000 psig. **Figure 1** shows the pressurization and vent systems on the LOX run tank. The GN2 storage cascade is not shown, but is on the left side of the figure, schematically, with the flow path indicated by the arrows.
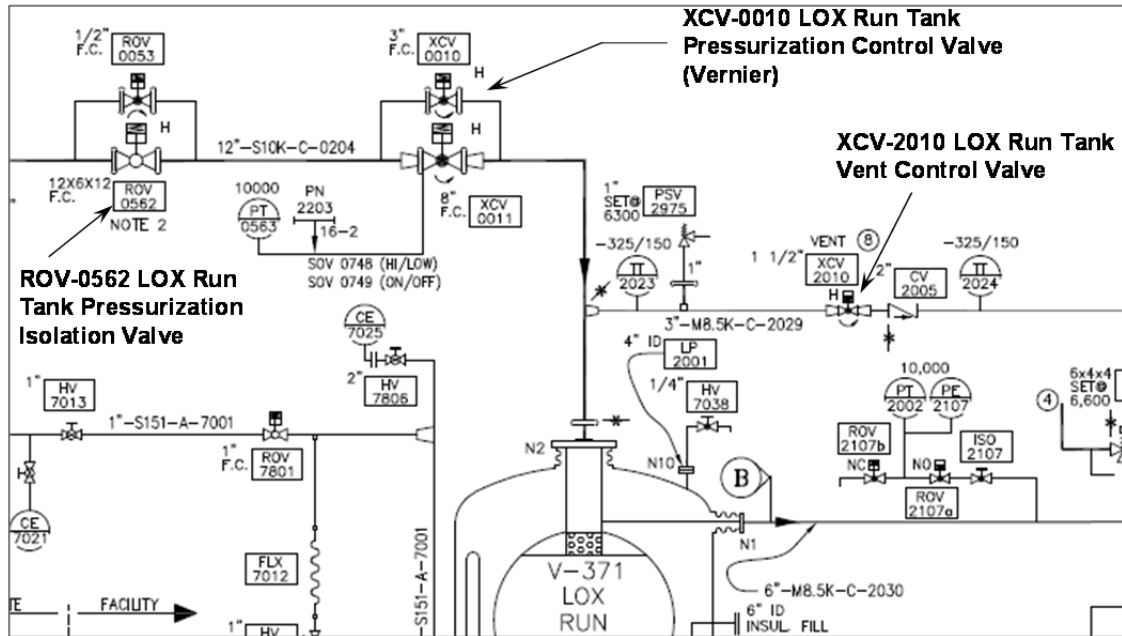
Figure 1. This schematic Process and Instrumentation Diagram (P&ID) shows how the LOX run tank is pressurized from the GN2 cascade, as well as the system used to vent the tank.

During initial blowdown tests, we started to notice anomalies in the tank vent rate data that made us suspect the vent control valve (XCV-2010) may have been leaking. The leak rate quickly became untenable once the pressurization system reached 5,000 psig. Not only was the leak now obvious in the data, but was clearly audible out on the test stand. Testing was temporarily suspended in order to fix the problem. Leaking control valves are not unusual, especially large metal-seated globe valves operating at very high pressure. We have a great deal of experience and expertise working with these valves, and therefore proceeded with our standard "fix" for the leak; minor mechanical adjustments, and control system calibration to ensure that the plug was properly seated in the body of the valve. (Note the mechanical engineers automatically default to the problem being a control system problem, and vice-a-versa). These fixes are generally successful; however, in this case, they proved to be futile. The leak continued until it abruptly increased during a blowdown test whereupon we (the mechanical engineers) admitted defeat and acknowledged that the cause could only be attributed to a significant mechanical failure. We performed a borescopic examination and discovered significant damage to the valve plug and seat. The valve was removed, examined, and determined to be severely damaged and irreparable. A substitute valve was installed and activation tests resumed.

Shortly after the activation tests restarted, we noticed that the tank pressure was rising quickly after the tank vent valve (replacement XCV-2010) was closed. This was noted while performing routine leak checks on the main tank Pressure Safety Valve (PSV). This pressure increase was significantly higher than anything we had seen in any previous operation. The only credible source of the gas that could pressurize the tank was via XCV-0010. (Note that XCV-0011, the main tank pressurization valve, was not used or required to support this test series. It was mechanically locked closed and previously leak checked). Based on the failure of XCV-2010, we (the mechanical engineers) bypassed blaming the controls and instrumentation folks, and proceeded under the assumption the valve had mechanically failed. This was not an unreasonable assumption for three reasons: 1) the failure was much more abrupt than the vent control valve failure, 2) the leak rate was very large, and 3) the valve was from the same manufacturer and of the same design as the vent valve. Borescopic examination confirmed significant damage to the valve plug and seat. We removed, examined, and determined the valve to be severely damaged and non-repairable. A substitute valve from a different manufacturer

was installed and activation tests resumed. The replacement valve has a proven design, had been previously used in high-pressure systems, and had an identical Cv to XCV-0010. Some mechanical modifications were required to accommodate the replacement valve.

FAILURE ANALYSIS

An analysis was performed to determine the failure mechanism of the two valves. Since the identical failure was observed in two previous valves of the same design, we felt it was necessary to determine the failure mode. Three other valves of the same design remained installed in the system. It was imperative that we understood how XCV-0010 and 2010 failed in order to evaluate the risk of leaving the remaining three valves in the system.

In **Figure 2**, we show the basic configuration of XCV-0010. The valve is a hydraulically actuated Y-pattern globe valve. The body of the valve is constructed from 304 stainless steel and the plug and stem are made from A286 Gr 660 Type 2 stainless steel. The plug is a three-piece design with a copper seat.
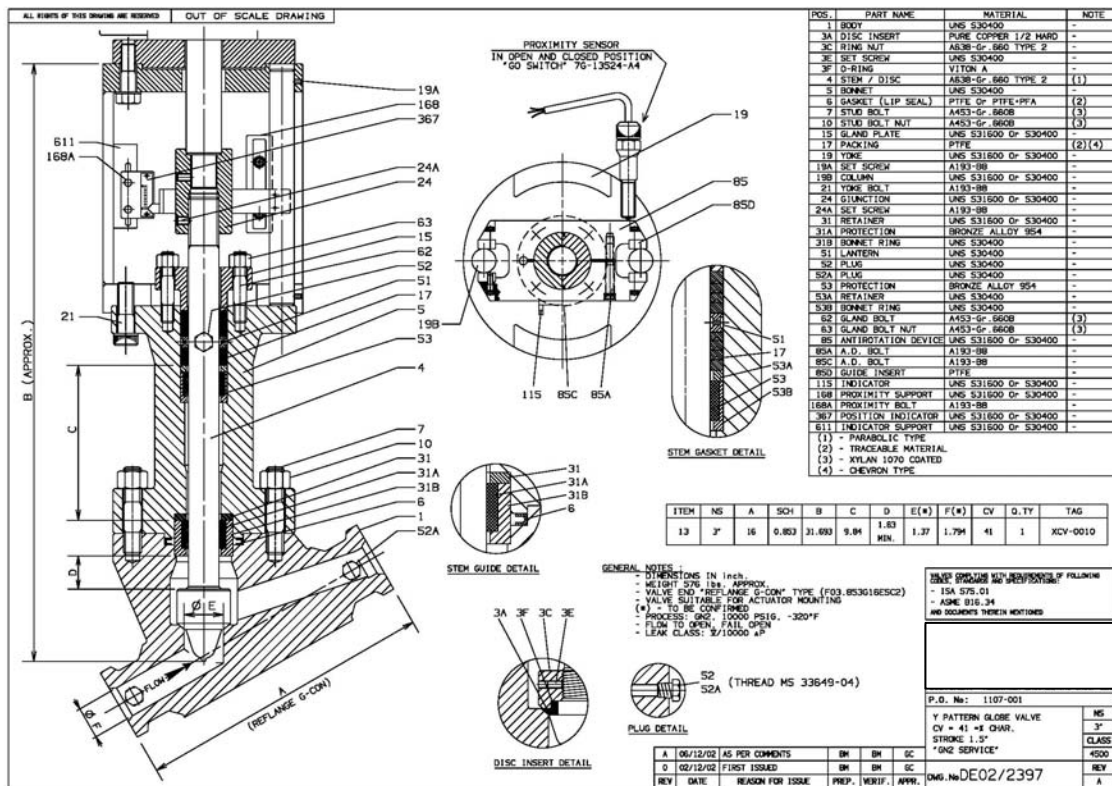
Figure 2. A cross sectional view of XCV-0010 shows the basic configuration of the valve. This configuration, consisting of a Y-pattern body, plug and stem, guides, and packing, is a common configuration for high-pressure globe valves, which has been used for over 60 years.

We undertook a simple three-phase approach to determine the cause of the valve failure(s): 1) disassemble the valve and perform a postmortem examination to evaluate the extent of the damage; 2) using the postmortem data, determine the most likely mode of failure, and develop a hypothesis for the failure; 3) build a model of the valve to see if the hypothesis was supported by the analysis.

POSTMORTEM EXAMINATION

Figures 3, 4, and 5 show the extent of the damage to the valve.

Removing the plug from the body required "tapping" it out. (Precision valve disassembly)

The guide bearing was seized in its race and the race seized in the valve body (hence the sledge hammer)

Figure 3. Removing the plug from the valve body was very difficult. We would not usually use a sledge hammer to get the plug/stem assembly out of the valve.
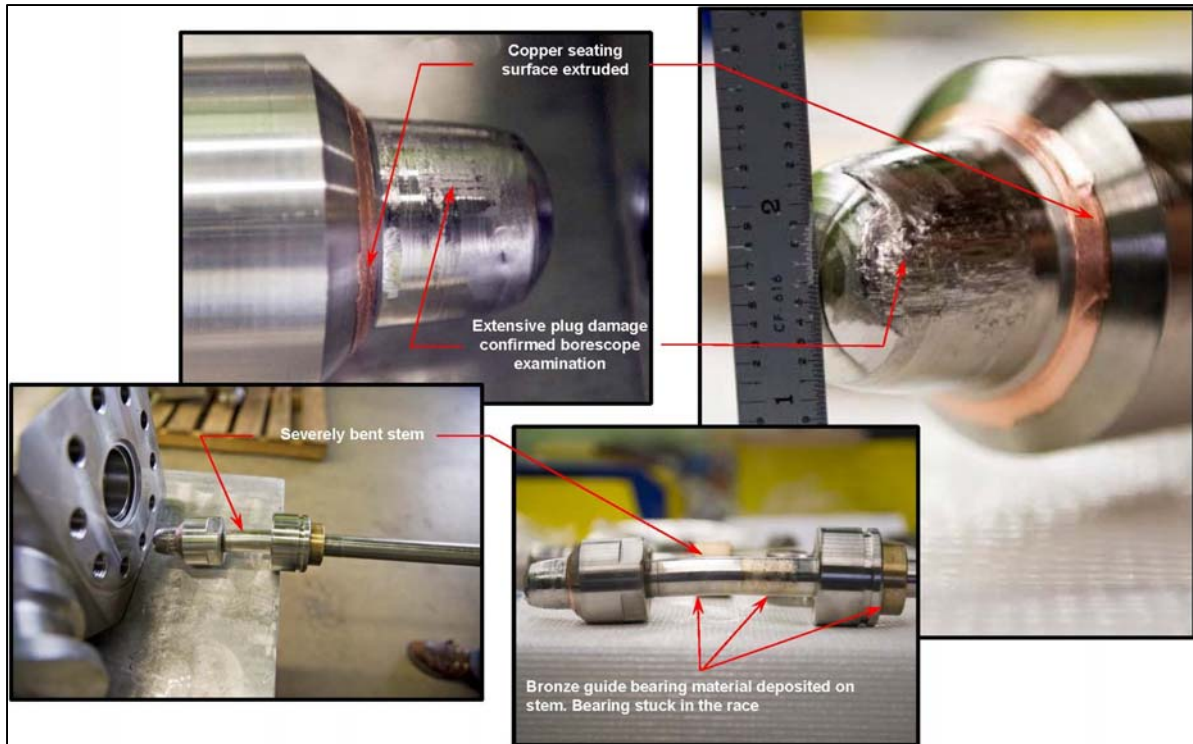
Figure 4. Extensive damage to the plug and stem was found. The stainless steel on the plug had been scraped off in the area where the plug crashed into the valve body seat. The copper seat on the plug was badly damaged and the stem was severely bent. We also found that the bronze guide bearing had rubbed so hard on the stem once it bent, that the bronze was deposited onto the stem itself.

Figure 5. What you are looking at is the seating area of the valve body. Note that stainless steel in the seating area has been scrapped so hard that the plug contact area has rolled over where the plug contacted. Also, you can see where we found copper from the valve plug deposited on the valve seat.

FAILURE HYPOTHESIS

Upon examination of the disassembled hardware, the failure was identified as a bent stem. At this point, we were faced with determining what could have caused the stem to bend. There are only two modes by which the stem could have bent in the manner it did: buckling and/or a cantilever load placing a bending moment on the stem. Now, intuitively based on the evidence, we thought that the stem bending could only have occurred due to a bending load; however, we have clearly demonstrated on other valves (much to our embarrassment) that it is entirely possible to buckle the stem by over-adjusting the stem projection into the valve seat and then applying an excessive load from the actuator. In the past, we have only seen this happen on very small valves (less than ¾ inch), where the l/d ratio is high. Unfortunately, there is no clear line of demarcation in regards to l/d ratio that distinguishes when buckling would occur that can assist us in making this determination. However, based on our experience, valves sized ¾ inch and up are not as prone to stem buckling. Valves the size of XCV-0010 and 2010 are highly unlikely to fail in this manner. Additionally, the bending that occurred was inconsistent with the manner in which a column with both ends fixed would bend, lending further evidence to our claim. Thus, we decided to start our analysis by building a model in SolidWorks and performing a finite element stress analysis. A cantilever load was placed on the stem to simulate the upstream pressure experienced during valve actuation.

MODELING AND ANALYSIS

Before we get into the specifics of the modeling and analysis, it's necessary to describe how the valve was being operated when it failed. Refer to **Figure 1** as necessary. The sequence of events is as follows:

1. Isolation valves (separating the GN2 supply source from the control valves) are opened in order to deliver pressure to the tank pressure control valve (XCV-0010). At this point, up to 9,000 psig GN2 is on the upstream side of XCV-0010. The downstream side of the tank is at ambient pressure, so the pressure differential on the valve is 9,000 psi. The position of the valve is such that the plug and stem are fixed at each end with the upper support being the guide bushing and actuator and the lower support being the valve seat.

2. The vent control valve (XCV-2010) closes in preparation for the LOX run tank pressurization. The tank isolation valve (on the outlet pipe) is already closed, as well as all other valves connecting the tank to all other systems. The tank is sealed off and ready to be pressurized.

3. The control system commands XCV-0010 to pressurize the tank to 6,000 psig and hold.

4. On command, XCV-0010 goes rapidly to the full open position, and then begins to throttle to achieve set pressure in the tank. Once target pressure is achieved, the valve closes, and waits for the test to start.

5. When the test starts, the tank isolation valve begins to open again, thus XCV-0010 experiences a pressure drop and begins to throttle open to maintain the tank pressure at the set point.

6. At the end of the test, XCV-0010 closes and XCV-2010 (the tank vent valve discussed earlier) begins its controlled venting of the LOX run tank.

7. Once the tank is vented, all the valves are secured to their normal positions, and the system is returned to a non-operating secure mode.

In step 4, the control system commands XCV-0010 to go full open. At that point, the plug comes out of its seat under the 9,000 psi delta pressure and the valve stem assumes the characteristics of a cantilever beam (now supported at only one end by the guide bushing). This is the point when the valve stem will see the maximum load at the far end of the plug, resulting in the maximum bending moment. We decided to start our analysis using these conditions as we felt that, based on our hypothesis, this would be the worst-case load on the stem.

We started with a "first order" conservative analysis. By "first order" conservative analysis, we mean that we modeled the valve to be "better" than it really was. The following constraints/assumptions used in the analysis will explain:

1. We did not have complete engineering drawings for the valves that failed. Thus, we modeled the stem and plug geometry from the failed hardware. We had to build our own solid models of the hardware as best we could from the existing parts, and by scaling from the assembly drawing.

2. We used the 9,000 psi differential pressure as the load force since we had test data to support that.

3. We assumed that the guide bearing was in intimate contact with the valve stem (which it was not; it was a rather sloppy fit), and the guide bearing was somewhat closer to the plug than it really was. It should be noted that the first cut of the FEA had to be modified to move the support bearing closer to the plug because the large displacement would not allow the model to converge. This was a conservative assumption made because we didn't have the fabrication drawings.

4. We assumed the bending force was acting on the plug only. This reduced the area of the application of the pressure, resulting in a smaller total force on the stem.

5. We used the high end of the material properties for A286 Gr 660 Type 2 stainless steel. (171 ksi Su, 112 ksi Sy)

We constructed the models based on these constraints and assumptions in SolidWorks, and performed a series of finite element analysis studies. FEA studies were performed for both bending and buckling. The valve failed in bending by a large margin consistent with the postmortem examination of the valve. The analysis indicated that the factor of safety (FOS) in buckling was 8 (which means, the load required to make the stem fail is eight times larger than the axial load applied to the stem by the actuator in the closed valve position), so we decided that no further buckling analysis was warranted. The results are presented in and discussed with **Figures, 6, 7, and 8**.
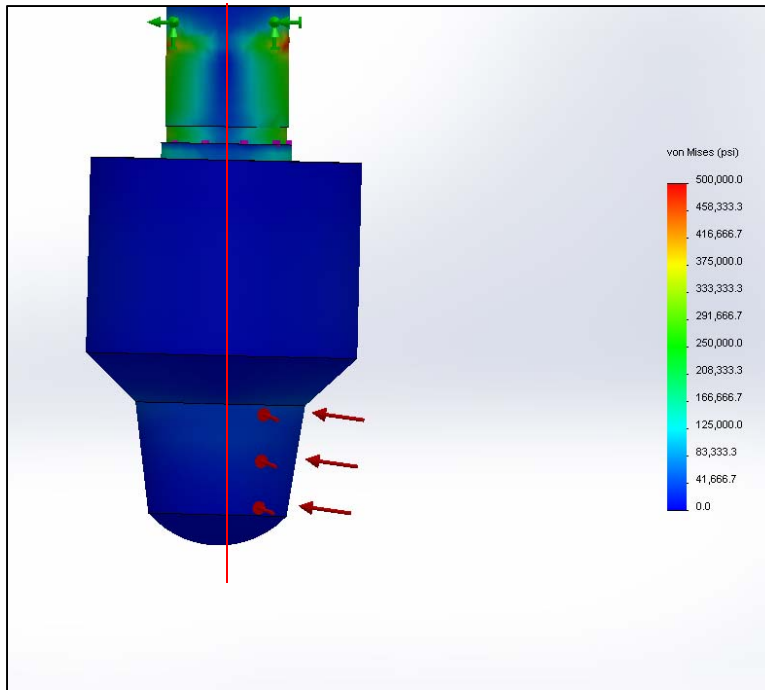
Figure 6. This graphic shows the output of the finite element solver displaying the maximum von Mises stress for the XCV-0010 stem/plug. Notice, as we stated in our assumptions, that the force of the pressure was placed orthogonal to the face of the plug, but to be conservative, only applied to the projection of the plug tip and not the whole wetted area of the plug (the large diameter portion). The results show that the maximum stress on the stem (487 ksi) occurs at the location of the cantilever support (guide bearing), and greatly exceeds the yield strength of the stem, resulting in permanent deformation. If you look carefully at the centerline, you will also notice slight bending of the stem. This is less than the real stem, but shows that even with this very conservative analysis, the stem would bend. If the maximum stress had not exceeded the material allowable, you would only see displacement on a greatly exaggerated (50x or greater) graphic (see next figure).

Displacement (Exaggerated) 10X
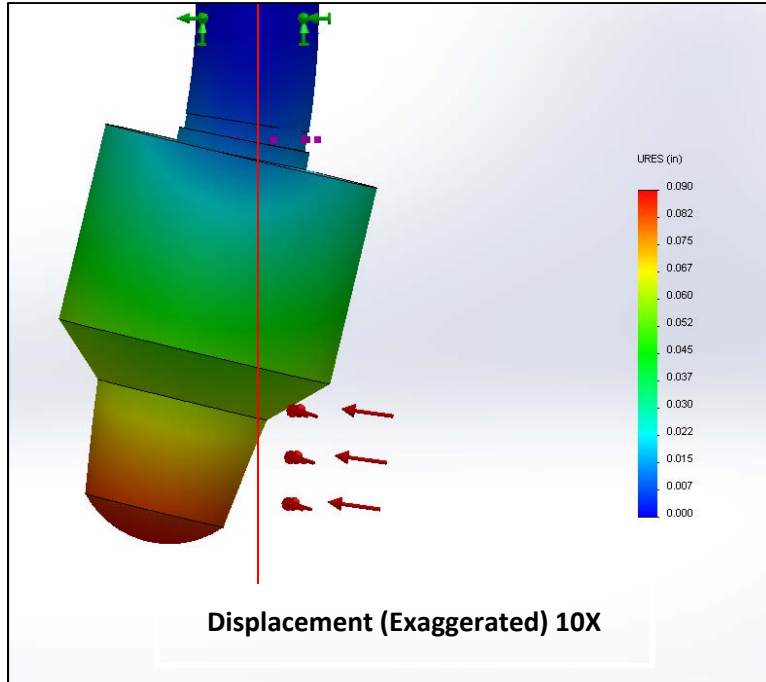
Figure 7. Seen here is the displacement exaggerated by 10 times to show the bending of the stem beginning at the location of the lower guide bearing. The maximum displacement according to the conservative model is approximately 0.09 inches. It doesn't seem like much, but permanent deformation of the stem on this magnitude indicates both an extreme overload, and in this case, certain interference between the plug and the valve seat.
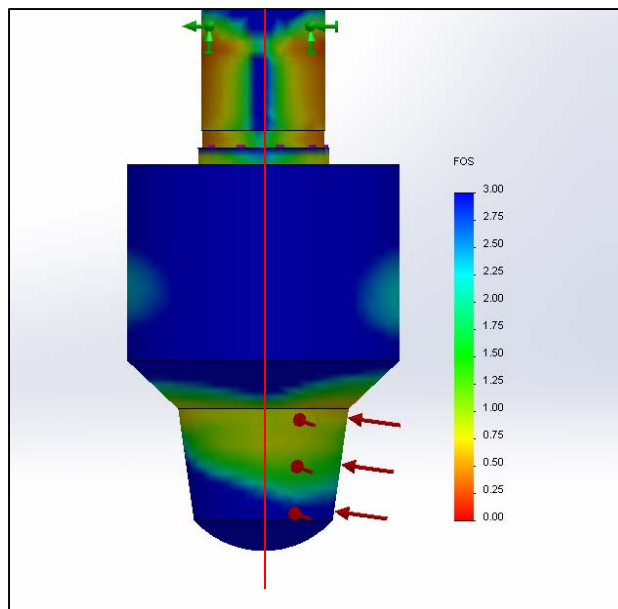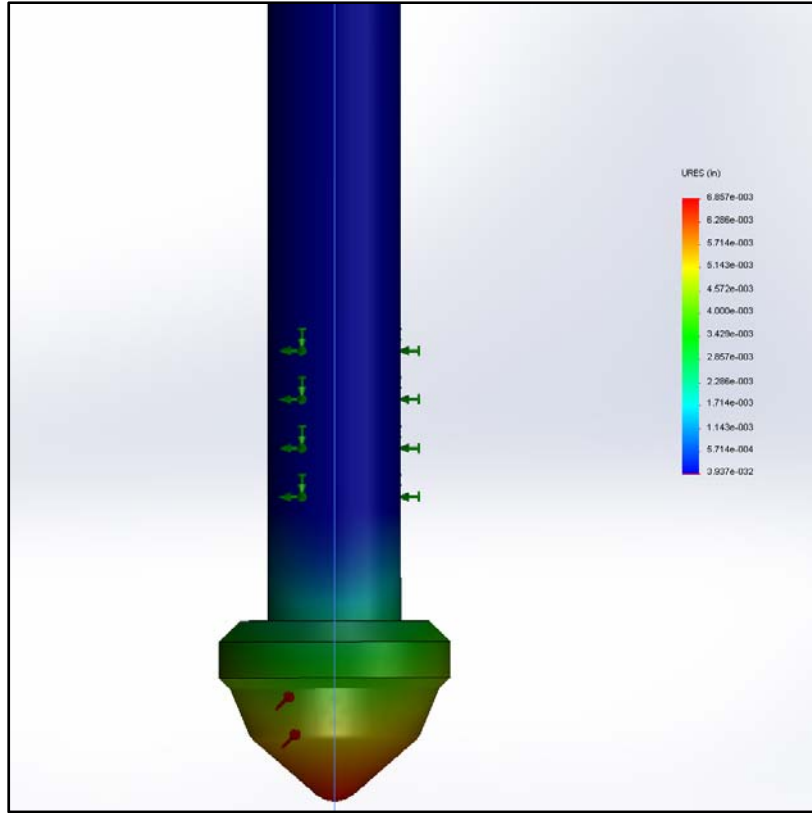

Figure 8. Here you can see the results of the finite element solver displaying the FOS for the XCV-0010 stem/plug. The results show that the FOS was less than one (1) in many areas of the structure with the minimum being 0.21. This means that permanent deformation occurred. No displacement is shown in this particular graphic.

For the sake of completeness an FEA was performed on the replacement valve (XCV-3003) that was taken from the test stand hydrogen system. The same analysis assumptions were made for this valve with two twists: First, we projected the pressure force on the full projection of the plug (as seen by the fluid flow), and second we used the low end of the material properties for the Nitronic 50 stainless steel stem in this valve (135 ksi Su, and 105 ksi Sy). We did this to see if the valve would pass using a similar analysis technique, but with much less conservatism. The results are presented in and discussed with **Figures, 9, 10, and 11**.

**Displacement (Exaggerated) 100X**
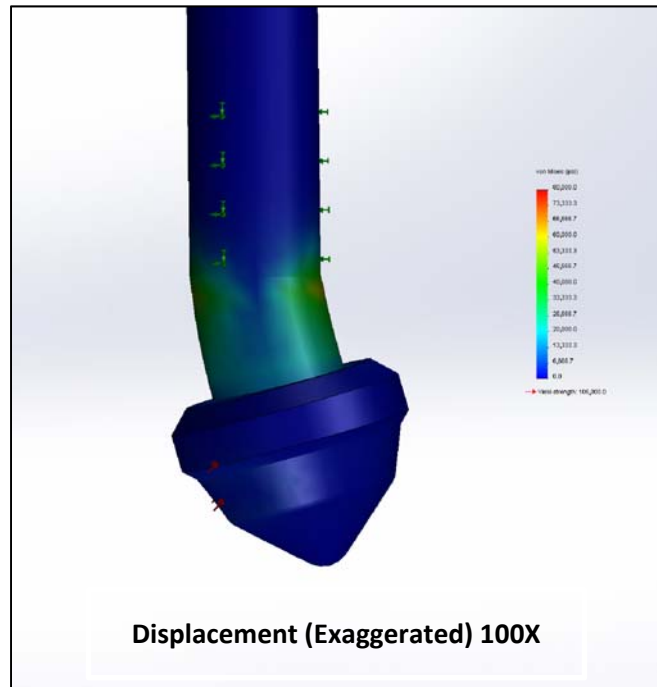
Figure 9. This graphic shows the output of the finite element solver displaying the maximum von Mises stress for the XCV-3003 stem/plug. Notice, as we stated in our assumptions, that the force of the pressure was placed orthogonal to the face of the plug, however in this less conservative case, applied to the whole projection of the plug tip. The results show that the maximum stress on the stem occurs at the location of the cantilever support (guide bearing), but in the case of this valve, the stress does not exceed the yield strength of the material. Note that the deflection in this graphic is exaggerated by 100 times. Otherwise, as in **Figure 6,** it is undetectable.

Figure 10. Seen here is the displacement results showing the maximum displacement is at least an order of magnitude less than XCV-0010, and would return to zero once the load is removed since no permanent yielding took place.

All of the work (disassembly, postmortem inspection, analysis, etc.) was performed for the tank vent control valve (XCV-2010). The failure was identical to that of XCV-0011, and the analysis showed similar results, so this paper will not discuss this particular valve any further.
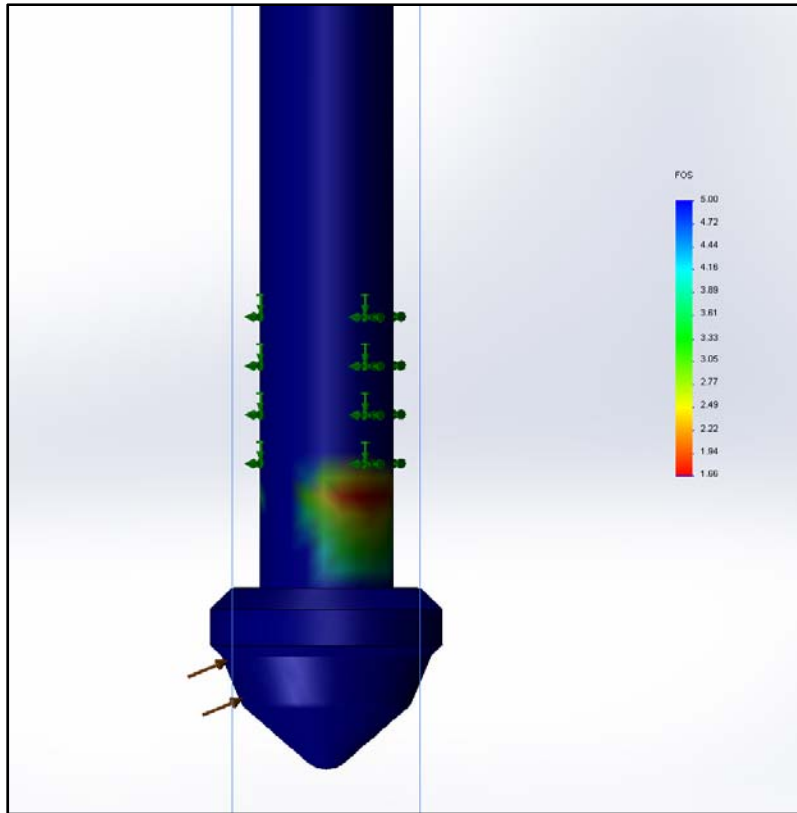
Figure 11. Here you can see the results of the finite element solver displaying the FOS for the XCV-3003 stem/plug. The results show that the FOS was greater than one (1) in all areas of the structure with the minimum being 1.60. This means that the FEA indicated no permanent yielding of the stem.

FAILURE CAUSE, PREVENTION, CORRECTIVE ACTIONS, AND RECOMMENDATIONS

The cause of the failure appears to be purely mechanical as there were no control system or operation errors that could be traced to the valve failures. Several obvious design flaws in the valve are considered to be major contributors to the failure of the valve. First, the large surface area projection of the plug into the flow stream subjects the stem and plug to an excessive load. Second, the distance from the stem guide bearing to the plug creates a long cantilever arm that promotes bending of the stem. This feature is not typical of other high pressure valves, which have guide bearings located much closer to the plug and have proven reliable in similar/identical environments. Third, the stem guide bearing tolerance was such that the stem was not well supported by the bearing, thus lending opportunity to produce greater displacements and higher stress concentrations on the valve stem. Finally, the ratio of the stem diameter to the plug diameter (d_{stem}/d_{plug}) is low. This feature creates a higher stress concentration at the stem-plug interface.

To prevent this from occurring again in the as-built system, we found the location of all the installed valves that have a similar design by the same manufacturer. There were two additional valves identified on the installed system, and one installed on the test skid for the HCB program. We took the following steps to determine the extent of the problem and mitigate the program impact:

1. We verified the service requirements for each valve in order to determine whether they were susceptible to a similar type of failure.

2. We removed and replaced "high risk" valves where possible and tagged those valves to identify that they are not approved for future use in similar high-pressure applications.

3. We located out-of-service valves of the same design and identified them for non-use pending an engineering evaluation for proposed future service.

The four high-pressure valves in the LOX facility that are of the same design, are the two previously discussed as well as two others that remain installed: the LOX run tank fill isolation valve, and the LOX run tank emergency vent and overfill valve. The LOX run tank fill isolation valve was determined to be very low risk of the same failure because this valve would never be opened against more than about 150 psi differential pressure. While seated, the valve should see no undue stress even when the tank is pressurized to 8,500 psig. No leaks or abnormal operations have been noted with this valve.

The LOX run tank emergency vent valve, on the other hand, is a whole different story. This valve is designed to be actuated, either manually or automatically, in the event of tank over-pressurization. This would potentially subject the valve to an 8,500 psi delta pressure. This valve is considered by us (TS2A) to have a high risk of failure in the event of an emergency tank vent. The valve would probably work one time, but may not be functional after a single use. There is also a possibility that the valve would not reseat after opening due to permanent material deformation, caused by the large pressure differential, thus potentially allowing contamination into the clean system from atmosphere – an extremely costly impact. The short term solution for dealing with this situation is, first, to do everything possible to prevent an emergency vent situation. Naturally, this is something we would always do, but because of the critical nature of this problem we are looking into adding backup controls and red-line cuts to the pressurization using redundant instrumentation. An intermediate solution (currently in design) is to install a temporary replacement at the conclusion of the current test campaign. This would allow for proceeding into the next phase of testing, but would limit run tank operations to about 6,000 psig. The long term solution we are pursuing is a brand new replacement valve; however, a 4" 8,500 psig rated Monel globe valve does not come cheap (several hundred thousand dollars) and would take a substantial amount of time to procure.

LESSONS LEARNED

There were several lessons we learned from the failure and subsequent analysis of these two control valves:

1. For initial testing of high-pressure systems, stop at incremental points on the way to MAWP and perform extensive system inspections. Doing so will afford operators, engineers and maintainers an opportunity to closely examine system integrity and performance and identify/correct any potential concerns prior to experiencing system failure. This practice could also potentially save a great deal of time, money and effort spent on your program.

2. If using a new or unproven supplier or design:

   - Insist on more analysis, testing, and design reviews to prove the design is adequate to meet the requirements.

   - Have the supplier available on site for inspection of the equipment installed in the system.

   - Talk to other supplier customers to get their input on the performance of similar equipment.

   - Spend more time performing critical evaluation of the design in order to identify potential design flaws prior to implementing new equipment.

3. Don't get target fixation (concentrate only on one failure mode):

- We thought it was a controls/tuning/calibration/adjustment issue because we have seen those many times.

- Didn't consider mechanical failure because we'd never seen something like this before and never imagined it could happen.

4. Perform more inspections of the valves between tests to monitor their condition until we have confidence they are operating within normal parameters.

5. Make <u>absolutely sure</u> the contractor understands the requirements:

- Perform a more thorough prequalification review.

- Review operating scenarios (not just the specifications) with the contractor.

- Ask them lots of questions to verify they understand your requirements.

- Don't assume the contractor understands your system operating procedures.

CONCLUSION

Poor valve design was the single largest contributor to the failure. There are systemic design problems with this particular valve configuration that were not detected during procurement. We had no experience with this valve brand and design, but the contract was awarded to the vendor due to their meeting the Scope of Work (SOW) qualifications, and because they were the low bidder. The issue was not detected until about 10 years after installation of the valves because we had never operated them at conditions anywhere near 9,000 psig. Previous experience with similarly designed valves from another supplier gave us no indication that we would have this sort of failure. Communication issues with the supplier on exactly how we operate the valves were also a contributing factor in the failures. The main supplier of our high pressure control valves (for the last 25 years) is intimately familiar with how we operate them in a high-pressure component test stand like TS2A. It was assumed that the supplier of these valves had an equal understanding of our operations. This was a bad assumption, which resulted in a great deal of time, money and effort being spent to correct what could have easily been prevented.