GoM Drilling, Completions and Interventions

Technical Assurance Report
Well Cap with Triple-Ram Stack

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1 Executive Summary

After the upper G-series flange of the Deep Water Horizon BOP flexjoint riser adapter has been removed, the riser will be mated to a transition spool and capped off with the Triple-Ram stack. The final assembly will have capability to shut-in the well, produce from the well or maintain some level of back pressure to assist in killing the well if necessary. A system and component level design assessment has been completed to evaluate the structural, pressure and sealing capability of the system against the following basic conditions:

Design life of ~ 3 months

Maximum differential pressure of 6,650 psi @ 40°F (shut-in conditions)

Maximum oil flowing temperature of 220°F

This technical assurance report covers the design, fabrication and testing of the Triple-Ram stack and transition spool as well as evaluation of the complete system including the existing flexjoint and BOP. The report supplements the original design calculations with additional design reports where appropriate and assumes standard industry proven components have been designed and tested based on manufacturer’s rated capacity.

Summary of results:

- After installation of the Triple-Ram stack, the system is considered structurally stable at the current inclination angle if the flexjoint is properly restrained. The weakest structural component is the transition spool. Analyses have been conducted which show the transition spool has adequate margin against plastic collapse (i.e. burst) and should be capable of short term operation provided the final inclination angle remains ≤3° from vertical with all six (6) G-series flange bolts in place. This analysis does not consider additional equipment or loads applied above the Triple-Ram stack.

- A system for jacking and stabilizing the flexjoint riser adaptor has been developed and tested.

- Buckling of the transition spool is not considered credible.

- Components with the highest potential of leakage are:
  - Elastomeric element in the flexjoint.
  - Face seal O-ring assembly of the mud boost valve.

- While the elastomeric element of the flexjoint and mud boost flange assembly O-ring cannot be inspected, analyses show the flexjoint elastomeric element and mud boost face seal should remain sealed during anticipated load conditions.
2 Scope of Work

2.1 General

The scope of this work is the assessment of the structural, pressure and sealing integrity of the assembly. The major components are shown in Figure 1, subsequent sections will further define key sub-components as required.

Figure 1 - Component Overview
The individual component analysis was carried out by different companies with integration and oversight by BP.

<table>
<thead>
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<th>Description</th>
<th>Analysis By</th>
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<tr>
<td>1</td>
<td>Triple Ram Stack</td>
<td>Transocean</td>
<td>Triple Ram</td>
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<tr>
<td>2</td>
<td>Transition Spool</td>
<td>IntecSEA/Frazer Nash</td>
<td>Transition Spool</td>
</tr>
<tr>
<td>3</td>
<td>Flexjoint</td>
<td>Oil States Industries</td>
<td>Flexjoint Assembly</td>
</tr>
<tr>
<td>4</td>
<td>Horizon LMRP</td>
<td>Stress Engineering Services (SES)</td>
<td>Design Criteria</td>
</tr>
<tr>
<td>5</td>
<td>Horizon BOP</td>
<td>SES</td>
<td>Design Criteria</td>
</tr>
<tr>
<td>6</td>
<td>Wellhead casing</td>
<td>SES</td>
<td>Design Criteria</td>
</tr>
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</table>

Table 1 - Component Analysis Responsibilities

The system is defined by a free standing Triple-Ram stack connected through a transition spool to the G-series flange at the top of the flexjoint which in turn is connected to the Horizon BOP attached to the wellhead casing. Particular attention has been paid to the structural stability, pressure containment and leak tightness of bolted joints. Detailed inspection of the existing components on the seabed is not possible and, where necessary, reasonable engineering judgment has been employed.

It has been necessary to make assumptions regarding the external loads applied during installation and operation. If there are significant changes from these assumptions, then this report should be reviewed and updated.

2.2 DOE Tri-Lab Assessment

A design review was held with the DOE Tri-Lab team on 23-Jun-10. Both BP’s presentations and DOE’s presentation, findings and recommendations are attached as Appendix A. Many of the findings and recommendations are addressed in this report. Where possible, Table 2 links specific items to relevant report section, and items not addressed in this report will be covered separately.
<table>
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<th>Report Section</th>
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<tr>
<td>F#1</td>
<td>BP must monitor the tilt of the combined stack to ensure that the induced load limit is never exceeded</td>
<td>Procedure</td>
</tr>
<tr>
<td>F#2</td>
<td>Since overpressure can lead to catastrophic failure, the selection of the burst disk must be approved by the DOE team.</td>
<td>NIS</td>
</tr>
<tr>
<td>F#3</td>
<td>Detailed bolt analysis (material property, preload and torque information, etc.) was not presented - only individual bolt load capability was available. This information must be provided and inserted into the record.</td>
<td>5.3.2</td>
</tr>
<tr>
<td>F#4</td>
<td>Since the Transition Spool is expected to be stressed near yield, BP must either present relevant quality data (material certification / inspection records and manufacturing) for the as-built component, or perform proof testing to demonstrate margin.</td>
<td>5.3</td>
</tr>
<tr>
<td>F#5</td>
<td>Details on glycol injection during installation were not provided during the review. These must be provided to the DOE team.</td>
<td>4.2</td>
</tr>
<tr>
<td>R#1</td>
<td>Ensure that an analysis of the allowable worst case combination of loads (e.g. largest tilt, highest temperature, highest pressure) is included in the record.</td>
<td>3.2</td>
</tr>
<tr>
<td>R#2</td>
<td>Measures should be taken to limit maximum pressure during well integrity testing and well shut-in operations.</td>
<td>Procedure</td>
</tr>
<tr>
<td>R#3</td>
<td>Given the many possible loading conditions, supporting analysis capability should remain engaged to quickly evaluate actual operational conditions as events unfold.</td>
<td>Procedure</td>
</tr>
<tr>
<td>R#4</td>
<td>Although not presented during the design review, BP’s evaluation of the ability of all added components to withstand the maximum expected internal pressure, should be included for the record (include as-built documentation to ensure proper bolt pre-load).</td>
<td>This Assurance Report</td>
</tr>
<tr>
<td>R#5</td>
<td>An elastic/plastic finite element model should be run to predict ultimate failure mode and pressure of the stack system (e.g. Flex Joint, G Flange and Transition Spool).</td>
<td>5.4</td>
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<tr>
<td>R#6</td>
<td>A calculation should be performed prior to installation in order to determine the effect of having five or fewer bolts restraining the G-flange and/or increased differential pressure.</td>
<td>5.4</td>
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<tr>
<td>R#7</td>
<td>Connection between the combined BOP stack to collection operations vessels on the surface should recognize new limitations in the shimmed Flex Joint.</td>
<td>NIS</td>
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<td>R#8</td>
<td>Use of shims to limit rotation of the flex joint will alter the load path and requires further analysis (including installation loads on the bolts) prior to installation.</td>
<td>3.2.2.2</td>
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<td>R#9</td>
<td>Installed bolt torque on Mud Boost valve flange should be confirmed, if possible via documentation of the as-built configuration.</td>
<td>6.3</td>
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<tr>
<td>R#10</td>
<td>BP should monitor for leaks around the Mud Boost valve flange during any high pressure operation.</td>
<td>Procedure</td>
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<tr>
<td>R#11</td>
<td>In addition, BP should consider injecting a pressure sensitive sealant into the Mud Boost valve flange in order to reduce the possibility of leakage at the flange.</td>
<td>Procedure</td>
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<tr>
<td>R#12</td>
<td>Generate a complete system level analysis of the estimated accuracy of pressure measurement.</td>
<td>NIS</td>
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<tr>
<td>R#13</td>
<td>Once a correlation between the three pressure measurements is made, transmit only one of the three pressure measurements so that higher frequency sampling rate can be obtained.</td>
<td>NIS</td>
</tr>
<tr>
<td>R#14</td>
<td>Attention should be given to potential bolt issues during riser flange removal and FCS installation such as tool access/engagement, possible need for captive nut replacement, and alternate means of providing flange clamping force.</td>
<td>Procedure</td>
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NIS – Not In Scope of this document  
Procedure – Item to be addressed in a Procedure

Table 2 - DOE Tri-Lab Assessment Actions
3 Design Criteria

The assessment work has been completed on the basis of the following current conditions and the future conditions.

3.1 Current Conditions for Horizon BOP

3.1.1 Stack Inclination

The Horizon BOP and flexjoint body are leaning 2.0°, ±0.5° in the direction of 310°, ±15°. Following the removal of the damaged riser, the flexjoint riser adaptor was leaning 4.5° (2.5° relative to the flexjoint body), ±0.5° in the direction of 310°, ±15°. The orientation of this inclination with respect to the top of the LMRP is shown in Figure 2.

![Figure 2 - BOP Orientation and Inclination](image)

3.1.2 Riser Adaptor Flange on Top of Flexjoint

The flexjoint used on the MC 252 well was from the Transocean Nautilus MODU as the Horizon unit was being refurbished. Transocean has informed BP that the flange at the
top of the riser extension pipe is a HMF G-Series flange. Figure 3 shows the existing G-Series flange after the riser pipe was cut.

![Figure 3 - G-Series Flange After Riser Removal](image)

### 3.1.3 Conditions During Riser Failure

An analysis was conducted to estimate loads in the riser just prior to the riser parting and falling to the seafloor. SES performed a structural analysis of the stack-up with the 21" marine drilling riser connected and including soil-structure interaction of the 36" casing with the 28" casing cemented inside (Appendix B). The four main components are the 36" conductor, the wellhead connector at the bottom of the BOP, the HC connector between the BOP and LMRP (Figure 1) and the telescopic joint outer barrel below the tension ring as shown in Figure 4.

---

1 E-mail from Steve Hand on 28-May-10

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Soil P-Y curves used for the soil structure interaction loads on the 36" casing were developed using API RP 2A. A sensitivity case was also evaluated where the undrained shear strength was increased by a factor of five for calculating the P-Y values. The analysis based on assumed undrained shear strength predicted large displacement at the mudline for the 36" casing (several diameters of displacement) at larger vessel offsets where failure was expected to occur. The actual soil deformation around the 36" casing of approximately one casing diameter (see Figure 5) does not support this prediction.
Two riser tension cases were also examined. The first case compared the recommended value in the Transocean Riser Management Plan report and a lower value (essentially the in-water weight of the riser) to determine if the flexjoint would bottom out and how far off location the rig could move before the tensioners bottom out. A second low tension case was also evaluated where the tensioners were assumed to have no stiffness, in an attempt to match the physical observation of the bottomed out flexjoint. Internal fluid content was taken as 2 pounds per gallon (ppg) to account for the well flowing with a high GOR fluid.

The yield capacity of the wellhead connector is 6.0 M ft-lbs at 1,500 kips tension and 15,000 psi pressure. The working capacity is 4.0 M ft-lbs at 1,500 kips and 15,000 psi pressure which is 66% of yield capacity.

A linear elastic analysis was conducted by statically offsetting the vessel as a percent of the water depth to evaluate stresses in the various components and determine the susceptible failure location. Table 3 shows a summary of the four (4) cases considered.

- Case 1 used reasonable parameters for setting up the analysis. Stresses in the 36” casing and wellhead connector are predicted to below yield. However, the flexjoint angle was less than observed and casing displacement at the mudline was predicted to be much higher than observed (Figure 5).
• Case 2 increased soil shear strength in an attempt to match the actual flexjoint and mudline displacement. Since the soil is much stiffer, the 36" casing is approaching a fixed boundary condition at the mudline, and the casing and wellhead stresses increase above yield. The flexjoint still does not bottom out but the mudline displacement is closer to that observed.

• Case 3 starts with the rig supporting the riser weight (i.e. low tension) assuming a tensioner or two could have failed. The initial tensioner stroke was 10' (vs. closer to mid-point) and they were allowed to stroke normally. The flexjoint did come close to bottoming out. The results indicate that when the telescopic joint reached yield, the 36" casing stresses were just above yield and wellhead stresses were below yield.

• Case 4 utilized the same low tension as Case 3 but zero stiffness in the tensioners. The results matched the observation; the flexjoint bottomed out prior to the riser parting. However, the displacement of the casing is predicted to be much less than observed. For this case, at the 14% vessel offset where the flexjoint would first bottom out, the stresses in the 36" casing were less than 15 ksi and the wellhead stresses were also below yield. However, this is not realistic as the tensioners require a finite stiffness to support the riser and with the low tension, failure of the riser is not expected.

<table>
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<tr>
<th>Case</th>
<th>Inner Tool Ring Tension at 98%</th>
<th>Inner Tool Ring Tension at 98%</th>
<th>Vessel Offset at Top</th>
<th>Maximum Lower Pipe Joint Load (ksi)</th>
<th>Vessel Offset at 14%</th>
<th>Maximum Lower Pipe Joint Load (ksi)</th>
<th>Yield Stress in the 36” Inner Casing (ksi)</th>
<th>Maximum Wall-thickness Bending Moment (ksi)</th>
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<td>1</td>
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<td>25</td>
<td>10</td>
<td>10</td>
<td>10</td>
<td>50</td>
<td>10</td>
<td>4.57</td>
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<td>10</td>
<td>**</td>
<td>.99</td>
<td>**</td>
<td>2.96</td>
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Table 3 - Results for Horizon BOP Stack & Riser Assessment

Although this was not an in-depth analysis, it suggests the telescopic joint outer barrel just below the tension ring is where failure initiated. Once the tensioners bottom out, the tension in the riser would increase very rapidly with vessel offset. The maximum riser tension would be located at the surface tensioner load ring. It is possible some of the tensioners also failed prior to the riser parting. From the survey of the riser on the seafloor, the riser is known to have failed near the drilling rig. From ROV video footage, it appears the flexjoint was bottomed out (reached its maximum stroke of 10°) prior to the riser parting. Upon parting and failing to the seafloor, the kink in the cross-over
section of the 21" marine riser connected to the top of the flexjoint riser adaptor occurred as seen in Figure 6

Figure 6 - Riser Kink

The results from this analysis are inconclusive; none of the cases reasonably represent the failure based on known observations. The analysis did not account for the rig listing and the riser likely parted when the rig was rolling over just prior to sinking. Furthermore, the rig center as surveyed on the seafloor is ~1,300' from the well location (offset of 26% of water depth) and the rig was not expected to have planed much as it sank. The analysis does however, reasonably bound the problem. The results range from the conductor and wellhead being below yield to going just above yielding but not expected to fail and the riser failing in the telescopic joint in the outer barrel just below the tension ring.

It is expected the Horizon BOP components have retained close to their original design capacity.
3.1.4 Current Operating Conditions

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<td>Hydrostatic head</td>
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<td>Differential pressure</td>
<td>&lt;100 psi at Flexjoint</td>
</tr>
<tr>
<td>Temperature</td>
<td>220 °F</td>
</tr>
<tr>
<td>Flexjoint rotation</td>
<td>4.5° ±1° from vertical</td>
</tr>
<tr>
<td>Flow rate</td>
<td>&gt;25 mbpd</td>
</tr>
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Table 4 - Current Operating Conditions

3.2 Future Conditions

3.2.1 Stack-up Analysis

The stack-up configuration was shown in Figure 1 with the Triple-Ram stack on top. SES has performed a structural analysis of the stack-up with loads imposed at the top of the BOP and includes the soil-structure interaction of the 36” casing with 28” casing cemented inside. The analysis was performed to determine system weak points under worst case loading conditions. The three areas of concern are the 36” conductor, the H4 wellhead connector at the bottom of the Horizon BOP and the HC connector between the Horizon BOP and Horizon LMRP (Figure 7). The loads from this system analysis were also used to evaluate the transition spool at the bottom of the Triple-Ram stack.

Analyses were completed for three inclination angles of the flexjoint. The 36” casing and Horizon BOP have an initial angle of inclination of 2°. The flexjoint riser adaptor was inclined to 3.5° relative to the Horizon BOP stack (2.5° relative angle as measured plus 0.5° uncertainty and 0.5° to allow space to install stabilization blocks) and further rotation prevented. A 10° relative flexjoint riser adaptor angle was also examined. The three load cases examined are summarized in Table 5.
Figure 7 - Areas of Concern

<table>
<thead>
<tr>
<th>Case #</th>
<th>Initial Angle of casing (deg)</th>
<th>Penetration Depth (m)</th>
<th>Wireline String Over which soil is removed (m)</th>
<th>Heading Angle (deg)</th>
<th>Final Angle of Inclination of Horizon (deg)</th>
<th>Final Angle of Inclination of Triple Elements (deg)</th>
<th>Bending Moment at Lower Connector (10° incl)</th>
<th>Bending Moment at Lower Connector (15° incl)</th>
<th>Wellhead Bending Moment at BOP (15° incl)</th>
<th>Wellhead Bending Moment at BOP (15° incl)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case N1</td>
<td>2</td>
<td>0</td>
<td>18.0</td>
<td>2.5</td>
<td>2.1</td>
<td>5.8</td>
<td>0.22</td>
<td>0.71</td>
<td>1.84</td>
<td>1.84</td>
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<tr>
<td>Case N2</td>
<td>2</td>
<td>0</td>
<td>19.0</td>
<td>3.3</td>
<td>14.0</td>
<td>4.5</td>
<td>0.61</td>
<td>0.90</td>
<td>1.17</td>
<td>2.18</td>
</tr>
<tr>
<td>Case N3</td>
<td>2</td>
<td>0</td>
<td>28.4</td>
<td>0.7</td>
<td>14.7</td>
<td>14.7</td>
<td>0.84</td>
<td>0.84</td>
<td>1.26</td>
<td>2.46</td>
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</table>

Table 5 - Load Cases for BOP Stack Analyses

The goose neck loads provided in Appendix B were applied to the hubs. Hub loads on the inclined side of the stack were analyzed as the worst case. The dry weight of the
Triple-Ram stack and transition spool is 150 kips with a center of gravity 15.63' above the G-series flange. The G-series flange is 7.31' above the flexjoint center of rotation.

Soil P-Y curves for use in the soil structure interaction loads on the 36" casing were developed using API RP 2A. However, the undrained shear strength was doubled for calculating the P-Y values based on input from the BP Geotechnical Engineer Technical Authority. The decision to double the undrained shear strength is further supported by reviewing the soil deformation around the 36" casing. Figure 5 depicts a gap at the mudline of approximately one casing diameter, while the drift off analysis predicted a much larger gap. To explore the effect of soil gap, a case was run where the upper 30' of the soil was removed. The moments along the length of the stack are shown in Figure 8.

![Bending Moment Distribution: Cases N1, N2, and N3](image)

**Figure 8 - Moment along Stack**

The 36" conductor is X65 material and the stress resulting from these moments are well below the allowable stress. The 28" casing is X56 and the resulting stresses also

---

<table>
<thead>
<tr>
<th></th>
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<tbody>
<tr>
<td>Authority:</td>
<td>Mark Nichols</td>
</tr>
<tr>
<td>Custodian/Owner:</td>
<td>Tim Bieri</td>
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<td>(if applicable):</td>
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<td>Page:</td>
<td>Page 19 of 48</td>
</tr>
</tbody>
</table>

Warning: Check DW Docs revision to ensure you are using the correct revision.
below the allowable stress. The working capacity of the HC connector at the bottom of
the LMRP at 10,000 psi and zero tension is 2.2 million ft-lbs. The working capacity of
the H4 connector at the bottom of the BOP at 10,000 psi and zero tension is 5.0 million
ft-lbs.

For all cases, the applied moments in the connectors are below the
capacity.

Figure 9 shows the flexjoint moment-rotation curve and the moment applied by the
Triple-Ram stack. The flexjoint moment capacity is essentially equal to the applied
moment from the Triple-Ram stack.

The Horizon BOP stack currently has a 2° inclination and the applied bending moments
from the Triple-Ram stack did not include the goose neck loads. Therefore, the flexjoint
will require stabilization in order to support the Triple-Ram stack. Further information on
the stack analysis can be found in Appendix A.

If left unstabilized under the applied load, the flexjoint will rotate until it
bottoms out.

![Figure 9 - Flexjoint Moment and Applied Bending Moment From Stack vs. Angle](image)

<table>
<thead>
<tr>
<th></th>
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<tbody>
<tr>
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<td>Mark Nichols</td>
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<td>Next Review Date:</td>
<td>(if applicable):</td>
</tr>
<tr>
<td>Page:</td>
<td>Page 20 of 48</td>
</tr>
</tbody>
</table>

Warning: Check DW Docs revision to ensure you are using the correct revision.
3.2.2 Flexjoint Riser Adaptor Jacking and Stabilizing

In order to improve the ease with which the transition spool can be inserted and to reduce self-weight moments once the capping stack has been landed, it was considered desirable to reduce the angle of the riser extension pipe to the vertical. A jacking system was designed to push the riser extension pipe into a more favorable angle, capable of effecting up to 5° movement from the current location. A stabilization system was then designed to hold the riser extension pipe in its new location.

3.2.2.1 Jacking and Stabilizing Design Loads

The jacking system was designed to be capable of overcoming the following potential sources of resistance (see Appendix C):

- The stiffness of the elastomeric element of the flexjoint, estimated to be a maximum of 417 kip-ft over 5°;
- The plastic moment capacity of two trapped and bent 6 5/8” 27 lb/ft drill pipes, estimated as a maximum of 70 kip-ft;
- Frictional effects at the nipple to wear sleeve interface measured on another flexjoint to be of the order of 500 kip-ft.

The total design moment for the jacking system is therefore about 1,000 kip-ft.

A convenient means of applying the load is to react off the bolt circle and the inside diameter of the flexjoint top ring, pushing on the thick section of the riser extension pipe where it tapers. This geometry offers a moment arm of 39.5” which gives a total load of 304 kips, achievable using hydraulic cylinders. In order to develop the required load in any direction using hydraulic cylinders of a practical size, the system uses 14 relatively compact hydraulic cylinders acting radially (see Figure 10), the combined force capability of which can be vectored to exceed the demand load of 304 kips (by 1.08 for 3 rams, or by 1.36 for 4 rams).
Once the riser adaptor has been positioned, it will need to be stabilized. The stabilization system is designed to resist installation and in-place loads without causing damage to either the system components or the flexjoint. Loads on the stabilization components arise from:

- Moment and deadweight of the stack components at their installed angle;
- Installation loads associated with landing the stack and its connections;
- Moments and forces from the CDP connections during production;
- Loads from an attached drilling riser, including potential drive-off loads

The stabilization system is described in Appendix C and comprises both wedges and solid pins to replace the jacking cylinders once the final position of the riser extension pipe is attained (Figure 11). The integrity of these components under the worst case loads is described below.
3.2.2.2 SIT Testing of the Jacking and Stabilization systems

SIT testing of both the jacking system and the stabilization system of wedges and pins was conducted at Oil States Industries using a spare flexjoint. These tests are reported in detail in Appendix C, with the findings summarized as follows:

For the jacking system

- Rams pressurized to 7,300 psi are capable of moving the flexjoint by 5°;
- Heavy contact between the assembled parts was noticeable but acceptable.

For the restraint system

- The restraint teeth penetrated through the flexjoint paint into the steel; however, in a dry friction condition the depth of penetration was shallow. This is not considered a problem for installation subsea;
- The effect of vertical movement of the riser extension was hard to demonstrate, as with 50 kips applied only got 0.050" lift. However, the use of both pins and wedges will make the assembly insensitive to vertical lift;
• Adequate clearances for blocks and wedges were demonstrated;
• Rams were pressurized to 10,000 psi opposite a single wedge without significant distortion.

3.2.2.3 Integrity of the Jacking and Stabilization systems

The integrity of the Jacking and Stabilization components was assessed using elastic-plastic FE analysis, with contact modeled between the various parts of the assembly (See Appendix C). This analysis recommended maximum load limits for the components as follows:

• 173 kips for the pin/ram holders, based on the strain concentration at the internal corner of the ram receptacle
• 215 kips for the pushers (which hook over the internal bore of the flexjoint lid) based on proximity to global plastic collapse

The potential for buckling of the riser extension adaptor due to being restrained by a single pair of wedges under a deadweight load of 170 kips and 1200 kip-ft of moment was also assessed using a shell finite element analysis and the method of ASME Code Case N-284-1 with a safety factor of 2 (see Appendix C).

Based on the loading above, the most probable buckling direction is predicted to be longitudinal with a design margin of 2.93. A sensitivity study including an external pressure of 1500 psi (corresponding to a flow to surface of a low density oil and gas mixture) resulted in hoop buckling becoming dominant, with a design margin of 2.61.

3.2.3 Future Operating Conditions

Table 6 shows the operating conditions of the flexjoint and Triple-Ram stack during 3 months operation.

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydrostatic head</td>
<td>2,250 psi</td>
</tr>
<tr>
<td>Shut in pressure</td>
<td>8,900 psi</td>
</tr>
<tr>
<td>Differential pressure</td>
<td>6,650 psi</td>
</tr>
<tr>
<td>Temperature</td>
<td>40°F Shut-in</td>
</tr>
<tr>
<td></td>
<td>220°F Producing</td>
</tr>
<tr>
<td>Hydrotect pressure on surface</td>
<td>7,500 psi</td>
</tr>
<tr>
<td>Triple-Ram stack angle</td>
<td>3° maximum</td>
</tr>
</tbody>
</table>

Table 6 - Future State Operating Conditions
4 Triple Ram

4.1 Description

The Triple-Ram stack comprises a Cameron HC collet connector, a Hydril single ram BOP, a Hydril dual-ram BOP, 2 WOM subsea double block gate valves, 2 Cameron mini-connector mandrels, a Vetco mandrel, BOP valve support frame (refer to Figure 12 and Appendix D). With the exception of the valve support frame, these components are industry standard components normally used for blowout prevention and bleed-off. In the proposed application, this stack may be required to shut-in the well, produce from the well or help provide back pressure to assist in the well kill operation.

![Figure 12 - Triple-Ram Stack Major Components](image)

The stack design, as configured, is inherently capable of holding back pressure to shut-in the well. To meet the requirements in this application the system design must also be able to meet the following requirements:

- Demonstrate pressure containing capability and functionality through hydro and functional testing with particular attention to the hydraulics and ram closure.
- Provide design assurance that the valves, mini-connectors, choke and bursting disk (when assembled) and the flowline loads are supported adequately for all operating conditions,
• Assure erosion and flow induced vibration (FIV) will not compromise equipment integrity during the anticipated period of operation.

A design assurance report had been generated and submitted by Transocean to address the design integrity of the system as well as the functioning and pressure testing of the assembly at Cameron’s Berwick facility and is presented as Appendix D. Key conclusions include:

- There is a low probability ductile burst, structural collapse or loss of pressure containment.
- Localized erosion in the capping stack is a possibility but unlikely to result in a leak for the limited duration when producing.
- Fatigue resulting from FIV is also considered unlikely, due to structural stiffness including the valve support frame.

4.2 Hydrate Management

The Triple-Ram stack will be installed in two phases: the transition spool with HC connector will be installed utilizing wireline, followed by the Triple-Ram stack on drill pipe. A hydrate mitigation plan has been developed and approved by BP’s Flow Assurance Engineer on the project and presented in Appendix D.

The mitigation of hydrates during installation of the transition spool is focused on possible formation of hydrates on the flange mating surfaces and seals. The hydrate formation temperature and pressure are 70°F and 2,250 psia respectively. At ambient seabed pressure and temperature (~2,250 psia, 40°F) seawater and hydrocarbon vapor will form hydrates. The transition spool temperature will be 40°F when lowered outside the plume. While approaching the plume, the spool piece could experience hydrate fouling from the hydrocarbon gas flowing from the well and mixing with cold sea water.

Oil and gas flowing from the well at 200°F is capable of warming the lower end of the transition spool above the hydrate formation temperature over a period of time. However, the approach of the transition spool in the plume right above the flexjoint riser adapter must be done as fast as practical to minimize the accumulation of hydrates on the flange and the other sensitive areas. It may be necessary to hold the transition spool for a short period to dissipate any hydrates that form prior to bolting up the G-series flange. Additionally, a special grease which is effective for hydrate disassociation will be applied to all sensitive areas of the assembly. Finally, an ROV mounted injection system will provide hydrate inhibitor locally to support the installation process as required (e.g. flush hydrate deposits from the bolt inserts where fouling might interfere with making the flange).
The Triple-Ram stack is installed after installation of the transition spool piece. The
Triple-Ram stack will be lowered remote from the plume. During the approach, the
outer surface of the structure may accumulate hydrate deposits. Therefore, as with the
transition spool, installation must be executed as quickly as practical in order to
minimize exposure to hydrate formation. An ROV mounted injection system will
provide hydrate inhibitor locally to support the installation process as required.

Once the Triple-Ram stack has approached the plume and is right above the HC
connector mandrel of the transition spool, the majority of the flow will be going through
the Triple-Ram stack and exiting from a perforated riser joint above it. There is a
possibility of hydrate formation in the stack internals due to water entrainment with the
buoyant oil and gas stream. The injection of the hydrate inhibitor will be necessary to
reduce hydrate formation.

The drill pipe used for lowering the Triple-Ram stack will provide hydrate inhibitor
through its bore to a 2" flexible pipe to one of the 3" mini hub Cameron collet
connectors. This will feed the bore of the stack with large quantities of hydrate inhibitor
to treat the entrained water. Additionally, the HC connector will be injected with
hydrate inhibitor provided by a flying lead from the IWOCs attached right above the
perforated riser on the drilling string.

- The probability of hydrate fouling during installation of the capping stack
  is significant and requires preventative measures.
- Rapid installation will reduce exposure to hydrate formation and
deposition.
- Hydrate inhibitor injected into the Triple-Ram stack bore and HC
  connector will suppress hydrate formation during installation.
- Strategic application of non-stick grease and hydrate inhibitors to the
  unit will prevent fouling of key areas.
5 Transition Spool

5.1 Description

The transition spool is shown in Figure 13 and is comprised of three elements:

- GE/Vetco Gray G-Series flange complete with male nose ring and 6 bolts;
- 21.5” OD x 1” WT X80 transition pipe;
- 18.75” API 15K weld neck flange.

![Diagram of Transition Spool Components](image)

**Figure 13 - Transition Spool Components**

The manufacturing drawings for the transition spool are attached as Appendix E, and the original design drawings for the riser adapter and the G-Series flange are attached as Appendix F. The transition spool is intended to provide an interface between the female G-Series flange at the top of the flexjoint riser adapter spool and a standard API 15K interface, with sufficient length in the spool to accommodate any drill pipe stubs that may remain when the existing G-Series flange is removed.

5.2 Loads

The design pressure of the transition spool is 8,900 psia, which is the estimated wellhead shut-in pressure of the Macondo well\(^3\). The spool is designed taking account

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\(^3\) Communication with Trevor Hill, BP, June 2010.
of the 2,250psi external hydrostatic pressure that exists at 4,992 feet of water depth, i.e. for a differential pressure of 6,650psi. The spool has also been hydrotested a pressure of 7,500psi.

In addition to maintaining a reliable seal at an internal pressure of 8,900psia, the spool must also provide a robust support foundation for the Triple Ram Stack. Depending upon the angle to the vertical at which the spool is ultimately installed, the loading on the spool may include a significant moment loading in addition to the pressure load and the deadweight of the Triple-Ram stack. It is desirable to minimize this moment by making the riser adapter as vertical as possible; nevertheless some non-zero installation angle is inevitable and the transition spool must therefore be capable of carrying moment. The extent of the moment loading under a range of scenarios has been established (Appendix B), from which the design basis for the transition spool may be summarized as:

- A dead weight of 150 kips which is slightly in excess of the anticipated mass of the Triple-Ram stack;
- A moment of 330 kip-ft which is the moment developed when the Triple-Ram stack is installed on the riser adapter at a stabilized angle of 2.5° from vertical, leading to an ultimate equilibrium angle of the Triple-Ram stack of 3°.

The integrity assessment in Section 5.4 considers the performance of the transition spool under these moment and deadweight loads in combination with the internal pressure.

The temperatures relevant to the transition spool are a hot oil flowing temperature of 220°F and a seabed ambient temperature of 40°F corresponding to the shut-in condition. It is normal to select low temperature materials downstream of any significant flow restrictions to avoid JT cooling associated with gas decompression; however, there are no such restrictions in the transition spool.

5.3 Materials of Construction

5.3.1 G-series flange

The flange at the top of the flexjoint mating onto the transition spool is a GE/Vetco Gray G-Series flange. This flange was designed in accordance with the requirements of API 16R. The flange has 6 bolts (See 5.3.2) for carrying axial tensile load and a male nose ring coupling with a 3 o-ring arrangement to achieve pressure sealing within the bore. The flange is rated for 5,000psi internal pressure and 3,000kips of tension (including the pressure end cap load). The pressure end cap load due to a differential pressure of

---

6,750 psi represents an axial tension load of 1,870 kips, so the flange and bolting has some residual capacity for reacting tension loads at this pressure.

The o-rings have been tested for 5,000 psi internal pressure at 180°F but should continue to seal effectively at 6,750 psi. A finite element assessment of the flange design carried out by GE/Vetco Gray (Appendix F) makes the following statement in relation to the performance of the seals at higher pressure:

"The o-ring type seals have been tested to a maximum of 5000 psi at 180 deg F. Under the normal conditions of use in this riser adapter/nose ring interface, higher internal pressure tends to decrease the seal extrusion gap between the nose ring and bore of the HMF connection."

Sealing at elevated internal pressure is discussed further in Section 5.4.

The Material Test Record (MTR) for the particular G-Series flange attached to the riser adapter fitted to the Deepwater Horizon flange joint has not been possible to locate. However, the G-Series flange used for the transition spool is attached at Appendix G. In both cases the material used is AISI 4130 with a specified minimum yield stress of 80ksi.

5.3.2 Flange Bolts

On the G-Series flange there are six 16½" long bolts with a 3-5/8" pitch 2G fit Stub Acme thread per ASME B1.8 that engage with captive elliptical nuts in the riser adapter flange (refer to Figure 14).

![Figure 14 - G-Series Flange Bolt Details](image)
The bolt heads have an integral washer under the hex head with a diameter of 5.13" to reduce bearing stresses. The bolts are pre-loaded to a total flange compression of 3,750 ksi or 625 ksi per bolt. The bolting chart and recommended lubricants to achieve the required pre-load for a given torque are given in Appendix F.

The normal make-up torque using the preferred TS70 Moly paste is 17,000 ft-lbs, which gives the preload of 625ksi per bolt when made up on the surface. TS70 Moly paste is a high performance lubricant with a coefficient of friction of about 0.07, giving approximately twice the preload as a mineral oil for the same applied torque. To account for uncertainty in the lubrication performance achieved when making the bolts up subsea, it is suggested to increase the applied torque by 5-10% (for further explanation, see Technical Note in Appendix F). The performance of the G-Series flange bolting is considered further in Section 5.4.

5.3.3 X80 Pipe section

The availability of materials dictated that the 21½ OD pipe needed to match the bore of the flexjoint connection be fabricated from 1" thick X80 riser pipe. This pipe would not normally be used for a design pressure of 6,650 psi. API 1111 has been used as a consistent basis for assessing the risk associated with using this pipe.

For comparison, using API 1111, typically the thickness of an X80 riser with an operating pressure of 6,650 psi would be 1.5 inch, and a pipeline would be 1.25". A pipeline or riser would normally be designed for at least a 20 year life.

The assessment methodology adopted calculates the burst pressure based on minimum, maximum and actual material properties. The code factors were then used to assess the allowable internal pressure and to make an assessment as to the risk associated with an operating pressure of 6,650 psi and to derive a suitable hydrotest pressure.

<table>
<thead>
<tr>
<th>Material properties</th>
<th>Minimum (ksi)</th>
<th>Maximum (ksi)</th>
<th>Measured (ksi)</th>
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<tr>
<td>Yield Strength</td>
<td>90</td>
<td>100</td>
<td>84.1</td>
</tr>
<tr>
<td>UTS</td>
<td>90</td>
<td>120</td>
<td>101.6</td>
</tr>
</tbody>
</table>

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It can be seen that the X80 pipe has an adequate margin against burst when assessed against the measured properties. Also using measured properties, the incidental overpressure is within 1% of the required operating pressure of 6,650 psi. Incidental overpressure is defined in API 1111 as:

“Incidental overpressure includes the situation where the pipeline is subject to surge pressure, unintended shut-in pressure, or any temporary incidental condition. The incidental overpressure should not exceed 90% of the hydrotest pressure. The incidental pressure may exceed MOP temporarily; but the normal shut-in pressure condition should not be allowed to exceed MOP.”

It can also be seen that an internal pressure of 6,650 psi represents about 80% yield whilst a hydrotest pressure of 7,500 psi represents rather less than 90% yield. A burst test on the X80 pipe gave an actual burst test result of 10,600 psi (Appendix G), and the design pressure of 6,650 psi thus represents about 65% of the burst test pressure. For this application, this is considered acceptable.

The original material certificates and full history of the X80 pipe are not available; however, samples have been tested and have properties that are typical for X80 pipe (Appendix G). The burst test result is also in line with that expected from X80 pipe.

One end of the X80 pipe is welded to the AISI 4130 API 15K flange, and this weld required PWHT. Tensile testing of specimens subject to the simulated PWHT met the API 5L requirements for X80 pipe and in fact exceeded the X80 plate values. The pipe that was used for the burst test was also subject to a simulated heat treatment.
5.3.4 18.75” API 15K Flange

The API 18.75” 15K weld neck flange is an industry standard design rated for 15ksi internal pressure. The material is AISI 4130 with a specified 80 ksi minimum yield. The MTR for the flange is attached in Appendix G.

5.4 Integrity Assessment

The transition spool has been subject to a number of independent assessments of its ability to withstand the required combined pressure, moment and deadweight loadings, as follows:

Appendix H: Spool Analysis 64053V issue 4
- Hand calculations performed by Frazer-Nash Consultancy Ltd
- ABAQUS finite element assessment by Frazer-Nash Consultancy Ltd
- ANSYS finite element assessment by Frazer-Nash Consultancy Ltd

Appendix I: Transition Spool Pipe Checks Rev B
- Code calculations to ASME B31.8/API RP1111/RP2A performed by IntecSea

In terms of establishing the transition spool pressure-moment capacity, Figure 15 shows the hand calculations for hoop burst and fully plastic moment which should bound the problem, together with a Miller curve for axial collapse of a pressurized pipe under bending.
Figure 15 - Hand Calculations

The finite element plastic collapse results are shown superimposed on the hand calculations in Figure 16, and it can be seen that there is good agreement. Note that the finite element results are for the specified minimum yield stress of 80 ksi in the transition spool and for nominal dimensions.
In order to determine a safe working envelope within the boundaries defined in Figure 16, it is informative to compare with the design margins intrinsic in relevant design codes. Figure 17 shows curves for combined pressure and moment with and without additional tension loading from API 2RD. The curves shown are based on yield stress, and therefore correspond to the “Survival” limit in API 2RD.

Also shown are buckling calculations to API RP1111. When a positive differential pressure exists the collapse pressure of the pipe is high; however when the internal pressure is lower than seabed ambient pressure, a buckling collapse mode becomes increasingly feasible. The curve shown is for 3% ovality, which is clearly a conservative assumption with respect to fabrication, as it would not be possible to weld a pipe of greater ovality to a machined flange.
Potential installation, well test, and production scenarios for the triple ram stack are overlaid on the various pressure-moment curves in Figure 18, for the configurations with and without a connection to Nakika (the Nakika tie-back configuration requires a 20 ton H4 connector to be latched onto the top of the Triple-Ram stack, which introduces considerable additional moment). The points all lie within the shaded area, which was constructed using a safety factor of x1.2 on differential pressure and x3 on moment relative to the plastic collapse finite element results. This is considered sufficient to produce a safe design, and permits minor scope creep in the design points if required.
Figure 18 – Various Potential Operating Scenarios

The results of the various assessments of pressure-moment capacity are in close agreement, and provide a clear picture of the design margins of the transition spool against a range of potential failure modes, summarized from Appendix H as follows:

- The most highly loaded portion of the riser adapter/X80 transition spool is the X80 portion above the G-Series flanges.
- For a scenario comprising a riser flowing to the surface filled with fluid of density 300 kg/m³ and 2% out-of-roundness in the transition spool, the margin on buckling under combined pressure and 590 kip-ft of moment is 1.27.
- The margin on yield stress in the bolt shank (including bending) for pressure and 330 kip-ft of moment is 1.31. If the moment load is increased by 75% to 590 kip-ft, this margin reduces to 1.21.
- If only 75% of design bolt preload can be achieved, the bolt load actually decreases indicating that the pre-load has not been overcome. The sensitivity of the bolt preload-torque relationship to potential uncertainty in the coefficient of friction that is achievable when the bolts are made up subsea suggests that it
may be prudent to increase the applied torque by 5-10% to ensure that the 75% minimum required preload is developed.

- The margin on **bearing stress under bolts** is estimated to be of the order of 1.43 under combined pressure and 330 kip-ft of moment.

- **Loss of any bolt** in either of the two moment orientations increases the stresses considerably above yield in both the bolts and the flanges.

- Under moment load the **seal interface at the bore** of the lower flange ovalises very slightly. The maximum difference in sealing diameter between the G-Series flange design case of 5,000 psi and 3 million pounds of axial force, and a combined pressure of 6,750 psi and moment of 330 kip-ft is about 0.005" to 0.007" which is considered acceptable. Further confidence is justified by virtue of the fact that the male seal pin portion comprises two seals, so increased internal pressure actually acts to reduce the seal clearances below those experienced at 5,000 psi.

- Elastic-plastic collapse analysis confirms that the ultimate limit state of the transition spool is governed by the X80 pipe in all cases. The reserve factor obtained by ramping up the design pressure of 6,750 psi if the moment is held constant at 300 kip-ft is 1.39. Similarly, the reserve factor obtained by ramping up the moment while holding the design pressure at 6,750 psi is estimated to be 7.27. Further analyses at 75% and 50% of design pressure also confirm that collapse of the X80 spool dominates.

The transition spool is considered to be robust against a design base case comprising 6,750 psi pressure and 330 kip-ft of applied moment and a range of prudent sensitivity studies.

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**Warning:** Check DW Docs revision to ensure you are using the correct revision.
6 Flexjoint Assembly

6.1 Description

The flexjoint was supplied by Oil States and is located on the top of the LMRP with the intent to allow compliance between the more flexible drilling riser and the rigid BOP stack (Figure 19 & Figure 20). The flexjoint attached to the Horizon LMRP was originally on the Nautilus and was changed out about one year ago. Oil States indicates the flexjoint was manufactured in 1999 and has been refurbished with no damage found. The most vulnerable elements of the flexjoint are the elastomeric element and the mud line valve assembly.

![Image of flexjoint](image)

**Figure 19 - Installed Flexjoint**
The flexjoint is rated for 5,000 psi internal pressure and is hydro-tested to 7,500 psi for 15 minutes. It has a tension capacity of 3,000 kips, at an angle of 10° from vertical and rated to 10,000 feet water depth. Refer to Appendix J for detailed assessment report.

Oil States has delivered over 1,000 drilling riser flexjoints with the first one entering service in 1976. To date, no failure has occurred in the 34 years operating history of drilling riser flexjoints under normal operating conditions. Where flexjoints have bottomed out (rotated to their maximum angle range) and been subsequently disassembled for inspection, no damage has been found.

6.2 Lower Flexjoint Assembly

6.2.1 Flexjoint Elastomeric Element

The flexjoint elastomeric element is designed to allow rotation of the riser and movement of the surface rig relative to the seabed. Normally, the flexjoint is in tension from the over-pull on the drilling riser and, when combined with internal pressure, keeps the flexjoint element in a state of compression and shear.
The flexjoint element is bonded to the body and the retaining ring assembly and the resulting lip seal forms part of the primary pressure boundary. There are two O-rings (Figure 20, item 8) behind the flex element. These O-rings act as environmental seals against ingress of seawater and do not form part of the pressure boundary. There is no mechanical connection between the wear ring and the riser adaptor and the adaptor will lift off the wear ring with low pressure and / or tension load applied. A low internal pressure will overcome the weight of the Triple-Ram stack. If the well was shut-in using the Triple-Ram stack, the flex element would return to the normal operating case of compression and shear.

After the riser parted near the MODU and fell to the seafloor, the flexjoint on the Horizon BOP rotated and bottomed out at its maximum angle. Since cutting the riser away from the BOP, the flexjoint has slowly recovered and appears to be returning back to near neutral position, currently at an angle of ~2.5° relative to the flexjoint body. Full recovery may never occur unless forced back to the neutral position by an external load. The flex element can exhibit a certain amount of set due to the sustained deformation for a prolonged time coupled with the temperature.

A method is being developed to jack the flexjoint back to a near neutral position. Shims will be placed between the top of the flexjoint body (around the bolt ring) and riser adaptor such that the unit cannot rotate any further under the weight of the Triple-Ram stack.

The potential flexjoint leak paths were identified in the report. However, since that report was issued, the expected operating conditions have changed and eight of the additional load cases have been analyzed. In total, 12 cases have been analyzed by Oil States as shown in Table 8.

<table>
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<tr>
<th>Parameter</th>
<th>Case 1</th>
<th>Case 2</th>
<th>Case 3</th>
<th>Case 4</th>
<th>Case 5</th>
<th>Case 6</th>
<th>Case 7</th>
<th>Case 8</th>
<th>Case 9</th>
<th>Case 10</th>
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**Table 8 - Flexjoint Load Cases Summary**

⇒ With the flex element over rotated to 12 degrees, results show that the flex element is sealing. No leak has been reported from the flexjoint.

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5 2200-T2-DO-RP-4003, Rev. 0, Technical Assurance Report on Pressure Containing Capability of the Swing Valve Assembly
The FEA work showed that all metal components remain well below yield strengths for all cases with the exception of the reinforcements between elastomeric shims which showed slight yielding at the tips. This is normal for this component.

The evaluation of the lip seal under these additional conditions cases indicate contact pressure remained above internal pressure for all load cases. This indicates seal integrity will be maintained.

Further details on Case 11 and 12 are provided in Appendix J as these cases represent most closely the expected upper bound operating condition once the Triple-Ram stack is installed. Case 11 would represent conditions right after shut-in for just a few hours and Case 12 would represent the long term condition as the fluids temperature cool to ambient seawater.

The metallic body of the flexjoint is not very highly stressed and the probability of ductile failure is considered low. The largest threat is probably due to degradation in service, such as wear due to the drill pipe on the inlet spool. It is not possible to further quantify this possibility with the current information.

Leakage from the non-metallic joints is considered a higher probability. The following sections address potential leak paths that have been identified.

6.2.2 O-rings between the inlet flange and the wear insert

The potential failure mode is extrusion of the O-ring (Figure 20, item 6) under high internal pressure. Test data shows that the O-ring is adequate for 9,000 psi at 160°F for 30 days. There is another O-ring (Figure 20, item 7) at the top of the wear ring, the design intent of this O-ring is to allow the space between the wear ring and the body to be packed with grease and it is not intended that this is a primary pressure containing seal. The wear ring is bolted in place and forms a metal to metal seal with the body.

The probability of O-ring extrusion and leakage is considered low.

6.2.3 Test port

There are two screwed test ports (Figure 20, item 13). These have previously been subject to the 7.5K hydrotest.

The probability of leakage from these ports is considered to be low.
6.3 Riser Adapter Assembly

The riser adapter assembly is supplied by GE/Vetco Gray and is bolted onto the riser adaptor via a 4 bolt arrangement. This arrangement has been assessed by the supplier, see Appendix F for the additional information:

"This arrangement utilizes an o-ring face seal that is preloaded by bolting. The bolt preload at assembled bolt torque values is equalized at approximately 6800 psi, which is the point where the face of the elbow may start to elastically separate from the riser adapter. However, additional separation due to a higher internal pressure only increases the separation by .001 inch for each 632 psi increase in pressure."

The complete riser adapter assembly is normally hydrotested in the shop to 9,000 psi prior to it being installed on the riser adaptor. During the hydrotest, the end plate is blanked off and stud bolts used. The installation on the flexjoint riser adaptor uses a threaded B8M Class 2 bolt. This is an austenitic bolt with better corrosion resistant properties than a standard B7 bolt, but a lower yield stress. The hydrotest gives confidence that the assembly is adequate for the pressure, but the final bolted joint to the riser is a less robust arrangement than that used during the shop hydrotest. Calculations show that the probability of leakage is low, provided the bolts were correctly tightened. It is not possible to inspect the bolts in-situ for residual bolt stress. No as-built records were located to verify bolt torque.

6.3.1 Mud Boost Valve

The valve assembly sits on the riser adapter of the flexjoint. The pressure boundary has been taken as the valve in the closed position for the purposes of this report.
The valve is standard EEC/HPT/T3 3 1/16" valve with hard facing seats and gates, provided in 2005/6. The valve is rated for 10,000 psi and is made from standard wellhead type materials, but with no corrosion resistant barriers or materials of construction. The actuation is by hydraulic, spring return fail closed, and the valve is provided with an additional hydraulic closure system which is intended to be a back up to assist the spring closure. It is understood that the back up system has been permanently connected, so effectively boosting the closure stroke each time the valve is closed.

The valve is regularly tested during operation and is rated to 10,000 psi. The probability of significant leakage through the valve is low as the seats and gate are hard faced. The probability of operation from closed to open is good, and the way the valve is presently configured the probability of closure is also good.

There is a possibility of external leakage during or following operation from upper or lower stem polymeric seals, but leak paths are small between shafts in metallic housings.
6.4 LMRP Package to Flexjoint Bolted Joint

The LMRP to flexjoint bolted joint is a Cameron supplied API6A style flange with a modified CX style gasket, which is a self energized metal ring type seal at the bore of the pipe. The capacity of the joint in combination with external loads has been assessed using API 6AE\textsuperscript{6}.

With the Triple-Ram stack installed there will be no additional tension on this joint, but there may be a bending moment if the flexjoint is not vertical. The load capacity of this joint is dependent on the initial bolt preload. The standard design bolt load will be between 40ksi and 52.5ksi.

Using the lower of these two figures, with a bore pressure of 6,650 psi the indicative bending moment capacity is 0.75 million ft-lb. Taking the flexjoint stiffness as 68 ft-kip/deg, this means that if the flexjoint returns to 5° rotation, then the applied bending moment will be 0.34 million ft-lb, well within the design capacity of a 6A flange.

This calculation was further refined in API6AF phase 2 report\textsuperscript{7}. This includes a hub stress criteria and a leakage criteria. The maximum bending moment for the leakage criteria with 40 ksi bolt preload, zero tension and 6,650 psi bore pressure is 0.75 million ft-lb.

- Probability of burst or leakage from this joint is low, provided that the flexjoint recovers to below 10° rotation.

\textsuperscript{6} API 6AF Technical Report on Capabilities of API Flanges under Combinations of Loading, API 2\textsuperscript{nd} Edition 1995.

\textsuperscript{7} API 6AF2 Technical Report on Capabilities of API Integral Flanges under Combinations of Loading - Phase II, API 3\textsuperscript{rd} Edition 2008.
7 Conclusions & Recommendations

This technical assurance report covers the design, fabrication and testing of the Triple-Ram stack and transition spool as well as evaluation of the complete system including the existing flexjoint and BOP. The report supplements the original design calculations with additional design reports where appropriate and assumes standard industry proven components have been designed and tested based on manufacturer's rated capacity.

Summary of results:

- After installation of the Triple-Ram stack, the system is considered structurally stable at the current inclination angle if the flexjoint is properly restrained. The weakest structural component is the transition spool. Analyses have been conducted which show the transition spool has adequate margin against plastic collapse (i.e. burst) and should be capable of short term operation provided the final inclination angle remains $\leq 3^\circ$ from vertical with all six (6) G-series flange bolts in place. This analysis does not consider additional equipment or loads applied above the Triple-Ram stack.

- A system for jacking and stabilizing the flexjoint riser adaptor has been developed and tested.

- Buckling of the transition spool is not considered credible.

- Components with the highest potential of leakage are:
  - Elastomeric element in the flexjoint.
  - Face seal O-ring assembly of the mud boost valve.

- While the elastomeric element of the flexjoint and mud boost flange assembly O-ring cannot be inspected, analyses show the flexjoint elastomeric element and mud boost face seal should remain sealed during anticipated load conditions.

The following are recommendations to be considered during the installation and operation of the Triple-Ram stack:

- Monitor the stack inclination.
- Monitor the stack for leakage.
- Maintain numerical models in as-built, inspected and operated conditions so quick evaluations can be made if operational conditions or any other conditions change.
- Inspect the flexjoint outer layer cover rubber for damage. This can be done once the debris shield is removed.
- Monitor the flexjoint extension pipe angle and the shim devices to confirm proper function.
8 Appendices

Appendix A. Summary Presentations from BP-DOE Design Review
   a. MC252 – BOP Connection Update
   b. Flange Connection Spool with Capping Stack
   c. Stress Engineering Analysis
   d. Oil States Flexjoint Overview
   e. DOE Tri-Lab Assessment

Appendix B. Summary Report Global Analysis Of Flanged Connection Option With Triple Ram BOP Assembly (For Macondo Well Intervention) 4992 Ft Water Depth, GOM, Stress Engineering Services, 28-Jun-10

Appendix C. Flexjoint Stabilization
   a. Flexjoint Jacks SIT Results, BP Document 2200-T2-DO-RP-4192
   b. Flexjoint Restraint Test Results, BP Document 2200-T2-DO-RP-4191
   c. Flexjoint Stabilizer Modeling, Oil States Industries, SV50548-248
   d. Flexjoint Holder/Pusher Cylinder Saddle Analysis, Oil States Industries, SV50548-248
   e. Flexjoint Stabilizer General Arrangement, Oil States Industries, Drawings SV50548-917 and SV50548-918.

Appendix D. Triple Ram Stack
   a. Cap Stack Design Report, Rev 1, Transocean, 27-Jun-10
   b. Function and Pressure Test Procedures for the Well Cap Assembly at Cameron Berwick Stack Pad, Transocean, 17-Jun-10
   c. Risk Assessment of Hydrate Formation on Triple Ram Capping Stack, BP, 24-Jun-10


Appendix F. G-Series Flange Information

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Warning: Check DW Docs revision to ensure you are using the correct revision.
a. VetcoGray Technical Note
b. VetcoGray Drawings
c. VetcoGray Operating and Service Procedure 14000, OSP14000C;
d. Engineering Bulletin, H0901584

Appendix G. Material and Related Test Records:
   a. Tensile test record for X80 pipe,
   b. Burst test record for X80 pipe,
   c. Hydrotant records for transition spool
   d. MTR for 18.75" API 15K flange,
   e. MTR for G-series Flange,

Appendix H. Transition Spool Integrity Assessment
   a. Structural Analysis of Spool Piece Frazer-Nash Consultancy Ltd, Spool Analysis 64053V Issue 2, June 2010
   b. Hand Calculations, Frazer-Nash Consultancy Ltd
   c. Technical Note – Bolt Torque, BP

Appendix I. Transition Spool Pipe Checks Rev B.xls, IntecSea, Inc, June 2010

MC252 – BOP Connection
Update for Science Team

23 June 2010  Trevor Smith
Topics

• Equipment

• Schedule
The Equipment

Flex Joint Alignment and Restraint Tools

- Jacking System – prototype proven
- Final Jacks SIT - June 24
- Shipment Offshore – June 26
- Restraint blocks – SIT June 29

Hydraulic jacks (50 tons) to create alignment

Passive restraint stops to hold in place
The Equipment: Jacks Trial
The Equipment

60" Chop Saw

- located on Boa Deep C
The Equipment

Torque Tools
- Subsea trials successful – tools proven

damaged bolt head – to be deburred
The Equipment

Flange Splitter

- Fitting larger cylinders
- SIT - June 24
- Shipping - June 26
The Equipment

Flange Puller
- SIT June 27
- Ship June 28

Overshot Flange Puller

Use pipe stubs to separate flanges
The Equipment

**Flange Spool**

- Fabrication essentially complete
  - Finishing guidance shoe
  - Transport stand being finalized
  - Lift Frame design/fab ongoing
- Final SIT & Handling trial – June 26
- Pressure Test – June 27
- Ship to Berwick June 27 for SIT with BOP
The Equipment

"Lasso" Drill Pipe

- Testing complete
3 Ram BOP

- Fabrication complete
- SIT Complete
- SIT with Flange Spool June 29
The Equipment

Top Hat #10

- Staged on Enterprise
The Schedule

• Equipment ready for shipment offshore – P_0- June 28
• Go/No Go for Removal of Top Hat #4 with Enterprise
  • Helix Producer in recovery mode and stable
  • Weather window assessment
  • All procedures approved and tabletop(s) complete
  • Flex joint prep work complete
  • SIMOPS identified
  • Top Hat #10 Staged on Enterprise
  • All tools in place – flange splitter; flex joint alignment and restraint system; flange puller, etc.

• Containment
  • Estimated 4.5 days of non containment to Enterprise
Flex Joint – Lower Flange Removal Feasibility

- 24 bolt API 10M flange
- 2" studs
- ROV access for tool deployment - very limited – **may not be feasible**
- high risk of seized nuts/studs
- Extremely lengthy bolt removal process, (providing achievable by ROV)
- Lengthy period of open flow
Flange Connection Spool with Capping Stack

23 June 2010

HIGHLY CONFIDENTIAL

TREX-120129.63
Capping Options

- Flex Joint Overshot: 4700 psia
- Flange Connection Spool: 9000 psia
- Latch Cap: 7250 psia

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Focus of this review

3 Ram Capping Stack

Flange Connection Spool

9000 psia
• **Overview of Design** - Mark Nichols  15mins
• **Engineering of combined stack** - Mark Nichols, Kevin Lanen/ Stress Engineering  60 mins
  - Loading conditions, Stress analysis & system Integrity
• **Review of Flex Joint** with OSI inc - David Petruska and OSI  60 mins
  - Overall design/materials of construction
  - FEA and stress analysis
  - Experience and testing to date
• **Instrumentation and Control features** Matt Gochenor  30 mins
  - P & ID
  - Pressure and Temperature transducers, location, accuracy, data transfer system and transmission to beach
  - Choke details Dave Brookes
• **Proposed method for controlling well flow and possible shut – in** - David Brookes 20 mins
  - Overall Capabilities, will not discuss how the overall process will be done
• **Update on installation techniques and testing** - Trevor Smith 30 mins
Schedule

- Equipment ready for shipment offshore – P₀ - June 28
- Go/No Go for Removal of Top Hat #4 with Enterprise
  - Helix Producer in recovery mode and stable
  - Weather window assessment
  - All procedures approved and tabletop(s) complete
  - Flex joint prep work complete
  - SIMOPS identified
  - Top Hat #10 Staged on Enterprise
  - All tools in place – flange splitter; flex joint alignment and restraint system; flange puller, etc.
- Containment
  - Estimated 4.5 days of non containment to Enterprise
Scope of Analyses

Perform global analysis - SES

Check connectors and casing - SES

Check capacity of transition spool - IntecSea

Check capacity of G-Class Flange - Vetco

Check capacity of flex joint - OSI
Horizon BOP Stack Inclination

- Stack leaning towards $310^\circ$
- Based on Roll/Pitch sensor readings and correlates with bulls eyes
- Reading Accuracy $\pm 0.5^\circ$

FJ extension Pipe leaning $4.5^\circ$

Casing, BOP, FJ housing leaning $2^\circ$
Flanged Connection Load Cases

0°

FJ Vertical

5.5° Case N1 Shimmed in Place

12° Case N2 FJ Bottoms Out

12° Case N3 FJ Bottoms Out 30' Soils Removed

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LNL004-026050

TREX-120129.71
IntecSea Spool Piece Analysis
Flange Connection Spool (FCS)

- Spool designed with new flange and seal assembly to mate with 3 ram stack

- Connected to mating flange on Horizon LMRP Flex Joint after removal of existing flange

- Design pressure is 9000 psia; subject to flex joint pressure limit with inclination

- Analysis conducted to understand structural/pressure capability at shut in pressure/temperature in rotated condition
• **Global Finite Element Analysis** provides the moment throughout the capping stack, LMRP, BOP and conductor casing for various restraint conditions of the flexjoint:
  - Riser adapter spool **restrained at 5.5 degrees** to the vertical
  - Riser adapter spool **free to move** within the +/- 10 degree range of the flexjoint

• Global analysis identified the 21.5” x 1”WT **transition spool** as a potential weak point under combined pressure load and applied moment

• **Local Finite Element Analysis** of transition spool carried out to determine the moment capacity when combined with other loads:
  - 6750psi differential pressure applied to the transition spool and flanges
  - 150kips of deadweight (current triple ram stack in-air weight = 143kips)
Moment distribution through transition spool

Distance from Mudline (ft)

Transition spool

900 kip-ft

330 kip-ft

Bending Moment (million lbf-ft)

-1.6 -1.4 -1.2 -1 -0.8 -0.6 -0.4 -0.2 0

50.0

Final angle: 6.9 degrees  Final angle: 14 degrees  Final angle: 14 degrees, 30m soil removed

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TREX-120129.77
Finite Element Model Description

- Horizontal half symmetry
- Contact surfaces on flange faces and under bolt and nut heads
- Bolt preload applied using pre-tension elements
- Fixed 2D below the G flange using a stub of un-pressurised 1” WT pipe to speed up model build

LOAD CASE: 1 Total bolt Pre tension - 6 bolts = 3750 kips
LOAD CASE: 2 Internal pressure 6750 psi
LOAD CASE 3a: AXIAL LOAD FZ = 2537.8 kips
LOAD CASE 3b: AXIAL LOAD FZ = -150 kips
LOAD CASE 4: BM UP to 900 kips-ft (10800 kips-in)

Transition spool

Riser adapter

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Finite Element Load Cases

- Maximum moment from global analysis is 900kip-ft for triple ram stack on unrestrained flexjoint – final angle of 14 degrees
- 900kip-ft applied together with pressure stresses and deadweight
- Three conservative assessment criteria:
  1. **Maximise compression stress** and check for buckling potential
      - Apply 6750psi differential, no end cap load and 150kips deadweight
  2. **Maximise tensile stress** and check against 90% yield
      - Apply 6750psi differential, and no deadweight
  3. Check for **excessive bolt load**
      - As 1 and 2.
Stress under compressive load

Bending strain < 0.83%
so no buckling at 2% ovality
(API RP1111)

Yield = 84.1ksi

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TREX-120129.80
Maximum tensile stress

Stresses are within 90% yield

Yield = 84.1 ksi

2537.8 k NET TENSION, 6.75 ksi NET PRESSURE, 10,800 inch-kip MOMENT
Bolt loads under moment loading

- G flange connection designed for a tensile load of 3million lbf carried by 6 bolts, i.e. **500kips in each bolt**
- Pressure end cap load for 6750psi differential pressure is 1.87million lbf or **312kips per bolt**
  - This leaves **188kips per bolt** for moment loading
- Conservative hand calculation – assume 2 bolts carry all moment
  - Bolt pitch circle is 30.75 inches (2.6 feet)
  - At 900kip-ft, the moment-induced bolt load is **351kips – Not OK**
  - At 330kip-ft, the moment-induced bolt load is **129kips - OK**
- Conclude that bolt loads are too high at 900kip-ft and acceptable at 330kip-ft with a reserve factor of **at least 1.4**
  - Finite element bolt loads will improve the reserve factor
Conclusions

- Transition spool assessed under a conservative range of combined moment, pressure and deadweight loads
- Results of finite element assessment
  - **Buckling** is not an issue for the loads considered
  - **Plastic collapse** is not an issue for the loads considered
  - **Bolt loads** are predicted to be the limiting factor
    - An applied moment of **900 kip-ft** corresponding to an unrestrained flexjoint is predicted to exceed the design capacity of the bolts
    - An applied moment of **330 kip-ft** corresponding to a flexjoint restrained at 5.5 degrees is within the design capacity of the bolts
- The **flexjoint should be restrained** to prevent movement beyond its current position of 5.5 degrees to the vertical thereby limiting the applied moment to 330kip-ft
Existing Flex Joint & Riser Adaptor

Riser Adaptor

G-flange

Mud Boost Valve Assembly

Flex Element (internal)

Flexjoint
G Flange

- 21" HMF- Class G Flange
  - Designed to API 16R
  - 6 bolts for axial load
    - Rated to 3,000,000 psi tension
    - Shut-in pressure creates 2,000,000 psi tension
  - Pin with 3 O-rings for pressure seal
    - Threaded into upper flange & 1 O-ring
    - Slip fit into lower flange & 2 O-rings
    - Rated to 5,000 psi, 180°F
      - Two hydrotests
        - Factory - 7500 psi for 15 min
        - Fabrication - 7500 psi for 30 min
        - Increasing pressure decreases extrusion gap
• Design for ±10° Movement in any direction
• The flex element is bonded to the body and retaining ring assembly
  - Resulting lip seal forms part of the primary pressure boundary.
• Expectation Flexjoint will return to near vertical once existing riser assembly removed
  - Analysis shows that the flex element seals against the internal pressure.
  - Possibility of leakage is considered low but possible.
Mud Boost Valve

- Mud Boost Valve
  - Rated for 10,000 psi
  - Hydraulic Open
  - Spring fail safe close (hydraulic back up)
- Mud Boost Valve Flange
  - Rated for 10,000 psi
- Elbow Face Seal
  - 4 Bolts and O-ring
  - 6,600 psi at bolt preload at torque values
  - Additional 632 psi increase for each 0.001" of separation
• Pressure Containment Assessment of DWH Flex joint, G flange & mud boost valve assembly
  - Low probability of burst failure
  - Low probability of significant leakage
Horizon Intervention
Flanged Connection Option w/ 3-ram BOP Stack
4992 ft Water Depth (Macondo Site)
Atul Ganpatye, Kenneth Bhalla
June 20, 2009

- Updated 3-Ram BOP Weight Included
- Gooseneck Loads Included

DRAFT

Taking on your toughest technical problems

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TREX-120129.95
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Updated Data and Key Analysis Inputs

- **Gooseneck Loads:**
  - Moment: 38.7 ft-kip
  - Vertical Load: 6.31 kip
  - Horizontal Load: 1 kip

- **3-Ram BOP Weight (including HC connector):**
  - \( 142,201 \text{ lbf} + 830 \text{ lbf (mini connectors)} = 143,031 \text{ lbf} \)

- **Initial Casing Inclination Angle with Vertical = 2°**

- **Initial Lower Flex Joint Angle = 3.5° (includes 0.5° excursion to account for uncertainty in measurement, and an additional 0.5° to account for shim tolerance)**
  - Thus, the total angle initial inclination of the 3-Ram BOP with vertical would be 5.5°
Updated 3-Ram BOP Weight

### 3 Ram BOP Weight Breakdown

<table>
<thead>
<tr>
<th>Component</th>
<th>Weight (lbs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Class F Riser Box</td>
<td>6,000 lbs</td>
</tr>
<tr>
<td>Yelco EvF Connector</td>
<td>17,200 lbs</td>
</tr>
<tr>
<td>Mandrel</td>
<td>6,000 lbs</td>
</tr>
<tr>
<td>Upper Frame</td>
<td>8,634 lbs</td>
</tr>
<tr>
<td>Double Ram Block</td>
<td>54,066 lbs</td>
</tr>
<tr>
<td>Lower Frame</td>
<td>1,484 lbs</td>
</tr>
<tr>
<td>Single Ram Block</td>
<td>29,262 lbs</td>
</tr>
<tr>
<td>Pancake Flange</td>
<td>3,100 lbs</td>
</tr>
<tr>
<td>Cameron HC Connector</td>
<td>23,100 lbs</td>
</tr>
<tr>
<td>4 x Valves</td>
<td>11,340 lbs</td>
</tr>
<tr>
<td>Pipework</td>
<td>0 lbs</td>
</tr>
<tr>
<td>ROV Panels</td>
<td>3,000 lbs</td>
</tr>
<tr>
<td><strong>Total Shipping Weight</strong></td>
<td><strong>175,201 lbs</strong></td>
</tr>
<tr>
<td><strong>Crane Lift Weight</strong></td>
<td><strong>198,401 lbs</strong></td>
</tr>
<tr>
<td><strong>Minus Skid</strong></td>
<td><strong>142,201 lbs</strong></td>
</tr>
</tbody>
</table>

### Center of Gravity Calculations

<table>
<thead>
<tr>
<th>Component</th>
<th>Weight (lbs)</th>
<th>Center of Gravity (ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mandrel</td>
<td>5,995 lbs</td>
<td>20.09 ft</td>
</tr>
<tr>
<td>Upper Frame</td>
<td>8,634 lbs</td>
<td>23.03 ft</td>
</tr>
<tr>
<td>Double Ram Block</td>
<td>54,066 lbs</td>
<td>20.09 ft</td>
</tr>
<tr>
<td>Lower Frame</td>
<td>1,484 lbs</td>
<td>17.36 ft</td>
</tr>
<tr>
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<td>17.60 ft</td>
</tr>
<tr>
<td>Pancake Flange</td>
<td>3,100 lbs</td>
<td>17.60 ft</td>
</tr>
<tr>
<td>Cameron HC Connector</td>
<td>23,100 lbs</td>
<td>17.60 ft</td>
</tr>
<tr>
<td>Pressure Test SPOOL</td>
<td>7,000 lbs</td>
<td>14.21 ft</td>
</tr>
<tr>
<td><strong>Total Shipping Weight</strong></td>
<td><strong>175,201 lbs</strong></td>
<td><strong>9.13 ft</strong></td>
</tr>
<tr>
<td><strong>Total Weight</strong></td>
<td><strong>198,401 lbs</strong></td>
<td><strong>99.2 Tons</strong></td>
</tr>
<tr>
<td><strong>Crane Lift Weight</strong></td>
<td><strong>142,201 lbs</strong></td>
<td><strong>71.1 Tons</strong></td>
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</table>

**Mini Connectors not included in this table**

### Center of Gravity on Skid

<table>
<thead>
<tr>
<th>Component</th>
<th>Weight (lbs)</th>
<th>Center of Gravity (ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mandrel</td>
<td>5,995 lbs</td>
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</tr>
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</tr>
<tr>
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<td>17.36 ft</td>
</tr>
<tr>
<td>Single Ram Block</td>
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<td>17.60 ft</td>
</tr>
<tr>
<td>Pancake Flange</td>
<td>3,100 lbs</td>
<td>17.60 ft</td>
</tr>
<tr>
<td>Cameron HC Connector</td>
<td>23,100 lbs</td>
<td>17.60 ft</td>
</tr>
<tr>
<td>Pressure Test SPOOL</td>
<td>7,000 lbs</td>
<td>14.21 ft</td>
</tr>
<tr>
<td><strong>Total Weight</strong></td>
<td><strong>198,401 lbs</strong></td>
<td><strong>99.2 Tons</strong></td>
</tr>
<tr>
<td><strong>Crane Lift Weight</strong></td>
<td><strong>142,201 lbs</strong></td>
<td><strong>71.1 Tons</strong></td>
</tr>
</tbody>
</table>

### Center of Gravity on LMRP

<table>
<thead>
<tr>
<th>Component</th>
<th>Weight (lbs)</th>
<th>Center of Gravity (ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mandrel</td>
<td>5,995 lbs</td>
<td>20.09 ft</td>
</tr>
<tr>
<td>Upper Frame</td>
<td>8,634 lbs</td>
<td>23.03 ft</td>
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<tr>
<td>Double Ram Block</td>
<td>54,066 lbs</td>
<td>20.09 ft</td>
</tr>
<tr>
<td>Lower Frame</td>
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<td>17.36 ft</td>
</tr>
<tr>
<td>Single Ram Block</td>
<td>29,262 lbs</td>
<td>17.60 ft</td>
</tr>
<tr>
<td>Pancake Flange</td>
<td>3,100 lbs</td>
<td>17.60 ft</td>
</tr>
<tr>
<td>Cameron HC Connector</td>
<td>23,100 lbs</td>
<td>17.60 ft</td>
</tr>
<tr>
<td>Pressure Test SPOOL</td>
<td>7,000 lbs</td>
<td>14.21 ft</td>
</tr>
<tr>
<td><strong>Total Weight</strong></td>
<td><strong>198,401 lbs</strong></td>
<td><strong>99.2 Tons</strong></td>
</tr>
<tr>
<td><strong>Crane Lift Weight</strong></td>
<td><strong>142,201 lbs</strong></td>
<td><strong>71.1 Tons</strong></td>
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LNL004-026078

TREX-120129.99
## Component Details

<table>
<thead>
<tr>
<th>Component Name</th>
<th># of Joints</th>
<th>Length (ft)</th>
<th>Distance of Flare</th>
<th>Outer Diameter (in)</th>
<th>Wall Thickness (in)</th>
<th>Hydrodynamic Drag Diameter (in)</th>
<th>Base Joint Weight (lb)</th>
<th>Component Joint Dry Weight (lb)</th>
<th>Component Joint Wet Weight (lb)</th>
<th>Skew Weight per Unit Length (lbs/ft)</th>
<th>Wall Weight per Unit Length (lbs/ft)</th>
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<tr>
<td>Low Pressure Housing</td>
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<td>3.47</td>
<td>5.0</td>
<td>37.26</td>
<td>3.22</td>
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<td>High Pressure Housing</td>
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<td>24.05</td>
<td>40.7</td>
<td>50.00</td>
<td>15.625</td>
<td>214</td>
<td>306715</td>
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<td>Horizon LMR®</td>
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<td>56.8</td>
<td>50.00</td>
<td>15.625</td>
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<tr>
<td>Riser Adaptor</td>
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<td>22</td>
<td>1401</td>
<td>1401</td>
<td>1253</td>
<td>219.4</td>
<td>192.9</td>
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<td>Spool</td>
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<td>5.92</td>
<td>70.0</td>
<td>21.50</td>
<td>1.0000</td>
<td>22</td>
<td>8800</td>
<td>8800</td>
<td>7021</td>
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<tr>
<td>Mandrel</td>
<td>1</td>
<td>1.43</td>
<td>71.4</td>
<td>32.00</td>
<td>6.2500</td>
<td>32</td>
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<td>9000</td>
<td>6500</td>
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<td>3-RAM BOP HC Connector</td>
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<td>75.1</td>
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<td>104530</td>
<td>8383.7</td>
<td>7298.2</td>
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</tbody>
</table>

Total Weight of the 3-Ram BOP Assembly (3-Ram BOP, HC Connector, Spool, and Mandrel) w/o Mini Connectors = 157,691 lbf

Total Weight of the 3-Ram BOP Assembly (3-Ram BOP, HC Connector, Spool, and Mandrel) w/ Mini Connectors = 158,521 lbf

Used in the Analysis

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Application of Gooseneck Loads

Fx = 1,000 lbf
M = 38,700 ft-lbf
Fy = 6,310 lbf
## Summary of Results

<table>
<thead>
<tr>
<th>Case #</th>
<th>Initial Angle of Inclination of the casing with the vertical (deg)</th>
<th>Penetration Depth below Mudline over which soil is removed (ft)</th>
<th>Max von Mises Stress in the 36&quot; x 2&quot; (X-65) Casing (ksi)</th>
<th>Flex Joint Angle (deg)</th>
<th>Final Angle of Inclination of the Horizon BOP with Vertical (deg)</th>
<th>Final Angle of Inclination of the 3-ram BOP with Vertical (deg)</th>
<th>Bending Moment at Lower Flex Joint (10^6 ft-lbf)</th>
<th>Bending Moment at Lower Flex Joint (10^6 ft-lbf)</th>
<th>Bending Moment at Lower Flex Joint (10^6 ft-lbf)</th>
<th>Wellhead Bending Moment (10^6 ft-lbf)</th>
</tr>
</thead>
<tbody>
<tr>
<td>CASE N1</td>
<td>2</td>
<td>0</td>
<td>16.0</td>
<td>3.5</td>
<td>3.1</td>
<td>6.9</td>
<td>0.33</td>
<td>0.47</td>
<td>0.71</td>
<td>1.64</td>
</tr>
<tr>
<td>CASE N2</td>
<td>2</td>
<td>0</td>
<td>19.0</td>
<td>10</td>
<td>3.3</td>
<td>14.0</td>
<td>0.61</td>
<td>0.90</td>
<td>1.17</td>
<td>2.18</td>
</tr>
<tr>
<td>CASE N3</td>
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<td>30</td>
<td>25.4</td>
<td>4.0</td>
<td>14.7</td>
<td>0.64</td>
<td>0.94</td>
<td>1.26</td>
<td>2.49</td>
<td></td>
</tr>
</tbody>
</table>

Capacity checked by others

- **HC connector at the Bottom of Horizon LMRP**
  - ✔ Working Capacity = 2.2 million ft-lbf (for 0 tension @10 ksi bore pressure)
- **SHD-H4 Connector (w/ H4 adapter kit) at the Bottom of Horizon BOP**
  - ✔ Working Capacity = 5.0 million ft-lbf (for 0 tension @15 ksi bore pressure)
Lower Flex Joint Behavior

For 2º casing inclination with vertical, flex joint is expected to bottom out, if not supported.

For 1º casing inclination with vertical, resisting moment provided by the undamaged flex joint is higher than the moment due to the weight of the components located above it. However, values are very close – and therefore, probably sensitive to any further disturbances or loadings.

(Note: These results do not include gooseneck load)
Lateral Displacement Profile

Initial and Final Lateral Displacement Profiles for Cases N1, N2, and N3

Angle of inclination of 3-rim BOP with vertical = 14.0 deg

Lower flax joint angle:
Case N1: 9.0 deg
Case N2 and N3: 1.0 deg

Distance from Mudline (ft)

Lateral Displacement (ft)

Case N1 Initial Configuration
Case N1; Final Configuration
Cases N2 and N3; Initial Configuration
Case N2; Final Configuration
Case N3; Final Configuration

Note: Angles shown on the graph are approximations.
Bending Moment Distribution: Cases N1, N2, and N3

- Bottom of Horizon LMRIP (HC Connector): 1.26 million ft-lbf
- Bottom of Horizon BOP (SHD-H4 Connector): 2.49 million ft-lbf
- Lower Flex Joint: -0.04 million ft-lbf

Distance from Mudline (ft)

Bending Moment (million ft-lbf)

- Bending Moment Distribution; CASE N1; Final Configuration
- Bending Moment Distribution; CASE N2; Final Configuration
- Bending Moment Distribution; CASE N3; Final Configuration

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Stress in 36" x 2" (X-65) Casing – Case N1

Stress Distribution in the 36" x 2" (X-65) Casing for Cases N1, N2, and N3

von Mises Stress (ksi)

Penetration Depth below Mudline (ft)

Max von Mises Stress = 16.0 ksi

Max von Mises Stress = 25.4 ksi

Max von Mises Stress = 19.0 ksi

Case N1 ——— Case N2 ——— Case N3

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Contact Information

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Houston, Texas 77041

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Fax: (281) 955-2638
www.stress.com
Supplementary Information
Additional Assumptions

- **Global** ABAQUS model of: lower flex joint, components of the free-standing stack system (3-ram BOP, Horizon BOP and LMRP), casing, wellhead, wellhead connector and soils
- Planar response
- Lateral P-Y Soil response
- No dynamic effects
- Linear elastic material model
Horizon Stack Used in the Analysis

BOP + Connector Length = 27.7 ft – from drawing alongside (excluding connector swallow);
BOP + Connector Weight = 335.3 kips
LMRP + Riser Adapter Length = 23.4 ft;
LMRP + Riser Adapter Weight = 290.0 kips

Source:
Drawing provided by BP in 2003
Mandrel Used in the Analysis
<table>
<thead>
<tr>
<th></th>
<th>SS-15 HP Housing</th>
<th>SS-15 LP Housing</th>
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</thead>
<tbody>
<tr>
<td>OD (in)</td>
<td>27.00</td>
<td>36.00</td>
</tr>
<tr>
<td>ID (in)</td>
<td>18.51</td>
<td>33.00</td>
</tr>
<tr>
<td>Length (ft)</td>
<td>6.63</td>
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</tr>
<tr>
<td>Weight (lbf/ft)</td>
<td>1189.6</td>
<td>553.2</td>
</tr>
<tr>
<td>Component Name</td>
<td>Length (ft)</td>
<td>Top from Mudline (ft)</td>
</tr>
<tr>
<td>----------------</td>
<td>------------</td>
<td>----------------------</td>
</tr>
<tr>
<td>36&quot; x 2&quot; (X-65)</td>
<td>121</td>
<td>6.53</td>
</tr>
<tr>
<td>36&quot; x 1.5&quot; (X-56)</td>
<td>166</td>
<td>-114</td>
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<tr>
<td>28&quot; x 0.75&quot; (X-52)</td>
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<td>0.00</td>
</tr>
<tr>
<td>22&quot; x 1.0&quot; (X-80)</td>
<td>286</td>
<td>6.38</td>
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Soil Data

Lateral resistance-deflection relationship comparison
(shown for example at 10 ft penetration depth)

<table>
<thead>
<tr>
<th>Penetration Depth (ft)</th>
<th>OLD Undrained Shear Strength (psf)</th>
<th>NEW Undrained Shear Strength (psf)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>30</td>
<td>60</td>
</tr>
<tr>
<td>10</td>
<td>50</td>
<td>100</td>
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<tr>
<td>100</td>
<td>700</td>
<td>1400</td>
</tr>
<tr>
<td>280</td>
<td>2000</td>
<td>4000</td>
</tr>
</tbody>
</table>

Used in the analysis in conjunction with API 2A-WSD guidelines for generating short term-static P-Y curves

(Note: values at 280 ft penetration depth are linearly extrapolated from values given at 10 ft and 100 ft)
Load Capacity Envelope: SS15 Wellhead System

RATED LOAD CAPACITIES FOR
THE DRIL-QUIP 18-3/4" SS-15 OR SS-10 (27" MANDREL)
WELLHEAD SYSTEM WITH RIGID LOCKDOWN
SAFETY FACTOR = 1.5
"AT YIELD" VALUES ARE HIGHER

RATINGS ARE BASED ON FEA CALCULATIONS WITH:
1. PRESSURE AREA IS EQUAL TO 18-3/4" FULL BORE
   (MID SHEAR RAMS CLOSED 0 PSI THROUGH 4000 PSI)
   PRESSURE AREA EQUAL TO 8.50" BORE X 5000-15000 PSI
2. RADIUS FOR BENDING MOMENT TAKEN
   AT LOCK RING FACE
3. IF WELLHEAD IS EMPTY MAX STRESS OCCURS AT 34" GROOVES
   OR 18-3/4" LANDING RING
   MAXIMUM STRESS OCCURS IN THE WELLHEAD TOP PROFILE.
4. MAXIMUM ALLOWABLE MEMBRANE STRESS = 67% OF YIELD
5. CAPACITIES WILL CHANGE IF CASING WEIGHT HANGING
   IN THE WELLHEAD OR PRESSURE AREA CHANGES.
6. PRELOAD = 1 MILLION LBS.
7. ANY CASING SIZE OR PRESSURE LOAD CASE GENERATES NO
   MORE THAN 1 MILLION LBS. TENSION.
8. CONDUCTOR HOUSING = 90 KSI YIELD
9. 18-3/4" LANDING RING = 140 KSI YIELD

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Wellhead Capacity

- High and Low Pressure Housing Wellhead is Dril-Quip SS-15 System
  - SS-15 is rated for 4 million ft-lb (15-ksi pressure and 1500-kips tension) at 2/3 yield: Working Capacity
  - SS-15 is rated for 6 million ft-lb (15-ksi pressure and 1500-kips tension) at yield: Survival Capacity
HD-H4 Connector or SHD-H4 with adapter kit - Capacity

Based on Typical "H-4" Profile Connector at 2/3 Yield
➢ @ 15 ksi pressure, 1500 kips tension: 4.0 million ft-lbf

Based on Typical "H-4" Profile Connector at Yield
➢ @ 15 ksi pressure, 1500 kips tension: 6.0 million ft-lbf

18-3/4" HD H-4 connector
18-3/4" HD H-4 connector shown with studded top and locking dogs engaged on MS-4002 2" outside diameter wellhead. Available with flanged top.

Technical specifications: 18-3/4" HD H-4

<table>
<thead>
<tr>
<th>Specification</th>
<th>Measurement</th>
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<tbody>
<tr>
<td>Bending load capacity</td>
<td>3,400 kips</td>
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<tr>
<td>Project</td>
<td>620 kips</td>
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<tr>
<td>Hydraulic capacity</td>
<td>19.20 US GPD</td>
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<tr>
<td>Lock fluid volume</td>
<td>15.16 US GPD</td>
</tr>
<tr>
<td>Overall diameter</td>
<td>52 ft</td>
</tr>
<tr>
<td>Weight</td>
<td>2.56.30 lbs</td>
</tr>
<tr>
<td>Size</td>
<td>32.25 in</td>
</tr>
<tr>
<td>Max. service pressure</td>
<td>15,000 psi</td>
</tr>
<tr>
<td>Max. test oper. pressure</td>
<td>3,000 psi</td>
</tr>
</tbody>
</table>
Cameron HC Connector

### Performance

<table>
<thead>
<tr>
<th></th>
<th>Model 70</th>
<th>HC</th>
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<tbody>
<tr>
<td>Pressure Rating</td>
<td>10,000/69.00</td>
<td>10,000/69.00</td>
</tr>
<tr>
<td>Bending @ 0 pressure, 0 tension* (H-lb)</td>
<td>3,645,000</td>
<td>4,048,000</td>
</tr>
<tr>
<td>Bending @ 0 pressure, 0 tension* (Nm)</td>
<td>4,342,000</td>
<td>5,488,000</td>
</tr>
<tr>
<td>Bending @ 10,000 psi, 0 tension* (H-lb)</td>
<td>1,796,000</td>
<td>2,319,000</td>
</tr>
<tr>
<td>Bending @ 10,000 psi, 0 tension* (Nm)</td>
<td>2,313,000</td>
<td>3,036,000</td>
</tr>
<tr>
<td>Bending @ 10,000 psi, 2,000,000 lb tension* (H-lb)</td>
<td>549,000</td>
<td>2,063,000</td>
</tr>
<tr>
<td>Bending @ 10,000 psi, 2,000,000 lb tension* (Nm)</td>
<td>744,000</td>
<td>2,797,000</td>
</tr>
<tr>
<td>Bending @ 15,000 psi, 2,000,000 lb tension* (H-lb)</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Bending @ 15,000 psi, 2,000,000 lb tension* (Nm)</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

For the purpose of this analysis, the bending moment capacity at 10 ksi pressure and 0 tension is assumed to be applicable.

Per the graph below, this is "at yield" value and hence representative of survival capacity.

This gives survival capacity of HC connector as 3.3 million ft-lb.

---

Source: Cameron Collet Connector Brochure

---

HIGHLY CONFIDENTIAL

TREX-120129.119
Sensitivity: Load Applied at Gooseneck Connection

Data provided by BP:

<table>
<thead>
<tr>
<th>9&quot;-15K API Flange @ 0-lbs Tension</th>
<th>Bending Capacity</th>
<th>Max Moment Induced from 180 deg gooseneck from chart below</th>
<th>Left over capacity of flange</th>
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<tbody>
<tr>
<td>0ksi</td>
<td>45000 ft-lbs</td>
<td>22,200 ft-lbs</td>
<td>22,800 ft-lbs</td>
</tr>
<tr>
<td>10ksi</td>
<td>28000 ft-lbs</td>
<td>22,200 ft-lbs</td>
<td>5,800 ft-lbs</td>
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<td>15ksi</td>
<td>14000 ft-lbs</td>
<td>22,200 ft-lbs</td>
<td>-8,200 ft-lbs</td>
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</table>
API 18-3/4" 15K Flange

Reference: Technical Report on Capabilities of API Integral Flanges Under Combination of Loading—Phase II  
API TECHNICAL REPORT 6AF2  
THIRD EDITION, SEPTEMBER 2008

Page 110
Prior to 1976, ball and socket type connectors were used in the offshore industry to create a flexible connection.

- Considered high maintenance and were unreliable.

Flexible joint connection requirements:

- High pressure fluid containment.
- Decoupling of the vessel motions from the riser and BOP stack with minimum transfer of bending stresses.
- High axial compression load resistance from riser tensions and internal pressure.

OSI introduced the first FlexJoint in 1976 for this application.

- Relies on the use of a spherically shaped, laminated elastomeric bearing.
- Since, over 1000 Drilling Riser FlexJoints are in use world-wide in the offshore industry.
Oilfield Laminated Bearing Applications

HYBRID RISER TOWERS

TLP TENDONS

DRILLING RISERS

STEEL CATENARY RISERS

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Laminated Elastomeric Bearings (Flex Element)

- A composite lamination of elastomer and typically metal reinforcements
- Supports high axial loads while allowing low stiffness rotational displacement
  - High bulk modulus but low shear modulus
- Examples include:
  - Bridge Bearings
  - Seismic Bearings
  - Helicopter Rotor Bearings
  - Solid Rocket Motor Nozzle Bearings
  - Off-shore Spherical Bearings (FlexJoints®)
  - Sound isolation couplings for the US Navy
Laminated Bearings
The Subsea FlexJoint is located at the top of the LMRP and couples the drilling riser to the BOP. The Subsea FlexJoint that is in place on the Horizon BOP was designed to meet the following requirements:

- Operating depths of 10,000 feet,
- Working pressure of 5000 psi
- Hydro-test pressure of 7500 psi
- Working axial riser tension of 2,000 kips
- Angular deflections of ± 10 degrees.
Subsea FlexJoint Testing

During the development stages of the Subsea FlexJoint, extensive testing under review from third party agencies such as DNV, ABS, and Lloyds were completed.

- The original Drilling Riser FlexJoint was cycled 300,000 times before entering service in 1976. The unit was in service for 11 years before successfully refurbished.

- Other OSI FlexJoints have been tested over 11,000,000 cycles without leakage or failure.

- Today, all of OSI's Drilling Riser FlexJoints have received third party design verification.
FlexJoint Design

- No failure has occurred in the 34 year history of Drilling Riser FlexJoints due to normal operating conditions.

- Thus, the primary mode of failure is considered the loss of pressure seal integrity via the primary seals:
  - API BX or CX ring joint gasket
  - Lip Seal
  - Ring groove insert O-Ring seal
  - Flex Element
  - Test port plug

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Lip Seal
- Over mold of the Flex Element elastomer that creates a seal between the Flex Element and the Body/Housing
  - Energized by riser tension and internal pressure
  - Has demonstrated to be a very effective seal with no evidence of extrusion even at hydro-test pressure conditions

Ring Groove Insert O-Ring
- Face seal using a 90 durometer nitrile O-ring material
  - Face seals do not provide a gap that could lead to O-ring extrusion and thus are inherently safe

Test Port plug
- Metal NPT threaded plug that passed the hydrotest
Flex Element

- Sizing of the flex element (laminated bearing) is performed based upon the elastomers capability

- Significant amount of surface area required to fail in the flex element before a leak path could develop

- As witnessed during qualification testing, continuous compressive load reduces the risk of extensive leakage
  - Even if elastomer has been fully compromised through the Flex Element
Structural assessment was completed with non-linear finite element analysis

- Model
  - 2D, Asymmetric-Axisymmetric finite element analysis using ABAQUS
    - CAXA ABAQUS element allows asymmetric loading to be applied to axisymmetric model
    - Useful for models that include rotation of the extension

- Material Properties
  - Linear, elastic / perfectly plastic metallic material properties
  - Non-linear, hyperelastic elastomer material properties
## Materials

<table>
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<tr>
<th>COMPONENT</th>
<th>MATERIAL</th>
<th>SPEC REQUIRED</th>
<th>AS-BUILT</th>
<th>DUROMETER</th>
<th>INCLUDED IN FEA</th>
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<tr>
<td>BODY/HOUSING</td>
<td>4130</td>
<td>60 KSI</td>
<td>78 KSI</td>
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<tr>
<td>BACK FLANGE</td>
<td>4130</td>
<td>80 KSI</td>
<td>91 KSI</td>
<td>N/A</td>
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<tr>
<td>RETAINER FLANGE</td>
<td>4130</td>
<td>80 KSI</td>
<td>82 KSI</td>
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<td>EXTENSION / NIPPLE</td>
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<td>80 KSI</td>
<td>83 KSI</td>
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<tr>
<td>WEAR SLEEVE / WEAR RING</td>
<td>4140</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
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<td>RING GROOVE INSERT</td>
<td>4130</td>
<td>60 KSI</td>
<td>85 KSI</td>
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<tr>
<td>RETAINER FLG STUDS</td>
<td>ASTM A193 B7</td>
<td>105 KSI</td>
<td>123 KSI</td>
<td>N/A</td>
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<tr>
<td>FLEX ELEMENT REINFORCEMENTS</td>
<td>8630</td>
<td>95 KSI</td>
<td>103 KSI</td>
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<td>FLEX ELEMENT ELASTOMER</td>
<td>NBR (OSI COMPOUND)</td>
<td>N/A</td>
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<td>YES</td>
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<td>LIP SEAL</td>
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<td>N/A</td>
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<td>RING GROOVE INSERT O-RING</td>
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<td>N/A</td>
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<td>SEA WATER CONTAINEMENT O-RING</td>
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<tr>
<td>GREASE CONTAINEMENT O-RING</td>
<td>BUNA-N</td>
<td>N/A</td>
<td>N/A</td>
<td>70</td>
<td>NO</td>
</tr>
</tbody>
</table>
Load Cases

- Design Load Cases
  - 5000 psi / 2000 kips / 10 degrees
  - 0 psi / 2500 kips / 10 degrees
  - 7500 psi

- Post-Incident Load Cases
  - Case 1 and 2 were based upon operating conditions after the incident occurred
  - Case 3 & 4 are potential future operating cases following corrective actions

- All load cases were provided by BP

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>POST-INCIDENT LOAD CASES</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1</td>
</tr>
<tr>
<td>PRESSURE</td>
<td>500 PSI</td>
</tr>
<tr>
<td>ANGLE</td>
<td>10°</td>
</tr>
<tr>
<td>RISER TENSION</td>
<td>0</td>
</tr>
<tr>
<td>TEMPERATURE</td>
<td>220°F</td>
</tr>
</tbody>
</table>
FEA showed all metal components to remain well below their yield strengths for all additional load cases.

Evaluation of the lip seal indicated that the seal contact pressures remained above the internal pressure for all load cases.
- Indicates that seal integrity will be maintained
- Key dimensions were confirmed to be within drawing tolerances

Ring groove insert seal was not included in the analysis.
- Face seal design is inherently safe from O-ring extrusion
- Ring groove insert is trapped between the FlexJoint and BOP mating surface preventing joint displacement that could lead to O-ring extrusion
Post-incident Load Case 1
Post-incident Load Case 2

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TREX-120129.141
Post-incident Load Case 3
Post-incident Load Case 4

OSI CONFIDENTIAL

HIGHLY CONFIDENTIAL

TREX-120129.143
Lip Seal Pressure – LC#4

Normalized distance along path

HIGHLY CONFIDENTIAL

OSI CONFIDENTIAL

TREX-120129.148
Flex Element elastomer

- Load cases 1 & 2 does not present a concern due to the low operating pressure.

- Load cases 3 & 4 present a higher risk for elastomer creep damage due to the high pressure and temperature conditions.

- However, damage would begin on side of the element exposed to the seawater allowing for ROV inspection.
Observations

All ROV images to date show no visible leakage from the FlexJoint.

After the incident, the FlexJoint extension was rotated to the maximum angle by the collapsed riser:
- Remained at this angle until the riser was cut;
- The extension is slowly returning back to its neutral position (0° rotation);
  - Full recovery to its neutral position may never occur unless forced back by an external load;
  - Recovery is slowed by cooler temperatures and the amount of set in the elastomer.
DOE Tri-Lab Assessment of BP Flange Connector Spool & 3 Ram Capping Stack

June 26, 2010

Los Alamos National Laboratory
Sandia National Laboratories
Lawrence Livermore National Laboratory
Assessment Report Outline

• This report will follow the same outline as the Flange Connector Spool Assembly and 3 Ram Capping Stack Design Review which was held at BP on June 23, 2010
  – Overview of Design
  – Engineering of Combined Stack
  – Review of Flex Joint (+ Transition Spool & Mud Boost Valve)
  – Instrumentation and Control
  – Installation Techniques and Testing

• Following the observations and comments in each technical area, we will include:
  – DOE Tri-Lab Team Findings and Recommendations
  – Summary
Overview of Design

• Various worst case loads were considered; not all possibilities were presented (they are available and should inserted into the record); we conclude that the likely load case space has been adequately explored.

• **Recommendation #1:** Ensure that an analysis of the allowable worst case combination of loads (eg. largest tilt, highest temperature, highest pressure) is included in the record.

• It should be noted that the margins of the design are highly sensitive to differential pressure

• **Recommendation #2:** Measures should be taken to limit maximum pressure during well integrity testing and well shut-in operations.
  
  – Limiting valve closing increments
  
  – Limiting rates of closure and thereby limiting maximum pressure
Overview of Design (Cont’d)

• The current design as shown will be adequate given the expressed limits on loading conditions (maximum 5.5 degree flex joint total tilt, etc.).
  – Requires lateral support of flex joint
  – Finding #1: – BP must monitor the tilt of the combined stack to ensure that the induced load limit is never exceeded

• Recommendation #3: Given the many possible loading conditions, supporting analysis capability should remain engaged to quickly evaluate actual operational conditions as events unfold.

• The proposed pressure and temperature instrumentation are adequate and provide sufficient redundancy for the criteria of a single pressure measurement.
Overview of Design (Cont’d 2)

• Installation techniques and testing as presented appear feasible and realistic but this work is ongoing and warrants additional review and evaluation.

• **Recommendation #4:** Although not presented during the design review, BP’s evaluation of the ability of all added components to withstand the maximum expected internal pressure should be included for the record (e.g. as-built documentation to ensure proper bolt pre-load).

• These conclusions are also based on the stated three-month design life; if operation of this equipment extends beyond this time then other issues of corrosion, fatigue, etc. will require investigation and evaluation.
Engineering of Combined Stack

- The beam element FE global model for the total stack is appropriate to determine overall deflections & applied moments.
- **Recommendation #5:** An elastic/plastic finite element model should be run to predict ultimate failure mode and pressure of the stack system (e.g., Flex Joint, G Flange and Transition Spool).
- Connectors & casing strings were evaluated under worst case load combinations and found to be acceptable with generous design margins remaining.
- **Finding #2:** Since overpressure can lead to catastrophic failure, the selection of the burst disk must be approved by the DOE team.
Engineering of Combined Stack (Cont’d)

• G-class flange (the flanges joining the flex joint and the flex joint connector spool) limitations are the connecting bolts.
  
  – Finding #3: Detailed bolt analysis (material property, preload and torque information, etc.) was not presented - only individual bolt load capability was available. This information must be provided and inserted into the record.

  – Assuming the stated 500 kips load capability per bolt is valid, at least two bolts are available to carry the moment, the flex joint is restrained to a 5.5 degrees or less total tilt and the differential pressure is limited to 6750 psi, the joint should be adequate.
    • If any of these qualifying conditions are not met, then the adequacy of this flange connection is in question.

  – Recommendation #6: a calculation should be performed prior to installation in order to determine the effect of having five or fewer bolts restraining the flange and/or increased differential pressure.
Review of Flex Joint

- Oil States Inc. (OSI), manufacturer of the flex joint, presented design, analysis, and testing of the current flex joint, as well as extensive field experience based upon 30+ years of fabricating designs that rely on similar spherically-shaped laminated elastomeric bearings.
- OSI presented detailed finite element analyses that included appropriate constitutive elements (elastic-plastic elements for metal components and hyper-elastic elements for elastomeric components)
  - Model results indicated no leakage under worst case conditions
Review of Flex Joint (Cont’d)

- The use of restraining shims to limit tilt will nullify the original capabilities of the flex joint to accommodate drilling riser and drill string movement
  - **Recommendation #7**: Connection of the combined BOP stack to collection operations vessels on the surface should recognize new limitations in the Flex Joint.

- **Recommendation #8**: Use of shims to limit rotation of the flex joint will alter the load path and requires further analysis (including installation loads on bolts) prior to installation.

- Future operations that could exceed the assumed loads must be analyzed prior to application.
Review of Transition Spool

- Transition spool design is stressed beyond standard engineering practice (to 90% of yield strength) under several of the presented loading cases; the compressive loading case is noted as being especially conservative. However, the stated limits on stack tilt will effectively lower the stress but not to within normal engineering practice (typically 2/3 of tensile yield strength).
  - It is noted that the adequacy of this component is highly sensitive to differential pressure.
- **Finding #4:** Since the Transition Spool is expected to be stressed near yield, BP must either present relevant quality data (material certification / inspection records and manufacturing) for the as-built component, or perform proof testing to demonstrate margin.
Review of Mud Boost Valve

- Mud Boost face seal has a potential leak vulnerability
- Bolt pre-load is estimated to be overcome at 6600 psig
- Variations in bolt installation pre-load could reduce this pressure
- O-ring damage may occur if bolt pre-load is overcome
  - This could result in a persistent leak
- **Recommendation #9:** Installed bolt torque should be confirmed, if possible via documentation of the as-built configuration.
- **Recommendation #10:** BP should observe to watch for leaks during any high pressure operation.
- **Recommendation #11:** In addition, BP should consider injecting a pressure sensitive sealant into the Mud Boost valve in order to reduce the possibility of leakage at the flange.
Instrumentation and Control

- Doubly redundant pressure measurement below the 3 Ram Capping Stack (mini-BOP) is appropriate to meet data requirements

- **Recommendation #12** - Generate a complete system level analysis of the estimated accuracy of pressure measurement

- **Recommendation #13** - Once a correlation between the three pressure measurements (at several different pressures) is made, transmit only one of the three pressure measurements so that higher frequency sampling rate can be obtained.
  
  - Sampling of a single gauge should be done at a rate of at least once every 5 seconds.
  
  - We recommend sampling the three gauges nominally once every 15 minutes to confirm there is no drift.
Installation Techniques & Testing

- Story boarding and land/pool testing is a best practice of great value and should continue.

- **Recommendation #14:** Attention should be given to potential bolt issues during riser flange removal and FCS installation such as tool access/engagement, possible need for captive nut replacement, and alternate means of providing flange clamping force.

- Planned chop saw approach for cutting riser from a variety of angles appears to be a robust approach and should be pursued.

- No attempt should be made to extract either of the drillpipes unless circumstances are significantly altered by future events. Review will be required at such time.
Installation Techniques & Testing (Cont’d)

• Consider using the “plume-inator” during FCS installation to improve the view of the drill pipes.

• Applied forces should be monitored (e.g. hydraulic ram pressure) and the operation halted if required levels approach limits of analyzed load cases.
  
  — Following hydraulic jacking to straighten Flex Joint, any deviations from the expected stiffness values should be investigated.

• Selection of the maximum pressure during well integrity testing should consider Mud Boost Face seal limitations.

• Finding 5: Details on methanol injection were not provided during the review. These must be provided to the DOE team.
Review Team Findings

- **Finding #1:** BP must monitor the tilt of the combined stack to ensure that the induced load limit is never exceeded.

- **Finding #2:** Since overpressure can lead to catastrophic failure, the selection of the burst disk must be approved by the DOE team.

- **Finding #3:** Detailed bolt analysis (material property, preload and torque information, etc.) was not presented - only individual bolt load capability was available. This information must be provided and inserted into the record.

- **Finding #4:** Since the Transition Spool is expected to be stressed near yield, BP must either present relevant quality data (material certification / inspection records and manufacturing) for the as-built component, or perform proof testing to demonstrate margin.

- **Finding #5:** Details on glycol injection during installation were not provided during the review. These must be provided to the DOE team.
Review Team Recommendations

- **Recommendation #1**: Ensure that an analysis of the allowable worst case combination of loads (e.g., largest tilt, highest temperature, highest pressure) is included in the record.

- **Recommendation #2**: Measures should be taken to limit maximum pressure during well integrity testing and well shut-in operations.

- **Recommendation #3**: Given the many possible loading conditions, supporting analysis capability should remain engaged to quickly evaluate actual operational conditions as events unfold.

- **Recommendation #4**: Although not presented during the design review, BP’s evaluation of the ability of all added components to withstand the maximum expected internal pressure, should be included for the record (include as-built documentation to ensure proper bolt pre-load).
Review Team Recommendations (Cont’d)

- **Recommendation #5:** An elastic/plastic finite element model should be run to predict ultimate failure mode and pressure of the stack system (eg Flex Joint, G Flange and Transition Spool).

- **Recommendation #6:** A calculation should be performed prior to installation in order to determine the effect of having five or fewer bolts restraining the G-flange and/or increased differential pressure.

- **Recommendation #7:** Connection between the combined BOP stack to collection operations vessels on the surface should recognize new limitations in the shimmed Flex Joint.

- **Recommendation #8:** Use of shims to limit rotation of the flex joint will alter the load path and requires further analysis (including installation loads on the bolts) prior to installation.
Review Team Recommendations (Cont’d 2)

- **Recommendation #9**: Installed bolt torque on Mud Boost valve flange should be confirmed, if possible via documentation of the as-built configuration.

- **Recommendation #10**: BP should monitor for leaks around the Mud Boost valve flange during any high pressure operation.

- **Recommendation #11**: In addition, BP should consider injecting a pressure sensitive sealant into the Mud Boost valve flange in order to reduce the possibility of leakage at the flange.

- **Recommendation #12**: Generate a complete system level analysis of the estimated accuracy of pressure measurement.
Review Team Recommendations (Cont’d 3)

- **Recommendation #13:** Once a correlation between the three pressure measurements is made, transmit only one of the three pressure measurements so that higher frequency sampling rate can be obtained.

- **Recommendation #14:** Attention should be given to potential bolt issues during riser flange removal and FCS installation such as tool access/engagement, possible need for captive nut replacement, and alternate means of providing flange clamping force.
Summary

• A comprehensive, four hour review was held by BP on June 23, 2010 at which the design, analysis and testing of the Flange Connector Spool Assembly and 3 Ram Capping Stack were presented.

• Reviewers included Subject Matter Experts from the three DOE design Labs as well as two DOE Science Advisors.

• Five Findings have been identified that the DOE Review Team feel must be closed before installation.

• Fourteen recommendations were generated that should increase our collective confidence in the proposed design, installation, and operation.

• The review did not generate any concerns that would warrant design modifications at this time.

A number of components have limited design margin, therefore BP must ensure that established limiting conditions (such as tilt angle and internal pressure) are never exceeded.
DOE Review Team

- Steve Black - LANL POC (DOE Team Lead)
- Arun Majumdar - DOE Science Advisor
- Alex Slocum - DOE Science Advisor
- Jim Sims - LANL Mechanical Engineer
- Nathan Bultman - LANL Mechanical Engineer
- Joel Bowers, Scott Perfect - LLNL Engineering
- Andy Shugard - SNL Engineering (Instrumentation)
- Rusty Escapule - SNL Engineering
SUMMARY REPORT
GLOBAL ANALYSIS OF
FLANGED CONNECTION OPTION
WITH TRIPLE RAM BOP ASSEMBLY
(FOR MACONDO WELL INTERVENTION)
4992 FT WATER DEPTH, GOM
PN 119681; PHASE 4

Prepared for:
BP
Houston, Texas

<table>
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<th>DATE</th>
<th>DESCRIPTION</th>
<th>ORIGINATOR</th>
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<th>APPROVER</th>
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GLOBAL ANALYSIS
OF FLANGED CONNECTION OPTION
WITH TRIPLE RAM BOP ASSEMBLY
4992 FT WATER DEPTH, GOM

PN 119681

Prepared for

BP
HOUSTON, TEXAS

Prepared By: Atul Ganpatye
Sr. Analyst

Reviewed By: Kenneth Bhalla, Ph.D.
Principal

Charles A. Miller, P. E.
Vice President

STRESS ENGINEERING SERVICES INC.
an employee-owned company

13800 WESTFAIR EAST DRIVE,
HOUSTON, TEXAS
77041-1101

JUNE, 2010
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PURPOSE

The purpose of the global analysis is to determine the response of the free standing system comprising of the casing, wellhead, Horizon BOP & LMRP, lower flex joint, and the triple RAM BOP assembly installed on top of the riser adapter. The system is loaded under self-weight of the components plus any additional externally applied gooseneck loads and a background current. The response is characterized primarily in terms of the maximum bending moments and/or stresses imparted on key components in the system. These response characteristics are compared back to the stress and moment allowables.

BP supplied SES with a problem description, a detailed description of the planned triple RAM BOP assembly, details of the Horizon BOP stack and wellhead components, representative site specific soil parameters, and general specifications for the analytical approach and specific cases to analyze.

ASSUMPTIONS

The analysis was performed with the following data/assumptions:

- Localized changes in the soil properties in the vicinity of the wellhead, due the Deepwater Horizon incident, have not been accounted for.

- The analysis precludes any damage (or permanent plastic deformation) in the casing or the wellhead system that may have resulted from the Deepwater Horizon incident.

- The analysis does not attempt to simulate the failure process of a specific mode at a specific component.

- A global ABAQUS beam model of the casing system, wellhead, Horizon BOP and LMRP, lower flex joint, triple RAM BOP assembly, and soils was used.

- Quasi-static analysis was performed (dynamic effects are not included).

- Linear elastic material properties are used.
- Initial casing system inclination, with respect to vertical, was assumed to be equal to 2° (based on information provided by BP); however, a few preliminary cases were analyzed with initial casing inclination, with respect to vertical, assumed equal to 1°, to understand the behavior of the lower flex joint under load.

- Initial flex joint angle of 3.5° (this includes 0.5° excursion to account for the uncertainty in measurement and an additional 0.5° to account for shim tolerance).

- All cases were analyzed with 0.2 knot background current.

- The triple RAM BOP assembly weight (including spool, mandrel, HC Connector and triple RAM BOP) was assumed to be equal to 158.5 kip (based on information provided by BP).

- Following gooseneck loads were provided by BP and are included in the model:
  - Moment: 38.7 ft-kip
  - Vertical Load: 6.31 kip
  - Horizontal Load: 1.00 kip

**SYSTEM CONFIGURATION**

The details of the components modeled in the analysis are listed in Table 1 and Table 2.

Table 1: Details of Components Modeled in the Global Analysis for the Flanged Connection Option with the Triple RAM BOP Assembly

<table>
<thead>
<tr>
<th>Component Name</th>
<th># of Units</th>
<th>Length (ft)</th>
<th>Weight (kip)</th>
<th>Weld Thrust (kip)</th>
<th>Horizontal Thrust (kip)</th>
<th>Bore/Body Size (in)</th>
<th>Component 1 Weight (kip)</th>
<th>Component 2 Weight (kip)</th>
<th>Component 3 Weight (kip)</th>
<th>Dry Weight (kip)</th>
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<tr>
<td>Upper Flange</td>
<td>1</td>
<td>3.47</td>
<td>30.00</td>
<td>3.068</td>
<td>42.1</td>
<td>15/20</td>
<td>55.00</td>
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<td>75.00</td>
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<td>Lower Flange</td>
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<td>3.02</td>
<td>27.00</td>
<td>2.863</td>
<td>37.8</td>
<td>16/20</td>
<td>50.00</td>
<td>60.00</td>
<td>70.00</td>
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<tr>
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<td>6.23</td>
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<td>4.915</td>
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<td>18/30</td>
<td>150.00</td>
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<td>65/20</td>
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<td>61.00</td>
<td>41.000</td>
<td>51.0</td>
<td>18/20</td>
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<td>180.00</td>
<td>762.0</td>
<td>650.0</td>
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</table>
### Table 2: Details of Casing System Modeled in the Global Analysis

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<thead>
<tr>
<th>Component Name</th>
<th>Length (ft)</th>
<th>Location of Top from Mudline (ft)</th>
<th>Location of Bottom from Mudline (ft)</th>
<th>Material Definition</th>
<th>Outer Diameter (in)</th>
<th>Inner Diameter (in)</th>
<th>Wall Thickness (in)</th>
<th>Wet Weight per Unit Length (lb/ft)</th>
<th>Dry Weight per Unit Length (lb/ft)</th>
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<tr>
<td>36&quot; x 2&quot; (X-65)</td>
<td>121</td>
<td>6.53</td>
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<td>X-65</td>
<td>36.00</td>
<td>32.00</td>
<td>2.00</td>
<td>726.93</td>
<td>631.99</td>
</tr>
<tr>
<td>36&quot; x 1.5&quot; (X-56)</td>
<td>166</td>
<td>-114</td>
<td>-280</td>
<td>X-56</td>
<td>36.00</td>
<td>33.00</td>
<td>1.50</td>
<td>553.22</td>
<td>480.97</td>
</tr>
<tr>
<td>28&quot; x 0.75&quot; (X-52)</td>
<td>280</td>
<td>0.00</td>
<td>-280</td>
<td>X-52</td>
<td>28.00</td>
<td>26.50</td>
<td>0.750</td>
<td>218.27</td>
<td>189.76</td>
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<tr>
<td>22&quot; x 1.0&quot; (X-80)</td>
<td>286</td>
<td>6.38</td>
<td>-280</td>
<td>X-80</td>
<td>22.00</td>
<td>20.00</td>
<td>1.00</td>
<td>224.28</td>
<td>194.99</td>
</tr>
</tbody>
</table>

A schematic of the entire assembly is shown in Figure 1. Additional details showing the casing inclination, flex joint angle, and application of gooseneck loads are shown in Figure 2.
Figure 1: Schematic Showing the General Arrangement of Components in the Flanged Connection Option with the Triple RAM BOP
Figure 2: Schematic Showing the Angles of Inclination and Gooseneck Loads Provided as Input to the ABAQUS Model
SOIL PROPERTIES AND MODELING

Soils data for the analysis were provided by BP and were used to calculate P-Y curves (lateral soil resistance curves), according to the procedure outlined in API 2A WSD. The key parameter in determining the soil lateral resistance is the undrained shear strength; with respect to Table 3, based on input and instructions from the BP geotechnical expert, the undrained shear strength listed under the column marked “NEW” was used in the analysis. Prior to this analysis SES was using the values listed under the column “OLD”.

Table 3: Soil Undrained Shear Strength Used in the Analysis

<table>
<thead>
<tr>
<th>Penetration Depth (ft)</th>
<th>OLD Undrained Shear Strength (psf)</th>
<th>NEW Undrained Shear Strength (psf)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>30</td>
<td>60</td>
</tr>
<tr>
<td>10</td>
<td>50</td>
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<tr>
<td>100</td>
<td>700</td>
<td>1400</td>
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<tr>
<td>280</td>
<td>2000</td>
<td>4000</td>
</tr>
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</table>

To understand the effect of decrease in soil lateral resistance due to soil plasticity (or decrease in the soil lateral resistance due to any other reason), a case was analyzed by removing a layer of soil up to a penetration depth of 30 ft. However, it should be noted that no attempt was made by SES to characterize soil damage or plasticity that may have been caused on account of the Deepwater Horizon incident. The removal of soil up to a penetration depth of 30 ft was merely an assumption primarily meant to understand the sensitivity of the results to the decrease in soil lateral resistance.

FLEX JOINT BEHAVIOR AND FINDINGS FROM PRELIMINARY ANALYSIS

Typically, the lower flex joint has a non-linear moment-angle of rotation relationship as shown in Figure 3. The non-linear curve in the figure can be interpreted as the resisting moment that the flex joint is capable of providing at a given angle of rotation.
Figure 3: Non-linear Moment-Angle of Rotation Relationship for the Lower Flex Joint

For example, with reference to Figure 3, if a moment load of about 300,000 ft-lbf is imparted on the flex joint, it would rotate to approximately 3 degrees, at which point it will have sufficient resisting moment to balance the imparted moment load. If the imparted moment load is more than 700,000 ft-lbf (slight more than the published 680,000 ft-lbf capacity at 10° rotation), the flex joint will tend to “bottom out” at 10° (approximately the maximum angle that the flex joint can physically achieve).

In light of the above information, preliminary analysis was conducted for the Flanged Connection Option with triple RAM BOP, to understand whether the moment imparted by the weight of the components above the flex joint would cause the flex joint to bottom out. To reinforce the understanding several cases with 1° initial casing inclination (with vertical) were also considered. The results of this preliminary analysis are summarized in Figure 4. It can be inferred from Figure 4 that, for the present casing inclination of 2°, the moment imparted by the weight of the components above the lower flex joint is more than the resisting moment that the flex joint can generate at any given angle of rotation. This implies that for a 2° casing inclination, with the current weight of the triple RAM BOP stack, the flex joint will always tend to bottom out at 10°.
Figure 4: Preliminary Analysis Summary Showing that the Flex Joint will Bottom Out for 2° Initial Casing Inclination with Vertical

For the given weight and inclination data, if it is determined that bottoming out the flex joint is not a feasible scenario (in terms of loads in the system after the flex joint bottoms out), then the preliminary results shown above predict that a physical restriction would be required to hold the flex joint at any other angle less than 10°.

CASES CONSIDERED

Based on the findings of the preliminary analysis discussed above, for an initial casing inclination of 2°, at least two cases would be required to adequately capture the response of the system, namely: 1. holding the flex joint at the current (last reported) position (3.5° flex joint angle including margin for measurement uncertainties, 5.5° with vertical), and 2. letting the flex joint bottom out to 10°, thus subjecting the system to the worst case scenario in terms of loads on the system. Adding a case to check for sensitivity to reduced lateral soil resistance gives rise to case matrix with three cases – this is shown in Table 4.
Table 4: Cases Considered for Global Analysis

<table>
<thead>
<tr>
<th>Case #</th>
<th>Initial Angle of Inclination of the casing with the vertical (deg)</th>
<th>Penetration Depth below Mudline over which soil is removed (ft)</th>
<th>Flex Joint Angle (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>CASE N1</td>
<td>2</td>
<td>0</td>
<td>3.5</td>
</tr>
<tr>
<td>CASE N2</td>
<td>2</td>
<td>0</td>
<td>10</td>
</tr>
<tr>
<td>CASE N3</td>
<td>2</td>
<td>30</td>
<td>10</td>
</tr>
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</table>

ANALYSIS TOOLS UTILIZED

A nonlinear finite element analysis, utilizing ABAQUS (Version 6.5), was used for the global analysis study. ABAQUS is a general-purpose finite element program, which can model non-linearity of geometry, material properties, and spring definitions.

The analysis employed:

- non-linear springs to model the soil characteristics
- beam elements to model all pipe sections.
- linear-elastic material assumptions

The finite elements utilized were the ABAQUS PIPE31 beam element. This is a three-dimensional beam element that models a pipe in space. This element accounts for internal and external hydrostatic pressure effects, using the HP option. The HP option calculates the correct hoop stresses in the pipe due to fluid pressure, which then calculates the correct von Mises stress without any further post-processing.

The finite element analysis first consists of pre-loading the riser with its distributed wet self-weight, followed by internal and external hydrostatic pressurization, and current loading.
MODELING AND ANALYTICAL ASSUMPTIONS

1. The BOP/LMRP was not modeled in detail. The stack was modeled as having an outer diameter of 50.00 inch and a wall thickness of 15.625 inch to represent an extremely stiff component.

2. At places in the wellhead where components overlap (i.e. low pressure and high pressure housings), structural dimensions of the inner and outer components were modeled individually. Nodes for the two components located within the overlapping region were constrained together using an *MPC, TYPE=TIE command. This allows for the determination of bending moments carried by the various wellhead/connector components in the overlapping region.

3. The bending moment carrying contributions of the 28 inch and 22 inch casings below the wellhead were included in the analysis. The 22 inch casing was assumed to be cemented to the 28 inch casing, which in turn was assumed to be cemented to the 36 inch casing. The casing strings were assumed coaxial.

4. The casing strings were modeled up to a depth of 280 ft below mudline.

5. Lateral soil support was assumed and modeled by equivalent soil springs (P-Y springs) acting against the conductor casing.

RESULTS AND DISCUSSION

A summary of the results from the global analysis of the Flanged Connection Option with the triple RAM BOP is shown in Table 5. The results in the table and the subsequent conclusions discussed below are based on the assumption that there is no prior damage to the casing/wellhead system from the Deepwater Horizon event.
Table 5: Summary of Results from Global Analysis for the Flanged Connection Option with Triple RAM BOP Assembly.

<table>
<thead>
<tr>
<th>Case #</th>
<th>Initial Angle of Indentation of the casing with the vertical (deg)</th>
<th>Penetration Depth below Mudline over which soil is removed (ft)</th>
<th>Max von Mises Stress in the 36&quot; x 2&quot; (X-465) Casing (ksi)</th>
<th>Flex Joint Angle (deg)</th>
<th>Final Angle of Indentation of the Horizontal BOP with Vertical (deg)</th>
<th>Final Angle of Indentation of the 3-rum BOP with Vertical (deg)</th>
<th>Bending Moment at Lower Flange Joint (10^6 ft-lb)</th>
<th>Bending Moment at Lower Flexible Joint (10^6 ft-lb)</th>
<th>Bending Moment at Horizon BOP (10^6 ft-lb)</th>
<th>Wellhead Bending Moment at Horizon BOP (10^6 ft-lb)</th>
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</thead>
<tbody>
<tr>
<td>CASE N1</td>
<td>2</td>
<td>0</td>
<td>10</td>
<td>3.5</td>
<td>3.3</td>
<td>12</td>
<td>0.61</td>
<td>0.9</td>
<td>1.71</td>
<td>2.16</td>
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<tr>
<td>CASE N2</td>
<td>2</td>
<td>0</td>
<td>19.0</td>
<td>10</td>
<td>3.3</td>
<td>14.0</td>
<td>0.61</td>
<td>0.9</td>
<td>1.71</td>
<td>2.16</td>
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<tr>
<td>CASE N3</td>
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<td>30</td>
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<td>10</td>
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<td>14.7</td>
<td>0.64</td>
<td>0.94</td>
<td>1.26</td>
<td>2.49</td>
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</table>

The results presented in Table 5 indicate the following:

a. The maximum predicted bending moment at the wellhead/wellhead connector (at the bottom of the Horizon BOP stack) is 2.49 million ft-lbf (for Case N3). Available data on Vetco SHD-H4 18-3/4 inch wellhead connector (with H4 adapter) on a 27" mandrel indicates that the bending moment capacity (working capacity) of the connector is 5 million ft-lbf (at 0 kip tension and 15 ksi bore pressure). Based on this, it can be concluded that the integrity of the wellhead connector is unlikely to be compromised.

b. The maximum predicted bending moment at the HC connector (at the bottom of the Horizon LMRP) is 1.26 million ft-lbf (for Case N3). Available data on 15 ksi rated Cameron HC connector indicates that the bending moment capacity (working capacity) is 2.2 million ft-lbf. Based on this, it can be concluded that the integrity of the HC connector is unlikely to be compromised.

c. The stress in the 36 inch casing is below yield for all the cases considered. Stresses in the other casing strings (28 inch and 22 inch) were also checked and were found to be lower than that in the 36 inch and below their respective yield stresses. Therefore, based on the above results, the integrity of the casing system is unlikely to be compromised.
d. Upon further investigation (not covered in this analysis), other components in the system may present a weaker link based on more detailed local (component level) analysis. However, this is beyond the scope of the present work. Accordingly, this report does not attempt to predict whether any of the components in the system would exceed their respective capacities, beyond what can be covered within the extent of the scope and assumptions for the global analysis detailed earlier in the report.

The bending moment distribution in the system for the three cases analyzed is shown in Figure 5.

![Bending Moment Distribution: Cases N1, N2, and N3](image)

Figure 5: Bending Moment Distribution Along the Casing/Stack System for the Cases Analyzed in this Study
The stress distribution along the 36 inch (X-65) casing is shown in Figure 6.

**Figure 6: Stress Distribution in the 36 inch Casing for the Cases Analyzed in this Study**
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### GoM MC252 - Containment and Disposal Project

**Flexjoint Jacks SIT Results**

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TREX-120129.189
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**Page**: Page 4 of 18

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LNL004-026170
1 Executive Summary

The Horizon flexjoint riser adapter is at an inclination angle of 4.5° ± 0.5° from vertical. This angle has potential to make installation of a capping stack more difficult as well as add bending moment loads to the system.

A system of pusher blocks and hydraulic jacks was developed in order to stabilize flexjoint riser adaptor. The hydraulic jacking system was found to be stable and effective at pushing the riser adapter through approximately 5° of inclination (i.e. near vertical).
2 Introduction

Correction of the Horizon flexjoint riser adapter angle is important for the installation of the Triple-Ram stack. For that purpose, a hydraulic jacking system was designed and built (see Figure 1. This system was tested at the Oil States Industries facility in Houston, TX on the original Deepwater Horizon flex joint (See Figure 2).

Figure 1 - Pusher General Arrangement
Figure 2 - Original Deepwater Horizon Flex Joint on Test Stand

The test will verify the hydraulic jacking system capability to rotate the riser adapter through approximately 4.5° of stroke.
3 Materials

3.1 Equipment List

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Figure 3 - Pusher Block Assembly
Figure 4 - Hydraulic Manifold and Digital Inclinometer Display
4 Pusher Test 06-27-2010

Three hydraulic rams were used to push the riser adapter over.

4.1 Procedure

- Install 3 hydraulic rams and pusher blocks adjacent to one another on the flex joint
- Install 3 additional hydraulic rams and pusher blocks adjacent to one another at 180°
- Install a digital inclinometer and display to the riser adapter
- Pressure up 3 adjacent hydraulic rams in 500psi increments to induce movement
- Record inclination and pressure
- Utilize the rams until either reach maximum stroke or near 10,000 psi working limit
- Release pressure
- Pressure up 3 adjacent rams at 180° to return the riser adapter to start location. Utilize same procedure as above.
- Release all pressure and observe impact to pusher blocks

4.2 Results

The flexjoint "popped" and began to move when hydraulic pressure reached 4,500psi. The pressure was increased incrementally until a pressure of 7,300 was reached.

Opposing 3 rams were pressured up, and the riser adapter was moved back toward its starting position.

Observations:

- The riser adapter took significant force to "break" it free and to begin movement indicating significant friction.
- The jacks were effective in moving the riser adapter approximately 4.5° and back again
- The blocks remained stable during use
- The blocks showed minimal signs of deformation after use

Table 1 shows the recorded test data and Figure 5 shows the plotted hysteresis and the design curve.
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Figure 7 - Pusher Blocks
Figure 10 - Bolt Bearing Area

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6 Appendix 2 - Contact Information

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Updated as per Comments: Kevin Lanan, Trevor Smith

Issued: Kevin Lanan, Trevor Smith

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**Authority:** Trevor Smith

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**Security Classification:** Project Confidential

Page: Page 3 of 30

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1 Executive Summary

Stabilization of the flex joint is an important aspect of capping the MC252 well. To this end, restraint blocks, wedges, and pins were designed, built and tested.

The restraint blocks, wedges, and pins performed as expected during the tests and were stable under load. The blocks, wedges, and pins are considered viable concepts for use in stabilizing the flex joint; subject to their design loading conditions.

Important observations:

- Little movement of the restraint block, wedge, and pin assemblies during engagement and loading
- Slightly increased movement of restraint block when Dry Moly lubricant used on critical surfaces. Increased indentation of teeth into retainer ring for this case.
- no deformation detectable in wedge or riser adapter
- vertical movement of the riser adapter was not simulated during restraint assembly loading

2 Introduction

2.1 Introduction

Correction of the riser adapter angle of the MC252 Flex Joint and its stabilization are necessary steps for the installation of the triple ram capping stack. For that purpose, Restraint block assemblies were designed and built (see Figure 1 and Figure 2). These assemblies were tested at the Oil States International facility in Houston, TX on the original Deepwater Horizon flex joint (See Figure 3).
14 Restraint assemblies to provide 360° support.

Figure 1 - Restraint General Arrangement

Figure 2 - Restraint General Arrangement
The tests will verify that the restraint block assemblies have appropriate geometry to support the riser extension and are stable while subjected to various load conditions.

2.2 Objective

These tests will identify the stability of the restraint block, pin, and wedge assembly when engaging the riser adapter and when subjected to load (see Figure 4). Also, the wedge clearance will be tested. Additionally, the vertical movement of the riser adapter when subjected to upward force will be evaluated.
3 Materials

3.1 Equipment List

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Figure 5 - Pusher Block and Ram Assembly

Figure 6 - Restraint Block
4 Restraint Wedge - Tested 07-01-2010

One restraint block with a 0° wedge was tested in the flex joint. 4 hydraulic rams and pusher blocks were used to push the flex joint extension onto the restraint block and wedge assembly.

4.1 High-Level Procedure

- Install restraint block with 0° attached
- Install 4 rams / pusher blocks opposite the restraint assembly
• Utilize rams to push riser adapter onto the restraint assembly
• Increase load on 4 pushers until maximum working pressure has been reached (10,000psi)
• Release load from cylinders
• Examine any movement or damage to the restraint assembly and riser adapter
• Repeat test with critical surfaces lubricated to reduce friction.

For full procedure see: Flex Joint Restraint Concept Test Procedure

4.2 Symmetric Loading - No Lubricant

The pushers were placed symmetric with respect to the restraint block assembly. Pressure was increased with no movement until about 6,800psi. At this point the flex joint creaked and began to move. The riser adapter was pushed over until it made contact with the restraint wedge. The wedge was loaded asymmetrically but did not move significantly. Pressure on the rams was increased until 10,000 psi was reached. No significant movement of the restraint block assembly or the riser adapter was observed. The rams were released and the riser adapter moved back approximately 5/8" and it stopped. Rams were used to push the riser adapter back away from the assembly, and it was removed. Upon examination, the teeth bit through the paint and engaged the steel retainer ring. This caused some localized damage to the teeth. The wedge was asymmetrically loaded, but did not yield. The riser adapter paint had been scratched, but the steel was not damaged. See Appendix 1 for photos / figures.

Other Observations:
• Although the teeth broke through the paint, they bit very little into the steel retainer ring
• Very little movement of the block - quite stable during engagement and loading
• Side of wedge scratched riser adapter paint - a radius will be ground on the edges to prevent potential damage to the riser adapter steel.
• Wedge loaded asymmetrically, but no deformation detectable in wedge or riser adapter.

NOTE: The flex joint moved asymmetrically most probably because the resultant of the rams was about 30° off from the pushing axis of previous pusher tests. There was likely some “memory” from the previous tests (e.g. the wear ring was a bit smoother in the other direction).

4.3 Asymmetric Loading - With Lubricant

The critical contact points of the restraint assembly were lubricated to simulate reduced friction. The rams were set up asymmetrically w.r.t. the restraint assembly. The riser adapter was pushed down to the wedge. As it contacted, the block moved sideways slightly and then engaged. The rams were pressured up to 10,000 psi and then released. The riser adapter was pushed off of the restraint assembly, and it was inspected. Again, the teeth bit through the paint and were slightly damaged. The wedge was loaded asymmetrically but did not yield. The paint on the riser adapter was scratched, but the steel was not damaged. See Appendix 2 for photos / figures.
Other Observations:
- Teeth broke through paint and bit into steel retainer ring. The residual grooves could be felt easily by hand.
- Some movement sideways during engagement, then very little while loading
- Assembly stable when lubricated

5 Riser Adapter Vertical Pull - Tested 07-02-10

The riser adapter was pulled vertically to simulate uplift force that could be exerted by internal pressure. The objective was to observe the 1/8" to 1/4" vertical play of the riser adapter w.r.t. the flex joint body. Measurements were taken, and there was very little vertical movement. See Appendix 3 for picture.

5.1 Procedure
- Affix laser survey dot on both flex joint body and riser adapter
- Utilize overhead crane and load cell to increase lift force on riser adapter in 5,000lb increments
- Monitor vertical displacement with laser survey equipment
- Max lift ~ 90% of assembly (29,000 lb flex joint + 25,000 lb SIT base ~ 55,000 lb assembly)

5.2 Results

The reference plane was set on the top of the retaining ring flange and was measured to be planar within .004". We measured the diameter of the retaining ring to be Ø58.040". Positive X is to the left in the picture and positive Y is coming straight out. Positive Z is going straight up the middle of the flex joint body.

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<td>45</td>
<td>-0.084</td>
<td>0.026</td>
<td>0.047</td>
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</table>
It can be seen from the table that very limited vertical movement was observed in the vertical direction (Z). Upon further inquiry of Oil States International, this is likely due to the tight fit of the flex element back flange and the body ID. During refurbishment work, OSI often end up lifting the entire flex joint when lifting the flex element out.

6 Restraint Wedge Test Fit - Tested 07-02-2010

Four restraint blocks with 0° wedges were placed at 4 quadrants of the riser adapter to demonstrate clearances. See Appendix 4 for pictures.

6.1 Procedure

- Move the riser adapter to vertical
- Test fit the block and 0° wedge assembly under the mud boost valve
- Test fit the block and 0° wedge assembly at four quadrants of the riser adapter.
- Observe fit-up

6.2 Results

The riser adapter was at an initial inclination of 1.5°. 2 rams were used to push the riser adapter to nearly vertical. Pressure was released, and the riser adapter stayed at vertical. A 0° wedge and restraint block assembly was placed into position under the mud boost valve. There was adequate clearance to get it in, but the block was not lowered all the way down, due to the fact that it would have been very difficult to remove with the mud boost valve in the way.

Next, the four restraint bocks and 0° wedges were placed at 4 quadrants of the riser adapter. These demonstrated adequate clearance for installation.

Other Observations:
- Riser adapter began to move with two cylinders at 3,000 psi
- 0° wedges fit on 4 quadrants demonstrating adequate clearance for installation
- Least clearance was about 1/16” and the most was about 1/2”
- Riser adapter pushed away from closest fitting wedge to facilitate removal without engaging teeth

7 Restraint Pin & Wedge - Tested 07-02-2010

One restraint block / 0° wedge / 0° pin assembly was tested in the flex joint. 3 Hydraulic rams were used to push symmetrically against the restraint assembly. See Appendix 5 for photos.
7.1 High Level Procedure

- Install restraint block with 0° pin and 0° wedge
- Utilize 3 pusher blocks and rams to push the riser adapter over to the assembly
- Observe engagement
- Pressure up the rams to 10,000 psi to load restraint assembly
- Release pressure
- Use rams to push riser adapter off the restraint assembly
- Examine any movement or damage to the restraint assembly and riser adapter
- Repeat test with critical surfaces lubricated to reduce friction and without wedge installed

7.2 Block + Pin + Wedge Symmetric Loading - No Lubricant

The 11” long 0° pin was slightly too long to install on the flex joint due to residual inclination of approximately 1°. A 1/4” shim plate was removed from the pin and it was inserted. Pushers were placed symmetric with respect to the restraint block and pin assembly. Pressure was increased with no movement until about 3,000 psi. At this point the flex joint creaked and began to move. The riser adapter was pushed over until it made contact with the restraint pin. Pressure on the rams was increased until 10,000 psi was reached. No significant movement or rotation of the restraint block and pin assembly or the riser adapter was observed. The rams were released, and then re-oriented to push the riser adapter back away from the assembly.

Other Observations:

- No significant movement of the restraint assembly during engagement and loading
- A little paint damage observed on the pin, block, wedge, and riser adapter
- No noticeable damage to steel on critical surfaces

7.3 Block + Pin No Wedge Symmetric Loading - With Lubricant

Critical surfaces were lubricated and the test procedure was repeated. Little to no movement was observed.

Other Observations:

- No significant movement of the restraint assembly during engagement and loading
- A little paint damage observed on the pin, block, wedge, and riser adapter
- No noticeable damage to steel on critical surfaces
Appendix 1 - Restraint Wedge, Symmetric, No Lubricant

Figure 9 - Symmetric Loading Configuration

Orientation of the 4 pusher blocks and the 1 restraint block assembly.

Figure 10 - Restraint Block and Wedge Assembly

---

Title of Document: Flex Joint Restraint Test Results
Authority: Trevor Smith
Custodian/Owner: Kevin Lanan
Retention Code: AAA0000
Security Classification: Project Confidential

Document Number: 2200-T2-DQ-RP-4191
Revision: 1
Issue Date: 7/6/2010
Next Review Date (if applicable): 7/6/2010
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Warning: Check DW Docs revision to ensure you are using the correct revision.
Pushers on right and restraint block on left.

Blue machinist’s ink sprayed on riser adapter to highlight bearing area of wedge.
When loaded, the wedge scraped some paint off the FJ nipple / riser adapter. Damage to steel will be prevented by slightly rounding the edges of the wedge block.
Figure 15 - Bearing Area of Restrained Wedge on FJ Nipple

Asymmetric bearing due to flex joint "memory" from other tests. This test was at about 30° off from previous pusher tests. There was no sign of deformation of the steel.

Figure 16 - Teeth of Back of Restraint Block with Paint

Teeth were damaged slightly from the test, but bit through the paint and engaged the retainer ring steel.
Figure 17 - Teeth Bearing Area

Teeth bearing area showing paint missing. There was very little indentation of the steel retainer ring.
Appendix 2 - Restraint Wedge, Asymmetric, with Lubricant

Figure 18 - Asymmetric Loading Configuration

Location of 4 pusher blocks and one restraint block with wedge.

Figure 19 - Bearing of Wedge on Flex Joint Nipple

Some paint scratching was observed as in previous test.
Figure 20 - Slight Movement of Block

Teeth moved slightly then engaged when lubricated and loaded asymmetrically.

Figure 21 - Bearing Area on FJ Nipple

No deformation of the FJ nipple steel was noticeable.
Bearing was uneven due to the asymmetry of the motion of the riser adapter as explained above. The line in this picture is due to uneven application of machinist's ink. Feeling this part by hand, there is no detectable sign of deformation.

Figure 23 - Bearing Area on Retainer Flange

Bearing area had noticeable steel deformation on ID of retainer flange due to teeth from retainer block. Indentation was on right side.
Figure 24 - Damage to Teeth from Testing

Damage to teeth from multiple tests. These were later cleaned up with a file. All load testing was done on the same retainer block.
 Appendix 3 - Vertical Pull test

Figure 25 - Riser Adapter Test Pull

The riser was pulled with an overhead crane to simulate uplift. Laser survey equipment was used to measure displacement. Uplift force was limited to the weight of the flex joint and testing base (~55,000 lbs).
Appendix 4 - 0° Restraint Wedge Test Fit

Figure 26 - Restraint Wedge Assembly Under Mud Boost

The wedge block assembly would fit under the mud boost valve, but was not lowered the last few inches.

Figure 27 - Four 0° Wedges at Quadrants

0° wedges were inserted at each quadrant to demonstrate fit-up.
Figure 28 - Minimal Play between Opposing 0° Wedges

There was minimal play between opposing wedges, yet enough clearance to install them.
Appendix 5 - Restraint Block with Pin; W & WO Wedge, W & WO Lubricant

Figure 29 - Restraint Block with Wedge and Pin Assembly

Figure 30 - Restraint Pin Seated

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Figure 31 - Pin Seated and Loaded by Riser Adapter

Figure 32 - Marks on Riser Adapter

No deformation of riser adapter steel detectable. Only paint scratches were evident.
Figure 33 - Marks on Pin

No significant difference between lubricated and non-lubricated and wedge and no wedge cases. Tip of pin had no deformation.
## Appendix 6 - Contact Information

<table>
<thead>
<tr>
<th>BP</th>
<th>Position</th>
<th>Office/Vessel Phone</th>
<th>Office/Vessel Fax</th>
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<tbody>
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<td></td>
<td></td>
<td>281-851-7686</td>
<td><a href="mailto:John.Schneble@bp.com">John.Schneble@bp.com</a></td>
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<td>Project Engineer</td>
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<td></td>
<td></td>
<td><a href="mailto:Vassilis.Gakas@bp.com">Vassilis.Gakas@bp.com</a></td>
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<td>Project Engineer</td>
<td>281-228-7423</td>
<td></td>
<td></td>
<td><a href="mailto:Chase.Breidenthal@bp.com">Chase.Breidenthal@bp.com</a></td>
</tr>
<tr>
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<td>Project Engineer</td>
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<td></td>
<td>281-222-8304</td>
<td><a href="mailto:Kevin.Lanan@bp.com">Kevin.Lanan@bp.com</a></td>
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<tr>
<td>John Stafford</td>
<td>Project Coordinator</td>
<td>713-445-2949</td>
<td></td>
<td>832-622-5643</td>
<td><a href="mailto:John.Stafford@oilstates.com">John.Stafford@oilstates.com</a></td>
</tr>
<tr>
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<td>Project Manager</td>
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<td></td>
<td>281-941-9465</td>
<td><a href="mailto:Paul.Centanni@oilstates.com">Paul.Centanni@oilstates.com</a></td>
</tr>
<tr>
<td>James Huber</td>
<td>Project Engineer</td>
<td>713-820-4583</td>
<td></td>
<td>817-459-1181</td>
<td><a href="mailto:James.Huber@oilstates.com">James.Huber@oilstates.com</a></td>
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## Document Authorization Form

This form to be used for authorizing new, revised and obsolete documents, please indicate clearly which category applies.

**Special Instructions**: MC252 Incident

### Document Details

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**Document Title**: Flex Joint Restraint Test Results

**Next Review Date**

**Reason for Issue** (check as applicable)

- New Document [X]
- Revised Document
- Obsolete Document

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</table>

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TREX-120129.235
HIGHLY CONFIDENTIAL
LNL004-026214
Elevation
Middle of Flange

Uniform Thickness
Assumed for Section 1.875"

ANSYS SHELL93 Elements
(8-node, 2nd-order, quads)
(3-translational, 3-rotational dof/node)

Number of Nodes = 6006
Number of Elements = 2601

Element Spacing < 0.75"
at Critical Locations
Fixed Except Radial Displacement
Uθ=U1=0 (CSYS=5)
Rθ=R1=R2=0 (CSYS=5)
Symmetry Boundary Condition
UZ=0
ROTX=ROTY=0
Rigid Spider (MPC) transfers UV/ROTZ only. Other DOF not affected.
V = 488.0 kips (Full)
    = 244.0 kips (Model)
Element Coordinate System
X = Hoop
Y = Axial
Z = Radial
XY = In-Plane Shear

Case 1:
Vertical (F) + Moment (M) + Shear (V)

Case 2:
Vertical (F) + Moment (M) + Shear (V) + Ext. Pressure (P)
Case 1:
Vertical (F) + Moment (M) + Shear (V)

Case 2:
Vertical (F) + Moment (M) + Shear (V) + Ext. Pressure (P)

Potential Governing Locations for Buckling Evaluation

Location A

Location B

Location C
### Case 1, First Five Eigenvalues

<table>
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<tr>
<th>SET</th>
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### Case 2, First Five Eigenvalues

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Case 1, Axial Stress

SV50548, BP MC252 - Vertical, Moment, Shear Eigen

HIGHLY CONFIDENTIAL
NODAL SOLUTION
STEP=1
SUB =1
TIME=1
S2 (AVG)
MIDDLE
RSYS=SOLU
DMX = 0.053604

Case 1, Radius Stress
100 psi used for calculation

SV50548, BP MC252 - Vertical, Moment, Shear Eigen
Case 2, Hoop Stress

SV50548, BP MC252 - Vertical, Moment, Shear, Pressure Eigen (psf)
NODAL SOLUTION

STEP = 1
SUB = 1
TIME = 1
SZ (AVG)
MIDDLE
RESOLV = SOLU
DMX = 0.056869
SMN = -750
SMX = -750
(ksi)

Case 2, Radial Stress

SV50548, BP MC252 - Vertical, Moment, Shear, Pressure Eigen
Case 3, In-Plane Shear Stress

SV50548, BP MC252 - Vertical, Moment, Shear, Pressure Eigen
Subject: Local Buckling Check per ASME Code Case N-284-1
Drawing Reference: Solidworks Model SV50548-917, Rev. 0

By: Dana E. Petroni  Revised By:
Date: July 2, 2010  Date:
Check: Check:
Date: Date:

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The purpose of these calculations is to check local buckling of the FJ Stabilizer BP MC 252.

Document Reference
ASME Boiler and Pressure Vessel Code, Code Case N-284-1

Geometry and Material Property Input

Maximum Outside Diameter: \( D := 24.79 \text{-in} \)

Maximum Outside Radius: \( R := 0.5 \cdot D \quad R = 12.395 \text{-in} \)

Minimum Thickness: \( t := 1.875 \text{-in} \)

Unstiffened Length: \( L := 40.50 \text{-in} \)

Elastic Modulus: \( E := 29.50 \cdot 10^6 \text{ psi} \)

Material Yield Strength: \( \sigma_Y := 85695 \text{-psi} \quad \text{MTR HT 08011708} \)

Required Factor of Safety: \( FS := 2.00 \quad \text{Section 1400(a)} \)

Required Minimum Corrected Eigenvalue: \( \lambda_{\text{min}} := 2.00 \)

Geometric Parameters

Buckling Parameter: \( M := \frac{L}{\sqrt{R \cdot t}} \quad M = 8.401 \quad \text{Section 1200} \)

\( R/t \) Ratio: \( X := \frac{R}{t} \quad X = 6.611 \)
**Capacity (Fabrication) Knockdown Factors**

Section 1511

Axial, Effect of R/t: \[ \alpha_{\phi L1} = 0.207 \]

\[ \alpha_{\phi L2} = 1.52 - 0.473 \cdot \log \left( \frac{R}{t} \right) \]

\[ \alpha_{\phi L3} = \frac{300 \cdot \sigma_Y}{E} - 0.033 \]

\[ \alpha_{\phi L4} = \min \left( \alpha_{\phi L2}, \alpha_{\phi L3}, \alpha_{\phi L1} \right) \]

Axial, Effect of Length:

\[ \alpha_{\phi LB} = \begin{cases} 1.5, & M < 1.5 \\ 0.627, & M < 1.73, 0.837 - 0.14 \cdot M \\ 0.826, & M < 10, 0.207 \\ 0.826, & M > 10 \end{cases} \]

Axial, Governing Number:

\[ \alpha_{\phi L} = \max \left( \alpha_{\phi LA}, \alpha_{\phi LB} \right) \]

Hoop:

\[ \alpha_{\theta L} = 0.800 \]

Shear:

\[ \alpha_{\phi 0L} = \begin{cases} 1.5, & R < 250 \\ 0.800, & 1.323 - 0.218 \cdot \log \left( \frac{R}{t} \right) \end{cases} \]

**Case 1, Loading**

Vertical Load = 170.0 kips
Bending Moment = 1200.0 kips*ft
Shear Load = 488.0 kips
No External Pressure

First Eigenvalue:

\[ \lambda_1 = 69.081 \] Location A Governs

Axial Stress:

\[ \sigma_\phi = 29171 \text{-psi} \]

Hoop Stress:

\[ \sigma_\theta = 16464 \text{-psi} \]

Radial Stress:

\[ \sigma_r = 100 \text{-psi} \] Nominal Radial Stress Assumed

Shear Stress:

\[ \sigma_{\phi \theta} = 672 \text{-psi} \]
Case 1. Factored Stress Components

Factored Axial Stress: \( \sigma_{\phi e L} := \lambda_1 \cdot \sigma_\phi \) \[ \sigma_{\phi e L} = 2015162 \text{ psi} \]

Factored Hoop Stress: \( \sigma_{\theta e L} := \lambda_1 \cdot \sigma_\theta \) \[ \sigma_{\theta e L} = 1137350 \text{ psi} \]

Factored Radial Stress: \( \sigma_{r e L} := \lambda_1 \cdot \sigma_r \) \[ \sigma_{r e L} = 6908 \text{ psi} \]

Factored Shear Stress: \( \sigma_{\phi \theta e L} := \lambda_1 \cdot \sigma_{\phi \theta} \) \[ \sigma_{\phi \theta e L} = 46422 \text{ psi} \]

Case 1. Plasticity Knockdown Factor Calculation

Section 1610 and Section 1713.1.1

Elastic Buckling Limits, Axial:
\[ \sigma_{xa} := \frac{\alpha_{\phi e L} \cdot \sigma_{\phi e L}}{FS} \] \[ \sigma_{xa} = 844831 \text{ psi} \]

Elastic Buckling Limits, Hoop:
\[ \sigma_{ha} := \frac{\alpha_{\theta e L} \cdot \sigma_{\theta e L}}{FS} \] \[ \sigma_{ha} = 454940 \text{ psi} \]

Elastic Buckling Limits, Radial:
\[ \sigma_{ra} := \frac{\alpha_{r e L} \cdot \sigma_{r e L}}{FS} \] \[ \sigma_{ra} = 2763 \text{ psi} \]

Elastic Buckling Limits, Shear:
\[ \sigma_{ra} := \frac{\alpha_{\phi \theta e L} \cdot \sigma_{\phi \theta e L}}{FS} \] \[ \sigma_{ra} = 18569 \text{ psi} \]

Plasticity Knockdown Parameters:

Axial:
\[ \Delta_{\phi} := \frac{\alpha_{\phi e L} \cdot \sigma_{\phi e L}}{\sigma_Y} \] \[ \Delta_{\phi} = 19.717 \]

Hoop:
\[ \Delta_{\theta} := \frac{\alpha_{\theta e L} \cdot \sigma_{\theta e L}}{\sigma_Y} \] \[ \Delta_{\theta} = 10.618 \]

Shear:
\[ \Delta_{\phi \theta} := \frac{\alpha_{\phi \theta e L} \cdot \sigma_{\phi \theta e L}}{\sigma_Y} \] \[ \Delta_{\phi \theta} = 0.433 \]
FJ Stabilizer Buckling Check
BP MC 252

Plasticity Knockdown Factors:

Axial: \( \eta_\phi := \begin{cases} 1 & \text{if } \Delta_\phi < 0.55, \ 1.000, \text{ if } \Delta_\phi < 1.6, \frac{0.45}{\Delta_\phi} + 0.18, \text{ if } \left( \Delta_\phi < 6.25, \frac{1.31}{1 + 1.15 \cdot \Delta_\phi}, \frac{1}{\Delta_\phi} \right) \end{cases} \)
\( \eta_\phi = 0.051 \)

Hoop: \( \eta_\theta := \begin{cases} 1 & \text{if } \Delta_\theta < 0.67, \ 1.000, \text{ if } \Delta_\theta < 4.2, \frac{2.53}{1 + 2.29 \cdot \Delta_\theta}, \frac{1}{\Delta_\theta} \end{cases} \)
\( \eta_\theta = 0.094 \)

Shear: \( \eta_\phi \theta := \begin{cases} 1 & \text{if } \Delta_\phi \theta < 0.48, \ 1.000, \text{ if } \Delta_\phi \theta < 1.7, \frac{0.43}{\Delta_\phi \theta} + 0.1, \frac{0.6}{\Delta_\phi \theta} \end{cases} \)
\( \eta_\phi \theta = 1.000 \)

**Case 1, Factored Minimum Eigenvalues**

Axial: \( \lambda_{\phi 1} \ := \ \lambda_1 \cdot \phi L \cdot \eta_\phi \)
\( \lambda_{\phi 1} = 2.938 \)

Hoop/Radial: \( \lambda_{\theta 1} \ := \ \lambda_1 \cdot \theta L \cdot \eta_\theta \)
\( \lambda_{\theta 1} = 5.205 \)

Shear: \( \lambda_{\phi \theta 1} \ := \ \lambda_1 \cdot \phi \theta L \cdot \eta_\phi \theta \)
\( \lambda_{\phi \theta 1} = 55.265 \)

**Case 1, Inelastic Buckling Limit**

Axial: \( \sigma_{xc} := \eta_\phi \cdot \sigma_{xa} \)
\( \sigma_{xc} = 42847 \text{-psi} \)

Hoop: \( \sigma_{hc} := \eta_\theta \cdot \sigma_{ha} \)
\( \sigma_{hc} = 42847 \text{ psi} \)

Radial: \( \sigma_{rc} := \eta_\theta \cdot \sigma_{ra} \)
\( \sigma_{rc} = 260 \text{-psi} \)

Shear: \( \sigma_{tc} := \eta_\phi \theta \cdot \sigma_{ta} \)
\( \sigma_{tc} = 18569 \text{-psi} \)
Case 1, Unity Checks, Unit Equations

\[ U_{C_{axial}} \left( \frac{\lambda_{\text{min}}}{\lambda_{\phi 1}} \right) \quad U_{C_{axial}} = 0.681 \quad U_{C_{axial}} = \frac{\sigma_{\phi}}{\sigma_{xc}} \quad U_{C_{axial}} = 0.681 \]

\[ U_{C_{hoop}} \left( \frac{\lambda_{\text{min}}}{\lambda_{\theta 1}} \right) \quad U_{C_{hoop}} = 0.384 \quad U_{C_{hoop}} = \frac{\sigma_{\theta}}{\sigma_{hc}} \quad U_{C_{hoop}} = 0.384 \]

\[ U_{C_{radial}} \left( \frac{\lambda_{\text{min}}}{\lambda_{\theta 1}} \right) \quad U_{C_{radial}} = 0.384 \quad U_{C_{hoop}} = \frac{\sigma_{r}}{\sigma_{rc}} \quad U_{C_{radial}} = 0.384 \]

\[ U_{C_{shear}} \left( \frac{\lambda_{\text{min}}}{\lambda_{\phi 1}} \right) \quad U_{C_{shear}} = 0.036 \quad U_{C_{shear}} = \frac{\sigma_{\phi \theta}}{\sigma_{\tau c}} \quad U_{C_{shear}} = 0.036 \]

Case 1, Unity Checks, Iteration Equations

Axial Plus Shear: \[ U_{C_{1}} := U_{C_{axial}} + U_{C_{shear}}^2 \quad U_{C_{1}} = 0.682 \]

Hoop Plus Shear: \[ U_{C_{2}} := U_{C_{hoop}} + U_{C_{shear}}^2 \quad U_{C_{2}} = 0.386 \]

Case 2, Loading

Vertical Load = 170.0 kips
Bending Moment = 1200.0 kips*ft
Shear Load = 488.0 kips
External Pressure = 1500 psi

First Eigenvalue: \[ \lambda_{1} := 41.477 \] Location B Governs

Axial Stress: \[ \sigma_{\phi} := 25114 \text{ psi} \]

Hoop Stress: \[ \sigma_{\theta} := 32457 \text{ psi} \]

Radial Stress: \[ \sigma_{r} := 750 \text{ psi} \]

Shear Stress: \[ \sigma_{\phi \theta} := 2010 \text{ psi} \]
Case 2. Factored Stress Components

Factored Axial Stress: \( \sigma_{\phi eL} = \lambda_1 \cdot \sigma_\phi \) \( \sigma_{\phi eL} = 1041653 \text{ psi} \)

Factored Hoop Stress: \( \sigma_{\theta eL} = \lambda_1 \cdot \sigma_\theta \) \( \sigma_{\theta eL} = 1346219 \text{ psi} \)

Factored Radial Stress: \( \sigma_{reL} = \lambda_1 \cdot \sigma_r \) \( \sigma_{reL} = 31108 \text{ psi} \)

Factored Shear Stress: \( \sigma_{\phi\theta eL} = \lambda_1 \cdot \sigma_{\phi\theta} \) \( \sigma_{\phi\theta eL} = 83369 \text{ psi} \)

Case 2. Plasticity Knockdown Factor Calculation

Section 1610 and Section 1713.1.1

Elastic Buckling Limits, Axial: \( \sigma_{xa} := \frac{\sigma_{\phi eL}}{FS} \) \( \sigma_{xa} = 436700 \text{ psi} \)

Elastic Buckling Limits, Hoop: \( \sigma_{ha} := \frac{\sigma_{\theta eL}}{FS} \) \( \sigma_{ha} = 538488 \text{ psi} \)

Elastic Buckling Limits, Radial: \( \sigma_{ra} := \frac{\sigma_{reL}}{FS} \) \( \sigma_{ra} = 12443 \text{ psi} \)

Elastic Buckling Limits, Shear: \( \sigma_{\tau a} := \frac{\sigma_{\phi\theta eL}}{FS} \) \( \sigma_{\tau a} = 33348 \text{ psi} \)

Plasticity Knockdown Parameters:

Axial: \( \Delta_\phi := \frac{\alpha L \cdot \sigma_{\phi eL}}{\sigma_Y} \) \( \Delta_\phi = 10.192 \)

Hoop: \( \Delta_\theta := \frac{\alpha L \cdot \sigma_{\theta eL}}{\sigma_Y} \) \( \Delta_\theta = 12.568 \)

Shear: \( \Delta_{\phi\theta} := \frac{\alpha L \cdot \sigma_{\phi\theta eL}}{\sigma_Y} \) \( \Delta_{\phi\theta} = 0.778 \)
Plasticity Knockdown Factors: Section 1611

**Axial:**
\[ \eta_\phi := \begin{cases} 
    1, & \text{if } \Delta_\phi < 0.55, \\
    0.91, & \text{if } 0.55 \leq \Delta_\phi < 1.0, \\
    0.79, & \text{if } 1.0 \leq \Delta_\phi < 1.6, \\
    0.64, & \text{if } 1.6 \leq \Delta_\phi < 1.9, \\
    0.5, & \text{if } 1.9 \leq \Delta_\phi < 2.5, \\
    0.45, & \text{if } \Delta_\phi \geq 2.5.
\end{cases} \]

**Hoop:**
\[ \eta_\theta := \begin{cases} 
    0.95, & \text{if } \Delta_\theta < 0.67, \\
    0.89, & \text{if } 0.67 \leq \Delta_\theta < 1.0, \\
    0.84, & \text{if } 1.0 \leq \Delta_\theta < 4.2, \\
    0.69, & \text{if } 4.2 \leq \Delta_\theta < 8.6, \\
    0.58, & \text{if } \Delta_\theta \geq 8.6.
\end{cases} \]

\[ \eta_\phi = 0.098 \quad \eta_\theta = 0.080 \]

**Shear:**
\[ \eta_{\phi\theta} := \begin{cases} 
    1, & \text{if } \Delta_{\phi\theta} < 0.48, \\
    0.96, & \text{if } 0.48 \leq \Delta_{\phi\theta} < 1.7, \\
    0.83, & \text{if } 1.7 \leq \Delta_{\phi\theta} < 4.2, \\
    0.58, & \text{if } \Delta_{\phi\theta} \geq 4.2.
\end{cases} \]

**Shear:**
\[ \eta_{\phi\theta} = 0.652 \]

**Case 2, Factored Minimum Eigenvalues**

**Axial:**
\[ \lambda_{\phi1} := \lambda_1 \cdot \alpha_\phi L \cdot \eta_\phi \]
\[ \lambda_{\phi1} = 3.412 \]

**Hoop/Radial:**
\[ \lambda_{\theta1} := \lambda_1 \cdot \alpha_\theta L \cdot \eta_\theta \]
\[ \lambda_{\theta1} = 2.640 \]

**Shear:**
\[ \lambda_{\phi\theta1} := \lambda_1 \cdot \alpha_{\phi\theta} L \cdot \eta_{\phi\theta} \]
\[ \lambda_{\phi\theta1} = 21.651 \]

**Case 2, Inelastic Buckling Limit** Section 1713.2.1

**Axial:**
\[ \sigma_{xc} := \eta_\phi \cdot \sigma_{xa} \]
\[ \sigma_{xc} = 42848-\text{psi} \]

**Hoop:**
\[ \sigma_{hc} := \eta_\theta \cdot \sigma_{ha} \]
\[ \sigma_{hc} = 42847 \text{ psi} \]

**Radial:**
\[ \sigma_{rc} := \eta_\theta \cdot \sigma_{ra} \]
\[ \sigma_{rc} = 990 \text{-psi} \]

**Shear:**
\[ \sigma_{\tau c} := \eta_{\phi\theta} \cdot \sigma_{\tau a} \]
\[ \sigma_{\tau c} = 21759 \text{-psi} \]
Case 2, Unity Checks, Unit Equations

\[ UC_{\text{axial}} := \frac{\lambda_{\text{min}}}{\lambda_{\phi 1}} \quad UC_{\text{axial}} = 0.586 \]
\[ UC_{\text{hoop}} := \frac{\lambda_{\text{min}}}{\lambda_{\theta 1}} \quad UC_{\text{hoop}} = 0.758 \]
\[ UC_{\text{radial}} := \frac{\lambda_{\text{min}}}{\lambda_{\phi 1}} \quad UC_{\text{radial}} = 0.758 \]
\[ UC_{\text{shear}} := \frac{\lambda_{\text{min}}}{\lambda_{\phi 1}} \quad UC_{\text{shear}} = 0.092 \]

Section 1713.2.1(a)

Case 2, Unity Checks, Interaction Equations

Axial Plus Shear:
\[ UC_1 := UC_{\text{axial}} + UC_{\text{shear}}^2 \quad UC_1 = 0.595 \]

Hoop Plus Shear:
\[ UC_2 := UC_{\text{hoop}} + UC_{\text{shear}}^2 \quad UC_2 = 0.766 \]

Section 1713.2.1(b) and Section 1713.2.1(c)
Flexjoint Tool Analysis
Holder/Pusher Cylinder Saddle

Subject: Strength Parameters for Elastic-Plastic FEA
Drawing Reference: Solidworks Model SV50548-248, Rev. 0
By: Dana E. Petroni
Date: June 22, 2010
Check: 
Date: 
Revised By: 

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The purpose of these calculations is to develop a true stress-true strain curve for use in elastic-plastic FEA per Part 5.2.4 and Part 5.3 of ASME Section VIII-2, 2007 Edition with 2008 Addenda.

Document Reference
ASME Section VIII, Division 2, 2008a Edition
Annex 3.D

Material: HT # 08011708

Notes: All properties taken at a single temperature of interest.
Temperature of interest taken as 70 deg. F unless stated otherwise.

Ferritic steel assumed for strain limit data. Use of ferritic steel data compared to other steels is conservative.

Material Property Input

Modulus of Elasticity: $E_y := 29.50 \cdot 10^6 \text{ psi}$

Engineering Yield Stress: $\sigma_y := 85695 \text{ psi}$

Engineering Tensile Stress: $\sigma_t := 105415 \text{ psi}$

0.2% Engineering Offset Strain: $\varepsilon_y := 0.002$

Curve Fit Parameter for Stress-Strain Curve: $\varepsilon_p := 2.00 \cdot 10^{-5}$ Table 3.D.1

Material Property Calculations

Engineering Yield to Tensile Stress Ratio: $R := \frac{\sigma_y}{\sigma_t}$ $R = 0.813$

Material Parameter for Stress-Strain Curve: $K := 1.5R^{1.5} - 0.5R^{2.5} - R^{3.5}$ $K = 0.317$

Curve Fit Exponent, Stress-Strain Curve: $m_1 := \frac{\ln(R) + (\varepsilon_p - \varepsilon_y)}{\ln(1 + \varepsilon_p)}$ $m_1 = 0.045$
Curve Fit Exponent, Stress-Strain Curve: \[ m_2 := 0.60 \cdot (1.00 - R) \]
\[ m_2 = 0.112 \]

Curve Fit Constant, Stress-Strain Curve Elastic Region:
\[ A_1 := \frac{\sigma_{ys} \cdot (1 + e_{ys})}{(\ln(1 + e_{ys}))^{m_1}} \]
\[ A_1 = 113871 \, \text{psi} \]

Curve Fit Constant, Stress-Strain Curve Plastic Region:
\[ A_2 := \frac{\sigma_{ults} \cdot e^{m_2}}{m_2 m_2} \]
\[ A_2 = 150751 \, \text{psi} \]

Function: Stress-Strain Curve Fitting Parameter
\[ f_H(\sigma_t, \sigma_{ys}, \sigma_{ults}, K) := \frac{2 \cdot [\sigma_t - [\sigma_{ys} + K \cdot (\sigma_{ults} - \sigma_{ys})]]}{K \cdot (\sigma_{ults} - \sigma_{ys})} \]

Function: True Plastic Strain, Micro-Strain Region of Stress-Strain Curve
\[ \varepsilon_{1}(\sigma_t, A_1, m_1) := \left( \frac{\sigma_t}{A_1} \right)^{m_1} \]

Function: True Plastic Strain, Macro-Strain Region of Stress-Strain Curve
\[ \varepsilon_{2}(\sigma_t, A_2, m_2) := \left( \frac{\sigma_t}{A_2} \right)^{m_2} \]

Function: True Strain, Micro-Strain Region of Stress-Strain Curve
\[ \gamma_1(\sigma_t, A_1, m_1, \sigma_{ys}, \sigma_{ults}, K) := 0.50 \cdot \varepsilon_{1}(\sigma_t, A_1, m_1) \cdot (1.0 - \tanh(f_H(\sigma_t, \sigma_{ys}, \sigma_{ults}, K))) \]

Function: True Strain, Macro-Strain Region of Stress-Strain Curve
\[ \gamma_2(\sigma_t, A_2, m_2, \sigma_{ys}, \sigma_{ults}, K) := 0.50 \cdot \varepsilon_{2}(\sigma_t, A_2, m_2) \cdot (1.0 + \tanh(f_H(\sigma_t, \sigma_{ys}, \sigma_{ults}, K))) \]

Function: True Strain
\[ f_{st}(\sigma_t, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{ults}, K) := \frac{\sigma_t}{E_y} \ldots \]
\[ + \gamma_1(\sigma_t, A_1, m_1, \sigma_{ys}, \sigma_{ults}, K) \ldots \]
\[ + \gamma_2(\sigma_t, A_2, m_2, \sigma_{ys}, \sigma_{ults}, K) \]
Function: Engineering Strain

$$f_{\text{eng}}(\sigma , E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) := e^{f_{\text{st}}(\sigma, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K)} - 1$$

Function: Engineering Stress

$$f_{\sigma_{\text{eng}}}(\sigma , E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) := \frac{\sigma}{1 + f_{\text{eng}}(\sigma, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K)}$$

Material Property Curve

Slope of stress-strain curve constant until proportional limit reached (approximately 60% of yield strength). Increment at increasing true stress values above proportional limit until maximum engineering stress value reached and make not of corresponding true stress. Assume zero tangent modulus beyond tensile strength.

$$f_{\text{st}}(0\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 0.000$$
$$f_{\text{st}}(52000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 1.762743794 \times 10^{-3}$$
$$f_{\text{st}}(56000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 1.898468353 \times 10^{-3}$$
$$f_{\text{st}}(60000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 2.035 \times 10^{-3}$$
$$f_{\text{st}}(63000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 2.138 \times 10^{-3}$$
$$f_{\text{st}}(66000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 2.243 \times 10^{-3}$$
$$f_{\text{st}}(69000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 2.355 \times 10^{-3}$$
$$f_{\text{st}}(72000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 2.482 \times 10^{-3}$$
$$f_{\text{st}}(75000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 2.644 \times 10^{-3}$$
$$f_{\text{st}}(78000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 2.885 \times 10^{-3}$$
$$f_{\text{st}}(80000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 3.134 \times 10^{-3}$$
$$f_{\text{st}}(82000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 3.510 \times 10^{-3}$$
$$f_{\text{st}}(84000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 4.105 \times 10^{-3}$$
$$f_{\text{st}}(86000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 5.084 \times 10^{-3} \quad \text{<----- Yield Stress}$$
$$f_{\text{st}}(87000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 5.810 \times 10^{-3}$$
$$f_{\text{st}}(88000\text{-psi}, E_y, A_1, A_2, m_1, m_2, \sigma_{ys}, \sigma_{uts}, K) = 6.771 \times 10^{-3}$$


\[ f_{ct}(89000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 8.040 \times 10^{-3} \]

\[ f_{ct}(90000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 9.675 \times 10^{-3} \]

\[ f_{ct}(91000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 11.659 \times 10^{-3} \]

\[ f_{ct}(92000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 13.849 \times 10^{-3} \]

\[ f_{ct}(93000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 16.019 \times 10^{-3} \]

\[ f_{ct}(94000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 18.014 \times 10^{-3} \]

\[ f_{ct}(95000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 19.833 \times 10^{-3} \]

\[ f_{ct}(96000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 21.570 \times 10^{-3} \]

\[ f_{ct}(97000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 23.330 \times 10^{-3} \]

\[ f_{ct}(98000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 25.191 \times 10^{-3} \]

\[ f_{ct}(99000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 27.199 \times 10^{-3} \]

\[ f_{ct}(100000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 29.383 \times 10^{-3} \]

\[ f_{ct}(101000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 31.760 \times 10^{-3} \]

\[ f_{ct}(102000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 34.343 \times 10^{-3} \]

\[ f_{ct}(103000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 37.145 \times 10^{-3} \]

\[ f_{ct}(104000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 40.178 \times 10^{-3} \]

\[ f_{ct}(105000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 43.455 \times 10^{-3} \]

\[ f_{ct}(106000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 46.992 \times 10^{-3} \]

\[ f_{ct}(107000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 50.804 \times 10^{-3} \]

\[ f_{ct}(108000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 54.909 \times 10^{-3} \]

\[ f_{ct}(109000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 59.325 \times 10^{-3} \]

\[ f_{ct}(110000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 64.072 \times 10^{-3} \]

\[ f_{ct}(111000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 69.171 \times 10^{-3} \]

\[ f_{ct}(112000\text{-psi}, Ey, A1, A2, m1, m2, \sigma_{sys}, \sigma_{uts}, K) = 74.644 \times 10^{-3} \]
Flexjoint Tool Analysis
Holder/Pusher Cylinder Saddle

\[
\begin{align*}
\text{f}_{\text{et}}(113000 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 80.516 \times 10^{-3} \\
\text{f}_{\text{et}}(114000 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 86.812 \times 10^{-3} \\
\text{f}_{\text{et}}(115000 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 93.557 \times 10^{-3} \\
\text{f}_{\text{et}}(116000 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 100.781 \times 10^{-3} \\
\text{f}_{\text{et}}(117000 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 108.512 \times 10^{-3} \\
\text{f}_{\text{et}}(117350 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 111.343 \times 10^{-3} \\
\text{f}_{\text{eng}}(117350 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 117.779 \times 10^{-3} \\
\sigma_{\text{eng}}(117350 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 104985 \text{ psi} \\
\text{f}_{\text{et}}(117700 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 114.242 \times 10^{-3} \\
\text{f}_{\text{eng}}(117700 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 121.023 \times 10^{-3} \\
\sigma_{\text{eng}}(117700 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 104993 \text{ psi} \\
\text{f}_{\text{et}}(117900 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 115.929 \times 10^{-3} \\
\text{f}_{\text{eng}}(117900 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 122.916 \times 10^{-3} \\
\sigma_{\text{eng}}(117900 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 104994 \text{ psi} \\
\text{f}_{\text{et}}(118100 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 117.639 \times 10^{-3} \\
\text{f}_{\text{eng}}(118100 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 124.838 \times 10^{-3} \\
\sigma_{\text{eng}}(118100 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 104993 \text{ psi} \\
\text{f}_{\text{et}}(118450 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 120.686 \times 10^{-3} \\
\text{f}_{\text{eng}}(118450 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 128.271 \times 10^{-3} \\
\sigma_{\text{eng}}(118450 \text{-psi}, \text{Ey, A1, A2, m1, m2, } \sigma_{\text{ys}}, \sigma_{\text{uts}}, K) &= 104994 \text{ psi}
\end{align*}
\]

Maximum Engineering Stress (Tensile Strength) at True Stress = 117900 psi
Tangent Modulus of Zero Beyond This Point
Elastic-Plastic Analysis

Von-Mises Plastic Strain
Load Multiplier = 1.00

File: ..\SV50548-249_CLAMP-FEA.x_t (in./in.)
Elastic-Plastic Analysis

Von-Mises Plastic Strain
Load Multiplier = 2.00

File: ..\8V50548-248_CLAMP-FEA.x_t

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HIGHLY CONFIDENTIAL

TREX-120129.274
Elastic-Plastic Analysis

Von-Mises Plastic Strain
Load Multiplier = 3.00
Elastic-Plastic Analysis

Von-Mises Plastic Strain
Load Multiplier = 5.00

File: ..\SV50548-248_CLAMP-PER.Ax_t

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HIGHLY CONFIDENTIAL

TREX-120129.276
Analysis Stopped at
10.0 x 110.4 k = 1104 k

Elastic-Plastic Analysis
Von-Mises Plastic Strain
Load Multiplier = 10.00

File: ..\SV50548-240_CLAMP-FEA.x_t
Elastic-Plastic Analysis
Von-Mises Plastic Strain
Load Multiplier = 2.00
Elastic-Plastic Analysis
Von-Mises Plastic Strain Load Multiplier = 3.00

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HIGHLY CONFIDENTIAL

TREX-120129.279
Analysis Stopped at
Design Load x 5.00

Elastic-Plastic Analysis
Von-Mises Plastic Strain
Load Multiplier = 5.00

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HIGHLY CONFIDENTIAL

TREX-120129.280
Elastic-Plastic Analysis

Von-Mises Plastic Strain
Load Multiplier = 10.00

Analysis Stopped at
10.0 x 110.4 k = 1104 k
Subject: Plastic Strain Check for Elastic-Plastic FEA
Drawing Reference: Solidworks Model SV50548-248, Rev. 2

By: Dana E. Petroni  
Date: June 23, 2010  
Check:  
Date:  

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The purpose of these calculations is to check plastic strain levels for an elastic-plastic FEA per Part 5.3.3 of ASME Section VIII-2, 2007 Edition with 2008 Addenda.

Document Reference  
ASME Section VIII, Division 2, 2008a Edition  
Part 5.3.3

Material:  
HT # 08011708

Notes:  
All properties taken at a single temperature of interest.  
Temperature of interest taken as 70 deg. F unless stated otherwise.

Ferritic steel assumed for strain limit data. Use of ferritic steel data compared to other steels is conservative.

Fatigue evaluation is not part of this evaluation. Calculation assumes only a limited number of load applications.

Material Property Input

Engineering Yield Stress:  
\( \sigma_{YS} := 85695 \text{ psi} \)

Engineering Tensile Stress:  
\( \sigma_{UTS} := 105415 \text{ psi} \)

Specified Elongation Percent:  
E := 19  
Assumed

Specified Area Reduction Percent:  
RA := 0.00  
Not Specified

Strain Limit Calculations

Engineering Yield to Tensile Stress Ratio:  
\( R := \frac{\sigma_{YS}}{\sigma_{UTS}} \)  
\( R = 0.813 \)

Uniaxial Strain Limit Parameter:  
\( \alpha_{GL} := 2.20 \)

Uniaxial Strain Limit Parameter:  
\( m_2 := 0.60 \cdot (1.00 - R) \)  
\( m_2 = 0.112 \)

Uniaxial Strain Limit Parameters per Table 5.7, ASME-VIII-2
Forwning Strain: $\varepsilon_{cf} := 0.000$

Assumed Heat Treatment in Accordance with ASME-VIII-2, Part 6 or equivalent standard

**Strain Limit Equations per Table 5.7, ASME-VIII-2**

**Uniaxial Strain Limit Value, Material Basis:**

$\varepsilon_{lu1} := m_2$

$\varepsilon_{lu1} = 0.112$

**Uniaxial Strain Limit Value, Elongation Basis:**

$\varepsilon_{lu2} := 2 \ln \left(1 + \frac{E}{100}\right)$

$\varepsilon_{lu2} = 0.348$

**Uniaxial Strain Limit Value, Area Reduction Basis:**

$\varepsilon_{lu3} := \ln \left(\frac{100}{100 - RA}\right)$

$\varepsilon_{lu3} = 0.000$

**Limiting Uniaxial Strain Limit:**

$\varepsilon_{lu} := \max(\varepsilon_{lu1}, \varepsilon_{lu2}, \varepsilon_{lu3})$

$\varepsilon_{lu} = 0.348$

**Equivalent Stress:**

$f_{\sigma c}(\sigma_1, \sigma_2, \sigma_3) := \frac{1}{\sqrt{2}} \sqrt{\left(\sigma_1 - \sigma_2\right)^2 + \left(\sigma_2 - \sigma_3\right)^2 + \left(\sigma_3 - \sigma_1\right)^2}$

Equation 5.1, ASME-VIII-2

**Limiting Triaxial Strain:**

$f_{\varepsilon L}(\sigma_1, \sigma_2, \sigma_3, \sigma_e, \sigma_{sl}, m_2, \varepsilon_{lu}) := \varepsilon_{lu} e^{\frac{\alpha_{sl}}{1 + m_2} \left[\\frac{\sigma_1 + \sigma_2 + \sigma_3}{3 - \sigma_e}\right]^{\frac{1}{3}}} - \frac{1}{3}$

Equation 5.6, ASME-VIII-2

**Strain Check:**

$f_{\varepsilon UC}(\varepsilon_{peq}, \varepsilon_{cf}, \varepsilon_L) := \frac{\varepsilon_{peq} + \varepsilon_{cf}}{\varepsilon_L}$

Equation 5.7, ASME-VIII-2
Flexjoint Tool Analysis
Holder/Pusher Cylinder Saddle

Design Load Multiplier = 1.70, Governing Region, Lower Radius

Principal Stresses:
\[ \sigma_1 := 137834 \text{ psi} \quad \sigma_2 := 85583 \text{ psi} \quad \sigma_3 := 74716 \text{ psi} \]

Total Equivalent Plastic Strain:
\[ \varepsilon_{\text{peq}} := 0.003126 \]
\[ \varepsilon_L := f_L(\sigma_1, \sigma_2, \sigma_3, \varepsilon_{\text{cf}}, \alpha_s, m_2, \varepsilon_{\text{Lu}}) \quad \varepsilon_L = 0.0233 \]

UC := \frac{\varepsilon_{\text{peq}}}{\varepsilon_{\text{cf}}} \quad UC = 0.134 \quad <---- Evaluation Passes Based on 1.70 Design Load Multiplier

Design Load Multiplier = 2.50, Governing Region, Lower Radius

First Principal Stress:
\[ \sigma_1 := 178608 \text{ psi} \quad \sigma_2 := 120634 \text{ psi} \quad \sigma_3 := 108863 \text{ psi} \]

Total Equivalent Plastic Strain:
\[ \varepsilon_{\text{peq}} := 0.007642 \]

Design Load Multiplier = 2.70, Governing Region, Lower Radius

First Principal Stress:
\[ \sigma_1 := 185021 \text{ psi} \quad \sigma_2 := 126672 \text{ psi} \quad \sigma_3 := 114578 \text{ psi} \]

Total Equivalent Plastic Strain:
\[ \varepsilon_{\text{peq}} := 0.009163 \]

Local Plastic Strain Evaluation Fails at Design Load Multiplier = 2.67 > 1.70, O.K.
Subject: Plastic Strain Check for Elastic-Plastic FEA
Drawing Reference: Solidworks Model SV50548-248, Rev. 2

By: Dana E. Petroni
Date: June 29, 2010
Check: 
Date: 

Flexible Tool Analysis
Holder/Pusher Cylinder Saddle

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The purpose of these calculations is to check plastic strain levels for an elastic-plastic FEA per Part 5.3.3 of ASME Section VIII-2, 2007 Edition with 2008 Addenda.

Document Reference
ASME Section VIII, Division 2, 2008a Edition
Part 5.3.3

Material: HT # 08011708

Notes: All properties taken at a single temperature of interest.
Temperature of interest taken as 70 deg. F unless stated otherwise.

Ferritic steel assumed for strain limit data. Use of ferritic steel data compared to other steels is conservative.

Fatigue evaluation is not part of this evaluation. Calculation assumes only a limited number of load applications.

Material Property Input

Engineering Yield Stress: \( \sigma_{YS} := 85695 \text{ psi} \)

Engineering Tensile Stress: \( \sigma_{UTS} := 105415 \text{ psi} \)

Specified Elongation Percent: \( E := 19 \) Assumed

Specified Area Reduction Percent: \( RA := 0.00 \) Not Specified

Strain Limit Calculations

Engineering Yield to Tensile Stress Ratio: \( R := \frac{\sigma_{YS}}{\sigma_{UTS}} \) \( R = 0.813 \)

Uniaxial Strain Limit Parameter: \( \alpha_{UL} := 2.20 \)

Uniaxial Strain Limit Parameter: \( m_2 := 0.60 \times (1.00 - R) \) \( m_2 = 0.112 \)

Uniaxial Strain Limit Parameters per Table 5.7, ASME-VIII-2
Forcing Strain: $\varepsilon_{cf} := 0.000$

Assumed Heat Treatment in Accordance with ASME-VIII-2, Part 6 or equivalent standard

Strain Limit Equations per Table 5.7, ASME-VIII-2

Uniaxial Strain Limit Value, Material Basis: $\varepsilon_{Lu1} := m^2$

$\varepsilon_{Lu1} = 0.112$

Uniaxial Strain Limit Value, Elongation Basis: $\varepsilon_{Lu2} := 2 \ln \left( 1 + \frac{E}{100} \right)$

$\varepsilon_{Lu2} = 0.348$

Uniaxial Strain Limit Value, Area Reduction Basis: $\varepsilon_{Lu3} := \ln \left( \frac{100}{100 - RA} \right)$

$\varepsilon_{Lu3} = 0.000$

Limiting Uniaxial Strain Limit: $\varepsilon_{Lu} := \max(\varepsilon_{Lu1}, \varepsilon_{Lu2}, \varepsilon_{Lu3})$

$\varepsilon_{Lu} = 0.348$

Equivalent Stress:

$$f_\sigma(\sigma_1, \sigma_2, \sigma_3) := \frac{1}{\sqrt{2}} \sqrt{\left(\sigma_1 - \sigma_2\right)^2 + \left(\sigma_2 - \sigma_3\right)^2 + \left(\sigma_3 - \sigma_1\right)^2}$$

Equation 5.1, ASME-VIII-2

Limiting Triaxial Strain:

$$f_\varepsilon(\sigma_1, \sigma_2, \sigma_3, \varepsilon_{eq}, \varepsilon_{cf}, m^2, \varepsilon_{Lu}) := \varepsilon_{Lu}^3 - \frac{\alpha_{sl}}{1 + m^2} \left[ \frac{\sigma_1 + \sigma_2 + \sigma_3}{3 \cdot \varepsilon_{eq}} \right]^{\frac{1}{3}}$$

Equation 5.6, ASME-VIII-2

Strain Check:

$$f_{UC}(\varepsilon_{eq}, \varepsilon_{cf}, \varepsilon_{Lu}) := \frac{\varepsilon_{eq} + \varepsilon_{cf}}{\varepsilon_{Lu}}$$

Equation 5.7, ASME-VIII-2
Design Load Multiplier = 9.00, Upper Radius  
\[ 110.4 \times 9.0 = 993.6 \text{ kips} \]

Principal Stresses:
\[ \sigma_1 := 144274 \text{-psi} \quad \sigma_2 := 78817 \text{-psi} \quad \sigma_3 := 65336 \text{-psi} \]

Total Equivalent Plastic Strain:
\[ \varepsilon_{\text{peq}} := 0.030845 \]
\[ \varepsilon_L := f_{\varepsilon_L}(\sigma_1, \sigma_2, \sigma_3, \varepsilon_{\text{cf}}, \varepsilon_L) \quad \varepsilon_L = 0.0500 \]
\[ \text{UC} := f_{\text{UC}}(\varepsilon_{\text{peq}}, \varepsilon_{\text{cf}}, \varepsilon_L) \quad \text{UC} = 0.618 \]

Design Load Multiplier = 9.50, Upper Radius  
\[ 110.4 \times 9.5 = 1048.8 \text{ kips} \]

First Principal Stress:
\[ \sigma_1 := 148714 \text{-psi} \quad \sigma_2 := 81194 \text{-psi} \quad \sigma_3 := 65924 \text{-psi} \]

Total Equivalent Plastic Strain:
\[ \varepsilon_{\text{peq}} := 0.041922 \]
\[ \sigma_e := f_{\sigma_e}(\sigma_1, \sigma_2, \sigma_3) \quad \sigma_e = 76310 \text{psi} \]
\[ \varepsilon_L := f_{\varepsilon_L}(\sigma_1, \sigma_2, \sigma_3, \sigma_e, \sigma_{\text{si}}, m_2, \varepsilon_L) \quad \varepsilon_L = 0.0522 \]
\[ \text{UC} := f_{\text{UC}}(\varepsilon_{\text{peq}}, \varepsilon_{\text{cf}}, \varepsilon_L) \quad \text{UC} = 0.803 \]

Design Load Multiplier = 10.00, Upper Radius  
\[ 110.4 \times 10.0 = 1104.0 \text{ kips} \]

First Principal Stress:
\[ \sigma_1 := 153963 \text{-psi} \quad \sigma_2 := 84418 \text{-psi} \quad \sigma_3 := 66745 \text{-psi} \]

Total Equivalent Plastic Strain:
\[ \varepsilon_{\text{peq}} := 0.059532 \]
\[ \sigma_e := f_{\sigma_e}(\sigma_1, \sigma_2, \sigma_3) \quad \sigma_e = 79862 \text{psi} \]
\[ \varepsilon_L := f_{\varepsilon_L}(\sigma_1, \sigma_2, \sigma_3, \sigma_e, \sigma_{\text{si}}, m_2, \varepsilon_L) \quad \varepsilon_L = 0.0542 \]
\[ \text{UC} := f_{\text{UC}}(\varepsilon_{\text{peq}}, \varepsilon_{\text{cf}}, \varepsilon_L) \quad \text{UC} = 1.099 \]

Per Second-Order Interpolation of Results Above, Approximate Load Where
\[ \text{UC} = 1 \text{ is about } 1080.0 \text{ kips.} \]

Oil States Industries, Inc.
Houston Operations
Houston, Texas
Flexjoint Tool Analysis
Holder/Pusher Cylinder Saddle

Subject: Plastic Strain Check for Elastic-Plastic FEA
Drawing Reference: Solidworks Model SV50548-248, Rev. 2

By: Dana E. Petroni  Revised By:
Date: June 29, 2010  Date:
Check:  Check:
Date:  Date:

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The purpose of these calculations is to check plastic strain levels for an elastic-plastic FEA per Part 5.3.3 of ASME Section VIII-2, 2007 Edition with 2008 Addenda.

Document Reference
ASME Section VIII, Division 2, 2008a Edition
Part 5.3.3

Material:
HT # 08011708

Notes:
- All properties taken at a single temperature of interest.
- Temperature of interest taken as 70 deg. F unless stated otherwise.
- Ferritic steel assumed for strain limit data. Use of ferritic steel data compared to other steels is conservative.
- Fatigue evaluation is not part of this evaluation. Calculation assumes only a limited number of load applications.

Material Property Input

Engineering Yield Stress:
\[ \sigma_{YS} := 85695 \text{-psi} \]

Engineering Tensile Stress:
\[ \sigma_{UTS} := 105415 \text{-psi} \]

Specified Elongation Percent:

\[ E := 19 \text{ Assumed} \]

Specified Area Reduction Percent:

\[ RA := 0.00 \text{ Not Specified} \]

Strain Limit Calculations

Engineering Yield to Tensile Stress Ratio:
\[ R := \frac{\sigma_{YS}}{\sigma_{UTS}} \quad R = 0.813 \]

Uniaxial Strain Limit Parameter:
\[ \alpha_{SL} := 2.20 \]

Uniaxial Strain Limit Parameter:
\[ m_2 := 0.60 \times (1.00 - R) \quad m_2 = 0.112 \]

Uniaxial Strain Limit Parameters per Table 5.7, ASME-VIII-2
Flexjoint Tool Analysis
Holder/Pusher Cylinder Saddle

Forming Strain: \( \varepsilon_{cf} := 0.000 \)
Assumed Heat Treatment in Accordance with ASME-VIII-2, Part 6 or equivalent standard

Strain Limit Equations per Table 5.7, ASME-VIII-2

Uniaxial Strain Limit Value, Material Basis:
\( \varepsilon_{Lu1} := m_2 \)
\( \varepsilon_{Lu1} = 0.112 \)

Uniaxial Strain Limit Value, Elongation Basis:
\( \varepsilon_{Lu2} := 2 \ln \left( 1 + \frac{E}{100} \right) \)
\( \varepsilon_{Lu2} = 0.348 \)

Uniaxial Strain Limit Value, Area Reduction Basis:
\( \varepsilon_{Lu3} := \ln \left( \frac{100}{100 - RA} \right) \)
\( \varepsilon_{Lu3} = 0.000 \)

Limiting Uniaxial Strain Limit:
\( \varepsilon_{Lu} := \max(\varepsilon_{Lu1}, \varepsilon_{Lu2}, \varepsilon_{Lu3}) \)
\( \varepsilon_{Lu} = 0.348 \)

Equivalent Stress:
\[
f_{\sigma}(\sigma_1, \sigma_2, \sigma_3) := \frac{1}{\sqrt{2}} \sqrt{\left(\sigma_1 - \sigma_2\right)^2 + \left(\sigma_2 - \sigma_3\right)^2 + \left(\sigma_3 - \sigma_1\right)^2}
\]
Equation 5.1, ASME-VIII-2

Limiting Triaxial Strain:
\[
f_{L}(\sigma_1, \sigma_2, \sigma_3, \sigma_e, \alpha_{sl}, m_2, \varepsilon_{Lu}) := \varepsilon_{Lu} e^{-\left(\frac{\alpha_{sl}}{1+m_2}\right)\left[\frac{(\sigma_1+\sigma_2+\sigma_3)}{3-\sigma_e}\right]^{1/3}}
\]
Equation 5.6, ASME-VIII-2

Strain Check:
\[
f_{UC}(\varepsilon_{peq}, \varepsilon_{cf}, \varepsilon_{L}) := \frac{\varepsilon_{peq} + \varepsilon_{cf}}{\varepsilon_{L}}
\]
Equation 5.7, ASME-VIII-2
Design Load Multiplier = 9.00, Cylinder Contact Region

\[ \sigma_1 := 156565 \text{-psi} \quad \sigma_2 := 81138 \text{-psi} \quad \sigma_3 := 47824 \text{-psi} \]

Total Equivalent Plastic Strain:

\[ \varepsilon_{\text{peq}} := 0.024323 \]

\[ \sigma_e := f_{\sigma e}(\sigma_1, \sigma_2, \sigma_3) \quad \sigma_e = 96498 \text{psi} \]

\[ \varepsilon_L := f_{\varepsilon L}(\sigma_1, \sigma_2, \sigma_3, \sigma_e, \alpha_{SI}, m_2, \varepsilon_{Lu}) \quad \varepsilon_L = 0.0956 \]

\[ UC := f_{UC}(\varepsilon_{\text{peq}}, \varepsilon_{cf}, \varepsilon_L) \quad UC = 0.254 \]

Design Load Multiplier = 9.50, Cylinder Contact Region

\[ \sigma_1 := 166038 \text{-psi} \quad \sigma_2 := 88764 \text{-psi} \quad \sigma_3 := 53404 \text{-psi} \]

Total Equivalent Plastic Strain:

\[ \varepsilon_{\text{peq}} := 0.032701 \]

\[ \sigma_e := f_{\sigma e}(\sigma_1, \sigma_2, \sigma_3) \quad \sigma_e = 99770 \text{psi} \]

\[ \varepsilon_L := f_{\varepsilon L}(\sigma_1, \sigma_2, \sigma_3, \sigma_e, \alpha_{SI}, m_2, \varepsilon_{Lu}) \quad \varepsilon_L = 0.0877 \]

\[ UC := f_{UC}(\varepsilon_{\text{peq}}, \varepsilon_{cf}, \varepsilon_L) \quad UC = 0.373 \]

Design Load Multiplier = 10.00, Cylinder Contact Region

\[ \sigma_1 := 178189 \text{-psi} \quad \sigma_2 := 98855 \text{-psi} \quad \sigma_3 := 60832 \text{-psi} \]

Total Equivalent Plastic Strain:

\[ \varepsilon_{\text{peq}} := 0.046117 \]

\[ \sigma_e := f_{\sigma e}(\sigma_1, \sigma_2, \sigma_3) \quad \sigma_e = 103712 \text{psi} \]

\[ \varepsilon_L := f_{\varepsilon L}(\sigma_1, \sigma_2, \sigma_3, \sigma_e, \alpha_{SI}, m_2, \varepsilon_{Lu}) \quad \varepsilon_L = 0.0785 \]

\[ UC := f_{UC}(\varepsilon_{\text{peq}}, \varepsilon_{cf}, \varepsilon_L) \quad UC = 0.587 \]

Per Second-Order Extrapolation of Results Above, Approximate Load Where

\[ UC = 1 \text{ is about 1170.0 kips.} \]
NODAL SOLUTION
STEP = 1
SUB = 2
TIMK = 1
EPFLEGV (AVG)
DMX = 0.004514
SMX = 0.001531

Elastic-Plastic Analysis
Von-Mises Plastic Strain
Load Multiplier = 1.00

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TREX-120129.300
NODAL SOLUTION

STEP=1
SUB =4
TIME=2
EPPEQV (AVG)
DMX = .010467
SMX = .021563

Elastic-Plastic Analysis

Von-Mises Plastic Strain
Load Multiplier = 2.00

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TREX-120129.301
Subject: Plastic Strain Check for Elastic-Plastic FEA
Drawing Reference: Solidworks Model SV50548-247, Rev. 3
By: Dana E. Petroni  Revised By:
Date: June 29, 2010  Date:
Check:  Check:
Date:  Date:

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The purpose of these calculations is to check plastic strain levels for an elastic-plastic FEA per Part 5.3.3 of ASME Section VIII-2, 2007 Edition with 2008 Addenda.

Document Reference
ASME Section VIII, Division 2, 2008a Edition
Part 5.3.3

Material:
HT # 08011708

Notes:
All properties taken at a single temperature of interest.
Temperature of interest taken as 70 deg. F unless stated otherwise.
Ferritic steel assumed for strain limit data. Use of ferritic steel data compared to other steels is conservative.
Fatigue evaluation is not part of this evaluation. Calculation assumes only a limited number of load applications.

Material Property Input

Engineering Yield Stress:
\( \sigma_{ys} := 85695 \text{-psi} \)

Engineering Tensile Stress:
\( \sigma_{uts} := 105415 \text{-psi} \)

Specified Elongation Percent:
\( E := 19.0 \)  Assumed

Specified Area Reduction Percent:
\( RA := 0.00 \)  Not Specified

Strain Limit Calculations

Engineering Yield to Tensile Stress Ratio:
\( R := \frac{\sigma_{ys}}{\sigma_{uts}} \)
\( R = 0.813 \)

Uniaxial Strain Limit Parameter:
\( \alpha_{sl} := 2.20 \)

Uniaxial Strain Limit Parameter:
\( m_2 := 0.60 \cdot (1.00 - R) \)
\( m_2 = 0.112 \)

Uniaxial Strain Limit Parameters per Table 5.7, ASME-VIII-2
Forming Strain: $\varepsilon_{cf} := 0.000$

Assumed Heat Treatment in Accordance with ASME-VIII-2, Part 6 or equivalent standard

Strain Limit Equations per Table 5.7, ASME-VIII-2

Uniaxial Strain Limit Value, Material Basis:

$$\varepsilon_{LU1} := \frac{m_2}{100}$$

$$\varepsilon_{LU1} = 0.112$$

Uniaxial Strain Limit Value, Elongation Basis:

$$\varepsilon_{LU2} := 0.348$$

Uniaxial Strain Limit Value, Area Reduction Basis:

$$\varepsilon_{LU3} := \ln\left(\frac{100}{100 - RA}\right)$$

$$\varepsilon_{LU3} = 0.000$$

Limiting Uniaxial Strain Limit:

$$\varepsilon_{LU} := \max(\varepsilon_{LU1}, \varepsilon_{LU2}, \varepsilon_{LU3})$$

$$\varepsilon_{LU} = 0.348$$

Equivalent Stress:

$$f_{\sigma}(\sigma_1, \sigma_2, \sigma_3) := \frac{1}{\sqrt{2}} \sqrt{\left(\sigma_1 - \sigma_2\right)^2 + \left(\sigma_2 - \sigma_3\right)^2 + \left(\sigma_3 - \sigma_1\right)^2}$$

Equation 5.1, ASME-VIII-2

Limiting Triaxial Strain:

$$f_{\varepsilon}(\sigma_1, \sigma_2, \sigma_3, \sigma_e, \sigma_{sl}, m_2, \varepsilon_{LU}) := \varepsilon_{LU} e^{\left[-\frac{\sigma_{sl}}{1 + m_2}\left[\frac{\sigma_1 + \sigma_2 + \sigma_3}{3 - \sigma_e}\right]\frac{1}{3}\right]}$$

Equation 5.6, ASME-VIII-2

Strain Check:

$$f_{UC}(\varepsilon_{peq}, \varepsilon_{cf}, \varepsilon_{LU}) := \frac{\varepsilon_{peq} + \varepsilon_{cf}}{\varepsilon_{LU}}$$

Equation 5.7, ASME-VIII-2
Design Load Multiplier = 1.70, Governing Region, Nut/Pusher Contact

Principal Stresses: $\sigma_1 := 169861 \text{ psi}$ \hspace{0.5cm} $\sigma_2 := 107290 \text{ psi}$ \hspace{0.5cm} $\sigma_3 := 60351 \text{ psi}$

Total Equivalent Plastic Strain: $\varepsilon_{\text{peq}} := 0.013721$

$\sigma_e := f_{\sigma e}(\sigma_1, \sigma_2, \sigma_3)$ \hspace{0.5cm} $\sigma_e = 95160 \text{ psi}$

$\varepsilon_L := f_{\varepsilon L}(\sigma_1, \sigma_2, \sigma_3, \sigma_e, \alpha_s, m_2, \varepsilon_{Lu})$ \hspace{0.5cm} $\varepsilon_L = 0.0649$

$UC := f_{UC}(\varepsilon_{\text{peq}}, \varepsilon_{cf}, \varepsilon_L)$ \hspace{0.5cm} $UC = 0.211$ \hspace{0.5cm} \text{Evaluation Passes Based on 1.70 Design Load Multiplier OK}$

Oil States Industries, Inc.
Houston Operations
Houston, Texas

Doc. No. SV50548-813
Revision: 0
Sheet XX of XX

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LNL004-026285
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Attachment: Function and Pressure Test Procedure for Well Cap Stack3 per test. pdf
Capping Stack Design Report

1.0 Introduction

This document reports the design information of the Cap stack components, bending loads on the BOP side outlets and design analysis of the BOP Valve Support Frame.

2.0 Scope

Design information is being provided on the following:

- Cameron 18 ¾-15ksi HC collet connector
- Hydril Single Ram BOP
- Hydril Double Ram BOP
- WOM Flanged Double Block Valves
- Cameron Mini-Connector Mandrels
- Vetco Mandrel
- Bending Loads on the Hydril BOP Outlets
- BOP Valve Support Frame and Design Analysis
- Choke and Burst Disk Load Considerations

3.0 Component Description and Design Information

- Cameron HC connector:

18 ¾ -15ksi working pressure designed, manufactured and tested to meet API 16A Specification for Drill-through Equipment Requirements with 18 ¾ 15ksi studded connection with Cameron proprietary CX gasket top x AX gasket bottom. This connector was pressure tested per API 16A to 1 ½ times working pressure in the factory and subsequently tested to 15ksi working pressure in the stack. This HC connector is a standard piece of equipment used throughout the industry for the last 30 years. The interface between the back sides of the collets and the actuating cylinder is a self-locking 4 degree taper so it is not necessary to maintain locking pressure after it is locked. However, as a safeguard the locking pressure to the actuating cylinder after we are latched will be locked in with valves on the panel.
HC Collet Connector
The HC Collet Connector is similar to the popular Model 70, but is designed to provide greater preload to withstand higher separating forces. Features include:
- Short "swallow up", tapered hub profile and positive unlocking system allow disconnection at angles up to 30°.
- Shortest "swallow up" of any major weldhead connector available on the market.
- Large actuating piston area creates a greater preload at a given hydraulic pressure than the Model 70.
- Higher applied loads can be tolerated without causing hub face separation because of the higher preloads.
- Uses annular cylinder with greater annular piston area for larger locking and unlocking force.
- Compact design minimizes height and weight.

All Cameron collet connectors have the following features:
- High strength and stiffness with a direct "load path" through the connector.
- Metal-to-metal sealing AX gaskets on the collet end ensure seal integrity. A new version of the AX gasket with a contingency seal may be used with the DMHC.
- Positively driven open collet segments during unlock ensure that no overpull is required to disconnect. A secondary unlock function is also included in the design.
- Insolent delays on all sealing surfaces exposed to seawater.
- Available with studded, flanged or clamped legs.
- Available with secondary unlock pistons.
- Inherent self-locking characteristics.

Structural Load Capacities, 15K Bore Pressures Based on membrane plus bending at yield

HC Collet Connector, 18-3/4" 15,000 psi WP

The published bending capacity at 10ksi is 3.38 million ft-lbs at zero tension. In our situation with no tensional loads applied during service the maximum bending moment applied is 1.28 million ft-lbs at 10ksi applied pressure.
- **Hydril 18 ¾ 15ksi Hydril Single and Double Ram BOP**

The Hydril Single and Double Ram BOP's are designed, manufactured and tested to meet API 16A Specification for Drill-through Equipment Requirements. Each BOP has an API 18 ¾ 15ksi flanged bottom connection and API 18 ¾ 15ksi studded top connection and is rated to 15ksi. Both the single and double ram have 4 1/16-15ksi API studded outlet connections.

These BOP's were pressure tested per API 16A to 1 ½ times working pressure in the factory and subsequently tested to 15ksi working pressure in the stack. This type single and double ram BOP is a standard piece of equipment used throughout the industry for well over 30 years.

These ram BOP's are hydraulically operated pistons to move the rams to the closed position. Also included in the hydraulic system are ratchet type locking mechanisms so that these BOP's once closed are locked in the close position. As a safeguard we will lock closing pressure on the rams to assure these rams, once closed, remain in the closed position until otherwise operated to open.

While our experience with these BOP's operating under producing conditions is minimal, these situations are not uncommon to the use of these BOP's during operating conditions where choke and kill situations occur. During these scenarios we are flowing through the choke lines back to surface to a choke during well control operations even though the flowrates are much smaller.
Hydrl Pressure Control
Compac™ Ram Blowout
Preventer

Applications & Benefits

The 18¾", 15,000 psi Hydrl Pressure Control Compact™ ram BOP delivers proven reliability for safe operation in the harshest environments. Rated for water depths to 15,000+ ft, the Compact ram has worked on the seabed continuously for one year during drilling of a 34,000-ft well. It can also add hundreds of hours of drilling time annually when configured to seal upside-down as a Subsea Stock Test Valve (SSTV) so that required pressure tests can be completed with the drill pipe in place.

This BOP has:
• Successfully held full rated working pressure at 300°F for eight hours using blind/shear rams; the tested unit then completed three additional closing cycles at low and high pressure.
• Passed API temperature testing requirements to 350°F using blind/shear rams.
• Passed API temperature tests to 500°F using fixed-bore rams.
• All fixed-bore ram sizes on the 18-15 Compact ram are rated for this temperature.
• Passed API pressure tests at temperatures as low as 30°F for 5½" to 7½" and 4½" to 7" variable rams.
• Sheared large-diameter pipe (16½, 109 lb-per-ft, P-110) with closing pressure of 4,000 psi.

Key Features:
• Field-replaceable upper seal seats, bottom wear plates and hydraulic manifold virtually eliminate the need to move the unit to an authorized repair facility for service.
• The bonnet seal carrier ring has a pressure-assisted design that enables the BOP to withstand external pressure differentials as high as 7,000 psi, equivalent to a depth of 15,000+ ft. This can occur in deep water in the event of large pressure drop in the well bore.
• The single set of high-performance elastomers handles both hot and cold temperature extremes.
• Multiple-position lock uses a reliable, mechanical clutch to automatically lock the rams in a seal-off position.
• Hydraulic manifold increases operating flexibility; it can be installed on either side of the BOP with the bonnets opening on the opposite side to fit various stack configurations.
• Corrosion-resistant alloy is used in ring grooves, bonnet seating area, seat bores and piston rod bore seal area for long life.

Because testing requirements are becoming increasingly severe, we have upgraded test center capabilities well beyond the HPHT limits sought by the industry today. Current capabilities include:
• High-temperature testing to 500°F
• Low-temperature testing to -60°F
• Shear testing for large-diameter and heavy-wall pipe
• Pressure testing with hang-off loads up to 600,000 lbs
• External Hydrostatic testing to 7,000 psi

26 June 2010

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TREX-120129.318

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LNL004-026297
WOM Flanged Double Block Valves
The WOM Subsea Dual Block Gate Valves are designed, manufactured, and tested to meet the requirements of API 16C Specification for Choke and Kill Systems. The block valves have a 4 1/16-15ksi flanged inlet with 3 1/16-15ksi thru bore and outlets and is rated to 15ksi.
This dual valve block gate valve was pressure tested per API 16C to 1 1/2 times working pressure in the factory and subsequently tested to 15ksi working pressure in the stack. This block valve is a standard piece of equipment used throughout the industry and been in service since 1991.
This type valve is also used in subsea production applications as well as test trees for well test operations.
These particular valves rather than fail close are fitted to be fail open in order to prevent over pressuring the well.
Cameron Min-Connector Mandrels
This Cameron mandrel has been designed, manufactured and tested to meet API 16C Specification for Choke and Kill Systems.
This mandrel has a API 3 1/6-15ksi Bottom flange x a Cameron proprietary hub connection and proprietary AX gasket preparation and is rated to 15ksi. These mandrels have been used in this industry for more than 30 years.
The mandrel has been tested to 1 1/2 times working pressure in the factory and tested to a working pressure of 15ksi.
Since the bending capacity of this bottom flange connection is limited by API 6A requirements the maximum bending capacity of this connection to preclude leaking is 14,000 ft-lbs therefore any side loads on this connection should be minimized with floatation.
Per Cameron, the bending capacity of the mini-collet connector hub is:

Bending Capacity of 3"-15K Mini-connector
at 0-ksi = 66,500 ft-lbs
at 10-ksi = 44,500 ft-lbs
at 15-ksi = 33,500 ft-lbs

Since a choke and burst disc assembly will be added to the top of the mini-connector mandrel with a weight of 2300 lbs and its center of gravity near the upright centerline, at 10ksi operating pressure the bending moment on connector hub and flange will be nil as compared to the maximum allowable bending moment capacity of the 3 1/6-15ksi flange.
- **Vetco Mandrel**

The Vetco Mandrel has been designed, manufactured and tested to meet the requirements for API 16A Specification for Drill-through Equipment Requirements. This mandrel has an API 18 ¾ 15ksi flanged bottom connection and a proprietary Vetco top connection complete with a proprietary VX ring gasket connection. This is a standard piece of Equipment on many BOP stacks throughout the industry and have been used for more than 30 years.

This mandrel was tested in the factory at 1 ½ times working pressure and has been tested to 15ksi working pressure on the stack.

4.0 **Bending Capacity on the BOP Outlets**

As per API Bulletin 6AF the maximum bending capacity at 10 ksi is 54,000 ft-lbs as is represented by the graph below. The center of gravity for the double blocks is approximately at the points of support on the valve support frame. The bolting on the side outlets of the BOP’s are ASTM A320 Gr L7 (105ksi yield strength) therefore the designation in the API 6AF tables specifying preload to 52.5ksi which is one half of yield strength which is typical for these connections. The valve assembly with spool and mandrel (33.56” to support) (2.8 ft) weighs approx. 6470 lbs. Considering the center of gravity at near the support the bending moment on the outlet of the BOP is 18,116 ft-lbs. The valve assembly with mandrel on the opposite side (23.07” to support) (2ft) weighs approx. 6070 lbs. Considering the center of gravity at near the support the maximum bending moment on the choke side outlet of the BOP is 12,140 ft-lbs.

If a choke valve and burst disc assembly is attached to the side outlet without the spacer spool with a weight of 2300 lbs at 37.94” then the new center of gravity for the block valve, mandrel and choke assembly is 27” (2.3 ft). The weight of the assembly becomes 8370 lbs and the bending moment on the choke outlet is 19,251 ft-lbs.

These bending moments are well within the capacity of the BOP flange outlets of 54,000 ft-lbs capacity at 10ksi. This is also within the capacity limitation at 15ksi which is approx. 27,000 ft-lbs.

All this is without considering the structural support of the valve support frame whose analysis is also attached. Any loads from addition of flowlines should be kept to within the structural limitation of the flange and minimized.
- BOP Valve Support Frame and Design Analysis

The Valve Support Frame is supported on the bonnets of the Single Ram BOP. It is then shimmed to zero clearance under the valve bodies to provide extra support for the valves to prevent unnecessary bending loads into the side outlets. The frame also attached to the lower bonnets of the upper double ram BOP with 2 each 1 1/2 " studs of ASTM A 193 Gr B7 (105 ksi yield strength) material which are screwed into the bonnets and supported to the beam to prevent cantilevering valves on one side to the other.
The valve frame support analysis actually considers from a design standpoint a load of 20,000 lbs per side with an anticipate maximum deflection of .022 inches from static load conditions.
Design Analysis for BOP Valve Support Frame  
Author: Josh Hilts – Transocean Design Engineer.

1. Introduction  
This report describes the structural analysis of the BOP valve support frame under loads of 20 kips per frame edge. The software used for the analysis is VersaFrame version 4, by Digital Canal Corporation.

The frame is constructed using W12x50 beams of material grade ASTM A992 (minimum yield strength 50 ksi). The frame is built in halves and is bolted together using 1” plates on each side of the connection, which is at the choke side and the kill side of the BOP. The frame is supported at the front and back sides along the length of the BOP bonnet (42-5/8”). The loads are applied at two points along each side of the frame, and are transferred from the valves to the frame by threaded studs on the front and back sides and by 2” vertical plates on the choke and kill sides.

2. Analysis Description  
The VersaFrame model uses simple supports at the center and the ends of the BOP bonnet. Two 10-kip loads are applied to each side of the frame, separated by 1 foot in each case. See Figure 1 below showing the loads and supports.

![Figure 1 - VersaFrame model loads and supports](image)

The loads are applied in two load combinations: static and dynamic. The static load combination applies all loads multiplied by a factor of 1.0 (10 kips applied at 8 locations). The dynamic load combination applies all loads multiplied by a factor of 1.5 (15 kips applied at 8 locations). For the dynamic analysis, allowable stresses are automatically increased by VersaFrame by a factor of 4/3 per the AISC Allowable Stress Design (ASD) manual.

3. Analysis Results
The maximum stress unity check ratio in the frame is 0.40, and is a vertical shear stress. The maximum allowable shear stress is 20 ksi (0.40 x 50 ksi), and the actual shear stress is 8 ksi. Therefore, the unity check ratio is 8 ksi / 20 ksi = 0.40.

The maximum combined stress ratio (axial + bending) in the frame is 0.18. The maximum allowable axial stress is 30 ksi (0.60 x 50 ksi), strong-axis bending stress is 33 ksi (0.66 x 50 ksi), and weak-axis bending stress is 37.5 ksi (0.75 x 50 ksi). Axial and weak-axis bending stresses are not present in the beam with the highest strong-axis bending stress, which is 7.96 ksi. Therefore, the unity check ratio is 7.96 ksi / (4/3 x 33 ksi) = 0.18.

See Figure 2 for a color-coded unity check view of the model, showing that the unity check values are less than 0.80 for all frame members.

Figure 2 – frame unity check

The maximum deflection under dynamic loads is 0.0334", and under static loads it is 0.0222", and occurs at the bolted connection for both cases. See Figure 3 for a deflection diagram, magnified by 250x.
Figure 3 – frame deflection

To check the bolts, half of the frame was removed from the model since the shear force approaches zero at the connection in the full model. See Figure 4. The maximum shear force at the connection is found to be 76.5 kips. The connections use (3) x 1.5" grade B7 bolts (minimum yield strength 105 ksi). Using these values and 42 ksi as the allowable stress (0.40 x 105 ksi), the unity check for bolt shear is 0.425, or a safety factor of 2.354.
Figure 4 – model of half of frame for bolt shear check

4. Conclusion
The frame is considered acceptable as designed for loads of 20 kips per side, applied as shown.
Function and Pressure Test Procedures for the Well Cap Assembly at Cameron Berwick Stack Pad

Criteria
Perform 15K test between test stump and lower ram, middle ram, upper ram, inner valves and outer valves.
In addition, apply 15K pressure to the outside of all valve gates.
Dual test to show 5 minutes @ 250 psi and 15 minutes @ 15,000 psi.
Times above to begin after pressure is stable.
5 minute line must be within a 15 minute period and 15 minute line within a 30 minute period.
Function all components through the ROV system to qualify same for correct hook up and functionality.

Failure Modes
There are to be no visible leaks of well bore fluid.
There are to be no visible leaks from operating circuits.
Pressure decline within operating chambers to be investigated and repaired if failures are present.
Pressure decline rate is not to exceed 20 psi over the 5 minute period for the 250 psi test.
Pressure decline rate is not to exceed 300 psi over the 15 minute period for the 15,000 psi test.
Per M.M.S. there is to be no pressure decline during the final 5 minutes of the charted test.

Set Up
Well bore test pressure entry through Autoclave valve at stump.
Certified Chart Recorder attached to well bore pressure source line.
ExF assembly located in close proximity to Well Cap stack.
5,000 psi supply from HPU to ExF assembly for Vetco connector tests and 3,000 psi supply for all other function tests.
Two (2) flying Leads to be used for all function test, one (1) with breech locks at both ends and the other with a breech lock at the panel end and a 17-D male probe at the function end.
HC Connector Latched as per OEM.
All rams Open and Vented.
Valve Gates in the relaxed and default position.
Vertical Mandrels fitted with Venting/Test hubs.
Upper blind flange removed.
17-H gauge probe installed with barrier valves Closed and vent valve Open.
Assembly filled with water with air purged or vented.

Procedures

( continued )
**E x F Connector (function test only)**
Test per procedure found in document “E x F Assembly & Panel OPD”

**HC Connector (function test only)**
Close the Surface ROV valve on the E x F panel.
Connect 3,000 psi supply to the surface supply breech lock on the E x F panel.
Connect the 17-D Flying Lead to the delivery breech lock on the E x F panel.
Install the 17-D into the Latch port on the HC panel.
Close the Flying Lead ROV vent valve and the Flying Lead supply valve on the E x F panel.
Open all four (4) paddle valves on the HC panel.
Open the Flying Lead ROV valve to apply Latch pressure to the connector.
Verify that stable pressure is present on the latch gauge.
Gauge Pressure = 3,100 psi.
Measure and record the position indicator stick up from upper body to top of indicator.
Position Indicator LATCHED = 3-3/8” inches.
Close the Latch paddle valve.
Close the Flying Lead supply valve.
Open the Flying Lead vent valve. (beware of venting fluid direction and force)
Remove 17-D probe.
Monitor Latch gauge pressure for 3 minutes.
Gauge pressure after 3 minutes = 2,950 psi.
Due to the area and volume related to the operating chamber it is common to see a small drop related to internal settling of dynamic components.
Open Latch ROV paddle valve.

Install the 17-D into the Primary UnLatch port on the HC panel.
Close the Flying Lead ROV vent valve.
Open the Flying Lead ROV valve to apply UnLatch pressure to the connector.
Verify that stable pressure is present on the unlatch gauge.
Gauge Pressure = 3,000 psi.
Measure and record the position indicator stick up from upper body to top of indicator.
Position Indicator UNLATCHED = 9-3/8” inches.
Close the Primary UnLatch paddle valve.
Close the Flying Lead supply valve.
Open the Flying Lead vent valve. (beware of venting fluid direction and force)
Remove 17-D probe.
Monitor UnLatch gauge pressure for 3 minutes.
Gauge pressure after 3 minutes = 2,850 psi.
Due to the area and volume related to the operating chamber it is common to see a small drop related to internal settling of dynamic components.
Open Primary UnLatch ROV paddle valve.

(continued)
Latch the connector.

Install the 17-D into the Secondary UnLatch port on the HC panel.
Close the Flying Lead ROV vent valve.
Open the Flying Lead ROV valve to apply UnLatch pressure to the connector.
Verify that stable pressure is present on the unlatch gauge.
Gauge Pressure = __3,000__ psi.
Measure and record the position indicator stick up from upper body to top of indicator.
Position Indicator UNLATCHED = __9-3/8"__ inches.
Close the Secondary UnLatch paddle valve.
Close the Flying Lead supply valve.
Open the Flying Lead vent valve. (beware of venting fluid direction and force)
Remove 17-D probe.
Monitor UnLatch gauge pressure for 3 minutes.
Gauge pressure after 3 minutes = __2,850__ psi.
Due to the area and volume related to the operating chamber it is common to see a small drop related to internal settling of dynamic components.
Open Secondary UnLatch ROV paddle valve.

AX Retainer (function test)
Install the 17-D probe into the AX port on the H.C panel.
Provide a regulated supply of 1,500 psi.
Close the AX paddle valve.
Close the Flying Lead supply valve.
Open the Flying Lead vent valve.
Monitor the trapped pressure which will be displayed on the Latch gauge.
Open AX paddle valve.

Latch
Latch the connector.
Close Flying Lead supply valve.
Operate Flying Lead vent valve to leave 1,000 psi to 1,500 psi on the Latch circuit.
Close the Latch paddle valve.
Open the Flying Lead vent valve.
Remove the 17-D probe.

Hydrate (function test)
Install 17-D probe into the Hydrate injection port.
Open the Flying Lead ROV valve to supply the hydrate flush system.
Witness fluid flow from the bottom of the connector.

Lower Ram (function and pressure test)
Connect the double breech lock Flying Lead to the delivery breech lock on the E x F panel.
Use the “Single Ram Panel OPD” for functional operation of the panel and ram.
(continued)
Ensure that P/T circuit Barrier valves on Valve panel are both closed.
Conduct a 250 psi with 5 minute hold followed by a 15,000 psi with 15 minute hold well bore test when ram is Closed and Vented.
Repeat test when ram is Closed and Isolated.
Pressure table when Close pressure is trapped.
3,000 psi @ 0 psi.
2,900 psi @ 280 psi.
2,200 psi @ 15,000 psi.
2,250 psi @ 10,000 psi.
2,550 psi @ 5,000 psi.
3,000 psi @ 0 psi.
Test Graphic

Middle Ram + Inner Gate Valves ( function and pressure test )
Connect the 17-D flying lead to the E x F panel.
Close the Blue Side Inner Gate valve and isolate with the ROV paddle valve.
Close the flying lead supply valve.
Vent the flying lead.
Close the Yellow Side Inner Gate valve and isolate with the ROV paddle valve.
Close the flying lead supply valve.
Vent the flying lead.
Remove the 17-D flying lead.
Connect the double breech lock Flying Lead to the delivery breech lock on the E x F ( continued )
panel.
Use the “Double Ram Panel OPD” for functional operation of the panel and ram.
Ensure that P/T circuit Barrier valves on Valve panel are both closed.
Conduct a 250 psi with 5 minute hold followed by a 15,000 psi with 15 minute hold well
bore test when ram is Closed and Vented.
Repeat test when ram is Closed and Isolated.
Pressure table when Close pressure is trapped.
3,150 psi @ 0 psi.
3,100 psi @ 280 psi.
2,850 psi @ 15,400 psi.
2,950 psi @ 10,000 psi.
2,980 psi @ 5,000 psi.
3,100 psi @ 0 psi.
Test Graphic

Upper Ram + Outer Gate Valves (function and pressure test)
Connect the 17-D flying lead to the E x F panel.
Close the Blue Side Outer Gate valve and isolate with the ROV paddle valve.
Close the flying lead supply valve.
Vent the flying lead.
Close the Yellow Side Outer Gate valve and isolate with the ROV paddle valve.
Close the flying lead supply valve.
Vent the flying lead.
(continued)
Remove the 17-D flying lead.
Connect the double breech lock Flying Lead to the delivery breech lock on the E x F panel.
Use the "Double Ram Panel OPD" for functional operation of the panel and ram.
Open the P/T circuit Barrier valves on Valve panel.
Ensure P/T circuit Vent valve is closed.
Conduct a 250 psi with 5 minute hold followed by a 15,000 psi with 15 minute hold well bore test when ram is Closed and Vented.
Repeat test when ram is Closed and Isolated.
Pressure table when Close pressure is trapped.
3,100 psi @ 0 psi.
3,050 psi @ 280 psi.
3,000 psi @ 15,000 psi.
3,000 psi @ 10,000 psi.
3,050 psi @ 5,000 psi.
3,050 psi @ 0 psi.
Monitor and record pressure at P/T gauge.
P/T gauge = __15,400__ psi.
Vent test pressure.
Close both P/T circuit barrier valves.
Open P/T circuit Vent valve.
Remove 17-H gauge probe.
Test Graphic

( continued )
Outer Gate Valves from Back Side (function and pressure test)
Install test manifold to both autoclaves valves on vertical mandrels.
Leave inner valve in the default Open position.
Close the outer Kill valve through the ROV panel.
Close the ROV paddle valve.
Vent at the E x F panel.
Close the outer Choke valve through the ROV panel.
Close the ROV paddle valve.
Vent at the E x F panel.
Conduct a 250 psi with 5 minute hold followed by a 15,000 psi with 15 minute hold pressure test when valves are Closed and Isolated.

Test Graphic

Inner Gate Valves from Back Side (function and pressure test)
Leave outer valve in the default Open position.
Close the inner Kill valve through the ROV panel.
Close the ROV paddle valve.
Vent at the E x F panel.
Close the inner Choke valve through the ROV panel.
Close the ROV paddle valve.
Vent at the E x F panel.
Conduct a 250 psi with 5 minute hold followed by a 15,000 psi with 15 minute hold pressure test when valves are Closed and Isolated.

(continued)
Procedure Approval

The above has been approved by Cameron, Berwick.

CAMERON __________________________________________

The above has been approved by Transocean, Houston.

TRANSOCEAN Rob Turlak

WITNESS DATE 6-17-10

Cameron __________________________________________

Transocean Robert Payne and Dean Williams

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be protected from fouling by hydrates or other contaminants. Any rate selected is not based on treatment of water; it is based on displacement of all seawater and continued protection from foreign material ingress. There are 24 gaps between the fingers of the collet. It was determined that a 0.5 gpm rate through each gap would be sufficient to maintain a contaminant-free mechanism until latched. Since there are 24 gaps the total requirement for the connector is a minimum of 12 gpm or 3 gpm per port.

**Triple Ram Stack**

During installation, the stack will be centered over the connector hub and stabbed. It will be subjected to (up to) full flow from the well at seabed conditions. If there is insufficient pressure drop through the assembly bore, significant volume of water may be entrained. If more than about 50% of the stream is seawater, mixture temperature will be reduced and hydrate formation conditions will be encountered within the stack. Accurate determination of the amount of entrained water during the installation is impossible, and overtreatment is not likely to be possible. Therefore the goal is to treat the flowing stream with as high a rate as practical during the time when the assembly is being positioned over the plume. Partial treatment will suppress hydrate formation conditions rather than eliminate risk. Once landed, the rate may be reduced to a nominal minimum until the drill string is detached. It’s important that the glycol inventory not be depleted prematurely. If the initial attempt to install the assembly has to be aborted, sufficient glycol must remain for further attempts.

Given this, the following schedule is recommended:

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<tr>
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<td>Remaining Inventory</td>
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The logic behind this schedule is to target roughly 1/3 of the available inventory. This should leave sufficient inventory to retry after making any adjustments, plus losses while idle.
GoM Drilling, Completions and Interventions

Hydrate Formation Assessment for Triple-Ram Capping Stack
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**Special Instructions:** MC252 Incident

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LNL004-026319
1 Introduction

A series of connection options to the flexjoint of the severed well of block MC252 have been developed for the containment and disposal of produced fluids. Among these options is the flanged connection to the flexjoint riser adapter (Figure 1).

![Figure 1 - G-Series Flange After Riser Removal](image)

These options require the removal of the upper flange including the cut riser joint pipe (Figure 1) and the subsea making of the flanged connection at the riser adapter flange with the Triple Ram stack (Figure 2).
Among the identified risks of these options is the formation of hydrates to such extent that could potentially prevent the successful installation. This document identifies these hydrate related risks and recommends mitigation.

The capping stack of the flexjoint developed for the “long term” containment and disposal of the produced fluids. “Long term” is defined as the time from installation of the stack to the final kill of the well with the relief wells that are currently drilled.
2 Component Description

2.1 Transition Spool Piece (Stage 1)

The transition spool (Figure 3) is designed with a 5M G-Series flange at the bottom to be connected to the flexjoint riser adapter G-Series flange. At the top of the spool piece there is 18¾” 15ksi flange connected to a HC connector mandrel. The HC connector enables the installation of the transition spool piece and the capping stack as a two step operation.

![Figure 3 - Transition Spool](image)

2.2 Triple Ram Stack (Stage 2)

The Triple-Ram stack consists of a lower ram, two opposing direction horizontal connections with two Cameron mini collet connector 3” hubs and two upper rams (Figure 2). Each horizontal “branch” is equipped with double isolation valves. The bottom of the Triple-Ram stack is connected to a 15ksi HC connector for the
engagement to the mandrel of the transition spool piece. The top of the Triple-Ram stack is attached to a H4 connector mandrel.

2.3 Perforated Riser

Based on the current working assumption the capping stack is set to be lowered for installation from a drillship attached to drill pipe. The Triple-Ram stack will be lowered into and connected to the HC mandrel of the transition spool piece with the bore open (all rams open) to vent the plume. A riser joint with a H4 connector attached at the bottom and a drill pipe adaptor spool piece at the top is plan to be used above the Triple-Ram stack during installation. In order to vent the plume flow through the open bore of the Triple-Ram stack, the riser joint is planned to be perforated. The length of the riser is about 75ft and will have 12 openings of 8” diameter each, 50” apart longitudinally and 60deg apart circumferentially (Figure 4).

![Figure 4 - Perforated Riser Joint with drill pipe adapter, 2" flexible pipe and H4 connector](image_url)
3 General Description of Installation Steps

The assumptions made regarding the installation procedure for the assessment of the hydrate risks are presented.

3.1 Transition Spool Piece Installation (Phase 1)

The transition spool piece and the capping stack are installed separately. The spool piece (as currently planned) will be lowered with wireline from the surface. The general steps of the spool piece installation can be seen in Figure 5.

![General Installation Sequence Steps - Spool Piece](image)

**Figure 5 - Installation Steps for Transition Spool Piece**

The spool piece is lowered at a safe distance from the plume (Lowering Step) to minimize the dropped objects risk. Once it has reach the desired height from the seal floor it is moved towards the riser adapter of the flex joint whose upper flange has already been removed and the drill pipes have been secured (Approaching Step). The spool piece is brought above the G flange of the riser adapter and the transition spool is entered over the two trapped drill pipes protruding from the lower flange (Stubbing step). The guidance pins are used for the correct orientation of the upper flange to the lower flange of the riser adapter. After the transition spool piece is landed the bolts are
fastened and the connection is completed. The slings are disconnected and the site is ready for the capping stack installation.

3.2 Capping Stack Installation (Phase 2)

The capping stack is planned to be installed after the spool piece is in place. The capping stack is lowered using drill pipe from a drilling ship. The general steps of the installation (for the triple ram or the lightweight ball valve) can be seen in Figure 6.

The capping stack is lowered at a safe distance from the plume (Lowering Step). At a second step the drill ship transports the capping stack right above the plume (Approaching Step). Due to the prior installation of the transition spool piece, the plume now flows from the top of the HC mandrel. Once in place, right above the connector mandrel, the capping stack is lowered and the HC connection is made. Operation of the rams from this point forward will be controlled via the detailed plans for well test and well kill operations.
4 Hydrate Formation Remediation Plan

4.1 Transition Spool Piece

The hydrate formation risks during installation of the transition spool piece are focused on possible interference of hydrates on making the subsea flanged connection to the riser adapter. Sensitive areas for hydrate formation are the G-Series flange sealing area (Figure 7), its flat bottom horizontal seating area, the threads of the flange bolts that will be suspended above their installed position as well as the bolt inserts on the flexjoint riser adapter flange.

![Figure 7 - Flange Sealing Surface (colored brown)](image)

At ambient seabed pressure and temperature (~2250 psia, 40°F) sea water and hydrocarbon vapor will form hydrates. The spool piece temperature will also be at 40°F. While approaching the plume the spool piece will be at risk of hydrate fouling from the hydrocarbon gas flowing from the well and mixing with cold sea water. Oil and gas are jetting out of the well at 200°F (above the hydrate formation temperature and pressure of 70°F and 2250 psia respectively). The hot flow is capable of warming up the spool piece lower components well above the hydrate formation temperature over time reducing the risk of hydrate formation. Nevertheless, the approach of the spool piece in the plume right above the flexjoint riser adapter must be made as quickly as possible to minimize the accumulation of hydrates on the flange and the other sensitive areas.

The results of a thermal CFD analysis (Figure 8 and Appendix A) conclude that the plume is warm enough to establish a 3 ft tall hydrate safe zone right above the flange of the riser adapter (Figure 11).
A six foot zone above the first three feet of the plume is recognized as the area that hydrates might form and the probability of hydrate formation increases above that.

Another step to minimize the amount of hydrates sticking at sensitive areas of the spool piece while approaching the plume is the application of grease over those areas. Grease does not help reducing the pressure-temperature envelop conditions of hydrate formation but lowers the ability of hydrate to stick on the steel structure. A study at the Colorado School of Mines on the identification of the most effective grease for hydrate disassociation from steel concluded that Grinsted Citrem SP70 and graphite are suitable candidates (Appendix B - Coatings to Reduce the Adhesion of Hydrate Particles inside the DD2 BOP Stack) among several candidates. Therefore Citrem grease has been procured and will be applied at the sensitive to hydrate areas of the spool piece.

Finally, an ROV mounted skid-pad/wand system will provide hydrate inhibitor locally to support the installation process. It will be used to flush hydrate deposits from the bolt inserts where fouling might interfere with making the flange.
4.2 Capping Stack

The capping stack is installed after installation of the transition spool piece. During the lowering step the same precautions as for the transition spool piece must be taken. The capping stack needs to be lowered at an area away from the plume.

During the approach the outer surface of the structure may accumulate hydrate deposits. Therefore, as with the transition spool, installation must be executed as quickly as practical in order to minimize exposure of the cold surface of the structure to the hydrocarbon gas plume and seawater mixture. The only external areas at risk to hydrate fouling are the ROV panels and for that purpose an ROV mounted methanol injection system is available during the capping stack installation. The ROV will spray hydrate inhibitor locally with a wand to the areas that need to be cleaned.

Once the capping stack has approached the plume and is right above the HC connector mandrel of the previously installed spool piece, the majority of the flow will be going through the capping stack and exiting from the perforated riser joint above it. There are two possible scenarios regarding the flow at the bottom of the capping stack around the HC connector. Either the pressure drop of the flow in the capping stack and perforated riser is low, enough to allow entrainment of water from the HC connector to the perforated riser, or, the pressure drop within the capping stack is such that all of the plume flow cannot go through the capping stack and therefore some of the plume escapes around the HC connector. The least desirable scenario for hydrate formation in the internals of the capping stack is the scenario of water entrainment.

In both scenarios the injection of hydrate inhibitor is necessary to reduce the risk of formation. The injection of glycol rather than methanol was decided based on the availability of the former on drill ship Enterprise as well as on the fact that glycol is safer to handle even though it is slightly less effective than methanol.

The glycol injection ports for the capping stack are the four ports of the Cameron HC connector and the main bore of the stack. The drill pipe used for lowering the capping stack will provide most of the glycol supply through its bore to a 2” flexible pipe (Figure 4) to one of the 3” mini hub Cameron collet connectors Figure 2. This will feed the bore of the capping stack with large quantities of glycol to treat the entrained water.

The HC connector will be injected with glycol provided by a flying lead from the IWOCs attached right above the perforated riser on the drilling string. The recommended volumes of glycol at the two injection points throughout the installation process are seen in Appendix C - Recommendations for Glycol Distribution during Installation of Capping Stack on MC-252.
5 Conclusions

The risk of hydrate fouling during installation of the capping stack is significant and requires preventative measures. Several measures will be taken to mitigate this risk. Rapid installation will reduce exposure to hydrate formation and deposition. In addition, strategic application of non-stick grease and glycol to the unit will prevent fouling of key areas and assure success.

The transition spool will be protected from hydrates using non-stick grease coating at sensitive areas around the bottom flange sealing areas. Because the transition is short, and the thermal mass of the oil plume is high, formation of hydrates on the shoe body is not expected. Further, any hydrate deposits that do form prior to entering the plume will be dissociated due to the high fluid temperature.

In order to protect the capping stack bore from fouling by hydrate deposits, provision for glycol injection into the bore in large quantities from the surface (installation vessel) has been included. The bottom of the capping stack at the HC connector will also have a glycol injection system in order to prevent fouling the connector should water entrainment occur during the alignment and linking of the connector to the capping stack mandrel.

Furthermore the installation will be supported with an ROV prepared to “power wash” accumulating deposits with hydrate inhibitor as needed in important areas prone to deposition such as the control panels or bolt inserts.
6 Appendix A

From: Lockett, Tim
Sent: Monday, May 17, 2010 5:21 PM
To: Hill, Trevor; Turnbull, Jon B; Wellings, James S
Cc: MC252_Email_Retention, Ravenscroft, Paul D
Subject: RE: Notes From Discussions with Exxon's Larry Talley, Hydrate Inhibition While Stabbing BOP on BOP

Trevor - as discussed earlier
Cc: Paul

John, James

Samir has completed a number of cases looking at the thermal aspects of a buoyant plume at flowrates 5 - 25 mbd coming off the top of the BOP stack at a temperature of 200 F, entraining sea-water at 40 F and cooling as a result of the mixing. The model is of the plume (in a steady state, 2-D axisymmetric) without looking at the way it interacts with any object placed within its path.

Please be aware that this is single-phase modelling of a condition which is multiphase. Nevertheless, I consider the results to be a helpful guide and I believe we can establish a workable target window for the operation.

I will summarise the results as follows:

1) Our expectation is that:

- equipment positioned within ~3 ft (vertically above) the top of the BOP is expected to be hot enough to be outside of the hydrate region

- equipment positioned 3 - 9 ft might be outside of the hydrate region depending on flowing conditions and entrainment

- at 9 - 15 ft there is the expectation of being inside the hydrate conditions

- more than 15 ft, should be inside the hydrate region.

2) There is quite a lot of uncertainty in the method by which we are representing the multiphase situation. In view of this a margin will be applied erring in the side of bringing the items which might be adversely impacted by hydrates closer to the BOP than the 1m range indicated above.

I understand that the closest positioning that is thought to be achievable (without endangering unexpected impact) is an offset of ~1 ft from the top of the BOP stack. I understand the aim is to bring the new item (valve or BOP) into position from the side of the jet and then hold for a time.
to allow it to warm up, funnelling the flow up through its middle, and injecting a hydrate inhibitor during this process to mitigate hydrate formation within the body.

These upper and lower bounds would seem to provide us with a workable target range of 1 ft to 3 ft for a hold time where the lower surfaces (where the sealing faces are located) can warm up to remove any hydrates which have deposited during the move-in from the side of the jet.

I trust this is helpful. Please can you advise if this is sufficient for planning this operation or if you would need to see further modelling (such as the BOP or valve placed in the jet at the hold position).

best regards
Tim
7 Appendix B - Coatings to Reduce the Adhesion of Hydrate Particles inside the DD2 BOP Stack

To: Trevor Hill
CC: Richard Chapman; Paul Ravenscroft
From: Simon Davies
Date: 26/05/10

In the case that the BOP stack from the DD2 is deployed to seal the leaking BOP stack from Deepwater Horizon, there is a risk of hydrate formation and plugging inside the DD2 BOP stack during deployment. If a hydrate plug were to form then it would make the mounting of the new BOP stack difficult.

To reduce the risk of forming a hydrate plug, it was decided to test whether grease-like coatings could reduce the tendency of hydrate particles to stick to the BOP stack. Experts from BP-Castrol in Pangbourne UK were consulted. They suggested eight readily available coatings that might reduce the adhesion of hydrate to steel. The suggested chemistries are listed below:

1. Oleamide solution.
2. Graphite from a pencil.
3. Grindsted Citrem SP 70
4. 1,9 nonane dithiol
5. Rain-X
6. Silicone oil
7. Castor Oil
8. GMO as an 80% solution

The efficacy of these coatings was then tested using the Micro Mechanical Force (MMF) apparatus at the Colorado School of Mines. The MMF apparatus works by
bringing a hydrate particle into contact with a steel surface with a known preload force, the particle is then pulled away from the steel and the force of adhesion is measured. The effectiveness of the coatings could be evaluated by performing similar tests on coated and uncoated steel surfaces and comparing the adhesion forces. The MMF experimental methods and detailed test results are provided in Attachment 1.

A summary of the results from the MMF apparatus are shown in Figure 1, the detailed results are provided in Attachment 1. The results indicated that Grindsted Citrem SP 70 was particularly effective at reducing the adhesion force between hydrate particles and steel by 97 - 98%.

**Figure 1:** Average Reduction in Adhesion Force from the Three Coatings (Error Bars Indicate the Range of the Reported Results)
Attachment 1: Detailed Experimental Results

As a next step, the durability of this coating will be tested at CSM by soaking the coated steel in salt water for 10 hours prior to the adhesion force test. CSM have also been asked to continue testing the remaining coatings that were suggested by BP-Castrol.

BP-Castrol advise that the Grindsted Citrem SP70 could be applied to the inside of the BOP using a wad or brush that is pulled through the BOP, sprayed on using an hvlp sprayer or perforated pipe, or by filling and turning the BOP stack.

Recommendations

Cameron who manufactured the BOP should be consulted on the compatibility of Grindsted Citrem SP 70 with the BOP stack. Cameron has already approved Castrol Biotac OG
8 Appendix C - Recommendations for Glycol Distribution during Installation of Capping Stack on MC-252

6/21/2010
Norman D. McMullen
Farah Saidi

Introduction
This document addresses glycol distribution in both the collet connector and triple ram stack components of the Capping Stack Assembly. Recommendations are made with the following assumptions:

- Glycol is delivered to the collet connector via an independent down line capable of at least 12 gpm.
- Glycol is delivered to the stack via 6-5/8” drill string and 3” hose (from the drill string to the stack).
- Total available glycol is 1000 barrels.
- Glycol to be used is 100% Monoethylene Glycol (MEG).
- Glycol is dyed with green (or any reasonable color) dye for visibility.
- Drill string is pre-filled with glycol.
- Maximum available pump delivery rate is 5 – 8 barrels per minute via the drill string.

Collet Connector
There are four injection ports on the connector. The purpose of these ports is to provide entry to the mechanism so that the moving parts and internal voids may be protected from fouling by hydrates or other contaminants. Any rate selected is not based on treatment of water: it is based on displacement of all seawater and continued protection from foreign material ingress.

There are 24 gaps between the fingers of the collet. It was determined that a 0.5 gpm rate through each gap would be sufficient to maintain a contaminant-free mechanism until latched. Since there are 24 gaps the total requirement for the connector is a minimum of 12 gpm or 3 gpm per port.

Triple Ram Stack
During installation, the stack will be centered over the connector hub and stabbed. It will be subjected to (up to) full flow from the well at seabed conditions. If there is
insufficient pressure drop through the assembly bore, significant volume of water may be entrained. If more than about 50% of the stream is seawater, mixture temperature will be reduced and hydrate formation conditions will be encountered within the stack.

Accurate determination of the amount of entrained water during the installation is impossible, and overtreatment is not likely to be possible. Therefore the goal is to treat the flowing stream with as high a rate as practical during the time when the assembly is being positioned over the plume. Partial treatment will suppress hydrate formation conditions rather than eliminate risk. Once landed, the rate may be reduced to a nominal minimum until the drill string is detached.

It’s important that the glycol inventory not be depleted prematurely. If the initial attempt to install the assembly has to be aborted, sufficient glycol must remain for further attempts.

Given this, the following schedule is recommended:

<table>
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<th>Glycol, gpm</th>
<th>Glycol Consumed</th>
<th>Remarks</th>
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<tr>
<td></td>
<td>min</td>
<td>Collet</td>
<td>Bore</td>
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<tr>
<td>120</td>
<td>60</td>
<td>12</td>
<td>12</td>
</tr>
<tr>
<td>60</td>
<td>120</td>
<td>0</td>
<td>24</td>
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<td></td>
<td>120</td>
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<td>24</td>
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<tr>
<td>Total</td>
<td></td>
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<tr>
<td>Remaining Inventory</td>
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</table>

The logic behind this schedule is to target roughly 1/3 of the available inventory. This should leave sufficient inventory to retry after making any adjustments, plus losses while idle.
GoM Development PU

MC 252 Recovery Plan

Transition Spool
Design Report
408024-00019-SE-REP-0001-0001
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1 Introduction

This design report documents the design of transition spool of the MC 252 recovery plan.

2 Scope

This report summarizes the results from preliminary analyses performed to design two spools to assist in the recovery effort. The results from axi-symmetric FE analyses performed for the 18 ¾-in 15K API flange and G-flange are also included in the report.

The following two spool options are as follows:

Option 1: Spool between bottom of Dual-Ram BOP and top of flexjoint. The spool will have an 18 ¾-in 15K flange at the BOP end and a “G-flange” at the flexjoint end.

Option 2: Spool between bottom of Dual-Ram BOP and top of LMRP (flexjoint has been removed). The spool will have an API 18 ¾-in 15K flange at both ends.

Preliminary sketches of the two spool options are included in the Appendix B of this report.

3 Assumptions

The assumptions made for design of transition spools are listed below:

- The weight of the dual-ram BOP and associated adaptors and running tools spools for installation to be installed above the transition spool weigh less than 150 kips.
- The maximum rotation of the BOP from the vertical is 10 degrees.
- The center of gravity of the BOP is together with any adaptors and equipment that may cantilever above the lower flange of the spool at 10 degrees will result in a moment of no more than 730 kip.feet at the lower flange face (equivalent to 150 kips at 34 ft COG, or 100 kips at 51 ft COG etc.). This will be compared against the moment from the selected stacked running and pressure isolation equipment once selected.
- The maximum tensile longitudinal loading in the system will not exceed the end cap force, and the equivalent loading in the bolts from end cap load and moment in the most critical flange will not exceed a direct tension load of 3000 kips on the flange.
- Maximum internal pressure is 8900 psi.
- Applicable water depth is 4992 ft.

4 Summary and Recommendation

Results from the preliminary analyses show that the design of the transition spools are acceptable for the anticipated loads during the recovery process.
5 Design Methodology

The design parameters used for design of transition spools are listed in Table 1.

<table>
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<tr>
<th>Parameters</th>
<th>Unit</th>
<th>Value</th>
<th>Reference</th>
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<tbody>
<tr>
<td>Outer Diameter of spool pipe</td>
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</tr>
<tr>
<td>Wall Thickness of spool pipe</td>
<td>in</td>
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</tr>
<tr>
<td>Water Depth</td>
<td>ft</td>
<td>4992</td>
<td></td>
</tr>
<tr>
<td>Maximum Internal Pressure</td>
<td>psi</td>
<td>8900</td>
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<tr>
<td>Steel Density</td>
<td>lb/ft³</td>
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<td></td>
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<tr>
<td>Pipe/Flange Material Yield Strength</td>
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<td>Ambient Reference Temperature</td>
<td>°F</td>
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<tr>
<td>Thermal Expansion Coefficient</td>
<td>in/°F</td>
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5.1 Spool Pipe Design in MathCAD

The design of the transition spool pipe was performed in accordance with formulae of ASME B31.8 [Ref. 1] thin and thick wall theories and API RP-1111 [Ref. 2]. The stresses adopted for checking the spool pipe design to ASME B31.8 equations are listed in Table 2 (note these are proposed for this project only and exceed normal code values). The MathCAD calculations are included in Appendix A.

<table>
<thead>
<tr>
<th>Stress Description</th>
<th>% of SMYS</th>
<th>Value (ksi)</th>
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<tbody>
<tr>
<td>Hoop Stress</td>
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<tr>
<td>Combined Stress</td>
<td>95%</td>
<td>76.0</td>
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5.2 Flange Welded Connection Axi-Symmetric Analyses in ANSYS

The welded connection between the spool pipe and the API flange and G-flange were analyzed in ANSYS for maximum anticipated tensile loads to confirm that any localized stresses in the flange do not exceed the design stresses recommended in API RP-2RD. Reference drawings of the flanges are included in Appendix B.

5.2.1 Allowable Stresses

As per the API RP-2RD Section 5.2 [Ref. 3], the stresses in the flange are governed by the following criteria:

a) \( \sigma_p \leq C_a C_f \sigma_y \)

b) \( (\sigma_p + \sigma_r) \leq 1.5 C_a C_f \sigma_y \)

c) \( (\sigma_p + \sigma_r + \sigma_\theta) \leq 3.0 C_a C_f \sigma_y \)

where,
\( \sigma_p \) = Primary membrane stress
\( \sigma_b \) = Primary bending stress
\( \sigma_q \) = Secondary normal or shear stress
\( \sigma_y \) = Flange material yield stress (80 ksi)
Ca = 2/3, Allowable stress factor for steel
\( C_f = 1.2 \), Design case factor (Table 2, Section 4.4 for temporary load and environmental condition)

<table>
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<tr>
<th>Stress Description</th>
<th>% of SMYS</th>
<th>Value (ksi)</th>
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<td>Primary Membrane Stress</td>
<td>80%</td>
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</tr>
<tr>
<td>Primary Membrane + Primary Bending Stress</td>
<td>120%</td>
<td>96.0</td>
</tr>
<tr>
<td>(Primary Membrane + Primary Bending Stress) + Secondary Stress</td>
<td>240%</td>
<td>192.0</td>
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</table>

5.2.2 Modeling in ANSYS

Axi-symmetric models of the flanges were developed in ANSYS to simplify the analyses and reduce the computational time. Figure 1 shows the axi-symmetric model of the API 18 3/4-in 15K flange. The model includes the API flange with part of spool pipe, a mirrored API flange with pipe, nuts and a bolt connecting the two flanges. This model ensures that the correct load path is modeled for the flanges.

![Axi-symmetric Model of API 18 3/4-in 15K Flange](image)

Figure 1: Axi-symmetric Model of API 18 3/4-in 15K Flange (Figure rotated to show entire model)

The model was meshed using axi-symmetric PLANER2 elements. The nuts were modeled to be integral with the flange since the bolt-preload ensures that the nuts are always in contact with the flange face. The bolt was modeled using BEAM3 element and a pre-load of 5509 kips was applied to the model using the initial strain option for the BEAM3 elements. The contact between the flanges was modeled using CONTA169 and TARGET172 elements.

The mesh was refined in the area of interest (transition areas adjacent to the welded connection between the pipe and the flange). Figure 2 shows the transition area in detail. The transition between the ID of the flange to the ID of the pipe was provided a slope of 4:1.
The maximum axial and bending stress in the pipe is 55.2 ksi. A uniform stress of 55 ksi was applied to the pipe section of the ANSYS model. Although this does not represent the true global behavior of the flange, it is an accurate representation of the cross-section of flange that sees the maximum combined axial and bending stress. A uniform internal pressure of 6.8 ksi was also applied to the inside of the flange and pipe. The axi-symetric model was restrained at the lower end of the mirrored flange.

![Figure 2: Close Up View of Transition Area](image)

Figure 2: Close Up View of Transition Area

Figure 3 shows the axi-symmetric model of the G-flange. The long transition area of the flange was modeled as a spline curve and was finely meshed for the analysis. The model was restrained at the lower end of the mirrored flange.

![Figure 3: Axi-symmetric Model of G-Flange (Figure rotated to show entire model)](image)

Figure 3: Axi-symmetric Model of G-Flange (Figure rotated to show entire model)
6 Results

6.1 Spool Pipe Design in MathCAD

The results from the MathCAD calculations for the spool pipe show that the spool will be operating at approximately 0.9\textsuperscript{th} SMYS, i.e. a hoop stress level that is normally used as the ASME B31.8 minimum pressure test limit (1.25 x the 0.72\textsuperscript{nd} SMYS hoop stress design limit), but significantly above the normal ASME B31.8 hoop stress limit 0.72\textsuperscript{nd} SMYS. However, this is mitigated slightly by tested material strength exceeding SMYS (84 to 95 ksi depending on heat treatment condition compared to 80 ksi).

Although the spool exceeds normal operational hoop stress, the combined stresses with arbitrary allowance for bending (up to 150 kip\textsuperscript{2} 28 ft\textsuperscript{2} sin 10\degree) do not exceed ASME B31.8 normal operational combined stress limit 0.9\textsuperscript{th} SMYS when using thick wall formulae.

Alternative checks per API 1111 (limit state design) show that the spool will be operating with a differential pressure above the normal (burst i.e. hoop) design stress. In fact it will be above normal incidental over-pressure limit, actually sitting just below (minimum) recommended test pressure, again mitigated by the tested material strength exceeding SMYS. The combined load design parameter is 0.9 (equal to the API 1111 requirement for normal operation) except if bending is treated as effective tension, in which case it just meets the requirement for “extreme loads” or hydrotest (0.96).

6.2 Flange Welded Connection Axi-Symmetric Analyses in ANSYS

6.2.1 18 \%\% in 15K API Flange

The equivalent stress in the flange in the area of interest is shown in Figure 4. The maximum combined stress in the flange is 106.1 ksi.
**Figure 4: Equivalent Stress in API Flange**

The equivalent stress was linearized at the location of the maximum stress. Figure 5 shows the stress linearization plot. The membrane stress in the flange is less than code allowable stress of 64 ksi. The combined membrane and bending stress is less than 96 ksi and the total stress in the flange is less than 192 ksi.
6.2.2 G-Flange

Results from the FE analysis of the G-flange indicate that its unique shape prevents development of hot-spot stresses along the length of the flange. The maximum equivalent stress in the flange is 76.7 ksi, as shown in figure 6.

The stress linearization of the G-flange stresses is shown in figure 7. The maximum membrane stress in the G-flange is approximately 71 ksi, which exceeds the code recommended value of 64 ksi. The combined membrane and bending stress in the flange is approximately 73.8 ksi, which is less than code recommended value of 96 ksi. The total stress in flange is also less than code recommended value of 192 ksi.
Figure 6: Equivalent Stress in G-Flange
6.3 Internal Guide Design in MathCAD and StruCAD 3D

6.3.1 MathCAD

The results from the MathCAD calculations for the internal guide show that at the point of highest bending moment on the full 18-inch diameter tubular cross section the stresses slightly exceed AIS/C API RP 2A allowable limits for a vertical load of 70 kips and lateral load of 7 kips in worst case combination (giving a combined unity check of 1.04 considering a 1/3 overstress for an accidental impact dynamic load condition). A bearing check for the “nose” of the tubular carrying all vertical load over a 9-inch arc length has a unity check of less than 0.95.

However, since the tubular has had strips cut out to allow fluid flow during deployment, the strength of the guide will be limited by the buckling capacity of the remaining strips of shell between the slots. The most highly stressed strip, just above the “nose” is 15-inches long, and is estimated to buckle for loads exceeding approximately 20 kips. This value is carried from the MathCAD calculation into the StruCAD analyses described in the next section.
6.3.2 StruCAD 3D

The results from the StruCAD 3D analysis for the internal guide consider the critical strip of shell above the nose to be on the point of buckling, with the strip ‘broken’, but with 20 kip loads acting downward on the adjacent lower surface and upward on the adjacent upper surface.

In this condition the nose of the guide experiences combined stresses of approximately 43 ksi under a vertical load of 70 kips acting without any associated lateral impact or frictional load (Load Case 1). Such stresses are above the minimum yield stress (35 ksi), but below the minimum tensile stress (60 ksi), so would be expected to show some permanent deformation, approximately the limit that may prevent the guide functioning.

Stress levels in the same location are approximately 40 ksi under a horizontal impact load of 7 kips acting without any associated vertical load (Load Case 2). Again, such stresses may lead to some permanent deformation, and again represent the limit that may prevent the guide functioning.

Stress levels in the same location are approximately 76 ksi under a vertical load of 70 kips acting in combination with an outward horizontal frictional or impact load of 7 kips (Load Case 3). Such a load combination would lead to significant permanent deformation, and exceeds limit for the guide functioning.

Stress levels in the same location are approximately 36 ksi under a vertical load of 70 kips acting in combination with an inward horizontal frictional or impact load of 7 kips (Load Case 4). Such a load combination would be below the limit for deformations that may prevent the guide functioning.

For the less likely combination of vertical and outward lateral loading on the nose the guide should be kept within the limits of the approximate formula:

\[
\text{Vertical Load} + \text{Outward Lateral Load} < 1
\]

70 kip

For the more likely combination of vertical and inward lateral loading on the nose the guide can accept 70 kip vertical and 7 kip lateral in combination.
Figure 8: Load Case 1 (vertical load acting alone)
Figure 9: Load Case 2 (lateral load acting alone)
Figure 10: Load Case 3 (vertical load acting with outward lateral)
7 References

3. American Petroleum Institute; “Design of Risers for Floating Production Systems (FPSs) and Tension Leg Platforms (TLPs)”, API RP-2RD.
Appendix A: Spool Pipe Code Check in MathCAD

(Documents not embedded in this report due to file size; included in PDF rendition)
BP MC 252 - Transition Spool

**INPUT VALUES**

<table>
<thead>
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<th>Symbol</th>
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<tr>
<td>t</td>
<td>1-in</td>
</tr>
<tr>
<td>E</td>
<td>29-10^6 psi</td>
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<tr>
<td>WD</td>
<td>4992-ft</td>
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<tr>
<td>t_c</td>
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<td>90-ksi</td>
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<tr>
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<tr>
<td>F_{d_{min}}</td>
<td>1700-kip</td>
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<tr>
<td>M_{max}</td>
<td>730-kip-ft</td>
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**Hand-Calc. Check Of 21" Pipe In Transition Spool Piece:**

**Case 1a**
- Max. Internal Pressure
- 1852 kip Max. Tension including end cap tension of 1852 kip
- 730 kip*ft moment, which for stack inclination of 10° allows up to 150 kips at up to 28 ft

**Case 1b**
- Max. Internal Pressure
- 1,700 kip Min. Tension including end cap tension thus 150 kip equivalent compression
- 730 kip*ft moment

**Case 2**
- Find Hydrotest Limit

Treat above as combined stress limit. Additionally, plan to keep hoop stress within the minimum for ASME B31.8 pipeline test, i.e. 1.25 x 0.72 = 0.9*SMYS.
Pipe Properties

\[ D_1 := D - 2t \]  \quad \text{Internal Diameter}  \quad D_1 = 19.5\text{-in}

\[ A_i := \frac{\pi}{4} (D_i)^2 \]  \quad \text{Internal cross-section Area of Pipe}  \quad A_i = 298.6\text{-in}^2

\[ A_o := \frac{\pi}{4} (D)^2 \]  \quad \text{External cross-section Area of Pipe}  \quad A_o = 298.6\text{-in}^2

\[ A_s := A_o - A_i \]  \quad \text{Steel Area of Pipe}  \quad A = 64.4\text{-in}^2

\[ I := \frac{\pi}{64} \left[ D^4 - (D_i)^4 \right] \]  \quad \text{Moment of Inertia of Pipe}  \quad I = 3391.2\text{-in}^4

\[ S_p := \frac{1}{\left( \frac{D}{2} \right)} \]  \quad \text{Elastic Modulus}  \quad S_p = 315.5\text{-in}^3

\[ a := \frac{D}{2} \]  \quad a = 10.75\text{-in}  \quad \text{Outer Radius}

\[ b := a - t \]  \quad b = 9.75\text{-in}  \quad \text{Inner Radius}
Load Effects

\[ P_e := W D - \gamma_{sw} \]  
External Pressure \[ P_e = 2219 \text{-psi} \]

\[ P_i - P_e = 6681 \text{-psi} \]  
Design Pressure Differential (thin wall calculations)

End Cap Force:

\[ \lambda_e := \frac{\pi}{4} \left( D^2 \right) \]  
External cap area \[ \lambda_e = 363.1 \text{-in}^2 \]

\[ \lambda_i := \frac{\pi}{4} \left( D_i^2 \right) \]  
Internal cap area \[ \lambda_i = 298.6 \text{-in}^2 \]

\[ F_p := \lambda_i \cdot P_i - \lambda_e \cdot P_e \]  
End cap force \[ F_p = 1852 \text{-kip} \]

Additional Bending Stress Due to up to 150 kips with COG up to 22 ft above 6 ft spool at 10 deg

\[ \sigma_b(r) := \frac{M_{max} \cdot r}{I} \]  
\[ \sigma_b(a) = 27.8 \text{-ksi} \] At outer pipe radius

\[ \sigma_b(b) = 25.2 \text{-ksi} \] At Inner pipe radius

Check "G" Flange capacity

\[ n := 6 \] Number of Bolts
\[ pcd := 37.5 \text{-in} \] Pitch Circle Diameter

\[ T_{\text{moment}} := \frac{4 \cdot M_{\text{max}}}{n \cdot pcd} \] Bolt tension due to applied moment \[ T_{\text{moment}} = 156 \text{-kip} \]

\[ T_{\text{endcap}} := \frac{F_p}{n} \] Bolt tension due to end cap force \[ T_{\text{endcap}} = 309 \text{-kip} \]

\[ T_{\text{total}} := T_{\text{moment}} + T_{\text{endcap}} \] Maximum bolt tension \[ T_{\text{total}} = 464 \text{-kip} \]

By inspection, this is below the design limit of the "G" flange with 3000 kips on a 6-bolt flange.
CASE Ia

Axial Stress Due to Vertical Load

\[ \sigma_a = \frac{F_{\text{dmax}}}{A} \]

\[ \sigma_a = 28.8\text{-ksi} \]

\[ \sigma_a + \sigma_h(a) = 56.5\text{-ksi} \]

Maximum tensile stress in pipe wall

\[ \sigma_a + \frac{(\sigma_h(a) + \sigma_h(b))}{2} = 55.2\text{-ksi} \]

Mid wall tensile stress in pipe wall

Thin Wall Pipe Hoop Stresses Due to Pressure

\[ \sigma_{h\text{-tw}} = \frac{(P_i - P_e)D}{2t} \]

Allowable:

\[ 0.9-F_y = 72.0\text{-ksi} \]

Thick Wall Hoop Stress (Internal Pressure + External Pressure)

\[ \sigma_h(r) = \frac{P_i \cdot b^2 \left( \frac{a^2}{r^2} + \frac{r^2}{a^2} \right)}{r^2 \left( \frac{a^2}{a^2 - b^2} \right)} + \frac{-P_e \cdot a^2 \left( \frac{b^2}{r^2} + \frac{r^2}{b^2} \right)}{r^2 \left( \frac{a^2}{a^2 - b^2} \right)} \]

\[ \sigma_h(a) = 59.7\text{-ksi} \]

\[ \sigma_h(b) = 66.4\text{-ksi} \]

Thick Wall Radial Stress (Internal Pressure + External Pressure)

\[ \sigma_r(r) = \frac{-P_i \cdot b^2 \left( \frac{a^2}{r^2} - \frac{r^2}{a^2} \right)}{r^2 \left( \frac{a^2}{a^2 - b^2} \right)} + \frac{-P_e \cdot a^2 \left( \frac{r^2}{b^2} - \frac{b^2}{r^2} \right)}{r^2 \left( \frac{a^2}{a^2 - b^2} \right)} \]

\[ \sigma_r(a) = -2.2\text{-ksi} \]

\[ \sigma_r(b) = -8.9\text{-ksi} \]

Thin Wall Combined (Von Mises) Stress: Simple Hoop & Axial

\[ \sigma_{vm\text{-tw}} = \sqrt{(\sigma_a + \sigma_h(a))^2 + \sigma_{h\text{-tw}}^2 - (\sigma_a + \sigma_h(a)) \cdot \sigma_{h\text{-tw}}} \]

Allowable:

\[ 0.95-F_y = 76.0\text{-ksi} \]

\[ \sigma_{vm\text{-tw}} = 65.3\text{-ksi} \]

\[ 0.95-F_y = 76.0\text{-ksi} \]
At inside wall (b)

Combined (Von Mises) Stress: Simple Hoop & Axial

\[ \sigma_{\text{vm}} := \sqrt{(\sigma_a + \sigma_b(b))^2 + \sigma_h(b)^2 - (\sigma_a + \sigma_b(b))\cdot \sigma_h(b)} \]

\[ \sigma_{\text{vm}} = 61.1 \text{ksi} \quad \text{Allowable: } 0.95 \cdot F_y = 76.0 \text{ksi} \]

Combined (Von Mises) Stress: Simple Axial & Shear

\[ \sigma_{\text{vm2}} := \frac{1}{2} \left[ \left( \sigma_a + \sigma_b(b) \right) - |\sigma_f(b)| \right]^2 + \left( \sigma_a + \sigma_b(b) \right)^2 + \sigma_f(b)^2 \]

\[ \sigma_{\text{vm2}} = 50.1 \text{ksi} \]

Combined (Von Mises) Stress: MAX

\[ \sigma_1 := \sigma_h(b) \quad \sigma_2 := (\sigma_a + \sigma_b(b)) \quad \sigma_3 := \sigma_f(b) \]

\[ \sigma_{\text{vm3}} := \sqrt{\frac{1}{2} \left( (\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 \right)} \]

\[ \sigma_{\text{vm3}} = 68.9 \text{ksi} \quad \text{Allowable: } 0.95 \cdot F_y = 76.0 \text{ksi} \]

At outside wall (a)

Combined (Von Mises) Stress: Simple Hoop & Axial

\[ \sigma_{\text{vm}} := \sqrt{(\sigma_a + \sigma_b(a))^2 + \sigma_h(a)^2 - (\sigma_a + \sigma_b(a))\cdot \sigma_h(a)} \]

\[ \sigma_{\text{vm}} = 58.2 \text{ksi} \quad \text{Allowable: } 0.95 \cdot F_y = 76.0 \text{ksi} \]

Combined (Von Mises) Stress: Simple Axial & Shear

\[ \sigma_{\text{vm2}} := \frac{1}{2} \left[ \left( \sigma_a + \sigma_b(a) \right) - |\sigma_f(a)| \right]^2 + \left( \sigma_a + \sigma_b(a) \right)^2 + \sigma_f(a)^2 \]

\[ \sigma_{\text{vm2}} = 55.4 \text{ksi} \]

Combined (Von Mises) Stress: MAX

\[ \sigma_1 := \sigma_h(a) \quad \sigma_2 := (\sigma_a + \sigma_b(a)) \quad \sigma_3 := \sigma_f(a) \]

\[ \sigma_{\text{vm3}} := \sqrt{\frac{1}{2} \left( (\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 \right)} \]

\[ \sigma_{\text{vm3}} = 60.4 \text{ksi} \quad \text{Allowable: } 0.95 \cdot F_y = 76.0 \text{ksi} \]
**CASE 1b**

Axial Stress Due to Vertical Load

\[
\sigma_a = \frac{F_{\text{dmin}}}{A}
\]

Thin Wall Combined (Von Mises) Stress: Simple Hoop & Axial

\[
\sigma_{\text{vm, tw}} = \sqrt{\left(\sigma_a - \sigma_b(a)\right)^2 + \sigma_{h, tw}^2 - \left(\sigma_a - \sigma_b(a)\right)\sigma_{h, tw}}
\]

Allowable:

\[0.95 \cdot F_y = 76.0 \text{ ksi}\]

At inside wall (b)

Combined (Von Mises) Stress: Simple Hoop & Axial

\[
\sigma_{\text{vm}} = \sqrt{\left(\sigma_a - \sigma_b(b)\right)^2 + \sigma_h(b)^2 - \left(\sigma_a - \sigma_b(b)\right)\sigma_h(b)}
\]

\[\sigma_{\text{vm}} = 65.8 \text{ ksi}\]

Combined (Von Mises) Stress: Simple Axial & Shear

\[
\sigma_{\text{vm2}} = \frac{1}{2} \left[ \left(\sigma_a - \sigma_b(b) - \sigma_f(b)\right)^2 + \left(\sigma_a - \sigma_b(b) + \sigma_f(b)\right)^2 \right]
\]

\[\sigma_{\text{vm2}} = 8.3607 \text{ ksi}\]

Combined (Von Mises) Stress: MAX

\[\sigma_{\text{vm3}} = \sigma_h(b) \quad \sigma_{\text{vm3}} = \sigma_a - \sigma_b(b) \quad \sigma_{\text{vm3}} = \sigma_f(b)\]

Allowable:

\[0.95 \cdot F_y = 76.0 \text{ ksi}\]

At outside wall (a)

Combined (Von Mises) Stress: Simple Hoop & Axial

\[
\sigma_{\text{vm}} = \sqrt{\left(\sigma_a - \sigma_b(a)\right)^2 + \sigma_h(a)^2 - \left(\sigma_a - \sigma_b(a)\right)\sigma_h(a)}
\]

\[\sigma_{\text{vm}} = 60.4 \text{ ksi}\]

Combined (Von Mises) Stress: Simple Axial & Shear

\[
\sigma_{\text{vm2}} = \frac{1}{2} \left[ \left(\sigma_a - \sigma_b(a) - \sigma_f(a)\right)^2 + \left(\sigma_a - \sigma_b(a) + \sigma_f(a)\right)^2 \right]
\]

\[\sigma_{\text{vm2}} = 3.1 \text{ ksi}\]

Combined (Von Mises) Stress: MAX

\[\sigma_{\text{vm3}} = \sigma_h(a) \quad \sigma_{\text{vm3}} = \sigma_a - \sigma_b(a) \quad \sigma_{\text{vm3}} = \sigma_f(a)\]

\[\sigma_{\text{vm3}} = 61.5 \text{ ksi}\]
CASE 2

Find onshore hydrotest pressure at 30 CFR 250 recommended upper limit of 0.95 x SMYS; i.e., above B31.8 minimum requirement of 1.25 x the pressure causing stress = 0.72 SMYS, equivalent 0.9 x SMYS minimum test.

Thin Wall Pipe Hoop Stresses Due to Pressure

\[ P_{\text{test}} = \frac{0.95 \cdot F_y \cdot 2 \cdot t}{D} \quad P_{\text{test}} = 7070 \text{-psi} \]

Thick Wall Hoop Stress (Internal Pressure \( \rightarrow \) External Pressure)

\[ P_{\text{stress}} = \frac{0.95 \cdot F_y \left[ b^2 \left( a^2 - b^2 \right) \right]}{b^2 \left( a^2 + b^2 \right)} \quad P_{\text{test}} = 7397 \text{-psi} \]
RE-CHECK TO API 1111: CASE 1a

\[ S := F_y \quad \text{SMYS (Specified Minimum Yield Strength)} \]
\[ U := F_t \quad \text{SMTS (Specified Minimum Tensile Strength)} \]
\[ f_d := 0.9 \quad \text{Burst pressure design factor} \]
\[ f_c := 1 \quad \text{Weld joint factor (only materials with factor 1.0 are acceptable)} \]
\[ f_t := 1 \quad \text{Temperature derating factor = 1.0 for temperature < 121 °C (250 °F)} \]

\[ P_b := 0.45(S + U) \ln \left( \frac{D}{D_t} \right) \quad \text{Burst pressure per API 1111 (2a)} \quad P_b = 7.469 \text{-ksi} \]
\[ P_t := f_d f_c f_t P_b \quad \text{Test pressure per API 1111 (1a)} \quad P_t = 6.722 \text{-ksi} \]
\[ P_d := 0.8 P_t \quad \text{Design pressure per API 1111 (1b)} \quad P_d = 5.378 \text{-ksi} \]
\[ P_a := 0.9 P_t \quad \text{Incidental overpressure per API 1111 (1c)} \quad P_a = 6.050 \text{-ksi} \]
\[ P_o := P_e \quad \text{API 1111 terminology for external pressure} \]
\[ P_t - P_o = 6.681 \text{-ksi} \quad \text{Differential Pressure on Spool} \quad \frac{(P_t - P_o)}{P_t} = 0.994 \]

**Spool will be operating above Incidental Overpressure limit, at 99.4% of API 1111 Test Limit**

\[ T_y := S \cdot A \quad \text{Yield tension of the pipe} \quad T_y = 5152 \text{-kip} \]
\[ \sigma_{max} := \frac{F_{d_{max}}}{A} \quad \text{Axial stress in pipe wall} \quad \sigma_{a} = 28.8 \text{-ksi} \]
\[ T_{eff} := \frac{\sigma_a \cdot A - P_t \cdot A_t + P_o \cdot A_o}{P_b} \quad \text{Effective tension in pipe} \quad T_{eff} = -0 \text{-kip} \]

Allowable effective tension in pipe API 1111 (3): \(0.6T_y = 3091 \text{-kip}\)

\[ \sqrt{\left( \frac{(P_t - P_o)}{P_b} \right)^2 + \left( \frac{T_{eff}}{T_y} \right)^2} = 0.895 \quad \text{API 1111 (4) combined load check} \]

0.9 For operational loads.
0.96 For extreme loads.
0.96 For hydrotest loads.

**Spool will be operating above operational load limit but below extreme/ hydrotest limit.**
RE-CHECK TO API 1111: CASE 1b

\[
\sigma_a = \frac{F_{\text{min}}}{A} \quad \text{Axial stress in pipe wall} \quad \sigma_a = 26.4\text{-ksi}
\]

\[
T_{\text{eff}} = \sigma_a \cdot A - P_i \cdot A_i + P_o \cdot A_o \quad \text{Effective tension in pipe} \quad T_{\text{eff}} = -152\text{-kip}
\]

\[
\sqrt{\left(\frac{P_i - P_o}{P_h}\right)^2 + \left(\frac{T_{\text{eff}}}{T_y}\right)^2} = 0.895 \quad \text{API 1111 (4) combined load check}
\]

---

Spool will be operating at 99.5% of API 1111 operational load limit

---

CHECK TO API 1111: As 1a but add bending to axial; not a formal API 1111 check

\[
\sigma_b = \frac{M_{\text{max}} r}{I} \quad \sigma_b = 27.8\text{-ksi} \quad \text{At outer pipe radius}
\]

\[
\sigma_a = \frac{F_{\text{max}}}{A} + \sigma_b \quad \text{Axial stress in pipe wall} \quad \sigma_a = 56.5\text{-ksi}
\]

\[
T_{\text{max}} = \sigma_a \cdot A - P_i \cdot A_i + P_o \cdot A_o \quad \text{Effective tension in pipe} \quad T_{\text{eff}} = 1788\text{-kip}
\]

Allowable effective tension in pipe API 1111 (3): \(0.6 T_y = 3091\text{-kip}\)

\[
\sqrt{\left(\frac{P_i - P_o}{P_h}\right)^2 + \left(\frac{T_{\text{eff}}}{T_y}\right)^2} = 0.959 \quad \text{API 1111 (4) combined load check}
\]

---

Spool will be operating at extreme hydrotest limit

---
Appendix B: Flange Analyses in ANSYS

/Documents not embedded in this report due to file size; included in PDF rendition/
ELEMENT SOLUTION
STEP=3
SUB =6
TIME=3
SEQV (NOAVG)
DMX = .057903
SMN = 19.089
SMX = 106.123
NODAL SOLUTION
STEP=3
SUB =6
TIME=3
USUM (AVG)
RXY=0
DMX = .089055
SMN = .016371
SMX = .089055
PRINT REACTION SOLUTIONS PER NODE

***** POSTI TOTAL REACTION SOLUTION LISTING *****

LOAD STEP= 3  SUBSTEP= 6
TIME=  3.0000  LOAD CASE=  0

THE FOLLOWING X,Y,Z SOLUTIONS ARE IN THE GLOBAL COORDINATE SYSTEM

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<td>5439</td>
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TOTAL VALUES

| VALUE | 0.0000 | -3343.4 | 0.0000 |

408024-00019-SE-REP-0001 Rev C  Page 5 of 5  TREX-120129.390
HIGHLY CONFIDENTIAL
Appendix C: Internal Guide Design

/Documents not embedded in this report due to file size; included in PDF rendition/

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<td>Page:</td>
<td>Page 21 of 22</td>
</tr>
</tbody>
</table>

Warning: Check DW Doc revision to ensure you are using the correct revision.
INPUT VALUES

D := 18-in Diameter of Pipe

t := 0.375-in Wall Thickness

E := 29-10^6 psi Modulus of Elasticity

F_{DR} := 70-kip Weight of Dual Ram

Hand-Calc. Check Of 18" Pipe Internal Guide:

Case 1

10% of 70 kip Dual Ram side load
100% of 70 kip DR vertical load on
9-inch arc of pipe.

Pipe Properties

\[ A = \frac{\pi}{4} \left( D^2 - (D - 2t)^2 \right) \]  
Steel Area of Pipe

\[ I = \frac{\pi}{64} \left[ D^4 - (D - 2t)^4 \right] \]  
Moment of Inertia of Pipe

\[ S = \frac{1}{\left( \frac{D}{2} \right)} \]  
Elastic Modulus

S_{MYS} := 35-ksi SMYS (Specified Minimum Yield Strength)

Bending Check Due to 70 kips up to 52' below with vertical load on one side

\[ M_b := F_{DR} \left( 0.1 \cdot 52 \cdot \text{in} + \frac{D}{2} \right) \]  
\[ M_b = 994.0 \text{kip-in} \]

\[ f_b := \frac{M_b}{S} \]

\[ f_b = 11.1 \text{-ksi} \]

\[ f_a := \frac{F_{DR}}{A} \cdot \frac{6}{\pi} \]

\[ f_a = 20.2 \text{-ksi} \]

\[ f_a + f_b = 31.3 \text{-ksi} \]

\[ UC := \left( \frac{f_a}{0.6F_{Y} \left( \frac{4}{3} \right)} \right) + \left( \frac{f_b}{0.75F_{Y} \left( \frac{4}{3} \right)} \right) \]

\[ UC = 1.04 \]
Bearing Check Due to 70 kips on 9" arc length

\[ f_p := \frac{F_{DR}}{9 \text{-in}\cdot\text{ft}} \quad f_p = 20.7 \text{-ksi} \]

\[ UC := \left[ \frac{f_p}{0.9 \cdot F_y \left( \frac{4}{3} \right)} \right] \quad UC = 0.49 \]

Check buckling capacity of strips of shell between vertical slots

\[ l_{\text{max}} := 1.5 \text{-in} \quad \text{Maximum length of most highly stressed slot} \]

\[ r_{\text{strip}} := \sqrt{\frac{r^2}{12}} \quad r_{\text{strip}} = 0.1083 \text{-in} \]

\[ k := 0.7 \quad \text{Assuming fixed-pinned behavior (AISC Table C-C2.1)} \]

\[ k \cdot \frac{l_{\text{max}}}{r_{\text{strip}}} = 97.0 \]

\[ C_c := \sqrt{\frac{2 \cdot \pi^2 \cdot E}{F_y}} \quad C_c = 127.9 \]

\[ F_a := \frac{\left( \frac{l_{\text{max}}^2}{k \cdot r_{\text{strip}}} \right)^2}{1 - \left( \frac{k \cdot r_{\text{max}}}{r_{\text{strip}}} \right)^2} \cdot F_y \quad (\text{AISC E2-1}) \quad F_a = 13.1 \text{-ksi} \]

\[ \left( \frac{2}{3} \right) + \frac{\left( \frac{l_{\text{max}}}{k \cdot r_{\text{strip}}} \right)^3}{8 \cdot C_c} - \frac{\left( \frac{l_{\text{max}}}{k \cdot r_{\text{strip}}} \right)^3}{8 \cdot C_c^3} \]

So allowable compressive load on a 3.2" wide strip with 1/3 overstress

\[ P_{\text{strip}} := 3.2\text{-in}\cdot\text{ft} \cdot F_a \cdot \frac{4}{3} \quad P_{\text{strip}} = 21.0349 \text{-kip} \]

Conservatively replace strip with opposing 20 kip reactions to simulate strip at point of buckling in structural analyses.
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**END**
Appendix D: Reference Drawings

(Documents not embedded in this report due to file size; included in PDF rendition)
PRELIMINARY
TRANSITION SPOOL - OPTION 1
API 18-3/4" 15KSI FLANGE x HMF G-FLANGE
SCALE: NONE
May 1, 2010
001

HIGHLY CONFIDENTIAL
MOCK WEAR BUSHING FOR TRANSITION SPOOL SIT
MACHINED FROM 20,000" OD X 0.825" WT, ASTM A106 GR B OR EQUAL

BUFF, SAND OR POLISH INSIDE SURFACES AND EDGES SMOOTH
0.15 x 45° EACH END
BREAK SHARP EDGE EACH END

SCALE NONE

May 5, 2010

INTECSEA
403824-00019-SE-REP-0001 Rev C
Appendix D
Page 5 of 17
HIGHLY CONFIDENTIAL

TREX-120129.416
Summary of Boost circuit, Riser Adapter, Main Tube, HMF Nose Ring:

**Metallic Components**
The calculations completed for the riser adapter and booster circuit component maximum pressure capabilities have resulted in the structural limitation due to pressure for these items being in the pipe section attached to the riser coupling pin. The calculated capacity at that location using 2.5 x 1.0 inch nominal dimensions is 6900 psi considering 90% of the minimum material yield strength as an allowable stress (testing allowable).

**Riser Adapter Elbow face seal**
This arrangement utilizes an o-ring face seal that is preloaded by bolting. The bolt preload at assembled bolt torque values is equalized at approximately 6600 psi, which is the point where the face of the elbow may start to elastically separate from the riser adapter. However, additional separation due to a higher internal pressure only increases the separation by .001 inch for each 632 psi increase in pressure.

**Elastomeric Components**
The o-ring type seals have been tested to a maximum of 5000 psi at 180 deg F. Under the normal conditions of use in this riser adapter/nose ring interface, higher internal pressure tends to decrease the seal extrusion gap between the nose ring and bore of the HMF connection.
These formulas assume normal operating conditions apply.

1. Booster Sub Box at Minimum Cross Section:

$$\sigma_{y,\text{box}} := 75000\text{-psi}$$  
Box material yield strength

$$C_d := .05\text{-in}$$  
Corrosion allowance

$$d_B := 2.88\text{-in}$$  
Sub Box ID at minimum cross section

$$t_B := .459\text{-in}$$  
Wall Thickness at minimum cross section.

$$d_{OD} := d_B + 2 \cdot t_B = 3.798\text{-in}$$  
Minimum OD based on minimum cross section for conservative est.

$$a_1 := \frac{d_{OD}}{2}$$  
Outer radius

$$b_1 := \frac{d_B}{2}$$  
Inner radius

$$p_1 := 6000\text{-psi}$$  
Working pressure at inner radius

$$p_0 := 0$$  
Working pressure at outer radius

$$t := a_1 - b_1$$  
Wall thickness

$$\sigma_h := \frac{p_1 \cdot b_1}{t} = 18824\text{-psi}$$  
Membrane hoop stress

$$\sigma_r := \frac{-p_1}{2} = -3000\text{-psi}$$  
Membrane radial stress

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\( \sigma_1 := 0.0 \text{-psi} \)  
\( \sigma_1 := \sigma_h - \sigma_t = 2.182 \times 10^4 \text{-psi} \)  
Minimum membrane longitudinal stress

\( \sigma_2 := \sigma_t - \sigma_1 = -3 \times 10^3 \text{-psi} \)  
Stress intensities

\( \sigma_3 := \sigma_1 - \sigma_h = -1.882 \times 10^4 \text{-psi} \)

\( \sigma_c := \max\left(\left|\sigma_1\right|, \left|\sigma_2\right|, \left|\sigma_3\right|\right) \)  
\( \sigma_c = 2.1824 \text{-psi} \)  
Maximum stress intensity

\( \text{UF4} := \frac{\sigma_c}{2 \sigma_y \text{box}} \)  
\( \text{UF4} = 0.436 \)  
Utilization factor for working pressure

\( P_{\text{maxB}} := \frac{P_i}{\text{UF4}} \)  
\( P_{\text{maxB}} = 13.747 \text{-ksi} \)  
Maximum allowable working pressure

NOTE: tested to 9000 psi
2. Booster pipe:

\[ \sigma_{\text{ytubing}} = 80000 \text{-psi} \]
Tube material yield strength

\[ OD_T := 5 \text{-in} \]
Tube OD

\[ t := .5 \text{-in} \]
Tube wall thickness

\[ ID_{\text{max}} := OD_T - 2 \ast t + 2 \ast C_a = 4.1 \text{-in} \]
Tube ID including corrosion effect

\[ a_2 := \frac{OD_T}{2} = 2.5 \text{-in} \]
Outside radius

\[ b_2 := \frac{ID_{\text{max}}}{2} = 2.05 \text{-in} \]
Inside radius

\[ p_1 = 6000 \text{-psi} \]
Working pressure at inner surface

\[ p_0 = 0 \text{-psi} \]
Pressure at outer surface

\[ \sigma_{\text{mm}} := \frac{p_1 \ast b_2}{t} = 24600 \text{-psi} \]
Membrane hoop stress

\[ \sigma_{\text{mr}} := \frac{-p_1}{2} = -3000 \text{-psi} \]
Membrane radial stress

\[ \sigma_{\text{mm}} := 0 \text{-psi} \]
Minimum membrane longitudinal stress

\[ \sigma_P := \sigma_h - \sigma_r = 2.76 \times 10^4 \text{-psi} \]
Stress intensities

\[ \sigma_P := \sigma_r - \sigma_1 = -3 \times 10^3 \text{-psi} \]

\[ \sigma_P := \sigma_1 - \sigma_h = -2.46 \times 10^4 \text{-psi} \]

\[ \sigma_{\text{max}} := \max\{\sigma_1, \sigma_2, \sigma_3\} \quad \sigma_c = 27600 \text{-psi} \]
Maximum stress intensity

\[ UF5 := \frac{\sigma_c}{3 \ast \sigma_{\text{ytubing}}} \quad UF5 = 0.517 \]
Utilization factor for working pressure

\[ P_{\text{maxT}} := \frac{p_1}{UF5} \quad P_{\text{maxT}} = 11.59 \times 10^3 \text{psi} \]
Maximum allowable working pressure

NOTE: tested to 9000 psi

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TREX-120129.432

HIGHLY CONFIDENTIAL

LNL004-026411
3. Riser Adapter Valve

Customer supplied valve with 10k flanges.

4. Riser Adapter Ell Face Seal

\[ S_{\text{bolt}} := 65 \text{ ksi} \]
\[ \frac{W}{A} := 0.5 \frac{S_{\text{bolt}}}{E} = 32.5 \text{ ksi} \]
\[ A_{\text{bolt}} := 0.999 \text{ in}^2 \]
\[ N_{\text{bolt}} := 4 \]
\[ \text{Preload} := T \cdot A_{\text{bolt}} \cdot N_{\text{bolt}} = 129870 \text{ lbf} \]
\[ D_{\text{seal}} := 5.005 \text{ in} \]
\[ A_{\text{seal}} := \frac{\pi}{4} \cdot D_{\text{seal}}^2 = 19.674 \text{ in}^2 \]
\[ P := \frac{\text{Preload}}{A_{\text{seal}}} = 6601 \text{ psi} \]

B8M Bolt yield strength
Bolt stress when preloaded to 1/2 yield
Area of bolt
Number of bolts
Preload resulting from bolts
Face seal diameter
Area of face seal
Pressure required to exceed preload

NOTES:
1. 6600 psi is the calculated value where the pressure end load may start to exceed the preload in the connection.

\[ L := 9.64 \text{ in} \]
\[ \delta := 0.001 \text{ in} \]
\[ E := \frac{\delta \cdot A_{\text{bolt}} \cdot N_{\text{bolt}} \cdot 3000000 \text{ psi}}{L} = 12436 \text{ lbf} \]
\[ P_2 := \frac{F}{A_{\text{seal}}} = 632 \text{ psi} \]

Length of bolt to 1/2 grip each side
deflection
Force to stretch bolting .001"
Pressure to stretch bolting .001"

This arrangement utilizes an o-ring face seal that is preloaded by bolting. The bolt preload at assembled bolt torque values is equalized at approximately 6600 psi, which is the point where the face of the elbow may start to elastically separate from the riser adapter. However, additional separation due to a higher internal pressure only increases the separation by .001 inch for each 632 psi increase in pressure.

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TREX-120129.433
HIGHLY CONFIDENTIAL
LNL004-026412
5. HMF Nose Ring and seals

The o-ring type seals have been tested to a maximum of 5000 psi at 180 deg F. Under the conditions of use in this riser adapter/nose ring interface, higher internal pressure tends to decrease the seal extrusion gap between the nose ring and bore of the HMF connection.

To evaluate the nose ring and include the support of material nearby the oring seal grooves an axisymmetric FEA was performed with 6500 psi internal pressure.
Calculate pressure capacity of nose ring ignoring backup from HMF flange connection (conservative).

\[ P_{\text{fe}} = 6500 \text{-psi} \]
\[ \sigma_{\text{mem}} = 39910 \text{-psi} \]
\[ \sigma_{\text{mb}} = 63840 \text{-psi} \]
\[ S_y = 80000 \text{-psi} \]

\[ P_{\text{allow1}} := \frac{2}{3} \frac{S_y}{\sigma_{\text{mem}}} \cdot P_{\text{fe}} = 8686 \text{-psi} \]

Calculate pressure capacity based on membrane stress.

\[ P_{\text{allow2}} := \frac{S_y}{\sigma_{\text{mb}}} \cdot P_{\text{fe}} \]

Calculate pressure capacity based on membrane plus bending stress.
6. Main Tube

The calculations completed for the riser adapter and booster circuit component maximum pressure capabilities have resulted in the structural limitation due to pressure for these items being in the pipe section attached to the riser coupling pin. The calculated capacity at that location using 21.5 x 1.0 inch nominal dimensions is 6900 psi considering 90% of the minimum material yield strength as an allowable stress (testing allowable).

**Input Data**

- **Outside Diameter, nominal**
  \[ \text{OD} := 21.5\text{-in} \]
  \[ h_w = 1.0\text{-in} \]

- **Inside Diameter**
  \[ \text{ID} := \text{OD} - 2 \cdot t = 19.5\text{-in} \]

- **Internal Pressure**
  \[ P := 5.22\text{-ksi} \]

- **Test Pressure**
  \[ P_{\text{test}} := 6.9\text{-ksi} \]

- **Material Yield Strength**
  \[ S_{\text{y}} := 80\text{-ksi} \]

- **Capped ends (1=Y,0=N)**
  \[ \text{CAP} := 0 \]

- **Axial Load**
  \[ F := 0\text{-lbf} \]

- **Moment**
  \[ M := \frac{0}{12}\text{-lbf-ft} \]

**Stress calculation Variables**

- Inside Radius
  \[ r_i := \text{ID} \cdot 0.5 \]
  \[ r_i = 9.75\text{-in} \]

- Outside Radius
  \[ r_o := \text{OD} \cdot 0.5 \]
  \[ r_o = 10.75\text{-in} \]

- Moment of Inertia
  \[ I := \pi \cdot 64^{-1} \left( \text{OD}^4 - \text{ID}^4 \right) \]
  \[ I = 3.391 \times 10^3\text{-in}^4 \]

**Principle Stress Calculations**

1) **Stress Calculations at ID, Design Pressure + Applied Loads**

- **Axial Stress**
  \[ \sigma_{ai} := \frac{\text{CAP} \cdot P \cdot r_i^2 + \frac{F}{\pi} \cdot \frac{r_o^2 - r_i^2}{I}}{r_o \cdot r_i} \]
  \[ \sigma_{ai} = 0\text{-ksi} \]

- **Radial Stress**
  \[ \sigma_{ri} := -P \]
  \[ \sigma_{ri} = -5.22\text{-ksi} \]

- **Hoop Stress**
  \[ \sigma_{\theta i} := P \cdot \frac{\left( r_o^2 + r_i^2 \right)}{r_o^2 - r_i^2} \]
  \[ \sigma_{\theta i} = 53.632\text{-ksi} \]
2) Stress Calculations at OD, Design Pressure + Applied Loads

Axial Stress
\[ \sigma_{ao} := \frac{C A P \cdot P \cdot r_1^2}{r_0^2 - r_1^2} + \frac{F}{\pi} + \frac{M \cdot r_0}{I} \quad \sigma_{ao} = 0\text{-ksi} \]

Radial Stress
\[ \sigma_{ro} := 0\text{-psi} \quad \sigma_{ro} = 0\text{-ksi} \]

Hoop Stress
\[ \sigma_{ho} := 2 \cdot P \cdot \frac{r_1^2}{r_0^2 - r_1^2} \quad \sigma_{ho} = 48.412\text{-ksi} \]
3) Average Stress on Area, Design Pressure + Applied Loads

Hoop Stress

\[ \sigma_0 := \frac{2 \cdot P \cdot r_1^2}{\left( \frac{r_0^2 - r_1^2}{2} \right)^2} \left[ \frac{\left( \frac{r_0^2 - r_1^2}{2} \right)}{r_0^2} \cdot \ln \left( \frac{r_0}{r_1} \right) \right] \]

\[ \sigma_0 = 50.853 \text{-ksi} \]

Radial Stress

\[ \sigma_r := \frac{2 \cdot P \cdot r_1^2}{\left( \frac{r_0^2 - r_1^2}{2} \right)^2} \left( \frac{r_0^2 - r_1^2}{2} - \frac{r_0^2}{2} \cdot \ln \left( \frac{r_0}{r_1} \right) \right) \]

\[ \sigma_r = -2.44 \text{-ksi} \]

Axial Stress

\[ \sigma_a := \frac{CAP \cdot P \cdot r_1^2}{r_0^2 - r_1^2} \left[ \frac{E}{\pi} \cdot M \left( \frac{OD^2 + ID \cdot OD + ID^2}{3 \cdot (OD + ID)} \right) \right] \]

\[ \sigma_a = 0 \text{-ksi} \]

4) Stress Calculations at ID, Test Pressure

Axial Stress

\[ \sigma_{ait} := \frac{CAP \cdot P_{test} \cdot r_1^2}{r_0^2 - r_1^2} \quad \sigma_{ait} = 0 \text{-ksi} \]

Radial Stress

\[ \sigma_{rit} := -P_{test} \quad \sigma_{rit} = -6.9 \text{-ksi} \]

Hoop Stress

\[ \sigma_{\theta it} := P_{test} \left( \frac{r_i^2 + r_0^2}{r_0^2 - r_i^2} \right) \quad \sigma_{\theta it} = 70.893 \text{-ksi} \]

5) Average Stress on Area, Test Pressure

Hoop Stress

\[ \sigma_{it} := \frac{2 \cdot P_{test} \cdot r_1^2}{\left( \frac{r_0^2 - r_1^2}{2} \right)^2} \left[ \frac{\left( \frac{r_0^2 - r_1^2}{2} \right)}{r_0^2} \cdot \ln \left( \frac{r_0}{r_1} \right) \right] \]

\[ \sigma_{it} = 67.219 \text{-ksi} \]

Radial Stress

\[ \sigma_{it} := \frac{2 \cdot P_{test} \cdot r_1^2}{\left( \frac{r_0^2 - r_1^2}{2} \right)^2} \left( \frac{r_0^2 - r_1^2}{2} - \frac{r_0^2}{2} \cdot \ln \left( \frac{r_0}{r_1} \right) \right) \]

\[ \sigma_{it} = -3.226 \text{-ksi} \]

Axial Stress

\[ \sigma_{it} := \frac{CAP \cdot P_{test} \cdot r_1^2}{r_0^2 - r_1^2} \quad \sigma_{it} = 0 \text{-ksi} \]
Comparison of values to code allowables

Membrane plus bending allowable
\[ S_{mb} = S_{y} \]
Membrane allowable, Design
\[ S_{md} = \frac{2}{3} S_{y} \]
Membrane allowable, Test
\[ S_{mt} = 9 S_{y} \]
\[ S_{mb} = 80 \text{-ksi} \]
\[ S_{md} = 54.323 \text{-ksi} \]
\[ S_{mt} = 72 \text{-ksi} \]

1) Membrane plus bending at ID, Design

\[ \sigma_{mb} = \begin{pmatrix} \sigma_{ai} \\ \sigma_{ri} \\ \sigma_{0i} \end{pmatrix} \]
\[ \sigma_{mb} = \begin{pmatrix} 0 \\ -5.22 \\ -53.632 \end{pmatrix} \text{-ksi} \]

Stress Intensity
\[ S_{intmb} := \max(\sigma_{mb}) - \min(\sigma_{mb}) \]
\[ S_{intmb} = 58.852 \text{-ksi} \]

Safety Factors
\[ SF_0 := \frac{S_{mb}}{S_{intmb}} \]
\[ SF_0 = 1.359 \]

2) Membrane plus bending at OD, Design

\[ \sigma_{mb} := \begin{pmatrix} \sigma_{ao} \\ \sigma_{ro} \\ \sigma_{0o} \end{pmatrix} \]
\[ \sigma_{mb} = \begin{pmatrix} 0 \\ 0 \\ 48.412 \end{pmatrix} \text{-ksi} \]

Stress Intensity
\[ S_{intmb} := \max(\sigma_{mb}) - \min(\sigma_{mb}) \]
\[ S_{intmb} = 48.412 \text{-ksi} \]

Safety Factors
\[ SF_1 := \frac{S_{mb}}{S_{intmb}} \]
\[ SF_1 = 1.652 \]

3) Membrane Stress, Design

\[ \sigma_{m} := \begin{pmatrix} \sigma_a \\ \sigma_r \\ \sigma_\theta \end{pmatrix} \]
\[ \sigma_{m} = \begin{pmatrix} 0 \\ -2.44 \\ 50.853 \end{pmatrix} \text{-ksi} \]

Stress Intensity
\[ S_{intm} := \max(\sigma_{m}) - \min(\sigma_{m}) \]
\[ S_{intm} = 53.293 \text{-ksi} \]

Safety Factors
\[ SF_2 := \frac{S_{m}}{S_{intm}} \]
\[ SF_2 = 1.001 \]

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4) Von Mises Stress at ID, Test Pressure

\[
\sigma_{mb} := \begin{pmatrix} \sigma_{ait} \\ \sigma_{rit} \\ \sigma_{\theta ii} \end{pmatrix} \quad \sigma_{mb} = \begin{pmatrix} 0 \\ -6.9 \\ 70.893 \end{pmatrix} \text{ksi}
\]

Von Mises Stress
\[
s_{\text{equiv}} := \sqrt{\frac{1}{2} \left( (\sigma_{ait} - \sigma_{rit})^2 + (\sigma_{ait} - \sigma_{\theta ii})^2 + (\sigma_{rit} - \sigma_{\theta ii})^2 \right)}
\]

Safety Factors
\[
SFT_0 := \frac{s_{mb}}{s_{\text{equiv}}} \quad SFT_0 = 1.073
\]

5) Membrane Stress, Test

\[
\sigma_m := \begin{pmatrix} \sigma_{alt} \\ \sigma_{rt} \\ \sigma_{\theta t} \end{pmatrix} \quad \sigma_m = \begin{pmatrix} 0 \\ -3.226 \\ 67.219 \end{pmatrix} \text{ksi}
\]

Stress Intensity
\[
s_{\text{min}} := \max(\sigma_m) - \min(\sigma_m) \quad s_{\text{min}} = 70.445 \text{ksi}
\]

Safety Factors
\[
SFT_1 := \frac{s_{\text{min}}}{s_{\text{equiv}}} \quad SFT_1 = 1.022
\]

Safety Factors
For Design Pressure-Tension-Bending case, all factors must be greater than 1 to satisfy API criteria

\[
\begin{pmatrix} 1.359 \\ 1.652 \\ 1.001 \end{pmatrix} \quad \text{ID m+b, field conditions}
\]

\[
\begin{pmatrix} 1.652 \\ 1.001 \end{pmatrix} \quad \text{OD m+b, field conditions}
\]

\[
\text{min}(SF) = 1.001
\]

For Test Conditions, one of the following criteria must be greater than 1 to satisfy API criteria

\[
\begin{pmatrix} 1.073 \\ 1.022 \end{pmatrix} \quad \text{Von Mises stress at ID membrane, test pressure}
\]

\[
\text{min}(SFT) = 1.022
\]
EQUIPMENT

HMF-G Marine Riser System and Handling Tools

TECHNICAL DESCRIPTION

Riser Joint - Type HMF-G

The VetcoGray HMF-G riser is designed for high tension and bending loads encountered in deep water. Locking bolts, used to secure multiple joints together in a string, are torqued so that the preload exceeds the rated working load. This prevents flange separation, increases fatigue life, and reduces the stresses in the flange less than that experienced by the pipe. A stepped diameter design of the pin and box connection facilitates engagement under severe vessel movement in rough seas. The HMF-G riser joint consists of a pipe body with an HMF-G flanged pin on one end and an HMF-G flanged box on the other. Each joint contains two (2) choke & kill lines, a hydraulic line, a booster line, and may have preparations for buoyancy modules. The auxiliary lines are secured to the riser joint at the pin and box flanges as well as along its length with clamp assemblies spaced approximately at 12 foot intervals.

Handling Tool - Mechanical

The VetcoGray manual riser handling/test tool is designed for handling and testing the 21" HMF class G riser joints. In addition, it can be used for handling the telescopic joint. The unit has a removable thread-on elevator shoulder on top for interfacing with elevator. The tool is secured to the riser joints by six HMF bolts which need only be torqued with an impact wrench to 1000 ft-lb (1 350 Nm) making the use of the hydraulic torque wrench not necessary for riser handling make-up.

The tool has integral test pins, which allow pressure testing of the choke and kill, booster and hydraulic lines when required during riser running operations. Each test pin is mounted in a keyed preparation in the tool flange, which allows them to be retracted when not in use. Extension of the test pins is accomplished by removing the handle retainer pin, rotating the test pin 90° to allow downward movement, then rotating back 90° to lock the pin in place. The test pins can also be extended prior to engaging the tool into a joint, if required.
Handling Tool - Hydraulic

The VetcoGray hydraulic riser handling/test tool is designed for handling and testing the 21" HMF class G riser joints. In addition, it is utilized for handling the telescopic joint and intermediate flex joint. The unit has a removable thread-on elevator shoulder on top for interfacing with elevator. The tool is hydraulically operated through an internal cylinder, which moves the tool stem up and down. The upward stem movement drives six dogs radially outward which engage a mating grooved profile in the inside diameter of each HMF flange box coupling. Downward stem movement allows the dogs to retract, releasing the tool from the coupling. The rising stem of the tool provides a fail-safe feature in that it cannot be released while under a load.

The tool has integral test pins, which allow pressure testing of the choke and kill, booster and hydraulic lines when required during riser running operations. Each test pin is mounted in a keyed preparation in the tool flange, which allows them to be retracted when not in use. Extension of the test pins is accomplished by removing the handling retainer pin, rotating the test pin 90° to allow downward movement, then rotating back 90° to lock the pin in place. The test pins can also be extended prior to engaging the tool into a joint, if required.

SPECIFICATIONS

- Refer to the Bill of Materials and Assembly Drawings for all major dimensions

- Recommended Lubricants:
  - Moving Parts: Jet Lube ALCO EP-73 Plus
  - Threads: Never-Seez, Regular Grade Paste
  - HMF Bolts and Inserts: Never-Seez, Regular Grade Paste or TS 70 Moly Paste

- Handling Tool - Mechanical:
  - Maximum tensile capacity: 2,000,000 lb (907 200 kg)
  - Weight: 6557 lb (2 974 kg)
  - Length: 104.63" (2.65m)
  - Maximum diameter: 42"
  - Bolt make-up torque: 1000 ft-lb (1 356 Nm)

- Handling Tool - Hydraulic:
  - Maximum tensile capacity 2,000,000 lb (907 200 kg)
  - Weight: 6267 lb (2 842 kg)
  - Length: (extended) 110" (2.8 m)
  - Maximum diameter: 42"
  - Max. Operating Pressure: 2000 psi (138 bar)
  - Max. Test Pressure: 3000 psi (207 bar)
## HMF-G MARINE RISER SYSTEM
### AND HANDLING TOOLS

### TABLE 1

<table>
<thead>
<tr>
<th>HMF Class</th>
<th>P/N</th>
<th>QTY</th>
<th>Torque with Never-Seez Regular Grade in ft-lb (+/- 250 ft-lb)</th>
<th>Torque with TS 70 Moly Paste in ft-lb (+/- 250 ft-lb)</th>
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</thead>
<tbody>
<tr>
<td>Class D</td>
<td>H10508-1</td>
<td>6</td>
<td>9000</td>
<td>7500</td>
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<td>6</td>
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<td></td>
<td>H141522-3</td>
<td>6</td>
<td>23000</td>
<td>19250</td>
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## RECOMMENDED FASTENER TORQUES

Material: B7 (105 ksi yield)
Assumed Coefficient of Friction: Never-Seez (Cf = 0.085)
TS 70 Moly Paste (Cf = 0.069)
Preload = 65% of bolt yield strength

<table>
<thead>
<tr>
<th>Bolt Size (UNC Series)</th>
<th>Torque Never-Seez (ft-lb)</th>
<th>Torque TS 70 Moly Paste (ft-lb)</th>
<th>Bolt Size (UN Series)</th>
<th>Torque Never-Seez (ft-lb)</th>
<th>Torque TS 70 Moly Paste (ft-lb)</th>
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<td>.250-20</td>
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<td>9040</td>
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<td>26152</td>
<td>21455</td>
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</table>

*Bolt sizes greater than 2.500" have a material yield strength of 95,000 psi
U.S. CUSTOMARY UNITS TO SI UNITS

<table>
<thead>
<tr>
<th>Measurement Type</th>
<th>English Unit</th>
<th>SI Units</th>
<th>SI Symbol</th>
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<td>PSI</td>
<td>Megapascals</td>
<td>MPa</td>
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<td>Weight</td>
<td>Pounds</td>
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<td>Torque</td>
<td>Foot-Pounds</td>
<td>Newton Meters</td>
<td>Nm</td>
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<tr>
<td>Volume</td>
<td>Gallons</td>
<td>Liters</td>
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<tr>
<td>Length</td>
<td>Inch</td>
<td>Millimeter</td>
<td>mm</td>
<td>25.4</td>
</tr>
</tbody>
</table>

Metric equivalents used in VetcoGray procedures are expressed in SI units as illustrated above.

ROUTINE OPERATIONS

PRE-OPERATIONAL CHECKS

Riser Joint
1. Visually inspect accessible riser pipe and welds for damage.
2. Remove the pin and box protectors.
3. Thoroughly clean the riser flange pin, flange box and threaded inserts on the riser joint for inspection. Clean all elastomer seals such that each may be visually inspected. Thoroughly clean the pin and box connections on the choke, kill, hydraulic, and booster lines for inspection. Replace all damaged or worn seals. Steam cleaning or high pressure water blast will provide the quickest and most thorough results. However, if buoyancy equipment is used, care must be taken not to damage the buoyancy equipment during cleaning.
4. Lubricate the riser pin and box flanges, choke, kill, booster and hydraulic pins with a light coat of grease. Also, lubricate the bolts, threaded inserts and washers with a light coat of Never-Seez, Regular Grade Paste.
5. Install the pin and box protectors. Pin and box protectors must be installed on a joint of riser prior to it being moved.

Handling Tool - Mechanical
1. Set the tool on the rig floor.
2. Remove the pin protector.
3. Push the locking bolts past the retainer exposing the bolt threads below the pin flange. Clean the handling tool pin flange and the locking bolt threads.
4. Visually inspect all of the parts for signs of damage. Repair or replace components that are damaged.
5. Lubricate the pin, flange and test pins with a light coat of grease.
6. Lubricate the locking bolt's face, threads and washers with a light coat of grease.
7. Inspect the lifting padeye for cracks or damage. Repair as required.

8. Inspect the square shoulder used by the elevator to lift the riser string. It should be clean and square. Rounded shoulders in this area may decrease the rated working load of the tool.

9. Lubricate the tool through all grease fittings.

Handling Tool - Hydraulic
1. Set the tool on the rig floor.
2. Attach a hydraulic source to the 3/8", 3000 psi (207 bar) quick disconnect fittings located on the stem of the tool. Cycle the stem up and down and verify that it moves freely and the dogs can be pushed in fully when the tool is in the "unlocked" position.
3. Lubricate the dogs and body with a light coat of grease.
4. Inspect the lifting padeye for damage. Repair as required.
5. Inspect the square shoulder used by the elevator to lift the riser string. It should be clean and square. Rounded shoulders in this area may decrease the rated working load of the tool.
6. Verify the slotted hex nut on the bottom of the tool is tight. This nut secures the stem and the hydraulic cylinder.
7. Lubricate the tool through all grease fittings.

RUNNING PROCEDURE

The procedure assumes the hydraulic handling tool is being used. The procedure for using the mechanical handling tool is the same except torque is used to attach the tool to the HMF-G coupling flange. (The bolt make-up torque for the running tool is 1000 ft-lb only when running riser.)

1. Position the riser spider and gimbal in the rotary table.
2. Pick up a riser joint and set it in the "V" door, box end up. Remove the box protector.

NOTE:

When handling riser joints, care must be taken to prevent risk from injury of personnel or damage to equipment. Foam buoyancy modules on risers are especially vulnerable to damage. If it is necessary to pick up and move riser with a crane when automatic handling equipment is not available, a spreader bar or sling should be utilized.

3. Pick up the riser handling tool and make-up to the elevator.
4. Stab the handling tool into the riser box flange.
5. Lock the handling tool using 1500 psi locking pressure. When using this mechanical tool, torque the six (6) bolts to 1000 ft-lb (1 350 Nm). After tool is locked in flange, engage the manual lock.

6. Pick up the riser joint with the handling tool using elevator. Do not use drillpipe elevator. It is recommended that the lower end (pin end) of the riser joint be restrained while the joint is being lifted into the derrick to prevent it from swinging when picked up clear of the rig floor.

7. Remove the pin protector from the pin flange.

8. Lower the riser joint and stab the pin into the riser box of the equipment on the spider, making sure to align the orientation pin prior to stabbing.

9. Using the hydraulic torque tool, spin the bolts up in a star pattern.

10. Apply full recommended torque to the bolts (see Table 1 in Specifications) using the hydraulic torque wrench from the torque wrench kit.

11. Riser connection may be pressure tested at this point using the test port on the OD of the pin flange (Autoclave SG-562-CX). The maximum test pressure is 5000 psi (345 bar).

12. Pick up the riser string and open the riser spider gates.

13. Lower the riser until the next riser box flange is positioned for landing in the spider. Close the spider gates and land the riser string on the spider.

14. Unlock the manual lock, then unlock the riser handling tool using 1500 psi unlock pressure and remove the tool from the riser joint.

15. Pick up and run additional riser joints as per steps 1 through 13.

16. Prior to breaking out the riser handling tool, pressure test the auxiliary lines with the test pins provided on the handling tool after the desired number of joints has been run.

When using the mechanical handling tool, engage the test pins by rotation of each 90° to lower and back 90° to lock in place. Test pins can also be engaged prior to stabbing the mechanical tool into the joint. Pressurize each line to the required test pressure.

**RECOVERING**

1. Install the riser spider and gimbal on the rotary table.

2. Retrieve the diverter using specified procedures, leaving the telescopic joint crossover box flange supported on the riser spider.

3. Retrieve the telescopic joint using the appropriate procedures.

4. Pick up the string and open spider gates and raise until spider gates can be closed under the upper box connector of the next riser joint.

5. Slack off slowly until string weight is fully supported on the spider gates. Back out all bolts on each HMF-G connection. Pick up the telescopic joint, install the pin protector then move the telescopic joint to the ramp or riser handling system.

6. Remove the handling tool and install a box protector before moving the telescopic joint to storage.
NOTE:

All subsea drilling equipment should be washed down with fresh water each time it is recovered to reduce corrosion and extend life.

7. Stab the handling tool into the riser joint box flange.
8. Pick up the string until the box connector of the next riser joint can be landed at the spider.
9. Lower the string and carefully land on the riser spider. Back out all bolts on the riser joint. Make certain each bolt is completely disengaged before picking up and recovering the joint. Pick up on the joint and install the pin protector.
10. Move the joint to the ramp or riser handling skid. Remove the handling tool and install a box protector on the riser joint before moving to the rack.
11. Repeat steps 7 through 10 until all of the riser joints are recovered.

POST OPERATIONAL MAINTENANCE

Riser
1. Remove the pin and box protector from the riser joints.
2. Push the locking bolts past the retaining ring (if so equipped) exposing the bolt threads below the pin flange.
3. Clean the pin, pin flange, auxiliary line pins and locking bolt threads. Visually inspect all of the parts for signs of damage. Repair or replace bolts and washers that are damaged or galled.
4. Lubricate the pin ends of the riser, termination joint, auxiliary lines and telescopic joint with a light coat of grease.
5. Clean the riser auxiliary line boxes and the threaded inserts. Visually inspect all of the parts and seals for signs of damage. Repair or replace components that are damaged.
6. Lubricate the box end of the riser joint and auxiliary boxes with a light coat of grease.
7. Reinstall the protectors.

NOTE:

The riser should be inspected per steps 8 through 9 after any three-year period or after experiencing any abnormal operating conditions prior to its re-use. Abnormal operating conditions can be described as: putting the riser string into compression, subjecting the riser to excessive bending moments or excessive tension, parting the riser, dropping the riser, etc. Individual components subject to stress and fatigue (i.e., bolts) should be subject to NDT examinations every 12 months.
8. Inspect coupling-to-pipe and pipe-to-pipe welds using a magnetic particle or ultrasonic inspection procedure. Magnetic particle inspection will require sandblasting to remove paint in the areas to be inspected. Defective areas, if found, must be ground and feathered out until completely removed. Areas that require grinding beyond API defined tolerances must be repaired. Consult VetcoGray for the detailed repair procedure.

9. Repaint sandblasted areas and/or areas where paint is damaged to VetcoGray paint specifications.

RECOMMENDED SPARES

Selection of actual quantities for spare parts should be determined by customer operating personnel. As a rule of thumb we offers the following suggestions.

Rubber Goods
- Static seals: 1 replacement
- Internal moving seals: 2 replacements
- External seals: 3 replacements
- Special seals: 3 replacements

Hardware
- Socket head cap screws:
  - with 1 to 10 in parts list: 2 spares
  - with 11 or more in parts list: 4 spares
- Socket set screws: full count

Hydraulic Hose
- Enough of each size for complete replacement

Hose Fittings
- Enough of each size for several complete replacements.

General
- 1 replacement for any subassembly which if damaged or lost could cause a rig down situation.

DISASSEMBLY/ASSEMBLY PROCEDURES

DISASSEMBLY PROCEDURE #1

Riser Joints
1. Remove the nose ring from the riser joint by rotating to the right the nose ring (left-hand 4 pitch thread).
2. Remove all elastomer seals from the nose ring and inspect the seal grooves for damage or deterioration. Emery cloth should remove any minor damage.
3. Inspect the threads inside the riser pin flange. Repair as required.
4. Inspect the grooves inside the end of the riser joint that is used by the hydraulic handling tool. Repair as required with emery cloth or small files.
5. Mark the position of the guide clamps or thrust collars. Remove the socket head cap screws securing them to the tube of the riser joint.
6. Remove the six Spirolox retaining rings securing the threaded inserts in the box flange of the riser joint.
7. Remove the six threaded inserts. Inspect the threads and OD for excessive wear.
8. Inspect all HMF-G bolts. Check the hex head for rounded off shoulders. **Do not use bolts with worn shoulders.** Inspect the bolt for galling. Repair all galled surfaces.
9. Inspect the flat washers for galling. Replace all damaged or galled washers. **Do not use washers that indicate galling.**
10. To remove any auxiliary lines, all line clamps must be removed. Back off the set screws that secure the locking nut to the line. This will allow the locking nut to be removed with counterclockwise rotation. The line can be removed through the box flange.

**ASSEMBLY PROCEDURE # 1**

**Riser Joints**

<table>
<thead>
<tr>
<th>NOTE:</th>
</tr>
</thead>
<tbody>
<tr>
<td><em>Lubricate all threaded parts with Never-Seez, Regular Grade Paste, during assembly.</em></td>
</tr>
</tbody>
</table>

1. Install one flat washer on each HMF-G riser bolt. Store all bolts and washers to prevent damage. **Do not install and store the bolts and washers in the riser joint.**
2. Install the threaded inserts in the sockets in the box flange end of the riser joint. Install the Spirolox ring to secure the threaded insert in the riser joint.
3. Install the guide clamps or thrust collars as applicable around the end of the riser joint per the assembly drawing. Torque the 1"-8UNC hex head screws to 387 ft-lb (512 Nm).
4. Install the lip seals on the nose ring. Lubricate the threads and install it in the pin end of the riser joint.
5. Install the auxiliary lines through the box flange allowing the line pin stab to mate with the appropriate hole in the pin flange. Secure the line to the riser joint by installing the locking nut on the pin end of the line. Ensure the locking nut is properly spaced in accordance with the applicable assembly drawing.
6. Tighten the set screws that secure the locking nut to the line.
7. Reinstall all clamps and/or thrust collars.
DISASSEMBLY PROCEDURE # 2

Riser Handling Tool - Mechanical
1. Remove the locking pins that secure the test pins to the HMF flange.
2. Remove all test pins and HMF bolts for cleaning and inspection.
3. Remove the elevator shoulder carried nut.
4. Back off the set screw that secures the retaining ring to the stem. This will allow removal of the retaining ring (nut) that secures the lifting ring to the stem. Use counterclockwise rotation (right-hand threads).
5. Slide lifting ring off the stem.
6. The stem is secured to the HMF flange with left-hand stub acme threads. Remove with clockwise rotation.

ASSEMBLY PROCEDURE # 2

Riser Handling Tool - Mechanical
1. Thoroughly clean all parts and inspect for damage, stem cracks, corrosion, etc. Pay particular attention to the stem, the elevator shoulder carried nut, the lifting ring and padeye and the HMF flange.
2. Minor damage, wear and corrosion may be corrected by hand buffing or emery cloth.
3. Severe corrosion, distortion and/or stress cracks are major concerns and must be corrected before placing the tool back in service.
4. Lubricate each part liberally during assembly.
5. Install O-ring seals in grooves at top and bottom of stem.
6. Thread the HMF flange onto the stem with counterclockwise rotation. Threads need not be torqued, but ensure that the flange is completely seated on the stem shoulder.
7. Slide the lifting ring over the stem and secure it by installing the retaining ring on the stem. Tighten the set screw in the retaining ring against the stem.
8. Thread on the elevator shoulder carried nut.
9. Ensure that the retaining rings in each bolt hole location are in good condition and properly installed. Replace them if necessary. Install the HMF bolts in the “up” position.
10. Inject lubricant through all grease fittings.
DISASSEMBLY PROCEDURE # 3

Riser Handling Tool - Hydraulic
1. Remove the cotter pin and slotted hex nut from bottom end of tool.
2. Remove six screws holding on lower reaction sleeve and remove sleeve.
3. Remove the elevator shoulder carried nut.
4. Back off the set screw that secures the retaining ring to the stem and un-thread the retaining ring. This will allow the lifting ring to slide free from the stem.
5. Remove stem by pulling downward through the bottom of the body.
6. All plumbing and other attached items cannot be removed.

ASSEMBLY PROCEDURE # 3

Riser Handling Tool - Hydraulic
1. Thoroughly clean all parts and inspect for damage, stress cracks, corrosion, etc. Pay particular attention to the stem, the box threads on the upper end of the stem, the lifting ring and padeye, the locking dogs and the dog windows in the body.
2. Minor damage, wear and corrosion may be corrected with hand buffing or emery cloth.
3. Severe corrosion, distortion and/or stress cracks are major concerns and must be corrected before placing the tool back in service.
4. Lubricate each part liberally during assembly.
5. Push stem through center hole of body.
6. Slide the lifting ring over the stem and secure it by installing the retaining ring on the stem. Tighten the set screw in the retaining ring against the stem.
7. Thread on the elevator shoulder carried nut.
8. Attach the hydraulic cylinder to the bottom of the stem.
9. Install the hydraulic plumbing to the cylinder and stem.
10. Install the locking dogs at each position in the body ensuring that the large bevel is toward the stem. Push dogs to the locked (extended) position.
11. Attach the reaction sleeve to the bottom of the housing with the six bolts. Make sure to align the indicator rods with the matching holes in the stem.
12. Thread the slotted nut onto the cylinder rod and torque sufficiently to remove all slack from the connection. Install a new cotter pin to secure the nut.
13. Cycle the tool several times to ensure proper operation. Pressure test both the “latch” and “unlatch” cylinder positions to full operating pressure for five (5) minutes. No leakage is permitted.
Engineering Bulletin

TS70 Moly Paste Lubricant for HMF Connector Bolting

1. Background

1.1. Lubricants

1.1.1. Never-Seez regular grade paste is the current standard for makeup of HMF drilling riser bolting.

1.1.2. TS70 Moly Paste was evaluated as an alternative for Never-Seez to reduce make-up and break-out torques. The evaluation included lab and field trials for performance.

1.2. Recommended alternative

1.2.1. The conclusion of the evaluation is that TS70 Moly Paste may be used instead of Never-Seez, with some adjustment of the torque values to maintain consistent bolt preload.

1.2.2. This bulletin provides instructions for rig use of both Never-Seez and TS70 Moly Paste.

2. HMF Connector Bolting

2.1. Make-up procedure

2.1.1. This procedure applies to the bolting only; all other riser makeup procedures still apply.

2.1.2. Thoroughly clean all old grease from the bolts, washers, inserts, and mating areas of the pin and box flanges. There can be some change in effectiveness if the previous grease contaminates the new grease.

2.1.3. After cleaning, ensure that all bolts and inserts are in good condition with no torn or damaged threads, etc.

2.1.4. Apply the new grease to the bolts and inserts, using standard greasing procedures. Never-Seez Regular Grade and TS70 Moly Paste are both approved alternatives.

2.1.5. Once riser run begins, make up bolting using standard sequences and procedures. Torque the bolts per Section 2.2.
2.2. Torque Recommendations

<table>
<thead>
<tr>
<th>Riser Class</th>
<th>Bolt</th>
<th>Bolt Size (in.)</th>
<th>Torque (ft-lbs) – Never Seez</th>
<th>Torque (ft-lbs) – TS70 Moly Paste</th>
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<tr>
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3. Subsequent Action

3.1.1. This engineering bulletin is being delivered in advance of updates to the rig manuals. Operating and Service Procedures for HMF risers will be updated with TS70 Moly Paste as an alternative as time permits.
Test Certificate

Acute Technological Svs.,
11925 Brittmoore Park Drive
Houston, TX
77041

Attn: Quentin Champ

Item
- 21.5" OD x 1.00" WT API 5L X80 Pipe, Seam Welded

Specification
- API 5L, Gr. X80

Heat Treatment - ASME VIII

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Item 01: Simulated Post Weld Heat Treatment.
Furnace record provided on page 2.

Tensile Test - API 5L

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Item 02: Fracture Location: Weld, Type: Ductile
Item 03: Fracture Location: Weld, Type: Ductile

Approved By Kevin J. Elliott

Kevin J. Elliott
For and on authority of Exova Inc.
Test Certificate

Acute Technological Svs,
11925 Brittmoore Park Drive
21.5"OD x 1.00"WT API 5L X80 Pipe, Seam Welded

Figures - In House Procedure

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Test Certificate

Acute Technological Svcs.,
11925 Brittmoore Park Drive
Houston, TX
77041

Attn: Quenton Champ

Item: 21" OD x 1.125" WT Test Pipe - Heat # xxxxxxx

Specification: API 5L

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Tensile Test - ASTM E8

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Tensile Test - ASTM E8

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Charpy Test - ASTM E23

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</table>

Item 08: Percent Shear: 100, 100, 100 / Mil Lat Exp: 99, 96, 98

The reverting of false, fictitious or fraudulent statements or omissions may be punished as a felony under federal law.
This document may not be reproduced other than in full, except with the prior written approval of the issuing laboratory.
These results pertain only to the items tested as specified by the client unless otherwise indicated.

TREX-120129.462

HIGHLY CONFIDENTIAL

LNL004-026441
Test Certificate

Acute Technological Svs.
11925 Brittmoore Park Drive
21" OD x 1.125" WT Test Pipe - Heat # xxxxxxx

Approved By  Huu Bui

Witnessed by  BP America

REF No  O101787  Issue  1
Page  2 of 2

Kevin J. Elliott
For and on authority of Exova Inc.
Calculating Burst Pressure of Pipe
(based on methods in API RP 1111 Appendix A)

Nomenclature
D = Outside diameter of pipe (inches)
I = Nominal wall thickness of pipe (inches)
E = Modulus of elasticity (psi)
S_y = Specified minimum yield strength of pipe (SMYS, psi)
U = Specified minimum ultimate strength of pipe (psi)
P_y = Elastic collapse pressure of pipe (psi)
P_c = Yield pressure at collapse (psi)
P_e = Collapse pressure of pipe (psi)

\[ P_e = \frac{1}{0.002} \left( \frac{\frac{I}{D}}{1 - \frac{I}{D}} \right) \]  
Elastic collapse pressure \( P_e = \) psi

\[ P_y = 2.25 \left( \frac{\frac{I}{D}}{1 - \frac{I}{D}} \right) \]  
Yield pressure at collapse \( P_y = \) psi

\[ P_c = \sqrt{P_y^2 + P_e^2} \]  
Collapse pressure of pipe \( P_c = \) psi

**Additional input values**
(Assumed 10% over minimum values)

Average measured yield strength of pipe (psi) \( S_{yield} = 38,000\) psi

Average measured ultimate tensile strength of pipe (psi) \( S_{U} = 50,000\) psi

Minimum measured wall thickness (inches) \( I_{min} = 0.10\) in

**Calculated Values for Appendix A**

Cross-sectional area of pipe steel (in²) \( A = \frac{\pi}{4} \left( b^2 - D^2 \right) \)  
\( A = \) in²

External cross-sectional area of pipe (in²) \( A_o = \frac{\pi}{4} D^2 \)  
\( A_o = \) in²

**Calculated Test Pressure Values**

Capped end yield pressure (psi) \( CYP = \frac{2A}{\sqrt{3}} \left( \frac{S_{yield}}{S} \right) \left( \frac{I_{min}}{I} \right) \)  
CYP = psi

Capped end burst pressure (psi) \( CBP = \frac{2Y_{actual}}{\sqrt{3}} \left( \frac{D}{D_t} \right) \left( \frac{S_{yield}}{S} \right) \left( \frac{I_{min}}{I} \right) \)  
CBP = psi
May 8, 2010

Gary E. Harrison
DWGoM Pipeline Technical Authority
BP
Westlake 4 Room 431C
200 Westlake Park Blvd.
Houston, TX 77079
Phone: 281-366-7667
Cell: 281-222-9485
E-mail: gary.harrison@bp.com

SUBJECT: Burst Test of 21.5-inch x 1.0-inch, Grade X80 Riser Pipe Material

Gary,

This letter report provides a summary of the burst test that was performed for BP by Stress Engineering Services, Inc. (SES) on Saturday, May 8, 2010. The test involved pressure testing a 21.5-inch x 1.0-inch, Grade X80 riser pipe (98 inches in length). Slip-in end caps were welded into the pipe sample prior to testing.

The detailed test program is attached to this letter report and provides the specific activities completed in the burst test. Two strain gages were installed at the axial center of the pipe: Gage #1 was positioned adjacent to the longitudinal weld seam, while Gage #2 was positioned 90 degrees circumferentially relative to Gage #1. The outside diameter / wall thickness measurements made at the two strain gage locations were 21.562 inches / 0.997 inches and 21.548 inches / 0.990 inches, respectively. Data for the strain gages and calibrated pressure transducer were recorded at a rate of 1 scan per second. Two 10-minute pressure holds were made at 6,650 psi (design pressure) and 7,500 psi (hydrotest pressure). The estimated capped end burst pressure per API RP 1111 was 10,914 psi (assuming actual yield and ultimate strengths 10 percent greater than the API 5L minimum specified values). The maximum test pressure reached during the burst test was 10,546 psi. This value is 1.59 times the designated design pressure of 6,650 psi. The failure occurred in the middle of the test sample and did not fail in the weld seam.

The following figures are provided on the attached pages.

- Figure 1 – Strain gage measurements as a function of internal pressure
- Figure 2 – Test sample in the SES test pit prior to burst
- Figure 3 – Test sample in the SES test pit after burst
- Figure 4 – Overall view of test sample after burst
- Figure 5 – Close-up view of the fracture surface
- Figure 6 – Pressure transducer calibration certificate

Please contact me if you have any questions.

Regards,

Chris Alexander, Ph.D.
chris.alexander@stress.com
(281) 897-6504 (direct phone) • (281) 450-6642 (cell phone)
Figure 1 – Strain gage measurements (hoop) as a function of internal pressure

Figure 2 – Test sample in the SES test pit prior to burst
Figure 3 – Test sample in the SES test pit after burst
Figure 4 – Overall view of test sample after burst

Figure 5 – Close-up view of test sample after burst
**Figure 6 – Calibration certificate for 30,000 psi pressure transducer**
BP 21.5-inch API-5L X80 Riser Pipe Burst Test Procedure

Project Number: PN1601045
Project Engineer: Chris Alexander (chris.alexander@stress.com | 281-450-6642, cell)
Project Manager: Brent Vyvial (brent.vyyvial@stress.com | 281-380-8506, cell)
Client: BP
Testing Date: May 8, 2010
Testing Description: Burst testing capped 21.5-inch x 1.0-inch, Grade X80 pipe sample

Project Description
This project involves burst testing a 21.5-inch x 1.0-inch, Grade X80 pipe sample. The intent of this effort is to destructively test the pipe to evaluate the integrity of the pipe material and determine its limit state capacity.

SES activities (unless otherwise noted) associated with the test program include the following tasks:
1. Weld end caps to the test sample (6-inch thick beveled disks using 50 ksi material). Per BP’s request, welds will be PWHT at 1150F for 1 hour.
2. Install two bi-axial strain gage rosettes at the axial center of the pipe positioned 90 degrees relative to one another.
3. Verify the pipe diameter and the wall thickness of the pipe using a UT meter (measure regions adjacent to each strain gage and record as appropriate).
4. Place the test assembly in the SES burst chamber and connect necessary hardware and instrumentation.
5. Perform pressure testing on test samples with holds at designated pressure levels (record strain and internal pressure at a rate of 1 scan per second during testing).
6. Perform post-failure inspection after burst testing including:
   a. Photo documentation
   b. Measuring location of failure including length of burst opening and distance of opening from end of pipe sample.
7. Provide a summary report that provides documentation and test results.

Equipment Needed
- SES burst chamber
- Pump for water in sample
- One (1) analog pressure dial pressure gauge
- 30 ksi pressure transducer
- Strain gages (2 bi-axial rosettes)
- DAQ system to record internal pressure and strain (versus time)
- Safety lighting (RED)

Safety Procedures
The following steps will be carried out to ensure the safety of SES lab personnel, guests, and equipment.
- Complete SES Engineering Safe Work Permit
- Place sample in pit and install all of the retaining bolts.
- Announce burst when imminent (after last pressure hold).
- A RED light will be running continuously during testing during the period of imminent failure to designate that testing is in progress.
Testing Protocol

Pre-testing procedures:
1. Install strain gages and connect cables (see Figure 1 for strain gage locations).
2. Measure pipe diameter and wall thickness using UT meter at strain gage locations. Record sample length.
3. Take photographs of sample prior to testing.
4. Connect all pressure equipment (including the dial pressure gauge) and measurement devices required for testing.
5. Place sample in test pit with wood end blocks.
6. Have proper safety indicator lighting available and notify lab personnel once testing is to start.
7. Verify calibration of pressure transducer and strain gages. Make sure all measurement devices zeroed prior to start of test.
   a. Transducer S/N: 267925
   b. Have the calibration certification on the P-drive.

Testing procedures:
1. Start the data acquisition system in order to record data. Record data at 1 scan per second.
2. Increase pressure at a rate not to exceed 25 psi per second.
3. Hold at the following pressure levels for 10 minutes (each hold)
   a. Design pressure: 6,650 psi
   b. Hydrotest pressure: 7,500 psi
4. Take sample to failure pressure, estimated to be 10,914 psi (burst pressure based on API RP 1111 CEBP assuming actual material properties are 10% greater than the minimum specified values as shown on the following page).

Post-testing procedures:
1. Perform failure inspection after burst testing including:
   a. Photo documentation.
   b. Measuring location of failure including length of opening and distance from end cap.
2. Provide a summary report that provides documentation and test results.

Figure 1 – Strain gage locations on test sample
Calculating Burst Pressure of Pipe
(based on methods in API RP 1111 Appendix A)

Nomenclature:
- D = Outside diameter of pipe (inches)
- t = Nominal wall thickness of pipe (inches)
- E = Modulus of elasticity (psi)
- \( S_y \) = Specified minimum yield strength of pipe (SMYS, psi)
- \( U \) = Specified minimum tensile strength of pipe (psi)
- \( P_e \) = Elastic collapse pressure of pipe (psi)
- \( P_y \) = Yield pressure at collapse (psi)
- \( P_c \) = Collapse pressure of pipe (psi)

\[
\begin{align*}
D &= 3.5\text{ in} \\
\frac{1}{t} &= 14000 \text{ psi} \\
\frac{S_y}{U} &= 0.5 \\
\frac{P_y}{P_c} &= 38000 \text{ psi}
\end{align*}
\]

Internal diameter of pipe (inches) \( D_i := D - 2t \)

\[
\begin{align*}
P_e &= 2E \left( \frac{t}{D} \right) \left( \frac{1}{1 - v^2} \right) \\
&P_e = 6634 \text{ psi} \\
P_y &= 2S_y \left( \frac{t}{D} \right) \\
&P_y = 7443 \text{ psi} \\
P_e &= \frac{P_y P_c}{\sqrt{P_y^2 + P_c^2}} \\
&P_e = 4953 \text{ psi}
\end{align*}
\]

Additional input values (Assumed 10% over minimum values)

Average measured yield strength of pipe (psi) \( S_{y,\text{actual}} = 58000 \text{ psi} \)

Average measured ultimate tensile strength of pipe (psi) \( U_{\text{actual}} = 95644 \text{ psi} \)

Minimum measured wall thickness (inches) \( t_{\text{min}} = 1.0 \text{ in} \)

Calculated Values for Appendix A

Cross-sectional area of pipe steel (in\(^2\)) \( A := \pi \left( \frac{D^2}{4} - D_i^2 \right) \)

External cross-sectional area of pipe (in\(^2\)) \( A_o := \pi \frac{D^2}{4} \)

Calculating Test Pressure Values

\[
\begin{align*}
\text{Capped end yield pressure (psi) CEYP} &= \frac{S - A}{\sqrt{A_o}} \left( \frac{S_{y,\text{actual}}}{S} \right)^{\frac{1}{2}} \\
&= \text{CEYP} = 20013 \text{ psi} \\
\text{Capped end burst pressure (psi) CEBP} &= \frac{2\cdot S_{y,\text{actual}}}{\sqrt{A}} \left( \frac{D}{D_i} \right) \left( \frac{S_{y,\text{actual}}}{S} \right)^{\frac{1}{2}} \\
&= \text{CEBP} = 10914 \text{ psi}
\end{align*}
\]
May 9, 2010

Gary E. Harrison
DW GoM Pipeline Technical Authority
BP
Westlake 4 Room 431C
200 Westlake Park Blvd.
Houston, TX 77079
Phone: 281-366-7667
Cell: 281-222-9485
E-mail: gary.harrison@bp.com

SUBJECT: Hydrotest of Valve and Transition Spools

Gary,

This letter report provides a summary of the hydrotests that were performed for BP by Stress Engineering Services, Inc. (SES) on Saturday, May 8, 2010 and Sunday, May 9, 2010.

The detailed test program is attached to this letter report and provides the specific activities completed in the hydrotests. An internal pressure of 7,500 psi was applied to the valve spool and the valve was inspected for leaks twice prior to the hydrotest. During the hydrotest of the valve spool the pressure decreased by 10 psi over a 30 minute period. Upon completion of the hydrotest, an internal pressure of 6,670 psi was applied to the upstream side of the valve to set the seats. The transition spool was tested to 7,500 psi, and the pressure decreased by 13 psi over the 30 minute hold period.

The following figures are provided on the attached pages:
- Figure 1 – Pressure plot for valve spool
- Figure 2 – Pressure plot for transition spool
- Figure 3 – Valve spool in the SES test pit
- Figure 4 – Transition spool in the SES test pit
- Figure 5 – Calibration certificate for upstream pressure transducer
- Figure 6 – Calibration certificate for downstream pressure transducer

Please contact me if you have any questions.

Regards,

Brent Vyvial
brent.vyvial@stress.com
(281) 897-1014 (direct phone)
(281) 955-2900 (office)
Figure 1 – Pressure plot for valve spool
Figure 2 – Pressure plot for transition spool
Figure 3 – Valve spool in the SES test pit

Figure 4 – Transition spool in the SES pit
**CUSTOMER:** MOHR Engineering Division  
13602 Westland East Blvd

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**CALIBRATION READINGS (as left)**

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All readings within manufacturer tolerance (+/- .5% F.S.)

**CONVERSION FACTORS (Reading - Offset)*gain**

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Calibration performed per STS document PTC1001 and traceable to N.I.S.T.  
Equipment used: Pressure sensors model M3800 SN:61205, Data Precision model 3600 Sn:1685  

Technician: L. Wilson  
DATE: June 8, 2009  
SIGNED: [Signature]  
RECALL: June 8, 2010

Figure 5 – Calibration certificate for upstream pressure transducer
Figure 6 – Calibration certificate for downstream pressure transducer
Hydrotest of Valve and Transition Spools

Project Number: PN1601045
Project Manager: Brent Vyvial
Client: BP
Testing Date: May 7 & 8, 2010

Project Description
This project involves hydrotesting of a valve spool and transition spool.

SES activities associated with the test program include the following tasks:
1. Make-up flanges on spools (BP to provide necessary bolts, nuts, gaskets, etc; SES to provide torque equipment required to make-up flanges).
2. Fill assembly with tap water.
3. Place the test assembly in the test pit and connect necessary hardware and instrumentation.
4. Conduct pressure tests on valve spool and transition spool.

Equipment Needed
- SES Waller burst pit
- Pump
- One (1) analog pressure dial pressure gauge
- 10,000 psi pressure transducer
- DAQ system to record internal pressure

Safety Procedures
The following steps will be carried out to ensure the safety of SES lab personnel, guests, and equipment.
- Cover on test pit

Note: Valve spool will be tested prior to testing of transition spool. Hydrotest spool to remain made-up to transition spool for shipment.
Transition Spool Test Procedure

Pre-testing procedures:
1. Make-up flanges on transition spool (Figure 1). Torque G-Flange bolts to 17,000 ft-lbs (per BP). Torque 18 ¾ in, 6BX, 15k flange bolts to 7,047 ft-lbs. Vetco will supply technician to assemble flanges.
2. Fill assembly with tap water and place assembly in test pit.
3. Connect pressure lines and instrumentation.
4. Place lids on pit.

Testing procedures:
1. Start the data acquisition system in order to record data. Record data at 1 scan per second.
2. Increase internal pressure to 7,500 psi at a rate not to exceed 25 psi/sec.
3. Once pressure stabilizes, hold for 30 minutes.
4. Remove internal pressure.
5. Stop data acquisition system.

Post-testing procedures:
1. Drain water.

Valve Spool Test Procedure

Pre-testing procedures:
1. Make-up flanges on valve spool (Figure 2). Torque G-Flange bolts to 17,000 ft-lbs (per BP). Torque 18 ¾ in, 6BX, 15k flange bolts to 7,047 ft-lbs. Vetco will supply technician to assemble flanges.
2. Position valve partially open.
3. Fill assembly with tap water and place assembly in test pit.
4. Connect pressure lines and instrumentation.
5. Place lids on pit.

Testing procedures:
1. Start the data acquisition system in order to record data. Record data at 1 scan per second.
2. Increase internal pressure to 7,500 psi at a rate not to exceed 25 psi/sec.
3. Once pressure stabilizes, hold pressure for 30 minutes.
4. Remove internal pressure.
5. Close valve.
6. Increase internal pressure on downstream side of valve to 6,670 psi at a rate not to exceed 25 psi/sec.
7. Hold pressure for 15 minutes.
8. Remove internal pressure.
10. Repeat steps 5-9.

Post-testing procedures:
1. Drain water.
2. Remove flanges.
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**Notes:**
- Geeks and dims good @ 4/15/08
- TREX-120129.483 HIGHLY CONFIDENTIAL LNL004-026462
RISER JT-DRLG PART, HMF-G, PIN-ROUGHOUT, R1. COUPLING

NOM. 21.000, CLASS G
(PURCHASE PER DRAWING)

AI-MATERIAL: PER VGS 5.110.3.1,
(FORGED, ROUGH-OUT TO PRINT PRIOR TO HEAT TREATMENT)

DIMENSION CATEGORIES-
CATEGORY 3: ALL DIMENSIONS
WEIGHT: 2155 LBS

E1-IDENTIFICATION: ON OD OF FLANGE
MARK P/N H224960-1 (REV.)
HEAT NO.
SPECIFICATION REVISION REFERENCE: REVISION VGS5.110.3 4130 MODIFIED LOW ALLOY STEEL 9

END OF SPECIFICATION
Material Test Report No. 302438

Customer: VETCO GRAY INC.

Spec: VGS 5.110.3.1 Rev. 9

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**Description**
CONTOUR RING
42.500" X 18.875" X 14.188"
MELTING PRACTICE: E. F. V. D. (ELECTRIC FURNACE VACUUM DEG.)

**HEAT: C1667 (9/12) PCS**

**CHEMICAL ANALYSIS**

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**HEAT TREATMENT**

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<th>Heat Treatment</th>
<th>Temperature (°F)</th>
<th>Time at Temp (Hrs)</th>
<th>Cooling Medium</th>
<th>Quench H. Temp. (°F)</th>
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<tr>
<td>9 PCS</td>
<td>162598</td>
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<td>1897</td>
<td>4.00</td>
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<td>9 PCS</td>
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**APPROVED**

INSPECTOR #10

SEP 18 2008

QUALITY ASSURANCE

TREX-120129.485

HIGHLY CONFIDENTIAL

LNL004-026464
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<td>H224960-1 Rev.C</td>
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Description: CONTOUR RING 42.500" X 18.875" X 14.188" MELTING PRACTICE E. F. V. D. (ELECTRIC FURNACE VACUUM DEG.)

TEST SPECIMEN AND PCS WERE HEAT TREAT TOGETHER
FORGING RATIO: 3.5 TO 1
TEST SPECIMEN REMOVED FROM A SACRIFICIAL PROD. PART AFTER HEAT TREAT
IMPACT TEST: 0.39" X .39" X 2.18" CVN SPECIMEN
TENSILE TEST: 0.500" DIAMETER SPECIMEN
ULTRASONIC EXAMINATION PER: VGS 8.1.2 S1A Rev. 10
IN ACCORDANCE WITH: DIN 50049.3.1B 04/92
FURNACE CALIBRATED PER: API 6A APPENDIX H 17TH EDITION
HARDNESS ON TEST SPECIMEN IS REPRESENTED BY TWO VALUES.
THICKNESS USED FOR HEAT TREATMENT TIMES 0 IN
FURNACE TEMP. MONITORING AND RECORDING METHOD: AIR THERMOCOUPLE
THE SPECIMEN FOR MECHANICALS WERE TAKEN TO PER DRAWING OR INSTRUCTION
CVN TEST SPECIMEN: LONGITUDINAL
POST-WELDING HEAT TREATMENT @ 1099 °F X 3 HRS
ROUGH MACHINE PRIOR TO HEAT TREAT
TESTING ACCORDING TO ASTM A370
NACE MR - 0175
LINE ITEM: 09
Serial Number Note:
HEAT#: C1665 SN: 3,4,14
HEAT#: C1987 SN: 5,6,7,8,9,10,11,13,15
MPP-0708-05 Rev.7

APPROVED
SEP 18 2008
INSPECTOR #10
A-46530
A-I

Arturo Leos Texas
QUALITY ASSURANCE

TREX-120129.486
HIGHLY CONFIDENTIAL
### REPORTE GENERAL DE INSPECCION MAQUINADOS

**No. de OP:** 261204  
**Piezas de la OP:** 24  
**Fecha:** 19/04/2008  
**Lote:**  
**Colada:**  
**Numero de parte:** H224960-1 Rev.C  
**Notas:**

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Desviación Aceptada por:  
Motivo de la desviación:  
Fecha:  
Recibió el reporte:  
Nombre y firma:  
Aprobado enero 15 2004

**Pedido:** A90903

---

**INSPREX-120129.487**

INSPECTOR #10

APPROVED  
SEP 18 2008

HIGHLY CONFIDENTIAL
## HARDNESS TEST REPORT (BHN)

**Job Order**: 261204

- **Customer**: VETCO GRAY INC.
- **PO**: A90903
- **Line Item**: 09
- **Part number**: H224960-1 Rev.C
- **Specification**: VGS 5.110.3.1 Rev. 9
- **Scope**: M-1
- **Date**: 17/04/2008

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### Comments:
- H1 Equipo: B-4, H2 Equipo: B-4, H3 Equipo: B-4

### Inspector:
- SPONCE

---

**INSPECTOR #10**

![Signature]

**APPROVED**

SER-1 8 2008

LN: PS507, PS265

---

**Desviacion:**

---

**TREX-120129.488**

HIGHLY CONFIDENTIAL

LNL004-026467
1.0 PURPOSE
To provide the methods for the ultrasonic inspection scanning of Riser Forgings.

2.0 GENERAL
This Scan Plan provides specific requirements for the ultrasonic inspection of riser flange forgings.

3.0 ULTRASONIC INSPECTION - SCAN DESCRIPTIONS
The following ultrasonic inspections shall be carried out in accordance with VGS 8.1.2 rev 10, S1 & scan plan in exhibit 1.

4.0 PRODUCT ACCEPTANCE

Basic Calibration Block(s). The basic calibration reflectors shall be used to establish a primary reference response of the equipment. The basic calibration reflectors may be located either in the component material or in a basic calibration block.

The DAC curve shall be constructed with at least 3 holes as follow:

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<th>FBH</th>
<th>Metal Thickness</th>
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<td>From</td>
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<tr>
<td>1/16&quot;</td>
<td>0&quot;</td>
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<tr>
<td>1/8&quot;</td>
<td>Greater than 1.5&quot;</td>
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<tr>
<td>¼&quot;</td>
<td>Over 6&quot;</td>
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Exhibit 1

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<tr>
<th>Scan #</th>
<th>Scan Surface</th>
<th>Calibration</th>
<th>Transducer</th>
<th>Frequency MHz</th>
<th>Size</th>
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<tr>
<td>Scan 1</td>
<td>Outside Diameter A</td>
<td>1/4&quot; FBH DAC</td>
<td>Normal</td>
<td>2.25 - 5</td>
<td>¼&quot; or 1&quot;</td>
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<tr>
<td>Scan 2</td>
<td>Flat End (Face B)</td>
<td>1/8&quot; FBH DAC</td>
<td>Normal</td>
<td>2.25 - 5</td>
<td>¼&quot; or 1&quot;</td>
</tr>
<tr>
<td>Scan 3</td>
<td>Flat End (Face C)</td>
<td>1/8&quot; FBH DAC</td>
<td>Dual</td>
<td>2.25  or 5.0</td>
<td>½&quot; x ¾&quot;</td>
</tr>
<tr>
<td>Scan 4</td>
<td>Inside Diameter D</td>
<td>1/8&quot; FBH DAC</td>
<td>Dual</td>
<td>2.25  or 5.0</td>
<td>½&quot; x ¾&quot;</td>
</tr>
<tr>
<td>Scan 5</td>
<td>Inside Diameter D Circumferential scanning, clockwise &amp; counterclockwise direction</td>
<td>Notch 3% thickness</td>
<td>Angular 45° Wedge</td>
<td>2.25  5</td>
<td>1&quot; x 1&quot; 5/8&quot; x ¾&quot;</td>
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<tr>
<td>Scan 6</td>
<td>Inside Diameter D Axial scanning in two opposite directions</td>
<td>Notch 3% thickness</td>
<td>Angular 45° Wedge</td>
<td>2.25  5</td>
<td>1&quot; x 1&quot; 5/8&quot; x ¾&quot;</td>
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Note: The angle shoe wedge shall be contoured to the surface of the part as necessary.
Frisa SCAN PLAN

Date: May 10th, 2007
Scan Plan No.: UT-S-06
Rev. A: May 10th, 2007

Title: ULTRASONIC INSPECTION SCAN PLAN FOR
HMF Flange Forgings - Connectors

Page: 2 of 2

---

Scan 1
Zone A

Scan 2
Zone B

Scan 3
Zone C

Scan 4
Zone D

Scan 5
Zone D

Scan 6
Zone D

---

Roy Delgatty
Vetco Gray Inc. Corporate NDE Level III, SCWI
APPROVED
SEP 18, 2008
INSPECTOR #10
H-46530
A-I

PREPARED BY
Samuel Kono Asaid
Laboratory Chief
ASNT Level II

---

TREX-120129.490
HIGHLY CONFIDENTIAL
LNL004-025469
ULTRASONIC TEST REPORT

Certificate: 20884

IDENTIFICATION PART

Job Order: 261204
Material: VGS S.110.3.1 Rev. 9
Specification: VGS 8.1.2.5IA Rev. 10

Dimensions: 42.350" x 18.75" x 14.188"
Heat: C1665 Qty: 3 PCS
Heat: C1667 Qty: 9 PCS

Description: CONTOUR RING

EQUIPMENT

Brand: Sonatest MasterScan Model: MasterScan Serial No: 1000613 Calibration Due: OCT/3/2008

STRAIGHT BEAM (LONGITUDINAL WAVE)

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<tr>
<td>Couplant</td>
<td>GEL</td>
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<tr>
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ANGLE BEAM (SHEAR WAVE)

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EVALUATION: Accepted

REMARKS: No detectable indications

Evaluated by: [Signature]

FRISA FORJADOS S.A. DE C.V.
OPEN DIE & ROLLED RING FORGINGS
PO Box 1273 Monterrey, NL
64000 Mexico Ph. (52 8) 153-0321 Fax (52 8) 336-3560
e-mail: frisa@frisa.com

APPROVED
SEP 1 8 2008

INSPECTOR #10
N-46580 A-I

TREX-120129.491

HIGHLY CONFIDENTIAL
LNL004-026470
HARDNESS TEST REPORT (BHN)

Job Order 261204

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<td>Specification: VGS 5.110.3.1 Rev. 9</td>
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<td>Procedure: Respetar Marcaje</td>
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APPROVED
SEP 18 2008
INSPECTOR #10
A-1

LN: PS507, PS265
Comments:
H1 Equipo: B-4, H2 Equipo: B-4, H3 Equipo: B-4

Desviación:

Inspector: SPONCE

TREX-120129.492

HIGHLY CONFIDENTIAL
LNL004-026471
DR. DAVID ALEJANDRO MARTINEZ FERNANDEZ
MEDICO CIRUJANO OFTALMOLOGO
CED. PROF: 2370336

REPORTE OFTALMOLOGICO:

NOMBRE: JOSE GUADALUPE HERNANDEZ
EDAD: 43 AÑOS
EMPRESA: FRISA FORJADOS

AGUDEZA VISUAL: O.D. NEUTRO 20/20
O.I. NEUTRO 20/20

VISION CROMATICA: NORMAL
JAEGER 1-2 A 12" DISTANCIA: NORMAL
FONDO DE OJO: NORMAL
ESTEREAGUDEZA: NORMAL
EXAMEN OFTALMOLOGICO: DENTRO DE LIMITES NORMALES

DIAGNOSTICO: EMETROPIA
PLAN: CITA ANUAL

OBSERVACIONES: CUMPLE CON REQUERIMIENTOS DE INSTRUCCION PARA NORMA PERSONAL-1-05

MONTERREY, N. L. A 24 DE JULIO DEL 2006

DR. DAVID ALEJANDRO MARTINEZ FERNANDEZ

APPROVED
SEP 18 2008
INSPECTOR #10
N-46030
A-I

MATRIZ:
AV PADRE MIER #202 PTE. CENTRO
MONTERREY, N. L. C. P. 64000
TEL. 8340-3212

CLINICA MONTERREY:
AV PADRE MIER #119 PTE.
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SUCursal ANAHUAC:
AV UNIVERSIDAS #326 PTE.
SAN NICOLAS DE LOS GARZA, N. L.
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AV EUGENIO GARZA SADA #2656 SUR
MONTERREY, N. L. C. P. 64440
TEL: 8337-0230 FAX 8337-0791

TREX-120129.493
HIGHLY CONFIDENTIAL LNL004-026472
SOLD TO: BP EXPLORATION
C/O PRICE WATERHOUSE COOPER
PO BOX 22024
TULSA, OK 74121

SHIP TO: ACUTE TECHNOLOGIES
11925 BRITTMORE PARK DRIVE
HOUSTON, TX 77041

ITEM CODE | QTY | HEAT CODE | DESCRIPTION | MATERIAL GRADE / HEAT TREAT
---|---|---|---|---
12831 | 1.00 | H3074 | 18.75" API 5K WN (BX) 15.5 ID | AISI 4130 API 75K
NORM @ 1650°F/32 HRS/AIR COOLED
QUENCH @ 1600°F/21 HRS/AIR COOLED
WATER TEMP INT 76°F/OUT 82°F
TEMPER @ 1200°F/20 HRS/AIR COOLED

CHEMICAL ANALYSIS
HEAT CODE | C | Mn | P | S | Si | Cu | Ni | Cr | N | Moly | V | Cb | Nb | Ti | Al | B | Sn
---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---
H3074 | 0.310 | 0.570 | 0.009 | 0.012 | 0.260 | 0.210 | 0.170 | 1.040 | 0.230 | 0.022 | 0.005 | 0.003 | 0.019 | 0.010

MECHANICAL PROPERTIES
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---|---|---|---|---|---|---
H3074 | .016/.015/.016 | .500 | 4 X 4 X 8 | 3.1:1 | SOREL | United States of America

COMMENTS
PROJECT: BP MC 252 INCIDENT
OTHER CHEMICALS: AS: 0.008; SB: 0.006; CA: 0.0002
CVN: LONGITUDINAL, MELT PRACTICE: E,P,V; GRAIN SIZE: 7
NO WELD REPAIR

WE CERTIFY THIS MATERIAL MEETS THE REQUIREMENTS OF NACE MR-0175/ISO 15156-1 DATED FEB 1, 2005.
WE CERTIFY THIS MATERIAL MEETS THE REQUIREMENTS IN ACCORDANCE TO EN10204-3.1.

HIGHLY CONFIDENTIAL

TREX-120129.494
**ITEM CODE** | **QTY** | **HEAT CODE** | **DESCRIPTION** | **MATERIAL GRADE / HEAT TREAT**
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QUENCH @1600°F/21 HRS/AIR COOLED
WATER TEMP IN/OUT/UP/FP:
TEMP @1200°F/20 HRS/AIR COOLED

**CHEMICAL ANALYSIS**

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**COMMENTS**

PROJECT: BP MC 252 INCIDENT
OTHER CHEMICALS: AS: 0.028; SB: 0.008; CA: 0.0002
CVN: Longitudinal, Melt Practice: EF-VD; Grain Size: 7
NO WELD REPAIR

WE CERTIFY THIS MATERIAL MEETS THE REQUIREMENTS OF NACE
MR-0175/ISO 15156-1 DATED FEB 1, 2005.
WE CERTIFY THIS MATERIAL MEETS THE REQUIREMENTS IN
ACCORDANCE TO EN10204-3.1.

HIGHLY CONFIDENTIAL

LNL004-026474

**AUTHORIZED SIGNATURE**

**TREX-120129.495**
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**CHEMICAL ANALYSIS**

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Unless otherwise stated, yield stress is 0.2% offset, gauge length is 2” for 1/2” bars or 1” for 1/4” bars. Unless otherwise stated, Charpy Impact specimens are V-notch 10 x 10 mm.

**COMMENTS**

WE CERTIFY THIS MATERIAL MEETS THE REQUIREMENTS OF NACE MR-0175/ISO 15166-1 DATED FEB 1, 2005. WE CERTIFY THIS MATERIAL MEETS THE REQUIREMENTS IN ACCORDANCE TO EN10204-3.1.

HIGHLY CONFIDENTIAL

LNL004-026475

TREX-120129.496
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Unless otherwise stated, yield stress is 0.2% offset. Gage length is 2" for 1/2" bars or 1" for 1/4" bars. Unless otherwise stated, Charpy Impact specimens are V-notch 10 x 10 mm.

Comments: 

**AUTHORIZED SIGNATURE**

We certify this material meets the requirements of NACE MR-0175/ISO 15156-1 DATED FEB 1, 2005. We certify this material meets the requirements in accordance to EN10204-3.1.

Highly Confidential
1. INTRODUCTION

This technical note presents the results of three different analyses of a spool piece that is to be fitted to the G-Series flange of the flex-joint of the GoM Horizon BOP stack.

The G-Series flange was originally designed to resist three million pounds-force of axial tension loading and 5,000psi differential pressure. In the current application, the differential pressure is increased to 6,750psi and the external axial loading is a modest compressive deadweight load due to installed equipment. However, the potential exists for this deadweight to be installed at a small angle to the vertical and this introduces a bending moment at the G-Series flange. The purpose of this assessment is to determine the moment capacity of the flange in the presence of the 6,750psi differential pressure and the deadweight.

The calculations reported are as follows:

2. Finite element assessment based on an ABAQUS model developed by Frazer-Nash.
3. Finite element assessment based on an ANSYS model originally developed by Ansys Inc. and subsequently modified by Frazer-Nash.
2. TRADITIONAL HAND CALCULATIONS

The following is a summary of scoping calculations carried out on the Transition Spool Piece Pipe, G Series Flange Bolts and Pipe Section Below the G Series Flange. The calculations are attached to this report. Note that no design / safety factors have been included in the calculations.

The design pressure of the system is assumed to be 9,000psi internal and a hydrostatic pressure of 2,250psi, giving a differential pressure of 6,750psi. The design basis for bending assumes the Triple Ram Stack fitted with the Gate Valve Assembly, which corresponds to a bending moment of 590kip-ft.

2.1 HOOP BURSTING CAPACITY

The hoop bursting capacity of the pipes has been estimated below as 8,211psi. A reserve factor of 1.22 therefore exists on the design differential pressure of 6,750psi.

\[ \sigma_p \ln \left( \frac{OD}{ID} \right) = 84,100 \times \ln \left( \frac{21.5}{19.5} \right) = 8,211 \text{psi} \]

2.2 AXIAL COLLAPSE

Axial plastic collapse of the pipes due to combined axial differential pressure stress plus bending moment has been estimated using a Miller formula (Reference 1). The following collapse conditions have been calculated:

- Differential pressure = 6,750psi, Bending moment = 4.0 x 590 kip-ft (i.e. a margin of 4.0 on bending moment, differential pressure constant).
- Differential pressure = 2.1 x 6,750psi, Bending moment = 590 kip-ft (i.e. a margin of 2.1 on differential pressure, bending moment constant).
- Differential pressure = 1.9 x 6,750psi, Bending moment = 2 x 590 kip-ft (i.e. a margin of 1.9 on differential pressure and bending moment).

The dominant loading is the differential pressure, and it is clear that hoop bursting is the limiting factor.

2.3 G-SERIES FLANGE BOLTS

The yield capacity of one bolt (assuming a yield of 130ksi) is estimated to be 1,170kips. The design pre-load (based on a total for six bolts of 3,750kips) is about half this (625kips). The load in one bolt due to a differential pressure of 6,750psi is 342kips. The load in one bolt due to a bending moment of 590kip-ft is 148kips. One can add conservatively together the pre-load, pressure load and moment load (1,115kips) and still be just below the yield capacity of the bolts (1,170kips). This takes no account of the fact that the bolts will not carry all of the pressure and moment loads because of the pre-loading of the bolts and flanges.
The bearing stresses under the bolt heads are high but below the likely bearing strength of the flanges, taken as 1.5 x yield strength.

2.4 BUCKLING

Buckling of the transition spool piece pipe was considered using API RP 1111, Section 4.3.2.2 (Reference 4). The following conditions were considered:

- An external seawater pressure of 2,250psi.
- An internal pressure of 675psi, corresponding to a flowing mixture density of 300kg/m³ in an attached riser to the surface.
- A bending moment of 590kip-ft.
- A 2% out-of-roundness.

The transition spool piece pipe is not seamless and therefore a collapse factor of 0.6 was used. A margin of 1.27 was obtained, which is dominated by the external pressure loading.

2.5 BOLT PRELOADS AND TIGHTENING TORQUES

Hand calculations have been carried out based on British Standard 3580:1964 (Reference 5) in order to investigate how the relationship between bolt torque and the resulting preload varies with the coefficient of friction between the mating threads and between the nuts and the flanges.

With a coefficient of friction for Never-Seez lubricant assumed to be 0.085 the required torque to achieve a preload of 625 kips per bolt is calculated to be 20,098 lbf-ft. Reference 6 provides an assumed coefficient of friction for Never-Seez of 0.085 and states that the required tightening torque for Class G H141522-2 bolt with Never-Seez is 20,500 lbf-ft. These values are in good agreement, increasing confidence in the calculations.

Figure 1 shows the tightening torque required to produce a bolt preload of 625 kips at various coefficients of friction.

Figure 2 shows the preload achieved given a tightening torque of 20,500 lbf-ft at various coefficients of friction.

2.6 SHEAR FORCE

The bending moment considered in the calculations above is due to the offset of the deadweight of the assembly due to the angle of the BOP stack. In addition to the bending moment and compressive force a shear component will be developed. Assuming an angle of tilt of 3 degrees and a compressive force of 150kips the shear stress developed in the spool piece is calculated to be 0.12ksi. This shear stress is not considered to be significant and the shear force was therefore omitted from the finite element analyses presented in Section 2.
3. ABAQUS FINITE ELEMENT MODEL

A finite element model of the transition spool, bolts, lower flange and a section of the riser adapter has been developed for analysis using the ABAQUS finite element software. The geometry was represented using 250,000 second-order modified tetrahedral elements (C3D10M). To ensure an appropriate representation of global and local bending stresses, a mesh density with at least three elements through the wall thickness of the pipes was used.

Views of the model mesh are shown in Figure 3 and Figure 4.

3.1 GEOMETRY

The geometry of the spool and bolts was taken from a SAT CAD file provided by BP, Reference 2. It should be noted that Frazer-Nash has not checked in detail the geometry supplied. However, it was found that the bolt heads in this geometry were too small and they were therefore increased to 5.25" outside diameter (Reference 3).

The following modifications were made to the lower riser adapter and flange section:

- The outer diameter of the lower pipe was modified to be 22 7/8"
- The wall thickness of the lower pipe was increased to 1.22".
- The lower pipe was cut to form the model boundary where the riser adapter section expands at 25° beneath the top face of the flange.

The contribution of the sealing pin in terms of pressure containment has conservatively been ignored, it is not included in the model.

3.2 MATERIAL PROPERTIES

The assumed material properties are summarised in the following table:

<table>
<thead>
<tr>
<th>Component</th>
<th>Material designation</th>
<th>Young’s Modulus</th>
<th>Poisson’s Ratio</th>
<th>Yield Strength</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spool</td>
<td>X-80</td>
<td>29000ksi</td>
<td>0.29</td>
<td>84.1ksi</td>
</tr>
<tr>
<td>Lower pipe and flange</td>
<td>AISI 4130</td>
<td>29000ksi</td>
<td>0.29</td>
<td>80ksi</td>
</tr>
<tr>
<td>Bolts</td>
<td>AISI 4340 modified</td>
<td>29000ksi</td>
<td>0.29</td>
<td>130ksi</td>
</tr>
</tbody>
</table>

3.3 BOUNDARY CONDITIONS

At the lower model boundary, the cut pipe section was constrained to prevent axial and circumferential translations. Radial translations were permitted, so as not to constrain hoop expansion of the pipe under differential pressure loading.

At the upper end of the model, a constraint was used to tie all nodes on the top surface of the upper spool flange to a single node at the centre of the pipe. The constraint ensures the flange remains planar, whilst allowing radial translations. The applied end cap and moment loads are applied to a single node in the centre of the flange top surface.
Contact was modelled between the contacting surfaces at the bolt to upper flange interface, upper flange to lower flange interface and the bolt to lower flange interface. A coefficient of friction of 0.3 was assumed. The applied boundary conditions and constraints are shown in Figure 5 and Figure 6 respectively.

3.4 LOADCASES

3.4.1 Linear Elastic Analyses

The bolt preload was applied by defining a coefficient of thermal expansion for the bolts and cooling them down over a load increment until sufficient bolt pre-load was developed.

The end cap and moment loads were applied to the nodal constraint at the top surface of the spool upper flange. The end cap load applied has been calculated based upon an internal pressure of 9,000psi and an external pressure of 2,250psi. The cross-section of the spool pipe was then used to determine the end cap force. A compressive dead weight load of 150kips was assumed to act downwards in an opposite sense to the end cap pressure loading on the top of the spool piece. The resulting upwards axial load is 1,721 kips, which corresponds to the pressure end cap load of 1,871 kips minus the 150 kips deadweight. The differential pressure was applied to all internal surfaces. The following table summarises the load cases considered:

<table>
<thead>
<tr>
<th>Load Case</th>
<th>Bolt Preload (per bolt) (kips)</th>
<th>Moment Load (kip-ft)</th>
<th>Differential Pressure (ksi)</th>
<th>End Load (kips)</th>
<th>Number of bolts</th>
<th>Orientation of moment load (axis of rotation)</th>
<th>Comment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base Case</td>
<td>625</td>
<td>590</td>
<td>6.75</td>
<td>1721</td>
<td>6</td>
<td>x</td>
<td>Preload reduced by 25%</td>
</tr>
<tr>
<td>Sensitivity 1</td>
<td>469</td>
<td>590</td>
<td>6.75</td>
<td>1721</td>
<td>6</td>
<td>x</td>
<td>Moment increased by 50%</td>
</tr>
<tr>
<td>Sensitivity 2</td>
<td>625</td>
<td>885</td>
<td>6.75</td>
<td>1721</td>
<td>6</td>
<td>x</td>
<td>One bolt at extremity on the tensile side removed</td>
</tr>
<tr>
<td>Sensitivity 3</td>
<td>625</td>
<td>590</td>
<td>6.75</td>
<td>1721</td>
<td>5</td>
<td>x</td>
<td>Moment load applied in different orientation</td>
</tr>
<tr>
<td>Sensitivity 4</td>
<td>625</td>
<td>590</td>
<td>6.75</td>
<td>1721</td>
<td>6</td>
<td>y</td>
<td>Moment load applied in different orientation. One bolt at extremity on the tensile side removed</td>
</tr>
<tr>
<td>Sensitivity 5</td>
<td>625</td>
<td>590</td>
<td>6.75</td>
<td>1721</td>
<td>5</td>
<td>y</td>
<td>Moment load applied in different orientation. One bolt at extremity on the tensile side removed</td>
</tr>
<tr>
<td>Sensitivity 6</td>
<td>625</td>
<td>330</td>
<td>6.75</td>
<td>1721</td>
<td>6</td>
<td>x</td>
<td>Moment reduced to 330kip-ft</td>
</tr>
<tr>
<td>Sensitivity 7</td>
<td>625</td>
<td>590</td>
<td>6.75</td>
<td>1721</td>
<td>5</td>
<td>y</td>
<td>Moment load applied in different orientation. One bolt near neutral axis on the tensile side removed</td>
</tr>
</tbody>
</table>

Hand calculations indicate that, based upon the assumption of a rigid flange, the worst orientation of the moment load with respect to the load transferred by the bolts is given by a moment load applied about the y axis (sensitivity load cases 4 and 5). However the difference in bolt loads due to orientation was not calculated to be great and given the
expected effect of the flange stiffness it was considered appropriate to analyse both orientation cases.

### 3.4.2 Ultimate Failure Analyses

Six elastic-plastic collapse analyses have been run using the same model as described above to enable a pressure vs. moment collapse envelope to be produced. A seventh elastic plastic collapse analysis has been analysed to consider the collapse load in the event that the vessel was forced to drive off with the riser still attached. In this case a fixed design pressure and moment has been considered with an increasing end cap force.

The material models used have been modified to have elastic-perfectly plastic behaviour using the properties defined in Section 2.2.

The loads and boundary conditions are applied in the same manner as for the linear elastic analyses. The loads applied are summarised below:

<table>
<thead>
<tr>
<th>Load Case</th>
<th>Bolt Preload (per bolt) (kips)</th>
<th>Moment Load (kip-ft)</th>
<th>Differential Pressure (ksi)</th>
<th>End Load (kips)</th>
<th>No. of bolts</th>
<th>Orientation of moment load (axis of rotation)</th>
<th>Comment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Collapse 1</td>
<td>625</td>
<td>Scaled</td>
<td>6.75</td>
<td>1721</td>
<td>6</td>
<td>x</td>
<td>Design pressure, increasing moment</td>
</tr>
<tr>
<td>Collapse 2</td>
<td>625</td>
<td>330</td>
<td>Scaled</td>
<td>150 constant compressive + Scaled</td>
<td>6</td>
<td>x</td>
<td>Design moment, increasing pressure</td>
</tr>
<tr>
<td>Collapse 3</td>
<td>625</td>
<td>None</td>
<td>Scaled</td>
<td>150 constant compressive + Scaled</td>
<td>6</td>
<td>x</td>
<td>No moment, increasing pressure</td>
</tr>
<tr>
<td>Collapse 4</td>
<td>625</td>
<td>Scaled</td>
<td>4.5</td>
<td>1049</td>
<td>6</td>
<td>x</td>
<td>75% design pressure, increasing moment</td>
</tr>
<tr>
<td>Collapse 5</td>
<td>625</td>
<td>Scaled</td>
<td>2.25</td>
<td>377</td>
<td>6</td>
<td>x</td>
<td>50% design pressure, increasing moment</td>
</tr>
<tr>
<td>Collapse 6</td>
<td>625</td>
<td>Scaled</td>
<td>8.5</td>
<td>2244</td>
<td>6</td>
<td>x</td>
<td>119% design pressure, increasing moment</td>
</tr>
<tr>
<td>Collapse 7</td>
<td>625</td>
<td>330</td>
<td>6.75</td>
<td>1721 constant tensile +Scaled</td>
<td>6</td>
<td>x</td>
<td>Design pressure, and moment, increasing axial force</td>
</tr>
</tbody>
</table>

(Note: 'Scaled' indicates that the load application is increased automatically within the analysis until collapse occurs. In the case of the pressure load being scaled, the differential pressure and its associated end cap load are both increased in proportion.)
3.5 RESULTS

3.5.1 Linear Elastic Analyses

The analysis results are shown in the following figures. In all plots the deformations have been scaled up significantly so as to be visible.

- For the base case analysis, Von Mises stresses for the full model, spool, lower flange and bolts are shown in Figure 7, Figure 8, Figure 9 and Figure 10 respectively.

- For sensitivity case 1, Von Mises stresses for the full model, spool, lower flange and bolts are shown in Figure 11, Figure 12, Figure 13 and Figure 14 respectively.

- For sensitivity case 2, Von Mises stresses for the full model, spool, lower flange and bolts are shown in Figure 15, Figure 16, Figure 17 and Figure 18 respectively.

- For sensitivity case 3, Von Mises stresses for the full model, spool, lower flange and bolts are shown in Figure 19, Figure 20, Figure 21 and Figure 22 respectively.

- For sensitivity case 4, Von Mises stresses for the full model, spool, lower flange and bolts are shown in Figure 23, Figure 24, Figure 25 and Figure 26 respectively.

- For sensitivity case 5, Von Mises stresses for the full model, spool, lower flange and bolts are shown in Figure 27, Figure 28, Figure 29 and Figure 30 respectively.

- For sensitivity case 6, Von Mises stresses for the full model, spool, lower flange and bolts are shown in Figure 31, Figure 32, Figure 33 and Figure 34 respectively.

- For sensitivity case 7, Von Mises stresses for the full model, spool, lower flange and bolts are shown in Figure 35, Figure 36, Figure 37 and Figure 38 respectively.

It can be noted that the highest stresses occur at stress concentrations and the bearing faces between the bolts and the flanges. However, away from these regions, the stresses are significantly lower. To determine the maximum stresses in the bolt shank, the spool pipe region and the lower flange, the maximum stresses in the regions shown in Figure 39, Figure 40 and Figure 41 respectively were obtained. These are presented in the table below:
<table>
<thead>
<tr>
<th>Loadcase</th>
<th>Description</th>
<th>Peak stress in bolt shank (ksi)</th>
<th>Peak stress in spool (away from flange) (ksi)</th>
<th>Peak stress in lower pipe (away from flange) (ksi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base Case</td>
<td></td>
<td>130</td>
<td>84.1</td>
<td>80</td>
</tr>
<tr>
<td>Sensitivity 1</td>
<td>Preload reduced by 25%</td>
<td>107.3</td>
<td>73.9</td>
<td>65.9</td>
</tr>
<tr>
<td>Sensitivity 2</td>
<td>Moment increased by 50% to 885kip-ft</td>
<td>117.6</td>
<td>81.2</td>
<td>72.9</td>
</tr>
<tr>
<td>Sensitivity 3</td>
<td>One bolt at extremity on the tensile side removed</td>
<td>165.6</td>
<td>89.6</td>
<td>81.7</td>
</tr>
<tr>
<td>Sensitivity 4</td>
<td>Moment load applied in different orientation</td>
<td>108.3</td>
<td>74.0</td>
<td>67.4</td>
</tr>
<tr>
<td>Sensitivity 5</td>
<td>Moment load applied in different orientation. One bolt at extremity on the tensile side removed</td>
<td>160.8</td>
<td>89.2</td>
<td>74.3</td>
</tr>
<tr>
<td>Sensitivity 6</td>
<td>Moment reduced to 330kip-ft</td>
<td>99.0</td>
<td>70.9</td>
<td>62.3</td>
</tr>
<tr>
<td>Sensitivity 7</td>
<td>Moment load applied in different orientation. One bolt near neutral axis on the tensile side removed</td>
<td>160.4</td>
<td>88.2</td>
<td>80.6</td>
</tr>
</tbody>
</table>

The stresses quoted for the bolt shanks include any bending components present. For comparison, the pre-load only stress in the bolt shanks is 70ksi.

The results show that the applicable yield stresses are exceeded for sensitivity cases 3, 5 and 7 in both the bolt shanks and the spool. All of these cases assume the loss of a single bolt which would have been loaded in tension.

The results from the FEA indicate that bearing stresses under the bolt heads are high. A simple calculation can be undertaken to investigate whether these stresses are acceptable or not.

The bolt head OD is 5.25" and the hole diameter is 3.75".

The overlapping bearing area is therefore \( \frac{\pi}{4} (5.25^2 - 3.75^2) = 10.6in^2 \)

Under the base load case, the maximum stress in a bolt shank is 107.3ksi. The bolt shank has an area of 9in\(^2\) and therefore this is equivalent to a bolt load of 966kips.

The bearing stress under the bolt head is therefore \( \frac{966}{10.6} = 91ksi \)

For steels, typical bearing strengths are about 1.5 times the yield strength. Taking a yield strength of the flanges of 80ksi, the reserve factor for the base load case is therefore:
\[
\frac{1.5 \times 80}{91} = 1.32
\]

The bearing reserve factors in the bolts themselves will be much higher because the yield strength of the bolts is much higher than for the flanges.

It can be seen that the limiting bolt load for a bearing reserve factor of one is \(1.32 \times 107.3 = 141.6\text{ksi}\), with no design / safety factor applied. This is larger than the bolt loads for all the load cases considered except the ones with missing bolts, and larger than the bolt yield of 130ksi. The bolt shank would therefore just yield before a bearing failure occurred in the flanges.

### 3.5.2 Ultimate Failure Analyses

The results for the collapse analyses carried out are summarised in the following table. The FE derived pressure vs. moment collapse envelope is shown in Figure 42. For comparison a pressure vs. moment curve has also been derived from the hand calculations presented in Section 2.1 and 2.2. These consider axial collapse (combined internal pressure and bending moment) and hoop burst (only internal pressure).

<table>
<thead>
<tr>
<th>Load Case</th>
<th>Description</th>
<th>Moment Load at Collapse (kip-ft)</th>
<th>Differential Pressure at Collapse (ksi)</th>
<th>Collapse Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>Collapse 1</td>
<td>Design pressure, increasing moment</td>
<td>(7.27*330=) 2399</td>
<td>6.75</td>
<td>See Figure 43</td>
</tr>
<tr>
<td>Collapse 2</td>
<td>Design moment, increasing pressure</td>
<td>330</td>
<td>(1.39*6.75=) 9.38</td>
<td>See Figure 44</td>
</tr>
<tr>
<td>Collapse 3</td>
<td>No moment, increasing pressure</td>
<td>None</td>
<td>(1.40*6.75=) 9.45</td>
<td>See Figure 45</td>
</tr>
<tr>
<td>Collapse 4</td>
<td>75% design pressure, increasing moment</td>
<td>(8.15*330=) 2890</td>
<td>4.5</td>
<td>See Figure 46</td>
</tr>
<tr>
<td>Collapse 5</td>
<td>50% design pressure, increasing moment</td>
<td>(8.95*330=) 2954</td>
<td>2.25</td>
<td>See Figure 47</td>
</tr>
<tr>
<td>Collapse 6</td>
<td>119% design pressure, increasing moment</td>
<td>(4.51*330=) 1488</td>
<td>8.5</td>
<td>See Figure 48</td>
</tr>
</tbody>
</table>

(Note: The figures referenced above show equivalent plastic strain at collapse, note that the deformations have been scaled up significantly so as to be visible).

The applied moments versus rotation angles at the top of the spool piece are plotted for Collapse Cases 1, 4, 5 and 6 in Figure 49, Figure 50, Figure 51 and Figure 52. Collapse Case 2 has a constant moment, and collapses at a small angle of 0.2°. Collapse Case 3 has no moment applied and therefore the rotation of the top of the spool at collapse is zero. The results from Cases 2 and 3 are similar, indicating that the design moment has little effect on collapse.

It can be seen that the results obtained agree well with the hand calculations in Section 2, increasing the confidence in the analysis. The hand calculations predict a bursting margin of 1.22, and the FEA predicts 1.40 (Collapse Case 3). The reason that the FEA predicts a
higher burst pressure than the hand calculations is because of the hoop stiffening effects of
the flanges. The hand calculations predict a collapse moment of \(4 \times 590 = 2360\text{kip-ft}\) in the
presence of the design pressure, the FEA predicts 2399kip-ft (Collapse Case 1). A simple
hand calculation of the plastic moment capacity of the spool pipe gives 2,800kip-ft, providing
further confidence in the results from the FEA.

Collapse case 7 was analysed to account for the possibility that the vessel may be forced to
drive off with the riser still attached. In this case an additional tensile load would be applied to
the riser. Constant design pressure and moment loads were assumed. The magnitude of the
additional tensile load (above that resulting from the end cap pressure load and deadweight)
at the point of collapse was found to be 3520kips. The collapse location is shown in Figure
53.
4. **ANSYS FINITE ELEMENT MODEL**

A finite element model of the spool, bolts and lower flange section was developed by ANSYS Inc. using the ANSYS finite element software. This model was passed on to Frazer-Nash Consultancy for review. During the course of the review, a number of errors in the model were identified. However, it should be noted that Frazer-Nash has not conducted a detailed check of the ANSYS Inc. model. The errors discovered were:

- The moment/boundary condition was incorrectly applied to the top of the flange.
- The differential pressure was not applied to all pressurised surfaces of the model.
- The riser adapter pipe was constrained radially, rather than permitting radial displacements, whilst constraining the pipe section to remain plane.
- The ANSYS Inc. model uses SOLID45 tetrahedral elements, which are first order tetrahedral elements and not recommended for bending problems.
- In some regions of the model, the mesh density is coarse, resulting in only a single element through the wall thickness. In these regions the model will not adequately predict bending stresses through the section. The results should therefore be viewed in this context. The model has not been re-meshed to improve accuracy by Frazer-Nash.

Except where noted, these errors were corrected and the base case loading defined in Section 2.4 was applied.

In the ANSYS Inc. model the dimensions of the lower pipe section have been assumed to be the same as those in the spool. It is understood that this is not the case and this was corrected in the ABAQUS model, but not corrected in the ANSYS Inc. model.

The results from the revised ANSYS Inc. model are shown in Figure 54 to Figure 60.

Whilst issues with the ANSYS model have been identified, the results indicate similar levels of predicted stress for the base case, providing further confidence in the results of the ABAQUS model presented in Section 2.
5. CONCLUSIONS

A range of hand calculations and finite element models have been used to assess the integrity of the G-Series flange and transition spool under combined pressure, moment and deadweight loadings.

The conclusions are as follows:

- For the base case moment load case of 590kip-ft, with all the bolts contributing as designed, the stresses in the bolts and pressure parts are below yield. The minimum margin in the pressure parts is 1.14, which occurs in the X80 transition spool. This reduces to 1.04 if the moment is increased by 50% to 885kip-ft. With the moment reduced to 330kip-ft, the margin increases to 1.19.

- Margins on the bolt yield stress range from 1.11 at 885kip-ft to 1.31 at 330kip-ft.

- The orientation of the moment relative to the bolt pattern only affects the bolt and pressure part stresses by a few percent.

- The bearing stresses in the flanges, under the bolt heads, are less than 1.5 times the yield strength of the flanges.

- The margin against buckling to API RP 1111, under a combined external pressure and moment loading, is 1.27. This assumes a worst-case scenario involving a riser to surface filled with a fluid of density 300kg/m³ and an out-of-roundness of 2%.

- Using an elastic-perfectly plastic collapse analysis and ramping up the moment, whilst holding the differential pressure constant at the design value of 6.75ksi, a collapse moment of 2399kip-ft is predicted. This represents a margin of 7.27 on the design moment of 330kip-ft.

- The margin estimated by ramping up the pressure load, whilst holding the moment constant at 330kip-ft, is 1.39 on the design differential pressure of 6.75ksi.

- Removal of any of the six bolts in either of the two moment orientations considered increases the stresses in the remaining bolts and pressure parts above yield.
6. REFERENCES


2. E-Mail from Julian Austin (BP) to Ian Bottomley (Frazer-Nash) dated 18th June 2010. Directions to Download Site ftp://11229202:kaskidar@ftp.intec.com

3. E-Mail from Pierre Beynet (BP) to Ian Bottomley (Frazer-Nash) dated 25th June 2010. "FW: Riser Flange Bolts".


6. GE Oil and Gas, Operating and Service Procedure 14000, 10 September 2009.
7. FIGURES

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Collapse 1 - 9kai Design Internal Pressure (Differential = 6.75kai), Increasing Moment

Figure 49: Collapse Case 1 - Design Pressure with Moment Increased Until Collapse
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Figure 56: Ansys model, base case, spool lower end, Von Mises Stress (ksi)
Figure 57: Ansys model, base case, lower flange and pipe, Von Mises Stress (ksi)
Figure 58: Ansys model, base case, lower flange and pipe, Von Mises Stress (ksi)
Figure 59: Ansys model, base case, bolts, Von Mises Stress (ksi)
Figure 60: Ansys model, base case, bolts, Von Mises Stress (ksi)
Sealing calcs on slope piece collapse

Look at collapse of cylindrical sections due to internal pressure and moment. Moment constant along slope piece.

Bolt C - Seepage Remnant (lower section)

22 7/8" OD

1 3/4" Wall

Mean R = 257 mm, t = 275 mm

\( \frac{t}{R} = 8.87 \) - too thick.

This is the lower part cut which the slope piece fits.
Use Miller Solution of Review of Limit Loads of Structures "3rd Ed."

Internal \( p = 6750 \text{ psi} \) (46.5 MPa)

diff

\[
p = \frac{PR}{2\sigma_y} = \frac{46.5 \times 25.7^{2.75}}{2 \times 5.7 \times 31}\]

\[
w = \cos \left( \frac{\pi P}{c} \right) = 0.37 \quad 0.399
\]

\[
m = 0.81
\]

With reference to the attached graph, allowable \( m = 0.84 \) (0.81 from formula)

\[
m = \frac{N}{N_{\text{mm}}}
\]

\[14^{2} \times 5\]

\[\text{mm}^{2} \times \text{mm}^{2} \times \text{mm}^{2}\]
\[ m = \cos \left( \frac{\pi \beta}{2} \right) \]

\[ \beta = 0.4 \]

\[ m = 0.81 \]

\[ m = 0.16 \]

\[ \beta = 0.90 \]

\[ \frac{0.90}{0.4} = 2.25 \]

**Operating Condition**

\[ RF = \frac{0.25}{0.16} = \frac{0.80}{0.4} = 2 \]

\[ \beta = 0.87 \]

**FIG. 9.5.1. Throughwall Defect Under Combined Bending and Pressure.**
\[ \beta = 0^\circ = \text{no defect} \]

Overall margin \( n \geq 2 \),

Agree.

Lee, 2.18.

\[ p = \frac{m}{\rho} \]
\[ m = \frac{63 (\pi R^3)}{2} \]
\[ \frac{m}{\rho} = \frac{330 \times 10}{6750} \]

\[ \frac{m}{\rho} = \frac{1.9 \times 10}{4 \times 283^2 \times 31 \times 547} = 0.175 \]

\[ p = 1.9 \times 10 \times 125 = 0.758 \]

\[ \frac{p}{1.9 \times 10 \times 547} = 0.144 \]

By process of iteration, the overall margin at collapse is here

\[ \frac{0.7}{0.39} = \frac{0.84}{0.44} = 1.9 \]

FIG. 9.5.1. Throughwall Defect Under Combined Bending and Pressure.

\[ \text{in collapse occurs at } 1.9 \times \beta \text{ and } 1.9 \times m. \]

TREX-120129.573

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Calc. Title: **Space PIECE COLA**

**Author:** Mike Anderson  
**Date:** 24/6/18

\[ M = 0.84 \times 4 \times 257 \times 3.1 \times 517 \]
\[ = 3.557 \times 10^7 \text{ Nm} = 3.927 \times 10^7 \text{ Nm} \]
\[ = 2620 \text{ kip-ft} = 1896 \text{ kip-ft} \]

So collapse under the proposed moment of 339 kip-ft is unlikely, agreed.

\[ RF = \frac{2620}{590} = 4.47 \]

This is an RF on the applied moment, i.e. 4.47 x The moment (plus original pressure) would be needed to collapse the pipe.

Noted also that using 6900 psi, it is possible to be ct (P_{int} = P_{int}A_0)

---

**Calc. No:** 37958  
**File Ref.**  
**Pages:** 4 of 11
**SPOOL PIECE COLLAPSE**

**Author:** Mike Anderson  
**Date:** 24/12/10  
**Approved Date:**

---

**SPOOL PIECE**

2½ OD 1 Wall

\[ \sigma_y = 84 \text{kN/m}^2 \] (579 MPa)

\[ t = 25.4 \text{mm} \]

\[ \beta = 46.5 \times 260 \degree = 0.41 \]

\[ m = \cos \left( \frac{\pi \beta}{2} \right) \]

\[ m = 0.81 \]

\[ M = 3.18 \times 10^7 \text{Nm} = 2346 \text{Kips-ft} \]

---

**RF = 2346 = 4.0**

**NB:** RF is an applied moment.
FIG. 9.5.1. Throughwall Defect Under Combined Bending and Pressure.
\[ \beta = 0.41 \]
\[ m = 0.8 \]

\[ m = \cos \left( \frac{\pi \beta}{2} \right) \]

---

\[ \beta = 0.86 \]
\[ \frac{0.86}{0.41} = 2.1 \]

\[ \beta = 0.76 \]
\[ \frac{0.76}{0.41} = 1.9 \]

---

**FIG. 9.5.1.** Throughwall Defect Under Combined Bending and Pressure.
Bolts - 6

\[ \sigma_y = 145 \text{ksi} = 896 \text{MPa} \]

\[ \theta = 3.38 \text{ deg} = 8.518 \text{ rad} \]

\[ A = 5789 \text{ mm}^2 \]

Yield capacity = 1000 x 5789

= 5.79 MN

5.18 MN per bolt

Bolt preload = \( \frac{2.78}{6} \text{ MN} \) (approx. 1/2 yield)

Assuming sealing occurs at the important base, the pressure load on each bolt is:

\[ 460.5 \times 500 \times 11 \times \frac{\pi}{4} \times 6 \times \frac{1}{2} \]

= 1052 MN (1.4 kips) per bolt

Pessimistic (500 MPA)

TREX-120129.578

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LNL004-026557
The orientation of the belts to the moment is correct.

\[ \text{Moment} = 330 \text{ kips-ft} \]

\[ = 447.4 \times 10^6 \text{ Nmm} \times \frac{800}{6} \times 10 \]

Load per belt = \(\frac{447.4 \times 10^6}{2 \times 340 \times 2}\)

\[ C = F = 0.0 \]

\[ = 0.33 \text{ MN} \times \frac{590}{330} \]

\[ = 0.59 \text{ MN} \]

(Bolt distance squared). Calc gives 0.37 MN.
Bending About Y-Y

Load per bolt = \( \frac{447.4 \times 10 \times 400}{4(200)^2 + 2(400)^2} \)  
\( = \frac{590000}{4 \times 40000 + 2 \times 160000} \)  
\( = \frac{590000}{330} \)  
\( = 0.182 \text{ MN} \)  

\( \times 330 = 0.182 \text{ MN} \)  

\( \text{Actual + Tension + Band} \)  
\( = 4.652 \text{ MN} \)  

So moment + pressure bolt load is less than 2 MN and margin on yield > 2. \( \checkmark \) Loads are safe.

Bending through a line A-B is marginally worse than X-X, but not as bad as Y-Y.
Theme not considered include the flanges. By inspection they will be stronger than the bolts or pipe sections. Let's of holes through (FEA).

Limiting feature of the bolts in tension may not be nux; tensile area (bearing under head, thread shear) but no details available.
Quick Check on Bearing Stress
Under Bolt Heads

Bolt head OD = 5.25" \sqrt{3} = 133.3 \text{ mm}

Hole = 3.75" = 95.3 \text{ mm}

\text{Area} = \pi \left( \frac{133.3^2 - 95.3^2}{4} \right) = 6822 \text{ mm}^2

At bolt capacity (5.18 MN) bearing stress = \frac{5.18 \times 10^6}{6822} = 759 \text{ N/mm}^2

Under pre-load = \frac{2.78 \times 10^6}{6822} = 408 \text{ N/mm}^2
Yield stress of spool = 552 MPa (80 ksi)

Typical bearing strength for steel
~ 1,589

RF = \frac{1.5 \times 552}{789} = 1.09
Buckling Calculation to

API RP 1111 (P + moment)

Due to combined bending and
external P. Internal P assumed
to be zero. (See later)

Spool piece will be limiting

\[ t = 25.4 \text{ in} \checkmark \]

\[ CR = 21\frac{1}{2}'' = 546\text{ mm} \checkmark \]

\[ DA = 19\frac{1}{2}'' = 495\text{ mm} \checkmark \]

\[ P_0 = 2250\text{ psi} = 15.5\text{ MPa} \checkmark \]

\[ P_c = 0 \checkmark \text{ (As stated above)} \]

\[ M = 330\text{ kips-ft} = 450\times10 \]

\[ = 800 \times 10^6\text{ Nmm} \]

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<table>
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**Calc. Title:** SPOOL PIECE COLLAPSE

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<tr>
<td>Mike Anderson</td>
<td>24/11/10</td>
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</tbody>
</table>

![Equations](image)

**Check external pressure**

\[
f_{0.9} P_e >= P_0 \quad (P_i = 0)
\]

\[0.9 \times 34.2 \geq 15.5 \text{ MPa} \quad \checkmark\]

\[\text{Largest to quotation MBL.}\]

**TREX-120129.586**

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LNL004-026565
Job No. 37958
File Ref.
Calc. No.

Calc. Title SPOOL PIECE COLLAPSE
Page 3 of 5 Pages

Author Mike Anderson
Date 24/6/18
Approved

\[ \varepsilon_b = \varepsilon_b = \frac{0.23 \times 10^6}{25} \]

\[ I = 1.415 \times 10^9 \text{mm}^4 \times \left( \frac{1.408 \times 10^9 \text{mm}^4}{800} \right) \]

\[ \varepsilon = 450 \times 10^{-4} \times 546 = 4.341 \]

\[ \delta = \text{Ovality} = 2\% = 0.02 \]

\[ 9(\delta) = \frac{1}{1 + 20\delta} = 0.714 \]

\[ \varepsilon_b + P_0 \leq 0.714 \]

\[ \varepsilon_b = f \frac{P_0}{f} \]

Take \( f_c = 0.6 \) (non-seamless)

TREX-120129.587

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POOL PIECE COLLAPSE

Author
Mike Anderson

Date
24/1/19

Approved

Date

4.194 x 10^-7
4.341 x 10^-4

15.5

(0.778)

4.2326
0.342

(0.758)

Which is not less than 0.714

\( f_e = 0.17 \) if pipe is seamless,

In which case the requirement is satisfied. NB the contribution from the bending is negligible

\( (0.019) \) in either case

\( (0.018) \)

Conclusion

Spool piece pipe meets the API requirements if it's seamless.

TREX-120129.588

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Assumption of zero internal pressure is too conservative. Use fluid density of 300 kg/m³ internal.

\[ P_0 = 2250 \text{ psi} \]
\[ P_i = 2250 \times 0.3 \]
\[ = 675 \text{ psi} \]
\[ m = 590 \text{kFPM} \]
\[ (P_0 - P_i) = 10.86 \text{ kPa} \]

\[ 7.7 \times 10^{-4} + \frac{10.86}{0.02326} = 0.1562 \]

\[ 0.16 \times 341.2 \]

Which is less than 0.714.

\[ \text{Margin} = \frac{0.714}{0.1562} = 1.27 \]
4.3.2.1 Collapse Due to External Pressure

The collapse pressure of the pipe shall exceed the net external pressure everywhere along the pipeline as follows:

\[ f_0 P_c \geq (P_a - P_i) \]  

(9)

where

\[ f_0 \] is the collapse factor;

- 0.7 for seamless or electric resistance welded (ERW) pipe;
- 0.6 for cold expanded pipe, such as double submerged arc welded (OSAW) pipe.

Under some circumstances in cold expanded pipe, credit can be taken for partial recovery of compressive yield strength by heat treatment to at least 233 °C (450 °F) for several minutes. Such heat treatment may be provided during the fusion bond epoxy coating process of the pipe, provided temperature and duration of heating is carefully controlled. In such cases, the collapse factor of 0.6 may be raised to no more than 0.7. The proposed increase in design factor should be validated through a testing program.

\[ P_c \] is the collapse pressure of the pipe, in N/mm² (psi).

The following equations can be used to approximate collapse pressure:

\[ P_c = \frac{P_e P_t}{\sqrt{P_t^2 + P_e^2}} \]  

(10)

\[ P_y = 2E \left( \frac{L}{D} \right) \]  

(11)

\[ P_y = 2E \left( \frac{L}{D} \right) \] 

(12)

where

\[ E \] is the modulus of elasticity, in N/mm² (psi);

\[ P_e \] is the elastic collapse pressure of the pipe, in N/mm² (psi);

\[ P_y \] is the yield pressure at collapse, in N/mm² (psi).

The collapse pressure predicted by these or other equations should be compared to the hydrostatic pressure due to water depth to ensure adequate wall thickness is chosen for the range of water depths to be encountered.

4.3.2.2 Buckling Due to Combined Bending and External Pressure

Combined bending strain and external pressure load should satisfy the following:

\[ \varepsilon_s + \frac{(P_e - P_i)}{f_0 P_c} \leq g(\delta) \]  

(13)
where

\( f_c \) is the collapse factor for use with combined pressure and bending loads;

recommended value for \( f_c \):

\( f_c = 0.6 \) for cold expanded pipe such as DSAW pipe;

\( f_c = 0.7 \) for seamless pipe.

For installation conditions, consideration can be given to higher collapse factors up to 1.0. Regardless of the selection of the value for \( f_c \), the conditions for collapse in Equation (9) need to be satisfied.

**NOTE** The collapse factor \( f_c \) was absent from Equation (13) in previous editions of this practice. This factor has been added to reflect consistency between DNV design codes, this practice, and API 2RD (refer to OTC 13013.10).

To avoid buckling, bending strains should be limited as follows:

\[ \varepsilon \leq f_1 \varepsilon_1 \]  \hspace{1cm} (14)

\[ \varepsilon \leq f_2 \varepsilon_2 \]  \hspace{1cm} (15)

where

\[ g(\delta) \] collapse reduction factor = \((1 + 20\delta)^{-1}\);

\[ \delta \] ovality = \( \frac{D_{\text{min}} - D_{\text{max}}}{D_{\text{min}} + D_{\text{max}}} \);

\[ \varepsilon \] is the bending strain in the pipe;

\[ \frac{1}{2D} \] = buckling strain under pure bending;

\[ \varepsilon_1 \] is the maximum installation bending strain;

\[ \varepsilon_2 \] is the maximum in-place bending strain;

\[ f_1 \] is the bending safety factor for installation bending plus external pressure;

\[ f_2 \] is the bending safety factor for in-place bending plus external pressure;

\[ D_{\text{max}} \] is the maximum diameter at any given cross section, in \text{mm} (\text{in}.);

\[ D_{\text{min}} \] is the minimum diameter at any given cross section, in \text{mm} (\text{in}).

**NOTE** Equation (13) is acceptable for a maximum \( D_{\text{min}} \leq 50 \). Refer to the OMAE article for utilizing ratios higher than 50.

Bending strains \( \varepsilon_1 \) and \( \varepsilon_2 \) are not simply nominal (global) bending strains and shall include an allowance for possible strain concentrations. For example, if a pipe is reeled, the nominal bending strain in the pipe on the reel or aligner is given by:

\[ \varepsilon_{\text{nom}} = \frac{d_{\text{pipe}}}{d_{\text{nom}} + d_{\text{out}}} \]  \hspace{1cm} (16)

where

\[ d_{\text{pipe}} \] is the outside steel pipe diameter;
Technical Note on Bolt Torque/Preload relationship

Using TS70 Moly paste, GE/Vetco Gray design basis is to apply 17,000 ft-lb of torque to achieve a preload of 625ksi.

Per BS3580, the Torque to achieve a desired Preload has three elements:

\[ T = P_0 \left[ \frac{p}{2\pi} + E_s \mu_1 / 2 \cos \alpha + (d_o + d_i) \mu_2 / 4 \right] \]

\[ T = P_0 \left[ \text{mechanical advantage in thread} + \text{friction in thread} + \text{friction under bolt head} \right] \]

Using the same lubricant, it can be assumed that the friction under the thread is the same as the friction under the bolt head. To get 625ksi preload for 17,000ft-lb of torque implies a friction coefficient of 0.072 - note that this applies to making the joint up on the surface. A typical steel to steel bolted joint made up by dipping the bolts in oil corresponds to a friction coefficient of about twice this value, say 0.15.

For the subsea installation condition, the mechanical advantage due to thread form is unchanged relative to the surface assembly condition as is the friction under the head of the bolt, since it can be prepared with a layer of Moly paste. The only uncertainty therefore is how effectively the Moly paste will transfer from the bolt threads to the internal threads of the captive elliptical nuts when the bolts are done up.

The torque required to achieve a 625ksi preload assuming that the effective friction coefficient in the threads is midway between the design value for TS70 Moly paste (0.072) and the value obtained for a bolt dipped in oil (0.15) is therefore given by:

\[ T = 625E3 \times [0.04 + 1.765 \times (0.072 + 0.15) / 2 + 0.16] \text{ in-lb} \]
\[ T = 20530\text{ft-lb, i.e. a 21\% increase} \]

The finite element analysis of the flange assembly under combined pressure and moment loading showed that even with a 25% reduction in the 625ksi preload, the bolts remain effective for the current design basis.

The margin on bolt yield for the current design basis is 1.31. Therefore it could be considered prudent to increase the torque on the bolts by about 5-10% to make up for the inability to affect the lubrication state of the nuts currently on the existing G flange.
### ASME B31.8 Results Summary

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<th>External Pressure</th>
<th>Axial Load</th>
<th>Moment</th>
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### API 1101 & API RP2A Results Summary

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**Page 1**

**Transition Spool Pipe Check** Rev B.4.4

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