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Updated Design Methods for HPHT Equipment

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Abstract

Drilling industry technology is advancing rapidly. Drillers are encountering downhole pressures over 20,000 psi and temperatures over 450°F. These high pressure high temperature (HPHT) conditions require drilling and completion equipment that is beyond the scope of today's API specifications.

API specifications 6A[1], 16A[2], 16C[3], and 17D[4] address the design and design verification methods for drilling equipment. These specifications currently reference the ASME Boiler and Pressure Vessel Code, Section VIII, Division 2[5] (ASME VIII-2) as one of the primary design verification methodologies. API specifications first referenced ASME VIII-2 as a design verification methodology nearly 20 years ago, because it was the best available at that time. This is called the "ASME Method".

The ASME Section VIII, Division 3[6] (ASME VIII-3) was developed to give the requirements for the construction of high pressure vessels. ASME VIII-3 was first issued in 1997 and is intended to be used in place of ASME VIII-2 for high pressure vessels, generally in excess of 10,000 psi.

The design of drilling and completion equipment should be done using first principles. The authors propose that the designs for 15,000 psi or higher service should be verified using the rules from ASME VIII-3. A significant difference between the two design verification methods is that the user specifies the loading criteria for the performance-based ASME VIII-3. ASME VIII-3 requires cyclic load (fatigue or fracture mechanics) analysis and has limitations in materials, fabrication and inspections, and testing requirements that specifically apply to thick-walled pressure vessels.

This paper discusses how the performance-based code methods of ASME VIII-3 can be integrated into API specifications for HPHT drilling and completion equipment.

Introduction

This paper originated from requests by major operators for HPHT development and completion well equipment in ultra-deepwater in the Gulf of Mexico. Gulf of Mexico deep gas, as well as deep gas exploration on land, will require the development of equipment rated for 25,000 and 30,000 psi in sizes larger than existing API specifications. Along with these higher pressures come increased temperatures, upwards of 450°F.

Drilling for oil and gas in these new frontiers is challenging the development of equipment. Along with the promise of increased rewards, however, come increased risks associated with equipment failures.

Operational Impacts

The need for higher working pressure equipment has operational implications as well as design impacts. Equipment with the same bore sizes, such as wellheads, valves, and BOP's, will grow substantially in outside dimensions as well as total weight.

For example, an 18 ¾"–20,000 psi BOP assembly could easily weight double that of a similar current design 18 ¾"–15,000 psi BOP assembly. Using conventional materials, the assembled weight of an 18 ¾"–20,000 psi BOP is about 1,500,000 lbs. This is based on the ability of the BOP manufacturer to qualify variable bore pipe rams for circulating temperatures at the BOP as high as 325°F to 350°F. It is unlikely in the near term that BOP manufacturers will be able to qualify variable bore pipe rams for these HPHT applications. A BOP outfitted for both drilling and completion operations may need six or more ram assemblies to cover the pipe sizes and the shear rams. This increases the weight of the BOP assembly to about 2,000,000 lbs. Few, if any, drilling vessels in today's offshore fleet can carry, handle, and run a BOP assembly of this weight and size.

18 ¾"–20,000 psi wellheads will need outside diameters of 30 inches or more using conventional sour service materials. For existing wellhead connectors, only the largest sizes available in the industry today have the lower body locking mechanisms large enough to accommodate this size mandrel.

All well control equipment components are similarly affected.

Manufacturers need to design flex loops mounted on the lower riser package rated for these higher pressures. These may be flexible pipes or steel loops that allow the riser to flex at its design limit. The added wall thickness for the loops will require larger bend radii to obtain the required flexibility.

Choke and kill (C&K) lines on the riser joints will have thicker walls and associated weights. There may be an incentive towards designs using alternative materials to reduce weight. The riser will need more buoyancy materials to offset the higher C&K line weights. The added steel and buoyancy weight will have significant impact on the vessel variable deck load (VDL) on board the drilling rig.

The droop hoses in the moonpool, which connect the riser to the vessel, will require flexible pipe that is not currently available. The flexible pipe will require a working pressure equal to the BOP and riser C&K system.

Vessel piping from the moonpool C&K lines to the manifold, the kill line from the cement unit to the manifold, and the handling equipment will all increase weight and VDL.

One alternative that some operators are reviewing is a two-stack approach. The BOP assembly will be changed out from an 18 ¾"–15,000 psi stack to a 13 5/8"–20,000 psi stack when the higher pressure rating is required. This will allow a 15,000 psi rated rig to drill the majority of the well and then the 20,000 psi rated equipment to finish the drilling operations and complete the well for production. This two-stack approach will reduce the weight and size of the BOP, but will require additional operations during the well construction process.

Equipment Impacts

The design of existing drilling equipment is based upon design verification methods established nearly two decades ago. The strength analysis is based on classical strength of materials equations or linear-elastic finite element analysis (FEA). The equipment designs for lower pressures and temperatures do not consider cyclic behavior, fatigue or fracture mechanics.

The current standards for the design and manufacture of equipment for HPHT applications fall under the auspices of API 6A, 16A, 16C, or 17D. One of the design verification methods used by these specifications was established in 1986 with the adoption of portions of ASME VIII-2 in the 15th edition of API 6A. This "ASME Method" uses portions of ASME VIII-2. The ASME Method uses linear-elastic stress analyses and considers the primary failure mode to be elastic or plastic deformation resulting in leakage before fracture. The current API specifications also do not require fatigue analysis as part of the design verification requirements.

Designs verified using the current methods are thought to be acceptable for all drilling applications and to have service lives of at least 30 years. In order to assure equipment safety when using these methods, large safety factors and high hydrostatic proof test pressures are used. As a result, equipment designed using current methods is usually heavy, robust, and perhaps more costly than necessary.

A single design that will fit all applications is simply not possible when high pressures and high temperatures are involved. Today's HPHT equipment requires custom designs for safe operation, which are built for specific service requirements that consider all loading scenarios, failure modes, and useful life of the equipment. The custom designs require a more detailed design verification analysis using advanced techniques.

Experience has shown that high pressure vessels usually have a failure mode based upon cyclic loading, crack propagation, and eventual fracture. Often, high pressure vessels appear to fail in a fast fracture mode due to the thick walls that prevent the material from necking down.

The ASME adopted rules for the design verification of high pressure vessels in 1997, specifically ASME VIII-3. This code uses elastic-plastic analyses and fracture mechanics. The intent of this method is to verify that the design is fit for the specified service requirements as defined by the user.

The resulting equipment will be lighter in weight, but will have a limited safe life since the materials will be more highly stressed.

User Design Specifications

A significant difference between current practice and ASME VIII-3 rules is that the user defines the service requirements up front. The user provides a document that describes the expected service life. An example of such a document is Figure 5. This is the basis for the manufacturer to use in the designs and verification analyses. This is in contrast to the current practice where manufacturers provide standardized designs thought to be good for all applications.

Design Verification Methods

Design methods that are appropriate for conventional equipment are not suited for the requirements of HPHT. Using appropriate design methods are important in the development and manufacture of HPHT equipment. This section of the paper describes the contrast between the current ASME Method and the ASME VIII-3 rules as applied to the design of an example BOP. Our objective is to show the technical merits for the adoption of ASME VIII-3 concepts in developing the next generation of API standards for HPHT design.

The analyses done for stress verification are only some of the factors that need to be considered in the design of HPHT equipment. Other factors such as deflections, sealability, and additional post-weld heat treatment (PWHT) cycles will also affect the final pressure rating of a component.

API

The current API design verification methods are based on linear-elastic stress analysis. The listed API specifications differ slightly in their allowable stress limits, but all follow the same basic criterion. The maximum allowable general primary membrane stress intensity in the pressure wall at hydrostatic test pressure is limited to either 83% (API 6A and 17D) or 90% (API 16A and 16C) of the material specified minimum yield strength.

General primary membrane stress intensity is the average stress intensity in a section, away from discontinuities. The stress intensity is twice the maximum shear stress or the largest algebraic difference between the principal stresses.

All four current API specifications state that fatigue analyses are beyond their scope. This is not appropriate for high pressure equipment, since the most likely failure mode is fracture due to cyclic loading.

ASME VIII- 3

The ASME VIII-3 rules go beyond a stress analysis based upon static loading conditions. The analysis verifies that the equipment is designed to be fit-for-purpose. This requires the user and the manufacturer to work together. The user defines the service requirements for the equipment. Then, the manufacturer completes the design and the design verification analysis to demonstrate the structural capacity of the equipment to meet the specified service requirements.

In addition to the linear-elastic methods used in ASME VIII-2 (and API), ASME VIII-3 uses elastic-plastic FEA and linear-elastic fracture mechanics analyses. Figure 1 is a flow chart that shows the overall design verification process that is proposed for the next generation of API codes for HPHT equipment. The design verification analyses include the calculation of linear-elastic stresses for the fracture mechanics calculations and an elastic-plastic load analysis.

The design verification process includes:

- Linear-elastic finite element modeling
- Fracture mechanics calculations including fatigue crack growth calculations
- Limit analysis using an elastic-plastic material model

A fatigue analysis using elastic stresses and conventional S-N fatigue curves is not appropriate for thick-wall pressure equipment. All mentions of cyclic behavior and fatigue in this paper concern crack growth as a function of cyclic loads and calculations based on linear elastic fracture mechanics, LEFM.

Linear-elastic Finite Element Modeling

In the ASME Method (and API), the stresses found in the linear-elastic FEA are compared directly to specified stress limits. In ASME VIII-3, the reason for doing the FEA is to get the stress profile for the fracture mechanics calculations.

Linear-elastic FEA calculates the stress profile in a body under load. The finite element analysis of high pressure BOP's uses solid continuum elements because of the thick walls. Figure 2 shows example geometry for a high pressure BOP body and the finite elements of the model. This example uses quarter-symmetry to reduce the calculation time. In this model, elastic materials were used and pressure was applied to the internal surfaces and pressure end loads to the openings.

When the FEA is complete, the analyst finds the location of maximum stress. At this location, the analyst develops an equation for the fracture mechanics calculations from the stress profile.

Figure 3 shows the von Mises stress contours in the example BOP body. The stresses are highest at the corner on the inside of the BOP. The stress distribution equation was written for this location.

Fracture Mechanics Calculations

LEFM was selected as the basis for ASME VIII-3 fatigue analysis because it was thought to best represent the behavior of high pressure vessels. Other fracture mechanics methods may be used if they are based on sound engineering principles and consistent with the safe design practices for pressure vessels such as those embodied in ASME VIII-3.

For this discussion, there are two parts to the fracture mechanics assessment. The first calculates the critical crack size based on material toughness, crack geometry, and stress level. The second is a fatigue analysis to follow crack growth based on the number of cycles at the various stress levels. Figure 4 is a flow chart that provides a list of the steps used in the LEFM analysis.

LEFM uses material toughness, often in the form of Charpy V-notch (CVN) values. Crack growth is calculated starting from an initial flaw size. The initial flaw size is often set by the sensitivity of the inspection method. The rate of crack growth is a function of stress level, cyclic operating loads, and material toughness properties.

The calculated final crack size must be less than the maximum allowable flaw size. ASME VIII-3 limits the flaw size to the smallest of 25% of the wall thickness or 25% of the critical crack size.

The steps used to determine the critical crack size are:

- From the elastic FEA stress results, find an equation for σ as a function of x/a [σ(x/a)] using the following relation:

$$\sigma\left(\frac{x}{a}\right) = A_0 + A_1 * \left(\frac{x}{a}\right) + A_2 * \left(\frac{x}{a}\right)^2 + A_3 * \left(\frac{x}{a}\right)^3$$

- Convert the CVN value to the fracture mechanics stress intensity, K_{IC}.
- Using an expression for K based on σ(x/a); calculate the critical crack size, a_c, using K_{IC}.

After the critical crack size is found, a fatigue analysis is done to calculate the crack growth due to cyclic service. The equation used to calculate crack growth, *da*, as a function of cycle number, *dN* is known as the *Paris Law*[10].

$$\frac{da}{dN} = C \cdot \Delta K^{-m}$$

where:

da	Crack growth (inches)
dN	Number of cycles
ΔK	Range of stress intensity (ksi)
C, m	Material constants

Figure 5 has a table that includes an example pressure service history.

- Using ASME VIII-3, the maximum crack size, a_{max}, is 25 percent of the critical crack size, a_c.
- Solve for *da/dN* using C and m from ASME VIII-3 and the expression for K that is a function of internal pressure and crack size. The iteration starts by assuming an initial flaw size based on the size of the maximum undetected flaw using the actual inspection technique. In the example, a surface crack at the cross bore is assumed to have an initial flaw that is the same size as the initial crack depth.
- Calculate total crack growth, a_{tot}, by summing *da* for each pressure and cycle number combination.
- The total crack growth, a_{tot}, must be less than the maximum crack size, a_{max}, to ensure that the rules of ASME VIII-3 are satisfied. [Note: ASME VIII-3 has additional limits on cycles that should be considered and assessed on each analysis.]

Based upon the ASME VIII-3 rules, the pressure rating of a component is usually limited by the fatigue analysis. The fatigue analysis considers all the cyclic loading in the user design specifications including hydrostatic pressure testing and thermal stresses. Any factor that influences the fatigue life, such as more load cycles or reduction of material toughness from added post weld heat treatment (PWHT) cycles, may affect the final pressure rating of a product.

Limit Load Analysis

A limit load analysis is a nonlinear elastic-plastic FEA. This section describes an example evaluation performed on a BOP body design. Only the Young's Modulus and the yield strength are used for material properties since ASME VIII-3 requires the use of an elastic, perfectly plastic material model in elastic-plastic analyses. The typical high strength low alloy steels used for BOP's behave much like the elastic, perfectly plastic material model. We used a small displacement relationship, consistent with conventional analyses of structures using limit load methods.

In this limit load analysis, you find the lower bound collapse load. The internal pressure and pressure end loads were increased to the level that convergence was improbable.

The load-deflection data from the analysis are plotted in Figure 6. These data were obtained by monitoring the deflection of a node located on the inner surface of the BOP body. The instability of the model occurred at a pressure of about 129,000 psi. This pressure is the *lower bound collapse load*.

ASME VIII-3 rules say that the design margin against collapse load may not be less than 1.732. Using the 129,000-psi lower bound collapse load, the resulting design pressure is limited to 74,500 psi.

As an additional check, ASME VIII-3 does not permit the design pressure, P_d, to exceed the value set by the following equation. Technically, this equation only applies to cylindrical shells and only at locations remote from discontinuities. However, it is appropriate as a calculation of a pressure limitation for the example BOP body along with the other limits.

$$P_{design} = \frac{2}{3} \cdot S_{yield} \cdot \ln(Y)$$

where "Y" is the ratio of the outer diameter to inner diameter of the pressure vessel

The maximum pressure from the limit load analysis and the design pressure limit from the equation above are clearly not the limiting factors in the final pressure rating. In most high pressure designs, pressures are limited based on requirements associated with the fatigue analysis.

Comments on Design Verification Methods

Information is presented on the merits of API's adoption of design verification methods used in ASME VIII-3 in developing the next generation of standards for HPHT equipment.

The basis of the current design verification methods are on linear-elastic material properties and analysis that has thin-wall shell theory at its foundation. Linear-elastic methods limit design pressures due to their inability to account for plasticity and the stress redistributions that result. Thick wall vessels intended for high pressure service require an evaluation using elastic-plastic material properties and, more importantly, fracture mechanics based fatigue calculations.

ASME VIII-3 requires an elastic-plastic analysis and does not permit linear-elastic analysis when the Y ratio (OD/ID) exceeds 1.5. HPHT equipment rated for 15,000 psi or higher will have Y ratios greater than 1.5 unless the metallurgists can find some miracle material. In addition, ASME VIII-3 requires fracture mechanics calculations for all but a select few, thin-wall, when doing fatigue evaluations.

Materials

The primary difference in material property requirements between API and ASME VIII-3 is the fracture toughness. API requires CVN values for three samples to exceed 15 ft-lb average at the minimum design temperature with no single sample falling below 10 ft-lb.

ASME VIII-3 requires either CVN or CTOD fracture toughness per ASTM E-1290[8], J_{IC} fracture toughness per ASTM E-1820[9], or K_{IC} fracture toughness per ASTM E-399[7] for the primary forging. The objective is to obtain the K_{IC} toughness value for the fracture mechanics evaluation. For CVN, ASME VIII-3 requires a minimum average value of 30 ft-lb transverse (50 ft-lb longitudinal) at the minimum design temperature and no single value less than 24 ft-lb transverse (40 ft-lb longitudinal). ASME based these minimums on the need for relatively high ductility for the materials of construction. A particular set of user requirements and a manufacturer's design may need higher toughness minimums to satisfy the fatigue analysis.

Detailed material requirements for HPHT components are the subject of other papers to be written.

Fabrication and Inspection Using ASME VIII-3

A variety of welding processes are usable for fabrication of HPHT equipment. However, only full penetration butt joints made by double welding are permitted. Other joint designs are permitted if they meet the requirement to assure that the deposited metals on both the inside and outside of the weld are of the same quality.

Welding procedure qualifications include testing the base metal, weld metal, and heat affected zones after all thermal treatments. The weld metal and heat affected zones are required to meet all the mechanical test requirements of the base metal including toughness testing,. This requirement extends to production test coupons on each component built.

Post weld heat treatment after all welding will usually be necessary in order to meet the material mechanical test requirements for the weld metal and heat affected zones. Both preheat and post weld heat treatment are usually required for fabrication of the high strength low alloys thought to be typical for this equipment.

Examination of all welds is done by sensitive ultrasonic methods (UT) that are qualified to reliably identify flaws down to a size that is consistent with the initial flaw size used in the fracture mechanics assessment.

All pressure containing components are required to be examined throughout 100 percent of their volume after the quench and tempering phase of the heat treatment. This means that each forging will be examined in detail in the semi-finished state before final machining. The examination shall be done using UT. If the component cannot be adequately inspected by UT, radiographic methods (RT) shall be used. In addition, all external surfaces and accessible internal surfaces shall be examined by either the magnetic particle method (MT) or liquid penetrant method. Acoustic emission methods may be used to supplement the volumetric examination.

After all heat treatment and hydrostatic testing, ASME VIII-3 requires an examination of all surfaces by either magnetic particle methods or liquid penetrant methods. The surface NDE acceptance requirements according to ASME VIII-3 are essentially the same as API Specification 16A.

Testing

ASME VIII-3 requires a hydrostatic pressure test to a level of at least 1.25 times the design pressure rating of the vessel. The user may require a higher pressure. For a 20,000 psi design, the test pressure would be at least 25,000 psi.

For single wall vessels, ASME VIII-3 places an upper limit on hydrostatic test pressure according to the following equation:

$$P_{test} = S_{yield} \cdot \ln(Y)$$

This pressure results in the outer fiber of a cylinder, remote from discontinuities, reaching yield stress.

Conclusions

Today's drilling industry is encountering new challenges. The HPHT environment of 20,000 psi and over and 350°F and over will require the design of a new class of equipment to meet these ratings.

Drilling equipment, such as wellheads, BOP's, valves, and C&K systems, is designed and manufactured under the auspices of API specifications. The design verification methods in these API specifications was adopted nearly 20 years ago. They use portions of the ASME VIII-2 rules. Since that time, ASME has adopted newer methods for the design verification of high pressure vessels by issuing the ASME VIII-3 rules.

ASME VIII-3 permits high stress levels but has a mandatory fatigue analysis. This results in lighter weight designs that have a limited safe life. The intent of the ASME VIII-3 methods is to verify that the design is fit for the specified service requirements as defined by the user.

API should integrate ASME VIII-3 rules into existing specifications for equipment rated for 15,000 psi and higher for improved performance with equivalent safety.

References

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10. Paris, P. and F. Erdogan, *A critical analysis of crack propagation laws*, Journal of Basic Engineering, Transactions of the American Society of Mechanical Engineers, December 1963, pp.528-534.

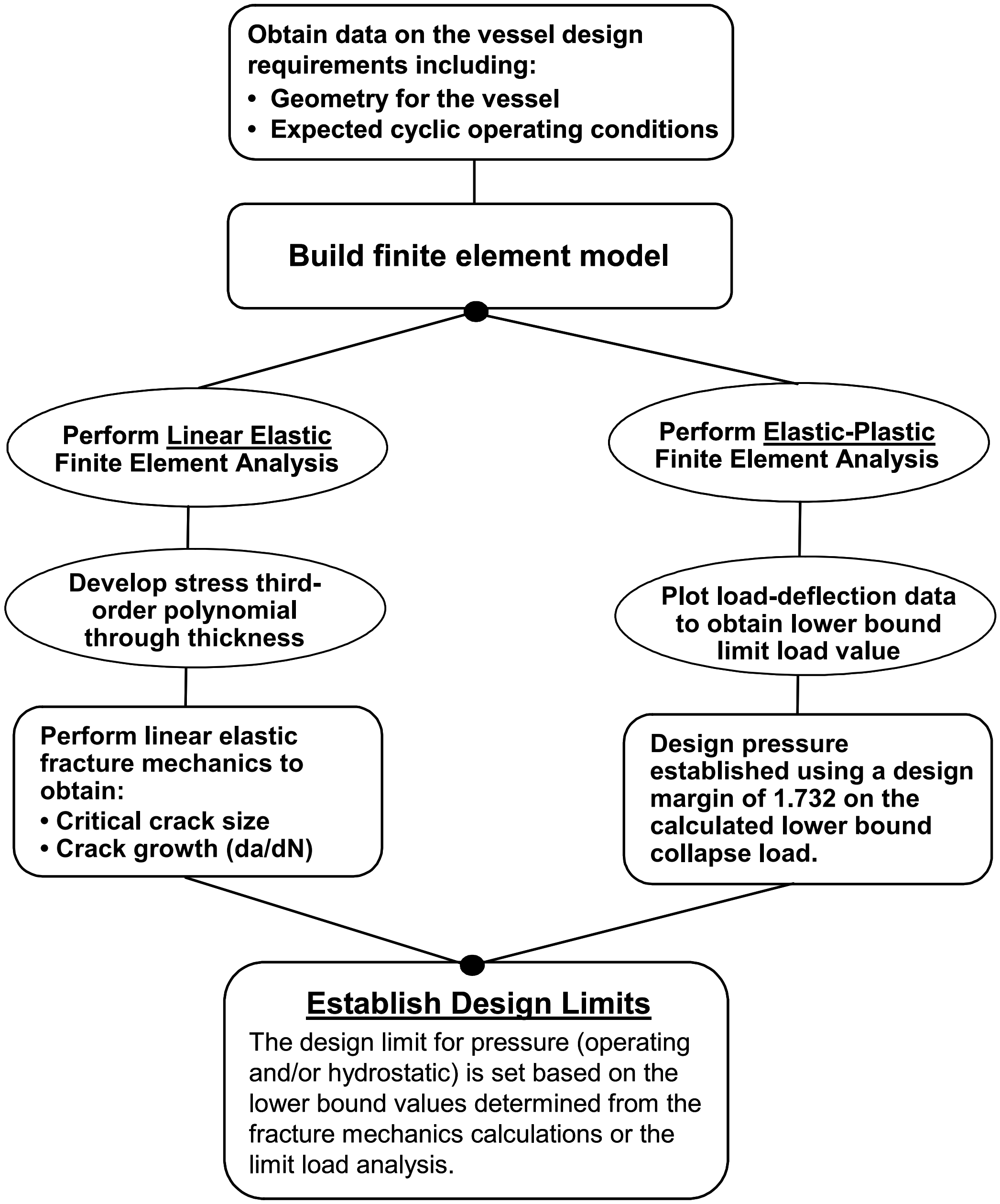

Figure 1 – Flow Chart for High Pressure Vessel Design Verification

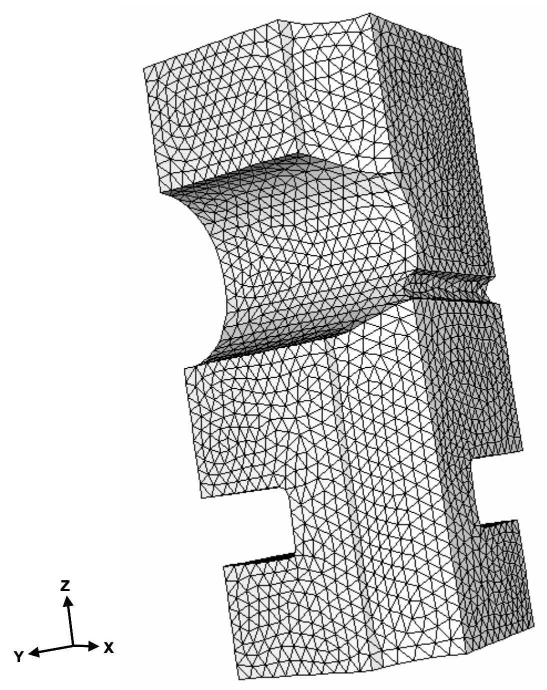

Figure 2 – Exemplar Geometry for a High Pressure BOP Model (Quarter-Symmetry)

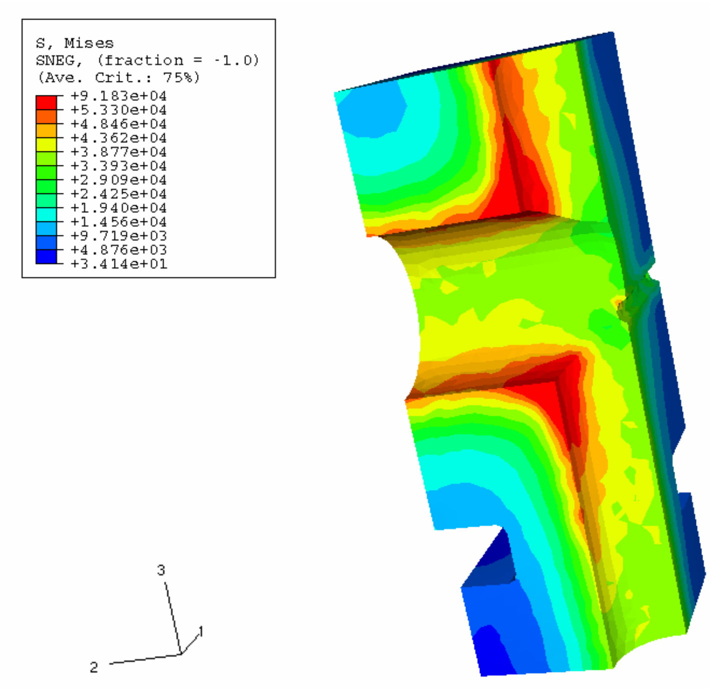

Figure 3 – von Mises Contour Plot for the Exemplar BOP at 129,000 psi

Linear Elastic Fracture Mechanics

Develop third-order polynomial expression for σ(x/a) using elastic finite element stresses. The variables A_0, A_1, A_2, and A_3 are calculated.

↓

From Table D-401.1 get G_0, G_1, G_2, and G_3 using *a/t*=0.0 and *a/l=0.5* (per D-402(a)) ratios associated with the crack geometry.

↓

Develop expression for K using appropriate internal pressure, A_P, and crack depth, *a*.

↓

$$K_I = [(A_0 + A_P)G_0 + A_1G_1 + A_2G_2 + A_3G_3] \cdot \sqrt{\frac{\pi a}{Q}}$$

$$Q = 1 + 4.593 \cdot \left(\frac{a}{l}\right)^{1.65} - q_y$$

$$q_y = \frac{1}{6} \cdot \left[\frac{(A_0 + A_P)G_0 + G_1A_1 + G_2A_2 + G_3A_3}{S_y}\right]^2$$

<u>Solve for critical cracks size, a_c</u>

Using maximum internal pressure (e.g. 25 ksi hydro) and minimum CVN of 30 ft-lbs, calculate critical crack size of **<u>0.453 inches</u>**.

$$K_C = \sqrt{5\sigma_y{}^2 \cdot \left[\frac{CVN}{\sigma_y} - 0.05\right]}$$

↓

<u>Solve for maximum flaw size</u>

Per KD-412.1 the maximum flaw size is limited to the lesser of 25% of the wall thickness or 25% of the critical crack size.

↓

<u>LEFM-based Fatigue Calculations</u>

Using the operating conditions (including load and frequency data), determine the growth of the initial crack size based upon material constants, *C* and *m*. Final growth should be less than maximum allowable flaw size (*0.25*a_c).

$$\frac{da}{dN} = C \cdot \Delta K^{-m}$$

Figure 4 – Linear-Elastic Fracture Mechanics Analysis Flowchart Based on ASME VIII-3

BOP Pressure Testing History

Assumptions

1. Six ram stack (3 doubles) and 2 annulars
2. Upper ram is casing shear ram (non-sealing)
3. Four year cycle between major overhauls
4. Twelve weeks per well
5. Pressure cyles based upon Lower Pipe Ram (worst case)

Types of Tests

1. Factory hydro pressure test
2. Factory wellbore pressure test
3. Factory stack wellbore pressure test
4. Surface stack wellbore pressure test
5. Subsea stack wellbore pressure test after landing stack
6. Bi-weekly stack pressure test to casing pressure
7. Bi-weekly ABOP pressure test to casing pressure or 15 ksi

Years / Equipment Cycle	4
Days / year	365.25
Total days	1461
Days / well	84
Wells / equipment cycle	17.39
Wells / equipment cycle	**17**
Number of ABOP's	2
Number of Ram BOP's	5
LPR pressure cycles / stack test	7

	Pressure Level (ksi) for Cycle Count					
	10	**15**	**18**	**20**	**30**	
Factory Testing Cycles						
Body hydro test					2	
BOP wellbore test				2		
Stack test		2		5		
Rig Testing Cycles / well						
Surface stack test		2		5		
Land stack on wellhead		2		5		
Week 2	7					
Week 4		7				
Week 6		2	5			
Week 8		2	5			
Week 10		2		5		
Week 12		2		5		
Pressure cycles / Equipment cycle	119	323	170	340	0	
Factory pressure cycles	0	2	0	7	2	
Total cycles	**119**	**325**	**170**	**347**	**2**	**963**

Figure 5 – Example of Equipment Service History

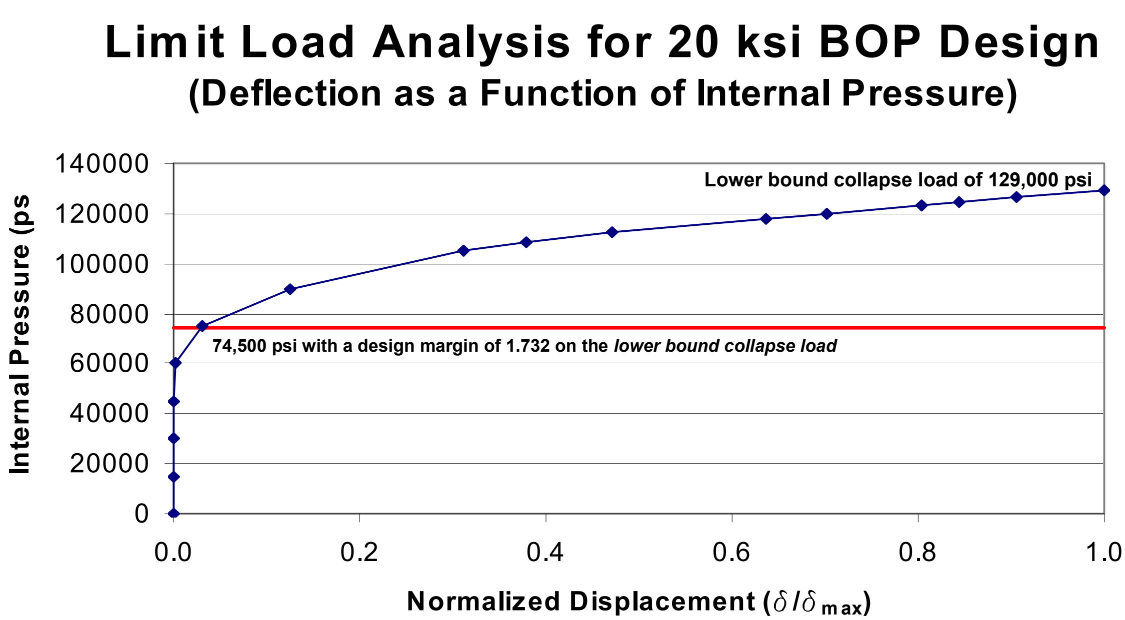

Figure 6 – Limit Load Analysis Results