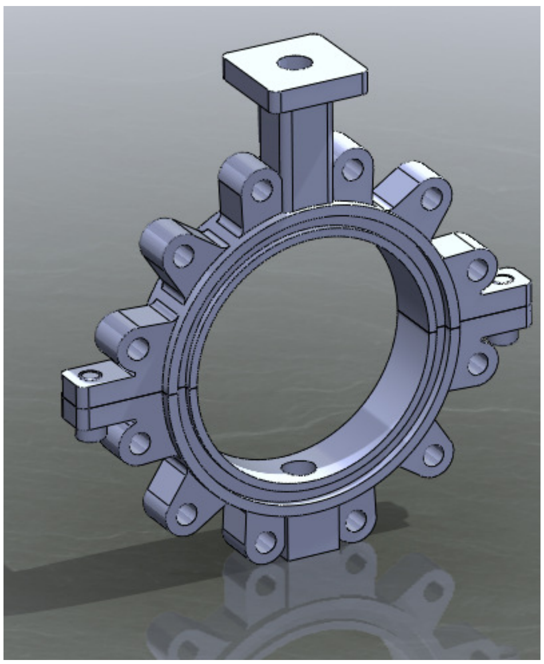
Ph: 520-265-3657
Fax: 888-241-3035
mailto:info@keydesigneng.com
http://www.keydesigneng.com/

Brief Discussion:
Split-Body 12in Butterfly valve,
Finite Element Analysis per ASME B31.3



Prepared by: Michael Rodgers, P.Eng.

Date: July 16, 2010

Key Design Engineering is a Canadian engineering firm located in Waterloo, Ontario, that specializes in ASME Code calculations and Canadian Registration (CRN):

- **Code Calculations** per ASME VIII-1 & ASME B31.3 for Pressure Vessel, Fittings, & Piping systems.
- **Finite Element Analysis (FEA)** of Pressure Vessels and Fittings in accordance with ASME VIII-2, as permitted by VIII-1, B31.3, & B31.1.
- **Canadian Registration (CRN)** preparation of documentation for submission of pressure vessels, fittings, or pressure piping for registration in one Jurisdiction or Canada-wide.

KEY DESIGN ENGINEERING
Ph: 520-265-3657
Fax: 888-241-3035
mailto:info@keydesigneng.com
http://www.keydesigneng.com/

Section Headings:

Introduction and Scope:

This brief discussion investigates the central joint of a 12" split-body butterfly valve using Finite Element Analysis (FEA) interpreted per the ASME Code. The observations are primarily of a qualitative rather than a quantitative nature. This is not a comprehensive report such as is used to submit to a Canadian Jurisdiction for CRN registration. Rather, it illustrates how FEA is used to fill in gaps in the classical ASME Code configurations and how it can replace burst testing for proving an acceptable design pressure. Please note as well that the effects of *dead-end service* and *flow-induced vibrations* are beyond the scope of the present study.

Background:

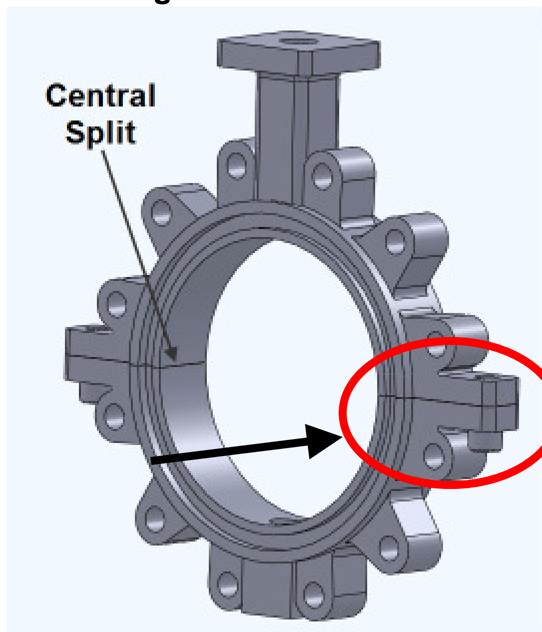
This 12-inch split-body butterfly valve cannot not covered by the classical Code calculations and thus the design must be proven either by burst testing per ASME VIII-1-UG-101(m) or by Finite Element Analysis (FEA) per ASME VIII-2. FEA does not fully replace physical testing. However, it is useful in proving the safety of the design and has several advantages over burst testing for the purpose of establishing an acceptable design pressure, such as:

- <u>Material Options</u>: substantially different materials can be covered by a single worst-case scenario,
- <u>Design iteration</u>: various design conditions and design options can be reviewed and changes can be quickly implemented if such are required,
- <u>Component preservation</u>: no destruction of the component is required,
- <u>Interpolation</u>: greater interpolation of sizes, particularly in cases where the stress is very low.

Ph: 520-265-3657
Fax: 888-241-3035
mailto:info@keydesigneng.com
http://www.keydesigneng.com/

Why it cannot be covered by the Classical Code Configurations:

The primary reason why this valve cannot be covered by classical Code Calculations is that the body is not a continuous circumference. This requires a bolted connection at the centre, as shown in the image to the right. This central joint provides various features that are worth mentioning and need to be carefully considered in the FEA model setup and interpretation:

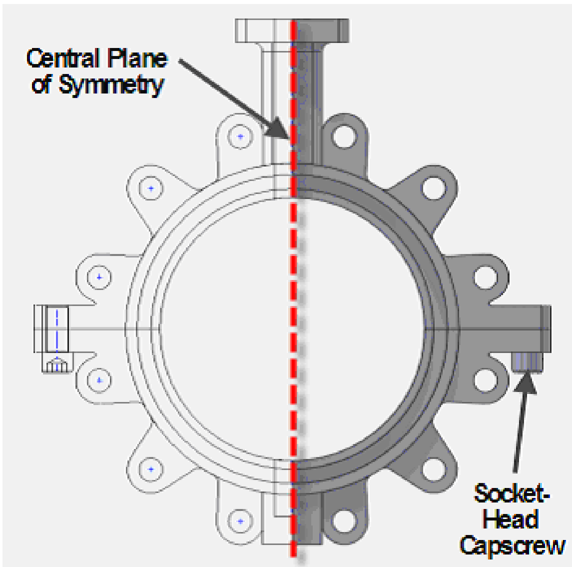


- *Metal to Metal contact*: The bolting preload must be sufficient to maintain joint contact, to avoid premature failure of the bolts during the pressure cycles.
- *Hinge Effect*: The central mating tabs experience rotation about the outer edge. Two factors that affect this are the preload and the stiffness of the tab. Higher preload mitigates this hinge effect. Contrariwise, a weaker tab reduces the required preload.
- *Tab Stress*: The tab must be able to withstand both the shear and bending stresses. If it is too thin to reduce the bolting load, this will increase the shear stress and peak stress at the connection with the body. If the tabs are too long, they will have greater bending stress.

These are some of the points that must be considered and require the use of FEA over classical Code Calculations.

Ph: 520-265-3657
Fax: 888-241-3035
mailto:info@keydesigneng.com
http://www.keydesigneng.com/

Setup, Results, and Discussion



Model Setup:
This 12-in model is both symmetrical about a vertical axis, as shown, and through the bolt centrelines. The analysis was performed on the right-hand side component only, shown solid, with a symmetry restraint on both the central cut-planes. A beam-based bolt connector feature is used to simulate the socket-head capscrew. The upper section is constrained vertically by the bolt connector.

Allowable stress for the materials covered, at 100F:

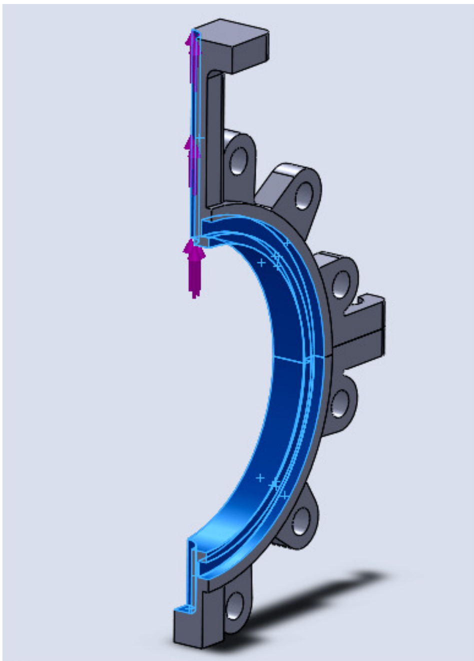
A395:	20 KSI
A216 WCB:	21.9 KSI
A351 CF8M:	20 KSI
A193-B7:	25 KSI (bolt)

Bolting pattern and drilling corresponds to ASME B16.5 for a 150# 12in flange.

KEY DESIGN ENGINEERING

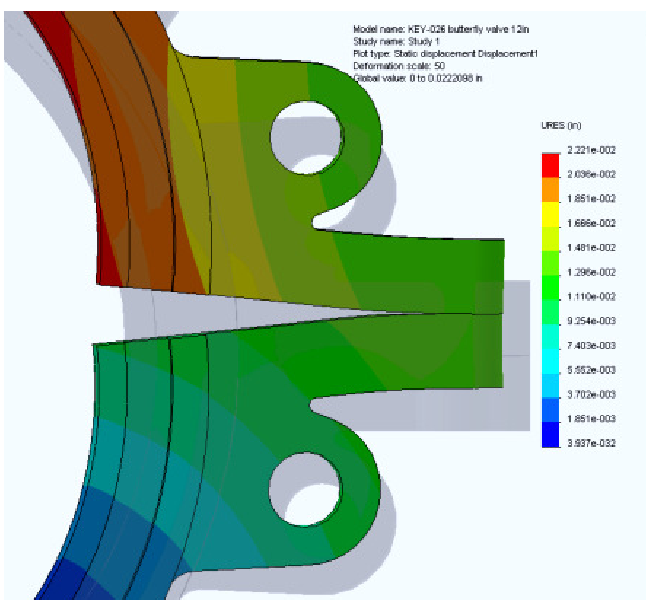
Ph: 520-265-3657
Fax: 888-241-3035
mailto:info@keydesigneng.com
http://www.keydesigneng.com/

Load Application:



- Pressure: The internal faces are pressurized with 275 PSI pressure, which is a reasonable worst-case scenario for the 150# flange group, covering various materials.
- Forces: The upper stem opening has a pressure-equivalent force applied to the bore.
- Preload: The bolt is preloaded to 23,200 PSI, which is later shown to be a conservatively low number.

Joint Displacement:



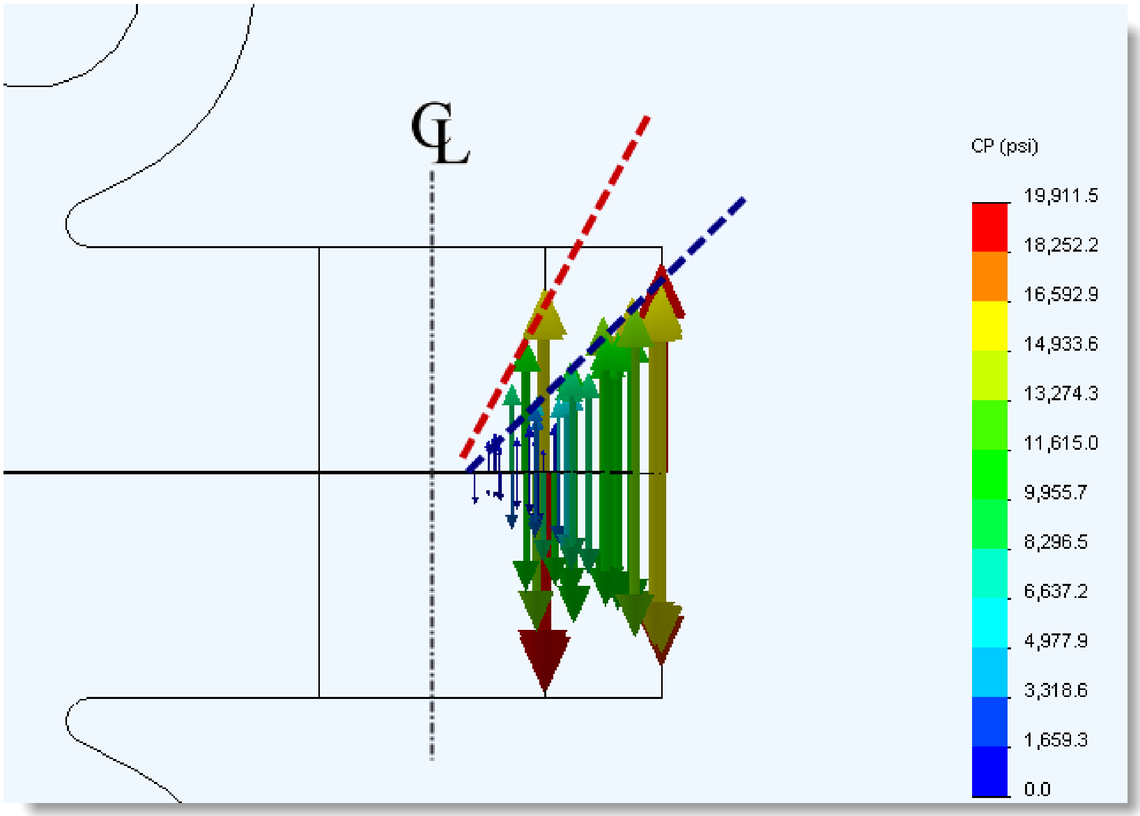
The image to the left shows the joint displacement:

- Separation: The joint separation at the inside surface proves that the sliding contact at the faces has been properly applied.
- Shape: The deformed shape looks like a cantilever-beam, which suitably reflects the preloaded bolt condition.

KEY DESIGN ENGINEERING

Ph: 520-265-3657
Fax: 888-241-3035
mailto:info@keydesigneng.com
http://www.keydesigneng.com/

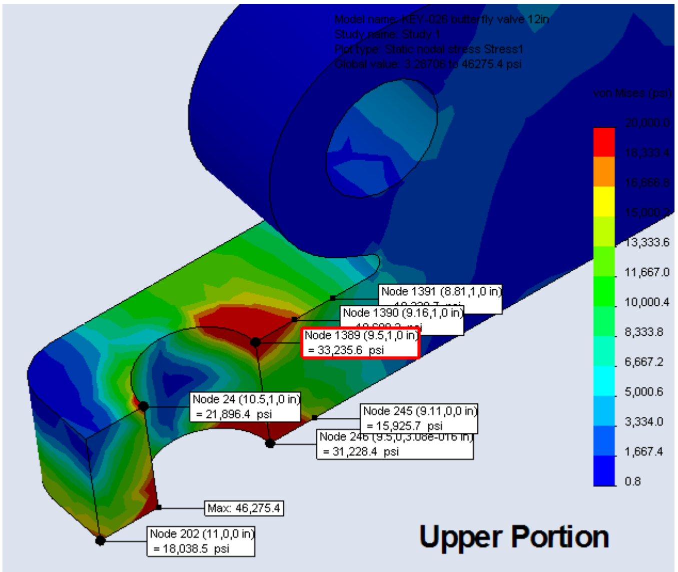
Contact Pressure:

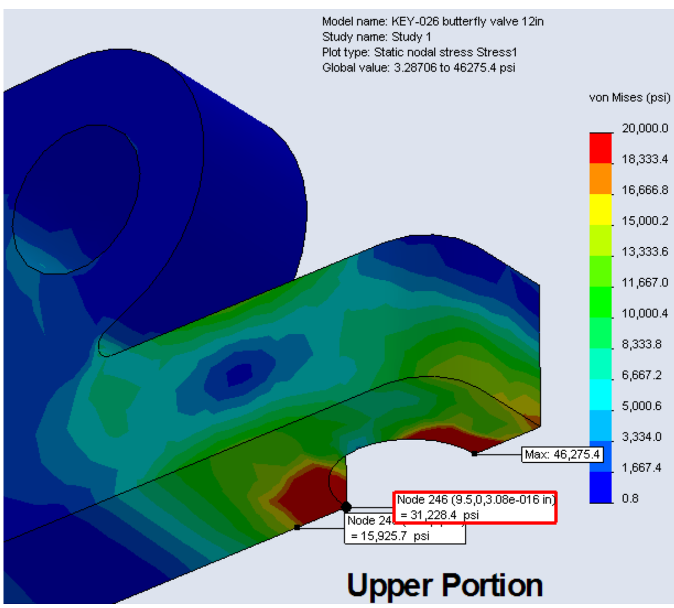


The Contact Pressure Plot above shows pressure around the outer edge of the bolthole, and out towards the outer edge of the tab, resulting from the face contact:

- The vertical dashed line corresponds to the bolt-hole centreline.
- The Red dashed line shows the sharp increase in stress from approximately ¾ of the bolthole till the outer edge of the same.
- The Blue dashed line shows the increasing contact pressure from about ¾ of the hole up to a maximum at the outer edge of the tab. The peak being is in line with the bolt centre, at the secondary cut plane.

Stress Results, Upper Portion:





The first figure on the left shows stress in the Upper Portion Tab, with the display capped at 20,000 PSI for convenience.

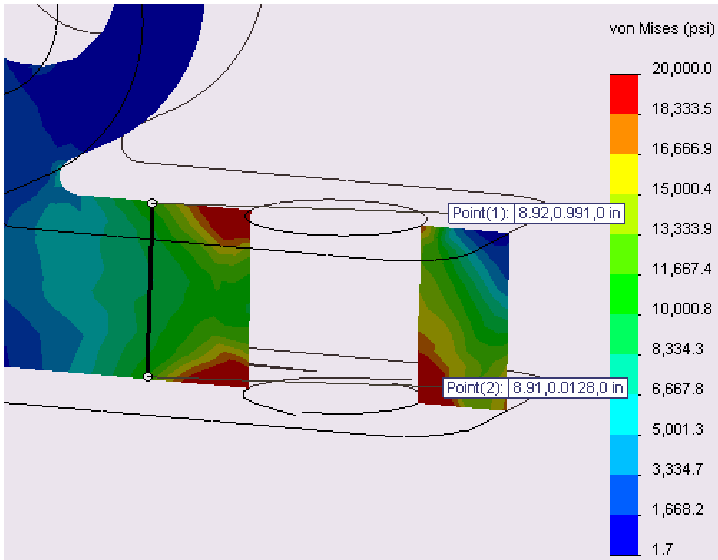
- General: The general areas display a stress below the Pm limit of 20,000 PSI and are therefore acceptable.
- Local: The inside face of the bolt hole has a stress of 33,235 PSI, which is even less than the Pl+Pb limit for the weakest material and is therefore acceptable. For this stress category, an additional allowance for Peak, F, is also permitted, increasing the limit.
- Peak Stress: The maximum stress is 46,896 PSI, at the outermost edge of the bolt hole. This bearing stress is located at a stress concentration and is covered by a fatigue-life analysis.

The lower figure displays the general stress on the outside of the tab, which is also below the 20,000 PSI limit and is therefore acceptable. The highest stresses have been covered in the discussion above.

There is a difference in the stress distribution between the Upper and Lower tabs (see subsequent sheets). This is due mostly to the threaded-hole configuration in the one tab and the through-hole in the other. The bolt head produces a conical compaction zone that is absent from the threaded tab.

Stress is acceptable.

Stress Linearization, Upper Portion:



Quantity, Units: psi	Normal X	Normal Y	Normal Z	Shear XY	Shear XZ	Shear YZ	von Mises
Membrane Stress	-230.34	-84.837	-166.71	-2992.6	26	65.518	5186.2
Bending (Point 1)	-12904	247.32	-1941.7	-251.74	-57.815	-96.72	12215
Membrane Stress + Bending (Point 1)	-13135	162.48	-2108.5	-3244.3	-31.815	-31.203	13541
Bending (Point 2)	12904	-247.32	1941.7	251.74	57.815	96.72	12215
Membrane Stress + Bending (Point 2)	12674	-332.16	1775	-2740.8	83.815	162.24	12993
Peak (Point 1)	-15.796	-173.66	-389.24	2987.9	3.1706	61.755	13564
Peak (Point 2)	-3914.5	328.41	2188.8	3125.9	-906.71	-243.91	13483

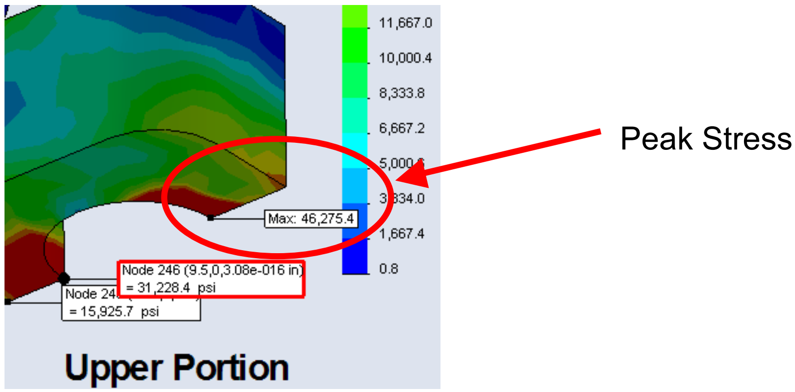
The images above show a typical Stress Classification Line through the centre plane of the tab, where the bending stress would be expected to be the greatest because of the threaded hole. The highest stresses are found at Point 1:

Membrane = 5,186 PSI < Pm = 20,000 PSI **Acceptable**
Membrane + Bending =13,541 PSI < Pm+Pb = 30,000 PSI **Acceptable**

KEY DESIGN ENGINEERING

Ph: 520-265-3657
Fax: 888-241-3035
mailto:info@keydesigneng.com
http://www.keydesigneng.com/

Cycle-Life Analysis, Peak Stress Upper Portion:



Fatigue analysis per ASME VIII-2, 2007

Table 3.F.1 for Carbon Steel, for UTS<=80ksi
The design number of cycles, N

N=	**=(10^x)*(Et/Efc) =**	**4.96E+04 cycles**

Where:

Et=	28300000	psi, elevated temperature modulus of elasticity
Efc=	28300000	psi, baseline modulus of elasticity
S=	46.275	ksi, measured peak stress value, alternating about "0 psi"

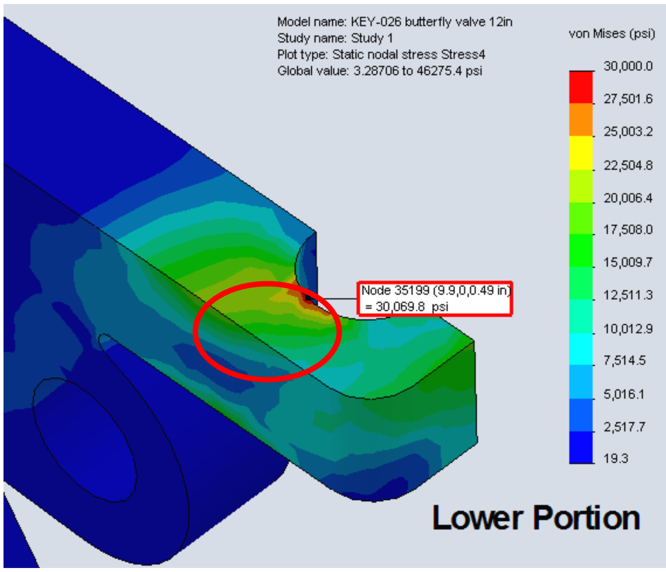
For, 7<= Sa < 31 ksi

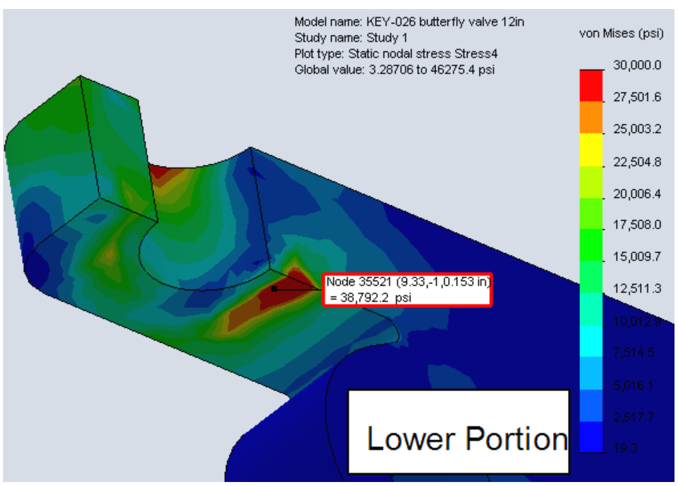
X=	4.6956182	
Numerator	38.161416	
Denominator	8.1270272	
C1	2.25454	
C2	-4.64E-01	
C3	-8.31E-01	
C4	8.63E-02	
C5	2.02E-01	
C6	-6.94E-03	
C7	-2.08E-02	
C8	2.01E-04	
C9	7.14E-04	
C10	0	
C11	0	
Sa	23.1375	Sa is within range
Cus	1	Conversion factor

The expected fatigue life is greater than 49,600 cycles at this level of stress. Acceptability depends on customer requirements. For many piping applications, this is acceptable.

Ph: 520-265-3657
Fax: 888-241-3035
mailto:info@keydesigneng.com
http://www.keydesigneng.com/

Stress Results, Lower Portion:





The first figure on the left shows stress in the Lower Portion Tab, with the display capped at 30,000 PSI for convenience.

- General: The general areas display a stress below the Pm limit of 20,000 PSI and are therefore acceptable.
- Local: The inside face of the bolthole has a stress of 30,069 PSI, which is over the Pl+Pb limit for the weakest material. For this stress concentration, the allowable Sa=3xS, which is acceptable.
- Bending Stress: The circled area has local stress that is greater than the Pl limit. Stress linearization is performed through this zone, on both faces of the tab, to demonstrate that it is a localized surface stress, due to the Stress Concentration (bolthole) and is discussed on subsequent sheets.

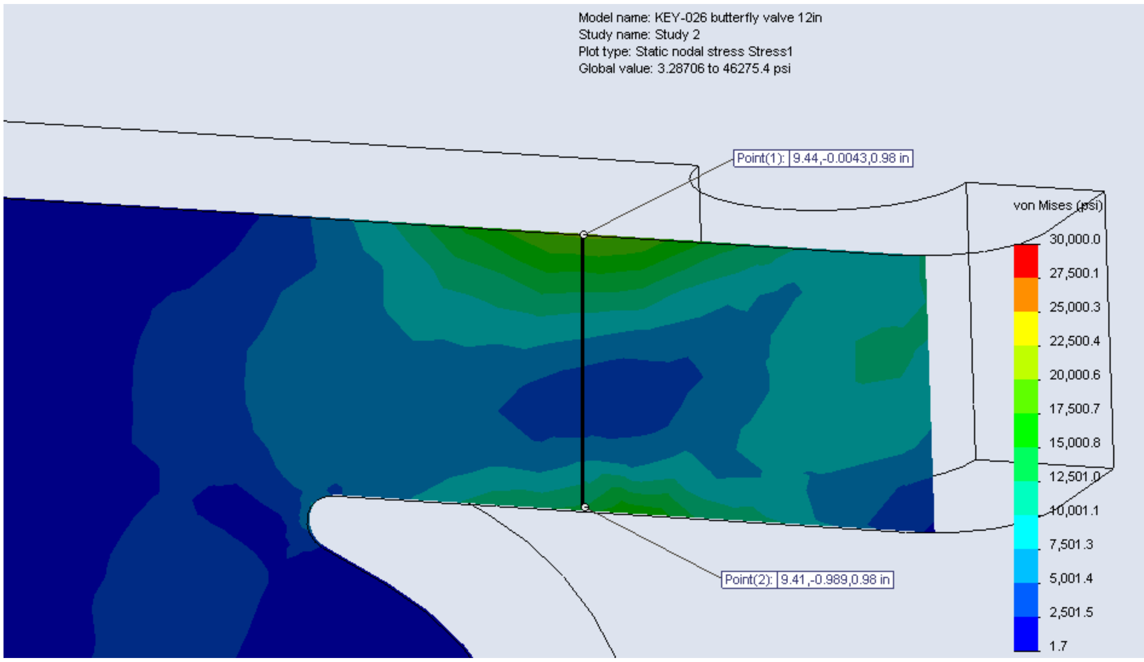
The lower figure displays the general stress on the bottom face of the tab. The stress of 38,792 PSI is due to the line application of the bolting forces and is subject to Sa=3xS, for Pl+Pb+Q+F and is therefore acceptable. Furthermore, it doesn't correspond to a physical stress such as would be present under a washer or bolt head.

Stress is acceptable.

KEY DESIGN ENGINEERING

Ph: 520-265-3657
Fax: 888-241-3035
mailto:info@keydesigneng.com
http://www.keydesigneng.com/

Stress Linearization, Lower Portion:



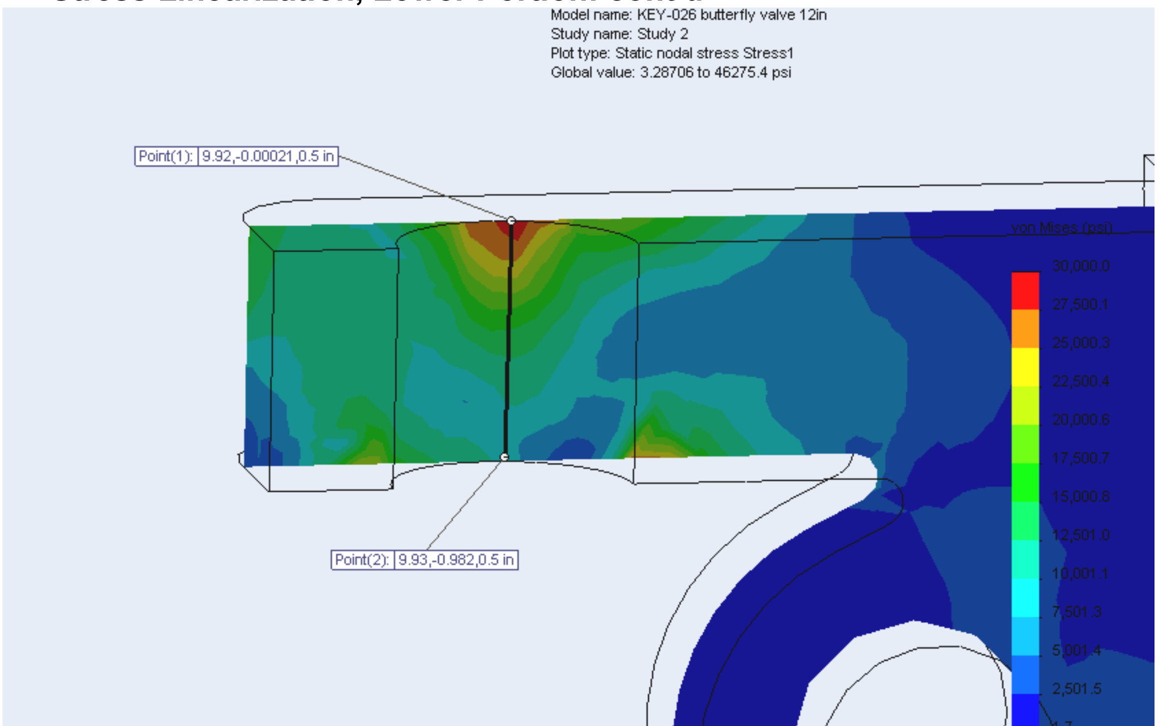
Quantity, Units: psi	Normal X	Normal Y	Normal Z	Shear XY	Shear XZ	Shear YZ	von Mises
Membrane Stress	3507.6	373.66	-170.71	85.299	-565.51	-60.691	3580
Bending (Point 1)	14947	-609.29	204.94	348.69	207.84	-106.79	15183
Membrane Stress + Bending (Point 1)	18455	-235.63	34.237	433.99	-357.66	-167.48	18585
Bending (Point 2)	-14947	609.29	-204.94	-348.69	-207.84	106.79	15183
Membrane Stress + Bending (Point 2)	-11440	982.94	-375.65	-263.39	-773.35	46.096	11887

The images above show a typical Stress Classification Line through the outside edge of the tab, where the surface stress is highest. This is a transition zone between the tightly-clamped end and the stiff connection with the valve body. The highest stresses are found at Point 1:

Membrane =	3,580 PSI < Pm = 20,000 PSI	**Acceptable**
Membrane + Bending =	18,585 PSI < Pm+Pb = 30,000 PSI	**Acceptable**

Ph: 520-265-3657
Fax: 888-241-3035
mailto:info@keydesigneng.com
http://www.keydesigneng.com/

Stress Linearization, Lower Portion: cont'd



Quantity, Units: psi	Normal X	Normal Y	Normal Z	Shear XY	Shear XZ	Shear YZ	von Mises
Membrane Stress	9233.8	159.61	874.95	-4796.8	617.43	-928.92	12212
Bending (Point 1)	15283	1746.7	1758.7	927.37	1076.1	481.61	13778
Membrane Stress + Bending (Point 1)	24517	1906.4	2633.7	-3869.5	1693.5	-447.32	23441
Bending (Point 2)	-15283	-1746.7	-1758.7	-927.37	-1076.1	-481.61	13778
Membrane Stress + Bending (Point 2)	-6049.7	-1587.1	-883.76	-5724.2	-458.64	-1410.5	11333

The images above show a typical Stress Classification Line through the inside face of the bolthole, where the high stress-point is found. This bending stress is compounded by a stress concentration (bolthole). The highest stresses are found at Point 1:

Membrane = 12,212 PSI < Pm = 20,000 PSI **Acceptable**
Membrane + Bending = 23,441 PSI < Pm+Pb = 30,000 PSI **Acceptable**

Ph: 520-265-3657
Fax: 888-241-3035
mailto:info@keydesigneng.com
http://www.keydesigneng.com/

Bolting Stress:

For the symmetrical bolt connector used in the analysis, the results are shown below. The Bolt connector models the bolt shank as a beam element, and rigid bar elements for the head parts. This type of arrangement takes its name from the spider-like arrangement of elements. For the Socket-head capscrew used, the head is on the lower portion face and the upper portion tab is threaded.

Calculated values, applying bolt characteristics, A 193-B7, S=25,000 PSI, Bolting Stress:

Stress (PSI)		Limit	Outcome
36,242	Tensile Stress	Tensile = 2xS = 50,000 PSI	Acceptable
18,335	Bending Stress	Tensile+Bending = 3xS = 75,000 KSI	Acceptable

Please note that Bolting Fatigue is not applicable here, since the Membrane+Bending stress is very low, when the preload stress is removed.

Recommendations & Comments:

- The valve's central joint satisfies the requirements of ASME VIII-2-2007 as permitted by ASME B31.3-2008 at a design pressure of 275 PSI with an expected fatigue life of over 49,600 cycles.
- Bolting preload can be increased to further stiffen the joint and reduce the hinging about the tab's outer edge. It is expected that this will increase fatigue life.
- The tabs can be stiffened to reduce body transition stress, by filling in some material at the lug. However, this will increase the bolting requirement.
- No consideration has been given to ASME B31.3 para.309.3 for tapped hole depth requirement, as this is a sample job only and dimensions were taken from information available on company's public site.

Key Design Engineering
55 Northfield Dr. E, Suite 194
Waterloo, ON N2K 3T6
http://www.keydesigneng.com/