Study of Drill String Safety Valves

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1

Executive Summary

This LSU study was funded by the Minerals Management Services U. S. Department of the Interior, Washington, D.C., under Contract Number 14-35-001-30749. This report has not been reviewed by the Minerals Management Service and approved for publication. Approval does not signify that the contents necessarily reflect the views and policy of the Service, nor does mention of trade names or commercial products constitute endorsement or recommendation for use.

A study of blowout preventer pressure test results by the Minerals Management Service (MMS) identified the Drill String Safety Valve (DSSV) as one of the least reliable components of the well control system. MMS sponsored a project at LSU to study this problem and to make recommendations for improving the prevention of blowouts through drill pipe. This final report presents the results that were obtained as part of this MMS project. The study was completed in several phases by groups of senior engineering students working on a senior design project. It was conducted as an interdisciplinary project which involved students and supervising faculty members from the LSU departments of Petroleum and Mechanical Engineering. The first phase of the study included a review of the common failure modes of drill string safety valves and a review of the patent literature and existing product descriptions. Failure to close against high flow rates and under high pressure was the most serious failure mode identified. After considerable brainstorming, the group returned to a quarter-turn ball valve as the most feasible concept that would allow a DSSV to be stripped in the borehole and wireline tools to pass through an open valve. The final design concept selected was a trunnion mounted ball with spring loaded seats. A new DSSV valve design was completed and a prototype valve was constructed. The valve was tested under static conditions at various levels of internal pressure. The closing torque was measured with no pressure differential across the valve seats for pressures up to 10,000 psi. The opening torque was measured with various levels of differential pressure across the seat up to 4,000 psi. The operating torque of the new design was less affected by the internal pressure level in the valve than the commercially available valves tested.

The second phase of the study was conducted in a summer term and consisted of a force analysis on the bearing surfaces for an upward pressure differential across a closed valve. Modifications to the prototype valve were recommended based on the results of the force analysis and an examination of the valve parts after Phase I testing was completed.

In the third phase of the study, which was conducted by a different team, the valve design was modified based on the work done in Phase I and Phase II. The design goal was a working pressure of 5000 psi and an operating torque of less than 400 ft-lbf for all conditions. The prototype was modified and a flow loop for testing drillstring safety valves under flowing conditions at a desired flow rate and internal pressure was also constructed. Theoretical operating torque versus differential pressure curves were calculated for the prototype design prior to testing. The prototype design was tested along with several of the most commonly used commercially available drillstring safety valves. Although the measured operating torque was higher than the theoretical values calculated, the new design met the design goal.

It was noted during the Phase III testing that the operating torque requirements of a commercially available DSSV that was tested with the new prototype had increased significantly over previously measurements made during early tests. It was hypothesized that mud debris from testing and weathering of valve components during periods of outside storage had increased friction between the valve moving parts. Internal clearances could also have changed after conduction a number of valve operations under high pressure and high torque conditions. Phase IV of the study was the design and construction of a valve storage system capable of maintaining a drillstring safety valve in a like new condition on the rig floor in an offshore environment. A storage stand was designed that could store up to four DSSV units submerged in "environmentally friendly" oil and allow any of the four valves to be retrieved with an air hoist in less than 10 seconds. Each DSSV is stored in a cylindrical chamber and rests on a piston that can quickly elevate the DSSV by opening a valve that allows hydraulic pressure to be applied beneath the piston. The storage stand was demonstrated at an LSU/MMS Well Control Workshop.

As a result of the study, the following recommendations are made regarding the use of Drill String Safety Valves:

1. The DSSV intended for use as a stabbing valve to stop flow through the drillstring during tripping operations should not be used in the drillstring for other operations. The stabbing valve should be maintained in a "like-new" condition and used only during periodic pressure testing with fresh water.

2. Operators and/or drilling contractors should check threads, valve wrench, and lift sub on the stabbing valve and actuate the stabbing valve close and open each tour.

3. Operators and/or drilling contractors should use a drillstring float whenever practical to provide redundant protection against a high-rate flow through the drill-string during tripping operations.

4. When floats are not used, shear rams are recommended for redundant protection against blowouts through the drillstring during tripping operations.

5. Drill String Safety Valves should not be the only means for stopping flow from the drillstring at the surface when reverse circulating the well during completion operations. Flow should be routed through hydraulically operated valves and a choke manifold.

6. Drill string safety valves should not be the only means for stopping flow through the drill string when significant piping and flow restrictions are present above the valve.

Acknowledgement

The work for this interdisciplinary project was supervised by Dr. A. T. (Ted) Bourgoyne, Jr. and Dr. L. S. (Scott) Stephens. The work was done by the following students:

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Chapter

2

Introduction

In this chapter, the basic drill string safety valve terminology will be reviewed. In addition, the current problem areas identified in this study from a literature review and from consultation with industry representatives will be presented.

Safety valves are ball valves used to stop flow through the drill string that begin when the drill string is being withdrawn from the well. Shown in Figure 1 is a photograph of a traditional TIW safety valve. This photograph was taken during a visit to a valve manufacturing facility. The valve has been disassembled here to show the main working components.

Figure 1: Photograph of the traditional ***TIW*** **Drill String Safety Valve.**

The portion of the safety valve shown on the right side of Figure 1 would accept the upper valve seat and spring and screw down over the ball.[1] After assembly, the ball "floats" between the upper and lower seats and seals when pressure is applied against the ball. The spring assists in providing a low-pressure seal. The valve is closed by rotating the ball by means of an operating crank or valve stem that fits into a circular hole in the valve body.

The patent has expired on this simple design which is now available from several manufacturers in addition to Texas Iron Works (TIW) from which it took its name. The name TIW valve is often used as the generic name for a Drillstring Safety Valve (DSSV).

[1] The portion on the right side of the safety valve is shown upside down to expose the female threads that make-up on the male threads located just below the ball and lower seat.

Shown in Figure 2 are the traditional locations of safety valves which include upper and lower kelly cocks as well as a stabbing valve. Government regulations require that a safety valve and operating wrench be maintained on the rig floor at all times.

Displayed in Figure 3 is a photograph of a safety valve made-up on top of a section of drillpipe which has been cutaway so that the ball and seats may be observed. The safety valve shown is a one-piece valve design that eliminates the need for threads in the valve body area. This not only decreases the number of possible leak paths, but also eliminates the problem of ball locking due to excessive torque. The valve is operated by means of an operating-wrench that is inserted into the valve stem and turned one-quarter turn.

Shown in Figure 4 are the blowout preventer component pressure test results for the U. S. Outer Continental Shelf during 1993 and 1994. Note that the pressure test failure rate for drillstring safety valves and inside blowout preventers was about 25%. This was especially troublesome, since the level of redundant protection for blowouts through the inside of

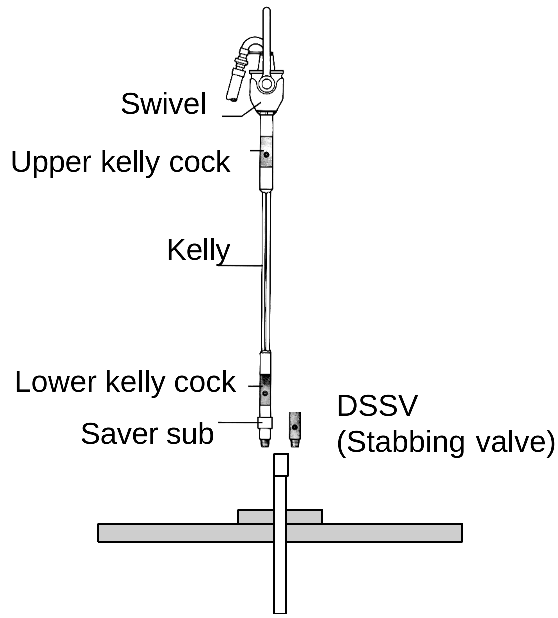

Figure 2: Schematic showing traditional locations of safety valves.

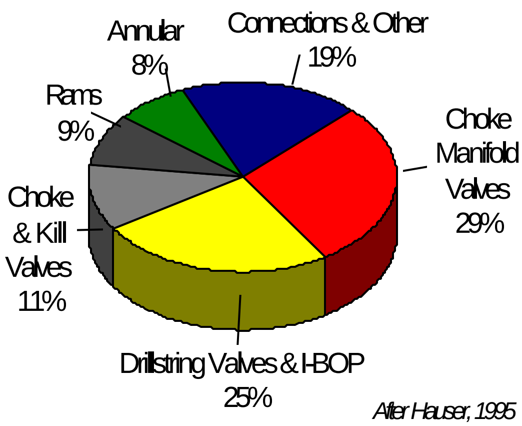

Figure 3: Results compiled from Blowout Preventer Component Pressure Tests for the U. S. Outer Continental Shelf during 1993 and 1994.

Figure 3: Photograph of a safety valve which has been cutaway and made-up on top of a section of drillpipe.

the drillstring can be much less than for flow through the annulus. Even though the choke manifold also had a high pressure test failure rate, a failure in this component is not as serious because these valves are not primary blowout barriers. Failure of one of these valves generally would not lead to a blowout.

Several Objectives were defined for this project as follows: (1) to verify that safety valve failures are a significant problem during actual well control operations, not just during routine testing, (2) to identify common modes of safety valve failure, (3) to identify alternative methods for preventing blowouts through drillpipe during tripping operations, (4) to design a test apparatus for evaluating alternative safety valve designs, and (5) to make preliminary recommendations for reducing the risk of blowouts through drillpipe. A secondary objective of the project was to provide meaningful educational experience for LSU Engineering students working on their senior design projects.

The approach used in this study had four major aspects:

1. The students first talked with industry experts to better define the safety valve problems that have occurred in the past. This aspect of the study dealt with problems occurring during well control operations rather than problems occurring during routine pressure tests. The students also visited valve manufacturers and talked with design engineers to get a better understanding of the design problems and the various features already available.

2. The students conducted a literature and patent review to determine alternatives to the ball valve design that had been studied in the past, and to identify auxiliary devices that could be used with a safety valve to improve reliability.

3. The students then developed an alternative valve design and a test apparatus for comparing the performance of the prototype valve design with several commercially available valves. As part of the evaluation of the test apparatus, experimental data were obtained on valve performance for several different valve designs.

4. Finally, the students developed a design of an improved storage stand for maintaining safety valves readily accessible and in a “like-new” condition in an offshore rig-floor environment.

The various aspects of the research task will be presented in separate chapters.

Chapter

3

Industry and Literature Survey

The results of consultation with industry experts provided much valuable information, some of which had been published in the literature

Mobil Oil Company conducted an industry survey in 1994 and identified 29 safety valve failures during well control operations over an unspecified period. The survey was conducted after Mobil experienced a number of problems in 1993 with stabbing valves leaking after being stripped into a well in a threatened blowout situation. The survey also identified several common failure modes for safety valves. The Mobil Oil study also agreed with experiences reported by other industry experts that were consulted.

Common Modes of Failure

The common modes of failure identified in the industry survey conducted by Mobil included:

1. Failure to close due to high torque (valve lock up);
2. Failure to seal due to flow erosion;
3. Failure to seal against pressure from below;
4. Failure to seal against pressure from above;
5. Failure to seal against pressure from outside; and
6. Failures to open due to high torque (valve lock up).

The most serious of these failure modes is failure of the valve to close due to high torque. Blowouts that occur due to this failure mode continue to occur. The author is aware of a blowout in 1996 and another in 2001 that resulted in loss of life after a DSSV could not be closed.[2] About 400 ft-lbf is generally regarded as an upper limit of torque that can be applied manually with an operating wrench. If the torque required to completely close the valve is exceeded before the valve is fully closed, the failures associated with partially closed valves occurs. High torque values are caused by the build up of pressure in the valve as the valve begins to restrict the flow. The pressure pushes the valve stem further into and against the valve body and the ball is forced against the upper seat. These two actions create can friction forces that can not be overcome, especially if the ball and stem dimensions are out of tolerance due to

[2] In both cases, other human errors also contributed to the loss of life.

wear. If the ball and stem are put under too much pressure, local stress deformations create metal to metal contacts with the associated high friction surface. The ball of a two-piece valve often locks if too much make-up torque is applied across the valve body. Tong placement is critical when tightening across this type of valve.

Amoco conducted a series of safety valve tests in 1990 at their research lab after a blowout in the Goldsby field due to multiple DSSV failures. A photograph of the Amoco Goldsby blowout is shown in Figure 4. Note the flow of gas and saltwater exiting the drillpipe. Amoco estimated the saltwater flow rate to be about 17 barrels per minute or 700 gallons per minute during this blowout. Three safety valves were available on this job. After the first valve failed, the second was successfully stabbed into the top of the first valve. The third valve was stabbed into the top of the second valve after it failed. It was reported that the valves failed during closure for two reasons. First, the large amount of torque required to close the valve under the existing flowing conditions prevented a quick closure. Second, the valve balls and seats eroded before the valves could be fully closed.

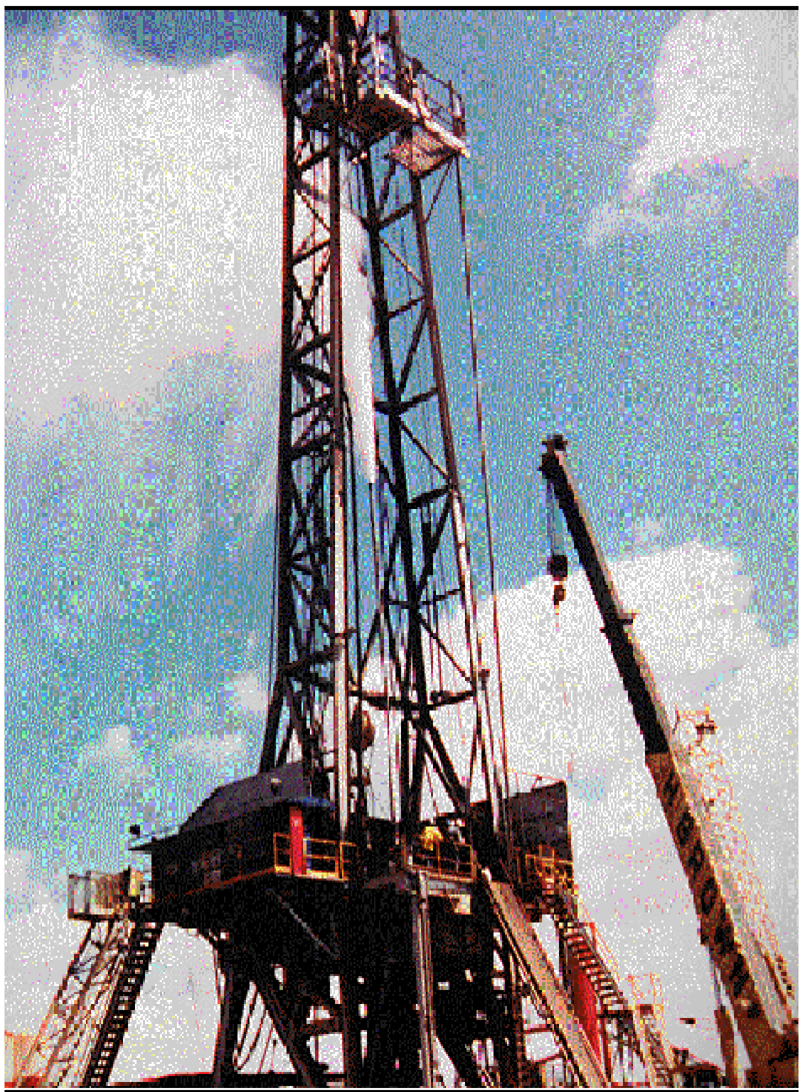

Figure 4: Photograph of Amoco Goldsby Blowout.

The Goldsby blowout let the high-pressure fluid move from below the valve, through the valve, and out the top. In addition, several other failure modes can occur. A failed safety valve installed on top of the drillpipe with additional piping connected above it can let high-pressure fluid move from below the valve, through the valve body, to the outside atmosphere. When pressure-testing equipment installed on top of the safety valve, a failure can occur allowing flow from above the valve, through the valve, and into the drillpipe below. This prevents a valid pressure test from being performed. After stripping a stabbing valve into the well, a failed safety valve can let pressure move from the annular space around the valve, through the valve body, and into the drillpipe. This can cause surface pressure readings to be misleading and could cause mistakes to be made during the well control operations. The common leak paths are illustrated in Figures 5-7.

One of the most common failure modes identified was the failure of the valve to open even after pressures are equalized across the ball. It is sometimes necessary to freeze a plug of ice-mud below the safety valve so that it can be replaced with pressure on the drillpipe. Ball lock-up is caused by pressure against the valve stem, by pressure forcing the ball into the seat, and by alignment problems that develop under stress.

Eroded balls and seats are also a common cause

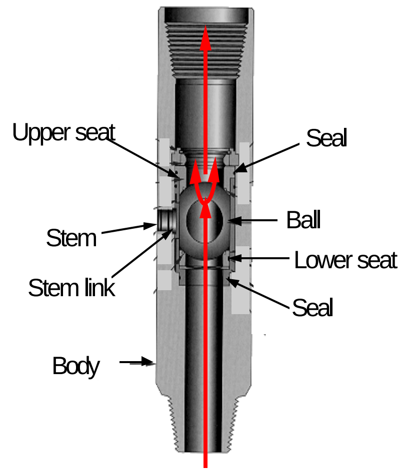

Figure 5: Fluid Escape past upper Seat

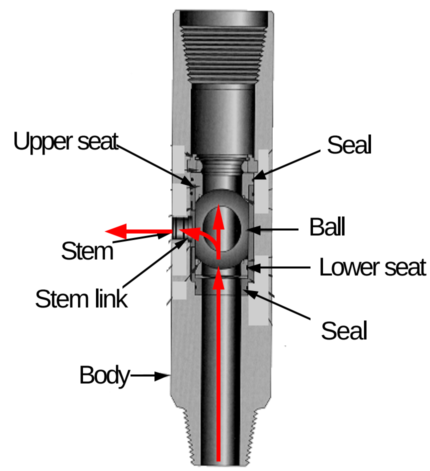

Figure 6: Fluid Escape through Valve Stem

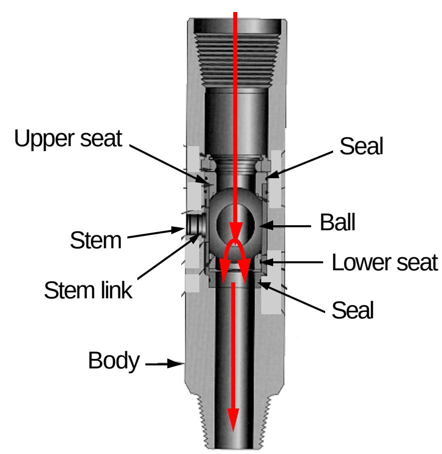

Figure 7: Fluid Escape through lower Seat

Caused by Human Error
(Partially Closed Valve)

Figure 8: Erosion of Ball and Seat by Drilling Fluid

of failure identified in this study. The erosion could be due by flow of mud solids through the valve as it is being closed. However, if the ball is not aligned perfectly in the open position, erosion in an upper or lower Kelly valve will also occur during normal drilling operations (Figure 8). In addition, erosion is caused by wireline work done through the valve. Failed elastomers can also cause leaks through the stem and around the valve seats. In addition, stress cracks in the valve body or threads also can occur.

A valve seat cut by fluid erosion due to a slightly over-closed valve is shown in Figure 9. Wear on the valve stem stop can sometimes allow too much rotation of the ball. Deformation of the valve stem stop can be caused by applying excessive torque when using a cheater bar. Design constraints resulting from maximum outside diameter, minimum inside diameter, and a high working pressure requirements make it difficult to design robust valve stops.

Wireline cuts to the inside diameter of the ball as shown in Figure 10 can reduce the amount of over-rotation that can be tolerated without leaking. Once fluid begins leaking past the valve seat, erosion can enlarge the leak path.

Figure 11 is an example of a failure in a DSSV stem. The valve stem accepts the valve wrench which then allows the valve to be operated. The valve stem is recessed into the valve body. Upon removal of the operating wrench, the DSSV can be stripped into the well without danger of the valve stem catching on a ledge or in a casing connection. The stem shown has wear in the metal that serves as the valve stem stop and is also cut by fluid erosion through the center of the stem.

Although not illustrated in a photograph, another common problem is valve lock-up due to hardened solids left in the valve after

Figure 9: Erosion of Seat by Drilling Fluid in over-closed valve.

Figure 10: DSSV Ball cut by wireline (slickline) work done through the valve.

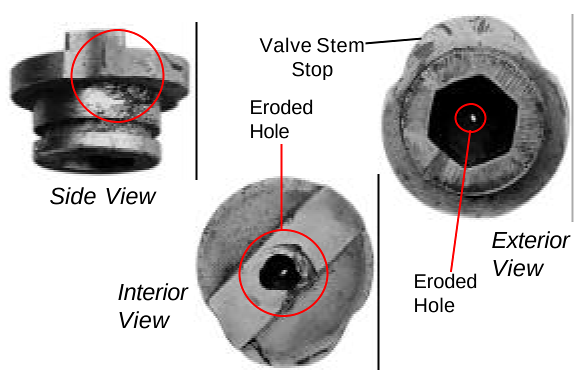

Figure 11: DSSV Stem showing wear on Valve Stem Stop and Erosion by Fluid

circulating mud and cement through the valve. Incomplete removal of cement from well control equipment can lead to a non-operational safety device.

Literature and Patent Search

There was good coverage of devices to prevent blowouts through the drillpipe in the patent literature. Twenty-three patents were reviewed and a number of alternatives to ball valves have been patented. The research team concluded that ball valves are best suited to the need for full opening valves with a small outside diameter that can be stripped into the well under pressure. Auxiliary equipment that compliments the use of safety valves and increases the number of barriers to a blowout through the drillstring was also found in the patent and literature review. Much of this auxiliary equipment had also been identified through discussions with industry experts. The auxiliary equipment identified for added blowout barriers included shear rams, floats or check valve placed in the drill collars near the bottom of the drillstring, a drop-in check valve, a wireline retrievable check valve, a velocity triggered check valve, and a double valve assembly. Shear rams can be used to cut through the drillpipe and close the well on top of the drillpipe if the safety valve fails. The disadvantage of shearing the drillpipe and dropping it to bottom is that it can make it more difficult to be able to eventually circulate kill mud to the bottom of the well.

Floats (check valves) installed in the drill collars are widely used by some operators to make it easier to stab and close safety valves at the surface. Both flapper and dart type check valves are available. Even if the check valve leaks, the flow rate is generally reduced enough so that the safety valve can be successfully closed without cutting out the valve. Some operators seldom use floats for the following reasons:

1. Extra time is needed to fill the inside of the pipe when lowering pipe into the well;

2. Higher surge pressures occur when pipe is lowered into the well;

3. Excessive loads of rock cuttings in the annular mud can be detected by back-flow through the drillstring when a connection is made; and

4. The shut-in drillpipe pressure is more difficult to read the after taking a kick.

The drop-in or wireline retrievable check valve (Figure 12) overcomes many of the objections to a float in the drill collars. A sub that will accept

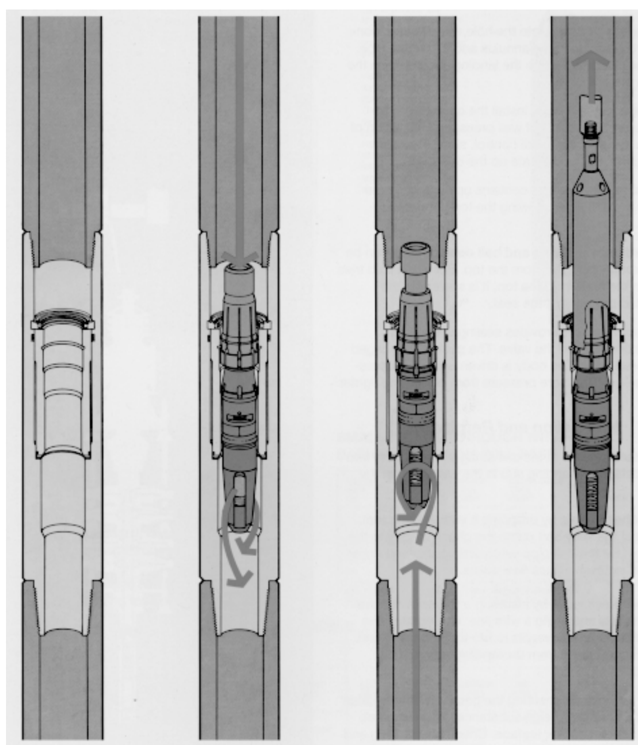

Figure 12: Schematic of a Drop-in / Wireline Retrievable Check Valve

a check valve is always run in the drill string near bottom. Just before it becomes necessary to pull the drillstring from the well, the check valve assembly is dropped into an open drillpipe connection and pumped to bottom where it latches into the sub. If the well tries to blowout during tripping operations, the check valve will stop the flow and make it easy to stab and close the surface DSSV as part of the shut-in procedure. In the event wireline work below the check valve becomes necessary, the drop-in check valve is wireline retrievable. However, the drop-in check valve would not provide a barrier to blowouts when making a connection during drilling if it is not deployed until tripping operations begin.

An example of a velocity triggered check valve is shown in Figure 13. This valve was designed and tested to a limited extent during the late 70's by Hughes Tool Company for Shell. It was lost in the shuffle of buy-outs during the 80's. Prototype valves are again being built by a new company.

An example of a double valve assembly is shown in Figure 14. For high flow rates, the lower ball may cut out, but the flow rate should be reduced enough to allow the upper valve to be successfully closed (if it is closed quickly before the bottom valve becomes too eroded). The bottom valve can also be used as a mud saver valve since a back-up valve is available. The disadvantage of this approach is that it is not well suited to stabbing valves, because of the extra weight that must be handled. Even a single stabbing valve for 4.5-in. or 5-in. drillpipe weighs more than 100lbs. To minimize the weight of a double valve, one manufacturer recently developed a double ball, single body design.

New Prototype Design

An interdisciplinary team was assembled which included seniors and supervising faculty in the LSU Petroleum Engineering and Mechanical Engineering Departments. The group was charged with the development of an improved DSSV design that would reduce the tendency for

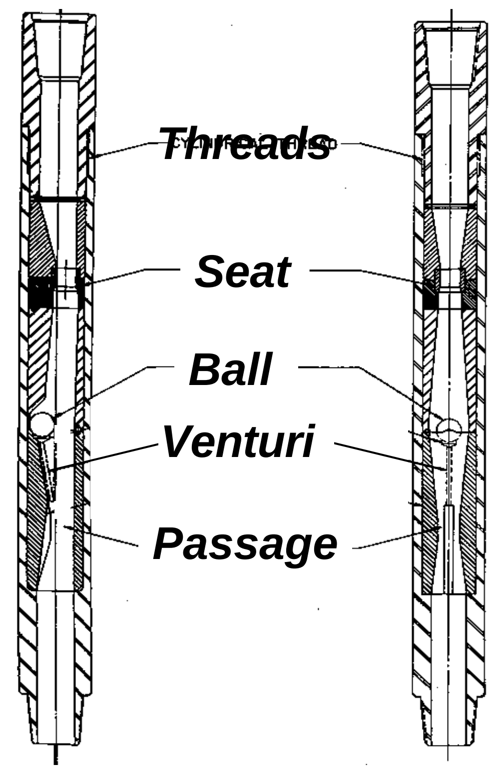

Figure 13: Schematic of a Velocity Triggered Trip Valve

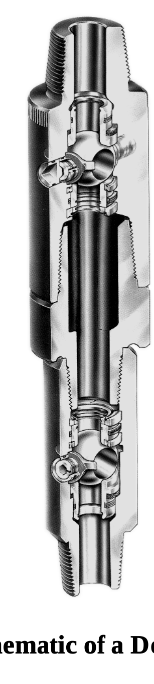

Figure 14: Schematic of a Double DSSV

valve lock-up when attempting to close or open against a high pressure differential or with a high internal pressure. After reviewing the patent literature and discussing the design constraints, the group concluded that the current approach of a quarter-turn ball valve was best. The main design constraints resulted from the need to be able to strip into a well under pressure and to be able to perform wireline work through the valve. It was decided that an effort would be made to design a trunnion-mounted ball with spring loaded seats in order to reduce the pressure effects on valve operation. It was also decided that prototypes of the new design would be constructed and tested together with commercially available valves to evaluate the effectiveness of the new design. The development of a new valve design and a valve test loop is presented in the next chapter.

Chapter

4

Development of Novel DSSV Design

Drill String Safety Valves (DSSVs) are used to prevent blowouts through the drillstring during tripping operations. Several case history reviews of well control events have shown evidence of severe problems with DSSVs. Of those problems, valve lock up is most significant resulting in failure to open or close due to high torque.

This Chapter describes the design and testing of a prototype low torque DSSV of the ball valve type. The prototype design relies upon a trunnion-mounted ball and pseudo-fixed seats to achieve low torque operation, with the idealized design goal being a constant actuation torque independent of valve internal pressure. Development of the novel DSSV took place in Phase II and Phase III of the project. Phase II included the design, construction, and testing of an initial prototype. In Phase III, the design was improved based on the results of Phase II and finite element modeling of stresses in the bearing surfaces. Additional testing was then completed. Commercially available valves were tested with the new prototype to provide a basis for evaluation of the new ball and seat design concept.

Phase II Design

Drilling operators have relied upon a manually operated DSSV over remotely actuated valves for stabbing valves [Chilingarian, 1981]. This simplifies stabbing and operating the valve, regardless of the height of the drillstring in the derrick. In addition, a hydraulic actuator could not be permanently affixed to the DSSV because of the need to be able to strip the valve into the well under pressure. One significant disadvantage of manually operated DSSVs is that the available actuating torque to open and close the valve is limited by the physical strength of the operator. When the torque required for operating the valve exceeds the torque applied by the operator, then the valve is said to lock up. Valve lock up is generally considered to occur when the torque that is required for actuating the valve is greater than 500 ft-lbs. Because of dimensional constraints, it is also difficult to manufacture valve stops that can tolerate excessive torque without permanent deformation.

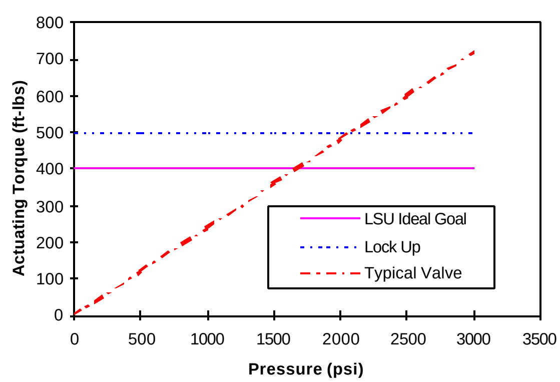

Figure 15: Typical Torque versus Internal DSSV Pressure with no Pressure above the DSSV.

In many ball valve designs, the torque that is required to either open or close the valve is largely a linear function of internal pressure and/or differential pressure across the ball. As such, at some critical pressure the ball valve locks up due to increased friction between internal components. Figure 15 illustrates the problem of lock up for a typical manually actuated ball valve. An ideal DSSV ball valve design is one where the required actuation torque is constant and independent of the valve pressure, up to the maximum design sealing pressure. For such a valve, lock up will never occur as long as the constant required actuation torque is below the lock up torque value. Figure 15 also illustrates this idealized concept.

The objective of Phase I was to design, construct and test a DSSV with an actuation torque vs. differential pressure curve that approaches the idealized, constant torque design. This work focuses upon the low torque ability of the DSSV to open under large differential pressures and to close under high flow conditions. It considers low torque ability of the DSSV to close under equalized pressure as a secondary design criterion, as much less torque is required when the pressure above the DSSV is equal to the internal DSSV pressure.

Ball valve designs commonly employ either a trunnion-mounted ball with floating seats or a floating ball with fixed seats [Piper, 1985]. In both cases, the pressure of the fluid being sealed generates the sealing force between the ball and the seats. The seats are then said to be energized. Of course, due to the friction between the ball and the seats, and under ideal conditions, the required actuation torque for these designs increases linearly with this pressure. In order to achieve the design goal of an actuation torque independent of differential pressure, the fluid being sealed cannot be used to energize the seats. One way of achieving this performance is to mount both the seats and the ball in the valve body. In this manner all forces which act on the ball and the seats are directly transferred to the valve body. Such a design then requires a separate means to energize the seats with a force large enough to provide adequate sealing up to the maximum rated pressure. This is the basis for the design resulting from this project.

Figure 16 shows an assembly drawing of the LSU Phase II prototype DSSV. The valve size is 6 3/8" OD x 2 1/4" ID x 25 3/4" tall with 17-4 stainless steel upper and lower seats and ball. No surface coatings were used on the valve to reduce friction. O-rings between the seats and the ball were used to create the major dynamic seal. O-rings are also used throughout the valve for the secondary static seals. The ball is mounted in a set of sleeve bearings on two sides, through the actuation

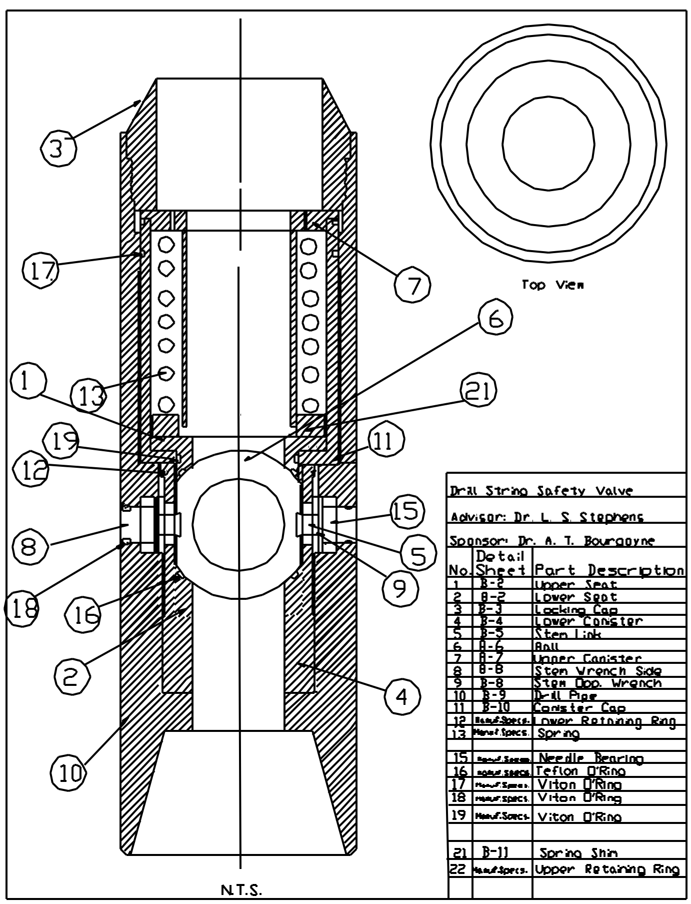

Figure 16: Phase II Prototype DSSV, 6 3/8" OD x 2 1/4" ID

stem and stem link. The tolerances between the ball, stem link, stem and bearings are kept small such that the ball can "float" only a few thousandths of an inch before engaging the sleeve bearings. In essence the ball is trunnion mounted. Using this mounting, the pressure acting against the ball is transferred to the valve body through the bearings. The bearing load capacity is thus a limiting factor for the valve design. Stresses are concentrated in the bearing bore rather than being distributed across the ball and seat. Commercially available bearings were reviewed for this service and those with the greatest load capacity for the given space constraint were selected. Even so, the bearing load capacity limits the differential pressure capability of the valve to 4000 psi. This is only 80% of the target design specification of a differential pressure of 5000 psi. It should also be noted that the torque required to actuate the valve must increase with differential pressure to the extent that the bearing frictional force increases. However, the pressure range was sufficient to allow testing of the new design concept.

The valve design uses a ball housed inside a canister for sealing. The canister assembly was provided by M&M International and is one of their patented design concepts. The bottom seat is part of the canister itself, while the top seat is a separate component that is loaded and held in place by a helical coil compression spring. This spring provides the force which energizes the top seat against the ball and provides a constant sealing force independent of differential pressure when the valve is sealing from below. For a target design specification of a maximum differential pressure of 4000 psi, the sealing pressure between the ball and seats is taken as 1.0-1.5 times this value. Based upon simple assumptions regarding the contact area between the seat O-rings and the ball in the loaded condition, the compression spring was designed to provide a maximum force of 4000 lbf. Shims were designed to adjust this sealing force as needed during testing. Figure 17 shows the resulting force-deflection curve for the spring over three cycles as tested using an Instron machine.

The canister is held into place by a locking cap which is threaded into the top of the valve body. Therefore, any pressure acting against the bottom of the canister is directly transferred to the valve body

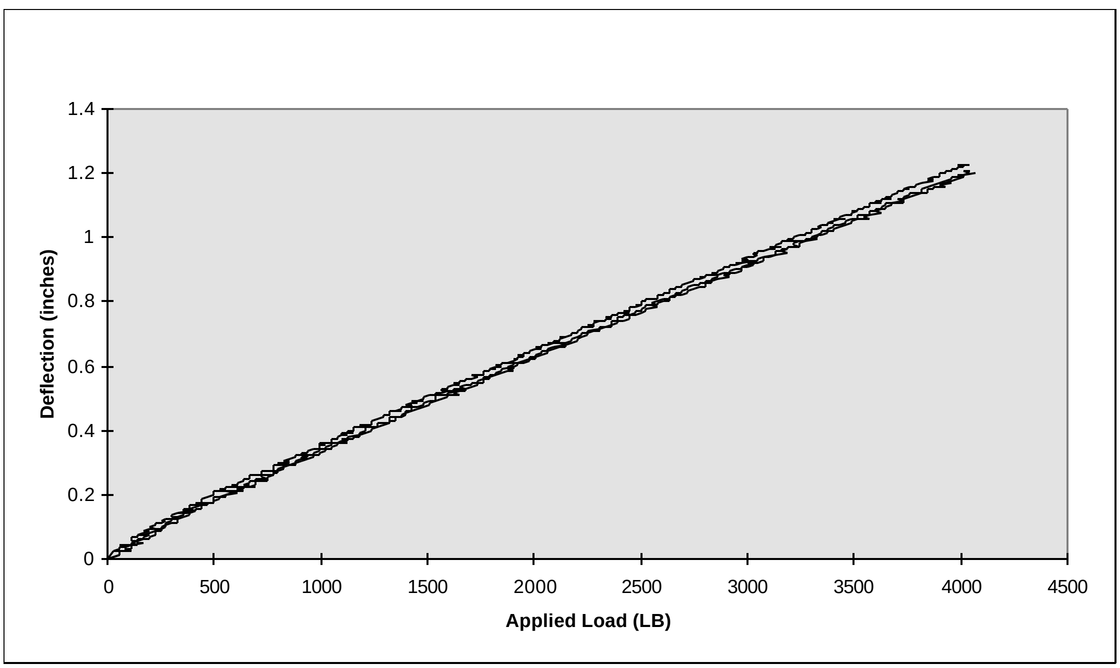

Figure 17: Load versus Deflection for Phase II Loading Spring for Top DSSV Seat

through the locking cap. Together, the bearings, canister, spring and locking cap provide a design where the differential pressure contribution to the actuation torque depends primarily upon the sleeve bearing internal friction. Optimal selection of frictionless bearings then results in a low torque DSSV design when sealing a differential pressure from the below. Another limitation of the prototype design shown in Figure 16 is that when pressure is applied from above, the low torque operation is lost. This is not considered to be a significant limitation as the pressure when sealing from above can be controlled by the operator. Finally, thrust bearings are mounted on the actuation stems to reduce the friction between internal components due to the pressure difference between the interior and exterior of the valve. This design can be termed a trunnion-mounted ball, pseudo-fixed seat design. Table 1 below summarizes the specification of the Phase II DSSV.

Table 1: Phase II Prototype Design - Trunnion Mounted Ball, Pseudo-Fixed Seats

Internal Component Material	17-4 Stainless Steel
Sealing Surface	Viton O-rings
Max. Body Pressure	10,000 psi
Max. Differential Pressure	4,000 psi (Sealing from Below)
Low Torque Operation	Sealing from Below Only
Primary Sealing Surface	Top Seat, sealing from above and below

Phase II Static Test Results

Three types of static pressure tests were performed on the prototype safety valve. These were a hydrostatic test on the valve body, a measurement of torque required to open under a differential pressure, and a measurement of torque required to close under 100% equalized pressure. No tests under flow conditions were conducted in this first series of tests. All tests were performed using water, pressurized by a hydraulic test stand at a local valve manufacturer's facility. These tests were similar to those performed by the commercial DSSV manufacturer. Torque readings were made by a calibrated digital torque wrench. The hydrostatic pressure test required the valve body and stem seals to effectively seal twice the maximum allowable working pressure of the valve for 5 minutes. The valve exceeded this performance by sealing a pressure of 10,000 psi for a 15-minute period. Torque measurements under differential and 100% equalized pressure were obtained as part of the following testing procedure:

1. Begin with valve in closed position;
2. Pressurize valve from below to the desired level;
3. Hold at this pressure for 5 minutes and check for evidence of leakage across the ball;
4. Open the valve using the torque wrench to obtain the torque required to open under differential pressure (The valve is now at 100% equalized pressure on both sides of the ball);
5. Close the valve using the torque wrench to obtain the torque required to close under 100% equalized pressure.

Torque to Open under Differential Pressure

The torque to open the Phase II prototype valve was measured for various internal pressures with no pressure above the DSSV. Similar measurements were made on two additional DSSV valves provided by two different commercial valve manufacturers. All of the valves are of 2 1/4" ID bores. Both commercial DSSVs tested were one-piece designs rather than the very low cost two-piece design patented by Texas Iron Works (TIW). One of the commercial DSSVs tested was a premium valve with several special features for reducing torque. As was discussed earlier, due to limitations in the bearing load capacity, the maximum rated differential pressure for the prototype LSU DSSV is 4000 psi. The results of the tests, which are shown in Figure 18, reflect this pressure rating as differential pressure tests were performed up to this maximum limit.

During these tests, the valve stem for the LSU DSSV yielded when excessive torque was accidentally applied while adjusting the pneumatic operator pressure in the test stand. The stem was re-machined, but this difficulty resulted in only four data points for these tests. The results indicate that the actuation torque to open the two commercial DSSVs was largely linear, while that for the LSU valve varied non-linearly. This suggests that certain components within the valve shifted under pressure due to incorrect tolerance during assembly, and that surfaces other than the sleeve bearings and O-ring seats were loaded and in contact. Finally, the data does show a cross-over between the endpoints, where at low pressures the LSU DSSV requires more actuation torque than the commercial DSSVs but at high pressures it requires less actuation torque.

Data for the commercial DSSV1 shown in Figure 18 was obtained after the valve had been used in a number of preliminary experiments and had been allowed to weather outdoors for several months. The data for DSSV2 was obtained with a new valve. Data published for DSSV2 [Tarr, 1998] after many

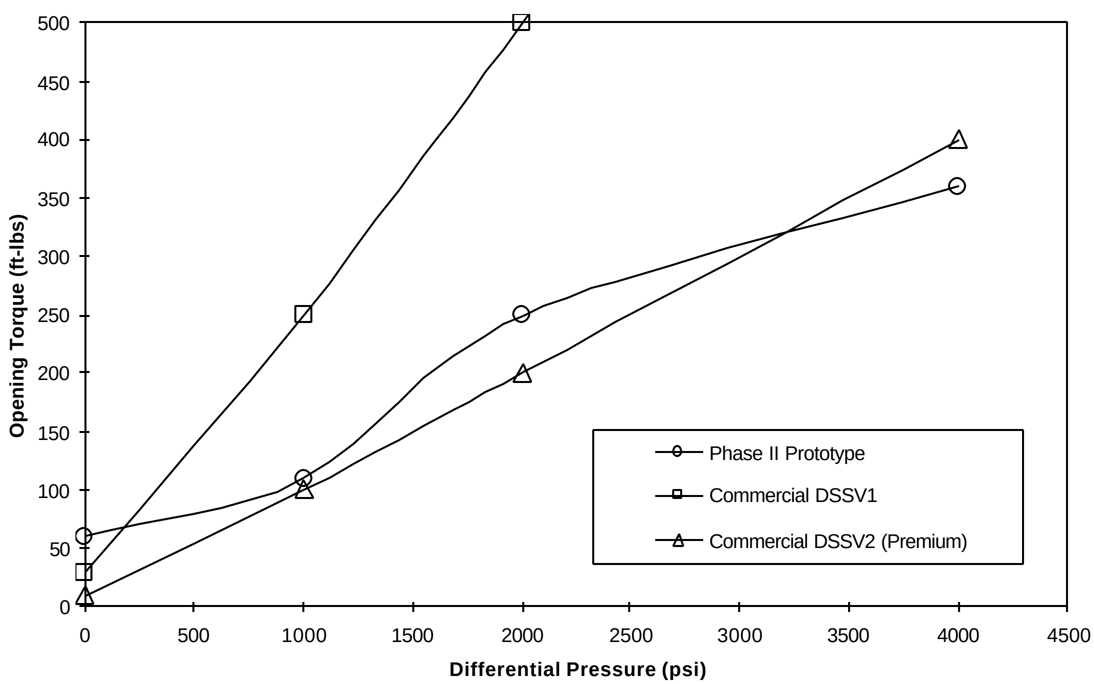

Figure 18: Torque to Open Pressurized DSSV with no Pressure above valve.

cycles and after exposure to a water-base mud containing sand fell slightly above the data for DSSV1. Experience during the development of the testing program indicated that the operating torque for DSSVs under a differential pressure depended to a great extent upon the previous history of the valve.

Torque to Close DSSV under Equalized Pressure

Figure 19 shows the torque required to close the Phase II prototype design as a function of pressure when the pressure above the valve is equal to the pressure within the valve. Similar data for another commercially available valve is provided for a qualitative comparison and is labeled "Commercial DSSV1." The data shown for the commercially available valve was taken from catalog curves published by the manufacturer. Both the Phase II Prototype valve and the commercially available valve are of 2 1/4" ID bore and have a one piece body design. This data indicates that the Phase II Prototype required a largely constant torque to actuate against equalized pressure. The required actuation torque varied between 60-80 ft-lbs over an equalized pressure range of 0-10,000 psi. This compared favorably to the data taken from product datasheets of the commercially available valve. These data shows that for 100% equalized pressure, the Phase II Prototype achieved the goal of an actuation torque which is largely independent of the internal valve pressure.

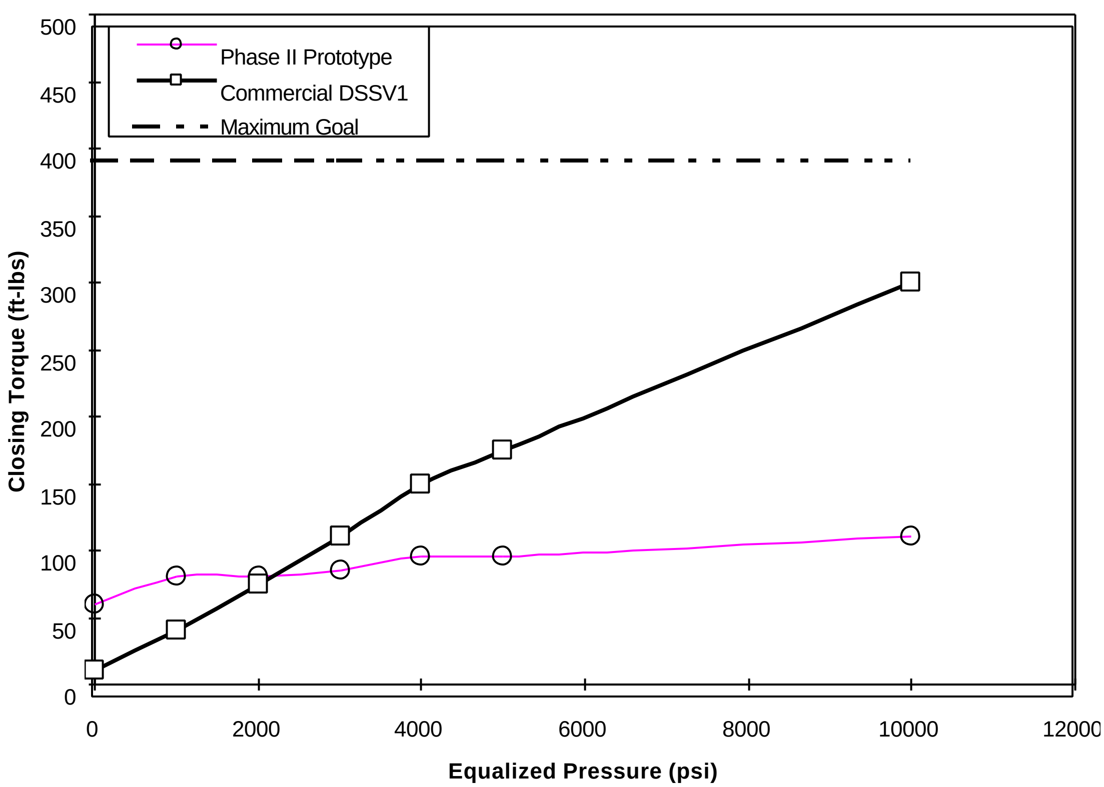

Figure 19: Torque to Close DSSV as a function of Equalized Pressure, 2-1/4" ID Valves (Pressure above Valve Equal to Pressure within Valve)

Post Test Valve Inspection

Upon completion of the testing, the valve was disassembled and inspected. Inspection revealed significant galling[3] between the lower seat and the ball, between the ball and stem link, and between the stem link and stem. This wear indicates incorrect dimensional tolerance after assembly of the parts. Indeed, an assembly review revealed that the O-ring groove for the bottom seat was machined to the wrong size. The wear pattern between the ball and the stem link indicated that the ball was slightly off center. Finally, wear between the stem and the canister windows indicated that the bearing tolerances were too loose and the canister acted as a bearing surface. Each of these deficiencies would be expected to increase actuation torque for the ball valve as a function of differential pressure.

The results achieved with the Phase II prototype indicated that the basic design concept was valid and the research team recommended that this design concept be kept for the Phase III design team. It was also recommended that additional analytical work be done to model stress and strain in the bearing assembly before doing the Phase III design work.

Finite Element Analysis of Bearing Bore

During the Phase II design of the valve, the bearing bore was identified as a key element to the success of the new design concept. The bearing bore was designed to fit into the outer shell or body of the valve. The double stem bearing bore assembly must handle loads as high as 20,000 psi to achieve a DSSV with a 10,000 psi working pressure and a 2.25-in internal diameter. In terms of load handling, the bearing surface must be able to handle high loads with very little deflection. A Finite Element Analysis (FEA) was used to better understand how displacements and stresses on the bearing surface act under certain load conditions. ANSYS 5.0, a commercial software package, was used to perform the finite element analysis.

The finite element model should approximately model the actual bearing assembly in two dimensions. The actual bearing assemble is shown in Figure 20. A review of how the stem and bearing bore fit together and the how the bearing bore and the casing fit together shows that they can be modeled in 2-

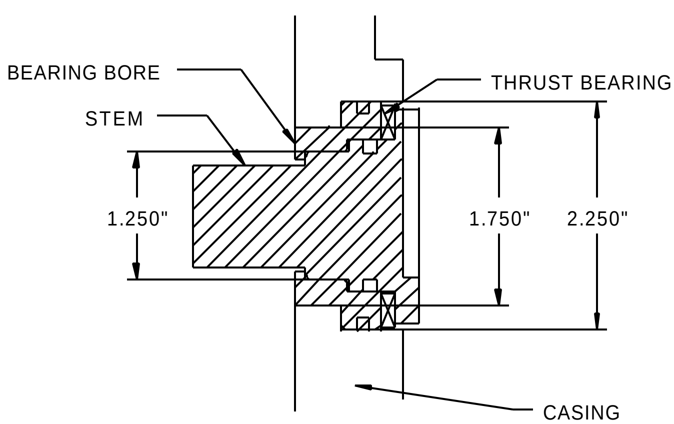

Figure 20: Bearing Assembly

[3] Galling was indicated by adhesive wear where incomplete cold welding occurred, leaving large streaks in the surface.

D as two concentric circles. The bearing was modeled as a “plane 82 2-D 8-node” structural-solid with the key-option set to three in the ANSYS software package. What this means is that there are eight nodes with each having two degrees of freedom to move in the x and y coordinate directions. The key-option of three allows the model to be plane stress with a thickness to be input. The two sets of o-rings were left out of the model because they are used only as a seal surface and not a bearing surface. The model assumes that the valve body is rigid, allowing no deflection of the outer surface. The bearing material was assumed to be homogeneous, having a modulus of elasticity of 30 Mpsi and a Poisson’s ratio of 0.292. The compressive strength of the bearing material was 135,000 psi in an annealed condition. The effect of a Melonite coating on the bearing surface was not included in the model analysis. Radial clearances of 0.001 to 0.003 inches were investigated.

The mesh used in the Finite Element Analysis can be seen in Figures 21 and 22. Figure 21 shows the calculated displacement for a load of 10,000 lbf (internal valve pressure of 5,000 psi) and a radial clearance between the bearing surfaces of 0.003 inches.

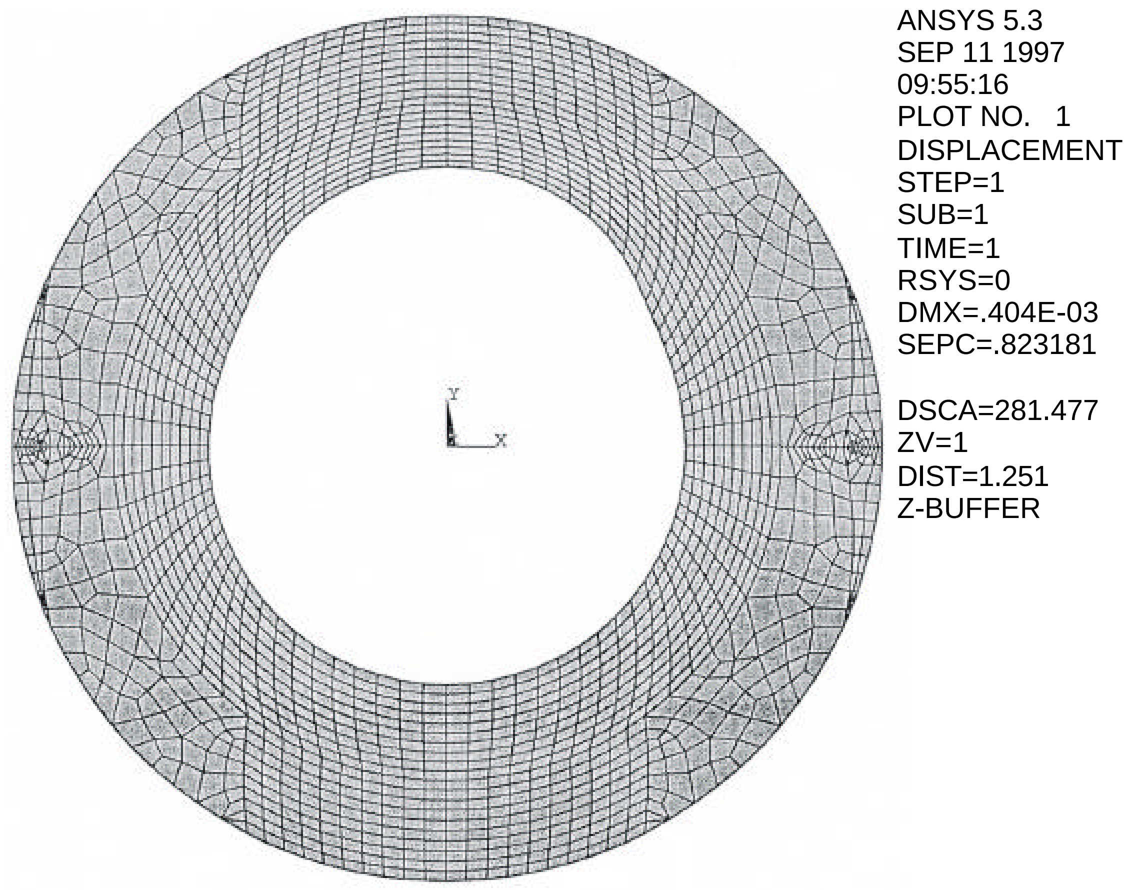

Figure 21 – Visualization of Calculated Displacements for Bearing Bore using a Radial Clearance of 0.003 inches and an Internal DSSV Pressure of 5000 psi

Figure 22 shows the calculated stress distribution on the bearing bore for the same conditions. Based on the results of the analysis, the bearing design should be capable of handling an internal pressure of 10,000 psi. However, it was decided that construction of a DSSV with a trunnion-mounted ball and a working pressure of 10,000 psi was beyond the scope of the current project.

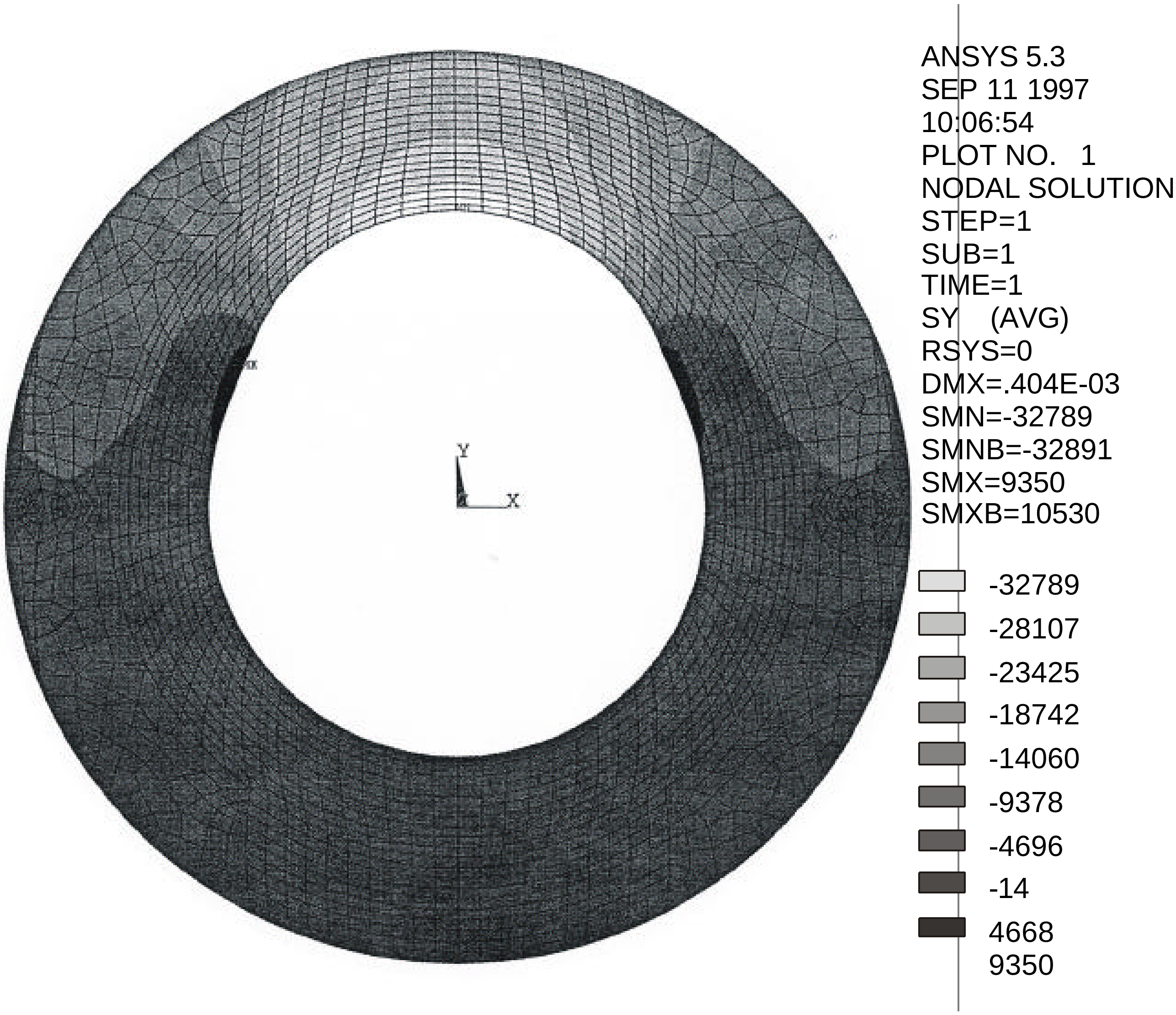

Figure 22 – Calculated Stress Distribution for Bearing bore using a Radial Clearance of 0.003 and an Internal DSSV Pressure of 5000 psi. (Stress shown by ANSYS in psi with gray scale)

Phase III Design

Figure 23 shows an assembled drawing of the Phase III valve. Detailed machine drawings are provided in Appendix A. The valve size is 6 3/8" OD x 2 ¼" ID x 27" tall. All internal components are 17-4 stainless steel coated using a nitride process. The surface coatings are used on the valve to reduce friction. The valve uses Teflon O-rings between the seats and the ball to create the major dynamic seal. Butyl O-rings are used throughout the valve for secondary static seals. The ball is mounted in a set of sleeve bearings (the stems and bearing bores). The tolerances between the ball, stem links, stems, and bearing bores are small so that the ball is allowed to "float" only three thousandths of an inch before engaging the bearing surfaces. This design is considered a trunnion mounted ball valve. Using this mounting, the pressure forces acting against a closed ball are transferred to the valve casing through the through the stems and bearing bores.

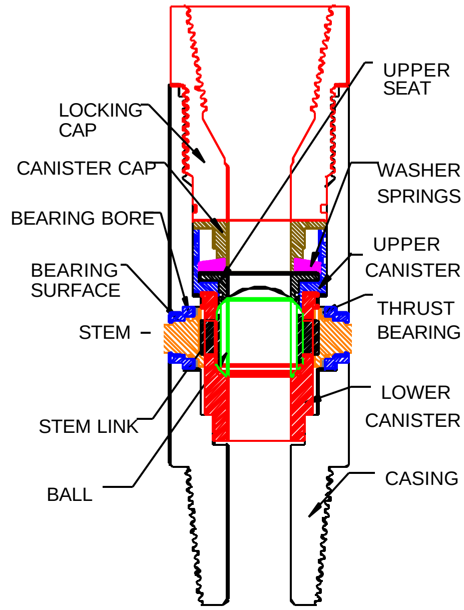

Figure 23: Phase III Prototype DSSV Design

The Phase III Prototype valve design, like the Phase II design, used a ball housed inside a canister for sealing. The bottom seat is part of the lower canister itself, while the upper seat is a separate component that is loaded by five Belleville springs. The springs provide the force which energizes the top seat against the ball and provides a constant sealing force independent of differential pressure when the valve is being sealed from below. For a design specification of 5000 psi for the maximum differential pressure, the sealing force between the seat and the ball was taken as 1.1-1.5 times this value. Based upon assumptions regarding the contact area between the seat, O-rings, and the ball in the loaded condition, the Belleville springs was designed to provide a maximum force of 5500 lbf.

The canister is held into place by a locking cap which is threaded into the top of the valve body. Therefore, any pressure acting on the bottom of the canister is directly transferred to the valve casing through the locking cap. Optimal selection of frictionless bearings then results in a low torque DSSV design when sealing a differential pressure from below. Finally, thrust bearings are mounted on the stems to reduce the friction between the internal components due to the pressure difference between the interior and exterior of the valve. The Phase III design was also a trunnion mounted ball, pseudo-fixed seat design. Table 1 below summarizes the performance specifications of the valve as it is presently designed.

Table 2: Phase III Prototype Trunnion Mounted DSSV with Pseudo Fixed Seats

Internal Component Material	17-4 Stainless Steel Nitride Process Coating
Sealing Surfaces	Teflon O-rings
Max. Body Pressure	7,500 psi
Max. Differential Pressure	5,000 psi (Sealing from Below)
Low Torque Operation	Sealing from Below Only
Primary Sealing Surface	Top and Bottom Seats (Sealing from Above and Below)

Design Changes

The bearing design was changed significantly from the Phase II design which had utilized a commercially available sleeve type bearing with two separate press-fitted bearing surfaces. One bearing surface was to be press-fit onto the stem (inner sleeve), while the other was to be press-fit into the casing (outer sleeve). A schematic of this can be seen in Figure 24.

The design was changed for several reasons. First, press-fitting a bearing surface into the valve body with perfect concentricity is difficult to accomplish. Second, replacing the bearing surfaces would be difficult. For example, during testing procedures if the bearing surface is galled or damaged, it would have to be replaced. Since the parts are press-fit and in a recess, the only way to remove them is to machine them off. It is very difficult to machine the bearing surfaces without damaging the stems. Also, it would be difficult to remove the bearing surfaces from the valve body. These would also have to be removed by machining.

For the required forces and size constraints no commercially available bearing could be found. Therefore, a new bearing was designed from scratch. Also, results from last years testing showed localized yielding on the bearing surfaces because they were not hard enough. Instead of using a press-fit bearing surface on the stem, the new bearing surface was designed to be an integral part of the stem. The original bearing surface ring was designed to be 17-4 stainless. This was not changed because the stems were also 17-4 stainless. However, we had to increase the size of the thrust bearing in order to compensate for the increased diameter of the stem. An isometric view of the new stem with its integral bearing surface is shown in Figure 25.

The second design change was with the outer bearing surface. Instead of using a press-fit ring with an internal bearing surface into the casing, we used a bearing bore with interference fit at a

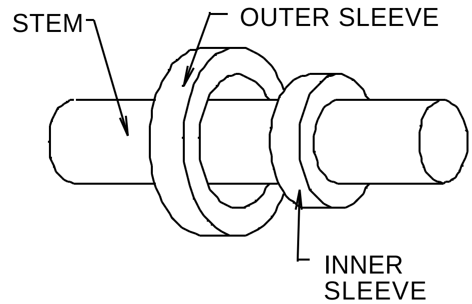

Figure 24: Phase II Bearing Design

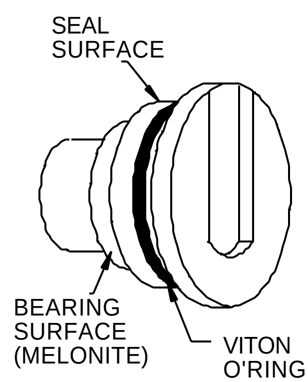

Figure 25: Phase III Stem Design

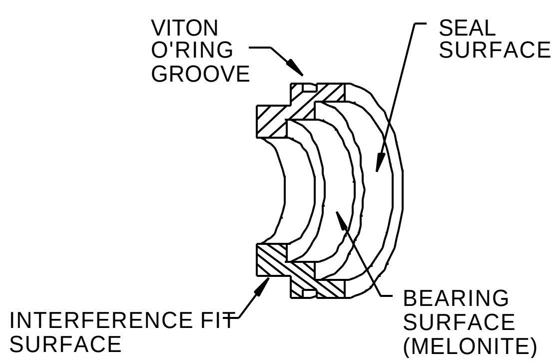

Figure 26: Phase III Bearing Bore Design

non-critical area. An isometric cross section of the bearing bore can be seen in Figure 26. This bearing bore design eliminated several problems. First, the radial thickness of the bearing bore eliminated any chance of deformation during the press-fit. Since the smallest outer diameter of the bearing bore is the only surface of the part that will be interference fit, a slight deformation at this point will not effect the bearing surface. Second, the new design of the bearing bore allows for easy removal. If the bearing bore needs to be removed from the valve, it can be pressed out of the valve casing with little difficulty. The bearing bore was constructed of 17-4 stainless steel. The new bearing bore and stem design[4] is illustrated in Figure 27.

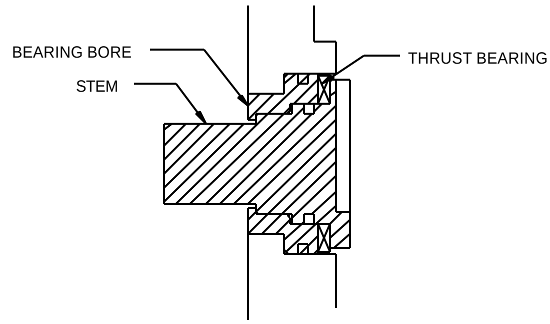

Figure 27: Mating Stem and Bearing Bore

With the new design changes, the possible stresses were re-calculated using bearing design equations and the factors of safety were checked. The Maximum Allowable Working Pressure for the valve is 5000 psi. With the valve in the closed position, this differential pressure causes a force of 10,000 lbs to act upon the two stems and bearing bores. The diametrical clearance between the bearing surfaces on the stem and the bearing bore is 0.003 inches. Hertzian contact stresses were used to calculate the maximum pressure on the bearing. This calculation provided an independent analysis of the Finite Element Analysis model done previously by the first design team. The contact area was calculated using the following equation:

$$P\max = \frac{2F}{p\ aL}$$

Where:

$$a = \left[\left(2\frac{F}{p\ \ L} \right) \frac{\frac{\left(1 - n_1^{\ 2}\right)}{E_1} + \frac{\left(1 - n_2^{\ 2}\right)}{E_2}}{\left(\frac{1}{d_1} + \frac{1}{d_2} \right)} \right]^{\frac{1}{2}}$$

F = Force between stem and bore;

[4] The valve stem used in the Phase III Prototype was modified to fit a pneumatic valve operator in the flow loop used for testing the valve against flow. The stem is shown protruding out of the valve body. The valve stem used on a working DSSV valve would be recessed and require the use of a valve wrench for manual operation.

L = Length of bearing surface;

ν = Poisson's Ratio;

E = Modulus of Elasticity;

d_1 = Diameter of bearing surface on stem;

d_2 = Diameter of bearing surface on bearing bore;

1 = Denotes stem material properties; and

2 = Denotes bearing bore material properties.

The static factor of safety was calculated by dividing the yield strength by the maximum pressure (Pmax). The factor of safety for a 5000 psi working pressure is 4.4. This is the ideal situation in which only the intended "bearing surface" is being used. For a 10,000 psi working pressure, the static factor of safety would be about 2.2.

There is a second scenario in which the factor of safety will change. The seal surfaces between the bearing bore and stem could come into contact when pressure is applied to the valve. This is because the required diametrical clearance between the sealing surfaces is only slightly above that of the bearing clearance (approximately 0.002"). In the case that the sealing surfaces and the bearing surface do come into contact, the contact area would increase thus causing the maximum pressure to decrease. The worst case would be if the sealing surface absorbed all loads and the intended bearing surface absorbed none. This could happen if the tolerances between the stem bearing surface and the bearing bore were not met. This case was also modeled using the equations given above. A factor of safety of 2.37 was calculated for a working pressure of 5000 psi. For a working pressure of 10,000 psi, the factor of safety would be about 1.18.

Testing of Belleville Springs

The Belleville springs were used in the Phase III design to force the seat against the ball. The Belleville washer springs were designed to have an inside height of 0.080", and a thickness of 0.150", giving a height over thickness ratio (the "h/t" ratio) of 0.533. This h/t ratio gives an approximately linear response when deflection is plotted against force. The springs as received from the manufacturer differed from the original design. The received inside height was 0.070" and the thickness is 0.105". The h/t ratio is 0.667 and the calculated spring response is less linear. The force of the received springs was computed to be 1115 lbf at 0.065" of deflection. Five springs should produce a force of 5575 lbf at the 0.065" of deflection specified in the design.

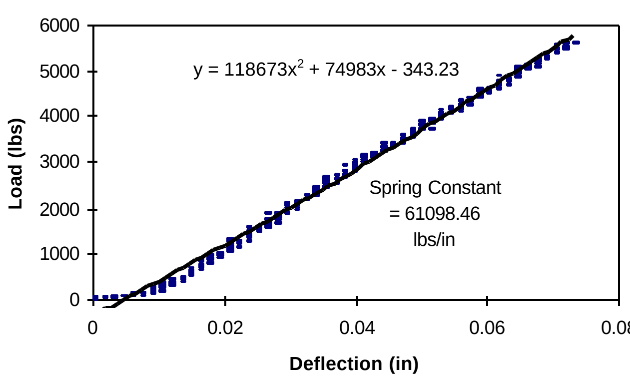

Figure 28: Load versus Deflection for the stack of Five Springs used in Phase III Prototype DSSV

To check the computed spring values, the as built springs were tested in an Instron compression tester. First, each spring was tested individually to determine the individual spring constants. The springs were loaded between flat steel plates. Second, a stack of five springs was tested. This test was done to verify that (1) the actual spring load was near the computed spring load and (2) that the total load could be computed using the sum of the individual spring constants and the same deflection for all five springs. The results of the tests showed that a force of 5032 lbf would be generated for the stack of five springs used in the Phase III Prototype. Shown in Figure 28 are the test results.

Assembly Procedure

The Drill String Safety Valve consists of many different integrating parts. Therefore, assembling the parts in the correct order is a must. The following is the procedure that was used to assemble the valve.

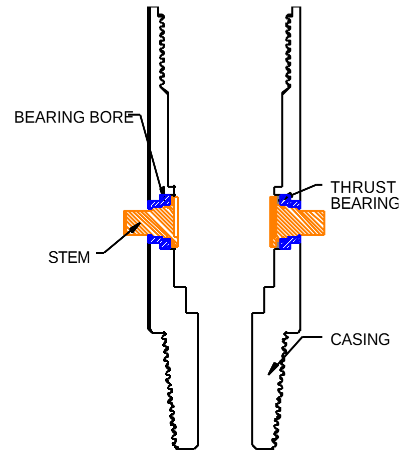

Figure 29: Step 1 of Assembly

1. The bearing bore o-rings were greased and then installed on the bearing bore. With the casing in a vertical position, the bearing bores were inserted as shown in Figure 29.

2. The thrust bearings were greased with a light coat of grease and placed onto the stems. The stem O-rings were placed into their respective grooves. The O-rings and bearing surfaces were also greased. The stems were inserted into the bearing bores. Finally, the slots on the stem that accept the stem links were positioned vertically. Figure 29 also illustrates this assembly.

3. The Teflon O-ring was inserted into the O-ring groove in the lower canister. It was "set" by placing the ball on the O-ring and striking the ball firmly with a rubber mallet. The ball was then removed. The next step was to take the lower canister and insert the stem links such that the stem links were also vertically positioned. Once the stem links were in place, the ball was then inserted into the lower canister with the slot in the ball aligned with the stem links. A schematic of this can be seen in Figure 30.

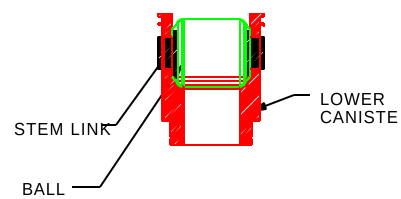

Figure 30: Step 2 of Assembly

4. The Teflon O-ring in the upper seat was inserted in the upper seat O-ring groove and was "set" using the same method used to set the lower Teflon o-ring. The upper canister was placed on top of the lower canister and then fastened together by using a snap ring. The upper seat was then placed into the upper canister with the Teflon o-ring in contact with the ball.

5. The next step was to insert the five springs with the concave sides down. Once the springs were in place, the upper canister cap was placed on the upper canister. For the upper canister cap to be attached to the upper canister, the springs were compressed using a suitable shop press.

6. The whole inner canister was placed into the press and then compressed until a snap ring could be inserted into a snap ring groove to connect the upper canister and the upper canister cap.

7. Once step 6 was completed, the inner canister was one integrated part. The inner canister was removed from the press and all of the necessary outer o-rings were greased and installed. A schematic of the inner canister can be seen in Figure 31.

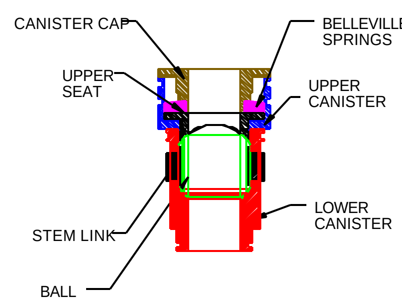

Figure 31: Assembled Inner Canister

8. The final step was to insert the inner canister into the valve body. Once the inner canister was situated properly in the casing, the upper casing cap with its attached and greased o-ring was screwed on and the assembly was thus completed.

Phase III DSSV Test Program

The initial test conducted on the Phase III Prototype were hydrostatic tests conducted at a commercial valve manufacturing facility. The test was similar to the test performed after manufacturer of a commercial valve. This test began with the water-filled valve in the closed position and one end of the valve fitted with a threaded cap. The threaded cap was ported to allow hydrostatic pressure to be applied. A high-pressure pump was used to increase the internal pressure of half of the valve. To pass the test, the pressure must be held for ten minutes with no significant leaks or drops in pressure. The opposite end of the valve is then tested in a similar manner. The Phase III Prototype was tested successfully in this manner to 7500 psi, which was 1.50 times working pressure of the valve. The DSSV was then removed from the test apparatus and actuated to ensure no deformation occurred with the moving parts of the valve. Finally, the valve was disassembled and inspected for internal damage or deformation of internal components.

The second type of tests conducted on the Phase III Prototype was a measurement of the torque to operate the valve with internal pressure. For these tests, the pressure above the ball and upper seat was equal to the internal pressure so that there was no pressure differential across the ball and seat. These tests were conducted at the LSU Research Well Facility operated by the Department of Petroleum Engineering. Similar tests were also performed on two commercially available DSSVs using the LSU facility. Additional test results of this type had been reported in the literature for other valves. This allowed the performance of the Phase III prototype to be compared to several other valve designs.

A schematic of the test stand used for the equalized pressure tests is shown in Figure 32. The bottom of the valve was threaded into the test stand. The bottom of the test stand is fitted with adapters that allow for the connection of a hydrostatic pressure source. Pressurized was provided by a pneumatic pump capable of providing pressure up to 15,000 psi. The test stand cap is threaded into the top of the valve, and pressure sensors were threaded into the top of the cap. Torque and position sensors were attached to the valve stem. The signals from these sensors are sent to a computer with a sample rate of three samples per second. The high-pressure pump was used to charge the fluid to the desired pressure. As the valve was closed and opened, the torque and position was recorded. Tests were repeated three times for pressures of 1000, 2000, 3000, 4000 and 5000 psi. When the tests were completed, the DSSV was disassembled,

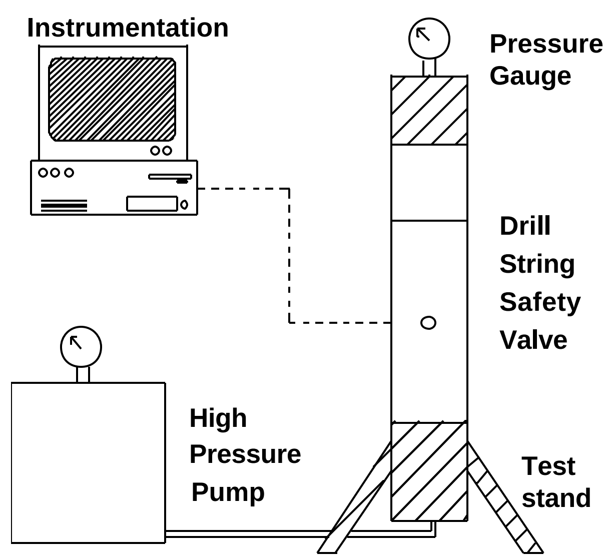

Figure 32: Equalized Pressure Test Apparatus

and the internal components were again inspected for damage and deformation. The test data is presented and compared graphically to other DSSVs in the next section of this report

The third type of test conducted on the Phase III DSSV measured the closing torque when the valve was closed with water flowing through the valve. A flow loop was constructed at the LSU Research Well Facility for these tests. A schematic of the flow loop is shown in Figure 33. A mud tank filled with water was connected to a Halliburton HT400 positive displacement pump. The pump was connected to a SWACO automatic choke in parallel with the design valve. Torque and position sensors were attached to the valve stem. The signals from these sensors were sent to the computer configured with

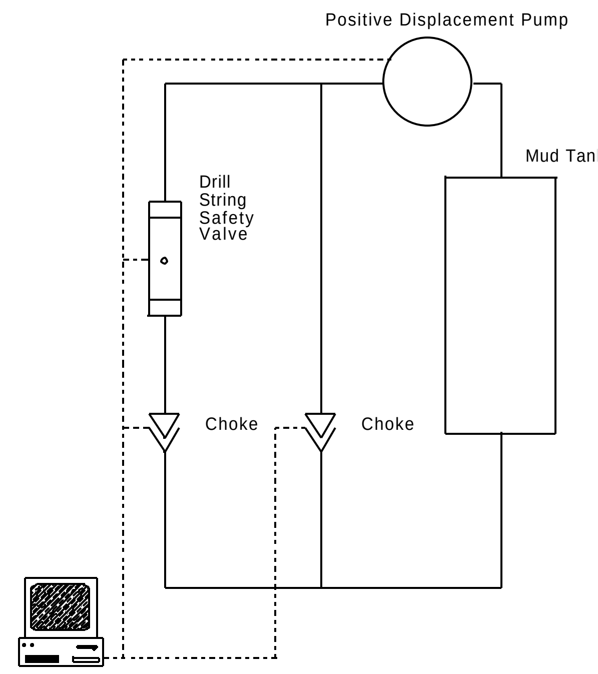

Figure 33: Schematic of Test Flow Loop for Closing DSSV against Flow

Figure 34: Photograph of Pump and Test Stand

instrumentation software. The pump was used to circulate the fluid at the desired flow rate. A downstream pressure was held against the DSSV by another choke places in series with the DSSV. The choke in parallel with the DSSV was used as a pressure release device. This choke was set so that it is normally closed at pressures below a set pressure psi. When the choke was closed, the entire flow passed through the DSSV until the DSSV closes and the pressure in the system builds to the set pressure. The automatic choke then opens and holds the pressure constant at the set pressure. This arrangement approximately simulates the pressure build-up under a DSSV after it closed a flowing well. Torque and position during closure were recorded three times for each of six flow rates ranging from 100 to 350 gallons per minute.

DSSV failure to close has also occurred during completion operations involving reverse circulation through the drillstring. When a DSSV is used as a safety valve with downstream piping attached to the top of the valve, significant pressure can develop at the DSSV due to flow through the downstream piping. The purpose of the choke in series with the DSSV (Figure 33) was to simulate the effect of downstream piping.

Theoretical Operating Torque

Calculations were made in order to predict the torque required for operation of the Phase III DSSV. These calculations were based on the design of the valve assuming that all design dimensions were achieved during construction.

Before calculating torque to operate the DSSV in a test condition, calculations were made for operating the valve with no pressure, as would be the case in a routine actuation test done on the rig floor. The only force applied to the valve in this situation is the 5,000-lbf force from the springs. Assuming that the ball is exactly centered, the contacting surfaces are the interfaces between the ball and Teflon O-rings, and the interface between the bearing O-ring seals and the bearing bores (Figure 35).The friction between the ball and the Teflon seals was calculated using the normal force and the coefficient of friction between the two surfaces.

The frictional force caused by one rubber bearing O-ring seal was calculated with the

$$F_{bs} = (f_c L) + (f_n A)$$

following equation [Warring, 1981]:

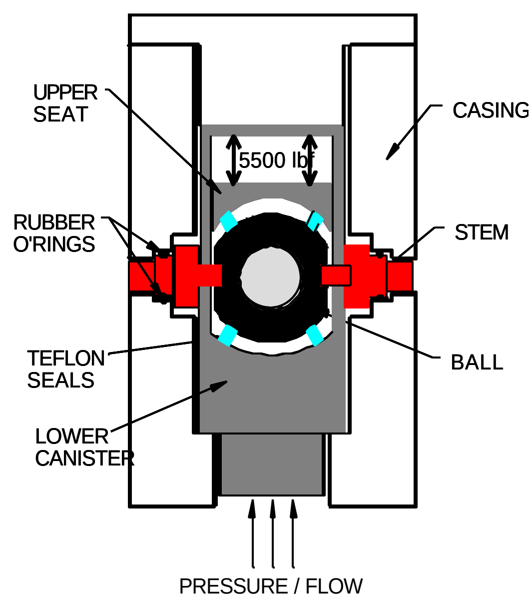

Figure 35: Phase III Prototype Schematic showing Force acting between Ball and Seat and in Trunnion Bearing

Where:

f_c = Friction per unit length due to O-ring compression (lbf/in)

L = Length of seal rubbing surface (in)

f_n = Friction per unit area due to fluid pressure (lbf/in2)

A = Projected area of seal (in2)

The torque needed to overcome the friction involved with the Teflon O-rings and the bearing O-rings at atmospheric pressure is given by:

$$T_{np} = 2(m_t F_{spring} R_{ball}) + 2(F_{bs} R_{seal})$$
$$= 2(0.04 * 5500 * \frac{1.8}{12}) + 2(0.8 * p * 1.498 * \frac{1.488}{2*12}) = 66 \text{ ft - lbs}$$

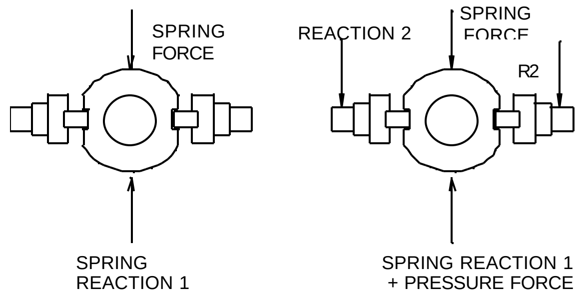

Figure 36: Force Balance with no Pressure Differential and with 5000 psi Pressure Differential

When an equalized pressure is applied inside the DSSV, the friction between the ball and Teflon O-rings is the same as the previous case. However, the stem will be forced axially into the needle thrust bearings. Since the needle thrust bearings allow for low torque operation at high forces, the operating torque was expected to increase only slightly with increasing equalized pressure applied to the valve.

When a pressure differential exists across the ball and upper seat such that the pressure inside the valve becomes greater that the pressure above the valve, additional load will be transferred to the trunnion bearings in the stem (Figure 36). When there is no pressure in the valve, the only forces on the ball are the “squeezing” forces caused by the spring. When pressure applied to the valve, the pressure causes the ball to push upward. With the tolerances set correctly in the valve, the ball will compress the spring slightly and crush the Teflon o-ring a little more causing the stem bearing surfaces to contact. The friction of the bearing surfaces was calculated assuming that all force due to an applied pressure differential is transferred to the stem bearing surfaces.

The actuation torque for an applied pressure differential was estimated using:

$$T_{NP} = 2(m_{Teflon} F_{spring}) R_{ball} + 2(f_{compression} L_{O-ring}) R_{O-ring} + 2(m_{Melonite} F_{bearing}) R_{bearing}$$

In older valves or when excessive pressure is applied to the ball, the ball and seat come into contact the coefficient of friction in the first term of this equation changes to that of non-lubricated Melonite. The only term that is a function of pressure is the friction in the bearings. The coefficient of friction given by the manufacturer for Melonite is 0.35 for dry surfaces and 0.04 for lubricated surfaces. The bearing surfaces for the manufactured valve were greased prior to assembly. The coefficient of friction was taken as the average of the dry and lubricated values because of the inability to lubricate the bearings after assembly. A plot of the actuation torque as function of differential pressure is shown in Figure 37. The plot shows that ideally the valve is expected to operate under 400 ft-lbf of actuation torque up to a 5000-psi differential pressure. Once closed, the valve would hold against higher pressure, but could not be opened without applying an equalizing pressure above the DSSV.

The required torque to close the valve during flow is a maximum during the last interval of rotation of the ball. This is due to the water hammer effect when flow is suddenly stopped. The maximum pressure experienced by the ball during a closing operation is given by:

$$\Delta P = \frac{\rho c \Delta V}{g}$$

Where:

ρ = Density of drilling mud.

c = Speed of Sound through mud.

ΔV = Mud Velocity through valve before closure (assume to be zero after closure).

g = Acceleration due to Gravity

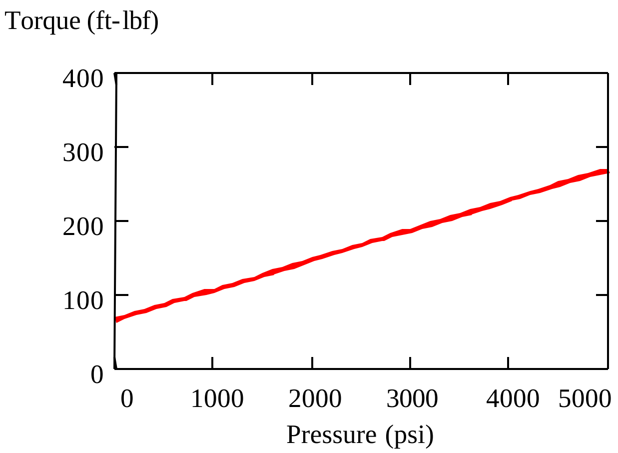

Figure 37: Theoretical Actuation Torque as a Function of Upward Differential Pressure

The differential pressures caused by closing the valve against a flow were calculated for an unweighted drilling fluid for the range of the actual test flow rates. The water hammer pressures calculated are valid for a stabbing valve as long as the closing time is less than the time required for a pressure wave to travel to the bottom of the drillstring and then be reflected back to the surface. This would be a valid assumption for most actual cases but not for a test loop. The required torque was calculated using the equations for ΔP and for T_{NP} given above. The results are shown in Figure 38. For flow rates up to 350 gal/min (8.3 bbl/min) the Phase III Prototype DSSV should close easily. The water hammer pressure at 350 gal/min is about 1750 psi. The calculated results vary linearly with mud density. For 16.7 pound per gallon mud, the torque required would be twice those shown in Figure 38.

The ball and seats, like the bearing surfaces, are both coated with Melonite. If the ball is not centered correctly because dimensional tolerances were not achieved or if the Teflon o-rings are not seated properly during assembly, the ball and the seats could come into contact at higher pressure differentials and flow rates. The result would be higher actuation torque because the reaction force from the pressure would be at a non-lubricated surface. The main reason it is desired that the force be transferred to the bearing surfaces is because they can be lubricated.

Test Results

The test results obtained in this study for the Phase III Prototype were compared to test results obtained for commercially available valves. Some of the test results for commercially available valves were obtained in this study and some were obtained from published results by Tarr et al [1998]. Some of the data published by Tarr et al were obtained after flowing a 16-ppg water base mud containing sand through the valve. All of the data obtained in this LSU study was done with water or unweighted gel mud.

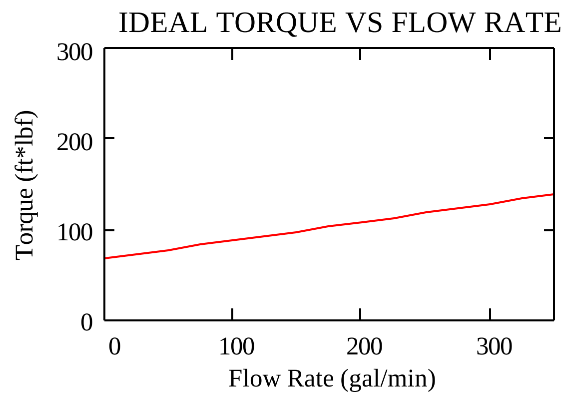

Figure 38: Calculated Torque required for closing Phase III Prototype against flow of unweighted Mud

Operating torque as a function of pressure are shown in Figure 39 for the Phase III Prototype and for five commercially available DSSV designs. The results of Figure 39 were obtained by closing the DSSV with pressure above the valve equal to the internal pressure of the valve (equalized pressure condition). DSSV0 was a valve with the traditional two-piece TIW design with a floating ball and fixed seats. DSSV1 was a more advanced design with a one-piece body, a floating ball, and fixed seats. DSSV2 was a premium design similar to DSSV1 but with additional needle bearings in the stem and special coatings for torque reduction. DSSV3 was a relatively new design with a trunnion-mounted ball and floating seats. DSSV4 was a single body, double ball design intended for use with a top drive.

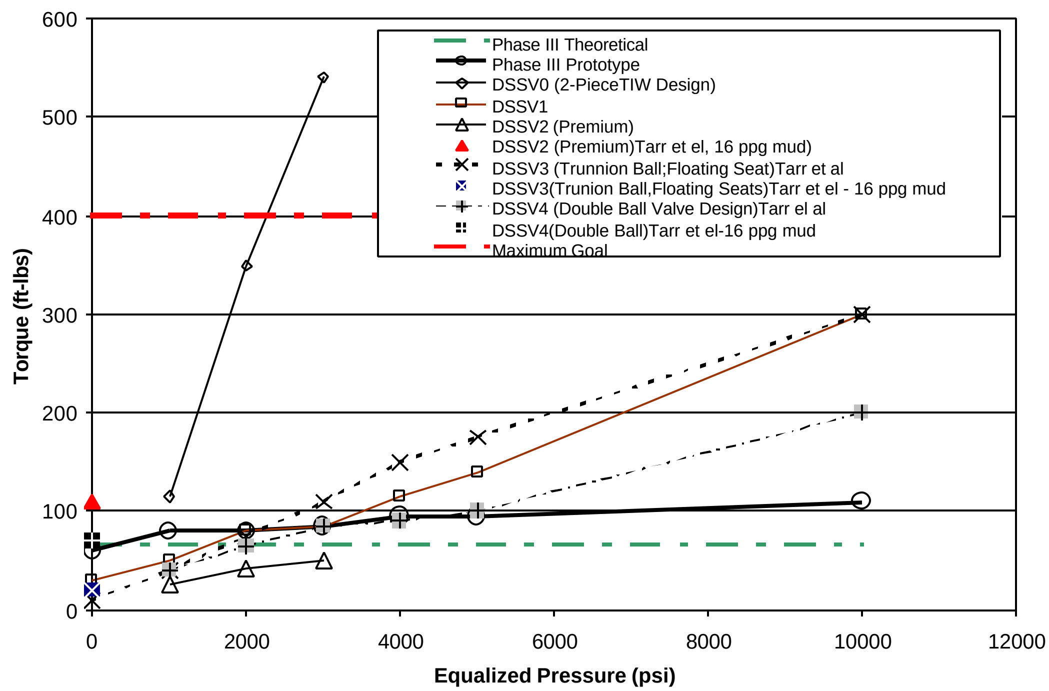

Figure 39: Maximum Closing Torque as a function of Internal Pressure with Pressure above DSSV equal to internal Valve pressure

Note that the operating torque for the Phase III Prototype was much less sensitive to pressure than the commercially available valves tested. At high pressures, the Phase III Prototype could be operated at much lower values of torque than the commercially available valves. At low internal pressures, the Phase III Prototype required a higher torque to close the valve than the commercially available valves. This was an expected effect of the high spring force squeezing the upper seat against the ball. There was good agreement between the calculated theoretical torque to close the valve and the measured torque to close the valve for internal pressures below 1000 psi. However, at high values of internal pressure the measured torque value were about 30% higher than the calculated theoretical values. It is believed that the reason for this discrepancy was because not all of the pressure loading was being transferred to the stem bearings as was assumed in the theoretical analysis. Compression of the spring with increased pressure likely increased the friction between the ball and the seat, which was not considered in the theoretical analysis. A more complex finite-element analysis, that was beyond the scope of this project, would be needed to consider these effects.

The maximum torque to open the Phase III DSSV with pressure inside the valve and atmospheric pressure above the closed ball and seat are shown in Figure 40. Note that in this test, the Phase III Prototype achieved the best results at the highest differential pressures used in the tests. DSSV2 (Premium one-piece design) was a close second at 4000 psi and gave slightly better results than the Phase III Prototype at the lower differential pressures used in the tests. However, data taken in sandy mud and reported by Tarr et al [1998] showed a much poorer performance for DSSV2. Although not subjected to the sandy mud test protocol, the Phase III design would be expected to perform better because the ball and seat are held in firm contact. Thus, solids cannot pass between the ball and seat and enter the valve body. Note that most DSSVs should not be expected to open with less than 400 ft-lbf of torque at differential pressures above about 1000 psi.

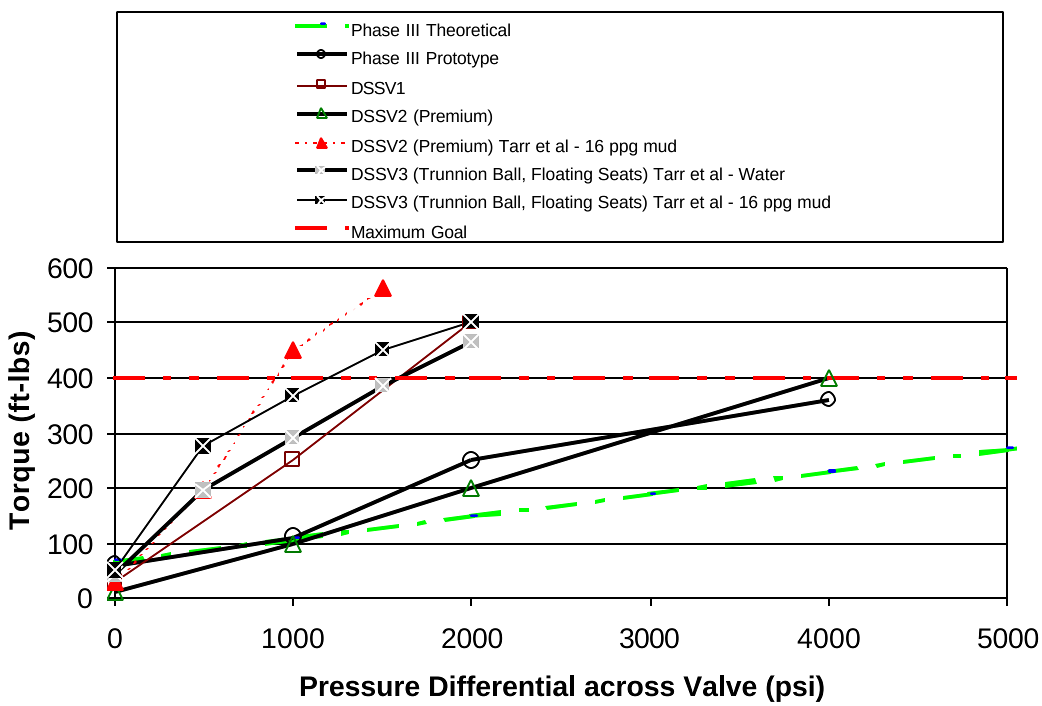

Figure 40: Maximum Torque to Open DSSV as a function of Internal Pressure with Atmospheric Pressure above DSSV

As in the previous tests, the observed maximum torque to operate the Phase III Prototype was close to the theoretical values calculated at low differential pressures, but about 30% higher than the theoretical values calculated at high pressure differentials. As discussed above, it is believed that the reason for this discrepancy was because not all of the pressure loading was being transferred to the stem bearings as was assumed in the theoretical analysis.

The maximum torque to open the Phase III Prototype DSSV with a higher pressure inside the valve than above, but with pressures above the valve higher than atmospheric is shown in Figure 41. This test was more severe than the test to produce Figure 40 because of the much higher internal pressures reached. The data points are grouped at differential pressure increments of 500 psi as described in the test procedure. The bias pressures shown are the pressures above the ball and upper seat. For example, at a pressure differential of 500 psi in Figure 40, the lowest point shown corresponds to the maximum torque to operate the DSSV with 500 psi above the valve and 1000 psi in the valve. The highest point shown corresponds to the maximum torque to operate the valve when the pressure above the ball and upper seat is 3500 psi and the pressure in the valve is 4000 psi. The theoretical design curve for differential pressure is for a zero pressure bias. Up to 4000-psi differential, the only data point that exceeded 400 ft-lbf was for a 2500-psi pressure bias.

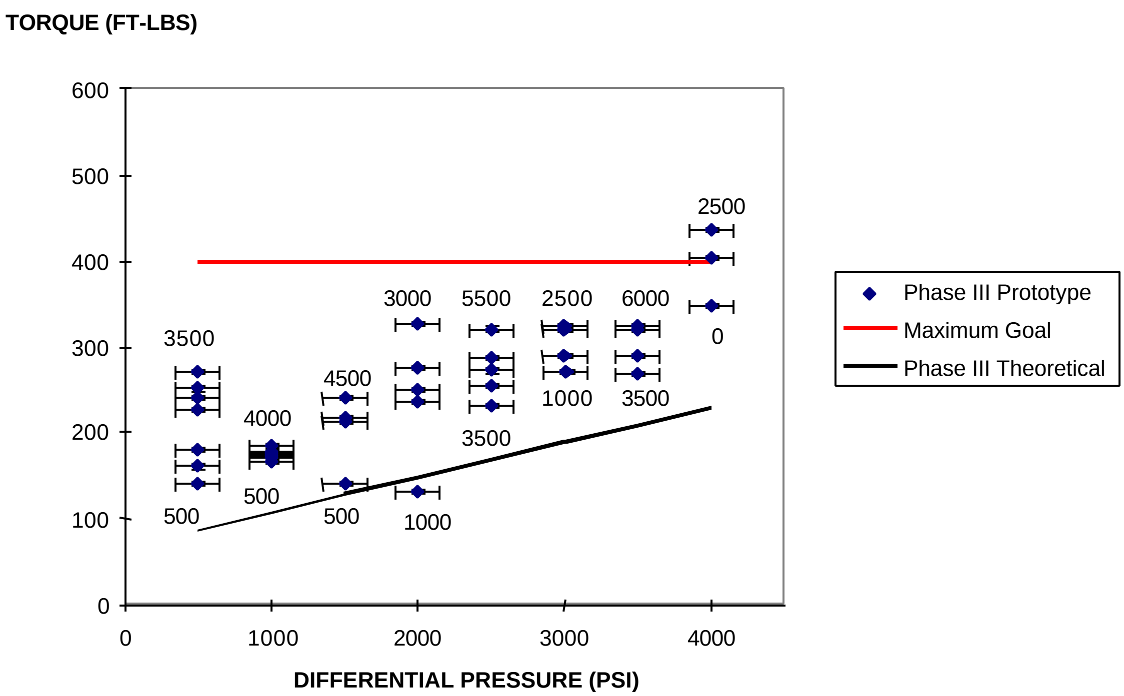

Figure 41: Maximum Torque required to Operate the Phase III Prototype as a function of Pressure Differential acting on the Ball and Upper Seat. (Various points shown at same differential pressure are for series of Bias Pressures held above Ball and Seat at 500-psi increments. The Numbers shown identify the Minimum and Maximum Pressures held above Ball and Seat.)

Figure 42 shows the maximum torque required for closing the Phase III Prototype when that valve has unweighted drilling mud flowing through it. Flow rates of 100 to 350 gallons per minute were used in the tests. The equalized pressure in the DSSV when it is initially open is 500 psi as set by the choke downstream of the valve (See Figure 33). When the DSSV is closed, the line pressure increases up to 2000 psi maximum before the automatic choke in parallel with the valve began operating to hold the pressure constant at 2000 psi. In addition, due to the quick closing of the DSSV under flow, a water hammer pressure may occur. This pressure depends upon the flow rate and is also given in Figure 42. It should be noted that this value is an upper bound as it is based upon instantaneous closing of the valve in a relatively short test loop. The valve was closed quickly (~ 1 sec) as recommended based on a previous study by BP-Amoco. This may have reduced the pressure build-up in the valve due to throttling. A commercial DSSV was also used in the flow tests to allow a comparison to be made to the performance of the Phase III Prototype.

The Design Goal for the Phase III Prototype was met over the full range of flow rates studied. However, the measured values of torque were much higher than those expected based on a theoretical analysis of the valve design. In addition, the commercial DSSV could be closed in this test at lower values of torque then the Phase III Prototype. It is believed that the unexpected results could have been due to a change in the internal condition of the Phase III Prototype. The prototype had been disassembled and reassembled several times prior to running this last series of tests. Erratic test results for commercially available DSSVs were also seen during other tests performed during the study.

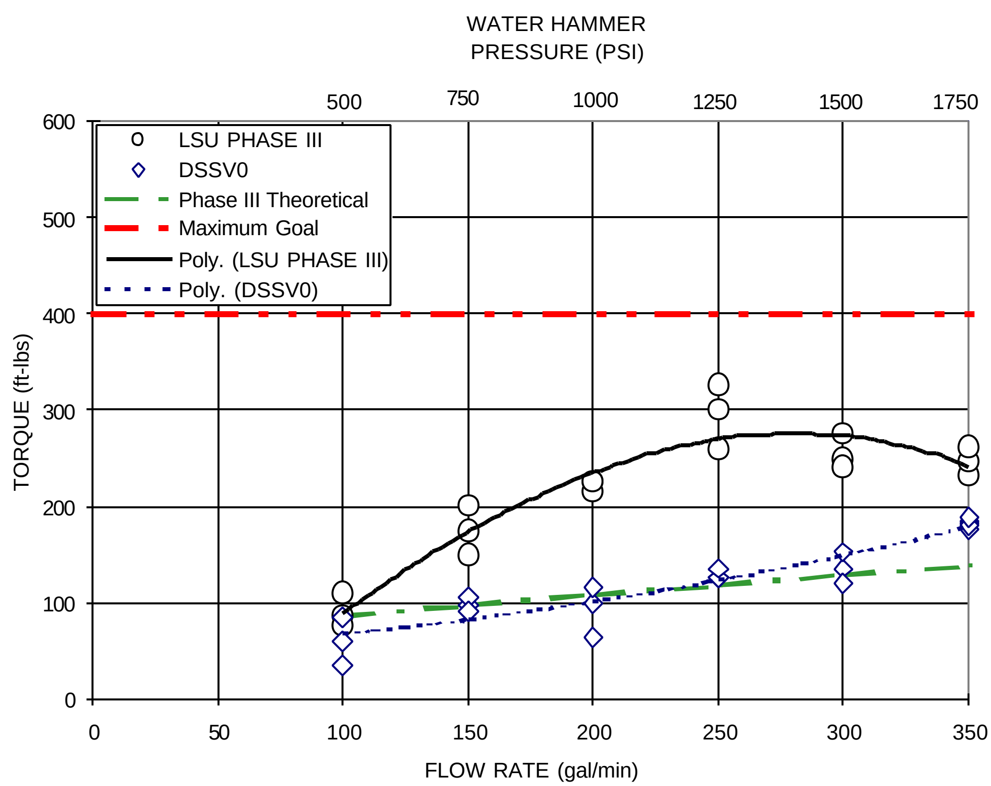

Figure 42: Maximum Torque required for Closing against Flow

Chapter

5

Development of DSSV Storage Stand

It was observed in the test program that the torque required for operating a DSSV changed with time and use. This is not surprising since the internal moving parts are exposed to the slurries periodically pumped through the valve and to weathering. In addition, the lack of lubrication of the mating parts between the stem and the ball more susceptible to binding if some parts are slightly out of tolerance. The purpose of this phase of the research program was to design a storage container that will preserve DSSVs in a like new working condition and allow for easy accessibility at the rig floor, thus resulting in improved reliability.

The design goals were determined by the student design team after talking with a number of different drilling operators. It was found that more than one size DSSV is often required on the rig floor at all times making it necessary for the stand to accommodate more than one DSSV. The storage container must hold the largest diameter of DSSVs listed in API Specification 7, which is 7.875 inches. In order to maintain the DSSVs in a like new condition, it is important that corrosion is prevented and that debris is not allowed to solidify on the moving components. In order to account for the quick retrieval of the DSSV, in less than 10 seconds, there must be a fast and easy way of removing it from storage. Since the valves are also used off shore, the container and its components must be environmentally safe. Due to the corrosive environment offshore the stand must be made to resist all conditions and maintain its integrity for a design life of five years. In addition, it was believed that industry would accept the new technology more readily if the cost of the unit were below $4,000.

Design

The design proposed is a storage structure, containing four storage cylinders. Each storage cylinder contains enough oil for a DSSV stored in the cylinder to be immersed in oil. A mechanism to quickly lift the DSSV out of the oil bath is driven by a hydraulic pump. In the prototype, the pump is manually actuated to supply the required fluid and pressure to lift each valve from its cylinder. A single fluid reservoir located centrally, supplies oil to the pump. Machine drawings for the prototype storage structure are given in Appendix B.

The storage stand consists of two basic components: the storage structure and the hydraulic system. The storage structure includes the all the components that act as containment, including a side plates, bottom plate, lid, and maintenance panel. The hydraulic system consists of all parts used to store the valves in the bath as well as those components which raise or lower the valves. The basic

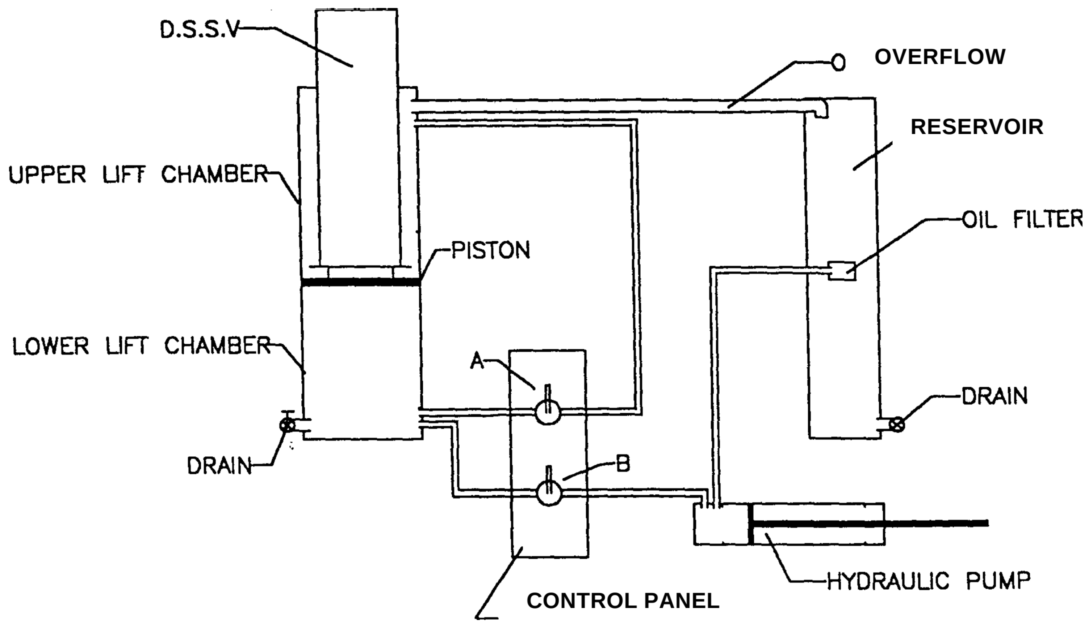

Figure 43 Schematic: for Hydraulic System for Proposed New DSSV Storage Stand

components of the hydraulic system include: the storage cylinders, pistons, pump, fluid reservoir, control valves located on a side control panel, and the tubing with the necessary fittings. A schematic of the hydraulic system is shown in Figure 43.

Hydraulic Pump

Because most prefabricated pumps are hand actuated and can be purchased at a much lower cost than a customized pump the design team decided to purchase a hand actuated hydraulic pump rather than fabricate a customized pump. In order to find the appropriate pump needed for the hydraulic system all of the forces needed to be accounted for. The first step was to determine the resistive forces on the piston. These forces include the weight of the safety valve, the weight of the piston, the weight of the fluid above the piston and, the resistance caused by the friction on the o-ring. The maximum weight of a DSSV was found to be 250 1bf. The piston that was designed weighs 29 lbs. The maximum that the fluid would weigh is when the piston is at its lowest position was found to be 59 lbs. The frictional force between the O-ring and cylinder wall was calculated to be about 20 lbf. Thus the total weight that needs to be overcome was found to be about 360 1bf, which corresponds to a pressure of five psi in the lower lift chamber. To allow for head losses, unexpected frictional forces in the system, and ease of pump operation, a pump that could handle a pressure of 25 psi was chosen.

The pump that was selected is a "Fill-Rite" hand pump intended for pumping fuel from 55-gallon drums. The Fill-Rite is easily attachable to any threaded mounting and is small enough to fit inside the storage structure between the storage cylinders. The fluid displacement of the pump is 35 cubic inches per stroke, which results in a 0.5-in lift of the valves per stroke. Only 10 strokes are required to raise the valves the desired 5 inches. Each stroke of the pump took less than 1 second to execute allowing the minimum lift time requirement to be met. Also, the Fill-Rite operates at maximum pressure of 25 psi.

A check valve was placed at the inlet of the control panel to relieve any pressure that the pump would experience while the valves were in their lifted positions. The check valve holds the pressure of the fluid and valve weight exerted when the valves are completely lifted. The maximum pressure the check valve can withstand is 250 psi making it more that sufficient to handle the pressure exerted. This was a precautionary measure implemented to improve the life and reliability of the hydraulic pump.

Hydraulic Cylinders

The hydraulic cylinders were designed to accommodate all sizes of DSSVs listed in API spec 7. The maximum height of a safety valve was found to be 2 inches. The maximum size that the cylinders can hold is 30 inches. The largest outer diameter of a safety valve was determined to be 7.875 inches so, the cylinders were made to hold valves up to 9.5 inches in diameter. An overflow pipe was welded on to the top of the hydraulic cylinders. One-inch pipe was welded below the over flow to allow for the flow from the lower chamber to the upper chamber. Also, one-inch pipe was welded at the bottom of the cylinder to allow the fluid to flow out of the lower chamber. To attach the cylinders to the bottom plate of the outer storage container, three steel feet were welded to the bottom of each of the cylinders.

For a piston cylinder devise to work properly there are many factors that must be considered during the design. A surface finish between 8 to 16 micro-inches was selected to minimize frictional forces. A Nitrile O-ring material was selected to be compatible with the oil used in the hydraulic system and oil bath. A gasket width of .210-in was selected for the pump and .275-in for the lift. Since the pressure requirements were very low for a piston, a large tolerance could be used. This helped provide a low-cost construction process.

One of the concerns of the proposed design was the possibility of the DSSV tipping over during removal or resting on the inside wall of the storage cylinder. To prevent lateral movement of the DSSV, a threaded shaft was inserted into the piston. The maximum allowable tilt angle with the rod inserted is 7 degrees or 0.75 inches laterally. The rod was designed against bending and shear failure. The rod is 314 stainless steel and the dimensions are 10 inches long with a minor diameter of 7/8 inch on the threaded part located at the base of the piston. Using the lateral load of 250 pounds and the dimensions and material properties of the rod, it was determined that the rod had a factor of safety of 1.3 against bending failure and 240 against shear failure.

Fluid Reservoir

The reservoir was built to store extra fluid that gets displaced when safety valves are inserted and to collect drilling fluid solids that settles out of the oil. The reservoir is made of 10-inch OD steel pipe with four two-inch openings for the overflow pipes that lead to each cylinder. There is also a one-inch drain at the bottom to allow for cleaning. The reservoir was attached to the pump intake using one-inch pipe located 10 inches from the bottom of the reservoir. It was put near the center of the reservoir so the only oil would get sucked out to the pump in the event water and solids began collecting in the bottom of the reservoir. Three steel feet were welded to the bottom of the reservoir to attach it to the bottom plate.

Hydraulic Fluid

The hydraulic fluid must bathe and lubricate internal valve parts and create a non-corrosive environment ensuring the valve is in a “like-new” condition if needed. Next, the fluid must be environmentally friendly, in order to be compatible with offshore use. The hydraulic fluid must also have the versatility to remain operable under the range of temperatures experienced and maintain a reasonable viscosity to allow for easy pumping. The viscosity of the hydraulic fluid is not a real factor in the performance of the system due to the low pressures and velocities. Additionally, the head loss the hydraulic fluid experiences while being pumped through the system is negligible due to the large diameters of tubing and valves.

The fluid selected is called EAL-224H Hydraulic Fluid, and is produced by Mobil. The EAL-224H has excellent corrosion and rust resistance creating an excellent environment for the valves. In addition, 224H is recommended for use between 0-180 degrees Fahrenheit. This temperature was felt to be suitable for rig floor conditions. The 224H is not only non-toxic, but is also bio-degradable. It is recommended in marine applications.

Control Panel

Since there is only one pump for the four hydraulic lifts, a control panel was needed to direct the fluid flow that will determine which of the DSSVs is accessed or stored. To do this, hydraulic lines were run to a control panel containing ball valves. In order to lift a DSSV, the ball valve for that cylinder is opened and the drain valve is closed. The valve positions are reversed when lowering a valve into the storage position.

Storage Structure

Since the DSSV Phase III valve is 27 inches long, and most commercial valves are no longer than 32 inches in length, a height for the storage stand of 3 feet was selected. In order to house a maximum of four valves with an outer diameter of 6 to 8 inches and have a small footprint on the rig floor area, a 3-foot cubic stand was chosen.

The materials considered for the stand construction included stainless steel, aluminum, and carbon steel. Aluminum would have been a good choice as far as weight and moderate strength, but failed in the area of corrosion resistance. Stainless steel was strongly considered because it is non-corrosive, has high strength, and is very durable. However, manufacturing costs would have caused the design goal to be exceeded. After consulting with several oil-field service personnel and steel distributors, it was discovered that the vast majority of containers used offshore are made from regular carbon steel that has been sandblasted and painted. The design chose carbon steel for construction of the prototype DSSV Storage Stand.

The components of the storage structure considered were the bottom plate, side plates, back wall, the lid, and maintenance panel. Due to the heavy weight of the hydraulic system, the bottom plate needed to be thicker than the side and back plates. After performing deflection calculations, one-quarter inch carbon steel was selected for the bottom plate. The remaining panels were constructed from 3/16-inch plates.

In order to make the stand portable, consideration was given to placing a set of locking wheels on the bottom of the stand. However, after consulting with industry personnel familiar with rig floor conditions, this idea was discarded in favor of the use of pad eyes that would allow the unit to be moved with an air hoist. Originally only two pad eyes were considered with one on each side panel. After the hydraulic system was designed, it was decided on to use four pad eyes (two on each side panel) to ensure that the stand could not tip while being moved with the air hoist. In addition to pad eyes for moving, there should be some means of moving the stand with a forklift. To accomplish this, the stand was elevated using two half-moon rubber skids. Lid accessories include the hinges, handle, latches and rubber seals. These accessories have all been designed previously by an oil-field service fabricator, and are standard for this type of application.

Figure 44: DSSV and Inside BOP on Rig Floor Ready to Stab on Drillstring if Needed

Assembly and Testing

The DSSV Storage Stand was assembled and tested at the LSU Petroleum Engineering Research Well Facility. It was tested and found to function as designed. All design goals were met.

Shown in Figure 44 is the top portion of a DSSV located on a rig floor among other equipment. It is in a ready-to-stab configuration with a lifting sub screwed into the top of the valve. This is the arrangement commonly used by the drilling industry today. Shown in Figure 45 is the prototype of the proposed DSSV Stand.

Figure 45: Proposed DSSV Storage Stand with Control Valves on Side.

Chapter

6

Conclusions and Recommendations

This study was undertaken by LSU under the sponsorship of the MMS to improve safety and reduce risk of blowouts through the drill string. MMS has been mandated by congress to insure that the best available and safest technology is utilized by industry in developing our natural resources on the outer continental shelf.

A robust Drill String Safety Valves DSSV is difficult to design because the DSSV must fit in the drillstring and be full-opening enough to allow wireline work to be done through the valve. The valves have been shown to have one of the highest failure rates of the blowout control equipment routinely used on drilling rigs. Blowouts have occurred on a number of occasions when a DSSV could not be closed to stop flow from the well up the drillstring. Other failure modes have also been reported that have not caused blowouts but have made well control difficult to achieve.

Most DSSVs that are commercially available are primarily quarter-turn ball valves in which the ball floats between fixed seats and depend on well pressure to force the closed ball against a seat to form a seal. This design minimizes the crushing force placed on the ball and distributes stresses through the seats over a wide area. However, interference between parts can result during ball rotation if dimensional tolerances are not met. A relatively new DSSV design has recently become available that utilizes a trunnion mounted fixed ball and floating seats. This design reduces the risk of interference between parts during ball rotation but results in more concentrated stresses in the valve stem and ball.

In this project, DSSV failure modes were studied in detail and procedures for reducing the risk of blowouts due to a DSSV failure were identified. A novel DSSV design was explored that used a trunnion-mounted ball and spring loaded seats was constructed and tested by an interdisciplinary research team composed of Petroleum Engineering and Mechanical Engineering seniors and faculty. The novel design results in operating torque being less affected by internal pressure but also results in more concentrated stresses in the valve stem. The test program indicated that the design objectives of the new design were achieved. The test program also revealed that the torque required to operate a DSSV type ball valve can change with use (especially after being operated in a high torque condition), with internal exposure to drilling slurries, and with non-use and exposure to the elements. In addition to a new valve design, a new DSSV storage stand was designed and tested that could maintain a DSSV on the rig floor in a “like-new” condition.

As a result of the work done in this study, the following conclusions and recommendations were made:

Conclusions

1. Failure of a DSSV to close against a high flow rate or under high pressure has resulted in a number of blowouts.
2. Test results obtained and this study and results published in the literature show that the torque required for operating a DSSV increase with internal valve pressure and can exceed 400 ft-lbf at pressures as low as 1000 psi.
3. The new prototype design developed in this work shows promise of reducing the operating torque at high pressures.
4. High pressure on a DSSV valve before closing can occur when there is a significant length of piping and or restrictions downstream of the DSSV.
5. The link between the ball and stem in a DSSV is generally a two piece design to facilitate disassembly and rebuilding the internal components of the valve. This design is susceptible to binding during rotation of the ball under flow or pressure if dimensional tolerances are not closely observed during manufacturing and assembly.
6. Low cost DSSVs utilize a threaded two-piece body construction to allow the valve to be disassembled and rebuilt in the field. The torque used to remake the body threads can affect critical dimensional tolerances of the component placement in the valve body. Applying excessive torque across the valve body when making up the valve in the drillstring can bind the ball between the seats.
7. Redundant barriers to blowouts are needed for the drillstring during tripping operations similar to the use of redundant barriers provided in the blowout preventer stack for stopping a flow up the annulus.
8. Experience has shown that maintaining multiple DSSVs available on the rig-floor as stabbing valves does not always prevent blowouts.

Recommendations

1. The DSSV intended for use as a stabbing valve to stop flow through the drillstring during tripping operations should not be used in the drillstring for other operations. The stabbing valve should be maintained in a “like-new“ condition and used only during periodic pressure testing with fresh water.

2. Operators and/or drilling contractors should check threads, valve wrench, and lift sub on the stabbing valve and actuate the stabbing valve close and open each tour.

3. Operators and/or drilling contractors should use a drillstring float whenever practical to provide redundant protection against a high-rate flow through the drill-string during tripping operations.

4. When floats are not used, shear rams are recommended for redundant protection against blowouts through the drillstring during tripping operations.

5. Drill String Safety Valves should not be the only means for stopping flow from the drillstring at the surface when reverse circulating the well during completion operations. Flow should be routed through hydraulically operated valves and a choke manifold.

6. Drill string safety valves should not be the only means for stopping flow through the drill string when significant piping and flow restrictions are present above the valve.

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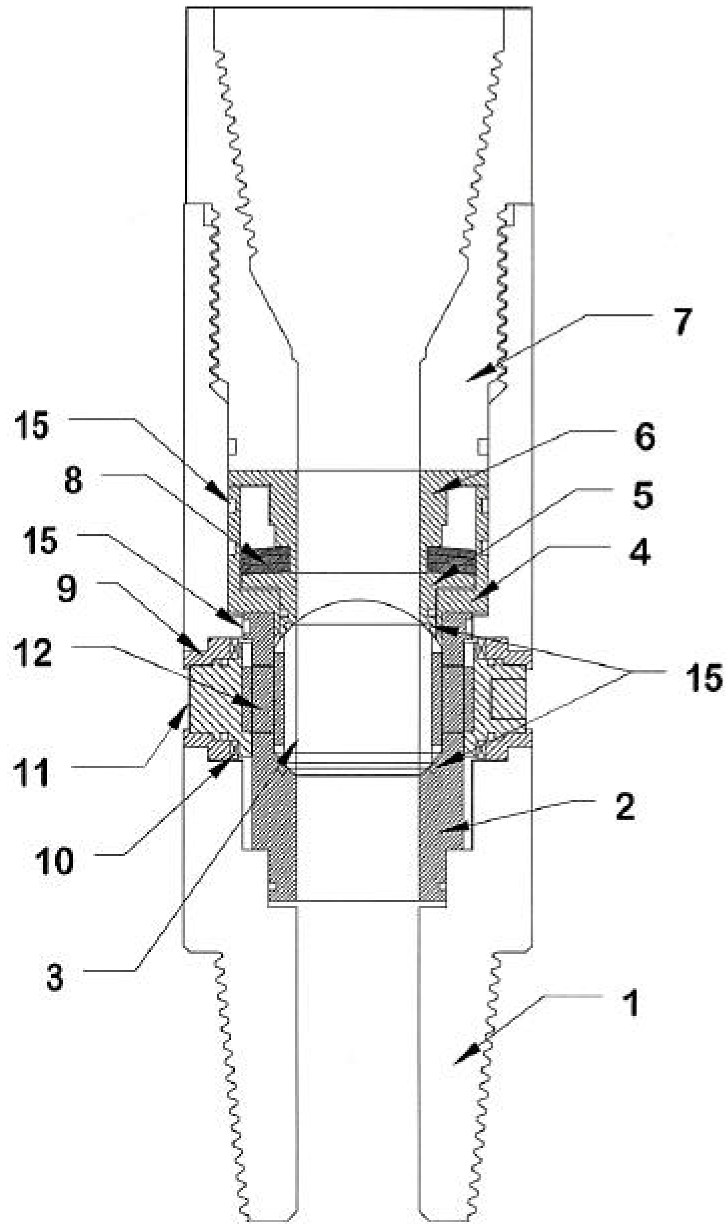
Appendix A

Phase III Prototype DSSV

This appendix contains the machine drawings for the Phase III prototype DSSV.

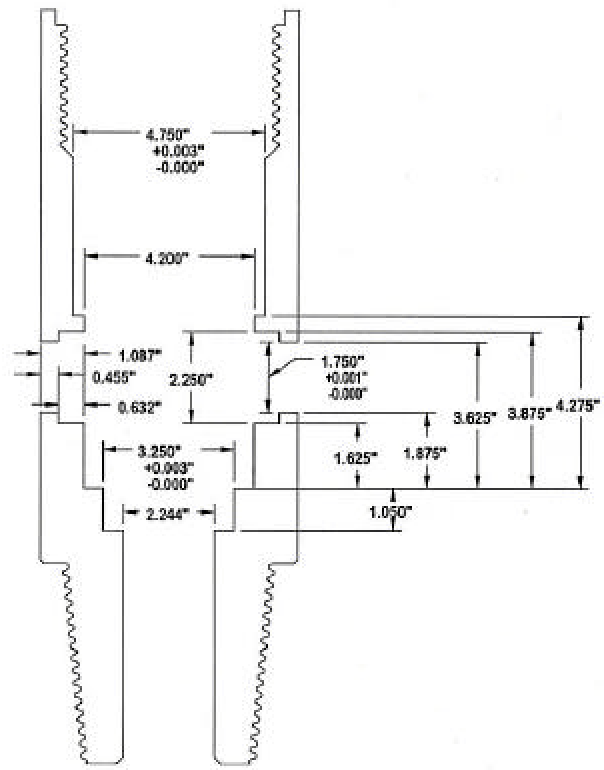


PART #	PART NAME
1	CASING
2	LOWER CANISTER
3	BALL
4	UPPER CANISTER
5	UPPER SEAT
6	UPPER CANISTER CAP
7	LOCKING CAP
8	WASHER SPRINGS
9	BEARING BORE
10	THRUST BEARING
11	STEM
12	STEM LINK
13	VITON O'RING
14	TEFLON O'RING
15	SNAP RING

DRILL STRING SAFETY VALVE

TEAM 2 : BRAD SMITH, MARK CASEMORE, ELLIOT COLEMAN, THOMAS CORE

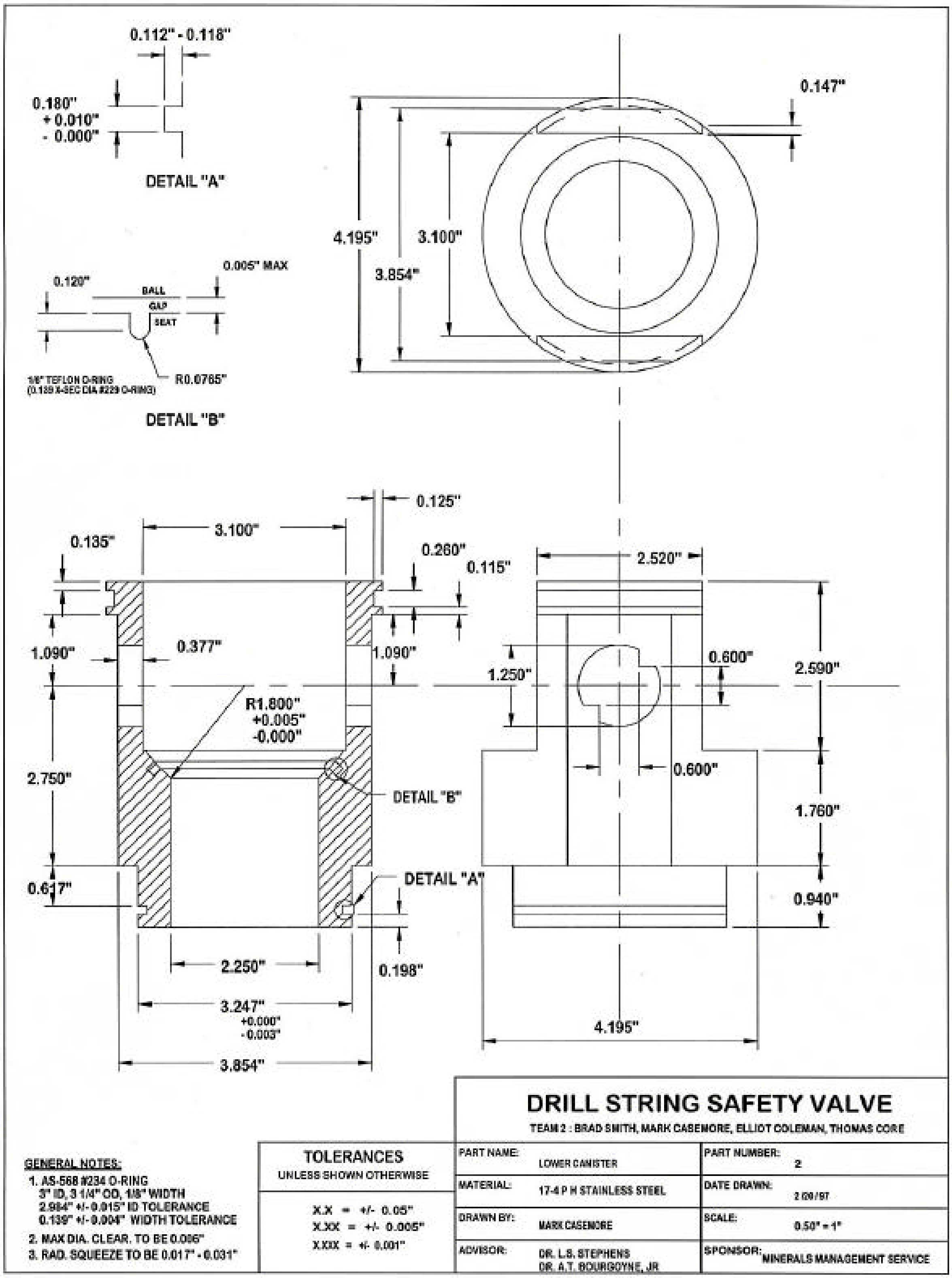
TOLERANCES UNLESS SHOWN OTHERWISE				
X.X = +/- 0.05" X.XX = +/- 0.005" X.XXX = +/- 0.005"	PART NAME:	NUMBERED ASSEMBLY	PART NUMBER:	0
	MATERIAL:	VARIOUS	DATE DRAWN:	2 / 27 / 97
	DRAWN BY:	MARK CASEMORE BRAD SMITH	SCALE:	0.333" = 1"
	ADVISOR:	DR. L.S. STEPHENS DR. A.T. BOURGOYNE	SPONSOR:	MINERALS MANAGEMENT SERVICE

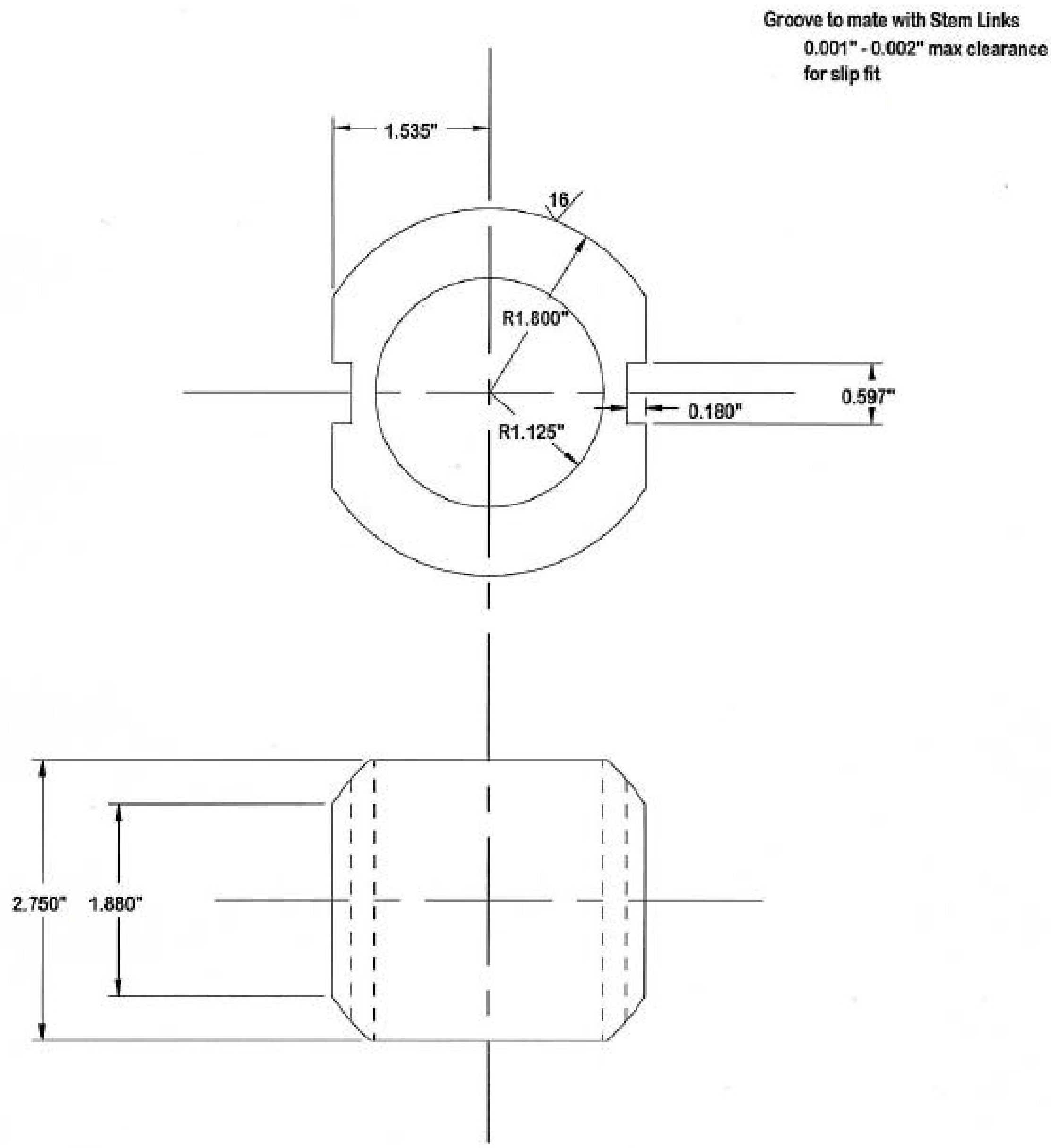


DRILL STRING SAFETY VALVE

TEAM 2 : BRAD SMITH, MARK CASEMORE, ELLIOT COLEMAN, THOMAS CORE

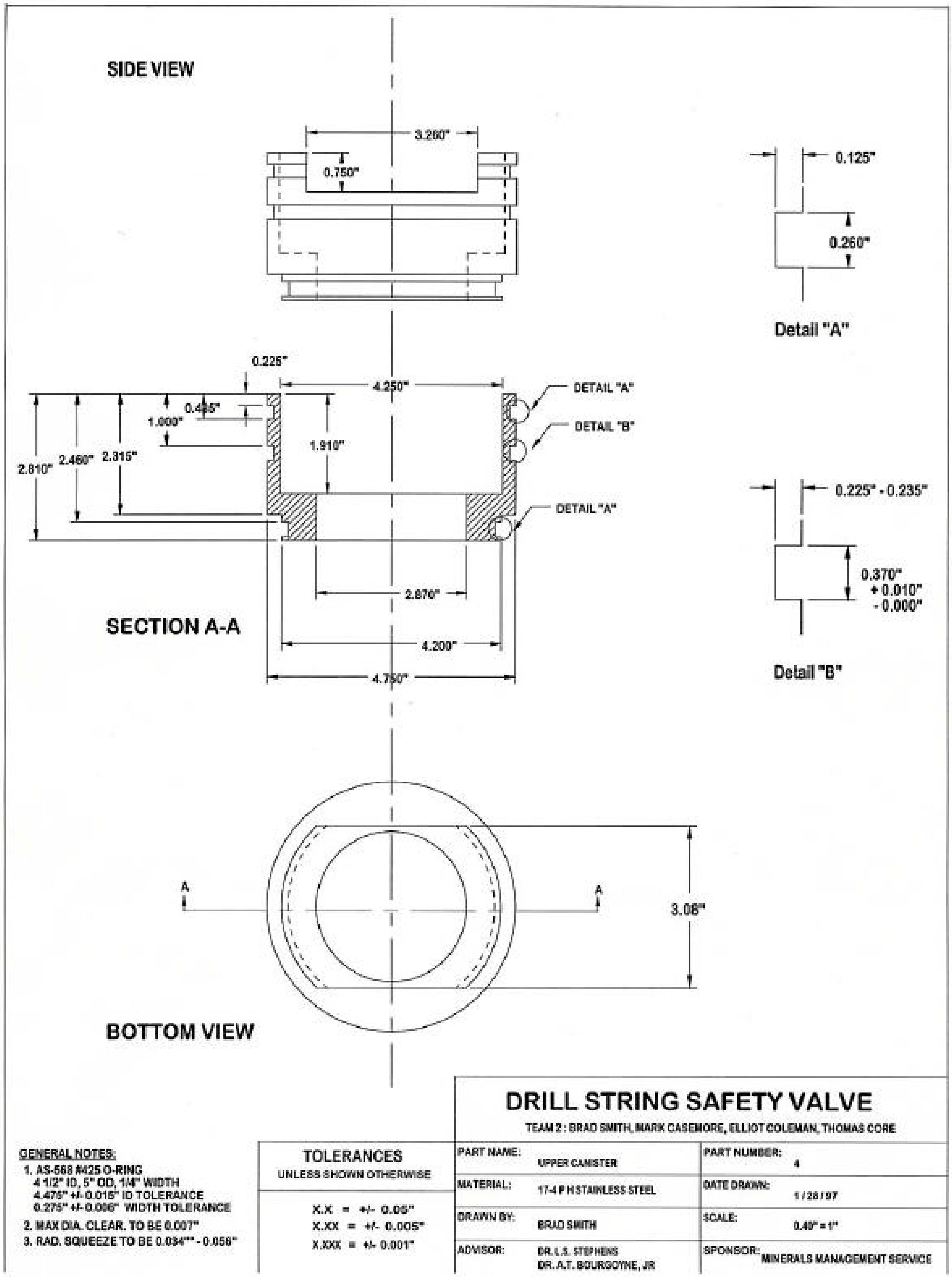
TOLERANCES UNLESS SHOWN OTHERWISE				
X.X = +/- 0.05" X.XX = +/- 0.005" X.XXX = +/- 0.001"	PART NAME:	CASING	PART NUMBER:	1
	MATERIAL:	AISI 4140	DATE DRAWN:	2/27/97
	DRAWN BY:	BRAD SMITH	SCALE:	0.25" = 1"
	ADVISOR:	DR. L.S. STEPHENS DR. A.T. BOURGOYNE, JR	SPONSOR:	MINERALS MANAGEMENT SERVICE

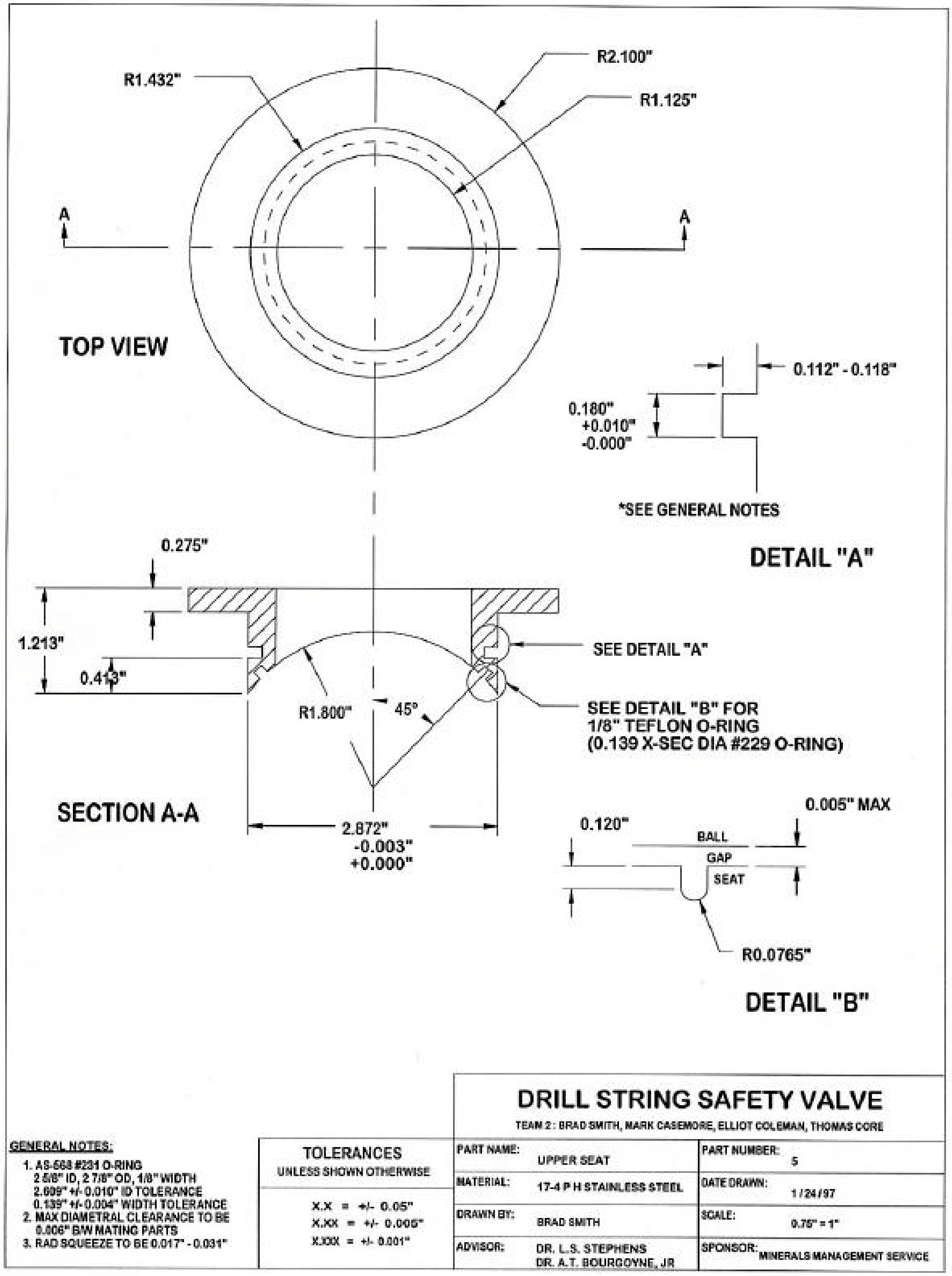

0.112" - 0.118"
0.180"
+ 0.010"
- 0.000"
DETAIL "A"
0.147"
4.195"
3.100"
3.854"
0.005" MAX
0.120"
BALL
GAP
SEAT
R0.0765"
DETAIL "B"
0.125"
3.100"
0.135"
0.260"
0.115"
2.520"
1.090"
0.377"
1.090"
1.250"
0.600"
2.590"
R1.800"
+0.005"
-0.000"
0.600"
2.750"
DETAIL "B"
1.760"
0.617"
DETAIL "A"
0.940"
2.250"
0.198"
3.247"
+0.000"
-0.003"
4.195"
3.854"
DRILL STRING SAFETY VALVE
TEAM 2 : BRAD SMITH, MARK CASEMORE, ELLIOT COLEMAN, THOMAS CORE
GENERAL NOTES:
1. AS-568 #234 O-RING
3" ID, 3 1/4" OD, 1/8" WIDTH
2.984" +/- 0.015" ID TOLERANCE
0.139" +/- 0.004" WIDTH TOLERANCE
2. MAX DIA. CLEAR. TO BE 0.006"
3. RAD. SQUEEZE TO BE 0.017" - 0.031"
TOLERANCES
UNLESS SHOWN OTHERWISE
X.X = +/- 0.05"
X.XX = +/- 0.005"
X.XXX = +/- 0.001"
PART NAME: LOWER CANISTER
MATERIAL: 17-4 P H STAINLESS STEEL
DRAWN BY: MARK CASEMORE
ADVISOR: DR. L.S. STEPHENS
DR. A.T. BOURGOYNE, JR
PART NUMBER: 2
DATE DRAWN:
SCALE: 0.50" = 1"
SPONSOR: MINERALS MANAGEMENT SERVICE



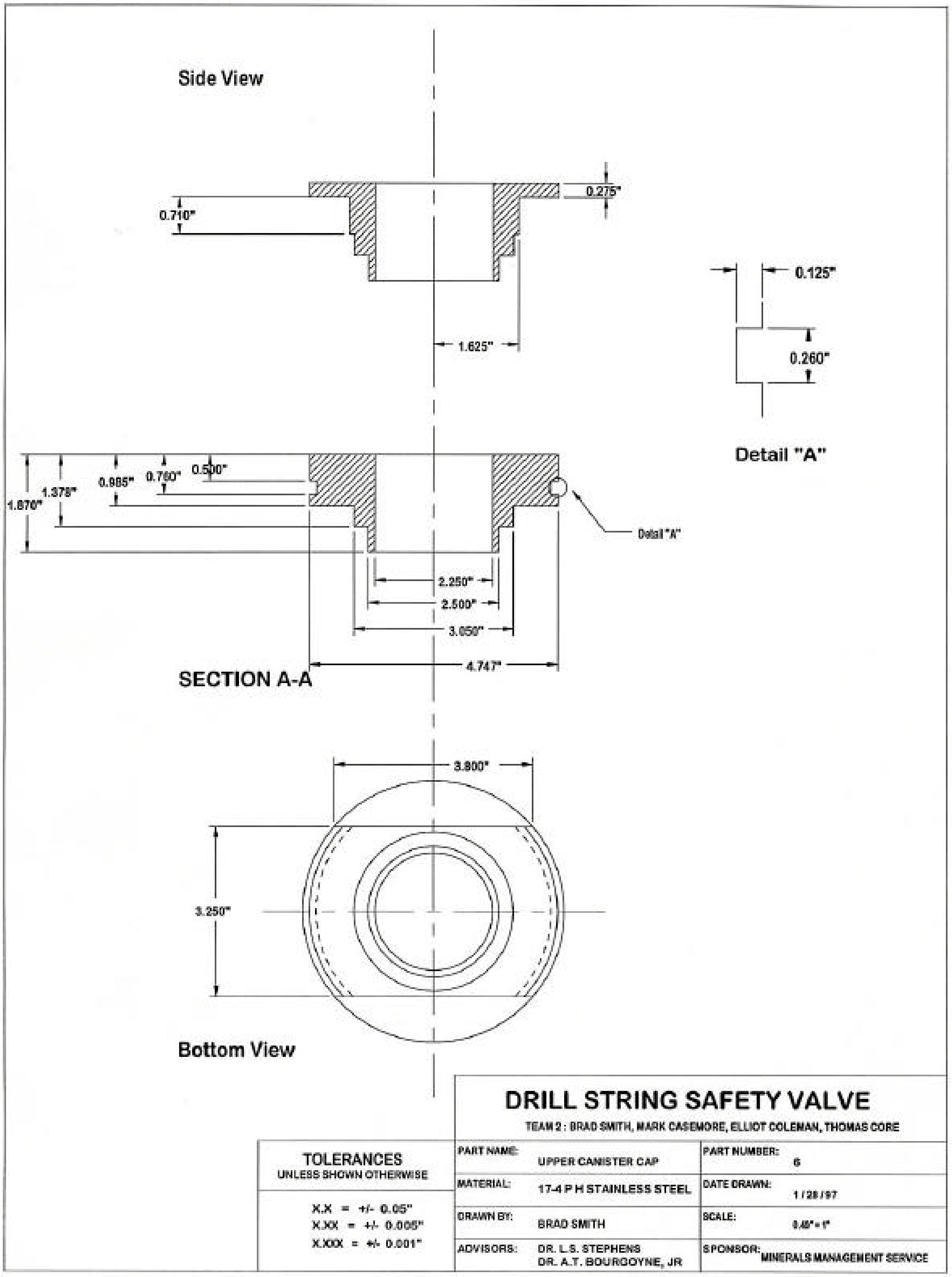
DRILL STRING SAFETY VALVE	
TEAM 2 : BRAD SMITH, MARK CASEMORE, ELLIOT COLEMAN, THOMAS CORE	
PART NAME: BALL	PART NUMBER: 3
MATERIAL: 17-4 P H STAINLESS STEEL	DATE DRAWN: 11 / 29 / 96
DRAWN BY: BRAD SMITH	SCALE: 0.75" = 1"
ADVISOR: DR. L.S. STEPHENS DR. A.T. BOURGOYNE, JR	SPONSOR: MINERALS MANAGEMENT SERVICE

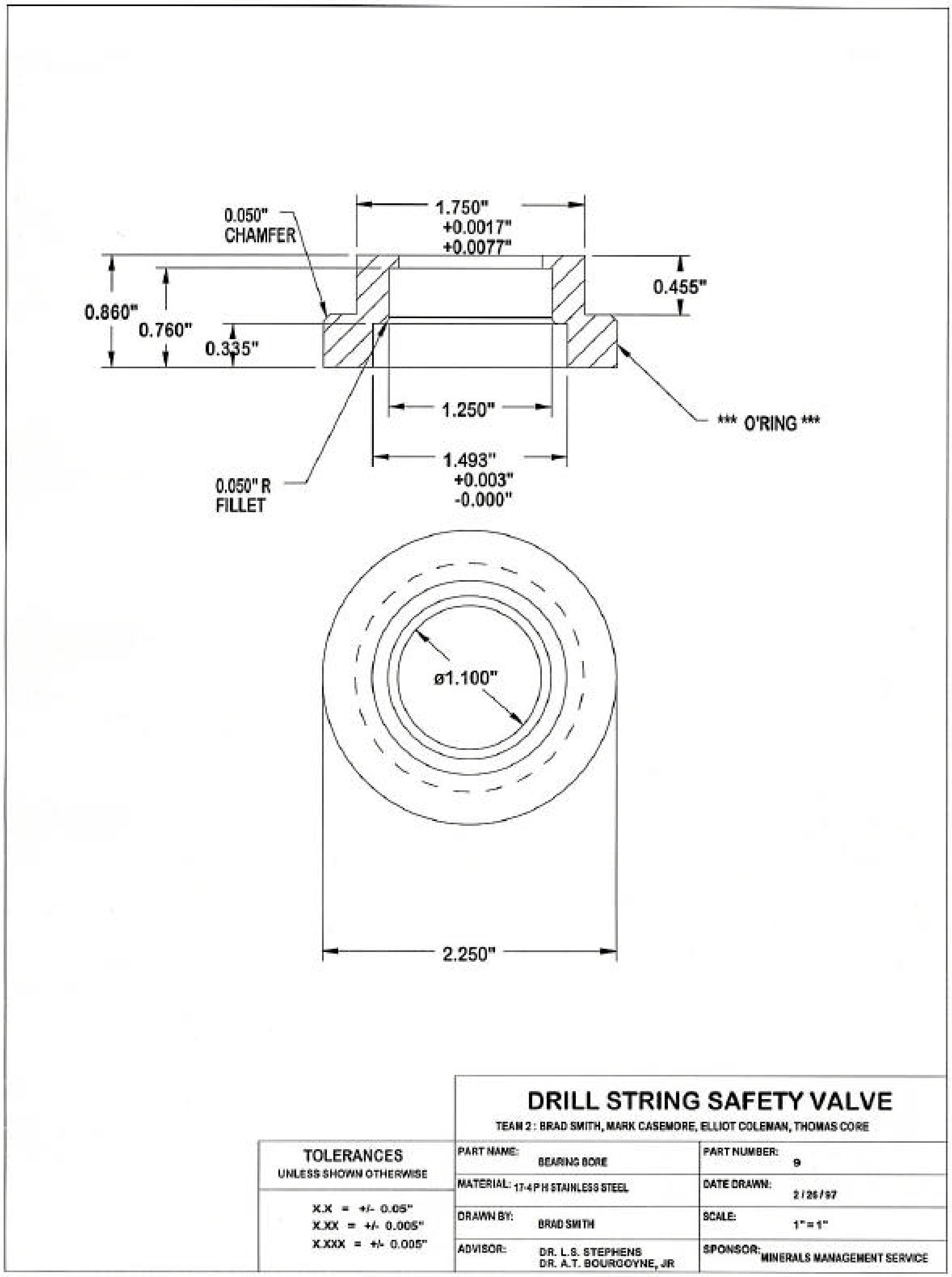
TOLERANCES UNLESS SHOWN OTHERWISE
X.X = +/- 0.05" X.XX = +/- 0.005" X.XXX = +/- 0.001"

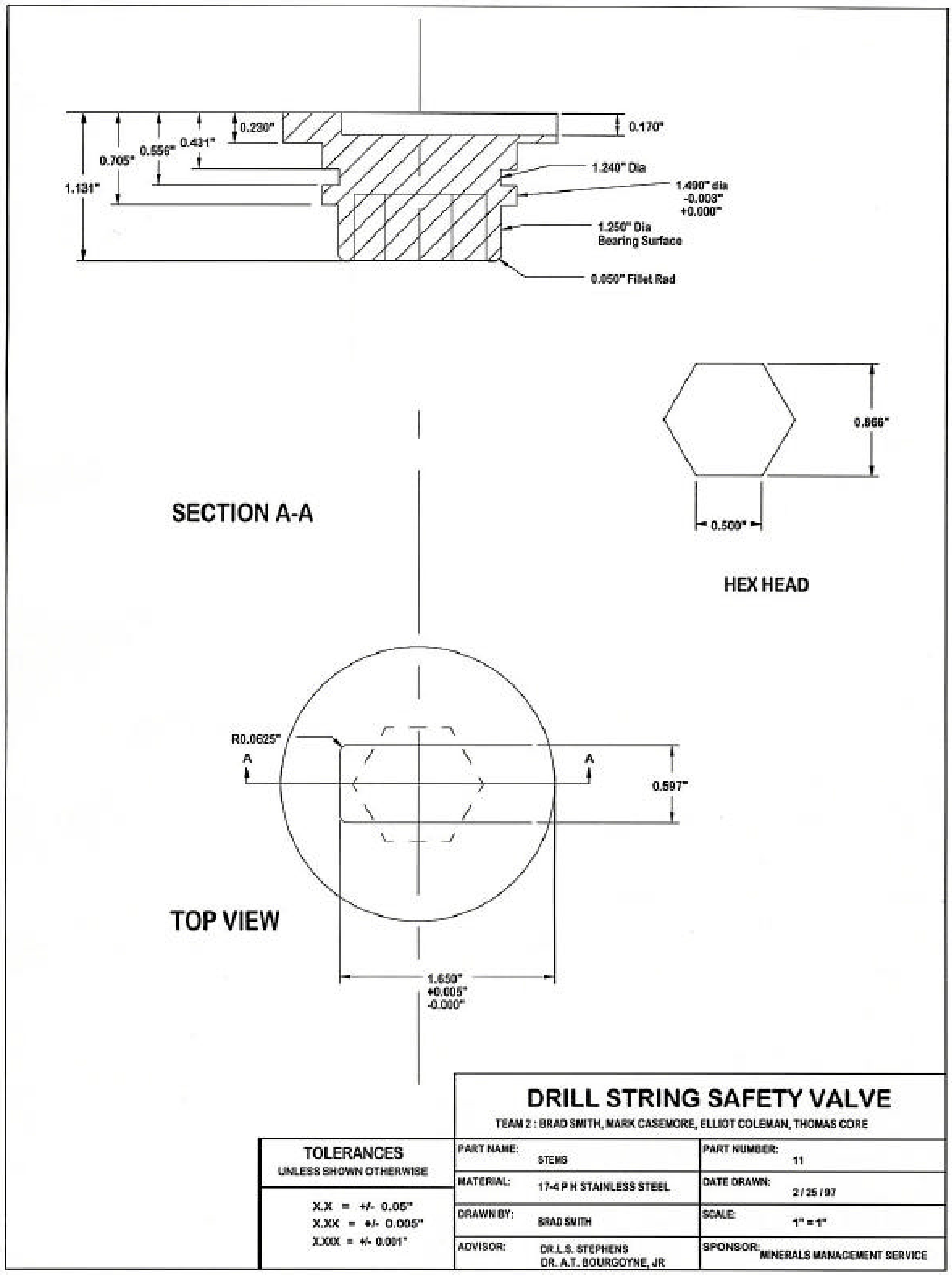

SIDE VIEW
3.260"
0.750"
0.125"
0.260"
Detail "A"
0.225"
4.250"
0.485"
1.000"
1.910"
2.810"
2.460"
2.315"
DETAIL "A"
DETAIL "B"
DETAIL "A"
0.225" - 0.235"
0.370"
+ 0.010"
- 0.000"
2.870"
SECTION A-A
4.200"
4.750"
Detail "B"
A
A
3.06"
BOTTOM VIEW
DRILL STRING SAFETY VALVE
TEAM 2 : BRAD SMITH, MARK CASEMORE, ELLIOT COLEMAN, THOMAS CORE
GENERAL NOTES:
1. AS-568 #425 O-RING
4 1/2" ID, 5" OD, 1/4" WIDTH
4.475" +/- 0.015" ID TOLERANCE
0.275" +/- 0.006" WIDTH TOLERANCE
2. MAX DIA. CLEAR. TO BE 0.007"
3. RAD. SQUEEZE TO BE 0.034"" - 0.056"
TOLERANCES
UNLESS SHOWN OTHERWISE
X.X = +/- 0.05"
X.XX = +/- 0.005"
X.XXX = +/- 0.001"
PART NAME: UPPER CANISTER
PART NUMBER: 4
MATERIAL: 17-4 P H STAINLESS STEEL
DATE DRAWN: 1 / 28 / 97
DRAWN BY: BRAD SMITH
SCALE: 0.40" = 1"
ADVISOR: DR. L.S. STEPHENS
DR. A.T. BOURGOYNE, JR
SPONSOR: MINERALS MANAGEMENT SERVICE



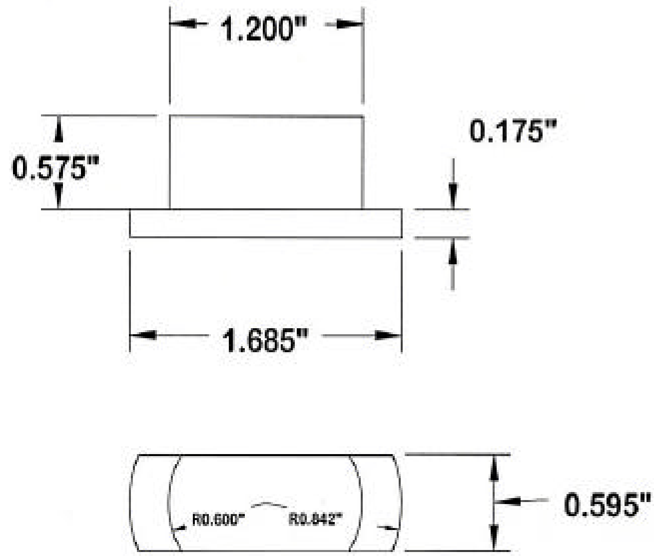
R1.432"
R2.100"
R1.125"
A
A
TOP VIEW
0.112" - 0.118"
0.180"
+0.010"
-0.000"
*SEE GENERAL NOTES
DETAIL "A"
0.275"
1.213"
0.413"
SEE DETAIL "A"
R1.800"
45°
SEE DETAIL "B" FOR
1/8" TEFLON O-RING
(0.139 X-SEC DIA #229 O-RING)
SECTION A-A
2.872"
-0.003"
+0.000"
0.005" MAX
0.120"
BALL
GAP
SEAT
R0.0765"
DETAIL "B"
DRILL STRING SAFETY VALVE
TEAM 2 : BRAD SMITH, MARK CASEMORE, ELLIOT COLEMAN, THOMAS CORE
GENERAL NOTES:
1. AS-568 #231 O-RING
2 5/8" ID, 2 7/8" OD, 1/8" WIDTH
2.609" +/- 0.010" ID TOLERANCE
0.139" +/- 0.004" WIDTH TOLERANCE
2. MAX DIAMETRAL CLEARANCE TO BE
0.006" B/W MATING PARTS
3. RAD SQUEEZE TO BE 0.017" - 0.031"
TOLERANCES
UNLESS SHOWN OTHERWISE
X.X = +/- 0.05"
X.XX = +/- 0.005"
X.XXX = +/- 0.001"
PART NAME: UPPER SEAT
PART NUMBER: 5
MATERIAL: 17-4 P H STAINLESS STEEL
DATE DRAWN: 1/24/97
DRAWN BY: BRAD SMITH
SCALE: 0.75" = 1"
ADVISOR: DR. L.S. STEPHENS
DR. A.T. BOURGOYNE, JR
SPONSOR: MINERALS MANAGEMENT SERVICE


Side View
0.275"
0.710"
0.125"
0.260"
1.625"
Detail "A"
1.870"
1.378"
0.985"
0.760"
0.500"
Detail "A"
2.250"
2.500"
3.050"
4.747"
SECTION A-A
3.800"
3.250"
Bottom View
DRILL STRING SAFETY VALVE
TEAM 2 : BRAD SMITH, MARK CASEMORE, ELLIOT COLEMAN, THOMAS CORE
TOLERANCES
UNLESS SHOWN OTHERWISE
X.X = +/- 0.05"
X.XX = +/- 0.005"
X.XXX = +/- 0.001"
PART NAME: UPPER CANISTER CAP
PART NUMBER: 6
MATERIAL: 17-4 P H STAINLESS STEEL
DATE DRAWN: 1 / 28 / 97
DRAWN BY: BRAD SMITH
SCALE:
ADVISORS: DR. L.S. STEPHENS
DR. A.T. BOURGOYNE, JR
SPONSOR: MINERALS MANAGEMENT SERVICE


0.050"
CHAMFER
1.750"
+0.0017"
+0.0077"
0.455"
0.860"
0.760"
0.335"
1.250"
*** O'RING ***
1.493"
+0.003"
-0.000"
0.050" R
FILLET
ø1.100"
2.250"
DRILL STRING SAFETY VALVE
TEAM 2 : BRAD SMITH, MARK CASEMORE, ELLIOT COLEMAN, THOMAS CORE
TOLERANCES
UNLESS SHOWN OTHERWISE
X.X = +/- 0.05"
X.XX = +/- 0.005"
X.XXX = +/- 0.005"
PART NAME: BEARING BORE
MATERIAL: 17-4 P H STAINLESS STEEL
DRAWN BY: BRAD SMITH
ADVISOR: DR. L.S. STEPHENS
DR. A.T. BOURGOYNE, JR
PART NUMBER: 9
DATE DRAWN: 2 / 26 / 97
SCALE: 1" = 1"
SPONSOR: MINERALS MANAGEMENT SERVICE



0.230"
0.431"
0.556"
0.705"
1.131"
0.170"
1.240" Dia
1.490" dia
-0.003"
+0.000"
1.250" Dia
Bearing Surface
0.050" Fillet Rad
0.866"
0.500"
SECTION A-A
HEX HEAD
R0.0625"
A
A
0.597"
TOP VIEW
1.650"
+0.005"
-0.000"
DRILL STRING SAFETY VALVE
TEAM 2 : BRAD SMITH, MARK CASEMORE, ELLIOT COLEMAN, THOMAS CORE
TOLERANCES
UNLESS SHOWN OTHERWISE
X.X = +/- 0.05"
X.XX = +/- 0.005"
X.XXX = +/- 0.001"
PART NAME: STEMS
PART NUMBER: 11
MATERIAL: 17-4 P H STAINLESS STEEL
DATE DRAWN: 2 / 25 / 97
DRAWN BY: BRAD SMITH
SCALE: 1" = 1"
ADVISOR: DR.L.S. STEPHENS
DR. A.T. BOURGOYNE, JR
SPONSOR: MINERALS MANAGEMENT SERVICE



DRILL STRING SAFETY VALVE

TEAM 2 : BRAD SMITH, MARK CASEMORE, ELLIOT COLEMAN, THOMAS CORE

PART NAME:	STEM LINK	PART NUMBER:	12
MATERIAL:	17-4 P H STAINLESS STEEL	DATE DRAWN:	1/21/97
DRAWN BY:	MARK CASEMORE	SCALE:	1" = 1"
ADVISOR:	DR.L.S. STEPHENS DR. A.T. BOURGOYNE, JR	SPONSOR:	MINERALS MANAGEMENT SERVICE

TOLERANCES
UNLESS SHOWN OTHERWISE

X.X = +/- 0.05"
X.XX = +/- 0.005"
X.XXX = +/- 0.001"

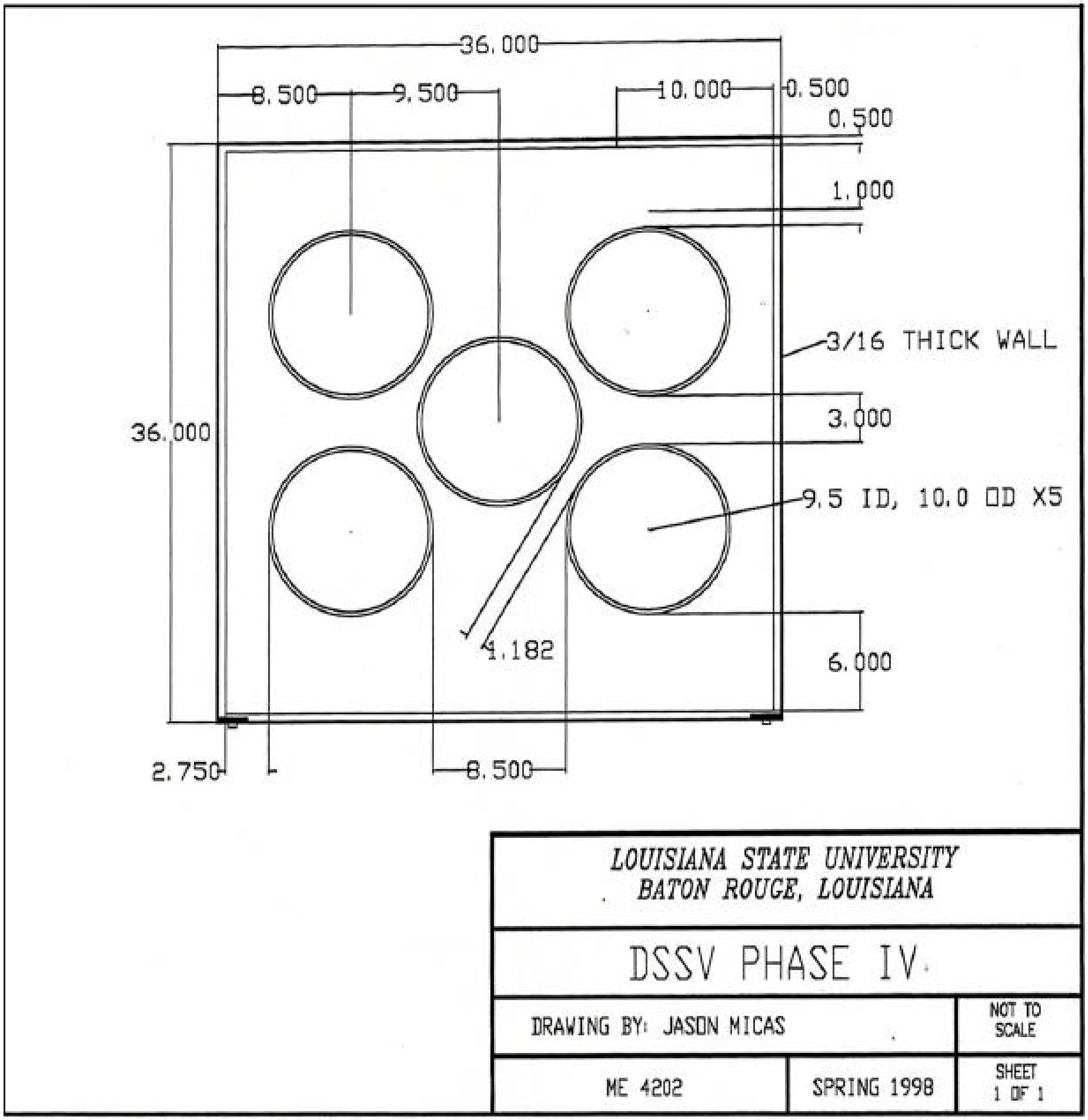
REVISIONS	
MADE BY	DATE

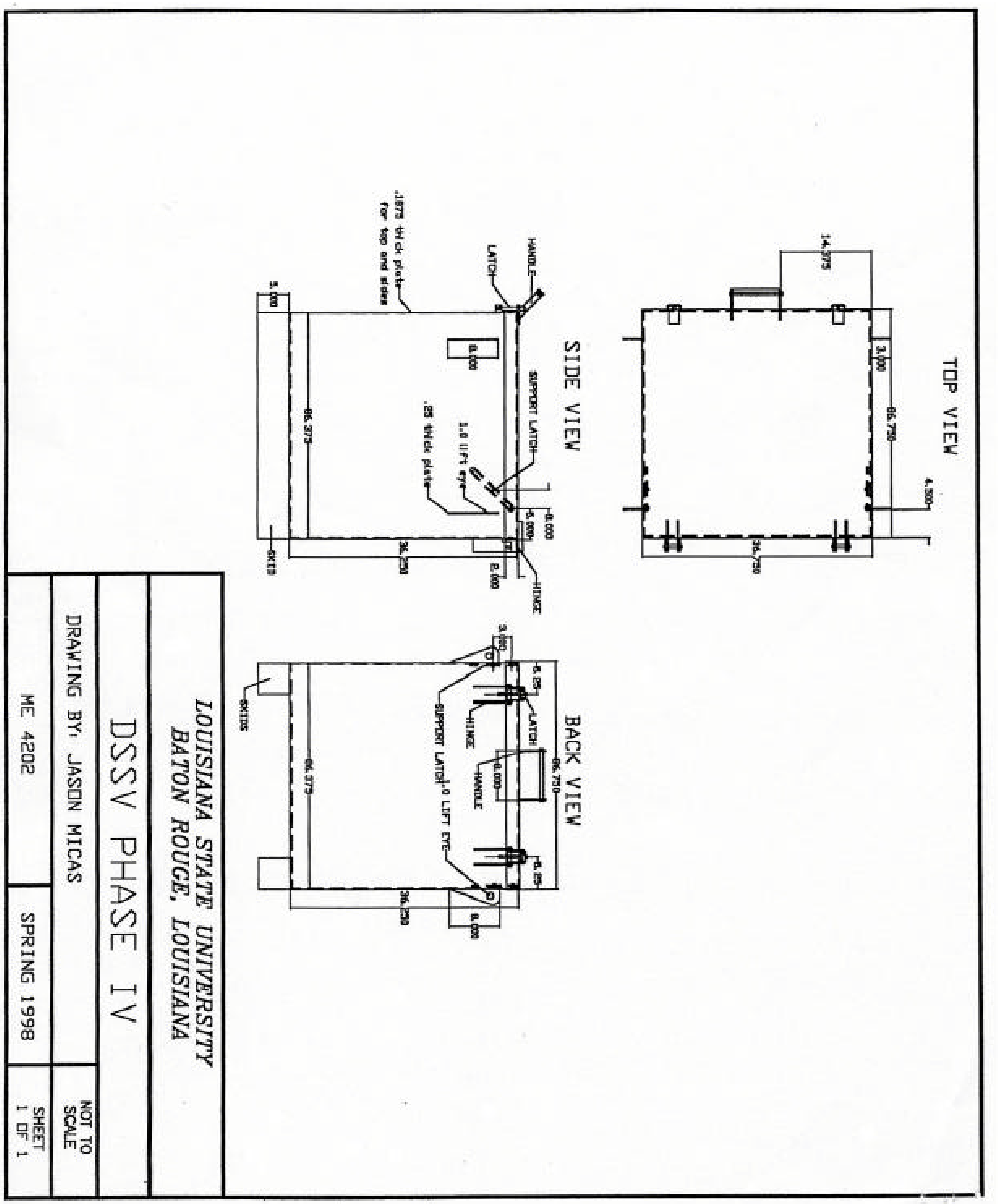
Appendix

B

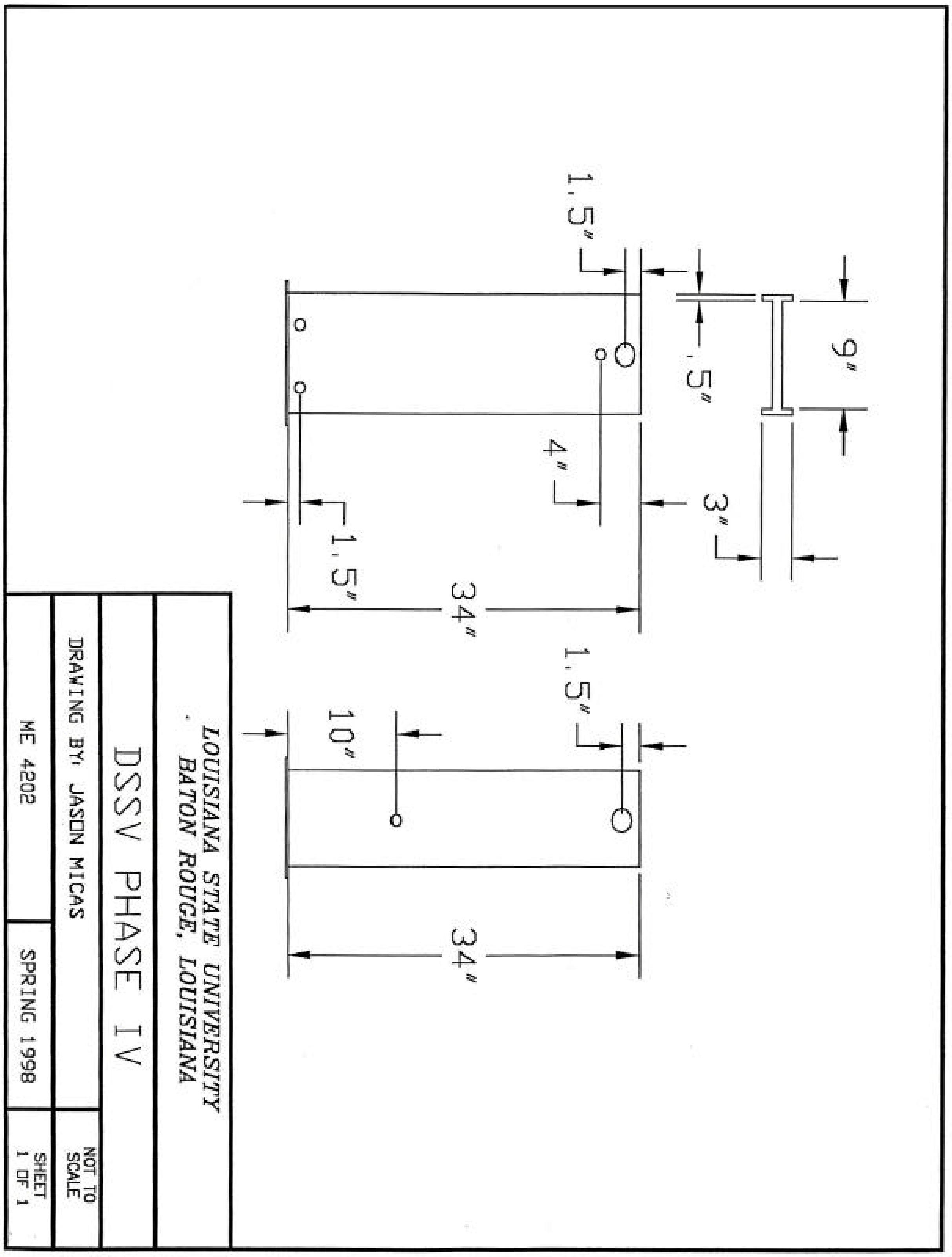
Phase IV Prototype DSSV Storage Stand

This appendix contains the machine drawings for the Phase IV prototype DSSV Storage Stand..


36.000
8.500
9.500
10.000
0.500
0.500
1.000
3/16 THICK WALL
3.000
36.000
9.5 ID, 10.0 OD X5
1.182
6.000
2.750
8.500
LOUISIANA STATE UNIVERSITY
BATON ROUGE, LOUISIANA
DSSV PHASE IV
DRAWING BY: JASON MICAS
NOT TO SCALE
ME 4202
SPRING 1998
SHEET 1 OF 1



TOP VIEW
SIDE VIEW
BACK VIEW
LOUISIANA STATE UNIVERSITY
BATON ROUGE, LOUISIANA
DSSV PHASE IV
DRAWING BY: JASON MICAS
ME 4202
SPRING 1998
NOT TO SCALE
SHEET 1 OF 1


LOUISIANA STATE UNIVERSITY
BATON ROUGE, LOUISIANA
DSSV PHASE IV
DRAWING BY: JASON MICAS
NOT TO SCALE
ME 4202
SPRING 1998
SHEET 1 OF 1
9"
.5"
3"
1.5"
4"
1.5"
34"
1.5"
10"
34"